Structural Integrity of Open-Cell Aluminium Foam Sandwich Panels for Lightweight Wing Structures

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Declaration of originality

I declare that the work presented in this thesis is my own and that any work of others has been appropriately referenced.
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Abstract

The overarching aim of this work was to concentrate on the mechanical modelling and experimental characterisation of novel open-cell aluminium foam core sandwich panels for prospective use as an airplane wing skin material.

A repeating unit cell 2D FE model was created to assess the mechanical behaviour of infinitely long, regularly tessellated honeycomb core sandwich panels. An analytical model using Timoshenko beam theory was developed to predict the Young’s modulus of a hexagonal honeycomb core; there is good agreement between the two models.

A microtensile test procedure was developed to determine the mechanical properties of individual foam struts. A FE model of the as-tested struts was created, using XMT scans of the undeformed struts to define the geometry, to establish a method that compensates for grip slippage inherent in the testing of the struts. Strut deformation was described by a calibrated continuum viscoplastic damage model.

The damage model was implemented into 3D FE models of an open-cell aluminium alloy foam core sandwich panel subjected to uniform compression to study the effect of varying the strut aspect ratio on the mechanical properties of the core. FE models of the panel subjected to three and four point bending were created to provide a virtual standardised test to assess the core elastic properties. The extent of structural damage in the panels was simulated for indentation loading indicative of a tool strike; an optimal strut aspect ratio was identified providing the best energy absorption per unit mass whilst ensuring core damage is detectable.

The effect of morphological imperfections on the mechanical properties and extent of detectable damage of the core was studied. The shear modulus of the core was greatly reduced under the presence of both fractured cell walls and missing cells. The extent of visible damage was largely unaffected by either type of defect.
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Nomenclature

Acronyms

ASTM American society for testing and materials specification
BCC Body-centred cubic
CAD Computer-aided design
CDM Continuum damage mechanics
DIC Digital image correlation
DOF Degree of freedom
EDM Electrical discharge machining
EDX Energy-dispersive X-ray spectroscopy
FBD Free body diagram
FE Finite element
LHS Left-hand side
MMC Metal matrix composite
MPC Multi-point constraint
NDT Non-destructive testing
PBCs Periodic boundary conditions
RHS Right-hand side
SEM Scanning electron microscope
UTS Ultimate tensile strength
XMT X-ray microtomography

Roman symbols

A Cross-sectional area
A_D Dorn constant
b Sandwich panel width
c  Foam core thickness
D  Sandwich panel bending stiffness
d  Sandwich panel thickness
d_0  Minimum distance between two adjacent nuclei in a regular Voronoi lattice
E  Foam Young’s modulus
E_1  Foam Young’s modulus in the in-plane horizontal direction
E_2  Foam Young’s modulus in the vertical direction
E_3  Foam Young’s modulus in the out-of-plane horizontal direction
E_f  Young’s modulus of sandwich panel facesheets
E_s  Young’s modulus of the solid of which the foam is made
f  Void volume fraction
f_c  Critical void volume fraction
f_f  Void volume fraction at final fracture
f_{growth}  Void growth volume fraction
f_{nucleation}  Void nucleation sites volume fraction
G  Foam shear modulus
G_s  Shear modulus of the solid of which the foam is made
I  Moment of inertia
K  Drag stress
k  Shear coefficient
L  Sandwich panel length
l  Foam cell strut length
M  Bending moment
n_{stress}  Stress exponent
P  Axial force
<table>
<thead>
<tr>
<th>Symbol</th>
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<tr>
<td>$Q$</td>
<td>Transverse force</td>
</tr>
<tr>
<td>$Q_{act}$</td>
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</tr>
<tr>
<td>$q(x)$</td>
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<td>Temperature</td>
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<tr>
<td>$t$</td>
<td>Foam cell strut thickness</td>
</tr>
<tr>
<td>$t_c$</td>
<td>Foam cell strut thickness at $x_c$</td>
</tr>
<tr>
<td>$U$</td>
<td>Sandwich panel shear rigidity</td>
</tr>
<tr>
<td>$U_1$</td>
<td>Horizontal displacement</td>
</tr>
<tr>
<td>$U_2$</td>
<td>Vertical displacement</td>
</tr>
<tr>
<td>$V_0$</td>
<td>Sandwich core volume</td>
</tr>
<tr>
<td>$w$</td>
<td>Indenter width</td>
</tr>
<tr>
<td>$w_c$</td>
<td>Initial transverse deflection of a cell wall at $x_c$</td>
</tr>
<tr>
<td>$w_o$</td>
<td>Amplitude of waviness of a cell wall</td>
</tr>
<tr>
<td>$x$</td>
<td>Distance from the mid-joint along a cell wall</td>
</tr>
<tr>
<td>$x_c$</td>
<td>Distance to plastic hinge from the mid-joint of a cell wall</td>
</tr>
</tbody>
</table>
Greek symbols

$\alpha_V$  Regularity parameter

$\Delta_1$  Midspan deflection of a sandwich panel subjected to three point bending

$\Delta_2$  Midspan deflection of a sandwich panel subjected to four point bending

$\delta$  Transverse deflection

$\delta_d$  Minimum distance between two adjacent nuclei in a Voronoi lattice

$\delta_{ij}$  Kronecker delta

$\varepsilon$  Engineering strain

$\varepsilon_2$  Engineering strain in the vertical direction

$\varepsilon_D$  Densification strain

$\varepsilon_{\text{eq}}$  Equivalent von Mises strain

$\dot{\varepsilon}_e$  Effective viscoplastic strain rate

$\varepsilon_{ij}^e$  Elastic strain tensor

$\varepsilon_{ij}^p$  Plastic strain tensor

$\varepsilon_{ij}^t$  Total strain tensor

$\lambda$  Lamé’s first parameter

$\mu_f$  Coefficient of friction

$\nu$  Foam Poisson’s ratio

$\nu_s$  Poisson’s ratio of the solid of which the foam is made

$\rho$  Foam density

$\bar{\rho}$  Normalised dislocation density

$\dot{\bar{\rho}}$  Rate of normalised dislocation density

$\rho_i$  Initial dislocation density

$\rho_{\text{max}}$  Saturated dislocation density
\( \rho_r \) Relative density  
\( \rho_s \) Density of the solid of which the foam is made  
\( \sigma \) Engineering stress  
\( \sigma_0 \) Matrix yield stress  
\( \sigma_2 \) Engineering stress in the vertical direction  
\( \bar{\sigma}_{cr} \) Crushing stress  
\( \sigma_e \) Equivalent von Mises stress  
\( \bar{\sigma}_{el} \) Elastic buckling stress  
\( \sigma_f \) Maximum principle stress  
\( \sigma_{MOR} \) Modulus of rupture of the foam strut  
\( \bar{\sigma}_{pl} \) Plateau stress  
\( \sigma_H \) Hydrostatic yield strength  
\( \sigma_H^0 \) Hydrostatic yield strength of a perfect honeycomb  
\( \sigma_{ind} \) Indentation strength  
\( \sigma_m \) Hydrostatic stress  
\( \sigma_{pl}^* \) Honeycomb plastic collapse stress  
\( \sigma_U \) Uniaxial yield strength  
\( \sigma_U^0 \) Uniaxial yield strength of a perfect honeycomb  
\( \sigma_y \) Foam elastic limit  
\( \sigma_{ys} \) Yield strength of the solid of which the foam is made  
\( \tau_{pl}^* \) Honeycomb plastic shear strength  
\( \Phi \) Flow potential  
\( \phi \) Rotational angle  
\( \psi_c \) Associated slope of initial transverse deflection of a cell wall  
\( \omega \) Damage parameter  
\( \omega_{crit} \) Maximum allowable damage
Chapter 1.

Introduction
1.1 Background and motivation

The design of commercial aircraft structures is consistently driven towards reducing weight (and hence operating costs) whilst at the same time achieving a satisfactory level of strength. This results in the need to stabilise thin panels to carry tensile and compressive loads and a combination of the two in tension, torsion, and bending (Herrmann et al., 2005).

Conventional wing box design utilises thin panels that comprise of a skin stiffened by stringers, as shown in Fig. 1.1. The outer wing box structure is manufactured from separately machined parts – spars, ribs, and skin/stringer panels. The ribs serve to stabilise the structure and transfer the local air load into the wing box. The panels and spars carry the global bending and torsional loads (Schuhmacher et al., 2002; Worsfold, 1998).

Fig. 1.1 A schematic layout of a British Aerospace outer wing structure, after (Worsfold, 1998)

Skin/stringer panels are typically fabricated by machining stiffeners from thick blanks and subsequently fastening to a sheet. The resultant panels are light

1 – Skin plating
2 – Ribs
3 – Strips, closing plates and seals
4 – Spars
5 – Main gear support structure complete
6 – Stringers and stiffeners
and stiff, but they are relatively expensive to produce due to high machining costs and the inefficient use of material. They also display significant anisotropy in the bending plane (i.e. they are not equally stiff about all axes of bending) (Sypeck, 2005). Indeed, the stabilisation of the thin sheets can often be more efficiently achieved by the use of twin skins with a stabilising medium between them (Herrmann et al., 2005).

Sandwich structures are defined as “structural members made up of two stiff, strong skins separated by a lightweight core” (Ashby and Gibson, 1988). The lightweight core serves to separate the skins, hence increasing the moment of inertia of the panel whilst keeping weight to a minimum. The result is a structure that is efficient in resisting both bending and buckling loads (Ashby and Gibson, 1988). As already noted, traditional stiffened skins comprise of discrete stiffeners. However, sandwich structures provide a continuous stiffness distribution within the skin panel. This integral stiffening leads to a reduced parts count for assemblies and hence less logistics, parts manufacturing, and assembly work (Herrmann et al., 2005).

Sandwich structures were first used in British aviation in the construction of the de Havilland Mosquito night bomber of World War II (Hoff, 1951). In 1943, the Vultee BT-15 fuselage was constructed from fibreglass-reinforced polyester as the face material with both a glass-fabric honeycomb and a balsa wood core (Vinson, 2005). Today, composite sandwich structures are used extensively in aircraft design (for example on the Airbus A380 – see Fig. 1.2) (Schuhmacher et al., 2002). 46% of the external surface of the Boeing 757/767 is honeycomb sandwich and the Boeing 747 comprises of a fuselage cylindrical shell that is primarily Nomex honeycomb sandwich (Vinson, 2005). It should be noted, however, that the use of sandwich structures in commercial aviation is at present limited to secondary structures only. To be applicable to primary structures (that is, structural components that are necessary to sustain design ultimate flight and
sandwich structures must ensure that any damage during the service life of the component does not result in failure prior to the damage being detected – this would need to be demonstrated by appropriate tests and analyses, and the definition of maximum allowable damages and their visibility – and it is here that the ability to accurately model the behaviour of sandwich panels is of critical importance. Sandwich structures suffer the disadvantage that structural failures cannot always be detected by standard methods such as visual inspection and ultrasonic pulse-echo. This may lead to the need to inspect the components using repeated non-destructive testing (NDT), whilst considering at the same time the economic requirements for in-service life (Herrmann et al., 2005). The key drivers for the use of a sandwich panel as a primary structure include its ability to fail-safe, the toughness of the materials, and its capability to withstand impact damages (Herrmann et al., 2005).

1 – Spoilers/ailerons
2 – Vertical tail plane: fin tip & fairings, dorsal fin, leading & trailing edge panels
3 – Horizontal tail plane: fairing, leading & trailing edge panels
4 – Fuselage belly fairing
5 – Flap track fairings
6 – Pylons/nacelles: secondary structure, access panels
7 – Nose landing gear doors
8 – Cabin: floor panels
9 – Wing: leading & trailing edge panels, access panels

Fig. 1.2 Example A380 sandwich panel applications, after (Herrmann et al., 2005)
Metal foams are a relatively new class of materials that display high values of $E^{1/3}/\rho$, where $E$ is the Young’s modulus of the foam and $\rho$ the foam density. This material index is derived in Ashby and Lu (2003) for a panel of specified length, width, and stiffness with the objective of minimising the mass. The greater the value of $E^{1/3}/\rho$, the lighter a panel can be for a given stiffness. $E^{1/3}/\rho$ therefore characterises the bending-stiffness of lightweight panels and suggests the use of metal foams as light, stiff panels (Ashby and Lu, 2003). The weight of metal foam sandwich structures is comparable to waffle-stiffened aluminium panels, but they have lower manufacturing costs (Ashby et al., 2000). Furthermore, metal foams have high values of $\sigma_y^{1/2}/\rho$, where $\sigma_y$ is the elastic limit of the foam. This material index is derived in Ashby and Lu (2003) for a panel of specified length, width, and strength with the objective of minimising the mass and hence characterises the bending-strength of lightweight panels. The greater the value of $\sigma_y^{1/2}/\rho$, the stronger a panel will be for a given mass. Consequently, a metal foam panel is stronger, for a given mass, than one of the same material which is solid. Strength limited foam core sandwich panels can also offer weight savings over traditional stringer-stiffened panels (Ashby et al., 2000).

Metal foams are therefore able to combine low density with good bending stiffness and strength. This makes them attractive as cores of lightweight sandwich structures. In addition, they can (unlike honeycombs) display isotropy in mechanical properties. They have an outstanding ability to absorb energy at almost constant pressure and can be readily formed into curved shapes (McCormack et al., 2001). They display a densification stage in a compressive stress-strain plot, where the stress rises rapidly with strain as the foam crushes – this has the implication that the integrity of a metal foam core sandwich panel is not necessarily compromised when subjected to impacts. Moreover, open-cell foams do not trap moisture (i.e. they are less susceptible to corrosion than honeycomb cores).
Metal foam core sandwich structures therefore have a great deal of potential applications in the aerospace industry. The Norwegian University of Science and Technology and EADS CCR have studied the damage performance of aluminium foam core sandwich structures subjected to bird-strike tests, and simulated the process using the finite element (FE) method; the results show that a shield made of aluminium foam core sandwich can effectively protect the equipment and people in an airplane from bird-strikes or other similar impacts (Hanssen et al., 2004). Furthermore, Boeing has evaluated the use of aluminium sandwich panels with aluminium foam cores for tail booms of helicopters, and some helicopter manufacturers have tried to use aluminium foam parts to replace some of the currently used honeycomb components (Banhart, 2001). Aluminium foam breather plugs made by ERG Aerospace Corporation have been used in the F-22 stealth fighter aircraft for pressure releases during rapid altitude changes, and aluminium foam core sandwich panels are used in the AMS-02 satellite to protect vital components against micrometeorite strikes in space (http://www.ergaerospace.com). Moreover, Alm Germany, in collaboration with the German Space Agency (DLR) and the French Space Agency (CNES), have explored the potential application of aluminium foam core sandwiches in space components, and succeeded in preparing an Ariane 5 rocket cone demonstrator (Schwingel et al., 2007; Gokhale et al., 2011).

Metal foam core sandwich panels also show promise in automotive applications due to their lightweight construction and energy absorbing capabilities. One example is German car maker Karmann’s concept car called the Ghia roadster, which consists of aluminium foam core sandwich panel parts that are lighter and stiffer than the conventional components made of stamped steel sheets that they replace (Phelan, 1998).
In addition, Fiat and the Norwegian University of Science and Technology have conducted studies that show car crash boxes (assemblies located at the front of a car, designed to crumple in the event of a collision to protect passengers and minimise vehicle damage) that comprise of a tube filled with a metal foam display an improved axial energy absorption capability compared to that of empty tubes, and more than the sum of the individual energies of the tube and the foam (Cheon and Meguid, 2004). There is also a marked improvement in the energy absorption in off-axis collisions of metal foam filled tubes due to the isotropic nature of the foams (Banhart, 2003). Other potential applications exist in shipbuilding, the railway industry, the biomedical industry (including bone augmentation implants), and civil engineering (Banhart, 2001; Banhart, 2003; Baumeister et al., 1997; Evans et al., 2001; Singh et al., 2010a).

1.2 Aims and objectives

Due to the potential of metal foams for lightweight structures and energy absorption as outlined in Section 1.1, the aim of this work is to concentrate on the mechanical modelling and experimental characterisation of novel open-cell aluminium foam sandwich panels (aluminium foam core with aluminium facesheets) for prospective use as a wing skin material to aid in the design and development of new lightweight aircraft wing designs lacking ribs and stringers.

The objectives of this work have been established in cooperation with the Airbus Future Projects Office and are as follows:

- To develop a novel microtensile test procedure to be able to directly determine the mechanical properties of individual aluminium alloy foam struts. The strength of an open-cell metal foam is strongly dependant on the individual cell strut properties. Usually, the material properties of the bulk alloy from which the foam is made are used to
predict the foam properties. However, due to the foaming process and length scale of the struts, there can be marked differences between the mechanical properties of the bulk alloy and the individual struts due to differences in both composition and microstructure (Zhou et al., 2005);

- To establish and calibrate a continuum viscoplastic damage model, using the results of the microtensile tests, in order to accurately model the material behaviour of the individual foam struts. This includes capturing damage softening and failure;

- To develop multiscale computational models using the damage model for predicting (characterising and quantifying) the performance of aluminium foam core sandwiches under uniform compression and bending loading scenarios. This includes determining the energy absorption levels;

- To develop a multiscale computational model using the damage model to predict the performance and damage of aluminium foam core sandwiches under indentation loading scenarios. These loading scenarios are intended to capture the effect of an accidental tool drop impact under ground repair conditions. As discussed in Section 1.1, to be applicable to primary airplane structures, aluminium foam core sandwich panels must ensure that any damage during their service life does not result in failure prior to the damage being detected. Low energy impacts can reduce the strength of sandwich structures, as well as cause considerable subsurface damage. This is a particular problem in aircraft structures which may be subjected to tools (e.g. spanners) being dropped during maintenance or foreign object damage during landing and take-off; and
To consider the effect of parameters including the foam core relative density (defined as \( \frac{\rho}{\rho_s} \), where \( \rho \) is the density of the foam and \( \rho_s \) is the density of the solid of which it is made), the extent of processing induced foam core morphological defects such as fractured cell walls and missing cells, and the angle of tool drop impact on the performance and damage of aluminium foam core sandwiches.

### 1.3 Thesis structure

This thesis is organized into seven chapters. This first chapter motivated the work and outlined the key advantages of aluminium foam core sandwich panels as a potential use as a wing skin material; notably, such panels obviate the need for discrete stiffeners. It laid out the objectives of the work; namely to predict the damage of the panels under a light impact scenario, along with the definition of maximum allowable damages and their visibility.

Chapter 2 provides a comprehensive review assessing the relative benefits of metal foam core sandwich panels with respect to honeycomb, polymeric foam and truss cores. The current analytical and FE modelling tools available for metal foams are also reviewed.

Chapter 3 establishes a repeating unit cell 2D FE modelling procedure with periodic boundary conditions (PBCs) to model the mechanical behaviour of infinitely long, regularly tessellated honeycomb (i.e. hexagonal, square, and equilateral triangle cell shapes) core sandwich panels. An analytical solution using Timoshenko beam theory is developed to predict the Young’s modulus of a hexagonal honeycomb core and is compared to the FE results. The comparative performance of each cell shape is investigated for various applications (e.g. energy absorption, lightweight structural applications, etc.).
Chapter 4 first characterises the aluminium foam studied in this work (e.g. chemical composition, average grain size, cell morphology, etc.). A microtensile test procedure is then developed to directly determine the mechanical properties of individual aluminium alloy foam struts.

Chapter 5 develops a realistic FE modelling procedure of the as-tested foam struts of Chapter 4 using X-ray microtomography (XMT) scans of the undeformed struts to define the geometry. Strut deformation is described by established continuum viscoplastic damage constitutive equations calibrated using the microtensile test data of Chapter 4. The as-tested strut FE model is used to determine the reasons for the considerable reduction in elastic stiffness observed during microtensile testing, and to develop a procedure that compensates for the effect of grip slippage inherent in the microtensile testing of aluminium foam struts.

Chapter 6 presents 3D FE models of an open-cell aluminium metal foam core sandwich panel subjected to (I) uniform compression, (II) three and four point bending, and (III) indentation loading scenarios indicative of an accidental tool strike. The continuum viscoplastic damage model calibrated in Chapter 5 is implemented into the uniform compression and indentation loading models. The effects of morphological imperfections on the mechanical properties and damage visibility (i.e. the extent of ‘hidden’, undetected core damage) of the sandwich panel are considered, and the radius of the indenter is varied so as to capture the influence of varying angles of tool drop impact on the damage visibility.

Finally, Chapter 7 comprises of a summary of the main findings and a discussion of future work.
Chapter 2.

Literature review
2.1 Introduction

Metal foams are a relatively new class of materials that show good potential for lightweight structures, energy absorption, and thermal management (Ashby et al., 2000; Banhart, 2001; Evans et al., 2001). The first attempts to manufacture metal foams date back to the 1940s when Benjamin Sosnick tried to foam aluminium with mercury (Sosnick, 1948). In the 1950s it was found that liquid metals could be foamed by pretreating them to enhance their viscosity, for example by oxidising the melt or adding oxide particles (Elliot, 1956). In the early 1970s the Ford Motor Company evaluated aluminium foam samples, but initial developments were unsuccessful, leading to a decline in R&D of metal foams post-1975 (Babcsán and Banhart, 2006; Banhart and Weaire, 2002). Research picked up again towards the end of the 1980s when the Shinko Wire Company developed the Alporas foam manufacturing process (Akiyama et al., 1986). In 1991, Joachim Baumeister brought the compacted-powder foaming process developed in the 1950s to a considerable level of sophistication (Baumeister, 1991).

Today, several different manufacturing techniques exist to make metal foams, five of which are commercially established – (I) melt gas injection, (II) gas-releasing particle decomposition in the melt, (III) gas-releasing particle decomposition in semi-solids, (IV) casting using a polymer or wax precursor as a template, and (V) metal deposition on cellular preforms. These techniques are reviewed by Banhart (2001) and by Ashby et al. (2000), which present a comprehensive review of metal foams, including their manufacturing methods and their advantages in industry. The different manufacturing techniques are used for different subsets of metals to create porous materials with a limited range of relative densities and cell sizes. Some methods produce closed-cell foams, others open-cell. The cost of each process varies significantly – from $7 to $12000 per kg.
Most commercially available metal foams are currently based on aluminium or nickel. Further details can be found in Banhart and Baumeister (1998), Baumeister (2001), Frei et al. (2000), Gergely and Clyne (2000), Koerner (2008), Koerner et al. (2006), and Leitmeier (2001).

The current manufacturing methods for producing sandwich panels that combine a metal foam core with metal facesheets can be classed under two distinct categories: (a) Ex-situ bonding is the process of bonding the facesheets directly onto a sheet of metal foam; and (b) In-situ bonding is the process of combining the metal foam manufacture with bonding to the facesheets (Banhart and Seeliger, 2008). The ultimate objective of the bonding process is to achieve a bond between the facesheets and foam core that is above the strength of the foam (Neugebauer et al., 2001). Technologies are emerging for creating syntactic metal foam structures (i.e. foams with an integrally shaped skin). This would allow cheap, lightweight structures to be moulded in a single operation and it is perhaps here that current metal foam technology holds the greatest promise (Ashby et al., 2000; Koerner 2008).

Characterisation and testing methods have been developed for metal foams, including uniaxial compression tests on cylindrical foam specimens, uniaxial tension tests on dogbone specimens, torsion tests, and four point bending for fatigue testing (e.g. ASTM C-273-61 and ASTM E8-96a). The foam structure can be examined using notably optical microscopy, scanning electron microscopy, and X-ray tomography (XMT) (Banhart, 2001; Bart-Smith et al., 1998).

The review work by Banhart (2001) and by Ashby et al. (2000) concentrates mainly on the manufacturing technologies and experimental methods for foam materials and structures. However, significant efforts have also been made to develop analytical as well as FE modelling techniques for assessing the material behaviour of foam materials (e.g. Ashby and Gibson, 1988; Chen et al., 1999; Gong et al., 2005; Hodge and Dunand, 2003; Huang and Gibson, 2003; Jang
and Kyriakides, 2009; Onck et al., 2001; Shulmeister et al., 1998; Silva et al., 1995; Simone and Gibson, 1998; Zhu et al., 2000). This Chapter aims to provide a comprehensive review of these techniques. The current modelling tools available for metal foams fall under three key categories (Betts, 2012):

- **Analytical methods**, utilising dimensional analysis that gives the dependence of the foam properties on the relative density but not the cell geometry;
- **Finite element methods utilising a repeating unit cell**; and
- **Finite element methods utilising the random Voronoi technique**. This approach gives a more accurate representation of the cell geometry of the foams (Zhu et al., 2000).

The detailed research results on the above techniques are reviewed, analysed and presented in Sections 2.3 and 2.4 respectively. Section 2.5 provides an assessment of the relative merits of each method, as well as discussing the future trends for the modelling of metal foams.

For completeness, the relative benefits of metal foam core sandwich panels with respect to honeycomb, polymeric foam and truss cores are summarised, discussed and presented in Section 2.2.

### 2.2 Sandwich structure core types and their relative benefits

The two most common sandwich core types in industrial applications are honeycomb and foam (polymeric or metallic) cores (Niu, 1996). An additional core type comprises of truss structures.

Honeycomb cores consist of “any array of identical prismatic cells which nest together to fill a plane” (Ashby and Gibson, 1998). The cells are typically hexagonal in section, though they can also be triangular, square, rhombic, or circular (Chung and Waas, 2002). Polymer and metal honeycombs are used as
sandwich panel cores in aerospace components, though honeycombs can also be made from ceramics and paper (Ashby and Gibson, 1998). Fig. 2.1(a) shows an aramid-fibre (Nomex) reinforced honeycomb structure (Sypeck and Wadley, 2002). The structure is effectively 2D and regular, making honeycombs easier to analyse than foams which have 3D cell arrangements (Lu and Yu, 2003).

The truss core sandwich panel includes a corrugated sheet or a truss core disposed between two facesheets.

### 2.2.1 Honeycomb cores

It can be seen from Fig. 2.1(a) that a typical honeycomb is made up of a set of hexagonal cells. The dimensions of the cell are defined by the cell wall lengths, l and c, the angle between the two cell walls θ, and the thickness of the cell walls h, as shown in Fig. 2.1(b) (Lu and Yu, 2003).

![Fig. 2.1 (a) Aramid-fibre (Nomex) reinforced honeycomb, (Sypeck and Wadley, 2002); (b) Definitions of parameters for a honeycomb cell, (Lu and Yu, 2003)](image)

Honeycomb properties are anisotropic – that is, the in-plane stiffnesses and strengths are different to the out-of-plane ones. When a honeycomb is compressed in-plane (i.e. the stress acts orthogonal to the axis of the cells – the plane $X_1X_2$ in Fig. 2.1(b)), the cell walls initially bend, and the deformation is linear elastic.
Depending on the material of the cells, they then collapse beyond a critical strain by either elastic buckling, plastic yielding, creep, or brittle fracture. Eventually, adjacent cell walls will touch, the cells will close up and the structure densifies, resulting in a sharp increase in stiffness. In tension, the cell walls initially bend and, depending on the material, either yield plastically or fracture in a brittle manner (Ashby and Gibson, 1988). When loading occurs in the out-of-plane direction (the X₃ direction in Fig. 2.1(b)), the cell walls either extend or compress and the moduli and collapse stresses are much higher – i.e. honeycomb structures are much stiffer and stronger in the out-of-plane direction (Ashby and Gibson, 1988).

Fig. 2.2(a) and (b) show compressive stress-strain curves for a honeycomb structure loaded in the in-plane direction and the out-of-plane direction respectively (Ashby and Gibson, 1988). It can be seen from Fig. 2.2(a) and (b) that there are three distinct stages in the compressive stress-strain curves: an initial linear elastic region, followed by a plateau region, and finally a densification stage where the stress rises rapidly with strain. Increasing the relative density of a honeycomb structure alters the stress-strain curves – see Fig. 2.2(a) and (b).

Analytical models have been developed to evaluate the moduli and collapse stresses of honeycombs in both uniaxial and biaxial loading for both in-plane and out-of-plane loading scenarios. These are presented by Ashby and Gibson (1988) and Lu and Yu (2003). These show that one of the governing criteria which determines the performance of a honeycomb structure is its relative density, defined as the overall density of the cellular material divided by the density of the solid of which the cellular material is made. FE methods have also been developed – these and the analytical methods are discussed in Sections 2.3 and 2.4.

Sandwich structures that comprise of honeycomb cores can display high strength and light weight. However, as mentioned previously, they are also highly anisotropic. They are also difficult to form into complex curved shapes because of
induced anticlastic curvature. Durability issues have been linked to moisture intrusion into the panels resulting in internal corrosion and facesheet debonding (Sypeck, 2005; McCormack et al., 2001).

![Diagram of stress-strain curves](image)

**Fig. 2.2** (a) In-plane direction honeycomb stress-strain curves; (b) Out-of-plane direction honeycomb stress-strain curves, (Ashby and Gibson, 1988)

### 2.2.2 Metal foam cores

As with honeycombs, the properties of a foam are largely controlled by its relative density. In addition, the material the foam is made from and its cell type (open or closed) also dictate the properties (Lu and Yu, 2003). Foam properties are further affected by anisotropy and defects – i.e. wiggly, buckled, or broken cell walls and cells of significant size (Ashby and Lu, 2003).

Metal foams can, unlike honeycombs, display isotropy in mechanical properties. They can be made with integral skins, which presents the possibility to make composite structures without adhesive bonding, and can be readily formed into curved shapes (McCormack et al., 2001). They display a densification stage when subjected to a compressive stress, where the stress rises rapidly with strain as the foam cells crush – this has the implication that the integrity of a metal foam
core sandwich panel is not necessarily compromised when subjected to impacts. Furthermore, open-cell foams do not trap moisture (i.e. they are less susceptible to corrosion than honeycombs) (Worsfold, 1998). Open-cell cores could provide a dual function, and potentially be used for the storage or drainage of fuel in aircraft wing structures. Honeycomb cores or traditional stringer-stiffened panels do not offer this advantage (Wicks and Hutchinson, 2001).

Metal foam cores can exhibit values of $E^{1/3}/\rho$ in the range of 2 to 5 (GPa)$^{0.33}$/(Mg/m$^3$) whereas steels are typically around 0.7 (GPa)$^{0.33}$/(Mg/m$^3$) and aluminium around 1.5 (GPa)$^{0.33}$/(Mg/m$^3$). Metal foam cores can also exhibit values of $\sigma_y^{1/2}/\rho$ in the range of 2 to 10 (MPa)$^{0.5}$/(Mg/m$^3$) whereas steels are typically around 1.8 (MPa)$^{0.5}$/(Mg/m$^3$) and aluminium around 3.7 (MPa)$^{0.5}$/(Mg/m$^3$) (Ashby et al., 2000). As noted in Section 1.1, this suggests their use as the cores of lightweight sandwich structures.

2.2.3 Polymeric foam cores

The analytical models to evaluate the moduli and collapse stresses of open-cell and closed-cell polymeric foams in both compression and tension are identical to those for metal foams, which are discussed in Section 2.3 (Ashby and Gibson, 1988).

Fig. 2.3 shows compressive stress-strain curves for a polyurethane foam (Lu and Yu, 2003). It can be seen that the curves exhibit the same trend as per metal foams and honeycombs.

Polymeric foams tend to be cheaper than their metallic counterparts (Sypeck, 2005). As with open-cell metal foams, open-cell polymeric foams do not trap moisture (Sypeck, 2005). Closed-cell polymer foam cores give increased thermal insulation at moderate weight, but creep even at ambient temperatures (McCormack et al., 2001).
The structure of polymeric foams is similar to that of metallic foams, but they do not exhibit metallic characteristics such as electrical conductivity. Unless protected, polymer structures used in aircraft construction suffer more damage from lightning strikes than metallic ones and allow significant proportions of lightning current to flow into onboard systems (e.g. electrical wiring) (Niu, 1996).

Fig. 2.3 Stress-strain curves for closed-cell rigid polyurethane foams of various densities, (Lu and Yu, 2003)

2.2.4 Truss cores

Work has been conducted to investigate the properties of miniature truss core sandwich panels (Sypeck and Wadley, 2002). These miniature truss core sandwich structures are similar in design to large engineering structures such as bridges and skyscraper frames. Fig. 2.4(a) shows a typical miniature truss core sandwich structure (Hyun et al., 2003).

Deshpande and Fleck (2001) have analysed tetragonal and pyramidal shaped trusses. They found that both cores display significant anisotropy and are susceptible to plastic buckling, resulting in bending asymmetry. Work has also been conducted to investigate the properties of 3D Kagomé truss core topologies, shown in Fig. 2.4(b) (e.g. Deshpande and Fleck, 2001; Hutchinson and Fleck, 2006; Hyun et al., 2003; Lim and Kang, 2006). It is found in Hyun et al. (2003)
that whilst tetragonal truss cores do not display isotropy after yielding, Kagomé cores do. The Kagomé core also has the greater load capacity and appears to be the superior core choice for ultra-light panels.

![Diagrams of sandwich panels](image)

Fig. 2.4 (a) LM22 truss core sandwich panel with solid facesheets, (Deshpande and Fleck, 2001); (b) Kagomé truss core panel, (Hyun et al., 2003)

Truss core sandwich panels tend to be advantageous for some applications since they may be fabricated with facesheets having a heavier gauge than those of honeycomb structures.

In the United States, around 40% of bridges are not able to handle present demands and require replacement. Approximately half the cost of bridge replacement comes from rerouting traffic during the construction process. Sandwich structures with truss cores offer a potential to provide pre-made deck panels that can be installed within days as opposed to the weeks required by traditional construction methods (McCormack, 2001).

Work by Wicks and Hutchinson (2001) as well as Deshpande and Fleck (2001) indicates that sandwich structures with periodic open-cell truss cores can be as stiff, strong, and light as hexagonal honeycomb core panels. Also, the open nature of truss cores means they do not trap moisture and could provide a dual function (e.g. they could potentially be used for the storage or drainage of fuel). Open-cell cores based upon tetrahedral truss concepts can allow fluids to easily flow through, making them less susceptible to internal corrosion and
depressurisation induced delamination (Sypeck and Wadley, 2002). However, truss core sandwich panels also tend to be expensive and difficult to manufacture, generally requiring batch type processing. Automated manufacture of miniature truss cores remains at present expensive (Sypeck and Wadley, 2002).

2.3 Analytical modelling methods of metal foams

The mechanical behaviour of metal foams is set by the cell structure and mechanical properties of the solid material. In an attempt to understand the mechanical response of foam materials under loading, analytical modelling tools have been developed for metal foams (Ryu et al., 2005).

The important length scale in metal foams is cell size, which is significantly large compared to the grain size that dictates properties in dense metals (Ramamurty and Paul, 2004). The unit cell of cellular/lattice materials is in the order of millimetres or micrometres, which allows them to be treated both as structures and materials. The lattices can be studied using traditional methods of mechanics, however one must also treat the lattice as a ‘material’ in its own right, with its own set of effective properties that allows a direct comparison with fully dense materials (Ashby, 2006).

Analytical methods to determine the basic properties of metal foams (e.g. the moduli and collapse stresses) are presented by Ashby and Gibson (1988). This analysis was extended to include size effects by Onck et al. (2001), in which the FE method was used – Onck et al.’s (2001) work is discussed in Section 2.4.1. Chen et al. (1999) examined the effects of periodic defects (i.e. cell waviness and non-uniform wall thickness) in altering the shape and size of the yield surface analytically, using a unit cell model for periodic hexagonal honeycombs. Ashby and Gibson’s (1988) as well as Chen et al.’s (1999) findings are summarised in Table 2.1.
Table 2.1 Unit cell structures and associated equations used in the analytical modelling of open-cell metal foams, after (Ashby and Gibson, 1988; Chen et al., 1999)

<table>
<thead>
<tr>
<th>Model and Unit Cell</th>
<th>Key Equations</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ashby and Gibson (1988) idealised cubic open-cell model</td>
<td>Bending-dominated</td>
</tr>
</tbody>
</table>
|  | \[
\frac{\bar{\sigma}_{pl}}{\sigma_{ys}}\alpha \left( \frac{\rho}{\rho_s} \right)^{3/2}
\] (2.1) |
|  | Buckling-dominated |
|  | \[
\frac{\bar{\sigma}_{el}}{E_s} \alpha \left( \frac{\rho}{\rho_s} \right)^2
\] (2.2) |
|  | Fracture-dominated |
|  | \[
\frac{\bar{\sigma}_{cr}}{\sigma_{MOR}}\alpha \left( \frac{\rho}{\rho_s} \right)^{3/2}
\] (2.3) |
| Chen et al. (1999) unit cell of a regular honeycomb with cell waviness | \[\delta = w_o\sin\left(\frac{2\pi x}{l}\right)\] (2.4) |
|  | \[
|Q\left(\frac{1}{2} - x_c\right) + Pw_c|
\] |
|  | \[-\sigma_{y}b\left[\frac{t^2}{4} - \frac{(P\cos\varphi_c + Q\sin\varphi_c)^2}{(2\sigma_{y}b)^2}\right] = 0\] (2.5) |
| Chen et al. (1999) unit cell of a regular honeycomb with non-uniform wall thickness | \[
|Q\left(\frac{1}{2} - x_c\right) + Pw_c| - \sigma_{y}b\left[\frac{t_c^2}{4} - \frac{p^2}{(2\sigma_{y}b)^2}\right] = 0\] (2.6) |
|  | \[t_c = t_1 - \frac{2x_c(t_1 - t_2)}{l}\] (2.7) |
|  | \[t_1 = t + \frac{l\tan\theta}{4}\] (2.8) |
|  | \[t_2 = t - \frac{l\tan\theta}{4}\] (2.9) |
For isotropic, open-cell foams Ashby and Gibson (1988) considered the cubic model shown in Table 2.1. It consists of solid struts surrounding a void space and connected at joints. Cellular solids are characterised by their relative density, which for the structure shown in Table 2.1 (with \( t \ll l \)) is given by:

\[ \frac{\rho}{\rho_s} \alpha \left( \frac{t}{l} \right)^2 \]  

(2.10)

where \( \rho \) is the density of the foam, \( \rho_s \) is the density of the solid of which it is made, \( l \) is the length of the cell edges, and \( t \) is the thickness of the cell edges.

Three possible collapse mechanisms exist under compressive loading: plastic bending of the cell edges, elastic buckling of the cell edges, or successive fracturing of the cell edges. The one that requires the lowest stress dominates. The analysis by Ashby and Gibson (1988) produced the equations shown in Table 2.1 when the foam is loaded in compression. In Eqns. (2.1) to (2.3), \( \sigma_{pl} \) is the plateau stress, \( \sigma_{ys} \) is the yield strength of the solid of which the foam is made, \( \sigma_{el} \) is the stress that causes the foam to collapse by elastic buckling, \( E_s \) is the modulus of the solid of which the strut is made, \( \sigma_{cr} \) is the crushing stress, and \( \sigma_{MOR} \) is the modulus of rupture of the solid strut. The constants of proportionality can then be determined by fitting experimental data. Ashby and Gibson (1988) found that experimental data for open-cell foams that collapse plastically are adequately described by Eqn. (2.1) when the constant of proportionality is approximately equal to 0.3. Experimental data for elastomeric open-cell foams are adequately described by Eqn. (2.2) when the constant of proportionality is approximately equal to 0.05, and those for brittle foams are adequately described by Eqn. (2.3) when the constant of proportionality is approximately equal to 0.2.

Most closed-cell foams also follow the above scaling laws, which is unexpected as the cell faces must carry membrane stresses when the foam is loaded. However, the cell faces are very thin and rupture or buckle at such low stresses that their contribution to stiffness and strength is small, leaving the cell
edges to carry most of the load (Ashby, 2006). Ashby and Gibson’s (1988) analysis suggests that the hydrostatic strength of an isotropic metal foam is governed by cell wall stretching and scales with \( \frac{p}{\rho_s} \), whilst the uniaxial strength is controlled by cell wall bending and scales with \( (\frac{p}{\rho_s})^{3/2} \). These predictions neglect the effect of imperfections in the microstructure (waviness of cells, non-uniform cell wall thickness, etc.).

The key equations used in Chen et al.’s (1999) analysis of the effect of cell wall waviness and non-uniform cell wall thickness in altering the shape and size of the yield surface are presented in Table 2.1. Eqn. (2.4) describes the initial transverse deflection \( \delta \) for a wavy imperfection along each cell. \( w_o \) is the amplitude of the waviness, \( n \) is the number of ripples in a length \( l/2 \) (assuming each beam is symmetrical – see Table 2.1), and \( x \) is the distance from the mid-joint O along the cell edge. By considering a wavy beam of length \( l/2 \) and thickness \( t \) which is clamped at one end and subjected to a transverse force \( Q \) and axial force \( P \) at the other end, Chen et al. (1999) produced Eqn. (2.5) for the yield locus in \( (P,Q) \) space. In Eqn. (2.5) it is assumed that plastic collapse is by the formation of a plastic hinge at a distance \( x_c \) from the fixed end O. \( w_c \) and \( \psi_c \) denote respectively the initial transverse deflection and associated slope at \( x_c \). The effect of wavy imperfections on the yield strengths of perfect honeycombs is shown in Fig. 2.5(a) – \( \sigma_U \) is the uniaxial yield strength and \( \sigma_H \) is the hydrostatic yield strength (hydrostatic loading causes yielding in honeycomb and foam structures in contrast to the assumption for homogeneous solids) of a honeycomb with cell wall waviness, and \( \sigma_U^0 \) and \( \sigma_H^0 \) are the corresponding yield strengths of a perfect honeycomb (Chen et al., 1999).

As can be seen from Fig. 2.5(a), Chen et al. (1999) found that cell wall waviness reduces the hydrostatic yield strength of the regular honeycomb structure significantly. This is because cell wall waviness changes the deformation...
mechanism from cell wall stretching to cell wall bending under hydrostatic loading. The uniaxial yield strength is only slightly reduced as the deformation mechanism is cell wall bending for both a perfect and wavy honeycomb when loaded uniaxially.

Chen et al. (1999) considered the effect of non-uniform wall thickness on the size and shape of the yield surface by using the geometric model shown in Table 2.1. They assumed that the cell wall thickness decreases linearly from the joint O to the mid-point of the cell edge and they used simple beam theory such that the predictions are limited to small variations in cell wall thickness. They determined the yield surface by analysing the plastic collapse of a clamped beam whose thickness varies linearly with length. The critical load for collapse of such a beam is given by Eqn. (2.6), where $x_c$ is the distance of the plastic hinge from the built-in end. The thickness $t_c$ at $x_c$ is given by Eqns. (2.7) to (2.9). The effect of non-uniform wall thickness on the yield strengths of perfect honeycombs is shown in Fig. 2.5(b) (Chen et al., 1999).

As can be seen from Fig. 2.5(b), Chen et al. (1999) found that non-uniform wall thickness slightly reduces the hydrostatic yield strength – they explained this by noting that under hydrostatic loading the deformation mechanism is cell wall stretching, but the yield strength is reduced due to the thinning of the cell walls towards the midpoints of the struts. The uniaxial yield strength increases slightly with non-uniform wall thickness as in this instance yield is due to the formation of plastic hinges in the vicinity of the joint, such that a redistribution of the cell wall material towards the joint will increase the plastic collapse moment and hence the yield strength.

Chen et al. (1999) also studied the effects of random defects (i.e. cell-size variations, fractured cell walls, cell-wall misalignments, and missing cells) using the FE method. This approach and their findings are discussed in Sections 2.4.1 and 2.4.2.
Finally, Zhu et al. (1997) analytically determined the elastic constants (Young’s modulus, shear modulus, and Poisson’s ratio) of an open-cell foam loaded in uniaxial tension, using tetrakaidecahedral cells on a BCC lattice as shown in Fig. 2.6, by considering the bending, twisting and extension of the cell edges.

It is assumed in Zhu et al. (1997) that the material of the cell edges is an isotropic elastic solid with a Poisson’s ratio of 0.5. It is found that the Young’s modulus, shear modulus, and Poisson’s ratio can be expressed as functions of relative density, for an equilateral triangle cross-section, as follows:

\[
E = \frac{0.726E_s\rho_r^2}{1 + 1.09\rho_r} \tag{2.11}
\]

\[
G = \frac{0.2333E_s\rho_r^2}{1 + 0.700\rho_r} \tag{2.12}
\]

\[
v = 0.5 \left( \frac{1 - 1.09\rho_r}{1 + 1.09\rho_r} \right) \tag{2.13}
\]

where \(\rho_r\) is the relative density of the open-cell foam, \(E\) is the Young’s modulus of the foam (note that for the isotropic foam analysed, \(E_1 = E_2 = E_3\)), \(E_s\) is the Young’s modulus of the solid of which the foam is made, \(G\) is the shear modulus.
of the foam, and \( \nu \) is the Poisson’s ratio of the foam. In Zhu et al. (1997) it is noted that the relationship between the relative density and the width of cell edges depends on the cross-sectional shape – indeed, the Young’s modulus is found to be 38% higher, for a given relative density, if the edge cross-sections are Plateau borders rather than equilateral triangles.

![Diagram of tetrakaidecahedral cells in a BCC lattice](image)

Fig. 2.6 Three tetrakaidecahedral cells in a BCC lattice, with a lattice repeat vector shown. Typical members under load, when a tensile stress is applied along the z axis, are shown in bold, (Zhu et al., 1997)

### 2.4 Numerical modelling methods of metal foams

In an attempt to understand the mechanical response of foam materials under loading, numerical modelling tools have been developed for metal foams. These can be separated under two distinct techniques, given below, that are now reviewed in turn (Betts, 2012):

- **Finite element methods utilising a repeating unit cell** such as a tetrakaidecahedron; and
Finite element methods utilising the random Voronoi technique. This approach gives a more accurate representation of the cell geometry of the foams (Zhu et al., 2000).

2.4.1 FE methods utilising a repeating unit cell

The second approach to studying the mechanical behaviour of metal foams is to analyse a repeating unit cell, such as a tetrakaidecahedron, using finite elements. Simone and Gibson (1998) used FE analysis of idealised 2D (hexagonal honeycomb) and 3D (closed-cell tetrakaidecahedral foam) cellular materials to consider the effect of the distribution of solid between cell faces and edges on the mechanical properties. Specifically, they used FE analysis to estimate the relative elastic modulus and relative plastic collapse strength as a function of relative density and the distribution of solid material. For the 3D case, they utilised the Kelvin tetrakaidecahedron unit cell as it is the lowest energy unit cell known consisting of a single polyhedron.

The tetrakaidecahedron cell is defined by six planar square faces and eight hexagonal faces that are non-planar, but have zero mean curvature. The analysis in Simone and Gibson (1998) makes the simplification that all cell faces are planar so as to eliminate non-linearities caused by wall curvature and isolate the effects of the solid distribution. An aggregate of the tetrakaidecahedral cells used to model the closed-cell foam is shown in Fig. 2.7(a).

It is found in Simone and Gibson (1998) that shifting material away from the cell faces of a closed-cell foam into Plateau borders along the edges has minimal effect on the elastic modulus and causes a reduction in the peak stress, which is shown in Fig. 2.7(b). This is accounted for in Simone and Gibson (1998) by noting that closed-cell foams deform primarily by the in-plane stretching of the
cell faces, and shifting material towards the cell edges reduces the net cell face area with respect to in-plane axial deformation – thus leading to a reduction in stress.

The foam behaves more like an open-cell foam as the volume fraction of the solid shifted to the edges increases to the point where almost all of the solid material is in the cell edges as opposed to the faces.

Fig. 2.7 (a) Idealised tetrakaidecahedral foam structure; (b) Normalised elastic modulus and normalised peak stress vs. fraction of solid in the Plateau borders for tetrakaidecahedral foams, (Simone and Gibson, 1998)

Onck et al. (2001) modelled the effect of the size of a rigid indenter on the indentation strength of a regular hexagonal honeycomb of unit depth using the FE analysis program ABAQUS. A honeycomb of sufficient size to eliminate any
influence of the boundaries was selected. Each cell wall was modelled using beam elements and the solid cell wall material was assumed to be elastic-perfectly plastic with $E_s = 70$ GPa, $\nu_s = 0.3$, and $\sigma_{ys} = 300$ MPa. The relative density was taken to be 0.09. The indenter was displaced uniformly into the honeycomb whilst the opposite edge of the honeycomb was fixed in the direction of indenter displacement and free to translate in the normal direction.

Fig. 2.8 shows Onck et al.’s (2001) results as a plot of indentation strength normalised with respect to compressive strength vs. the ratio of indenter width to cell size.

![Normalized indentation strength plot vs. w/S ratio](image)

**Fig. 2.8 Normalised indentation strength plotted against the ratio of indenter width to cell size, (Onck et al., 2001)**

It can be ascertained from Fig. 2.8 that the indentation strength decreases as the ratio of indenter size to the cell size increases. Onck et al. (2001) explain that this trend may be understood by noting that the total load on the indenter is equal to the sum of that required to crush the honeycomb beneath the indenter and that required to fully yield the cell walls at the perimeter of the indenter, resulting in the following equation:
\[
\frac{\sigma_{\text{ind}}}{\sigma_{\text{pl}}} = 1 + C_2 \frac{\tau_{\text{pl}}^* S}{\sigma_{\text{pl}}^* W}
\]  
\tag{2.14}

where \(\sigma_{\text{ind}}\) is the indentation strength, \(\sigma_{\text{pl}}^*\) is plastic collapse stress of the honeycomb, \(\tau_{\text{pl}}^*\) is the plastic shear strength, and \(S/w\) is the ratio of cell size to indenter width. Eqn. (2.14) has been plotted in Fig. 2.8, and it can be seen that there is good agreement with the FE results when \(C_2 = 7.23\).

Moreover, Andrews et al. (2001) compared Eqn. (2.14) developed in Onck et al. (2001) to experimental results for axisymmetric indentation tests carried out on a closed-cell aluminium foam, and found that the indentation data are well described by Eqn. (2.14).

Hodge and Dunand (2003) developed a 3D FE model to predict the creep properties of nickel-rich NiAl foams. They used a repeating unit cell consisting of three orthogonal sets of four parallel hollow or solid struts with square cross-section, connecting at joints arranged on a square lattice – see Fig. 2.9(a). They considered two relative densities for both types of struts (hollow and solid). It should be noted that the actual architecture of the NiAl foam is much less regular than the simple cubic lattice shown in Fig. 2.9(a). Whilst the model is a simplified representation of the general geometry of the NiAl foam, it does capture important features such as hollow struts and 3D periodicity.

In Hodge and Dunand (2003), creep of the NiAl material within the struts and joints is assumed to take place according to the power-law equation:

\[
\dot{\varepsilon} = A_D \sigma^{n_{\text{stress}}} \exp \left( \frac{-Q_{\text{act}}}{R_g T} \right)
\]  
\tag{2.15}

where \(A_D\) is the Dorn constant, \(n_{\text{stress}}\) is the stress exponent, \(Q_{\text{act}}\) is the activation energy, \(R_g\) is the gas constant, \(T\) is the temperature, \(\dot{\varepsilon}\) is the steady-state strain rate, and \(\sigma\) is the uniaxial applied stress.

It is found in Hodge and Dunand (2003) that the 3D FE model predicts creep rates in reasonable agreement with experimental data from creep tests.
between 1073 K and 1375 K with compressive stresses between 0.1 MPa and 1.5 MPa of NiAl foams consisting of open-cells with hollow struts – see Fig. 2.9(b).

Based on the numerical results from the FE model, a simplified analytical model is proposed in Hodge and Dunand (2003) whereby struts parallel to the applied stress deform by creep in a purely compressive mode, whilst perpendicular struts prevent buckling but provide no directional load bearing capacity. It is found that the analytical model produces results that are very similar to the predictions of the numerical model and in good agreement with the experimental data.

![Graph](image)

Fig. 2.9 (a) Geometric FE model for foam with hollow struts (relative density = 5 %). In the solid strut model, struts have smaller width but same cross-sectional area; (b) Compressive strain rate vs. stress curves at 1173 K as measured experimentally on 20 pores per linear inch (ppi) foams and as calculated by FE for hollow struts, (Hodge and Dunand, 2003)

A significant limitation of the above unit cell modelling approach is that it does not capture the natural variations in microstructure that are observed in most cellular materials – for instance, cell-size variations. FE models that allow for these to be accounted for are discussed in Section 2.4.2.

Chen et al. (1999) considered the effect of cell wall misalignments using the FE method by displacing in random directions the joints of a perfect hexagonal
honeycomb by a constant distance $\alpha l$, where $l$ is the length of each side of the honeycomb, and the fraction $\alpha$ gives the magnitude of the imperfection. The displaced cellular structure for the case $\alpha = 0.2$ is shown in Fig. 2.10(a). The uniaxial yield strength $\sigma_U$ and the hydrostatic yield strength $\sigma_H$ of a honeycomb with cell wall misalignments, normalised by the corresponding yield strengths $\sigma_U^0$ and $\sigma_H^0$ of a perfect honeycomb, are plotted as functions of the imperfection measure $\alpha$ in Fig. 2.10(d) for a honeycomb of relative density 0.10. For completeness, the ratio of uniaxial to hydrostatic yield strength $\sigma_U/\sigma_H$ of the imperfect honeycomb is included in Fig. 2.10(d). It is concluded from Fig. 2.10(d) that the cell wall misalignments lead to a large reduction in hydrostatic strength. Chen et al. (1999) also noted that by varying the bending to stretching strength ratio of a beam element, it was found that under hydrostatic stressing the deformation mechanism of cell-wall bending dominates over cell-wall stretching as $\alpha$ increases.

Chen et al. (1999) also considered the effect of fractured cell walls on the strength of foams by randomly removing 1%, 5%, and 10% of the cell edges of a perfect honeycomb. Fig. 2.10(b) shows the FE mesh of a perfect honeycomb with 1% of its cell edges removed randomly. Chen et al.’s (1999) results are shown in Fig. 2.10(e), which plots the uniaxial and hydrostatic yield strengths of a honeycomb with fractured cell walls, normalised by the corresponding strengths for a perfect honeycomb, as a function of the percentage of fractured cell walls for an initially perfect honeycomb of relative density 0.10. For completeness, the ratio of uniaxial to hydrostatic yield strength $\sigma_U/\sigma_H$ of the imperfect honeycomb is included in Fig. 2.10(e). It can be seen from Fig. 2.10(e) that the removal of cell walls results in pronounced weakening. Chen et al. (1999) noted that by varying the bending to stretching strength ratio of a beam element, the bending of cell walls was found to be the dominant deformation mechanism under hydrostatic loading for perfect honeycombs with fractured cell walls. This differs from the hydrostatic
compressive behaviour of honeycombs without fractured cell walls, which is dominated by cell-wall stretching.

Finally, Chen et al. (1999) analysed the effect of missing cells by removing a cluster of adjacent cells within a perfect hexagonal honeycomb structure – see Fig. 2.10(c). Their results are shown in Fig. 2.10(f), where it can be deduced that the presence of a single hole reduces the hydrostatic strength significantly – the presence of a hole induces bending of the cell walls for hydrostatic loading, which produces the reduction in strength.
Fig. 2.10 Typical FE mesh for honeycombs with (a) cell wall misalignment ($\alpha = 0.2$); (b) fractured cell walls (number fraction = 1%); (c) 1 cell missing. Effect on uniaxial and hydrostatic yield strengths of 2D foams with relative density 0.10 of (d) cell wall misalignments; (e) fractured cell walls; and (f) missing cells, after (Chen et al., 1999)
2.4.2 FE methods utilising the random Voronoi technique

The third approach to studying the mechanical behaviour of metal foams is to model the real foam structure (which can be represented as a stacking of randomly distributed cells of various shapes and sizes which fill the space completely) by the random Voronoi technique. This technique can produce a geometrically more realistic model of the foam structure (Shulmeister et al., 1998).

The objective is to create FE models in a manner that is similar to the way metal foams are produced in reality, and to make the models large enough to provide reliable input for generation of homogenised engineering properties (Hallstrom and Ribeiro-Ayeh, 2005). This is achieved by generating distributions of cell nuclei (or points) in space numerically, and simulating cell growth around the nuclei through generation of Voronoi tessellations. Voronoi tessellations are a form of space decomposition; given a set of N points in a plane, a Voronoi tessellation divides the domain in a set of polygonal regions, the boundaries of which are the perpendicular bisectors of the lines joining the points. Each polygon contains only one of the N points. The resulting structure is a Voronoi foam. If the nuclei in the model are distributed periodically, the cellular foam microstructure is then regular. A random distribution of nuclei is modelled with a random space tessellation. The randomly distributed points then become the centres of the foam cells; flat cell boundary faces appear where two neighbouring cells come into contact. Open-cell foams can be modelled by locating the struts where three cell faces meet and subsequently removing the cell faces. The initial distribution of the nuclei completely determines the final geometry of the Voronoi tessellation and hence the foam microstructure (Shulmeister et al., 1998).

Zhu et al. (2000) investigated how cell irregularities affect the elastic properties of open-cell foams using 3D FE analysis. They constructed 3D random structures by first defining a periodic cubic control volume (or representative
volume element). A distribution of virtual cell nuclei was then placed in the representative volume element by generating x, y, and z coordinates independently from the pseudo-random numbers between 0 and 1, starting at one corner of the cube. Once the first point is specified, each following random point is accepted only if it is greater than a minimum allowable distance $\delta_d$ from any existing point, until N nuclei are seeded in the cube. The cell nuclei are defined as the centre points of solid spheres that are packed in the space (Zhu et al., 2000; Hallstrom and Ribeiro-Ayeh, 2005). The regularity of a 3D Voronoi tessellation can be measured by Eqn. (2.16), viz.:

$$\alpha_V = \frac{\delta_d}{d_0}$$  \hspace{1cm} (2.16)

where $d_0$ is the minimum distance between any two adjacent nuclei in a regular lattice with N identical cells. To construct a random tessellation, the maximum $\delta_d$ should be less than $d_0$, else it is impossible to obtain N cells. For a regular lattice, $\delta_d = d_0$. For a totally random tessellation, $\delta_d = \alpha_V = 0$ (Zhu et al., 2000).

All struts in the foam are represented mechanically by beams rigidly connected in vertices. It is assumed for simplicity in Zhu et al. (2000) that all the struts have the same and constant cross-section, and thus the analyses were limited to models having low relative density. The results of Zhu et al. (2000) are given in Fig. 2.11 and suggest that, for low density foams, highly irregular foams have a greater Young’s modulus and shear modulus, and smaller bulk modulus than a perfect foam. It was also found that the Poisson’s ratio does not change with cell regularity, but does reduce gradually with increasing relative density (see Fig. 2.11 and Fig. 2.12).
Fig. 2.11 Effects of cell regularity on reduced (a) Young’s modulus, (b) bulk modulus, (c) shear modulus, and (d) Poisson’s ratio of random Voronoi foams having a constant relative density of 0.01 – diamond points represent theoretical results, and the bars represent computational results, after (Zhu et al., 2000)

Fig. 2.12 Effects of relative density on the Poisson’s ratio of random Voronoi foams with varying degrees of regularity parameter $\alpha_V$, (Zhu et al., 2000)
Huang and Gibson (2003) created an open-cell Voronoi structure and inputted it into ABAQUS to analyse the steady state creep response of a foam. The creep response of foams is of interest when they are used at high temperatures relative to their melting point. For instance, metal foams are attractive for heat transfer devices and lightweight structural sandwich panels, both of which may require them to be used at high temperatures. It was found that the creep response of a damaged Voronoi foam with struts randomly removed increases rapidly as the fraction of struts removed increases – see Fig. 2.13.

Chen et al. (1999) studied the effects of cell-size variations using the FE method. They considered the multi-axial yield response of 2D honeycombs with both Γ-Voronoi and δ-Voronoi distributions of cell size (for the former, the minimum distance separating two adjacent generation points is unconstrained, whilst for the latter it must be larger than a minimum prescribed value). Their findings are shown in Fig. 2.14 – it can be seen that the hydrostatic strengths of these two structures are less than that of a perfect honeycomb by a factor of 2 or 3, though the microstructures are not sufficiently dispersed in cell size to switch the deformation response from cell-wall stretching to cell-wall bending under hydrostatic loading. Furthermore, they found that the uniaxial elastic and plastic properties of random Voronoi models are well described by those of a perfect honeycomb, regardless of whether the cells are distributed according to the Γ- or δ-law. Comparable results for the elastic properties of a δ-Voronoi distributed honeycomb are presented by Silva et al. (1995) for 2D foams, and by Grenestedt and Tanaka (2000) for 3D foams.

Chen et al. (1999) went on to consider the effect of fractured cell walls in Γ-Voronoi structures, and it was found that the yield behaviour of a Γ-Voronoi structure with relative density 0.10 is essentially the same as that shown in Fig. 2.10(e) for a perfect honeycomb. This can be explained by the fact that fractured
cell walls have a much stronger effect than variations in cell size on reducing the yield strength of 2D foams.

Fig. 2.13 The creep rate of a damaged Voronoi foam, normalised by that of the intact foam at the same nominal relative density, plotted against the fraction of struts removed, (Huang and Gibson, 2003)

Fig. 2.14 Effect of size variations and cell wall misalignments on the hydrostatic yield strength of 2D cellular foams, (Chen et al., 1999)
2.5 Modelling of metal foams: relative benefits and future trends

The ability to accurately model metal foams is of prime importance to their application in industry. For example, as noted in Section 1.1, the use of sandwich panels in commercial aviation is at present limited to secondary structures only. The capability to accurately model the behaviour of metal foams is necessary if metal foam core sandwich panels are to be used as primary aircraft structures, enabling, for instance, the extent of subsurface core damage to be investigated for different loading scenarios.

FE models that utilise repeating unit cells are unable to capture the natural variations in the microstructure, however they do still capture some of the important features of foams such as 3D periodicity and the cross-sectional shape of the cell walls; such models have been shown to provide good agreement with experimental data and theoretical predictions (Hodge and Dunand, 2003; Onck et al., 2001). FE models utilising the random Voronoi technique can produce a geometrically more realistic model (and are hence more representative) of the foam structure than analytical methods or FE methods that utilise a repeating unit cell (Huang and Gibson, 2003; Zhu et al., 2000). However, they can require longer model construction and running times and hence cost.

Analytical methods do not take into account the effect of imperfections in the microstructure, and are simplified representations of foam structures. However, they do provide a means to quickly assess the mechanical properties of a foam before proceeding to more complicated numerical methods, and can provide a benchmark for validation of a new FE model. Moreover, theoretical models of foam mechanics permit the identification of the deformation mechanisms that control mechanical behaviour – in FE analysis, there may be no physical understanding of the dominant deformation mechanisms (Zhu et al., 1997).
FE methods are also being developed that utilise a 3D tomographic image (a non-destructive visualisation of a foam at the scale of its cellular microstructure obtained by XMT) of a real foam as the geometric description of the model (see Fig. 2.15) (Maire et al., 2003; Youssef et al., 2005). Such techniques could model both open-cell as well as closed-cell foams and could prove to be useful in predicting the mechanical response of cellular materials.

Fig. 2.15 3D tomography image of a closed-cell foam, (Youssef et al., 2005)
Chapter 3.

2D finite element and analytical modelling of honeycomb core sandwich panels
3.1 Introduction

This Chapter aims to shed light on the mechanics of complex 3D foams by conducting FE and analytical modelling of 2D regular honeycomb structures. Foams consist of cell walls that form an intricate 3D network which distorts during deformation in ways which are difficult to identify; honeycombs are much simpler. This Chapter sets out to establish a repeating unit cell 2D FE modelling procedure to predict the mechanical behaviour of infinitely long, regularly tessellated hexagonal honeycomb core sandwich panels (e.g. Young’s modulus, energy absorbed, etc.).

The length and diameter of individual open-cell metal foam struts can vary significantly between different foams; for instance, the commercial open-cell metal foam ERG Duocel has a strut aspect ratio (length to thickness) of approximately 8.3 (Onck et al., 2004), whilst the open-cell metal foam studied in this work (see Chapter 4) has a strut aspect ratio of approximately 3.0. Therefore, the 2D FE models in this Chapter utilise Timoshenko beam elements (as opposed to Euler-Bernoulli beam elements) as these are efficient for both thin and thick beams.

The FE results are compared to an analytical model developed in Section 3.5 that utilises Timoshenko beam theory to determine the Young’s modulus of a hexagonal honeycomb core.

The 2D FE model is then used in a comparative study of optimal cell shapes for a given application (e.g. energy absorption, lightweight structural applications, etc.) using the three existing 2D regular space-filling tessellations: those constructed from squares, equilateral triangles, and hexagons.
3.2 Hexagonal honeycomb FE model development

3.2.1 Overview of FE model

FE modelling of regular honeycombs has been conducted in this work using the software ABAQUS 6.11. The model consists of a set area divided into a regular hexagonal tessellation. The honeycomb is enclosed at the top and bottom by solid 0.8 mm thick facesheets to create a sandwich structure. The dimensions of the hexagonal cells were based on those of the open-cell foam ERG Duocel (http://www.ergaerospace.com): the strut length was set to 1.5 mm, and the thickness of the struts was set to 0.18 mm (Onck et al., 2004). The relative density of the regular hexagon tessellation can be approximated as follows (Ashby and Gibson, 1988):

$$\frac{\rho}{\rho_s} = \frac{2}{\sqrt{3}} \left( 1 - \frac{1}{2\sqrt{3}} \frac{t}{l} \right) = 13.4 \%$$

where \( t \) is the cell strut thickness and \( l \) is the strut length (with \( t \ll l \)).

For practical applications, the thickness of the sandwich panel will be of a specified value in the order of several millimetres to centimetres. However, the length of the sandwich panel could be up to several metres long; it would be both impractical and computationally inefficient to physically model the full panel length. Therefore, the length of the sandwich panel was progressively increased within the FE model until convergence of the stress-strain plots was achieved so as to identify the smallest length possible that could be modelled while taking into account edge effects. Fig. 3.1 illustrates this principle.
3.2.2 Applied boundary conditions and loads

The applied boundary conditions are shown in Fig. 3.1. A symmetry boundary condition was applied across the horizontal centreline of the sandwich panel. The left face of the upper facesheet was constrained in the horizontal direction.

A uniform compressive load was modelled by applying a multi-point constraint (MPC) along the top face of the upper facesheet, whereby all nodes along that face were tied to the central node. Compressive load vs. displacement
plots were obtained by moving this central node in the vertical direction under a controlled, linear displacement.

### 3.2.3 Material model

The cell walls of the honeycomb and the facesheets were assigned the material properties of aluminium alloy Al-7075-0: \( \rho_s = 2800 \text{ kg/m}^3 \), \( E_s = 71.7 \text{ GPa} \), \( \sigma_{ys} = 145 \text{ MPa} \), \( \nu_s = 0.33 \). The flow stress was assumed to be given by (El-Domiaty et al., 1996):

\[
\sigma = 400e^{0.17} \text{ MPa}
\]  

(3.2)

where \( \sigma \) is the engineering stress, and \( \varepsilon \) the engineering strain.

### 3.2.4 Element type, profiles, and time step

The walls of the honeycombs were modelled as beam elements having solid square cross-section, in an analogous manner to work by Onck et al. (2001), Silva et al. (1995), and Zhu et al. (2000). A beam element is a 1D line element in the X-Y plane that has stiffness associated with deformation of the line (the beam's “axis”). These deformations consist of axial stretch/compression and curvature change (bending). The main advantage of beam elements is that they are geometrically simple and have few degrees of freedom.

Specifically, the Timoshenko beam B21 element was used. This allows for transverse shear deformation (Timoshenko, 1956). ABAQUS assumes that the transverse shear behaviour of Timoshenko beams is linear elastic with a fixed modulus and, hence, independent of the response of the beam section to axial stretch and bending. These elements in ABAQUS are formulated so that they are efficient for thin beams – where Euler-Bernoulli theory is accurate – as well as for thick beams: because of this they are the most effective beam elements in ABAQUS (ABAQUS, 2012).
The B21 element linearly interpolates the displacement field. Multiple beam elements were necessary to model each strut in order to adequately capture the deflection behaviour of the struts and the mechanical response of the honeycomb. Mesh sensitivity analysis established that 15 beam elements were necessary to model each strut. The upper facesheet was modelled as a shell planar feature comprising of elements of dimensions 0.1 x 0.1 mm (so, for a model length of 10 mm the upper facesheet comprised of 800 elements).

3.2.5 Connector assignments, constraints, and surface interactions

The honeycomb struts were tied to one another. The joints between the honeycomb struts were constrained in the U1, U2, and U3 translational directions, as well as the UR1, UR2, and UR3 rotational directions. In ABAQUS, this was specified as follows:

- Translational connector type: Join
- Rotational connector type: Align

The struts at the honeycomb/facesheet interface were tied to the facesheet. Each strut and the facesheet was assigned a tangential frictionless surface interaction property to all part instances in their line of sight.

3.3 Hexagonal honeycomb FE results

Fig. 3.2 shows the hexagonal honeycomb stress-strain graph for a relative density of 13.4 % (i.e. for a strut length of 1.5 mm and a strut thickness of 0.18 mm), assuming an elastic-plastic material model. The graph displays a trend associated with elastic-plastic honeycombs (Ashby and Gibson, 1988). There are three distinct regions: a linear-elastic regime, followed by a plateau of roughly constant stress, and finally a regime of steeply rising stress. This behaviour is also typically observed in commercial open-cell foams (Ashby et al., 2000).
The stress-strain behaviour of the honeycombs is described by the different mechanisms of deformation for each region, and can be observed directly from the FE simulations. For the hexagonal honeycomb, the processes are as follows:

- The cell walls initially bend elastically, resulting in linear-elasticity;
- Once a critical stress is reached the cells begin to collapse. The cell walls collapse due to the formation of plastic hinges at the section of maximum moment in the bent members;
- Finally, the cells collapse to such an extent that opposing cell walls touch one another. This explains the densification region of the load-displacement graph.

The effect of increasing the length of the sandwich panel can be observed from Fig. 3.2. As the length is increased from 9 mm to 90 mm, so too are the effective Young’s modulus and peak stress. The increase in effective Young’s modulus is notable, varying from 36 MPa to 182 MPa. The initial loading peak stress also varies significantly, from 166 kPa to 522 kPa. There is less variation in the plastic properties, and the densification strain is roughly the same for all the models (Betts et al., 2012).

As the stress-strain plots do not converge even at a sandwich panel length of 90 mm, it is necessary to investigate implementing PBCs at the left and rightmost nodes of the model to describe a sandwich panel of infinite length. This is discussed in Section 3.4.
Fig. 3.2 Stress-strain curves determined using FE models with different sandwich panel lengths (L). The stress-strain relationships can be divided into three regions as shown: (I) Elastic region; (II) Plastic collapse followed by plateau; (III) Densification, (Betts et al., 2012)
3.4 Implementation of PBCs

PBCs have been applied to previous FE models of metal foams and honeycombs to simulate an infinite array of cells connected to each other (e.g. Hodge and Dunand, 2003; Huang and Gibson, 2003; Zhu et al., 2000). PBCs effectively eliminate edge effects from the mechanical analysis.

For the 2D case, PBCs assume that for any two corresponding beam nodes on the vertical boundaries of the model, the nodes have the same relative displacement in the vertical and horizontal directions and the same rotational angle in the X-Y plane. This is represented by Eqn. set (3.3).

\[
\begin{align*}
U_1^{\text{LHS}} - U_1^{\text{RHS}} &= 0 \\
U_2^{\text{LHS}} - U_2^{\text{RHS}} &= 0 \\
\varphi^{\text{LHS}} - \varphi^{\text{RHS}} &= 0
\end{align*}
\]  

(3.3)

where the superscript LHS denotes a node on the left vertical boundary, and RHS is the corresponding node on the right vertical boundary (see Fig. 3.3). The subscripts 1 and 2 denote the respective degree of freedom (DOF) of the node.

The above PBCs were applied to the FE model outlined in Section 3.2, for a facesheet length of 9 mm. PBCs were then applied to the same model for a facesheet length of 45 mm to verify the convergence of the results. Fig. 3.4 shows the stress-strain plots for the two models, and it can be seen that there is a good agreement between the two. The effective Young’s modulus varies by 0.2 %, whilst the peak stress differs by 0.9 % – it is therefore concluded that an infinitely long sandwich panel may be modelled by a facesheet length of 9 mm with PBCs (Betts et al., 2012).
Fig. 3.3 Hexagonal honeycomb FE model with PBCs enclosed by facesheets, (Betts et al., 2012)

Fig. 3.4 Comparison of stress-strain relationships calculated using different FE model sizes (L = 9 mm and 45 mm) with PBCs to demonstrate convergence, (Betts et al., 2012)

Fig. 3.5 plots the effective Young’s modulus and peak stress for the FE models of Fig. 3.2 – i.e. a facesheet length of 9 mm, 36 mm, and 90 mm without PBCs – and compares these to the values obtained using PBCs. It can be ascertained that as the model length increases, it tends towards the solution with PBCs.
Fig. 3.5 Convergence of FE models towards PBC solution for (a) effective Young’s modulus, and (b) peak stress, (Betts et al., 2012)
3.5 FE model validation

Ashby and Gibson (1988) have previously predicted the Young’s modulus of a regular hexagonal honeycomb using Euler-Bernoulli beam theory. In an analogous method to that of Ashby and Gibson (1988), Timoshenko beam theory is now used to analytically determine the Young’s modulus of a honeycomb in the $X_2$ direction (see Fig. 3.6). Timoshenko beam theory is preferred as it accounts for the effects of transverse shear strain, which are not captured by Euler-Bernoulli beam theory (Timoshenko, 1956). The latter therefore under-predicts deflections and thus over-predicts beam stiffness. For a homogeneous beam of constant cross-section, Timoshenko beam theory provides the following differential equation to describe the relationship between the beam’s deflection and the applied load:

$$E_s I \frac{d^4 w}{dx^4} = q(x) - \frac{E_s I}{kAG_s} \frac{d^2 q}{dx^2}$$  \hspace{1cm} (3.4)

Consider a honeycomb comprising of regular hexagons of square cross-section and compressed in the $X_2$ direction, as shown in Fig. 3.6.

Fig. 3.6 Free body diagram (FBD) of an individual strut subjected to uniaxial compression for use in the Timoshenko analytical solution, (Betts et al., 2012)
By equilibrium, \( Q = 0 \). Separating the load \( P \) into components parallel and normal to the beam (denoted \( P_p \) and \( P_n \) respectively), the following loading equation for the beam can be written (using discontinuity functions for beam equations, see Soutas-Little et al. (2008)):

\[
q(x) = P_n(x - 0)_{-1} - P_n(x - l)_{-1} - M(x - 0)_{-2} - M(x - l)_{-2}
\]  
(3.5)

This equation is valid for all values of \( x \) from minus infinity to plus infinity, although the beam only exists between \( x = 0 \) and \( x = l \). Inserting the expression for \( q(x) \) of Eqn. (3.5) into Eqn. (3.4) and integrating gives:

\[
E_s I \frac{d^3w}{dx^3} = P_n(x - 0)^0 - P_n(x - l)^0 - M(x - 0)_{-1} - M(x - l)_{-1}
\]

\[
- \frac{E_s I}{kA G_s} (P_n(x - 0)_{-2} - P_n(x - l)_{-2} - M(x - 0)_{-3}) - M(x - l)_{-3} + C_1
\]

Integrating Eqn. (3.6) gives:

\[
E_s I \frac{d^2w}{dx^2} = P_n(x - 0)^1 - P_n(x - l)^1 - M(x - 0)^0 - M(x - l)^0
\]

\[
- \frac{E_s I}{kA G_s} (P_n(x - 0)_{-1} - P_n(x - l)_{-1} - M(x - 0)_{-2}) - M(x - l)_{-2} + C_1 x + C_2
\]

The constants \( C_1 \) and \( C_2 \) can be evaluated by noting that at \( x = 0_- \) (i.e. at a point just below \( x = 0 \)):

\[
\frac{d^3w(0_-)}{dx^3} = 0 = C_1
\]

\[
\frac{d^2w(0_-)}{dx^2} = 0 = C_2
\]

Integrating Eqn. (3.7) twice provides expressions for the slope and deflection of the beam, given by Eqns. (3.9) and (3.10) respectively.
\[ E_s I \frac{dw}{dx} = \frac{P_n}{2} (x - 0)^2 - \frac{P_n}{2} (x - l)^2 - M(x - 0)^1 - M(x - l)^1 \]

\[ - \frac{E_s I}{kAG_s} (P_n (x - 0)^0 - P_n (x - l)^0 - M(x - 0)_{-1} - M(x - l)_{-1}) + C_3 \] (3.9)

\[ E_s I \delta = \frac{P_n}{6} (x - 0)^3 - \frac{P_n}{6} (x - l)^3 - \frac{M}{2} (x - 0)^2 - \frac{M}{2} (x - l)^2 - \]

\[ \frac{E_s I}{kAG_s} (P_n (x - 0)^1 - P_n (x - l)^1 - M(x - 0)^0 - M(x - l)^0) + C_3 x + C_4 \] (3.10)

The constants \( C_3 \) and \( C_4 \) can be evaluated by noting that at \( x = 0_- \) (i.e. at a point just below \( x = 0 \)):

\[ \frac{dw(0_-)}{dx} = 0 = C_3 \] (3.11)

\[ w(0_-) = \delta(0_-) = 0 = C_4 \]

Hence, from Eqn. (3.10) at \( x = 1 \) the deflection of the beam can be described as follows:

\[ E_s I \delta = \frac{P_n}{6} l^3 - \frac{M}{2} l^2 - \frac{E_s I}{kAG_s} (P_n l - M) \] (3.12)

Now, the moment tending to bend the cell wall is given by:

\[ M = \frac{P_l \sin \theta}{2} \] (3.13)

Inserting Eqn. (3.13) in Eqn. (3.12), and noting \( P_n = P \sin \theta \), gives:

\[ E_s I \delta = \frac{P l^3 \sin \theta}{6} - \frac{P l^3 \sin \theta}{4} - \frac{E_s I}{kAG_s} \left( P \sin \theta - \frac{P l \sin \theta}{2} \right) \] (3.14)

So:

\[ |\delta| = \frac{P l^3 \sin \theta}{12E_s I} + \frac{P l \sin \theta}{2kAG_s} \] (3.15)

Now:

\[ P = \sigma_2 t l (1 + \sin \theta) \] (3.16)

A component \( \delta \sin \theta \) of the deflection is parallel to the \( X_2 \) axis, giving a strain:
For a beam of square cross-section, \( I = \frac{t^4}{12} \) and the Young’s modulus is given by Eqn. (3.18) (using Eqn. (3.17)).

\[
\varepsilon_2 = \frac{|\delta| \sin \theta}{l \cos \theta} = \frac{\sigma_2 tl(1 + \sin \theta)\sin^2 \theta}{\cos \theta} \left( \frac{l^2}{12E_s I} + \frac{1}{2kG_s} \right)
\]

(3.17)

And noting that for a regular hexagon, \( \theta = 30^\circ \):

\[
E_2 = \frac{\sigma_2}{\varepsilon_2} = \frac{\cos \theta}{tl(1 + \sin \theta)\sin^2 \theta} \left( \frac{l^2}{E_s t^4} + \frac{1}{2kG_s} \right)
\]

(3.18)

For the honeycomb model of Section 3.4, \( t = 0.18 \) mm, \( l = 1.5 \) mm, \( E_s = 71.7 \) GPa, and \( G_s = 27.0 \) GPa. The shear coefficient, \( k \), is defined by ABAQUS for a rectangular (or square) cross-section to be equal to 0.85 (ABAQUS, 2012). So, from Eqn. (3.19) \( E_2 = 279 \) MPa (Betts et al., 2012).

The FE model of Section 3.4, with PBCs, displayed an effective Young’s modulus of 268 MPa, which is in good agreement with the Timoshenko solution (a 4 % difference). Fig. 3.7 shows the stress-strain plots for the FE models with facesheet lengths 9 mm, 36 mm, and 90 mm without PBCs, as well as that obtained using PBCs. The Timoshenko analytical solution has been superimposed on the plots.
Fig. 3.7 Comparison of stress-strain plots obtained from different FE model sizes (L = 9, 36 and 90 mm) as well as the solution with PBCs. The Timoshenko analytical solution is super-imposed to show the theoretical effective Young's modulus in the insert, (Betts et al., 2012)

3.5.1 Variance between Euler-Bernoulli and Timoshenko beam theory

Using Euler-Bernoulli beam theory, the deflection of a regular hexagonal honeycomb cell wall is given as follows (Ashby and Gibson, 1988):

\[ |\delta| = \frac{Pl^3 \sin\theta}{12E_s I} \]  \hspace{1cm} (3.20)

Comparing this to Eqn. (3.15), it is apparent that Euler-Bernoulli beam theory under-predicts deflections. The magnitude of the variation can be expressed by the ratio (for a square cross-section):

\[ \frac{|\delta|_{\text{Euler}} - |\delta|_{\text{Timoshenko}}}{|\delta|_{\text{Timoshenko}}} = \frac{E_s t^2}{2kG_s l^2 + E_s t^2} \]  \hspace{1cm} (3.21)
For the FE model of Section 3.4, and varying the beam thickness, this ratio exceeds 10% for values of $\frac{t}{l} > 0.27$. Beyond this value, Euler-Bernoulli beam theory inadequately approximates the deflection of the honeycomb.

Similarly, using Euler-Bernoulli beam theory, the Young’s modulus of a regular hexagonal honeycomb cell wall with square cross-section is given as follows (Ashby and Gibson, 1988):

$$E_2 = \frac{2.3}{\left(\frac{l^3}{E_s t^3}\right)} \quad (3.22)$$

Comparing this to Eqn. (3.19), it is apparent that Euler-Bernoulli beam theory over-predicts the Young’s modulus. The magnitude of the variation can be expressed by the ratio (for a square cross-section):

$$\frac{E_2^{\text{Euler}} - E_2^{\text{Timoshenko}}}{E_2^{\text{Timoshenko}}} = \frac{E_s}{2k_G s} \left(\frac{t}{l}\right)^2 \quad (3.23)$$

For the FE model of Section 3.4 this ratio exceeds 10% for values of $\frac{t}{l} > 0.25$ and Euler-Bernoulli beam theory then inadequately approximates the Young’s modulus of the honeycomb.

The above findings are of relevance to industrial applications of commercial open-cell metal foams. Indeed, XMT scans have been conducted in this work on individual struts of a commercial open-cell metal foam acquired from BPE International, Germany (a metal matrix composite (MMC) fabricated from an Al-Zn-Mg-Cu (7xxx series) alloy with TiC particles) – see Section 4.2.4 for full details. From the 3D render of each scanned strut, and using the imaging software ImageJ (U.S. NIH, Bethesda, Maryland), it has been determined that the average strut length and diameter are 1.7 mm and 0.562 mm respectively (i.e. $\frac{t}{l} = 0.33 > 0.25$). From Eqn. (3.23) it can be deduced that Euler-Bernoulli beam theory therefore over-predicts the Young’s modulus of an equivalent 2D hexagonal
honeycomb representation of the foam by 17% with respect to Timoshenko beam theory (Betts et al, 2012).

### 3.6 Equilateral triangle and square honeycomb FE model development

Equilateral triangle and square tessellated FE models have been created with the same relative density and strut thickness as the hexagonal FE model of Section 3.2.1. PBCs have been implemented in both models as described in Section 3.4, and the material model is as described in Section 3.2.3. The only variable between the different models is the length of the struts.

The relative density of the equilateral triangle tessellation can be approximated as follows (Ashby and Gibson, 1988):

$$\frac{\rho}{\rho_s} = 2\sqrt{3} \frac{t}{l} \left(1 - \frac{\sqrt{3} t}{2 l}\right)$$  \hspace{1cm} (3.24)

where \(t\) is the cell wall thickness and \(l\) is the strut length (with \(t \ll l\)). To achieve a relative density of 13.4%, as for the hexagonal tessellation, the strut length was set to 4.5 mm.

Fig. 3.8(a) shows the equilateral triangle tessellated model along with the applied boundary conditions. The loading direction was chosen to be that in which the stiffness of the cells was greatest – for equilateral triangles, the maximum stiffness is in the \(E_2\) direction.

The relative density of the square tessellation can be approximated as follows (Ashby and Gibson, 1988):

$$\frac{\rho}{\rho_s} = 2 \frac{t}{l} \left(1 - \frac{t}{2 l}\right)$$  \hspace{1cm} (3.25)

To achieve a relative density of 13.4%, as for the hexagonal tessellation, the strut length was set to 2.6 mm.
Fig. 3.8(b) shows the square tessellated model along with the applied boundary conditions. The loading direction was chosen to be that in which the stiffness of the cells was greatest – for squares, the maximum stiffness is in the $E_1$ and $E_2$ directions.

![Fig. 3.8 FE model of (a) equilateral triangle, and (b) square honeycomb with PBCs enclosed by metal facesheets](image)

### 3.7 Equilateral triangle and square honeycomb FE results

Fig. 3.9 shows the stress-strain curves for the regular equilateral triangle and square tessellations, for a relative density of 13.4 % and a strut thickness of 0.18 mm. The results for a regular hexagonal honeycomb tessellation are also included. The graphs display a trend associated with elastic-plastic honeycombs (Ashby and Gibson, 1988). There are three distinct regions: a linear-elastic regime manifested by bending of the cell walls, followed by a plateau of roughly constant stress once a critical stress is reached at which the cells begin to collapse, and finally a regime of steeply rising stress where the cells collapse to such an extent that opposing cell walls touch one another and further deformation compresses the cell wall material itself. This behaviour is also typically observed in commercial open-cell foams (Ashby et al., 2000).
A second peak is observed for the square honeycomb after an initial plateau region at a strain of approximately 64%. This is due to additional beams yielding and the subsequent plastic collapse of additional struts.

Fig. 3.9 Stress-strain curves determined using FE models with different tessellations. The stress-strain relationships can be divided into three regions as shown: (I) Elastic region (see insert); (II) Plastic collapse followed by plateau; (III) Densification.
3.8 Comparative study of optimal cell shapes for a given application

Table 3.1 records the effect of honeycomb cell shape on the measured Young’s modulus, peak stress, plateau stress, modulus of resilience, and plastic strain energy density (measured up to the densification region). The modulus of resilience and plastic strain energy density were directly determined from the area under the stress-strain plots of Fig. 3.9. The area under each curve was found using the trapezium rule, viz.:

$$\text{Area} = 0.5h[(y_0 + y_n) + 2(y_1 + y_2 + \cdots + y_{n-1})]$$  \hspace{1cm} (3.26)

where \( h \) is the constant difference between adjacent strain steps, and \( y_0, y_1, \) etc. the corresponding value of stress at each strain step.

The data in Table 3.1 shows that the square and triangular honeycombs are less good for energy absorbing applications which require a long, flat plateau of elevated stress. The plateau stress is lower for both the square and triangular honeycombs compared to the hexagonal one (for the same relative density), as is the plastic strain energy density.

It is observed that the Young’s modulus of the square and triangular honeycombs is much greater than that of the hexagonal one (for the same relative density), while the initial collapse stress is similar. This makes them the best choice for lightweight structural applications.
Table 3.1 FE results for the three regular 2D tessellations. The effect of honeycomb cell shape on Young’s modulus, peak stress, plateau stress, modulus of resilience, and plastic strain energy density is recorded.

<table>
<thead>
<tr>
<th></th>
<th>Hexagon</th>
<th>Triangle</th>
<th>Square</th>
</tr>
</thead>
<tbody>
<tr>
<td>Young’s modulus (MPa)</td>
<td>268</td>
<td>333</td>
<td>493</td>
</tr>
<tr>
<td>Peak stress (kPa)</td>
<td>998</td>
<td>947</td>
<td>1008</td>
</tr>
<tr>
<td>Plateau stress (kPa)</td>
<td>285</td>
<td>185</td>
<td>101</td>
</tr>
<tr>
<td>Modulus of resilience (kJm³)</td>
<td>2.6</td>
<td>1.4</td>
<td>1.1</td>
</tr>
<tr>
<td>Plastic strain energy density (kJm³)</td>
<td>192</td>
<td>123</td>
<td>68</td>
</tr>
</tbody>
</table>
Chapter 4.

Characterisation and in-situ microtensile testing of open-cell metal foam struts
4.1 Introduction

Analytical methods to determine the basic properties of metal foams (e.g. the moduli and collapse stresses) have been extensively studied; these have been reviewed in Section 2.3. For example, for isotropic, open-cell foams Ashby and Gibson (1988) considered a cubic model consisting of solid struts surrounding a void space and connected at joints. As noted previously, cellular solids are characterised by their relative density, which is given by (with $t << l$):

$$\frac{\rho}{\rho_s} \alpha \left(\frac{t}{l}\right)^2$$

(4.1)

The foam properties are then a function of the relative density. For example, the plastic collapse strength of a foam is given by (Ashby and Gibson, 1988):

$$\frac{\bar{\sigma}_{pl}}{\sigma_{ys}} \left(\frac{\rho}{\rho_s}\right)^{3/2}$$

(4.2)

where $\sigma_{ys}$ is the strut yield strength.

It is therefore apparent that the foam strength is also strongly dependant on the individual cell strut properties. Typically, the material properties of the bulk alloy from which the foam is made are used to predict the foam properties. However, due to the foaming process (Ashby et al., 2000; Banhart, 2001; Banhart and Baumeister, 1998; Baumeister, 2001; Frei et al., 2000; Gergely and Clyne, 2000; Koerner et al., 2006; Leitmeier and Flankl, 2001) and length scale of the struts, there can be notable differences between the mechanical properties of the bulk alloy and the individual struts due to differences in both composition and microstructure.

Few studies have been previously done to assess the mechanical properties of individual metal foam struts, especially with regards to their direct measurement, e.g. Markaki and Clyne (2001), Simone and Gibson (1998), Zhou et
al. (2002), and Zhou et al. (2005). This Section aims to further the direct measurement of metal foam strut mechanical properties by developing a novel microtensile testing technique to measure the tensile properties of metal foam struts. XMT is employed as a means to accurately measure the strut cross-sections prior to deformation, so as to enable the experimental force readings to be converted to stress.

4.2 Characterisation of metal foam samples

4.2.1 Foam morphology and manufacturing process

The metal foam tested in this work was acquired from BPE International, Germany. The bulk foam has an open-cell structure, and is a metal matrix composite (MMC) fabricated from an Al-Zn-Mg-Cu (7xxx series) alloy with TiC particles. Fig. 4.1(a) and (b) show the foam morphology.

(a)  
(b)

Fig. 4.1 Morphology of open-cell Al-Zn-Mg-Cu (7xxx series) alloy foam with TiC particles. (a) Photograph of bulk foam, and (b) micrograph taken in SEM

The bulk foam is manufactured using an investment casting technique, similar to that used for the ERG Duocel range of foams (http://www.ergaerospace.com). An open-cell polymer foam with the required
relative density and cell size is used as a template to create an investment casting mould. The polymer mould is then coated with a mould casting (ceramic powder) slurry which is subsequently dried and embedded in casting sand. Next, the mould is baked, which causes the casting material to harden and decomposes the polymer template. A negative image of the polymer foam is thus produced. The mould is then filled with the aluminium MMC and cooled. After directional solidification and cooling, the mould materials are removed and a metal equivalent of the initial polymer foam is produced (Ashby et al., 2000). Fig. 4.2 illustrates the approach.

![Diagram](image)

**Fig. 4.2 Investment casting technique used to manufacture open-cell metal foams, (Ashby et al., 2000)**
4.2.2 Chemical composition

The chemical composition of the metal foam was determined using energy-dispersive X-ray spectroscopy (EDX). For this purpose, a Hitachi S3400N SEM fitted with an Oxford Instruments INCA system was used. Two metal foam samples were mounted into separate conductive Bakelite blocks and then progressively ground to finer levels using silicon carbide grinding paper as follows: first, 800 grit paper was used, followed by 1200 grit paper, then 2400 grit paper, and finally 4000 grit paper. A 3 μm polish was then performed on the samples, before a final polish was carried out using colloidal silica suspension.

Each sample was investigated in turn using EDX, with four readings taken for each. The chemical composition of the Al-Zn-Mg-Cu alloy with TiC particles is shown in Table 4.1, taken as an average of the eight readings (Betts et al., 2013).

Table 4.1 Chemical composition of Al-Zn-Mg-Cu alloy with TiC particles metal foam strut (in weight percent). Average based on 8 readings from 2 separate samples, (Betts at al., 2013)

<table>
<thead>
<tr>
<th>Element</th>
<th>Mg</th>
<th>Al</th>
<th>Si</th>
<th>Ti</th>
<th>Cu</th>
<th>Zn</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average</td>
<td>1.09</td>
<td>Balance</td>
<td>0.68</td>
<td>6.49</td>
<td>1.46</td>
<td>4.18</td>
</tr>
</tbody>
</table>

4.2.3 Ageing process

Al-Zn-Mg-Cu 7xxx series aluminium alloys demonstrate a particularly high response to precipitation hardening, and would generally be used in applications in an optimally age-hardened condition. The optimal ageing treatment for this foam is needed so that samples can be aged before the microtensile testing. Ashby and Jones (2013) and Polmear (1995) provide extensive background theory on the ageing response of aluminium alloys.

To achieve peak strength, the foam struts were solution heat treated at 480 °C for 2 hours, prior to water quenching to achieve a supersaturated solid solution.
Then, they were aged by heating to 120 °C for 4 hours to allow finely dispersed precipitates to form, after which they were left to cool slowly at room temperature. This heat treatment process is illustrated schematically in the temperature-time graph of Fig. 4.3.

Fig. 4.3 Schematic diagram of the T6-strengthening process

The optimal ageing time of 4 hours was determined by conducting microtensile tests on struts aged for 0, 2, 3, 4, 5, 6, 7 and 14 hours respectively using the test procedure outlined in Section 4.3.1, from which the ultimate tensile strength (UTS) could be ascertained. Three repeats were conducted for each ageing time. Fig. 4.4 plots the ultimate tensile strength normalised against that for the as-cast condition vs. ageing time – it is apparent that the peak strength occurs at an ageing time of 4 hours.

It should be noted that the T6 ageing condition does not significantly impair the ductility of the struts; i.e. strengthening is not associated with a notable reduction in ductility (Betts et al., 2013).
Fig. 4.4 UTS normalised against that for the as-cast condition vs. ageing time for the Al-Zn-Mg-Cu (7xxx series) alloy foam with TiC particles

4.2.4 XMT of open-cell metal foam struts

X-ray micro-computed tomography (XMT, Phoenix X-ray Systems and Services GmbH, Wunstorf, Germany) scanning was used to obtain full 3D renders of individual struts extracted using electrical discharge machining (EDM) from the bulk foam, following methodologies previously applied to whole structure analysis (Singh et al., 2010b; Zhang et al., 2009). Although various experimental set-ups can be used to perform XMT, the basic principles of the technique remain the same and a detailed description of the process can be found in both Buffière et al. (2008) and Buffière et al. (2010). A comprehensive theoretical description of the technique can be found in the classical book of Kak and Slaney (1988).

The 3D renders were then imported into the analysis software Avizo 5 (Visage Imaging, GmbH, Berlin), where cross-sectional slices of the struts could be exported at specific increments (between 15.79 µm and 18.25 µm, depending on
The typical grain structure of an individual strut is shown in Fig. 4.6, as observed using the Hitachi S3400N SEM. The bulk foam samples were mounted into conductive Bakelite blocks and then ground and polished as described in Section 4.2.2. The samples were then etched with Keller’s reagent in order to display the grain boundaries. The average grain has an area of 0.03 mm$^2$, determined using the imaging software ImageJ (U.S. NIH, Bethesda, Maryland) and based on an average taken from ten cross-sections.
4.3 Experimental procedures

4.3.1 Microtensile test set-up

The Gatan Microtest 300 rig was used for microtensile testing of the metal foam struts in the SEM. The rig is a screw-driven, dynamic testing stage module which can be operated either within or outside the SEM. The rig is fitted with a 300 N load-cell that has an accuracy ± 1 % of full-scale, and a resolution 0.1 % of full-scale. The rig can operate in tension or compression, and can be fitted with horizontal 3-pt or 4-pt bending options. The rig has a stroke of 10 mm, and the minimum separation between the standard grips is also 10 mm; as the struts are in the order of 1 mm to 2 mm in length, an elongated flat-surface steel grip was custom made and attached to the rig using four screws – Fig. 4.8 shows the design of this grip. Fig. 4.7 shows photographs of the microtensile test set-up (Betts et al., 2013).

The struts were clamped with parallel, flat-surface steel grips. The tests were conducted at a speed of 0.5 mm/min, with a sampling time of 500 ms. The Gatan Microtest 300 rig has a speed range of 0.1 mm/min to 1.5 mm/min, which
corresponds to a strain rate range of $1.7 \times 10^{-3} \text{ s}^{-1}$ to $2.5 \times 10^{-2} \text{ s}^{-1}$ for the tested struts in this work. In this range, aluminium alloys have little sensitivity to the strain rate at room temperature (Davis, 2004; Hatch, 1984).

The Gatan Microtest software outputted force vs. displacement plots, and a video of each test was recorded (the Gatan Microtest 300 rig comes with comprehensive software that has a MTVideo option, allowing synchronized movies and data of the tests to be captured).

![Image of test setup](image)

**Fig. 4.7** Photographs of the microtensile test set-up showing Gatan Microtest 300 rig and SEM, (Betts et al., 2013)
4.3.2 Determination of stress and strain

In order to convert the force readings from the Gatan Microtest software into engineering stress, the undeformed cross-sectional area of the struts is required. Furthermore, a detailed measure of the tested strut geometry is required for accurate calibration of the viscoplastic damage constitutive equations in Chapter 5. Hence, each strut was scanned using XMT prior to testing as outlined in Section 4.2.4 so as to obtain full 3D renders of the samples – see Fig. 4.5(a). As noted previously, the analysis software Avizo was then used to export cross-sectional slices of the struts at specific increments.

Twenty cross-sectional slices were taken along the central region of each strut and their area was measured using the imaging software ImageJ. The force readings were then converted to engineering stress by dividing each value by the average area of the twenty cross-sectional slices.
Engineering strain was determined from the crosshead displacement of the Gatan Microtest 300 rig. The original, undeformed length of the strut was determined by measuring the distance between the grips prior to any load being applied. This was achieved by taking a picture of the rig in the SEM and using ImageJ to determine the distance (Betts et al., 2013).

It is worth noting that machine compliance is not an issue in these tests due to the very small cross-sectional area of the specimens (~0.25 mm$^2$ as shown in Fig. 4.5(b) and Fig. 4.10). The total cross-sectional area of the grips is ~300 mm$^2$ (see Fig. 4.8); i.e. a factor of 1200 difference in area with respect to the tested struts. Simplistically, the test set-up can be viewed as three elastic cylinders bonded together in series. The cylinders at each end (representing the machine/grips) have a much larger cross-sectional area than the central cylinder (representing the sample); the stress in the central cylinder will therefore be much greater for a given load than that in the grips. Furthermore, the grips are made of steel and hence have a greater Young’s modulus than the aluminium sample (approximately by a factor of three). By combination of these two factors, the strain in the sample will be much greater than that in the grips (approximately by a factor of 3600), and the strain measure will be negligibly affected by machine compliance.

4.4 Microtensile test results and analysis

4.4.1 Microtensile properties of the struts

Ten metal foam struts were heat treated according to the T6 condition outlined in Section 4.2.3. The struts were then scanned using XMT and microtensile tested. Several specimens fractured at the grips and were therefore discounted from the analysis. Fig. 4.9 gives the experimental microtensile test stress-strain graphs for
the struts that did not fracture at the grips (Betts et al., 2013). Fig. 4.10 shows the measured variation of strut cross-sectional area along the length of these struts.

Fig. 4.9 Experimental microtensile test stress-strain graphs for the metal foam struts aged at 120 °C for 4 hours. Four repeats were carried out under the same conditions, (Betts et al., 2013)

Fig. 4.10 Measured variation of strut cross-sectional area along their length
From Fig. 4.9, it can be seen that the gradient of the initial elastic region of the stress-strain plots is much lower than the expectation of roughly 70 GPa for corresponding aluminium alloys. The average gradient measured from the experimental results is 14 GPa. A much lower than expected initial elastic region was also reported in Zhou et al. (2005). The reasons for this disparity are discussed in Chapter 5. The tested struts showed reasonable repeatability in the results, with the UTS varying from 398 MPa to 435 MPa and the failure strain varying from 8.4 % to 9.8 % (Betts et al., 2013).

### 4.4.2 Corrected stress-strain graphs of the struts

From the observed microtensile test results in Fig. 4.9, it is clear that the elastic behaviour of the struts was not correctly measured by the tests. Previous work (Zhou et al., 2005), in which a similarly low elastic slope was measured (~10 GPa for a T6 heat treatment), has suggested this may be due to the initial curvature of the struts which may reduce the effective initial strut stiffness. This proposal is further assessed in Chapter 5, as are strut slippage effects.

In an attempt to improve upon the machine reported values, 2D digital image correlation (DIC) analysis was conducted using GOM software (GOM, mbH, Braunschweig) to analyse the video frames captured during the microtensile tests, based on identifiable features in the surface structure. However, due to the unavoidable changes in contrast between images in the same stack, the strain results from the DIC analysis exhibited a high noise level in the region ± 0.1 %. The strain contour plots from the DIC analysis (see e.g. Fig. 4.11(a)) are therefore not suitable with regards to obtaining an accurate strain distribution across the strut surface. A statistical analysis of the strain field was instead created by considering the strain along three longitudinal sections of the strut, taken at evenly distributed intervals across the strut width. The average strain obtained from the three sections...
for each image in the same stack is denoted $\varepsilon_Y$. Fig. 4.11(b) plots $\varepsilon_Y$ and the corresponding standard deviation for each image in the stack for experiment 3. The onset of yielding occurs at stage 51 in the image sequence, and it can be determined from Fig. 4.11(b) that the average strain at yield is 0.4 ± 0.1 %. Based on a measured yield stress of 196 MPa, this implies a Young’s modulus range of 39 GPa to 65 GPa, which is insufficiently accurate; calculating strain based on the measurement of the distance between two identifiable points on the sample was insufficiently accurate for the same reasons.

Fig. 4.11 DIC analysis for experiment 3. (a) Strain contour plot at onset of yielding. (b) Average longitudinal strain and standard deviation obtained for each strain stage in the image stack.
An unloading slope test was also attempted. The Gatan Microtest 300 rig is not capable of loading and unloading a specimen in one continuous test, so a tensile load was first applied and the test manually stopped at a point soon after yielding. The test rig was then set to compression mode and unloading was applied to the specimen. This produced two load-displacement graphs: one for loading, and one for unloading. The two graphs were combined by subtracting each load and displacement of the unloading curve from the maximum load and displacement of the loading curve. However, this unloading test produced a measured Young’s modulus very similar to that reported in Fig. 4.9, and it is expected that some reverse slippage occurs when the direction of the screw-driven loading is reversed, since the part of the sample within the grips which has slipped will also elastically unload. Fig. 4.12 shows the unloading test stress-strain curve superimposed on those for the microtensile tests of Fig. 4.9.

![Unloading test stress-strain curve superimposed on the microtensile test results of Fig. 4.9](image)

Hence, the following correction is proposed: at the onset of yielding, the struts experience uniaxial tensile testing conditions and slippage is at that point
significantly reduced. The stress-strain plots of Fig. 4.9 have consequently been modified as follows (Betts et al., 2013):

- The elastic region of the plots has been modified so that the Young’s modulus is equal to 70 GPa;
- The displacement due to slippage has been subtracted from the total measured displacement so as to correct the reported strain values.

The validity of this approach is investigated in Chapter 5. Fig. 4.13 shows the corrected stress-strain graphs.

![Stress-strain Graphs](image)

**Fig. 4.13 Corrected stress-strain graphs adjusted to address the observed significant reduction in strut stiffness seen in Fig. 4.9, (Betts et al., 2013)**

### 4.4.3 Strut fracture

Micrographs of the strut fracture surface were captured after testing using the SEM, and are presented in Fig. 4.14(a) – (d), with overall fracture shown in Fig. 4.14(a). Microvoids and dimples can be observed at higher magnifications, indicative of ductile fracture (Fig. 4.14(b) – (d)). TiC particles are also visible at higher magnifications. These particles are on a scale small enough to affect the
ageing time of the material (Fig. 4.14(d)) – BPE International has confirmed that the TiC particles reduce the required ageing time of the material.

Fig. 4.14 (a) Fracture of strut after microtensile testing, observed using SEM. (b)-(d) Fracture surfaces of tested strut at varying levels of magnification showing microvoids, ductile dimples, and TiC particles.
Chapter 5.

3D X-ray tomography based finite element modelling of open-cell metal foam struts
5.1 Introduction

In Chapter 4, a microtensile test procedure was developed to directly determine the mechanical properties of individual metal foam struts. It was found that the measured strut properties show a significant reduction in elastic stiffness compared to the typical value of 70 GPa for aluminium alloys.

The reasons for this observed reduction in stiffness are now investigated via realistic FE modelling of the as-tested struts, using the XMT scans of the undeformed struts presented in Chapter 4 for the strut geometry. A set of continuum mechanics-based viscoplastic damage constitutive equations are used to model the material behaviour of the struts. The equations are calibrated with the microtensile test data of the aluminium alloy’s optimally aged condition presented in Chapter 4 and are implemented into ABAQUS through the user-defined subroutine VUMAT.

The FE simulations are used to determine the effect of strut curvature as well as slippage between the test rig grips and foam struts on the recorded stress-strain plots. The FE simulations are also used to develop a procedure, initially proposed in Chapter 4, that compensates for the effect of grip slippage inherent in the microtensile testing of metal foam struts.

5.2 Unified viscoplastic damage constitutive law and calibration

5.2.1 Background on continuum modelling of ductile damage

Damage is the collective name given to the degrading defects observed when a metal is subjected to continued plastic deformation: discreet microvoids or cracks will nucleate and grow within the material until they eventually coalesce to form macrocracks that lead to material failure (Lin et al., 2005a). Besson (2010)
provides a comprehensive review of both the micromechanics-based and phenomenological material constitutive equations that have been developed to simulate ductile damage and rupture.

Among the first micromechanical models created to describe the development of ductile damage are those by McClintock (1968) and Rice and Tracey (1969). These describe the ductile growth of isolated cylindrical or spherical voids in a rigid, perfectly plastic matrix due to stress triaxiality. For instance, Rice and Tracey (1969) produced the following equation for the rate of variation of the radius of a spherical void with high stress triaxiality:

\[
\frac{\dot{R}_v}{R_v} = \alpha \exp \left( 1.5 \frac{\sigma_m}{\sigma_0} \right) \dot{\varepsilon}_{eq} \tag{5.1}
\]

where \(R_v\) is the void radius, \(\alpha\) is a numerical factor, \(\varepsilon_{eq}\) the von Mises equivalent strain, \(\sigma_0\) the matrix yield stress, and \(\sigma_m\) the hydrostatic stress.

As noted by Besson (2010), the definition of a rupture criterion stating that fracture occurs when the normalised void radius has reached a critical value (Marini et al., 1985) stemmed from the Rice and Tracey (1969) model, and is given by Eqn. (5.2).

\[
\frac{R_v}{R_{v0}} = \left( \frac{R_v}{R_{v0}} \right)_c \tag{5.2}
\]

where \(R_{v0}\) is the initial void radius and \(\left( \frac{R_v}{R_{v0}} \right)_c\) is a material dependent parameter defining the critical value for void growth.

However, the work by Rice and Tracey (1969) does not consider the effect of void growth on softening. The Gurson model (1977) generates a yield function to accommodate for this effect; damage is represented by the void volume fraction (i.e. porosity). The plastic yield surface produced by Gurson (1977) is represented by Eqn. (5.3).

\[
\Phi = \frac{\sigma_e^2}{\sigma_0^2} + 2f \cosh \left( \frac{\sigma_m}{2\sigma_0} \right) - 1 - f^2 \tag{5.3}
\]
where $\Phi$ is the flow potential and $\sigma_{e}$ is the equivalent stress. The void volume fraction, $f$, is equal to the summation of the volume fraction of new nucleating voids, $f_{\text{nucleation}}$, and that of the growth of pre-existing voids, $f_{\text{growth}}$, and its evolution with respect to time is given by Eqn. (5.4).

$$\dot{f} = \dot{f}_{\text{growth}} + \dot{f}_{\text{nucleation}}$$

Tvergaard and Needleman (2001) further developed the Gurson model to improve its ability to predict porosity and damage-related material softening at final failure by modifying Eqn. (5.3) as follows:

$$\Phi = \frac{\sigma_{e}^{2}}{\sigma_{0}^{2}} + 2q_{1}f^{*} \cosh \left(\frac{q_{2}\sigma_{m}}{2\sigma_{0}}\right) - 1 - (q_{1}f^{*})^{2}$$

$$f^{*} = \begin{cases} 
  f & \text{for } f < f_{c} \\
  f_{c} + \left(\frac{1}{q_{1}} - f_{c}\right)(f - f_{c})/(f_{f} - f_{c}) & \text{for } f \geq f_{c}
\end{cases}$$

where the bilinear function $f^{*}$ accounts for the effects of rapid void coalescence at failure. It is assumed that when a critical porosity, $f_{c}$, is reached, damage increases at a greater rate due to coalescence; fracture occurs when $f = f_{f}$ and $f^{*} = 1/q_{1}$. $q_{1}$ and $q_{2}$ account for different-shaped voids.

A more phenomenological approach to damage modelling, based on macroscopic considerations, was first developed by Kachanov (1958). Kachanov (1958) introduced a continuous variable related to the density of microcracks and voids in a material, with its evolution described by constitutive equations written in terms of stress or strain. The constitutive equations can be used to predict the initiation of macrocracks when used in structural calculations. This approach is labelled continuum damage mechanics (CDM). Extensive treatments of CDM are presented in books by Lemaitre and Chaboche (1990) and Rabotnov (1969).

For example, Lemaitre’s work (1984) uses a scalar damage variable $\omega$ that is based on the area fraction of voids in a given cross-section normal to the applied stress:
\[ \omega = \frac{S_D}{S_T} \]  

(5.7)

where \( S_D \) is the area of voids along the cross-section and \( S_T \) the total cross-sectional area. As \( 0 \leq S_D \leq S_T \), \( 0 \leq \omega \leq 1 \). By considering an effective stress tensor, \( \tilde{\sigma} \), which corresponds to the stress acting on a fictitious undamaged volume of smaller cross-sectional area, given by Eqn. (5.8), Lemaitre (1984) produces the constitutive Eqn. (5.9) that relates directly to isotropic ductile damage.

\[ \tilde{\sigma} = \frac{\sigma}{(1 - \omega)} \]  

(5.8)

\[ \dot{\omega} = \left[ \frac{K^2}{2E\sigma_0^3} \left( \frac{2}{3}(1 + v) + 3(1 - 2v) \left( \frac{\sigma_m}{\sigma_e} \right)^2 \right) \right] (\dot{\varepsilon}_{eq})^{2/m} \dot{\varepsilon}_{eq} \]  

(5.9)

where \( K \) is the drag stress, \( E \) the Young’s modulus, \( \dot{\varepsilon}_{eq} \) the equivalent plastic stain, \( S_0 \) is a material and temperature dependent coefficient, and \( m \) is the hardening exponent.

Finally, Lin et al. (2005a) reviewed a series of constitutive equations for viscoplasticity that have been previously developed to capture the effects of various time-dependant phenomena such as dislocation-associated hardening and damage. It is assumed that once the values of specified damage variables reach certain levels, the material can no longer sustain the applied load and failure takes place.

**5.2.2 Multi-axial constitutive equation set**

The following CDM multi-axial constitutive equation set has been previously developed to model the ductile behaviour, including damage softening, of a wide range of metals (see e.g. Lin and Dean, 2005; Lin et al., 2005a; Wang et al., 2011) and is proposed to describe the damage behaviour of the metal foam struts studied in this work (Betts et al., 2013). The equations enable a range of time-dependent phenomena, such as dislocation-associated hardening and damage, to be modelled.
An evolving damage parameter based on micromechanisms, \( \omega \), is established by the equation set. This feature is employed in Chapter 6 to predict the level of damage in the struts of an aluminium metal foam core sandwich panel subjected to low energy impacts, enabling the extent of subsurface damage to be ascertained.

Eqn. (5.11) represents the development of normalised dislocation density due to plastic strain and the dynamic recovery of the dislocation density, as presented in Lin et al. (2005b). Static recovery is neglected as the loading scenarios are all at room temperature. Eqn. (5.12) describes isotropic hardening, defined as the increase in the static yield surface beyond the initial yield point, and stems from observations that the mean slip length is governed by the inverse of the square root of the dislocation density (Nes, 1998). Eqn. (5.13) describes the evolution of plastic strain controlled damage nucleation and growth, and determines material failure (Lemaitre and Chaboche, 1990; Lin et al., 2005a).

\[
\dot{\varepsilon}_e^p = \left( \frac{\sigma_e - k_s - R}{K} \right)^n
\]  

\[
\ddot{\rho} = A_1 (1 - \bar{\rho}) \dot{\varepsilon}_e^p
\]  

\[
R = B \bar{\rho} \dot{m}
\]  

\[
\dot{\omega} = \frac{\alpha C \varepsilon_e^p}{(1 - \omega)^{n_0}}
\]  

\[
\sigma_{ij} = 2G\varepsilon_{ij}^e + \lambda \delta_{ij} \varepsilon_{kk}^e
\]  

\[
\varepsilon_{ij}^e = \varepsilon_{ij}^l - \varepsilon_{ij}^p
\]  

\[
\bar{\sigma}_{ij} = \frac{\sigma_{ij}}{(1 - \omega)}
\]  

\[
\dot{\varepsilon}_{kl}^p = \frac{3}{2} s_{kl} \dot{\varepsilon}_e^p
\]  

\[
\ddot{\rho} = \frac{\rho - \rho_i}{\rho_{max} - \rho_i}
\]
where $\dot{\varepsilon}_e^p$ is the effective viscoplastic strain rate; $\dot{\varepsilon}_e^b$ is given by Eqn. (5.10) when $\bar{\sigma}_e - k_1 - R \geq 0$ and $\dot{\varepsilon}_e^p = 0$ when $\bar{\sigma}_e - k_1 - R < 0$. $\dot{\rho}$ is the rate of normalised dislocation density. Normalised dislocation density, $\bar{\rho}$, varies from 0 to 1; $\rho_i$ is the dislocation density for the virgin material (the initial state) and $\rho_{\text{max}}$ the maximum (saturated) dislocation density that the material can have. $R$ describes the hardening caused by dislocations within the material. $\omega$ is damage and varies from 0 (no damage) to 1 (full damage); $\alpha = 1$ when $\sigma_{kk} \geq 0$ or $\alpha = 0$ when $\sigma_{kk} < 0$, which ensures damage is only included in the constitutive equation set when loading produces a state of hydrostatic tension. $\varepsilon_{ij}^e$, given by Eqn. (5.15), is the elastic strain, $\varepsilon_{ij}^b$ is the total strain and $\varepsilon_{ij}^p$ is the plastic strain. $k_1$ is the yield stress and $K$ is the drag stress. $G$ and $\lambda$ are the Lamé parameters (where $G$ is the shear modulus).

The von Mises stress is given by $\sigma_e = \sqrt{\frac{3}{2} s_{ij} s_{ij}}$, and the deviatoric stress is given by $s_{ij} = \sigma_{ij} - \frac{1}{3} \sigma_{kk} \delta_{ij}$. Tildes denote effective tensors in terms of damage, such as in Eqn. (5.16). $\delta_{ij}$ is the Kronecker delta, equal to 1 when $i = j$ and equal to 0 when $i \neq j$ (all indices – i, j, k, l – follow the usual tensor conventions). $n$, $A_1$, $B$, $m$, $C$, and $n_0$ are material constants.

### 5.2.3 Calibration of multi-axial constitutive equation set

The material constants $E$, $v$, $k_1$, $K$, $m$, $n$, $A_1$, $B$, $C$, and $n_0$ in the multi-axial constitutive equation set outlined in Section 5.2.2 have been determined by calibrating the damage model using the corrected microtensile test results presented in Section 4.4.2. The group of non-linear ordinary differential equations outlined in Section 5.2.2 can be solved using numerical integration techniques such as the forward Euler method.

From a physical understanding of each material constant, a known value or a bounding range of values was initially assigned to each unknown; $E$, $v$, $k$, $K$ and
were set to 70 GPa, 0.33, 185 MPa, 100 MPa and 2, respectively, and n, A₁, B, C and n₀ to the ranges 1 to 8, 0.1 to 5, 200 MPa to 2000 MPa, 0.2 to 2 and 4 to 10, respectively. An optimisation technique for determining the final values of the material constants was then used, based on minimising the sum of the squares of the errors between the computed and corrected microtensile experimental data; details of the optimisation scheme used are given in Li et al. (2002) and Lin and Yang (1999). Table 5.1 gives the set of parameters determined by the optimisation process for the tested struts.

**Table 5.1 Constants used in the multi-axial constitutive equation set for the T6 heat treated metal foam struts, (Betts et al., 2013)**

<table>
<thead>
<tr>
<th>E (GPa)</th>
<th>ν</th>
<th>k₁ (MPa)</th>
<th>K (MPa)</th>
<th>n</th>
<th>A₁</th>
<th>B (MPa)</th>
<th>C</th>
<th>m</th>
<th>n₀</th>
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<tbody>
<tr>
<td>70</td>
<td>0.33</td>
<td>185</td>
<td>100</td>
<td>1.4</td>
<td>0.55</td>
<td>1450</td>
<td>0.8</td>
<td>2</td>
<td>6</td>
</tr>
</tbody>
</table>

Fig. 5.1 shows the calibrated damaged model superimposed on the corrected microtensile test stress-strain plots (Betts et al., 2013). It can be seen that there is evident damage softening in some of the experimental curves; the micrographs of the strut fracture surfaces presented in Section 4.4.3 provide further evidence that the struts fail in a ductile manner with some softening before failure.
5.3 FE Modelling of as-tested struts

5.3.1 Overview of FE model

FE modelling of the as-tested metal foam struts was conducted using the software ABAQUS 6.11. The models consist of a 3D meshed geometry of identical dimensions to the as-tested struts. This has been achieved by importing the XMT scans outlined in Section 4.3.2 into the analysis software Avizo, where the geometry was meshed with a 4-node tetrahedral grid; Avizo uses a marching cubes algorithm whereby a smooth surface domain of discretised triangles is first generated before the solid volume with tetrahedral elements is created (Wang et al., 2005).

The mesh was then imported into ABAQUS with element type C3D4 selected. A comparison of Fig. 5.2(a) and (b) demonstrates that the part geometry in ABAQUS is identical to that of the same as-tested strut. To achieve mesh convergence, 3386 elements were required (Betts et al., 2013).
5.3.2 Applied boundary conditions, loads, and material model

The applied boundary conditions are shown in Fig. 5.2(b). The bottom face of the strut was constrained in all degrees of freedom. A uniform tensile load was modelled by applying a MPC along the top face of the strut, whereby all nodes along that face were tied to the central node. Tensile load vs. displacement plots were obtained by moving this central node in the vertical direction under a controlled, linear displacement.

The viscoplastic damage constitutive equation set presented in Section 5.2 was implemented into the ABAQUS model through the user-defined subroutine VUMAT.

5.3.3 FE results for as-tested strut model

Fig. 5.2(c) shows a comparison of the stress-strain graphs for the calibrated damage model of Section 5.2 and the FE model of one of the as-tested struts. There is good agreement between the two curves. This indicates not only that the applied damage model captures the behaviour of the material, that it has been successfully implemented into ABAQUS and the calibration is valid, but more markedly that any strut curvature present had a negligible effect on the experimentally measured strut stiffness for the tested metal foam; otherwise, the FE predicted initial elastic slope would be less than that of the corrected experimental curve (equal to 70 GPa).

In other words, any effect of the struts straightening from an initially curved shape by bending before uniaxial tension has minimal impact on the force vs. displacement readings. The large reduction in the gradient of the initial elastic region of the experimental stress-strain plots in relation to that for corresponding aluminium alloys is therefore primarily due to some other factor. Consequently, the
effect of slippage between the tested struts and the grips is the proposed cause, as considered in Section 5.4; it is further proposed that slippage was also the cause of the lower initial elastic slope reported in Zhou et al. (2005), rather than strut curvature (Betts et al., 2013).

(a) Metal foam strut during microtensile testing, observed using SEM. (b) FE model of same as-tested metal foam strut with calibrated damage model set as material definition. (c) Comparison of stress-strain graphs for the calibrated damage model and the FE model of the as-tested strut, (Betts et al., 2013)
5.4 FE analysis of slippage effects

5.4.1 Overview of FE model

To investigate the reasons for the observed reduction in stiffness for the experimental microtensile stress-strain plots of the metal foam struts (see Section 4.4), an FE model has been created to capture the effect of slippage between the Gatan Microtest 300 grips and the struts. Fig. 5.3 shows the FE model of the microtensile test with grips. The strut geometry was constructed as described in Section 5.3. The grips were modelled as rigid bodies, and were each assigned a reference point for application of the boundary conditions and loads (Betts et al., 2013).

5.4.2 Applied boundary conditions, loads, and material model

The applied boundary conditions are shown in Fig. 5.3. For the first dynamic, explicit time step of the analysis, the grips were brought together by applying a vertical displacement to a reference point on each of the grips. All other degrees of freedom were constrained.

In the second step, the left pair of grips was constrained in all degrees of freedom and a horizontal displacement was applied to the right pair of grips with all other degrees of freedom constrained. This was in order to replicate the microtensile test conditions. The material model is as used in Section 5.3.2.

To model the effect of slippage between the grips and the strut, the tangential surface interaction between the grips and strut was assigned a penalty friction formulation, with a constant coefficient of friction, $\mu_f$, equal to 0.25. This value was selected through trial and error so as to achieve the best fit with the experimental data, and lies within the typical range for aluminium/steel surface interactions – see e.g. Chaplin and Chilson (1986) (Betts et al., 2013).
5.4.3 FE results for slippage effects

Fig. 5.3 shows a comparison, for one tested strut, of the stress-strain graphs for the FE model with slippage for three different values of the coefficient of friction, $\mu_f = 0.15, 0.25$ and $0.40$, and the experimental microtensile test result of the same strut. The best agreement is achieved in the observed stiffness between the FE model with slippage and the experimental result when $\mu_f = 0.25$. Fig. 5.3 indicates that slippage between the strut and grips is the principal cause for the measured reduction in strut stiffness during microtensile testing. As expected, a lower value of $\mu_f$ reduces the observed stiffness in the FE model with slippage, and a higher value of $\mu_f$ increases it.

Fig. 5.4(b) compares the strain field along the same longitudinal section between the FE model with slip (with $\mu_f = 0.15, 0.25$ and $0.40$) and the FE model with no slip for a grip displacement of $0.002$ mm (i.e. with loading still in the
elastic region, and corresponding to an average strain of 0.2 % for the no slip condition). It can be seen from Fig. 5.4(b) that the effective gauge length (defined as the region where appreciable strain occurs) is greater for the FE model with slip than that with no slip. This furthers the explanation as to why the observed strut stiffness during microtensile testing is lower than expected; i.e. strain based on the apparent gauge length in a scenario where slippage occurs is too large, which contributes to a lower than expected elastic slope. In addition, increasing the coefficient of friction in the model with slip reduces the effective gauge length towards that of the no slip case, and the measured strains increase and tend towards that for the case with no slip, as observed in Fig. 5.5 (Betts et al., 2013).

Fig. 5.4 (a) Path for strain measurements (horizontal black lines) shown for both the FE model with no slip and the FE model with slip. (b) Comparison of strain field along the same longitudinal section between the FE model with slip, with $\mu_f = 0.15$, $\mu_f = 0.25$, and $\mu_f = 0.40$, and the FE model with no slip for a grip displacement of 0.002 mm, (Betts et al., 2013)
Fig. 5.5 Trend of FE model with slip with increasing $\mu_t$ towards the FE model with no slip for average strain across the gauge length and effective gauge length, (Betts et al., 2013)
Chapter 6.

3D finite element modelling of open-cell metal foam core sandwich panels
6.1 Introduction

In this Chapter, a 3D FE model of an open-cell aluminium alloy metal foam core sandwich panel subjected to uniform compression is first created to study the effect of varying the foam strut aspect ratio (length to thickness) on the elastic properties of the core; the FE model incorporates the continuum mechanics-based viscoplastic damage model previously calibrated in Section 5.2. In addition, FE models of the open-cell metal foam core sandwich panel subjected to three point and four point bending are produced in accordance with ASTM C-393-00, hence providing a virtual standardised test to assess the foam core elastic properties. The bending models use a tabulated elastic-plastic material model and their results are compared to those from the uniform compression model for validation purposes.

Furthermore, as noted in Chapter 1, metal foams show promise as the cores of lightweight sandwich structures for use in the design of aircraft wing boxes, which are at present typically fabricated utilising thin panels that comprise of a skin stiffened by stringers. One of the key drivers for the use of a sandwich panel as a primary wing structure includes its capability to withstand low energy impacts, such as an accidental tool strike under ground repair conditions. Therefore, the damage model calibrated in Section 5.2 is used in indentation loading scenario FE models to investigate the extent of structural damage in metal foam core sandwich panels used as an airplane wing skin material when subjected to tool drop impacts. The FE simulations are also used to identify an optimal foam strut aspect ratio that provides the greatest energy absorption per unit mass whilst ensuring core damage is accurately reflected by facesheet deformation, which is necessary for detection and repair. The effect of varying the indenter radius on the extent of visible structural damage is also considered so as to capture the influence of varying angles of tool drop impact.
As observed in Chapter 2, the mechanical properties of a foam are primarily dictated by its composition, relative density, and structure (open-cell vs. closed-cell). Moreover, most commercially available foams contain processing induced morphological defects that further influence their mechanical properties. These include fractured cell walls and cells of exceptional size (formed by the amalgamation of adjacent missing cells) (Chen et al., 1999). However, only several attempts have been made to account for the effects of fractured cell walls and missing cells on the mechanical properties of cellular solids (e.g. Chen et al., 1999; Guo et al., 1999; Silva and Gibson, 1997), and these have been limited to 2D models. This Chapter aims to further this work by investigating the effect of fractured cell walls and missing cells on the mechanical properties of metal foam cores by using the aforementioned 3D uniform compression FE model.

Finally, the effect of fractured cell walls and missing cells on the extent of visible structural damage of the foam core is investigated for indentation loading scenarios.

6.2 3D uniform compression FE model

6.2.1 Overview of FE model

Commercial open-cell foams are typically available in a range of relative densities. For example, the aluminium alloy foam Duocel, manufactured by ERG using a directional solidification route, has a relative density varying from 0.05 to 0.1 (Ashby et al., 2000). As noted in Section 4.1, the properties of metal foams depend most directly on those of the material from which they are made and their relative density. The relative density of the foam is in turn dependent on the aspect ratio (length to thickness) of the individual foam struts. It is therefore important to be able to accurately assess the effect of varying the strut aspect ratio on the
mechanical properties of the foam, so as to be able to select the most appropriate ratio for a given application of desired strength and weight.

To this purpose, a 3D uniform compression FE model has been created to assess the effects of varying the strut aspect ratio on the elastic properties of an open-cell foam – see Fig. 6.1(b). The metal foam core has been modelled using the 3D Kelvin structure, which consists of a regular network of space-filling idealised tetrakaidecahedrons. This unit cell was chosen as it is the lowest energy unit cell (i.e. that consisting of the least surface area between the cells for a given cell volume) known consisting of a single polyhedron (Zhu et al., 2000).

Fig. 6.2 presents a micrograph (taken in the SEM) showing the typical morphology of the open-cell metal foam studied in this work (a MMC Al-Zn-Mg-Cu (7xxx series) alloy with TiC particles) and characterised in Section 4.2. The shape of the cells indicates the suitability of using the idealised Kelvin cell model. Studies investigating Kelvin cell foams have previously been conducted, for example, in Gong et al. (2005) and Jang and Kyriakides (2009). Fig. 6.1(a) shows a unit cell. The cell is a 14-sided polyhedron that consists of 6 squares and 8 hexagons with all edges being of the same length; as it is an open-cell foam, the foam is comprised of only struts along the cell edges joined where they intersect, i.e. the cell walls are open. The struts are circular in cross-section. For a given strut aspect ratio, with corresponding cross-sectional area A, the relative density of the core can be determined using:

$$\rho_r = \frac{A \sum_{i=1}^{N} l_i}{V_0}$$

(6.1)

where $l_i$ are the cell strut lengths, $N$ is the total number of cell struts, and $V_0$ is the sandwich core volume containing all the cells.

Five relative densities were investigated: $\rho_r = 0.05, 0.06, 0.07, 0.08,$ and $0.09$ by varying the cross-sectional area of the struts. This range was selected based on that for Duocel. The length of the struts is based on the MMC Al-Zn-Mg-Cu
alloy foam with TiC particles tested in this work, and was found to be equal to 1.7 mm from the XMT scans of the struts as described in Section 4.2.4. The average strut cross-sectional area of the MMC Al-Zn-Mg-Cu alloy foam with TiC particles was found to be 0.562 mm, which corresponds to a relative density of 0.09 for the Kelvin cell core of Fig. 6.1(b).

The Kelvin core geometry was constructed using the CAD program SolidWorks (Dassault Systèmes SolidWorks Corp, Vélizy) and then imported into ABAQUS 6.11 as a single part to be subsequently meshed. The Kelvin core mesh was constructed using element type C3D4. To achieve mesh convergence, 505553 elements were required. The Kelvin core was assigned a self-contact tangential frictionless surface interaction property.

![Fig. 6.1](image1.png)

**Fig. 6.1** (a) Space filling idealised unit cell, modelled using the Kelvin structure (a network of regular tetrakaidecahedrons). (b) Uniform compression FE model showing applied boundary conditions

![Fig. 6.2](image2.png)

**Fig. 6.2** Micrograph showing the typical morphology of the open-cell foam fabricated from an Al-Zn-Mg-Cu (7xxx series) alloy with TiC particles. Adjacent cells A (hexagonal in shape) and B (approximately square in shape) indicate suitability of idealised Kelvin cell model
6.2.2 Applied boundary conditions, loads, and material model

The applied boundary conditions are shown in Fig. 6.1(b). The bottom face of the sandwich panel was constrained in all degrees of freedom. Two symmetry boundary conditions were applied so that only one quarter of the entire panel needed to be modelled – this was achieved by applying symmetry boundary conditions along the mid-points of the panel’s width and depth.

A uniform compressive load was modelled by applying a MPC along the bottom surface of the upper facesheet, whereby all nodes along that surface were tied to the central node. Compressive load vs. displacement plots were obtained by moving this central node in the vertical direction under a controlled, linear displacement. The MPC was applied to the bottom surface (as opposed to the upper surface) to ensure that only the properties of the core were determined (i.e. by ensuring no deformation of the upper facesheet). Facesheets were used as opposed to rigid bodies as they resulted in a more computationally efficient model with shorter run times due to less complex surface interactions.

The foam core was assigned a solid, homogenous cross-section. The viscoplastic damage constitutive equation set previously presented and calibrated in Section 5.2 was implemented into the ABAQUS model for the metal foam core through the user-defined subroutine VUMAT. The facesheets were assigned the material properties of aluminium alloy Al-7075-0, as given in Section 3.2.3.

6.2.3 FE results for 3D uniform compression model

Fig. 6.3(a) shows the stress-strain plots up to a strain of 0.1 for varying relative densities, $\rho_r = 5, 6, 7, 8$ and 9 %, for the foam core sandwich panel subjected to uniform compression. The insert shows the full stress-strain plot for $\rho_r = 9$ %. The graph displays a trend associated with open-cell metal foams, with three distinct
regions: a linear-elastic regime, followed by a plateau of roughly constant stress, and finally a regime of steeply rising stress as the core crushes.

Fig. 6.3(b) plots the variation of Young’s modulus, $E$, of the foam core with increasing relative density. As $\rho_r$ increases from 5 % to 9 %, $E$ increases from 173 MPa to 490 MPa. Ashby et al. (2000) provide the following scaling relationship for commercially available metal foams:

$$E \approx \alpha_2 E_s \left(\frac{\rho}{\rho_s}\right)^n$$  \hspace{1cm} (6.2)

where $n$ has a value between 1.8 and 2.2 and $\alpha_2$ between 0.1 and 4, depending on the structure of the foam. By fitting this equation to the FE results of Fig. 6.3(a), it is found that there is good agreement for $n = 2$ and $\alpha_2 = 0.964$ with a maximum error between the numerical results and the fitted equation of 11 %.

The range of $\rho_r$ between 5 % and 9 % reflects that of commercially available open-cell foams. FE simulations were also conducted for a core relative density of $\rho_r = 1$ % and 35 %. The effective Young’s modulus was found to be 18.8 MPa and 2372.1 MPa respectively, however using Eqn. (6.2) with $n = 2$ and $\alpha_2 = 0.964$ predicts 6.7 MPa and 8266.3 MPa respectively. Therefore, the scaling relationship of Ashby et al. (2000) does not hold for extreme values of $\rho_r$.

As the foam core is isotropic, its shear modulus, $G$, can be determined using:

$$G = \frac{E}{2(1+\nu)}$$  \hspace{1cm} (6.3)

where $\nu$ is the Poisson’s ratio of the core (approximately equal to 0.33). The validity of this approach is confirmed in Ashby et al. (2000). Therefore, when $\rho_r = 5$ %, $G = 65.0$ MPa and when $\rho_r = 9$ %, $G = 184.2$ MPa. This lies within the range of ERG/Duocel foams (Ashby et al, 2000; http://www.ergaerospace.com). 

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Fig. 6.3 (a) Stress-strain plots for the uniform compression FE model for varying relative densities = 5, 6, 7, 8 and 9 % showing initial response. Insert shows full stress-strain plot for a relative density of 9 %, with an elastic region followed by plastic collapse with a long plateau and finally densification. (b) Plot of effective Young’s modulus vs. relative density: comparison of uniform compression FE results with Ashby et al. (2000) fit. Fitted equation also shown
As shown in the insert of Fig. 6.3(a), the idealised open-cell Kelvin foam has a long, well-defined plateau stress that continues up to the densification strain, $\varepsilon_D$. The plateau stress scales with relative density as follows:

$$\bar{\sigma}_{pl} \approx \sigma_{ys} \left( \frac{\rho}{\rho_s} \right)^{3/2} \quad (6.4)$$

This agrees well with the scaling law for commercially available metal foams given in Ashby et al. (2000) for the plateau stress and there is a maximum error between the numerical results (see Table 6.1) and the fitted Eqn. (6.4) of 11 %.

The variation of strain energy density up to the densification strain as $\rho_r$ increases from 5 % to 9 % is shown in Table 6.1. The strain energy density was directly determined from the area under the stress-strain graphs using the trapezium rule (see Eqn. 3.26), but can also be roughly approximated using the index $\bar{\sigma}_{pl}\varepsilon_D$.

<table>
<thead>
<tr>
<th>Relative density (%)</th>
<th>Plateau stress (MPa)</th>
<th>Strain energy density (MJm$^{-3}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>2.24</td>
<td>1.54</td>
</tr>
<tr>
<td>6</td>
<td>3.05</td>
<td>2.07</td>
</tr>
<tr>
<td>7</td>
<td>3.62</td>
<td>2.49</td>
</tr>
<tr>
<td>8</td>
<td>4.39</td>
<td>2.98</td>
</tr>
<tr>
<td>9</td>
<td>4.79</td>
<td>3.30</td>
</tr>
</tbody>
</table>

Table 6.1 FE results of plateau stress and strain energy density as a function of relative density for an idealised open-cell Kelvin foam subjected to uniform compression loading
6.3 Three and four point bending FE models

6.3.1 Overview of FE models

This Section sets out to validate (a) the implementation of the damage model of Section 5.2 into the FE simulations presented in this work, and (b) the results of the uniform compression FE model presented in Section 6.2.3. To achieve this, the shear modulus of the Kelvin core with $p_r = 9\%$ is determined in this Section via numerical simulations of three and four point bending loading scenarios.

The three and four point bending FE models have been produced in accordance with ASTM C-393-00 Standard Test Method for Flexural Properties of Sandwich Constructions, so as to provide a virtual standardised test to assess the foam core elastic properties – see Fig. 6.4(a) and (b). To ensure simple sandwich beam theory is valid when analysing the results of the models, the span length was set to be greater than 20 times the sandwich thickness (equal to 236 mm and 11.782 mm respectively) with the ratio of facing thickness to core thickness less than 0.1 (equal to 0.8 mm and 10.182 mm respectively). The width and length of the specimens followed the guidance of Section 5 of ASTM C-393-00 and were equal to 29.7 mm and 292 mm respectively.

The FE models were run in ABAQUS standard as ABAQUS explicit exhibited prohibitively long run times when trying to capture quasi-static test conditions. The standard simulations diverged when the measured load vs. displacement plots reached a maximum due to high non-linearities in the solutions, however this is not of major importance as the results of this Section are concerned with the elastic properties of the core only. To achieve mesh convergence, 1449648 elements were required. Fig. 6.5(a) and (b) show the convergence trends for the slope of the elastic region of the load-midspan deflection plots of the three and four point bending models respectively with increasing number of mesh elements.
6.3.2 Applied boundary conditions, loads, and material model

Fig. 6.4(a) and (b) show the applied boundary conditions for the three and four point bending FE models. Two symmetry boundary conditions were applied so that only one quarter of the entire panel needed to be modelled in each instance – this was achieved by applying symmetry boundary conditions along the mid-points of the panel’s width and depth. The panels were supported by rigid pins on the bottom face that were constrained in all degrees of freedom via a reference point. Vertical loading was applied via a reference point associated with the rigid pins on the upper face, ensuring all other degrees of freedom were constrained; this reference point was translated vertically under a controlled, linear displacement.

In order to validate the implementation of the damage model of Section 5.2 through the user-define subroutine VUMAT into the uniform compression FE
model of Section 6.2, the tabulated stress-strain data from the calibrated damage model for an individual strut (see Fig. 5.1) was used as the material model for the foam core in the bending simulations. Hence, the bending simulations were effectively run with the same material properties as the uniform compression model but without the implementation of a VUMAT subroutine that uses a viscoplastic damage constitutive equation set (and as such any analysis of damage within the core was limited). The facesheets were assigned the same material properties as those of Section 3.2.3.

Fig. 6.5 Convergence trend for the slope of the elastic region of the load-midspan deflection plots of (a) the three and (b) the four point bending model respectively with increasing number of elements
6.3.3 FE results of three and four point bending simulations

Fig. 6.6(a) and Fig. 6.7(a) show the load-midspan deflection plots for the three and four point bending loading scenarios respectively. Fig. 6.6(b) and Fig. 6.7(b) show the collapse behaviour of the sandwich panel under both loading scenarios at varying displacements. No localised core crushing effects were observed in the vicinity of the loading pins, hence the diameter of the loading pins (= 24 mm) was appropriate.

The uniform compression FE model of Section 6.2 gave a value of $G = 184.2$ MPa for $\rho_r = 9\%$. Using Eqns. (6.5) and (6.6), in conjunction with Eqns. (6.7) and (6.8), presented in ASTM C-393-00, allows this value to be superimposed on the load-midspan deflection plots up to the onset of plastic deformation (determined from the contour plots for stress of the FE results).

\[
\Delta_1 = \frac{PL^3}{48D} + \frac{PL}{4U} \tag{6.5}
\]

\[
\Delta_2 = \frac{11PL^3}{768D} + \frac{PL}{8U} \tag{6.6}
\]

\[
D = \frac{E_f(d^3 - c^3)b}{12} \tag{6.7}
\]

\[
U = \frac{G(d + c)^2b}{4c} \tag{6.8}
\]

where $\Delta_1$ and $\Delta_2$ are the midspan deflections of the panel subjected to three and four point bending respectively and are determined from the sum of the bending and shear components, $P$ is the measured load, $L$ is the span length, $D$ is the panel bending stiffness, $U$ is the panel shear rigidity, $d$ is the sandwich thickness, $c$ is the core thickness, $E_f$ is the facing modulus, $G$ is the core shear modulus, and $b$ is the sandwich width.
Fig. 6.6 (a) Load-midspan deflection plot for the three point bending loading scenario. The ASTM result predicted using $G = 184.2$ MPa from the uniform compression FE model is superimposed. (b) Collapse behaviour of the metal foam core sandwich panel subjected to three point bending at varying midspan deflections ($= 0.00, 1.75, 3.50, 5.25$ mm)
Fig. 6.7 (a) Load-midspan deflection plot for the four point bending loading scenario. The ASTM result predicted using $G = 184.2$ MPa from the uniform compression FE model is superimposed. (b) Collapse behaviour of the metal foam core sandwich panel subjected to four point bending at varying midspan deflections ($= 0.0, 2.5, 5.0, 7.5$ mm)
It can be seen in Fig. 6.6(a) and Fig. 6.7(a) that there is a good agreement between the load-midspan deflection plots of the bending simulations and that produced using Eqns. (6.5) to (6.8) in conjunction with the shear modulus determined from the uniform compression FE results.

Eqns. (6.5) to (6.8) also enable the shear modulus of the foam core to be determined for the three and four point bending simulations. This gives $G = 190.6$ MPa and $G = 194.5$ MPa respectively, which agrees well with $G = 184.2$ MPa from the uniform compression simulation and thus indicates that the damage model was correctly implemented into the uniform compression FE model of Section 6.2.

The shear moduli determined from the three and four point bending simulations can be cross-checked with the equations for determining the flexural stiffness and core shear modulus when knowledge of the deflections of the same sandwich panel are known for both three and four point bending loading scenarios as follows (ASTM C-393-00):

$$D = \frac{P_1 L_1^3}{48\Delta_1} \left[ 1 - \left( \frac{11L_2^2}{8L_1^2} \right) \right]$$ \hspace{1cm} (6.9)

$$G = \frac{P_1 L_1 c \left( \frac{8L_1^2}{11L_2^2} - 1 \right)}{\Delta_1 b (d + c)^2 \left( \frac{16P_1 L_1^3 \Delta_2}{11P_2 L_2^3 \Delta_1} - 1 \right)}$$ \hspace{1cm} (6.10)

where $P_1$ and $P_2$ are the loads determined for three and four point bending respectively, and $L_1$ and $L_2$ are the spans for the three and four point bending scenarios respectively (in this instance they are equal).

Eqns. (6.9) and (6.10), in conjunction with the results of Fig. 6.6(a) and Fig. 6.7(a), give $D = 109.3$ MNmm$^2$ (vs. $D = 102.9$ MNmm$^2$ from Eqn. (6.6)) and $G = 183.5$ MPa. This is in good agreement with the values obtained previously.
6.4 Airplane wing skin tool drop impact FE studies

6.4.1 Overview of FE model

As noted in Chapter 1, low energy impacts can reduce the strength of sandwich structures as well as cause considerable subsurface damage. This is a problem in aircraft structures which may be subjected to tools being dropped during maintenance or foreign object damage during landing and take-off. Hence, an FE model has been developed in this Section, using ABAQUS 6.11, to simulate an indentation loading scenario on a metal foam core sandwich panel. By loading and then unloading the indenter, the objective is to replicate a tool drop impact. Fig. 6.8 illustrates the loading scenario, whereby facesheet stretching and core compression occurs.

An impact energy of 20 J has been selected in order to represent a 2 kg mass being dropped from a 1 m height, which corresponds to an impact velocity of 4.5 ms\(^{-1}\). As shown in Deshpande and Fleck (2000), shock wave effects are negligible for typical aluminium foams for impact velocities less than 50 ms\(^{-1}\). Deshpande and Fleck (2000) found that the dynamic behaviour of the open-cell aluminium foam Duocel is very similar to its quasi-static behaviour for strain rates in the range \(10^{-3}\) s\(^{-1}\) to 5000 s\(^{-1}\) (with impact velocities < 50 ms\(^{-1}\)), with no elevation of the dynamic stress-strain graphs compared to the corresponding quasi-static graphs. Lankford and Dannemann (1998) also found that the compressive strength of Duocel is insensitive to the applied strain rate in the range \(10^{-3}\) s\(^{-1}\) to 1200 s\(^{-1}\). Similar observations are reported in Ma et al. (2009). These findings indicate that the indentation FE model developed in this Section is indeed representative of a tool drop impact under ground repair conditions.

The metal foam core has been modelled using the 3D Kelvin structure, as described in Section 6.2.1. The length and diameter of the struts are based on those
of the real foam dimensions of the MMC fabricated from an Al-Zn-Mg-Cu (7xxx series) alloy with TiC particles that is studied in this work, and have been obtained from XMT scans of the struts as described in Section 4.2.4. The length and diameter are equal to 1.7 mm and 0.562 mm respectively. This gives a strut aspect ratio of 3, which corresponds to a relative density $\rho_r = 0.09$ determined by the cross-sectional area $A$ and specified by Eqn. (6.1).

![Fig. 6.8 FE model showing the indentation loading scenario used to simulate a tool drop impact on a wing sandwich panel configuration](image)

Three other strut aspect ratios were investigated, 1.5, 6 and 9, with corresponding relative densities $\rho_r = 0.35, 0.02$ and 0.01 by varying the cross-sectional area of the struts. In addition, a strut aspect ratio of 1 was modelled by replacing the Kelvin core structure with a continuous solid core.

Edge effects were avoided by ensuring that the indentations were at least one indenter diameter away from the edges of the plate, as outlined in Ashby et al. (2000). The thickness of the sandwich plate was set as a design constraint, equal to 11.8 mm. The facesheets each had thickness 0.8 mm. The indenter diameter was set to 21.7 mm (corresponding to 4 cell lengths) to represent a realistic tool size and was modelled as a 3D analytical rigid part.

The Kelvin core mesh was constructed using ABAQUS element type C3D4. To achieve mesh convergence, 406403 elements were required. The Kelvin core was assigned a self-contact tangential frictionless surface interaction property.
A tangential frictionless surface interaction property was also assigned between the indenter and the upper facesheet of the sandwich panel.

6.4.2 **Applied boundary conditions, loads, and material model**

The applied boundary conditions are shown in Fig. 6.9. The bottom face of the sandwich panel was constrained in all degrees of freedom. Two symmetry boundary conditions were applied so that only one quarter of the entire panel needed to be modelled – this was achieved by applying symmetry boundary conditions along the mid-points of the panel’s width and depth.

To establish the performance of different metal foam cores for a given tool drop impact scenario, the impact energy (i.e. the area under the loading curve) of each of the models (strut aspect ratio = 1, 1.5, 3, 6 and 9) was set to the same value. This was achieved by indenting the sandwich panels by different displacements according to the strut aspect ratio (greater forces are expected for lower strut aspect ratios); this can be observed in Fig. 6.12. Once the indenter reached its maximum displacement the sandwich panel was then fully unloaded. Load vs. displacement plots were obtained from a reference point on the indenter.

The foam core was assigned a solid, homogenous cross-section. The material model of the foam core was as described in Section 6.2.2. The facesheets were assigned the material properties of aluminium alloy Al-7075-0, as given in Section 3.2.3.
6.4.3 Definition of maximum allowable damage and visibility

The maximum allowable damage, \( \omega_{\text{crit}} \), for the struts in the metal foam core was determined from the stress-strain graph of an individual strut as given by the calibrated damage model in Section 5.2 and Fig. 5.1. Failure of the struts is deemed to occur when the UTS is reached. Fig. 6.10 shows the damage-strain graph for one strut (\( \omega \) varies from 0, no damage, to 1, full damage) – it can be seen that at the UTS, the damage is equal to 0.105 (i.e. \( \omega_{\text{crit}} = 0.105 \)).

Visible damage is defined as the region along which the top facesheet has deformed by an angle of at least 10° from the horizontal (see Fig. 6.11). Critical damage is defined as the region in the foam core where struts with a damage of at least 0.105 are found; the level of strut damage after indentation was determined
from contours of $\omega$ in the ABAQUS results viewport. For a given impact, if the facesheet visible damage region (represented by a radius in the horizontal plane, $r_{vd}$) extends at least as far as the critical damage region of the foam core (also represented by a radius in the horizontal plane, $r_{cd}$), then it may be concluded that all of the critical internal core damage is revealed by corresponding facesheet deformation. Fig. 6.11 illustrates the approach.

Fig. 6.10 Stress-strain graph for an individual metal foam strut as determined using the calibrated damage model, and corresponding damage-strain graph. Method for determining the critical damage value is shown.

Fig. 6.11 Method for determining the visible facesheet deformation radius, $r_{vd}$, and the foam core damage radius, $r_{cd}$.
6.4.4 FE results for indentation loading scenarios

Fig. 6.12 shows the force-displacement plots for varying strut aspect ratios = 1 (solid block), 1.5, 3, 6 and 9 for the foam core sandwich panel subjected to loading and unloading of an indenter. The impact energy for each model is the same (equal to 20 J) – this has been achieved by increasing the peak displacement of the indenter for the cores with greater strut aspect ratios to balance the reduction in measured peak force (see insert in Fig. 6.12). Fig. 6.13 shows the collapse behaviour of the metal foam core sandwich panel for a strut aspect ratio of 3 at varying displacements. Localised crushing is observed in the region directly beneath the indenter, with deformation symmetric about the tip of the indenter.

Fig. 6.14 plots both the energy absorbed per unit mass and the damage visibility ratio (defined as the ratio of facesheet visible damage radius to foam core critical damage radius) as a function of varying strut aspect ratios. As the strut aspect ratio increases, so too does the energy absorbed per unit mass, however the damage visibility ratio decreases. There is hence an optimal design range for the strut aspect ratio where a compromise between energy absorbed per unit mass and damage visibility ratio is obtained, and is represented indicatively by the shaded region in Fig. 6.14. As the energy absorbed per unit mass increases at a faster rate than the damage visibility ratio decreases with increasing strut aspect ratio, some sacrifice in damage visibility (i.e. selecting a damage visibility ratio slightly lower than 1) may be desired for a marked improvement in energy absorbed per unit mass (indeed, as the damage visibility ratio decreases from 1 to 0.83 the energy absorbed per unit mass increases from 2 kJ/kg to 4 kJ/kg); in other words, the optimum design range maximises the energy absorbed per unit mass whilst ensuring that damage remains mostly visible. The lower bound of the optimum design range is where the damage visibility ratio is equal to 1; in this instance, this occurs at a strut aspect ratio of 3.2. The upper bound of the optimum design range
is a choice to be made depending on the application, and is not fixed; in Fig. 6.14, it is taken indicatively at a strut aspect ratio of 4.3 as this gives a doubling in energy absorbed per unit mass as the damage visibility ratio decreases by 17%.

![Force-displacement plots for varying strut aspect ratios](image)

**Fig. 6.12** Force-displacement plots for varying strut aspect ratios, \( l/t = 1 \) (solid block), 1.5, 3, 6 and 9. Insert shows corresponding dependence of peak displacement and peak force with varying \( l/t \)

![Collapse behaviour of the metal foam core sandwich panel](image)

**Fig. 6.13** Collapse behaviour of the metal foam core sandwich panel for \( l/t = 3 \) at varying displacements (= 0, 1.25, 2.5, 3.75, and 5 mm)
6.4.5 Effect of varying indenter radius

The effect of varying angles of tool drop impact is now considered by varying the indenter radius in the model described in Section 6.4.1, for a relative density of 9%. Three indenter radii have been considered: 10.84, 21.69, and 32.53 mm.

Fig. 6.15 shows the force-displacement plots for varying indenter radii for the foam core sandwich panel subjected to loading and unloading of an indenter. The impact energy for each model is the same (equal to 20 J, following the studies in Section 6.4.4) – this has been achieved by increasing the peak displacement of the indenter for smaller indenter radii to balance the reduction in measured peak force (see insert in Fig. 6.15).

The damage visibility ratio for the three models with varying indenter radii was found to be, using the method outlined in Section 6.4.3, 1.02, 1.04, and 1.08 for a radius of 10.84, 21.69, and 32.53 mm respectively. Hence, there is a small improvement in the damage visibility ratio for increasing indenter radius, and damage always remains visible (i.e. $\geq 1$).
6.5 Effect of imperfections on the mechanical properties and damage visibility of open-cell metal foam sandwich panels

6.5.1 Effect of fractured cell walls

Fractured cell walls represent a common morphological defect found in commercial metal foams (Guo et al., 1999; Silva and Gibson, 1997). Fractured cell walls have been introduced into the foam core of the uniform compression FE model of Section 6.2 (with $\rho_r = 9 \%$) by removing individual struts regularly throughout the model using the CAD program SolidWorks. Four foam core scenarios have been investigated: 0.0, 1.8, 3.7, and 7.3 % fractured cell walls, which is a similar range to that considered by Chen et al. (1999) for the 2D case. Fig. 6.16 shows the location of the fractured cell walls for a number fraction of 3.7 %.
Fig. 6.16 Foam core with fractured cell walls – individual struts are missing through the depth of the core (number fraction = 3.7 %)

6.5.1.1 Effect of fractured cell walls on the mechanical properties of the foam

Fig. 6.17 shows the stress-strain plots for the uniform compression FE model for varying proportions of fractured cells walls = 0.0, 1.8, 3.7, and 7.3 %. The insert of Fig. 6.17 plots the variation of effective shear modulus, $G$, and plateau stress, $\bar{\sigma}_{pl}$, with increasing % fractured cell walls. There is a significant drop in both $G$ and $\bar{\sigma}_{pl}$ as the number fraction of fractured cell walls increases from 0.0 % to 7.3 %. $G$ decreases from 184.2 MPa to 61.2 MPa (a 66.8 % decrease), whilst $\bar{\sigma}_{pl}$ decreases from 4.62 MPa to 2.06 MPa (a 55.4 % decrease).

Fig. 6.17 Stress-strain plots for the uniform compression FE model for varying proportions of fractured cell walls = 0.0, 1.8, 3.7, and 7.3 % showing initial response. Insert shows the reduction in $G$ and $\bar{\sigma}_{pl}$ (normalised with respect to the values for 0.0 % fractured cell walls – $G_0$ and $\bar{\sigma}_{pl0}$) with increasing % fractured cell walls
6.5.1.2 Effect of fractured cell walls on the damage visibility

This Section aims to further the work of Section 6.4 by assessing the effect of fractured cell walls on the extent of visible structural damage in metal foam core sandwich panels used as an airplane wing skin material when subjected to a tool drop impact. The FE model for the indentation loading scenarios in this Section is the same as that presented in Section 6.4. The metal foam core has been modelled as described in Section 6.5.1.1, with varying degrees of fractured cell walls = 0.0, 1.8, 3.7, and 7.3%.

To establish the performance of different metal foam cores for a given tool drop impact scenario, the impact energy in each of the models (fractured cells walls = 0.0, 1.8, 3.7, and 7.3%) was set to the same value (equal to 20 J) – this has been achieved by increasing the peak displacement of the indenter for the cores with greater % fractured cell walls to balance the reduction in measured peak force (see insert in Fig. 6.18). Fig. 6.18 shows the force-displacement plots for varying % fractured cell walls = 0.0, 1.8, 3.7, and 7.3% for the foam core sandwich panel subjected to loading and unloading of an indenter.

The damage visibility ratio for the four models with % fractured cell walls = 0.0, 1.8, 3.7, and 7.3% was found to be, using the method outlined in Section 6.4.3, 1.02, 0.93, 0.94, and 0.78 respectively. The damage visibility ratio is therefore only notably impeded at a high number fraction of fractured cell walls.
Fig. 6.18 Force-displacement plots for varying % fractured cell walls = 0.0, 1.8, 3.7, and 7.3 
%. Insert shows corresponding dependence of peak displacement and peak force with varying 
% fractured cell walls

6.5.2 Effect of missing cells

Missing cells appear in commercial metal foams as large holes (Chen et al., 1999). 
Missing cells have been introduced into the foam core of the uniform compression 
FE model of Section 6.2 (with $\rho_r = 9\%$) by removing groups of adjacent cells 
throughout the model using the CAD program SolidWorks. Three foam core 
scenarios have been investigated: 2, 4, and 8 adjacent missing cells in each quarter 
of the model, which is a similar range to that considered by Chen et al. (1999) for 
the 2D case. Fig. 6.19 shows the location of the missing cells for the uniform 
compression FE model.
6.5.2.1 Effect of missing cells on the mechanical properties of the foam

Fig. 6.20 shows the stress-strain plots for the uniform compression FE model for varying proportions of missing cells = 0, 2, 4, and 8 per quarter of the model. The insert of Fig. 6.20 plots the variation of effective shear modulus, \( G \), and plateau stress, \( \bar{\sigma}_{\text{pl}} \), with increasing number of missing cells. It can be seen that there is a larger drop in \( G \) than \( \bar{\sigma}_{\text{pl}} \) as the number of missing cells increases from 0 to 8. \( G \) decreases from 184.2 MPa to 101.9 MPa (a 44.7 % decrease), whilst \( \bar{\sigma}_{\text{pl}} \) decreases from 4.62 MPa to 4.03 MPa (a 12.8 % decrease). Comparing these to the results of Section 6.5.1.1, it can be seen that missing cells have less of a knock-down effect than fractured cell walls on both \( G \) and \( \bar{\sigma}_{\text{pl}} \).
Fig. 6.20 Stress-strain plots for the uniform compression FE model for varying number of missing cells = 0, 2, 4, and 8 per quarter of the model to assess the effect of large holes present in the foam; initial response shown. Insert shows the reduction in $G$ and $\bar{\sigma}_{pl}$ (normalised with respect to the values for 0 missing cells – $G_0$ and $\bar{\sigma}_{pl0}$) with increasing number of missing cells.

6.5.2.2 Effect of missing cells on the damage visibility

Initially, the foam core with missing cells shown in Fig. 6.19 that was used in the uniform compression FE model was also used to investigate the effect of missing cells on the extent of visible structural damage in the indentation model described in Section 6.4.1. However, it was found that the missing cells had no effect on either the indentation force-displacement plots of the foam core or on the damage visibility. This is because they were located far enough away from the indenter to have no effect.

Therefore, as a more appropriate study, the presence of 8 missing cells in the foam core was considered for the indentation loading scenario at varying distances from the indenter location. 8 missing cells were considered (I) directly under the indenter, (II) at a radius of 13.3 mm from the centre of the indenter, and (III) at a radius of 20.1 mm from the centre of the indenter – see Fig. 6.21.
Fig. 6.21 Schematic showing the three locations (I), (II), and (III) of the 8 missing cells in the foam core geometry for the indentation loading scenarios. Cells are missing through the entire core thickness at each location. Indenter location shown by black circle

Fig. 6.22 shows the force-displacement plots for the varying locations of the 8 missing cells in the foam core sandwich panel subjected to loading and unloading of an indenter. The impact energy for each model is the same (equal to 20 J, as per the fractured cell wall studies) – this has been achieved by increasing the peak displacement of the indenter for the cores with the missing cells located closest to the indenter so as to balance the reduction in measured peak force (see insert in Fig. 6.22).

The damage visibility ratio for the three models with varying locations of the 8 missing cells was found to be, using the method outlined in Section 6.4.3, 1.00, 1.00, and 1.02 for locations (I), (II), and (III) respectively. The damage visibility ratio is therefore unimpeded by the location of the missing cells, and damage always remains visible (i.e. ≥ 1).
6.5.3 Combined effect of fractured cell walls and missing cells

It is likely for different types of imperfections to coexist in a commercial metal foam (Chen et al., 1999). Therefore, the combined effect of fractured cell walls and missing cells is now investigated for both uniform compression and indentation loading scenarios. A foam core of relative density 9 % with 3.7 % fractured cell walls (located as described in Section 6.5.1) and varying extents of missing cells has been analysed. The location of the missing cells is outlined in Sections 6.5.2.1 and 6.5.2.2.

6.5.3.1 Combined effect of fractured cell walls and missing cells on the mechanical properties of the foam

Fig. 6.23 shows the stress-strain plots for the uniform compression FE model with 3.7 % fractured cell walls and varying proportions of missing cells = 0, 2, 4, and 8
per quarter of the model. The insert of Fig. 6.23 plots the variation of $G$ and $\bar{\sigma}_{pl}$ with increasing number of missing cells.

![Graph showing stress-strain plots for the uniform compression FE model with varying number of missing cells and fractured cell walls.](image)

**Fig. 6.23** Stress-strain plots for the uniform compression FE model with 3.7% fractured cell walls and varying number of missing cells = 0, 2, 4, and 8 per quarter of the model; initial response shown. Insert shows reduction in $G$ and $\bar{\sigma}_{pl}$ (normalised with respect to the values for 0 missing cells and 3.7% fractured cell walls – $G_0$ and $\bar{\sigma}_{pl0}$) with increasing number of missing cells.

It can be seen that the simultaneous effect of missing cells and fractured cell edges is such that the foam is not particularly sensitive to the presence of large isolated holes when subjected to uniform compression. $G$ decreases from 81.6 MPa to 73.5 MPa (a 9.9% decrease) as the number of missing cells increases from 0 to 8. This is a significantly lower percent decrease than that observed as the number of missing cells increases in a foam core with no fractured cell walls ($G$ was found to drop by 44.7%, as seen in Section 6.5.2.1 and Fig. 6.20). These results are in general agreement with other theoretical as well as experimental findings (Chen et al., 1999; Olurin et al., 2000). $\bar{\sigma}_{pl}$ decreases from 3.37 MPa to 3.02 MPa (a 10.4% decrease), which is similar to the decrease observed in the foam core with no fractured cell walls (see Section 6.5.2.1).
6.5.3.2 Combined effect of fractured cell walls and missing cells on the damage visibility

Fig. 6.24 shows the force-displacement plots for varying locations of 8 missing cells in a foam core sandwich panel with 3.7 % fractured cell walls subjected to loading and unloading of an indenter. The impact energy for each model is the same (equal to 20 J, as per Section 6.5.2.2) – this is illustrated by the insert of Fig. 6.24.

By comparing Fig. 6.24 with Fig. 6.22, it is observed that when the foam core comprises of 3.7 % fractured cell walls and missing cells located directly under the indenter, the recorded peak force is lower than that for the same core with no fractured cell walls. There is also a corresponding increase in the peak displacement as the impact energy is the same in all models.

The damage visibility ratio for the three models with 3.7 % fractured cell walls and varying locations of the 8 missing cells was found to be 1.00, 0.97, and
0.97 for locations (I), (II), and (III) respectively. The damage visibility ratio is therefore similar to that observed for a core with 3.7% fractured cell walls and no missing cells (see Section 6.5.1.2), as well as that observed for a core with no fractured cell walls and varying locations of the 8 missing cells (see Section 6.5.3.2), and is always close to 1 and hence mostly visible.
Chapter 7.

Conclusions
7.1 Summary and key results

The overarching aim of this work was to investigate the potential use of aluminium foam sandwich panels as a wing skin material as part of the design and development of new lightweight aircraft wing designs lacking ribs and stringers. Sandwich structures that comprise of metal foam cores (Betts, 2012):

- Can display isotropy in mechanical properties;
- Can be manufactured with integral skins;
- Can be readily made into curved shapes;
- Open-cell foams do not trap moisture;
- Combine low density with good bending stiffness and strength;
- Display a densification stage in a compressive stress-strain plot, with the implication that the integrity of a metal foam core sandwich panel is not necessarily compromised when subjected to impacts.

7.1.1 Analytical modelling of honeycomb cores

An analytical model that uses Timoshenko beam theory (Timoshenko, 1956) has been developed in this work to predict the in-plane Young’s modulus of a regular 2D hexagonal honeycomb core, determined as follows:

\[ E_2 = \frac{2.3}{\left( \frac{i^3}{E_s t^3} + \frac{1}{2ktG_s} \right)} \] (7.1)

This model has been compared to predictions made by the analytical model of Ashby and Gibson (1988) that employs Euler-Bernoulli beam theory. Euler-Bernoulli beam theory was found to be inadequate for predicting the Young’s modulus of the modelled honeycomb at values of \( \frac{t}{i} > 0.25 \), for which Timoshenko beam theory becomes necessary (Betts et al., 2012).
The open-cell metal foam tested in this work (see Section 7.1.3) has an average strut length and diameter of 1.7 mm and 0.562 mm respectively (i.e. $\frac{L_i}{d_i} = 0.33 > 0.25$). It has been established in this work that Euler-Bernoulli beam theory therefore over-predicts the Young’s modulus of an equivalent 2D hexagonal honeycomb representation of the foam by 17% with respect to Timoshenko beam theory (Betts et al, 2012).

### 7.1.2 2D FE modelling of honeycomb cores

A repeating unit cell 2D FE modelling procedure has been established in this work to model the in-plane mechanical behaviour of infinitely long honeycomb core sandwich panels (e.g. Young’s modulus, energy absorbed, etc.). The FE model can shed light on the mechanics of more complex 3D metal foams. PBCs have been implemented within the model to eliminate edge effects from the mechanical analysis; this avoids the need to physically model the actual sandwich panel length, which is advantageous in terms of both the time to (I) construct the model, and (II) run the model.

The model was validated by comparing the results to those predicted by the analytical Timoshenko beam theory model of Section 7.1.1; it was found that there is good agreement between the two with a 4% difference in the recorded effective Young’s modulus. The load-displacement plots display three distinct regions: a linear-elastic regime, followed by a plateau of roughly constant stress, and finally a regime of steeply rising stress. This behaviour is also typically observed in commercial open-cell foams (Ashby et al., 2000).

FE modelling of regular tessellated honeycombs (i.e. hexagonal, square, and equilateral triangle) has been conducted using the established 2D model. It was found that for a given value of relative density the hexagonal honeycomb absorbed the most energy, whilst the square and triangular honeycombs displayed a similar
value of initial collapse strength and much higher value of Young’s modulus than the hexagonal honeycomb, making them the best choice for lightweight structural applications (Betts et al., 2012).

7.1.3 Microtensile testing of open-cell metal foam struts

A novel microtensile test procedure has been developed in this work to directly determine the mechanical properties of individual aluminium alloy foam struts. This was motivated by the observation that due to the foaming process and length scale of metal foam struts, there can be marked differences between the mechanical properties of the bulk alloy and the individual struts caused by differences in both composition and microstructure. The conversion of the force data to stress has been achieved using XMT scans of the undeformed struts.

The foam tested in this work is a MMC fabricated from an Al-Zn-Mg-Cu (7xxx series) alloy with TiC particles. Micrographs of the strut fracture surface were captured after testing using the SEM. Microvoids and dimples can be observed at higher magnifications, indicative of ductile fracture.

The measured strut properties showed a significant reduction in elastic stiffness compared to the typical value of 70 GPa for aluminium alloys, in the order of 14 GPa. As a correction, it was proposed that at the onset of yielding the struts experience uniaxial tensile testing conditions and slippage between the grips and tested strut is at that point significantly reduced (Betts et al., 2013). This proposal was investigated using a realistic FE modelling procedure of the as-tested struts as discussed in Section 7.1.4.

7.1.4 XMT based FE modelling of open-cell metal foam struts

A FE model of the as-tested struts of Section 7.1.3 has been developed using XMT scans of the undeformed struts to define the geometry. Strut deformation was
described by continuum viscoplastic damage constitutive equations calibrated using microtensile test data for the aluminium alloy’s optimally aged condition, and implemented into ABAQUS through the user-defined subroutine VUMAT.

This FE model was used to determine the reason for the considerable reduction in elastic stiffness observed during microtensile testing compared to the typical value of 70 GPa for aluminium alloys.

It has been established that, for the metal foam material investigated in this work, strut curvature has a minimal impact on the measured strut stiffness. Slippage between the grips and the strut during microtensile testing appears to be the chief factor in the recorded reduction in stiffness. The FE model was subsequently used to develop a procedure, described in Section 4.4.2, that compensates for the effect of grip slippage inherent in the microtensile testing of metal foam struts (Betts et al., 2013).

7.1.5 3D FE modelling of open-cell metal foam core sandwich panels

The calibrated constitutive equations outlined in Section 5.2.2 have been implemented in this work into ABAQUS repeating unit cell FE models of open-cell metal foam core sandwich panels through the user-defined subroutine VUMAT.

A 3D FE model of an open-cell aluminium alloy metal foam core sandwich panel subjected to uniform compression has been created to study the effect of varying the foam strut aspect ratio on the elastic properties of the core. It was found that as the relative density increases from 5 % to 9 %, the effective Young’s modulus increases from 173 MPa to 490 MPa. These results were compared to a scaling relationship relating the effective Young’s modulus to the relative density for commercially available metal foams presented in Ashby et al. (2000), and it
was found that there is a good agreement between the numerical results and the fitted equation with a maximum difference of 11 % in the recorded effective Young’s modulus for a given relative density.

In addition, FE models of the open-cell metal foam core sandwich panel subjected to three point and four point bending were produced in accordance with ASTM C-393-00, hence providing a virtual standardised test to assess the foam core elastic properties. These bending FE models used a tabulated elastic-plastic material model and were employed to verify the implementation of the damage model in the uniform compression FE simulations. The core shear modulus determined from knowledge of the deflections of the same sandwich panel subjected to both three and four point bending loading scenarios was found to be 183.5 MPa, which is in good agreement with the value of G = 184.2 MPa obtained from the uniform compression simulation.

The extent of structural damage in a metal foam core sandwich panel was simulated for indentation loading scenarios indicative of an accidental tool strike under ground repair conditions, which is one important design consideration for potential alternative airplane wing skin materials. A range of optimal strut aspect ratios of $3.2 \leq \frac{h}{t} \leq 4.5$ was identified through simulation that provides the best energy absorption per unit mass whilst ensuring core damage is accurately reflected by facesheet deformation, which is necessary for detection and repair.

The effect of two different types of morphological imperfections – fractured cell walls and missing cells – on the elastic properties of the foam core has been studied for uniform compression. Furthermore, the effect of these imperfections on the extent of visible structural damage of the foam core has been studied for indentation loading scenarios. The shear strength (and hence the Young’s modulus) of the foam core experienced a significant knock-down effect under the presence of both fractured cell walls and missing cells; G decreased by
67% as the proportion of fractured cell walls increased from 0% to 7.3% and G decreased by 45% as the proportion of missing cells increased from 0% to 11.1%. The simultaneous effect of missing cells and fractured cell edges is such that the shear strength of the foam core is not particularly sensitive to the presence of large isolated holes. The extent of visible structural damage was largely unaffected by either type of defect.

7.2 Future work

7.2.1 DIC analysis of metal foam struts

In this work (Section 4.4.2), DIC analysis was conducted on the video frames captured during the microtensile tests of the foam struts in an attempt to improve upon the Gatan Microtest 300 rig reported values of strain, based on tracking identifiable features in the surface structure. However, due to the unavoidable changes in contrast between images in the same stack, the Young’s modulus determined from the analysis was insufficiently accurate to be of practical use.

For a full DIC analysis that provides a high accuracy, full-field strain measurement of the strut, the specimen surface must have a random high contrast surface pattern (i.e. a speckle pattern) which deforms together with the specimen surface as a carrier of deformation information (Pan et al., 2009). The speckle pattern would need to be made by first spraying white paint and then fine black dots over the entire strut surface. It may also be necessary to first polish the strut surface. This would remove the problem encountered due to changes in contrast between images in the same stack. However, the facilities available at Imperial College during the preparation of this thesis were not suitable for such a small-scale DIC analysis.
7.2.2 Metal foam core sandwich panel experimental tests

In this work, the implementation of a continuum viscoplastic damage model into the 3D FE models of open-cell metal foam core sandwich panels was validated by using (I) experimental data for a range of commercially available metal foams in the literature (e.g. Ashby et al, 2000; http://www.ergaerospace.com) as well as scaling relationships derived from the experimental data, (II) three and four point bending FE models produced in accordance with ASTM C-393-00 and employing a tabulated elastic-plastic material model, and (III) FE results in the literature (e.g. Chen et al., 1999).

Macroscopic tests (uniform compression, three and four point bending, and indentation loading) on aluminium facesheet sandwich constructions with the cores made using the same metal foam analysed in this work (a MMC fabricated from an Al-Zn-Mg-Cu (7xxx series) alloy with TiC particles) were originally intended to be conducted. However, this was not feasible due to a lack of available samples. Such tests would have enabled the FE modelling work to be further validated and aided in identifying failure mechanisms, mechanical properties and energy absorption levels.

7.2.3 Effect of cyclic loading on metal foam core sandwich panels

Airplane wings are subjected to repeated, cyclic loads. The resulting cyclic stresses can lead to microscopic physical damage that can eventually develop to form a crack or other macroscopic damage that leads to failure of the component. Fig. 7.1 shows an example of the irregular, cyclic load vs. time history encountered during one flight of a fixed-wing aircraft.

The effect of cyclic loading on the degradation of strength should therefore be simulated for aluminium foam core sandwiches for potential use as a wing skin
material. The analyses could be used to predict probable locations of damage within the structure under fatigue loading. The results could then be compared to appropriate known experimental data in the literature to validate the outcomes, e.g. Harte et al. (1999), McCullough et al. (2000), Olurin et al. (2001), and Sugimura et al. (1999).

Fig. 7.1 (a) Actual loads for one flight of a fixed-wing aircraft, and (b) a simplified version of this loading. Working loads occur due to take-offs, manoeuvres, and landings, and there are vibratory loads due to runway roughness and air turbulence, as well as wind gust loads in storms, (Dowling, 1999)

7.2.4 Improvements in the manufacturing process of metal foams

The various application fields for metal foams, including concepts that are still in the verification phase, have been extensively reviewed by Banhart (2001). These comprise of three overarching categories: (I) structural applications, e.g. light, stiff metal foam core sandwich panels with excellent damping behaviour that show promise in ship building, (II) functional applications, e.g. highly conductive open-cell foams based on copper or aluminium that can be used as heat exchangers; the open-cell structure allows gases or liquids to flow through them, allowing heat to be added or removed, and (III) decorative applications, e.g. aluminium foams have
been used to build designer furniture and foams based on gold or silver have potential to be a new material for jewellery.

However, despite these potential applications, mass production of metal foams has been limited primarily due to (I) an insufficient ability to produce high quality materials with good reproducibility of their properties, due to a lack of control over the structure and morphology of the foams, and (II) high manufacturing costs (see e.g. Ashby et al. (2000) and Banhart (2001)). In addition, for most manufacturing processes, there is currently no applicable analytical or numerical model that can predict the effect of parameter changes – at present, improvements are typically made by trial-and-error.

With improvements in the manufacturing processes of metal foams there is a potential for future industrial mass market applications. The effects of foaming parameters including temperature, pressure, and cooling rate on the final foam properties (including the extent of morphological imperfections) need be studied in greater detail and optimised.
References


Appendix A.

Scholarly output
A.1 Journal papers


A.2 Conference papers


A.3 Book chapters


A.4 Awards

Awarded the commended review of the 2011 Materials Literature Review Prize of the Institute of Materials, Minerals and Mining.