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Exhibit H

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The Use of Honeycomb in the Design of Innovative Helmets

by

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Abstract

i

Motorbike riders are among the most vulnerable road users. The improvement of the protection offered by motorcycle helmets through use of non-conventional energy absorbing materials could significantly reduce the number of motorcyclists' fatalities.

This thesis investigates the coupling of hexagonal aluminium honeycomb with polymeric foams for the design of innovative and safer motorbike helmets.

The compressive behaviour and energy absorption properties of two layered foamhoneycomb composites are assessed experimentally. The experiments include quasistatic and impact compressive tests. Experimental outcomes show an increase of the energy absorbed by the two-layered materials with respect to the one provided by foams currently used for the manufacturing of helmets, tested under the same conditions. A finite element model representing the two-layered materials is also proposed. The model is validated against the experimental results. An accurate reproduction of the experiments is attained.

A commercially available helmet is then modified to accommodate aluminium honeycombs in the energy absorbing liner, and standard tests are performed. The investigation includes also the testing of unmodified helmets, presenting same geometry and material properties of the prototypes. The experiments consist of impacts against a flat and kerbstone surfaces, as prescribed by standards. The dynamical responses of the prototypes and their commercial counterparts are compared. It is found that for impacts against the kerbstone anvil, the prototypes offer a noticeable reduction of the accelerations transmitted to the head, compared to the commercial helmets. For impacts against the flat surface, commercial helmets generally provide better protection to the head, which highlights a non optimum design of the prototype helmet and the limitations of using aluminium honeycombs as reinforcement materials.

Experimental findings are later used to validate a finite element model of the prototype, where the two-layered model presented in this thesis is implemented. Numerical results are in good agreement with experimental findings.

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Declaration of Originality

I hereby declare that the work presented in this thesis is my own and based on research carried out at Imperial College London. No part of this thesis has been submitted elsewhere for any other degree or qualification. Where information is derived from other studies, the sources are indicated in the text and appropriately referenced.

Gaetano Caserta

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iv

Dissemination

The research presented in this thesis has continuously been disseminated within the scientific as well as industrial community during the course of the project. The dissemination has been pursued by means of:

Journal papers

- G. Caserta, L. Iannucci, U. Galvanetto. "Static and Dynamic Energy Absorption of Aluminium Honeycombs and Polymeric Foam Composites". *Mechanics of Advanced Materials and Structures*, 17 (5), 2010, pp. 366-376.
- G. Caserta, L. Iannucci, U. Galvanetto. "Shock absorption performances of a motorbike helmet with honeycomb reinforced liner". Composite Structures, 93, 2011, pp. 2748-2759;

Conference papers

- Caserta, G., Iannucci, L., Galvanetto, U. "Micromechanics analysis applied to the modelling of aluminium honeycomb and EPS foam composites". *Proceedings* of the 7th European Ls-Dyna Users conference, 14th -16th May Salzburg, Austria (2009).
- Caserta, G., Iannucci, L., Galvanetto, U. "Static and Dynamic Energy Absorption of Aluminium Honeycombs and Polymeric Foam Composites". *Proceedings of the 15th International Conference on Composite Structures* (ICCS), 15th – 17th July Porto, Portugal (2009);
- Caserta, G., Iannucci, L., Galvanetto, U. "The use of aluminium honeycombs for the improvement of motorbike helmets". *Proceedings of the 9th International Conference on Sandwich Structures (ICSS 9, Caltech)*, 14th – 16th of June, Pasadena, California (2010);

Technical reports

- 1. **Caserta, G., and Galvanetto, U.** Design of protective equipment. MYMOSA EU research training network, Report no. WP3.2a, 2010
- Ghajari, M., Caserta, G., and Galvanetto, U. Comparison of safety helmet testing standards. MYMOSA EU research training network, Report no. WP3.1, 2008

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xv Nomenclature

Nomenclature

Lower case Latin letters

a_{max}	Maximum acceleration tolerable by the head
b	Specimen width
с	Propagation of the speed of sound
$d_{\rm cell}$	Honeycomb cell diameter
g	acceleration of gravity
h	Impact tests drop height
h_0	Initial height
\mathbf{h}_{f}	EPS foam height
$h_{\rm h}$	Honeycomb height
\mathbf{h}_{s}	Two-layered material height
1	Distance between the centre of gravity of the headform and the reference
	plane
1	Honeycomb cell wall length
1	Specimen length
m	mass of the helmeted headform
m	Falling mass
m	Parameter related to development of failure in unidirectional composites
m	EPS foam material constant
р	Strain rate parameter
r	Shear strain rate ratio
\mathbf{S}_{ij}	i, $j = x, y, z$. Deviatoric stresses
t	Honeycomb cell wall thickness
\mathbf{t}_{i}	i = 1, 2, Thickness of i-th lamina in the outer shell lay-up
u	Cross-head displacement
ua	Displacement experienced by the actuator
um	Machine displacement
$\dot{u_0}$	Displacement rate under pure compressive loads
ü	Compressive displacement rate
ν̈́	Shear displacement rate
\mathbf{x}_{e}	Displacement measured in the elastic regime

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xvi Nomenclature

Upper case Latin letters

А	Load distribution area at the interface head/headform
А	Load carrying area
А	EPS foam material constant
A_0	Initial cross-sectional area
В	EPS foam material constant
С	EPS foam material constant
С	Strain rate parameter
D	EPS foam material constant
Ε	Young's modulus
$\mathbf{E}_{\mathbf{i}}$	i = T, L, W. Honeycomb Young's modulus along the tubular, longitudinal
	and transverse direction
$\dot{E_0}$	Energy absorption rate under pure compressive loads
Ė	Energy absorption rate
\overline{E}	Normalised energy absorption rate
\mathbf{F}_{m}	Maximum force at the end of elastic regime
\mathbf{F}_{\max}	Critical force
G	Shear modulus
G_{ij}	i = L,W; j = T. Honeycomb shear modulus in the ij plane
Η	Length of a honeycomb fold
${ m I}_{ m ij}$	ij = x, y, z. Moments of inertia of the headform
L	Distance travelled by trigger instrument
L_{e}	Finite element characteristic length
Ν	Compressive force
Р	Load
\mathbf{P}_0	Internal pressure
P_{max}	Maximum recorded load
R	Shear stress ratio
R	Foam relative density
R	Resultant force
\mathbf{S}	Shear force
Stt	Honeycomb plateau stress
Т	Total thickness of the laminated outer shell
V	Impact speed

xvii Nomenclature

V_{f}	Volume of EPS foam
V_h	Volume of honeycomb
$V_{\rm s}$	Volume of two-layered material
W	energy absorbed per unit volume

Lower case Greek letters

α	Angle between loading direction and honeycomb out-of-plane direction
в	Angle between loading direction and honeycomb in-plane direction
γ	Engineering shear strain
δ	Compressive displacement
3	engineering strain
$\epsilon_{\rm d}$	Densification strain
$\mathcal{E}0$	Nominal failure strain
Ė	Strain rate
v	Poisson's coefficient
$ ho_{ m f}$	EPS foam density
$ ho_{h}$	Honeycomb density
$ ho_{s}$	Two-layered materials density
σ	Engineering stress
σ	Crush stress
σ _b	Honeycomb bare compressive strength
σ_{c}	Crush strength
σ _e	Stress in the elastic regime
σ_{i}	i = T, L, W. Honeycomb crush strength along the i-th direction
σ_{i}	Composite nominal stress. $i = C, T. C = compression; T = tension$
$\widehat{\sigma}_i$	i = C, T. Effective stress in a unidirectional lamina; $C = compression; T =$
σ _{max}	Maximum stress tolerable by human head
σ 0	Plateau stress
$\sigma_{\rm ys}$	Yield stress
$\hat{ au}$	Effective shear stress in a unidirectional lamina
τ	Engineering shear stress
$ au_{ij}^{max}$	i = L, W; j = T. Honeycomb ultimate shear strength in the L-T and W-T
	planes

xviii Nomenclature

 ω_i Composite damage parameter; i = L, T, S; L = longitudinal direction; T = transverse direction; S = in-plane shear directions

Upper case Greek Letters

ΔP	Load interval measured in the elastic region
Δt	Time interval used to measure impact speed
Δt_e	Solution time step for finite element analyses
Δu	Displacement interval measured in the elastic region
Φ	Inclination of loading direction with respect to honeycomb out-of-plane direction

Acronyms and Abbreviations

ABS	Acrylonitrile-Butadiene-Styrene
AIS	Abbreviated Injury Scale
C.G.	Centre of gravity of the headform
CFC	Channel Frequency Class
EPS	Expanded Polystyrene
GRP	Glass fibre reinforced polyester resin
HIC	Head Injury Criterion
LA	Linear Acceleration
LVDT	Linear Variable Displacement Transducer
MAT20	<i>Rigid</i> material for Ls-Dyna
MAT24	Piecewise Linear Isotropic Plasticity material for Ls-Dyna
MAT58	Laminated Composite material for Ls-Dyna
MAT63	Crushable Foam material for Ls-Dyna
\mathbf{PC}	Poly-Carbonate
PET	Polyethylene-terephthalate
PLA	Peak Linear Acceleration
UD	Unidirectional
UN	United Nations
WSTC	Wayne State Tolerance Curve

1 Chapter 1 . Introduction

Chapter 1 Introduction

Motorcycle riders are among the most vulnerable road users. Statistical data showed that in the past years in various countries worldwide, although motorcycles shared a small percentage of all motorised vehicles used, motorcyclist fatalities had been accounting for a relatively high percentage of all road fatalities,. According to the European COST 327 Database (2001), which reports the analysis of 253 accidents occurred in Germany, Finland and United Kingdom from July 1995 to June 1998, motorcycles comprised only 6.1% of all motorised vehicles. Nevertheless, motorcycle fatalities comprised the 16% of the total number of road fatalities. Similar trends were observed in the United States, where from 2000 to 2002 motorcycles comprised only the 2.1% of all motorised vehicles, but motorcycle fatalities accounted for the 9% of the total road fatalities (NCSA, 2004).

According to the COST 327 (2001), 66.7% of the total motorcyclist deaths were due to the occurrence of severe or untreatable head injuries. In the same document, it is also stated that head injury severity increased quite remarkably with the head impact speed, and it is related to the shape of the struck object.

Injuries are classified through the Abbreviated Injury Scale (AIS), which is an anatomic based scoring system (Gennarelli and Wodzin, 2005) and a tool commonly used to determine the severity of single injuries. The AIS scale ranges from 0 to 6, and indicates the severity of injuries as follows:

0 - not injured; 1 - minor; 2 - moderate; 3 - serious; 4 - severe; 5 - critical; 6 - fatal

An additional AIS code of 9 is adopted to indicate those injuries that cannot be classified, for example because of insufficient information available.

With regard to the effect of the impact speed, in the COST 327 it is stated that that impact speeds below 10 km/h were likely to cause minor injuries (for example, AIS 1

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injuries occurred in 27.3% of cases, while the remaining percentage of motorcyclist did not suffer any injury). Conversely, impact speeds in the range 61-80 km/h resulted to cause the majority of severe or fatal head injuries observed. For example, the 66.3% of the total AIS 5 injuries occurred in this speed range. Similarly, the 50.7% of the total AIS 6 injuries occurred in the same speed range. Some important conclusions were made also regarding the influence of the shape of the struck object. It was found that most of the impacts (79%) occurred against a round surface, and head injuries caused by such impacts were found to be evenly distributed in the full AIS scale. On the other hand edge objects, like for example a kerbstone, were the least likely to be struck but the most likely to cause severe injuries (AIS 5 occurred with 40% probability). Flat objects were found to be struck in 9% of the total accidents and to cause moderate injuries, mainly in the range AIS 1-3.

Motorcycle helmets are currently the only protective system for the head.

Evidence of the usefulness of helmets in reducing head injuries had been shown in some previous accident reconstruction analyses. In a review of 921 motorcycle accidents, occurred in Europe during the period 1999 to 2000 (MAID, 2004), it was concluded that wearing motorbike helmets resulted in an efficient prevention or reduction of head injuries severity in approximately 68% of the total accident cases. Similar trends were confirmed in the NCSA (2004), where it was stated that between 2000 and 2002 the number of motorcycle deaths would have been reduced by 37% if all the motorcyclists wore an helmet. Because of the effectiveness of helmets in protecting the head, many countries worldwide have introduced helmet use laws in recent years, which resulted in a further reduction of head injuries severity and frequency (Kraus et al., 1994; Chiu et al., 2000; Ferrando et al., 2000; Servadei et al., 2003).

1.1 Originality and motivation

Although current helmets have been optimised to offer best protection to the wearer, more work is needed to overcome the difficulty of reducing the occurrence of motorcyclist's fatalities.

It is generally believed that the increase of the energy absorbed by helmets of 30% would reduce by 50% the occurrence of severe or fatal head injuries in case of accident (COST 327, 2001).

3 Chapter 1 . Introduction

A way to improve the protection offered to the head could be the use of non-conventional materials capable of more energy absorption than the one offered by EPS foams, while keeping the accelerations transmitted to the head at safe levels and the weight of the helmet unchanged. The solution proposed in the present thesis consists in the substitution of parts of the EPS energy absorbing liner with layers of hexagonal aluminium honeycomb.

Honeycomb materials have been used extensively in a wide range of applications as core of sandwich panels, including vehicle crash test barriers, aeronautic and space structures (Goldsmith et al., 1997; Papka and Kyriakides, 1999a,b), due to their exceptional energy absorption capabilities combined with light weight.

The peculiar characteristic of these materials is their mechanical response when subjected to pure compressive loadings along the tubular direction which consists of a progressive buckling of the cell walls. In a typical stress versus strain curve (Fig 1.1) such behaviour is represented by fluctuations of the stress around a constant value, which endure up to relatively high deformation strains (typically 80%), resulting in consistent energy absorption per unit volume levels (shaded area in Fig. 1.1).



Figure 1.1 - Energy absorbed by honeycomb subjected to compressive loadings. (from <u>www.universalmetaltek.com</u>)

Alternative solutions could have been the design of liners entirely made of aluminium honeycombs or the use of EPS foams with higher densities, with respect to current foams used for the manufacturing of helmets. However, a full honeycomb liner would not provide multi-directional protection to the head, due to the anisotropic nature of

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honeycombs and the manufacturing difficulties in giving them doubled curvature shapes. The use of negative Poisson's ratio honeycombs could be a valid alternative to overcome the problem related to double curvatures, since these materials are specifically designed to be adapted to round surfaces (Scarpa et al., 2003). However, the cost of production of such honeycombs is prohibitive for the manufacturing of innovative helmets. In addition to such disadvantages, scalp injuries or skull fracture could occur from direct contact of the honeycomb cell walls with the head in case of impact. On the other hand, the use of higher density foams would lead to higher energy absorption at expenses of the increase of the accelerations transmitted to the head.

Composites made by the combination of aluminium structures with polymeric or metallic foams have been already studied for energy absorption applications (Kavi et al, 2006; Hannsenn et al., 2000; Hannsenn et al., 2001). Kavi *et al.* (2006) tested foam-filled aluminium tubes under quasi-static compressive loadings applied along the tubular direction. Aluminium foam with density 270 kg/m³ and expanded polystyrene (EPS) foam with 32 kg/m³ density were used to fill the aluminium tubes. Same tests were performed on the tubes and the filling materials singularly. Force versus displacement curves were plotted for each material tested, and compared. From experimental outcomes, the authors observed that foam-filled aluminium tubes offered higher energy absorption levels (i.e. a larger area under the forces versus displacement curve) than the sum of those of the foam and the tubes considered alone, due to an *interaction effect* between the two materials (Figure 1.2 shows for example, the results obtained from tests on tubes filled with Al foam).



Figure 1.2 - Interaction effect in Al foam filled tube (from Kavi et al., 2004)

5 Chapter 1 . Introduction

In a similar study Hannsenn *et al.* (2000) investigated the influence of the loading speed on the mechanical response of foam filled aluminium tubes. The authors performed a set of impact compressive tests, with impact speed varying from v = 11.2 m/s to v = 22.3 m/s, and compared the mechanical response with the one obtained from quasi-static compressive tests. They observed that, under dynamic loading conditions, the energy absorption provided by such materials increased with loading speed and such increase varied from 10 to 20% with respect to the one offered when loaded quasi-statically.

It is therefore believed that the combination of honeycombs and foams as two-layered structures, which is the main subject of this thesis, can provide similar advantages to those offered by foam filled aluminium tubular structures, through for example penetration of the honeycomb cell walls in the polymeric liner and friction between the two materials. In addition to this, the solution proposed does not require excessively production costs and can be easily adapted to existing helmet manufacture.

1.2 Outline

In the next chapter, the impact behaviour of commercial motorbike helmets is discussed with emphasis on the function of its most relevant components: the inner liner and the outer shell. Some brief highlights on materials commonly adopted for the helmet manufacturing, and on the European standard regulations for testing helmets are also provided, since this methodology is the one adopted in the present investigation for the testing of helmet prototypes (chapter 6).

The mechanical properties of hexagonal aluminium honeycombs subjected to compressive loadings are reviewed in chapter 3. Particular focus is also given to the behaviour of honeycombs subjected to inclined compressive dominant loadings, since this loading condition is the most representative of impacts occurring during motorbike accidents (COST 327, 2001).

The experimental testing of two-layered materials made of EPS foams and aluminium honeycomb, which is one of the main contributions provided in the present research, is discussed in chapter 4 for potential application to helmet design. Quasi-static and impact compressive loadings were applied on three combinations of honeycomb and foams, and the energy absorbed was calculated as the area under the force versus displacement curve, up to the onset of the densification of the materials. Standard 6 Chapter 1 . Introduction

procedures were followed for the collection of materials properties, and the same tests were performed on EPS foams currently used for manufacturing of helmets and aluminium honeycomb singularly. Quasi-static standard shear tests were also performed on honeycombs for a complete characterisation of the honeycomb mechanical properties.

The material properties collected from experimental tests were later used for the validation of a finite element (FE) model representative of the two-layered materials. During this phase of the research, FE models of EPS foams and honeycomb were generated and validated singularly. A mesh convergence study was also performed to assess the effect of the mesh size on the FE outcomes (Appendix B). The two models were then used to create the two-layered materials tested in the present investigation. The techniques adopted for the modelling of the two-layered materials and the results obtained from numerical simulations are discussed in chapter 5.

On the base of the outcomes obtained from the experimental tests on two-layered materials, a prototype helmet was produced and tested following UNECE 22.05 (2002) standard regulations. In the prototypes, layers of the polymeric inner liner were substituted by layers of honeycomb in the front, top and rear area. Impacts were performed in the areas covered by honeycombs. Additional impacts were performed on the side of the helmet to assess the extent to which honeycomb materials can provide protection to the head, even when loading is applied on uncovered areas. Same impact tests were performed on unmodified helmets, presenting same geometry, dimensions and materials of the prototypes (except of the honeycomb inserts), and the dynamical responses were compared. A detailed description of the helmets, the methodology adopted and results obtained are discussed in chapter 6. The design and testing of aluminium honeycomb reinforced helmets has never been explored before, and the work described in this section is the major contribution provided by the present thesis to the current state of the art of motorcycle helmet design.

A FE model of the helmet prototype is described in chapter 7, and validated against the experimental results obtained in the present research. The model was later used to assess the influence of the honeycomb crush strength on the overall dynamic response of the prototype helmets (Appendix D). This thesis is concluded with a conclusion section (chapter 8), and recommendations for future research (chapter 9).

7 Chapter 2 . Experimental investigation of two-layered honeycomb-foam structures

Chapter 2 On helmet design and testing

Modern commercially available helmets present four main components: a stiff outer shell, an inner protective liner, a comfort pad of soft foam and fabric, and a retention system. In full-face helmets, protective foam layers are also placed in the chin and the side of the face to provide extra protection. Fig.2.1 shows a schematic section view of a full-face helmet and its principal components



Figure 2.1 - Schematic of a full-face motorcycle helmet and its main components (from UNECE 22.05, 2002)

The protective function is mainly performed by the inner liner, the outer shell and the additional polymeric paddings, while the remaining parts are designed to offer comfort and ensure a stable fitting of the helmet on the head. However, in this document major attention is given to the first two components, since they are of the highest importance for the safety of motorcycle riders.

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8 Chapter 2 . Experimental investigation of two-layered honeycomb-foam structures

2.1 The inner liner

The main function of the inner liner is the absorption of the impact energy through crushing of the cells within the material from which the liner is fabricated, usually an expanded polystyrene (EPS) foam.

Previous studies (Mills, 1991; Gilchrist and Mills, 1994; Willinger et al., 2000) have shown that there are two main load paths through which the impact force is transmitted to the head (Fig. 2.2). The first load path is through the elastic deformation of the shell in the surrounding areas of the impact, and hence to the liner (which is assumed to deform elastically) and the headform. The second load path is directed straight to the head through compression of the shell and the yielded foam (shaded area) that is below the contact area with the object impacted. The magnitude of the force transmitted to the head, and so the energy absorbed, is determined by the shape (i.e. local curvature), the material properties and the thickness of the helmet components in the impact region.



Figure 2.2 - Load transmission paths during impact on helmet (From Willinger et al., 2000)

EPS foams are preferred among the energy absorbing materials commercially available because of their capability to provide multidirectional resistance to impacts, combined

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9 Chapter 2 . Experimental investigation of two-layered honeycomb-foam structures

with light weight and relatively low costs of production (Miltz, 1990; Di Landro et al., 2002; Shuaeib et al., 2002; Gilchrist and Mills, 1994). Polystyrene foams belong to the category of closed cell foams (Gibson and Ashby, 1997) and when subjected to compressive loadings, they deform exhibiting three characteristic deformation regimes, which occur with the following sequence: linear elastic (I), plateau (II) and densification (III). Figure 2.3 shows a typical compressive stress versus strain response of closed cell foams, where the deformation regimes are also indicated.



Figure 2.3 - Typical compressive stress vs strain response of closed cell foams (from www.posterus.sk).

Linear elastic regime is short compared to the other two deformation regimes, and ends at relatively low strain values (typically 3-5 %). It consists of a nearly linear increase of stress values with strain. Previous researches based on the micromechanical analysis of closed cell foams (Gibson and Ashby, 1997), have shown that this deformation phase is controlled by the elastic bending of the polymeric cell walls. For increasing values of the strain, the foam cells start collapsing plastically at an approximately constant stress (plateau regime). The densification regime is associated with large compressive strains, when the foam cell walls completely crush and the constituent material is compressed. Such phenomenon in the stress versus strain curve is represented by a steep increase of the stress values with strain.

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10 Chapter 2 . Experimental investigation of two-layered honeycomb-foam structures

The area under the stress versus strain curve up to a given value of the strain is the energy per unit volume W dissipated by the foam in straining the material to that value (Roylance, 2001):

$$W = \int_0^\varepsilon \sigma(\varepsilon) d\varepsilon \tag{2.1}$$

where σ and ε are the engineering stress and strain. Therefore, beyond the densification regime the material is not considered as energy absorber anymore, due to the fact that further energy absorption is at the expense of the transmission of excessively high loads. Thus the maximum absorbable energy per unit volume is obtained by integrating the stress versus strain curve up to the densification strain.

Experimental data available in literature indicated that the amount of energy absorbable by EPS foam materials can be regulated either by varying the density of the foam (Gibson and Ashby, 1997; Di Landro et al., 2002; Doroudiani et al. 2003a,b; Bosch, 2010) or their thickness (Lye et al., 1998). However, the maximum thickness of the liner in current helmets is limited to approximately 40mm, so not to compromise aerodynamic performances and comfort provided to the wearer. On the other hand, the maximum EPS density is limited by weight requirements and, as explained later, by maximum accelerations transmittable to the head. The density of EPS foams can be controlled by varying the foaming processing conditions (Doroudiani et al., 2003a). It is known that Young's modulus and crush strength increase with foam density (Gibson and Ashby, 1997; Di Landro et al, 2002; Doroudiani et al., 2003b).

Di Landro et al. (2002) performed dynamic compressive tests on different EPS foam densities typically adopted for the production of helmet liners. Impact loadings were applied by dropping a flat indenter on to representative cubic samples from a height so that the impact speed was approximately equal to 2 m/s. The forces experienced by the anvil versus its vertical displacement were plotted and compared (Fig 2.4).

As it can be noted, the higher the density the higher the load required to crush the material (thus the energy absorbable during the plateau regime), but the earlier the occurrence of the densification regime. Note also that the initial slope of the curve increases with density.

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Figure 2.4 - Effect of density on EPS foams compressive response (from Di Landro et al., 2002)

Similar trends were observed by Doroudiani et al. (2003b), who tested a wide range of EPS foam densities, ranging from 33 kg/m³ to 624 kg/m³, under impact loading conditions. From experimental outcomes it was observed that the impact strength increased remarkably with density. For example, foams densities equal to 150 kg/m³ provided impact crush strength equivalent to those offered by the bulk polystyrene (1040 kg/m³ density) from which the foam was fabricated and tested under the same conditions. For densities equal to 624 kg/m³, the foam crush strength was double the bulk polystyrene strength.

In helmet design, EPS liners are generally chosen to meet standard impact test requirements (of which some highlights are reported in section 2.3). In general, the materials should be able to absorb the whole impact energy mgh (m = mass of the helmeted headform, in kg; g = acceleration of gravity; h = drop height prescribed by standards, in m), while keeping the accelerations transmitted to the headform at safe levels. The mass of the helmeted headform vary depending on the size of the helmet to be tested, and on average is equal to 5 kg. In current standard tests, the impact energy is of the order of 150 J (Snell, 2005; UNECE 22.05) and maximum thresholds for the accelerations vary in magnitude between different standards. Such limits are based on decades of studies on the biomechanics of the head, which established maximum tolerances of the human head to accelerations and their duration. Such information were gathered from testing on live primates and human cadavers (Nahum et al. 1997).

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12 Chapter 2 . Experimental investigation of two-layered honeycomb-foam structures

It is generally agreed that the human head can sustain maximum linear accelerations ranging from 250g to 300g (Newmann et al., 2000; Shuaeib et al., 2002a) without suffering severe or fatal injuries, provided such accelerations are applied for a few milliseconds. In the COST 327 (2001) for example, from an analysis based on the reconstruction of 21 accident cases it was concluded that maximum accelerations equal to 260g are likely to cause moderate head injuries (AIS 3). Further details regarding the effect of accelerations on the human head, and some of the head injury predictors commonly adopted in the literature are discussed in Appendix A.

Critical forces related to maximum accelerations can be simply calculated as $F_{max} = ma_{max}$ (Gibson and Ashby, 1997; Shuaeib et al., 2002). The maximum stress is therefore calculated as $\sigma_{max} = F_{max}/A$, where A is the load distribution area at the interface liner/headform. Past researches (Gibson and Ashby, 1997; Gilchrist and Mills, 2001; Shuaeib et al., 2002) have established that the contact area between the helmet and the head can be reasonably assumed to be equal to 0.01 m². As result, the average stress that must be sustained by the foam is of the order of 1 MPa.

Fig. 2.5 shows an example of foam selection for helmet liners (Avalle et al. 2001). The diagram shows the compressive stress versus strain response of three different foam densities, which could be potentially adopted for the manufacturing of a motorcycle helmet. It is assumed that the shaded area under the curves represents the impact energy to be absorbed during a standard drop test. As it can be noted, if the density is excessively low (ρ_1), densification could be reached and high forces would be transmitted to the head. Conversely, if the density is too high (ρ_3), critical forces could be reached even prior to complete exhaustion of the energy absorbing capabilities of the material (i.e. excessively high loads are transmitted while the foam is still crushing).

The best foam for the stated impact energy input is the foam with an intermediate density ρ_2 , which is able to provide maximum energy absorption while keeping loads below critical values.

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Figure 2.5 - Energy per unit volume absorbed by foams with different densities (adapted from Avalle et al., 2001)

In general, the densities currently adopted for helmet liners fit in the range of 50-60 kg/m³. However, in modern helmet design it is common to insert softer foam layers (density generally of the order of 30 kg/m³) to compensate the excessive stiffness of the outer shell in some regions of the helmet, such as the crown (Gilchrist and Mills, 1994).

2.2 The outer shell

The main functions of the external shell are the protection of the head from penetration of sharp objects and the distribution of the impact load along a wider surface, which also results in an increase of the energy absorption capability offered by the underlying polymeric liner. Two main types of shells are currently used for motorbike manufacturing, which provide remarkably different mechanical properties:

- Thermoplastic shells, typically made of Polycarbonate (PC) or acrylonitrilebutadiene-styrene copolymer (ABS);
- Composite shells, typically made of glass fibre reinforced polyester resins (GRP), or Kevlar fibres or carbon fibres reinforced epoxy resins;

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In general, thermoplastic shells are used for low-cost helmet manufacturing. They are viscoelastic isotropic materials and are highly deformable. However, they do not yield when undergo high loads and only deform exhibiting non-linear behaviour, so that little energy absorption is provided by such materials. In a previous study on large strain cyclic deformation of PC thermoplastics (Rabinowitz and Beardmore, 1974), it was observed that under cyclic stress loadings, a hysteresis loop developed (Fig. 2.6).



Figure 2.6 - Cyclic stress-strain curve for polycarbonate (From Rabinowitz and Beardmore, 1974)

As stated in section 2.1, the area inside the curve represents the energy absorbed per unit volume of the material, and it is due to the viscoelastic behaviour of the material itself. As it can be noted, once maximum stress is reached, the unloading of the material occurs following a pattern which is very close to the loading counterpart. This suggests that most of the input energy is stored elastically during the loading phase and then returned to the system during the unloading phase (area underneath the unloading curve).

For motorbike helmets, when impact occurs against flat surfaces this material characteristic results in a localised non-linear elastic bending and stretching of the shell material next to the impact site, and high helmet rebound velocities (Ghajari, 2010; Gilchrist, 1994). Impacts against round surfaces are generally considered more severe due to the concentration of the impact load on a relatively restricted area of the helmet, and ABS shells exhibit buckling (Gilchrist and Mills, 1994). According to the authors,

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impacts against convex surfaces can be considered equivalent to a single fatigue cycle. Due to the fact that thermoplastic shells are highly deformable, the load distribution is mainly determined by the shape of the impacted object. To compensate the low energy absorption provided by thermoplastic shells, in helmet manufacturing it is common to adopt high density EPS liners which are, as explained earlier, capable of higher energy absorption than their lower density equivalents.

In recent years, composite materials have been preferred to thermoplastics because of their ability to preserve the helmet integrity even when extensive damage is sustained (Kostopoulos et al., 2002; Aiello et al., 2007). In comparison to thermoplastic shells, composite shells can also absorb large amounts of impact energy through fibre breaking, delamination and matrix cracking (up to 30% of the total impact energy), and spread the impact loading along a larger area. Fig. 2.7 shows a typical stress versus strain curve for a fibre composite subjected to flexure loadings. The shaded area represents the energy returned to the system, which is small compared by the energy absorbed through elastic deformation (initial slope) and delamination. In helmet impacts this characteristic results in lower rebound velocities compared to the ones observed from similar impacts on ABS helmet shells (Gilchrist and Mills, 1994). However, one of the main drawbacks of using composite shells is due to the fact that if delamination does not occur, other failure mechanisms occur only at relatively high loads (Kostoupoulos et al., 2002), which might results in the transmission of high forces to the head.



Figure 2.7 - Stress-strain curve for a fibre composite in flexure (from Gilchrist and Mills, 1994)

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2.3 Helmet standard testing

2.3.1 Introduction

The effectiveness of motorcycle helmets is currently assessed through sets of standard tests specifically defined to reproduce load accident conditions and to verify the functionality of helmet components. Worldwide, different countries have defined their own regulations and among them the United Kingdom, Australia, United States, Japan and European Union provided some of the most selective standards currently available.

All the standard assessment procedures are formulated on the base of the following principles:

- The helmet shall absorb the impact energy;
- The helmet shall protect the head from penetration of sharp objects;
- The helmet must remain efficiently fastened to the head in case of accident;

Tests aiming to assess the impact absorption properties consist of dropping a helmeted headform from a specific height onto a rigid anvil. Directions provided by the standards generally include the apparatus to be used, the positioning of the helmet during the impact, the shape of the surfaces to be impacted, the drop height and the environmental conditions (i.e. in terms of humidity and temperature) under which the helmets shall be tested. The dimensions and weight of the headform, which is usually metallic, may vary depending on the size of the helmet tested. The resultant accelerations transmitted to the centre of gravity (C.G.) of the headform during impacts are measured and used as evaluation criteria. Helmets are considered safe if the maximum recorded acceleration remains below a prescribed maximum threshold, which varies depending on of different standards. Such parameter is often referred in literature to as to *Peak Linear Acceleration* (PLA). Some regulations include also restrictions about the duration of the accelerations (FVMSS 218, Snell 2010), or prescribe a maximum threshold for the Head Injury Criterion (HIC) which is a worldwide accepted injury predictor. The HIC is defined as

$$H.I.C. = \left[\frac{1}{t_2 - t_1} \int_{t_2}^{t_1} a(t) dt\right]^{2.5} (t_2 - t_1)$$
(2.2)

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where a(t) is the acceleration at the time t measured in g, t_1 and t_2 are the times (in seconds) of beginning and ending of the interval chosen in such a way to make the HIC maximum. Further details regarding head injury indexes are provided in Appendix A.

Table 2.1 reports the acceleration limits recommended by some of the most relevant standard procedures.

	1				
	UNECE	BS	FMVSS 218	AS/NZS1968	Snell M2010
	22.05	6658	(USA)	(Australia)	(USA)
	(UN)	(UK)			
PLA limit	275g	300g	400g	300g	300g
Acceleration	-	-	LA > 200g for not	-	LA > 200g for
duration			more than		not more than
[ms]			3ms		3ms
			LA > 150g for not		LA > 150g for
			more than 6ms		not more than
					4ms
HIC	$2400~{ m g}^{2.5}~{ m s}$	-	-	-	-
PLA: Peak Linear Acceleration; LA: Linear Acceleration; HIC: Head Injury Criterion					

Table 2.1 - Standard acceleration limits

The assessment of the resistance to penetration of sharp objects is performed through guided fall of a conical spike onto the surface of a helmeted headform, fixed on a rigid base (Fig. 2.8). Standards generally provide directions regarding the shape, dimensions and mass of the spike, the drop height and the impact site (Snell 2010, FMVSS 218). The helmet is considered efficient if the striking object does not achieve contact with the headform at any time during the impact.

Efficient fastening of the helmet to the head is assessed through tests on the chin strap. These tests include dynamic tensile tests and roll-off tests. Dynamic loads are applied through drop of a mass, connected to one end of the chin strap, from a certain height (in UNECE 22.05 a mass of 10 kg is dropped from 0.75m). The chin strap must withstand the force and its length must not increase by more than 30mm. In roll-off tests, the falling mass is connected to neck curtain of the helmet (Fig. 2.8), and left to drop from a specific height. To pass the test, the helmet should remain on the headform.

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Figure 2.8 - Standard penetration test and roll-off test (from Mlyajlma et al., 1999; UNECE 22.05)

2.3.2 The UNECE 22.05 standards

In this thesis, only the impact tests prescribed by UN standards (UNECE 22.05) are briefly described, since these tests are among the most important for the assessment of the performance of the helmet. The standard is also used in the present investigation for the testing of the helmet prototypes. In addition, UNECE 22.05 standards are currently among the most widely used worldwide (over 50 countries according to <u>www.WebBikeWorld.com</u>, 2008) and considered amongst the most discerning regulations. Further information regarding other testing methods and a comparison between standard regulations can be found in a study published during the development of the present research (Ghajari, Caserta and Galvanetto, 2008).

2.3.2.1 Apparatus

The impact test apparatus prescribed by UNECE 22.05 (Fig. 2.9) must include four main components:

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- a **rigid base** made of steel and with a mass equal to 500 kg. The base must be built in a way that its resonance frequency does not affect the results;
- a **free fall guidance system**, consisting of metallic cables, along which the helmet is dropped. Friction along the guides shall be such as the impact speed is not less than 95% of the theoretical speed;
- a mobile system, which is designed to support the helmet during the impact.
- a **rigid anvil**, mounted on the rigid base and against which the helmet is dropped;



Figure 2.9 - Example of impact test apparatus scheme (from UNECE 22.05)

2.3.2.2 Headforms

The headforms must be made of metal and their minimum resonance frequency should be not less than 3000 Hz, in order not to influence acceleration recordings. The standards prescribe five headform topologies, which differ in size (expressed in terms of head circumference) and weight, as presented in Table 2.2. Depending on the size of the headform, a set of detailed geometrical dimensions, such as the extent of the skull above the reference plane (Fig. 2.11) and the dimensions of the facial features below the reference plane, are also prescribed. However such information is not reported here, due

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to the large amount of details provided by the standards. The interested reader is referred to the UNECE 22.05 regulations for further informations.

Headform type	Size (in cm)	Weight (in kg)
А	50	3.1 ± 0.10
Ε	54	4.1 ± 0.12
1	57	4.7 ± 0.14
Μ	60	5.6 ± 0.16
0	62	6.1 ± 0.18

Table 2.2 - UNECE 22.05 standard headforms

The centre of gravity (C.G.) of the headform, where accelerometers shall be placed, must lie as nearest as possible to the point G (see Fig. 2.11) defined on the central vertical axis, at a distance l below the reference plane. Such distance ranges from 11.1mm to 13.7mm, depending on the headform size. The accelerometers must weight not more than 50 grams and must be able to withstand a maximum acceleration of 2000g without sustaining damage.

2.3.2.3 Anvils

Two anvils are used for impact tests: flat anvil and kerbstone anvil.

The flat anvil is a rigid cylinder made of steel, with diameter equal to 130mm. The kerbstone anvil is a triangular prism, as shown in the schematic in Fig. 2.10. The two sides of the kerbstone anvil must be symmetrical to the vertical axis (dashed line in Fig. 2.10), and form an angle of 105° (±3). A striking edge with a radius of 15 ± 0.5 mm connects the two inclined sides at the top of the anvil. The height and the length of the kerbstone must be not less than 50mm and 125mm respectively.

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Figure 2.10 - Anvil shapes prescribed by UNECE 22.05 standards (from Cernicchi et al., 2008).

2.3.2.4 Energy absorption tests

Prior to testing, the helmet must be exposed to the hygrometry and temperature conditions listed in Table 2.3. As can be seen, the standards prescribe three conditioning types. For each helmet type, two helmets shall be ambient conditioned, one helmet shall be heat conditioned and another shall be low temperature conditioned. Impacts shall be performed against either anvil. For the largest headform size, impacts on helmets heat conditioned shall be performed against the kerbstone anvil, while impacts on helmets conditioned at low temperature shall be performed against the flat anvil (Table 2.3).

Conditioning	Temperature	Relative	Anvils that	Exposure time
type		humidity	may be used for	
			test	
Ambient	25 °C		Flat and	At least 4 hours
			kerbstone	and not more
Heat	50 °C	65%	Kerbstone*	than 6 hours
Low	-20 °C		Flat*	
temperature				
*The indicated anvil shall be used only for the largest headform size. For smaller				
headforms, both anvils can be used				

Table 2.3	- Helmet	conditioning	types
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After conditioning, impacts shall be performed in 5 points on the surface of the helmet, which are named as follows, and indicated by standards as in Fig. 2.11:

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- B, in the front region;
- P, in the crown region;
- R, in the rear region;
- X, on the side of the headform, either left or right;
- S, on the chin area.



Figure 2.11 - Identification of impact points on the headform (adapted from UNECE 22.05).

Each set of impacts must be performed respecting the following sequence: B, X, P, R, S. The helmeted headform shall be dropped from a height such as the impact speed is equal to 7.5 m/s for all the impact points and anvils used, except the point S, for which the impact speed shall be equal to 5.5 m/s. Measurements of the impact velocity shall be made at a height ranging from 10mm to 60mm from the anvil surface.

When the kerbstone anvil is used, its orientation must be such as the striking edge forms a 45° angle with the *vertical median plane* (intended as the headform symmetry plane), for impacts on the B, P, R, and S points. For impacts on the sides (point X), the 45° angle is formed with the basic plane.

Helmets are considered capable of dissipating the impact energy if the maximum resultant acceleration recorded from the C.G. of the headform and the HIC do not exceed 275 g and 2400 g^{2.5} s respectively.

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Chapter 3 On the compressive behaviour of aluminium honeycombs

3.1 Introduction

Honeycomb materials can be defined as an array of identical prismatic cells which nest together to fill a plane (Gibson et Ashby, 1997). The shape of the honeycomb may vary depending on the required use and is usually hexagonal (Fig 3.1), squared, or circular. Currently available honeycombs can be either made of fibreglass, carbon fibre reinforced plastics, Nomex aramid paper reinforced plastics, or metals (usually aluminium).



Figure 3.1 - Aluminium honeycomb structure (from <u>www.sae.org</u>)

Different fabrication processes are currently adopted for the manufacturing of honeycombs. However, the basic and most adopted method is the *expansion process*, due to the ease in being implemented in automated development procedures. The expansion method consists in the stacking of material sheets where adhesive is printed along

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parallel straight lines (highlighted by black lines in Fig 3.2a). The distance between two consecutive lines on a sheet determines the dimension of the honeycomb cells. The stacking sequence is such as each glue line in the first sheet is glued to the space in the middle of two consecutive lines in the second sheet. In the third sheet the glue lines are aligned with the ones in the first sheet and placed between the lines of the fourth sheet, and so forth. The stacking process is repeated until a honeycomb block with desired dimensions is built. The block is then cured to activate the bonding agent (Fig. 3.2b), and prepared for expansion. Slices of honeycomb may then be cut to the desired height (T dimension in Fig. 3.3a) and then expanded. Finally, the honeycombs can be trimmed to the desired L dimension (or *double cell wall direction*) and W dimension (or *direction of expansion*).



Figure 3.2 - Schematic of honeycomb stacking sequence. a) aluminium stacking prior to curing; b) cured aluminium block



Figure 3.3 - Schematic of honeycomb structure. a) Expanded aluminium honeycomb (adapted from Doyoyo and Mohr, 2003); b) top view of honeycomb microstructure (adapted from Wilbert, 2011). t = cell wall thickness; d_{cell} = cell size; h_c = honeycomb height; l = cell wall length.

The honeycomb cell size d_{cell} is defined as the distance between two parallel honeycomb cell walls in a single cell, while the honeycomb height h_c is the length of the honeycomb along the T direction. The honeycomb density ρ_h is defined as the ratio of the weight of the honeycomb over the volume occupied by the material, assumed as solid.

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As result of the expansion process, commercial honeycombs present cell walls with doubled thickness (thick black lines in Fig. 3.3), and their mechanical behaviour is generally considered orthotropic (Gibson and Ashby, 1997; Balawi and Abot, 2008). Maximum strength and stiffness is offered when honeycombs are loaded along the alignment of the honeycomb prisms (T direction, also referred in literature to as to *out-of-plane* direction), since the honeycomb cell walls are subjected to pure compression or tension (Gibson and Ashby, 1997). When loaded along the W and L directions (also known in literature as *in-plane* directions), the honeycomb elastic stiffness and strength are in general two orders of magnitude lower than the ones offered when loaded along the T direction (Zhou and Mayer, 2002; Gibson and Ashby, 1997; Zhu and Mills, 2000). Same behaviour is observed comparing the shear in-plane behaviour (i.e., shear loadings applied in the W-L plane) with the out-of-plane shear response (shear loadings applied in the W-T and L-T planes).

The stiffness and energy absorption amount provided by honeycombs can be controlled by varying the thickness of the cell walls, the aluminium alloy and the cell size (Wu and Jiang, 1997). In general, increasing the thickness of the aluminium foil or using a stronger aluminium alloy causes an increase of the axial and bending stiffness of the honeycomb cell walls (Ashby et al., 1997), resulting in higher strength offered along any loading direction. Conversely, honeycomb presenting large cell size are weaker than honeycomb with smaller cells.

In this chapter a general overview of the mechanical response of aluminium hexagonal honeycomb, subjected to compressive loadings along different directions, is provided. Particular focus is also given to the behaviour of honeycombs subjected to compressive dominant loadings with respect to the out-of-plane direction, since this loading case is of particular interest for engineering application where protection from multidirectional impact loadings is required, such as motorcycle helmets design.

3.2 In-plane compressive response

In spite of the fact that honeycombs are mainly investigated for their out-of-plane mechanical properties, their behaviour under in-plane compressive loadings has been also extensively studied in past researches (Gibson and Ashby, 1997; Zhu and Mills, 2000; Zhou and Mayer, 2002; Said and Tan, 2008), for the representation of the in-plane

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properties of honeycomb core sandwich materials. Zhou and Mayer (2002), performed quasi-static tests on large aluminium honeycombs used for vehicle impact tests. Loads were applied along the L and W direction at a strain rate equal to 0.056 s⁻¹. From the experimental stress versus strain curves (Fig. 3.4), the authors identified three main deformation regimes for the two evaluated loading directions *linear elastic*, *plateau* and densification. During the first deformation regime, it was observed that the stress rose linearly with strain up to strain values approximately equal to 15% for the W-direction loading case, and 10% for the L-direction loading case. According to Gibson and Ashby (1997), who conducted an extended analysis of the mechanical behaviour of honeycomb structures subjected to different compressive loadings, this phase of the deformation is controlled by elastic bending of the cell walls. For further increase of the compressive strain, the honeycomb cell walls started bending plastically and at macroscopic level, collapse bands developed in the honeycomb structure (Fig. 3.5), which propagated to the surrounding cells as the compressive strain increases (plateau regime). The collapsing process continued at relatively constant stress values, until all rows were completely crushed and opposite honeycombs cell walls were in touch (Fig. 3.6). At this point the material acted nearly as a solid, resulting in a steep increase of stress values (densification regime).

The resistance offered by honeycombs in the L-direction is always higher than the one offered along the W-direction, as suggested by the stress versus strain curves plotted in Fig. 3.4, for example. Such difference is attributed to the loading of the doubled thickness cell walls. Gibson and Ashby (1997), explained this phenomenon through solution of equilibrium equations applied to a single honeycomb cell wall. They showed that when honeycombs are loaded along the W-direction, the compressive stress in the honeycomb faces with doubled thickness is negligible. Hence, the honeycomb resistance is provided only by the bending stiffness of the adjacent cell walls. Conversely, when loaded along the L-direction, the honeycomb cell walls with doubled thickness carry the compressive load, giving a consistent contribution to the overall resistance of the honeycombs.

Similar trends were observed by Zhu and Mills (1998) and Hönig and Stronge (2001), who tested aluminium honeycomb materials under quasi-static loads applied along the honeycomb in-plane directions. Fig. 3.5a shows a typical deformation sequence of honeycombs subjected to compressive loading along the L-direction. According to Zhu

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and Mills (1998), the honeycomb in-plane deformation modes consist in rotations and bending of the honeycomb cell walls. Another difference between the two main loading cases consists in the occurrence of the densification regime. As suggested by Fig. 3.4, densification occurs earlier when honeycombs are loaded along the L-direction. Such phenomenon was attributed again to the contribution of the honeycomb cell walls with doubled thickness. Fig. 3.6 shows a comparison between fully compressed honeycombs for loadings applied along W and L directions.



Figure 3.4 - In-plane honeycomb compressive stress-strain response (from Zhou and Mayer, 2002).



Figure 3.5 - Deformation of honeycomb subjected to compressive in-plane loadings; a) Compression along L; b) Compression along W (from Said and Tan, 2008)

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Figure 3.6 - Honeycomb specimens fully crushed after in-plane compression (from Zhou and Mayer, 2002). a) Loading along W-direction; b) Loading along L direction.

3.2.1 Effect of loading speed

Little information was found in literature regarding the experimental investigation of the in-plane impact response of aluminium honeycombs. Indeed, because of the low energy absorption offered along any in-plane direction, any impact test is likely to result in damaging of the testing equipment, so that most of the studies are based on use of numerical methods such as the finite element method (Honig and Stronge, 2002a,b; Ruan et al., 2003).

Ruan et al. (2003) virtually tested aluminium honeycombs under different impact compressive loadings applied along the L direction. Impact loadings were simulated through use of a planar rigid wall, moving at constant speed. Different velocities were prescribed, ranging from v = 3.5 m/s to v = 35 m/s, and force versus displacement curves experienced by the moving wall were plotted and compared with a quasi-static theoretical value obtained from an equation proposed by Gibson et al. (1997):

$$\sigma_0 = \frac{2}{3} \sigma_{ys} \left(\frac{t}{l}\right)^2 \tag{3.1}$$

where σ_0 is the honeycomb plateau stress, σ_{ys} is the yield stress of the material with which the honeycomb is made, h_c and l are the height and the length of the cell wall.

From numerical outcomes (Fig. 3.7a) the authors observed an increase of the simulated crush strength with impact speed. Such increase ranged from 8% (v = 7 m/s) to 55% (v = 35 m/s).

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Figure 3.7 - Honeycomb in-plane dynamic response. a) Force-displacement response (from Ruan et al., 2003); Stress-strain response (from Hönig and Stronge, 2002).

However, the oscillations in the force values observed in numerical outcomes (Fig. 3.7a) were not discussed. In addition to this, no information regarding the influence of the strain rate on the linear elastic regime were provided.

Such information were provided by Hönig and Stronge (2002a), who assessed the dynamic behaviour of honeycombs subjected to in-plane impact loadings through finite element analyses. In their simulations, the impact velocities ranged from v = 1 m/s to v = 30 m/s and stress-strain responses (Fig. 3.7b) were considered as evaluation criteria. From numerical outcomes it was concluded that strain rate sensitivity became more pronounced for impact speeds above 5 m/s, while dynamic responses obtained for impacts below 1 m/s can be considered equivalent to quasi-static response. The authors

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linked the oscillations in the force values to waves propagation along the honeycomb structure. One complete oscillation is due to the travelling of the impact wave from the impact surface (peak values) to the bottom of the honeycomb (minimum values). The authors also observed an increase in the initial slope of the stress-strain curve with increasing speed, suggesting that the in-plane Young's modulus increases with strain rate. The model was later validated against experimental results obtained from drop impact tests on large honeycomb specimens (Hönig and Stronge, 2002b), which confirmed the trends predicted numerically.

3.3 Out-of-plane compressive mechanical response

Literature survey indicated that the honeycombs out-of-plane behaviour has been widely investigated in past researches (Goldsmith and Sackman, 1992; Wu and Jiang, 1997; Gibson and Ashby, 1997; Zhao and Gary, 1998; Zhou and Mayer, 2002; Mohr and Doyoyo, 2003a, b; Hong et al., 2006; Hong et al., 2008) under both quasi-static and dynamic compressive loadings. Doyoyo and Mohr (2004) tested aluminium hexagonal honeycombs under quasi-static compressive loadings applied along the T direction. From the experimental stress-strain curves, the authors identified five different characteristic deformation regimes (Fig. 3.8) named as *elastic, non linear elastic, softening, crushing regime* and *densification regime*.

The elastic regime consisted in a conventional linear elastic response. For increasing values of the strain, stress values increased following non-linear trends (non linear elastic regime). In this phase of the deformation, the honeycomb structure exhibited elastic buckling of the honeycomb cell walls (Gibson and Ashby, 1997; Doyoyo and Mohr, 2004; Hong et al., 2006). The end of non-linear elastic regime was determined when the local stress in the honeycomb cell walls reached the yield point of the aluminium material with which the honeycomb is made. Such phenomenon is represented in the stress-strain response by a peak in the stress values. Afterwards, the honeycomb structure loses its loading carrying capabilities and starts collapsing plastically, (softening). The peak stress value is also known in literature as *bare compressive strength* (Gibson and Ashby, 1997), and it is commonly used for the identification of honeycomb materials in commercially available catalogues (www.hexcel.com).

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However, in sandwich structures designed for energy absorption applications this peak stress might result in the transmission of undesired high forces, so that it is common practice to slightly pre-crush the honeycomb surface prior to its use. This operation results in the complete removal of the initial peak, without changing significantly the energy absorption properties of the honeycomb.

The softening phase is characterised by a rapid drop of stress levels, and the development of buckling folds in the honeycomb structure, often at the interface between the loading surface and the honeycomb (Wu and Jiang, 1997; Gibson and Ashby, 1997). The crushing phase consists in a progressive buckling of the honeycomb cell walls (Fig. 3.9), which results in stress oscillations around a nearly constant value, named in literature as *plateau stress* or *crush strength*, and used as main parameter for the evaluation of energy absorption properties of honeycombs. In a previous study similar to the one presented by Doyoyo and Mohr (2004), Wu and Jiang (1997) discussed the axial crush behaviour of six different honeycomb materials, subjected to both quasistatic and impact compressive loadings applied along the T direction. From experimental observations the authors concluded that the waveform of the curve is due to the continuous formation (peak values) and crushing (minimum values) of the buckling folds. According to Gibson and Ashby (1997) and Zhu and Mills (2002) plastic hinges originate in proximity of the ends of each fold (see schematic in Fig. 3.9) during the collapsing process. The length of each fold (indicated by H in Fig. 3.9) is approximately equal to half the length of the cell wall *l* (Fig. 3.3b).

As the strain continues to increase, the honeycomb becomes completely folded and the densification regime occurs, indicated by a sudden and sharp increase of the stress values. In this phase, the honeycomb is fully crushed and acts as a solid material. The slope of the stress-strain curve in this region (Fig. 3.8) tends asymptotically to the elastic modulus of the bulk material with which the honeycomb is made (Goldsmith and Sackman, 1992, Hong et al. 2006). Once the densification regime is reached, the honeycomb does not act as energy absorber anymore.

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Figure 3.8 - Typical tress-strain response of metallic honeycombs subjected to out-of-plane compressive loadings (from Mohr and Doyoyo, 2004a). S_{TT} = plateau stress; ϵ_d = densification strain



Figure 3.9 - Schematic of folding mechanism of an axially loaded honeycomb cell (from Goldsmith and Sackman, 1992). D = honeycomb cell diameter, t = honeycomb cell wall thickness, P = load, H = length of the fold



Figure 3.10 - Fully compressed honeycomb subjected to out-of-plane compression (from Yamashita and Gotoh, 2005). a) top view; b) lateral view; c) detail perspective view.

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3.3.1 Effect of loading speed

Numerous experiments have established that honeycomb crush strength is loading rate dependent and proportional to impact speed (Goldsmith and Sackman, 1992; Wu and Jiang, 1997; Zhao and Gary, 1998, Zhou and Mayer, 2002). Goldsmith and Sackman (1992) fired a flat projectile on honeycomb specimens at impact velocities ranging from 20 to 28 m/s. From the experimental outcomes, the authors observed an increase of the honeycomb crush strength ranging from 30% to 50% with respect to the quasi-static equivalent. Similar trends were observed by Wu and Jiang (1997) and Zhao and Gary (1998). Wu and Jiang performed impact tests on six different honeycomb typologies. The authors confirmed that the loading rate has significant influence on the honeycomb strength and energy absorption capabilities. Zhao and Gary (1998) performed impact tests on two honeycomb typologies. Loads were applied along the out-of-plane direction and the impact speed ranged from 2 to 28 m/s. The results obtained were compared with quasi-static outcomes (Fig. 3.11). The authors observed an evident increase of the dynamic crush strength with respect to the quasi-static counterpart (40%), but little difference between crush strength values recorded at the chosen impact velocities.



Figure 3.11 - Influence of the loading rate on the honeycomb mechanical response (from Zhao and Gary, 1998).

Different trends were observed by Hong et al. (2008), who tested aluminium honeycomb under compressive loadings at impact speed ranging from 4.8 m/s to 18 m/s. From experimental outcomes, the authors observed a remarkable increase of the honeycomb crush strength with impact speed. Fig. 3.12 shows the normalised crush strength (intended as the ratio between the dynamic and static crush strength) in

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function of the impact speed. As it can be noted, the honeycomb crush strength increased with impact speed following linear trends.



Figure 3.12 - Honeycomb dynamic crush strength in function of the impact velocity (from Hong et al., 2008)

The authors concluded that the discrepancies among available results in literature might depend on the honeycomb material properties and loading conditions adopted by various researchers. It is also commonly agreed that honeycomb loading speed dependence is due to strain rate material sensitivity and lateral micro-inertia forces (Zhao and Gary, 1998; Goldsmith and Sackman, 1997), which develop during the impact and have a stabilisation effect on buckling mechanisms characterising the collapse of the honeycomb structure. However, other researchers (Zhou and Mayer, 2002) attributed strain-rate sensitivity to the effect of the air trapped within the honeycomb cells. As matter of fact, the loading rate sensitivity of aluminium honeycombs still remains an open question.

3.4 Honeycombs subjected to combined out-of-plane loading

As seen in the previous paragraph, honeycomb structures offer high resistance when loaded along their out-of-plane direction. However, in some applications such as vehicle crash tests, the honeycombs in moving or stationary barriers are often subjected to combinations of normal and shear loadings. Therefore, in recent research most attention has been given to the assessment of the mechanical behaviour of honeycombs subjected

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to compressive dominant out-of-plane loadings (Doyoyo and Mohr, 2003; Doyoyo and Mohr, 2004a; Doyoyo and Mohr 2004b; Hong et al., 2006; Hong et al. 2008; Hou et al., 2011), under both quasi-static and dynamic loading conditions.

Some studies (Doyoyo and Mohr, 2003; Doyoyo and Mohr, 2004a; Doyoyo and Mohr 2004b) have focused only on the effect of the inclination of the load with respect to the out-of-plane direction, on the overall mechanical response of the honeycombs. Other studies (Hong et al., 2006; Hong et al., 2008) explored also the influence of the in-plane orientation on the response provided by honeycombs subjected to inclined loadings (Hong et al., 2006; Hong et al., 2008).

Fig. 3.13 shows a schematic of the honeycomb loaded under compressive dominant inclined loadings.



Figure 3.13 - a) Schematic of a honeycomb specimen loaded under compressive dominant loading (please note that honeycombs are represented in sandwich configuration); b) Top view of a single honeycomb cell and in-plane loading angle. (adapted from Hong et al., 2006).

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N denotes the compressive force applied along the T direction, while S denotes the shear force applied along the in-plane direction. R is the resultant force. In the studies presented in this thesis, the out-of-plane loading angle was defined either as the angle a between R and S (Doyoyo and Mohr, 2004; Hou et al., 2011), or the angle Φ between R and N (Hong et al., 2006; Hong et al., 2008). For clarity, both the angles are indicated in Fig. 3.13a, and will be always specified in the next paragraphs. The in-plane orientation angle β was defined as the angle between the shear force S and the honeycomb Ldirection (Fig. 3.13b).

3.4.1 Effect of out-of-plane inclination

The effect of the inclination of the loading angle α with respect to the out-of-plane direction was widely investigated by Doyoyo and Mohr (2004a), who tested aluminium honeycombs under a consistent range of combinations of applied out-of-plane stress and shear stress in the L-T plane. To achieve the desired combination of shear and compressive loadings, honeycombs specimens were inclined with respect to their tubular direction, and quasi-static force (R in Figure 3.13) was applied through use of a special apparatus designed for biaxial testing of sandwich materials on the L-T plane.

The loading angle a (Fig. 3.13a) ranged from 0 degrees (equivalent to pure shear loading in the L-T plane) to 90 degrees (equivalent to pure compressive loading along T direction). From experimental outcomes (Fig. 3.14), the authors observed that normal stress-strain curves obtained from tests on inclined specimens exhibited similar characteristics to the ones obtained from tests under pure out-of-plane compressive loadings (i.e. similar shapes). As expected, the stress values decreased with the loading angle. The authors also performed two unloading-reloading cycles at large strains in each test, which in Fig. 3.14 are represented by relaxation drops of stress values. The linear elastic (elastic I) and non linear elastic (elastic II) regimes, as discussed in section 3.3, were identified in all the curves. Pictures taken at this stage of the deformation (Fig. 3.15b) showed the formation of shallow buckling patterns in the honeycomb structure. The end of the elastic regime was determined when stress values reached a peak, of which the magnitude decreased for decreasing loading angles. Afterwards, a softening phase was observed, characterised by a drop in stress values until a minimum, which did not result significantly affected by the loading angle. At this stage, the formation of a collapse band in the honeycomb microstructure (Fig. 3.15c) was observed.

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For increasing values of the normal strain, honeycomb specimens continued collapsing at nearly constant stress levels (crushing regime). Unlike the case of out-of-plane pure compression, the buckling patterns were irregular and originated in proximity of the centre of the honeycomb structure (Fig. 3.15 c-f).



Figure 3.14 - Mechanical response of aluminium honeycombs subjected to high biaxial loading angles (from Doyoyo and Mohr, 2004). a) normal stress – strain curves; b) Shear stress-strain curves



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Figure 3.15 - Deformation sequence of honeycomb subjected to biaxial loading at 80° (from Doyoyo and Mohr, 2004)

For low biaxial loading angles ($0^{\circ} < \alpha < 30^{\circ}$), a transition regime from compressive to tensile stress was observed (Fig. 3.16a). The strain values at which such transition occurred were found to be influenced by the loading angle, and the lower the angle the earlier the occurrence of the transition.

Regarding the shear mechanical response (Fig. 3.14b), it was observed that stress-strain curves presented similar shape and deformation regimes observed in normal stressstrain curves. However, a more pronounced softening was observed and it was found that the minimum stress (point c in Fig. 3.14b) was considerably dependent on the loading angle. Note that shear strength increases for decreasing loading angles. Shear stress-strain curves at low loading angles presented similar characteristic to the ones recorded from tests at high loading angles. However, at low loading angles a hardening phase was observed, characterised by an increase of the stress values, to which followed fracture. The hardening phenomenon was justified by the fact that for such low loading angles, the cell walls aligned with the L-direction are stretched rather than compressed, providing a significant contribution to the shear resistance of the honeycomb structure. Fig. 3.17 shows the typical deformation patterns observed from pure shear loadings

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(point e). As it can be noted from Fig. 3.17f, fracture occurred in proximity of the bond between the honeycomb and the loading plate (highlighted by a circle).



Figure 3.16 - Mechanical response of aluminium honeycombs subjected to low biaxial loading angles (from Doyoyo and Mohr, 2004). a) normal stress – strain curves; b) Shear stress-strain curves

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Figure 3.17 - Deformation sequence of honeycomb subjected to pure shear loading (from Doyoyo and Mohr, 2004)

3.4.2 Effect of in-plane orientation

The effect of the in-plane orientation angle was investigated experimentally by Hong et al. (2006) and Hong et al. (2008) under both quasi-static and dynamic loading conditions. Two honeycomb materials were tested (here referred as Type I and Type II) Different out-of-plane loading angles (in these studies referred to as to the angle Φ and showed in Fig. 3.3a) were also assessed. However, the authors did not provide information regarding the range of out-of-plane angles tested. As reference, only results obtained from $\Phi = 15^{\circ}$ were discussed. Quasi-static and impact loadings were applied for $\beta = 0^{\circ}$ (shear loading S aligned with the strong shear axis L), $\beta = 90^{\circ}$ (shear loading S aligned with the weak shear axis W) and $\beta = 30^{\circ}$, to assess the honeycomb when the shear component of the resultant loading is applied along a random in-plane direction. The normalised out-of plane crush strength was plotted against normalised shear strength for the given in-plane loading conditions (Fig. 3.18). Normalised stresses were obtained as the ratio of the normal and shear stresses experienced by the honeycombs subjected to inclined loadings, over the stress experienced by honeycombs subjected to pure compressive loadings. Quadratic fitting curves were also generated to provide a better visualisation of general experimental data trends. As suggested by experimental outcomes (Fig. 3.18), the authors observed an increase of the shear strength at the expenses of the normal crush strength as the load inclination became more pronounced, confirming the trends previously observed by Doyoyo and Mohr (2004). With reference to the effect of the in-plane orientation angle, maximum increase of shear strength with minimum reduction of the normal crush strength was achieved when the shear load was

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aligned along the L-axis ($\beta = 0^{\circ}$), while shear loadings aligned with the W direction ($\beta = 90^{\circ}$) resulted in minimum shear strength and maximum reduction of the normal crush strength. Results obtained from random in-plane orientation ($\beta = 30^{\circ}$) were found to be enclosed between the two limit loading angle conditions (Fig. 3.18b).



Figure 3.18 - Honeycomb normalised normal crush strength versus normalised shear strength for different in-plane orientation angles (from Hong et al., 2006); a) $\beta = 0^{\circ}$ and $\beta = 90^{\circ}$; b) $\beta = 30^{\circ}$

3.4.3 Energy absorption

The energy absorbed by honeycombs subjected to inclined loadings can be calculated as the sum of the work done by the normal load and the work done by the shear load (Hong et al., 2008). The energy absorption rate per unit crush area under combined loads can be defined as (Hong et al., 2006):

$$\dot{E} = \sigma \dot{u} + \tau \dot{v} \tag{3.2}$$

Where σ and τ are the normal and shear strengths and \dot{u} and \dot{v} are the compressive and shear displacement rates. The energy absorption rate can be normalised by the energy absorption rate under pure compressive loads, defined as

$$\dot{E_0} = \sigma_0 \dot{u_0} \tag{3.3}$$

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where σ_0 is the normal crush strength under pure compressive load and $\dot{u_0}$ is the normal displacement rate. As result of the normalisation process:

$$\bar{E} = \frac{\dot{E}}{\dot{E}_0} = \frac{\sigma \dot{u} + \tau \dot{v}}{\sigma_0 \dot{u}_0} \tag{3.4}$$

Under quasi-static loading conditions it can be assumed $\dot{u}_0 = \dot{u}$, and σ_{cr} independent from \dot{u}_0 . Thus

$$E = \frac{\sigma}{\sigma_0} (1 + Rr) \tag{3.5}$$

Where $R = \tau/\sigma_{cr}$ is the shear stress ratio and r is the shear strain rate ratio. The shear stress ratio can be also expressed in terms of the ratio of the magnitude of the shear force S to the normal force N, and so in terms of the loading angle Φ , as:

$$R = \frac{S}{N} = \tan \Phi \tag{3.6}$$

This suggests that the energy absorption depends on both the out-of-plane and the inplane orientation angles. Fig. 3.19 shows the normalized energy absorption rate versus the shear stress ratio. Least square fitted lines were also added for a better representation of the results.



Figure 3.19 - Normalised energy absorption per unit crush area for honeycombs loaded under different shear stress ratios (out-of-plane loading angles) and different in-plane orientation angles (from Hong et al., 2004)

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It was found that when honeycombs are loaded in the L-T plane (i.e. $\beta = 0^{\circ}$), the normalised energy absorption rate was higher than 1 and increased for increasing values of the shear stress ratio R. This suggests that the energy absorption rate of honeycombs subjected to inclined loadings is higher than the one provided when loaded under pure compressive loadings, for this particular loading condition. When loads were applied in the W-T plane ($\beta = 90^{\circ}$), the normalised energy absorption rates were lower than 1 and decreased for the given set of shear ratios R's. As the in-plane orientation angle decreases, such reduction becomes less pronounced, as suggested by results obtained from tests on honeycombs with $\beta = 30^{\circ}$.

Fig. 3.20 a-d show the comparison between the deformation patterns observed from tests under pure compressive loadings (Fig. 3.20a), and the ones observed from tests under inclined loads (Fig. 3.20 b-d). The shear load direction is marked in all the figures and results shown were obtained for $\Phi = 15^{\circ}$ and $\beta = 0^{\circ}$ (Fig. 3.20b), $\beta = 30^{\circ}$ (Fig. 3.20c) and $\beta = 90^{\circ}$ (Fig. 3.20d). Honeycombs loaded under inclined loadings showed inclined stacking folds (of which an example is marked by a circle in Fig. 3.20d), because of the asymmetry introduced by the presence of the shear load (Hong et al., 2006).



Figure 3.20 - Top view of crushed honeycombs under dynamic loadings at V_{imp} = 14.8 m/s. a) Honeycomb subjected to pure compressive loading ($\Phi = 0^{\circ}$); b) Honeycomb subjected to inclined loading with $\Phi = 15^{\circ}$ and $\beta = 0^{\circ}$; c) $\Phi = 15^{\circ}$, $\beta = 30^{\circ}$; d) $\Phi = 15^{\circ}$, $\beta = 90^{\circ}$

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3.4.4 Effect of loading speed

Little information regarding the behaviour of aluminium honeycombs subjected to dynamic compressive dominant loadings is currently available in literature, due to the difficulties in testing honeycombs under these loading conditions. Some recent experimental studies (Hong et al. 2008; Hou et al., 2011) suggested that crush strength increases with impact speed, while decreases with inclination angle. Hong et al.(2008) applied dynamic loadings to honeycombs for $\Phi = 15^{\circ}$ and $\beta = 0^{\circ}$, 30° , 90° . The impact velocity ranged from v = 2.5 m/s to v = 9 m/s. The authors, by comparing their results to their quasi-static counterparts (Hong et al., 2006), observed that while normal strength increased with loading speed, shear strength remained unaffected. Fig. 3.21 a-c show the normalised compressive strength and shear strength (normalisation is by the corresponding quasi-static counterparts) in function of the impact speed. The results presented in Fig. 3.21 were obtained for $\Phi = 15^{\circ}$, and $\beta = 0^{\circ}$, 30° , 90° .



Figure 3.21 - Normalised crush strength and shear strength in function of impact speed and in-plane orientation angles (from Hong et al., 2008). a) $\beta = 0^{\circ}$; $\beta = 30^{\circ}$; $\beta = 90^{\circ}$

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In a similar study, Hou et al. (2011) performed a series of dynamic shear-compression tests on aluminium honeycombs. The out-of-plane loading angle varied from 0 (pure out-of-plane compression) to 60°. Impact velocity was approximately equal to 15 m/s. From experimental outcomes, the authors observed a decrease of the normal crush strength with increasing loading angle, confirming the trends discussed by Hong et al. (2008). In addition to this, the authors noted that the initial slope and initial peak load decreased for increasing loading angles.

3.5 Conclusions

In this chapter the mechanical response of aluminium honeycombs subjected to compressive loadings was reviewed. It is generally agreed that because of their cellular structure consisting of arrays of joined hexagons, the honeycomb behaviour is strongly orthotropic. The three orthotropic axes are commonly identified in literature as the out-of-plane direction T (or the tubular direction), and the in-plane directions W (direction of expansion) and L (direction transverse to expansion). Literature survey indicated that maximum strength and resistance to quasi-static and impact loadings are offered when honeycombs are loaded along their tubular direction. Conversely, minimum resistance and stiffness is offered when honeycomb are loaded along the W or the L directions. However, it was found that the L-direction is always stronger than the W direction, due to the contribution of the doubled thickness cell walls to the overall resistance of the honeycombs. A review of the in-plane deformation mechanisms observed in literature indicated that crushing of the honeycombs occur as formation and expansion of collapse bands in the honeycomb structure. At microscopic level, it was observed that in-plane deformation modes consist of rotations and bending of the honeycomb cell walls.

With regard to the out-of-plane direction, the main crushing mechanism consists in a progressive buckling of the honeycomb cell walls, which often originates at the interface between the honeycomb and the loading surface, and propagates through the height of the honeycomb. This deformation mode occurs at nearly constant loads and endures up to high deformation levels, resulting in extended plateau regime and energy absorption per unit volume.

Experimental and numerical studies on the dynamic compression of aluminium honeycombs indicated that such materials exhibit to some extent strain rate sensitivity,

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often manifested as an increase of the crush strength and Young's modulus with increasing impact speed. Although the extent to which strain rate influences the mechanical response of aluminium honeycomb is still an open question, it is generally agreed that strain rate sensitivity of aluminium honeycombs depends on three main factors:

- Development of micro inertial forces which have a stabilising effect on the buckling of the cell structure;
- Increase of the pressure of the air trapped within the honeycomb cells
- Strain rate sensitivity of the aluminium alloy with which the honeycomb is made.

Some studies on the behaviour of aluminium honeycombs subjected to combination of normal and shear stresses in the T-W and T-L planes, indicated that such materials can still provide excellent energy absorption performances even when loaded under inclined loadings. This is due to the fact that the shear resistance compensates the loss of compressive strength as the loading inclination becomes more pronounced. Some experiments indicated that the energy absorption depends also on the direction of the shear load with respect to the in-plane direction. It was found that when the shear load was aligned with the L direction, the reduction of the normal strength was minimum and the shear strength was maximum for increasing loading angles. In this particular loading configuration, the honeycomb exhibited higher energy absorption rates than the ones offered when loaded under pure compressive out-of-plane loadings. On the other hand, when the shear load was aligned with the W direction, the honeycomb exhibited remarkably lower energy absorption rates compared to the ones related to out-of-plane compression. Case 1:21-cv-02112-CMA-SKC Document 40-8 Filed 09/15/21 USDC Colorado Page 67 of 224

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Chapter 4 Experimental investigation of two layered honeycomb-foam structures

In Chapter 3 the mechanical properties of aluminium honeycombs subjected to compressive loadings were reviewed.

The next step is the study of the coupling of hexagonal honeycomb with expanded polystyrene (EPS) foams for the potential application in motorbike helmet design. The starting point of this investigation is the assessment of the compressive properties of EPS foams currently adopted for the manufacturing of helmets, and aluminium honeycomb tested alone. Standard shear tests are also performed on honeycomb samples for a complete material characterisation. Then, foam and honeycomb are coupled as two layered structures (here generally referred to as to *configuration*) and their energy absorption properties under compressive loadings are assessed. The experimental outcomes of this study are later used for the characterisation and validation of FE models representing the material configurations tested (Chapter 5).

4.1 Introduction

As stated in Chapter 1, the substitution of parts of the helmet energy absorbing liner with layers of aluminium honeycombs could significantly improve the safety levels provided by commercially available helmets. In the present investigation, three different configurations of two-layered structures made of aluminium honeycomb and EPS foams were tested under both quasi-static and impact compressive loadings. The same tests were performed on EPS foams and aluminium honeycombs alone, each material presenting same dimensions. The objectives of these experiments were to:

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- Compare the energy absorption properties of such materials, subjected to different compressive loading conditions;
- Observe the deformation shapes of the two-layered structures during compression, and the interaction between their components;
- Collect the required input parameters for the FE modelling of EPS foams and honeycombs in Ls-Dyna environment;

4.1.1 Materials and test samples

All the materials tested under quasi-static and impact compressive loads were prismatic samples of square cross section with 50 mm x 50 mm area and 40 mm height. The dimensions of the cross sectional area were chosen in accordance to standards for the determination of compression properties of rigid cellular plastics (BS ISO 844:2007). According to these standards, the specimen base shall be either circular or squared, with a minimum area of 25 cm^2 and maximum of 230cm^2 . In the present investigation, the minimum dimensions were chosen to fit the dimensions of the indenter used for the impact tests (see section 4.2.3 and Fig. 4.5). To avoid the occurrence of size effects in the honeycomb layers, it was ensured that the chosen dimensions could also meet the standards for the determination of the compression properties of honeycomb core sandwich structures (ASTM C364 and ASTM C365). According to such regulations the length L (see Fig. 3.3a). of the honeycombs samples shall be less than eight times the height T, while the width W shall not be smaller than 50 mm and not higher than the length L. No recommendations are made for the height T, which was then chosen as to equal to the height of the two-layered materials for the intended use. The EPS foams used in this investigation were supplied by Dainese s.p.a. (Campodoro, Italy), while the aluminium honeycombs were supplied by Cellbond Composites Ltd (Huntingdon, UK). The EPS foams layers were cut from large panels, approximately 380mm (width) x 380mm (length) x 40mm (height) using a circular saw with a blade with 300mm diameter and 3 mm width. Particular attention was given to the cutting process to ensure that allowances for machining did not significantly affect the regularity of the specimens. The honeycomb layers were instead manually and accurately cut from larger honeycomb panels (400mm x 400mm area), obtained by expansion of glued aluminium sheets. Particular care was given to the cutting process not to damage the honeycomb cells or alter their shape.

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The EPS foam densities chosen for all the experiments were those currently adopted at Dainese s.p.a. for the manufacturing of motorbike helmet liners: 40, 50 and 60 kg/m³. The honeycombs used were the 5.2 Al 3003 cores produced at Cellbond composites, which presented a relative density (intended as the weight divided by the volume occupied by the honeycomb as homogenised material) $\rho_h = 80 \text{ kg/m}^3$, crush strength $\sigma_{\theta} = 1.6 \text{ MPa}$, cell diameter $d_{cell} = 6.35 \text{ mm}$, foil thickness t = 0.075 mm. The aluminium alloy used for the production of such honeycombs was the Al 3008 H18, of which the mechanical properties are publicly accessible in online material databases (www.matweb.com). As result of the expansion process, the honeycombs presented walls common to two cells with doubled thickness.

The energy absorption provided by the two-layered configurations was compared to the one offered by EPS densities 50 and 60 kg/m³. The EPS foam densities 40 and 50 kg/m³ were used as base for the two-layered materials. The density of the EPS foams used for the two-layered configurations was always lower than the density of the foam to which the materials were compared. However, the distribution of the foam and honeycomb thickness was assigned so that the two-layered structures overall density was equal to those of the foams to which they were compared, so that the weight of the helmet would not increase. The honeycomb layers were assumed as homogeneous materials, so that the volume V_s of the two-layered materials was given by the sum of the volumes of its components. Their overall density ρ_s was assumed as the weighted average of its components densities instead. Thus

$$\begin{cases} V_s = V_f + V_h \\ \rho_s = \frac{\rho_h V_h + \rho_f V_f}{V_h + V_f} \end{cases}$$
(4.1)

where the subscripts f and h refer to the foam and honeycomb materials respectively. Due to the fact that all the two-layered components presented same cross sectional area, the equations (4.1) can be rewritten in function of the material height h as

$$\begin{cases} h_s = h_f + h_h \\ \rho_s = \frac{\rho_h t_h + \rho_f t_f}{t_h + t_f} \end{cases}$$
(4.2)

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Two material configurations (listed as configuration 2 and 3 in Table 4.1) were compared to EPS 60 kg/m³ density, and one configuration (configuration 1) was compared to EPS 50 kg/m³.

Table 4.1 lists the thicknesses assigned to each foam and honeycomb layer according to Eq. 4.2, to match the overall two-layered material densities with the ones of the foams to which the materials were compared.

Configuration	Overall	EPS foam density	Honeycomb	Foam
	density	[kg/m ³] used as base	height [mm]	height
	[kg/m ³]	layer		[mm]
1	50	40	10	30
2	60	40	20	20
3	60	50	14	26

Table 4.1 – Two-layered configurations

As example, Fig 4.1a and b show the top and lateral view of configuration 1.







As stated previously, one of the purposes of this research was the observation of the interaction between honeycombs and foams under compressive loading conditions. Thus, no bonding agents or other types of constraints were used to connect the honeycombs layers to the EPS foams, to facilitate the penetration of the honeycombs themselves in the foams. It is believed that friction between the penetrating honeycomb walls and the foam material could further contribute to energy dissipation.

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4.2 Tests and apparatus description

4.2.1 Quasi-static tests

In order to investigate the quasi-static compressive behaviour of the materials, the standard compressive INSTRON machine 4505 100kN (shown in Fig 4.2) was used. The apparatus consisted of a twin column, high stiffness structural frame. A shaft perpendicular to the columns was moved along the vertical direction by two hydraulic actuators, incorporated in the columns. Two loading frames, each one mounting a circular steel plate with 150 mm diameter, were connected to the moving shaft and the upper edge of the machine, in alignment with the central axis of the apparatus. The Instron data logger, connected to a desktop computer, was used to collect and process the force and displacement signals. A high resolution camera was employed for capturing the deformation shapes of the material tested.



Figure 4.2 - Quasi-static test set up

The specimens were placed on top of the circular steel plate attached to the moving shaft (Fig. 4.3), and crushed against the fixed plate at a loading rate of 2 mm/min. The loading force was recorded using a sandwich load cell INSTRON 2518-801, which offers load capacity of 100kN and accuracy equal to 0.25% of the indicated load. The load cell was placed above the fixed plate.

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Figure 4.3 - Experiment set up

Five tests for each material were performed, hence a total of 35 tests were carried out. All the tests ended when the load reached a value of 15kN, at which all the specimens showed densification. Force-displacement curves were then plotted from the load cell recordings, and the energy absorbed and compressive properties were calculated. Details about data processing and energy calculation methodology followed during this investigation are provided in section 4.3.

4.2.2 Shear tests

The honeycomb specimens were tested under shear loading according to the plate shear test method, prescribed by standards tests for shear properties of sandwich core materials (ASTM C 273/C 273M – 07a). The plate shear test method consists in the application of a compressive or tensile load through diagonally opposite corners of the material to be tested. Therefore the technique does not produce pure shear loadings. Nonetheless, the standard suggests the minimum cross sectional dimensions of the specimens in function of their height, to minimise the effect of secondary stresses (i.e. normal stresses) that occur during the loading phase. According to standards, the width and the length of the specimen shall not be less than 3 times and 12 times the height of the core material. The height of the honeycombs tested in this investigation was 16 mm, so that the specimen cross-sectional area was 190 mm (length) x 76 mm (width). Particular care was given to the specimen preparation process not to damage the
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honeycomb cell walls, and to ensure that the standard prescriptions were respected. A digital calliper, which offered accuracy within 1% of the indicated measurement, was used to check the dimensions of the honeycomb panels in three points along any of the three directions of the cores (L, W, T, as described in section 3.1).

The INSTRON machine described in the previous paragraph was adopted, and the experiment set up was changed to reproduce the loading scheme suggested by standards. Fig 4.4a and b compare the schematic view of the shear plate testing prescribed by standards, and the test set up used during this investigation.



Figure 4.4 - Shear plate test set up; a) standard set up; b) experimental set up; t = height of the core material; L = length; b = width

The specimens were placed between two steel loading plates, of which the width was equal to the width of the honeycombs and the length equal to 360mm. The thickness of the loading plates is prescribed by standards to prevent the bending of the plates themselves during the loading of the honeycombs. According to a honeycomb materials database (<u>www.hexcel.com</u>), which reports a wide range of aluminium properties obtained from the application of the shear plates testing method, such thickness must not be less than 25mm. In this investigation, the thickness of the loading plates was

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chosen equal to 50mm. The high strength epoxy adhesive ARALDITE 2015 was used to bond the honeycombs to the loading plates. This structural paste can provide shear lap resistance higher than 15 MPa for metal-to-metal bonding (<u>www.Huntsman.com</u>) if curing times are longer than six hours. To ensure that adequate shear strength was achieved, the bonded specimens were cured at room temperature overnight. According to Hong et al. (2006), if the penetration of epoxy adhesives into the honeycomb cell walls is kept between 0.5 and 1 mm, the debonding of the honeycomb from the steel plates can be avoided, whilst minimising possible influence of the glue in the honeycomb mechanical response. For the shear testing of the honeycombs presented in this thesis, particular care was given to the glue application process, to ensure a uniform distribution of the epoxy agent as suggested by the authors.

Shear stresses were applied in the L-T and W-T planes, since the honeycomb shear resistance in these planes is a characteristic of most engineering applications (<u>www.hexcel.com</u>; Mohr and Doyoyo, 2004; Hong et al., 2006). Three specimens were tested for each loading orientation so that a total of six tests were performed. All the samples were tested at room temperature. A sampling rate of 0.5 data recordings per second was adopted. In all the tests, the loading rate was set equal to 1mm/min and the deformation stages were recorded using a digital high resolution camera. Images were taken at each displacement increase of 1mm. All the tests ended when the total failure of the honeycombs occurred, which was manifested by the detachment of the right loading plate. Please note that the chain in the set up was added for safety purposes. Force-displacement curves were then plotted from the load cell recordings and the shear material properties were obtained. Further details regarding data processing are reported in section 4.3.2.

4.2.3 Impact tests

For the experiments, the drop test set up built at Imperial College and depicted in Fig 4.5 was adopted. A metallic block with a flat circular indenter (75 mm diameter) mounted at its bottom was used to impact the specimens. The weight of the total falling mass was equal to 5 kg in all the impacts, except of impacts on EPS foam 40 kg/m³, for which the mass was reduced to 3kg. The dropping mass was left to fall along two vertical guides from approximately 3m height, so that the impact speed was close to the one prescribed by UNECE 22_05 standard tests: 7.5 m/s. The specimens were placed at

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the bottom of the two-rail guide, and aligned with the geometric centre of the anvil. A metallic block was used as base support for the specimens. The impact load was recorded using a strain gauges 32 kN load cell fitted into the indenter and positioned just above the impact head (see Fig. 4.5). The load signal was sampled at a frequency of 100 kHz, using the oscillator Sigma 60 manufactured by LDS Nicolet, and then filtered using a Channel Frequency Class (CFC) 1000 digital filter, to remove electrical noise. This particular filter offers a 3dB limit frequency equal to 1650 Hz and requires a minimum sampling frequency equal to 10 kHz. The sampling frequency used in this investigation was equal to 100 kHz. A charge amplifier was also used for the load cell. No constraints were applied to the sides of the specimens, due to the slight Poisson ratio presented by the EPS foams (Di Landro et al., 2002) and aluminium honeycombs (Gibson and Ashby, 1997). However, to avoid potential dislocations of the specimens during the contact with the anvil, an adhesive tape was applied between the bottom surface of each material and the support base, to provide a stable bonding between the two parts during the impacts. With regard to the two-layered configurations, the foam layer was always the component in contact with the base. Therefore, the adhesive was applied to the bottom surface of the foam. Five tests were performed for each twolayered material configuration, EPS foam density and honeycomb, so that a total of 35 impacts were carried out.



Figure 4.5 - Impact drop tower

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4.3 Data analysis

4.3.1 Quasi-static compressive response

The compressive force – displacement response of each specimen was determined by measuring the cross-head displacement. According to Kalidindi et al. (1997), since the loading system is subjected to the same loads applied to the material tested, the total displacement recorded by the actuator u_a is given by the sum of the true material displacement u and the displacement of the machine fixtures u_m . Therefore, in the present investigation the material displacement u can be obtained as

$$u = u_m - u_a \tag{4.3}$$

An alternative and efficient approach could have been the use of a Linear Variable Displacement Transducer (LVDT) which would have provided a more accurate measurement of the materials deflection. However, because of the lack of availability of such instrumentation, a machine compliance scatter plot (provided by Imperial College and shown in Fig. 4.6) was used to obtain the material displacement from Eq. 4.3. This curve was determined by applying the direct method proposed by Kalidindi et al. (1997), which consists in performing a quasi-static compressive test without any material between the compressive plates, and recording the forces and displacements experienced by the loading plate. The main advantage of using this method is the possibility to measure the machine displacements directly from the actuator recordings. A linear law was established to fit the experimental data, as showed in Fig 4.6 and the machine displacement u_m was therefore calculated as:

$$u_m = ax + b \tag{4.4}$$

where a = 0.0079 mm/kN, x is the load (in kN) and the term b = 0.0241 mm is representative of the initial settling compliance (Kalidindi et al. 1997).

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Figure 4.6 - Machine compliance curve

The load-displacement responses of each of the foams tested in this investigation were then used to determine the average compressive Young modulus E and the average compressive strength o_m , defined by standards (BS ISO 844:2007) as

$$E = \sigma_e \frac{h_0}{x_e}$$
 and $\sigma_m = \frac{F_m}{A_0}$ (4.5)

where F_m is the maximum force reached at the end of the elastic regime of the material in kN, A_0 is the initial cross-sectional area of the specimen in m², h_0 is the initial thickness in mm, and

$$\sigma_e = \frac{F_e}{A_0} \tag{4.6}$$

 F_e and x_e must be intended as the value of the force and the related displacement measured within the elastic zone of the curve.

The honeycomb load-displacement data were used instead to obtain the bare compressive strength o_b and the crush strength σ_c , defined in section 3.3.

4.3.2 Shear response

The machine compliance curve was also applied to determine the actual shear displacement experienced by the honeycomb layers tested under shear loadings. The force-displacement curves were then used to determine the ultimate shear strength and the shear modulus in the L-T plane and the W-T plane.

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The ultimate shear strength τ^{max} was calculated as

$$\tau^{\max} = \frac{P_{\max}}{lb} \tag{4.7}$$

Where P_{max} is the maximum recorded force on the specimen, l and b are the length and the width of the specimen (Fig. 4.4a).

The core shear modulus G was determined as

$$G = \frac{\left(\frac{\Delta P}{\Delta u}\right)h_h}{(lb)} \tag{4.8}$$

where h_h is the initial height of the honeycomb core and $\Delta P_{\Delta u}$ is the slope of the forcedisplacement curve in N/mm, measured in the range in which the engineering shear strain (defined as $\gamma = u/h_h$), assumed a value between 0.002 and 0.006. Due to the nature of the test method, the inclination of the specimens with respect to the loading axis determined the combination of applied shear and normal loads. However, in this investigation it was assumed that the shear loading acting on the honeycomb layers was equal to the resultant loading force applied by the actuators. Such assumption was justified by the fact that the inclination of the specimens with respect to the loading direction was very small, so that the compressive component of the stress was negligible compared to the shear component.

4.3.3 Impact compressive response

As stated in section 4.2.3, the force signals were filtered using a Class Frequency Channel (CFC) low pass digital filter, to remove electrical noise. This particular filter offers a 3dB limit frequency equal to 1650 Hz and requires a minimum sampling frequency equal to 10 kHz. The sampling frequency used in this investigation was equal to 100 kHz. The filtering frequency adopted was equal to the 3dB limit allowed by the filter. The results of such operation are showed in Fig 4.7, where an example of filtered impact response of EPS foams 60kg/m³ density is illustrated. It can be observed that the filtered curve still presents a series of oscillations, which were attributed to the vibrations of the loading system during the impacts, and could not be removed. Indeed, the application of lower filtering frequencies to remove such oscillations resulted in an 59 Chapter 4 . Experimental investigation of two-layered honeycomb-foam structures

excessively smooth loading history, not representative of the real dynamical response of the materials tested.



Figure 4.7 - Example of filtering applied to the impact response of EPS foam 60 kg/m³

In all the impact tests, the material displacement and the impact speed were tracked by use of high speed camera. In addition to such measurements, the deformation shapes and the penetration of the honeycombs in the foams during the testing of the two-layered configurations were observed. Fig 4.8 shows the experiment set up for the recording of the impact sequences of each test. The Phantom v12 camera was used to record the impacts. A sampling frequency of 11.000 frames per second was adopted to capture the anvil movements and material deformations, so that the time interval Δt between two consecutive frames was equal to 90.9 μ s. The acquired images were then processed via use of the Phantom 675.2 software (www.visionresearch.com), and the deformation of the materials was measured through the use of built-in tracking functions. The impact speed was calculated as the difference between the vertical distance of the anvil from the base in the last two consecutive frames prior to contact, divided by the time interval Δt . The average impact speed recorded from all the tests was equal to v = 7.55 m/s. Two halogen beam lights disposed on both sides of the camera were used to provide consistent brightness for the recordings.

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Figure 4.8 - High speed camera set up

For the evaluation of foams and honeycombs impact compressive properties, the same procedure described for the quasi-static case was adopted.

4.3.4 Energy absorbed

The energy absorbed by the specimens was determined from integration of the experimental load-displacement curves up to the displacement to which the onset of the densification regime occurred. The Riemann sum with midpoint approximation was adopted as integration method. Considering the two-layered materials, it was reasonable to assume that the onset of the densification regime occurred when the last folding of the honeycomb cell walls layer was observed (Fig. 4.18e). In the force-displacement curve of two-layered materials such phenomenon is represented by the last minimum reached by the force values in the region representing the deformation of the honeycomb which determined also the densification of the two-layered structure (Fig. 4.18). The EPS foams tested in this investigation showed densification at different values of the displacement upon their density. In addition, the passage between the plateau regime and the densification regime was smooth in all the tests, so that it was hard to establish with precision the onset of the densification regime from experimental observations. Literature survey confirmed some ambiguity in the definition of onset of densification regime for cellular solids (Li, 2006). Thus, for energy absorption

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calculation, it was assumed that densification occurred when the compressive load was equal to the densification load value of the two-layered material to which the foams were compared. Fig 4.9 shows an example of curve integration for EPS foam 50 kg/m³ and configuration 1, loaded quasi-statically. The region marked with small crosses correspond to the energy absorption provided by the two-layered configuration, while the area highlighted in yellow corresponds to the energy absorbed by the foam.



Figure 4.9 - Energy absorption calculation. Quasi-static case example

For the honeycombs tested alone, the onset of the densification regime was assumed coincident with the last minimum value of the force at the end of the plateau regime.

4.4 Results

4.4.1 EPS foams

Fig 4.10 shows the typical compression force-displacement curves obtained from experiments on EPS foams. As it can be seen from the graph, the curves exhibit the three distinguished regions mentioned in section 2.1: linear elastic regime (a), plateau regime (b) and densification (c). In the elastic regime, the load rises linearly with the displacement up to approximately 5% of the initial thickness. It can be noted that in this

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phase the slope of the curve is influenced by the density of the foam, and the higher the density, the higher the slope. Such trends are in accordance with existing experimental results (Gibson and Ashby, 1997; Di Landro et al., 2002). In the same way, the crushing load increased with the density of the material, at expenses of a shorter duration of the plateau regime, in accordance to what observed by Saha et al. (2006).



Figure 4.10 - Quasi-static load versus displacement response of EPS foams

Figs 4.11 a - c illustrate the effect of the loading speed on the compressive behaviour of EPS foams tested alone. As it can be noted, EPS foams compressed at an impact speed equal to 7.5 m/s showed an increase of the crush strength and an earlier occurrence of the densification regime, with respect to the static case (continuous line). Little variation of the initial slope, and so Young's modulus, of the impact curve was also observed. These results are in line with those reported in literature (Di Landro et al., 2002; Gibson and Ashby, 1997; Saha et al., 2005; Ouellet et al., 2006; Bosch, 2006), and are attributed to strain rate effects.

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Figure 4.11 – Loading speed effect on EPS foams; a) EPS 40 kg/m³; b) EPS 50 kg/m³; c) EPS 60 kg/m³

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Table 4.2 reports the average Young's modulus and compressive strength of the EPS foams, calculated according to Eq. 4.5 for both the quasi-static and dynamic loading case, and used for the characterisation of the foam finite element models created during this investigation (Chapter 5 and 7). The table includes also, the calculated standard deviation and coefficient of variation. As it can be seen, the Young's modulus was not significantly affected by the impact speed, and the observed increases with respect to the static case were within 5-10 %. On the other hand, the crush strength was consistently affected by the dynamic loading conditions (up to 62.2% increase).

	EPS 40 kg/m ³		EPS 50 kg/m ³		EPS 60 kg/m ³	
	Static	Impact	Static	Impact	Static	Impact
Young Modulus [MPa]	15.00	16.5	25.2	27.1	34.4	36.3
Standard deviation	0.5	0.7	0.450	0.6	0.3	0.43
Coefficient of variation %	0.03	0.04	0.017	0.014	0.008	0.012
Compressive strength [MPa]	0.22	0.352	0.328	0.532	0.44	0.677
Standard deviation	0.04	0.72	0.07	0.8	0.06	0.77
Coefficient of variation %	0.19	1.98	0.22	1.5	0.13	1.15

Table 4.2 - EPS foam compression properties

4.4.2 Aluminium honeycombs – compressive tests

Fig. 4.12 illustrates the averaged load-displacement curve of the honeycomb specimens subjected to pure out-of plane compressive loading (loading along T direction). The general shape of the curve shown in Fig. 4.12 is in agreement with existing experimental results (Hong et al., 2006; Wu and Wu-Shung, 1997; Zhou and Mayer, 2002). As it can be seen, initially the load increased sharply with the displacement up to a peak value, approximately equal to 9.5kN. During this compressive phase the honeycomb cell walls buckled elastically. As the displacement further increased, the bottom edges of the honeycomb cell walls started folding plastically, which resulted in a sudden drop of the loading force, as indicated in Fig. 4.12. Afterwards, the honeycomb collapsed plastically by a progressive plastic-buckling of the cell walls. The collapsing wave front always moved from the bottom edge upwards in all the tests. The plastic collapse of the honeycomb is represented in Fig.4.12 by a regular fluctuation of the load around a nearly constant value, which in average was equal to 3.95 kN. Finally, when the honeycomb cell walls were completely folded, the specimens acted as a solid material and the load increased steeply due to densification.

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Figure 4.12 - Out of plane compressive response of aluminium honeycomb

Fig. 4.13 and Fig. 4.14 show the representative load-displacement curves of the honeycombs subjected to mono-axial in-plane compressive loadings (loading along W and L directions).



Figure 4.13 - Mechanical response of honeycomb compressed along the Ldirection

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Figure 4.14 – Mechanical response of honeycomb compressed along the Wdirection

As it can be noted, in both cases the load increased linearly with the displacement up to approximately 2.5mm. With respect to the out of plane loading case, no peak load was observed during the linear elastic deformation regime. Afterwards, the honeycomb collapsed plastically at a nearly constant loading force, until densification. In this phase, in both the loading cases it was observed the formation of localised collapsing bands in proximity of the central rows of honeycomb cells in the L-W plane, and the subsequent extension to the adjacent cells. However, when loaded along the L direction, the honeycombs offered an increase of the crush strength approximately equal to 25% and a shorter duration of the plateau regime, with respect to the loading along the W direction case. Such difference was linked to the contribution of the doubled thickness cell walls to the overall stiffness of the honeycomb, when loaded along L direction.

Comparing Fig. 4.12 with Figs. 4.13 - 4.14 and referring to Table 4.3, it can be observed that the in-plane crush strength is approximately two orders of magnitude lower than the one evaluated from the out-of-plane loading case, in accordance to what observed in literature (Lamb, 2007; Zhou and Mayer, 2002).

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Fig. 4.15 shows the comparison between the out of plane quasi-static and dynamic compressive response of aluminium honeycomb. Table 4.4 lists the average crush strength and Young's modulus recorded from impacts, their standard deviation and coefficient of variation. As evident from the graph, the crushing load (average 4.3 kN) increased slightly with the loading speed. Little increase of the initial peak load (10.5 kN) was also observed during this investigation, with respect to the quasi-static case. However, due to its very short duration (0.03 msec), in post-processing phase such peak was eliminated by the application of the CFC low-pass filter. From Fig. 4.15 it can be also observed that densification regime was not reached during impacts, suggesting that the honeycombs tested in this investigation dissipated the whole impact energy.



Figure 4.15 - Effect of loading speed on aluminium honeycomb

4.4.3 Aluminium honeycombs - shear tests

Fig. 4.16 and Fig. 4.17 show the shear load-displacement response of the aluminium honeycombs tested in this investigation. In general, results from shear tests showed very similar trends and repeatability, so that only one representative force-displacement curve is provided here for each loading case. The shape of the curves is in agreement with existing results (Zhou and Mayer, 2002; Mohr and Doyoyo, 2004). Initially, in both cases, the load increased linearly with the displacement up to a peak. Pictures taken at

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this stage showed the formation of a pattern of shear superficial buckles (pictures a in Figs. 4.16 and 4.17). As the displacement increased, the shear load suddenly dropped to a minimum, approximately equal to 13.8kN for the L-T loading case, and 11kN for the T-W loading case. This phase was identified as the onset of the plastic crushing of the honeycombs, which consisted in the formation of collapse bands through the whole length of the specimens (picture b). Then the load increased again up to a maximum when the honeycombs were loaded in the L-T plane, while it remained nearly constant in the W-T loading case. Further increase of the displacement determined the crack initiation of the honeycomb cell walls and subsequent propagation of the damage (pictures c and d). Comparing Fig. 4.16d with Fig. 4.17d it can be noted that for the L-T loading case, damage developed approximately in proximity of the collapse bands, while for the W-T loading case, the fracture of the cell walls occurred in proximity of the surface next to the loading plate. In this phase of the deformation the honeycomb cell walls torn and the load values dropped gradually to nearly zero values. Afterwards, total failure of the specimen occurred, which was manifested by the detachment of the specimens from the loading plates. Table 4.3 includes the honeycomb shear properties obtained in the present investigation, following the procedure described in section 4.3.2, and the calculated standard deviations and coefficients of variation.



Figure 4.16 - Shear force-displacement curve for loading in the L-T plane

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Figure 4.17 - Shear force-displacement curve for loading in the W-T plane

Property	Average	Standard	Coefficient of
		deviation	variation %
Bare compressive strength σ_b [MPa]	3.800	0.200	0.052
Crush strength or [MPa]	1.580	0.080	0.051
Crush strength ow [MPa]	0.025	0.006	0.249
Crush strength o _L [MPa]	0.033	0.006	0.193
Young modulus E _T [MPa]	283	7.550	0.026
Young modulus Ew [MPa]	0.450	0.150	0.330
Young modulus E _L [MPa]	0.650	0.060	0.090
Ultimate shear strength L-T plane, τ_{LT}^{max} [MPa]	2.460	0.464	0.180
Ultimate shear strength W-T plane, τ_{WT}^{max} [MPa]	1.390	0.360	0.250
Shear modulus L-T plane <i>G</i> _{LT} [MPa]	3.320	0.150	0.045
Shear modulus W-T plane G _{WT} [MPa]	2.770	0.270	0.100

 Table 4.3 - Honeycomb mechanical properties (static tests)

Table 4.4 - Honeycomb mechanical	properties	(impact tests)
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Property	Average	Standard	Coefficient of	
		deviation	variation %	
Crush strength or [MPa]	1.600	0.110	0.068	
Young's modulus E _T [MPa]	290	9.220	0.031	

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4.4.4 Two-layered configurations

Fig. 4.18 shows an example of average force-displacement curve obtained from tests on configurations 2. The particular shape of the curve is due to the significant difference between the stiffness of the materials used. Indeed, from all the quasi-static tests, it was always observed that the honeycomb did not deform during the first phase of the compression, and its function was limited to the transmission of the load to the underlying foam. Initially, the load increased linearly up to 5% of the total deformation (a). As the displacement increased further, the foam began collapsing plastically at nearly constant load values. Pictures taken at this stage showed localised deformations in the upper surface of the foam (circled with red dotted line in Fig. 4.18), caused by the transmission of concentrated carrying loads by the honeycomb cell walls (b). Once the foam was completely crushed, the compressive load was then entirely carried by the honeycomb. In this transition phase, the load rose steeply up to a peak value and the honeycomb exhibited elastic buckling of the cell walls (c), while the underlying foam bottomed out. For further increase of the displacement values, the honeycomb collapsed by a progressive folding of the cell walls along the loading direction (d), in the same way as observed during experiments on honeycombs alone. Once the honeycomb was completely folded (e), the whole material acted as a solid and further compression led to a sharp increase of the load carrying values.

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Figure 4.18 - Typical configuration 2 force-displacement curve

The same behaviour was observed from all tests on configurations 1 and 3. Fig. 4.19 and Fig. 4.20 illustrate the comparison between the compressive force-displacement curves of EPS foams 50kg/m³ and 60kg/m³ (continuous lines), with those of the twolayered materials of equivalent density (dotted lines). In the graphs, the forcedisplacement curve of the aluminium honeycomb tested alone is also included. As it can be noted, the honeycomb presented the highest crush strength among all the materials tested but the shortest plateau regime. All the two-layered materials presented the longest plateau regime instead and, as expected, lower load values than the foam to which they were compared during the first phase of the compression. It must be stressed indeed that the density of the EPS foams used as base layer for the two-layered configurations was always lower than the density of the foams used for the comparison (see Table 4.1). This also resulted in lower loads required to crush the foam layer.

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Figure 4.19 - Comparison between configuration 1, EPS 50 kg/m³ and honeycomb



Figure 4.20 - Comparison between configuration 2, configuration 3, EPS 60 kg/m³ and honeycomb

Figs. 4.21 a-c show the comparison between the quasi-static and dynamic response of two-layered materials subjected to compressive loads. As it can be noted, all the forcedisplacement curves obtained from impact tests are higher in magnitude than their quasi-static counterparts, and exhibit an earlier occurrence of the densification regime. With reference to the part of the curve representing the deformation of the honeycomb, it can be observed that the average load values are generally higher than the ones exhibited by honeycombs tested alone, as depicted in Fig 4.15. This phenomenon was attributed to strain rate effects and to the interaction between the two materials. Indeed high speed camera recordings showed that during the impacts, the honeycombs penetrated slightly the underlying polystyrene foams, as can be observed from the 73 Chapter 4 . Experimental investigation of two-layered honeycomb-foam structures

sequence of photographs taken at the times 0.45, 1.8, 3.15, 4.5 msec from each set of tests, and showed in Table 4.5.



Figure 4.21 - Effect of loading speed on two-layered materials; a) Configuration 1; b) Configuration 2; c) Configuration 3

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Table 4.5 - Photographic sequence of impacts on two-layered configurations; t = time, δ = compressive displacement



4.4.5 Energy absorption

Table 4.5 resumes the energy absorption values evaluated from both the quasi-static and impact experimental outcomes, according to the procedure described in section 4.3.4. As it can be seen, the two-layered materials provided an increase of the energy absorbed with respect to the one provided by the EPS foams to which they were compared. In the table, such percentage increase is showed between brackets. It can be noted that the values ranged from 18.5% to 39.1% for the static case, and from to 22.65

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to 40.02 % for the dynamic case. It can be also noted that, the higher the honeycomb (Table 4.1), the higher the energy absorbed. The higher difference in percentage of energy absorbed by two-layered configurations under impact loadings might be due to either strain rate effects or the interaction effect between the honeycomb and the foam. From this investigation, it was not possible to determine the extent to which the interaction effect gave significant contribution to the energy absorbed.

Density	Material	Energy absorbed	Energy absorbed	
[kg/m ³]		quasi-static loads [J]	impact loads [J]	
50	EPS foam	46.11	52	
50	Configuration 1	54.66 (+18.5 %)	63.78 (+22.65 %)	
	EPS foam	55.36	63.49	
60	Configuration 2	77.01 (+ 39.1 %)	89.02 (+4 0.02%)	
	Configuration 3	68.97 (+ 24.6 %)	80.77 (+ 27.21 %)	
80	Honeycomb	115.75	134	

 Table 4.6 - Energy absorbed by the materials

4.5 Conclusions

The energy absorption properties of aluminium honeycombs and EPS foams composites subjected to quasi-static and dynamic loads were studied and compared with those of their components tested alone. The main aim of this work was to assess whether the two-layer configurations are potentially suitable for the improvement of commercially available helmets. Among the materials tested in this investigation, aluminium honeycombs provided the highest amount of energy absorbed under both quasi-static and dynamic loadings. Therefore, the use of honeycombs only for the production of innovative liners is compromised by the difficulties in giving them domed curvatures, and by their insufficient strength when loaded "in-plane". Two-layered materials were preferred as potential new energy absorbers, due to the higher energy absorption provided than the one of foams currently used for the manufacture of helmets. In addition to this, the EPS foam layer in two-layered configurations can still ensure multidirectional protection to the head against impacts, compensating the lack of in-plane resistance of honeycombs. Case 1:21-cv-02112-CMA-SKC Document 40-8 Filed 09/15/21 USDC Colorado Page 96 of 224

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4.6 Publications

The work described in the present chapter resulted in the following publication:

- Caserta, G., Iannucci, L., Galvanetto, U. "Static and Dynamic Energy Absorption of Aluminium Honeycombs and Polymeric Foam Composites". Proceedings of the 15th International Conference on Composite Structures (ICCS), 15th – 17th July Porto, Portugal (2009);
- G. Caserta, L. Iannucci, U. Galvanetto. "Static and Dynamic Energy Absorption of Aluminium Honeycombs and Polymeric Foam Composites". Mechanics of Advanced Materials and Structures, 17 (5), 2010, pp. 366-376.

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Chapter 5 Finite element modelling of two layered honeycomb-foam structures

5.1 Introduction

In Chapter 4 the compressive behaviour of aluminium honeycomb and EPS foams, combined as two-layered materials, was analysed and the experimental outcomes led to some interesting conclusions. The next step is the development and validation of a 3D FE model of the two-layered materials tested in this investigation, to be later used for the modelling of honeycomb reinforced helmets (Chapter 7).

The starting point of this chapter is the development of the EPS foam and honeycomb models separately, and their validation against the experimental results presented in Chapter 4. In this phase of the study, most attention goes to the assessment of the mesh density that provides best convergence between numerical and experimental outcomes (Appendix B), the methods adopted for the material characterisation of the models and the contact algorithms used. The two models are then merged to simulate the compressive behaviour of the two-layered materials. Validation of the models is attained through comparison of force versus displacement curves generated from numerical outputs with the experimental counterparts.

In the present analysis, all the finite element models are generated using the mesher software Hypermesh 9.0 (Hyperworks, 2008). The explicit solver Ls-Dyna v.971 (Livermore Software Technology Corporation) is used to simulate the experimental tests discussed in this thesis. 78 Chapter 5 . Finite element modelling of two layered honeycomb-foam structures

5.2 Simulation of quasi-static and impact compressive tests.

As explained in Chapter 4, quasi-static and impact compressive tests consisted in the crushing of material samples between two flat surfaces, where one surface was moving and the other one used as support. In all the FEA performed in the present investigation, a rigid planar wall was defined at the bottom surface of each model (Fig. 5.1), to simulate the support surface used during experiments. A solid cylinder of 20 mm of height, placed on top of each material model, simulated the loading anvil used during the experiments. The diameter of the cylinder was equal to 150mm for quasi-static analyses, and equal to 75mm for impact analyses, since these diameters were the ones of the anvils used for the experiments (sections 4.2.1 and 4.2.3). To generate the anvil models, 920 eight - noded solid elements of average length equal to 10 mm, were used.

The cylinders were assumed as infinitely rigid and the Ls-Dyna material card MAT_20_RIGID (Hallquist, 2007b) was adopted to model the anvil material properties. In Ls-Dyna environment, it is common to assign such material algorithm to those parts of the model which are considerably stiff or have little influence on the overall dynamical response of the simulated structures (<u>www.dynasupport.com</u>). By using MAT20 material card, rigid elements are bypassed during model processing, so that no further computational costs are required. However, it is necessary to implement realistic values of the Young's modulus E and Poisson ratio v, to avoid numerical instabilities when contact with other parts is defined (Hallquist, 2007b). For the purposes of this study, the material properties of the stainless steel alloy AISI4000 (www.matweb.com) were used ($\rho = 7850 \text{ kg/m}^3$; E = 196 GPa; v = 0.27), since this alloy is very similar to the one adopted for the manufacturing of the anvils used during experiments. To replicate the quasi-static loading rate adopted during the experiments (section 4.2.1), a homogeneous unidirectional displacement field was assigned to the anvil using the algorithm CONSTRAINED_RIGID_MOTION (Hallquist, 2007b), and the direction of motion was oriented along the height of the model (z-axis in Fig 5.1) and towards the bottom rigid wall. Virtual constraints were also applied to the anvil so that only translation along the loading direction was permitted. No restraints were applied to the material models instead, in accordance to testing conditions described in sections 4.2.1 and 4.2.3. Penalty stiffness contact algorithms were defined at the interfaces material/anvil (see section 5.3.4 for further details). A static coefficient of friction equal to 0.5 was assigned at the interface material/rigid wall.

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To reduce computational time, the simulated displacement rate was increased to 50 mm/min (25 times higher than the actual loading rate adopted for the experiments). In addition to this, non-structural mass was added to the models to increase explicit time step. It is known that in Ls-Dyna environment, computational time is strongly dependent on the number of elements, dimension of the elements and material properties (Hallquist, 2007a). Explicit time step is generally calculated for each element as follows

$$\Delta t_e = \frac{L_e}{c} \tag{5.1}$$

where L_e is a characteristic length of the element considered, and c is the propagation of the speed of sound within the element, defined as (Hallquist, 2007a):

$$\begin{cases} c = \sqrt{\frac{E(1-v)}{(1+v)(1-2v)\rho}} & for solid elements \\ c = \sqrt{\frac{E}{\rho(1-v^2)}} & for shell elements \end{cases}$$
(5.2)

where *E* is Young's modulus, *v* is Poisson's ratio and ρ is the density of the material. The definition of the characteristic length varies depending on the type of element considered (Hallquist, 2007a). The solution time step is automatically computed by Ls-Dyna as the time step of the element for which the rate (L_e/c) is minimum, in order to avoid numerical instabilities.

Mass scaling techniques are commonly accepted for solutions of quasi-static problems by use of FE explicit solvers (<u>www.dynasupport.com</u>). However, particular care must be used in the application of these techniques. Indeed, it is agreed that to keep quasi-static loading conditions, the kinetic energy must be very small relative to the peak internal energy (<u>www.dynasupport.com</u>; Mohr and Doyoyo, 2004) for the whole duration of the simulation. In the present analysis, minimum computational time while keeping quasistatic conditions (i.e. the kinetic energy was less than 0.01% of the peak internal energy) was achieved by applying a scaling factor equal to 10,000 to each material density. Further increase of the mass resulted in a non-realistic reproduction of the mechanical behaviour of the simulated materials.

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Impact compressive loadings were simulated following a different approach. The density of the anvil material was adjusted ($\rho = 22,728 \text{ kg/m}^3$) so that the overall anvil mass was equal to the mass of the drop assembly used during the experiments (5 kg, as reported in section 4.2.3). Initial velocity equal to 7.55 m/s was then assigned to all the nodes of the anvil through Ls-Dyna algorithm INITIAL_VELOCITY (Hallquist, 2007b), since this loading rate was equal to the average impact speed adopted for experiments.



Figure 5.1 - Finite element compressive loading scheme

5.2.1 Data analysis

In both quasi-static and impact analyses, forces experienced by the bottom rigid wall over time were collected in Ls-Dyna output files RWFORC (Hallquist, 2007b), and processed in Ls – Prepost software. The output interval was equal to the sampling frequency adopted during the experiments (100kHz, as reported in section 4.3.3). High frequency oscillations in the force values were then removed by applying a built-in Butterworth digital filter (<u>www.lstc.com</u>, Ls-Prepost online support), whose frequency was set equal to the one of the CFC filter during the experiments (1650 Hz, as reported in section 4.3.3). No filters were applied to quasi-static outcomes instead. The compressive displacement was computed as the vertical displacement (z-axis in Fig 5.1) of a reference node, chosen on the surface of the simulated material in contact with the loading anvil (see Fig. 5.1), and collected in Ls-Dyna output file NODOUT (Hallquist, 2007). Force versus displacement curves were then plotted for every material model, 81 Chapter 5 . Finite element modelling of two layered honeycomb-foam structures

mesh size and loading speed, and compared with the experimental results reported in this thesis (Chapter 4).

5.3 Finite element modelling of the EPS foams.

5.3.1 Mesh

The model of the foam consisted of a solid block with same dimensions as those of the EPS foam specimens tested during this investigation (50mm length x 50mm width x 40mm height). Tetrahedral four - noded solid elements were used to generate the EPS foam models. These elements are usually preferred to other element typologies due to the ease in modelling complex shapes, such as helmet liners (Cernicchi, et al., 2008). However, a mesh convergence study is recommended, due to the fact that tetrahedral elements often lead to excessive rigidity (Puso and Solberg, 2006). In the research presented in this thesis, four different mesh densities were generated to assess the element dimensions that provide best agreement between numerical and experimental results. The mesh densities were named as *coarse, medium, fine* and *extra fine*, on the base of the average dimension of each element, as listed in Table 5.2. Figures 5.2a - d illustrate a perspective view of the four EPS foam meshes generated.



Figure 5.2 - EPS foam models – a) coarse mesh; b) medium mesh; c) fine mesh; d) extra fine mesh

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5.3.2 Material properties

The three EPS foam densities $(40, 50 \text{ and } 60 \text{ kg/m}^3)$ tested in the present investigation were simulated. The Ls-Dyna material card MAT 63 CRUSHABLE FOAM (Hallquist, 2007b), specifically designed for the modelling of closed-cells isotropic foams such as EPS foams, was adopted. Under compressive loadings, the complete range of deformation stages of the foams (i.e. linear, plateau and densification regimes) is simulated through the introduction of user defined Young's modulus and stress versus volumetric strain curves. Lateral deformation is also considered through use of a user defined Poisson's ratio. However, previous studies showed that EPS foams subjected to mono-axial compressive loadings do not exhibit significant lateral deformations (Rinde, 1970; Di Landro et al. 2002) and that their average Poisson's ratio is of the order of 0.01. Thus, such value was introduced in the material card. For tensile loadings, the material is modelled as linear elastic and failure is considered when a user defined cut-off tensile stress is reached. The yielding surface of the material is defined at each time step through computation of the volumetric strain. If the magnitude of one of the principal stress components exceeds the values defined by the surface, a scaling factor is then applied so that the overcoming stress values are reduced to the yield surface. In the present investigation, the compressive behaviour of the simulated foams was defined through implementation of the experimental data reported in this thesis (Table 4.2), for both quasi-static and impact FEA. However, to avoid numerical instabilities, the stress strain curves to be introduced in the model must not present irregularities or oscillations. Thus, it was immediately clear that the experimental curves obtained in this investigation could not be directly adopted. Equivalent curves, free from oscillations, were then extrapolated from the experimental outcomes by using the closed cell foam model proposed by Gibson et al. (1997). According to the authors, the three compressive deformation regimes of foams can be adequately represented by the following equations

$$\begin{cases} \sigma = E\varepsilon + \frac{P_{0}\varepsilon}{1 - \varepsilon - R} & \text{linear elastic regime} \quad 0 < \varepsilon < \varepsilon_{y} \\ \sigma = \sigma_{y} + \frac{P_{0}\varepsilon}{1 - \varepsilon - R} & \text{plateau regime} \quad \varepsilon_{y} < \varepsilon < \varepsilon_{D} \left(1 - \frac{1}{D}\right) \\ \sigma = \sigma_{y} \frac{1}{D} \left(\frac{\varepsilon_{D}}{\varepsilon_{D} - \varepsilon}\right)^{m} + \frac{P_{0}\varepsilon}{1 - \varepsilon - R} & \text{densification regime} \quad \varepsilon > \varepsilon_{D} \left(1 - \frac{1}{D}\right) \end{cases}$$
(5.3)

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where σ and ε are the engineering stress and strain, E is the Young's modulus, σ_y is the compressive yield stress (here assumed as the compressive strength defined in section 4.3.1), ε_y is the strain value corresponding to the yield stress, ε_D is the full densification strain, P_0 is the internal initial pressure (equal to the atmospheric pressure 0.1 MPa), and R is the foam relative density defined as the ratio between the density of the foam and the density of the solid polymer with which the foam is made. D and m are constants equal to 2.3 and 1 (Gibson and Ashby, 1997; Cernicchi et al., 2008).

In addition to this model, the authors provided definitions of the Young's modulus, yield stress and full densification strain in function of the relative density R as follows:

$$E = AR^2 + BR \tag{5.4}$$

$$\sigma_{\gamma} = CR^{1.5} \tag{5.5}$$

$$\varepsilon_D = 1 - 1.4R \tag{5.6}$$

where A, B and C are material constants, that can be obtained from micromechanics analysis of the foam cells (Cernicchi et al., 2008). Nevertheless, the performance of a microscopic analysis in the present investigation would have resulted in a complicated process and because of time constraints, the least square method (Aldrich, 1998) was adopted to obtain the constants. In a previous study on the FE modelling of EPS foams for motorbike helmets subjected to UNECE 22.05 standard tests (2002), conducted by Cernicchi et al. (2008), such constants were obtained through curve fitting of experimental foam data published by Di Landro et al. (2002). Cernicchi et al. (2008) demonstrated that good agreement between experimental data and the model proposed by Gibson et al. (1997) could be obtained for $A = 6.64 \times 10^9$ Pa, $B = 2.58 \times 10^8$ Pa and C = $3.37 \ge 10^7$ Pa. In the present investigation, the least square method was applied to the outcomes listed in Table 4.2, to find A, B and C. Assuming the density of the bulk polystyrene equal to 1050 kg/m³ (www.matweb.com), from Eq. 5.4-5.6 it was obtained approximately $A = 7 \times 10^9$ Pa and $B = 2.7 \times 10^8$ Pa from both quasi-static and impact results. With regard to the constant C, the value attained from impact outcomes was approximately 33.6% higher than the one obtained from quasi-static equivalents, and 40% higher than the one suggested by Cernicchi et al. (Table 5.1). Such difference was attributed to strain rate effects. It is known that the mechanical response of EPS foams

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is strain rate independent up to strain rates of the order of $10s^{-1}$ (Dean and Read, 2001). For higher loading rates, foams exhibit higher crush strength and earlier occurrence of the densification regime (Gibson and Ashby, 1997). Experimental data published by Di Landro et al. (2002) were obtained from relatively low loading speed (2.1 m/s), while in the tests performed in the present investigation the impact speed was similar to the one prescribed by standards for testing helmets (UNECE 22.05, 2002; Snell, 2005), to which strain rates of the order of $150s^{-1}$ may occur (Brands, 1996). At these loading rates, EPS foams can exhibit an increase of the crush strength up to 50% with respect to the one offered under low rate loading conditions (Subhash et al., 2006). With regard to the coefficient D (Eq. 5.3), it was found that a reduction of 35% with respect to the value suggested in literature (D = 2.3) could adequately model the onset of the densification regime in the impact outcomes represented in Fig. 4.11.

Table 5.1 - Coefficients introduced in Gibson et al. model for the modelling of EPS foams

EPS	R	A [Pa]	B [Pa]	C [Pa]		D	
density				static	dynamic	static	dvnamic
[kg/m ³]				Statio	aynanne	Statio	aynanno
40	0.038						
50	0.047						
60	0.057	$7 \ge 10^{9}$	$2.7 \ge 10^8$	$3.54 \ge 10^8$	$4.73 \ge 10^8$	2.3	1.5
Cernicchi							
et al.		6.64×109	958 ± 108	9.97 ± 108		0.0	
(2008)		0.04 X10°	$2.00 \times 10^{\circ}$	0.07 X 10°		-	2.0

Fig 5.3 shows, as example, the comparison between the EPS 40 kg/m³ foam impact compressive stress-strain curve, reconstructed using Eq. 4.3 - 4.6, and the equivalent experimental counterpart.

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Figure 5.3 - Comparison between EPS foam 40 kg/m³ experimental compressive stress-strain curve and mathematical model proposed by Gibson et al. (1997)

5.3.3 Contact

In Ls-dyna environment, surfaces in contact are distinguished in *slave surface* and *master surface*. Such distinction originates from the fact that nodes lying on the slave surface are constrained to slide on the master surface, as soon as contact occurs. Afterwards, the slave nodes are forced to remain on the master surface until tensile force is developed. One of the methods adopted in Ls-Dyna to treat sliding and impact along interfaces, is the penalty stiffness method. When using this method, penetration of the slave nodes in the master surface is continuously checked during the whole simulation. If penetration of the slave nodes occurs, a spring element is generated between all the penetrating nodes and the master surface, on the penetration point. The interface force is proportional to the entity of the penetration, the minimum bulk modulus between the parts in contact, and the dimensions of the master element which is penetrated.

In the present investigation, the contact algorithm AUTOMATIC_SURFACE_TO_SURFACE (Hallquist, 2007b) was defined at the interface foam/anvil. By using automatic contact algorithms, distinction between master and slide surfaces is automatically performed by Ls-Dyna. Friction is modelled according to the Coulomb friction model. Static coefficient of friction was set equal to 1

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(<u>www.engineersedge.com</u>). Dynamic coefficient of friction was assumed as 1/3 of the static coefficient.

Soft penalty formulation (Hallquist, 2007a) was activated. Such formulation is specifically designed to avoid excessive penetration between parts with dissimilar stiffness, such as the simulated foams and steel materials simulated in the present investigation. This is attained through calculation of an additional stiffness, proportional to the masses of the slave and master nodes, to the initial solution time step and to a user defined scaling factor. Usually, the stiffness calculated through soft penalty method is consistently higher than the stiffness assigned through traditional penalty method. However, the two values are always checked, and the maximum between the two is assigned to the contact force. In the present analysis, it was observed that the use of a scaling factor 0.1 (default value), resulted in the lack of penetration at the foam/anvil interface.

5.3.4 Results

The compressive behaviour of three EPS foam densities was simulated. The quasistatic and impact tests described in section 4.1.1 were reproduced. Numerical loaddisplacement curves were generated for each foam density and loading speed, and compared with the experimental counterparts reported in section 4.4.1. The main purpose of this work was to assess the performances of the material card MAT63, when Eq. 5.3 - 5.6 are used for the implementation of EPS foam material properties. In addition to this, the model sensitivity to mesh size was tested. In general, good agreement between numerical and experimental results was obtained from all the tested mesh densities (Appendix B). However, the use of medium mesh provided the lowest scatter between numerical results and experimental counterparts, whilst keeping computational costs low (each simulation required an average of 5 minutes for a complete solution, when a 2 CPU computer was used). Figs 5.4 - 5.5 depict the comparison between numerical load-displacement curves and their experimental counterparts, for all the simulated foam densities and loading rates. As evident from the plots, the model followed the foam compressive response up to displacement approximately equal to 30 mm in all the cases. Afterwards, load increases following different patterns. This was associated to modelling of densification regime through use

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of Eq. 5.3, which did not perfectly match experimental data, as can be also observed in Fig 5.3.



Figure 5.4 - FEA results of the EPS foams subjected to pure quasi-static compressive loadings. a) EPS 40kg/m³; b) EPS 50kg/m³; c)EPS 60kg/m³

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Figure 5.5 - FEA results of the EPS foams subjected to pure impact compressive loadings. a) EPS 40kg/m³; b) EPS 50kg/m³; c) EPS 60kg/m³
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Although this effect is only minimum, numerical outcomes suggested that particular care should be given to the implementation of semi-empirical equations (Eq. 5.3 - 5.6) for the modelling of EPS foams.

Fig. 5.6 a-e show the numerical deformation sequence of foam 50 kg/m³ subjected to impact loadings.



Figure 5.6 - EPS finite element deformation sequence. FE impact on EPS 50 kg/m³

A comparison between deformation sequences obtained from all the mesh sizes investigated in the present research is provided in Appendix B, Table B.1

5.4 Finite element modelling of aluminium honeycombs

5.4.1 Mesh

The aluminium honeycombs were modelled as three-dimensional hexagonal cell structures using two-dimensional shell elements. The dimensions of the model and number of cells were the same as those of the specimens tested experimentally (50mm length x 50mm width x 40mm height, 63 cells). Squared 4-noded shell elements were used for the generation of the honeycombs (Hallquist, 1997a). Due to the complex and highly non-linear behaviour of hexagonal honeycombs, five through wall thickness

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integration points were assigned to each shell element (<u>www.dynasupport.com</u>). The cell wall thickness was assigned to each honeycomb element through Ls-Dyna algorithm SECTION_SHELL (Hallquist, 2007b) and was set equal to the thickness of the aluminium foils (t = 75 microns) used for the manufacturing of the honeycombs tested in this thesis. The model included also cell walls with double thickness (red walls in Fig. 5.7), to represent the glued aluminium cell walls of the specimens described in Chapter 4. Analogously to the modelling of the EPS foams, four different mesh densities were generated to perform a convergence study, aimed to find the mesh size which provides best agreement between numerical and experimental results. Each mesh density (here referred to as to *coarse, medium, fine* and *extra fine*) was defined by the element length and number of elements, as listed in Table 5.1. Fig. 5.7 depicts a schematic view of the honeycomb simulated in the present investigation, which includes also a detailed view of the four mesh densities generated.



Figure 5.7 - FE Honeycomb model

To prevent hourglass energy problems, the standard Ls-Dyna control card HOURGLASS (Hallquist, 2007b) was activated and the hourglass coefficient was set equal to 0.1 (Ls-Dyna standard value). Prior to the simulation of the compressive tests described in section 4.2, geometrical imperfections were introduced in the honeycomb models. According to different studies on the finite element modelling of the collapse behaviour of thin walled structures (Craig and Roux, 2008; Chryssanthopoulos et al.,

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1991), imperfections of geometry and material properties which may originate from the manufacturing processes must be taken into account in FE analyses, to achieve a more faithful reproduction of the compressive collapse of such structures. These imperfections can be virtually introduced in the models as geometrical distortions replicating the natural deformation modes of the structure (Mohr and Doyoyo, 2004). This methodology was adopted in the study reported in this thesis. The lowest frequency deformation modes of the honeycomb subjected to unit compressive loadings along the T-direction were obtained using the Ls-Dyna control card EIGENMODE (Hallquist, 2007b). Then, an output file including the nodal distortions representing the first deformation mode of the honeycomb was created. Finally, the nodal coordinates of the "deformed" configuration were introduced in the honeycomb model, as initial geometrical condition. For accuracy reasons, and according to what suggested by Doyoyo and Mohr (2004), the maximum displacement of the nodes from their "undeformed" configuration was scaled to the order of the cell wall thickness.

5.4.2 Material properties

The material properties of the honeycombs were modelled using the Ls-Dyna material card MAT_24_PIECEWISE_LINEAR_ISOTROPIC_PLASTICITY (Hallquist, 2007b). This material model allows the definition of arbitrary stress versus strain curve and strain rate dependency. Different stress versus volumetric strain curves for various strain rates can be introduced. Strain rate dependency is taken in to account through interpolation between curves. When stress versus strain curves are not available, it is possible to introduce in the material model arbitrary values of the yield stress σ_y , Young's modulus *E* and Poisson's ratio *v*. The yield surface is defined through the Von Mises flow rule (Kazimi, 2001):

$$\phi = \frac{1}{2} s_{ij} s_{ij} - \frac{\sigma_y^2}{3} \le 0$$
(5.7)

where s_{ij} is the deviatoric stress and σ_y is the yield stress. At each time step, the update of the deviatoric stresses is assumed as linear and the yield function is checked. If the Von Mises rule is satisfied, then the deviatoric stresses are accepted. If the yield function is not satisfied, then the overcoming stresses are scaled back to the yielding surface. For shell elements, strains normal to the mid surface of the elements are

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assumed negligible. Because the honeycombs were reproduced as array of twodimensional hexagonal cells, the material properties introduced in the model were those of the bulk aluminium alloy Al 3003 H18 ($\rho = 2730 \text{ kg/m}^3$, E = 68.9 GPa, $\sigma_y = 186 \text{ MPa}$, v = 0.33), since this alloy was the one adopted for the production of the honeycombs tested in this investigation.

Strain rate dependency is alternatively treated through use of a mathematical model proposed by Cowper-Symonds (1983). According to the authors, the strain rate sensitivity of metallic alloys can be adequately represented by the following equation:

$$\sigma = \sigma_0 \left[1 + \left(\frac{\dot{\varepsilon}}{C}\right)^{\frac{1}{p}} \right]$$
(5.8)

where σ is the dynamic stress at uniaxial strain rate $\dot{\varepsilon}$, σ_0 is the material yield stress measured at strain rate 1s⁻¹, *C* and *p* are parameters that can be obtained from experimental tensile tests. Previous studies on the strain rate sensitivity of steel alloys (Paik and Thayamballi, 2003), have shown that such model can accurately reproduce rate sensitivity at both low (10⁻⁴ s⁻¹) and high strain rates (1000 s⁻¹).

It is known that aluminium strain rate effects are also dependent on alloy (Smerd et al., 2005). For aluminium 3003 alloys, it was found that $C = 2.5 \times 10^5 \text{ s}^{-1}$ and p = 8 (Guoxing and Tongxi, 2003). In the present investigation, the formulation proposed by Cowper-Symonds was used, and the coefficients proposed by Guoxing et al. (2003) were adopted.

5.4.3 Loading conditions and contact algorithms

Quasi-static compressive loadings were applied along the three honeycombs main directions L, W and T, while impact loadings were applied along the T direction only, in accordance to what performed experimentally (Chapter 4). Force versus displacement curves were then generated following the same procedures described in section 5.2.1, and compared with the experimental counterparts reported in section 4.4.2.

Two automatic penalty stiffness contact logics were defined at the interface honeycomb/anvil:

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- AUTOMATIC_NODES_TO_SURFACE, to model the contact between the edges of the honeycomb cells and the surface of the anvil.
- AUTOMATIC_SURFACE_TO_SURFACE, to model the contact between the upper surface of the honeycomb and the anvil, during the progressive folding of the cell walls.

A third algorithm, AUTOMATIC_SINGLE_SURFACE, was defined to avoid selfpenetration of the honeycomb cell walls during the plastic deformation. This contact algorithm does not require definition of a master surface. The static coefficient of friction was set equal to 0.45, which is a typical value for dry contact between aluminium and steel (<u>www.engineersedge.com</u>). The dynamic coefficient of friction was assumed equal to 1/3 of the static coefficient. Soft penalty option was activated and the penalty scaling factor was set equal to 0.1.

The thickness of the shell was taken in to account in all contact algorithms.

5.4.4 Results and discussion

Quasi-static and impact compressive tests, described in section 4.2.1 and 4.2.3, were simulated. Quasi-static loadings were virtually applied along the three honeycomb main directions, while impact loadings were applied along the direction of the alignment of the honeycomb cell walls. The main purpose of this research was to create and validate a FE model of aluminium honeycomb to be used for the FE modelling of two layered foam-honeycomb structures (section 4.5), and honeycomb reinforced motorcycle helmets (Chapter 7). The performances of the material algorithm MAT24 were assessed with implementation of bulk aluminium 3003 alloy properties. Strain rate effects were taken into account through use of Cowper-Symonds model. Different mesh densities were adopted for the generation of the honeycomb model and it was observed that numerical results were strongly dependent on element size. Best correlation between experimental and numerical outcomes was obtained when 0.3 mm elements were used (fine mesh, Table 5.1). Further details are discussed in Appendix IV.

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Figure 5.8 - FEA results of hexagonal honeycomb subjected to in-plane quasistatic compressive loadings. a) loading along W direction; b) loading along L direction

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Figure 5.9 - FEA results of hexagonal honeycomb subjected to out-of-plane pure compressive loadings a) Quasi-static loading; b) impact loading

Force versus displacement curves obtained from quasi-static FE simulations are compared with experimental counterparts in Fig 5.8 a, b and Fig. 5.9a. Numerical impact response of honeycomb is compared with experimental results in Fig. 5.9b. As it can be observed, the numerical model generally follows the trends observed experimentally, for all the simulated loading conditions. Considering the L loading case (Fig. 5.8b), it can be however noted that forces attained from FEA followed similar

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patterns to those observed experimentally until displacement approximately equal to $\delta = 20 \text{ mm}$. Afterwards, experimental data exhibit higher values of the force and earlier occurrence of the densification regime, with respect to numerical outcomes. It was concluded that the use of fine mesh and added structural mass may have altered the stiffness and deformation modes of the honeycomb model, for the particular loading condition considered.

Fig. 5.9 a and b show FEA results of quasi-static and impact compression along T direction in comparison with experimental outcomes. Considering the quasi-static case, it can be observed that the initial peak exhibited by numerical results is considerably higher than the experimental counterpart. Such difference was attributed to the differences between the imperfections introduced in the FE model and the ones of the specimens adopted for experimental analyses. It is inferred that an increase of the scaling factor adopted for the nodal distortions could provide better convergence between numerical and experimental peak loads. After the initial peak, numerical results follow experimental trends up to displacement equal to 30mm. In this deformation phase, the honeycomb model deformed through progressive folding of the cell walls, starting from the loaded edge (Fig 5.10), in agreement with existing FE results (Doyoyo and Mohr, 2003; Yamashita, 2005).



Figure 5.10 - FE deformation sequence of the honeycomb. Impact along T

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The honeycomb crush strength (see definition given in section 3.3) was evaluated as 1.6 MPa, which is consistent with the value obtained from experimental tests (1.58 MPa). However, from Fig. 5.9a it can be noted a slight difference in the oscillations of force values. Such discrepancy was attributed to addition of large amount of non-structural mass to the model, which has influence on the propagation of perturbations within the honeycomb elements (Eq. 5.2), on the deformation modes of the structure and so the forces transmitted between the honeycomb and the bottom rigid wall. However, for the purposes of this study, such discrepancy was considered a minor issue. Indeed, such problems can be simply solved by reducing the amount of added mass, at the expense of an increase in CPU time.

With respect to the impact loading case (Fig. 5.9b), it is evident that the FE model provided a very good prediction of the observed experimental trends. Note that for both curves, the initial peak load was eliminated by signal filtering. However, from both numerical and experimental non filtered data it was observed that the initial peak loads presented approximately identical values (10.5 kN). In addition, the simulated crush strength (1.64 MPa) was in very good agreement with the one calculated from experimental data (Table 4.4). As stated already in Section 4.4.2, during experiments the honeycomb layer absorbed the whole impact energy amount prior to densification. Same results were observed from FEA. Comparing Fig 5.9a with Fig 5.9b, it could be inferred that the FE model provided better prediction of impact compressive response, than the quasi-static case.

	EPS foam		Honeycombs	
Mesh type	Number of	Average length	Number of	Average
	elements	[mm]	elements	length [mm]
Coarse	104	20	38,592	0.92
Medium	230	16	$154,\!370$	0.46
Fine	475	10	358,640	0.3
Extra fine	3022	5	617,480	0.2

Table 5.2 - Honeycomb and foam model mesh densities

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5.5 FE modelling of two-layered materials

In Section 5.3 and Section 5.4 the FE models of the EPS foams and hexagonal honeycombs were presented and validated against experimental results. From the numerical outcomes, it was observed that best agreement with experimental results was obtained when using a medium mesh size for the modelling of EPS foams, and a fine mesh size for modelling of honeycombs (Table 5.2). The two models were then combined to simulate all the two-layered configurations tested in the present investigation, and described in section 4.1.1. The dimensions of the FE models where the same as those of the experimental equivalents (50 mm length x 50 mm width x 40 mm height). The lay-up of the foam and honeycomb layers was the same as the one chosen for the experiments, and listed in Table 5.3, where the number of elements used is also indicated.

Configuration	EPS foam	Honeycomb	Foam	Number of elements	
	density	height	height		
	[kg/m ³] used	[mm]	[mm]		
	as base layer	[]	[]	shell	solid
1	40	10	30	89,660	180
2	40	20	20	179,320	117
3	50	14	26	125,524	152

Table 5.3 – FE two-layered configurations

The mechanical properties of the two-layered materials were modelled according to what reported in 5.3.25.4.2.in section and Penalty contact algorithm AUTOMATIC NODES TO SURFACE (Hallquist, 2007b) was defined at the interface honeycomb/foam and honeycomb/anvil. Soft penalty option was activated for both interfaces. Contact between the deformed honeycomb cell walls and the foam and anvil models was considered by defining AUTOMATIC_SURFACE_TO_SURFACE (Hallquist, 2007) algorithm. The self-penetration of the honeycomb cell walls during collapse was prevented by defining automatic single surface contact algorithm. As example, Fig. 5.11 shows the FE model of configuration 2.

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Figure 5.11 – Two-layered FE model.

Note that in Figure 5.11 contact algorithms are also shown. Contact parameters (i.e. coefficient of friction, penalty scale factors, etc.), element typology, number of through-thickness integration points, are reported previously in section 5.3 and section 5.4. Quasi-static and impact compressive loading conditions, described in section 4.2, were applied along the alignment of the cell walls of the honeycomb layer. Numerical force versus displacement curves were then generated and compared with the experimental data reported in section 4.4.4.

An alternative approach could be the modelling of two-layered structures as representative unit volumes, in order to reduce computational cost. In the present investigation, a unit cell model representing two-layered structures was generated and validated against the experimental results reported in this thesis. The shape of the unit cell was chosen in agreement with previous studies on the modelling of aluminium honeycombs as unit cells (Yamashita and Gotoh, 2005; Asadi et al., 2006; Shi and Tong, 1995). Fig. 5.14a and b show a schematic view of the FE unit cell model generated in this investigation

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Figure 5.12 - Unit cell model. a) top view; b) perspective view

Numerical results obtained with the unit cell model resulted in good agreement with the experimental counterparts. Further details about this study are reported in Appendix IV.

5.5.1 Results FE two-layered materials

Fig. 5.12 and Fig. 5.13 show force versus displacement curves obtained from FE simulations in comparison with experimental counterparts, for both the static and dynamic cases. As can be seen from the plots, the results obtained from numerical analyses on two-layered panels are generally in good agreement with experimental results. With respect to the numerical outcomes representing the compressive response of configuration 1, it can be noted that all the numerical curves exhibit an earlier densification regime with respect to the experimental counterparts. Such discrepancy might be due to difference of mesh size between the parts.

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Figure 5.13 - FEA results of two-layered materials subjected to pure compressive quasi-static loadings a) Configuration 1; b) Configuration 2; c) Configuration 3

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Figure 5.14 - FEA results of two-layered materials subjected to pure compressive impact loadings a) Configuration 1; b) Configuration 2; c) Configuration 3

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The numerical load curves representing the mechanical response of configurations 2 and 3 exhibit very good agreement with experimental curves. It was observed that use of contact AUTOMATIC_NODES_TO_SURFACE algorithm played a crucial role in this regard. Without use of this algorithm non-realistic transmission of forces between the parts was observed, and excessive penetration of the honeycomb material into the underlying foam material.

5.6 Conclusions

FE models of two layered foam-honeycomb composites were generated and validated in the Ls-Dyna environment against quasi-static and impact compressive tests. Generally, the results are in good agreement with those observed experimentally outcomes obtained in the present investigation. To the knowledge of the author, modelling of the interaction between honeycombs and polymeric foams is novel and original work. The mechanical properties of the foams were implemented through use of Gibson and Ashby (1997) closed cell foam model, fitted to the experimental data reported in this thesis. Numerical outcomes from simulations of quasi-static and impact compressive tests on EPS foam models resulted in good agreement with experimental outcomes, and in line with what reported in similar earlier FEA (Cernicchi et al., 2008; Ghajari, 2010). However, numerical results were found to be slightly dependent on mesh density, as confirmed by (Cernicchi et al., 2008). In the present analysis, best agreement between numerical and experimental results was obtained when using solid tetrahedral element size approximately equal to 15 mm. Such mesh size was chosen for the modelling of two-layered structures and prototype helmets described in this thesis.

The mechanical properties of the honeycomb were modelled by implementing the material properties of the bulk aluminium alloy (Al 3003 H18) used for the manufacturing of the honeycombs. Strain rate effects were taken into account through use of the Cowper-Symonds model (Kazimi, 2001). Initial imperfections were modelled through introduction of geometrical distortions replicating the lowest natural deformation mode of the honeycombs, in line with existing FE studies (Mohr and Doyoyo, 2004). Quasi-static and impact compressive simulations indicated that the FE model tends to give predictions that are in agreement with the experimental data from corresponding tests. However, some discrepancies were observed for compressive loadings along the L and T direction. Such discrepancies were attributed to mesh

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density sensitivity and to the addition of excessive non-structural mass for quasi-static analyses. In the present study, use of shell element size equal to 0.3 mm was found to provide best agreement between numerical and experimental outcomes.

The outcomes of the study presented in this chapter are later taken into account for the FE modelling of innovative helmets, where aluminium honeycomb is used as reinforcement material for the polymeric energy absorbing liner (Chapter 7). The model proposed can be also used for the design of other personal protective equipment, and applications where impact energy absorption is required.

5.7 Publications

The work presented in this chapter resulted in the following publications:

- G. Caserta, L. Iannucci, U. Galvanetto. "Static and Dynamic Energy Absorption of Aluminium Honeycombs and Polymeric Foam Composites". *Mechanics of Advanced Materials and Structures*, 17 (5), 2010, pp. 366-376
- Caserta, G., Iannucci, L., Galvanetto, U. "Micromechanics analysis applied to the modelling of aluminium honeycomb and EPS foam composites". *Proceedings of* the 7th European Ls-Dyna Users conference, 14th -16th May Salzburg, Austria (2009).

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Chapter 6 Experimental assessment of a helmet prototype

6.1 Introduction

In Chapter 4, the energy absorption properties of two-layered materials made of EPS foams and aluminium honeycomb were assessed experimentally. The next step is the application of the two-layered material concept to the production of innovative and safer helmets. In this chapter, the impact response of a modified version of the AGV Gp-Tech helmet (Fig. 6.1) manufactured by Dainese SpA (Italy, a partner of the MYMOSA network), and here referred to as *prototype*, was investigated following UNECE 22_05 (2002) standards for testing helmets. The same tests were performed on unmodified Gp-Tech helmets, presenting same dimensions, material properties and overall weight. The dynamic responses of both the helmet typologies were compared, and the recorded Peak Linear Acceleration (PLA) and Head Injury Criterion (HIC) were used as evaluation criteria.



Figure 6.1 - AGV Gp-Tech full face helmet

In addition to the assessment of the shock absorption properties, the aim of this study was the collection of validation data for the FE modelling of honeycomb reinforced helmets (Chapter 7). Case 1:21-cv-02112-CMA-SKC Document 40-8 Filed 09/15/21 USDC Colorado Page 126 of 224

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6.2 The helmet prototypes

The un-modified helmets tested in this investigation, which already meet the relevant standard, consisted of a fibre reinforced outer shell, a multi-density foam energy absorbing liner, two lateral cheek pads, a protective chin moulding pad and a retention system.

Hexagonal aluminium honeycomb layers were introduced in the helmet prototypes to enhance the energy absorption properties offered by the polymeric inner liner, and so improve the protection of the head against impacts. The honeycombs were inserted in the front, top and rear surfaces of the liner, as showed in Fig. 6.2 a-c. Recesses were created in the prototype liner to accommodate the honeycomb layers.



Figure 6.2 - Helmet prototype liner; a) perspective front view; b) top view; c) perspective rear view

The depth of the recesses was assigned in accordance to the study on the impact compressive behaviour of the two-layered foam-honeycombs composites described in Chapter 4. Although the results presented in section 4.4 suggested that the higher the depth of the honeycomb the higher the energy absorbed, during preliminary attempts at prototype manufacturing, excessive reduction of the thickness of the liner to accommodate higher honeycomb layers caused the breaking of the liner itself. Therefore, for the manufacturing of the prototype helmets the height of the honeycombs was limited to 20mm in the front and rear surfaces, and to 16mm in the crown region. No modifications were made to the lateral and chin surfaces of the helmet, due to manufacturing difficulties encountered. A computer based cutting process was adopted to ensure that the dimensions and the positioning of the hollows were the same for all the helmet liners. A layer 1mm thick of polymeric glue was uniformly distributed at the

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bottom surfaces of the hollows, to provide a homogeneous bond between honeycombs and foams. To ensure the loading of the honeycombs along their out-of-plane direction, and so achieve maximum energy absorption (Gibson and Ashby, 1997), the layers were oriented so that the plane containing the cell walls was perpendicular to the impact direction in all of the three sites selected, as illustrated in Fig. 6.3.



Figure 6.3 - Schematic section of the prototype liner

However, because the UNECE standards do not prescribe any constraints either to the headform or the helmet, the impact direction is not always perfectly perpendicular to the helmet surface, and even more so in real road crashes. In the helmet prototype, this would result in a combined shear-compressive loading of the honeycombs, which could alter their out-of-plane deformation modes and energy absorption capabilities. However, as already discussed in section 3.4, honeycomb materials can still provide good energy absorption capabilities even when loads are inclined with respect to their tubular direction (Hong et al., 2006; Hong et al., 2008). In the helmet prototypes, to provide maximum shear resistance in the symmetry plane, the honeycombs were oriented so that in all the impact sites the doubled cell walls were parallel to the symmetry plane of the helmet, as showed in Fig 6.4.

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Figure 6.4 - Orientation of the honeycombs with respect to the symmetry plane of the prototype liner

6.3 Materials

6.3.1 The outer shell

A microscopic analysis of a section of the outer shell in the crown region revealed a twolayered composite structure underneath a uniform painting coat layer, as showed in Fig. 6.5.



Figure 6.5 - Section of the outer shell in the crown region

The first upper layer was a woven hybrid material made with threads of Kevlar, carbon and fibreglass fibres (CKBEX). The second layer was a woven composite fabric made of Kevlar fibres. A third additional layer, a short fibre glass composite with

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random oriented threads 3 cm long, was added on the external front surface of the helmet, right above the upper edge of the visor. An epoxy resin solution was used to impregnate the layers prior to pressure bag moulding. According to Kostoupoulous et al. (2002) the use of composite layers with significant difference in stiffness can promote delamination of the outer shell, introducing further energy dissipation mechanisms. It is believed that the extra fibre glass composite layer was added in the Gp-Tech for similar purposes, since the front point is often subjected to severe impacts in road accidents (COST 327, 2001). Due to confidentiality, the material properties of the composite layers, their lay-up and thicknesses are not reported in this thesis.

6.3.2 The inner liner, the cheek and chin pads

The inner liner was made of expanded polystyrene (EPS) foam 50 kg/m³ density and its thickness ranged from 35 to 40 mm throughout the surface, except for the crown region, where the thickness was lower to accommodate a lighter layer of EPS foam (35 kg/m³). Mills et al. (2009) stated that the use of lighter foams in the top area compensates the excessive rigidity of the shell in the crown, attributed to the local pronounced double curvature and lack of free edges in proximity (Gilchrist and Mills, 1994), resulting in a better protection of the head. The cheek pads and the chin pad were made of EPS 70 kg/m³ density. The thickness of the lateral cheek pad ranged from 20 to 35 mm, while the thickness of the chin pad varied from 15 to 20mm. All the EPS helmet components were manufactured by means of the injection moulding process.

6.3.3 The honeycombs

The honeycombs used for the assembly of the helmet prototypes were the hexagonal 5.2 Al 3003 cores, produced by expansion of glued aluminium sheets at Cellbond Composites (UK, a partner of the MYMOSA network) and described in Chapter 4.

However, prior to the production of the innovative helmets, a FE model of the helmet prototype was created and a preliminary FE study was conducted. The aim of this work was to determine the honeycomb crush strength which provides maximum energy absorption, while minimising the accelerations transmitted to the head, and the HIC values (Caserta et al., 2009). The outcomes of this study showed that best results could be obtained when the crush strength of the honeycombs was equal to 0.7 MPa. Further

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details are reported in Appendix D. The honeycomb layers used in the present investigation were then chemically treated to achieve the desired crush strength. The etching process consisted in the exposition of the honeycombs to acid, and their collection at regular time intervals to assess the new crush strength. Once the value suggested by numerical analyses was achieved, the honeycombs were then sent to Dainese SpA for the assembly of the helmet prototypes. Particular care was given to the assembly process not to damage or modify the shape of the honeycomb cells. Since the weight of the honeycombs layers was similar to the weight of the quantity of liner removed, the prototype helmets presented same overall weight of their commercial counterparts (1.150kg).

6.3.4 The headform

Each helmet was extra large size, so that the conventional ISO 62 magnesium headform prescribed by standards (UNECE 22.05, 2002) and shown in Fig. 6.6, was used to fit the helmets. ISO headforms are specifically designed so that their lowest natural frequency is higher than 3000 Hz, to avoid interference with acceleration signals during standard tests.



Figure 6.6 - ISO 62cm rigid headform used for drop impact tests. a) lateral view; b) front view.

The measured weight of the headform was 6.1kg, as recommended by UNECE 22.05 regulation.

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6.4 The experiments

The impact tests were conducted at Dainese SpA, following UNECE 22.05 procedures, which are described in section 2.3.2.

Figs. 6.7 a-b show the impact rig used in this investigation. The apparatus mainly consisted of a rigid vibration free steel base, an anvil mount connected to the steel base, a circular helmet support aligned with the anvil base, a monorail guide system 5m high and a hydraulic system used to control the movements of the helmet support along the vertical rail.



Figure 6.7 - Apparatus used. a) Impact rig; b) Drop tower

The impacts were performed against both the flat and kerbstone anvils prescribed by standards, and showed in Figs. 6.8 a and b. The kerbstone anvil was rotated so that the upper edge was inclined by an angle of 45 degrees with respect to the symmetry plane of the helmet, as prescribed by standards.



Figure 6.8 - Impact anvils: a) kerbstone; b) flat

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All the helmets used in our investigation were tested at ambient temperature T = 23.5 °C and hygrometry percentage 62.5%, as suggested by standards. Prior to conditioning, the helmets were placed onto a fake headform for the marking of the impact points on the shell surface. A set of laser beams (showed by red dotted lines in Fig. 6.9) were used to ensure that the marking of the impact points on the surface of the shell was consistent with standard prescriptions, minimising any possible positioning error.



Figure 6.9 - Laser positioning system

The helmets were then placed in a conditioning machine where they were left for 4 hours, to ensure a uniform distribution of the temperature and humidity among the helmet parts. After conditioning, the helmets were fitted with the headform and placed in the testing rig showed in Fig. 6.7a. The helmet chin strap was then strongly tightened to firmly link the helmet to the headform. The rig straps were used to steadily connect the helmeted headform to the testing rig. A laser ray pointing out from the centre of the anvils (showed as a dotted line in Fig. 6.7a) and directed vertically, was used to adjust the position of the helmets on the circular rig so that the end of the beam pointed the impact point marked on the shell prior to any test. By doing this, it was possible to align the impact points with the centre of the anvil with consistent precision. The helmet rig was then raised up to approximately 3m height (measured distance between the impact

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point and the centre of the anvil surface) using the hydraulic control system, and then dropped on the anvil. Six prototype helmets and six Gp-Tech commercial helmets were tested: three prototypes and three commercial helmets were dropped against the flat anvil, whereas the remaining helmets were dropped against the kerbstone anvil. Each helmet was impacted on the points prescribed by standards, except of the point S (chin area), according to the following sequence:

- Front (B);
- Left side (Xsx);
- Top (P);
- Rear (R);
- Right side (Xdx);

Figs 6.10 a-d show the positioning of the helmet on the rig prior to impact, for all the configurations tested.



Figure 6.10 - Positioning of helmets prior to impact; a) front; b) top; c) rear; d) right side

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A total of 60 drop tests were carried out. However, because the results obtained from impacts on the lateral surfaces showed very similar trends, only the right side loading case is discussed. The measured weight of the drop assembly was equal to 0.6 kg so that the total falling mass (headform + drop assembly + helmet) was equal to 7.85kg. The impact energy was approximately equal to 220 J. In all the tests, the acceleration histories recorded from the centre of gravity of the headform, the maximum peak accelerations and the Head Injury Criterion values were calculated and compared.

After the sequence of impacts, the outer shell of the helmet prototypes was removed and the liners were sectioned along their mid-plane. A high resolution camera was used to acquire images of the post-impact deformation of the honeycombs. A digital calliper, offering precision equal to 0.01% of the indicated length, was adopted to measure the height of the deformed honeycombs in proximity of the upper and lower edges.

6.4.1 Data analysis

The accelerations were recorded using three piezoelectric mono-axial accelerometers M353B17, produced by PCB Piezotronics, oriented along the three main headform axes. Such accelerometers offer a measurement range equal to \pm 500g and 3dB limit frequency range 0.35 to 70000 Hz (<u>www.pcb.com</u>, PBC Piezotronics inc.). The acceleration signals were sampled at a frequency equal to 50 kHz, using the data acquisition system PCI-6024E, produced by National Instruments (www.ni.com). Data were low-pass filtered using a 4th order Butterworth filter, as prescribed by ISO 6487 standards (2002). The impacts were recorded over a period of 25ms. The filtering frequency used for data analysis was equal to 1.7 kHz. Fig. 6.11 shows an example of the results of filtering operation on the acceleration signal recorded from an impact of prototype helmet on the front region.

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Figure 6.11 - Example of filtering applied to the impact response of a prototype helmet on the front region;

The impact speed was recorded using the DATASENSOR SR31 photocell, which offers a detection point depth equal to 12mm, a maximum response time equal to 50 microseconds and switching frequency equal to 10 kHz. A triggering system, consisting of two metallic bars oriented perpendicularly to the monorail guide, and placed on the bottom left side of the drop assembly, as indicated in Fig. 6.7a, was used to activate the photocell prior to impact. The height of the photocell was adjusted so that in any test, the impact speed was recorded at approximately 3 cm above the surface of the anvils, in accordance to what prescribed by UNECE 22.05 regulations. The photocell produces an on/off electrical signal as an object is detected within the detection point field. The passage of the two metallic bands during the fall determined the production of two electrical impulses at the times t_1 and t_3 , as showed in the scheme depicted in Fig. 6.12. Case 1:21-cv-02112-CMA-SKC Document 40-8 Filed 09/15/21 USDC Colorado Page 136 of 224

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Figure 6.12 - Trigger functionality

The acceleration signals and the photocell outputs were collected and processed using the software DLS 9000 produced by AD engineering. The impact velocity calculated through software built-in functions as

$$v = \frac{L}{\Delta t} \tag{6.1}$$

where *L* is the distance between the lower edges of the metallic bands and $\Delta t = t_3 \cdot t_1$ is the time interval during which the distance L was travelled by the triggering system. The impact velocity recorded from all the tests ranged from v = 7.52 m/s to v = 7.55 m/s, which resulted within the tolerance limits prescribed by UNECE 22.05. The acceleration histories were then used to calculate the PLA and HIC.

6.5 Results

As outlined in the introduction, the protection function of helmets is linked to the shock absorption capabilities of the materials used for the manufacturing of the energy absorbing liner and the outer shell. The main purpose of this work was to assess the effectiveness of an innovative helmet where an aluminium hexagonal honeycomb was

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introduced in the liner to enhance its shock absorption properties. UNECE 22.05 standard test were performed on both the prototype helmets and the commercial counterparts, in an attempt to determine whether the prototype helmet can provide improved protection to the head. In this section, the acceleration histories recorded from all the impacts (grouped by impact point and anvil used), the average peak resultant accelerations and the HIC experienced by the headform were compared, from both the prototype and commercial designs. In general, it was observed that prototype helmets provided better protection to the head from impacts against the kerbstone anvil in the front and rear surfaces, and from impacts against the flat anvil in the top surface. Significant improvements could be also observed from impacts on the lateral surfaces, not modified in this investigation, for both the anvils used. In some cases, no differences were observed between the commercial and prototype dynamical responses. Other results highlighted the limitations of the strategy adopted in this research instead. Thus, to provide a better understanding of the prototype energy absorption mechanisms, a post impact deformation analysis of the prototype liner was also carried out and discussed in the present section.

Each graph contains six acceleration histories, three curves representing the dynamical response of the helmet prototype and the other three the impact behaviour of the commercial counterparts. Helmets tested against flat anvil were numbered from 1 to 3, whereas those tested against the kerbstone anvil were numbered 4 to 6. Two tables (Table 6.1 and Table 6.2), listing the average PLA and HIC recorded from impacts, are also provided. Values between brackets represent the percentage variation of the prototypes impact parameter with respect to the one offered by the commercial helmets, for a given impact point and anvil used.

In Table 6.3, the standard deviation (SD) of the PLA values is also provided. In the majority of the cases, repeatability of results obtained from impacts against the flat anvil was considerably better than the one attained from impacts on the kerbstone anvil.

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6.5.1 Impacts against the flat anvil





Figure 6.13 - Headform resultant accelerations – time traces for impacts against the flat anvil; front region



Figure 6.14 - Post-impact deformation of the front region. a) Section view; b) Front view

Fig 6.13 shows the resultant translational acceleration histories of the centre of mass of the headform, recorded from impacts on the front surface. As evident from the graph, the reproducibility of the results is quite consistent, and the dynamical response of the prototype helmets strongly matched the one of the commercial Gp-Tech, for the whole duration of the impacts. The calculated average PLA and HIC values (Table 6.1), computed from the translational acceleration traces, suggest that no apparent improvements could be achieved by the introduction of the honeycombs in the front area for this particular impact configuration. A section of the impacted liner of the prototype 2 (Fig. 6.14a) revealed that the aluminium honeycomb layer did not crush completely.

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The residual thickness of the honeycomb varied from 50% to 75% of the initial thickness, as suggested by the values measured at the upper and lower boundary of the honeycomb (Fig 6.14a). As evident from the snapshot, the honeycomb did not completely fold during impact. It was concluded that the honeycomb reached the plateau regime, but its full energy absorption capabilities were not exploited. Such phenomenon was attributed to either strain rate effects, which increased the stiffness of the honeycombs, or to a non-uniform contact between the outer shell and the liner surface, as highlighted by the final deformed shape of the honeycomb.

Top impact



Figure 6.15 - Headform resultant accelerations – time traces for impacts against the flat anvil; top region



Figure 6.16 - Post-impact deformation of the top region. a) section view; b) top view

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Fig. 6.15 illustrates the helmet dynamic responses recorded from impacts on the crown region. As can be seen from the plot, the acceleration histories of the commercial helmet present double peak shape curves. In a past study focused on the fit effect on the dynamic response of motorbike helmets (Chang et al., 2001), this characteristic was attributed to a linear oscillation of the shell mass on the EPS liner, which acts as a spring during such impacts. With reference to the results presented in this thesis, it can be noted that both the prototype and commercial helmets at the initial part of the acceleration traces showed similar trends until the first peak, occurred at approximately t = 4.5 ms. Afterwards, the prototype helmets showed a consistent drop of the acceleration values, with respect to the one offered by the commercial helmet design. From t = 7.5 ms, all the acceleration decreased following same trends. The consistent reduction of the second peak acceleration provided by the prototypes, resulted also in a considerable decrease of the HIC values (average 15%), as showed in Table 6.1. It is therefore supposed that the presence of the honeycomb in the top area contributed significantly to the absorption of the oscillations of the shell. Maximum PLA provided by both the helmet designs resulted in very similar values (Table 6.1). Section of the top area of the prototype liners (Fig. 6.16a and b) revealed an almost uniform and complete crushing of the honeycomb layer, extending symmetrically from the impact point to the sides of the honeycomb layers. Such results suggested that not only the contact between the shell and the honeycomb was uniform, but also that the full honeycomb energy absorption capabilities were exploited.

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Figure 6.17 - Headform resultant accelerations – time traces for impacts against the flat anvil; rear region



Figure 6.18 - Post-impact deformation of the rear region. a) section view; b)rear view

As evident from Fig. 6.17, the helmet prototypes presented higher values of the maximum acceleration, and a slightly shorter duration of the acceleration than the commercially designed helmet. Such results suggested that the introduction of the honeycombs in the rear region led to a worsening of the protective performances of the helmet. Indeed, as reported in Table 6.1, both the averaged PLA and HIC were relatively higher (19.5% and 18.4% respectively) than the ones offered by the commercial Gp-Tech. From observations of the impacted rear region (Fig. 6.18 a and b), it was concluded that the contact between shell and honeycombs was not uniform at the time of the impact, as indicated by the presence of a u-shaped deformation region

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localised on the top of the layer, probably caused by the contact between the rear ventilation area and the honeycombs. It was also observed that the remaining impact surface of the honeycomb presented small or non-existent crushing deformation. It was not possible to establish with certainty the causes of such tests failure. It was therefore inferred that the difference in the curvature between the shell and the liner, the excessive space between the two parts, and the non-optimum positioning of the honeycombs inside the liner might have caused a non-uniform loading of the honeycomb layer itself, leading to high load concentrations and so densification of the underling foam layer.

Side impact



Figure 6.19 - Headform resultant accelerations – time traces for impacts against the flat anvil; side region

Figure 6.19 shows the resultant headform accelerations for impacts on the right side. As can be seen, generally the dynamic responses of both helmet types presented very similar trends and accelerations duration. However, the average PLA and HIC showed by the helmet prototypes (Table 6.1) were lower than the ones offered by the commercial designs. It was not possible to establish whether such improvement was due to the presence of the hollow cutouts and the honeycombs in the liner, that might have influenced the overall load spreading capabilities of the helmet, and so the energy absorption. Sectioning of shell at the lateral impact point revealed a non uniform distribution of the thickness of the shell. In particular, it was observed that the thickness is consistently higher in the region adjacent to the connection between the

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visor and the shell. Small rotations might have occurred during the helmet fall so that the anvils hit the regions where the thickness was higher, leading to higher accelerations. However, due to the limited number of available helmets, further work could not be carried out to confirm this hypothesis.

Table 6.1 - Average HIC and PLA recorded from impacts against the flat anvil

Impact point	HIC [g ^{2.5} s]		Peak Linear Acceleration [g]	
	Prototype	Commercial	Prototype	Commercial
Front	1524 (+2%)	1494	182 (+1.5%)	179.3
Тор	1681 (-15%)	1972	206 (-1.2 %)	208.3
Rear	1654 (+18.4%)	1397	202 (+19.5%)	169
Side	1577 (-11.5%)	1782	182.8 (-9.2%)	201.3

6.5.2 Impacts against kerbstone anvil

Front impact



Figure 6.20 - Headform resultant accelerations – time traces for impacts against the kerbstone anvil; front region

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Figure 6.21 - Post-impact deformation of the front region. A) section view; b)front view

From Fig. 6.20 it can be seen that in all impacts, the acceleration histories of the helmet prototypes were generally lower in magnitude with respect to the ones showed by the Gp-Tech commercial versions, while the duration was similar. The acceleration traces show similar trends up to the time t = 6ms. Afterwards, the acceleration values of the Gp-Tech commercial helmets rose suddenly up to a distinct peak, at t = 8ms(particular emphasis of the phenomenon showed by the Gp-Tech 6). Existing finite element and experimental researches on the load spreading capabilities of polystyrene liners (Mills and Gilchrist, 1991; Gilchrist et al., 2003) subjected to impacts against round surfaces, have shown that kerbstone anvils cause concentration of high loads on the liner, which may reach densification and so transmit high accelerations to the head. With regard to the prototype dynamical response, it can be noted that a similar peak could be observed in only one case (Prototype 6), although its magnitude was distinctly lower than the one observed from the commercial helmets testing. It was concluded that the use of aluminium honeycombs increased the shock absorption capabilities of the Gp-Tech for this particular loading condition. Such conclusion was also confirmed by the significant reductions of both the average peak resultant accelerations (-27%) and the HIC (-14%), compared to the unmodified helmets (Table 6.2). Observations of the deformed liner highlighted a localised honeycomb densification area, that matched remarkably the shape and positioning of the kerbstone anvil (Fig. 6.21 a and b), suggesting that for this impact condition the whole shock absorption capabilities of the helmet prototype were exploited.
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Figure 6.22 - Headform resultant accelerations – time traces for impacts against the kerbstone anvil; top region



Figure 6.23 - Post-impact deformation of the top region. A) section view; b)top view

The resultant acceleration profiles vs time are plotted in Fig. 6.22. It is clear that both the prototype helmets and commercial Gp-Tech presented similar duration of the impacts (approximately 11 ms) and curve shapes. It can be, however, noted that in one test, the prototype helmet exhibited a higher value of the maximum acceleration than the one offered by the commercial helmets design. In the remaining two impacts, the prototype helmets provided a moderate reduction of both the maximum acceleration and the HIC values instead. Because of the variability of the results and the limited number of helmet tested, it was not possible to establish whether the prototype can provide better protection to the head for the given impact condition. Similar average PLA and

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HIC were also observed. Inspections of the crushed prototype liner (Fig. 6.23) revealed a non-uniform deformation of the honeycomb and different failure modes, including shear failure and tearing of the honeycomb cell walls. It was not possible to establish with certainty the cause of such phenomenon, and to confirm whether an optimum exploitation of the honeycomb energy absorption properties would have led to better results. In addition to this, the area covered by honeycomb surface was limited, and it is believed that this factor may have contributed negatively to energy absorption performances. Nevertheless, results obtained from all the tests were widely within standard limits.

Rear impact



Figure 6.24 - Headform resultant accelerations – time traces for impacts against the kerbstone anvil; rear region



Figure 6.25 - Post-impact deformation of the rear region. A) section view; b)rear view

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As evident from Fig. 6.24, the prototype helmets exhibited slightly shorter impact durations and lower acceleration magnitudes, than the ones showed by the commercial Gp-Tech helmets. The dynamic responses of the helmets follow similar trends until t = 4 ms. Afterwards, it can be noticed a change in the slope of the acceleration history in commercial helmets, not observed in prototypes. According to Gilchrist and Mills (1991), such change is due to deflection of the shell during the impact. Thus, it was inferred that the honeycomb absorbed the oscillations of the outer shell. From the graph, it can be also noted that commercial Gp-Tech helmets exhibited a peak in the acceleration values, not shown by helmet prototypes. A section of the impacted liner (Fig. 6.25a and b) showed that the honeycomb did not crush completely, suggesting that the impact energy was dissipated by the liner prior to densification. Similar deformation patterns to the ones observed in the front area were also detected. Enhancement of the energy absorption properties was also confirmed by the considerable reduction of the averaged PLA and HIC (24.5% and 15%), with respect to the commercial design values.

Side impact



Figure 6.26 - Headform resultant accelerations – time traces for impacts against the kerbstone anvil; side region

Fig. 6.26 depicts the headform resultant acceleration histories for impacts on the right lateral surface. As it can be clearly distinguished, all the dynamical responses consist of doubled peak shape curves. Generally, the curves representing helmet prototypes exhibited lower acceleration magnitudes than the ones showed by the commercial Gp-

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Tech helmets, but similar duration. From Table 6.2 it can be also noted that, analogously to what observed from impacts against the flat anvil, the helmet prototype offered reduced average values of both the HIC (- 12%) and PLA (- 12.5%) with respect to its commercial version. Analogously to what stated for the case of impacts against the flat anvil, it was not possible to establish whether the enhanced energy absorption capability offered by the prototypes was due to the presence of the honeycombs.

Table 6.2 - Average PLA and HIC recorded from impacts against the kerbstone anvil

Impact point	HIC [g ^{2.5} s]		Peak Linear Acceleration	
	Prototype	Commercial	Prototype	Commercial
Front	991 (-14%)	1151	141 (-27%)	192
Тор	1248 (- 4%)	1301	184 (+7.4%)	171
Rear	957 (- 15 %)	1126	136 (-24.5%)	180
Side	1098 (-12%)	1236	155 (-12.5%)	177

Impact point	Anvil hit			
	Flat		Kerbstone	
	Gp-Tech	Prototype	Gp-Tech	Prototype
Front	6.8	2.0	32.0	13.0
Тор	3.5	2.0	6.4	34.3
Rear	2.1	12.2	15.6	5.4

Table 6.3 - Standard deviation of PLA values

6.6 Discussion and Conclusions

The coupling of aluminium honeycombs and EPS foams was considered for the design of an innovative motorbike helmet. The impact behaviour of a modified version of a commercial helmet, where aluminium honeycombs were introduced in the front, top and rear region of the energy absorbing liner, was assessed following UNECE 22_05 standards. Unmodified helmets, presenting same geometry and material properties (except for the honeycomb inserts), were also tested under the same conditions. The dynamic responses were compared and the peak linear acceleration and the Head Injury Criterion were used as evaluation criteria.

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The approach is very demanding since the comparison is carried out with the performance of a commercial helmet which has already undergone a stringent optimisation procedure. All acceleration histories reported in section 6.5 and validation criteria listed in Tables 6.1 and 6.2 are widely within the UNECE 22_05 standard limits. Therefore it is clear that any improvement of the performance is rather difficult.

Comparing the results from different impact sites and anvils used, different trends were observed for the two evaluated helmet designs. Generally, the prototype helmets provided better protection to the head from impacts against the kerbstone anvil, in particular by significantly reducing the PLA and HIC during impacts on the front and the rear surfaces. Sensitivity of results to the anvil shape has been already observed in a previous experimental study on the dynamic behaviour of helmets (Gilchrist and Mills, 1991). Different typologies of helmets were tested against flat and hemispherical surfaces. It was observed that forces transmitted to the head are linked to the load spreading properties of the shell (the stiffer the shell, the larger the load spreading area). They observed that deformable shells provide better protection against flat surfaces, at expenses of protection against round surfaces. Conversely, stiff shells (such as the one used in our investigation) provide better protection against round surfaces at expenses of protection from impacts against flat ones. In addition to this, it was observed that the magnitude of the forces transmitted to the head was also dependent on the stiffness of the underlying energy absorbing liner and the curvature of the shell in the impacted point. It was concluded that helmets cannot be optimised for all shapes of struck objects. In the research presented in this thesis, the trends observed are generally in agreement with results presented in literature. The achievements obtained were linked to the capacity of honeycombs to offer extended and constant plateau regime, which makes them capable of providing good shock absorption properties even at very high deformation stages. Some little improvements were also observed from impacts in the top region, but because of the variability of the results (Table 6.3) and the limited number of experiments carried out, it was not possible to confirm this trend. When impacts were performed against the flat anvil, the prototype top area provided best protection to the head, in terms of HIC. No significant improvements were observed from impacts on the front region, while impacts on the rear region highlighted inferior performances in comparison with the ones offered by the helmet commercial design. From observations of deformed prototype liners, it was concluded that the honeycombs in the front and rear areas did not contribute significantly to the impact energy

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absorption. This was attributed to a non-uniform contact between the outer shell and the honeycombs during the impacts, to strain rate effects, which increased the honeycombs resistance, and to a non-optimum design of the prototype liner. Surprisingly, significant reductions of the PLA and HIC were observed from impacts on the lateral surfaces, not modified because of manufacturing difficulties, against both the anvils. It was assumed that the presence of honeycombs and the recesses in the liner might have influenced the load spreading capabilities of the helmet, and so the energy absorption. Nevertheless, observations of the damaged shell suggested that impacts did not always occur on the marked impact points, and that higher accelerations were observed when the impact occurred in proximity of the interface visor edge/shell, where the thickness of the shell was higher than in the surrounding areas. Thus, it was not possible to establish accurately the causes of such improvements and it is then believed that both the factors might have contributed to the difference between the prototype and commercial Gp-Tech dynamical responses.

However it must be noted that due to research time and budget restrictions, the manufacture of the prototype helmets was carried out following a non-industrialised process prone to imperfection. Moreover, such constraints did not allow for more prototypes to be made, so that there was no possibility to carry out an optimisation of the prototypes.

On the basis of the results presented in this chapter it can be concluded that the use of aluminium honeycombs, as reinforcement material for the energy absorbing liner, can lead to an improvement of the safety levels provided by current commercial helmets without increasing their weight. Conversely, results from impacts against the flat anvil indicated to some extent the limitations of the strategy adopted in this research. Future work should be addressed to the optimisation of honeycombs reinforced helmets for impacts against flat surfaces. Finite element analyses should be addressed to the design of prototype helmets where the gap between the outer shell and the inner liner is reduced to minimum, especially in the rear region. Moreover it would be interesting to assess the prototype impact protection when more severe impact conditions or different standard regulations are considered. Future designs should also consider the extension of the areas covered by the honeycombs to the remaining surface of the liner, including the lateral surfaces. Most notably, this is the first study to the knowledge of the author to investigate the effectiveness of helmets where aluminium honeycombs are introduced in the liner. Results presented in this chapter could provide the framework for future 131 Chapter 6 . Experimental assessment of a helmet prototype

research on the design of the honeycomb reinforced helmets, and to assess their performance characteristics. The approach proposed in this research can be also applied to a wider range of personal protective equipment design and energy absorption applications.

6.7 Publications

The research work presented in this chapter resulted in the following publications:

- 1. Caserta, G., and Galvanetto, U. Design of protective equipment. MYMOSA EU research training network, Report no. WP3.2a, 2010.
- Caserta G., Iannucci, L., Galvanetto, U. Shock absorption performances of a motorbike helmet with honeycomb reinforced liner. Composite Structures, 93, 2011, pp. 2748 - 2759;

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Chapter 7 Finite element modelling of the helmet prototype

7.1 Introduction

The experimental results discussed in Chapter 6 led to relevant conclusions for the design of innovative and safer helmets where aluminium honeycomb could be used as reinforcement material for the energy absorbing liner. However, some outcomes suggested that optimisation of the prototype for some loading conditions is needed. The use of FEA could play an important role for the optimum design of helmets, or the prediction of the prototype dynamical response when changes in the geometry, different honeycomb crush strengths, impact speed, impact points and surfaces struck are considered.

Literature survey suggests that helmet models can be grouped in two categories:

- Lumped mass models
- Finite element models

Lumped mass models consist of rigid masses connected by massless springs and dampers. Rigid masses represent the helmet and head components, while springs and dampers represent the material properties of these components. Not many studies on the use of lumped mass models were found in literature. An example of such model was proposed by Gilchrist and Mills (1993), where four masses represented the outer shell, the helmet liner, the headform and the striker, connected by springs and dampers (Fig. 7.1).

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Figure 7.1 - Lumped mass model of a helmet (from Gilchrist and Mills, 1993)

The authors investigated the effect of different shell materials and different foam materials on the forces transmitted to the head for impacts against an hemispherical anvil. They concluded that the use of stiffer shells in combination with lower density foam liners results in a reduction of the force transmitted to the head during the impact. One drawback of these models is the fact that they are very simplistic and do not include information regarding the geometry of the helmet parts and do not allow the modelling of interface forces. In addition to this, calibration is required for each impact configuration, which means that they can only be developed if helmets are available for testing. Hence, they can only be used for trend analyses and provide limited information in comparison to FE models.

Finite element models allow the prediction and analysis of more complex mechanisms, such as material non-linearity, strain rate sensitivity, large deformations, etc. Another advantage of Finite element models with respect to lumped mass models is the major flexibility provided in the design phase.

Yettram et al. (1994) developed one of the first FE helmet models reported in literature. The model was very simplistic and represented an open-face helmet (Fig. 7.2a). The authors simulated the helmet dynamic response for impacts on the crown region, combining three EPS foam densities (24, 44 and 57 kg/m³) with four plastic shell materials (glass-fiber polymeric reinforced shell, polycarbonate shell, high density and

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low density polyethylene shell). Three impact speeds were simulated, 3.27, 5.1 and 6.9 m/s, and from numerical outcomes the authors concluded that the use of stiff shells and the higher density foams caused higher peak accelerations transmitted to the head and HIC values, confirming previous experimental findings (Hopes and Chinn, 1989).

Later, FE models of the helmet evolved in more sophisticated and realistic reproductions of commercially available helmets (Fig. 7.2b and c).



Figure 7.2 - Evolution of FE models of motorcycle helmets. a) Yettram et al. (1994); b) Kostoupoulos et al. (2001); Cernicchi et al. (2008)

Kostoupoulos et al. (2001) conducted a FE analysis on helmets featuring three different fibre reinforced woven shells: carbon fabric reinforced polyester, glass fabric reinforced polyester and Kevlar fabric reinforced polyester. The liner simulated in their investigation was made of EPS foam 50 kg/m³. The innovation introduced by the authors consisted in the modelling of delamination. Fibre tensile, matrix tensile and matrix compressive failure were also modelled. Snell Standard tests (1998) were reproduced. As stated in section 6.3.1, the authors observed that delamination was more pronounced in those shell systems where the composite layers presented significant difference in inplane shear stiffness. The only drawback of the model proposed by the authors is the fact that validation against experimental results was not carried out.

Cernicchi et al. (2008) proposed an FE model of a commercially available composite helmet and studied the influence of the mesh size on the results. One of the innovation introduced in the FE modelling of the helmet consisted in the modelling of the retention system, which was approximated as two elastic springs which connected the chin of the headform to the outer shell. The UNECE 22.05 standard tests were simulated on the 135 Chapter 7 . Finite element modelling of the helmet prototype

front, top, rear and side regions of the helmet. From numerical outcomes the authors observed that the presence of the chin strap improved the prediction of the experimental results. Further details regarding the influence of the mesh size are described later.

This chapter discusses the development and validation of a Finite element model representative of the helmet prototype presented in chapter 6 of this thesis. The model described in this section is intended to provide the framework for future research, of which some suggestions are provided in Chapter 9. Major focuses of this study are the material characterisation of the helmet parts, and the effect of the honeycomb crush strength on the impact behaviour of the prototypes, which is explained in detail in Appendix D. The explicit solver Ls-Dyna 971 is used to simulate the experiments performed in the present investigation.

7.2 The Gp-Tech prototype model

The model consisted of the outer shell, the inner liner (intended as the assembly of the main liner, the top layer and the lateral cheek pads), the honeycomb layers, the chin strap and the rigid headform (Fig. 7.3). Because the visor and the comfort pad do not provide significant contribution to the energy absorption provided by helmets (Cernicchi et al., 2008; Ghajari, 2010;), such parts were not included in the model. Also, to reduce computational costs and considering that impacts on the chin area were not considered in the present investigation, the chin pad was not included in the model. Prior to the generation of the prototype FE model, an accurate virtual version of the Gp-Tech helmet geometry was reconstructed using a 3D digitiser scanner over the commercial Gp-Tech parts. The acquired IGES format images were then imported in CAD software for modifications. Cavities were virtually created on the front, top and rear surface of the liner, to accommodate the honeycomb layers. The dimensions and the depth of the hollows were the same as those created in the prototype helmets tested during this investigation (section 6.2). The virtual Gp-Tech parts were finally imported in Hypermesh 9.0 (Altair Engineering Inc.) for the generation of the FE model.

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Figure 7.3 - Prototype finite element model. a) Perspective view; b) Section view

7.2.1 The headform

The helmet model was fit with a finite element version of the ISO 62 rigid headform prescribed by UNECE standards (2002) and used at Dainese s.p.a for helmet testing. For the generation of the headform model (Fig. 7.4), 10,961 solid tetrahedral elements (Hallquist, 1997a) were used, with average length equal to 16mm. As stated in section 2.3, in order to avoid potential influence of the headform vibrations on the test results, the UNECE regulations require the lowest eigenfrequency of the headform to be larger than 3000 Hz. Also, in helmet impact tests, the deformation of the headform is negligible compared to the deformation of the helmet (Aiello et al., 2010). Thus, in the present investigation the headform was modelled as infinitely rigid material, in line with previous researches on the FE modelling of helmets (Bosch, 2010; Ghajari, 2010; Cernicchi et al., 2006;). The material algorithm 20_RIGID (Hallquist, 1997b) was used to characterise the headform mechanical properties. However, as stated in section 5.2, realistic values of the Young's modulus and Poisson's ratio are required to avoid contact instabilities, when using this particular material algorithm. For the purposes of this study, the material properties of the Magnesium K1A (Table 7.1) were used, because this material is commonly adopted for the production of ISO standard headforms (www.cadexinc.com). The Ls-Dyna keyword PART_INERTIA (Hallquist, 1997b) was activated to automatically compute inertial properties (Table 7.1), and to assign initial speed (v = 7.5 m/s) and translational mass (m = 6.1 kg, as prescribed by standards for the given headform size). A node (Fig. 7.4a) was generated in the headform model to define the centre of gravity (C.G.) of the headform. Such node was positioned in the

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virtual headform in a way to result coincident with the G point prescribed by standards (see Fig. 2.11), where accelerations shall be measured.



Figure 7.4 - ISO 62 standard headform model and centre of gravity node

Moments of inertia [kg·m ²]	Density [kg/m³]	Young's Modulus [GPa]	Poisson's ratio
Ixx = 0.017345			
Iyy = 0.011550			
Izz = 0.008541	1740	38	0.34
Ixy = 3.88e-5			
Ixz = 3.665e-4			
Iyz = -4.039e-4			

Table 7.1 - Mechanical properties assigned to the headform model

7.2.2 The energy absorbing liner

As already stated in Chapter 2, the helmet liner is the component that absorbs most of the energy during an impact. The liner of the Gp-Tech used in this investigation is made of a multi-layered EPS foam material. Tetrahedral elements were used to generate the models of the polymeric parts of the liner, as suggested by Cernicchi et al. (2008). The dimensions of the elements were chosen on the base of the results obtained from the

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convergence study carried out during the present investigation, and described in Appendix B. Medium mesh density (consisting of elements with average side equal to 16mm) was found to provide best agreement between numerical and experimental outcomes (section 5.3.4). As result of the meshing process, 20,550 elements were used to generate the helmet main liner, 6,444 elements were used to generate the top layer and 5,950 elements were adopted to generate the cheek pads.

As stated in section 6.3.2, the main liner is made of EPS foam with 50 kg/m³ density, the top layer is made of EPS foam with 35 kg/m³ density, while the cheek pads are made of EPS foam with 70 kg/m³ density. The material card 63_CRUSHABLE_FOAM (Hallquist, 1997b), of which a short description is provided in section 5.3.2, was adopted for the modelling of the mechanical properties of the liner parts. However, due to lack of experimental information regarding the compressive behaviour of EPS foams 35 kg/m³ and 70 kg/m³, the semi-empirical approach proposed by Gibson et al. (1997), and already used in the present investigation to model the behaviour of two-layered materials (section 5.3.2), was adopted to generate mechanical properties of such foams. These properties were then used to characterise the impact behaviour of the cheek pads and the top layer. Recalling Eq 5.3, Table 5.1 (dynamic values) and assuming the density of the bulk polymer equal to 1050kg/m³, the following properties and stress versus strain curves were obtained and introduced in the model. Poisson's effect was modelled by assigning a very small Poisson's ratio to all the liner components (Table 7.2), analogously to the modelling of the two-layered structures presented in this thesis (section 5.5).

Table 7.2 - EPS foam material properties. ρ = foam density; R = foam relative density; E = Young's modulus, σ_y = crush strength

Helmet component	ρ [kg/m³]	R	E [MPa]	σ _y [MPa]	v
Top layer	35	0.0333	11.8	0.29	0.01
Main energy absorbing liner	50	0.0476	27.1	0.54	0.01
Cheek pads, chin pad	70	0.0667	46.4	0.9	0.01

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Figure 7.5 - Stress versus strain curves representing the numerical compressive behaviour of the energy absorbing liner parts.

The honeycomb layers used during the experiments were modelled using twodimensional shell elements. Analogously to the honeycomb FE models presented in section 5.4, four-noded squared shell elements were used to generate the prototype honeycomb components. Cell walls with doubled thickness were included. The honeycombs were oriented in a way that all the honeycomb cell walls with doubled thickness were parallel to the symmetry plane of the helmet (Fig 6.4). The dimensions of the honeycomb cell walls were the same as those of the honeycombs used for the experiments. Five through-thickness integration points were assigned to each element of the honeycomb model, and fine mesh density (Table 5.1) was adopted, since this mesh size is the one that provided best convergence between experimental and numerical results in the present investigation (Appendix B). The geometry of the layers was such as to faithfully reproduce the shape and dimensions of the honeycombs used during the experiments (Fig 7.6). As result of the meshing process, the front honeycomb layer consisted of 155,271 elements, while the top and the rear layers consisted of 117,418 elements and 254,833 elements respectively. Initial imperfections were introduced in the models, according to what reported in section 5.4.1. The material properties of the Al 3003 H18 alloy ($\rho = 2730 \text{ kg/m}^3$, E = 68.9 GPa, $\sigma_v = 186 \text{ MPa}$, v = 0.33) were implemented using the Ls-Dvna algorithm 24_PIECEWISE_LINEAR_ISOTROPIC_PLASTICITY (Hallquist, 1997b), of which a short description is provided in section 5.4.2.

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Figure 7.6 - Finite element model of the energy absorbing liner (The top layer is represented in yellow).

Prior to simulating the impact tests described in Chapter 6, a parametric study aimed to assess the influence of the honeycomb crush strength on the prototype dynamic response was performed. Different honeycomb crush strengths were simulated by varying the thickness of the honeycomb cell walls, while the dimensions of the cells (6.3 mm) and the material properties were not changed. The honeycomb crush strengths taken in consideration ranged from 1.5MPa (achieved with cell wall thickness equal to 70 microns) to 0.5 MPa (achieved with cell wall thickness 20 microns). Further details and results obtained from this parametric study are provided in Appendix D. In this section, only results obtained from simulations of prototype including honeycombs with crush strength equal to 0.7 MPa (achieved with cell wall thickness equal to 30 microns) are compared to experimental results, since this strength was the one of the actual honeycombs adopted for testing. The polymeric glue used to fix the honeycombs to the helmet liner during the prototype manufacturing (section 6.2) was simulated by fully constraining the bottom nodes of each of the honeycomb layers to the underlying foam model.

7.2.3 The outer shell

As stated in Chapter 6, the outer shell of the prototype consisted of a two-layered woven composite structure. A third additional layer was placed in the front area of the helmet,

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to improve protection to the head. Because of its small thickness (from 1 to 1.5mm through the whole surface), the outer shell was modelled using 10,524 four-noded shell elements. The stacking ply sequence of the composite structure was simulated using the Ls-dyna keyword PART_COMPOSITE (Hallquist, 1997b). When adopting such algorithm, each layer is identified by an integration point, to which Ls-Dyna users can assign thickness and material properties. For unidirectional laminas, the in-plane orientation of the fibres with respect to a user defined local coordinate system can be also assigned. However, due to the fact that all the composite layers of the Gp-Tech either presented a woven or randomly oriented fibres structures, such option was not considered. The sequence of integration points is given starting from the bottommost layer (in this case, the layer in contact with the energy absorbing liner), and the total thickness T of the composite is given by the sum of the thicknesses of each integration point. Fig. 7.7 shows, as example, the simulated stacking sequence of the outer shell in the front region.



Figure 7.7 - Simulated stacking sequence of the outer shell in the front region

The material card 58_LAMINATED_COMPOSITE (Hallquist, 1997b), which is designed for the modelling of the behaviour of orthotropic composites, unidirectional composites and woven composites, was adopted to characterise the mechanical properties of the Gp-Tech shell components. Such model is formulated for plane stress conditions, and it is based on a continuum damage mechanics model proposed by Matzenmiller et al. (1995). According to this model, the effective stress $\hat{\sigma}$ (i.e. the stress carried by the net 142 Chapter 7 . Finite element modelling of the helmet prototype

undamaged area) and effective shear stress $\hat{\tau}$ are related to the nominal stress σ through a damage parameter ω_i .

For a unidirectional lamina,

$$\begin{bmatrix} \hat{\sigma}_{C} \\ \hat{\sigma}_{T} \\ \hat{\tau} \end{bmatrix} = \begin{bmatrix} \frac{1}{1 - \omega_{L}} & 0 & 0 \\ 0 & \frac{1}{1 - \omega_{T}} & 0 \\ 0 & 0 & \frac{1}{1 - \omega_{S}} \end{bmatrix} \begin{bmatrix} \sigma_{C} \\ \sigma_{T} \\ \tau \end{bmatrix}$$
(7.1)

where the subscript L refers to the longitudinal direction (intended as the direction of alignment of the fibres), T to the transverse direction and S refers to in-plane shear. For an undamaged lamina $\omega_i = 0$, while for a fully damaged lamina $\omega_i = 1$. The constitutive law is given by

$$\begin{bmatrix} \hat{\sigma}_{C} \\ \hat{\sigma}_{T} \\ \hat{\tau} \end{bmatrix} = \frac{1}{D} \begin{bmatrix} (1 - \omega_{L})E_{L} & (1 - \omega_{L})(1 - \omega_{T})\nu_{TL}E_{L} & 0 \\ (1 - \omega_{L})(1 - \omega_{T})\nu_{LT}E_{T} & (1 - \omega_{T})E_{T} & 0 \\ 0 & 0 & D(1 - \omega_{LT})G_{LT} \end{bmatrix} \begin{bmatrix} \varepsilon_{C} \\ \varepsilon_{T} \\ \gamma \end{bmatrix}$$
(7.2)

where

$$D = 1 - (1 - \omega_L)(1 - \omega_T)\nu_{LT}\nu_{TL} > 0$$
(7.3)

To determine the stress-strain mechanical response within the laminate, Ls-Dyna user can define in-plane compressive and tensile Young's moduli, shear moduli and Poisson's ratio. Damage is then reproduced as degradation of the in-plane stiffness matrix components, trough the variables ω_i . Four failure modes are taken into account for each lamina:

- tensile fibre failure (fibre rupture);
- compressive fibre failure (fibre buckling);
- matrix cracking under transverse tensile and shear loadings;
- matrix cracking under transverse compressive and shear loadings.

The damage parameter ω is implemented in MAT 58 as follows (Xiao, et al., 2009):

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$$\omega = 1 - \exp\left[-\frac{1}{m} \left(\frac{\varepsilon}{\varepsilon_0}\right)^m\right] \tag{7.4}$$

where ε_0 is the nominal failure strain, and *m* is a parameter which is related to development of failure. High values of m result in faster propagation of damage, which in a typical stress versus strain curve results by a steeper drop of stress values once damage initiates (Fig. 7.8).



Figure 7.8 - Stress versus strain curve of a simulated unidirectional lamina for different values of m (from Schweizerhof et al., 1998)

One of the drawbacks of using this material card is the fact that a calibration through experimental tests is required for each simulated material, especially for the modelling of the softening part of the stress-strain curve, resulting often in time consuming processes (Schweizerhof et al., 1998). In addition to this, the two damage parameters ω_L and ω_T are generally different for compressive and tensile loadings, which increase the number of experimental tests needed. The shear damage parameter ω_S is independent on the sign of the shear stress. The maximum strengths in tension, compression and shear must be also defined with their correlated strain values. Failure in an element is considered when the local strain reaches a failure value defined as ERODS parameter (Hallquist, 1997). Upon ERODS value is reached, Ls-Dyna users can choose whether the elements are either removed or their moduli are decreased to near zero values. The second option is known to result in a more stable contact between the shell and the other parts in the model (Ghajari, 2010), hence it was adopted in this investigation.

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The collection of material properties and calibration of the material cards was beyond the scopes of this research, and could not be performed due to time constraints. Thus, the material properties provided by the helmet manufacturer were introduced in the FE model. Such parameters were obtained from quasi-static compressive, tensile and shear tests prescribed by ASTM standard regulations (D 3039/D 3039M; 4255/D 4255M; 5467/D 5467M), performed at MAVET s.r.l., on flat coupons made with the composite materials used for the production of the Gp-Tech. For confidentiality reasons, such properties cannot be reported in this thesis.

It must be also noted that in FE analyses, softening behaviour is subjected to mesh size sensitivity (Zienkiewicz and Taylor, 2000). With particular reference to the continuum damage model used in the present investigation, the smaller the elements used, the lower the energy required to initiate damage. Thus, excessively small elements would lead to unrealistic behaviour. Previous studies on the FE modelling of motorcycle helmets (Cernicchi et al., 2008; Ghajari, 2010), where Ls-dyna MAT58 was adopted for the modelling of the outer shell, suggested that mesh dependence analyses are needed to obtain good agreement between experimental and numerical results. Cernicchi et al. (2008) simulated impact on the front surface of a commercial helmet against the kerbstone anvil prescribed by UNECE 22.05 regulations. Six mesh densities were adopted, where the element average dimension ranged from 2mm to 15mm. The force experienced by the anvil was plotted over time and numerical outcomes suggested that convergence between results was obtained only for meshes where the average side ranges from 2mm to 5mm. In a similar study, Ghajari (2010), simulated UNECE 22.05 standard impact tests on a FE model of an AGV full face helmet produced at Dainese s.p.a. The resultant acceleration of the centre of gravity of the headform was considered as evaluation criteria, and the mesh sensitivity of the shell was investigated by using four-noded elements presenting three different average lengths: 3mm, 6mm and 10mm. From numerical outcomes, the author observed that use of 6mm and 3mm elements resulted in very similar acceleration histories, both in terms of magnitude and duration, while a consistently different dynamical response was observed from use of 10mm elements. On the base of these results, in the present investigation the outer shell was reconstructed using elements with average dimension equal to 3mm.

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Figure 7.9 - Outer shell model

7.2.4 The chin strap/retention system

The chin strap, a 300 mm long, 25 mm wide and 1.5 mm thick polyethylene terephthalate (PET) woven band, was modelled using four-noded shell elements. The initial shape created, passed through the holes of the cheek mouldings and closely fitted the headform chin. A hundred 4-noded shell elements were used for the generation of the chin strap shape. The material card MAT24 (Hallquist, 1997b) was used to model the chin strap mechanical properties ($\rho = 800 \text{ kg/m}^3$, E = 1.83 GPa, v = 0.2, $\sigma_y = 47 \text{ MPa}$). A preliminary FE simulation was carried out to pull the ends of the chin strap through the shape conformed to the chin of the headform. To achieve this aim, a force equal to 10 N was applied at both the ends of the chin strap model and directed towards the top of the model (Fig. 7.10 a and b). Then, the deformed mesh of the chin strap was introduced in the prototype model with no pre-stress conditions, in accordance to previous studies on the modelling of the chin strap in crash helmet simulations (Mills and Gilchrist, 2008; Ghajari, 2010). The link between the retention system and the shell of the Gp-Tech was simulated by constraining the nodes at the ends of the chin strap model to the surface of the outer shell.

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Figure 7.10 - Virtual tightening of the chin strap (the complete model is not shown); a) front view; b) side view

7.2.5 The anvils

The flat and kerbstone anvil prescribed by standards, and used in the present investigation, were created through use of pre-built Ls-Dyna rigid wall algorithms. A cylindrical surface (with 130mm diameter and 50mm thickness) was used to simulate the flat anvil, while a combination of a cylindrical surface (with 30mm diameter and 125mm lenght) and two flat surfaces (125mm length x 80mm width) was used to generate the kerbstone shape (Fig. 7.11).



Figure 7.11 - UNECE 22.05 finite element anvil shapes. a) flat anvil; b) kerbstone anvil

Friction between the anvils and composite shell helmets during impact tests was measured by Mills et al. (2009) who performed a series of oblique impact tests on to flat anvils covered with abrasive paper. From experimental outcomes, the authors measured 147 Chapter 7 . Finite element modelling of the helmet prototype

a dynamic coefficient of friction approximately equal to 0.55. Although in the present investigation no abrasive paper was used, the value proposed by Mills et al. was adopted.

7.2.6 Contact logics

Three typologies of penalty stiffness contact algorithm (see section 5.4.3) for a generic description) were used to model contact between the helmet parts.

AUTOMATIC_SURFACE_TO_SURFACE (Hallquist, 1997b) contact was defined at the interfaces shell/liner, shell/honeycombs, honeycombs/liner, liner/headform and at the interface between the chin strap and the helmet components with which contact could have occurred during the simulations (i.e. headform, cheek pads and shell). Soft penalty formulation was activated for contact between parts exhibiting significant difference in stiffness, such as shell/liner, liner/headform and honeycombs/shell. The soft penalty coefficient was set equal to 0.1 (Ls-Dyna standard value). For a more realistic reproduction of the transmission of forces between the helmet parts, the thickness of the shell and the honeycomb walls was set to be taken into account.

AUTOMATIC_NODES_TO_SURFACE (Hallquist, 1997) algorithm was defined at the top and bottom nodes of each honeycomb layer model, to avoid penetration of the honeycomb edges in the surfaces of the shell and the liner during the simulations. Soft penalty option was activated at the interface honeycomb/shell and the scaling factor was set equal to 0.1.

Finally, the contact algorithm AUTOMATIC_SINGLE_SURFACE (Hallquist, 1997) was defined for each honeycomb layer, to prevent self-penetration of the honeycomb cell walls during the buckling of the honeycombs.

Static and dynamic friction at the interfaces between the helmet parts was modelled through Ls-Dyna built-in functions, which are based on the Coulomb friction model. For the contact between the polystyrene components, the static coefficient of friction was set equal to 0.5 (Cernicchi et al., 2008; <u>www.matweb.com</u>). For contact between foams and metallic parts (i.e. at the interfaces liner/honeycomb and liner/headform), due to lack of available data in literature, the coefficient of friction between polystyrene and steel (1.05) was taken as reference. For contact between the shell and the other parts of the helmet, and between the chin strap and the cheek pads and headform, a unique

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coefficient of friction equal to 0.3 was adopted, in line with existing FE studies (Ghajari, 2010). In all the interfaces, the dynamic coefficient of friction was set equal to one third of the value of their static counterparts.

7.2.7 Simulations

Impacts were simulated in the front, top and rear region of the helmet against both the kerbstone and flat anvil. In each impact configuration, the kerbstone anvil was inclined by a 45 degree angle with respect to the plan of symmetry of the helmet, in accordance to standard prescriptions. Prior to simulation, the virtual helmet was positioned in a way such as the impact point was aligned to the centre of the surface of the anvil. Impact speed was simulated by assigning initial velocity equal to 7.5 m/s to all the nodes of the model (excluding the headform, for which PART_INERTIA was used), by using the LS-Dyna algorithm INITIAL_VELOCITY (Hallquist, 1997).

Acceleration histories of the C.G. of the headform were recorded and processed using the software LS-PrePost. A virtual version of the Butterworth filter adopted for the experiments (section 6.4.1) was applied to the numerical acceleration signals to remove undesired oscillations. The filtering frequency was equal to the one adopted during the experiments (1.7 kHz) for the removal of undesired numerical oscillations caused by contact instabilities. In each simulation, the solving time step was calculated on the size of the honeycomb elements and it was of the order of 10⁻⁹s. Due to the high number of elements employed for the modelling of the helmet, each simulation required an average of 72 hours for complete solution (time recorded when a 8 CPUs computer was used).

7.3 Results

In this section, the FE model of the helmet prototype is validated against the experimental results obtained from impacts in the front (B), top (P) and rear (R) region presented in Chapter 6. In each graph, the numerical resultant acceleration histories of the C.G. of the headform are compared with their experimental counterparts. The peak linear acceleration is also considered as a validation criteria and compared to the average values recorded during experiments (Table 7.3). Snapshots of the post-impact deformation of the honeycombs were also taken in the three selected areas and compared with the experimental equivalents presented in Chapter 6.

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7.3.1 Front region (impact point B)

Figure 7.12 a and b depict the acceleration histories recorded from impacts in the front area, against the flat and kerbstone anvil.



Figure 7.12 - FE results from impacts in the front area. a) flat anvil; b) kerbstone anvil

In general, the FE model provided very good agreement with the trends observed experimentally, both in terms of shape of the acceleration histories and magnitudes. However, there are some minor discrepancies that need discussion. Firstly, it can be noted that initially the experimental accelerations rose following different patterns compared to the numerical predictions, until a maximum value. Comparing the peak linear accelerations (Table 7.3), it can be also observed that the numerical predictions

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are relatively higher (9.4% for the impacts against the flat anvil and 6% for impacts against the kerbstone anvil) than the average attained from experimental results. These discrepancies were attributed to the difference between the simulated material properties of the helmet parts and the actual material properties of the components used for the manufacturing of the prototypes. For example, it is known that environmental factors such as temperature and humidity might have a significant degrading effect on the mechanical properties of polystyrene foams (Gibson and Ashby, 1997; Liu et al., 2003). With particular reference to the effect of humidity on the compressive properties of EPS foams, experimental data available in literature (Liu et al., 2003), it was observed that the plateau stress of EPS foams compressed in normal conditions (25 °C and relative humidity 30%) decreased by approximately 20% when the same materials were tested at relative humidity equal to 85%. It is therefore possible that because of not ideal storing conditions, the foams tested in the present analysis (section 4.4) for the characterisation of the FE liner might have weakened due to exposure to humidity, compared to the foams used for the manufacturing of the prototypes, contributing to the difference between numerical and experimental outcomes. It must be also stressed that the material characterisation of the shell was attained through quasi-static tests on flat material coupons, a methodology commonly adopted in literature for the FE modelling of the outer shell (Kostopoulos et al., 2002; Aiello et al., 2007; Cernicchi et al., 2008), but not actually representative of real helmet loading conditions. In addition to this, because of local curvature and imperfection in the manufacturing processes, the mechanical response of composite shells significantly varies from the one offered by same materials in a flat form. Mills et al. (2009) for example, in a FE investigation on the impact performances of motorcycle helmets, demonstrated that the use of material properties obtained from test performed on flat composites leads to overestimation of the shell bending stiffness (i.e. the loading spreading capabilities of the shell are altered, and so the accelerations transmitted to the head).

A second major discrepancy that can be observed from Fig 7.12a and b (and also on the other results presented in this section) consists in the duration of the numerical accelerations, which is in general shorter than the one observed experimentally. Evident difference between the curves can be observed in the unloading region (i.e. the region of the curve after the maximum peak acceleration), where numerical resultant acceleration traces drop following a steeper pattern compared to the experimental counterparts. This phenomenon was more marked in impacts against the kerbstone anvil (Fig. 7.12b). In

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existing FE results (Ghajari, 2010) such behaviour was attributed to the modelling of the unloading of the foams in material model MAT63, of which a short description is here provided in section 5.3.2. When using such material card, the unloading of the foam in the stress versus strain curve is assumed to follow a straight line, whose slope is by default equal to the user defined foam Young's modulus (Hallquist, 1997). However, due to the fact that the foam densification region exhibits higher slopes than the one typical of the elastic region (see for example Fig. 2.3), Ls-Dyna automatically adjusts the value of E in a way that the slope of the unloading curve is higher than the steepest slope present in the user input curve. By doing this, the unloading occurs without selfintersection of the curve itself and numerical instabilities are avoided, at expenses of a more unrealistic unloading behaviour. In all the simulations performed in the present investigation, the value of E was automatically increased by two orders of magnitude compared to the value defined as input. As consequence, the foam elastic behaviour was also affected, but because of its short duration compared to plateau and densification regimes (Gibson and Ahsby, 1997), it was believed that its influence on the accelerations transmitted to the head was negligible. In spite of the discrepancies observed in the present analysis, the predictions provided by the model are accurate, and differences in the magnitudes and durations of accelerations are in line with existing FE studies (Cernicchi et al. 2008, Ghajari, 2010).

Fig. 7.13 a and b show the comparison between the simulated deformation of the honeycomb layers and experimental observations.

A comparison between sections of the helmet in the front region is also provided in Fig. 7.14 a and b. As it can be noted, the FE model could generally reproduce the deformation shapes observed experimentally. However, considering the impacts against the flat surface (Fig. 7.14a), it can be noted that in numerical outcomes the honeycomb presented a localised densification region, not shown by honeycombs used for the experiments. This phenomenon was attributed to the assumptions made for the modelling of the strain rate dependency of aluminium honeycombs (section 5.4.2). Indeed, the formulation used in the present investigation (Cowper and Symonds, 1958) does not take into account the air trapped within the honeycomb cell walls, which is known to increase the strength of honeycombs subjected to impact loadings (Zhou and Mayer, 2002). In addition to this, it is possible that shear stresses developed in honeycomb structure might not have been as consistent as those developed in

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honeycombs during the experiments. Indeed, one of the limitations of the work presented in this thesis is the lack of validation of the honeycomb model against quasistatic shear test results. Therefore, densification of the honeycomb occurred, contributing to the transmission of higher accelerations to the headform, compared to the ones observed experimentally.



Figure 7.13 - Post impact deformation of the front region. Comparison between FE simulations and experiments. a) impact against the flat anvil; b) impact against the kerbstone anvil



Figure 7.14 - Post-impact section of the front region. Comparison between FE simulations and experiments; a) impact against the flat anvil; b) impact against the kerbstone anvil

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7.3.2 Top region (Point P)

The results obtained from FE impacts in the crown area are shown in Fig 7.15 a and b, for the two evaluated impact surfaces.



Figure 7.15 - FE results from impacts in the top area; a) flat anvil; b) kerbstone anvil

With regard to the impact against the flat anvil (Fig. 7.15a), the model could reasonably reproduce the shape of the experimental accelerations, and provided exceptional agreement in terms of peak linear accelerations (Table 7.3). However, it can be noted that the discrepancies observed from impacts in the front region are here more pronounced. Such effect was attributed to the more pronounced doubled curvature of the shell in the crown area, which might have further altered the simulated mechanical response of the outer shell.

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Regarding impacts against the kerbstone anvil (Fig. 7.15b), it can be noted that numerical accelerations presented similar trends offered by their experimental equivalents. However, while the experimental outcomes showed a characteristic double peak shape, FE accelerations where characterised by a single peak followed by oscillations around a nearly constant value, until unloading occurred. Such effect resulted also in a quite consistent difference between numerical maximum accelerations and experimental equivalents (16% as reported in Table 7.3).

These discrepancies were mainly attributed to the assumptions made for the modelling of the composite shell using material card MAT58.

In existing similar FE results (Ghajari, 2010), analogous phenomena were observed from simulations of impacts against the kerbstone anvil on the rear area of a commercial helmet, whose shell was made of similar materials to the ones of the shell presented in this thesis. The author linked such behaviour to higher amounts of energy dissipated by the shell in FEA, compared to the one dissipated by the shell during experiments. Such conclusion was confirmed from comparison of the sequence of numerical deformation of the helmet with experimental counterparts. In FEA, the rebound of the helmet occurred slightly later than in experiments because of the more pronounced deformation of the shell, which also had a stabilising effect on the maximum accelerations transmitted to the head. On the other hand, the earlier rebound of the helmet in experiments suggested that most of the energy stored by the outer shell during the impact was released as kinetic energy during the unloading phase, resulting also in a peak of the acceleration values. In the present investigation, similar conclusions are assumed to justify the discrepancies observed in Fig. 7.15b.

Fig 7.16 and Fig. 7.17 compare the post impact deformation of the honeycomb layers with FE equivalents. As it can be noted, the model could faithfully reproduce the deformation shapes observed experimentally.

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Figure 7.16 - Post impact deformation of the front region. Comparison between FE simulations and experiments. a) impact against the flat anvil; b) impact against the kerbstone anvil



Figure 7.17 - Post-impact section of the front region. Comparison between FE simulations and experiments; a) impact against the flat anvil; b) impact against the kerbstone anvil

7.3.3 Rear region (point R)

Fig. 7.17a and b show the comparison between FEA results and experimental outcomes attained from impacts on the rear surface. As it can be noted, the model provided a very good agreement with experiments, both in terms of acceleration magnitudes and duration. Comparing the peak linear accelerations, it can be observed that the simulated values were also very close to the average recorded from experiments (Table 7.3). Best results in terms of shape of acceleration curves, magnitudes and duration over time were obtained from impacts against the flat anvil (Fig 7.17a), while best prediction

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of PLA was obtained from impacts against the kerbstone anvil (2.9% difference as reported in Table 7.3).

Comparing the deformation shapes of post-impacted honeycombs with experimental equivalents (Fig. 7.19 and Fig. 7.20), it can be also noted that the model could faithfully reproduce the deformation shapes observed experimentally.



Figure 7.18 - Acceleration histories from impacts at v = 7.5 m/s in the rear area, comparison between numerical and experimental results. a) impacts against the flat anvil; b) impacts against the kerbstone anvil

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Figure 7.19 - Post-impact deformation of the rear region. Comparison between FE simulations and experiments. a) Impact against the flat anvil; b) impact against the kerbstone anvil.



Figure 7.20 - Post-impact section of the rear region. Comparison between FE simulations and experiments; a) impact against the flat anvil; b) impact against the kerbstone anvil

Table 7.3 - Maximum accelerations of the centre of gravity of the headform.Comparison between numerical and experimental results.

Impact point	Flat anvil		Kerbstone anvil	
	FE prediction	Experimental	FE prediction	Experimental
		outcome		outcome
В	199 (+ 9.4 %)	182	150 (+ 6.0 %)	141
Р	203 (-1.5%)	206	156 (- 16.0 %)	184
R	194 (- 4.0 %)	202	140 (+ 2.9 %)	136

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7.4 Conclusions

A FE model of an innovative helmet, where aluminium honeycomb is used as reinforcement material, was generated in Ls-Dyna environment. The UNECE 22.05 standard impact tests in the front (B), top (P) and rear (R) region of the helmet were simulated, and numerical outcomes were compared to experimental results attained during the present investigation. The present study is similar to recent published investigations on the FE modelling of motorbike helmets (Cernicchi et al., 2008; Ghajari, 2010). However, to the knowledge of the author of this thesis, the introduction of honeycombs in the helmet liner model is new and has not been addressed before. The mechanical behaviour of the outer shell was modelled through use of an algorithm based on a continuum damage mechanics model. The dimensions of the shell elements were chosen on the base of existing FE results. Material properties of the shell components were obtained from tests on representative flat coupons and provided by the helmet manufacturer. Analogously to existing FE researches (Chang et al., 2001; Kostoupoulos et al., 2002; Cernicchi et al., 2008), the polymeric liner parts of the helmet were modelled as isotropic materials, and their mechanical behaviour was modelled through use of the semi-empirical equations proposed by Gibson and Ashby (1997). These equations were calibrated with experimental results obtained in the present analysis from compressive tests on expanded polystyrene samples. The foam model provided good agreement with experimental observations.

The mechanical response of the honeycomb layers was approximated through use of a material algorithm based on piecewise linear elasto-plasticity principles. The honeycomb alloy material properties were retrieved from available data in literature, and the FE model was validated against experimental tests performed in the present investigation, on aluminium honeycomb samples. Good agreement was observed between numerical and experimental outcomes.

The model could realistically reproduce the impact response of the prototype helmets tested in this investigation, for the three evaluated loading sites and the two anvils used. Particular good agreement with experimental results was observed from impacts on the front and rear region, against the kerbstone anvil.

However, FE results related to impact in the crown region highlighted the limitation of the strategy adopted in the present research, and although the prediction of the maximum accelerations fall well within the range of values recorded experimentally,

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further work is needed to improve the modelling of the helmet. The discrepancies were attributed to the use of composite materials properties obtained from tests on flat coupons for the modelling of the shell, which are known to alter the actual shell load spreading capabilities. The validation of the shell model over tests performed on doubled curvature composite materials could improve the accuracy provided by the helmet model. In addition to this, failure due to delamination and sensitivity of the composite materials to strain rate are not included in the material model used in this investigation. This is believed to have further contributed to increase the differences between numerical and experimental outcomes. However, the modelling of delamination would have resulted in excessive computational time costs, and previous studies on the FE modelling of motorbike helmets (Kostoupoulos et al., 2001) showed that in carbon, Kevlar and glass fibre epoxy composites, delamination failure takes approximately only 10% of the total impact energy absorption share. With regard to the strain rate sensitivity of laminate composites, no specific material algorithms are currently available in Ls-Dyna, although some energy based material models including strain rate effect are under development (Iannucci and Ankersen, 2006).

7.5 Publications

The work presented in this chapter resulted in the following publication:

 Caserta, G., Iannucci, L., Galvanetto, U. "The use of aluminium honeycombs for the improvement of motorbike helmets". *Proceedings of the 9th International Conference on Sandwich Structures (ICSS 9, Caltech)*, 14th – 16th of June, Pasadena, California (2010); 160 Chapter 8 . Conclusions

Chapter 8 Conclusions

Although current helmets have been optimised to offer best protection to the wearer, more work is needed to overcome the difficulty of reducing the occurrence of motorcyclist's fatalities. Indeed, statistical data showed that motorcycle riders are still among the most vulnerable of road users, and that a considerable number of deaths is due to occurrence of severe or fatal head injuries. It is generally believed that an increase of the energy absorbed by helmets of 30% would reduce by 50% the occurrence of severe or fatal head injuries in case of accident (COST 327, 2001).

The main aim of this research work was to increase the protection offered to the head through the use of non-conventional materials, capable of more energy absorption than the one offered by EPS foams, and able to keep the accelerations transmitted to the head at safe levels. In addition to this, it was in the scopes of the present research not to significantly increase the weight of the helmets, in order not to compromise comfort provided to the wearer or increase their rotational inertia. This could potentially result in the transmission of higher loads to the neck, or higher rotational accelerations to the head, which are known to cause the most severe head injuries (Gennarelli et al., 1993).

The solution proposed in the present thesis consisted in the substitution of parts of the EPS energy absorbing liner with layers of hexagonal aluminium honeycomb of similar overall density. Such strategy was inspired by previous studies on the compressive behaviour of aluminium foam-filled tubes (Hannsen et al., 2000; Kavi et al., 2006). It was shown that such materials can provide higher energy absorption levels than the sum of those of the foam and the tubes considered alone, due to an *interaction effect* between the two components.

It was therefore believed that the combination of honeycombs and foams as two-layered structures, which is one of the main subjects of this thesis, could provide similar advantages to the ones offered by foam filled aluminium tubular structures. In two-
layered materials, the interaction effect could consist in the penetration of the honeycomb cell walls in the polymeric liner or friction between the two materials, or a combination of both. Another advantage of the solution proposed in this thesis is the simplicity of the concept, which does not require excessive costs of production and can be easily implemented in helmet manufacturing processes.

Aluminium honeycombs have been extensively used in a wide range of applications as core of sandwich panels, including vehicle crash tests, aeronautic and space structures (Goldsmith et al., 1997; Papka and Kyriakides, 1999a,b). In this thesis, the compressive behaviour of aluminium honeycomb was reviewed in chapter 2. The special feature of such materials is their mechanical response when subjected to pure compressive loadings along the tubular direction, which consists of a progressive buckling of the cell walls. In a typical stress versus strain curve, such behaviour is represented by fluctuations of the stress around a constant value, which endures up relatively high deformation strains (typically 80%). This results in consistent energy absorption per unit volume. Experimental and numerical studies on the dynamic compression of aluminium honeycombs also indicated that such materials exhibit to some extent strain rate sensitivity, often manifested as an increase of the crush strength and Young's modulus with increasing loading speed, while the duration of the plateau regime remains unchanged. Other studies (Hong et al., 2006; Hong et al., 2008), indicated that such materials can still provide excellent energy absorption even when loaded under inclined loadings with respect to the out-of-plane direction. This is due to the fact that the development of shear stresses in the honeycomb structure compensates the loss of compressive strength as the loading inclination becomes more pronounced. Some experiments (Hong et al., 2006) indicated that the energy absorption depends also on the orientation of the shear component of the resultant applied force with respect to the inplane direction. In particular, it was found that when the shear force is aligned with the L direction the honeycombs offer maximum energy absorption rates. Because of such characteristics, honeycombs are among the best candidate materials for the improvement of the safety levels provided by current helmets. Alternative solutions could be the design of liners entirely made of aluminium honeycombs or the use of EPS foams with higher densities, with respect to current foams used for the manufacturing of helmets. However, a full honeycomb liner would not provide multi-directional protection to the head, due to the anisotropic nature of honeycombs and the manufacturing difficulties in giving them doubled curvature shapes. Indeed, one of the main drawbacks

of aluminium honeycombs for energy absorption applications is the poor resistance offered when loaded "in-plane", which is typically two orders of magnitude lower than the one offered when loaded along the tubular direction. In addition to such disadvantages, scalp injuries or skull fracture could occur from direct contact of the honeycomb cell walls with the head in case of impact. On the other hand, the use of higher density foams would lead to higher energy absorption at expenses of the increase of the accelerations transmitted to the head. Thus, the partial substitution of a layer of EPS foam with a layer of honeycomb is a way to exploit the honeycomb characteristics without losing multi-directional protection provided to the head.

8.1 The two-layered materials

The next step of this research was the study of the compressive behaviour of honeycomb and EPS foam combined as two-layered structures (chapter 4). The main aim of this work was to quantify the energy absorption provided by the two-layered configurations and compare it with the one offered by current EPS foams used for helmet manufacturing. The lay-up of the materials and weight of their components were such as their overall density was equal to the one of the EPS foams to which they were compared. Quasi-static standard compressive tests and impact compressive tests were performed on cubic representative samples. Experimental results obtained from tests on EPS foam specimens were in agreement with existing outcomes reported in literature. Two-layered materials exhibited strong non-linear characteristics. Deformation mechanisms under quasi-static loads consisted in the crushing of the bottom EPS foam layer, followed by the collapse of the upper aluminium honeycomb. This was mainly due to the consistent difference in strength between the two materials. No interaction was observed between the two materials at this stage of the experimental testing. During impact tests instead, high speed camera recordings indicated that small penetration of the honeycomb layers on the underlying foams occurred. This mechanism was believed to have further contributed to the energy absorption provided by two-layered materials. However, in the present investigation it was not possible to quantify the amount of energy dissipated through this phenomenon because of time research constraints. Comparison of energy absorption with EPS foams suggested that two-layered configurations can provide increased energy absorption ranging from 18.5% to 39.1% for quasi-static loadings, and from 22.6% to 40% for impact loadings.

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The experimental results obtained at this stage of the research were later used to validate a finite element model representative of the two-layered materials tested, and described in chapter 5. Foams were modelled as solid blocks while aluminium honeycombs were modelled as an array of two-dimensional plans nested together to form the hexagonal honeycomb structure. The mechanical properties of the foams were implemented through use of existing semi-empirical equations (Gibson and Ashby, 1997) fitted to the experimental data obtained during experiments described in chapter 4. Numerical outcomes from simulations of quasi-static and impact compressive tests on EPS foams resulted in good agreement with experimental outcomes, and in line with what reported in similar earlier FE analyses (Cernicchi et al., 2008; Ghajari, 2010). However, the results were found to be slightly dependent on mesh density. Best agreement was obtained when using solid tetrahedral element size approximately equal to 15 mm. The mechanical properties of the honeycomb were modelled by implementing the material properties of the bulk aluminium alloy (Al 3003 H18) used for the manufacturing of the honeycombs. Strain rate effects were taken into account through use of the Cowper-Symonds model (Kazimi, 2001). Initial imperfections were modelled through introduction of geometrical distortions replicating the lowest natural deformation mode of the honeycombs, in line with existing FE studies (Mohr and Doyoyo, 2004). Quasi-static and impact compressive simulations indicated that the FE model tends to give predictions that are in agreement to the experimental data from corresponding tests. However, some discrepancies were observed for compressive loadings along the L and T direction. Such discrepancies were attributed to the addition of excessive non-structural mass for quasi-static analyses. Numerical results were also found to be influenced by mesh size. In this study, use of shell element size equal to 0.3mm provided best agreement between numerical and experimental outcomes. The methodology and results presented in chapter 5 were later used for the FE modelling of the prototype helmets tested in the present investigation (Chapter 7). An original and alternative approach for FE modelling of two-layered materials as representative unit volumes was also proposed in this thesis, to reduce computational cost. Results obtained from simulations on the unit cell model were in good agreement with experimental counterparts. Further details regarding the methodology adopted were reported in Appendix C.

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8.2 The helmet prototypes – experimental testing

The impact behaviour of a modified version of a commercial helmet, where aluminium honeycombs were introduced in the front, top and rear region of the energy absorbing liner, was assessed following UNECE 22_05 standards. Unmodified helmets, presenting same geometry and material properties (except for the honeycomb inserts), were also tested under the same conditions. The dynamical responses were compared and the peak linear acceleration and the Head Injury Criterion were used as evaluation criteria.

Comparing the results from different impact sites and anvils used, different trends were observed for the two evaluated helmet designs. Generally, the prototype helmets provided better protection to the head from impacts against the kerbstone anvil, in particular by significantly reducing the PLA and HIC during impacts on the front and the rear surfaces. Conversely, results from impacts against the flat anvil indicated to some extent the limitations of the strategy adopted in this research. Indeed, when impacts were performed against the flat anvil the prototype top area provided best protection to the head, in terms of HIC. No significant improvements were observed from impacts on the front region, while impacts on the rear region highlighted inferior performances in comparison with the ones offered by the helmet commercial design. From observations of deformed prototype liners, it was concluded that the honeycombs in the front and rear areas did not crush completely and so gave only minor contributions to the impact energy absorption, which was therefore carried out by the outer shell and the polymeric liner. This was attributed to a non uniform contact between the outer shell and the honeycombs during the impacts, to strain rate effects, which increased the crush honeycomb resistance, and to a non optimum design of the prototype liner. It must be noted that due to research time and budget restrictions, the manufacture of the prototype helmets was carried out following a non-industrialised process prone to imperfection. Moreover, such constraints did not allow for more prototypes to be made, so that there was no possibility to carry out an optimisation of the prototypes. Surprisingly, significant reductions of the PLA and HIC were observed from impacts on the lateral surfaces, not modified because of manufacturing difficulties, against both the anvils. It was assumed that the presence of honeycombs and the hollows in the liner might have influenced the load spreading capabilities of the helmet, and so the energy absorption. On the basis of the results presented in chapter 5 it can be concluded that the use of aluminium honeycombs, as reinforcement material for the

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energy absorbing liner, can lead to an improvement of the safety levels provided by current commercial helmets without increasing their weight.

8.3 The helmet prototypes – FE modelling

The final stage of the research presented in this thesis consisted in the generation and validation of a FE model representative of the helmet prototypes. This study is similar to recent published investigations on the FE modelling of motorbike helmets (Cernicchi et al., 2008; Ghajari, 2010). However, to the knowledge of the author of this thesis, the introduction of honeycombs in the helmet liner model is new and has not been addressed before. The mechanical behaviour of the outer shell was modelled through use of an algorithm based on a continuum damage mechanics model. The dimensions of the shell elements were chosen on the base of existing mesh convergence results (Cernicchi et al. 2008; Ghajari, 2010). Material properties of the shell components were obtained from tests on representative flat coupons and provided by the helmet manufacturer. Analogously to existing FE researches (Chang et al., 2001; Kostoupoulos et al., 2002; Cernicchi et al., 2008), the polymeric liner parts of the helmet were modelled as isotropic materials, and their mechanical behaviour was modelled through use of semi-empirical equations proposed by Gibson and Ashby (1997). These equations were calibrated with experimental results obtained in the present analysis from compressive tests on expanded polystyrene samples, and discussed in Chapter 4. The mechanical response of the honeycomb layers was approximated through use of a material algorithm based on piecewise linear elasto-plasticity principles. The honeycomb alloy material properties were retrieved from available data in literature, and the FE model was validated against experimental tests performed in the present investigation (Chapter 5). The prototype model could realistically reproduce the impact response of the prototype helmets tested in this investigation, for the three evaluated loading sites and the two anvils used. Particularly good agreement with experimental results was observed from impacts on the front and rear region, against the kerbstone anvil. However, FE results related to impact on the crown region highlighted the limitation of the methodology adopted for the modelling of prototype helmets, and although the prediction of the maximum accelerations was reasonably in agreement with the range of values recorded experimentally, further work is needed to attain a more realistic behaviour. The discrepancies were attributed to the use of composite materials properties obtained from

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tests on flat coupons for the modelling of the shell, which are known to alter the actual shell load spreading capabilities (Mills et al., 2009). The validation of the shell model over tests performed on doubled curvature composite materials could improve the accuracy provided by the helmet model presented in this thesis. In addition to this, failure due to delamination and sensitivity of the composite materials to strain rate are not included in the material model used in this investigation. This is believed to have further contributed to increase the differences between numerical and experimental outcomes. However, the modelling of delamination would have resulted in excessive computational time costs, and previous studies on the FE modelling of motorbike helmets (Kostoupoulos et al., 2001) showed that delamination failure takes approximately only 10% of the total impact energy. With regard to the strain rate sensitivity of laminate composites, no specific material algorithms are currently available in Ls-Dyna, although some energy based material models including strain rate effect are under development (Iannucci and Ankersen, 2006). 167 Chapter 9. Recommendations for future work

Chapter 9 Recommendations for future work

To the knowledge of the author, the use of aluminium honeycombs for reinforcement of motorbike helmets has not been explored in previous research. The study presented in this thesis is novel and provides significant contribution to the passive safety research field. The results presented in this thesis demonstrate the feasibility of using aluminium honeycombs as reinforcement materials for motorbike helmets and could provide the framework for future research, focused on the improvement of the protection provided by Personal Protective Equipment (PPE). However, further work is necessary to improve some aspects of the methodology adopted in the present investigation, and to assess the potential of two-layered materials and prototype helmets under a wider set of loading conditions.

9.1 Investigation of two-layered honeycomb-foam structures

The coupling of aluminium honeycombs and EPS foams as two layered structures provided higher energy absorption amounts than those offered by EPS foams of equivalent densities tested under the same conditions. However, it is necessary to assess the extent to which the interaction effect between the two materials might contribute to the overall energy absorption provided by two-layered configurations. Moreover, in the present investigation the mechanical response of two-layered materials was assessed only under compressive loadings. The evaluation of the mechanical behaviour under shear loading or compressive dominant loadings is necessary to achieve a full understanding of the potential offered by two-layered configurations, especially with regard to protection of the head. 168 Chapter 9 . Recommendations for future work

In addition, the number of configurations proposed in this thesis was relatively limited. It would be of interest to carry out an optimisation of the two-layered materials and explore their behaviour when a wider variety of foam densities and honeycomb typologies is considered.

9.2 Finite element modelling of two-layered materials

The FE model of the two-layered materials proposed in this research work predicted the trends observed experimentally with a good level of accuracy. However, the use of mass scaling factors for quasi-static simulations in Ls-dyna environment is critical for the reproduction of the mechanical behaviour of materials, and excessively high factors might lead to unrealistic behaviours. Some results presented in this thesis suggested that structural mass should be carefully optimised to reduce to minimum computational costs, while attaining a realistic mechanical behaviour. Conventional static finite element analysis could be used to simulate static tests in a more effective way.

Moreover, due to time research constraints, the honeycomb model was not validated against the shear experimental results presented in this thesis, and failure was not considered. This might have also had consequences on the simulation of the prototype tested in the present analysis. It is therefore suggested that future research shall address these topics.

9.3 Prototype helmet design

The introduction of aluminium honeycombs in motorbike helmets may lead to significant improvements of the protection offered to the head. However, the number of prototype tested in the present investigation was relatively restricted, and results showed consistent variability in some cases (Table 5.3). Moreover, impacts against the flat surface highlighted the limitations of the strategy adopted in the present investigation.

Strain rate has some influence on the crush strength offered by honeycombs, and so on the maximum accelerations transmitted to the head. In the present analysis, the honeycombs used to reinforce the helmet liner did not crush completely, although their stiffness was very similar to the one of the EPS foam liner. This resulted in maximum 169 Chapter 9 . Recommendations for future work

accelerations higher than those transmitted by commercial helmets. It was believed that the enclosure of the honeycombs between the outer composite shell and the polymeric liner might have caused entrapment of the air within the honeycomb structure, therefore resulting in an increase of the honeycomb stiffness during the impacts.

Future work should be addressed to the followings:

- Testing of higher number of prototypes to achieve more consistent statistical information;
- Optimisation of prototype helmets for impacts against flat surfaces.
- Design of helmets where the gap between the outer shell and the underlying liner is reduced to a minimum;
- Determine the extent to which the air trapped within the honeycomb structure affects the prototype dynamic response;
- Investigation of the prototype deformation mechanisms through use of advanced investigation techniques, such as for example the x-ray high speed cameras used by Bosch (2010);
- Assessment of the prototype impact protection when more severe impact conditions or different standard regulations are considered;
- Extension of the areas covered by the honeycombs to the remaining surface of the liner, including the lateral surfaces;

Most of these subjects could be addressed through use of FEA. The model proposed in the present investigation can be used to achieve this aim. However, such model needs some minor improvements, which can be resumed as follows:

- Characterisation of the outer shell through use of mechanical properties obtained from tests performed on outer shells taken singularly, rather than flat representative specimens, as suggested by Mills et al. (2009);
- Improvement of the honeycomb model by including damage and shear failure mechanisms;

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183 APPENDIX A. Head injuries, PLA and HIC

APPENDIX A Head injuries, PLA and HIC

Head injuries can be generally defined as temporary or permanent damage of the head or one of its components (Suaeib et al, part I, 2002). Head injuries can be distinguished in four types: scalp damage, skull fracture, brain damage and neck injury.

Neck injuries are found to occur with very low frequency compared to other injuries (Hume et al., 1995; Cost 327, 2003) and scalp damage can be considered far less serious than skull fracture and brain damage.

Skull fracture might occur from impacts against rigid objects such as tree branch, road posts, motorcycle parts, etc. However, in general the outer shell of the helmet is able to prevent skull fracture by spreading the impact load over a larger area (assuming that the impact occurs within the helmet coverage area).

Nevertheless, the head is subjected to a combination of translational and rotational accelerations, which induce stress and deformations of the brain.

Previous studies on the biomechanics of the head have shown that different injury typologies can be associated to transmission of linear or rotational accelerations (Holbourn et al., 1943; Gennarelli et al., 1983; King et al., 2003). The main injury mechanisms related to linear accelerations are:

- generation of pressure gradients through deformation of the skull;
- propagation of pressure waves through the brain;
- relative movements between the brain and the skull.

The effects of linear accelerations are only local (King et al., 2003) and mainly consist of fractures of the skull, concussions and haemorrhages (Gennarelli et al., 1987, Gurdjian, 1972). The effects of rotational accelerations can be either localised or diffused. Rotational accelerations were considered for first time by Holbourn (1943), who hypothesised that rotational accelerations induce shear and tensile strain to the brain, resulting in concussions and countercoup contusions. Gennarelli et al. (1983, 1987), from experimental tests on live primates, confirmed that rotational accelerations play a major role in the generation of diffused axonal injuries (DAI), concussions and subdural

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haematomas (SDH). DAI consists is a distributed damage of the axonal components of neurons in the brain structure, while SDH is defined as "Bleeding into the space between the dura (the brain cover) and the brain itself" (www.MedicineNet.com).

DAI and SDH are currently considered amongst the most severe head injuries (Gennarelli, 1983). With particular reference to DAI, it is reported that such injury is one of the most common injuries observed in motorcycle accidents (Bandak and Eppinger, 1994), and often results in fatalities, permanent vegetative state or permanent impaired conditions (www.Wikipedia.org). SDH introduce pressure gradients to the head, of which the effects might consist of dizziness, slurred speech and may progress to coma and eventual death, depending on the severity of the haematoma.

Different head injury predictors based on both linear and rotational accelerations have been proposed and used by most researchers and standard tests in the past decades, due to the ease with which such accelerations can be measured during experiments.

Other injury predictors had been also formulated based on evaluation of stress and strains developed within the brain structure.

However, in this section only injury predictors based on measurements of translational accelerations are briefly discussed, since such indexes are used for the evaluation of the dynamical response of the prototype helmets designed during the present research (Chapter 7 and 8).

A.1 Peak Linear Acceleration

The Peak Linear Acceleration (PLA) is defined as the maximum value of the resultant acceleration transmitted to the centre of gravity of the headform during standard tests.

A.2 Head Injury Criterion

Besides the maximum tolerable accelerations, it is known that the severity of head injuries is also proportional to the duration of the accelerations transmitted to the head (King, 2000), so that low accelerations sustained by the head for a relatively long time interval can have more severe consequences than high accelerations sustained for short times. To establish a relationship between accelerations and their respective time intervals required to produce skull fracture, the Wayne State Tolerance Curve (WSTC) was generated on 1960 (Lissner et al., 1960). This curve (Fig. A.1) was first built on the 185 APPENDIX A. Head injuries, PLA and HIC

base of experimental data obtained from tests on human cadavers and was later enriched by results obtained from tests on embalmed animals (Patrick, 1965).



Figure A.1 - Wayne State Tolerance Curve (from McElhaney, 1976)

The WSTC was used by Versace (1971) to define the Head Injury Criterion (HIC):

$$H.I.C. = \left[\frac{1}{t_2 - t_1} \int_{t_2}^{t_1} a(t) dt\right]^{2.5} (t_2 - t_1)$$
(1.1)

where a(t) is the resultant translational acceleration at the time t measured in g, t_1 and t_2 are the times (in seconds) of beginning and ending of the time interval, chosen in such a way to make the HIC maximum. The maximum time interval $t_2 - t_1$ is limited to 15 ms for practical reasons (NHTSA, 1998). According to Horgan (2005), life threatening injuries are likely to occur for values of the HIC between 1000 (16% probability) and 3000 (99% probability).

The use of the HIC as evaluation criteria of head injuries is still widely accepted by most researchers and standard regulations. However, its use had been criticised (Gennarelli, 1987) because of the fact that the HIC does not take into account rotational accelerations, which as explained earlier, represent a major head injury mechanism. 186 APPENDIX B. FE modelling of two-layered material. Mesh convergence results

APPENDIX B FE modelling of twolayered materials. Mesh convergence results

In this appendix, the FEA results of a mesh convergence study performed using the honeycomb and EPS models are presented.

Four different mesh densities were generated to assess the element dimensions that provide best agreement between numerical and experimental results. The mesh densities were named as *coarse, medium, fine* and *extra fine*, on the base of the average dimension of each element, as listed in Table B.1. The models were generated according to procedures provided in Chapter 5.

Force versus displacement curves were generated for each mesh density and compared with experimental results reported in Chapter 4. With regard to EPS foams, comparison was made with experimental impact outcomes reported in section 4.4.1.

With regard to aluminium honeycomb, comparison was made with experimental outcomes obtained from quasi-static compressive tests applied along the two honeycomb in-plane principal directions and from impacts loads applied along the tubular direction (section 4.4.2).

A sequence of snapshots taken at different stages of the deformation is also provided for all the mesh densities. 187 APPENDIX B. FE modelling of two-layered material. Mesh convergence results

	EPS foam		Honeycombs	
Mesh type	Number of	Average length	Number of	Average
	elements	[mm]	elements	length [mm]
Coarse	104	20	38,592	0.92
Medium	230	16	154,370	0.46
Fine	475	10	358,640	0.3
Extra fine	3022	5	617,480	0.2

Table B.1 - Honeycomb and foam model mesh densities

B.1 EPS foams

Fig. B.1 a-c show the FE results obtained from simulations on EPS foams. In general, it was observed that the coarser the mesh, the higher the load required to crush the foam. However, this effect was minimum for the range of mesh densities assessed. Table B.2 illustrates the distribution of the Von Mises stresses in the foam model for the four evaluated mesh densities, at different deformation stages. As it can be observed, medium, fine and extra fine meshes showed a similar and uniform distribution of the stress through the material structure, except of the bottom region. Coarse mesh exhibited non-uniform stress distributions instead.

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Figure B.1 - Effect of mesh density on numerical response of EPS foams subjected to impact compressive loadings. a) EPS 40kg/m³; b) EPS 50 kg/m³; c) EPS 60 kg/m³

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Numerical results suggest that convergence is obtained for elements whose minimum average length is equal to 16mm. Therefore, medium mesh density provides best agreement with experimental results, whilst keeping minimum computational time.

B.2 Honeycombs

The results obtained from FEA of quasi-static loadings applied along the in-plane directions are showed in Fig. B.2 a-b. It is evident that mesh size had a significant influence on the numerical outcomes. With reference to the in-plane loading case, it was observed that use of coarse elements resulted in significant overestimation of the load required to crush the honeycomb (up to two times the one recorded experimentally). Fine mesh provided best agreement with experimental results. Use of finer meshes resulted in higher load values compared to experimental counterparts. With regard to FE simulations of out-of-plane compressive response (Fig. B.3), it can be noted that all the assessed mesh densities provided similar trends to the ones observed experimentally. However, Fig. B.3a suggests that use of coarse mesh resulted in an extended initial peak load, which leads to significant overestimation of the energy absorbed by the honeycomb. In addition, snapshots of the deformation sequence of honeycombs (Table B.3) highlighted unrealistic deformation shapes. Conversely the use of medium mesh resulted in a more realistic reproduction of the deformation shapes observed experimentally, at the expenses of an erroneous prediction of the crushing load required to crush the honeycomb. Use of fine and extra fine mesh resulted in a very good agreement with experimental counterparts.

It was concluded that use of fine mesh provides best results for both in-plane and outof-plane loading conditions.

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Figure B.2 - Effect of mesh density on numerical response of honeycombs subjected to in-plane compressive loadings. a) Load along W-direction; b) load along L-direction



Figure B.3 - Effect of mesh density on the numerical response of honeycombs subjected to out-of-plane compressive loadings

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Table B.3 - Influence of the mesh size on the stress distribution on the honeycomb model

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APPENDIX C The unit cell model

General techniques for exploiting periodicity and symmetries in micromechanics of composite materials have been applied to find the smallest unit cell to be used as model for the two-layered materials described in chapter 4. To reduce computational costs, the inner symmetries of the unit cell chosen were exploited to find an even smaller region to be used for the analyses. Fig. C.1a shows a top view of the unit cell and the subcell chosen for this investigation. These shapes were chosen similar to the ones adopted in previous studies on the modelling of honeycomb panels as small representative volumes (Yamashita et al., 2005; Asadi et al., 2006).

A rigid stonewall with prescribed motion was used to simulate a quasi-static compressive loading. A second rigid stonewall was used as base for the hybrid model. Forces recorded from the fixed rigid wall were plotted against the vertical displacement of the moving rigid wall, and compared with those obtained from experimental tests.

C.1 Mesh

The honeycomb model was generated using four-noded shell elements, while the foam was modelled using tetrahedral constant stress solid elements. Due to the complex mechanical behaviour of honeycombs, three through-thickness integration points were defined for each shell element. The dimensions of the honeycomb elements were chosen according to results of the mesh convergence study carried out during the present investigation and reported in Appendix B. The thickness of the shell elements was chosen equal to the actual thickness of the aluminium foils used for the manufacturing of the honeycombs used in this investigation (t = 75 micron). The dimensions of the solid elements used for the foam were chosen referring to a study conducted at Imperial College on the influence of the element dimensions in the numerical analysis of EPS

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foams (Cernicchi et al., 2008). To simulate the penetration of the honeycomb in the foam, phenomenon observed during impact tests, the foam elements aligned with the honeycomb cell walls were bisected by planes coincident with the honeycomb cell walls, up to one third of the total thickness of the foam model. As result of this operation, couples of unmerged nodes occupying the same position were generated, as highlighted by small black circles in Fig. C.1b



Figure C.1 - Hybrid unit cell and subcell models. a) top view; b) perspective view

C.1.1 Pre-crush of the honeycomb

The honeycombs used for the experiments were pre-crushed using a quasi-static standard compressive machine. The aim of this work was to facilitate the plastic collapse of the honeycomb layer during the compression of the hybrids. This condition was reproduced in the model as a distortion of the honeycomb geometry. In particular, a preliminary numerical compression was applied to the honeycomb unit cell alone. The force history was recorded and the deformation of the honeycomb was observed. In the post processing phase, it was generated an output file containing the honeycomb node coordinates at the stage of the deformation in which the honeycomb began collapsing plastically. This file was then introduced in the hybrid model as initial geometry, so that the honeycomb presented the desired pre-crushing effect. Fig.2 shows the result of this operation.

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Figure C.2 - Pre-crush effect

C.2 Contact

Due to the interaction between very thin shell elements and thick solid elements, the correct choice of the contact logics was crucial for the correct reproduction of the mechanical behaviour of hybrids. Four contact algorithms were used in this model. The keyword AUTOMATIC_NODES_TO_SURFACE was used to take in to account the first contact between the edges of the honeycomb cell walls and the surface of the foam. Preliminary analysis performed without this contact card showed a sudden failure of the contact when honeycomb edges touched the foams surface. The contact logic AUTOMATIC_SURFACE_TO_SURFACE was used to model the sliding of the honeycomb inside the foam. Some difficulties arose from the consistent difference between the element dimensions and stiffness, which leaded to unrealistic behaviours. To overcome this problem, SOFT = 1 penalty option was activated for all of the contact logics mentioned and the soft penalty scaling factor was set equal to 0.1.

The contact logic AUTOMATIC_SINGLE_SURFACE was used to correctly reproduce the progressive folding of the honeycomb, without incurring in the self-penetration of the cell walls.

The INTERIOR contact algorithm was used to avoid self-penetration (and so negative volumes in the model) of the foam elements, when high local deformations occurred.

C.3 Boundary conditions

C.3.1 Unit cell

To introduce these constraints in Ls-Dyna, additional coordinate systems were defined. Figure C.3 shows a top view of the unit cell and the reference systems defined. In particular, the boundary conditions prescribed to the nodes lying in the face 2 were

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the same as those applied to nodes on the face 3 and referred to the coordinate system I. In the same way, boundary conditions prescribed for the nodes on the face 1 were also applied to nodes lying on the face 4, with reference to the coordinate system II. Please note that in all of the systems, the z axes are perpendicular to the plane of the page, the positive direction being pointing to the reader. Further boundary conditions were applied to the central nodes (highlighted by circles) and to all of the nodes at the corners and the bottom of the hybrid.

The constraints applied to the set of nodes lying on the corners and those in the centre of the unit cell (highlighted by a circle in Fig.3) are referred to the coordinate system G.

For the nodes on the faces 1-4, the following degrees of freedom were removed:

- displacement along y;

For the nodes in the centre and the nodes in the corners, the following degrees of freedom were removed:

- displacement along x;
- displacement along y

Concerning the nodes at the bottom of the hybrid, it was chosen to eliminate any degree of freedom.



Figure C.3 - Unit cell and sub-cell local coordinate systems

C.3.2 Sub-cell
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The same boundary conditions prescribed for the unit cell were applied to the nodes lying on the sides 3 and 4, the nodes in the centre, the nodes at the corners and at the bottom of the hybrid. The only difference consists in the removal of the translational degree of freedom along y for the nodes lying in the face 5, with reference to the coordinate system G.

C.4 Material properties

The isotropic material model CRUSHABLE_FOAM was used to model EPS properties, while PIECEWISE_LINEAR_PLASTICITY was used to model the honeycomb material properties. Table C.1 shows the mechanical properties introduced in the model.

Mechanical property	Material				
]	EPS foan	Aluminium		
			3003 H18		
Density [kg/m ³]	40	50	60	2730	
Young's modulus [MPa]	16	16	24	6890	
Poisson Ratio	0.01	0.01	0.01	0.33	
Cut-off tension [MPa]	0.21	0.32	0.42	-	
Yield stress [MPa]	-	-	-	186	
Plastic hardening modulus [MPa]	-	-	-	5.5	

Table C.1 - Honeycomb and foams mechanical properties

The foam material properties were obtained from quasi-static compressive tests performed on EPS foam samples at Imperial College London. The aluminium properties were instead obtained from the online database <u>www.matweb.com</u>.

C.4.1 The strain rate effect

The aluminium strain rate dependence was modelled by introducing an arbitrary curve in the material card adopted for this investigation, showed in Fig. 4. This curve represents the scaling factor to be applied to the quasi-static crush strength of the bulk material with which the honeycomb is made, to obtain the correspondent dynamic crush strength when the strain rate is known. A bilinear law was used, on the base of Cellbond Composites experience. Case 1:21-cv-02112-CMA-SKC Document 40-8 Filed 09/15/21 USDC Colorado Page 218 of 224

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Figure C.4 - Strain rate effect

C.5 Loading conditions

In the FE analyses, the loading conditions were simulated using a rigid moving stonewall with prescribed speed equal to the compressive rate (2 mm/min) adopted for the experiments. A second rigid stonewall was placed underneath the hybrid. The mass of the model was scaled to increase the minimum time step and so reduce computational time. Force versus displacement curves were plotted from post-processing procedures and compared with those obtained from tests on real specimens. A scaling factor, equal to rate of the cross-sectional area of the two-layered materials used for the experiments and the cross-sectional area of the unit cell, was applied to take into account the difference between the model scale and the real dimensions of the specimens used.

C.6 Results

Figures 5a, 5b and 5c show force versus-displacement obtained from FE simulations for each of the hybrid configurations tested, in comparison with those recorded from experiments. In addition, a lateral view of a hybrid unit cell with the loading direction is showed. The experimental curves are the mean of five tests results carried out on each of the hybrids treated.

As it can be seen, there is good agreement between the results obtained from numerical analyses and those obtained experimentally, confirming the pertinence of the contact logics, boundary conditions and material properties chosen. Numerical results indicate a little increase in the slope of the linear force curve at the beginning of the compression. This might be due to a slight excessive coefficient of friction adopted in the contact NODES_TO_SURFACE. Case 1:21-cv-02112-CMA-SKC Document 40-8 Filed 09/15/21 USDC Colorado Page 219 of 224

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Figure C.5 - Force versus displacement curves. a) Configuration 1; b) Configuration 2; c) Configuration 3

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It can be also noted that the numerical curves present a higher peak in the force value before the series of oscillations prior the densification of the specimen, with respect to that showed by experimental data. This phenomenon was found to be strongly dependent on the initial pre-crush of the honeycomb.

C.6.1 Deformation shapes

During experiments, it was observed that hybrids subjected to compressive loads deform following a precise sequence:

- Elastic deformation of the foam;
- Plastic collapse of the foam;
- Densification of the foam and elastic buckling of the honeycomb;
- Plastic collapse of the honeycomb
- Densification of the hybrid

Fig. 6 shows a side view sequence of the deformation of the hybrid 3 unit cell. The deformation shapes showed by other hybrids are not reported, since they are similar to those illustrated in the figure. The deformed shapes of hybrids were recorded when the compressive displacement was equal to 5, 10, 15, 20, 25 and 30 mm. A picture showing the fully densificated hybrid was also included.





Results showed very good agreement with experimental observations, confirming the pertinence of the shape of the unit cells and the boundary conditions chosen. In

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particular, the latter had a crucial influence in the correct reproduction of the deformation modes.

It can be also noted that the use of unmerged nodes allowed the representation of the penetration of the honeycomb in the foam.

Results from analyses of sub-cells presented similar results.

C.7 Conclusions

Innovative composites made of aluminium honeycombs and EPS foams were modelled as unit cells in Ls-Dyna environment. A sub-model was also created to further reduce computational costs. Results from numerical analyses showed good agreement with experimental observations. In addition to this, the simulated deformation shapes of the two-layered materials were in substantial agreement to those observed during experimental tests. It was concluded that both the unit cell and the sub-cell could be used for the prediction of the quasi-static compressive behaviour of hybrids, when the boundary conditions are applied. The use of the contact logic correct AUTOMATIC_NODES_TO_SURFACE had a crucial role in the modelling of the contact between honeycomb and foam. The pre-crush of the honeycomb had a significant influence on the peak force recorded before the collapse of the honeycomb. The correct choice of the boundary conditions allowed the reproduction of the real deformation modes of hybrids and the use of a smaller model, which introduced a saving in the computational costs. In particular, the analysis of the sub-cell required a half of the time necessary to perform simulations using the full model.

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APPENDIX D Influence of the honeycomb strength on the impact response of the helmet prototypes

In this appendix, the FE impact response of helmet prototypes is investigated for different values of the honeycomb crush strength, ranging from 0.5 MPa to 1.5 MPa.

Fig. D.1 – D.3 represent the resultant accelerations obtained from numerical simulations, for impacts in the front, top and rear area, against the flat and kerbstone anvile prescribed by UNECE 22.05 standards. The helmet prototype model was built according to the methodology described in Chapter 6. Each honeycomb crush strength was simulated through application of a scaling factor to the thickness of the honeycomb cell walls, which was equal to the rate of the simulated crush strength over the honeycomb nominal crush strength (1.6 MPa). The geometry and material properties of the honeycombs were not modified. Maximum accelerations experienced by the headform, in g, were also recorded and compared in Table D.1.

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Figure D.1 - Acceleration histories from impacts on the front region; a) flat anvil; b) kerbstone anvil.



Figure D.2 - Acceleration histories from impacts on the top region; a) flat anvil; b) kerbstone anvil.



Figure D.3 - Acceleration histories from impacts on the rear region; a) flat anvil; b) kerbstone anvil.

From the numerical outcomes it can be noted that the honeycomb crush strength had a significant influence on the dynamic response of prototype helmets. In all the impacts, except of impacts on the top region against the kerbstone anvil (Fig. D.2b), the initial slope of the acceleration curves and maximum accelerations increased with increasing crush strength, while the duration of the accelerations became shorter. The curves in

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red are representative of the prototypes used during the experiments and were used to validate the model against experimental data (section 7.3).

From Table D.1, it can be noted that minimum accelerations transmitted to the head are achieved when the honeycomb crush strength was in the range 0.7-0.9 MPa (minimum values are highlighted as bold characters).

Honeycomb crush strength [MPa]	Flat anvil			Kerbstone anvil		
	Front	Тор	Rear	Front	Top	Rear
0.5	207.8	202.8	173.1	159.0	160.2	147.3
0.7	199.0	202.1	173.7	150.4	154.8	140.7
0.9	208.4	215.8	205.7	143.6	153.8	154.5
1.1	231.9	232.8	220.1	157.4	155.4	164.8
1.3	253.9	248.6	242.0	172.8	164.9	182.3
1.5	269.8	263.3	260.5	192.6	165.4	198.5

Table D.1 - Maximum accelerations in function of the honeycomb crushstrength, impact point and surface hit