IMPACT ENERGY ABSORPTION ANALYSIS
OF DIFFERENT THIN-WALLED TUBES
WITH AND WITHOUT REINFORCEMENT

A THESIS SUBMITTED TO THE UNIVERSITY OF MANCHESTER FOR THE
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The School of Mechanical, Aerospace and Civil Engineering
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Abstract

For an ideal impact energy absorber, the initial peak force should be low and the average crushing force should be high. Also, a long stroke and a stable force history are expected. The thin-walled tube under axial loads is a kind of energy absorber that can produce controlled progressive collapse during a crash. It is a promising collapse mechanism for energy absorption with demonstrated success in industry. But the conventional thin-walled tubes still have high initial peak force and force fluctuations during a crushing process. To help to achieve a better energy absorbing structure, a research work has been carried out in this thesis.

The aim of the present research is to achieve an improved understanding of the crushing behaviour of thin-walled tubes under axial loads. In the study, the entire crushing process, including the initial stage of collapse, its localization and the subsequent progressive folding has been carefully investigated by experiment. The relation between the localized plastic deformation and the corresponding crushing force is built by comparing the cross section of series of specimens and their load-displacement curves, which give a deep insight of the collapse mechanism of circular thin-walled tube under axial loads. Then some trigger systems are proposed, which is proved to be a good way to reduce the initial peak force and influence the collapse behaviour. To achieve higher energy absorbing efficiency, the multi-cell thin-walled tube has been investigated. Finally, based on the analysis in this study, a new multi-cell profile which is composed of coaxial tubes with different lengths and dented grooves is proposed. The new design is proved to be a good energy absorber with low initial peak force and very high energy absorption efficiency.

Keywords: specific energy absorption, average crushing force, initial peak force
Declaration

No portion of the work referred to in the thesis has been submitted in support of an application for another degree or qualification of this or any other university or other institute of learning.

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Shuo Lu
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CHAPTER 1

INTRODUCTION

Energy absorption capability of the structure is one important aspect in modern structural design, which has many applications in engineering, in particular, in the design of various kinds of energy-absorbing devices to minimize human injuries while collision is occurred in transportation systems. Even for the structure with no purpose on energy absorption, it also needs to meet integrity requirements for the sake of safety. Within all these energy absorbing structures, thin-walled metal tubes subjected to axial loads are the most employed structural elements due to their high efficient energy absorption characteristics, low costs and easy to manufacture.

Clearly, it is important to understand the actual collapse behavior of the thin-walled tube to ensure the structure can dissipate the kinetic energy in a controlled, effective energy absorbing mode. It is the main objective of this study to identify and understand the main factors that influence the collapse behavior of thin-walled metal tubes under axial loads and examine the role of each factor played in the crushing process.

This chapter begins with the introduction of some background knowledge related to energy absorption and structural crashworthiness. Then the objectives and the contributions of the thesis are presented. Finally, a brief description of the thesis structure is given.

1.1 Background of the thesis

In this thesis, the work was mainly concentrated on structural crashworthiness analysis
of thin-walled metal tubes with various cross sections, which have been widely employed in engineering as energy absorbing devices. In the design of energy absorbing devices, there are a wide range options, mainly based on their applications. Fig 1.1 shows some designs of energy absorbers, which normally can be seen in the real life. These energy-absorbing devices have different energy absorbing characteristics and can dissipate kinetic energy in a wide variety of ways like plastic deformation, friction and fracture.

![Image](a) ![Image](b)

**Fig 1.1 The design of different energy absorbers**

(a): the bumper of the car  (b): the structure in the helmet

In all these kinds of devices, thin-walled metal tubes are most popular adapted in the design of energy absorbing structures, especially in transportation vehicles, because of their efficient energy absorbing characteristics under axial crushing loads.

For thin-walled metal tubes, their energy absorption capacity will mainly depend on the amounts of plastic deformation which takes place during the axial crash. Because metal materials can absorb a large amount of kinetics energy in the course of large plastic deformation, structures whose function is to absorb mechanical energy in the event of a
dynamic collision or accident are commonly constructed of ductile metal. With regard to the loading condition, basically tubes can be subjected to lateral compression or axial compression. The lateral compression modes can produce the relatively low force-long stroke deformation characteristics. When subjected to an axial load, these tubes can collapse either in symmetrical buckles or in non-symmetric patterns. These kinds of collapse modes are expected to provide a relatively long stroke and a fairly stable reaction force throughout the entire crushing process, which makes them inherently suitable for use in the design of energy absorbing devices.

In this study, structural crashworthiness of thin-walled metal tubes is the main concern. The term crashworthiness refers to the ability of a structure to protect its occupants or contents during an impact. Considering a car accident, the less the injury suffered by the occupants, and the less the damage to the total vehicle, the better is the crashworthiness of the vehicle. From the viewpoint of crashworthiness, the topic seeks to improve the collision resistance of structural systems by sacrificing the structure to absorb collision energy and, thereby, to protect the passengers or cargo. In the case of passengers, it is required that a survivable volume remains and the decelerations and forces are limited so that they remain at survivable levels for specified accident scenario [1].

Generally speaking, the essence of crashworthiness is energy management and the control of crash deceleration pulses. In order to better understand this essence, let’s review how to assess crashworthiness in practice. To assess crashworthiness, several criteria are used, which include the human tolerance to deceleration experienced by the vehicle during an impact, the deformation patterns of the vehicle structure, and the probability of injury predicted by human body models.

Human injury tolerance to impact varies with size, age, sex, and specific conditions. The main injuries include head injuries, spinal injuries and chest injuries. Among them,
head injuries are responsible for a significant loss of life and serious injuries in transportation accidents.

According to European New Car Assessment Programme (Euro NCAP) [106], the tolerance level of head injuries is the limitation of the acceleration, with a value of 80g (standard gravity acceleration) over a time period of more than 3 ms that should not be exceeded. In order to explain the relation between the acceleration and the injury level, several criteria have been developed. Gadd Severity Index (GSI) was defined as [52]:

$$SI = \int_{0}^{T} A_{v}^{2.5} dt \quad (1.1)$$

where $A_{v}$ is the dimensionless acceleration $a_{v}/g$, $a_{v}$ is the average head acceleration which may vary throughout the loading pulse.

Another criterion HIC (Head Injury Criteria) is also used on a worldwide basis to assess safety of vehicles in crash test.

$$HIC = \left( T_{2} - T_{1} \right) \left\{ \int_{T_{1}}^{T_{2}} A_{v} dt/(T_{2} - T_{1}) \right\}^{2.5} \quad (1.2)$$

where any time interval $T_{2} - T_{1}$ is selected to maximise the right-hand side of equation 1.2. This means that the HIC includes the effects of head acceleration and the duration of the acceleration. HIC can be recast the form as:

$$HIC = V_{2} a^{1.5} / g^{2.5} \quad (1.3)$$

which shows the relation between the Head Injury criterion and the impact velocity and acceleration. HIC $\geq 1000$ is considered as life threatening.

The influence of pulse length and the shape of deceleration curve also play an important role in the severity of injury. Agaram [107] investigated the phenomenon and pointed out that for different shapes of deceleration curves with the same level deceleration, the
relevant HIC calculated can have different values and lead to different occupant responses. The severity of damage is higher if the larger the value of the deceleration and the longer the deceleration lasts.

Over the past decades, many research efforts have been made to understand the collapse behaviour of thin-walled metal tubes under static or dynamic axial loading. The early researches mainly focus on the theoretical and experimental studies. The first study on this topic began fifties years ago by Alexander [2]. He proposed a simple, kinematically admissible collapse mechanism, which formed the basis of the crush calculations. Since then, many improvements and extensions have been made to his approach. More realistic folding mechanisms were formulated by researchers. Through these theoretical studies, a good understanding of the collapse mechanism was achieved. Even today, with the availability of highly sophisticated numerical methods and advanced experimental techniques, simplified models still provide valuable assistance with the interpretation of experimental results and for preliminary design purposes. Experimental studies were also carried out by some researchers over the years. These tests examined various designs of thin-walled structures with different materials and loading conditions and help to give a sound understanding of impact phenomena in relation to different applications.

Furthermore, in recent years, with the development of finite element method and also due to the availability of increased computing power, numerical simulations were played a more important role in impact analysis by many researchers.

Although a lot of studies have been conducted by various researchers, the problem remains challenging. The deformation history is complex; it involves large plastic deformations including load reversals, sharp radii of curvature, contact and other challenges. There seems to be a consensus that much more can be done to lessen the
potential danger of impact accidents and improve the energy absorption capability of the structures.

1.2 Objectives of the thesis

The main purpose of this thesis is to grasp the complex crushing behavior of thin-walled ductile metal tubes under axial loading by experimental study and simulation. Based on the understanding of the collapse behavior, the new structures and trigger systems can be developed and make it possible to better predict and control the collapse process, improve the stabilization and achieve higher energy absorbing capability.

In this thesis, a series of experiments were firstly carried out, in which all the relevant variables are examined. In the experiments the tube diameter, thickness were varied to establish trends of the effect of these variables on the results. In addition, several models are evaluated by comparing their predictions to the experimental results.

Through this study, a clear picture of the collapse behaviour of these structures should be achieved, which is vital for the design of energy absorbers. The main factors which affect the energy absorption characteristics will be decided and systematically analyzed for all these structures in a wide data range.

In this thesis, based on the analyses, ideal energy absorbers are expected to be designed. Firstly, an ideal energy absorber should provide high mean crushing force, which means higher energy absorbing capability. Secondly, to minimize the injury and damage, the initial peak force should be eliminated and the crushing force should be kept as stable and smooth as possible, thereby, the deceleration during the crash can be below some critical values. For example, in the case of a car accident, the soft crush of sheet metal produces low deceleration pulses in the vehicle. The structural member of the frame creates more deceleration. The series of deceleration pulses first affect the vehicle, the
interior components, and then the occupants. The more severe and abrupt the
deceleration, the more severe the possible injuries. In a collision, slowing down the
deceleration by even a few tenths of a second can create a drastic reduction in the force
involved.

To achieve these goals, a new design of energy absorbers is proposed and analyzed.
This new energy absorber eliminates the drawbacks of the existing energy absorbers and
provides some valuable instructions to engineering design. Also, several methods of
reducing the initial peak forces have been examined.

1.3 Contributions to knowledge

There are several new contributions in the work reported in this thesis, which improve
the understanding of the crushing behaviour of thin-walled tube and give useful
instructions for engineering designs. A brief summary of these contributions is given as
follow.

(1) A very careful experimental study has been carried out to examine how the localized
plastic folds form and develop during the crushing process. The relationship
between the localized plastic deformation and the corresponding crushing force has
been checked in detail. The knowledge obtained can be very useful for the research
in collapse mechanism of thin-walled structures.

(2) In previous investigations [3,4,5], the important geometrical factor, namely the ratio
of diameter/thickness (D/t), generally ranges between 4 and 150. In this study, the
range of this dimensionless parameter is extended beyond those reported and
reaches to 500. Therefore, a more rigorous assessment for the dynamic progressive
axial buckling of circular thin-walled tubes has been achieved, and it gives useful
instructions for the further experimental studies.

(3) A thoroughly investigation of multi-cell tubes is presented in chapter 6. Several new cross-section profiles of multi-cell tubes are proposed and analyzed. An important conclusion has been drawn from the analyses, namely: the high energy absorption capability of multi-cell tubes is mainly determined by the ratio of the longer side width of each cell to the tube wall thickness, while the number of the cells in the multi-cell tube has only minor effect on specific energy absorption. The main contribution is to improve the stability of the structure. This is different from the conclusions presented in the literature that the impact energy absorption increases with the number of cells.

(4) Several new methods are proposed to reduce the initial peak force, which include drilling some holes on the tube wall, the circumferential dented trigger on the tube wall and a taper on the end of the tube. These trigger designs are proved to be an effective way to reduce the initial peak force and can seriously affect the crushing behaviour.

(5) A new design of impact energy absorber, which is based on the investigation in this thesis, has been proposed. The design is a multi-cell circular tube which is composed of different number of circular coaxial tubes and supporting ribs. The height of each tube is different and there are circumferential dented triggers on each tube. Comparing with the conventional thin-walled tube, this new design can provide very stable reaction force and high energy absorption capability. Furthermore, the initial peak force completely disappears in the reaction force.

1.4 Outline of the thesis

This thesis consists of seven chapters. A brief description of its outline is given as
Chapter 1 provides some background about structural crashworthiness and identifies the contributions and objectives of this research before describing the structure of the thesis.

Chapter 2 briefly reviews the state-of-the-art of knowledge on structural crashworthiness and summaries the available literature on the plastic collapse behaviour of thin-walled tubes subject to axial loadings.

Chapter 3 investigates the crushing behavior of thin-walled circular tubes experimentally. The influences of geometrical properties on the collapse modes are examined. The plastic deformation in the local folding area is carefully investigated. The static and dynamic tests are compared to reveal the effect of different loading conditions.

Chapter 4 numerically investigated the main factors involved in energy absorption of the single thin-walled metal tube when subjected to axial impact loads and demonstrates how the different geometrical parameters, material properties and loading conditions can affect the impact process and energy absorption capability.

Chapter 5 presents an experimental study to examine the development of the first peak force and introduces different trigger systems to reduce the first peak force and check how the triggers can affect the energy absorption characteristic of the thin-walled tube.

Chapter 6 systematically analyses the crash behaviour of multi-cell thin-walled metal tubes under axial impact. The research demonstrates why higher energy absorption capability of the multi-cell thin-walled tubes can be achieved. In the last part, a new
multi-cell structural design has been proposed and examined.

Chapter 7 summarizes the work presented in the thesis and then presents the conclusions which have been drawn from the study. Finally, some topics for future research are suggested.
CHAPTER 2

Literature review

As an efficient energy absorbing structure, thin-walled metal tubes are widely studied by researchers over the years. This chapter is intended to provide a comprehensive review of the investigations to date on the collapse mechanism of thin-walled tubes under axial loading.

This chapter begins with a brief introduction to thin-walled energy absorbing structures in Section 2.1, which includes various types of energy absorption structures and the related research work. The definitions of terms and variables used later in the thesis are also introduced. Section 2.2 reviews the main research on the crushing behavior of thin-walled tubes under axial loads and Section 2.3 explains all the main theoretical models on the circular thin-walled tube and the square tube. Section 2.4 summarizes the key points in this chapter.

2.1 Introduction to thin-walled energy absorbing structures

In practice, there are different kinds of energy absorbers which can apply in various situations based on their applications, such as in nuclear engineering [6], vehicle design [7,8], collision protection for highway safety [9,10]. Basically, an energy absorber is a device that converts kinetic energy into some other form of energy. The requirements of energy absorbers are usually to achieve high energy absorption with less mass, while keeping the reaction force low enough to minimize the injury and damage to people and cargo. As a good energy absorber, it is also expected to provide a relatively long stroke and a fairly stable reaction force throughout the entire crushing process.
According to their energy absorbing characteristics, energy absorbers can be divided into two types. The first type absorbs kinetic energy in a reversible way, like spring-based vehicle suspensions, which can be used repeatedly for a long term [11,12,13]. The second type is aimed at absorbing the majority of the kinetic energy of impact within the device itself in an irreversible manner. This type of energy absorbers are normally one-shot items, i.e., once crushed, they are discarded and replaced. By sacrificing itself, this kind of energy absorber ensures that human injuries and equipment damage are minimized and are widely applied in aerospace and automotive engineering. A typical example of this kind of energy absorber is the front bumper using in the car.

Thin-walled metal tubes with various cross section profiles belong to the second type of energy absorbers, which are extensively employed as energy absorbing structures in industry due to their high strength-to-weight ratio, low cost, and excellent energy absorption capability. For example, they are used in automobile bodies, and as aircraft fuselages and ship hulls. Over the past three decades, enormous efforts have been made by industry and university researchers to understand the mechanisms of structural collapse in axial crushing of thin-walled metal tubes. In the work reported in this thesis, circular thin-walled metal tubes have been chosen as the basic structure to study their collapse mechanism and energy absorption characteristics of the folding process.

Thin-walled metal tubes can have different geometry profiles, such as circular and square thin-walled tubes, hat-type cross-section tubes, corrugated tubes, frusta and tapered tubes. Many researchers have made great efforts to examine all these energy absorbing structures.

The circular and squared thin-walled tubes are most popular studied by researchers. Johnson et al. [14,15] reviewed and identified the dominant collapse modes of simple
structural elements in their review papers. Andrews et al. [3] pointed out that the collapse mode is governed by the geometrical properties and investigated the effect of the tube length to wall thickness (L/t) and the D/t ratios on the mode of collapse. The experimental observations show that the thicker tube normally collapses in a concertina mode, whereas the relatively thinner tube tends to buckle in a diamond mode. Abramowicz et al. [16,17,18] conducted a series of detailed experimental study to characterize the various collapse modes of circular and square thin-walled steel tube. They observed that reasonable agreement existed between the experimental results and the predictions of average crush loads based on their theoretical model.

Gupta et al. [19] performed an axial compression test on thin-walled aluminum and mild steel circular tubes in both annealed and as received conditions and combined all their results and developed an empirical model for determining the load history. Experimental studies have also been carried out by Galib [20] and Guillow [21] at different test conditions.

The effect of heat treatment on the mode of collapse of axially crushed tubes has also been studied. Such a phenomenon has been experimentally studied by Reddy and Zhang [22] who concluded that the removal of strain hardening not only changes the mode of deformation but, in some cases, causes an overall loss of stability of the crushed tubes.

The other geometry profiles were also studied by different researchers. Kim et al. [23] investigated various ways of reinforcing the hat-type cross-section to address the design aspects of a front side rail structure of an automobile body. White [24,25] also investigated the crushing behaviour of hat-type thin-walled section theoretically and experimentally. Mamalis et al. [26,27] carried out experimental studies on the collapse behaviour of thin-walled circular cylinders and frusta under axial compression and give a basis of comparison between the cylinder and the frusta. EI-Sobky [28] also
conducted similar experimental study. The crushing behavior of corrugated tubes was examined by Singace et al. [29]. The studies concluded that corrugated tubes would be a favorable choice, if an energy absorption device is expected to achieve a controlled behavior. Tapered thin-walled metal inverbuckle energy absorbing tubes were investigated by Chirwa [30]. In his study, the tapered tube with various external loadings and boundary conditions was studied. In particular, the influence of tapered dimension was explored. An approximate theoretical prediction is compared with the experimental results and shows good agreement.

In order to improve the energy absorbing capability of the thin-walled metal tube, some other materials can be filled in the structures, such as polyurethane foam, metal foam, concrete and wood.

The earlier work on foam-filled tubes is focused on steel tubes filled with polyurethane foam fillers. The first attempt to understand the force-deformation response of foams goes back to work done by Shaw and Sata [31]. They identified the well-known three-region stress–strain curve characteristics of foams: a linear elasticity at small strains, a long distinct plateau region with almost constant stress, and a final densification region at very large strains (60-90% nominal strain). The long plateau region makes the aluminum foam an ideal material for energy absorption. Reid [32] examined the tubes both empty and filled with polyurethane foam of various densities and point out that filling the tubes with foam is found to improve their crush strength not only because of its own strength but also through an interaction with the enveloping tube sheet which transforms the deformation mode from a non-compact to a compact form. Some theoretical models are proposed to explain and quantify the interaction between the foam and the sheet metal tubes. Abramowicz et al. [33] extended the analytical basic folding mechanism approach to account for the polymeric foams. However, Thornton et al. [34] summarized the effect of polyurethane foam by
concluding that even though a considerable increase of collapse load was achieved, thickening of the tube wall was still more weight efficient than polyurethane foam filling.

Metal foam filler is a new and promising cellular material developed in recent years. Gibson and Ashby [35,36] give a comprehensive review of various properties of metal foam. The main contribution to the available well-organized database of experimental results for statically and dynamically axially loaded foam-filled tubes can be attributed to the comprehensive work done by Hanssen et al. [37,38]. In addition, some useful design formulas have been introduced for practical application based on the experimental results. The axial crushing of aluminium foam-filled tubes has been studied by Seitzberger et al. [39,40] with emphasis on the experiment. Various cross-sections were considered in their studies, including square, hexagonal, octagonal and bitubal arrangements of these cross-sections. They reported that considerable mass efficiency improvements with respect to energy absorption were obtained by the foam filling, particularly the bitubal arrangements.

Tubes filled with concrete were examined by Tao [41] and Chen [42] and prove that the concrete filler can greatly enhance the buckling loads and energy absorption capability. The collapse behavior of tubes filled with wood was investigated by Reddy [43] experimentally and theoretically. Experiments show that the mode of elastic buckling is changed by the presence of the wood filler. And their energy absorbing capacity shows a considerable enhancement. The theoretical model proposed also shows good agreement with the experimental results in mean loads and the fold length. Sandwich plate is another widely used structure as energy absorbers. Many research work have been conducted by various researchers [44,45].

Some research works on tubes under axial crushing are related to various loading
conditions. Some of the most well-known arrangements involve, axial crushing of tubes, oblique loading of tubes[46], transverse loading [47], lateral crushing of tubes [48] and tube inversion[49]. Each energy absorber system has its own characteristics. The kinetic energy can be absorbed by plastic bending, stretching and tearing [50,51]. All of these contributed towards a better understanding of the modes of failure and the energy dissipation patterns during impact in such structures.

In this thesis, thin-walled metal tubes are the main structures to be reviewed and studied as widely used energy absorbers. In the following sections, the available information in the literature on the plastic collapse behaviour of thin-walled tubes subject to axial loading is summarized. It is arranged broadly in chronological order.

### 2.2 Introduction to thin-walled metal tubes under axial loads

#### 2.2.1 Axial collapse modes

In this thesis, the study mainly focuses on the crushing behaviour of the thin-walled metal tubes under axial loads. A thin-walled metal tube, when subjected to a large axial impact load, may undergo different deformation modes, i.e. symmetrical mode, asymmetrical mode, mixed mode and global buckling [3], as shown in Fig 2.1.

- **The symmetrical mode** is often known as the concertina (ring) mode.
- **The asymmetrical mode** is referred to as diamond mode.
- **The mixed mode**: in this mode, the tube starts to deform in the concertina mode but then revert to the diamond mode as collapse progresses.
- **Euler (global) buckling mode** is an inefficient mode of energy absorption with potentially catastrophic consequences and needs to be avoided in crashworthiness applications.
The axial crushing force related to the concertina and diamond collapse modes exhibits similar characteristics as that reported by Johnson and Reid [14]. Andrews [3] pointed out that the concertina mode shows slightly higher specific energy absorption than the diamond mode. This is most probably due to the greater degree of plastic deformation in the concertina mode. The diamond mode is characterized by a number of lobes, which can vary from three to nine for most practical tubes. A typical load-displacement curve of thin-walled tube axial collapse is shown in Fig 2.2. When a thin-walled circular tube under axial loads is fully crushed, its crushing process usually consists of three stages. First, the axial crushing force reaches the initial peak to overcome the initial resistance of the tube. Second, the force is followed by a sharp drop and then fluctuations about a mean load during the crushing progresses. These fluctuations are a result of formation
of the successive plastic folding, the amplitude of fluctuations being sometimes as high as 50% of the mean load. A stroke of about 70–75% can be obtained at a steady mean load. Third, the force increases rapidly with relatively small increase of crushing.

![Graph](image)

**Fig 2.2 A typical load-displacement curve of thin-walled tube under axial loads**

The matter of which modes dominate the collapse process depends primarily on the section geometry i.e. ratios of the dimensions of the tube, namely length, diameter and thickness. The collapse modes can affect the energy absorption of the structures. The following section will introduce the closed-form solutions for the concertina mode and the diamond mode separately.

### 2.2.2 Main variables in impact analysis

Many parameters have been introduced in impact analysis to evaluate the energy-absorbing characteristics and efficiencies of the structures. These parameters express the characteristics of energy absorbers in terms of their mass, volume, and other important properties. The main parameters included in this study are listed below:

- average crush force
- initial peak force
Average crush force, $P_m$ is one of the most significant parameters for quantifying the behaviour of axially compressed tubes. It is defined as the initial kinetic energy divided by final reduction in axial length. $P_m$ is usually divided by the plastic moment $M$ to produce a ratio $P_m/M$. When calculating the plastic moment, $M$, researchers have used various measures for the flow stress.

Initial peak force is an important parameter in the design of an energy absorber, since the maximum force level is associated with safety. The initial peak force should be kept below a certain level when designing an energy absorber even if the absorber can absorb the required energy.

There are two dimensionless parameters included in this study. The first one is structural effectiveness, which is defined as $\mu = P_m/A\sigma_u$ where $P_m$ is the average crush force and $\sigma_u$ is the ultimate tensile stress. Structural effectiveness is a ratio between the average crushing force and the force required to sustain plastic deformation at a particular strain, which is used to describe how much the structure can contribute the energy absorption [52]. Many researchers assess the crushing force by using the normalized force $P_m/M$. Comparing these two parameters, structural effectiveness $\mu$ not only include the ratio between the crushing force and the material strength, but also include the cross-section area of the structure, which reflects the strength of the structure. Therefore, structural effectiveness $\mu$ is an important variable to represent the energy absorbing capability of the structure.

The second dimensionless parameter is called the relative density, or solidity ratio,
which is defined as $\delta = A/A_1$, where $A = 2\pi Rh$ is the cross sectional area of the thin-walled tube and $A_1 = \pi R^2$ is the cross sectional area which is enclosed by the cross section.

Specific energy absorption, SEA (defined as Energy absorbed divided by the weight of the structure) is a parameter used in connection with energy absorption capability. The area under the load-deflection curve for a structure is a measure of the energy absorbed by a structure for a particular direction of application of the force [53].

$$SEA = \frac{\text{Energy Absorbed}}{\text{Weight of Structure}}$$

### 2.3 Theoretical models

In all these thin-walled energy absorbing structures, circular and square thin-walled metal tubes are most widely used geometry profile. The theoretical analysis on this kind of tube has been carried out by researchers for over five decades. Several important theoretical models are proposed by researchers. Alexander (1960) [2] and Pugsley [54,55] were the first investigators to analyze the collapse behaviour of the thin-walled tube. Alexander presents a theoretical model of the crushing process of thin-walled metal tubes deforming in a concertina mode, while Pugsley develops a theoretical model for the tubes with the diamond mode. Their theoretical work was based on the final deformed shape of the crushed tubes without considering the effect of the loading paths. Since then, a number of authors have contributed to the problem of thin-walled structures and many improvements and extensions have been made to their models.

Andronicou et al. [56] modified the model by Alexander and considered the interaction between axial bending and circumferential stress resultants for the formation of the first hinge. In his study, the plastic bending moment and the distance between plastic hinges
become dependent on the axial force.

A more realistic radially outward folding mechanism was formulated by Abramowicz et al. [16,17]. They proposed an improved model by introducing curvature in the deforming fold length. They also introduced the important concept of effective crushing distance and included the effective crushing distance and material strain rate effects into the crushing model.

Grzebieta [57] proposed a collapse mechanism model for axi-symmetric mode, which was a modification of Alexander’s. In his model, a fold consisted of three equal length, the two curved regions are separated by a straight region where each region is one-third of the fold leg length.

Wierzbicki et al. [58] modified Alexander’s solution and replaced stationary plastic hinges with moving hinges. This led to a realistic deformed shape and improved prediction of the mean crushing force. They pointed out that the crushing process features a stiffening phase of the tube resistance which follows the softening phase during the formation of each buckle. In their later work [59], they pointed out some inaccurate assumptions made by Alexander and introduced some new concepts related to the crushing process, which are the super folding element, the transition zone model and the eccentricity factor (m). Using these concepts, the model captures several important features in the crushing process.

In the analysis produced by Wierzbicki, the eccentricity factor m relating the inward and outward parts of the folds was an arbitrary value. Singace et al. [60] re-examined the problem and verified the value of the eccentricity factor m. In another paper [61], they investigate the effect of the tube material and its D/t ratio on the value of m and showed that m is slightly sensitive to the strain hardening characteristics of the compressed tube.
material. Singace [62] also extended his research on the eccentricity factor $m$ to the diamond mode and concluded that the value of $m$ is independent of the tube geometric ratios and material properties.

Furthermore, some excellent reviews of the research work on the collapse behaviour of the thin-walled tube have been given by Jones [63], Johnson [14], Alghamdi[64], Olabi [65] and Reid [66]. In addition, several important books related to structural impact are published, such as Crashworthiness of Vehicles by Johnson et al. [67], Structural Crashworthiness by Jones et al. [68], Structural Impact and Crashworthiness edited by Davies et al. [69], Structural Failure by Wierzbicki et al. [70], Structural Crashworthiness and Failure by Jones et al. [71], Structural Impact by Jones [52], Energy absorption of structures and materials by Lu et al. [53], Introduction to Impact Engineering by Macaulay [72], Engineering Plasticity by Johnson [73], Metal Forming and Impact Mechanics edited by Reid [74], Impact on Composite Structures by Abrate [75], Impact Strength of Materials by Johnson [76]. These papers and books help to bring together the understanding of the collapse mechanism of the energy absorbing structures.

2.3.1 Theoretical prediction for the single circular tube

In all these theoretical models, four models proposed by Alexander, Abramowicz, Wierzbicki and Singace will be discussed below in detail, which can help to understand the main works done by the researchers. These four models represent the main development of the model of the thin-walled tube. In this process, several new concepts are presented. More details of these models are explained below.
2.3.1.1 Alexander model

It was Alexander [2] in 1960 who firstly developed a theoretical analysis for the axial crushing of a thin-walled cylindrical tube subjected to a static axial load. In his model, Alexander assumed the tube collapses in the form of a concertina mode with straight-side convolutions, which are facilitated by a kinematic mechanism with three circumferential plastic hinges, as illustrated in Fig 2.3.

The external work done to achieve this concertina mode is determined, which can be divided into two parts, namely the bending at three stationary plastic hinges (extensional deformation) and the circumferential stretching of the metal between the hinges (inextensional deformation). It transpires that the extensional deformations, though highly localized, dissipate at least one-third of the total energy.

![Fig. 2.3 Theoretical model for symmetric crushing mode by Alexander [2]](image)

As shown in Fig 2.3, during an increment \(d\theta\) of the angle \(\theta\), the increment of work done for bending at the three joints is
\[
\begin{align*}
\text{d}W_b &= 2M \cdot \pi 2R \cdot \text{d}\theta + 2M \cdot \pi (2R + 2H \sin \theta) \cdot \text{d}\theta \\
\end{align*}
\]

(2.1)

The von Mises yield condition is used to generate the fully plastic bending moment. The collapse moment M will be \(2/\sqrt{3} (\sigma_0 t^2 / 4)\), where \(\sigma_0\) is the flow stress. Then, \(\theta\) increasing from 0 to \(\pi/2\), the total work done is

\[
\begin{align*}
W_b &= 2M \cdot \pi 2R \cdot \frac{\pi}{2} + 2M \cdot \int_0^{\pi/2} \pi (2R + 2H \sin \theta) \cdot \text{d}\theta \\
&= 2\sqrt{3} \sigma_0 t^2 \cdot \pi (\pi R + H) \\
\end{align*}
\]

(2.2)

The increment of work done in stretching the metal between the hinges will be

\[
\begin{align*}
\text{d}W_s &= 2\pi \sigma_0 H^2 t \text{d}\theta \cos \theta \\
\end{align*}
\]

The integrated stretching work done is

\[
W_s = 2\pi \sigma_0 H^2 t \\
\]  

(2.3)

From the energy balance view, the external work has to be dissipated by the bending work and the stretching work. The corresponding mean crushing forces are determined by equating the external work to the mean collapse force multiplied by the distance through which one complete folding form. Consequently

\[
\begin{align*}
P_m \cdot 2H &= W_b + W_s \\
\end{align*}
\]

(2.4)

where \(P_m\) is the mean crushing force over a complete collapse of the fold. Thus the following theoretical equation is obtained for normalized average crush force:

\[
\frac{P_m}{M} = 20.73 \sqrt{\frac{2R}{t}} + 6.283 \\
\]

(2.5)
The plastic half-wavelength, $H$ is determined by minimizing this expression as follows:

$$\frac{H}{R} = 1.905 \sqrt{\frac{t}{2R}}$$  \hspace{1cm} (2.6)

Although simple, this model seems to include the underlying physical processes involved. It provided the basis for many further developments.

### 2.3.1.2 The Abramowicz model

In Alexander’s model, some simplifications are introduced, for example, the deforming tube wall between the plastic hinges is assumed to be the straight line. But this assumption does not agree with the experimental observations. Experiments show that the final shape of the fold is the curved surface. Alexander’s model also assumes that the material is rigid, perfectly plastic. These assumptions simplify the theoretical analysis, but result in some inaccuracy. In an attempt to produce a more realistic fold, an improved model was presented by Abramowicz and Jones [16,17]. They conducted axial compression tests on a range of thin-walled circular and square steel tubes and introduced two new concepts into the crushing model, which are the effective crushing distance and material strain rate effects.

The effective crushing distance is the first important concept introduced. Abramowicz examined the effective crushing distance of crushing process, using two arcs joined together to represent the deformed tube wall, as shown in Fig 2.4, where a fold consisted of two equal segments of length $H$, curved in opposite directions and the material had finite thickness. It was recognized that the deforming tube wall bends in the meridian direction instead of the straight line between the plastic hinges. This leads to an effective crush length which is smaller than $2H$. Consequently, a slightly higher
average force is obtained. As seen in Fig 2.4, an effective crushing distance is

\[ \delta_e = 2H - 2X_m - t \]  \hspace{1cm} (2.7)

where from Abramovicz [77] \( X_m = 0.28(H/2) \). Thus,

\[ \delta_e = 1.72H - t \]  \hspace{1cm} (2.8)

![Figure 2.4 Effective crushing distance [77]](image)

An alternative estimate of the mean crushing load \( P_m \), which takes more realistic account of the mode of deformation, is given as:

\[ \frac{P_m}{M_o} = \frac{20.97(\frac{2R}{h})^{0.5} + 11.90}{0.86 - 0.568(h/2R)^{0.5}} \]  \hspace{1cm} (2.9)

The second concept introduced to the model is material strain rate effects. Under dynamic loading conditions, there are two factors which are not included in the static collapse. They are the inertia effect and the strain rate effect. Jones [52] points out that the inertia forces in the wall of a tube can affect the mode of deformation under rapid
acceleration, but it doesn’t play an important role during the dynamic progressive buckling of tubes struck by masses having $M\gg m$ and with low velocities. When the influence of inertia effects of the tube becomes important in a practical problem, with larger axial impact velocities, then the phenomenon is known as dynamic plastic buckling, which is related to more complex mechanism. Therefore, in the study, if the velocity is limited to 10 m/s, the influence of inertia forces can be ignored.

However, in the dynamic progressive buckling analysis, although inertia effects may be ignored, the influence of material strain rate sensitivity must be retained for many materials, especially for the strain-rate dependent materials. In mild steel, for example, its value may be larger than the quasi-static value by a factor of about 2 at strain-rates which commonly occur in practice in the collision of vehicles. Hence, Abramovicz modifies the plastic flow stress in his model, in order to cater for the enhancement of the flow stress with strain rate, which is given as:

$$\frac{P_m}{M_0} = \left[ 20.97 \left( \frac{2R}{h} \right)^{0.5} + 11.90 \right] \left[ 1 + \left( \frac{0.25V}{684R\left(0.86-0.568h/2R^{0.5}\right)} \right)^{1/3.91} \right]$$  \hspace{1cm} (2.10)

In the model, the first section includes the effective crushing distance and the second section represents the material strain-rate effect by using Cowper-Symonds constitutive equation.

2.3.1.3 Wierzbicki model

In Alexander and Abramowicz’s models, the region between the plastic hinges was assumed to move either completely outwards or completely inwards. But the experiments have shown that the wall of an axisymmetric deformed tube will be laid down partly to the inside and partly to the outside of the tube. The concept of the ratio of the outward part of the folding length to the total folding length was introduced by
Wierzbicki et al. [59]. It is known as the eccentricity factor and denoted by m.

Wierzbicki et al. introduced the new model for the symmetric collapse mechanism which allows for both inwards and outwards radial displacement. In addition, by considering energy rate equations, Wierzbicki et al. develop equations for not only determining average crush load but also a representative load–deflection history.

Wierzbicki also pointed out that Alexander’s model was based on some assumption that one fold goes through the entire crushing process before the next one begins to deform. But he reckoned that these assumptions were inaccurate based on actual observations. In order to better explain the crushing process, Wierzbicki introduced three new concepts:

1. The super folding element (SE)
2. The transition zone model
3. The eccentricity factor (m)

Using these concepts, the model captures with great realism several effects. These include description of a softening, followed by a stiffening phase, alternating lower and higher peaks and a reduced crush distance.

**Super Folding Element**

The Super Folding Element is a representative element of the crushed zone from which the entire deformed tube can be assembled by translation, rotation and mirror reflection. Fig 2.5 shows a basic super folding element, which is isolated from the tube by two horizontal cuts and developed as a basic folding wavelength. Then the active zone of plastic deformation can be represented by two super folding elements.
Fig 2.5 Undeformed, partially deformed and fully crushed individual superfolding element [59]

Transition zone model

Transition zone is a region between the crushed part of the tube and the undeformed part, which is composed of two super folding elements. Wierzbicki use this zone to precisely represent the crushing process.

Fig 2.6 illustrates the basic requirements of the transition zone of plastic deformation. It is assumed that the crushed part of the tube is composed of alternating, densely packed circles. The undeformed part of the tube is straight. The point A of the already crushed tube must be connected with point B of the undeformed zone. Point A is assumed to move vertically with a constant velocity to simulate the crushing process. Point B is stationary until a new contact between the lobes occurs.

Fig 2.7 demonstrates the crushing process of the two-element model. Note that after element 1 is completely crushed so that touching occurs, the transition zone shifts downward by one SE and the process is repeated with participating elements 2 and 3. The two element transition zone is relatively simple and represents the geometry of actual columns quite closely.
The concept eccentricity factor is firstly introduced by Wierzbicki based on the experimental observations and theoretical analysis. He pointed out that an axisymmetrically deformed tube will fold partly to the inside and partly to the outside of the original radius of the tube. This geometric eccentricity feature can be described by a factor, $m$, which is defined as the outward portion over the whole folding length. The introduction of the eccentricity factor, $m$, in the analysis leads to the successful
qualitative reproduction of many of the features characterizing the physical behaviour of tubes folding in a concertina mode.

Wierzbicki’s model captures several features of the crushing processing which were unaccounted for in all previous computational models of progressive folding. Then the formula for the normalized mean crushing force can be recast into the form:

$$\frac{P_m}{M} = \frac{8\pi}{t} H + \frac{\pi^2 R}{H}$$  \hspace{1cm} (2.11)

The formula for the wavelength is

$$\frac{H}{R} = \sqrt[4]{\frac{\pi}{2}} \frac{t}{\sqrt{2R}} = 0.886 \frac{t}{\sqrt{2R}}$$  \hspace{1cm} (2.12)

Wierzbicki’s new model captures several important factors which show very good agreement with experimental results.

2.3.1.4 Singace model

In Wierzbicki’s model, the value of the eccentricity factor $m$ is arbitrary and indeterminate, which withheld further clarification of the critical inclination angles of the folding elements corresponding to inward and outward folds. Singace [60] take a further step, based on Wierzbicki’s model, to verify that the value of the eccentricity factor $m$. The analysis also introduce the critical angles $\alpha_0$, and $\beta_0$, which characterizes the critical position for the next fold and corresponds to the formation of the outward and the inward parts of the folds, as shown in Fig 2.8.
In the first phase of collapse, the eccentricity factor, $m$, is related to the critical angle $\alpha_0$, as follows:

$$\cos \alpha_0 = m$$

During the second phase of folding the eccentricity is defined as $n = 1 - m$ and the radii of the plastic hinges are not equal to those in the first phase. Similarly to the first phase, the eccentricity factor, $n$, is related to the critical angle $\beta_0$ as follows:

$$\cos \beta_0 = 1 - m = n$$

Singace et al. [61] attempted to verify the theoretical values of $m$ and the relevant values of the critical angles $\alpha_0$ and $\beta_0$ by experiments. The theoretical value of $m$ is equal to 0.65.

### 2.3.1.5 Theoretical model for the diamond mode

Theoretical models for the diamond mode are more complex and less successful than those for the ring (concertina) mode. Pugsley and Macaulay [54,55] were among the first researchers to consider the asymmetric folding mode, their study being largely
empirical. They proposed the normalized mean crushing force:

\[
\frac{P_m}{M} = 0.326 \left(\frac{2R}{t}\right) + 217.7
\]  

(2.13)

where the coefficients are selected to agree with some experimental test results on stainless steel and soft aluminum cylindrical shells.

Johnson et al. [78] attempted to develop a theory for the asymmetric mode based on the actual geometry of folding, with the tube material at the mid-surface being considered inextensional. Wierzbicki [79] further modified the model and gave the approximate expression

\[
\frac{P_m}{M} = 62.88 \left(\frac{2R}{t}\right)^{1/3}
\]  

(2.14)

and obtained good agreement with experimental results.

Abramowicz et al. [17] pointed out that the diamond mode has different multiple lobes for different D/t ratio and they give the equation as

\[
\frac{P_m}{M} = A_1 \left(\frac{2R}{t}\right)^{1/2} + A_2
\]  

(2.15)

Coefficients \(A_1\) and constants \(A_2\) can are associated with the number of lobes \(N\) respectively. For example, it may be shown that the coefficients \((A_1)\) in Equation (19) are 21.07, 20.61, 20.40 and 20.30 with associated constants \((A_2)\) 32.66, 60.70, 96.72 and 140.74 when \(N=3, 4, 5\) and 6, respectively. The difficulty with theoretical equations developed for asymmetric mode collapse is they require a knowledge of the number of lobes, \(N\), for a given D/t ratio. It must also be noted, however, that a successful model needs to take into consideration the large deflection theory and the strain hardening effect.
2.3.2 Theoretical prediction for the single square tube

Compared with the circular tubes, the crushing behavior of square tubes is more complex. For a square tube, there are five different collapse modes which might occur during axial compression. They are the symmetric mode, the asymmetric mixed collapse mode A, the asymmetric mixed collapse mode B, the extensional collapse mode and the global buckling mode. Abramowicz et al. [17] have identified two basic collapse elements (Type I and Type II) which have been used to study the dynamic progressive buckling of square tubes, as shown in Fig. 2.9.

![Fig. 2.9 Basic collapse elements of collapse square tube [80]](a) Type I (b) Type II

The symmetric collapse mode consists of four type I elements. The asymmetric mixed deformation mode A consists of two layers with a total initial height 4H and six type I and two type II basic folding elements. The asymmetric mixed deformation mode B consists of two layers with a total initial height 4H and seven type I and one type II basic folding elements. The extensional collapse mode consists of one layer with four type II elements. The extensional collapse mode governs the progressive behavior of thick square columns. The mean collapse force for extensional behaviour is considerably larger than that associated with the other three collapse modes.
For a square column with the symmetric mode of deformation, the normalised mean crushing load is

\[ \frac{P_m}{M} = 52.22 \left( \frac{c}{h} \right)^{1/3} \]  
(2.16)

for asymmetric mixed collapse modes A and B, respectively, the normalised mean crushing load are

\[ \frac{P_m}{M} = 43.61 \left( \frac{c}{h} \right)^{1/3} + 3.79 \left( \frac{c}{h} \right)^{2/3} + 2.6 \quad \text{(mode A)} \]  
(2.17)

\[ \frac{P_m}{M} = 46.16 \left( \frac{c}{h} \right)^{1/3} + 2.14 \left( \frac{c}{h} \right)^{2/3} + 1.3 \quad \text{(mode B)} \]  
(2.18)

For a square column with an extensional collapse mode, the normalised mean crushing load is

\[ \frac{P_m}{M} = 36.83 \left( \frac{c}{h} \right)^{1/2} + 10.39 \]  
(2.19)

### 2.3.3 Generalized folding mechanisms

A basic folding mechanism was developed by Wierzbicki and Abramowicz [79,80,81] for predicting crushing behavior of thin-walled tubes with an arbitrary central angle, which combines the features of the two super-folding elements. The mechanism considered a model with the "elements in a series" in which the quasi-inextensional mode persists during the first phase of deformation up to the intermediate configuration whereupon the extensional mode takes control of the crushing process.

The super folding element consists of four trapezoidal elements which are connected by two horizontal cylindrical surfaces and two inclined conical surfaces. The central region...
bounded by four moving circular arcs must form a section of a toroidal surface, as shown in Fig 2.10. This deformation mode will be called a basic folding mechanism.

The initial geometry of the super folding element is defined by the height $2H$, total width $C$, which is the length of the segments AB and BC, and the angle $2\phi_0$ between two adjacent plates. The current geometry is described either by the crushing distance $\delta$ or the angle of rotation of the side panels $\alpha$ or the horizontal displacements of the point B.

The plastic folding of the element involves four different deformation mechanisms. These are (referring to Fig 2.10 c, below):

- Bending along stationary and moving hinge lines.
- Rolling deformations.
- Deformation of a ‘floating’ toroidal surface.
- Opening of conical surfaces.

Wierzbicki pointed out that two-thirds of the plastic energy is always dissipated through inextensional deformations at stationary and moving plastic hinge lines. The extensional deformations, confined to the small fraction of the total area of the shell, are responsible for the remaining one-third of the dissipated energy.
Using the super folding element method, they calculated the mean crushing force $P_m$ as

$$P_m = 9.56\sigma_0 b^{1/3}t^{5/3}$$

(2.20)

where $\sigma_0$ denotes the flow stress of the material, $b$ the sectional width and $t$ the wall thickness. Abramowicz later changed the constant 9.56 to 13.06. The half wavelength $H$ for the folding deformation can be calculated by

$$H = 1.276b^{2/3}t^{1/3}$$

(2.21)

A list of closed-form solutions of the progressive crushing of thin-walled tubes by various researchers are summarized, below in Table 2.1.
Table 2.1 List of theoretical models

<table>
<thead>
<tr>
<th>Author</th>
<th>Equation</th>
</tr>
</thead>
</table>
| Alexander (1960)        | \[
\frac{P_m}{M} = 20.73 \sqrt{\frac{2R}{t}} + 6.283 \]
|                         | \[
\frac{H}{R} = 1.905 \sqrt{\frac{t}{2R}}\]                                                  |
| Johnson (1972)          | \[
\frac{P_m}{M} = 2\sqrt{3\pi}(R/t)(H/R) + 2\pi(R/t)(H/R)^2/\sqrt{3} + 2\pi^2(R/H) + 2\pi \\
\frac{H}{R} = [(\pi t/\sqrt{3R})(1 + 2(H/R)/3)^{-1}]^{1/2}\] |
| Abramowicz and Jones    | \[
\delta_e = 2H - 2X_m - t\]                                                                   |
|                         | \[
\frac{P_m}{M} = 20.79 \sqrt{\frac{2R}{t}} + 11.9 \]
|                         | \[
\frac{H}{R} = 1.76 \sqrt{\frac{t}{2R}}\]                                                  |
| Wierzbicki and Abramowicz (1992) | \[
\frac{P_m}{M} = \frac{8\pi}{t}H + \frac{\pi^2R}{H}\]
|                         | \[
\frac{H}{R} = \sqrt{\frac{\pi}{4}} \sqrt{\frac{t}{2R}} = 0.886 \sqrt{\frac{t}{2R}}\] |
| Singace (1995)          | \[
\frac{P_m}{M} = 31.49 \sqrt{\frac{R}{t}} + 5.632\]                                       |
| Pugsley (1979)          | \[
\frac{P_m}{M} = 0.326(\frac{2R}{t}) + 217.7 \text{ (diamond mode)}\]                     |
| Wierzbicki (1983)       | \[
\frac{P_m}{M} = 62.88(\frac{2R}{t})^{1/3} \text{ (diamond mode)}\]                       |
| Abramowicz and Jones    | \[
\frac{P_m}{M} = A_1(\frac{2R}{t})^{1/2} + A_2 \text{ (diamond mode)}\]                  |

The theoretical analysis outlined here was developed for the thin-walled tubes subjected to a static axial load. In the case of the dynamic progressive buckling of thin-walled tubes, although inertia effects may be neglected, the influence of material strain rate sensitivity must be considered for many materials [82]. Thus, if a tube is made from a
strain-rate sensitive material, then it is necessary to modify the plastic flow stress in the equation in order to cater for the enhancement of the flow stress with strain rate.

2.3.4 Theoretical prediction for multi-cell thin-walled tubes

While single thin-walled tubes have still been addressed by various authors, new structures, such as multi-cell tubes, composite thin-walled tubes and foam filled tubes, are becoming the focus of many researchers in recent years. A number of authors have contributed to these topics.

High weight efficient energy absorption is one of the major objectives in impact structure design. As a relatively new class of sectional configuration, multi-cell thin-walled tubes exhibit exceptionally high capacity of energy absorption, which have recently drawn increasing attention in the research community and automotive industry. But due to the complexity of the problem, only a limited literature exists on such structures.

In the Super Folding Element method [80], Wierzbicki and Abramowicz pointed out the number of angle elements on the cross section of a tube decides, to a large extent, the efficiency of energy absorption. It is therefore desirable to design thin-walled multi-cell tubes for weight efficient energy absorption.

Chen and Nardini [83] carried out experimental studies on the axial crushing behavior of single-hat and double-hat absorbers with an internal flange sections, and found that the latter improves the Specific Energy Absorption (SEA, energy absorption per unit structural mass) by about 20% compared to the former.
Chen and Wierzbicki [84] adopted a simplified approach to derive the analytical solution for mean crushing force of multi-cell sections. Rather than building a model consisting of trapezoidal, toroidal, conical and cylindrical surfaces with moving hinge lines, they proposed a basic folding element consisting of 3 extensional triangular elements and 3 stationary hinge lines, as shown on Fig 2.11. The crushing force can also be divided into three parts: the average crushing force of empty column, the internal web and the interaction effect between them.

The membrane energy $W_m$ dissipated during one wavelength crushing can be evaluated by integrating the extensional and compression area (shaded area in Fig. 2.11)

$$W_m = 2M \cdot H^2 / t$$

(2.22)

The bending energy $W_b$ was calculated by summing up the energy dissipation at stationary hinge lines. For each flange, three horizontal stationary hinge lines are developed. Then

$$W_b = 2\pi \cdot M \cdot L$$

(2.23)

where $L$ denotes the total length of all flanges. Assuming that the effective crush distance is about 70-75% of the wavelength, the mean crushing force $P_m$ can then be
obtained

\[
\frac{P_m}{M} = \frac{4}{3} \left( \frac{H}{\gamma} + \pi \frac{L}{H} \right)
\]  

(2.24)

The wavelength $H$ can be determined by the stationary condition of the mean crushing force $\frac{\partial P_m}{\partial H} = 0$, which leads to

\[
H = \sqrt{\frac{\pi L t}{N}}
\]  

(2.25)

where $N$ is the number of contributed flanges.

Chen and Wierzbicki [84] pointed out that the double-cell section is more efficient than single section, but the triple-cell section proved to be no better than the double-cell section. Zhang and Cheng [85] then developed a theoretical solution to the square multi-cell columns by dividing the section into three basic components.

The membrane energy $W_m$ was derived by integrating the extensional and compressional areas of three basic components: the corner part, the crisscross part and the T-shaped part, as shown in Fig 2.12.

![Fig 2.12 The division of the multi-cell section [85]](image)
1. The corner part
After deformation, three membrane elements (one in extension and two in compression) were developed for each flange. The membrane energy dissipated by the corner part during one wavelength crushing was:

\[ W_c = 4M \cdot H^2 / t \]  
(2.26)

2. The crisscross part
The deformation mode of the crisscross part is much more complicated than the corner part. The membrane energy dissipated by the crisscross part during one wavelength crushing was:

\[ W_c = 16M \cdot H^2 / t \]  
(2.27)

3. The T-shape part
The shape of the membrane element of T-shaped part is the same as crisscross part. For all three flanges, two flanges were in compression and another one was unaffected. The membrane energy dissipated by the T-shaped part during one wavelength crushing was:

\[ W_c = 8M \cdot H^2 / t \]  
(2.28)

The whole energy dissipated by membrane deformation is

\[ W_c = (4N_C + 16N_O + 8N_T)M \cdot H^2 / t \]  
(2.29)

where \( N_C, N_O \) and \( N_T \) denote the number of corner, crisscross and T-shape patterns in the cross section, respectively; As shown in Fig 2.12, \( N_C, N_O \) and \( N_T \) are 4, 9 and 12, respectively, for a 4_4 multi-cell section.

Applying the stationary condition \( \partial P_m / \partial H = 0 \), the mean crushing force was given as
\[ P_m = \frac{2}{\eta} M \sqrt{\frac{2(N_c + 4N_D + 2N_T)\pi L}{t}} \]  

(2.30)

where \( H \) is the half-length of a single fold, \( P_m \) denotes the average crushing force, \( \eta \) is the effective crush distance factor.

Kim [86] also derived the analytical solution for the mean crushing force of a multi-cell profiles with four square elements at the corner. He pointed out that the severe deformation of combined bending and membrane deformation takes place near the corners of the column. Then the idea of adding a further folding element to the corner part of a cross section was proposed. As shown in Fig 2.13, the square corner and the circular corner elements were added to the conventional tube.

![Square corner and Circular corner](image)

**Fig 2.13 Variation of cross section of multi-cell tubes [86]**

Compared with the conventional square and circular profiles, the SEA of the new multi-cell structures are reported as increasing by 190% and 165% for the square corner tube and circular tube, respectively.

These solutions show increased energy absorption efficiency when thin-walled single tube was divided into multi-cell tube, which means that multi-cell tubes are more attractive than single tubes. However, due to the complexity of theoretical analysis, how
to design the cross-section profiles for better energy absorption efficiency is still a matter which needs to be analyzed.

2.4 Summary

In this chapter, the investigations to date on the collapse mechanism of thin-walled tube under axial impact are reviewed. It is intended to give a clear, comprehensive scenario of the collapse behaviour of thin-walled tubes under axial impact, which provides a good basis for the further analysis in the later chapters.

The review firstly introduces various designs of energy absorbing structures and the related research work and then introduces the collapse behavior of the thin-walled tube in axial loading and the various theoretical models of single thin-walled circular tubes. Wherever possible, the various contributors to the research in this field are cited and the particular methodologies and the conclusions drawn from their work are described and evaluated.

Through this review, several key points about the collapse mechanism of the thin-walled structures are illustrated, which are presented as follows.

1. The progressive buckling collapse of the thin-walled tube is an efficient energy absorption mechanism, while the Euler buckling is the deformation mode that should be avoided in the engineering design.

2. The mean crushing force is an important parameter to evaluate the energy absorption capability of the thin-walled structures. To achieve high energy absorption, the mean crushing force of the energy absorbers should has a high value.

3. The energy absorbed in the crushing process is mainly decided by the plastic
deformation of the metal shell, which includes bending at plastic hinges and circumferential stretching of the metal between the hinges. The energy absorbers which can produce more local buckling normally have higher energy absorption capability.
Chapter 3

Experimental study of the collapse behavior of the thin-walled metal tube under axial loads

To understand how the thin-walled metal tubes behave under axial crushing load, an experimental study of the collapse behaviour of circular thin-walled metal tubes was carried out in this chapter. The main objective of this experiment was to capture the main features of the crushing behaviour and study the energy absorption characteristics of circular thin-walled metal tubes under axial static and dynamic loads and compare the experimental results with existing theoretical predictions.

3.1 Introduction

As introduced in the last chapter, thin-walled metal tubes under axial loads will collapse in different modes due to various parameters such as geometrical shapes, material properties, boundary and loading conditions. By controlling these parameters, energy absorption capability of the structure can be improved. To achieve the target, we need to understand the main feature of the collapse process of the structure and how the crushing energy is absorbed. From the viewpoints of structural crashworthiness, a favourable energy absorption structure should be collapsed in a predicted and controlled manner and provide a fairly stable reaction force throughout the entire crushing process. Therefore, it is vital to conduct an experimental study to investigate all these characteristics of thin-walled metal tubes.

In this chapter, a series of axial crushing tests on aluminium alloy circular thin-walled tubes have been conducted and compared with various theoretical predictions and empirical relations. The main objectives of this experiment are:
• Understanding how the localized plastic fold forms and develops during the axial collapse

• Comparing the experimental results with existing theoretical predictions to validate these models.

• Comparing the static and dynamic behaviour of circular thin-walled tubes.

3.2 Experimental detail

This experiment study is composed of two parts, the static test and the dynamic test. The quasi-static test was carried out in order to understand how the localized plastic fold forms and develops during the axial collapse and study the relationship between the localized plastic deformation and the corresponding crushing force. Also, the difference between the dynamic and static behaviour has been investigated. In this section, a detailed description is given of the experimental facility and test procedure, with the intention of providing an understanding of the measurements involved and the background needed.

3.2.1 Test procedure

The tube specimens were cut from commercially available aluminium alloy tubes with circular cross sections. The end surfaces need to be perpendicular to the longitudinal axes of the specimens. The dimensions of all the test specimens are presented in Table 3.1. There are over 14 different tubes examined, with non-dimensional ratios D/t from 8 to 70, L/D from 1.5 to 8. The sizes of the tested tubes were as received from the manufacturer.

The static tests were carried out by using an Instron (Model 4507) hydraulic testing machine, with 200 kN capacity, as seen in Fig 3.1. A flat steel-plate was fitted to the moving cross-head of the test machine. This plate was parallel to the base plate of the
test machine. Prior to the start of each experiment, the specimen was placed between the parallel plates of the test machine, in a position perpendicular to the base plate, and was held in place with a small axial compressive load of about 20 to 30 N.

Axial loading was applied by a cross-head moving at a preselected speed 6 mm/min, which produced an average strain rate of the order of $10^{-3}$/s in the plastic deformation zones. The data acquisition rate is set as 5 pts/sec. This slow loading is of a quasi-static nature and variations of deformation rate within this range had no influence on any aspects of the tube crushing.

![Fig 3.1 Instron (Model 4507) 200 kN hydraulic testing machine](image)

The dynamic tests were performed on specimens on a drop-hammer rig, as seen in Fig 3.2. The specimens were struck by rigid masses weighed at 150 kg and at impact velocities up to 9.4 m/s. A flat cylindrical head was made from mild steel. The specimens simply rested on the base of the drop-hammer rig, with the longitudinal axis perpendicular to the base surface.
The impact force was measured directly by a Load cell, which was fixed to the base plate. A laser sensor was fixed on the test rig to measure the axial shortening of the impacted tube. The impact test data acquisition and measurement system used in this study is a 32 channel Nicolet DAQ system. To meet all crash test requirements, the DAQ system records impact at a sampling frequency of 200 kHz. The ADC resolution is set as 10 bits. The total duration of the recording (sweep time) is set as 5 microseconds * 10000 = 50 milliseconds. In order to filter the high frequency noise from the test rig vibration, the cut-off frequency is set at 100 kHz.

The initial impact energies are achieved by selecting different impact mass and height. In the test rig, the heights are chosen 1 and 4.5 meters, which mean the velocities of the impact mass are 4.43 and 9.4 m/s, respectively.

**Fig 3.2 Drop-hammer rig**

### 3.2.2 Material properties

The aluminium alloys used in the tests were made up of alloy A6082 temper T6. All the specimens have been annealed before the test to get higher deformation levels. The
procedure of anneal is that the specimens were heated to temperature 270°C and held there for 40 minutes to relieve stresses in the metal, then the specimens were cooled down slowly in the furnace.

The engineering tensile stress–strain curves of the material were found by standard tensile testing. The tensile testing was conducted using an Instron 200 kN hydraulic testing machine. Three material samples were tested. Because the large-diameter tube in the test cannot be tested in full section, longitudinal tension test specimens were cut from the sidewalls of the test tubes parallel to the direction of extrusion. The samples were machined to the proper dimensions required for the test, according to ASTM standards E8 [87], as shown in Fig 3.3. Through the test, the yield strength, ultimate strength and elongation can be determined.

![Fig 3.3 Geometry and dimensions of material test specimens](image)

(a) The specimen for material test (b) the dimensions of the specimen
During the test, the reduced gage section ensured that the highest stresses occurred within the gage and not near the grips of the Instron load frame, preventing strain and fracture of the specimen near or in the grips. Fig 3.4(a) shows the self-adjusting grips having a surface contour corresponding to the curvature of the tube, which ensure the reduced section not to subject any deformation or cold work. Extensometers were used and verified to include the strains corresponding to the yield strength and elongation at fracture, as shown in Fig 3.4(b).

Fig 3.5 shows the measured engineering stress–strain curve, where the yield stress (0.2% proof stress) of 266 MN/m² and an ultimate stress of 282 MN/m² were obtained from the test conducted at a strain rate of $10^{-4}$ s⁻¹.

![Fig 3.4 Gripping devices and strain measurement](image)

(a) Self-adjusting grips   (b) Extensometers
In this study, the effect of strain rate on the mechanical properties of the aluminium alloy is almost negligible. In contrast to various other ductile metals, such as the steel, aluminium alloys exhibit much less 'strain-rate sensitivity. Jone [52] gave a clear explanation of the strain rate of different metal materials in his book. McGregor [88] conducted a detailed study on the aluminium alloy structure performance and pointed out that the strain-rate sensitivity of the aluminium alloy was very small in comparison with the steel.

### 3.2.3 Static test

The static test was conducted for the purpose of observing the whole crushing process and understanding how the localized plastic fold forms and develops during the axial collapse and study the relationship between the localized plastic deformation and the corresponding crushing force and also study any relationship between the dynamic and static behaviour.

#### 3.2.3.1 Experimental results

A summary of the experimental data and experimental results from the static tests on the circular tubes is presented in Table 3.1 & 3.2.
Table 3.1 The experimental data from the static tests

<table>
<thead>
<tr>
<th>Specimen No.</th>
<th>L (mm)</th>
<th>t (mm)</th>
<th>D (mm)</th>
<th>D/t</th>
<th>L/D</th>
<th>M (kg)</th>
<th>A (mm$^2$)</th>
<th>A1 (mm$^2$)</th>
</tr>
</thead>
<tbody>
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<td>A1</td>
<td>101.6</td>
<td>1.6</td>
<td>50.8</td>
<td>31.75</td>
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<td>0.07</td>
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<td>76.2</td>
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<td>0.32</td>
<td>778</td>
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</tbody>
</table>

Table 3.2 The experimental results from the static tests

<table>
<thead>
<tr>
<th>Specimen No.</th>
<th>Structural effectiveness $\mu$</th>
<th>Solid ratio $\delta$</th>
<th>$\delta_T$ (mm)</th>
<th>$F_{mean}$ (KN)</th>
<th>$F_{max}$ (KN)</th>
<th>$F_{ratio}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>0.342</td>
<td>0.126</td>
<td>66.5</td>
<td>24.6</td>
<td>46.2</td>
<td>1.87</td>
</tr>
<tr>
<td>A2</td>
<td>0.345</td>
<td>0.256</td>
<td>66.5</td>
<td>50.4</td>
<td>71.6</td>
<td>1.42</td>
</tr>
<tr>
<td>A4</td>
<td>0.221</td>
<td>0.084</td>
<td>116.0</td>
<td>23.9</td>
<td>57.5</td>
<td>2.40</td>
</tr>
<tr>
<td>A5</td>
<td>0.359</td>
<td>0.171</td>
<td>99.6</td>
<td>78.7</td>
<td>135.9</td>
<td>1.73</td>
</tr>
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<td>A7</td>
<td>0.158</td>
<td>0.063</td>
<td>148.9</td>
<td>22.8</td>
<td>57.7</td>
<td>2.54</td>
</tr>
<tr>
<td>A8</td>
<td>0.260</td>
<td>0.128</td>
<td>149.0</td>
<td>76.2</td>
<td>135.2</td>
<td>2.08</td>
</tr>
<tr>
<td>A11</td>
<td>0.342</td>
<td>0.126</td>
<td>146.5</td>
<td>24.6</td>
<td>47.1</td>
<td>1.91</td>
</tr>
<tr>
<td>A12</td>
<td>0.355</td>
<td>0.256</td>
<td>89.6</td>
<td>51.9</td>
<td>72.9</td>
<td>1.41</td>
</tr>
<tr>
<td>A14</td>
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<td>232.4</td>
<td>25.5</td>
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<td>2.31</td>
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<td>A15</td>
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<td>228.2</td>
<td>80.5</td>
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<td>1.74</td>
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<td>A18</td>
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<td>0.126</td>
<td>224.0</td>
<td>24.3</td>
<td>45.4</td>
<td>1.87</td>
</tr>
<tr>
<td>A20</td>
<td>0.0639</td>
<td>0.126</td>
<td>329.0</td>
<td>4.6</td>
<td>44.8</td>
<td>9.63</td>
</tr>
<tr>
<td>AA1</td>
<td>0.457</td>
<td>0.126</td>
<td>106</td>
<td>41.1</td>
<td>64.8</td>
<td>1.58</td>
</tr>
<tr>
<td>G5</td>
<td>0.393</td>
<td>0.118</td>
<td>75.5</td>
<td>51.7</td>
<td>69.6</td>
<td>1.53</td>
</tr>
</tbody>
</table>

Fig 3.6 The load-displacement curve and the deformed profile of the specimen A1
Fig 3.7 The load-displacement curve and the deformed profile of the specimen A2

Fig 3.8 The load-displacement curve and the deformed profile of the specimen A4

Fig 3.9 The load-displacement curve and the deformed profile of the specimen A5
Fig 3.10 The load-displacement curve and the deformed profile of the specimen A7

Fig 3.11 The load-displacement curve and the deformed profile of the specimen A8

Fig 3.12 The load-displacement curve and the deformed profile of the specimen A11
Fig 3.13 The load-displacement curve and the deformed profile of the specimen A12

Fig 3.14 The load-displacement curve and the deformed profile of the specimen A14

Fig 3.15 The load-displacement curve and the deformed profile of the specimen A15
Fig 3.16 The load-displacement curve and the deformed profile of the specimen A18

Fig 3.17 The load-displacement curve and the deformed profile of the specimen A20

Fig 3.18 The load-displacement curve and the deformed profile of the specimen G5
Fig 3.19 The load-displacement curve and the deformed profile of the specimen AA1

Fig 3.6 - 3.19 show the load-displacement curves of all the tested specimens, together with the final deformed profiles. As can be seen, certain tubes exhibit the concertina mode, while some other tubes develop the concertina mode firstly, and then transfer to the diamond mode. For the tube A20, a global buckling occurs at the early stage and the average crushing force is only 4.6 kN, which is over five time lower than the other tubes with the same cross section. It proves that the global buckling is an inefficient mode of energy absorption with potentially catastrophic consequences and needs to be avoided in crashworthiness applications.

Fig 3.6 illustrates the load–displacement curve of the tested tubes A1 (D/t = 31.75 and L/D = 2), which is deformed in the concertina mode. When the specimen was fully compressed, five folds were formed. The first fold was formed at the top end of the tube. Each pair of peak in the load-displacement curve was associated with the development of one full fold cycle. The load-displacement curve oscillates about a mean load. A stroke of about 75 % can be obtained at a steady mean load. The first peak force of A1 was measured to be 46.2 kN, which is 87% larger than the value of the average crushing force and about 21% higher than the other peak forces. For most specimens, the first peak force is larger than the other peak forces. This is due to the fact that for the formation of the first peak, the structure is free of all deformations, while the formation of the next one will be influenced by the deformations already formed during the last fold; so the other peaks will be, due to this fact, lower than the first one, because the second lobe is already initiated.
Fig 3.20 shows the relation between the D/t ratio and the F_{ratio} for all the specimens. It can be seen that the force ratios increase with the D/t ratio. It means that as the tube wall is thinner, the initial peak force will be much higher than the average crushing force.

For the specimen A1, A11 and A18, which have the same cross-section and different length, the average crushing forces are almost same. For all the other tubes with the same cross-section area, like A2, A12 and G5, A4 and A14, A5 and A15, the average crushing forces are also very close. It indicates that if the collapse mode is same, the length of the tube has no effect on the crushing force.

Comparing the specimen A1, A4 and A7, they have the same thickness and different diameters. From A1 to A7, their diameters are 50.8 mm, 76.2 mm and 101.6 mm respectively, and their wall cross-section areas are increasing from 255.3 mm² to 510.7 mm². But their average crushing forces are almost same. It means that from A1 to A7, their energy absorbing efficiency drop. The structural effectiveness of these three specimens are 0.342, 0.221 and 0.158, respectively. It indicates that the dimensionless parameter structural effectiveness can effectively capture the energy absorbing characteristics.

Fig 3.21 presents the comparison of the experimental results and some theoretical predictions. The equation 3.1 is proposed in by Mamalis and Johnson [26] and the
equation 3.2 is proposed in reference [89] by Thornton. The equation 1 has a better agreement with the experimental result than the equation 2. There is quite considerable scatter in the experimental results which is probably due to different factors, such as the initial imperfection of the tested tube, different material properties and the different test arrangement.

\[ \mu = \frac{7\delta}{4 + \delta} + 0.07 \]  
\[ \mu = 2\delta^{0.7} \]  

(3.1)  
(3.2)

Fig 3.22 presents the relation between the D/t ratio and the structural effectiveness. Although there are some scatters, it can be seen that as the D/t ratio increases, the structural effectiveness will reduce, which means that the energy absorption capability decreases.

Fig 3.21 Comparison of theoretical prediction and the experimental results

Fig 3.22 The relation between the structural effectiveness and the D/t ratio
Fig 3.23 shows the relation between the D/t ratio and the solidity ratio. When the D/t ratio increases, the solidity ratio will drop. It is easy to understand that as the tube wall is thinner, the value of the solidity ratio will be smaller.

### 3.2.3.2 A typical crushing process

This section will investigate the correspondence between the changing geometry of the collapsing tube and the resulting crushing force in some detail, mainly based on the experimental observations, which will help to understand the collapse mechanism of the thin-walled tube.

Understanding the process of formation of the crushing folds is crucial. In order to understand how the localized plastic fold forms and develops during the axial collapse and study the relationship between the localized plastic deformation and the corresponding crushing force, the crushing process of the specimen H1 to H6 have been presented to describe the typical crushing behaviour. These specimens (H1 to H6) with the same geometries have been crushed and stopped at the different crushing stages in a sequence. Then the deformed tubes were cut into two halves and the surfaces of the cut were polished and photographed. From these sectioned tube, the localized plastic folds can be studied in some detail. All the geometric characteristics of the folds are carefully measured.
Fig 3.24 - 3.29 show the load-displacement curves of all the specimens H1 to H6, together with the final deformed profiles. The specimen H1 to H6 has the same geometry ($D/t = 25.4$ and $L/D = 1.97$). All the tubes are 150 mm in length and 3 mm in thickness. The outside diameter of all the tubes is 76.2 mm. When the specimen was fully compressed, four folds were formed. The first fold was formed at the top end of the tube. Each pair of peak in the load-displacement curve was associated with the development of one full fold cycle.

**Fig 3.24** The load-displacement curve and the final deformed profile for H1

**Fig 3.25** The load-displacement curve and the final deformed profile for H2
Fig 3.26 The load-displacement curve and the final deformed profile for H3

Fig 3.27 The load-displacement curve and the final deformed profile for H4

Fig 3.28 The load-displacement curve and the final deformed profile for H5
Fig 3.29 The load-displacement curve and the final deformed profile for H6

Fig 3.30 shows a sequence of photographs of the crushing stage which correspond to the points on the load-displacement curve marked with the numbers. The crushing process can be divided into two distinct stages: the initial stage and the post-buckling stage. Based on these photographs and curves, the detailed folding process will be examined below.
Chapter 3

Fig 3.30 The photos of the crushing process and the corresponding load-displacement curve
(a) The cross-section view (b) The front view (c) Load-displacement curve

**Initial stage of folding process**

In the initial stages of crushing, it is observed that the first fold will always form at one of the two ends of the tube with a radially outward buckle. The tube wall tends to move radially outwards. Initially, the tube behaves as if its edges are fixed because of the square end faces and the friction between the surfaces in contact. The response is elastic. As loads rise, radially inward forces are generated by the expansion of the radially outward moving parts of the tube. When these forces are large enough to overcome the friction force between the tube end surface and the plate, the edge region undergoes radially inward movements and the tube begin to develop the first local fold, which corresponds to the first maximum load at point (1). In this pre-failure phase the load is increasing up to the point of maximum strength; but little change in the deformed shape of the tube occurs and the strains remain small.

From this point on a dramatic change in collapse mode occurs. The end of the tube slides inwards from its initial position until it is nearly flat, lying partly to the outside and partly to the inside of the tube. The final deformed shape of the tube edge is shown in Fig 3.31 It can be seen that the end surface of the tube will develop severe plastic
deformation in one side of the edge, changing the profile from square surface to round surface, which help to absorb some kinetic energy.

![Severe plastic deformation](image)

**Fig 3.31 The local plastic deformation of the wall end edge**

From point (1), the load will continues to drop until the upper wall of this buckle comes into contact with the rigid supporting plate at point (2), collapse is temporarily halted causing the small stress peak corresponding to point (3), as shown in Fig 3.30. Then, the collapse continues with the formation of an inward fold. Collapse is halted when the inner walls come into contact at point (4). The first fold is completely formed at this point.

**Postbuckling stage of folding process**

When the first fold complete, the next fold start to develop. The load rises to a new local maximum which occurs at point (5) when a new outward fold starts to form and the load will drop again. The first half of the new folding is completed by point (6) when the outer knee formed comes into contact with the first fold. Then the load increases once more to point (7) when the inward fold starts to form and the load drop again. At point (8), the inward fold is fully formed and a folding cycle is completed.
The formation of the first fold is different from the post-buckling stage. The form of the first fold experiences elastic compression. The buckling needs to overcome the elastic resistance. The later plastic buckling can develop from the shape of the inward fold, which make the buckling start easily. That explains why the initial peak is larger than the other peak forces. Also, it can be observed that the value of m is a little bigger for the first fold.

The third and fourth folding period is similar to the second one. The two stress peaks and valleys are at about the same levels and their extent is also approximately the same. By the fifth cycle the lower stress peak has almost disappeared. This tendency for distortion after 3–5 cycles was observed in most of the experiments. Only the specimen A7 and A8 keep the two peaks throughout the crushing process, which have the largest diameter in all the specimens.

Comparing the cross section of all the specimens and their corresponding load-displacement curves, it was found that the lower stress peak is related to the formation of the inward fold. If the inward fold form more difficultly, this lower stress peak may be not drop and keep rising. For most of tubes, after a 3-5 cycles, the folds stack up, which cause the inward fold force is close to the elastic buckling force. Then the reaction force will not drop and keep rising. And the inward fold and outward fold will form simultaneously. For the tube with high D/t ratio and large diameter, this effect is small and the lower stress peak will disappear after more folds are formed, as seen in the specimen A7 and A8.

In Chapter 2, Wierzbicki’s model [59] predicts the lower and higher peak by using the transfer zone model and corresponds very closely to our experimental observation. But in his model, the lower peak values are related to the eccentricity parameter m, which is not right, according to our experimental observation. It has been observed that the value of m is about 0.62 for most of the tubes, but the ratios of higher peak and lower peak are very different for these tubes. According to our experimental observation, the lower and higher peak in the model should be related to geometry variables, like the D/t ratio and the tube diameter.
The value of $m$ has also been investigated. The definition of $m$ is shown in Fig 3.32. In Singace’s paper [60,61,62], the theoretical value of $m$ is equal to 0.65, which is slightly higher than the measured values in this test. All the measured values are listed in Table 3.3. The average measured value is about 0.62. Furthermore, the value $m_1$ in the first fold is larger than the value of other folds, which may be due to the different between the initial folding process and the post-buckling folding process.

Table 3.3 Measured geometrical results for all the specimens

<table>
<thead>
<tr>
<th>Specimen No.</th>
<th>$t$ (mm)</th>
<th>$D$ (mm)</th>
<th>$m$</th>
<th>$m_1$</th>
<th>$t_{inner}$ (mm)</th>
<th>$t_{outer}$ (mm)</th>
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</thead>
<tbody>
<tr>
<td>A1</td>
<td>1.6</td>
<td>50.8</td>
<td>0.60</td>
<td>0.71</td>
<td>2.4</td>
<td>2.0</td>
</tr>
<tr>
<td>A2</td>
<td>3.25</td>
<td>50.8</td>
<td>0.61</td>
<td>0.72</td>
<td>4.8</td>
<td>4.0</td>
</tr>
<tr>
<td>A4</td>
<td>1.6</td>
<td>76.2</td>
<td>0.59</td>
<td>0.70</td>
<td>2.3</td>
<td>1.9</td>
</tr>
<tr>
<td>A5</td>
<td>3.25</td>
<td>76.2</td>
<td>0.62</td>
<td>0.72</td>
<td>4.7</td>
<td>3.8</td>
</tr>
<tr>
<td>A7</td>
<td>1.6</td>
<td>101.6</td>
<td>0.63</td>
<td>0.72</td>
<td>2.2</td>
<td>1.9</td>
</tr>
<tr>
<td>A8</td>
<td>3.25</td>
<td>101.6</td>
<td>0.62</td>
<td>0.71</td>
<td>4.6</td>
<td>3.7</td>
</tr>
<tr>
<td>A11</td>
<td>1.6</td>
<td>50.8</td>
<td>0.62</td>
<td>0.72</td>
<td>2.4</td>
<td>2.0</td>
</tr>
<tr>
<td>A12</td>
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<td>50.8</td>
<td>0.60</td>
<td>0.69</td>
<td>4.7</td>
<td>3.7</td>
</tr>
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<td>A14</td>
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<td>76.2</td>
<td>0.61</td>
<td>0.72</td>
<td>2.3</td>
<td>1.8</td>
</tr>
<tr>
<td>A15</td>
<td>3.25</td>
<td>76.2</td>
<td>0.62</td>
<td>0.72</td>
<td>4.7</td>
<td>3.8</td>
</tr>
<tr>
<td>A18</td>
<td>1.6</td>
<td>50.8</td>
<td>0.62</td>
<td>0.71</td>
<td>2.4</td>
<td>1.9</td>
</tr>
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<td>0.62</td>
<td>0.70</td>
<td>2.4</td>
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<tr>
<td>AA1</td>
<td>2</td>
<td>50.8</td>
<td>0.62</td>
<td>0.72</td>
<td>2.3</td>
<td>1.9</td>
</tr>
<tr>
<td>G5</td>
<td>3</td>
<td>50.8</td>
<td>0.61</td>
<td>0.71</td>
<td>4.7</td>
<td>3.9</td>
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<tr>
<td>H7</td>
<td>3</td>
<td>76.2</td>
<td>0.62</td>
<td>0.72</td>
<td>4.6</td>
<td>3.7</td>
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</tbody>
</table>

Another interesting feature is that the inner thickness of each fold is larger than the corresponding values of the outer one. After each experiment, the deformed inner and
outer thickness of the folds was measured. The thickness at the inner fold is about 35% larger than that of the outer fold.

### 3.2.4 Dynamic test

A series of impact tests have been carried out on the test rig introduced above. In this test, the velocities of the impact mass are 4.43 and 9.4 m/s, respectively. All the specimens tested have the same geometry as the one using in the static test, in order to compare the collapse behaviour of the thin-walled tube under static and dynamic loading condition.

### 3.2.4.1 Experimental results

A summary of the experimental results from the dynamic tests on the circular tubes is presented in Table 3.4.

**Table 3.4 The experimental results from the dynamic tests**

<table>
<thead>
<tr>
<th>Specimen No.</th>
<th>Structural effectiveness µ</th>
<th>Solid ratio δ</th>
<th>δf (mm)</th>
<th>$F_{\text{mean}}$ (kN)</th>
<th>$F_{\text{max}}$ (kN)</th>
<th>$F_{\text{ratio}}$ (kN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>1.08</td>
<td>0.126</td>
<td>63.1</td>
<td>24.9</td>
<td>47.4</td>
<td>1.91</td>
</tr>
<tr>
<td>A2</td>
<td>1.05</td>
<td>0.256</td>
<td>26.9</td>
<td>48.7</td>
<td>77.3</td>
<td>1.59</td>
</tr>
<tr>
<td>A4</td>
<td>0.76</td>
<td>0.084</td>
<td>127.4</td>
<td>26.1</td>
<td>68.2</td>
<td>2.61</td>
</tr>
<tr>
<td>A7</td>
<td>0.55</td>
<td>0.063</td>
<td>173.1</td>
<td>25.1</td>
<td>69.6</td>
<td>2.78</td>
</tr>
<tr>
<td>A8</td>
<td>0.77</td>
<td>0.128</td>
<td>52.3</td>
<td>72.1</td>
<td>167.6</td>
<td>2.32</td>
</tr>
<tr>
<td>A11</td>
<td>1.15</td>
<td>0.126</td>
<td>61.2</td>
<td>26.3</td>
<td>49.9</td>
<td>1.89</td>
</tr>
<tr>
<td>A12</td>
<td>1.13</td>
<td>0.256</td>
<td>87.9</td>
<td>52.5</td>
<td>74.8</td>
<td>1.42</td>
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<tr>
<td>A15</td>
<td>1.12</td>
<td>0.171</td>
<td>79.6</td>
<td>78.1</td>
<td>146.3</td>
<td>1.87</td>
</tr>
<tr>
<td>A20</td>
<td>1.12</td>
<td>0.126</td>
<td>176.4</td>
<td>25.6</td>
<td>49.4</td>
<td>1.91</td>
</tr>
</tbody>
</table>

**Fig 3.33** Comparison between the static and dynamic test for A1 and the deformed profile
Fig 3.34 Comparison between the static and dynamic test for A2 and the deformed profile

Fig 3.35 Comparison between the static and dynamic test for A4 and the deformed profile

Fig 3.36 Comparison between the static and dynamic test for A7 and the deformed profile
Fig 3.37 Comparison between the static and dynamic test for A8 and the deformed profile

Fig 3.38 Comparison between the static and dynamic test for A11 and the deformed profile

Fig 3.39 Comparison between the static and dynamic test for A12 and the deformed profile
**Fig 3.40** Comparison between the static and dynamic test for A15 and the deformed profile

**Fig 3.41** Comparison between the static and dynamic test for A20 and the deformed profile

**Table 3.5** Comparison between static and dynamic tests

<table>
<thead>
<tr>
<th>Specimen No.</th>
<th>$F_{\text{mean}}$ (static test) (kN)</th>
<th>$F_{\text{mean}}$ (dynamic test) (kN)</th>
<th>Increase of $F_{\text{mean}}$ (%)</th>
<th>$F_{\text{max}}$ (static test) (kN)</th>
<th>$F_{\text{max}}$ (dynamic test) (kN)</th>
<th>Increase of $F_{\text{max}}$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>A1</td>
<td>24.6</td>
<td>24.9</td>
<td>1.2</td>
<td>46.2</td>
<td>47.4</td>
<td>2.6</td>
</tr>
<tr>
<td>A2</td>
<td>49.4</td>
<td>49.7</td>
<td>0.6</td>
<td>71.6</td>
<td>77.3</td>
<td>8.0</td>
</tr>
<tr>
<td>A4</td>
<td>23.9</td>
<td>26.1</td>
<td>9.2</td>
<td>57.5</td>
<td>68.2</td>
<td>18.6</td>
</tr>
<tr>
<td>A7</td>
<td>22.8</td>
<td>25.1</td>
<td>10.0</td>
<td>57.7</td>
<td>69.6</td>
<td>20.6</td>
</tr>
<tr>
<td>A8</td>
<td>76.2</td>
<td>77.1</td>
<td>1.2</td>
<td>156.2</td>
<td>167.6</td>
<td>7.4</td>
</tr>
<tr>
<td>A11</td>
<td>24.6</td>
<td>25.3</td>
<td>2.8</td>
<td>47.1</td>
<td>49.9</td>
<td>5.9</td>
</tr>
<tr>
<td>A12</td>
<td>51.9</td>
<td>52.5</td>
<td>1.2</td>
<td>72.9</td>
<td>74.8</td>
<td>2.6</td>
</tr>
<tr>
<td>A15</td>
<td>80.5</td>
<td>78.1</td>
<td>-3.0</td>
<td>139.8</td>
<td>146.3</td>
<td>4.6</td>
</tr>
<tr>
<td>A20</td>
<td>4.6</td>
<td>25.6</td>
<td>457</td>
<td>44.8</td>
<td>49.4</td>
<td>10.3</td>
</tr>
</tbody>
</table>
Fig 3.33 - 3.41 show the load-displacement curves of all the dynamically tested specimens, together with their final deformed profiles. The load-displacement curve of the corresponding static test is also plotted in the same figure. There are high frequency contents on the load-displacement curves, which were assumed to be associated with the vibration of the test rig.

The collapse modes of deformation are generally the same as in the static tests. The only difference occurs at specimens A7 and A20. In the static test, A7 is crushed in a concertina mode throughout the process. While in the dynamic test, it firstly develops three concertina folds, then transfer to the diamond mode. For the specimen A20, in the static test, it develops a global buckling from the early stage. Therefore, it doesn’t provide much resist force. But in the dynamic test, it firstly develops some folds with the concertina mode and the diamond mode at both end of the tube. Then it begins to develop a global buckling.

A comparison is made between the static crushing loads and the dynamic experimental results, as shown in Table 3.5. It is observed that there is no significant increase in the force-displacement curves at these two different loading conditions. For the specimens A1, A2 and A11, which were crushed at the velocity of 4.43 m/s, the increase of the crushing force is only about 1-2%. For all the other specimens, which were crushed at the velocity of 9.4 m/s, the increase of the velocity is about 1-10 %. The experimental results exhibit some scatter which is possibly caused by the initial geometric imperfection and experimental conditions. The comparison proves that the aluminium alloy is a strain-rate insensitivity material. Therefore, in the numerical analysis, the material strain rate effect can be ignored in the definition of the material properties.

3.3 Summary

In this chapter, an experimental study was carried out to study the crush behaviour of the thin-walled metal tube statically and dynamically. The entire crushing process, including the initial stage of collapse, its localization and the subsequent progressive folding, were investigated. The experimental results include the careful measurements
of the geometric characteristics of the folds and the crushing response. Analytical models were used to compare their predictions to the experimental results. The following conclusions can be drawn from the study:

1. A typical steady-state folding period involves two load peaks and valleys associated with buckling and self-contact of the walls inside and out. Under the testing conditions used, it was found that after a few cycles the steady-state folding reverted to a single load peak and valley. This phenomenon is due to the way the folds stacked up, which results in the inward fold more difficult to form. Then the lower peak force will not drop and the inward fold and outward fold will form simultaneously.

2. The relation between the localized plastic deformation and the corresponding crushing force is built by comparing the cross section of series of specimens and their load-displacement curves, which give a deep insight of the collapse mechanism of circular thin-walled tube under axial loads.

3. The relation between the energy absorption capability and the geometrical properties was discussed. Some interesting conclusions are drawn, for example, the structural effectiveness of the thin-walled tube will fall with the rising D/t ratios.

4. The static and dynamic crushing behaviour have been compared. The load-displacement curves are very close. It demonstrates that the aluminium alloy is strain-rate insensitivity material. Therefore, in the numerical analysis, the material strain rate effect can be ignored in the definition of the material properties.
Chapter 4

Numerical simulation of the collapse behaviour of single-cell thin-walled metal tubes under axial loads

The single thin-walled metal tube has been widely adopted as the energy absorber by industry because of its efficient energy absorption characteristics and its low cost. Furthermore, many complex engineering systems consist largely of this simple structural component. Therefore, a full understanding of its collapse behaviour is important for revealing the dynamic behaviour of a much more complex system. In this chapter, numerical investigations of the single thin-walled circular metal tube under axial impact were carried out by using the nonlinear explicit finite element codes ABAQUS/Explicit.

4.1 Introduction

The achievement of the desired energy absorption capability is a major concern in the design of real energy absorbing devices. Even for a structure which is not designed for energy absorption, its energy crashworthiness also needs to be considered, in case some accidents occur. A good energy absorption device should include low initial peak force, high average crushing force and high specific energy absorption (SEA). To achieve these, the main factors that can affect energy absorption must be analyzed. The energy absorption that accompanies structural collapse is mainly affected by three factors:

1. Material properties
2. Structural geometry
3. Loading condition
Material properties under dynamic loading condition are an important factor which affects structural energy absorption capability. The strength of the material is directly related to the resistance capability of the structure. Also, the heat treatment of the material affects the crushing behaviour seriously. Moreover, as the strain rates increase, many materials show an increase in their yield strength which is known as material strain rate sensitivity [90]. Therefore, a good understanding of the mechanical properties of materials under dynamic loading conditions is vital to predict the response of structures under such dynamic loading condition.

The selection of geometrical parameters, such as the ratio of diameter to thickness (D/t) and length to diameter (L/D) and the cross section profile, is another important aspect in determining the energy absorption efficiency of the structural design. They not only decide the different collapse modes, but also affect the average crush force and maximum crush force.

The mode of deformation of the structure under dynamic loading depends strongly on the impact velocity. This may be significantly different from quasi-static collapse. In addition, the velocity sensitivity will also vary for different types and sizes of structure. Therefore, it is necessary to access the velocity effects in structural collapse.

A lot of researches related to single thin-walled tube have been carried out in recent years by various authors [91,92]. However, the complexity of the problem causes the knowledge in this subject still scattered and further researches is needed. The aim of the work reported in this chapter was to investigate the collapse behaviour of the single thin-walled metal tube under axial impact and demonstrate how material properties, section geometry and loading parameters can affect the collapse process and improve energy absorption capability. Firstly, the strain rate sensitivity behaviour of materials is discussed. Then, a series of numerical simulations are reported which were conducted to investigate the effects of geometrical parameters and then develop a collapse mode classification chart which predicted the mode of collapse for the range of given D/t and L/D combinations. Thirdly, effects of the mass and impact velocity of the impactor are discussed.


4.2 Material properties under dynamic loading conditions

4.2.1 Background

The early research on the crushing behaviour of the thin-walled tube was static analysis [93]. Because the energies involved in the tube collapse are much greater than the maximum amount of elastic energy which could be absorbed by the tube, the plastic properties of materials become important in impact analysis. Thus, the rigid-plastic approximation [94] is an appropriate idealization for the material characteristics and this assumption has simplified the theoretical analyses greatly.

In fact, impact is really a dynamic process. When materials are subjected to dynamic loading conditions following high velocity impact, a wide range of strains and strain rates will be experienced. The dynamic plastic collapse of energy-absorbing structures is more difficult to understand than the corresponding quasi-static collapse, on account of two effects which can be described as the "strain-rate factor" and the "inertia factor" respectively [95,96].

Jones [52] systematically discussed the material properties under dynamic loading conditions and summarized the following rules:

1. The material properties will change under the dynamic load
2. Different materials have different sensitivity

The strain rates are an essential consideration in solving practical engineering problems. Jones [8] gives the definition of average axial strain rate in a bar as:

\[
\ddot{\varepsilon} = \varepsilon / t = (\delta / L) / t = V / t
\]  \hspace{1cm} (4.1)

where \(\delta\) is the final deflection, \(L\) is the length of the bar, which is generated in a bar when neglecting any stress wave effects. This effect becomes important in many materials when the strain rates range between 10 and 100 per second, which are characteristic values for high-energy dynamic events. Fig 4.1 shows the stress-strain
curves for different strain-rates of mild steel [97]. This indicates that the plastic flow stress corresponding to a strain rate of 55 s\(^{-1}\) is approximately double the static flow stress. Therefore, it is important to include the strain rate dependence when the steel is defined in the model.

There are different constitutive equations to describe the strain-rate-sensitive behaviour of materials. The empirical Cowper-Symonds uniaxial constitutive equation and Johnson Cook constitutive equation [98,99,100] are commonly used to assess material strain rate effects in structures analysis and are included in ABAQUS material modelling.

### 4.2.2 Cowper-Symonds model

Cowper-Symonds constitutive equation is widely used to describe strain rate sensitivity effects in structures and is defined as

\[
\dot{\varepsilon} = D \left( \frac{\sigma_d}{\sigma_0} - 1 \right)^q
\]
where $\sigma_o$ is the static flow stress, $\sigma_o^{d}$ is the dynamic flow stress at a uniaxial plastic strain rate; $\dot{\varepsilon}$ is the current strain rate and D and q are constants for a particular material. A description of the material properties is given in Table 4.1.

### Table 4.1 Physical properties for different materials

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (kg/m$^3$)</th>
<th>Elastic modulus (GPa)</th>
<th>Poisson’s ratio</th>
<th>0.2% proof stress (MPa)</th>
<th>UTS (MPa)</th>
<th>D</th>
<th>q</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mild steel</td>
<td>7800</td>
<td>209</td>
<td>0.3</td>
<td>220</td>
<td>370</td>
<td>40.4</td>
<td>5</td>
</tr>
<tr>
<td>Aluminium (AA6082-T6)</td>
<td>2700</td>
<td>70</td>
<td>0.33</td>
<td>266</td>
<td>282</td>
<td>6500</td>
<td>4</td>
</tr>
<tr>
<td>Aluminium (AA6060-T5)</td>
<td>2700</td>
<td>70</td>
<td>0.33</td>
<td>180</td>
<td>212</td>
<td>6500</td>
<td>4</td>
</tr>
</tbody>
</table>

High values of D imply low strain rate sensitivity. Fig 4.2 shows the relationship between non-dimensional flow stress and strain rate for aluminium alloy and mild steel defined by Cowper-Symonds equation. It is interesting to note that the dynamic flow stress of mild steel is doubled at a strain rate of 40 s$^{-1}$, while the dynamic flow stress of aluminium alloy only increases by 20%. Doubling the flow stress of aluminium alloy needs a strain rate of 6500 s$^{-1}$. It is evident that the plastic behaviour of mild steel is highly sensitive to strain rate. But the aluminium alloys are insensitive to strain rate.

![Fig 4.2 Strain-rate sensitivity by Cowper-Symonds equation for aluminium alloy and mild steel](image-url)
4.3 Crash energy absorption characteristics of single thin-walled tubes

It has been stated that the thin-walled metal tube is widely used as the energy absorber. For example, the thin-walled tubes with closed hat and rectangular cross-sections are utilized as automobile body structures. Therefore, a full understanding of the collapse behaviour of the thin-walled metal tube is important for building safer structures and also in evaluating existing ones.

4.3.1 Typical collapse characteristics of single thin-walled tubes

As mentioned earlier, the thin-walled circular metal tube, when subjected to an axial force, may develop different modes of collapse, i.e. axisymmetrical mode, non-symmetrical mode, mixed mode or global buckling. Which deformation modes occur in the crushing process depends primarily on the section geometry.

The typical force–displacement curves of thin-walled tubes under axial crushing are shown in Fig 4.3(a). The axial force reaches an initial peak force, followed by a sharp drop and then exhibits a repeated pattern. The initial peak force is governed by initial elastic-plastic buckling and each pair of repeated peaks is associated with the development of one full wrinkle or buckle. Usually, the buckles develop sequentially from one end of the tube so that the phenomenon is known as progressive crushing.

If a thin-walled tube is subjected to a sufficiently severe dynamic axial load, then structural inertia effects produce the phenomenon of dynamic plastic buckling. In this circumstance, the deformed shape of the structure will wrinkle over the entire length when buckled dynamically, unlike the dynamic progressive buckling case with buckling confined to one end.

The value of the first peak is larger than the value of the other peaks. This is due to the fact that for the formation of the first peak, the structure is free of all deformations, the deformations already formed during this fold influence the formation of the next one by producing a local bending at the plastic hinges level; so due to this the other peaks will
be, lower than the first one. The global buckling takes place when the length of the tube increases to a specific value. This is a collapse mode which should be avoided in the engineering design, because the crushing force can drop suddenly when the global buckling takes place, which is quite dangerous in practice, as shown in Fig 4.3(b).

![Fig 4.3 Axial crushing behaviour of the thin-walled circular tube](image)

Johnson and Reid [14] identified the dominant modes of deformation in their paper and pointed out that it depends on geometrical parameters such as the ratios of diameter/thickness (D/t) and length/diameter (L/D) and also on material properties.
Andrews [3] then conducted a comprehensive series of tests on annealed aluminium alloy tubes to investigate the relation between deformation modes and structural geometry and developed a classified chart for collapse modes. Although many researchers surveyed the effect of the structural geometry, a wider range of geometry and materials need to be considered systematically.

4.3.2 Modelling details and some considerations

4.3.2.1 Modelling details

In the work reported in this chapter, a series of numerical simulations of axial impact on tubes were conducted by using the explicit dynamics FE code ABAQUS/Explicit [101]. The full section of tube wall was modelled using shell elements S4R with five integration points through the thickness. The striker is applied as a solid body with an initial velocity. Clamped boundary conditions were applied at the bottom of the column. The contact between the tube and striker was defined using general contact. In ABAQUS/Explicit this allows very simple definitions of contact with very few restrictions on the types of surfaces involved. During progressive formation of plastic folds, using a self-contact interface prevented interpenetration between two folds.

The range of impact velocities covered are similar to those in many practical structural crashworthiness problems for which transverse inertia does not influence the buckling mode. For example, the velocity of 10 m/s is equal to a car driving at a speed of 23 miles per hour. Then the velocity between 10 and 30 m/s is the practical driving speeds required in most of the driving regulations.

Aluminium alloy AA 6082 T6 was adapted as the material of the metal tube wall. The constitutive behaviour of aluminium alloy was based on the Cowper-Symonds elastic-plastic material model, which is appropriate for modelling moderate-rate impacts involving metals [102]. The elastic properties and the Cowper-Symonds material parameters for the elastic-plastic behaviour are the same as in Table 4.1, which is the data generated from the material test in Chapter 3.
When defining plasticity data in ABAQUS, true stress and true strain must be used. ABAQUS requires these values to interpret the data correctly. The first piece of data given defines the initial yield stress of the material and, therefore, the plastic strain value should be zero. ABAQUS approximates the smooth stress-strain behaviour of the material with a series of straight lines joining the given data points. Any number of points can be used to approximate the actual material behaviour; therefore, it is possible to use a very close approximation of the actual material behaviour.

According to the shape of the stress-strain curve, a yield point is not easily defined for aluminium alloys, then an offset yield point is set at 0.2% strain. And because the post-yield hardening is almost linear for aluminium alloys, two data points (yield stress and ultimate tensile stress) are used to define the plasticity behaviour, which is thought that can properly approximate the actual material behaviour, as seen in Fig 4.4. In order to verify this plastic definition, two models are compared, as shown in Fig 4.5. These two models have same geometries and loading conditions and the only difference is the plastic definition. One model uses two data points (offset yield stress and UTS), the other model uses 10 test data points. It can be seen that the load-deflection curves are almost same. Therefore, it proves that the plastic definition of two data points is a proper definition in our simulation.

![Fig 4.4 Stress-strain curves with different plastic definitions](image)
Furthermore, ABAQUS interpolates linearly between the data points and assumes that the response is constant outside the range defined by the input data, when the stress in the material reaches the last data point, the material will deform continuously until the stress is reduced below this value [101].

**4.3.2.2 Modelling considerations for impact simulation**

FEA is a useful and powerful method for structural analysis and design, provided that an accurate and reliable finite element model is obtained, especially for dynamic analysis. The structural impact process is a high speed dynamic event which has some special characteristics. Firstly, the load is applied rapidly and is very severe and the response of the structure changes rapidly. Accurate tracking of stress waves through the plate is important for capturing the dynamic response. Secondly, the structures have large deformation and involve complex contact interaction. Thirdly, the stiffness of the structure changes drastically as the loads are applied and the plastic properties must be defined for the materials.

All of these characteristics are complex nonlinear problems and the predicted dynamic response can be very sensitive to the modelling. Therefore, modelling the impact event...
is a big challenge and must be conducted carefully and the results need to be validated. The following section summaries some important considerations in modelling the impact process.

(1) Material definition

When defining plasticity data in ABAQUS, true stress and true strain should be used. Quite often material test data are supplied using values of nominal stress and strain. In such situations, the expressions presented below should be used to convert the plastic material data from nominal stress/strain values to true stress/strain values. The relationship between the true strain and the nominal strain:

\[
\varepsilon_{\text{true}} = \ln(1 + \varepsilon_{\text{norm}}) \tag{4.4}
\]

\[
\sigma_{\text{true}} = \sigma_{\text{norm}}(1 + \varepsilon_{\text{norm}}) \tag{4.5}
\]

There is a paucity of adequate experimental data on the strain rate sensitive properties of materials with large strains which develops during the crushing of thin-walled tubes and other structural members. This is an area which requires further study since most data generated in the past is for small strains.

(2) Mesh refinement

It is important to use a sufficiently refined mesh to ensure that the results from the simulation are adequate, especially in the dynamic impact analysis. It is good practice to perform a mesh convergence study, simulating the same problem with a finer mesh and comparing the results.

A mesh refinement study has been carried out by analyzing the thin-walled crash using five different mesh densities. The deformed contours used for different mesh sizes are indicated on Fig 4.6 and the force histories are shown in Fig 4.7. The mesh sizes are 0.0075, 0.005, 0.004, 0.003, 0.0025 mm respectively.

It can be seen that the deformation modes and force histories are different by choosing the different mesh size. The tube with course mesh deforms in diamond mode, but the
tubE with refined mesh deforms in ring mode. That means the simulation for thin-walled tube crash is sensitive to the mesh design. Therefore, it is important to choose the right mesh size for models to achieve the accuracy results.

![Deformed shapes of the tube with different mesh designs](image)

**Fig 4.6 Deformed shapes of the tube with different mesh designs**

![Load-time history of the tube with different mesh sizes](image)

**Fig 4.7 Load-time history of the tube with different mesh sizes**

(3) **Contact definition**

Contact simulations in Abaqus/Explicit can utilize either the general contact algorithm or the contact pair algorithm. When surfaces are in contact, they usually transmit shear as well as normal forces across their interface. Thus, the analysis may need to consider
frictional forces, which resist the relative sliding of the surfaces. Often the friction coefficient at the initiation of slipping from a sticking condition is different from the friction coefficient during established sliding. The former is typically referred to as the static friction coefficient, and the latter is referred to as the kinetic friction coefficient. In Abaqus an exponential decay law is available to model the transition between static and kinetic friction.

![Deformation shapes of the tube model with different friction coefficients](image1)

Fig 4.8 Deformation shapes of the tube model with different friction coefficients

![Load-time history of the tube model with different friction coefficients](image2)

Fig 4.9 Load-time history of the tube model with different friction coefficients

![Deformation shapes of tubes with different friction coefficients](image3)

Fig 4.10 Deformation shapes of tubes with different friction coefficients (a) $\mu=0$, (b) $\mu=0.15$
Fig 4.11 Force history for different friction coefficients (a) $\mu = 0$, (b) $\mu = 0.15$

Figs 4.8 and 4.9 show deformed shapes and force history for the same model with different friction coefficient $\mu = 0$ and $\mu = 0.15$. It can be seen that although the initial peak forces are the same, the force history is different for different friction coefficients. Figs 4.10 and 4.11 show another model with different friction coefficient $\mu = 0$ and $\mu = 0.15$. The first model deforms in mix mode and the second model deforms in diamond mode.

The force history in Fig 4.9 and Fig 4.11 show that the friction coefficients have more effect on the diamond mode than the ring mode. It can be explained that there are more self-contact occurred for the diamond mode. It means that friction coefficients have effect on the simulation results and the effect is different for tubes with different geometries. Therefore, when a model is built, the friction coefficient chosen must be close to that in the real application.

4.3.3 Comparison of experiment and simulation results

In order to guarantee that the simulation can accurately predict the response, the simulated results are directly compared with the dynamic test results. Fig 4.12 – 4.17
show the load-displacement curves of the dynamical test specimens and the corresponding simulation result, together with their final deformed profiles.

Fig 4.12 Comparison between the simulation and dynamic test for A1

Fig 4.13 Comparison between the simulation and dynamic test for A2

Fig 4.14 Comparison between the simulation and dynamic test for A4
Fig 4.15 Comparison between the simulation and dynamic test for A7

Fig 4.16 Comparison between the simulation and dynamic test for A8

Fig 4.17 Comparison between the simulation and dynamic test for A12

It can be seen from the curves that the onset of collapse agrees very well with the
experiment results. The difference for the two results is the wavelength of the later folds and the collapse mode for some specimens, as can be seen in specimen A4 and A12. But for all the specimens, the initial peak force and the average crushing force can be correctly predicted. Since the simulation results correlate well with the dynamic test results, it can be useful for evaluating energy absorption in the following study.

4.3.4 Effect of cross sections of tubes

Various cross sections of tubes can influence the energy absorption capability to some extent, because the collapse modes are different for different cross section of tube. The cross sections investigated in this work are triangular, square and circular as shown below.

![Fig 4.18 Deformed shapes of tubes with different cross section profiles](image)

![Fig 4.19 Load-time histories of tubes with different cross section profiles](image)
Figs 4.18 and 4.19 show the deformation shapes of the three different cross section profiles (triangular, square and circular) and their force histories. The mass and the height of the tubes are the same. It can be seen that the deformation modes and the crushing forces are different. The circular-section tube has the largest mean crushing force and the triangular-section tube has the smallest one. This demonstrates that the cross section profile has effects on the crushing process and the energy absorption capability of the tube. In the subsequent investigations carried out, the circular-section tube and multi-cell tubes based on the circular-section tube are analysed systematically.

4.3.5 Effect of geometry of circular section tubes

One of the most important factors that control the tube performance is the section geometry. The geometrical parameters of thin-walled circular tubes include thickness, diameter and length which will affect the energy absorption capability of the structure. In order to study the effect of geometrical parameters, a series of simulations were performed as listed in Table 4.2.

(1) Effect of the tube wall thickness

Firstly, the effect of thickness will be discussed here. Conventionally, thickness increase is considered a method to improve impact resistance. But the thickness increase isn’t unlimited, because thickness also relate to the maximum peak force, which must be below some level. To verify the effect of thickness, a series of models were built and run. In these simulations, the weight of the tubes was kept constant at 0.55 kg, the initial impact velocity was at 10 m/s, and the initial impact energy was equal to for all these models. The variables were the wall thickness and the diameter of the tube.

It is very clear from the simulation results in Figs 4.20 that energy absorption capability of the tube increase for the thicker tubes, because the weight of all the tubes in this simulation are same and the mean crushing force is higher for the tube with the thicker wall. And energy absorption capability will improve as the mean crushing force increases. So this is an advantage for energy absorption. But the initial peak force also
increases with the thickness. Since the maximum force level is associated with safety, the peak force should be kept below a certain level for an absorber even if it can absorb the required energy. Therefore, the thickness of the tube should be controlled in a reasonable range to guarantee a limited peak force and enough energy absorption capability.

(2) Effect of Diameter-to-Thickness ratio

In impact analysis, many variables contribute to the response of complex structures subjected to dynamic loads which produce large deformations and inelastic material behaviour. Thickness is not the only factor that can affect the energy absorption. Diameter and length of the tube also affect the deformation mode. It transpires that dimensionless groups of variables are a better choice to represent the structural response. Therefore, the ratios of diameter to thickness (D/t) and length to diameter (L/D) are used to investigate the effect of the geometrical parameters. The selection of these geometrical parameters is an important aspect of determining the energy absorption efficiency of the structural design.

To investigate the relation between the energy absorption capability and the ratio of diameter to thickness (D/t), a series of simulations were carried out. Research in the past on circular tubes has generally concentrated on tubes with D/t ratios between 4 and 65.
This is common industrial practice. In order to have a broader picture, it was decided to extend the range of research up to approximately D/t = 500. The ranges of nominal tube sizes considered were listed in Table 4.2.

Table 4.2 Model data for dynamic simulation on circular tubes

<table>
<thead>
<tr>
<th>Model No.</th>
<th>Constant</th>
<th>Variable</th>
</tr>
</thead>
<tbody>
<tr>
<td>Model 1</td>
<td>Diameter=0.05m</td>
<td>Thickness: 0.000625m</td>
</tr>
<tr>
<td></td>
<td>Length=0.2m</td>
<td>0.000833m</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.00125m</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.0025m</td>
</tr>
<tr>
<td>Model 2</td>
<td>Thickness=0.0005m</td>
<td>Diameter: 0.0125m</td>
</tr>
<tr>
<td></td>
<td>Length=0.2m</td>
<td>0.025m</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.0375m</td>
</tr>
<tr>
<td></td>
<td></td>
<td>0.05m</td>
</tr>
<tr>
<td>Model 3</td>
<td>Length=0.2m</td>
<td>Wall thickness: 0.0002 m - 0.01 m</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Diameter D: 0.01 m - 0.1 m</td>
</tr>
<tr>
<td></td>
<td></td>
<td>L/D ratio range: 2 to 20</td>
</tr>
<tr>
<td></td>
<td></td>
<td>D/t ratio range: 10 to 500.</td>
</tr>
</tbody>
</table>

Fig 4.21 shows the relationship between the SEA and the D/t ratio, where the impact velocity was kept constant at 10 m/s and the ratio of D/t range from 10 to 500 and the
ratio of L/D range from 4 to 16. It can be seen that as the D/t ratio decreases, SEA increases. It means that the low D/t ratio can improves the energy absorption capability of the thin-walled tube.

(3) Mode classification

Model 3 also investigated the L/D ratio effects. The results in Fig 4.21 show that the ratio of L/D does not affect SEA, but can affect the collapse modes of the tube. In order to get a clear picture on how the geometry parameters affect the collapse process, models with different range of L/D and D/t ratios were built in Model 3. Then a mode classification chart related to the ratios was produced.

A mode classification chart is produced from the simulation results for aluminium thin-walled tubes, as shown in Fig 4.22. The range of D/t considered is D/t = 10 - 500. Collapse modes are observed for L/D = 2 - 20. This chart is divided up into areas which correspond to the different modes of collapse. Note that a logarithmic scale is used for D/t on the chart in order to cover the wider range of D/t values considered.

The blue dashed lines in Fig 4.22 represent the transition zone of the collapse modes. It indicates that the different collapse modes are not only related to the D/t ratios, but are also related to L/D ratios. The figure shows that when L/D = 2 the tube will deform in ring mode for D/t < 50 but it will deform with mixed mode for D/t > 50. When 4 < L/D < 8, the deformation modes are the ring mode and the mix mode type for low D/t ratio and Euler buckling occurs when D/t ratio reaches some limiting value. When 10 < L/D < 14, the deformation changes to the mix mode for D/t < 100 and the Euler buckling will occur for D/t > 100. As the L/D ratio reaches 18, the collapse mode starts with the mixed mode and the Euler critical buckling mode occurs when the D/t ≥ 90.

It can be summarized broadly from the chart that the effective collapse modes occur when D/t < 100. For D/t > 100 and L/D > 8, the Euler buckling occurs. For L/D < 10, the tube is more stable and the Euler buckling only occurs at D/t > 150. Therefore, in the engineering design, the selected D/t ratio should be less than 100 and the L/D ratio should be less than 8.
Fig 4.22 Mode classification chart for the circular tube

The influence of D/t and L/D ratios on the collapse modes has also been investigated by various researchers. Abramowicz and Jone [16,18] proposed their model based on the experimental study. Their experiments show that thicker tubes with D/t less than 80 deform in concertina mode, while those with larger values of D/t deform in diamond mode. Andrews et al. [3] present modes of deformation and load-compression curves of circular tubes of D/t = 4 – 62.5; L/D = 0.17 – 8.75 and reveal that the concertina mode occurs in tubes of D/t varying from 10 to 62.5 and the maximum value of L/D being about 5 -6. The red line in Fig 4.22 shows the transition zone from Andrews’ experiment, where the deformation changes from the concertina mode to diamond mode or mix mode.

Comparing our simulation results to their experimental results, the transition zones are quite different, especially for the global buckling. Several factors can contribute to this difference. Firstly, the extent of work hardening during manufacture and subsequent annealing has a big influence on the collapse mode. Secondly, the process of manufacturing by the extrusion of aluminium leads to geometric imperfections, which can affect the deformation mode and the mean crushing force. The geometric imperfection can induce the global buckling to occur much early than the simulation.

Another interesting phenomenon observed in the simulation is the diamond mode which
has multiple corners (or lobes). It was observed that for tubes with an increasing D/t ratio, the number of circumferential lobes also increased from 3 up to 5 or 6, as seen in Fig 4.23. At high values of D/t (>200), the number of lobes often varied (in one case erratically between 3, 4 and 5 lobes). The number of lobes, N, was not always an integer—for example, in some cases it was observed that a relatively stable pattern was with 3.5 lobes in a spiralling arrangement. In other cases the lobes were simply incompletely formed.

For all tubes within the range simulated, the concertina collapse mode absorbs more energy per unit length of tube than other modes. The diamond mode and mix mode absorb slightly less energy than the concertina mode. This is because the energy absorption mainly depends on the amount of plastic deformation which takes place under axial loading. When the tube length is greater than the critical length for the given tube, it deforms in the global buckling mode, which is an inefficient mode of energy absorption. The resistant force of this mode will fall sharply and needs to be avoided in crashworthiness designs.

![Diamond modes with circumferential lobes for different D/t ratios](image)

**Fig 4.23 Diamond modes with circumferential lobes for different D/t ratios**
4.3.6 Effect of impact mass and initial impact velocity

The collapse process under axial impact is significantly different from quasi-static collapse. Karagiozova [103,104,105] demonstrated that the inertia characteristics of the tube, together with the material properties, determine particular patterns of the axial stress wave propagation, thus, causing either dynamic plastic or dynamic progressive buckling to develop during the initial phase of the shell response.

Calladine [95,96] pointed out there are two types of structures as shown in Fig 4.24. Type I has a relatively "flat topped" static load-deflection curve, while type II has a "steeply falling" curve. The deformation of type II specimens is much more sensitive to impact velocity than that of type I specimens. The typical load-deflection curve for thin-walled tubes belongs to the type II structure. This means the thin-walled metal tubes are sensitive to impact velocity. Therefore, it is important to examine the effect of mass and initial impact velocity. In order to assess these effects, a series of simulations were performed. The thin-walled tube of 50 mm diameter, 100 mm length and 4 mm thickness is used in the simulation. The weight of the tube is 0.49 kg. The description of the impactor is given in Table 4.3.

![Fig 4.24 Idealized type I and type II structures [95]](image)

(a) Load deflection (b) energy deflection curves
Table 4.3 Dimensions, Velocities and Energies of Impactors

<table>
<thead>
<tr>
<th>Model</th>
<th>Series</th>
<th>Velocity (m/s)</th>
<th>Mass (kg)</th>
<th>Side length (m)</th>
<th>Height (m)</th>
<th>Initial kinetic energy (J)</th>
<th>Energy absorbed (J)</th>
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<tr>
<td>D1</td>
<td>D1_1</td>
<td>10</td>
<td>15.6</td>
<td>0.2</td>
<td>0.05</td>
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<td></td>
<td>D1_2</td>
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<td>31.2</td>
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<td>0.1</td>
<td>1560</td>
<td>1560</td>
</tr>
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<td></td>
<td>D1_3</td>
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<td>0.2</td>
<td>0.2</td>
<td>3120</td>
<td>2432</td>
</tr>
<tr>
<td></td>
<td>D1_4</td>
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<td>0.4</td>
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<td>2954</td>
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<td></td>
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<td>0.8</td>
<td>12480</td>
<td>3281</td>
</tr>
<tr>
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<td>499.2</td>
<td>0.2</td>
<td>1.6</td>
<td>24960</td>
<td>3398</td>
</tr>
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<td></td>
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<td></td>
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<td>3476</td>
</tr>
<tr>
<td>D2</td>
<td>D2_1</td>
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<td>0.2</td>
<td>0.8</td>
<td>3120</td>
<td>2311</td>
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<tr>
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<td>0.2</td>
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<td>12480</td>
<td>3281</td>
</tr>
<tr>
<td></td>
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<td>0.2</td>
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<td>49920</td>
<td>3950</td>
</tr>
<tr>
<td></td>
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<td>249.6</td>
<td>0.2</td>
<td>0.8</td>
<td>112320</td>
<td>3979</td>
</tr>
</tbody>
</table>

(a)                                (b)                               (c)                                 (d)

Fig 4.25 Deformation shape of single thin-walled tube crushed using impactors of different impact mass m
(a) m = 31.2 kg  (b) m = 124.8 kg  (c) m = 249.6 kg  (d) m = 3993.6 kg

The effect of the masses is firstly investigated. Fig 4.25 shows the deformation shapes of the single thin-walled tube crushed by impactors with different masses. Fig 4.26 shows the results for Model D1, where the impact velocity was kept constant at 10 m/s. The masses chosen range from 15.6 kg to 3993.6 kg, which include masses for which the tube was not fully crushed and the mass for which the tube was fully crushed. Thus
Chapter 4

the complete effect of crushing masses can be covered. The geometries of the impactors are listed in Table 4.3.

![Figure 4.26 Force histories for different impact mass m](image)

It should be noted that the masses of the impactors are stated to one decimal place because they are based on the selected geometrical dimensions. It can be seen that there is slight increase in energy absorbed when the mass of the impactor is increased from 62.4 to 3993.6 kg. For the masses 15.6 kg and 31.2 kg, all the kinetic energy are absorbed in the crushing process and the single tube can not be fully crushed by the impactor. The mass chosen in the analyses should be large enough to produce the full crushing.
Fig 4.27 Variation of impact energy absorbed (SEA) with initial kinetic energy (IKE)

Fig 4.27 shows the variation of the specific energy absorbed (SEA) with the initial kinetic energy (IKE) for Model D1. The figure shows that the SEA is linearly proportional to the IKE up to a value of IKE of about 2500 J. Above this energy, the SEA begins to diminish in value and eventually levels out approximately as IKE increases. The value of IKE of 2500 J at which the SEA begins to depart from linearity indicates the maximum IKE required to completely crush the model D1 tube. For an impact velocity of 10 m/s, the corresponding impactor mass required is 62.4 kg. Therefore, the use of this mass at a velocity of 10 m/s will ensure that the tube is completely crushed. When the impactor mass is increased beyond 62.4 kg, the IKE is correspondingly increased, the tube is fully crushed but the SEA is only slightly increased. In fact, this slight increase in SEA is due to an increased axial deformation of the tube as a result of increased impact energy when the mass of the impactor is increased.

Effect of impact velocity is investigated secondly. Figs 4.28 and 4.29 show the results from model D2. In this case, the mass was kept constant at 249.6 kg, while the impact velocity was varied in the range 5–40 m/s. It was observed that as the impact velocity increases, the impact energy absorbed increases slightly. These results also reveal that when the initial impact velocity is increased, the initial peak force and each postbuckling peak force increased and the deformation modes are also different. Fig 4.30 shows the variation of the specific energy absorbed (SEA) with the initial kinetic energy (IKE) for Model D2.
Fig 4.28 Deformation shapes for different impact velocities

(a)                                   (b)                                    (c)                                  (d)

Fig 4.29 Load-time curves for different impact velocities

Fig 4.30 Variation of impact energy absorbed (IEA) with initial kinetic energy (IKE)
Figure 4.31 shows a comparison of the force history of a single thin-walled tube model D1 and D2 crushed by impactors of different masses, velocities and initial kinetic energies. In case (c), the tube was crushed by an impactor of 2246.4 kg at an impact velocity of 10 m/s. The initial kinetic energy was 112 kJ. In case (a), the tube was crushed with the same IKE of 112 kJ but by a mass of 249.6 kg at an impact velocity of 30 m/s. In case (d), the tube was crushed by an impactor of 3993.6 kg at an impact velocity of 10 m/s, which gives an IKE of 199.7 kJ. In case (b), the tube was crushed with the same IKE of 199.7 kJ but with a mass of 249.6 kg and an impact velocity of 40 m/s.

The force histories show that for the same IKE, the initial peak force and mean crushing force are higher when a higher impact velocity is used. Also, the secondary peaks, which are due to the folding of the tube during the deformation process, are much sharper and higher in magnitude at the higher impact velocity. This means that higher impact velocity can cause more damage in the crushing process because of the higher peak forces. For case (c) and (d), their force histories are almost same, which demonstrates that when impactor mass are much larger than the mass of the tube, the initial effect can be ignored.
4.4 Summary

In this chapter an investigation of the collapse behaviour of single-cell thin-walled metal tube using finite element method has been reported. The numerical analysis presented here has explored the effect of critical parameters on the energy absorption capability, including material properties, section geometry and loading conditions, which govern the collapse behaviour for circular thin-walled metal tube subjected to dynamic axial loads.

The study is intended to gain a clear understanding of the effects of these factors on the energy absorption capability of this structure. Of three main factors, the section geometry is the most important factor to control the collapse behaviour of the tube. The geometry parameters include the thickness of the tube wall, the diameter of the tube and the length of the tube. It is demonstrated that the thicker tube wall can achieve higher specific energy absorption. However, to decide the deformation modes of the tube, its length and the diameter must also be considered. Using dimensionless parameters such as L/D (length/diameter) and D/t (diameter/thickness) ratios is a suitable choice and can reveal much extra information.

In this study, the ratio of diameter to thickness is confirmed to be the most important factors to affect the energy absorption efficiency. The range of tubes considered was D/t = 10 – 500 and L/D = 2 - 20. A collapse mode classification chart for A6082-T6 aluminium tubes has been produced.
Chapter 5

The trigger systems of thin-walled tubes under axial loading

In this chapter, attempts are made to reduce the initial peak force and improve energy absorption characteristics of thin-walled tubes by artificially introducing various types of triggering systems. In the study, quasi-static compression tests were carried out to investigate the energy absorption characteristics of the tubes with different trigger designs. By properly introducing a trigger system on the tube wall, the initial peak force can be effectively reduced. Also the collapse behaviour of the structure can be predicted and controlled.

5.1 Introduction

Reducing the initial peak force is one of the most important considerations in the design of an energy absorber, since the maximum force level is associated with safety. The initial peak force should be kept below a certain force level for an energy absorber even if it can absorb the required energy.

The impact behaviour of thin-walled tube under axial loading condition consists of two stages, the initial compression stage and the subsequent post-buckling stage with large strains and deformations. In chapter 3, the initial stage of the crushing was carefully investigated. It demonstrates that the initial peak force is closely related to the initial compression process. From the viewpoint of energy absorption, the initial compression process is less important, but the initial peak force is governed by this initial elastic-plastic buckling process. In this early stage, the energy is converted firstly to elastic strain energy in the un-deformed tube. Then, at a specific force limit point,
plastic buckling occurs in some local region, where is normally near the end of the tube. From this point on the energy is mainly dissipated by plastic deformation of the tube wall metal. Therefore if the elastic compression can be passed quickly or avoided, the initial peak force will be decreased effectively. This is the reason why a trigger system can effectively eliminate the initial peak force.

In chapter 3, it is observed that the value of the initial peak force is larger than the value of the other peaks. This is due to the fact that for the formation of the first peak force, the thin-walled tube is free of all deformations. But for all the other peak forces, the deformations have already formed during the last fold, which influence the formation of the next one by producing a local bending at the plastic hinges level. Introducing a trigger to the tube is similar to this process. By introducing some kind of dents or removing some materials from the tube wall, the plastic buckling can be induced at very early stage. Therefore the elastic compression which causes the initial peak force can be passed very quickly or avoided.

In this chapter, three different trigger designs are introduced, including the holes or dented grooves on the tube wall and the taper at the end of the tube, which are expected to affect the crushing process and make it possible to reduce the initial peak force and control the collapse mode.

### 5.2 Experimental study

A series of axial crushing tests on thin-walled tubes with different trigger designs are carried out in this section. Three trigger designs are proposed, which include holes on the tube wall, the circumferential indented groove on the tube wall and the taper on the end of the tube. In the test, 19 different tubes are tested, as seen in Fig 5.1. The detailed experimental procedure is introduced below.
5.2.1 Test procedure

The tube specimens were cut from commercially available aluminium alloy tubes with circular cross sections. The dimensions of the test specimens are presented in Table 5.1. The end surfaces of each tube need to be perpendicular to the longitudinal axes of the specimens. All the tests are conducted on an Instron 200 kN hydraulic testing machine. A flat steel plate was fitted to the moving cross-head of the test machine and was parallel to the base plate of the test machine.

Prior to the start of each experiment, the specimen was placed between the parallel plates of the test machine, in a position perpendicular to the base plate, and was held in place with a small axial compressive load of about 20 to 30 N. Axial loading was applied by a cross-head moving at preselected speeds 6 mm/min, which produced an average strain rate of the order of $10^{-3}$/s in the plastic deformation zones.

The aluminium alloys used in the tests were made up of alloy A6082 temper T6. All the specimens have been annealed before the test to get higher deformation levels. The procedure of anneal is that the specimens were heated to temperature 270°C and held there for 40 minutes to relieve stresses in the metal, then the specimens were cooled.
down slowly in the furnace. The engineering tensile stress–strain curves of the material were found by standard tensile testing, which is same as the test in Chapter 3.

5.2.2 First test: A thin-walled tube with holes on the tube wall

The first proposed trigger design is to introduce some holes on the tube wall with different number, size and locations. The main purpose to add these holes on the tube wall is to reduce the initial peak force and control the collapse mode by inducing folds occurring at the specific location.

Fig 5.2 Geometries of the tubes with holes

Depending on the size and location of the holes, a total of nine triggered configurations were defined, as shown in Fig 5.2 and Table 5.1, where L, D, b, c and R denote the key dimensions. All the tubes are 150 mm in length and 2 mm in thickness. The outside diameter of all the tubes is 50.8 mm. The only difference among all the specimens is the
size of the holes and their position on the tube wall. The tube without the hole is referred to as specimen AA1, working as the reference. In specimens B1 to B3, four holes are introduced near the top end with different diameters 6, 8 and 14 mm. The distance between the hole center to the top end of the tube is 20 mm. In models C1 to C3, 4 to 6 layer holes are drilled on the tube wall at even intervals. The holes on each layer are same with the holes on the tube B1. In models D1 to D3, the holes are located at different heights. The diameter of the hole is also same with the tube B1. The experimental data are listed in Table 5.1.

Table 5.1 Experimental data for the first test

<table>
<thead>
<tr>
<th>No.</th>
<th>L (mm)</th>
<th>t (mm)</th>
<th>D (mm)</th>
<th>Volume (mm³)</th>
<th>Mass (kg)</th>
<th>Eₐ (kJ)</th>
<th>ơₐ (mm)</th>
<th>Fmean (kN)</th>
<th>Pfirst (kN)</th>
<th>SEA (kJ/kN)</th>
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<td>AA1</td>
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<td>50.8</td>
<td>37095.9</td>
<td>0.1002</td>
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<td>41.1</td>
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<tr>
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<td>50.8</td>
<td>36914.6</td>
<td>0.0997</td>
<td>3.60</td>
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<td>36.0</td>
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5.2.2.1 Experimental results for the specimen AA1, B1, B2 and B3

Fig 5.3 shows the load–displacement curves of the specimens AA1, B1, B2 and B3, together with the un-deformed shapes and the final deformed profiles. The purpose of these four tests is to examine how the size of the hole can influence on the collapse behavior of the tube.
Fig 5.3 Test results for specimen AA1, B1, B2 and B3
(a) Un-deformed and deformed tube profiles   (b) Load-displacement curve

Model AA1 collapses in the concertina mode and forms seven plastic folds. But the specimens B1, B2 and B3 collapse in the diamond mode from the beginning of the
crushing, being quite different from specimen AA1. Fig 5.4 shows a series of photos of the specimen B2 to exhibit the crushing process of the tube. Clearly, it is the holes that induce the formation of the folds and cause the tube to collapse in the diamond mode. For all the tubes with the holes, the first plastic hinges are always formed at the location of the holes. It is also observed that there are some ductile tearing on the hole region, which may be caused by the severe plastic deformation in the local area.

Comparisons in the mass, the initial peak force, energy absorption and SEA are summarized in Table 5.1. It can be seen that the specimen AA1 has the highest initial peak force. For the specimen B1 to B3, the initial peak forces reduce in a sequence, according the diameter of the holes. The larger is the diameter, the more is the reduction. It is understandable that the plastic fold develops at the hole, the larger the hole is, the less material left to carry the crushing load. Considering the absorbed energy and SEA, the specimen AA1 has the highest value of SEA. For the tubes with the holes, their SEA
all decreases slightly. The specimen B2 has the largest reduction in SEA (12%). This test demonstrates that the initial peak force can be effectively removed by drilling some holes on the tube wall and the value of this force can be controlled by changing the diameter of the holes. But the energy absorbing capability of the tube will be sacrifice a little bit.

5.2.2.2 Experimental results for the specimen AA1, B1, D1, D2 and D3

Fig 5.5 shows the load–displacement curves of the specimens AA1, B1, D1, D2 and D3, together with the un-deformed shapes and the final deformed profiles. The purpose of these five tests is to examine how the position of the hole affects the collapse behavior of the thin-walled tube.

For the tube B1, D1, D2 and D3, the only difference is the position of the hole on the tube wall. From the deformed profiles of these five tubes, it can be seen that the hole position on the tube wall has a big effect on the collapse mode. For the specimen D3, it collapses in the concertina mode and forms six and half plastic folds. While for the specimen D1 and D2, they absorb energy mainly by the axial splitting, other than develop concertina or diamond modes. Fig 5.6 shows a series of photos of the specimen C2 to exhibit the crushing process of the tube. Plastic hinges are initially formed at the hole position, and continue to develop gradually axial splitting. The splitting involves the plastic bending and ductile tearing, which is more complicated energy absorbing mechanisms and absorbs energy in a long stroke and with an almost constant load.

Comparing the initial peak force and SEA, D1 and D2 are slight higher in energy absorption efficiency than the specimen AA1. The increases of SEA are 2.27% and 0.25%, respectively. The initial peak forces are same for the specimen B1, D1, D2 and D3, which are lower than the specimen AA1. It means that the position of the hole has
not effect on the initial peak force, which is only influenced by the size of the hole. Furthermore, the test also demonstrates that tube splitting is an effective way to absorb energy.

![Diagram of tube with dimensions labeled: D, L, R, b]

**Fig 5.5 Test results for specimen B1, D1, D2 and D3**

(a) Un-deformed and deformed tube profiles  (b) Load-displacement curve
5.2.2.3 Experimental results for the specimen AA1, B1, C1, C2 and C3

In order to further investigate the influence of the holes on the tube wall, a series of tubes with more holes on the wall have been tested. Fig 5.7 shows the load–displacement curves of the specimens AA1, B1, C1, C2 and C3, together with the un-deformed shapes and the final deformed profiles. The purpose of these five tests is to examine how to control the collapse behavior of the thin-walled tube by adding different layer holes.

For the tubes B1, C1, C2 and C3, they have different layer holes on the tube wall. In each layer, there are 4 identical holes evenly positioned on the tube wall. All the holes on the tube wall have the same diameter 6 mm. For the specimens C1, C2 and C3, the
collapse process involves the severe plastic deformation and tearing and splitting of the tube wall. Fig 5.8 shows a series of photos of the specimen C2 to exhibit the crushing process of the tube, which can help to understand the collapse behavior of this kind of tubes.

It can be seen from the test that the plastic hinges are initially formed at the center line of one layer holes, as shown in Fig 5.8 (2), and then the plastic hinges on one side of holes begin to move downwards the center line, while the other hinges move symmetrically above the center line on the opposite side of the holes, as can be seen in Fig 5.8 (3-4).

The hole between the two plastic hinges is crushed to form a line and the angle between the line and the horizontal plane increases gradually from 0 degree to 90 degree. When the angle reach about 30 degree, axial splitting occurs in the middle point of the line and continues to develop until the new plastic hinges forms in another layer and develops gradually another axial splitting.

Comparing the initial peak force and SEA, the specimen C1, C2 and C3 are slight lower in energy absorption efficiency than the specimen AA1. Their SEA are 11.3%, 6.5% and 13.1% lower than that of AA1, respectively. The initial peak forces are same for the specimen C1, C2 and C3, which are lower than the specimen AA1. It demonstrates again that the collapse behavior can be seriously affected by drilling some holes on the tube wall.

Normally, the initial peak force can be effectively reduced. How much forces can be reduced will mainly depend on the size of the holes, but not depend on the numbers of holes on the tube wall, because the initial peak force is mainly related to the first buckling occurred and all the other holes will only affect the collapse behaviour in the
later stage. The collapse behavior can be totally different by adding these holes. The tube wall not only can develop the localized plastic deformation, but also can generate some ductile splitting, based on the location of the holes.

Fig 5.7 Test results for specimen B1, C1, C2 and C3
(a) Un-deformed and deformed tube profiles  (b) Load-displacement curve
Through the experimental study, it shows that this trigger design can effectively reduce the initial peak force and affect the collapse behaviour of the thin-walled tube. The initial peak force mainly depends on the size of the trigger, but isn’t affected by locations of the trigger. But triggers will weak the strength of the tube. The bigger trigger size means the loss of energy absorption capability. The effect to the collapse mode mainly bases on the position of the trigger. By controlling the position of the trigger, the collapse mode can be fully different. The energy absorption capability can be improved in some specific positions. Therefore, by choosing the trigger with the right size and position, an energy absorbing structure with higher energy absorbing capability and lower initial peak force can be achieved.
5.2.3 Second test: A thin-walled tube with the taper on the tube wall

In this section, the tapered tube were introduced and investigated, which is another proposed design of the trigger system. A total of four tapered configurations were defined as shown in Fig 5.9 and Table 5.2, depending on dimensions of the taper. All the tubes are 100 mm in length and 3 mm in thickness. The outside diameter of all the tubes is 50.8 mm. The only difference among these specimens is the dimensions of the taper on one end of the tube. The tube without the taper is referred to as specimen F5. The purpose of these five tests is to examine how the dimensions of the taper can affect the collapse behavior of the thin-walled tube. Fig 5.9 shows the load–displacement curves of the specimens F1 to F5, together with the un-deformed shapes and the final deformed profiles. It can be seen that the initial peak forces of all the tapered tubes are reduced considerably than that of specimen AA1. This is mainly due to the folding deformation is easily induced by the thinner tube wall. In this early stage of the crushing, the energy is converted firstly to elastic strain energy in the un-deformed tube. Then, at the limiting point, buckling occurs in some local place. When the initial contact area is smaller, the resist force provided by the elastic compression will be reduced. Comparing the specimen F1, F2 and F3, when the values of b are 1.25, 1.5 and 2, the corresponding first peak forces are decreasing 61.5%, 48.1% and 35.9%, respectively. It demonstrates that the initial peak force can be reduced by decreasing the initial contact area.

Table 5.2 Experimental data for the second test

<table>
<thead>
<tr>
<th>No.</th>
<th>a (mm)</th>
<th>b (mm)</th>
<th>Volume (mm³)</th>
<th>Mass (kg)</th>
<th>Eₐ</th>
<th>δ_f (mm)</th>
<th>F_mean (kN)</th>
<th>P_first (kN)</th>
<th>Reduction of P_first (%)</th>
<th>SEA (kJ/kN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>F1</td>
<td>30</td>
<td>2</td>
<td>42688.0</td>
<td>0.1153</td>
<td>3.46</td>
<td>67.5</td>
<td>41.1</td>
<td>44.6</td>
<td>35.9%</td>
<td>30.0197</td>
</tr>
<tr>
<td>F2</td>
<td>25</td>
<td>1.25</td>
<td>41639.5</td>
<td>0.1124</td>
<td>2.87</td>
<td>67.5</td>
<td>32.9</td>
<td>26.8</td>
<td>61.5%</td>
<td>25.5278</td>
</tr>
<tr>
<td>F3</td>
<td>30</td>
<td>1.5</td>
<td>41530.3</td>
<td>0.1121</td>
<td>3.04</td>
<td>67.5</td>
<td>36.2</td>
<td>36.1</td>
<td>48.1%</td>
<td>27.1110</td>
</tr>
<tr>
<td>F4</td>
<td>40</td>
<td>1.5</td>
<td>40356.9</td>
<td>0.1090</td>
<td>3.00</td>
<td>67.5</td>
<td>34.8</td>
<td>34.2</td>
<td>50.9%</td>
<td>27.5321</td>
</tr>
<tr>
<td>F5</td>
<td>0</td>
<td>3</td>
<td>45050.4</td>
<td>0.1216</td>
<td>3.48</td>
<td>67.5</td>
<td>52.3</td>
<td>69.6</td>
<td>0%</td>
<td>28.6099</td>
</tr>
</tbody>
</table>
In terms of the absorbed energy and SEA, the specimen F1 is more effective in energy absorption than the un-tapered tube F5. While the SEA of the specimen F2, F3 and F4 are lower slightly than that of the specimen F5. Comparing to their final deformation profiles, the specimen F1 and F5 both collapse in a concertina mode, while the specimen F1 forms three and half folds and F5 form three folds. When a fixed length
tube can form more folds, it means it develops more plastic deformation and absorbs more energy. For the specimen F2, F3 and F4, they collapse in a diamond mode. Fig 5.10 shows a series of photos of the specimen F4 to exhibit the crushing process of the tube, which can help to understand the collapse behavior in this kind of tubes. Therefore, it can be concluded that the energy absorption capability of the tapered tube is related to the deformation mode and the initial peak force is decided by the initial contact area of the end surface.

Fig 5.10 The crushing process of the specimen F4

5.2.4 Third test: A thin-walled tube with the circumferential groove on the tube wall

In this section, circumferential indentation triggers were introduced and investigated, which is the third proposed design of the trigger system. The indentation is manufactured by cutting a groove on the tube wall. Three different triggered configurations were defined as shown in Fig 5.11 and Table 5.3, where \( a = 3.5 \) mm, \( R \) is
5 mm, the value of b depends on the location of the groove. All the tubes are 150 mm in length and 3 mm in thickness. The outside diameter of all the tubes is 76.2 mm. The only difference among all the specimens is the location of the groove on the tube wall. The tube without the groove is referred to as specimen H7.

Fig 5.11 Test results for specimen H7, E1, E2 and E3
(a) Un-deformed and deformed tube profiles  (b) Load-displacement curve

Fig 5.11 shows the load–displacement curves of the specimens H7, E1, E2 and E3,
together with the undeformed shapes and the final deformed profiles. The purpose of these four tests is to examine how the position of the groove affecting the collapse behavior of the tube.

In terms of the initial peak force and SEA, it can be seen that the initial peak forces of the specimen E1, E2 and E3 are reduced by 22.3%, 24.2% and 23.9% respectively, comparing to the specimen H7. This is mainly due to the folding deformation is easily induced by the groove. The value of the initial peak force for these three specimens are almost same, which means that the position of the groove have no effect on the initial peak force. The specimen E1, E2 and E3 are slight lower in energy absorption efficiency than the specimen H7. Their SEA are 2.87%, 11.92% and 2.34% lower than that of H7, respectively. The difference in SEA reduction may due to the different collapse modes of these specimens.

Table 5.3 Experimental data for the third test

<table>
<thead>
<tr>
<th>No.</th>
<th>L (mm)</th>
<th>t (mm)</th>
<th>D (mm)</th>
<th>b (mm)</th>
<th>Volume (mm³)</th>
<th>Mass (kg)</th>
<th>Eₙ (J)</th>
<th>δₖ (mm)</th>
<th>Fₘₑₜₖ (kN)</th>
<th>First peak force (kN)</th>
<th>SEA (kJ/kN)</th>
</tr>
</thead>
<tbody>
<tr>
<td>H6</td>
<td>150</td>
<td>3</td>
<td>76.2</td>
<td>0</td>
<td>103484</td>
<td>0.2794</td>
<td>9.54</td>
<td>106</td>
<td>97.8</td>
<td>149.7</td>
<td>34.1438</td>
</tr>
<tr>
<td>E1</td>
<td>150</td>
<td>3</td>
<td>76.2</td>
<td>20</td>
<td>101744</td>
<td>0.2747</td>
<td>9.11</td>
<td>106</td>
<td>86.4</td>
<td>116.3</td>
<td>33.1624</td>
</tr>
<tr>
<td>E2</td>
<td>150</td>
<td>3</td>
<td>76.2</td>
<td>75</td>
<td>101744</td>
<td>0.2747</td>
<td>8.26</td>
<td>106</td>
<td>78.6</td>
<td>113.4</td>
<td>30.0682</td>
</tr>
<tr>
<td>E3</td>
<td>150</td>
<td>3</td>
<td>76.2</td>
<td>130</td>
<td>101744</td>
<td>0.2747</td>
<td>9.16</td>
<td>106</td>
<td>86.6</td>
<td>113.9</td>
<td>33.3444</td>
</tr>
</tbody>
</table>

Fig 5.12 shows a series of photos of the specimen E2 to exhibit the crushing process of the tube. It can be seen that the first fold occurs just below the groove, which is a concertina mode. The second fold develops above the groove and the collapse mode change to a diamond mode. Then the third fold is formed below the first fold in a concertina mode. For the specimen E1 and E3, their first folds are both in a concertina mode and occur just near the groove. But E1 develops a concertina mode in the later stage, while E3 develops a diamond mode.
Fig 5.12 The crushing process of the specimen E2

To further investigate the influence of the circumferential indented groove on the tube wall, a comparison between the experimental results and FE simulation results is carried out. The FE models are built based on the geometries of tested tubes. Fig 5.13 – 5.15 show the load-displacement curves of the test specimens E1, E2, E3 and the corresponding simulation results, together with their final deformed profiles in the tests and simulations.

Fig 5.13 Comparison of test & simulation results of specimen E1
Fig 5.14 Comparison of test & simulation results of specimen E2

Fig 5.15 Comparison of test & simulation results of specimen E3

Generally speaking, the test results in Fig 5.13 – 5.15 show agreement with the simulation results. It means that the simulation can be effective method for the trigger analysis. However, if we compare these three figures, it can be seen that the test results for specimen E2 and E3 show more different than the specimen E1. Considering the final deformation profiles, E1 deforming in concertina mode and E2, E3 deforming in diamond mode, we can tell that the diamond mode is more sensitive to the geometries. If a diamond mode is induced by the trigger, a small geometry imperfection or difference can results in more difference in force history and collapse behaviour.
In order to better compare the test and the simulation, Fig 5.16 shows a series of frames of the simulation results of the specimen E2, which can be compared with the test results shown in Fig 5.12. The comparison shows the general collapse processes are quite close, but the deformed shapes during the crash are slightly different, which explain the different force-deflection curves in Fig 5.14.

5.3 Summary

In this chapter, an experimental study has been carried out to examine how a trigger system can influence the collapse behaviour of the thin-walled tube. Three different trigger designs were proposed, which include holes on the tube wall, the circumferential indented groove on the tube wall and the taper on the end of the tube. The following conclusions can be drawn from the study:

All the trigger designs can effectively reduce the initial peak force and affect the collapse behaviour of the thin-walled tube. How the trigger design affect these features will directly relate to the geometrical properties. For the tube with holes, the initial peak force mainly depends on the diameter of the hole, but isn’t affected by locations of the hole. The bigger the hole is, the lower is the initial peak force. But the strength of the structure will be sacrificed. The energy absorbing capability will be reduced slightly.
Furthermore, the hole in different positions can result in the totally different collapse mode. In some case, material splitting occurs. The initial material failure always takes place at the hole. For the tube with dented groove, the first fold always occurs near the groove. The next fold can be a concertina model or a diamond mode, mainly depending on the position of the groove. For the tapered tube, the initial peak force can be controlled by the dimensions of the taper. When the initial contact area is smaller, the initial peak force will be lower. The collapse mode can also be different due to the different taper dimensions. Generally speaking, the trigger is an effective way to reduce the initial peak force, but slightly sacrifice the energy absorption capability. The bigger trigger size means the loss of energy absorption capability. The collapse behavior of the tube is quite sensitive to the trigger position. But the trigger position doesn’t affect the initial peak force. By controlling the position of the trigger, the collapse mode can be fully different. The energy absorption capability can be improved in some specific position. Therefore, if properly choose the trigger size and the position, a thin-walled energy absorbing structure with higher energy absorbing capability and lower initial peak force can be achieved.
Chapter 6

Numerical simulation of the collapse behaviour of multi-cell thin-walled metal tubes under axial loads

To improve the energy absorption efficiency of single thin-walled tubes, one simple and effective approach is to divide the cross-sectional area of conventional tubes into multiple cells. Such multi-cell tubes are different from foam and honeycomb structures that are currently used by industry and they have the potential to be more efficient as impact energy absorbers. However, because the collapse behaviour of the multi-cell thin-walled tube under axial crash conditions is more complex than that of the single thin-walled tube, this structure has not been widely studied by researchers. In order to better understand their energy absorption capability, numerical studies of the axial crushing behaviour of the multi-cell thin-walled metal tubes have been performed and are reported in this chapter. In addition, a new design of the energy absorption structure based on the multi-cell tube is proposed in the later part.

6.1 Introduction

For thin-walled tubes, there are various ways of improving high specific energy absorption, such as increasing the tube wall thickness, using new materials, putting fillers into the structures and adding internal support ribs to form a multi-cell cross-section. These methods have all been widely adopted by industry. In this chapter, the focus is on the study of the multi-cell thin-walled tube.

The multi-cell thin-walled tube is a kind of structure that is composed of several single
tubes which are connected by supporting ribs by different methods. Fig 6.1 shows some profiles of multi-cell tubes, which will be discussed and compared in this chapter. It should be noted that all these profiles are specific to this thesis; they are not found in any published literature. Moreover, the conventional method of making the multi-cell thin-walled tube is using spot-welding or bonding to connect the sheet metal. In recent years, with the development of new technologies, almost any arbitrary shaped cross-sections can be produced by the extrusion process. Such developments have enabled multi-cell thin-walled structures to become increasingly popular as means of absorbing kinetic energy following a crash.

![Fig 6.1 Different cross-section profiles of multi-cell tubes](image)

Despite being used by industry, the conventional multi-cell cross-section profiles have not been optimized in terms of crash energy absorption and weight efficiency. The aim of the work reported in this chapter is to investigate the crash behaviour of the multi-cell thin-walled metal tubes and compare the various factors which can be used to achieve high energy absorption efficiency.
6.2 Crash energy absorption characteristics of multi-cell circular section tubes

For a multi-cell thin-walled metal tube, it can be understood that the crushing force may consist of two parts: the average crushing force of the empty tube and the interaction effect between the side wall and the internal supporting ribs. The collapse modes of empty columns can be changed drastically by introducing internal ribs. Because the energy absorption depends mainly on how severe plastic deformation is, this can to some extent explain the much higher energy absorption efficiency of multi-cell tubes than the single tube.

In this chapter, in order to investigate how the multi-cell tube can improve energy absorption capability, various multi-cell cross-sections are considered in the analysis and a series of numerical investigations which were conducted are reported. In the analyses, a comparison of the single and the multi-cell tubes was considered firstly and the effect of the number of cells is examined. Then the effects of section geometry are discussed. Thirdly, the effect of the cross-section profiles in the multi-cell tube is studied. Finally, a new structural design was proposed to achieve the optimal energy absorption.

6.2.1 Modelling details

The structures considered in this study are thin-walled multi-cell tubes with various cross-sections. The full section of tube wall was modelled using shell elements S4R with five integration points through the thickness. The loading condition is a striker of mass 62.4 kg crashing the tube at an initial speed of 10m/s. The mass 62.4 kg can guarantee the tube to be fully crushed. Clamped boundary conditions are applied at the bottom of the column. Aluminium alloy AA 6082 T6 was adapted as the material of the
metal tube wall. The material parameters for the elastic-plastic behaviour are the same as in Table 4.1.

In this study, in order to compare the specific energy absorption conveniently, the weights of all the structures are kept the same. The geometrical parameters include wall thickness, diameter and the side length of each cell.

### 6.2.2 A comparative study of single and multi-cell tubes

Several factors can affect the energy absorption of the multi-cell tube, such as the number of cells and the cross-section profiles in the tube. The following section will examine the effect of these factors.

Firstly, comparison is made between a single tube and a multi-cell tube which consists of a same single tube with internal ribs. The thickness of the single circular tube is 1.25mm. To maintain the same weight of tube, the thickness of the multi-cell tube is 0.764 mm. The original and deformed shapes and the crushing force responses are shown in Fig 6.2 and 6.3, respectively.

For the single tube without internal ribs, the force history curve fluctuates after the initial peak force. For the tube with internal ribs, the peak force is the same as that of the single tube, but the load history is much flatter than the single tube and the specific energy absorption increases slightly. For the example shown in Fig 6.3, the SEA of the multi-cell tube with 4 ribs is greater than the SEA of the single tube by about 5%. Because both tubes are equal in weight and length, the contact areas between the tube and the impactor are the same. This means that the initial peak force does not relate to the cross-section profile, but is related to the contact area between the impactor; the tube and the internal rib helps to reduce the oscillation and to make the local buckling more
efficient.

In order to further understand the effect of the internal ribs, a series of simulations with different D/t ratios were carried out. Fig 6.4 shows the SEA of the single tubes and the multi-cell tubes with different D/t ratios. It can be seen that the single tubes with internal ribs can increase SEA by 5-10% depending on the D/t ratios compared to the single tubes without supporting ribs.

Fig 6.2 Deformation shapes of tubes with or without internal reinforcements
(a) Single tube without internal ribs    (b) Single tube with internal ribs

Fig 6.3 Load-time histories of single tube with and without internal reinforcements
(a) Single tube without internal ribs    (b) Single tube with internal ribs
Figs 6.5 and 6.6 show further comparisons between double coaxial multi-cell tubes with and without internal ribs. It can be seen that the initial peak forces for both tubes remain the same but the mean crush force is much higher for the tubes with ribs. Fig 6.7 compares the SEA of the double coaxial multi-cell tubes with and without internal ribs which have different D/t ratios. It can be seen that the tubes with internal ribs can increase SEA by 20-30% depending on the D/t ratios. It illustrates that the double coaxial tube with internal ribs has higher specific energy absorption than single-cell tubes with and without ribs.

It is also observed from Fig 6.3 and Fig 6.5 that the deformation modes are different for the tubes with and without ribs. There are more local buckling produced on the tube with internal ribs. The severe deformation of combined bending and membrane deformation also occurs for the tube with the internal supporting ribs. This can be explained that the internal ribs cause interaction between the tube walls and the internal ribs; they also cause more severe local buckling and hence, absorb more energy.
Fig 6.5 Deformation shapes of double coaxial tubes with or without internal reinforcements

Fig 6.6 Load-time histories of double tubes with and without internal reinforcements

Fig 6.7 SEA of the double tubes with and without internal webs for different D/t ratios
Fig 6.7 shows the SEA of the tubes with and without internal ribs with different D/t ratios. It can be seen that the tubes with internal ribs can increase SEA by 20-30% depending on the D/t ratios compared to the single tubes without supporting ribs.

6.2.3 Effect of angle of internal ribs

The angle between neighbouring ribs seems to affect crushing energy absorption. Three different internal rib design comprising 3, 4 and 6 ribs where the angles between neighbouring ribs are 60°, 90°, 120°, are compared in Figure 6.8. The figure also shows the predicted deformed shapes of the three rib designs after they were subjected to impact loading. The weight of these rib designs are kept the same in order to compare the the energy absorption capability easily. The ribs were analyzed with their axis aligned with the axis of the impact mass of 62.4 kg. One end of each rib was fixed while the other end was subjected to impact loading. The velocity of the impact mass at the time of impact was 10 m/s. General contact was assumed between the impacting face of the mass and the ribs.

![Fig 6.8 Deformation shapes of different angle ribs](image)
In Fig. 6.9, three rib designs are compared. Clearly, the average crushing forces are similar for different angle elements. However, as the neighbouring angle decrease, which means more local buckling is involved in the crushing, the force curves become smoother and stable. This means that changing the angle of neighbouring ribs can not improve energy absorption efficiency, but can help improve the stability of the structures.

In Fig. 6.10, deformation shapes of different angle ribs are shown. (a) 3 internal ribs, (b) 4 internal ribs, and (c) 6 internal ribs.
Figs 6.10 and 6.11 further compare the models that include the 3 rib design inside the single thin-walled tube. The weights of new models are still the same. It can be seen that the force histories are quite similar. It means that the angle of the ribs in the structures only has very small effect on the energy absorption capability of the structure.

6.2.4 Effect of the number of radial & circumferential cells

In order to investigate the effect of the number of cells in thin-walled circular-section tubes, a series of models with different number of coaxial tubes were studied, as illustrated in Fig 6.12. Five of the models had 4 circumferential ribs while the sixth model had 8 circumferential ribs. The diameter range of the cross-section of the tubes is 10-50 mm, and the length of the tubes is 200 mm. The weight of all the multi-cell tubes was kept the same; the thickness of the tubes ranges from 0.56 mm to 3.12 mm.

It can be seen from Fig 6.13 that the 5 coaxial tubes with 8 ribs can produce highest crushing force and the force curve is very smooth. The single tube has the lowest crushing force. It means that, in a fixed space, the number of cells in the tube cross section determines, to a large extent, the efficiency of the energy absorption, but it does
not affect the initial peak force. It helps to explain why a thin-walled tube with multiple cells is an efficient energy absorber. It seems that the mean crushing force of the tubes tends to a certain limit as the number of cells is increased.

Fig 6.12 Multi-cell tubes with different number of radial & circumferential cells

Fig 6.13 Load-time histories of multi-cell tubes with different number of radial & circumferential cells
6.3 Crash energy absorption characteristic of multi-cell square section tubes

The geometrical effect is clearly an important factor to decide the energy absorption capability of a thin-walled tube. In order to further investigate the geometrical effect, the rectangular/square tubes are adopted as the models to analysis the effect. Compared to the circular tube, the rectangular/square can give fixed ratios of different dimensions, which can give a better understanding of the geometrical effect.

6.3.1 Effect of the number of rectangular/square cells

The analysis above shows that the double coaxial tube with internal ribs has better energy absorption capability than the single tube with internal ribs. It means that the number of the cells can affect energy absorption capability. In order to explore this effect, a series of models of square tubes are built and compared, as shown in Fig 6.14. The weight and the size of all the tubes are kept the same.

Fig 6.14 shows the deformation shapes of the tubes with different number of cells and their crushing forces are shown in Fig 6.15. Fig 6.16 shows the energy absorbed during the crush. It can be seen that as the number of the cells increases, the initial peak force drops slightly, but the mean crushing force increases. The tube with 16 cells doubles the mean crushing force of the single tube.
Fig 6.14 Deformation shapes of multi-cell tubes with different number of cells

Fig 6.15 Load-time histories of multi-cell tubes with different number of cells
Comparing the crushing process of the tubes with different cells, it is noted that the tube with more cells can produce much more smooth force curve than single tube. It is also observed that the length of the longer side of a cell determines, to some extent, the deformation shape of the tube and the mean crushing force. For example, for the tubes with 4 cells and 8 cells, as seen in Fig 6.14 (c) and (d), the longer side of each cell is the same and the mean forces of the 8 cell tube is only slightly higher than that of the 4 cell tube. But for the tubes with 8 cells and 16 cells, the length of each of the cell of the 16 cell tube is half of the longer side of the cells of the 8 cell tube. Therefore, the tube with 16 cells has much higher mean crushing force than the tube with 8 cells. It also can be explain from the deformation shapes of the tube. For the tube with 16 cells, there are more local buckling produced than the tube with 8 cells and these buckling are smaller in size. Normally, the less length of the buckling, the more resistance force produced by the buckling. Therefore, the tube with more cells can has higher energy absorption capability.

6.3.2 Effect of section geometry

As mentioned above, the multi-cell thin-walled tube can achieve higher energy
absorption efficiency than the single tube for the same spatial volume and the same weight. As the number of the cells in the tube increases, the energy absorption capability can be improved. The possible reason for that may be related to the ratio of the side length to the thickness (B/t) each cell of the multi-cell tube. Chapter 4 confirms that the ratio of diameter to thickness (D/t) is the main geometrical parameter to affect the SEA for single-cell thin-walled tubes.

In order to illustrate the geometrical profiles can improve the energy absorption capability, a series of simulations are carried out, which include the models with different number of square cell components as shown in Fig 6.17. The thickness and the length of each cell are the same, which means the B/t ratios (side length/thickness) are the same for these tubes. Fig 6.18 plots the SEA for single-cell tubes and multi-cell tubes with different B/t ratios and different number of cells. The B/t ratios are chosen between 10 and 30. The number of the cells ranges from 1 to 12.

![Cross-section profiles of square multi-cell tubes with different number of cells](image)

Fig 6.17 Cross-section profiles of square multi-cell tubes with different number of cells

It can be seen from Fig 6.18 that for the multi-cell thin-walled tube, if the B/t ratios are the same, the SEA only increases slightly as the number of cells increases. For example,
the SEA of the tube with 12 cells increases by 10% more than that of the single tube of the same B/t ratio. But when the B/t ratio decreases, the SEA increases evidently. For example, compared with the 12 cell tube with B/t = 30, there is an increase in SEA of up to 200% for the 12 cell tube with B/t = 10. This demonstrates that the B/t ratio significantly affects the SEA.

Therefore, the main reason that the multi-cell tube can produce higher SEA than single-cell tube in the same volume of space is that multi-cell tubes can produce lower ratios of B/t than the single tube. The number of the cells in the multi-cell tube is a minor reason to improve the energy absorption efficiency. This conclusion is different from some explanation by other researchers, who think it is the number of the cells determines the energy absorption capability of the multi-cell tube.

It can be understood that more metal materials in the multi-cell tube is involved in the local plastic buckling deformation in comparison with the single-cell tube. The local plastic buckling deformation, to a large extent, decides the energy absorption capability of the structures. This is the main reason why the multi-cell tube produces higher energy absorption efficiency.
absorption efficiency.

6.3.3 Effect of reinforcement for square tubes

An important aspect in structural design is choosing an optimal cross-section profile. As mentioned in chapter 2, in single thin-walled metal tubes, the energy is mainly dissipated by membrane deformation and bending deformation along the bending hinge line during the tube crushing process. It means that the energy absorbed depends mainly on two factors, which are how much material is involved in the plastic buckling deformation and how severely these plastic deformations are. Commonly, it is observed that the severe deformation of combined bending and membrane deformation takes place near the corners of a square tube. Then it is natural to consider adding more materials to the corner part to absorb more crash energy. Therefore, the idea of adding ribs to the corner parts of a cross section is proposed.

![Deformation shapes of tubes with corner reinforcements](image)

**Fig 6.19 Deformation shapes of tubes with corner reinforcements**

Fig 6.19 shows the cross-section profiles of the square tubes with supporting ribs at the corners. Fig 6.19(a) shows the original empty square tube in which the side length is 50 mm. Fig 6.19 (b) to (e) show several square tubes where internal supporting ribs are...
added to the corners of the square tube. The variable B represents the distance from the corner to the joint point between the rib and the tube wall, which ranges from 3 mm to 25 mm in the models. The thickness of the tubes changes from 1.46 mm to 2.5 mm, to guarantee that the weight of the structures is kept the same.

It is observed from the simulation results that the more severe deformation happens in the corner of the tube with supporting ribs and the folding length decreases. It means the tube with supporting ribs in the corner can absorb more crash energy and improve the energy absorption efficiency. Fig 6.20 shows the force history of the tubes with and without supporting ribs. The mean crushing force for the tube without supporting ribs is clearly lower than that for other tubes with supporting ribs. For those tubes with supporting ribs, as the position of the supporting ribs moves from the corner to the centre of the tube, the mean force increases firstly, then the mean force begins to decrease at a particular value of B.

![Fig 6.20 Load-time histories of tubes different cross-section profiles](image)

It means that the location of the ribs can affect the energy absorption capability and that
there is an optimal profile existing. In order to further examine the effect of these corner ribs, more models with different ratios of width to thickness were built to find out the optimal position of the supporting ribs added to the tube.

Fig 6.21 compares the mean crushing force of the tubes with different rib positions and B/t ratios. It show that the mean crushing force increases as the ribs location gets closer to the corners at first. But when the distance reaches a specific value, the mean crushing force of the tubes starts to fall. It means that there is an optimal position for best energy absorption efficiency.

![Fig 6.21 Mean crushing force of tubes with different cross-section profiles](image)

**6.3.4 Transition condition between progressive and global buckling**

In Chapter 4, it is demonstrated that the highest SEA can be achieved as the ratio of D/t is about 10 for the single tube. Lower than this value, progressive buckling will not occur. Then global buckling may occur or the initial peak force may be too high. For the multi-cell tube, the transition condition between progressive buckling and global buckling also needs to be considered and compared with the single-cell tube.
The global buckling is an inefficient collapse mode which should be avoided in the structural design. There is a critical tube length which is related to the initiation of the global buckling of the tube. Figs 6.22 to 6.24 show the collapse shape and force histories for the single tube and the multi-cell tube with same width and thickness. The critical length for the single-cell tube with $B/t = 40$ for which the global buckling happens is 230 mm. For the multi-cell tube, the critical length is 110 mm. It can be seen that the single tube has a longer length for global buckling than the multi-cell tube. Figs 6.22 to 6.24 demonstrate that the multi-cell tube has a higher structural energy absorption capability than a single tube of the same mass. However, the stability of the multi-cell tube is less than that of the single tube.

Fig 6.22 The critical length of global buckling of single and multi-cell tubes
Fig 6.23 Load-time histories of single tubes with different lengths h

Fig 6.24 Load-time histories of multi-cell tubes with different lengths h

6.4 A new design of impact energy absorber

A new design of the energy absorber is proposed in this section, based on the analysis presented above. The basic component is composed of two coaxial circular tubes with different diameters and lengths. The tubes are connected by the supporting ribs, as show in Fig 6.25. The procedures of the new design are introduced as follows.
For the double coaxial tubes, adding a circumferential trigger to the large diameter tube will effectively decrease the initial peak force as compared to the undented tube. However, the initial peak force, produced by the small diameter tube, still exists. In order to further reduce this peak force, another circumferential trigger can be introduced to the small diameter tube, which reduces the initial peak force further in this new design, as shown in Fig 6.26.

**Table 6.1 Geometrical dimensions of the double coaxial tube**

<table>
<thead>
<tr>
<th>Models</th>
<th>a</th>
<th>b</th>
<th>c</th>
<th>d</th>
<th>e</th>
</tr>
</thead>
<tbody>
<tr>
<td>h_{outer} (mm)</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>100</td>
<td>100</td>
</tr>
<tr>
<td>h_{inner} (mm)</td>
<td>100</td>
<td>95</td>
<td>85</td>
<td>75</td>
<td>65</td>
</tr>
</tbody>
</table>
In order to control the force-displacement curve and further decrease the initial peak force. A new approach is introduced, namely, adjusting the length of the small diameter tube. Fig 6.27 shows double coaxial tube structures, in which the two tubes are connected by internal rib and the large diameter tube is slightly higher than the tube of small diameter. The lengths of the tubes are listed in Table 6.1. Thus, because only the tube with the large diameter produces the contact force, the initial force will decrease. As the crash process continues, the contact area increase gradually, then the contact force will increase gradually.

Fig 6.27 Coaxial tubes with different inner tube lengths h
(a) h = 100 mm, (b) h = 95 mm, (c) h = 85 mm, (d) h = 75 mm, (e) h = 65 mm

Fig 6.28 Effect of the length h of the inner tube on the force history
The graphs in Fig 6.28 show the effect of the different height of the inner tubes. In the model, the height of the thinner tube will change from 0.065m to 0.095m. This will affect the position of the second peak force, which occurs when the striker starts to make contact with the inner tube. By controlling the height of the thinner tube, the load-deflection curve can be adjusted to desired shape.

(a)                    (b)

Fig 6.29 Comparing tubes with and without trigger

If a circumferential dent trigger is added to the inner tube, as shown in Fig 6.29, the
second peak force also can be effectively eliminated. Then the desired load-deflection curve can be achieved. Comparisons are made between arrangements with and without a circumferential dent trigger on the inner tube, as shown in Fig 6.30. The force deflection curve is almost ideal for energy absorption. The force increases gradually to a certain value when impact occurs and on reaching this value, it will remain almost constant throughout the crash process.

Comparing the double coaxial tubes and triple coaxial tubes and fourth coaxial tubes, as shown in Figs 6.31 and 6.32, the fourth coaxial section showed the highest specific energy absorption, and the double-cell section is more efficient than the single tube. Comparing the crushing force between the fourth coaxial section and the single tube, the SEA increases up to 200%. But as the ratio D/t larger than 150, the mean force decreases to very low level and can not provide enough energy absorption capability. Therefore, achieving highest specific energy absorption must involve considering several factors.

![Fig 6.31 Coaxial tubes with internal ribs and circumferential dent trigger](image)

(a) double coaxial tube,  (b) triple coaxial tube,  (c) quadruple coaxial tube

This new concentric tube structure design includes concentric tubes of different heights (with circumferential dent trigger), connected by internal webs. The advantage of the new structure is that the contact force will gradually increase to a stable value in a
controlled way. The initial peak force almost disappears. And the axial force only fluctuates slightly.

![Force history of coaxial tubes](image)

**Fig 6.32 Force history of coaxial tubes with internal ribs and circumferential dent trigger**

(a) double coaxial tube, (b) triple coaxial tube, (c) quadruple coaxial tube

### 6.5 Summary

The crushing behaviour of multi-cell thin-walled metal tubes has been studied and the results obtained have been described and discussed in this Chapter. Several conclusions can be drawn from the analysis.

Firstly, the energy absorption efficiency can be greatly improved by introducing internal ribs to the tubes. The severe plastic deformation in the local buckling is responsible for the higher energy absorption. The B/t ratio is the key factor that decides the energy absorption efficiency. The number of cells of a multi-cell tube only slightly affects the efficiency of the energy absorption, and it does not affect the initial peak force. But in a fixed space, the number of cells in the tube cross section determines, to a large extent,
the efficiency of the energy absorption.

Secondly, when adding supporting ribs to the corner of the tube, the energy absorption capability can be effectively improved and there is an optimal position to add the supporting ribs to achieve best energy absorption.

Thirdly, changing the angle of neighbouring ribs does not improve the energy absorption efficiency, but can help the stability of the structure.

Fourthly, for the same length of single and multi-cell tube, the global buckling will firstly occur on the multi-cell tube. The stability of single thin-walled tube decreases when adding internal ribs inside the tube.

A new energy absorption structure is proposed. By choosing different cross-section profiles connected by internal webs, different heights of tubes and circumferential dent triggers, an optimal energy absorption structure, which has low initial peak force and higher specific energy absorption, can be achieved.
CHAPTER 7

General discussion and conclusions

This chapter is intended to draw a brief and clear picture for the whole study and summarize all of the work done in this thesis. Firstly, a general discussion is given to summarize all the models examined in this study. Secondly, the main conclusions drawn from this study are listed. Finally, some recommendations for future work are suggested.

7.1 General discussion

There are three main objectives in this thesis. The first objective is to investigate the relation between the localized plastic folding deformation and the corresponding crushing force and give a deep insight of the collapse mechanism of circular thin-walled tube under axial loads. The second objective is to investigate how to reduce the initial peak force and control the collapse behaviour by introducing trigger designs in the thin-walled metal tube. The third objective is to propose a new structural design which is based on the analysis given in this thesis, to achieve high energy absorption efficiency.

The thin-walled tubes examined in this study involve the single circular thin-walled tube and the multi-cell thin-walled tube. The single thin-walled circular metal tube is the main structure to be investigated because it is widely used as the basic components of impact energy absorbers by industry and most theoretical analysis is based on this geometry profile. Also, its profile is relatively simple and can help to understand the collapse behaviour in the research.
The collapse behaviour of a single thin-walled tube, when subjected to an axial impact load, is normally characterized by a lot of progressive buckling folds which are considered to be the most efficient energy absorption modes. These deformation modes can be symmetric or asymmetric, depending primarily on the section geometry.

The study is intended to gain a clear understanding of the effects the section geometry. The geometry parameters include the thickness of the tube wall, the diameter of the tube and the length of the tube. It is demonstrated that the thicker tube wall can achieve higher specific energy absorption. However, to determine the deformation modes of the tube, its length and the diameter must also be considered. Using dimensionless parameters like L/D (length/diameter) and D/t (diameter/thickness) is a suitable choice and can reveal much extra information. In this study, the ratio of diameter to thickness is confirmed to be the most important factor to affect the energy absorption efficiency. Furthermore, the effect of material properties, impact velocity and the mass of impactors are also investigated in some detail and several important conclusions are drawn from the analyses.

To achieve higher energy absorption efficiency, multi-cell thin-walled tubes are proposed and examined. It is demonstrated that the severe plastic deformation in the local area affects the energy absorbed. The multi-cell thin-walled tube can cause more material to be involved in the plastic deformation, produce more severe local plastic buckling, and achieve even higher energy absorption efficiency. The interaction effects between the tube walls contribute a lot to the structural strength and deformation mode. In comparison to the single tube, the specific energy absorption can increase extensively, depending on the profiles of the tube.

The third objective of the thesis is to propose a new structural design. In the study, the new design is a multi-cell coaxial tube with different tube length, which can effectively
reduce the initial peak force and produce high energy absorption efficiency.

Moreover, in order to reduce the initial peak force, three trigger systems are suggested in our study. These trigger systems can effectively keep the initial peak force below a certain level and improve the system safety. These triggers can also change the collapse mode and affect the energy absorption characteristics of the thin-walled tube.

### 7.2 Conclusions

The main conclusions of the thesis are summarized in this section.

Chapter 3 investigates the entire crushing process, including the initial stage of collapse, its localization and the subsequent progressive folding. The experimental results include the careful measurements of the geometric characteristics of the folds and the crushing response. The relation between the localized plastic deformation and the corresponding crushing force is built by comparing the cross section of series of specimens and their load-displacement curves, which give a deep insight of the collapse mechanism of circular thin-walled tube under axial loads. Each peak force in the load-displacement curve has been related to the corresponding fold shape. The relation between the energy absorption capability and the geometrical properties has also been examined. Furthermore, the static and dynamic behaviour has been found very similar, which prove that the aluminium alloy is strain-rate insensitivity material.

Chapter 4 focuses on the crushing behaviour of the single thin-walled circular metal tube. It is demonstrated the effect of the section geometry. A mode classification chart is established, which describe the relation between the deformation mode and the geometry parameters. The ratio of diameter to thickness is proved to be the most important factor to determine the deformation modes. It demonstrates that the higher
impact velocity can cause more damage because of the higher peak forces in the
crushing process.

Chapter 5 proposes three different trigger designs, which include holes on the tube wall,
the circumferential indented groove on the tube wall and the taper on the end of the tube.
All the trigger designs can effectively reduce the initial peak force and affect the
collapse behaviour of the thin-walled tube. The experimental study shows that the initial
peak force mainly depends on the size of the trigger, but isn’t affected by locations of
the trigger. But trigger will weak the strength. The bigger trigger size means the loss of
energy absorption capability. The trigger also can affect the collapse mode, mainly
based on the position of the trigger. By controlling the collapse mode, the energy
absorption capability can be improved. Therefore, by choosing the trigger with the right
size and position, an energy absorbing structure with higher energy absorbing capability
and lower initial peak force can be achieved.

Chapter 6 thoroughly investigates the crushing behaviour of the multi-cell thin-walled
tube. The main factors examined include the number of the cell and section geometry. It
is demonstrated that the ratio of the side width of the cell to the thickness (B/t) is the
main factor to determine the energy absorption capability of multi-cell thin-walled tube.
The number of the cells in the multi-cell tube only can slightly affect the specific energy
absorption. This study also proves that single thin-walled tube has better stability than
multi-cell thin-walled tube. A new multi-cell profile which is composed of coaxial tubes
with different lengths is proposed, which can improves energy absorption capability
evidently.

7.3 Future works

The study presented in this thesis is mainly based on the experiment and numerical
simulation. One main merit of numerical methods is that the range of data chosen can be wide. Many extreme conditions, which are difficult to be achieved in the experimental condition, can be tested by numerical method. Therefore, many results can give useful instructions for the experimental study. In the future work, if more experimental study based on this numerical analysis can be conducted, it can help to improve the reliability of this study, which still needs to be refined and developed further.

Structural failure like ductile tearing isn’t included in the present study. However, the tearing and splitting of the thin-walled metal tube in the collapse process commonly occurs in practice. Therefore, the assessment of the amount of energy dissipated by ductile tearing is an important research topic for the future research, especially in numerical study.

While metal structures were still investigated by various researchers, in recently years, different kinds of new materials, such as functional gradient materials and fiber-reinforced composites, are becoming the focus of many authors. These kinds of materials absorb kinetic energy by various mechanisms and exhibit great energy absorption characteristics. Furthermore, with the development of new manufacturing technologies in the present day, the costs of manufacturing are reduced dramatically, which make the applications of these new materials in industry possible. However, the research on these new materials, although gain much attention, many understandings are still limited. Therefore, it is necessary to do more in-depth study to better understand the energy absorption characteristics of these new materials, experimentally and theoretically.

Another interesting and open question in structural crashworthiness is how to maximize the capability of the energy absorption of structures. Although optimal works was
carried out in the thesis, it is only limited to some comparative study. To achieve the
most weight-efficient design of crush members, structural optimization technique
should be employed, with the help of the clear physical understanding of the crushing
mechanics of thin-walled tubes. However, limited effort has been devoted so far to the
design optimization of crashworthiness criteria despite its great practical importance. So
it would be interesting and helpful to apply the structural optimization to improve the
design of the energy absorption devices.
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