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Numerical Analysis on Crashworthiness Design of Thin Walled Columns under Dynamic Impact Loading Conditions

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Declaration

I certify that this research work titled "*Numerical Analysis on Crashworthiness Design of Thin Walled Columns under Dynamic Impact Loading Conditions*" is my own work. The work has not been presented elsewhere for assessment. The material that has been used from other sources it has been properly acknowledged / referred.

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Sajjad Hussain

2020-NUST-MS-Mech16

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This thesis has been read by an English expert and is free of typing, syntax, semantic, grammatical and spelling mistakes. Thesis is also according to the format given by the university.

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Signature of Supervisor

Copyright Statement

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This accomplished is only achieved with the help of Allah Almighty Who has guided me throughout the duration of this thesis and made me capable to fulfill all the requirements to complete this task.

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Abstract

Transportation and automobiles have become a part of our daily lives and the risks associated with them of unwanted collision have also increased with automobiles being manufactured at a very high level all over the world. With the increasing trend of fatalities and injuries caused by the road traffic accidents, making the vehicles safer has become more critical. In the early years vehicle were made of solid metals and heavy frames which provided enough occupant protection but with time the need of making vehicles fuel efficient the low weight requirement has changed the manufacturing era of automobiles. Although we cannot avoid the occurrence of unwanted collisions but the impact and severity of crash can be reduced. Thin Walled Columns are extensively being used in various engineering applications for the purpose of safety requirements due to their high energy absorption capacity. Their primary purpose is to absorb the excess amount of impact energy during an unwanted collision undergoing deformation. Numerical analysis of crushing behavior using Finite Element Module LS-DYNA is carried out to study the best performance parameters which are specific energy absorption (SEA), mean crushing force (MCF) and low peak force (PF) by varying different design variables including cross sectional geometry, thickness and length. The objective is to find the best optimal crash box which undergoes progressive deformation to absorb collision energy for the application in automobiles to enhance the safety for passenger cabin. To study the effect of number of corners on crashworthiness parameters, polygons having 5 to 8 corners are numerically analyzed and the best of them is further numerically investigated with circular, rectangle and square cross sectional shape. Thin walled tube having hexagonal cross section is found to be the best for energy absorption under axial impact loading while comparing selected crashworthiness parameters.

Key Words: *Crashworthiness, Thin walled tube, Finite element method, Mean crushing force, Specific energy absorption, Peak force*

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1 INTRODUCTION

1.1 Background , Scope and Motivation:

There has been an enormous increase in the production of automobiles for the last 50 years. With the increase in number of vehicles on roads, the vulnerability of unwanted collision of vehicles has also increased resulting in one of the leading cause of deaths worldwide. Fatalities due to road traffic accidents are estimated to be 1.5 million and as well as 50 million peoples are injured each year[1]. The global share of fatalities due to crash accidents is predicted to be the third leading cause in 2020 which was on the ninth position in 1990. This can be explained as the major contribution in increasing road traffic injuries is from low and middle income countries. Developed countries are showing a declining trend in number of accidental deaths over the past two decades due to the implementation of enhanced safety measurements. It has been observed in low and middle income countries that 30-70% orthopedic beds in hospitals are usually served to the patients of road traffic accidents and it has also been the major factor of causing mental disabilities to the affected people[1]. The percentage of occurrence of traffic fatalities is 93% in low and middle income countries and these countries produce 60% of the world's vehicles (WHO). It is estimated that road traffic accidents cost 3% of the gross domestic product for most of the countries worldwide (CDC).

Road traffic injuries accounts for 9% of world deaths which is 1.7 times the deaths caused by HIV/AIDS, tuberculosis and malaria combined [2].

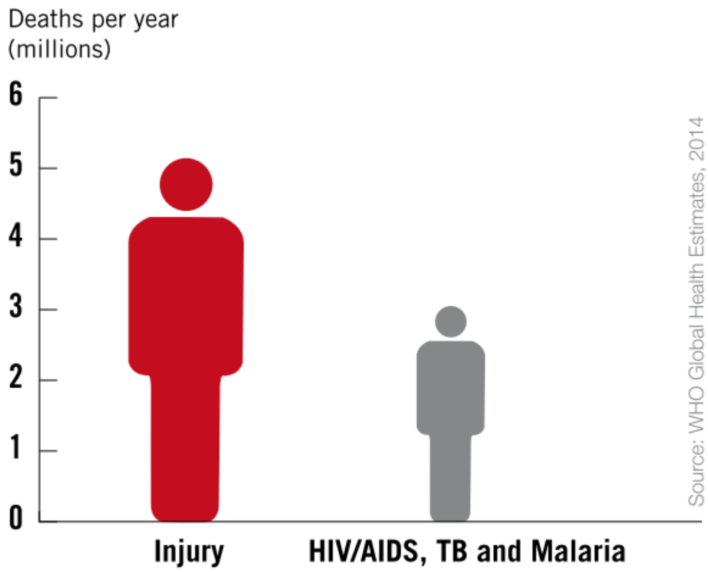


Figure 1-The Scale of Road Fatalities

In 2012 deaths due to road traffic injuries are 24% on the global level and stands as leading cause of the fatalities worldwide[2].

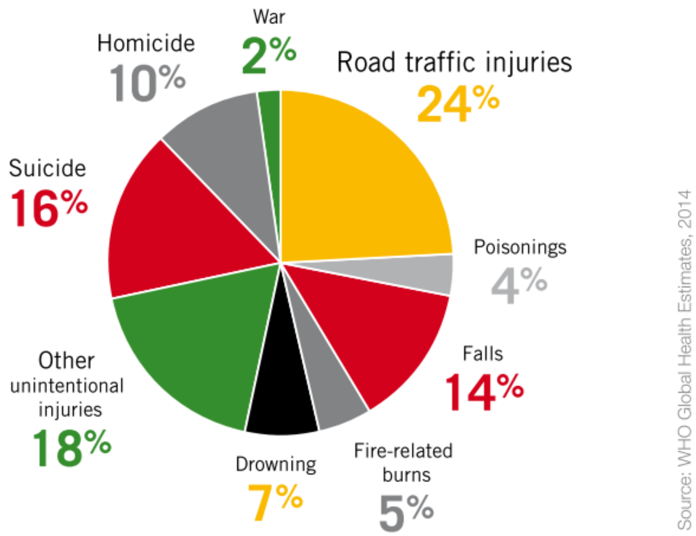


Figure 2-Causes of Injury Deaths

There is an increasing trend of deaths by road traffic injuries in developing countries and it is recorded that death toll due to road crashes have increased from 2.5% to 12.5% in India and Cambodia [2]. This pattern is similar for many countries where implementation of safety strategies is not as better as in developed countries.

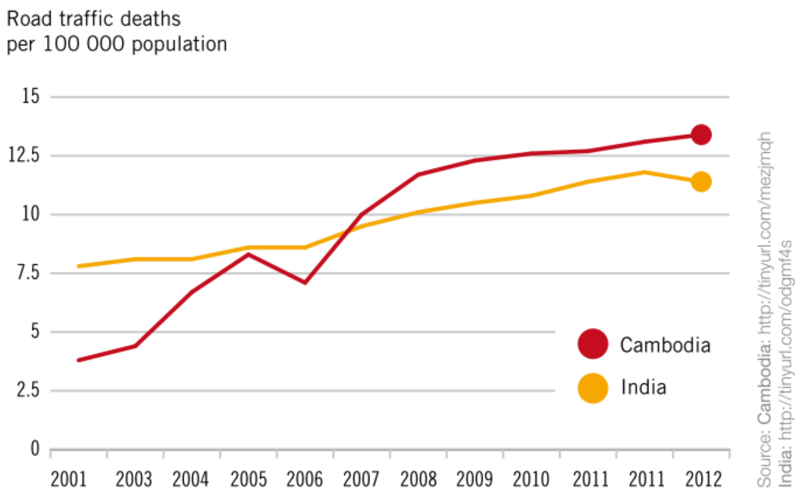


Figure 3-Rising Road Traffic Injuries

Keeping in view the statistical data of developing and under developing countries it is concluded that the road traffic injuries can be minimized with the implementation of safety regulations. It can be made possible by finding out an efficient way of importing the safety regulation from developed countries to the developing countries. A major portion almost 48% of the vehicles being manufactured all around the world is being manufactured in developing countries. There is a dire need of implementing and regulating the manufacturing processes and meeting the required standard parameters for automobiles. Awareness to the general public and incorporation of comprehensive safety program into national planning is inevitable. Making the automobiles safe is as much as necessary as making the transportation system to the standards which eventually lead to reducing the possibility of road accidents.

There are several types of unwanted collision in real world which contributes toward road traffic fatalities and the most abundant is passenger car collision among them[3].

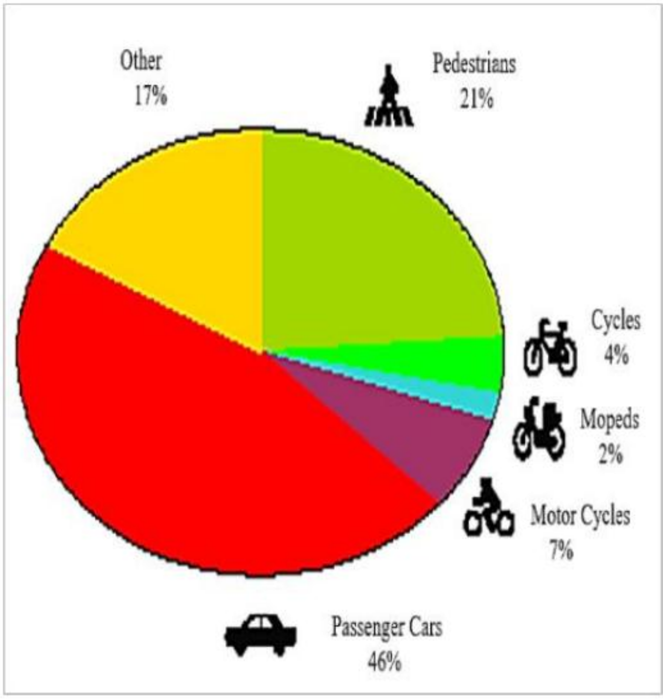


Figure 4-Various Types of Crash Collisions (NHTSA, 2014). In

1.2 Types of Passenger Car Collisions:

There are several types of collisions can occur among the vehicles on road but it can be categorized by analyzing the major scenarios of accidents. There are different types of vehicles running around and the crash events can occur by a large truck hitting a car or bike can collide to a compact car and SUV vehicles,

The four main types of passenger car collisions are:

- Rear Collisions
- Side Collisions
- Front Collisions
- Rollover Crashes

According to the statistical data front collisions constitute more than 50 percent of all accidents and serious injuries associated with it. On the other side rear end collisions constitute of approximately 29 percent of all accidents (NHTSA, 2016).

1.2.1 Front Collision:

Front collision is the most occurring real life scenario and it is the most dangerous in terms severity of impact. Automobiles are made safe from the start of their manufacturing by installing heavy bars and lateral beam at front of the vehicles and with time to time automobiles are equipped with more innovative components to reduce the effect of front collision. The ratio of injuries causing death in case of accidents is the most for the front collision as compared to the other possible collisions. It has been observed that the deaths due to the front collision is still at the top as the vehicles are being made more safe and secure.

According to the statistical data by the Insurance Institute for Highway Safety (IIHS), fatality rate for the passengers in a vehicle due to frontal impact crashes is calculated 58% of the fatalities occurring in a vehicle.

1.3 Problem Statement:

A separate component is installed in between of bumper beam and main assembly to absorb impact energy in case of unwanted collision which is called crash box. Thin walled tubes of metal extrusion are best suitable for this purpose.

The main objective of this study is to investigate the effect of different cross sectional shapes of thin walled columns on energy absorption characteristics subjected to axial impact loading conditions. There are several scenarios in real life occurrence of collisions but the frontal collision is the major one which requires the automobiles to be crashworthy for axial impact.

1.4 Automobile Safety:

From the start of the era of transportation, automobiles are manufactured to provide a safe and secure compartment for its users by using very strong structural elements which includes heavy metals and stiffer frames. The automobiles were heavy in weight and less vulnerable to deformation in case of colliding. With the advent of time the need of vehicles to be more fast has led to the direction of reducing its weight which was also desirable to consume less energy for powering them. These needs have made the vehicles less in weight and safety is ensured by the use of technology. This use of technology has evolved in many forms and set forth the way of introducing very innovative components in vehicles. Now a day's passenger cars have very innovative components for the sake of safety and it includes seat belts, airbags, hydraulic energy absorber and much more.

1.4.1 World Forum for Harmonization of Vehicle Regulation:

The World Forum for Harmonization of Vehicle Regulations is a separated forum which works under the “Sustainable Transport Division” of the “United Nations Economic Commission for Europe (UNECE)” for making the transportation safe while contributing with defining a set of rules for the automobile manufacturers.

World Forum for Harmonization of Vehicle Regulation under the UN Economic Commission for Europe is devoted to make automobiles safe from the initial point of their design. A set of regulations and requirements is documented and fulfillment of these requirements by undergoing certain tests is necessary for all the automobile manufacturers. Automobiles have to meet the criteria in order to get launch as a commercial product for the general public.

There are six basic requirements which an automobile must have to include in its specifications. The requirements are listed below;

- Seat belts and anchorages
- Occupant protection in frontal or side collisions
- Electronic stability control (ESC)
- Pedestrian protection
- Motorcycle anti-lock braking systems

Most countries accept the UN Regulations and apply it the same way in their territories for production of automobiles or they reflect the main subject of the UN Regulations content to show compliance with their unavoidable national requirements to meet the general deviation in the transportation system of their specific region. The countries which do not have their own specific requirements for the transportation system allow the use of UN type-approved vehicles in their region of control. These countries then do not have any objection for the imported vehicles which have met the certain safety standards set forth by the UN Regulation and the registration is permitted in every aspect in their region of control. The most significant countries where UN Regulation are mirrored and compiled in their own set of standards are United States and Canada.

1.4.2 National Highway Safety Traffic Administration (NHTSA):

United States is one of the major exceptions which do not regulate the UN Regulations for their transport system and automobile production. The national highway safety traffic administration (NHTSA) is a body which is dictated by the department of transportation (DOT) to ensure the safety features of vehicles and setting the general rules for the flow of traffic. The national highway safety traffic administration (NHTSA) formulates a set of rules and regulation for the production of automobiles. The automobile manufacturers have to meet the criteria defined within federal motor vehicle safety standards (FMVSS).

The federal motor vehicle safety standards (FMVSS) are the legislative regulations for production of automobiles as a replacement of the UN Regulations which are developed by the "World Forum for Harmonization of Vehicle Regulations". UN Regulations are not as fully applicable in the United States but they are presented by applying some changes in the form of federal motor vehicle safety standards (FMVSS).

The set of rules and tests described in FMVSS are the standards for evaluating the performance of automobiles. This set of requirements are essentials for every commercial vehicle in terms of ensuring that the safety standards are being met which are defined by department of transportation. Satisfying this criterion is a form maximum guarantee that the occupants travelling inside the vehicles are not vulnerable to the high risks if an unwanted collision occurs and the severity of the crash would be minimized.

The standards are defined in a very broad manner so they can cover every aspect of the performance of automobiles. The performance criteria include almost every factor including the pollution control standards as well as making the vehicles safer by assessing the structure to be crash worthy and cost effective.

There are several articles which are designed to cover in detail every single aspect of collision. As stated earlier the front collision is more prone to occur and for that purpose article 208 is the most detailed chapter which is defined for the protection of occupants and main passenger cabin.

1.4.2.1 FMVSS-208:

The objective of a crash test for Federal Motor Vehicle Safety Standard (FMVSS) No. 208 is to check the crashworthiness of a passenger vehicle while protecting its occupants in case of an unwanted real world frontal impact[5].

There are different types of tests are prescribed at 48kph which are listed below;

- Fixed barrier test which covers the full-frontal scenario
- Fixed barrier test which covers the possible oblique impact from front
- Sled test for general impact case
- Fixed deformable barrier having an offset at the front of vehicles
- Deformable barrier which is moving on perpendicular direction
- Deformable barrier which is moving at an oblique angle
- Fixed deformable barrier which is occupying the full front of vehicle

1.4.3 National Automotive Sampling System (NASS):

In the United States a department has been established to arrange the statistical data of crash events by investigating around 4,500 crashes per year. The investigation of these crashes let to know the nature of crash event and the fatalities occurring in the corresponding crash events. The data of this investigation is sampled into categories to compare with the tests regulated by the concerning automotive safety departments for the purpose of knowing how much these tests replicate the real world crashes.

1.5 Crash Management System:

One of the major aspects in automobiles design is to minimize the occurrence and severity of unwanted collision. Safety requirements of automobiles in industries have got much more attention to minimize the repair cost and keeping safe the passengers cabin[6].

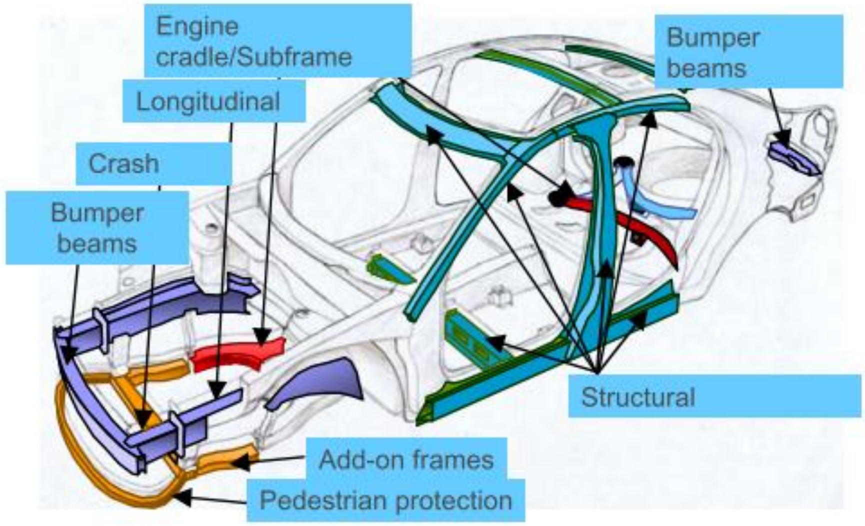


Figure 5: Crash management system of typical vehicle

The different approaches in design to enhance and achieve Automotive safety can be classified in two categories namely “Active” and “Passive” measures.

1.5.1 Active Safety:

Active safety measures in automobiles contain the use of innovative technology which reduces the chances of accidents to minimal level and contributes towards the prevention of a crash in the first place. It can be seen as the active systems in the automobile like Antilock Braking System (ABS), Proximity Sensors to avoid collision by applying auto braking, infrared cameras and Traction Control System.

1.5.2 Passive safety:

Passive safety measures refer to the system consisting of a set of components which contributes to reduce the forcefulness and severity of the crash event.

Crash protection primacies are different with respect to the travelling speed of the vehicles at which the unwanted collision occurs. The speed of the vehicle is much more important as it decides how harsh the impact would be and also categorize the occurring of crash event with respect to the possible surroundings at that speed.

The crash safety priorities are classified to the three main basic categories as mentioned below;

Speed(Km/h)	Main goal of Passive Safety
=<15	Minimizing repair cost
=<40	Protecting pedestrian
>40	To make sure occupant protection

Table 1: Classification with respect to impacting velocity

The term “Crash Management System” is used to define the structural subsystems specifically designed for the purpose of absorbing impact energy in case of accidents while undergoing elastic as well as plastic deformation and keeping safe the main assembly and the passenger’s compartment.

Crash Management System has evolved through different stages in the past 30 years to meet certain performance parameters. It usually consists of the bumper beam and related attaching components to the longitudinal beams on which the main assembly of the automobile is mounted. The usual practice in industry is to attach the rear bumper beam directly to the longitudinal beam while the front bumper beam is connected to the longitudinal beam through the installation of a separate deformation element in between. The installation of deformation element has evolved with time to attain the required safety and termed as “Crash Box”. The purpose of the “Crash Box” is to absorb energy by undergoing plastic deformation and can be replaced easily.

The main resolutions for installing a subsystem of bumper and beam at the front and rear end of the vehicles can be summarized as:

- To absorb the kinetic energy of collision as the unwanted collision happens and make a guided path for the remaining extra energy to the specific designed points in the main frame of the vehicle.

- Reducing the cost of repairing the deformable parts of the vehicle which occurs at low speed so the insurance companies have to bear the minimum penalty

- To provide a defined directed path for the impact load into the main frame of the automobiles by keeping in view that the intrusion of the structural parts is as less as possible into the passenger cabin and the chances of the survival of passenger inside the vehicle are maximized

- To show the compliance of the vehicle to the safety legislative requirements regarding the energy absorbing ability of the system.

1.6 Front Crash Management System:

The front crash management system consists of different components which do perform their balanced functions individually to fulfill the certain needs and working on the purpose for which each of the part is designed.

It generally consisted four to five components which include the energy absorbing elements like crash box in shape of thin walled tube, the main bumper beam in lateral direction, the connecting elements like bolt panel and brackets and possible installation of foam or honeycomb structure in between the bumper beam and the visible plastic fascia.

In case of an impact the crash energy is firstly absorbed by the elastic deformation of the plastic fascia and the foam installed after it. As soon as the impact energy exceeds it is consumed by the plastic deformation of the reinforcement beam which is in fact the bumper beam. If the crash load is still higher then there comes the deformation element (Crash box) which is mainly designed to reduce the impact at the start and then consume excess energy uniformly by plastic deformation and save the main assembly ahead of it from any damage and ensuring the safety of occupants. Only if the energy absorption capability of the crash box is limited and overloaded then the load is transferred to the main longitudinal beams and deformation of these further structural parts start to occur.

1.7 Rear Crash Management System:

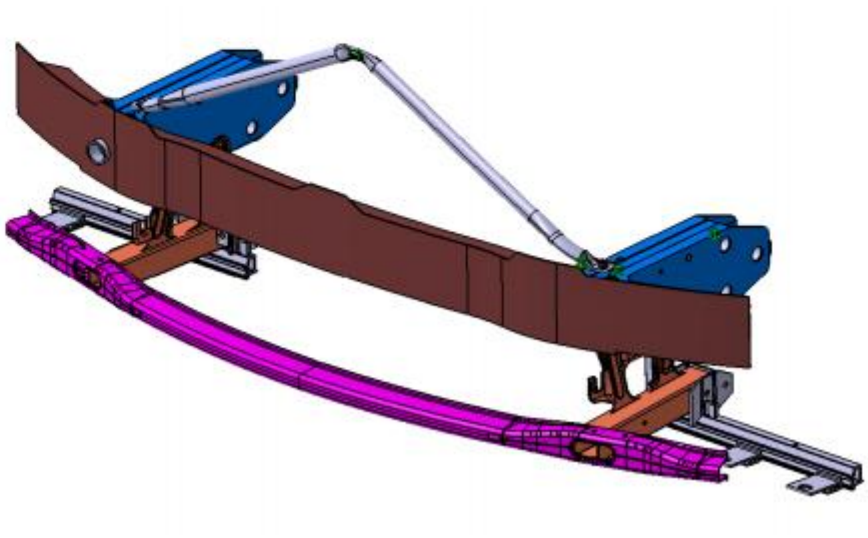
The rear crash management system is almost identical to the front crash management system in older versions of automotive safety. Automobiles were guarded with high metallic beams in lateral axis to the longitudinal beams of main frame of the automobiles. In modern car thin walled beams are attached to the longitudinal beams with the help of fixing plate. There are certain safety requirements are established by the regulatory automotive safety departments to ensure the safety of occupants in case of unwanted collision from the rear side of the vehicles.

1.8 Roll Over Crash Management System:

Roll over crash management system lies in the structural crashworthiness of the main frame constituting elements. The main frame of the vehicle which is also called body in white (BIW) is designed specifically with the channelization of energy which travel through the main frame and avoid any penetration in the passenger compartment in case of roll over scenario.

1.9 Pedestrian Protection:

Making the automobiles safer for the occupants traveling inside the vehicle required some stiffness of the structure to avoid too much deformation of the overall structure but the objective of the crash management system do also require the safety of pedestrians. This protection being incorporated into automobile safety is complex and getting more and more severe. One of the points under consideration is to deploy air bags with the help of integrating some impact sensors which lead to the use of those airbags just prior to the impact. This approach is labeled as the kinetic approach. The static approach towards the pedestrian protection introduces the induction of an additional component which is installed on the front of vehicle as lower bumper stiffener[7].



Front crash management system of the BMW 3 series models
(Source: Constellium)

Figure 6: Lower bumper stiffener for pedestrian protection

The idea behind the installation of this additional component is to introduce a divided load path in the upper and lower channel which would lead to reduce the interference by absorbing the impact energy through multiple energy absorbers and to manage the pedestrian leg impact. The key challenges of this idea being implemented are constrained by weight and vehicle styling. The potential of being low weight of aluminum allow the installation and let the manufacturers to utilize this approach to ensure pedestrian protection.

1.10 Evolutions of bumper design:

There are four basic bumper design principles which have been evolved keeping in view the purpose of its installation in addition to the main structure of the automobiles.

1. Installation of an evident metallic lateral beam at the front and rear end of the vehicle. This beam is subjected to reduce the impact of crash by absorbing the kinetic energy of the colliding object. This has been vastly used practice in the past but the low weight requirement of the vehicle has put it out of the list.

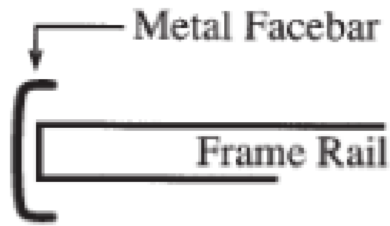


Figure 7: Stage 1 of bumper design

2. For the styling purpose and avoiding heavy structural part, plastic frame is introduced which is followed by the reinforcing beam system which is connected to the front and rear longitudinal beam. This design principle contributes to the overall crashworthiness of the vehicle structure while reducing the performance of the bumper performance.

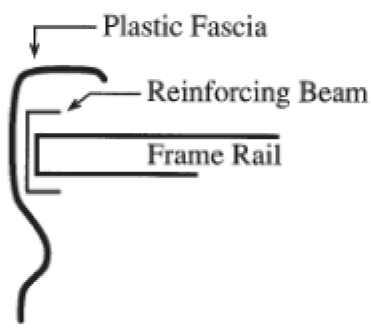


Figure 8: Stage 2 of bumper design

3. The installation of a mechanical energy absorber in connection to the main longitudinal beam in addition to the plastic fascia and reinforcing beam. The mechanical energy absorber can be of different types including a reversible type and deformation element. The reversible type can be a shock absorber which retains to its original shape after the impact while the deformation element (Crash Box) undergoes plastic deformation and can be replaced after a crash.

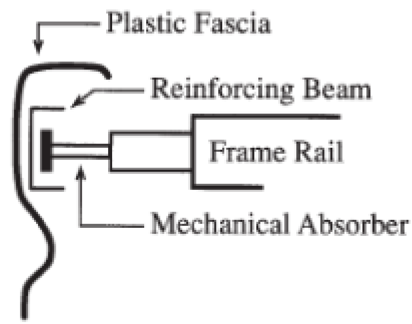


Figure 9: Stage 3 of bumper design

4. Installation of a foam or honeycomb energy absorber as a replacement to the mechanical energy absorber which is a complex subsystem. This is designed to get installed between plastic component and reinforcing beam which is eventually connected to the main longitudinal beam. This design principle provides the best pedestrian protection while compromising on level of energy absorption in case of medium or higher speed collision.

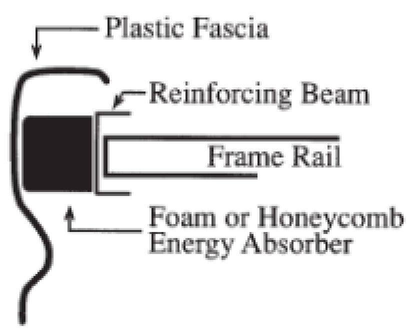


Figure 10: Stage 4 of bumper design

1.11 Design criteria of Crash Management System:

Taking into account the performance parameters of crashworthiness of the system, the design is constrained by multiple factors including reduced weight, smaller size, low cost and fabrication method along with meeting the certain legislative regulations.

There are certain properties of aluminum alloys which best provides the basic requirements of crash management system while being cost effective. Aluminum alloys offers very high stiffness while satisfying the constraint of weight. Structural components of aluminum alloys exhibit remarkable energy absorptions characteristics and being very lightweight naturally.

Aluminum rolled sheets and aluminum extrusion can be best utilized in the manufacturing of bumper beams and attaching components like brackets and fillets.

1.11.1 Working of Crash Management System:

The process of making safer vehicles there are certain considerations which have made the way of inclusion of front and rear crash management system into the structural load path. This has led to a conceptual approach of designing the overall main structural frame which best utilize these crashworthy subsystems at their best potential.

This conceptual approach has set forth to ensure compatibility of designs of main longitudinal beams and other working components of vehicles with the subsystems of vulnerable energy absorption.

The impact energy of the unwanted collision is channelized among the different components of cash management system as described below;

- The reinforcing beam which is designed in a certain able to absorb the collision energy of impact while there is an elastic and as well as plastic deformation

- Thin walled tubes which is also called crash boxes are designed to go under progressive buckling so the maximum of energy is dissipated by plastic deformation

- Regenerative mechanical energy absorbers like shocks and hydraulic systems which absorb the impact energy in a reversible manner and come back to their original shape as the load is removed

- Connecting components in the form of two-sided brackets which may also go under deformation

- Installing a foamed material which is compressed in event of an axial load and absorb the kinetic energy

1.12 Legislative Regulations:

There are different legislative regulations all over the world where a specific test for crash protection is designed which measure the crash energy absorbing capability and acceptable damage is defined for low, medium and high speed levels. So this becomes as one of the major design evaluator and as well as constraint in the designing and manufacturing of automobiles for industrial production.

In addition to this legislative regulation for safety requirements, a very important test is also requested by the insurance companies which require minimizing the repair cost and the cheaper replaceable parts after a crash.

Every legislative regulation authority has their own standardized test parameters depending over the region in which they are necessary to meet. Detail of some of the global level safety regulations authority is given below;

- North America (IIHS Part581)
- Europe (ECE-R42)
- AZT
- Insurance Companies (RCAR)

Keeping in view the broad spectrum of the objectives of a crash management system there are several design solutions came under consideration. The feasibility of these design solutions are evaluated on technical basis and constrained by the level of cost and production at commercial level which is affordable by the general public.

1.13 Deformation element (Crash Box):

Crash box is an important element of crash management system of automobiles which is installed in between the main longitudinal beams of assembly and the bumper beam. The purpose of this element is to absorb kinetic energy by collapsing in case of an unwanted collision. It is designed with appropriate design parameters to protect the expensive parts like radiator and other expensive parts of vehicle by reducing the impact of collision[7].

Figure 11: Crash box installation

1.13.1 Regenerative energy absorbers:

Regenerative energy absorbers are getting attention in recent years with the emergence of electric vehicles. This is a type of shock absorber which converts the kinetic energy and vibrational mechanical energy into useful form of energy. The absorbed energy is converted into other form like electric energy which can be utilized for powering some of the components of the vehicle. In crash management system this concept of energy absorber is utilized and researched to some extent with the use of springs. The advantage of using this kind of mechanism is not replacing the energy absorber in each incident of energy absorbing. The energy absorber returns to its original shape after absorbing the access kinetic energy which is dissipated in compressing the spring and damping. This kind of energy absorber is not cost effective and requires a bit more complex subsystem in order to installing it after the bumper beam.

1.13.2 Non-Regenerative energy absorbers:

Non-regenerative energy absorber is typical thin walled tubes which absorbs energy by going into pre-designed deformation mode. It does not reverse back to its original shape after absorbing access amount of energy. This is the most popular subsystem of modern crash management system and being more studied due to its cost effectiveness and easy to install. There has been a lot of evolutionary designed implemented in combination with bumper beam and separately with the help of fixing brackets.

Non-regenerative crash boxes are widely used of aluminum extrusions and different aluminum alloys are used due to its low weight and high strength. Composites are also being studied for their effectiveness of being low weight and showing high energy7 absorptions characteristics with the reinforcements of fibers.

A typical non-regenerative crash box is shown in the figure below.

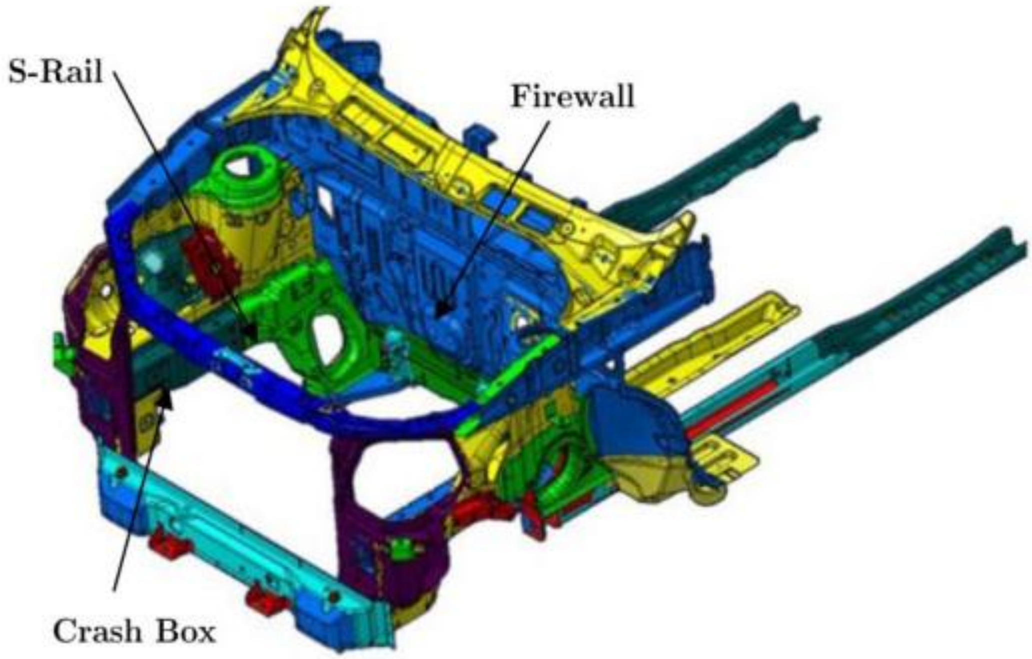


Figure 12: Non-regenerative crash box

2 LITERATURE REVIEW

The energy absorption capabilities of thin walled tubes is best utilized when the low weight is also desired. The study on thin walled tubes concentrates on the deformation pattern while collapsing undergoing an impulsive impact.

Theoretical and experimental procedures were the main techniques in the early stages for engineering design. Experimental methods are quiet expensive in nature and require very highly capable equipment to apply the loading conditions and measure the results. Over the years with the advent of computers, simulations techniques take the place in every engineering branch for the design, analysis and optimization for a wide range of design variables[8]. This era of computer simulations of physical models instead of carrying out the whole experiment is made possible only with finite element method. This has been proved as a very useful and powerful tool to understand the deformation pattern and energy absorption capabilities of thin walled tubes. Applications of computer simulations for the design and development of the best optimized model have reduced the need to manufacture all the options available of prototypes which require a lot of energy and time as well.

Due to very high energy absorption capacity along with having a very low mass, thin walled tubes have been extensively studied in automotive industry, navy products and maritime development like ship building and aerospace industry. Now a days components based on this energy absorbing principle is being widely used in trains, cars, nuclear reactors and various other industrial products where access and unwanted energy needs to be absorbed in a controllable manner.

Analytical solution for the axial compression of thin walled columns was studied a few decades ago by Alexander. An approximate theory was presented for the axial compression of thin walled tubes which take into account the thickness of shell, diameter and collapse load[9]. The finding of the analytical solution does not consider the phenomenon of superimposed axial stress on yield criterion and does not have any effect of deformation mode as well as equilibrium conditions. The following relationship was developed.

$$P = Ct^{1.5}\sqrt{D}$$

Where P is the crushing load when the collapsing starts, t is thickness of the shell and D is the diameter.

One of the earliest studies on the energy absorption of thin walled columns was carried out by Abramovics and Jones. They presented a theory for the axial crushing of thin walled tubes. According to this theory two third of the plastic energy is dissipated through inextensional deformations and the residual one third of the energy is dissipated through extensional deformations[10]. They also showed that the mean crushing force depends on the thickness of the shell.

The crushing process of PVC cylindrical panels was carried out to study the mechanism of formation of folds under axial compression loads[11]. In this study the crushing process is formulated as a problem of isometric transformation of surfaces. It has been concluded that the relative positions of hinges at the shell surfaces must be pre-determined with the executions of experiments in order to identify the buckling modes. It is studied that process of formation of progressive buckling modes generates in a proportional way in the elastic range and after that a whole wave is formed across the length of the shell.

An experimental study on the energy absorption capacity of thin walled tubes made of mild steel, glass reinforced composites and Kevlar reinforced composites were carried out by Hamaouda. Axial crushing of the thin walled tubes under quasi-static loading conditions was carried out experimentally by selecting the square cross section. Effect of thickness and length of the specimen has considerable effect on the energy absorption of thin walled tubes. In case of using composites, the length of reinforcing fibers do affect the crushing behavior of the tubes[12].

Energy absorption capacity of square thin walled columns with introducing buckling initiators was carried out to minimize the initial peak force. Initial collapse force which caused the thin walled column to start buckling is of vital importance because it has considerable affect in the application of thin walled tube. It has been studied that the introduction of triggering mechanism in thin walled columns let the reduction of initial peak load but the decreases the overall amount of energy absorbed by the column,. The overall energy absorbed by the thin walled tube reduces with the buckling imitators as it lead towards the more deflected length of the column. This leads towards the consequence that in case of more heavy impact load the specimen would be bottomed out[13].

Energy absorption capacity of square thin walled columns with buckling initiators under oblique impact loading conditions has been also studied. There is a remarkable reduction in the peak load which causes the deformation to start. It has been found that the buckling initiators tend to alter the deformation mode form global buckling to progressive buckling[14]. This study showed that there is better trend in crush force efficiency with introducing buckling initiators at the point where the buckling initiates in case of plane thin walled column.

Axial crushing of thin walled columns having circular cross section studied with a triggering mechanism near the impact end. There is a reduction in the peak load was highlighted under a series of quasi-static analysis. It has been observed that there is no change in the overall geometric stiffness of the thin walled columns. The results of the experiments showed that the buckling initiators changed the large progressive deformation mode to mixed or diamond mode deformation[8]. It is found that the peak load can be reduced by 30% with the help of triggering mechanism. The study showed a good agreement between the theoretical calculation and the results of experiments.

One of the earliest studies which include the experimental analysis of crash behavior of the thin walled columns under oblique loading was carried out for aluminum alloy for three different angles. Aluminum alloy AA6060 was subjected to quasi-static oblique loading conditions. In this study factorial analysis was carried out to study the mean crush load. The parameters such as length of the specimen, angle of applied load, thickness of the walls, temper by heat treatment of the alloy and velocity at which the impact occurs are selected as design parameters and varied to study the effect[15].

The numerical analysis of windowed and multi cell square tubes for the calculation the effect on energy absorption and peak load was carried out using ABAQUS explicit solver[16]. In this study the overall weight is kept same for comparative study of the thin walled columns. Loading condition selected for the analysis were axial and oblique as well. The study showed that the multi cell tube had higher mean crushing force as compared to windowed thin walled tube. Keeping in view the initial peak load, the windowed thin walled tube had a lower initial peak load as compared to the multi cell tube. It has been also observed that with the increase of loading angle the effectiveness of both altered tubes decreased and if the deformation mode changes from progressive buckling to global bending then the conventional thin walled has much more better results in comparison of these altered geometries.

Axisymmetric thin-walled square tubes were analyzed to observe the effect on energy absorption with straight and tapered geometries and having two different cross sections. Numerical analysis of the thin walled tubes having single cell and multi-cell cross section was carried out in LS DYNA. The study of four different types of thin walled tubes under oblique loading condition was critically analyzed using peak force (PF) and specific energy absorption (SEA) as two main crashworthiness parameters. It has been observed that the multi-cell thin walled tube has the best results under oblique loading conditions in terms of selected crashworthiness parameters. In addition to this multi-objective optimization design by using multi-objective particle swarm optimization (MOPSO) algorithm technique was used for the multi-cell thin walled tube to achieve the maximum of crashworthiness parameters by using the dimensions of the cells, length and width of the thin walled columns[17].

Crashworthiness optimization study of foam filled thin walled columns having square cross section under oblique loading conditions was studied with variation in load angle, geometry and material properties[18]. In this study PAM-CRASH was used to find the numerical solution. Dynamic finite element analysis (FEA) was carried out after validating by theoretical solutions and existing experimental literature. Specific energy absorption (SEA) and peak force (PF) as the selected crashworthiness parameters were analyzed for empty and foam filled tubes. The analysis was carried out for both the axial and oblique loading conditions. It has been observed that the optimal design vary under different load angles for either empty or foam-filled columns. Under pure axial loading condition, it is found that the foam filled square thin walled tube may have better crashworthiness as compared to empty thin walled columns, but a different approach must be used to enhance the crashworthiness of empty thin walled tubes under oblique loading conditions.

A numerical investigation is carried out to study the displacement of center of gravity (COG) of various polygonal cross sectional structures under axial impact loading[19]. The finite element code used in this study is LS DYNA and a subroutine code is developed to calculate the center of gravity of thin walled tube during the crash event and after total deformation. It is studied that an even number of polygonal edges causes a more symmetric displacement of the center of gravity. The symmetric deformation is more prominent with the increase of polygonal edges and as the initial wall thickness decreases. It is concluded the mass moment of inertia of thin walled in lateral axis can be neglected as compared to the mass moment of inertia of thin walled tube in the direction of axial compression.

Thin walled tubes having Cee-shaped cross sectional are studied find out the center of gravity under axial crushing by using the finite element code LS DYNA. The effect of wall thickness on the displacement of center of gravity is investigated. It has been observed that the effect of opening angle of Cee has an increasing trend as the wall thickness of the structure decreases and the with increase of wall thickness the displacement of center of gravity in the direction of crush almost stabilize for all the opening angle of Cee shaped thin walled tubes in the range of 10 to 90 degrees[20].

Corrugated thin walled tubes are also investigated experimentally and numerically and their energy absorption characteristics are compared to the flat thin walled tubes. Various designs of corrugated tubes are examined by conducting finite element analysis using LS DYNA[21]. Initially axial crushing behavior of the corrugated thin walled tube is studied theoretically by using the super folding element theory and then the numerically investigating is conducted. The selected material of the thin walled tube was aluminum alloy AA6060 having temper T4 and the selected crashworthiness parameters are initial peak force and crush force efficiency. It has been concluded that the corrugated thin walled tubes have advantages as compared to the flat thin walled tube in terms of lower initial collapse force and do also have lower crush force fluctuation frequency.

Thin walled tubes of composites of cotton fiber are studied for energy absorption characteristics using the finite element code Altair Hyperworks of RADIOSS. In this study effect of radial corrugation on tubes of selected composites are analyzed under axial impact loading. The collapse procedures are simulated after the validation of finite element model. Introducing radial corrugation along a shell element generator enhance the crashworthiness performance of energy absorbing composites tube. It is concluded that the as the number of radial corrugation increases the amount of absorbed energy also increases significantly[22].

Axial deformation behavior of open and end capped cylindrical tubes is studied at quasi static and impact loading It has been found that the initial peak force can be reduced by 15 to 30% by using end capped cylindrical tubes as compare to open ended tubes while the energy absorption capacity is not compromised[23]. Cylindrical tubes were manufactured using rolled sheets of aluminum alloy and their properties were determined by performing uni-axial tensile test.

Forming data is incorporated in the numerical software for better simulation results and four stage deep drawing process is carried out for this purpose. The properties and dimensions of the tools required is carried out in modeling software CATIA and all the pre-processing is carried out in HyperForm and the finite element analysis is carried out in LS DYNA.

Quasi-static analysis of thin walled walled tubes for nine different geometric shaped with the installation of polyurethane closed cell-foam inside them was carried out to study the crashworthiness of the tubes. Two different thicknesses are introduced inside the thin walled tubes and the analysis is carried out experimentally. It has been concluded that presence of polyurethane foam has reduced the delamination process and fracturing of fibers which in result has reduced the energy absorption and higher peak loads are calculated[24]. Although the crushing phenomenon is more stable for the square and hexagonal tubes with the presence of the foam but other crashworthiness parameters are not much desirable.

Conical tubes have also been studied for the energy absorption purposes. Dynamic computer simulation procedures are carried out for the empty and foam filled conical tubes and it has been observed that foam filled conical tubes are better designs for the purpose of energy absorption[25]. This study was carried out to simulate the response of conical tubes under oblique impact loading and energy absorption due to progressive deformation and bending was critically analyzed. The primary variables in this analysis were the initial wall thickness, load angle and semi apical angle which were studied in depth for their effect on energy absorption characteristics. The particular aspect of interest of these variables in oblique loading condition was to find out the transition of progressive buckling to the global bending for this specific tube geometry. In experimental analysis the foam used was composed of aluminum with a density of 40kg/m^3 and an angle of 10 degrees was kept to validate the finite element model. This simple experimental setup was just carried out for the validation purposes. This validated finite element model was then further utilized for the study of different geometries and their parametric study was made possible under numerically approximated solutions. Computer modeling was conducted finite element code LS DYNA 971 and isotropic plasticity model and crushable foam material models were used for the modeling of conical tubes and the foam filler respectively Foam filled tubes are found to be more effective for energy absorption as they can withstand oblique and axial impact with a minimal reduction in energy absorption It has also been observed that the tubes filled with foam and having semi apical angle of 5 degrees can withstand impact more efficiently as compared to the empty conical tubes.

There has been a lot of research carried out which primarily focus on enhancing the material properties which eventually lead to the better energy absorption of the thin walled columns. On the other hand geometries of thin walled columns are also studied in wide range which includes the use of multi-cell and different geometric patterns. Different loading conditions produce different deformation modes in the thin walled columns which is also a broad spectrum of study in the field of thin walled tubes.

In the present study thin walled tubes of different cross sections are analyzed by numerical investigation using the explicit dynamic solver LS DYNA. The FEM model being applied to all the cases of different geometries is validated against the existing experimental literature to ensure the accuracy of the results. The verification of finite element is also carried out to making the results independent of the mesh sizes and other variables in the element formulation.

3 THEORETICAL ANALYSIS

3.1 Mechanical Properties

The most important phenomenon is the deformation of the thin walled columns which can be elastic and plastic in nature to absorb energy. In this section the basic properties of materials are discussed to better understand the energy absorption.

Elastic deformation:

Elastic deformation is a temporary deformation of a material. It is defined as the shape change of a material under very low stress which is reversible. Energy absorbed in elastic deformation is recovered as the load is removed.

Considering an isotropic material and calculating the strain in x-direction.

$$\varepsilon_x = \frac{\sigma_x}{E}$$

Here ε_x is the strain in x-direction and σ_x is the stress in the same direction. E is the Young's modulus for the material. This relationship is known as Hooke's law.

This uniaxial load causes deformation in other two direction also which is described by the following relationship.

$$\varepsilon_y = \varepsilon_z = -v\varepsilon_x$$

Here v is the Poisson's ratio for the material. The above two equations are combined together to obtain general Hooke's law which take into account the Poisson shrinkage:

In terms of shear stress and torsional stress the elastic behavior is defined by the following relation.

$$\tau = G\gamma$$

Here τ is the shear stress and γ is the shear strain. G is the shear modulus of the material.

Elastic deformation is proportional to the applied load up to the point of yield in the stress strain diagram[26].

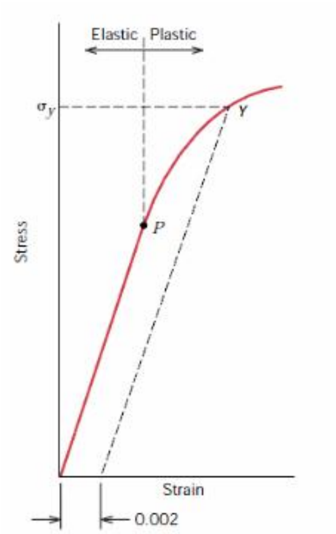


Figure 13: Stress strain diagram

The material is elastic as load is applied and reaches the value at point P and beyond that point the plastic deformation begins to take place. It is not so much obvious to find the exact point of transition from elastic to plastic deformation. For this obstacle to overcome a series of experimentation led to the definition of a point Y on curve at which the yield phenomenon for the most of the materials occur. This point is obtained by drawing a parallel line to the elastic curve which originates by introducing an offset on the abscissa of 0.002. The intersection of these two lines defines the yield point of the material. This relationship does not account for the nonlinearities of the material and in case of complex loading this cannot be clearly obtained.

3.1.1 Plastic deformation:

In crash analysis there is a complexity of the load being applied as it is impulsive in nature. This complexity of load contains much nonlinearity in it which makes the elastic theory no more useful. Every software uses plasticity theory to predict the structural response of the material.

To understand the plasticity theory it is necessary to know the true stress strain curve which is different than the engineering stress strain curve on which the elasticity theory of material relies.

Engineering stress strain curve consider the initial dimensions of the specimen. It does not account for the change in dimensions of the specimen as the load applies and causes a change in length.

The following relationships are the basics of engineering stress strain curve.

$$\varepsilon_{\mathrm{eng}} = \frac{\Delta \mathrm{L}}{\mathrm{L}_0}$$

$$\sigma_{eng} = \frac{F}{A_0}$$

Where L_0 and $A_0$is the initial length and area respectively.

The true stress strain curve considers the change in length and area of the specimen at each instant. True stress and strain is presented by the following relationship.

$$\varepsilon_t = \ln\frac{L}{L_0} = \ln(\varepsilon_{eng} + 1)$$

$$\sigma_t = \frac{F}{A} = \sigma_{eng}(\varepsilon_{eng} + 1)$$

In the above relations change in load and area at each is considered which results in plotting the true stress strain curve. There is a relationship between engineering and true stress curve for both stress and strain[26].

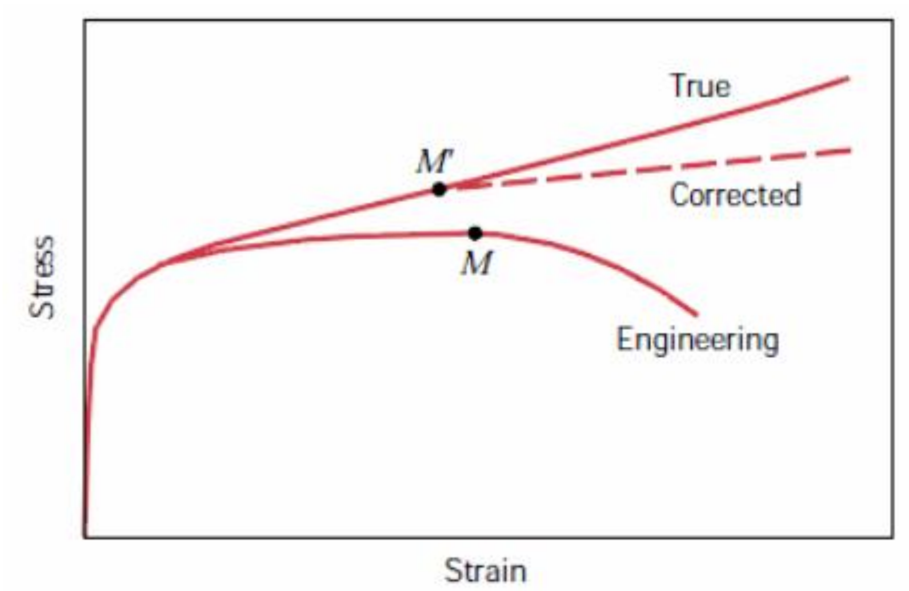


Figure 14: Engineering and true stress-strain curves

In the above diagram the curve beyond the M point represent the corrected curve which considers the necking phenomenon. This true stress strain curve is necessary when using any numerical software. When material model is imported into the software it utilizes the true stress strain relation because the deformation is calculated at very small time steps and changed length and load is taken into account rather than the initial dimensions of the specimen.

True stress strain curve of a material is also called flow curve for the material. A mathematical relation for the plastic deformation is developed which is called Hollomon law which is applicable from the start of plastic deformation up to the point of maximum load which causes necking in the specimen.

$$\sigma = K\varepsilon^{n}$$

In the above relation K is the stress value at unit deformation and n is the strain hardening coefficient. These two values are different for every alloy.

There are some necessary simplifications are made to obtain the true stress strain curve. These simplifications are made by assuming that there is not any initial elastic deformation and Bauschinger effect is also ignored [27]. It is assumed that the behavior of material is perfectly rigid below the yield stress as there is not any elastic deformation and it is considered that the in plastic region the recorded stress value would be less in any case than the initial yield stress. This is also defined as the perfectly rigid-plastic material.

Plasticity theory relies of yield criterion which incorporates all the stress states which lead to the plastic deformation. Yield criterion is defined as[27]

$$f\left(\sigma_x, \sigma_y, \sigma_z, \tau_{xy}, \tau_{yz}, \tau_{zx}\right) = C$$

$$f(\sigma_1, \sigma_2, \sigma_3) = C$$

The second equation is for the isotropic material and principal stresses are taken into account. C is the material constant.

The yield surface in the stress plane is shown in below figure (Figure 3). It is clear that the plastic flow occur only when “f =0”. For all other values of “f” the material is in elastic deformation.

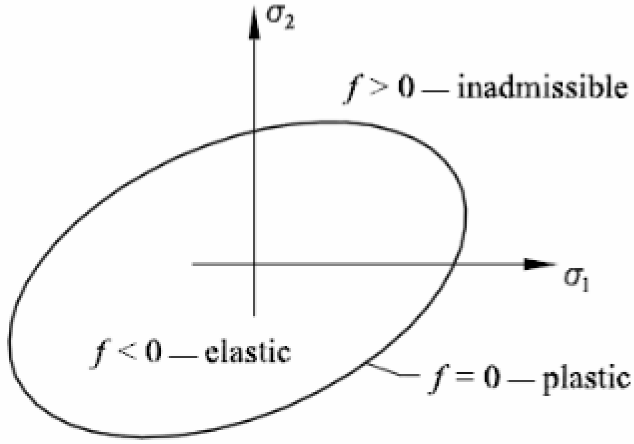


Figure 15: Yield surface

3.2 Crashworthiness parameters:

To assess the quality of the thin walled tubes in term of crashworthiness there are several parameters which must be analyzed critically to get the desired behavior of the specimen. One of the important set of parameters is derived from the force-deflection graph. Area under the force-deflection curve is the energy absorbed by the thin walled tube.

3.2.1 Energy Absorbed:

The total work done during the axial crushing of the thin walled tubes is equal to the area under the force-deflection curve.

$$W = \int P \, d_s$$

Where, P is crushing force which is making the thin walled tube to deform. Integration of this force over the crushed length of the thin walled tube gives the total work done by the impact load.

Total work done by the crushing force is equal to the energy absorbed by the thin walled tube. In case of progressive buckling which is the most desirable deformation mode in the study of thin walled tube, almost all of the input kinetic energy is absorbed by the thin walled column. In order to absorb all the input kinetic energy the length of the thin walled tube must be large enough so it must not be bottomed out.

3.2.2 Specific Energy absorption:

One of the more important crashworthiness parameter than energy absorbed is the specific energy absorption. It is calculated by dividing the total energy absorbed by the total mass of the energy absorbing element.

$$\mathrm{E} = \frac{\mathrm{W}}{\mathrm{m}}$$

Higher values of specific energy absorption indicate the high capability of thin walled tube to absorb kinetic energy.

3.2.3 Peak Force(PF):

Peak force is the initial collapse before after the impact which causes the thin walled tube to start buckling. In designing a thin walled tube for crash energy absorption it is desired to keep the initial collapse force as low as possible. This is necessary to reduce the severity of impact as it would lead to high deceleration of occupants inside the passenger cabin.

3.2.4 Mean crushing force (MCF):

Mean crushing force is obtained by dividing the total absorbed energy by the maximum deformation of the thin walled tube.

$$MCF = \frac{E_a}{\delta}$$

Where $E_a$is the total absorbed energy and δ is the maximum deformed length of the specimen.

4 OBJECTIVES AND METHODOLGY

4.1 Objectives:

The main objective of this study is to analyze the thin walled tubes of different cross section using numerical approximate solutions. Finite element model is validated against experimental literature. The validated finite element model of the thin walled tubes is applied to different geometries and base on specific crashworthiness parameters their energy absorption capacity is critically analyzed.

4.2 Numerical Methodology:

The design and numerical analysis of the thin walled columns is divided into four stages. At the very first stage the thin walled tubes are designed in CAD software. After this the CAD models are imported to the numerical software for pre-processing which includes the discretization of the geometry for application of FEM analysis. This second stage also includes the application of initial and boundary condition.

The third stage is to import date to the numerical software for the solving of the mathematical model. In this study explicit solver LS DYNA is used for the numerical solution.

The last stage is visualization of the equations solved by importing the data to post-processor interface. Simulation results for the deformations are obtained in post-processing. For plotting the results the tabular data is imported to excel and values for different cases are plotted on the same graph to get the better understanding of the results.

4.2.1 Initial design:

Computer aided design (CAD) models of the thin walled tubes are carried out in Solidworks.

4.2.2 Pre-processing:

CAD models of thin walled tubes are imported into ANSYS Design Modeler for further pre-processing. This stage includes the application of initial conditions and boundary conditions necessary to model the crash event.

Using the ANSYS Design Modeler, the geometry is treated with necessary processing so it can be further meshed with desired elements. Crash box which is the primary subject in the study is converted into surfaces so it can be discretized into finite elements using shell elements. The essence of using the shell elements is described in the chapter of finite element studies.

4.2.2.1 Meshing:

Meshing is carried out in ANSYS Workbench by applying desired mesh controls. Working in this GUI interface the mesh size for different geometrical aspects is much simple and more visible. Quadrilateral four node shell is used to mesh the energy absorber and triangular elements are avoided. Rigid bodies are modeled by rigid brick elements which are hexahedral having eight nodes.

4.2.2.2 Boundary conditions:

Applying the right boundary conditions is the backbone of the finite element analysis. After the meshing of the model boundary conditions are applied to accurately describe the physical phenomenon of the crash scenario.

Rigid base plate is placed on one end of the thin walled tubes and fixed support is applied at this end. The fixed support is applied to constraint the degree of freedom of elements in all directions.

On the striking end of the thin walled tube a rigid plate is placed with a minimal distance so the initial impact force is measured without any numerical noise. A defined point mass is placed on the striking plate to compensate the low weight of plate.

4.2.2.3 Contact properties:

To get accurate results its mandatory to have precise contact modeling which described the contact properties of physical problem to be studied.

The contact between the impactor and the thin walled tube is modeled as “Nodes to Surface algorithm” as shown in the figure below. This contact region is made frictional with a friction coefficient of 0.3.

The contact between the thin walled tube and base plate is defined as simply bonded. This contact is not supposed to go under any changes during the impact as this end of the thin walled tube is constrained in every direction by apply a fixed support at the base plate. The contact is shown in the figure below.

When there is a contact happening between two surfaces with each other there is a possibility of nodes penetration of elements into other. Progressive deformation of the thin walled tubes is possible only with the formation of lobes so there is a need to model the self-contact of surfaces coming into contact with each other for the thin walled tube. LS DYNA provides a contact model for this scenario which is precisely designed for crash simulation. The prescribed contact model is “AUTOMATIC_SINGLE_SURFACE” contact modeling for the thin walled tube. This contact is made frictionless to avoid interpenetration of surfaces into each other. The software introduces a penalty method for taking into account this phenomenon of interpenetration. This penalty method works by introducing linear springs between every node that penetrates the surface and protects the master surface[26].

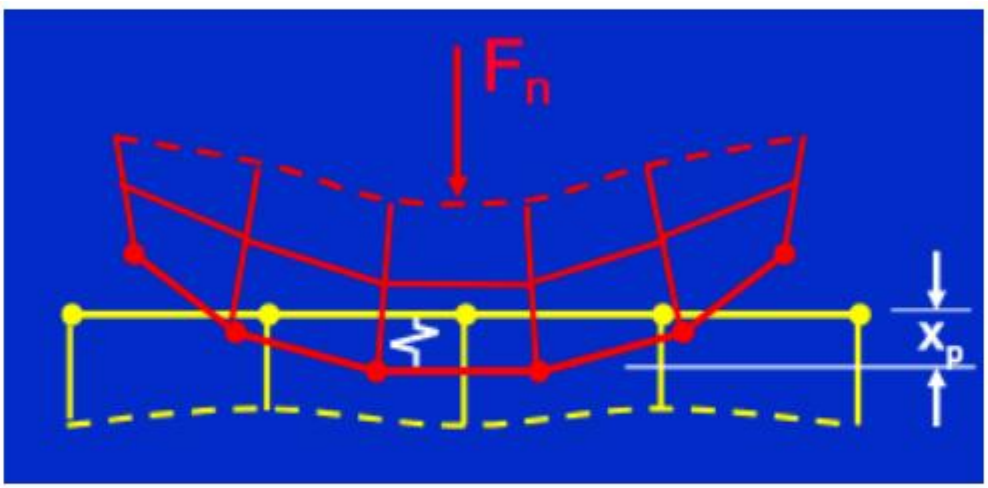


Figure 16: Penalty method

This method works by the introduction of a contact force which enforce a equilibrium by pushing back the node to avoid mingling into the other surface element. The force can be calculated as;

$$F_c = K_c x_p$$

In the above equation K_c is the contact or penalty stiffness and x_p is for the displacement of the penetrated node. To avoid instabilities the magnitude of the force must not be as much as needed to push the back the element to avoid penetration. But this has to be under consideration that the higher values of this force would cause a separation of the surfaces which would also lead towards instability and wrong results of simulation would come out.

There is no need to specify the target and base faces or surfaces while defining this contact in LS DYNA. The software automatically apply this contact algorithm when the two surfaces of thin walled tube would come into contact with each other.

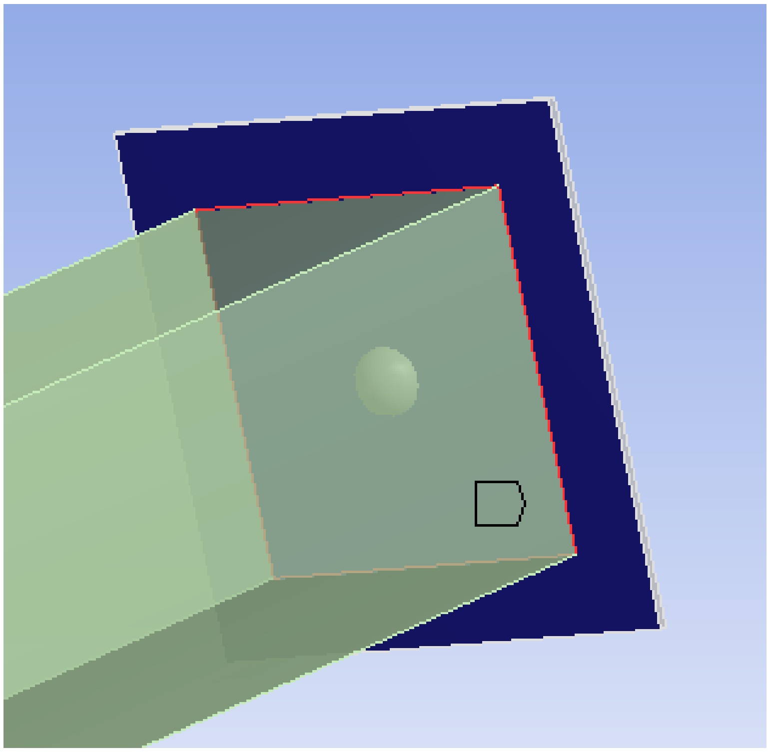


Figure 17: Contact region between impactor and energy absorber

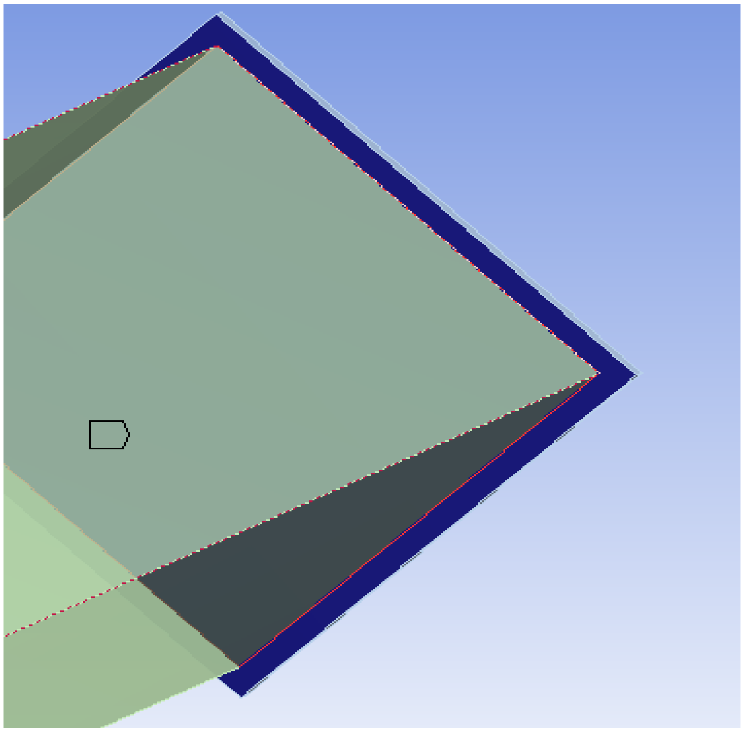


Figure 18: Contact region between energy absorber and base plate

4.2.3 Solver:

Explicit dynamics LS DYNA is used to explicitly solve the partial differential equations which describes the entire model. This is done by the solver by converting the differential equations into algebraic equations and then numerically finding a solution for the governing equations.

4.2.4 Post-processing:

From the simulation results force and deflection values with respect to time are obtained. Force is calculated at the impactor end and the deflection is calculated of the node at the upper corner of the energy absorber. Plots of these variables are available in ANSYS Workbench with time on horizontal axis. So for further processing to obtain force-deflection curve the tabular data is exported to excel. With original tabular data force-deflection curve is plotted and desired crashworthiness parameters are calculated.

5 FINITE ELEMENT STUDIES

5.1 Numerical devices:

There are several computer softwares available for the numerical analysis of a discretized FEM model of a physical model. It includes RADIOSS, ABAQUS, NASTRAN and LS DYNA. All of these solvers produce accurate results depending on the area of application and type of analysis.

The numerical solver being used in this study is LS DYNA. It is developed by Livermore Software Technology Corporation (LSTC) in 1976 for the simulation of blast of nuclear bomb. It became the public domain software in 1978 and now is being widely used for automotive and aerospace industries. Early releases of this solver provided a separate interface for pre-processing and post-processing (LSprepost) of the numerically solved equations which has been widely used in numerical analysis research studies in various engineering products. A few year ago this solver has been released with mutual agreement with ANSYS and provided with the workbench interface. Now all the GUI capabilities can be utilized for pre-processing and post-processing of the results while using a this explicit dynamic solver.

CAD models can be imported with different file formats and further geometry cleanups for better meshing are easy to perform. For meshing there are different softwares available like HYPERMESH and MeshPro but in the present study models are meshed using the standard mesh capabilities of ANSYS software.

5.2 FEM method:

To solve the complex problem in the fields of stress analysis, heat transfer, fluid flow and many other engineering applications finite element method was first introduced in 1960. Analytical solutions are only available to basic engineering models which are not applicable where a compound of different geometrical aspects and nonlinearities in the governing equations of the models are present.

Finite element method technique is the foundation of simulation softwares for assessing the design. This method is used on the geometry designed on a CAD software which considers all the surfaces continuous. This continuous geometry is then discretized which is dividing the main domain into simple elements[26]. Differential equations are easy to calculate for these simple elements and all the equations are assembled to obtain a system of equations which describes the complex model for which no analytical solution can be calculated.

5.3 Implicit vs. explicit method:

In FEM analysis nodal displacement is the primary value calculated at each of the finite element. This nodal displacement is used to calculate the strain of the element from which the stress is calculated at each node.

Finite elements solver use iterative integration process to find a converged numerical solution. Every CAE software use two different approaches to find the numerical solution for approximation of the partial differential equation. These two different approaches are implicit and explicit.

Explicit method and implicit method differ only by the enforcement of equilibrium at each time increment of the problem under consideration.

For example, the governing equation for structural crashworthiness is as;

$$[M]\{y''\} + [C]\{y'\} + [K]\{y\} = \{f\}$$

Implicit method calculates the values of unknown $\{y\}$ by inversion of stiffness matrix. In this method the values which are to be calculated are expressed in terms of other parameters which are also unknown at the beginning of this time increment. So an iterative process in performed to calculate the values of these unknown variables. In implicit method after each time increment the analysis also perform Newton-Raphson iteration process to establish equilibrium of the internally calculated forces to the externally applied displacement steps which is in fact the incremental load being applied. This enforced equilibrium at each time step requires the inversion of stiffness matrix through iterative process which leads towards the high computational cost in terms of more computational time, memory and disk space. In addition to this procedure if there arises any non-linearity then it takes more iterative process to account for that in terms of accuracy. Backward and central difference methods are used in implicit method and no stability condition is required.

Explicit method used to calculate the state of a system a later stage from the solution of the state of the system at the current stage. In this method the values of acceleration $\{y''\}$ are calculated at each time increment. In explicit analysis the stiffness matrix is updated at each step based on changes in material and geometry. This results into the generation of a new stiffness matrix at each time increment and again the procedure of applying the load at this increment is repeated. This method does not require iterative processes for the inversion of stiffness matrix as there is no need to enforce equilibrium condition at each time increment. This eventually reduces the computational cost in terms of computation time, disk memory and space. The other difference due to this approach lies in the conditional stability of the numerical process. Grid sizes are not allowed to be chosen freely in explicit analysis as we have to ensure the stability of the solution.

The stability of explicit dynamics is considered by the equation below;

$$\Delta t_{stable} = \frac{L_{element}}{v_{wave}}$$

Where $L_{element}$the smallest length of the element in the mesh is generated and v_{wave} is the speed of sound for the material being used.

For any material v_{wave} is calculated by using elasticity modulus E and density of the material. Given as;

$$v_{wave} = \sqrt{\frac{E}{\rho}}$$

From the above equations the limit imposed on the time increment which is summarized as Courant stability is the limit that the time increment must be smaller as compared to the time taken by the affective wave which is travelling through the material to travel through the element size.

Implicit method used to calculate the displacement at each node of the element and need the solution to be converged at each time increment. This is done by applying global equilibrium at each time step. While in explicit dynamics there is no need to establish this extra calculation at each step so saving computational time and cost.

5.4 Element formulation:

There are different types of element formulation available for the elements being used for meshing of the model. Element formulation is selected by analyzing the application of the problem.

In our study of crash analysis we have used thin shell elements to model the energy absorber for numerical analysis. The best element formulation and algorithm for the thin shell elements which are impulsively loaded for the very short time resulting in nonlinear response of material and undergoing large deflection is studied by Belytschko and Tsay[28]. This element formulation is widely applied by the researchers in numerical analysis of thin walled structure at comparatively low speeds due to its capabilities of using reduced integration scheme and accuracy in predicting material response in large deformations. This element formulation is available in LS DYNA library and used for the thin shell elements.

5.5 Hourglass control:

Hourglass is commonly known phenomenon in finite element analysis. It is zero energy mode of the elements. Elements are distorted in such a way that it contains zero strain at each node of the element which results in no stress values at that node.

There are various hourglass control options are available in LS DYNA. They can be categorized as stiffness based and viscous based hourglass control. Thin shell elements with reduced integration points need no hourglass control but for more than 2 integration points hourglass control is recommended. The viscous based hourglass control are useful for high velocity impacts in structural parts but for impact lying in low velocity mode stiffness based hourglass control is suggested[29]. In this study the stiffness base hourglass control namely Type 4 in LS DYNA is applied to the geometry meshed with thin shell elements with an hourglass coefficient of 0.3.

6 FINTE ELEMENT MODEL VALIDATION AND VERIFICATION

Finite element model needs to be validated against some experimental analysis so its accuracy and convergence to right solution can be identified. Validation of the FE model can be defined as the solving the right equations which are mathematically describing your model.
Verification of the FE model is a check to the mesh sensitivity of the model. This is a core need of the numerical approximate solution of a physical problem to make the it independent of the refinement of discretization of the complex problem into small elements. Verification of the finite element model can be described as a proof of solving the equation in the right way which describes the physical model when assembled together.

6.1 Base model:

The base model of thin walled tube is of square cross section with dimension of 80mm×80mm. The length of the tube is 310mm. The thickness of the wall is kept 2.5mm.

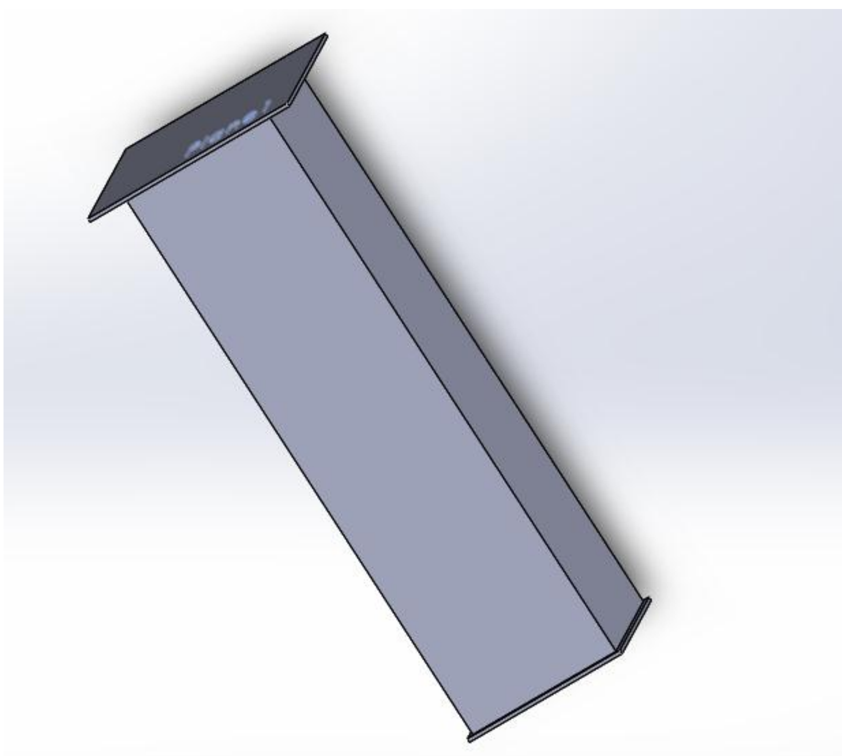


Figure 19: Base model

The dimensions of the base model are kept same as of the model used in experiments. In experimental setup a projectile with an initial velocity of 17.7m/s is impacted on the thin walled tube. For this purpose the impactor plate is not made connected with the thin walled tube in the initial CAD model of the physical problem. There is an offset of 0.05mm between the impactor plate and the thin walled tube.

This CAD model is imported in ANSYS Design Modeler and the walls of the thin walled tube is converted into surfaces by applying “midsurfacing” tool. This step is carried out in Design Modeler as it keep record of the thickness of the wall with it and let the mesh controls to be applied for meshing with thin shell elements.

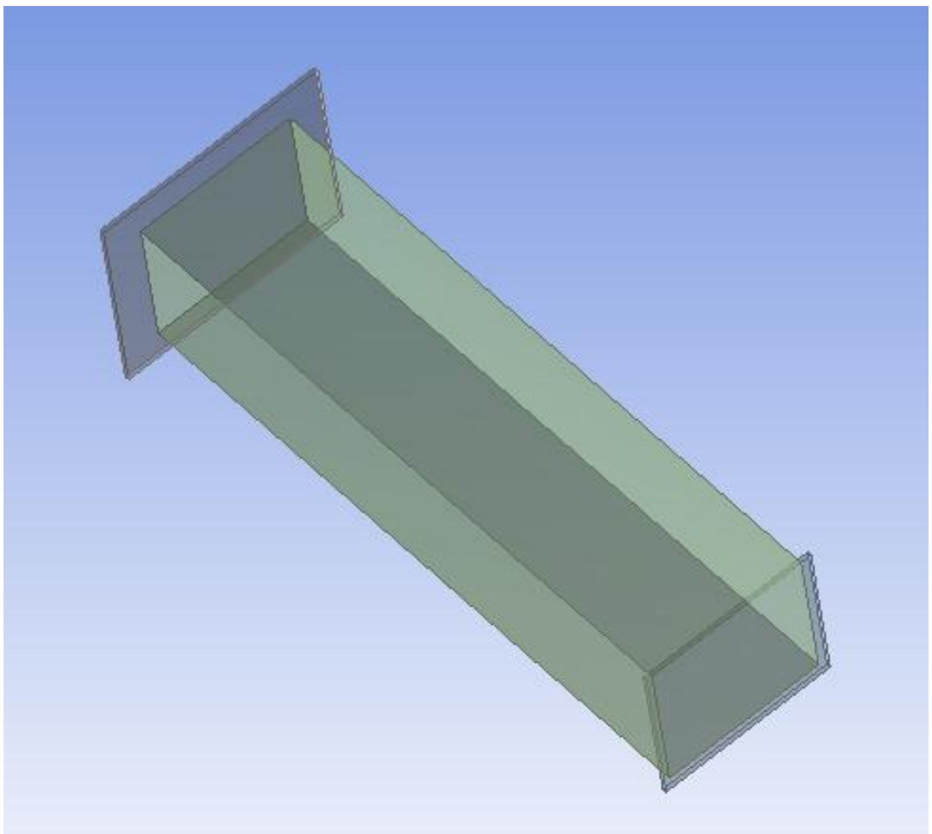


Figure 20: Converted to surfaces

Converted surfaces are meshed with thin shell elements.

The meshing of imapctor and base plate is carried out with rigid brick elements and element size is kept 8.0mm. Further refinement of mesh is not applied to these both parts as there is no deformation is going to occur on these two parts. The main parameter to study on the impactor’s striking face is the crushing force which needs to be calculated.

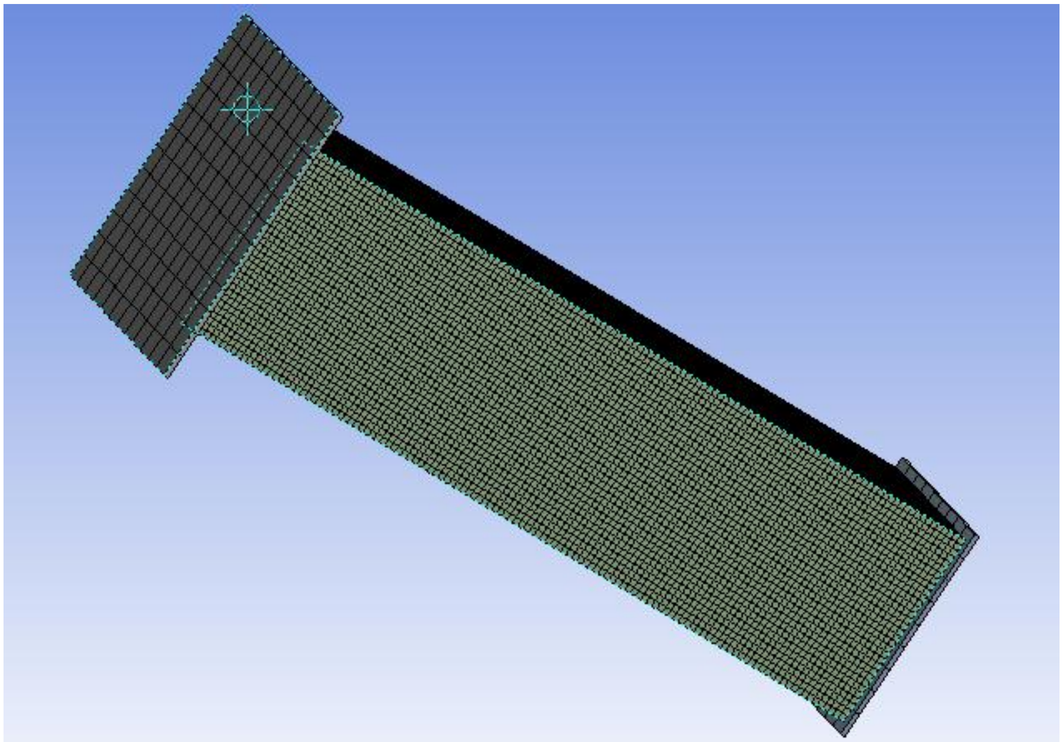


Figure 21: Meshed base model

Application of the boundary and initial conditions include the velocity of striker plate in the impacting direction and constraining the base plate in all degrees of freedom.

As there is earth gravity cannot be avoided so the standard earth gravity is applied to all the bodies in the model.

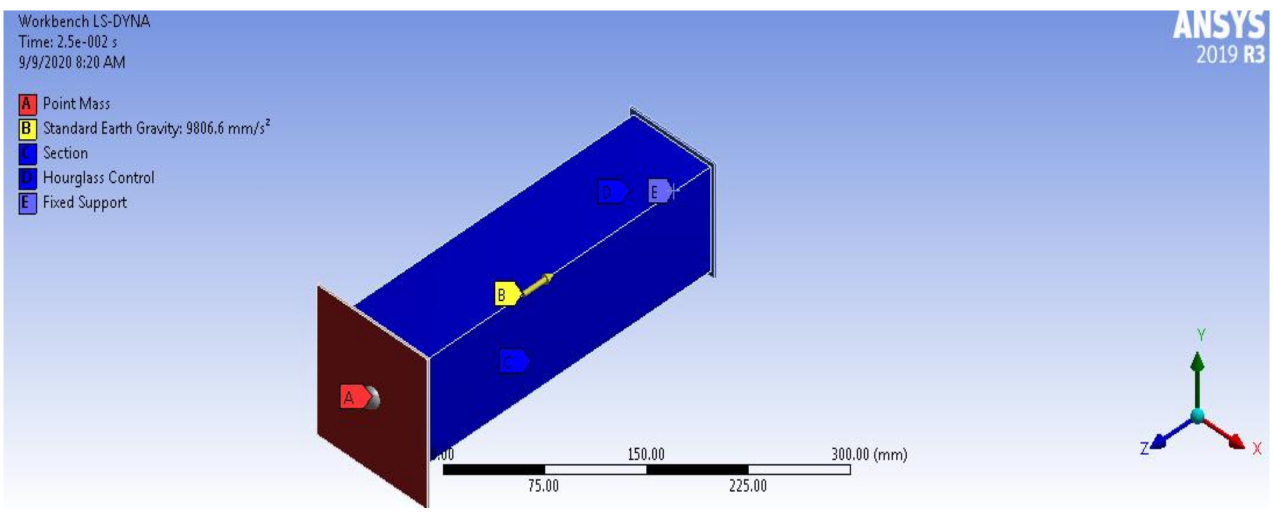


Figure 22: Application of Initial and Boundary conditions

6.2 Verification of FEM:

The sensitivity analysis of the finite element model while varying the element size of the thin shell elements and the number of integration points through thickness is carried out to find out the best suitable values of these parameters.

At very first step the model is meshed using default element size selected by the program based on the overall size of the model. After that the meshed model is discretized again with reducing the element size to half of the default values and the output variables like peak force and total deformation is analyzed. This refinement of element size tends to saturate the value of peak force as the element size is reduced up to 3.0mm.

To study the effect of element size upon further refinement three separate analysis are carried out by using the element size of 2.0×2.0mm, 2.5×2.5mm and 3.0×3.0mm while the number of integration points are kept same with a value of 5 integration points through the thickness of the shell element. The results for the force-deflection curves of these three analysis for different element sizes are plotted on the same graph to check out the insensitivity of the finite element model toward element size.

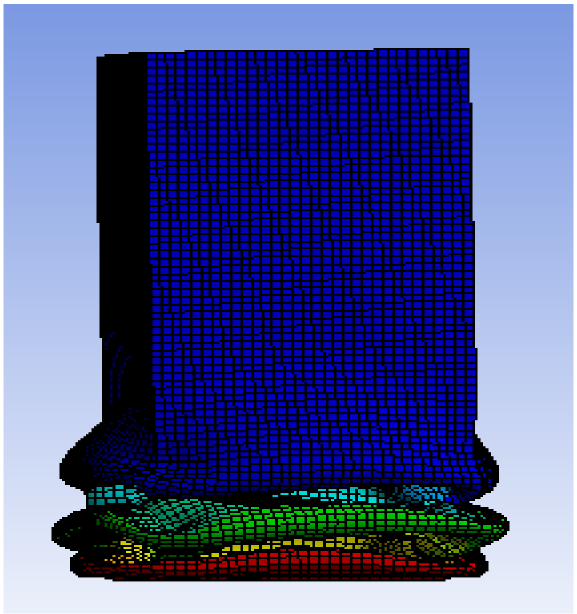


Figure 23: Element size 2.0×2.0mm

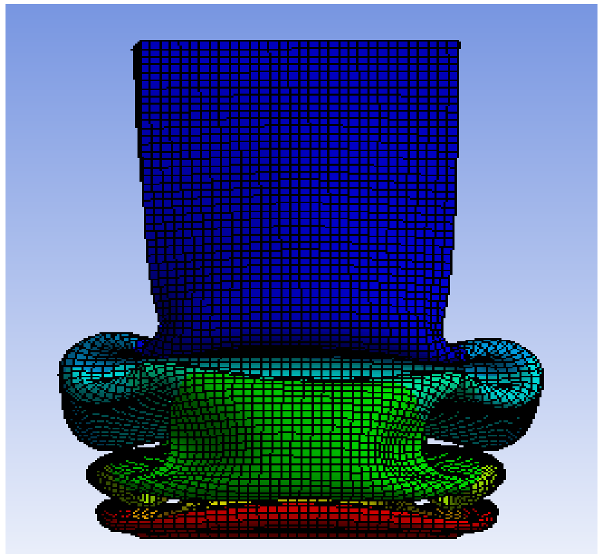


Figure 24: Element size 2.5×2.5mm

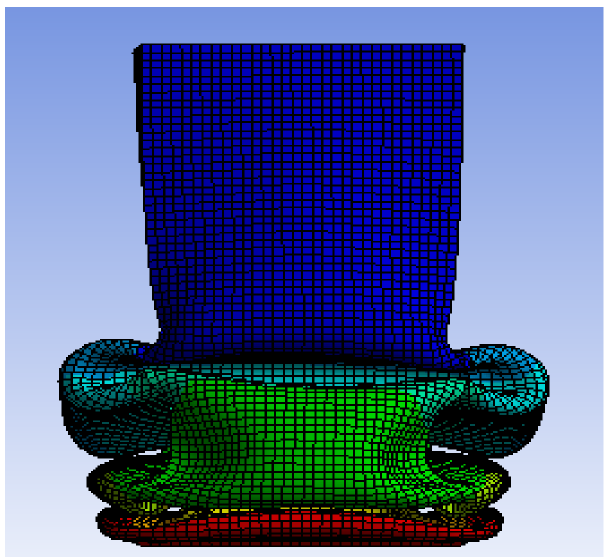


Figure 25: Element size 3.0×3.0mm

The above figure shows the total deformation of the energy absorber with respect to the selected element size. It can be clearly seen that the thin walled tube goes under progressive deformation with the application of dynamic loading.

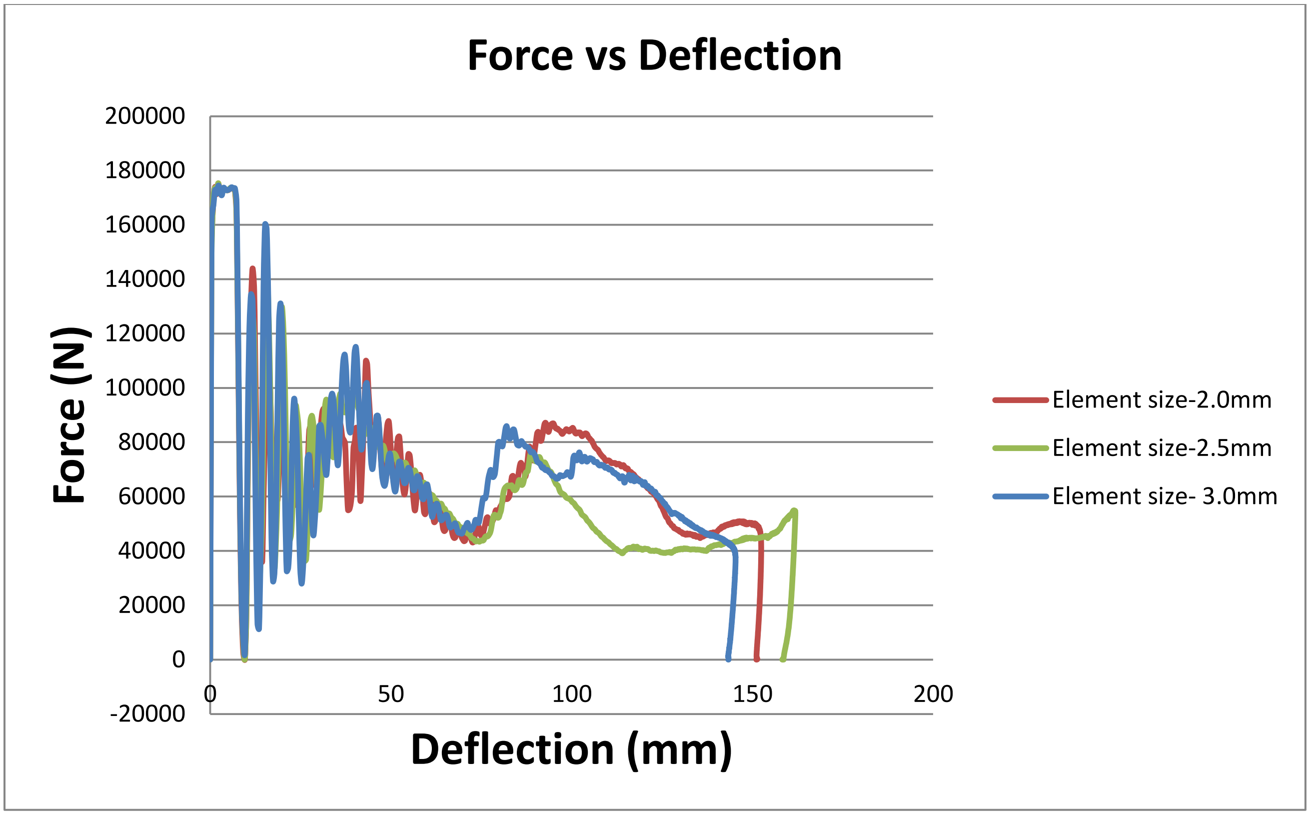


Figure 26: Force vs. deflection curves for different element sizes

It has been observed that after the element size is reduced to 3.0mm, the further refinement does not making any remarkable changes in the results. The peak dynamic force and trend of the crushing force is almost identical but there lies minor changes in the total deformation of the energy absorber. This minor change which is calculated to be within a range of below 5% variation is acceptable due to the numerical noise. Based on this observation the element size of 2.5×2.5mm is selected for further analysis of the crash boxes of different geometries.

After the selection of element size the number of integration points is varied through the thickness of the shell element to find out their effect on the results. For this purpose the number of integration points through thickness is kept 3, 5 and 7 with the selected element sizes. Comparison of the force-deflection is shown in the figure below.

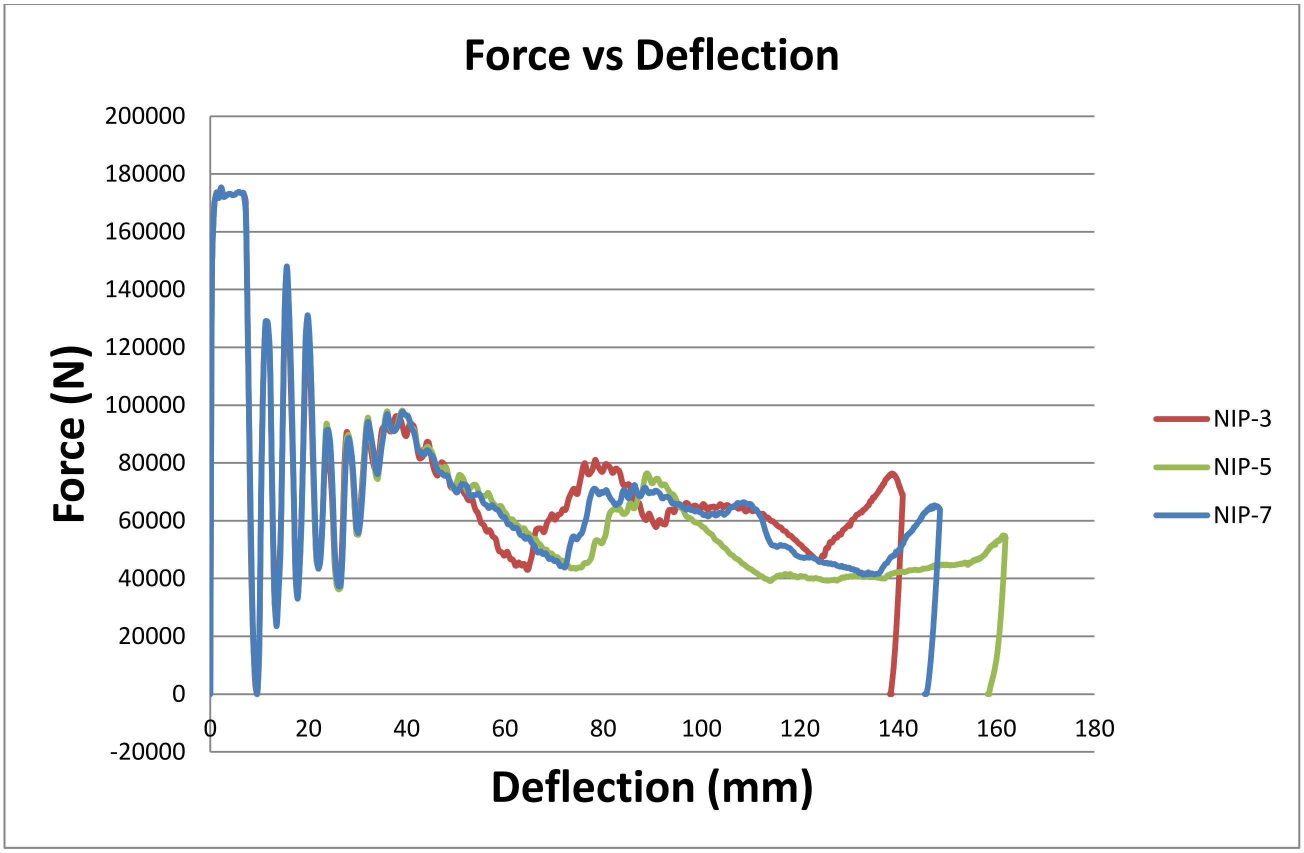


Figure 27: Varying number of integration points through thickness

The observation has been made that there is no major changes in the results with variation of number of integration points through thickness. So 5 number of integration points through thickness is selected with element size of 2.5×2.5mm for the final finite element model of the thin walled tube.

6.3 Validation of FEM:

The results of the numerical approximated solution of the square thin walled tube are compared with existing experimental literature for validation purposes. As mentioned earlier the base model is designed on the same dimensions as of the specimen used in experiments. All other physical parameters are modeled as to replicate the physical problem so the accuracy and trustworthiness of the results of finite element model is ensured. This step ensures that the results of the other different geometries under this validated model are reliable and crashworthiness parameters can evaluate their performance.

The force-deflection curves of the experimental setup and simulated model are compared by plotting on the same graph. The tabular data for the experimental setup is obtained with the help of graph digitizer software “GetData”.

The graphs of force and deflection with respect to time are shown in the figures below;

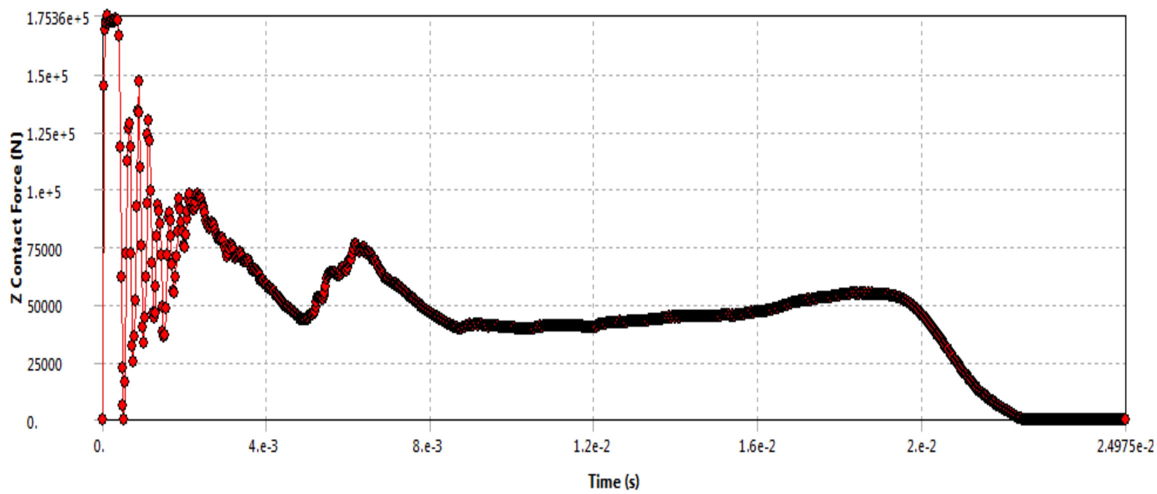


Figure 28: Force vs. time

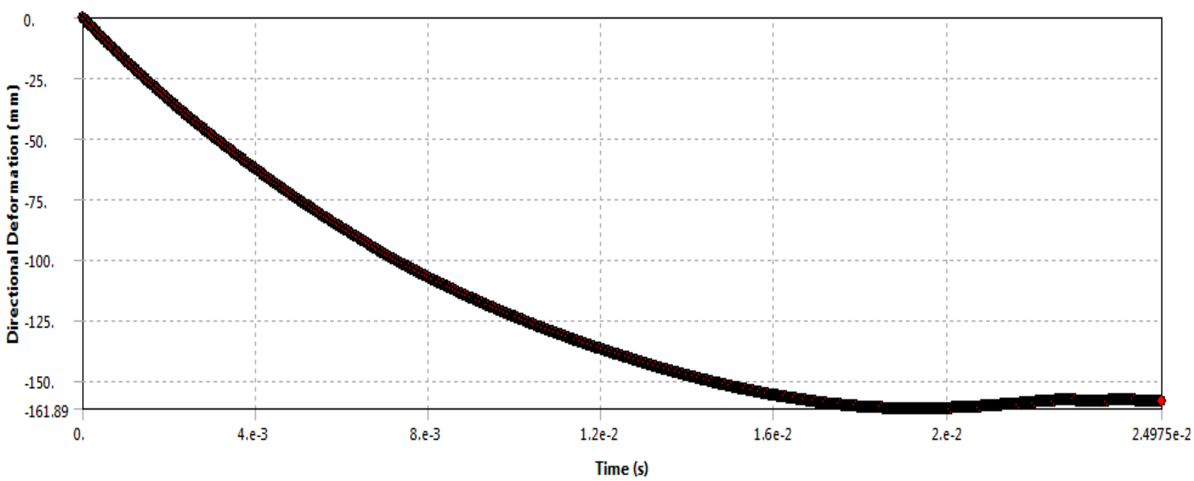


Figure 29: Deflection vs. time

The results of the simulation are validated through the experimental analysis carried out of square thin walled tube made of the aluminum alloy “AA6060 T6”. This experimental study was performed to check the effect of using different tempers of aluminum alloys and keeping their tracks on the selected crashworthiness parameters[30]. Force vs. deflection curves of different cases were plotted in this study. In our study the selected material of the crash box is “AA6060 T6” and the force-deflection curve for this case is shown below.

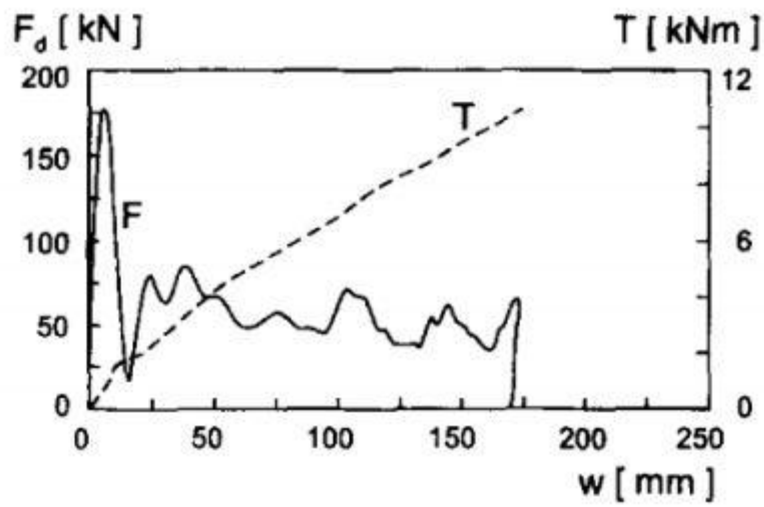


Figure 30: Force vs. deflection-Experimental

Tabulated data is exported to the "MS Excel" for both of the cases and plots are compared as shown below.

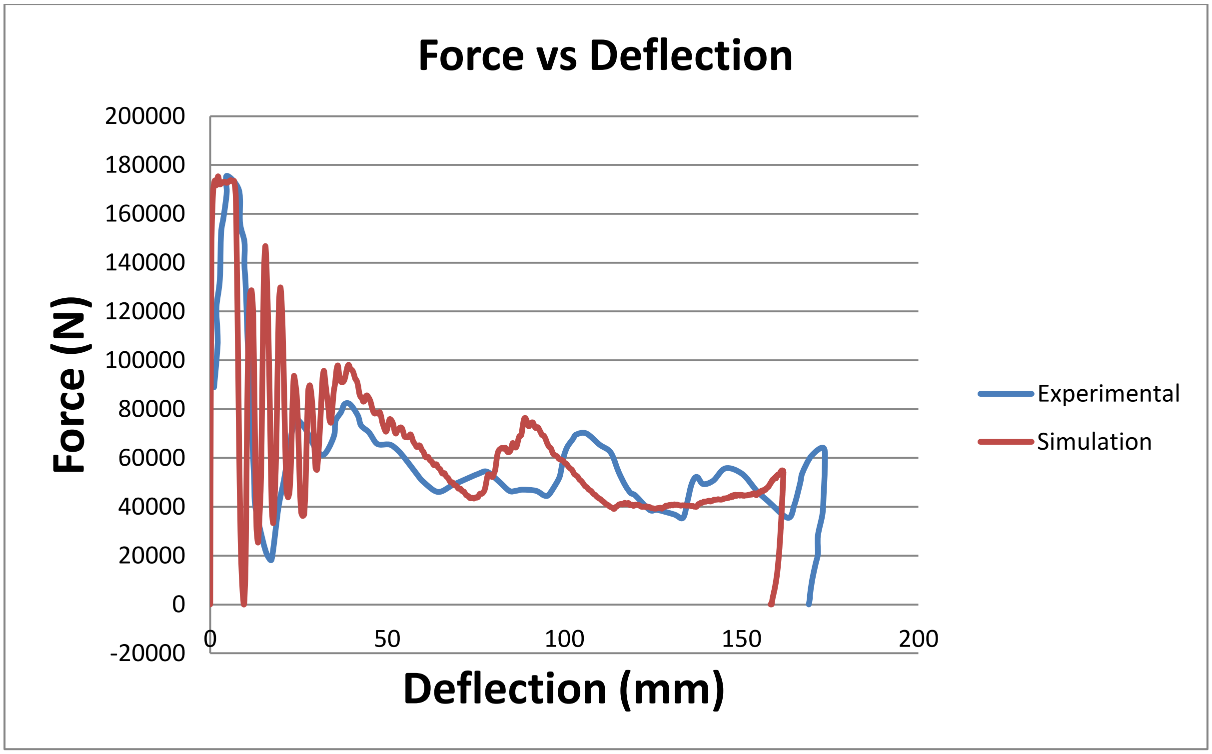


Figure 31: Comparison of experimental and simulated plots

The discrepancy in the results is very minimal which ensures the validation of finite element model.

Values are compared in the table below.

Table 2: Comparison of experimental and simulation results

	Peak Force (PF)	**Deflection**
Experimental	175kN	169.4mm
Simulation	175.4	161.8mm
Difference	0.22%	4.48%

The difference in the results is due to the approximation of experimental data for plotting and the numerical noise in plotting the simulation results.

7 NUMERICAL ANALYSIS OF DIFFERENT GEOMTRIES

Different cross sectional shapes of thin walled tubes are designed with the dimensions calculated by having approximately the same cross sectional area of all the cases.

7.1 Rectangular cross section:

Dimensions of the thin walled tube of rectangular cross section are 100mm×64mm and length is 310mm. Thickness of the walls is varied from 1.5mm to 4.0mm with an increment of 0.5mm. The deformed specimens are shown in the figures.

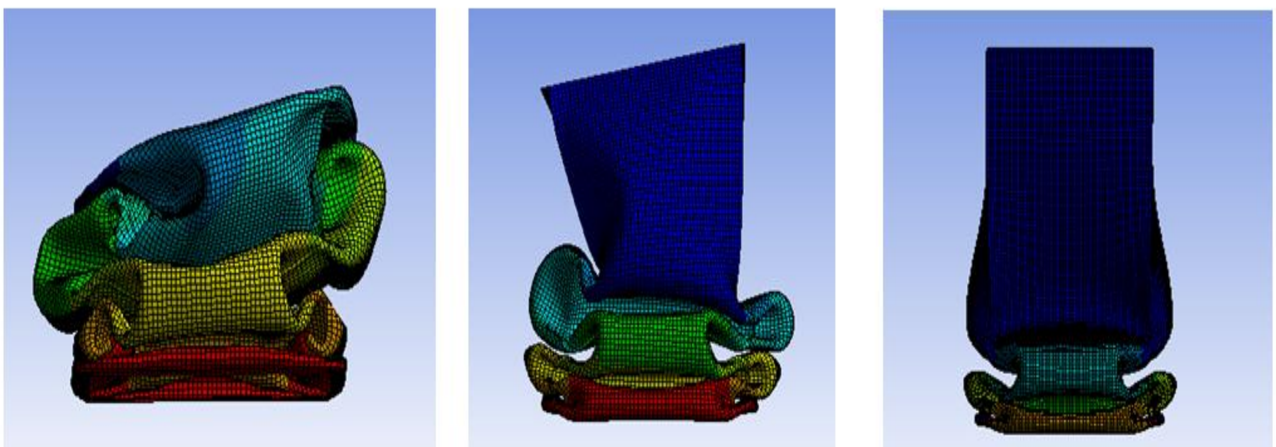


Figure 32: Rectangular- t=1.5mm, 2.0mm, 2.5mm

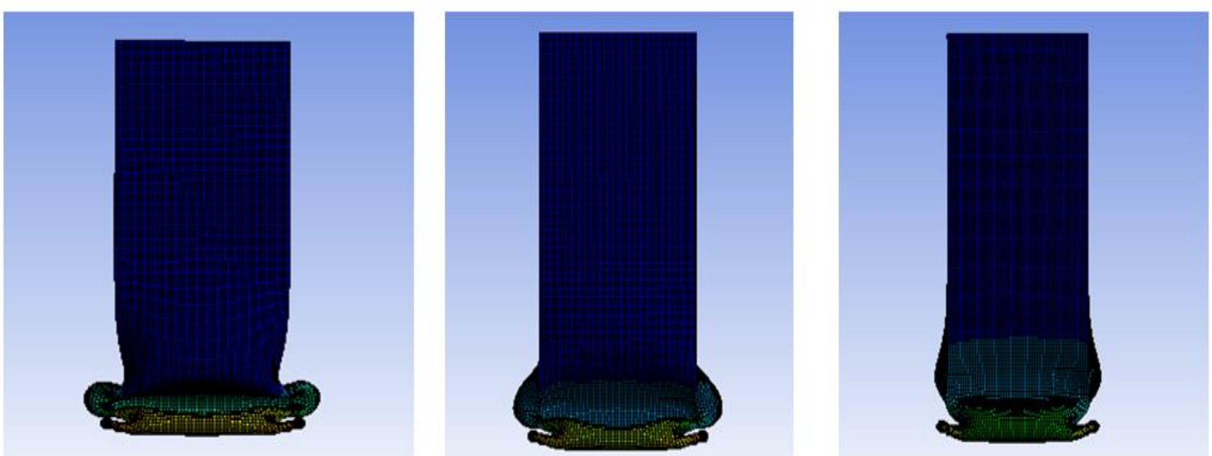


Figure 33: Rectangular- t=3.0mm, 3.5mm, 4.0mm

7.2 Results:

Force-deflection curves are compared in the figure below.

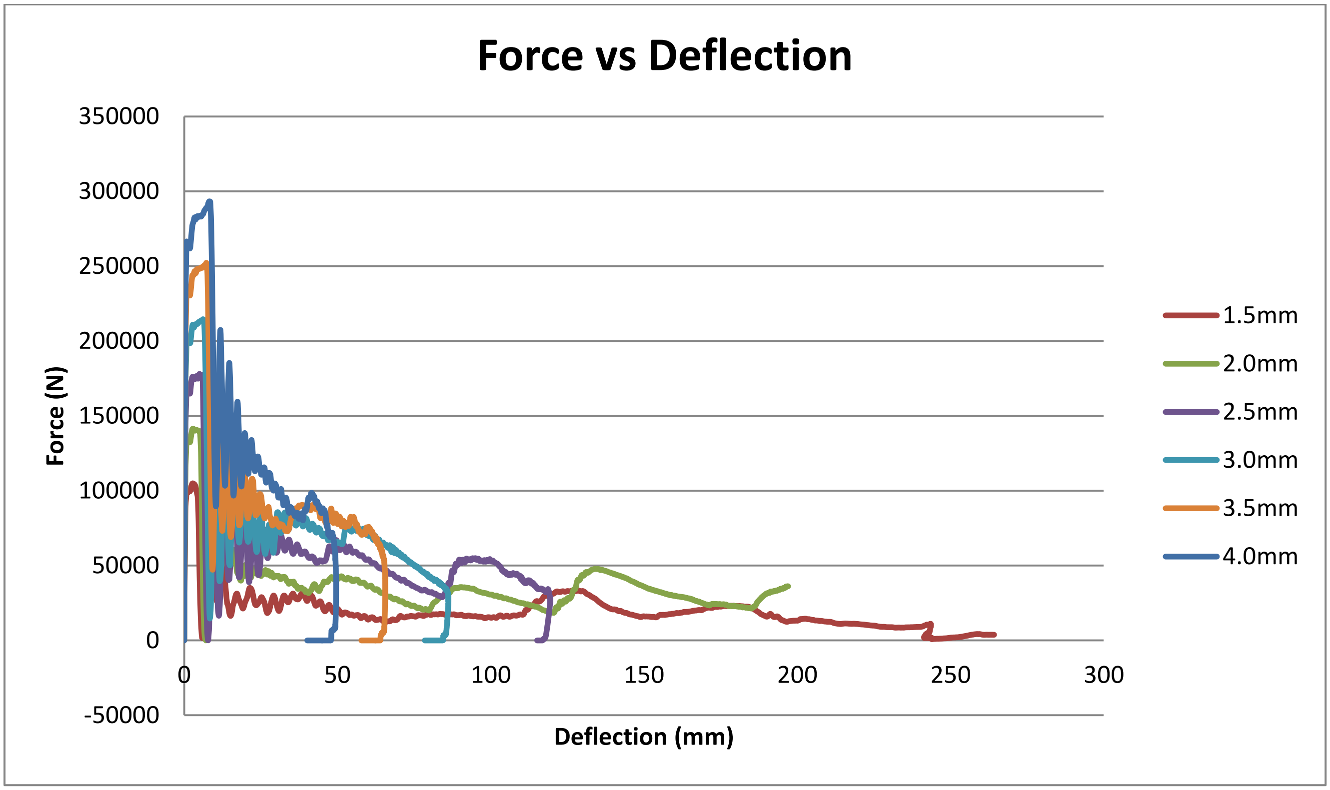


Figure 34: Comparison of force deflection curves

Force-deflection curves showed the trend of thickness on peak force (PF). The initial collapse force is increasing with the increase in thickness. The total deformation of the specimen is decreasing with the increase in wall thickness. Total deformation does have a net effect on mean crushing force which is important crashworthiness parameters.

The value of peak crushing force for thickness value of 1.5mm thin walled tube has the minimum value of peak crushing force which is 104.8kN. But the deformation length of the specimen is maximum of 264.2mm which has caused the mean crushing force of 19.83kN. The increasing trend of peak crushing force has reached to the maximum value of 293.2kN for the thin walled tube having thickness of 4.0mm. Specific energy absorption is maximum of 12.78kg/kJ for the thin walled tube having minimum thickness of 1.5mm and is minimum at 6.26kg/kJ for the thin walled tube having maximum wall thickness of 4.00mm. The value of mean crushing force has an increasing trend with the increase of wall thickness but in contrast to that the value of load uniformity has the maximum value of 5.28 for the case of wall thickness of 1.5mm and have an decreasing trend with the increase in wall thickness of thin walled tube.. Crashworthiness parameters are evaluated and shown in the table below.

t (mm)	M (kg)	EA (kJ)	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU	MD (mm)
1.5	0.41	5.24	12.7804878	104.8	19.8334595	5.284	264.2
2	0.54	6.94	12.85185185	141.2	35.210553	4.010161	197.1
2.5	0.68	6.92	10.17647059	177.8	57.9564489	3.067821	119.4
3	0.82	6.87	8.37804878	214.5	79.6983759	2.691397	86.2
3.5	0.96	6.85	7.135416667	251.9	104.580153	2.408679	65.5
4	1.09	6.83	6.266055046	293.2	137.979798	2.124949	49.5

Table 3: Crashworthiness parameters- Rectangular case

7.3 Circular Cross section:

Dimensions of the thin walled tube of circular cross section are as diameter of 90mm and length is 310mm. Thickness of the walls is varied from 1.5mm to 4.0mm with an increment of 0.5mm. The deformed specimens are shown in the figures.

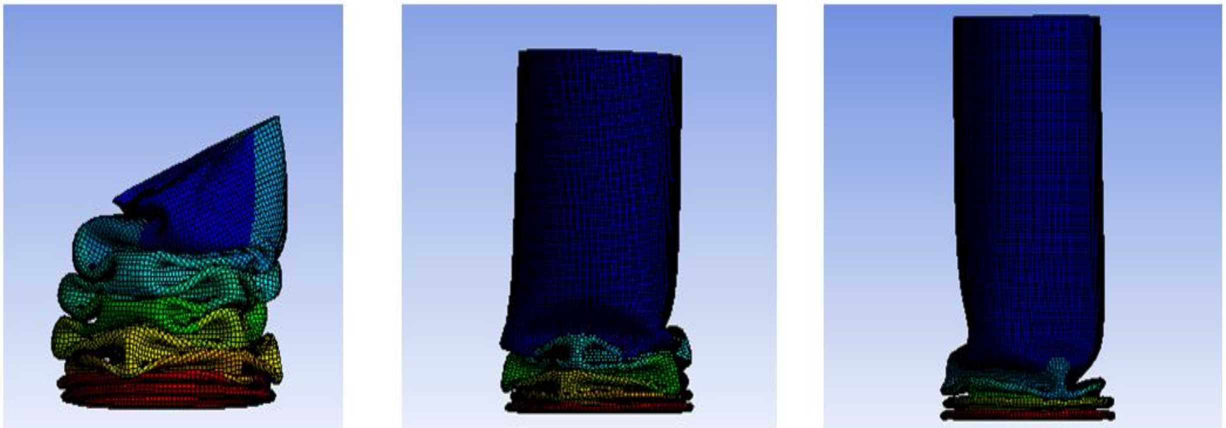


Figure 35: Circular- t=1.5mm, 2.0mm, 2.5mm

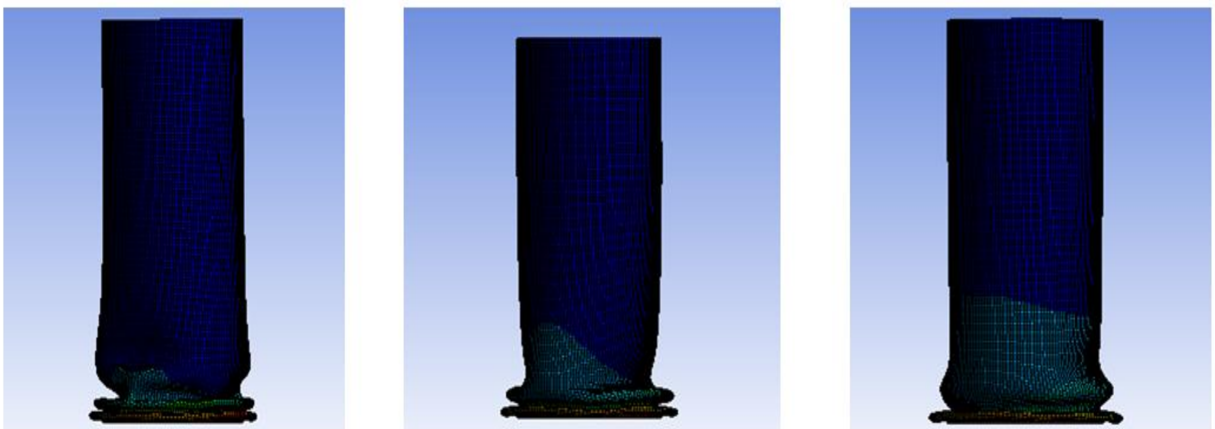


Figure 36: Circular- t=3.0mm, 3.5mm, 4.0mm

7.4 Results:

Force-deflection curves are compared in the figure below.

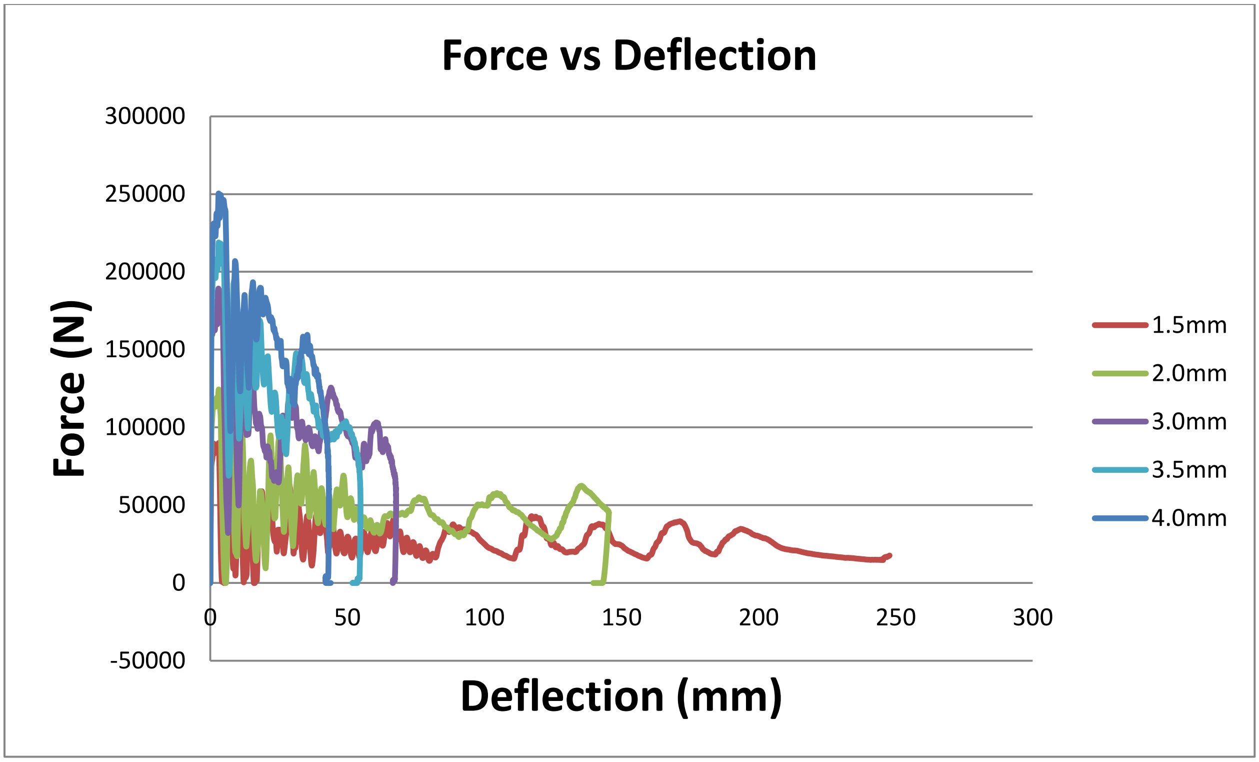


Figure 37: Comparison of force deflection curves

Force-deflection curves showed the similar trend as for the case of rectangular cross section. It is showing same trend for the peak force (PF) with respect to the wall thickness. The initial collapse force is increasing with the increase in thickness. The total deformation of the specimen is decreasing with the increase in wall thickness. Total deformation does have a net effect on mean crushing force which is important crashworthiness parameters.

The value of peak crushing force for thickness value of 1.5mm thin walled tube has the minimum value of peak crushing force which is 89.7kN. But the deformation length of the specimen is maximum of 274.8mm which has caused the mean crushing force of 24.78kN. The increasing trend of peak crushing force has reached to the maximum value of 250.4kN for the thin walled tube having thickness of 4.0mm. Specific energy absorption is maximum of 16.60kg/kJ for the thin walled tube having minimum thickness of 1.5mm and is minimum at 7.35kg/kJ for the thin walled tube having maximum wall thickness of 4.00mm. The value of mean crushing force has an increasing trend with the increase of wall thickness but in contrast to that the value of load uniformity has the maximum value of 3.61 for the case of wall thickness of 1.5mm and has an decreasing trend with the increase in wall thickness of thin walled tube.

Crashworthiness parameters are evaluated and shown in the table below.

t (mm)	M (kg)	EA (kJ)	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU	MD (mm)
1.5	0.41	6.81	16.60976	89.7	24.78166	3.619612	274.8
2	0.47	6.97	14.82979	124.4	47.93673	2.595088	145.4
2.5	0.59	6.89	11.67797	151.2	74.16577	2.038676	92.9
3	0.7	6.86	9.8	189.1	101.3294	1.866191	67.7
3.5	0.82	6.83	8.329268	218.9	124.635	1.756328	54.8
4	0.94	6.91	7.351064	250.4	157.4032	1.590819	43.9

Table 4: Crashworthiness parameters- Circular case

7.5 Polygons:

To study the effect of number of corners, a comparatively smaller in length, thin walled tubes are designed with increasing number of corners from 5 to 8. Crashworthiness parameters of these different polygons are analyzed and the best among them is selected for further analysis.

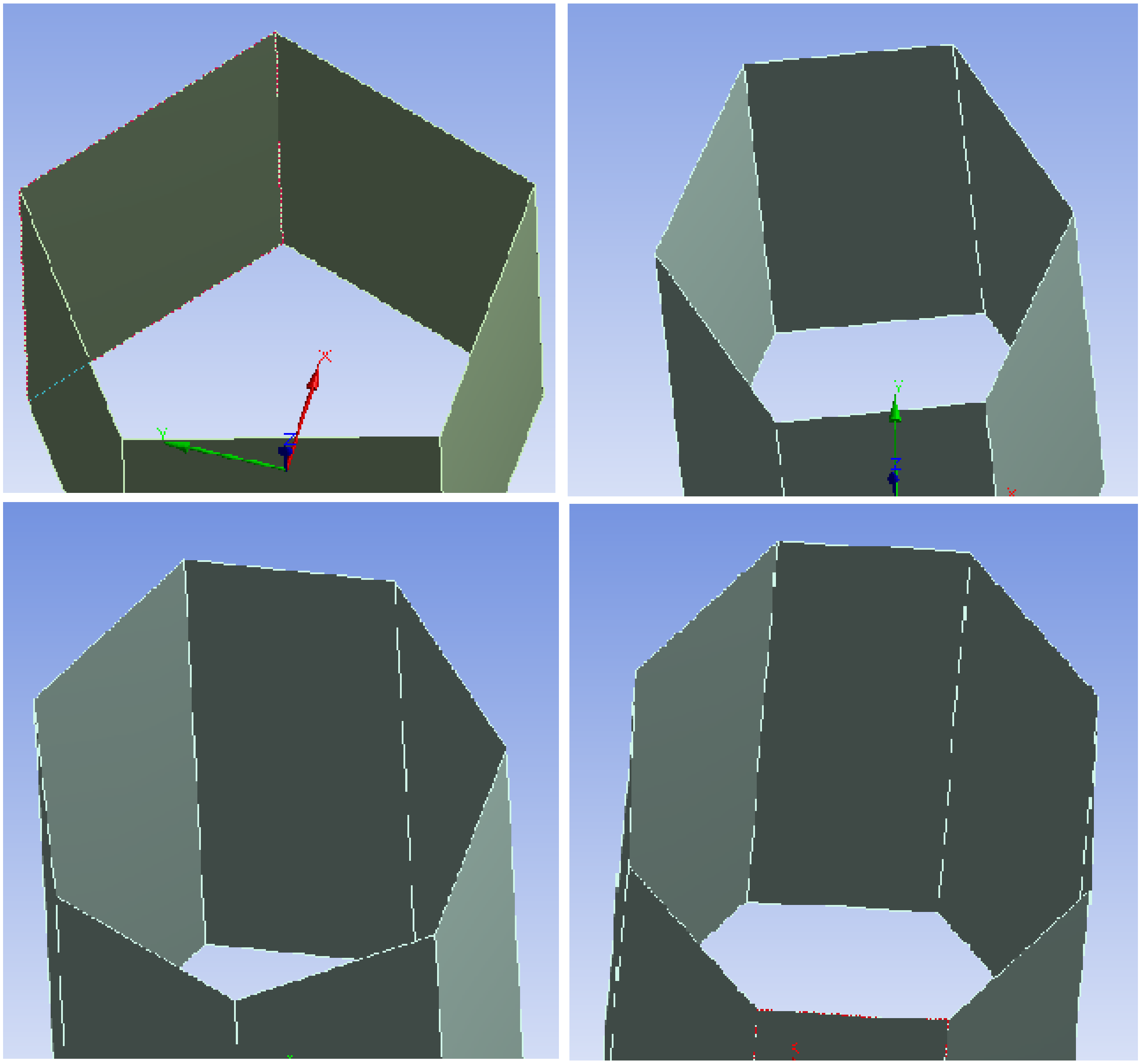


Figure 38: Polygons

The outer diameter of these polygons is 90mm and length is 200mm.

7.5.1 5 Croners:

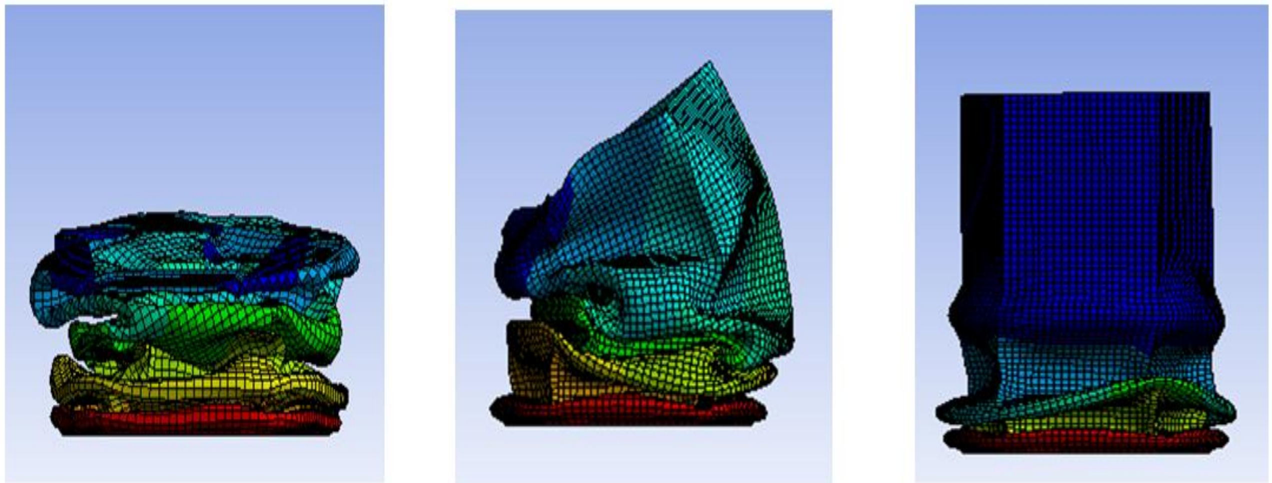


Figure 39: 5 corners- t=1.5mm, 2.0mm, 2.5mm

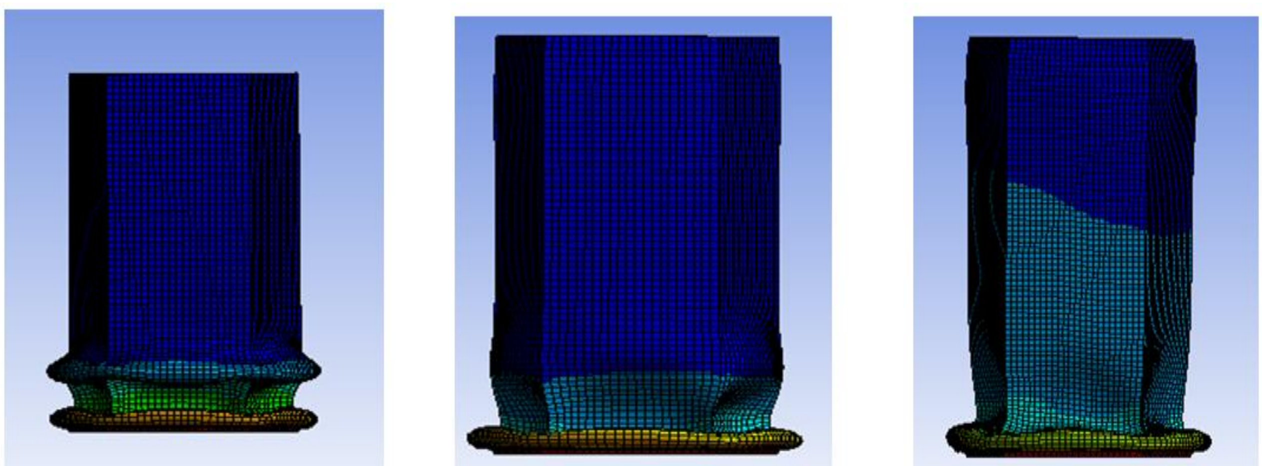


Figure 40: 5 corners- t=3.0mm, 3.5mm, 4.0mm

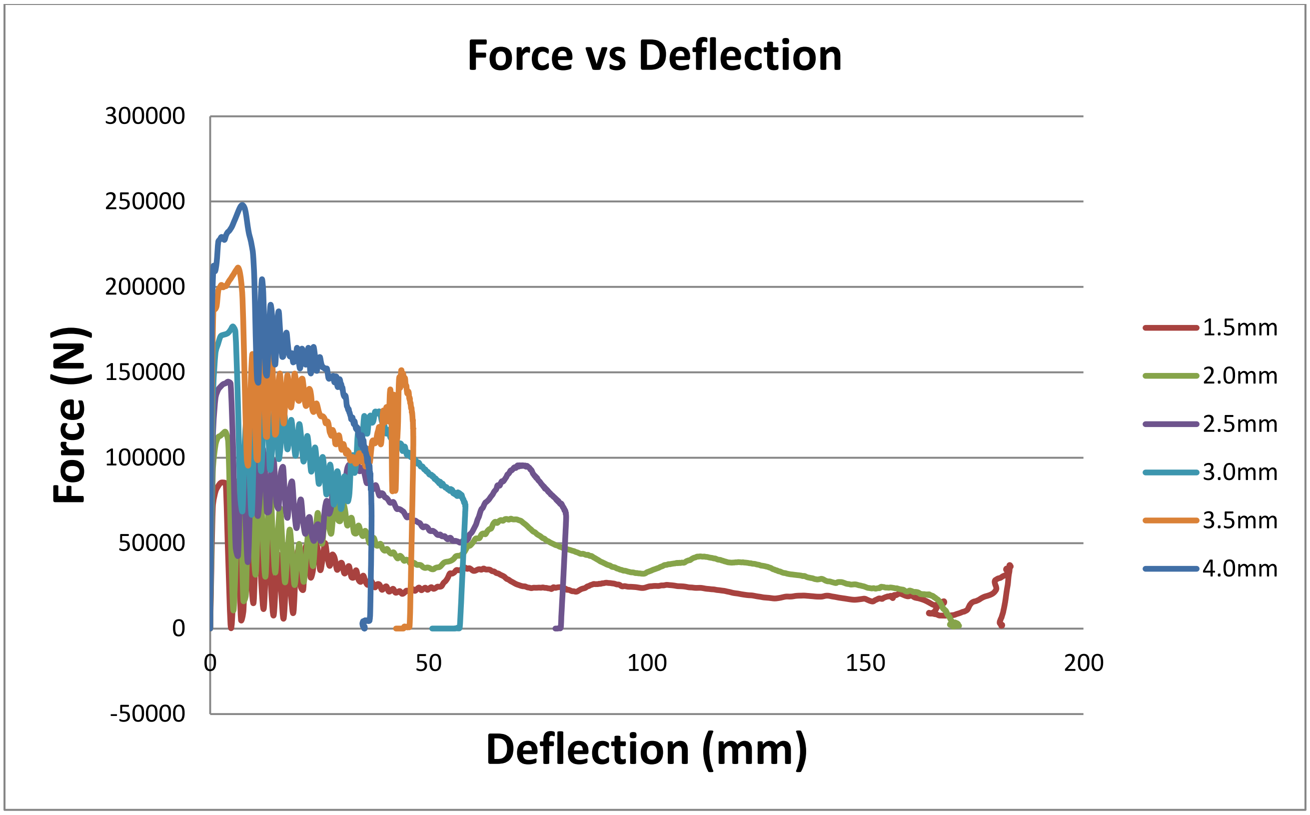


Figure 41: Comparison of force deflection curves

t (mm)	M (kg)	EA (kJ)	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU	MD (mm)
1.5	0.21	4.61	21.95238	85.5	25.15003	3.399599	183.3
2	0.28	6.96	24.85714	115	40.58546	2.833527	171.49
2.5	0.35	6.11	17.45714	144.5	74.96933	1.927455	81.5
3	0.42	6.26	14.90476	176.8	107.1918	1.64938	58.4
3.5	0.49	6.24	12.73469	211.3	134.1935	1.574591	46.5
4	0.57	6.3	11.05263	248	170.7317	1.452571	36.9

Table 5: Crashworthiness parameters- 5 corners

7.5.2 6 Croners:

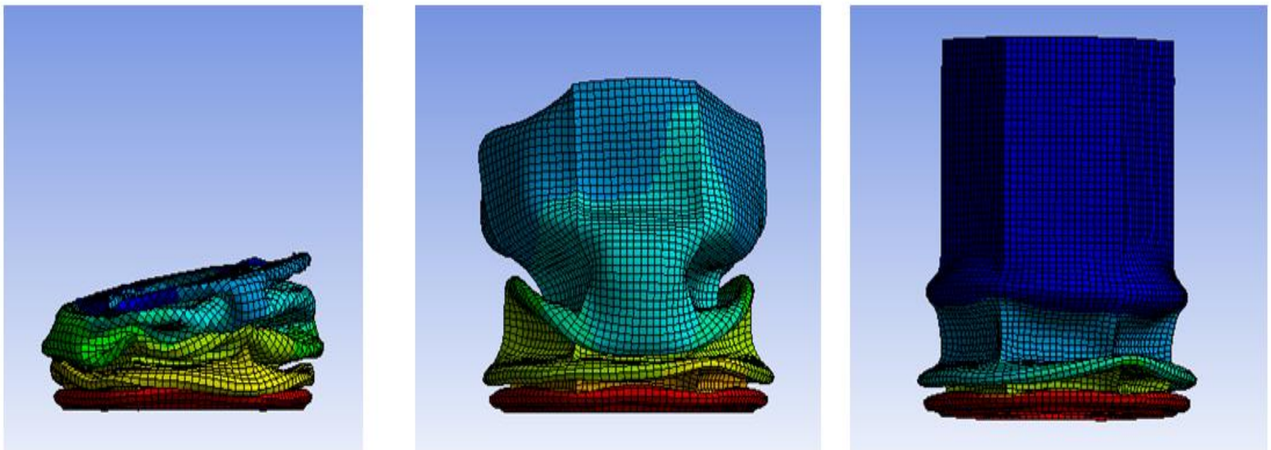


Figure 42: 6 corners- t=1.5mm, 2.0mm, 2.5mm

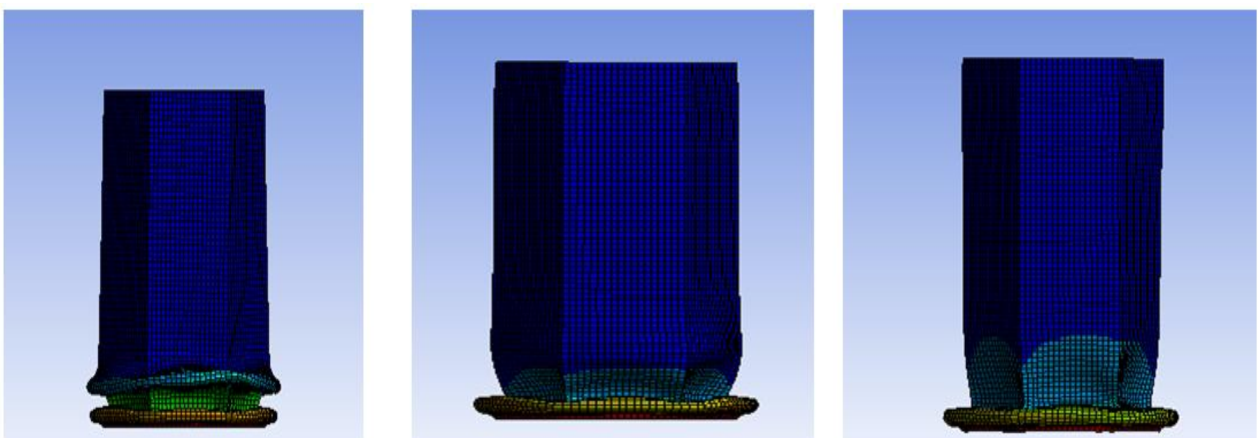


Figure 43: 6 corners- t=3.0mm, 3.5mm, 4.0mm

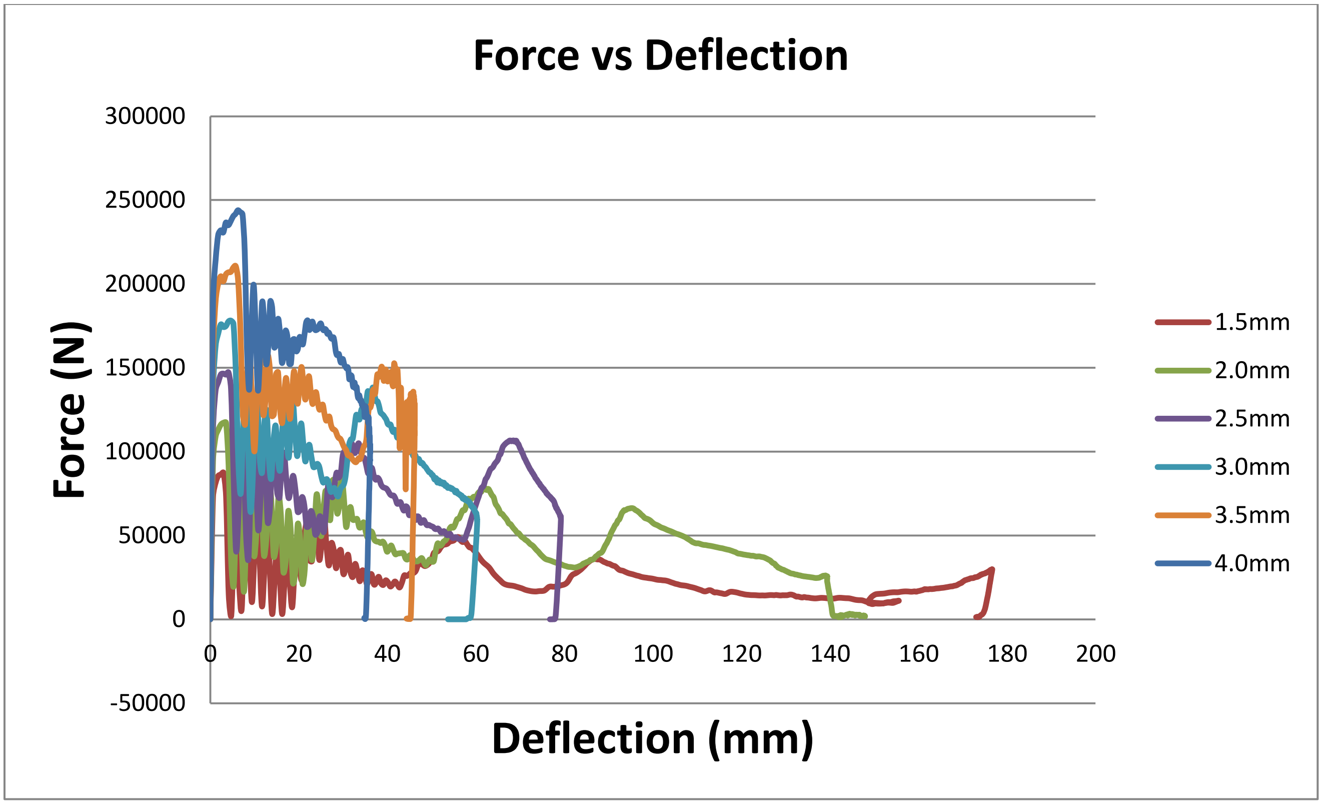


Figure 44: Comparison of force deflection curves

t (mm)	M (kg)	EA (kJ)	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU	MD (mm)
1.5	0.21	4.49	21.38095	87.4	25.41461	3.438966	176.67
2	0.29	6.96	24	117.2	41.61933	2.815999	167.23
2.5	0.36	6.3	17.5	147.22	91.04046	1.617083	69.2
3	0.43	6.29	14.62791	178.13	104.485	1.704837	60.2
3.5	0.51	6.28	12.31373	210.6	135.9896	1.548648	46.18
4	0.58	6.27	10.81034	243.9	174.1183	1.400772	36.01

Table 6: Crashworthiness parameters- 6 corners

7.5.3 7 Corners:

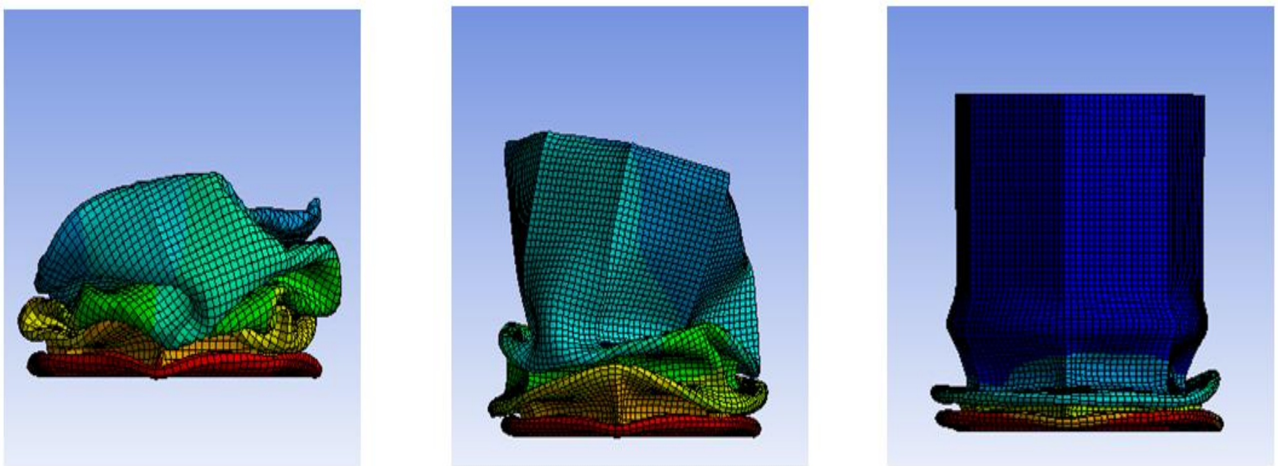


Figure 45: 7 corners- t=1.5mm, 2.0mm, 2.5mm

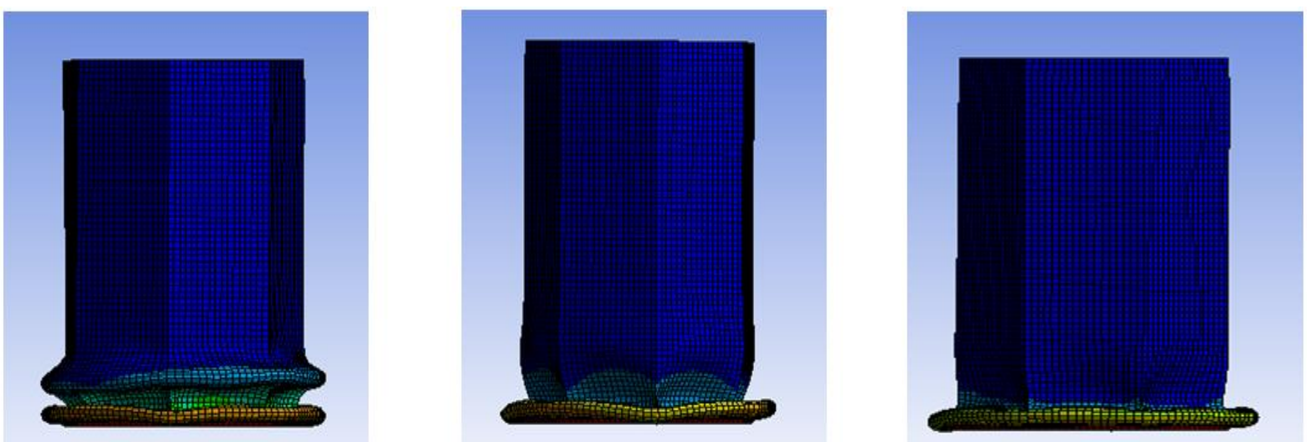


Figure 46: 7 corners- t=3.0mm, 3.5mm, 4.0mm

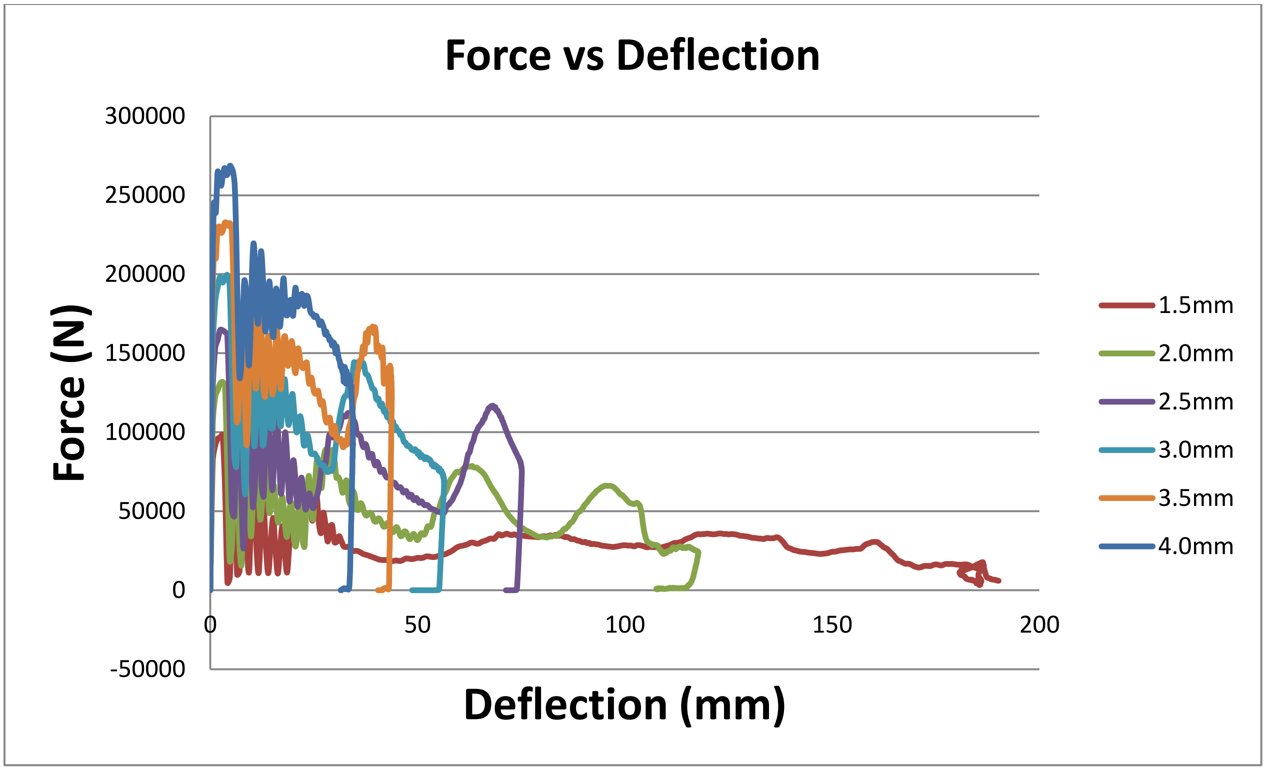


Figure 47: Comparison of force deflection curves

t (mm)	M (kg)	EA (kJ)	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU	MD (mm)
1.5	0.24	5.58	23.25	98.8	28.62272	3.451803	194.95
2	0.32	6.21	19.40625	131.8	52.74333	2.498894	117.74
2.5	0.4	6.29	15.725	164.6	83.75499	1.965256	75.1
3	0.49	6.3	12.85714	199.6	111.623	1.788163	56.44
3.5	0.57	6.34	11.12281	232.6	144.8481	1.605821	43.77
4	0.65	6.34	9.753846	268.5	184.3023	1.456845	34.4

Table 7: Crashworthiness parameters- 7 corners

7.5.4 8 Croners:

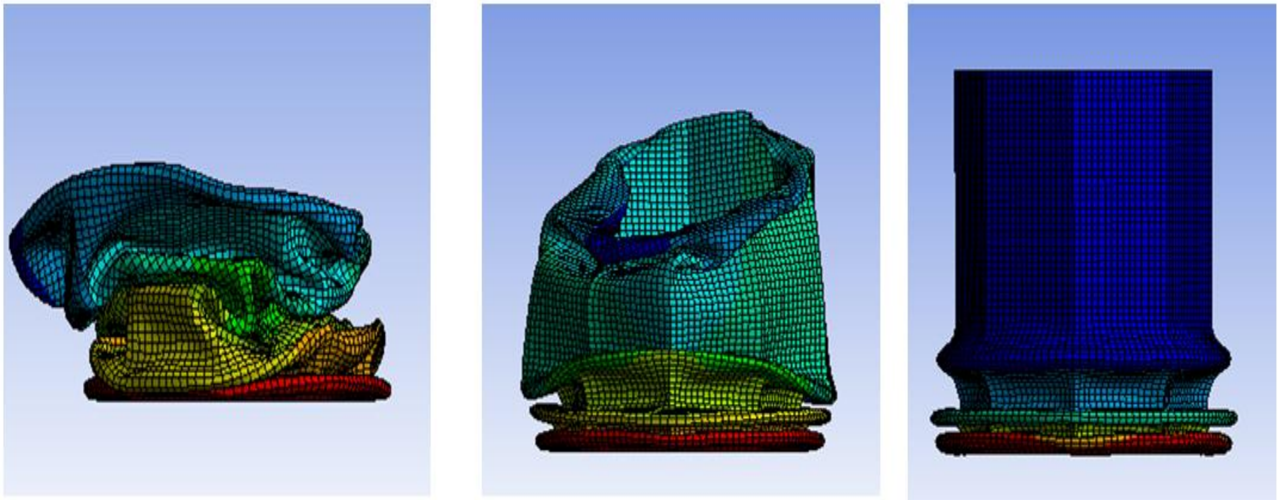


Figure 48: 8 corners- t=1.5mm, 2.0mm, 2.5mm

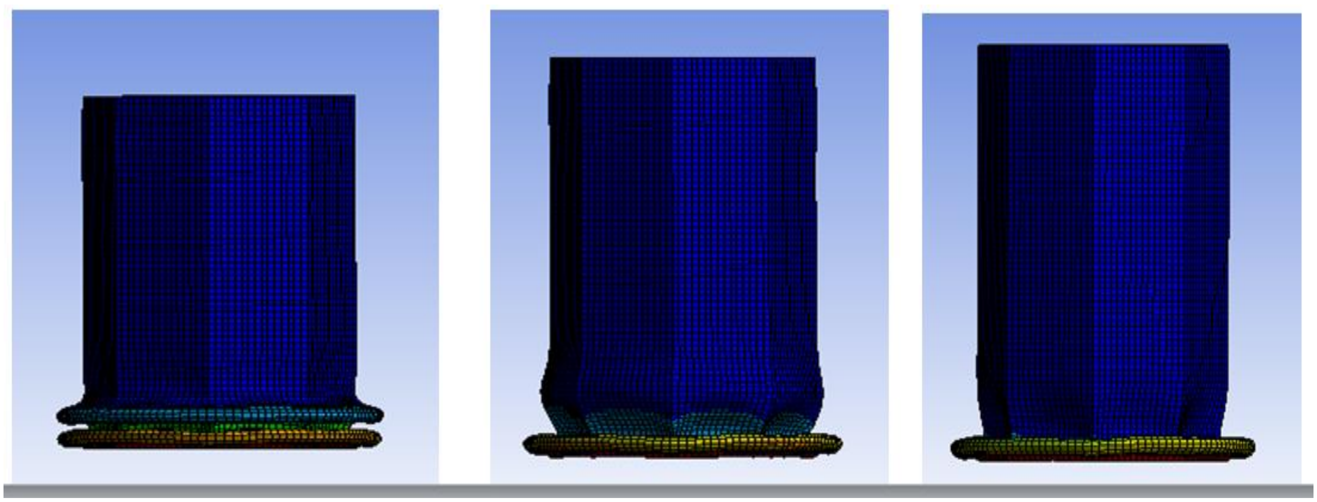


Figure 49: 8 corners- t=3.0mm, 3.5mm, 4.0mm

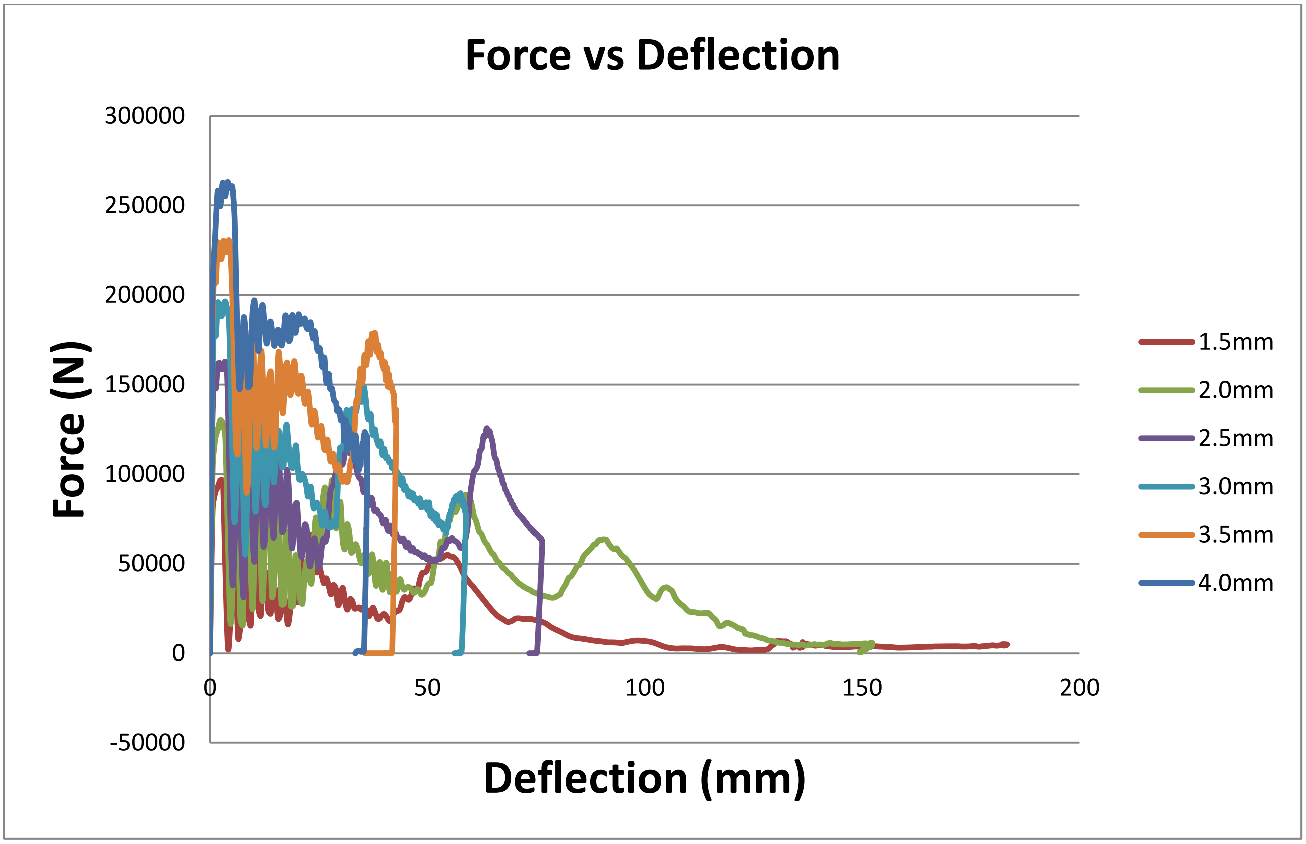


Figure 50: Comparison of force deflection curves

t (mm)	M (kg)	EA (kJ)	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU	MD (mm)
1.5	0.24	4.78	19.91667	96.5	25.72105	3.751791	185.84
2	0.32	6.23	19.46875	130.1	40.87658	3.182751	152.41
2.5	0.4	6.3	15.75	162.7	82.41758	1.974093	76.44
3	0.48	6.28	13.08333	196.4	106.8027	1.838904	58.8
3.5	0.56	6.28	11.21429	230.6	146.729	1.571605	42.8
4	0.64	6.3	9.84375	263	174.5152	1.507032	36.1

Table 8: Crashworthiness parameters- 8 corners

7.6 Results:

For all the polygons force-deflection curves showed the trend the similar trend as of for circular and rectangular cross section. The effect of thickness on peak force (PF) showed that it has a proportional effect. The initial collapse force is increasing with the increase in thickness. The total deformation of the specimen is decreasing with the increase in wall thickness. Total deformation does have a net effect on mean crushing force which is important crashworthiness parameters.

Peak crushing force have an increasing trend with the increase of number of corners from 5 to 7 but the cross sectional shape having 8 corners has a slightly less value than the cross sectional shape having 7 corners. For the case of thickness values of 1.5mm having 5 corners the value of peak crushing force is 85.5kN which goes to the maximum of 98.8kN for the cross sectional shape having 7 corners. The value of peak force for cross sectional shape having 8 corners is slightly less which is 96.5kN. The mean crushing force have the same trend as of the peak crushing force with respect to the number of corners.

The values of other crashworthiness parameters are analyzed for the case of wall thickness of 2.5mm The case of wall thickness of 1.5mm is bottomed out design as it does have the lowest value of energy absorbed for all cases. The specimen having wall thickness of 2.0mm have the maximum specific energy absorption of 24.85kg/kJ for 5 corners but have lower values of mean crushing force and load uniformity. The cross sectional shape of 6 corners with wall thickness value of 2.5mm has the maximum specific energy absorption of 17.5kg/kJ among all the thin walled tube of same thickness. This case has the value of peak crushing force of 147.22kN which is slightly more than the case of 5 corners which have 144.5kN but it is less than the other two cases of 7 and 8 cornes which have values 164.7kN and 162.4kN respectively.

Comparing other crashworthiness parameters for this thickness value of 2.5mm, the values of mean crushing force and load uniformity are 91.04kN and 1.61 respectively which are better than all other cases. These two values make this case the best optimum shape among the polygons under consideration. So the hexagonal shape of thickness value of 2.5mm is selected for further analysis and comparison with other basic cross sectional shapes.

7.7 Hexagonal Cross section:

Thin walled tube with hexagonal cross section is modeled with the dimension as similar to the base model so the thin walled tube of different cross sections can be compared with respect to the selected performance parameters of crashworthiness,

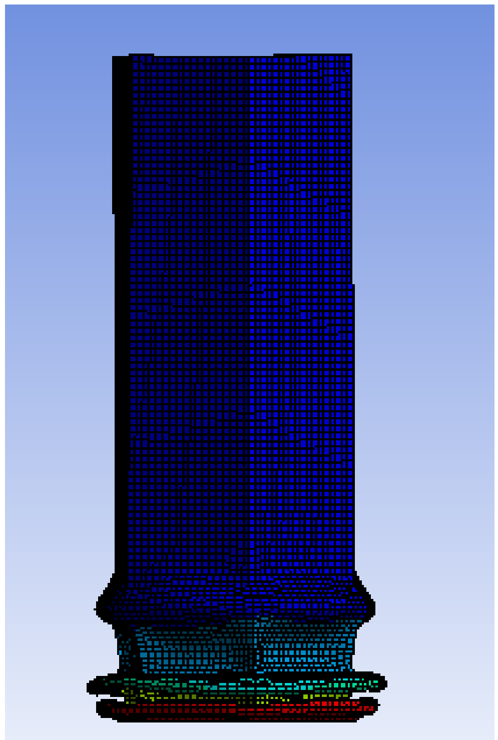


Figure 51: Deformed shape. Hexagonal cross section

Crashworthiness parameters are evaluated and tabulated below;

t (mm)	M (kg)	EA (kJ)	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU	MD (mm)
2.5	0.56	6.32	11.28571	146.3	80.20305	1.82412	78.8

Table 9: Crashworthiness parameters – Hexagonal shape

Force-deflection curves for the four selected shapes are plotted for comparison below.

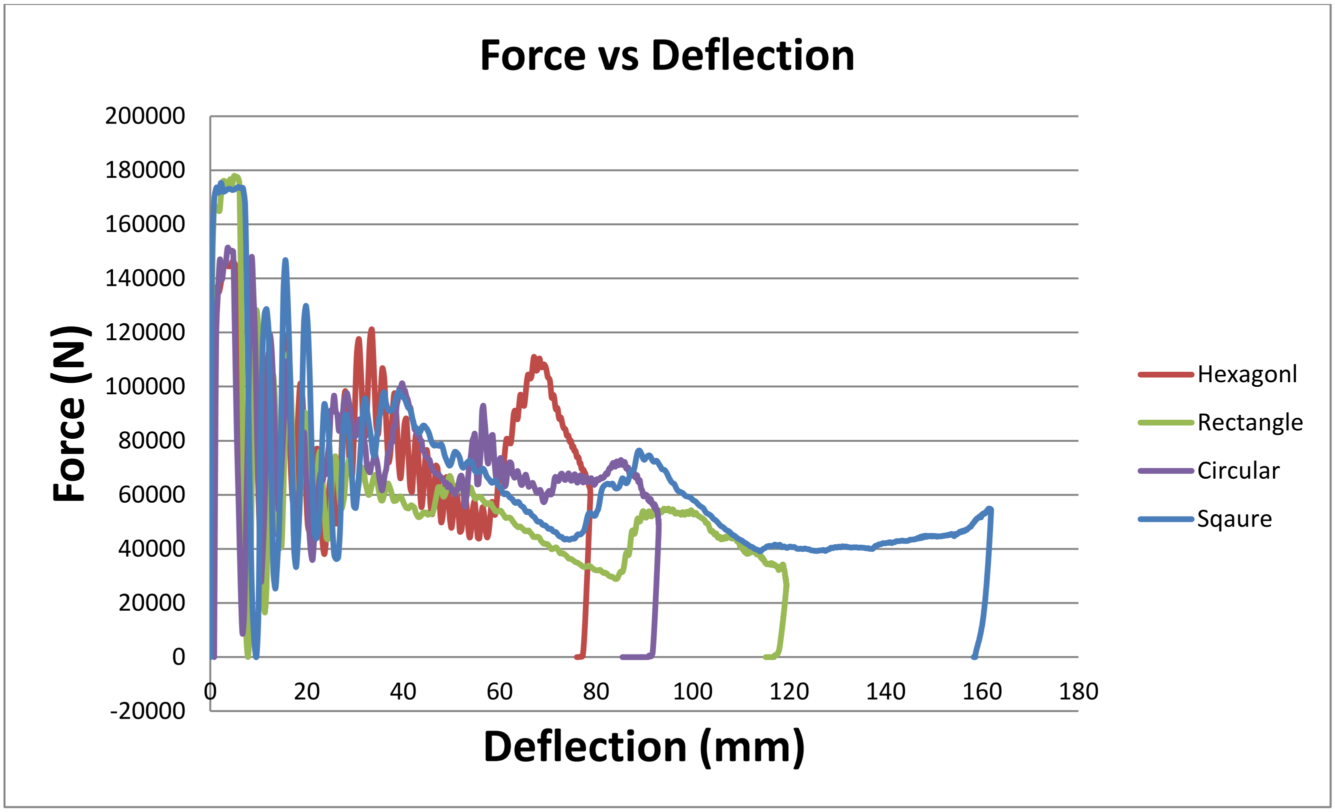


Figure 52: Force-deflection for four cross sectional shapes

Crashworthiness parameters of thin walled tubes of different cross sections are tabulated below.

Geomtery	SEA (kg/kJ)	PCF (kN)	MCF (kN)	LU
Square	15.36	175.3	62.94	2.78
Hexagonal	11.28571	146.3	80.20305	1.82412
Rectangle	10.17	177.8	57.95	3.06
Circular	11.67	151.2	74.16	2.03

Table 10: Crashworthiness parameters of different cross section

7.8 Results and Discussion:

The four selected cross sectional shapes are examined under same dynamic impact loading conditions have clearly showed that the hexagonal shape is the best optimum solution. Hexagonal shape have specific energy absorption of 11.28 kg/kJ which is more than the rectangular and circular cross sectional shape which have specific energy absorption of 10.02kg/kJ and 11.57kg/kJ respectively. The value of specific energy absorption for square cross sectional shape is 15.36kg/kJ which is higher than the hexagonal cross sectional shape but in contrast to this the values of other crashworthiness parameters are less suitable as compared to the hexagonal cross sectional shape. Mean crushing force has the highest value of 80.20kN for the hexagonal shape which is desirable crashworthiness parameters. The value of load uniformity is 1.82 which is also the lowest among the four selected shapes. The value of unity for load uniformity is the ideal case for a thin walled tube absorbing crash energy.

8 CONCLUSION

Thin walled tubes of aluminum alloy are excellent product in era of crashworthiness for their privilege of having high strength to weight ratio. Progressive deformation mode is the best collapsing scenario of thin walled tube under axial impact loading. Increase in number of corners tends to make progressive deformation more uniform as compared to rectangular and other cross sectional shapes. The even number of polygonal edges leads to the progressive deformation and eventually has more capability to absorb energy. In real life world the road traffic accidents are not pure axial in nature but the collision occur at some angle so thin walled columns can be studied under oblique loading condition as a future work of this study.

9 APPENDIX A

10 REFERENCES

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CERTIFICATE OF COMPLETENESS

It is hereby certified that the dissertation submitted by NS Sajjad Hussain, Reg No. **00000172303**, Titled: ***Numerical Analysis on Crashworthiness Design of Thin Walled Columns under Dynamic Impact Loading Conditions*** has been checked/reviewed and its contents are complete in all respects.

Supervisor's Name: **Lt Col Dr Syed Gul Hassan Naqvi** Signature: _______________

Date: _____________________