

ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY
School of Mechanical, Chemical and Materials Engineering



**Analysis of the full frontal crashworthiness of four wheel light
weight vehicles with and without crashing load absorber**

A thesis submitted in partial fulfillment of the requirements for the award of

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Master of Science

In

Automotive Engineering

By

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SCHOOL OF MECHANICAL, CHEMICAL AND MATERIALS ENGINEERING
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CANDIDATE'S DECLARATION

I hereby declare that the thesis work which is being presented in the paper entitled **“Analysis of the full frontal crashworthiness of four wheel light weight vehicles with and without crashing load absorber”** in partial fulfillment of the requirements for the degree of master of science in Automotive Engineering is an authentic record of my own work carried out from Sep 2015 to January 2016 under the supervision of Mr. N. Ramesh Babu, Associate Professor, Department of Mechanical and Vehicle Engineering, Adama Science and Technology University, Adama, Ethiopia.

The matter embodied in this paper has not been submitted by me for the award of any other degree or diploma. All relevant resources of information used in this paper have been duly acknowledged.

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This is to certify that the above declaration made by candidate is correct.

Mr. N. Ramesh Babu (Associate Professor)

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God has everything to do. All things were made through him, and nobody was made without him. He is always the sponsor of my existence and gave a full health through this thesis work.

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DEDICATION

*To my wife Melkam Tadie, who take the responsibility of looking after our
‘pioneer’ daughter; Tsion Ermias without my involvement*

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ACRONYMS AND SYMBOLS

2D	two dimensional
3D	three dimensional
HIC	head injury criteria
FMVSS	federal motors vehicle safety standard
CMVSS	Canadian motor vehicles safety standard
FEM/A	finite element method/analysis
CAD/E	computer aided design/engineering
SOFI	small overlap frontal impact
IIHS	insurance institute highway safety
PDOF	principal directions of force
RRB	rear roof bow
RSR	rear side rail
SWR	structure-to-weight ratio
FRB	front roof bow
CRIS	Controlled Rollover Impact System
DOE	design of experiment
NHTSA	national highway traffic safety administration
SPOC	several partial overlap collisions
CMVDR	Canadian motor vehicle driver regulations
ODB	overlap deformation beam
ABS	acrylonitrile butadiene styrene
PEI	plastic-elastic isotropy
GMT	glass mat thermoplastic
SMC	sheet molding compound
EPP	expanded poly-propylene
ETP	extended tri-propylene
IPM	ideal point method
RTE	restraint energy

CIREN	crash injury research engineering network
CDC	collision deformation classification
ECE	European crash engineering
NASS-CDS	national automotive sampling system-crashworthiness data system
SAE	society of automotive engineers
NCAP	new car assessment program

ABSTRACT

Road traffic accident is the major alarming issue in the world. The crashworthiness enhancement of vehicle structures is a very challenging task during the vehicle design process due to complicated nature of vehicle design structures that need to comply with different conflicting design task requirements. Although different safety agencies and researchers have issued and modified standardized crash tests to guarantee structural integrity and occupant survivability, there is continued rise of fatalities in vehicle crashes especially the passenger cars. This research explores the analysis of a full frontal crashworthiness of providing energy absorbing systems as a function of the impact speed to achieve enhanced occupant as well as driver safety. The study is conducted starting from design of different parts of the crashing load absorbers, where the performance potentials of different concepts in energy absorbing elements are investigated. The models of the existing and modified vehicle body-over-frame with the crashing load absorber are prepared using Catia software. In the final phase, the proposed crashing load absorber concept is implemented in a high fidelity non-linear finite element model of a small passenger car in the explicit dynamics platform. In this case before proceeding through the FE explicit dynamics analysis of the existing and modified vehicle models, the static structural analysis of the crashing load absorber piston, crashing load absorber rear end supporter and crashing load absorber assembly is performed at design load. The analysis of the models in explicit dynamics with and without energy absorbing system, obtained at their impact speeds depending on the kinetic energy that produced in a real vehicle at 80 km/h, are compared with those of the baseline model to illustrate the crashworthiness enhancement and energy absorbing properties of the existing one and the proposed concept. The results shown the crashing load absorber piston, rear end supporter and the absorber assembly itself at 11274.65 N given the better result on deformation and stress relative to the material properties of structural steel. The existing vehicle and modified vehicle models analysis result shown at their initial velocity of 26.984 m/s and 26.011 m/s respectively, the total deformation of the existing vehicle model is much greater than the modified vehicle model of total deformation. Thus the vehicle crashworthiness can be greatly enhanced using the crashing load absorber system.

1.0 INTRODUCTION

1.1 Background of the study

Transportation is one of the basic requirements for the proper functioning of societies as its demand is highly related to the movement of people from one place to another. It is difficult to conceive of a situation where transport does not play a major role in the life of an individual. It is an accepted fact that of all modes of transportation, road transport is the nearest to people out of all other transportation systems. And its major advantage compared with others, its flexibility, which allows it to operate from door-to-door over short distance at the most competitive prices. In Africa over 80% of goods and peoples are transported by roads where as in Ethiopia road transport accounts for over 90% of all the inter-urban freight and passenger movements in the country. Therefore, transportation has a direct impact on the day-to-day activities of the people, especially in large cities where the distance to be traveled is too far to cover on foot or bicycle within a reasonable time. Cities in the developing nations are not only showing a rapid population growth, but also a change in their resident's way of life. This obviously implies that there is a need for a corresponding expansion of infrastructures and services. But due to inadequate road network, slow road construction and maintenance, rapid traffic growth, shortage of parking space in the narrow streets, as well as ineffective traffic management and enforcement, there is a rapid growth of road traffic accidents. The constraints associated with transport include the risk of traffic congestion, traffic accident, pollution, noise and the like. Serious injuries and mortality in the road collisions are a public health problem with consequences similar to those of major disease such as cancer and cardiovascular disease. Worldwide, about 1.2 million persons were killed on the roads and an additional 20-50 million were injured. Road traffic injuries were the 11th leading cause of death and accounted for 2.1% of all deaths. High income countries reduced fatalities from road traffic accidents by more than 25% during 1968-1998, and another drop of 30% will be recorded by 2020. While in low and middle income countries, where 81% of the world's population live and own about 20% of the world's vehicles, and the annual deaths and disabilities from road traffic accidents will raise considerably by 2020 [1].

Historically, styling and packaging considerations drove the design of vehicle body structures. The final design was the product of a long evolution guided primarily by testing, supported by simple linear strength of material methods. With the advances in computer hardware and software, several analytical design capabilities evolved, providing engineers

with a variety of tools to design modern vehicle structures that can meet the growing customer demands for better crashworthiness performance, quieter ride and reliability. These tools include simple spring-mass models, beam element models, hybrid models and finite element models. Although these tools vary in complexity, each is based on structural mechanics principles that require conservation of mass, momentum and energy. The selection of a particular analysis tool depends on the task at hand and on the particular design phase. For example, a simple beam model may be more appropriate at concept initiation to determine section geometry, size and thickness, whereas a full-scale finite element analysis of the vehicle may be more suitable when a prototype is built. Correspondingly, the time required to develop a model and the amount of information provided vary by model complexity. In recent years, the auto industry has experienced the greatest demand from customers, regulators, and the media to provide safer vehicles. This translates into better crashworthiness of the vehicle structure and effective restraint systems. Collapsing structure on the vehicle's deceleration-time history, and learning about the interaction between the occupants and the vehicle during crash are examples of the needed knowledge. Efficient methods and tools are necessary to enable initiation of a sound design at the early concept phases of development [2].

The road traffic accident on the vehicle occurs mostly in the form of head-on collision or rollover is a consequence of circumstances either internal or external constraints that produce abnormal operating conditions for the vehicle. Whether the collision occurs with another vehicle or with a stationary obstacle, it subjects the vehicle structure to forces and deformations. If the forces involved exceed the energy absorbing capability of the vehicle structure, occupants may be injured or killed. This early period focused on basic improvements such as adopting an all-steel body structure is better occupant protection. In addition, the vehicle structure crash tests can be made by using computer simulation or experimental work. These tests involved rollover simulations and car-to-barrier frontal impact. Vehicle safety improvements over the past seven decades have focused on crash avoidance technology, structural crashworthiness, and occupant protection devices. Nowadays, transportation safety efforts focus on crashworthiness, crash avoidance, driver performance, and highway construction too. Over the past decade automakers have added many features to help the driver avoid a crash, such as anti-lock braking systems, traction control devices and daytime running lamps. Vehicles also include many crashworthiness

features such as rigid steel occupant-cells surrounded by strategically placed, energy absorbing components [3].



Figure 1.1: Head-on bad collision situation of lighter vehicles [4]

In all types of crash accidents, frontal crashes cover about 30 % of the total number of accidents. The injury and lethality rate are also very high due to the external and internal problems on the vehicle. Most drivers are convinced that they can avoid such troublesome situations. However the statistics shows that ten thousand dead and hundreds of thousands to million wounded each year. Hence, improvement in the safety of automobiles is prerequisite to decrease the numbers of accidents. Automotive bumper system is one of the key systems in passenger as well as lorry cars. Bumper systems are designed to prevent or reduce physical damage to the front or rear ends of passenger motor vehicles in collision condition. It protects the hood, trunk, grill, fuel, exhaust and cooling system as well as safety related equipment such as parking lights, headlamps and taillights, etc. A good design of car bumper must provide safety for passengers and should have low weight. Different countries have different performance standards for bumpers. Under the International safety regulations originally developed as European standards and now adopted by most countries outside North America, a car's safety systems still function normally after a straight-on pendulum or moving-barrier impact of 4 km/h (2.5 mph) to the front and the rear, and to the front and rear corners of 2.5 km/h (1.6 mph) at 45.5 cm (18 in) above the ground with the

vehicle loaded or unloaded. In North America (FMSS) and Canada (CMVSS), it should be meet 4 kmph pendulum and barrier impacts [5].

The function of automotive bumpers has changed considerably over the past 70 years. The later performance is achieved by a combination of careful design, material selection to obtain a particular balance of stiffness, strength and energy absorption. Stiffness and Energy absorption are essential criterion. Stiffness is important because vehicle design consideration limits the packaging space for the bumper design to deform under load and Energy absorption, because bumper must limit the amount of the impact force transmitted to the surrounding rails and vehicle frame. Automotive bumper plays a very important role in absorbing impact energy (original purpose of safety) and styling stand point/aesthetic purpose. Now a days, automotive industry concentrates on optimization of weight and safety. Therefore, how to improve vehicle passive safety performance in the crash and how to avoid or alleviate injury and death of passengers in frontal crash to the maximum have become an important subject of vehicle passive safety performance research in many countries or in the world. In automobile design, crash analysis is the most important engineering processes in developing a high quality safety in the vehicle. Computer simulation technologies have greatly endorsed the safety, reliability and comfort of today's automobiles [6].

The primary concern for drivers and passengers is safety and come back to their residence without any disturbance. Designers and manufacturers have respond to this key concern and expectation with an increasing number of regulations. Although the details may vary slightly from country to country, the fundamental requirements are almost the same or similar. A vehicle is expected to provide adequate protection to driver and passengers in a catastrophe. To protect the occupants of a car, there are many new perceptible safety features such as airbags, antilock brake system, and traction control. A less tangible feature that cannot easily be seen by drivers and passengers is the crash response behavior. In a well-designed automobile, the car body and varies components have the protective layer for the occupant of the vehicle. They serve as the crumbling zone to absorb the energy of impact and next to crumbling zone towards the driver to or passenger there will be crashing load damper. In this area most automobiles have not bumper around the crumbling zone. In automobiles (passenger) and heavy vehicles bumper is used in front and rear of vehicle as safety purpose. It made in such a way that it takes maximum impact of vehicle and after

accident it goes towards chassis. The lot of research is done in this field on material basis. Most of the vehicle use bumper which made from steel, light material such as aluminum, and plastic and many more [7].

The regulatory and consumer crash test programs introduced over the last 15-30 years have enhanced safety and crashworthiness of cars and light trucks. The automakers have responded to these tests by developing innovative countermeasures to reduce risk of injuries to occupants in those crash modes. However, the crashes that occur in the real world are diverse and may not always be addressed by the current regulatory and consumer crash tests. This study investigates one such crash scenario that 91 fatal frontal crashes in Sweden involving belted occupants seated in the front row of a vehicle. In this study they defined a series of load paths, which act upon a vehicle during a collision. Evidence from real-world crash data was reviewed to establish the load paths associated with occupant fatalities in frontal collisions. For frontal crashes, a researcher found, "...small overlap crashes account for 48% of the fatalities." They identified that in approximately 60 percent of fatality cases, there was loading of the left side structure, left wheel, and left shotgun beam. These loading areas are essentially outside of the vehicle's longitudinal rails, although fatalities were mainly attributable to small overlap impacts located on the driver's side. As the researcher found that crashed vehicles were difficult to classify when the degree of longitudinal rail (frame) involvement was unknown. A study focused on the analysis of Narrow offset / overlap and Corner impacts through the NASS-CDS and the CIREN. The characteristics of vehicle damage are defined in NASS-CDS using the CDC as defined in the SAE Standard J224. In the Pintar study, narrow offset damaged vehicles were selected using the CDC, where the involved vehicle sustained frontal, corner impact type damage that was restricted to the left-most or right-most third of the end plane. They found that the extent of head, thoracic, and upper extremity trauma in CIREN was similar to that found in the NASS dataset of narrow offset frontal impacts. Analysis of CIREN occupants identified that over half of the involved occupants sustained serious or greater injuries to the lower extremities. It was noted that the change of velocity reported in the NASS-CDS did not appear to be well correlated with the severity of the narrow offset crash [8].

Now a days the bumper that mostly seen on vehicle is directly connected to chassis, so there is solid to solid contact between bumper and chassis, furthermore no space is kept between bonnet and bumper, all they make closed (fixed) contact. Vehicle that use bumper system that connected to chassis of vehicle with no gap between bonnet and bumper so

that after accident impact travel in linkage, and affect to bonnet, engine and finally goes to driver cabin and passenger. Currently technology has been change and research is going on impact load absorbing bumper system, it involves in automobile industry in 19th century with rigid and complex form.

In this study in addition to crumple zone there will be a slight analysis on the vehicle front structure with crashing load absorber to improve the efficiency of crashworthiness using computer simulation. Before testing crashworthiness of the frontal impact, there must be added crashing load absorber at two places of the vehicle front. It must be designed with the appropriate material and applied load on it. After designing this absorber, the 2D or 3D drawing is done and it will assemble inside the front part of the vehicle. In the methodology part tools for designing structures for crashworthiness are described with emphasis on frontal crashes. The reason for emphasis on designing for frontal crashes is that these are considered to be responsible for more traffic fatalities and injuries than any other crash mode.

1.2 Statement of the problem

Road traffic accident or vehicle crash is the major alarming issue in the world. This problem or issue in the world including our country Ethiopia making the citizens afraid of their journey. This is because during any vehicle crash or accident, there was at least one death or injury and lose of wealth. In developing countries the death rates per vehicle can be in the order of 50 times that of European countries [9]. Currently Ethiopia, one of the developing countries in the world, loses around 400 million Birr in each year due to road accident [10]. This does not include the socio economic costs associated with the accidents. This alarming accident rate is recorded as the third killing vector in the past years. Even those figures declared based on statistical data from the authorities showing a very hazardous situation; however there does not seem a sufficient work being done to reduce this alarming accident rate. The rate of traffic accidents in the world especially in Ethiopia goes up together with the increase of motor vehicles and population size. According to Fanuel S. [10], the pre-conditions that lead to vehicle accident are the followings:

- Shortage and low quality of transport services and facilities
- Poor quality of roads, pedestrian walkways
- High rate of congestion at peak hours and hence high rate of traffic accidents
- Lack of lane for each motor vehicles in their size and functionality

- Traffic road signs /signals invisibility and lack of regular maintenance
- Drivers and pedestrians behavior and knowledge on road safety
- Poor and traditional procedure of vehicle maintenance

Due to these existing problems that mentioned above, different types of motor vehicle accidents will be happened on the road. So there is a need to improve crashworthiness of the vehicles to reduce damage and injury caused due to collisions.

1.3 Objective of the study:

1.3.1 General objective

The general objective of this thesis work is to analyze the full frontal crashworthiness of four wheel light weight vehicles with and without crashing load absorber.

1.3.2 Specific objectives

- To design the crashing load absorber components such as
 - spring design
 - piston buckling design and analysis
- To model the sample vehicle frame with and without crashing load absorber using Catia software; Catia V5 R19
- To analyze the crashworthiness using ANSYS R15.0 software
 - static structural analysis
 - crashing load absorber rear end supporter deformation analysis
 - piston buckling analysis
 - crashing load absorber deformation analysis by applying point load at the front end of the absorber
 - explicit dynamics analysis
 - the existing vehicle model stress and deformation analysis
 - the modified vehicle model stress and deformation analysis
- finally to compare the crashworthiness of the structure with crashing load absorber and without crashing load absorber or compare the stress and deformation values on the existing and modified vehicle models

1.4 Delimitations of the study

To understand complete crashworthiness of the vehicle, the detail analysis in all directions is necessary. But in this work the analysis is delimited only frontal region of the vehicle. The

crashing load absorber is installed horizontally parallel to the chassis frame. It is tried to see the effectiveness of the installed absorber on crashworthiness of the vehicle in frontal collision. So in this work, it is mainly focused on the front region of the vehicle.

1.5 Limitations of the study

Shortage of time limited the work towards the frontal collision only. The collision analysis in all directions needs more time. Non availability of high capacity computers is another factor which limited the work. Analysis using ANSYS software with small size element and fine adjustment needs high capacity computers. Due to non-availability of computers with high processing capacity, the analysis is limited to big size of elements up to 10 mm with coarse adjustment.

1.6 Significance of the study

This work helps in improving the crashworthiness of the vehicle during frontal collisions. The drivers and passengers feel safe and enjoy the tension free drive or journey. It minimizes the injury of passengers and drivers by reducing the forward velocity of the passengers as well as the drivers. It also reduces the damage of vehicle front parts during collision. It helps the future researchers in their work.

1.7 Thesis Organization

This thesis consists of four chapters. The present chapter provides a general introductory part together with statement of the problem, objectives of the study, delimitation or scope, limitation of the study and significance of the study. The second chapter provides literature review of the most recent research works that are related to the field of study. In Chapter three, a crashing load absorber components are designed and modeled using CAD. The existing vehicle model is prepared using Catia software taking the dimensions directly from the automotive workshop. In addition to this, the modified vehicle model is also done using the crashing load absorber design specifications and existing vehicle model. In this chapter the general idea on the ANSYS software package is out lined. Chapter 4 contains the result and discussion portion which shows the ANSYS software results on the deformation and stress in static structural and explicit dynamics analysis of crashing load absorber, crashing load absorber piston, crashing load absorber rear end supporter, existing vehicle model and modified vehicle model. Finally conclusions, the most significant suggestions and the future works are briefly described in Chapter 5.

2.0 LITERATURE REVIEW

2.1 Crashworthiness

Crashworthiness is an engineering term used to define the ability of vehicle structure to protect its occupants during an impact/collision [11]. Crashworthiness is not limited to automobiles only, it is also applied to other transportation vehicles, such as ships, planes, and trains. In fact, the first systematic and scientific investigation of the subject was applied to railway axles between 1879 and 1890 by Thomas Andrews [12]. In other words, crashworthiness is the process of improving the crash performance of a structure by sacrificing it under impact for the purpose of protecting occupants from injuries [13]. To improve the structure design for crashworthiness, it is required to understand the different factors affecting the crash process. In the following, different fundamental aspects of design for crashworthiness have been described and pertinent works have been reviewed.

2.1.1 Crash Characteristics

Accidents occur in a random manner. An automobile can be impacted from any direction at different speeds. It can also include an automobile impacting another automobile, which in turn can be the same or different from the first automobile. This shows how automobiles affect and being affected by each other in crash situations. An automobile can also impact a rigid barrier, a tree, a light post . . . etc., which may lead to severe deceleration and high loads, as a rigid body can't deform to absorb part of the impact energy. An automobile can also impact a pedestrian which leads to the importance of design for pedestrian safety as well, and finally an automobile can go into rollover accidents. According to Galganski [14], crashworthiness problems can be characterized by:

1. Displacement and energy: Frontal structure length is being reduced by modern design styles and at the same time, it is required to absorb most of the impact energy and to minimize intrusion into the compartment.
2. Crash pulse: Crash pulse is the deceleration induced by impact on the human body. HIC is used to measure the damage from crash pulse on the brain, and it should be less than a certain limit by regulations [15].
3. Crash position: The structure should be able to mitigate injuries in different crash positions such as full frontal impact, offset frontal impact, side impact, rear impact, and rollover.

4. Automobile compatibility: With different automobile models, the structure should be able to mitigate injuries resulting from an accident involving two different automobiles, which can differ in size and/or weight.

2.1.2 Crash Types

2.1.2.1 Frontal crash or impact

Frontal impact can be realized in two stages: In the first stage, the vehicle strikes a barrier or another vehicle which causes front to end crush and the kinetic energy is dissipated into deformation of the structure. In the second stage, the occupant continues to move freely against the interior if not restrained, or interacts with restraint system, if restrained. The kinetic energy is then transformed into interior deformation of the structure and compression to the occupant's body. Finally, the remaining kinetic energy is dissipated as the occupant decelerates with the vehicle. Injuries may occur during the second stage in case the impact loading is high enough beyond the safety limits. A good design should ensure that the kinetic energy is dissipated gradually to minimize injuries. Safety measures, such as using energy absorbing materials to cover the interior parts, seat belts, and air bags are important in reducing injuries due to interaction between the occupant and the interior. To provide the minimum level of safety, automobile manufacturers are obliged by law to ensure that their designs comply with governmental regulations. Automobile manufacturers must demonstrate that their vehicles are in compliance with safety standards before they are sold. For frontal collisions, vehicle designs are regulated by FMVSS 208 in the U.S., by CMVSS 208 in Canada, and by ECE R-12 in Europe [16].

2.1.2.2 Side Impact

Side impact is the second most frequent mode of accidents. It is defined as the incident when a striking vehicle hits a target vehicle in the area of one or more of its pillars, during which the kinetic energy is transformed into deformation of both vehicles. The side impact can be described in two stages as has been described in frontal impact. However, in side impacts, restraint systems play less role in protecting the occupant due to the proximity of the occupant to the interior (doors). The stiffness of the side structure plays a major role in occupant's safety, i.e., it must be strong to resist intrusion. The cross bars are also important in controlling the deformation to protect the occupant. Finally, the inside layout is an important factor as well in minimizing injuries due to interaction between the occupant's

body and the vehicle interior. For side impacts, vehicle designs are regulated by FMVSS 214 in the U.S., by CMVSS 214 in Canada, and by ECE R-59 in Europe [17].

2.1.2.3 Rear Impact

Although the number of rear impacts is much less than the number of front or side impacts, the whiplash injuries caused by this types of accidents are complicated, hard to recover from, and can cause fatalities. For safety against rear impact, special attention must be given to the structural integrity of the fuel tank and fuel lines, seat resistance especially seat backs, and to head restraints. For rear impacts, vehicle designs are regulated by FMVSS 224 in the U.S., by CMVSS 224 in Canada, and by ECE R-42 in Europe [18].

2.1.2.4 Rollover impacts

In rollover impacts or accidents, the whole vehicle structure along with vehicle interior must protect occupants from injuries. Seat belts are particularly important in these accidents and the roof must provide resistance to deformation. During rollover accidents, the doors must not open by themselves and special valves must ensure that fuel lines are closed and fuel ventilation is directed to a charcoal canister to prevent fire hazards. The study of rollover is of special interest to heavy vehicles and trucks since their relatively higher center of gravity makes them more susceptible to this kind of accidents. This type of accidents can be prevented by controlling vehicle dynamic responses during driving and braking performance, as reported by El-Gindy. For rollover accidents, vehicle designs are regulated by FMVSS 201 and 208 in the U.S., by CMVSS 201 and 208 in Canada, and by ECE R-21 in Europe [19].

2.1.3 Structural Requirements

By definition, crashworthiness is the measure of the structure's ability to protect the occupants during impact, therefore the structure must fulfill some requirements that are summarized in the following:

- The structure must absorb as much impact energy as possible by plastically deforming in a controllable manner to minimize the remaining impact energy which can then be handled by the restraint system.
- The structure must preserve at least the minimum survival space to keep injury and fatality levels as low as possible.

A good design should achieve these two goals simultaneously and at the same time, it must adhere to other design objectives, e.g. comfortability, accessibility, weight, fuel consumption,

etc. Thus, the full design process is very complicated and requires high levels of communication between different disciplines. In order to achieve these goals, a good understanding of the structural deformation process and its mechanism should be maintained. Generally, the deformation (collapse) modes of the structure can be divided in two modes:

1. Axial collapse mode characterized by regular accordion type folding or irregular crumpling of the walls of the structure.
2. Bending collapse mode where discrete plastic hinges are formed and the structure collapses around them in a linkage type fashion.

Pure axial collapse as shown in Figure 2.1 is the most desirable collapse form, as it includes the absorption of the maximum amount of energy. An axial collapse, also called progressive buckling, involves formation of complete folds along the beam/tube. However, it is the most difficult to achieve and it can be realized only during a head on collision, direct front-rear accidents, or slightly off-angle ($5^\circ - 10^\circ$) impacts.



Figure 2.1: Axial collapse mode

On the other hand, the bending collapse mode is the most frequent to occur as it has the least energy path during an impact, and the structure will follow this path unless it is well designed to be forced into the axial collapse mode. As shown in Figure 2.2, this mode involves a global bending initiated by building up of stress concentration at a weak point until yield is exceeded and the structure bends around this point. Therefore, the design must not allow this building up of stress concentration and maintains a uniform deformation along the component length, which leads to the issue of the stability of the axial collapse.

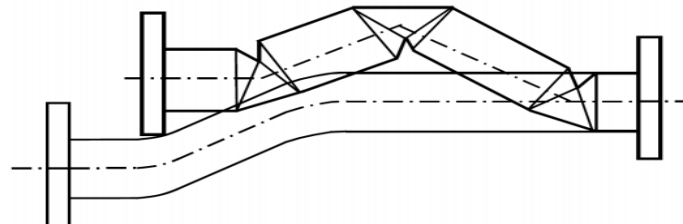


Figure 2.2: Bending collapse mode

Simulated crash-testing is being increasingly by various institutes to study the outcome of a vehicular in various situations under different conditions. The advantage of simulation is that the FE models can be reused again and again and also the user has the freedom to change any of the parameters of the test and also the user can vary the material properties as well as the type of material of the parts in the vehicle. The FE model was then used to simulate crash test. The FE software used here to carry out the simulation was Explicit Dynamics. One of the tests carried out was the Frontal crash at optimum/moderate speed. Before the simulation could be carried out, several other pre-processing conditions have to be specified. The test results were verified using results from existing bumper crash-test. Research works done so far related to this topic were discussed here under one by one.

A study was carried out by Cheg, where the aim of the study was to reverse engineer a 1997 Honda Accord DX sedan and to develop a FE model of the vehicle to be that can be successfully used in computational simulations of full frontal, side, and oblique car-to-car impact testing. The crashworthiness was then compared to existing physical data of a 2007 Jeep Wrangler that has been manufactured with all safety standards and technology. These comparison were made to evaluate the crashworthiness of the pre safety standards [20].

NGUYEN, depend on the IIHS, introduced the small-overlap frontal impact test in 2012 in order to consider one of the most serious crash scenarios and difficult engineering challenges. During the experiment, the vehicle hits a rigid wall at 64kmph with 25% overlap of the front vehicle on the driver's side. This was to simulate a vehicle colliding with a pole or flat object but only at the part of the left headlight. In small-overlap frontal crashes, the crash forces are applied outboard of the vehicle's longitudinal frame rails. In addition, forces are concentrated on the front suspension, at the left front wheel, and at the base of the A-pillar. These areas place the significant role in dissipating force during offset impact. Thus to Optimize the crashing level of the vehicle , the researcher put their main intrusion structure in six major variables such as: outer A pillar (x1), A-pillar reinforcement (x2), inner A-pillar (x3), door frame (x4), rocker panel (x5), and lower hinge pillar (x6), which were the values of overlap length reduced from 25% to 20% [21].

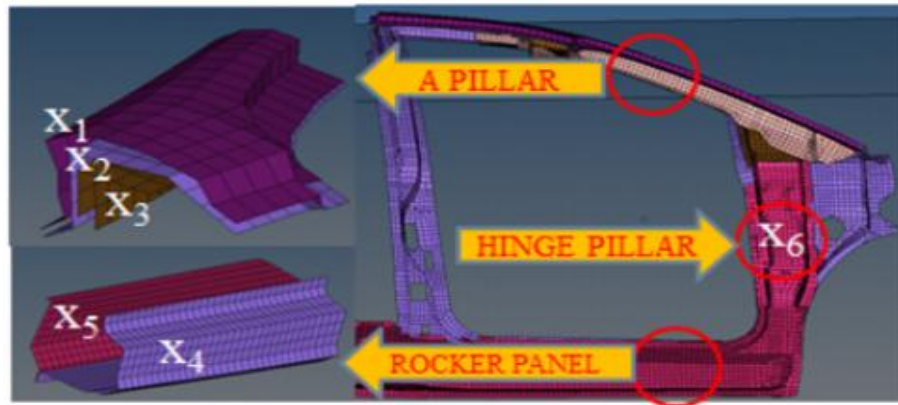


Figure 2.3: Six major structural variable to reduce overlap size

In recent studies on small-overlap impacts, found that the rails absorbed over 50% of the crash energy in the IIHS test. Finite element analysis was performed to characterize the frontal pole impact compared to the full frontal rigid barrier test and the IIHS 40% offset frontal impact test. The analysis investigated the distribution of the crash energy absorption by the front-end structure. Two vehicles that have different cross-section of the rails were simulated to compare the efficiency of crash energy absorption by different types of rails. His research was conducted depend on the material property on the existing vehicle [22].

Sherwood, the barrier model was designed following the real small-overlap crash test of IIHS which are used in SOFI simulations. The purpose of this study was to compare the overall ratings for vehicle structural performance in the event of a SOFI on the basis of crash test simulations. A thickness optimization method is proposed to improve a vehicle's structural resistance to a SOFI. The thicknesses of the A-pillar, rocker panel, and hinge pillar will be changed to reinforce the vehicle structure. The optimized designs with the best crashworthiness intrusion results were chosen for comparison with the original model [23].

The researcher Park suggested that the offset-frontal crash test can be used to complement the full-frontal crash test. He and his peer was done by considering the vehicle with Dummy. Frontal-offset and full-frontal testing was conducted using both the mid-size (50th percentile male Hybrid III) and the small stature (5th percentile female Hybrid III) dummies. Five vehicle models were used in the testing: Dodge Neon, Toyota Camry, Ford Taurus, Chevrolet Venture and Ford Contour. In the crash tests, all dummies were restrained with the available safety belt systems and frontal air bags. Both dummy sizes often exceeded the allowable lower leg injury criteria both in the offset tests and the full-frontal tests. Lower leg injury occurrences in real world crashes, based on National Automotive Sampling System data, showed that many lower leg injuries occur in both full-frontal and offset type crashes. The

results also showed that vehicles with a stiffer occupant cage did well in the offset test. In addition, the test data suggested that the small female dummy may be exposed to a high risk of neck injury if positioned close to the deploying air bag. Thus the researcher focused area was comparing the frontal crash accident leads to a serious injuries either in male or female driver by representing with Dummy [24].

Halloway proposed classifiers to indicate the number of longitudinal members. The purpose of the study was to distinguish the role of vehicle structure in frontal impacts in published coded NASS-CDS data. The criteria used: CDC coding rules, crush profile locator data and the projected location of longitudinal structural members in models of vehicle class sizes used by NASS-CDS. Two classifiers were developed to augment the CDC system. The Coincidence classifier indicated the relationship between the quadrant of the clock face the crash vector originates in and the aspect of the end plane the center of damage was located. It has three values: Linear (12 o'clock impacts) Consistent and Variant ("oblique" PDOFs) [25].

Hallquist, in other study, a vehicle computer model will be applied to real IIHS crash test using LS-DYNA software for the crash test simulations. As automobiles, this used to be simply for transportation, have become essentials for well-being and leisure life, the number of automobiles has increased with every year. When selecting an automobile, consumers take account of its design, performance, convenience and safety. Among these factors, safety is regarded as important by manufacturers as well as by consumers. Accordingly, the NCAP conducts various car safety tests in every country where the program is run. Such tests include the frontal and side crash test; rear-end collision test and rollover safety test. Tests are done to assess the crash safety of automobiles, with the scores of each model disclosed publicly to induce manufacturers to design and produce safer automobiles [26].

In this paper, Adams was used to conduct the multi-body dynamics analysis. The basic model that the researcher was used an existing vehicle, the stiffness of the upper body was changed, and the effects were examined. The results of conducting a parametric study while changing the stiffness of the members of the upper body by reinforcing the B-pillar (B), A-pillar (A), FRB, RRB, and RSR. As the SWR value obtained from the static roof crush resistance test increases, the displacement of A-pillar top-end in the CRIS test tends to decrease. The A-pillar had a very sizeable effect on the displacement of the roof in the dynamic test, and the effects of the front roof-bow and the rear side-rail must be taken into

consideration. It must be kept in mind that it was completely different than the effect of each member on the roof crush resistance. The DOE confirmed that the roof crush resistance SWR value and the displacement of A-pillar top-end in a dynamic rollover test were highly correlated. However, if the stiffness of the A-pillar, which had a sizeable effect on the displacement of the roof in the dynamic test, was inappropriate, the dynamic roof crush resistance was reduced even though the roof crush resistance is excellent. In conclusion, the NHTSA's standard requiring that the static roof crush resistance be SWR 3.0 or higher was shown to be the minimum safety standard necessary to secure the survival space. It was desirable that the minimum requirement for protecting passengers in real rollover accidents is greater than SWR 4.0 as required by IIHS. However, additional studies are needed to identify if the current result would hold when the pitch and yaw angle was changed [27].

When a car impacted with a barrier, the structure of the car was deformed in a characteristic way. The vehicle's kinetic energy was transformed into energy of deformation. The impact triggered the airbag and the occupant began its movement forward. When a seat belt was used, the occupant movement is restrained. In about 30 milliseconds, the air bag deployment will reach its maximum level and will help reduce the kinetic energy of the person. As this happens, the vehicle continues its deformation as the structure protects passengers from dangerous intrusions in the cabin. The severity of a crash can be measured by the damage caused to its occupants. The head injuries are the leading cause of death and disability in humans, and about half of the head injuries are caused by car accidents. In some of these works, the approach followed was to modify the design of a component (or several) and measure changes in energy absorbed or developed peak force when subjected to impact. However, these techniques were difficult to apply for the case of crashworthiness analysis, because they require a considerable number of variables. Under such circumstances, the total processing time can be very high. Other authors have used genetic algorithms combined with response surface methods. Table 2.1 compares several of these studies.

Table 2.1: Optimization studies for crashworthiness, application of genetic algorithms and response surface methods.

Reference	Optimization methodology	Number of runs	Number of design variables	Nature of design variables	Number of objective variables	Nature of objective functions
Chen and Li, 2009	Response surface method	136	8	Thickness	1	Acceleration peak of B-pillar
Hilmann et al., 2007	Monte Carlo + genetic algorithm	>100	15	Thickness and bumper section parameters	1	Plastic strain (minimize)
Hamza and Saitou, 2005	DOE + genetic algorithm	54	18	Sectional height and width (4), thicknesses (14)	2	Passenger compartment intrusion, maximum passenger point acceleration
Fang et al., 2005	Response surface method	27	10	Thickness	3	Energy absorbed at 20ms and 40ms and peak acceleration at engine top
Wu and Xin, 2009	Response surface method	12	3	Sectional height and width and angle of S-rail	1	Internal energy

An important conclusion can be derived from table 2.1: with more design variables, a major number of runs is required. Cao prepared a review that supports the division of an optimal deceleration pulse in 3 phases which is also explained by Witteman. These 3 phases are defined as follows:

1. In this first stage the collision is produced and the passenger starts his/her movement forward. Because there is no impact between the passenger and the interior of the vehicle during this phase, high deceleration of the vehicle can be allowed.
2. The airbag expands and contact with the passenger begins. During this stage the vehicle deceleration should be kept low to reduce the amount of damage.
3. Contact between passenger and airbag is fully developed. The airbag must perform its task and prevent potentially dangerous impacts and high decelerations in the passenger's head and chest.

The deceleration of the vehicle during this phase can be moderately high.

Witte man obtained the optimized deceleration pulses at 32, 56 and 64 km/h. According to his work, for a crash at 64km/h, for phase 1 (first 18 cm of deformation), a deceleration of 45g's is desirable. For phase 2 (frontal deformation between 18 cm and 59 cm), a deceleration of 9g's is preferable. For phase 3 (a length of 59 cm of deformation to 76 cm), a deceleration of 23g's is desirable. The duration of each phase is almost constant regardless of the initial velocity, and corresponds to between 10 and 30 m/s for the first phase and approximately 35 m/s for the second phase. This will limit the analysis as well as reduce simulation time and number of variables to modify. The decomposition time can be based on the phases explained before. The delimitation in space can be achieved by the choice of certain components that are important in the observed crash mode.

In the work of Cao an experimental design was conducted to identify the parameters of the restraint system (airbag and seat belt) that have the most important effects from the perspective of occupant damage in a frontal collision. Their results suggest that the force on the seat belt retractor and the area of the vent hole in the air bag are the most important parameters. Also, they proposed an optimization of the parameters of the restraint system along with the deceleration pulse as a design variable. This methodology differs from the "traditional approach" which first makes a structure as efficient as possible from the perspective of energy absorbed and then, the resultant pulse is introduced as an input in the restraint system optimization. This work builds on the procedure proposed by Cao, who combined design of experiments with optimization of restraint system parameters along using the deceleration pulse. This work goes further by including the deceleration pulse among the design variables and quantifying the improvements obtained in the optimization of the resulting HIC15. The pulse divided into three or more phases of constant deceleration, represents a simplification of the pulses obtained in real life. However, the works cited previously do not provide a measure of the error or deviations incurred by making this simplification. The present study tests the simplifications of the deceleration pulse and analyzes its effect on HIC15. The 40% offset crash test is an important test included in FMVSS 208 and used by the Euroncap and the IIHS. However, some documents state that in the full frontal crash test major damage is produced by head and chest decelerations and this test is the best to prove restraint systems. For this reason, this work focuses in the existing material frontal crash test only [28].

Alen John had done the paper on bumper system of ambassador car. It was modeled by using CATIA V5 R18. Further static analysis was done by using FEM package of ABAQUS

6.10. And for dynamic he used ANSYS Workbench 11.0. the material used here such as Aluminum B390 alloy, carbon composite and chromium coated mild steel. For mention three material he did static, dynamic and impact analysis. The author selected best material out of the mention material above. And it applies for his bumper system, for analysis of these material different boundary conditions are set for static, dynamic and impact analysis. Experimentation of these materials was done by considering speed and forces of impact at different boundary conditions act on bumper of vehicle. The isometric model of bumper figure 2.4, on which different tests are performed to select best material. Results were drawn in different parameters such as after impact analysis, under loading condition. Thus depending on the above parameters and values of deformation and von misses stress value, carbon composite material was selected. Material was selected because of fulfilling the requirement such as high strength to weight ratio, crashworthiness, high stiffness to weight ratio. Material study has been done but new design does not take in to account which result in solid to solid contact between bumper and bonnet which result in no energy absorption and sudden failure [29].

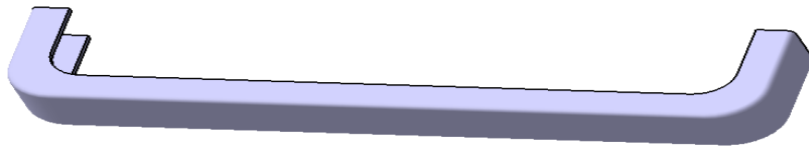


Figure 2.4: Isometric model of bumper

Jacobson also described the significance and developments of two decades of car design for applications to severe frontal collisions with partial overlap. Frontal SPOC also called small overlap crashes pose special challenges with respect to structural design as well as occupant protection. Important design principles include front side members orientated along the wheel envelopes together with a strong support structure utilizing a space frame principle with beams loaded mainly in tension and compression. Among the design principles are multiple front side members on each side, helping energy absorption efficiency and robustness. The upper side members absorb energy and also transfer the forces via the A-pillars into the roof and the reinforced body and door structures. The lower side members are connected by a cross member with balanced design combining strength and energy absorption using truss principles. The overall design is made to address loads in multiple directions as well as mitigate front wheel intrusion into the passenger compartment. Real world data, complemented with laboratory testing, show that the improvements have substantial effect on occupant protection, reducing overall injuries over the years and

reducing foot-well intrusion related injuries specifically. Thus the researcher focusses on the frontal impact crashworthiness with adding a cross member bar at the front of the vehicle without separating the bumper with main frame [30].

Zhida sen had done work on new design of longitudinal beam for front of the car, a frontal crash FEA model is done using CMVDR-294, and simulation for the same using LS-DYNA. In this paper they have done work on 100% crash simulation, and 40% ODB crash simulation for front of the car, but key focus of this paper on side impact of car. After accident simulation is done for front only but side parameters such as side door, pillar roof, door frame driver seat not take in to account, in this paper they work for the same parameters. For that he divided 3 division crash energy absorption areas, front, middle and rear. Longitudinal beam, the part who takes maximum impact after accident, so he did deformation study at certain forces on longitudinal beam. Finally result showed that improvement in longitudinal beam will improve car safety, and the same area he suggested for future work. In present paper he worked on transfer of forces to different areas of vehicle, and related calculations were done. But the key area that is bumper of vehicle is didn't study on this research. The force will be transferred first to the bumper and then to other parts. So the bumper design is much important to reduce transmission of forces [31].

S.Prabhakaran had done work on composite bumper which was applicable to light passenger vehicle, i.e. totally focused on materials of existing bumper, further fabrication was also done by using hand layup process. In this method they used E-glass epoxy bidirectional laminate and glass fiber. The focus of paper based on new design, material and cost saving. The thickness of the bumper according to material properties was calculated theoretically. Bumper was made of composite and was tested by Charpoy test. Results are good compared to steel material. Experiment was done on same area (40 mm^2) impact value and impact energy are 294 J and 74 J/mm^2 . Further analysis was done on composite and steel bumper by using FEA. Results showed that reduction in weight and cost by 53.78% and 77.22%. Impact strength is good for composite that is 7.25 compared to steel 3.25. Maximum stress produced on the material is 369.168 N/mm^2 , compared to steel 142.471 N/mm^2 . In case of factor of safety, it is 3.4 compared to steel with 1.2. So alternately there was weight reduction of bumper and less weight improved fuel efficiency of vehicle. The bumper model analysis was very old design, not advanced one. No impact

absorbing material was added to bumper, which made solid to solid contact with vehicle [32].

Mohit Tomar had done work on new design of bumper system in order to minimize the collision force which produces by other car. Test results were found by using simulation process (Autodesk). As compared to conventional bumper he added shock absorber to the front bumper in order to reduce impact. The load of 11240.450 lb was applied on the bumper and test results are drawn in table 2.2. Results showed improvement with that small change of adding shock absorber to bumper in order to absorb shock during impact. Results are good for all the stresses which are mentioned in the table 2.2. So in the above research work a detail design basic study was done but type of materials and weight were not considered. Changes were done in only in the frame (chassis). The forces were applied on the bumper only. Bonnet area which is the force transfer medium was not taken in to account. Force transfer to different areas after collision were not considered [33].

Table 2.2: Conventional and modified test results

Type	Applied load	Von misses	1 st principal stress	3 rd principal stress
Conventional design	11240.45lb	118.4 Max	32.71 Max	2.1Max
Modified Design	11240.45lb	235.8 Max	65.09Max	4Max

Nitin S. Motgi had done work on impact analysis of front bumper. For that he designed a new bumper system and model was analyzed using CAD/CAE software as shown in fig. 2.5. Stress and deformation results were found for aluminum and composite bumper fascia. Internal energy indicates energy absorption capacity of bumper. From results he concluded that aluminum fascia had poor absorption capacity of impact while composite bumper fascia showed better results than aluminum. Figure 2.6 (a) and (b) shows that internal energy absorption in case of aluminum and composite bumper fascia [34].

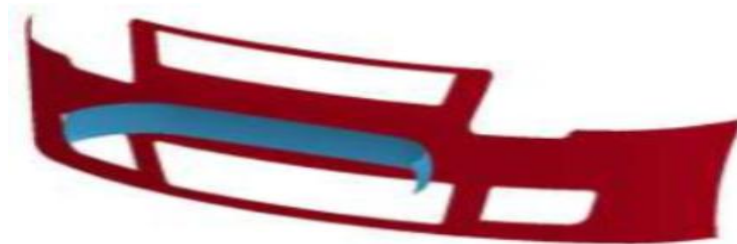


Figure 2.5: CAD model of bumper system

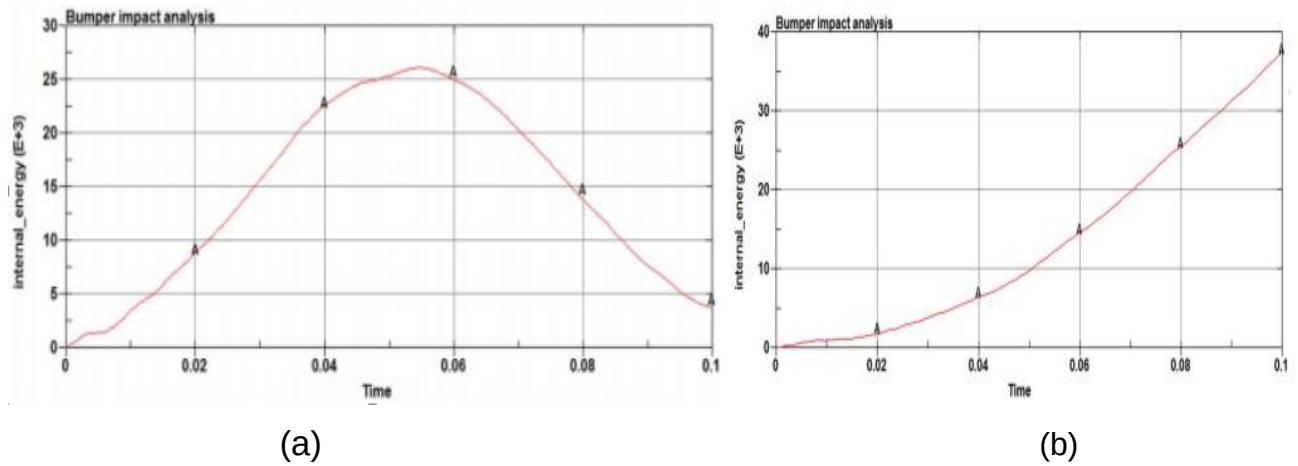


Figure 2.6: a) Aluminum bumper fascia, b) Composite bumper fascia

Pradeep Kumar Uddandapu had done work on bumper system of vehicle by considering variables like material, structure and shape as shown in fig. 2.7. Impact condition was studied for analysis of bumper beam to improve crashworthiness during collision. More priority was given to selection of material, and he performed analysis for various speeds according to their regulation, and finally came up with certain conclusions. The material property of ABS and PEI plastic were good and they could take over conventional material that used such as steel fiber. The material properties of ABS Plastic and PEI are given in table 2.3.

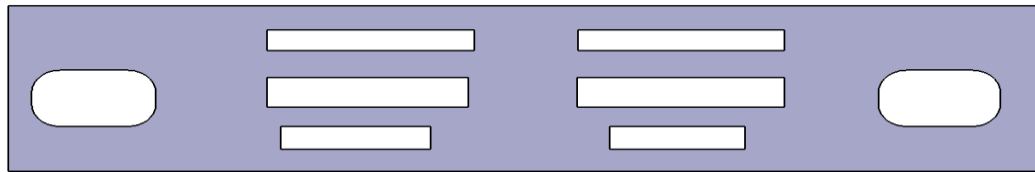


Figure 2.7: CAD model of bumper system

TABLE 2.3: Material property for ABS Plastic and PEI

Parameter	ABS plastic	PEI
Model type	Linear elastic isotropic	Linear elastic isotropic
Tensile strength	$3 \times 10^7 \text{ N/m}^2$	$2.41 \times 10^8 \text{ N/m}^2$
Yield strength	-	$2.41 \times 10^8 \text{ N/m}^2$
Elastic modulus	$3.38 \times 10^7 \text{ N/m}^2$	$3.38 \times 10^7 \text{ N/m}^2$
Poisson's ratio	0.394	0.4
Mass density	1020 kg/m^3	1480 kg/m^3
Shear modulus	$3.189 \times 10^8 \text{ N/m}^2$	$3.189 \times 10^8 \text{ N/m}^2$

Result showed that modelling and impact analysis is done by using PRO-E software, analysis was carried out at different speeds followed by stress, displacement and strain, as compared to steel property it was found that ABS and PEI are good material, but no shock

absorbing material take in to consideration. So the problem of transfer of forces in a linkage remain as it is [35].

Galal A. Hassaan had done work on bumper system of vehicle by introducing spring damper system. It is applicable for both front and rear of vehicle. Initial parameters are considered for bumper such as 100 mm dynamic motion. Results were obtained by considering vehicle colliding with other vehicle at different speeds. With the results obtained, he tried to show the plot of vehicle buffer time response and arbitrary buffer parameter. On that basis, he found graph between vehicle buffer time responses to arbitrary buffer parameter as shown in fig. 2.8. MATLAB code is written in order to get the result at different speeds and masses. Exact optimal design of buffer was presented in this paper by considering active and semi active approach. Exact design of bumper was not taken in to consideration. Application of it on vehicle was not optimal and it could not avoid the transfer of impact from bumper to bonnet. So it reduces the impact energy but at higher speed it definitely damage bonnet, engine and transmission parts of vehicle [36].

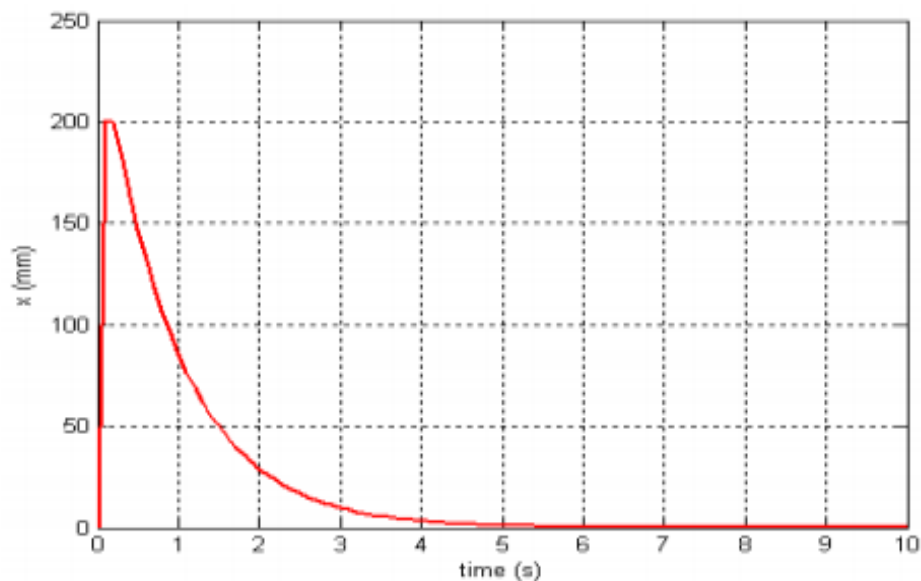


Figure 2.8: Vehicle-Buffer Time Response to Arbitrary Buffer Parameters

Aman N. Charkha had done research on bumper material. He replaced old bumper made up of steel by Composite bumper system. He concentrated on weight reduction. He used fiber reinforced composite material that resulted in to less cost and 30% weight reduction. They added new composite structure material such as glass chopped strand mat, core mat and epoxy resin to absorb most kinetic energy. From the analysis it was observed that composite impact bumper beam was most suitable over steel impact beam. The cost and

weight of composite bumper were low compared to steel. Impact and equivalent strength was also good. So this research was done in material point of view, but no any study was done on new design or structure of bumper [37].

In Hosseinzadeh RM research work mentioned that bumper beams were one of the main structures of passenger cars that protect them from front and rear collisions. In this paper, a commercial front bumper beam made of GMT is studied and characterized by impact modeling using LS-DYNA ANSYS 5.7. Three main design factors for this structure: shape, material and impact conditions are studied and the results are compared with conventional metals like steel and aluminum. Finally the aforementioned factors are characterized by proposing a high strength SMC bumper instead of the current GMT [38].

In this paper, Marzbanra, discussed the most important parameters including material, thickness, shape and impact condition for design and analysis of an automotive front bumper beam to improve the crashworthiness analysis in low-velocity impact. The simulation of original bumper under conditions of impact was according to the low-speed standard of automotive stated in ECE United Nations Agreement Regulation number 42, 1994. In this research, a front bumper beam made of three materials: aluminum, GMT and high-strength SMC was studied by impact modeling to determine the deflection, impact force, stress distribution and energy-absorption behavior. The mentioned characteristics were compared to each other to find best choice of material, shape and thickness. The results showed that a modified SMC bumper beam could minimize the bumper beam deflection, impact force and stress distribution and also maximize the elastic strain energy. In addition, the effect of passengers in the impact behavior was examined [39].

A bumper is a car shield made of steel, aluminum, rubber, or plastic that is mounted on the front and rear of a passenger car. When a low speed collision occurs, the bumper system absorbs the shock to prevent or reduce damage to the car. The car bumper was designed to prevent or reduce physical damage to the front and rear ends of passenger motor vehicles. Generally, a bumper is attached to either end of a vehicle to absorb impact in a collision, thereby protecting passenger. As shown in Figure 2.9, a conventional bumper system comprises a bumper cover 1 defining an outer appearance of the bumper system, an energy absorber 2 formed of an elastic material such a polypropylene foam body or an urethane foam body to absorb energy, an impact beam for supporting the energy absorber 2, and a stay 4 for connecting the impact beam 3 to a vehicle body [40].

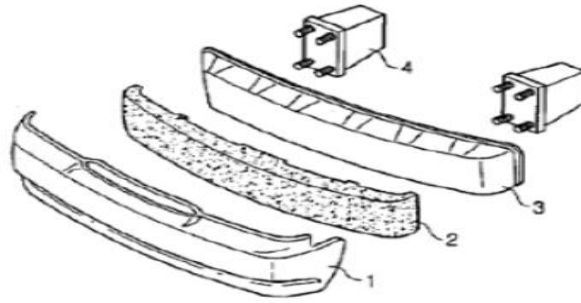


Figure 2.9: Conventional bumper cover

Andersson R. disclosed was a bumper system including a bumper cover, an energy absorber formed of a synthetic resin material through a foam molding process, an impact beam for supporting the energy absorber, the impact beam being formed of a glass mat thermoplastic sand having a "C-shaped" section, and a stay for connecting the impact beam to a vehicle body. Tip insertion grooves in which the tips are inserted are formed on an inner surface of the energy absorber, and a pressure receiving surface corresponding to the web portion is formed on the inner surface of the energy absorber [41].

Butler M. focused on increasing crash performance in automotive vehicles. It was necessary to use new techniques and materials. Components linked to crash safety should transmit or absorb energy. The energy absorbing capability of a specific component was a combination of geometry and material properties. For these components the chosen material should have high yield strength and relatively high elongation to fracture. These demands have led to increasing interest in the use of high strength stainless steels [42].

Carley ME done the research work on this area and the objective of this study was to design efficient epoxy structural foam reinforcements to improve the energy absorption of front and rear automotive bumper beams. Three bumper structural performance criteria were studied in this work [43].

Evans D and Morgan T as vehicle manufacturers continued to became more aggressive with the styling of new vehicles, bumper system technologies was required to find new solutions that fit into the reduced package spaces while continuing to meet the vehicle performance and cost requirements. The purpose of this paper was to introduce new and innovative EPP foam technologies and techniques [44].

Witteman WJ who was automotive styling trends point to reduced bumper overhang, greater sweeps, and reduced overall package space for the bumper system. This paper will review the industry trends associated with bumper energy absorbers and explore the potential fit of this new prototype energy absorber design as an alternative to EPP foam. Also he has

included a review of the simulated performance of the prototype ETP energy absorber and a comparison of its actual test results for 8kmph FMVSS Part 581 impact series to the performance of EPP foam packaged in the same environment [45].

Masoumi A. and his peers, in their thesis described the design of a new frontal vehicle structure that directs the asymmetric crash load of an offset collision as an axial load to the second unloaded longitudinal member. Only by using both longitudinal members and through a progressive folding pattern, enough energy can be absorbed in the front structure to prevent a deformation of the passenger compartment. To prevent a premature bending collapse, the new longitudinal members consist of two functional components: an inside square crushing column for a normal stable axial force level and a stiff outside sliding supporting structure that gives the necessary extra bending resistance. An integrated cable system transmits the force to the other longitudinal member. With this novel design concept, a vehicle has similar energy absorption in the front structure for the entire range of collision situations (full, offset, oblique) [46].

Zonghua Z. Discussed that material selection for automotive closures was influenced by different factors such as cost, weight and structural performance. Among closures, the automotive bonnet must fulfill the requirements of pedestrian safety which is evaluated by child and adult head from impacts. The mechanisms of injury were complex, therefore; the HIC which shows a measure of the likelihood of head injury arising from an impact was developed. HIC includes the effects of head acceleration and the duration of the acceleration. In this paper a new finite element model has been developed which was capable to simulate head impact phenomenon between head form impacts and composite bonnet. Then the behavior of three identical bonnets made of steel, aluminum and composite have been investigated by the developed model [47].

In this paper, O. G. Lademo, discussed a rib-reinforced thin-walled hollow tube-like beam (named as rib-reinforced beam) is presented for potential application in vehicle bumper. Through numerical simulation of the bending behavior under impact loads, the rib-reinforced beam was compared with thin-walled hollow tube-like beams filled with and without foam materials (empty beam and foam filled beam) in crashworthiness. The effects of the shape of the reinforced rib are investigated and the shape optimization design is performed for increasing energy absorption and reducing the initial peak force. A multi-objective crashworthiness optimization formulation including maximum energy absorption, maximum

specific energy absorption and minimum initial peak force is constructed based on the IPM. The optimum configuration of the reinforced rib is given with a normalized cubic spline function. Numerical example results show that, compared with the empty and foam-filled beams with same weights, the optimized rib-reinforced beam has higher energy absorption performance and lower initial crash force. It is found that for the rib reinforced beam little rumple is formed around the compressed indentation, which helps to retard the collapse of the side wall and means more energy absorption [48].

Nitin S. Manufacturing of a bumper system from aluminum extrusions often involves series of forming operations performed in the soft W-temper condition, and then artificially age-hardening of the components to the material's peak hardness T6 condition. It is probable that proper FE modeling of the crash performance of the resulting systems must rely upon a geometry obtained from an FE model following the process route, i.e., including simulation of all major forming operations. The forming operations also result in an inhomogeneous evolution of some internal variables (among others the effective plastic strain) within the shaped components. Results from tensile tests reveal that plastic straining in W-temper leads to a significant change of the T6 work-hardening curves. In addition, the tests show that the plastic pre-deformation causes a reduction of the elongation of the T6 specimens. In the present work, these process effects have been included in a user-defined elastoplastic constitutive model in LS-DYNA incorporating a state-of-the-art anisotropic yield criterion, the associated flow rule and a non-linear isotropic work hardening rule as well as some ductile fracture criteria [49].

CHEN and his corporate authors worked on the improvement of cab crashworthiness for front pillar impact test and structure improvement for side 20° pendulum impact test. They have observed the deformation of the various parts of the cab that can be found on it. The main and first one was the A pillar, which directly contacts with the pendulum. From their software simulation test and direct test, A-pillar was collapsed, the maximum dynamic deformation in X direction was 288.9mm in the middle of the A-pillars. They have inserted the structure foam in the middle of A-pillar which contacts with the pendulum in order to improve crashworthiness. The second part needs to be improved was the rear wall horizontal beam which has the 307.3mm maximum dynamic deformation in X direction. A reinforcing plate was added in the horizontal beam covering the serious deformed part. The third part needs to be improved was the connecting part between the B-pillar and the floor

beams. Its collapse initiates an inhomogeneous distribution of deformation on the rear wall, therefore inducing the 295.9 mm maximum dynamic deformation of the rear wall in X-direction. The other one that they have used to improve in the side 20° pendulum impact test was done on the top of left side panel. In this test they have got 63.1 mm maximum deformation in the lateral direction. To minimize this deformation value, they had inserted the structure foam in side panel between the inner and outer panels to help the inner panel absorb impact energy. Thus their work depends on the thickness of the material and type to strengthen [50].

Design, fabrication and testing of steel and composite bumper (using glass fiber material) were completed and also composite bumper was analyzed and compared with steel bumper. Cost of composite bumper was also Rs. 820/- which was 80% less than steel bumper. Impact strength of composite bumper was 7.35 J/mm² where steel bumper is 3.25 J/mm². The existing and composite bumper were analyzed in ANSYS10.0 and the Maximum stress induced in the composite bumper was 159.36 N/mm² where steel is 292.669 N/mm². Factor of safety for composite bumper is increased by 64%. From the study, it was concluded that the researcher focused on the material properties and made their conclusion for bumper which one of the material is the best depend on the stress values comes from FEA. The results obtained were described in table 2.4 below [51].

Table 2.4: Results of finite element analysis

Description	Steel bumper	Composite bumper
Max. stress (N/mm ²)	369.168	142.471
Min. stress (N/mm ²)	0.003924	0.000302
FS	1.2	3.4

This paper tried to show the three areas: front, middle and rear energy absorption areas, as shown in Figure 2.10 (a) below, in which, the front energy absorption area is mainly made up of front bumper beam, bumper beam buffer block, engine hood front end and crash box. Such components were made of high performance plates, which can absorb the energy produced in the impact as much as possible through their deformation, and also continued to split the energy to the left and the right by using the force borne by the structure and transfer the energy rearward. The rear energy absorption area is composed mainly of the

driver's cab that is both strong and rigid. In design of rear energy absorption area, deformation that may cause injury to the passengers must be avoided as much as possible so as to guarantee the passengers' safety by reducing the intrusion into the front floor and the crash speed as well as the acceleration of frontal crash. Crash force transmission routes: the front, middle and rear energy absorption areas were the basis of multiple force transmission routes in frontal crash. Setting of multiple force transmission routes in frontal crash for frontal crash may effectively absorb energy and transmit crash force. Multiple force transmission routes in frontal crash were generally divided into three layers as shown in Figure 2.11(b) below, which are typical force transmission routes in the course of frontal crash. Finally they have made their modification on the basis of existing structure, in order to reinforce the bending rigidity at this position, two 20 mm stiffeners were added in the weak position in the middle of the right longitudinal beams. The paper did not show any intermediate damper thing between the bumper and front part of the car [52].

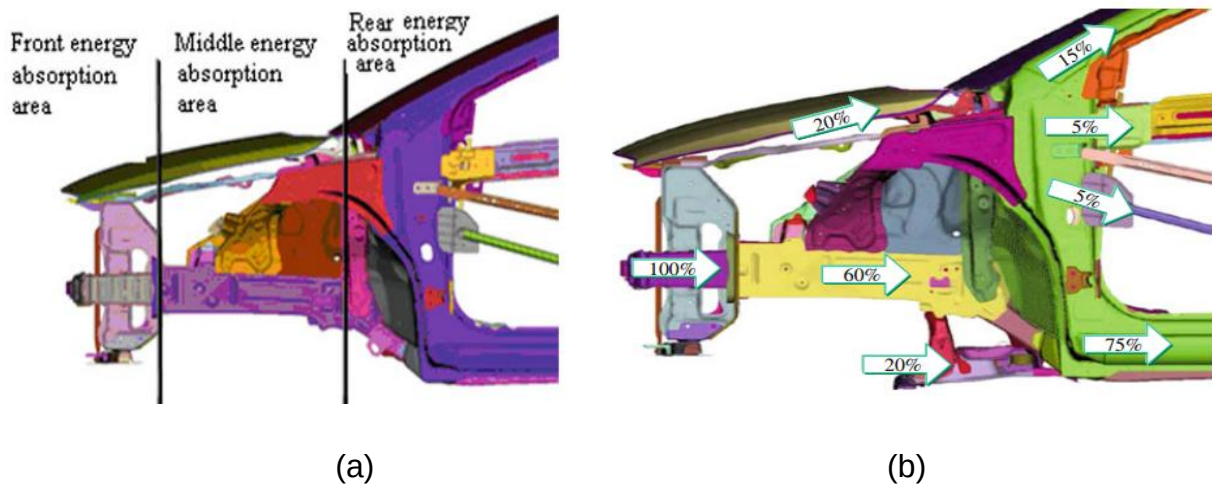


Figure 2.10: (a) Energy distribution areas, (b) Force transmission routes during collision

The third stage of reverse engineering was model validation. Initially, the model was divided into several major components. These included the engine cradle, the front bulkhead, the A-pillar, the B-pillar, the firewall, the floor, and the rear end. This separation into components allowed a smaller model to be used during the assembly phase, since each component required significantly less computer resources than the entire automobile. This parsing also allowed the entire model to be meshed and assembled on personal computers. To aid in the verification process, each subcomponent was subjected to a simulated 35 mph impact into a rigid wall. These simulations allowed the stability of each component to be tested and modified if necessary. Two components, the front driver's side door and the steel door

beams inside were selected for component testing. A large (100 ton) tension compression test machine was selected to load these components in three point bending. High strength steel reinforcing beams (approximately 32 mm in diameter with a wall thickness of 3 mm) are built horizontally inside each door to satisfy the FMVSS 214 Side Impact Protection requirements. Each beam was anchored at the door latch and door hinge assemblies. Quasi-static three point bending tests were conducted on several beams. A -45° , 0° and 45° strain gage rosette was placed on the tensile side of the beam. These tests were then used to estimate the elastic modulus for the high strength steel as 208 GPa. Results of several full-vehicle crash tests were available to the authors. Insight into the physical crash behavior of the car is helpful in making a comparison and interpretation of the FE analysis Figure 2.12. Load vs. displacement comparison for high strength door beam. For a comparison, a primary quantity of interest was the velocity of certain locations in the structure. Although elemental strain and stress are of interest, they are of questionable reliability since in the FE analysis they are derived from severely under-integrated elements.

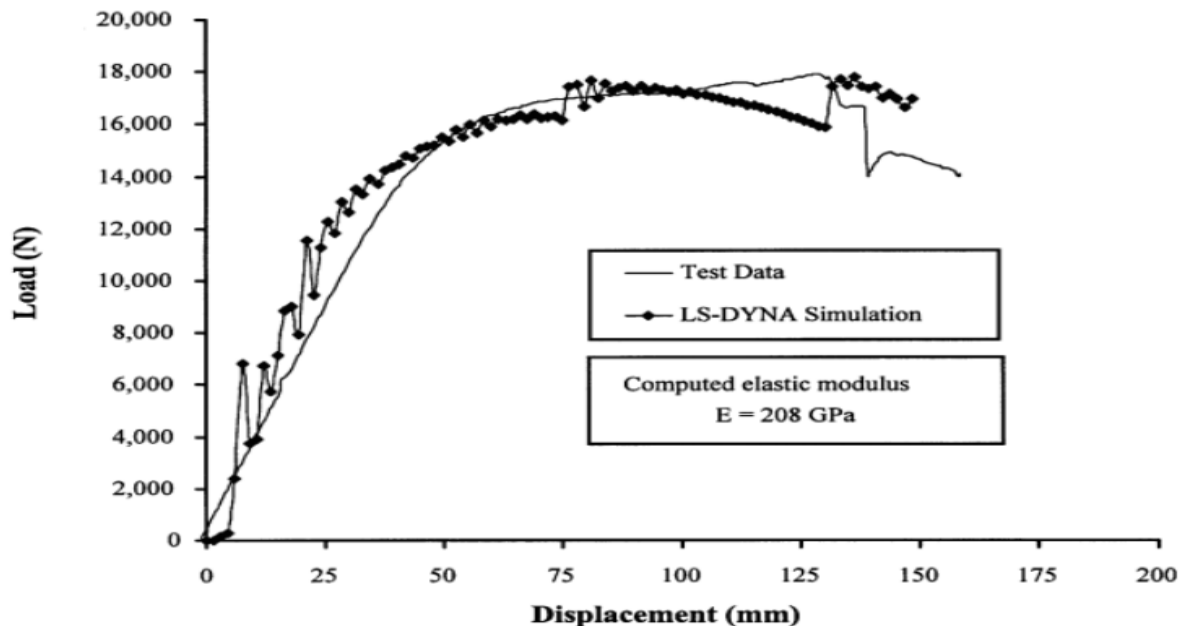


Figure 2.11: Load vs. displacement comparison for high strength door beam

As an illustration of one type of model validation, the full-vehicle test results were compared relative to the FE analysis calculations for the velocity of the center of gravity. In order to conduct the simulated impact, the model was supplemented with a ground plane and a frontal rigid barrier overlapping the driver's side 41% of the car's width. In general the research focusses on the existing vehicle model impact test. He tried to show the test impact at different angles such as -45° , 0° and 45° . He added some bars at the front and

sides around the doors to reinforce the pillars that to resist the impact load or decreasing the deformation level. Out of these works, he did not show any absorber mechanism that added on the vehicle [53].

In order to derive guideline about vehicle structure crash property and occupant package layout, this study analyzed top-rated vehicle in US NCAP. Based on proposed severity function which depends on RTE and layout factors of occupant package, this study obtained results as follows: in the first case, through severity function, they found relationship among package space, belt angle, and RTE (which is used to evaluate the crash pulse severity in this study). Using these relationship, they have suggested design guideline so as to achieve better safety performance in front NCAP. In the second case, they proposed the initial conceptual evaluation process in order to evaluate crashworthiness performance at early development stage. In order to guarantee same occupant injury performance, much more restraint space is required for the vehicle which has higher crash pulse severity [54].

Evans D and Morgan T. as vehicle manufacturers continue to become more aggressive with the styling of new vehicles, bumper system technologies will be required to find new solutions that fit into the reduced package spaces while continuing to meet the vehicle performance and cost requirements. The purpose of this paper is to introduce new and innovative EPP foam technologies and techniques [56].

Euro-NCAP and US-NCAP organized crash test using frontal and side collisions and publish test results that provided accurate information to motoring consumers based on a realistic and independent assessment of safety performance for most popular cars sold in Europe. In the European NCAP for frontal impact, a vehicle moving at 65 kmph strikes a deformable barrier that was offset to 40% of vehicle width .This test was very severe on the vehicle front-end structure as only one-half of the structure absorbs most of the energy of impact. US-NCAP conducted a frontal impact test and a side impact test to investigate occupant safety. A rigid barrier crash test was applied for full width frontal impact at 56 kmph (35 mph). Both Euro and US NCEP prepared to give emphasis to drivers or car owners to take care for their vehicle during collision at up to 64 kmph and 56 kmph resp. Their suggestion was focused on the vehicle that manufactured without any impact load absorber mechanism. They did not show any impact load absorber on their works [57].

3.0 MATERIALS AND METHODS

METHODS

3.1.1 Spring design and size determination

The function of the spring is to absorb the crashing load during impact or collision. In this study, the system contains two springs in each impact load absorber. The first spring is external spring which surrounds the whole length of the piston-cylinder and the second one is the smallest internal spring called inner spring which is located inside the cylinder. The springs are generally made from plain carbon steel. To go further on the analysis of the overall system, first step is the determination of the size of each spring which depends on the applied impact load. Thus the applied force or load on each absorber represented by 'F' which is called the axial compressive force. The design of each absorber is as follows:

3.1.1.1 External helical coil spring (outer spring)

The load applied during vehicle collision can be obtained using the mass and deceleration of the car itself. From Newton's second law:

$$F = ma \quad (3.1)$$

Where, m = mass of modified vehicle and

a = the deceleration of the vehicle during impact

Assume this car is hitting a fixed wall and it will stop within a second.

$$a = \frac{u-v}{t} \quad (3.2)$$

Where u = initial velocity of the car model

v = final velocity of the car during impact is zero

t = time after which vehicle stopped in seconds.

For this analysis it is assumed that the kinetic energy of the modified model KE_m is equal to the kinetic energy of the real vehicle at its velocity and mass. Therefore, the mass of the modified model m_m and the initial velocity of this model V_m can be obtained from equating the kinetic energy of the real vehicle KE_r with this one.

$$KE_r = KE_m \quad (3.3)$$

Then the force acting during collision is calculated using equation (3.1). This force or load is equally divided between two absorbers due to full impact load to the vehicle front. When this load is distributed to the two impact load absorbers, it is also distributed in to two springs in each absorber. Because the system contains two springs; inner spring and outer spring.

Then the maximum deflection of the outer spring at axial compressive load will be as follows [58]:

$$\delta = \frac{F}{K} \quad (3.4)$$

Where F = axial impact load

k = spring stiffness

Spring Wire Diameter (d):

The diameter of a spring wire (d), as shown in Figure 3.1, can be calculated using the following relations,

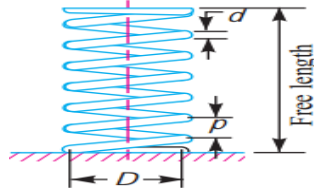


Figure 3.1: Helical coil spring parameters

$$\tau_{sp} = k \times \frac{8 \times F \times C}{\pi \times d^2} \quad (3.5)$$

$$k = \frac{4C - 1}{4C - 4} + \frac{0.615}{C} \quad (3.6)$$

$$C = \frac{D}{d} \quad (3.7)$$

Where τ_{sp} = shear stress of the spring material, 420 MPa

k = spring constant

C = spring index ≈ 5 to 10

d = spring wire diameter

D = spring mean diameter

Spring Coil Mean Diameter (D):

$$D = C \times d \quad (3.8)$$

Spring Coil Outer Diameter (D_o):

$$D_o = D + d \quad (3.9)$$

Spring Coil Number of Turns (n):

Number of turns of the coil from axial deflection equation,

$$\delta = \frac{8 \times F_e \times C^3 \times n}{G \times d} \quad (3.10)$$

$$n = \frac{\delta}{F_e} \times \frac{G \times d}{8 \times C^3}$$

Where δ = maximum spring deflection

G = spring material rigidity taken as approx. 84MPa

For square and ground end, the total number of turns (n'):

$$n' = n + 2 \quad (3.11)$$

Spring Coil Free Length (L_F):

L_F = solid length + Maximum compression + clearance between adjacent coils

$$L_F = n' \times d + \delta + 0.15 \times \delta \quad (3.12)$$

Pitch of the Coil (P):

$$p = \frac{L_F}{n' - 1} \quad (3.13)$$

Table 3.1: Size specification of outer spring

No.	parameter	value
1	Initial velocity of modified vehicle model	93.636 kmph
2	Mass of the modified vehicle model	875.88 kg
3	Deceleration of the modified vehicle model after impact	26.01 m/s ²
4	Force acting on the outer spring	11274.65 N
5	Maximum Deflection of outer spring	127.11 mm
6	Spring stiffness	88.7 kN/m
7	Spring index	5
8	Spring constant	1.3105
9	Wire diameter	25 mm
10	Spring mean diameter	125 mm
11	Spring outer diameter	150 mm
12	Total number of turns	26
13	Free length of the spring	796.18 mm
14	Pitch of the coil spring	31.85 mm

3.1.1.2 Inner helical coil spring

One quarter of the total impact load during car collision will be transferred to the inner spring. As in the above mentioned, the impact load equally divided in to the absorbers and this load, 11274.65N, is firstly accepted by the outer spring and next to this, this load also divides in to two and the inner spring take 5637.325 N impact load. The maximum deflection of the spring under axial compressive load and size determination using eqns. (3.4) to (3.13) are show in table 3.2.

Table 3.2: Specification of inner spring

No.	parameter	value
1	Initial velocity of real vehicle	80 kmph
2	Initial velocity of modified vehicle model	93.636 kmph
3	Curb weight of real vehicle in kilogram	1200 kg
4	Mass of the modified model	875.88 kg
5	Deceleration of the modified vehicle after impact	26.01 m/s ²
6	Force acting on the inner spring	5637.32 N
7	Maximum Deflection of the absorber	63.55 mm
8	Spring stiffness	88.7 kN/m
9	Spring index	5
10	Spring constant	1.3105
11	Wire diameter	15 mm
12	Spring mean diameter	75 mm
13	Spring outer diameter	90 mm
14	Total number of turns	17
15	Free length of the spring	328 mm
16	Pitch of the spring	20.5 mm

3.2 Buckling analysis of crashing load absorber piston

In certain types of problems, and especially those involving compressive loads, a structural member may develop relatively large lateral distortions under certain critical loading conditions. Such structural members are said to buckle, or become unstable, at these critical

loads. As an example of elastic buckling, consider firstly the buckling of a slender column under an axial compressive load.

3.2.1 Flexural buckling of a pin-ended strut

A perfectly straight absorber piston act as a bar of uniform cross-section with two axes of symmetry C_x and C_y in the cross section as shown in Figure 3.2. End thrusts P are applied along the centroidal axis C_z of the absorber piston and EI is its uniform flexural stiffness for bending about C_x .

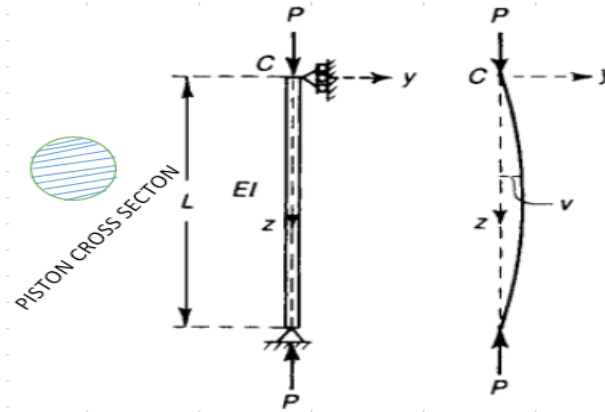


Figure 3.2: Uniform cross section of absorber piston

Now C_x is the weakest axis of bending of the absorber piston, and if bowing of the compressed absorber piston occurs we should expect bending to take place in the yz -plane. Consider the possibility that at some value of P , the end thrust, the strut can buckle laterally in the yz -plane. There can be no lateral deflections at the ends of the strut; suppose ' v ' is the displacement of the center-line of the absorber piston parallel to C_y at any point. There can be no forces at the hinges parallel to C_y , as these would imply bending moments at the ends of the bar. The only two external forces are the end thrusts P , which are assumed to maintain their original line of action after the onset of buckling. The bending moment at any section of the bar is then,

$$M = P v \quad (3.14)$$

This is the general equation of columns. From this general equation, the following formula with some rearrangement obtained [59].

$$P_{cr} = \frac{\pi EI}{l \times l}, \quad (3.15)$$

P_{cr} = the critical load that leads to column buckle can also expressed by P .

E = elastic modulus of the material,

I = moment of inertia

I = overall length of column or in this case the length of absorber piston that obtained from the overall structural mounting assemblies of crashing load absorber in Catia software.

$$I = \frac{\pi \times d^4}{64} \quad (3.16)$$

Where d = diameter of the piston

Now, to make a safe absorber piston design, the critical stress must be less than the allowable or yield strength of the material.

i.e. $\sigma_c < \sigma_y$

$$\text{Where } \sigma_c = \frac{P_{cr}}{A} \quad (3.17)$$

σ_y = allowable yield strength of the steel material

σ_c = critical stress

The circular cross sectional area of the piston can be calculated as follows.

$$A = \frac{\pi d^2}{4} \quad (3.18)$$

The values of various parameters related to piston buckling are given in table 3.3 [59].

Table 3.3: parameters under piston buckling analysis

No.	parameter	values
1	P_{cr}	11274.65 N
2	E	210 GPa
3	I	409 mm
4	d	25 mm
5	σ_y	250 MPa
6	σ_c	22.97 MPa

As the critical stress and yielding stress values are shown in the table 3.3, the critical stress is less than the yielding stress of the material. Thus the material of the piston can withstand the impact loads up to the critical load of 11274.65 N.

3.3 Crashing Models

3.3.1 Crashworthiness Model

3.3.1.1 Design Intent

The crashing load absorber is to be designed to keep the occupants safe in a collision up to a speed of 80 kmph. This structure is chosen to install at two places of the vehicle chassis frame outer side near to the front end of the frame. These two absorbers are an energy absorbent mounts that would absorb and slow the vehicle safely during collision with fixed barrier. This two absorbers are light weight, simple and do not increase the frontal area of the vehicle.

3.3.1.2 Mounting structure of absorber

To allow for a safe enclosure around the rider, a dynamic mounting model is designed to properly assemble and disassemble. The design intent of this system is to absorb energy of an impact when the vehicle collides the wall with its running velocity until full compression of the structure. This front mount crashing load absorber is of light weight and simplistic as well as stiffer to avoid failure. Structures being tested in the simulation are discussed in later section. The model of absorber mounting structure and line diagram are shown in figure 3.3.

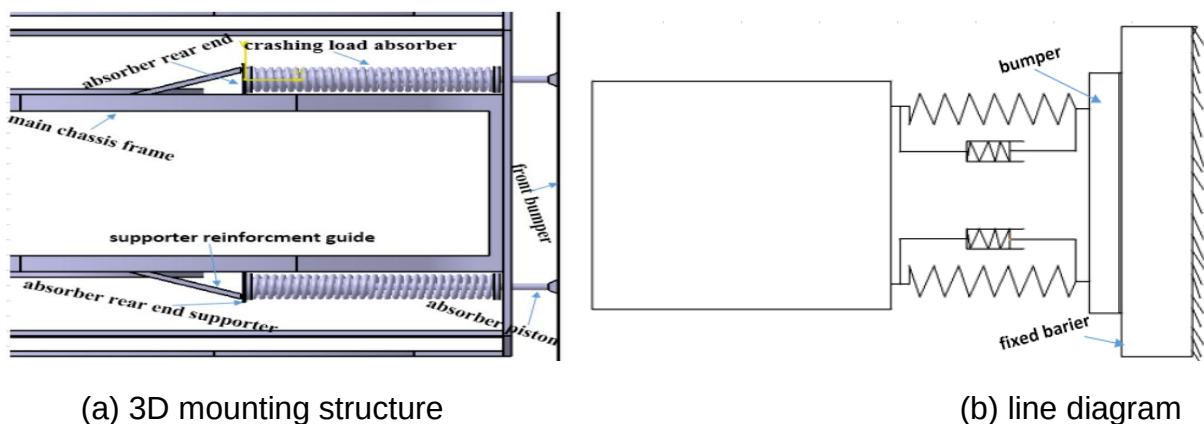


Figure 3.3: Absorber mounting structures

3.3.2 Modeling using Catia V5 R19

Catia V5 R19 employs a user based environment to graphically build 3D models of the desired structure. It is a vertically integrated system used to add, subtract or sweep material to the desired shape. This format is most desirable for the application being studied because a vertically integrated design means that additional features or trials can be run using a suppressed version of the final construct. A second strong advantage using Catia V5 R19 is its ability to save file types in common formats such as .step or .iges and it is easily imported

to the ANSYS software. These formats are what allows the translation of not only the given solid from Catia V5 R19 but transfers vital information such as part mates and contact regions where the laws of physics will automatically be applied in ANSYS.

3.3.2.1 Impact load absorber model

3.3.2.1.1 Inner spring

This part is a helical coil spring type shown in fig.3.4 that gives the light service, mounted inside the cylinder. The dimensions obtained from design calculations are used for preparing the model. The model is prepared using Catia software. To draw this part, there are two alternatives:

- Following start, shape then 'generative shape design' or
 - Directly in mechanical design using 'part design or sketcher' workbench
- and all necessary parameters of the spring are shown in section 3.1.1.2.

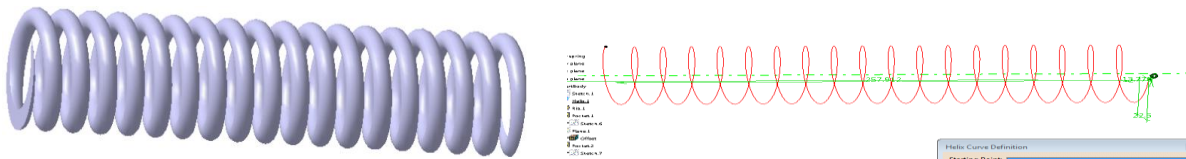


Figure 3.4: 3D Inner spring model and its helix

3.3.2.1.2 Outer spring

This part is a helical coil spring type shown in fig.3.5 that gives the light service, mounted outside the cylinder. The dimensions of this part obtained from design specification. The model is prepared using Catia software. To draw this part, there are two alternatives too:

- Following the procedure 'start', 'shape' then 'generative shape design' or
- Directly in mechanical design using 'part design or sketcher'

and all the parameters are given in the size determination analysis of section 3.1.1.1.



Figure 3.5: 3D Outer spring model and its helix

The detail drawings of the inner and outer springs is shown in Appendix-A.

3.3.2.1.3 Piston

This is prepared by using Catia V5 R19 depending on the design specifications or parameters as described or obtained in section 3.2. This part of absorber contains cylindrical rod, outer spring front end bracket or seat and front end; ring like shaped part that used to fix the absorber with the bumper as shown in figure 3.6. The ring outer diameter is equivalent to the piston diameter. All necessary parameters are obtained under section 3.2. To prepare this model in Catia, the general steps are shown in the flow chart of figure 3.8. The detailed drawing is given in the Appendix B.

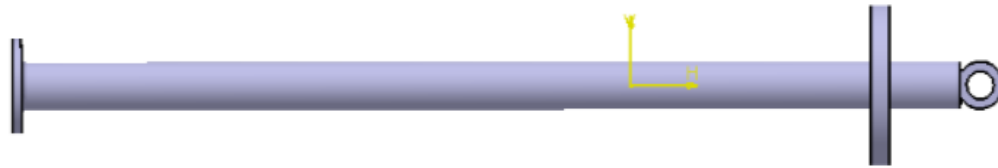


Figure 3.6: crashing load absorber piston model

Table 3.4: measured major dimensions of piston

No	parameter		Measured value (mm)
1	Piston rod diameter		25
2	Ring diameter	Outer	25
		inner	15
3	Total length		409
4	Bracket (spring seat) diameter		150
5	Seat depth		10

3.3.2.1.4 Cylinder

It is also prepared using Catia software. The size is depending on the size of the inner spring and outer spring. So the minimum length, innere diameter, outer diameter and other parameters are determined depending on the sizes of the two springs. Ring like part used to join the overall absorber with the supporter that fixed lateraly on the outer side of the chassis frame. The ring diameter is obtained from the the diameter of the supporter of the absorber. The steps to draw this part is shown in the flow chart of figure 3.8. The detail drawing of the part is attached in the Appendix C.

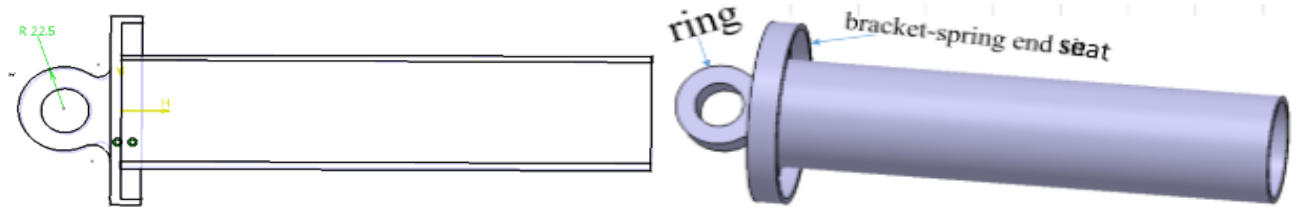


Figure 3.7: Crashing load absorber cylinder 2D and 3D model

Table 3.5: Cylinder dimensions

parameter	Values (mm)
Total length	350
Inner diameter	91
Outer diameter	98.5
Ring outer diameter and	45
Ring thickness	10
Outer spring seat diameter	156
Seat depth	10

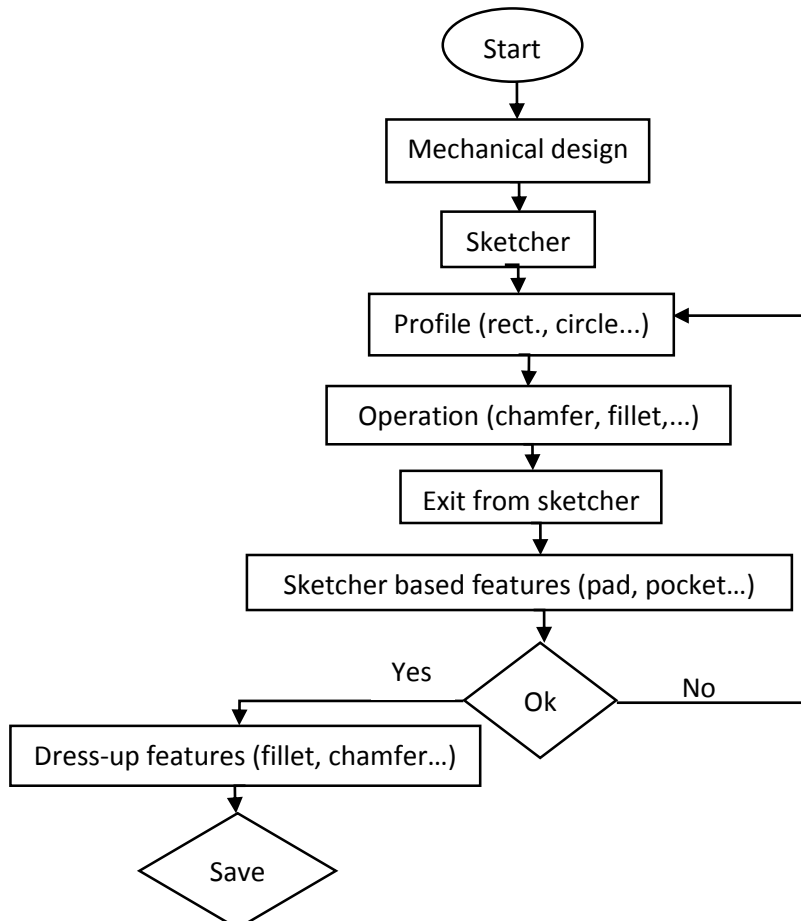


Figure 3.8: Flow chart that shows the general steps to draw cylinder, piston

3.3.2.2 Crashing load absorber assembly

This assembly is done using Catia software in the assembly workbench. To get the assembly workbench; click start, mechanical design, assembly design and after that import each component by clicking “existing componenet with positioning”. The assembly was done orderly starting from cylinder as a base for others. Then bring the inner spring and join its axis and one end ground seat with the cylinder inner base surface. After this assembly as shown in fig. 9, bring piston and outer spring turn by turn following the same procedure. The detail assembly drawing is attached in the Appendix D. fig. 10, shows the flow chart how to assemble the detail components

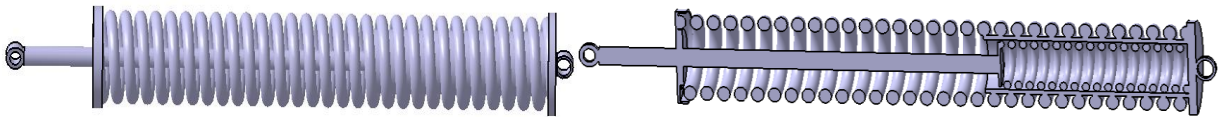


Figure 3.9: Crashing load absorber assembly with sectional view

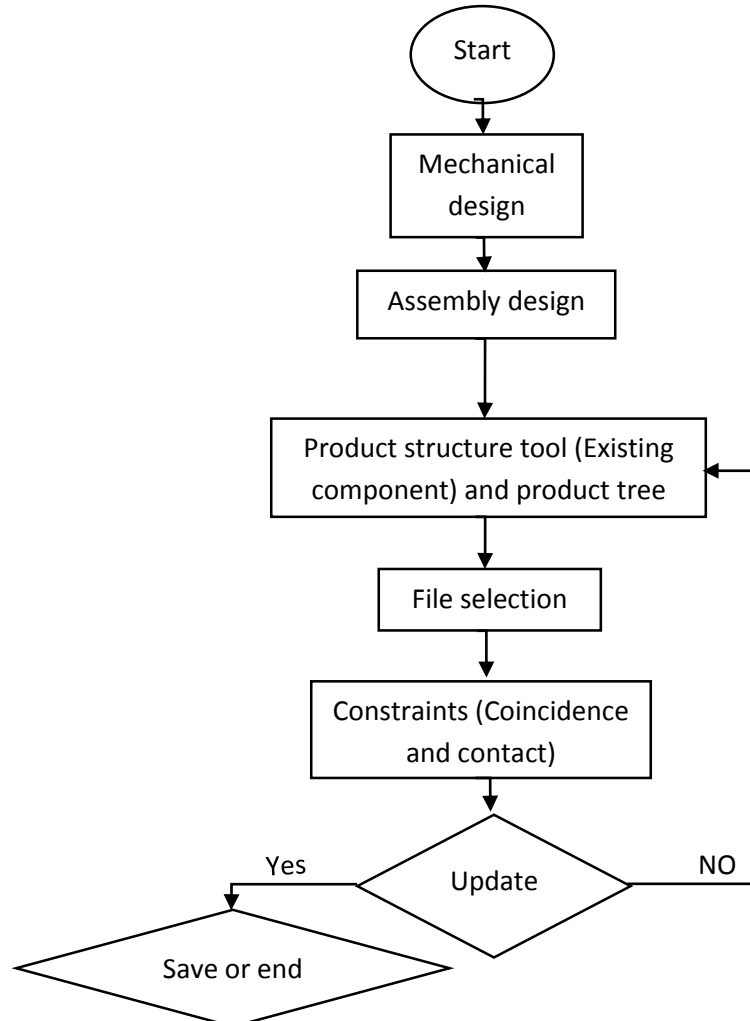


Figure 3.10: flow chart that shows the general assembly of models

3.3.2.3 Overall frame of the vehicle

The overall frame of the vehicle is drawn using the dimensions obtained by direct measurement of the existing vehicle in Adama science and Technology University automotive workshop. The vehicle used for the measurements is Toyota HILUX LN106-0136567. There is a gap between bumper and the front chassis frame and joined by a rigid metal bar with thickness around 3 mm. From various literatures, it is understood that most light weight vehicle bumpers are hinged on to the front end of the chassis frame. This situation may minimize the overall weight and size of the vehicle, but during collision especially front collision due to increase in overlap length, the injury level is increases. The track width and overall length of the vehicle are taken from the existing real vehicle in the workshop as given in the table below. The lower end of the bumper is 60 mm far from the upper layer of the front end chassis frame. The overall height of the bumper is also considered in the preparation of the model of bumper and the crashing load absorber front end is attached at 78.5 mm from the lower edge of the bumper. The two crashing load absorbers are mounted with a distance of 784 mm between them. The general assembly drawing of these two models such as the existing vehicle model and modified vehicle model is given in Appendix E.

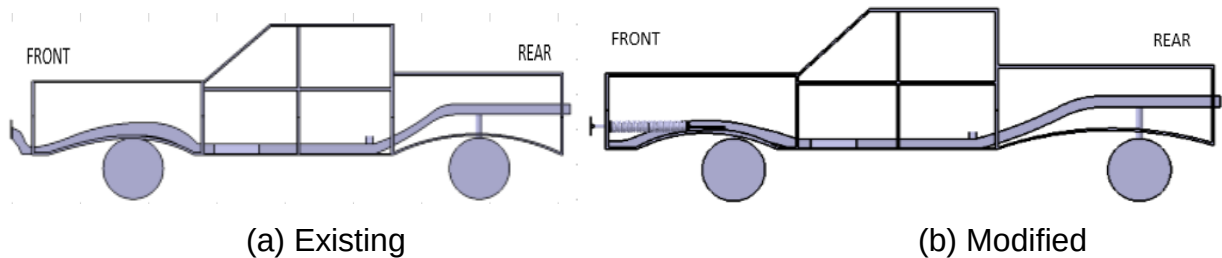


Figure 3.11: Overall frame of vehicle models

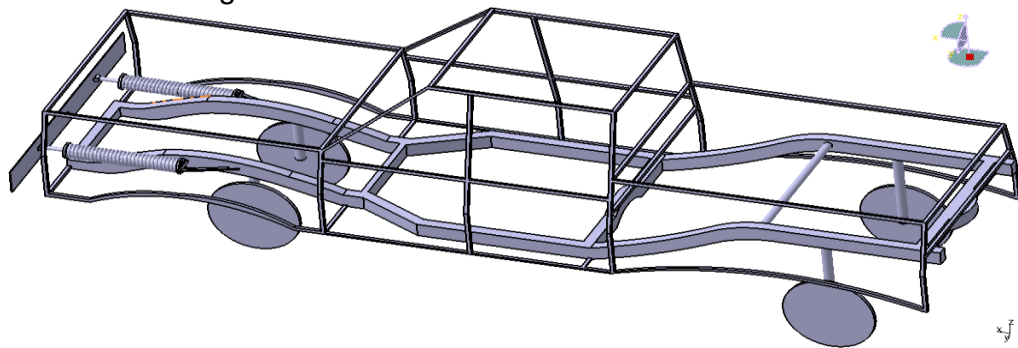


Figure 3.12: Isometric view of overall frame model of the vehicle with absorber



Figure 3.13: Real vehicle (source: photo captured from real vehicle)

Table 3.6: Existing and modified vehicle models parameters

parameter	Real vehicle	Existing vehicle model	Modified vehicle model
Weight	1200 (Kerb) kg	813.85 kg	875.88 kg
Wheel base	2840 mm	2840 mm	2840 mm
Height	1760 mm	1760 mm	1760 mm
length	4690 mm	4690 mm	4690 mm
Gap b/n chassis frame and bumper	-	115.5 mm	115.5 mm

3.3.2.4 Crashing load absorber rear end Supporter

Crashing load absorber is designed to be simple and effective in absorbing the energy without transferring to the front chassis frame as well as to the occupants and driver. To give rigid support at the rear end, a supporting structure is designed with two members, one attached perpendicularly with chassis frame laterally, and another which is one end pinned with the supporter free end and other end is attached with the chassis frame outer side as shown in the figure 3.14. This structure which is the unfixed supporter tip guide pinned with the chassis frame and unfixed supporter tip conventionally depend on the applied force on the supporter. The angle between the chassis frame and unfixed supporter tip guide is more acute that helps to minimize the bending of chassis frame at that point.

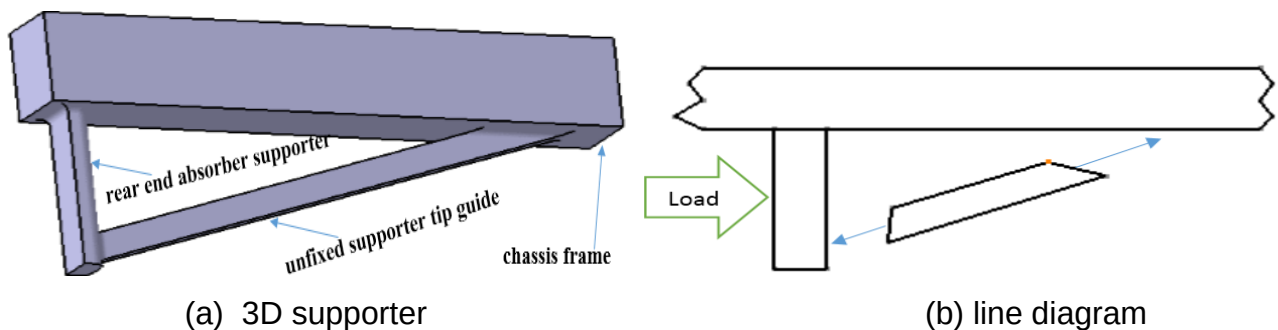


Figure 3.14: Supporter mounting structure with reaction forces

3.3 Finite element analysis

ANSYS R15 is a powerful numerical solver that is used due to its abilities in FEA given the mechanical nature of this thesis. The details of the simulation are mentioned later in this report, however, the basic format of the program is a parceled interface with sub windows and subroutines that define each given detail of the simulation separately and combined using the master window. This interface is ideal for the application at hand due to its ability in running multiple steps because each parameter is defined in its own subroutine and can be shared between simulations. In this case explicit dynamics analysis case is applied for the crashing load analysis and static structure analysis for crashing load absorber rear end supporter deformation analysis. On the other hand, static structural analysis also used for checking buckling of absorber piston and deformation values of overall crashing load absorber system with point load. In explicit dynamics, velocity is the initial parameter to proceed to the deformation and stress analysis. In the static structural part, point load is the initial parameter instead of velocity [61].

3.3.1 Meshing

Meshing is a process used commonly by numerical solvers, breaking a complex geometry part into small geometric shapes. These shapes are then mathematically coupled and solved as individual parts for stiffness and deflection being combined in the end for a global result of the effect on the part or assembly as a whole. These elements come in many shapes and sizes limited only by the user's imagination to fit specific needs. A smaller mesh size means having more, or smaller elements filling the part allowing for a more refined definition of how the stress is passing through the part. Having more elements also increases the computational time linearly with elements and to the power of two in relationship to the area being covered. Refinements to a standard mesh include techniques such as sizing, sweeping and shaping, amongst others not used here. Sweeping is a technique that places a planar mesh surface in the thickness of the part shown in Figure 3.15. This in turn simulates two separate parts joined together to further the accuracy in the thickness direction of the part being studied. Shaping can be utilized much like swapping coordinate systems, for instance a hexagonal mesh is beneficial for round parts and a triangular mesh is better for planar parts. This can also get more complex with the introduction of trapezoids and other simple polygons and can be found to be more beneficial for a given part by trial and error. The meshing type is triangular as shown in the figures below.

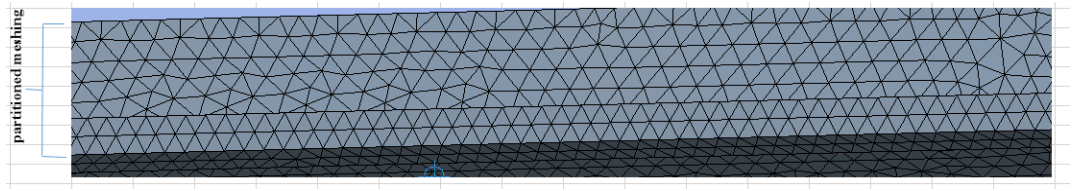
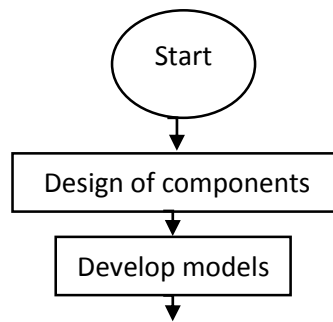


Figure 3.15: Mesh Partition

As the solver progresses past the elastic limit of the part, the solver will refine the mesh in these areas to gain a detailed evaluation because this area is so sensitive to loading changes at this time. This particular model had no abrupt geometry changes such as sharp corners or small raised sections, so meshing is not a complication in the simulation. Element sizes are taken 1.5mm for static structural analysis and 8 to 10mm for explicit dynamics analysis were kept surrounding the solid with refinements. Square elements were set to prefer but not forced due to the need for triangular elements in the structural steel and surrounding layers. Each layer was swept uniformly and kept a uniform mesh although minor refinements were needed at the structural steel layer during large contours to avoid contacting nodes. The steps to mesh the given geometry are as follows:

- On the model workbench, click 'mesh' on the project
- Then on the detail edit the relevance from 0 to 1
- Under sizing the element size is edited depending on the geometry size and the capacity of computer
- The relevance center should edited to either coarse, medium or fine. From these alternatives fine is the recommended one for suitable a smaller element size depend on the computer capacity but in this analysis coarse is selected due to lower capacity of my computer processor.
- The smoothing item should be edited to either high, medium or low. High is better.
- Such other necessary things should be edited.
- Finally after all necessary things are edited, click 'update' or 'general mesh' and wait up to get the required result.



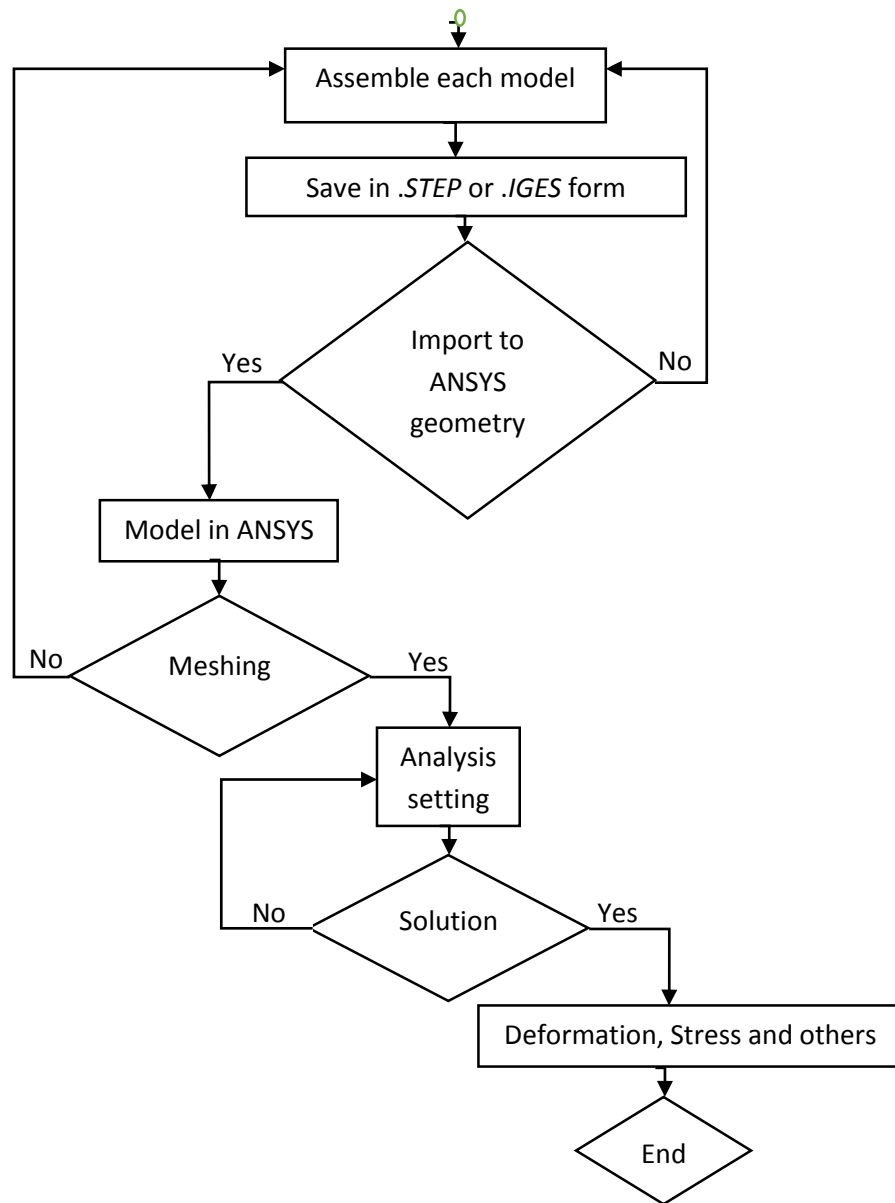


Figure 3.18: Procedures for modeling in ANSYS workbench

➤ Material selection

Selection of the suitable material for mechanical engineering areas is a key aspect for every researcher. Even a small amount in weight reduction of the vehicle, may have a wider economic impact. Composite materials are proved as suitable substitutes for steel in connection with weight reduction of the vehicle. The material which is capable of absorbing more energy in the form of elastic deformation would be preferred for crashing load absorber and other vehicle body and chassis frames. Scope of this work does not include the analysis of the effect of type of material. Therefore this analysis is performed with the same material which is structural steel. Performance of existing vehicle model and modified vehicle model both made of structural steel are compared on their deformation size and equivalent stress.

4.0 RESULTS AND DISCUSSION

4.1 Finite element analysis

The manufacture of automobile structures has recently evolved to satisfy both car manufacturer's interests and consumer demand. However, the evolution process has been subjected to several potentially conflicting constraints such as: minimum structural weight, sufficient occupant protection according to safety regulations, and efficient power and the corresponding vehicle acceleration. These had led to many necessary vehicle structural modifications that are designed to apply to different car models pending minimum changes, and to cope with engineering technology and manufacturing capabilities. Current vehicle body structures are classified into two main categories: the body-over-frame structure (or ladder frame) and the unitary-body (or integral) structure. The second category, known as a *monocoque* structure, is widely used in manufacturing all passenger, SUV, minibus, and bus vehicles. Experimental crashworthiness investigations of vehicle structures through crash tests are extremely costly; it costs more than \$25,000 per direction crash test and could reach to \$200,000 for a full crash test. In addition to the high costs involved, they are time consuming process that does not always provide definitive information [60]. However, with improvements in the computational power and the accuracy of simulation models, virtual testing can be extensively used as an alternative to the time consuming and costly full-scale tests, especially when testing severe maneuvers. As a result, different types of computational models, especially Finite Element (FE) model, have emerged to simulate the response of vehicle structures under crash events. Crash simulation of vehicle's structure is a highly nonlinear and short event duration problem and generally includes the following nonlinearities:

- (i) geometric (concerned with changes in configuration of each structural member's shape);
- (ii) material (concerned with changes in material properties such as plasticity);
- (iii) Boundary or contact (concerned with changes in contact area due to frictional or applied force) [61].

This implies that specialized software is required for simulating crash events. Among the available FE software packages, the explicit dynamics is one of the most widely used programs and is well suited for conducting crash analysis.

In this work, a multi-purpose FE model of a light-weight pick-up car with a ladder frame structure-type is used to investigate frontal impact crashworthiness enhancement by implementing the proposed crash energy absorber.

4.1.1 Nonlinear Finite Element Modeling for Crashworthiness

The FE methods have been extensively used in simulating different dynamic problems through the discretization of the whole structure into finite elements, development of the equations of motion for each element, assembly of all equation of motions based on continuity and solving the set of governing equations of motion using the appropriate techniques [61]. The generalized dynamic equations of motion of a structure in the finite element form can be represented as:

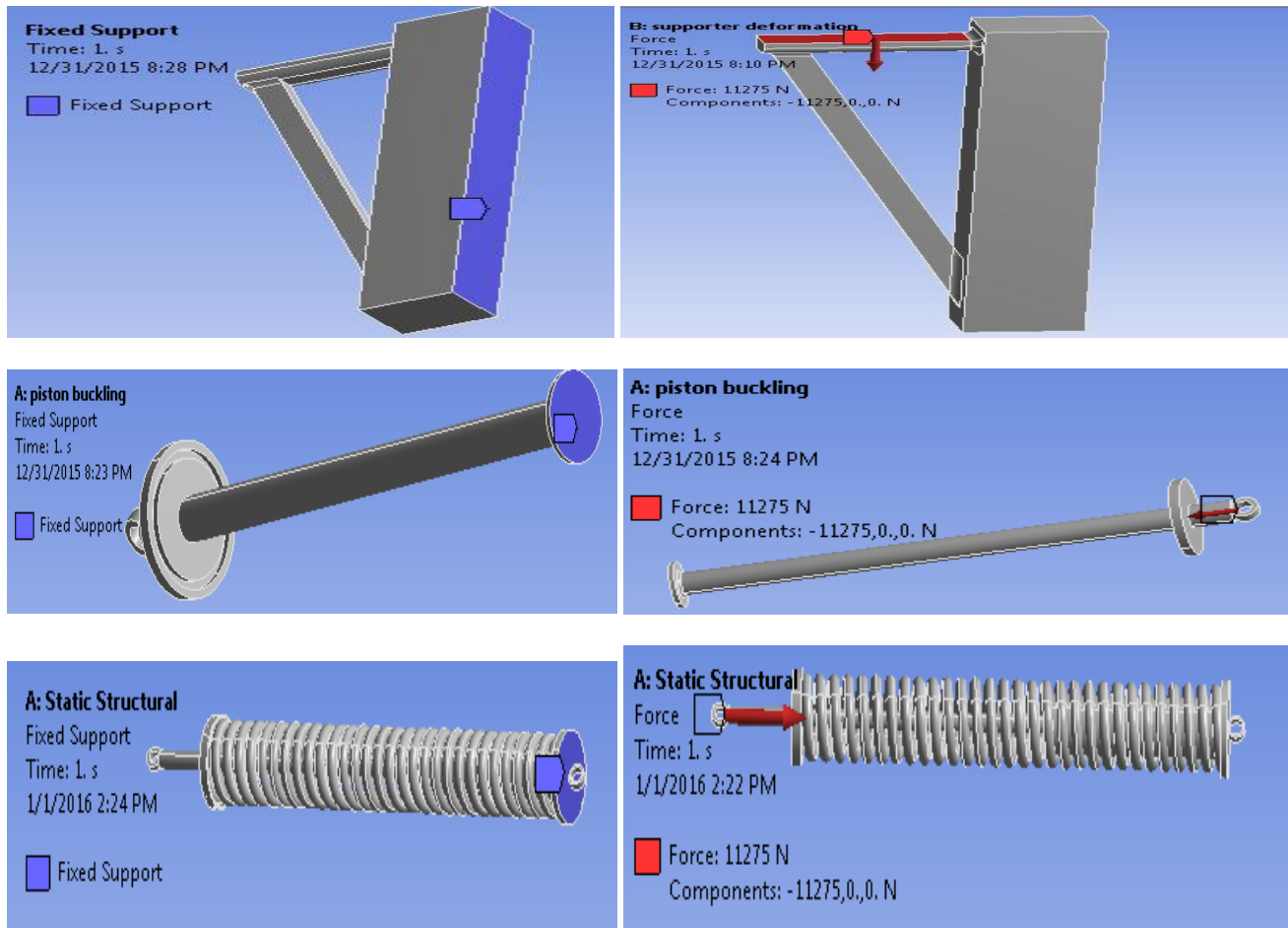
$$[M]\{X''\}+[C]\{X'\}+[K]\{X\}=\{F^{ext}\} \quad (4.1)$$

Where $[M]$, $[C]$ and $[K]$ the system structural mass matrices, damping matrices and stiffness matrices respectively. $[K]\{X\}$, which equals $\{F^{int}\}$, represents the internal force resulting from the plastic deformation of a structural part, $\{X\}$, $\{X'\}$ and $\{X''\}$ are the nodal displacements, velocity and deceleration vectors respectively and $\{F^{ext}\}$ is the external nodal load vector. It should also be noted that for problems that involve material and geometrical nonlinearity, such as crash simulation problems, the stiffness matrix $[K]$ is not constant; instead, it is a function of displacement and time. There are two methods of analysis that can be used in solving Eqn. (4.1), namely: implicit and explicit methods due to the behavior of dynamic analysis. An explicit method is more appropriate for impact problem or nonlinear transient dynamic problems with very short time durations as in the case of a crash simulation event. This method is widely used in analysis of phenomena of energy absorbing. Due to its large capabilities in a field of highly nonlinear and very short term actions it is one of the main tool in designing either whole vehicle body or particular energy absorbing elements [62]. On the other hand, the implicit method is more suitable for dynamic problems in which the duration of the applied load is relatively large compared with the fundamental period of structure and it is not applicable for this analysis [63]. Now before discussing the explicit dynamics analysis of the existing vehicle model and modified vehicle model, the results of static structural analysis of crashing load absorber, crashing load absorber piston and crashing load absorber rear end supporter will be discussed.

4.1.2 Static structural analysis

4.1.2.1 Boundary conditions

In finite element analysis 3D models of crashing load absorber, crashing load absorber rear end supporter and absorber piston which are developed in CATIA V5 are imported to ANSYS workbench to conduct the analysis. The boundary conditions given are fixed support at one end as shown in the figure 4.1 and point load is applied on the front face of each model. The details of meshing are given table 4.1 and figure 4.2.



(a) Fixed support

(b) point load

Figure 4.1: Fixed support and point load representation on crashing load absorber systems

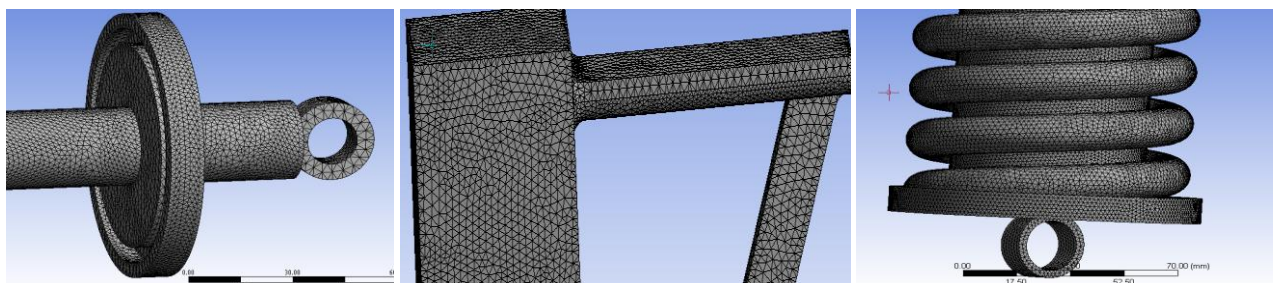


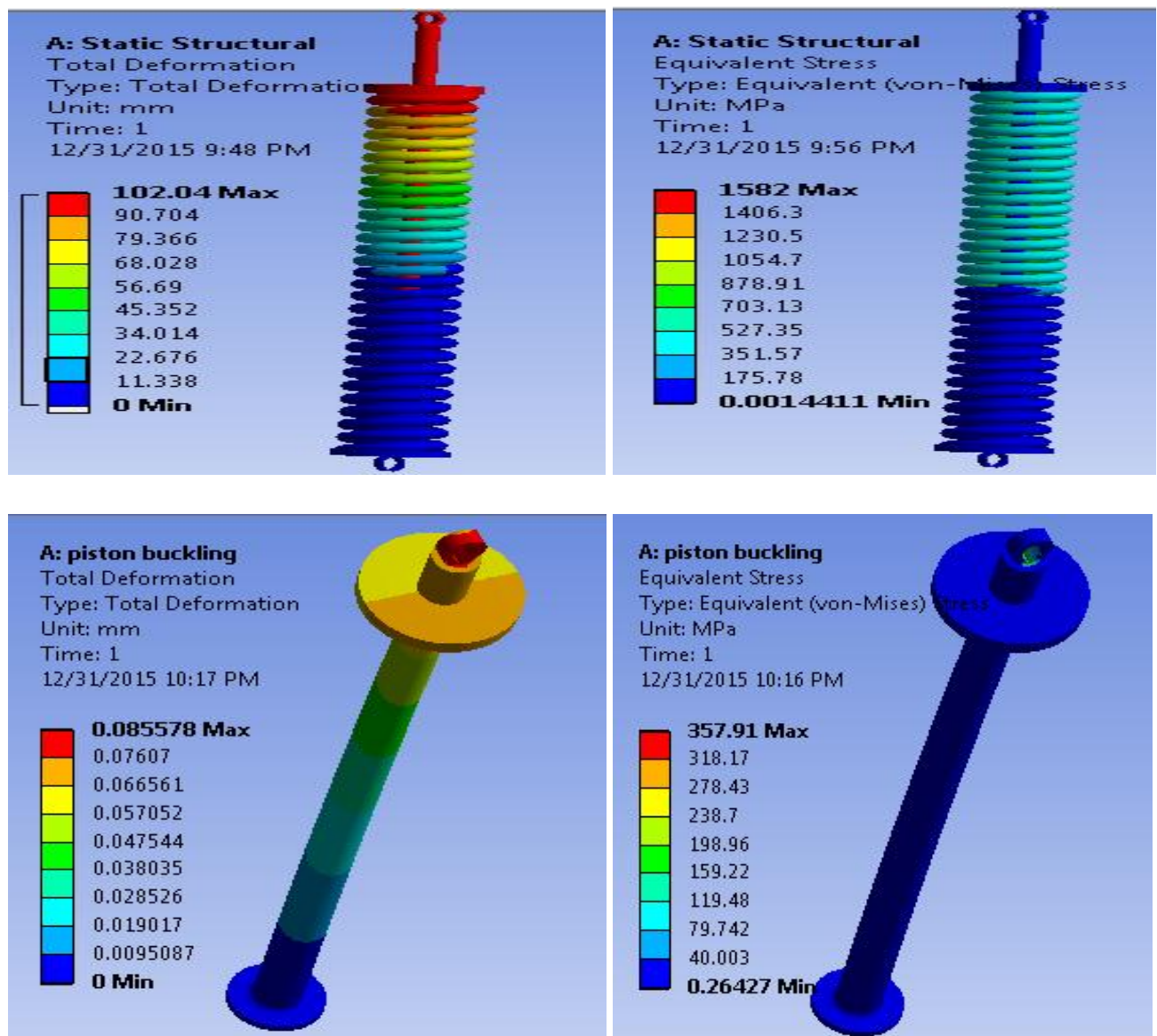
Figure 4.2: Some portion of the Meshed absorber piston, absorber rear end supporter and absorber assembly respectively

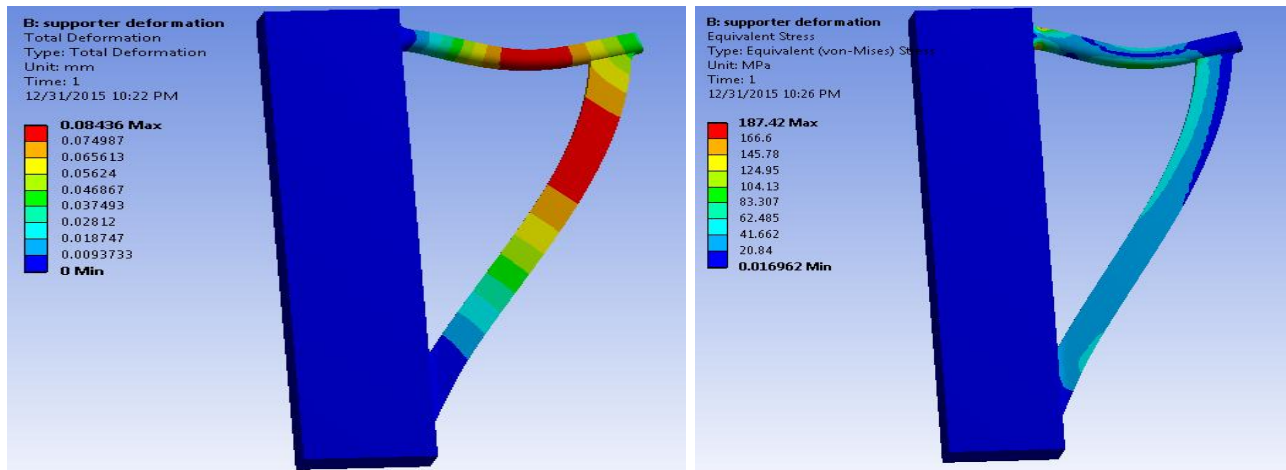
Table 4.1: Meshing statistics of the static structural analysis

Parts are analyzed	Number of nodes	Number of elements	Element size (mm)	Analysis type
Absorber rear end supporter	30731	131561	1.5	Static structural
Absorber piston	31663	116859	1.5	>>
Crashing load absorber	492766	2275708	1.5	>>

4.1.2.2 Stress and deformation results

The total deformation (all directional displacement) and Equivalent stress (von-misses stress) are obtained from the static analysis of the models shown in figure 4.3.





(a) Total deformation

(b) Equivalent stress

Figure 4.3: Total deformation and equivalent stress analysis of absorber, piston and supporter respectively

The maximum value (maximum deformation) is indicated with red color which is 102.04 mm, 0.085578 mm and 0.08436 mm for crashing load absorber, crashing load absorber piston and crashing load absorber rear end supporter respectively as shown in figure 4.2 (a) and figure 4.2 (b) also shows maximum and minimum value of equivalent (von-misses) stresses when maximum impact load is applied on them. From the analysis the maximum Equivalent (von-misses) stress of the crashing load absorber is 1582 MPa, crashing load absorber piston is 357.91 MPa and for crashing load absorber rear end supporter is 187.42 MPa. Table 4.2 is tabulated based on the maximum equivalent stress and total deformation analysis results of components in crashing load absorber mechanism during static structural impact analysis. The results help in understanding whether the components can withstand the maximum impact load during collision or not.

Table 4.2: Result of deformation and stress analysis of absorber, piston and supporter

Parameter	Crashing load absorber	Absorber piston	supporter
Load (N)	11275	11275	11275
Total deformation (mm)	102.04	0.085578	0.08436
Equivalent stress (MPa)	1582	357.91	187.42

The results of this analysis are based on the design criteria called Von misses stress criteria, which arises from distortion energy failure theory. According to distortion energy

theory, failure occurs when the distortion energy in actual case is more than the distortion energy in a simple tension case at the time of failure.

From table 4.2, the results of this analysis are compared with the material (structural steel) properties. From structural steel general properties, its ultimate strength is 460 MPa [59]. As the results indicate, the crashing load absorber equivalent stress; 1582 MPa greater than the material property of ultimate strength whereas the crashing load absorber piston and supporter equivalent stress values of 357.91 MPa and 187.42 MPa respectively are less than the ultimate strength of structural steel.

The purpose of static structural analysis on the crashing load absorber is to know how much the deflection length of the absorber is when statically maximum concentrated load is applied at the front end of absorber and see what it with is in the gap between the front bumper and front end chassis frame. The gap between the front bumper and the front end chassis frame is 115.5 mm as obtained by direct measurement on existing vehicle.

Depending on the maximum impact load applied on the absorber and the gap between the front bumper and front end chassis frame, the maximum deformation value of crashing load absorber is tolerated.

When the static structural analysis is performed independently on the crashing load absorber piston, the equivalent stress value is less than the ultimate strength value of the material. Even the maximum deformation value is almost negligible i.e. 0.085578 mm. Therefore, the crashing load absorber piston can play a role without any buckling during vehicle crash with the fixed barrier.

During vehicle frontal impact with fixed barrier or any other object, there is load transfer starting from the front bumper to the rear of the vehicle parts. The crashing load absorber is used as an intermediate between front bumper and front end chassis frame. This crashing load absorber is hinged on the supporter which is fixed at the side of the chassis frame laterally as shown in figure 3.14. Therefore, the impact load is directly transferred to the supporter during the crashing load absorber spring get its solid length. At this time the supporter gets the maximum impact load of 11275 N. Using this load, the static structural analysis result shows the maximum deformation value of 0.08436 mm which is almost negligible value. The maximum equivalent stress value as shown in table 4.2 is less than the material ultimate strength value. So the supporter can support the crashing load absorber up to this maximum impact load without failure.

4.1.3 Explicit dynamics analysis

4.1.3.1 Existing vehicle model and modified model analysis

This is done by taking the full model of the vehicle developed using Catia software. The meshing element size is taken as 8 to 10 mm. The analysis is performed using explicit dynamics at its initial velocities to determine the deformation and stress values. The vehicle model is assembled with the wall; act as a barrier, for this analysis. The material of wall is similar to the vehicle model but its thickness is much greater than the bumper. This is stiffer than the front part of the vehicle.

The displacements in explicit dynamics scheme are solved by direct numerical integration of the equation with a short period of time step. This method is not accurate for modeling quasi-static processes that occur over a long period of time. To be more specific, Δt must be smaller than the time required for an elastic wave to propagate from one side of an element to the other. An elastic wave is a type of mechanical wave that transmits energy which propagates in elastic or viscoelastic materials. The elasticity of the material provides the restoring force of the wave. The end time of crashing load analysis in analysis setting is calculated from the formula as given in eqn. (4.3). This end time is feed at explicit dynamics model workbench 'analysis setting' on the detail part. The software by itself asked to fed the 'end time' when the user click on 'analysis setting'.

The one-dimensional stress-wave speed traveling through the material during collision is calculated first as follows in the eqn. (4.2).

$$C_{stress} = \sqrt{E(1 - \nu)/[(1 + \nu)(1 - 2\nu) \times \rho]} \quad (4.2)$$

C_{stress} = is the one-dimensional stress-wave speed traveling through the material during collision.

E = Elastic modulus of the material, taken as 210 GPa

ν = Poison's ratio, which is 0.3

ρ = Density of steel material, 7850 kg/m³

The end time of the collision of the model is calculated either eqn. (4.3) or (4.4). In other words, the minimum time step size is related to the time it takes for stress wave to travel across a block element or deformation element.

$$\Delta T = \frac{l}{C_{stress}} \quad (4.3)$$

where l = overall effective length of the vehicle model, which is taken 4690 mm

Explicit dynamics analysis method needs an initial velocity and end time from its motion or the end of deformation time on the elements. Therefore the initial velocity for existing vehicle and modified vehicle model is 26.98 m/s and 26.01 m/s that obtained using eqn. (3.3) with a

mass of 813.85 kg and 875.88 kg respectively. The density and modulus of elasticity of the structural steel material are 7850 kg/m^3 and 210 GPa respectively. These velocity variation comes from the mass difference and the mass of each model is as shown in the figure 4.4.

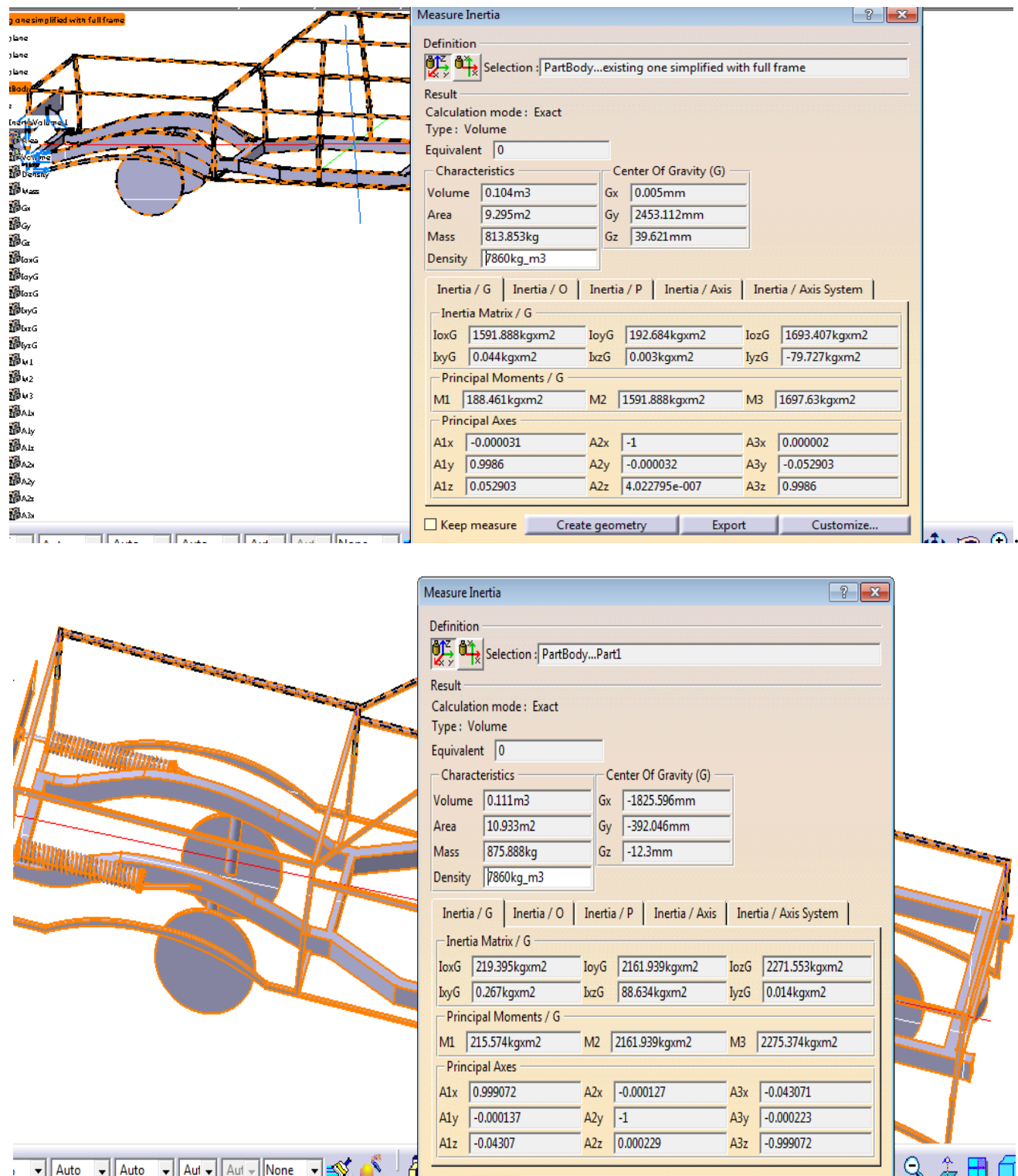


Figure 4.4: Inertia measurement of existing vehicle model and modified vehicle model respectively

Figure 4.5 shows the geometry of the two vehicle models with wall that is imported from the Catia software in to the ANSYS R15.0. These models are imported to the explicit dynamics Geometry tool.

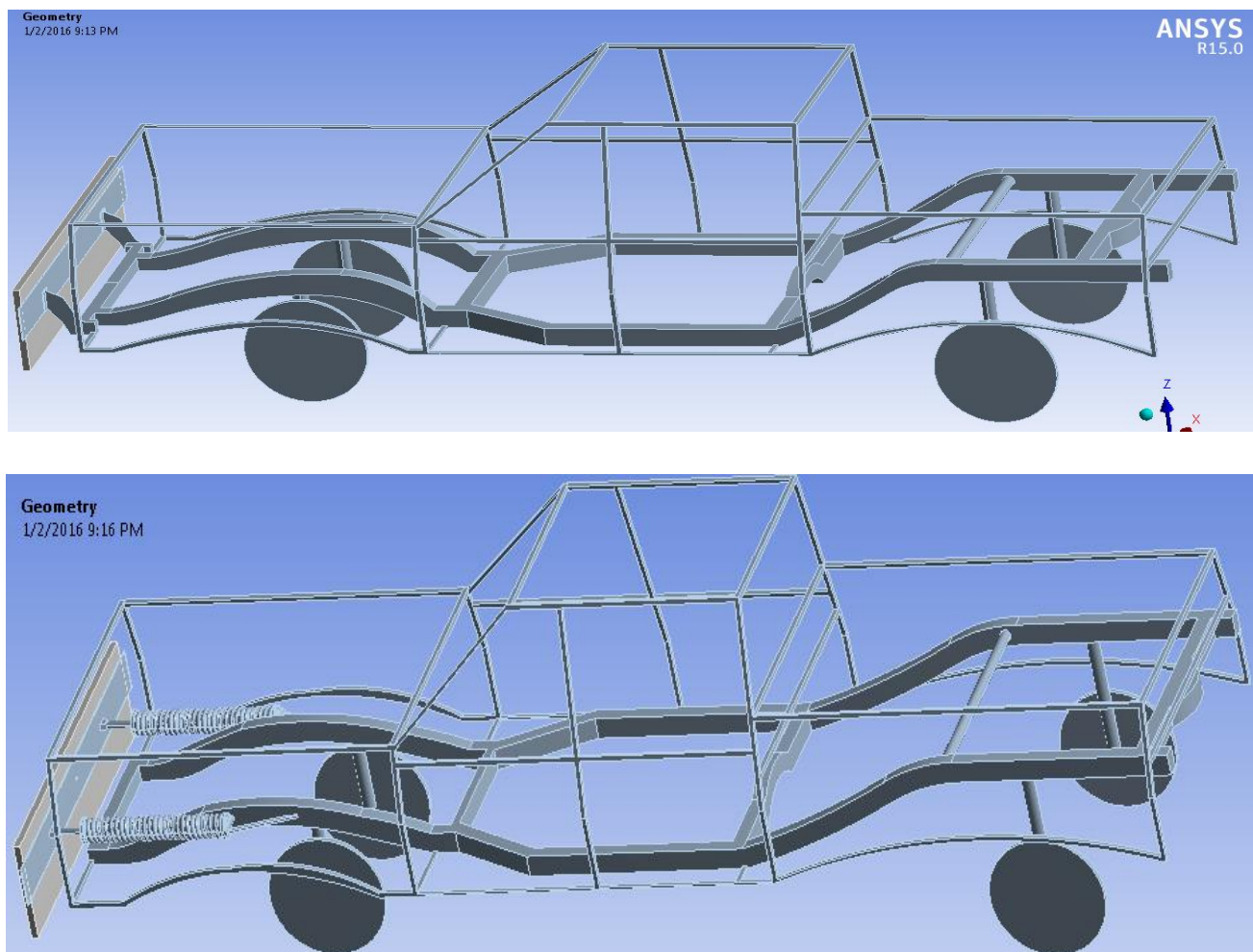
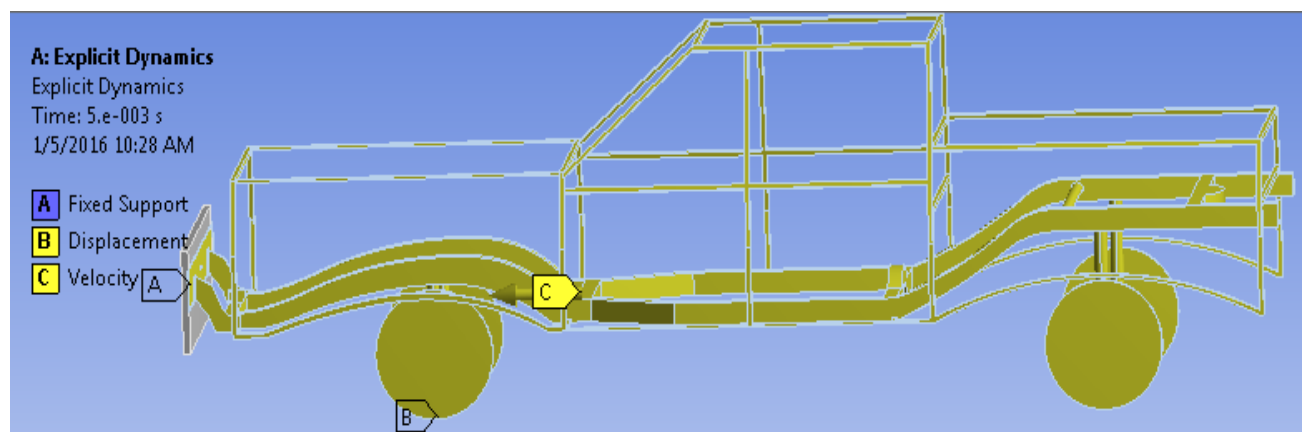
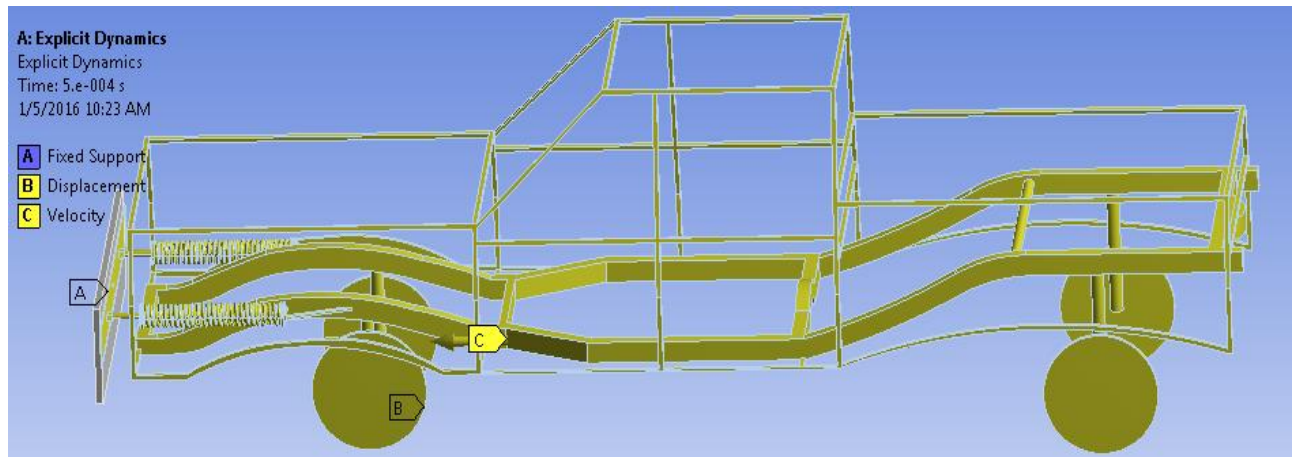


Figure 4.5: Geometry of the .existing and modified vehicle models with rigid barriers

4.1.3.1.1 Results of crash analysis of existing and modified vehicle model



(a) Existing vehicle model with rigid barrier



(b) Modified vehicle model with rigid barrier

Figure 4.6: Boundary conditions

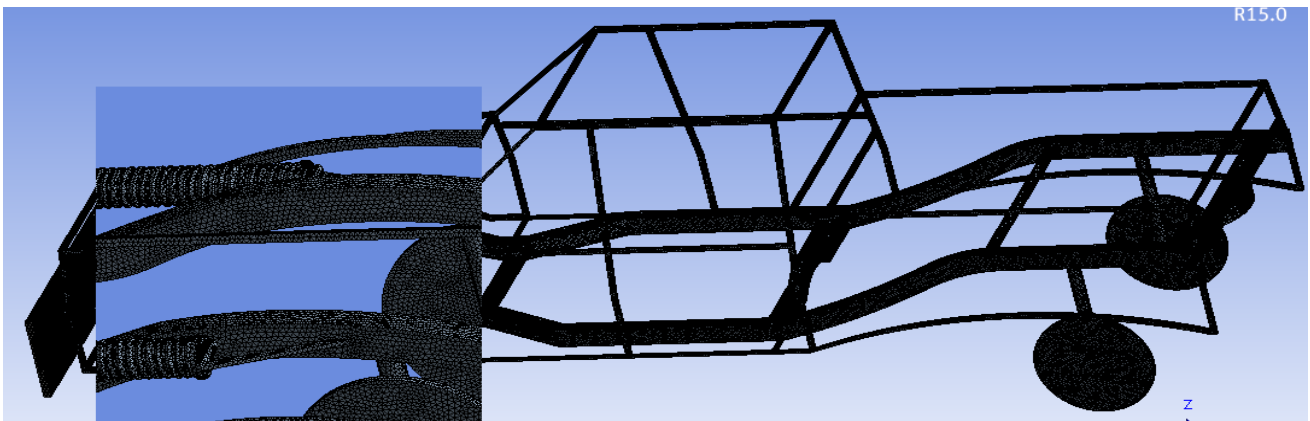
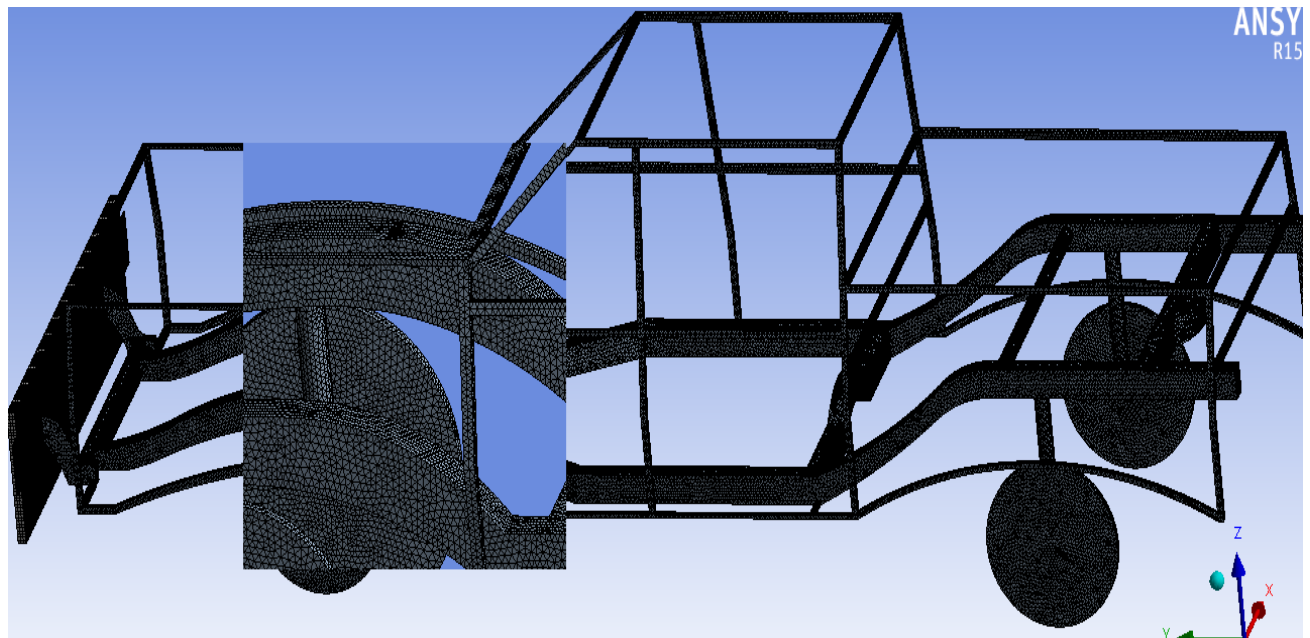
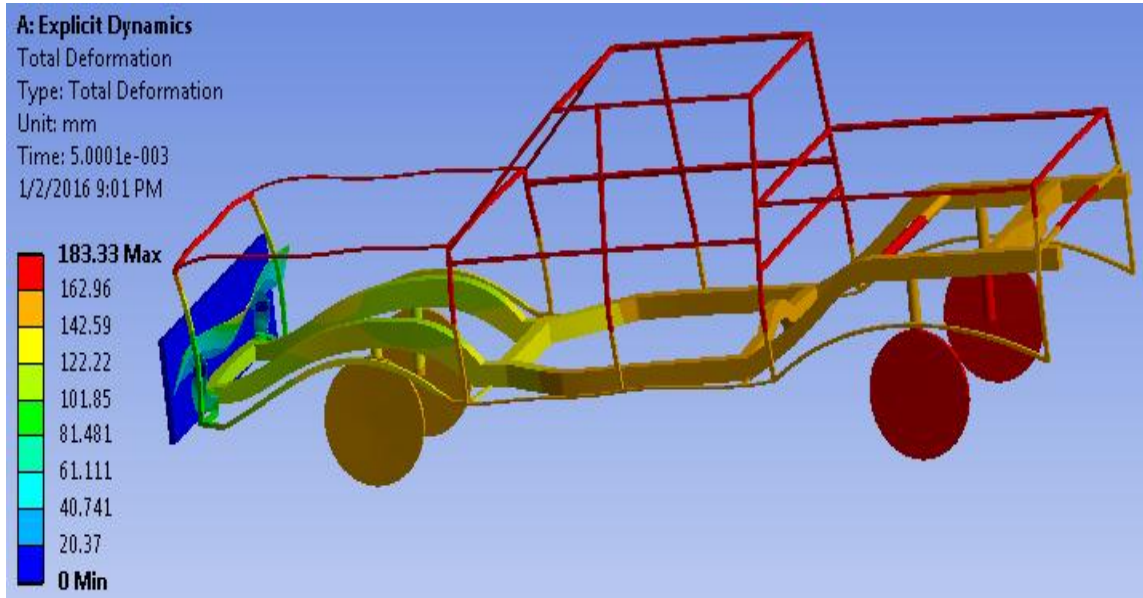


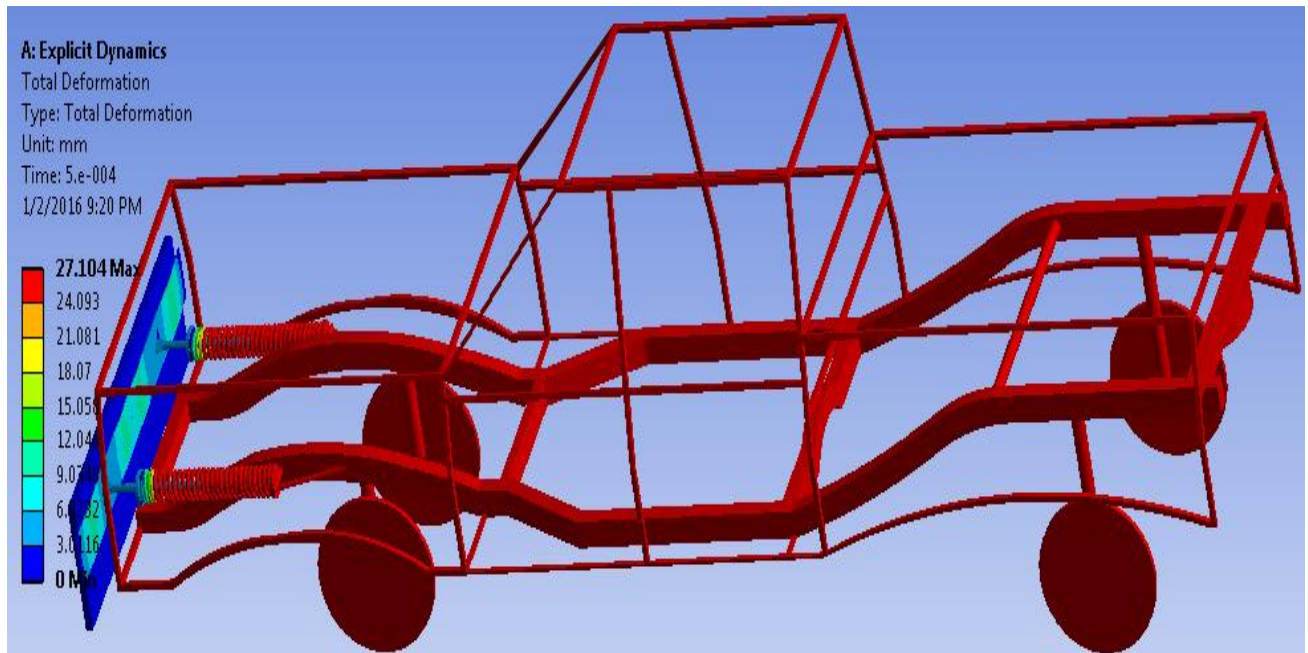
Figure 4.7: Meshed model with partial magnification of existing and modified vehicle models with rigid barrier or wall

Table 4.3: Meshing statistics for explicit dynamics analysis of the two models

Models	Number of nodes	Number of elements	Element size (mm)	Analysis type
Existing model	156204	513497	8	Explicit dynamics
Modified model	179443	564115	10	>>

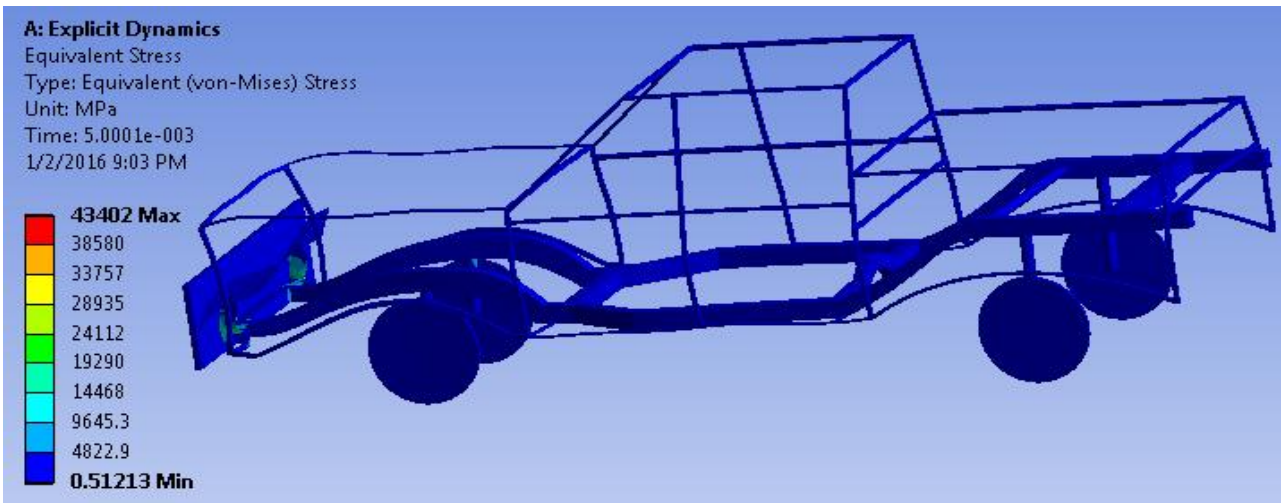


(a) Existing vehicle models

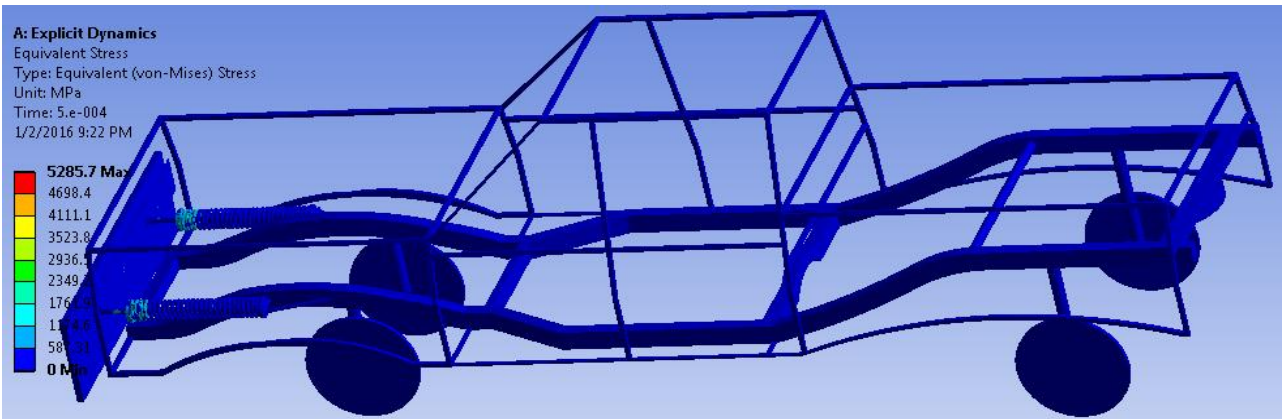


(b) Modified vehicle models

Figure 4.8: Total deformation analysis



(a) Existing vehicle models



(b) Modified vehicle models

Figure 4.9: Equivalent (Von-misses) stress analysis

Table 4.4 is tabulated on the basis of stress and deformation analysis results of existing vehicle model and modified vehicle model during frontal crash.

Table 4.4: Stress and Deformation analysis results of existing vehicle model and modified vehicle model during frontal crash

Parameters	Existing vehicle model	Modified vehicle model
Initial speed (m/s)	26.98	26.01
Mass (kg)	813.85	875.88
Total deformation (mm)	183.33	27.104
Equivalent stress (MPa)	43402	5285.7

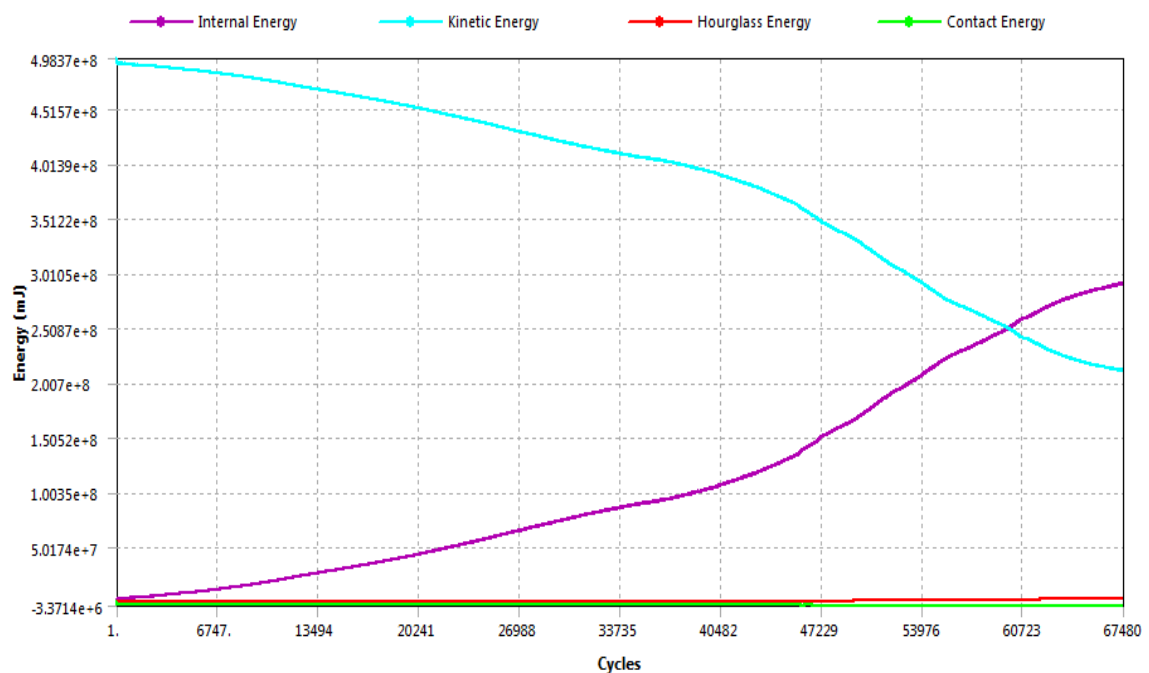
Table 4.4 shows frontal crash analysis of Existing vehicle model and modified vehicle model using explicit dynamics at 26.98 m/s and 26.01 m/s speed respectively equivalent to the real

vehicle speed at 22.22 m/s which is the real vehicle and the two models have equal kinetic energy during collision with the fixed barrier or wall. After analysis it is observed that the total deformation (27.104 mm) and equivalent stress (5285.7 MPa) at 26.01 m/s of the modified vehicle model are less relative to the existing vehicle model as shown in table 4.4. As the mass of two models shown in table 4.4, the modified vehicle model is somewhat heavier than the existing vehicle model due to the additional features of crashing load absorber and crashing load absorber supporter. But this weight difference is not that much large and it has negligible effect compared with the accident that occurred on the vehicle without using any absorber mechanism as shown in the deformation size and equivalent stress value of the existing vehicle model.

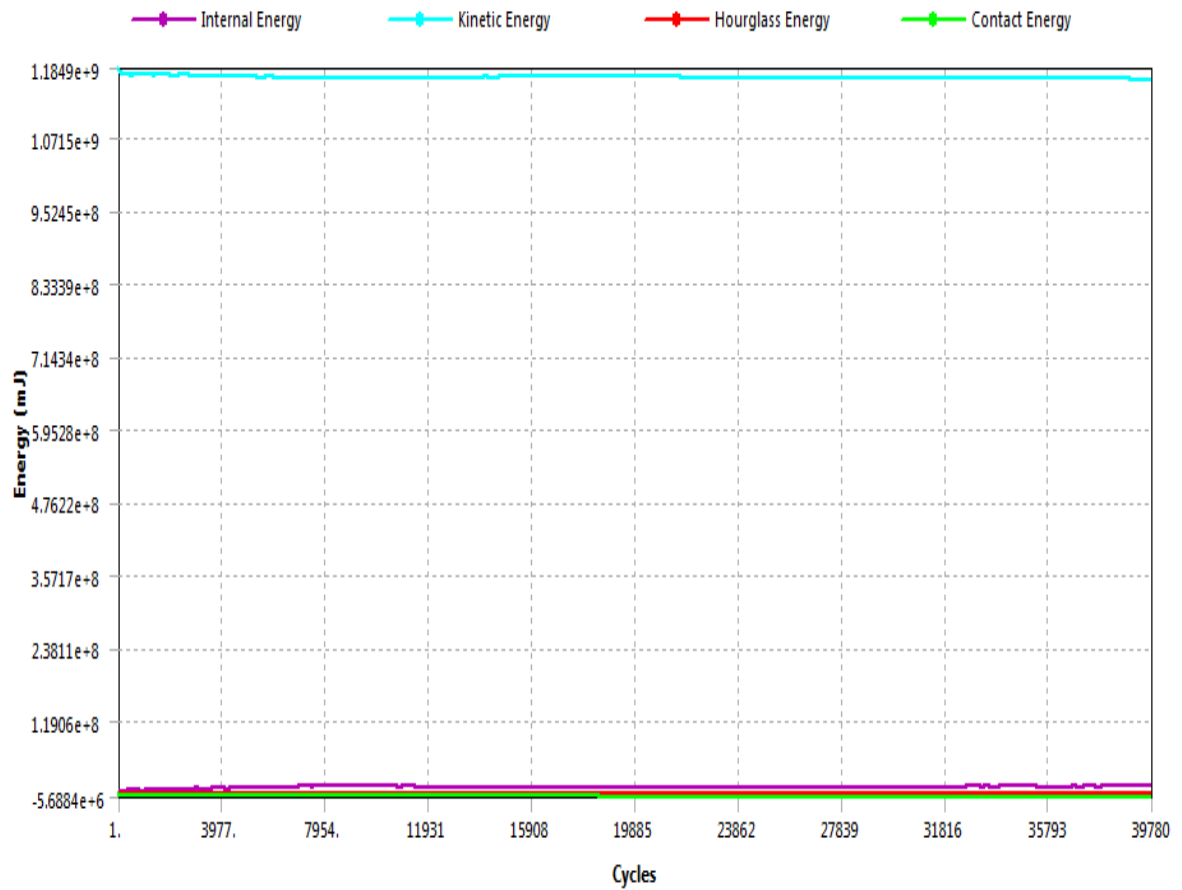
Therefore, the modified vehicle model have better results on total deformation and equivalent stress values compared with the existing vehicle model.

Figure 4.10 shows how energy is converging or induced. The internal energy of the models starts to increasing as it is been crashed and the kinetic energy of the models starts to decreasing because of impact with the rigid barrier. The amount of internal energy and kinetic energy of the two models is different depending on the size of deformation as shown in the figure 4.10.

Energy Summary



(a) For existing vehicle model

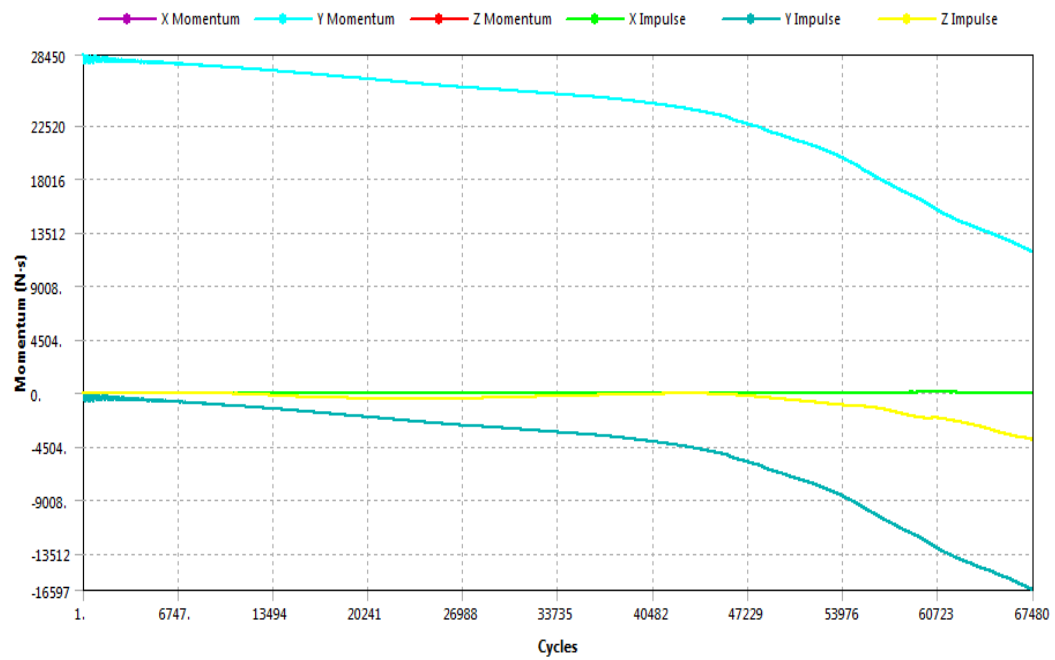


(b) For modified vehicle model

Figure 4.10: Energy summary of crash analysis

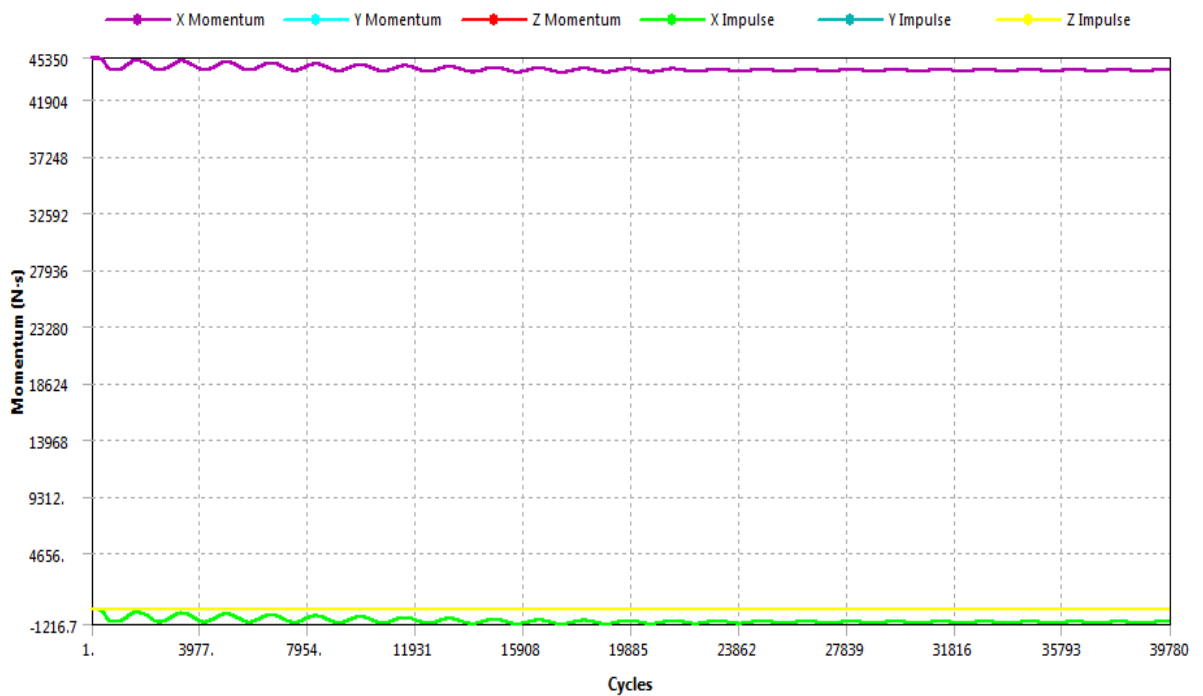
Figure 4.11 shows the impulse-momentum summary of the crashing analysis. The directions of motion for the existing and modified vehicle models are in the Y-axis and X-axis as shown in the figure. Due to the impact of the two models with the rigid barrier, the momentum value starts to decrease in the direction of motion. The impulse for the two models also decrease from zero to negative value. The negative value comes from the change in momentum due the final velocity is less than the point of impact initial velocity. Figure 4.11 also shows an impulse in Z-direction. This value comes from the unsteady impact condition and the surface roughness condition.

Momentum Summary



(a) For existing vehicle model

Momentum Summary



(b) For modified vehicle model

Figure 4.11: Impulse-Momentum summary of crash analysis

5.0 SUMMARY, CONCLUSION AND RECOMMENDATION

5.1 Summary

In this thesis work the frontal crash analysis of Toyota HILUX LN106-0136567 is done. The analysis is performed on the existing vehicle and modified vehicle models. Both the existing and newly modified models are modeled using CATIA V5 and imported into ANSYS workbench in the form of .IGES or .STEP file for detail analysis. Then this .IGES or .STEP files are analyzed using static structural and explicit dynamics successfully.

The modified vehicle model which contains in addition to the overall chassis and body frame, the crashing load absorber assembly and crashing load absorber rear end supporter. The crashing load absorber assembly contains cylinder, piston and two springs as expressed in the above sections. The crashing load absorber, crashing load absorber piston and crashing load absorber rear end supporter bending or deformation and stress analysis performed by using static structural analysis in the ANSYS workbench. The overall frontal crash analysis of existing vehicle model and modified vehicle model is done using explicit dynamics.

5.2 Conclusion

As the results shown in the analysis;

- The piston, rear end supporter and crashing load absorber assembly can resist or withstand the coming impact load up to 11275 N without any failure.
- The explicit dynamics analysis result shows, the modified vehicle model total deformation and equivalent stress values are less than the existing vehicle model.
- Therefore, the modified vehicle model is preferred than the existing one.
- The forward motion of drivers and passengers will be reduced during vehicle collision.
- This is due to the installation of the absorber mechanism which absorbs the impact load slowly and as a result the vehicle decelerate moderately.
- The other thing in this analysis using (installing) crashing load absorber minimizes the overlap sizes of vehicle front parts during collision.

5.3 Recommendation

- During crashing load absorber installation on the vehicle must be consider structural comfort; crashworthiness enhancement, good aerodynamic and ergonomic features.
- The material effect is not considered in this study. The absorber mechanism or all the newly added systems must made from stiff, higher in absorbing impact load and light in weight material.

5.4 Future Work

- ✓ Human life is priceless and fragile. To take care for this fragile and priceless human life, many researchers tried to put their findings on the vehicle front, side, rear and rollover impact analysis especially on appropriate material depend on the vehicle type and other considerations.
- ✓ The research work considers only inline frontal collision of the vehicle. Thus offset, side impact as well as roll-over impact analysis will be considered.
- ✓ On the other hand this paper is only based on computational impact simulation scenarios, from practical perspective it is required to test the model practically.

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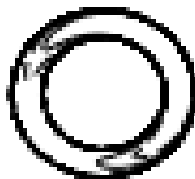
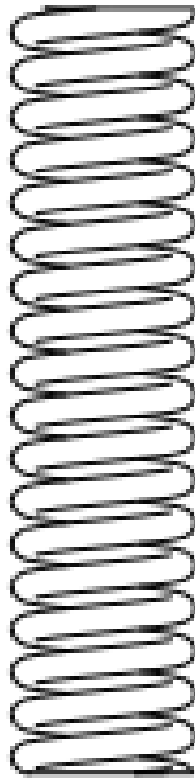
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APPENDIX-A

APPENDIX - A

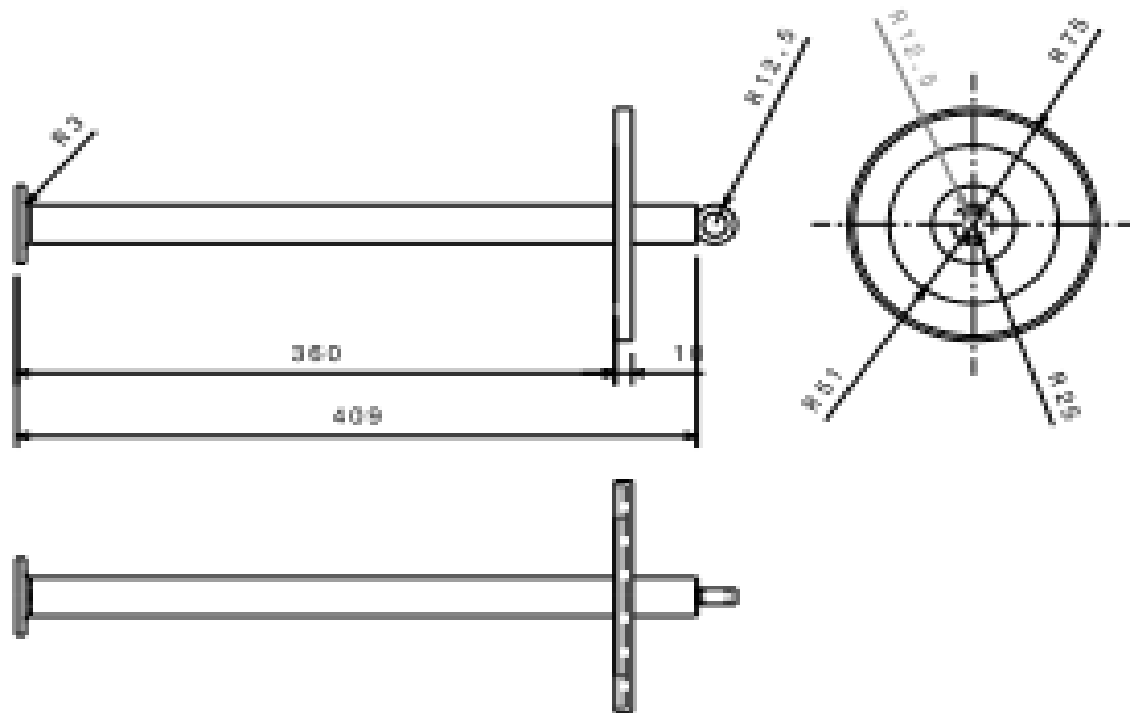


Parameter	inner spring	outer spring
mean dia.	75	125
outer dia.	90	150
wire dia	15	25
free length	328	796.18
pitch	20.5	31.8
no. of coil	17	26

spring		ASTU			
		INNER AND OUTER SPRING			
DRAWN BY Ermas	DATE 1/18/2018				
CHECKED BY XXX	DATE XXX	SIZE A4	DRAWING NUMBER 3		REV X
DESIGNED BY XXX	DATE XXX	SCALE 1:2	WEIGHT(kg) 0.16		SHEET 1/1

APPENDIX-B

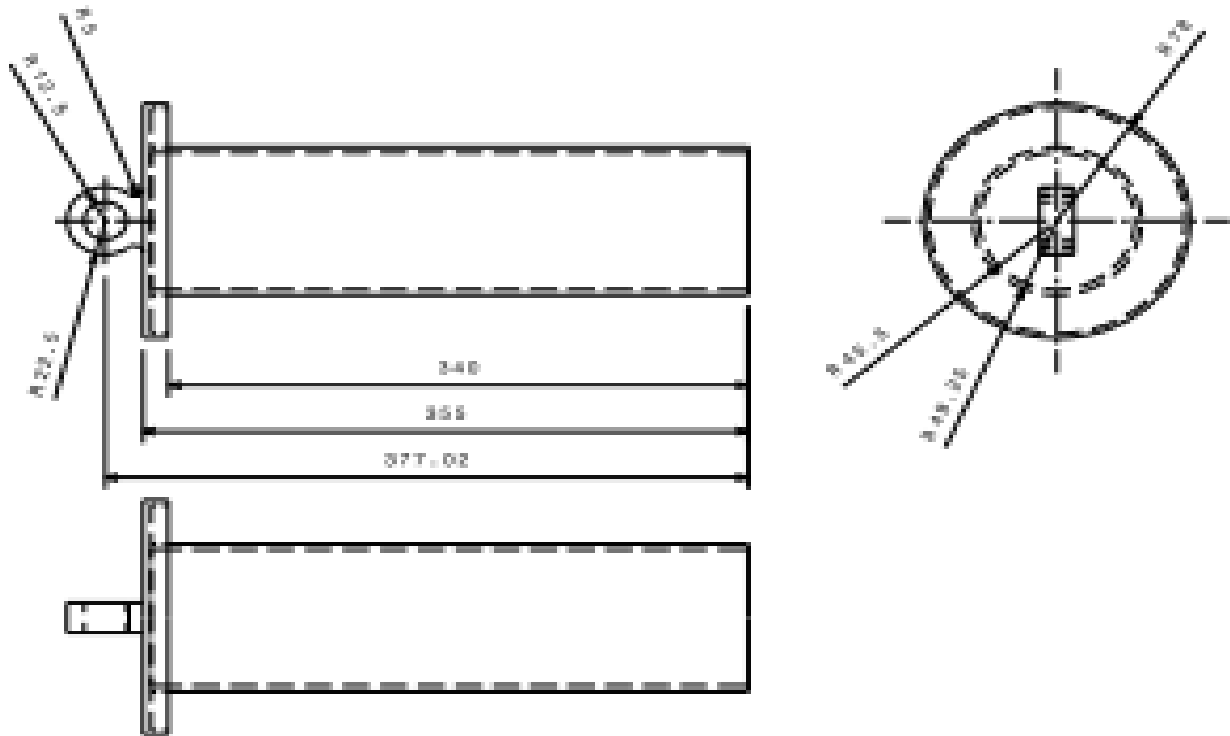
APPENDIX-B



DESIGNED BY: Ermiyas				I	—	
DATE: 1/15/2016				H	—	
CHECKED BY: XXX				G	—	
DATE: XXX				F	—	
SCALE: A4		PART MODELING		E	—	
				D	—	
SCALE	REVISION (No)	DRAWING NUMBER		SHEET	C	—
1:4	2.63	Absorber piston		1/1	B	—
This drawing is our property. It can't be reproduced or communicated without our written agreement.				A	—	

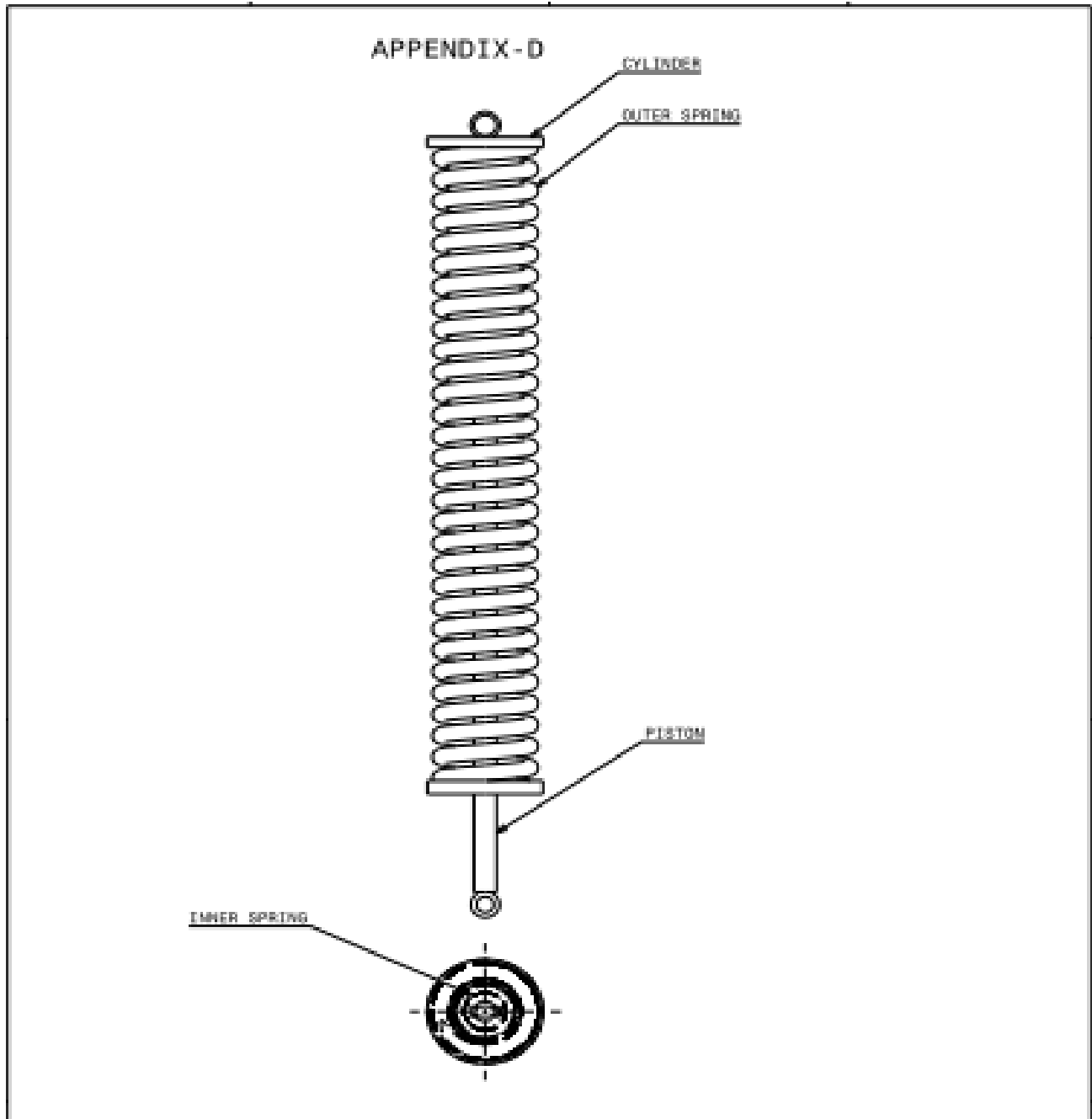
APPENDIX-C

APPENDIX-C



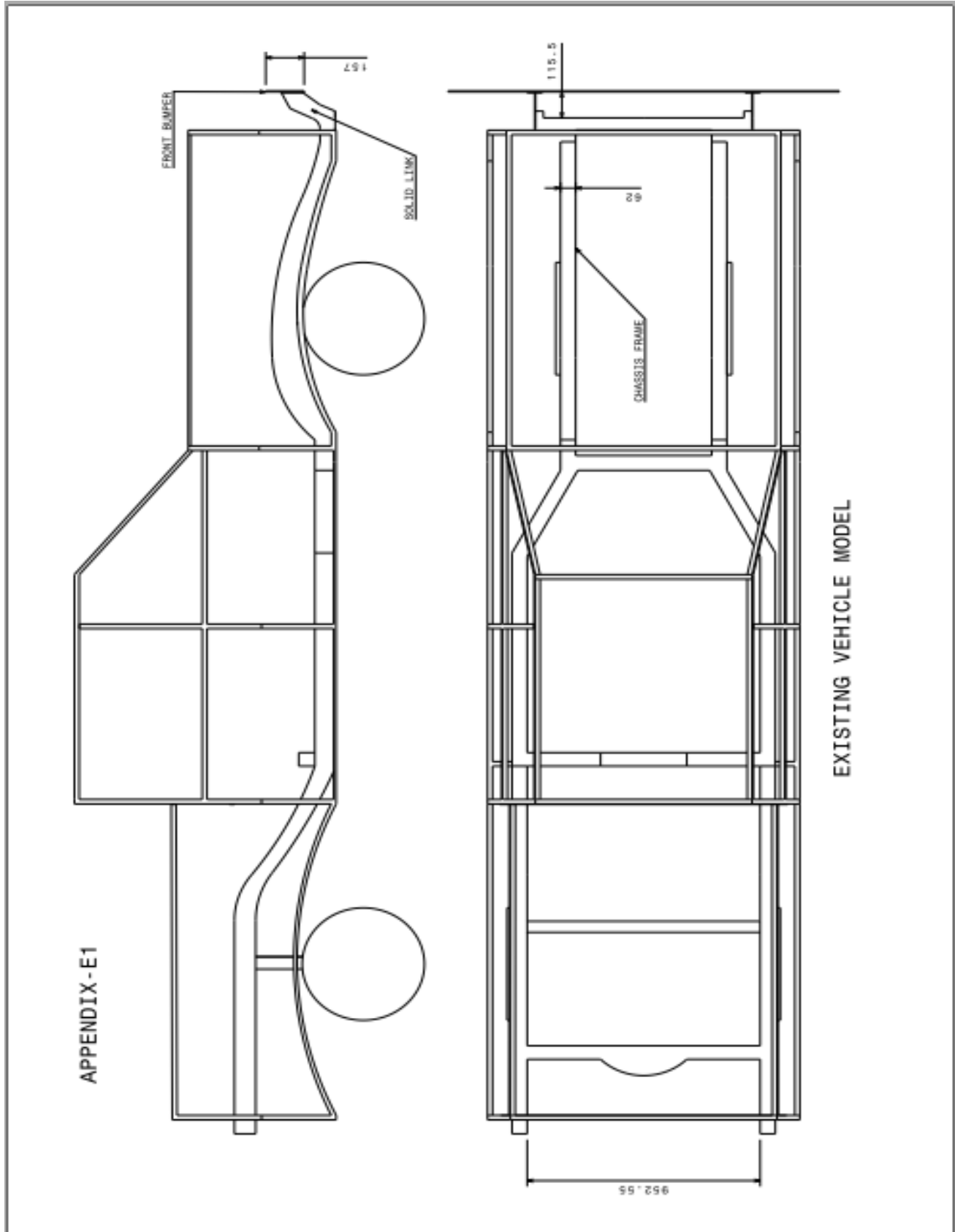
FOR PARTIAL FULFILMENT OF MSc THESIS		ASTU			
		ABSORBER CYLINDER			
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CHECKED BY XXX	DATE XXX	SCALE 1:4	WEIGHT (kg) 4.12	SHEET 1/1	
DESIGNED BY XXX	DATE XXX				

APPENDIX-D



ASSEMBLY		ASTU			
DRAWN BY Ermas		ABSORBER ASSEMBLY			
CHECKED BY XXX		DATE 1/18/2018	SIZE A4	DRAWING NUMBER 6	REV X
DESIGNED BY XXX		DATE XXX	SCALE 1:5	WEIGHT(kg) 15.82	SHEET 1 / 1

APPENDIX-E1



APPENDIX-E2

