Addis Ababa Institute of Technology

School of Mechanical and Industrial Engineering

Structural Design, Modeling and Rollover Stability Analysis of Four Wheeler Bajaj (Bajaj Qute)

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(Mechanical Design)

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Declaration

This is to certify that the thesis presented by Abiy Debebe Sime, entitled as "***Structural Design, Modeling and Rollover stability analysis of Bajaj Qute***" and submitted to the School of Mechanical and Industrial Engineering in the partial fulfillment of the requirements for the award of the degree of masters of science in Mechanical Design Engineering with the regulations of the university, and meet accepted standards with respect to originality and quality.

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By

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Preference

The task described in this thesis was carried out at Addis Ababa University, School of Mechanical and Industrial Engineering as a part of the education to Masters of Science in Mechanical Design. The basis of this thesis was to analyze the existing Bajaj qute shortcomings and to perform structural design, rollover stability, and impact analysis that can assuage existing shortcomings. The Bajaj Qute main body structure is a key element which going to be analyzed in this thesis.

Table of content

List of Table

List of Figure

Acknowledgment

Foremost, I would like to give gratitude to God for the endless blessings and gifts. I would like to give thanks to Dr. Ermias G. Koricho for his Professional, technical guidance and support throughout this research, with the absent of his continuous corrections and positive comments this thesis work wouldn't have been fortunate. I am grateful to Mr. Hairedin Ismael for his valuable advice and for always having time for discussions about my research works. I want to express my sincere appreciation and gratitude to Mr.Araya, chair of mechanical design stream for continues support in any manners. Last but certainly not least, I would like special thanks all members of SMiE vehicle research center and friends for their encouragement, and support with different materials and ideas.

Abstract

Four-wheeled Bajaj Qute becoming a major means of transportation for short distances in Ethiopia. The current structure of the Bajaj Qute extends the Vehicle not be used in a most populated area like Addis Ababa which indicate that there is strong room for improvement in this huge market of the vehicle. The primary objective of this research is structural design, modeling and rollover stability analysis of four-wheeler Bajaj (Bajaj Qute) sated objective primarily geometrical modeling using SOLID WORK 18 software of existing Bajaj Qute structure was developed with wheel base vs truck width ratio of 1.47 and following the same procedure model-2 of Bajaj Qute structure with wheel base vs truck width ratio of 1.27 was developed.

Based on formulated geometry static structural analysis for four basic loading case in addition to torsional stiffness has been performed for two models of Bajaj Qute structure by employing ANSYS 19.1 software and for impact analysis ANSYS 19.1 with explicit dynamic was utilized and three model of frontal panel with the same dimensions but having different number of holes with hole size of 10mm has been developed.

Generally, from static structural analysis of models there is increment of torsional stiffness of Bajaj Quit structure with 33%. Using static stability factor and height of center of gravity relation for rollover stability analysis, it is found that angle of tilt (critical angle) for existing Bajaj Qute structure was ϴ=53.230 whereas for new model it becomes ϴ=56.650 which indicate that new model of Bajaj Qute have capable to withstand rollover if accident encountered and for impact simulation of frontal panels of three models, it is found that internal energy developed for models varies while of thin-walled tube (hollow) in frontal panel and the variation is accordingly that is 10.4KJ , 14.82KJ and 19.4KJ respectively .

Key Element: Bajaj Qute, torsional stiffness, rollover stability, SOLID WORK 18.0, ANSYS workbench 19.1, static stability factor, internal energy.

CHAPTER ONE

INTRODUCTION

Transportation industry plays a major role in the economic development of many nations. At present days' Vehicle industries are in great competition in design and modeling of a most effective Vehicles to satisfy the requirements brought forth by strength, stability, crashworthiness, safety, fuel consumption, cost, weight, and materials. To carry through those gaps different types of new models of Vehicles has been coming out into the market for better strength and fuel efficiency Consequently, Tetracyclic Vehicle or specifically four wheelers Bajaj Qute has been built up for the purpose of the populace as well as individual transport particularly in developing countries like Ethiopia.

Despite the fact that there are many changes and improvements done on the Vehicles structure even there are strength, stability, crashworthiness, safety etc. defects. The major means of transport in ethiopa incluide Bajaj,minibus, bus,etc[1]. The Bajaj Quite was launched in 2012 by Bajaj Auto, one of India's largest automotive manufacturers. The Qute comes under the Quadricycle category, which is a whole new category in India. The main components of the Bajaj structure parts include right side, left side, front side, a rear side, roof, and floor. The body structure, which is the main part of the body, should support the engine, seats, fuel tank and passengers[2].

Also, the structure is subjected to dynamic and impact loads which could be detrimental to the strength and rigidity of the overall structure. To make the Bajaj structure strong enough, the Bajaj manufacturer use steel of varieties of the cross-section. A simple modification of structural parts is limited by the body strength, stiffness, and crashworthiness and in order to make an improvement on the design of the structure, it is basic to analyze the structure for stresses and deformations to find parts which are subjected to maximum and minimum stresses and deformations. This indirectly tells whether parts are highly loaded or not. Therefore, understanding of this lays the foundation for the improvement of the Bajaj in weight [2,3].

One of the most important requirements in the vehicle body is that the vehicle must have a desirable strength, stability and low price and specifically in all small passengers car and in light utility vehicles(LUV) there is repetitive accident are encountered due to rolling over of vehicle[4] so, it is essential to consider four-wheeled Bajaj Qute for this research to be conducted, which is a one means of public transport for short distance in particular for middle-income families and city in Ethiopia and still continue. Four wheeler Baja Qute also is known in the rest of the world for public transport particularly in the entire Africa and Asia continent [3].

1 Background

1.1 Basics Definition of terms used in any Vehicles

The Automotive structure developed from different component beyond those portion of Vehicle there are also supportive parts including light, air conditioner or heater, stereo, wiper, etc. Of the Automotive which enables the vehicle to a movie in good manners. The followings are the basic and main element that exist in any vehicles.

i. The Body

The body shell is a fairly complex assortment of large steel sections. These sections have been stamped into specific shapes which make up the body any vehicles. The body also has one other job which is usually important to the owner[5].

Figure 1 shows the shell body of a typical vehicle structure

ii. Frame

The frame is defined as a fabricated structural assembly that supports all functional vehicle systems. This assembly may be a single welded structure, multiple welded structures or a combination of composite and welded structures. Depending upon the application of loads and their direction, chassis is deformed in a respective manner briefed as follows it is the strong metal structure that provides a mounting place for the other parts of the vehicle[6].

iii. Chassis

The chassis is formed by the frame with the frame side members and cross members. supports the body and often used when referring to a vehicle's frame and everything mounted to it except the body tires, wheels, engine, transmission, drive axle assembly, and frame. It is the most crucial element that gives strength and stability to the vehicle under different conditions. Automobile frames provide strength and flexibility to the automobile. The backbone of any automobile, it is the supporting frame to which the body of an engine, axle assemblies are affixed. Tie bars, that are essential parts of automotive frames, are fasteners that bind different auto parts together [6].

Figure 2 shows a typical vehicle chassis structure

1.1.1 Vehicle Design

The very first stage of vehicle production has to be designed. Design can be considered as an activity to find the best (optimal) solution to an engineering problem within certain constraints. The requirements for modern Cars and heavy vehicles cause many tasks in vehicle design. Beside the fundamental tasks as proper identification of engine, transmission system, steering, suspension, brakes in terms of safety, utility and comfort the material properties and structure geometry become more and more important. It has to be emphasized the role of endurance and durability in design and manufacture of the reliable vehicle[7].

a) Vehicle Structural Requirements

The structural requirements of any vehicle structure can be described and summarized as follows.

- ✓ The structure must be sufficiently stiff to react the static loads and Dynamic loads without excessive deformation.
- ✓ The structure must be sufficiently strong to withstand many cycles of the applied loading without suffering from fatigue or other forms of material failure.
- ✓ The structures hold deformation in such a manner under impact load Conditions[8].

1.1.1 Type of structure of vehicles

1.1.1.1 Chassis Frame

Chassis is a French term and was initially used to denote the frame parts or Basic Structure of the vehicle. It is the backbone of the vehicle.

A vehicle without a body is called Chassis. The components of the vehicle like Power plant, Transmission System, Axles, Wheels and Tires, Suspension, Controlling Systems like Braking, Steering etc., and also electrical system parts are mounted on the Chassis Frame. It is the main mounting for all the components including the body. So it is also called as Carrying Unit[4,6].

The Chassis of a car is built to support the body. It includes the frame, wheels and working components of the vehicle. The rigid structure of the frame provides protection during an accident. There are several types of vehicle structures. Structure selection is an important point since it will define all the following stages of vehicle development. There are different choices, monocoque, semi-monocoque, body on frame. Other kinds of structures are the backbone and ladder chassis. Each one has advantages and disadvantages[8].

1.1.1.2 Monocoque Structure

Monocoque Structures consist of sheets, Beams or other kinds of metal components united by Welding and resulting in a rigid Structure in one piece. Monocoque Chassis are used now a day for the majority of the Vehicles. It is efficient in terms of using space, so the interior space is wide. It is not cost effective to manufacture them in small quantities; however, it is easy to mass produce. They might be economically produced due to the high automation which allows their manufacture[9].

Figure 3 typical passenger's car of monocoque structure

In a Monocoque Chassis, the Stress generated by the Vehicle during motion is being distributed among the Structure and does not form localized stress which may have a higher value of deformation. The depth of a Structure such as a space frame can improve the stiffness and in the integral Structure, the whole side frame with its depth and the roof are made to contribute to the vehicle bending and torsional stiffness. Typical passenger car integral structure from Monocoque structure uses more metal material than any other type of structure, consequently, it is heavier[8].

This structure is the best configuration to protect the passengers in a rollover, and generally in any kind of collision because it provides more stability in case of an accident. Nevertheless, after having an accident the reparation is quite difficult. Less noise and vibration are produced compared with any other type of structure. One disadvantage is that is difficult to design different bodyworks from the same vehicle and any kind of reform in the structure are almost impossible to implement. Basic different between Unibody and Monocoque Structure[10].

i. Unibody Structure

Structure Frames are the preferred construction for most cars on the market today. In a Unibody design, the frame, the floor plan, and the chassis of the vehicle are made as a single structure. This combination allows for a lighter and more rigid frame than the body-on-frame architecture that was used in the past[9,10].

ii. Monocoque Structure

A monocoque frame is basically a "skin" that supports its load by distributing tension and compression across its surface. It lacks a load carrying internal frame. Monocoque frames have been used sparingly in racing and luxury vehicles. In 1992, the McLaren F1 was designed with a carbon-fiber monocoque body[7].

1.1.1.3 Space Frame (Body on Frame)

The body on the frame is the oldest structure system used and nowadays it only applies in the Construction of commercial vehicles and off-road vehicles. It consists of two different structures that are the frame and the body the frame is a structure consisting of one or more metal beams along the vehicle, connected by crossbars welded, screwed or riveted, arranged transversely or diagonally[8].

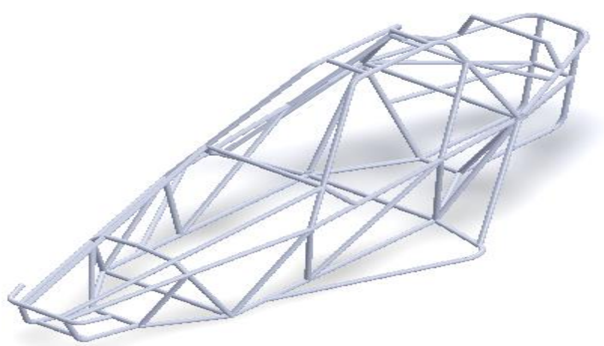


Figure 4 shows structure of space frame of vehicles

This element has a high strength and the mechanical components and the bodywork are attached to it. The body forms the outer shell of the vehicle and is screwed to the frame. It has not any strength feature. This system allows some advantages.

It has a great strength to carry heavy loads and a high rigidity to withstand high elastic and dynamic stresses. In all frames, till now length in one dimension is very less compared to the other two dimensions. Increasing depth increases bending strength. Beam elements carry either tension or compressive loads[9].

1.1.1.4 Backbone Structure

A backbone Structure is the simplest structure design. It consists of a sturdy tubular backbone that joints the front and rear axle. These chassis are fully enclosed to be rigid structure and handle all loads. The space within the structure is used to place the driveshaft in case of a front engine, rear-wheel drive layout. Further, the drive train, engine, and suspensions are all connected to each of the ends of the chassis[8].

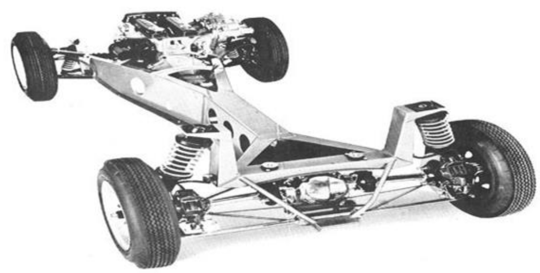


Figure 5 shows structure of Backbone chassis made from triangulated tubes

The body is built on the backbone usually made of glass-fiber. Almost rear wheel drive and front engine vehicles use backbone chassis. Main backbone is a closed box section. Rotated beams at front and the rear extent to suspension mounting points. Transverse beams resist lateral loads; Backbone frame: bending and torsion beam; Splayed beams: bending; Transverse beams: tension or compression.

1.1.1.5 Ladder Frame

A ladder Frame is a simplest and oldest frame used in modern Vehicle construction. It was originally adapted from “horse and buggy” style carriages as it provided sufficient strength for holding the weight of the components. Larger beams could be used if there were higher weight capacity required. The engine of the vehicle using this ladder frame is placed in the front or sometimes in the rear and supported at suspensions points. Their constructions consist of two longitudinal rails interconnected by many lateral or cross braces, typically made from round or rectangular tubing or channel[8].

Figure 6 shows the structure of the leaf spring ladder frame assembly

The longitude members are the main stress member. They deal with the load and also the longitudinal forces caused by acceleration and braking. It can use straight or curved members. The lateral and cross members provide rigidity to the structure because it provides resistance to lateral forces and further increase torsion rigidity. Body mounts are usually integral outriggers from the main rails, and suspension points can be well or poorly integrated into the basic design. Most SUV's are still used ladder chassis[9].

1.1.2 Joining method of Vehicles Structure

The joining mechanism of the automotive structure differs from type to type, a method to methods of vehicles. The processes that may be employed to join the side and cross-members are welding, riveting, and bolting.

1.1.2.1 Welding

gives the most rigid joint, but this method has so far been generally uneconomic for the production members involved, especially since machine Welding would be required to guarantee the necessary weld uniformity of individual cross- members. The most common welding type used in unibody or monocoque vehicle structure development is spot welding[9,11].

1.1.2.1.1 Spot welding

Spot welding is the most preferred and widely used method for joining metal sheets in automotive and many other industrial assembly operations. The body of a car is typically joined by thousands of spot welds. One of the many geometrical factors affecting the final geometrical outcome of the metal part assemblies is the welding process considering welding sequence used when the parts are welded together[9].

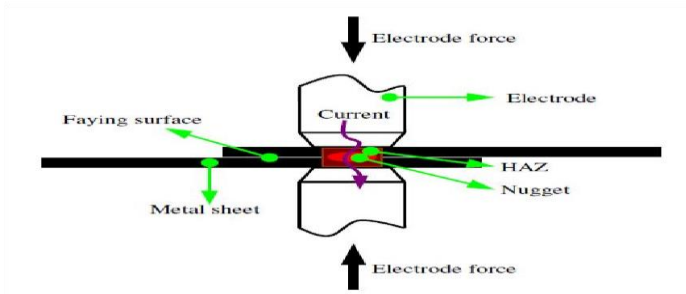


Figure 7 Schematic representation of the spot welding process

The Spot welds guarantee the strength of the Car, but their positions also affect the geometrical quality of subassemblies and the final product. In practice, the positions of the Weld points often deviate from the nominal position. A spot weld is materialized by clamping the sheets with two pincers while applying force and transmitting current, the electrical resistance of the contacting sheets generates sufficient heat at the faying surfaces to melt the metal; eventually, a nugget develops and the interface locally disappears. A spot welding gun has two electrodes, which are applied from either side of the sheet metal parts. When the parts are in contact, an electric current is applied and the result is a small spot, heated to the melting point, in which the parts are joined[13].

The loads imposed on the chassis or body structure of a passenger car or light commercial vehicle due to normal running conditions are needed to be considered in any vehicle structural design.

1.1.3 Loads imposed on vehicle body structure

1.1.3.1 Vertical symmetrical ('pure bending case') causes bending

This is loading in a vertical plane, the x–z plane due to the weight of components distributed along the vehicle frame which cause bending about the y-axis, the bending conditions depend upon the weights of the major components of the vehicle and the payload. The first consideration is the static condition by determining the load distribution along the vehicle [10].

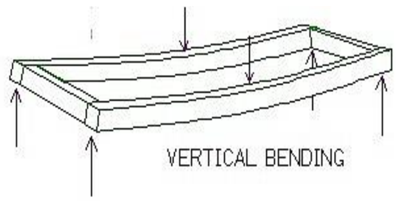


Figure 8 a representation of pure bending loading case in vehicles

The axle reaction loads are obtained by resolving forces and taking moments from the weights and positions of the components (i.e. the equations of statics). The structure can be treated as a two-dimensional beam as the vehicle is approximately symmetric about the longitudinal X-axis. The unspring masses consisting of wheels, brake discs/ drums, and suspension links are of course not included as they do not impose loads on the structure. Generally, the weight of vehicle components mounted to the frame and driver's weight accounts for vertical bending[14].

1.1.3.2 Pure Torsion loading case

The vehicle body is subjected to a moment applied at the axle centerlines by applying upward and downward loads at each axle in this case. These loads result in a twisting action or torsion moment about the longitudinal x-axis of the vehicle, the case of pure torsion can be considered simply as being applied at one axle line and reacted at the other axle[8].

The condition of pure torsion cannot exist on its own because vertical loads always exist due to gravity, as mentioned in the introduction. However, for ease of calculation, the pure torsion case is assumed. The maximum torsion moment is based on the loads at the lighter loaded axle, and its value is the wheel load on that lighter loaded axle multiplied by the wheel track.[10]

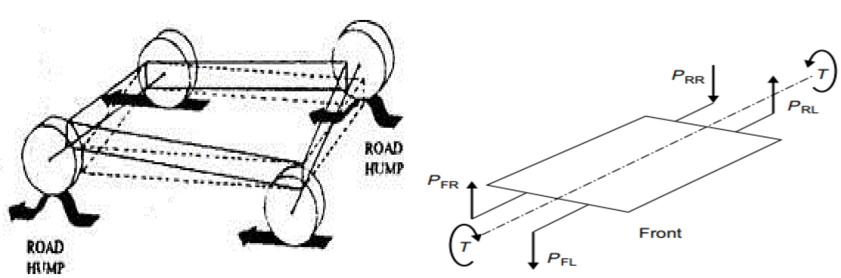


Figure 9 a representation of pure torsion loading cases in vehicles

These loads occur due to roll over bumps acting on the diagonally opposite front and rear wheels of a vehicle. Torsional loading affects the handling of the vehicle. Here, the frame can be considered as a torsion spring connecting two ends on which suspension load acts [9,14].

1.1.3.3 Combined loading case (pure bending and pure torsion)

In practice, the torsion case cannot exist without bending as gravitational forces are always present. Therefore, the two cases must be considered together when representing a real situation[10,14].

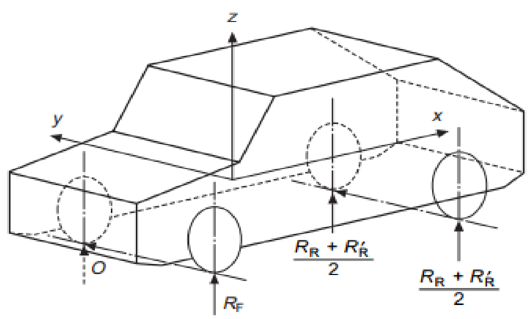


Figure 10 a representation of combination loading for bending and torsion loads

1.1.3.4 Horizontal Lozenging

When diagonally opposite wheels of a vehicle are subjected to forward and backward forces, the frame is distorted to parallelogram shape. These loads may be due to vertical variations on the roads[15].

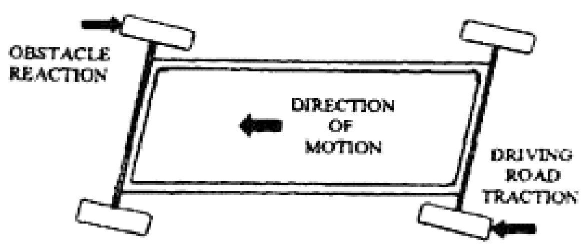


Figure 11 Illustrates that loading cases for horizontal Lozenging

Or it can occur when the vehicle is driven around a corner or when it slides against a kerb that is loads along the y-axis.

1.1.3.5 Lateral loading

This type of bending occurs due to various reasons such as side wind loads, road camber and the centrifugal forces acting due to cornering. These lateral side forces are opposed by adhesive side reactions on the wheels[15].

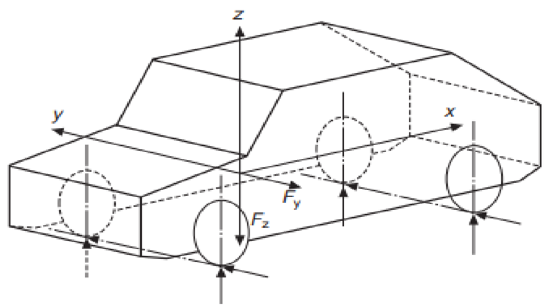


Figure 12 shows the load distribution in case of lateral loadings

1.1.3.6 Fore and aft loading

During acceleration and braking longitudinal forces are generated (along with the x-axis). Traction and braking forces at the tire to ground contact points are reacted by mass time's acceleration (deceleration) inertia forces[10]. The most important loading cases mostly considered in vehicle chassis and structural design are: Bending, Torsion and combination (bending and torsion) as these are paramount in determining a satisfactory structure.

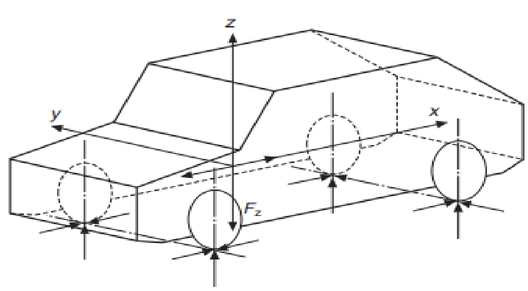


Figure 13 shows the load distribution in case of lateral loadings

The lateral loading and fore-aft loading cases require attention when designing the suspension mounting points to the structure but are less significant on the structure as a whole. Other localized loading conditions such as loads caused by door slamming, seat belt loads, etc., are not considered in this work[14].

1.1.4 Stability of automotive structure

Stability of the automobile refers to the ability that the automobile is able to run following the way is given by the driver through steering system and steering wheel when the driver doesn't feel excessively tension and fatigue, and the ability to resist interference and keeping traveling stably when encountered outside interferences. The stability of the automobile is very important to the high-speed vehicles because it can not only affect the manipulation of convenience but is also a major performance that decides the safety of the high-speed vehicles[16].

1.1.4.1 Center of gravity of vehicle structure

Center of gravity, also known as the center of mass, is that point at which a system or body behaves as if all its mass were centered at that point. Where the weight, and also all accelerative forces of acceleration, braking and cornering act through it[17].Center of gravity location can be defined as:

1.) The balance point of an object including the automotive structure.
2.) The point through which a force will cause pure translation.
3.) The point about which gravity moments are balanced.
4.) The point which if the body is hanged from it will stay balanced (leveled as it is on the ground).

1.1.4.1.1 Important of center of gravity

When attaining an analysis of the forces applied on the car, the CG is the point to place the car weight, and the centrifugal forces when the car is turning or when accelerating or decelerating. Any force that acts through the CG has no tendency to make the car rotate. The center of mass height, relative to the track, determines load transfer, (related to, but not exactly weight transfer), from side to side and causes body lean[18].

When tires of a vehicle provide a centripetal force to pull it around a turn, the momentum of the vehicle actuates load transfer in a direction going from the vehicle's current position to a point on a path tangent to the vehicle's path. This load transfer presents itself in the form of body lean. Body lean can be controlled by lowering the center of weight or the widening the car track, it can also be controlled by the springs, anti-roll bars or the roll center heights[17,18].

1.1.4.2 Rollover stability of the automotive structure

The roll stability of vehicles is one of the most important measures of vehicle safety. With the advent of new configurations of vehicles that are longer, wider and higher, conventional methods of evaluating stability are proving to be more difficult and expensive[20]. Rollovers are among the most severe traffic crashes and are of particular concern for occupants of light trucks and vans (LTVs) including pickup trucks, sport utility vehicles (SUVs), minivans and full-size vans up to 10,000 pounds gross vehicle weight rating (GVWR). While only about 3 percent of all passenger vehicle (passenger car and LTV) crashes involve rollover, according to the 2003 Fatality Analysis Reporting System (FARS), one-third of all passenger vehicle occupants who lost their lives were in vehicles that rolled over, a total of 10,376 rollover deaths. Of these, 4,433 were in passenger cars, 2,639 in SUVs, 2,569 in pickup trucks, 724 in vans, and the remaining 11 in other or unknown types of light trucks[19].

Rollover is considered when the vehicle, due to a strong collision or to the loss of control after a difficult maneuver, tumbles laterally, causing large deformations to the structure, because of all the vehicle weight pressures of the vehicle lateral in its interior direction, causing several injuries to the occupants[21].

1.1.4.2.1 Static stability factor of the automotive structure

The static stability factor (SSF) of a vehicle is an at rest calculation of its rollover resistance based on its most important geometric properties and it measures how top vehicle is.Also, it predicts the rollover propensity. The SSF is known as a vehicle truck width (t) divided by twice of the height of mass center above the ground while the vehicle is empty and at rest. This measure of rollover tendency reflects only the most fundamental relationship. It is obtained under the assumption that a vehicle is a rigid body and ignores all higher order effects, in particular, the effects of suspension and tire compliance.

In reality, vehicle suspension allows for significant movements of wheels with respect to the body, resulting in changes in halftrack width and position of the vehicle center of gravity during large lateral acceleration. For vehicles with low static stability factors, the cumulative effect of secondary factors may be sufficient to reduce the lateral acceleration threshold to the value achievable during emergency handling maneuvers[22].

1.2 Motivation

There is frequently increasing the demand for Bajaj Qute Vehicle in Ethiopia due to its cost and simplicity. But it's clear and visible that frequently toppling or tipping happened to the Bajaj Qute Structure due to the height of the Center of gravity or small in truck width of the vehicle which indicates that analysis should be performed before supply the product to customers. Even there is no research carried out on the structure of Bajaj Qute in Ethiopian context including crashworthiness behavior of the Vehicle body.

There are prominent numbers of Bajaj Qute importer in Ethiopia, primarily settled in Addis Ababa. Almost all, simply assemble the Qute body using the manual launched by manufacturing company. Still, there are no analysis performs on the structure to minimize some risks related to accident and stability problems. The motivation of this research is to study the rollover stability and impact behavior of Bajaj Qute to find better and safe structure by analyzing the structure for the case of static and explicit dynamic conditions.

1.3 Statement of the Problem

According to Fatality Analysis Reporting System (FARS) in the 2003, One-third of all passenger vehicle who lost their lives were in vehicles that rolled over, and are of particular concern for occupants of light trucks, vans (LTVs) including pickup trucks, sport utility vehicles (SUVs) and small passengers vehicles (minivans)[4] and from that fact, Bajaj Qute structure is considered for rollover stability analysis since, it's under category of small passengers vehicle and mostly available in our country and from euro test result of Bajaj Qute April 2016 structure of the Qute was judged to be less in resistance to impact load in frontal crash test[2] and referring above test, the energy absorption behavior (crashworthiness) of four wheeler vehicle (Bajaj Qute) is selected for analysis.

1.4 Objectives of the Research

1.4.1 General Objective

The main objective of this research is structural design, modeling and rollover stability analysis of four-wheeler Bajaj (Bajaj Qute). This is done by investigating the current Bajaj structure and come up with improved design to assuage the shortcomings.

1.4.2 Specific Objectives

- To develop and determine appropriate three-dimensional models of Bajaj structure using SOLIDWORKS 18.0
- To design Bajaj Qute body under different types of loads and loading conditions.
- Estimate the height of center of gravity both for existing and new geometry of Bajaj Qute structure and compare the result.
- Estimate the static stability factor (SSF) both for existing and new geometry of Bajaj Qute structure.
- To predict the deflection, stress, reaction force, developed both in existing and new model of Bajaj Qute structure using ANSYS Workbench.
- Compare the torsional rigidly existing and new geometry of Bajaj Qute structure and compare the results.
- Develop three different types of frontal panel to check impact resistance behavior of the frontal part of the vehicle structure.
- Study the crashworthiness or the energy absorption behavior of the frontal panel of Bajaj Qute structure.
- Study the effect of adding hollows in frontal panel (crash box) in energy absorption capacity of structure.

1.5 Outcomes of the Research

The overall structural modeling of Bajaj based on specifications that collected from importers of vehicles is analyzed and presented.Analyzing the Bajaj Qute structure for rollover stability and crashworthiness based on finite element methods will be examined then recommendations on farther modification on the structure of Bajaj Qute will be presented.

1.6 Methodology

The methodology that has been applied by the study are chosen in order to acquire information and derive conclusions about the structural analysis four-wheeled Bajaj Qute.

1.6.1 Data collection (Research materials)

For the purpose of this research, and in order to achieve the objectives listed in the subtopic of specific objective of the research the data collected through secondary data collection method:

- Through investigation of different published papers and journals related to structural modeling and stability analysis of vehicles.
- By browsing different motor vehicles structure and analysis that support the basic inputs gained from journals and published papers.
- By visiting conventional Bajaj Qute in different sub-city in Addis Ababa, Ethiopia to get specifications and to take detail measures of the overall dimension of the vehicle.

The research will include stability analysis, stiffness analysis, explicit dynamic analysis (impact simulating) and simulation of Bajaj frame for different loading conditions. Stiffness analysis is concerned with stress and deformation analysis of Bajaj frame using ANSYS workbench finite element analysis. The explicit Dynamic analysis incorporates crashworthiness behavior of Bajaj Qute. The research begins with modeling the Bajaj structure with the data that derived from the Bajaj Qute imports. This model is then analyzed for rollover stability and for impact property i.e. total energy developed and deformations.

1.7 Research procedures

The detail Steps followed in research preparation are listed in the following ways.

1.7.1 Data collection

The real detail drawing data as well as the specifications of Bajaj body structure components is collect from Hora corporate groups that locate in Addis Ababa.

1.7.2 CAD modeling of the structure

Based on the data collected from the company the CAD modeling is performed. For modeling of the structure of Bajaj Qute SOLID WORK 18.0 was used.

1.7.3 Finite element (FEM) analysis

The finite element analysis of structure for the required results using ANSYS Workbench 19.1 software performed and steps followed to get the output result is illustrated as follow.

- ✓ Divide structure into pieces (elements with nodes)
- ✓ Describe the behavior of the physical quantities on each element.
- ✓ Assemble the elements at the nodes to form an approximate system of equation for the Whole structure.
- ✓ Solve the system of equations involving unknown quantities at the nodes (e.g. displacements)
- ✓ Apply boundary condition according to load distribution on Bajaj Qute structure.
- ✓ Apply the boundary condition for the case of impact simulation to get the desired solution like deformation, energy absorption behaviors etc.
- ✓ Calculate the desired quantities (e.g. deformation and stresses) at selected elements.

1.7.4 Solution procedure using ANSYS

The geometry of the Bajaj Qute to be analyzed is imported from solid modeler including

1.) The element type and materials properties specified.

2.) Meshing the three-dimensional solid model will generate based on element types and property of Bajaj Qute.

3.) The boundary conditions like fixed supports, the center of gravitation, acceleration etc. And external loads are applied.

4.) The solution like total deformation, strain energy, directional deformation, torsional rigidity and etc. Is generated based on the previous input parameters. Finally, the solution is viewed in a variety of displays like in simulation or graphics display.

1.7.5 Steps of the analysis

The diagram below describes the method followed in the overall analysis of the Bajaj structure on the base of bending load and torsional stiffness. The modeling is made until the stress result is reasonably below the allowable stress for both loading cases.

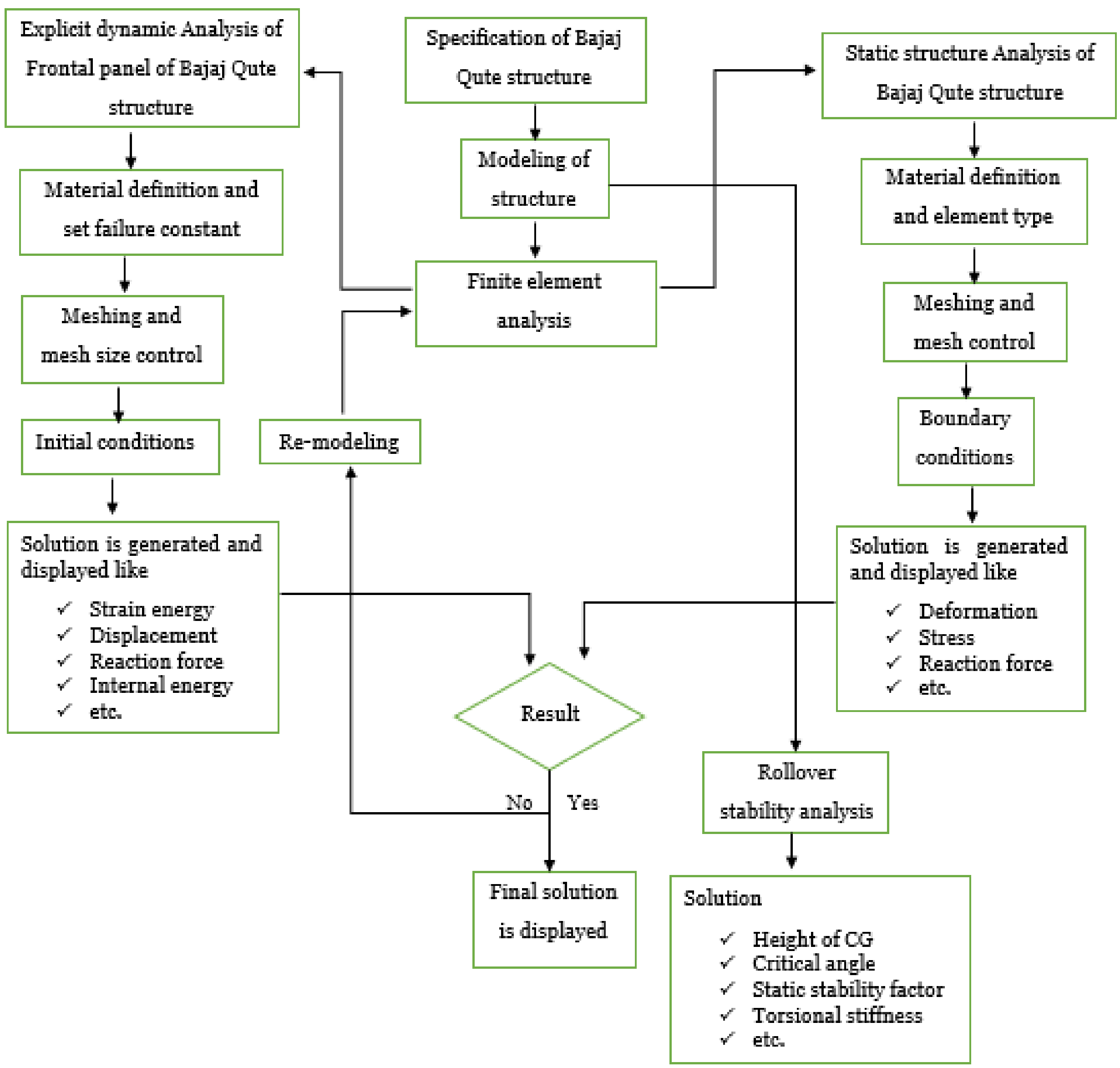


Figure 14 shows the flow diagram of a step in Analysis of Bajaj Qute

CHAPTER TWO

LITERATURE REVIEW

There have been researches on different type's vehicle structural modeling analysis, and a large number of literature on structural modeling and analysis has been published. The vehicle body structural stress analysis and dynamic loads with different loading conditions and the optimal design for any vehicles are always major concerns in the structural design of vehicles. But Bajaj Qute body has received much less research attention than other types of vehicle structural analysis due to their availability and utilization in the developed countries but its known that most Asia and Africa continent used as one basic means of transportation for short distance[2].

A four-wheeled Bajaj Qute and the word has been in use since the early 19th century. To build the structure of vehicle they initially conceive the concept from two-wheel motor bicycle and they design three wheel Bajaj auto having a capacity to carrying only three passengers at a time. While they design the structure of Bajaj consider comfortable seating for three passengers involves visual comfort, openings, access to Surroundings emerge space and location for passenger and luggage or baggage, better lighting, larger windscreen area, and better cover structure, adapting vehicle for tropical climate, weather protection in the form of canopy, door attachment and accessories modularity in Construction for assembly and disassembly for service[1,22].

To make more ergonomic they use of sheet metal and soft, flexible materials for body design but they did not consider the stiffness and crashworthiness properties of materials that indirectly affect the overall functions of structure that they built up[2]. The design of the vehicle body has evolved from a simple, all steel structure that meets the basic requirement of strength and functionality, to the current day complex and efficient structure.

2.1 Preceding Works related to load distribution on vehicle body and chassis

In *"Design and structural analysis of an off-road vehicle "*by P. Rastogi, I. Sharma, 2016 studied designing a frame for an ATV by considering the following basic loading conditions I*) "Longitudinal Torsion*" loading cases roll over bumps acting on diagonally opposite front and rear wheels of vehicles II) "*Vertical bending* "lading case when weight of vehicle components mounted to the frame and driver's weight III) "*Lateral Bending* "occurs due to various reasons such as side wind loads, road camber and the centrifugal forces acting due to cornering IV) "*Horizontal Lozenging*" is when diagonally opposite wheels of a vehicle are subjected to forward and backward forces.

"In the study of "*the frame analysis and testing for student formula*" by Tanawat Limwathanagura, Chartres Sithananun 2011, studies for the analysis and testing for determining the torsional stiffness of the student formula's space frame. The space frame for Chulalongkorn University Student Formula team used in 2011 TSAE Auto Challenge Student Formula in Thailand was designed by considering required mass and torsional stiffness based on the numerical method and experimental method. The numerical result was compared with the experimental results to verify the torsional stiffness of the space frame. It can be seen from the large error of torsional stiffness of 2011 frame that the experimental result cannot verify by the numerical analysis due to the difference between the numerical model and experimental setting.

In the study, "the numerical analysis and experiment of the space frame" same 2011 frame model are performed by improving the model setting. The improvement of both numerical analysis and experiment are discussed to confirm that the models from both methods are same. After the frame was analyzed and tested, the results are compared to verify the torsional stiffness of the frame. It can be concluded that the improved analysis and experiments can be used to verify the torsional stiffness of the space frame.

"*Design and analysis of car Chassis* "by Mohamad sazuan bin sarifudin, 2012 was to devolve car chassis frame to study the effect of load that applied in term of driver weight, the car body, and the equipment. According to the study to avoid any possibilities of failure of the structure and thus to provide enough supporting member to make the region stronger in term of deformation. Finite element analysis enables to predict the region that tends to fail due to loading. Besides that, need to utilize the feature of CAE software.

In "*Modelling and Dynamic Analysis of Bus Body Built in Ethiopia*" by Besufekad Getachew describe the way of analyzing of the structure of Bus for two main loading cases, bending load and torsional stiffness for the improvement on the structure of Bus. This study examines the stresses and deformation responses of a typical bus structure during application of the load in service. To do so mass of passengers for dynamic analysis was taken as 65 kg.

"Design and Analysis of a 3-Wheeler Integrated Monocoque Chassis" by g. Sawan kumar1, Shuvendra Mohan, 2016 the design and development of integrated chassis of three wheeler prototype vehicle chassis is done, to convert the three-wheeler passenger vehicle chassis into a load carrier chassis The parameters checked in the analysis are the displacement of the chassis structure and stresses under static condition and for analysis case three types of loading case pure bending, pure torsion and combined loading are considered. The modeling of new chassis is done by using PRO-E and FEA by using ANSYS. Specifications of materials selection become a priority in order to construct the new chassis. The best design with minimum self-weight, maximum load capacity, and minimum deflection under static loading were then identified based on the results obtained through FEA.

2.2 Preceding Works related to rollover stability analysis of vehicle

"*Trends in Static Stability Factor of Passenger Cars, Light Trucks, and Vans Rollover"* by N. hignway traffic safty Adminstration 2005. Paper describe basic means of assessing rollover risk is the static stability factor (SSF), a measurement of a vehicle's resistance to roll over. The higher the SSF, the lower the rollover risk. Most passenger cars have values in the 1.30 to 1.50 range. Higher-riding SUVs, pick-up trucks, and vans usually have values in the 1.00 to 1.30 range. This report tracks the trend in SSF over time, looking in particular at changes in various passenger vehicle types.

"*Static stability factor test results for road and transport safety"* by Raphael Grzebieta, Illustrates that the lower the SSF number, the lower the vehicle's resistance is to roll over if the applied side force is sufficiently high. Lower critical angle more topple and contrariwise (and is also a function of wheelbase and track width, depending on tip-over direction) this is determined by using static stability factor (SSF) as input. Lateral SSF values across all road going light passenger vehicle types typically range from around 1.00 to 1.50. Most passenger cars have values in the 1.30 to 1.50 range. Higher-riding sports utility vehicles (SUVs), US pick-up trucks, and vans usually have values in the 1.00 to 1.30 range. Heavy Trucks are in the range of 0.35 to 0.5 depending on loading Stability of vehicles also largely depend up on the ratio of wheel base vs truck width or in common sense ''Golden ratio "and for most passengers' car it's range between 1.2 to 1.7 and commonly golden ratio 1.68 preferred to use.

2.3 Preceding Works related to impact analysis on the Vehicle body.

"*Crash simulation using Ansys explicit dynamics*" by Badita, Oana-Georgiana si Rautescu, Diana-Iulia, 2016 deals the development, modification, and analysis of a finite element model of the car body. A simple FE model is developed in ANSYS, cleaned in ANSA and it is solved for full frontal impact in ANSYS LS-DYNA explicit code.

"*Novel Safety Requirements and Crash Test Standards for Lightweight Urban Vehicles*" by David Egertz,2011 state and examine road safety for Urban Light-weight Vehicle (ULV) to find critical crash scenarios from which future crash testing methods for urban vehicles can be derived. The term ULV is specific to this report and is the title for all engine-powered three- and four-wheeled vehicles categorized by the European Commission. Other attributes than the wheel geometry is engine power and the vehicles unlade mass and according to the standard for Light Quadricycle Four-wheeler with a maximum mass of 350-400 kg with an internal combustion engine capacity of 50cc or less, an internal combustion or electric engine with maximum power of 4kW and/or a the maximum impact speed for analysis should not be more than 45km/h (12.5m/s).

"*Design optimization of vehicle components for full frontal crash* "by Pulkit Sharma, Ram Mohan Telikicherla, Sai Nizampatnam, 2016. State that energy absorption plays an important part during a frontal crash for passenger safety. Optimization of the frontal components is the key to increasing energy absorption due to large parameters. A full frontal crash of Volvo V40 model has been done in this project using ANSYS Explicit Dynamics module. Simplified geometry to represent major energy absorption mechanism of the individual components involved in the frontal crash have been modeled on ANSYS modeler and analyzed under standard test condition.

In the study "*Design and Analysis of an Automotive Front Bumper Beam for Low-Speed Impact* "by Mahesh Kumar V. Dange, B. Buktar,2015 simulation of bumper beam is done under low-velocity impact as per the standards of automotive stated in E.C.E. United Nations Agreement, Regulation no. 42, 1994. Characteristics are compared with each other to find the best choice of material. The results show that an M220 material can minimize the bumper beam deflection, the effect of passengers in the impact behavior is examined.

Passenger cars had the lowest rollover fatality rate (23 percent of fatalities were in vehicles that rolled over), while (small utility vehicles) SUVs and (small passenger cars) vans had the highest, 59 percent. Similarly, according to the General Estimates System (GES), 6 percent of passenger car occupants who were injured were in vehicles that rolled over. LTV rates of rollover-related injured occupants were higher, 9 percent of those injured in vans, 13 percent in pickup trucks, and 20 percent in SUVs. Looking at occupant fatalities per 100,000 registered vehicles, passenger cars had the lowest rate at 3.69, with vans similarly low at a rate of 3.83. The rates for pickup trucks (7.18) and SUVs (10.22) were much higher.1 clearly, rollover crashes are a major safety problem for all classes of light vehicles, particularly LTVs.

In "*A novel tailor-made technique for enhancing the crashworthiness by multistage tubular square tubes*" by Zahran Pu Xue M.S. Esa, M.M. Abdelwahab, 2018.Study and demonstrate that energy absorption systems reveal various techniques for enhancing the crashworthiness of a structure and proposes tailor-made technique for crashworthiness design by performing a combination of two or more energy absorption techniques to fulfill the crashworthy designer requirements. Here verities of structure is used to increase the energy absorption behavior of structure like hollows.

Here hollows (thin-walled tube) used as an energy absorber which is one of efficient techniques for enhancing the crashworthiness of vehicle's elements such as, crash boxes which are fixed on the front rail, as shown in. Thin walled tube has high energy absorption characteristics as it dissipates the kinetic energy in various forms such as plastic deformation, friction, fracture, shear, bending, tension, torsion, and etc.

2.4 Conclusion from all preceding works

From all literatures, it is visible that in structural design and simulation of any vehicle there are a variety of loading condition that should be considered (pure bending load, pure torsion load, combined loading (pure bending pure torsion loading), horizontal Lozenging and etc.) to conclude whether the given structure is stiffer or not without exciding the yield strength of the material.

Rollover stability of any vehicle can be affected by basically by critical angle of vehicle, height of center of gravity and by static stability factor so, while considering the rollover stability analysis of any vehicle those parameter should be kept according with types of designed vehicles.

In impact analysis of frontal rail (frontal panel) approximately up to 55.3% of kinetic energy in the vehicle will be absorbs in case of frontal crashes as a result it is named the most effective parameter in the design of vehicle's safety and adding extra energy absorber is one of the efficient techniques for enhancing the crashworthiness of vehicle's elements such as, crash boxes which are fixed on the front rail and stated that hollows(Thin walled tube) has high energy absorption characteristics as it dissipates the kinetic energy in various forms.

CHAPTER THREE

MATERIALS, CONDITIONS, AND METHODS

3.1 Material used to Design Bajaj Qute Structure

The material used for analyzing the structure of Bajaj Qute is Structural steel sheet and other material properties are listed below in the table.

Table 1 material properties table of Structural steel sheet (St-60) for static structure[24]

Structural steel sheet (St-60)		
No.	Mechanical properties	Values
1	Yield strength	550 Mpa
2	Ultimate tensile strength	630 MPa
3	Young's modulus(E)	200000 MPa
4	Poison 's ratio(v)	0.3
5	Density(ρ)	7850 kg/m3
6	Elongation	20%

N.B:-The material specified in the table used for both static structure and explicit dynamic analysis. But for the case of explicit dynamics analysis (impact analysis), there is an additional parameter that is Johnson cook failure, that is those Johnson cook failure constant will be added to material properties of Structural steel sheet.

3.1.1.1 Johnson cook material constant for Structural steel sheet (St-60)

3.1.1.1.1 Introduction to Johnson-cook strength model

The expression for the equivalent stress-plastic strain curve of the material depends on the current plastic strain rate and the temperature[25].

$$\sigma_e = \left[A + B(\varepsilon_e^p)^n\right]\left[1 + C\ln\left(\frac{\dot{\varepsilon}_e^p}{\dot{\varepsilon}_{to}^p}\right)\right]\left[1 - \breve{T}^m\right]$$ ------------------------Equation-1

Where:

ε - Equivalent plastic strain,

A-Elastic limit, n – Exponent of strain hardening,

B – Modulus of strain hardening, m – Exponent of thermal weakening,
$\dot{\varepsilon}_e^p$- Reference strain rate, $\dot{\varepsilon}$ - The non-dimensional speed of plastic strain,

$$\breve{T} = \frac{T - T_r}{T_m - T_r}\text{, --- Equation-2}$$

Where:

$\breve{T}$ - Homological temperature, T_r – Reference temperature
T - Current temperature, T_m - Melting temperature

3.1.1.1.2 Introduction to the Johnson-Cook Failure Model

The Johnson-Cook failure model can be used to model the ductile failure of materials experiencing large pressures, strain rates, and temperatures. This model is constructed in a similar way to the Johnson-Cook plasticity model in that it consists of three independent terms that define the dynamic fracture strain as a function of pressure, strain rate, and temperature[32,33].

The ratio of the incremental effective plastic strain and effective fracture strain for the element conditions is incremented and stored in custom results variable, damage. The material is assumed to be intact until damage = 1.0. At this point, failure is initiated in the element. An instantaneous post-failure response is used. Since the evolution of damage is related to the plastic strains, it is modeled as given below It is modeled as given below[26].

$$D=\begin{cases} 0, \text{when } \varepsilon p \leq \ \varepsilon p, d \\ \frac{Dc}{\varepsilon f - \varepsilon p,d}, \text{when } \varepsilon p > \varepsilon p, d \end{cases}\text{-------------------------------Equation-3}$$

Where Dc is critical damage, εp, d is the damage threshold and ε_f is a fracture strain given by Johnson Cook as

$$\varepsilon_f=(D1 + D2EXP\left(D3\left(\frac{p}{\sigma Y}\right)\right)(1 + D4Ln\acute{\varepsilon})(D5T*)\text{------------------------Equation-4}$$

In which P/σy is stress triaxiality parameter and ε is strain rate. The constants D1 through D5 are material Constants and obtained from the experiment. The quantity described by $D= \sum \frac{\Delta\varepsilon}{\varepsilon f}$, Called Damage parameter is a function of strain rate and stress triaxiality coefficient. In this relationship [26].

$$\Delta\varepsilon^p=\sqrt{2}/3\ [\sum(d\varepsilon 1 - d\varepsilon 2)2]1/2\text{-------------------------Equation-5}$$

Is the plastic strain increment in each repetition and ε f is the fracture strain. When Damage parameter reaches unity failure will occur and the failed element will vanish.[26]

N.B:- Considering the above derivation as an input for the Johnson cook strength and failure values and constants are taken from the experimental result which summarized in the table below.

Table 2 shows Johnson cook strength and failure constant value for structural steel(St-60) [25]

Yield stress, A	448 Mpa	D1	0.0705
Strain hardening parameter, B	782 Mpa	D2	1.732
Strain hardening exponent n	0.562	D3	-0.54
Strain rate sensitivity parameter, c	0.024	D4	-0.015
Reference strain rate, $\dot{\varepsilon}$ (s^{-1})	0.001	D5	0
Temperature exponent, m	0.454		
Melting point, T_{melt} (C)	900 ^{o}f		
The room temperature T_o	293^{o}K		

3.2 Structural Modeling and analyzing

Traditionally, formal modeling of systems has been through a mathematical model, which attempts to find analytical solutions to problems and thereby enable the prediction of the behavior of the system from a set of parameters and initial conditions. While computer simulations might use some algorithms from purely mathematical models and can combine simulations with reality or actual events, such as generating input responses, to simulate test subjects who are no longer present[27].

3.2.1 Specification Qute for Geometrical and Mathematical Modeling

Here, Shell parts are built as a single unit. Each component is assembled together with spot welding to hold the body to the rigid chassis. But all these tasks may depend on the type of model that going to analyze.

For the case, Bajaj Qute vehicle having the specifications shown in the table below is regarded. The data and specification of Bajaj Qute have been collected from hora cooperative trading P.L.C Company and from field measurements, basic dimensions that are approximately equal to the specification data were obtained.

Table 3 shows Specifications of Bajaj Qute for modeling and analysis purpose [2]

Frame type	Monocoque Chassis	Ground Clearance	180mm
Length*Width*Height	2752mm*1312mm*1652mm	Kerb Weight	399Kg
Wheel Base	1925mm	Fuel tank (Reserve)	8L (including 1 liter reserve)
Suspension Front	Independent Suspension. Twin leading arm	Suspension Rear	Independent Suspension.
Turning radius	3.5 meters	Front and Rear Track	1143mm
Engine Type	Four stroke, spark ignition, single cylinder engine	Max. Speed	70km/h ,5th gear
Max. Power	9.9 kw at 5500Rpm	Displacement	217cc
Brake type	Hydraulic with H-split	Front Tire Size	101mm/R203mm
Brake Size	All Drum Dia,180mm	Rear tire size	101/R203mm
Seating capacity	4 (1+3),3 passangers	Number of seating row	2 Rows, front and back rows
Number of Doors	4 Doors	Boot space	44 liter ,for oil
Electric System	12 volt DC –ve earth	Battery	12volt, 26AH
Head Lamps	35*35 WHS1	Tail Lamp	Tail Lamp

3.2.2 Geometrical Modeling of Bajaj Qute for static analysis

The modeling of the Bajaj Qute structure is made with data collected from Hora cooperative group in addition to field measurements. Before measuring the structure, drawing of the Bajaj Qute structure is made. This helps for easy placement of the measured length on the corresponding members on the drawing.

During the data collection of the structure, all members of the structure are measured with the length measuring tape or tools. Modeling of the Bajaj Qute structure is made with SOLIDWORKS 18.0 software. The Bajaj Qute structure is made with structural steel sheet with different size and shapes that is according to the area that presence on the structure of Bajaj Qute. This Bajaj Qute has a capacity of carrying 4 passengers at a time [2].

3.2.2.1 Steps followed for Geometrical modeling of Bajaj Qute structure

Table 4 shows the basic Dimensions for existing and new models of Bajaj Qute structure

Model-1	**Dimensions**	**Value**	**Model-2**	**Dimensions**	**Value**
	Length	2752mm		Length	2752mm
	Height	1652mm		Height	1652mm
	Width	1312mm		Width	1512mm

3.2.2.1.1 Breakdown structure for 3 dimensional modeling

The initial formulation of the model as a function of lines, for the development of two dimensional model of structure accordance with the specification.

For the purpose of analysis, there are two-dimensionally different three-dimensional models of Bajaj Qute are developed and the only difference between them is the truck width (width of the vehicle). i.e. Rollover stability of Bajaj Qute is attain using those two model of vehicle and truck width of model-2 is obtained from ratio of wheel base vs truck width of passengers' vehicles.

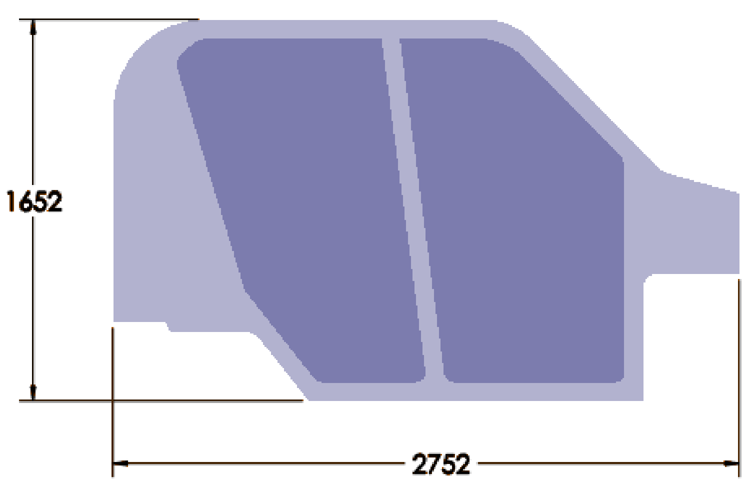


Figure 15 shows the initial two dimensional formulated model for Bajaj Qute

3.2.2.1.2 Defining main sections and modeling of body exterior panels

Here the three-dimensional modeling for each part independently developed using SOLID WORK surface modeling tools interlinking with u-channel of the structure. Surfacing modeling is a method of creating planar and non-planar complex geometry which has zero thickness.

<u>Advantages of Surfaces modeling:</u>

- ✓ Surfaces give much more flexibility when creating complex shapes that cannot be done using solid features.
- ✓ Surfaces can be used to build a shape face-by-face rather than all at once.
- ✓ The surface can be used as reference geometry.
- ✓ Surfaces can be more efficient than solid features depending on the object[28].

<u>Disadvantages of Surfaces modeling:</u>

- ✓ Modeling with surfaces required more work than modeling with solids because number of steps and element are joined together to form one single surface but solid is not.
- ✓ There is more complex workflow when creating surfaces and then converting to solids[28].

3.2.2.1.2.1 Total Assembly of Bajaj Qute with their detail descriptions

The vehicle main structure entirely consists of 17 parts which assembled together to form the main structure of Bajaj Qute.

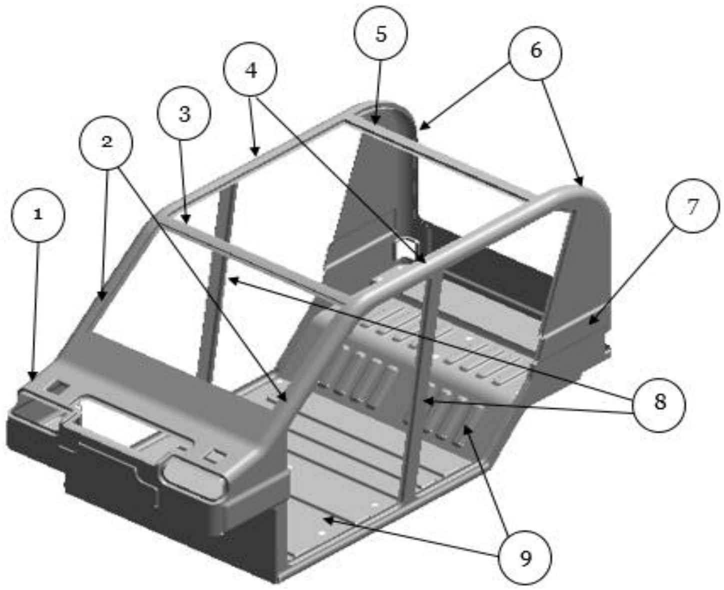


Figure 16 shows the assembly of Bajaj Qute with respective part description

3.2.2.1.2.2 Parts formulation of Bajaj Qute using surface modeling tools

1.) Frontal Panel Structure of Bajaj Qute (1)

The frontal part should be strong enough to withstand frontal impact and the external components like gril, shield, bumper, headlight, indicator light, and license plate are attached to it[29].

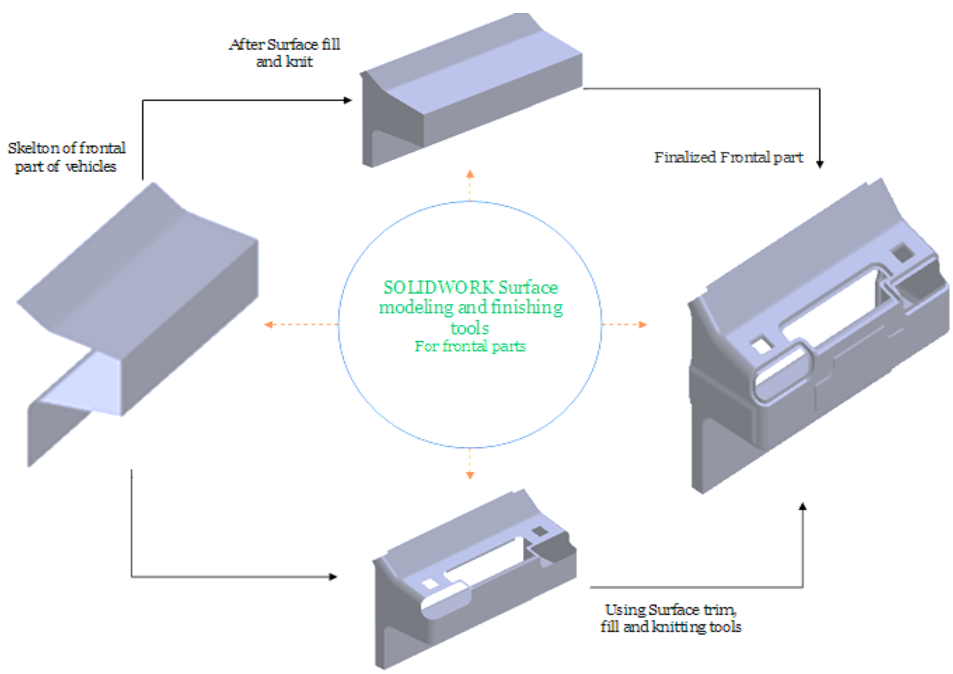


Figure 17 shows the modeling procedure for frontal part of Bajaj Qute structure

2.) Upper two Pillars-A of Structure of Bajaj Qute (2)

They are developed basically to withstanding the wind force that is the way they called as windshield pillar and used to protect occupants of a vehicle.

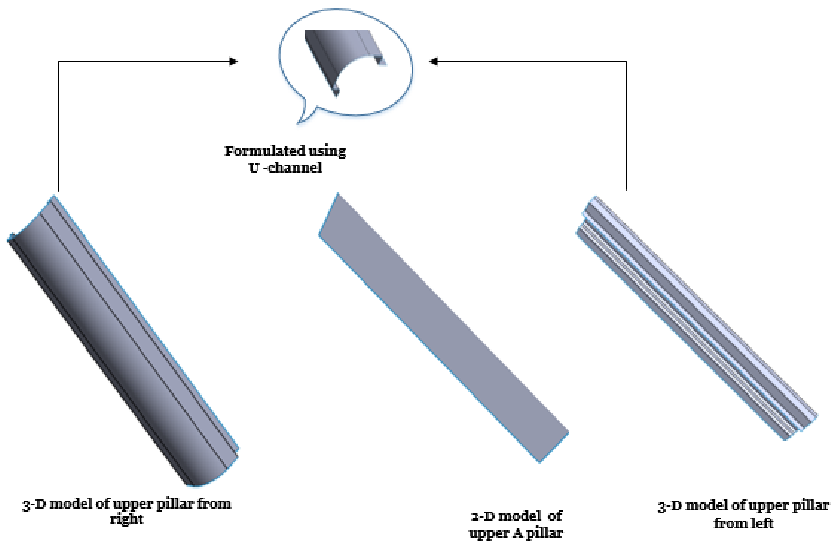


Figure 18 shows the model of pillars of A of Bajaj Qute structure

3.) Header Rail (front header)(3)

This component support and connect the upper two pillar with the Side roof rail of both rights and left parts with the help spot welding for the case of Bajaj Qute also it protects the occupants from head impact with the help of upper sheet cover.

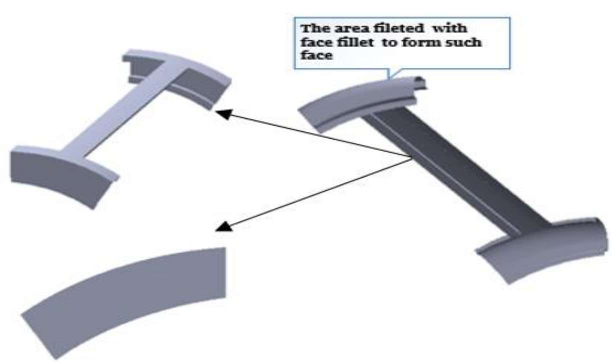


Figure 19 shows the model of Header Rail of Bajaj Qute structure

4.) Side roof rail or Cantrail (4):- Side roof rail or vehicle roof carrier is primarily helping the driver to balance the weight of the vehicle by transferring loads to the top[8].The side roof rail devolved using the dimensions equal to the existing model.

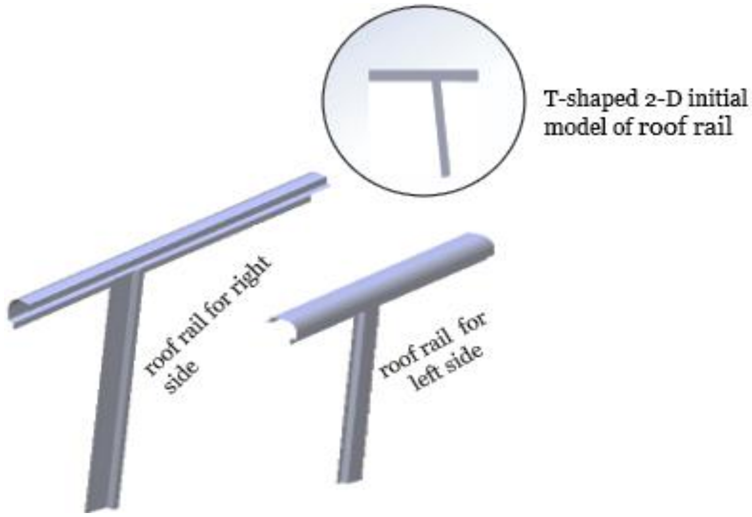


Figure 20 shows the model of Side roof rail of Bajaj Qute structure

5.) Lower B-pillars and sills (center pillars, rocker panels)(8)

B-pillars starts at the end of the first door and they usually offer substantial structural support for a vehicle side component .most of time they are bound to the body frame and wrap all the way into the floor. Depend on design, there may be multiple pieces attached to increase the integrity of the structure [29].

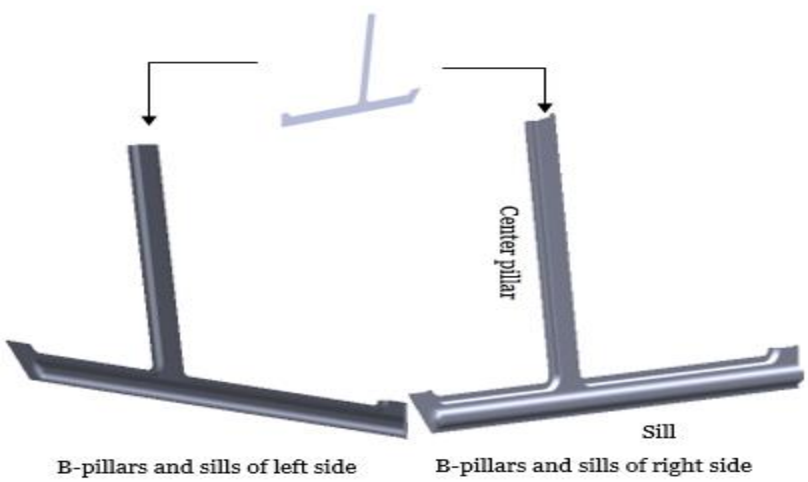


Figure 21 shows the model of B-pillars and sills of Bajaj Qute structure

6.) Back Header Rail (back header)(5)

Back header support and connect the Lower B-pillars and sills with the C-pillars of both rights and left parts with the strong spot welding in the case of Bajaj Qute[29].

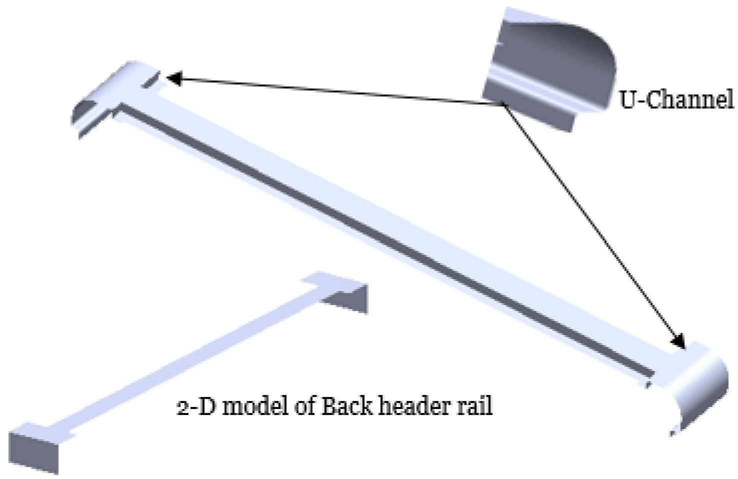


Figure 22 shows the model of Back Header Rail of Bajaj Qute structure

7.) C-pillars of the vehicle (6)

C-pillar is the third type of pillar that hold the sides of vehicles rear windows screen and it is vertical structure behind the rear door[29].

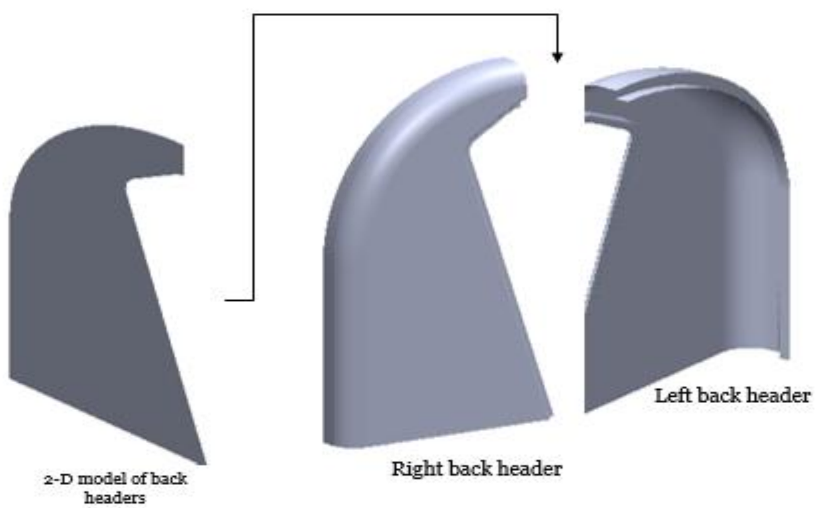


Figure 23 shows the model of C-pillars of both right and left parts of Bajaj Qute

8.) Rear quarter panel (7)

It is body panel of the vehicle between a rear door and the trunk and they typically a welded on a component of the unibody structure. Some quarter panels are one large piece that serves both as area fender and roof section[29].Here as shown in figure 24 the quarter panel developed using the trauck width of the vehicle structure.

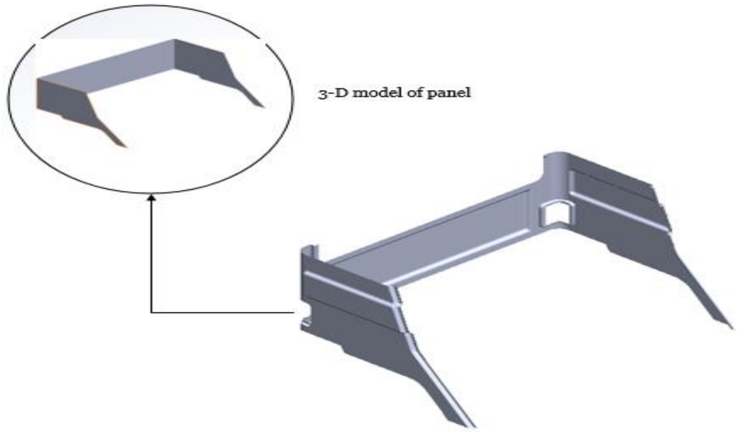


Figure 24 shows the model of a Rear quarter panel of Bajaj Qute structure

9.) Floor panel(Floor pan)(9)

Floor panel is a big sheet of metal stamping that incorporates several smaller stampings welded together to form the floor or underside of vehicles and corrugated sheet has been used to model some parts that is to increases the energy absorption of structure . It is also consist of a structural and external panel of automotive [29].

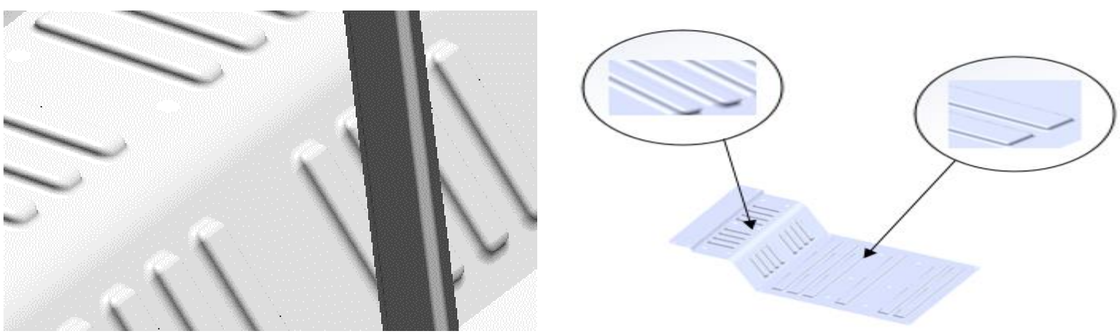


Figure 25 shows the model of Floor panel of Bajaj Qute structure

3.2.2.1.3 Detail Assembly of each pillar, headers, panels of Bajaj Qute

The total components of Bajaj Qute structures are designed on the foot of the manufacturing process of the vehicle. The structure generally consists of about 17 parts welded together with the help of Spot welding and for exterior constituent another type of joining mechanisms has been used like rivet joints, bolt, and nut etc.

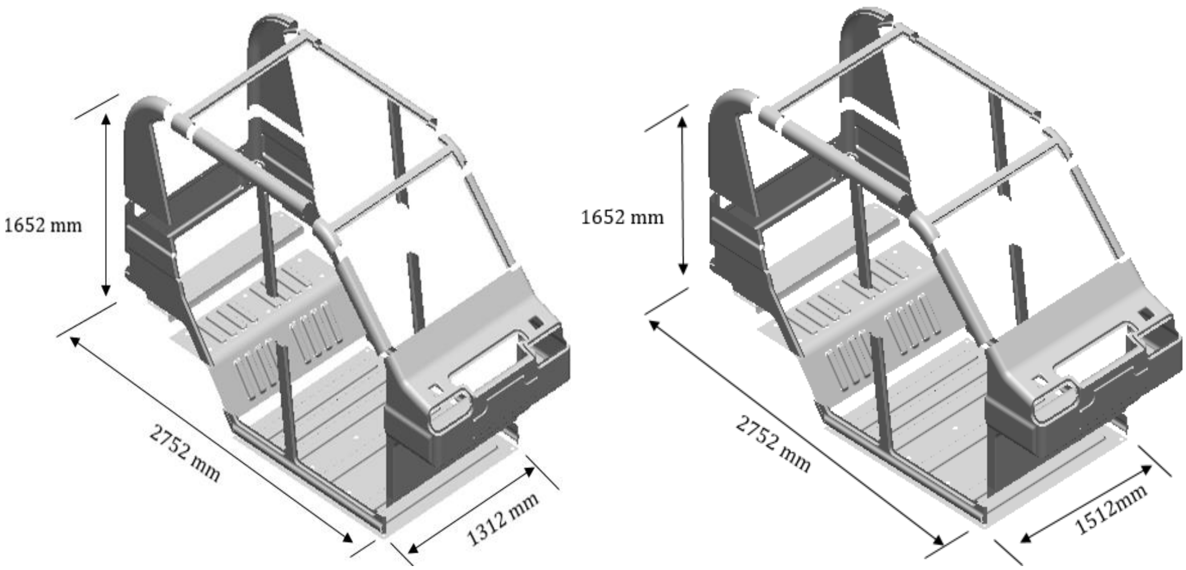


Figure 26 shows the exploded view two-dimensionally models of Bajaj Qute structure

3.2.3 Geometrical Modeling of Bajaj Qute for Frontal Panel Impact analysis

3.2.3.1 Frontal Panel model (crash box)

For explicit dynamic analysis of Bajaj Qute frontal panel used to represent the other component of the vehicle.

Table 5 shows the basic input dimensions for frontal panel

Basic Dimensions	Input Values
Length	504mm
Height	1131mm
Width	1512mm

Hence, using the primary formulated model as a reference two different models of frontal panel developed and this model of a frontal panel assembled together with other component to form the assembly model of the Bajaj Qute.

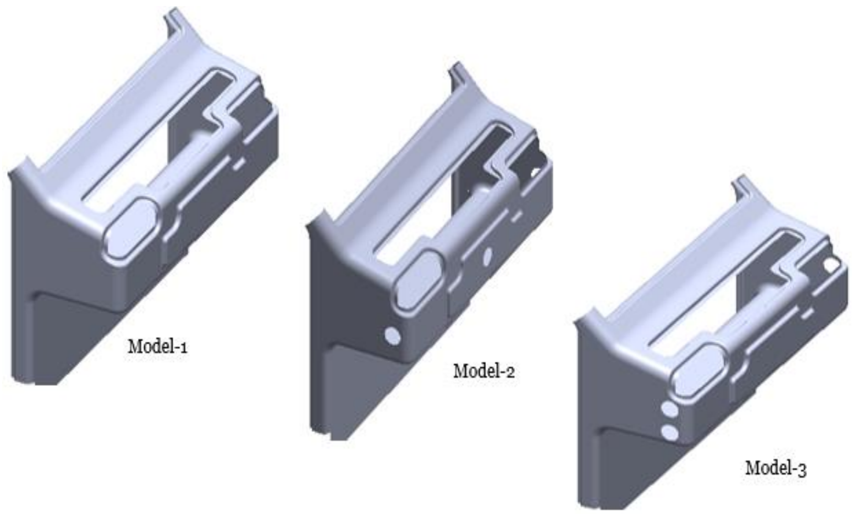


Figure 27 shows the three different models of Frontal Panel of Bajaj Qute structure

3.2.3.2 Model of the rigid wall for impact analysis

Model Developed as a rigid wall for each frontal panels are similar and designed using surfacing tool especially through with surface filling tool. Rigid wall developed merely by creating two-dimensional rectangle that dimensionally greater than frontal Panel and filling it with surface filling tools to create the model.

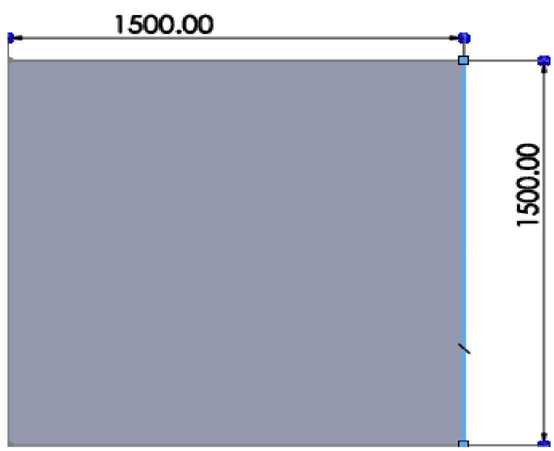


Figure 28 shows a model of the rigid wall of Bajaj Qute structure

The rigid wall is design in such way that to withstand the crash during impact simulation of vehicle structure and have non deformable behavior in addition same material properties with the frontal panel.

3.2.3.3 Assembly model of the frontal panel and rigid wall for impact analysis

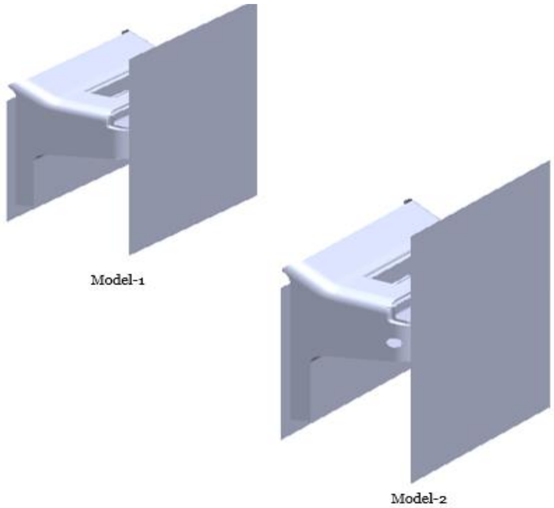


Figure 29 Shows Assembly of models of the frontal panel and rigid wall

3.2.4 Mathematical Modeling of Bajaj Qute structure

Mathematical modeling is the process of constructing mathematical objects whose behaviors or properties correspond in some way to a particular real-world system. It is the art of translating problems from an application area into tractable mathematical formulations whose theoretical and numerical analysis provides insight, answers, and guidance useful for the originating application[30].

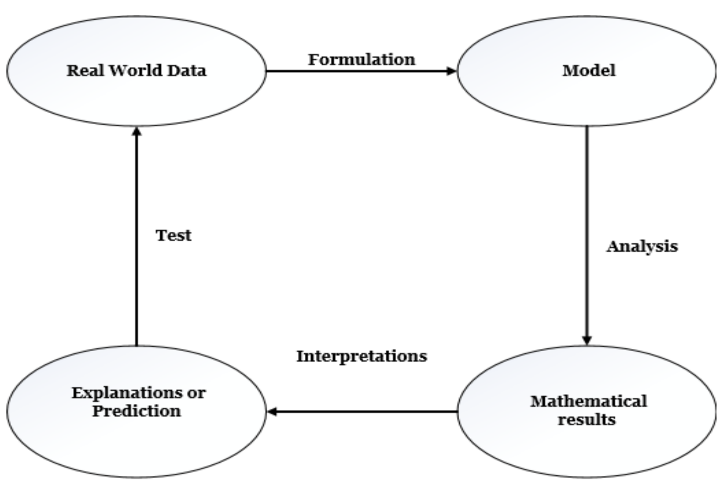


Figure 30 Schematic flow of the modeling process of the certain system

3.2.4.1 Motivation behind mathematical modeling

Mathematical modeling increase understanding about real-world situation typically, if we have a mathematical model which accurately reacts some behavior of a real-world system of interest, we can often gain an improved understanding of that system through analysis of the model[30].

3.2.4.2 Mathematical Modeling of Bajaj Qute structure for ANSYS static Structure analysis

3.2.4.2.1 Assumptions and Restrictions of ANSYS static structural analysis

- ✓ The static analysis solution method is valid for all degrees of freedom (DOF).
- ✓ Inertial and damping effects are ignored, except for static acceleration fields.

3.2.4.2.2 Derivation of the Structural matrix for ANSYS static structural analysis

The principle of virtual work states that a virtual (very small) change of the internal strain energy must be offset by an identical change in external due to the applied load[31].

$$\delta U = \delta V \text{-- Equation-6}$$

Where:

U = Strain energy (internal work) = U1 + U2

V = External work = V1 + V2 + V3, δ = Virtual operator

The virtual strain energy is defined as

$$\delta U_1 = \int_{vol} \{\delta\varepsilon\}\{\sigma\} d(vol)^T \text{--------------------------------------- Equation-7}$$

Where:

$\{\varepsilon\}$ = Strain vector, $\{\sigma\}$Stress vector, Vol = Volume of element and T=transpose of matrix

Taking into account that linear material and geometry and combining derivation of both prprinciplesf virtual work and virtual strain energy together to have the following expression[31].

$$\delta U_1 = \int_{vol} (\{\delta\varepsilon\}^T[D]\{\varepsilon\} - \{\delta\varepsilon\}^T[D]\{\varepsilon^{the}\})\,\{\sigma\}d(vol)^T \text{------------ Equation-8}$$

The strain related to the nodal displacement given by

$$\{\varepsilon\} = [B]\{u\} \text{-- Equation-9}$$

Where:

[B] = Strain-displacement matrices, based on the element shape function

{U} = Nodal displacement vector

It will be assumed that all effect is in the global Cartesian system. Combining strain related nodal displacement and virtual strain energy and nodal displacement vector does not vary over the volume the virtual strain energy become [31].

$$\delta U_1 = \{\delta u\}^T \int_{vol} \{B\}^T[D][B]d(vol)\{u\} - \{\delta u\}^T \int_{vol} \{B\}^T[D]\{\varepsilon^{the}\}d(vol) \text{----Equation-10}$$

Another form of virtual strain energy is when a surface against a distributed resistance, as a foundation stuffiness and express as follows

$$\delta U_2 = \int_{area_f} \{\delta w_n\}^T[\sigma]d(area_f) \text{--------------------------------------Equation-11}$$

Where:

$\{W_n\}$ = motion normal to the surface, {σ} = Stress carried by the surface

$Area_f$= area of the distributed resistance,

Both {wn} and {σ} have only one nonzero component. The point-wise normal displacement is related to the nodal displacement given by

$$\{W_n\} = [N_n]\ \{u\} \text{-- Equation-12}$$

Where: [Nn] is matrices of shape function for normal motions at the surface. The stress,

$$\{\sigma\} = k\ \{wn\}$$

Where: k is the foundation stuffiness in units of force per unit area and constant across the area. Another form of virtual strain energy formulation can be achieved by combining stress and the previous virtual strain energy that is

$$\delta U_2 = \{\delta w_n\}^{T}k \int_{area_f} \{N_n\}^T[N_n]d(area_f)\,\{u\} \text{--------------- Equation-13}$$

Following, the external virtual work considered. The inertial effect will be considered first

$$\delta V_1 = \int_{vol} \{\delta w\}^T \frac{\{F^a\}}{vol} d(vol) \text{ ------------------------------ Equation-14}$$

Where: $\{w\}$ = Vector of displacements of general point, $\{F^a\}$ = Acceleration (D'Alembert) force vector and according to Newton's second law[31]

$$\frac{\{F^a\}}{vol} = \rho \frac{\partial^2}{\partial t^2} \{w\} \text{---Equation-15}$$

Where: ρ, t are density and time respectively. The displacement within the element is related to the nodal displacement by[31]

$$\{W\} = [N]\ \{u\}$$

Where: [N] is the matrix of shape function and it is also the density of the structure is constant throughout the surface then[31]

$$\delta V_1 = -\{\delta u\}^T \rho \int_{vol} [N]^T [N] d(vol) \frac{\delta^2}{\delta t^2} \{u\} \text{-------------Equation-16}$$

The pressure force vector formulation starts with

$$\delta V_2 = \int_{area_p} \{\delta w_n\}^T \{P\} d(area_p) \text{------------------------ Equation-17}$$

Where: {p} and the applied pressure vector (normal constants only one non zero component) and $area_p$ = Area over which pressure acts. And then the equation become

$$\delta V_2 = \{\delta w_n\}^T \int_{area_p} [N_n] \{P\} d(area_p) \text{--------------- Equation-18}$$

Unless otherwise noted, pressures applied to the outside surface of each element and area normal to the curved surface, if applicable. The nodal force applied to the element can be accounted for by[31]

$$\delta V_2 = \{\delta u_n\}^T \{F_e^{nd}\} \text{-----------------------------------Equation-19}$$

Where: $\{F_e^{nd}\}$ is a nodal force applied to the element and finally, combine the above all equations to form

$$\{\delta u\}^T \int_{vol} [B]^T [D] d(vol) \{u\} - \{\delta u\}^T \int_{vol} [B]^T [D] \{\varepsilon^{the}\} d(vol) + \{\delta u\}^{T_k} \{u\} + \{\delta u\}^T \int_{area_f} [N]^T [N] d(area_f) \{u\} = -\{\delta u\}^T \rho \int_{vol} [N]^T [N]\, d(vol) \frac{\delta^2}{\delta t^2} \{u\} + \{\delta u\}^T \int_{area_p} [N]^T \{p\}\, d(area_p) + \{\delta u\}^T \{F_e^{nd}\} \text{------ Equation-20}$$

<u>Nothing that</u>: $\{\delta_u\}$ T vector is a set of arbitrary virtual displacement common in all of the above terms, the condition required to satisfy the equation above and reduced and This equation represents the equilibrium equation on a one element basics[31]

$$([K_e] + [K_e^f])\{u\} - \{F_e^{the}\} = [M_e]\{\ddot{u}\} + \{F_e^{pr}\} + \{F_e^{nd}\} \text{---- Equation-21}$$

Where:

$[K_e] = \int_{vol} [B]^T [D] d(vol)$ =element stiffness matrix

$[K_e^f] = \int_{area_f} [N]^T [N] d(area_f)$ = element foundation stiffness matrix

$\{F_e^{the}\} = \int_{vol} [B]^T [D] \{\varepsilon^{the}\} d(vol)$ = element thermal load vector

$[M_e] = \rho \int_{vol} [N]^T [N]\, d(vol) \frac{\delta^2}{\delta t^2}\{u\}$= element mass matrix

$\{\ddot{u}\} = \frac{\delta^2}{\delta t^2}\{u\}$= acceleration vector (such as gravitational effect)

$\{F_e^{pr}\} = \int_{area_p} [N]^T \{p\}\, d(area_p)$ = element pressure load vector

Description of Structural Systems the overall equilibrium equations for linear structural static analysis are:

$$\{F\} = [K]\{U\} \text{ Or } \{F^a\} + \{F^r\} = [K]\{U\} \text{ --------------Equation-22}$$

Where:

[K]= Total stiffness matrix

{U} = Nodal displacement vector

N =Number of elements

[Ke] =Element stiffness matrix (may include the element stress stiffness matrix)

$\{F^r\}$ = Reaction load vector

$\{F^a\}$=The total applied load vector, is defined by:

$$\{F^a\} = \{F^{nd}\} + \{F^{ac}\} + \sum_{m=1}^{N} [F]_e^{th} + \{F_e^{pr}\} \text{---Equation-23}$$

Where: $\left\{F^{nd}\right\}$ = Applied nodal load vector

$\left\{F^{ac}\right\}$ = - [M] {a_c} = Acceleration load vector

$\{M\}$ = Total mass matrix = $\sum_{e=1}^{N} M_e$

$[M_e]$ = Element mass matrix

$\{a_e^{th}\}$ = The Total acceleration vector

$\left\{F_e^{th}\right\}$ = Element thermal load vector

$\{F_e^{pr}\}$ = Element pressure load vector

3.2.4.2.3 Derivation of the Structural matrix for Shell element in ANSYS for static structure.

A shell structure carries loads in all directions and therefore undergoes bending and twisting, as well as in-plane deformation.[32] The plate structure can be treated as a special case of the shell structure, the shell element developed in this section is applicable for modeling plate structures. In fact, it is common practice to use a shell element offered in a commercial FE package to analyses plate structures[33].

3.2.4.2.3.1 Elements in Local Coordinate Systems for shell element

Shell structures are usually curved. The curvature of the shell is then followed by changing the orientation of the shell elements in space. Therefore, if the curvature of the shell is very large, a fine mesh of elements has to be used. This assumption sounds rough, but it is very practical and widely used in engineering practice[33].

Similar to the frame structure, there are six DOFs at a node for a shell element: three translational displacements in the x, y and z directions, and three rotational deformations with respect to the x, y and z-axes and the generalized displacement vector for the element

Can be written as[33].

$$\mathbf{d_e}=\begin{Bmatrix} de1 \\ de2 \\ de3 \\ de4 \end{Bmatrix} \text{ node 1, 2, 3} \text{------------------------- Equation-24}$$

Where: d_{ei} (i = 1, 2, 3, 4) are the displacement vector at node i

$$\mathbf{d_{ei}}=\begin{Bmatrix} ui \\ vi \\ wi \\ \Theta xi \\ \Theta yi \\ \Theta zi \end{Bmatrix} \text{----------------------------------- Equation-25}$$

Where: ui= displacement in x the direction, vi= displacement in the Y direction, wi= displacement in z direction, Θxi= rotation about x-axis, Θyi= rotation about y-axis, Θzi = rotation about z-axis

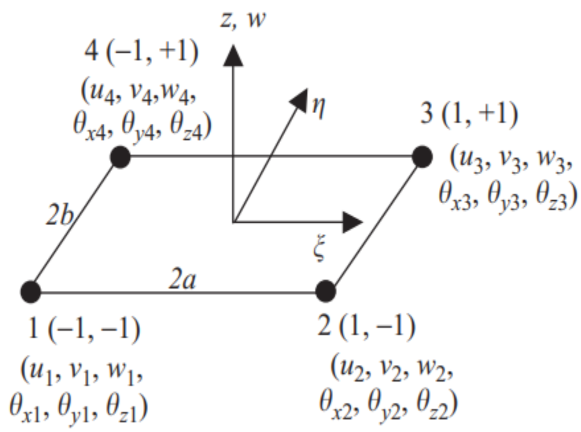


Figure 31 Schematic representation for the middle plane of a rectangular shell element

The membrane stiffness matrix can thus be expressed in the following form using sub-matrices according to the nodes:[42] where the superscript m stands for the membrane matrix. Each sub-matrix will have a dimension of 2 × 2 since it corresponds to the two

DOFs u and v at each node.

$$\boldsymbol{K_e^m} = \begin{bmatrix} k_{11}^m & k_{12}^m & k_{13}^m & k_{14}^m \\ k_{21}^m & k_{22}^m & k_{23}^m & k_{24}^m \\ k_{31}^m & k_{32}^m & k_{33}^m & k_{34}^m \\ k_{41}^m & k_{42}^m & k_{43}^m & k_{44}^m \end{bmatrix} \text{-------------------- Equation-26}$$

Note again that the matrix above is actually the same as the stiffness matrix of the 2D rectangular, solid element, except it is written in terms of sub-matrices according to the nodes and the stiffness matrix for a rectangular plate element is used for the bending effects, corresponding to DOFs of w, and θx, θy. The bending stiffness matrix can thus be expressed in the following form using sub-matrices according to the nodes[9,40].

$$\boldsymbol{k_e^b} = \begin{bmatrix} k_{11}^b & k_{12}^b & k_{13}^b & k_{14}^b \\ k_{21}^b & k_{22}^b & k_{23}^b & k_{24}^b \\ k_{31}^b & k_{32}^b & k_{33}^b & k_{34}^b \\ k_{41}^b & k_{42}^b & k_{43}^b & k_{44}^b \end{bmatrix} \text{-------------------- Equation-27}$$

Where the superscript b stands for the bending matrix. Each bending sub-matrix has a dimension of 3 × 3. The mass matrix for the shell element in the local coordinate system is then formulated by combining Equations of bending stiffness matrix and membrane stiffness matrix[34].

$$\mathbf{k_e}=\begin{bmatrix}
k_{11}^{m} & 0 & 0 & k_{12}^{m} & 0 & 0 & k_{13}^{m} & 0 & 0 & k_{14}^{m} & 0 & 0\\
0 & k_{11}^{b} & 0 & 0 & k_{12}^{b} & 0 & 0 & k_{13}^{b} & 0 & 0 & k_{14}^{b} & 0\\
0 & 0 & 0 & 0 & 0 & & 0 & 0 & 0 & 0 & 0 & 0\\
k_{21}^{m} & 0 & 0 & k_{22}^{m} & 0 & 0 & k_{23}^{m} & 0 & 0 & k_{24}^{m} & 0 & 0\\
0 & k_{21}^{b} & 0 & 0 & k_{22}^{b} & 0 & 0 & k_{23}^{b} & 0 & 0 & k_{24}^{b} & 0\\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0\\
k_{31}^{m} & 0 & 0 & k_{32}^{m} & 0 & 0 & k_{33}^{m} & 0 & 0 & k_{34}^{m} & 0 & 0\\
0 & k_{31}^{b} & 0 & 0 & k_{32}^{b} & 0 & 0 & k_{33}^{b} & 0 & 0 & k_{34}^{b} & 0\\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0\\
k_{41}^{m} & 0 & 0 & k_{42}^{m} & 0 & 0 & k_{43}^{m} & 0 & 0 & k_{44}^{m} & 0 & 0\\
0 & k_{41}^{b} & 0 & 0 & k_{42}^{b} & 0 & 0 & k_{43}^{b} & 0 & 0 & k_{44}^{b} & 0\\
0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0
\end{bmatrix} \text{------ Equation-28}$$

The stiffness matrix for a rectangular shell matrix has a dimension of 24 × 24. Note that in the above equation, the components related to the DOF θz, are zeros. This is because there is no θ_z in the local coordinate system. If these zero terms are removed, the stiffness matrix would have a reduced dimension of 20×20[39,40].

Similarly, the mass matrix for a rectangular element can be obtained in the same way as the stiffness matrix. The mass matrix for the 2D solid element is used for the membrane effects, corresponding to DOFs of u and v. The membrane mass matrix can be expressed in the following form using sub-matrices according to the nodes[32]:

$$\boldsymbol{m}_{\boldsymbol{e}}^{\boldsymbol{m}} = \begin{bmatrix}
m_{11}^{m} & m_{12}^{m} & m_{13}^{m} & m_{14}^{m}\\
m_{21}^{m} & m_{22}^{m} & m_{23}^{m} & m_{24}^{m}\\
m_{31}^{m} & m_{32}^{m} & m_{33}^{m} & m_{34}^{m}\\
m_{41}^{m} & m_{42}^{m} & m_{43}^{m} & m_{44}^{m}
\end{bmatrix} \text{-------------------- Equation-29}$$

Where the superscript m stands for the membrane matrix. Each membrane sub-matrix has a dimension of 2 × 2. The mass matrix for a rectangular plate element is used for the bending effects, corresponding to DOFs of w, and θx, θy. The bending mass matrix can also be expressed in the following form using sub-matrices according to the nodes[33].

$$m_e^b = \begin{bmatrix} m_{11}^{b} & m_{12}^{b} & m_{13}^{b} & m_{14}^{b} \\ m_{21}^{b} & m_{22}^{b} & m_{23}^{b} & m_{24}^{b} \\ m_{31}^{b} & m_{32}^{b} & m_{33}^{b} & m_{34}^{b} \\ m_{41}^{b} & m_{42}^{b} & m_{43}^{b} & m_{44}^{b} \end{bmatrix}$$ -------------------- Equation-30

Where the superscript b stands for the bending matrix. Each bending sub-matrix has a dimension of 3 × 3. The mass matrix for the shell element in the local coordinate system is then formulated by combining the above two equations to form the following matrix[34].

$$\mathbf{k_e}=\begin{bmatrix} m_{11}^{m} & 0 & 0 & m_{12}^{m} & 0 & 0 & m_{13}^{m} & 0 & 0 & m_{14}^{m} & 0 & 0 \\ 0 & m_{11}^{b} & 0 & 0 & m_{12}^{b} & 0 & 0 & m_{13}^{b} & 0 & 0 & m_{14}^{b} & 0 \\ 0 & 0 & 0 & 0 & 0 & & 0 & 0 & 0 & 0 & 0 & 0 \\ m_{21}^{m} & 0 & 0 & m_{22}^{m} & 0 & 0 & m_{23}^{m} & 0 & 0 & m_{24}^{m} & 0 & 0 \\ 0 & m_{21}^{b} & 0 & 0 & m_{22}^{b} & 0 & 0 & m_{23}^{b} & 0 & 0 & m_{24}^{b} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ m_{31}^{m} & 0 & 0 & m_{32}^{m} & 0 & 0 & m_{33}^{m} & 0 & 0 & m_{34}^{m} & 0 & 0 \\ 0 & m_{31}^{b} & 0 & 0 & m_{32}^{b} & 0 & 0 & m_{33}^{b} & 0 & 0 & m_{34}^{b} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ m_{41}^{m} & 0 & 0 & m_{42}^{m} & 0 & 0 & m_{43}^{m} & 0 & 0 & m_{44}^{m} & 0 & 0 \\ 0 & m_{41}^{b} & 0 & 0 & m_{42}^{b} & 0 & 0 & m_{43}^{b} & 0 & 0 & m_{44}^{b} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix}$$ ----- Equation-31

3.2.4.2.3.2 Elements in Global Coordinate Systems for shell element

The matrices for shell elements in the global coordinate system can be obtained by performing the transformations.

$K = \mathrm{TKeT^T}$ -- Equation-32

$$\mathrm{Me} = \mathrm{TMeT^T}, \qquad \mathrm{Fe} = \mathrm{T^T\,Fe}$$

Where T is the transformation matrix, given by

$$\mathbf{T} = \begin{bmatrix} T_3 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & T_3 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & T_3 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & T_3 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & T_3 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & T_3 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & T_3 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & T_3 \end{bmatrix} \text{------------------------------------ Equation-33}$$

In which

$$T_3 = \begin{bmatrix} l_x & m_x & n_x \\ l_y & m_y & n_y \\ l_z & m_z & n_z \end{bmatrix} \text{-- Equation-34}$$

Where l_k, mk, and n_k (k= x, y, z) are directional cosines defined by

$$l_x = \cos(x, X) , m_x = \cos(x, Y) , n_x = \cos(x, Z) \; l_y = \cos(y, X) , m_y = \cos(y, Y) , n_y = \cos(y, Z)$$

$$l_z = \cos(z, X) , m_z = \cos(z, Y) , n_z = \cos(z, Z)$$

To define these direction cosines, the position, and the three-dimensional orientation of the frame element must define first. With nodes 1 and 2, the location of the element fixed on the local coordinate frame, and the orientation of the element has also been fixed in the x-direction. However, the local coordinate frame can still rotate about the axis of the beam. [32]

3.2.4.3 Mathematical Modeling of Qute structure for Ansys explicit dynamic analysis

3.2.4.3.1 Basic formulation of explicit dynamic analysis

The partial differential equations to be solved in an Explicit Dynamics analysis express the conservation of mass, momentum, and energy in Lagrangian coordinates. These, together with a material model and a set of initial and boundary conditions, define the complete solution of the problem. For the Lagrangian formulations currently available in the Explicit Dynamics system, the mesh moves and distorts with the material it models and conservation of mass are automatically satisfied. The density at any time can be determined from the current volume of the zone and its initial mass[35].

$\frac{\rho_0 V_0}{V} = \frac{m}{V}$ -- Equation-35

The partial differential equations that express the conservation of momentum relate the acceleration to the stress tensor σ_{ij}.

$$\rho\ddot{x} = b_x + \frac{\partial\,\sigma xx}{\partial x} + \frac{\partial\,\sigma xy}{\partial y} + \frac{\partial \sigma xz}{\partial z}$$ -- Equation-36

$$\rho\ddot{y} = b_y + \frac{\partial\,\sigma yx}{\partial x} + \frac{\partial\,\sigma yy}{\partial y} + \frac{\partial \sigma yz}{\partial z}$$ -- Equation-37

$$\rho\ddot{z} = b_z + \frac{\partial\,\sigma zx}{\partial x} + \frac{\partial\,\sigma zy}{\partial y} + \frac{\partial \sigma zz}{\partial z}$$ --Equation-38

And Conservation of energy is expressed according to

$\dot{e} = \frac{1}{\rho}(\sigma xx\dot{\varepsilon}_{xx} + \sigma yy\dot{\varepsilon}_{yy} + \sigma zz\dot{\varepsilon}_{zz} + 2\sigma xy\dot{\varepsilon}_{xy} + 2\sigma yz\dot{\varepsilon}_{yz} + 2\sigma yz\dot{\varepsilon}_{zx}$----------------- Equation-39

These equations are solved explicitly for each element in the model, based on input values at the end of the previous time step. Small time increments are used to ensure the stability and accuracy of the solution. Note that in Explicit Dynamics we do not seek any form of equilibrium; we simply take results from the previous time point to predict results at the next time point. There is no requirement for iteration[35].

3.2.4.3.2 Basic formulation explicit Time Integration

The Explicit Dynamic solver in ANSYS uses a central difference time integration scheme to proceed to another step. After forces have been computed at the nodes of the mesh (resulting from internal stress, contact, or boundary conditions), the nodal accelerations are derived by equating acceleration to force divided by mass.

$$\ddot{X}_i = F_i/m + b_i$$ -- Equation-40

Where: $\ddot{X}_i$ = Components of nodal acceleration (i=1, 2, 3), b_i =are the components of body acceleration, F_i = Forces acting on the nodal points, m= is the mass attributed to the node. With the accelerations at time n determined, the velocities at time $n + 1/2$ are fa ound from[35].

$\dot{x}_i^{n+1/2} = \dot{x}_i^{n-1/2} + \ddot{x}_i^{n} \Delta t^{n}$-- Equation-41

And finally, the positions are updated to time n+1 by integrating the velocities

$x_i^{n+1} = x_i^{n} + \dot{x}_i^{n+1/2} + \Delta t_i^{n+1/2}$--- Equation-42

3.2.4.3.3 Basic formulation Mass Scaling in ANSYS Explicit analysis

The maximum time step that can be used in explicit time integration is inversely proportional to the sound speed of the material, hence directionally proportional to the square root of the mass of material in an element[35].

$\Delta t \sim \frac{1}{c} = 1/\sqrt{Cii/\rho} = \sqrt{m/vcii}$--- Equation-43

Where: C_{ii} is the material stiffness (i=1, 2, 3), ρ is the material density m is the material mass and V is the element volume and by artificially increasing the mass of an element, one can increase the maximum allowable stability time step, and reduce the number of time increments required to complete a solution[35].

3.2.4.3.4 Basic Step Controls in ANSYS Explicit analysis

1.) **Maximum Energy Error**

Energy conservation is a measure of the quality of an explicit dynamic simulation. Bad energy conservation usually implies a less than optimal model definition. This parameter allows you to automatically stop the solution if the energy conservation becomes poor. Enter a fraction of the total system energy at the reference cycle at which you want the simulation to stop. For example, the default value of 0.1 will cause the simulation to stop if the energy error exceeds 10% of the energy at the reference cycle.[43]the global energy is accounted as follows

$$\text{Reference Energy} = [\text{Internal Energy} + \text{Kinetic Energy} + \text{Hourglass Energy}]$$
At reference cycle

$$\text{Current Energy} = [\text{Internal Energy} + \text{Kinetic Energy} + \text{Hourglass Energy}]$$
At current cycle

Work Done = Work done by constraints + Work done by loads + Work done by body forces + Energy removed from system by element erosion + Work done by contact penalty forces.

$$\text{Energy error} = \frac{[\text{current energy} - \text{refrence energy} - \text{workdone}]}{\max([\text{current energy}],[\text{refrence energy}],[\text{kinetic energy}])} \text{----------------- Equation-44}$$

3.2.4.3.5 Rigid Body in ANSYS Explicit analysis

Rigid materials can be modeled in an explicit dynamics system by selecting geometry, "Stiffness behavior equal to rigid" on a body. In such cases only the density property of the material associated with the body will be used. For explicit dynamics systems all rigid bodies must be discretized with a full mesh. This will be specified by default for the explicit meshing physics preference. The mass and inertia of the rigid body will be derived from the elements and material density for each body [38,41].

By default, a kinematic rigid body is defined in explicit dynamics and its motion will depend on the resultant forces and moments applied to it through interaction with other parts of the model. Elements filled with rigid materials can interact with other regions via contact. Constraints can only be applied to an entire rigid body. Displacement cannot be applied to one edge of a rigid body; it must be applied to the whole body[38,42].

3.2.5 Rollover stability Analysis of Bajaj Qute by varying truck width

Before proceed to rollover stability analysis of Bajaj Qute it is better to study terms and concepts related with rollover stability analysis like center of gravity, height of center of gravity and static stability factor of vehicles.

3.2.5.1 Derivation for Center of Gravity and height of center of gravity of vehicles.

One of the most important of having and knowing the center of gravity of vehicles gravity and moment balancing or center of gravity is a point in which moment and gravity are balanced.

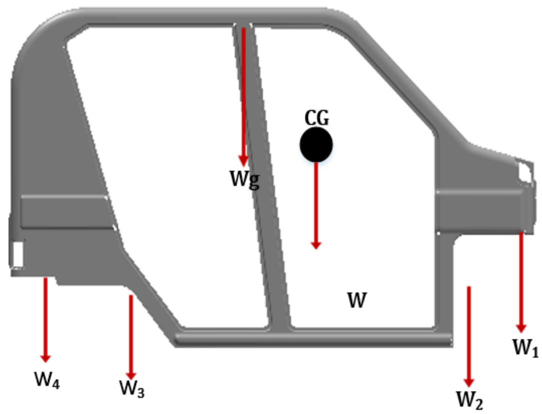


Figure 32 shows the summation of moment created in vehicles from CG position

Summation of moments of parts weights around any point is equal to the moment of the summation of weights around this point.

$(w_i . X_{Cg} = (w_1 . X_1 + w_2 . X_2 + w_3 . X_3 + w_4 . X_4 + w_5 . X_{5+\cdots\ldots})$---------------------- Equation-45

$X_{Cg} = \sum(w.X) / \sum(W)$--- Equation-46

Where: (Wi)=weight of components of vehicles, where Xi is the distance in x direction between the point i and that point. [17]

3.2.5.2 Effect of height of center of gravity, truck width on the Rollover stability of vehicle.

The center of mass height, relative to the wheelbase, determines load transfer between front and rear. The car's momentum acts at its center of mass to tilt the car forward or backward, respectively during braking and acceleration. Since it is only the downward force that changes and not the location of the center of mass, the effect on over/under steer is opposite to that of an actual change in the center of mass[18].

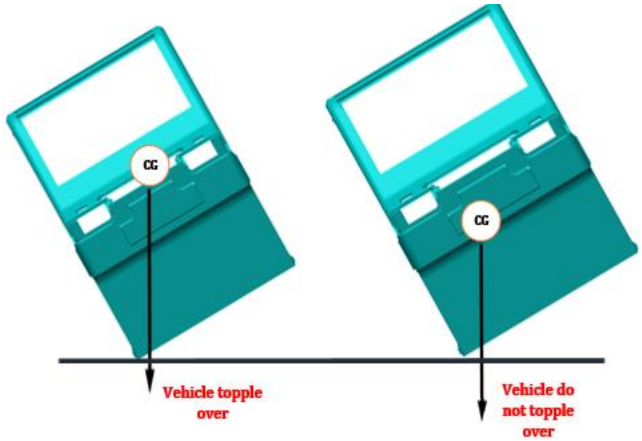


Figure 33 illustrate the effect of height of the CG on rollover stability of vehicle

A lower center of mass is a principal performance advantage of sports cars, compared to sedans and (especially) SUVs. Some cars have body panels made of lightweight materials partly for this reason[17].

3.2.5.2.1 Obtaining height above ground of CG (h)

Center of gravity (CG) height is an important parameter for lightweight vehicles (LWV). Because of the inherently smaller weight and size, an LWV's CG height is more easily affected by loading conditions compared with conventional vehicles. This paper proposes a novel tire instant effective radius (TIER) method for real-time estimation of the CG height for LWVs. The method utilizes the mathematical correlation between the tire vertical load transfer that is proportional to the CG height and the TIER variation[36].

As the sown figure below the weight of the rear axle (Wr1) will be weighed while the front pair of wheels is raised up quite a small distance H (or h1) as shown figure below[18].

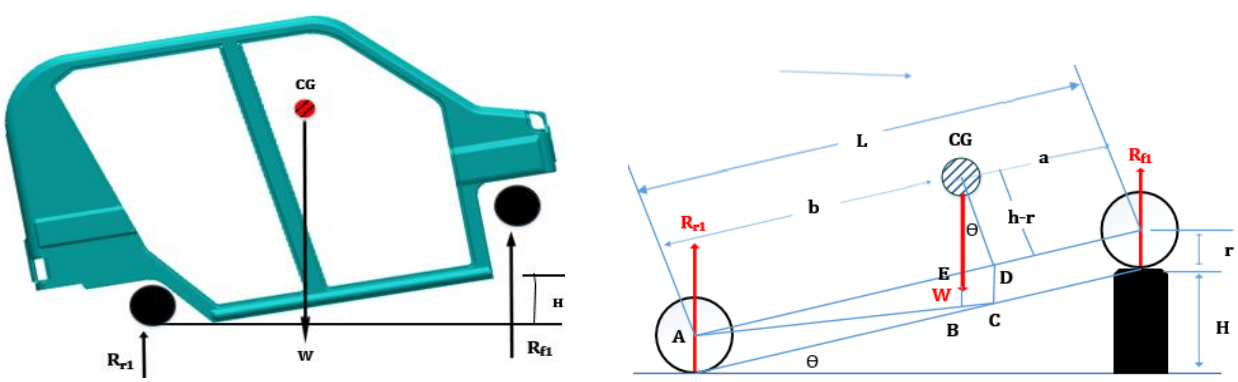


Figure 34 shows the mathematical expression to find the height above the CG

From above figure, the front wheels are raised up with a small distance H (h1) and r is the wheel radius with a distance of the CG height (h). Using equilibrium equation to solve height above the center of gravity from the ground as follows[16,17].

1.) The summation of vertical forces in the y-direction is equal to 0($\sum Fy = 0$) referring above figure we have the following

$$\sum \mathrm{Fy} = 0 \text{ , -- Equation-47}$$

$$(\mathrm{R_{r1}} + \mathrm{R_{f1}} - \mathrm{W}) = 0, \mathrm{R_{f1}} = (\mathrm{W} - \mathrm{R_{r1}}) \text{ ------------------------- Equation-48}$$

2.) The summation of moments about any point is equal to 0, ($\sum M_A = 0$) and we have the following expression.

$$\sum \mathrm{M_A} = 0 \text{ ,}$$

$$(\mathrm{R_{f1}}(\mathrm{Lcos\Theta}) - \mathrm{W(AB)} = 0,$$

$$(\mathrm{R_{f1}}(\mathrm{Lcos\Theta}) = \mathrm{W(AB)} \text{--Equation-49}$$

From the free body diagram that presence on the left side of the above figure, we have the following expression. $\mathrm{B} = \mathrm{AC} - \mathrm{BC}$, Where:

$$A\mathrm{C} = \mathrm{bcos\Theta} \text{ And } \mathrm{BC} = \mathrm{ED} = (\mathrm{h} - \mathrm{r})\mathrm{sin\Theta} \text{------------------------- Equation-50}$$

Where: (Θ) is the critical angle, the angle created when the center of gravity is directly over the pivot point. Then referring the above figure and using those equations derived from both moment and reaction forces at equilibrium condition,

$$A\mathrm{B} = \mathrm{AC} - \mathrm{BC} = \mathrm{bcos\Theta} - (\mathrm{h} - \mathrm{r})\mathrm{sin\Theta} \text{----------------------------Equation-51}$$

Substitute the value of AB form the above equation in equation 1 to have the following relationship,

$$(\mathrm{R_{f1}}(\mathrm{Lcos\Theta}) = \mathrm{W}(\mathrm{bcos\Theta} - (\mathrm{h} - \mathrm{r})\mathrm{sin\Theta}) \text{ ------------------------ Equation-52}$$

$$(\mathrm{R_{f1}}(\mathrm{Lcos\Theta}) = \mathrm{Wbcos\Theta} - \mathrm{W}(\mathrm{h} - \mathrm{r})\mathrm{sin\Theta}) \text{-------------------------Equation-53}$$

To get the unknown distance of the center of gravity from the ground, substitution of the equation is done as follows.

$$\left(W(h-r)\sin\theta = Wb\cos\theta - R_{f1}(L\cos\theta)\right) \text{-------- Equation-54}$$

$$\left((h-r) = (b - L(\frac{R_{f1}}{W}))\cot\theta\right) \text{-------- Equation-55}$$

$$\left(h = b - L(\frac{R_{f1}}{W}))\cot\theta + r\right) \text{-------- Equation-56}$$

$$\text{Where: } \theta = \sin^{-1}(\frac{H}{L})$$

$(h-r)$ Is the distance of CG above the axle plane and h is the distance of CG above the ground.[18] But if we consider the height of center of gravity from ground from side location of CG.$\theta = \tan^{-1}(\frac{t}{2H})$, Where: t=truck width and h= height of center of gravity from ground[18].

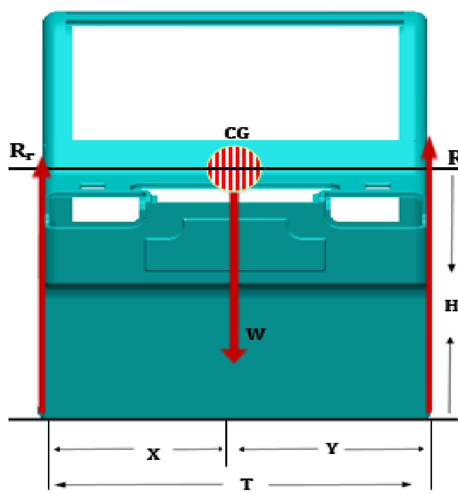


Figure 35 Illustrate frontal view to finding height of CG mathematically

3.2.5.3 Derivation for static stability factor (SSF) of vehicles.

The problem of vehicle stability is then described based on the Static Stability Factor (SSF), the ratio of the one-half the track width T divided by the height of the center of gravity H of the vehicle and One of the primary means of assessing rollover risk is the static stability factor (SSF), a measurement of a vehicle's resistance to rollover. The higher the SSF, the lower the rollover risk[4].

The static stability factor (SSF) of a vehicle is an at rest calculation of its rollover resistance, based on its most important geometric properties. Basically, SSF is a measure of how top-heavy a vehicle is. A vehicle's static stability factor (SSF) is calculated using the formula shown below[3,44].

$$\mathrm{SSF} = \frac{\mathrm{T}}{2\mathrm{H}} \text{------------------------------------ Equation-57}$$

The angle of tilt of the vehicle prior to rollover or critical angle (θ)

$$\theta = \tan^{-1}\left(\frac{\mathbf{T}}{\mathbf{2H}}\right) \text{------------------------------------ Equation-58}$$

Where: (H) Height of the center of gravity of the vehicle and track width (T) is the distance between the centers of the right and left tires along the axle.

Table 6 Shows typical values of static stability factor (SSF) of some vehicles

No.	Vehicle types	SSF values
1	Large vehicle types	1.00-1.50
2	Most passenger cars	1.30-1.50
3	Higher-riding SUV, vans	1.00-1.30

Note that:-The location of the center of gravity is measured in a laboratory to determine the average height above the ground of the vehicle's mass and the lower the SSF number, the more likely the vehicle is to roll over in a tripped single vehicle crash whereas higher SSF value equates to a more stable, less top-heavy vehicle[4].

3.2.5.4 Rollover Stability Analysis of Bajaj Qute Model-1 with the original truck width

3.2.5.4.1 Input data of Model-1 for rollover stability analysis

Here as input static stability factor for Bajaj Qute taken as 1.35 since, for passenger's car SSF is in range between 1.3-1.5 so

Table 7 Shows values of static stability factor (SSF) and basic dimensions for Model-1

	Overall dimensions(m)		SSF Value
Model-1	Length(L)	2.752	1.35
	Height(h)	1.652	
	Width (T)	1.312	

a. Height above ground of center of gravity(H) for model-1

Using **SSF** $= \frac{\mathbf{T}}{\mathbf{2H}}$, it is easy to calculate the height above ground of CG .first substitute SSF=1.35, $1.35 = \frac{1.312}{2\mathrm{H}}$ form this height above CG (H) is derived as

$\mathrm{H} = 0.49\mathrm{m} = 490\mathrm{mm}$-- (Ans.)

b. Critical angle or angle of tilt (θ) for model-1

$\theta = tan^{-1}(\frac{t}{2H})$, $\theta = \tan^{-1}(\frac{1.312}{2*0.49})$

$\theta = \tan^{-1}(1.338)$,
$\theta = 53.23\ ^0$-- (Ans.)

3.2.5.5 Rollover Stability Analysis of Bajaj Qute Model-2 with the Modified truck width

3.2.5.5.1 Input data of Model-2 for rollover stability analysis

Here as input static stability factor for Bajaj Qute taken as 1.35 for rollover stability analysis for model-2

Table 8 Shows values of static stability factor (SSF) and basic dimensions for model-2

	Overall dimensions(m)		SSF Value
Model-2	Length(L)	2.752	1.35
	Height(h)	1.652	
	Width (T)	1.512	

a. Height above ground of center of gravity (H) for model-2

Using $\mathrm{SSF} = \frac{\mathbf{T}}{\mathbf{2H}}$, it is easy to calculate the height above ground of CG .first substitute SSF=1.35, $1.35 = \frac{1.512}{2\mathrm{H}}$ form this height above CG (H) is derived as

$\mathrm{H} = 0.56\mathrm{m} = 560\mathrm{mm}$-- (Ans.)

b. Critical angle or angle of tilt (θ) for model-2

$\theta = tan^{-1}(\frac{t}{2H})$, $\theta = \tan^{-1}(\frac{1.512}{2*0.56})$

$\theta = \tan^{-1}(1.52)$, $\theta = 56.65^0$--- (Ans.)

3.2.6 Computing Torsional stiffness for Models of Bajaj Qute

The stiffness of a vehicle structure has important influences on its handling and vibrational behavior. It is important to ensure that deflections due to extreme loads are not as large as to impair the function of the vehicle. Low stiffness can lead to unacceptable vibrations, such as ‘scuttle shake’. Again, different load cases require different stiffness definitions, and some of these are often used as ‘benchmarks’ of vehicle structural performance like that of torsional stiffness (K_T) and mathematically

$$\mathrm{K_T} = \frac{\mathbf{T}}{\theta},$$ -- Equation-59

Where: T = torque and θ =angle of twist

Torsion stiffness is an important characteristic in chassis design with an impact on the ride and comfort as well as the performance of the vehicle.[38] The acceptable torsional stiffness can be evaluated for specific criteria while for other criteria it is based on experience and development, as described in the previous section. A typical medium-sized saloon fully assembled can have a torsional stiffness of 8000 to 10 000 N-m/degree. That is when measured over the wheel-base of the vehicle. Experience shows this to be acceptable for a road going passenger car. If the stiffness is low, driver perception is that the front of the vehicle appears to shake with the front wing structures tending to move up and down[39].

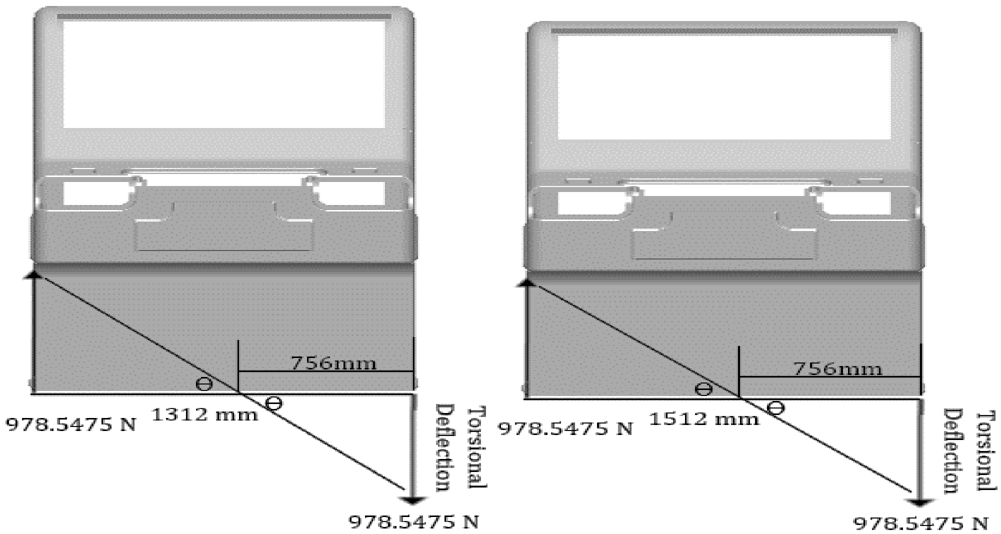


Figure 36 shows the mathematical expression to find torsional stiffness of two models

3.2.6.1 Torsional stiffness for Model-1 of structure of Bajaj Qute

Here referring figure for model-1, the total moment (M) calculated as follows.

$\mathrm{M} = (\,978.547\mathrm{N} * 1.312\mathrm{m})/2 + (978.547\mathrm{N} * 1.312\mathrm{m})/2 = 978.547\mathrm{N} * 1.312\mathrm{m} =$ 1283.84448Nm=T (torque)

Using the above result as an input to calculate the torsional stiffness of model-1.before that it is important to calculate the angle of twist or angle of deflection using the above figure. And from that, it is visible $\theta = \tan^{-1}\left(\frac{\text{torsional deflection}}{\frac{L}{2}}\right)$ =1.5828 mm/656mm

Hence the torsional deflection is captured from the static analysis for pure torsion loading case of structure for model-1,

$$\theta = \tan^{-1}(0.0024) \text{ Then } \theta = 0.138^0\,,\quad K_T = \frac{1283.84448\mathrm{Nm}}{0.138}$$

$$K_T = 9303.2\mathrm{Nm/degree}$$

3.2.6.2 Torsional stiffness for Model-2(standard table)

$$M = (978.547\mathrm{N} * 1.512\mathrm{m})/2 + (978.547\mathrm{N} * 1.512\mathrm{m})/2 = 978.547\mathrm{N} * 1.512\mathrm{m} =$$
$$1479.6\mathrm{Nm} = \mathrm{T}\ (\text{torque})$$

Using above result as an input to calculate torsional stiffness of model-2.before that it is important to calculate angle of twist or angle of deflection using above figure.

And from that, it is visible $\theta = \tan^{-1}\left(\frac{\text{torsional deflection}}{\frac{L}{2}}\right)$ = 1.399mm/756mm Hence the torsional deflection is captured from the static analysis for pure torsion loading case of structure for model-1, $\theta = \tan^{-1}(0.00185)$ then $\theta = 0.106$

$$K_T = \frac{1479.6\text{Nm}}{0.106} = 13958.5\text{Nm/degree}$$

Remark: the above torsional stiffness result for both model-1 and model-2 are under range of standard so the design of structure to stiffness is safe and the standard torsional stiffness for different vehicles[40] are illustrate in table bellows.

Table 9 Torsional Stiffness of different vehicles

Vehicle types	**Torsional stiffness (Nm/degree)**
Formula SAE car	1000-5000
Passenger car	5000-20000
Winston cup racing car	15000-30000
Sports car	15000-40000
Formula one car	10000-100000

3.3 Finite Element Modeling and analysis

Geometrical modeling of Bajaj Qute was carried out in SOLIDWORKS 18.0 and after performing structural modeling the next step is to define and prepare the developed model to further analysis in ANSYS Space claim and Workbench. To do so first the techniques attachment of SOLIDWORKS model into ANSYS should be defined according to the way model was formulated.

3.3.1 Importing or attachment of model to ANSYS

ACIS (*.sat, *.sab) format is suited and the best format to import the geometry to both in ANSYS space claim and Workbench according to the following criteria.

Table 10 Import Preference Support for ACIS geometry interface

Basic characteristics of ACIS format	
Import Solid Bodies	Yes
Import Surface Bodies	Yes
Import Line Bodies	Yes
Attribute Processing and prefix	Yes - Color, Layer, and Publication
Named Selection Processing and prefix	Yes - Color, Layer, and Publication
Material Processing	Yes
Analysis Type	2-D yes, 3-D yes

3.3.2 Setting Spot weldments for developed models

The first step of finite element modeling and analysis after importing assembled model to ANSYS is fixing the connection of each part of the assembled model by means of weldment according to the model type. Spot welds provide a mechanism to rigidly connect two discrete points in a model and can be used to represent welds, rivets, bolts, etc.

The points usually belong to two different surfaces and are defined on the geometry during the solver initialization process, the two points defining each spot weld will be connected by a rigid beam element. Additionally, rigid beam elements will be generated on each surface to enable transfer of rotations at the spot weld location[35].

For the case of Bajaj, Qute structure shell element would be defined by Spot weldments. Spot welding for Bajaj Qute structure was defined in ANSYS Space Claim with the following basic Spot welding parameters.

- ✓ Start offset =0 as a default
- ✓ End offset =0 as a default
- ✓ Number of points=8mm as a default
- ✓ Increment=19.69mm
- ✓ Model thickness=2mm

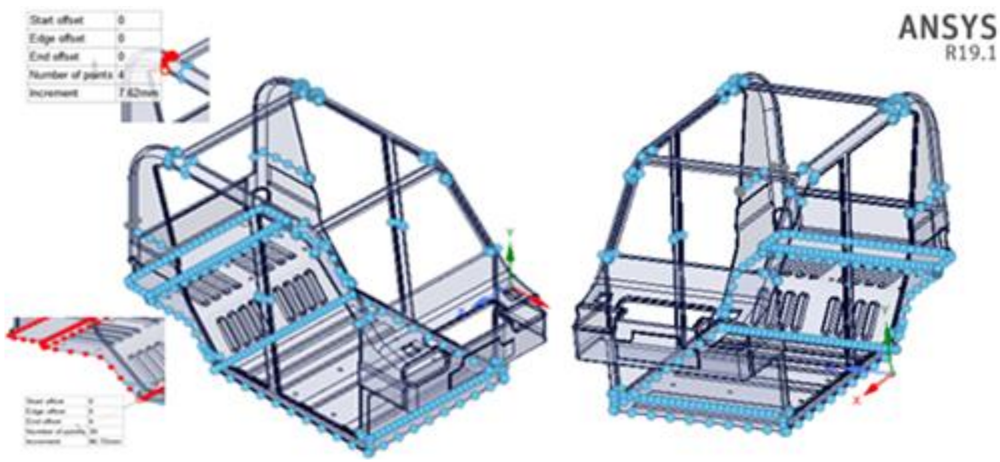


Figure 37 Shows Schematic representation of spot weldments for models of Bajaj Qute

3.3.3 Finite element modeling and static structure analysis for model-1 of Bajaj Qute structure

3.3.3.1 Basic input data for analysis of model-1 from literatures

- ✓ The mass of passengers in vehicle is taken as 65kg (for single person) and applied as appoint mass in the structure of vehicle[41].
- ✓ Total mass of vehicle is m=399Kg and the weight of vehicle become

$$w = m * g \text{ --- Equation-60}$$

$$w = 399kg * 9.81m/s2 \text{ And } w = 3914.19N$$

$$w = \frac{3914.19N}{4} = 978.5475N(\text{Distributed load on the structure})$$

3.3.3.2 Setting spot weldments for model-1

Previously, spot weldment was defined in ANSYS Space Claim so, no need to bear on here about contact between parts. But to be confidential concerning with weldment defined before checking is recommended in ANSYS workbench like described below

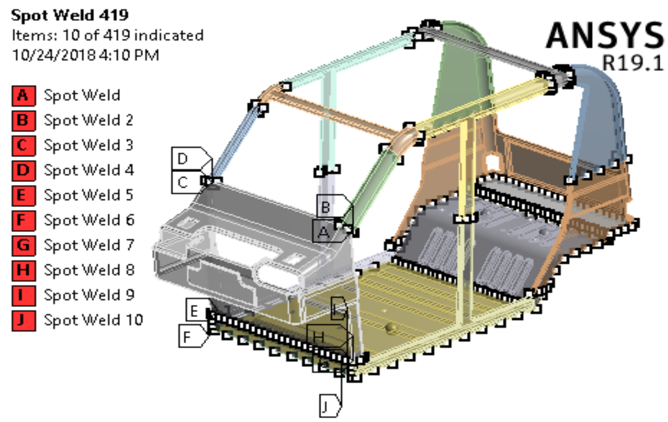


Figure 38 Shows ANSYS Workbench spot weldments for models-1

3.3.3.3 Mesh and mesh size control in Model-1

To have an efficient mesh and result it's better to control the mesh size and element of the given model hence, triangle methods is preferred for shell elements and elements order is taken as linear elements.

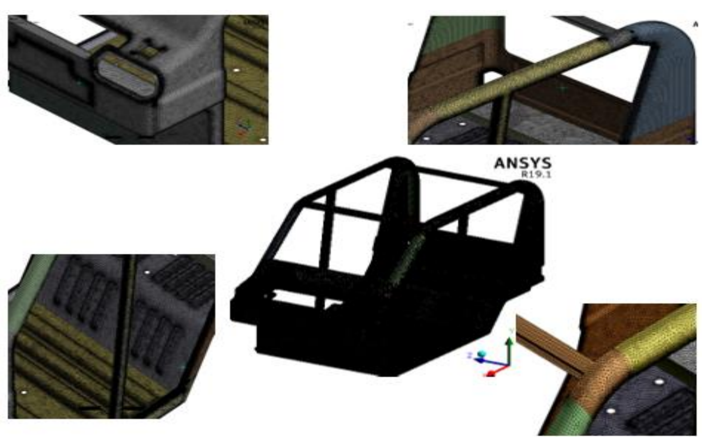


Figure 39 Shows meshing and mesh control size for models-1

Table 11 shows the number of nodes and element generated in meshing of model-1

Number of Nods for model-1	533820
Number of Elements for model-1	284741

3.3.3.4 Boundary conditions for static structural analysis model-1

In static structure analysis of any vehicles, load distribution in a vehicles body or structure is the key element to predict the deformation, equivalent von-misses stress, stiffness, strain etc. Of vehicles. Indeed, the boundary condition for static structural analysis of model-1 of Bajaj Qute is described using those loading confronted in a vehicle body as described in chapter one of this thesis paper.

Indeed, the boundary conditions varies according to the loading considerations. There are around seven loading types are exist for structural analysis of any vehicles but this research exclusively study the basic four loading type and for each loading types the boundary conditions are stated below.

1. Model-1 boundary conditions for pure bending Loading case

In pure bending loading conditions, the mass of passengers and standard earth gravity are the only loads that applied on the structure and fixed support on four points in which the four wheels are climbed.

Note that: model- 1 and model-2 differ only by their width so the same procedure are employ to analysis the structure of Bajaj Qute of boundary conditions.

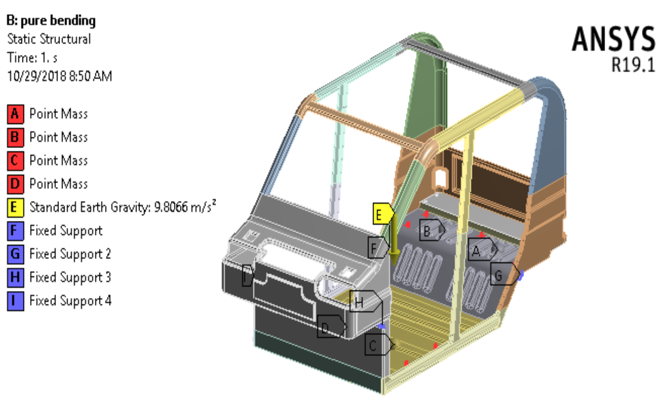


Figure 40 shows the boundary conditions for models-1 case of pure bending

And the hole defined to represent the mass of each passengers are 30mm diameter used to mount the seat to the main structure of Bajaj Qute.

Table 12 shows the boundary conditions seated for analysis of pure loading of model-1

Boundary conditions	**Values/description**
mass of passengers (m_p)	65kg for single person
Number of seats	3(passengers) +1(driver)
Standard earth gravity(g)	9.81m/s^2(- y direction)
Fixed support at	Four points on wheel mounting areas

As shown figure above in the ANSYS WORKBENCH the mass of each passengers is defined as point in 12 holes, holes are used to climb up the seat to the structure and 3 holes are used to define one point mass.

2. Model-1 boundary conditions for pure Torsion loading case

For the case of pure torsion loading conditions, the total mass of vehicle multiplied with standard earth gravity(g) to have total weight of vehicle(w) and the weight of vehicle is equally distributed to four point in which the wheels are mounted.

$$w = 978.5475\text{N}$$

Is applies in opposite direction of two wheels and other two wheels are fixed and there is twisting effect on the structure.

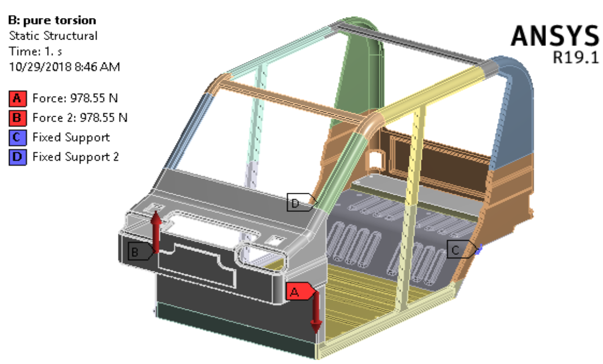


Figure 41 shows the boundary conditions for models-1 case of pure torsion

Table 13 shows the boundary conditions seated for pure torsion of model-2

Boundary conditions	**Values/description**
Weight of vehicles/4	978.5475N applied at two wheels mount area of opposite direction.
Fixed support	On two wheels mounting areas

3. model-1 boundary conditions for Horizontal Lozenging case

In Horizontal Lozenging loading case there is only one free wheel mounting area and the rest three are fixed and standard earth gravity is applied in –y direction.

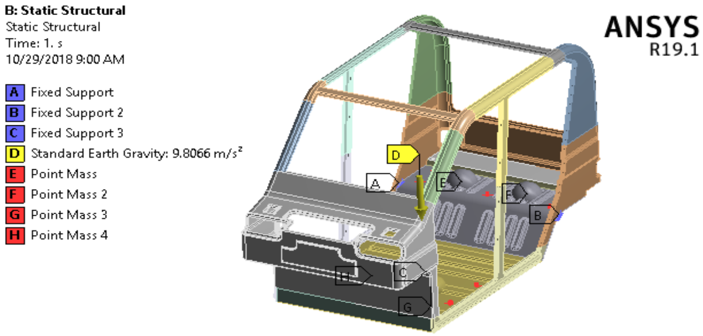


Figure 42 shows the boundary conditions for models-1 case of horizontal Lozenging

Table 14 shows the boundary conditions seated for horizontal Lozenging of model-1

Boundary conditions	Values/description
mass of passengers (m_p)	65kg for single person
Number of seats	3(passengers) +1(driver)
Standard earth gravity(g)	9.81m/s^2(- y direction)
Fixed support at	Three points on wheel mounting areas and one is free

4. model-1 boundary conditions for Combined loading case

Since combined loading is a combinations of loads of pure bending and pure torsion cases so, those boundary conditions used for pure bending and torsions are apply here as a boundary conditions. Two of wheel mounting areas are fixed plus equal and opposite forces are applied on the other two wheel mounting surfaces.

Besides this the mass of each passengers are defined as a point mass and applied on those 12 holes of seat mounting areas. Likewise the standard earth gravity at center of mass is applied in the negative Y direction.

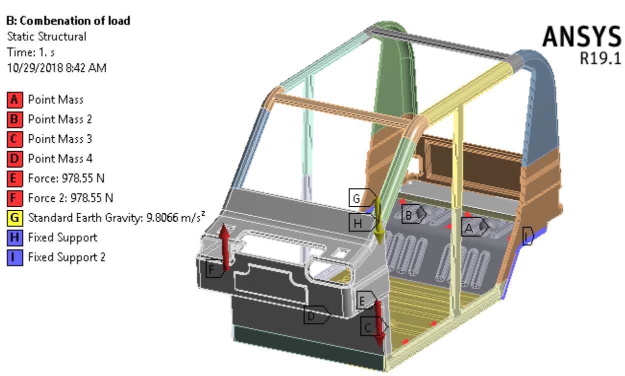


Figure 43 shows the boundary conditions for models-1 case of combined loading

Table 15 shows the boundary conditions seated for combined loading of model-1

Boundary conditions	**Values/description**
Weight at each wheel	978.55 N
Number of seats	3(passengers) +1(driver)
Standard earth gravity(g)	9.81m/s²(- y direction)
Fixed support at	two points on wheel mounting areas

3.3.4 Finite element modeling and static structure analysis for model-2

3.3.4.1 Define basic inputs data for model-2

Like that of model-1, model-2 used the same material and input data for analysis only width of model-2 is different than model-2

- ✓ The mass of passengers in vehicle is taken as 65kg (for single person) and applied as appoint mass in the structure of vehicle[41].
- ✓ Total mass of vehicle is m=399Kg and the weight of vehicle become

$$\mathrm{w} = \mathrm{m} * \mathrm{g} \text{ --- Equation-61}$$

$$\mathrm{w} = 399\mathrm{kg} * 9.81\mathrm{m/s2} \text{ And } w = 3914.19\mathrm{N}$$

$$w = \frac{3914.19\mathrm{N}}{4} = 978.5475\mathrm{N}(\text{Distributed load on the structure})$$

3.3.4.2 Setting spot weldments for model-2

The weldment for model-2 of Bajaj Qute described below and it's similar to the previous model.

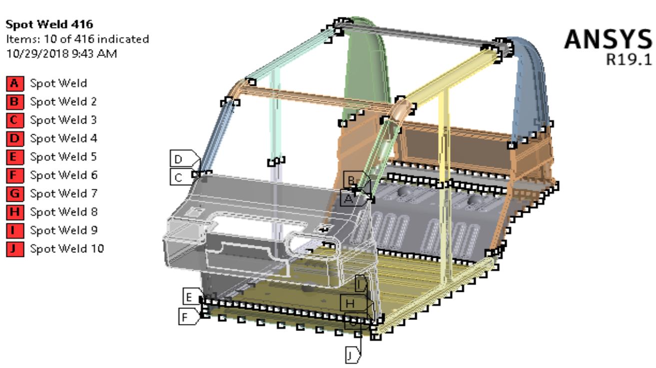


Figure 44 Shows ANSYS Workbench spot weldments for models-2

3.3.4.3 Mesh and mesh size control in Model-2

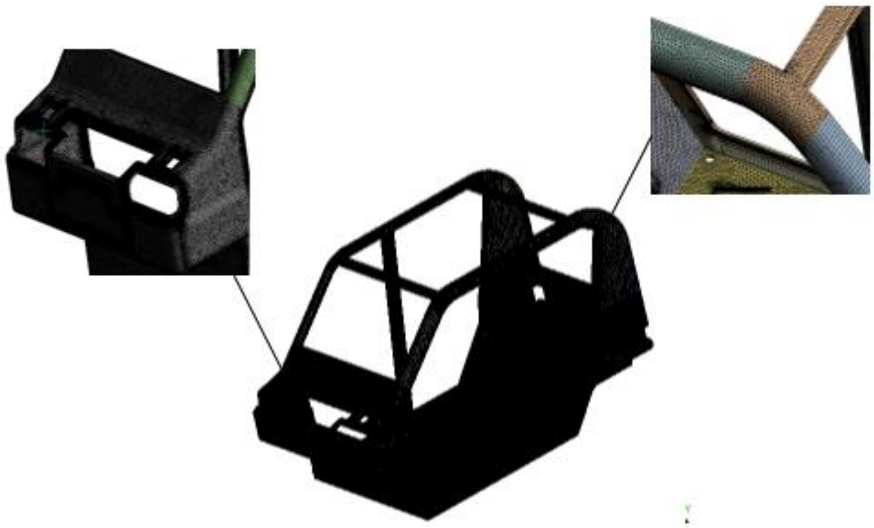


Figure 45 Shows meshing and mesh control size for models-2

Table 16 Shows the number of nodes and element generated in meshing of model-2

Number of Nods for model-2	533820
Number of Elements for model-2	284741

1. **Model-2 boundary conditions for pure bending Loading case**

Like that of model-1, for model-2 pure bending loading conditions, the mass of passengers and standard earth gravity are the only loads that applied on the structure and fixed support on four points in which the four wheels are climbed.

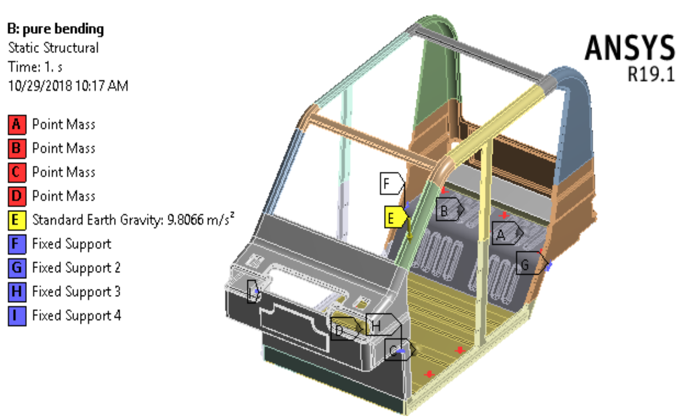


Figure 46 shows the boundary conditions for models-2 case of pure bending

Table 17 shows the boundary conditions seated for analysis of pure loading of model-2

Boundary conditions	**Values/description**
mass of passengers (m_p)	65kg for single person
Number of seats	3(passengers) +1(driver)
Standard earth gravity(g)	$9.81m/s^2$(- y direction)
Fixed support at	Four points on wheel mounting areas

2. Model-2 boundary conditions for pure Torsion loading case

For the case of pure torsion loading conditions, the total mass of vehicle multiplied with standard earth gravity(g) to have total weight of vehicle(w) and the weight of vehicle is equally distributed to four point in which the wheels are mounted.

$$w = \frac{3914.19N}{4} = 978.5475N$$

Here like model-1 torsion loading case the value $978.5475N$ applies in opposite direction of two wheels and other two wheels are fixed and there is twisting effect will produce.

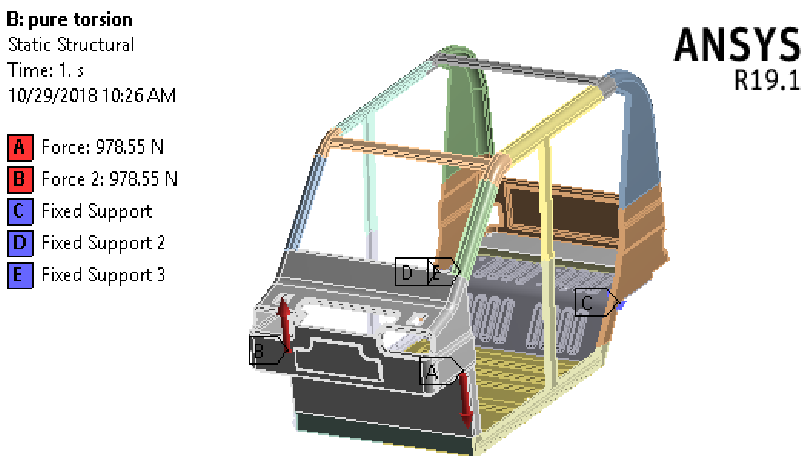


Figure 47 shows the boundary conditions for models-2 case of pure torsion

Table 18 shows the boundary conditions seated for pure torsion of model-2

Boundary conditions	**Values/description**
Weight of vehicles/2	3914.19N /2=978.5475N applied at two wheels mount area of opposite direction.
Fixed support	On two wheels mounting areas

3. model-2 boundary conditions for Horizontal Lozenging case

In Horizontal Lozenging loading case there is only one free wheel mounting area and the rest three are fixed and standard earth gravity is applied in –y direction.

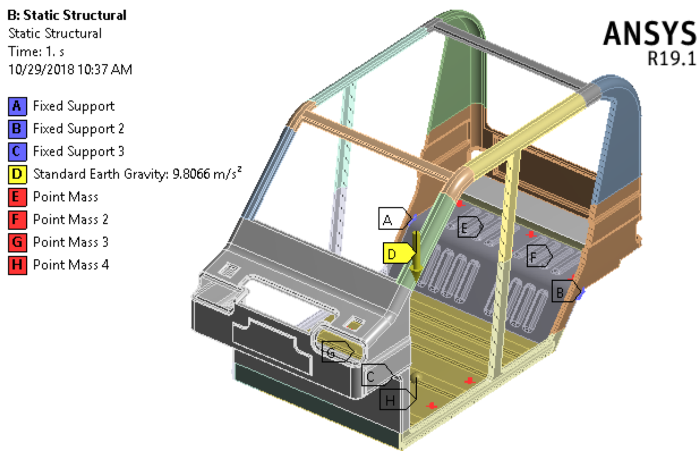


Figure 48 shows the boundary conditions for models-2 case of horizontal Lozenging

It's also including the mass of passengers as a boundary condition for horizontal Lozenging cases.

Table 19 shows the boundary conditions seated for horizontal Lozenging of model-2

Boundary conditions	**Values/description**
mass of passengers (m_p)	65kg for single person
Number of seats	3(passengers) +1(driver)
Standard earth gravity(g)	9.81m/s²(- y direction)
Fixed support at	Three points on wheel mounting areas and one is free

4. Model-2 boundary conditions for Combined loading case (pure bending +pure torsion)

Since combined loading is a combinations of loads of pure bending and pure torsion cases so, those boundary conditions used for pure bending and torsions are apply here as a boundary conditions. Two of wheel mounting areas are fixed plus equal and opposite forces are applied on the other two wheel mounting surfaces.

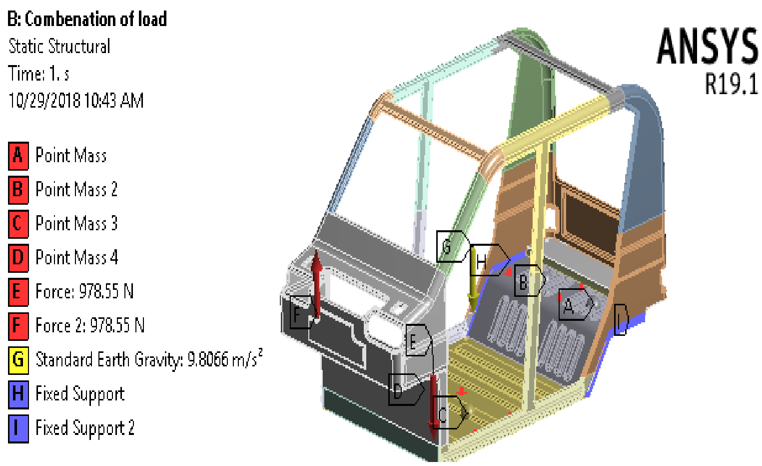


Figure 49 shows the boundary conditions for models-1 case of combined loading

Besides this the mass of each passengers are defined as a point mass and applied on those 12 holes of seat mounting areas.

Table 20 shows the boundary conditions seated for combined loading of model-2

Boundary conditions	**Values/description**
Weight at each wheel	978.55 N
Number of seats	3(passengers) +1(driver)
Standard earth gravity(g)	9.81m/s²(- y direction)
Fixed support at	two points on wheel mounting areas

3.3.5 Finite Element Modeling and impact analysis of Frontal Panel

The vehicle structural elements consist of front panel (rails), bumpers, pillars and rockers among those elements, the front panel are the structural components which play a main role in the energy absorption and can be considered as the most effective parameter in the design of vehicle's safety because it absorbs approximately up to 55.3% of the kinetic energy of the vehicle in case of frontal crashes[42].that is the reason why the impact simulation is performed using frontal panel.

3.3.5.1 Basic inputs Parameters

For the finite element modeling and impact analysis of frontal panel, three models are developed and they only different between them is the number of holes created on the surface of structure.

- ✓ Impact velocity for frontal panel(crash box) is taken as 12.5m/s that applied on frontal panel impact simulation [43].
- ✓ The hole size that created on the frontal panel structure have a size of 10mm for each holes[42].

Table 21 shows the basic input parametrizes used to model frontal panel

	Basic Dimensions	**Input Values**
Model -1 **Model-2** **Model-3**	Length	504mm
	Height	1131mm
	Width	1512mm

3.3.5.2 Material properties of Frontal panels for three models

Material used impact analysis is the same with static analysis and the three model thickness of stated in table-24

Table 22 Johnson cook and material properties for frontal panel impact analysis [31]

Structural steel sheet (St-60)	
Mechanical properties	Values
Yield strength	550 Mpa
Ultimate tensile strength	630MPa
Young's modulus(E)	200000 MPa
Poison 's ratio(v)	0.3
Density(ρ)	7850 kg/m3
Elongation	20%
Thickness of model	3mm
Johnson cook failure constant (D1,D2,D3,D4,D5)	(0.0705,1.732,-0.54.-0.015,0) respectively

3.3.6 Finite Element Modeling and impact analysis of Frontal Panel -1(model-1)

Model-1 have hollows on the structure that created to increase the energy absorption behavior of the frontal panel by converting most of the kinetic energy during a crash situation into other forms of energy in a predictable and controllable manner.

3.3.6.1 Meshing and mesh size control for frontal panel-1 (model-1)

Size controlling tools of meshing enables to reduce the time take to finish one single simulation of given model so, for model-1 the following mesh size are used and applied.

Table 23 shows meshing, method and mesh size control for frontal panel-1

Mesh Size applied part	Mesh Method	Element size
Frontal panel	Triangular method	20mm
Rigid well	Triangular method	40mm

Meshing method for model-1 perform with linear element order for both frontal panel as well as for rigid well.

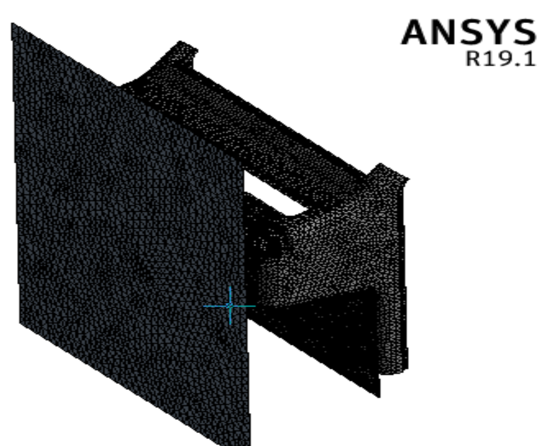


Figure 50 Shows meshing and mesh control size for models-1 of panel

3.3.6.2 Boundary conditions for impact analysis of frontal panel-1 (model-1)

For impact analysis of frontal panel-1 total mass of a vehicles is defined on panel, velocity of vehicle is taken as 12.5m/s[43] that apply on frontal panel and the wall is taken as rigid(non-deformable) and it is also fixed

$$\text{Total mass of vehicle(Mt)} = \text{mass of vehicle} + \text{mass of passangers}\text{------------- Equation-62}$$
$$\text{Mt} = 399\text{Kg} + 65\text{Kg} * 4$$
$$\text{Mt} = 659\text{Kg}$$

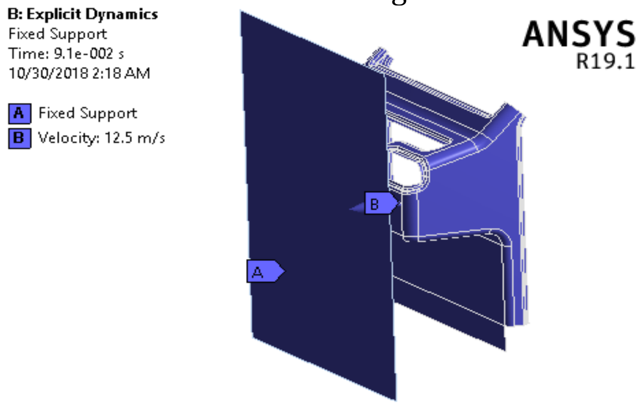


Figure 51 shows the boundary conditions for impact analysis of models-1

Note that: - the total kinetic energy for frontal panel-1 can be calculate as follows

$$\text{KE} = \frac{1}{2}(\text{Mv} + \text{Mp})\text{V}^2\text{--- Equation-63}$$

Where Mv=mass of vehicle, Mp=passengers mass, $\text{KE} = \frac{1}{2}(65 * 4 + 399) * 12.5^2 = 51.5\text{KJ}$

3.3.7 Finite Element Modeling and impact analysis of Frontal Panel -2(model-2)

Model-2 have hollows on the structure that created to increase the energy absorption behavior of the frontal panel by converting most of the kinetic energy during a crash situation into other forms of energy in a predictable and controllable manner.

3.3.7.1 Meshing and mesh control for frontal panel-2 (model-2)

Size controlling tools of meshing enables to reduce the time take to finish one single simulation of given model so, for model-2 the following mesh size are used and applied.

Table 24 shows meshing, method and mesh size control for frontal panel-2

Mesh Size applied part	Mesh Method	Element size
Frontal panel	Triangular method	20mm
Rigid well	Triangular method	40mm

Meshing method for model-2 carry through with linear element order for both frontal panel as well as for rigid well.

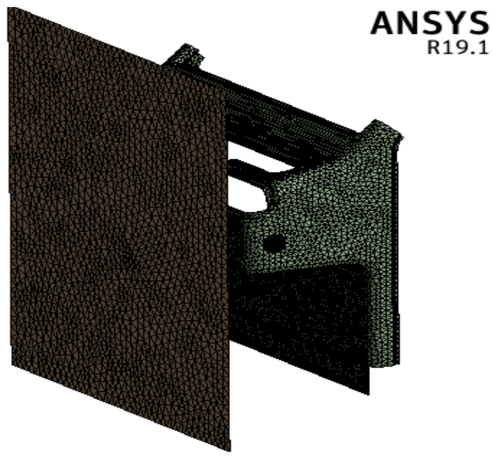


Figure 52 Shows meshing and mesh control size for models-2 of frontal panel

3.3.7.2 Boundary conditions for impact analysis of frontal panel-2 (model-2)

For impact analysis of frontal panel-2 total mass of a vehicles is defined on panel, velocity of vehicle is taken as 12.5m/s[43] that apply on frontal panel and the wall is taken as rigid(non-deformable) and it is also fixed

$$\text{Total mass of vehicle(Mt)} = \text{mass of vehicle} + \text{mass of passangers}\text{------------- Equation-64}$$

$$\mathrm{Mt} = 397.2\mathrm{Kg} + 65\mathrm{Kg} * 4$$

$$\mathrm{Mt} = 657.2\mathrm{Kg}$$

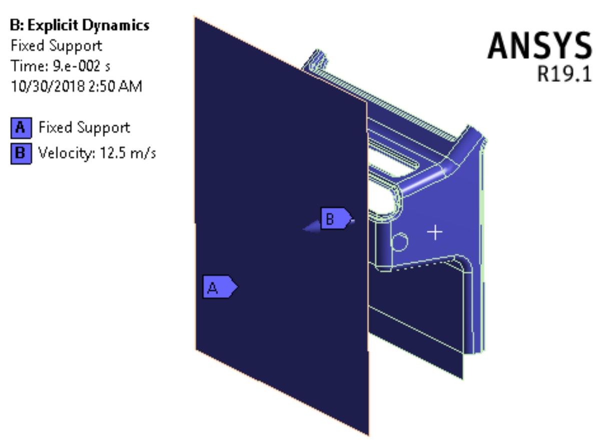


Figure 53 shows the boundary conditions for impact analysis of models-2

<u>Note that</u>: - the total kinetic energy for frontal panel-2 can be calculate as follows

$$KE = \frac{1}{2}(Mv + Mp)V^2 \text{-- Equation-65}$$

Where, Mv=mass of vehicle, Mp=passengers mass, $KE = \frac{1}{2}(65 * 4 + 397.2) * 12.5^2 = 51.4KJ$

3.3.8 Finite Element Modeling and impact analysis of Frontal Panel -3(model-3)

Compared to other two Models of frontal panel, Model 3 have a large number of holes in its surface and those holes are increase the energy absorption behavior of the panel.

3.3.8.1 Meshing and mesh size control for frontal panel (model-3)

Table 25 shows meshing, method and mesh size control for frontal panel-3

Mesh Size applied part	Mesh Method	Element size
Frontal panel	Triangular method	20mm
Rigid well	Triangular method	40mm

Meshing method for model-3 perform with linear element order for both frontal panel as well as for rigid well.

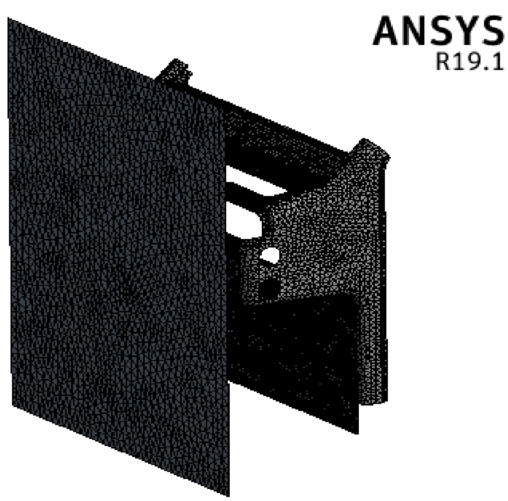


Figure 54 Shows meshing and mesh control size for models-3 of frontal panel

3.3.8.2 Boundary conditions for impact analysis of frontal panel-3 (model-3)

For impact analysis of frontal panel-3 total mass of a vehicles is defined on panel, velocity of vehicle is taken as 12.5m/s[43] that apply on frontal panel and the wall is taken as rigid(non-deformable) and it is also fixed

$$\text{Total mass of vehicle(Mt)} = \text{mass of vehicle} + \text{mass of passangers}$$

$$\text{Mt} = 396.6\text{Kg} + 65\text{Kg} * 4$$
$$\text{Mt} = 656.6\text{Kg}$$

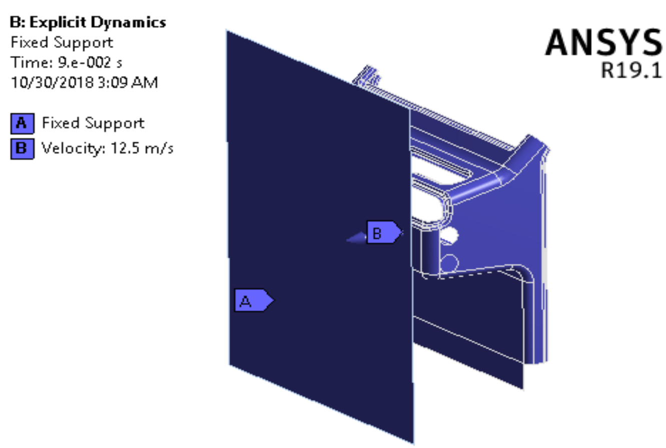


Figure 55 shows the boundary conditions for impact analysis of models-3

<u>Note that</u>: - The total kinetic energy for frontal panel-3 can be calculate as follows

$$\text{KE} = \frac{1}{2}(\text{Mv} + \text{Mp})\text{V}^2 \text{-- Equation-66}$$

$$\text{where Mv} = \text{mass of vehicle}, \text{Mp} = \text{mass of passaners},$$

$$\text{KE} = \frac{1}{2}(65 * 4 + 396.6) * 12.5^2 = 51.3\text{KJ}$$

CHAPTER FOUR

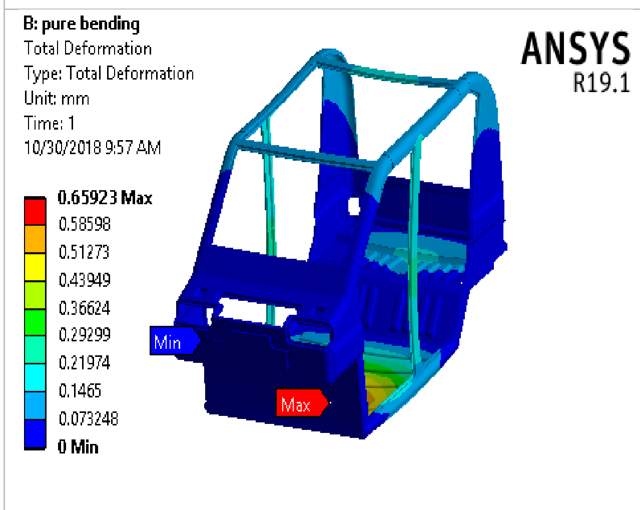
RESULTS AND DISCUSSION

4.1 Result for Static structure analysis of Bajaj Qute body

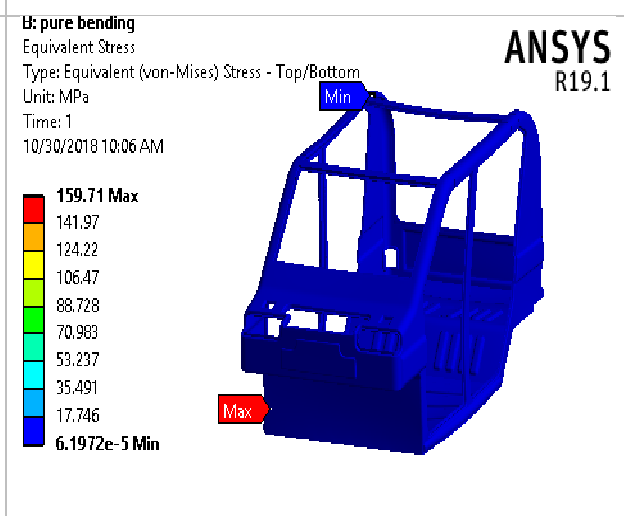
4.1.1 Result for Static structure of Bajaj Qute Model-1

4.1.1.1 Result for Pure bending loading case of model-1

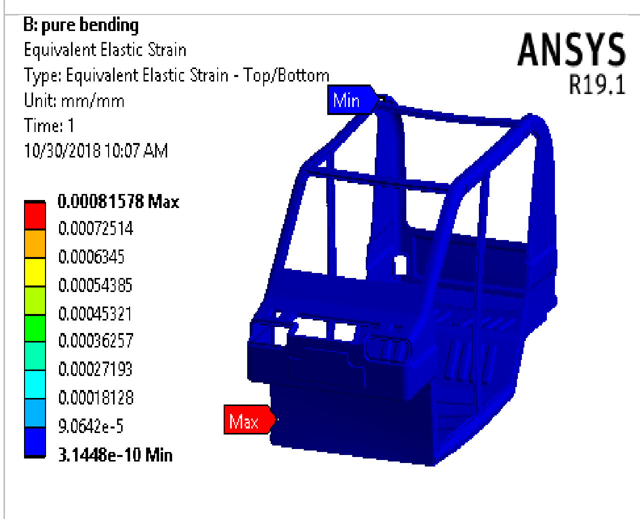
Total Deformation



Equivalent (von-Miss) Stress



Equivalent Elastic Strain



Strain Energy

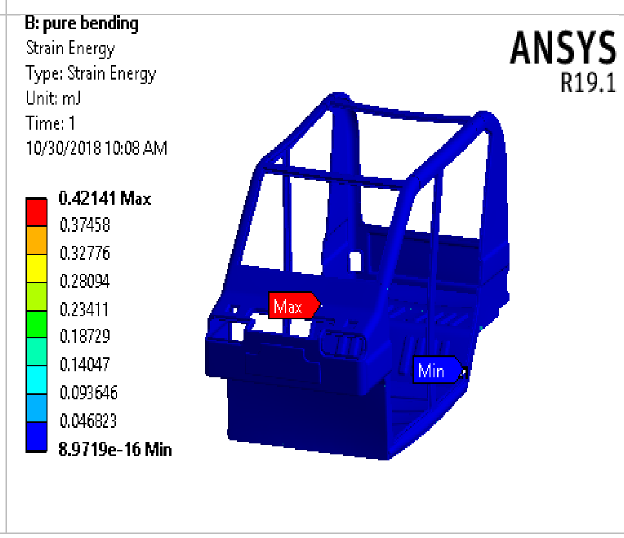


Figure 56 shows the result for pure bending loading for models-1 of Qute structure

Table 26 Shows reaction forces and moment result for pure bending of model-1

Reaction Force in the X [F_X]	**Reaction Force in the Y [F_Y]**	**Reaction Force in the Z [F_Z]**
255.2N	1635.8N	1047.9N
Moment in the X [M_X]	**Moment in the Y [M_Y]**	**Moment in the Z [M_Z]**
744.59N.mm	6530.6N.mm	655.82N.mm

4.1.1.2 Result for Pure torsion loading case of model-1

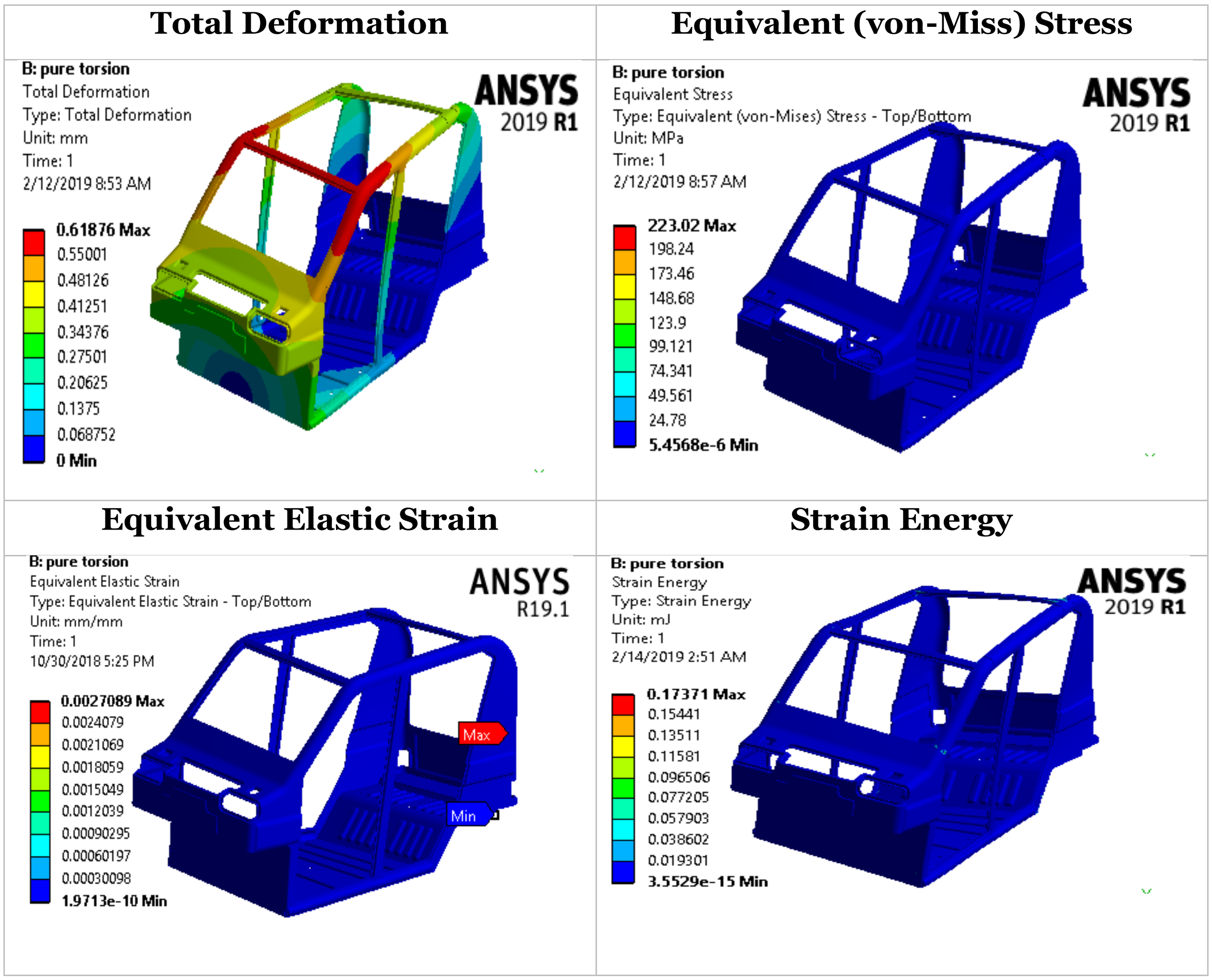


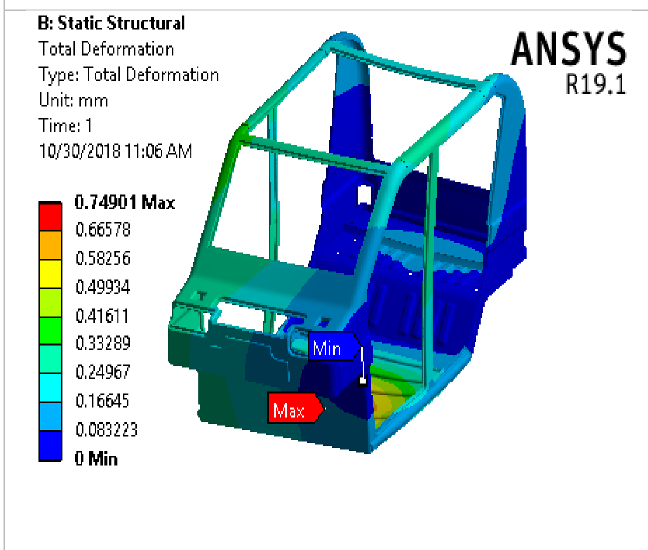
Figure 57 shows the result for pure torsion loading for models-1 of Qute structure

Table 27 Shows reaction forces and moment result for pure torsion of model-1

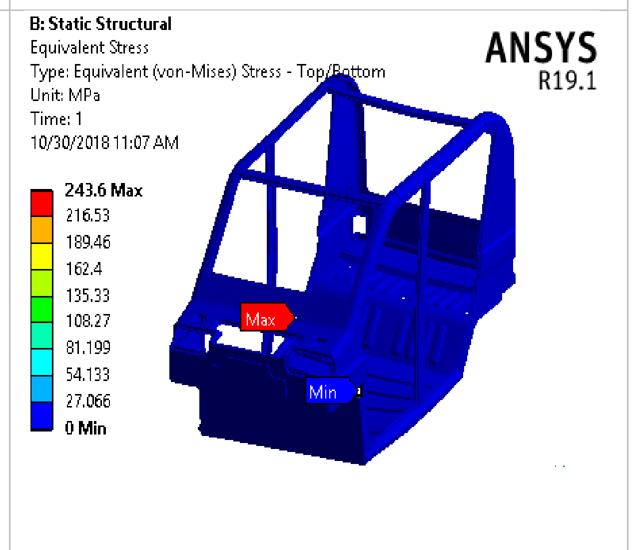
Reaction Force in the X [F_x]	**Reaction Force in the Y [F_Y]**	**Reaction Force in the Z [F_Z]**
1036.N	438.91N	41.52N
Moment in the X [M_x]	**Moment in the Y [M_Y]**	**Moment in the Z [M_Z]**
10833N.mm	0.5584N.mm	5166.9N.mm

4.1.1.3 Result for Horizontal Lozenging loading case of model-1

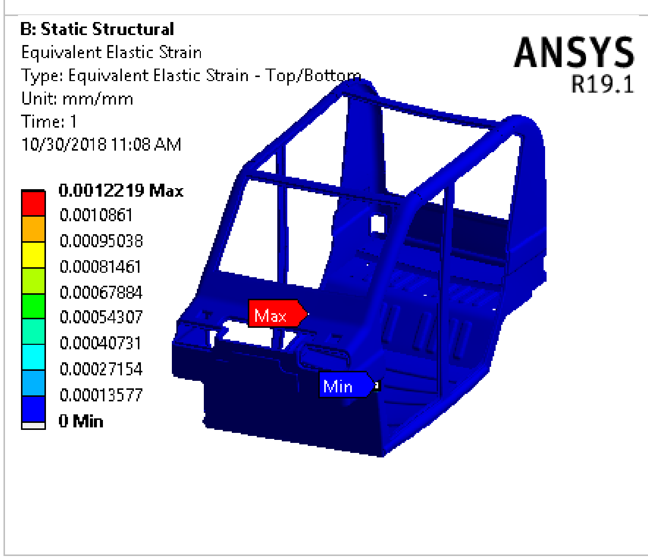
Total Deformation



Equivalent (von-Miss) Stress



Equivalent Elastic Strain



Strain Energy

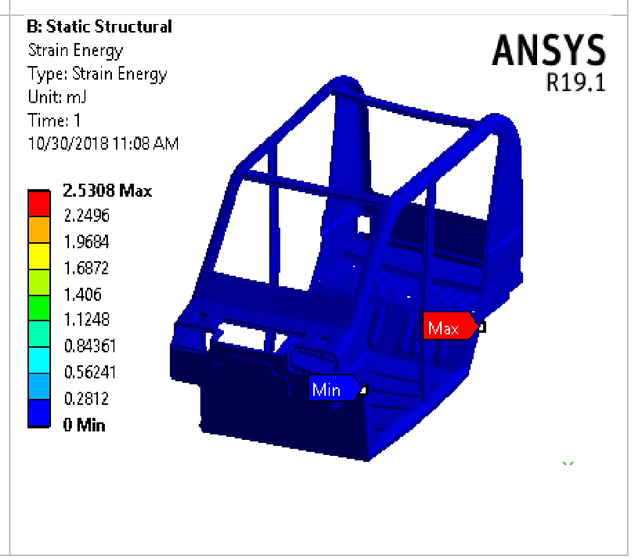


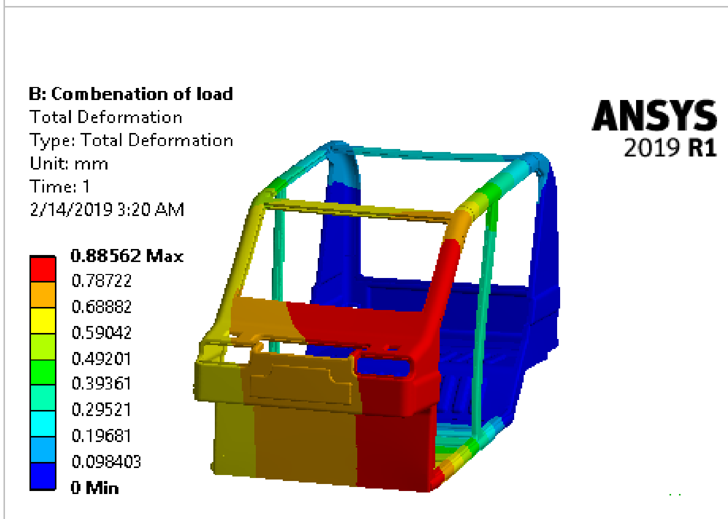
Figure 58 shows the result for Horizontal Lozenging of models-1 of Qute structure

Table 28 shows reaction forces and moment result for horizontal Lozenging of model-1

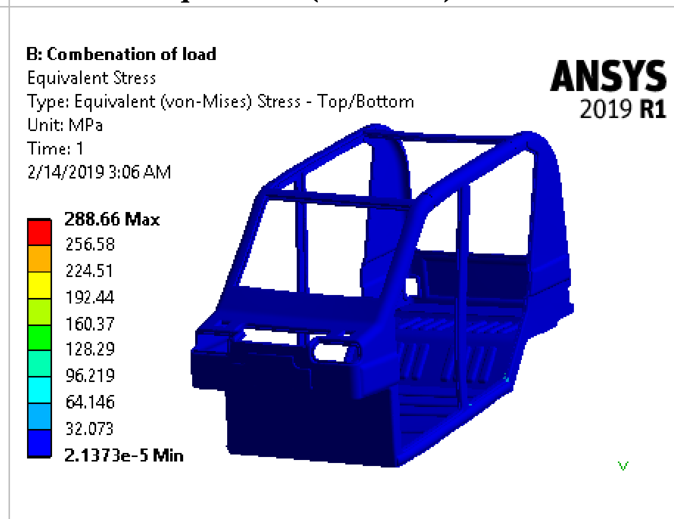
Reaction Force in the X [F_x]	**Reaction Force in the Y [F_Y]**	**Reaction Force in the Z [F_Z]**
2794.7N	2725.4N	3513.1 N
Moment in the X [M_x]	**Moment in the Y [M_Y]**	**Moment in the Z [M_Z]**
17216N.mm	10327N.mm	2588.9N.mm

4.1.1.4 Result for Combinations of loading case of model-1

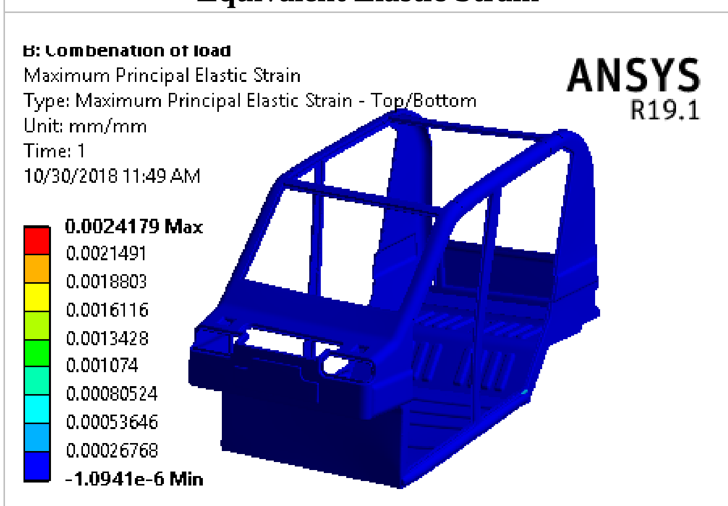
Total Deformation



Equivalent (von-Miss) Stress



Equivalent Elastic Strain



Strain Energy

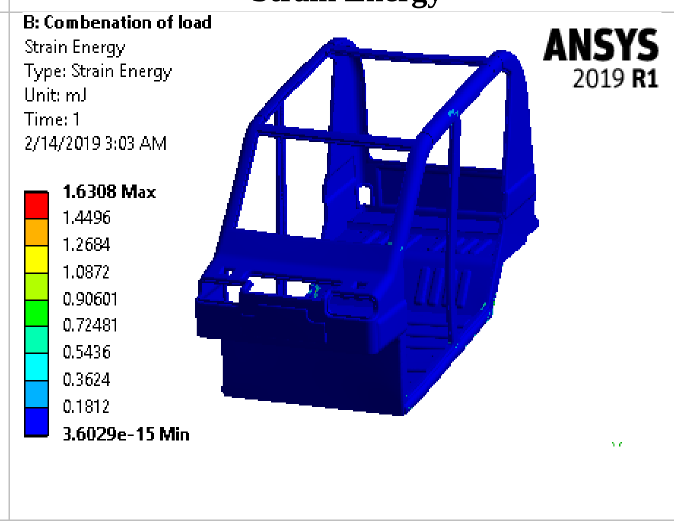


Figure 59 shows the result for Combined loading of models-1 of Qute structure

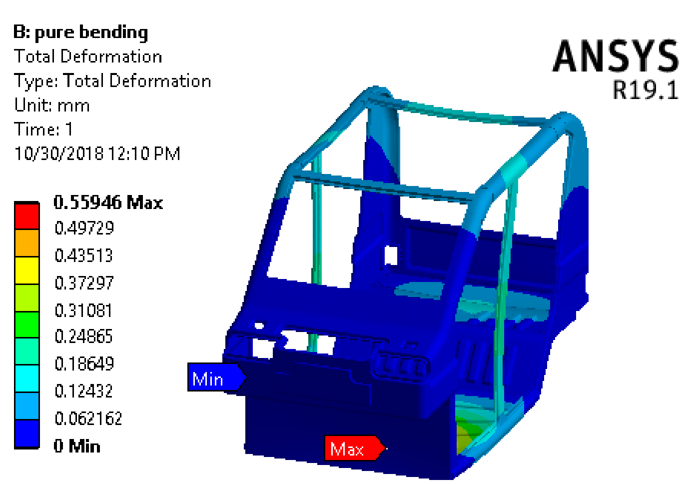
Table 29 shows reaction forces and moment result for combined loading of model-1

Reaction Force in the X [F_x]	Reaction Force in the Y [F_Y]	Reaction Force in the Z [F_Z]
14931N	1588.2N	1485.5 N
Moment in the X [M_x]	**Moment in the Y [M_Y]**	**Moment in the Z [M_Z]**
42426N.mm	99713N.mm	29572N.mm

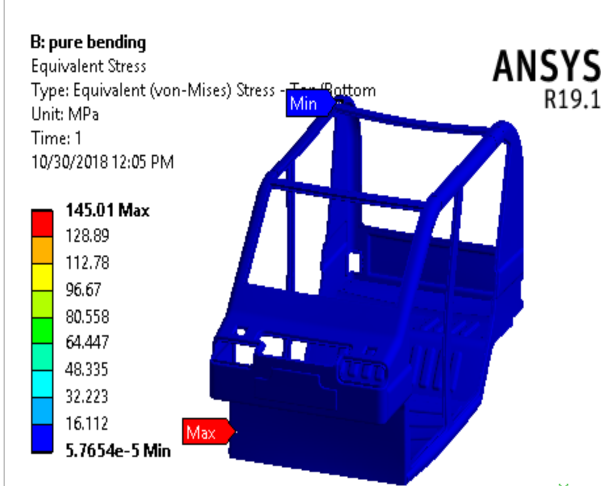
4.1.2 Result for Static structure of Bajaj Qute of Model-2

4.1.2.1 Result for Pure bending loading case of model-2

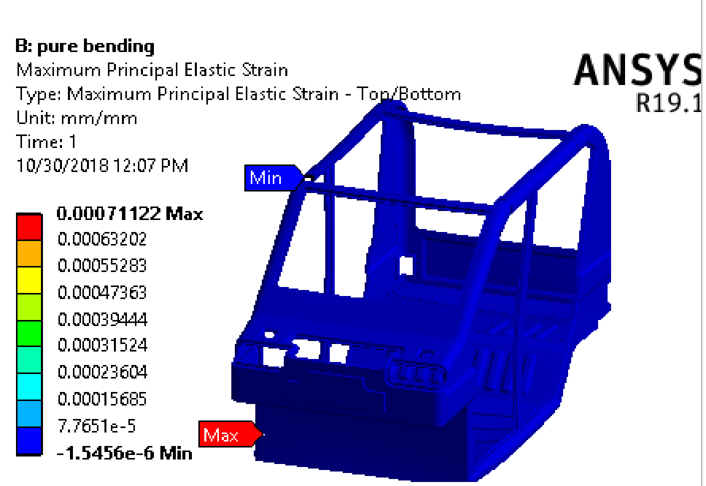
Total Deformation



Equivalent (von-Miss) Stress



Equivalent Elastic Strain



Strain Energy

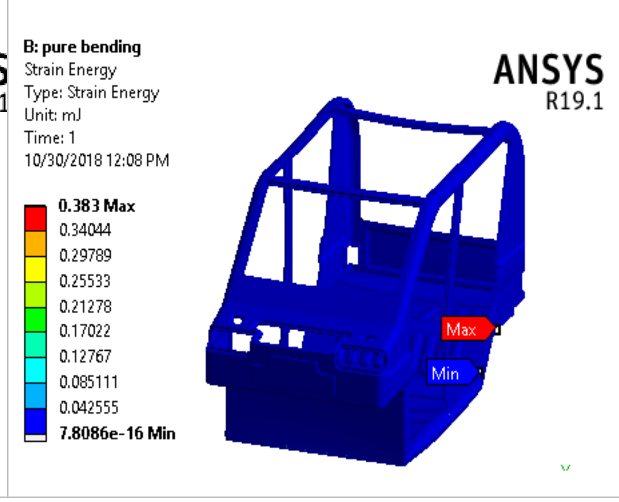


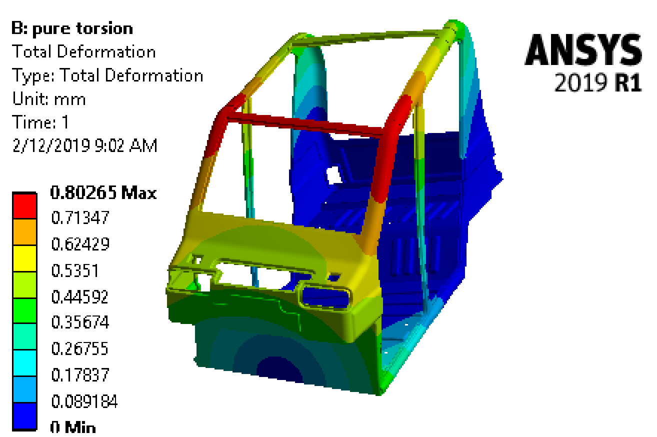
Figure 60 shows the result for pure bending loading for models-2 of Qute structure

Table 30 shows reaction forces and moment result for pure bending of model-2

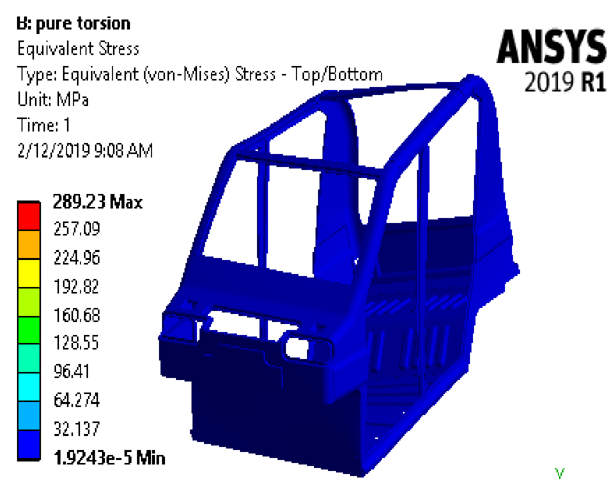
Reaction Force in the X [F_x]	**Reaction Force in the Y [F_Y]**	**Reaction Force in the Z [F_Z]**
291.9N	1654.2N	1089.8N
Moment in the X [M_x]	**Moment in the Y [M_Y]**	**Moment in the Z [M_Z]**
827.53N.mm	6486.N.mm	801.26N.mm

4.1.2.2 Result for Pure torsion loading case of model-2

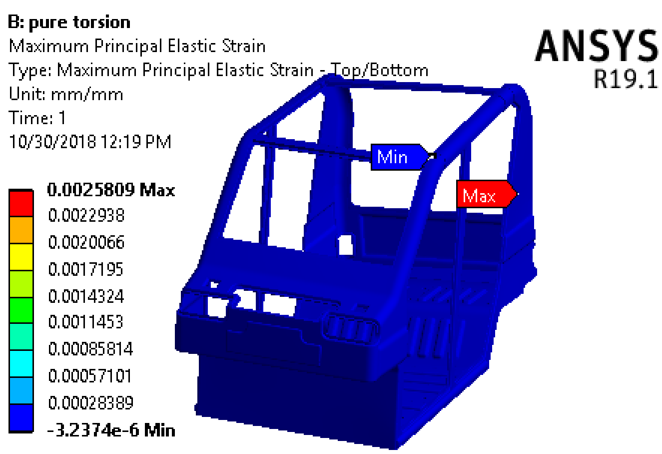
Total Deformation



Equivalent (von-Miss) Stress



Equivalent Elastic Strain



Strain Energy

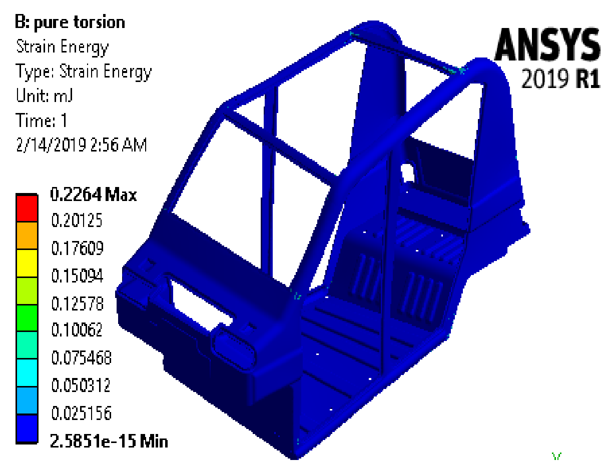


Figure 61 shows the result for pure torsion loading for models-2 of Qute structure

Table 31 shows reaction forces and moment result for pure torsion of model-2

Reaction Force in the X $[F_X]$	Reaction Force in the Y $[F_Y]$	Reaction Force in the Z $[F_Z]$
852.39N	640.33N	180.4N
Moment in the X $[M_X]$	**Moment in the Y $[M_Y]$**	**Moment in the Z $[M_Z]$**
10143N.mm	1889.4N.mm	401.35N.mm

4.1.2.3 Result Horizontal Lozenging loading case of model-2

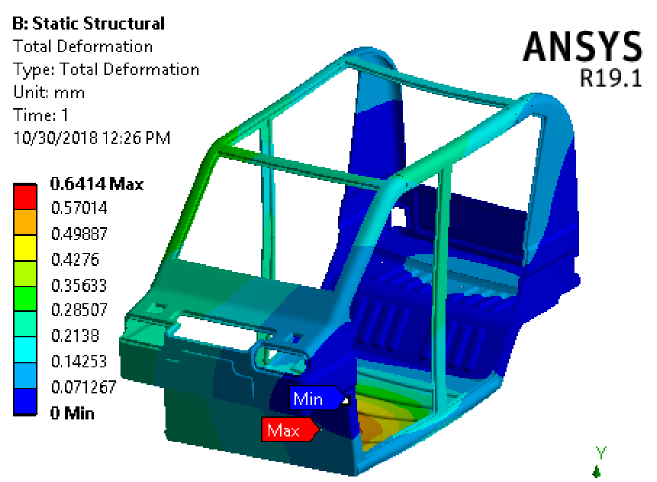
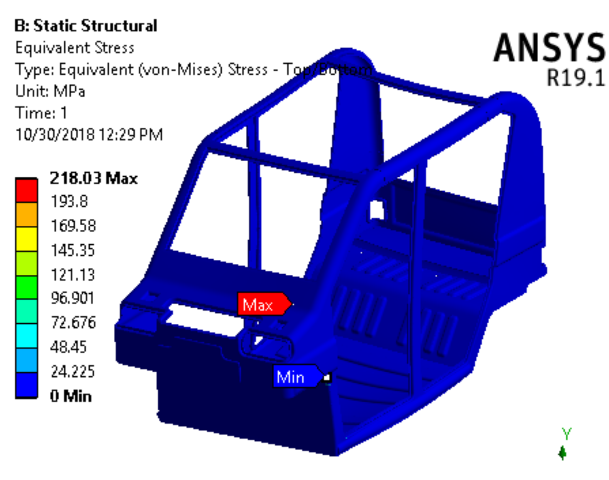
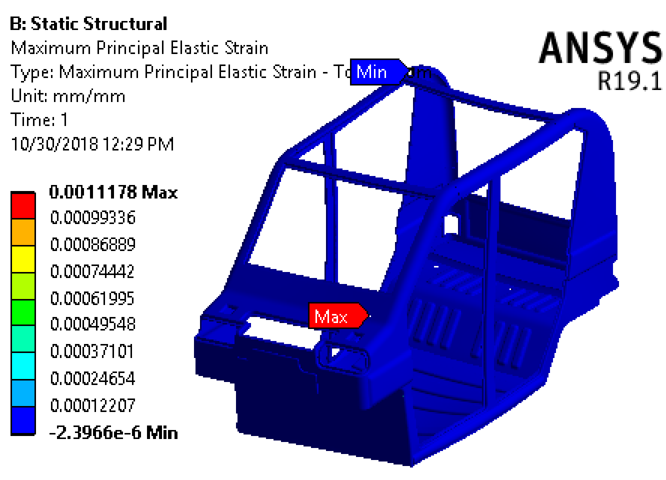
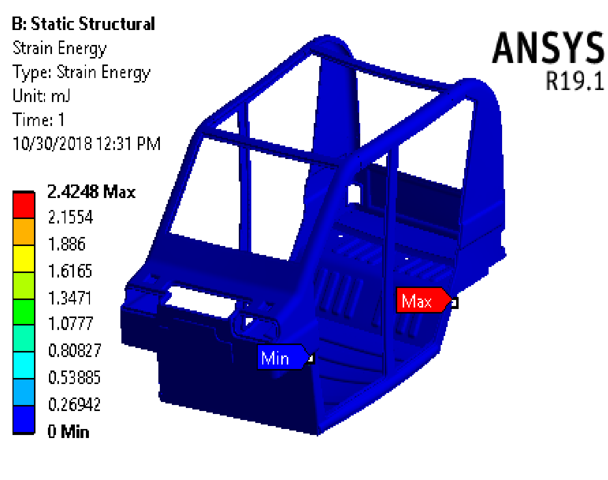
Total Deformation	Equivalent (von-Miss) Stress
	
Equivalent Elastic Strain	**Strain Energy**
	

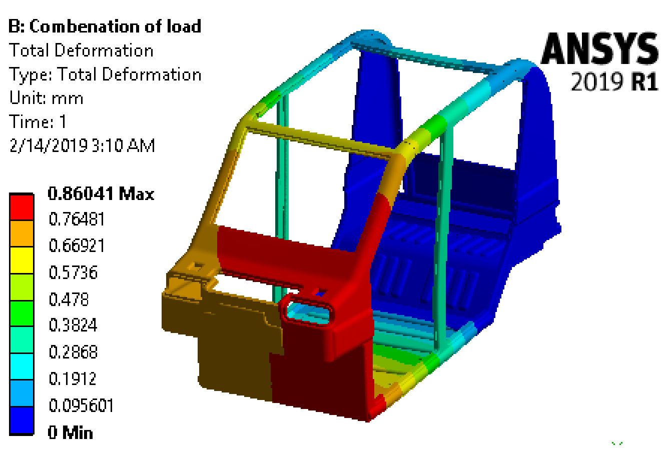
Figure 62 shows the result for Horizontal Lozenging of models-2 of Qute structure

Table 32 shows reaction forces and moment result for horizontal Lozenging of model-2

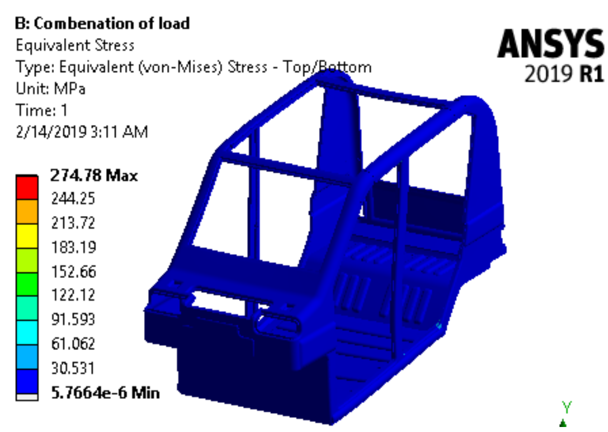
Reaction Force in the X [F_x]	**Reaction Force in the Y [F_Y]**	**Reaction Force in the Z [F_Z]**
2955.7N	2821.8N	3647.9N
Moment in the X [M_x]	**Moment in the Y [M_Y]**	**Moment in the Z [M_Z]**
19265N.mm	-10297N.mm	-46.044N.mm

4.1.2.4 Result Combination of loading case of model-2

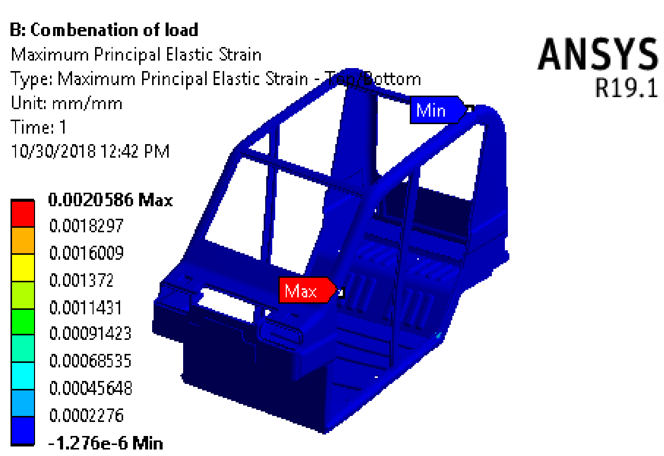
Total Deformation



Equivalent (von-Miss) Stress



Equivalent Elastic Strain



Strain Energy

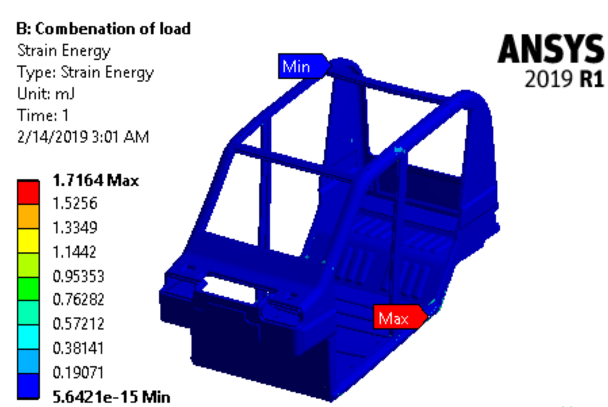


Figure 63 shows the result for Combined loading of models-2 of Qute structure

Table 33 shows reaction forces and moment result for combined loading of model-2

Reaction Force in the X [F_X]	**Reaction Force in the Y [F_Y]**	**Reaction Force in the Z [F_Z]**
15236N	1729.3N	1527N
Moment in the X [M_X]	**Moment in the Y [M_Y]**	**Moment in the Z [M_Z]**
38404N.mm	101650N.mm	30233N.mm

4.1.3 Result for impact analysis of frontal panel model-1

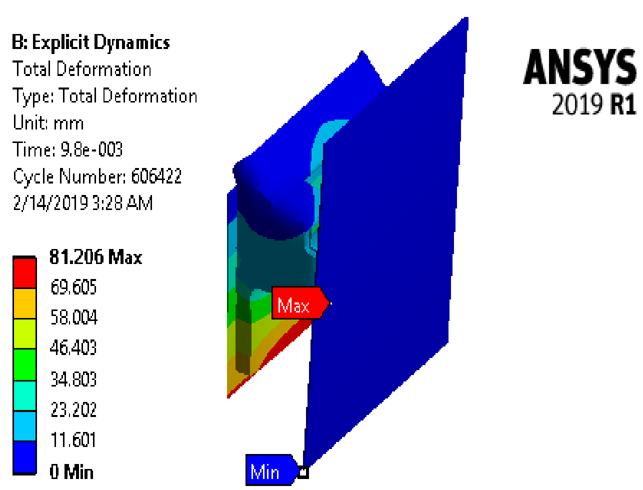
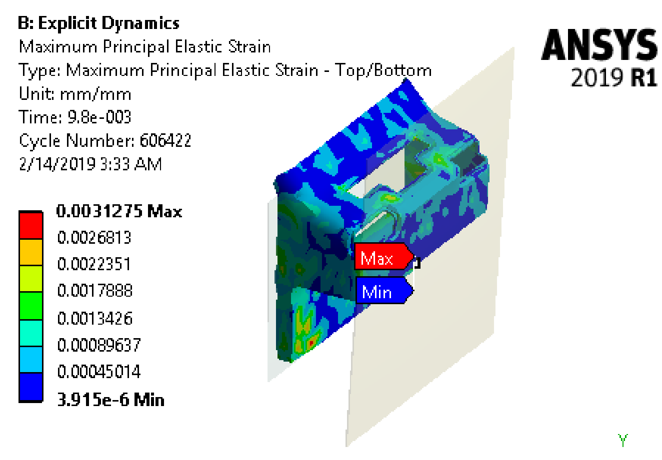
Total Deformation	**Maximum principal Elastic Strain**
	

Figure 64 shows the result for impact analysis of frontal panel model-1

4.1.3.1 Graphical representation result for energy developed and released during impact analysis of frontal panel model-1

The energy developed and released during impact simulation for frontal panel or model-1 is described in the graph below. It illustrate the way how Energy is converted from one types to other from and describe the conservation of energy.

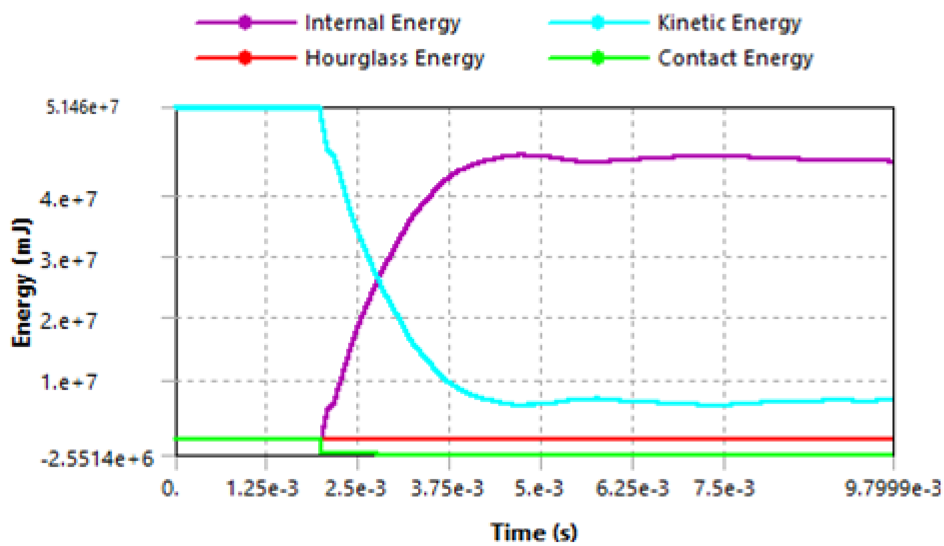


Figure 65 illustration of total kinetic energy developed during impact simulation

4.1.4 Result for impact analysis of frontal panel model-2

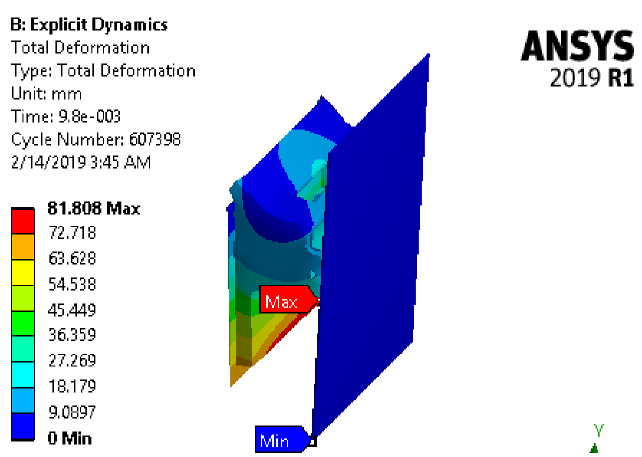
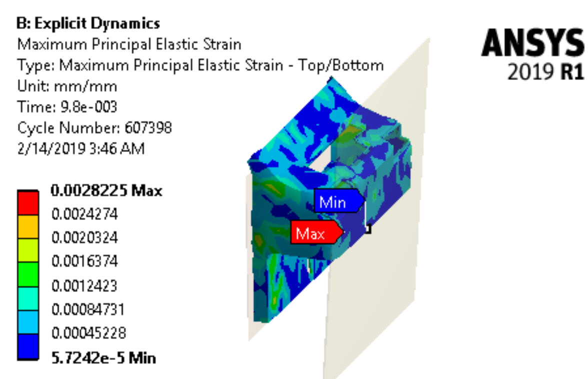
Total Deformation	Maximum principal Elastic Strain
	

Figure 66 shows the result for impact analysis of frontal panel model-2

4.1.4.1 Graphical representation result for energy developed and released during impact analysis of frontal panel model-2

The energy developed and released during impact analysis for frontal panel or model-2 is described in the graph below. It illustrate the way how Energy is converted from one types to other from.

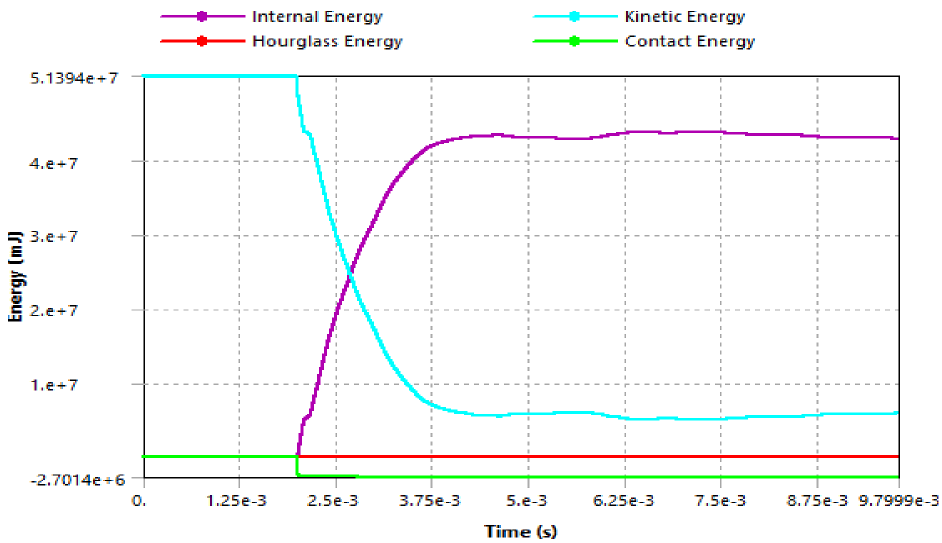


Figure 67 illustration of total kinetic energy developed during impact simulation

4.1.5 Result for impact analysis of frontal panel model-3

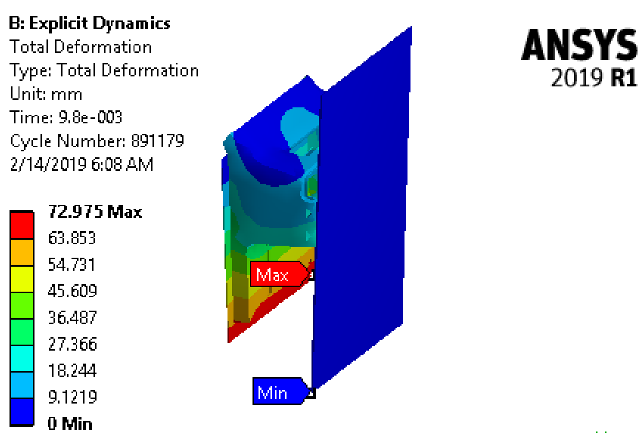
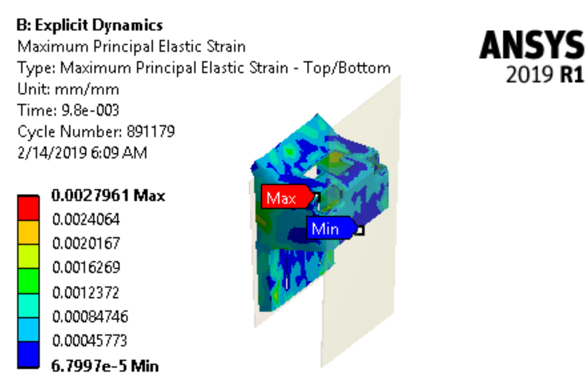
<table>
<tr><th>Total Deformation</th><th>Maximum principal Elastic Strain</th></tr>
<tr><td>
</td><td>
</td></tr>
</table>

Figure 68 shows the result for impact analysis of frontal panel model-3

4.1.5.1 Graphical representation result for energy developed and released during impact analysis of frontal panel model-3

The energy developed and released during impact analysis for frontal panel or model-3 is described in the graph below. It illustrate the way how Energy is converted from one types to other from.

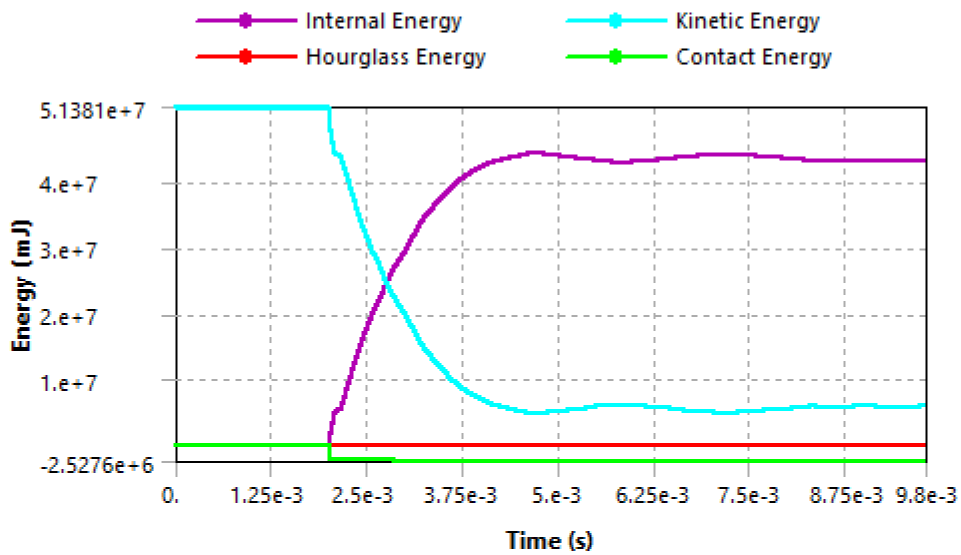


Figure 69 illustrate of total kinetic energy developed during impact simulation

4.2 Energy absorption indicators

In order to evaluate the crushing capability and energy absorption characteristics of energy absorbing structures ,itis necessary to define energy absorption's indicators[42].The energy absorption of three different model of frontal rail(panel) can be evaluate using above indicator as follows

a. Specific energy absorption (SEA)

SEA is defined as the ratio of the absorbed energy by a structure to its mass .SEA is a key criterion to evaluate energy absorption capacities of different structures [42].

$$SEA = \frac{EA}{Mt}$$ --Equation-67

Where, EA=energy absorption (KJ)

Mt=mass of vehicle (Kg)

- ✓ For Bajaj Qute frontal structure energy absorption is calculated using specific energy absorption (J/Kg) and total mass of a vehicle as an input.
- ✓ The specific energy absorption is found from ansys dynamic analysis of frontal panel for each step of cycles.

4.3 Graphical illustrations for three models of frontal panel impact characteristics

1. Displacement vs time graph

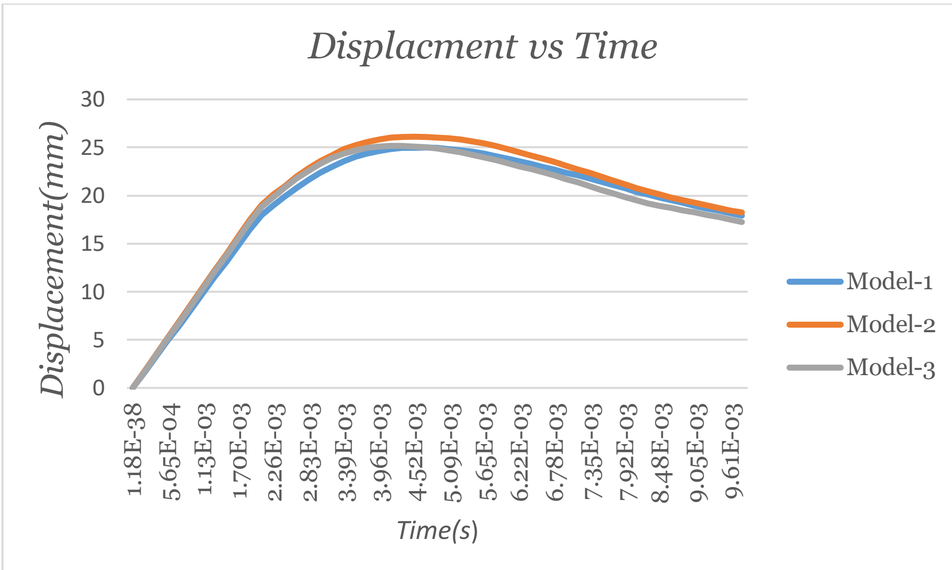


Figure 70 shows the displacement Vs time graph for each models of impact analysis

2. Acceleration vs time graph

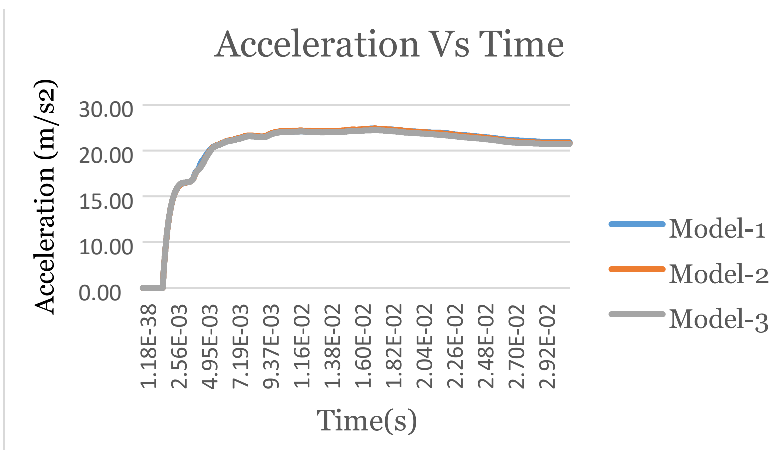


Figure 71 shows the acceleration vs time for each models of impact analysis

3. Force vs time graph

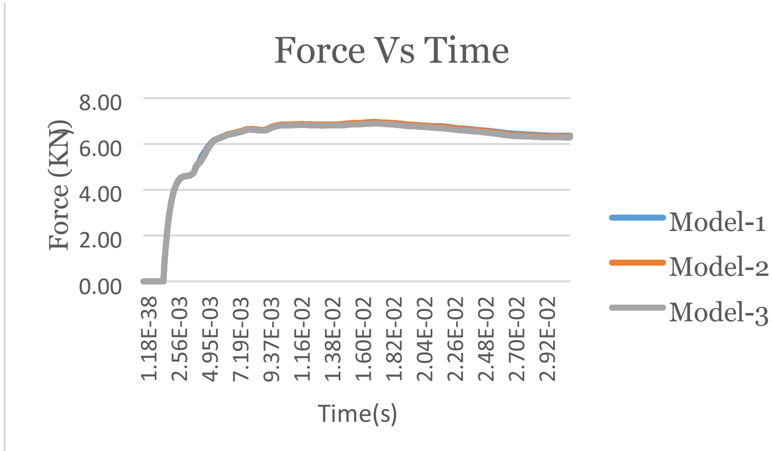


Figure 72 shows the Force vs time for each models of impact analysis

4. Energy developed in each model

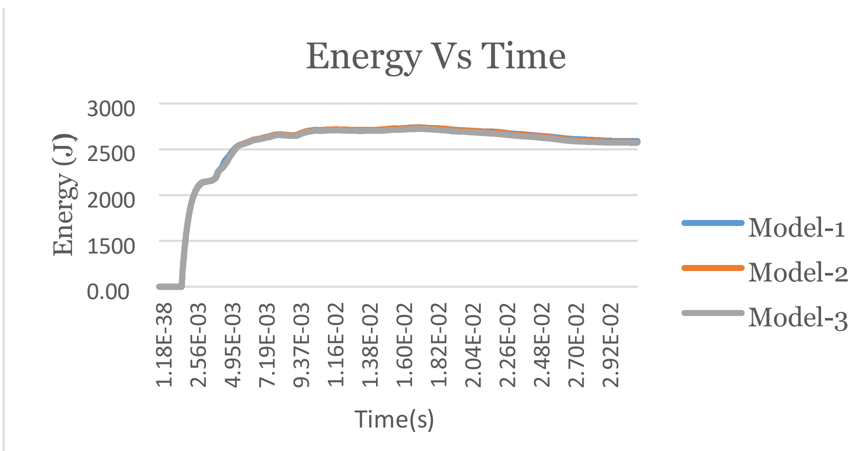


Figure 73 shows the Energy vs time for each models of impact analysis

5. Force Vs Displacement

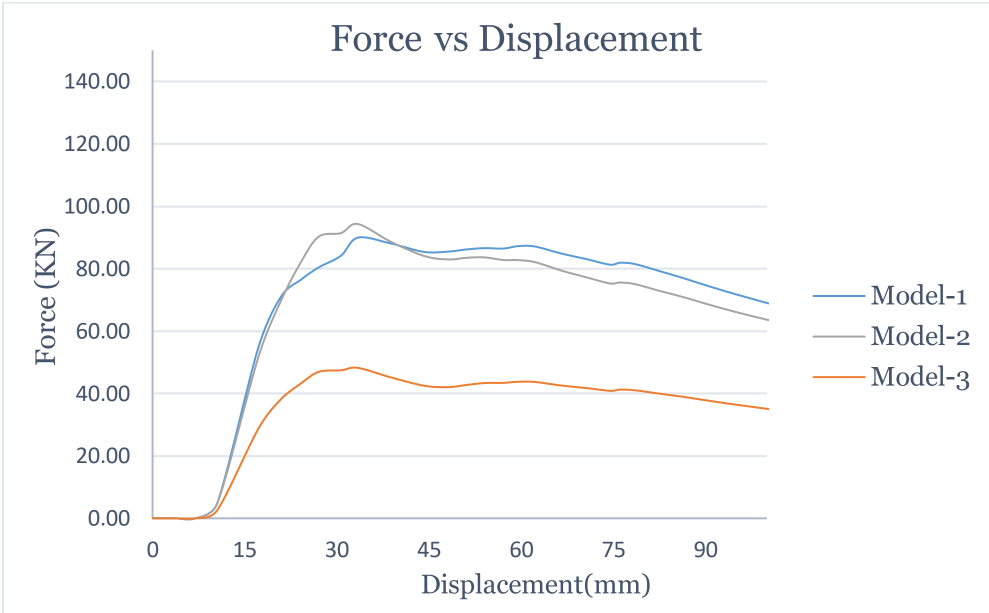


Figure 73 shows the Force Vs Displacement for each models of impact analysis

4.3 Discussion for Bajaj Qute structural analysis and impact analysis of frontal panel

Static structure analysis of Bajaj Qute employed using two models of vehicles that have different truck width one have 1312mm and other have 1512mm and from this the varieties of loads distribution for both models are computed. In model-1 of vehicle structure the von-miss stress for pure bending, pure torsion, horizontal Lozenging and combined loading are calculated and they are below the yield strength of the material so, the design is safe which means

Table 34 shows the comparison of von-miss stresses with that of material yield strength for model-1

	Loading types	**Von-miss stress**	**Yield strength of material**
Model-1	pure bending load	159Mpa	**550Mpa**
	Pure torsion load	223.02Mpa	
	horizontal Lozenging	243.6Mpa	
	Combined loading	288.60MPa	

Table 35 shows the comparison of von-miss stresses with that of material yield strength for model-2

	Loading types	Von-miss stress	Yield strength of material
Model-2	pure bending load	145.01Mpa	**550Mpa**
	Pure torsion load	289.23Mpa	
	horizontal Lozenging	218.03Mpa	
	Combined loading	274.78Mpa	

From the illustrative table it is seeable that the von-miss (equivalent stress) for all types of loading case are less than the material yield strength in two models so, the structural analysis of Bajaj Qute for static structure is safe.

Impact simulation and Graphical result of frontal panel of each models, it is clear that creating holes and increasing number of holes makes the model to be deform highly and increase the energy absorption behaviors of the model.

Table 36 Show deformation and internal energy developed for each model

	Deformation(mm)	Absorbed Energy (kJ)
Model-1	81.206	10.5
Model-2	81.808	14.83
Model-3	72.975	19.4

According to the given model result frontal panel three or model-3 have more energy absorption behavior compared to the rest two models of frontal panels and from table -36 it is visible that the deformation of model increases while the number of holes created in model increases and internal energy developed inside the model during impact increases with increment in number of hole created in each model.

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

1. 5.1 Conclusion

From the structural modeling, impact and stability analysis of Bajaj Qute vehicles different kinds of conclusions are brought out and from structure due to design change of model there is increment of torsional stiffness of Bajaj Quit structure with an incremental rate from 9303.2Nm/degree to 13958.5Nm/degree. There is also increment in rollover stability of vehicle due to increment in truck width of a vehicle using wheelbase vs truck width ratio as an input and the increment can be expressed in terms of both height of center of gravity and critical angle (angle of tilt) numerically, height above the ground of center of gravity was (h1=490mm) then increased to h=560mm and the angle of tilt of model-1 was ($\Theta 1=53.23^{o}$) then changes to ($\Theta 1=56.65^{o}$) and those increasing in critical angle(title angle) makes the vehicle to withstand rollover. It is also inquired that developing thin-walled tube (hollow) in the given geometry can increase the energy absorption capacity of model and comparatively model-3 have high energy absorption relative to model-2 and model-2 to have high energy absorption capability relative to model-1.

Recommendation

As a recommendation it is better considering the other types of loading in a vehicle structure and consider other boundary conditions should be conceive to have thoroughly results on the structural static analysis of Bajaj Qute and for impact analysis of Bajaj Qute structure side impact should be perform because side impact is one of the biggest accident that currently happens in any vehicles.

Future works

- ✓ Lateral stability of Bajaj Qute structure, to reduce overall instability of vehicle structure.
- ✓ Topology optimization of Bajaj Qute structure because the structure need to be reduce in terms of mass and other parameters.

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