Addis Ababa University

Addis Ababa Institute of Technology

Addis Ababa, Ethiopia

Strength analysis of three wheeled vehicle's chassis and body frame assembled in Ethiopia

By

Damtie Enawgew

A thesis submitted to

School of Mechanical and Industrial Engineering

Presented in partial fulfillment of the requirement for the Degree of Master of Science

Advisor
Dr.-Ing.Zewdu Abdi

October, 2013

Addis Ababa University

School of graduate studies

This is to certify that this thesis prepared by Damtie Enawgew, entitled: Strength analysis of three wheeled vehicles chassis and body frame assembled in Ethiopia and submitted in partial fulfillment of the requirements for the degree Master of Science compiles with regulations of the university and meets the accepted standards with respect to originality and quality.

Signed by the examining committee:

Internal Examiner________________________ signature______________ date______________

External Examiner________________________ signature______________ date______________

Adviser ________________________________signature_______________ date_____________

Chairman of the School__________________ signature _______________date_____________

Chair of department of graduate program coordinator

Dedication

Dedication to my beloved family and my wife for their immeasurable support, encouragement, and patience

Preface

The work reported in this thesis was performed at Addis Ababa University, School of Mechanical and Industrial Engineering as a part of the education to Masters of Science in Mechanical Design Stream. The foundation of this thesis came up as the thesis was to analyze the existing three wheeled vehicle model short coming and to remodel a new frame and structure that can alleviate existing short comings. The three wheeled vehicle frame and main structure is a key factor in this thesis.

Acknowledgement

First of all, I would like to thank God for his endless blessings and gifts!

I would like to thank my adviser Dr.-Ing. Zewdu Abdi for his scientific and technical guidance and support throughout this thesis, without his continuous corrections and constructive comments this thesis wouldn't have been successful. My advisor Dr. -Ing.zewdu abdi helped me through this thesis even starting from title selection.

.

I am grateful to all staff of AAU, School of Mechanical and Industrial Engineering for giving me the opportunity to create this thesis work and assigning advisor. Specially, I want to express my sincere appreciation and gratitude to Mr. Behailu for his valuable advice and for always having time for collecting data for me.

I would also like to thank my father Enawgew, and my mother Mulualem Assefa, for their endless comment, understanding, support and encouragement in my life.

Last but certainly not least, I would like to thank all of my friends for their encouragement, and support with different materials and ideas. They helped me through the data collection by finding manuals as well as guidance through difficult steps.

Abstract

Three-wheeled vehicles are becoming a popular and common means of short distance transport in Ethiopia. It is significant to know the current strength situation of three wheeled vehicle frames to be confident on their load carrying capacity, service and use. They are antiquated, unsafe and highly polluting, so there is substantial room for improvement in this huge market. A new three wheeled vehicle is designed, safer, more modern and easier to manufacture. This had achieved by modeling and analyzing the existing three wheeled vehicle model and by modeling a new three wheeled vehicle that alleviate the shortcomings in the existing model using different software like CATIA AND ANSYS. The main thing done was changing semi-monocoque type of the existing structure to a space frame. The result that we obtained was also such a better in strength and stiffness, easy to manufacture and cost effective. The purpose of this project is to document procedural details of such remodeling, with a view to presenting more research and development oriented investigations in the future. The contents of this report may also be of interesting to practicing engineers engaged in prototype development and field testing of three wheeled vehicles.

Table of contents

Content pages

List of tables

List of figures

Chapter One

Introduction

The automotive industry is facing tremendous engineering challenges over several years to meet the requirements generated by the fuel shortage, fuel cost, pollution control, raw material cost, and costumer safety. In meeting these challenges a wide variety of new models has been emerged or entered into the market in search for better strength and safety and fuel efficiency. As a result, three wheeled motor vehicles have been developed for the purpose of public and private transport, all over the world and in particular in developing countries. Three wheeled motor vehicles, typically used in India and most of the developing countries have their front steering with one wheel similar to those of motor cycles and motor scooters and the two rear wheels are driving wheels with a differential and suspension, which are similar to those of automobiles. Even though there are many modifications and improvements done still there are strength shortcomings [1].

1.1 Background

The three-wheeled vehicles (TWV) are popular means of transportation in Ethiopia because of their low priced have initial purchase, small sized and low fuel consumption. They generally characterized by a sheet-metal body or open frame resting and have one wheel in front and two wheels in their rear. One problem of these vehicles is the lowest strength of the frame and main structure.

The Driver is placed in a small cabin in the front of the vehicle and the passengers sit in the rear. These vehicles also can transport all kind of goods. The assignment of this thesis is to design and model a modern and better three-wheeled vehicle.

One of the most important requirements is that the vehicle must have an optimum strength and low price. In order to achieve this goal, it is necessary to use the optimal material and suitable frame type to ensure the required quality with the lowest cost and simple construction. This Master thesis is aimed to strengthen analysis of the existing three-wheeled vehicle and to remodel a new three-wheeled vehicle for the Ethiopia market.

In order to build the most energy efficient car, consideration of the types of the chassis selection must be made. Basically, the types of existing chassis design consist of backbone, space frames, monocoque, ladder frame, and semi monocoque. Each of chassis designs has their own strengths and weaknesses. Every chassis types are considered between weight, component size, complexity, vehicle intent, and ultimate cost. Even within a basic design method, strength and stiffness can vary significantly, depending on the chassis type. An ideal chassis is the one that has high stiffness with low weight and cost [4].

The chassis has to contain the various components required for the three wheeled vehicle as well as being based around a driver's cockpit. The safety of the chassis is a major aspect in the design, and should be considered through all stages. The design also has to meet strict requirements and regulations set [5].

Three wheeled vehicle is selected for this project to be designed, which is a popular passenger vehicle for middle-income families and city low fee taxi. Three-wheeled taxis form an essential part of public transport for the urban middle class population of Ethiopia to move short distance. Apart from Ethiopia, they are also being used in the world, for public transport and for carrying freight. Obviously, these low cost vehicles will continue as a major mode of travel in our country and also the whole Africa and Asia. Even now a day, their frames are being assembled in our country, Ethiopia. Also their market demand is increasing fast. So as to decrease unnecessary wastage of money as well as to save passengers life and property from risk of accident on taxi, it is essential to analyze strength of the frames and remodel. This is done by investigating the current three wheeled vehicle structure assembled in Ethiopia and come up with improved design to alleviate the shortcomings. Surely, the result of the analysis and the implementation of the design will contribute to the improvement of the locally built tri-wheeler. In conventional practice the three-wheeled vehicle body is produced from overlapping sheet metal, fastened by a multiplicity of spot welds often robotically applied. And their construction is highly complex and impossible for technologically less developed countries. . Therefore, the first task in this thesis will be modeling and analysis of the existing three- wheeled vehicle frame. After that the result will be interpreted and depending on the result the new model will be created in order to avoid the shortcomings of the existing three wheeled vehicle. To do so, I have used different software's like that of CATIA, for 3D modeling and ANSYS, for FEM analysis.

1.2. Problem statement

There is a frequent accident occurring on three wheeled taxis and community dishonest using these vehicles due to their strength uncertainty and instability. Hence, I decided to do strength analysis of three wheeled vehicle's frame and body structure assembled in Ethiopia and recommend on shortcomings of current design and remodel a new three wheeled vehicle which can alleviate the society from fear and accident. These shortcomings include over strength of structure, which leads to overweight, high fuel consumption and speed reduction and also under strength of the current design, may lead to accidents and losses. But the most critical fear of the population is the level of risk that occurs if the vehicle crashes accidentally. Because there is no protective cover for the passenger the risk level is thinking high in the society. This thesis will try to identify the types of shortcomings really exist in the three wheeled taxis we see running on the roads and solve these problem in a right way.

1.3. Objective of the Study

Main objective of this thesis is to investigate the modeling and strength analysis of frame and main structure of three wheeled vehicles assembled in Ethiopia.

Specifically we are going to:

- Study the existing frame and structure of three wheeled vehicle.
- Study load caring and load distribution capacity.
- Model a frame and structure using FEM analysis of load casing.
- Evaluate the existing frame and structure and to recommend structure change taking for improvement.
- Prepare and propose a new frame and structure of three wheeled vehicle.

- Design the new frame of three wheeled vehicle which able to withstand the load applied on the chassis structure with maximum strength, simple in construction, low cost and better reliability.

1.4. Project methodology

In order to complete this project, there were several steps which had been followed through.

First Data collection has been done from manuals of local assembling company and by measuring of the real three wheeled vehicle frame and structure components. Then CAD modeling of the structure had made by using the data recorded from the existing vehicle body. CATIA software was used to build the 3Dcomputer model. Also Finite element analysis using ANSYS and analytical solution has been performed for analyzing of the existing frame and structure for the required results i.e. static analysis and crash simulation. After that, Re-modeling of the frame and structure using new size of components and new frame type has been done. Next to that Finite element analysis of the new modeled structure was performed. After these works Comparison of the original and improved structure In terms of strength parameters of the original and improved structure was performed. Finally, Cost Analysis has been carried out. Recommendation was set on the existing frames or on of relevant improvements of the design of new three wheeled vehicle frame. The project methodology in the form of a flow chart is graphically shown bellow in figure 1.

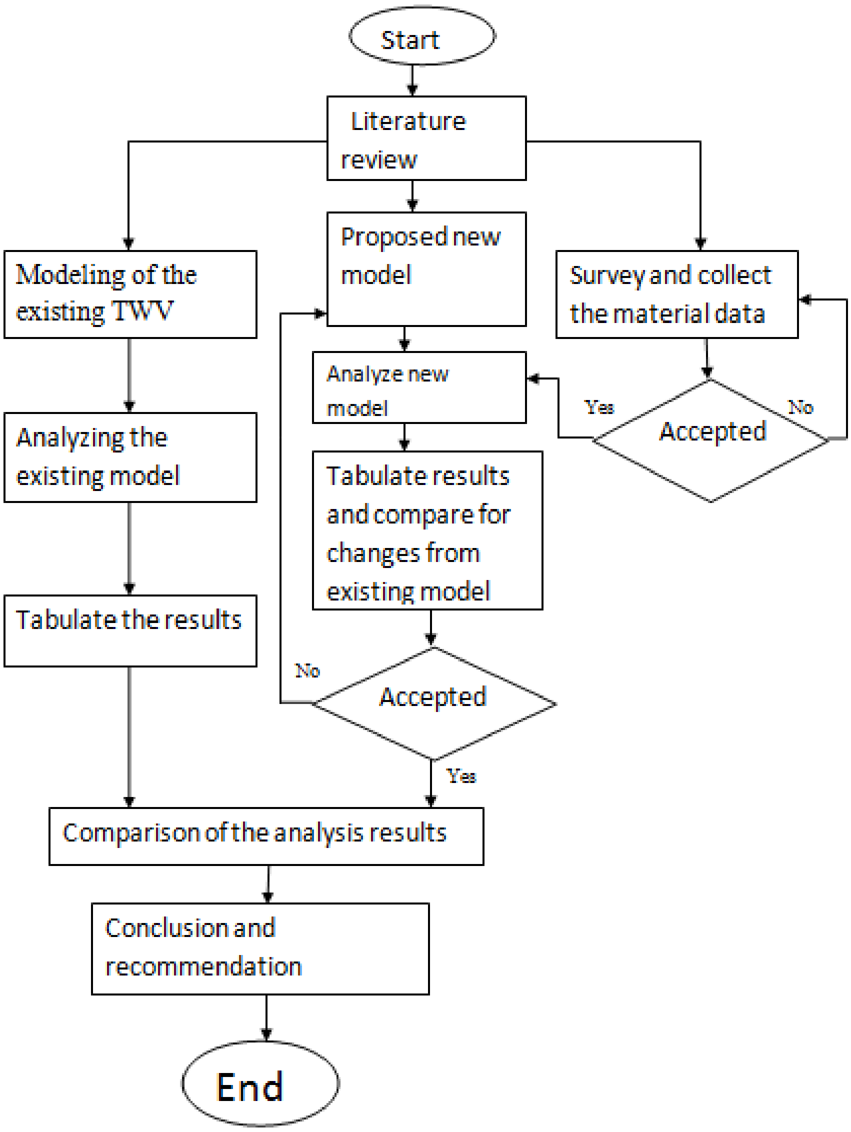


Figure 1: project flow chart

Materials required

Materials used to accomplish this research are software like, CATIA and ANSYS ; measuring instruments like, meter and calipers; Flash; digital Camera; stationary materials; books and journals; Computer and other complimentary services.

Chapter Two

Literature review

2.1. Introduction

In conventional practice the passenger car body is produced from overlapping sheet metal, fastened by a multiplicity of spot welds often robotically applied. The body frame work is constructed by joining together thin-welded closed-section members, and where these are subjected to particularly large torsion and bending stresses, such as in the case of body sills, they may be reinforced by the inclusion of longitudinal partitions.

2.2. Theory

Types of three wheeled passenger vehicles

Some common three wheeled vehicles on the market and their suspension solutions are discussed below.

Piaggio Ape

The first "auto rickshaw" was produced in 1948 by the Italian company Piaggio & C. SpA. The model was named Ape and they are produced in three different types; van, pick-up and rickshaw. It has got a semi-monocoque frame and a steel body in the front that fits either one or two peoples, while the rear is built as a single load-bearing chassis in sheet metal.
The rear suspension consists of trailing arms, spring coils and shock absorbers and in the front there is a leading link suspension [5].

Bajaj

Bajaj Auto is the fourth largest manufacturer of two- and three-wheelers in the world and the largest three wheeled vehicle producer in India. The Bajaj auto rickshaws are very common on the Ethiopian market. One common version is the Bajaj RE [5].

The first rickshaw built by Bajaj was licensed under Piaggio/Vespa and appeared on the market in 1950. They used same mechanics as the Piaggio Ape but changed its appearance and even introduced a taxi version which Piaggio Ape did not have [5]. This means that they still used the trailing arms together with shock absorbers and springs as rear suspension and leading link as front suspension. It was taken as my concern in this project.

Monika Motors

Second largest company that specializes on auto rickshaws in Thailand, it's a part of the B.Grimm concern and was founded in 1883. First auto rickshaw came into production in 1960 and was named Monika L5. The L5 uses the same suspension combination as Tuk Tuk Forwarders auto rickshaws, i.e. a dual telescopic fork and a rigid rear axle with leaf springs.
In other hand you can say that trailing arms are common as a rear suspension and trailing and leading link are common in the front. Those equipped with leading link has the advantage of anti-dive [5].

Tuk Tuk Forwarder Co., Ltd

This company is only aimed on rickshaws and was founded in 1993. They did not start their production until 1996 where they had a yearly capacity of 12.000 units. In 1994 the first two electrical Tuk Tuks were produced. Suspension wise this company goes in another direction than the rest as they use leaf springs and double shock absorbers together with a rigid axle for the rear suspension. In the front they use a telescopic fork with coil springs and dual shock absorbers [5].

Mahindra & Mahindra

Mahindra & Mahindra is a company that doesn´t just aim for the commercialized vehicle but also to personal vehicle such as SUV, tractors, two-wheelers etc. The company was started in 1945 and is an underlying company of Mahindra group. The first three-wheeler to reach the market was the Alfa which was introduced in 2008. The Alfa is equipped with leading link, coil spring and hydraulic shock absorber as the front suspension. The rear suspension consists of independent trailing arms together with hydraulic shock absorbers and rubber springs.

TVS King

The first auto rickshaw developed by TVS Motor was named TVS King and was introduced in 2008; it was the first auto rickshaw model in India with a four stroke 200 cc engine. The TVS King uses trailing link as front suspension and individual trailing arms as rear suspension and it has got constant rate coil springs with co-axial hydraulic dampers [5].

2.3 Type of structure

Introduction of 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 back bone of the vehicle. A vehicle without 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].

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 the vehicle development. There are different choices, monocoque, semimonocoque, body on frame. Other kinds of structures are suerleggera or backbone and ladder chassis. Each one has advantages and disadvantages.

Monocoque or integral structure

Monooque 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.

Monocoque structure uses more metal material than any other type of structure, consequently, it is heavier. This structure is the best configuration to protect the passengers in 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. Because it is geometrically very complicated, Sheet steel pressings and spot welds used to form an integral structure. Side frames + depth + roof give good bending and torsion stiffness. Stress distribution is function of applied loads and relative stiffness between components. Stress distribution can be analyzed by FEM [4]. Its advantages are stiffer in bending and torsion; lower weight; less cost and quiet operation.

Semi monocoque

Semi monocoque structure is quite similar to monocoque structure. The only difference is that semimonocoque has longerons and stingers to make up a strength base. The other components are attached to this base. Sometimes, it is difficult to differentiate it from monocoque structure. Nowadays, auto-rickshaws have a semi-monocoque chassis. It is a very simple structure that consists on a combination of longerons and stingers supporting the weight of the metal sheets, and the round tubes where the roof is attached. The chassis of the current auto-rickshaws is not monocoque or independent chassis [4].

Space frames or Body on frame

Body on 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: the frame and the bodywork.

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. This element has a high strength and the mechanical components and the bodywork are attached on 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. This system also protects the bodywork from bending or torsion efforts. The body on frame system has a great facility to use different bodies in the same frame. Reforms in the body can also be performed easily. 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.

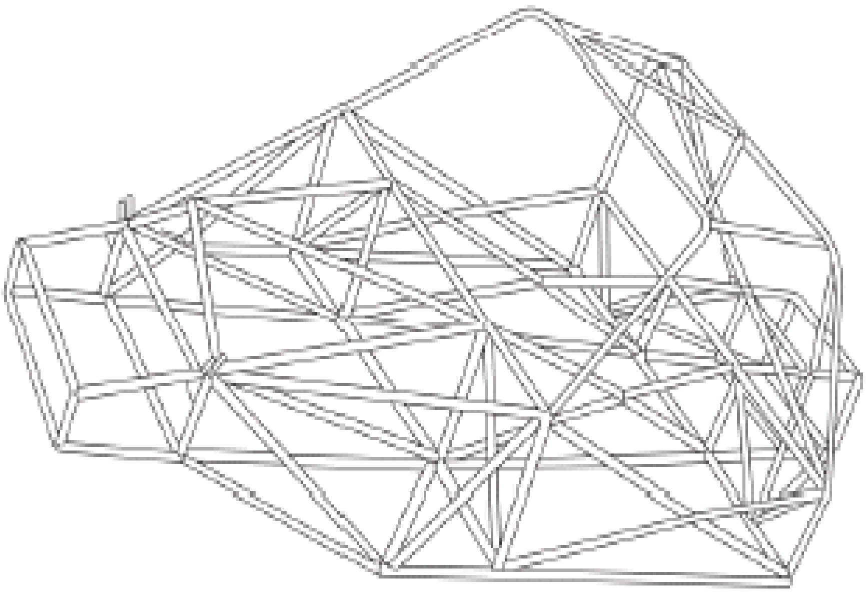


Figure 2: space frame chassis

Backbone

A backbone chassis is the simplest structure design. It consists of a sturdy tubular backbone that joints the front and rear axle. These chassis is fully enclosed to be rigid structure and handle all loads. It should be noted that the backbone chassis can be built through many types of construction. The space within the structure is used to place the driveshaft in case of front-engine, rear-wheel drive layout. Further, the drive train, engine and suspensions are all connected to each of the ends of the chassis. The body is built on the backbone usually made of glass-fiber.
Almost rear wheel drive and front engine vehicles use backbone chassis [4]. Main back bone is a closed box section. Splayed beams at front and rear extent to suspension mounting points. Transverse beams resist lateral loads; Back bone frame: bending and torsion beam; Splayed beams: bending; Transverse beams: tension or compression.

Ladder frame

A ladder frame is the 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/cross braces, typically made from round or rectangular tubing or channel. 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 use ladder chassis.

Generally, Functions of the chassis frame are:

1. To carry load of the passengers or goods carried in the body.
2. To support the load of the body, engine, gear box etc.
3. To withstand the forces caused due to the sudden braking or acceleration
4. To withstand the stresses caused due to the bad road condition.
5. To withstand centrifugal force while cornering

2.4 frame member cross sections

So far the commercial vehicle ladder frame has been identified as comprising basically two members joined by a series of cross-members. It is now necessary to examine this structure in a little more detail, especially with respect to the material selections used, the different methods of joining the frame members, and the means by which they may be reinforced for greater rigidity, as follows [7].

Channel section steel pressing of constant web height are usually employed for the chassis frame side-members. Although the use of channel section provides adequate resistance to vertical bending loads, it is much less efficient in resisting side-ways bending and torsion (twisting) loads. However it does lend itself to the economic production of commercial vehicle frames for the numbers required, and also provides for easy attachment of cross-members and the various other mounting brackets.

The cross members must resist torsion deflection of the frame as a whole imposed by rocking movements of the sprung axles, and also prevent individual twisting of the side members arising from overhung loads such as the fuel tank. Various material sections are used for the cross-members, including I-section, top –hat section, channel section and tubular-section beams. I section cross members are often used towards the front of the frame, because they can better resist bending loads in horizontal and vertical directions, thereby maintaining front end alignment and supporting the power unit. Top –hat- sections cross-sections may be used towards the middle of the frame, since they more readily lend themselves to special profiles. These can be required to provide clearance for a component, such as an intermediate bearing for the propeller shaft. Tubular- section cross- members are normally found towards the extreme rear end of the frame, where they offer good resistance against torsion loads.

Joining methods of the frame members

The processes that may be employed to join the side and cross- members are welding, riveting, and bolting. 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. Riveting is widely used for joining the frame members, and it may be done cold rather than hot so as to avoid the rivets contracting during cooling and developing clearances in their holes. Bolting is also popular and can be more effective than riveting as regards rigidity because the tightness with which the frame members are clamped together can be accurately controlled. A possible disadvantage is that screwed connections may loosen if not properly installed. A combination of riveting and bolting is sometimes employed; for example, a bolted connection may be confined to the front cross- members so that it can be readily removed to facilitate engine replacement [7].

Beams are the basic parts in case of collision. They have to protect the passengers around any direction of the space. There were also some different options with respect to beams. First option was made the beams external parts. That means they should be painted and prepared as the external sheets, so it makes them more expensive. One advantage is that the sheets are directly joined to the beams; consequently, there are not joins between two sheets. However, external beams have more complexity shapes. In the other hand, the internal beams have easier shapes and probably higher stiffness, so that is the solution taken.

2.5 Materials

One of the most important decisions is to choose the type of material to be used. Depending on the manufacturing process, the material used will vary. We can distinguish basically two groups of parts with different requirement in the vehicle structure. The structural beams, whose goal is to get a structure as strong and robust as possible, and the sheets, whose aim is, apart from providing resistance to the whole, enclose the interior volume to protect it from the outside.

2.5.1 Structural beams materials

Materials used for beams should be strength, so metals are the most adequate material. Common materials of structural beams are [7]:
Conventional steel: It is a ductile material that can be manufactured with complex forms and it present good welding properties. It is very economical. Nevertheless, it is necessary to use greater thicknesses than other materials, which leads to a weight increasing. HSS (High speed steel): It has higher strength than conventional steel, so it is possible to reduce the weight and increase the rigidity of the structure. They are lightly more expensive and less ductile. Boron alloyed steel: This kind of steel might be formed by hot-stamping. This method allows good ductility during the process and high final rigidity. It is more expensive than conventional steel. Aluminum: It is lighter than steel and gets more strength with less weight. It presents better corrosion resistance. However is more expensive than steel and it is more difficult to weld. Many people think that the chassis which built from aluminum is the path to the lightest design, but this is not necessarily true. Aluminum is more flexible than steel. In fact, the ratio of stiffness to weight is almost identical to steel, so an aluminum chassis must weigh the same as a steel one to achieve the same stiffness. Aluminum has an advantage only when there are in very thin sheet sections where buckling is possible but that are not generally the case with tubing. The uses of aluminum and FRP in designing the chassis are helped to reduce overall vehicle weight, thereby reducing fuel consumption as well.

2.5.2 Exterior sheets materials

Materials used for external sheets have less structural requirements. Plastic material; It is a cheap material and easy to manufacture, but cannot be welded to the structure and do not provide any strength to the assembly. Fiberglas; It is a light material, but harder than plastic. It cannot be welded to the structure and it is more expensive than plastic or sheet metal. Sheet metal; It is a cheap material, provides strength and can be welded the assembly. However, it is heavier than plastic.

2.6 Vehicle axis system

The vehicle moves around an axis system, with x, y and z-axles, with its origin in a line perpendicular to the road through the CG (centre of gravity) of the whole vehicle and at the intersection of the vehicle roll axis. A body in space is determined by six degrees of freedom (DOF), one rotation and one translation about each axis. Since the vehicle chassis can be modeled as a rigid body its rotations can be described [5]. Roll angle: it is the rotation around the x-axis; the x-axis is pointed in the longitudinal direction of the vehicle, in other words the driving direction of the vehicle. When the vehicle rolls the body leans as when cornering. The vehicles positive longitudinal direction is defined along the x-axis according to ISO 8855. Pitch angle: it is the rotation around the y-axis; the y-axis is transverse to the vehicle. When the vehicle brakes or accelerates there is a pitch change. Yaw angle: it is the rotation around the z-axis; the z-axis is perpendicular to the road pointing down through the vehicle in the SAE vehicle axis system.

2.7 Software

CATIA (computer aided three dimensional interactive application) is used to model the three wheeled vehicle 3D drawing. It is used to do both part drawing and assembly drawing.

The finite element method (FEM) is used for seeking an approximated solution of the distribution of field variables in the problem domain that is difficult to obtain analytically. Known physical laws are then applied to each small element, each of which usually has a very simple geometry. A continuous function of an unknown field variable is approximated using piecewise linear functions in each sub-domain, called an element formed by nodes. The unknowns are then the discrete values of the field variable at the nodes.

2.8 modeling

Traditionally, formal modeling of systems has been via 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. Many research works had been done on tricycles, computer aided design of various engineering structures and assembly. And the unique specialty of the work is the integration of computer aided design, modeling and simulation through the use of relevant computer programs to aid the assembly procedure of Tricycle. This will help the accuracy, save time and make it flexible thereby improving quality and efficiency of the assembly procedure [8].

In the study of "Stability of Three-Wheeled Vehicles with and without Control System" by M.A.Saeedi R.Kazemi, 2013, stability control of a three-wheeled vehicle with two wheels on the front axle, a three-wheeled vehicle with two wheels on the rear axle, and a standard four-wheeled vehicle are compared. For vehicle dynamics control systems, the direct yaw moment control is considered as a suitable way of controlling the lateral motion of a vehicle during a severe driving maneuver. In accordance to the present available technology, the performance of vehicle dynamics control actuation systems is based on the individual control of each wheel braking force known as the differential braking. Also, in order to design the vehicle dynamics control system the linear optimal control theory is used. Then, to investigate the effectiveness of the proposed linear optimal control system, computer simulations are carried out by using nonlinear twelve-degree-of-freedom models for three-wheeled cars and fourteen-degree-of-freedom model for a four-wheeled car [9].

"Development of an auto rickshaw vehicle suspension" by Thomas Gyllendah and David, 2012 was to develop a vehicle suspension intended for an auto rickshaw. A variety of different suspension types were investigated and evaluated until two suspension types were chosen; one type for the front and one type for the rear. These suspension types were then simulated and tested in Siemens NX in different critical scenarios to gain useful information. The information was then evaluated to draw a conclusion if the developed suspension obtained good performance. The final solution was simulated and partly verified and still work remains to get a full overview of the performance. And this study has printed in journal [5].

"Dynamic Behavior of Three-wheeled Vehicle under Random Road Characteristics Using FEM" was done by R. Koona1, M.K. Naidu and B. Satyanarayana, 2004. In this work the analysis is performed using power spectral density analysis. A PSD spectrum is a statistical measure of the response of a structure to random dynamic loading conditions. It is a graph of power spectral density versus frequency, where the PSD may be a displacement PSD, velocity PSD, acceleration PSD or force PSD. Similar to response spectrum analysis random vibration analysis may be single point or multi point. In a single point random vibration analysis, only one PSD spectrum is specified at different points in the model. In the present paper single point response analysis has been used [1].

Chapter three

Modeling and analysis of the existing three wheeled motor vehicle

3.1 Introduction

Here, the chassis and body are built as two separate units. The body is then assembled in to the chassis with welds to hold the body to the rigid chassis. But all these tasks may depend on the type of model we are going to analyze. For this project a three-wheeled vehicle having the specifications shown in table 1 bellow is considered.

3.2 Three wheeled vehicle Technical specification

The data and specification of three wheeled vehicle has been collected from TAGRO BAJAJ P.L.C Company and from field measurement of real three wheeled vehicle. Even from field measurements, basic dimensions that are approximately equal to the specification data were obtained.

Engine and transmission
Name----------------------------------Bajaj RE 4-stroke
Type-----------------------------------four stroke, spark ignition, forced air cooled
No of cylinders----------------------one
Bore-----------------------------------57 mm
Stroke---------------------------------68 mm
Engine displacement----------------173.52 cc
Maximum net power----------------8 Hp at 5000 rpm
Maximum net torque----------------1.3 kg.m at 4000 rpm
Compression pressure---------------12+ 0.5 kg/cm2
Compression ratio--------------------9:1
Idling speed-------------------------1200+-150 rpm
Ignition system----------------------electronic, d.c. type

Ignition timing----------------------3^0 BTDC upto1500 rpm and 17^0 BTDC up to 2500 rpm onward
Fuel-----------------------------------87 octane petrol
Carburetor---------------------------side draft 18mm equivalent venture
Spark plug---------------------------MICO WR7BC4
Spark plug gap----------------------0.6 up to 0.7mm
Lubrication--------------------------positive lubrication ,wet sump
Starting------------------------------electric starter motor and hand lever
Clutch--------------------------------wet, multi disk type
Transmission------------------------4-speed forward+ reverse
Differential--------------------------integral with engine
Differential gear ratio--------------forward 2.192:1 ,reverse 1.925:1
Overall gear ratios
1st gear--------------------------------32.558:1
2nd gear--------------------------------19.065:1
3rd gear---------------------------------11.862:1
4th gear---------------------------------8.25:1
Reverse gear--------------------------28.59:1
2. chassis and body
Frame type -----------------------constructed from pressed steel sheets and sections welded together (semi-monocoque)
Suspension
front-------------------------------anti drive type, constant rate coil spring and double acting hydraulic shock absorber
rear--------------------------------independently sprung wheels by trailing arms with helical springs and hydraulic damper
brakes
front and rear----------------------hydraulic expanding shoe type size 170mm * 29.5 wide
parking------------------------------------mechanically operated on rear brakes
wheel rims(front and rear)------------- 8in×3in

wheel outer diameter-------------------- 400mm
Tire (front and rear) --------------------- 4-8,4P.R.
TYRE PRESSURE
Front-- 2.1kg/cm2(30 psi)
Rear--- 2.4 kg/cm2(34 psi)
Fuel tank capacity------------------------- 8 liter including 1 liter reserve
3.electricals
System-------------------------------------- 12 volt d.c. negative ground
Battery-------------------------------------- 12v ,32 ah
Head lamp---------------------------------- 35/35 w
Tail/stop lamp------------------------------ 5/21 w
Side indicator lamp------------------------ 10 w
Pilot lamp------------------------------------ 5w
Reverse lamp-------------------------------- 10w
Battery charging, Neutral, head light hi beam, turn pilot
Indicator-------------------------------------- 2w
Horn-- 12v d.c
Wiper system---------------------------------electrical wiper motor
4.DIMENSIONS
Length-------------------------------------- 2625mm
Width-------------------------------------- 1300mm
Height-------------------------------------- 1710mm
Wheel base-------------------------------- 2000mm
Wheel track------------------------------- 1150mm
Ground clearance----------------------- 180mm this is the distance between the ground and the underside of the vehicle chassis
Turning circle radius--------------------- 5760
5.WEIGHTS
Kerb weight------------------------------ 330 kg (the weight of motor car without occupant or baggage)

Gross vehicle weight---------------------- 640kg
Maximum permissible weights (the total weight of the vehicle and load, including the weight of fuel, and the driver and passenger if carried) for goods vehicles and trailers depend on their wheelbase, the number of axles, the outer axle spread (the distance between the center of the wheels on the front and rearmost axles) and the relevant axle spacing in the case of articulated vehicles.
Maximum play-load including driver----------- 310kg
Maximum front axle weight--------------------- 210kg
Maximum rear axle weight---------------------- 430kg
6.PERFORMANCE
Maximum speed--------------------------------- 60km/h
Climbing ability-------------------------------- 18% maximum

Table 1 : specification data of the existing three wheeled vehicle

3.3 CATIA Modeling of existing three wheeled vehicle

The model consists of main chassis elements and body. The mass of the components like engine, gear box, propeller shaft etc. is considered as 3-D mass elements. Any problem becomes complex, if the real situations are considered, it becomes very difficult to analyze the problem with such complexities. In order to simplify the problem, some assumptions have to be made. In the present analysis the following assumptions are considered.

1. The curvature of the chassis members where cross section changes takes palace is neglected
2. The tires are considered as linear springs with stiffness in vertical direction alone
3. The mass of the engine gear box and other components are lumped at appropriate nodes
4. Some small members which do not have any significant load carrying function are neglected.

The CATIA V5R16 is used for model of the existing three wheeled vehicle. It is made by using chassis cross-section of 90×50×3mm steel beam rectangular hallow section(RHS) at rear part and top hat cross section at front with steel sheet metal of 8mm thickness spot welded with it. Chassis is a semi- monocoque type (figure 3).

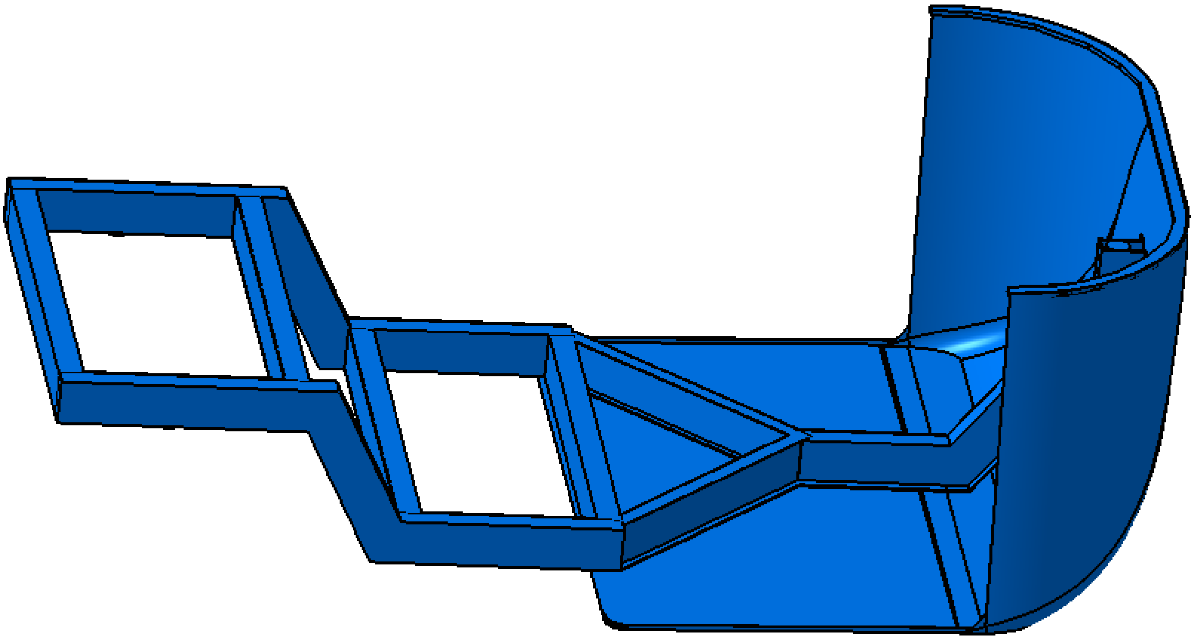


Figure 3: semi monocoque chassis of the existing model

Generally the model has been done using the overall dimensions of 2625mm Length, 1300mm Width and1710mm Height. Bars with circular cross sections and diameters of 25mm and 1.5mm thickness have been modeled in CATIA for roofing purpose. Finally each part assembled together before going to finite element analysis (figure 4).

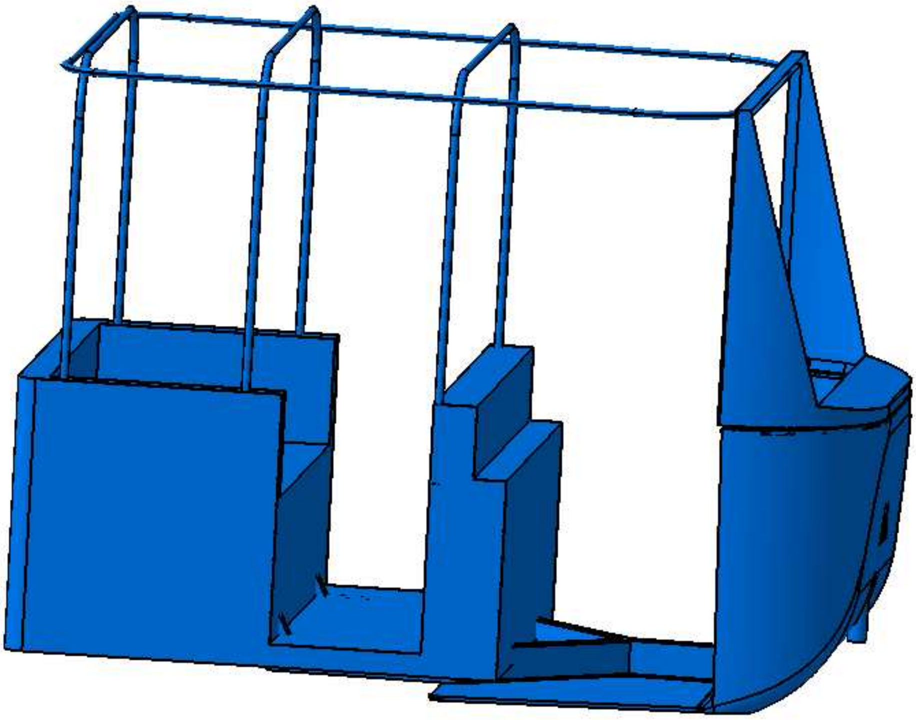


Figure 4: 3D model of the existing three wheeled vehicle

3.4. Finite element analysis of the existing TWV structure

The reason for using a FEA software program is to analyze and compare the performance varying design iteration, with the process of transferring the model from the CAD program CATIA to the FEA program ANSYS. Next, proper principles are followed to establish equations for the elements, after which the elements are tied to one another. This process leads to a set of linear algebraic simultaneous equations for the entire system that can be solved easily to yield the required field variable.

The FEA of the model, once imported into ANSYS is a simple process, consisting of a number of simple steps;

1. selecting the element type(s) to use,
2. Inputting material properties,
3. Entering real constants, thickness values or section properties,
4. Meshing elements

5. Applying loads and boundary conditions,
6. Running Analysis and viewing results.

Materials

Material selection of the chassis plays crucial part in providing the desired strength, endurance, safety and reliability to the vehicle. To choose the optimal material we need an extensive study on the properties of different materials. The material for the existing three wheeled vehicle model is the structural steel ASTM 106a grade B with the following parameters shown in table 2 bellow.

No	material	Yield strength	Ultimate tensile strength	Young's modulus(E)	Polson 's ratio(v)	Elongation
1	Structural steel	300MPa	441MPa	200000MPa	0.3	20%

Table 2: material for the existing three wheeled vehicle model

3.4.1 Static analysis

A static strength analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those caused by time-varying loads. This analysis determines the displacements, stresses, strains, and forces in structures or components caused by loads that do not induce significant inertia and damping effects. For carrying out the FEM Analysis of chassis as per standard procedure first it requires to create merge part for assembly to achieve the connectivity and loading and constraining is required to be applied. Simplified CAD model of existing three wheeled vehicle frame is created using CATIA and it is imported in ANSYS as an external geometry file.

Meshing

An important process of any FEA is to refine the mesh that is used in the analysis, a process that involves reducing mesh or element size a number of times and plotting the results as to ensure the analysis is stable. The mesh influences the accuracy, convergence and speed of the solution. Furthermore, the time it takes to create a mesh model is often a significant portion of the time it takes to get results from an ANSYS solution. Tetrahedral mesh elements are used in meshing of the three wheeled vehicle structure. A tetrahedron element is 3-D solid element which has four triangle faces and six triangle cross-sections (figure 5).

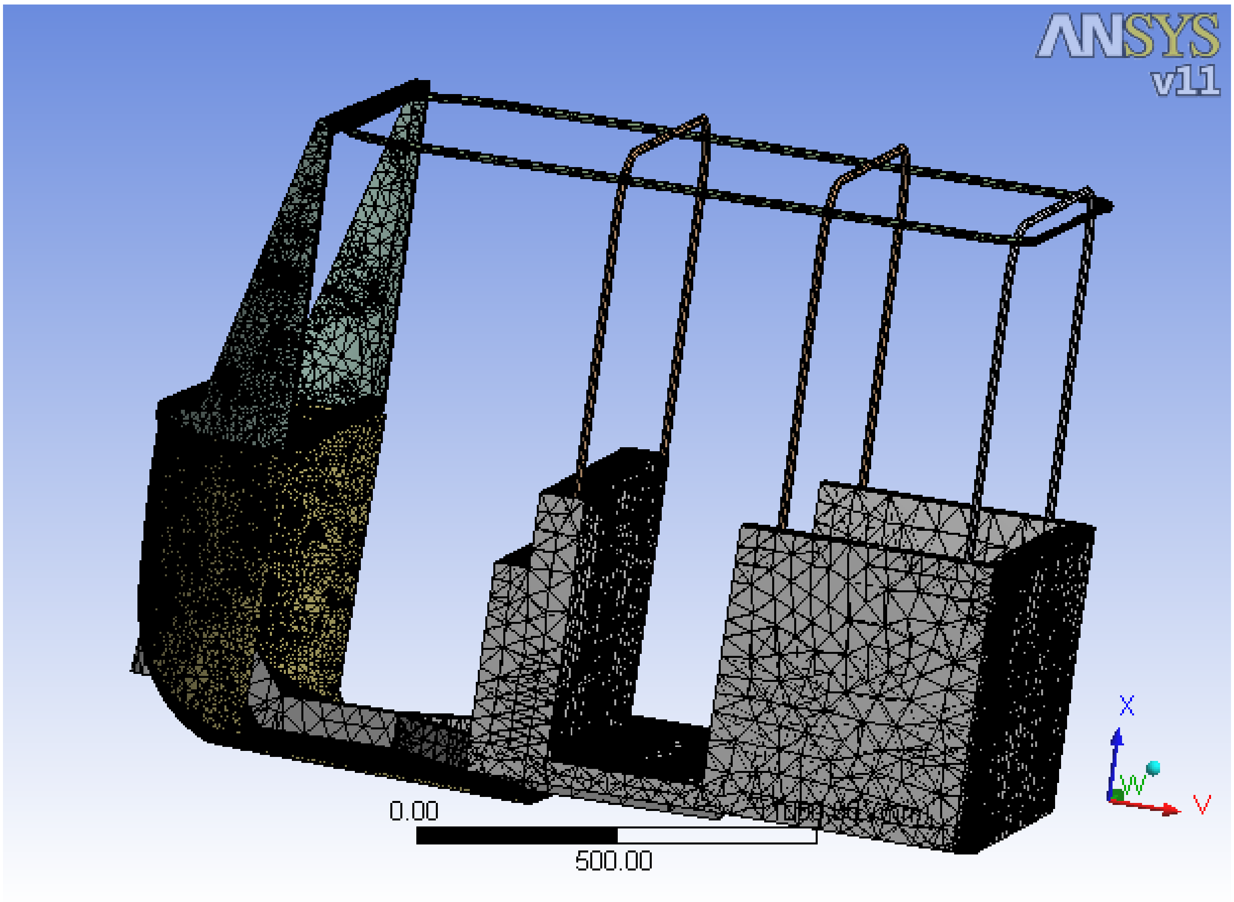


Figure 5: FEM mesh of the existing model

Element and Nodes

The meshed TWV chassis model has number of elements and number of nodes. The element is tetrahedral. In order to get a better result, locally finer meshing applied in the region which is suspected to have the highest stress. Finally, 240395 numbers of Nodes and 124768 numbers of Elements has been found.

Loading and Boundary condition

Total weight, the weight of three wheeled motor car having all components without occupant or baggage is, 330 kg or 3237.3N. This will be applied at the center of gravity of the frame. And Maximum play-load including driver is 320kg applied at seat points of each passenger and driver. For each will have 80kg (784.8N) weights. The TWV chassis model is loaded by static forces from its own body and load from passengers, driver, and other baggage weights. For this model, the maximum loaded Gross vehicle weight is 640 kg. The load is assumed as a uniform distributed obtained from the maximum loaded weight divided over the total length of chassis frame. Maximum front axle weight is 210kg (2060.1N), which is the part of the Gross vehicle weight that loaded on front tire so that it uses as a vertical reaction force at front tire. Maximum rear axle weight is 430kg which is the part of the Gross vehicle weight that loaded on rear left and rear right tires, so that it uses as a reaction force. For the two rear tires there will have 215kg (2109.8N) reaction load each. Detail loading of model is shown in Figure 6. Earth gravity is also considered for the chassis frame as a part of loading. The location of the CG of the driver, passengers and TWV's frame are calculated using the three wheeled vehicle drawing in the x, y and z coordinate system. These are as follows in x, y and z axis order.

- ✓ Driver= [-1350, 500,400] mm
- ✓ Passenger 1= [-550, 200, 400] mm
- ✓ Passenger 2= [-550, 512, 400] mm
- ✓ Passenger 3= [-550, 825, 400] mm
- ✓ Body CG= [-900, 500,148] mm

There are three displacement boundary conditions of model. The first boundary condition is applied in front part of the chassis, the second and the third boundary conditions are applied in rear part of chassis. For all displacement boundary conditions the z component of the displacement had set to zero.

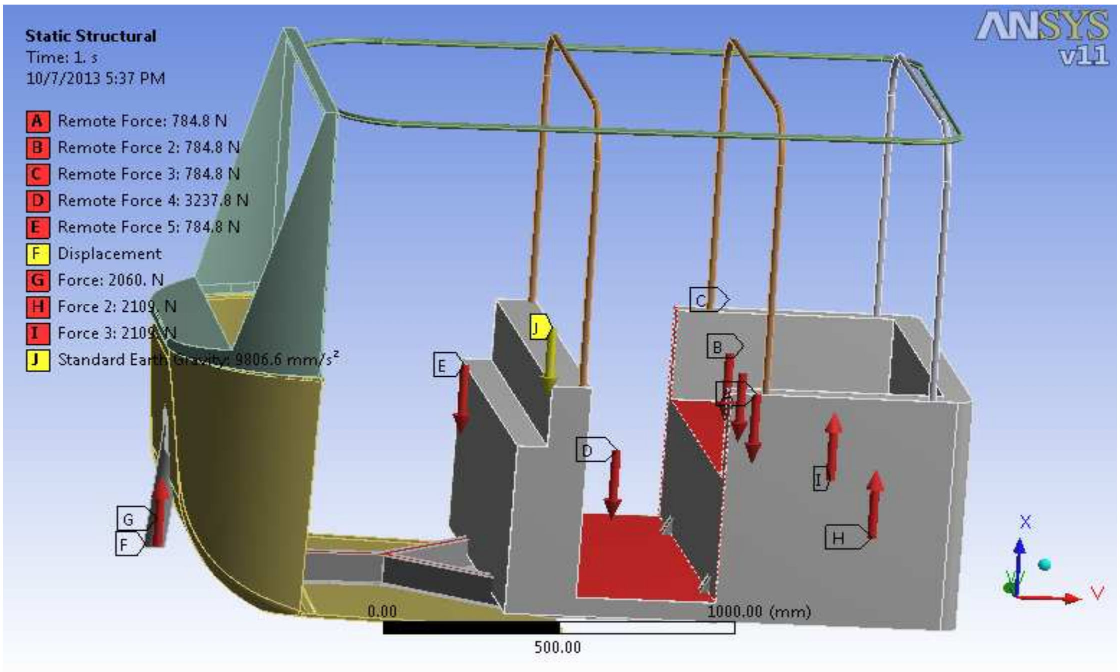


Figure 6: loads and boundary conditions of the existing model

Simulation Results

The overall solution progress is indicated by a status bar. In addition we can use the Solution Information object which is inserted automatically under the Solution folder. This object allows us to view the actual output from the solver, graphically monitor items such as convergence criteria for nonlinear problems and diagnose possible reasons for convergence difficulties by plotting Newton-Raphson residuals.

The analysis type determines the results available for us to examine after solution. For this project the analysis type is structural analysis and we are interested in equivalent stress results or maximum shear results. The result types section lists various results available to us for post processing. In this case the solution shown bellow in table 3 is obtained.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	2.9166e-003	1.6302e-003	-58,389	1.6799	0	-0.62973
Maximum	178.58	103.08	124.91	-	64.393	2.4887

Table 3: solution table of existing model static analysis

Maximum Von Misses stress and maximum shear stress

The Maximum Equivalent Stress safety tool is based on the maximum equivalent stress failure theory for ductile materials, also referred to as the von Mises-Hencky theory. The theory states that a particular combination of principal stresses causes failure if the maximum equivalent stress in a structure equals or exceeds a specific stress limit. The Maximum Shear Stress safety tool is based on the maximum shear stress failure theory for ductile materials. The theory states that a particular combination of principal stresses causes failure if the Maximum Shear equals or exceeds a specific shear limit, where the limit strength is generally the yield strength of the material.

From the table 3 the maximum Von Misses stress magnitude obtained is 178.58 MPa and the maximum shear stress magnitude is 103.08 MPa of critical point is around the side surfaces of the TWV about 600mm from rear as shown in figure 7 (a) and(b). But factor of safety is 1.6799 obtained as shown in figure 7 (c) and this is very reliable value. So it is safe mode of the chassis frame to resist the load because the stress results are all bellow frame material yield strength.

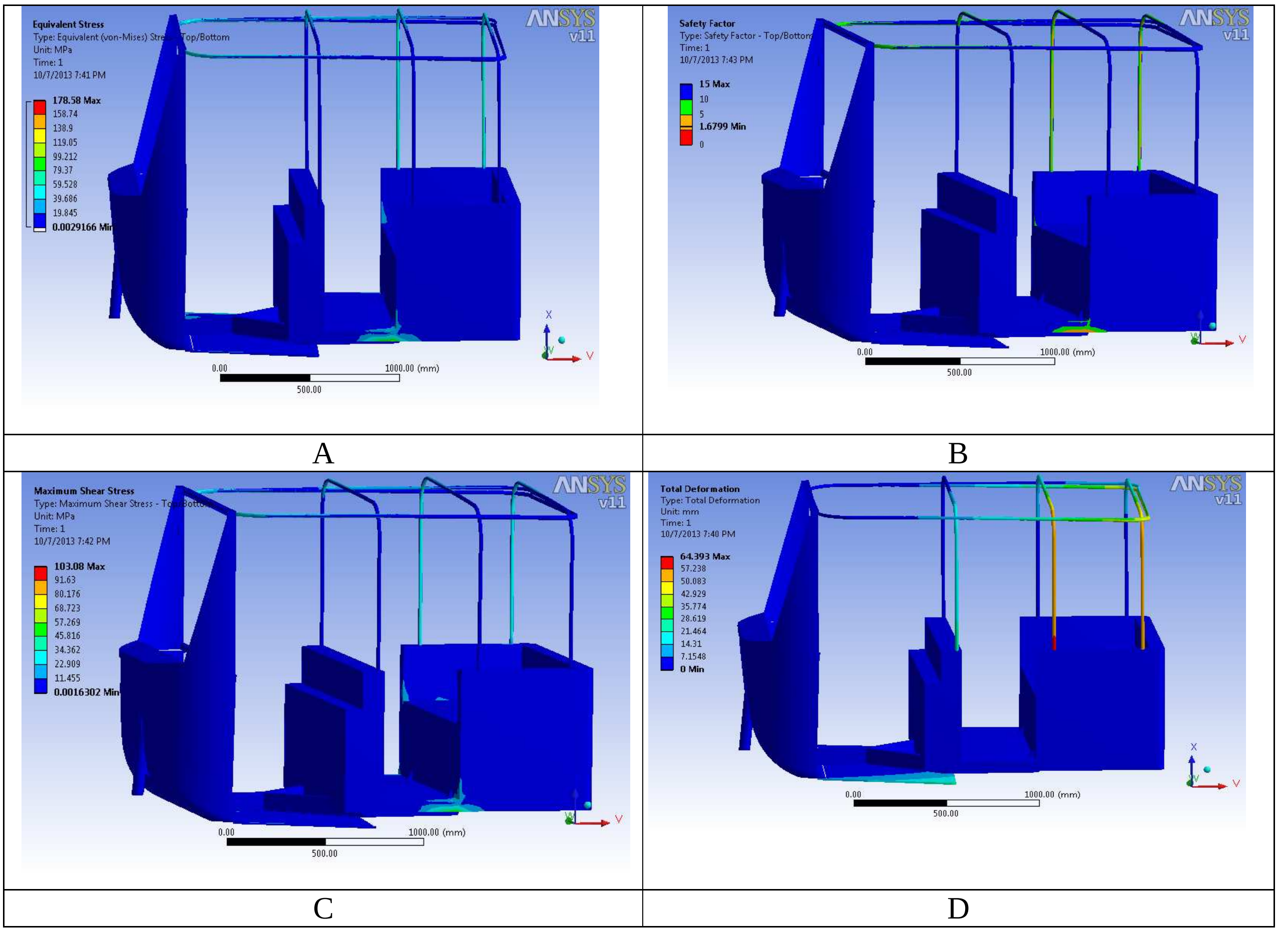


Figure 7: results of a) equivalent stress b) maximum shear stress c) safety factor respectively d) total deformation

Displacement results

The maximum displacement of chassis and location of maximum displacement is 64.39mm obtained around the rear part on the roofing tubes. The magnitude of maximum displacement is too large to expect that the model is safe. From this result we can see as the model needs some improvement around the tube structures.

3.4.2 Torsion loading condition

One of the most important FEA analyses of the frame will be the Torsion Rigidity analysis. That is because with this analysis, we are going to check the rigidity of the frame in all the cases that the vehicle is running, breaking, turning and dumping. All this dynamic actions will provoke serious torsion stresses on the frame. In torsion (simulate the wheel run up) the torque can be confirmed by axle load, suspension stiffness and the rough of road surface. For that, making a correct analysis of the deformation and torsion stresses on the frame is very important.

In order to design a car of maximum torsion stiffness the generalized equation for torsion must be examined. Figure 8 below is a basic shaft constrained at one end and an applied torque T at the other, with Φ denoting the resultant twist of the shaft.

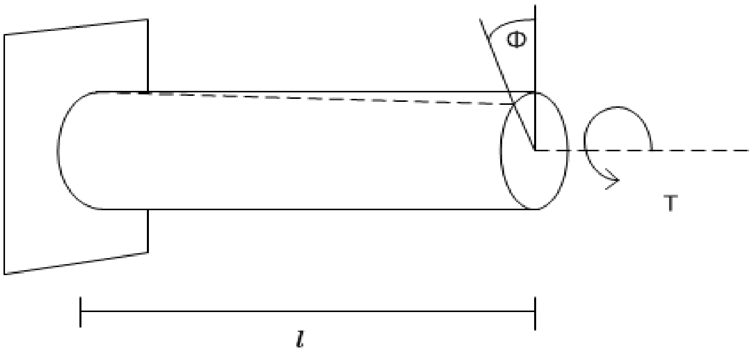


Figure 8: Simple Torsion of Shaft

Equation is the simple formula that relates this angle of twist to the applied torque, with J representing the shafts polar moment of inertia, G representing the shear modulus of the material and *l is* being the length of the shaft.

$$T = \frac{\emptyset JG}{L}$$

This equation can then be rearranged to express torsion stiffness,

$$\frac{T}{\emptyset} = \frac{JG}{L}$$

This expression displays that torsion stiffness in proportional to both the polar Moment inertia and material shear modulus, whilst being inversely proportional to the length. The stiffness of the chassis can be helped to be maximized, by using these key relationships and placing them in the context of the chassis.

- Length – relates to the wheelbase of the car.
- Shear modulus – is related to the Young's modulus of a material and will be examined.
- Polar moment of inertia – is a section property that is defined by equation above and is increased by having more cross-sectional area located further away from the neutral axis.

- o In the case of three wheeled vehicles torsion load will occur when one of the three wheels run up (bump) due to road roughness. Only one case may subject the TWV for torsion that is when one of the rear wheels bump. But, when the front wheel bumps up, it will create a bending load effect on the three wheeled vehicle chassis.

Method of loading for bump

To do the FEA analysis of Torsion Rigidity, we have to create a structure that will simulate the wishbones of the suspension, just in the line where the front axe will be. That's because we want to simulate the efforts that the road make on the wheels. This effort will be transmitted to torsion efforts on the frame. An assumption is made that when the vehicle passes over a bump, the entire weight of the vehicle will turn into two point loads at the two points where the wheel force is transmitted to the frame, through the suspension. The worst case will be when the suspension fails and the entire force is transmitted. As the requirement is not for the frame to fail in case the suspension fails. These two point loads will be equal to the weight of the frame.

$$2F = m \times g$$

Where, F is force on each wheel, m is total mass of the vehicle including driver and passengers, and g is the gravitational acceleration.

3.4.2.1 Rear right Wheel Bump case

Model used is full model of the three wheeled vehicle and the load is the total weight (Vehicle +Driver + passengers weight) is applied on the model. Boundaries conditions are all DOF (Ux, Uy, Uz) are zero at front wheel and rear left wheel but free at the rear right wheel attachment point. That means the frame will be fixed selecting the joints at the rear left wheel position of the frame and at the front wheel position where the suspension would mount.

Stress results

The maximum Von Misses stress magnitude, 295.31 MPa of critical point is around 600mm from rear of TWV and the maximum shear stress magnitude is 170.45 MPa and 600mm from rear as shown in the figure 9. But factor of safety is 1.0159. As we see on the stress plot (table 4), the maximum stress is under the material resistance. But it is almost equal. That is impossible by a correct design of the frame. One of the important factors is that the car is very heavy at all, maximizing the efforts. Other factor is that the suspension will not absorb the great part of the impacts produced by bumps on the road. That provokes that the load that we use to make the test is maximized.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	3.2588e-003	1.8263e-003	-75.25	1.0159	0	-1.103
Maximum	295.31	170.45	164.69	-	64.449	2.7419

Table 4: simulation results of rear right wheel bump case

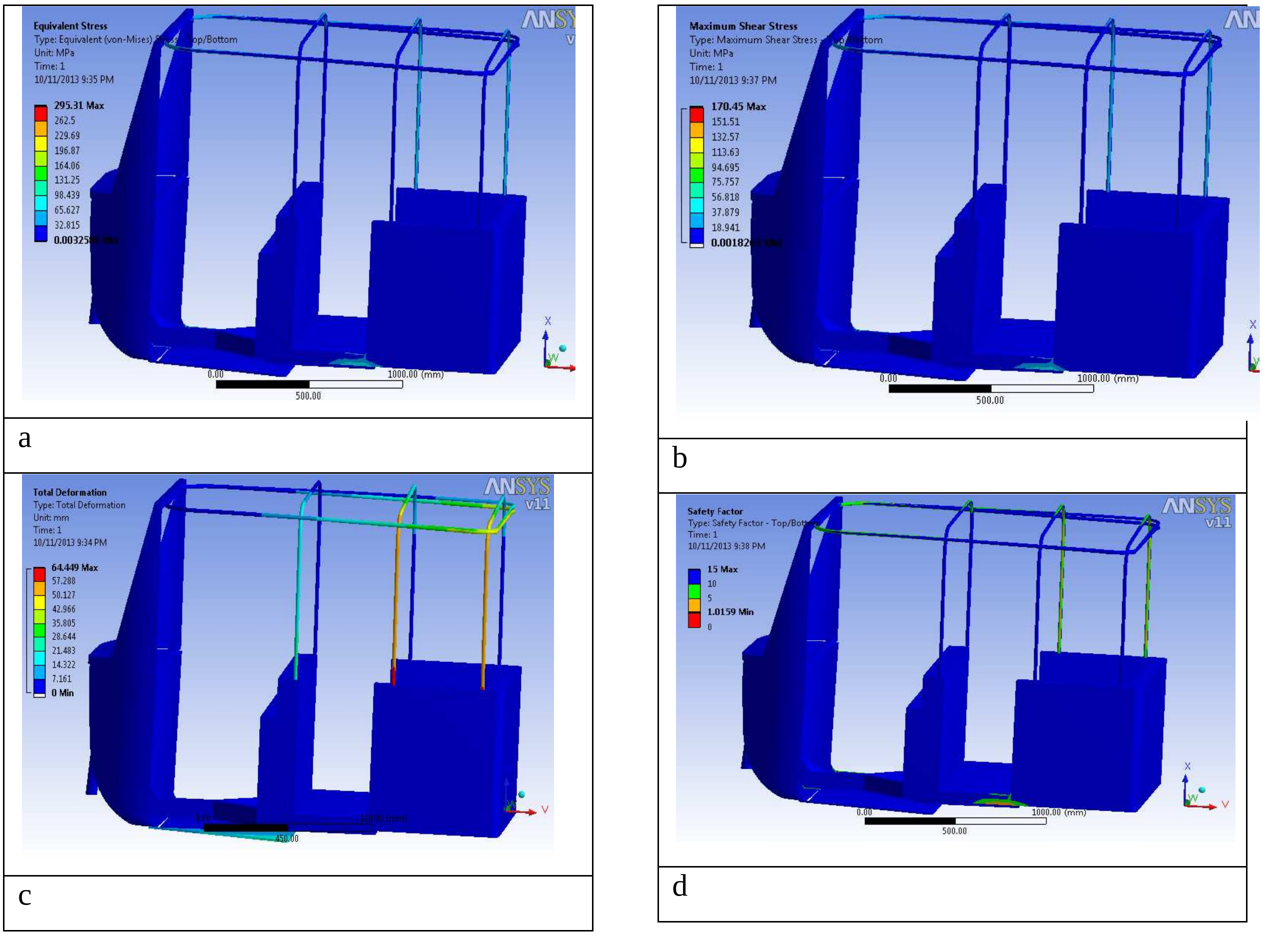


Figure 9: show a) equivalent stress b) maximum shear stress c) total deformation d) safety factor respectively

Deformation results

As we can see on the displacement plot in figure 9(c), the maximum displacement in any components of the frame, under a torsion effort will be about 64.449 millimeters.

3.4.2.2 Front wheel bump case

The model used is full model of the three wheeled vehicle and the load applied on vehicle is similar to the rear bump case. Boundary conditions are all displacement values had set to zero at rear right wheel and rear left wheel but free at the front wheel attachment point. That means all loads will be transmitted to frame through rear wheels.

Stress results

The maximum Von Misses stress magnitude is 276.74 and the maximum shear stress magnitude is 159.77 MPa. But factor of safety is 1.084. As we see on the stress result plot in table 5, the maximum stress is under the material resistance. But it is almost equal.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	2.2975e-003	1.3259e-003	-64.855	1.084	0	-0.62386
Maximum	276.74	159.77	139.4	-	64.421	2.4187

Table 5: simulation results of front wheel bump case

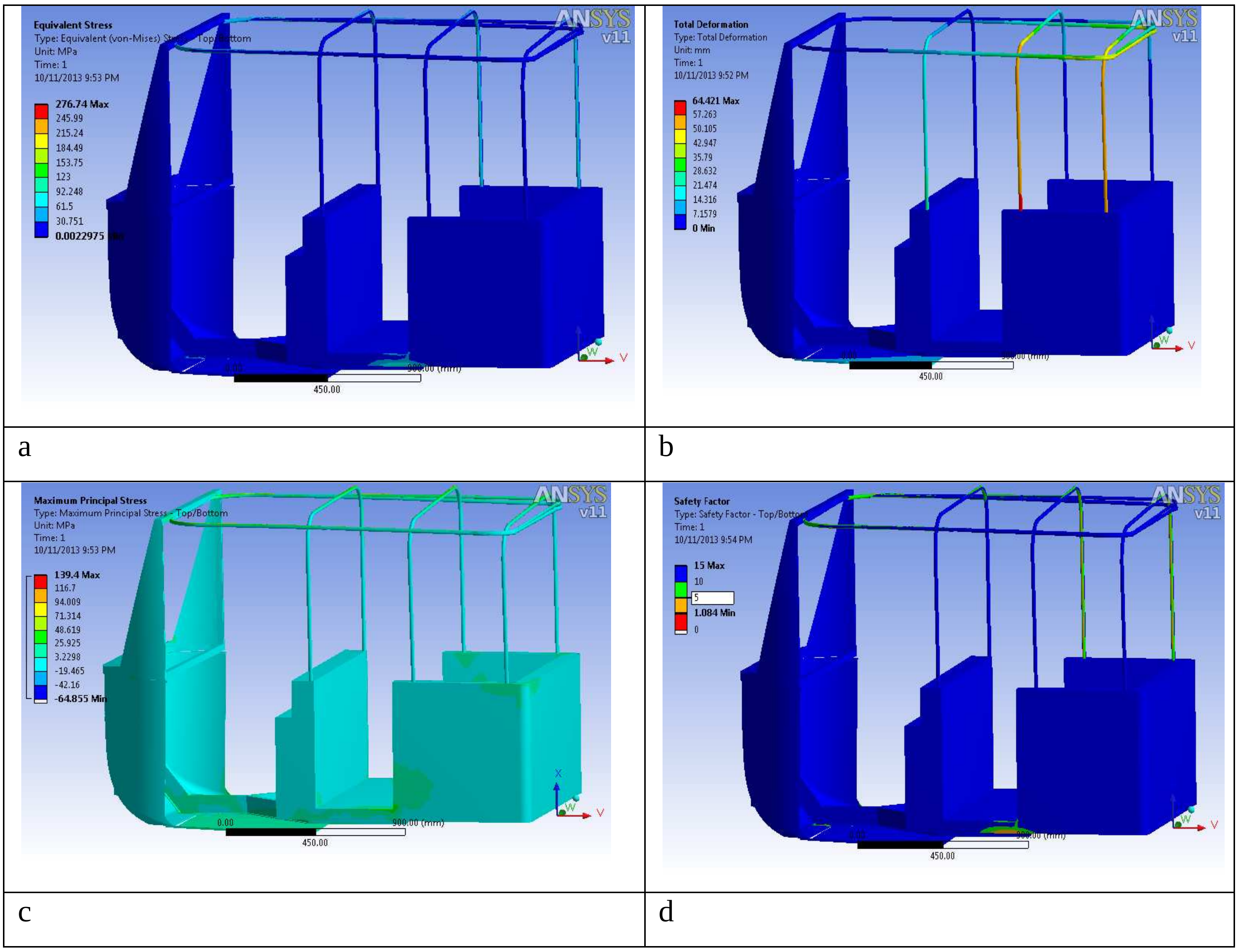


Figure 10: show a) equivalent stress b) total deformation c) maximum principal stress d) safety factor of front wheel bump test respectively

Displacement Result

As we can see in figure 10 (b) the maximum displacement on the main frame, after the application of the forces described, and fixing the frame correctly is 64.421 mm. The figure tells us this displacement has been occurred on the roofing tubes. This means that these tubes are sensitive to displacement and have no contribution for strength because they have highly displaced with a load of under the material strength.

3.4.3 Impact simulation

The force applied on the front head direction is pretty big, because is simulating a front impact. Then we need a very tough structure to resist this kind of impact. We can get the result of the test that the maximum displacement under these conditions is occurred by using the following dynamics load. For a perfectly inelastic collision, energy transferred,

$$\mathrm{Et} = \frac{\mathrm{m1} \times \mathrm{m2} \times (\mathrm{u1} - \mathrm{u2})^2}{2(\mathrm{m1} + \mathrm{m2})}$$

Where, m1 & m2 are the two colliding vehicle masses with velocities u2 & u1 respectively. Since both m1 & m2 are two three wheeled vehicles with similar masses & the vehicle m2 is at rest. Implies that, m1 = m2 & u1 =0 then,

$$E_t = 0.25 \times \mathrm{m1} \times \mathrm{u1}^2$$

Now, $F = \frac{\mathrm{Et}}{\mathrm{t}}$ where, t is impact time and F is impact force.

$$F = \frac{0.25 \times m1 \times u1^2}{t}$$

Total weight of the vehicle (including driver and passengers) is m1 which is 640 Kg .maximum sped of the vehicle is 60 km/h that is 16.7 m/s. in most crashes t is the order of 1 second. Now in our case of three wheeled vehicle crash force becomes,

$$F = \frac{0.25 \times 640kg \times (16.7m/s)^2}{1s} = 44622.4N$$

Also the design is for no plastic deformation, the vehicle should remain in the elastic region. The design factor of safety should remain above unity taking in to account uncertainty in the nature of forces.

3.4.3.1 Front impact FEM analysis

To do the FEA analysis of the front Bulkhead, we are going to apply the load of 44622.4N at the front in addition to static loads we have seen before in the static condition analysis. The point of application of these loads will be the actual attachment points between the impact front heads (figure 11). The frame will be fixed displacement but not rotation of the bottom nodes of both sides and both locations. Model used is full model of the three wheeled vehicle with boundary conditions at suspension mounting points Uy = Uz = 0 and at rear corner points all DOF=0.

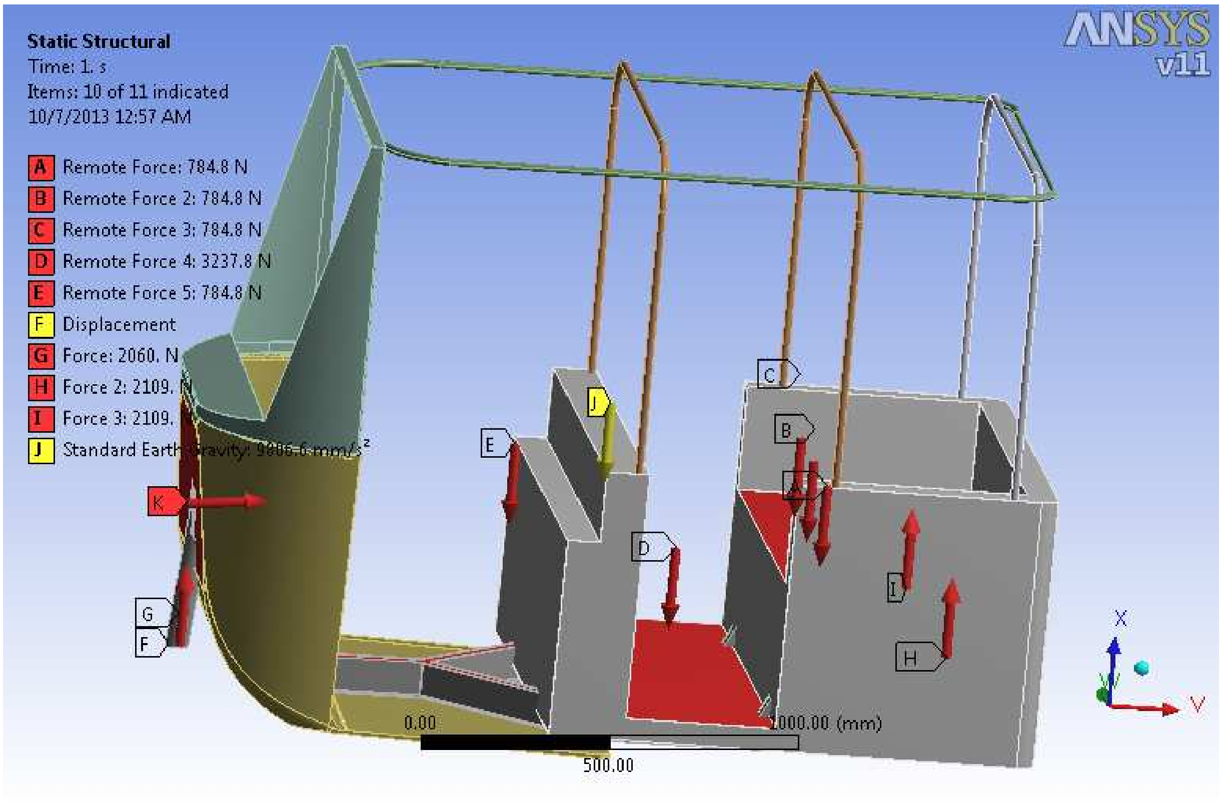


Figure 11: loads and boundary conditions for front impact

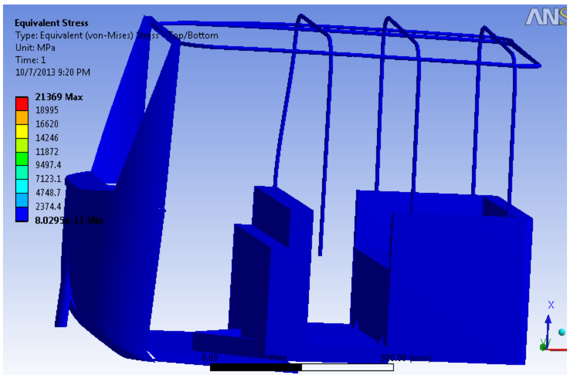
Stress results

Since the force applied on the front bulkhead, is pretty big, the location of maximum Von Misses stress and maximum shear stress are at front. The Von Misses stress magnitude of critical point is 21369 MPa and the maximum shear stress magnitude is 12337 Mpa as we see from table 6. As we can see, the result of the test is that the maximum displacement under these conditions is 106.92 mm, which is totally unacceptable. This result has happened because the circular bars are in incorrect position and not giving a high rigidity to the frame. Safety factor is 0.014039 hence; the Chassis will not safe under Frontal Impact. Figure 12 (d) shows this truth in a very clear way.

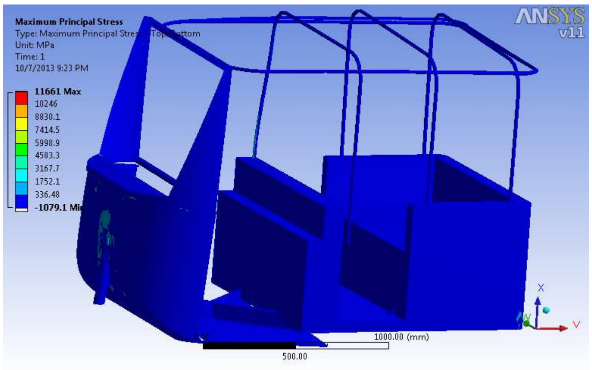
Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	8.0295e-011	4.1838e-011	-1079.1	1.4039e-002	0	-5.0448
Maximum	21369	12337	11661	-	106.92	86.948

Table 6: simulation results of front impact case

a



b



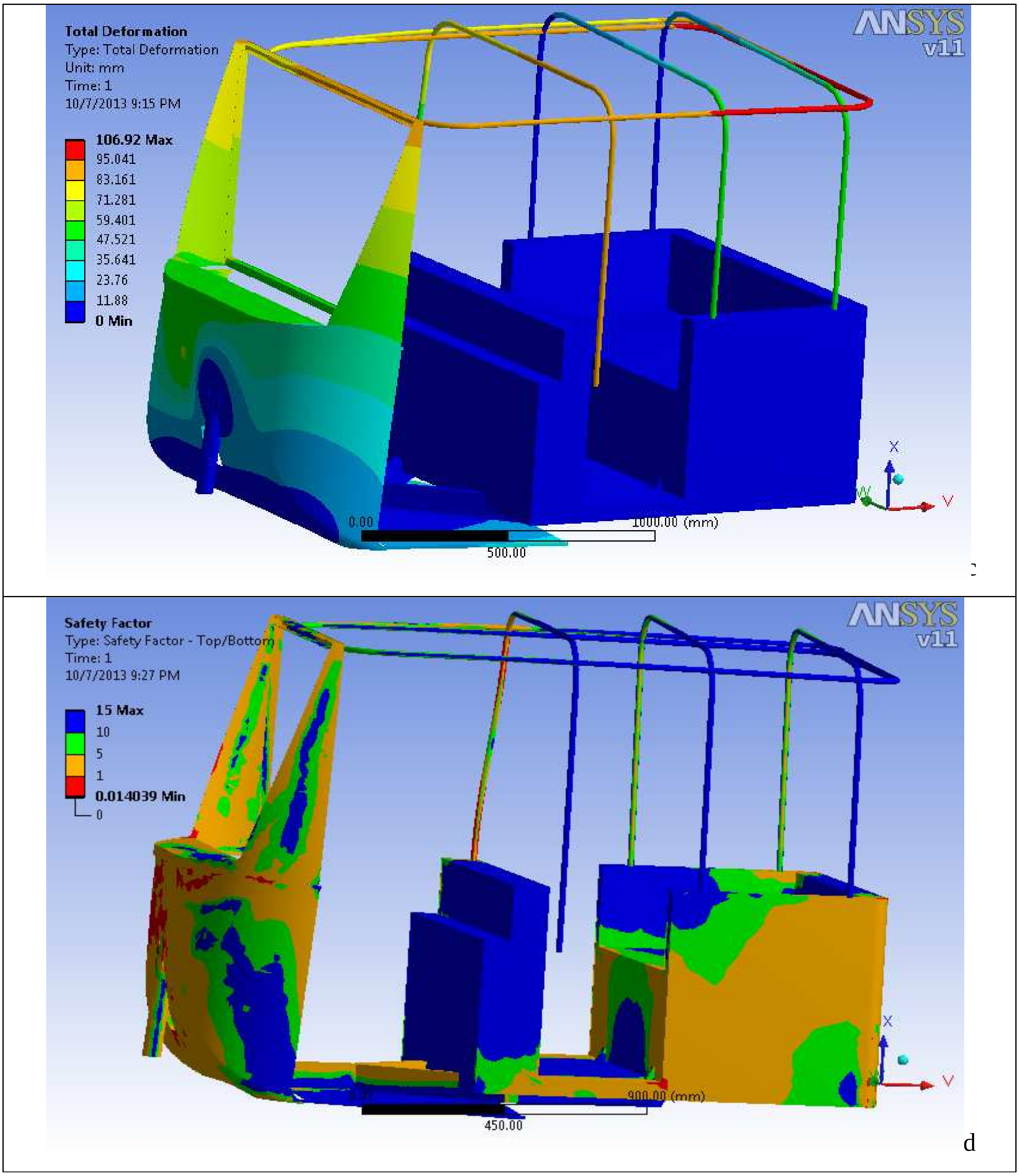


Figure 12: show a) equivalent stress b) maximum principal stress c) total deformation d) safety factor of front impact test respectively

3.4.3.2 Rear impact FEM analysis

To do the FEA analysis of the rear head impact, we are going to apply the load of 44622.4N at the rear (figure 13) in addition to static loads we have seen before in the static condition analysis. The boundary conditions are used at Suspension mounting points Uy = Uz = 0 and at front corner points all DOF=0.

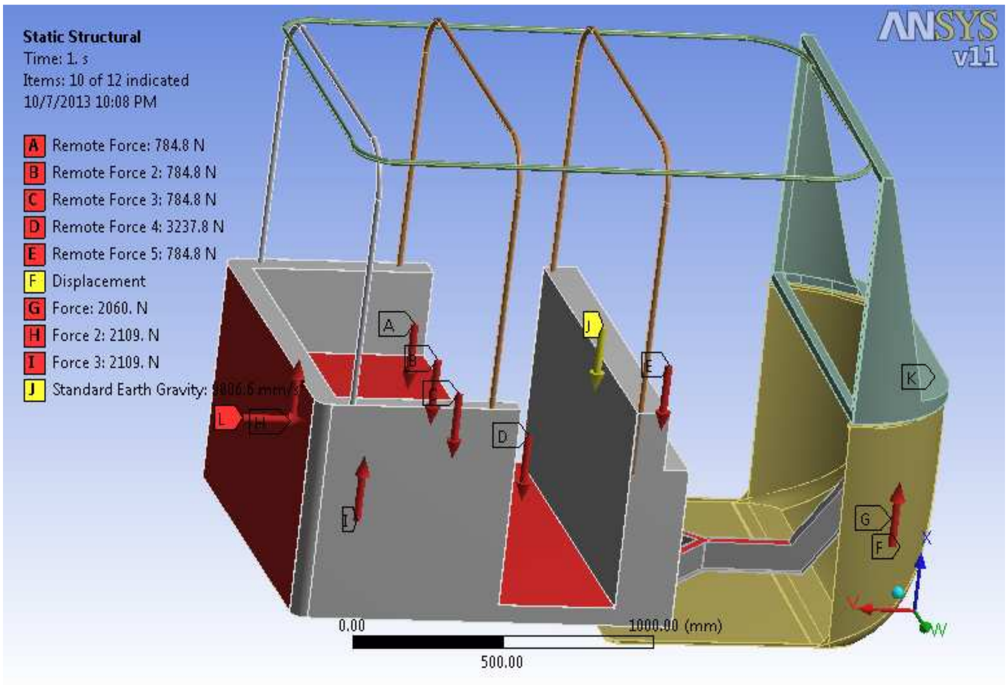


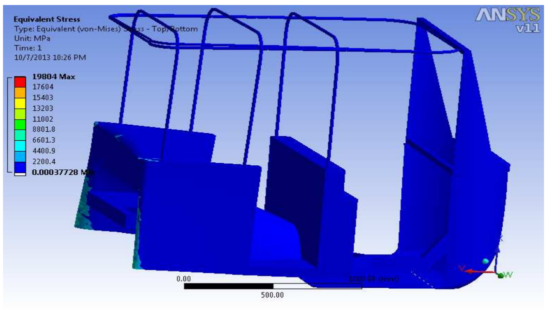
Figure 13: loads and boundary conditions for rear impact

Simulation results

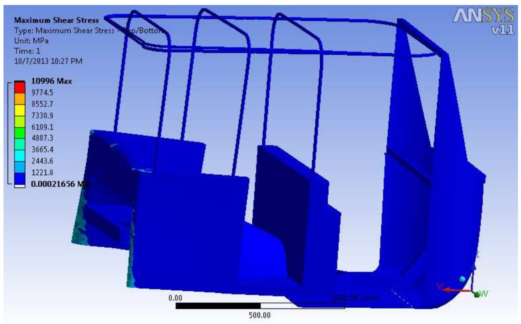
The location of maximum Von Misses stress and maximum shear stress are at rear. Table 7 shows the Von Misses stress magnitude of critical point is 19804 MPa and the maximum shear stress magnitude is 10996 Mpa due to Impact. As we can see the table, the result of the test is that the maximum displacement under these conditions is 1436.8 mm, which is totally unacceptable. This result will be done because the frame could not resist the load, has not enough rigidity. Safety factor is 0.015148 shown in figure 14(d) hence, the chassis will not safe under frontal impact.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	3.7728e-004	2.1656e-004	-7034	1.5148e-002	0	-1435.8
Maximum	19804	10996	22060	-	1436.8	2.7133

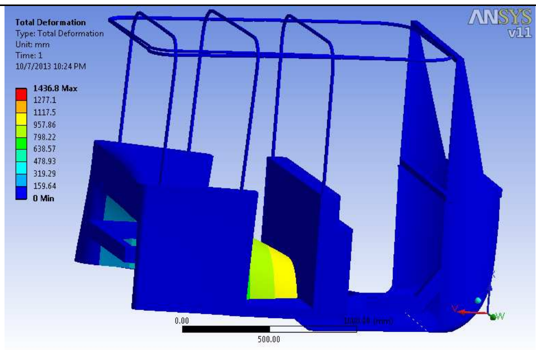
Table 7: simulation results of rear impact case



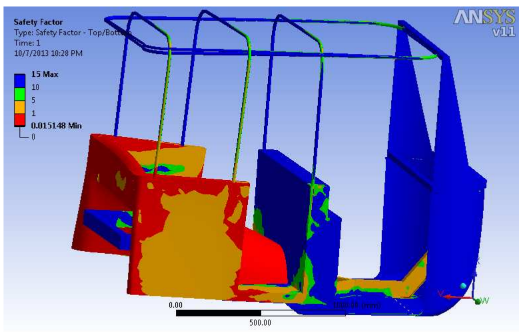
a



b



c



d

Figure 14: show a) equivalent stress b) maximum shear stress c) total deformation d) safety factor of rear impact test respectively

3.4.3.3 Side impact FEM analysis

To do the FEA analysis of the side impact, we are going to apply the Load of 44622.4N at the side in addition to static loads we have seen before in the static condition analysis. The point of application of these loads will be the actual attachment points between the impact front and side heads. Boundary conditions applied at suspension mounting points are Uy = Uz = 0 and at rear corner points all DOF are zero.

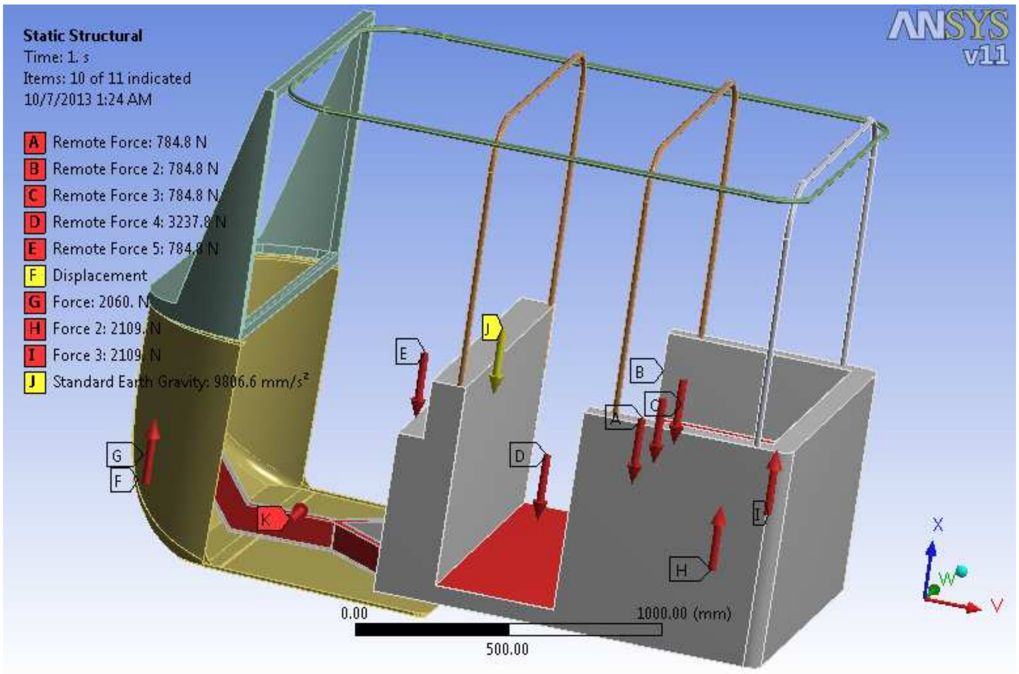


Figure 15: loads and boundary conditions for side impact

Simulation results

The location of maximum Von Misses stress and maximum shear stress are at sides around passenger's seat. The Von Misses stress magnitude of critical point is 345.36 MPa and the maximum shear stress magnitude is 197.39 MPa obtained. As we can see, the result of the test is that the maximum displacement under these conditions is 66.116 mm, which is totally unacceptable. Safety factor is 0.86942. Hence, the chassis will not safe under frontal impact. See table 8 and figure 16 for details.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	1.3414e-007	7.7217e-008	-69.32	0.86942	0	-1.0997
Maximum	345.08	197.39	228.73	-	66.116	1.2501

Table 8: simulation results of side impact case

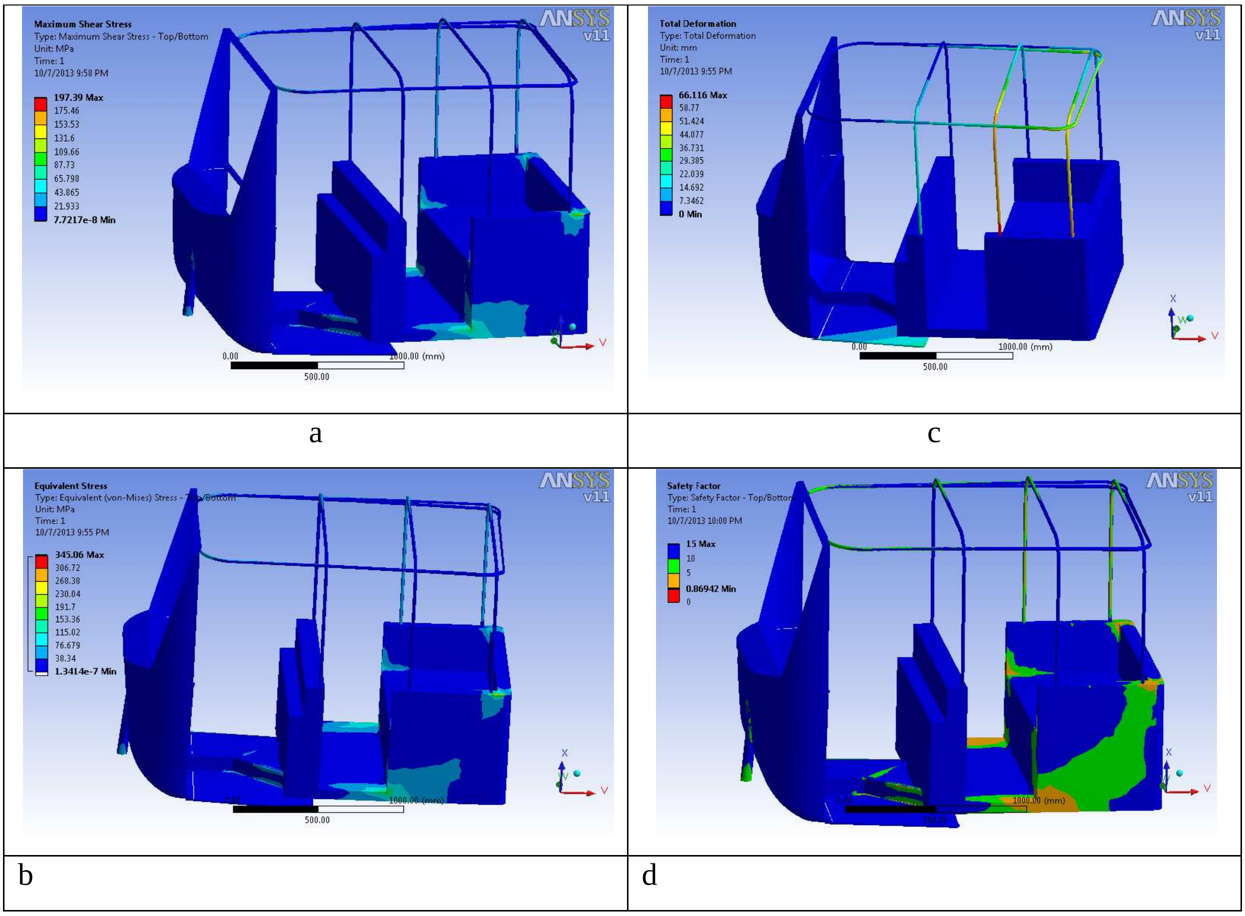


Figure16: shows a) maximum shear stress b) equivalent stress c) total deformation d) safety factor

3.5 Result verification and conclusion for the existing model

The highest von-miss stress occurred is 21369 MPa by FEM analysis in case of front impact. The material yield strength is 300 MPa. The result of FE analysis is bigger 70 times than the material strength. The maximum displacement of numerical simulation result is 1436.8 mm and the minimum safety factor is 0.015148 obtained in case of rear impact. The result of numerical simulation is unexpected and unimaginable. These values tell us that the model could not resist the crash loads and could not give a protection to the passengers and to driver also. It indicates that it is necessary to model a new three wheeled vehicle structure.

The conclusion that we can get of this FEA analysis is that the frame is not correctly dimensioned for the weight of the car. That will be because the strength is only for static case but during impact the load resistance reliability suggested is not optimum. Other factor is that the sheet metal cover of the car is making a frame heavier for a vehicle; maybe it will be better to improve the frame and think about making the vehicle lighter and strong. In that case the space frames will make better strength, but reducing the weight of the car. As we have seen the sheet metal has no value to protect driver and passengers. Using the space frame has a great advantage to alleviate this shortcoming and to protect passengers from risk.

Chapter four

Modeling and analysis of new TWV model

4.1 Frame design criteria

A frame of a vehicle can be defined like a structure that has the finality to connect in a rigid way the front suspension with the rear one and at the same time brings anchorage points for the different parts of the vehicle. Its finality is also protecting the driver and passengers in case of impacts or rollovers. The design of a frame is a commitment between rigidity, weight and volume, without forgetting the final economic cost. Have to be considered the static resistance and the fatigue, the loads limits on the unions, the building and the assembly. In this project only takes into account the static stresses and the impacts, leaving for a higher level of complexity the fatigue calculations. In the frame design, regarding to stiffness, there are some aspects to take in count [2].

- Items that are not part of the structure that give stiffness to the frame.
- The anchor points that support large loads must be triangulated, like the suspension attachments.
- Although is interesting to the frame to allow deformation due to an impact, the part that protects the legs has to be totally rigid.

Rigidity criteria

We must differentiate between bending stiffness and torsion stiffness. The first refers to the deformation of the frame under the weight of the different elements. Not a problem while designing the chassis. The second refers to the frame deformation when is under an asymmetric load, such as when one wheel goes through a hole.

Several studies show that the deformation due to axial forces is much smaller than due to bending moments and torsion. That is why it is better to load the bars with axial efforts than with bending moments or torsion moments. Regard with the axial effort type, is better the compression efforts than the traction efforts to avoid buckling problems [2].

Mass distribution criteria

- It is interesting to maintain the rigidity of the frame, trying to get the overall weight of the frame lower as possible to take advantage of the engine power.
- Regard to the suspension studies, it is convenient to keep the center of gravity as low as possible [2].

Space criteria

- Leave space for the engine and its components (exhaust, air intake…) and the transmission (chain and differential).
- The frame needs to allow the entry on the cockpit of a 95th percentile male with a helmet.
- The size of the pedals, the angle and length of the legs and feet determine the length of the pilot's compartment (cockpit).

Cost criteria

- The selection of the bars has not to be very varied in terms of diameter and thickness
- Will be bending the low number of bars and trying to bend only bars of one determinate diameter.
- Minimum number of joints.

In general the Requirements of bodies for various types of vehicles: The body of the most vehicles should fulfill the following requirements [3]:

1. The body should be light.
2. It should have minimum number of components.
3. It should provide sufficient space for passengers and luggage.

4. It should withstand vibrations while in motion.
5. It should offer minimum resistance to air.
6. It should be cheap and easy in manufacturing.
7. It should be attractive in shape and color.
8. It should have uniformly distributed load.
9. It should have long fatigue life
10. It should provide good vision and ventilation.

4.2 Basic decisions

Some basic decisions must be taken at the beginning of the project development, such as the type of structure, the manufacturing processes and the materials used. These three decisions must be taken together. The materials must be adequate to use the manufacturing processes selected. Type of structure is Space frame structure will be selected. Using this structure the TWV will be safer and adequate for mass production.

The space frame design is a very popular option for chassis type, as it is a low cost, simple to design and easy to construct. The design is governed by the Pakistan Standards and Quality Control Authority (PSQCA) for three wheeler auto vehicles. PSQCA rules, although they leave enough freedom to see a wide verity of space frame shapes, sizes and design techniques. Generally there are two main options for the material used in the space frame, these are plain carbon steel and AS 4130, commonly known as chrome –molly because of its chromium and molybdenum content. One of the main advantages of using the space frame design is its easy and logical construction process, of which can be performed by person with intermediate knowledge and experience using basic welding and metal working equipment [3]. Generally, the space frame chassis is the most suitable chassis type used in the prototype car construction in world compared to others chassis types, hence I decide to apply this concept in my thesis.

There are many advantages obtained from space frame. Since the space frame systems are triangulated format, it will provide maximum strength and minimum deflection of the design compare to the other chassis types due to the support from tubular pipes. Space frame chassis systems are lighter than traditional steel. Therefore, it provides significant economy in foundation costs. Using a space frame chassis in a three wheeled car, the high torsion rigidity can be achieved as well as its light weight. It means that, space frame chassis designs will enhance the rigidity per weight ratio.

4.3 New three wheeled vehicle 3D CATIA model

The CATIA V5R16 is used for model of the new three wheeled vehicle. It is made by using rectangular hallow section (RHS) of 40×40×3mm steel beam welded together. Chassis is a space frame type with the same length width and height value as previous.

parameter	**Value in mm**
length	2625
width	1300
height	1710

Table 9: basic dimensions of the new model

The Pakistan standard of three wheeled vehicle has many rules that should be fulfilled during the design of TWV. These rules are listed bellow in the table10 and used in thesis as guidance.

Rule numbers	n a m e	specification
1	seating capacity	Three (3) or Seven (7) persons including driver.
2	safety bar	There shall be a partition between the driver and passengers and with safety bars.
3	provision of doors	(OPTIONAL)
4	protection from	The passengers shall be provided protection from weather

	weather	by means of fiberglass or any suitable water-resistant material.
5	over hang	Rear over hang shall not be more than 68 cm (27 inches).
6	seat	Every passenger shall be provided with at least 38 cm (15″) x 38 cm (15″) seats.
7	seating height	Shall not be less than 30 cm (12 inches) measured from floor board.
8	backrest	Every passenger shall be provided with backrest of 40 cm (16 inches) height minimum.
9	roof	Watertight roof shall be provided which shall cover completely the area from windscreen to the extreme rear edge of the vehicle.
10	height of roof	The internal height of the roof measured from the seat level to any point vertically above each seat shall not be more than 1060mm.
11	leg room back	Every passenger shall be provided with legroom not less than 32.5 cm (13 inches).
12	floor board	Shall be strong made of suitable material so closely fitted covered to prevent smoke or dust from entering in.
13	entrance level	In case of empty vehicle the entrance level (step of floor board) shall not be less than 30 cm (12 inches) and more than 55 cm (22 inches).
14	mirrors	Shall be provided internally and externally so fitted as to enable the driver to have view of the road in the rear of the vehicle.

Table 10: rules and standards of Pakistan [3]

Using data in table 9 and rules and standards of Pakistan in table 10 the CATIA model of the new vehicle structure has been developed. This model in figure 17 shows the model of the new three wheeled vehicle that is ready for FEM analysis.

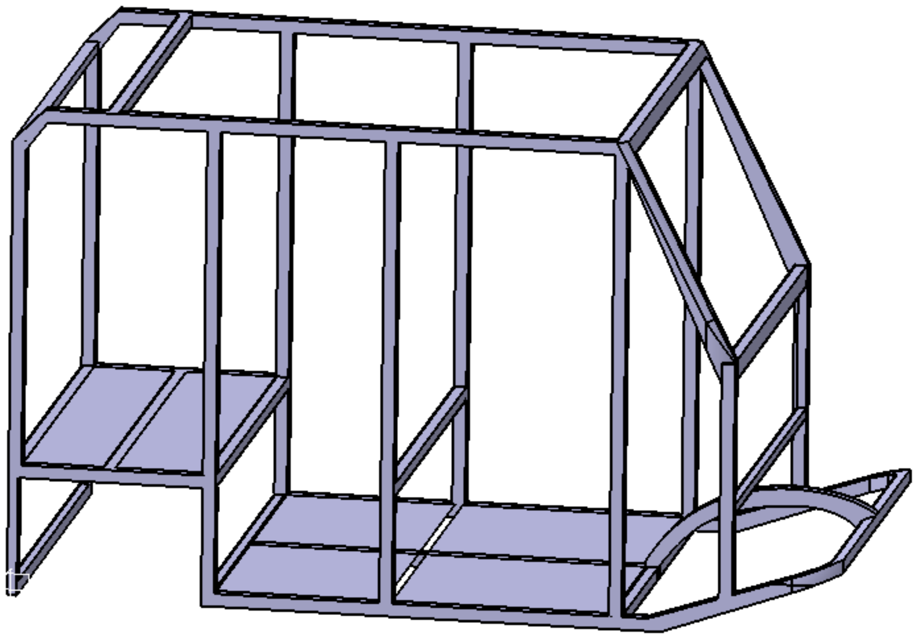


Figure 17: CATIA model of the new three wheeled vehicle

4.4 FEM analysis of the new vehicle model

The element types selected were BEAM 181 for the tubing sections and shell element being used for skin floor. ANSYS only required the input of Young's modulus and Poisson's ratio, with the values of 200GPa and 0.3 used respectively; these were used for both the tubing and sheet steel.

An important process of any FEA is to refine the mesh that is used in the analysis, a process that involves reducing mesh or element size a number of times and plotting the results as to ensure the analysis is stable as shown in figure 18.

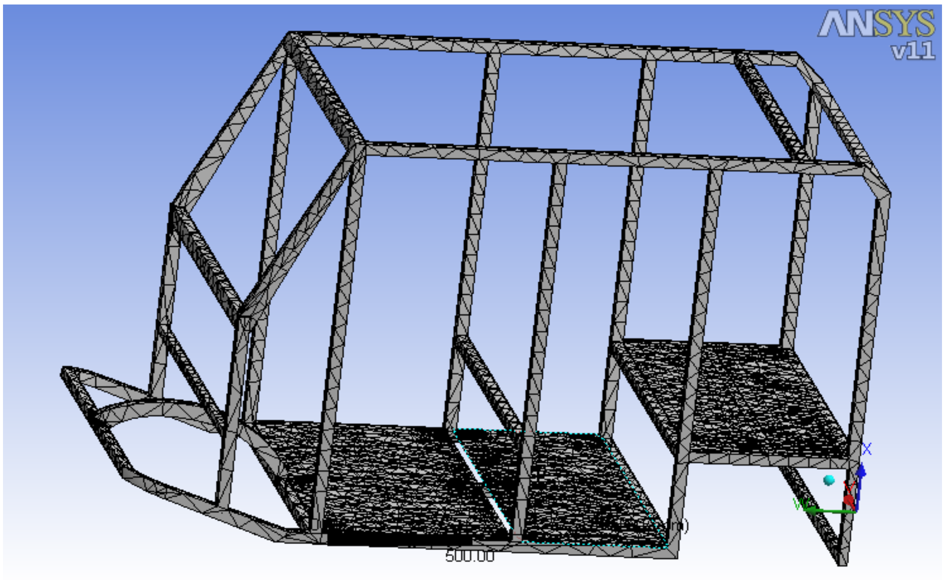


Figure 18: FEM mesh of the new model

Element and Nodes

The meshed new model of TWV chassis has number of elements and number of nodes. The element is BEAM 181. In order to get a better result, locally finer meshing applied in the region which is suspected to have the highest stress that is around the sheet metal. Finally the model has 39943 numbers of Nodes and 18992 numbers of Elements.

4.4.1 Static analysis

Loading and Boundary condition

Total weight, the weight of motor car having all components without occupant or baggage is, 330 kg or 3237.3N. This has been applied at the center of gravity of the frame. And Maximum play-load including driver is 320kg applied at seat points of each passenger and driver (figure 19). For each will have 80Kg (784.8N) weights. Actually no modification applied to loads and boundary conditions since our aim is to get better strength and stiffness in comparison to the existing model.

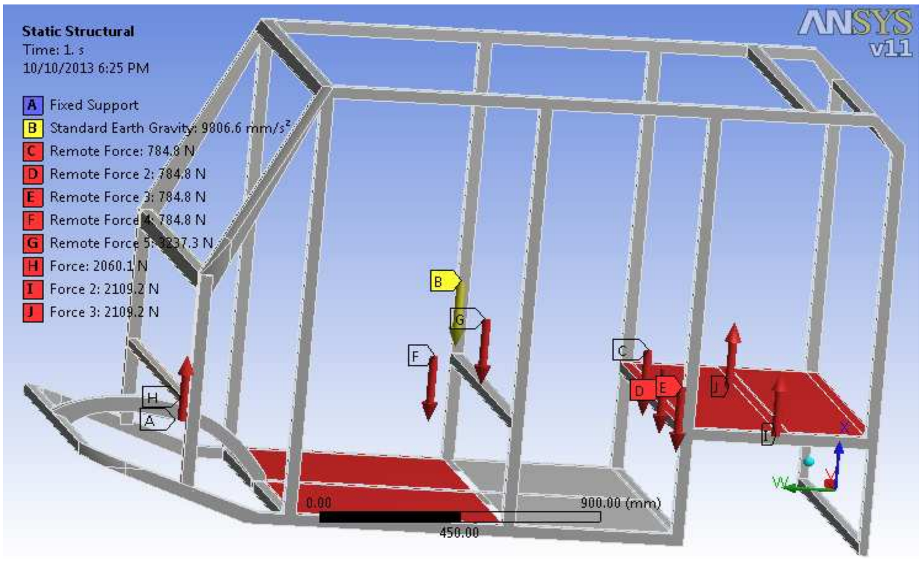


Figure 19: loads and boundary conditions for new model static analysis

Analysis results review

Automobile chassis usually refers to the lower body of the vehicle including the tires, engine, frame, driveline and suspension. Out of these, the frame provides necessary support to the vehicle components placed on it. Also the frame should be strong enough to withstand shock, twist, vibrations and other stresses. The chassis frame consists of side members attached with a series of cross members. Stress analysis using finite element method (FEM) can be used to locate the critical point which has the highest stress. This critical point is one of the factors that may cause the fatigue failure. The magnitude of the stress can be used to predict the life span of the vehicle chassis. The accuracy of prediction life of vehicle chassis is depending on the result of its stress analysis.

Maximum Von Misses stress and maximum shear stress

For the new model, the maximum Von Misses stress magnitude obtained is 221.31 MPa (figure 20(a) and table 11). Its critical point is around floor of 1100mm from rear of TWV and the maximum shear stress magnitude is 125.26 MPa (figure 20(b)). But factor of safety is 1.3556 (figure 20(d)). So it is safe for stiffness and thickness of the chassis frame at reliable stress level. It indicates as it is necessary to accept a new three wheeled vehicle.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	6.2787e-003	3.5104e-003	-32.338	0.35557	0	-2.4154e-002
Maximum	221.31	125.26	132.97	-	0.44868	1.3635e-002

Table 11: static analysis results of new model

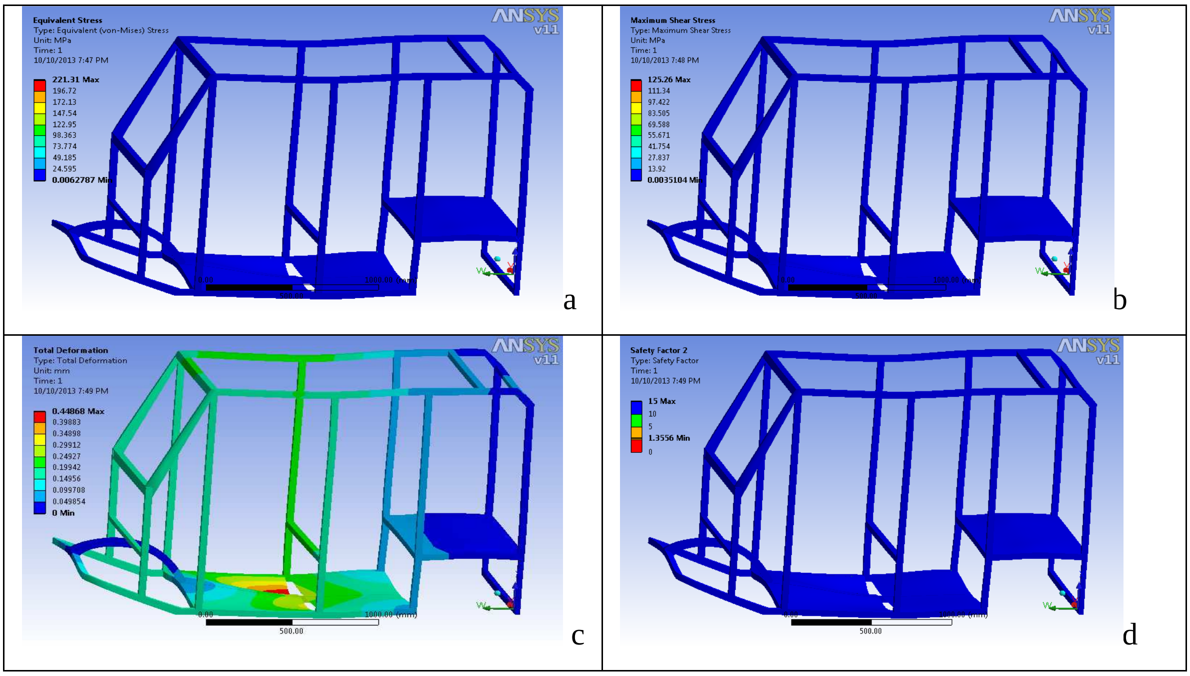


Figure 20: shows a) equivalent stress b) maximum shear stress c) total deformation d) safety factor

Displacement results

As we can see in figure 20 (c), the maximum displacement on the main frame, after the application of the forces described, and fixing the frame correctly is 0.44868 mm. Then we have got minimum displacement and that occurs on sheet metal. That has been got thanks to a correct designing of the shape of the TWV structure. The magnitude of maximum displacement is too exact to expect that the model is safe.

4.4.2 Crash analysis of new TWV model

4.4.2.1 Front crash simulation

To do the FEA analysis of the front impact for the new TWV, we are going to apply the load of 44622.4N at the front in addition to static loads. The frame has displacement boundary conditions at suspension mounting points Uy = Uz = 0 and at rear corner points all DOF=0.see figure 21.

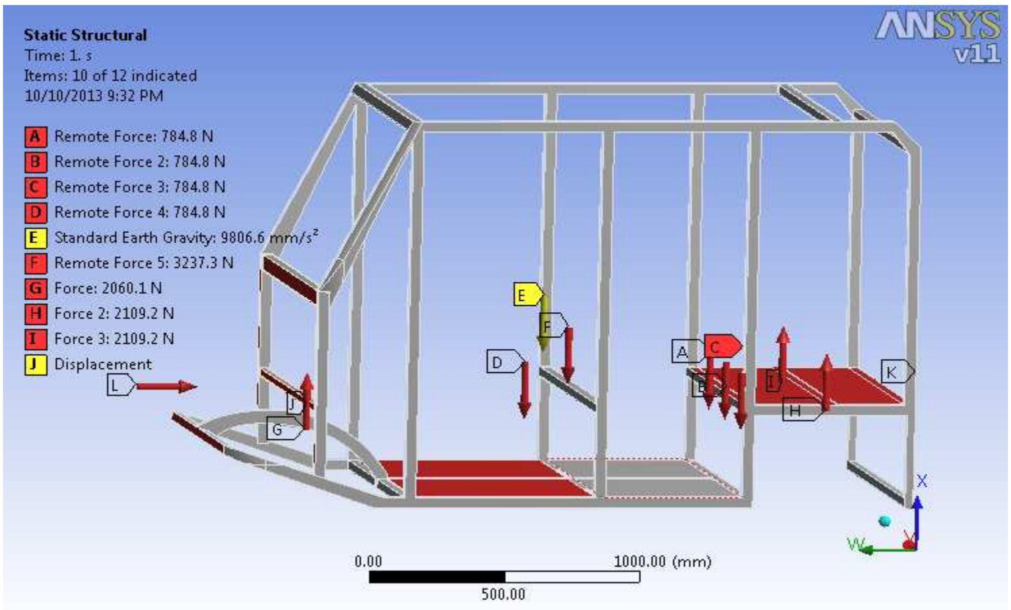


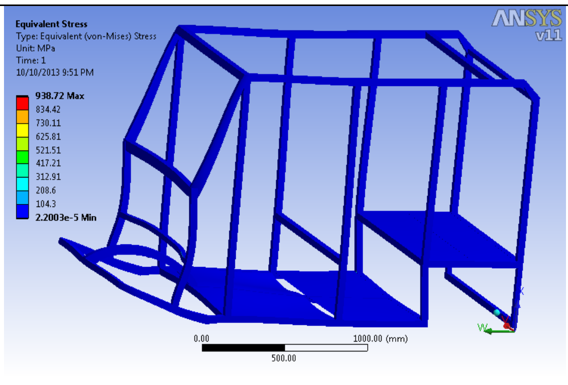
Figure 21: loads and boundary conditions for front impact analysis of the new TWV model

Analysis stress results

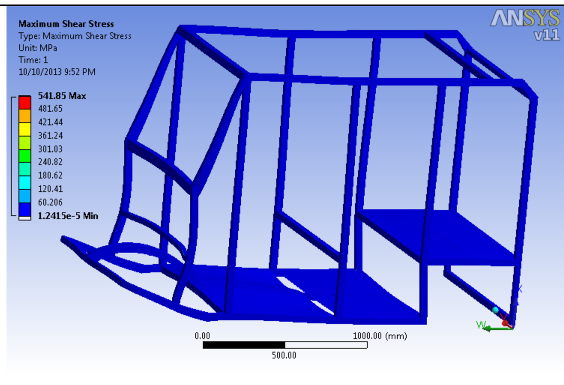
Since the force applied on the front head the location of maximum Von Misses stress and maximum shear stress occurred at front (figure 22). The Von Misses stress magnitude of critical point is 937.72 MPa and the maximum shear stress magnitude is 543.85 MPa (table 12). Even if the stress values obtained are above the material strength we should not forget as the analysis is for collision. This result has found because the bars are in a correct position, and giving a high rigidity to the frame. Safety factor is 0.31958 hence, the Chassis will not safe under frontal impact but compared to the existing model it is more reliable under crashing condition.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	2.2003e-005	1.2415e-005	-56.232	0.31958	0	-4.3832e-002
Maximum	938.72	541.81	558.16	-	1.0883	2.3177e-002

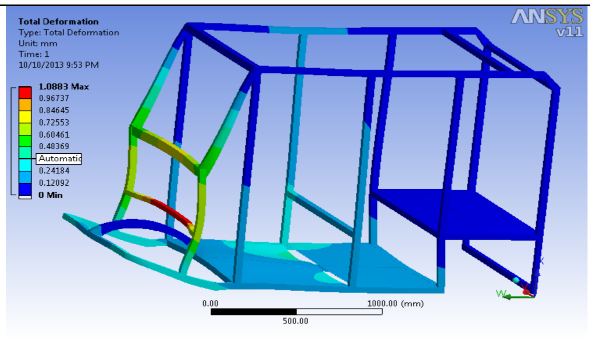
Table 12: front crash results of new model



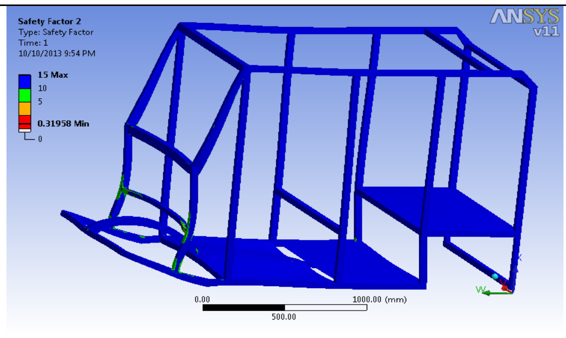
a



b



c



d

Figure 22: shows a) equivalent stress b) maximum shear stress c) total deformation d) safety factor

Deformation results

As we can see in figure 22 (c), the maximum displacement under these conditions is 1.0883 mm, which is totally acceptable. Especially figure shows us the front beams are deflected inward due to the large amount of load applied in case of collision.

4.4.2.2 Rear crash simulation

To do the FEA analysis of the rear impact load applied is 44622.4N at the rear in addition to static loads we have seen before in the static condition analysis. The point of application of these loads will be the actual attachment points between the impact front and rear heads and boundary conditions are: at suspension mounting points Uy = Uz = 0 and front corner points all DOF=0.

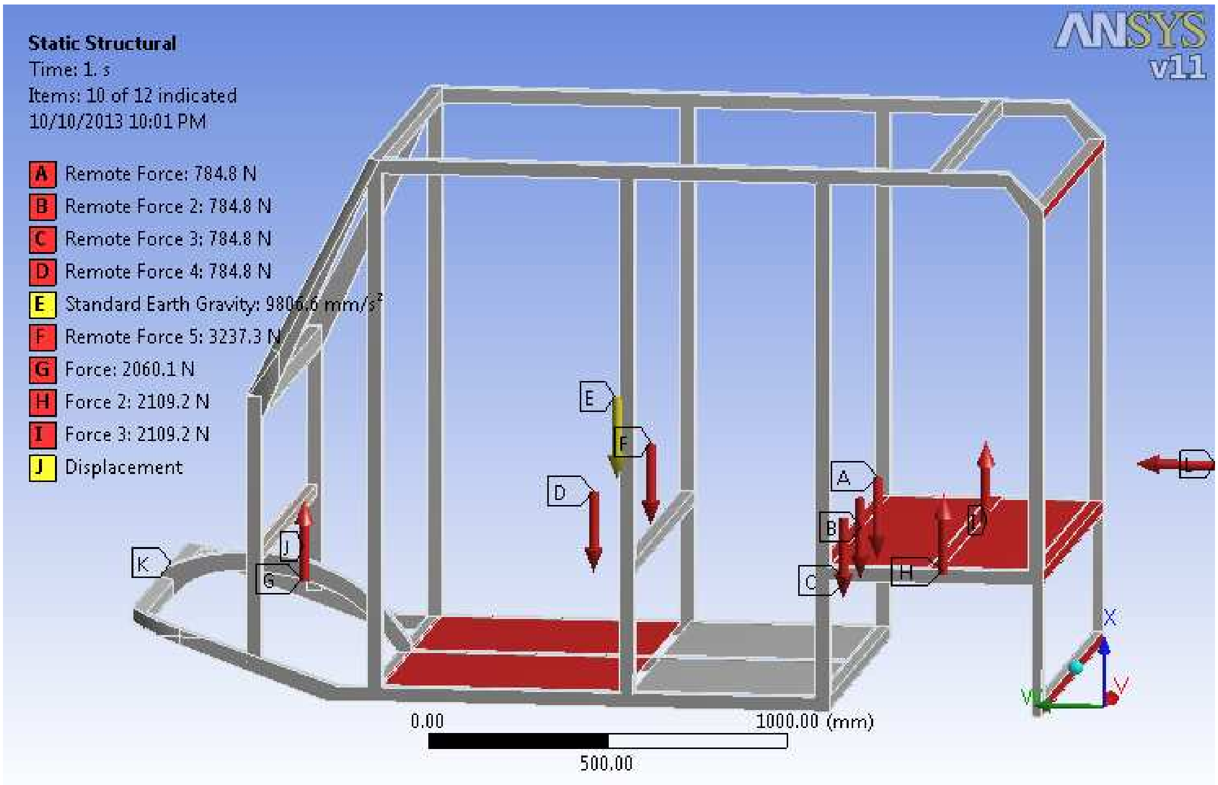


Figure 23: rear crash loads and boundary conditions of new model

Analysis results

The location of maximum Von Misses stress and maximum shear stress are at rear (figure 24). The Von Misses stress magnitude of critical point is 281.03 MPa and the maximum shear stress magnitude is 162.05 MPa. As we can see table 13, the result of the test is that the maximum displacement under these conditions is 2.0697 mm, which is totally acceptable. This result will be done because the frame could resist the load, and giving a high rigidity. Safety factor is 1.0675 hence; the chassis will be safe under rear impact.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	1.287e-003	6.8609e-004	-32.657	1.0675	0	-5.1976e-002
Maximum	281.03	162.05	180.17	-	2.0697	7.1932e-002

Table 13: analysis results of rear crash for new model

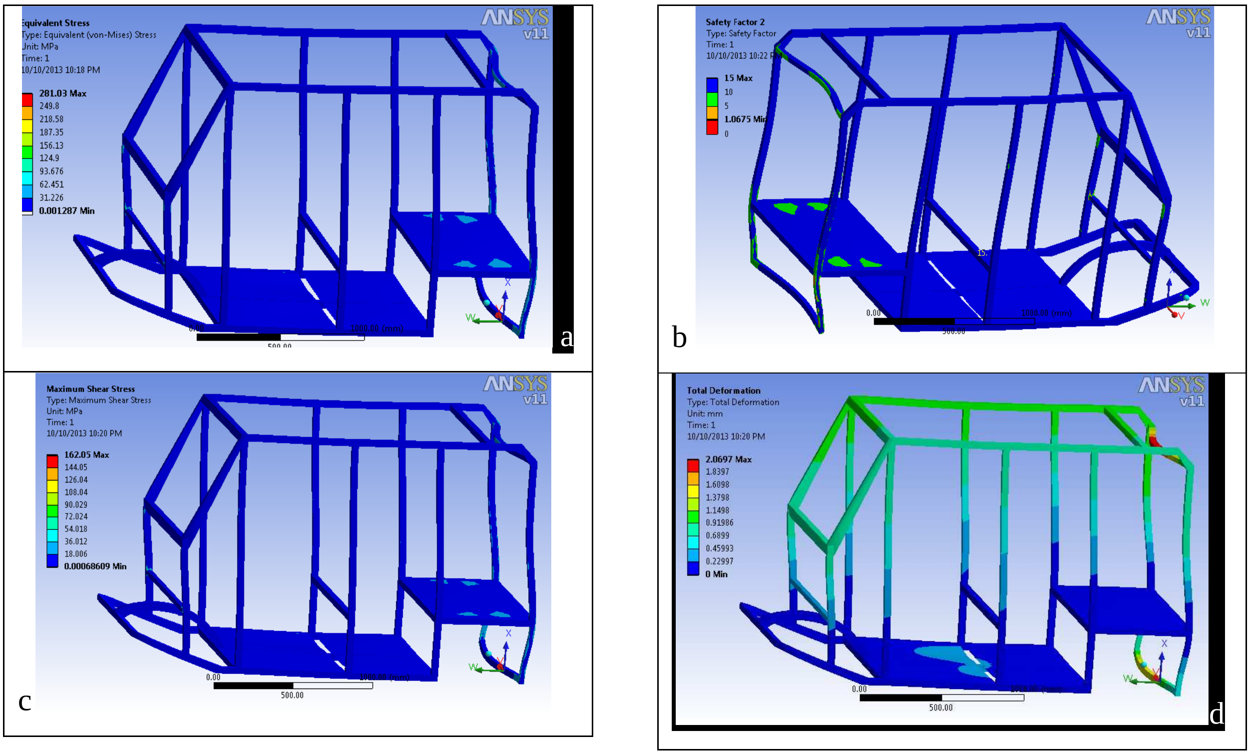


Figure 24: a) equivalent stress b) safety factor c) maximum shear stress d) total deformation of analysis results of rear impact

4.4.2.3 Side impact FEM simulation

Side impact of the vehicle may occur due to crash one vehicle with another vehicle that comes perpendicular to the side of this vehicle. To do the FEA analysis of the side impact, we are going to use the load applied of 44622.4N at the side. The model used is full model of the three wheeled vehicle and boundary conditions at suspension mounting points are Uy = Uz = 0 and at Rear Corner Points All DOF=0.

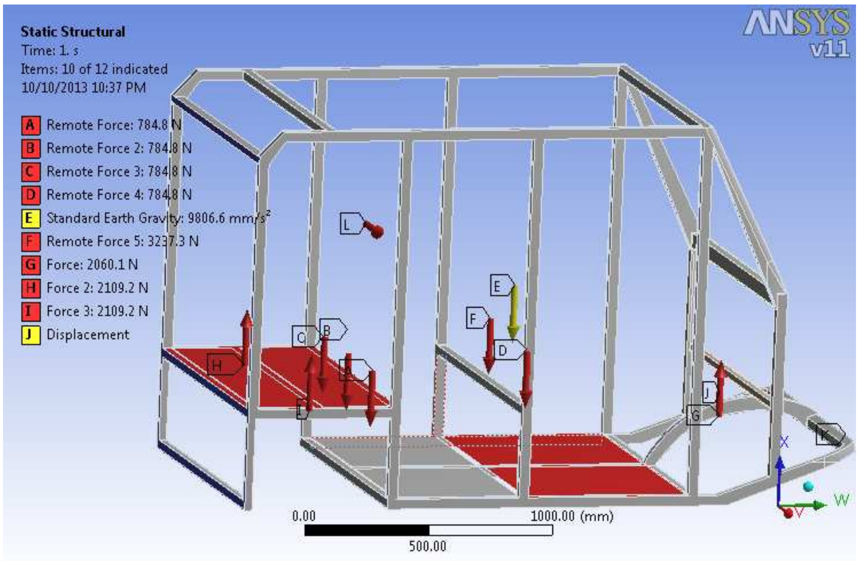


Figure 25: shows loads and boundary conditions of side impact simulation for new model

Analysis results

The location of maximum Von Misses stress and maximum shear stress are at top sides around driver seat (figure 26). The Von Misses stress magnitude of critical point is 853.33 MPa and the maximum shear stress magnitude is 477.63 MPa (table 14). As we can see, the maximum displacement under these conditions is 1.378 mm, which is totally acceptable. This result will be done because the bars are in a correct position, giving a high rigidity to the frame. Safety factor is 0.35156 hence, the Chassis will not safe under side Impact but still better than the existing model.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	1.8342e-005	1.0394e-005	-96.391	0.35156	0	-7.2955e-003
Maximum	853.33	477.63	802.73	-	1.378	1.3761

Table 14: analysis results of side impact

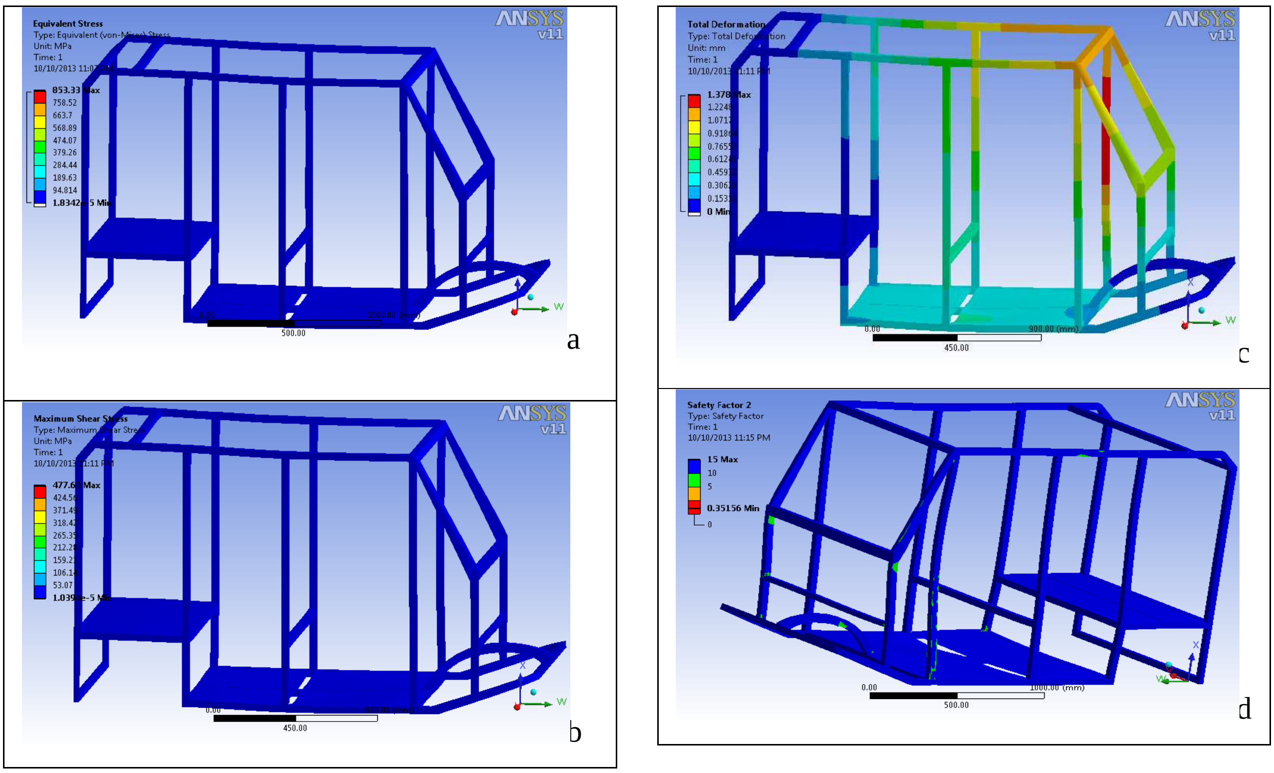


Figure 26: shows a) equivalent stress b) maximum shear stress c) total deformation d) safety factor

4.4.3 Bump analysis of the new model

4.4.3.1 Front wheel bump analysis

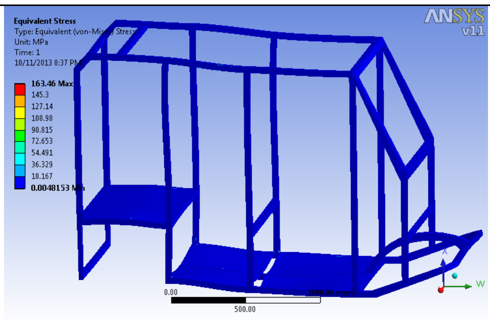
In bump analysis of the new model due to road roughness, loads and boundary conditions applied are similar to the existing model analysis. In this front wheel bump case of analysis load of 6150.4N (Vehicle +Driver + passengers Weight) is applied on vehicle. Boundary conditions are All DOF =0 at rear right wheel and rear left wheel but free at the front wheel attachment point. That means all loads will be transmitted to frame through rear wheels.

Stress results

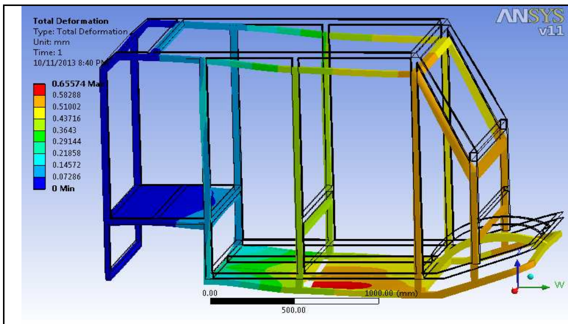
Using the above input and doing the FEM analysis the maximum Von Misses stress magnitude of 163.46 MPa is obtained and its critical point is around front from rear of TWV and the maximum shear stress magnitude is 90.912 MPa (figure 27). The maximum deflection obtained is 0.65574mm. But factor of safety is 1.8353. As we see on the stress plot table 15, the maximum stress is under the material resistance. That is possible by a correct design of the frame. One of the important factors is that the car is not heavy at all. Other factor is that the frame will absorb the great part of the impacts produced by bumps on the road. That provokes that the load that we use to make the test is minimized.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	4.8153e-003	2.7306e-003	-34.398	1.8353	0	-2.5931e-002
Maximum	163.46	90.912	245.74	-	0.65574	1.5974e-002

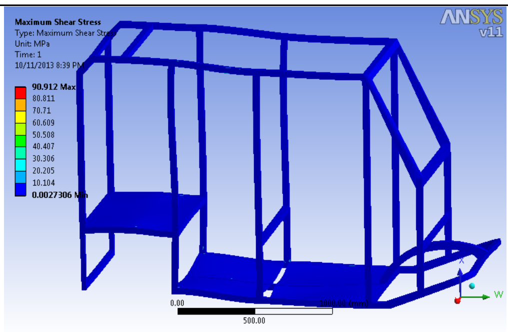
Table 15: front wheel bump simulation results of the new model



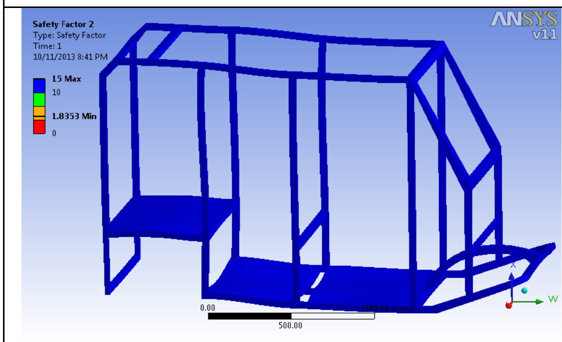
a



c



b



d

Figure 27: a) equivalent stress b) maximum shear stress c) total deformation d) safety factor of front wheel bump simulation results of the new model

4.4.3.2 Rear right Wheel Bump Test

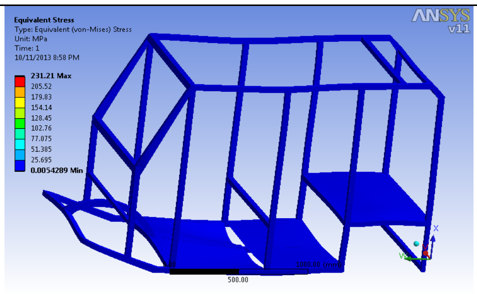
Keeping applied load similar to pervious, the frame will be fixed selecting the joints at the rear left wheel position of the frame and at the front wheel position where the suspension would mount. The maximum allowable deflection and stresses of failure must not occur anywhere in structure.

Stress results

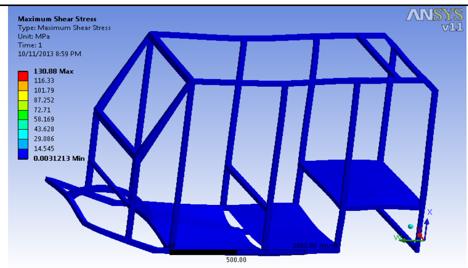
The maximum Von Misses stress magnitude, 231.21 MPa of critical point is around the top rear corner of TWV and the maximum shear stress magnitude is 138.88 MPa on sheet metal 1100mm from rear. The maximum deflection obtained in this case is 0.49403mm (figure 28). But factor of safety is 1.2975 (table 16). As we see on the stress plot, the maximum stress is under the material resistance. That is possible by a correct design. One of the important factors is that the car is very light at all, compared to frame strength. Other factor is that the suspension will absorb the great part of the impacts produced by bumps on the road. That provokes that the load that we use to make the test is minimized.

Result type	Equivalent or von-misses stress (MPa)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Safety factor	Total deformation (mm)	Directional deformation (mm)
Minimum	5.4289e-003	3.1213e-003	-31.531	1.2975	0	-2.9547e-002
Maximum	231.21	130.88	139.91	-	0.49403	1.0546e-002

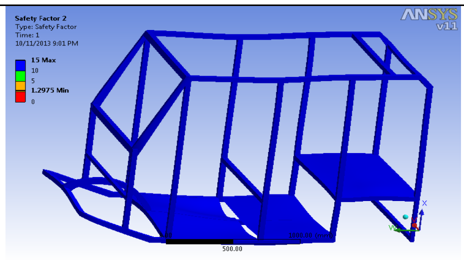
Table 16: rear right wheel bump simulation results of the new model



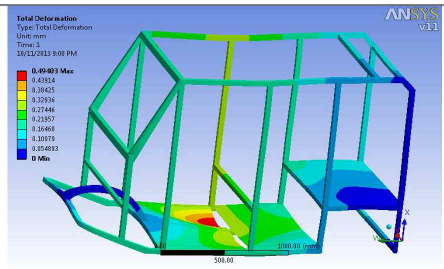
a



b



c



d

Figure 28: shows a) equivalent stress b) maximum shear stress c) safety factor d) total deformation of new model in rear wheel bump case

4.5 Analytical Verification

The following method is used as verification for the method used. The method is verified by using a simply supported overhanging beam of cross section used in the structure. Total load acting on chassis is the sum of capacity of the chassis and Weight of body and engine, which is 640 kg. Chassis has two beams. So load acting on each beam is half of the total load acting on the chassis. That is 320 kg mass.

I. Calculation for reaction

Chassis is simply clamp with Shock Absorber. So Chassis is a Simply Supported Beam with uniformly distributed load. Acting on Entire span of the beam is 320kg (3139.2N). Length of the beam is 2000 mm and uniformly distributed load is 3139.2 / 2625 =1.1959 N/mm.
Side bar of the chassis are made from rectangular channels with 40mm x 40 mm x 3 mm and,

- ✓ Front Overhang (a) = 325 mm
- ✓ Rear Overhang (c) = 300 mm
- ✓ Wheel Base (b) = 2000 mm
- ✓ Total length (l) = 2625 mm

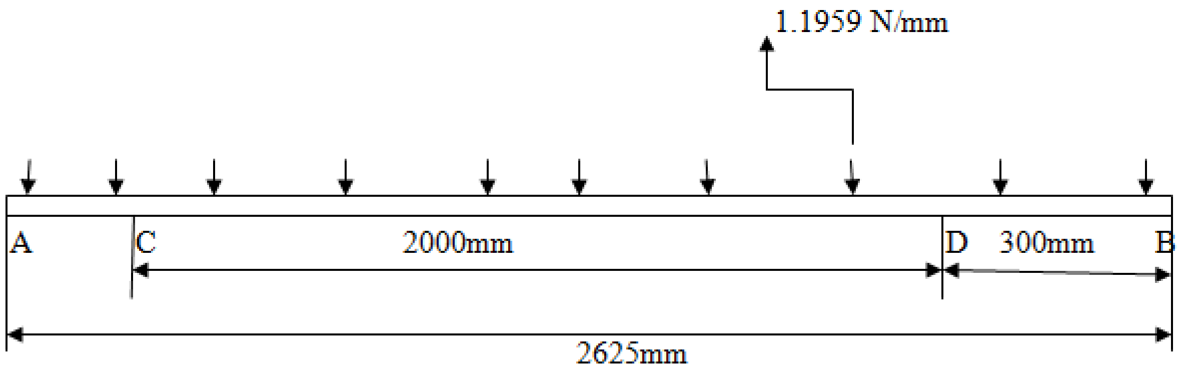


Figure 29: Chassis as a simply supported beam with overhang

Now taking the moment around the support D,

$$Rc = \frac{w \times l \times (l - 2 \times c)}{2 \times b}$$

$$= \frac{1.1959 \text{ N/mm } \times 2625mm \times (2625mm - 2 \times 300mm)}{2 \times 2000mm}$$

$$= 1589.24N$$

And taking the moment around the support C,

$$Rd = \frac{w \times l \times (l - 2 \times a)}{2 \times b}$$

$$= \frac{1.1959 \text{ N/mm } \times 2625mm \times (2625mm - 2 \times 325mm)}{2 \times 2000mm}$$

$$= 1550N$$

II. Calculation for Shear Force and Bending Moment

Shear Force calculation

V_1 = w× a

= 1.1959 N/mm×325 mm

= 388.67 N

V_2 = Rc – V1

= $1589.24N$ −388.67 N

= 1200.57 N

V_3 = Rd – V4

= 1550N – 358.77 N

= 1191.23 N

V_4 = w×c

= 1.1959 N/mm×300mm

= 358.77 N

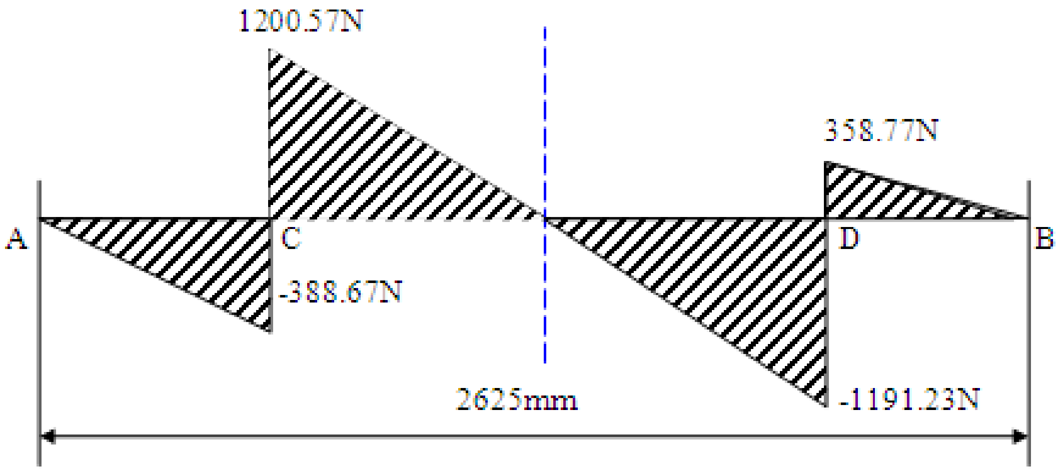


Figure 30: Shear force diagram

Bending Moment

To find bending moment between A & C,

$$M1 = \frac{-w \times a^2}{2}$$

$$= \frac{-1.1959N/mm \times (325mm)^2}{2}$$

$$= -63158.46875Nmm$$

To find bending moment between D & B,

$$M2 = \frac{-w \times c^2}{2}$$

$$= \frac{-1.1959N/mm \times (300mm)^2}{2}$$

$$= -53815.5Nmm$$

To find bending moment between C & D,

$$M3 = Rc \times (\frac{Rc}{2 \times w} - a)$$

$$= 1589.24N \times (\frac{1589.24N}{2 \times 1.1959N/mm} - 325mm)$$

$$= 539473.159Nmm$$

M3 is the maximum bending moment is calculated.

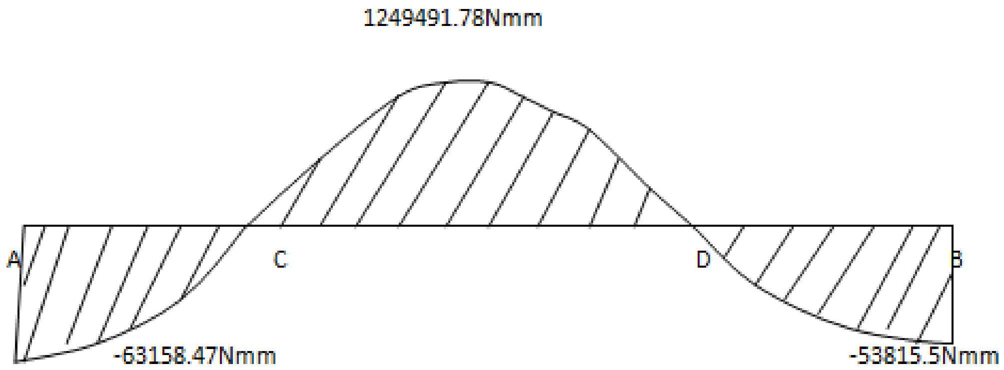


Figure 31: bending moment diagram

III. Stress calculation

The stress is given by

$$б = \pm \frac{M \times y}{Iz}$$

Where M= Bending moment, Iz = second moment of area and Б is the stress at distance y from the N.A. (z-z axis) of the beam cross-section

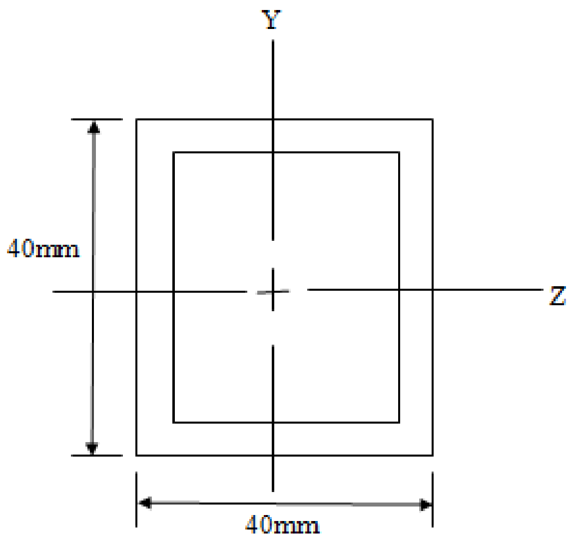


Figure 32: Cross section of the beam used in the verification

The second moment of area is given by

$$Iz = \frac{b \times h^3}{12}$$

$$= \frac{40 \times 40^3}{12} - \frac{34 \times 34^3}{12}$$

$$= 101972mm^4$$

The maximum tensile stresses occur at bottom periphery of the cross section, i.e. y = -20mm whereas the maximum compressive stress occurs at the top periphery of the cross section, i.e. y=20mm.

$$б = \pm \frac{1249491.078Nmm \times 20mm}{101972mm4}$$

$$б = \pm 105.81\text{MPa}$$

The compressive stress occurs at the top side of the cross section and its value is calculated to be -105.81MPa. The tensile stress occurs at the bottom section of the beams cross section and its value is found to be $+105.81$MPa. Now to find the von miss stress (equivalent stress), we should first find the principal stresses.

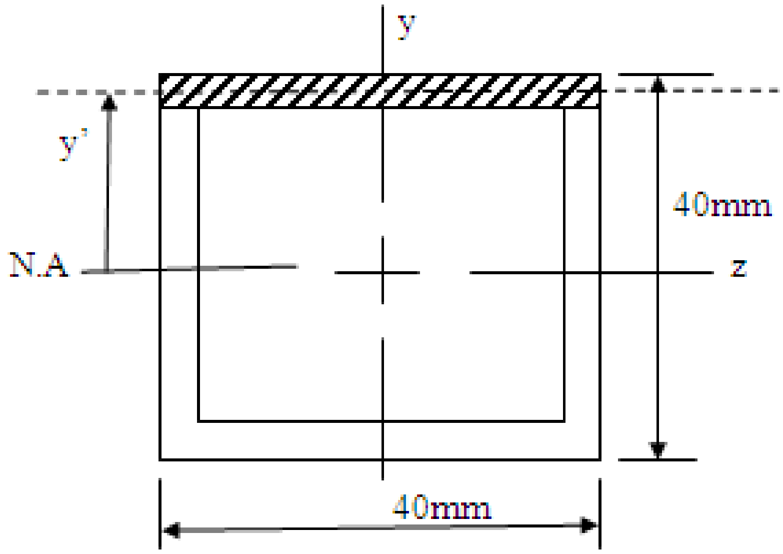


Figure 33: Cross section of beam for shear stress

For the shear stress at section, considering the shaded area above

Q=y'×A'

Where, y' is the distance from the neutral axis to the centroid of the area considered and A' is the area of the considered section.

As we have seen from shear force diagram shear force at the middle between points C and D of the beam becomes zero.

$$\tau_{xy} = \frac{V \times Q}{I}$$

Since shear force has a zero value, the shear stress becomes zero, i.e. $\tau_{xy} = 0$. To find the principal stresses, the following equation is used.

$$\sigma 1, \sigma 2 = \frac{\sigma x + \sigma y}{2} \pm \sqrt{(\frac{\sigma x - \sigma y}{2})^2 + {\tau_{xy}}^2}$$

$$\sigma x = -105.81\text{MPa}, \qquad \sigma y = 0$$

Then, it will give us,

$$\sigma 1 = 0, \qquad \sigma 2 = 0 \; and \; \sigma 3 = -245.06552\text{MPa}$$

$$\sigma_{von-miss} = \sqrt{\frac{(\sigma 1 - \sigma 2)^2 + (\sigma 2 - \sigma 3)^2 + (\sigma 3 - \sigma 1)^2}{2}}$$

$$\sigma_{von-miss} = 105.81\text{MPa}$$

It is the von-miss stress ai the middle point between supports.

IV. Analytical verification for deflection of the beam

Macaulay's method is used to find the bending moment at any general section X-X of the beam. The Macaulay's method helps to find a single expression for bending moment throughout the complete span of the beam. It is used to cover all loading conditions. Finding moment at x—x, taking clockwise direction as positive,

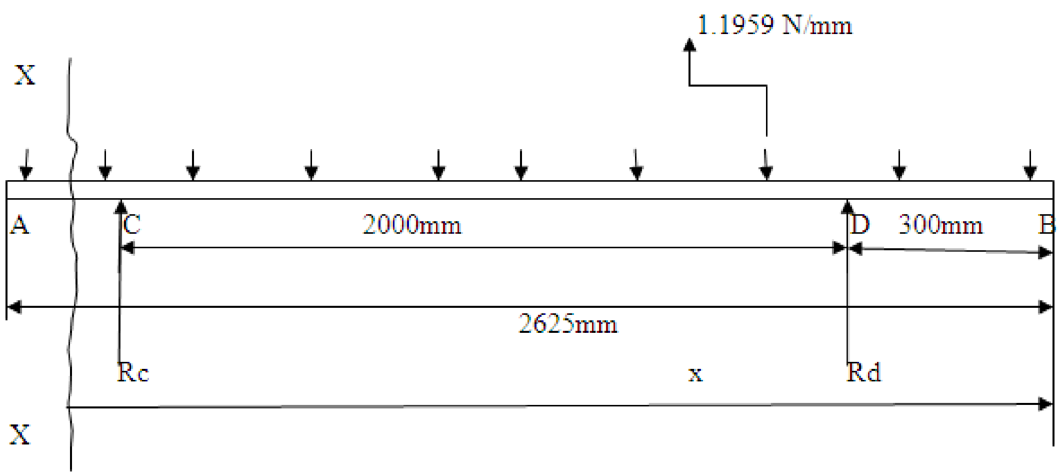


Figure 34: Section X-X of Beam

M_{XX}=-1.1959N/mm× (2625-x)-R_C×(x-2300)-Rd×(x-300) +1.1959N/mm×(x/2)

=-1.1959N/mm× (2625-x) - 1589.24N ×(x-2300)-1550N×(x-300) +1.1959N/mm×(x/2)

=-1.1959N/mm× (2625-x) - 1589.24N ×(x-2300)-1550N×(x-300) +1.1959N/mm×(x/2)

=-3139.2375+1.1959×x-1589.24 ×x+3655252-1550×x+465000+0.59795×x

=4117112-3137 ×x

But the bending moment is given by the following equation

$$M = EI\frac{d^2y}{d^2x}$$

By integrating this equation, slope is found.

$$\theta = \frac{dy}{dx}$$

The second integration of this equation gives deflection (δ) of the beam.

$$\delta = y$$

Therefore, integrating twice the above bending moment equation will provide the following deflection equation along the length of the beam.

$$EI\delta = -522.8x^3 + 2058556x^2 + Ax + B$$

Where, A and B are integrating constants to be solved from the boundary conditions.

When $x = 300, y = 0 \; and \; x = 2300, y = 0$

Therefore, the values of A and B is $A = -21.89 \times 10^8 \quad and \; B = 48.25 \times 10^{10}$

$EI = 200 \times 10^3 \times 101972 = 2039440000Nmm^2$

Substituting these values in the deflection equation, the maximum deflection can be found at the center between supports C and D at x=1325mm.

$$EI\delta max = -522.8 \times (1325)^3 + 2058556 \times (1325)^2 - 21.89 \times 10^8 \times 1325 + 48.25 \times 10^{10}$$

$$EI\delta max = -121.6139 \times 10^{10} + 361.405 \times 10^{10} - 290.0425 \times 10^{10} + 48.25 \times 10^{10}$$

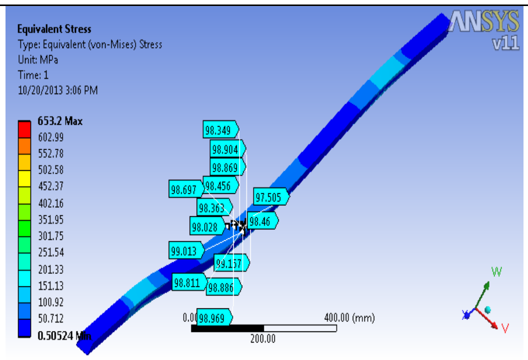
$$EI\delta max = -1.185 \times 10^{10}$$

$$\delta max = -5.81mm$$

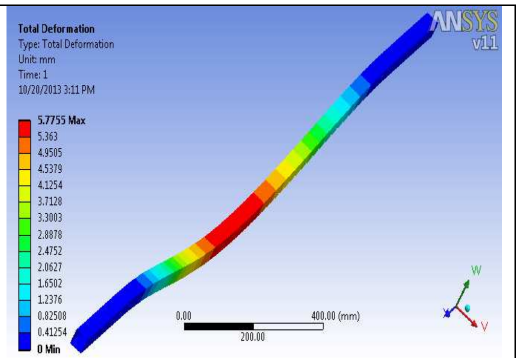
This result indicates that the deformation is 5.81mm downward.

4.6 ANSYS verification

Using this beam of 2625 mm in length and having zero displacement values at its supports attachment points the ANSYS verification is done. The result extracted is the stress and displacement values at the middle point between supports because the analytical verification is done at this point. This happens since we are verifying our analytical results and ANSYS results in comparison. The ANSYS result gives a maximum equivalent stress of 100.92MPa (figure 35).this result is less than the analytical result of 105.81MPa. And the maximum displacement obtained in analytical calculation is 5.81mm but ANSYS gives the result of 5.7755mm (figure 35b).The difference is caused by simplification of model and uncertainties of numerical calculation but values approaches to each other. This implies that our analysis was correct.



a



b

Figure 35: ANSYS verification results

Chapter five

Comparison of the two models

Here we will see the results of the existing model and the new model in comparison. Different cases of the analysis are tabulated and compared in each detail cases and remarks have been given. The remark is given to the new model strength status when compared to the existing model strength.

5.1 In terms of static case results

As we can see the results in table 17 bellow the results of equivalent stress, maximum shear stress and maximum principal stress are bellow the material strength for both existing and new models. Even though the values of the new model stresses are higher than the existing model stress values the safety factor value indicates that the model is safe. This shows that the new model is designed to have high strength without material wastage. But when we see the displacement values the new model is extremely reliable than the existing model.

Analysis type	Existing model	New model	remark
Von-miss stress (MPa)	178.58	221.31	safe
Maximum principal stress(MPa)	124.91	132.97	safe
Maximum shear stress(MPa)	103.08	125.26	safe
Total deformation(mm)	64.393	0.44868	better
Safety factor	1.6799	1.3556	safe

Table 17static case result table

5.2 In terms of impact case

Front impact

Table 18 sows that the new model is much safer than the existing model in the case of front crash. Even if the new model stress level is above the material strength the safety factor is about 30 times of the existing model. When we see the deformation results there is a105mm difference between. This indicates that the new model design using the space frame is advanced one step forward in front crash case.

Analysis type	Existing model	New model	remark
Von-miss stress (MPa)	21369	937.72	better
Maximum principal stress(MPa)	11661	558.16	better
Maximum shear stress(MPa)	12337	541.85	better
Total deformation(mm)	106.92	1.0883	better
Safety factor	1.4039×10^{-2}	0.31958	better

Table 18 front impact result table

Rear impact case

The advancement of the new model in the rear impact case is special and unexpected. All stress results of the new model are under the material resistance. As we remember from the figure of the existing model result the rear part of the model was diminished. But in the new model this case is extremely changed. Even the safety factor becomes above one. From table 19 we can observe the deformation difference that is unbelievable.

Analysis type	Existing model	New model	remark
Von-miss stress (MPa)	19804	281.03	better
Maximum principal stress(MPa)	22060	180.17	better
Maximum shear stress(MPa)	10996	162.05	better
Total deformation(mm)	1436.8	2.0697	better
Safety factor	1.5148×10^{-2}	1.0675	better

Table 19 rear impact result table

Side impact case

This case of the analysis is somewhat different from the previous results. All the stress results of the new model are above the material capacity but some the stress results of the existing model are all under the material resistance value. The safety factors values of both models are bellow one. As we see from table 20 even if the new model is not safe in terms of side impact the deformation value is highly improved.

Analysis type	Existing model	New model	remark
Von-miss stress (MPa)	345.06	853.33	Not better
Maximum principal stress(MPa)	228.73	802.73	Not better
Maximum shear stress(MPa)	197.39	477.63	Not better
Total deformation(mm)	66.116	1.378	better
Safety factor	0.86942	0.35156	Not better

Table 20 side impact result table

5.3 In terms of bump

Front wheel bump case

Table 21 shows the results of the two models in case of front wheel bump. The values indicate that the stress levels for both models are bellowing the material load capacity. But when we see the deformation values there is a earth and sky difference. This is very big and critical change. The safety factor is above one for both cases of models but in the existing model case the value is very close to one and it indicates that the design is not reliable for front wheel bump case.

Analysis type	Existing model	New model	remark
Von-miss stress (MPa)	276.74	163.46	better
Maximum principal stress(MPa)	139.4	245.74	good
Maximum shear stress(MPa)	159.77	90.912	better
Total deformation(mm)	64.421	0.65574	better
Safety factor	1.084	1.8353	better

Table 21front wheel bump result table

Rear wheel bump case

All the stress values and the deformation results of the existing model are higher than the new model results. But the values of both cases are bellowing the martial resistance. When we see the safety factor results the new model has a very good safety factor result. But the existing model safety factor is close to one. It indicates that the new model design is more reliable than the existing model.

Analysis type	Existing model	New model	remark
Von-miss stress (MPa)	295.31	231.21	better
Maximum principal stress(MPa)	164.69	139.91	better
Maximum shear stress(MPa)	170.45	130.88	better
Total deformation(mm)	64.449	0.49403	better
Safety factor	1.0159	1.2975	better

Table 22 rear wheel bump case

Chapter six

Cost estimation

Cost analysis is carried in order to estimate the cost of the new three wheeled vehicle model. The analysis is based on the current cost of the materials in the market. Figure 36 shows the numbers given to each parts of the space frame to identify each other. Members have the same cross section but they are different in their length. Each member has a rectangular hallow cross section of $40 \times 40 \times 3mm$.

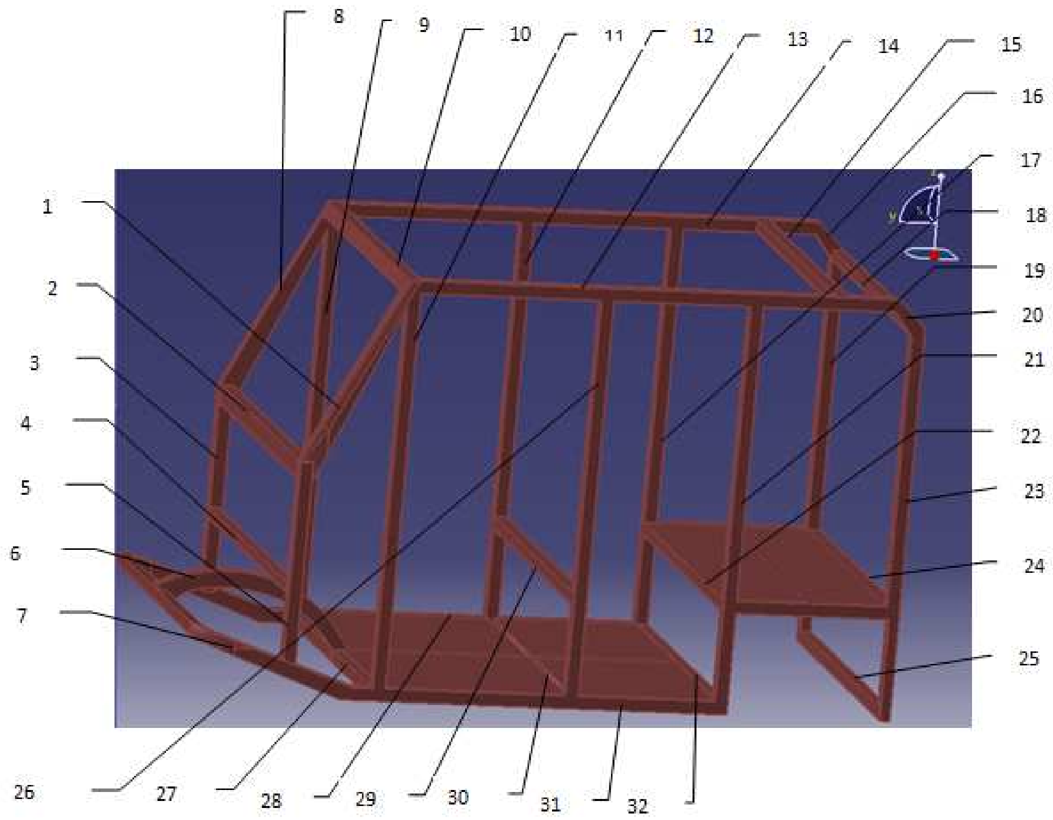


Figure 36: identifications number for each member

In order to estimate cost, we need to find the total length of beam used in the three wheeled vehicle structure. To do this we have make a table by grouping members with the same length in one column. Table 23 bellow shows grouped members and their total length.

Group number	Identification numbers	Unit length(mm)	Number of members	Total length(mm)
1	2,4,10,15,18,22,24,25,27,29,30,32	1220	12	14640
2	9,11,12,17,21,26	1530	6	9180
3	1,8	410	2	820
4	3,5	380	2	760
5	6	725	1	725
6	7	2040	1	2040
7	28,31	1420	2	2840
8	13,14	1840	2	3680
9	19,23	1430	2	2860
10	16,20	140	2	280
The sum of total length				37825mm

Table 23: cost analysis table

From the above table the total lengths of the rectangular hallow cross section with $40 \times 40 \times 3mm$ required is 37825mm. The current cost of the steel with the above cross section obtained from market is 80 Birr per meter. Now multiplying the total lengths of the steel required with its cost per meter the total cost of the new model frame can be obtained.

$$total\ cost\ eastimated = 37.825m \times 80Birr/m$$
$$= 3026\ Birr$$

This cost is so small comparing to the current cost of TWV in the market, which is around 85 thousand birr. Even if this cost of the new model does not include the cost of the engine, the axle, wheels and electrical systems it will obviously decrease the manufacturing cost and simplify the level of technology requirement.

Chapter seven

Conclusions and Recommendations

7.1 Conclusion

This report thoroughly dealt with various load analysis on chassis and optimization has been achieved by changing the chassis type and by increasing its resistance to loads. The chassis and body frame was designed so as the vehicle can withstand better crash and static types of loads and is capable of moving on bumpy road. I also believe that the design is quite successful and have the opportunity to design a vehicle with these characteristics is very important. The frame has been improved by optimizing the number of bars and tubes that are part of it and repeating over and over FEA analyses. In this way could reduce the weight of our vehicle and increase its strength. One of the main improvements to be made in such designs would be to build the frame in such as low cost, easy to manufacture and highly strong. An attractive option is to be performed using the space frame. That greatly facilitates the assembly and may even lead us to manufacture in our country, which represents a major advance to Ethiopia. In this thesis we have seen that the space frame is a better option in terms of strength and stiffness and cost for the TWV frame and body design. In the existing model all the total displacement values obtained were above 60mm. bit in the new model designed using space frame the maximum displacement obtained was 2.07mm. This indicates that the stiffness value has been improved. Also in terms of the crash analysis the maximum von-misses stress obtained in the existing model was about 70 times larger than the results obtained in the new model. This shows that the new model can withstand impact loads so as the risk level has been minimized.

7.2 Recommendation

As a recommendation I put here my feelings to Ethiopian investors, researchers, vehicle body manufacturing companies and to the government also to invest their money and knowledge on the sector of three wheeled vehicle to minimize the loss of our currency.

7.4 Future works

- The dynamic response of the model should be performed in the future.
- The sheet metal which is covered on the structure should be included in the finite element analysis in order to find a more refined result.
- Further redesign of the TWV structure considering individual members is recommended.
- Prototype development and test should be performed.
- Stability analysis should be prformed.

References

1. R. Koona, M.K. Naidu , and B. Satyanarayana, Dynamic Behavior of Three-wheeled Vehicle Under Random Road Characteristics Using FEM, SAE Technical paper series , Andhra Pradesh, India,2004.

2. Formula Student DIT Team, frame and body design, Institution of Mechanical Engineers (I. Mech. E), paper submitted for competition, Europe, 1999.

3. Pakistan Standards and Quality Control Authority (PSQCA) Standards Development Centre, Pakistan standard specification for three wheeler auto vehicles (SDC), paper for standards and rules, Pakistan, 2005.

4. Jornsen Reimpell, Helmut Stoll, and Jurgen W. Betzler, The Automotive Chassis: Engineering Principles, second edition, book translated from the German by AGET Limited, 2001.

5. Thomas Gyllendahl, and David Tran, Development of an auto rickshaw vehicle suspension, research paper, India, February, 2012.

6. Alan Babu, Akhil Gupta, Prateek Bansal, Ankur Wadhwa, Sahil Arora, Raghib raja, Ankit Garg, Shubham Garg, Sagar Gupta and MD. Areeb, Hybrid human powered electric vehicle, Jamia Millia Islamia University, Delhi, India, 2013.

7. David Botana Olañeta, Part Realization Process from CAD Models to Physical Parts, paper for Master Thesis, Lulea, May 2012.

8. Anindya Deb, transforming a lecture-oriented course on design of automotive systems into a unique design experience, International Conference on Engineering Education, Manchester, U.K., 2002.

9. M.A.Saeedi, Stability of Three-Wheeled Vehicles with and without Control System, research paper, Tehran, Iran, March 2013.

10. Besufekad Getachew, Modeling and Dynamic Analysis of Bus Body Built in Ethiopia, thesis paper, Addis Ababa, Ethiopia, February 2012.

11. Ashutosh Dubey and Vivek Dwivedi, Vehicle Chassis Analysis: Load Cases & Boundary Conditions for Stress Analysis, journal article, 2001.

12. B.L. Boada, A. Gauchia, M.J.L. Boada, V. Diaz, A genetic-based optimization of a bus structure as a design methodology. Paper presented at the 12th IFToMM World Congress, Besancon, France, 2007.

13. Johanssan & S, Eslund, Optimization of Vehicle Dynamics in Truck by use of Full Vehicle FE Models, journal article, IMechE.- C466/016/93, pp 181-193,1993 .

14. A. Van Berkum, Chassis and suspension design FSRTE02, Master's Thesis, Eindhoven, March 2006.

15. Jacob I. Salter, Design, Analysis and Manufacture of 2011 REV Formula SAE Vehicle Chassis, thesis paper, The University of Western Australia, October17, 2011.

16. Christopher Daven Larsen, Comparison of Three Degree of Freedom and Si x Degree of Freedom Motion Bases Utilizing Classical Washout Algorithms, Graduate Thesis and Dissertations. Paper 10139, Iowa State University,Ames, Lowa,2011.

17. Md. Shahidul Islam, Zaheed Bin Rahman, and Nafis Ahmad, Designing Solar Three-Wheeler for Disable People, International journal o f Scientific & Engineering Research Volume 3, Issue1, January 2012.

18. Sairam Kotari, and V.Gopinath, static and dynamic analysis on tatra chassis, International Journal of Modern Engineering Research (IJMER) Vol.2, Issue.1, pp-086-094.

19. Kassahun Mekonnen, static and dynamic analysis of a commercial vehicle with van body, Thesis paper, Addis Ababa University, Ethiopia, May 2008.