TRIBHUVAN UNIVERSITY

INSTITUTE OF ENGINEERING

THAPATHALI CAMPUS

"DESIGN, ANALYSIS AND EVALUATION OF AN ELECTRIC ALL TERRAIN VEHICLE"

SUBMITTED BY

Gagan Acharya (073/BAME/611)
Prakash Koirala (073/BAME/626)
Purna Khatri (073/BAME/629)
Rahul Jamkatel (073/BAME/630)
Sandeep Khanal (073/BAME/634)
Saurab Ghimire (073/BAME/636)
Subin Tandukar (073/BAME/641)
Susmita Parajuli (073/BAME/645)

A PROJECT REPORT

SUBMITTED TO THE DEPARTMENT OF AUTOMOBILE AND MECHANICAL ENGINEERING

IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE

DEGREE OF BACHELOR IN AUTOMOBILE ENGINEERING

DEPARTMENT OF AUTOMOBILE AND MECHANICAL ENGINEERING

KATHMANDU, NEPAL

March 2021

Head

Department of Automobile and Mechanical Engineering

Thapathali Campus, Institute of Engineering

Kathmandu, Nepal

TRIBHUVAN UNIVERSITY

INSTITUTE OF ENGINEERING

THAPATHALI CAMPUS

DEPARTMENT OF AUTOMOBILE AND MECHANICAL ENGINEERING

The undersigned certify that they have read, and recommended to the Institute of Engineering for acceptance, a project report entitled "DESIGN, ANALYSIS AND EVALUATION OF AN EATV" submitted by Gagan Acharya, Prakash Koirala, Purna Khatri, Rahul Jamkatel, Sandeep Khanal, Saurab Ghimire, Subin Tandukar and Susmita Parajuli in partial fulfillment of the requirements for the Bachelor in Automobile and Mechanical Engineering.

Supervisor, Er. Subodh Kumar Ghimire
Lecturer

Supervisor, Er. Sunil Adhikari
Lecturer
Department of Automobile and Mechanical Engineering
Thapathali Campus, Institute Of Engineering

External Examiner, Er. Dipesh Poudel
Training Manager
Sipradi Trading Pvt. Ltd.
Naikap, Kathmandu

External Examiner, Dr. Krishna Pd. Shrestha
Assistant Professor
Department of Mechanical Engineering
Kathmandu University

Er. Subodh Kumar Ghimire
Head of Department
Department of Automobile and Mechanical Engineering
Thapathali Campus, Institute Of Engineering

Date: ………………………

ACKNOWLEDGEMENT

We are indebted to the Department of Automobile and Mechanical Engineering, Thapathali Campus for providing us the opportunity to undertake this project. We are grateful to our department head Er. Subodh Kumar Ghimire for all support and guidelines he had provided during the entire project duration.

We are also thankful to our supervisors Er. Sunil Adhikari and Er. Subodh Kumar Ghimire for supervising and guiding us throughout the project. They have played the role of guardian for helping us to carry out the project.

We are grateful to Er. Chiranjivi Dahal, Er. Mahesh Neupane, Er. Manoj Adhikari for their valuable suggestions during the project duration in simulation and report making process.

We would like to thank our friends of Automobile 4th year 073 batch of Thapathali campus their support, coordination, time, motivation and assistance directing this project to this day.

Finally, we thank anyone associated with this project directly or indirectly for their efforts on bringing this project towards completion.

ABSTRACT

The source for the petroleum product has been declining day by day. To recover from the possible condition of energy scarcity many alternative resources are being developed nowadays, among which electricity is regarded as primary alternative. Since the automobile are taken as the primary resource of the petroleum product consumer, many electric vehicle models are being developed and manufactured in these days.

Almost every electric vehicle that is being manufactured in present scenario is either luxury or heavy vehicles that cost a consumer a lot of investment even for manufacture. So the project EATV is the concept of the developing a light weight vehicle that can be used for short runs in local area for the transportation of the person or it can be used as the attraction point for the tourist. The use of the EATV can be made on various conservational area where the topography of the land is not similar and use of the IC engine vehicle causing the pollution and noises for the animals living in that area. For such use the projected manufacturing cost of the EATV is way below the normal electric vehicle cost and this will also promote the use of local resource as for the manufacturing and maintaining the vehicle while reducing the usage of petroleum product which is imported from other nations and are in the verge of ending the resource.

At first, our project was mainly focused on fabrication. But, considering the impact of the COVID-19 pandemic, the fabrication seemed to be difficult and inconvenient. In order to fulfill our requirement for Bachelor's in Automobile Engineering, we modified our project from fabrication to computer modeling various systems in MATLAB, Simulink and ANSYS to find ideal requirement of parts involved.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIST OF ACRONYMS AND SYMBOLS

ATV	:	All-Terrain Vehicle
ATT	:	All-Terrain Tire
AGM	:	Absorbent Glass Mat
AISI	:	American Iron and Steel Institute
BMS	:	Battery Management System
BLDC	:	Brushless Direct Current
BMD	:	Bending Moment Diagram
CAD	:	Computer Aided Design
EATV	:	Electric All-Terrain Vehicle
FOS	:	Factor of Safety
FWD	:	Front Wheel Drive
GPa	:	Giga-Pascal (10^9 Pa)
HEV	:	Hybrid Electric Vehicle
HID	:	High-intensity discharge lamps
IRJET	:	International Research Journal of Engineering and Technology
Kg	:	Kilogram
KPI	:	King Pin Inclination
LCO	:	Lead Cobalt Oxide
LED	:	light-emitting diode
MATLAB	:	Matrix Laboratory
mm	:	Millimeter

MPa	:	Mega-Pascal (10^6 Pa)
NHTSA	:	National Highway Traffic Safety Administration
N	:	Newton
NCA	:	Nickel Cobalt Aluminum
NCM	:	Nickel Cobalt Manganese
PMSM	:	Permanent Magnet Synchronous Motor
PM	:	Permanent Magnet
Pa	:	Pascal
SFD	:	Shear Force Diagram
SAE	:	Society of Automotive Engineers
SRM	:	Switched Reluctance Motor
SOC	:	State of Charge
UTV	:	Utility-Terrain Vehicle
UN	:	United Nations
US	:	United States

CHAPTER ONE
INTRODUCTION

1.1 Overview

According to the American National Standards Institute, an ATV is a vehicle that runs on low pressure tires, with operator straddled seats and steering wheel handlebars. As the name suggests, it is designed to climb a broader range of terrain than most of the other vehicles available commercially. According to the American National Standards Institute, an ATV is a vehicle that runs on low pressure tires, with operator straddled seats and steering wheel handlebars. As the name suggests, it is designed to climb a broader range of terrain than most of the other vehicles available commercially.

Electric all-terrain vehicles (EATV) are small motor vehicles with three, four or six wheels for use on various types of terrain. They are originally planned for off-road use. The first three wheeled all-terrain vehicles (ATV) was the Sperry-Rand Tri-cart. It was designed in 1967 as a graduate project of John Plessinger at the Cranbrook Academy of Arts near Detroit. They travel on low –pressure tires, with a seat that is straddled by the operator, along with handlebars for steering control. All-terrains are first began as farm equipment and were awkward, laughable, and mostly operated by men. Today, EATVs have had their look, shape, and size changed and has different names such as quad, three-wheeler, and four-track, a four-wheeler, or quad cycle. They are also used as sports equipment in team sports or individual sports, outdoor addicts like off-road touring, hiking, crossing beaches, crossing snow, etc. EATVs are best suited for quick turns. They operate well in tight woods, and are great for situations that call for quickly hopping on and off the vehicle or hauling small cargo loads.

The global electric ATV and UTV market accounted for $468.9 million in 2019 and is expected to reach $4.30 billion by 2030. The market is anticipated to grow at a Compound annual growth rate (CAGR) of 23.37% during the forecast period 2020 to 2030. The market growth is mainly attributed to the rising number of government policies for electric vehicles, better availability of charging infrastructure, and increasing need to minimize the level of carbon dioxide emissions. In addition, electric ATVs and UTVs have low cost of ownership and low noise emissions, which are expected to drive the market growth during the forecast period. (BIS Research, 2020)

1.2 Background

All-terrain vehicles are firstly developed for farm equipment then they are planned mainly for off-road use. The active development in the field of ATV in recent years has led to its increase in versatility in terms of its functionality and range of application .Early ATVs development is generally divided into two separate categories on their purpose: off-road four wheelers and street four wheelers. As the name implies off-road four-wheeler is meant to be used in off road terrain only. ATVs are also used in agriculture to bridge the advantages of trucks and tractors. The ATV is more physically demanding to ride, and the rider must use balance to manipulate and control the vehicle. They are small enough to really manhandle around. They are not only useful for various types of terrains but also in sports.

ATVs are best suited for making quick turns. They are highly capable, can tow heavy loads and can also be ridden on the trails, and modified for high performance riding. They are mostly used to get sportsmen out to their hunting land, or plowing snow. The ATV was very suitable for the project as it had all-wheel drive through each of its six wheels. This enabled the vehicle to navigate rough terrain with a minimal risk of getting stuck. Maneuvering the vehicle was effected through a skid steering system.

1.3 Purpose of EATV

The EATV is planned to be capable to use in most of the place in Nepal. The EATV is mostly planned to be used as the tourist attraction parameter for tourist areas. For example the quad-bikes used in the Chitlang, Makwanpur District becoming the attraction point for the local tourist. With acting as the attraction for the tourist destination, the EATV can be used in the world heritage sites and many more sites and museums like Lumbini and National park like Chitwan National Park, Bardiya National Park for safari. The use of EATV in National park can be increased to security personal for patrolling in the area also as the EATV is designed to be environment friendly and can travel in multiple terrain like swamps, broken wooden logs etc. The use of EATV can be made for the local travel also by foe people who live in off-road conditions.

1.4 Objectives

1.4.1 Main Objective:

The main objective of the project is to design roll-cage as well as model and simulate the systems of Electric All-Terrain Vehicle (EATV).

1.4.2 Specific Objective

The main objective as stated above, following specific objectives will be obtained:

- To design the roll-cage frame considering various static and impact loading.
- To study and analyze the performance of the vehicle in various road condition using computer model.
- To design the weld and find welding factor of safety.
- To find the appropriate power requirement of electric motors and battery.
- To design, model and simulate the suspension system.
- To design and analyze the performance of coil spring.
- To determine the criteria to select the appropriate tire suitable for EATV.

1.4.3 Scope of the work

Considering the impact of COVID-19, the fabrication phase of the electric all-terrain vehicle (EATV) was difficult and inconvenient. The budget for EATV cannot be met in this crisis. In order to fulfill our requirements for the final year course in Bachelors in Automobile Engineering, we will perform the modeling, simulation and analysis of electric all-terrain vehicle. The design conditions and requirements will be based upon the SAE India BAJA ATV Rule-Book.

The project include conducting model design for the following systems of EATV.

i) Power requirement modeling
ii) Suspension modeling

We made use of the design and simulation software to perform model design and simulation of the systems. There are many modeling and simulation software present in the market. We intend to make use of software like ANSYS, MATLAB.

MATLAB is a proprietary multi-paradigm programming language and numerical computing environment developed by Math Works. MATLAB allows matrix manipulations, plotting of functions and data, implementation of algorithms, creation of user interfaces, and interfacing with programs written in other languages.

Simulink is a MATLAB-based graphical programming environment for modeling, simulating and analyzing multi-domain dynamical systems. Its primary interface is a graphical block diagramming tool and a customizable set of block libraries. Simulink is widely used in automatic control and digital signal processing for multi-domain simulation and model-based design.

The benefits of using modeling and simulation software are of following reasons.

a. Performing equation based modeling approach.
b. Simulating various power train configurations and determine if design goals are achieved.
c. Simulation is faster and cost effective.
d. Performance can be analyzed in advance.

CHAPTER TWO
LITERATURE REVIEW

2.1 Reviews of Previous Reports and Works Published

To design an EATV roll cage, we must understand the various aspects of the EATV like about its working, components needed in it and the way of doing it. For this knowledge we need to have knowledge about the amount of research or work done on it previously. To get the knowledge of the process of designing the roll cage and other work or development made on the all-terrain vehicle we have gone through following article or reports.

a. "Optimization of Chassis of an All-Terrain Vehicle", by Junaid Mohammed Farooq, studied chassis of ATV. The paper deals with design of chassis frame for an All-Terrain Vehicle and it's Optimization. Various loading tests like Front Impact, Rear Impact, Side Impact, and Roll over test etc. have been conducted on the chassis and the design has been optimized by reducing the weight of the chassis.

b. Rahul Dev Gupta, Rakesh Kumar Phanden and other in their paper "Design and Development for Roll Cage of All-Terrain Vehicle" studied aims to give an introduction to the material selection procedure, pipe size selection and various tests that need to be done before finalizing the design. Various factors such as impact force determination, loading points, the mesh size dependence of generated stress, Von-Misses Stress, Deformation and Factor of Safety (FOS) are studied.

c. "Design & Manufacturing of All Terrain Vehicle (ATV) - Selection, Modification, Static & Dynamic Analysis of ATV Vehicle" in the journal presented by Upendra S. Gupta, Sumit Chandak, Devashish Dixit provided detail description of the design considerations, static & dynamic analysis and mathematical data involved in the design of an ATV Vehicle.

d. "Modal and Static Analysis of a Standard All-Terrain Vehicle Chassis" by Shaik khajamoinuddin, B.Balaji did chassis analysis using software and comparing it with theoretical calculations.

e. Denish S. Mevawala et. al. in their paper “Stress Analysis of Roll Cage for an All-Terrain Vehicle” discussed about the roll cage of ATV. The paper deals with design of roll cage for an ATV and Various loading tests like Front Impact, Side Impact and rear impact have been conducted. The modeling and stress analysis is done by ANSYS software.

f. The paper “Simulation of Roll cage of an All-Terrain Vehicle considering inertia, using Transient Multi-body analysis” by A.V.S. Abhinav have developed an approximate model using different elements in ANSYS APDL and a transient multi-body analysis is carried out to check for failure in the roll cage due to inertia of major components. The paper thus provides the method to study the effect of inertia of individual components on the roll cage during an impact and helps in designing a safer vehicle.

"Design of an Electric Powertrain for an All-Terrain Vehicle" by Gabriel Vicente Cerqueira Ribeiro this dissertation developed and implemented of a fully electric functional traction system for an All-Terrain Vehicle (ATV). Each of the subsystems went through a phase of research in the state of the art, followed by a simulation experiment, in several different software (MATLAB®, Simulink ®, MotorSolve ®, Inventor , and Solidworks). In this thesis, a control algorithm typically used in high efficiency SRM drives is tested and implemented. All of the circuits needed for testing the electrical machine are operational. (Riberio, 2016)

"Modeling and torque control for a 4-wheel- drive electric vehicle" by Carlos Montero, David Macros and et.al presented A dynamic model based on the connection of the mechanical devices in the vehicle is implemented as a CAD model and as a simulator using Simmechanics in the MATLAB/Simulink framework. This model is validated using data from real experiments and it can be used to test new torque controllers before their implementation. (Carlos Montero, 2015)

"A Four-Wheel-Drive Fully Electric Vehicle Layout with Two- Speed Transmissions" by De Pinto S., Camocardi P.and et al. This paper presents a novel fully electric vehicle layout, consisting of two drivetrains, each of them including atwo-speed transmission, for improving vehicle acceleration and gradeability performance. The adoption of two-speed transmissions allows eight different gear state combinations, increasing the possibility of selecting a high-efficiency state for each operating condition. A torque-fill controller is developed for the compensation of the torque gap during gearshift through the variation of the torque on the other axle. Finally, a simulation-based analysis of the acceleration, gradeability and gearshift performance of this vehicle layout is discussed, in comparison with alternative fully electric vehicle configurations (De Pinto S., 2014).

"Drive Selection and Performance Evaluation of Electric and Hybrid Electric Vehicles" by Ms. Vaishali Bakshi, Prof. Mrs. V.S. Jape. In this paper, EV and HEV models are analyzed for their performances by considering the cases of Permanent magnet Synchronous Motor (PMSM) and Induction Motor (IM) drivetrains. First, the EV and HEV models are developed in MATLAB/SIMULINK environment along with PMSM and Induction motor drives in different cases. The motor controllers and supervisory controllers are appropriately tuned to achieve the required speed-torque demands and vehicle performance. The results of simulation of EV and HEV under the two case examples of motor types are compared to analyze the relative performance in terms of efficiency, response and fulfilling load demands. (Ms. Vaishali Bakshi, 2017)

“Optimization of Chassis of an All-Terrain Vehicle” by Junaid Mohammed Farooq, A.S.N. Saitejaetal studied chassis of ATV. The paper deals with design of chassis frame for an All-Terrain Vehicle and its Optimization. Various loading tests like Front Impact, Rear Impact, Side Impact, and Roll over test etc have been conducted on the chassis and the design has been optimized by reducing the weight of the chassis.

“Modal and Static Analysis of a Standard All-Terrain Vehicle Chassis” by Shaik khaja moinuddin, B.Balaji did chassis analysis using software and comparing it with theoretical calculations.

“Design & Manufacturing of All Terrain Vehicle (ATV) - Selection, Modification, Static & Dynamic Analysis of ATV Vehicle” in the journal presented by Upendra S. Gupta, Sumit Chandak, Devashish Dixit provided detail description of the design considerations, static & dynamic analysis and mathematical data involved in the design of a ATV Vehicle.

"Comparison Of Characteristics Of Various Motor drives Currently Used In Electric Vehicle Propulsion System" paper presented by Gagandeep Luthra in the International

Journal of Mechanical And Production Engineering, ISSN: 2320-2092, Volume- 5, Issue-6, June 2017 compared different types and the characteristics of electric motor drives used in the electric vehicle. The conclusion drawn was that the most widely accepted as well as used Electric Motor Drive is the Induction Motor Drive and Permanent Magnet (PM) Brushless Motor Drive. The Induction Motor Drive is the most cost effective Motor Drive. The most Energy Efficient is the Permanent Magnet (PM) Brushless Motor Drive. Most mature technology is of the D.C Motor Drive as a lot of research work and development has been done on them over the past. Most reliable are ones being the Induction Motor Drive and the Switch Reluctance Motor Drive. (Luthra, 2017)

2.2 Engineering Design

The engineering design process is a common series of steps that engineers use in creating functional products and processes. The process is highly iterative - parts of the process often need to be repeated many times before another can be entered - though the part(s) that get iterated and the number of such cycles in any given project may vary. Engineering design has many steps that should be followed in order to get the final object or component as per requirement.

In mechanical design, almost every new design is inspired by some existing design or the theories. In other word, a new design is not entirely new. These new design will either be inspired by previously existing design, in terms of the aesthetic or the shape, and change in the properties of the material used for better performance. For example the change of the shape for better handling and operational activities of any previously object is also a type of design while the change of the material without altering the size and shape of the item same for better durability is also a design.

2.2.1 Steps of Engineering Design

In general, the design of the new item is the result of the need or the crisis caused by the existing design. To provide the better alternative of the existing design or to develop the new design to solve the crisis, the design engineer must go through a set of steps or process of the design for better outcome. The steps of the engineering design are briefly discussed below:

a) Research
b) Problem Statement

c) Feasibility
d) Concept Generation
e) Preliminary design
f) Detailed design
g) Production planning

a. Research

While designing the object or item huge amount of the time is invested in the research and locating information. If the designing process is being done to enhance the existing items, then the reverse engineering is termed as the proper way of gathering information. In this step of the designing process, it is necessary to gather all relevant data and information related to the item that is being designed. Consideration should be given to the existing applicable literature, problems and successes associated with existing solutions, costs, and marketplace needs.

b. Problem Statement

Defining the problem is one of the most important elements in the design process and this task is often performed at the same time as a feasibility analysis. The problem statement is important to make the control of the designing process. This step includes the functions, attributes, specifications of the user. Some of the problem statement includes the hardware and software parameter, availability, maintainability and testability.

c. Feasibility

The feasibility study is an evaluation and analysis of the potential of a proposed project to support the process of decision making. It outlines and analyses alternatives or methods of achieving the desired outcome. The feasibility study helps to narrow the scope of the project to identify the best scenario.

The purpose of a feasibility assessment is to determine whether the engineer's project can proceed into the design phase. This is based on two criteria: the project needs to be based on an achievable idea, and it needs to be within cost constraints.

d. Concept Generation

A concept study (conceptualization, conceptual design) is often a phase of project planning that includes producing ideas and taking into account the pros and cons of implementing those ideas. This stage of a project is done to minimize the likelihood of error, manage costs, assess risks, and evaluate the potential success of the intended project. In any event, once an engineering issue or problem is defined, potential solutions must be identified. These solutions can be found by using ideation, the mental process by which ideas are generated. Various generated ideas must then undergo a concept evaluation step, which utilizes various tools to compare and contrast the relative strengths and weakness of possible alternatives.

e. Preliminary design

The preliminary design, or high-level design includes (also called FEED or Basic design), often bridges a gap between design conception and detailed design, particularly in cases where the level of conceptualization achieved during ideation is not sufficient for full evaluation. So in this task, the overall system configuration is defined, and schematics, diagrams, and layouts of the project may provide early project configuration. During detailed design and optimization, the parameters of the part being created will change, but the preliminary design focuses on creating the general framework to build the project on.

f. Detailed design

Following preliminary design is the Detailed Design (Detailed Engineering) phase, which may consist of procurement of materials as well. This phase further elaborates each aspect of the project/product by complete description through solid modeling, drawings as well as specifications.

Computer-aided design (CAD) programs have made the detailed design phase more efficient. For example, a CAD program can provide optimization to reduce volume without hindering a part's quality. It can also calculate stress and displacement using the finite element method to determine stresses throughout the part.

g. Production planning

The production planning and tool design consists of planning how to mass-produce the product and which tools should be used in the manufacturing process. Tasks to complete in this step include selecting materials, selection of the production processes, determination of the sequence of operations, and selection of tools such as jigs, fixtures, metal cutting and metal or plastics forming tools. This task also involves additional prototype testing iterations to ensure the mass-produced version meets qualification testing standards

2.3 Introduction to Automobile Frame

Frame of automobile is the rigid support that holds the whole body and other accessories required for the smooth operation of the vehicle. At past or in initial conditions a separate frame was used as the base to which other components are attached to it. In present scenario for small vehicles or light vehicles a unibody are mostly used. But for all the buses and trucks still uses separate frame development process to which other body parts are attached.

The frame of an automobile is like the skeleton of the human body that gives strength to the whole structure and gives housing to all the parts that are necessary.

2.3.1 Construction of the frame

Generally the frame should be able to hold all the components that are fastened to it for the operation along with the all dynamic loads. So the frames are made of rigid materials like steels. The profile section of the material that is being used for the frame constructions are as below:

a) Channel-section

The channel section had been popular type of design at the starting period as it has strength and has easy manufacturability. For making of these section a thick metal plate was taken, thickness ranging from 1/36 inch to ½ inch regarding the weight of the vehicle. (Kenworth Truck Company, 2012)

This type of frame is better for long member of frames and can absorb bending forces.

b) Box type

Originally, boxed frames were made by welding two matching C-rails together to form a rectangular tube. Modern techniques, however, use a process similar to making C-rails in that a piece of steel is bent into four sides and then welded where both ends meet.

This type is used more often as short members and is better for bending and torsional load.

c) Tubular type

This is modern profile or section that is being used in three wheeler and motorcycle and scooters. These are good at absorbing torsional load.

2.3.2 Function of frame

The major function of the frame is to support the vehicle load and prevent the deflection when sudden impacts. The main functions of a frame in motor vehicles are:

a) To support the vehicle's mechanical components and body
b) To deal with static and dynamic loads, without undue deflection or distortion.

These include:

i. Weight of the body, passengers, and cargo loads.
ii. Vertical and torsional twisting transmitted by going over uneven surfaces.
iii. Transverse lateral forces caused by road conditions, side wind, and steering the vehicle.
iv. Torque from the engine and transmission.
v. Longitudinal tensile forces from starting and acceleration, as well as compression from braking.
vi. Sudden impacts from collisions.

2.3.3 Classification of frame

The frame could be classified on majorly three categories as:

a) Conventional

Conventional frame is used to be popular for the starting phase of the automobiles. In these types of the frames the frame is constructed using two long member and 5-6 short members attached in between long members.

b) Integral

Integral types are popular nowadays in cars and light vehicles. In this type of frame, all the vehicle parts are attached to the body, so it is also called as unibody structure. Since long members are removed, these are light in weight and cheaper but since all are attached in one body, maintenance is difficult.

c) Semi-integral

This type of the frame is mixed form of the above where half of the frame is fixed. Generally front part including gear box, engine, and front suspension are covered this type of frame is popular in American and European vehicle.

2.3.4 Types of frame

The automobile frame is classified in different categories on the basis of the layout of the members attached. The major popular types of the frames are as below:

a) Ladder Frame

The ladder frame is the simplest and oldest of all designs. It consists of two symmetrical rails, or beams, and cross member connecting them. Originally seen on almost all vehicles, the ladder frame was gradually phased out on cars around the 1940s and is now seen mainly on trucks. This design offers good beam resistance because of its continuous rails from front to rear, but poor resistance to torsion. Also, the vehicle's overall height will be higher due to the floor pan sitting above the frame instead of inside it.

b) Backbone Frame

Backbone frame is a type of an automobile construction frame that is similar to the body-on frame design. Instead of a two-dimensional ladder type structure, it consists of a strong tubular backbone (usually rectangular in cross section) that connects the front and rear suspension attachment areas. A body is then placed on this structure.

c) X-frame

This is the design used for the full-size American models of General Motors. In which the rails from alongside the engine seemed to cross in the passenger compartment, each continuing to the opposite end of the cross member at the extreme rear of the vehicle.

It was specifically chosen to decrease the overall height of the vehicles, and to increase in the space for transmission. The X-frame was claimed to improve on previous designs, but it lacked side rails and thus did not provide adequate side impact and collision protection. So this design was replaced by perimeter frames.

d) Perimeter Frame

Similar to a ladder frame, but the middle sections of the frame rails sit outboard of the front and rear rails. This was done to allow for a lower floor pan, and therefore lower overall vehicle in passenger cars. In addition to the perimeter frame allows lower seating positions when that is desirable, and offers better safety in the event of a side impact. However, the design lacks stiffness, because the transition areas from front to center and center to rear reduce beam and torsional resistance.

e) Platform Frame

This is a modification of the perimeter frame in which the passenger compartment floor and often the luggage compartment floor were permanently attached to the frame, for extra strength. Neither floor pieces were sheet metal straight off the roll, but had been stamped with ridges and hollows for extra strength.

This was used by the Germans on the Volkswagen Beetle and the Mercedes-Benz "Ponton" cars of the 1950s and 1960s, where it was called in English-language advertisements as the "frame floor".

f) Unibody (or) Unit body

In a unibody (also unit body, unitary construction, or unitized construction) design, the frame and body are constructed as a single unit. This became the preferred construction for mass market automobiles and crossovers especially in the wake of the two energy crises of the 1970s and the mid-2000s oil price increases.

g) Sub Frame

A sub-frame is a structural component of a vehicle. Such as an automobile or an aircraft, that uses a separate structure within a larger body-on-frame or unit body to carry certain components, such as the engine, drivetrain, or suspension. The sub frame is bolted and/or welded to the vehicle.

The principal purposes of using a sub-frame are, to spread high chassis loads over a wide area of relatively thin sheet metal of a monocoque body shell, and to isolate vibration and harshness from the rest of the body.

2.3.5 Load supported by frame

The major function of the frame is to support the load of the vehicle. So the frame should be rigid enough to tackle the entire load. The various loads that act on the vehicle are as below:

a) Short duration Load – While crossing a broken patch.
b) Momentary duration Load – While taking a curve.
c) Impact Loads – Due to the collision of the vehicle.
d) Inertia Load – While applying brakes.
e) Static Loads – Loads due to chassis parts.
f) Over Loads – Beyond Design capacity

2.3.6 Difference between frame and chassis

A frame is the skeleton of a car without the mountings, were as a chassis is a mounted frame.

Simply speaking a frame supports the components added to the vehicle and gives the vehicle more rigidity i.e., it consists of only the chassis without any extra components such as engine, drivetrain etc. ,But, the chassis (rolling chassis) consists of the frame plus the "running gear" like engine, transmission, drive shaft, differential, and suspension.

2.4 Welding

Welding is a fabrication process whereby two or more parts are fused together by means of heat, pressure or both forming a join as the parts cool. Welding is usually used on metals and thermoplastics. The completed welded joint may be referred to as a weldment. The parts that are joined are known as a parent material. The material added to help form the join is called filler or consumable. The form of these materials may see them referred to as parent plate or pipe, filler wire, consumable electrode (for arc welding), etc. Consumables are usually chosen to be similar in composition to the parent material, thus forming a homogenous weld, but there are occasions, such as when welding brittle cast irons, when filler with a very different composition and, therefore, properties is used. These welds are called heterogeneous.

Welding is a high heat process which melts the base material. Typically with the addition of filler material heat at a high temperature causes a weld pool of molten material which cools to form the join, which can be stronger than the parent metal. Pressure can also be used to produce a weld, either alongside the heat or by itself. It can also use a shielding gas to protect the melted and filler metals from becoming contaminated or oxidized. (TWI Ltd., n.d.)

2.4.1 Types of Welding Process

There are a variety of different processes with their own techniques and applications for industry, these include:

1. **Arc**

This category includes a number of common manual, semi-automatic and automatic processes. These include metal inert gas (MIG) welding, stick welding, tungsten inert gas (TIG) welding also known as gas tungsten arc welding (GTAW), gas welding, metal active gas (MAG) welding, flux cored arc welding (FCAW), gas metal arc welding (GMAW), submerged arc welding (SAW), shielded metal arc welding (SMAW) and plasma arc welding.

These techniques usually use a filler material and are primarily used for joining metals including stainless steel, aluminum, nickel and copper alloys, cobalt and titanium. Arc welding processes are widely used across industries such as oil and gas, power, aerospace, automotive, and more.

2. **Friction**

Friction welding techniques join materials using mechanical friction. This can be performed in a variety of ways on different welding materials including steel, aluminum or even wood.

The mechanical friction generates heat which softens the materials which mix to create a bond as they cool. The manner in which the joining occurs is dependent on the exact process used, for example, friction stir welding (FSW), friction stir spot welding (FSSW), linear friction welding (LFW) and rotary friction welding (RFW).

Friction welding doesn't require the use of filler metals, flux or shielding gas. Friction is frequently used in aerospace applications as it is ideal for joining otherwise 'non-weldable' light-weight aluminium alloys. Friction processes are used across industry and are also being explored as a method to bond wood without the use of adhesives or nails.

3. Electron Beam

This fusion joining process uses a beam of high velocity electrons to join materials. The kinetic energy of the electrons transforms into heat upon impact with the work pieces causing the materials to melt together. Electron beam welding (EBW) is performed in a vacuum (with the use of a vacuum chamber) to prevent the beam from dissipating.

There are many common applications for EBW, as can be used to join thick sections. This means it can be applied across a number of industries from aerospace to nuclear power and automotive to rail.

4. Laser

Used to join thermoplastics or pieces of metal, this process uses a laser to provide a concentrated heat ideal for barrow, deep welds and high joining rates. Being easily automated, the high welding speed at which this process can be performed makes it perfect for high volume applications, such as within the automotive industry. Laser beam welding can be performed in air rather than in a vacuum such as with electron beam joining.

5. Resistance

This is a fast process which is commonly used in the automotive industry. This process can be split into two types, resistance spot welding and resistance seam welding.

Spot welding uses heat delivered between two electrodes which are applied to a small area as the work pieces are clamped together.

Seam welding is similar to spot welding except it replaces the electrodes with rotating wheels to deliver a continuous leak-free weld.

2.4.2 Common Joint Configurations

1. Butt Joint

A connection between the ends or edges of two parts making an angle to one another of 135-180° inclusive in the region of the joint.

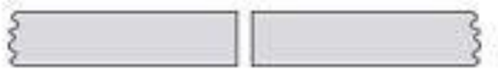


Figure 2-1
Butt Joint

2. T Joint

A connection between the end or edge of one part and the face of the other part, the parts making an angle to one another of more than 5 up to and including 90° in the region of the joint.

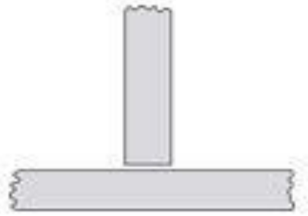


Figure 2-2
T-joint

3. Corner Joint

A connection between the ends or edges of two parts making an angle to one another of more than 30 but less than 135° in the region of the joint.

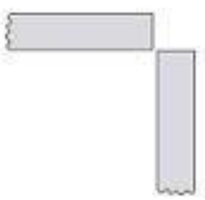


Figure 2-3
Corner joint

4. Edge Joint

A connection between the edges of two parts making an angle to one another of 0 to 30° inclusive in the region of the joint.

Figure 2-4
Edge Joint

5. Cruciform Joint

A connection in which two flat plates or two bars are welded to another flat plate at right angles and on the same axis.

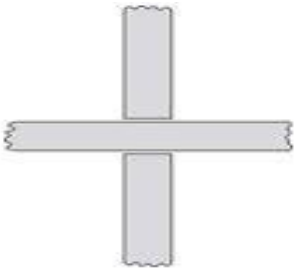


Figure 2-5
Cruciform Joint

6. Lap Joint

A connection between two overlapping parts making an angle to one another of 0-5° inclusive in the region of the weld or welds.

Figure 2-6
Lap Joint

2.5 Introduction to Tires

Tires are the solid support that holds the whole vehicle weight. It is the medium by which the vehicle makes constant ground contact. In general the function of the tires is to support and balance the vehicle weight in upright position and give the linear motion to the vehicle with the rotatory motion of the vehicle itself. In other word tire can be described as the cushion to the vehicle.

A rubber cushion that usually contains compressed air and fits around a wheel (as of an automobile) (Merriam-Webster. (n.d.)., n.d.).

Along with the function of making constant ground contact and making the movement of the vehicle possible in linear motion, the tires also absorbs certain amount of the shock provided by the road conditions. The compressed air is used in the tires which act as the primary shock absorber for the vehicle.

2.5.1 Important Features of tires:

Tire is used as one of the major important aspect of the vehicle. Tire is essential for the vehicle movement. For the use of the tires for the longer period and n a safer way various aspects of the tires should be taken care of during the selecting of the tire. Some of the features or the consideration points for the selection of the tires are briefly described below:

a) Tire tread:

Tread refers to the pattern that has been made around the surface of the tire face or width. The tread is very much essential for the tractive and braking function of the tires. For the urban drive a regular smooth tread pattern can be used while for the rough road or off-road drive it is important to go with the deep and aggressive looking tread pattern for the better grip to the roads.

b) Sidewall:

Sidewall of the tire refers to the part between the wheel and the tread. For the sports car the slick profile for racing which looks super cool but these sidewalls will not be suitable for the off-road drive. So we have to look for the condition where the tire is to be used.

c) Rating:

Rating is the representation of size, load capacity, speed limit, and type of the tire. Rating generally shows the features or the characteristics of the tire. Rating shows either the tire could be used for the road conditions like firmer tire are more effective for the on-road or smooth road and for the off-road condition smooth tire are preferred, if both are altered then it affects the fuel economy and the life of the tires.

2.5.2 Tire rating:

Rating refers to the describing various parameter to define tire performance, capacity and characteristics. The tire rating is indication that is embedded on the tire sidewalls which shows the types of the ply used in the tire, diameter of the rim, weight, speed properties or limitations.

An example of the decoding the tire rating for rating

P215/65R15 89H,

can be decoded as

P	=	Tire class (P indicates passenger car, LT indicates light truck)
215	=	nominal width of tire in millimeter
65	=	Ratio of height to width (aspect ratio)
R	=	Radial ply tire
15	=	Rim diameter code
89	=	Load index
H	=	Speed symbol.

These are the basic rating symbol indicated on the tire. Along with these rating index, other symbols are used in the tire sidewalls to show other parameter like type pressure, tube or tubeless ire etc.

2.5.3 Types of Tires

The tires available on the market can be classified on the various groups on the basis of their purpose or the ply used. Some categorizations of the tire are briefly described below:

a. **On the basis of the ply used:**

A tire is made up of various layers of fabrics, those fabric are called as plies. The plies support and hold the firm nature of the tires. Polyester cord is basically popular or most used fabrics for making of the plies. On the basis of how the plies are place in layer they are classified as:

i) Radial ply

The cord plies are arranged at 90 degrees to the direction of travel, or radially (from the center of the tire). This assures less heat buildup and a softer ride. Moreover, radial tires are manufactured with the plies laid radially which results in a more flexible tire wall. Some key features of Radial ply tires are as below:

- flexible sidewalls
- low rolling resistance
- less vibration
- extended life

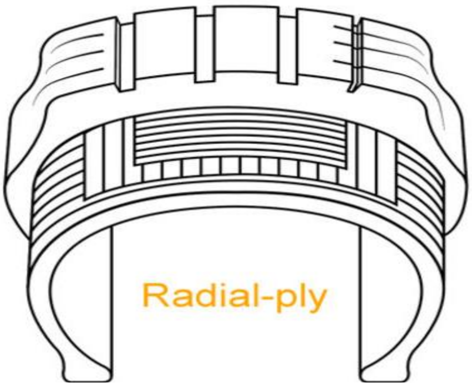


Figure 2-7 Cross-Section of tire showing radial ply (source: tyreplex)

ii) Bias ply

Bias ply, also known as cross-ply, tires are manufactured with the plies laid out diagonally, which causes the walls to not be as flexible. Bias ply tires have stiffer sidewalls, so if your rig tends to sway, they may help reduce this problem. They also have advantages for carrying heavy loads. Some key features of Bias tires are as below:

- Stiff in nature
- Can bear heavy loads

Figure 2-8 Cross-Section of tire showing cross- ply (source: tyreplex)

b. On the basis of use:

The conventional tire is needed to change for every season. That means the tire suitable for dry road is not suitable to use on wet road. So, on the purpose of use of the weather the tires are classified as below:

i) Summer tire

Summer tires are designed to offer high levels of performance and are optimized to cope in temperatures above 7°C. They have a softer rubber compound to enhance stability and grip and they sport a tread pattern that provides resistance against aquaplaning.

ii) Winter tire

Winter tires have been designed with a large number of grooves and sipes to offer greater traction and grip on snowy, icy and wet surfaces. They are optimized to remain flexible when temperatures drop to provide safety.

c. On the basis of tube

Generally there are two categories of tires on basis of tube used, one is tubed and another is tubeless.

i) Tubed

This is the traditional types of tire in which a rubber tube is used inside the tire. Air is filled at high pressure in the tube, not on the tires so that the leakage can be protected. The inflated inner tube provides the structural support and cushioning. As the tube is not attached to the tire, the tires are comparatively lighter. Some features of these tires are as below:

- It is cheaper
- Puncture of the tire is more often
- Easy set-up

ii) Tubeless

When a tube is attached or combined to the tires inner wall rather than using external tubes for the better performance then it is called tubeless. Air is filled directly to the tires after fitted in the rim using the sealants. The sealant is the special type of liquid used inside the tubeless tires to prevent the leakage of the air through small punctures. The key features of the tubeless tires are as below:

- It is expensive
- Small puncture can be sealed by the sealant, so less chance of going flat.
- Little complex to install
- Frequent change of sealant is necessary.

2.5.4 All terrain tires

All terrain tires are that tire which can be used irrespective of the terrain or road condition. These types of the tire are suitable to road conditions like urban or smooth road, rough or off road, muddy road, wet roads, and steep roads. They combine the open-tread design of off-road tires with the good handling of street tires. It's important to remember that as this type of tires is all-purpose. So, it's not the best option for people who drive only on highways and paved roads or only off-road.

All-terrain tires show an adequate performance with a bit more noise and a little more time to perform the turning and braking. They are suppler when it comes to minor sharp impacts. However, on continuous bumps and multiple impacts, all-terrains start to jitter more than well-controlled highway all-seasons. Fuel efficiency is also less in AT tires

All-terrains will provide you with sufficient grip and performance on snowy and icy roads. However, in temperatures colder than 7-10 °C, the tire's performance may become worse due to the compounds that suppose warmer weather use. The braking tests on snowy surfaces from 40 mph showed AT tires need 56 meters to stop. (Utires, 2017)

2.5.5 Pros and cons of all terrain tires

The use of All-terrain tire (ATT) has various pros and cons. Some of the pros of using the all-terrain tires are as below:

a) Open tread design, which improves the traction of the tire on the off-road conditions.
b) Reinforced sidewalls present on some models of all-terrain tires, providing more load carrying capacity. These are usually more aggressive tires made for heavier trucks and more off-road use.
c) As all-terrain tires are considered all-purpose, they also provide traction on snowy and icy surfaces. If regular adequate performance is enough for your car, you don't have to change from summer to winter tires.

Some Cons of using the ATT are as below:

a) The tires are noisier than regular all-season tires due to the tread design
b) The softer rubber of the tires means a shorter tread life, though it isn't lower than the average of 50,000 km. Such compounds help all-terrain tires provide adequate performance on all surfaces.

All-terrain tires rank in the middle in terms of fuel efficiency, as regular street tires use less fuel, and off-road tires use much more. Mechanical friction, wind and rolling resistance, and tread patterns all influence the economy.

2.6 Accessories

The vehicle should be moving at various condition, lie in day light, in foggy condition and in night time also. To operate in this condition vehicle needs to use various types of the lights for easy visibility and to eliminate the confusion on the crossover roads. The use of headlight is essential for driving at night while fog light is needed to minimize accident risk at foggy weather condition. Turn signals lights or indicator lights are necessary to show the direction that the vehicle will be following from the crossover. Tail lights are needed to indicate the vehicles operations and presence to the vehicle coming from the behind of the vehicle. The necessary lighting system and accessories with circuit diagram are discussed below.

2.6.1 Headlamp

Headlamp, also called as headlight, is a lamp or light source that has been placed in front of the vehicle to illuminate the road at the night road conditions. Headlamp is the device or the set of the light source and reflector, while headlight is the beam of the light that is produced by the headlamp and distributed to the road for illumination. For the source of the light the vehicle can use various types of the bulb such as:

a. Tungsten bulb
b. LED bulbs
c. High-intensity discharge lamps (HID)
d. Laser lights

To use headlight on road, operators are provided with the dimmer switch so that they can choose between dim light and high beam where dim light is directed downward and in some countries directed to the driving lane i.e. in left lane driving system, light is directed in left side and downward so that traffic coming from right side does not affected from the light and light is directed to right bottom side so that traffic coming from left side does not affected. Standard halogen replacement bulbs are generally 55 watts for most vehicles while HID bulbs run at around 35 watts. If headlamp of more power is used than these, more brightness may be obtained but that will hamper the traffic coming from opposite side.

The basic symbol for low beam light and high beam light are shown as below

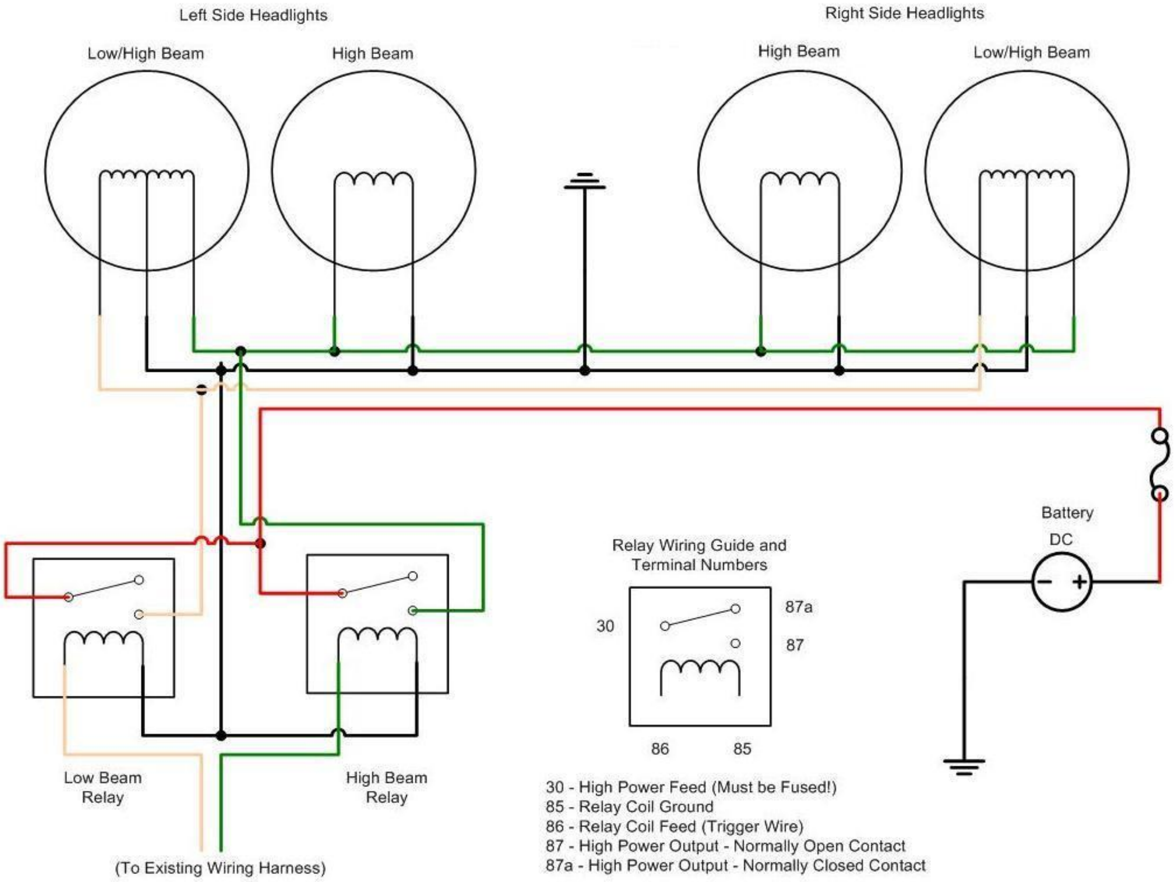


Figure 2-9 Circuit diagram of Head light (source: rowand)

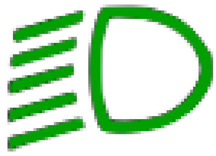


Figure 2-10 Symbol for Low beam light

Figure 2-11 Symbol for High beam light

The circuit diagram for the dual headlamp with dimmer switch and relay control is described briefly with the help of the basic diagram in the above figure.

In general, the power from the battery is always keep supplied to the relays. Relays are switches that open and close circuits electromechanically or electronically. Relays control one electrical circuit by opening and closing contacts in another circuit. When the headlight switch is operated then it activates the relay connected to either high beam relay or low beam relay depending upon the position of the dimmer switch.

When the relay operates, it passes the power or current from the battery to the bulbs. When, the low beam relay is operated, the high beam relay stays on OFF position, so that current is passed to only low beam lamps. When dimmer switch is operated to high beam, the current passing through low beam relay cuts off and current then passes through high beam relay and lamps. When headlights are not necessary, the headlight switch is turned off, which disconnects both relays and no current flows to lamps.

2.6.2 Turn indicator

Turn indicator often, called a turn signal, is a flashing light that is used in the vehicle to indicate the direction that the vehicle is intending to move. mounted near the left and right front and rear corners of a vehicle, and sometimes on the sides or on the side mirrors of a vehicle, activated by the driver on one side of the vehicle at a time to advertise intent to turn or change lanes towards that side. In general the turn signal is yellow in color. 21 watt bulb will be used as the turn indicator at four positions, at right and left of front and rear side of the vehicle. The international signal light symbol is as:

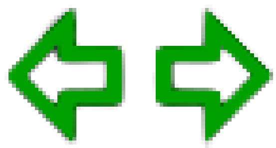


Figure 2-12 Symbol of Turn signal

In turn indicator, the flasher or blinker plays a vital role to make the signal light blinking so that it is easily noticeable. Some of the flasher also provided with the sound alert system. The basic flasher circuit is shown below:

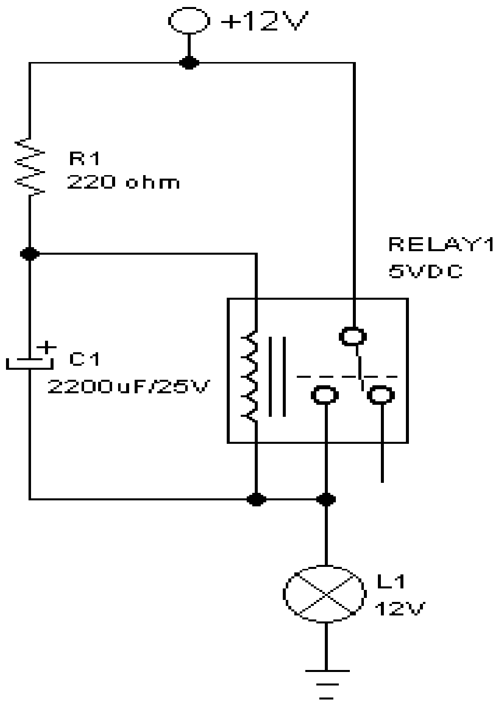


Figure 2-13 Basic Flasher circuit (Source: Deeptronic)

The flasher relay works on principles of electro-mechanics to appropriately power turn and hazard signals on most automobiles manufactured since the late 1930s. When the circuits is supplied with the rated voltage (6 or 12 volts) the switch, which is normally closed, is made to turn on using the electro-magnetic force created by the coils. This turning causes the current to pass through the bulb rather than the coil due to less resistance, then the electromagnetic force disappears and the circuit is returned to closed position. This causes the electricity to pass from the coil and again magnetizing the circuit and operating switch. This process is repeated over the time until the current is supplied to flasher unit and blinking output is obtained when connected in designed circuits. The basic circuit of the turn signal is shown below:

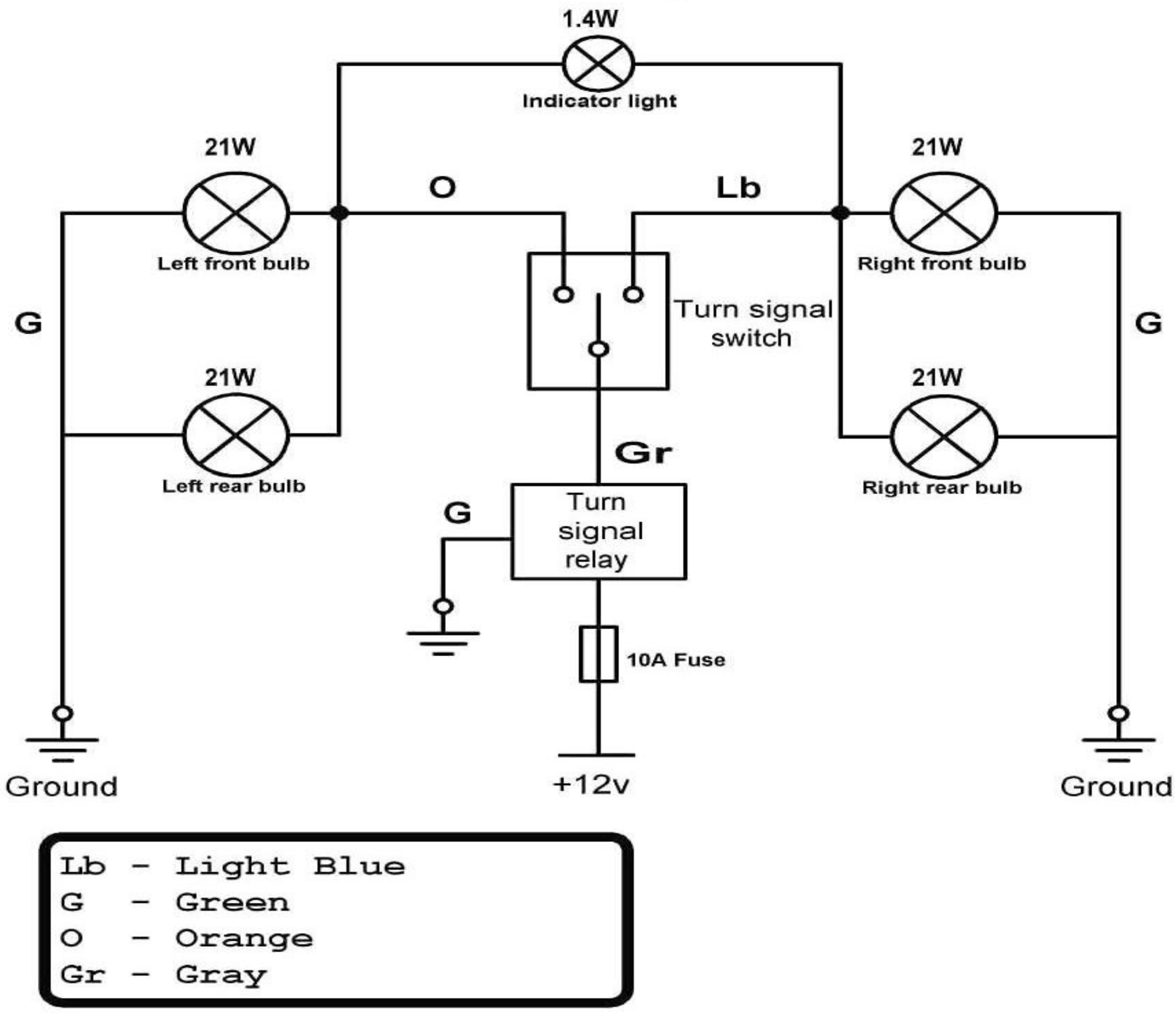


Figure 2-14 turn signal circuit with color code

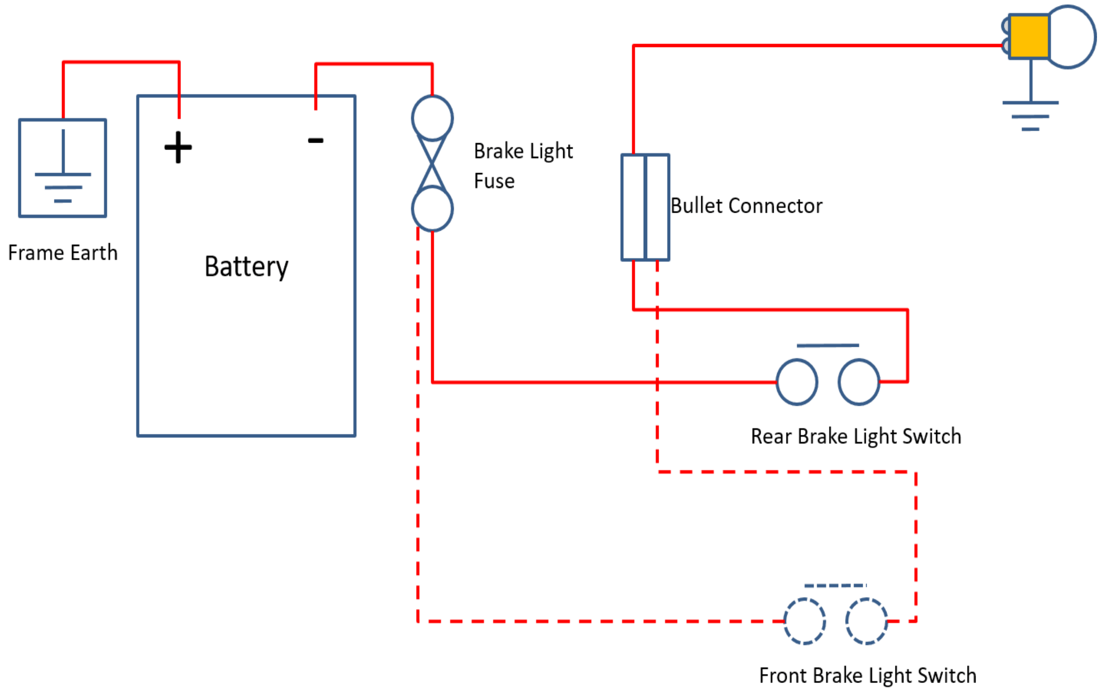
The current is supplied to the flasher unit or relay through a fuse. The flasher unit develops a blinking or regularly varying ON and OFF current which is transferred to the bulbs through wires. In between the flasher unit and bulbs, a turn signal switch is used which is used to direct the current to either right side or left side, wherever needed.

iii) Brake light

Red steady-burning rear lights, brighter than the rear position lamps, are activated when the driver applies the vehicle's brakes and warn vehicles behind to prepare to stop. These are formally called stop lamps in technical standards and regulations and in the Vienna Convention on Road Traffic, though informally they are sometimes called "brake lights". They are required to be fitted in multiples of two, symmetrically at the left and right edges of the rear of every vehicle. International UN regulations No. 7 specify a range of acceptable intensity for a stop lamp of 60 to 185 candelas. In North America where the UN regulations are not recognized, the acceptable range for a single-compartment stop lamp is 80 to 300 candelas. We will use 27watt bulb for brake light.

Figure 2-15 Brake light circuit block diagram (source: wiring.yenpancane)

The brake light bulb are connected to the headlight also, so that when the headlight is turned on it automatically turns the light generally known as tail light which is slightly



low bright than brake light. When the brake is applied through pedal, he brake light switch place near to pedal activates and passes the current from battery to the brake light.

iv) Reverse gear indicator

To warn adjacent vehicle operators and pedestrians of a vehicle's rearward motion, and to provide illumination to the rear when backing up, each vehicle must be equipped with one or two rear-mounted, rear-facing reversing (or "backup") lamps. These are required to produce white light by US and international UN Regulations. However, some countries have at various times permitted amber reversing lights. In Australia and

New Zealand, for example, vehicle manufacturers were faced with the task of localizing American cars originally equipped with combination red brake/turn signal lamps and white reversing lights. Those countries' regulations permitted the amber rear turn signals to burn steadily as reversing lights, so automakers and importers were able to combine the (mandatorily amber) rear turn signal and (optionally amber) reversing light function, and so comply with the regulations without the need for additional lighting devices. Both countries now require white reversing lights, and the combination amber turn/reverse light is no longer permitted on new vehicles. The US state of Washington currently permits reversing lamps to emit white or amber light.

As the reverse light of the EATV we will use two 21 watt white led bulb at the back side of the vehicle whose working is described using the basic circuit diagram below:

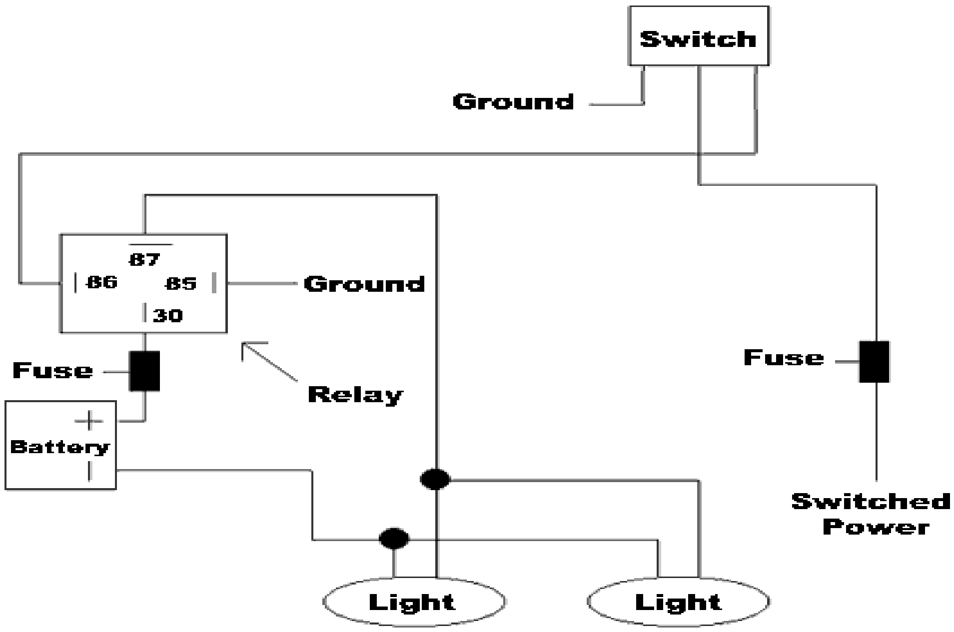


Figure 2-16 Reverse light circuit (source: rangerforums)

When the gear box is shifted to reverse gear, it activates the switch which powers the four-way relay connected to the battery and reverse lights. The switch operates the relay whenever the reverse gear is activated so that the battery power is transferred to the reverse lights at the back of the vehicle.

v) Horn

A horn is a sound-making device that can be equipped to motor vehicles, buses, bicycles, trains, trams (otherwise known as streetcars in North America), and other types of vehicles. The sound made usually resembles a "honk" (older vehicles) or a "beep" (modern vehicles). The driver uses the horn to warn others of the vehicle's approach or presence, or to call attention to some hazard. Motor vehicles, ships and trains are required by law in some countries to have horns. But some countries also set limits to the horn sound level. For our EATV we will use the 100watt powered single piece horn at front of the vehicle.

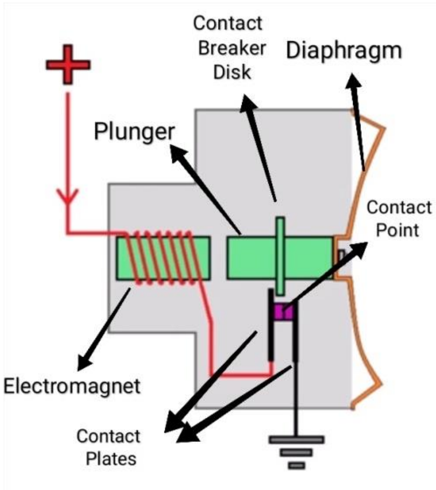


Figure 2-17 Horn construction (source: autocurious)

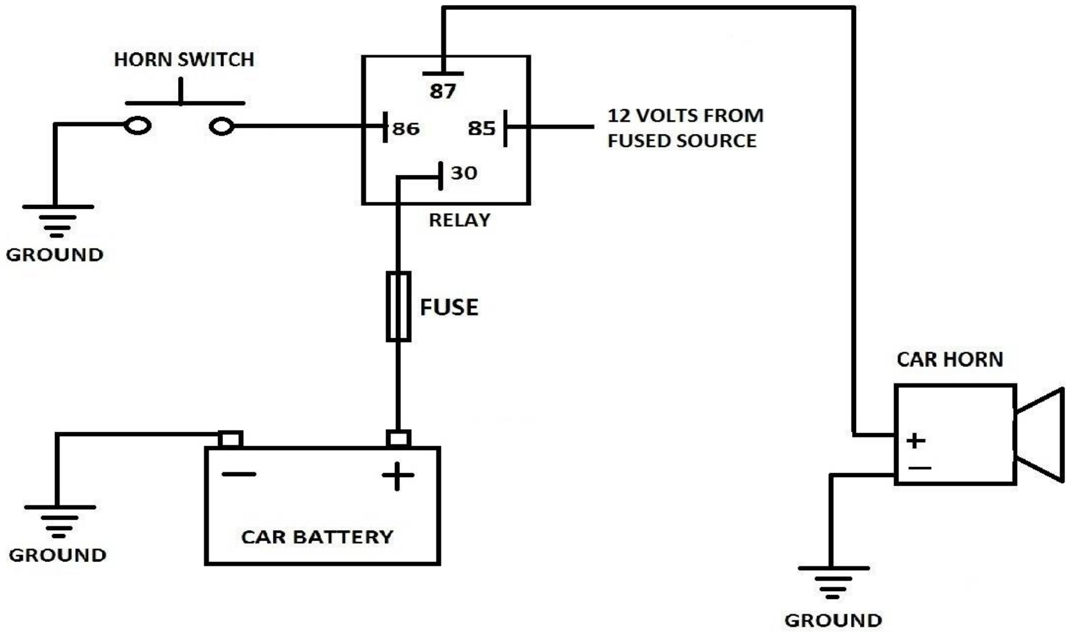


Figure 2-18 Circuit diagram for Horn (source: Baseballdiagram)

In actual operation, one presses the horn button and at that moment the electrical contact allows current flow to the relay, which in turn furnishes high current to the horn's electromagnet. That, in turn, attracts the diaphragm, which flexes to its mechanical limit. This disengages the contact, which stops current flow to the electromagnet. The diaphragm is released to travel back past neutral position closing the switch again, and thereby pulling the diaphragm back, setting up an even oscillation. Horns come in an endless variety of notes, or frequencies. The note of a horn is determined by the flexibility of the diaphragm its physical size; the power of the electromagnet; the mass of the diaphragm mechanical arrangement of the switch contact; size and shape of the horn's case and a number of other contributing factors.

2.7 Motor

The electric motor has the primary purpose of supplying mechanical torque to the drive wheels of the automobile. The secondary function of the motor is to generate electric energy through a generation mode by accepting mechanical torque from the drive wheels. There are different types of electrical motors that can be used for EVs and HEVs.

DC machines are the simplest motors to control as the source of electrical energy is already stored in a dc state (Sen, 1996). The dc machine has many mechanical components such commutator and brushes. These added mechanical components add to higher maintenance requirements over ac machines. The highest efficiency point of dc machines is near the maximum rated speed.

AC or alternating current machines offer great efficiency over a broad range of speeds; however they require more complex and sophisticated control systems and drives than their dc counterparts. There are multiple variants of ac machines available, including induction, synchronous and permanent magnet. Induction and permanent magnet machines can be controlled with a similar converter whereas a synchronous machine requires an added dc excitation system to create a magnetic field on the rotor.

2.7.1 Comparative study of different motors

a. Induction motor

These are some of the most popular AC motors. Induction motors are usually just called as AC motors owing to their popularity.

The efficiency of the induction motor is also pretty high in the range of 85-97% which is excellent. But that does not mean that induction motors are perfect. A low starting torque, an increase in the rotor losses at higher speeds, a reduction in efficiency as well as a lower efficiency when compared to permanent magnet synchronous due to the presence of rotor windings. Induction motors also has an extremely narrow speed across which useful power output can be obtained. Induction motors are actually used by Tesla in their cars such as the Tesla model S. Induction motors are commonly used in high power applications which explains its presence in electric vehicles. Induction motors also have a market share in variable speed drive applications such as air conditioning systems, elevators and escalators.

A single phase induction motor, unlike a 3 phase induction motor, does not have a self-starting torque. Auxiliaries are required to start a single phase motor.

Speed control of an induction motor is very difficult to attain. This is because a 3 phase induction motor is a constant speed motor and for the entire loading range, the change in speed of the motor is very low.

Due to poor starting torque, the motor cannot be used for applications which require high starting torque.

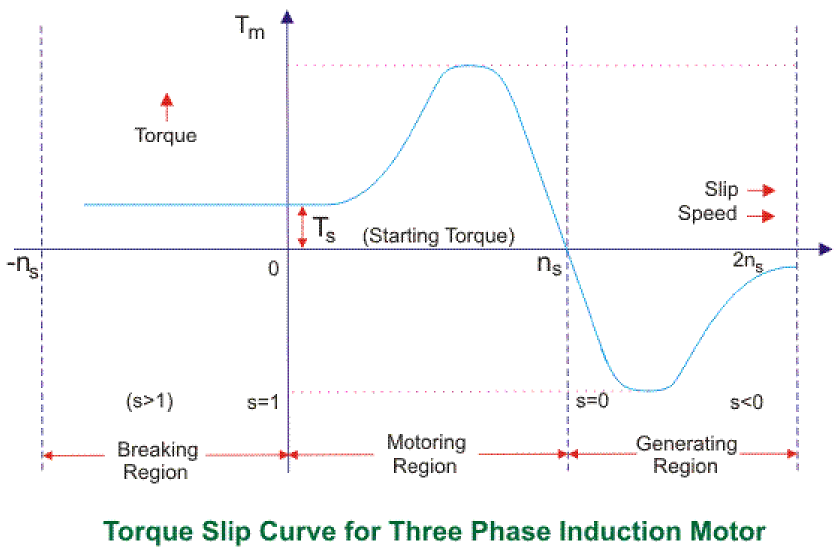


Figure 2-19 Torque Slip Curve for Three Phase Induction Motor (Electrical4u, 2017)

b. Permanent Magnet Synchronous Motor

Permanent magnet synchronous motors are also another serious contender for the usage in electric vehicles and rightfully so. It is also a brushless motor thereby the robustness is increased and therefore the maintenance issues are extremely low.

- They are much costly as compared to other types of large commercial and industrial fans
- Irreversible Demagnetization: These motors use permanent magnets, and everything depends on the magnetization of the magnets. Due to any problem in magnetization, the fan will not perform well or stop. Due to irreversible demagnetization, the performance degrades. It even becomes unstable in some cases.
- Controlling a PMSM is not that easy as other motors. It requires a complex controlling system.

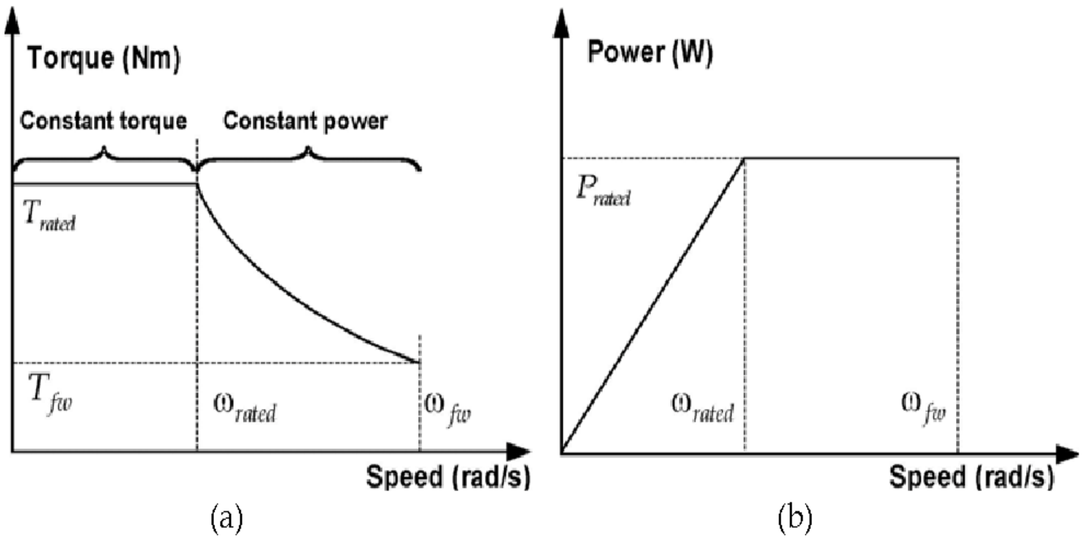


Figure 2-20 Torque/Power vs. Speed Curve for PMS Motor (Semanticscholar, 2015)

c. Brushless DC Motor

BLDC motors possess excellent speed torque characteristics, a wide operating speed range compared to induction motors, high power densities and lesser maintenance issues due to the lack of brushes. The lack of brushes also means that there are no sparking issues as well that conventional brushed DC motors usually suffer from.

- A BLDC technically can be considered as a type of synchronous motor as the magnetic field produced by the rotor and the stator rotate at the synchronous speed. The rotation occurs due to the pair poled being energized in a sequential manner such that the rotation is started and maintained. It is also lighter, easier to control, and less prone to failures that usually occur with other motors

- BLDCs can be a viable option to be used in the electric vehicles in low power electric vehicles such as electric auto-rickshaws and two-wheelers.
- Better speed versus torque characteristics
- High dynamic response
- High efficiency
- Long operating life due to a lack of electrical and friction losses
- Noiseless operation
- Higher speed ranges

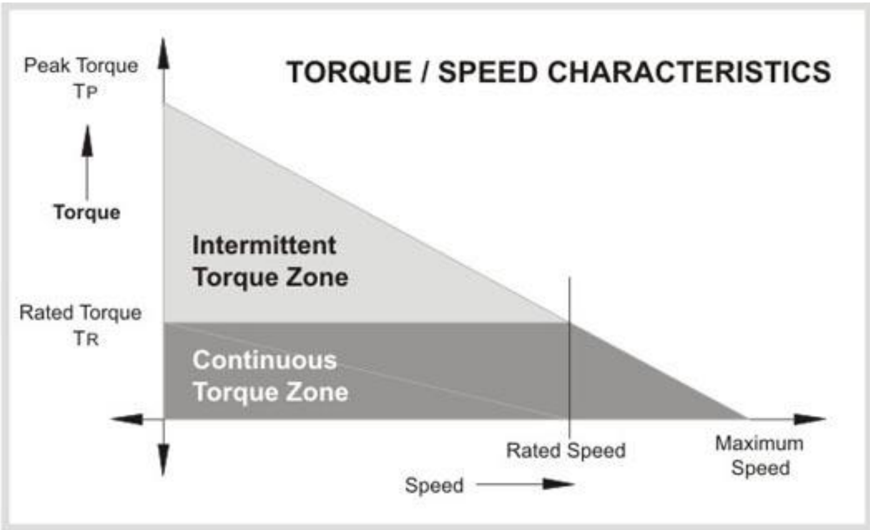


Figure 2-21 Torque-Speed Characteristics of BLDC Motor (Exchange, 2013)

2.8 Battery

2.8.1 Lead acid battery

These batteries are easy to identify because they are clear containers with a black top that has caps fastened to it. These batteries have multiple cells separated by lead plates that are charged by battery acid to power ATV. ATV conventional batteries, unlike most car batteries, take maintenance to make sure they are able to function and run properly. The primary responsibility as an ATV owner is to make sure the acid levels stay within the fill lines. (J.L.Lowry, 2012).

There are two types of lead acid battery namely FLA (flooded lead acid) and SLA (sealed lead acid). For the purpose of EATV, FLA is more popular than SLA because of long life (up to 4 years) and it cheaper amp-hour than other available batteries. The number of cycles over the depth of discharge can be shown here. As seen it has only 950 -1100 cycles over 30 percent depth of discharge. The number of charging and discharging cycle determined the battery life span.

Flooded batteries must be regularly serviced in order to maintain efficiency and battery life. The maintenance is done by filling the individual cells of the battery with distilled water to maintain an optimum specific gravity, which is measured by a hydrometer. To keep the battery running efficiently, the case must be taken off and this procedure must be done about once a month. With a sealed battery this is not necessary, and as the name suggests, the battery is permanently sealed. A safety problem arises when considering a flooded battery; even if proper precautions are taken, any time exposed acid is involved, an accident is subject to higher probability. Even though sealed batteries are approximately twice as expensive as flooded, safety is more important. With all of this information taken into account, it was determined that if a lead acid battery was to be used, it would be a deep cycle sealed battery.

2.8.2 Lithium ion battery

This is the newest and most specialized type of battery and with that comes a more substantial price tag. These batteries come pre-sealed and ready to charge and install. Unlike lead acid and AGM batteries, there is no liquid in a lithium battery. This makes them lighter, smaller and able to be mounted in any position. Lithium batteries are the latest in ATV battery technology, but that does not make them necessary for all ATVs. Lithium-ion batteries are lighter than Lead acid and Nickel metal. These are also the batteries used in digital cameras and smart phones. Most Lithium batteries, regardless of the cathode material, can be cycled well over one thousand times because Lithium has the largest electrochemical potential and specific energy, pound for pound, of any metal.

There are many types of Lithium ion batteries, so the first process in evaluating Lithium ion was deciding which type would be used. Overall, six types of Lithium-ion batteries are in common usage: Cobalt Oxide (LCO), Nickel Cobalt Aluminum Oxide (NCA), Nickel Cobalt Manganese Oxide (NCM), Manganese Oxide (LMO), Titanate (LTO), and Iron Phosphate (LFP). Lithium ion batteries are named for the active materials, cathodes, that give the battery its unique characteristics. All of these different cathode materials offer different advantages to the battery. While examining the types of batteries, it was apparent that the cost and performance of the three were relatively similar, so instead, emphasis was put on life span and safety. Superiority was denoted as the Iron Phosphate battery, which is intuitive as it was first developed in an effort to replace lead acid batteries. It is by far the safest option since it was designed to be used at high voltages for a long time; other batteries would falter in such conditions. If the battery is used, and stored, properly it can be used for approximately 2000 cycles. This is much higher than Nickel Cobalt Manganese Oxide and Manganese Oxide. In addition to advantages that come with safety and life span, Iron Phosphate also has relatively high specific energy and excellent specific power. Specific energy is the amount of energy that the battery has per unit mass, and specific power is the amount of energy per unit volume. There are two important characteristics to mention: firstly, the vehicle has a weight limit, secondly, the area where the battery will be stored is relatively small. This will allow the vehicle to have maximum power while not weighing down the vehicle or taking up much room. After all the factors had been evaluated, it was clear that Lithium Iron Phosphate (LiFePO4) was the best battery option if the vehicle was to run using a Lithium-ion battery. (Kretchmer, April 26, 2017)

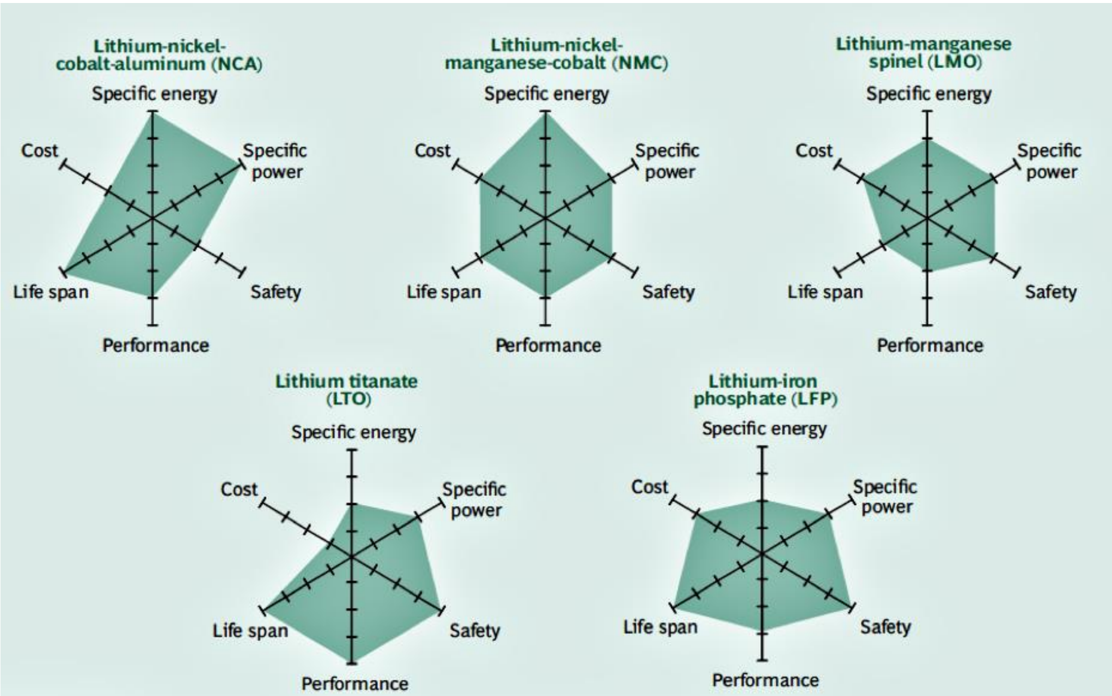


Figure 2-22 Comparison of different lithium ion battery

In most cases, lithium ion battery technology is superior to lead acid due to its reliability, efficiency and other attributes such as: -

a. Capacity: We can store more energy within a lithium ion battery using the same physical space. So, we can discharge more energy and thus power appliances for longer periods of time which is required for EATV.

b. Efficiency: Higher efficient batteries charge faster and similarly to the depth of discharge, improved efficiency means a higher effective battery capacity. Most lithium ion batteries are 95 percent efficient or more than lead acid battery whose efficiencies vary between 80 to 85 percent.
c. Lifespan: Lithium ion batteries generally last for several times the number of cycles as lead acid batteries, leading to a longer effective lifespan for lithium ion products.

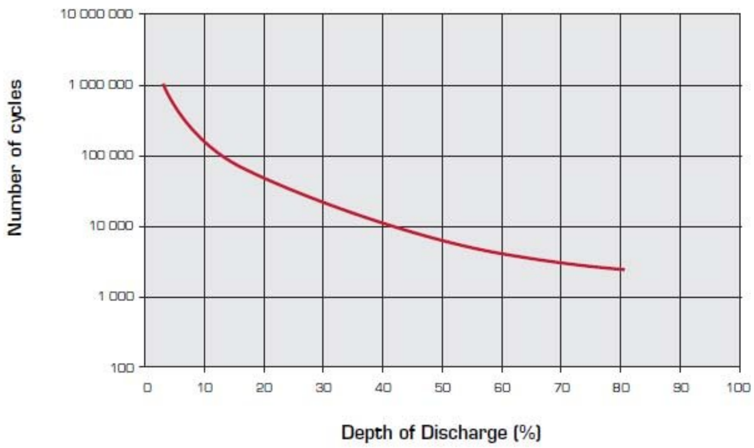


Figure 2-23 Cycle vs % depth of discharge of lithium ion battery

Li-ion battery price is the major challenge to companies, investors and researchers but thanks to the researchers and due to mass production, its price has been lowered down 14 percent from 2010 (NRs. 127,600 /kwh) to 2019 (NRs. 17,710 /kwh). If this price trend goes on it is expected that the price of per kilowatt hour of Li-ion battery in 2030 will be NRs 6,820.

2.8.3 Nickel Metal Hydride

Lead-acid batteries have been around since the earliest days of electric vehicles. The energy density of lead-acid batteries is often used as a base when comparing different battery technologies. Lead acid batteries can supply and absorb a large current. As the energy density of lead acid is not very high, the mass and volume of the batteries required to store a functional amount of energy is a problem. Also, lead acid batteries do not have a long deep cycle life. Ni-mH batteries can have up to three times the energy density of lead-acid batteries per unit volume and weight. The batteries can supply a large current. However, when charging the batteries only a fraction of the supply current can be applied. The limitation on the charge current results in an extended re-charge period. Ni-mH batteries have a much better life when deep cycle discharged when compared to lead-acid.

Li-Ion batteries have approximately four to five times the energy density of lead acid batteries for the same unit and weight. The volume of Li-Ion batteries is also much smaller than their lead-acid counterparts. Similar to Ni-mH batteries, they can supply a large current but can only absorb a fraction of this current when being charged. The discharge characteristic of a Li-Ion battery is practically a constant voltage. This is in general a desirable characteristic to have for a battery, but adds complexity when trying to monitor the state-of-charge (SOC). This constant discharge characteristic brings about the requirement of a battery management system to monitor the energy in and the energy out of the battery. This management technology enables excellent deep cycle discharge life and can monitor additional items, such as the temperature, to extend their service life. Li-Ion batteries are currently extremely expensive when compared to other battery types (Ceraolo & Pede, 2001). Table 3 is a comparison of the different battery technologies (Miller, 2004), (Rosenkrans, 2003)

Table 2-1 Comparison of EV batteries at "deep cycle" condition

High Energy Design in Deep Cycle Applications(Units)	Lead Acid	Nickel Metal Hydride	Lithium-Ion
Energy density (W-hr/Kg)	35	70	>90
Power density (W/Kg)	150	200	600
Life time (number of cycles) At 80% depth of charge	125	3,000	2,500

2.9 Battery management system

2.9.1 Introduction

We are using lithium ion battery for our EATV. The lithium ion batteries can be used only in specified conditions. These batteries are large capacity and are rechargeable. The process of charging and discharging should be constantly monitored and controlled. So battery management system should be used to monitor battery state of charge and ensured safety of operations. BMS is an electronic system that manages the rechargeable batteries to ensure it operates safely and efficiently. It has number of electronic components embedded in a circuit board. Battery management system are equipped with some sensors to determine several factors like very battery cells terminals voltage, state of charge of the battery, the temperature of the battery.

2.9.2 Advantages of Battery Management System

- Battery management system continuously monitors parameters such as temperature, voltage and current in and out of the pack to ensure it is being operated in safe conditions the entire time. Mainly lithium ion batteries have high chance of catching fire so BMS is most in such batteries.
- Battery management system helps to maintain charge state in between the maximum and minimum limit set by manufacturer. During charging it determines how much current can go safely in the battery. It protects the battery from overcharging, overheating and from explosion. It also protects the battery from deep charging. If the cell is discharged below threshold of 5%, total capacity of the cell can be permanently reduced. BMS helps to protect from this also.
- BMS helps for energy management, consumption rate, travel range, monitor battery level indicator, voltage indicator. It also helps to minimize leak current
- BMS helps in cell balancing by maintaining the cells at equal voltage level and maximize the capacity utilization of the battery pack. It measures relative voltage and calculate how much charge equalization is required.
- BMS is also responsible for communicating with other ECU (electronic control unit) in vehicle. It provides the necessary data about the battery conditions to the motor controller to ensure smooth running.
- It powers the load with the minimum supply voltage irrespective of the battery voltage.

2.9.3 General BMS system

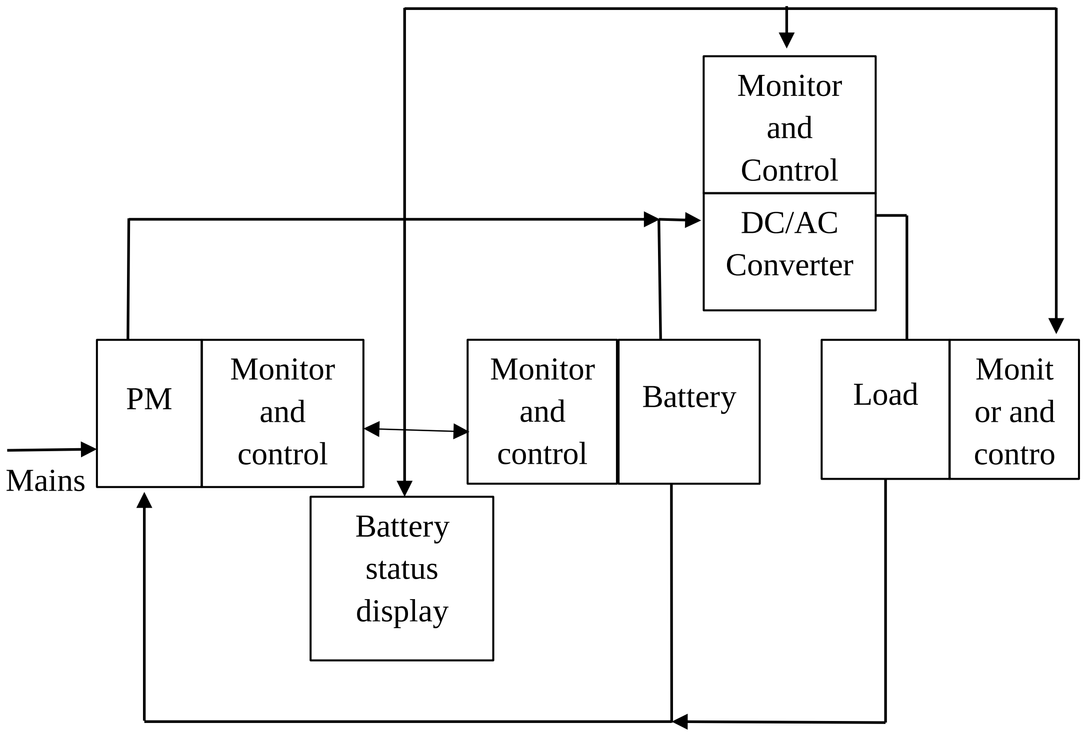


Figure 2-24 General BMS System

2.9.4 Basic components of BMS

a. Power module (PM)

Power module charges the battery by converting electrical energy from the mains into electrical energy suitable for use in battery. PM can also be used to power the portable device directly when the battery is low or damaged. PM can either be a separate device or be integrated with portable device. The monitor and control process of the power module can be of two types i.e. monitoring and controlling for energy conversion process and for battery charging process.

b. Battery

Battery's main objective is to store energy obtained from the mains or some other external power source and to release energy to the load when needed. The portable devices or vehicles can be operated without connection to any power source. Different types of battery can be used. Two or more than two battery can be used as a battery pack if high voltage is required. For the vehicles detachable battery is used. For our EATV high lithium ion battery is used. In lithium ion battery safety switch is used. The electronic safety switch monitors battery voltage, current, and temperature. It also controls charging and discharging process.

c. DC/DC converter

The DC/DC converter is used to connect a battery to the various system parts when the battery voltage doesn't match the voltage needed. When the battery voltage is too low, it will increase the voltage and if the voltage is too high, down conversion is performed. The efficiency can be improved by minimizing the supply voltage up to need. By the use of DC/DC converter the design of the load can be optimized for minimum supply voltage instead of whole voltage which will increase the efficiency of load.

d. Load

The basic task of the load is to convert the electrical energy supplied by a battery into an energy form that will fulfill the load's function. In vehicle also there are different types of load which use different supply voltage. For example, electric motor, it receives current from the battery source and then uses it to power the power train. It coverts electrical energy into the mechanical energy in the form of rotation. Other loads can be light, horn and so on.

e. Communication channel

The communication channel is used to transfer monitor and control signals from one BMS part to another. The type and complexity of communication channel in a battery management system depends on the complexity of the system and the intelligence needed for battery management. The battery can also communicate their status to the system host and other system parts such as power module or charger.

2.10 Steering system

2.10.1 Introduction

Steering system is one of most important part of any vehicle, which is used to provide directional stability to the vehicle. The primary function of a steering system is to steer the wheels in order to take the driver command as an input and provides an overall directional control for the vehicle. A steering system must provide sufficient comfort for the driver to actually sense what is happening with the front tires contact patch as well as enough "feel" to sense the approach to the cornering limit of the front tires. The steering must be fast enough for an EATV, so that the vehicle's response to steering and to steering correction to happen instantaneously and it must also have some self-returning action. The feel, feedback and self-returning action depend on the kingpin inclination, scrub radius, castor angle of the front tire.

Positive caster angle improves the self-returning action of the steering wheel, but increases a little steering effort, whereas the negative castor angles reduce the steering effort, of the driver but produce some wheel wandering problems. Higher the Kingpin Inclination (KPI) angle helps in improving self-returning action of the steering wheel and also decreasing the steering effort. Negative camber angle produces higher lateral forces to improve the cornering ability of the vehicle. Moreover, Toe in angle of the wheels help in improving the straight-line stability, whereas the toe-out angle help in improving the cornering stability. Negative scrub radius helps to stabilize the effect of vehicle handling. Hence, proper alignment of the wheels helps in achieving better steer ability and handling characteristics of the vehicle.

2.10.2 Design of steering system

a. Requirements of steering system:

i. The steering mechanism should be very accurate and easy to handle.

ii. The effort required to steer should be minimum and must not be tiresome to the driver.

iii. The steering mechanism should also provide the directional stability. This implies that the vehicle should have tendency to return to its straight ahead position after turning.

iv. It should provide pure rolling motion to wheel.

v. It should be designed in such a manner that road shocks are not transmitted to driver.

b. Geometry selection and set up:

Traction is an important aspect in off-road racing as compared to speed. Since Ackerman steering geometry gives high stability at lower speed. This type of geometry is appropriate for Electric All-Terrain Vehicle where the speed limit is 40 kmph because of the terrain therefore we choose Ackerman steering geometry.

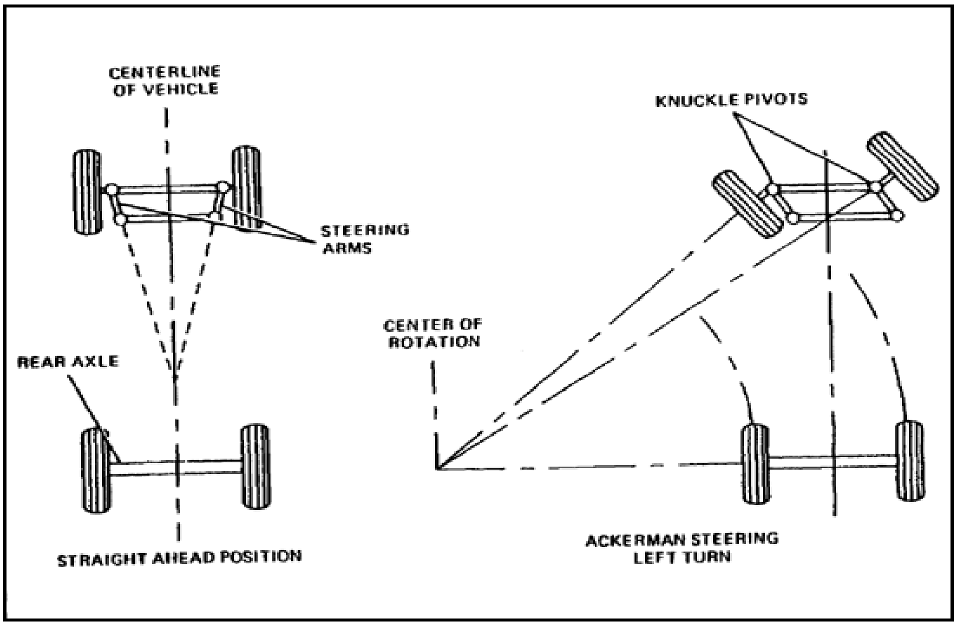


Figure 2-25 Ackermann Steering Geometry (Source: https://www.pinterest.com/)

Ackerman Principle state that the extended axes of steering arm should meet at the center of the rear axle. While taking turns, the condition of perfect rolling is achieved if the axes of the front wheels when produced meet the rear axis at one point. This is the instantaneous center of the vehicle. When vehicle is following a curved path the inner wheel deflects by a greater angle than the outer wheel to effectively complete the cornering without skidding.

The steering and suspension systems are crucial for successful operation of any variety of cars. Due to the large responsibility that these two major components share coupled with the fact that EATVs are required to be capable of taking over toughest of the terrains possible, it is obvious that consequences of failure or improper setup of the suspension and/or steering could be quite catastrophic. So combined efforts in suspension geometry and steering geometry were taken in order to design efficient system. Directional control of a road vehicle is normally achieved by steering the front wheels. This is mainly the result of steering wheel movement by the driver, but partly the result of suspension characteristics. (Akshya Pawar, 2018)

c. Steering Design Parameters:

Wheelbase and track width are selected considering suspension geometry, handling and stability of vehicle. Kingpin offset was decided by considering the packaging of wheel assembly and caster angle selected such that it gives straight line stability and optimum self-returning action for better handling. Position of rack were chosen so as considering pedal packaging and to avoid significant amounts of bump steer.

Table 2-2 Input Parameters

Wheelbase	1600 mm
Pivot to pivot distance	989.6 mm
Track width	1389.6 mm
Caster Angle	11°
Rack dimensions length travel	150mm

2.10.3 Classification of Steering System:

Steering system can be classified mainly into two types. They are manual steering system and power steering system. These can be further classified into several types.

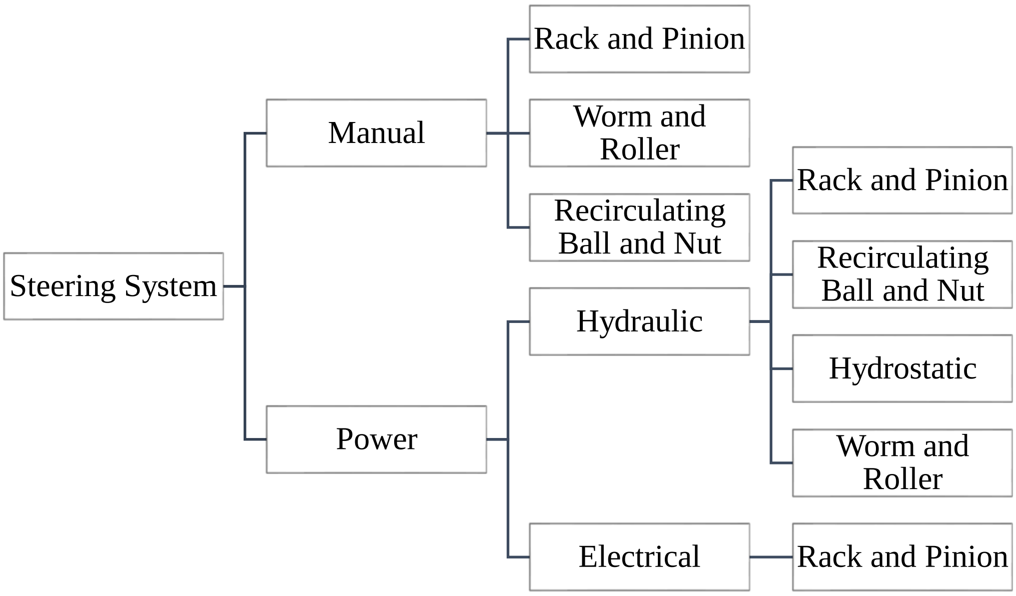


Figure 2-26 Classification of Steering System

Manual steering is considered to be entirely adequate for smaller vehicles. It is tight, fast, and accurate in maintaining steering control. However, larger and heavier engines, greater front overhang on larger vehicles, and a trend toward wide tread tires have increased the steering effort required.

Power steering is an advanced form of steering system in which the overall effort required by the driver is reduced through an increase in the force applied on the steering wheel.

2.11 Suspension system

2.11.1 Introduction to Suspension

Suspension is of extreme importance in any vehicle. It is the term given to the system of springs, shock absorbers and linkages that connects the vehicle to the wheel. It performs the function of shock absorption and allows relative motion between the vehicle and wheel. It serves a dual purpose contributing to the vehicle's handling and braking for good active safety and driving pleasure, and keeping vehicle occupants comfortable and reasonably well isolated from road noise, bumps and vibrations. The suspension has to withstand major shock and uneven train. The suspension systems absorb such shock and bump through the arrangement and linkages of spring damper. The absorption force mainly depends upon the unsprung mass at each wheel. The more the ratio of sprung mass to unsprung mass the more will be absorption force and less affected by bumps, dips and road imperfections.

2.11.2 Function of Suspension System

a) Suspension system supports the weight of the frame, transmission, drive train, passengers and cargo.
b) It helps to keep the tires in contact with the road.
c) It helps to maintain correct vehicle ride height.
d) It allows tires and wheels to move up and down with minimum vertical movement of vehicle thus provide a comfortable and safe ride.
e) It contributes in good active safety, driving pleasure and keeping vehicle occupant comfortable.
f) It helps to keep the wheels in correct alignment by assisting steering system.
g) It controls the direction of travel of vehicle and allows proper cornering of vehicle.
h) It controls excessive body dive while braking.

2.11.3 Basic terms related to suspension

a. Caster Angle

   Caster angle is defined as the angle made by kingpin in the side view with the vertical axis passing through the wheel center. It is measured in longitudinal direction. Caster angle is adjusted to provide good handling and steering characteristics of vehicle.

b. Kingpin Inclination

   Kingpin inclination can be defined as the angle at which kingpin axis is inclined to the vertical axis passing through the wheel center. The higher the kingpin angle, the more will vehicle lift when steered. When the kingpin axis is extended to the ground, the distance between that and the center of contact of wheel and road is called scrub radius. It increases the wear and tear of tires.

c. Camber

   Camber is defined as the angle between the vertical axis and the inclination of wheel tire view from front of the vehicle. There are two types of camber; positive camber and negative camber. Camber is considered as positive if top of the wheel tilted outwards and negative if top of the wheel tilted inwards. Camber affects the stability of the vehicle. Positive camber is for stability of the vehicle whereas negative camber is for high performance vehicle for better cornering. A negative camber could potentially lock the steering system.

d. Roll Center

Roll center can be defined as the point in the transverse vertical plane through the wheel centers at which lateral forces may be applied to the sprung mass without producing suspension roll. It is the point about which the vehicle rolls during cornering. It is the location at which lateral forces developed by the wheels are transmitted to the sprung mass. It affects the behavior of both the sprung and unsprung masses and thus directly influences cornering of the vehicle. We will discuss more about ball joint later in our report.

e. Ball Joint

Ball joints are made of a bearing stud and socket that fit snugly inside a lubricated bearing. It connects the control arm to the steering knuckles and allow for smooth and solid movement in the suspension. Almost in every automobiles ball joint is vital component.

f. Sprung and Unsprung Mass

Sprung mass is the mass supported by the suspension whereas unsprung mass is the mass that is mainly connected to the suspension. Sprung mass includes body frame, the internal components, passengers and cargo. Unsprung mass includes mass of wheel axles, wheel bearings, wheel hubs and tires and portion of spring and shock absorbers.

g. Anti-dive

Anti-dive is the characteristics of a vehicle to dive to the front on the application of brakes. It affects the amount of suspension travel when the brakes are applied. When the brakes are applied, the weight transferred from rear of the vehicle to the front caused vehicle to dive front. Anti-dive can be achieved by adjusting the pivot points of the suspension. 100% anti-dive refers to the no change in height of front suspension during braking. All the inertial force during acceleration is taken by the suspension force and zero force by the spring so spring won’t get compressed at all. 80% anti-dive will mean that 80% force is taken by suspension links and only 20% by spring.

h. Anti-Squat

Anti-squat is a term of suspension system to determine how much the suspension system is resisting suspension compression. Amount of squat vehicle experiences depends upon vehicle properties like center of gravity height, total weight, and acceleration rate and wheel base. 100% anti-squat refers to no change in rear suspension height under acceleration. Achieving 100% anti -squat will help in steadiness of the vehicle

2.11.4 Types of suspension system

According to the function and geometry, suspension system can be broadly categorized into two types. They are discussed below:

a. Dependent Suspension System

In this type of suspension system, both right and left wheels are attached to the same solid axle so it acts as a rigid structure. The movement of one wheel will slightly affect another wheel in the rear or in the front. The movement of one wheel will slightly affect another. When one wheel hits the bump in the road its upward movement will cause slight movement or tilt in another wheel. In this type of suspension, when camber of one wheel changes than the camber of opposite wheel also changes in the same way. If one wheel has positive camber change than another has negative camber change. This type of suspension is mainly used in heavy vehicles. Depending upon the location of linkages, the dependent suspension system has following configuration:

i. Watts linkage
ii. Pan hard rod
iii. Satchell link
iv. Live axle
v. Leaf spring
vi. Twist beam
vii. Beam axle
viii. WOBLink

b. Independent Suspension

Independent suspension is the type of suspension system that allows each of the wheels on the same axle to move independently of each other. This type of suspension provides better ride quality. The main advantages of such system are that they require less space, provide better steer ability and low weight compared to another. Suspensions with other devices, such as anti-roll bars that link the wheels are also classified in independent suspension systems. The various independent suspension systems are as follows:

i. Double wishbone suspensions
ii. McPherson struts and strut dampers
iii. Rear axle trailing-arm suspension
iv. Semi-trailing-arm rear axles
v. Multi-link suspension

2.11.5 Double Wishbone Suspension System

The double wishbone suspension is a very popular type of suspension found on mid-range to high-end cars. It is an independent suspension design using two (occasionally parallel) wishbone-shaped arms to locate the wheel. The double-wishbone suspension can also be referred to as "double A-arms," though the arms themselves can be A-shaped, L-shaped, or even a single bar linkage. A single wishbone or A-arm can also be used in various other suspension types, such as MacPherson strut and The upper arm is usually shorter to induce negative camber as the suspension jounces (rises). Each wishbone or arm has two mounting points to the chassis and one joint at the knuckle. The shock absorber and coil spring mount to the wishbones to control vertical movement. Double wishbone designs allow the engineer to carefully control the motion of the wheel throughout suspension travel, controlling such parameters as camber angle, caster angle, toe pattern, and roll center height, scrub radius, scuff and more.

There are various advantages and disadvantages of double wishbone suspension system it provides the engineer more free parameters than some other types do. It is fairly easy to work out the effect of moving each joint, so the kinematics of the suspension can be turned easily and wheel motion can be optimized. It also provides increasing negative camber gain all the way to full jounce travel. Its disadvantages are that it consists of large number of components within the suspension setup it takes much longer to service and is heavier than an equivalent MacPherson design. It takes more space and is complex than the other suspension system.

Double wishbones are usually considered to have superior dynamic characteristics as well as load-handling capabilities, and are still found on higher performance vehicles. Examples of makes in which double wishbones can be found include Alfa Romeo, MG, Pontiac, Honda and Mercedes-Benz. Short long arms suspension, a type of double wishbone suspension, is very common on front suspensions for medium-to-large cars such as the Honda Accord (replaced by MacPherson struts in 2013+ models), Peugeot 407, or Mazda 6/Atenza, and is very common on sports cars and racing car.

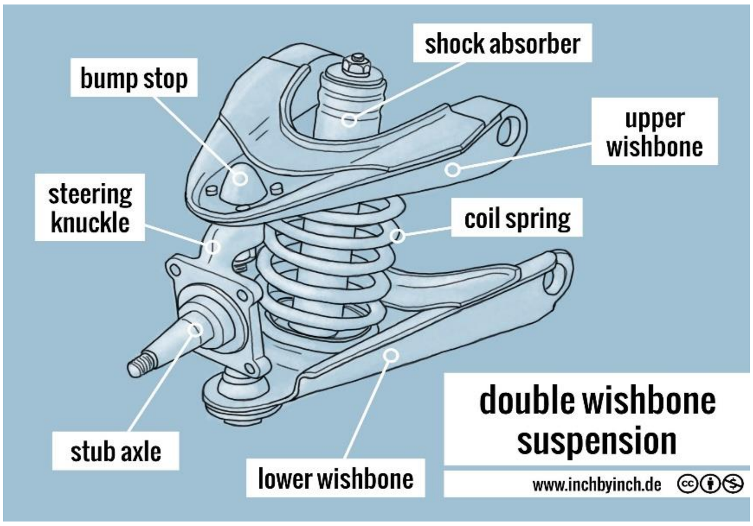


Figure 2-27 Double Wishbone Suspension System

2.11.6 Quarter car model

The sprung mass resting on the suspension and tire springs is capable of motion in the vertical direction. An Isolated Quarter Car Model is as shown in the figure below which can be used for determination of spring and damper effect. Here ‘Xs’ denotes the displacement of sprung mass, ‘Xu’ denotes the displacement of unsprung mass. The values ‘Ks’, ‘Cs’ and ‘Kt’ denotes the suspension stiffness, suspension damping coefficient and tire stiffness respectively.

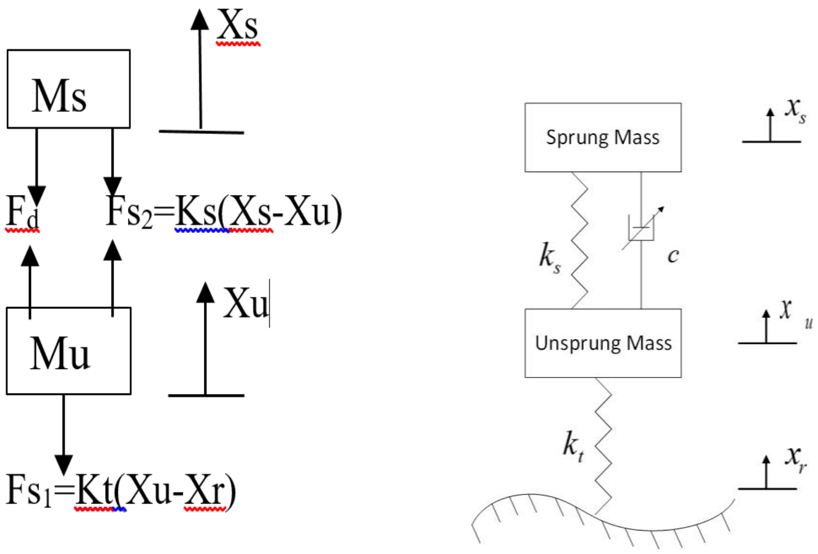


Figure 2-28 Free Body Diagram of Quarter Car Model

From Newton's 2nd law;

$\sum F = ma$

For sprung mass "M_s"

$\sum F = -F_d - F_s = Ms\ddot{X}_s$

$$= -C_S(\dot{X}_s - \dot{X}_u) - k_S(X_s - X_u) = Ms\ddot{X}_s$$

So,

$$\boxed{Ms\ddot{X}_s + C_S(\dot{X}_s - \dot{X}_u) + k_S(X_s - X_u) = 0}$$

The above equation is car body bounce governing equation.

For wheel bounce:

For un sprung mass,

$\sum F = ma$

$\sum F = F_d + F_{s2} - F_{s1} = Mu\ddot{X}_u$

$$C_S(\dot{X}_s - \dot{X}_u) + k_S(X_s - X_u) - k_t(X_s - X_r) = Mu\ddot{X}_u$$

$$Mu\ddot{X}_u + k_t(X_s - X_r) - k_S(X_s - X_u) - C_S(\dot{X}_s - \dot{X}_u) = 0$$

So,

$$\boxed{Mu\ddot{X}_s + C_S(\dot{X}_u - \dot{X}_s) + k_S(X_u - X_s) + k_t(X_s - X_r) = 0}$$

Where,

X_r= Road displacement

X_u= Unsprung mass displacement

X_s=Sprung mass displacement

So,

$$\ddot{X}_s = \frac{1}{Ms} C_S(\dot{X}_u - \dot{X}_s) + k_S(X_u - X_s) \qquad \text{--- (i)}$$

And,

$$\ddot{X}_u = \frac{1}{Mu} C_S(\dot{X}_s - \dot{X}_u) + k_S(X_s - X_u) + k_t(X_r - X_u) \qquad \text{--- (ii)}$$

These equations (i) and (ii) are integrated twice so that we can obtain the displacement of sprung and unsprung masses respectively.

2.12 Braking system

2.12.1 Introduction to Braking System

In an automobile vehicle, a braking system is an arrangement of various linkages and components like brake lines or mechanical linkages, brake drum or brake disc, master cylinder or fulcrums and so on (Mishra, 2018). These components are arranged in such a way that it converts vehicle's kinetic energy into heat energy which in turn stops or decelerate the vehicle. Braking system in any vehicle is arguably the most important subsystem of a vehicle. Brakes are used to stop a moving vehicle, to prevent it from moving or to control its speed while in motion. A quality braking system enables driver with range of braking effort. This will allow him to feather the pedal for obtaining required amount of braking, getting desired motion while navigating corners and curves. In this way, braking system contributes majorly in terms of safety and handling property of the vehicle.

2.12.2 Need of braking system

Braking system in any automobile is of great importance. The need of effective braking is increases more in our EATV. The needs of braking system in automobile are given below:

a. To decelerate the moving vehicle
b. To completely stop the moving vehicle
c. To prevent the vehicle from any damage due to severe road conditions.
d. For stable parking of vehicle either on the flat surface or on a slope.
e. As a precaution for accident.
f. To maintain vehicle speed during downward operation.

2.12.3 Classification of Braking system

Over the year since the development of automobile various types of brake system are being used. As the automobile model becomes more complex the demand in high effective braking system increases. With present technology and design braking system can be used to generate power. The system is classified on various basis but we don't require every one of them. So some of the classification important to our project are given below (Mishra, 2018):

a. On the basis of power source

i. Mechanical Braking System

Mechanical brake system is used by various old vehicles but they are obsolete nowadays due to their less effectiveness. They are often used for hand brakes and emergency brakes in modern automobiles also. Mechanical braking system is the type of braking system in which the brake force applied on the brake pedal is carried to the final brake drum or disc rotor by the various mechanical linkages like cylindrical rods, fulcrums, springs etc. In order to stop the vehicle. Since various mechanical components are used in this system so its effectiveness highly depends upon those components. So any damage of any components will have direct effect on braking efficiency so this type of braking system is not applicable for rough terrain vehicle and heavy automobiles.

ii. Hydraulic Braking System

Most modern cars have hydraulic brake system. This system works on the pressure applied by the brake fluid. In this type of brake system, the brake force applied by the driver on brake pedal is first converted into hydraulic pressure by master cylinder than this hydraulic pressure from master cylinder is transferred to the final brake drum or disc rotor through brake lines. On the contrast to mechanical system which uses various linkages to carry brake force applied by driver it uses brake fluid in order to stop or retard the vehicle. The force generated in the hydraulic braking system is higher when compared to the mechanical braking system. Almost all the bikes and cars on the road today are equipped with the hydraulic braking system due to it high effectiveness and high brake force generating capability. The chance of brake failure is very less in case of the hydraulic braking system. The direct connection between the actuator and the brake disc or drum makes very less chance of brake failure.

iii. Pneumatic Braking System

Pneumatic braking system is the type of braking system in which compressed air is used to transfer the pedal force from the brake pedal to the final brake drum or disc rotor. Compressed air is obtained from the compressor which takes the atmospheric pressure, purify it and then compressed it. Pneumatic brake generated higher brake force than the hydraulic brake. As hydraulic brake fails to transmit high brake force to the greater distance, heaviest automobiles like busses, truck and other construction equipment uses pneumatic brake.

Pneumatic brake system is usually equipped with a reserve air tank which comes in action when there is a brake failure due to leakage in brake lines so it has less chance of brake failure.

iv. Vacuum Brakes

For the braking purpose this system uses various components like exhauster, main cylinder, brake lines, valves along with disc rotor or drum. In this system the vacuum created in the brake lines cause the brake pad to move to stop or decelerate the vehicle. Vacuum brakes were used in old trains but now replaced by pneumatic brake system because of its less effectiveness and slow braking.

Vacuum brakes are cheaper than pneumatic brake but not often used because they are less safe than air brakes.

v. Magnetic Braking System

In this type of braking system the braking of vehicle is caused by the magnetic field generated by permanents magnet. It works on the principle that when we pass a magnet through a cooper tube, eddy current is generated and the magnetic field generated by this eddy current provide magnetic braking. This is the friction less braking system thus there is less or no wear and tear. This serves to increase the life span and reliability of brakes. This type of braking systems can be found in many modern and hybrid vehicles. Also, it is quite modest in size compared to the traditional braking systems. It is mostly used in the trams and trains. The response to the braking in this is quite quick as compared to other braking systems.

b. On the basis of frictional contact

i. Drum Brakes (Internal Expanding Brake)

Drum brakes were the first types of brake used on motor vehicles. Nowadays also over 100 years of its invention, drum brakes are still used in rear wheels of most vehicles. They are widely used as a rear brakes on small car and motorcycle. Drum brakes are also occasionally fitted as the parking and emergency brake even when the rear wheels use disc brakes as the main brakes A drum brake is a traditional break in which the friction is caused by a set of shoes or pads that press against a rotating drum-shaped part called a brake drum. It is generally made of cast iron that rotates with the wheel. Given below is the schematic diagram for drum brake showing its major components.

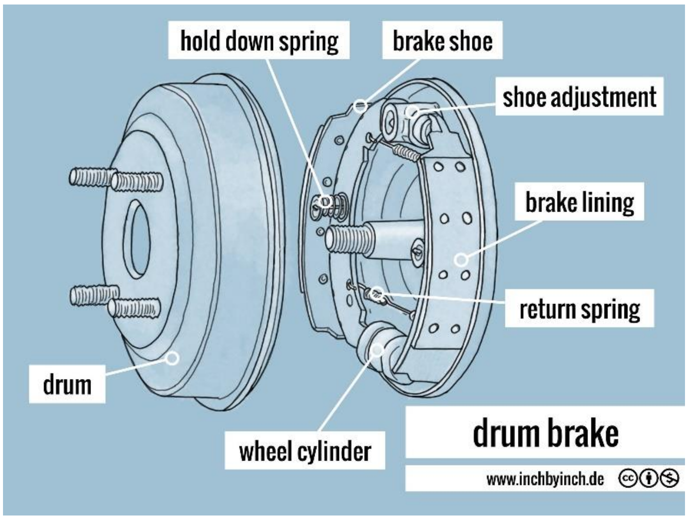


Figure 2-29 Drum Brake System

ii. Disc Brakes (External Expanding Brake)

A disc brake is a type of brake that uses the calipers to squeeze pairs of pads against a disc or a rotor. It is the types of braking system in which instead of a drum assembly a disc rotor attached to the hub of the wheel in such a fashion that it rotates with the wheel, this disc rotor is clamped in between the caliper which is rigidly fixed with the knuckle or upright of the vehicle. When the brakes are applied the actuation mechanism contracts the attached brakes pads which in turn makes the frictional contact with the rotating disc rotor and causes the braking of the vehicle. The brake disc is usually made of cast iron, but in some cases be made of composites such as reinforced carbons or ceramic matrix components. It provides efficient braking than the drum brakes. All the modern vehicles have disc brake on front and disc or drum on the rear wheel. In contrast to the drum brake which has servo effect, brake disc has no self-servo effect and are less prone to brake fade. It recovers more quickly from immersion. Given below is the schematic diagram for disc brake showing its major components.

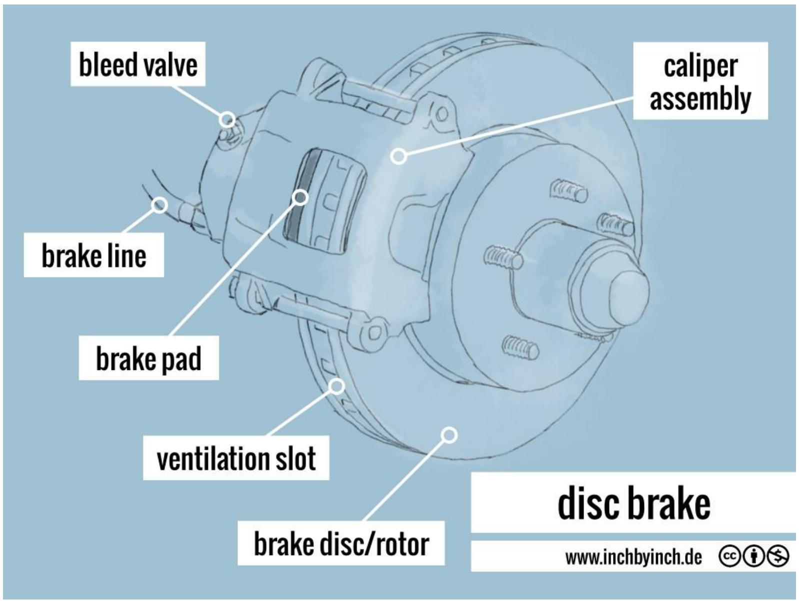


Figure 2-30 Disc brake

2.12.4 Brake Biasing

Brake biasing is the percentage of total braking force applied to the front wheels. It means applying variable force for both the front and rear axles of vehicle. This is done to obtain maximum braking efficiency and to have good stability and reduce stopping distance in account for variable road conditions. The braking force should increase as the weight on wheels increased. More braking force is required to the wheels having more weight. While applying brake the weight transfers from the rear to the front. This increases the traction in the front end reduces it in the rear. If we have equal brake balance than rear wheels will lock up first and vehicle tends to spin. Optimal braking is only obtained when all the four wheels lock up at the same time so for that brake biasing is important. It helps the driver maintain control while braking. A 60-70% bias is common on RWD street vehicles. This means that the front brakes provide 60-70% of the total braking force. FWD vehicles cars have up to 80% of bias. The biasing depends upon the dynamic weight transfer during braking. If we are using two master cylinders than bias bar also called as balance bar is used. But we are using only one cylinder so for proper biasing we are using larger size of caliper in front wheels and rear wheels. Biasing value can also be used in some cases so it creates complexity and increase our expenses. Other methods of increasing front bias are:

a. Larger front rotors
b. Front pads with more friction
c. Larger front calipers pistons
d. Smaller rear rotor
e. Rear pads with loss friction
f. Lowered suspension
g. More weight on the rear axle
h. Less sticky tires on the front wheels.

2.13 Tools Used

We have used some software for designing, modeling and simulation of Electric All-Terrain Vehicle Systems. These includes Solidworks, ANSYS, MATLAB and Simulink.

a. Solidworks

   It is a computer-aided design (CAD) and computer-aided engineering (CAE) computer program for solid modeling. It runs primarily on Microsoft windows and does not support MacOS. Solidworks software was founded by Massachusetts Institute of Technology graduate Jon Hirschtick in December 1993. The software now encompasses a number of programs that can be used for both 2D and 3D design. Solidworks is used to develop mechatronics systems from beginning to end. At the initial stage, the software is used for planning, visual ideation, modeling, feasibility assessment, prototyping, and project management. The software is then used for design and building of mechanical, electrical, and software elements. Finally, the software can be used for management, including device management, analytics, data automation, and cloud services. Its solution is used by mechanical, electrical, automobile engineers to form a connected design. In our EATV also various components are designed using solidworks. Steering system, suspension system and chassis frame are designed. After designing they are assembled together by mating of every component.

b. ANSYS

   Ansys, Inc. is an American company based in Canonsburg, Pennsylvania. ANSYS was founded in 1970 by John Swanson. Swanson sold his interest in the company to venture capitalists in 1993. It is a simulation software for design, testing and operation of product. It facilitates for fluid dynamics, electronics design and other physics analysis. is a software package that lets you digitally model real world phenomena. It uses computer-based numerical techniques to solve physics problems. The range of problems ANSYS can solve is immense and could be anything from fluid flow, heat transfer, stress analysis and more. It is also used to simulate computer models of structures, electronics, or machine components for analyzing strength, toughness, elasticity, temperature distribution, electromagnetism, fluid flow, and other attributes. Its main advantage is that it can solve problems that are not amenable to an analytical approach. That is, they don't have standard formulae. With the development of ANSYS software in recent years it has become more user friendly and are equipped with more options and simulation. We have used ANSYS for finding the deformation of spring, stress developed and factor of safety of spring. Chassis frame analysis is also carried out using ANSYS software.

c. MATLAB

MATLAB is a proprietary multi-paradigm programming language and numerical computing environment developed by Math Works. MATLAB allows matrix manipulations, plotting of functions and data, implementation of algorithms, creation of user interfaces, and interfacing with programs written in other languages.

d. Simulink

Simulink offers a way to solve equations numerically using a graphical user interface, rather than requiring code.

Models contain blocks, signals and annotation on a background.

i. Blocks are mathematical functions, they can have varying numbers of inputs and outputs.
ii. Signals are lines connecting blocks, transferring values between them. Signals are different data types, for example numbers, vectors or matrices. Signals can be labelled.
iii. Annotations of text or images can be added to the model, and while not used in the calculations they can make it easier for others to understand design decisions in the model.

We have used several blocks from Simulink library. These blocks in this library are mostly used for viewing data from the model.

i) Sinks Library (Scope block): These blocks in this library are mostly used for viewing data from the model. It plots input against time.

ii) Sources Library (Source block): Source blocks provide different signals for your model. The simplest source is simply a constant. A block that outputs a constant value.

iii) Math Operations Library :

- Add, Subtract and Sum Blocks: The Add, Subtract and Sum blocks are all essentially the same. By changing the Icon shape and List of signs in the block parameter we can convert one into the other.
- Gain Block: The Gain block can be used to multiply a signal by a constant value. You must configure the block parameters to perform matrix or element-wise (array) multiplication.
- Integration (Integrator block): We have used continuous integrator which outputs the integral of its input at the current time step. The following equation represents the output of the block 'y' as a function of its input 'u' and an initial condition y_0, where y and u are vector functions of the current simulation time t.

$$y(t) = \int_{t_0}^{t} u(t)dt + y_0$$

CHAPTER THREE
METHODOLOGY

3.1 Flowchart

The flow chart for the overall design is shown in Figure. This flow chart will serve as the general guidance to steer this project in the right direction. The flowchart is divided into two parts, in which the first part concentrates on the design and the second part focus on the fabrication and road tests. The outcome of this project is to study the performance of the proposed E-ATV by analysis of ride data recorded during test.

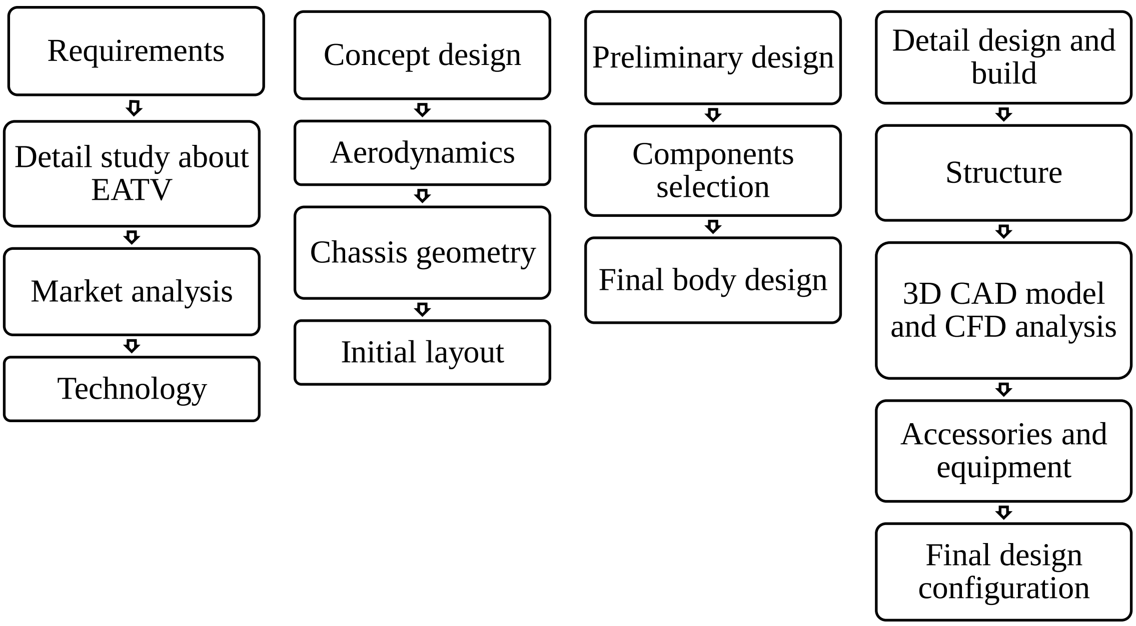


Figure 3-1 Design Methodology

The requirements of the proposed E-ATV are established after considering the information obtained from a literature reviewed. These requirements are set in a way that the E-ATV is practically feasible in term of design, fabrication and ride tests.

There are a few basic E-ATV requirements established in this project. The E-ATV should have long travel range and ability to reach the desired location as fast as possible at the reasonable time. The E-ATV should be suitable at smooth and rough surfaces. Maximum load of 150Kg is to be allowed which may consist of imaging equipment. The E-ATV should be capable of performing demanded operations with acceptable safety and stability characteristics. All-Up-Weight (AUW) of less than 200 kg. The wheel span of about 4m.The expected velocity range of approximately 50-60 km/hr. Lastly, it would be equipped with versatile controller unit able to handle manual controls.

Similarly the methodology which are required to build a symbolic design project model is explained below.

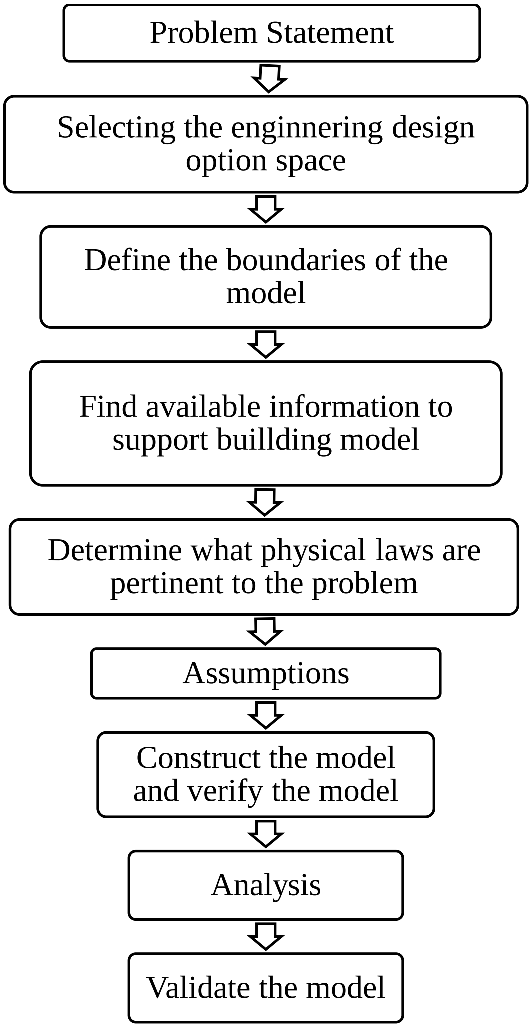


Figure 3-2 Methodology for project model design

Problem Statement

The purpose of the model is determined. The amount of resources spent on the model will depend on the importance of the decision that needs to be made.

Select the engineering design decision space

In this step, we need to move from general, fuzzy ideas to specific questions that can be answered by engineering analysis. We write the questions that we expect the model to help us answer.

Define the boundaries of the model

Here the boundaries of the model is determined. The boundary of the design system determines what is part of model and what is not. The information we can get from the model depends on the resolution of the model.

Find available data to support building the model

We separated step 4 and step 5 which are closely related in the listing model building process for emphasis. With the thought that has gone into defining the problem we should now know what physical knowledge we use to represent the physical situation.

Assumptions

In building a model we should be aware that the model is an abstraction of reality. . One way to achieve simplification is to minimize the number of physical quantities that must be considered in the model to make it easier to achieve a mathematical solution.

Construct and verify the model

A good first step in building the model is to make a careful sketch of the physical elements of the problem. Usually the analytical description of the model starts with either appropriate conservation laws or balance equations.

Analysis

The solution of the mathematical equations, analytically, numerically, or graphically, is the next to last step in the modeling process. Because only the simple cases can be solved with analytical (closed form) methods it is often necessary to use numerical methods.

Validate the model

It is important to validate the model to determine whether it is a faithful representation of the physical system. Validation also shows the level of accuracy that might be expected in predictions made with the model.

3.2 Structural Design Parameters

To design a small-scale E-ATV, it is important to limit certain parameters of sizes to a practically reasonable figure so that other parameters concerning dimensions would be feasible for fabrication. The design parameters chosen should also obey known theoretical concepts and equations to confirm that the E-ATV will be able to accomplish stable and safety flights across all its flight modes. The general procedures to determine basic structural design parameters are:

a. Estimate the All-Up-Weight of the vehicle
b. Chassis Design: Determine the size of the rod loading, wheel span, aspect ratio etc.
c. Selection of battery, motor, wheel hub, suspension, braking etc.

These general procedures will be used to estimate the size of the prototype. After fabrication and road tests, the actual dimensions will be adjusted accordingly.

3.3 Material Selection

Effective and economic project is the outcome of best material selection. Every single part of the smallest item such as a screw to the largest part such as a chassis, weight is the critical factor when coming to selection. Ideally, the high strength material and yet roust against mechanical (drop, shock) and environment (waterproof, salt spray compliant, altitude /low pressure, oil/chemical contamination/corrosion) shall be selected. Such EATV that normally uses in highway shall comply with some national standard of complaint to ensure the product is robust and can be used in any kind of application or situation. High investment in design/fabrication and qualification test shall be anticipated. However, the final choice of material and choice of manufacturing or fabrication techniques is also depending on cost affordability and quantity to be produced. For this project which is considered a prototype, the choice of material is not at all compliant with the above Standard. What most important in this study is the E-ATV is really working and noticeable. Material for the Chassis should be strong enough and easily available. In this study, wing material is made out of iron alloys which is very lightweight, easy to shape and low cost.

If we look into the rulebook of the SAE BAJA, on the basis of which the roll cage frames and other components will be chosen, has stated clearly that the material to be used for the frame of the roll-cage must contain the carbon 0.18% or more for the strength properties. This rule allows us to make our choices one among the four materials available in the market. The available materials, taken through various platforms, reports and study, are as below:

1. AISI 1018 - 0.18%
2. AISI 1020 -0.20%
3. AISI 4130 -0.30%
4. AISI 4140 -0.40%

All of these four material listed here are suitable to be used for the use of the roll cage fabrication. The differences in properties of these materials are very few and almost similar considering the manufacturability. Considering the calculations and suggestions mentioned on the journal “Design and Development of Roll Cage” by Aditya Kumar Mohanty, Ankit Jambhulkar, Prof. Bhupesh Sarode on International Research Journal of Engineering and Technology (IRJET), we selected the AISI 1020 as the material of our roll cage. (Aditya Kumar Mohanty, 2018)

AISI 1020 has low hardenability properties and is a low tensile carbon steel. It has high machinability, high strength, high ductility and good weldability. It is normally used in turned and polished or a cold drawn condition. The mechanical properties of the AISI 1020 are shown in following table:

a. Chemical Composition

The chemical composition of AISI 1020 carbon steel is outlined in the following table.

Table 3-1 chemical composition of the AISI 1020

Element	Content (%)
Manganese, Mn	0.30-0.60
Carbon, C	0.18-0.23
Sulfur, S	0.05 (max)
Phosphorous, P	0.04 (max)
Iron, Fe	Balance

Source: (AZoM, 2012)

b. Mechanical Properties

The following table shows mechanical properties of cold rolled AISI 1020 carbon steel.

Table 3-2 Mechanical properties of AISI 1020

Properties	Metric
Tensile strength	420 MPa
Yield strength	350 MPa
Modulus of elasticity	200 GPa
Shear modulus (typical for steel)	80 GPa
Density	7.87 g/cm^3
Poisson's ratio	0.29
Elongation at break (in 50 mm)	15%

Hardness, Brinell	121
Hardness, Knoop (converted from Brinell hardness)	140
Hardness, Rockwell B (converted from Brinell hardness)	68
Hardness, Vickers (converted from Brinell hardness)	126
Machinability (based on AISI 1212 steel. as 100 machinability)	65

Source: (AZoM, 2012)

3.4 3D Modelling

The 3D modeling will be done in Solidworks software. Basically, the critical parts that need to be modeled are the chassis. Other parts such as battery, motor, tires, steering wheel and column, would be designed with selected specification from the market because these parts are known for their reliability and strength during actual road test. The steps for the 3D modeling are:

1. Determine the dimension and design of the frame.

2. Project the chassis cross-section into 3D model.

3. Assemble the different parts with the frame.

The Solidworks designs of purposed two models for the EATV are attached in the annex-A and the mass properties of the designs with the center of mass is shown in annex-B

CHAPTER FOUR
MATHEMATICAL CALCULATIONS

4.1 Calculation of Tube Diameter

4.1.1 Conceptual Base Design

We assume the entire load applied to the vehicle transferred to the base of the EATV. So, the critical point for the stress is located at the base. Hence we will calculate the tube diameter for the base to bear the entire load and use same diameter tube in upper body fabrication to make a safe design.

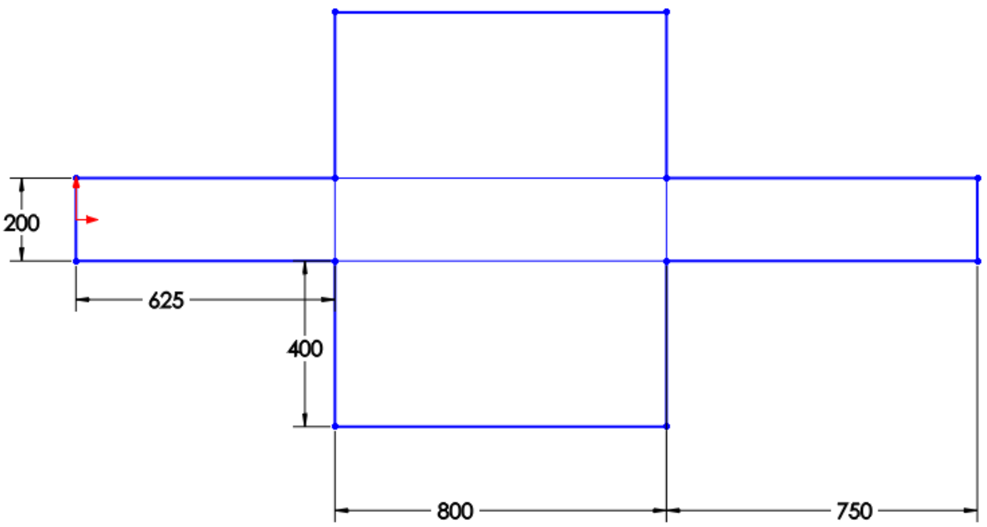


Figure 4-1 Conceptual base design dimension

4.1.2 Circular cross section over square cross section

The circular cross section has higher strength to flex and torsional twisting than square cross section. Circular tube requires less material making it light and cheaper. Circular tubes are easier to bend, thread, seal, and insulate, making them ideal for variety of applications. Cylinders are stronger than squares, giving them consistent strength all around. The major disadvantages with circular tubes are fitting and welding which requires a lot of time and patience.

Compared with solid shaft, hollow shaft offers following advantages:

- The stiffness of the hollow shaft is more than that of solid shaft with same weight.
- The strength of hollow shaft is more than that of solid shaft with same weight.
- The natural frequency of hollow shaft is higher than that of solid shaft with same weight.

Disadvantages over solid shafts:

- Hollow shaft is costlier than solid shaft.
- The diameter of hollow shaft is more than that of solid shaft and requires more space.

4.1.3 Design of Hollow tube

The design of hollow shaft consists of determining the correct inner and outer diameters from strength and rigidity considerations. Such shafts are subjected to axial tensile force, bending moment, torsional moment or combination of these loads.

Let us assume,

$$d_i/d_0 = C$$

where, d_i = inside diameter of the hollow shaft (mm)

d_0 = outside diameter of the hollow shaft (mm)

C = ratio of inside diameter to outside diameter

d. Maximum stress

When the hollow shaft is subjected to bending moment, the bending stresses are given by,

$$\sigma_b = M_b*y/I$$

where, M_b is bending moment

For hollow circular cross-section,

$$I = \pi*(d_0^4 - d_i^4)/64$$

or, $$I = \pi* d_0^4*(1 - C^4)/64$$

and $$y = d_0/2$$

So, $$\sigma_b = 32\ M_b / (\pi* d_0^3*(1 - C^4))$$

Similarly, when the hollow shaft is subjected to torsional moment, the torsional shear stress is given by,

$$\tau = 32\ M_t / (\pi * d_0^3 * (1 - C^4))$$

where, M_t is torsional moment

Now, we apply "Maximum Principal Stress Theory" of failure.

i.e., $\sigma = \{16\ M_b / (\pi * d_0^3 * (1 - C^4))\} + [\{16\ M_b / (\pi * d_0^3 * (1 - C^4))\}^2 + \{16\ M_t / (\pi * d_0^3 * (1 - C^4))\}]^{1/2}$

or, $\sigma = 32\ M_b / (\pi * d_0^3 * (1 - C^4))$ Since, $M_t = 0$

Also, $\sigma = S_{yt}$ / (factor of safety)

Therefore, S_{yt} / (factor of safety) $= 32\ M_b / (\pi * d_0^3 * (1 - C^4))$ …………………. (a)

e. Bending Moment

We assume that the base of the frame experiences the maximum load at static normal loading conditions of accessories along with the weight of passenger. We take half of the symmetrical preliminary design of frame base.

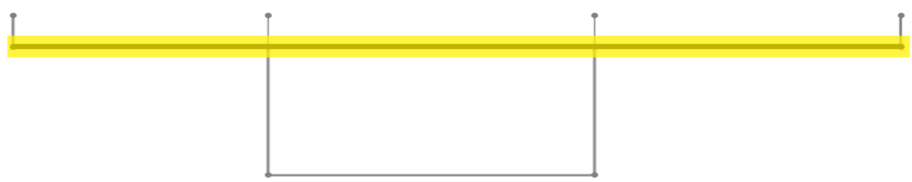


Figure 4-2 Half symmetric view of the base

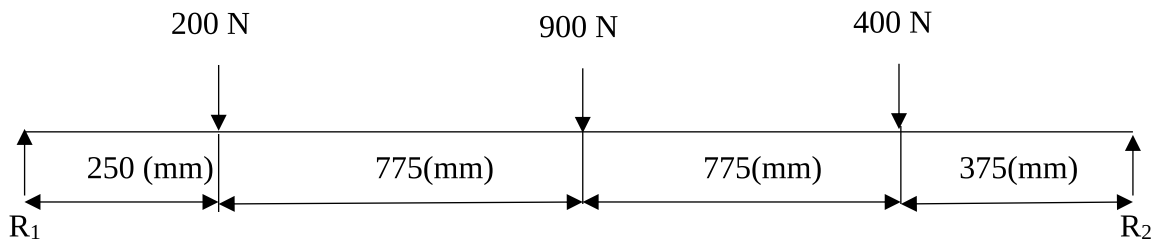
The highlighted straight beam will experience the maximum stress. So, we assume it as a single linear unit.

The weight of rack and pinion in automobile steering assembly varies with the size of the vehicle. The average approximate weight of steering assembly is found to be 15 kilograms in light 4 wheels vehicles. So, the point load of 200 Newton is experienced

250 millimeters right of the left end. The load of 800 Newton to 775 millimeters right then is due to seat assembly and weight of a passenger. The weight of differential and battery setup is assumed to create a point load of 400 Newton at 375 millimeters left of the right end. The loading conditions of the frame could be summarized as:

- 400 N at 0.25 m from front end
- 1800 N at 1.025 m from front end
- 800 N at 1.8 m from front end.

So, the free body diagram for the loading conditions is:



According to the laws of equilibrium,

$R_1 + R_2 = 200\ N + 900\ N + 400\ N$

$R_1 + R_2 = 1500\ N$

Taking Moment at R_1 (clockwise)

$0 = (200 * 250 + 900 * 1025 + 400 * 1800 - R_2 * 2175)$ N-mm

Therefore, $R_2 = 778\ N$

$R_1 = 722\ N$

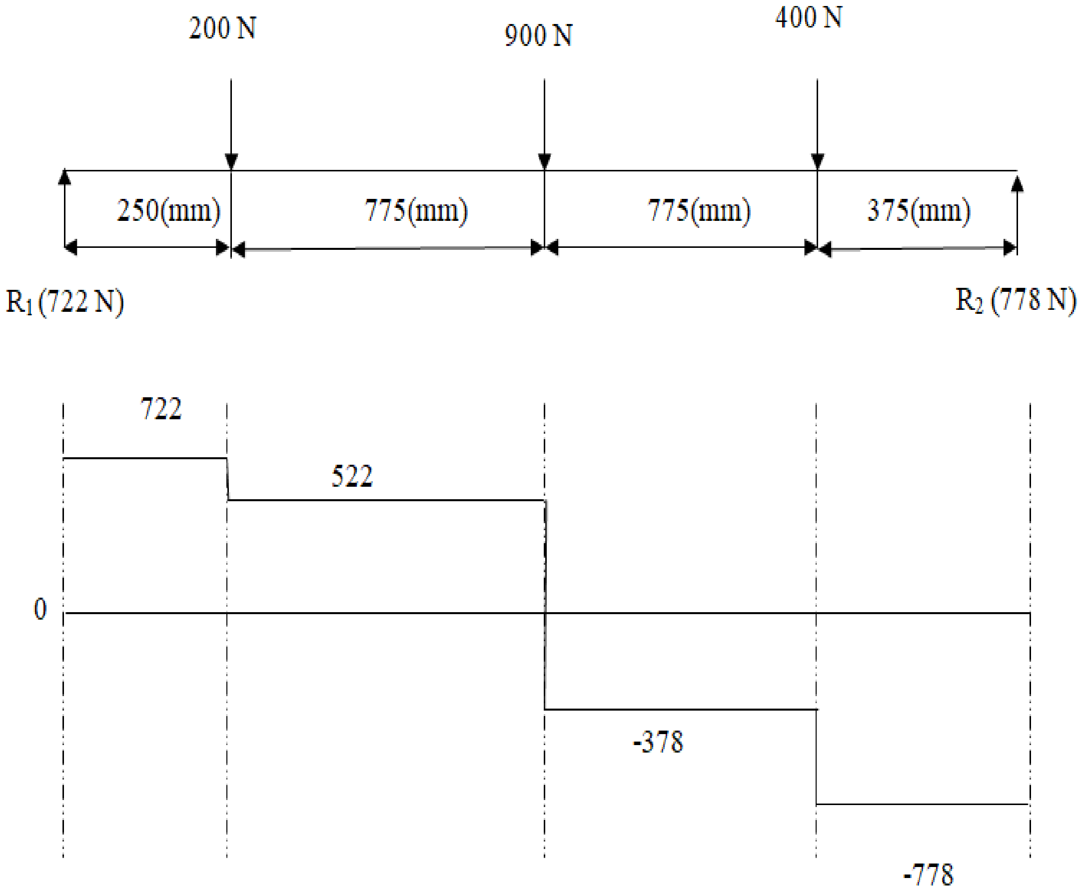


Figure 4-3 Shear force diagram (N)

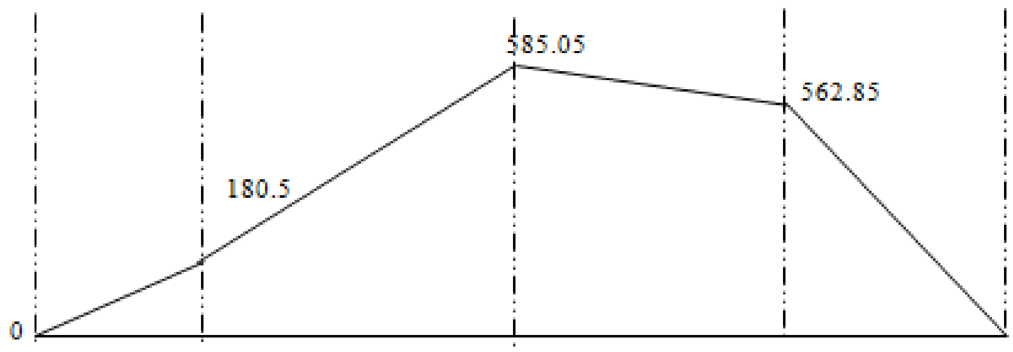


Figure 4-4 Bending Moment diagram (N -m)

f. Calculation of minimum diameter:

Let us assume the ratio of diameter of tube C be 0.8 and consider factor of safety to be 2.

The maximum moment of 585.05 N-m is seen on the frame.

Now, we have,

M_b = 585.05 N-m = 585050 N-mm

S_{yt} = 351.57 MPa = 351.57 N mm^{-2} (material selection)

Now, putting these values in …………………. (a)

We get,

S_{yt} / (factor of safety) = 32 M_b / (π* d_0^3*(1 - C^4))

351.57 / 2 = 32 * 585050 / (π* d_0^3*(1-0.8^4))

Or, d_0 = 30.62 mm and thickness 3 mm

So, the minimum outer diameter of 30.62 millimeters is required for the design of the frame. Considering the availability in the market, the external diameter of 33.7 mm and thickness 2.60 mm will be used for the fabrication of frame.

4.2 Calculation of Bending Strength

AISI 1020

Outer diameter (Do) = 33.75 mm

Inner diameter (Di) = 28.55 mm

E = 200 GPA

SY = 351.57 MPA

Thickness (t) = 2.6 mm

Radius (R) = 16.875 mm

4.2.1 Moment of Inertia

$I = \pi/64\ (Do^4 - Di^4)$

$I = \pi/64\ (33.75^4 - 28.55^4) = 31075.814\ mm^4$

4.2.2 Bending Strength

Bending strength = (Syt * I /C)

= (351.57 * 31075.814)/ (16.875) N mm

= $6.4 * 10^5$ N mm

4.3 Weld Design

4.3.1 Electrode Selection

The following points should be considered when selecting the proper coated electrode:

1. Match the mechanical properties of the base metal.
2. Match the chemical composition of the base metal as closely as possible.
3. Match electrode to available power supply — AC or DCEN (electrode negative) or DCEP (electrode positive).
4. Match the electrode to the position of the weld.
5. Observe joint design — use deep penetrating electrodes for tight fit-up and unbeveled joints. Use light penetration electrodes for poor fit-up and thin material.
6. Observe service conditions and select electrode accordingly - high temperature, low temperature, corrosive atmosphere, impact loading. These conditions are best met by using low hydrogen electrodes.
7. Consider welding costs - highest deposition rate is in the flat position. Use high iron powder electrodes to further increase deposition. By far, the largest factor is labor and overhead.

4.3.2 Safety Factor for welding

Normal stress developed on the weld bead is given by:

$$\frac{\sigma}{r} = \frac{M}{I}$$

Where, $I = 0.707\ \pi r^3 t$

Shear stress developed on the weld bead is given by

$$\frac{\tau}{r} = \frac{T}{J}$$

Where, $J = 2I = 2 * 0.707\ \pi r^3 t$

Torque (T) = F*r

So,

$$\sigma = \frac{585.05*10^9}{\pi\left(\frac{33.75}{2}\right)^2 *0.707*2.60}$$

=355 MPa

Also,

$$\tau = \frac{T*r}{J}$$

$$= \frac{(144*109)}{\left(2\pi\left(\frac{33.75}{2}\right)^2 *0.707*2.60\right)}$$

= 43.78 MPa

Now, maximum principle stress on the weld is given by the resultant of the shear stress and normal stress and is calculated as

$$\sigma_p = \frac{\sigma}{2} + \sqrt{\left(\left(\frac{\sigma}{2}\right)^2 + \tau^2\right)}$$

$$= \frac{355}{2} + \sqrt{\left(\frac{355}{2}\right)^2 + (43.78)^2}$$

= 360 MPa

Now, using this principle stress and finding factor of safety for different welding material. We have assumed the weld material or filler rod material indicated as 60Ksi, 70 ksi and 80 ksi. These ranges of the welding material are being suggested for the welding process of pipes or tubes.

a. Using 60 ksi material,

Ultimate strength (σut) = 413.7 MPa

So, factor of safety = ($\sigma ut/\sigma p$) = (413.7 / 360) = 1.149

b. Using 70 Ksi material

Ultimate strength (σut) = 482.67 MPa

So, factor of safety = ($\sigma ut/\sigma p$) = (482.67 / 360) = 1.34

c. Using 80 Ksi material

Ultimate strength (σut) = 551.6 MPa

So, factor of safety = ($\sigma ut/\sigma p$) = (551.6 / 360) = 1.53

Among these suggested categories of the welding material, the highest rating material is used i.e. 80 Ksi rated material as it provides high safety factor.

4.4 Analysis of Vehicle Frame

The frame is so designed to sustain significant loads and impacts (front, side, rollover and torsional). It should provide optimum damage protection to the drivers as well as to the vehicle itself in case of various impacts as mentioned below. To be absolutely sure about the amount of safety a frame provides to the drivers, the CAD models of the frames employing the above four materials are simulated using software for-

- Front impact
- Side impact

• Roll over

• Torsion

Assumptions considered

• The frame material is considered to be isotropic and homogenous.

• The tube joints are considered to be perfect

4.4.1 Frontal Impact Analysis

Assumptions & Considerations

Front impact test was carried out assuming that the vehicle is having a mass of 450 kg (including the mass of both the drivers and dead weight) and travelling with a velocity of 40 kmph colliding head on with a stationary wall.

- All the points at wheel mountings are fixed with zero degree of freedom constraints.
- The load is applied at the front nodes
- The mesh generated during this process is beam mesh with defined mesh control and fine density.
- As per the rule specified by NHTSA (National Highway Safety Traffic Administration), the minimum impact time is of the order 250ms.

Calculation of Impact Forces

Mass of vehicle including mass of drivers (m) =450kg

At the time of impact, velocity of the vehicle (v_{in}) = 40kmph = 11.11 mps

Final velocity of the vehicle (v_f) = 0 kmph

Impact time (Δt) = 0.25 sec

We know,

Work done (W) = Change in Kinetic Energy

$W = ½ * m * (v_{in}^2 – v_f^2)$

$W= ½* 450 * (11.11^2 - 0)$

W=27772.223 J

Acceleration (a) = Δv/t

a = 11.11/0.25

a = 44.44 m/s^2

Applying second equation of motion

$s = ut – ½at^2$

$s = 11.11*.25 - 0.5*44.44*0.25^2$

s = 1.3887 m

Now,

Work done = Force(F) * Displacement(s)

W = F * s

F = W/s

F = 27772.223/1.3887

F = 19998.72 N

So, let front loading impact Force is 20000 N

For analysis, we will be using the rounded-up value i.e. F = 21000 N

4.4.2 Rear Impact Analysis

Assumptions & Considerations

- Rear impact test was carried out assuming that the vehicle is having a mass off 450 kg (including the mass of both the drivers and dead weight) and hit by the vehicle travelling with a velocity of 40 kmph on its rear end.
- All the points at wheel mountings are fixed with zero degree of freedom constraints.
- The load is applied on 6 nodes at any one of the sides of the frame
- The mesh generated during this process is beam mesh with defined mesh control and fine density.
- As per the rule specified by NHTSA (National Highway Safety Traffic Administration), the minimum impact time is of the order 250ms.

Calculation of Impact Forces

- During side impact, it is assumed that the efficycle is at rest and another efficycle hits the first efficycle sideways.
- Impact time in this case is taken as 0.25 seconds.
- Mass of the efficycle including mass of drivers (m) = 450kg
- Impact time (Δt) = 0.25 sec

We know,

Work done (W) = Change in Kinetic Energy

$W = ½ m (v_{in}^2 – v_f^2)$

$W= ½ * 450 * (11.11^2 - 0)$

At the time of impact, velocity of the vehicle (v_{in}) = 40kmph = 11.11 mps

W= 27772.223J

Acceleration (a) = $\Delta v/t$

a = 11.11/0.25

a = 44.44 m/s^2

Applying second equation of motion

$s = ut – ½at^2$

$s = 11.11*0.25 - 0.5*44.44*0.25^2$

s = 1.3887 m

Also,

Work done = Force (F) * Displacement (s)

W = F * s

F = W/s

F =27772.223 /1.3887

F = 19998.72 N

So, let front loading impact Force is 21000 N

F = 21000N

4.4.3 Roll Over Analysis

Assumptions & Considerations

- Rollover analysis was carried out assuming that the vehicle is having a mass of 450 kg (including the mass of both the drivers and dead weight) and the vehicle rolls upside down because of any external impact.
- All the points at wheel mountings are fixed with zero degree of freedom constraints.
- The vehicle is assumed to have been dropped from a height of 10 feet onto its top surface on the ground. This assumption is made to stimulate roll over condition
- The mesh generated during this process is beam mesh with defined mesh control and fine density.
- It is assumed that the impact time of roll over is 0.25 seconds.

Calculation of Impact Forces

Mass of the vehicle including mass of drivers (m) = 450 kg

Height from which vehicle is assumed to be dropped (h) = 10 feet = 3.048 m

Since the vehicle is dropped from a height, its potential energy changes into kinetic energy at the time of impact.

i.e. $mgh = \frac{1}{2} mv^2$

$v = (2*g*h)^{1/2}$

$v = (2*9.81*3.048)^{1/2}$

$v = 7.73$ m/s

Now,

Work done (W) = $\frac{1}{2} mv^2$

$W = \frac{1}{2} * 450 * 7.73^2$

W = 13444.4025J

Displacement (s) = t * v = 0.25 * 7.73 = 1.9325 m

We know,

Work done (W) = Force (F) * Displacement (s)

W = F * s

F = W/s

F = 13444.4025/1.9325

F = 6957 N

So, let the top loading force is 7000 N

Number of nodes at top = 4

Load per nodes at front = 7000/4 = 1750 N

4.4.4 Torsional Analysis

a) Assumptions & Considerations

- Torsional analysis was carried out assuming that the vehicle is having a mass of 240 kg (including the mass of both the drivers and dead weight). The front wheels' experience bump on one side and drop on another.
- The point at the rear swing arm hub is fixed with zero degree of freedom constraints.
- One fourth of the load applied in front impact is applied at the two side nodes in equal magnitude and opposite direction.
- The mesh generated during this process is beam mesh with defined mesh control and fine density.

b) Calculation of Impact Forces

For Torsion, the twisting effect is assumed to be one fourth of the force in front impact.

Force = (front impact force)/4 = 21000/4 = 5250 N

Number of nodes at side = 2

Load per node at side = 5250/2 = 2625 N

For the analysis, we will use the rounded-up value of 3000 Newton force.

CHAPTER FIVE
POWER REQUIREMENT

In order to calculate tractive power requirement, the net force required after subtracting the external forces acting on the vehicle is calculated. The electric motor torque is calculated from the tractive force required. The external forces like rolling resistance, road grade are subtracted. The tractive effort that can be applied to the ground is directly related to the properties of the ground. Different surfaces such as sand, asphalt and ice will have different value of friction which will limit the maximum force that can be applied to accelerate the vehicle without spinning the wheels. The tire revolution per minute (rpm) of a vehicle is dependent only on the gear ratio.

5.1 Theoretical Considerations

Sprung mass of vehicle with driver and passenger (Ms) = 450kg

Unsprung mass of vehicle (Mu) = 90kg

Total mass of vehicle (M) = 540kg

Coefficient of rolling resistance (Cr) = Cr = 0.045 (Chevrefils, 2008)

Rolling resistance (F_r) = C_r * W

= 0.045 * 540* 9.81 =238.38 N

Take Max. Velocity (v) = 60 km/hr = 16.67 m/s

Average running speed (v_{avg}) = 25 km/hr = 6.94m/s

Time to acquire (t_{avg}) = 10 sec

Acceleration (a) = $\frac{V(avg)}{t}$ = 6.94/10 = 0.694m/s^2

Acceleration Force (F_a) = M * a = 540 * 0.694 = 374.76 N

Grade (G) = 25 % = 14.036° (Assumed)

Grade force (F_g) = M*g*sin(α)

= 540 * 9.81 * sin(14.036)= 1284.79N

Total resistive force (F_{res}) = $F_a + F_g + F_r$

a. Calculation of Starting Torque

Initial force to start a vehicle at 25% grade

$F_i = F_r + F_a + F_g = (238.38 + 374.76 + 1284.79) = 1897.93$ N

Size of wheel = 0.3175 m

Torque required to move the wheel (T_{axle})

$T_{axle} = F_i$ * Radius of wheel = 1897.93 * 0.3175 = 602.60 Nm

b. Climbing the grade

Assume, Grade = 25 % at V= 30 km/hr

Grade Force (F_g) = M*g*sin(α) = 1284.79 N

Rolling Resistance (F_r) = M*g* C_r = 238.38 N

Total force (F_t) = $F_g + F_r$ = (1284.79 +238.38) = 1523.17 N

V = 25 km/hr = 6.944 m/s

Power required to climb the grade ($P^1_{required}$) = F_t * V

= 1523.17* 6.944

= 10572Watt

=10.57W

c. Continuous power required in plain road

Average Speed (V_{plain}) = 40 km/hr = 11.11 m/s

The force required to overcome resistance (F_t) = $F_a + F_r$

= 374.76 + 238.38 = 613.14N

Power required ($P^2_{required}$) = F_t* V_{avg} = 613.14 * 11.11 = 6811.985 W= 6.81 KW

From the calculation above, the overall tractive power requirement to run the EATV at average velocity of 25 km/hr on 25% grade is found to be 10.57kW. This value does not account for the losses accumulated in the drive train, heat losses and motor losses and thus a margin must be added to account for these losses.

d. Motor Torque and Rated Speed Calculation

Total Gear ratio of transmission (G) = 12.44 (Dana, 2020)

Efficiency of transmission (η_G) = 0.7

Wheel rotational Speed rpm @ 40 km/hr (V)

$V = \frac{\pi DN}{60}$

$N_{axle} = \frac{V*60}{\pi D} = 332rpm$

Torque provided by the motor (T_{motor}) = $(1/G*\eta_G)*T_{axle}$ = (1/12.44*0.7)*456.576

= 52.43 Nm

Angular Velocity of the motor shaft (N_{motor}) = $G*N_{axle}$ = 12.44 * 332 = 4130 rpm

5.2 Selection of Motor

Based on the calculation of Power and torque requirement, the selection of the motor is carried out. Since the minimum power requirement is found to be 10.563kW, the BLDC motor of rated power of 12kW is selected. The table below describes the electrical and mechanical parameter of the selected motor (Electric Motorsport, 2021).

Table 5-1 Specification of Selected BLDC Motor

Parameters	**Values**
Rated Voltage	96V
Current	125 Amps (180 Amps DC into the motor control)
Peak Current	420 Amps for 1 minute (600 Amps DC into the motor control)
Rated Power	12kW
Peak Power	30kW
Speed	5000rpm
Rated Torque	80Nm
Peak Torque	160Nm
Efficiency	> 90%
Dimensions	30*30*25cm
Weight	16kg
Cooling	Open Frame

5.3 Selection of Motor Controller

The motor drive has multiple components that require selection. Based on axial flux motors Sinusoidal (sine wave) controllers such as the Sevcon Gen4 are to be used. The Gen4 range represents the latest design in compact controllers. These reliable controllers are intended for on-road and off-road electric vehicles and feature the smallest size in the industry for their power capacity.

Due to the high efficiency it is possible to integrate these controllers into very tight spaces without sacrificing performance. The design has been optimized for the lowest possible installed cost while maintaining superior reliability in the most demanding applications. (Electric Motorsport, 2021) . Other details of the motor controller is annexed.

Table 5-2 Specification of the Sevcon Gen4 S4 110V 300A UVW Motor Controller 634A13210

Nominal Voltage	110 VDC
Operating Voltage	48-150 VDC
Current (120s)	300A
Boost (10s)	360A
Cont. (60min)	120A

5.4 Sizing of battery bank

Current drawn by motor to run at maximum power is given by the relations given below.

I = P/V

= Power of motor/ battery voltage

= 12000/96 =125 Ampere (A)

The range of the E-ATV is assumed to be 40km in once full charged. The average speed of the E-ATV is taken as 25 km/hr. The operating time of the vehicle can be calculated to be 1.6 hour. Since, the battery is discharged to only about 80% of its rated capacity. Discharging it below this point can damage the battery pack.

Take, Efficiency (η) = 80%

Battery Capacity required = (Amp*operating hour)/ η = (125*1.6)/0.8 = 250 A-hr

Battery Energy Capacity (kW-hr) = (Ah*V)/1000 = (250*96)/1000 = 24kW-hr

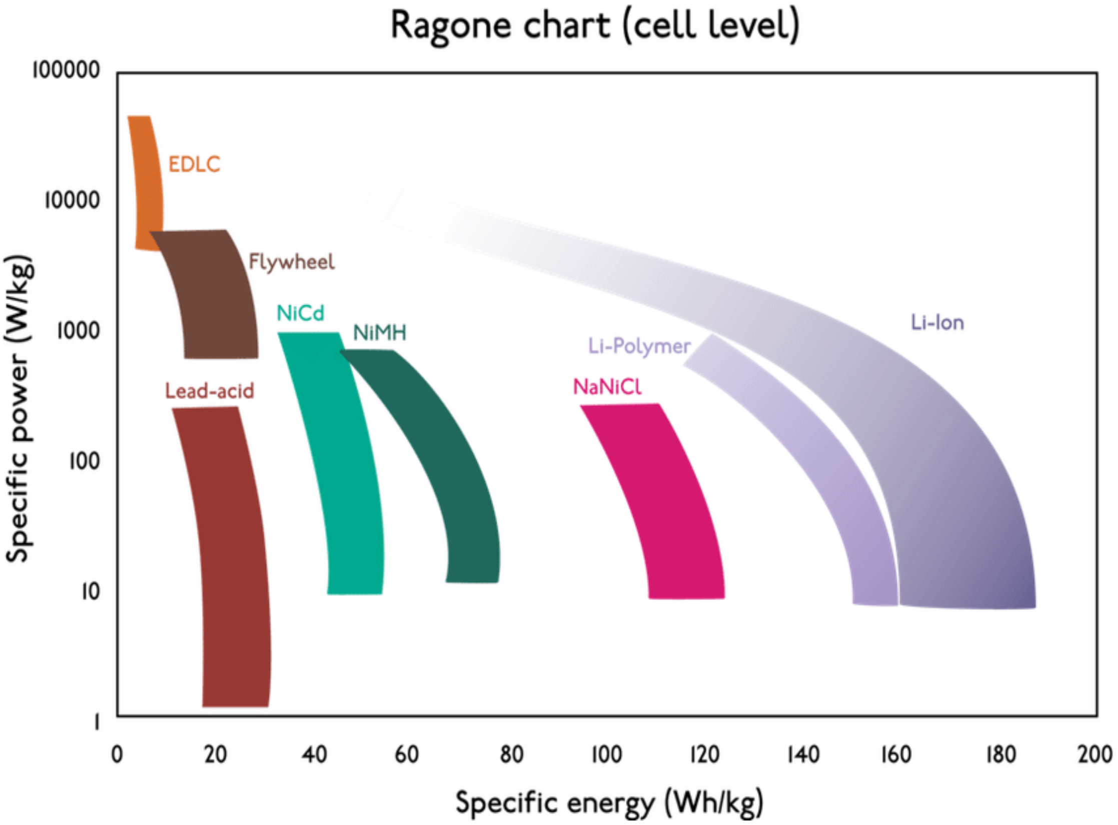


Figure 5-1 Ragone diagram cell level adapted from Van Den Bossche 2009 (Bernardini, 2015)

CHAPTER SIX
SUSPENSION SYSTEM

6.1 Selection of suspension system

Suspension systems are selected on the basis of requirement and geometry for our E-ATV, availability and ease of manufacture. All - terrain vehicles are used in uneven terrain and off roads so suspension comes into major play. They require independent suspension system. So considering all the parameters double wishbones system is selected.

Double wishbone suspension system is independent suspension system consists of two lateral arms (upper arm and lower arm). The two arms are used to strengthen the suspension in double wishbone geometry. The upper arm is usually shorter to induce negative camber as the suspension jounces. It consists of a knuckle attached on the wheel hub. The knuckle has two mounting points for A-arms. The first is the upper kingpin point and second is lower kingpin point. The two arms are attached to those points. The two arms have ball joints in the end to allow the movement in multiple directions. On the other side the arms are connected to the chassis at two points. Now the vertical movement is controlled by the shock absorber and coil spring which are generally mounted on the lower arm. This type of suspension is mostly used in rear wheel drive vehicles. Unlike MacPherson strut this type of suspension provides negative camber gain to fulfill bump travel. It has got superior load handling characteristics.

Design of the geometry of double wishbone suspension system along with design of spring plays a very important role in maintaining the stability of the vehicle. For the spring selection, the various types of spring geometry were explored in the market. Due to the various factors like simplicity, affordability and simplicity helical spring was chosen and design was based on that. The helical spring is easy to manufacture and of high reliability. Deflection of spring is linearly proportional to the force acting on the spring

6.2 Material selection for wishbone suspension

The most primary need for design and fabrication of suspension system is the material selection for the wishbones. Initially the materials are selected based on the various aspects like strength, availability, affordability and so on. The selection also depends upon the various factors like carbon content, ease of welding, less weight and high strength. Different types of carbon steel were compared for the wishbones. Various factors were considered. The comparison between several carbon steel is given below:

Table 6-1 Comparison between different steels

Parameters	AISI 1020	AISI 1018	AISI 1040	AISI 4130
Tensile strength	294.74	440	620	560
Yield Strength	294.4	317	415	460
Hardness(BHN)	211	126	201	217
Carbon Content	0.23	0.18	0.4	0.3

After performing the comparison, the best steel selected is AISI 1040. Based on its properties, the design of wishbones is carried out.

For the manufacture of the spring, spring steel is selected. It is the used in the manufacture of automotive and industrial suspension springs. These are easily formed, shaped ad post heat treated. They have high yield strength and tensile strength.

6.3 Design procedure

The design procedure for the chosen suspension system is divided into two stages:

a. Primary design
    i. Basic design and development of suspension system components.
    ii. Modified design parameters based on approximation of dynamic condition.
    iii. Static testing and analysis.
b. Secondary design
    i. Dynamic testing and analysis on ANSYS and MATLAB.
    ii. Modification of design parameters based on dynamic testing results.

6.4 Design of the suspension spring

The various design consideration made for our EATV suspension are as follows:

Total sprung mass of EATV = 450 Kg

Mass distribution (Front: Rear) =40:60

Mass per Front Wheel = 40% of 180

=180 Kg

Mass per each front wheel = 180/2

=90 Kg

Mass per rear wheel = 60 % of 450 Kg

= 270 Kg

Mass per each rear wheel = 270/2

= 135 Kg

Track Width = 1389.6 mm

Wheel base = 1600 mm

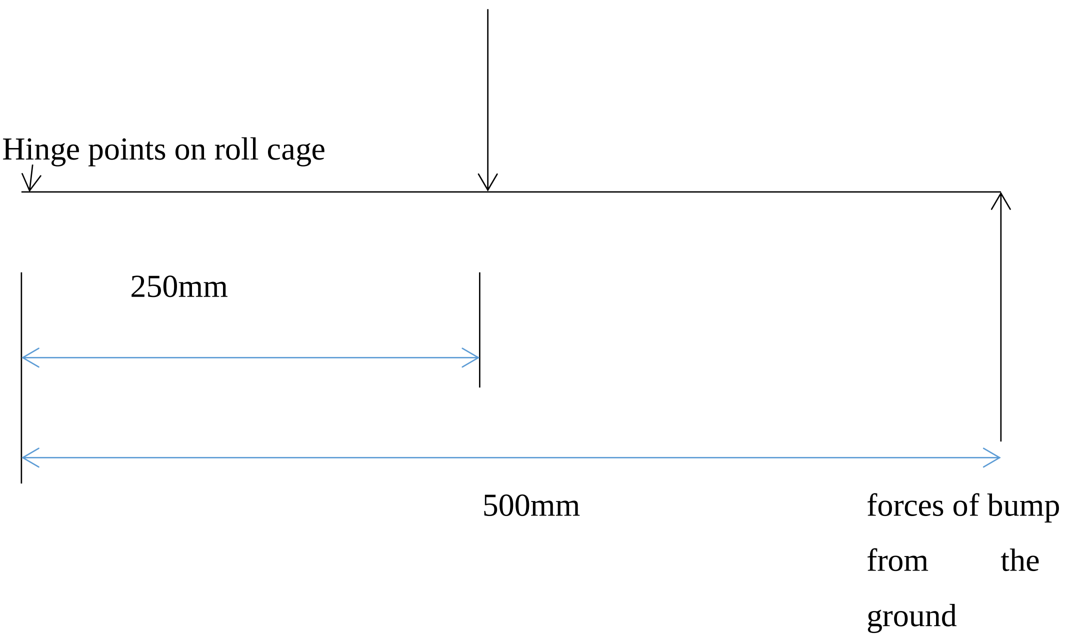
Static to dynamic factor = 2.5

Angle of inclination of the strut from the horizontal = 60^o

Point of attachment of strut from the chassis end = 250 mm

a) Design of front Spring

Dynamic load acting on the spring



Reaction force acting from the ground on wheel = Mass per wheel * 9.81

= 90 * 9.81 = 882.9 N

Horizontal distance of reaction force from hinge point = 500 mm

By taking moment about the hinge point,

882.9 * 500 = spring force * 250

Therefore, Spring force = 1765.8 N

Now, Dynamic spring force = 2.5 * 1765.8 = 4414.5 N

Now for our EATV spring travel should be minimum of 4 inch (101.6 mm) (Akshay G Bharadwaj, 2016)

Required spring stiffness = Dynamic spring force / spring deflection

= 4414.5 / 101.6

= 4.45 N/mm

Material : oil hardened steel wire of grade 2 (helical spring)

Ultimate tensile strength = 11000 (Aditya Sinha, 2018)

Shear stress = 0.5 * tensile strength

= 0.5 * 1100 = 550 N/mm^2

Taking spring index(C) = 8

By Wahl's factor,

$$K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$$

= 1.184

Now, Shear stress = $K * \frac{8PC}{\pi d^2}$

On solving, we get d = 8.799 mm ≈ 10mm

Spring wire diameter = 10mm

Mean coil diameter(D) = C * d = 8 * 10

= 80 mm

Modulus of rigidity (G) = 84000 N/mm^2

Spring deflection = $\frac{8PD^3N}{Gd^4}$

where N is the number of coils.

Now, substituting the value of all parameter we get,

N = 11.799 ≈ 12 coils

Assuming spring has square and ground ends so,

Total number of coils (N_t) = N + 2 = 12+2

= 14 coils

Solid length of coil = N_t * d = 14 * 10

= 140 mm

Total axial gap = (N_t – 1) * 1 = (14 – 1) * 1

= 13 mm

Free length = solid length + total axial gap + spring deflection

= 140 + 13 + 101.6 = 254.6 mm

Pitch of coil = $\frac{\text{Free length}}{N_{t-1}}$ = 19.58 mm

Outside diameter of coil (D_O) = D + d = 80+10 = 90 mm

Inside diameter of coil (D_i) = D - d = 80-10 =70 mm

b) Design of Rear spring

Maximum number of required parameter are same as front spring for the design of rear suspension spring. Only the reaction force acting on the wheel is different since there is unequal weight distribution in front and rear suspension.

Reaction force acting on the ground from wheel = Mass per wheel * 9.81

= 135 * 9.81 = 1324.35 N

By taking moment about hinge point,

1324.35 * 500 = spring force * 250

spring force = 2648.7 N

Dynamic spring force = 2.5 * 2648.7 = 6621.75 N

Required spring stiffness = Dynamic spring force / spring deflection

= 6621.75 / 101.6

= 65.17 N/mm

By previous calculation, k = 1.184

Now, Shear stress = $K * \frac{8PC}{\pi d^2}$

on solving, we get d = 10.78 mm ≈ 12mm

Therefore, Mean coil diameter = C * d

D = 8 * 12 = 96 mm

Spring deflection = $\frac{8PD^3N}{Gd^4}$

where N is the number of coils.

Now, substituting the value of all parameter we get,

N = 9.44 ≈ 10 coils

Assuming spring has square and ground ends so, Total number of coils (N_t) = N + 2 = 10+2

= 12 coils

Solid length of coil = N_t * d = 12 * 12

= 144 mm

Total axial gap = (N_t – 1) * 1 = (12 – 1) * 1

= 11 mm

Free length = solid length + total axial gap + spring deflection

= 144 + 11 + 101.6 = 256.6 mm

Pitch of coil = $\frac{\text{Free length}}{N_{t-1}}$ = 23.33 mm

Outside diameter of coil (D_O) = D + d = 96 + 12 = 108 mm

Inside diameter of coil (D_i) = D - d = 96 - 12 = 84 mm

CHAPTER SEVEN
STEERING SYSTEM

7.1 Selection of steering system

We selected rack and pinion manual steering system over power steering although it has several advantages which includes:

a. Preventing the wheels from transferring the load to steering column.
b. Reducing the driver's fatigue.
c. Low input torque and continuous steering function.
d. Oil output directly proportional to the steering speed.

Although the power steering system is advantageous, it has some complications regarding design and increase in cost. Fluid leakage problem is also seen in power steering system. So, for EATV we have used rack and pinion manual steering system which overcome the limitations of power steering. The power steering system also requires several components to operate (hydraulic or electric). Hence, manual steering system is used keeping all those criteria in consideration.

7.2 Rack and Pinion Manual Steering system

Rack-and-pinion steering is the most common type of steering on cars, small trucks and SUVs. It is a simple mechanism. A rack-and-pinion gear set is enclosed in a metal tube, with each end of the rack protruding from the tube. A rod, called a tie rod, connects to each end of the rack.

The pinion gear is attached to the steering shaft. When you turn the steering wheel, the gear spins, moving the rack. The tie rod at each end of the rack connects to the steering arm on the spindle.

Basically, a rack and pinion are two gears which make up a gear set. These gears are positioned within a metal tube. On each side of the tube, you can see the rack coming out. There is also a component called a tie rod which ties off the ends of the rack and connects the steering arm and spindle together. The function of the tie rod is to relay the force of the rack gear (or steering center link) to the steering knuckle. This is done as the tie rod connects with the steering arm. As a result, the steering wheel can turn as you try to rotate it with your hands.

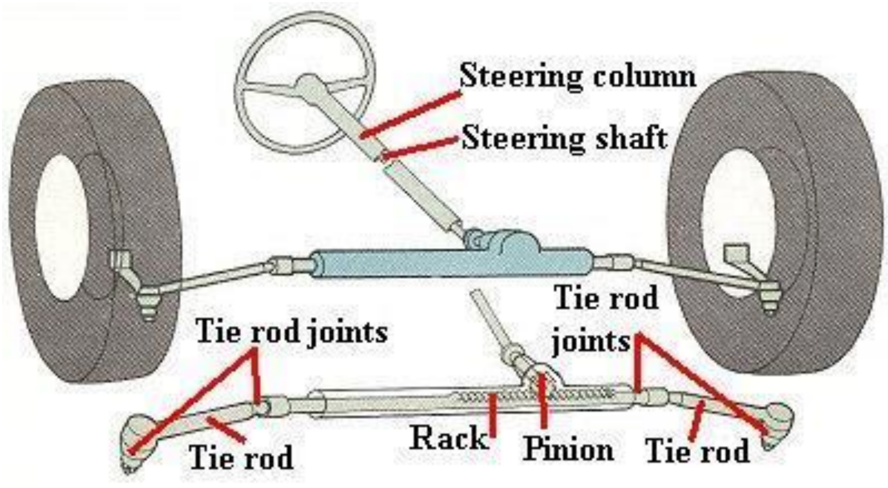


Figure 7-1 Rack and Pinion Steering System (The Motorom Budsman, 2019)

Components of rack and pinion steering system:

The rack and pinion steering system consist of following main parts.

i) Steering wheel: It is a driving wheel which is a type of steering control in vehicles. It is manipulated by the driver.

ii) Steering shaft: It is a long shaft which connects the steering wheel to pinion gear. It transfers the rotation of steering wheel to rotation of pinion.

iii) Rack and Pinion assembly: A rack-and-pinion gearset is enclosed in a metal tube, with each end of the rack protruding from the tube.
The rack-and-pinion gearset does two things:
   - It converts the rotational motion of the steering wheel into the linear motion needed to turn the wheels.
   - It provides a gear reduction, making it easier to turn the wheels.

iv) Tie rod: A rod, called a tie rod, connects to each end of the rack. The tie rods are the connection from your steering system to your wheels. They are responsible for moving your wheels when you steer and for making turning possible.

7.3 Calculations

Table 7-1 Input Parameters

Wheel base (b)	1600 mm
Track width (a)	1389.6 mm
Pivot to pivot distance (c)	989.6 mm

i) For inner and outer turning radius

Ackermann angle (α) = $\tan^{-1}\frac{c}{2b}$

$= \tan^{-1}(989.6/3200)$

$= 17.18^\circ$

For perfect Ackermann,

$\cot(\text{outer angle}) - \cot(\text{inner angle}) = \frac{\text{Track width}}{\text{Wheel base}}$

or, $\cot\phi - \cot\theta = \frac{c}{b}$

Assuming the inner steer angle to be 35˚.

So,

$\cot\phi - \cot 35^\circ = \frac{989.6}{1600}$

or, $\cot\phi = 2.0466$

and,

$\phi = 26.04^\circ$

Hence the outer turning angle is 26.04˚.

Also,

The total steering angle = outer angle + inner angle

$= 35^\circ + 26.04^\circ$

$= 61.04^\circ$

Now,

Turning radius of inner wheel (Ri) = $\frac{L}{\sin\theta}$

$= \frac{1600}{\sin(35^\circ)}$

$= 2789.5$ mm

Turning radius of outer wheel (Ro) = $\frac{L}{\sin\phi}$

$$= \frac{1600}{\sin(26.04°)}$$

$$= 3644.66 \text{ mm}$$

Also from geometrical interpretation, we found the value of turning radius of center of gravity.

So, Turning radius of C.G. = 2900mm

Table 7-2 Technical Specifications of Steering System

Velocity at turning	5 m/s
Turning radius of C.G. (Rcg)	2900 mm
Turning radius of inner wheel (Ri)	2789.5 mm
Turning radius of outer wheel (Ro)	3644.66 mm
C.G. height	595 mm
Radius of wheel	317.5 mm
Caster Angle	11˚
Load on each tire	90 kg
Scrub radius	48.95 mm
Coefficient of friction	0.6
Diameter of steering wheel	300 mm
Pinion radius	16 mm
Angle between tie rod and rack in top view	41.6330˚
Angle between tie rod and rack in front view	16.50˚
Ackermann Angle	17.18˚

ii) Rack and pinion Calculations: (Aksh Patel, 2019)

The value of rack travel was taken as 150 mm and assume the number of turns (lock to lock).

For the dimension of rack and pinion, we have the formulae below;

$$\text{Rack travel} = \pi \times d_{Pi} \times n$$

where,

n = no. of steering wheels turns = 1.5

d_{pi} =diameter of pinion

$$150 = \pi \times d_{Pi} \times 1.5$$

$$d_{Pi} = 32 \text{ mm}$$

Now,

From the standard values of module (as per the convenience). Let us assume 'm'=1.5

From this, the teeth number on pinion is easily calculated as;

$$\text{Module} = \frac{\text{diameter of pinion}}{\text{no. of teeth of pinion}}$$

$$M = \frac{d}{t_{pi}}$$

$$1.5 = \frac{32}{t_{pi}}$$

$$t_{pi} = 22 \quad \text{(teeth on pinion)}$$

The module for rack and pinion will be same in order for messing of both the teeth:

$$M = \frac{d_{pi}}{t_{pi}} = \frac{d_{ra}}{t_{ra}}$$

The pitch circle diameter for rack is infinite but we have the travel of pinion (i.e. Rack travel). From that value, converting the rack travel length into circular pitch diameter (only for calculation purpose).

$$\text{Rack travel} = \pi \times d(t)$$

$$d(t) = \frac{150}{3.14} = 48 \text{ mm}$$

The value of $d(t)$is determined. So, from that gear ratio of rack and pinion is obtained. Thus obtained gear ratio is used to find the total no. of teeth on rack as;

$$\text{Gear ratio} = \frac{\text{no. of teeth on pinion}}{\text{no. of teeth on rack}} = \frac{\text{diameter of pinion}}{\text{diameter of rack}}$$

i.e.

$$\frac{32}{22} = \frac{48}{t_{ra}}$$

$$t_{ra} = 33 \text{ (teeth on rack)}$$

iii) Steering Force Calculations (Mr. Kshitij N Sable, 2019)

For Top view:

The angle between Steering Arm & Axle=43.30˚

The angle between tie rod & rack=41.6330˚

For Front view:

The angle between Steering arm & axle=0˚ (Horizontal)

The angle between tie rod & rack=16.50˚

The force required to push/pull tire at static conditions is μmg.

$$F_{tyre} = \mu \text{ (mg)} \qquad \text{------ (i)}$$

$$= 0.6 \times 90 \times 9.81$$

$$= 529.74 \text{N}$$

Resolving the obtained force into steering arm & tie rod inclinations.

From equation (i),

Steering arm force = 529.74/cos(43.3˚) ------- (ii)

= 727.89 N

From equation (ii)

Tie rod force = [steering arm force/ Top cosine difference of angle between steering arm &axle and tie rod &rack]/ Front cosine of angle between tie rod and rack ------- (iii)

= [727.89/cos(43.3˚- 41.633˚)] / cos(16.5˚)

= 759.47N

From equation (iii)

Force on rack and pinion = Tie rod force× Front cosine of angle between tie rod and rack × Top cosine of angle between tie rod and rack ------ (iv)

= 759.47 × cos(16.5˚) × cos(41.633˚)

= 544.26 N

From equation (iv)

Torque on pinion = Force × diameter of pinion

= 544.26 × 32

= 17416.32 N-mm

Torque on steering wheel = Force applied × Radius of steering wheel

or, 17416.32 = F_{wheel} × 150

So, Force on steering wheel (F_{wheel}) = 116.11 N

CHAPTER EIGHT
BRAKING SYSTEM

8.1 Design of Braking System for EATV

Braking system in any vehicle is the most important subsystem of the vehicle. The needs of braking system are to increase the safety and maneuverability of the vehicle by dynamically locking all the four wheels on any surfaces. The all-terrain vehicle requires efficient braking system based on the space availability inside the vehicle. It requires proper selection of components and their placing in minimum space possible. The design of braking system of our EATV is based on the braking system of SAE Baja. The main objective of our design is to produce light weight brake assembly, reliable throughout the different terrains and effective to satisfy the wheel lock conditions.

The first things to look at while designing the brake system is the selection of actuation system to transmit the braking force from driver to the wheels. Actuation system in brake can be mechanical, hydraulic or pneumatic as discussed earlier. Hydraulic brakes are easily available, cheap, reliable, and effective and provide larger breaking force. After selecting hydraulic system, we must decide whether to use disc brake or drum brake. We have already discussed about the working mechanism and components used of these brakes in details. We are using disc brake for our EATV due to the following advantages of disc brake over drum brake:

a. Disc brakes can dissipate heat better than drum brakes can. It works by converting motion energy into heat energy. Disc brake is placed outside the so can interact easily with atmospheric air.
b. A condition known as brake fade happens very less in disc brake.
c. Disc brakes also perform better in wet weather, because centrifugal force tends to fling water off the brake disc and keep it dry.
d. Longer life due to fewer moving parts and more effective heat dissipation.
e. Much less sensitive to premature lock-up or wheel skid than corresponding uni-servo or duo-servo hydraulically operated drum brakes.
f. Easy maintenance compared to drum brakes. Entire rotor can be removed for maintenance without removing hub.

For our EATV, we are certain that disc brakes are going to work much better. EATV will have to go through long, rough, terrain, wet endurance course which will give challenges to our braking system and from research and study it has been proved that the disc brake will work much better for such applications. There are various parts in our braking system. The detail description of the parts is discussed below.

a. Brake Pedal

   The brake pedal uses leverage to transfer the effort from the driver's foot to the master cylinder. The mounting of brake pedal needs various considerations. Brake pedals should be mounted securely, free from any excessive sideways movement, and at a height and angle that will allow the driver to quickly move from pressing the accelerator pedal to applying the brakes. The length of pedal is defined by pedal ratio to be used and the space available below and behind the pedal for swinging motion. We are using the pedal with the brake pedal ratio of 5:1. It means pedal will amplify the effort applied by the driver to 5 times.

b. Master Cylinder

   Master cylinder pressurizes breaking fluid with the help of drivers input. It is a control device that converts non-hydraulic pressure (commonly from a driver's foot) into hydraulic pressure. The various types of master cylinder provide designer with range of choices for intended applications. One of the variant is tandem master cylinder. The tandem master cylinder has two separate hydraulic chambers. It is characterized by two pistons operating in series within a common bore. Tandem master cylinders generate the hydraulic pressure for the two separate brake circuits. It provides split circuiting for each pair of wheels. This allows driver to break even if one of the circuit fails due to leakage. BAJA uses master cylinder having bore diameter 19.05mm. Hence for our EATV tandem master cylinder with bore diameter 0.75 inch or 19.05mm is select.

c. Brake Rotor

   Since we are using disc brake, brake rotor or disc is used which is mounted on the wheel hub. Its main function is to retard the vehicle by applying torque to the wheel. It's usually made of cast iron because it's hard-wearing and can resist high temperatures. The major parameter while selecting the size of rotor is diameter of rim. The rotor has to fit properly in packaging space available. Ventilated disc rotor can be used to improve cooling. These slots are designed to use centrifugal force to cause airflow when the disc is rotating. Disc rotor can be customized according to the needs or can also be purchased from the market. In our EATV we are using the disc rotor of diameter 300mm and effective radius is 125mm. We have selected the size of disc by market analysis.

d. Brake Calipers

Once the master cylinder and size of brake disc is decided another important things to decide is brake calipers. The brake caliper houses brake pads and pistons. Its job is to slow the car's wheels by creating friction with the brake rotors. The brake caliper fits like a clamp on a wheel's rotor to stop the wheel from turning when you step on the brakes. Due to dynamic load transfer to the front wheels while braking, calipers should be selected so it provides proper brake biasing on the front and rear wheels. The size of calipers on the front should be greater than rear to lock all the wheels at same time. So according to our calculation size of calipers on front and rear wheels are 38mm and 20mm respectively.

e. Brake pads

Disc brake pads consist of friction material bonded onto a steel backing plate. The backing plate has lugs that locate the pad in the correct position in relation to the disc. Calipers are usually designed so that the condition of the pads can be checked easily once the wheel has been removed, and to allow the pads to be replaced with a minimum of disassembly. The materials of brake pads have direct effect on the braking torque. So it must be selected which provides higher frictional force to the rotor. We are choosing the pads having coefficient of friction 0.4.

f. Brake Fluid

Brake fluid is a type of hydraulic fluid used in the hydraulic brake and hydraulic clutch application in automobiles, motorcycles and light trucks. It is used to transfer force into pressure and to amplify braking force. It carries braking force from master cylinder to the brake pads. There are different types of brake fluid used in automobiles vehicles. Brake fluid used in automobiles with its properties is given below.

Property	DOT 3	DOT 4	DOT 5	DOT 5.1
Dry Boiling Point (°C)	≥ 205	≥ 230	≥ 260	≥ 260
Dry Boiling Point (°F)	≥ 401	≥ 446	≥ 500	≥ 500
Wet Boiling Point (°C)	≥ 140	≥ 155	≥ 180	≥ 180
Wet Boiling Point (°F)	≥ 284	≥ 311	≥ 356	≥ 356
Viscosity at 100 °C (cSt)	≤ 1.5	≤ 1.5	≤ 1.5	≤ 1.5
Viscosity at -40 °C (cSt)	≥ 1500	≥ 1800	≥ 900	≥ 900
Color	Colorless to Amber	Colorless to Amber	Purple	Colorless to Amber
Typical Chemistry	Glycol Ether	Glycol Ether + Borate Ester	Silicone	Borate Ester + Glycol Ether

Figure 8-1 Properties of Brake Fluid (Mototribology, 2018)

8.2 Brake Biasing

Brake biasing is the percentage of total braking force applied to the front wheels. It means applying variable force for both the front and rear axles of vehicle. This is done to obtain maximum braking efficiency and to have good stability and reduce stopping distance in account for variable road conditions. The braking force should increase as the weight on wheels increased. More braking force is required to the wheels having more weight. While applying brake the weight transfers from the rear to the front. This increases the traction in the front end reduces it in the rear. If we have equal brake balance than rear wheels will lock up first and vehicle tends to spin. Optimal braking is only obtained when all the four wheels lock up at the same time so for that brake biasing is important. It helps the driver maintain control while braking. A 60-70% bias is common on RWD street vehicles. This means that the front brakes provide 60-70% of the total braking force. FWD vehicles cars have up to 80% of bias. The biasing depends upon the dynamic weight transfer during braking. Biasing can be obtained by using two master cylinders of different bore sizes. If we are using two master cylinders than bias bar also called as balance bar is used. Disc calipers are easily available and are easily replaceable also so we are considering using different size brake caliper for biasing. And also we are using only one cylinder so for proper biasing we are using larger size of caliper in front wheels and smaller size in rear wheels. Other methods of increasing front bias are:

- Larger front rotors
- Front pads with more friction
- Larger front calipers pistons
- Smaller rear rotor
- Rear pads with loss friction
- Lowered suspension
- More weight on the rear axle
- Less sticky tires on the front wheels.

8.3 Brake Calculations

Total mass of the EATV= 450 kg

Weight of vehicle= 450 × 9.81= 4414.5N

Wheel base (L) = 1600mm

Longitudinal distance of Centre of gravity from the front axle (b) = 900mm

Longitudinal distance of Centre of gravity from the rear axle (c) = (1600 - 900) mm

= 700 mm

The weight on the front axle and rear axle in the static condition can be calculated as:

Front axle static load (W_f) $= w\times\frac{b}{L} = 4414.5\times\frac{900}{1600}$

$$W_f = 2483.16 \text{ N}$$

Rear axle static load (W_r) $= w\times\frac{c}{L} = 4414.5\times\frac{700}{1600}$

$$W_r = 1931.34 \text{ N}$$

Dynamics:

Height of Centre of gravity (h) = 595 mm

Coefficient of friction between road and tires (μ_r) = 0.6 (Aman Sharma, 2018)

Diameter of tire = 25 inch

Radius of tire (R_t) = $^{25}/_{2}$ = 12.5 inch

$R_{t=}$ 12.5×25.4

$R_{t=}$ 317.5mm

Now, based on our calculations;

Maximum velocity of our EATV = 40 $\mathrm{km/hr}$

$=11.11\mathrm{m/s}$

Let's us assume we are considering 2 seconds to stop our vehicle while braking to maximum efficiency.

Deceleration (α) = $\frac{v}{t}$ = $\frac{11.11}{2}$ = 5.55 $\mathrm{m/s}^2$

During braking, weight is transferred to front axle. So,

Front axle dynamic load (W_{fd}) = $W_f + \frac{w\alpha h}{gL}$

Rear axle dynamic load (W_{rd})=$W_r - \frac{w\alpha h}{gL}$

Where,

h = height to CG

α = deceleration of vehicle

L= length of wheel base

Front axle dynamic load (W_{fd}) = $W_f + \frac{w\alpha h}{gL}$

$$= 2483.16 + \frac{4414.5\times5.55\times0.595}{9.81\times1.6}$$

W_{fd} =3412.75 N

Rear axle dynamic load (W_{rd})=$W_r - \frac{w\alpha h}{gL}$

$$= 1931.34 - \frac{4414.5 \times 5.55 \times 0.595}{9.81 \times 1.6}$$

$$W_{rd} = 1001.75 \text{ N}$$

Now, we can calculate the required frictional torque on brake disc;

Frictional torque to lock front wheels (T_f) = $\mu_r \times R_f \times W_{fd}$

Where,

μ_r= Coefficient of friction between tire and road

R_f = Radius of front tire

$$T_f = \mu_r \times R_f \times W_{fd}$$

$$= 0.5 \times 3412.75 \times 0.3175$$

$$T_f = 541.77 \text{ NM}$$

Similarly,

Torque required to lock rear wheels (T_r)=$\mu_r \times R_r \times W_{rd}$

$$= 0.5 \times 1001.75 \times 0.3175$$

$$T_r = 159.03 \text{ Nm}$$

<u>Selection of caliper</u>

Average force applied by driver to pedal = 300 N

For maximum hydraulic brake pedal ratio used is 5: 1 (Vivek Singh Negi, 2017)

So,

Brake pedal force (F_{pd}) = $300 \times \frac{5}{1}$

$$= 1500 \text{ N}$$

We are using Bosc Tandem master cylinder of bore diameter $3/4$ inch(0.75) equals to 19.05 mm

Area of master cylinder (A_{mc}) = $\frac{\pi d^2}{4}$

$$= \frac{\pi \times (19.05)^2}{4}$$

$$A_{mc} = 285.02 \times 10^{-6}\ \text{mm}^2$$

Master cylinder pressure (P_{mc}) = $\frac{F_{bp}}{A_{mc}}$

$$= \frac{1500}{285.02}$$

$$= 5.26 \times 10^{6}\ \text{N/mm}^2$$

Here, we are selecting two different caliper piston for front and rear wheels for brake biasing.

Let A_{cp} be the area of caliper piston.so,

Force generated by caliper (F_{cal}) = $P_{mc} \times A_{cp}$

$$= 5.26A_{cp} \times 10^{6}\ \text{N}$$

Here we use a double piston fixed caliper.

So,

Caliper clamp load (F_{cl}) = $2 \times F_{cal}$

$$= 2 \times 5.26A_{cp} \times 10^{6}\ \text{N}$$

$$F_{cl} = 10.52A_{cp} \times 10^{6}\ \text{N}$$

Let coefficient of friction between brake pad and rotor disc be 0.4 i.e. $\mu_{bp} = 0.4$

Force on disc by brake pad ($F_{friction}$) = $\mu_{bp} \times F_{cal}$

$$= 0.4 \times 10.52A_{cp} \times 10^{6}\ \text{N}$$

$= 4.208A_{cp} \times 10^6$ N

Outside usable diameter of disc (D) = 300mm

Inside usable diameter of disc (d) = 200mm

Effective radius = (D+ d)/4

= 125mm

Effective radius of brake disc (R_{eff}) =125mm = 0.125m

Torque of rotor = $F_{friction} \times R_{eff}$

$= 4.208A_{cp} \times 10^6 \times 0.125$

$= 526 \times 10^3 A_{cp}$

For front wheel,

The frictional torque on rotor must be greater than the torque required to lock the wheels

i.e. $526 \times 10^3 A_{cp} > 541.77$

$A_{cp} > 10.3 \times 10^{-4}$

$$\frac{\pi d_f^2}{4} > 10.3 \times 10^{-4}$$

$$d_f^2 > \frac{10.3 \times 10^{-4} \times 4}{\pi}$$

$d_f > 0.03621\text{m}$

$d_f > 36.21\text{mm}$

Similarly, for rear wheel

$526 \times 10^3 A_{cp} > 159.03$

$$\frac{\pi d_f^2}{4} > 3.02 \times 10^{-4}$$

Also,

$$d_f^2 > \frac{3.02 \times 10^{-4} \times 4}{\pi}$$

$d_f > 0.01960\text{m}$

$d_f > 19.60\text{mm}$

So, for proper effectiveness of brake and to lock both the wheels at same time we use different caliper at front and rear wheels.

For front wheel larger caliper size of 38 mm is used and for rear wheel smaller caliper of 20 mm are used. This is considered taking in mind the availability in market.

Now,

Total torque on rotors $= T_f + T_r$

$$= 715.36 \times 10^3 \times \frac{\pi 0.038^2}{4} + 715.36 \times 10^3 \times \frac{\pi 0.020^2}{4}$$

$= 761.80\text{N}$

Also,

Total braking force applied on wheel on wheel $(F_B) = \frac{T}{R_w}$

$= \frac{761.80}{0.3175}$

$= 2399.37\text{N}$

Deceleration (a) = $\frac{F_B}{M}$

$= \frac{2399.37}{450}$

$= 5.33\ \text{m/s}^2$

Also,

Stopping time (T) = $\frac{v}{a}$

$=\frac{11.11}{5.33}$

$=2.08s$

Which is close to our assumption:

Stopping distance = $\frac{v^2}{2a}$

$=\frac{11.11^2}{2\times 5.33}$

$=11.58m$

Kinetic energy developed = $\frac{1}{2}mv^2$

$=\frac{1}{2}450 \times 11.11^2$

$= 2777.222$ j

Power generated $=\frac{KE}{T}$

$= \frac{2777.222}{2.08}$

$= 1335.20$ Watt

CHAPTER NINE
FRAME ANALYSIS

9.1 Grid Test Analysis

For the better result analysis in the commercial analysis software it is necessary to make the fine elements or mesh in the frames. To determine the better mesh size for the appropriate result, first the frame geometry is meshed for the various sizes using the meshing software. Then the frames are simulated under same loading on commercial analysis software, until the simulated results are similar to each other. The result of the grid test analysis performed for our frames are shown in table below.

Table 9-1 Grid test analysis result

Element size	Number of elements	Max. stress (Mpa)	Max. deformation(mm)
30	50434	1192	4.5767
25	73183	2021.6	4.9897
14.5	98123	2674.4	7.8382
14	100566	2951	8.8769
13.5	124902	3053	9.0767

The above obtained result for the grid test result is plotted in the graph for maximum stress developed against number of the elements, and maximum deflection against number of the elements used for the analysis. The graphs are shown in figures below:

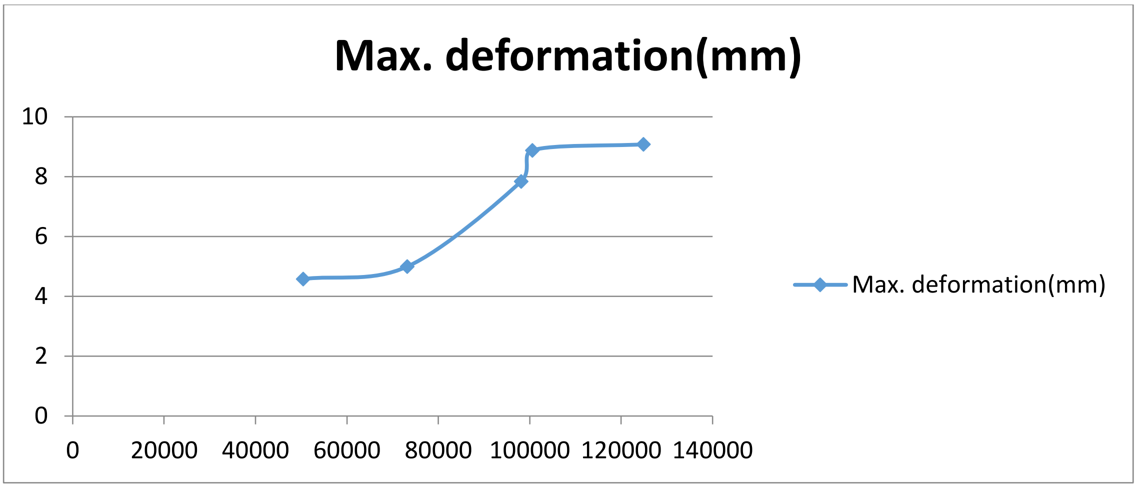


Figure 9-1 Deformation vs number of element graph

In the figure above, the graph is plotted between the maximum deformations in the frame for the certain forces. The graph shows that as the number of the elements increases for the analysis increases the maximum deformation developed also increases with high slope until the number of elements increased to 100566 elements. Then on further increasing the number of elements the increase in the deformation is very low that means the increase in number of elements from here doesn't affect the result very much. So we proceed with the corresponding element size i.e. 14mm for complete meshing and other analysis done on impact tests.

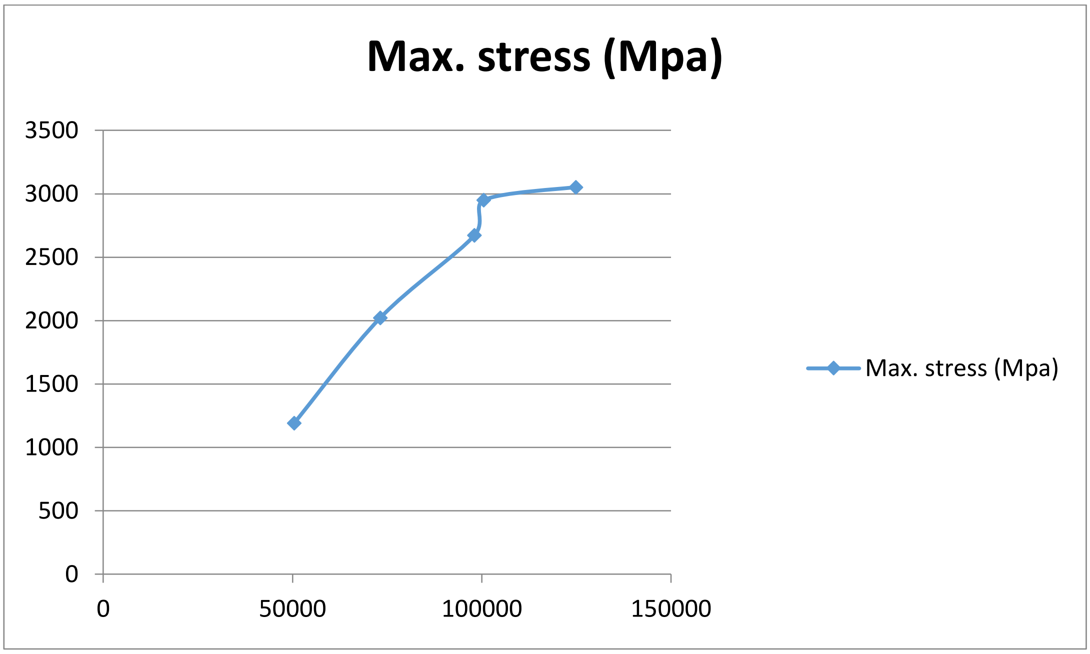


Figure 9-2 Stress vs. number of element graph

In the figure above, the graph is plotted between the maximum stresses generated in the frame for the certain forces. The graph shows that as the number of the elements increases for the analysis increases the maximum stress developed also increases with high slope until the number of elements increased to 100566 elements. Then on further increasing the number of elements the increase in the stress is very low that means the increase in number of elements from here doesn't affect the result very much. So we proceed with the corresponding element size i.e. 14mm for complete meshing and other analysis done on impact tests.

9.2 Normal Loading

The base of the EATV is subjected to the static loading of the passengers, batteries, steering system and many other components which provides various stress and deflections. The stress and deflection developed on the base of two modal of the EATV's are shown in following figures and summarized in table.

a. Model-One

For the analysis of the static dynamic loading condition of the model one, following mesh parameter and number of nodes and elements are used for the analysis software

i. Mesh size: 14mm
ii. Number of elements:100566
iii. Number of nodes: 244276

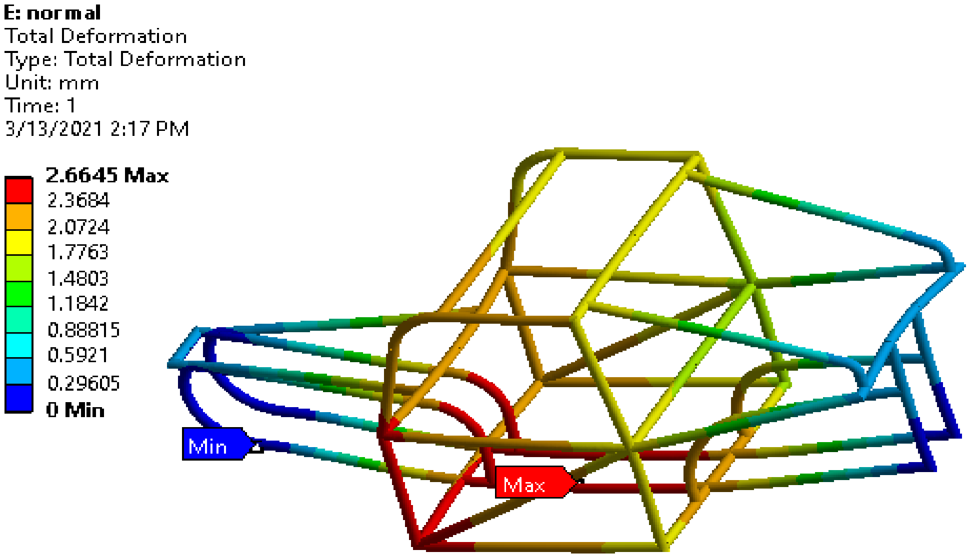


Figure 9-3 Total deformation for normal loading

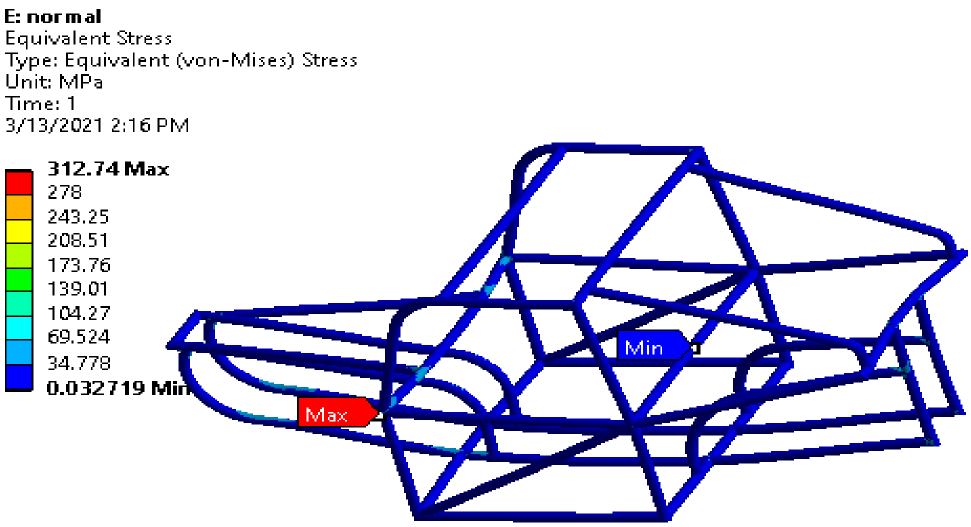


Figure 9-4 Stress distribution for Normal loading

b. Model-Two

For the analysis of the static and dynamic loading condition of the model to following mesh parameter and number of nodes and elements are used for the analysis software

i. Mesh size 14mm
ii. Number of elements 168597
iii. Number of nodes 325209

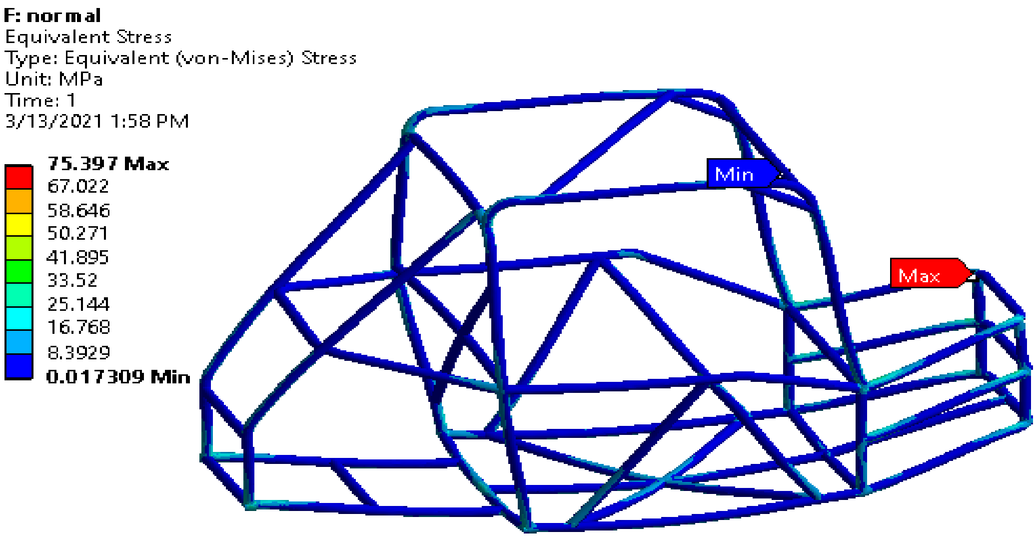


Figure 9-5 Stress distribution on normal static loading

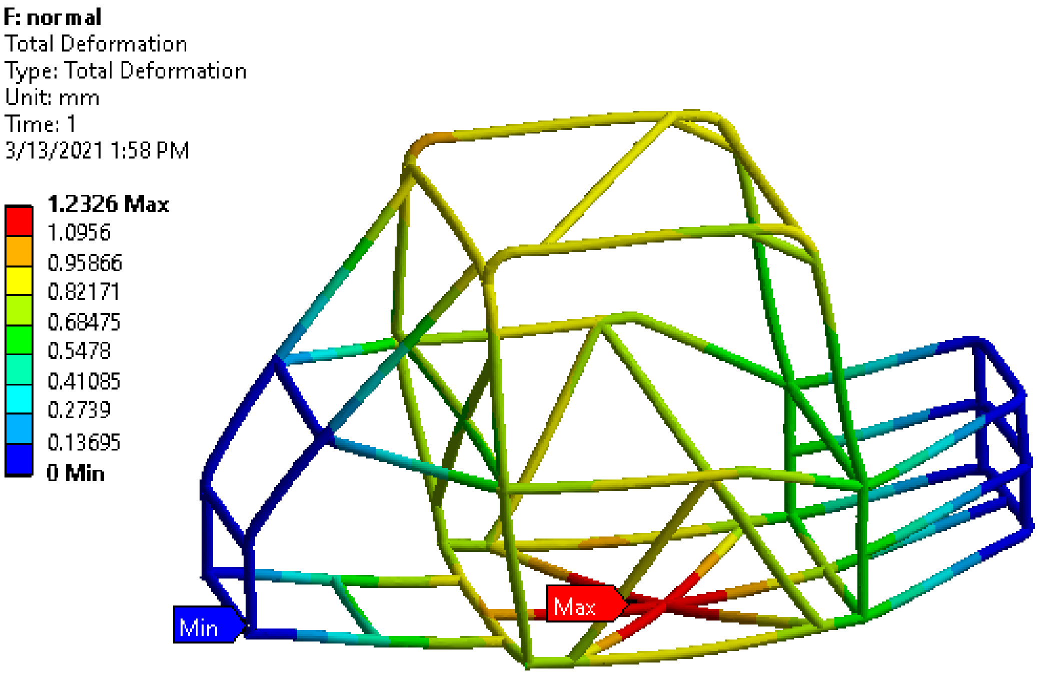


Figure 9-6 Total deflection for Normal loading

The maximum Von-Mises stress distribution and maximum total deformation derived above are tabulated for the two models and are shown below:

Table 9-2 Normal loading to the base of EATV models

Type	Model-One	Model-two
Von-Mises Stress (MPa)	312.74	75.397
Deflection	2.66	1.2326
FOS	1.122	4.655

From the above table and the diagrams showing the stress distribution and the maximum total deformation, we calculated the factor of safety for the roll cages for the normal static loadings which shows that the model-two has better result regarding the factor of safety when comparing to the maximum yield strength of the material used i.e. 351MPa.

9.3 Model-One Roll Cage

For the model-one roll cage we used the impact forced calculated in the chapter 4.4 for various impacts like frontal impact, rear impact, roll over and torsional impact. For the analysis the mesh properties is used similar to the static analysis. And the result obtained for the test is tabulated.

9.3.1 Front Impact

a. Von-Mises stress distribution

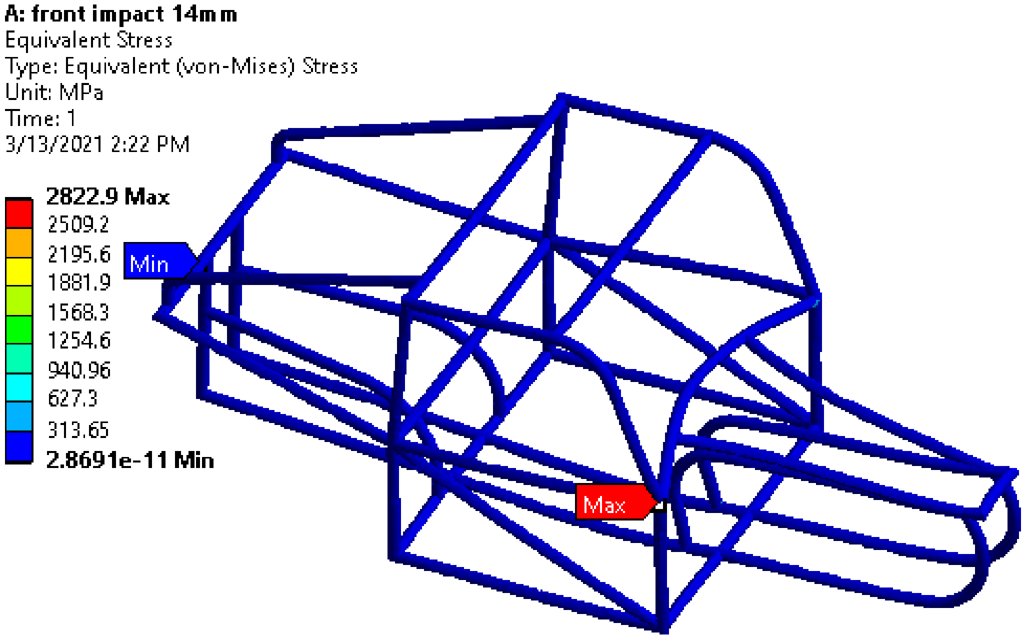


Figure 9-7 Front Impact Von-Mises Stress distribution

b. Deformation Analysis

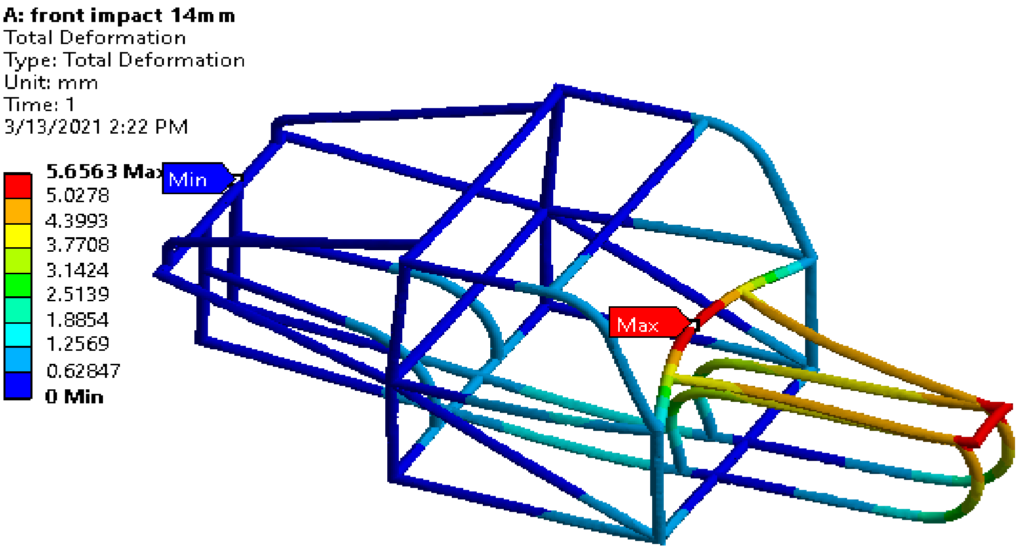


Figure 9-8 Front Impact Deformation Test

c. Stress Distribution in Cabin area

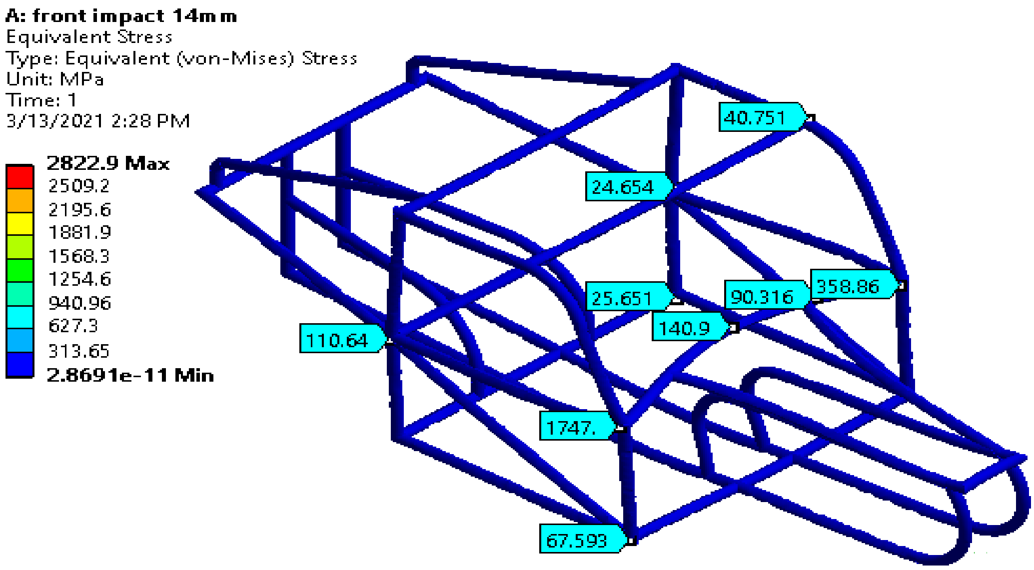


Figure 9-9 Stress distribution in Cabin

Analysis Results:

Maximum stress in Roll cage = 2822.9 MPa

Maximum Deformation in Roll cage = 5.65 mm

Maximun Cabin stress = 2822.9 MPa

9.3.2 Rear Impact

a. Deformation Analysis

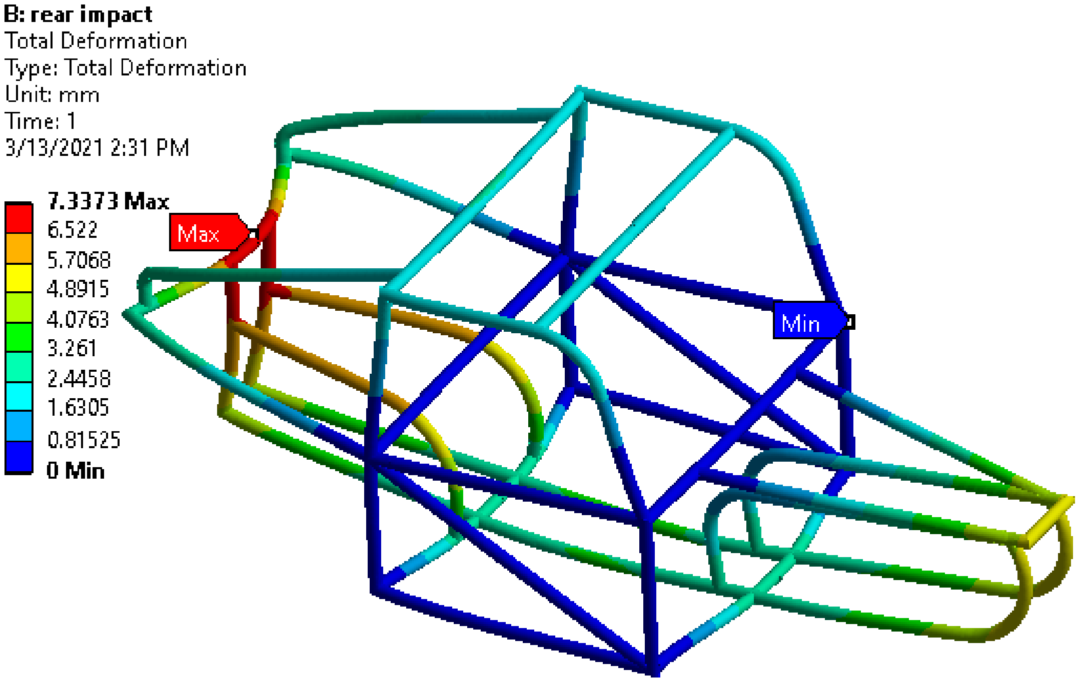


Figure 9-10 Rear Impact deformation Test

b. Von-Mises stress distribution

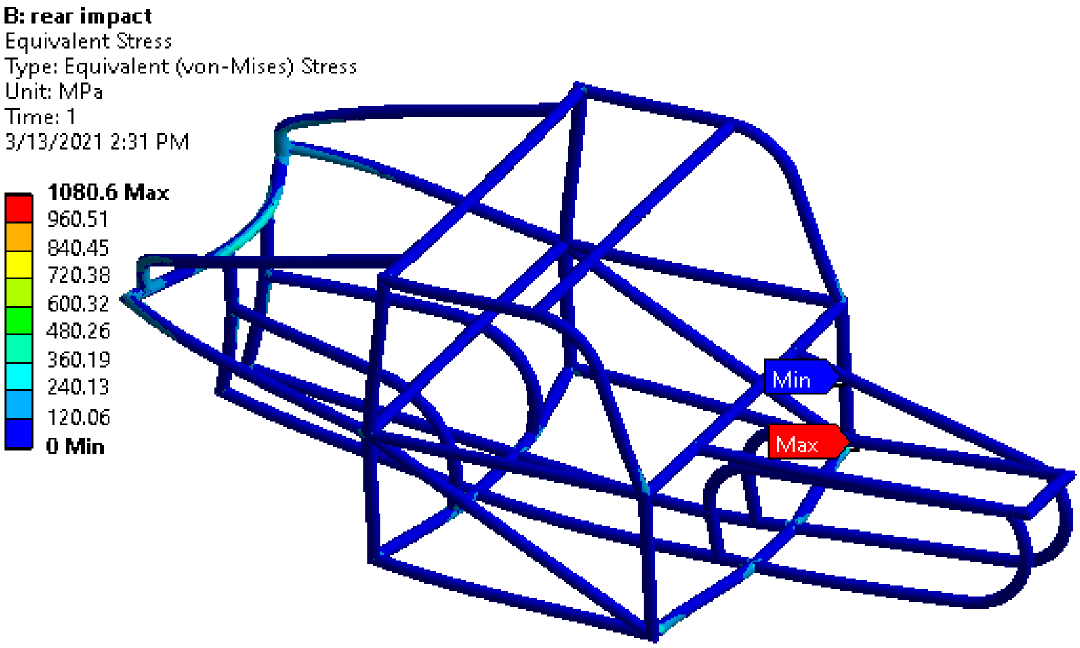


Figure 9-11 Rear impact Von-Mises stress distribution

c. **Stress Distribution in Cabin area**

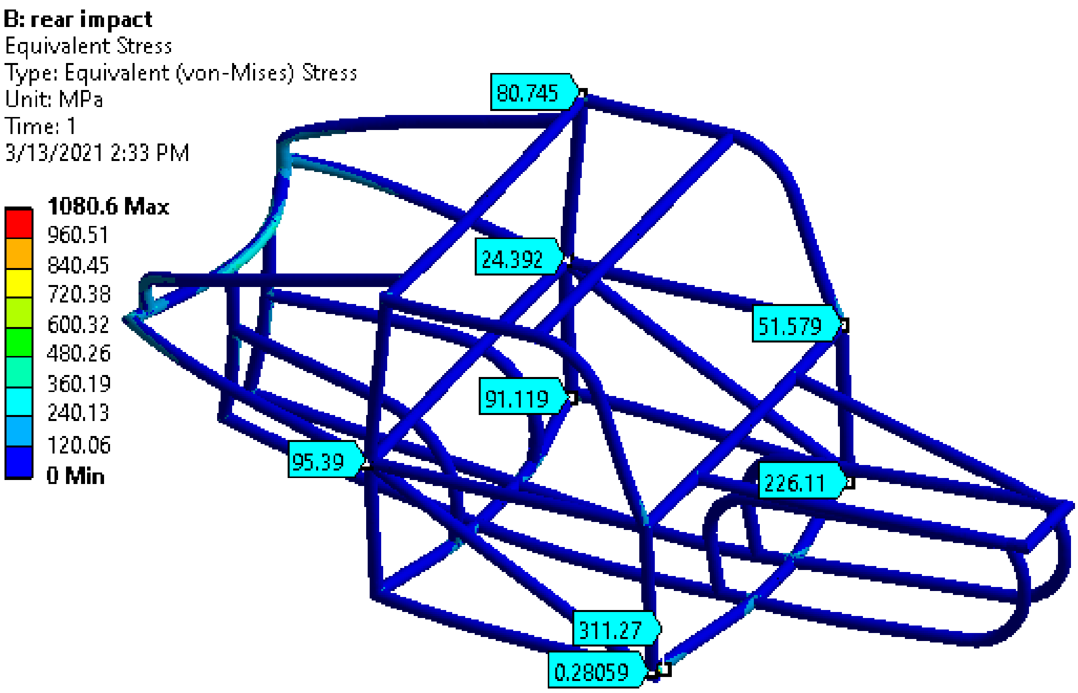


Figure 9-12 Stress distribution in cabin for rear impact

Analysis Results:

Maximum stress in Roll cage = 1080.6 MPa

Maximum Deformation in Roll cage = 7.337 mm

Maximimun Cabin stress =311.27 MPa

9.3.3 Roll Over

a. Deformation Analysis

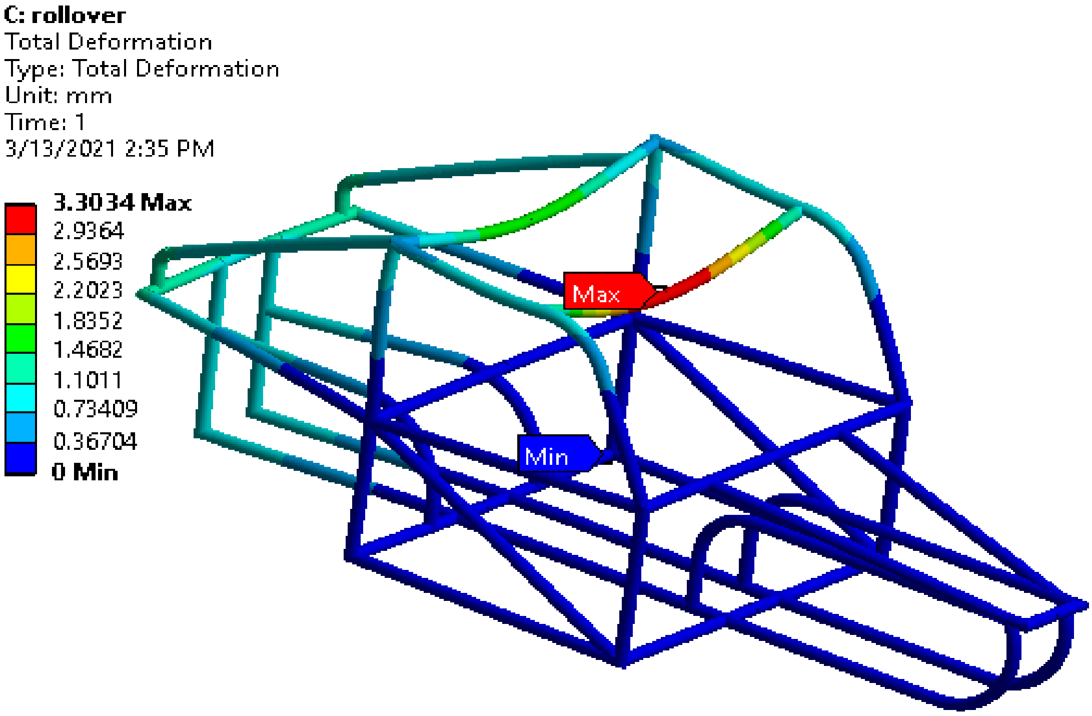


Figure 9-13 Roll over deformation test

b. Von-Mises stress distribution

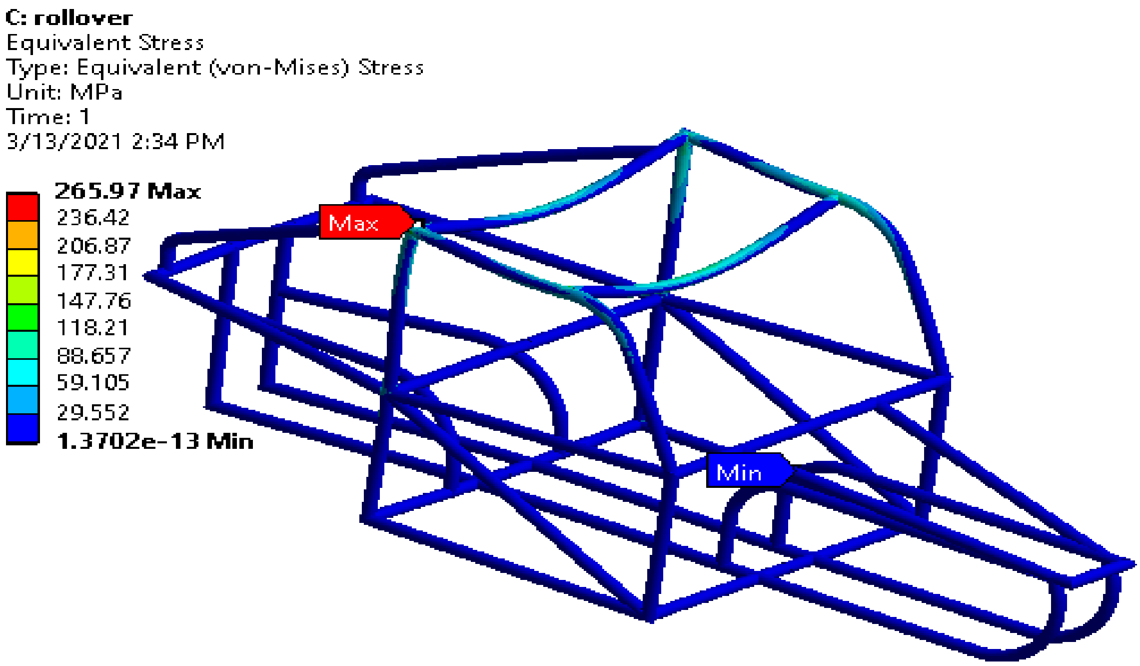


Figure 9-14 Rollover Von-Mises stress distribution

c. Stress Distribution in Cabin area

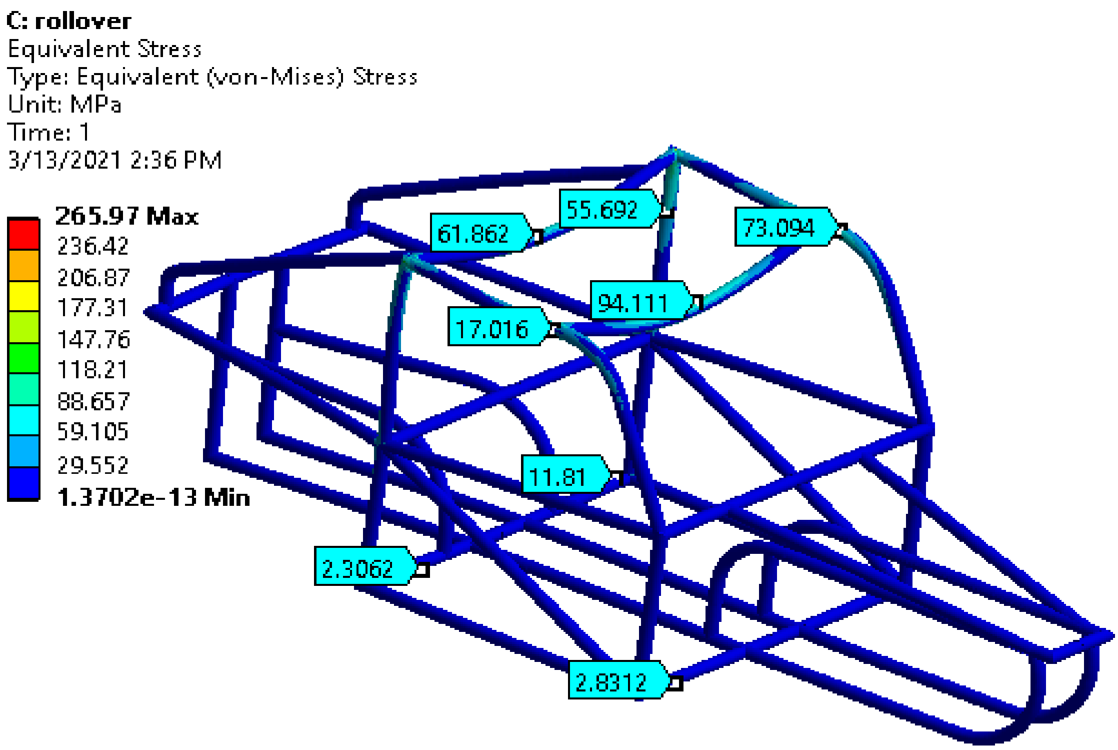


Figure 9-15 Stress distribution in Cabin for roll over

Analysis Results:

Maximum stress in Roll cage = 265.97 MPa

Maximum Deformation in Roll cage = 3.30 mm

Maximimun Cabin stress = 265.97 MPa

9.3.4 Torsional Impact

a. Deformation Analysis

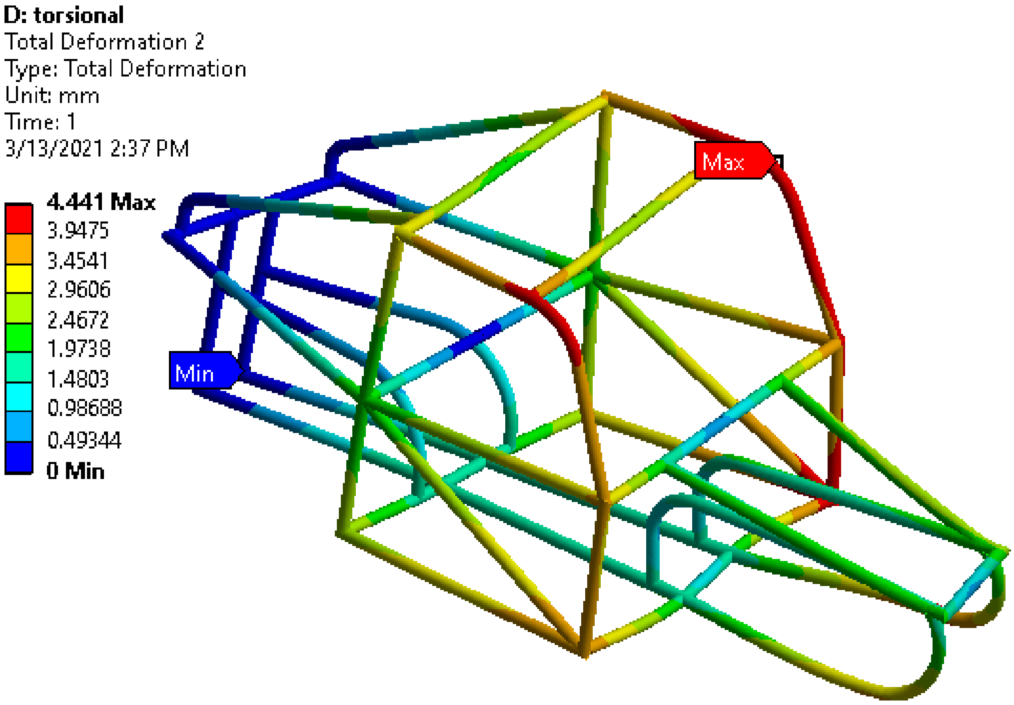


Figure 9-16 Torsional impact deformation test

b. Von-Mises stress distribution

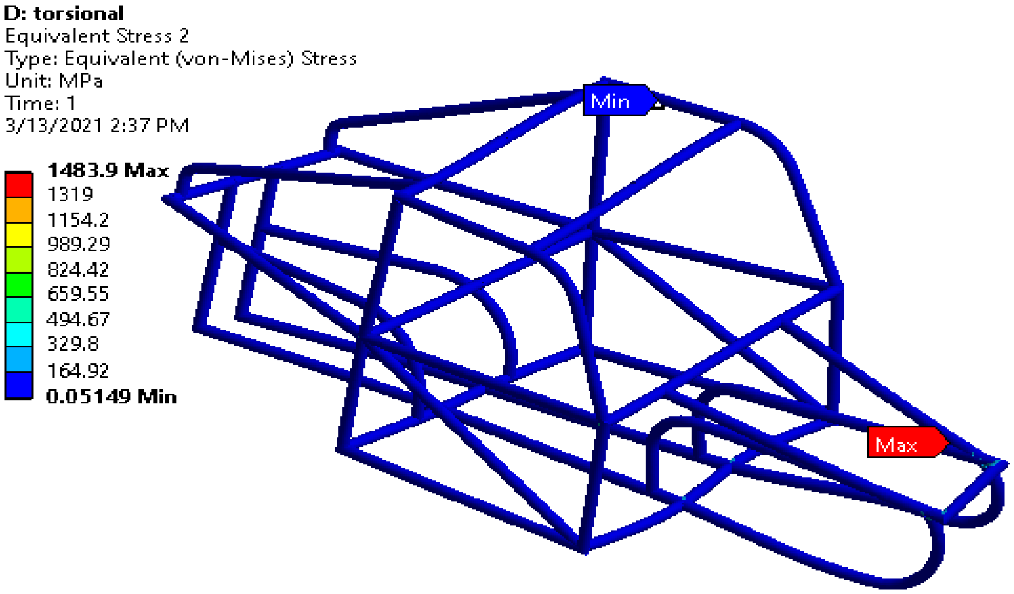


Figure 9-17 Torsional impact Von-Mises stress distribution

c. Stress Distribution in Cabin area

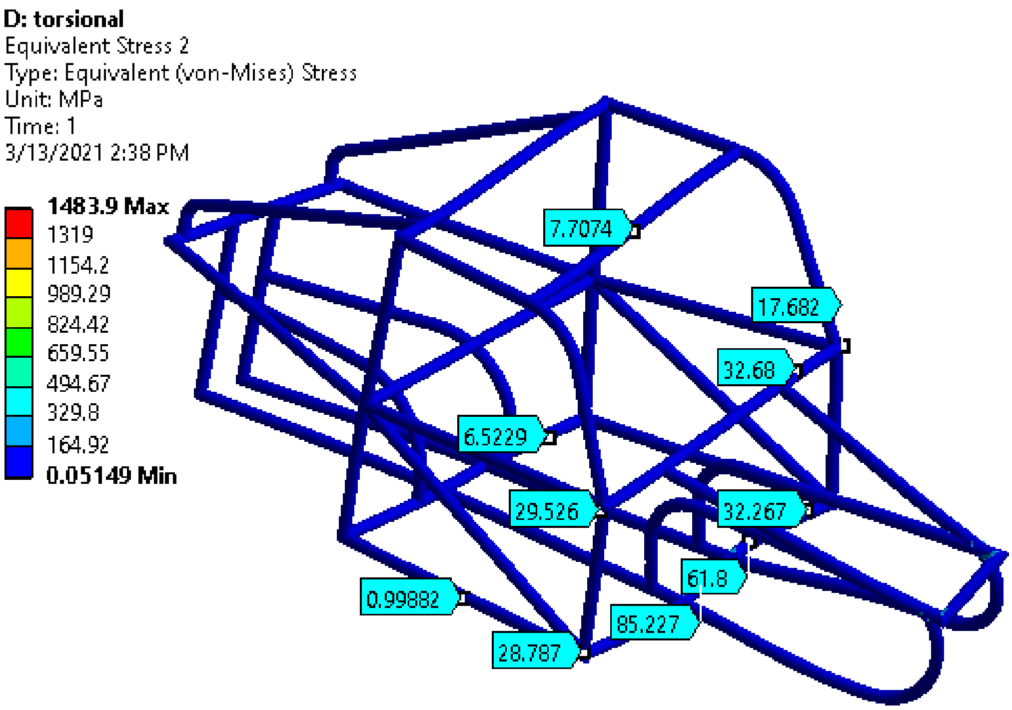


Figure 9-18 Stress distribution in Cabin for torsion test

Analysis Results:

Maximum stress in Roll cage = 1483.9MPa

Maximum Deformation in Roll cage = 4.441 mm

Maximum Cabin stress = 85.227 MPa

9.4 Model-Two Roll Cage

For the analysis the mesh properties is used similar to the static analysis. And the result obtained for the test is tabulated.

9.4.1 Front Impact

a. Deformation Analysis

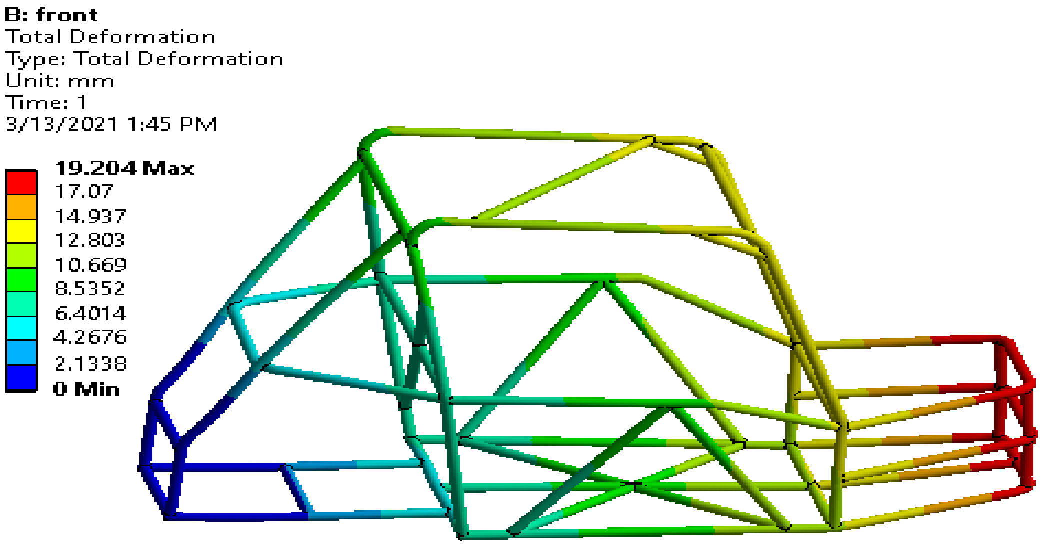


Figure 9-19 Front Impact Deformation Test

b. Von-Mises stress distribution

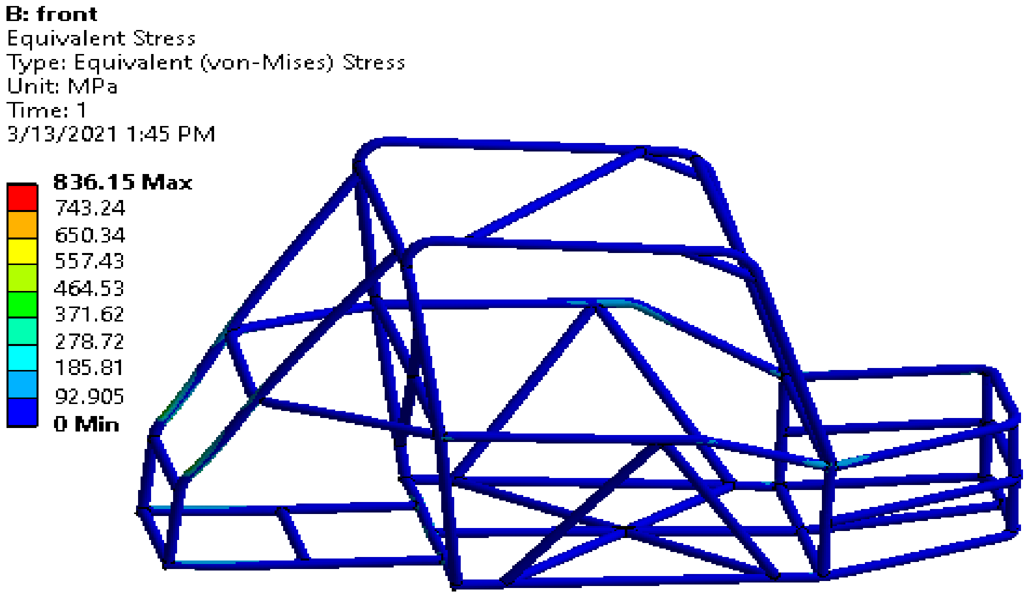


Figure 9-20 Front Impact Von-Mises Stress distribution

c. Stress Distribution in Cabin area

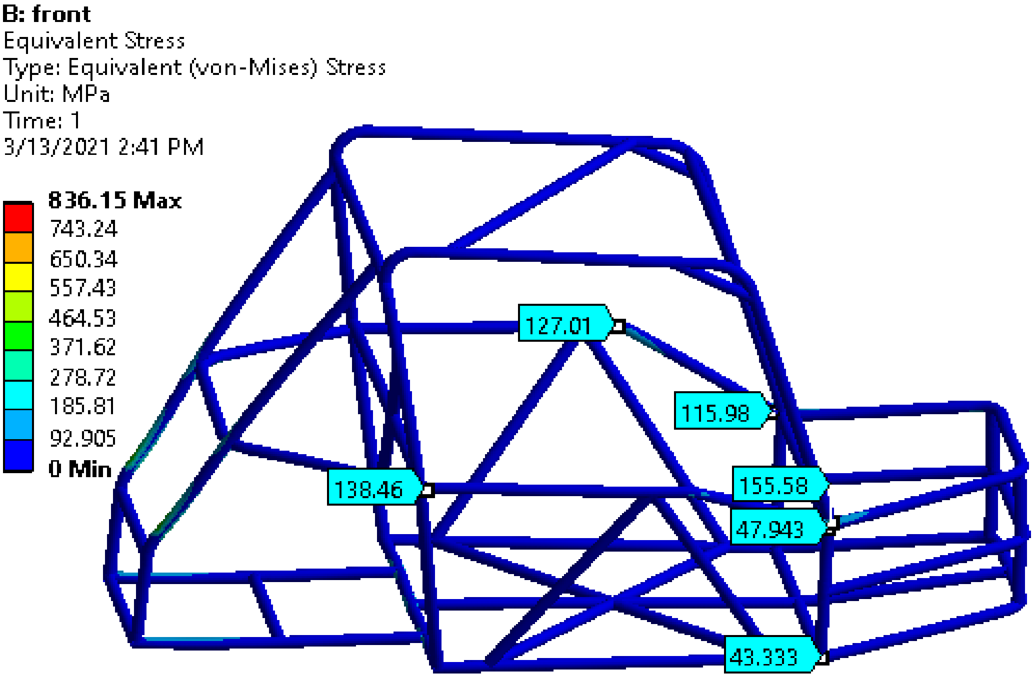


Figure 9-21 Stress distribution in for Cabin for Front impact

Analysis Results:

Maximum stress in Roll cage = 836.15 MPa

Maximum Deformation in Roll cage = 19.204 mm

Maximum Cabin stress = 155.58 MPa

9.4.2 Rear Impact

a. Deformation Analysis

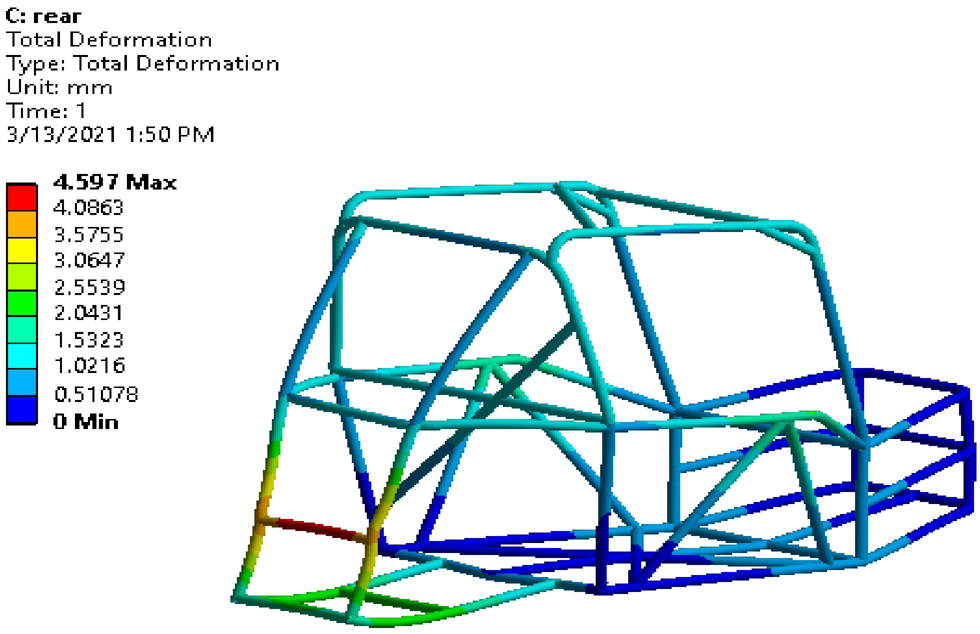


Figure 9-22 Rear Impact deformation Test

b. Von-Mises stress distribution

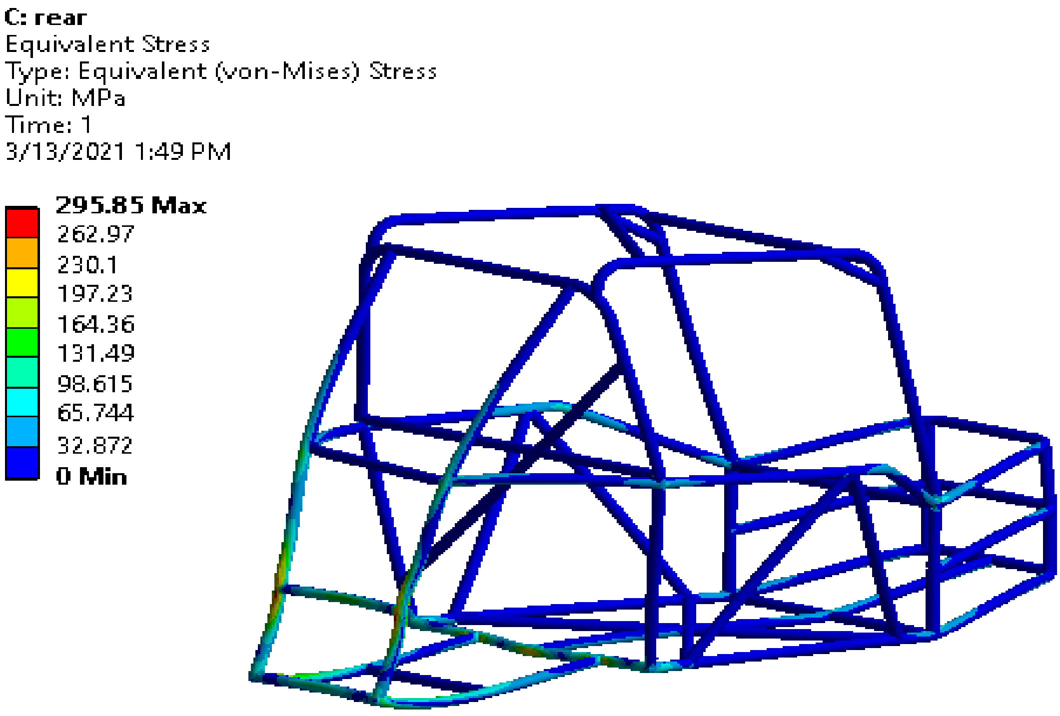


Figure 9-23 Rear impact Von-Mises stress distribution

c. Stress Distribution in Cabin area

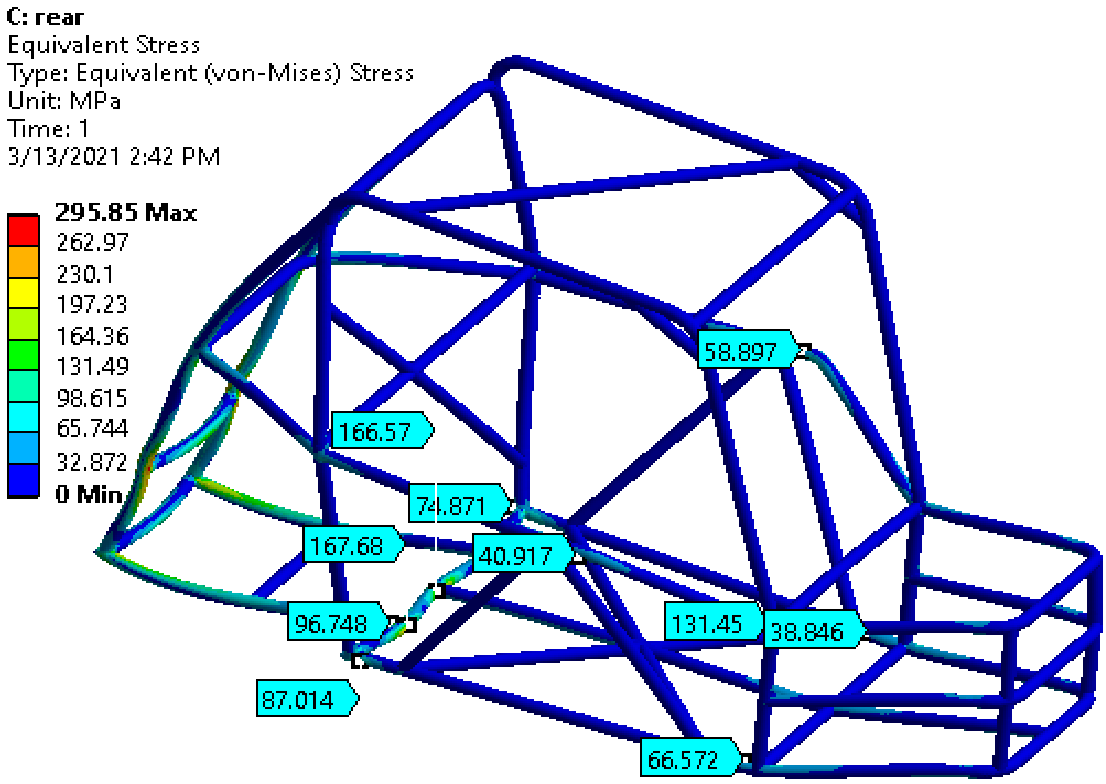


Figure 9-24 stress distribution in Cabin for Rear impact

Analysis Results:

Maximum stress in Roll cage = 295.85 MPa

Maximum Deformation in Roll cage = 4.597 mm

Maximum Cabin stress = 166.57 MPa

9.4.3 Roll Over

a. Deformation Analysis

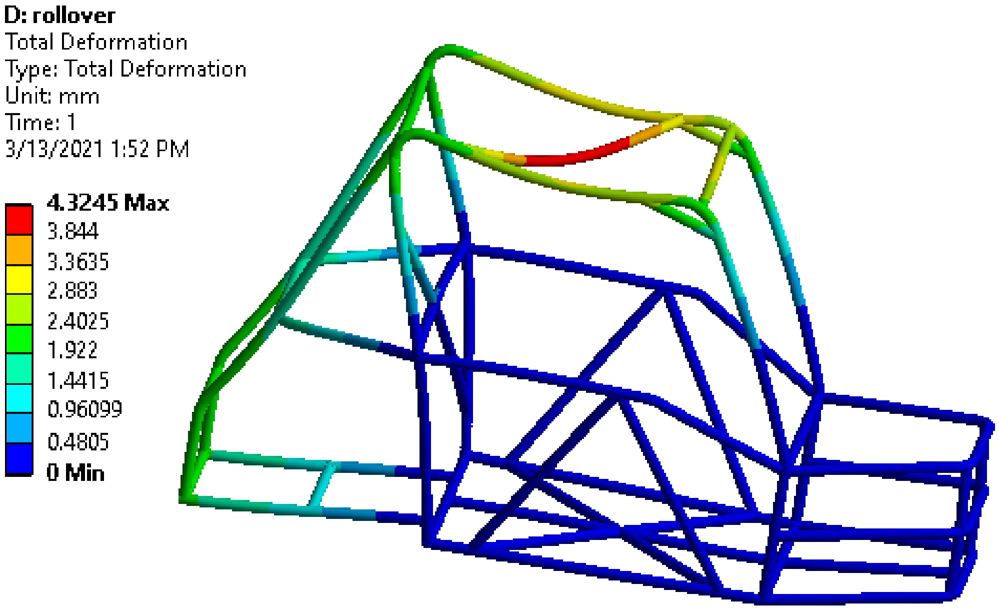


Figure 9-25 Roll over deformation test

b. Von-Mises stress distribution

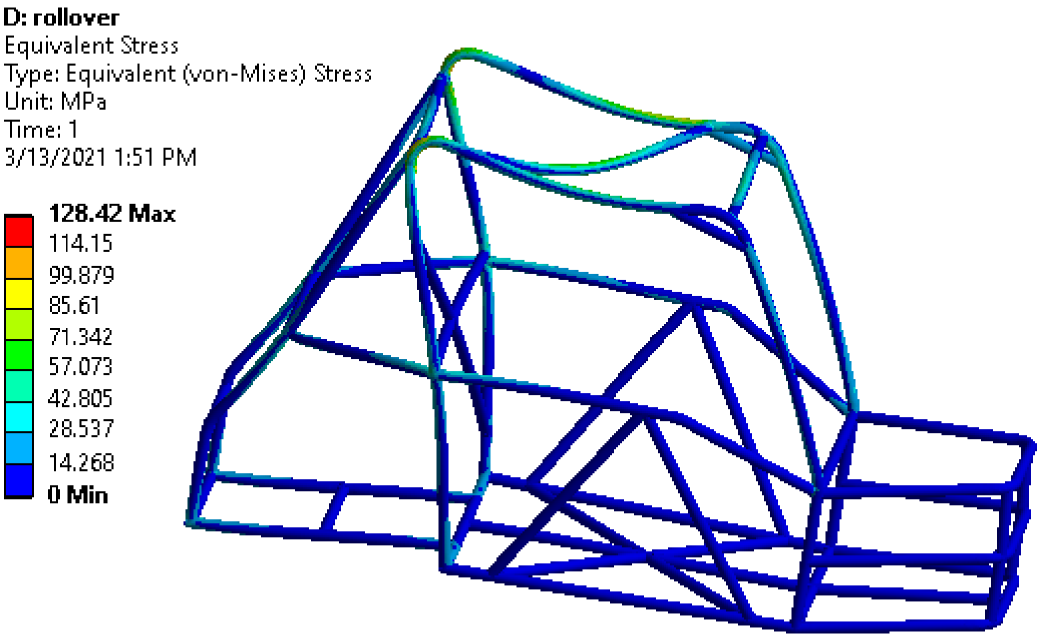


Figure 9-26 Rollover Von-Mises stress distribution

c. Stress Distribution in Cabin area

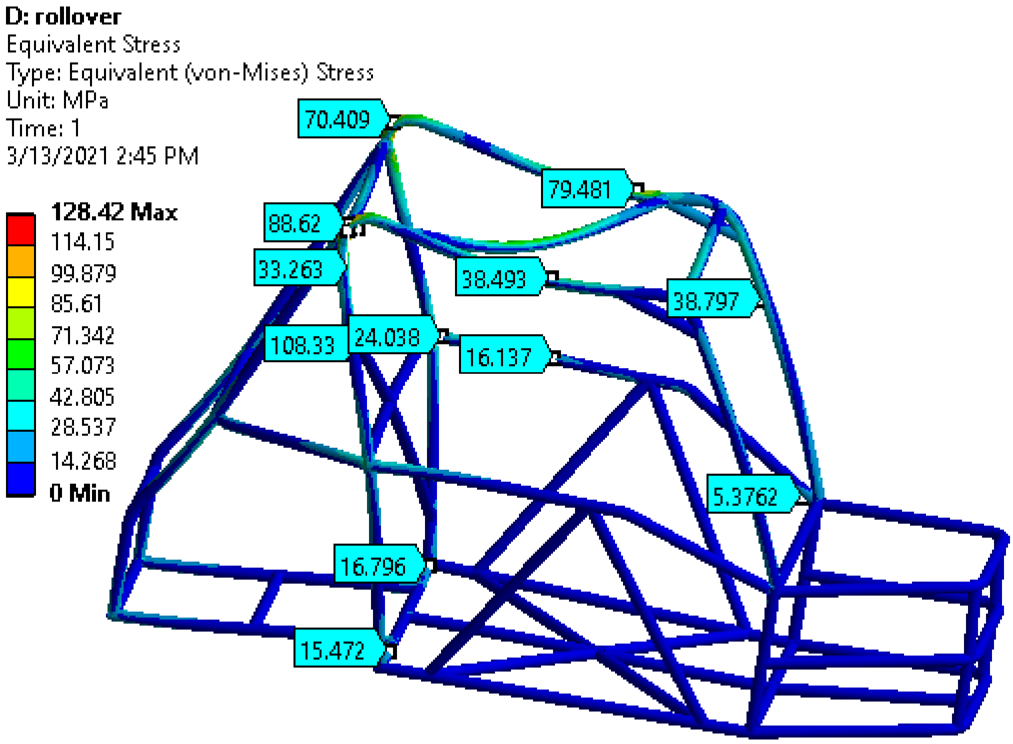


Figure 9-27 Stress distribution in Cabin for Roll over

Analysis Results:

Maximum stress in Roll cage = 128.42 MPa

Maximum Deformation in Roll cage = 4.3245 mm

Maximum Cabin stress =108.33 MPa

9.4.4 Torsional Impact

a. Deformation Analysis

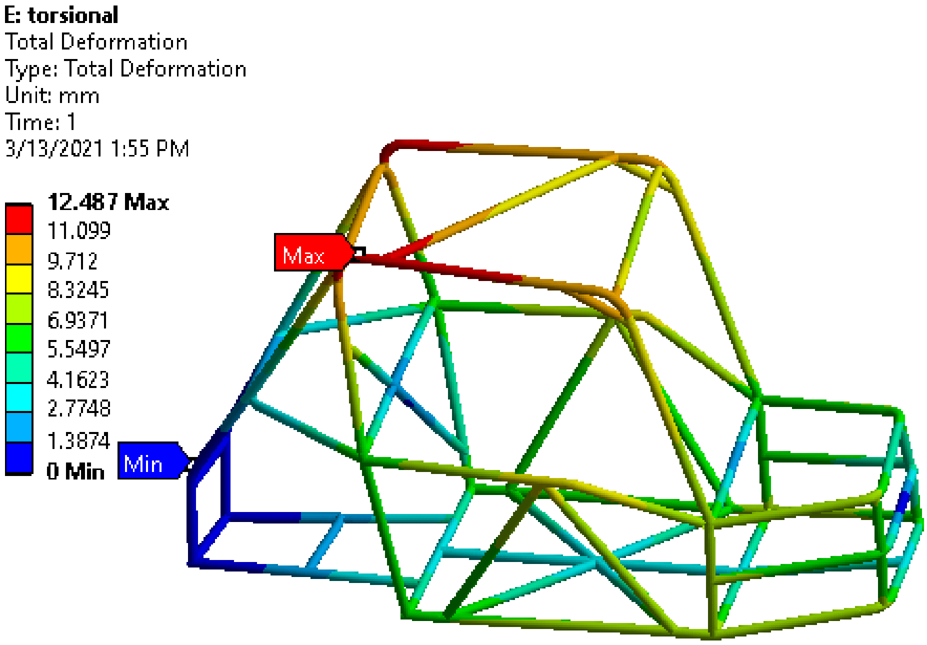


Figure 9-28 Torsional impact deformation test

b. Von-Mises stress distribution

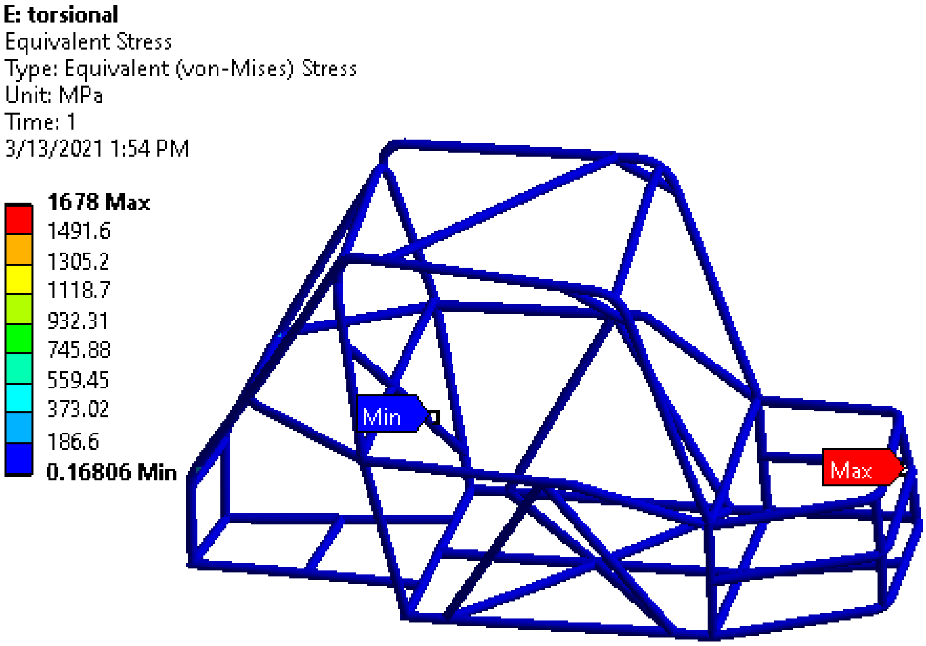


Figure 9-29 Torsional impact Von-Mises stress distribution

c. Stress Distribution in Cabin area

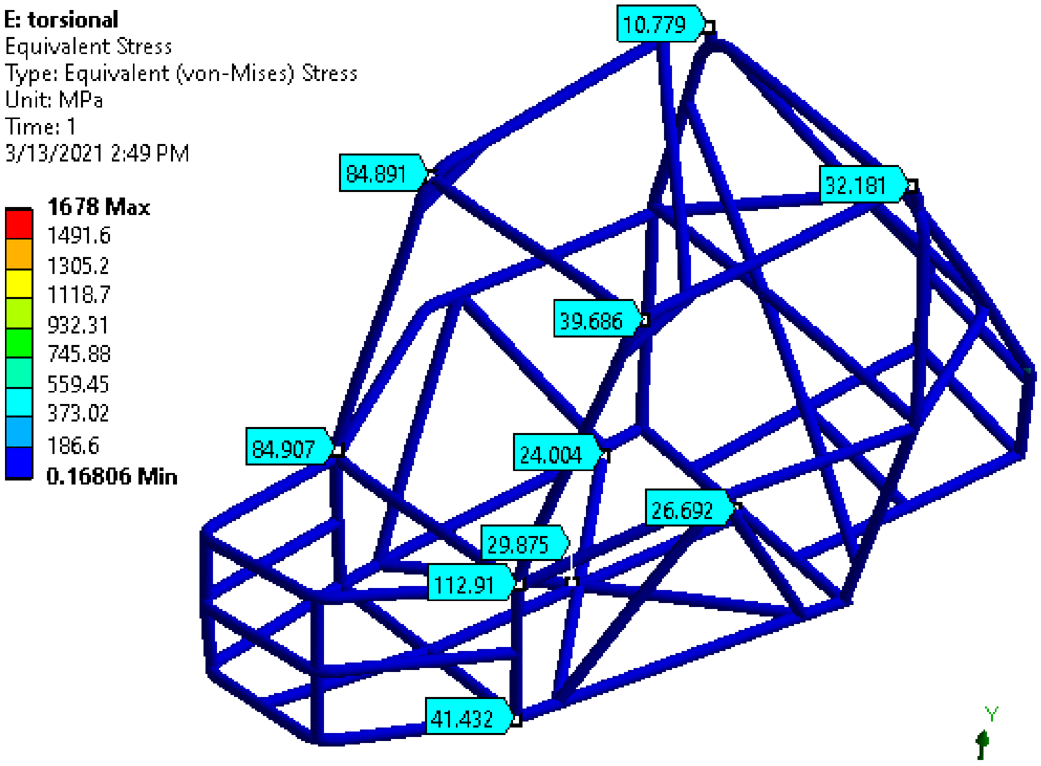


Figure 9-30 Stress distribution in Cabin for torsional impact

Analysis Results:

Maximum stress in Roll cage = 1678 MPa

Maximum Deformation in Roll cage = 12.487 mm

Maximum Cabin stress = 84.891 MPa

Min Factor of safety = 0.0434

Minimum Life Cycle = 45

9.5 Comparison Table

9.5.1 Summary Table

Now, all of the analysis details of the model-one and model-two for stress and deformation test is summarized in the table shown as below. The analysis results of the factor of safety for the whole body are included in annex.

Table 9-3 Comparison of the ANALYSIS testing

Analysis Type	**Roll Cage**	**Deformation (mm)**	**Stress (MPa)**	**FOS**
Front Impact	Model-one	5.65	2822.9	0.124
	Model-two	19.204	836.15	0.419
Rear Impact	Model-one	7.337	1080.6	0.325
	Model-two	4.597	295.85	1.186
Roll Over	Model-one	3.30	265.97	1.319
	Model-two	4.3245	128.42	2.733
Torsion Impact	Model-one	4.441	1483.9	0.236
	Model-two	12.487	1678	0.209

9.5.2 Discussion

The factor of safety is seen below one at some critical point which means there is the high chance of the breaking of the members or the joint when subjected to the impact force or accident. But the maximum areas of the frame has high safety factor, above 2, which is shown in the safety factor determination of the cabin in table below, that indicated those member can withstand the high impact fore than the considered forces or accident force. In some areas the members with low stress developed and high factor of the safety can be replaced with the smaller dimension tubes or removing them so that the weight of the vehicle can be reduced.

From the above table showing the comparison between the two models, the maximum deformation of the model-two is seen to be higher for front impact, roll over and torsion impact than the maximum deformation of the model-one while the maximum deformation of the model-one is higher than the model-two maximum deformation rear impact analysis. The maximum stress developed on the model-two on front impact, rear impact and roll over is minimum while the maximum stress on model-one is minimum than the maximum stress developed on second model in torsion impact. While comparing the minimum life and minimum safety factor, the model two has better result than model one in front impact, rear impact test and roll over test and the result is better for the model-one. With these comparison, the model two is seen better for the most of the test. So we choose the model-two for our final design of the EATV.

9.5.3 Cabin Analysis

The factor of the safety for the cabin of the Analysis software is for different loading impacts are shown in following table.

Table 9-4 FOS of Cabin of EATV

Analysis Type	**Roll Cage**	**Max. Stress (MPa)**	**Factor of Safety**
Front Impact	Model-one	2822.9	0.124
	Model-two	155.58	2.25
Rear Impact	Model-one	311.27	1.127
	Model-two	166.57	2.107
Roll Over	Model-one	265.97	1.319
	Model-two	108.33	3.24
Torsion Impact	Model-one	85.227	4.118
	Model-two	84.891	4.134

This shows that the cabin of model-two is far better to than the cabin of the model-one in the case of the crash testing. So for the safety reason we will proceed with the model two for the fabrication process.

CHAPTER TEN
DESIGN, VALIDATION AND SIMULATIONS

10.1 Modeling and Simulation for Power Train

The Figure below shows the Simulink model for calculation of energy consumption of the Electric All-terrain vehicle. The model shows how to model vehicle dynamics to obtain high level information about tractive force requirements for a given weight and drive cycle. A MATLAB Script is coded to initialize the Simulink model which contains all the main parameter of the vehicle. The Script is placed in the Annex section of the report.

The Simulink block diagram model is run for 2474 s, which is to total duration of the FTP75 drive cycle. The time step of 1 second is set because the input data (FTP 75) is sampled at 1 s. The speed profile is read from FTP75 drive cycle which is reduced by 2.25 to match maximum velocity requirement of our Electric ATV. Before running the simulation we need to initialize the MATLAB script.

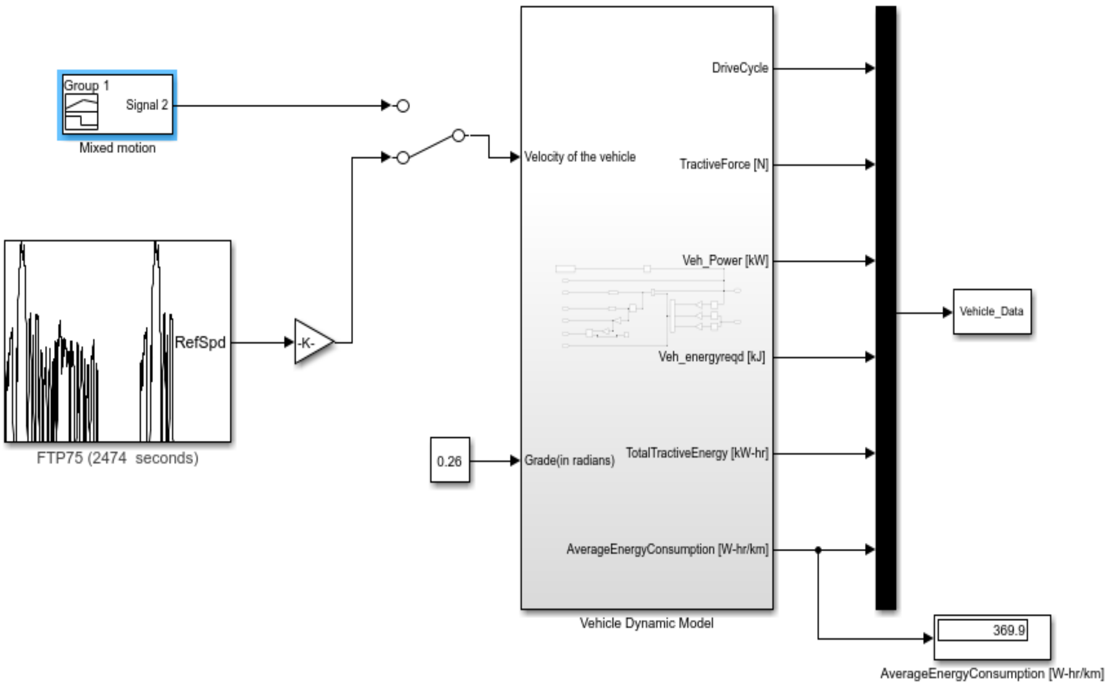


Figure 10-1 MATLAB Simulink Model for Energy Consumption

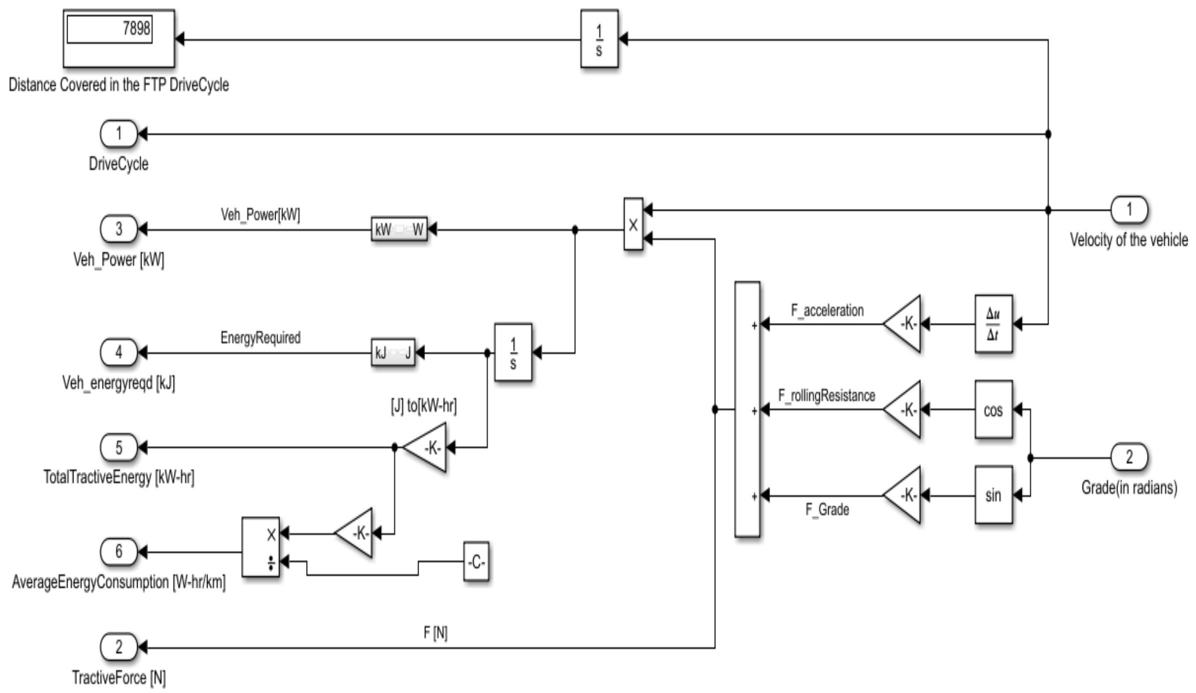


Figure 10-2 Simulink Vehicle Dynamic Model for Energy Consumption

From the above model, we have found the total length of the drive cycle was 7898 m in the time interval of the 2474 seconds of the drive cycle.

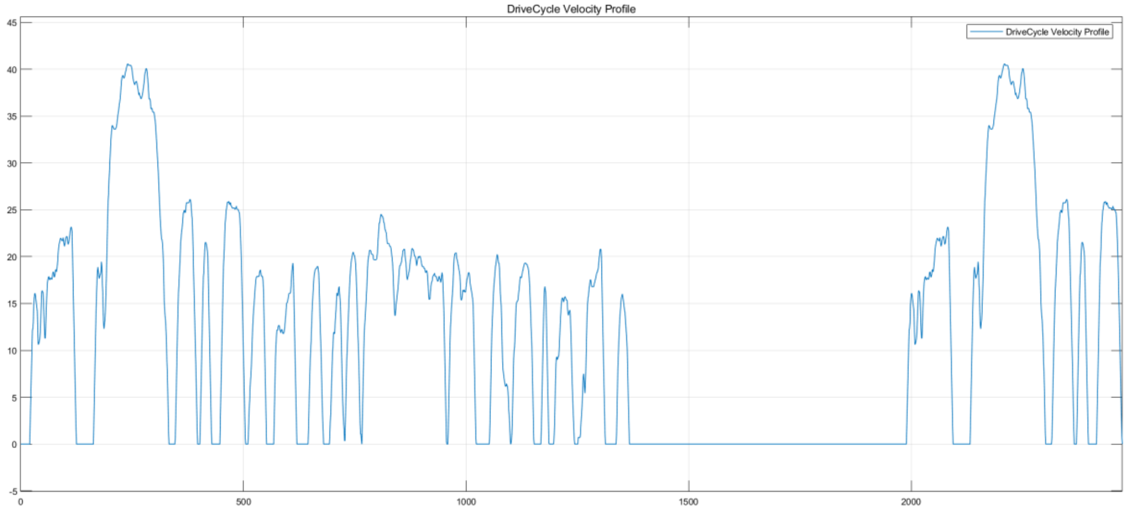


Figure 10-3 Reduced FTP75 Velocity Drive-cycle Profile

The above drive cycle profile shows that the maximum velocity is taken as 40 km/hr.

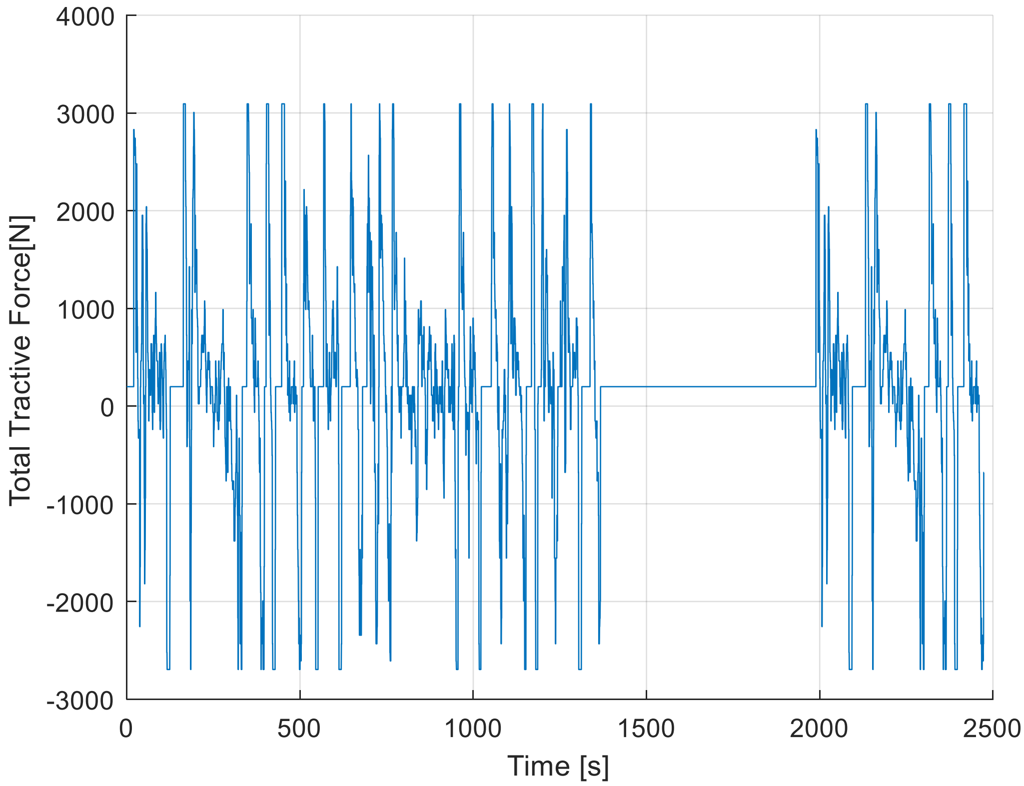


Figure 10-4 Tractive Force versus time at Grade 0%

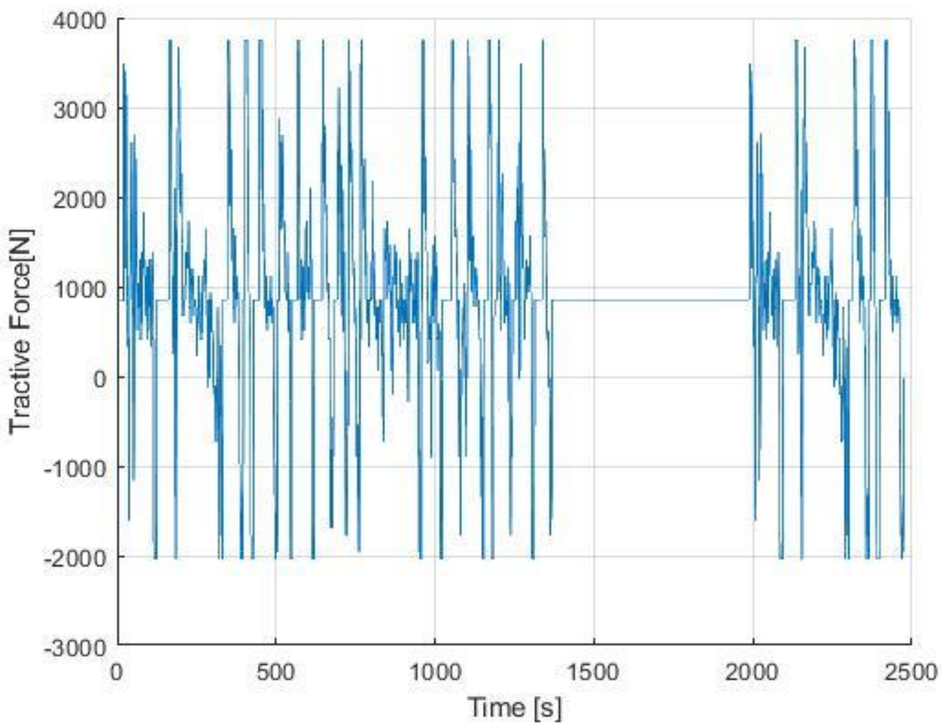


Figure 10-5 Tractive Force versus Time Graph at Grade 25%

The above graph show the tractive forces and time graph. The maximum tractive force is observed to be 4179N at 169 seconds of the drive cycle. The negative tractive force of 1568N at 116 seconds is due to deceleration of the vehicle during the drive cycle.

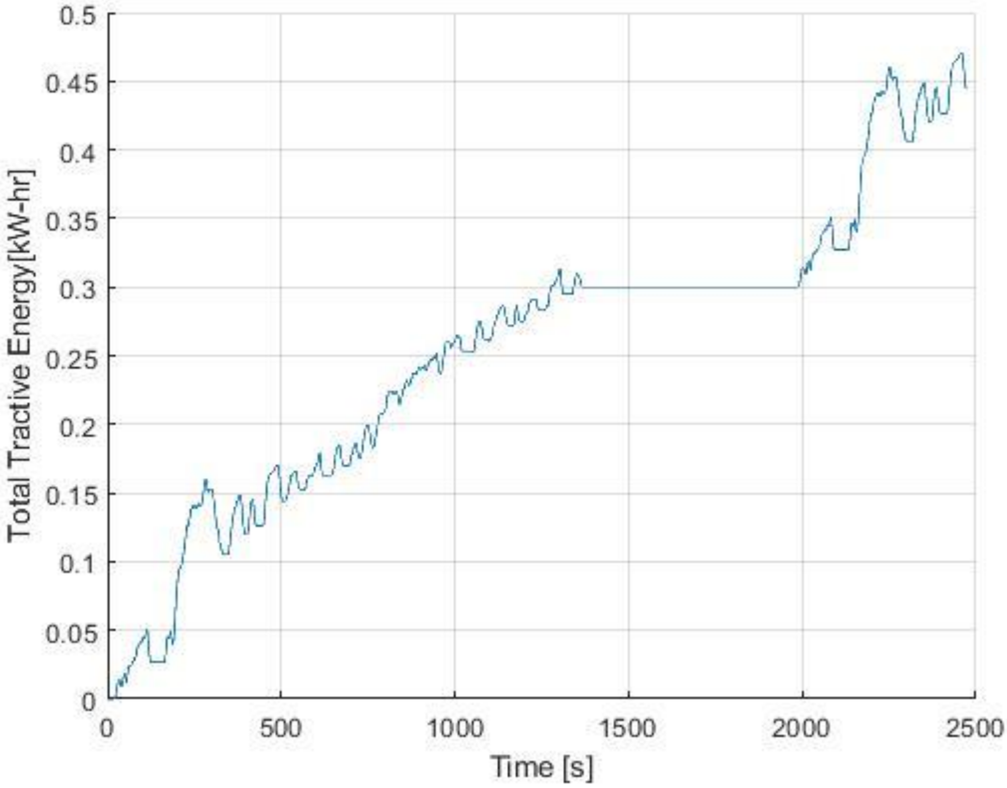


Figure 10-6 Total Tractive Energy at Grade 0%

From the above graph, it is observed that the maximum tractive energy requirement in the given drive cycle is found to be 0.4702 kW-hr. By dividing the last calculated value of the total energy (470.2 W-h) to the total length of the FTP75 drive cycle (7.898km), we get the average energy consumption during the given drive cycle provided the 0 % grade throughout the cycle was found to be 59.53 W-hr/km.

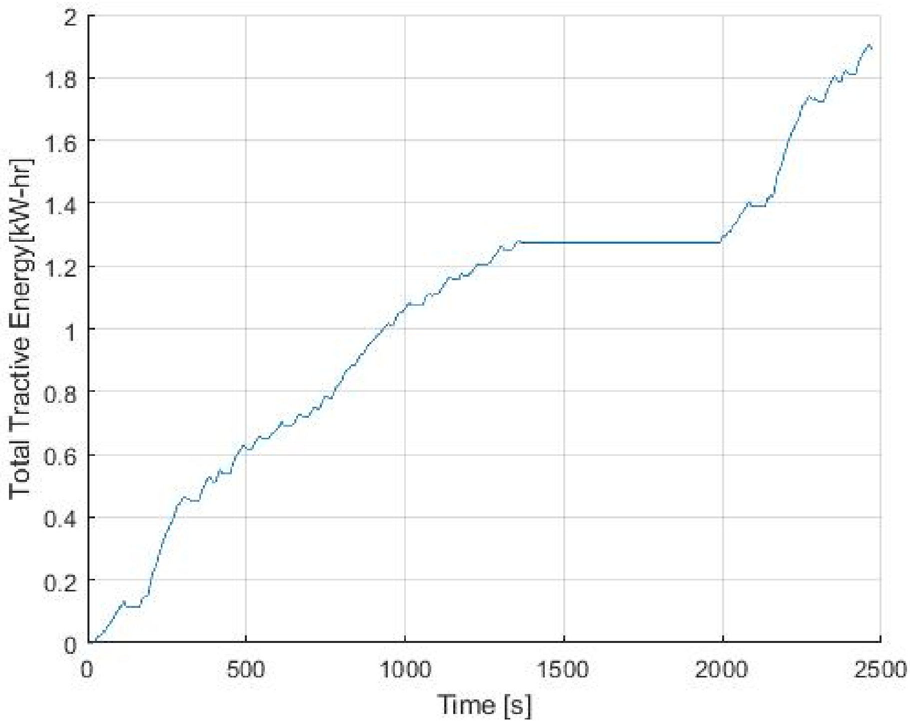


Figure 10-7 Tractive Energy vs. Time Graph at 25% Grade

By integrating the total power over time (for the whole duration of the cycle), we get the total energy consumption (J). From the above graph, it is observed that the maximum tractive energy requirement in the given drive cycle is found to be 2.932 kW-hr.

By dividing the last calculated value of the total energy (2932W-h) to the total length of the FTP75 drive cycle (7.898km), we get the average energy consumption during the given drive cycle provided the 25% grade throughout the cycle was found to be 371.23 W-hr/km. This value will be used to calculate the total energy required for the high voltage battery.

10.2 Result

From the Simulink Simulation, it is found that the average energy consumed by the Electric ATV for the provide drive cycle is 59.35 Wh/km for 0 % grade condition and 371.23 Wh/km for uphill 25% grade condition.

10.3 Design and Analysis of coil spring for Suspension system

The helical coil spring is designed by using Solidworks and analyzed by ANSYS 17.2 software. In this the spring behavior will be observed by applying certain load. Model of the spring will be first created by using Solidworks where its inner and outer diameters were specified. The pitch is calculated by free height of coil spring divided by the number of turns.

We have analyzed the spring under the loading conditions: 4414.5N (for front spring) and 6621.75N (for rear spring). Under the load, the deformation of spring is observed. Along with this, stresses are also developed in the coils. Safety factor of the spring is also obtained in order to ensure the minimization of possible risks or failure. Thus, we have analyzed total deformation, equivalent stresses, equivalent elastic strain, shear stress and safety factor of the coil spring.

10.3.1 For front spring under load 4414.5N

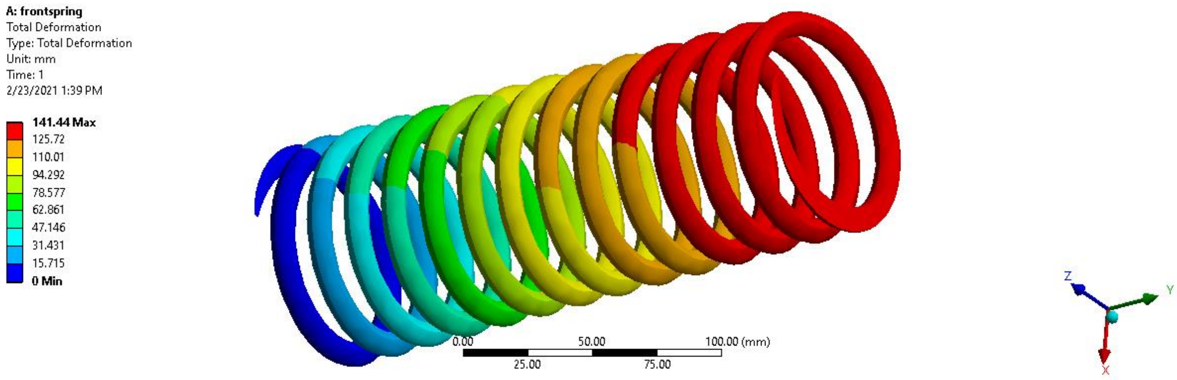


Figure 10-8 Total Deformation for front spring

The load of 4414.5N is applied on the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum deformation obtained. The maximum deformation obtained is 141.44mm and lowest deformation obtained is 0 at the fixed support. The middle portion experiences slight deformation as shown above.

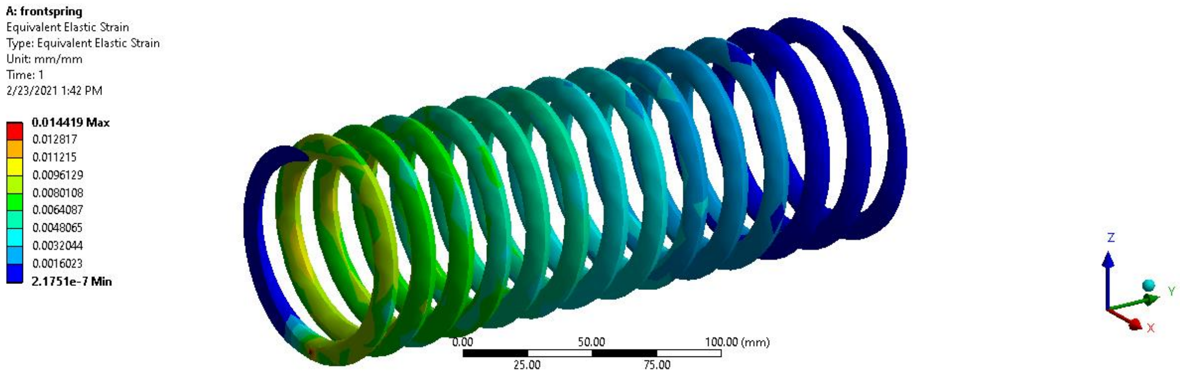


Figure 10-9 Equivalent Elastic Strain for front spring

The load of 4414.5N is applied on the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum strain is 0.014419 and the minimum is 2.1751e-7. The blue part (uppermost part) has maximum deformation, so the ratio of deformation to total length would be minimum and vice versa.

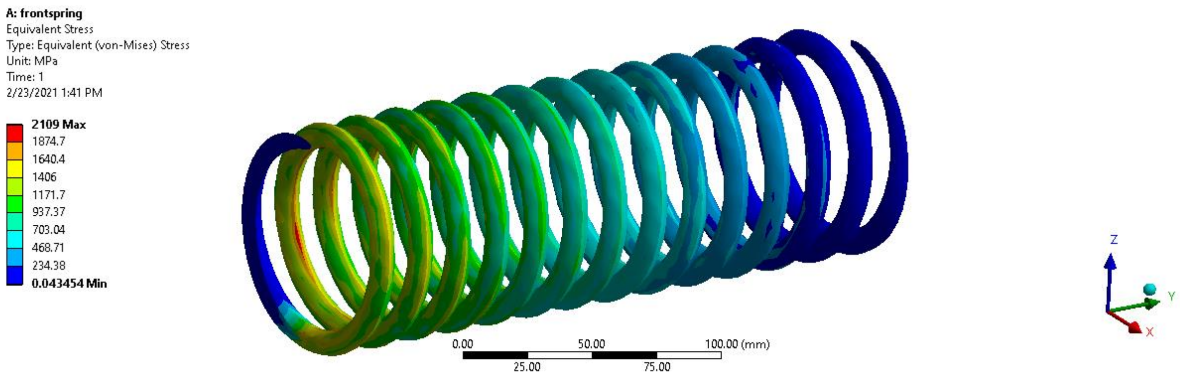


Figure 10-10 Equivalent Stress for front spring

The load of 4414.5N is applied on the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum stress is found to be 2109 MPa and minimum to be 0.043454MPa.

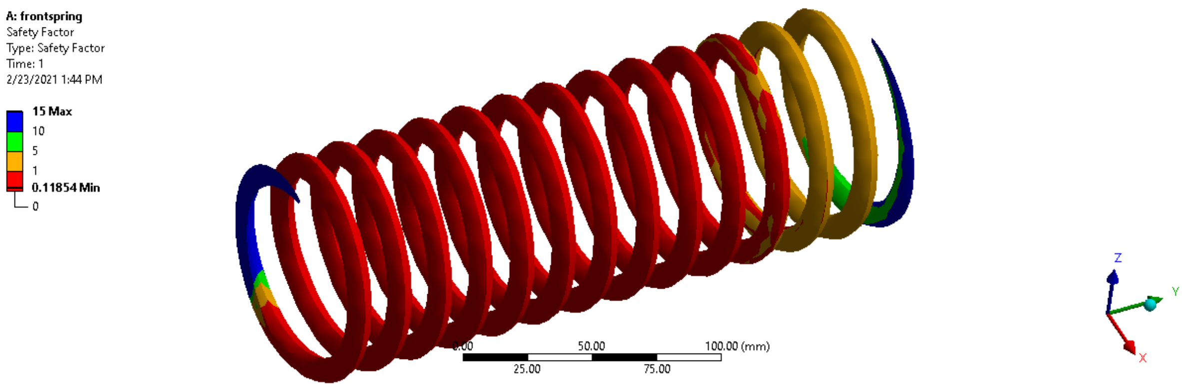


Figure 10-11 Safety Factor for front spring

The load of 4414.5N is applied on the upper part of the spring and lower part is fixed to the wishbone mounted to frame. The maximum safety factor is 15 which is allowed on upper (blue) portion since it gets maximum deformation so may have the chance of failure. Similarly, the minimum safety factor would be zero at the lower region (red) of the spring.

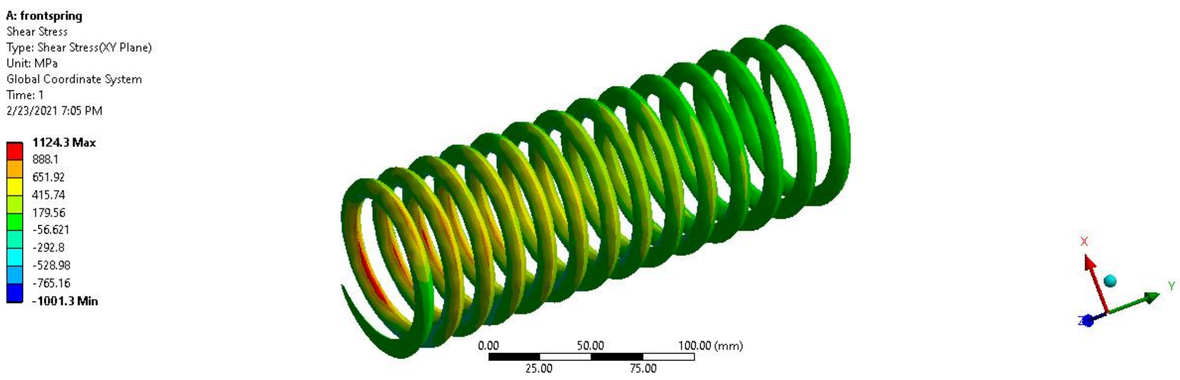


Figure 10-12 Shear Stress for front spring

The load of 4414.5N is applied on the the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum shear stress is 1124.3MPa and the minimum is in negative. The blue part has minimum stress.

10.3.2 For rear spring under load of 6621.75N

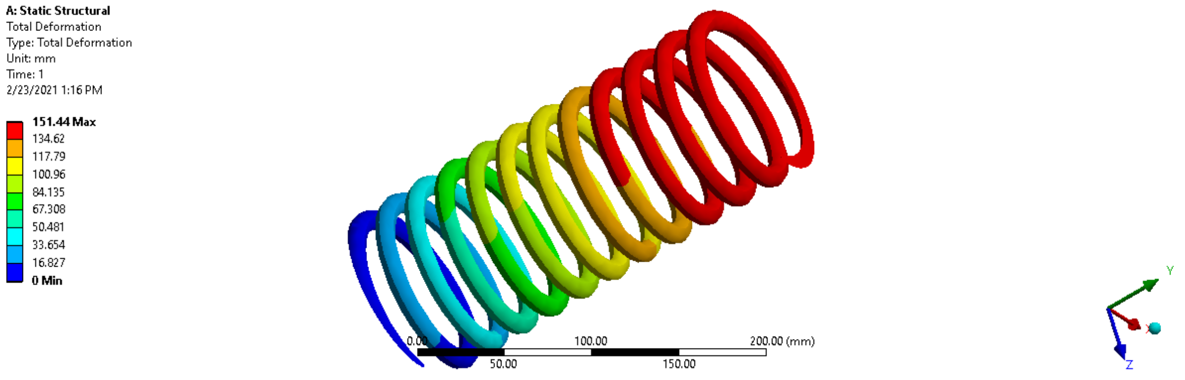


Figure 10-13 Total Deformation for rear spring

The load of 6621.75N is applied on the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum deformation obtained. The maximum deformation obtained is 151.44mm and lowest deformation obtained is 0 at the fixed support. The middle portion experiences slight deformation as shown above.

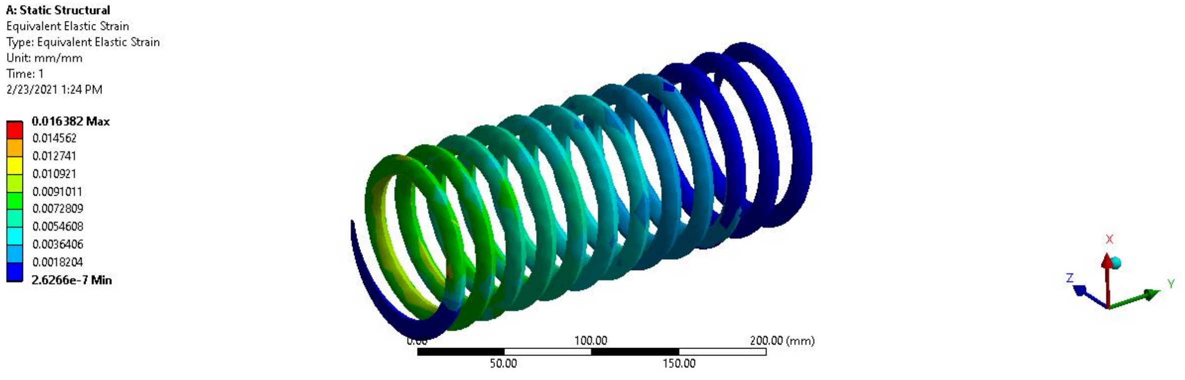


Figure 10-14 Equivakent elastic strain for rear spring

The load of 6621.75N is applied on the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum strain is 0.016382 and the minimum is 2.6266e-7. The blue part has minimum deformation, so the ratio of deformation to total length would be minimum and vice versa.

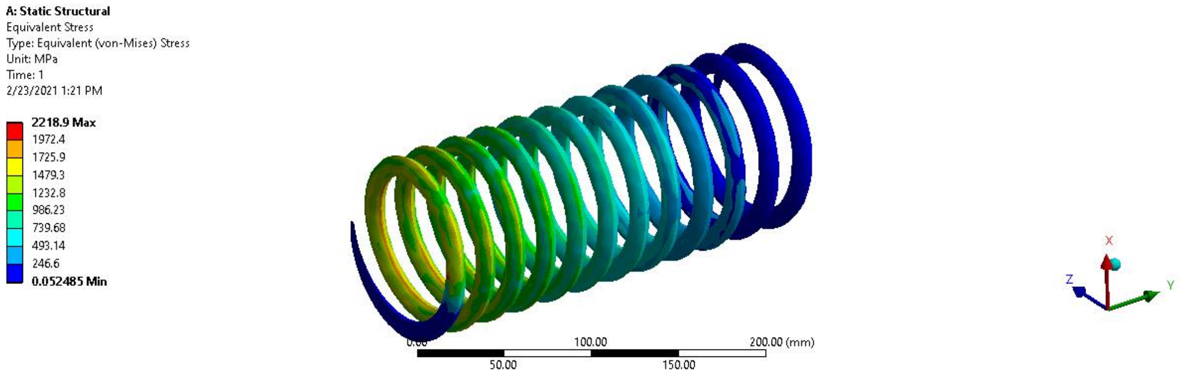


Figure 10-15 Equivalent Stress for rear spring

The load of 6621.75N is applied on the the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum stress is 2218.9MPa and the minimum is 0.052485MPa. The blue part has minimum stress.

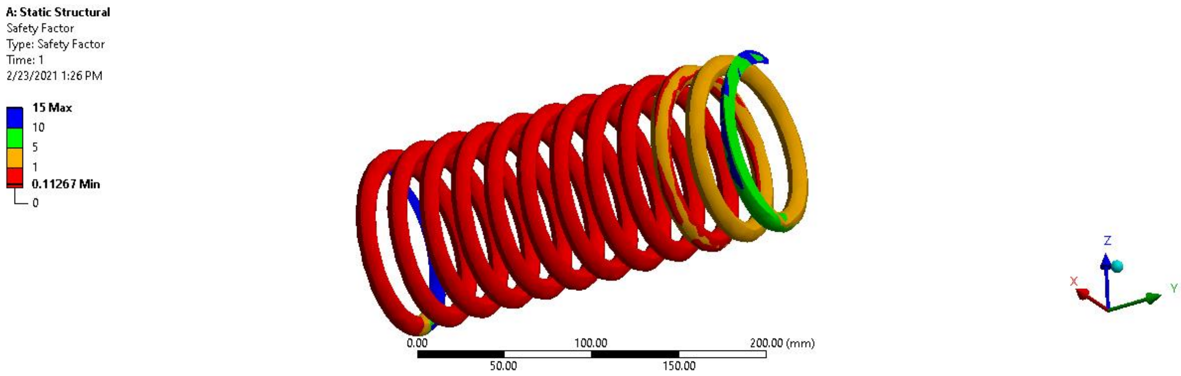


Figure 10-16 Safety factor for rear spring

The load of 6621.75N is applied on the upper part of the spring and lower part is fixed to the wishbone mounted to frame. The maximum safety factor is 15 which is allowed on upper (blue) portion since it gets maximum deformation so may have the chance of failure. Similarly, the minimum safety factor would be zero at the lower region of the spring.

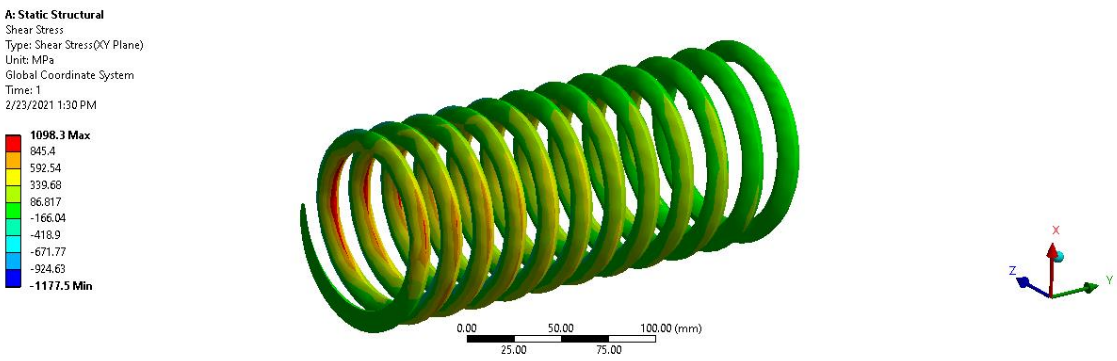


Figure 10-17 Shear Stress for rear spring

The load of 6621.75N is applied on the the upper part of the spring and lower part is fixed to the wishbone mounted to frame. Under analysis, the maximum shear stress is 1098.3MPa and the minimum is in negative. The blue part has minimum stress.

The maximum parameter result from the ANSYS for front and rear spring could be tabulated as below.

Table 10-1 Result of spring analysis from ANSYS

Parameters	Front Spring(maximum)	Rear Spring(maximum)
Total Deformation(mm)	141.44	151.44
Equivalent Stress(MPa)	2109	2218.9
Shear Stress(MPa)	1124.3	1098.3
Safety Factor	15	15
Equivalent elastic strain	0.014419	0.016382

10.3.3 Error Calculation

a. For front Spring

Calculation of shear stress

Dynamic Force (F) = 4414.5N

Mean Spring diameter (D) = 80mm

Diameter of spring (d) = 10mm

Here, C = D/d = 80/10 = 8

$K_S = \frac{2C+1}{2C} = \frac{2\times8+1}{2\times8} = \frac{17}{18}$

Shear stress = $K_S \frac{8FD}{\pi d^3}$

$= \frac{17}{18} \times \frac{8\times4414.4\times0.08}{\pi\times0.01^3}$

= 849.35 MPa

From ANSYS Simulation Shear Stress calculated = 1124.3 MPa

Error can be calculated as:

Error = $\frac{1124.3-849.35}{1124.3} \times 100\%$

= 24.45 %

b. For rear spring

<u>Calculation of shear stress</u>

Dynamic Force (F) = 6621.75N

Mean Spring diameter (D) = 96mm

Diameter of spring (d) = 12mm

Here, C = D/d = 96/12 = 8

$K_S = \frac{2C+1}{2C} = \frac{2\times8+1}{2\times8} = \frac{17}{18}$

Shear stress = $K_S \frac{8FD}{\pi d^3}$

$= \frac{17}{18} \times \frac{8\times6621.75\times0.096}{\pi\times0.12^3}$

= 884.74 MPa

From ANSYS Simulation Shear Stress calculated = 1098.3 MPa

Error can be calculated as:

Error = $\frac{1098.3-884.74}{1098.3} \times 100\%$

= 19.44 %

10.4 Suspension Quarter car model representation in Simulink

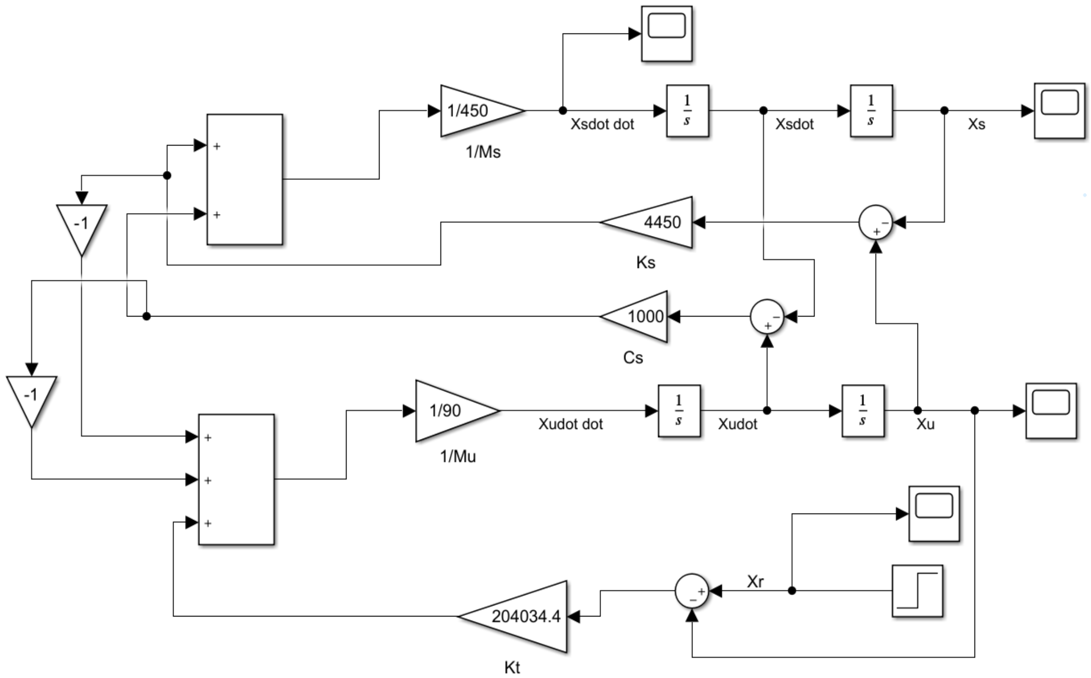


Figure 10-18 Simulink Model of Quarter Car Model

The quarter car model is a mass spring damper system having two masses, un-sprung mass and sprung mass interconnected by spring and damper.

Each sprung mass and unsprung mass have their own independent displacement. So, quarter car will have two independent displacement making the degree of freedom of the quarter car model as two.

Here, for the simulation of the model in Simulink, we use the following parameters:

Mass of sprung mass (Ms) = 450kg
Mass of unsprung mass (Mu) = 90kg
Tire stiffness (Kt) = 204034.4 N/m
Damping coefficient (Cs) = 1000Ns/m
Spring stiffness (Ks) = 4450 N/m

When we run the model in Simulink, the results will be obtained as below.

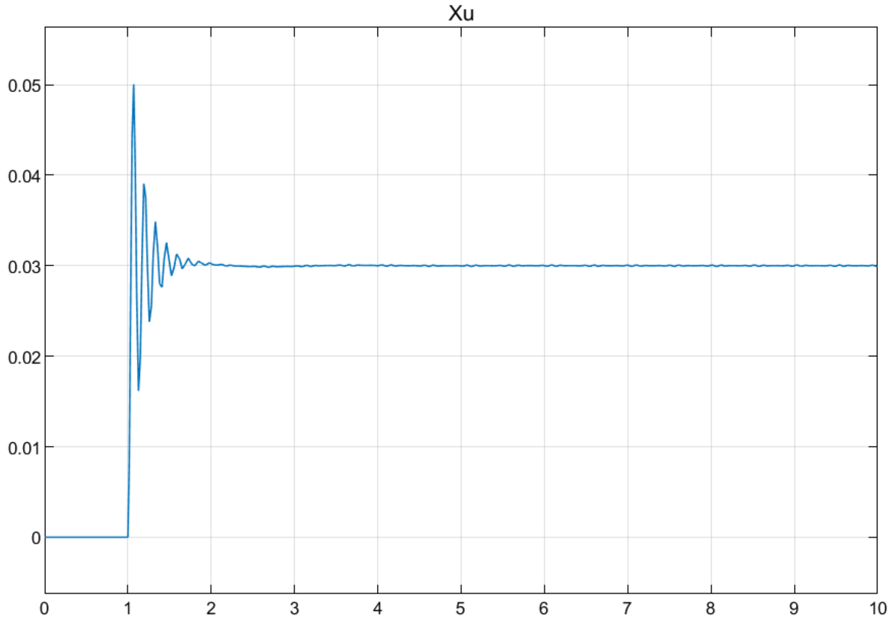


Figure 10-19 Plot of unsprung mass displacement vs time

The above graph shows the unsprung mass displacement with time. The displacement of unsprung mass (m) is plotted along Y-axis and the time(in sec) is plotted along X-axis. The sudden impact on road is felt at 1 second (step input), which displaces the unsprung mass upto 0.05m. Suddenly, the motion dampens and displaces down to 0.015m. Slowly, the damping of vibration starts and the unsprung mass displacement ends within a period of 1 second. (i.e. upto 2^{nd} second).

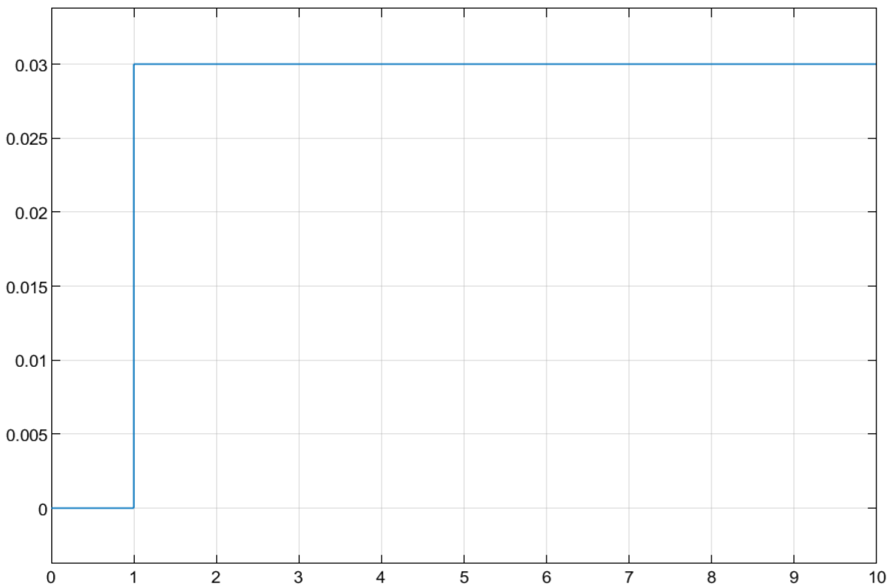


Figure 10-20 Step input signal

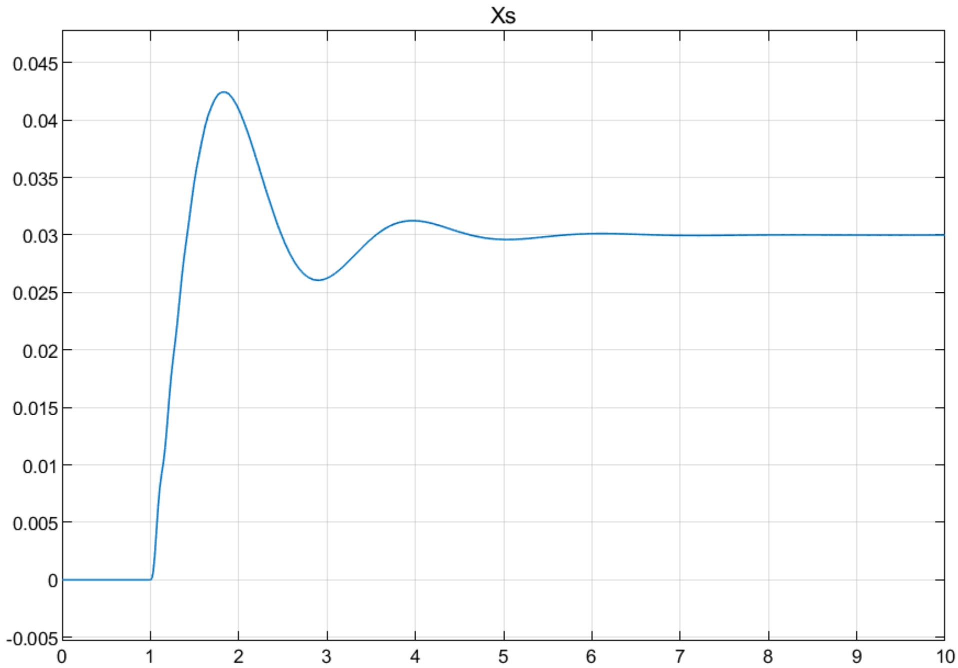


Figure 10-21 Plot of sprung mass displacement vs time

The above graph shows the sprung mass displacement with time. The displacement of sprung mass (m) is plotted along Y-axis and the time(in sec) is plotted along X-axis. The sudden impact on road is felt at 1 second (step input), which displaces the sprung mass upto 0.0425m. Suddenly, the motion dampens and displaces down to 0.027m. Slowly, the damping of vibration starts and the sprung mass displacement ends within a period of 5 seconds. (i.e. upto 6th second).

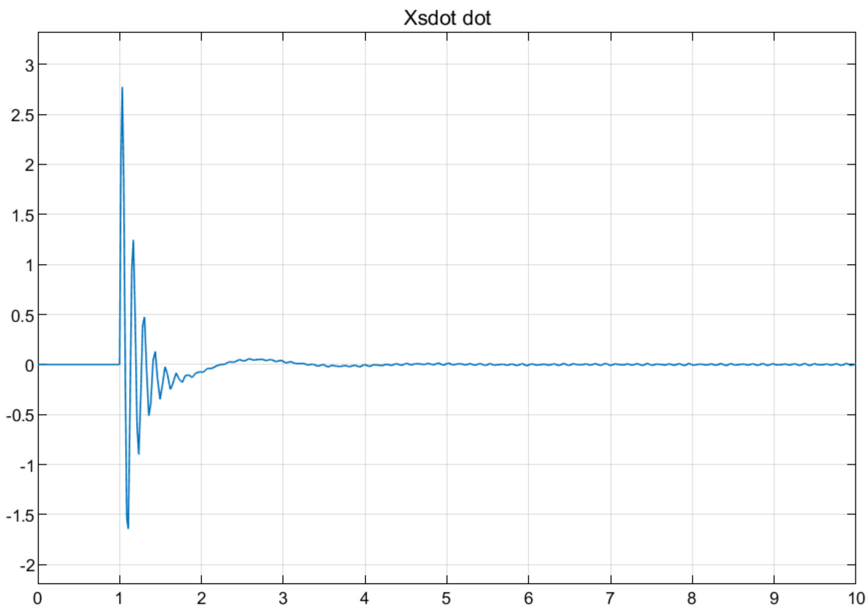


Figure 10-22 Plot of sprung mass acceleration with time

The above graph shows the sprung mass acceleration with time. The acceleration of sprung mass (m/s^2) is plotted along Y-axis and the time (in sec) is plotted along X-axis. The sudden impact on road is felt at 1 second (Step input), which displaces the sprung mass upto $2.7 m/s^2$. Suddenly, the motion dampens and decelerates down to $1.4 m/s^2$. Slowly, the damping of vibration starts and the sprung mass acceleration ends within a period of 3 seconds. (i.e. upto 4^{th} second).

10.5 Design of steering system

We have done the design of steering system in Solidworks. The design is below. All the dimensions are used as calculated.

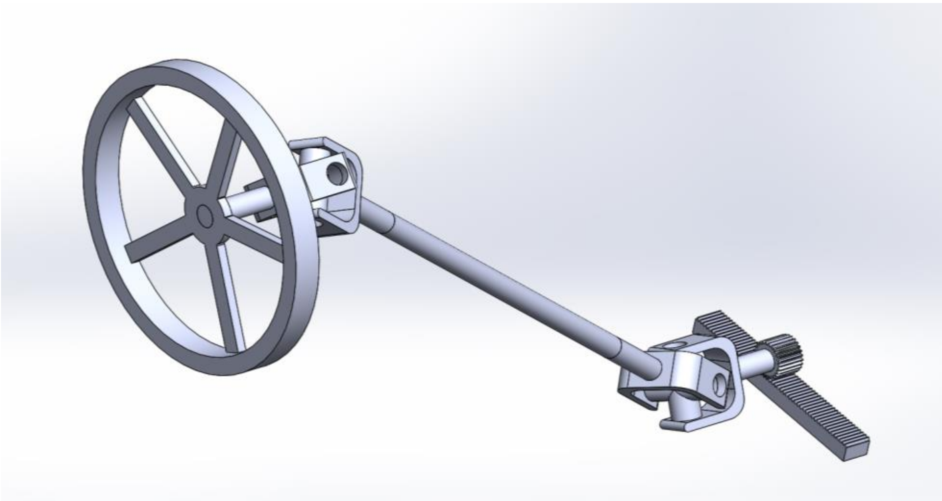


Figure 10-23 Isometric View of Steering System

The different components used to assemble the steering system were designed as below.

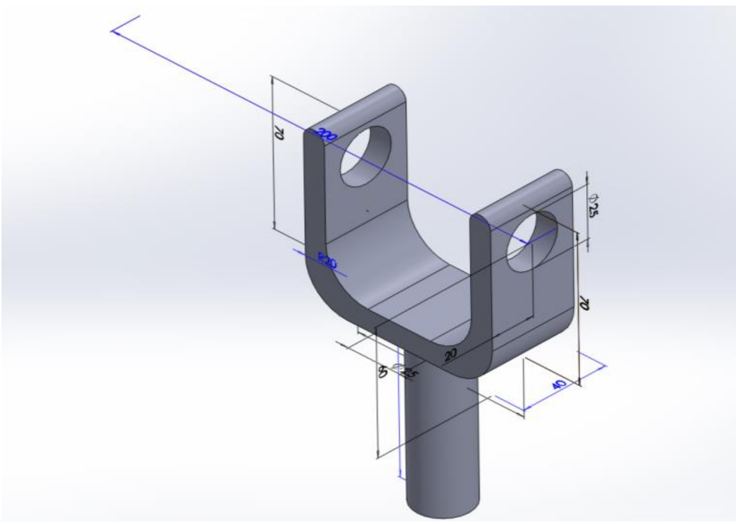


Figure 10-24 Universal Joint

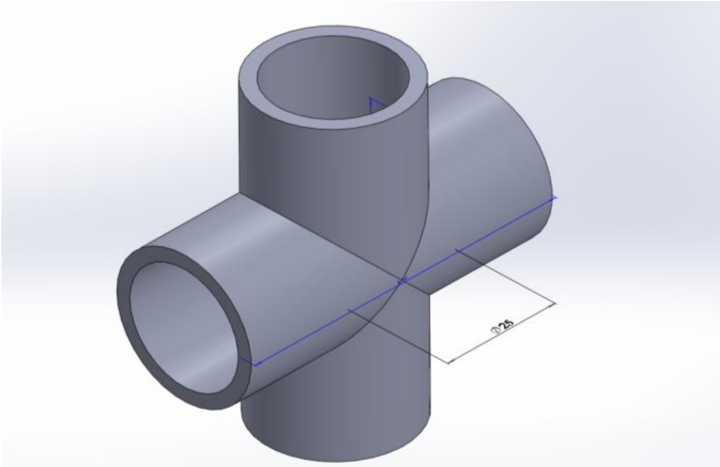


Figure 10-25 Cross Tube

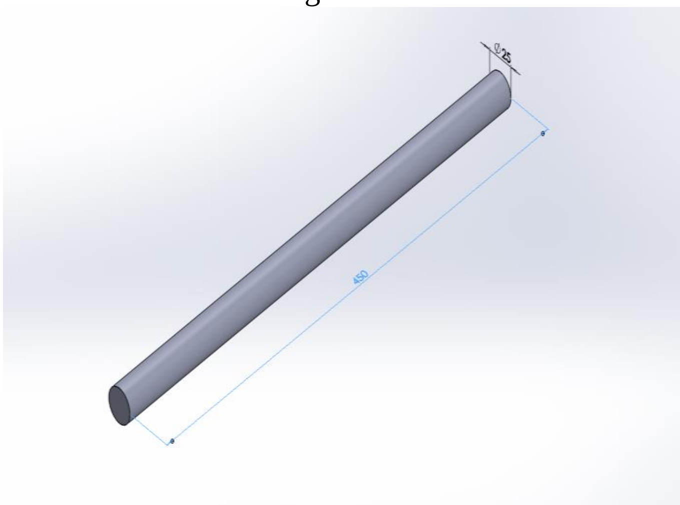


Figure 10-26 Steering Shaft

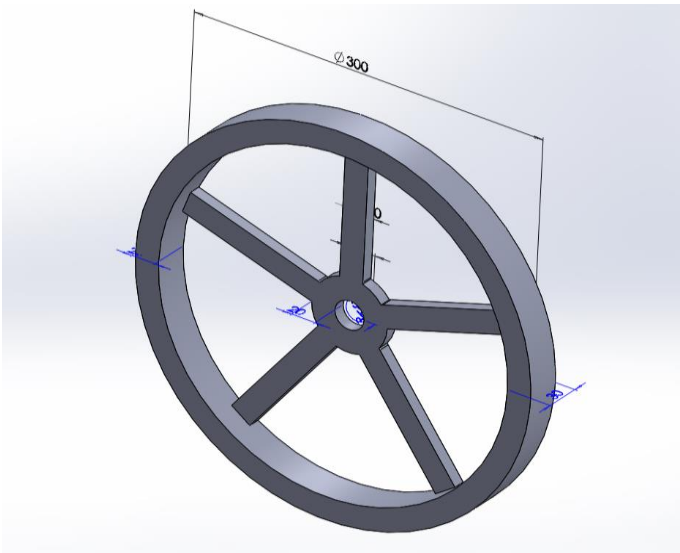


Figure 10-27 Steering Wheel

CHAPTER ELEVEN
CRITERIA OF TIRE SELECTION

For the use of All-terrain vehicle the tire must be chosen very carefully as the tire also determine the performance of the vehicle. For the purpose of the ATV, the tire must be suitable to travel on all season, all road condition like paved road or on-road, off-road, steep roads, sand dunes, muddy roads, wet roads etc. so the properties of the ATT should be as below:

a) Tread pattern

For the purpose of the ATV, there are number of tread pattern like square, rectangular or lugs. But the common thing that are present in any sort of the tread pattern is the pattern are close to each other to give higher contact patch. Although choosing only a single tire for all year and for all terrain will only give average performance in every terrain, with a tread depth of under an inch, and we could get good traction with a smooth ride. The primarily suggested tread pattern for the tire is shown below.

Figure 11-1 Recommended tire tread pattern of tire

b) Suitable for all season

As discussed earlier the ATT should be used for all the year and all terrain like snow, mud, off-road, wet road etc. It is quite expensive to keep separate tire for every type of terrain and change them according to the situation. So the selection of the tire must be done keeping all possible situations on consideration. This may not give perfect result for the use of the tire but will be usable enough.

c) Rating

Rating is important for selecting the tires. It shows the characteristics of the tire. Since we are planning to operate the ATV in off-road conditions more often then we must choose the tire likewise. For such condition soft tires are more suggested by the expert along with the radial plies more than 4 plies. Other rating parameter like speed, load rating etc. must be chosen as per our vehicle specification of what speed and load will it take.

d) Size

The size selection of the tire is also most important. For the suitable size of the ATT, Hilltop tire services suggested following sizes:

Tire Size 25x8-12* – Height x Width - Wheel Diameter

The (25) is the overall height of the tire when it is mounted and inflated to factory recommended air pressure.

The (8) is the overall width of the tire when mounted and inflated to factory recommended air pressure.

The (12) is the diameter of the wheel that the tire will mount to.

*All measurements are in inches. (Hilltop tire services, 2019)

e) Brand and Pricing

We must go for known tire manufacturer but the pricing of the tire must also be compared to it before buying. Cheap tires will run you less money, but paying a price that’s way above the rest of the pack likely isn’t worth it. Instead, look for tire prices that are on par with all the popular brands and models.

We must check either warranty is available or not. If we can get similar tire at other prices like some exhibitions, discounted price sales etc. we should choose them.

CHAPTER TWELVE
DRIVER ERGONOMICS

12.1 Introduction

Ergonomics is the science of equipment design intended to maximize productivity by reducing driver fatigue and discomfort. Ergonomic consideration should not compromise the design of our ATV. People vary in their heights and weights widely. But our vehicle should be able to keep any person safe and provide good comfort for all.

12.2 Factors Affecting Driver Ergonomics

The essential factors to be considered for driver ergonomics are elaborated below:-

a. Seat flexibility

A lot of drivers suffer from back pain, spinal cord injuries due to inflexible and uncomfortable seat. So, in our project, we are considering this factor as an essential parameter for designing. We use seat which is more flexible and comfortable for driver. The shape and size of seat is chosen according to our design and driver's easiness.

b. Position of steering wheel

Another factor to be considered for proper ergonomics is the position of steering wheel in relation to various drivers. The optimal placement with respect to different person should be found before placing it. The position of steering wheel is selected in such a way that it is suitable for each person. For the confortable grip and operation we will be using the steering wheel of 300mm diameter.

c. Position of accelerator pedal and brake pedal

Based on the design, the position of pedal which suits the design is decided. Then the reach of different person to the pedal shall be ensured. The pedal shall also be provided to indicate the driver the appropriate time of shift.

d. Height and width of chassis frame

The height and width of the chassis frame should be designed in such way that the driver as well as passenger should feel comfortable while boarding and riding on a vehicle.

The table shown in the below shows the various standard values for the different parameter of the driver's ergonomics and the desired or designed valued for the parameter that is used for the frame design and manufacture.

Table 12-1 Drivers ergonomics parameter

Parameter	Standard Range	Designed values
Angle of elbows	125°- 140°	128°
Angle of knee	120°- 150°	125°
Angle of back	8°- 15°	12°
Steering wheel diameter (mm)	-	300
Angle of steering wheel	40°- 50°	40

CHAPTER THIRTEEN
BUDGET ESTIMATION

The estimated budget for the making of the EATV roll-cage is highly dependent or affected by the material to be used and the manpower cost. The cost of the accessories to be installed in the frame will be dependent on the brand of the accessories chosen and its quality. The budget shown in above table is an estimates made with discussion with the technical person working in the field of mechanical workshops. The tentative estimation of the budget needed to make the roll cage with above stated accessories is shown in table below:

Table 13-1 Estimated budget for EATV Chassis fabrication

S. No.	Item	Estimated cost (NRs)	Remarks
1	Design of the model	15000	To be done by ourselves
2	Human Resource	25000	Cost of manpower involved (4 person * 50hr * 125 per hour)
3	Rent of the workshop	10000	For private workshop
4	Fabrication material for frame	25000	For the shafts that will be used for the fabrication.
5	Equipment used for fabrication	10000	Cost of Equipment for cutting , welding, bending machine etc.
6	Tires	10000	Cost for the purchase of tires
7	Seat	16000	2 seat of Rs.8000 each
8	Accessories	15000	Headlight, Turn indicators, wires, fuses, relays etc.
9	Other expenses	10000	Cost of transportation, clips, nuts and bolts etc.
	Total cost	1,36,000/-	Cost for making roll cage

Also the budget estimation for the cost of components and cost of resources that are required are tabulated as below.

i) Cost of Components

Table 13-2 Cost of Components required

S. N.	Components	Rate(NRs.)	Specifications	Qty	Cost(NRs.)
1	FNR Gear Box with differential	80,000	Gear ratio =12.44 Efficiency = 0.7	1	1,00,000
2	Electric Motor (12KW)	75,000	Rated power=12kw Rated Torque=80Nm Speed = 5000rpm Rated Voltage=96 V	1	75,000
3	Battery	1,00,000	Battery Capacity = 150Amp-h	2	2,00,000
5	Battery Charging Kit	25,000		1	25,000
6	Motor Controller	5,000		1	5,000
7	Brake System	30,000	Master Cylinder Bore Diameter = 19.05mm Rotor diameter=300mm Calipers diameter =38mm(front),20mm(rear)	1	30,000
8	Steering System	15,000	Diameter of steering wheel = 300mm Diameter of pinion = 32mm	1	15,000
9	Suspension System	10,000	Wishbone length = 500mm Spring diameter=10mm(front),12mm (rear)	4	40,000
10	Miscellaneous	-		-	40,000
Total Cost					5,30,000

iii) Cost of resources

Table 13-3 Cost of resources

S.N.	Resources	Cost(NRs.)
1	Human Resources	30,000
2	Space	20,000
3	Machining processes	20,000
4	Welding	5,000
5	Designing	10,000
6	Designing software	10,000
7	Transportation	5,000
Total cost		1,00,000

Therefore, the total cost of our project is the total cost of making roll cage, total cost of components and resources which will be:

=NRs.1,36,00 + NRs.5,30,000 + NRs.1,00,000

= NRs. 7,66,000/-

CHAPTER FOURTEEN
LIMITATIONS AND FUTURE SCOPE

Although the EATV is designed considering various loading conditions, still there exist some limitations. The limitations of the vehicle are enlisted as below:

1. The fine meshing of the elements for the analysis or grid independent test could not be done due to the insufficient capabilities of the laptop available.
2. The weight of the passenger and driver is limited to 150 kg in total. No extra loading or extra loading is considered.
3. The dimension of the tubular shaft is used same over the entire frame which increase the weight and cost of the fabrication of the.
4. The minimum factor of safety at the critical point during the impact is low to one and at some point below one. This means there is high chance of breaking of the members during breakdown.

To improve the limitations of the EATV frames, we suggest following activities to be done to the EATV frames during the fabrication process.

1. A improved quality of processor could be used for the mesh quality of the geometry by doing grid independent test.
2. The frame could be provided with bigger dimensional tubes at the base of the material so that it could support higher weight of the passenger and the driver in combination.
3. The dimension of the tubes to be used over the frames can be changed according to the stress distribution is low so that the weight of the vehicle can be decreased. The members used in the top part of the frame can be fabricated using small tubes.
4. The portion or members having the low factor of safety can be eliminated or FOS can be increased using the extra member to support the critical members and sharing the maximum stress developed over the longer area.
5. For the optimization of the frame regarding the weight distribution of the material used, we can select the material based on the ratios of Young's modulus to yield stress of the material. That is

Mass of frame or material (m) α (Young's modulus (E) / Yield stress (σ))

CHAPTER FIFTEEN
CONCLUSION

The utilizations of the EATV can be done for the conservational areas for the safaris and for other local tourist destination where it can attract the local tourist and promote the use of the electricity and other local resource. The strength of the vehicle depends on its frame, so we have tested our conceptual frames for various loading conditions and found that the use of the 33.7mm hollow tube with the thickness of 2.6mm will be enough to carry 2 passenger of average weight of 75Kg each along the other weight of batteries, motor and other components when the manufacture is done using the AISI1020 steel. Using this dimension and material, for the normal loading of the vehicle we have got the factor of safety above 8 for model-two frame which is selected as suitable design for the manufacturing of the EATV.

While considering the various impacts loading or the crash testing of the both models, we find that the model two could resist the impact loading. In chapter 5, we saw that the cabin where the passenger sits for the travel is safe for the use even during the crash. There is present of certain cracks or failures at some joints which does not affect the passenger's safety and those failure members can be replaced after the crash. We also tested the welding of the joints in which, butt joint is used and the electrode of rating 80KSi is suitable for the use as it gives the higher factor of safety for the use among the range that is used for the pipe welding. In case of the tire of the EATV we suggest to use the radial ply tire of minimum 4 plies and size of 25x8-12* – Height x Width - Wheel Diameter.

In this way we can conclude that our designed roll cage model-two is suitable to be used for the EATV and is safe for the passenger to use for the various impacts loads.

The power requirement of the vehicle for the system was calculated. For the vehicle of 540kg mass, the overall tractive power requirement to run the EATV at average velocity of 25km/hr on 25% grade was found to be 10.57kW. The torque and rpm requirement of the motor was found to be 52.43Nm and 4130 rpm revolution. Assuming the range of the E-ATV to be 40km in once full charged and the average speed of the E-ATV is taken as 25 km/hr, the operating time of the vehicle can be calculated to be 1.6 hour. For the aforementioned condition, the battery energy capacity was required to be 24kW-hr analytically.

From the Simulink Simulation, it is found that the average energy consumed by the Electric ATV for the provide drive cycle is 59.35 Wh/km for 0 % grade condition and 371.23 Wh/km for uphill 25% grade condition.

Suspension systems was selected on the basis of requirement and geometry for our E-ATV, availability and ease of manufacture. All - terrain vehicles are to be used in uneven terrain and off roads so suspension comes into major play. They require independent suspension system. So considering all the parameters double wishbones system was selected. The analysis of coil spring of suspension system was done in ANSYS and simulated in Simulink.

For braking system, disc brakes were selected. Brake prime concern is to lock both the wheels at same time. So, different caliper at front and rear wheels were used. For front wheel larger caliper size of 38 mm was used and for rear wheel smaller caliper of 20 mm was used. This was considered taking in mind the availability in market.

Similarly, for steering system we have used rack and pinion manual steering system rather than power steering system. We have also designed the rack and pinion steering system in Solidworks.

REFERENCES

Aditya Kumar Mohanty, A. J. (2018). Design and Development of Roll Cage :-A Review. *International Research Journal of Engineering and Technology (IRJET)*, 1736-1740.

Aditya Sinha, A. J. (2018). Design, Analysis and Simulation of Double Wishbone Suspension System for Formula Student Vehicle. *International Research Journal of Engineering and Technology*, 1427-1432.

Aksh Patel, N. B. (2019, June 6). Design Methodology and Manufacturing of Rack and Pinion for All Terrain Vehicle. *International Journal of Engineering Research and Technology*.

Akshay G Bharadwaj, S. M. (2016). Design, Analysis, Simulation and Validation of Suspension System for an Electric All-Terrain Vehicle. *International Journal of Innovative Research in Science, Engineering and Technology*, 21228-21238.

Akshya Pawar, S. Z. (2018, March). Design of Steering System for All terrain Vehicle. *International Research Journal of Engineering and Technology (IRJET)*.

Aman Sharma, P. A. (2018, May). DESIGN, ANALYSIS AND FABRICATION OF BRAKING SYSTEM WITH REAR INBOARD BRAKES IN BAJA ATV. *International Research Journal of Engineering and Technology, 5*(5), 4340-4345.

Arthur P. Boresi, R. J. (2003). *ADVANCED MECHANICS OF MATERIAL*. JOHN WILEY AND SONS, INC.

AZoM. (2012, 07 05). *AZO MATERIALS*. Retrieved 04 22, 2020, from azom.com: https://www.azom.com/article.aspx?ArticleID=6114#:~:text=AISI%201020%20has%20low%20hardenability,high%20ductility%20and%20good%20weldability.

Bansal, R. (2009). *A textbook of Strength of Material.* Laxmi Pubications Pvt. Ltd.

Bernardini, A. B. (2015). *Technological solutions aiming at recovering metro bramulti-criteria analysis case study*. BDC – Bollettino del Centro Calza Bini – Università degli Studi di Napoli Federico.

BIS Research. (2020). *Electric All-Terrain Vehicle (ATV) and Utility-Terrain Vehicle (UTV) Market - Analysis and Forecast, 2019-2030.* BIS Research.

Carlos Montero, D. M. (2015). *Modeling and torque control for a 4-wheel- drive electric vehicle.* Retrieved from ResearchGate.

Cartrade Editorial Team. (n.d.). *Cartrade* . Retrieved t, from www.cartrade.com: www.cartrade.com/authors/cartrade-editorial-team

Chevrefils, A. R. (2008). *Modeling and Design of an Electric All-Terrain Vehicle.* Winnipeg, Manitoba: Department of Electrical and Computer Engineering, University of Manitoba.

Dana, L. (2020, December). Retrieved from Dana: https://www.dana.com.br/arquivos/fora-de-estrada/dext-OPECondSP.pdf

De Pinto S., C. P. (2014). A Four-Wheel-Drive Fully Electric Vehicle Layout with Two- Speed Transmissions" . Retrieved from https://www.researchgate.net/publication/282567099_A_Four-Wheel-Drive_Fully_Electric_Vehicle_Layout_with_Two-Speed_Transmissions

Electric Motorsport, I. 2. (2021, January). Retrieved from Electricmotorsport: https://www.electricmotorsport.com/customer/account/index/

Electrical4u. (2017). Torque Slip Characteristics.

Exchange, E. S. (2013). Continuous Rating of Electric Motors.

FS2017MODS. (2019, september 14). *MODSFS2017*. Retrieved january 25, 2020, from FS2017MODS

George E. Dieter, L. C. (2009). *Engineering Design.* McGraw Hill HigherEducation.

grimmermotors. (n.d.). *grimmermotors*. Retrieved january 25, 2020, from grimmermotors: https://grimmermotors.co.nz/clutch-platedisc-repair-replacement/

Hibbeler, R. (2011). *Mechanics of Material.* Pearson Prentice Hall.

Hilltop tire services. (2019, september 25). *Hilltop tire services*. Retrieved november 09, 2020, from www.hilltoptireservice.com: https://www.hilltoptireservice.com/Blog/ArticleID/394/How-to-choose-the-right-tires-for-your-ATV

Howacarworks. (n.d.). *howacarworks*. Retrieved january 24, 2020, from howacarworks.com: www.howacarworks.com/basics/how-the-steering-system-works

Kenworth Truck Company. (2012, August). Kenworth Heavy Duty Body Builder Manual - 2012. *Kenworth Heavy Duty Body Builder Manual*.

Luthra, G. (2017). Comparison Of Characteristics Of Various Motor drives Currently Used In Electric Vehicle Propulsion System. *International Journal of Mechanical And Production Engineering*.

Merriam-Webster. (n.d.). (n.d.). *Merriam-Webster. (n.d.).* Retrieved 11 07, 2020, from Merriam-Webster. (n.d.).: https://www.merriam-webster.com/dictionary/tire

Mishra, P. (2018, june 15). *Types of Braking System in Automobile*. Retrieved 1 18, 2021, from Mechanical Booster: https://www.mechanicalbooster.com/2018/06/types-of-braking-system.html

Mototribology. (2018). Brake Fluid.

Mr. Kshitij N Sable, M. A. (2019). Design and Analysis of the Steering System for All-Terrain Vehicle. (IJSRD, Ed.) *International Journal for Scientific Research and Development*.

Ms. Vaishali Bakshi, P. M. (2017). Drive Selection and Performance Evaluation of Electric and Hybrid Electric Vehicles. *International Journal of Mechanical And Production Engineering, ISSN: 2320-2092, Volume- 5, Issue-6*.

Naveen Kumar, S. V. (2020). Roll Cage and Material Selection. *INTERNATIONAL JOURNAL OF ENGINEERING RESEARCH & TECHNOLOGY (IJERT)*.

Riberio, G. V. (2016). *Design of an Electric Powertrain for an All-Terrain Vehicle.*

Richard G. Budynas, J. K. (2011). *Shigley's Mechanical Engineering Design.* Mcgraw Hill.

Semanticscholar. (2015, October). Permanent Magnet Synchronous Motor.

The Motorom Budsman. (2019, November).

TWI Ltd. (n.d.). Retrieved 02 12, 2021, from TWI Ltd. website: www.twi-global.com

Utires. (2017, June 16). *Utires*. Retrieved november 08, 2020, from www.utires.com: https://www.utires.com/articles/what-all-terrain-tires-are-used-for-and-how-they-differ-from-other-tires/

Vivek Singh Negi, N. D. (2017, November). Design of Braking System of BAJA Vehicle. *International Journal of Advance Engineering and Research, 4*(11), 1059-1064.

ANNEX A
MATLAB CODE

```
clear
clc
%Define Model Parameters
g                 = 9.81;     %Acceleration due to gravity [m/s^2]
massVeh           = 540;      %Mass of the E-ATV[kg]
Wheel_radius      = 0.3175;   %Radius of Wheel [m]
G                 = 12;       %Gear Ratio of Transmission
n_G               = 0.7;      %Efficiency of Transmission
c_r               = 0.045;    %Coefficient of rolling resistance
drivecycle_dist   = 7.898;    %DriveCycle distance covered [km]

%Define Simulation Parameters
tFinal     = 100;
tStepMax   = 10;
tStepMin   = 0.5;

%Define Initial Conditions

%Run the Simulink Model using the 'sim' Command
sim('EnergyConsumptionCalc.slx')

%Extract the data generated by the 'To Workspace' block using a
%time series structure
t                 = Vehicle_Data.time;
vel               = Vehicle_Data.Data(:,1);
Tractive_force    = Vehicle_Data.Data(:,2);
veh_Power         = Vehicle_Data.Data(:,3);
veh_energyreqd    = Vehicle_Data.Data(:,4);
kWhr              = Vehicle_Data.Data(:,5);
avgEnergyConsump  = Vehicle_Data.Data(:,6);
```

```
%Displaying the result of data extraction
figure
hold on
plot(t, kWhr)
xlabel('Time [s]')
ylabel('Total Tractive Energy[kW-hr]')
grid on
```

ANNEX B
TECHNICAL DRAWING OF THE ME1115 12KW BLDC MOTOR (ELECTRIC MOTORSPORT, 2021)

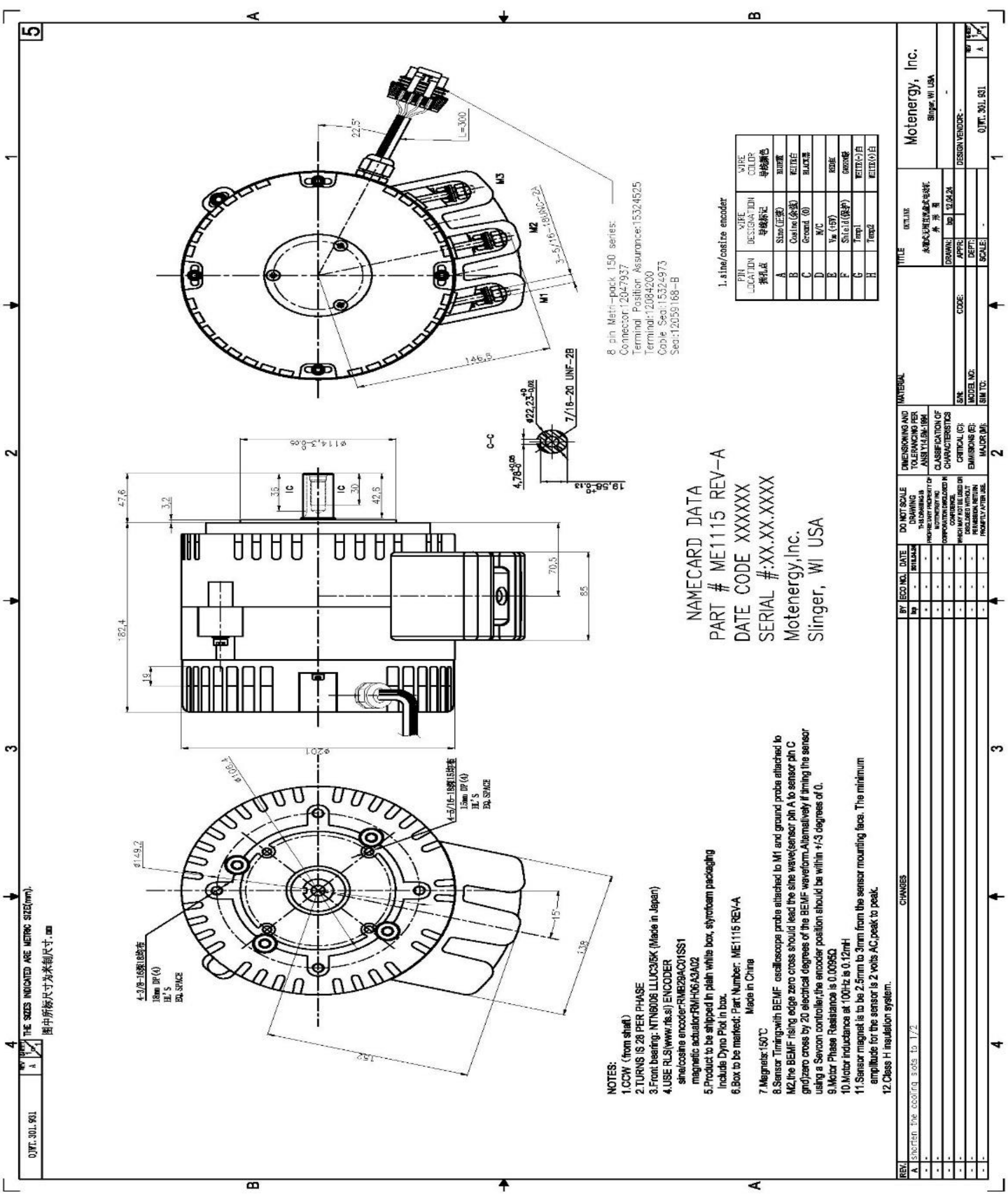
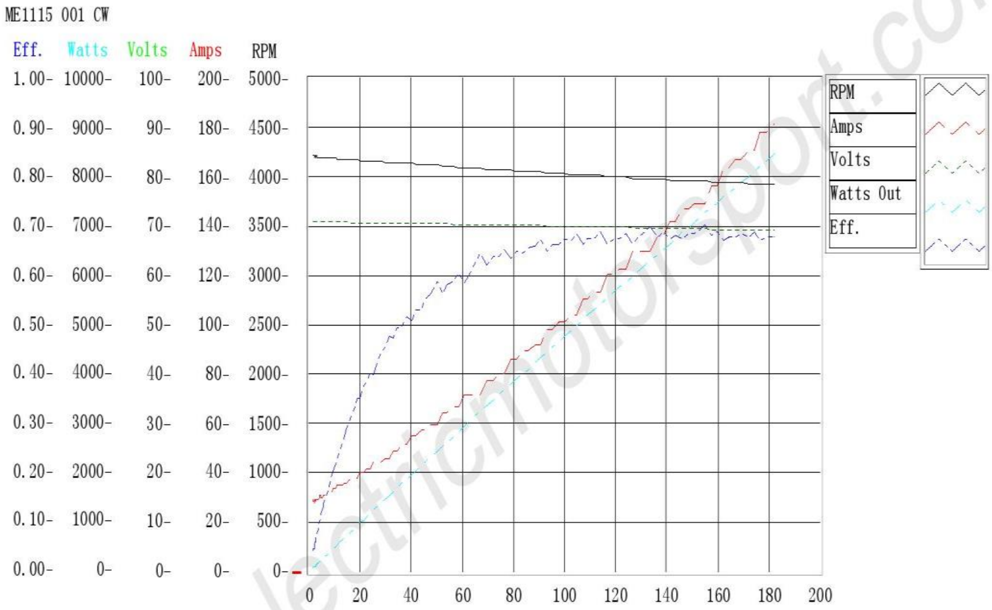


Figure 18-1 Technical Drawing of 12KW BLDC Motor

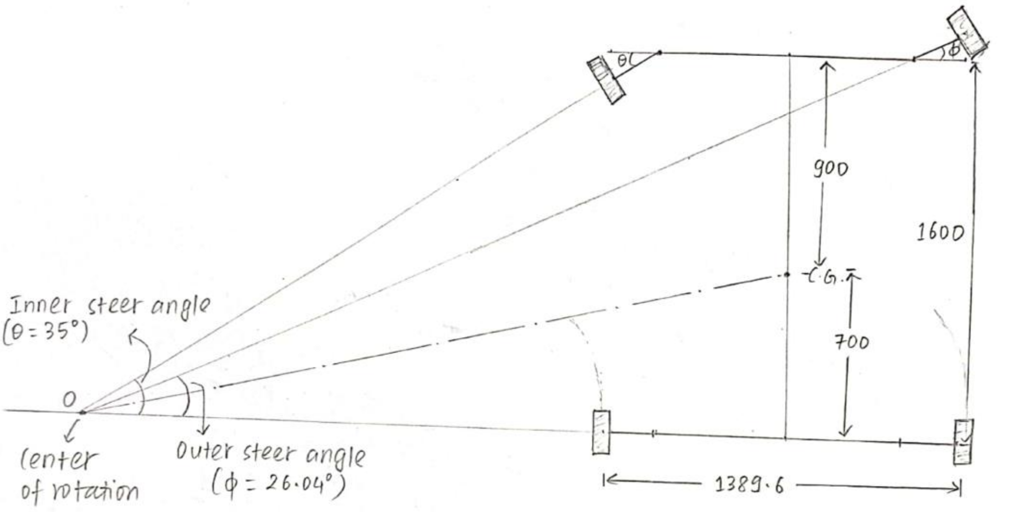
ANNEX C
ME1115 PERFORMANCE CURVES (ELECTRIC MOTORSPORT, 2021)



ANNEX D
GEOMETRICAL INTERPRETATION OF TURNING RADIUS

Geometrical Representation of Turning Radius:

Scale : 200mm = 1cm
Dimensions in millimeters.



Here, Inside turning radius (R_i) = 2789.5 mm
Outside turning radius (R_o) = 3644.66 mm.
and,
From geometry; we obtain,
Turning radius of CG (R_{cg}) = 14.5 cm
= 14.5 × 200
= 2900 mm.
∴ R_{cg} = 2900mm.

Figure D.1 Geometrical Interpretation of Turning Radius

ANNEX E
DIFFERENT VIEWS OF STEERING SYSTEM

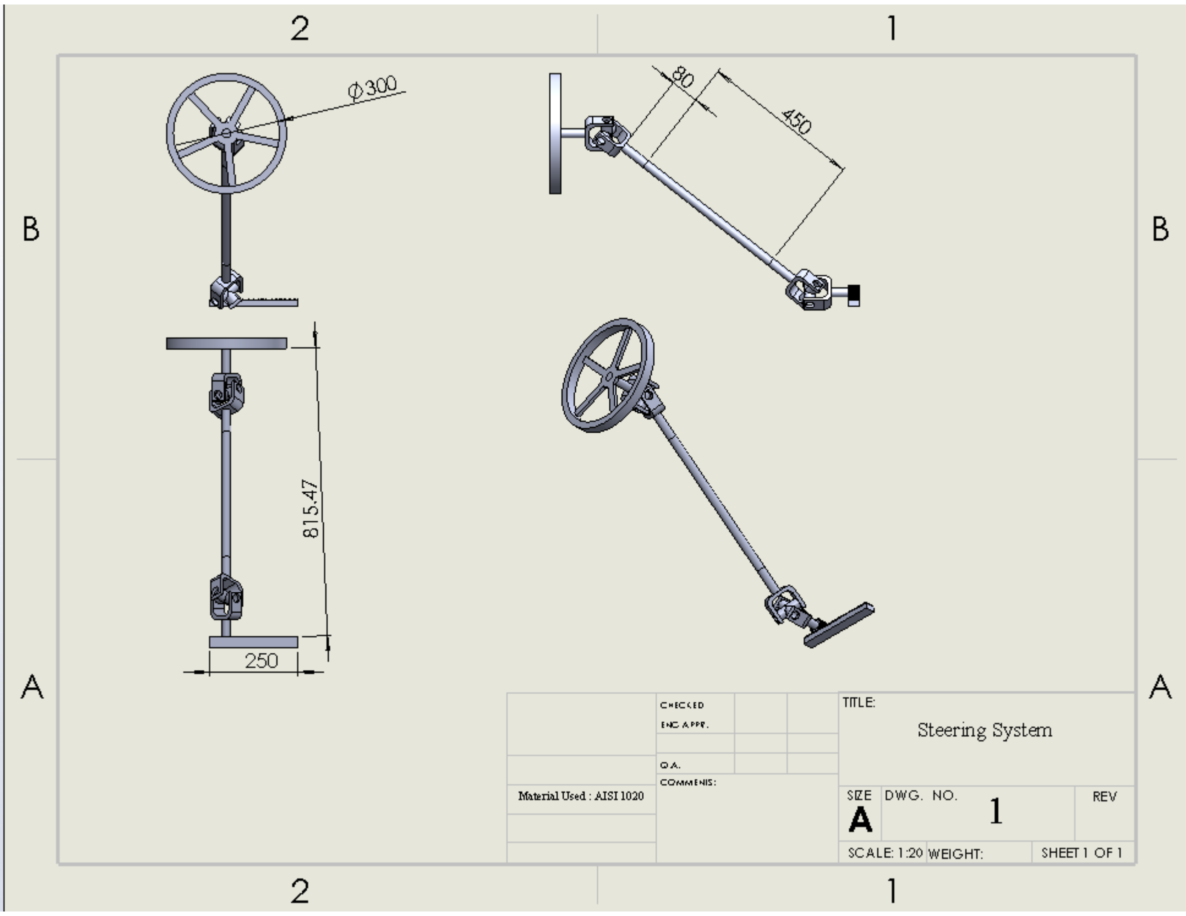


Figure 21-1 Different views of Steering System

ANNEX F
3D MODELS

The designed two models of the roll-cage are shown using the front view, top view, side view and isometric view in the figure below.

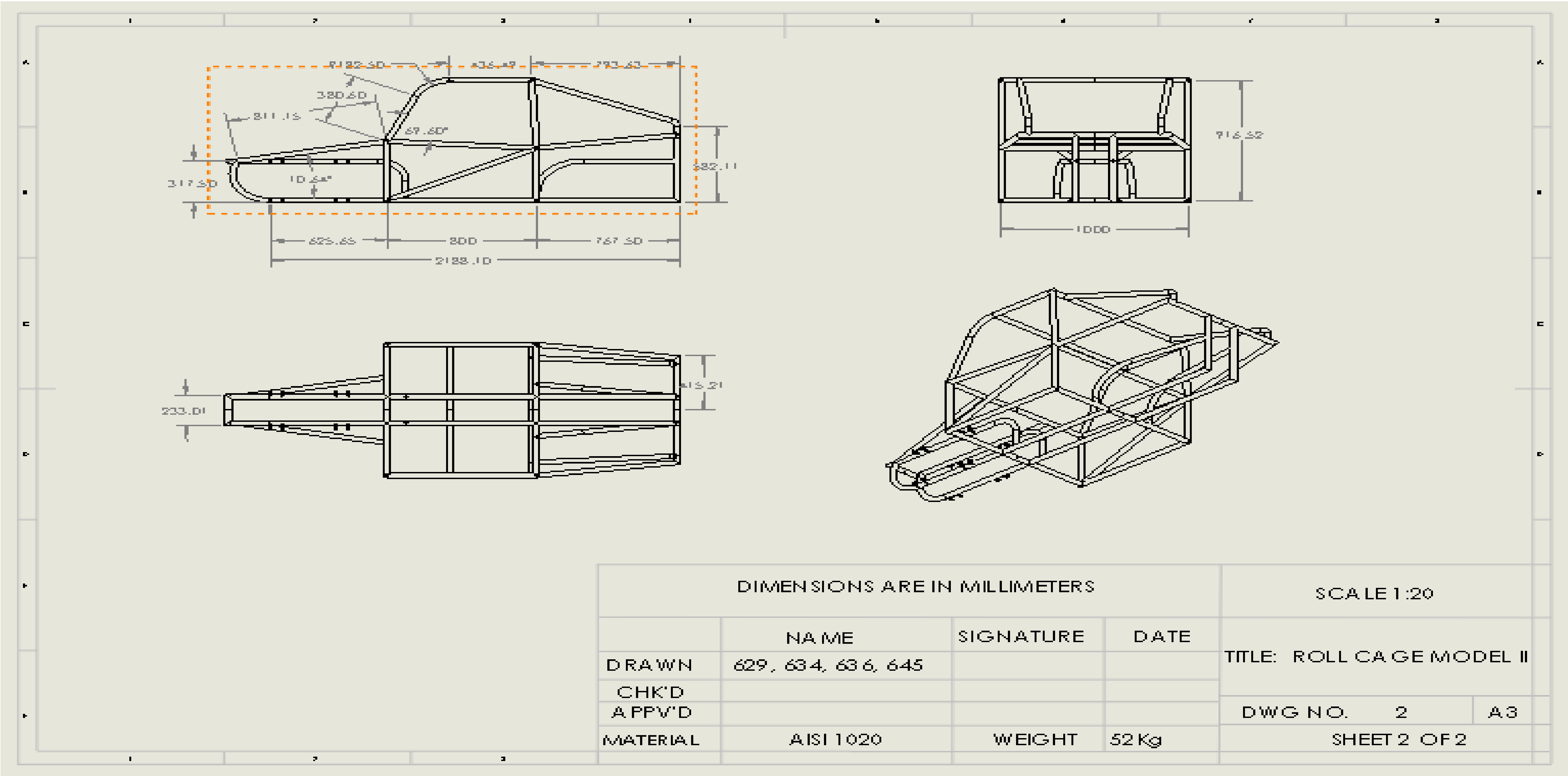


Figure 22-1 Solidworks Design of Model-one

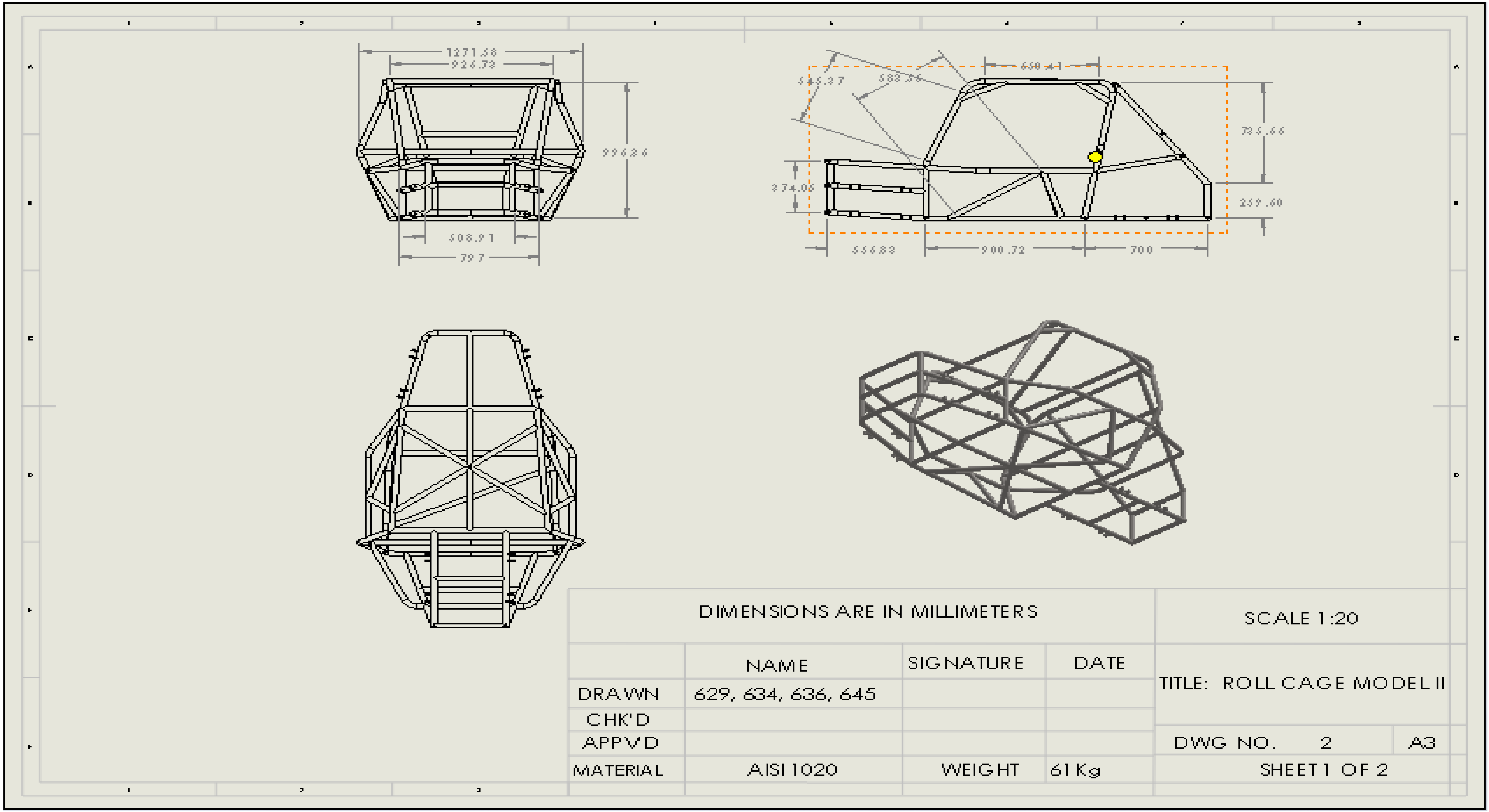
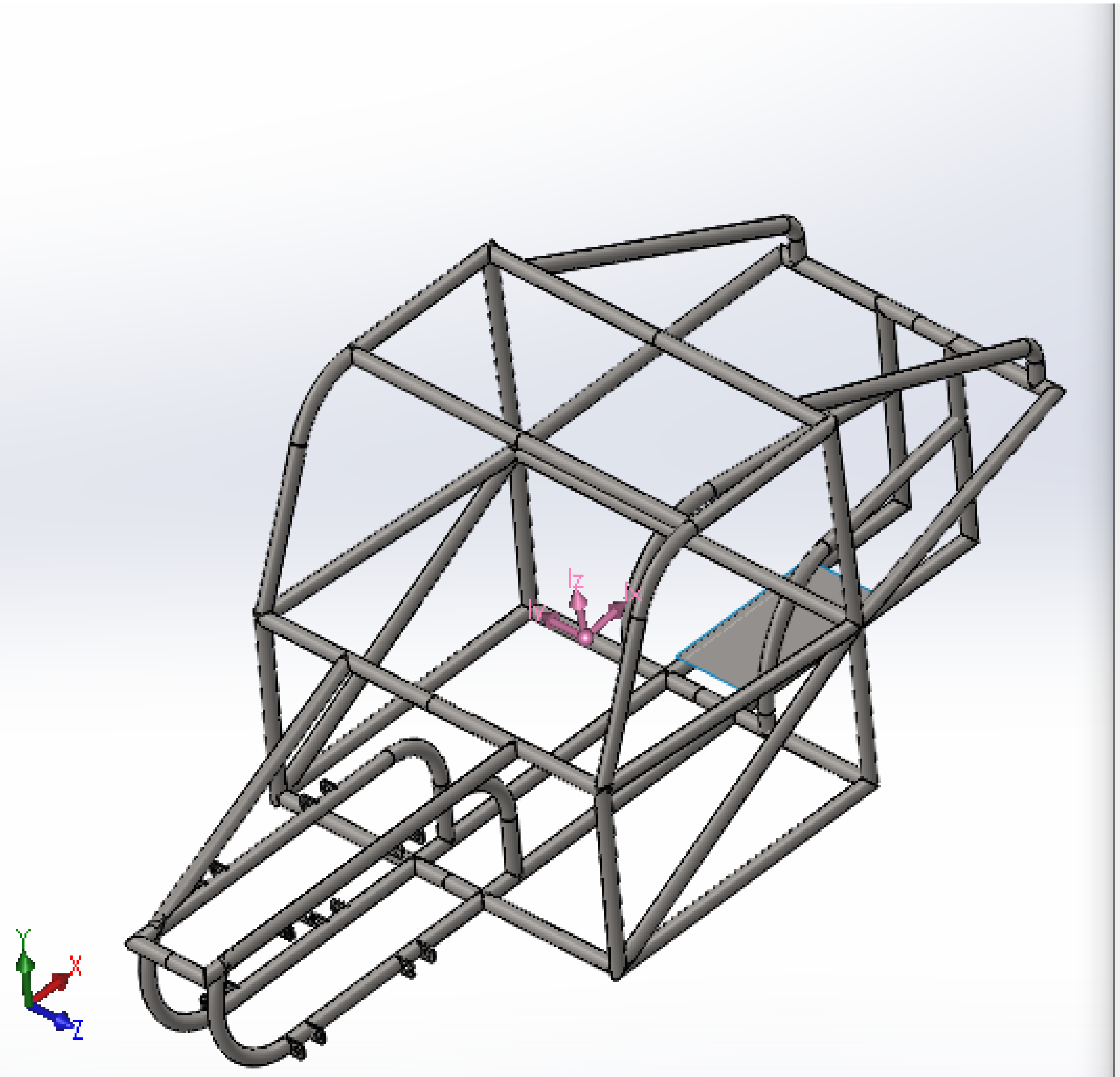


Figure 22-2 Solidworks Design of Model-two

ANNEX G
MASS PROPERTIES

The mass properties of the two models designed using the software is derived using the Solidworks itself through which the model is drawn. The center of mass is also shown in the figure below



Mass properties of model I
Configuration: Default<As Machined>
Coordinate system: -- default --

Density = 0.01 grams per cubic millimeter

Mass = 52554.52 grams

Total weld mass = 0.00 grams

Volume = 6652470.87 cubic millimeters

Surface area = 6637219.98 square millimeters

Center of mass: (millimeters)
X = 1122.78
Y = 338.31
Z = -99.85

Principal axes of inertia and principal moments of inertia: (grams * square millimeters)
Taken at the center of mass.

Ix = (0.99, 0.11, -0.00)	Px = 10397725355.64
Iy = (-0.00, -0.00, -1.00)	Py = 25653064699.21
Iz = (-0.11, 0.99, -0.00)	Pz = 27251184199.90

Moments of inertia: (grams * square millimeters)
Taken at the center of mass and aligned with the output coordinate system.

Lxx = 10617842905.02	Lxy = 1913428750.47	Lxz = -8589182.32
Lyx = 1913428750.47	Lyy = 27031071422.33	Lyz = -625313.54
Lzx = -8589182.32	Lzy = -625313.54	Lzz = 25653059927.41

Moments of inertia: (grams * square millimeters)
Taken at the output coordinate system.

Ixx = 17156899397.57	Ixy = 21876135756.25	Ixz = -5900590484.51
Iyx = 21876135756.25	Iyy = 93807032419.10	Iyz = -1775973310.76
Izx = -5900590484.51	Izy = -1775973310.76	Izz = 97920088012.45

Figure 23-1 Mass properties of model-one

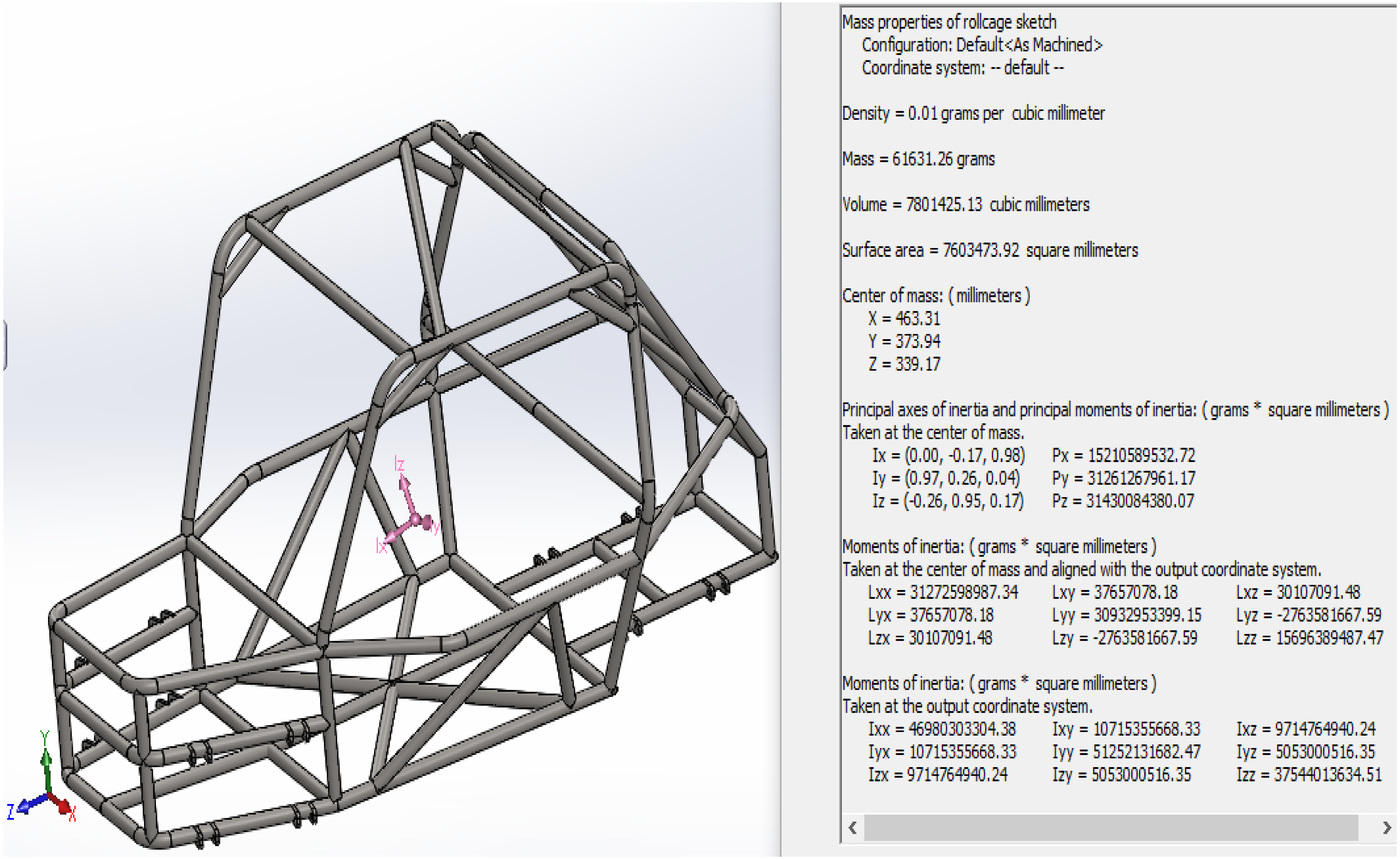


Figure 23-2 Mass properties of model-two