University of Khartoum

Faculty of Engineering

Mechanical Engineering Department

Dune Buggy Design

A thesis submitted in partial fulfillment of the requirements for the degree of B.Sc. in Mechanical Engineering.

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Dedication

To our Parents who have always been there for us in our highest and lowest

To our Brothers and sisters who have always shown us support in and stood by us in all times.

Our Teachers who were always guide us to success and gave us advises and help to excel above others.

Our friends who cheered, rooted and gave us all the help we need to accomplish our goals.

And to Every person who shared their knowledge and Ideas with us.

Wishing you all the best in life.

Thank You ..

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Firstly thanks to Allah, our creator above anything for creating everything and giving us the ability and strength to accomplish our task.

We are grateful to our teacher DR. Muddawi Ibrahim Adam. Who gave us moral support and guided us in different situation in difficult matters. He had been very kind and patient while providing us the outlines of this project and correcting our information.

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And all the workers at Lambda engineering.

Abstract:

With the continuous increment of prices in Sudan the cost of personal transport continues to rise. This increase in costs prevents people from owning their own personal transport media.

This project is attempting to solve this problem by designing a vehicle that is simple, adequate, cost saving with appropriate life time. This problem was solved using design concepts, computerized simulation for behavioral observation with the help of the commercial package SOLIDWORKS.

المستخلص:

مع استمرار الزيادة في الأسعار في السودان فإن تكلفة وسائل الإنتقال الشخصية في إرتفاع مستمر. هذه الزيادة في الأسعار تمنع بعض المواطنين من امتلاك وسائل انتقالهم الشخصية.

هذا المشروع يهدف لأن يحل هذه المشكلة عن طريق تصميم عربة بسيطة، مناسبة، قليلة التكلفة و ذات عمر مناسبز هذه المشكلة تم حلها بإستخدام أفكار تصميمية وعمليات محاكاة محوسبة لمراقبة نمط تصرفات العربة في الواقع بإستخدام الحزمة SolidWorks

Table Of Contents

Dedication……ii
Acknowledgment……iii
Abstract……iv
المستخلص……v
List of symbols and abbreviations……x
List of Figures……xi
List of Tables……xii

1. Introduction……1

1.1 Background……1

1.2 Problem statement……2

1.3 Solution of the problem……2

1.4 Scope……2

2. LITTERATURE REVIEW……3

2.1 Introduction to Dune Buggy……3

2.1.1 F1 Race Cars……3

2.2 History of the Dune Buggy……4

2.3 Methods of Designing the Dune Buggy……5

3. Design……6

3.1 Introduction to CAD……6

3.1.1 The History of CAD……6

3.1.2 What is Solidworks……7

3.1.2.1 Solidworks Simulation Solver……8

3.1.2.2 Mesh definition……..8

3.1.2.3 Finite Element Analysis……9

3.2 Vehicular Design……10

3.2.1 Conceptual Design……10

3.2.2 Chassis Frame……10

3.2.2.1 Functions Of The Chassis Frame……10

3.2.2.2 Types of the Chassis Frame……11

3.2.2.3 Chassis Design……11

3.2.3 Engine……19

3.2.3.1 Engine selection……19

3.2.3.2 Engine Mounting……25

3.2.4 Drive Train……26

3.2.4.1 Differential Gearbox……26

3.2.4.2 DriveTrain Elements……27

3.2.4.2.1 Engine-Differential Torque transmitter……29

3.2.4.2.2 Differential Gearbox Output shaft……29

3.2.4.2.3 Constant Velocity Joints……30

3.2.4.2.4 Wheel Hub……30

3.2.4.2.5 Bearing Selection……33

3.4.4.2.6 Brake Selection……35

3.2.5 Suspension System……39

3.2.5.1 Types of Suspension……40

3.2.5.2 Suspension Design……..41

3.2.5.2.1 Springs and Shock Absorbers Selection……..41

3.2.5.2.2 Rear Suspension Simulation……..44

3.2.5.2.3 Front Suspension Simulation……..52

3.2.5.3 Steering Design and Selection……60

3.3.5.3.1 Steering Concerns……60

3.2.5.3.2 Steering Solution……61

4. MANUFACTURING……63

4.1 Chassis……63

4.1.1 Elements……63

4.1.2 Manufacturing Processes……63

4.1.3 Method……64

4.2 Drive Train……64

4.2.1 Manufactured Elements……64

4.2.2 Purchased Elements……64

4.2.3 Manufacturing Processes……65

4.2.4 Method……65

4.3 Suspension……65

4.3.1 Manufactured Elements……65

4.3.2 Purchased Elements……65

4.3.3 Manufacturing Processes……66

4.3 4 Method……66

4.4 Steering……67

4.4.1 Manufactured Elements……67

4.4.2 Purchased Elements……67

4.4.3 Manufacturing Processes……67

4.4.4 Method……67

4.5 Assembly……68

4.5.1 Drive Train……68

4.5.2 Suspension……68

4.5.3 Steering……69

5.RECOMMENDATIONS AND FUTURE WORK……70

Refrences……71

Appendices……72

A-1 Bearing Table……72

A-2 Material Table……74

A-3 Engineering Drawings……75

List of symbols and abbreviations:

M:	Mass	(kg)
I:	Inertia	$(kg.m^2)$
ω:	Rotational speed	(rad./s)
P:	Power	(watt)
P:	Pressure	(Pa)
μ:	Friction coefficient	
A:	Area	(m^2)
c_d:	Drag coefficient	
T:	Torque	(N.m)
τ:	Shear stress	(MPa)
ξ:	Damping ratio	
f:	Frequency	(Hz)
Cc:	critical damping constant	$(N.s/m)$
C:	damping constant	$(N.s/m)$
K:	Spring stiffness	(N/m)
OD:	outer diameter	(m)
ID:	Inner diameter	(m)
r_e:	Effective radius	(m)
r_i:	inner radius	(m)
r_o:	Outer radius	(m)
t:	thickness	(m)
v:	velocity	(m/s)
α:	linear acceleration	(m/s^2)
t:	time	(second)
K.E:	kinetic energy	(Joule)
g:	gravity	(m/s^2)
f_f:	friction force	(N)
f_d:	drag force	(N)
F:	force	(N)
L:	length	(m)
l_h:	life time	(Hour)
L_d:	design life time	(rev)
E:	energy	(Joule)
θ:	angle	(rad)

List of figures:

Figure 2-1: Example of a Dune Buggy…….. 4
Figure 3-1: Ariel Atom……..11
Figure 3-2: Model Isometric View……12
Figure 3-3: Chassis Model Meshing……16
Figure 3-4: Chassis Model Simulation Stress plot……..17
Figure 3-5: Chassis Model Simulation Deflection plot……17
Figure 3-6: Chassis Model Simulation Factor of Safety Plot……..18
Figure 3-7: Selected Engine……..25
Figure 3-8: Differential Gearbox general Layout……27
Figure 3-9: Torque transmission path to wheels……..27
Figure 3-10: Splined shaft……..28
Figure 3-11: wheel hub……31
Figure 3-12: Brake Configuration……..38
Figure 3-13 : Independent Suspension Advantages over dependant……..41
Figure 3-14: rear suspension meshing……..48
Figure 3-15: Rear Suspension simulation Stress plot……..49
Figure 3-16: Rear Suspension Simulation Deflection plot……50
Figure 3-17: Rear Suspension Simulation Factor of Safety Plot……51
Figure 3-18 : Front suspension Simulation meshing……57
Figure 3-19 : front suspension simulation stress plot……57
Figure 3-20: front suspension simulation Deflection plot……58
Figure 3-21: front suspension simulation Strain plot……..59
Figure 3-22: Turning radius difference……..60
Figure 3-23: Ackermann true geometry……..61
Figure 3-24: Steering hand……62

List of tables:

Table 3-1: Chassis Study Properties……13
Table 3-2: Chassis Material Properties……14
Table 3-3: Chassis Fixture Location……14
Table 3-4: Chassis Forces……15
Table 3-5: Chassis Meshing Information……16
Table 3-6: Chassis Simulation Mesh Information……16
Table 3-7: Masses and centre of gravity location……20
Table 3-8: product of masses and their distances……21
Table 3-9: distance between parts CoG and buggy's CoG……22
Table 3-10: Resultant Inertia……23
Table 3-11: rear suspension simulation properties……44
Table 3-12: Rear suspension simulation parts and materials……45
Table 3-13: rear suspension simulation units……46
Table 3-14: Rear suspension simulation fixtures……46
Table 3-15: rear suspension simulation loads……46
Table 3-16: rear suspension simulation mesh information……47
Table 3-17: rear suspension simulation mesh details……47
Table 3-18: Front suspension simulation units……52
Table 3-19: front suspension simulation properties……53
Table 3-20: front suspension material properties……54
Table 3-21: Front suspension simulation fixtures……55
Table 3-22: front suspension simulation loads……55
Table 3-23: front suspension simulation mesh information……56
Table 3-24: front suspension simulation mesh details……56

CHAPTER 1

INTRODUCTION:

1.1 Background:

Cars. Are the First thing to come in mind when talking about mechanical engineering, it is considered the general representative of mechanical engineering, it started with the need of a way to reduce the time consumed to move from location to location

Vehicles have been an important part of our daily life as a transportation medium, there is no middle or higher class house with a good income of money that doesn't have at least one car, it saves both time to reach to a certain destination and the effort required to do so. Vehicles also made travelling through forests and deserts easier for those who are travelling enthusiastic.

Some people depend on vehicles for their income such as taxi drivers and public transportation drivers. It also helps other fields of work most companies and factories depend semi completely on cars in transporting their products, it also helped inviting other fields of careers such as Showrooms that Both display and Sell special types of cars such as supercars or Classics. Beside workshops which are purposed for vehicular maintenance, factories which produce them and their parts.

Some countries are known for their production quality around the world, what most of those countries have in common are their local vehicles manufacturing such as BMW and Mercedes in Germany, Lexus and Toyota in Japan and Cadillac and Ford in the united state of America. We can say that cars in general represent the form of industrial quality the country have

It also used in entertainment such as racing for example as the formula 1 which brings both excitement and entertainment and one of the ideas and object of the project is to bring this excitement in regular and daily used cars .

In the end cars become one of the parts that we cannot give it in our routine and daily live.

1.2 Problem statement:

In this project using local material and components is suggested to build and actual functioning vehicle of a dune buggy class at an economic value, by using only available and high supply assured parts to build a vehicle, Selecting suitable material, dimensions and assembly in attempt to solve the vehicles high price crisis.

1.3 Solution of the problem:

To Solve this problem the Dune Buggy is build in a workshop using construction steel strong enough but cheap for a chassis, A small but power appropriate engine as prime mover. Few options will be considered and the one yielding suitable design and economic overall cost will be chosen.

1.4 Scope:

The scope of this project is divided into:

1. search , Review and gather information about available resources
2. Conceptual Design
3. Computer Aided Design (CAD)
4. Selection of different suitable dimensions, and parts. Analysis of data gathered on all possible scenarios
5. Documentation of the Results
6. Analysis of the result
7. Recommendations

CHAPTER 2

LITTERATURE REVIEW:

2.1 Introduction to Dune Buggy:

Dune buggy cars firstly used as race cars in one of most famous racings and single seat auto racing , the formula 1 racings shows the great capability of the single seat auto racing especially the dune buggy cars.

2.1.1 F1 Race Cars:

The highest class of single-seat auto racing that is sanctioned by the Fédération Internationale de l'Automobile (FIA). The FIA Formula One World Championship has been the premier form of racing since the inaugural season in 1950, although other Formula One races were regularly held until 1983.

Formula One cars are the fastest road course racing cars in the world, owing to very high cornering speeds achieved through the generation of large amounts of aerodynamic downforce. Formula One cars race at speeds of up to 360 km/h (220 mph) with engines currently limited in performance to a maximum of 15,000 RPM. The cars are capable of lateral acceleration in excess of 5g in corners. The performance of the cars is very dependent on electronics – although traction control and other driving aids have been banned since 2008 – and on aerodynamics, suspension and tyres. The formula has radically evolved and changed through the history of the sport.

While Europe is the sport's traditional base, and hosts about half of each year's races, the sport's scope has expanded significantly and an increasing number of Grands Prix are held on other continents. F1 had a total global television audience of 425 million people during the course of the 2014 season. [1]

We can generally say that the F1 race cars are technically enhanced dune buggies.

A dune buggy is a recreational motor vehicle with large wheels, and wide tires, designed for different type of terrains the design is usually a modified vehicle and engine mounted on an open chassis. The modifications usually attempt to increase the power-to-weight ratio by either lightening the vehicle or increasing engine power or both.

Fig 2-1: Example of a Dune Buggy

2.2 History of the Dune Buggy:

The origin of the dune buggy is still blurry to the current day but some say evidence points to them being created around the 1950's. During this time, everyday cars were stripped down to their chassis and larger wheels added to make them safer and easier to use on a beach. In 1958, Pete Beirning of Oceano, CA took a Bug and stripped it down to create a short pan dune buggy. A short pan dune buggy is a buggy which has intentionally been shortened – usually by several inches. A roll cage is often added too. Dune buggies became more common in the 50's and 60's, when there was a huge trend for dune buggy racing. Back then, the buggies were large and fairly unattractive. It was a question of performance over aesthetics. Design was not really a big thing, but the trend for these types of buggies took off tremendously and this attracted lots of attention from thrill seekers and racers looking for something different.

The story which people favor and tend towards is that is originated in the late 70's by Honda Motors in Japan. It started with Honda Odyssey Dune Buggy in 1977 which was a single seat four wheeled all terrain vehicle with a 350cc two stroke engine. In the year 1989 the production of the buggy stopped for no obvious reason without registering a patent which by the time it stopped the production it turned into an open source for everyone to put their ideas and thoughts into it.[2]

2.3 Methods of Designing the Dune Buggy

Dune buggies are usually created using one of three methods.

i. The first involves alteration of an existing vehicle, most notably the original car or chassis, often a Volkswagen Beetle or a purpose-built chassis. The model is nicknamed Bug, therefore the term "buggy"

ii. The second method involves construction of a vehicle frame from scratch; formed and welded using steel tubing. The advantage of this method is that the fabricator can change fundamental parts of the vehicle (usually the suspension) which is the method used in this project

iii. The third method represents a mix of the above two design philosophies, typically constructed from a converted vehicle that has sustained damage from age, hard use, or accidents. This type of creation is called The Boston-Murphy style.[3]

CHAPTER 3

Design

3.1 Introduction to CAD:

Computer-aided design (CAD) is the use of computer systems to assist in the creation, modification, analysis, or optimization of a design. CAD software is used to increase the productivity of the designer, improve the quality of design, improve communications through documentation, and to create a database for manufacturing. CAD output is often in the form of electronic files for print, machining, or other manufacturing operations.

Computer-aided design is used in many fields. Its use in designing electronic systems is known as electronic design automation, or EDA. In mechanical design it is known as mechanical design automation (MDA) or computer-aided design (CAD), which includes the process of creating a technical drawing with the use of computer software.

CAD software for mechanical design uses either vector-based graphics to depict the objects of traditional drafting, or may also produce raster graphics showing the overall appearance of designed objects. However, it involves more than just shapes. As in the manual drafting of technical and engineering drawings, the output of CAD must convey information, such as materials, processes, dimensions, and tolerances, according to application-specific conventions.

CAD is an important industrial art extensively used in many applications, including automotive, shipbuilding, and aerospace industries, industrial and architectural design, prosthetics, and many more. CAD is also widely used to produce computer animation for special effects in movies,

3.1.1 The History of CAD:

In the mid-1960 time frame there were no commercial graphics systems on the market except for the Control Data Digigraphics system The need for computer-based graphic systems to improve the productivity of engineers and drafters was slowly being recognized by large manufacturing companies, especially those in the automotive and defense and aerospace industries. This early work fell into two categories. On one hand, automotive companies such as Renault and Ford focused on the mathematical definition of complex surfaces while other companies, such as Lockheed California focused on improving drafting productivity. The Renault work eventually evolved into Dassault Systèmes's CATIA while Ford's PDGS software is probably still used on occasion today. Lockheed's work, of course, resulted in the CADAM product The decade of the 1980s was perhaps the most significant period regarding the evolution of the CAD industry. At the start of the decade, the industry was dominated by five companies – Applicon, Auto-trol Technology, Calma,

Computervision and M&S Computing (Intergraph). Other companies starting to make themselves felt included McDonnell Douglas Automation, SDRC and IBM
significant industry consolidation began to occur around the mid-1990s and continues as this is written. The major vendors now see themselves as offering more than just CAD and document management
In very simple terms, virtually no product, building, electronic component or system or factory is designed today in a developed country without the use of this technology. It has resulted in more reliable products that are less expensive to produce and are more attractive to potential customers. It has changed technical education and to a significant extent, the practice of numerous professions. Design engineers do analysis today that a few years ago was only done by highly specialized professionals [4]

3.1.2 What is Solidworks:

SolidWorks (stylized as SOLIDWORKS), is a solid modeling computer-aided design (CAD) and computer-aided engineering(CAE) software program that runs on Microsoft Windows. The SolidWorks is produced by the Dassault Systèmes— a subsidiary of Dassault Systèmes, S. A. based in Vélizy, France— since 1997.

SolidWorks is currently used by over 2 million engineers and designers at more than 165,000 companies worldwide.

SolidWorks takes advantage of the familiar Microsoft Windows graphical user interface. This, easy-to-learn tool, makes it possible for you to quickly sketch out ideas, experiment with features and dimensions, and produce models and detailed drawings.

This is CAD software, which helps to create 2D or 3D solid models without any complexity, faster and in the cost effective way. The main advantage of the solid modeler is that very easy to use, simple graphics user interface and much more friendly, as compared with other CAD solid modeling softwares. It contains Solid modeling, Motion, Simulation, Toolbox, Tool Analyst, Circuit Works, Photo View 360, ScanTo3D, e-drawings and DWG editor.

According to MIT Survey in 2006, Solidworks selected as High productivity (Up to 95% than others) CAD software. It provides better design visualization, design better products, faster design iterations, improved communications, design with fewer errors, create more aesthetic design &products and also meets the clients/customer requirements.

The Solidworks have wide range of applications in industries such as Aerospace, Defense, Automotive, Transportation, Consumer products, Electronics, Process Plant, Energy conservation, Construction, Heavy equipment, Machinery, Medical tools, Mold & Tools design, Product design and other engineering services. It helps to design various products

and services, testing them in very cost effective way like Model and prototype testing as you learned earlier in engineering degree class.

3.1.2.1 Solidworks Simulation Solver:

COSMOSWorks is a design analysis system fully integrated with SolidWorks.
COSMOSWorks provides one screen solution for stress, frequency, buckling, thermal, and optimization analyses. Powered by fast solvers, COSMOSWorks enables you to solve large problems quickly using your personal computer. COSMOSWorks comes in several bundles to satisfy your analysis needs. COSMOSWorks shortens time to market by saving time and effort in searching for the optimum.
COSMOSWorks offers different solvers to handle different types and sizes of problems more efficiently. The solvers exploit a new technology for the solution of large systems of simultaneous equations to reduce solution time, disk space, and memory requirements. COSMOSWorks offers the following solvers:

- The Direct Sparse solver
- The FFE solver (iterative)
- The FFEPlus solver (iterative)

SOLIDWORKS Simulation uses the displacement formulation of the finite element method to calculate component displacements, strains, and stresses under internal and external loads. The geometry under analysis is discretized using tetrahedral (3D), triangular (2D), and beam elements, and solved by either a direct sparse or iterative solver. SOLIDWORKS Simulation also offers the 2D simplification assumption for plane stress, plane strain, extruded, or axisymmetric options. SOLIDWORKS Simulation can use either an h or p adaptive element type, providing a great advantage to designers and engineers as the adaptive method ensures that the solution has converged.

3.1.2.2 Mesh definition:

Finite Element Analysis (FEA) provides a reliable numerical technique for analyzing engineering designs. The process starts with the creation of a geometric model. Then, the program subdivides the model into small pieces of simple shapes (elements) connected at common points (nodes). Finite element analysis programs look at the model as a network of discrete interconnected elements.
The Finite Element Method (FEM) predicts the behavior of the model by manipulating the information obtained from all the elements making up the model. Meshing is a very crucial step in design analysis. The automatic mesher in
COSMOSWorks generates a mesh based on a global element size, tolerance, and local mesh control specifications. Mesh control lets you specify different sizes of elements for components, faces, edges, and vertices.
COSMOSWorks estimates a global element size for the model taking into consideration its volume, surface area, and other geometric details. The size of the generated mesh (number of nodes and elements) depends on the geometry and dimensions of the model,

element size, mesh tolerance, mesh control, and contact specifications. In the early stages of design analysis where approximate results may suffice, you can specify a larger element size for a faster solution. For a more accurate solution, a smaller element size may be required. Meshing can generate 3D tetrahedral solid elements or 2D triangular shell elements depending on the selected option when the study is created. Shell elements are naturally suitable for modeling thin parts (sheet metals). Using tetrahedral (solid) elements in meshing thin parts can be very inefficient due to the large number of elements that can be
generated.

3.1.2.3 Finite Element Analysis:

finite element method (FEM) is a numerical technique for finding approximate solutions to boundary value problems for partial differential equations. It uses subdivision of a whole problem domain into simpler parts, called finite elements, and variational methods from the calculus of variations to solve the problem by minimizing an associated error function. Analogous to the idea that connecting many tiny straight lines can approximate a larger circle, FEM encompasses methods for connecting many simple element equations over many small sub-domains, named finite elements, to approximate a more complex equation over a larger domain.

Since the majority of industrial components are made of metal, most FEA calculations involve metallic components. The analysis of metal components can be carried out by either linear or nonlinear stress analysis. Which analysis approach you use depends upon how far you want to push the design

If you want to ensure the geometry remains in the linear elastic range (that is, once the load is removed, the component returns to its original shape), then linear stress analysis may be applied, as long as the rotations and displacements are small relative to the geometry. For such an analysis, factor of safety (FoS) is a common design goal.
Evaluating the effects of post-yield load cycling on the geometry, a nonlinear stress analysis should be carried out. In this case, the impact of strain hardening on the residual stresses and permanent set (deformation) is of most interest.
The analysis of nonmetallic components (such as, plastic or rubber parts) should be carried out using nonlinear stress analysis methods, due to their complex load deformation relationship. SOLIDWORKS Simulation uses FEA methods to calculate the displacements and stresses in your product due to operational loads such as: Forces, Pressures, Accelerations, Temperatures, Contact between components.
Loads can be imported from thermal, flow, and motion Simulation studies to perform multi-physics analysis.

3.2 Vehicular Design

A vehicle is an integrated assembly of different parts and subassemblies provided to perform a variety of tasks including distance travelling, loads carrying, provide comfort zones for passengers.

3.2.1 Conceptual Design:

Main parts of a vehicle are chassis, engine, suspension, steering on. Having all mentioned combined together integrally on four wheels will create a vehicle.

Overall Dimensions:

3 meters length, 1.8 meter width, and 0.7 meter height. The chassis will conform of structural steel tubes welded together. Dimensions are based on the comfort of passengers, having adequate crumble zones in the front and the back. And allow ability to adjust dimensions if necessary.

The Engine will be chosen by compromise between power, size, spare parts availability, fuel consumption and overall cost.

Due to roads condition independent suspension will be designed so that oscillation at one wheel are not transferred to the other.

3.2.2 Chassis Frame:

Chassis is a French term and was initially used to denote the frame parts or Basic Structure of the vehicle. It is the back bone of the vehicle. A vehicle without body is called Chassis. The components of the vehicle like Power plant, Transmission System, Axles, Wheels and Tires, Suspension, Controlling Systems like Braking, Steering etc., and also electrical system parts are mounted on the Chassis frame. It is the main mount in which all the components including the body. So it is also known as a Carrying Unit.

3.2.2.1 Functions Of The Chassis Frame:

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

3.2.2.2 Types of the Chassis Frame:

1. Conventional frame: It has two long side members and 5 to 6 cross members joined together with the help of rivets and bolts. The frame sections generally used:
a. Channel Section - Good resistance to bending.
b. Tabular Section - Good resistance to Torsion.
c. Box Section - Good resistance to both bending and Torsion.

2. Integral Frame: This frame is used nowadays in most of the cars. There is no frame and all the assembly units are attached to the body. All the functions of the frame carried out by the body itself. Due to elimination of long frame it is cheaper and due to less weight most economical also. The only disadvantage is that repairing it is difficult.

3. Semi - Integral Frame: In some vehicles half frame is fixed in the front end on which engine gear box and front suspension is mounted. It has the advantage when the vehicle is met with accident the front frame can be taken easily to replace the damaged chassis frame. This type of frame is used in FIAT cars and some of the European and American cars.

3.2.2.3 Chassis Design:

For this project the conventional type of chassis will be used constituting of structural members connected through welding

A chassis will determine the general look of the vehicle and we will be selecting a model that is appealing to people

The general design of the chassis will be a replica of the Ariel Atom.

figure 3-1: Ariel Atom

Using only the construction steel we were able to replicate the design since the analysis was difficult to perform analytically we will be using SOLIDWORKS Simulation.

Circular structure member with inner dia of 45mm and 2mm thickness is used for primary parts.

Rectangular structure member with dimension of (60*30mm) is used for the seat area.

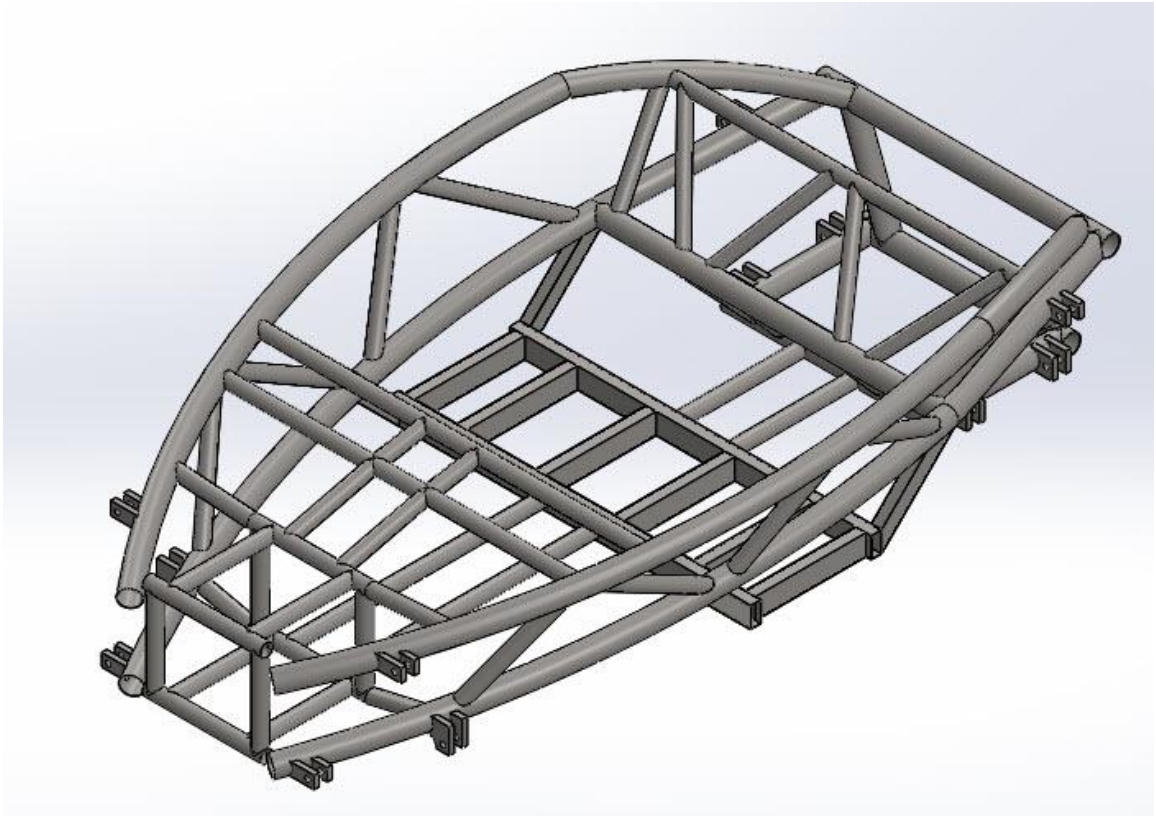


Fig 3-2: Model Isometric View

Analysis Assumptions:

1- Loads are applied normal to Surface
2- Frame Subjected to Bending Only
3- Welded parts are treated as casted
4- Tensile Strength is same for compression and tension
5- Supports are Hinged
6- Forces are Multiplied by a factor to simulate Dynamic Loading
7- Weight of the Chassis itself is included

Study Properties:

Table 3-1: Chassis Study Properties

Study name	Chassis Simulation
Analysis type	Static
Mesh type	Shell Mesh Using Surfaces
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SolidWorks Flow Simulation	Off
Solver type	Finite Element Analysis Plus
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	On
Compute free body forces	On
Friction	On
Use Adaptive Method:	Off

Material Properties:

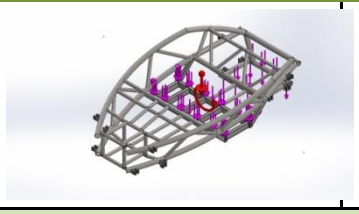
Table 3-2 : Chassis Material Properties

Model Reference	Properties	
	Name:	AISI 1020
	Model type:	Linear Elastic Isotropic
	Default failure criterion:	Max von Mises Stress
	Yield strength:	351.571 MPa
	Tensile strength:	420.507 MPa
	Elastic modulus:	201.1 GPa
	Poisson's ratio:	0.29
	Mass density:	7800 kg/m^3
	Shear modulus:	7.7e+010 MPa
	Thermal expansion coefficient:	0.000015 m/Kelvin

Fixtures:

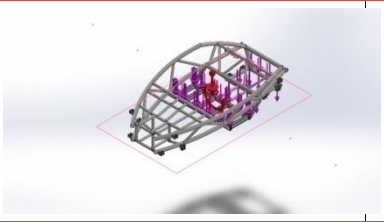
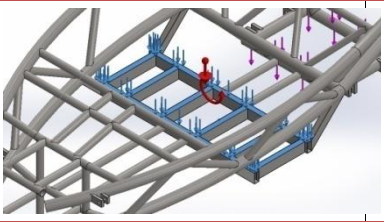
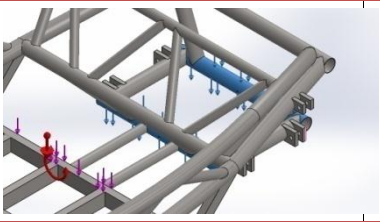
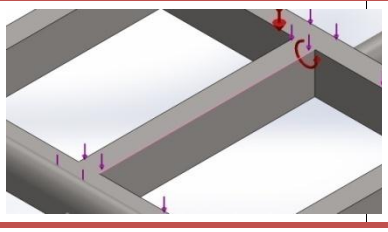
The chassis is supported by suspension separating it from the road and is mounted at the suspension control arms location

Table 3-3 : Chassis Fixture Location

Fixture name	Fixture Image	Fixture Details
Suspension Mounting		Entities: Suspension Mounting Type: Fixed Geometry

Loads:

Table 3-4 : Chassis Forces

Load name	Load Image	Load Details
Gravity		**Reference:** Z Axix **Values:** 9.81 **Units:** SI
Passenger Loads		**Entities:** Seat Frame **Type:** Apply normal force **Value:** 9000 N
DriveTrain Loads		**Entities:** Engine Support Frame **Reference:** Z Axix **Type:** Apply force **Values:** 3600 N
Cornering Load		**Centrifugal, Ref:** Z Axix **Angular Acceleration:** 6.17 rad/s^2

Mesh Information

Table 3-5 : Chassis Meshing Information

Mesh type	Shell Mesh Using Surfaces
Mesher Used:	Curvature based mesh
Jacobian check for shell	On
Maximum element size	0.0834124 m
Minimum element size	0.0278039 m
Mesh Quality	Draft Quality Mesh

Mesh Information – Details

Table 3-6 : Chassis Simulation Mesh Information

Total Nodes	111604
Total Elements	55880
Time to complete mesh(hh;mm;ss):	00:00:15

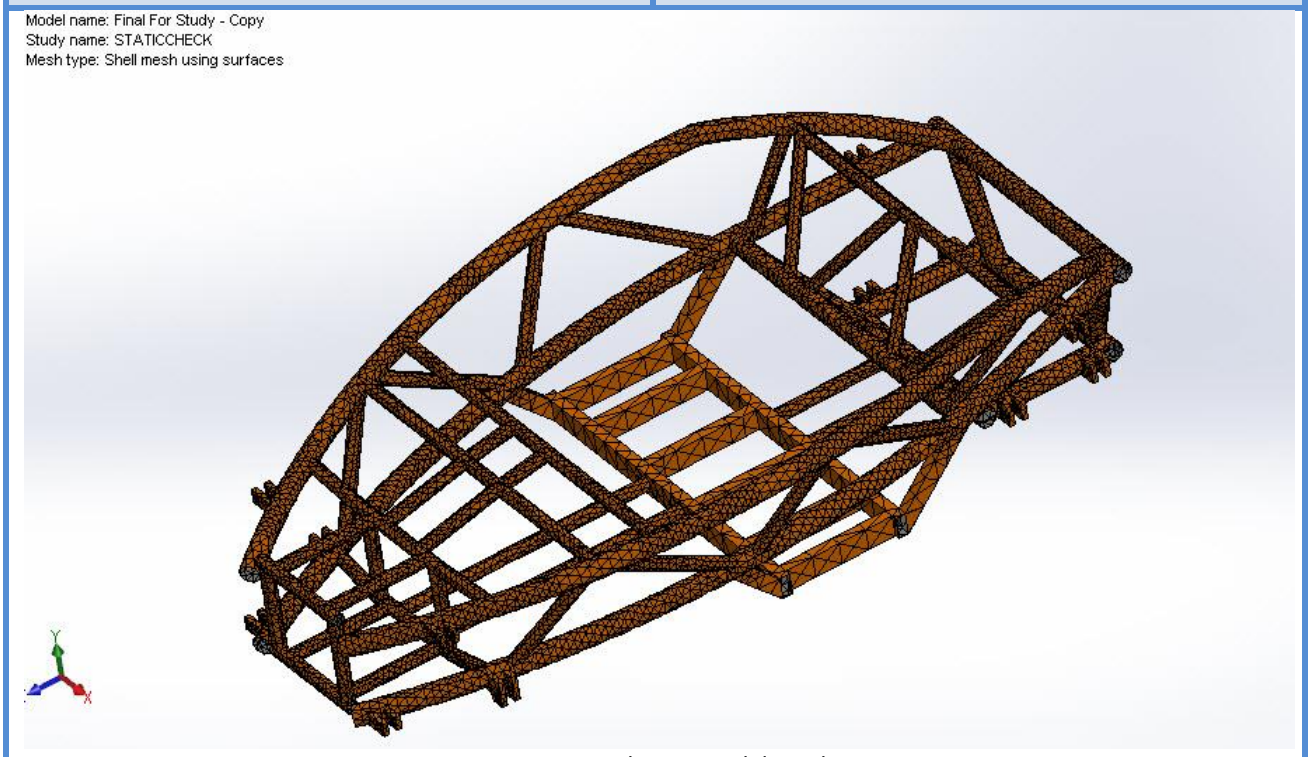


Fig 3-3 : Chassis Model Meshing

Simulation Results:

i. Stress

Highest Value : 120 MPa

Lowest Value 0 Mpa

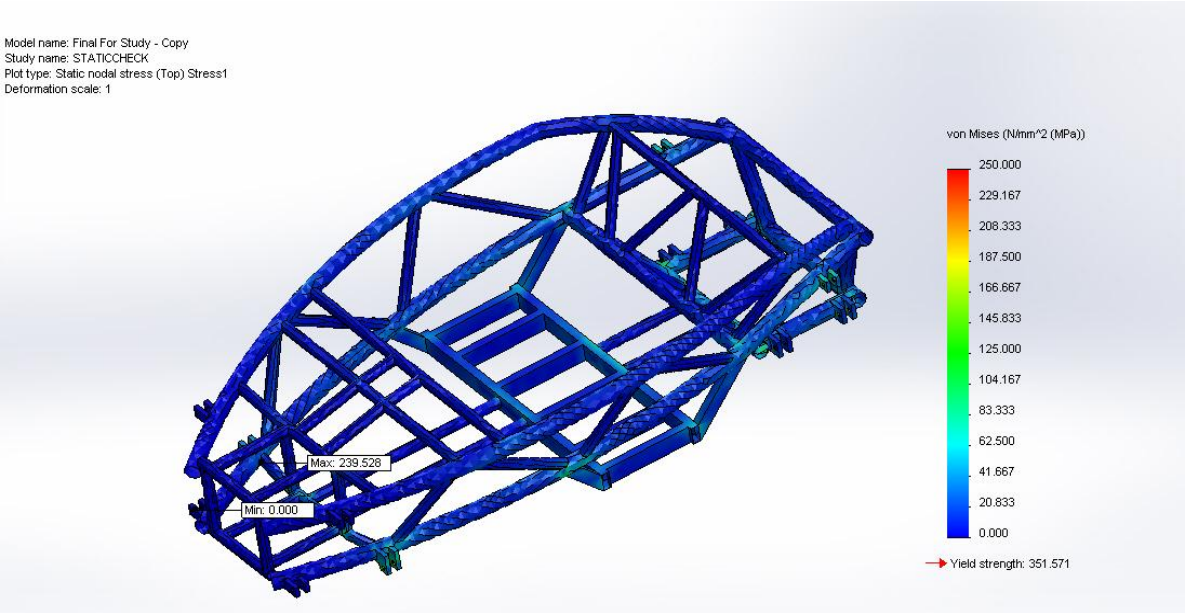


Fig 3-4 : Chassis Model Simulation Stress plot

ii. Deflection

Highest Value : 1.4 mm

Lowest Value : 0 mm

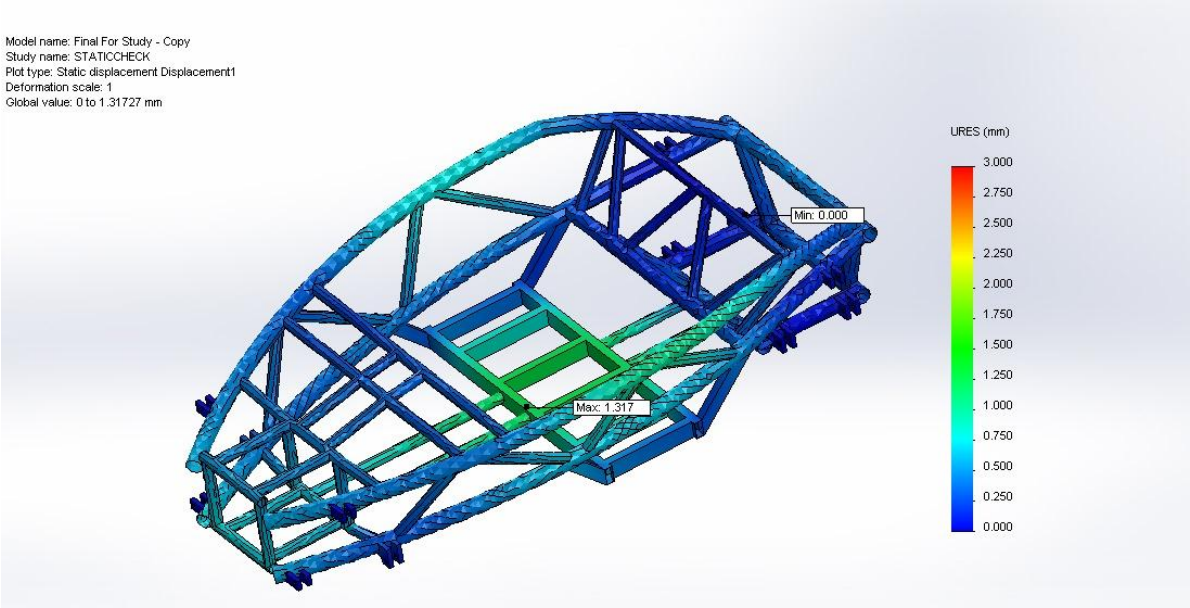


Fig 3-5 : Chassis Model Simulation deflection plot

iii. Factor of safety plot

Highest Value : 20+
Lowest Value : 1.5

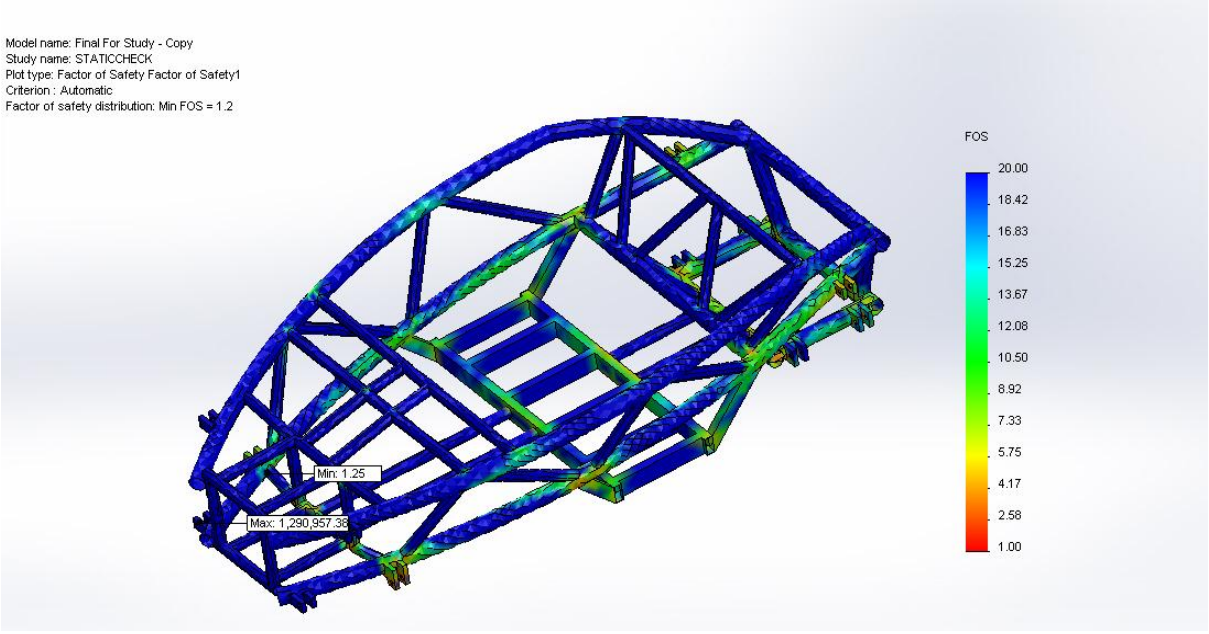


Fig 3-6 : Chassis Model Simulation Factor of Safety Plot

3.2.3 Engine:

Engine is a prime mover depends on conversion of a given energy into kinetic energy through rotational or translational motion. Vehicles commonly use internal combustion engines which convert the chemical energy in a fuel into rotational motion in the crank shaft.

3.2.3.1 Engine Selection:

To select an engine we need to determine the kinetic energy required by figuring the mass and inertia of rotating parts. Using inertia to determine the required engine power

Inertia: Is a property of an object that is its tendency to resist change in its velocity. The mass of an object is a measure of its inertia. If an object has a large mass it requires a large force to set it in motion or to change its velocity. If an object has a large mass it requires a large force to stop it if it is moving.

To determine the inertia we need to define the centre of gravity first.

The center of gravity is a geometric property of any object. The center of gravity is the average location of the weight of an object. We can completely describe the motion of any object through space in terms of the translation of the center of gravity of the object from one place to another, and the rotation of the object about its center of gravity if it is free to rotate. If the object is confined to rotate about some other point, like a hinge, we can still describe its motion

Assumptions:

1- Objects are Symmetrical.
2- Centre of Gravity of each object is at it is centre.
3- Dimensions and masses are approximated.
4- Dimensions of gravity centres are measured at the extreme front end.
5- Motion between tires and ground is rolling and co-efficient is constant
6- Total Vehicle Weight is 600 kg (including driver and passenger)
7- Various parts are well fastened and act as a single unit

Inertia Calculation:

Table 3-7 : Masses and centre of gravity locations

Components	Mass (kg)	Center of gravity (m)		
		xi	yi	zi
Chassis	120	1.70	0.00	0.45
Engine and Gearbox	40	2.70	0.00	0.40
Driver	120	1.70	-0.40	0.45
Front Right passenger	120	1.70	0.40	0.45
Tank	25	2.70	0.00	0.60
Battery	10	2.60	0.30	0.40
Driver seat	15	1.65	-0.40	0.49
Front rigt passenger seat	15	1.59	0.40	0.49
Misc.	50	1.50	0.00	0.40
TOTAL SUSPENDED MASS	**515**			
FL Wheel	20	0.50	-0.90	0.35
FR Wheel	20	0.50	0.90	0.35
RL Wheel	20	2.75	-0.90	0.35
RR Wheel	20	2.75	0.90	0.35
TOTAL NON SUSPENDED MASS	**80**			
TOTAL	**595**			

First, we need to find the C.G of the BUGGY using this equation:

$$Xg = \sum_{i=1}^{n} \frac{mi*xi}{M} \quad , Yg = \sum_{i=1}^{n} \frac{mi*yi}{M} \text{ and } \quad Zg = \sum_{i=1}^{n} \frac{mi*zi}{M} \ldots^{[1]}$$

Where:

m≡componenet mass

(X_i,Y_i,Z_i) the coordinates of the C.G of each component.

(X_g,Y_g,Z_g) the coordinates of the C.G of the buggy.

And M ≡ the total mass of the buggy.

From the table bellow we can find the C.G of the buggy.

Table 3-8 : product of masses and their distances

Mixi	Miyi	mizi
204.0	0	54
108.0	0	16
204.0	-48	54
204.0	48	54
67.5	0	15
26.0	3	4
24.8	-6	7.35
23.9	6	7.35
75.0	0	20
10	-18	7
10	18	7
55	-18	7
55	18	7
1067	**3**	**260**

$$\sum_{i=1}^{13} xi * mi = 1055\ kg.m \quad and \quad Xg = \frac{1055}{595} = 1.77m$$

$$\sum_{i=1}^{13} yi * mi = 3\ kg.m \quad and \quad Yg = \frac{3}{595} = 0.01m$$

$$\sum_{i=1}^{13} zi * mi = 260\ kg.m \quad and \quad Zg = \frac{260}{595} = 0.44m$$

Center of gravity coordinates		
xg (m)	yg (m)	zg (m)
1.77	0.01	0.43

Secondly; we have to calculate the distances between the vehicle C.G and the C.G of each component in the three directions as the following:

Table 3-9 : distance between parts CoG and buggy's CoG

Xi	Yi	Zi
xi-xg	yi-yg	zi-zg
-0.07	-0.01	0.02
0.93	-0.01	-0.03
-0.07	-0.41	0.02
-0.07	0.39	0.02
0.93	-0.01	0.17
0.83	0.29	-0.03
-0.12	-0.41	0.06
-0.18	0.39	0.06
-0.27	-0.01	-0.03
-1.27	-0.91	-0.08
-1.27	0.89	-0.08
0.98	-0.91	-0.08
0.98	0.89	-0.08

Now we can calculate the Inertia caused by each component using these equations:

- $I_{xxi}=m_i*X_i^2=m_i(Y_i^2+Z_i^2)$
- $I_{yyi}=m_i*Y_i^2=m_i(X_i^2+Z_i^2)$
- $I_{zzi}=m_i*Z_i^2=m_i(X_i^2+Y_i^2)$...[5]

The inertia I_x ,I_y ,I_z is the sum of each component's inertia

$$Ix = \sum \mathrm{mi}(\mathrm{Yi}^2 + \mathrm{Zi}^2)$$

$$Iy = \sum \mathrm{mi}(\mathrm{Xi2} + \mathrm{Zi2})$$

$$Iz = \sum \mathrm{mi}(\mathrm{Xi2} + \mathrm{Yi2})\ldots[5]$$

As listed in the following table:

Table 3-10 : Resultant Inertia

Ixx	Iyy	Izz	Ixz
mi(Yi²+Zi²)	mi(Xi²+Zi²)	mi(Xi²+Yi²)	miXiZi
0.051	40.39	40.37	0.942
0.037	99.9	99.9	-2.305
19.735	33.73	53.395	0.860
18.767	33.73	52.43	0.860
0.723	63.08	62.4	6.46
0.879	21.917	22.7	-0.540
2.515	4.256	6.67	0.426
2.394	3.356	5.65	0.377
0.046	7.29	7.22	-0.693
		24.07	
		23.7	
		69.52	
		69.16	
45	**308**	**537**	**6**

For the inertia around the Y and X axis, only the suspended mass has to be taken into account, but the inertia around Z axis is calculated with all the vehicle components (suspended and non-suspended mass)

The inertia of a DISK:

We can find the inertia of the wheels using this equation:

$$I = \frac{4m * r^2}{2}\ kg.m^2$$

Where:

M ≡ mass of a wheel.

R ≡ radius of wheel.

Then:

$$I = 4 * \frac{20 * 0.35^2}{2} = 4.9\ kg.m^2$$

Vehicle will accelerate from 0 to 80 km/h (22.2 m/s) in 16 seconds and engine will have to supply energy to match the kinetic energy, overcome drag energy and friction forces.

1- Kinetic Energy

Taking wheel radius of 0.35 meters yields 64 rad/s for v of 80km/h

$K.E = \frac{1}{2} I_{xx} {\omega_{xx}}^2$

I ≡Inertia in x direction

ω_{xx} ≡rotational velocity in x direction

$\frac{1}{2} * (45 + 4.9) * 64^2 = 10240\ J$

$Power = \frac{10240}{16} = 6400\ Watt$

2- Rolling Resistance

Tires are in contact with asphalt and the rolling Co-efficient is 0.02 (Assumption of an Old asphalt)

$F_f = \mu * m * g$

μ ≡coeffitient of friction

$F_f = 0.02 * 600 * 9.81 = 88.3\ N$

3- Air Drag

Frontal Area of vehicle is approximated to be equal to that of a small Sedan Automobile A = 1.5 m²

$F_d = C_d * \frac{1}{2} * \rho * A * v^2\ \ N$

c_d ≡drag coeffitiet

A ≡ Frontal Area of vehicle

$F_d = 0.3 * \frac{1}{2} * 1.15 * 1.5 * 22^2 = 125.1N$

For Air Drag and Rolling Ressistance

Power $= (F1 + F2) * v$

Power $= (\ 125.1 + 88.3) *\ 22 = 4695\ Watts$

Overall Power:

$$\frac{P_1 + P_2}{746} = \frac{4695 + 6400}{746} = 14.8\ hp$$

The Selected Engine is a single Cylinder 200cc engine generating 18 hp rated at 8700 rpm and a torque of 20 N.m at 7500 rpm. The Engine and gearbox are combined in a single unit

Fig 3-7: Selected Engine

3.2.3.2 Engine Mounting

Engine and Transmission are often considered as one single unit referred to as “PowerTrain“ which is mounted on the chassis of a vehicle using “ Mounts “

A bushing or rubber bushing is a type of vibration isolator. It provides an interface between two parts, damping the energy transmitted through the bushing.

These mounts are based on a rubber metal connection and range in complexity from conventional Rubber Metal Bushings and hydraulic dampers to sophisticated steel rubber mounts , they are used at the attachment points to serve as a counteract to the static load of the powertrain unit and to limit the maximum displacement caused by load shifts or high torques. The engine low frequency oscillations can be considerabily reduced by using specifying frequenct selective dampers. These mounts also help reduce and limit the powertrain excitations to the chassis which reduces structure borne noise this allows noise inside the passenger cabin to be controlled to increase ride comfort over the engine’s entire rotational speeds range

Thus we can summarize engine mounts fuctions to be

- Support
- Damping " Low Frequency, Large Excitation Amplitudes "
- Isolation " High Frequency, Small Excitation Amplitudes "

3.2.4 Drive Train:

The function of the DriveTrain is to couple the engine that produces the power to the driving wheels that consume this mechanical power. Through a gearbox which increases or decrease the speed of the engine as required.

The output shaft from the engine is connected to a differential gearbox which connected to the driving wheels.

3.2.4.1 Differential Gearbox:

In this project the engine and drive train will be mounted on the back of the buggy and power must be distributed to back wheels while allowing them to rotate at different speeds.

The differential is a device that splits the engine torque two ways, allowing each output to spin at a different speed. Car wheels spin at different speeds, especially when turning. You can see that each wheel travels a different distance through the turn, and that the inside wheels travel a shorter distance than the outside wheels. Since speed is equal to the distance traveled divided by the time it takes to go that distance, the wheels that travel a shorter distance travel at a lower speed. Also note that the front wheels travel a different distance than the rear wheels.

The differential has three jobs:

- To aim the engine power at the wheels
- To act as the final gear reduction in the vehicle, slowing the rotational speed of the transmission one final time before it hits the wheels
- To transmit the power to the wheels while allowing them to rotate at different speeds (This is the one that earned the differential its name.)

The Selected Differential Gearbox has a reduction ratio of 2.667

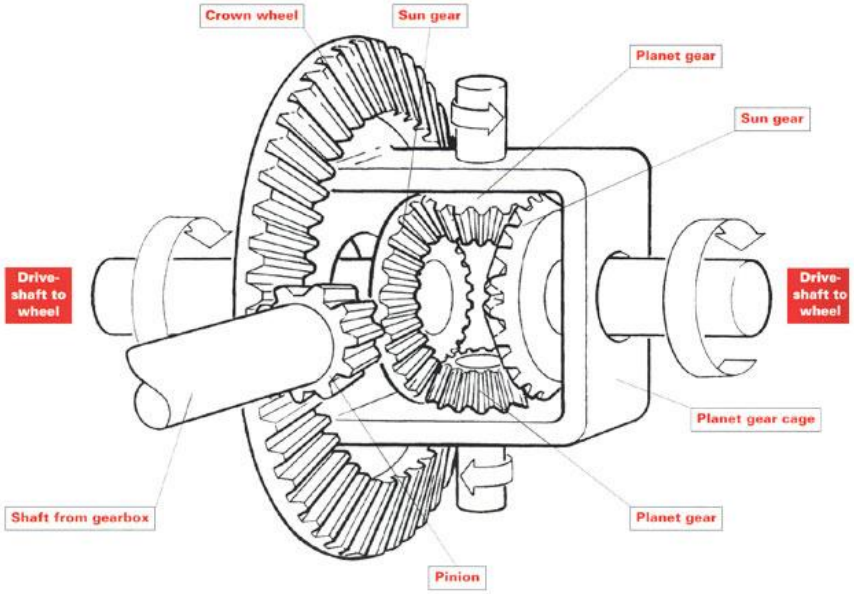


Fig 3-8: Differential Gearbox general layout

3.2.4.2 : DriveTrain Elements:

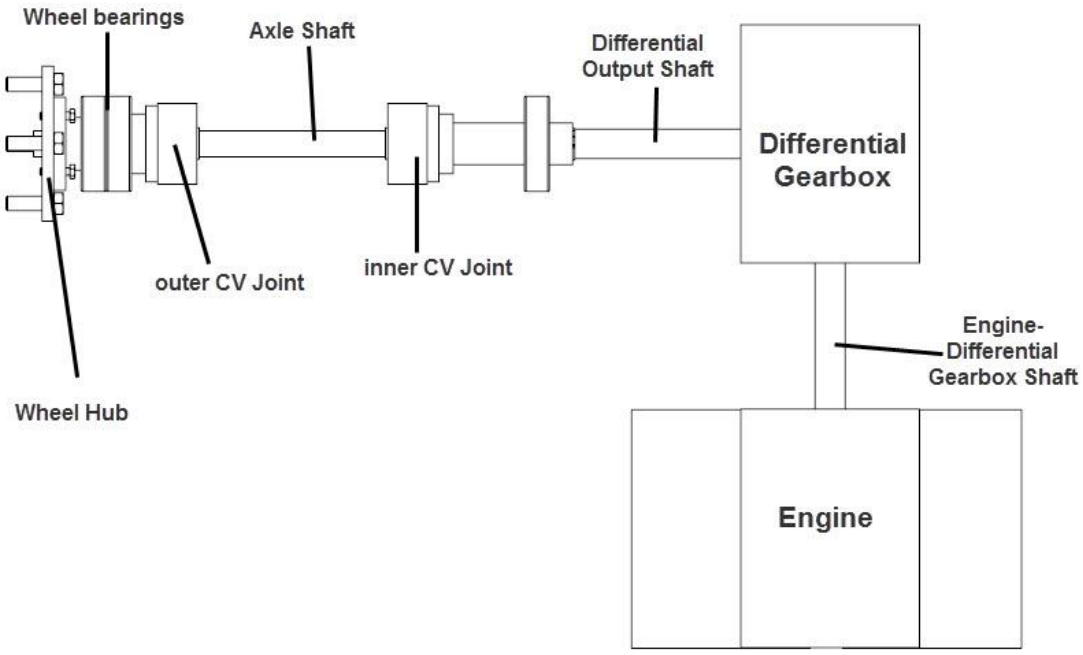


Fig 3-9: Torque Transmission Path to wheel

Engine transmits power through Gearbox to splined shaft. Spline Connections are used to permit relative axial motion the shaft and hub of the connected member. The splines are keys made integral with the shaft and usually consist of four, six, ten, or sixteen in number as standard for a given shaft it can be internal or external. splines can be thought of as a series of axial keyways with mating keys machined onto a shaft

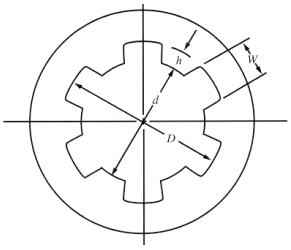


Fig 3-10: splined shaft

The torque capacity for the spline in shear is

$$T = \tau * A * r_m$$ [6]

Where
τ : Allowable Shear Stress Mpa
A : Spline tooth area mm^2
r_m : Mean radius of the spline mm
D : Outer diameter of spline
d : inner diameter of spline
w : width of spline
h : depth of spline
n : number of splines

Analysis:

$F = \tau * A$

$F = 2T/D_m$

$\frac{2T}{D_m} = \tau *$

$A = \frac{(D-d)}{2} * L * N$

$\tau = {}^{T}/_{A * r_m}$

$\tau = T / \left(\frac{D-d}{2}\right) * L * n * r_m$ [6]

3.2.4.2.1 Engine-Differential gearbox spline Torque transmitter:

Engine transmits power through a gearbox
Assuming the reduction ratio in the first gear which is the highest and is equal to 3.5
And knowing that the engine produces 20 N.m of Torque maximum
the Developed torque in the splines is 70 N.m

For the available engine and differential gearbox

$$\tau = \frac{T}{A * r_m}$$

$$A = \left(\frac{0.02 - 0.0165}{2}\right) * 0.07 * 6 = 0.000735\ m^2$$

$$r_m = \left(\frac{0.02 + 0.0165}{4}\right) = 0.009125\ m$$

$$\tau = \frac{70}{0.000735 * 0.009125} = 10.5\ MPa$$

Where:
$A \equiv$ spline contact area
$r_m \equiv$ spline mean radius

Engine-Differential Gearbox shaft Design Summary:

Shaft outer diameter:	30 mm
Splines Inner Diameter:	17 mm
Splines Outer Diameter:	21 mm
Number of splines:	6
Width :	4 mm
Length:	70 mm
Material:	C1020

3.2.4.2.2 Differential Gearbox Output shaft:

Consisting of 2 portions the first is in the differential and the other is connected to the inner CV Joint
For the portion connected to the differential gearbox it is splined with the following dimensions.
Outer diameter 25.5 mm
Inner diameter 20.5 mm

Spline length 25 mm
No. of splines 6

The torque is now multiplied after going through the differential gearbox
The new torque is

$$70\ N.m * 2.667 = 189\ N.m$$

$$\tau = \frac{T}{A * r_m}$$

$$A = \left(\frac{0.0255 - 0.0203}{2}\right) * 0.025 * 6 = 0.0003525\ m^2$$

$$r_m = \left(\frac{0.0255 + 0.0203}{4}\right) = 0.01145\ m$$

$$\tau = \frac{190}{0.0003525 * 0.01145} = 47.1\ MPa$$

Where:
$A \equiv$ spline normal area
$r_m \equiv$ spline mean radius

Using the same material of C1020 the design is safe.

3.2.4.2.3 Constant Velocity Joints:

(Also known as Homo-kinetic or CV joints) allow a drive shaft to transmit power through a variable angle, at constant rotational speed, without an appreciable increase in friction or play. They are mainly used in front wheel drive vehicles, and many modern rear wheel drive cars with independent rear suspension

CV Joint and the Axle shaft are available on market they are designed to transmit power in a commercial vehicle with 60 hp engine and hence they should withstand the 20 hp supplied by the buggy's engine.

3.2.4.2.4 Wheel Hub:

Wheel hub has the shape of a half flange coupling with the other half being the wheel power is transferred to

Fig 3-11: wheel hub

Wheel hubs come in different dimensions depending on application Some of them come with 4, 6, 8 or 10 Lug bolts
In designing the hub, standards taken will be the ones used by Hyundai Motors Company For a small sedan use of 4 bolts at a 100 mm bolt circle diameter with M12 bolts.

- *Designing the Hub:*

The hub is designed by considering it as a hollow shaft, transmitting the same torque (*T*) as that of a solid shaft.

$$T = \frac{\pi}{16} * \tau_s * (\frac{D^4 - d^4}{d})^{[7]}$$

Taking material C1020 with allowable stress of 60 MPa.

$$190 \times 10^3 = \frac{\pi}{16} * \tau_s * (\frac{35^4 - 27^4}{27})$$

$$\tau_s = 27\ MPa$$

- *Designing the Flange:*

The flange at the junction of the hub is under shear while transmitting the torque. Therefore, the Torque transmitted,

T = Circumference of hub × Thickness of flange × Shear stress of flange × Radius of hub

$$T = \pi D_f * \tau_s * t_f * r_f{}^{[7]}$$

$$190 = \pi * 0.13 * \tau * 0.010 * 0.065$$

$$\tau = 7.5\ MPa$$

Where:
D_f ≡*Dia. Of hub*
t_f ≡ Thickness of flange

- Safety of the Bolts:

The bolts are subjected to shear stress due to the torque transmitted. The number of bolts (*n*) depends upon the diameter of shaft and the pitch circle diameter of bolts ($D_{b.c}$)

$$T = A_{bolt} * \tau * R_{b.c} * N^{[7]}$$

$$T = \frac{\pi * {d_b}^2 * \tau * n * D_{b.c}}{4 * 2}$$

$$190 = \frac{\pi * 0.012^2 * \tau * 4 * 0.1}{8}$$

$$\tau = 8.4\ MPa$$

Where:
A_{bolt} ≡ Bolt cross-sectional area
$R_{b.c}$ ≡Bolt circle radius
n ≡ Number of bolts

-The Splined portion inside the Hub:

Inside the wheel hub the power is transmitted through a CV joint which has 25 spline at 26 mm outer diameter and 24 mm inner diameter

$$T = A * \tau * r$$

$$\tau = \frac{190 * 10^3}{\left(\frac{26 - 24}{2}\right) * 40 * 25 * 12.5}$$

$$\tau = 15.2\ MPa$$

Where:

A≡ spline area
r≡ spline radius

Design Summary:

Inner Diameter : 26 mm
Hub O.D : 35 mm
Hub Length : 75 mm
Flange Diameter : 130 mm
Flange thickness : 10 mm
Bolt Circle Diamter : 100 mm
Bolt Diameter : 12 mm (M12)
Material : C1020

3.2.4.2.5 Bearing Selection:

To select bearing we need to measure the load applied to it and Radially and Axially, Set a suitable Lifetime in Hours

First we measure Static load applied to bearing through the weight
As measured Before Center of Gravity location Co-ordinates are
(1.77, 0.005, 0.44) and The Total mass is roughly 600 kg
Resulting in load of 6890 N at the Center of Gravity.

Distance from front to Rear axis (Wheel-base) is 2.5 m
Distance between Center of Gravity and Front Axis 1.4 m
Distance between Center of Gravity and Rear Axis is 1.1 m

Resolving weight to find that

$$\sum F_y = 0$$

$$R_f + R_r = 6890\ N$$

$$\sum M_r = 0$$

$$R_f * 2.5 = 6890 * 1.1$$

$$R_f = 3032\ N\ And\ R_a = 3858\ N$$

Now to resolve the load on the left and right axles
Distance between wheels (Track-width) is 1.8 m
Distance between center of gravity and ends is 0.9 meters
Resolving yields:

Load at the right ends
Front Axis: 1516 N
Rear Axis: 1929 N
Load at the left ends
Front Axis: 1516 N
Rear Axis: 1929 N
This is the static load on each bearing.

Since vehicle is subjected to road bumps and holes with possible left and return to ground we simulate dynamic action loading and cornering by introducing dynamic load factor of 3

Thus Loads are:

Left and right Front Axis: 4548 N
Left And tight Rear Axis : 5787 N

Selected Life time is 5 Years

$$L_h = 5years * 365days * 3hours/day = 5475\ Hours$$

Assuming a person is to drive at 80 km/h average
80 km/h yields a angular velocity of 83 rad/s using a wheel of 0.28 m radius
This is translated to 792 rpm
For design purpose we assuming moderate speed of 800 rpm

$$L_d = 5475 * 800 * 60 = 262800000\ revs.$$ [8]

Equivalent load

$$P = VXR + YT$$ [8]

In this application Radial loads are much greater than axial loads we assume that axial load is 25% the value of radial loads
Axial Loads:
Front Axis: 1137 N
Rear Axis: 1446 N

For front axis

$$P = (1 * 0.56 * 4548) + (1.5 * 1137) = 4253\ N$$

$$C = 4253 * \left(\frac{262800000}{1*10^6}\right)^{\frac{1}{3}} = 27.3\ kN$$ [8]

For the Front Axis on both sides bearing number (6307)

For rear axis

$$P = (1 * 0.56 * 5787) + (1.5 * 1446) = 5410\ N$$

$$C = 5410 * \left(\frac{262800000}{1 * 10^6}\right)^{1/3} = 34.6\ kN$$

Where:
P≡ bearing equivalent load
C≡ bearing Dynamic load

Rear axis both sides will use bearing number (6307) is selected

Dimensions of bearing:

Outer Diameter	: 80 mm
Inner Diameter	: 35 mm
Width	: 21 mm
Shoulder required	: 1.5 mm

3.4.4.2.6 Brake Selection:

A brake is a device by means of which artificial frictional resistance is applied to a moving machine member, in order to retard or stop the motion of a machine. In the process of performing this function, the brake absorbs either kinetic energy of the moving member or potential energy given up by objects being lowered by hoists, elevators etc. The energy absorbed by brakes is dissipated in the form of heat. This heat is dissipated in the surrounding air (or water which is circulated through the passages in the brake drum) so that excessive heating of the brake lining does not take place.
Brakes are used to stop translational motion of suspended mass and Rotational motion of wheels and other non suspended mass

Energy Absorbed by Brakes:-

$$E = \sum_{i=1}^{n} \frac{1}{2} m_i v_i + \sum_{i=1}^{n} I_i \omega_i\ Joule$$

Standard Stopping distance for a modern sedan traveling at 80 km/h is 50 meters

To find the time of stopping we measure the required deceleration

$$v = v_0 + \alpha t \qquad -1$$
$$d = v_0 t + \frac{1}{2}\alpha t \qquad -2$$

Combining both equations yields

$$d = \frac{1}{2} * \left(\frac{v_0 - v}{t}\right) * t^2$$

Where:
v ≡final velocity
v_o ≡ initial velocity
α ≡ linear acceleration
t ≡breaking time
d ≡ breaking distance

For a vehicle traveling at 80 km/h and stopping in 50 meters

$$50 = \frac{1}{2} * \left(\frac{23 - 0}{x}\right) * x^2$$

$$x = 4.5\ second$$

Equation yields a stopping time of 4.5 seconds

I_w = polar moment of inertia of all wheels and axles ≈ 6 [kg m^2]

Now to measure the Braking power:

$$E = \left(\frac{1}{2} * 515 * 23^2\right) + \left(\frac{1}{2} * 6 * 85^2\right) = 157.9\ kJ$$

Where:

E ≡ Braking power

Using Brakes on all wheels (i.e 4 brakes)

$$\frac{160}{4} = 40\ kJ$$

This Energy is to be dissipated in nearly 5 seconds.

Thus the required breaking power by all brakes is

$$\frac{160}{5} = 32\ kW$$

Each brake has a braking power of 8 kW

Brakes available are Caliper Disk Brakes with the following characteristics:

- Brake Disk:

  1- Outer diameter : 220 mm
  2- Inner Diameter : 185 mm
  3- Thickness : 5 mm
  4- Material : Medium Carbon Steel

- Annular Brake Pads:

1- Width : 47 mm
2- Height : 31 mm
3- Material : Cast Iron

Coefficient of friction assumed 0.25 (as low as possible)

Braking Torque in each brake is 40 N.m

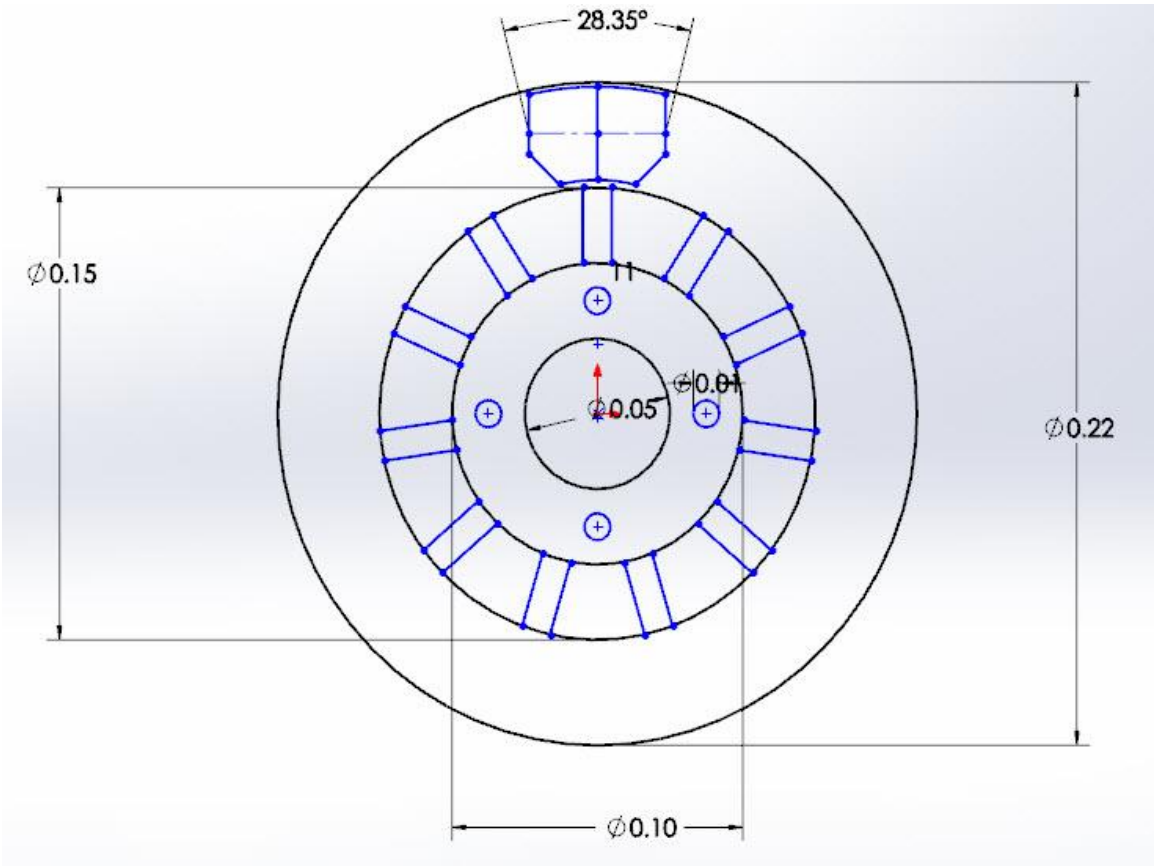


Fig 3-12: Brake configuration

$$T = \mu\, F\, r_e$$ [9]

$$r_e = \frac{r_o + r_i}{2} = \frac{110 + 92.5}{2} = 101.25\ mm$$ [9]

$$F = {}^{T}/_{\mu * r_e} = {}^{40}/_{0.25 * 0.10125} = 1580\ N$$

Where:

μ ≡Coefficient of friction

r_e ≡effictive radius

r_o ≡Outer radius of brake

r_i ≡Inner radius of brake

Maximum Contact Pressure

$$P_{max} = {}^{F}/_{\theta * r_i(r_o - r_i)}$$ [9]

$$P_{max} = \frac{1580}{28.35 * \frac{\pi}{180} * 0.15 * (0.22 - 0.15)} = 0.304\ MPa$$

Where:

p_{max} ≡ Maximum Contact Pressure

θ ≡Contact angle

This is force at the piston inside the Caliper

Brake Calipers transfer Force as a hydraulic system to increase the force by increasing diameters

$$\frac{F_1}{A_1} = \frac{F_2}{A_2}$$

area is increased 3 times so the force required by operator is

$$F_1 = \frac{F_2 A_1}{A_2} = \frac{1580}{3} = 526.6\ N$$

Further reduction in force required is done in the Pedals

3.2.5 Suspension System:

Suspension is the system of springs, shock absorbers and linkages that connects a vehicle to its wheels and allows relative motion between the two. Suspension systems serve a dual purpose

a- contributing to the vehicle's road holding /handling and braking for good active safety and driving pleasure by maximizing the friction between the tires and the road and Keep the tires in contact with the ground, because it is the friction between the tires and the road that affects a vehicle's ability to steer, brake and accelerate.

b- keeping vehicle occupants comfortable and a ride quality reasonably by allow the vehicle body to ride undisturbed while traveling over rough roads. well isolated from road noise, bumps, and vibrations ,etc.

3.2.5.1 Types of Suspension:

- Solid Axle: The solid, or beam, axle is used in the rear suspension of many front-engined, rear-wheel drive cars, and light commercial vehicles, and as the front suspension on many heavy commercial vehicles.
- Independent suspension : Independent suspension helps keep unsprung mass low. Also, if a wheel hits an irregularity, it won't upset the opposite wheel on the same axle. It allows wheel camber to be adjusted, or designed into the suspension geometry.
- Rear Independent Suspension : For independent suspension on the rear of a vehicle, many front-wheel drive cars use a McPherson strut at the rear. On rear-wheel drive vehicles, the suspension has to allow for the external drive shafts
- Rear Wheel Drive Independent Suspension : On rear-wheel drive vehicles with independent suspension, the final drive unit is fixed to the vehicle frame. Drive is transmitted to each wheel by external drive shafts.

Difference between Independent and Axle suspension:

1- One side is not affected when the other is actuated
2- Allow ability to Adjust Camber angles
3- Tires maintain in contact with ground at all angles
4- Offers more Grip
5- Less likely to roll over by use of Sway Bars

The Only disadvantage is that is has more moving parts

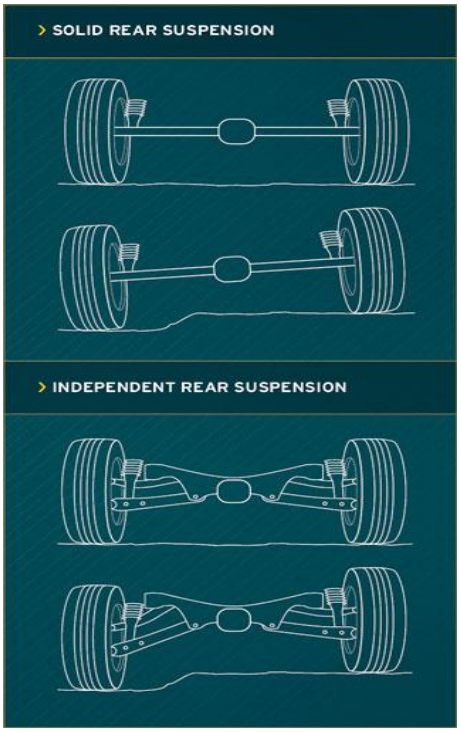


Fig 3-13: Independent suspension advantage over dependant

3.2.5.2 Suspension Design:

To increase ride comfort in vehicle independent suspension will be used.

3.2.5.2.1 Springs and Shock Absorbers Selection:

In a vehicle, shock absorbers reduce the effect of traveling over rough ground, leading to improved ride quality and vehicle handling. While shock absorbers serve the purpose of limiting excessive suspension movement, their intended sole purpose is to damp spring oscillations. Shock absorbers use valving of oil and gasses to absorb excess energy from the springs.

-For the front axis:

Mass(m)= $\frac{1520}{9.81} = 155\ kg$

Assuming that:
1/ the amplitude is to be reduced to one-fourth in one half cycle ($x_{1.5} = x_1/4$) and $x_2 = x_{1.5}/4 = x_1/16$

2/he damped period of vibration is given to be 3 s

$$\delta = \ln\left(\frac{x1}{x2}\right) = \ln(16) = 2.7726 = \frac{2\Pi\zeta}{\sqrt{1-\zeta^2}}$$[10]

Then $\zeta = 0.4037$

Damping time = $t_d = \frac{2\pi}{\omega d} = \frac{2\pi}{\omega n\sqrt{1-\zeta^2}} = 3s$[10]

Then $\omega n = 2.29\ \mathrm{rad/s}$

Where $\omega n \equiv the\ natural\ frequancy$

The critical damping constant can be obtained:

$$Cc = 2m\omega n = 2 * 155 * 2.29 = 710\ N.s/m$$[10]

And the damping constant is given by:

$$C = \zeta * Cc = 0.4037 * 710 = 306\ N.s/m$$[10]

And the stiffness by:

$$\mathrm{K} = m(\omega n)^2 = 815\ N/m$$[10]

-For the rear axis:

Mass (m) = $\frac{1930}{9.81} = 197\ kg$

Assuming that:
1/ the amplitude is to be reduced to one-fourth in one half cycle ($x_{1.5} = x_1/4$) and $x_2 = x_{1.5}/4 = x_1/16$

2/he damped period of vibration is given to be 3 s

$$\delta = \ln\left(\frac{x1}{x2}\right) = \ln(16) = 2.7726 = \frac{2\Pi\zeta}{\sqrt{1-\zeta^2}}$$

Then $\zeta = 0.4037$

Damping time = $t_d = \frac{2\pi}{\omega d} = \frac{2\pi}{\omega n\sqrt{1-\zeta^2}} = 3s$

Then $\omega n = 2.29$ rad/s

Where $\omega n \equiv the\ natural\ frequancy$

The critical damping constant can be obtained:

$$Cc = 2m\omega n = 2 * 197 * 2.29 = 903\ N.s/m$$

And the damping constant is given by:

$$C = \zeta * Cc = 0.4037 * 903 = 365\ N.s/m$$

And the stiffness by:

$$\mathrm{K} = m(\omega n)^2 = 1035\ N/m$$

Where:

δ ≡Logarithmic decrement

ζ ≡Damping coefficient

Cc ≡Critical damping constant

C ≡Damping constant

ωn ≡Natural frequancy

ωd ≡Frequency of damped vibration

K ≡Spring stiffness

3.2.5.2.2 Rear Suspension Simulation:

in the back the wheels will travel only up and down, there left and right motion is constrained by the use of multi location holder with pins as will be shown.

Parts:

i. Knuckle
ii. Upper Knuckle holder
iii. Lower Knuckle holder
iv. Wheel Hub
v. Wheel Bearing
vi. Bearing Cover
vii. Control Arms

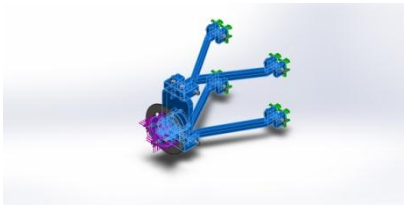
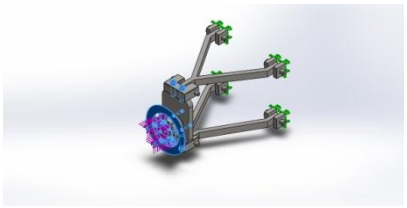
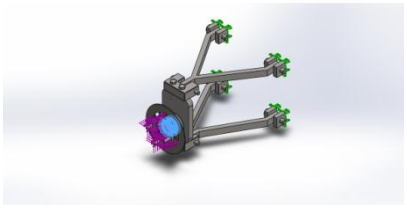
Simulation Properties:

Table 3-11: rear suspension simulation properties.

Study name	Rear Suspension
Analysis type	Static
Mesh type	Mixed Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SolidWorks Flow Simulation	Off
Solver type	FFEPlus
Soft Spring:	Off
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Use Adaptive Method:	Off

Material Properties:

Table 3-12 : Rear suspension simulation parts and materials

Model Reference	Properties	Components
	Name: **AISI 1020** Model type: **Linear Elastic Isotropic** Default failure criterion: **Unknown** Yield strength: **351 MPa** Tensile strength: **420 Mpa** Elastic modulus: **201 Gpa** Poisson's ratio: **0.29** Mass density: **7900 kg/m^3** Shear modulus: **77 Gpa** Thermal expansion coefficient: **1.5e-005 m/Kelvin**	Control Arms Suspension Mounts Wheel Hub Upper knuckle holder Lower Knuckle Holder
	Name: **AISI 1045 Steel, cold drawn** Model type: **Linear Elastic Isotropic** Default failure criterion: **Unknown** Yield strength: **530 Mpa** Tensile strength: **625 Mpa** Elastic modulus: **205 Gpa** Poisson's ratio: **0.29** Mass density: **7850 kg/m^3** Shear modulus: **80 Gpa** Thermal expansion coefficient: **1.2e-005 m/Kelvin**	Bolts Brake disks
	Name: **Alloy Steel** Model type: **Linear Elastic Isotropic** Default failure criterion: **Max von Mises Stress** Yield strength: **620 Mpa** Tensile strength: **723 Mpa** Elastic modulus: **201 Gpa** Poisson's ratio: **0.28** Mass density: **7700 kg/m^3** Shear modulus: **7.9e+010 N/m^2** Thermal expansion coefficient: **1.3e-005 m/Kelvin**	Bearings

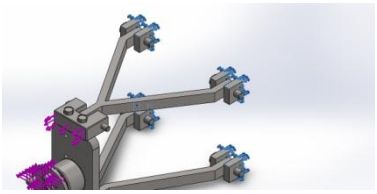
Units:

Table 3-13 : rear suspension simulation units.

Unit system:	SI (MKS)
Length/Displacement	mm
Temperature	Kelvin
Angular velocity	Rad/sec
Pressure/Stress	N/m^2

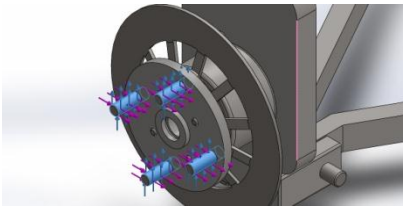
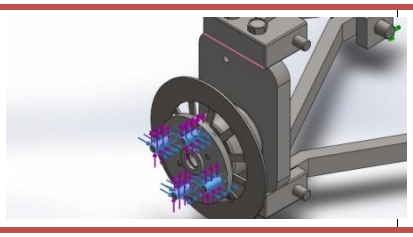
Fixtures:

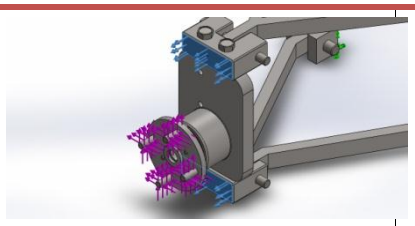
Table 3-14: Rear suspension simulation fixtures.

Fixture name	Fixture Image	Fixture Details
Suspension mounts		**Entities:** **Mounts inner Face** **Type:** **Fixed Geometry**

Loads:

Table 3-15: rear suspension simulation loads.

Load name	Load Image	Load Details
Bump Load		**Entities:** **Bolts** **Reference:** **X Axis** **Type:** **Apply force** **Values:** **(3000, 0, 0,) N** **Moments:** **(0, 0, 0) N·m**
Vehicle Weight		**Entities:** **Bolts** **Reference:** **Z Axis** **Type:** **Apply force** **Values:** **(0, 0, 3680) N** **Moments:** **(0, 0, 0) N·m**

Axial Loads		**Entities:** **Knuckle Holder** **Type:** **normal force** **Value:** **2000 N**

Mesh Information:

Table 3-16: rear suspension simulation mesh information.

Mesh type	Mixed Mesh
Mesher Used:	Curvature based mesh
Jacobian points	4 Points
Jacobian check for shell	On
Maximum element size	0.0154079 m
Minimum element size	0.00308159 m
Mesh Quality	Draft Quality Mesh

Mesh Information – Details:

Table 3-17: rear suspension simulation mesh details.

Total Nodes	68214
Total Elements	250677

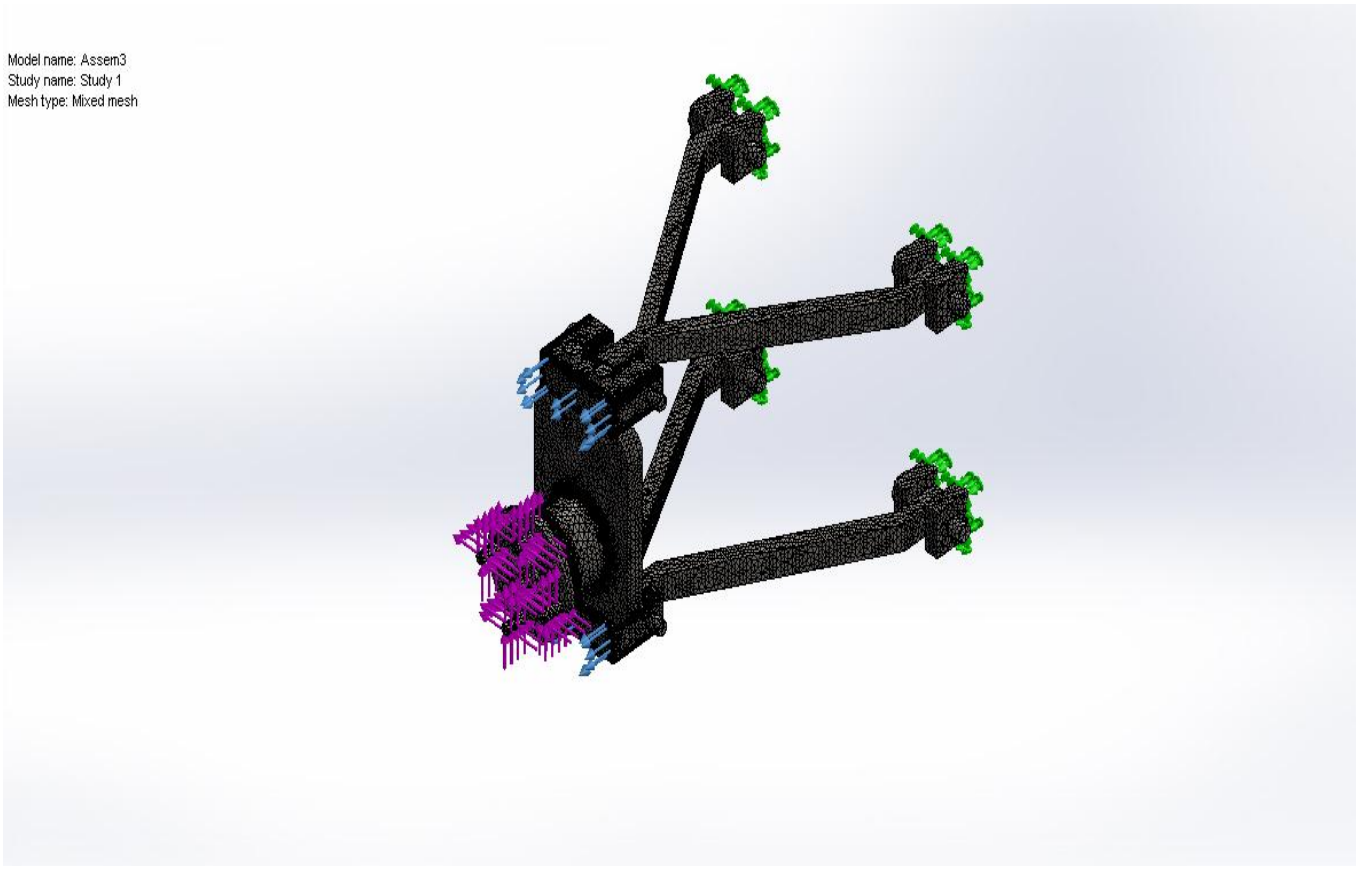


Fig. 3-14: rear suspension meshing.

Study Results:

i. Stress:

Highest Value : 120 MPa

Lowest Value : 0 MPa

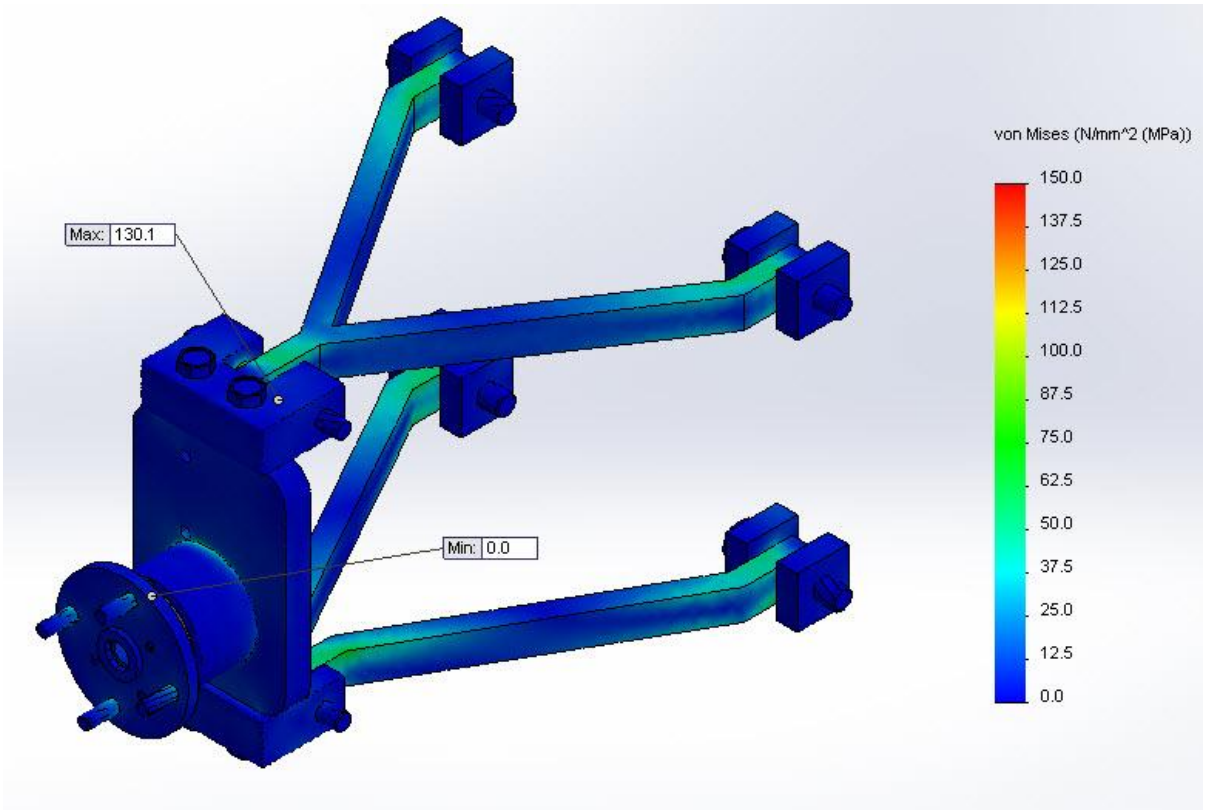


Fig 3-15 : Rear Suspension Simulation Stress Plot.

ii. Deflection:

Highest Value : 1 mm
Lowest Value : 0 mm

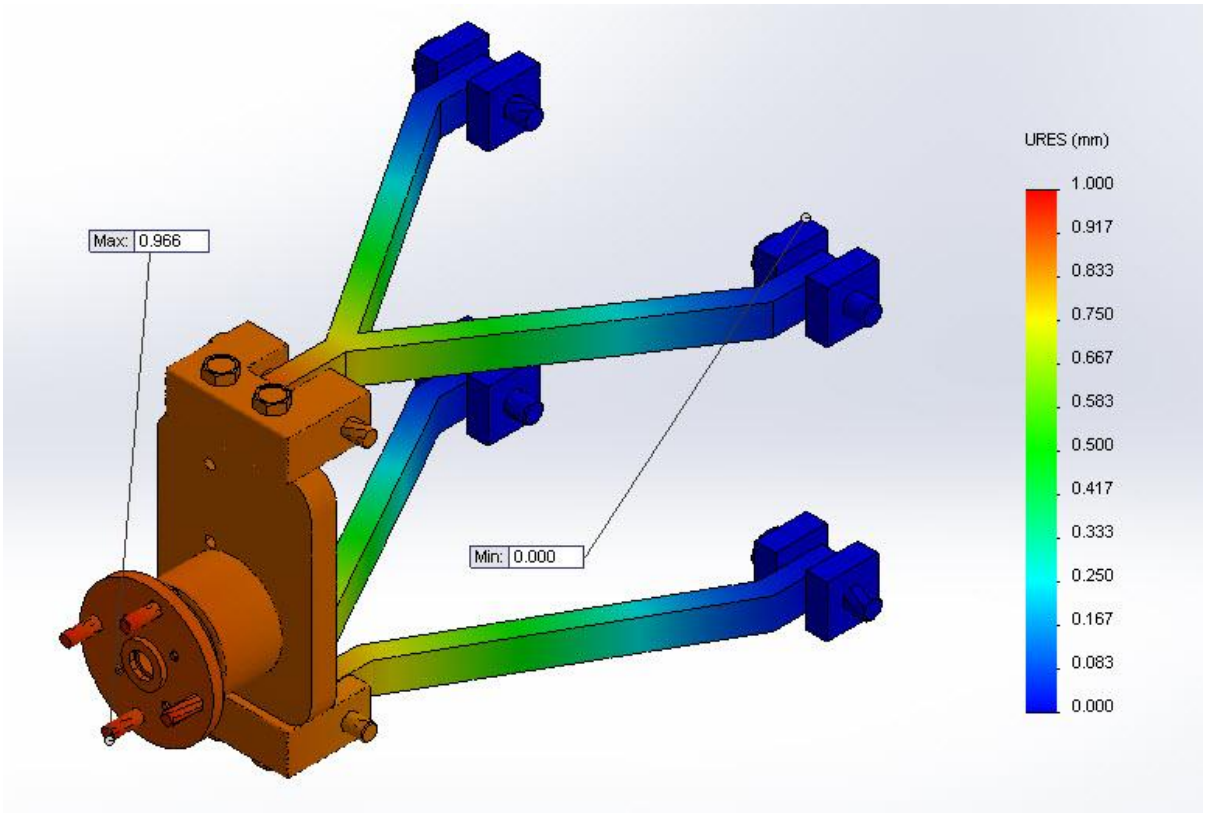


Fig 3-16 : Rear Suspension Simulation Deflection plot.

i. Factor of Safety :

Highest Value : 20+
Lowest Value : 4

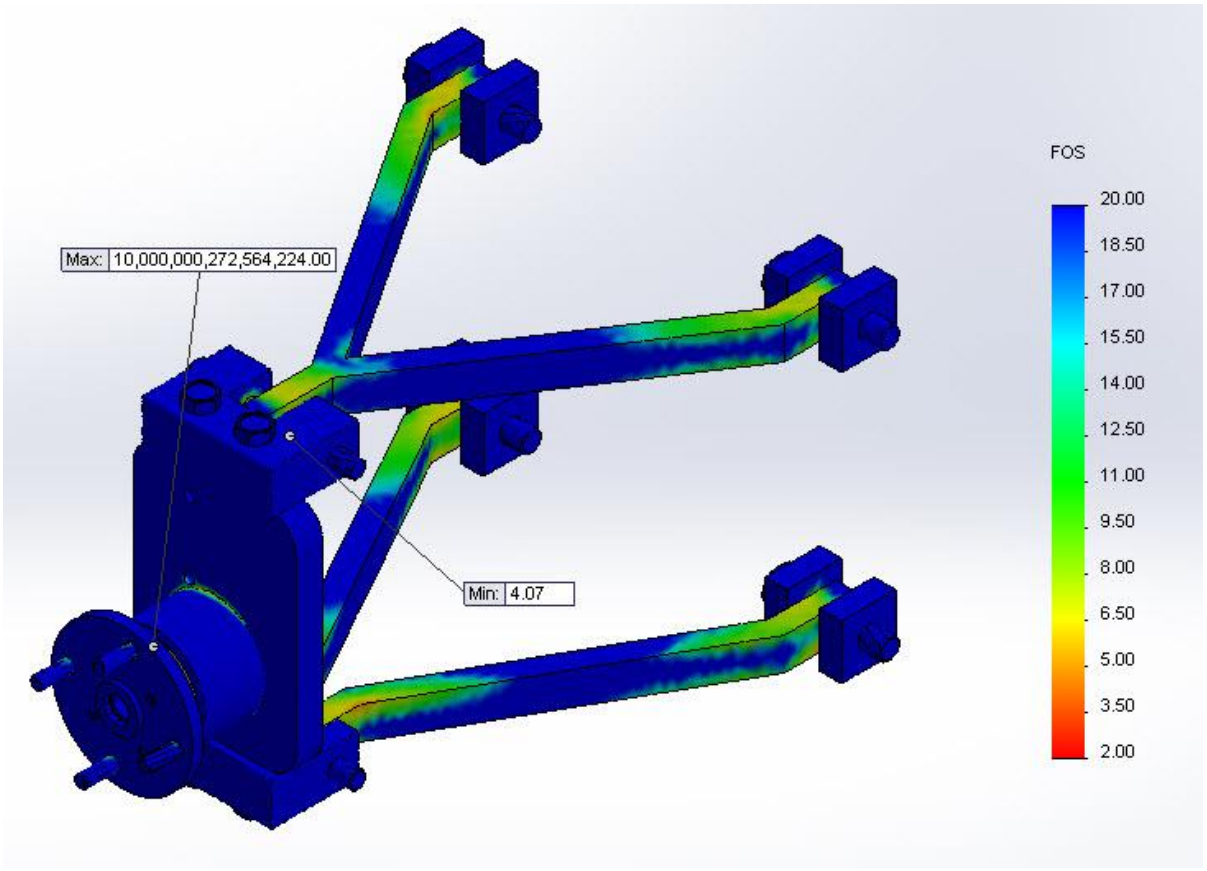


Fig 3-17 : Rear Suspension Simulation Factor of Safety Plot

3.2.5.2.3 Front Suspension Simulation:

Same Process is applied to the front suspension same parts but the only difference is that it will have a steering component and allow ability of rotation around the pin axis to accommodate driver's steering requirement.

Single pin at the top and the bottom of the knuckle will constrain one degree of freedom while the steering hand used to move the knuckle around the pin axis.

Parts:

i. Steering Knuckle
ii. Upper Knuckle Holder
iii. Lower Knuckle Holder
iv. Steering Hand
v. Wheel Bearings
vi. Wheel Hub
vii. Control Arms

Units:

Table 3-18: Front suspension simulation units.

Unit system:	SI (MKS)
Length/Displacement	Mm
Temperature	Kelvin
Angular velocity	Rad/sec
Pressure/Stress	N/m^2

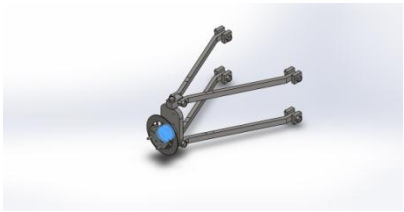
Study Properties:

Table 3-19: front suspension simulation properties.

Study name	Study 1
Analysis type	Static
Mesh type	Solid Mesh
Thermal Effect:	On
Thermal option	Include temperature loads
Zero strain temperature	298 Kelvin
Include fluid pressure effects from SolidWorks Flow Simulation	Off
Solver type	FFEPlus
Inplane Effect:	Off
Soft Spring:	Off
Inertial Relief:	Off
Incompatible bonding options	Automatic
Large displacement	Off
Compute free body forces	On
Friction	Off
Use Adaptive Method:	Off

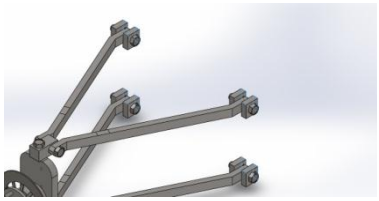
Material Properties:

Table 3-20: front suspension material properties.

Model Reference	Properties		Components
	Name: **Model type:** **Default failure criterion:** **Yield strength:** **Tensile strength:** **Elastic modulus:** **Poisson's ratio:** **Mass density:** **Shear modulus:** **Thermal expansion coefficient:**	AISI 1045 Steel, cold drawn Linear Elastic Isotropic Max von Mises Stress 5.3e+008 N/m^2 6.25e+008 N/m^2 2.05e+011 N/m^2 0.29 7850 kg/m^3 8e+010 N/m^2 1.15e-005 /Kelvin	Bolts
	Name: **Model type:** **Default failure criterion:** **Yield strength:** **Tensile strength:** **Elastic modulus:** **Poisson's ratio:** **Mass density:** **Shear modulus:** **Thermal expansion coefficient:**	AISI 1020 Linear Elastic Isotropic Max von Mises Stress 3.51571e+008 N/m^2 4.20507e+008 N/m^2 2e+011 N/m^2 0.29 7900 kg/m^3 7.7e+010 N/m^2 1.5e-005 /Kelvin	Control Arms Knuckle Knuckle Holder Suspension Mounts
	**Name:** **Model type:** **Default failure criterion:** **Yield strength:** **Tensile strength:** **Elastic modulus:** **Poisson's ratio:** **Mass density:** **Shear modulus:** **Thermal expansion coefficient:**	Alloy Steel Linear Elastic Isotropic Max von Mises Stress 6.20422e+008 N/m^2 7.23826e+008 N/m^2 2.1e+011 N/m^2 0.28 7700 kg/m^3 7.9e+010 N/m^2 1.3e-005 /Kelvin	Wheel Bearings

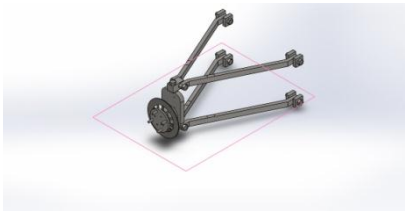
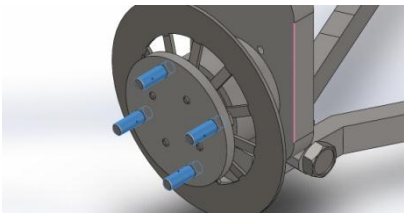
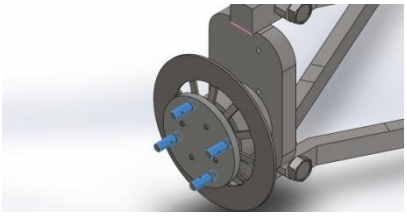
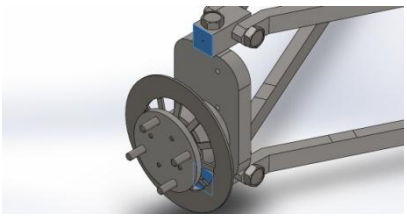
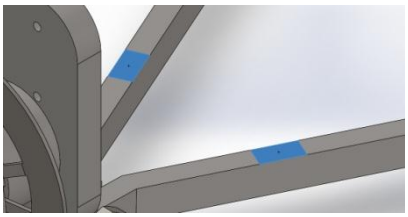
Fixtures:

Table 3-21: Front suspension simulation fixtures.

Fixture name	Fixture Image	Fixture Details
Suspension Mount		**Entities:** Inner Suspension Mount Face **Type:** Fixed Geometry

Loads

Table 3-22: front suspension simulation loads.

Load name	Load Image	Load Details
Gravity-1		**Reference:** Z Axis **Values:** 0 , 0, -9.81 **Units:** SI
Bump Load		**Entities:** Lug Bolts **Reference:** X Axis **Type:** Apply force **Values:** 0, 0, -4000 N
Vehicle Weight		**Entities:** Lug Bolts **Reference:** Z Axis **Type:** Apply force **Values:** 0 ,0, -3000 N
Axial Loads		**Entities:** 2 face(s) **Type:** normal force **Value:** 3000 N
Strut reaction		**Entities:** Strut Location **Type:** normal force **Value:** 3000 N

Mesh Information:

Table 3-23: front suspension simulation mesh information.

Mesh type	Solid Mesh
Mesher Used:	Curvature based mesh
Jacobian points	4 Points
Maximum element size	0.0174689 m
Minimum element size	0.00349378 m
Mesh Quality	Draft Quality Mesh
Remesh failed parts with incompatible mesh	On

Mesh Information – Details:

Table 3-24: front suspension simulation mesh details.

Total Nodes	16748
Total Elements	56155
Maximum Aspect Ratio	32.596
% of elements with Aspect Ratio < 3	84.6
% of elements with Aspect Ratio > 10	0.0819

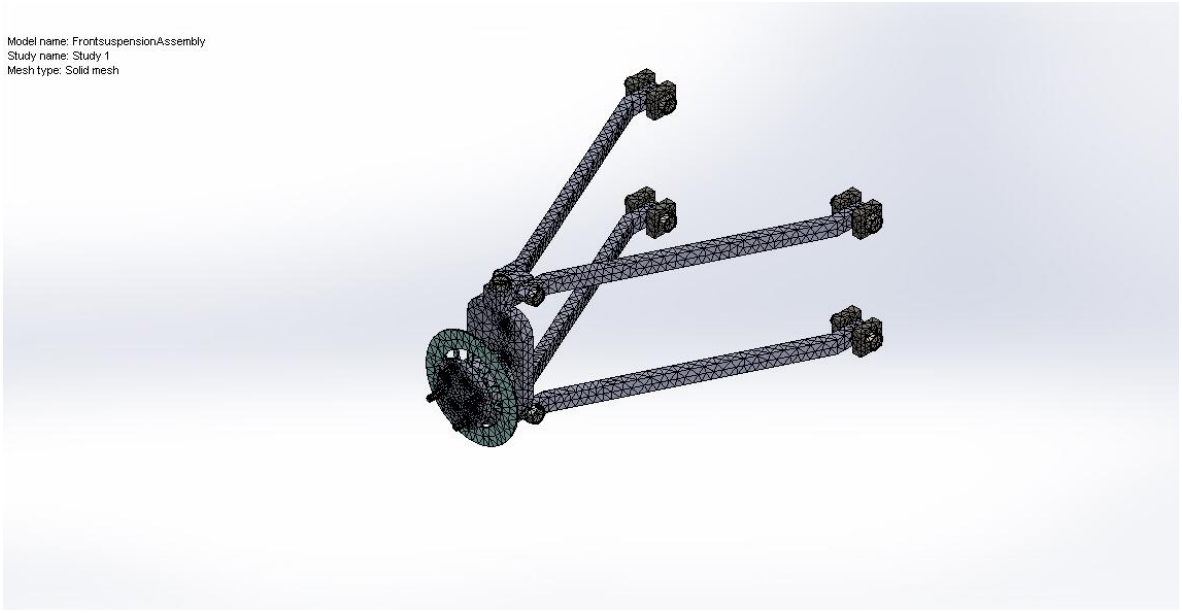


Fig 3-18: front suspension simulation meshing.

Study Results:

i. *Stress:*

Highest Value : 76 MPa

Lowest Value : 0 MPa

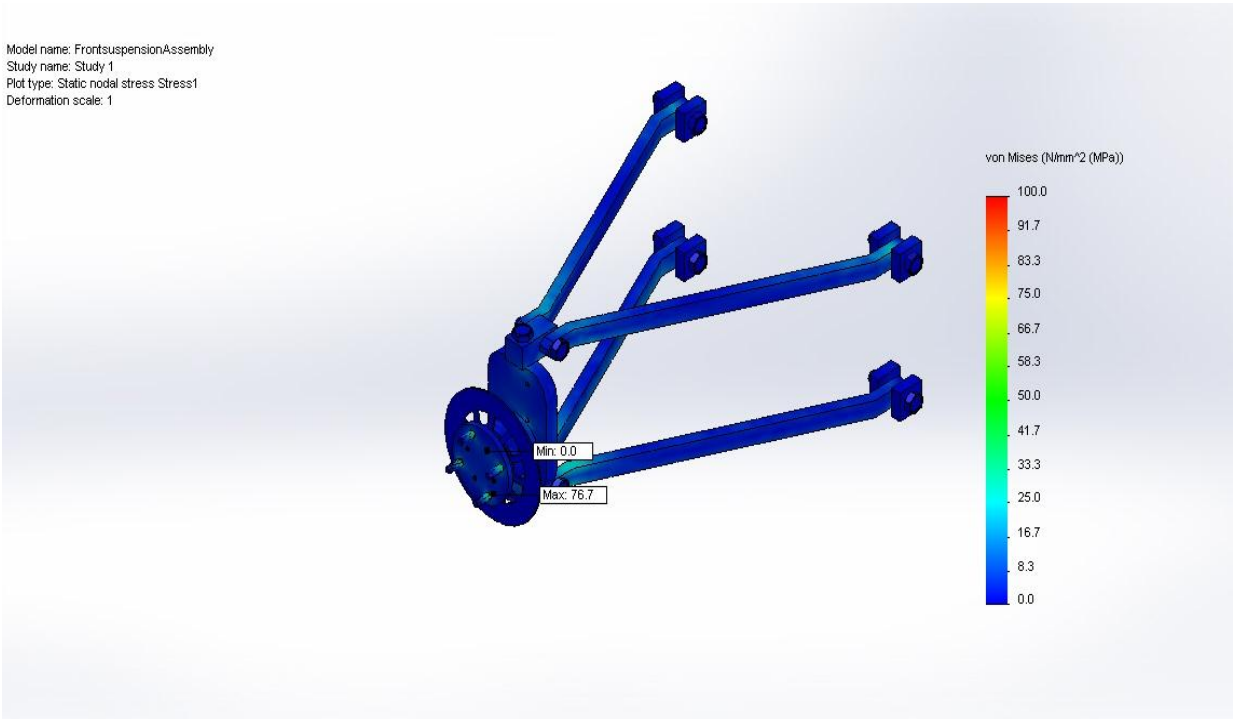


Fig. 3-19: front suspension simulation stress plot.

ii. *Deflection:*

Highest Value : 0.6 mm
Lowest Value : 0 mm

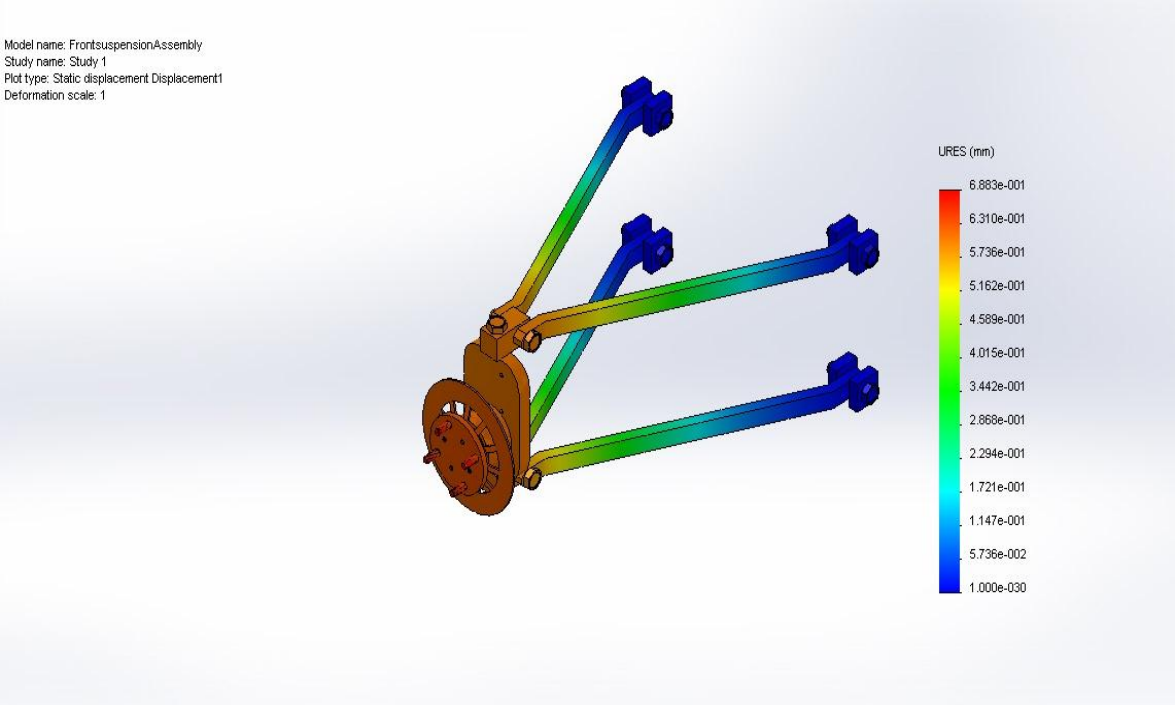


Fig. 3-20: front suspension simulation Deflection plot.

iii. *Strain :*

Highest Value: 0.000454365

Lowest Value: 0

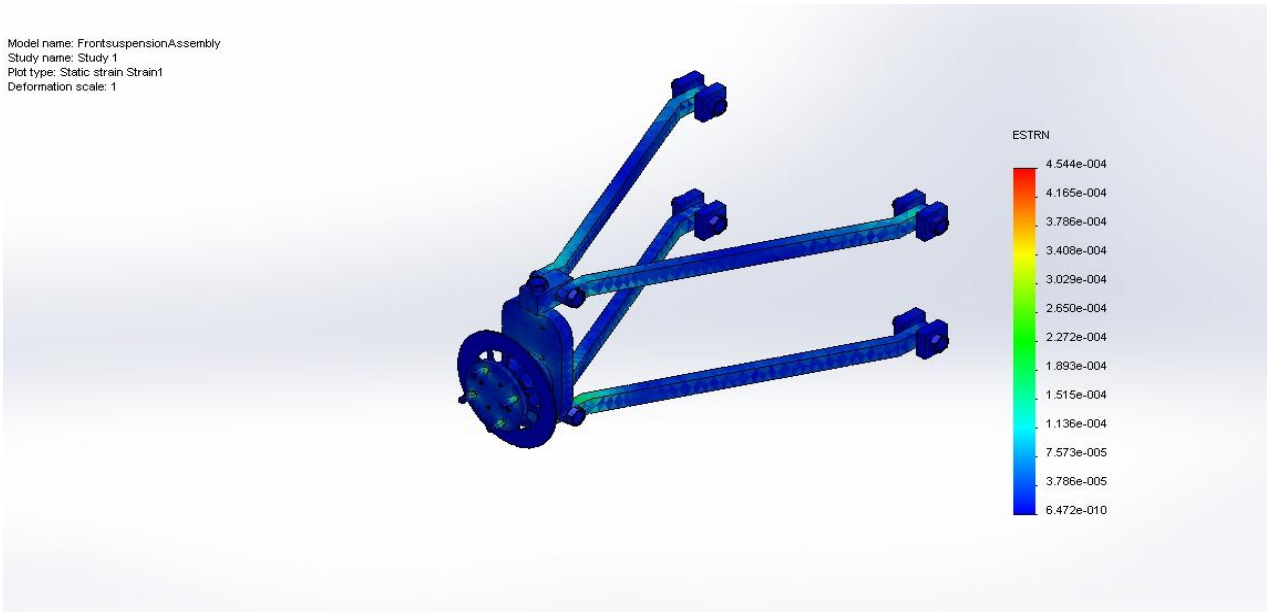


Fig. 3-21: front suspension simulation Strain plot.

3.2.5.3 Steering Design and Selection :

The most conventional steering arrangement is to turn the front wheels using a hand–operated steering wheel which is positioned in front of the driver, via the steering column, which may contain universal joints to allow it to deviate somewhat from a straight line.

3.3.5.3.1 Steering Concerns:

When talking about steering we are concerned with the Steering angles of each wheel and the turning circle which is the circle the vehicle will rotate about when the steering wheel is turned.
Each wheel is rotating at a different radius thus the wheels need to be turned at different angles and operated at different speeds.
Since the wheels are not driven and allowed to rotate freely, inertia forces will automatically adjust the output speed in the wheels, thus we are concerned with the steering angles since each wheel turns at a different radius.

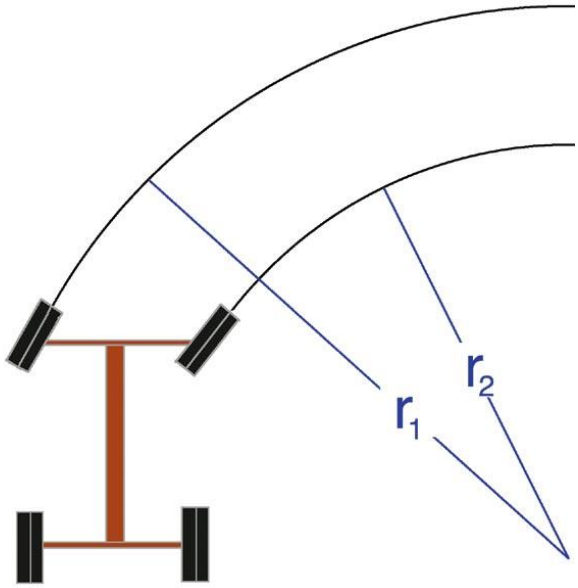


Fig. 3-22: Turning radius difference.

3.2.5.3.2 Steering Solutions:

The problem of the required angular difference is solved by introducing the Ackermann steering geometry.

Ackermann steering geometry is a geometric arrangement of linkages in the steering of a car or other vehicle designed to solve the problem of wheels on the inside and outside of a turn needing to trace out circles of different radius.
The intention of Ackermann geometry is to avoid the need for tyres to slip sideways when following the path around a curve.
To achieve this Geometry the Steering A simple approximation to perfect Ackermann steering geometry may be generated by moving the steering pivot points inward so as to lie on a line drawn between the steering kingpins and the centre of the rear axle.[2] The steering pivot points are joined by a rigid bar called the tie rod which can also be part of the steering mechanism, in the form of a rack and pinion for instance. With perfect Ackermann, at any angle of steering, the centre point of all of the circles traced by all wheels will lie at a common point.

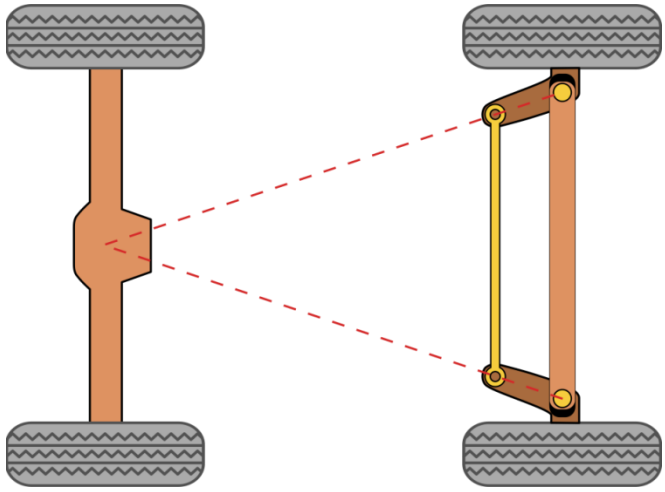


Fig. 3-23: Ackermann true geometry.

We will be using a Rack and Pinion Steering suited for the Hyundai Elantra.

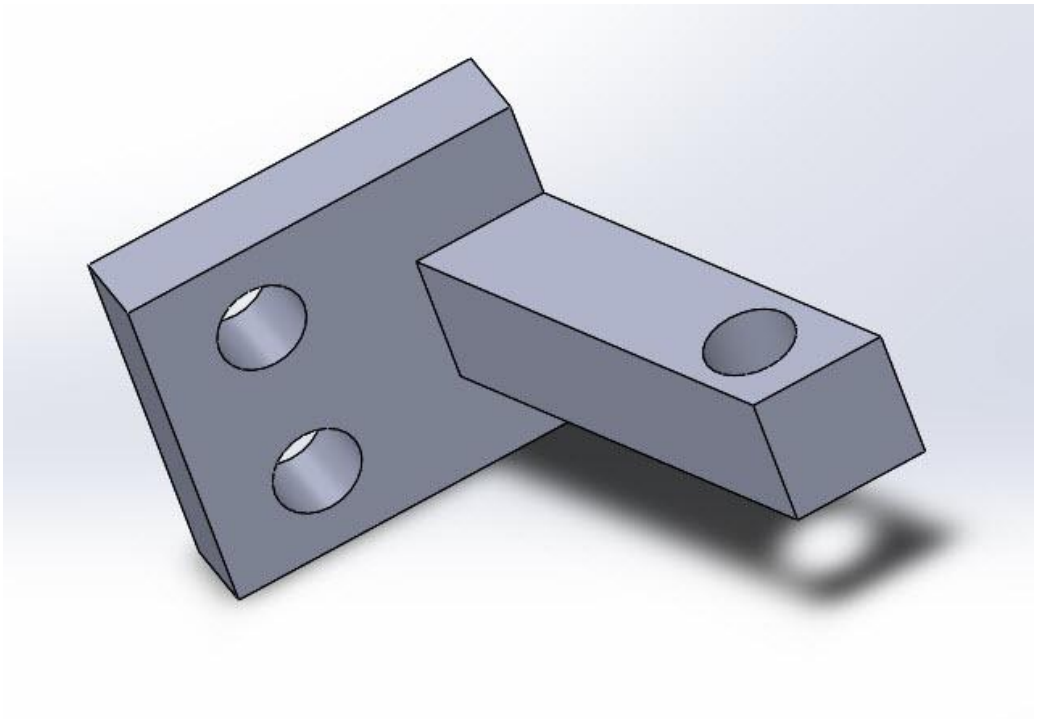


Fig. 3-24: Steering hand

Steering hand is designed with angular deviation to meet the Ackermann steering.

CHAPTER 4:

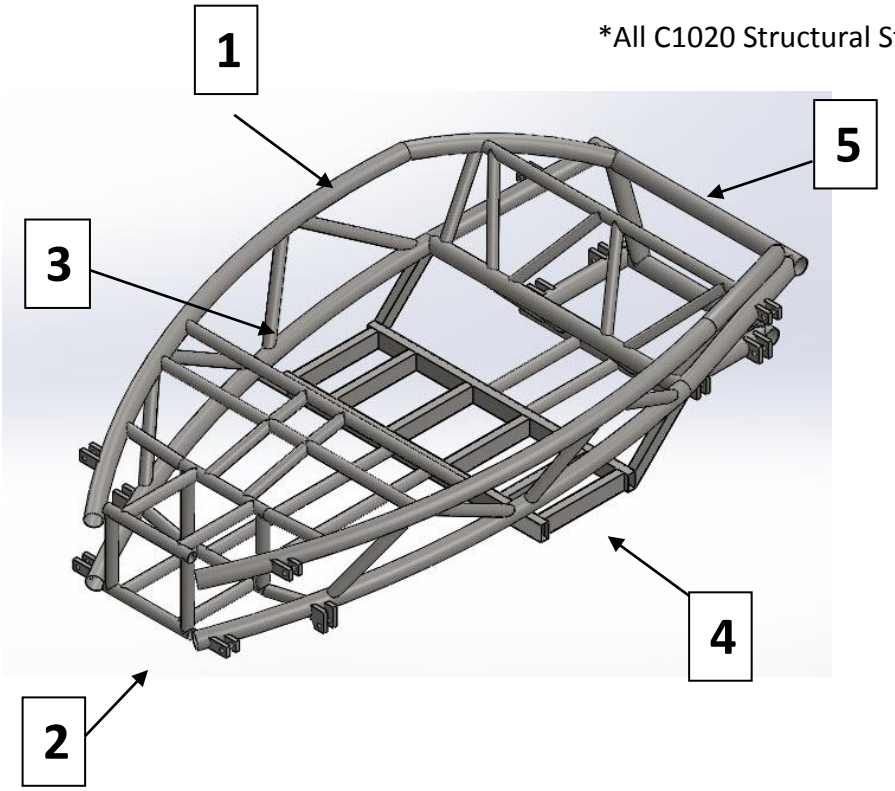
MANUFACTURING:

4.1 Chassis:

4.1.1 Elements:

1. Main members: 49mm steel pipe (2mm thick)
2. *Crumble Zone* members: 49mm steel Pipe (1 mm thick)
3. Supportmembers: 25mm steel Pipe (1 mm thick)
4. Seating support: 60x30mm rectangular cross section tube (1 mm thick)
5. Suspension mounts: Steel Plate (1 mm thick)

*All C1020 Structural Steel



4.1.2 Manufacturing Processes:

1. Cutting
2. Arc Welding
3. Bending
4. Drilling
5. Blacksmithing

4.1.3 Method:

1. Main, support and crumble zone pipes were measured according to design dimensions and cut into required lengths by means of a (cutting) grinding wheel.
2. Main (side) pipe members were bent into required curvature using a roll bending machine
3. Members were fixed together according to required end shape design by means of arc welding
4. Suspension mounts were cut into shape using a manual shear cutter, and required openings were created using heating and blacksmithing. Mounts were then welded into place

4.2 Drive Train:

4.2.1 Manufactured Elements:

1. Engine differential torque transmitter: (1 part)
   C1020 Steel Shaft
   D_o = 30mm
   D_i = 16mm
   6 slots of uniform size (d_o = 20mm, d_i = 18mm)

2. Differential gearbox output shaft: (2 parts)
   C1020 Steel shaft
   $D_{o,1}$ = 30mm, $D_{o,2}$ = 25.5mm
   6 slots of uniform size (d_o = 26mm, d_i = 20mm)
3. Output shaft - CV joint torque transmitter: (2 parts)
   C1020 Steel shaft
   D_o = 40mm, $D_{o,2}$ = 25mm
   25 slots of uniform size (d_o = 26mm, d_i = 24mm)

4.2.2 Purchased Elements:

1. Engine: Single cylinder 200 CC (gearbox included)
2. Differential gearbox: Reduction ratio = 2.667
3. 4 CV Joints: 49 degrees
4. 8 Wheel bearings: Bearing no. 6307
5. 4 Wheels: 16 inch (complete)
6. 2 Seats
7. 4 complete brakes: Disc type
8. Bolts: M8, M10, M18

4.2.3 Manufacturing Processes:

1. Lathing
2. Splining
3. Drilling
4. Threading

4.2.4 Method:

1. Engine and differential were mounted and positioned suitable to design requirements.
2. 30mm solid shaft was made hollow with D_i of 20mm, by means of a lathing machine, slotting was done according to required geometry by means of a splining machine to create the engine differential torque transmitter, which was then used to connect the engine and the differential.
3. 30mm solid shaft was reduced to 25.5mm by lathing, splined on both ends to appropriate dimensions and cut to suitable length before being used to connect the differential to the CV joint. (i.e differential gearbox output shaft)
4. Output-CV Joint torque transmitter was made by the reduction and splining of a 40 mm solid shaft to 25mm with 25 slots(d_o = 26mm, d_i = 24mm), to connect the output shaft and the CV Joint.
5. Remaining purchased components were then joined and assembled according to conceptual design.

4.3 Suspension:

4.3.1 Manufactured Elements:

1. Knuckle: 30mm thick steel plate, 110mm solid steel shaft (both C1020)
2. Bearing Cover: 110mm solid shaft
3. Holders: 30mm thick steel plate (C1020)
4. Control Arm: 4 layers of 5mm thick steel plate (C1020)

4.3.2 Purchased Elements:

1. Springs
2. Shock absorbers

4.3.3 Manufacturing Processes:

1. Lathing
2. Drilling
3. Arc Welding
4. Acetylene torch cutting
5. Facing
6. Threading

4.3.4 Method:

1. 30mm thick plate was cut into 4 knuckle plates and 8 holders (both rectangular with different dimensions) by means of an acetylene torch. Circular openings of appropriate diameter were made also using by acetylene. 110 mm shaft was then fit into knuckle plate opening and welded into place by means of arc welding.

2. Facing was then done to repair irregular surfaces caused by cutting. Lathing, drilling and threading onto the knuckles as well as drilling of the holders was then completed, according to design specifications.

3. 110 mm shaft was machined into bearing cover by means of lathing and drilling with respect to design specifications.

4. 4 layers of 5mm thick plates were welded according to desired length to achieve required rectangular cross section. Completed piece was then bent and drilled to attain final design of control arm.

5. Control arms were then positioned on suspension mounts on car chassis and kept in place by means of pins. Holders and knuckles were then joined by bolts and attached to control arms. Bearings were pressed into hollow opening in knuckle and subsequently mounted by bearing cover which was bolted onto the knuckle.

6. Wheel hub was pressed inside bearing and closed off axially by means of a crown nut and pin.

7. Shock absorbers were mounted onto control arms and chassis on either side.

4.4 Steering:

4.4.1 Manufactured Elements:

1. Steering Column: 20mm Solid steel shaft (C1020)
2. Steering Hand: 1mm thick steel plate (C1020)

4.4.2 Purchased Elements:

1. Bearings: Bearing no. 6105
2. Steering wheel + Air-bag
3. Steering shaft: (Rack and Pinion enclosed in housing)

4.4.3 Manufacturing Processes:

1. Cutting
2. Drilling
3. Lathing

4.4.4 Method:

1. Steering shaft was positioned and mounted on chassis appropriate to design requirements.
2. Steering wheel was placed adequately in front of driver seat.
3. Steering wheel and column were joined by drilling and insertion of a pin in drilled opening.
4. Other end of steering column was mounted onto the steering shaft in order to transmit motion.
5. Bearing was placed and position appropriately in order to support and govern the steering column.
6. Steering hand was formed by cutting two steel plate into required shape and dimensions, welding them at a 110 degree angle, and was then connected to front knuckle.

4.5 Assembly:

4.5.1 Drive Train:

1- Chassis is placed at an adequate height.
2- Engine is first mounted on it is specified location
3- The 1st torque transmitter is installed in the engine shaft
4- The differential gearbox is mounted behind the engine in a way their shafts are parallel and then connected with the 1st torque transmitter
5- The differential gearbox shafts are now meshed in the differential output.
6- On the other side of the output shaft the 2nd torque transmitter is connected
7- A pillow block bearing is used to act as a motion governor
8- On the other side of the torque transmitter the inner CV joint is mounted
9- The drive axle is now connected to the inner CV joint
10- Outer CV joint is now connected to the drive axle
11- The outer CV joint is meshed with the inner slots of the wheel hub.

4.5.2 Suspension:

1- Wheel bearings are pressed inside the knuckle until reaching the shoulder
2- Bearing cover is now placed and bolted to the knuckle to stop axial motion of bearing.
3- Lug bolts are bolted on the wheel hub as well as the brake disk
4- A spacer is now used to separate the bearings outer race from wheel hub (i.e. deny contact between rotating parts and stator parts)
5- Wheel hub is now pressed in the inside hole of the bearing
6- Outer CV joints are now meshed with the slots inside the wheel hub
7- The threaded portion of the CV joint is now located with a crown nut and a pin.
8- Upper and lower knuckle holders are now bolted at their location in the knuckle.
9- control arms are now inserted at their location and located with a pin.
10- The other side of the control arms are placed at their locators in the chassis frame
11- Springs and shock absorbers are mounted vertically connecting the lower control arms and the chassis frame b means of pins.

4.5.3 Steering:

1- Steering shaft is placed at the adequate height inside the front cube portion of the chassis
2- Steering wheel is placed at an appropriate height in front of the driver seat.
3- Steering column now connects the steering wheel to the steeing shaft
4- A pillow block bearing is now used govern the vertical motion of the steering colomn.

CHAPTER 5

RECOMMENDATIONS AND FUTURE WORK:

- SolidWorks has only static simulations. And to achieve better results Softwares having dynamic simulations should be used

- Materials were Selected based solely on their strength, further material properties should be considered

- Use of CNC machines to achieve better and accurate manufacturing of parts, human error may cause fatality of parts.

- Strength of the welded parts should be revised.

- CFD softwares should be used to accurately estimate the Co-efficient of drag.

Refrences:

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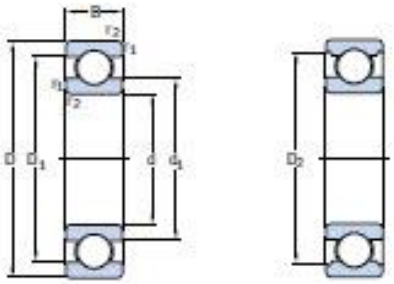
Appendices

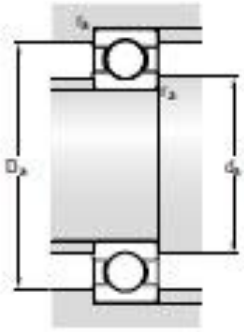
A-1 : Bearing Table:

1.1 Single row deep groove ball bearings
d 12 – 22 mm



Principal dimensions			Basic load ratings dynamic	static	Fatigue load limit	Speed ratings Reference speed	Limiting speed	Mass	Designation
d	D	B	C	C_0	P_u				
mm			kN		kN	r/min		kg	–
12	21	5	1,74	0,915	0,039	70 000	43 000	0,0063	61801
	24	6	2,91	1,46	0,062	67 000	40000	0,011	61901
	28	8	5,4	2,36	0,1	60 000	38 000	0,021	* 6001
	30	8	5,07	2,36	0,1	60 000	38 000	0,026	16101
	32	10	7,28	3,1	0,132	50 000	32 000	0,037	* 6201
	37	12	10,1	4,15	0,176	45 000	28 000	0,06	* 6301
15	24	5	1,9	1,1	0,048	60 000	38 000	0,0065	61802
	28	7	4,36	2,24	0,095	56 000	34 000	0,016	61902
	32	8	5,85	2,85	0,12	50 000	32 000	0,03	* 16002
	32	9	5,85	2,85	0,12	50 000	32 000	0,03	* 6002
	35	11	8,06	3,75	0,16	43 000	28 000	0,045	* 6202
	42	13	11,9	5,4	0,228	38 000	24 000	0,082	* 6302
17	26	5	2,03	1,27	0,054	56 000	34 000	0,0075	61803
	30	7	4,62	2,55	0,108	50 000	32 000	0,016	61903
	35	8	6,37	3,25	0,137	45 000	28 000	0,038	* 16003
	35	10	6,37	3,25	0,137	45 000	28 000	0,038	* 6003
	40	12	9,95	4,75	0,2	38 000	24 000	0,065	* 6203
	40	12	11,4	5,4	0,228	38 000	24 000	0,064	6203ETN9
	47	14	14,3	6,55	0,275	34 000	22 000	0,11	* 6303
	62	17	22,9	10,8	0,455	28 000	18 000	0,27	6403
20	32	7	4,03	2,32	0,104	45 000	28 000	0,018	61804
	37	9	6,37	3,65	0,156	43 000	26 000	0,037	61904
	42	8	7,28	4,05	0,173	38 000	24 000	0,05	* 16004
	42	12	9,95	5	0,212	38 000	24 000	0,067	* 6004
	47	14	13,5	6,55	0,28	32 000	20 000	0,11	* 6204
	47	14	15,6	7,65	0,325	32 000	20 000	0,098	6204 ETN9
	52	15	16,8	7,8	0,335	30 000	19 000	0,14	* 6304
	52	15	18,2	9	0,38	30 000	19 000	0,14	6304 ETN9
	72	19	30,7	15	0,64	24 000	15 000	0,41	6404
22	50	14	14	7,65	0,325	30 000	19 000	0,13	62/22
	56	16	18,6	9,3	0,39	28 000	18 000	0,18	63/22

* SKF Explorer bearing



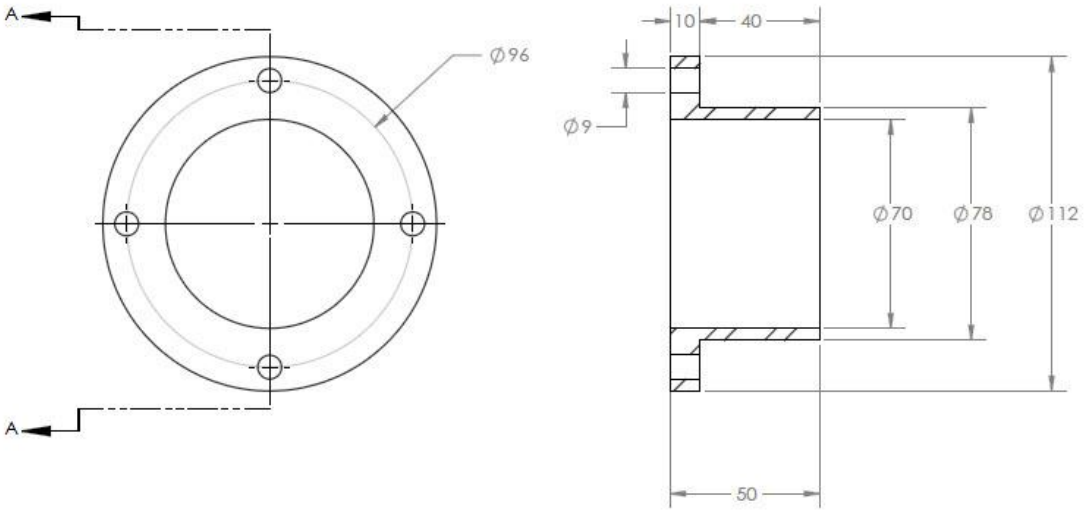
Dimensions					Abutment and fillet dimensions			Calculation factors	
d	d_1 ~	D_1 ~	D_2 ~	$r_{1,2}$ min.	d_a min.	D_a max.	r_a max.	k_r	f_0
mm					mm			–	
12	14,8	18,3	–	0,3	14	19	0,3	0,015	13
	16	20,3	–	0,3	14	22	0,3	0,02	15
	17	23,2	24,8	0,3	14	26	0,3	0,025	13
	17	23,4	24,8	0,3	14,4	27,6	0,3	0,025	13
	18,4	25,7	27,4	0,6	16,2	27,8	0,6	0,025	12
	19,5	29,5	31,5	1	17,6	31,4	1	0,03	11
15	17,8	21,3	–	0,3	17	22	0,3	0,015	14
	18,8	24,2	25,3	0,3	17	26	0,3	0,02	14
	20,5	26,7	28,2	0,3	17	30	0,3	0,02	14
	20,5	26,7	28,2	0,3	17	30	0,3	0,025	14
	21,7	29	30,4	0,6	19,2	30,8	0,6	0,025	13
	23,7	33,7	36,3	1	20,6	36,4	1	0,03	12
17	19,8	23,3	–	0,3	19	24	0,3	0,015	14
	20,4	26,6	27,7	0,3	19	28	0,3	0,02	15
	23	29,2	31,2	0,3	19	33	0,3	0,02	14
	23	29,2	31,2	0,3	19	33	0,3	0,025	14
	24,5	32,7	35	0,6	21,2	35,8	0,6	0,025	13
	24,5	32,7	–	0,6	21,2	35,8	0,6	0,03	12
	26,5	37,4	39,6	1	22,6	41,4	1	0,03	12
	32,4	46,6	48,7	1,1	23,5	55,5	1	0,035	11
20	23,8	28,3	–	0,3	22	30	0,3	0,015	15
	25,5	31,4	32,7	0,3	22	35	0,3	0,02	15
	27,3	34,6	–	0,3	22	40	0,3	0,02	15
	27,2	34,8	37,2	0,6	23,2	38,8	0,6	0,025	14
	28,8	38,5	40,6	1	25,6	41,4	1	0,025	13
	28,2	39,6	–	1	25,6	41,4	1	0,025	12
	30,3	41,6	44,8	1,1	27	45	1	0,03	12
	30,3	42,6	–	1,1	27	45	1	0,03	12
	37,1	54,8	–	1,1	29	63	1	0,035	11
22	32,2	41,8	44	1	27,6	44,4	1	0,025	14
	32,9	45,3	–	1,1	29	47	1	0,03	12

A-2 : Material Table:

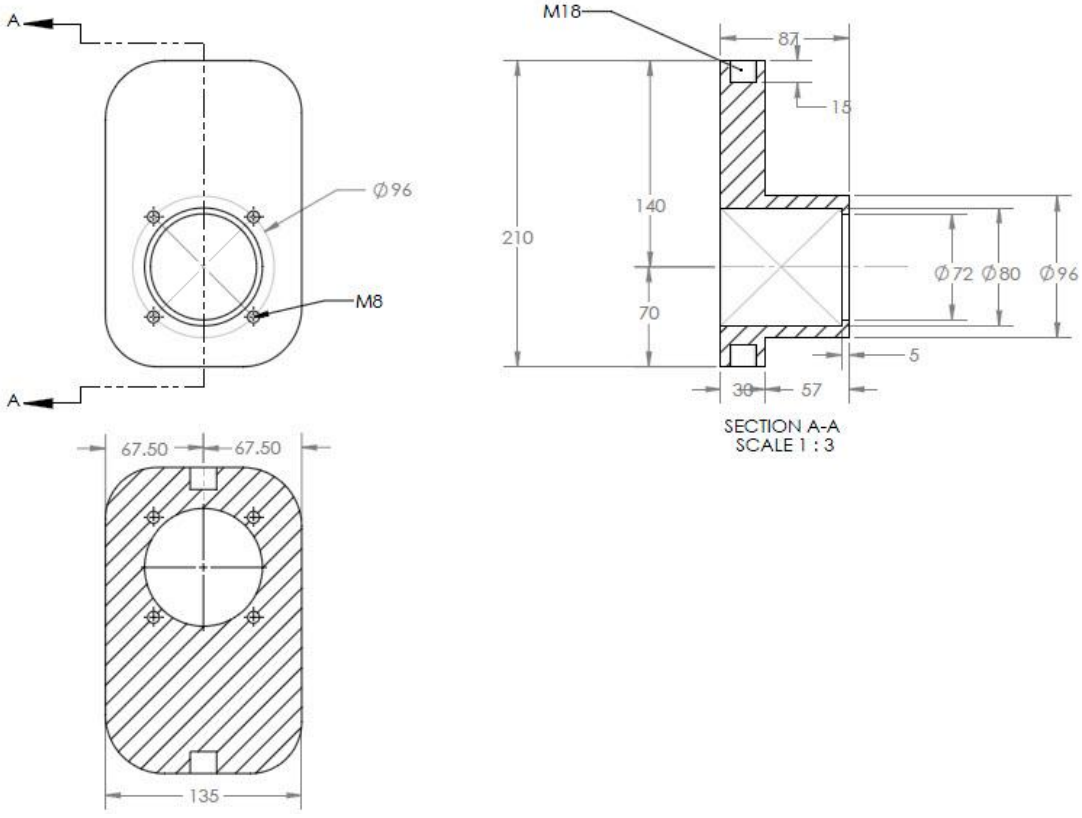
1 AISI No.	2 Treatment	3 Temperature °C (°F)	4 Tensile Strength MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation, %	7 Reduction in Area, %	8 Brinell Hardness
1030	Q&T*	205 (400)	848 (123)	648 (94)	17	47	495
	Q&T*	315 (600)	800 (116)	621 (90)	19	53	401
	Q&T*	425 (800)	731 (106)	579 (84)	23	60	302
	Q&T*	540 (1000)	669 (97)	517 (75)	28	65	255
	Q&T*	650 (1200)	586 (85)	441 (64)	32	70	207
	Normalized	925 (1700)	521 (75)	345 (50)	32	61	149
	Annealed	870 (1600)	430 (62)	317 (46)	35	64	137
1040	Q&T	205 (400)	779 (113)	593 (86)	19	48	262
	Q&T	425 (800)	758 (110)	552 (80)	21	54	241
	Q&T	650 (1200)	634 (92)	434 (63)	29	65	192
	Normalized	900 (1650)	590 (86)	374 (54)	28	55	170
	Annealed	790 (1450)	519 (75)	353 (51)	30	57	149
1050	Q&T*	205 (400)	1120 (163)	807 (117)	9	27	514
	Q&T*	425 (800)	1090 (158)	793 (115)	13	36	444
	Q&T*	650 (1200)	717 (104)	538 (78)	28	65	235
	Normalized	900 (1650)	748 (108)	427 (62)	20	39	217
	Annealed	790 (1450)	636 (92)	365 (53)	24	40	187
1060	Q&T	425 (800)	1080 (156)	765 (111)	14	41	311
	Q&T	540 (1000)	965 (140)	669 (97)	17	45	277
	Q&T	650 (1200)	800 (116)	524 (76)	23	54	229
	Normalized	900 (1650)	776 (112)	421 (61)	18	37	229
	Annealed	790 (1450)	626 (91)	372 (54)	22	38	179
1095	Q&T	315 (600)	1260 (183)	813 (118)	10	30	375
	Q&T	425 (800)	1210 (176)	772 (112)	12	32	363
	Q&T	540 (1000)	1090 (158)	676 (98)	15	37	321
	Q&T	650 (1200)	896 (130)	552 (80)	21	47	269
	Normalized	900 (1650)	1010 (147)	500 (72)	9	13	293
	Annealed	790 (1450)	658 (95)	380 (55)	13	21	192
1141	Q&T	315 (600)	1460 (212)	1280 (186)	9	32	415
	Q&T	540 (1000)	896 (130)	765 (111)	18	57	262

A-3 : Engineering Drawings:

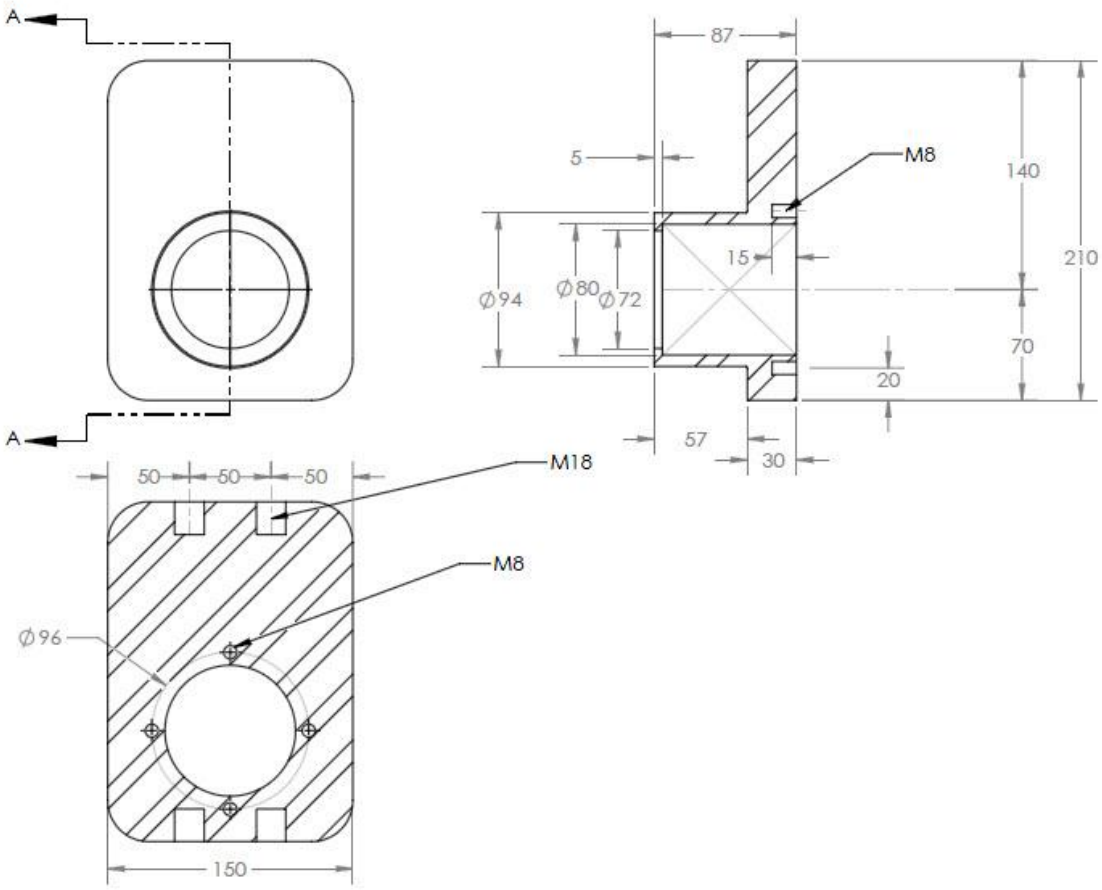
1- Bearing cover:



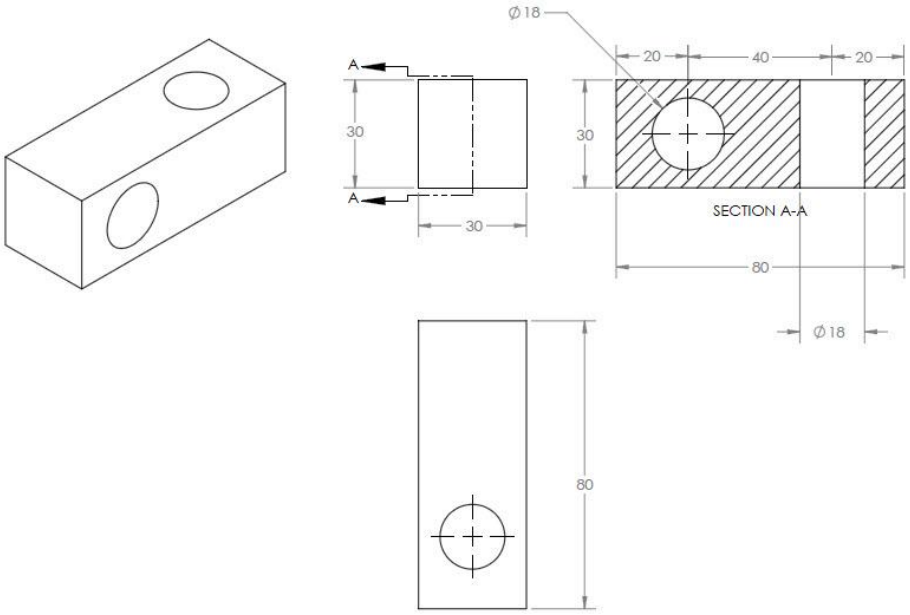
2- Front Knuckle:



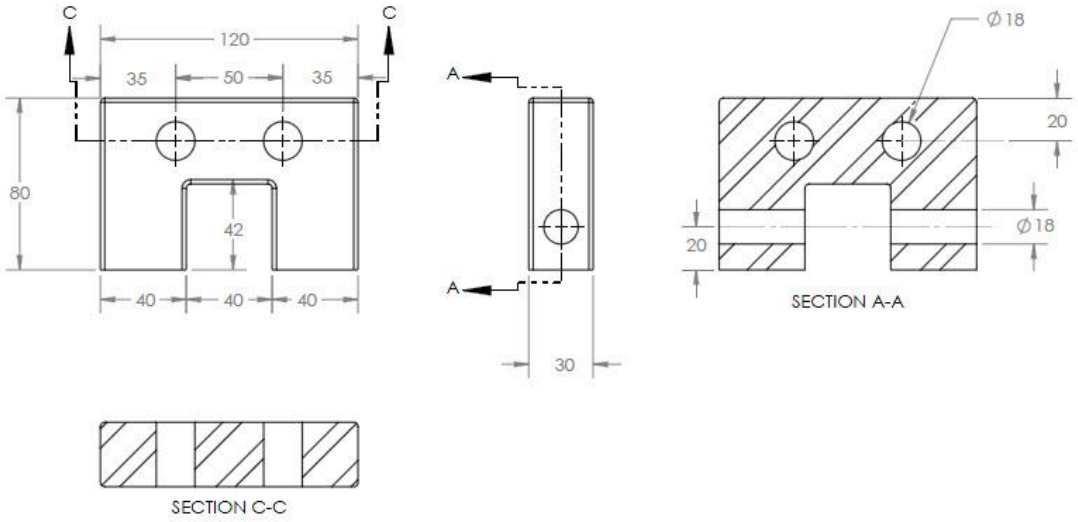
3- Rear Knuckle



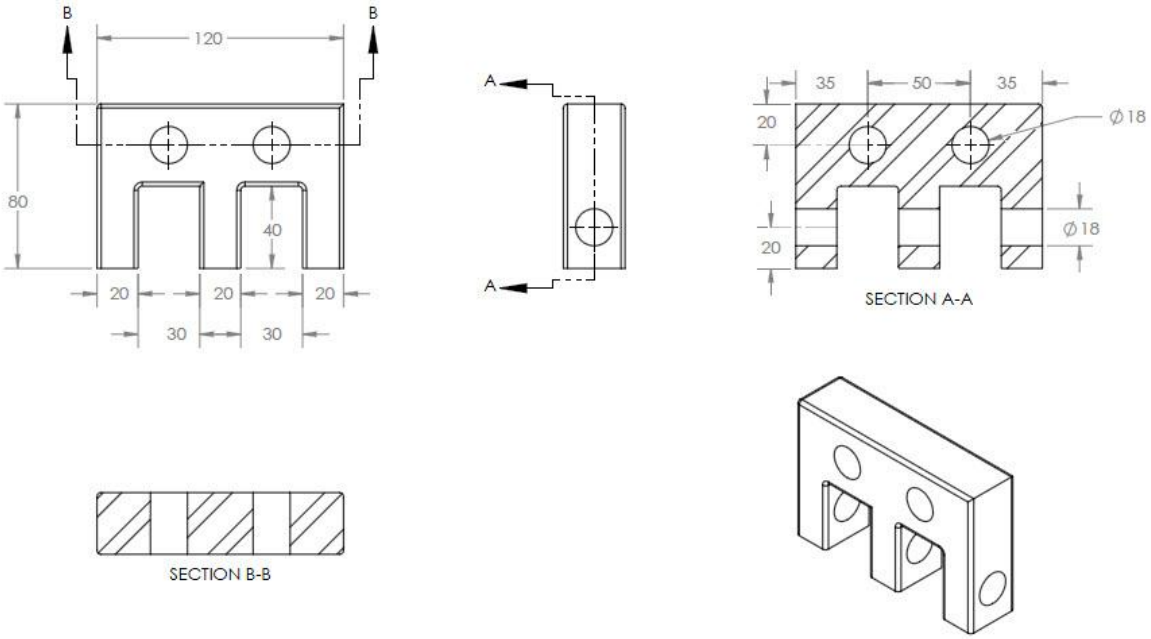
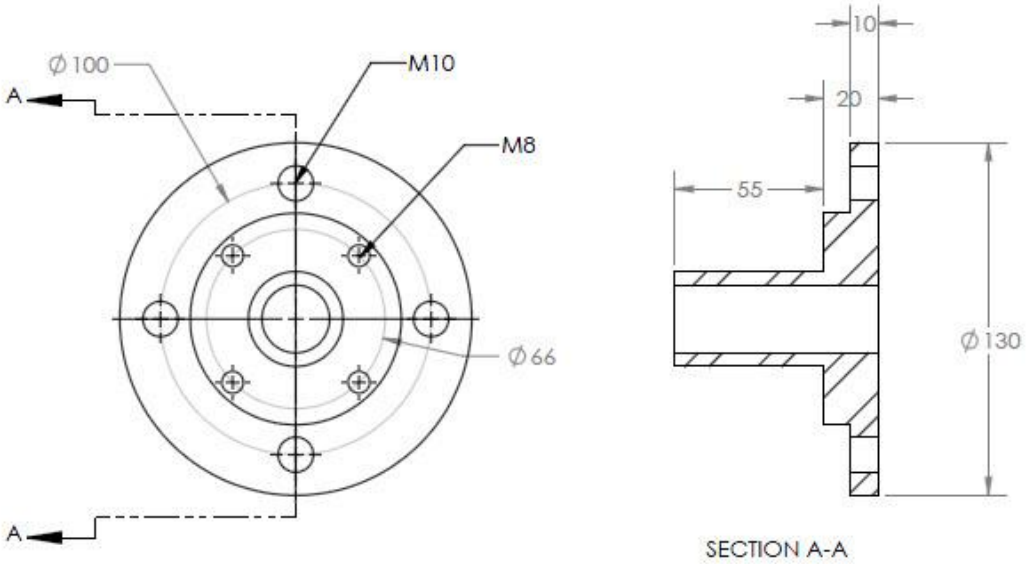
4- Front knuckle holder



5- Upper Rear knuckle holder:



6- Rear Lower Knuckle Holder:



7- Wheel Hub: