ADAMA SCIENCE AND TECHNOLOGY UNIVERSITY

SCHOOL OF MECHANICAL CHEMICAL AND MATERIALS ENGINEERING

DESIGN OF MACPHERSON STRUTS SUSPENSION SYSTEM FOR BAJAJ QUTE FOUR WHEELER SMALL VEHICLE

A Thesis Submitted in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Engineering

By

YONAS HAILEMARIAM

Advisor: Mr. RAMISH BABU (Ass. Professor)

MECHANICAL SYSTEM AND VEHICLE ENGINEERING PROGRAM

September, 2017

ADAMA-ETHIOPIA

CANDIDATE'S DECLARATION

I hereby declare that the work which is being presented in the thesis entitled "**Design of Macpherson struts Suspension System for Bajaj Qute four wheeler small vehicle"** in partial fulfillment of the requirements for the award of the degree of Master of Science in Automotive Engineering is an authentic record of my own work carried out from February, 2017 to September, 2017 under the supervision of Mr. RAMISH BABU (Associate Professor), Department of Mechanical and Vehicle Engineering, Adama Science and Technology University-Ethiopia.

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

Name	Signature	Date
Yonas Hailemariam	____________	________________
Candidate		

This is to certify that the above statement made by the candidate is correct to the best of my knowledge and belief. This thesis has been submitted for examination with my approval.

Name	Signature	Date
______________________	______________	________________
Advisor		

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Approved by board of Examiners

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External Examiner

ACKNOWLEDGEMENT

All praises to the ALMIGHTY OF GOD for His unseen hand, which guided and strengthened me, even at times when I felt; I were not capable of doing it ourselves.

I would like to express my special thanks to my advisor, Mr. Ramish Babu (Associate Professor) of Adama Science Technology University, for his valuable suggestion, comments, continuous support and guidance throughout the entire thesis work.

Finally, my sincere thanks should be forwarded to my beloved family and I would like to thank all my friends and classmates who are either directly or indirectly involved during the process of this thesis was conducted.

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ACRONYMS AND ABBREVIATION

CATIA V5:	Computer Aided Three Dimensional Interactive Application version 5
ANSYS:	Analysis of System
CAD:	Computer Aided Design
3D:	Three dimensional
2D:	Two dimensional
AISI:	American Iron and Steel Institute
FBD:	Free body diagram
FEA	Finite element analysis
FEM	Finite element method
FS	Factor of safety

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ABSTRACT

An Automotive Suspension is the system of parts that give a vehicle the ability to absorb (avoid) shock loads. And it is placed between the frame and the wheel. A vehicle suspension maximizes the road holding, provides steering stability with good handling and ensures the comfort of the passengers. The suspension system duty is to absorb or dissipate energy in addition to supporting the weight of the “Bajaj” three wheeler and four wheeler vehicles. Small vehicles are being imported and used for urban commuting in Ethiopia. The suspension system of these vehicles is found to have design drawbacks due to their design mismatch to Ethiopian road conditions. So, there is a need to improve the design of suspension system to increase its durability. Here in this work a design of Macpherson strut suspension system for Bajaj Qute four wheeler small vehicle was done in order to solve the existing problem of these vehicles in addition there is a pressing need for developing indigenous technology to avoid importation of spare parts which leads to industrial growth in the country. In order to complete this project, there were several steps to be followed. Data collection was done using different methods. Then CAD modeling of the structure was made by using the data recorded from the existing vehicle suspension system. Then design of each component of Macpherson strut and existing suspension system was made. CATIA V5 software was used to draw the 3D model of the suspension system. Also analytical analysis was performed for analysis of the improved suspension system for the required results and Finite element analysis was done using ANSYS software and the result was compared with the existing system. The new models becomes safe than the existing in all aspects. So, the problem in lateral direction which is flexible camber curve is solved, because the Macpherson suspension system have good Camber curve not as flexible as other suspension system and also the stress in existing system leads the system to have flexible camber, this is corrected in the new model. Also, the steering arm position is corrected to the middle of the knuckle to balance the steering force distribution. The vehicle also has lower ground clearance which is 180 mm when the vehicle is- unloaded, with this ground clearance it is difficult to drive in unpaved road. So, in this design the ground clearance of the vehicle is increased to 220mm. Therefore, the vehicle can be driven in all road conditions. After this prototype fabrication of the knuckle and lower control arm was done for demonstration purpose and discussing the results. Finally, I conclude that as per the analysis using different material for different parts of the suspension system is best and also the model of Macpherson suspension system for Bajaj Qute small vehicle is safe.

Keywords: *ANSYS, Shock absorber, Model analysis, Control arm, Vehicle handling*

CHAPTER ONE

1. INTRODUCTION

1.1 Introduction

Today all automotive industries are demanding more efficient vehicles such like Vehicle must be light weight, high durability, less expensive and good ride qualities. To design such automotive vehicles, automotive design engineers are implementing new ideas and technologies [1].

If we all think about vehicle performance, we always think of mileage of vehicle, torque, fuel efficiency and acceleration. But all power generated in engine is waste, if the vehicle driver cannot control the vehicle. For this reason only the automobile engineers turned their attention on vehicle suspension system.

The suspension system of a vehicle refers to the group of mechanical components that connect the wheels to the frame or body. A great deal of engineering effort has gone into the design of suspension systems because of an unending effort to improve vehicle ride and handling along with passenger safety and comfort. The vehicle suspension system provides comfort ride and safety to the passengers from road shocks, apart from this suspension system always maintains a contact between road surfaces and wheels to provide steering stability with good handling. If road is flat, there is no irregularities on the road, suspension system is not necessary to the vehicle. But in reality roads are not free from bumps, path holes and many imperfections. These can interact with the vehicle wheels. When vehicle moves on bump roads, bumps in the road causes the vehicle movement in up and down directions [1].

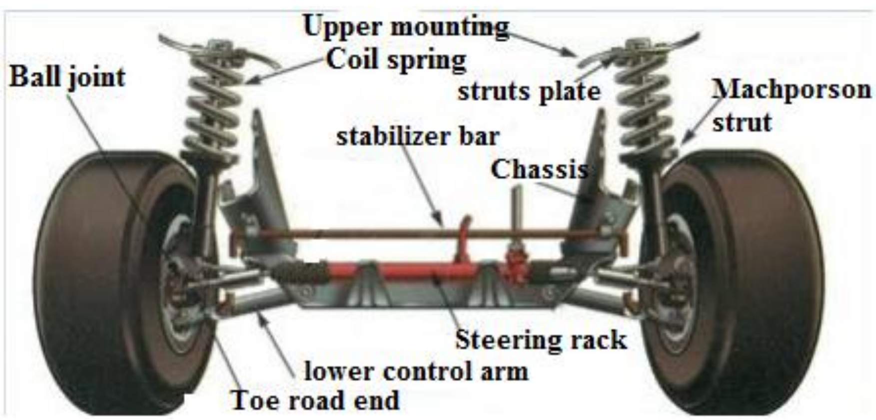
In certain time vehicle with high speed clashes the road bump and experiences very high shock loads, due to this shock load vehicle components are un-controlled from their function and damages will happen. To avoid this automobile engineers introducing suspension system to the vehicle.

Mainly Automobile Suspension systems are two types, Rigid Suspension System and Independent Suspension [1].

In Rigid Suspension System Both Right and left Wheel attached to the same solid axle. When one of the hits the road bumps, its upward movement causes a slight tilt of other side wheel. But In Independent Suspension System allows one wheel to move up and down with minimum effect to the other wheel. McPherson Strut Suspension is type of Independent Suspension System.

The Macpherson strut suspension was created by Earl Macpherson in 1949 for the ford company. Due to its light weight and size compatibility this kind of suspension is widely used in different vehicles. Moreover this kind of vehicle is more popular to be found in the front of the car even though it was also used as a rear suspension [9].

Performance requirements for a suspension system are to adequately support the vehicle weight, to provide effective ride quality which means isolation of the chassis against excitations due to road roughness, to maintain the wheels in the appropriate position so as to have a better handling and to keep tire contact with the ground. However it is well known that these requirements are conflicting, for instance to achieve better isolation of the vehicle chassis from road irregularities, a larger suspension deflection is required with soft damping, while a large damping yields better stability at the expense of comfort. Thus, the idea of incorporating of active or semi-active suspensions can be considered so as to reach these specifications more than those passive one[8].



Link: www.rapid –racer /suspension.php

Figure 1. 1: Macpherson Strut Suspensions system

The Roll of Suspension system

Suspension system is referred to the springs, shock absorbers and linkages that connect the vehicle to the wheels and allows relative motion between the wheels and the vehicle body. Suspension system also keeps the driver or operator isolated from bumps, road vibrations, etc. Also, the most important role played by the suspension system is to keep the wheels in contact with the road all the time. The main role of suspension system is described below:-

- To provide Ride & Handling Performance-
  - **Ride** - vehicle's ability to smooth out a bumpy road
  - **Handling** - vehicle's ability to safely accelerate, brake and corner.
- To ensure that steering control is maintained during maneuvering.

- To support the vehicle weight
- React the control force produced as a result of
    - Longitudinal braking & acceleration forces
    - Lateral (cornering) forces
    - Braking & acceleration torques [10].

Basic Function

One of the functions of suspension system is to maintain the wheels in proper steer and camber attitudes to the road surface. It should react to the various forces that act in dynamic condition. These forces include longitudinal (acceleration and braking) forces, lateral forces (cornering forces) and braking and driving torques. It should resist roll of the chassis. It should keep the wheels follow any uneven road by isolating the chassis from the roughness of the road.

The components of the suspension system perform six basic functions:

- Maintain correct vehicle ride height
- Reduce the effect of shock forces
- Maintain correct wheel alignment
- Support vehicle weight
- Keep the tires in contact with the road
- Control the vehicle's direction of travel[10]

Types of Suspension Components

There are three basic types of suspension components: linkages, springs, and shock absorbers. The linkages are the bars and brackets that support the wheels, springs and shock absorbers. Springs cushion the vehicle by dampening shock loads from bumps and holes in the road. Shock absorbers use hydraulic pistons and cylinders to cushion also the vehicle from shock loads. They also serve to damp spring oscillations, thus bring the vehicle back to a neutral position soon after being shock loaded by a road obstruction [8].

Links: There are a number of various shaped links that are used for the different types of suspension systems. They vary from straight bars to forged, cast or stamped metal shapes that best fit to support the springs, shocks and wheels onto vehicle frames or body structures. The simplest linkage is a straight bar that connects one wheel to the other on the opposite side of the vehicle. Others can be intricately shaped to connect springs, shock absorbers and wheels to vehicles as explained later [8].

Springs: There are three different spring types that are used in suspension systems: coil, leaf and torsion bar. Coil springs are merely wound torsion bars. They are commonly used because they are compact, easily mounted and have excellent endurance life properties. Leaf springs are long thin members that are loaded in bending. They are used as an assembly being comprised of several layers of thin metal to obtain the correct spring rate. Leaf springs serve as both the damping member and the linkage. Torsion bars rely on the twist of a long bar to provide a spring rate to dampen car shock loading. Torsion bars mount across the bottom portion of a vehicle and are more difficult to package than others [6].

Shock Absorbers: Shock absorbers use a piston and cylinder along with adjustable valves to control the flow of hydraulic fluid to set the damping force in both the retract (jounce) and extend (rebound) positions. Shock absorbers are set to retract under a lower force than to extend. This action absorbs road bump forces and dampens spring oscillations resulting in better vehicle ride and control [5].

It is very important to choose suspension type according to the use of vehicle. Most of the Bajaj Qute small vehicles are used in the cities, considering the use of this vehicle on roads in cities Macpherson strut suspension system is selected. This is an independent suspension system, so it increases the ride comfort, stability of vehicle, traction and also reduces the un-sprung mass.

Also Macpherson strut suspension system is light, offers easy packaging with high degree of freedom in design of suspension geometry such as very easy to alter the camber, toe angle etc. of the vehicle according to the requirements [16].

1.2 Ethiopian Transportation System and road condition

Ethiopia does not manufacture automotive vehicles locally at present; The country imports those vehicles from various countries of the world. Automobile importing companies in Ethiopia are importing different types of vehicles to the country's vehicle market. In Ethiopia, road transport is the major means of passenger and different vehicles are used for this purpose and this time Bajaj Qute four wheeler small vehicles are widely used for short distance taxi service. And Bajaj Qute four wheeler small vehicles are one of the highly imported vehicles in the country. In 2006, Ethiopia had 33,300 km of roads (5,400 km paved and 27,900 unpaved). So, due to undesirable road conditions of the country the suspension system of the Bajaj Qute four wheelers is mostly not matching and have impact in underside of the vehicle due to the lower ground clearance of the vehicle. So, improvement for the suspension system of the vehicle is necessary in order to match the vehicle with the road condition and to correct the vehicle flexible camber curve with respect to the roll center of the vehicle at lateral direction [25].

1.3 Problem Statement

The existing type of suspension system which is the front single wishbone and rear semi trailing arm of the Bajaj Qute have some limitation in lateral direction during steering this leads the vehicle to excessive vibration doe to its flexible camber curve and steering arm position with respect to roll center and there is impact under side of the vehicle doe to lower ground clearance with respect to the road conditions of Ethiopia. The existing front knuckle also have small thickness this leads the system to bend easily and the rear existing semi trailing arm needs high space and weight. So, to solve these problems and to improve the vehicle camber angle during lateral direction, road handling, and passenger comfort, ground clearance, designing of Macpherson strut suspension system is essential.

1.4 Objective

1.4.1 General objective

The general objective of this work is to design Macpherson strut suspension system for Bajaj Qute four wheeler small vehicles.

1.4.2 Specific objective

- ✓ To collect data related to existing front and rear suspension system
- ✓ To design and model the existing and new suspension system components
- ✓ To analyze and simulate the models of suspension system
- ✓ To fabricate prototype of the new knuckle and lower control arms of suspension system

1.5 Significance of the Project

This work helps in improving the suspension system in order to have correct camber curve and steering stability of the vehicle when the vehicle is riding on lateral direction with respect to the vehicle roll center and in order to have correct ground clearance. And it helps improving indigenous technology. Beneficiaries: automobile industries, researchers, society and country as a whole.

1.6 Project Scope

In order to reach the objectives, there is the scope of project which can be a benchmarking, starts from concept generation, design and modeling of suspension structure using force balance calculation and CAD, finite element analysis of suspension parts and lastly, to make prototype fabrication of the new front and rear Macpherson strut suspension components of knuckle and lower control arm.

1.7 Limitations

The limitation that can be encountered when conducting the research was lack of chrome Vanadium helical spring manufacturing factory in the country to manufacture the spring for the prototype.

1.8 Organization of the thesis

This thesis consists of eight chapters. It begins with the introduction chapter. This chapter gives the brief description of the problem statement and introduction of the Macpherson strut suspension systems, the objectives, scope, and significance of the work, limitation and organization of thesis.

Chapter Two discusses the literature review and the others related works from previous existing modeling paradigm and technique to identify a suitable system and estimate the parameters to achieved a good performance of the Macpherson strut suspension system for Bajaj Qute four wheeler small vehicle.

Chapter Three discusses methodology and procedures which is used to finalize the work. This includes Design and Modeling using CATIA V5 software. In this design and modeling of all components of suspension system of the existing and the new one was done. And also different analytical analysis has been done which is un input for the finite element analysis. This chapter also deals with Finite element analysis using ANSYS software and different analysis was done to check the visibility of this work.

Chapter Four includes preliminary, Results and discussion of the design of Macpherson strut suspension system for Bajaj Qute four wheeler small vehicles.

Finally, chapter Five describes summery, conclusion and recommendation of the thesis.

CHAPTER TWO

2. LITERATURE REVIEW

2.1 Introduction

There are eight main areas of interest in literature that are studied in the following sections. These parts of interest are; suspension system, Macpherson strut suspension, data collection, design of components, CAD modeling, optimization, analytical analysis and finite element analysis. According to what has already been discussed, parametric objective function is critical for optimization of imposed force on the vehicle by the road. Hence, by making a model of vehicle and its suspension system, it is possible to determine analytical analysis of the system and make desired objective function for optimization in order to perform CAD modeling.

2.2 Suspension system

Suspension is the term given to the system of springs, shock absorbers and linkages that connects a vehicle to its wheels and allows relative motion between the two parts. Suspension systems serve a dual purpose contributing to the vehicle's road holding, handling and braking for active safety and driving pleasure. In addition, Suspension systems are used for keeping vehicle occupants comfortable and reasonably well isolated from road noise, bumps, and vibrations.[16]

Types of Suspension system

Factors which primarily affect the choice of suspension type at the front or rear of a vehicle are;

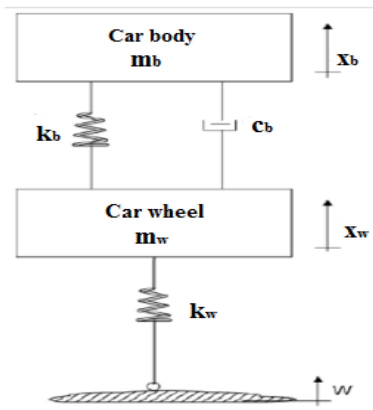
- Engine Location
- Whether the wheels are
    - Driven or undriven and
    - Steered or unsteered

In general, Suspensions can be broadly classified as

I. Passive suspension
II. Semi-active suspension
III. Active suspension

i. Passive Suspension

The conventional suspension system is passive suspension system. It has two elements one of them is damper and another is spring. In this passive suspension, the purpose of the dampers is to dissipate the energy and the spring is to store the energy. Often a spring design function is expressed in terms of energy storage capacity. In machines, springs are frequently used to store kinetic energy from moving components during deceleration and release this energy during acceleration to reduce peak loads. If a load exerted to the spring, it will compress until the force produced by the compression is equal to the load force. , the spring will oscillate around its original position for a period of time when the load is troubled by an external force. Dampers can absorb this oscillation; therefore, it would only jump for a short period of time. For this type of suspension system damping coefficient and spring stiffness are fixed characteristics therefore this is the main weakness as parameters for ride comfort and good handling vary with different road surfaces, vehicle speed and disturbances [8].



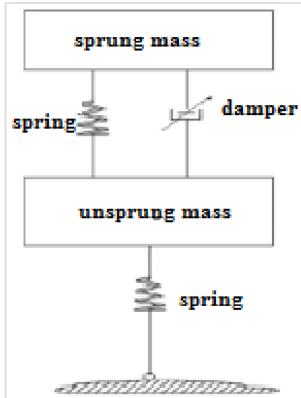
Link: www.researchgate.net/passive suspension system

Figure 2. 1: Passive suspension systems

The performance of passive suspension system can be improved by using the semi-active suspension system.

ii. Semi-active Suspension

The component in the semi-active suspension system is similar with passive suspension system and where external energy is needed in the system, it uses the same function of the active suspension system. The differentiation with passive suspension system is that the damping coefficient can be controlled .The fully active suspension is modified thus the actuator is only capable of dissipating power rather than supplying it as well. The actuator then becomes a continuously variable damper which is theoretically capable of tracking force demand signal independently of instantaneous velocity across it. While having low system cost, light system weight and low energy consumption this suspension system exhibits high performance [8].



Link: www.researchgate.net/semi-active suspension system

Figure 2. 2: The semi-active suspension system

Disadvantage of semi-active suspension: The high frequency harshness that has been observed in road tests and reported in analytical studies of semi active suspension is a significant feature that the performance of 'semi active' suspension is not suitable.

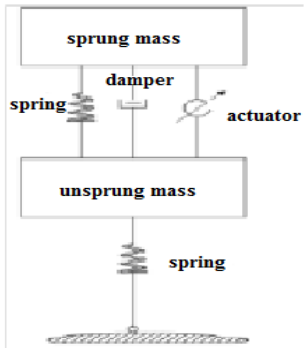
To get good body isolation for low frequency inputs a good semi active system should provide high damping, for good comfort, low damping in the mid - frequency range , adequate damping to control the wheel hop, especially under conditions of motion that requires the development of lateral forces and finally increased damping of structural modes. Semi active suspension that use feedback of modal variables reduces structural vibrations in comparison to the corresponding systems with rigid body based controllers [8].

iii. Active Suspension

As early as 1958, the conception of active suspension system was introduced. The differentiation compare to conventional suspension is active suspension system able to add energy into vehicle dynamic system by the use of actuators rather than dissipate energy. Active suspension can use further degrees of freedom in assigning transfer functions and therefore get better performance. The active suspension system consists an additional element in the conventional suspension, which the main component of it, is an actuator that is controlled by a high frequency response servo valve and which involves a force feedback loop. A control law, which is normally obtained by application of various forms of optimal control theory, governs the demand force signal, which typically generated in a microprocessor.

In theory, this suspension provides best possible ride and handling characteristics. By maintaining an around stable tire make contact with force, maintaining level vehicle geometry and by minimizing vertical accelerations to the vehicle it is done. On the other hand, due to its complication, cost and power requirements, it has not yet put into mass production [8].

In analysis of suspension system, there are varieties of performance criterion, which require becoming optimal. In designing a suspension system there are three performances criterion, which we should consider carefully; they are body acceleration, suspension travel and wheel deflection. By introducing the appropriate controller into the active suspension system, the performance of the system can be more improved.



Link: www.researchgate.net/active suspension system

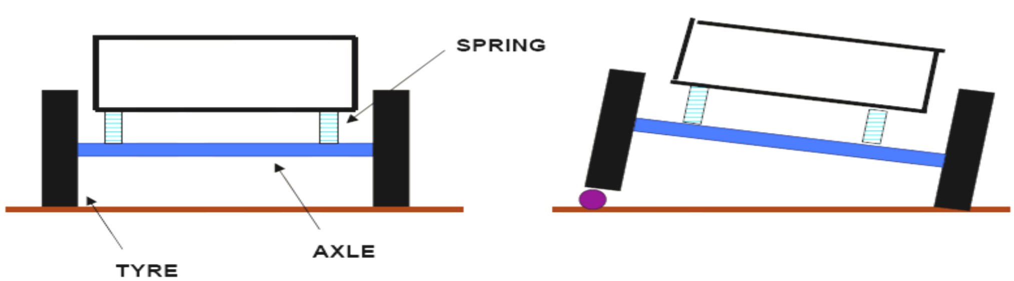
Figure 2. 3: The active suspension system

So, the conventional suspension system can be broadly classified as;

A. Dependent Suspension

This type of suspension system acts as a rigid beam such that any movement of one wheel is transmitted to the other wheel. Also, the force is transmitted from one wheel to the other. It is mainly used in rear of many cars and in the front of heavy trucks. Different types of dependent suspensions are

- Leaf Spring Suspension
- Pan hard rod
- Watt's Linkage



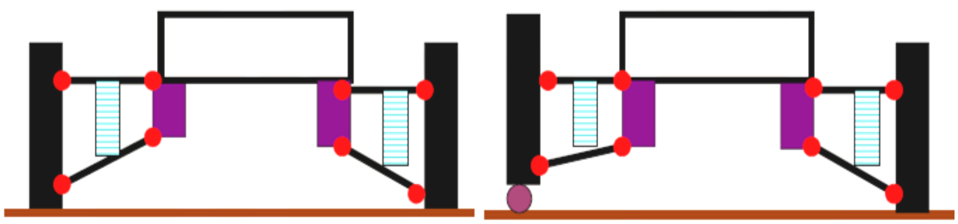
Link: www.slideshare.net/automobile suspension system

Figure 2. 4: Dependent Suspension system

B. Independent Suspension

This type of suspension allows any wheel to move vertical without affecting the other wheel. These suspensions are mainly used in passenger cars and light trucks as they provide more space for engine and they also have better resistance to steering vibrations. Different types of independent suspensions are;

- Swing Axle Suspension
- Macpherson Strut Suspension
- Double Wishbone Suspension
- Trailing Arm Suspension
- Semi-trailing Arm Suspension
- Transverse Leaf Spring Suspension



Link: www.slideshare.net/automobile suspension system

Figure 2. 5: Independent Suspension system

Out of all the above mentioned independent suspension systems, McPherson suspension system is selected as typical suspension system for analysis.

2.3 Macpherson Strut

Macpherson strut suspensions are among the simplest and most commonly used suspension systems globally, in passenger cars. It is

- One of the most popular systems
- One Control Arm
- Light weight
- Economical
- Good ride quality and handling characteristics
- Used for both front and rear suspensions
- Camber curve not as flexible as wishbone.

- It is mechanically simple, requires very little lateral space, its unsprung mass is small. In addition, its up-and-down motion causes very little camber change.
- It requires considerable vertical space. Lateral loads in the strut increases damper friction [20].

Components of Macpherson Suspension System

In McPherson Suspension System apart from the linkages, the basic components of any suspension are springs, dampers and stabilizer also called anti-roll bars. For unsprung masses such as knuckles and control arms, light weight presents additional advantages. Thus a reduction of the weight of the unsprung masses will also remain a most important phenomenon for future developments in suspension system [20].

- **Knuckle**
    - Function of the knuckle is to attach rotating components to suspension components; distribute load from road to body.
    - Vertical, lateral, longitudinal and torque loads are carried by the knuckle.
    - Also known as “wheel carrier”, “hub carrier”, “spindle”.

Link:www.indiamart.com/steering-nuckle

Figure 2. 6: Knuckle

- **Control Arm**
    - Function of the control arm is to attach the knuckle to the chassis, react wheel loads, and to guide the knuckle providing for correct suspension geometry
        - e.g. Camber , Caster, Toe, SAI

Link: www.autopartswarehouse.com/control_arms

Figure 2. 7: Control arm

- **Spring, damper and strut**
    - Function of spring is to support vehicle weight.
    - Function of shock absorber (damper) is to absorb energy from road, to support spring and to reduce sprung mass vibrations.
    - Function of strut is to carry lateral and vertical loading.

Link: www.amazon.com/strut-spring assembly

Figure 2. 8: Spring, damper and strut

Literature survey

Different literatures have been written depending on Macpherson suspension system some are:-

Mr.Sushilkumar P.Taksande (2005), This paper presents a case study of a Macpherson strut automotive suspension analysis, and valuates the uncertainties in the modeling of this complex dynamic problem using a simplified analytical model and a complex computational model. In both cases, variability in design variables is characterized using probabilistic design methods. The results are used in order to accumulate evidence of uncertainties in analytical and computational methods, to correlate predicted results to experimental data for vehicle chassis top mount force and to derive sensitivity measures. A response surface function is approximated which is useful for parametric studies for new variants of the system studied. Sources of uncertainty in this case study and methods for improving the correlations are then suggested [21].

Madhu Kodati, K. Vikranth Reddy (2008), This project focused on the kinematics study to appraise and optimize a Macpherson suspension system which used in passenger car by mean of Multi-body system approach using ADAMS/View software. The results of this research focused on the kinematic characteristic of the vehicle. There was several output result which not archived the target such as the camber angel changed. After the optimization, the camber angel value was increased and these were influenced the vehicle ride stability and performance [20].

M.S. Fallah, R. Bhat andW.F. Xie(2009), The main purpose of this paper is to propose a new nonlinear model of the Macpherson strut suspension system for ride control applications. The model includes the vertical acceleration of the sprung mass and incorporates the suspension linkage kinematics. The performances of the nonlinear and linearised models are investigated and compared with those of the conventional model. Besides, it is shown that the semi-active force improves the ride quality better than active force, while the opposite is true in terms of improving the performance of the kinematic parameters. The results of variations of the kinematic parameters based on the linear model subject to road disturbances are compared with those of a virtual prototype of Macpherson suspension in ADAMS software. The analytical results in both cases are shown to agree with each other [30].

D. Colombo (2009), In this paper that the investigation of cause of premature failure of the upper strut mounts of a McPherson suspension is done. Both experimental tests and numerical analyses had been carried out in order to estimate the service life of the component. The fatigue life of the component has been assessed by a defect tolerant analysis. The result of the investigation is that the upper strut mount failure was due to an impulsive load that cannot be justified by the static and dynamic loads acting on the component caused by road irregularities and vehicle maneuvering during usual working conditions. From the results obtained in the author work it can be therefore concluded that the failure of the upper strut mount was caused by an impulsive load that cannot be justified by the normal working condition for which both the component and the vehicle were designed [29].

Gadhia Utsav D. (2012), Presents that Suspension system is a very essential part of the automobile vehicle. Also, Stability and comfort is totally depended on the suspension system of vehicle. Where, suspensions are used to deal with hump in road surface, in other words, enhancing ride comfort from pits and other irregularities. The function of suspension system is to absorb vibration due to irregularities of road conditions. So in every car suspension system must be designed for the capacity of the vehicle. In the paper it is decided that existing car is for 5 people or not. After doing comparison and calculation researcher concluded that car cannot be used for 5 people rather it can be used only by 2 to 4 persons [27].

S. Pathmasharma et al (2013), in this paper has discussed about the existing suspension system and improved design of suspension system. Author has done modeling using UG and dynamic analysis done in the Automatic dynamic of Mechanical System in which analysis of suspension is carried out by author. In the paper main components like mounting head, track width and other parameters changed by author. Finally, author concluded that modification parameter of the suspension system improves the performance of the suspension system [28].

A. Purshotham (2013), This paper proposes a systematic and comprehensive development of a two-dimensional mathematical model of a McPherson suspension. The paper offers an implementation of the model using Matlab- Simulink, whose dynamics have been validated against a realistic two dimensional model developed with the ANSYS software. In this paper author concluded that the MacPherson suspension system has been modeled after studying dynamic equations to study vibration characteristics of sprung mass of the automobile system with the inclusion of various design parameters such as stiffness, damping, masses, moment of inertia, etc. The results obtained from ANSYS model are compared with the mathematical model implemented on Simulink. It is observed that the displacement and acceleration of the chassis of the automobile obtained in ANSYS are nearer to the values of mathematical model [15].

Javad Meshkatifar, Mohsen Esfahanian(2014), In this paper, the effects of roll center height of McPherson suspension mechanism on dynamic behavior of a vehicle are first studied, and then the optimum location of roll center of this suspension system is determined for a conceptual Class A vehicle. ADAMS/Car software was used for the analysis of vehicle dynamic behavior in different positions of suspension roll center. Next, optimization process has been done using ADAMS/Insight. Results show significant effects of roll center location on body roll angle, body roll rate and steering response. Also, the contradiction between body roll and steering response can be observed [9].

D.V. Dodiya (2014), This research paper presents that the suspension systems have been an economical choice for the vehicle designers. The leading arm has been previously checked for the static characteristics of the vehicle and the analysis of the suspension system, in this they have been considering the dynamic analysis of the suspension system and the consideration of the system in ADAMS and the failure and running conditions and the load and the transfer of forces have been considered to allow the suspension system to simulate and get the results of the working of the leading arm. After analysis author concluded that design of the leading arm is done to reduce the weight and leading arm suspension system works well with the system. Author concluded that modification will give better result in the suspension system and gives effective suspension system [26].

Shrikant S Dodamani, C Suresh, Nagaraj D.(2016), The main objective of this paper is to Design a Coil Spring for Macpherson Strut Suspension System. And analyze this Modified model in Static, Fatigue Loading Conditions. They chose good fatigue strength material, among the existing materials by its Mechanical Properties. Here modeling has been done in CATIA-V5/, Pre-processing of meshing has been done in hyper mesh & solver is used as ANSYS Workbench [1].

To summarize this the literatures surveyed above are many of which focus on case study, kinematics study and optimization, non linear model, effect of roll center height and e.t.c. of Macpherson strut suspension system. So, that is found to be different from the goal of this thesis. Because:- In my work design of Macpherson strut suspension system for Bajaj Qute four wheeler small vehicles was adapted and this work has not done before.

CHAPTER THREE

3. MATERIALS AND METHODS

3.1 Methods

The project process is split up by several minor steps in order to improve the final result of the project. First Data collection was done from manuals of local assembling company and by measuring of the real Bajaj Qute four wheeler vehicle suspension system components. Then CAD modeling of the structure was made by using the data recorded from the existing vehicle suspension system. Then design of each components of Macpherson strut suspension system was done. CATIA V5 software was used to draw the 3D model of the suspension system. Also analytical analysis was performed for analysis of the improved suspension system for the required results and Finite element analysis was done using ANSYS software. After this prototype fabrication of the spring, knuckle and control arm was done using cutting, welding, drilling and forging, spring coiling machines for demonstration purpose and discussing the results. Finally, Recommendation and future work was set on the improved suspension system. The project methodology in the form of a flow chart is graphically shown below in figure 3.1.

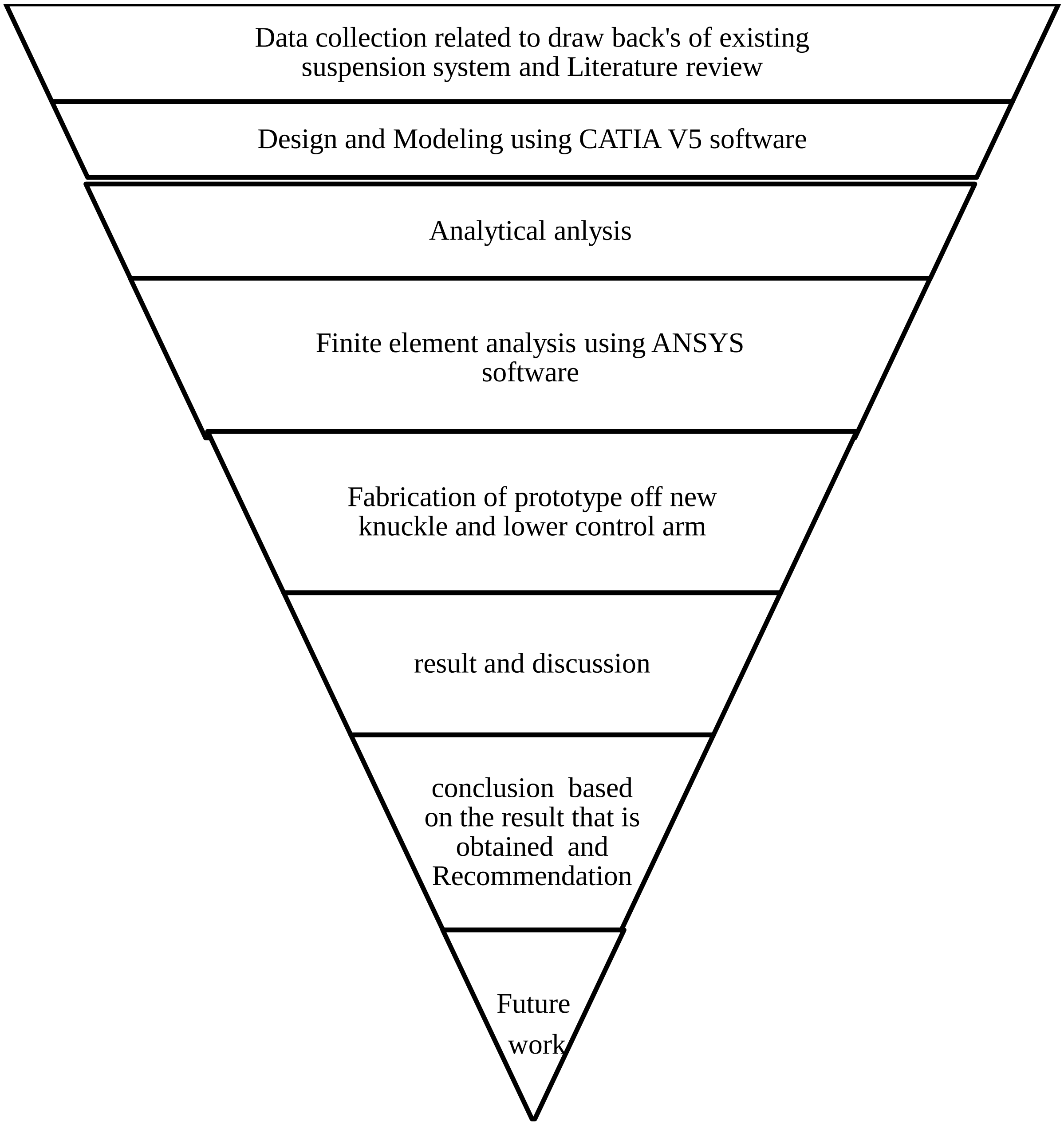


Figure 3. 1: Stages of the methodology

3.2 Materials

Here is the materials, instruments, equipments and software’s which have been used to finalize the thesis and to made prototype of the new knuckle and lower control arm of the suspension system.

Table 3. 1: Materials required

Materials required				
SNO	Item	part	specification	Quantity
1	AISI 1040 pipe steel	Control arm	(35 mm*40 mm)*10 m	
2	Structural steel	Damper components		
3	Beryllium copper Spring wire	spring	12 mm diameter *3 m	
4	Forge steel EN47	knuckle		

Table 3. 2: Instruments and equipments

Instruments and equipments	
1	Material tester
2	Caliper
3	Tape meter
4	Cutting and spring coiler machine
5	Welding and drilling machine
6	Flash
7	Stationary materials
8	Books and journals
9	Computer

Table 3. 3: Software

Software	
1	ANSYS
2	CATIA V5

3.3 DESIGN AND MODELING USING CATIA V5 SOFTWARE

CATIA V5 is modeling software which allows 3D- modeling and 2-D drafting of elements. In order to perform the analysis of the suspension system in ANSYS, it is necessary to model the system in any of the modeling software's such as Pro-Engineers, CATIA or Solid Works, etc. CATIA V5 modeling software is preferred for the modeling in this thesis.

3.3.1 3D-Models of the Existing Suspension System

I. **3D- Model of the Shock absorber**

- ✓ **Top part of the shock absorber**

Figure 3. 2: Top part of the shock absorber

- ✓ **Bottom part of the shock absorber**

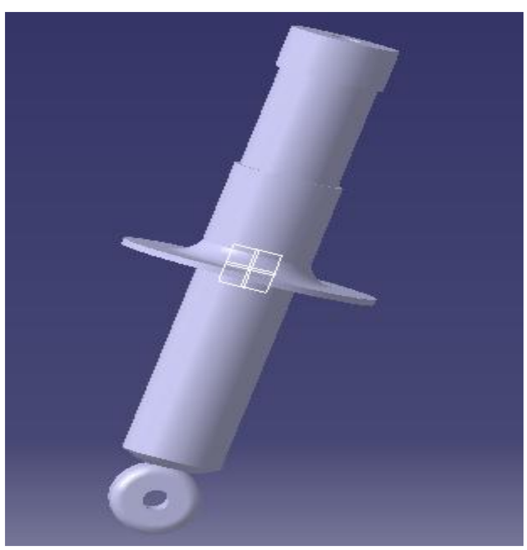


Figure 3. 3: Bottom part of the shock absorber

- **Spring**

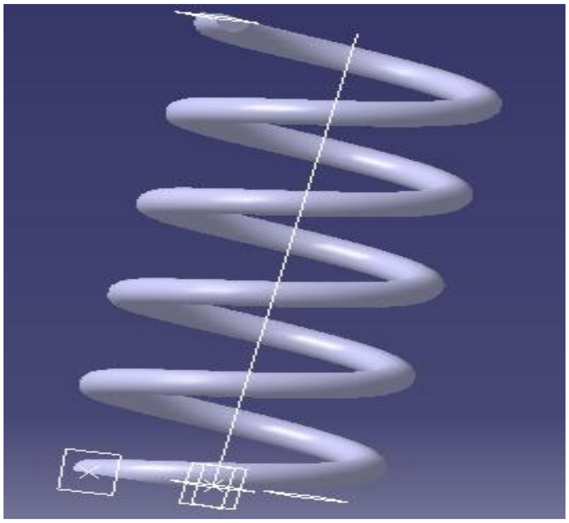


Figure 3. 4: Spring

- **Assembly of the Shock Absorber**

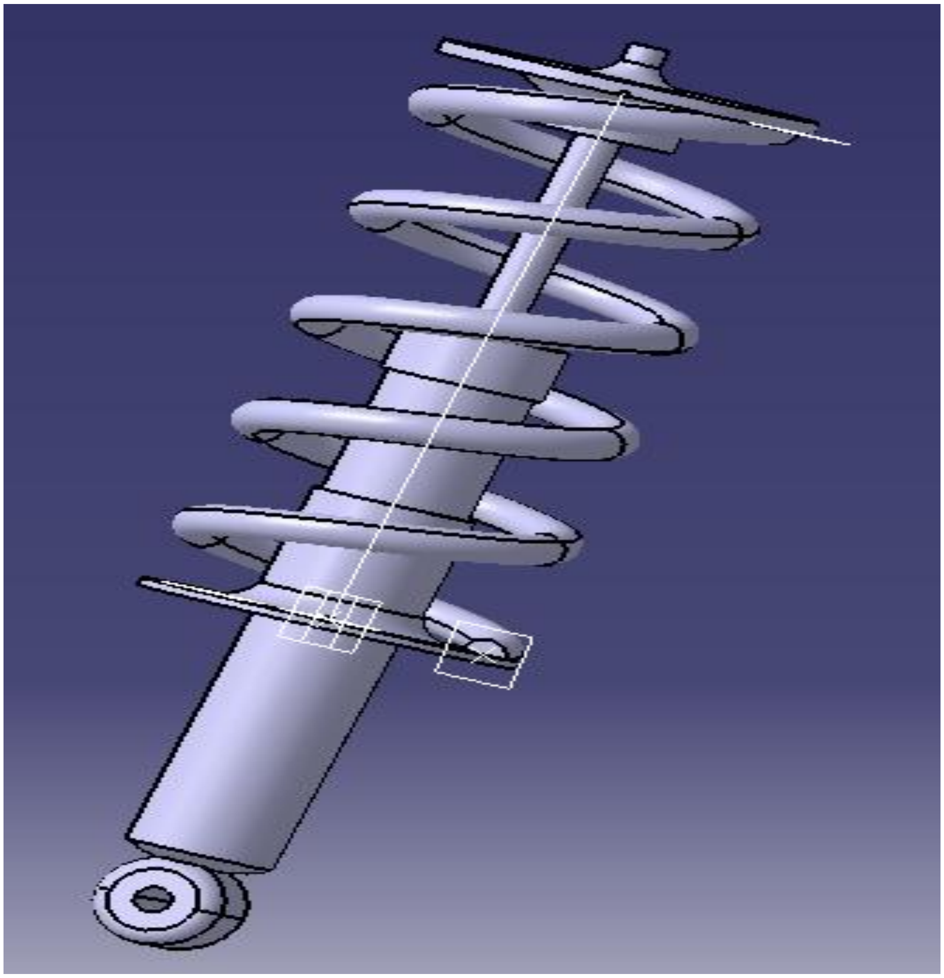


Figure 3. 5: Assembly of the Shock Absorber

II. **3D-Models of the Existing Front Suspension System**

- **Front Lower Control Arm**

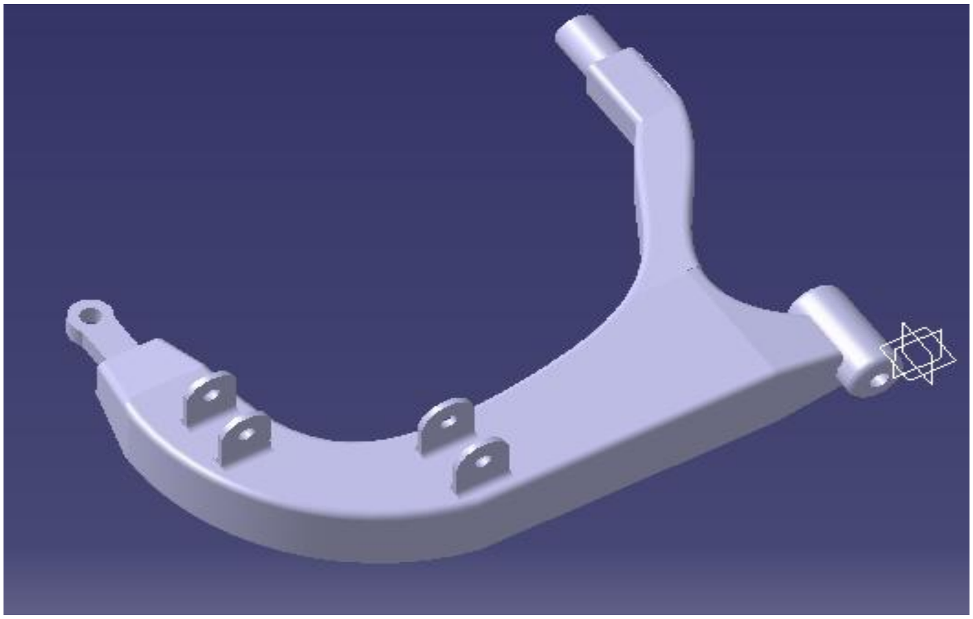


Figure 3. 6: Front Lower Control Arm

- **Front Knuckle**

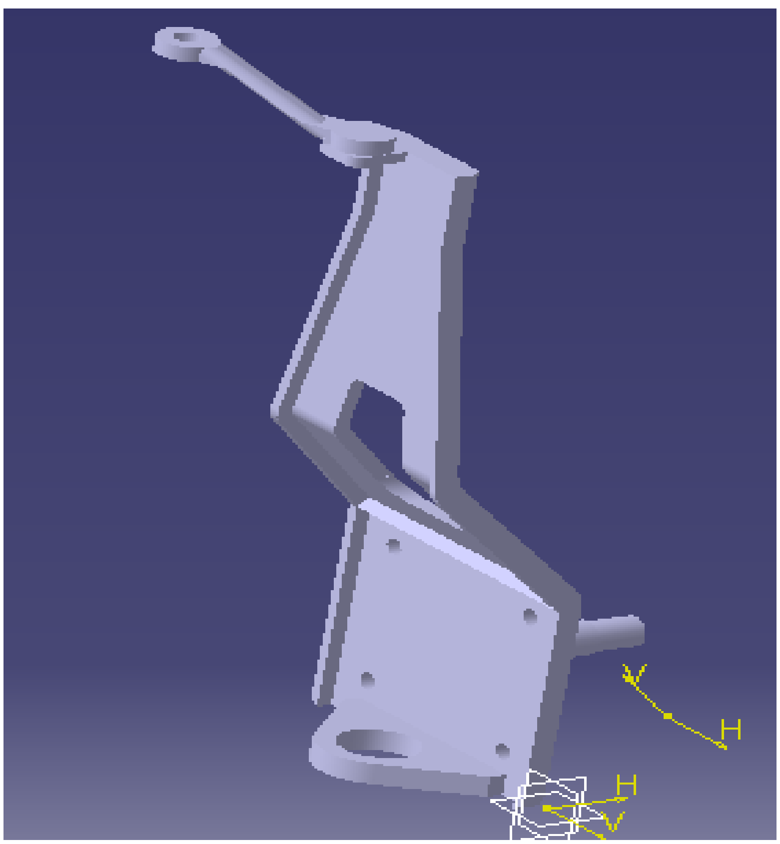


Figure 3. 7: Front Knuckle

- **Assembly of the Existing front suspension system**

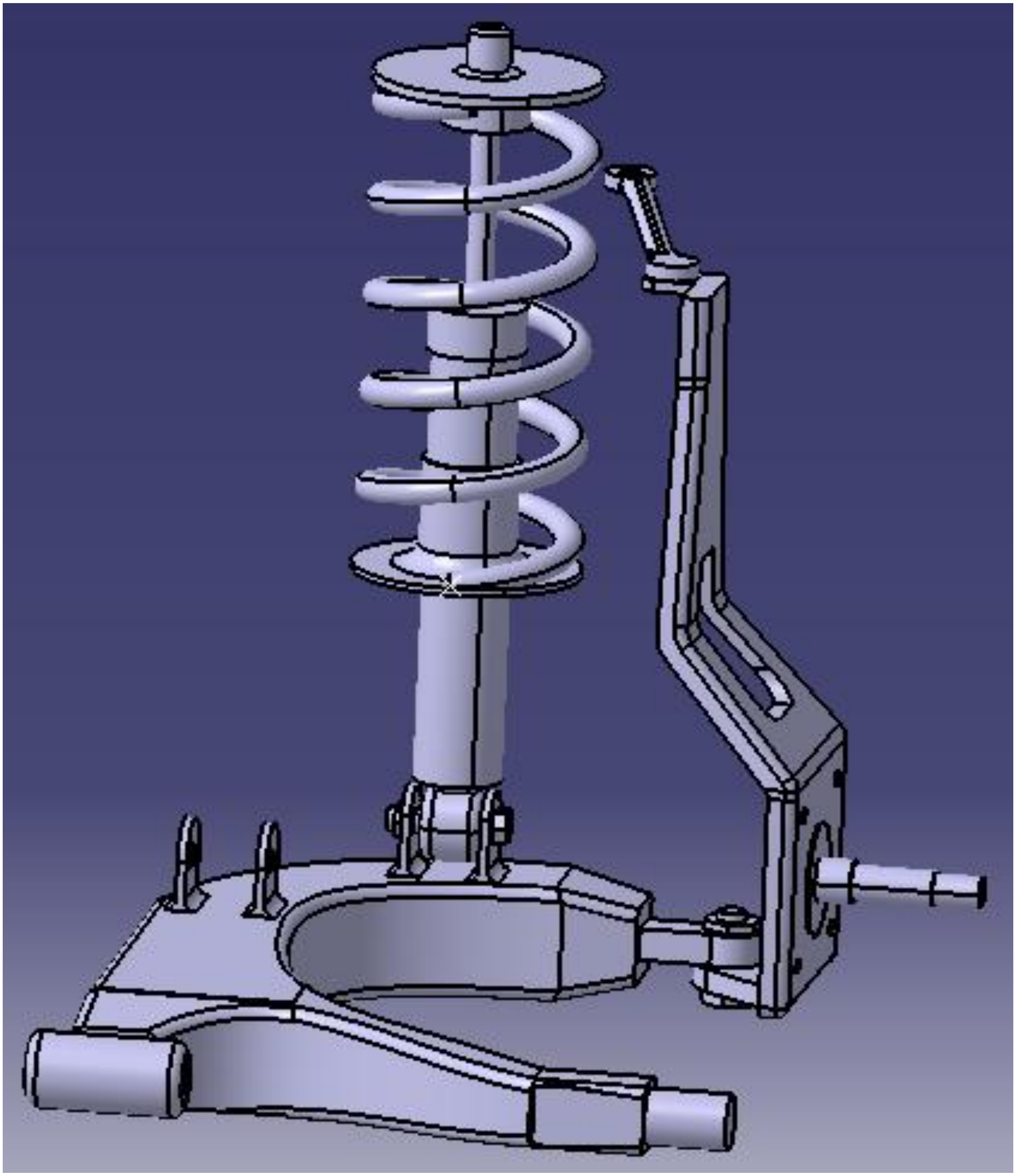


Figure 3. 8: Assembly of the conventional front suspension system

III. **3D-Models of the Existing Rear Suspension system**

- **Rear Semi Trailing Arm**

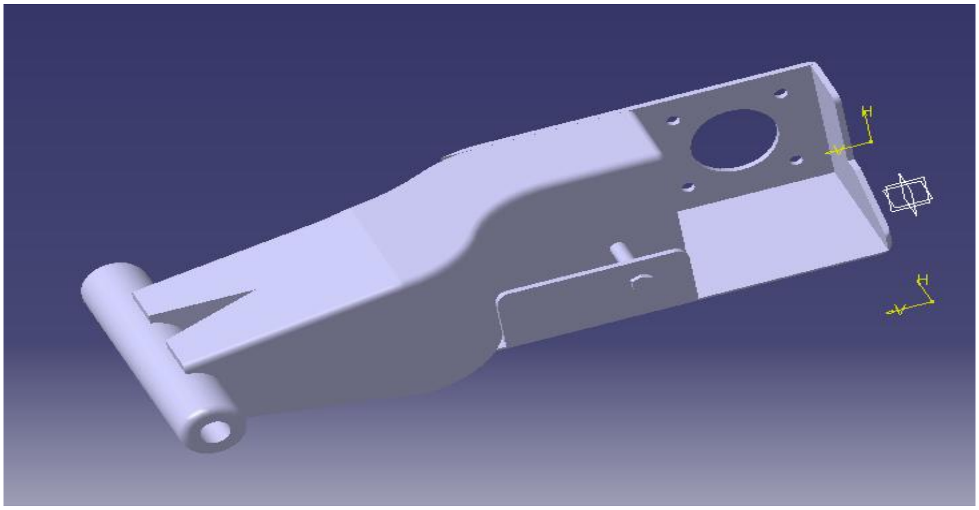


Figure 3. 9: Rear Semi Trailing Arm

- **Assembly of the Existing Rear Suspension System**

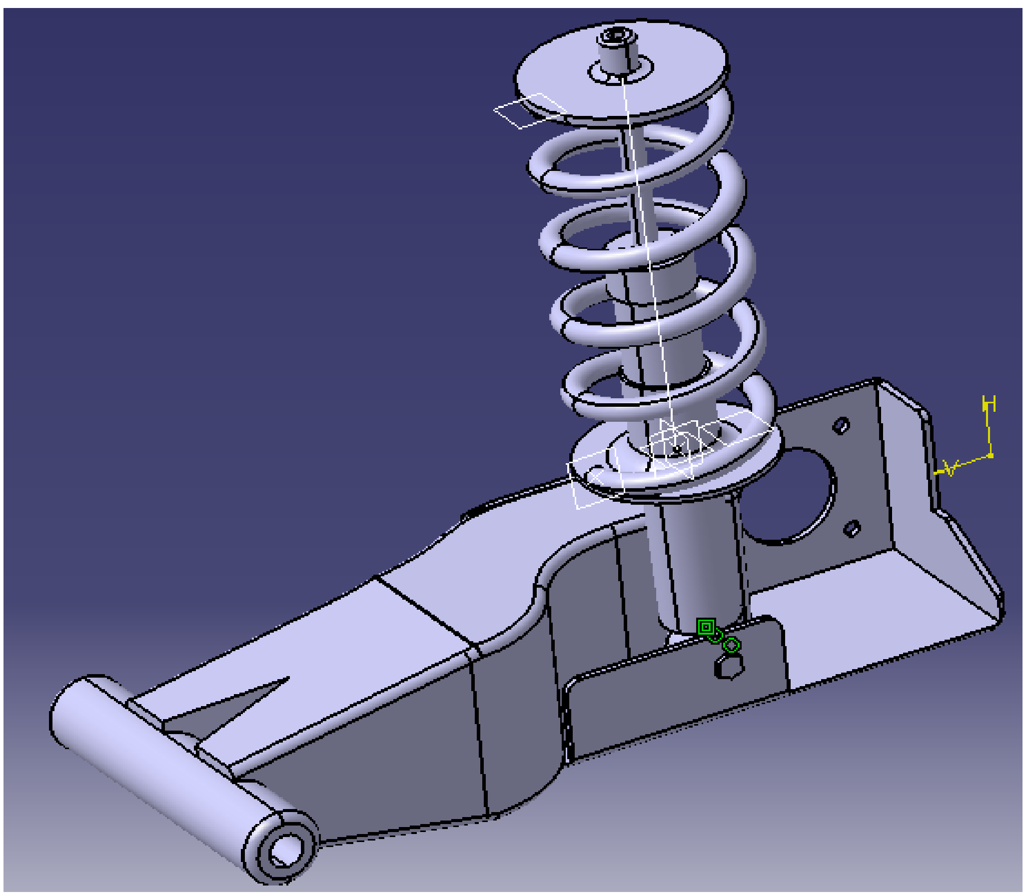


Figure 3. 10: Assembly of the Conventional Rear Suspension System

3.3.2 3D-Models of the New Suspension System

I. **3D-Models of New Shock Absorber**

Shock absorbers are used to damp the vibrations which occur during movements of the suspension. Shock absorbers convert the vibration energy into heat by means of friction. One end is connected to the body or frame, the other to the axle or control arm. Mono tube with Compression Piston type damper is used for this design. Because of:-

- Oil is separated from the gas.
- Having Compression Piston, means lower compression pressures and typically lower gas pressure can be used.

Main components of shock absorber consist of the following part

- Piston rod: It is made of high tensile steel harden and corrosion resistant.
- Piston ring: It is hardened for long life.
- Pressure chamber: It is made from hardened alloy steel machined from solid with closed rear end to with stand internal pressure up to 1000 bar.
- Outer body: It is heavy duty one piece fully machined from solid steel to ensure total reliability.
- Spring: Its duty is to absorb impacts

➢ **Bottom part of the new shock absorber**

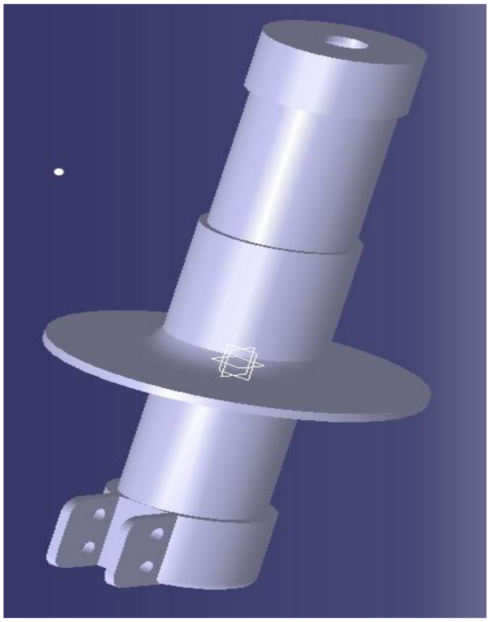


Figure 3. 11: Bottom part of the new shock absorber

- **Top part of the new shock absorber**

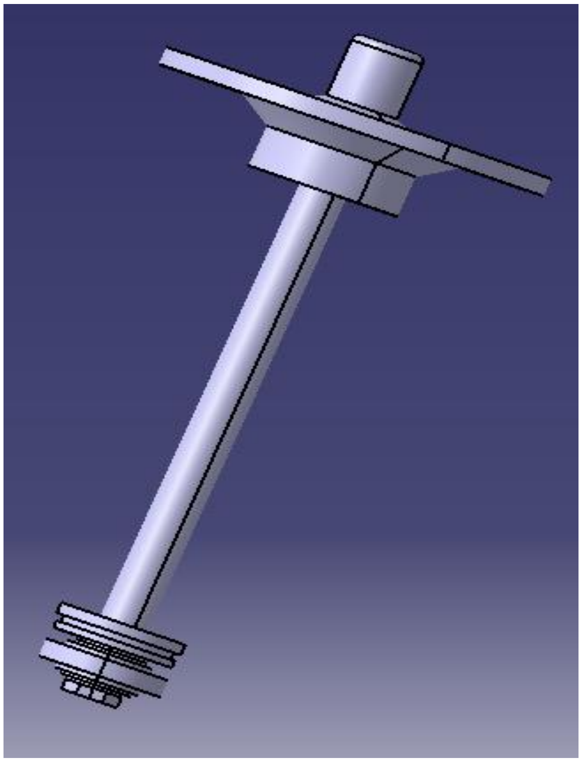


Figure 3. 12: Top part of the new shock absorber

- **Spring**

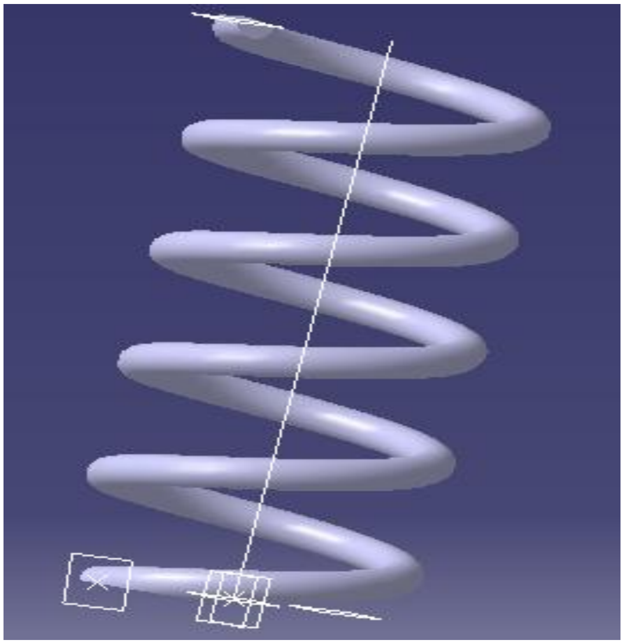


Figure 3. 13: Spring

- **Assembly of the new shock absorber**

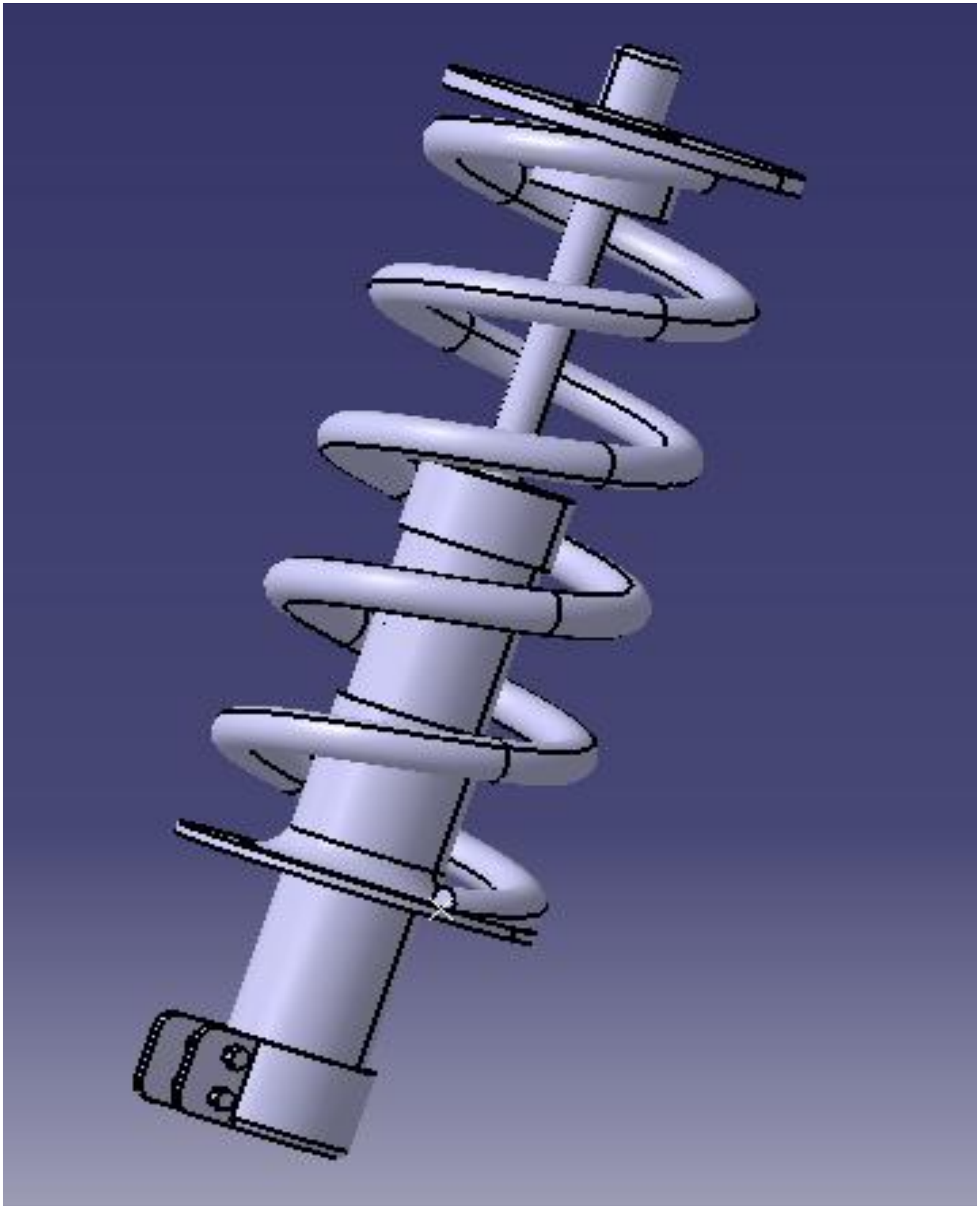


Figure 3. 14: Assembly of the new shock absorber

II. **3D-Models of the New Front Suspension System**

- **Front lower control arm**

The lower control arm is the most vital component in a suspension system. Lower control arm allows the up and down motion of the wheel, attaches the knuckle to the chassis, and to guide the knuckle providing for correct suspension geometry. E.g. Camber, Caster, Toe.

It is usually a steel bracket that pivots on rubber bushings mounted to the chassis. The other end supports the lower ball joint. The lower control arm carries a large portion of the lateral and longitudinal loading from the wheel**.**

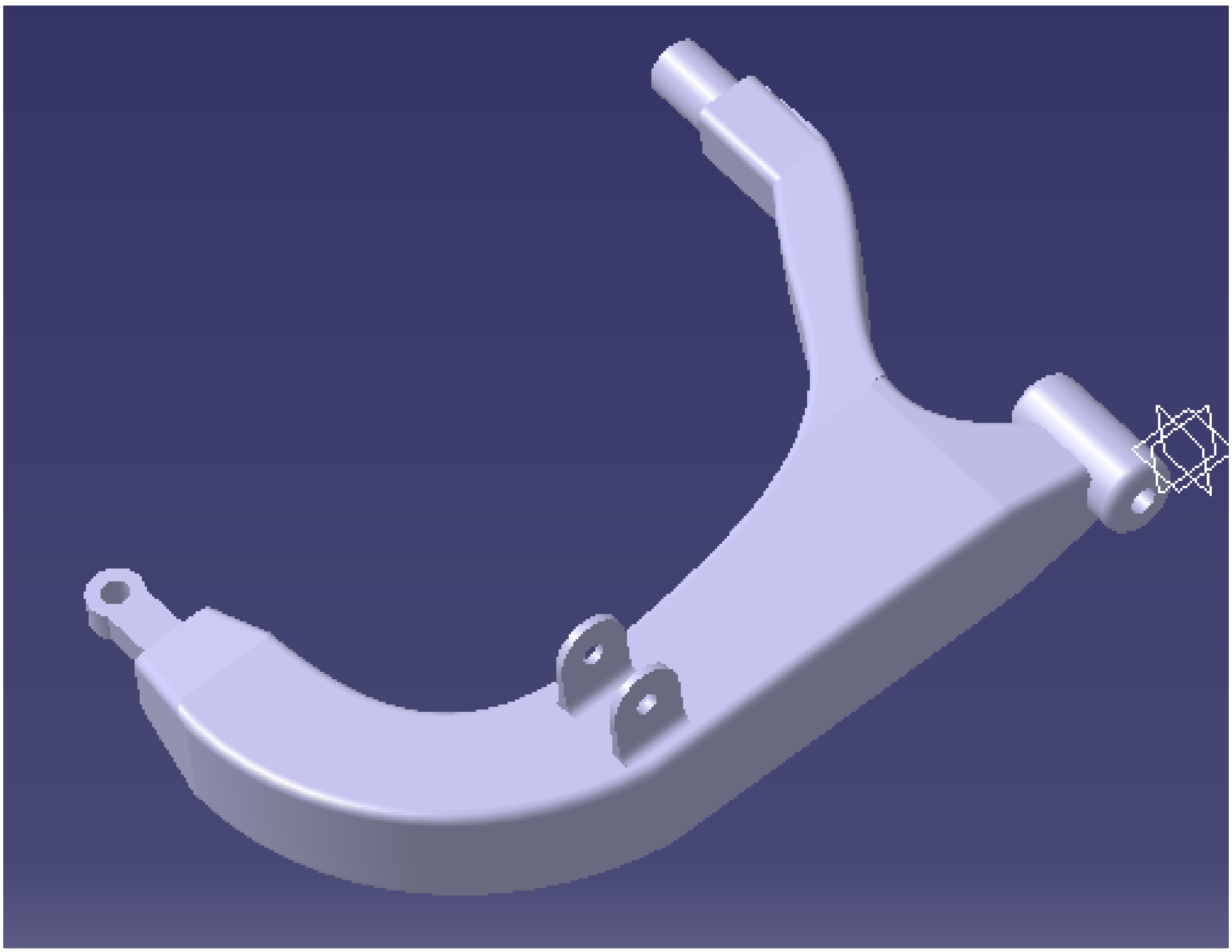


Figure 3. 15: Front lower control arm

➢ **Front Knuckle**

A Steering Knuckle is one of the critical components of vehicle which connects brake, suspension, wheel hub and steering system to the chassis. The forces applied on this component are of cyclic nature as the steering arm is turned to maneuver the vehicle to the left or to the right and to the centre again. The linking parts of the steering system and the suspension system have a direct impact on the performance of the vehicle's ride, durability, and steerability. So, the performance of these parts is directly compared to the quality of the vehicle. The knuckle is vital component that delivers all the forces generated at the Tier to the chassis by means of the suspension system. The design of the knuckle is usually done considering the various forces acting on it which involves all the forces generated by the road reaction on the wheel when the vehicle is in motion.

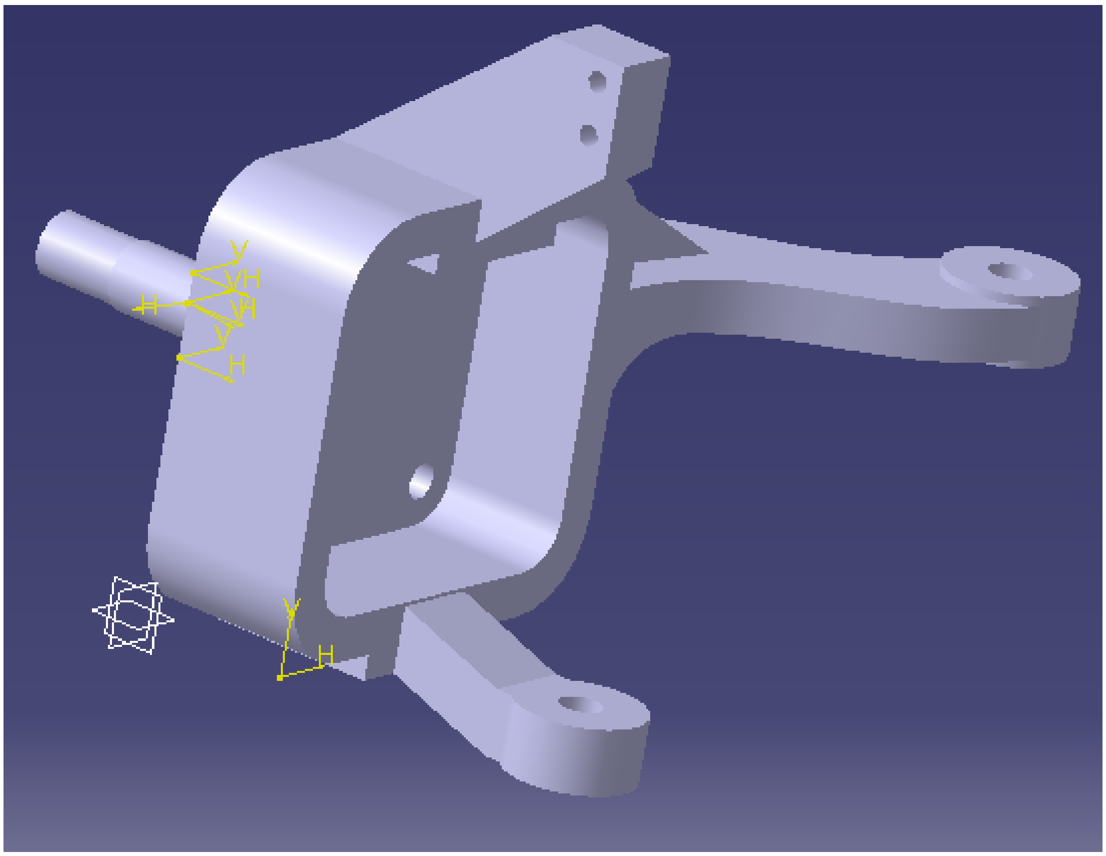


Figure 3. 16: Front Knuckle

- **Assembly of the new front suspension system**

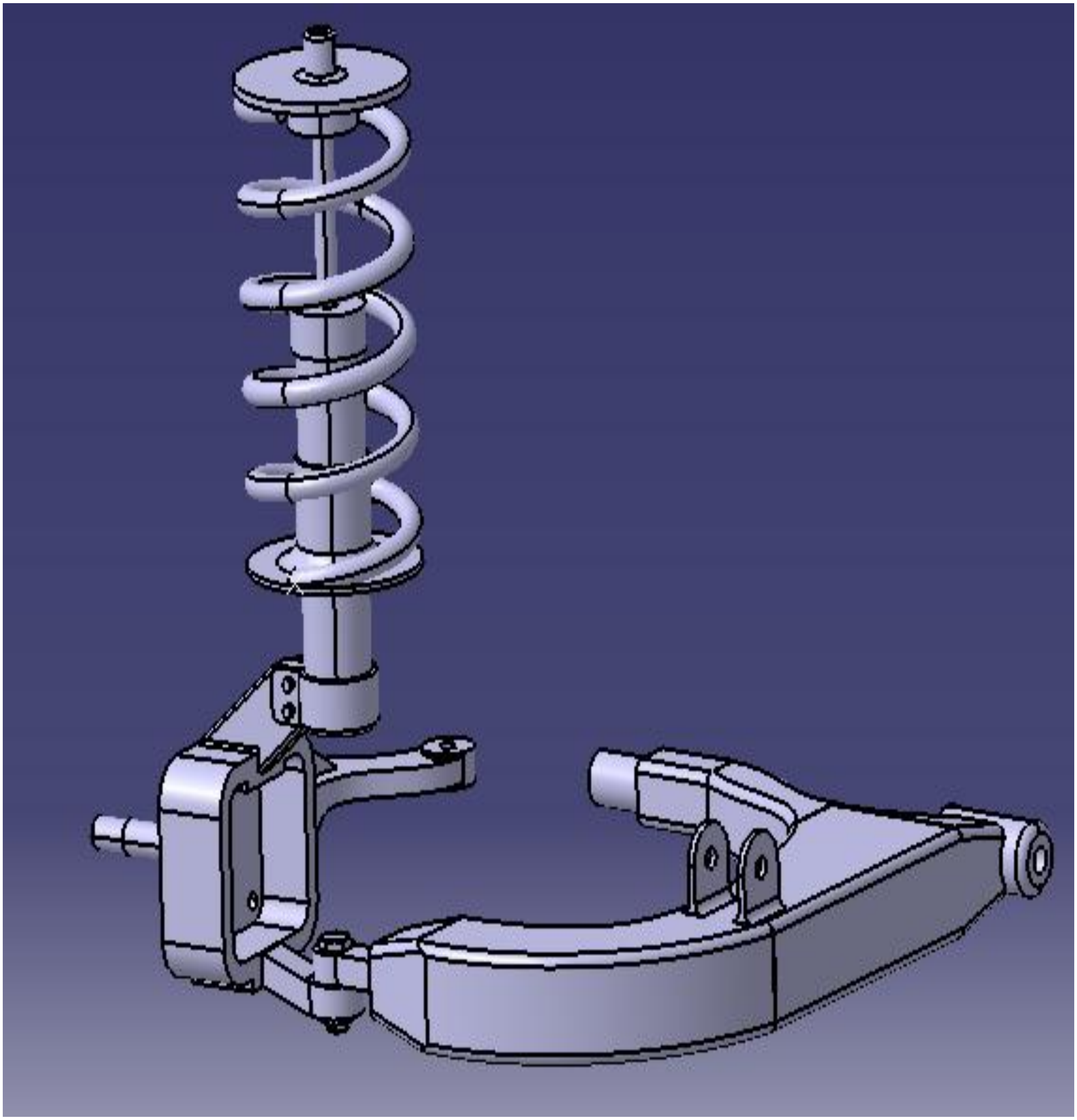


Figure 3. 17: Assembly of the new front suspension system

III. 3D-Models of the New Rear Suspension System

➢ Rear Lower control arm

The lower control arm is the most vital component in a suspension system. Lower control arm allows the up and down motion of the wheel, attaches the knuckle to the chassis, and to guide the knuckle providing for correct suspension geometry. E.g. Camber, Caster, Toe.

It is usually a steel bracket that pivots on rubber bushings mounted to the chassis. The other end supports the lower ball joint. The lower control arm carries a large portion of the lateral and longitudinal loading from the wheel**.**

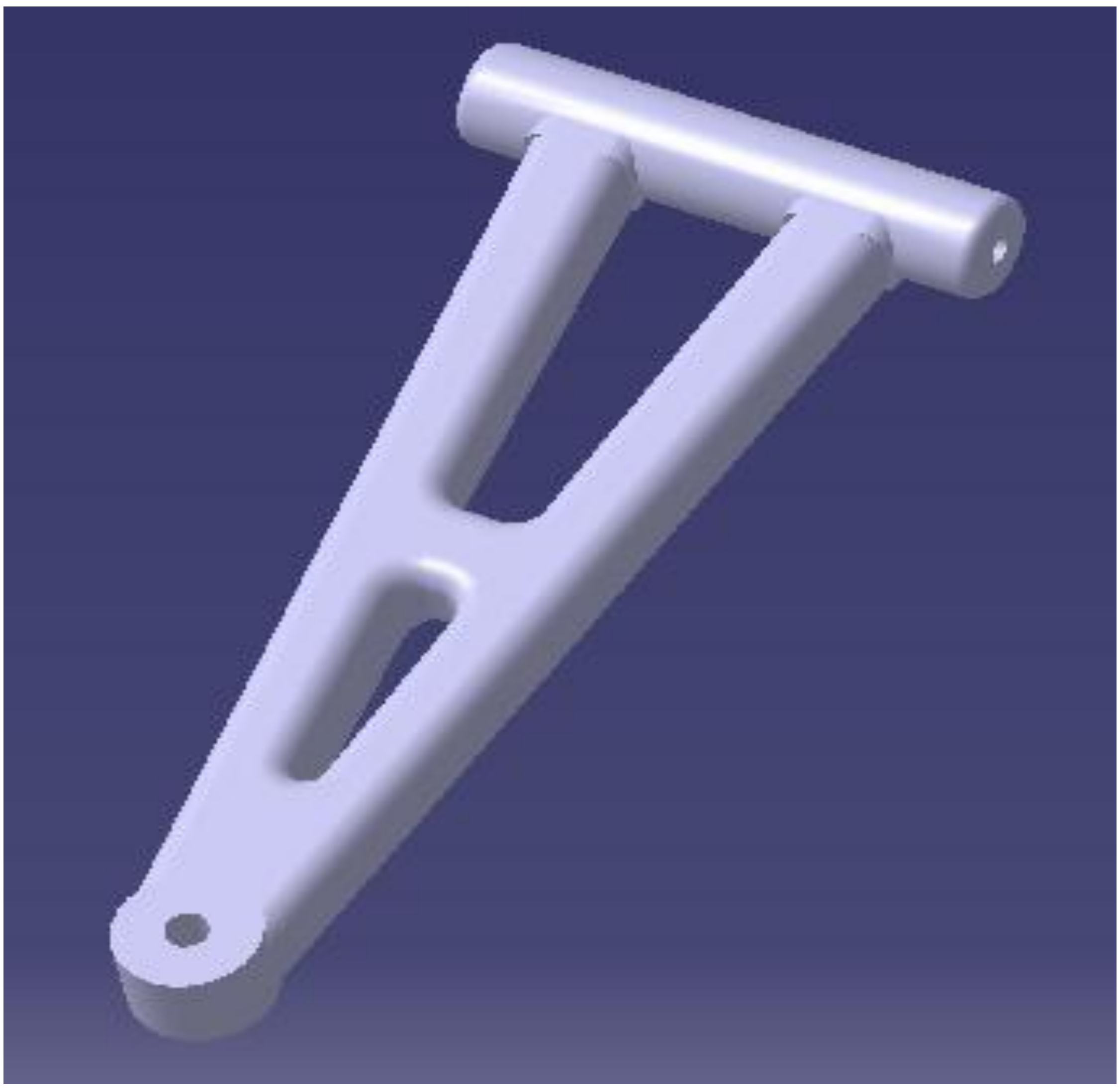


Figure 3. 18: Rear Lower control arm

- **Rear Knuckle**

Rear knuckle is the critical component of the vehicle which is linked with shock absorber and lower control arm. The shape and size of steering knuckle component depends upon the vehicle weight because vertical load of the vehicle is directly acts on it.

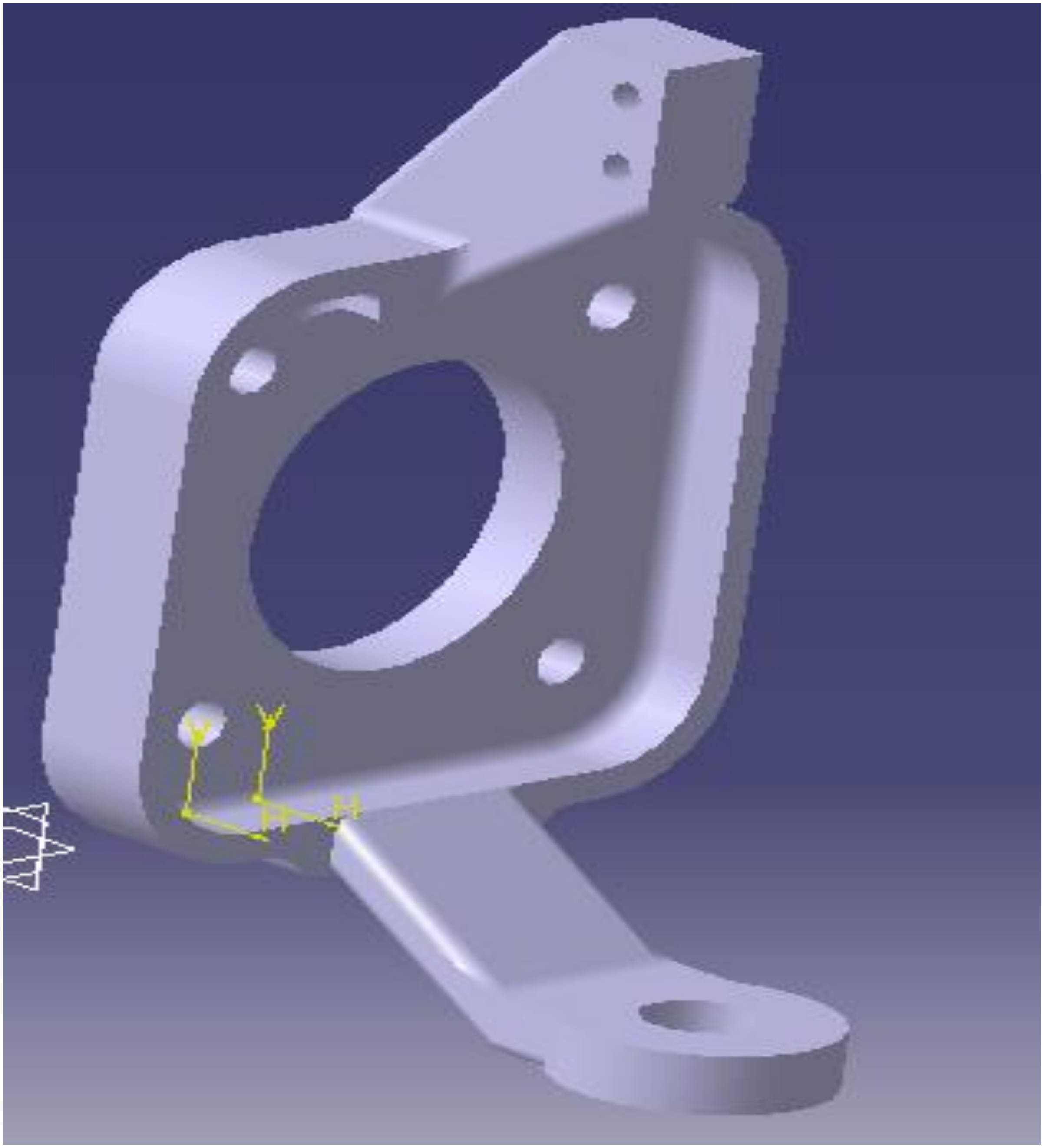


Figure 3. 19: Rear Knuckle

- **Assembly of the rear suspension system**

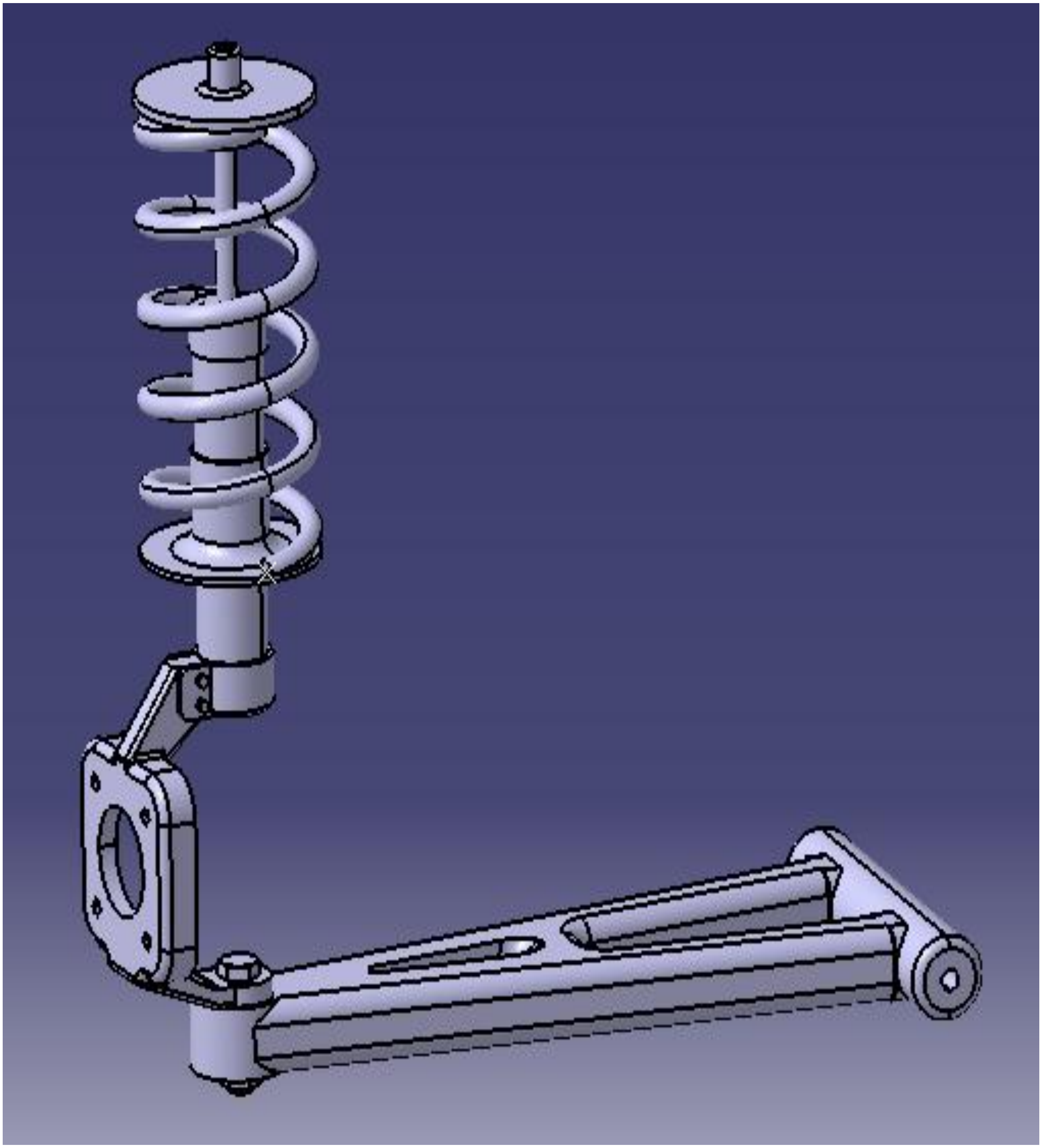


Figure 3. 20: Assembly of the rear suspension system

3.4 ANALYTICAL ANALYSIS

To design the new Macpherson strut suspension system the analysis have been done in two ways which is the simple analysis have been done in analytical analysis method and the complicated analysis have been done in ANSYS workbench 16.0.

Vehicle specification

Table 3. 4: Vehicle specification

Engine	270cc petrol
Max speed	70 Kmph
Kerb weight	399 kg
Seating capacity	driver + 4
Length	2752 mm
Width	1312 mm
Height	1652 mm
Ground clearance	180 mm - unloaded

3.4.1 Design of Shock absorbers

a. **Design of spring**

Material: chrome vanadium

G = 77200$\frac{N}{mm^2}$ = modulus of rigidity

Mean diameter of a coil = D = 86mm

Diameter of wire d =12 mm

Total no of coils n_1 = 5

Height h = 215mm

Outer diameter of spring coil D_0 = D + d = 98mm

Mass of vehicle =399 Kgs

Let average mass of 1 person = 50Kgs

The vehicle carries 5 persons including the driver = 250Kgs

Mass of vehicle + persons = 649Kgs

Let the vehicle front and rear wheels have the same distance from the center of the vehicle. So, divide this to 4 wheels

m = 649Kgs/4=162.25Kgs

Considering dynamic loads it will be double [31].

W = 324.5Kgs * 9.81 = 3183.345N

For single shock absorber weight = 3183.345N = W

Spring index: - The spring index is defined as the ratio of the mean diameter of the coil to the diameter of the wire. Mathematically

C = D/d = 86/12 =**7.2**

Deflection of Helical Springs of Circular Wire

We know that, compression of spring (δ) =$\frac{8*w*C^3*n}{G*d}$=$\frac{8*3183.35N*7.2^3*5}{\frac{77200N}{mm^2}*12mm} = 82mm$ ……… [12]

Solid length: - The solid length of a spring is the product of total number of coils and the diameter of the wire. Mathematically, Solid length of the spring,

$L_S = n'*d=5*12=60mm$

Free length: - is equal to the solid length plus the maximum deflection or compression of the spring and the clearance between the adjacent coils (when fully compressed). Mathematically,

L_F = Solid length + Maximum compression + Clearance between adjacent coils (or clash allowance)

$L_F = n'*d + \delta max + 0.15\ \delta max = 60+82+0.15*82 = 154.3mm$

Spring rate: - The spring rate (or stiffness or spring constant) is defined as the load required per unit deflection of the spring. Mathematically

$K = W / \delta$=3183.35/82=38.82N/mm

Pitch: - The pitch of the coil is defined as the axial distance between adjacent coils in uncompressed state. Mathematically,

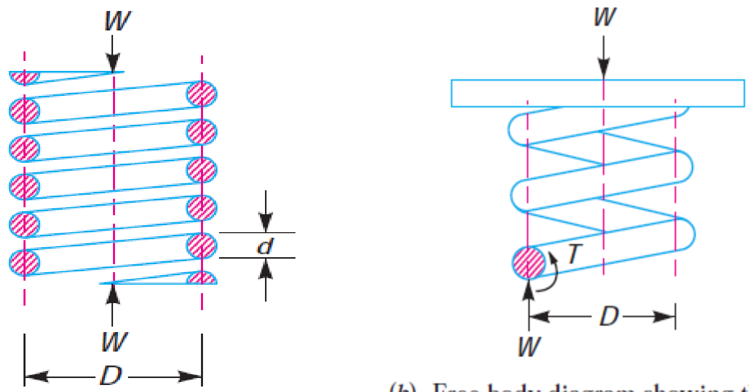
$P=\frac{\text{Free length}}{n'-1}= 154.3/4 = 38.6\ mm$

Stresses in Helical Springs of Circular Wire

Consider a helical compression spring made of circular wire and subjected to an axial load *W*, as shown in Fig below.

D = Mean diameter of the spring coil,
d = Diameter of the spring wire,
n = Number of active coils,
G = Modulus of rigidity for the spring material,
W = Axial load on the spring,

τ= Maximum shear stress induced in the wire,
C = spring index = D/d,
p = Pitch of the coils, and
δ = Deflection of the spring, as a result of an axial load W.



(*a*) Axially loaded helical spring.

(*b*) Free body diagram showing that wire is subjected to torsional shear and a direct shear.

Link: "Textbook of machine design"

Figure 3. 21: FBD spring

A little consideration will show that part of the spring, as shown in Fig. is in equilibrium under the action of two forces *W* and the twisting moment *T*. We know that the twisting moment,

$T=W * \frac{D}{2} = \frac{\pi}{16} * \tau_1 * d^3$

$\tau_1 = \frac{8*w*D}{\pi d^3} = \frac{8*3183.35*86}{\pi*12^3} = 403.65N/mm^2$ .... [12].

In addition to the tensional shear stress (τ_1) induced in the wire, the following stresses also act on the wire:

1. Direct shear stress due to the load *W*, and
2. Stress due to curvature of wire.

We know that direct shear stress due to the load *W*,

$$\tau_2 = \frac{load}{cross - sectional\ area\ of\ the\ wire}$$

$$\tau_2 = \frac{w}{\frac{\pi}{4} * d^2} = \frac{4 * w}{\pi d^2} = \frac{4 * 3183.35}{\pi * 12^2} = 28.16Mpa$$

We know that the resultant shear stress induced in the wire,

$$\tau_= \tau_1 \pm \tau_2 = \frac{8 * w * D}{\pi d^3} \pm \frac{4 * w}{\pi d^2} = 403.65MPa \pm 28.16MPa = 431.81MPa\ or 375.49MPa$$

The *positive* sign is used for the inner edge of the wire and *negative* sign is used for the outer edge of the wire. Since the stress is maximum at the inner edge of the wire, therefore

Maximum shear stress induced in the wire,

$$= \text{Torsional shear stress} + \text{Direct shear stress}$$

$$= \frac{8*w*D}{\pi d^3} + \frac{4*w}{\pi d^2} = \frac{8*w*D}{\pi d^3}\left[1+\frac{d}{2D}\right]$$

$$= \frac{8*w*D}{\pi d^3}\left[1+\frac{1}{2C}\right] = K_s * \frac{8wD}{\pi d^3} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots [12]$$

$$K_s = \text{Shear stress factor} = 1+\frac{1}{2C} = 1+\frac{1}{2*7.2} = 1.07$$

$$\tau_{max} = 1.07 * \frac{8*3183.35*86}{\pi * 12^3} = 431.9Mp\boldsymbol{a}$$

In order to consider the effects of both direct shear as well as curvature of the wire, a Wahl's stress factor (*K*) introduced by A.M. Wahl may be used.

Maximum shear stress induced in the wire,

$$\tau = K * \frac{8wD}{\pi d^3} = K * \frac{8w.C}{\pi d^2}$$

Where $K = K_s * K_c = \frac{4C-1}{4C-4} + \frac{0.615}{C} = 1.206$

$$\tau = 1.206 * \frac{8*3183.35*7.2}{\pi * 12^2} = 489.1Mpa$$

Buckling of Compression Springs

The critical axial load (*Wcr*) that causes buckling may be calculated by using the following relation, *i.e*.

$W_{cr} = k \times K_B \times L_F$

Where k = spring rate or stiffness of the spring = W/δ,

L_F = Free length of the spring, and

K_B = Buckling factor depending upon the ratio

L_F / D=1.79 from table using this ratio K_B=0.7121

$W_{cr} = 38.82 \times 0.7121 \times 154.3 = 4264.43$N

Dimensions of spring

Table 3. 5: Dimensions of spring

Mean diameter (D)	86 mm
Diameter of wire (d)	12 mm
Total no. of spring	5
Height of the spring	215 mm
Outer diameter of the spring coil	98 mm
Spring index	7.2
Solid length	60mm
Free length	155 mm
pitch	38.6

Design of spring

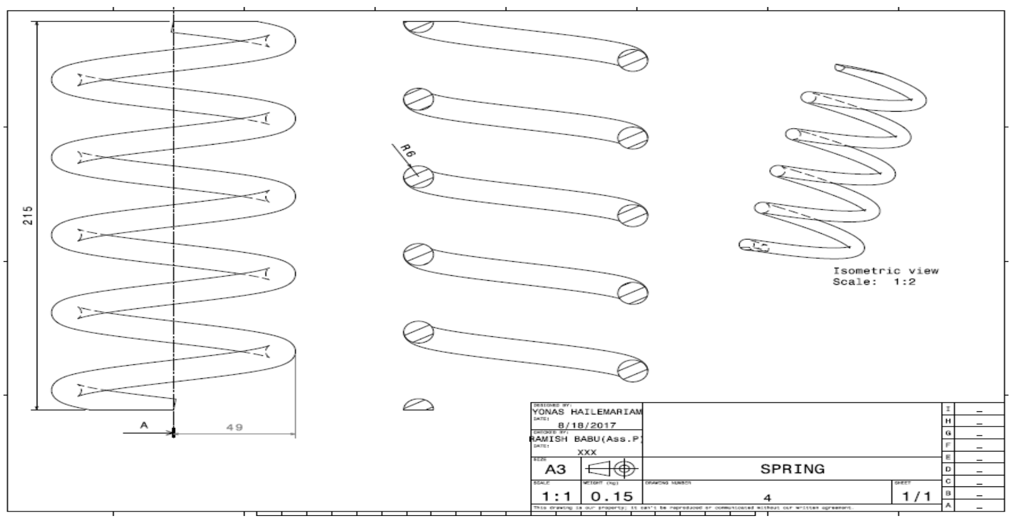


Figure 3. 22: 2D-drawing of spring

b. **Design of damper**

Damping force calculation for mono tube damper has been done simply in analytical method the rest have been done in ANSYS workbench.

Internal dameter of the damper cylinder = d_i =32 mm

Diameter of the rod = 12 mm

Force=Pressure*Area

Area=$\frac{\pi di^2}{4} = \pi * \frac{32^2}{4} = 804.25\ mm^2$

Compression pressure $P_2 = \frac{F}{A} = \frac{3183.35}{804.25} = 3.96\ \frac{N}{mm^2}$

$$A_{rebound} = A_{Tube} - A_{rod}$$

$A_{rebound} = \frac{\pi * 32^2}{4} - \frac{\pi * 12^2}{4}$ =690.743 mm^2

Pressure of rebound $P_1 = \frac{F}{A} = \frac{3183.35}{609.743} = 5.22\ \frac{N}{mm^2}$

$F_{rebound} = (P_1 - P_2) * A_{rebound}$

$$F_{rebound} = (5.22 - 3.96) * 690.743 = 870.336N$$

$F_{\text{Compression}} = (P_2 - P_1) * A_{\text{Compression}}$

$$F_{\text{Compression}} = (3.96 - 5.22) * 804.25 = -1013.4N$$

Gas force $F_{\text{Gas}} = Area * P_{Gas}$

Considering the pressure of the nitrogen gas from 25-30bar. The gas force can be calculated as,

$$F_{\text{Gas}} = Area * P_{Gas} = 113.097mm^2 * 2.5N/mm^2 = 282.7425N$$

Stress in pressurized cylinder

Let an internal pressure *p* be exerted on the wall of a cylinder of thickness *t* and inside diameter *di*. The force tending to separate two halves of a unit length of the cylinder is *pdi*. This force is resisted by the tangential stress, also called the *hoop stress*, acting uniformly over the stressed area. We then have *pdi* = 2*tσt,* or

$$(\sigma_t)av = \frac{Pd_i}{2t} = \frac{3.96 * 32}{2 * 3} = 21.12N/mm2$$

This equation gives the *average* tangential stress and is valid regardless of the wall thickness. For a thin-walled vessel an approximation to the maximum tangential stress is

$$(\sigma_t)max = \frac{P(d_i + t)}{2t} = \frac{3.96(32 + 3)}{2 * 3} = 23.1N/mm^2$$

where *di* + *t* is the average diameter.

In a closed cylinder, the longitudinal stress *σl* exists because of the pressure upon the ends of the vessel. If we assume this stress is also distributed uniformly over the wall thickness, we can easily find it to be

$$\sigma_l = \frac{Pd_i}{4t} = \frac{3.96 * 32}{4 * 3} = 10.56N/mm^2$$

The calculation shows the maximum stress obtained in this analysis is less than the yield strength of the material selected. So, this indicates the design is safe.

Dimensions of damper

Table 3. 6: Dimensions of damper

Length of piston rod and valve assembly	193 mm
Length of bottom body	191 mm
Diameter of piston rod	12 mm
Diameter of bottom body	38 mm
Diameter of cylinder	32 mm
Diameter of piston valve	31 mm
Thickness of the valve shims	1 mm

Design of the valve

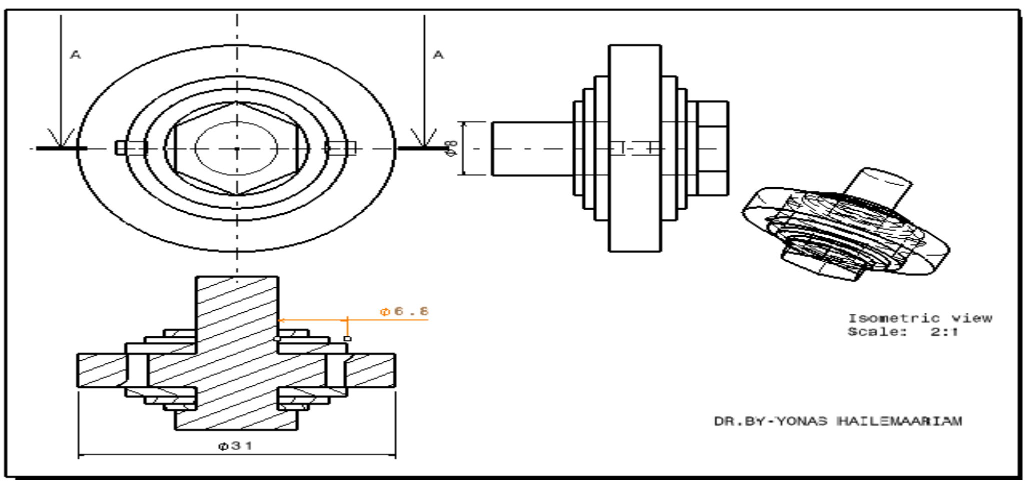


Figure 3. 23: 2D-Drawing of valve assembly part of shock absorber

Design of piston rod and valve assembly

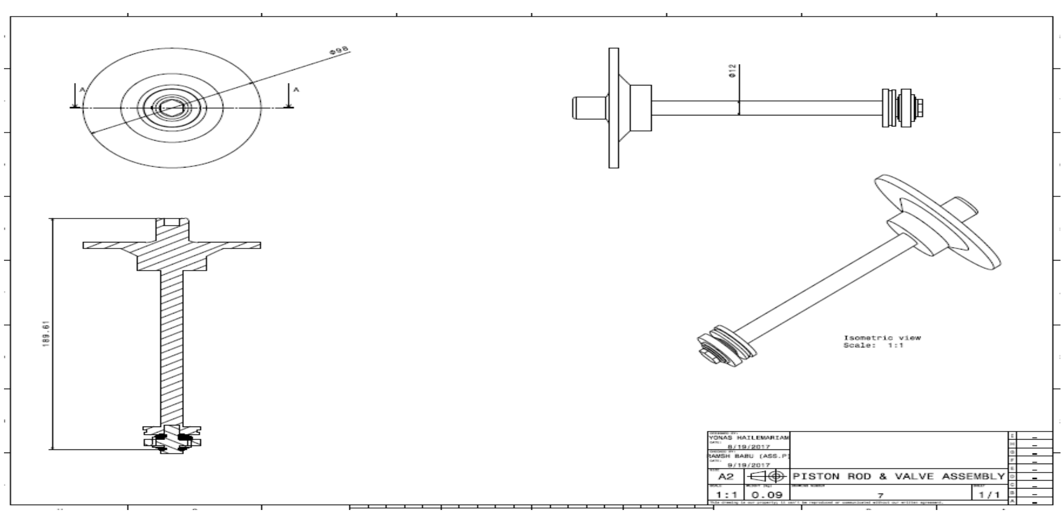


Figure 3. 24: 2D-Drawing of piston rod and valve assembly part of shock absorber

Design of bottom part

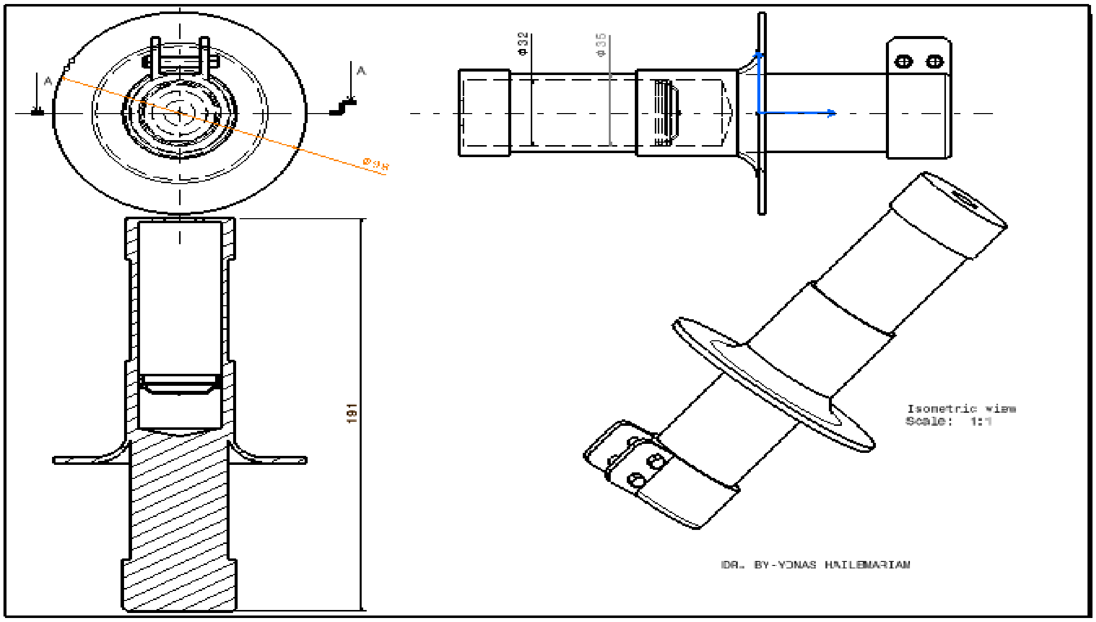


Figure 3. 25: 2D-Drawing of bottom part of shock absorber

Design of assembled shock absorber

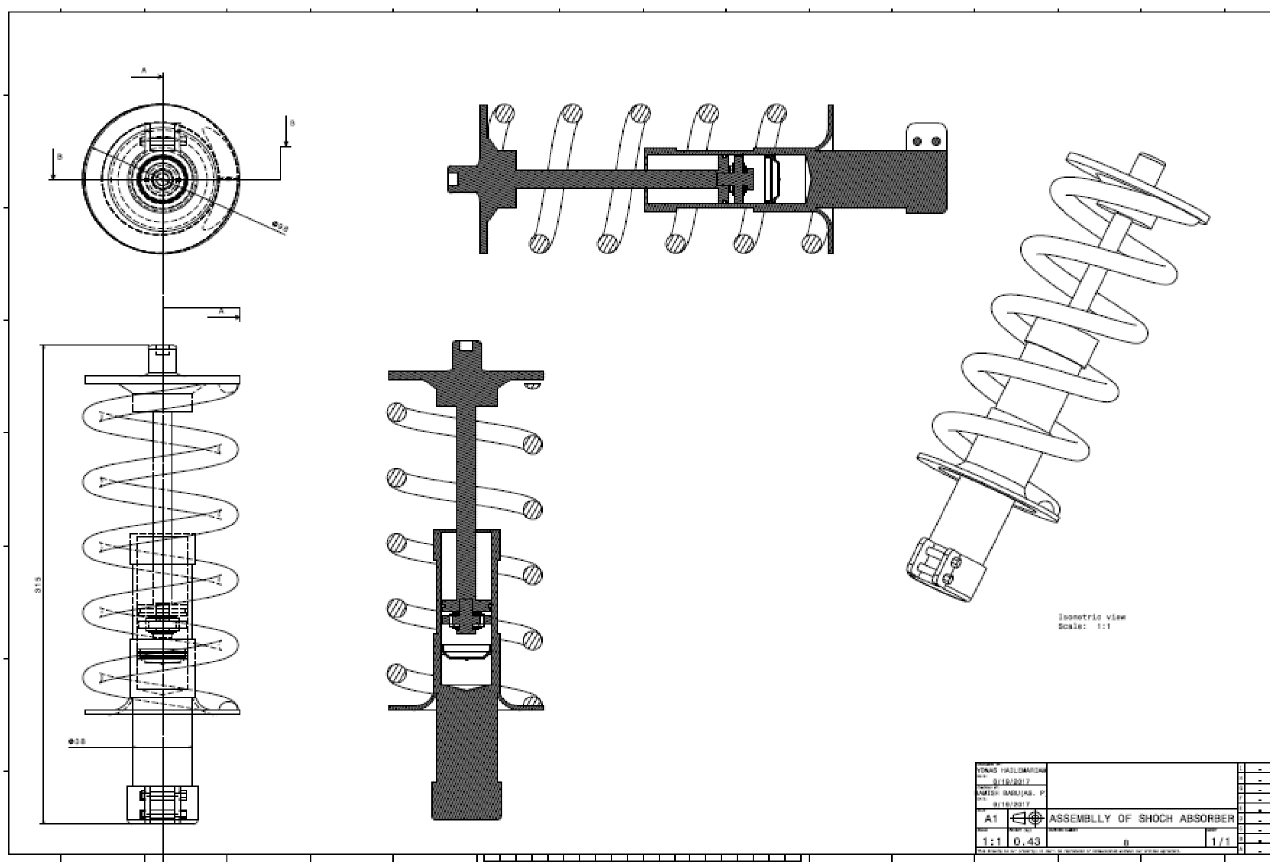


Figure 3. 26: 2D-Drawing of assembled shock absorber

3.4.2 Design of Knuckle

a) Calculation of Load

i) Axial Loads

There are two axial load acting on the steering knuckle such as tensile load and compressive load. The stress due to this load can be calculated using the following formula,

Tensile Load (Pt) = Tensile stress × Area

Compressive Load (Pc) = Compressive stress ×Area

ii) Inertia Load

This load acts on Steering Knuckle due to the inertia of the moving parts. The inertia load can be find out using the following formula,

Inertia Load (F_a) $= w^2 R[cos\theta + \frac{R}{L}\cos{(2\theta)}$ …..... [2]

Where:-w =load per unit length

R=radius of curvature

θ=slop of deflection

L= length

iii) Bending Load

This load acts on the steering knuckle due to the weight of the vehicle and this tends to bend the steering knuckle outward away from the centerline. Total inertia bending load is given by,

Bending Load (Fb) = $\frac{\rho A L^2 \sin{(\theta+\emptyset)}}{2} * N$……………….. [2].

Where: - ρ=density

A= cross sectional area

N= normal load

θ=slop of deflection

$\emptyset$= angle of twist

b) Calculation of Stresses

i) Stress Due to Axial Loads

The force of resistance per unit area, offered by a body against deformation is known as stress. This is given by,

Stress (**σ**) =P/A

ii) Stress due to Inertia bending force [2]

Inertia bending load sets up a stress which would be tensile on one side of the knuckle and compressive on another side and that these stress change sign each half revolution. The bending moment at any section 'X' *m* from the small end is given by,

$$M = \frac{X}{a}[1 - \frac{X^2}{L^2}]$$

The Stress is calculated by,

$$\sigma_b = \frac{M}{Z}$$

$$M = \frac{I}{2.5 * t}$$

$$I = 419 * t^4$$

C) Loading Condition on Knuckle

For the calculation of load acting on steering knuckle component, the required loading condition which is follows:

Table 3. 7: Loading condition of steering knuckle [2]

Link: "design and analysis of steering knuckle"

LOADING CONDITION	
Braking force	1.5mg (N)
Lateral force	1.5mg (N)
Steering force	45-50N

If the knuckle is design for the vehicle of 649*kg* weight .Thus braking force acting on it produced moment. This calculated as,

Braking force = 1.5*mg=1.5*162.25*9.81=2387.5N*

For calculation braking force acting on one wheel i.e.649/4=162.25*kg* for one wheel and perpendicular distance is 100*mm*considered.

Moment = Braking force × perpendicular distance=2387.5*100=238750N-mm

Dimensions of front knuckle

Table 3. 8: Dimensions of front knuckle

Vertical length of front knuckle	155 mm
Horizontal length of front knuckle	175 mm
Diameter of knuckle point	12 mm
Length of steering arm	100 mm
Length of knuckle hub	100 mm

Design of front knuckle

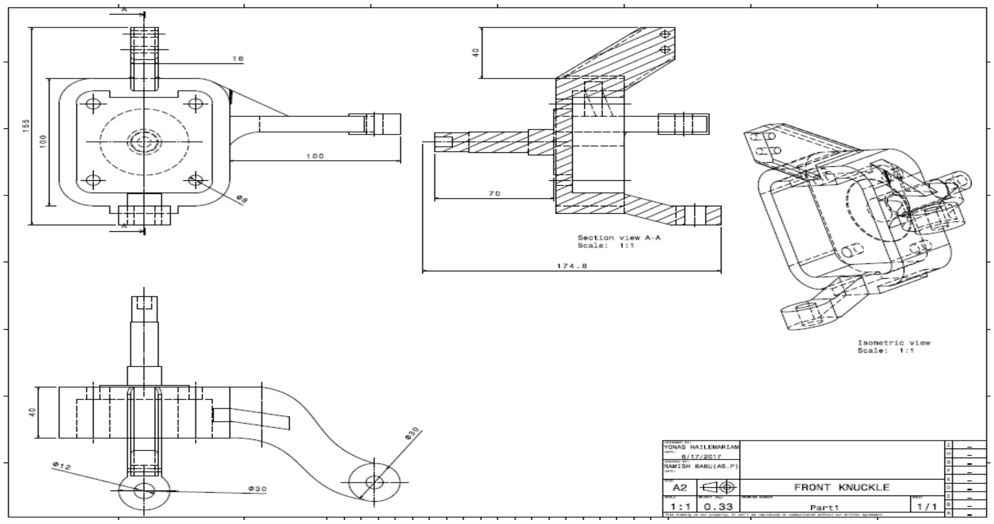


Figure 3. 27: 2D-Drawing of front knuckle

Dimensions of rear knuckle

Table 3. 9: Dimensions of rear knuckle

Vertical length of front knuckle	165 mm
Horizontal length of front knuckle	97 mm
Diameter of knuckle point	15 mm
Diameter of wheel hub	52 mm
Length of knuckle hub	100 mm

Design of rear knuckle

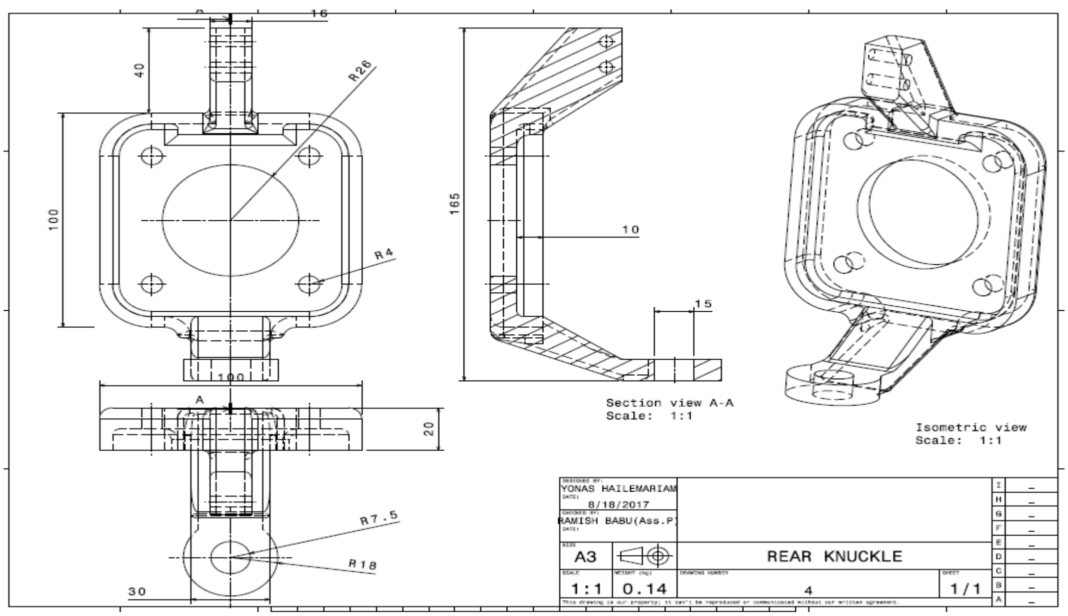


Figure 3. 28: 2D-drawing of rear knuckle

3.4.3 Design lower control arm

Dimension of lower control arm

Table 3. 10: Dimension of lower control arm

Overall length of front lower control arm	374 mm
Diameter of knuckle point	12 mm
Diameter of the arm end pipe	34 mm
Thickness of the arm	50 mm
Overall length of rear lower control arm	420 mm
Diameter of knuckle point	15 mm
Diameter of the arm end pipe	40 mm
Thickness (diameter) of the arm	35 mm

Design front lower control arm

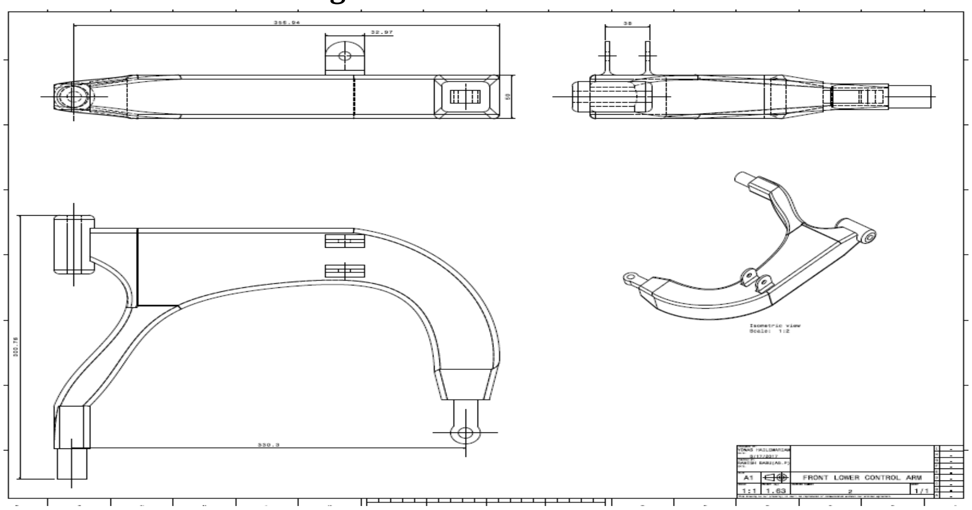


Figure 3. 29: 2D-Drawing of front lower control arm

Design rear lower control arm

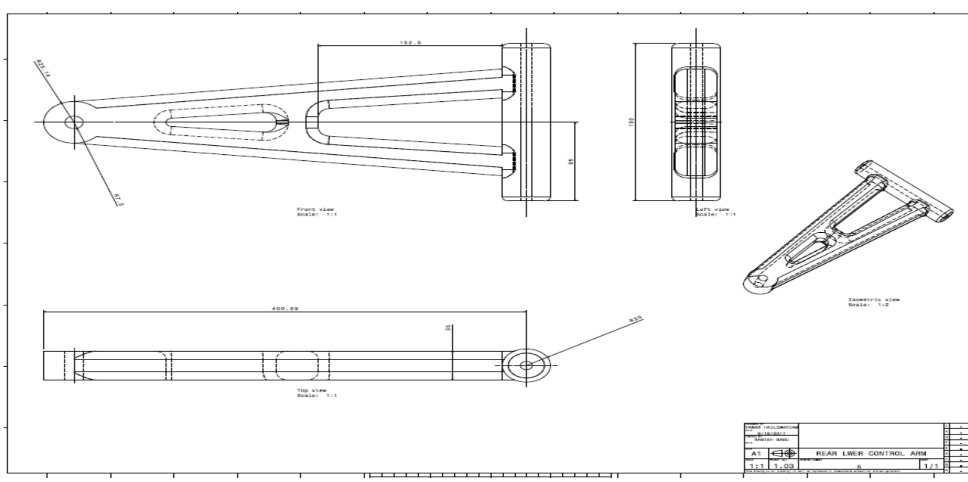


Figure 3. 30: 2D-Drawing of rear lower control arm

3.5 FINITE ELEMENT ANALYSIS

Finite Element Analysis (FEA) was first developed in 1943 by R. Courant, who utilized the Ritz method of numerical analysis and minimization of variation calculus to obtain approximate solutions to vibration systems. Shortly thereafter, a paper published in 1956 by M. J. Turner, R. W. Clough, H. C. Martin, and L. J. Topp established a broader definition of numerical analysis. The paper centered on the "stiffness and deflection of complex structures".

By the early 70's, FEA was limited to expensive mainframe computers generally owned by the aeronautics, automotive, defense, and nuclear industries. Since the rapid decline in the cost of computers and the phenomenal increase in computing power, FEA has been developed to an incredible precision. Present day supercomputers are now able to produce accurate results for all kinds of parameters [5].

FEA consists of a computer model of a material or design that is stressed and analyzed for specific results. It is used in new product design, and existing product refinement. A company is able to verify a proposed design will be able to perform to the client's specifications prior to manufacturing or construction. Modifying an existing product or structure is utilized to qualify the product or structure for a new service condition. In case of structural failure, FEA may be used to help determine the design modifications to meet the new condition [6].

FEA uses a complex system of points called nodes which make a grid called a mesh. The process for generating a mesh of nodes and elements consists of three general steps:

- CATIA offers a large number of mesh controls from which you can choose as needs dictate.
- Set the element attributes.
- Set mesh controls (optional).

It is not always necessary to set mesh controls because the default mesh controls are appropriate for many models. If no controls are specified, the program will use the default settings to produce a free mesh. Alternatively, you can use the Smart Size feature to produce a better quality free mesh. The various analysis results are shown in figures which are given below:

3.5.1FINITE ELEMENT ANALYSIS OF NEW SUSPENSION SYSTEM

I. **NEW SHOCK ABSORBER:**

- **Material Selection**

A great variety of spring materials are available to the designer, including plain carbon steels, alloy steels, and corrosion-resisting steels, as well as nonferrous materials such as phosphor bronze, spring brass, beryllium copper, and various nickel alloys. So, Chrome vanadium is selected for the spring and structural steel is selected for the top, cylinder and bottom part of the shock absorber in this design. Because, Chrome vanadium is the most popular alloy spring steel for conditions involving higher stresses than can be used with the high-carbon steels and for use where fatigue resistance and long endurance are needed. Also good for shock and impact loads. The mechanical, physical and chemical properties of Chrome vanadium materials are given below:

Table 3. 11: Mechanical and physcal property of Chrome vanadium

PROPERTIES		
QUALITY	**VALUE**	**UNIT**
density	7.8	g/cm^3
Tensile Strength, Ultimate	940	MPa
Tensile strength, yield	620	MPa
Young's Modulus	80	GPa
Poisson Ratio	0.29	---

Table 3. 12: Chemical composition of Chrome vanadium

Element	% Weight
Carbon, C	0.48-0.53
Chromium, Cr	0.80-1.10
Vanadium, V	0.15 min

Link: www.springhouseton.com/chrome-vanadium-spring

Boundary conditions:

- Material: chrome vanadium
- One end is fixed

- In the other end required load is given F=3183N and analyzed
- The analysis results are shown in figures which are given below:

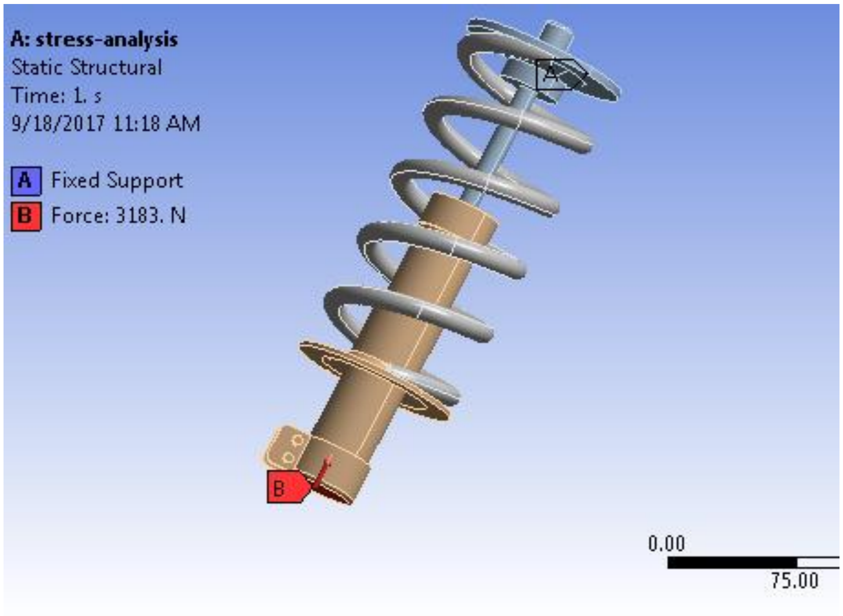


Figure 3. 31: Loads and boundary conditions of the shock absorber

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 225,731 numbers of Nodes and 140,017 numbers of Elements has been found.

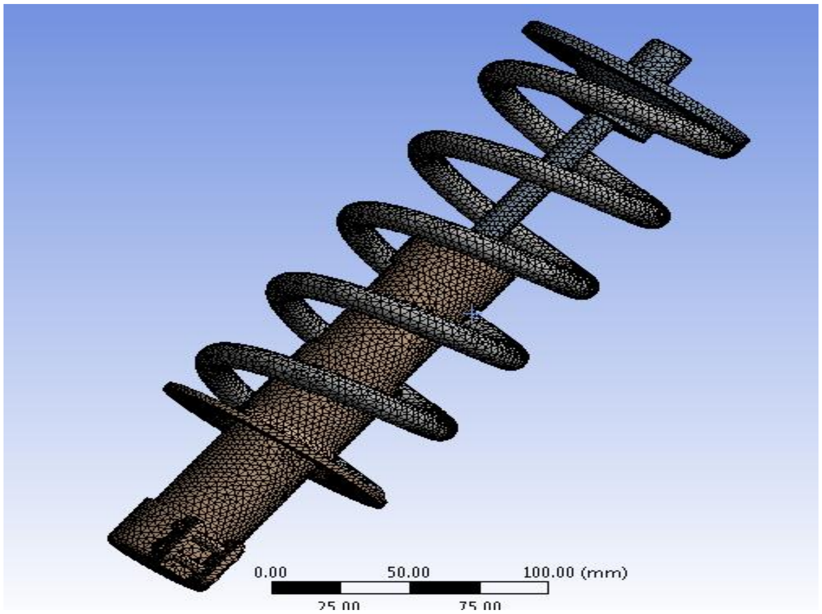
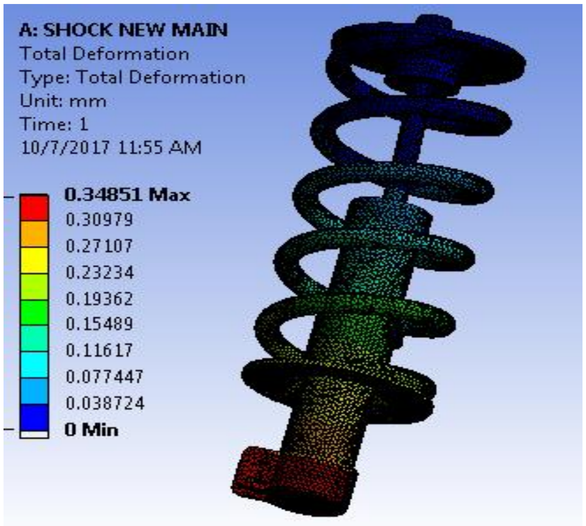
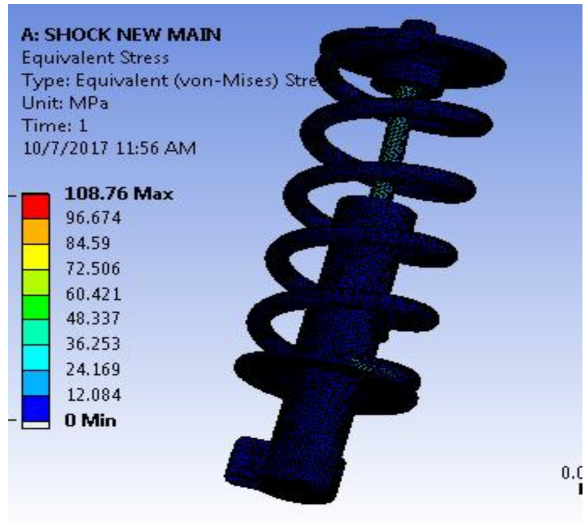


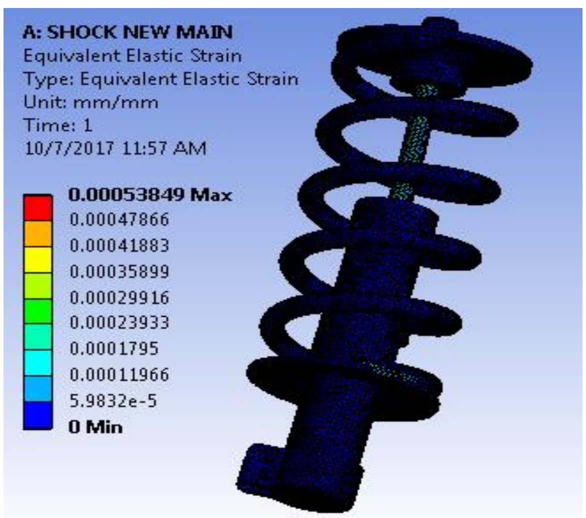
Figure 3. 32: Meshing of the shock absorber



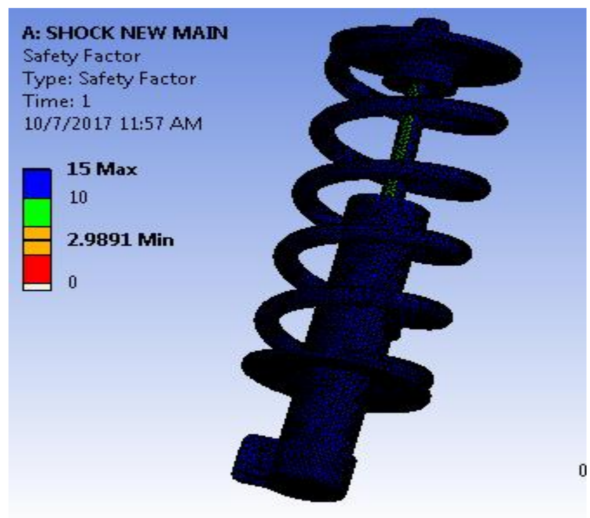
(A)



(B)



(C)



(D)

Figure 3. 33: Results of (A) total deformation diagram, (B) Equivalent (von-mises) stress diagram(C) Equivalent elastic strain, (D) Factor of safety diagram

Table 3. 13: Summarized results from shock absorber analysis

Analysis	Shock absorber results
Total deformation	0.3485mm
Equivalent (von-mises) stress	108.76Mpa
Equivalent elastic strain	0.0005385
Factor of safety	2.9891

II. **FRONT KNUCKLE:**

- **Material Selection**

The steering knuckle is made by different type of materials such as Cast iron, Mild steel and Aluminum. Cast Iron and Mild steel have a good strength but it contributes more weight to the vehicle. Forged steels are the most demanding materials for the steering knuckle in future. Due to the low weight of the materials fuel consumption can be stretch to the optimum level. Forge steel EN 47 is used as it offers a better strength and lightweight for the steering knuckle component. Forge Steel materials are generally used where high tensile strength and toughness is required thus the Forge Steel EN 47 materials are suited for the steering knuckle component. Due to this low weight of materials, it can decrease the fuel consumption and it has low density and sufficient yield strength. The mechanical, physical and chemical properties of Forge Steel materials are given below:

Table 3. 14: Mechanical and Physical property Forge steel EN 47

PROPERTIES		
QUALITY	**VALUE**	**UNIT**
density	7.7	g/cm^3
Tensile Strength, Ultimate	650	MPa
Tensile strength, yield	350	MPa
Young's Modulus	200	GPa
Poisson Ratio	0.3	---

Table 3. 15: Chemical composition Forge steel EN 47

Element	**% Weight**
Carbon, C	0.45-0.55
Manganese, Mn	0.50-0.80
Silicon, Si	0.05 Max
Chromium, Cr	0.80-1.20
Sulphur, S	0.05Max
Phosphorous, P	0.05max

Link: www.ssroundbar.com/EN47-alloy-steel

Boundary condition

- The knuckle hub is fixed
- Required forces are given at knuckle point and steering arm such as, opposite vertical forces, steering force and strut force also applied at the strut bracket arm.
- The analysis results are shown in figures which are given below

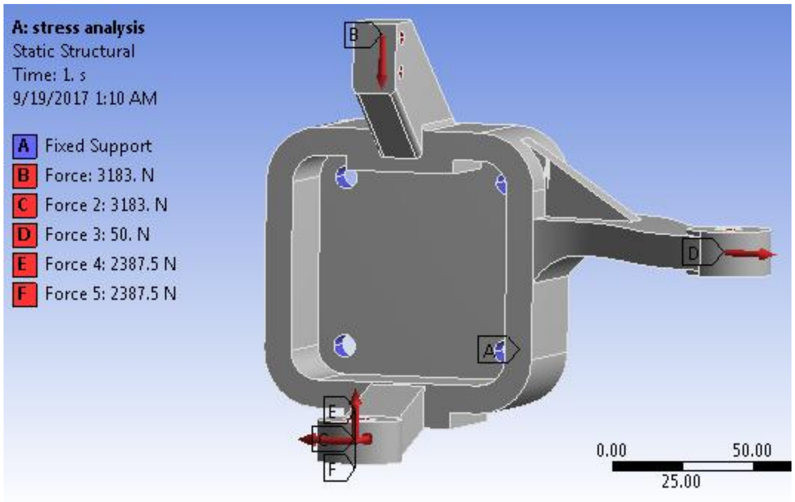


Figure 3. 34: Loads and boundary conditions of the front knuckle

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 162,502numbers of Nodes and 108,246 numbers of Elements has been found.

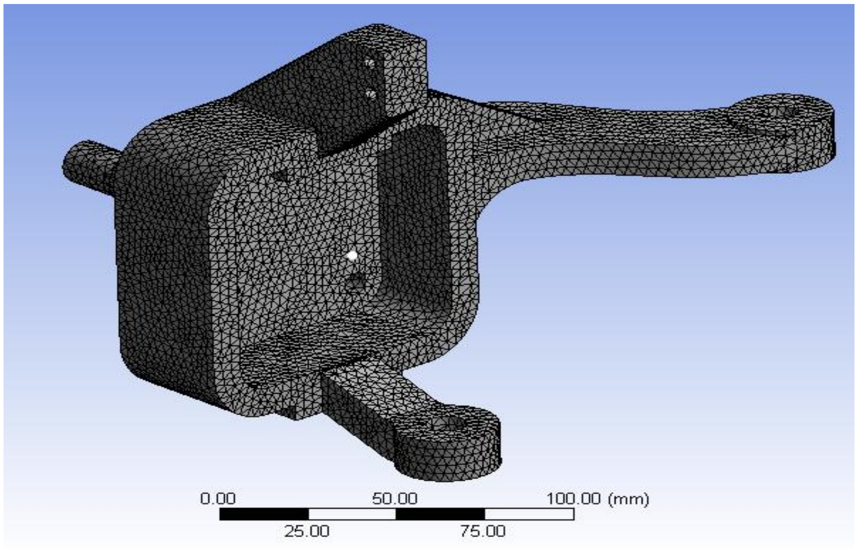
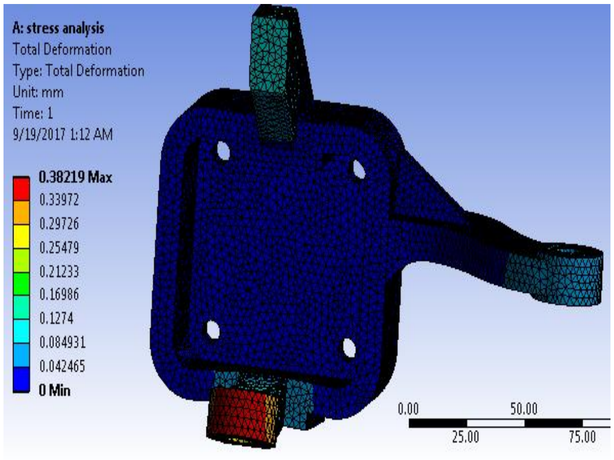
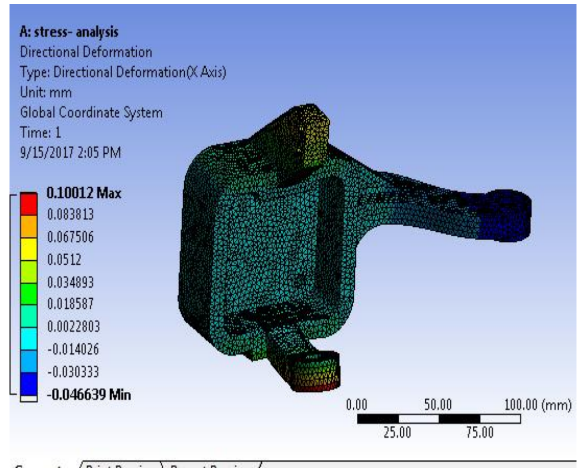


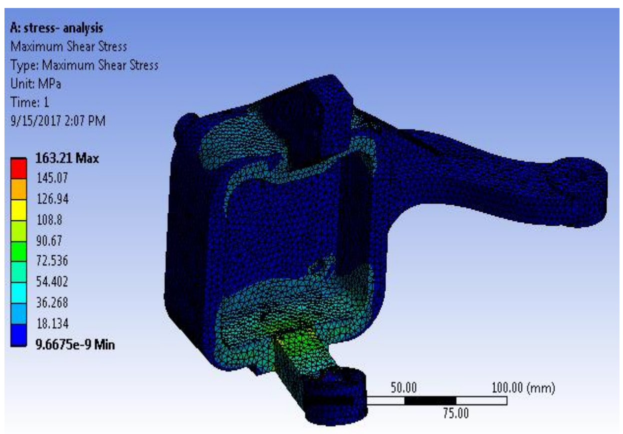
Figure 3. 35: Meshing of the front knuckle


A: stress analysis
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
9/19/2017 1:12 AM
0.38219 Max
0.33972
0.29726
0.25479
0.21233
0.16986
0.1274
0.084931
0.042465
0 Min
0.00
50.00
25.00
75.00

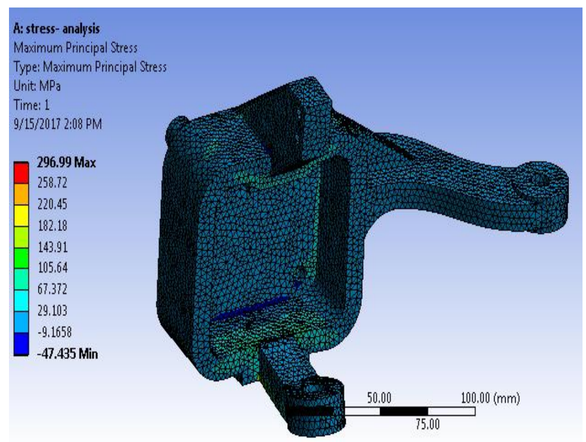
(A)


A: stress- analysis
Directional Deformation
Type: Directional Deformation(X Axis)
Unit: mm
Global Coordinate System
Time: 1
9/15/2017 2:05 PM
0.10012 Max
0.083813
0.067506
0.0512
0.034893
0.018587
0.0022803
-0.014026
-0.030333
-0.046639 Min
0.00
50.00
100.00 (mm)
25.00
75.00

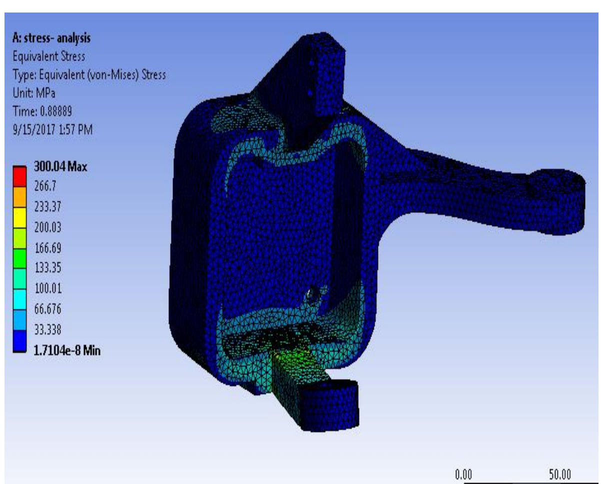
(B)


A: stress- analysis
Maximum Shear Stress
Type: Maximum Shear Stress
Unit: MPa
Time: 1
9/15/2017 2:07 PM
163.21 Max
145.07
126.94
108.8
90.67
72.536
54.402
36.268
18.134
9.6675e-9 Min
50.00
100.00 (mm)
75.00

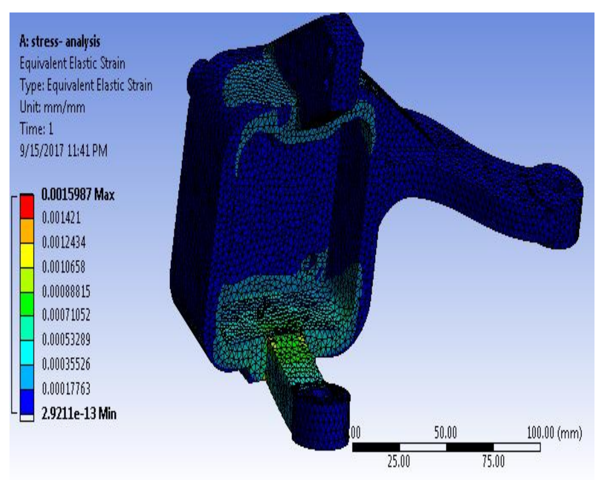
(C)


A: stress- analysis
Maximum Principal Stress
Type: Maximum Principal Stress
Unit: MPa
Time: 1
9/15/2017 2:08 PM
296.99 Max
258.72
220.45
182.18
143.91
105.64
67.372
29.103
-9.1658
-47.435 Min
50.00
100.00 (mm)
75.00

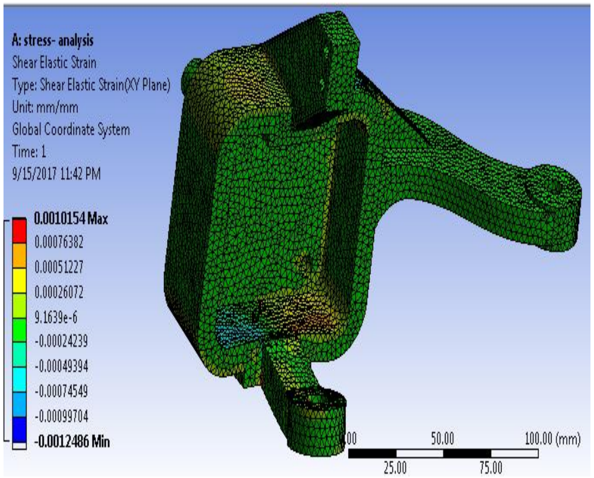
(D)

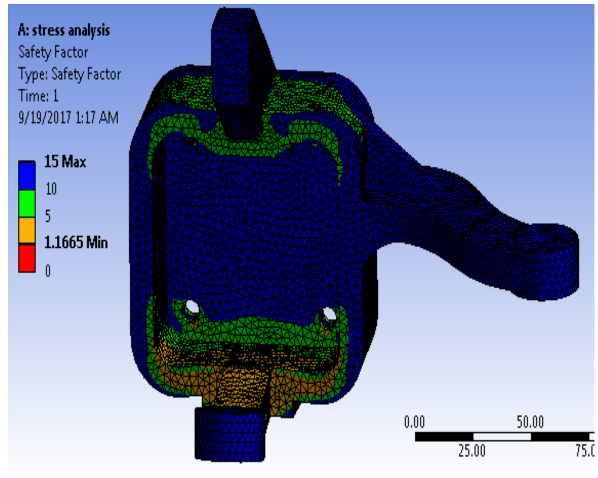

A: stress- analysis
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 0.88889
9/15/2017 1:57 PM
300.04 Max
266.7
233.37
200.03
166.69
133.35
100.01
66.676
33.338
1.7104e-8 Min
0.00
50.00

(E)


A: stress- analysis
Equivalent Elastic Strain
Type: Equivalent Elastic Strain
Unit: mm/mm
Time: 1
9/15/2017 11:41 PM
0.0015987 Max
0.001421
0.0012434
0.0010658
0.00088815
0.00071052
0.00053289
0.00035526
0.00017763
2.9211e-13 Min
50.00
100.00 (mm)
25.00
75.00

(F)





(G) (H)

Figure 3. 36: Results of (A) total deformation diagram, (B) directional deformation diagram,(C) maximum shear stress diagram,(D) maximum principal stress diagram (E) Equivalent (von-Mises) stress diagram(F)equivalent elastic strain diagram,(G) shear elastic strain diagram(H) safety factor

Table 3. 16: Summarized results from front knuckle

Analysis	**Front knuckle results**
Total deformation	0.3829
Directional deformation	0.10012
Maximum shear stress	163.21MPa
Maximum principal stress	296.99Ma
Equivalent (von-mises) stress	300MPa
Equivalent elastic strain	0.0015987
Shear elastic strain	0.0010154
Safety factor	1.1665

III. FRONT LOWER CONTROL ARM

- **Material Selection**

The material for the lower control arm is AISI 4130 low alloy steel. This contains chromium and molybdenum as strengthening agents. The steel has good strength and toughness, weldability and machinability. This low alloy steel finds many application as forgings in the aerospace and oil and gas industries as valve bodies and pumps as well as in automotive, agricultural and defense industries. The mechanical, physical and chemical properties of Forge Steel materials are given below:

Table 3. 17: Mechanical and Physical property of AISI 4130

PROPERTIES		
QUALITY	**VALUE**	**UNIT**
density	7.87	g/cm^3
Tensile Strength, Ultimate	670	MPa
Tensile strength, yield	460	MPa
Young's Modulus	205	GPa
Poisson Ratio	0.29	---

Table 3. 18: Chemical composition of AISI 4130

Element	% Weight
Carbon, C	0.28-0.33
Manganese, Mn	0.4-0.6
Silicon, Si	0.2-0.35
Chromium, Cr	0.80-1.10
Sulphur, S	0.04max
Phosphorous, P	0.035 max
Molybdenum, Mo	0.15-0.25

Link: www.azom.com/articleID=6742

Boundary conditions:

- By assuming as cantilever beam the ends of the arm which connects to frame is fixed.
- Required forces are given at knuckle point such as cornering and braking force
- In lower control arm addition to those forces, strut force is also acted

- The analysis results are shown in figures which are given below

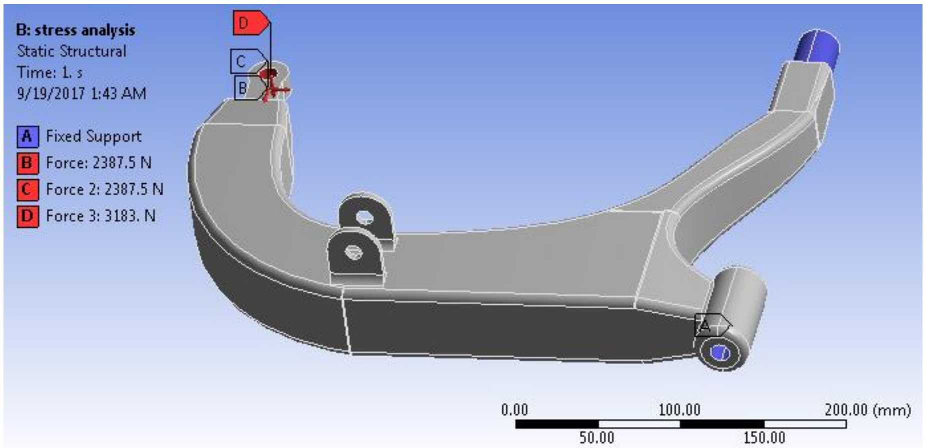


Figure 3. 37: Loads and boundary conditions of the front lower control arm

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 736,246numbers of Nodes and 525,028numbers of Elements has been found

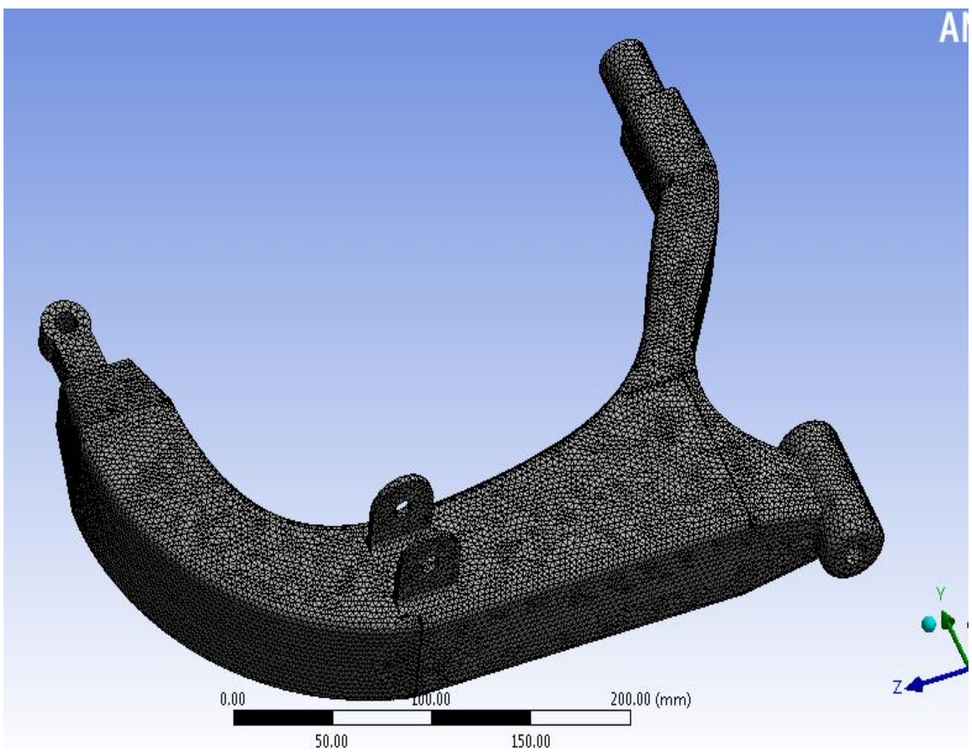
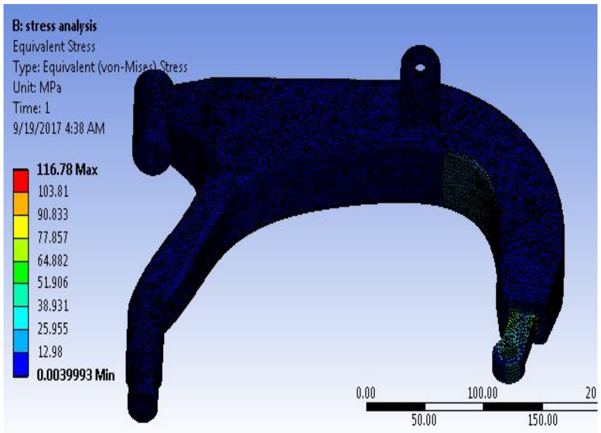
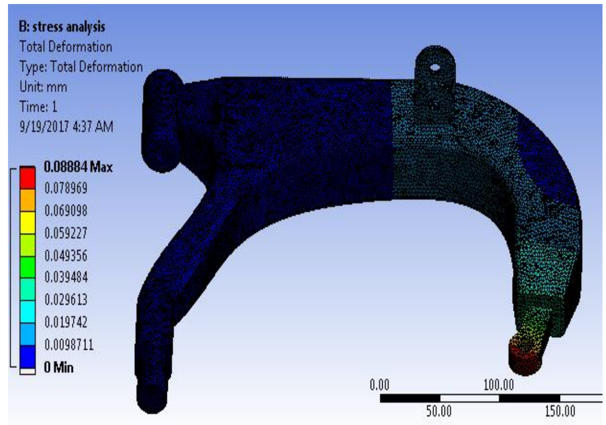


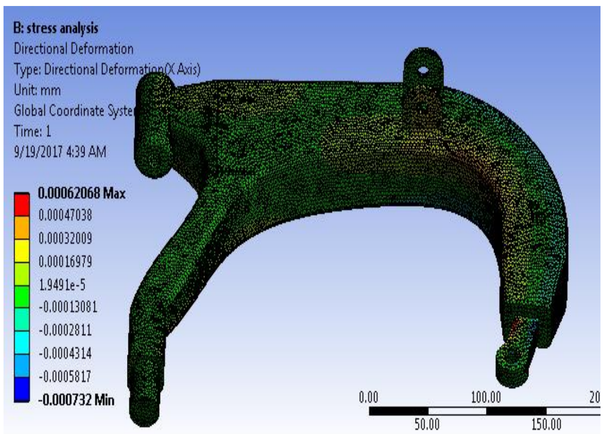
Figure 3. 38: Meshing of front lower control arm


B: stress analysis
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
9/19/2017 4:38 AM
116.78 Max
103.81
90.833
77.857
64.882
51.906
38.931
25.955
12.98
0.0039993 Min
0.00
100.00
50.00
150.00

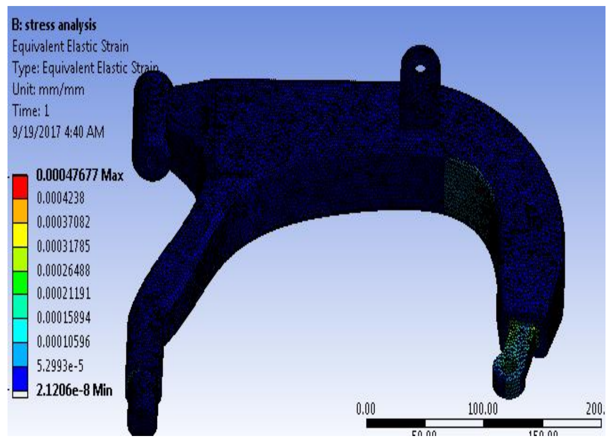
(A)


B: stress analysis
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
9/19/2017 4:37 AM
0.08884 Max
0.078969
0.069098
0.059227
0.049356
0.039484
0.029613
0.019742
0.0098711
0 Min
0.00
100.00
50.00
150.00

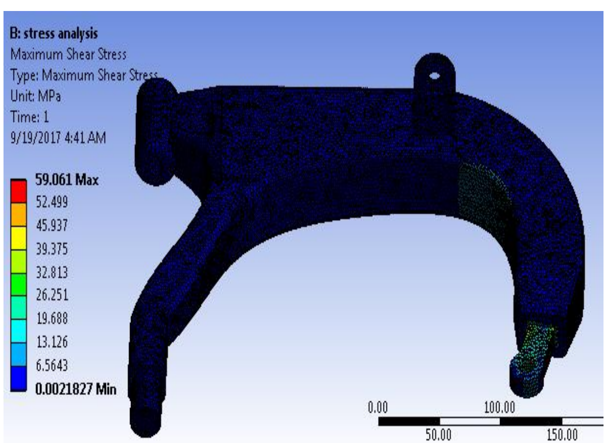
(B)


B: stress analysis
Directional Deformation
Type: Directional Deformation(X Axis)
Unit: mm
Time: 1
9/19/2017 4:39 AM
0.00062068 Max
0.00047038
0.00032009
0.00016979
1.9491e-5
-0.00013081
-0.0002811
-0.0004314
-0.0005817
-0.000732 Min
0.00
100.00
50.00
150.00

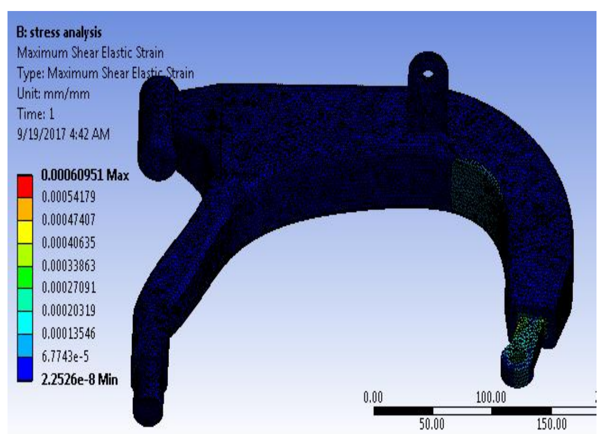
(C)


B: stress analysis
Equivalent Elastic Strain
Type: Equivalent Elastic Strain
Unit: mm/mm
Time: 1
9/19/2017 4:40 AM
0.00047677 Max
0.0004238
0.00037082
0.00031785
0.00026488
0.00021191
0.00015894
0.00010596
5.2993e-5
2.1206e-8 Min
0.00
100.00

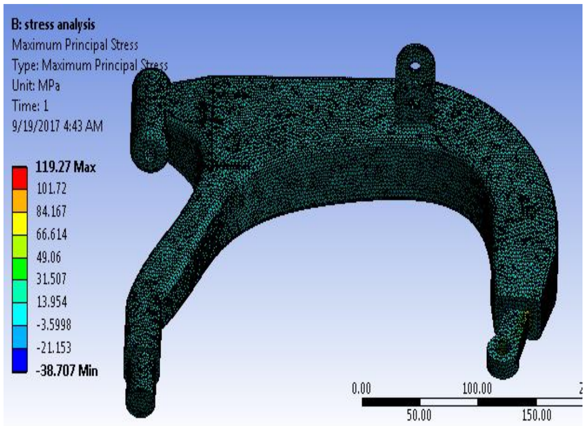
(D)

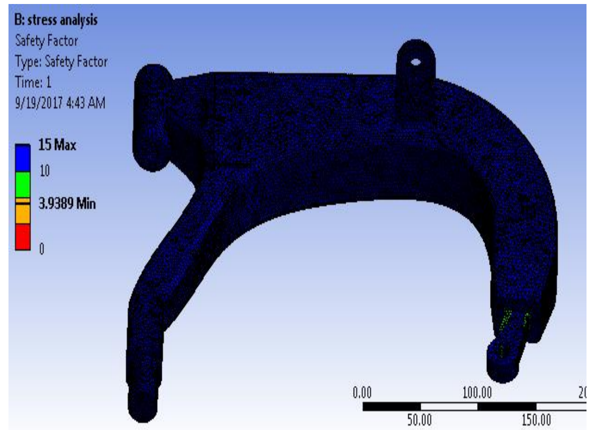

B: stress analysis
Maximum Shear Stress
Type: Maximum Shear Stress
Unit: MPa
Time: 1
9/19/2017 4:41 AM
59.061 Max
52.499
45.937
39.375
32.813
26.251
19.688
13.126
6.5643
0.0021827 Min
0.00
100.00
50.00
150.00

(E)


B: stress analysis
Maximum Shear Elastic Strain
Type: Maximum Shear Elastic Strain
Unit: mm/mm
Time: 1
9/19/2017 4:42 AM
0.00060951 Max
0.00054179
0.00047407
0.00040635
0.00033863
0.00027091
0.00020319
0.00013546
6.7743e-5
2.2526e-8 Min
0.00
100.00
50.00
150.00

(F)





(G) (H)

Figure 3. 39: Results of (A) equivalent (von-Mises) stress diagram (B) total deformation diagram (C) Directional deformation diagram (D) equivalent elastic strain diagram (E) Maximum shear stress diagram (F) maximum shear elastic strain diagram (G) maximum principal stress diagram (H) safety factor diagram

Table 3. 19: Summarized results from front lower control arm

Analysis	Front lower control arm results
Total deformation	0.08884mm
Directional deformation	0.00062
Maximum shear stress	59.061Mpa
Maximum principal stress	119.27Mpa
Equivalent (von-mises) stress	116.78Mpa
Equivalent elastic strain	0.000477
Shear elastic strain	0.00061
Safety factor	3.94

IV. **REAR KNUCKLE:**

- **Material is forged steel EN 47**

Boundary condition

- The knuckle hub is fixed
- Required forces are given at knuckle point and strut bracket arm and moment at the Ball Bearing Location / Stub Hole.
- The analysis results are shown in figures which are given below

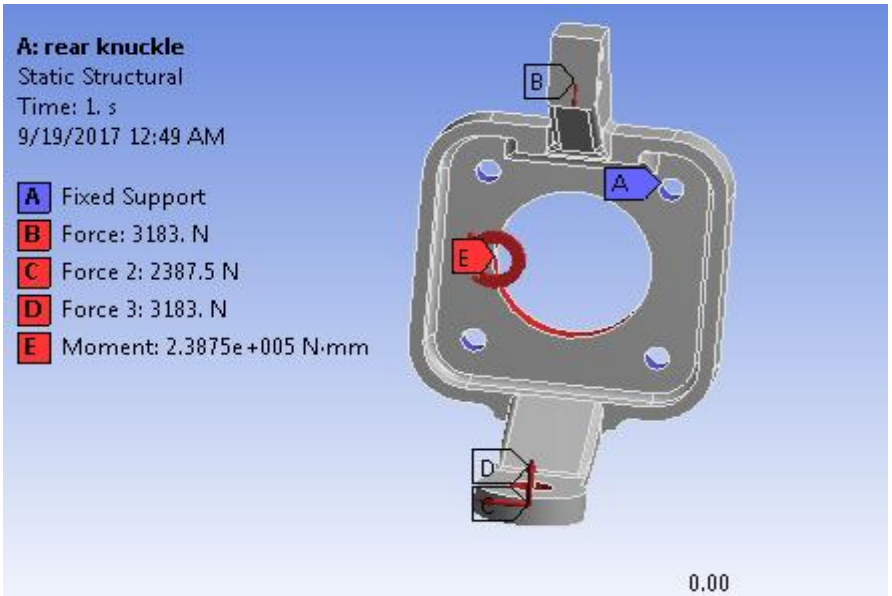


Figure 3. 40: Loads and boundary conditions of the rear knuckle

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 74,897numbers of Nodes and 48,342 numbers of Elements has been found.

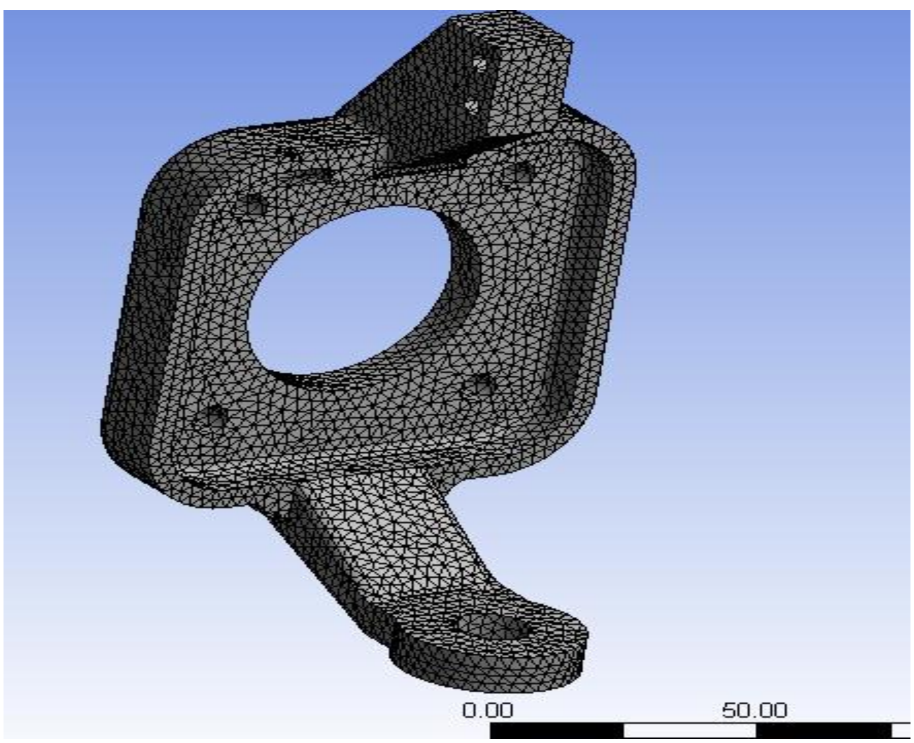
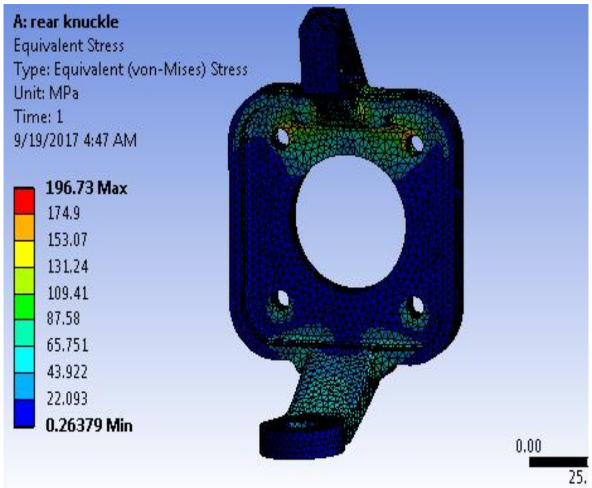
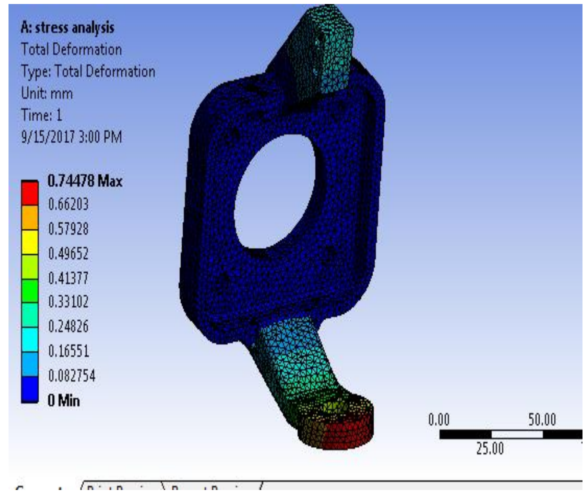


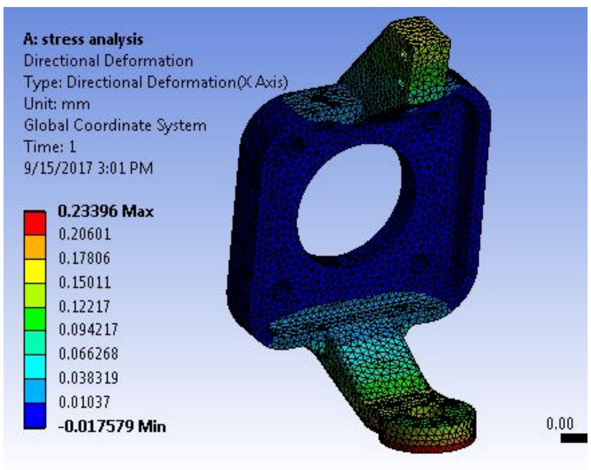
Figure 3. 41: Meshing of rear knuckle


A: rear knuckle
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
9/19/2017 4:47 AM
196.73 Max
174.9
153.07
131.24
109.41
87.58
65.751
43.922
22.093
0.26379 Min
0.00

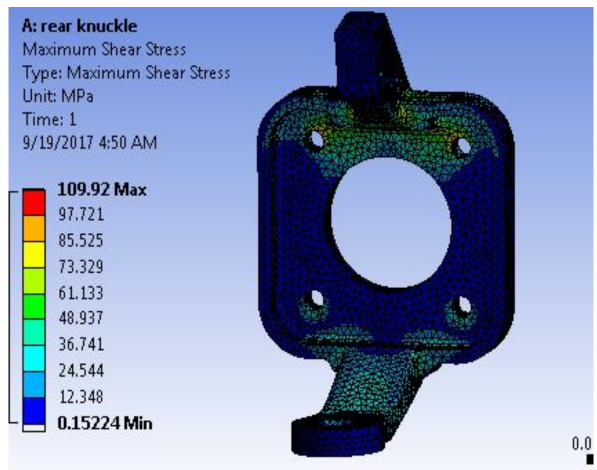
(A)


A: stress analysis
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
9/15/2017 3:00 PM
0.74478 Max
0.66203
0.57928
0.49652
0.41377
0.33102
0.24826
0.16551
0.082754
0 Min
0.00
50.00
25.00

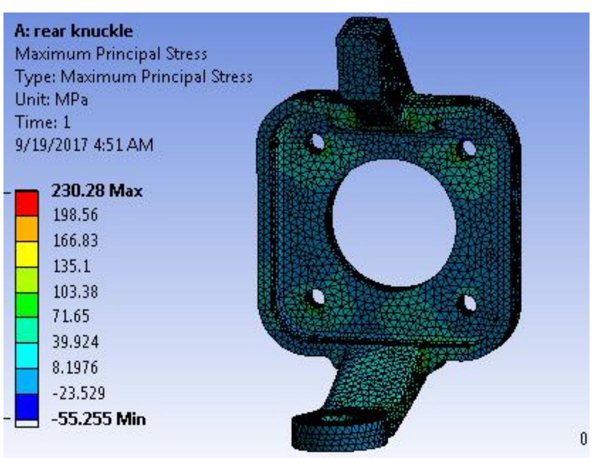
(B)


A: stress analysis
Directional Deformation
Type: Directional Deformation(X Axis)
Unit: mm
Global Coordinate System
Time: 1
9/15/2017 3:01 PM
0.23396 Max
0.20601
0.17806
0.15011
0.12217
0.094217
0.066268
0.038319
0.01037
-0.017579 Min
0.00

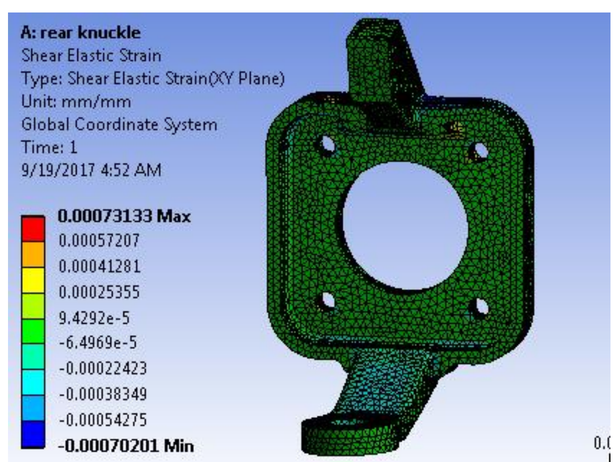
(C)


A: rear knuckle
Maximum Shear Stress
Type: Maximum Shear Stress
Unit: MPa
Time: 1
9/19/2017 4:50 AM
109.92 Max
97.721
85.525
73.329
61.133
48.937
36.741
24.544
12.348
0.15224 Min

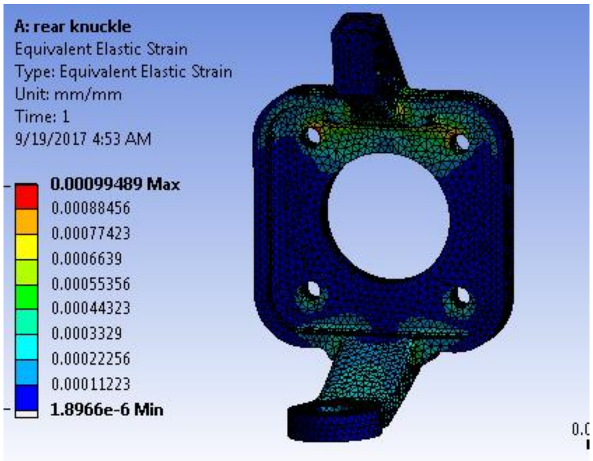
(D)


A: rear knuckle
Maximum Principal Stress
Type: Maximum Principal Stress
Unit: MPa
Time: 1
9/19/2017 4:51 AM
230.28 Max
198.56
166.83
135.1
103.38
71.65
39.924
8.1976
-23.529
-55.255 Min

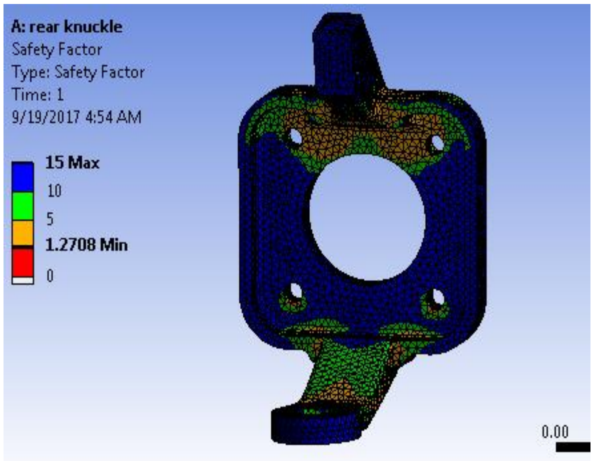
(E)


A: rear knuckle
Shear Elastic Strain
Type: Shear Elastic Strain(XY Plane)
Unit: mm/mm
Global Coordinate System
Time: 1
9/19/2017 4:52 AM
0.00073133 Max
0.00057207
0.00041281
0.00025355
9.4292e-5
-6.4969e-5
-0.00022423
-0.00038349
-0.00054275
-0.00070201 Min

(F)



(G)



(H)

Figure 3. 42: Results of (A) Equivalent (von-mises) stress diagram, (B)Total deformation diagram, (C) Directional deformation diagram, (D) Maximum shear stress diagram, (E) Maximum principal stress diagram, (F) Shear elastic strain diagram, (G) Equivalent elastic strain diagram, (H) Safety factor diagram

Table 3. 20: Summarized results from rear knuckle analysis

Analysis	Rear knuckle results
Total deformation	0.74478mm
Directional deformation	0.23396mm
Maximum shear stress	109.92Mpa
Maximum principal stress	230.28Mpa
Equivalent (von-mises) stress	196.73Mpa
Equivalent elastic strain	0.000995
Shear elastic strain	0.0007313
Safety factor	1.2708

V. **REAR LOWER CONTROL ARM:**

- **Material selected is low alloy steel AISI 4130**

Boundary conditions:

- By assuming as cantilever beam the ends of A-arm which connects to frame is fixed.
- Required forces are given at knuckle point such as wheel load and braking force
- In lower control arm addition to those forces, strut force is also acted
- The analysis results are shown in figures which are given below

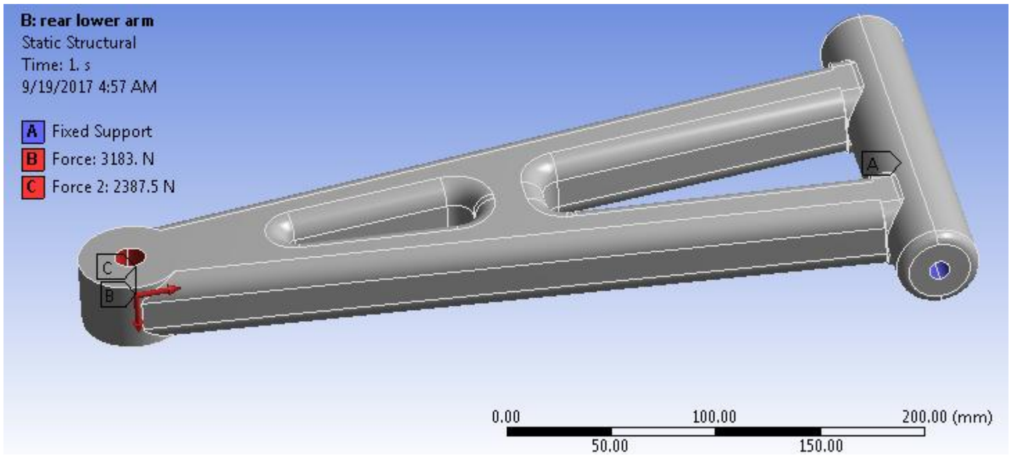


Figure 3. 43: Loads and boundary conditions of the rear lower control arm

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 471,828numbers of Nodes and 331,180numbers of Elements has been found

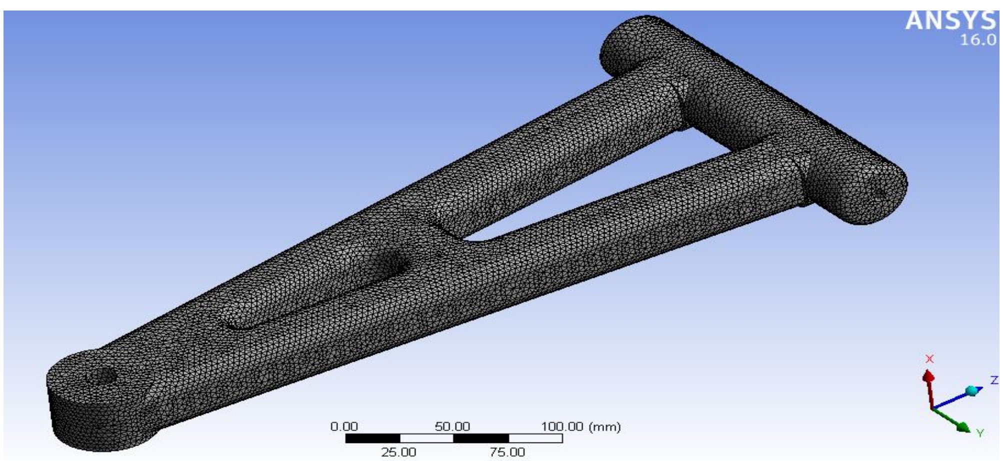
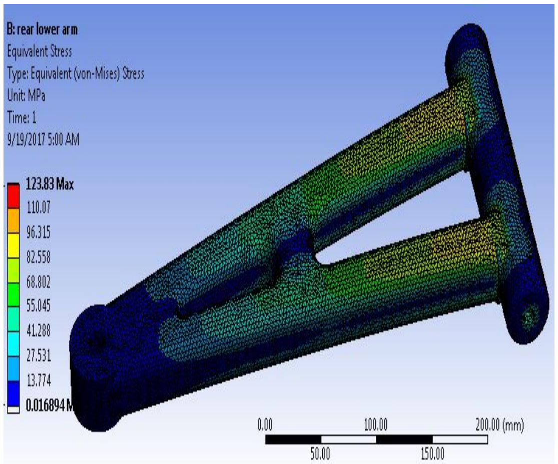
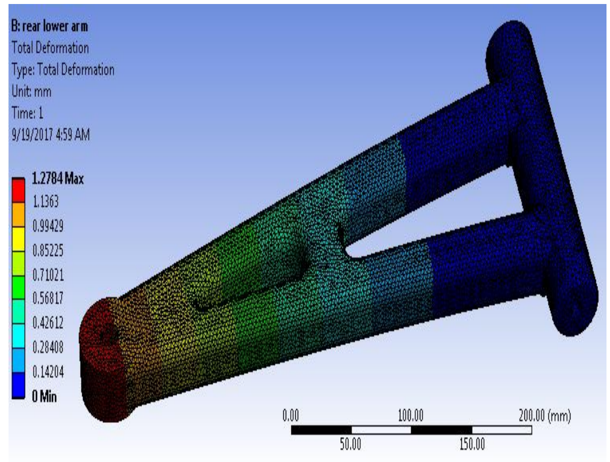


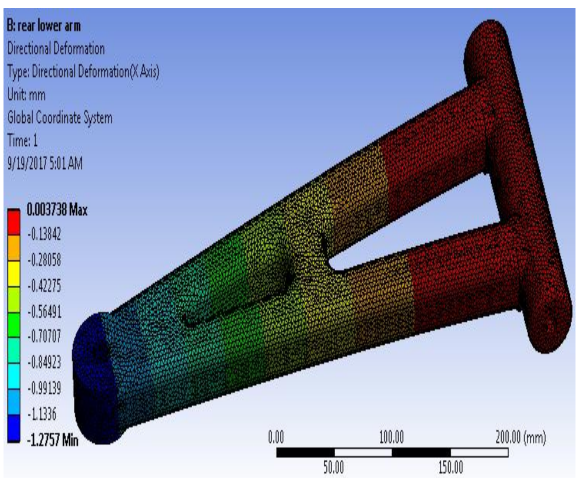
Figure 3. 44: Meshing of rear lower control arm


B: rear lower arm
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
9/19/2017 5:00 AM
123.83 Max
110.07
96.315
82.558
68.802
55.045
41.288
27.531
13.774
0.016894
0.00
100.00
200.00 (mm)
50.00
150.00

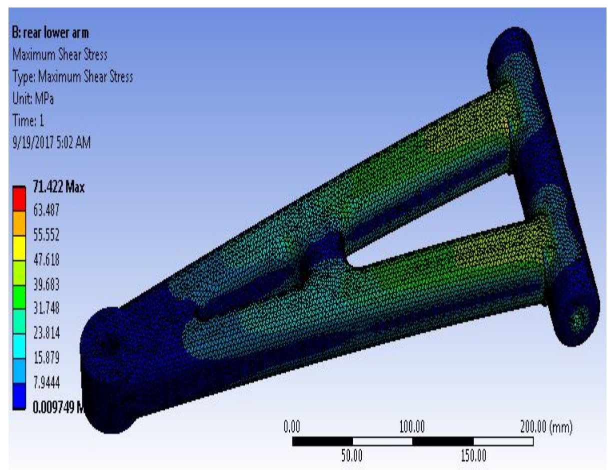
(A)


B: rear lower arm
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
9/19/2017 4:59 AM
1.2784 Max
1.1363
0.99429
0.85225
0.71021
0.56817
0.42612
0.28408
0.14204
0 Min
0.00
100.00
200.00 (mm)
50.00
150.00

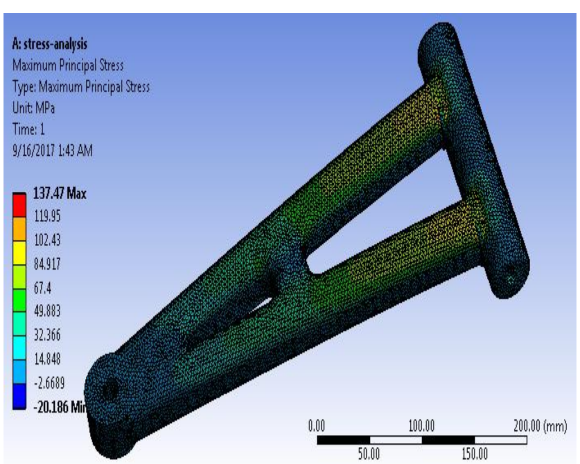
(B)


B: rear lower arm
Directional Deformation
Type: Directional Deformation(X Axis)
Unit: mm
Global Coordinate System
Time: 1
9/19/2017 5:01 AM
0.003738 Max
-0.13842
-0.28058
-0.42275
-0.56491
-0.70707
-0.84923
-0.99139
-1.1336
-1.2757 Min
0.00
100.00
200.00 (mm)
50.00
150.00

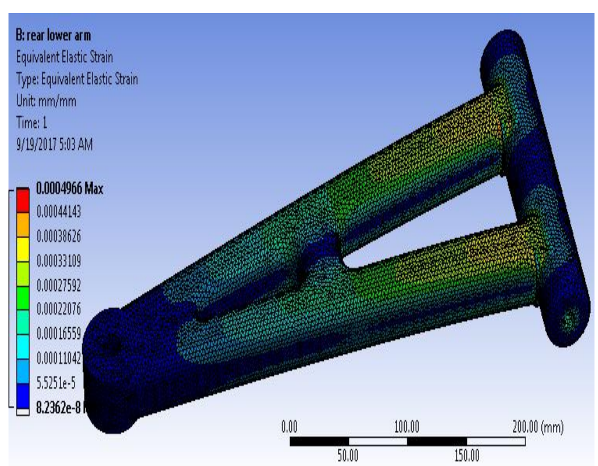
(C)


B: rear lower arm
Maximum Shear Stress
Type: Maximum Shear Stress
Unit: MPa
Time: 1
9/19/2017 5:02 AM
71.422 Max
63.487
55.552
47.618
39.683
31.748
23.814
15.879
7.9444
0.009749
0.00
100.00
200.00 (mm)
50.00
150.00

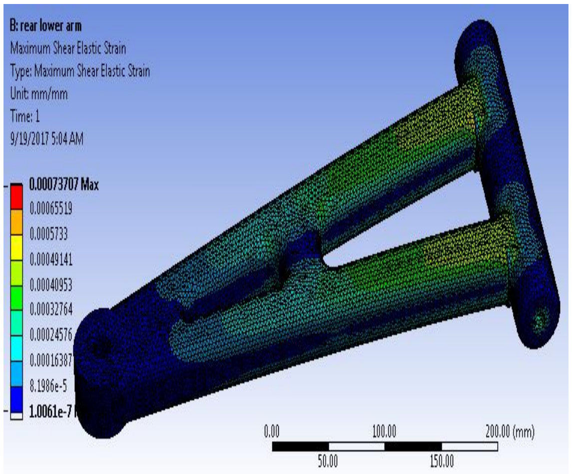
(D)


A: stress-analysis
Maximum Principal Stress
Type: Maximum Principal Stress
Unit: MPa
Time: 1
9/16/2017 1:43 AM
137.47 Max
119.95
102.43
84.917
67.4
49.883
32.366
14.848
-2.6689
-20.186
0.00
100.00
200.00 (mm)
50.00
150.00

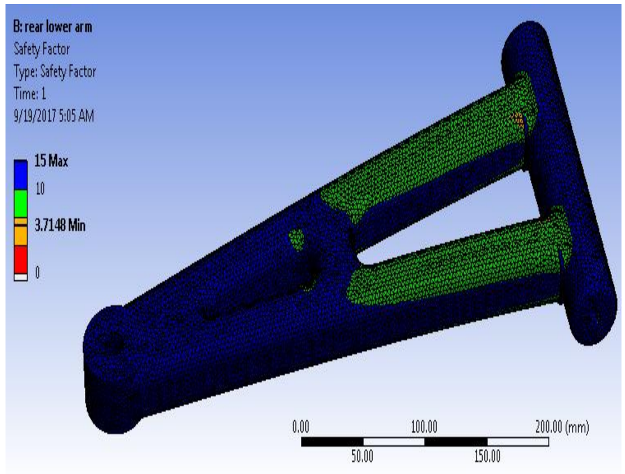
(E)


B: rear lower arm
Equivalent Elastic Strain
Type: Equivalent Elastic Strain
Unit: mm/mm
Time: 1
9/19/2017 5:03 AM
0.0004966 Max
0.00044143
0.00038626
0.00033109
0.00027592
0.00022076
0.00016559
0.00011042
5.5251e-5
8.2362e-8
0.00
100.00
200.00 (mm)
50.00
150.00

(F)



(G)



(H)

Figure 3. 45: Results of (A) Equivalent (von-mises) stress diagram, (B) Total deformation diagram, (C) Directional deformation diagram, (D) Maximum shear stress diagram, (E) Maximum principal stress, (F) Equivalent elastic strain diagram, (G) Maximum shear elastic strain diagram, (H) Safety factor diagram

Table 3. 21: Summarized results from rear lower control arm

Analysis	**Rear lower arm results**
Total deformation	1.2784mm
Directional deformation	0.003738mm
Maximum shear stress	71.42Mpa
Maximum principal stress	137.47Mpa
Equivalent (von-mises) stress	123.83Mpa
Equivalent elastic strain	0.0004966
Shear elastic strain	0.0007370
Safety factor	3.72

3.5.2 FINITE ELEMENT ANALYSIS OF EXISTING SUSPENSION SYSTEM

I. EXISTING SHOCK ABSORBER

Boundary conditions:

- Material: chrome vanadium
- One end is fixed
- In the other end required load is given F=3183N and analyzed
- The analysis results are shown in figures which are given below:

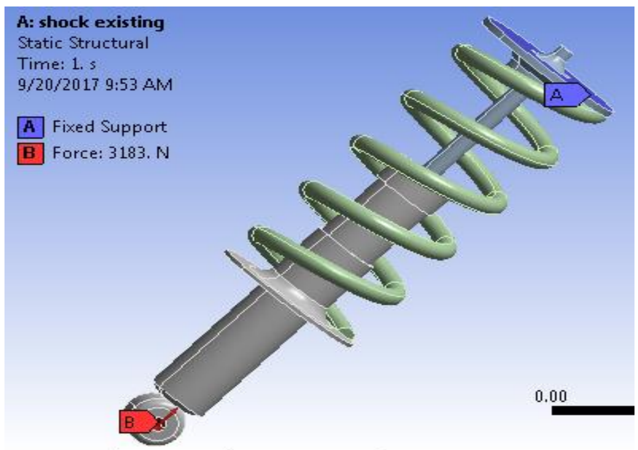


Figure 3. 46: Loads and boundary conditions of shock absorber

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 227,758 numbers of Nodes and 144,684 numbers of Elements has been found

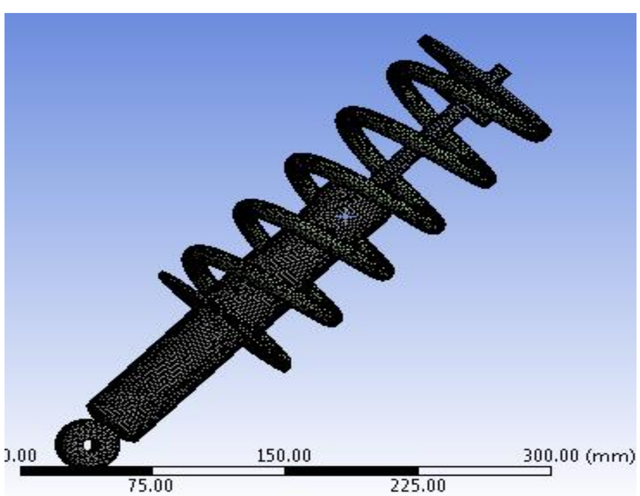
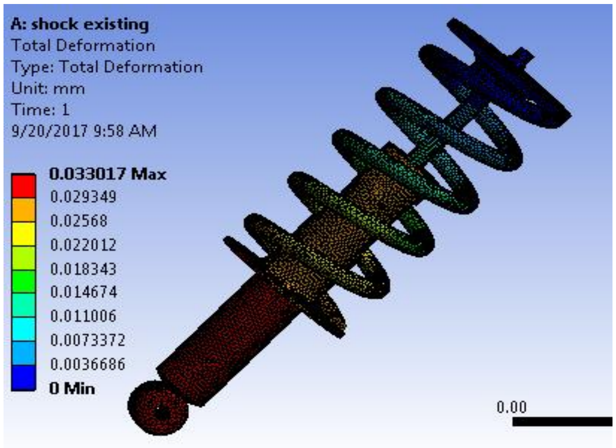
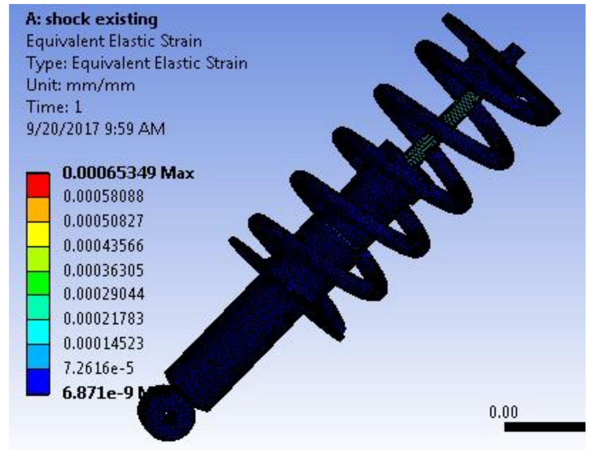


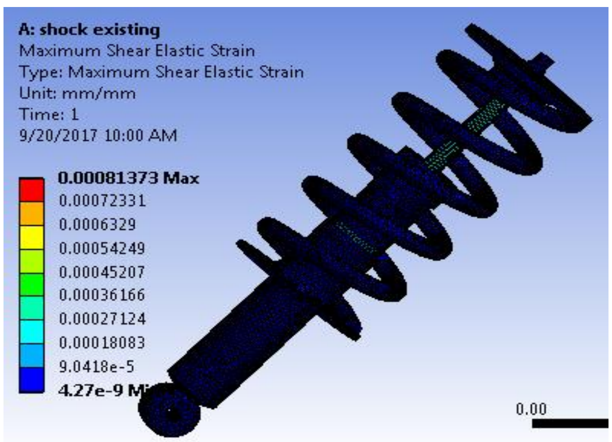
Figure 3. 47: Meshing of shock absorber



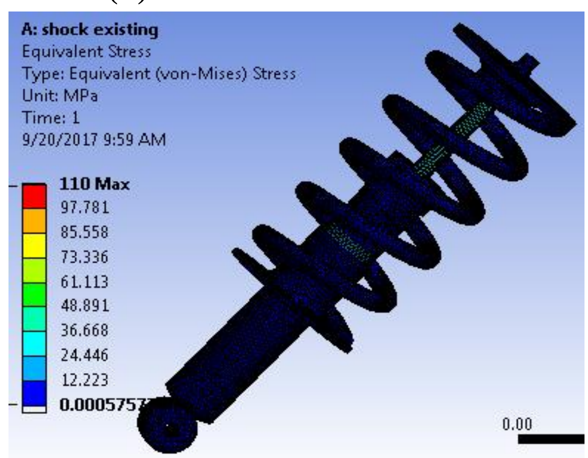
(A)



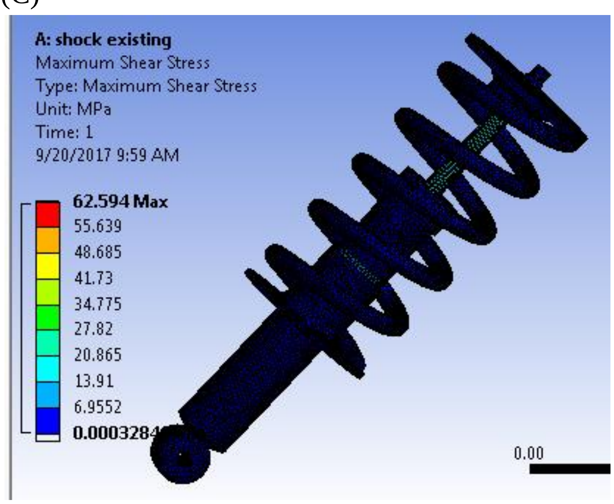
(B)



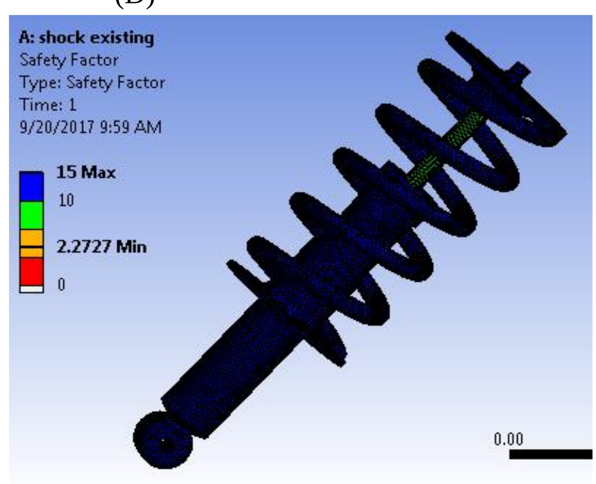
(C)



(D)



(E)



(F)

Figure 3. 48: Results of (A)total deformation, (B)equivalent elastic strain ,(C) shear elastic strain(D) Equivalent (von-Mises) stress (E) maximum shear tress (F) safety factor

Table 3. 22: Summarized results of existing shock absorber

Analysis	existing shock absorber results
Total deformation	0.033mm
Maximum shear stress	62.59Mpa
Equivalent (von-mises) stress	110Mpa
Equivalent elastic strain	0.0006535
Shear elastic strain	0.00081373
Safety factor	2.2727

II. EXISTING FRONT KNUCKLE

Boundary condition

- Material: Forge steel EN 47
- The knuckle hub is fixed
- Required forces are given at knuckle point and steering arm such as, vertical forces, brake force and steering force.
- The analysis results are shown in figures which are given below

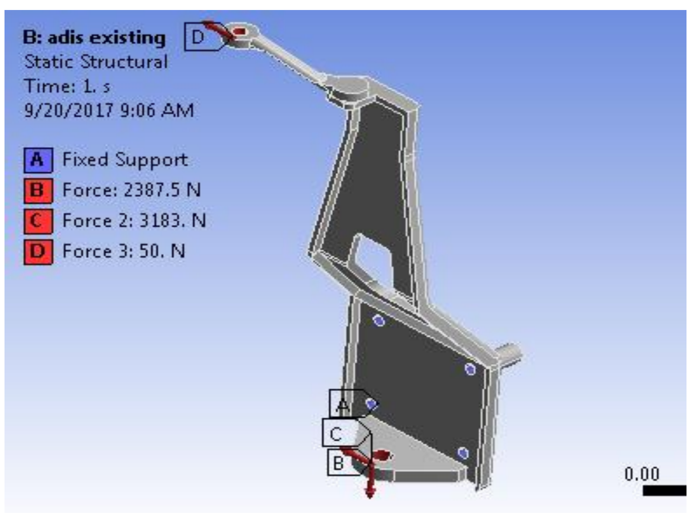


Figure 3. 49: Loads and boundary conditions of existing front knuckle

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 126434numbers of Nodes and 78756numbers of Elements has been found.

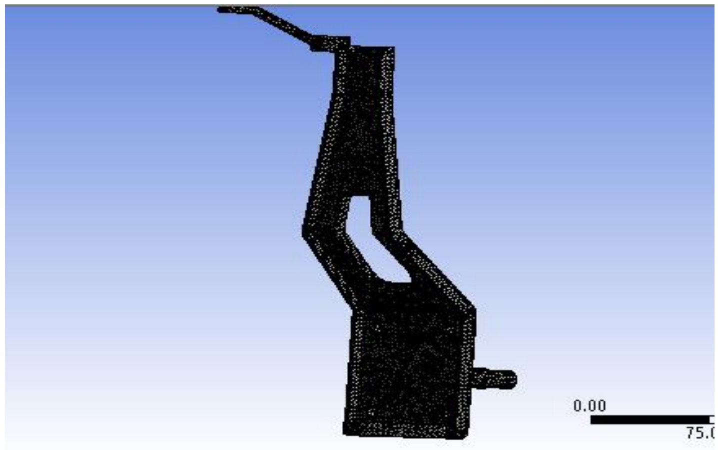
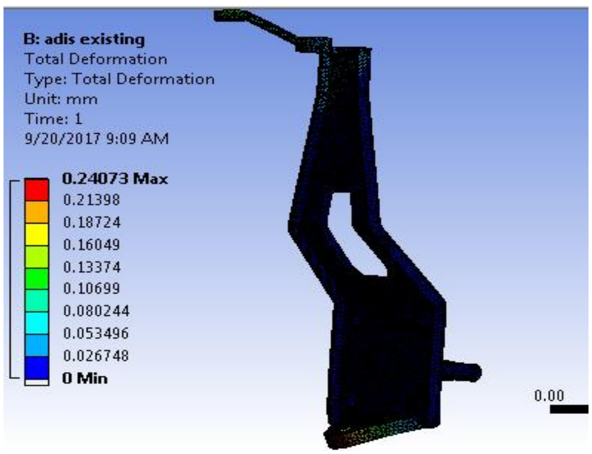
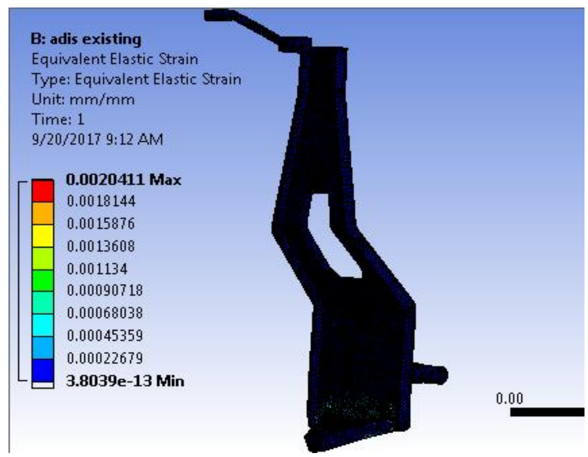


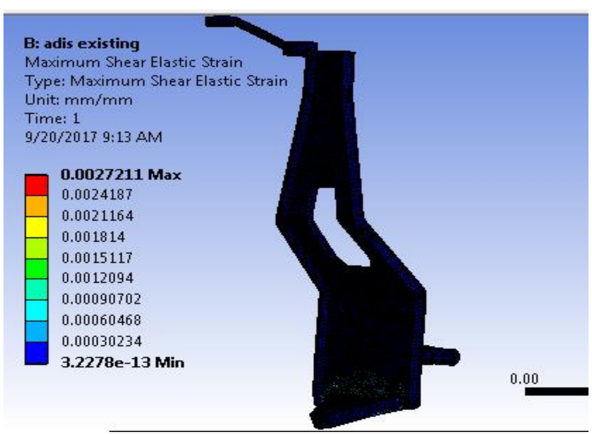
Figure 3. 50: Meshing of existing front knuckle



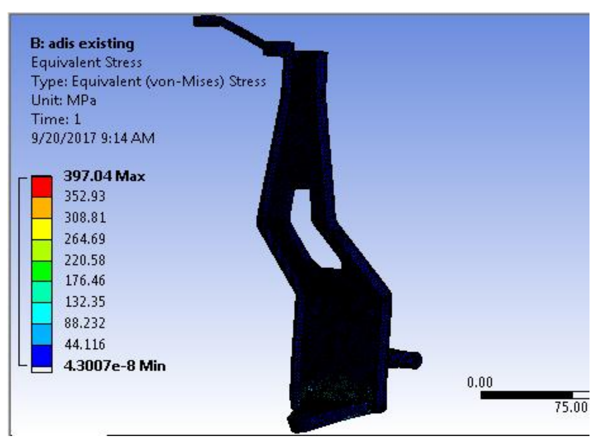
(A)



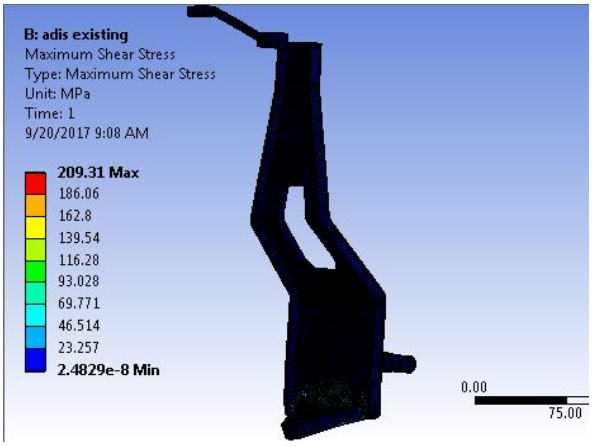
(B)



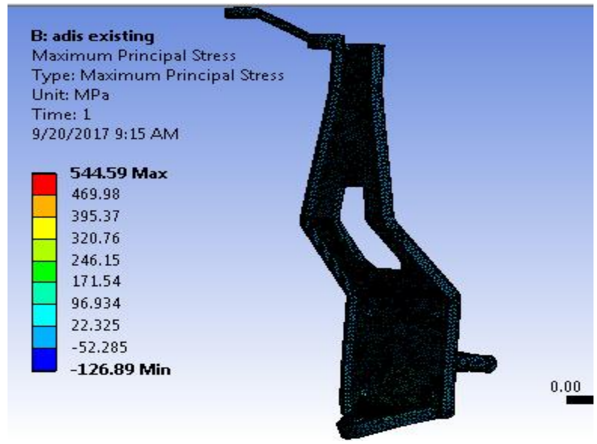
(C)



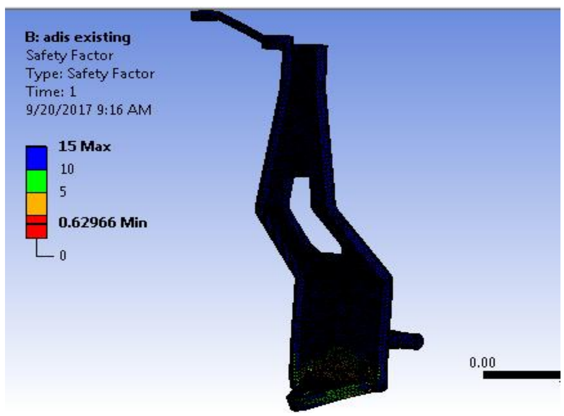
(D)



(E)



(F)



(G)

Figure 3. 51: Results of (A)total deformation, (B)equivalent elastic strain ,(C) shear elastic strain(D) Equivalent (von-Mises) stress (E) maximum shear tress (F)maximum principal stress, (G) safety factor

Table 3. 23: Summarized results of existing front knuckle

Analysis	Existing Front knuckle results
Total deformation	0.2407mm
Maximum shear stress	209.31Mpa
Maximum principal stress	533.39Mpa
Equivalent (von-mises) stress	397.04Mpa
Equivalent elastic strain	0.002041
Shear elastic strain	0.002721
Safety factor	0.6142

III. EXISTING FRONT LOWER CONTROL ARM

Boundary conditions:

- Material: low alloy steel AISI 4130
- By assuming as cantilever beam the ends of the arm which connects to frame is fixed.
- Required forces are given at knuckle point
- The analysis results are shown in figures which are given below

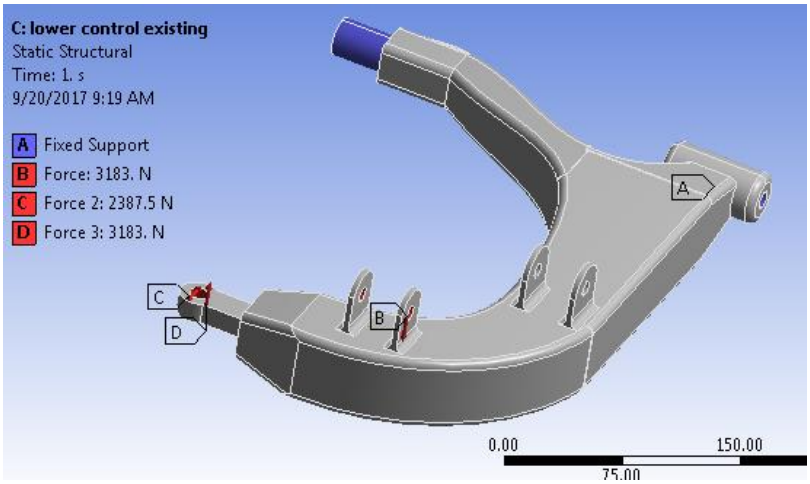


Figure 3. 52: Loads and boundary conditions of existing lower control arm

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 743058numbers of Nodes and 529148numbers of Elements has been found.

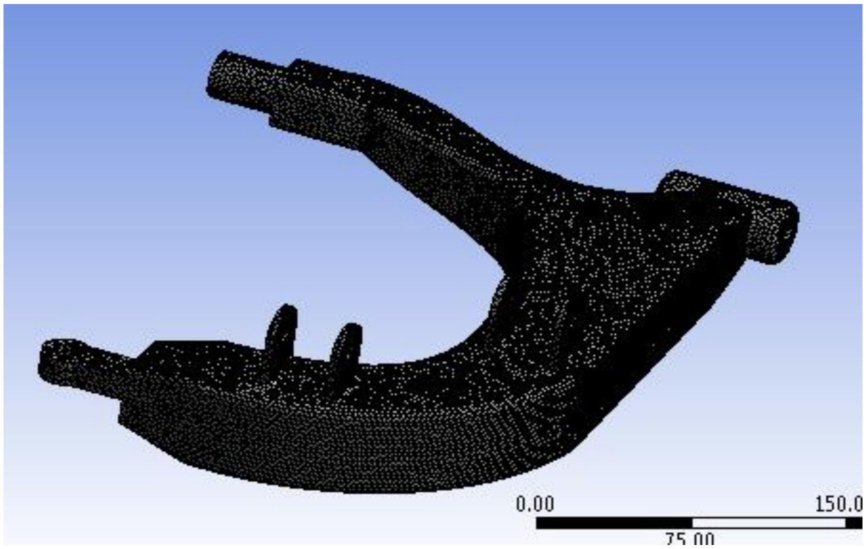
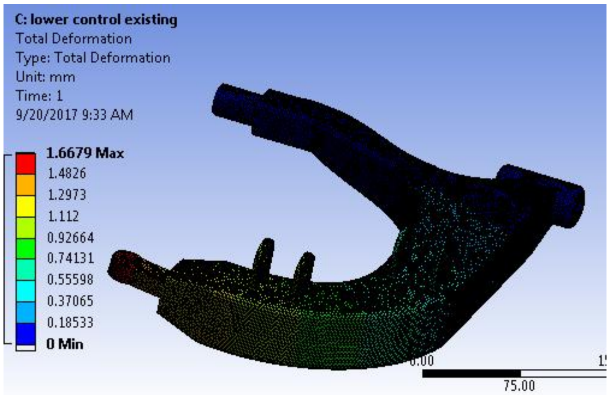
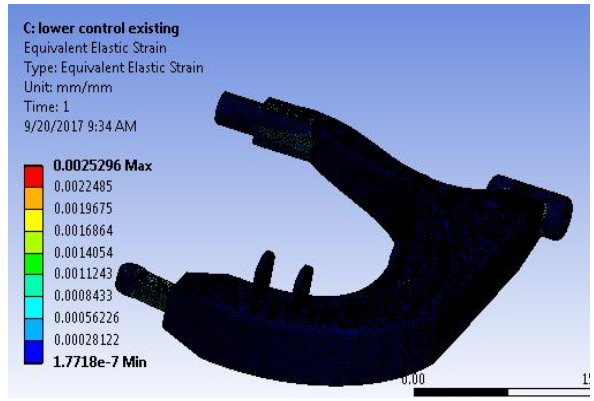


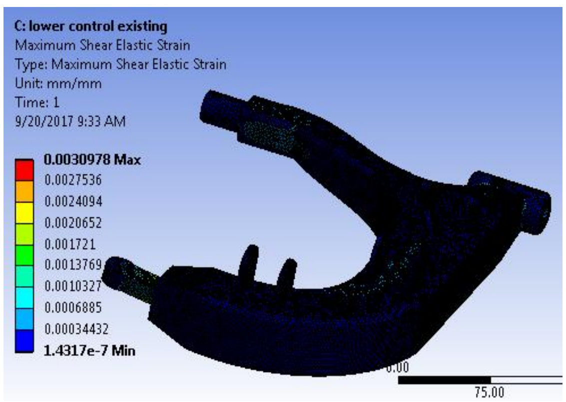
Figure 3. 53: Meshing of existing lower control arm


C: lower control existing
Total Deformation
Type: Total Deformation
Unit: mm
Time: 1
9/20/2017 9:33 AM
1.6679 Max
1.4826
1.2973
1.112
0.92664
0.74131
0.55598
0.37065
0.18533
0 Min
0.00
75.00

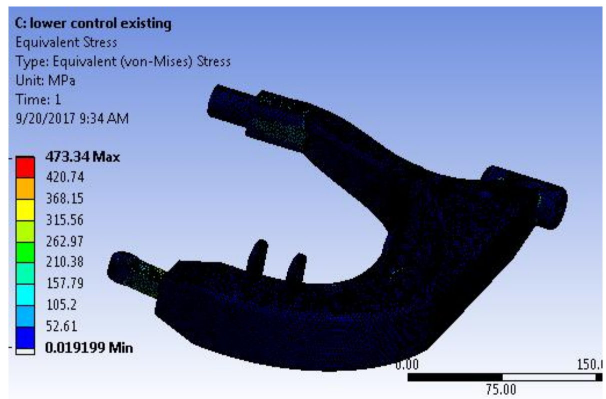
(A)


C: lower control existing
Equivalent Elastic Strain
Type: Equivalent Elastic Strain
Unit: mm/mm
Time: 1
9/20/2017 9:34 AM
0.0025296 Max
0.0022485
0.0019675
0.0016864
0.0014054
0.0011243
0.0008433
0.00056226
0.00028122
1.7718e-7 Min
0.00

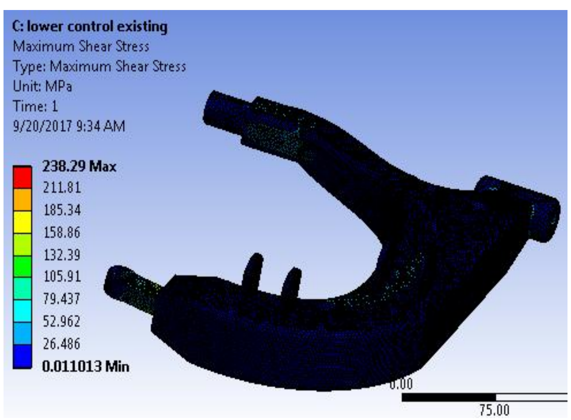
(B)


C: lower control existing
Maximum Shear Elastic Strain
Type: Maximum Shear Elastic Strain
Unit: mm/mm
Time: 1
9/20/2017 9:33 AM
0.0030978 Max
0.0027536
0.0024094
0.0020652
0.001721
0.0013769
0.0010327
0.0006885
0.00034432
1.4317e-7 Min
0.00
75.00

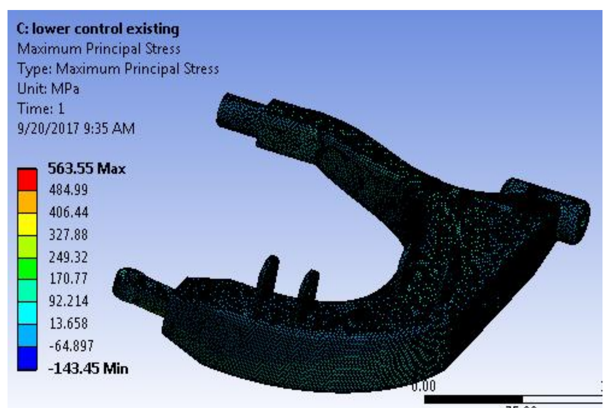
(C)


C: lower control existing
Equivalent Stress
Type: Equivalent (von-Mises) Stress
Unit: MPa
Time: 1
9/20/2017 9:34 AM
473.34 Max
420.74
368.15
315.56
262.97
210.38
157.79
105.2
52.61
0.019199 Min
0.00
75.00

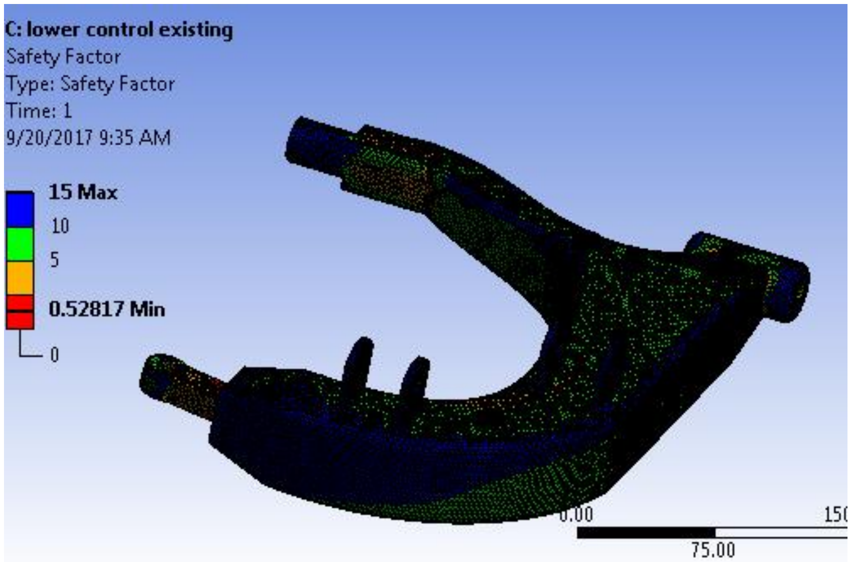
(D)


C: lower control existing
Maximum Shear Stress
Type: Maximum Shear Stress
Unit: MPa
Time: 1
9/20/2017 9:34 AM
238.29 Max
211.81
185.34
158.86
132.39
105.91
79.437
52.962
26.486
0.011013 Min
0.00
75.00

(E)


C: lower control existing
Maximum Principal Stress
Type: Maximum Principal Stress
Unit: MPa
Time: 1
9/20/2017 9:35 AM
563.55 Max
484.99
406.44
327.88
249.32
170.77
92.214
13.658
-64.897
-143.45 Min
0.00
75.00

(F)



(G)

Figure 3. 54: Results of (A)total deformation, (B)equivalent elastic strain ,(C) shear elastic strain(D) Equivalent (von-Mises) stress (E) maximum shear tress (F)maximum principal stress, (G) safety factor

Table 3. 24: Summarized results of existing front lower control arm

Analysis	Existing Front lower control arm results
Total deformation	1.6679mm
Maximum shear stress	238.2Mpa
Maximum principal stress	563.55Mpa
Equivalent (von-mises) stress	473.34Mpa
Equivalent elastic strain	0.002529
Shear elastic strain	0.003078
Safety factor	0.5218

IV. EXISTING REAR SEMI TRIALING ARM

Boundary conditions:

- Material: low alloy steel AISI 4130
- By assuming as the two ends of the arm which connects to frame and wheel is fixed.
- Required forces are given at shock absorber connecting point
- The analysis results are shown in figures which are given below

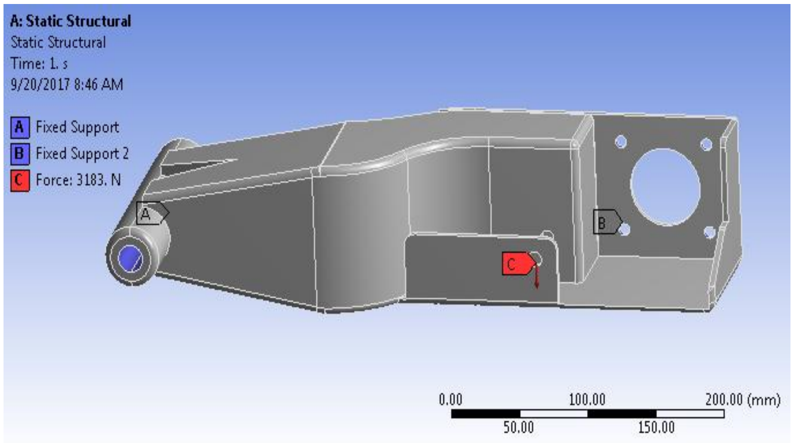


Figure 3. 55: Loads and boundary conditions of existing rear semi trailing arm

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 12386numbers of Nodes and 6619numbers of Elements has been found.

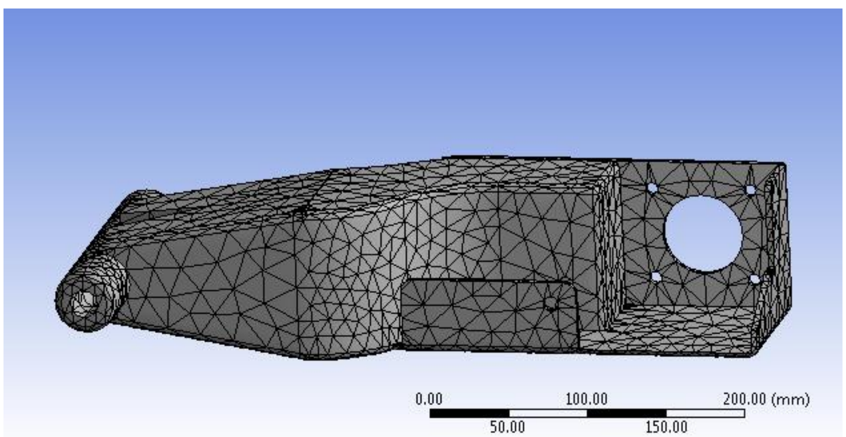
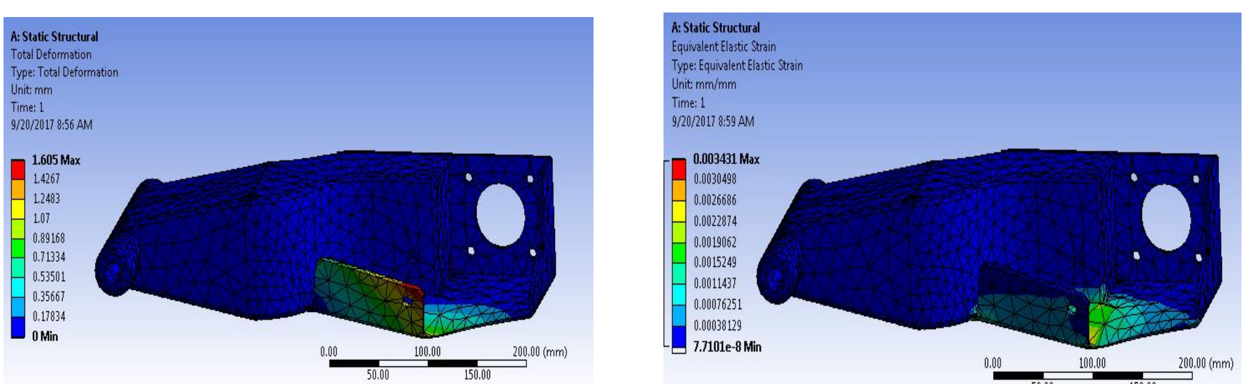
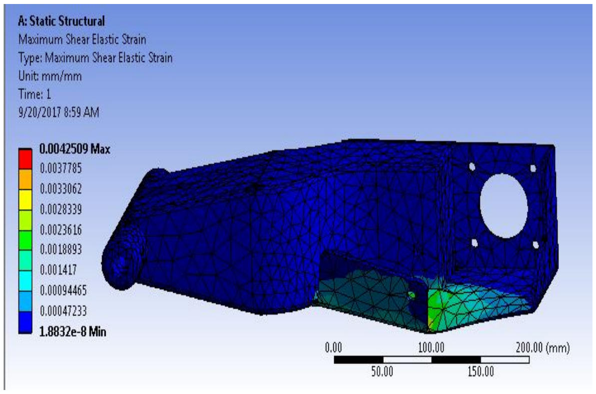


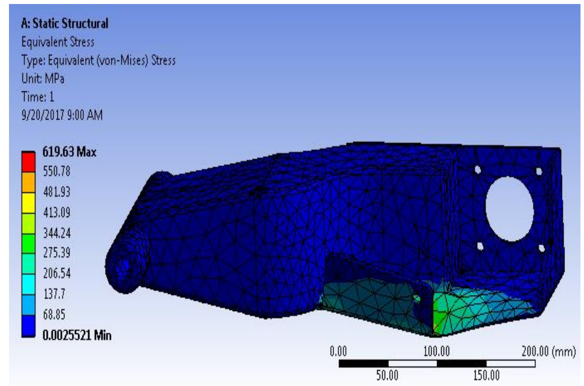
Figure 3. 56: Meshing of existing rear semi trailing arm



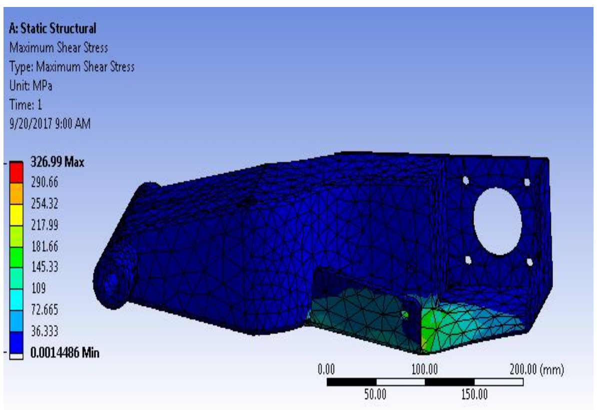
(A) (B)



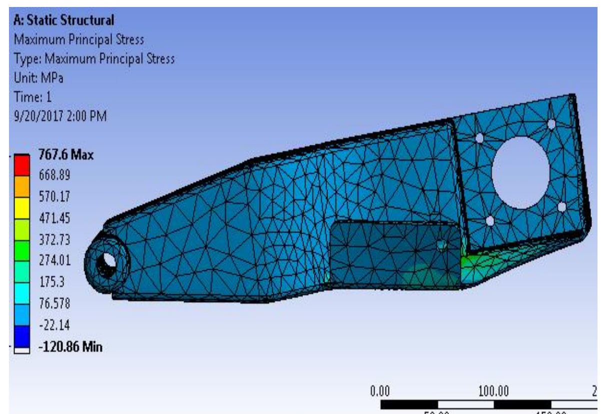
(C)



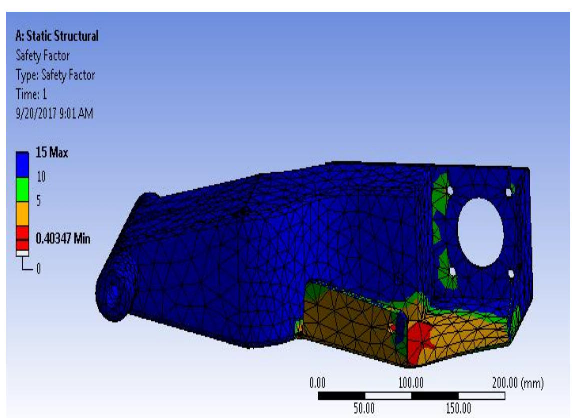
(D)



(E)



(F)



(F)

Figure 3. 57: Results of (A)total deformation, (B)equivalent elastic strain ,(C) shear elastic strain(D) Equivalent (von-Mises) stress (E) maximum shear tress (F) safety factor

Table 3. 25: Summarized results of existing rear semi trailing arm

Analysis	Existing rear semi trailing arm results
Total deformation	1.605mm
Maximum shear stress	326.99Mpa
Equivalent (von-mises) stress	619.63Mpa
Maximum principal stress	767.7MPa
Equivalent elastic strain	0.003411
Shear elastic strain	0.0042509
Safety factor	0.43

3.5.3 FEA OF THE ASSEMBLED NEW SUSPENSION SYSTEM

I. NEW FRONT MACPHERSON SUSPENSION SYSTEM

Boundary conditions:

- Material: low alloy steel AISI 4130 for the lower control arm, forged steel EN47 for knuckle and chrome vanadium for the spring.
- By assuming as the one end of the arm and top of the shock absorber are fixed to the frame.
- Required forces are given in different direction
- The analysis results are shown in figures which are given below

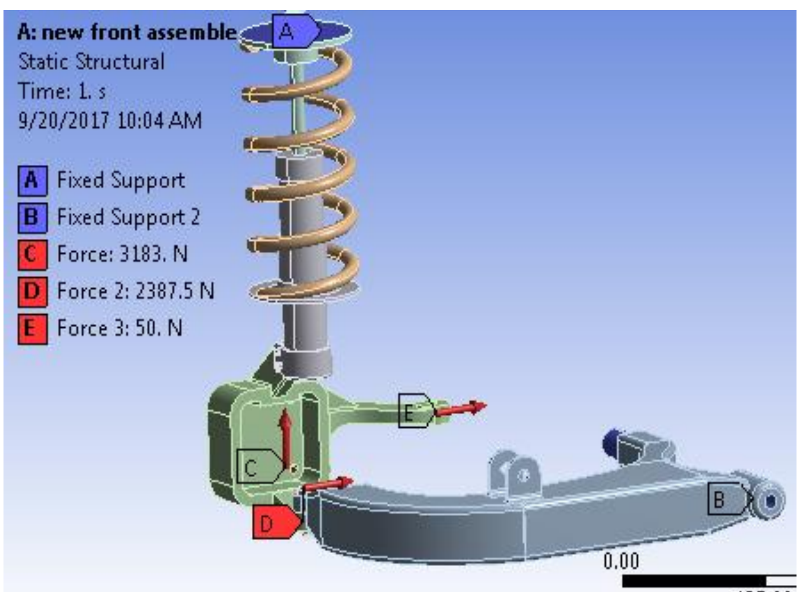


Figure 3. 58: Loads and boundary conditions of new assembled system

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 34707numbers of Nodes and 7748numbers of Elements has been found.

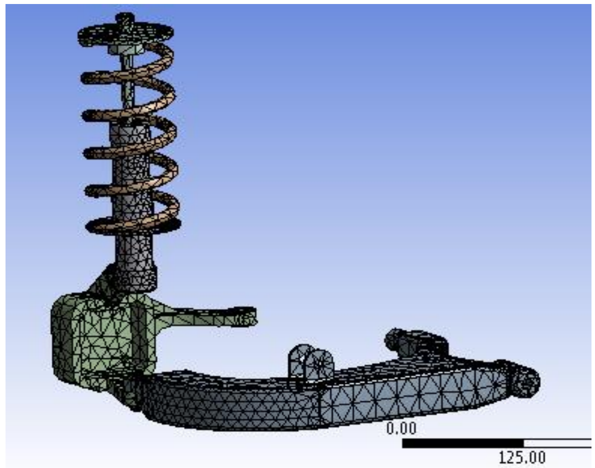
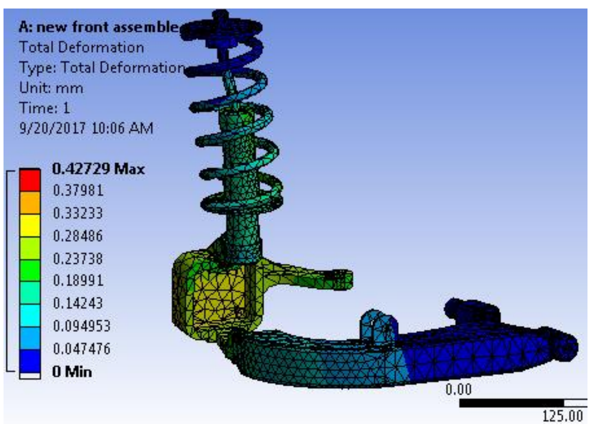
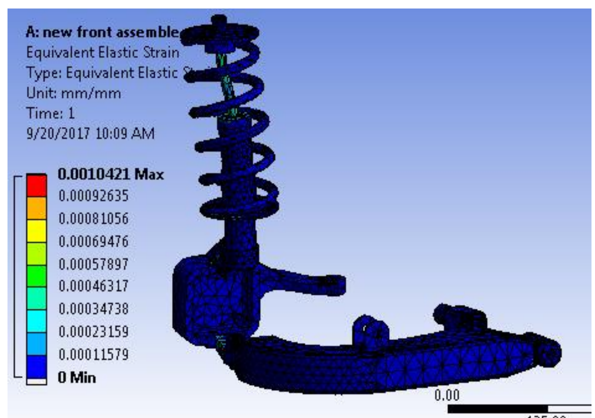
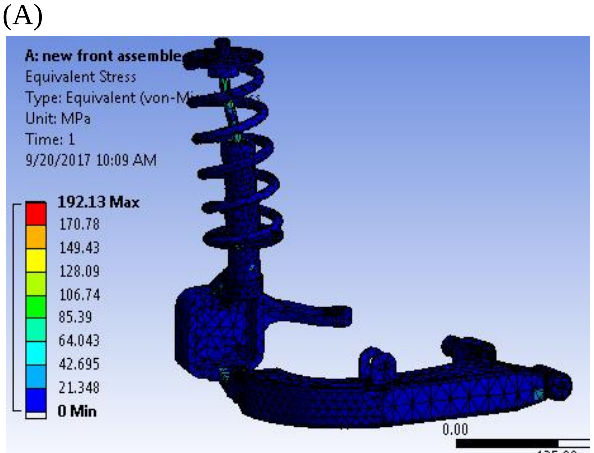


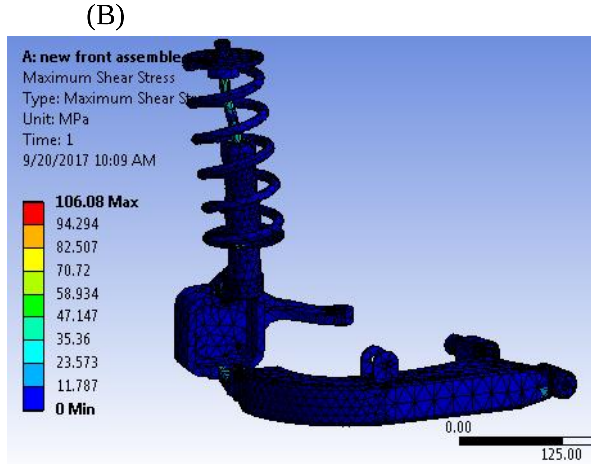
Figure 3. 59: Meshing of the new assembled system



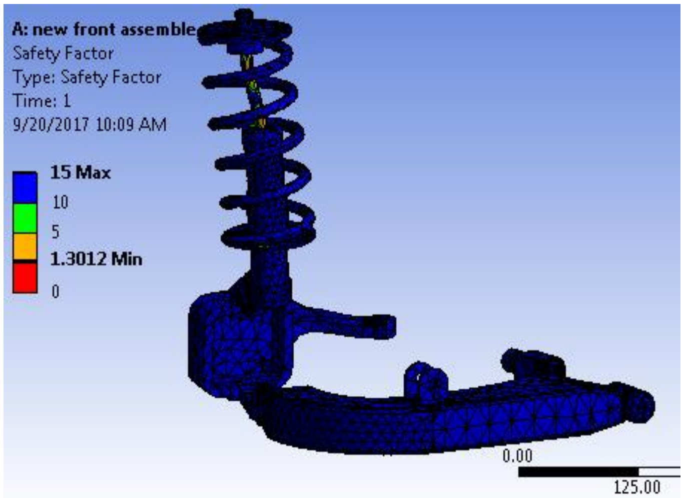


(A) (B)





(C) (D)



(E)

Figure 3. 60: results of (A) total deformation, (B) equivalent elastic strain, (C) Equivalent (von-Mises) stress (D) Maximum shear tress (E) safety factor

Table 3. 26: Summarized results of new front Macpherson suspension system

Analysis	new front Macpherson suspension system results
Total deformation	0.42729mm
Maximum shear stress	106.08Mpa
Equivalent (von-mises) stress	192.13Mpa
Equivalent elastic strain	0.0010421
Safety factor	1.0301

II. NEW REAR MACPHERSON SUSPENSION SYSTEM

Boundary conditions:

- Material: low alloy steel AISI 4130 for the lower control arm, forged steel EN47 for knuckle and chrome vanadium for the spring.
- By assuming as the one end of the arm and top of the shock absorber are fixed to the frame.
- Required forces are given
- The analysis results are shown in figures which are given below

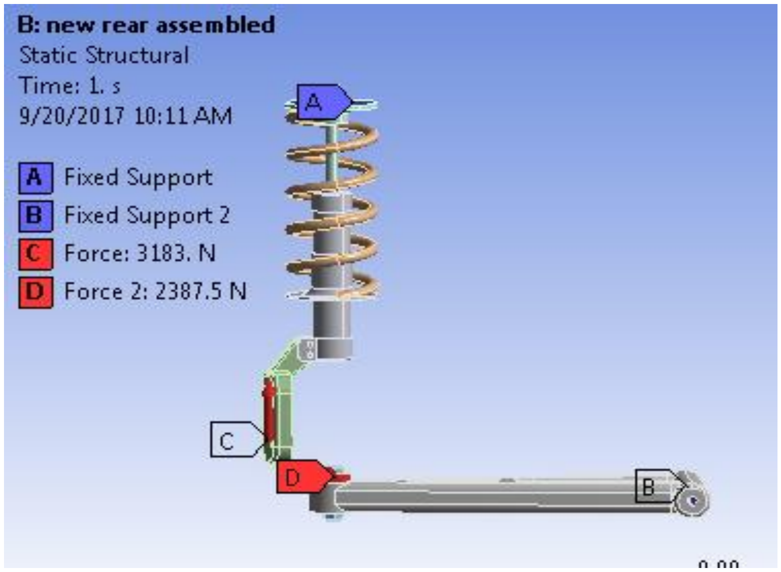


Figure 3. 61: Loads and boundary conditions of new rear assembled system

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 34707numbers of Nodes and 7748numbers of Elements has been found.

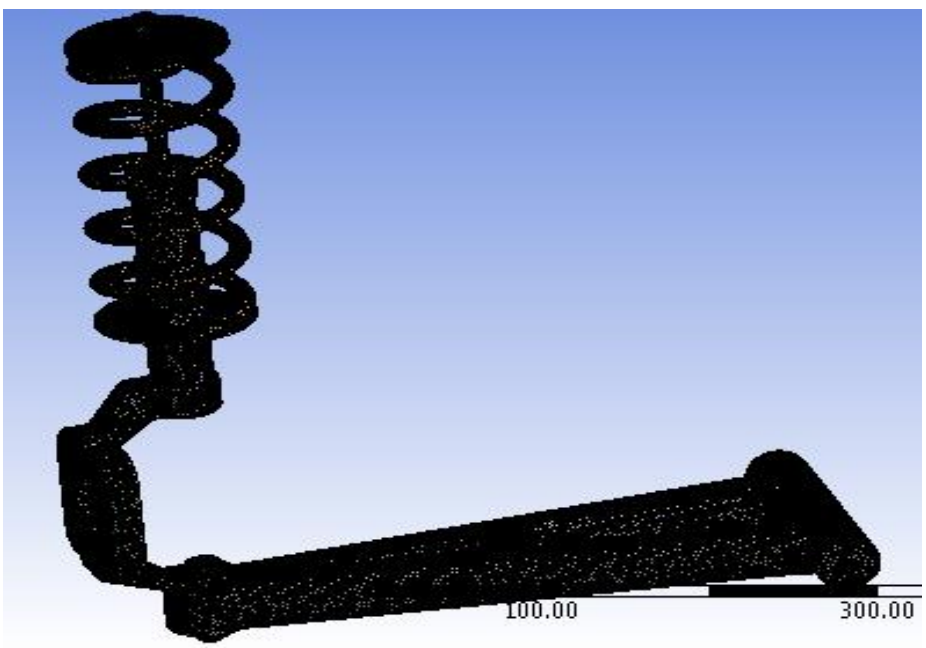
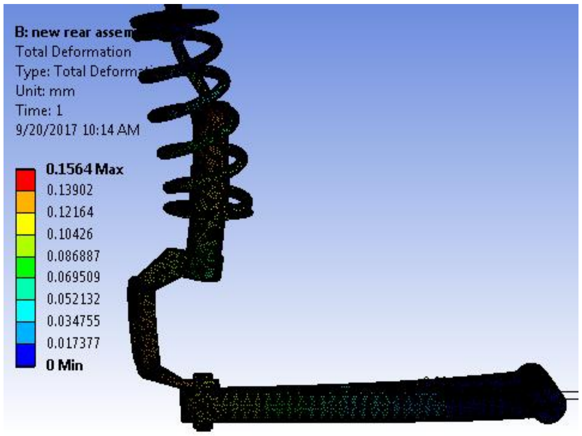
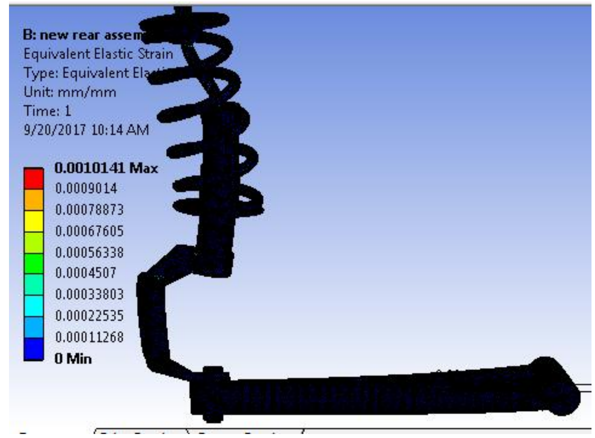


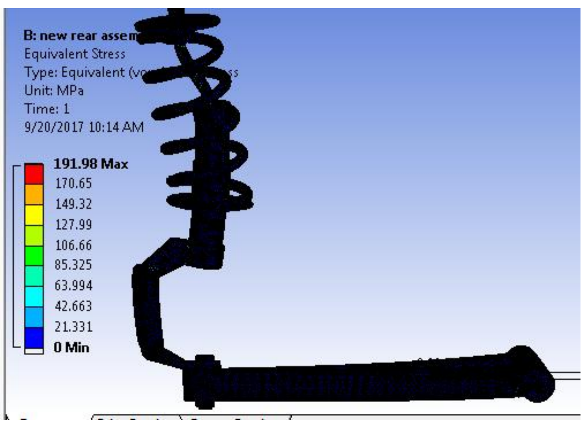
Figure 3. 62: Meshing of the new rear assembled system



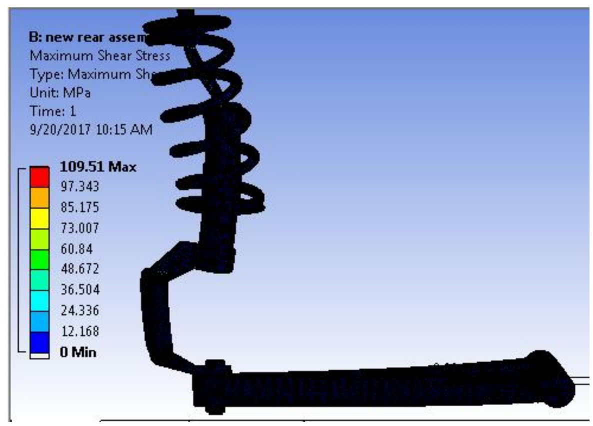
(A)



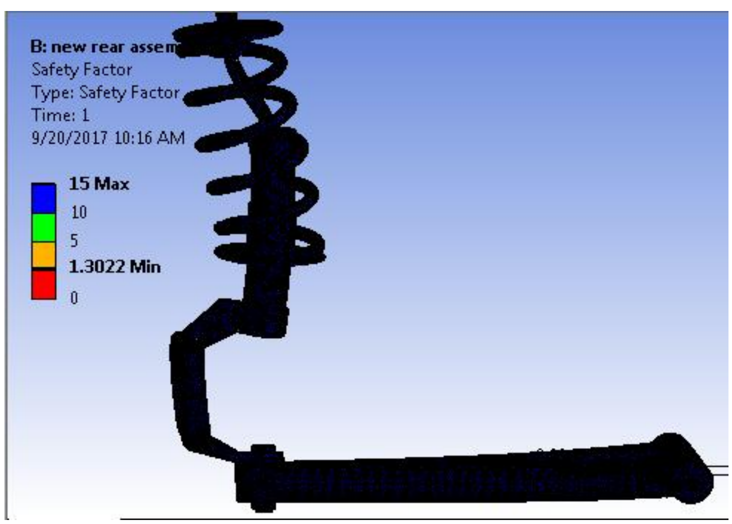
(B)



(C)



(D)



(E)

Figure 3. 63: Results of (A) total deformation, (B) equivalent elastic strain, (C) Equivalent (von-Mises) stress (D) Maximum shear tress (E) safety factor

Table 3. 27: Summarized results of new rear Macpherson suspension system

Analysis	New rear Macpherson suspension system results
Total deformation	0.1564mm
Maximum shear stress	109.51Mpa
Equivalent (von-mises) stress	191.98Mpa
Equivalent elastic strain	0.0010141
Safety factor	1.3022

3.5.4 FEA OF THE ASSEMBLED EXISTING SUSPENSION SYSTEM

I. EXISTING FRONT SUSPENSION SYSTEM

Boundary conditions:

- Material: low alloy steel AISI 4130 for the lower control arm, forged steel EN47is for the knuckle and chrome vanadium for the spring.
- By assuming as the one end of the arm and top of the shock absorber are fixed to the frame.
- Required forces are given in different direction
- The analysis results are shown in figures which are given below

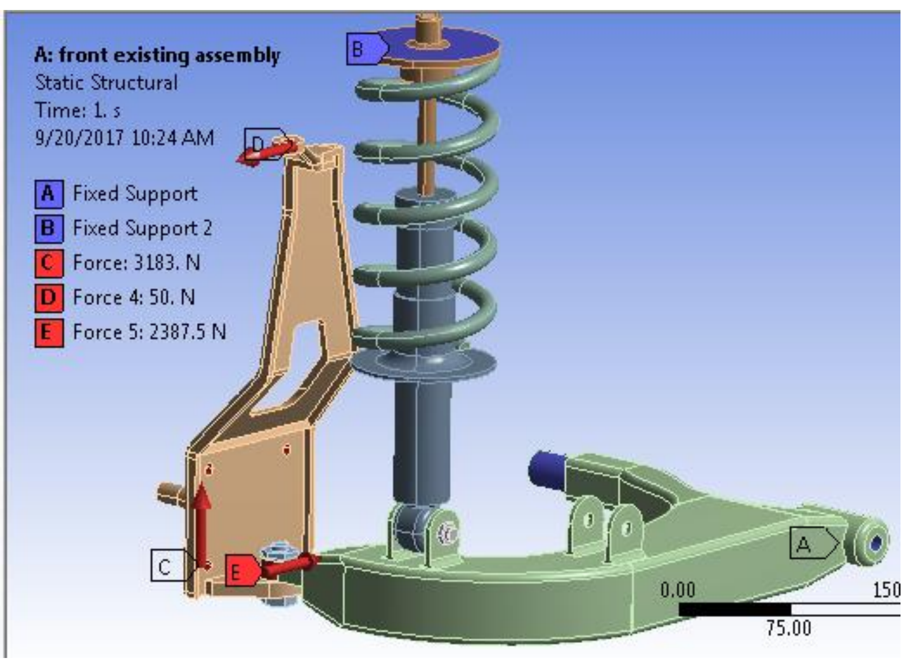


Figure 3. 64: Loads and boundary conditions of new front assembled system

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 1113869numbers of Nodes and 761949 numbers of Elements has been found.

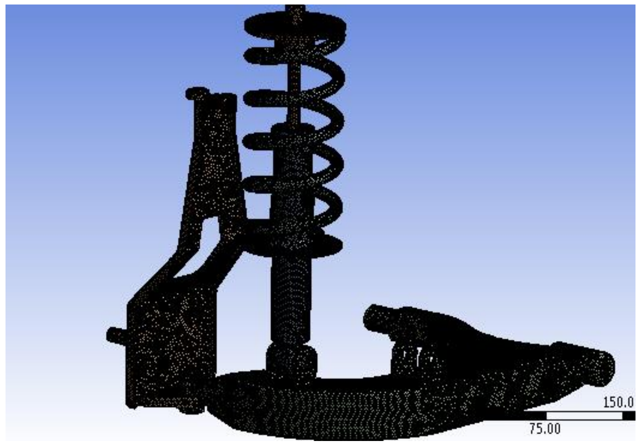
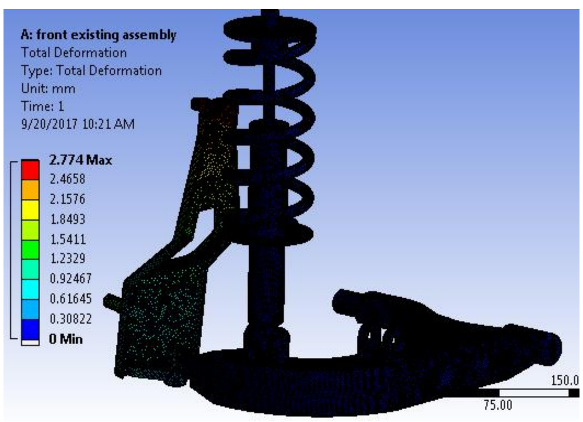
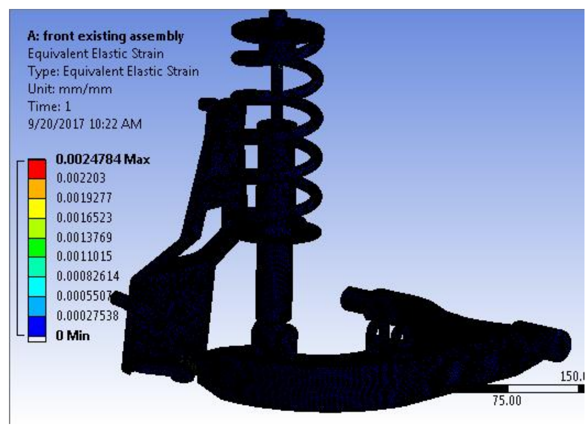


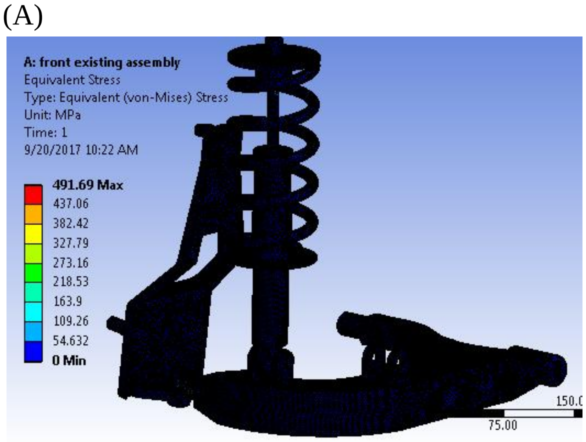
Figure 3. 65: Meshing of existing front system



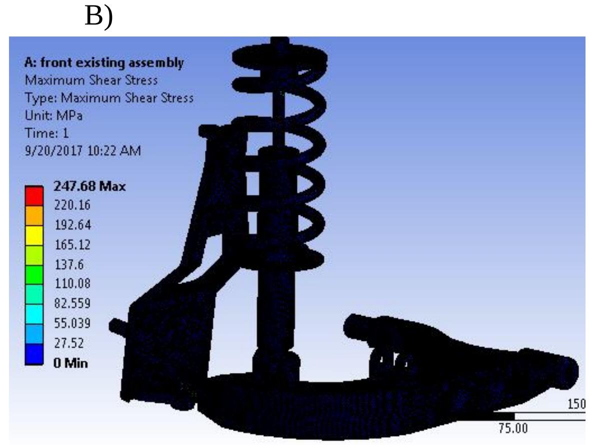
(A)



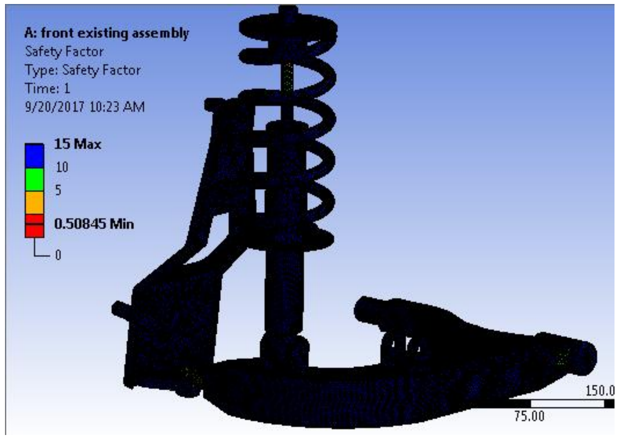
B)



(C)



(D)



(E)

Figure 3. 66: Results of (A) total deformation, (B) equivalent elastic strain, (C) Equivalent (von-Mises) stress (D) Maximum shear tress (E) safety factor

Table 3. 28: Summarized results of existing front suspension system

Analysis	Existing front suspension system results
Total deformation	2.774mm
Maximum shear stress	247.68Mpa
Equivalent (von-mises) stress	491.69Mpa
Equivalent elastic strain	0.0024784
Safety factor	0.50845

II. EXISTING REAR SUSPENSION SYSTEM

Boundary conditions:

- Material: low alloy steel AISI 4130 for the semi trailing arm and chrome vanadium for the spring.
- By assuming as the one end of the arm and top of the shock absorber are fixed to the frame.
- Required forces are given in different direction
- The analysis results are shown in figures which are given below

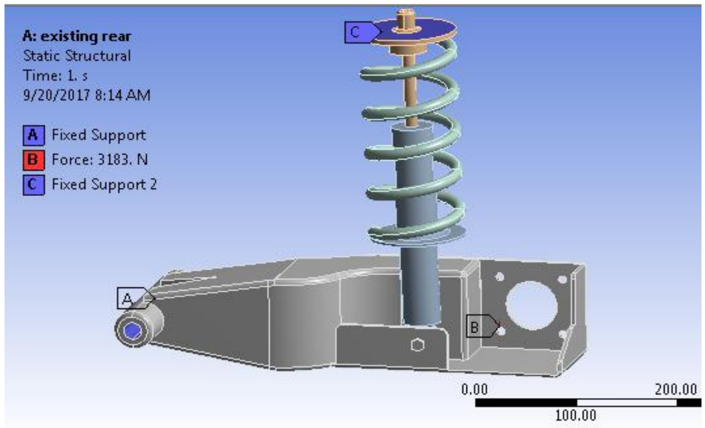


Figure 3. 67: Loads and boundary conditions of new rear assembled system

MESHING:

Tetrahedral mesh elements are used in meshing of the shock absorber. In order to get a better result, locally fine mesh is applied in the region which is suspected to have the highest stress. Finally, 27484numbers of Nodes and 13904 numbers of Elements has been found.

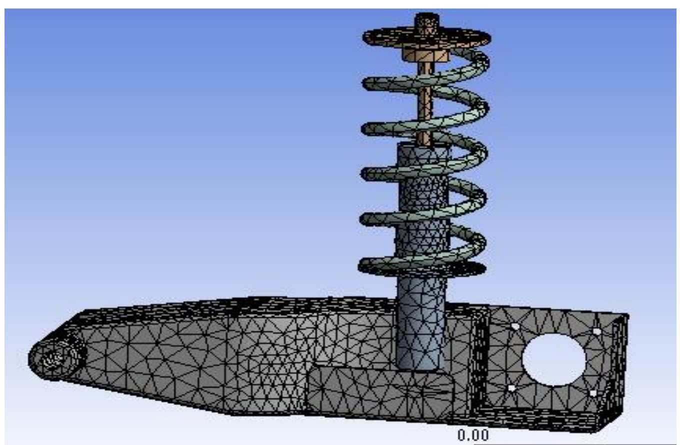
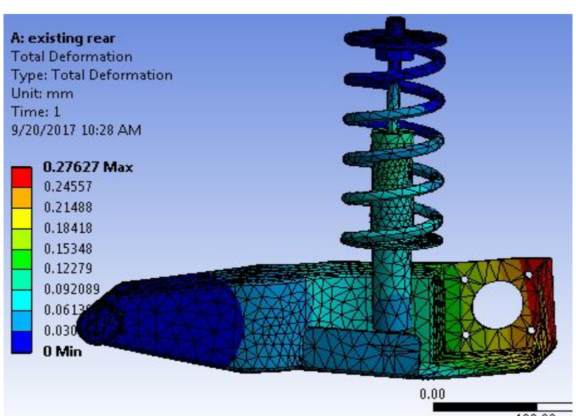
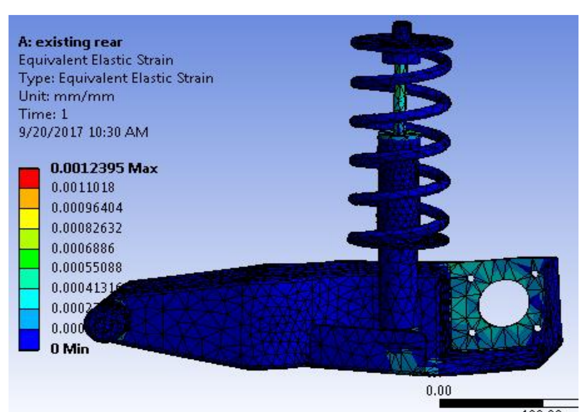


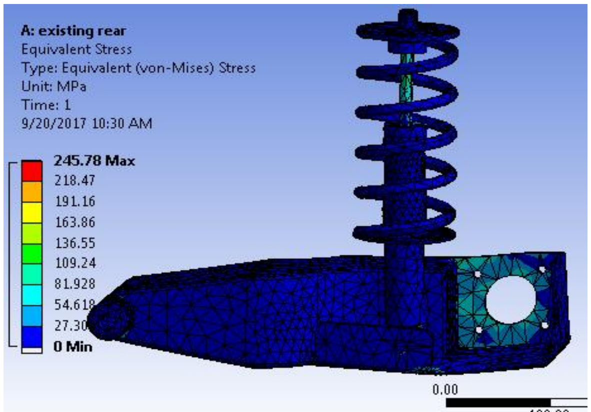
Figure 3. 68: Meshing of new rear assembled system

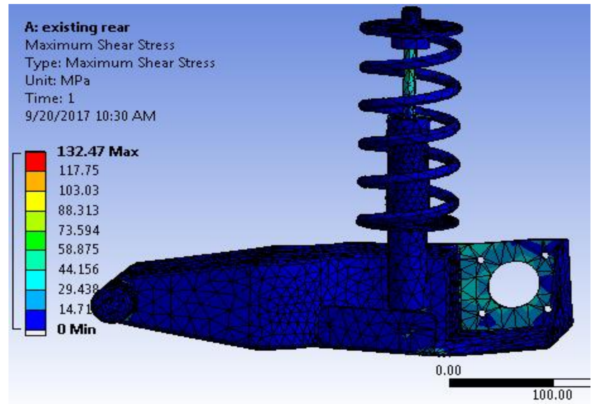


(A)

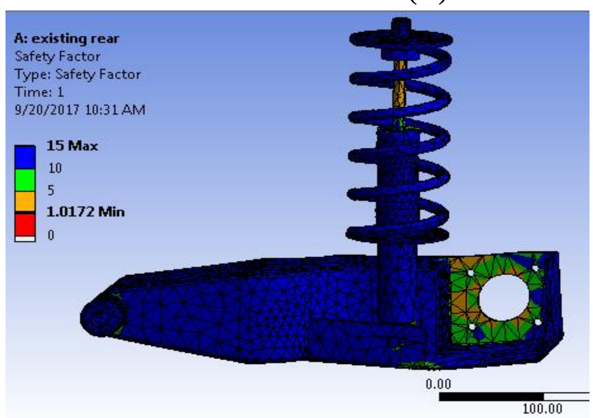


(B)





(C) (D)



(E)

Figure 3. 69: Results of (A) total deformation, (B)equivalent elastic strain, (C) Equivalent (von-Mises) stress (D) maximum shear tress (E) safety factor

Table 3. 29: Summarized results of existing rear suspension system

Analysis	**Existing rear suspension system results**
Total deformation	0.27627mm
Maximum shear stress	132.47Mpa
Equivalent (von-mises) stress	245.78Mpa
Equivalent elastic strain	0.001295
Safety factor	1.0172

CHAPTER FOUR

4. RESULT AND DISCUSSION

4.1 RESULT AND DISCUSSION OF NEW SUSPENSION SYSTEM

Table 4. 1: Summarized FEA results of new suspension components

SUSPENSION PARTS	FINITE ELEMENT ANALYSIS RESULTS								
	Total deformation (mm)	Directional deformation (mm)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Equivalent (von-Mises) stress (MPa)	Equivalent elastic strain	Shear elastic strain	Safety factor	Remark
Shock absorber	0.3485	0.0000908	27.127	79.223	108.76	0.0005383	0.000338	2.9891	Safe
Front knuckle	0.38219	0.10012	163.21	296.9	300	0.0015987	0.0010154	1.1665	Safe
Front lower control arm	0.0884	0.00062	59.06	119.3	116.78	0.000477	0.00061	3.94	Safe
Rear knuckle	0.74478	0.23396	109.2	230.3	196.73	0.000995	0.0007313	1.2708	Safe
Rear lower control arm	1.2784	0.00374	71.425	137.7	123.83	0.0004966	0.000437	3.72	Safe

4.1.1 Shock absorber

The shock absorber component has been modeled using CATIA V5 and analyzed using ANSYS workbench16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =225,731 and elements =140,017 as it gives better results. After meshing, the shock absorber top part was assumed as fixed support and force is applied on the strut. Then various parameters such as total deformation, equivalent (von-mises) stress, equivalent elastic stain, safety factor are completely analyzed. So, the result shows, equivalent (von-mises) stress is 108.76MPa, equivalent elastic stain is 0.0005383, safety factor is 2.98which is safe and the total deflection is of 0.3483mm. So, as the yield strength of the selected material is 620MPa, the analyzed stress is much less than the allowable stress of the selected material, it means that the design is safe under such loading.

4.1.2 Front knuckle

The steering knuckle component has been modeled using CATIA V5 modeling software and analyzed using ANSYS workbench16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =162,502 and elements =108,246 as it gives better results. After meshing, the knuckle hub was assumed as fixed support and moment was applied on the Ball Bearing Location / Stub Hole and force is applied on the strut bracket arm, knuckle point and steering arm(axial tensile force) in different axis. Then various parameters such as total deformation, directional deformation, Maximum principal stress, equivalent (von-mises) stress, equivalent elastic stain, shear elastic strain, safety factor And Maximum Shear Stress are completely analyzed based on the applied force. This paper represent that the areas where the stress concentration is maximum due to the applied load. So, the result shows, Maximum shear stress is 163.21MPa, Maximum principal stress is 296.99MPa, equivalent (von-mises) stress is 300MPa, equivalent elastic stain is 0.0015987, shear elastic strain is 0.0010154, safety factor is 1.1665 which is safe and the total deflection is of 0.38219mm. So, as the yield strength of the selected material is 350MPa. So, the analyzed stress is less than the allowable stress of the selected material, it means that the design is safe under such loading.

4.1.3 Front Lower control arm

In Finite Element Analysis different criteria’s was selected for design and analysis of front lower control arm. He model was created in CATIA V5 modeling software. This model was imported in to ANSYS workbench 16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =736,246 and elements =525,028 as it gives better results. After meshing, the ends of the arm which connects to frame was assumed as fixed support and various loads such as cornering force, braking force and strut forces are applied on the knuckle point in different axis. Then various parameters such as total deformation, directional deformation, Maximum principal stress, equivalent (von-mises) stress, equivalent elastic stain, shear elastic strain, safety factor And Maximum Shear Stress are completely analyzed based on the applied force. The result of maximum shear stress is 59.06MPa, Maximum principal stress is 119.27MPa, equivalent (von-mises) stress is 116.78MPa, equivalent elastic stain is 0.000477, shear elastic strain is 0.00061, safety factor 3.94

and the total deflection is of 0.0884mm. So, as the yield strength for this material is 460MPa, it means that the design is safe under such loading.

4.1.4 Rear knuckle

The knuckle component has been modeled using CATIA V5 modeling software and analyzed using ANSYS workbench16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =162,502 and elements =108,246 as it gives better results. After meshing, the knuckle hub was assumed as fixed support and moment was applied on the Ball Bearing Location / Stub Hole and force is applied on the strut bracket arm and knuckle point in different axis. Then various parameters such as total deformation, directional deformation, Maximum principal stress, equivalent (von-mises) stress, equivalent elastic stain, shear elastic strain, safety factor And Maximum Shear Stress are completely analyzed based on the applied force. This paper represent that the areas where the stress concentration is maximum due to the applied load. So, the result shows, Maximum shear stress is 109.92MPa, Maximum principal stress is 230.28MPa, equivalent (von-mises) stress is 196.73MPa, equivalent elastic stain is 0.000995, shear elastic strain is 0.0007313, safety factor 1.2708 which is safe and the total deflection is of 0.74478mm. So, as the yield strength of the selected material is 350MPa. So, the analyzed stress is less than the allowable stress of the selected material, it means that the design is safe under such loading.

4.1.5 Rear lower control arm

In Finite Element Analysis different criteria's was selected for design and analysis of rear lower control arm. The model is created in CATIA V5 modeling software. This model was imported in to ANSYS workbench 16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =471,828 and elements =331,180 as it gives better results. After meshing, the ends of the arm which connects to frame was assumed as fixed support and various loads such as braking force and strut forces are applied on the knuckle point in different axis. Then various parameters such as total deformation, directional deformation, Maximum principal stress, equivalent (von-mises) stress, equivalent elastic stain, shear elastic strain, safety factor And Maximum Shear Stress are completely analyzed based on the applied force. The result shows, Maximum shear stress is 71.425MPa, Maximum principal stress is 137.47MPa, equivalent (von-mises) stress is

123.83MPa, equivalent elastic stain is 0.0004966, shear elastic strain is 0.0004370, safety factor 3.72 which is safe and the total deflection is of 1.2784mm. So, as the allowable stress of the selected material is 460MPa, the analyzed stress is less than the allowable stress of the selected material, it means that the design is safe under such loading.

4.2 RESULT AND DISCUSSION OF THE ASSEMBLED NEW SUSPENSION SYSTEM

Table 4. 2: Summarized FEA of the assembled new suspension system

SUSPENSION PARTS	FINITE ELEMENT ANALYSIS RESULTS					
	Total deformation (mm)	Maximum shear stress (MPa)	Equivalent (von-Mises) stress (MPa)	Equivalent elastic strain	Safety factor	Remark
New front Macpherson suspension system	0.42729	106.8	192.13	0.0010421	1.3012	Safe
New rear Macpherson suspension system	0.1564	109.51	191.98	0.0010141	1.3022	Safe

For the new model, the maximum Von Misses stress magnitude obtained at front and rear suspension system is 192.13MPa and 191.98 MPa and the maximum shear stress magnitude is 106.8MPa and 109.51MPa. But factor of safety is 1.3012 and 1.3022 respectively. So it is safe for stiffness and thickness of the structure is at reliable stress level. Therefore, it indicates as it is necessary to accept a new model of suspension system or the vehicle.

4.3 RESULT AND DISCUSSION OF EXISTING SUSPENSION SYSTEM

Table 4. 3: Summarized FEA results of existing suspension components

SUSPENSION PARTS	FINITE ELEMENT ANALYSIS RESULTS							
	Total deformation (mm)	Maximum shear stress (MPa)	Maximum principal stress (MPa)	Equivalent (von-Mises) stress (MPa)	Equivalent elastic strain	Shear elastic strain	Safety factor	Remark
Existing Shock absorber	0.033017	62.594	151.91	110	0.00065349	0.0008	2.2727	Safe
Existing Front knuckle	0.2407	209.31	533.39	397.04	0.002041	0.002721	0.613	Not Safe
Existing Front lower control arm	1.667	238.29	563.55	473.34	0.002197	0.003078	0.52817	Not Safe
Existing rear semi trailing arm	1.605	326.99	767.7	619.63	0.003431	0.0042509	0.43	Not Safe

4.2.1 Existing Shock absorber

The shock absorber component has been modeled using CATIA V5 and analyzed using ANSYS workbench16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =227758 and elements =144684 as it gives better results. After meshing, the existing shock absorber top part was assumed as fixed support and force is applied on the bottom. Then various parameters such as total deformation, equivalent (von-mises) stress, maximum shear stress , equivalent elastic stain and safety factor are completely analyzed. So, the result shows, equivalent (von-mises) stress is 110MPa,maximum shear stress is 62.594 equivalent elastic stain is 0.000653, safety factor is 2.2727which is safe but not as the new one and the total deflection is of 0.033017mm. So, as the yield strength of the selected material is 620MPa, the analyzed stress is much less than the allowable stress of the selected material, it means that the design is safe but the new is mre safe than this.

4.2.2 Existing Front knuckle

The steering knuckle component has been modeled using CATIA V5 modeling software and analyzed using ANSYS workbench16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =126434 and elements =78756 as it gives better results. After meshing, the knuckle hub was assumed as fixed support and force is applied on the knuckle point and steering arm(axial tensile force) in different axis. Then various parameters such as total deformation, Maximum principal stress, equivalent (von-mises) stress, equivalent elastic stain, shear elastic strain, safety factor And Maximum Shear Stress are completely analyzed based on the applied force. This paper represent that the areas where the stress concentration is maximum due to the applied load. So, the result shows, Maximum shear stress is 209.31MPa, Maximum principal stress is 533.39MPa, equivalent (von-mises) stress is 397.4MPa, equivalent elastic stain is 0.002041, shear elastic strain is 0.00272, safety factor is 0.61482 which is less than 1 that's not safe and the total deflection is of 0.2407mm. So, as the yield strength of the selected material is 350MPa. So, the analyzed stress is greater than the yield strength of the selected material, it means that the design is not safe under such loading.

4.2.3 Existing front lower control arm

In Finite Element Analysis different criteria's was selected for design and analysis of existing front lower control arm. The model was created in CATIA V5 modeling software. This model was imported in to ANSYS workbench 16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =743058 and elements =529148 as it gives better results. After meshing, the ends of the arm which connects to frame was assumed as fixed support and various loads such as cornering force, braking force and vehicle load are applied on the knuckle point and shock absorber bracket in different axis. Then various parameters such as total deformation, Maximum principal stress, equivalent (von-mises) stress, equivalent elastic stain, shear elastic strain, safety factor And Maximum Shear Stress are completely analyzed based on the applied force. The result of maximum shear stress is 238.29MPa, Maximum principal stress is 563.55MPa, equivalent (von-mises) stress is 473.34MPa, equivalent elastic stain is 0.002197, shear elastic strain is 0.003078, safety factor 0.52817-not safe and the total deflection is of 1.6679mm. So, as the yield strength for this material is 460MPa, it means that the design is not safe under such loading.

4.2.4 Existing rear semi trailing arm

In Finite Element Analysis different criteria's was selected for design and analysis of existing rear semi trailing arm. The model was created in CATIA V5 modeling software. This model was imported in to ANSYS workbench 16.0. Then the selected material was added to the engineering data in ANSYS workbench 16.0. Then meshing has been done with the statistics of nodes =12386 and elements =6619 as it gives better results. After meshing, the ends of the arm which connects to frame was assumed as fixed support and various loads such as braking force and vehicle load are applied on the shock absorber bracket in different axis. Then various parameters such as total deformation, Maximum principal stress, equivalent (von-mises) stress, equivalent elastic stain, shear elastic strain, safety factor And Maximum Shear Stress are completely analyzed based on the applied force. The result of maximum shear stress is 326.99MPa, Maximum principal stress is 767.7MPa, equivalent (von-mises) stress is 619.63MPa, equivalent elastic stain is 0.003431, shear elastic strain is 0.004251, safety factor 0.43-not safe and the total deflection is of 1.605mm. So, as the yield strength for this material is 460MPa, it means that the design is not safe under such loading.

4.4 RESULT AND DISCUSSION OF ASSEMBLED EXISTING SUSPENSION SYSTEM

Table 4. 4: Summarized FEA of the assembled existing suspension system

SUSPENSION PARTS	FINITE ELEMENT ANALYSIS RESULTS					
	Total deformation (mm)	Maximum shear stress (MPa)	Equivalent (von-Mises) stress (MPa)	Equivalent elastic strain	Safety factor	Remark
Existing front suspension system	2.774	247.68	491.69	0.0024784	0.50845	Not Safe
Existing rear suspension system	0.27627	132.47	245.78	0.001295	1.0172	Better Not Safe

For the existing model, the maximum Von Misses stress magnitude obtained at front and rear suspension system is 491.69MPa and 245.78MPa and the maximum shear stress magnitude is 247.68MPa and 132.47MPa. But factor of safety is 0.50845 and 1.0172 respectively. So it is not safe for stiffness and thickness of the structure is at not reliable stress level. Therefore, it indicates as it is necessary to change the suspension system of the vehicle.

CHAPTER FIVE

5. SUMMERY, CONCLUSION AND RECOMMENDATION

5.1 SUMMERY

In this thesis Macpherson struts suspension system is designed for Bajaj Qute four wheeler small vehicles and modeled in 3D CATIA software. The main dimensions of the suspension system components are taken from the existing system of front single wishbone and rear semi trailing arm. The existing system is replaced by Macpherson suspension system by increasing the ground clearance by 40mm, from 180mm in the existing system when the vehicle is unloaded and the new is 220mm when the vehicle is unloaded. The new system is of less weight and needs small space relative to the existing system. Static structural simulation is done for the new model and existing model in ANSYS Workbench 16.0. Finally, the result of the analysis was checked weather it is safe or not

5.2 CONCLUSION

In this paper the design of Macpherson strut suspension system for Bajaj Qute four wheeler small vehicles is adapted in order to solve the limitation in lateral direction which is flexible camber curve with respect to roll center and lower ground clearance with respect to the road conditions of Ethiopia. Additionally, the steering arm of the conventional knuckle was vertically above the tyre. So, during steering this may affect the vehicle camber curve due to unbalance steering force distribution. In this paper all the components of new and existing suspension systems are modeled using CATIA V5 modeling software and imported in to ANSYS workbench 16.0 in order to analyze different parameters and the result was compared.

- The new models becomes safe than the existing in all aspects. So, the problem in lateral direction which is flexible camber curve is solved, because the Macpherson suspension system have good Camber curve not as flexible as other suspension system and also the stress in existing system leads the system to have flexible camber, this is corrected in the new model.

- Also, the steering arm position is corrected to the middle of the knuckle to balance the steering force distribution.

- The vehicle also has lower ground clearance which is 180 mm when the vehicle is-unloaded, with this ground clearance it is difficult to drive in unpaved road. So, in this design the ground clearance of the vehicle is increased to 220mm. Therefore, the vehicle can be driven in all road conditions.

- Finally, I conclude that as per the analysis using different material for different parts of the suspension system is best and also the model of Macpherson suspension system for Bajaj Qute small vehicle is safe. So, I recommend the vehicle company, the importers and the owners to use this.

5.3 RECOMMENDATION AND FUTURE WORKS

As a recommendation to Adama Science and Technology University (ASTU), Ethiopian investors, researchers, vehicle suspension manufacturing companies, microenterprises and to the government related to future work are as follows:

Recommendation:-

- To invest their knowledge and money on further study of the proposed suspension system
- Further redesign of the frame of the vehicle is recommended.

Future work:-

- Analysis using different materials under Different loads considering many other factors should be done in detail
- To study on the Stability of the vehicle.
- To design the new system using kinematics software

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APPENDEX

Questionnaire

General information

Age	
Gender	
Occupation	
Job location	
Resident location	
Date	

1. If there is a noise underside of the vehicle when it is ride in unpaved road condition (thick appropriate cell)

	No	Some	Much	Too much
Underside Noise				

Specify the problem-

2. If there is a problem of stability during riding in lateral direction (thick appropriate cell)

Yes (specify the problem)	
No	

3.If you see a problem specially in the suspension system and steering stability (thick appropriate cell)

Yes (specify the problem)	
No	

4. If you see a design problem in the suspension system of the vehicle?	
Yes (specify the problem)	
No	

5. Which component of the suspension system of the vehicle is failed mostly?	
knuckle	
Front Control arm	
Shock absorber	
spring	
Semi trailing arm	

6. During maintenance of the system what kind of failure have you observed?

7. In which underside component of the vehicle occurs the damage mostly?
where (specify the problem)