DESIGN AND ANALYSIS OF SUSPENSION SYSTEM FOR A FORMULA CAR

MEB4441 – PROJECT & VIVA-VOCE REPORT

Submitted by

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BONAFIDE CERTIFICATE

Certified that this Project Report titled "DESIGN AND ANALYSIS OF SUSPENSION SYSTEM FOR A FORMULA CAR" is the bonafide work of Mr. MANNE BHARGAVA SAI (18127008), Mr. SIMUNI VAMSIKRISHNA REDDY (18127026), Mr. SAMANTHAVADI TEJA (18127028), Mr. AMBATI ANAND REDDY (18127044) who carried out the project work under my supervision during the academic year 2021-2022.

2/5	02/05/22
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INTERNAL EXAMINER **EXTERNAL EXAMINER**

ABSTRACT

In automobiles, a double wishbone suspension is an independent suspension design that uses two wishbone- shaped arms to attach the wheel. Each wishbone or arm has two mounting points to the chassis and one joint at the knuckle. The coil over shock absorber mount to the wishbones to controls the vertical movement of the wheel. Double wishbone designs allow the engineer to carefully control the motion of the wheel throughout suspension travel, controlling such parameters as camber angle, caster angle, toe pattern, roll center height, scrub radius, scuff and more. In this project, the modelling of SUSPENSION SYSTEM OF A FORMULA CAR is done using modelling software CATIA and analysis is carried out using analysis tool ANSYS. The suspension system is analyzed using ANSYS Workbench.

Keywords: *Suspension system, Wishbone, Shock Absorbers, CATIA, ANSYS*

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MEB4441 – Project & Viva -voce

Project Batch No	**:**2
Title of the Project	:DESIGN AND ANALYSIS OF SUSPENSION SYSTEM FOR A FORMULA CAR
No of students in the Batch	**:**4

Students' Individual contribution

Name of the student	: MANNE BHARGAVA SAI
Register No	: 18127008

Myself, Bhargava Sai contributed the cad model using solid works but due to some errors in the design, the mesh got failed. Later the design got carried out using CATIA based on SAE standards. Design calculations were done for the coil over shock absorber. The knowledge for performing the calculations was gained by various tutorial videos and journals. Later, the information required for introduction of this project is gathered and the final alignment and all other corrections have been done in the report work. The information required for the PPT had been collected and results were tabulated and all other corrections have been done in the PPT.

Name and Signature of Student

Department of Mechanical Engineering

MEB4441 – Project & Viva -voce

Project Batch No	**:**2
Title of the Project	:DESIGN AND ANALYSIS OF SUSPENSION SYSTEM FOR A FORMULA CAR
No of students in the Batch	**:**4

Students' Individual contribution

Name of the student	: SIMUNI VAMSIKRISHNA REDDY
Register No	: 18127026

Myself, Vamsikrishna Reddy contributed all the literature papers and evaluated and took all the necessary information and values which are suitable for 3-D modelling and Analysis. Later, the analysis was performed using ANSYS software, where the fine mesh is applied and various parameters like Total deformation, Directional deformation, equivalent shear stress of the Suspension components were found and the results were compared among different materials and selected the best material. Later, the information required for introduction of 3-D modelling and Analysis software required for this project is gathered and all other corrections have been done in PPT.

Name and Signature of Student

Department of Mechanical Engineering

MEB4441 – Project & Viva -voce

Project Batch No **:**2

Title of the Project :DESIGN AND ANALYSIS OF SUSPENSION SYSTEM FOR A FORMULA CAR

No of students in the Batch **:**4

Students' Individual contribution

Name of the student : SAMANTHAVADI TEJA

Register No : 18127028

Myself, Teja Contributed all the literature papers and evaluated and took all the necessary information and values which are suitable for 3-D modelling and Analysis. Design calculations were done for the coil over shock absorber. The knowledge for performing the calculations was gained by various tutorial videos and journals. Finally, the analysis was done using ANSYS software, where the fine mesh is applied and various parameters like Total deformation, stress induced in the various suspension componentswere found and the results were compared among different materials. The information required for the PPT had been collected and results were tabulated and all other corrections have been done in the PPT.

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Students' Individual contribution

Name of the student	: AMBATI ANAND REDDY
Register No	: 18127044

Myself, Anand Reddy contributed the cad model using solid works but due to some errors in the design, the mesh got failed. Later the design got carried out using CATIA based on SAE standards. the analysis was done using ANSYS software, where the fine mesh is applied and various parameters like Total deformation, stress induced in the various suspension components were found and the results were compared among different materials. Later, the information required for introduction of 3-D modelling and Analysis software required for this project is gathered and all other corrections have been done in the report work.

Name and Signature of Student

CHAPTER 1

INTRODUCTION

n automobiles, a double wishbone suspension is an independent suspension design that uses two wishbone shaped arms to attach the wheel. Each wishbone or arm has two mounting points to the chassis and one joint at the knuckle. The coil over shock absorber mount to the wishbones to controls the vertical movement of the wheel. Double wishbone designs allows the engineer to carefully control and observe the motion range of the wheel throughout suspension travel, controlling various parameters as camber angle, caster angle, toe pattern, roll center height, scuff and more. The double wishbone suspension can also be called as "double A arms," because the arms themselves can be A shaped, L shaped, or even a single rod linkage.

Advantages include that it provides the engineer freer parameters than some other types dot is very easy to calculate the effect of moving each joint, so you can easily adjust the kinematics of the suspension and greatly optimize the movement of the wheel. In addition, the stress applied to various parts can be easily calculated, enabling the design of more optimized lightweight parts. They also provide increasing negative camber gain all the way to full jounce travel, unlike the MacPherson strut, which provides negative camber gain only at the beginning of jounce travel and then reverses into positive camber gain at high jounce amounts.

The design and analysis of few suspension system components is crucial. The components are designed from the calculations obtained from the various design parameters. Then, in the analysis phase, fixation is assumed depending on the function of the component, and the load is applied based on the load transfer calculation table.

1.1 Types of Suspension System

The suspension systems are basically classified into two types.

- Dependent suspension systems
- Independent suspension systems

1.2 Dependent Suspension System:

A dependent suspension typically has live axles (simple beams or "carriage" axles) that keep the wheels parallel to each other and perpendicular to the axle. They can be distinguished by the system of links used to place them both vertically and horizontally. Often, both features are combined into a set of links. If the camber of one wheel changes, so does the camber of the other wheel.

Examples of location linkages include:

- Trailing arms
- Satchell link
- Panhard rod
- Watts linkage
- Inboard
- Mumford linkage
- deDion axle

For front-engine, rear-wheel drive vehicles, the subordinate rear suspension can be either a "live axle" or a DeDion axle, depending on whether the differential is mounted on the axle. Solid axles are simpler, but unsprung weight contributes to wheel bounce.

Dependent (and semi-independent) suspensions need to carry a large load relative to the weight of the vehicle, have relatively soft springs, and are active suspensions (for cost and simplicity) because it provides constant camber. Most common in vehicles that do not use. The use of subordinate front suspension was restricted to heavier commercial vehicles

1.3 Independent Suspension System:

Independent suspension is a broad term for any vehicle suspension system that allows each wheel on the same axle to move vertically independently of each other (that is, react to road bumps). This is in contrast to rigid axles, rigid axles, or DeDion axle systems where the wheels are linked to each other. One movement affects the other wheel. Note that "independent" refers to the movement or path of movement of the wheel / suspension. The left and right sides of the suspension are typically connected to anti-roll bars or other such mechanisms. The anti-roll bar connects the left and right spring rates of the suspension, but not their movements.

Most modern vehicles have a separate front suspension (IFS). Many vehicles also have a separate rear suspension (IRS). As the name implies, the IRS independently bounced off the rear wheels. A completely independent suspension has an independent suspension on every wheel. Some early independent systems used swing axles, while modern systems use Chapman or MacPherson struts, trailing arms, multilinks or wishbones. Independent suspensions typically have lower unsprung weight, allowing each wheel to reach the road uninterrupted by the movement of the other wheels of the vehicle, improving ride quality and handling characteristics. Independent suspension requires additional engineering work and cost to develop compared to beam or live axle placement. Highly complex IRS solutions can also lead to higher manufacturing costs.

The main reason for the lower unsprung weight compared to the driven axle design is that the driven wheel differential unit does not form part of the unsprung element of the suspension system. Instead, bolt it directly to the vehicle chassis, or more commonly to the subframe. Relative movement between the wheels and the differential is achieved through the use of oscillating drive shafts connected by universal joints (U), similar to constant velocity (CV) joints used in front-wheel drive vehicles

The variety of independent systems includes:

- Swing axle
- deDion axle
- Macpherson strut
- Wishbones
- Multi-links
- Torsion beam
- Semi-trailing arm

1.4 Suspension Kinematics:

The motion of the linkages is of primary importance in designing a suspension system for a race car. The linkages must be positioned such that the linearity is maintained throughout the operating ranges of input and output parameters of the system. For example, during the suspension actuation there must not be high changes and sudden changes in the parameters like camber angle, roll center position, caster angle, mechanical trail, kingpin inclination angle etc.

1.5 Roll Center Height:

The most important parameter in the lateral suspension design is the roll center. If lateral force is applied at the roll center the chassis will not roll.

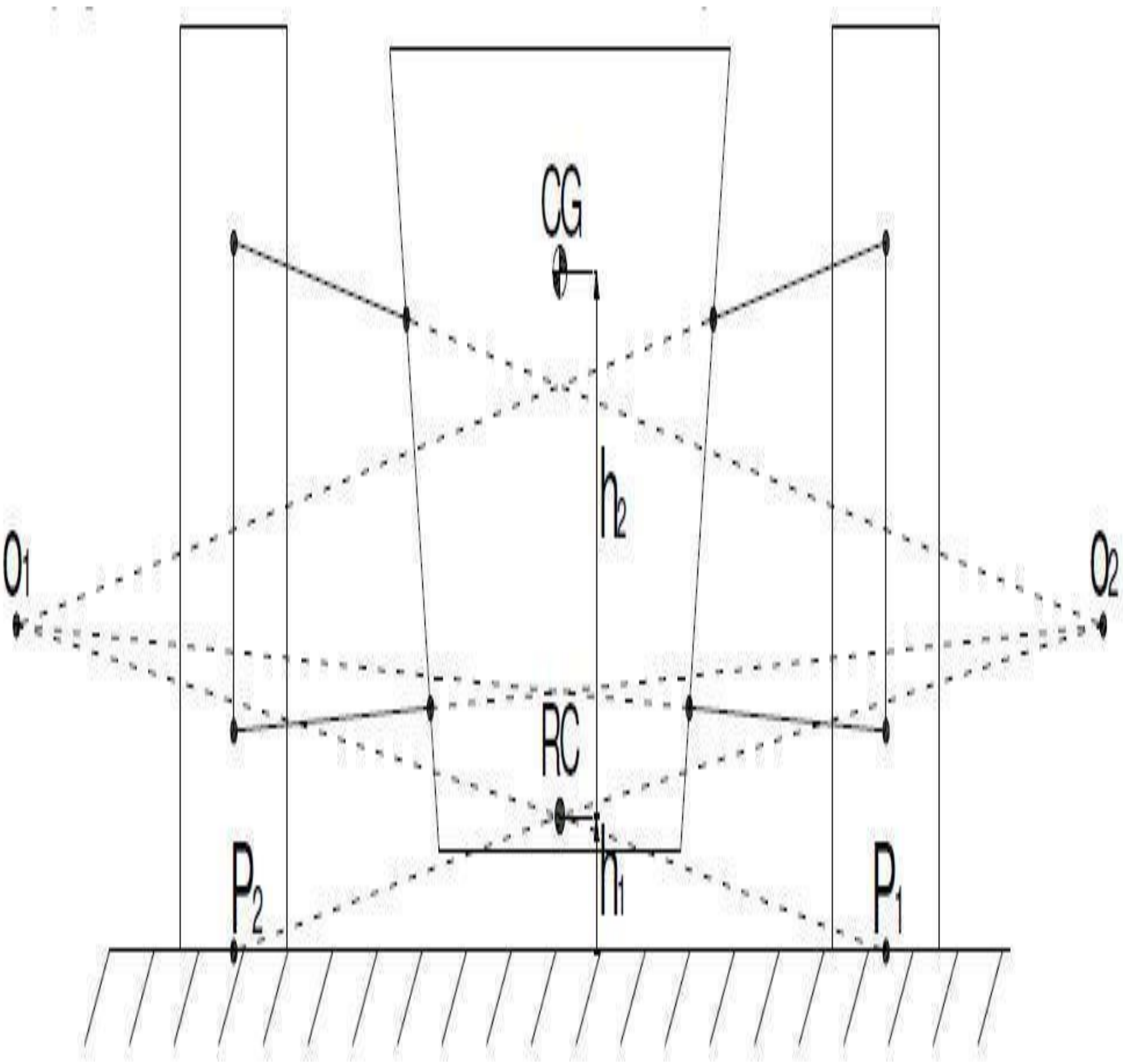

Fig 1.1 Depiction of ICs and RC and their height

Constructing a roll center above the center of gravity CG will create a negative $h2$ *and* a negative roll moment causing the chassis to roll into the curve like a motorcycle.

If the roll center is placed on the center of gravity CG the roll moment during cornering will be zero. And no roll stiffness has to be added to suppress the chassis roll.

Creating a roll center below the center of gravity CG will create a positive $h2$ and a positive roll moment causing the chassis to roll out of the curve, which is a natural movement when driving the car. But what will happen when the roll center is placed below the road surface? Therefore, two lateral force equilibriums for static curving are drawn in figure. In the first situation the roll center lies above the road surface and in the second situation the roll center lies below the roa1d surface which means $h1$ is negative. The used forces are, the lateral tire forces $Flat$, the tire normal forces FN and the so-called load transfer force FLT.

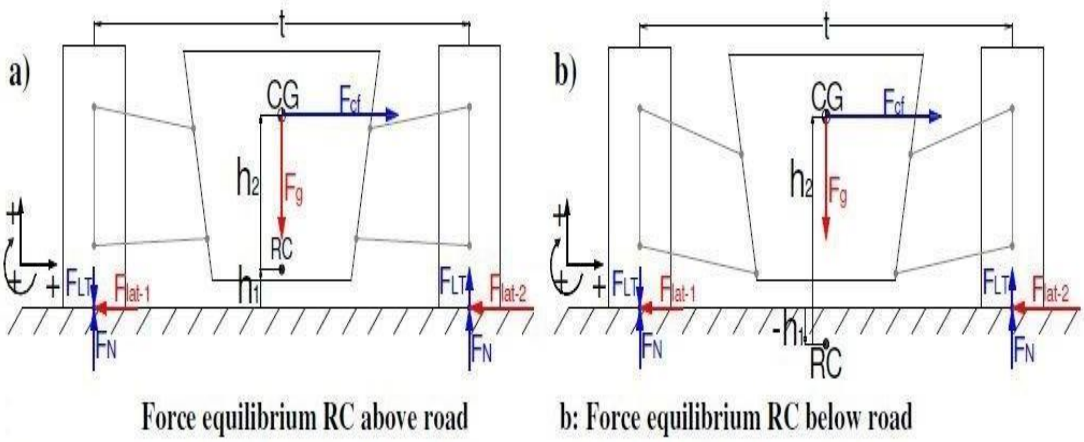

Fig 1.2 RC above and below ground

1.6 Camber

Camber is defined as the angle between the tire and the road in rear view. Figure shows three situations for the outer wheel; in the left situation the wheel is not cambered in other words the camber angle is zero. The middle situation shows a negative camber angle. The right wheel is cambered positive

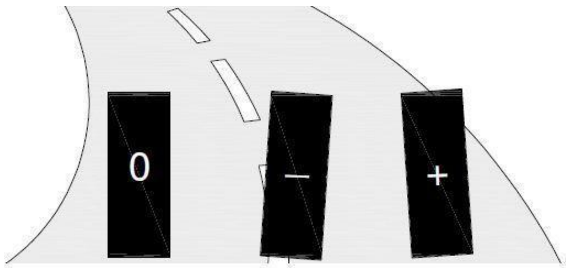

Fig 1.3 Camber angles

Lateral front tire force rises as the camber angle rises due to the addition of the camber thrust to the lateral force. This increases the lateral grip of the vehicle and helps in quick cornering maneuver. Large camber angles are not useful. Because the contact patch starts to become smaller.

Also, the camber change rate is very important. The camber change rate must be uniform and linear; else the handling of the car is affected badly. This may also lead to the breakage of linkages.

For the camber change rate to be minimum, the instantaneous centers are ensured to be far from the wheel center.

1.7 Kingpin Inclination and Scrub Radius:

The kingpin is built by connecting the top and bottom upright pivot points. Extend this line to the floor to see the scrub radius. The radius of friction causes steering torque when driving on bumps. If the kingpin is tilted inward when viewed from the front, this is known as a kingpin tilt or KPI. This can be used to edit the scrub radius. *Figure* shows a schematic section of the front tire, the chosen rim has an offset of 21 mm from the rim midline, and then the brake disk is placed with an inward offset of 9 mm. The clearance between the rim's inside and the brake disk is necessary to mount the braking caliper. The lower pivoting point is always placed as close as possible to brake disk (38 mm from midline). If the upper pivoting point is kept straight above the lower pivoting point the scrub radius will be 38 mm. Some KPI is applied which has reduced the scrub radius to 21 mm.

The king pin inclination angle is **arc tan (21/240) =5.0^{o}**.

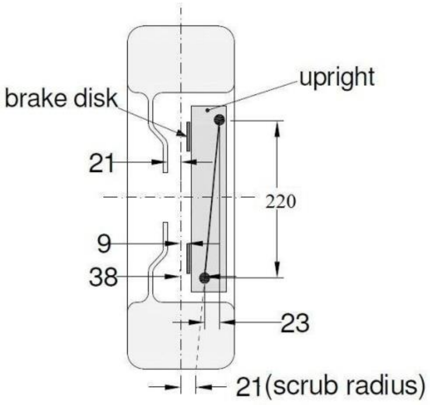

Fig 1.4 Wheel front view

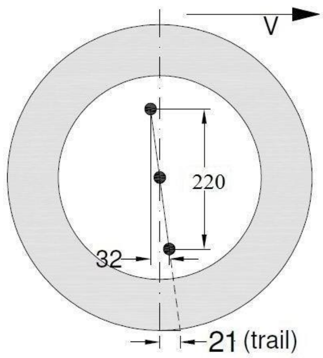

Fig 1.5 Wheel side view

1.8 Caster and Trail

The caster angle is defined as the angle of the kingpin when viewed from the side. Affects mechanical trucks. Mechanical casters are used to improve steering stability. Mechanical casters also generate steering torque. The lateral force of the tire is applied to the tire's tread, and the mechanical track acts as an arm that rotates around the kingpin, creating a moment in the kingpin. This moment is felt by the driver through the steering system. In addition to mechanical trucks, there are pneumatic trucks. This truck is caused by tires. It decreases as the slip angle increases. This is the feeling when the front tires start to slip. Mechanical casters must be on the order of pneumatic casters for the steering system to be self-centering. In addition, casters can be used to compensate for changes in camber angle due to the rake angle of the kingpin. The front wheel side view is depicted in *figure*. The pneumatic trail lies is in the order of 35 mm. So, the mechanical trail has been taken as **35 mm** too.

The king pin follows the front wheel axis. The caster angle is arc tan (21/240) =**5^{o}**.

1.9 Ride Rate, Roll Rate, Wheel Rates and Spring Rates Definitions:

Spring Rate: Force per unit displacement for the suspension spring alone. For coil springs, this is measured axially along the centerline. For bar springs it is measured by the attachment arm. Unit is Newton/Meter

Wheel Rate: Vertical force per unit vertical displacement at the location alongside the spindle to the wheel centerline measured relative to the vehicle chassis.

Roll Rate: Moment resisting body roll per degree of body roll. The resistance to the body role is provided by the ride rates, axle track width, and anti-roll bar.

Ride Rate: Vertical force per the unit vertical displacement of the tire ground contact reference point relative to the vehicle chassis. For infinitely stiff tire ride rate and wheel rate are equal.

1.10 Introduction to Upright of an Automobile:

In automotive suspension, an upright is that part which contains the hub of the wheel or spindle, and attaches to all the suspension components. It is called an upright, spindle or steering knuckle.

The wheel and tire assembly attaches to the hub or spindle of the steering knuckle and the tire / wheel rotates while being held by the post on a stable surface of motion.

The stanchions are shown attached to the upper control arm and the lower control arm. The wheel is shown attached to the steering knuckle at its midpoint. Note the protruding arm of the steering knuckle and the steering knuckle to which the steering assembly is attached to turn the steering knuckle and wheels.

Fig 1.6 Upright held between the A-arms at wheel end

In a non-drive suspension, the upright usually has a spindle onto which the brake drum or brake rotor attaches. The wheel/tire assembly then attaches to the supplied lug studs, and the whole assembly rotates freely on the shaft of the spindle.

In a drive suspension, the upright has no spindle, but rather has a hub into which is affixed the bearings and shaft of the drive mechanism. The end of the drive mechanism would then have the necessary mounting studs for the wheel/tire and/or brake assembly. Therefore, the wheel assembly would rotate as the drive Shaft dictates. It would not turn freely by itself, but only if the shaft was disengaged from the transaxle or differential.

1.11 Introduction of Springs:

A spring is an elastic object for storing mechanical energy. Springes are usually made of spring steel. Small springs can be coiled from pre-cured material, while large springs are made from annealed steel and cured after manufacture. Some non-ferrous metals are also used, such as phosphor bronze and titanium for parts that require corrosion resistance, and beryllium copper (because of their low electrical resistance) for springs that carry current.

When a spring is compressed or stretched, the force exerted by the spring is proportional to the change in length. The velocity or spring constant of a spring is the change in force exerted by the spring divided by the change in spring deflection.

That is, the gradient of the force-deflection curve. Tension or compression springs have a unit of force divided by distance, for example lbf / in or N / m. Some torsion springs are the unit of force and distance divided by the angle. B. N m / rad or ftlbf / degree. The reciprocal of the spring constant is slack. That is, if the spring elongation is 10 N / mm, the slack is 0.1 mm / N. The stiffness (or velocity) of a series spring is additive, as is the compliance of a series spring.

Depending on the design and required operating environment, any material can be used to construct a spring, so long as the material has the required combination of rigidity and elasticity: technically, a wooden bow is a form of spring

1.12 Introduction to A-Arms:

In automotive suspension, an automobile's control arm or wishbone or A-arm or A-frame is a nearly flat and roughly triangular suspension member (or sub-frame), that pivots in two places. The base of the triangle attaches at the frame and pivots on a bushing.

Fig 1.7 A-arms between chassis and wheels

The upper control arm can clearly be seen at the top portion of the suspension components in the Fig 1.7, where it is the silver part horizontally attached to the frame inside the red body portion and connecting to the steering knuckle near the side of the tire's wheel rim.

Note the roughly A-shaped design with the top of the A near the tire and the bottom two points connected to the frame inside the body's space.

1.13 Introduction of Rocker:

In a suspension system, specific type of mechanism used for transferring load through the push/pull rod coming from the wheel assembly and transfers this load to the spring damper assembly by pivoting at a point on the chassis is a rocker.

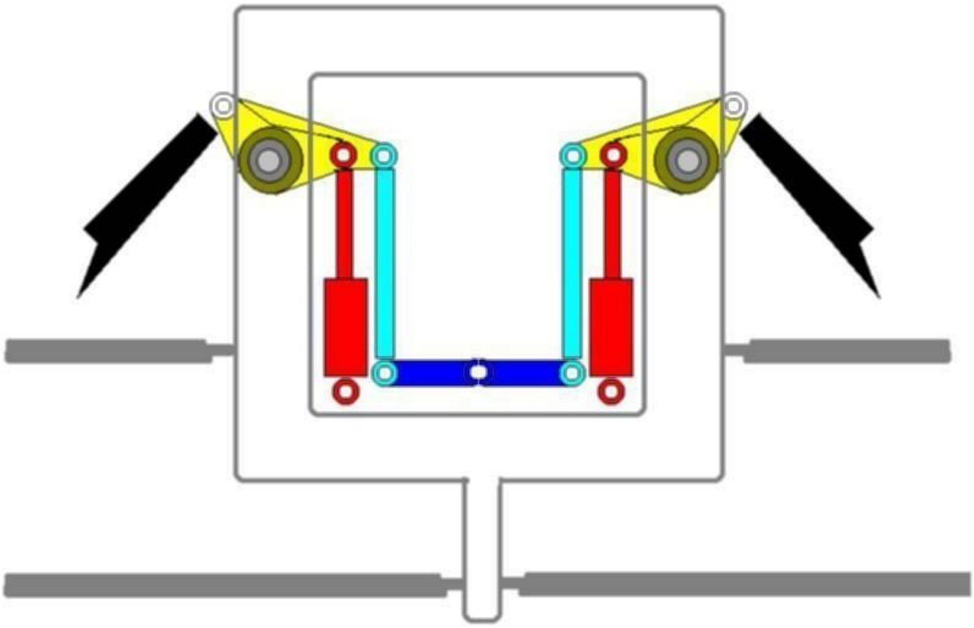

Fig 1.8 Basic function of rocker A-arm shown in yellow color

From Figure 1.8, the yellow body is a seesaw connected to a push rod (black), and the seesaw rotates with the frame. The adjacent node in the figure is for the spring damper assembly (red) and the last node is for the stabilizer bar (blue). The following arrangements do not require the nodes to be fixed in the above order, but can be done according to your suspension design needs.

From the figure, the working of the rocker can be explained as when there is any bump or droop at the wheel assembly, the push rod rotates the rocker about the pivot created on the chassis and transfers this load on to the spring damper assembly therefore absorbing the shocks and avoiding any roll created with the help of ARB.

Thus, the functioning of suspension components was understood. Along with the suspension components various parameters like scrub radius, king pin angle, wheel rate, ride rate etc. were understood and calculated. The calculations were later used in design calculations chapter. The suspension components will be discussed more in detail in the next chapter literature surveys. The knowledge of this chapter will be more helpful in the upcoming chapters to get better view at the working of suspension system.

CHAPTER 2
LITERATURE REVIEW

Dr. Htay Htay win, Dr. Nwe Ni tun et al., the purpose of this paper is about the shear stress and deformation produced in the spring at the loading condition is in limit, so it is safe. Relative error in maximum shear stress to applied load compared to values calculated using a simple analytical formula in the bibliography. Shear stress shows the maximum value inside each coil and clearly shows the stress distribution. From the above analysis, it was observed that the rigidity of the suspension springs increased and the load bearing capacity of the system increased. Reducing the number of coils will reduce the weight of the spring. It has been observed that the weight of the system is reduced under the same load conditions. Therefore, we have realized a lightweight system that helps improve the fuel efficiency of the vehicle. Based on modeling and analysis, it can be concluded that the new design is suitable for implementation. Since the model is from an existing vehicle, different load factors can be used to propose and analyze new designs.

Asad Ahmad, Md. Hassaan et al., the purpose of this thesis is giving the basic design and analytical concepts of the suspension system used in student formula car. After designing the control arm, steering knuckle and spring, we used CATIA to create a 3D model and analyzed it on the ANSYS workbench. The integrated assembly was simulated with LOTUS software. The results obtained in the simulation correspond to the designed parameters. And after analyzing the different suspension components, this paper establishes a methodology for analyzing the different components.

Julian Wisnu Wirawan, Ubaidillah et al., the purpose of this thesis is the characteristics of the designed suspension, which vary depends on the purpose of the vehicle. The suspension provided achieved a 1: 2 and 1: 1 travel ratio for the front and rear suspensions.

The designed front suspension was still too soft for a racing car. A racing car needs to be about 1: 1 for the car to be stable. The conclusion of the study is that the suspension is still too soft. This problem can be solved by giving the linkage more space to work best with better geometry.

Anshul Kunwar, Mohit Nagpal et al., the primary objective of this paper is to identify the design parameter of a vehicle suspension system with the proper study of vehicle dynamics. This paper will also help you study and analyze the impact parameters of suspension systems. A thorough analysis of the suspension system reveals that the front roll center is 20mm and the rear roll center is 40mm. The results are ideal for improving the stability of the car on the truck and the comfortable ride of the driver. Analysis reveals that the design is safe enough to enter the manufacturing process.

This paper concludes about the stability of the car based on roll centre at front and rear sections. But, only considering the roll centre to determine the stability of the car is not appropriate. Thus, including the other factors like centre of gravity etc. can provide better understanding about the stability of the vehicle.

Ajay Kumar, Rahul Rajput et al., the purpose of this thesis work is to explain the idea for some critical components in which more than one method are employed to ensure the durability of design. These analysis steps need to be followed many times in single component to obtain a perfect model. Since, the paper covers methodology for analysis of different components along with introduction to load calculation in most of the cases. This document will help you design and analyze suspension components for better design results, and help you perform analysis of various suspension components in ANSYS for optimal results. Finally, you get the best design with high factor of safety and minimal stress on the components.

This document describes the different parameters used in the design and different methods for performing analysis of suspension components. The design parameters were taken from the FSAE standard, but adding theoretical calculations will give you a better understanding.

Abishek.R.K, Aswin Krishna.M et al., The objective of this paper is to design the whole suspension system which Includes double wishbone system coil springs, wishbones and dampers for student formula cars. First, calculate the spring constant using ride comfort, wheel speed, and kinetic ratio. Then, with that help, set the dimensions of the spring. The sprue is then CAD modeled with creopametric. Then fix the UCA, LCA axes, UBJ, and LBJ points and sketch the arm. Then required damper should be selected according to the spring stiffness. First the wheel base and track, camber and castor angles in the car is set and then the wheel and spring rates are calculated. Then with those values springs and control arms are designed and structural analysis is done in springs and control arms. Along with this modal analysis in the springs is performed. Then the response is generated in MATLAB and multi body dynamics is performed in hyper works and graphs are generated and are found to be smooth. Thus, the design and analysis of suspension system is completed

H.G. Phakatkar, Chinmay potdar et al., This paper summarizes the design for suspension of Formula student car. The proposal made in this paper is very extensive and includes the problems of racing car design in knowledge of the rules and basics of vehicle chassis construction. This paper details the concepts of analysis for design verification. The project's Student Formula focuses on student collaboration. Important for communication between members of the racing team. Participation at the event is the final result and goal of the year-round endeavors.

This paper only discuses about the design of chassis and suspension but the load factors, design parameters were not discussed. Even the design procedure and theoretical calculations need to be added.

CHAPTER 3

DESIGN CALCULATIONS

3.1 Coil Spring Designing:

Coil over shock absorber:

1. It helps in reducing the center of gravity, while cornering, braking and acceleration.

Working principle:

- Work done = Force x Displacement

$$W = F \times Y$$

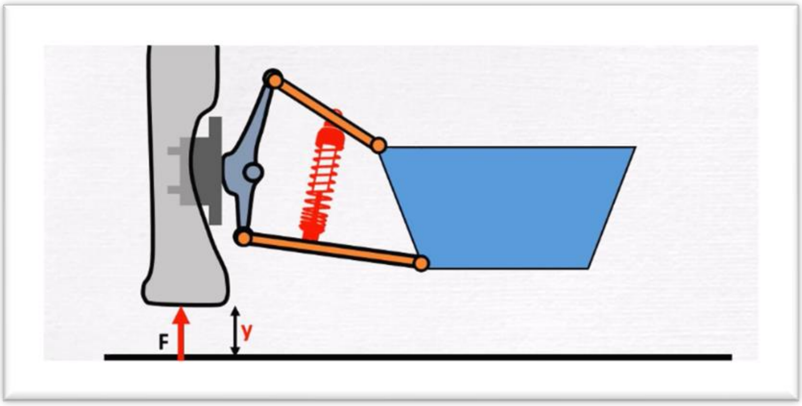

Fig 3.1 Coil Spring

- Here, the Fig 3.1 shows the attachment of coil over shock absorber
- Elastic potential Energy (Spring) (i.e., work done stored in the spring)

$$U = 0.5 \times K \times Y_1^2$$

- K = Spring Stiffness
- Y_1 = Displacement of Spring

Now during the expansion condition:

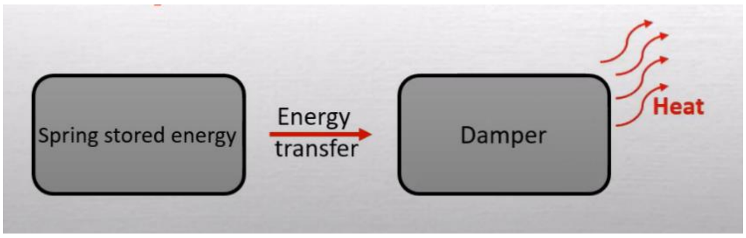

Fig 3.2 Energy Potenial

3.2 Damping Calculations:

Installation Ratio:

- Installation ratio relates the change in length of a force producing device to the change in vertical wheel centre movement.
- From the Fig 3.3,

I.R = Y_2/Y_1

Or

I.R = X_2/X_1

- It is the ratio of Spring displacement to the Wheel displacement.

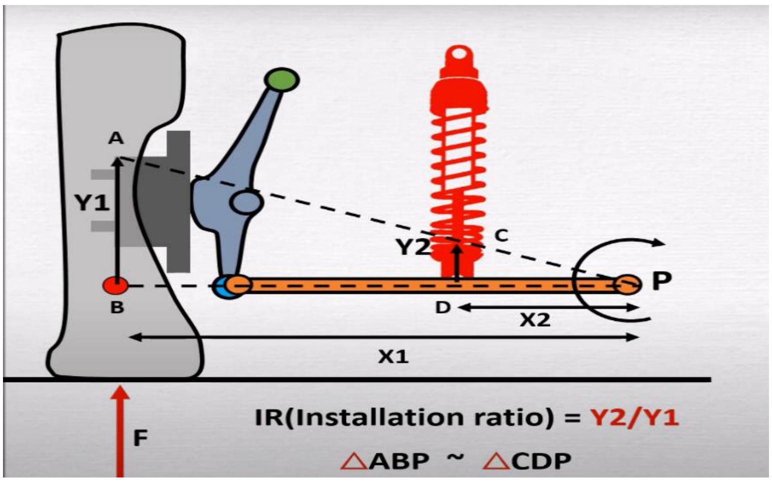

Fig 3.3 Spring Displacement

Spring rate (K_s):

- Amount of weight needed to compress the spring by 1 inch or 1 mm.

Wheel rate (K_w):

- Vertical force per unit vertical displacement of wheel centre.

Relationship between $K_{w,}$ IR and K_s:

- $Kw = Fs \times (\Delta IR/\Delta\delta) + K_s \times (IR)^2$
  - Kw = Wheel rate (N/mm)
  - Fs = Spring force (N/m)
  - Ks = Spring rate (N/mm)
  - IR = Installation ratio

Case 1:

Installation ratio is constant

- $\Delta IR = 0 \rightarrow K_w = K_s \times (IR)^2$

Case 2:

Coil over inclined at angle θ

- $K_w = K_s \times (IR)^2 \times \cos\theta$

Ride rate (K_r):

- $$K_r = \frac{K_w \times K_t}{K_w + K_t}$$
  - K_t = Tire rate (Can be taken from available tire data)

Damping Calculation Steps:

Determination of spring rate:

The spring rate depends upon the installation ratio. In the given FSAE car the installation ratio is approximately 1. i.e., the ratio of the travel of spring to the travel of the wheel is approximately 1.

$\omega = 1/(2\ \pi)*\surd\{K_s/M\}$

- M = Sprung mass on particular wheel

Higher the frequency, the stiffer the ri

Vehicle Type	Ride frequency
Passenger cars	0.5-1.0Hz
Rally Cars	1.5-2.0Hz
Non-Aero, moderate downforce Formula cars	1.5-2.5Hz
Race cars with up to 50% total weight in max downforce capability	2.5-3.5Hz
High downforce race cars with more than 50% of their weight in max downforce	3.5-5.0Hz

Fig 3.4 Ride frequency table

- Generally, from the Fig 3.4 the frequency for the race cars vary from 1.9Hz to 3.5Hz.
- So, we are assuming the value to be 1.93Hz.
- From the models of various F1 cars like pertronix and red bull, we can take the values of M to be 78 Kgs to 81 Kgs for the front section and 113 Kgs to 117 Kgs for the rear section.
- Therefore, we can take the values of M as
    - M (front section) = 79.2 Kgs
    - M (rear section) = 115.8 Kgs

Therefore, the front spring rate, $\mathbf{K_{SF} = 12N/mm}$ and the rear spring rate,

$\mathbf{K_{SR} = 17N/mm}$

Determination of Wheel Rate:

Tire and the suspension coil spring are two springs in series. The series combination of two springs, one acting between the chassis and the wheel center, the other acting between wheel center and ground. The first represents what we call as wheel rate.

- $K_S = K_W \times (IR)^2$

As the value of IR is equal to 1, the values of Wheel rate will be same as the value of spring rate

- Hence, the Front wheel rate, $\mathbf{K_{wf} = 12N/mm}$ **or 64.4847lbs/in.**
- The Rear wheel rate, $\mathbf{K_{wr} = 17N/mm}$ **or 94.2721lbs/in.**

3.3 Determination of Ride Rate:

The ride rates are chosen compatible to the wheel loads, that are rates stiff enough so that the outside suspension does not bottom or contact the bump stops. The important point is to avoid the bottoming the suspensions as this will cause a sudden change in wheel loading and upset the balance of the car.

From the FSAE rules the car must have at least 25.4mm jounce travel and 25.4mm rebound travel. Taking factors of safety into consideration, the total travel is chosen to be 60mm, i.e., 30mm in jounce and 30mm in rebound, with a ground clearance of 150mm. using the total ride travel the calculations are done.

- $K_r = \dfrac{K_w \times K_t}{K_w + K_t}$

- K_t = Tire rate (Can be taken from available tire data)
- K_t = 1250.8 lbs/in (From Hoosier tire specifications)

- So, putting the values K_w and K_t
  - K_{rf} = 11.9098 N/mm or 67.99 lbs/in
  - K_{rr} = 17.8558 N/mm or 101.9566 lbs/in

3.4 Free Vibration under Damping:

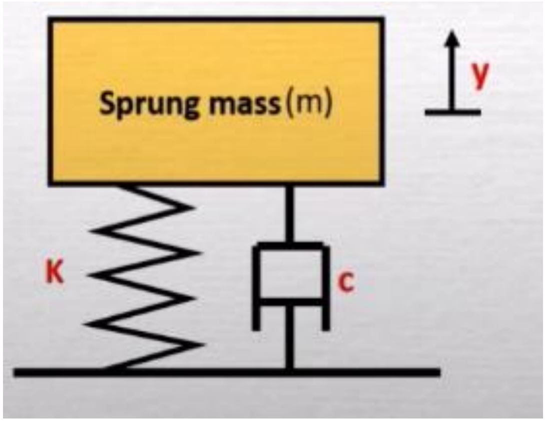

Fig 3.5 Free Vibration under damping

From the Fig 3.5, the following conditions are derived

- $C^2 < 4mK$ → Under damping
- $C^2 = 4mK$ → Critical damping
- $C^2 > 4mK$ → Over damping

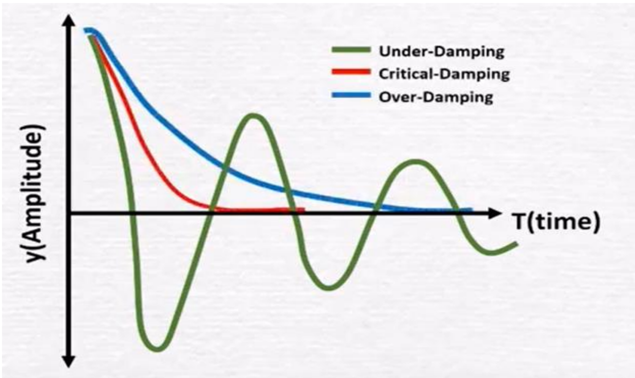

Fig 3.6 Graph b/w Amplitude and Time

So, from the Fig 3.6 the Oscillations of the displacement of spring mass becomes zero fastest in the case of critical damping. So, The damping constant is taken near critical damping constant

- $C_C = \sqrt{(4mK_s)}$

Damping ratio (ζ) = C / C_C

- $\zeta = 1 \rightarrow$ Critical damping
- $\zeta < 1 \rightarrow$ Under damping
- $\zeta > 1 \rightarrow$ Over damping

3.5 Terminology of Coil spring:

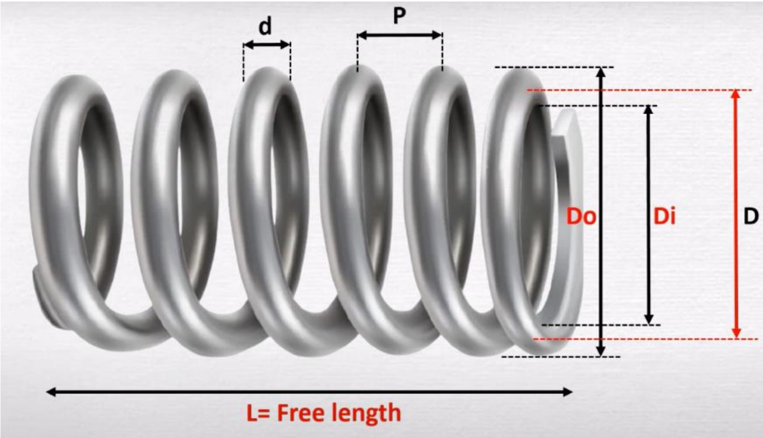

Fig 3.7 Coil Spring

- From the Fig 3.7,
- L = Free length
- D_0 = Outer diameter of spring
- D_i = Inner diameter of spring
- D = Mean diameter of spring
- d = Wire diameter
- P = Pitch

Active coils: Active coils are the coils that will be a part in deflection, when load is applied on material.

The types of ends of coil spring:

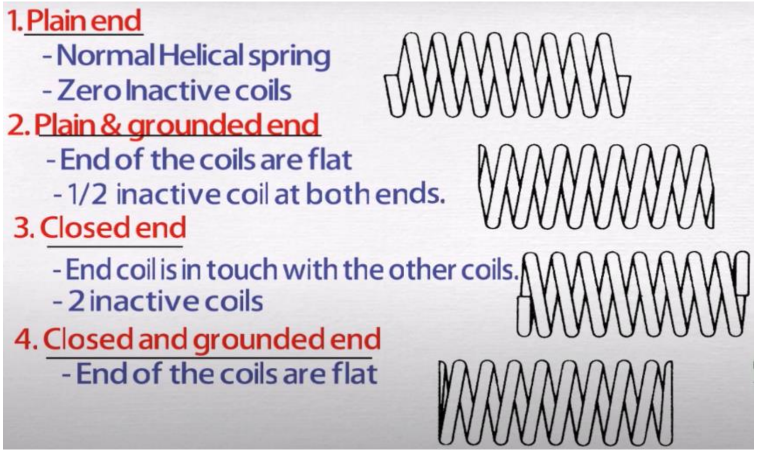

Fig 3.8 Types of Coil

3.6 Designing of Coil spring:

Spring Index:

- C = D/d = Spring index

Spring Index	Condition
0 - 3.9	Spring Cannot Manufacture
4 - 5	Spring Falls in the Difficult Category, higher cost
6-12	Best Manufacturing Range Lower Cost
13 - 15	Spring is Just Above Good But Not Quite in The Difficult Range
15 - 25	Spring Falls in the Difficult Category, higher cost
Above 25	Very expensive and difficult to manufacture, tolerances are larger

Fig 3.9 Spring Index

- Spring index basically tells us, how tight the spring and strength of the spring.
- From the Fig 3.9, we can assume the value of C as 9.
- So, we have ratio of mean diameter of spring and wire diameter.
  - $D = (D_0 + D_i)/2$ ------ (1)
  - $d = (D_0 - D_i)/2$ ------ (2)

Dividing the equation 1 by equation 2, we get

$D_0 / D_i = 5 / 4$

- The number of active coils will be in the range between 8 to 12
- So, we assume it to be 9

Deflection:

- $$\delta = \frac{8WD^3 \times N}{G \times d^4}$$

Spring rate → $K_s = W/ \delta$

$$K_s = \frac{G \times d^4}{8D^3 \times N}$$

- N = Active coils
- G = Shear modulus of material

Replacing the values of K_s, G and N as 12N/mm, 80KN/mm^2 and 9 (AISI 1040 Steel), value of d will be found.

- d = 16.33 mm
- D = 146.97 mm

- Total Coils = N_t = Active coils + Inactive coils
- Solid length = $L_s = N_t \times d$

$= L_s = 9 \times 16.33 = 146.97$ mm

- Pitch = L / N

$= 376.97 / 9 = 41.8$ mm

- Free length = $L = L_s + 0.15(\delta_{max}) + \delta_{max}$

  $= 146.97 + 200 + 0.15(200)$

  $= 376.97$ mm

CHAPTER 4

DESIGN OF SUSPENSION SYSTEM

4.1 Introduction to Modelling Software (CATIA V5):

Computer Aided Design (CAD) is the use of computer software to design a product or an object.

Computer Aided Manufacturing (CAM) is the use of computer software and hardware to plan, manage, and control the operation of a manufacturing facility.

Computer Aided Engineering is the use of computer software to solve engineering problems and analyse products created using CAD.

CATIA is an acronym for Computer Aided Three-Dimensional Interactive Application. This is the most capable, powerful and popular CAD software. H. Computer-aided design software. It is created, developed and owned by Dassault Systèmes in France. IBM was CATIA's leading marketer until 2010.

Because of its high usability, CATIA certification is one of the most popular and sought-after certifications in market.

4.2 Application of CATIA:

CATIA is among the very few software which has its application in about every industrial sector. Mainly used by the designer team. The design team for each organization needs to make a digital copy of each object it manufactures. This digital copy is very easy to make using CATIA. It is mostly found in companies who are associated with design and manufacturing of products.

4.3 Importance of CATIA:

The two recent versions of CATIA that is CATIA V5 & V6 are renowned as the world's leading design product suite. This was possible because of two important factors; one was the industrial usability and other was the huge and constant investment made for product development and innovation. CATIA has a wide range of uses in the industry, so demand for this product is increasing.

CATIA was originally used in automotive and aerospace engineering to design various parts and designs throughout the structure. Due to technological advances and various uses of CATIA, it can be used not only for designing parts and structures, but also for various purposes.

As a result, the fields of application of CATIA have expanded. Mass production and ease of use make CATIA affordable in the industry. Many big brands use CATIA as the basic engineering design platform.

4.4 Load Calculations:

4.4.1 Static loads:

Each wheel supports a share of total mass at its corner. Hence it is necessary to know the amount of load on each wheel. This can be calculated based on the position of center of gravity. The longitudinal position of the center of gravity and the height of the centre of gravity from the ground are recorded to calculate the loads acting on each wheel.

The overall mass of the car approximately came out to be 400kg. Initially the longitudinal position of the centre of gravity is taken into consideration. It is at a distance of 0.63m from the rear axle and is 0.34m above the ground.

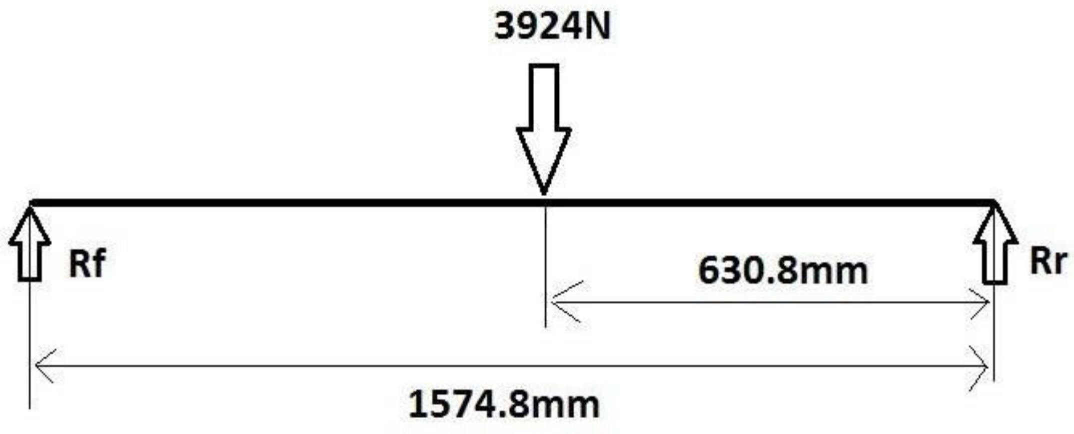

Fig 4.1 Free body diagram of car in rest in its side view

The procedure for calculating the reaction force of a simple support beam determines Rf (front end reaction force) and Rr (rear end reaction force). The reaction of the front wheels and the reaction of the rear wheels are shared by the two wheels. Since the vehicle is symmetrical with respect to the vertical axis, the front wheel load is evenly distributed across both wheels. Therefore, the static load for each wheel is shown below.

Table 4.1 Fore acting on a component

S. No	Component Name and Force symbol	Force Acting
1	FZFL (Front left wheel)	785N
2	FZFR (Front right wheel)	785N
3	FZRL (Rear left wheel)	1177N
4	FZRR (Rear right wheel)	1177N

4.4.2 Loads on wheel during a Bump:

The impact force of F1 can be up to 10G. That is, the load on each wheel can reach 10 times the static load on that wheel. The impact force of FSAE vehicles is assumed to be 2G or less. The main reason is that FSAE cars do not have a lot of aerodynamic force. In addition, go-kart trucks are much slower, which slows down bumps. Table 2 shows the maximum impact force values per wheel.

Hence maximum bump force is tabulated below for each wheel.

Table 4.2 Normal Loads during bumps

Front wheel (each)	1570N
Rear wheel (each)	2354N

4.5 Procedure for Modeling of Suspension System in CATIA

STEP 1: Go to Start in that Mechanical Design and it shows some modules then click Part Design for part design window.

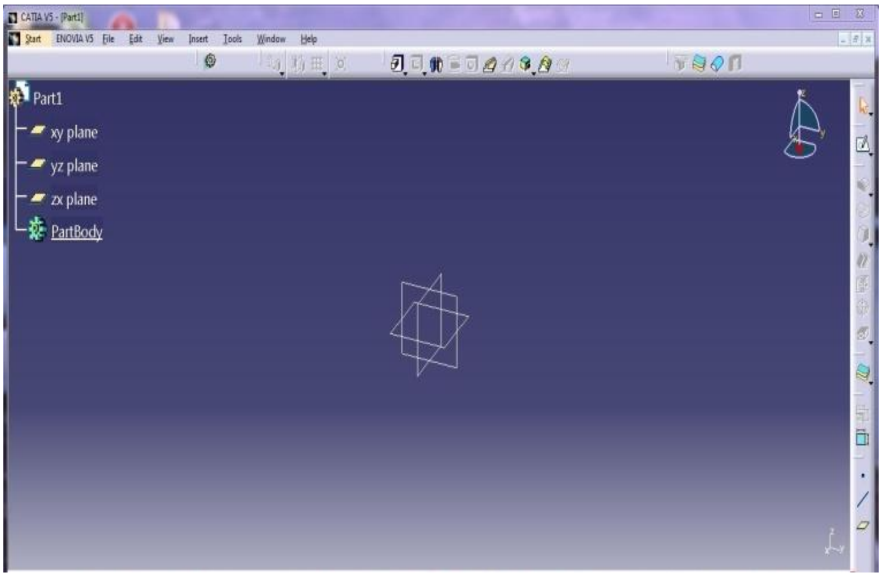

Fig 4.2 Part design window

STEP 2: Select the front plane and click on sketcher icon then sketcher window will get highlighted where we can draw 2D sketch.

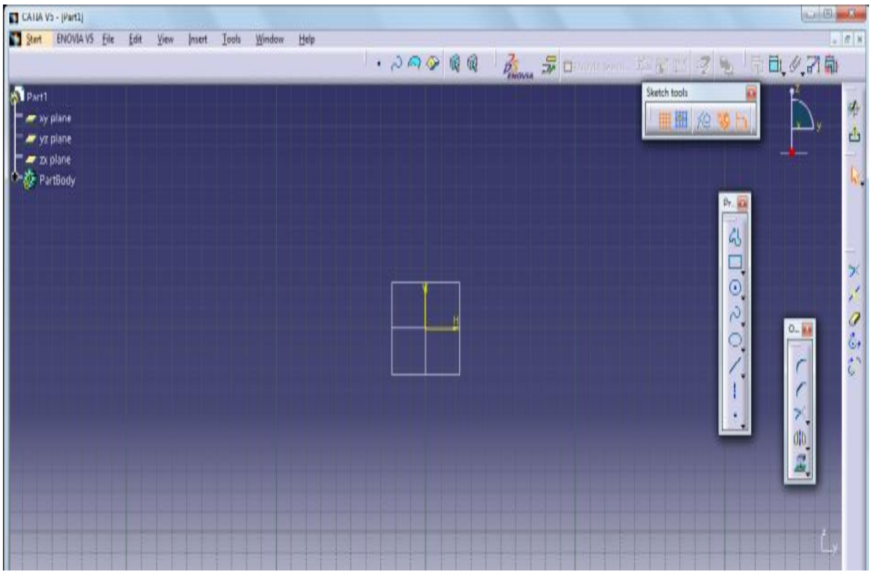

Fig 4.3 Sketcher window

STEP 3: Create the outline of the lower arm and upper arm using the dimensions based on FSAE standards.

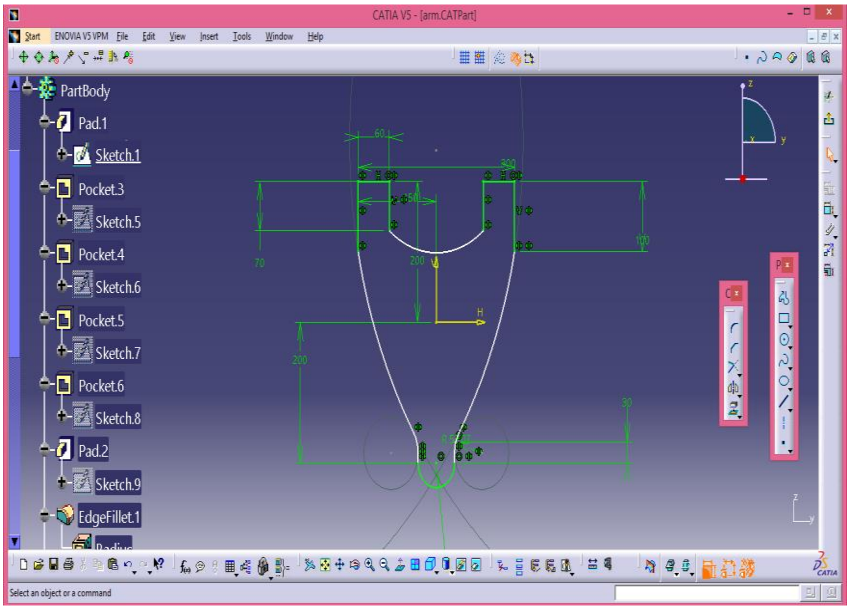

Fig 4.4 Lower Arm Sketch

STEP 4: Convert the 2D model into 3D solid model by using the extrude command.

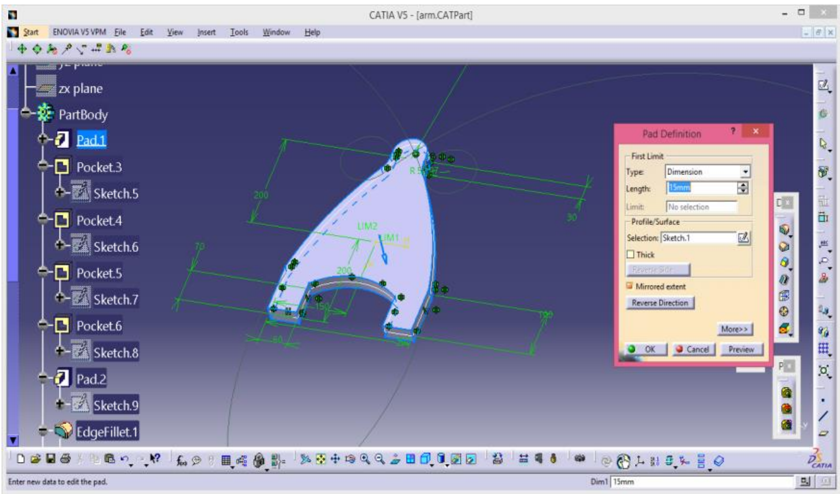

Fig 4.5 Converting into 3D Solid model

STEP 5: Now create a pocket at the necessary sections of the arm by applying the shell option.

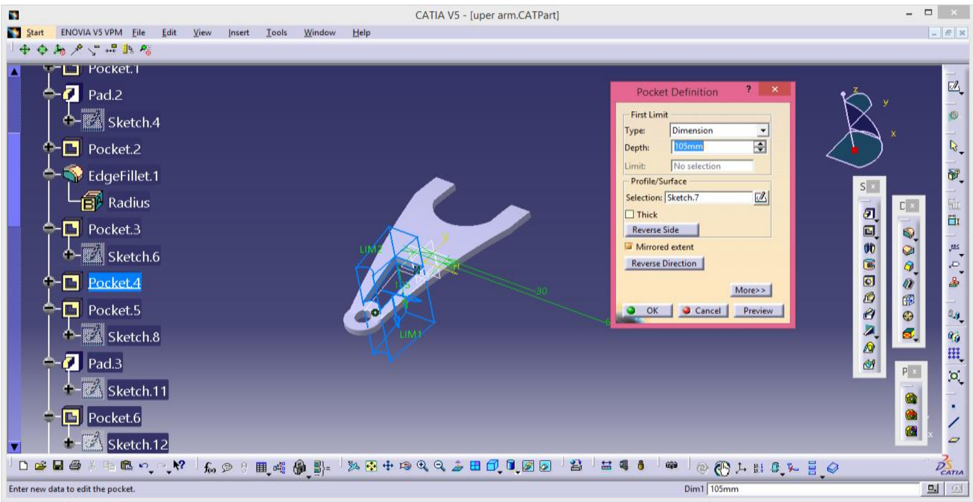

Fig 4.6 Upper arm

STEP 6: After completion of the CAD model, considering various views of the component to get better understanding.

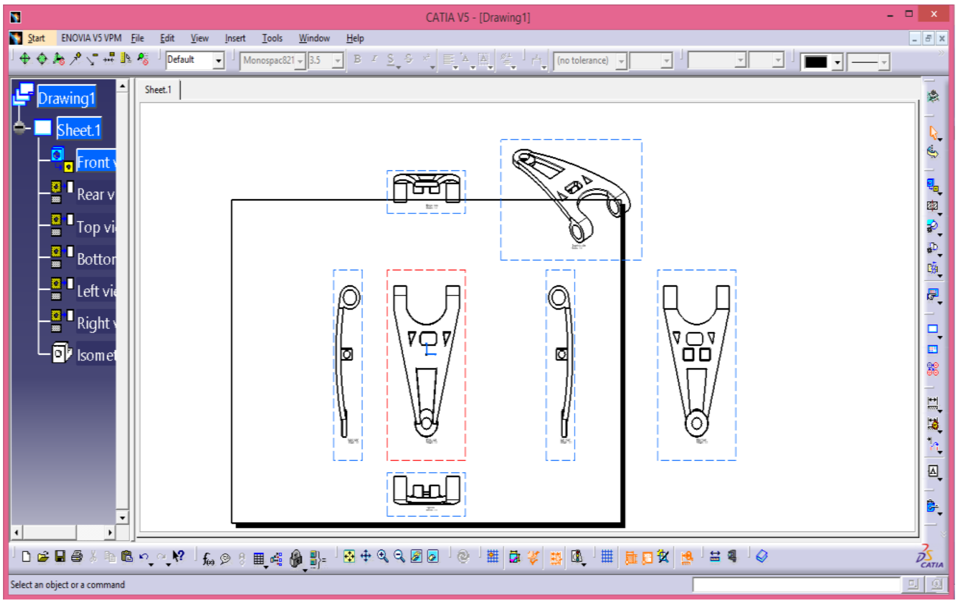

Fig 4.7 Different Views of Upper arm

4.6 Design of Coil Over Spring Absorber:

STEP 1: Select the front plane and click on sketcher icon then sketcher window will get highlighted where we can draw 2D sketch.

STEP 2: Create a 2D sketch for coil over shock absorber.

STEP 3: Use ‘Helix’ command for drawing the spring profile.

STEP 4: Now select the part design and make a space on the individual starting point.

STEP 5: Now draw the cross-section spring and complete the spring model by ‘Rib’ command.

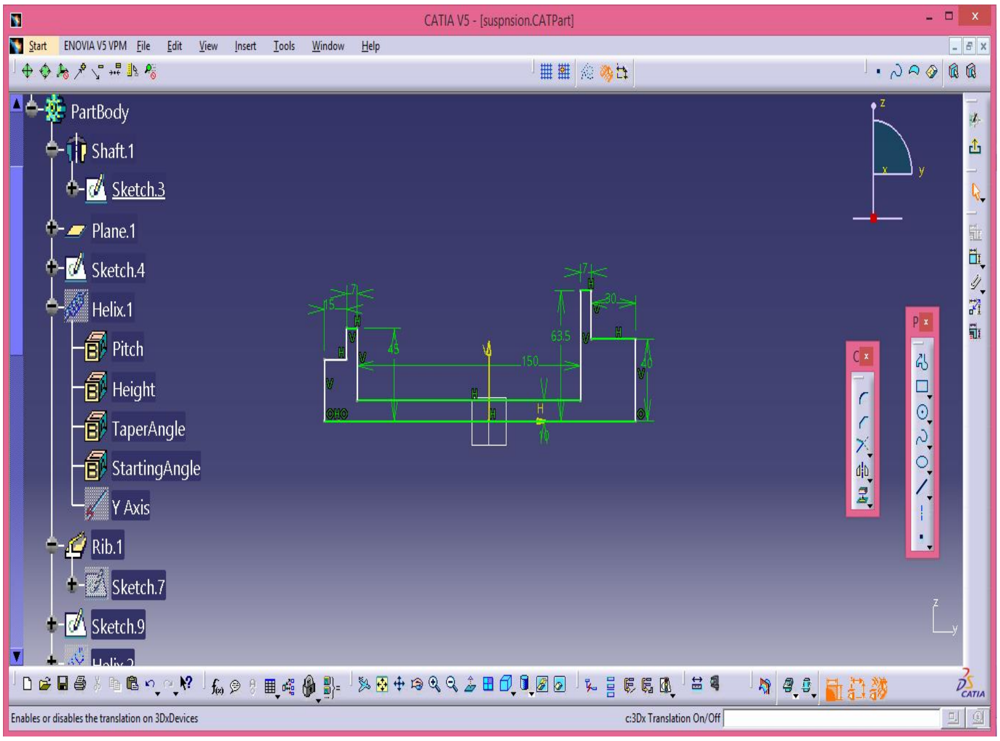

Fig 4.8 Sketch of coil over spring absorber

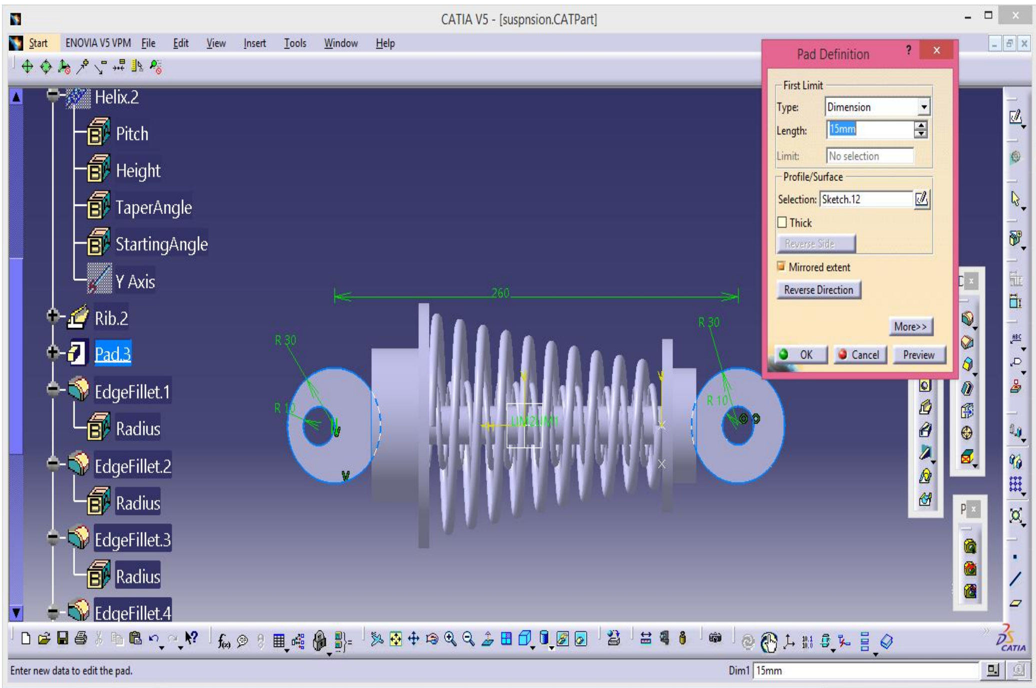

Fig 4.9 3D model of coil over spring absorber

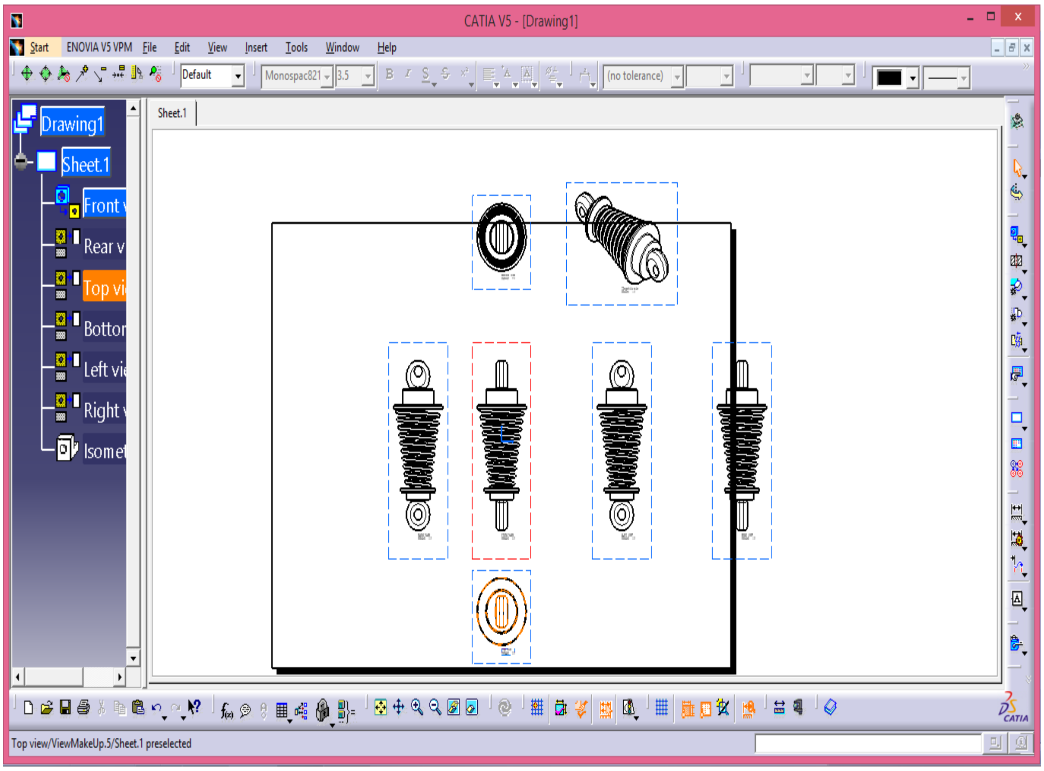

Fig 4.10 Different views of coil over spring absorber

Thus, the modelling of the suspension components is successfully completed. The suspension components like lower wishbone, upper wishbone and hub section are modeled based on FSAE standards with standard dimensions. The coil over shock absorber is modeled based on theoretical calculations performed. And the shock absorber is attached to lower and upper wishbone unlike race cars. The structural analysis of the suspension components will be carried out in the next chapter.

CHAPTER 5
ANALYSIS OF THE SUSPENSION SYSTEM

5.1 Introduction to ANSYS:

ANSYS 19.2 helps engineering teams accelerate innovation in any environment and create cutting-edge designs by harnessing new workflows and dynamic capabilities across ANSYS flagship suites. Updates in ANSYS Cloud offerings, such as virtual desktop infrastructure support, unites ANSYS flagship powerful workflows deliver a streamlined user experience with data and configuration management, dependency visualization and decision support, and advanced workflow features that are easy to use. These resources help users create large, complex designs more easily and quickly than ever before, increase productivity and drive the development of quality products and equipment on the market.

5.2 ANSYS 2020 R2Capabilities:

3D Design, Additive Manufacturing, Electronics, Embedded Software, Fluids, Materials, Optical, Platform, Semiconductors, Structures, Systems, Metaphysics and rapid Design Exploration for the Next Generation of Product Design.

5.3 Introduction to Ansys Meshing:

A mesh is a network of cells and points. Meshing is part of engineering simulation and can be used as a discrete local approximation for larger regions by decomposing complex geometries and models into simple elements. The net can be of any shape and size, depending on its shape and placement.

5.3.1 Purpose of Meshing:

In general, the continuous object to be analyzed has an infinite number of points, and it is virtually impossible to obtain a finite number of equations and converge to a solution. Therefore, the first step in finite element analysis is to perform the basic calculations of a problem with a finite number of points, and then interpolate the results obtained over the entire range. Each cell in the network represents a separate solution to the equation. Combining this across the network makes for a complete network solution

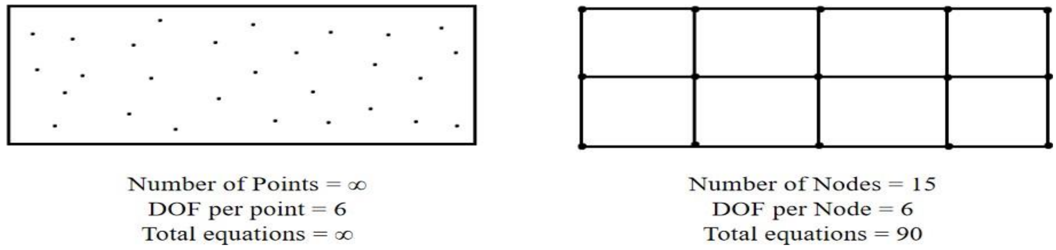

Fig. 5.1 Showing the possible number of equations before and after meshing

5.3.2 Classification of Elements: Based on Type of Grids

Based on the types of grids, depending on the layout of the mesh it can be classified into structured and unstructured mesh.

- **Unstructured Mesh**
- Unstructured networks are networks with general connectivity that are arbitrary (random) in structure, so connectivity between elements must be defined and stored. Unstructured element types include triangular (2D) or tetrahedral (3D) elements.
- **Structured Mesh**

Structured meshes are meshes with implicit (absolute) connectivity whose structure allows for easy identification of elements and nodes. In General, the structured meshes have quadrilateral elements (2D) or hexahedral (3D) elements.

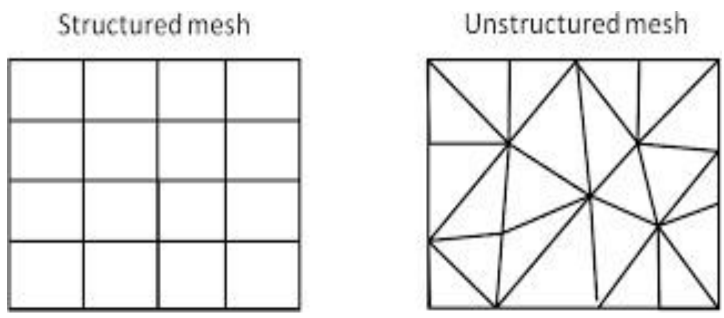

Fig.5.2 Structured and Unstructured mesh

5.3.3 Factors affecting the element types:

The element type for meshing mainly depends on three factors.

- Geometry Shape and size.
- Type of Analysis.
- Time Allotted for the project.

5.3.4 Meshing in ANSYS:

Working with the Mesh Module

- The First step in Working with the Mesh module is to open the module by selecting the type of solver or the component from the tool box.
- After Importing (the solver in project schematic page), the Mechanical module can be accessed by double clicking on the Model.

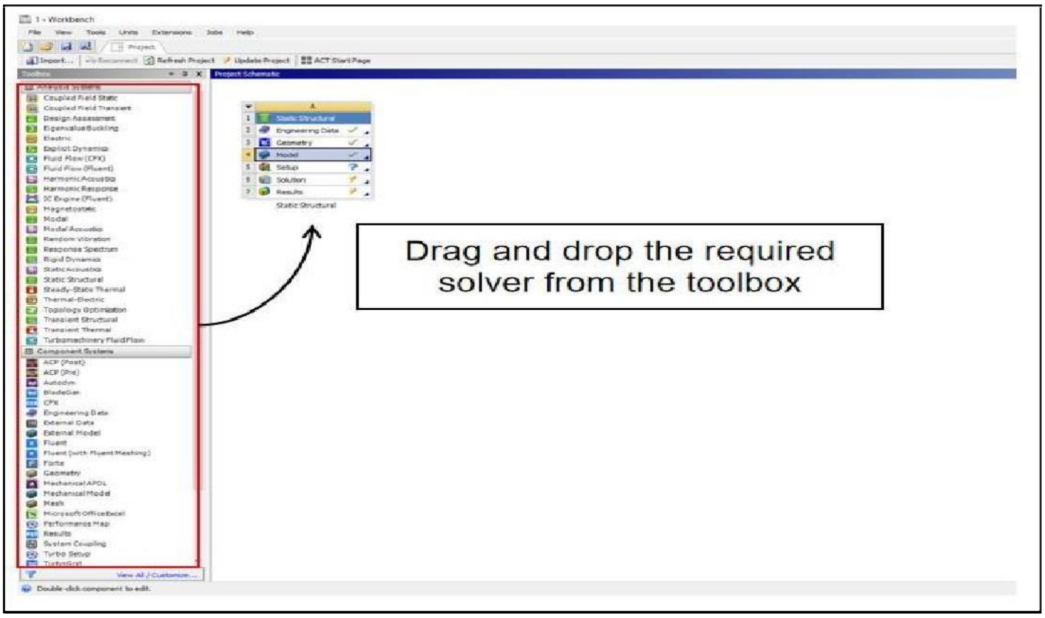

Fig.5.3 Image showing the layout of the project schematic page in ANSYS

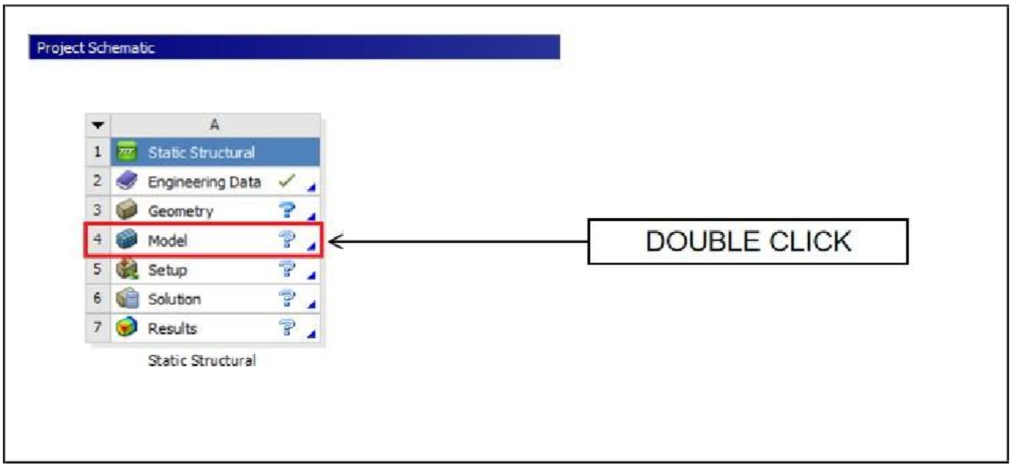

Fig.5.4 Image showing the path to open Mechanical Module in ANSYS

- Once the Mechanical module is open, the mesh settings can be accessed by clicking on the mesh from the project tree or by right clicking on the mesh for applying the necessary settings.

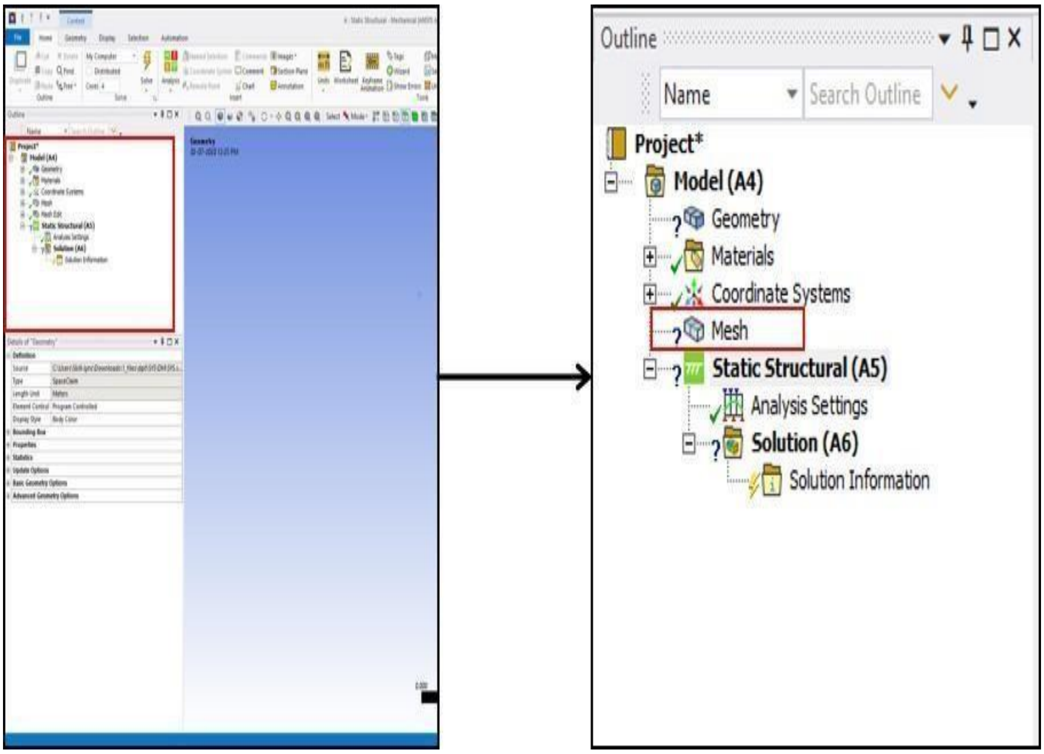

Fig.5.5 Image highlighting the location of Mesh Settings

5.3.5 Various types of local mesh controls

The Method can be inserted for the geometry from the Model tree outline by right clicking on **Mesh → Insert → Method.**

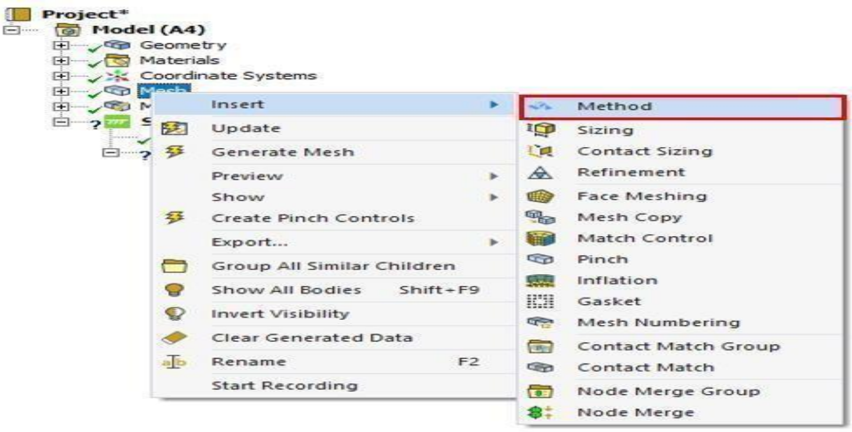

Fig.5.6 Image of path to insert a method for the mesh in ANSYS

The Method of mesh selected plays a vital role in the accuracy and reliability of the analysis performed. The algorithm tends to use hexahedral elements for the solid models and Quad elements by default for surface models. If the algorithm finds that the model is un sweep able, then it uses tetrahedral elements as default mesh. When we open the Method, the following appears in the details tab area.

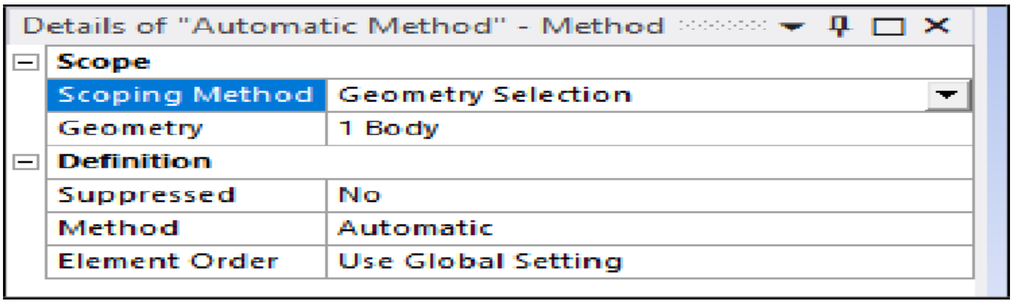

Fig.5.7 Image of details available for the ‘Method’ to be applied

5.3.6 Meshing images:

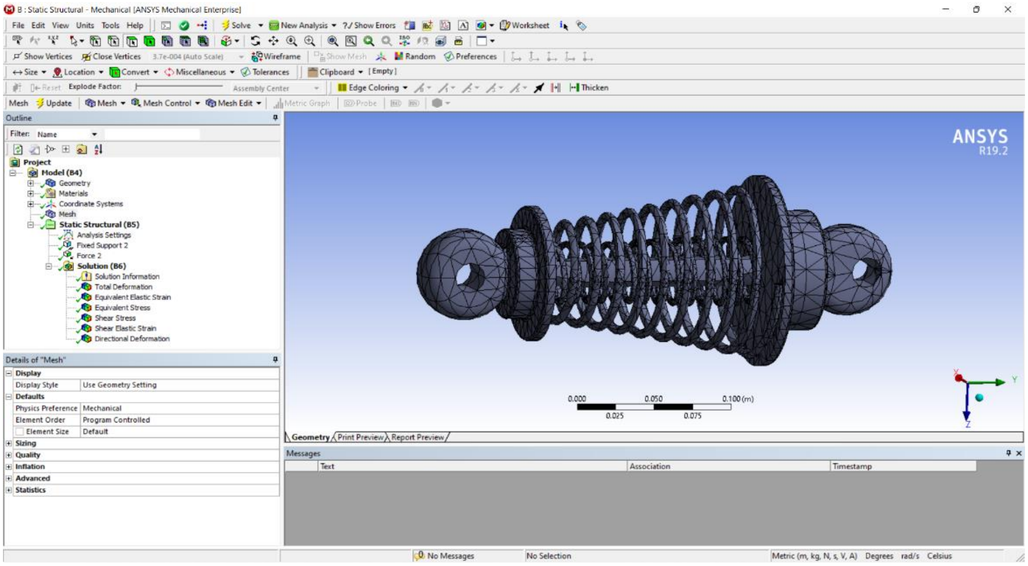

Fig.5.8 Meshing 1

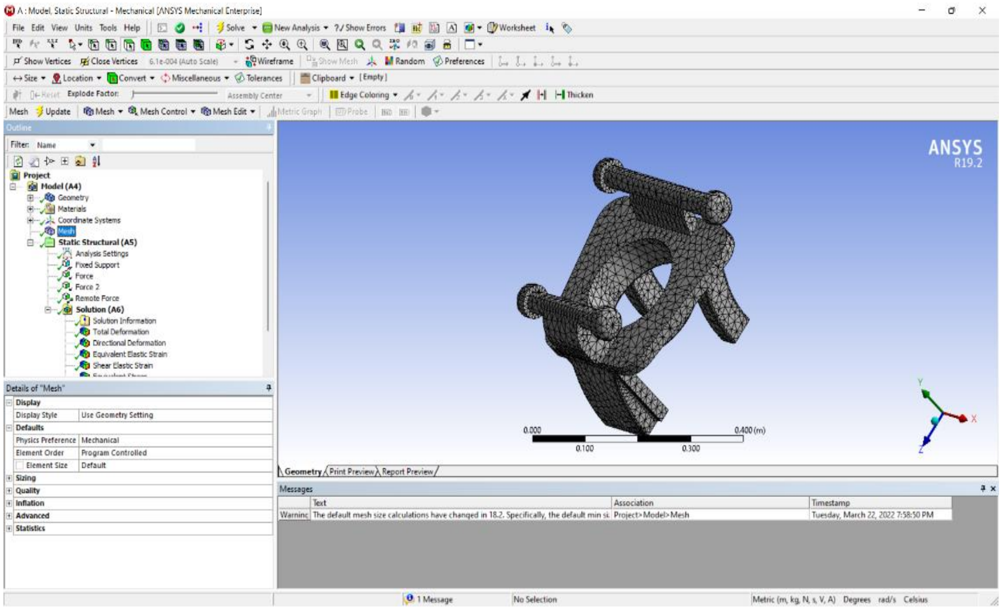

Fig.5.9 Meshing 2

Here in this investigation structural analysis of Suspension system components is carried out by using coarse mesh. After meshing 4 different types of materials were considered for the suspension components and 6 different types of materials were considered for coil over shock absorber. Thus, the structural analysis is carried out on all the suspension components with various materials.

5.4 Materials Selected to Perform Analysis:

5.4.1 Arms:

1. AISI 1040 Steel
2. Aluminum
3. Gray Cast Iron
4. Stainless Steel

5.4.2 Hub section:

1. Al 7075 Alloy
2. CA6NM Steel
3. Gray Cast Iron
4. Mild Steel

5.4.3 Coil over shock absorber:

1. AISI 302 Steel
2. Titanium Alloy
3. Structural Steel
4. Chrome Vanadium
5. Beryllium Copper
6. Phosphor Bronze

CHAPTER 6
ANALYSIS OF SUSPENSION COMPONENTS

6.1 Analysis of Coil Over Shock Absorber:

6.1.1 AISI 302 Steel:

Table 6.1 Analysis using AISI 302 Steel for Spring

Material/ Properties	AISI 302 Steel		
Min/Max/Avg	**Minimum**	**Average**	**Maximum**
Total Deformation (m)	0	1.6036×10^{-6}	3.3297×10^{-6}
Directional Deformation (m)	-1.4586×10^{-7}	-6.491×10^{-9}	6.7643×10^{-8}
Equivalent Elastic Strain	2.9202×10^{-9}	8.9587×10^{-7}	1.8755×10^{-5}
Shear Elastic Strain	-1.2672×10^{-5}	-1.0635×10^{-8}	1.1472×10^{-5}
Equivalent Stress (Pa)	102.87	1.5625×10^{5}	3.6101×10^{6}
Shear Stress (Pa)	-9.5532×10^{5}	-801.81	8.6487×10^{5}

6.1.2 Result:

The material used is AISI 302 Steel and the average deformation got during this analysis is 1.603×10^{-6} m, and the average equivalent elastic strain is 8.958×10^{-7}, average stress is 1.5625×10^{5} Pa, average shear stress is -801.81 Pa and average directional deformation is -6.491×10^{-9} m.

6.1.3 Titanium Alloy:

Table 6.2 Analysis using Titanium Alloy for spring

Material/ Properties	**Titanium Alloy**		
Min/Max/Avg	**Minimum**	**Average**	**Maximum**
Total Deformation (m)	0	3.1985×10^{-6}	6.6517×10^{-6}
Directional Deformation (m)	-2.8183×10^{-7}	-9.1987×10^{-9}	1.6574×10^{-7}
Equivalent Elastic Strain	5.6256×10^{-9}	1.8297×10^{-6}	3.8545×10^{-5}
Shear Elastic Strain	-2.7306×10^{-5}	-2.3381×10^{-8}	2.4728×10^{-5}
Equivalent Stress (Pa)	117.54	1.5966×10^{5}	3.6868×10^{6}
Shear Stress (Pa)	-9.6375×10^{5}	-825.22	8.7274×10^{5}

6.1.4 Result:

The material used is Titanium Alloy and the average deformation got during this analysis is 3.198×10^{-6} m, and the average equivalent elastic strain is 1.829×10^{-6}, average stress is 1.5966×10^{5} Pa, average shear stress is -825.22 Pa and average directional deformation is -9.1987×10^{-9} m.

6.1.5 Structural Steel:

Table 6.3 Analysis using Structural Steel for Spring

Material/ Properties	Structural Steel		
Min/Max/Avg	**Minimum**	**Average**	**Maximum**
Total Deformation (m)	0	$1.5448x10^{-6}$	$3.2087x10^{-6}$
Directional Deformation (m)	$-1.3971x10^{-7}$	$-5.9207x10^{-9}$	$6.8747x10^{-8}$
Equivalent Elastic Strain	$2.7816x10^{-9}$	$8.6716x10^{-7}$	$1.8185x10^{-5}$
Shear Elastic Strain	$-1.2448x10^{-5}$	$-1.0503x10^{-8}$	$1.1275x10^{-5}$
Equivalent Stress (Pa)	105.41	$1.5698x10^{5}$	$3.6266x10^{6}$
Shear Stress (Pa)	$-9.5752x10^{5}$	-807.93	$8.6733x10^{5}$

6.1.6 Result:

The material used is Structural Steel and the average deformation got during this analysis is $1.544x10^{-6}$ m, and the average equivalent elastic strain is $8.671x10^{-7}$, average stress is $1.5698x10^{5}$ Pa, average shear stress is -807.93 Pa and average directional deformation is $-5.9207x10^{-9}$ m.

6.1.7 Chrome Vanadium:

Table 6.4 Analysis using Chrome Vanadium for spring

Material/ Properties	**Chrome Vanadium**		
Min/Max/Avg	**Minimum**	**Average**	**Maximum**
Total Deformation (m)	0	1.495×10^{-6}	3.1048×10^{-6}
Directional Deformation (m)	-1.3561×10^{-7}	-5.8976×10^{-9}	6.4789×10^{-8}
Equivalent Elastic Strain	2.7068×10^{-9}	8.3713×10^{-7}	1.7541×10^{-5}
Shear Elastic Strain	-1.193×10^{-5}	-1.0039×10^{-8}	1.0804×10^{-5}
Equivalent Stress (Pa)	104.07	1.5661×10^{5}	3.6181×10^{6}
Shear Stress (Pa)	-9.5643×10^{5}	-804.88	8.6614×10^{5}

6.1.8 Result:

The material used is Chrome Vanadium and the average deformation got during this analysis is 1.495×10^{-6} m, and the average equivalent elastic strain is 8.3713×10^{-7}, average stress is 1.5661×10^{5} Pa, average shear stress is -804.88 Pa and average directional deformation is -5.8976×10^{-9} m.

6.1.9 Beryllium Copper:

Table 6.5 Analysis using Beryllium Copper for spring

Material/ Properties	**Beryllium Copper**		
Min/Max/Avg	**Minimum**	**Average**	**Maximum**
Total Deformation (m)	0	2.3766×10^{-6}	4.9364×10^{-6}
Directional Deformation (m)	-2.1494×10^{-7}	-9.1087×10^{-9}	1.0576×10^{-7}
Equivalent Elastic Strain	4.2795×10^{-9}	1.3341×10^{-6}	2.7977×10^{-5}
Shear Elastic Strain	-1.915×10^{-5}	-1.6159×10^{-8}	1.7347×10^{-5}
Equivalent Stress (Pa)	105.41	1.5698×10^{5}	3.6266×10^{6}
Shear Stress (Pa)	-9.5752×10^{5}	-807.93	8.6733×10^{5}

6.1.10 Result:

The material used is Beryllium Copper and the average deformation got during this analysis is 2.3766×10^{-6} m, and the average equivalent elastic strain is 1.3341×10^{-6}, average stress is 1.5698×10^{5} Pa, average shear stress is -807.93 Pa and average directional deformation is -9.1087×10^{-9} m.

6.1.11 Phosphor Bronze:

Table 6.6 Analysis using Phosphor Bronze for spring

Material/ Properties	**Phosphor Bronze**		
Min/Max/Avg	**Minimum**	**Average**	**Maximum**
Total Deformation (m)	0	2.8007×10^{-6}	5.8218×10^{-6}
Directional Deformation (m)	-2.4944×10^{-7}	-9.1396×10^{-9}	1.3809×10^{-7}
Equivalent Elastic Strain	4.9536×10^{-9}	1.5909×10^{-6}	3.3465×10^{-5}
Shear Elastic Strain	-2.3457×10^{-5}	-1.9993×10^{-8}	2.1252×10^{-5}
Equivalent Stress (Pa)	112.57	1.5869×10^{5}	3.6647×10^{6}
Shear Stress (Pa)	-9.6172×10^{5}	-819.71	8.7132×10^{5}

6.1.12 Result:

The material used is Phosphor Bronze and the average deformation got during this analysis is 2.8007×10^{-6}m, and the average equivalent elastic strain is 1.5909×10^{-6}, average stress is 1.5869×10^{5} Pa, average shear stress is -819.71 Pa and average directional deformation is -9.1396×10^{-9} m.

6.1.13 Analysis Pictures:

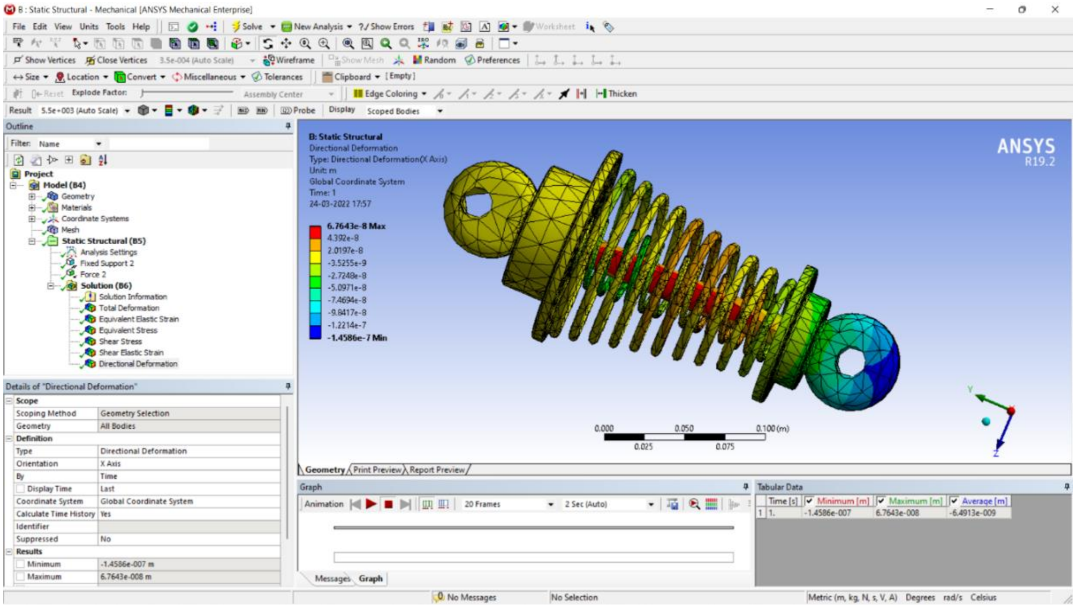

Fig 6.1 Analysis of Spring -View 1

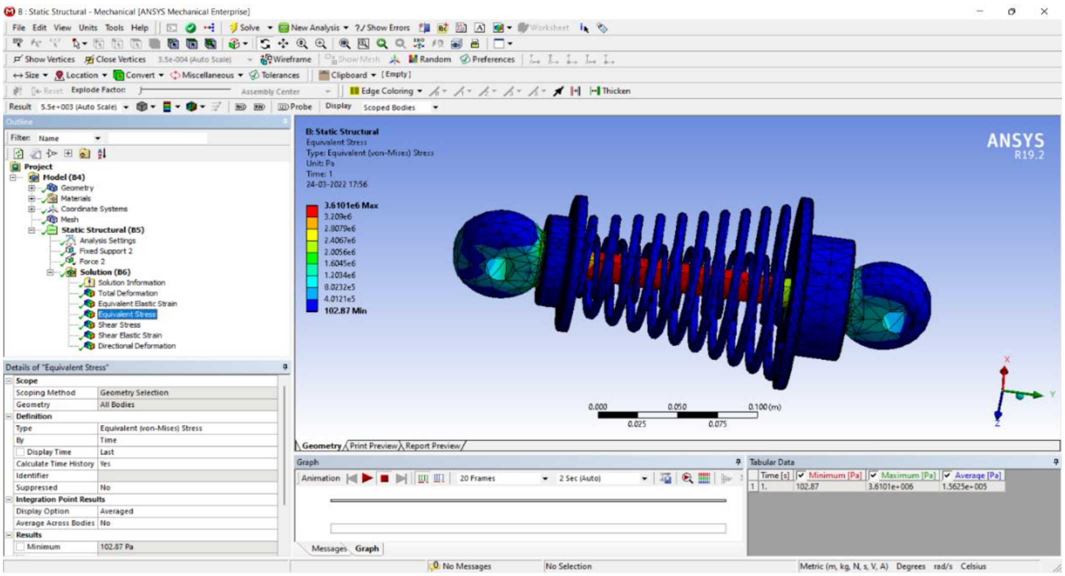

Fig 6.2 Analysis of Spring -View 2

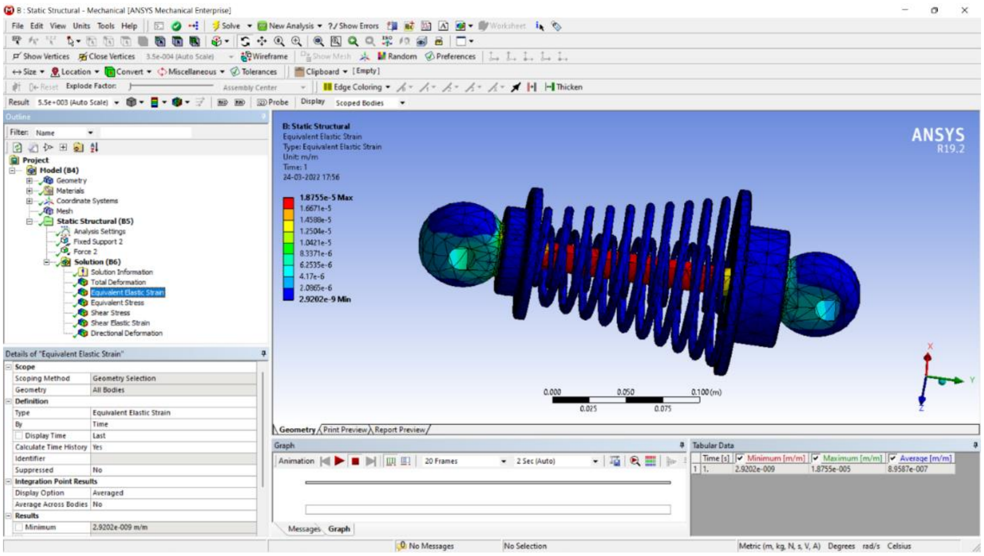

Fig 6.3 Analysis of Spring -View 3

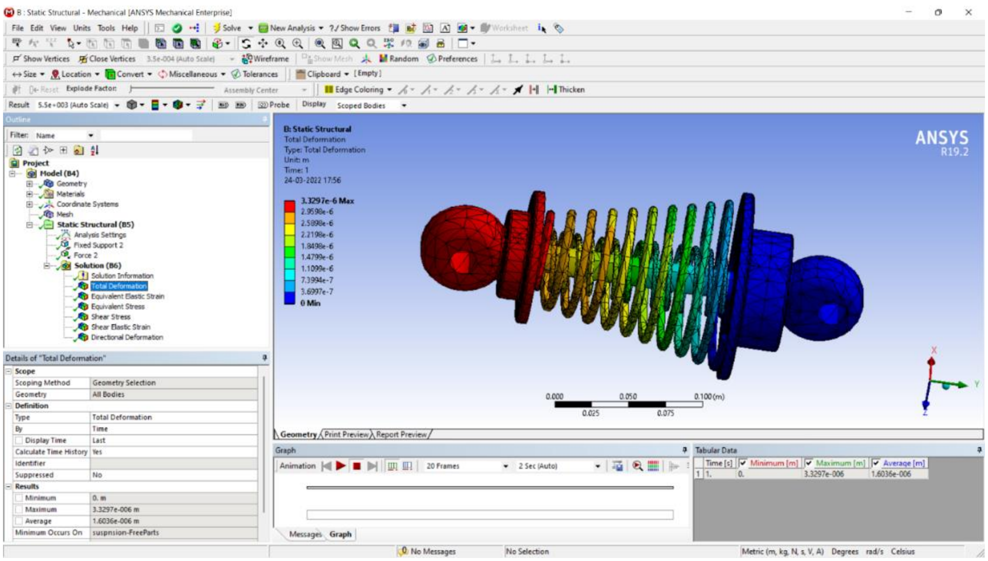

Fig 6.4 Analysis of Spring -View 4

6.2 Analysis of Upper Arm:

6.2.1 AISI 1040 Steel:

Table 6.7 Analysis using AISI 1040 Steel for upper arm

Material/ Properties	**AISI 1040 Steel**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	5.4411×10^{-3}
Directional Deformation (m)	-8.3423×10^{-6}	4.3856×10^{-5}
Equivalent Elastic Strain	5.7151×10^{-7}	1.6688×10^{-3}
Shear Elastic Strain	-9.7703×10^{-4}	6.4989×10^{-4}
Equivalent Stress (Pa)	95485	3.2307×10^{8}
Shear Stress (Pa)	-7.891×10^{7}	5.2491×10^{7}

6.2.2 Result:

The material used is AISI 1040 Steel and the maximum deformation got during this analysis is 5.4411×10^{-3} m, and the maximum equivalent elastic strain is 1.6688×10^{-3}, maximum stress is 3.2307×10^{8} Pa, maximum shear stress is 5.2491×10^{7} Pa and maximum directional deformation is 4.3856×10^{-5} m.

6.2.3 Aluminium:

Table 6.8 Analysis using Aluminium for upper arm

Material/ Properties	**Aluminium**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	1.6803×10^{-2}
Directional Deformation (m)	2.5763×10^{-5}	1.3544×10^{-4}
Equivalent Elastic Strain	1.765×10^{-6}	5.1537×10^{-3}
Shear Elastic Strain	-3.0173×10^{-3}	2.007×10^{-3}
Equivalent Stress (Pa)	95485	3.2307×10^{8}
Shear Stress (Pa)	-7.8914×10^{7}	5.2491×10^{7}

6.2.4 Result:

The material used is Aluminium and the maximum deformation got during this analysis is 1.6803×10^{-2} m, and the maximum equivalent elastic strain is 5.1537×10^{-3}, maximum stress is 3.2307×10^{8} Pa, maximum shear stress is 5.2491×10^{7} Pa and maximum directional deformation is 1.3544×10^{-4} m.

6.2.5 Gray Cast Iron:

Table 6.9 Analysis using Gray Cast Iron for upper arm

Material/ Properties	**Gray Cast Iron**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	1.0387×10^{-2}
Directional Deformation (m)	-1.5175×10^{-5}	8.1765×10^{-5}
Equivalent Elastic Strain	1.0522×10^{-6}	1.0522×10^{-6}
Shear Elastic Strain	-1.8632×10^{-3}	1.224×10^{-3}
Equivalent Stress (Pa)	91679	3.2345×10^{8}
Shear Stress (Pa)	-8.0058×10^{7}	5.2592×10^{7}

6.2.6 Result:

The material used is Gray Cast Iron and the maximum deformation got during this analysis is 1.0387×10^{-2} m, and the maximum equivalent elastic strain is 1.0522×10^{-6}, maximum stress is 3.2345×10^{8} Pa, maximum shear stress is 5.2592×10^{7} Pa and maximum directional deformation is 8.1765×10^{-5} m.

6.2.7 Stainless Steel:

Table 6.10 Analysis using Stainless Steel for upper arm

Material/ Properties	**Stainless Steel**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	5.9206×10^{-3}
Directional Deformation (m)	-9.2955×10^{-6}	4.8343×10^{-5}
Equivalent Elastic Strain	6.3363×10^{-7}	1.8132×10^{-3}
Shear Elastic Strain	-1.0629×10^{-3}	7.1187×10^{-4}
Equivalent Stress (Pa)	97507	3.2292×10^{8}
Shear Stress (Pa)	-7.8294×10^{7}	5.2439×10^{7}

6.2.8 Result:

The material used is Stainless Steel and the maximum deformation got during this analysis is 5.9206×10^{-3} m, and the maximum equivalent elastic strain is 1.8132×10^{-3}, maximum stress is 3.2292×10^{8} Pa, maximum shear stress is 5.2439×10^{7} Pa and maximum directional deformation is 4.8343×10^{-5} m.

6.2.9 Analysis Pictures:

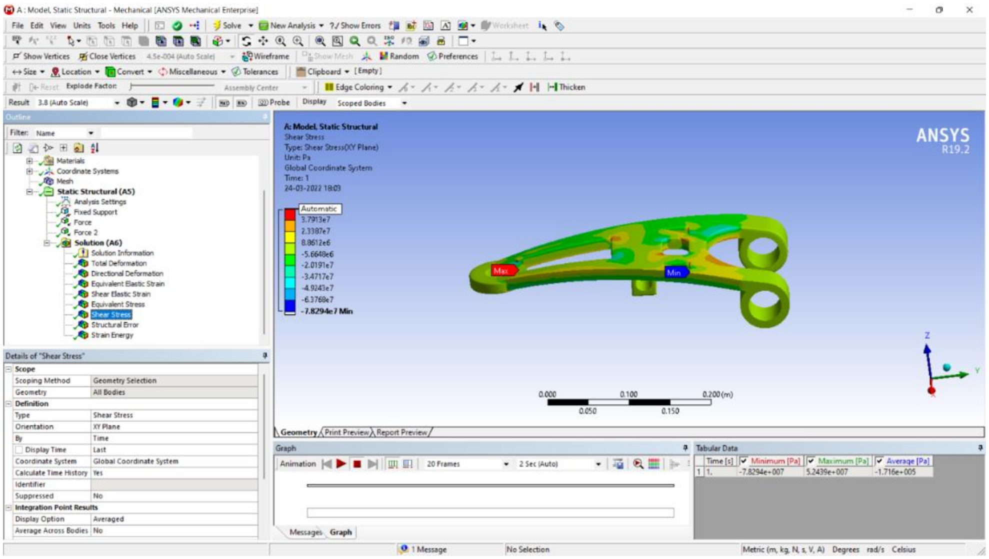

Fig 6.5 Analysis of Upper Arm -View 1

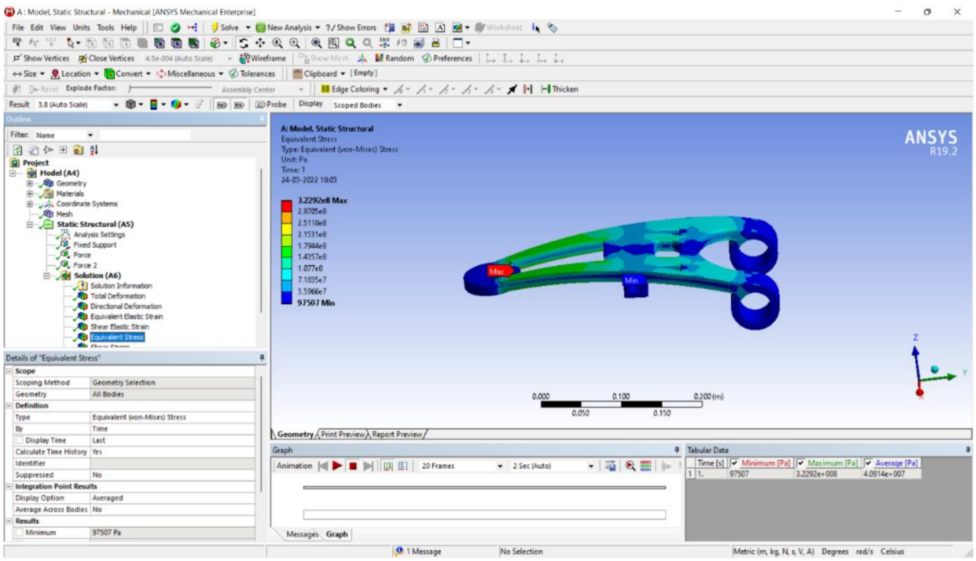

Fig 6.6 Analysis of Upper Arm -View 2

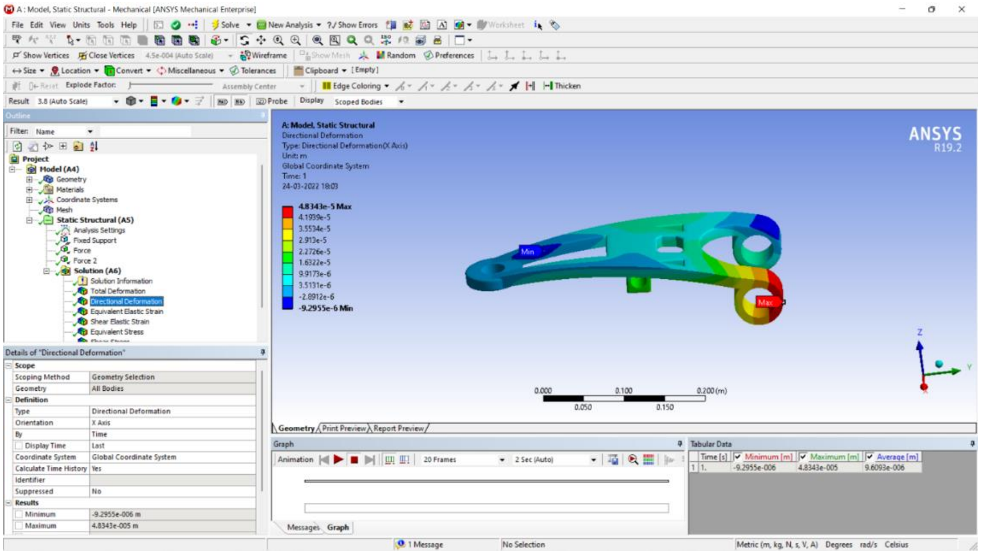

Fig 6.7 Analysis of Upper Arm -View 3

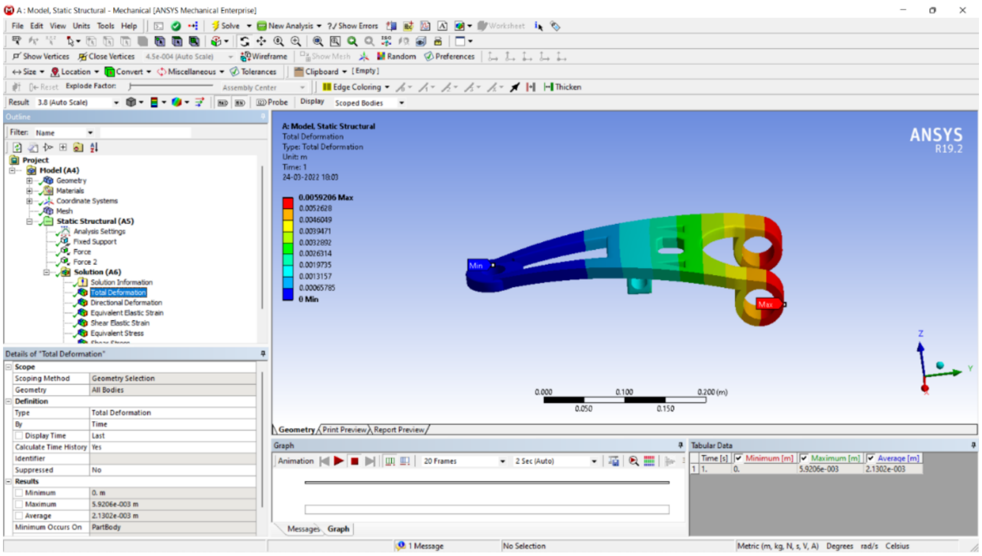

Fig 6.8 Analysis of Upper Arm -View 4

6.3 Analysis of Lower Arm:

6.3.1 AISI 1040 Steel:

Table 6.11 Analysis using AISI 1040 Steel for lower arm

Material/ Properties	**AISI 1040 Steel**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	$2.1902\text{x}10^{-5}$
Directional Deformation (m)	$-3.4193\text{x}10^{-7}$	$1.9257\text{x}10^{-5}$
Equivalent Elastic Strain	$4.6604\text{x}10^{-8}$	$5.4013\text{x}10^{-5}$
Shear Elastic Strain	$-1.3048\text{x}10^{-5}$	$1.5856\text{x}10^{-5}$
Equivalent Stress (Pa)	5745.7	$1.1025\text{x}10^{7}$
Shear Stress (Pa)	$-1.0538\text{x}10^{6}$	$1.2807\text{x}10^{6}$

6.3.2 Result:

The material used is AISI 1040 Steel and the maximum deformation got during this analysis is $2.1902\text{x}10^{-5}$ m, and the maximum equivalent elastic strain is $5.4013\text{x}10^{-5}$, maximum stress is $1.1025\text{x}10^{7}$ Pa, maximum shear stress is $1.2807\text{x}10^{6}$ Pa and maximum directional deformation is $1.9257\text{x}10^{-5}$ m.

6.3.3 Aluminium:

Table 6.12 Analysis using Aluminium for lower arm

Material/ Properties	Aluminium	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0.	$6.7639x10^{-5}$
Directional Deformation (m)	$-1.056x10^{-6}$	$5.9472x10^{-5}$
Equivalent Elastic Strain	$1.4392x10^{-7}$	$1.668x10^{-4}$
Shear Elastic Strain	$-4.0294x10^{-5}$	$4.8967x10^{-5}$
Equivalent Stress (Pa)	5745.7	$1.1025x10^{7}$
Shear Stress (Pa)	$-1.0538x10^{6}$	$1.2807x10^{6}$

6.3.4 Result:

The material used is Aluminium and the maximum deformation got during this analysis is $6.7639x10^{-5}$ m, and the maximum equivalent elastic strain is $1.668x10^{-4}$, maximum stress is $1.1025x10^{7}$ Pa, maximum shear stress is $1.2807x10^{6}$ Pa and maximum directional deformation is $5.9472x10^{-5}$ m.

6.3.5 Gray Cast Iron:

Table 6.13 Analysis using Gray Cast iron for lower arm

Material/ Properties	**Gray Cast Iron**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	4.1716×10^{-5}
Directional Deformation (m)	-6.4991×10^{-7}	3.6626×10^{-5}
Equivalent Elastic Strain	8.8949×10^{-8}	1.0391×10^{-4}
Shear Elastic Strain	-2.4877×10^{-5}	3.0291×10^{-5}
Equivalent Stress (Pa)	5808.7	1.1058×10^{7}
Shear Stress (Pa)	-1.0689×10^{6}	1.3016×10^{6}

6.3.6 Result:

The material used is Gray Cast Iron and the maximum deformation got during this analysis is 4.1716×10^{-5} m, and the maximum equivalent elastic strain is 1.0391×10^{-4}, maximum stress is 1.1058×10^{7} Pa, maximum shear stress is 1.3016×10^{6} Pa and maximum directional deformation is 3.6626×10^{-5} m.

6.3.7 Stainless Steel:

Table 6.14 Analysis using Stainless Steel for lower arm

Material/ Properties	**Stainless Steel**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	$2.3857x10^{-5}$
Directional Deformation (m)	$-3.7294x10^{-7}$	$2.0992x10^{-5}$
Equivalent Elastic Strain	$5.0604x10^{-8}$	$5.8541x10^{-5}$
Shear Elastic Strain	$-1.4203x10^{-5}$	$1.7241x10^{-5}$
Equivalent Stress (Pa)	5717.1	$1.101x10^{7}$
Shear Stress (Pa)	$-1.0463x10^{6}$	$1.2701x10^{6}$

6.3.8 Result:

The material used is Stainless Steel and the maximum deformation got during this analysis is $2.3857x10^{-5}$ m, and the maximum equivalent elastic strain is $2.0992x10^{-5}$, maximum stress is $1.101x10^{7}$ Pa, maximum shear stress is $1.2701x10^{6}$ Pa and maximum directional deformation is $2.0992x10^{-5}$ m.

6.3.9 Analysis Pictures:

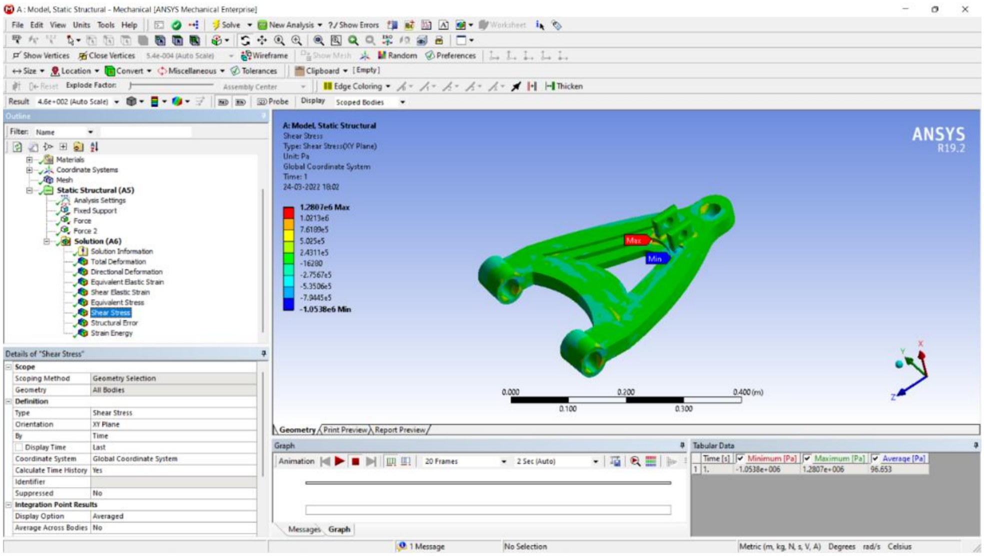

Fig 6.9 Analysis of Lower Arm -View 1

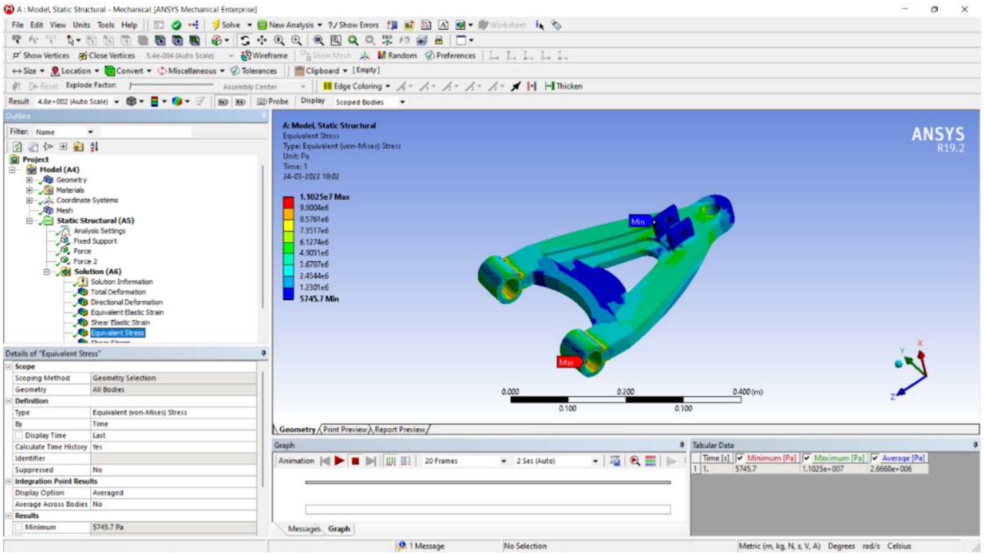

Fig 6.10 Analysis of Lower Arm -View 2

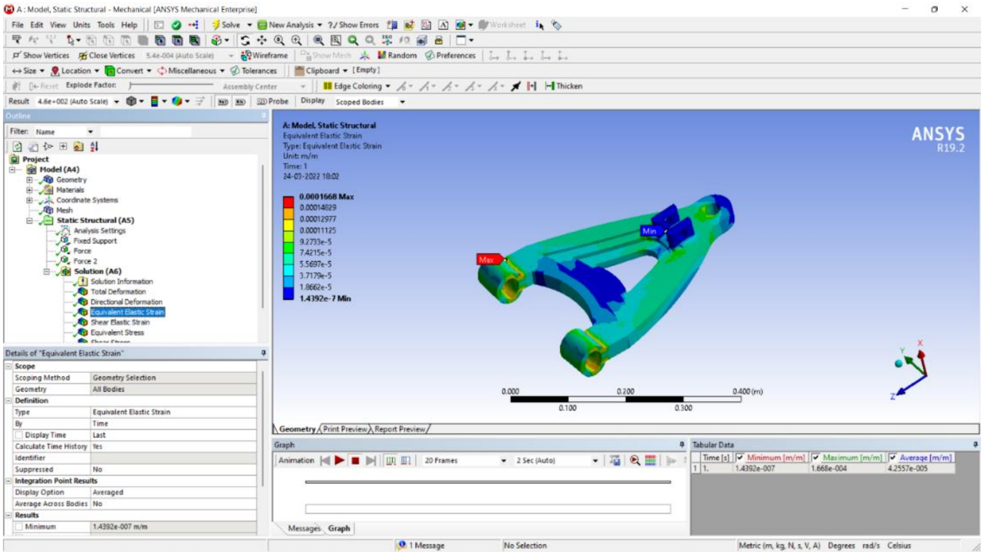

Fig 6.11 Analysis of Lower Arm -View 3

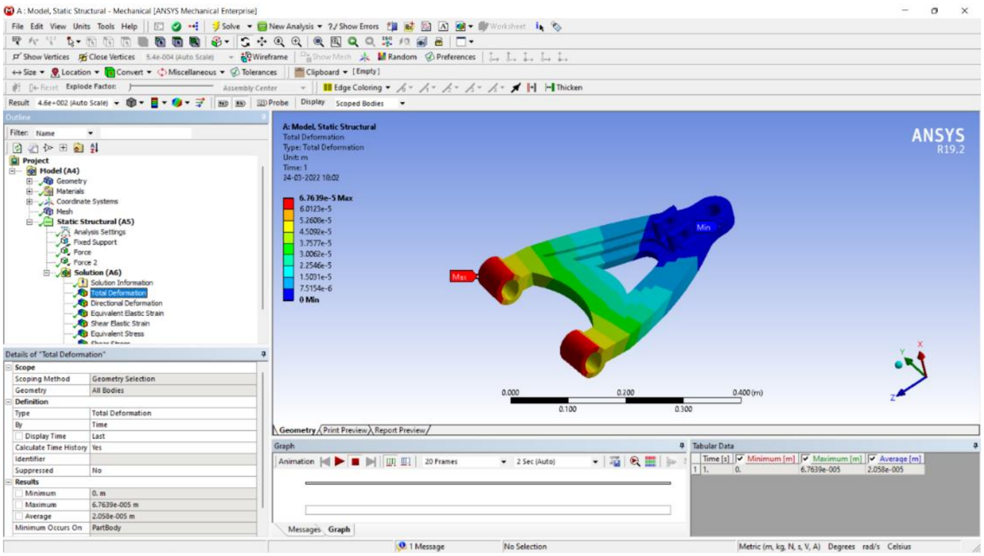

Fig 6.12 Analysis of Lower Arm -View 4

6.4 Analysis of Hub section:

6.4.1 Al 7075 Alloy:

Table 6.15 Analysis using Al 7075 Alloy for hub

Material/ Properties	**Al 7075 Alloy**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	$4.2507 x 10^{-5}$
Directional Deformation (m)	$-2.6246 x 10^{-6}$	$2.6301 x 10^{-6}$
Equivalent Elastic Strain	$2.4967 x 10^{-9}$	$8.0781 x 10^{-5}$
Shear Elastic Strain	$-5.0898 x 10^{-5}$	$5.8946 x 10^{-5}$
Equivalent Stress (Pa)	87.044	$5.6311 x 10^{6}$
Shear Stress (Pa)	$-1.3586 x 10^{6}$	$1.5734 x 10^{6}$

6.4.2 Result:

The material used is Al 7075 Alloy and the maximum deformation got during this analysis is $4.2507 x 10^{-5}$ m, and the maximum equivalent elastic strain is $8.0781 x 10^{-5}$, maximum stress is $5.6311 x 10^{6}$ Pa, maximum shear stress is $1.5734 x 10^{6}$ Pa and maximum directional deformation is $2.6301 x 10^{-6}$ m.

6.4.3 CA6NM Steel:

Table 6.16 Analysis using CA6NM Steel for hub

Material/ Properties	**CA6NM Steel**	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	$1.5144\text{x}10^{-5}$
Directional Deformation (m)	$-9.0006\text{x}10^{-7}$	$9.0204\text{x}10^{-7}$
Equivalent Elastic Strain	$8.9029\text{x}10^{-10}$	$2.748\text{x}10^{-5}$
Shear Elastic Strain	$-1.6903\text{x}10^{-5}$	$1.9433\text{x}10^{-5}$
Equivalent Stress (Pa)	87.705	$5.3889\text{x}10^{6}$
Shear Stress (Pa)	$-1.3306\text{x}10^{6}$	$1.5298\text{x}10^{6}$

6.4.4 Result:

The material used is CA6NM Steel and the maximum deformation got during this analysis is $1.5144\text{x}10^{-5}$ m, and the maximum equivalent elastic strain is $2.748\text{x}10^{-5}$, maximum stress is $5.3889\text{x}10^{6}$ Pa, maximum shear stress is $1.5298\text{x}10^{6}$ Pa and maximum directional deformation is $9.0204\text{x}10^{-7}$ m.

6.4.5 Gray Cast Iron:

Table 6.17 Analysis using Gray Cast Iron for Hub

Material/ Properties	Gray Cast Iron	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	2.7519×10^{-5}
Directional Deformation (m)	-1.643×10^{-6}	1.6467×10^{-6}
Equivalent Elastic Strain	1.6162×10^{-9}	5.0278×10^{-5}
Shear Elastic Strain	-3.1067×10^{-5}	3.5774×10^{-5}
Equivalent Stress (Pa)	87.531	5.4256×10^{6}
Shear Stress (Pa)	-1.3349×10^{6}	1.5372×10^{6}

6.4.6 Result:

The material used is Gray Cast Iron and the maximum deformation got during this analysis is 2.7519×10^{-5} m, and the maximum equivalent elastic strain is 5.0278×10^{-5}, maximum stress is 5.4256×10^{6} Pa, maximum shear stress is 1.5372×10^{6} Pa and maximum directional deformation is 1.6467×10^{-6} m.

6.4.7 Mild Steel:

Table 6.18 Analysis using Mild Steel for Hub

Material/ Properties	Mild Steel	
Min/Max	**Minimum**	**Maximum**
Total Deformation (m)	0	1.5113×10^{-5}
Directional Deformation (m)	-9.1857×10^{-7}	9.2047×10^{-7}
Equivalent Elastic Strain	8.8659×10^{-10}	2.8239×10^{-5}
Shear Elastic Strain	-1.7666×10^{-5}	2.0424×10^{-5}
Equivalent Stress (Pa)	87.136	5.5439×10^{6}
Shear Stress (Pa)	-1.3486×10^{6}	1.5591×10^{6}

6.4.8 Result:

The material used is Mild Steel and the maximum deformation got during this analysis is 1.5113×10^{-5} m, and the maximum equivalent elastic strain is 2.8239×10^{-5}, maximum stress is 5.5439×10^{6} Pa, maximum shear stress is 1.5591×10^{6} Pa and maximum directional deformation is 9.2047×10^{-7} m.

6.4.9 Analysis Pictures:

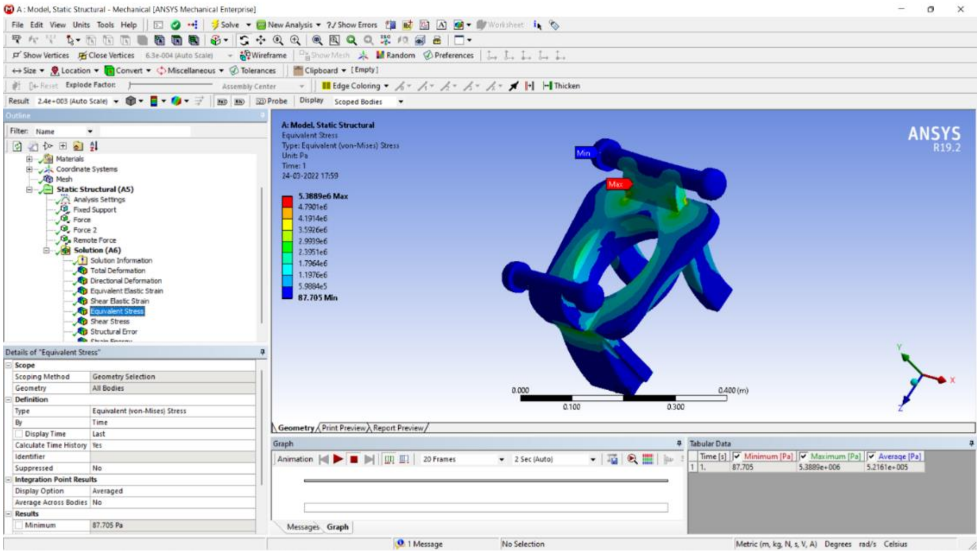

Fig 6.13 Analysis of Hub – View 1

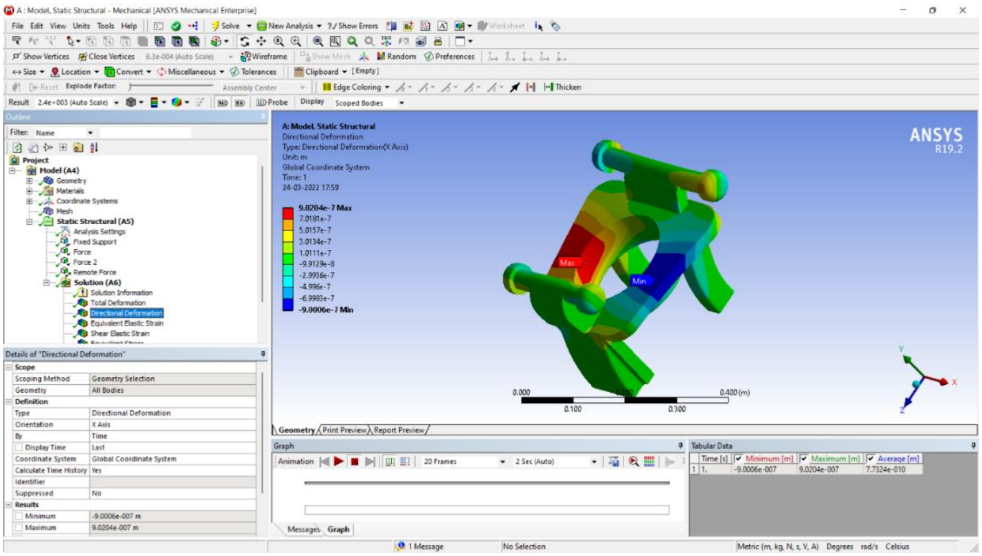

Fig 6.14 Analysis of Hub – View 2

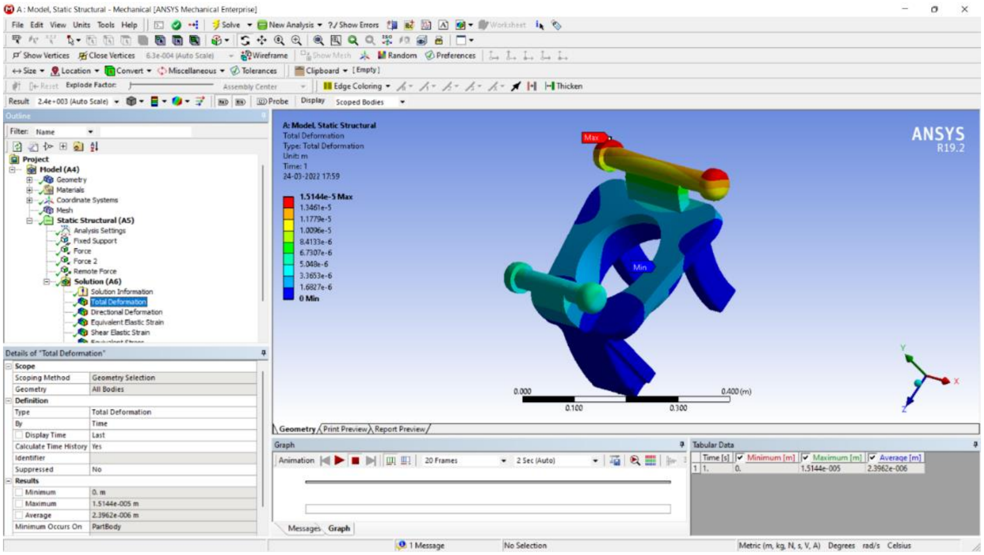

Fig 6.15 Analysis of Hub – View 3

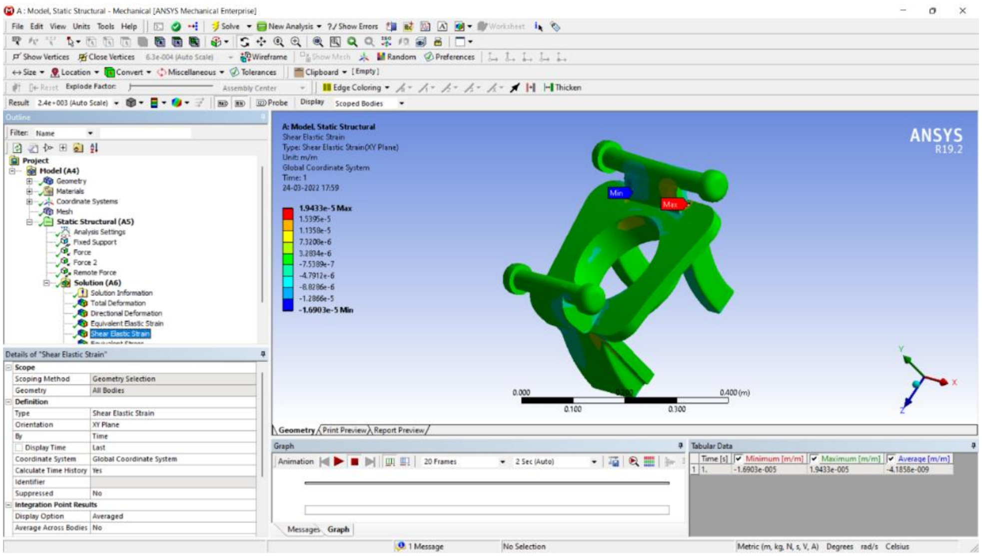

Fig 6.16 Analysis of Hub – View

CHAPTER 7

RESULTS AND DISCUSSION

7.1 Results of Spring Analysis

Table 7.1 Results of Spring Analysis

Material/ Properties	AISI 302 Steel	Titanium Alloy	Structural Steel	Chrome Vanadium	Beryllium Copper	Phosphor Bronze
Total Deformation (m)	1.603×10^{-6}	3.198×10^{-6}	1.5448×10^{-6}	1.495×10^{-6}	2.376×10^{-6}	5.821×10^{-6}
Directional Deformation (m)	-6.49×10^{-9}	-9.19×10^{-9}	-5.920×10^{-9}	-5.897×10^{-9}	-9.108×10^{-9}	1.380×10^{-7}
Equivalent Elastic Strain	8.958×10^{-7}	1.829×10^{-6}	8.6716×10^{-7}	8.3713×10^{-7}	1.334×10^{-6}	3.346×10^{-5}
Shear Elastic Strain	-1.06×10^{-8}	-2.33×10^{-8}	-1.050×10^{-8}	-1.003×10^{-8}	-1.615×10^{-8}	2.125×10^{-5}
Equivalent Stress(Pa)	1.5625×10^{5}	1.5966×10^{5}	1.5698×10^{5}	1.5661×10^{5}	1.5698×10^{5}	3.6647×10^{6}
Shear Stress (Pa)	-801.81	-825.22	-807.93	-804.88	-807.93	-819.71

7.1.1 Comparison of results:

By comparing the above results, the total deformation is less in Chrome vanadium. Whereas the directional deformation and equivalent elastic strain is comparably equal in Chrome vanadium and Structural steel. The equivalent stress induced in the Chrome vanadium is less. Therefore, we can choose Chrome vanadium as optimum material for the coil over shock absorber.

7.2 Results for Upper Arm Analysis

Table 7.2 Results of Upper Arm Analysis

Material/ Properties	AISI 1040 Steel	Aluminium	Gray Cast Iron	Stainless Steel
Min/Max	**Maximum values**			
Total Deformation (m)	$5.4411x10^{-3}$	$1.6803x10^{-2}$	$1.0387x10^{-2}$	$5.9206x10^{-3}$
Directional Deformation (m)	$4.3856x10^{-5}$	$1.3544x10^{-4}$	$8.1765x10^{-5}$	$4.8343x10^{-5}$
Equivalent Elastic Strain	$1.6688x10^{-3}$	$5.1537x10^{-3}$	$1.0522x10^{-6}$	$1.8132x10^{-3}$
Shear Elastic Strain	$6.4989x10^{-4}$	$2.007x10^{-3}$	$1.224x10^{-3}$	$7.1187x10^{-4}$
Equivalent Stress (Pa)	$3.2307x10^{8}$	$3.2307x10^{8}$	$3.2345x10^{8}$	$3.3292x10^{8}$
Shear Stress (Pa)	$5.2491x10^{7}$	$5.2491x10^{7}$	$5.2592x10^{7}$	$5.2439x10^{7}$

7.2.1 Comparison of results:

By comparing the above results, the total deformation is less in AISI 1040 steel when compared to other materials. Whereas the directional deformation is also less in AISI 1040 steel. On the contrary the equivalent elastic strain is less in Gray cast iron. The stress induced in the material is same for AISI 1040 steel and Aluminium. Therefore, we can choose AISI 1040 Steel as optimum material for upper wishbone.

7.3 Results of Lower Arm Analysis

Table 7.3 Results of Lower Arm Analysis

Material/ Properties	**AISI 1040 Steel**	**Aluminium**	**Gray Cast Iron**	**Stainless Steel**
Min/Max	**Maximum values**			
Total Deformation (m)	2.1902×10^{-5}	6.7639×10^{-5}	4.1716×10^{-5}	2.3857×10^{-5}
Directional Deformation (m)	1.9257×10^{-5}	5.9472×10^{-5}	3.6626×10^{-5}	2.0992×10^{-5}
Equivalent Elastic Strain	5.4013×10^{-5}	1.668×10^{-4}	1.0391×10^{-4}	5.8541×10^{-5}
Shear Elastic Strain	1.5856×10^{-5}	4.8967×10^{-5}	3.0291×10^{-5}	1.7241×10^{-5}
Equivalent Stress (Pa)	1.1025×10^{7}	1.1025×10^{7}	1.1058×10^{7}	1.107×10^{7}
Shear Stress (Pa)	1.2807×10^{6}	1.2807×10^{6}	1.3016×10^{6}	1.2701×10^{6}

7.3.1 Comparison of results:

By comparing the above results, the total deformation is less in AISI 1040 steel when compared to other materials. Whereas the directional deformation and equivalent elastic strain are also less in AISI 1040 steel. The stress induced in the material is same for AISI 1040 steel and Aluminium. Therefore, we can choose AISI 1040 Steel as optimum material for lower wishbone.

7.4 Results of Hub Analysis

Table 7.4 Results of Hub Analysis

Material/ Properties	**Al 7075 Alloy**	**CA6NM Steel**	**Gray Cast Iron**	**Mild Steel**
Min/Max	**Maximum values**			
Total Deformation (m)	4.2507×10^{-5}	1.5144×10^{-5}	2.7519×10^{-5}	1.5113×10^{-5}
Directional Deformation (m)	2.6301×10^{-6}	9.0204×10^{-7}	1.6467×10^{-6}	9.2047×10^{-7}
Equivalent Elastic Strain	8.0781×10^{-5}	2.748×10^{-5}	5.0278×10^{-5}	2.8239×10^{-5}
Shear Elastic Strain	5.8946×10^{-5}	1.9433×10^{-5}	3.5774×10^{-5}	2.0424×10^{-5}
Equivalent Stress (Pa)	5.6311×10^{6}	5.5889×10^{6}	5.4256×10^{6}	5.5439×10^{6}
Shear Stress (Pa)	1.5734×10^{6}	1.5298×10^{6}	1.5372×10^{6}	1.5591×10^{6}

7.3.1 Comparison of results:

By comparing the above results, the total deformation is less in Mild steel when compared to other materials. Whereas the directional deformation and equivalent elastic strain comparably equal and less in CA6NM steel and Mild steel. The stress induced in the Mild Steel is less when compared to other materials. Therefore, we can choose Mild steel as optimum material for hub section.

CHAPTER 8
CONCLUSION

In automobiles, a double wishbone suspension is an independent suspension design using two (occasionally parallel) wishbone- shaped arms to locate the wheel. Each wishbone or arm has two mounting points to the chassis and one joint at the knuckle. Due to this the maximum dampening may not occur in some cases. In this paper, the problem is solved by shifting the upper mounting point of coil over spring absorber from knuckle to the upper wishbone. So that the maximum dampening will occur. After the successful design of suspension components, the structural analysis of the components is carried in ANSYS. And the optimum material is chosen for the suspension components i.e., for Coil over shock absorber the chrome vanadium is chosen as the optimum material. For lower and upper wishbone AISI 1040 Steel and for Hub section Mild Steel is chosen as optimum materials.

Thus, the suspension system for a formula car has been designed and analyzed using FSAE standards.

CHAPTER 9

FUTURE SCOPE OF THE PROJECT

The suspension system design discussed in this paper is limited to coil over shock absorber and the remaining suspension components are designed directly based on FSAE standards. The structural analysis of the components is done by using few materials of similar properties. Various suspension designs can be proposed for better efficiency and stability. The theoretical calculations can be done by including the other parameters like roll center, center of gravity etc. to improve the stability of the vehicle at cornering and braking. All the suspension components can be designed by calculating and varying design parameters. Various materials can be tested and analyzed for better performance.

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