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Design and Optimization Formula SAE Drivetrain Components

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Abstract: The overall objective of this paper is to refine the drivetrain system design, selection of suitable materials and analysis techniques for FSAE vehicles. The main objective of this work is to optimize the hub and Upright for 10-inch rim, in terms of reducing the overall weight of the system, additional accessories for variable camber and appropriate selection of bearings. To achieve this, the safety deformation factor, the fatigue resistance of the system and the fit of the entire wheel set were taken into account. Software used in design and validation are Solidworks, Ansys
Keywords: FSAE, Wheel Hub, Upright, Shims, Brackets

I. INTRODUCTION

Formula SAE is a student-level design competition where teams from various colleges from around the world participate to compete in static and dynamic events. The teams are divided into subsystems where the role of each engineering department is to design, build and test their own part of the system considering formula SAE rules . The rules suggest that the steering will be placed on the front uprights and must ensure the tyres do not come into contact with the suspension, body or frame, standard wheel lug bolts and studs must be made of steel and are considered engineering fasteners, Aluminum wheel nuts may be used, but they must be hard anodized and in pristine condition . The scope of drivetrain departments is to design, validate and manufacture transmission systems and wheel assembly. Transmission systems include sprocket, differential, axle and CV joints, wherelse wheel assembly includes the upright, hub and wheel. The whole drivetrain system focuses on transmitting power from the engine output shaft to the wheels by minimizing the friction losses and uniform distribution of power . It also ensures the safety of moving parts of the engine.

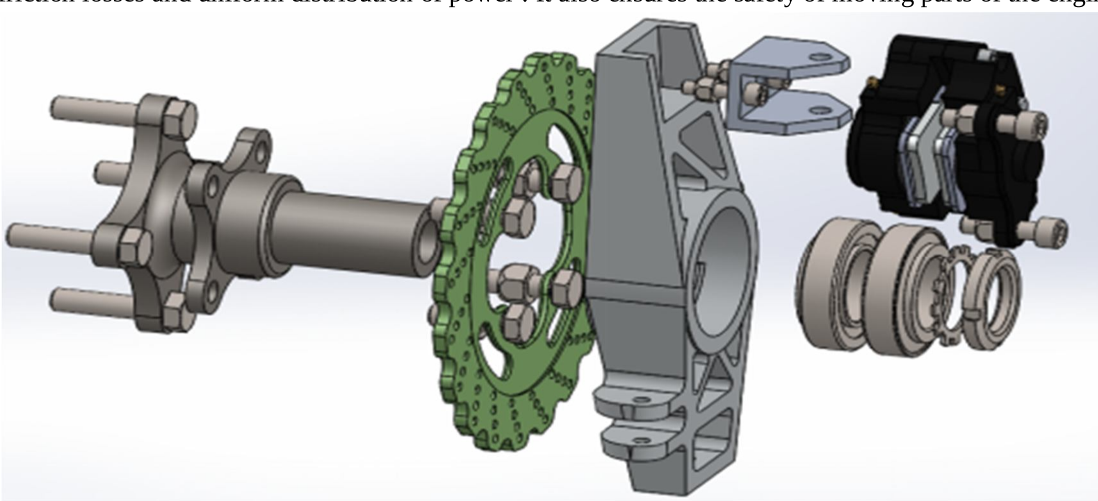

Fig. 1 Wheel Assembly layout

II. BEARING SELECTION

It is necessary to select the bearing before starting the design of hub and knuckle as some dimensions are dependent upon size and diameter of the bearing. For front and rear wheel assembly tapered roller bearing is used as bearing has to withstand radial as well as axial loads. To meet these requirements a pair of single row tapered roller bearings is used in back-to-back arrangements. Axial loads can be accommodated in both directions, although only by one bearing at a time. It is more effective than face-to-face arrangement because the pressure centre lies more far in back-to-back arrangement. It will provide better stability. Also back-to-back arrangement of bearing can withstand a greater amount of tilting moment as compared to face-to-face arrangement.

In this arrangement (back-to-back arrangement), it can accommodate larger moments even if the centre distance of the bearing is relatively shorter and this type of assembly of bearing will provide greater distance between the pressure centres of the bearing which will eventually help to provide better loads accommodation radial as well as axial at the same time. For our application, 320-32X bearing is selected. For front and rear wheel assembly the same set of taper roller bearings is used rather than ball bearing in the rear assembly.

TABLE I
PROPERTIES OF SELECTED BEARING

Designation	320/32X
Bore (d)	32 mm
Outside diameter (D)	58 mm
Width (B)	17 mm
Dynamic load (C)	45100 N
Static load (Co)	46500 N
Mass	0.2 Kg

III.HUB

A. Introduction

The hub contacts three different components within the wheel assembly which are the brake disk, the wheel and the wheel bearing. The main duty of the hub is to connect the suspension system of the car to the wheels. The design of the hub should be more reliable and sturdy as all the loads from the tire are going to be transferred via hub to the upright and then to the suspension system. So the hub must have to withstand all the forces and must prevent excessive deflections and bending. Components of wheel assembly are critical components so an excessive amount of deflection can bring many different problems regarding car's handling. Also due to this, some angles may vary i.e. camber, caster which make it difficult to steer the car properly. While designing the hub, we also have to look after the length of the spindle because more length increases the chances of bending which can affect the tire contact patch. Hub also consists of mounting the brake rotor.

The design of the wheel hub is dependent on the following factors

1) PCD of the wheel mounting plate
2) Bearing used
3) PCD of brake rotors
4) Locking methods of bearing
5) Front or rear hub
6) Size of brake calliper

B. Design procedure

Design of the hub started after the selection of wheels. As per the number of lug bolts and PCD of selected wheels from there we can start designing the hub. After that concerning the brake department as per the design of brake rotor and PCD of rotor mounting HUB design will proceed further considering PCD of rotor mounting. The length of the hub shoulder is dependent on the width of the bearing and locking system used for bearing. The diameter of the shoulder is dependent upon the ID of selected bearing.

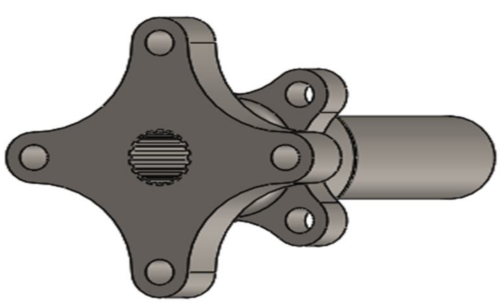

Fig. 2 3D CAD of Hub

C. Front Hub and rear Hub

Front hub is not involved in any direct power/torque transmitting system so there is no any providence of spline on hub. The sole function of the front hub is to support the wheels and to connect with the suspension system. In the front wheel assembly we have to provide a proper locking system to restrain the axial motion of the bearing. So the locking of the bearing is done by locking nut and tab washer. The thread for the lock nut is provided on the hub shoulder to lock the position of the bearing. The primary function of the rear hub is to transmit power from the engine via axle to wheel. As it is a power transmitting component it will have spline on the internal surface of the hub to fit the axel. The locking of bearing in the rear hub is done by a retaining ring and for that groove is provided on the hub shoulder. The method of locking of the bearing is different in rear wheel assembly because the hub is connected with the axle which helps to restrict the axial movement of the bearing. The geometric design of the front and rear hub is almost similar. There are no major changes in the design of both front and rear hub except for the length of hub shoulder. The length of the hub varies as per data from suspension geometry and packaging of wheel assembly in wheel.

D. Material Selection for hub

Material selection, in general, implies selecting materials with optimal costs and performance that meet the component's designed service life. As for a formula car, weight and capital are the most crucial aspects needed to be taken into consideration. While the team was primarily concerned with optimizing weight and cost, they also took into account mechanical properties such as tensile strength, compressive stress, endurance limit, impact resistance etc. Along with that, the space considerations, castability, weldability, and regional material availability were put into consideration..The AISI 4130 being strong and tough, the steel is also weldable and machinable. The AISI 4130 grade is a versatile alloy with good atmospheric corrosion resistance and a reasonable strength. Overall, it offers a good combination of strength, toughness, and fatigue resistance. Although carbon fiber and Mild Steel are both substantial materials, mild steel is typically more efficient than carbon fiber in terms of cost. Also, producing carbon fiber is a very precise and time-consuming process. The price of titanium is much higher than stainless steel, which makes it more cost-prohibitive and complex to manufacture.

TABLE II
PROPERTIES OF AISI 4340

Young’s Modulus	190 Gpa
Elongation at brake	12%
Fatigue strength	330 Mpa
Poisson’s ratio	0.29
Shear strength	430 Mpa
Tensile strength	690 Mpa
Yield Strength	470 Mpa

E. Force calculation and analysis

The hub must have to withstand following forces

1) Acceleration and deceleration
2) Force due to braking
3) Forces at the cornering
4) Brake torque

The force due to braking in the longitudinal direction is calculated as follows:
Stopping distance, s = 5m
Initial velocity, u = 13.5m/s
Calculating deceleration by v2 –u2 = 2*a*s
a = -18.22m/s2. Assuming 60% weight transfer to the front,
Force due to the braking on front hub = 3500N
Force due to cornering in lateral direction:
Skid pad track diameter, Ø=15.25 m ≥ r = 9.125 m
Width of track= 3 m

Travelling distance = d = 2*π*r 2*3.14*9.125 = 57.33m
Taking t = 7 sec for one lap
= 57.33*7 = 8.19 m/sec
= 8.192 9.125 = 7.35 m/sec2
f = 2352 N
Considering the weight of the car, bump force = 2000 N.
Moment generated at wheel mounting points during braking is calculated as follows
Force due to braking is longitudinal direction = 3500 N
Tire radius = 0.2286 m
Moment generated ≈ 800 Nm
Moment generated at brake rotor mounting points while braking is calculated as follows
After consulting the brake department, clamping force was found to be = 1076.5 N
Moment generated at the front wheels = 123 Nm

F. FEA Analysis

To validate the design of the hub we have to run the simulations on the basis of force which we have calculated . For validation, three different analysis were performed to ensure that the design is safe in all loading conditions. The methods of analyses are discussed below along with its results. The boundary conditions are shown in Table III, Table IV, Table V and results of simulations can be seen from Fig 3 to Fig 11.

1) 1^{st} *Analysis*

TABLE III
PROPERTIES OF AISI 4340

Constraints	Wheel mounting points
Force	Hub shoulder
Equivalent stress	1.81e+08 N/m2
Maximum deformation	9.2e-2 mm
FOS (Ultimate)	2.6

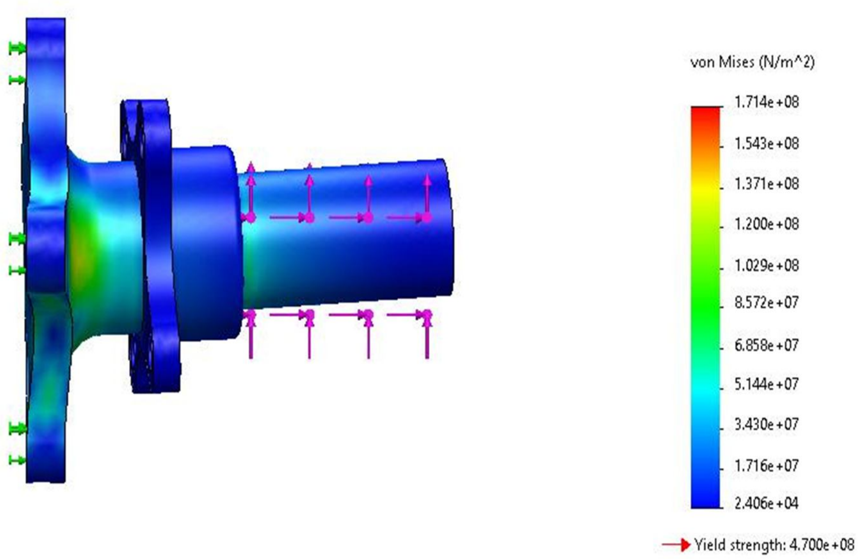

Fig. 3 Stress (Von Mises)

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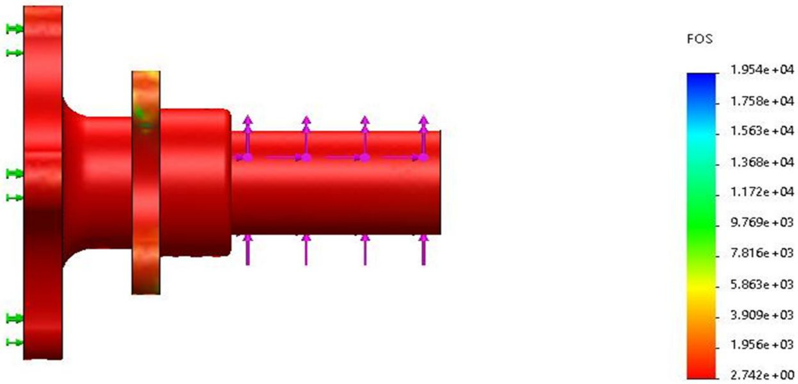

Fig. 4 Factor of safety

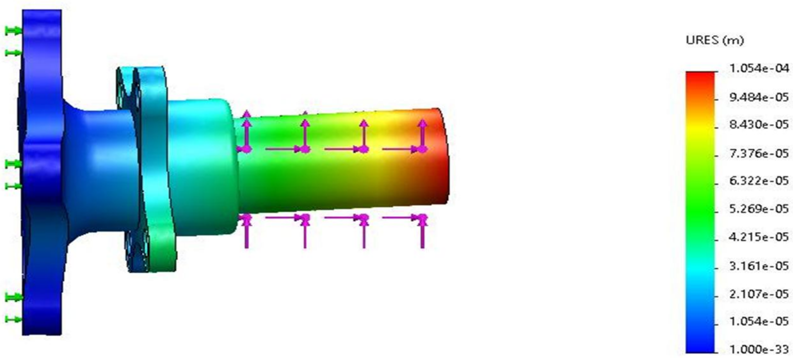

Fig. 5 Total Deformation

2) 2^{nd} *Analysis*

TABLE IV
PROPERTIES OF AISI 4340

Constraints	Brake mounting points
Force (Torque)	Wheel mounting points
Equivalent stress	1.76e+08 N/m2
Maximum deformation	8.66e-2 mm
FOS (Ultimate)	2.7

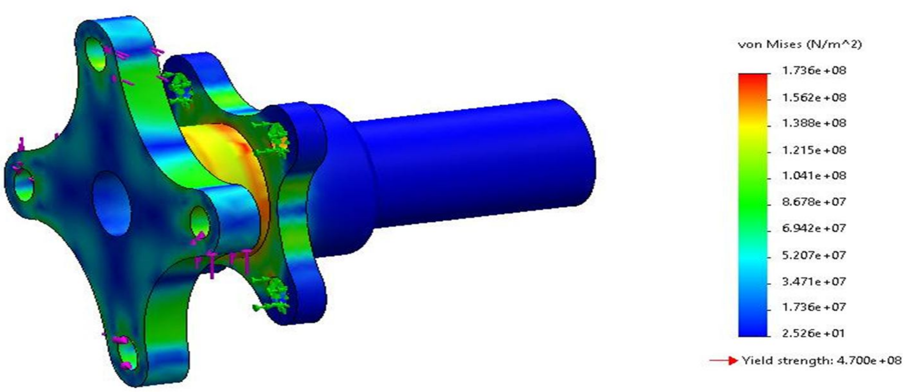

Fig. 6 Stress (Von Mises)

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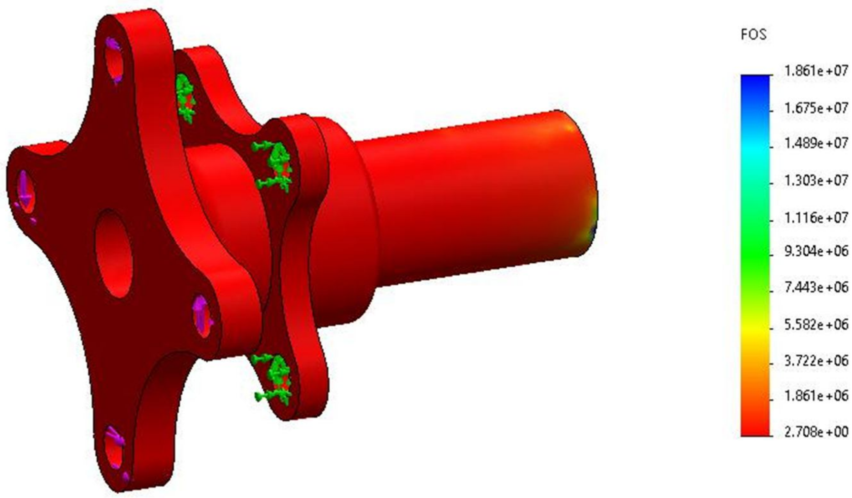

Fig. 7 Factor of safety

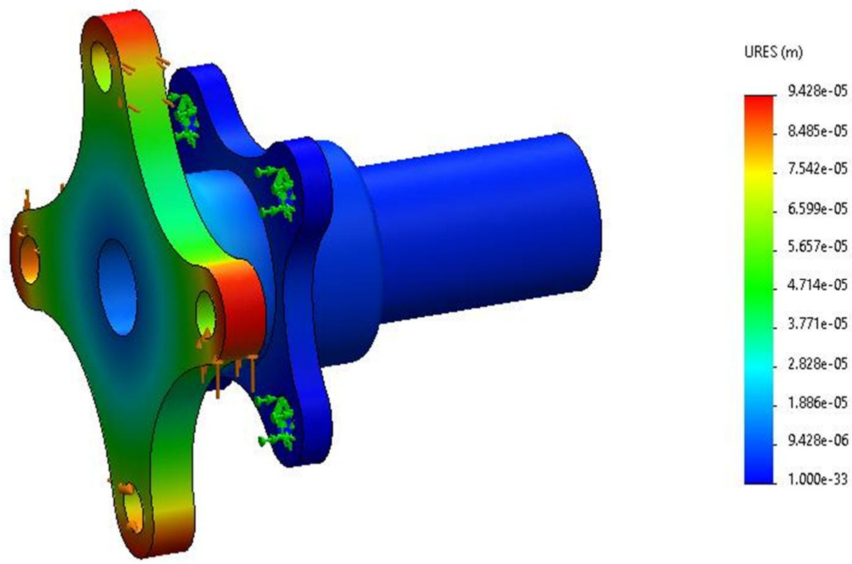

Fig. 8 Total Deformation

3) 3rd Analysis

TABLE V
PROPERTIES OF AISI 4340

Constraints	Wheel mounting points
Force (Torque)	Brake rotor mounting
Equivalent stress	1.177e+08 N/m2
Maximum deformation	3.4e-2 mm
FOS (Ultimate)	4

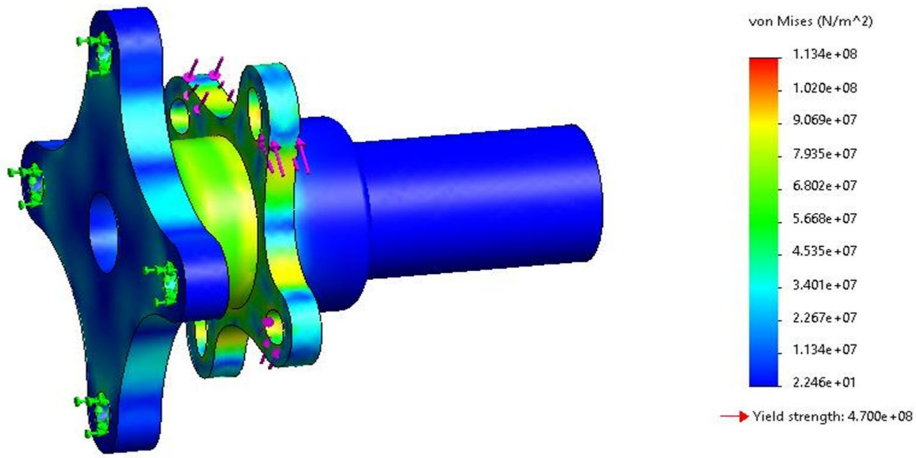

Fig. 9 Stress (Von Mises)

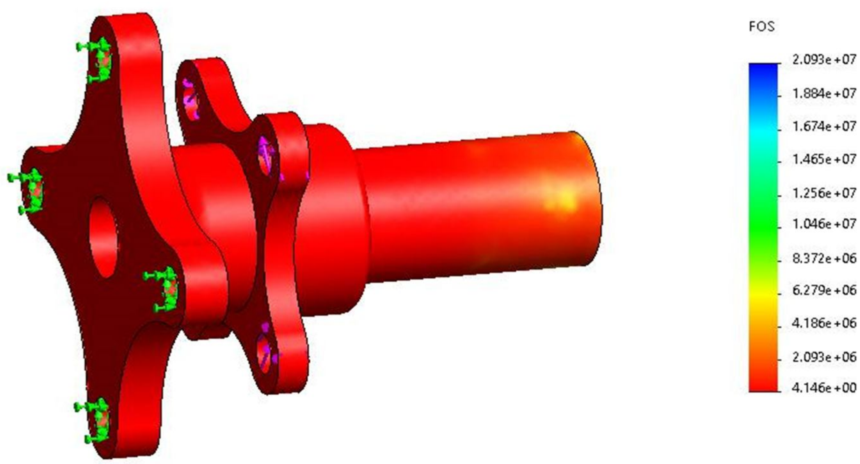

Fig. 10 Factor of safety

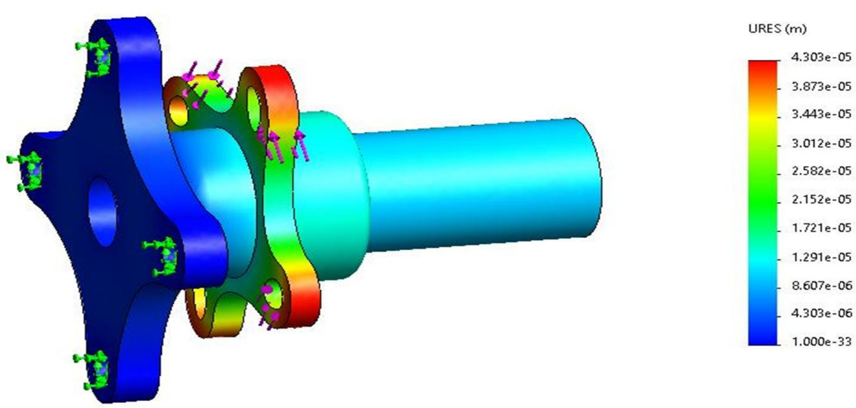

Fig. 11 Total Deformation

Hence, From all the analyses and from the results which we obtained, the total deformation is very low and also the stress generated are in the permissible limit of the selected material. In all the analysis the FOS is more than 2 so overall the design of the hub is safe.

IV. UPRIGHT

A. *Introduction*

The uprights are a vital element of the car because it connects the brakes, steering, wheels, and suspension. The uprights are the central part of the suspension of a Formula SAE car. All suspension parts as well as the control arms, steering arms, springs, shock absorbers, brakes, tires, and in the case of the rear upright the axles are connected to the uprights. There are several parameters that we've to contemplate very rigorously while designing the upright. A-arm is connected to the upright thus very little error in design might change the suspension geometry which can reflect in the car’s performance. The bearings are placed in the bearing house and the hub is connected through the bearings. In the bearing house, some space is provided between two bearings for restricting axial movement and fixing the bearing position. Upright additionally consist of brake calipers, therefore the position of the caliper should be effective for braking, and additionally, it shouldn't contact the rim while the rim spins. The front upright is connected to the steering rack with the steering arm in the upright. Whereas the rear upright is connected with the tie rod. The bracket design is provided for upper ball joint as well as lower ball joint for easy assembly and disassembly of A-arm with upright and other reason for bracket design is to introduce shims which allow varying in camber angle in step with our requirement of the car’s handling performance. All forces that the car can encounter due to the road and tire interaction can undergo the uprights. The uprights should be sufficiently robust to resist several of those forces occurring at the same time, also as any forces which will happen as a result of a crash or different kind of emergency without failure. Any failure of the uprights would render the automobile un-drivable. Factors that are considered while designing Upright are as follows

1) Caster angle
2) Kingpin inclination
3) Camber angle
4) Bearings
5) Distance between LBJ and UBJ
6) Brake calliper
7) Tie rod mounting

B. *Vehicle Dynamics*

The forward direction and each of the cars can have some extent of positive caster. The result of rotating the kingpin axis clockwise is that's causing the wheels to remain pointed forward once rolling and dampens the comeback of the wheel to center. This combined with trail is why an automobile can usually track during a straight line rather than turning to at least one side while no driver input. The trail may be a measurement that's often caused by adding a caster to an upright design, however, it may be accumulated or decreased manually. It's the measurement of the offset from the wheel center laterally to the kingpin axis once viewed from the side and is measured within the forward direction. The trail has a similar result as the caster, however, a lot is directly associated with the return ability to center. Finally, KPI or steering axis inclination is the angle between the ball joints from the vertical when looking from the front of the vehicle. This is often measured from the upper ball joint to the wheel center and is employed to help in return the ability of the wheel and to reduce the scrub radius of the front wheels. The scrub radius is the distance between the tire center and kingpin axis when viewing from the front of the vehicle. Having a large scrub radius makes the car way tougher in cornering because the wheels are being dragged around the kingpin axis. It's necessary to create the scrub radius as little as doable, to reduce fatigue on the driver and all navigating through turns.

TABLE VI
DATA FROM VEHICLE DYNAMICS (INPUT)

Value	Front Upright	Rear Upright
KPI	3.79º	2º
Caster	5º	0
Distance between UBJ and LBJ	171.17 mm	170.90 mm

C. Design Procedure

1) The first step of designing upright is from consideration of KPI and caster angle as per given by the suspension department and from distance of UBJ and LBJ.
2) Then after designing a bearing house which is dependent on selected bearing and considering tolerance. The space is provided between two bearings for the purpose of ease of assembly and removal and fixes the position of the bearing in the proper manner in the housing.
3) The next step after designing of housing the clamps are designed based on KPI and caster angle. After that with the help of a cad model of calliper and mounting dimensions' brake mountings are designed.
4) The brackets were designed according to the upright design for UBJ and LBJ.
5) The final step of the design of upright is the design of tie-rod mounting and steering arm.
6) Bracket design is also an integral part of upright design. Bracket should withstand the longitudinal force while braking, latitudinal force while cornering, bump force, moment generated at brake calliper mountings while braking, and force on tie rod mounting while cornering.

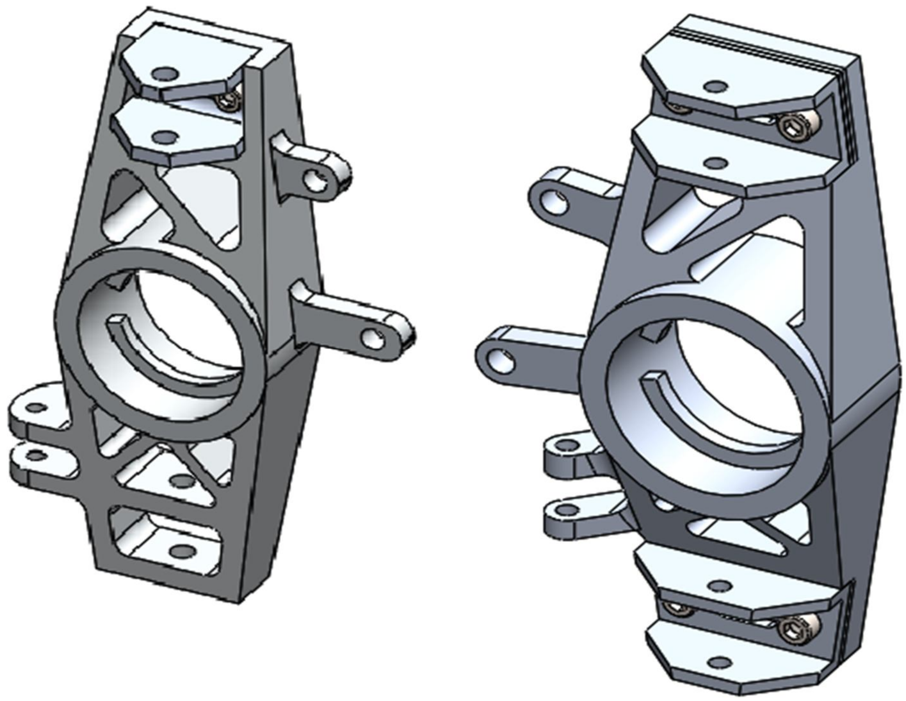

Fig. 12 3D CAD of Rear Upright (Left) and Front Upright (Right)

D. Material selection for upright

Density of the aluminum is around one third to that of general stainless steel so our idea of reducing in weight is achieved. Aluminum has a much longer life than steel. For the manufacturing of upright AL 7076-T6 is used as it is light weight and it provides better tensile strength and fatigue strength as compared to other aluminium alloy.

TABLE VII
PROPERTIES OF AISI 4340

Young's modulus	70 Gpa
Elongation at break	7.9%
Fatigue strength	160 Mpa
Poisson's ratio	0.32
Shear strength	330 Mpa
Tensile strength	560 Mpa
Yield strength	480 Mpa

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E. Force Calculation and Analysis of Knuckle

For the FEA analysis of the knuckle we have taken the same forces as we calculated before for the analysis of the hub as all the forces are going to transfer from hub to knuckle and then to suspension system.

As we've calculated before,

Longitudinal force during braking = 3500 N

Latitudinal force during cornering = 2352 N

Bump force = 2000 N

The FEM analysis was done on SolidWorks and the results were carried out as follows:

To validate the upright design and to check the sustainability of the design we have done various simulations and various loading scenarios. The calculation of the loads is done previously which are the same as the forces acting on the hub. The methods of analyses are discussed below. The constraints and results are shown in tabular form.

1) 1st analysis

TABLE VIII

PROPERTIES OF AISI 4340

Constraints	Bearing housing
Force	Bracket mounting points
Equivalent stress	1.92e+08 N/m2
Maximum deformation	3.09e-01 mm
FOS (ultimate)	2.6

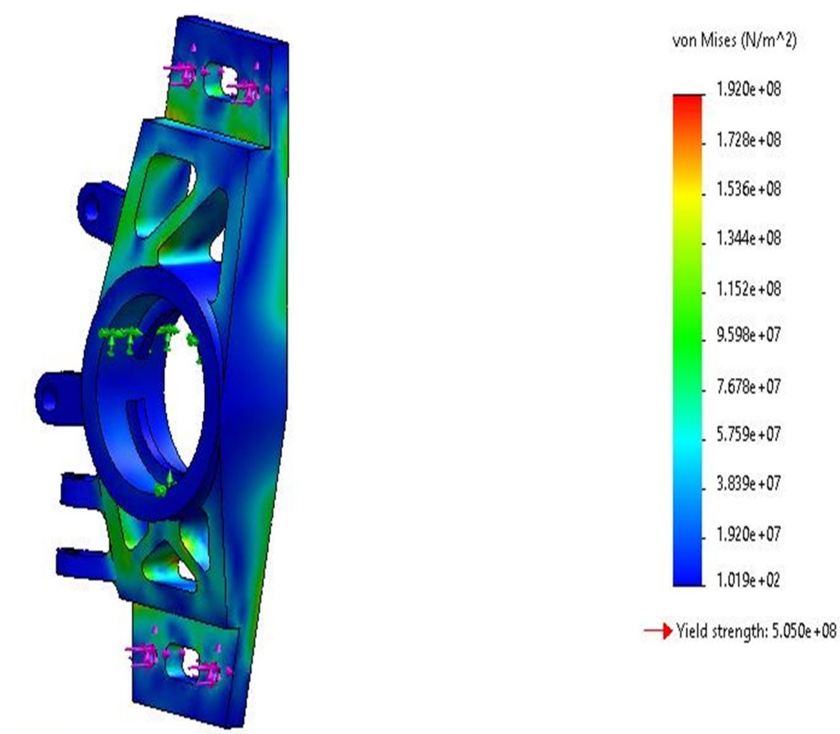

Fig. 13 Stress (Von Mises)

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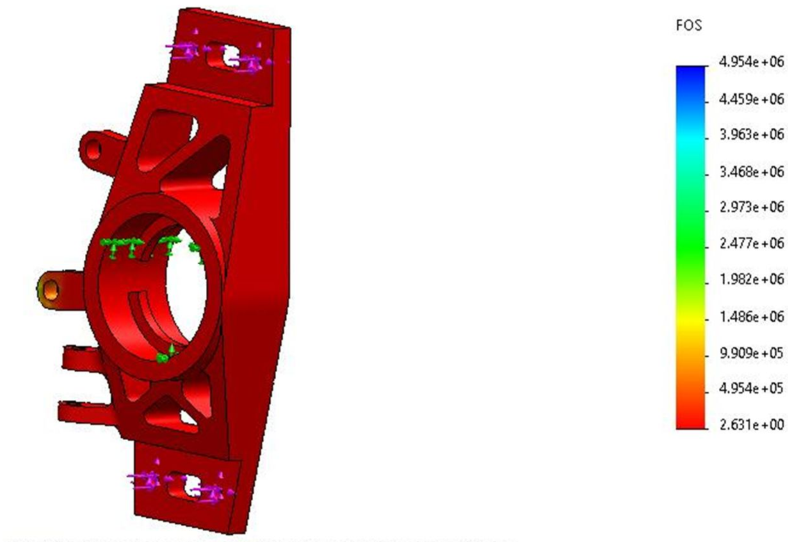

Fig. 14 Factor of safety

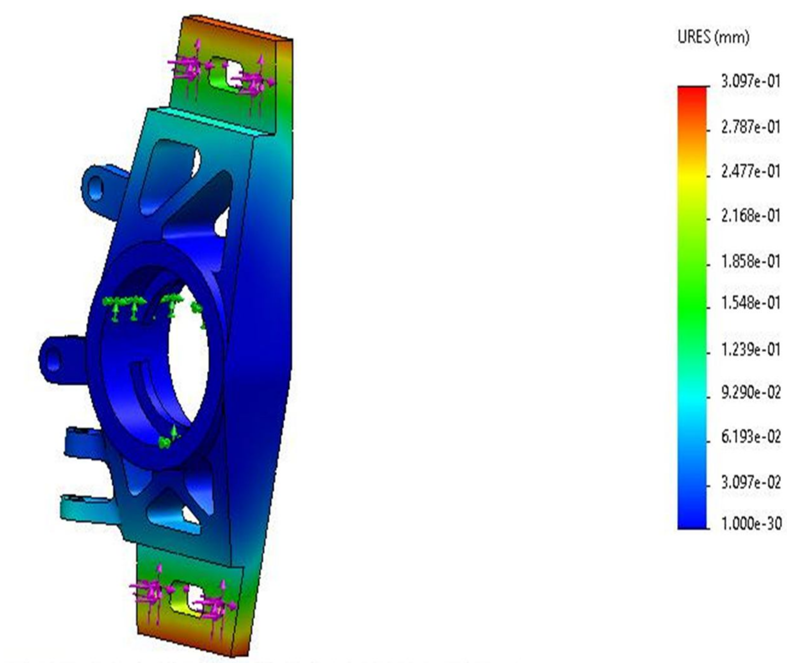

Fig. 15 Total Deformation

2) 2nd analysis

TABLE IX
PROPERTIES OF AISI 4340

Constraints	Bearing housing
Force	Steering arm
Equivalent stress	2.078e+08 N/m2
Maximum deformation	2.24e-01 mm
FOS (ultimate)	2.4

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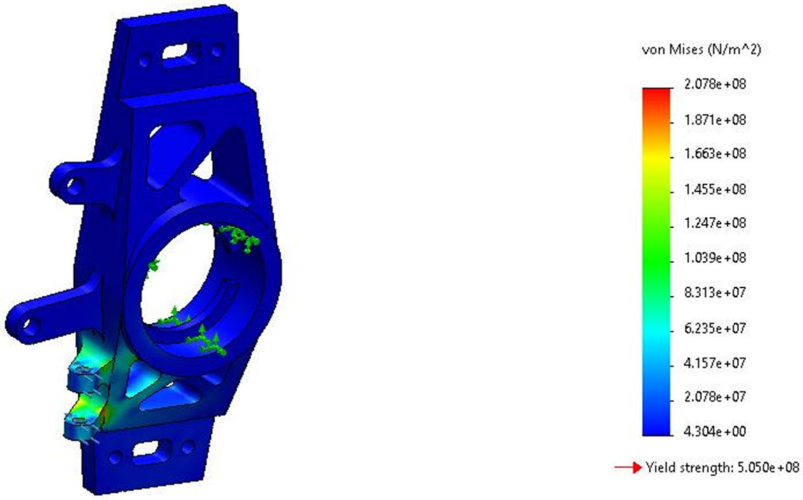

Fig. 16 Stress (Von Mises)

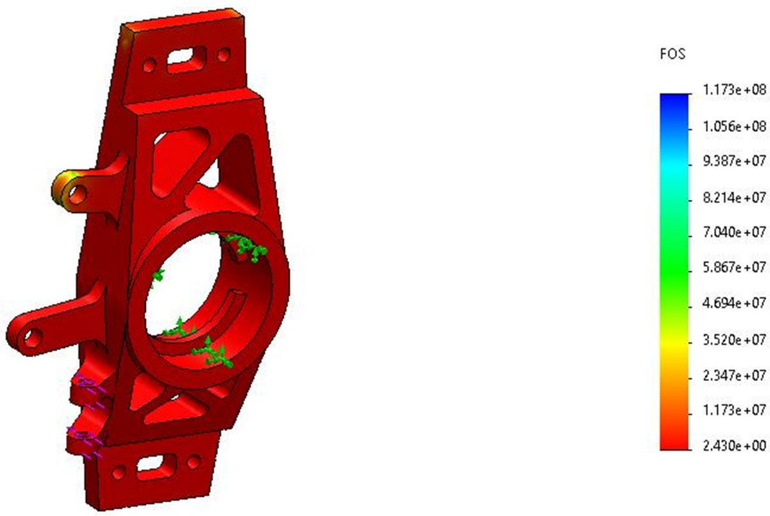

Fig. 17 Factor of safety

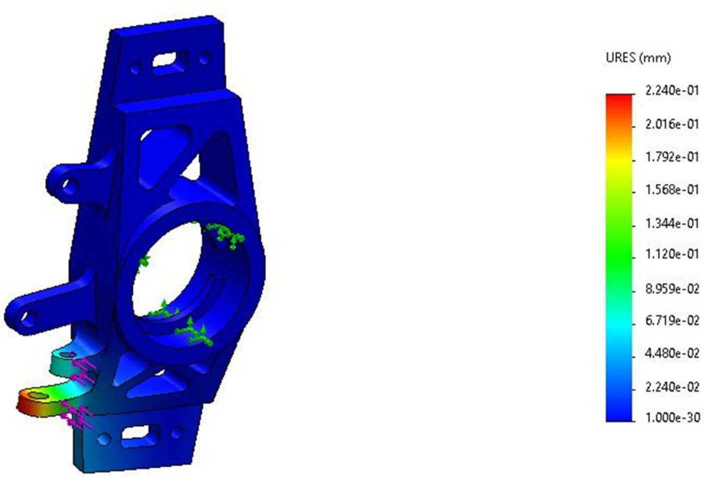

Fig. 18 Total Deformation

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3) 3rd analysis

TABLE X
PROPERTIES OF AISI 4340

Constraints	Bracket mounting points
Force (Torque)	Calliper mounting
Equivalent stress	1.90e+08 N/m2
Maximum deformation	2.4e-01 mm
FOS (ultimate)	2.6

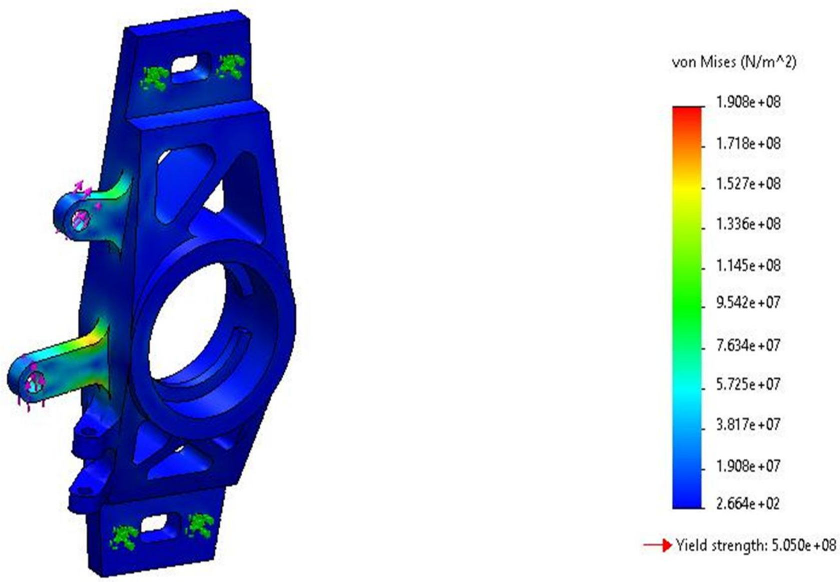

Fig. 19 Stress (Von Mises)

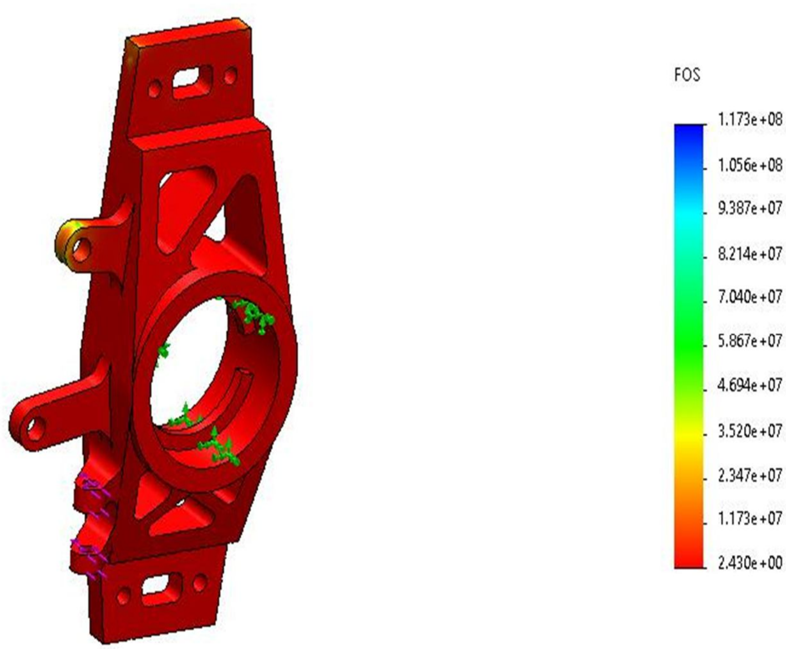

Fig. 20 Factor of safety

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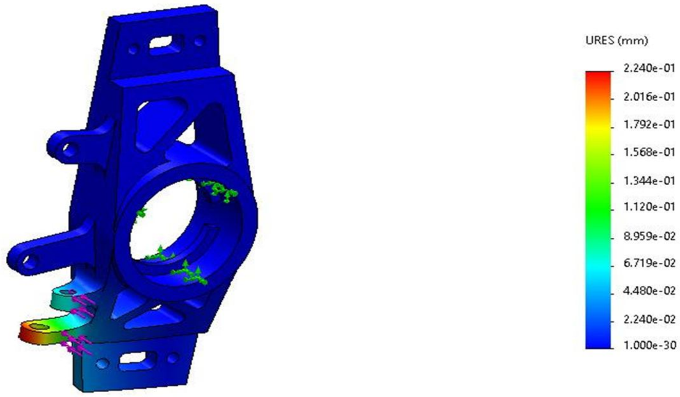

Fig. 21 Total Deformation

From the above results, the design of upright is safe and stresses are in the permissible limit of the selected material.

F. Bracket Analysis

Brackets are c shape type; it is used for mounting the suspension A-arm inside it to restrict the vertical and horizontal movement while moving. Its overall length depends upon the

- Bolt and nut used to fasten it to upright
- Minimal space required for easy movement of an arm wafer so that they do not touch each other
- Thickness of brackets

To ensure the strength of the bracket we have applied all the three forces which we have considered during analysis of the bracket because those forces are going to be transferred via ball joints to the bracket. The method of analyzing the bracket is shown in the form of tabular and the results are discussed below. The analysis of brackets is carried out for both upper and lower bracket design.

1) Analysis of upper bracket

TABLE XI

PROPERTIES OF AISI 4340

Constraints	Bolt mounting and body surface
Force	Remote load
Equivalent stress	1.48e+08 N/m2
Maximum deformation	1.46e-01 mm
FOS (ultimate)	3.4

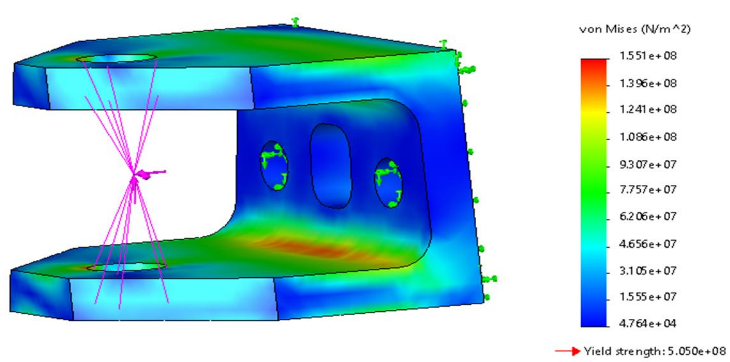

Fig. 22 Stress (Von Mises)

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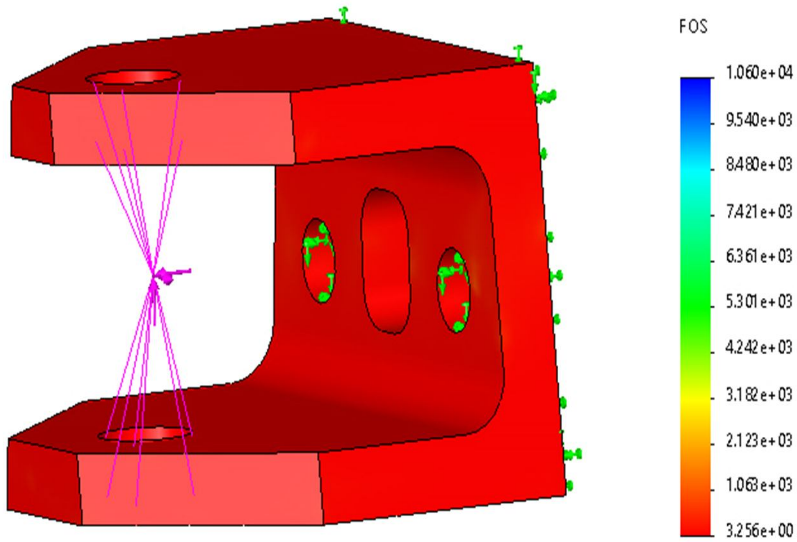

Fig. 23 Factor of safety

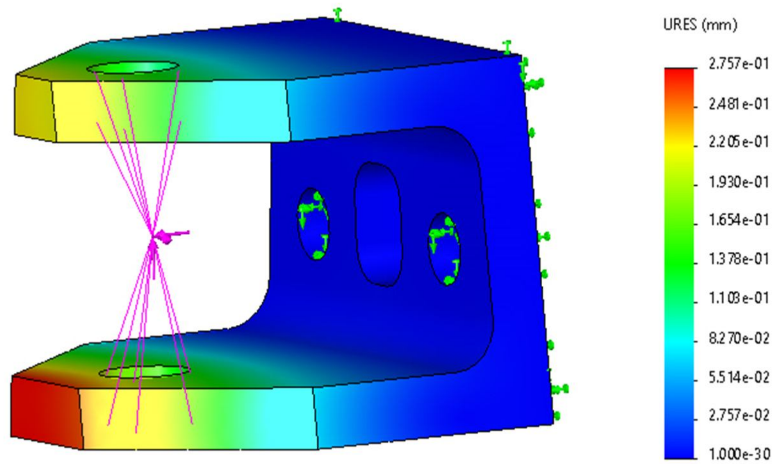

Fig. 24 Total Deformation

2) Analysis of Lower Bracket

TABLE XII
PROPERTIES OF AISI 4340

Constraints	Bolt mounting and body surface
Force	Remote load
Equivalent stress	1.55e+08 N/m2
Maximum deformation	1.22e-01 mm
FOS (ultimate)	3.2

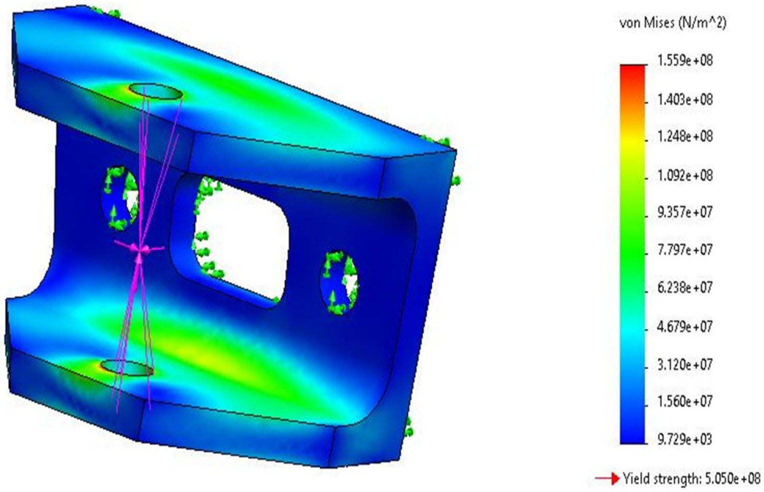

Fig. 25 Stress (Von Mises)

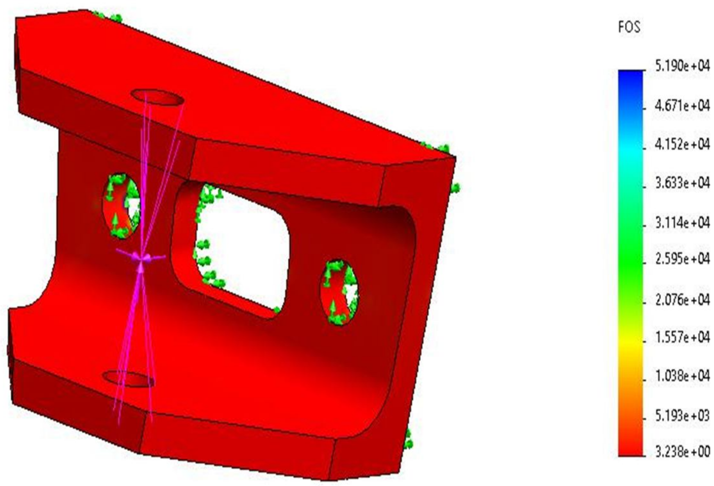

Fig. 26 Factor of safety

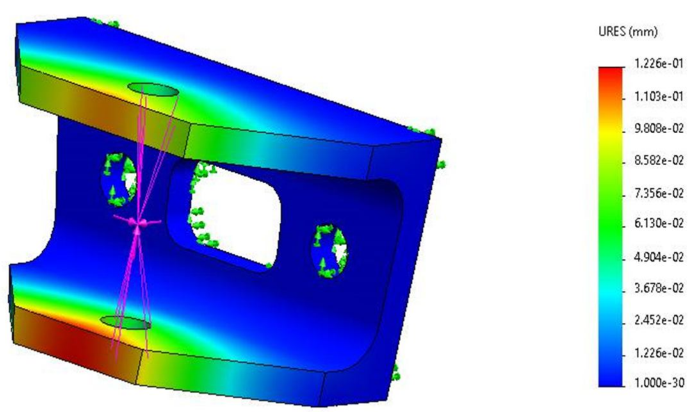

Fig. 27 Total Deformation

Here in both upper and lower brackets the value of FOS is more than 3 and deformation is very less. So, the design of both the brackets is safe and stresses are in permissible range.

V. DIFFERENTIAL

For any vehicle to move, torque is required to be supplied by the engine. This is transmitted by the transmission to the drive shaft leading to the final drive unit which contains the differential. A differential is a mechanical device which allows the transmission of torque and rotation from the gearbox to the wheels. Its utility comes into effect when the vehicle goes through a corner. While cornering, the inner tyre travels less as compared to the outer wheel. So in order to prevent scrubbing, the former needs to move more slowly than the latter. Without a differential, both wheels would rotate at same speeds through the corners resulting in unpredictable handling, damage to the tyres and roads as well as strain on the entire drivetrain systems.

A. Selection

1) There are, broadly speaking, three types of differential available in the market
   a) Locked Axle or spool
   b) Open Differential
   c) Limited – Slip Differential
2) Apart from this, various other options such as viscous differential and Electronic traction control units are also available
3) Open differential was considered after carefully going through all the concerning parameters. It allows the constant average rotational velocities for both wheels even when any one of them is rotating fast and the either one slow. It will allow equal torque split on both wheels as compared to the locked axle and is cost efficient as compared to the limited – slip differential.

B. Axles

1) Axles are the elements of the drivetrain responsible for the transmission of power to the Hub of the wheels. These shafts were splined on both ends to attach to the plunge joint on the differential side and rzeppa cv joint on the hub side.
2) There were two forms of shafts that were available for this purpose : Solid shaft OR Hollow shafts
3) Small diameter solid shafts of high quality AISI 4340 Normalised Steel was selected for this purpose considering the following factors
   a) The hollow shaft will require the large diameter for the same strength with larger volume which was a concerning factor given the packaging space of the vehicle.
   b) Also, the ends of the shafts were meant to be splined and there was a decent chance of failure of bonding with the joints.

C. Calculations

The axle shaft will undergo a shearing effect when subjected to the torsion. The maximum shear stress due to this should be less than the allowable stress

$$\text{Allowable stress} = \frac{\text{Proof stress}}{\sqrt{3}}$$

$$\boldsymbol{\sigma_y} = \boldsymbol{\tau_y} / \boldsymbol{\sqrt{3}}$$

$$\sigma_y = 470 * 0.58 = 272.6\ \text{Nmm}^{-2} \qquad (\text{yield strength} = 470\ \text{MPa})$$

The maximum stress subjected to a solid circular shaft under torque **T** is given by :

$$\boldsymbol{\tau_{max}} = \mathbf{2T / \pi R^3}$$

$$R^3 = 2T / \pi\, \tau_{max}$$

Design torque T = 1200 Nm was assumed after considering peak torque, FDR ratio and Primary reduction with sufficient safety factors. Hence,

$$R^3 = (2 \times 1200 \times 10^3 / \pi \times 470 \times 0.58\ )$$

$$R^3 = 2802.43$$

$$R = 14.10$$

Therefore, the standard size of shaft with **radius 20 mm** was considered finally.

D. Chain Drive

1) Chain drives are one of the most reliable and efficient methods of power transmission from an engine. It consists of a chain, driving sprocket and driven sprocket. The driving sprocket is connected to the output shaft of the engine which is engaged with the chain to transfer power to the driven sprocket located at some distance.
2) Roller chain drive was selected for the transmission of power. In this, the chain is in contact with the number of sprocket teeth. There is a percentage or part of total load carried by the chain to which each engaged tooth is subjected. The chain will also experience tension which will increase from slack side strand to the tight side strand of the chain drive. This chain tension and the load on each tooth of the sprocket depends on the no. of engaged teeth, elastic and frictional properties, tooth pressure angle, etc
3) After calculating chain tension and other forces, a standard 525 chain was selected. It is due to the fact that the parent engine has a chain of the same size and the parts of this size are easily available in the market which would be helpful in case of failure.
4) The no. of sprocket teeth were considered on the basis of the acceleration requirements for the vehicle taking acceleration thrust, gross wt. of the vehicle, peak torque of the engine, driveline efficiency and gear ratio into consideration.
5) Sprocket are subjected to high rotational speeds and power which makes it more prone to failure as compared to the other elements of the drivetrain.Therefore, static analysis was done on the sprocket to ensure that the stress induced due to maximum load on the sprocket teeth are less than the yield strength of the sprocket material which was Mild steel in this case.
6) Since the sprocket will also undergo cyclic loads, it was necessary to carry out fatigue analysis to ensure that the minimum fatigue life of the sprocket is of sufficient time period for the use.
7) These analyses were carried out using FEA to ensure the reliability and safety of proposed design in ANSYS software. Later, the results from this analysis were used for weight reduction using topology optimization and several modified designs were proposed. Further analysis was also done on the modified design before finalising.

VI. CONCLUSIONS

This work has provided a comprehensive literature review of existing analysis carried out in terms of design, stress, fatigue, optimization analysis of the hub and upright assembly. The design procedure was done by keeping the suspension geometry as primary reference for upright design. The hub is created with minimum weight criteria for enduring the loads encountered by formula student vehicle.

The purpose of this project isn't solely to design and manufacture the upright assemblies for the automobile, however additionally to provide an in-depth study within the method to arrive at the final design. In terms of quantitative enhancements, the design illustrates a major weight reduction.

The FEA result indicates that the upright assembly is able to perform safely in real track condition as per performance demand. The FEA could be a useful tool since it provides correct results to assess strength and fatigue life of the hub and upright. The same may also be applied for shape optimization of hub and upright.

Principally FEA of hub and upright assembly were done on mentioned objectives, simulating actual operating conditions in software system. The review shows that hub and upright are designed for various kinds of vehicle according to their use. Applied boundary conditions are calculated analytically.

This paper provides a basic layout encompassing key factors for designing all elements concerned within the design of wheel assembly which might be instrumental for improvement within the future design.

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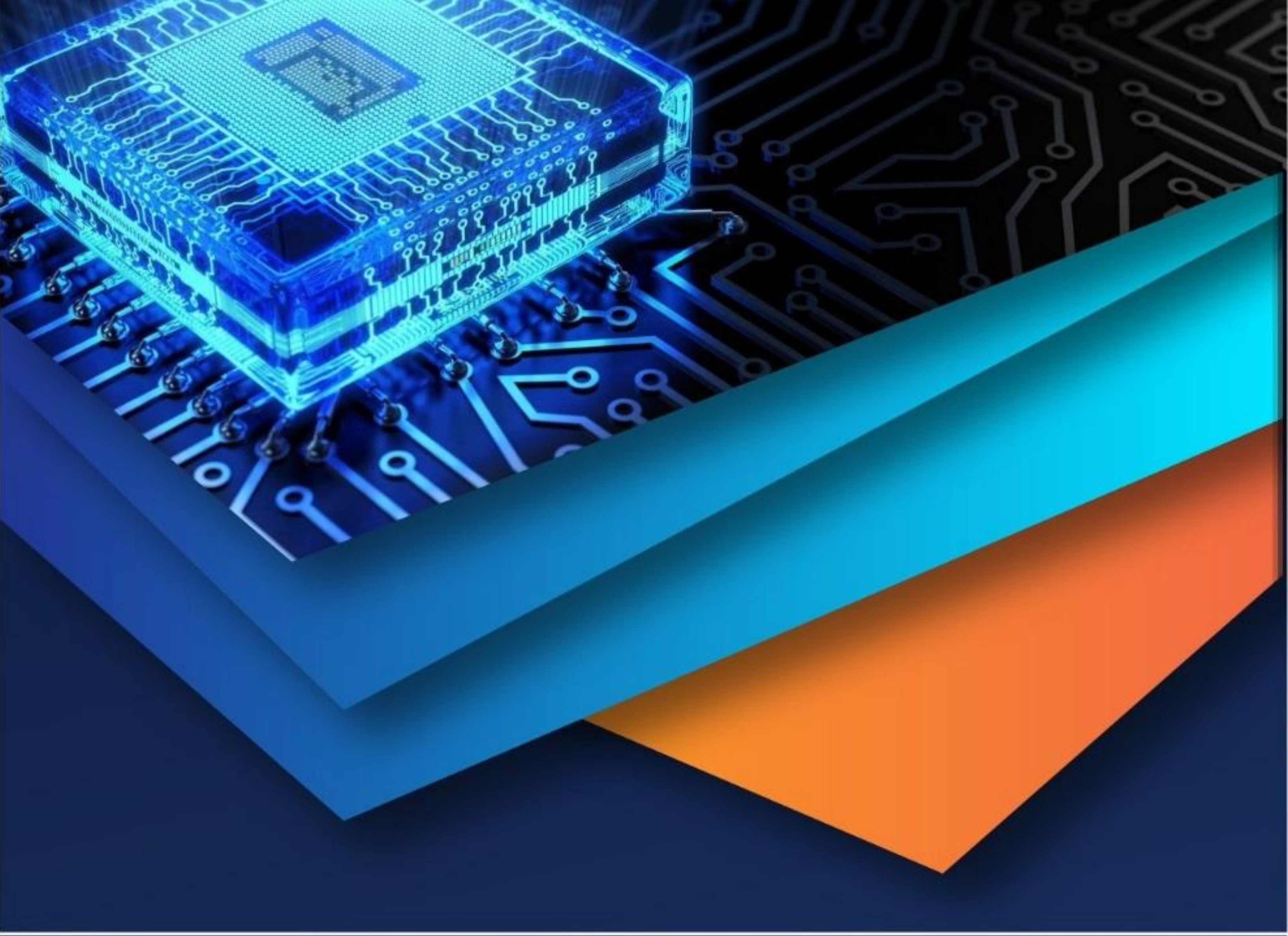

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