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# A REPORT ON DESIGN OF DRIVETRAIN FOR FSAE CAR

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## ABSTRACT

Wheel assembly is a major part of automotive design which connects the wheel to the suspension system and therefore transfers the force from road to the suspension components. It also holds the brake system as well as facilitates steering of the vehicle. Power train department is also concerned with the transmission of power from engine to wheels.

Aim of the report is to describe the process and simulation that goes into designing of hub, upright and differential mounting of a formula student car as well as calculating the size of sprocket for maximum acceleration.

## 1. INTRODUCTION

### 1.1 Problem Definition

Power train is that department of the car concerned with delivering the power from engine to the wheels. Every power transmitting element from engine to wheels are included in the department. It deals with the design, manufacturing and assembly of hubs, uprights, differential mounts, axle, differential, wheels and tires. These components play an important role in deciding the speed, acceleration and mass of the car. Components are designed and assembled after co-ordinating with suspension, chassis, brake and steering department.

### 1.2 Project Objectives

The project aimed to reduce the difficulties faced during the assembly in 2018 car and to completely eliminate the failures incurred in the assembly. To accomplish that object, a series of measures were taken from changing the arrangement of bearing to manufacturing hubs of steel instead of aluminium.

### 1.3 Design Constraints and considerations

The project was designed in compliance with the rules of Formula Bharat 2020 for participation in the same. The major design considerations in this year's car are as follows

- Packaging in 10'' wheels
- Reduction in unsprung mass
- Better acceleration
- Ease in assembly and disassembly
- Better locking of bearing to reduce backlash
- Adjustable differential mounting
- Ease of manufacturing of components

## 2. BEARING SELECTION

A pair of single row tapered roller bearing is used in back-to-back arrangement compared to previous car's face-to-face arrangement. For both tapered roller and angular contact ball bearings, the distance L between the pressure centres is longer when the bearings are arranged back -to -back compared with bearings arranged face to face. This means that bearings arranged back-to -back can accommodate relatively large tilting moments even if the distance between the bearing centres is relatively short. They are designed to accommodate combined loads, i.e. simultaneously acting radial and axial loads. Matched bearings arranged back-to-back have load lines that diverge toward the bearing axis to provide a relatively stiff bearing arrangement that can also accommodate tilting moments. Axial loads in both directions can be accommodated, but only by one bearing in each direction.

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After careful consideration, it was decided that 320-32X bearing be used as the load ratings of bearing were well above the load on hubs and uprights and the size of bearing was optimum.

**Bearing details**

ID - 32mm, OD – 58mm, Width – 17mm
Dynamic load rating – 36.9 KN, Static load rating – 46.5 KN

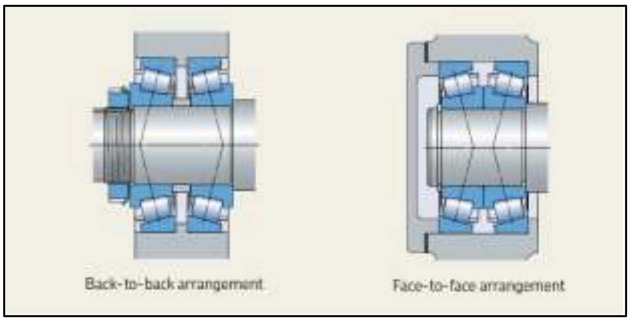


***Fig.1***

## 3. 3.WHEELS AND TIRES

Keizer 10'' aluminium wheels are used for lighter weight, lower CG and higher rollover stability. Wheels are custom designed for required offset, lug bolts size and PCD. PCD is 90mm which reduces the weight of hub.

Hoosier soft compound wet tires are being used for better grip and all weather purpose. We also have a set of Carlisle wet tires (205/50-10).

## 4. DIFFERENTIAL AND DIFFERENTIAL MOUNTING

Dry open type differential is being used instead of limited slip differential as it serves the purpose of providing grip and better handling and stability on a dry track and it is cost efficient at the same time. Sprocket mounted on the differential have teeth machined to our needs. Below is the calculation for required number of teeth.
400kg assumed mass
Coefficient of friction = 1
F = un = 4000 N
Tire radius = 0.2286m
T = 4000*0.2286 = 914.4
Front sprocket teeth = 16
Rear sprocket teeth = x
60N.m = torque produced by engine
77/39 = primary drive ratio
39/14 = 1$^{st}$ gear ratio
60*(77/39)*(39/14)*(x/16) = 914.4
X = 45
There are 52 teeth in the sprocket for better acceleration.

Differential mountings are adjustable with fixed mounting at one end and adjustable at other end with rod ends. They are made of Al 6061 (yield strength – 275 mpa) which suffice our strength requirements and is lightweight at the same time. Load acting on the differential mounting is only the weight of differential. Radial ball bearing 61907 is used to support the differential as axial loads acting on the mountings are very low and it can easily withstand load of differential. The bearings are stopped by providing retaining rings on either side of bearings.

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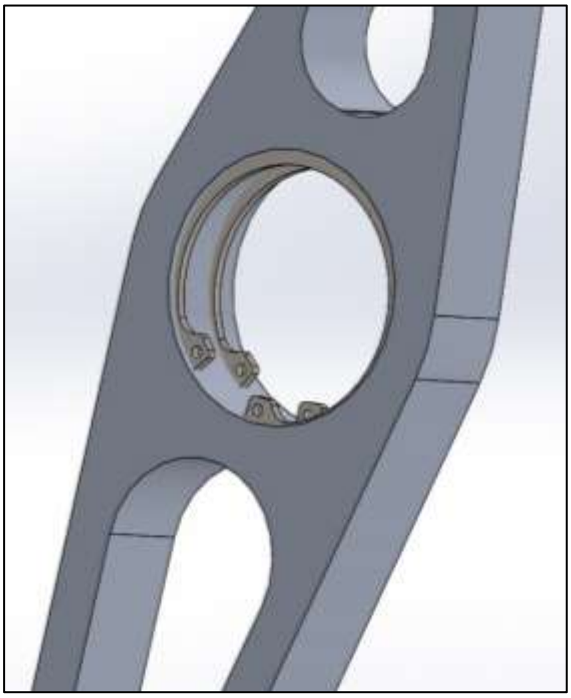

***Fig.2***

***Table.1***

| | |
|---|---|
| Constraints | Mounting surfaces |
| Forces | Bearing surface |
| Magnitude | 500 N |
| deformation | 5.9e-6 m |
| Equivalent stress | 4.67e6 Pa |

Using spool was another option but the main shortcoming of spool is that it can cause the rear of the vehicle to spin out, fish-tale or can cause lot of noise because of twisting and release of axles. This can also lead to breaking of axles eventually. It can also lead to wear in tires. On the other hand, open differential is advantageous for following reason.

- Allows for completely different wheel speeds on the same axle, meaning no wheel slip will occur while going around a corner, as the outside tyre will travel further.

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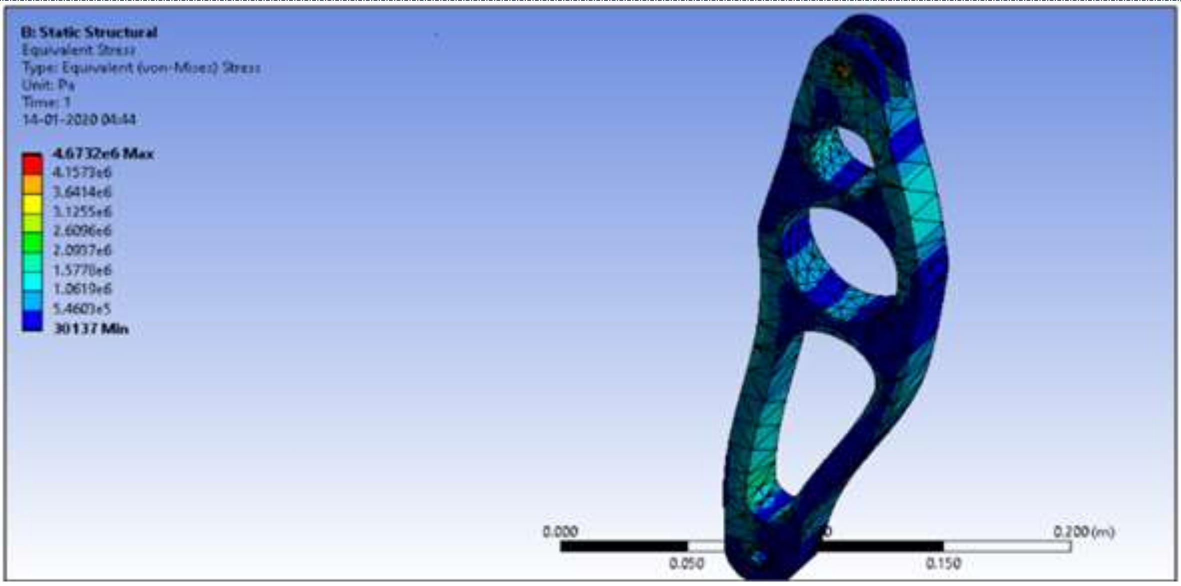


***Fig 3***

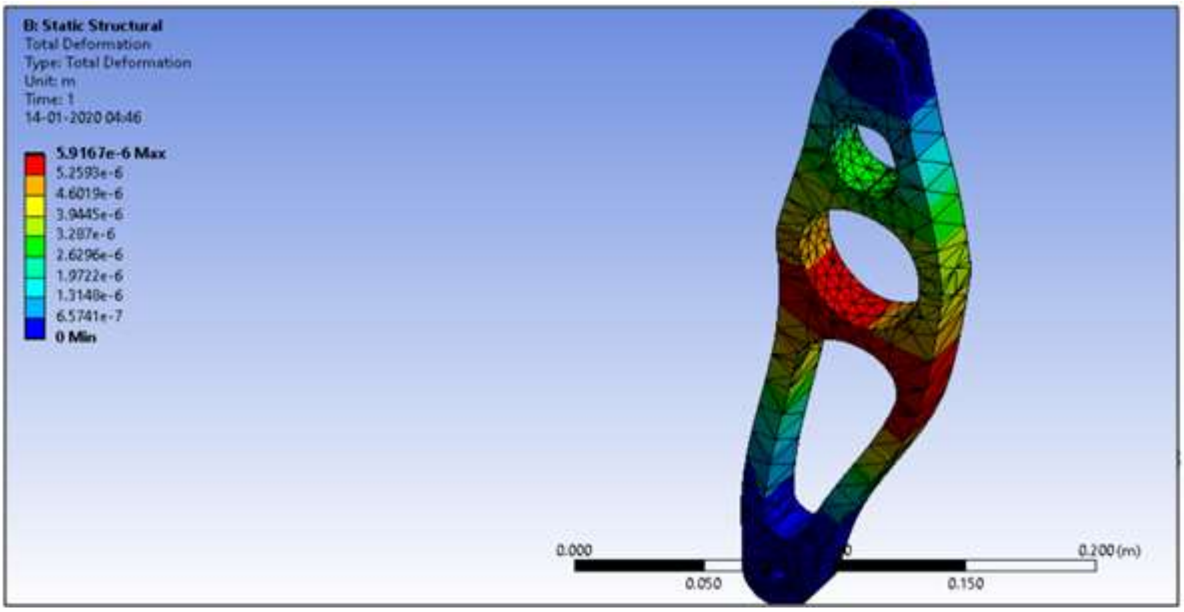


***Fig 4***

## 5. AXLE

Axle used in the prototype vehicle are plunge joint and rzeppa cv joint on the same shaft. Plunge joint is used on the differential side which provides axial adjustment of the axle. Rzeppa joint is used on the wheel side of the axle providing angular adjustments for better fitting while acting as cv joint at the same time.

## 6. HUBS

### 6.1. Introduction

Hub is that component of the assembly responsible for connecting wheels to the suspension system. Hubs are directly mounted to the wheels and transfers the power from axles to wheels. Hubs are supported by wheel bearings on uprights. Brake rotors are also mounted on hubs and it has internal splines to fit the axle.

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**6.2. Method**

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

- PCD of the wheel mounting plate
- Bearing used
- PCD of brake rotors
- Locking methods of bearing
- Front or rear hub
- Size of brake caliper
  a. wheel mounting plate is decided by the PCD of lug bolts. This is the first step in creating the design of hub in 3D modelling software.
  b. Shaft connecting the wheel mounting plate and brake rotor mounting plate. Width isdecided by the size of brake caliper and diameter by performing iterations in simulation software for required strength.
  c. Brake rotor mounting plate is decided by PCD of brake rotors.
  d. Bearing collar for mounting of bearing. Width is dependent on size of brake caliper and diameter on ID of bearing used.
  e. Bearing shaft used to mount the bearings and to transfer the load to suspension system. Diameter depends on ID of bearing and width on locking method used.

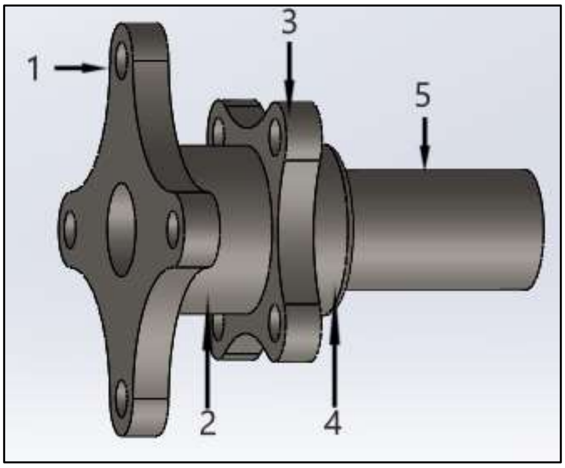


***Fig.5***

The wheel hub has to be strong enough to withstand the following forces

- Force due to acceleration or deceleration
- Cornering
- Wheel travel or bump
- Brake torque or torque from the axles

**6.3. Front hub**

The sole function of the front hub is to support the wheels and connect it to the suspension system. Front hub is non-power transmitting component hence it doesn't have splines to fit the axle. Bearings in the front assembly are locked with the help of a lock nut for which threads are provided on the shaft of front hub and a slot for tab washer.

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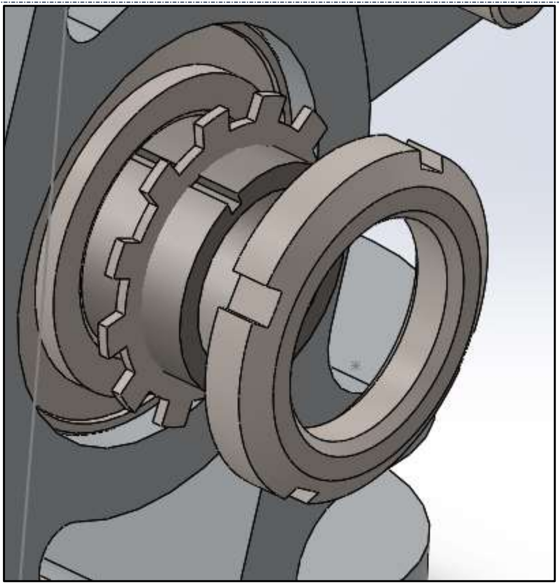

***Fig.6***

**6.4. Rear hub**

The function of the rear hub is to support the wheels and connect it to the suspension system as well as transmitting the power from axle to wheel. For that purpose, rear hubs have internal splines to fit the axle and to transmit the power. Bearing in the rear assembly is locked by retaining ring hence a groove is machined on the shaft to place ring.

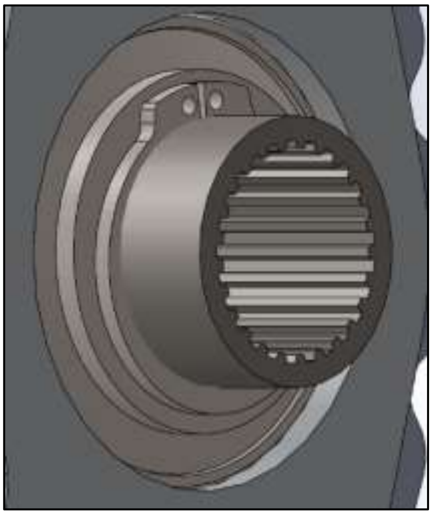

***Fig 7***

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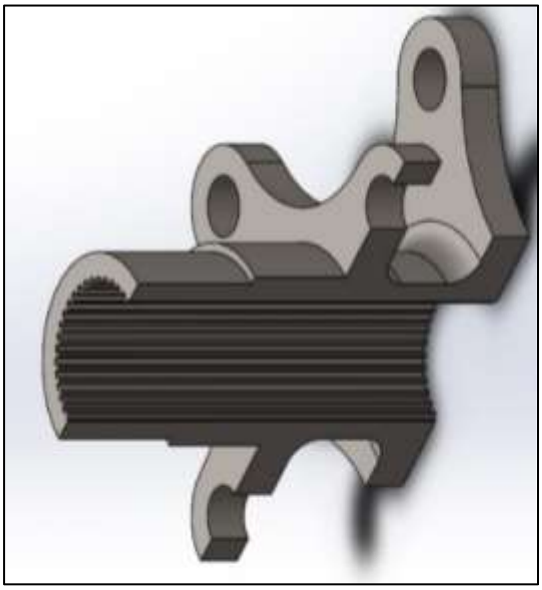

***Fig.8***

**6.5. Material selection**

SS 304

***Table.2***

| | |
|---|---|
| Young's modulus | 200Gpa |
| Elongation at break | 8% |
| Fatigue strength | 210Mpa |
| Poisson's ratio | 0.28 |
| Shear strength | 400Mpa |
| Tensile strength | 580Mpa |
| Yield strength | 230Mpa |

AISI 4340

***Table.3***

| | |
|---|---|
| Young's modulus | 190Gpa |
| Elongation at break | 12% |
| Fatigue strength | 330Mpa |
| Poisson's ratio | 0.29 |
| Shear strength | 430Mpa |
| Tensile strength | 690Mpa |
| Yield strength | 470Mpa |

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Hubs are made of AISI 4340 as it has high tensile and fatigue strength which are desirable properties for manufacturing of splines.

**6.6. Analysis**

The force due to braking in the longitudinal direction is calculated as follows
Stopping distance, s = 5m
Initial velocity, u = 13.8m/s
Mass of the car, m = 350kg
Calculating deceleration by $v^2$-$u^2$ = 2*a*s
a = -19m/$s^2$
assuming 60% weight transfer to the front, force due to the braking on front hub = 3990N
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

$v = \frac{d}{t} = \frac{57.33}{7} = 8.19$ m/sec

$a = \frac{v^2}{r} = \frac{8.19^2}{9.125} = 7.35$ m/sec$^2$

f = 2600N
Considering the weight of the car, bump force = 2000N.

The FEM analysis was done on ANSYS workbench and the results were carried out as follows
Quadratic method is used to generate the mesh with an element size of 5mm.

***Table.4***

| Details of "Force" | |
|---|---|
| **Scope** | |
| Scoping Method | Geometry Selection |
| Geometry | 2 Faces |
| **Definition** | |
| Type | Force |
| Define By | Components |
| Coordinate System | Global Coordinate System |
| X Component | 3990. N (ramped) |
| Y Component | 2000. N (ramped) |
| Z Component | 2600. N (ramped) |
| Suppressed | No |

***Table.5***

| | |
|---|---|
| Constraints | Wheel mounting points |
| Force | Shaft |
| Equivalent stress | 2.736e+008 Pa |
| Maximum deformation | 1.309e-004 m |
| FOS (ultimate) | 2.72 |

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Maximum stress is generated at the edge of wheel mounting points and maximum deformation occurs at the tip of the shaft. The design is safe.

Moment generated at wheel mounting points during braking is calculated as follows
Force due to braking is longitudinal direction =3990 N

tire radius = 0.2286 m
Moment generated ≈ 900Nm
Quadratic method is used to generate the mesh with an element size of 5mm.

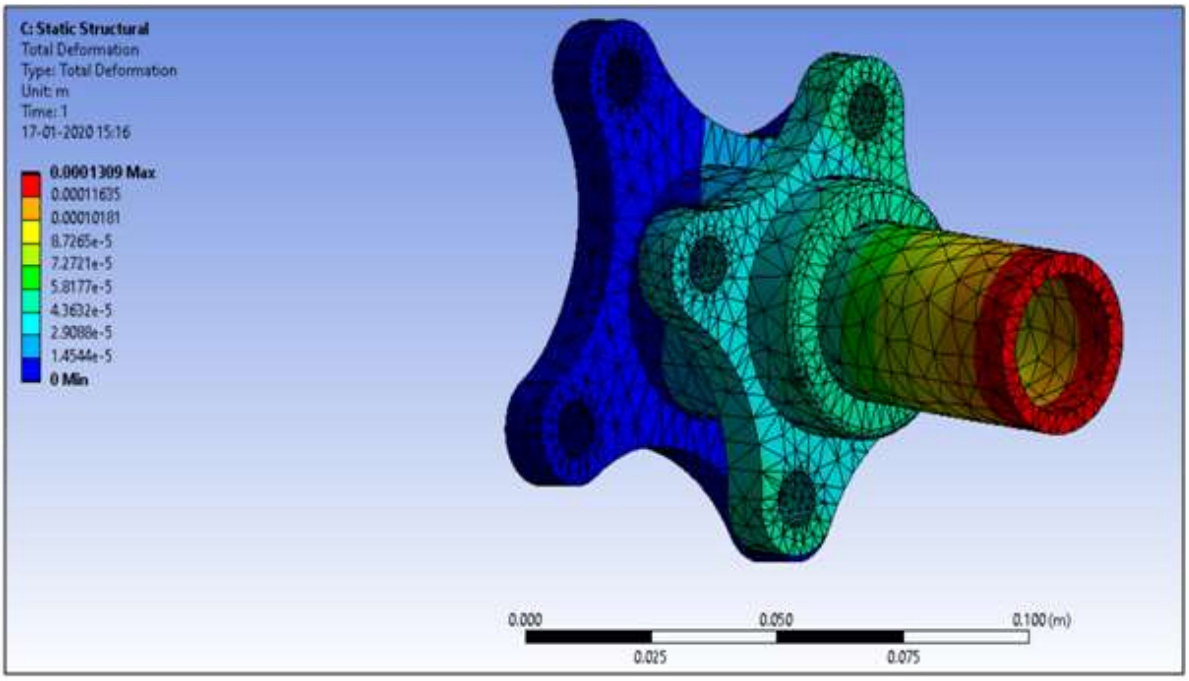


***Fig 9***

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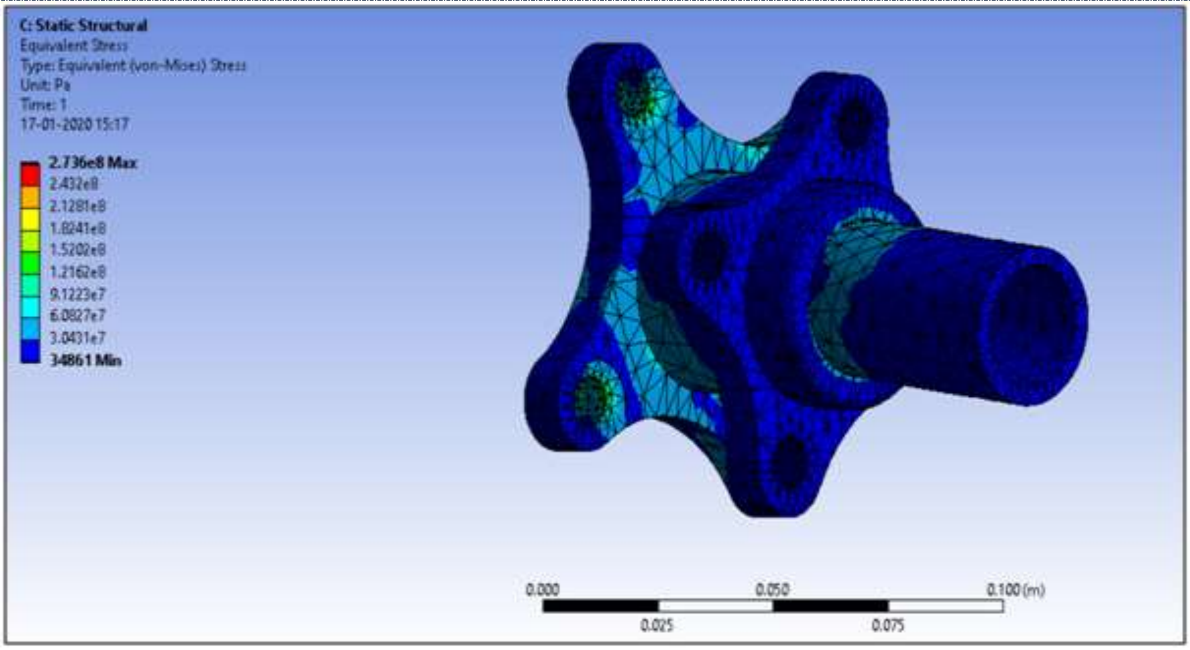


*Fig 10*

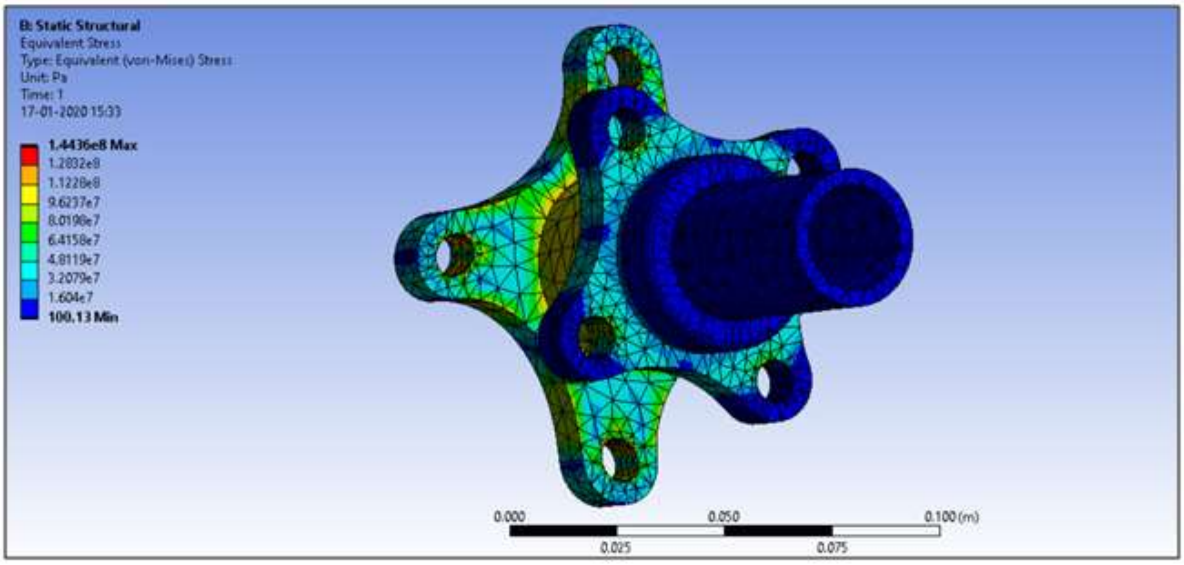


*Fig 11*

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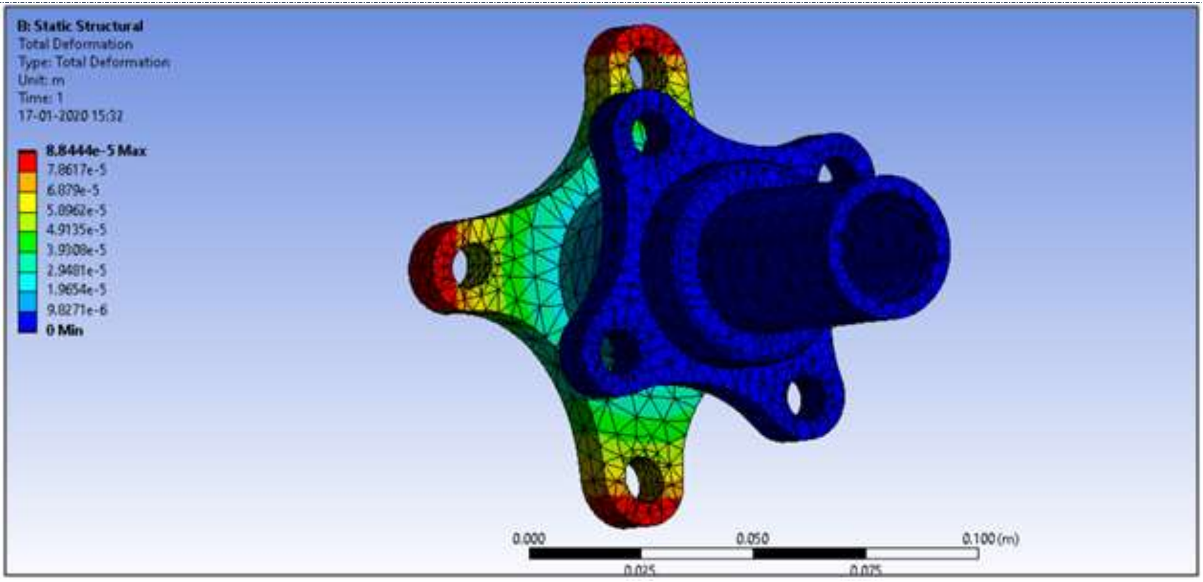


*Fig 12*

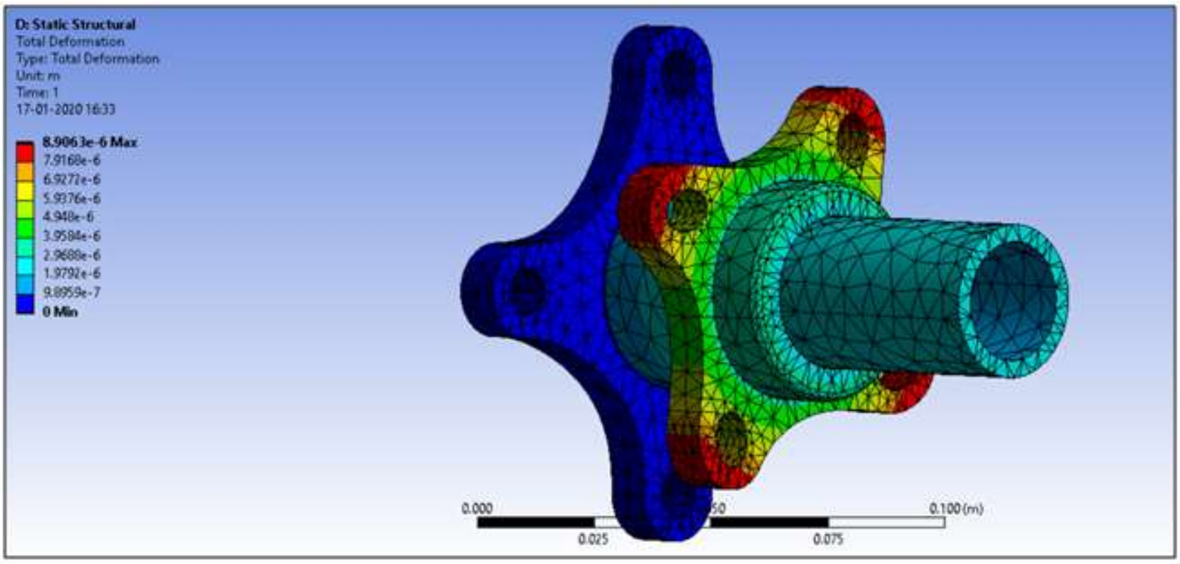


*Fig 13*

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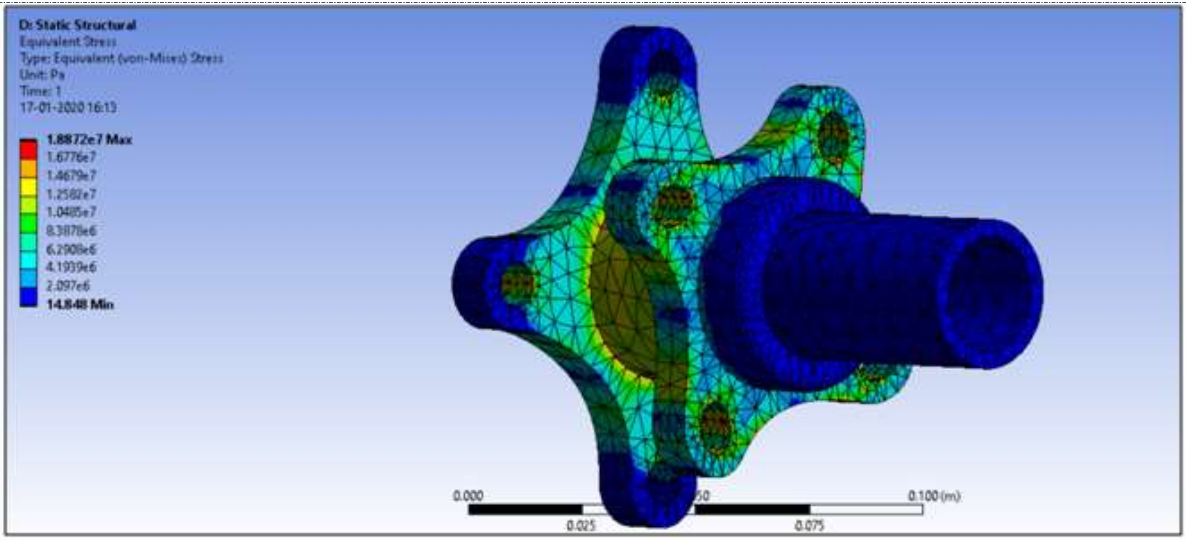


*Fig 14*

***Table.6***

| Details of "Moment" | |
|---|---|
| **Scope** | |
| Scoping Method | Geometry Selection |
| Geometry | 8 Faces |
| **Definition** | |
| Type | Moment |
| Define By | Vector |
| Magnitude | 900. N·m (ramped) |
| Direction | Click to Change |
| Suppressed | No |
| Behavior | Deformable |
| **Advanced** | |

***Table.7***

| | |
|---|---|
| Constraints | Brake rotor mounting points |
| Force | Wheel mounting points |
| Equivalent stress | 1.4436e+008 Pa |
| Maximum deformation | 8.8444e-005 m |
| FOS (ultimate) | 5.32 |

Maximum stress is generated at wheel mounting points and maximum deformation occurs at outer edge of wheel mounting points.
Moment generated at brake rotor mounting points while braking is calculated as follows
After consulting the brake department, clamping force was found to be = 1302 N
Moment generated at the front wheels = 115.5Nm
Quadratic method is used to generate the mesh with an element size of 5mm.

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***Table.8***

| Details of "Moment" | |
|---|---|
| **Scope** | |
| Scoping Method | Geometry Selection |
| Geometry | 8 Faces |
| **Definition** | |
| Type | Moment |
| Define By | Vector |
| Magnitude | 115.5 N·m (ramped) |
| Direction | Click to Change |
| Suppressed | No |
| Behavior | Deformable |
| **Advanced** | |

***Table.9***

| | |
|---|---|
| Constraints | Wheel mounting points |
| Force | Brake rotor mounting points |
| Equivalent stress | 1.8872e+007 Pa |
| Maximum deformation | 8.9063e-006 m |
| FOS (ultimate) | 41.8 |

The value of FOS is much greater than required value because the thickness of mounting plate is greater than required. This was necessary to fit the caliper because increase in shaft length would have led to more deformation.

## 7. UPRIGHTS

### 7.1. Introduction

Upright is that component of the assembly which houses wheel bearings, supports hub, brake caliper and connects wheel to suspension and steering system. All the forces travels from hub to upright to the suspension system. It also has to withstand torque from brake caliper and latitudinal force on tie-rod mounting. Hence upright has to have enough strength to withstand these forces.

### 7.2. Method

The design of upright is dependent on following factors.

- Kpi
- Caster
- Bearings
- Distance between lbj and ubj
- Locking method of bearing
- Brake caliper
- Tie rod mounting

The design process starts by deciding the kingpin inclination, caster and distance between upper ball joint and lower ball joint. The next step is designing the bearing housing after deciding the tolerances, bearing arrangement and locking method. A step is provided in the middle of the bearing housing to axially locate the bearing and the step is discontinuous to facilitate easy removal of bearing.

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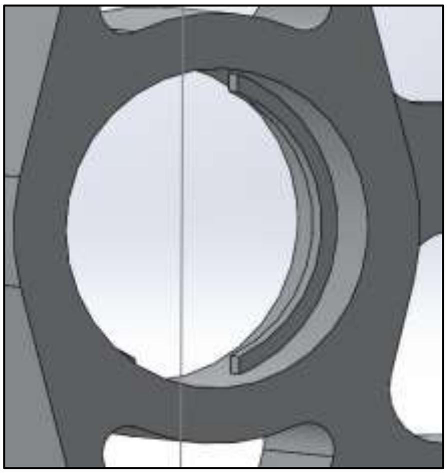

*Fig.15*

After the design of housing, clamps are designed and holes are provides for the wishbones from KPI and caster. Then comes the mounting for brake caliper which are designed with the help of CAD model of the caliper. The final step is to design the mounting for the tie-rods which are decided from the steering geometry.
The upright has to be strong enough to withstand following forces

- Longitudinal force while braking
- Latitudinal force while cornering
- Bump force
- Moment generated at brake caliper mountings while braking
- Force on tie rod mounting while cornering

**7.3. Front upright**
The front upright is deigned to withstand more cornering forces due to steering. Hence it was decided to use 2 single row tapered roller bearing with back to back arrangement. Also the tie rod mounting was designed in accordance with ackerman geometry.

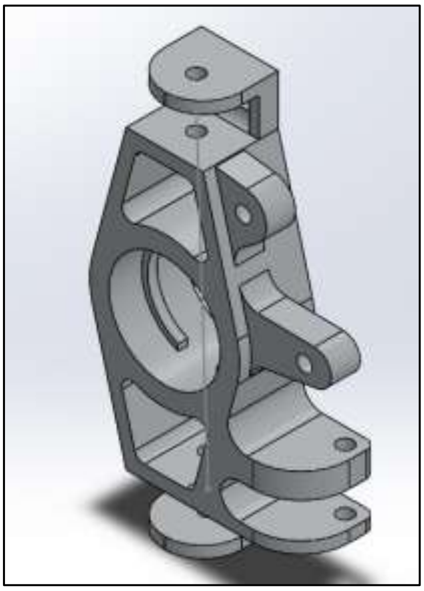

*Fig.16*

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**7.4. Rear upright**

The design of rear upright started by deciding to use radial ball bearing as axial forces on the rear wheel are low. The initial design looked as follows.

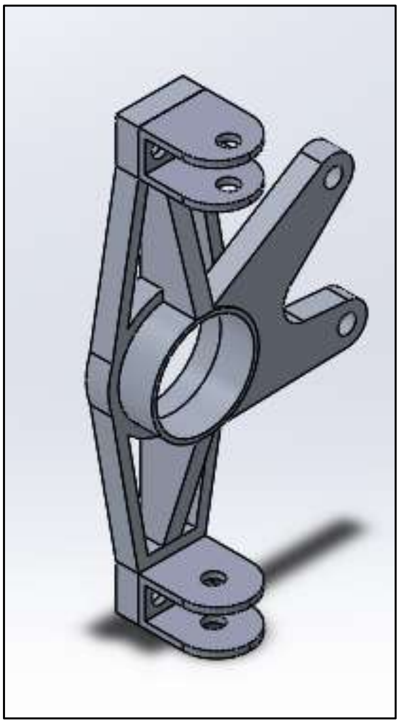

***Fig.17***

But there were some drawbacksin the design. The strength of the upright was very low and the length of brackets had to be long to fit the wishbones which decreased the strength even further, so it was decided to use the design as same as front with some changes in the tie rod mounting.

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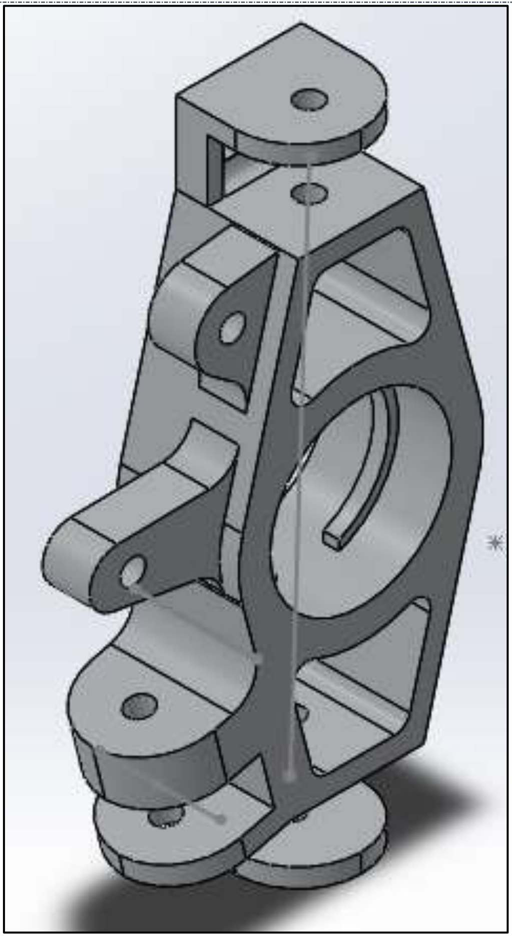

*Fig.18*

**7.5. Material selection**

***Table.10***

Al 7050

| | |
|---|---|
| Young's modulus | 70 Gpa |
| Elongation at break | 2.2% |
| Fatigue strength | 130 Mpa |
| Poisson's ratio | 0.32 |
| Shear strength | 280 Mpa |
| Tensile strength | 490 Mpa |
| Yield strength | 390 Mpa |

***Table.11***

Al 7075-T6

| | |
|---|---|
| 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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Uprights are manufactured from 7075-T6 billets as it's tensile strength and fatigue strength is greater than that of 7050.

**7.6.Analysis**

As we've calculated before,
longitudinal force during braking = 3990 N
latitudinal force during cornering = 2600 N
bump force = 2000 N

The FEM analysis was done on ANSYS workbench and the results were carried out as follows
Quadratic method is used to generate the mesh with an element size of 5mm.

***Table.12***

| Details of "Force" | |
|---|---|
| **Scope** | |
| Scoping Method | Geometry Selection |
| Geometry | 2 Faces |
| **Definition** | |
| Type | Force |
| Define By | Components |
| Coordinate System | Global Coordinate System |
| X Component | 3990. N (ramped) |
| Y Component | 2000. N (ramped) |
| Z Component | 2600. N (ramped) |
| Suppressed | No |

***Table.13***

| | |
|---|---|
| Constraints | Wishbone mounting points |
| Force | Bearing housing |
| Equivalent stress | 8.5549e+007 Pa |
| Maximum deformation | 5.3979e-005 m |
| FOS (ultimate) | 6.7 |

Moment generated at brake caliper mounting points = 115.5Nm
Quadratic method is used to generate the mesh with an element size of 5mm.

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***Table.14***

| Details of "Moment" | |
|---|---|
| **Scope** | |
| Scoping Method | Geometry Selection |
| Geometry | 4 Faces |
| **Definition** | |
| Type | Moment |
| Define By | Components |
| Coordinate System | Global Coordinate System |
| X Component | 0. N·m (ramped) |
| Y Component | 0. N·m (ramped) |
| Z Component | -115.5 N·m (ramped) |
| Suppressed | No |
| Behavior | Deformable |
| **Advanced** | |

***Table.15***

| | |
|---|---|
| Constraints | Wishbone mounting points |
| Force | Brake caliper mounting |
| Equivalent stress | 4.7813e+007 Pa |
| Maximum deformation | 5.346e-005 m |
| FOS (ultimate) | 12.12 |

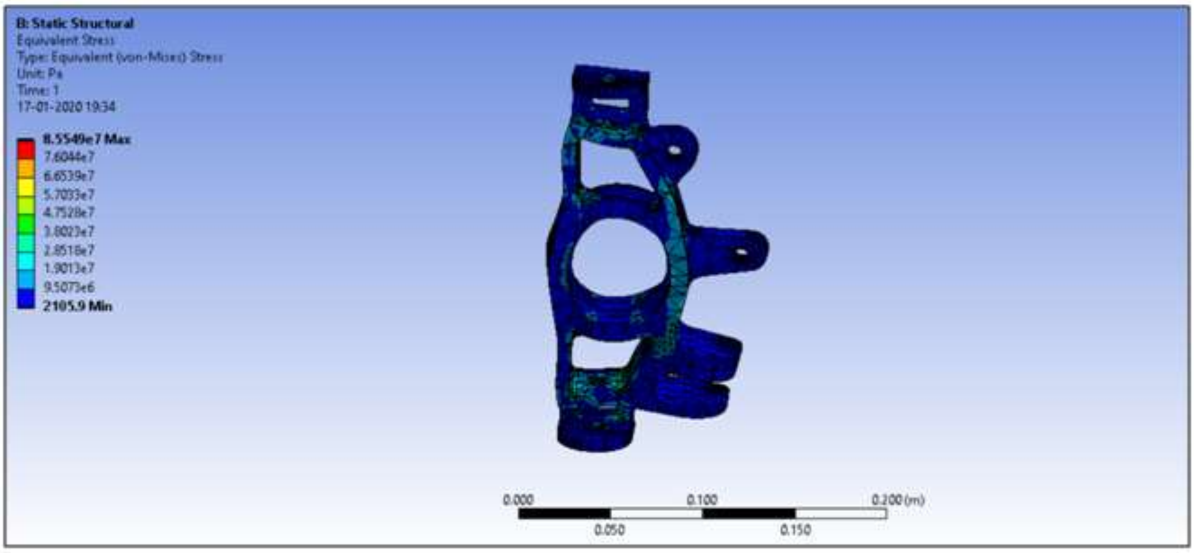


***Fig 19***

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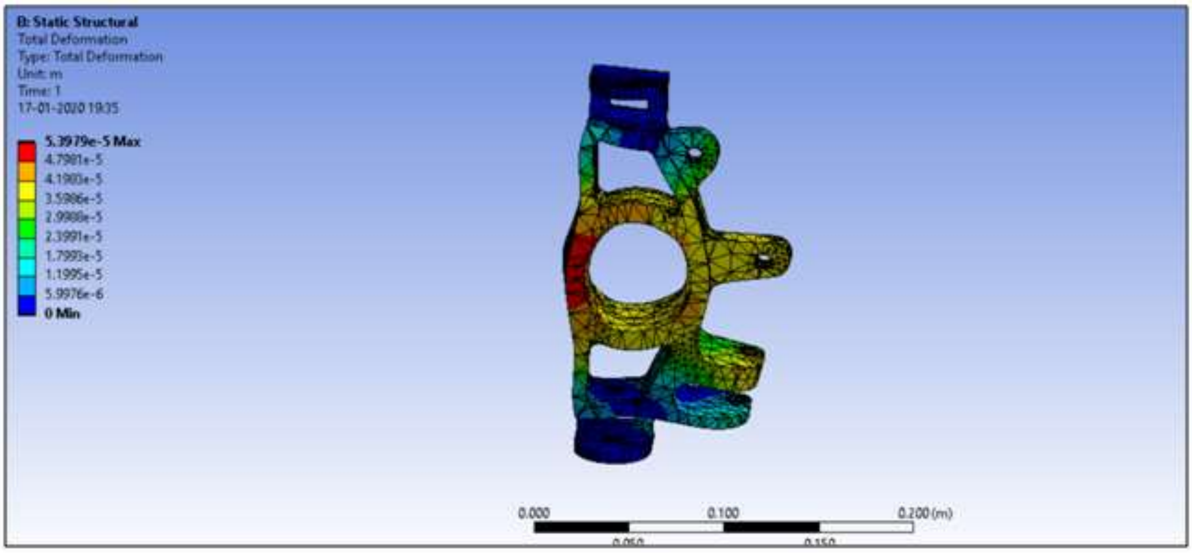


*Fig 20*

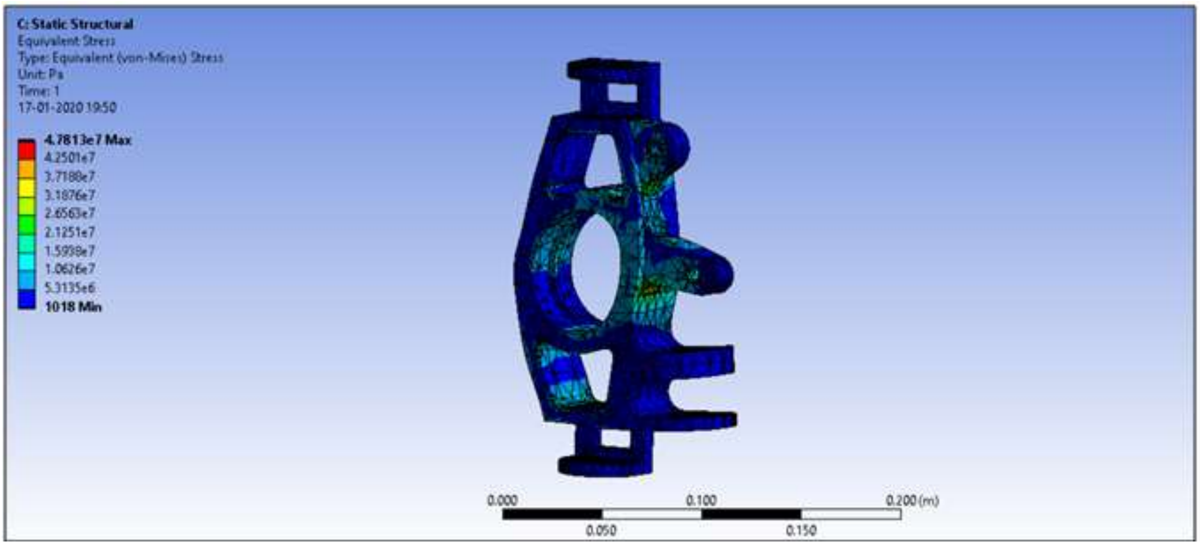


*Fig 21*

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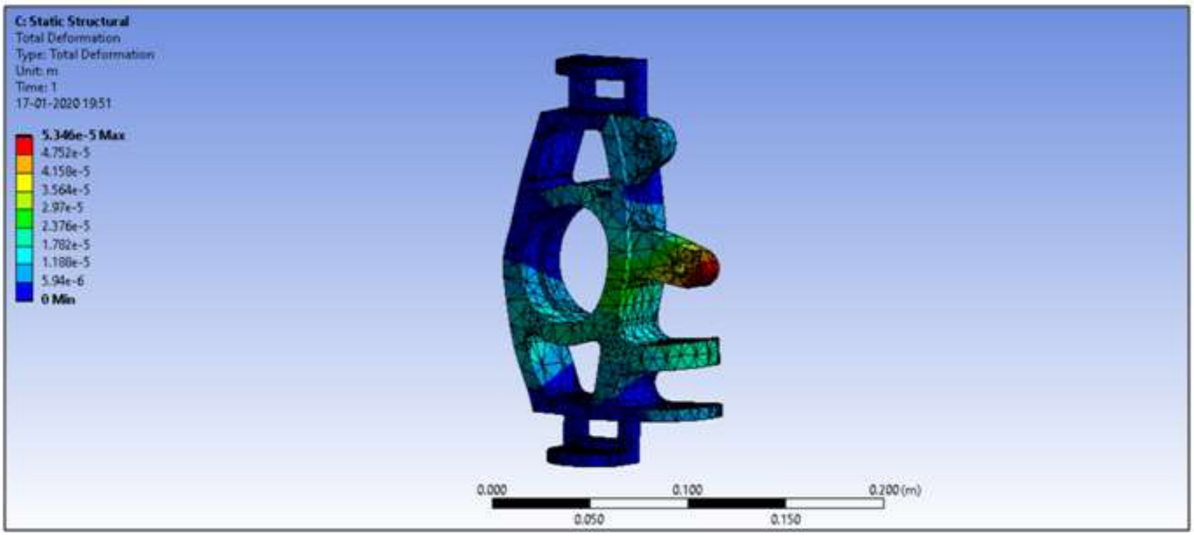


*Fig 22*

Force due to cornering on
tie rod mounting = 2600N
Quadratic method is used to generate the mesh with an element size of 5mm

***Table.16***

| | |
|---|---|
| Constraints | Bearing housing |
| Force | Tie rod mounting |
| Equivalent stress | 5.8464e+007 Pa |
| Maximum deformation | 7.2529e-005 m |
| FOS (ultimate) | 9.8 |

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***Table.17***

Details of "Force"

| | |
|---|---|
| **Scope** | |
| Scoping Method | Geometry Selection |
| Geometry | 4 Faces |
| **Definition** | |
| Type | Force |
| Define By | Components |
| Coordinate System | Global Coordinate System |
| X Component | 0. N (ramped) |
| Y Component | 0. N (ramped) |
| Z Component | -2600. N (ramped) |
| Suppressed | No |

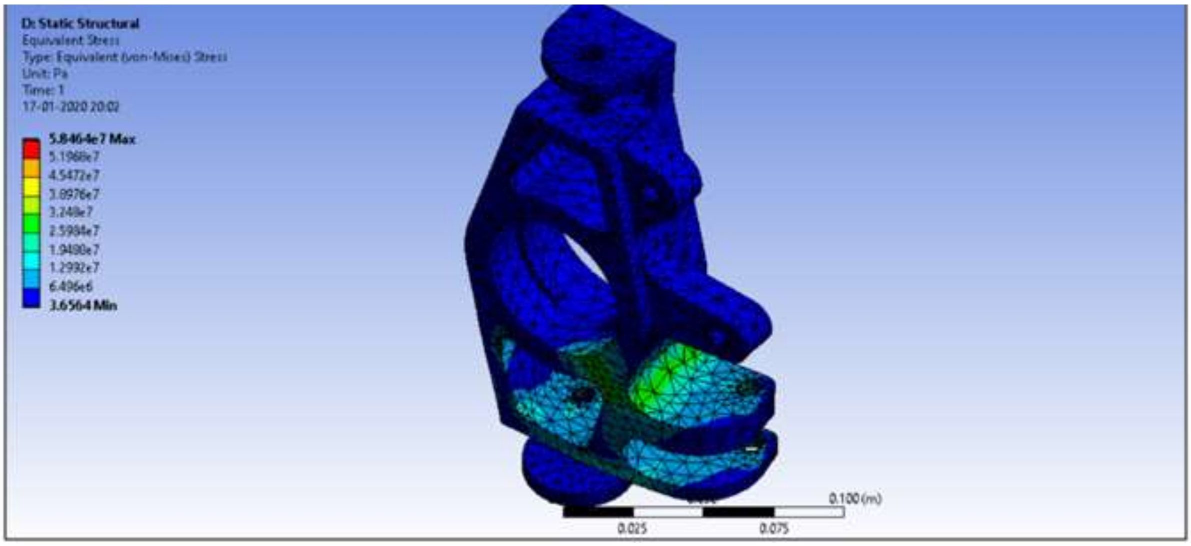


***Fig 23***

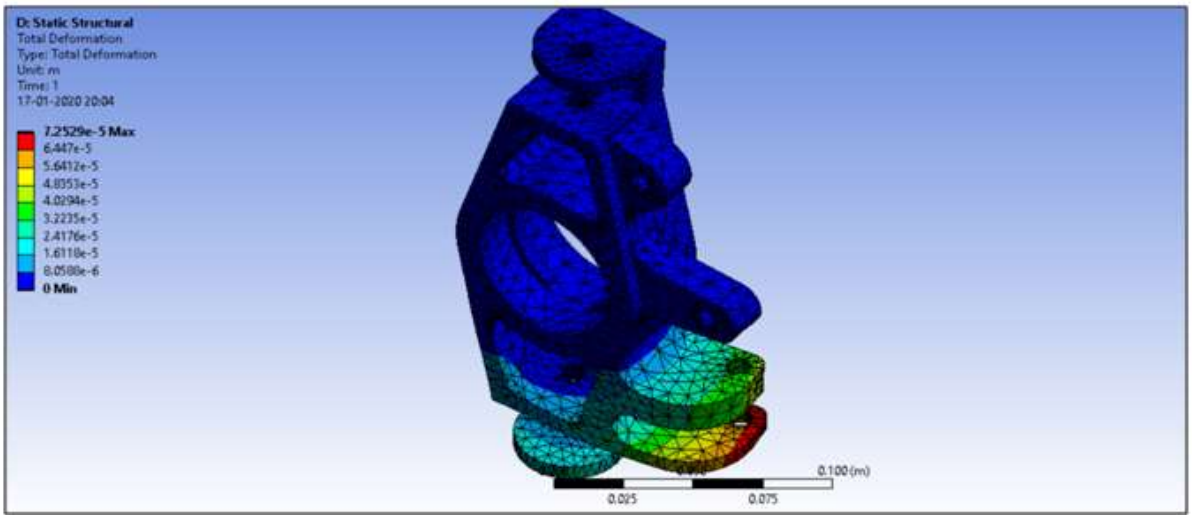


***Fig 24***

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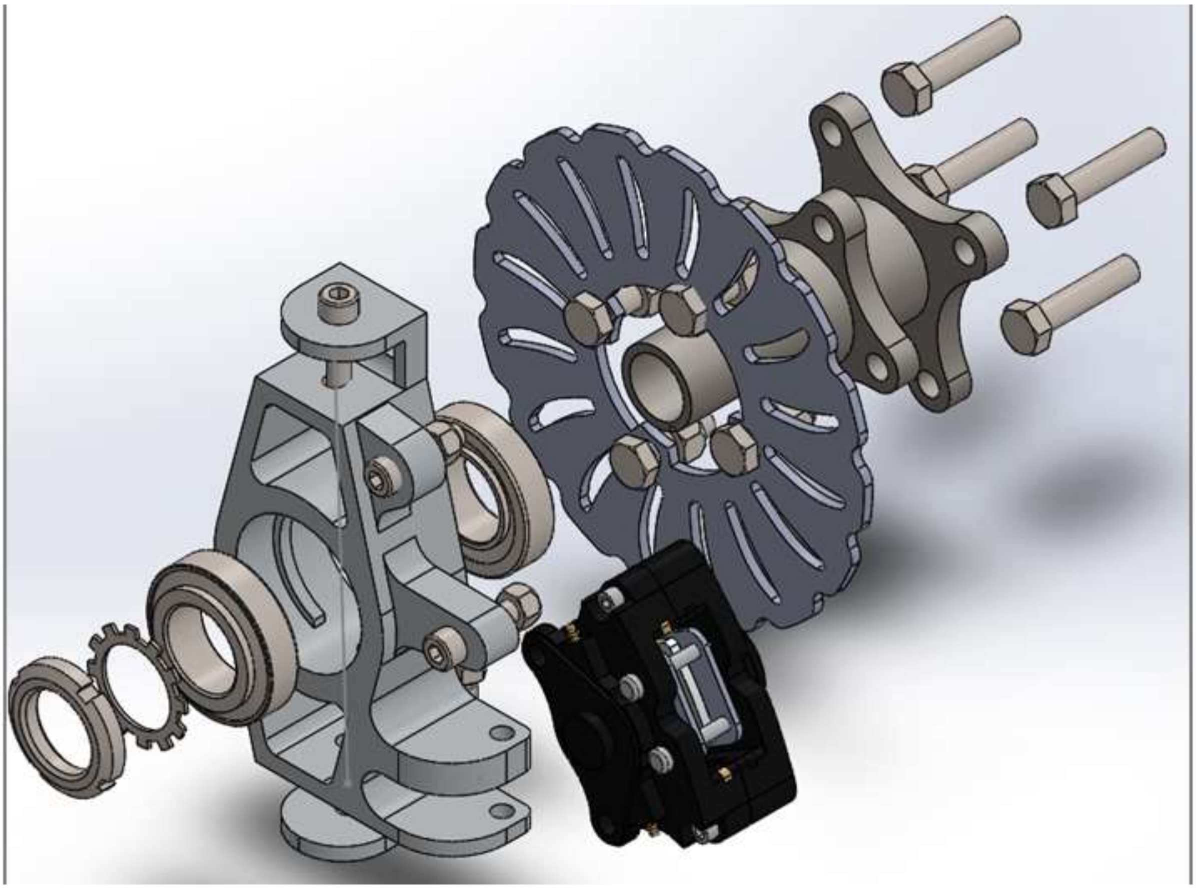

***Fig 25***

ISSN: 2277-9655
Impact Factor: 5.164
CODEN: IJESS7

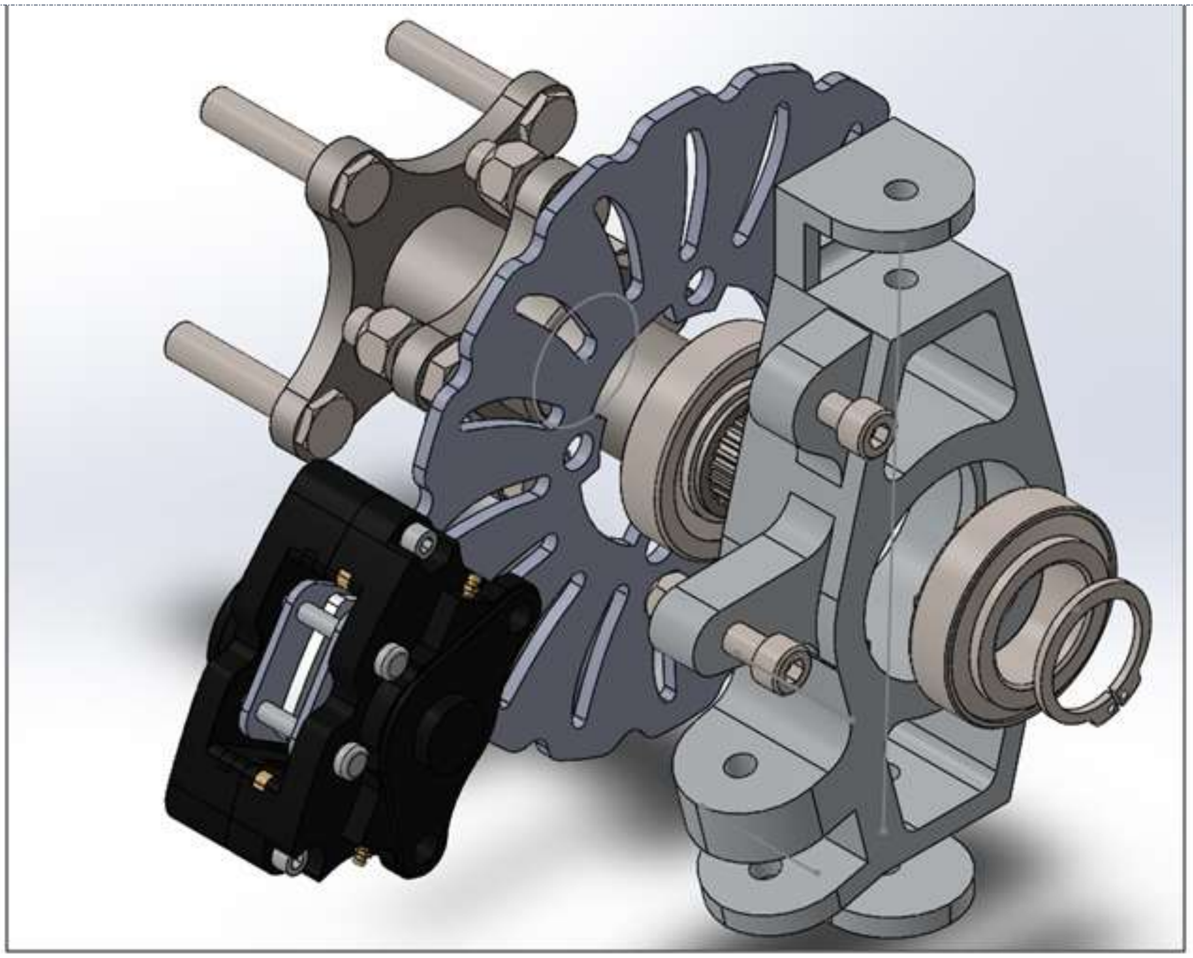

***Fig 26***

## 8. CONCLUSION

The following results were obtained from the report

1. Better acceleration by calculating number of teeth on sprocket
2. Efficient locking and placement of bearing to reduce backlash
3. Development of accurate FEA model with the help of calculations
4. To design adjustable differential mounting.

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