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Power Train Designing for Formula styled Racing Car

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Abstract: A wheel assembly is an integral part of a vehicle's design that connects the wheel to the suspension system and transfers pressure from the road to the suspension system. It also holds the brake system and facilitates steering. Power transmission is also addressed in the powertrain department.
We describe the process and simulation that result in the hub, upright, and differential mounting of a formula student car and the size of the sprocket for maximum acceleration in this report.
As a result of the work done on this project, the resulting car has improved acceleration, is easy and reliable to assemble, and has fewer breakdowns than the previous model. The report includes all the calculations that support the simulations and a validating statement about the bearing selection.

I. INTRODUCTION

A. Problem definition

The powertrain is that car department concerned with delivering the power from the engine to the wheels. Every force transmitting element from the engine to wheels is included in the department. It deals with the design, manufacturing, and assembly of hubs, uprights, differential mounts, axles, wheels, and tires. These components play an essential role in deciding the speed, acceleration, and mass of the car. Pieces are designed and assembled after coordinating with the suspension, chassis, brake, and steering department.

B. Project Objectives

As part of the project, several measures were taken to reduce the difficulties faced during assembly and avoid failures in the body. It took a series of measures, from changing the arrangement of bearings to manufacturing hubs of steel instead of aluminum.
The project was designed in compliance with the rules of Formula Bharat 2020 for participation in the same. The primary design considerations in this year's car are as follows.

1) Packaging in 10" wheels
2) Reduction in unsprung mass
3) Better acceleration
4) Ease in assembly and disassembly
5) Better locking of bearing to reduce backlash
6) Adjustable differential mounting

II. SELECTION

A pair of single-row tapered roller bearings are used in back-to-back arrangements compared to the previous car's face-to-face account. For both tapered roller and angular contact ball bearings, the distance L between the pressure centers is longer when the paths are arranged back-to-back compared with approaches put face to face. This means that back-to-back bearings can accommodate relatively large tilting moments even if the distance between the bearing centers 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 accommodate tilting moments. Axial loads in both directions can be adjusted, but only by one bearing in each.
Direction. After careful consideration, we decided that 320-32X bearing be used as the load ratings of bearing were well above the load on hubs and uprights, and the bearing size was optimum.
Bearing details
ID - 32mm, OD – 58mm, Width – 17mm
Dynamic load rating – 36.9 KN, Static load rating – 46.5 KN

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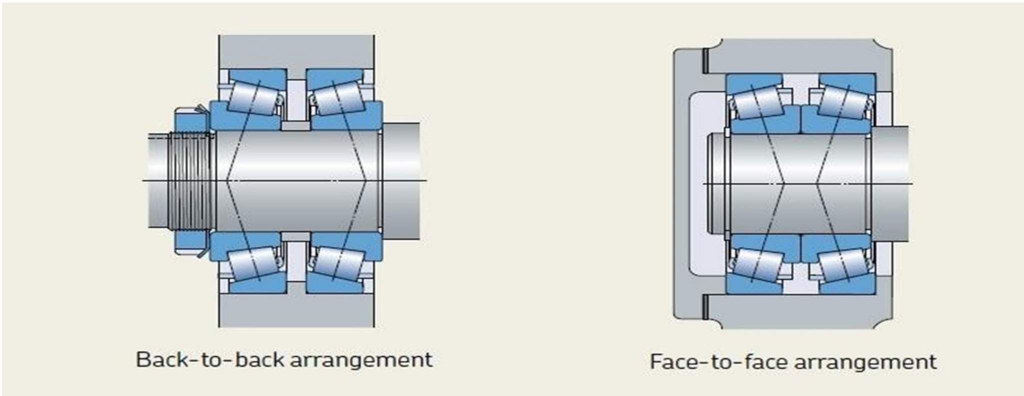

Fig. 1 1

III. WHEELS AND TIRES

Keizer 10" aluminum 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 the hub.

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

IV. DIFFERENTIAL AND DIFFERENTIAL MOUNTING

Dry open type differential is being used instead of the limited-slip differential as it provides grip and better handling and stability on a dry track and is cost-efficient at the same time.

Sprocket mounted on the differential have teeth machined to our needs. Below is the calculation for the required number of teeth.

400kg assumed mass Coefficient of friction = 1 F = un = 4000 N Tire radius = 0.2286m

T = 3500*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 = 1st 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 the fixed support at one end and flexible 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 mounts are very low and can easily withstand the differential load. The bearings are stopped by providing retaining rings on either side of the bearings.

TABLE I

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

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

The axle used in the prototype vehicle is the plunge joint and Mazeppa cv joint on the same.

The axle used in the prototype vehicle is the plunge joint and rzeppa cv joint on the same shaft. A plunge joint is used on the differential side, which provides axial adjustment of the axle. Rzeppa joint is used on the axle's wheel side, providing angular adjustments for better fitting while simultaneously acting as a cv joint.

A. *Hubs*

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

Method

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 caliper the PCD of lug bolts decides wheel mounting plate. It is the first step in creating the design of the hub in 3D modeling software.

The shaft connects the wheel mounting plate and brake rotor mounting plate. Width is decided by the size of the brake caliper and diameter by performing iterations in simulation software for required strength.

PCD of brake rotors decides brake rotor mounting plate.

They are bearing collars for mounting of bearing. Width is dependent on the size of the brake caliper and the diameter on the ID of the bearing used.

The bearing shaft is used to mount the bearings and to transfer the load to the suspension system. Diameter depends on ID of bearing and width on locking method used.

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

a) Force Due To Acceleration Or Decelerationfigures 1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18 and 19
b) Cornering
c) Wheel Travel Or Bump
d) Brake Torque Or Torque From The Axles

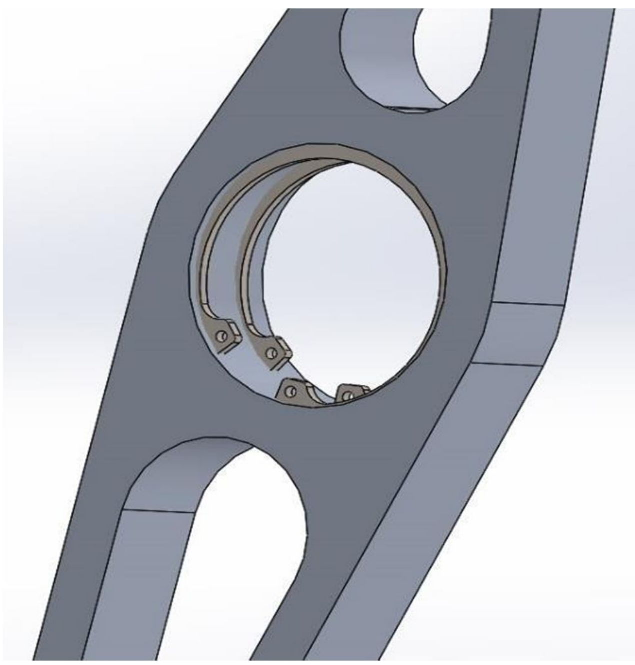

Fig. 2 2

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

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B. Rear Hub

The function of the rear hub is to support the wheels and connect them to the suspension system, as well as transmit 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 the ring.

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

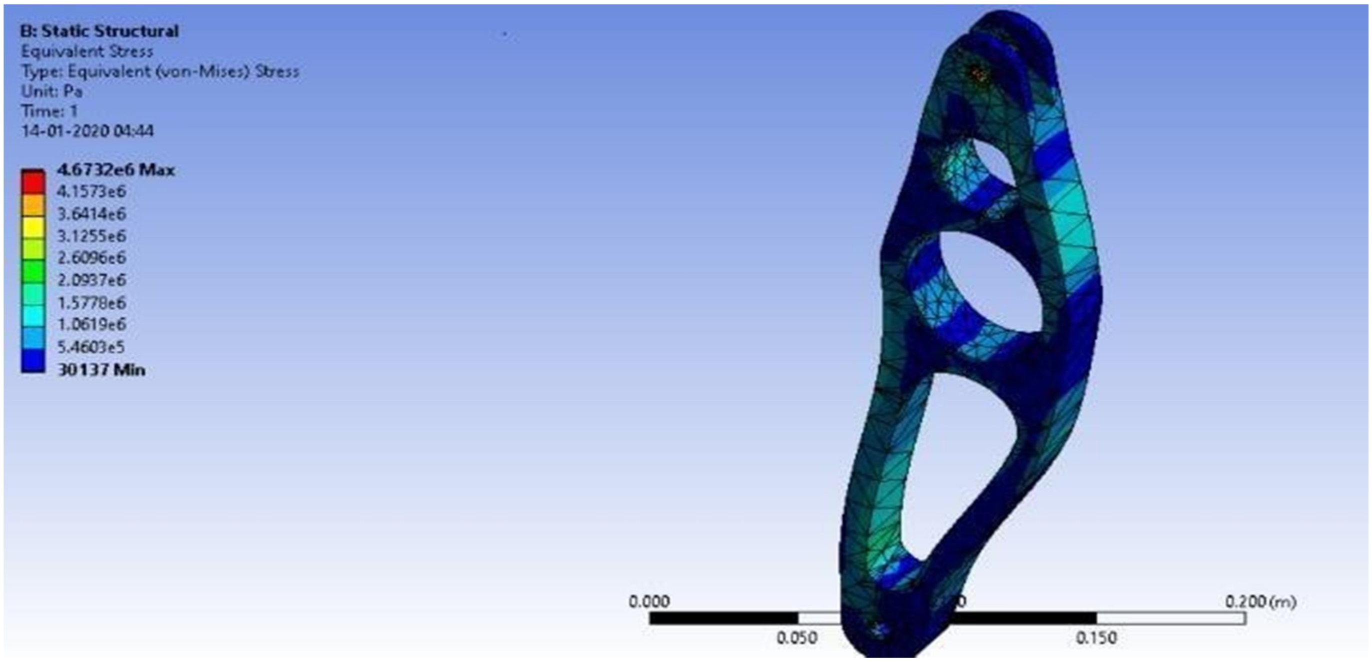

Fig. 3 3

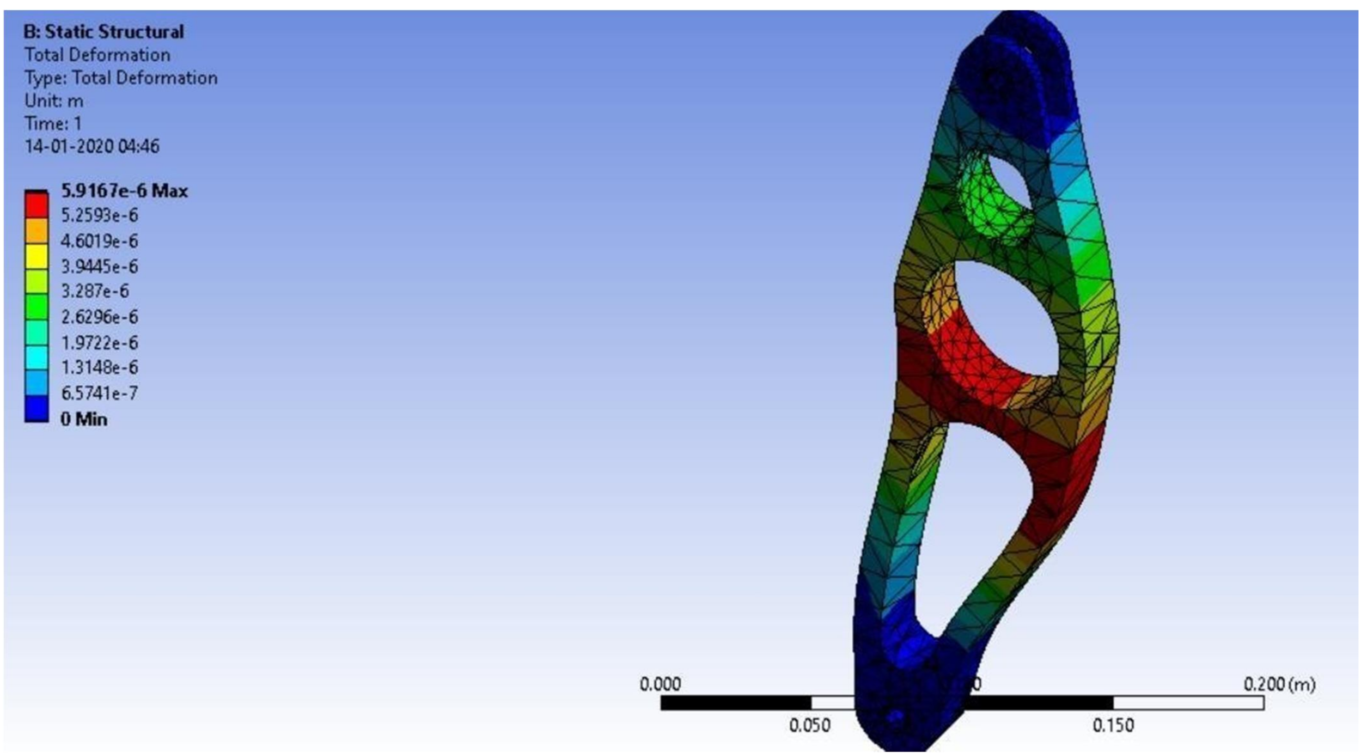

Fig. 4 4

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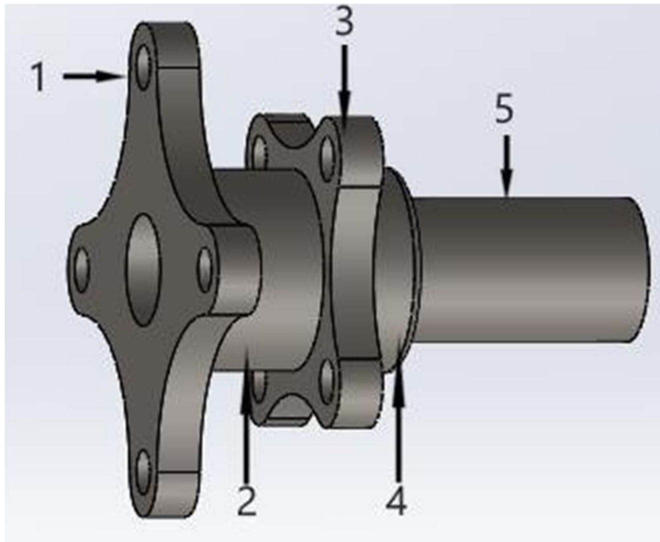

Fig. 5 5

Table II I

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

VI. ANALYSIS

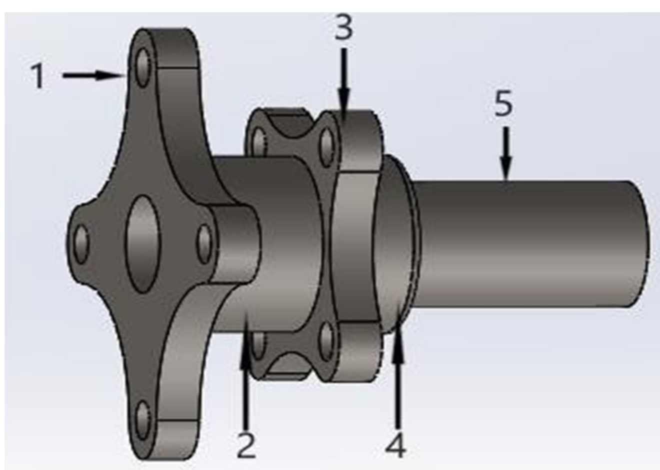

Fig. 6 6 TABLE III

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 v2-u2 = 2*a*s a = -19m/s2 assuming 60% weight transfer to the front, the force due to the braking on front hub = 3990N force due to cornering in lateral direction

TABLE III

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

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 = d/t = 57.33/7 = 8.19 m/sec
A = v*v/r = 8.19*8.19/9.125 = 7.35 m/sec2 f =2600N

Considering the weight of the car, bump force=2000N. The FEM analysis was done on ANSYS work bench and the results were carried- out as follows Quadratic method is used to generate the mesh with an element size of 5mm.

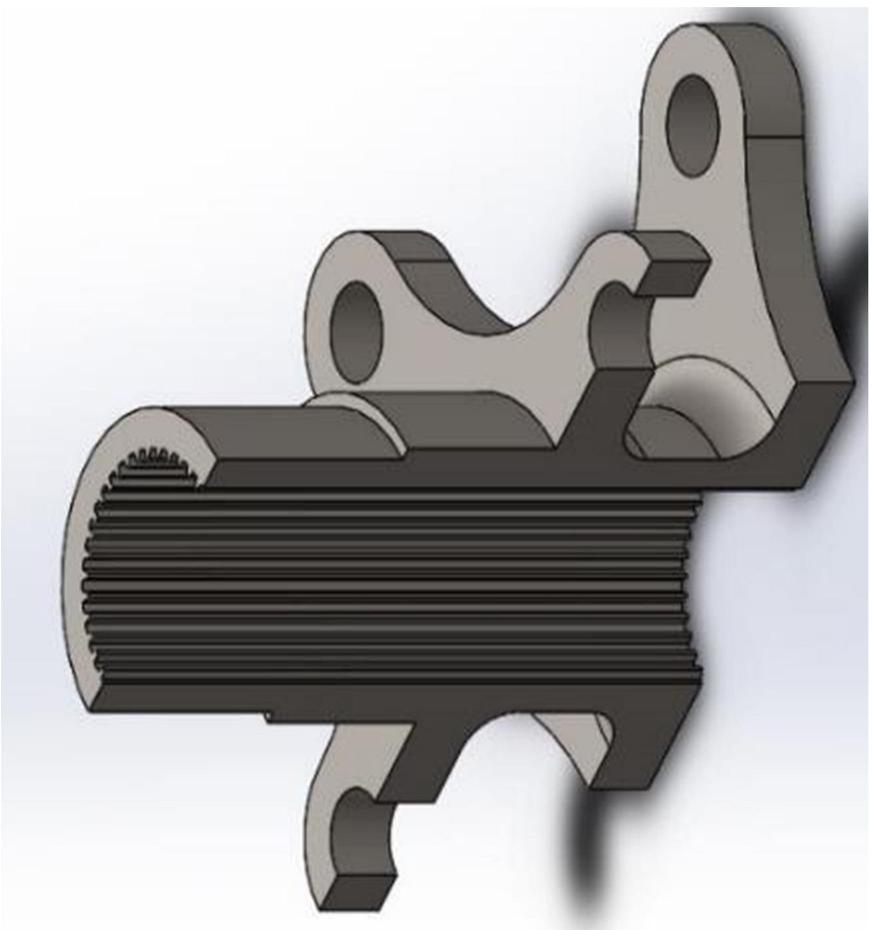

Fig. 7 7

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

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Moment generated at wheel mounting points during braking is calculated as follows

Table IV V

Constraints	Wheelmountingpoints
Force	Shaft
Equivalent stress	2.736e+008Pa
Maximum deformation	1.309e−004m
FOS(ultimate)	2.72

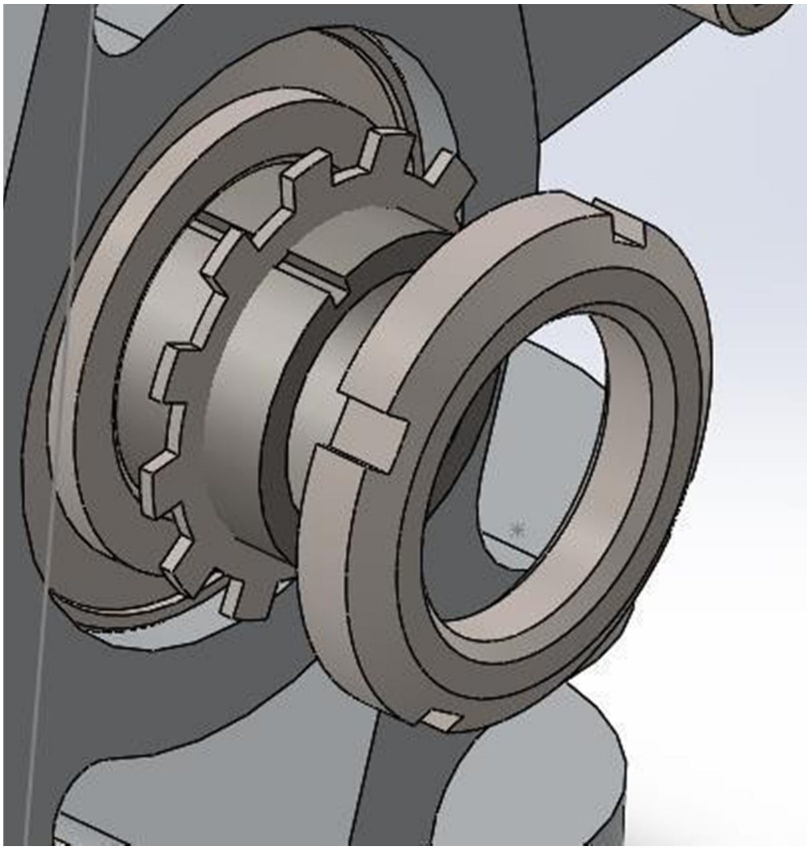

Fig. 8 8

Force due to braking is longitudinal direction= 3990 Ntirera dius=0.2286m Momentgenerated≈900Nm Quadratic method isused to generate the mesh with an element size of 5mm.

TABLE V

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

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Maximum stress is generated at wheel mounting points, and maximum deformation occurs at the outer edge of wheel mounting points. The moment generated at brake rotor mounting points while braking is calculated as follows. After consulting the brake department, clamping found the force to be = 1302 N Moment generated at the front wheels = 115.5Nm

The quadratic method is used to generate the mesh with an element size of 5mm.

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

Fig. 9 9

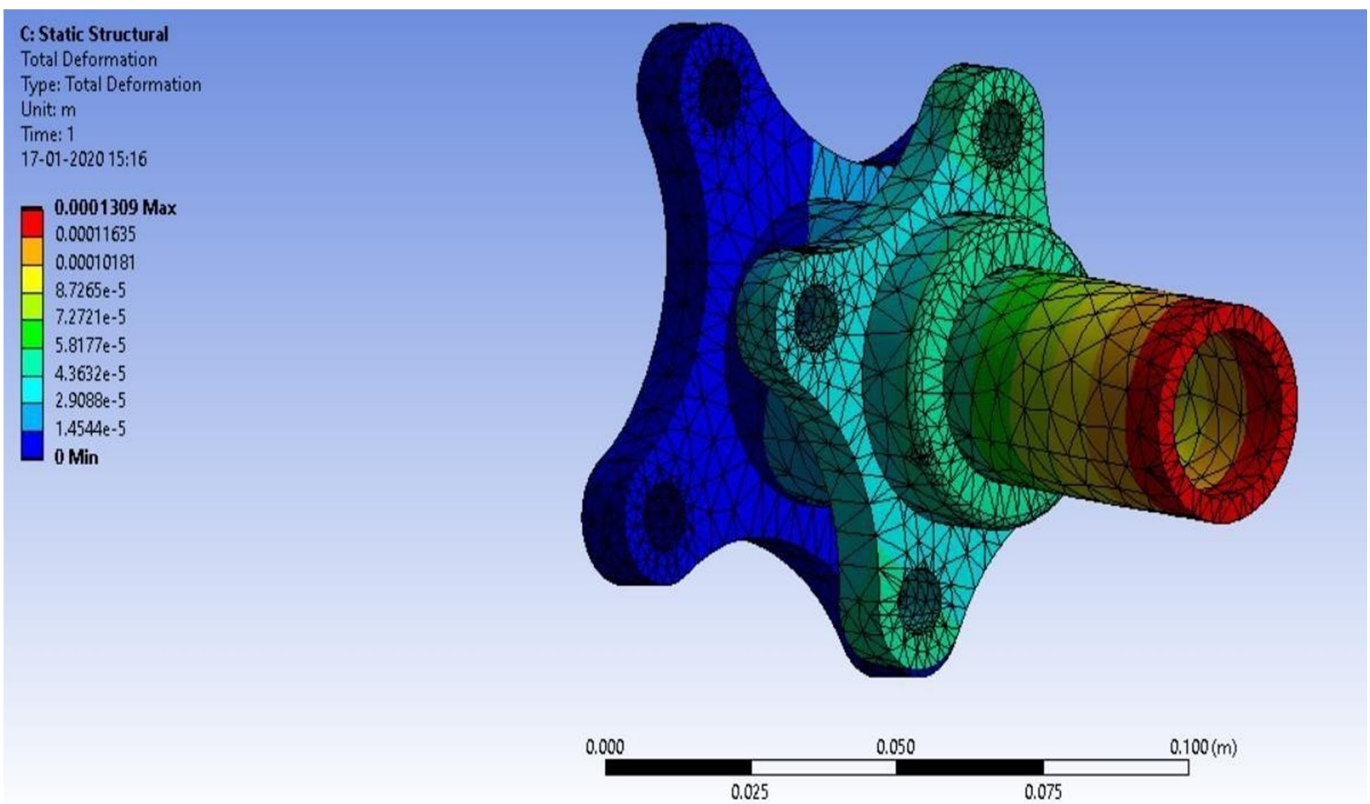

Fig. 10 10

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

VII. UPRIGHT

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

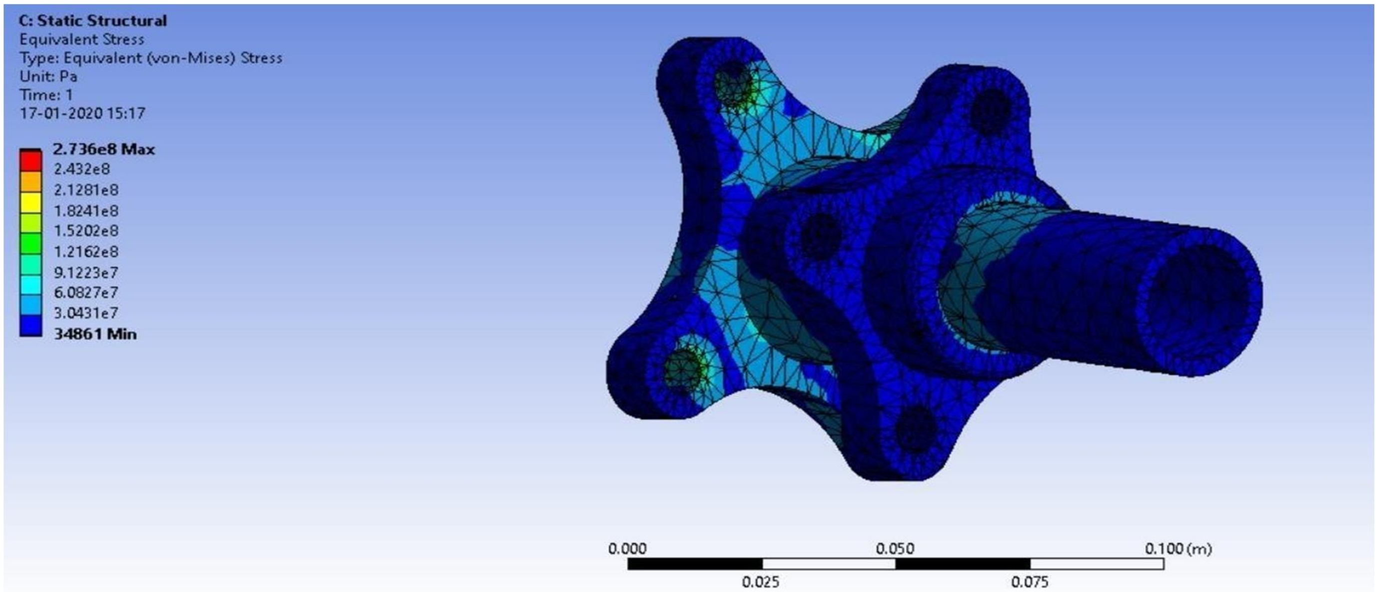

Fig. 11 11

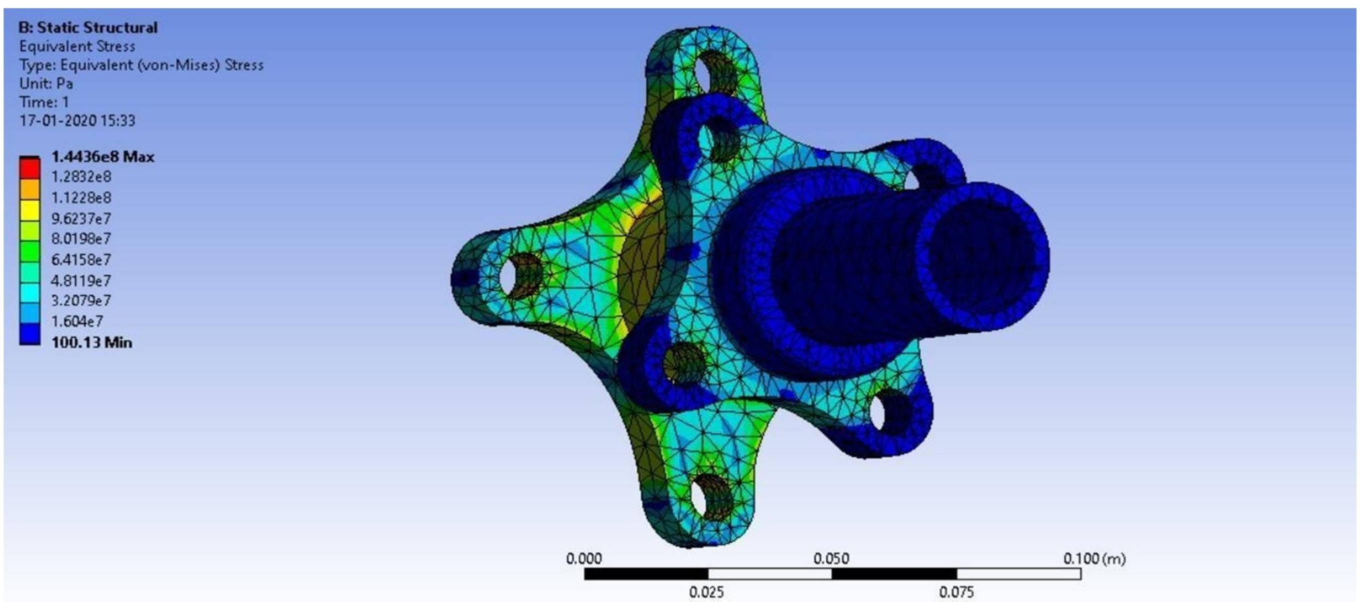

Fig. 12 12

TABLE VI I

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

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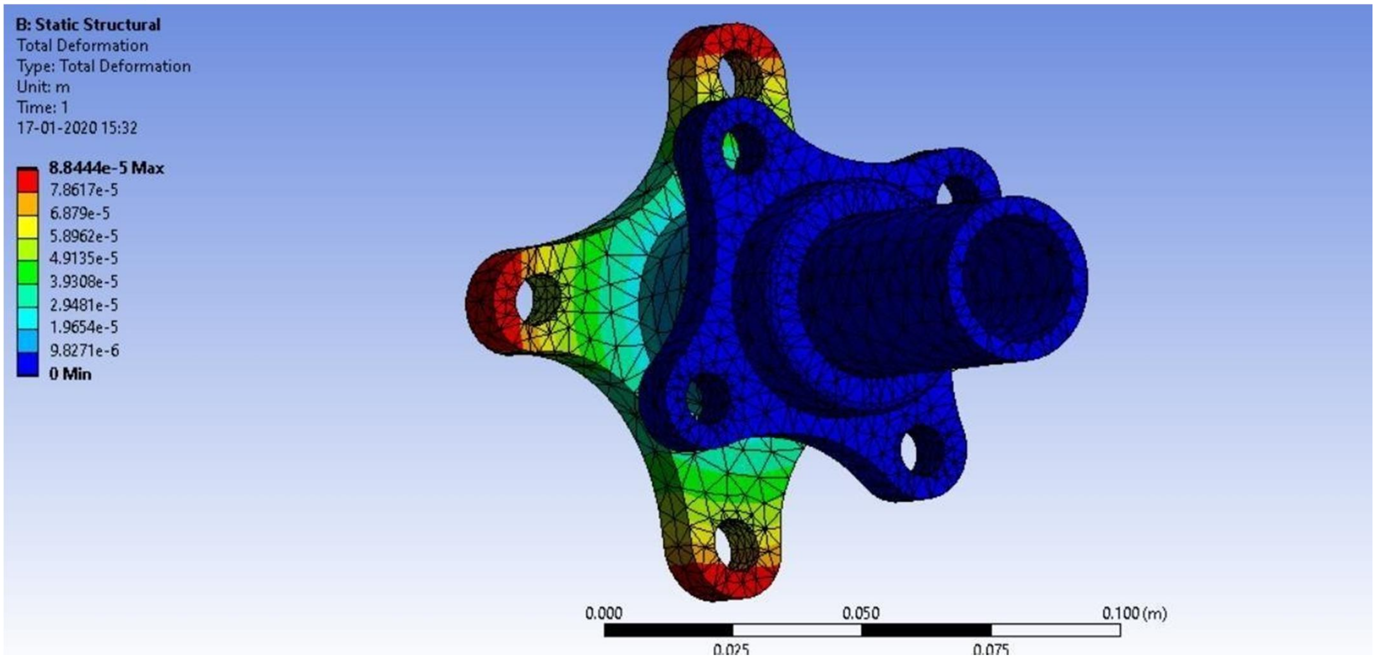

Fig. 13 13

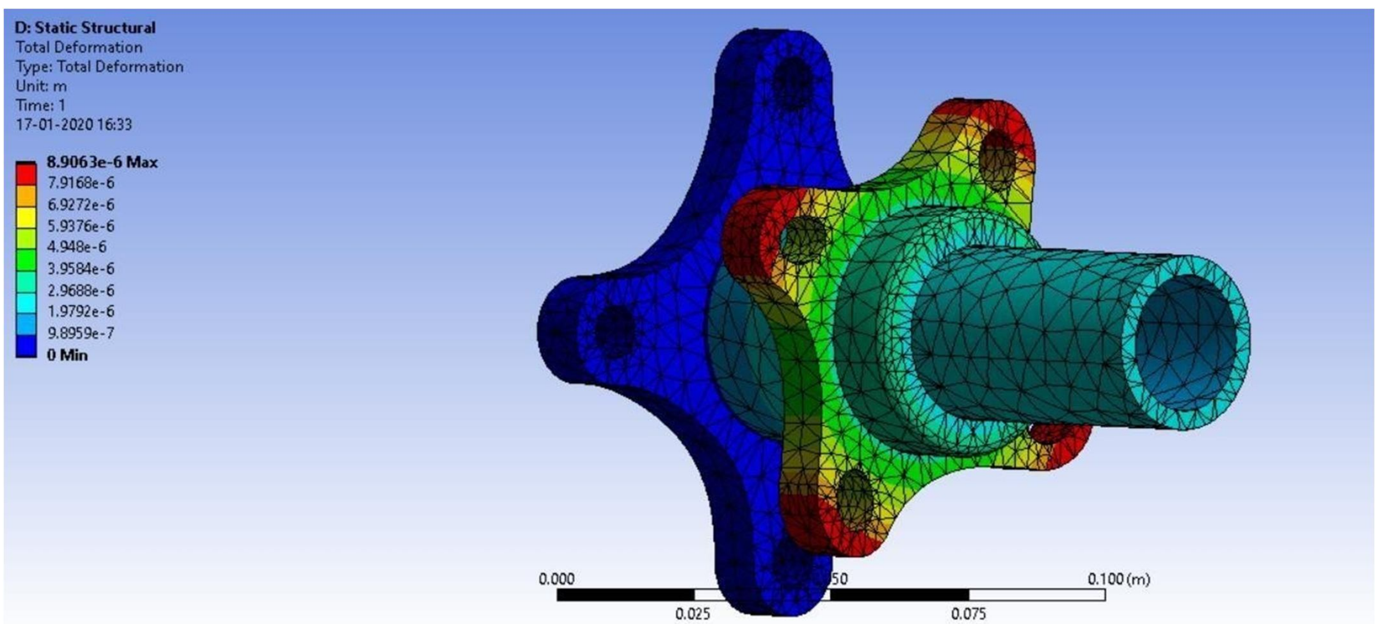

Fig. 14 14

VIII. METHOD OF DESIGNING

The design of upright is dependent on the following factors.

1) Kpi
2) Caster
3) Bearings
4) Distance between LBJ and ubj
5) Locking method of bearing
6) Brake caliper
7) Tie rod mounting

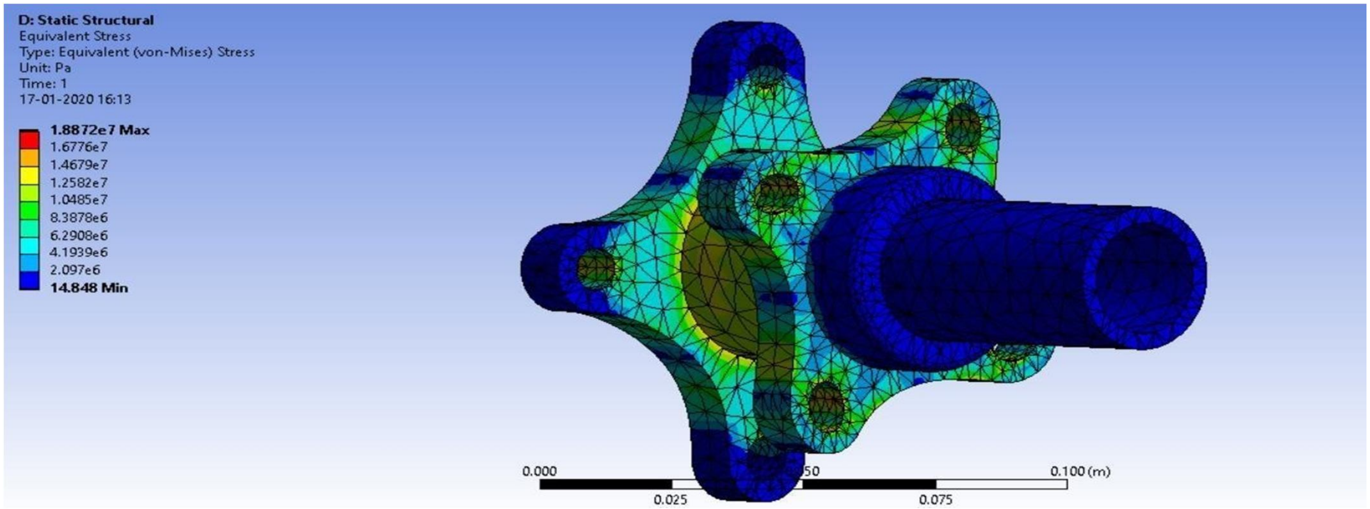

Fig. 15 15

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	

Fig. 16 16

The design process starts by deciding the kingpin inclination, caster, and distance between the upper and lower balls. The next step is designing the bearing housing after deciding the tolerances, bearing arrangement, and locking method. Finally, a step is provided in the middle of the bearing housing to locate the bearing axially, and the stage is discontinuous to facilitate easy bearing removal.
After the housing design, clamps are designed, and holes are provided for the wishbones from KPI and caster. Then comes the mounting for the brake caliper, designed with the help of a CAD model of the caliper. The final step is to prepare the mount for the tie-rods, which are decided from the steering geometry.

The upright has to be strong enough to withstand the following forces
a) Longitudinal force while braking
b) Latitudinal force while cornering
c) Bump force

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	

Fig. 17 17

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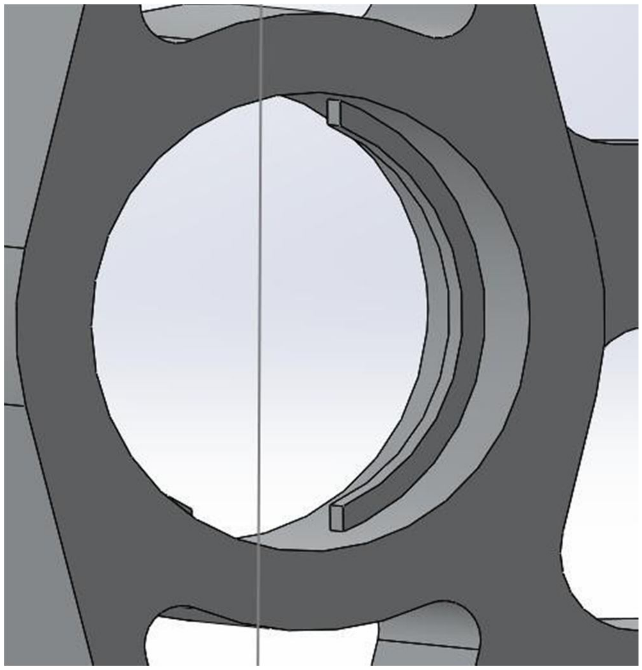

Fig. 18 18

d) The moment generated at brake caliper mountings while braking
e) Force on tie rod mounting while cornering

A. Front upright

The front upright is designed to withstand more cornering forces due to steering. Hence decided to use two single-row tapered roller bearing with back-to-back arrangements. Also, the tie rod mounting was designed following Ackerman Geometry.
Uprights are manufactured from 7075-T6 billets as its tensile strength and fatigue strength is greater than that of 7050

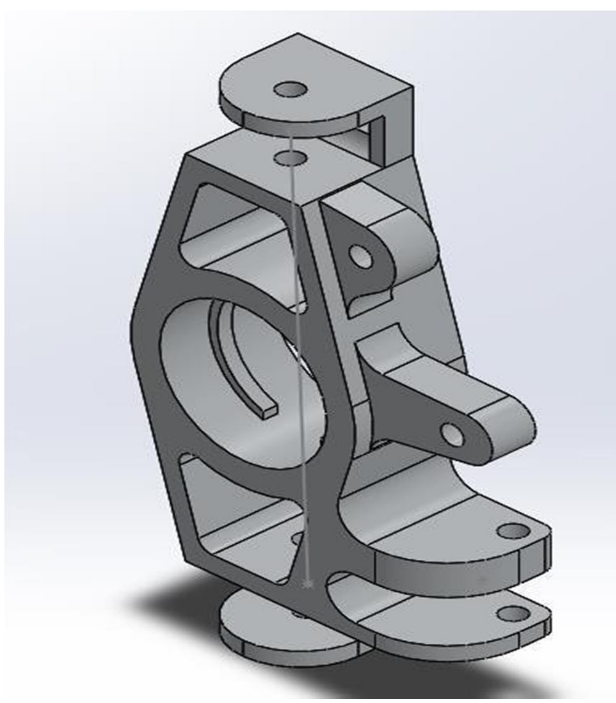

Fig. 19 19

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IX. MATERIAL SELECTION

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X. ANALYSIS

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

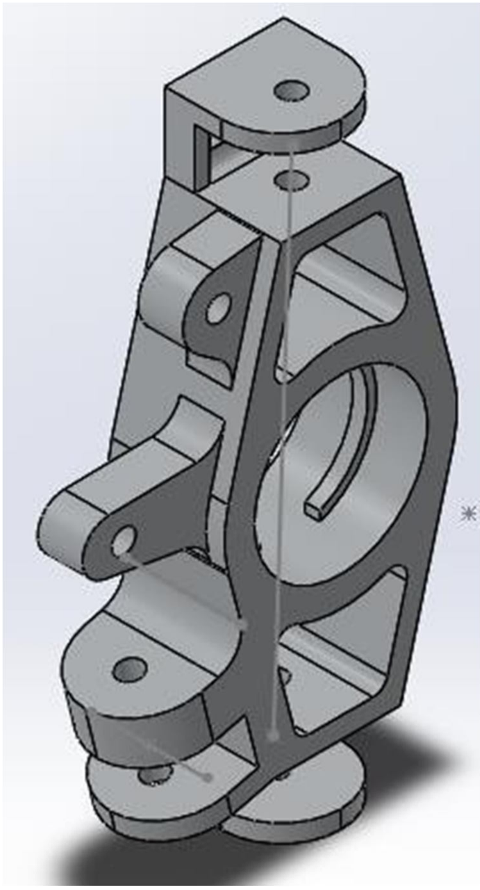

Fig. 20 20

TABLE VII II

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

We did the FEM analysis on the ANSYS workbench, and the results were carried out as follows. First, the quadratic method is used to generate the mesh with an element size of 5mm.

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

Force due to cornering on tie rod mounting = 2600N

A quadratic method is used to generate the mesh with an element size of 5mm CONCLUSION

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The following consequences had been acquired from the report

1) Better acceleration through calculating a wide variety of teeth on the sprocket
2) Efficient locking and location of bearing to lessen the backlash

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

Fig. 21 21

TABLE VIII III

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

TABLE IX X

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

TABLE X

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

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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	

Fig. 22 22

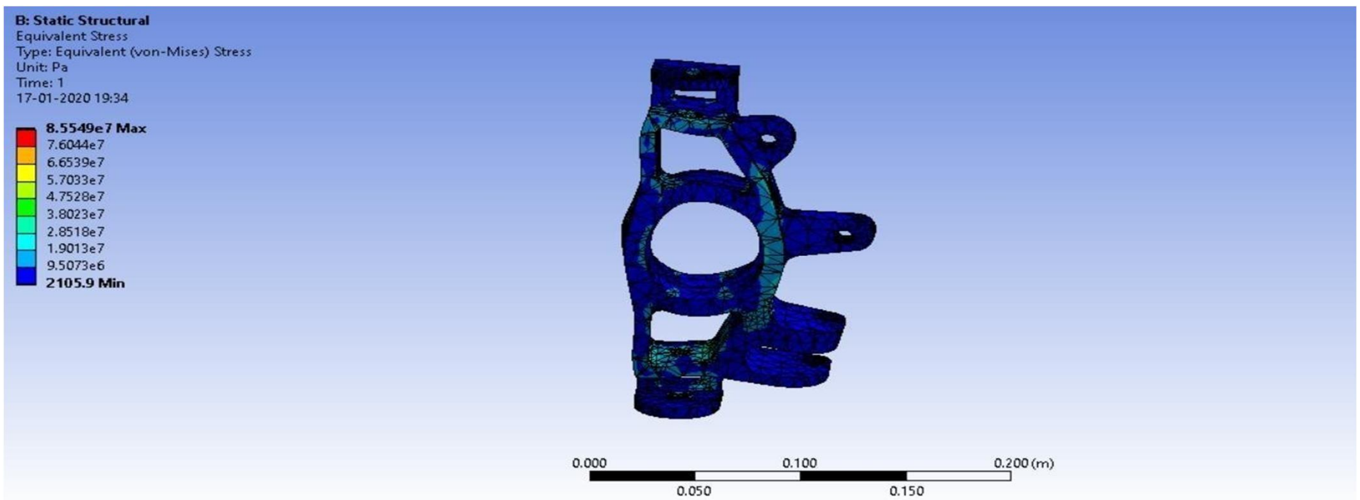


Fig. 23 23

3) Development of correct FEA version with the assist of calculations
4) To layout adjustable differential mounting

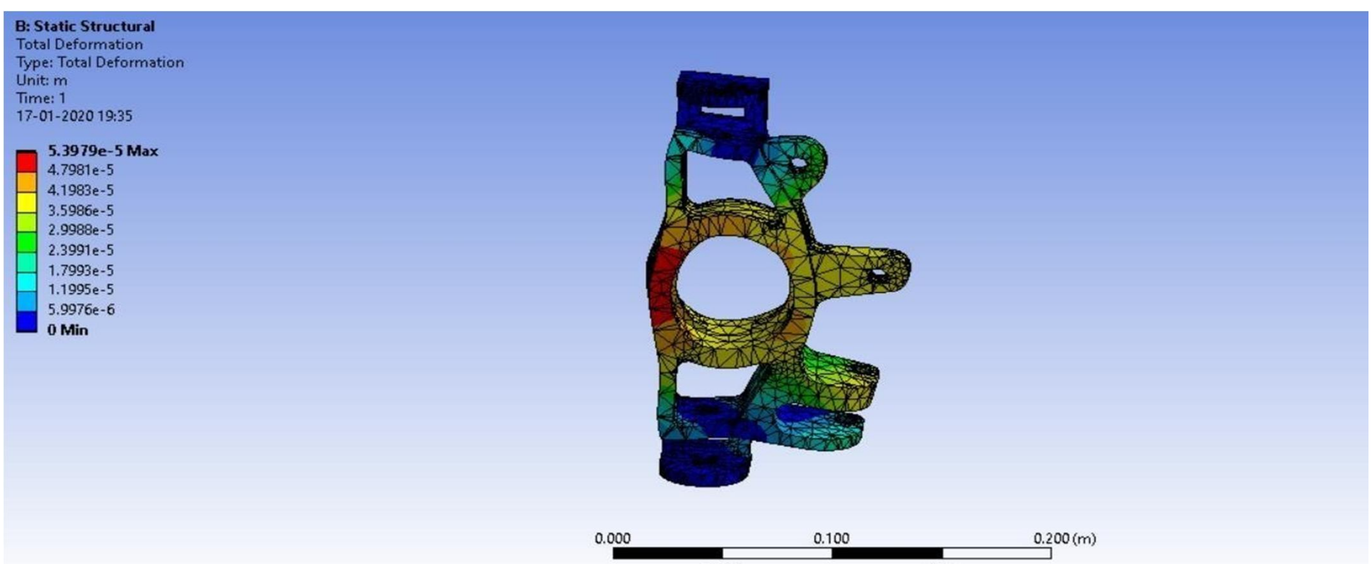


Fig. 24 24

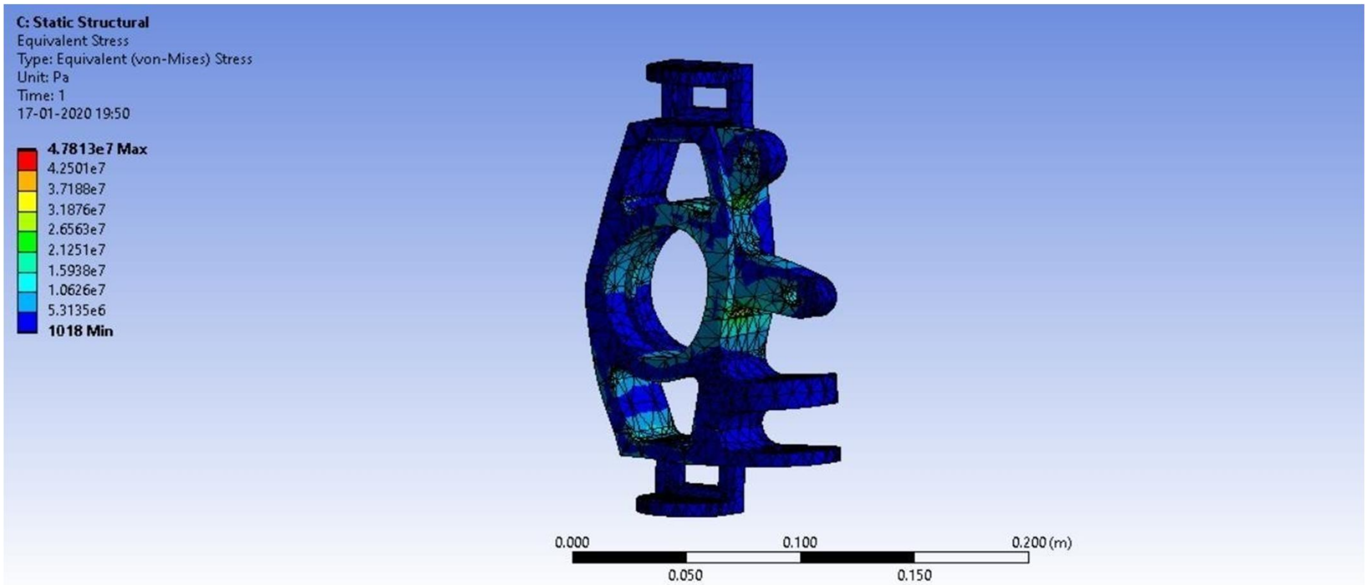


Fig. 25 25

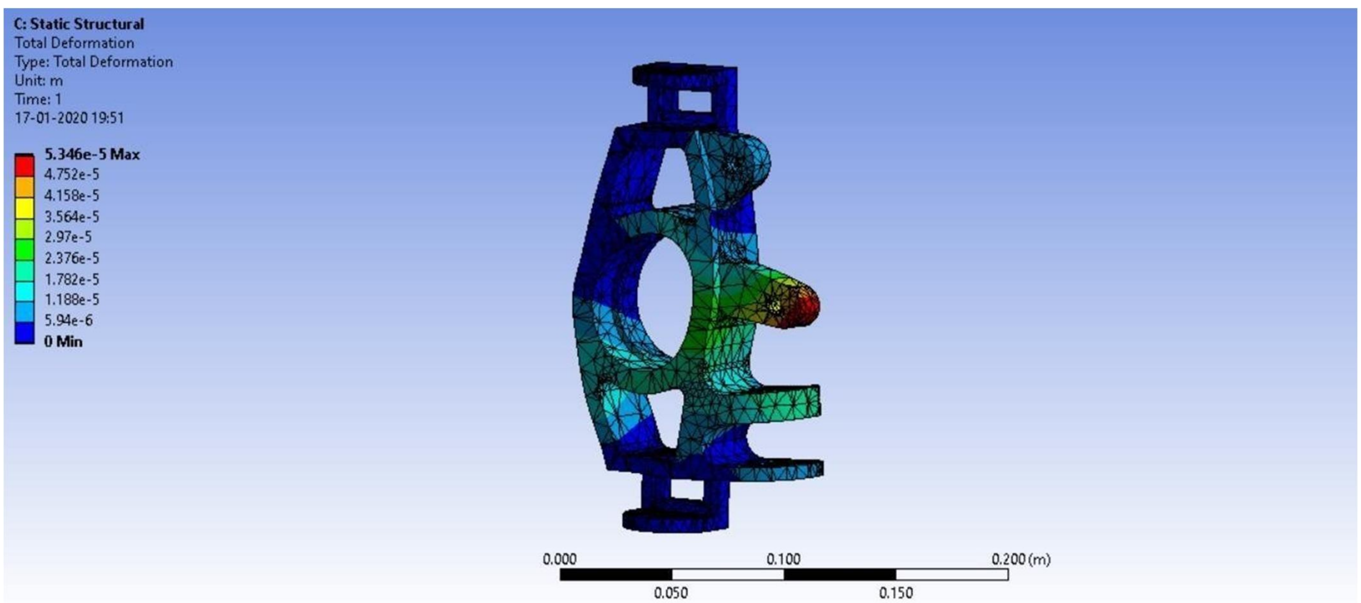


Fig. 26 26

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. 27 27

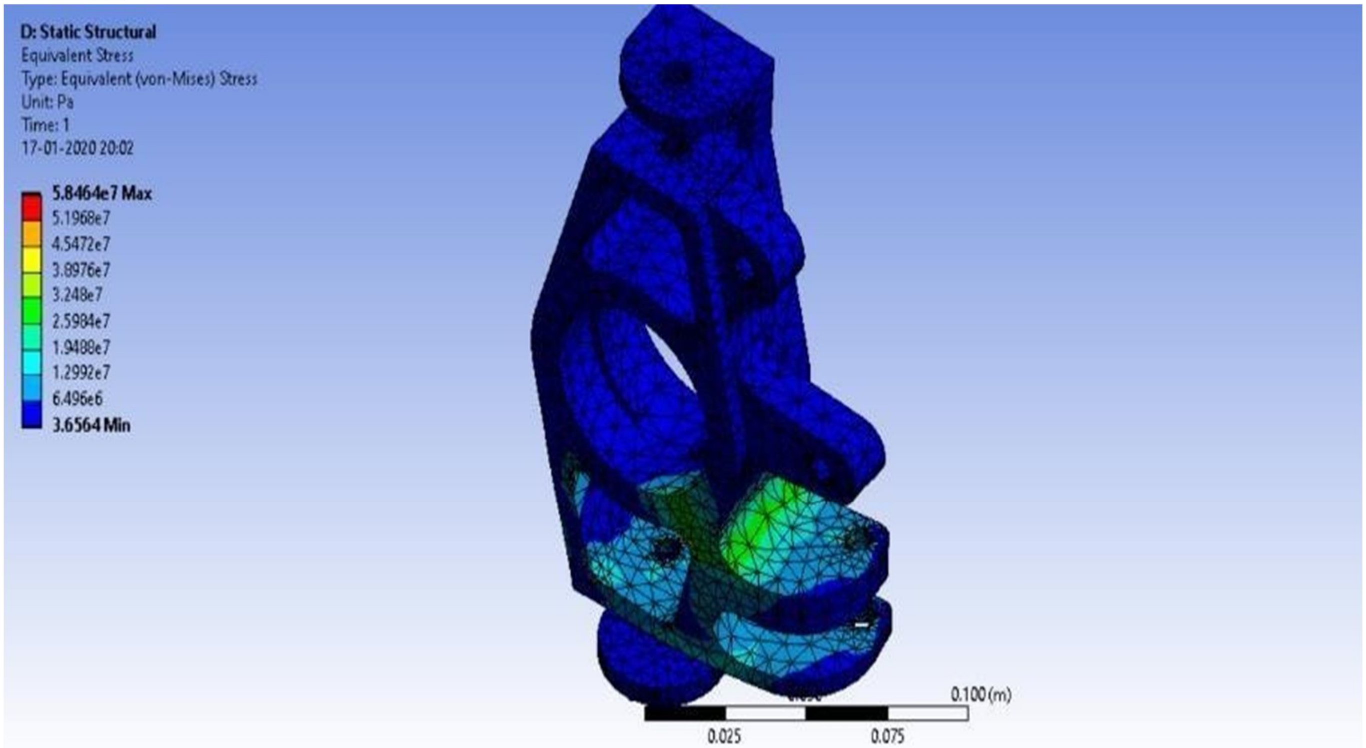


Fig. 28 28

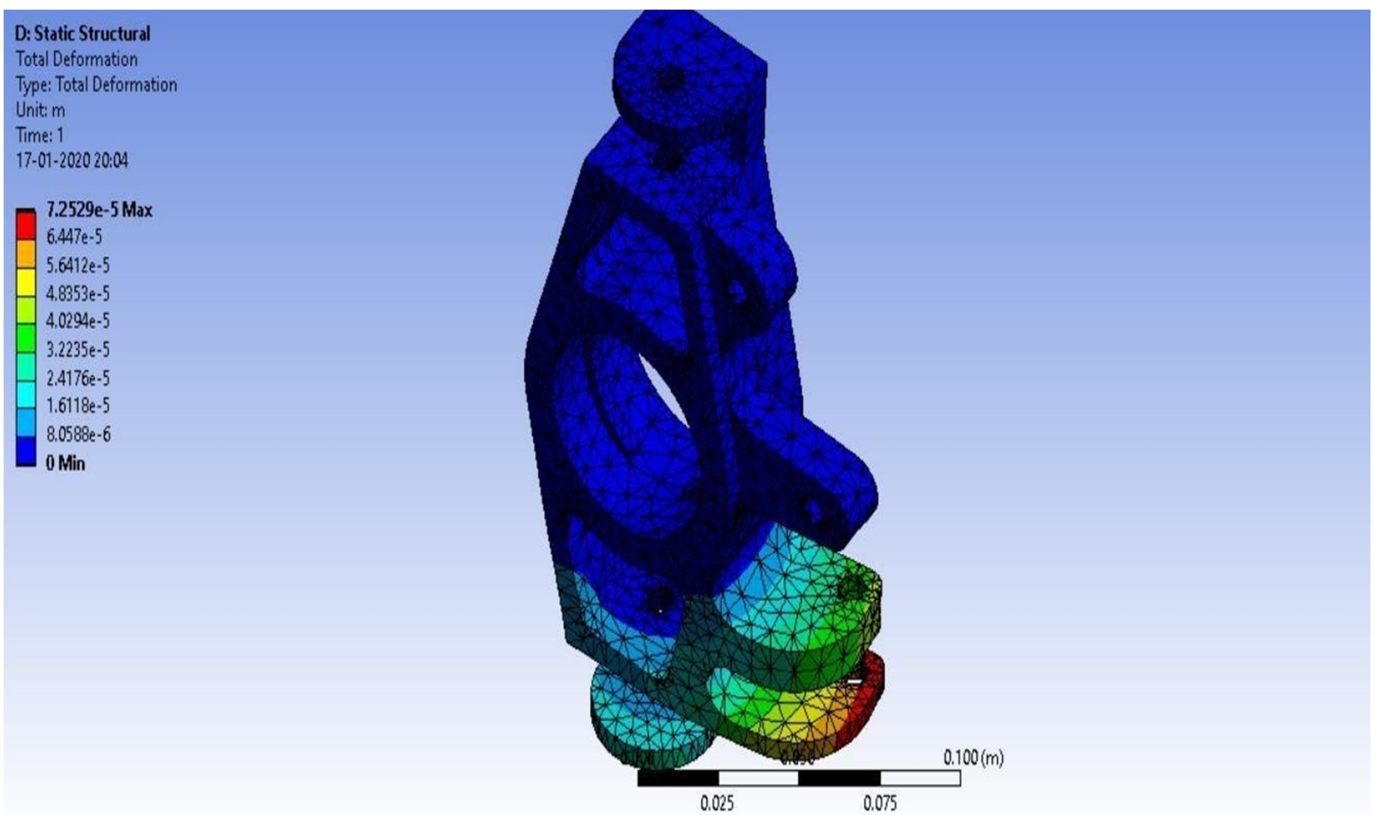


Fig. 29 29

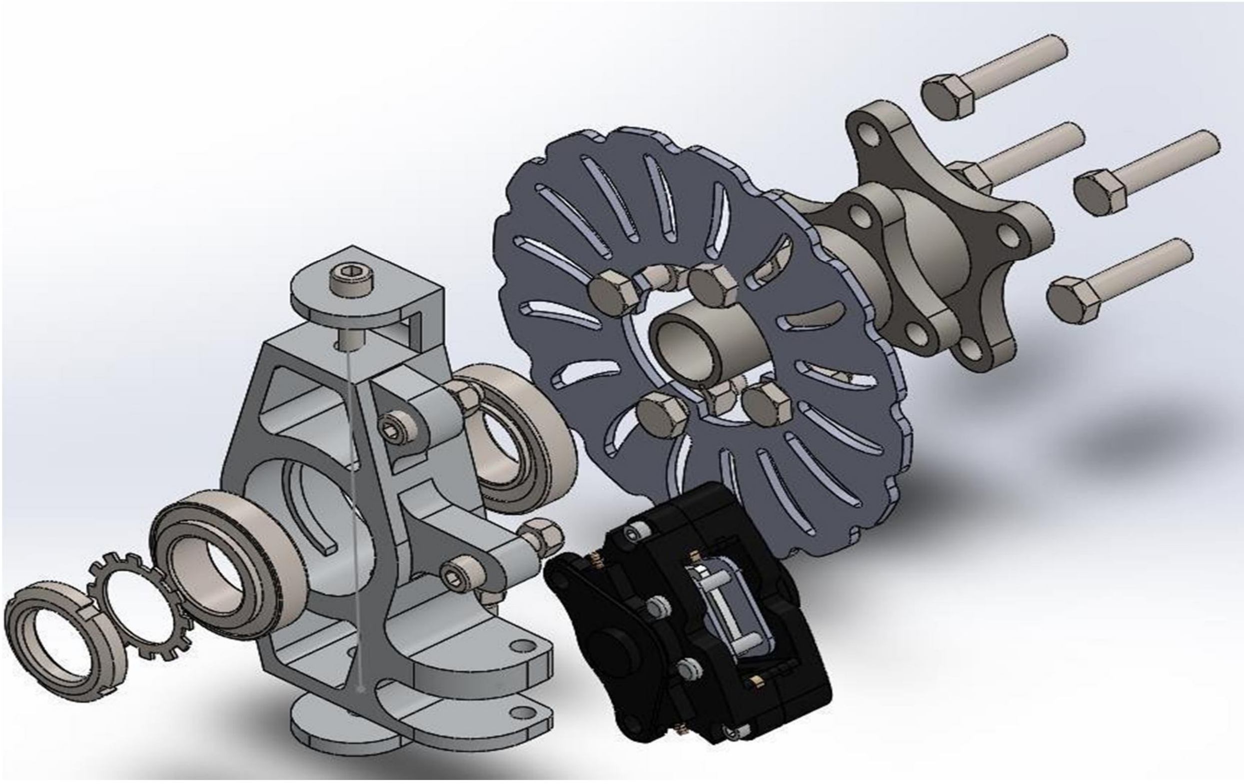


Fig. 30 30

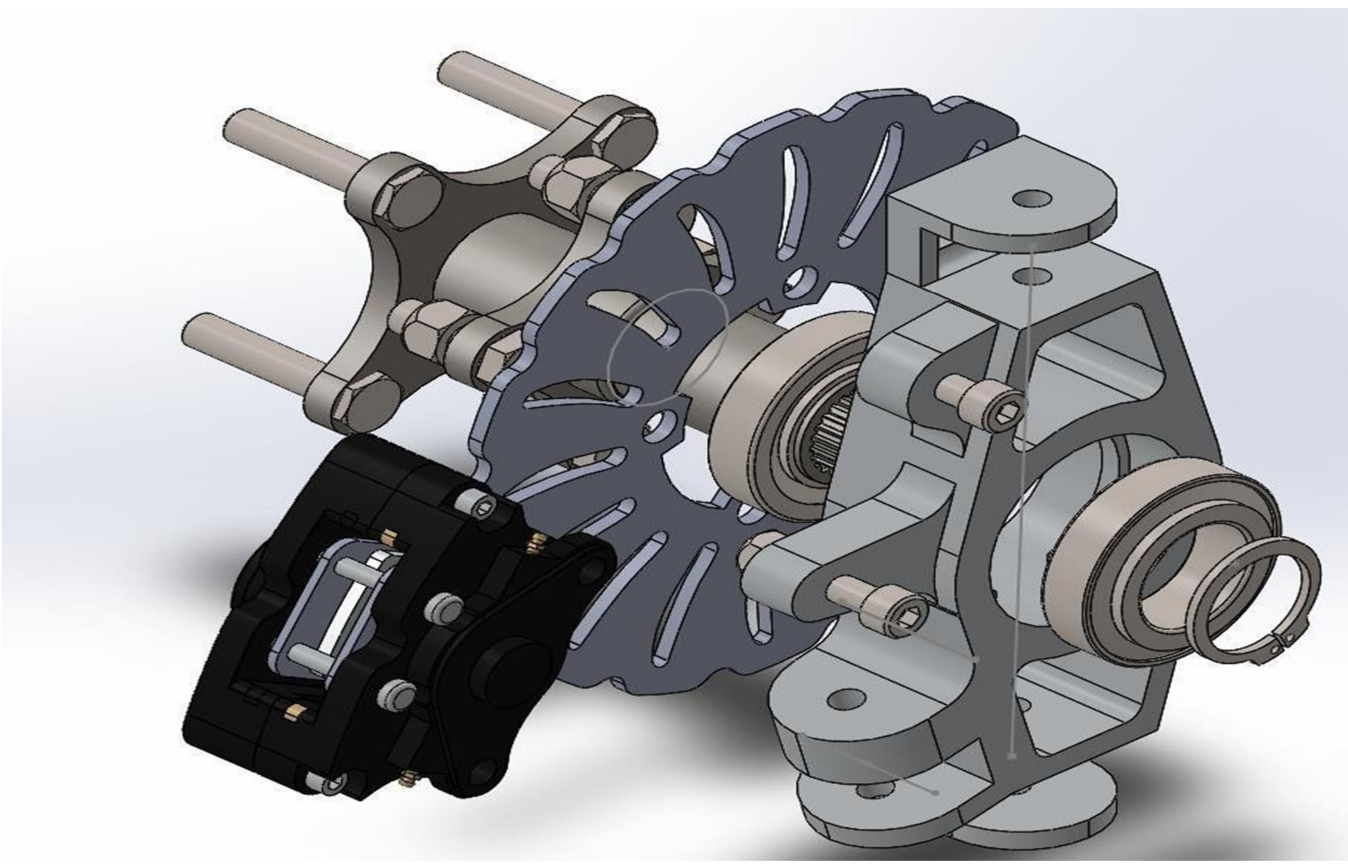
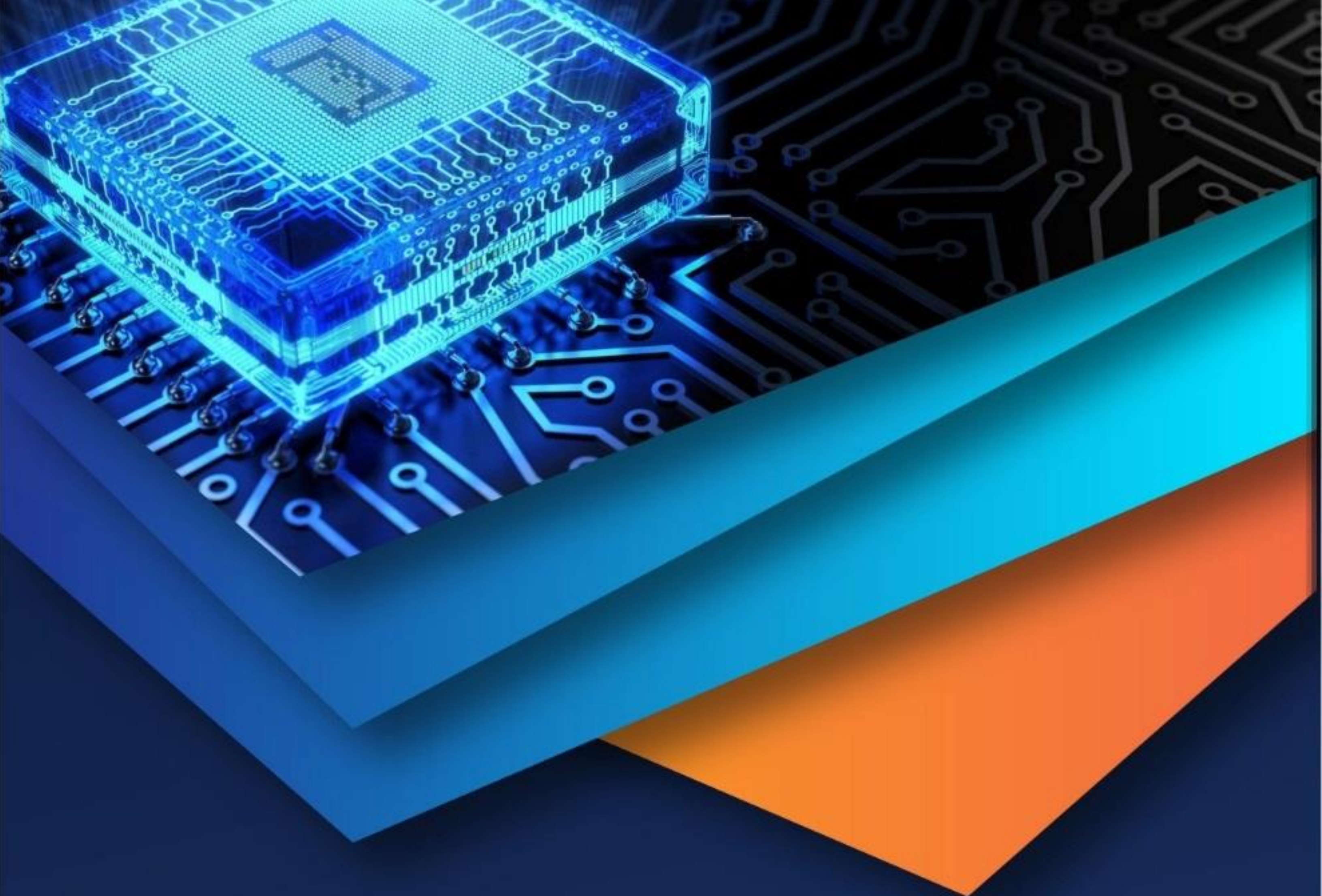


Fig. 31 31

XI. CONCLUSION

A. Improve acceleration by calculating the number of teeth on the sprocket
B. Efficient locking and placement of bearings to minimize backlash
C. Calculations are used to develop an accurate FEA model
D. To design a mounting system for adjustable differentials



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