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Research Article

Design and ANSYS analysis of Components of Wheel Assembly of SAE Car

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Abstract

In automobile industry it is essential to produce the light weight assembly in order to increase the vehicle performance. Also, lots of forces during braking, acceleration and bump conditions are also applied directly during dynamic condition. So, this paper deals with calculation of various loads and their simulation. The FEA result indicates that the upright assembly is able to perform safely in real track condition as per performance requirement.

Keywords: *Upright, hub, load transfer, ANSYS, dynamics terms, bearing selection.*

1. Introduction

There are always two types of masses in an automobile – sprung and unsprung mass. All the mass of the vehicle that is damped by the spring is called as the sprung mass. As the Wheel Assembly mass is not damped by the spring, it comes into the unsprung mass category. We know that the unsprung mass must be lower than the sprung mass and also should be as least as possible to provide proper drive stability and load balancing of the vehicle. Thus, it becomes important to reduce the mass of the wheel assembly and the rims and tires. But while doing this care must be taken that the mass of the wheels, tires and the wheel assembly must be enough to prevent the lateral toppling of the vehicle at the time of cornering or impact.

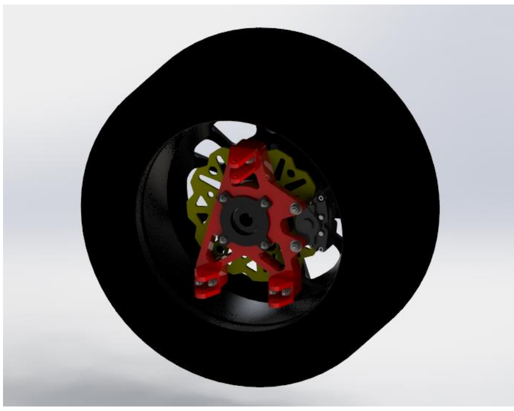

Fig 1 CAD file of assembly

There are a lot of forces acting on the wheels in the static and especially in the dynamic condition. As the Wheel assembly is directly connected to the wheels, all these forces also have an impact on the designing of the Wheel Assembly. A lot of forces act on the wheel assembly during accelerating, braking, cornering and tilting.

A good Wheel Assembly is one which can sustain such forces over a longer period of time. Thus, it is required to design the wheel assembly considering all these factors. A failure of any component of the Wheel Assembly means a breakdown of the automobile and in some cases might also be hazardous for the driver. Thus, utmost care must be taken while designing the Wheel Assembly. The objective of Optimization is always to find the best possible and suitable dimension. This is because optimization does not always mean reducing dimensions it also means finding out the dimensions which will just enough to sustain the forces.

The upright connects the control arms to the hub which connects the upright to the wheels, allowing the vehicle to move. The uprights also connect to the steering arm, allowing the driver to steer the vehicle, and the caliper, allowing the driver to stop the vehicle. The hub is directly connected to the wheel and is connected to the upright. The upright is to remain stationary relative to the chassis while the hub is to rotate with the wheel. This is done by placing a bearing between the hub and upright. Typically, a spindle is pressed into the upright and does not rotate and a bearing is pressed into the hub, and the spindle is pressed into the bearings allowing the hub to rotate about the spindle. Unsprung mass is the mass of the wheel, hub, rotor, caliper, uprights, spindle, and brake

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pads. Essentially unsprung mass is the mass that is not supported by the shocks (for example the chassis and everything supported by the chassis is sprung mass). It is important to reduce unsprung mass in order to increase acceleration. The greater the unsprung mass, the slower the accelerations.

2. Problem definition

In Formula SAE, the competition centers not only on the performance of the vehicle, but also, on the design itself. The concept of FSAE is that each collegiate chapter assumes the role of a fictitious company who is set on building a prototype formula style race car. This prototype is set to go on full production and is marketed towards the average weekend racer. The design is weighed as much as the performance aspect of the competition in terms of judging and scoring previously, a solid design methodology that uses the time, resources, talent and research available to produce a competitive vehicle has not been established. This is attributed to the infancy of the program; many hours have been spent acquiring resources and research for the betterment of the organization.

3. Project objectives

- In order to place well, as previously mentioned, the vehicle must be designed with good engineering practice and must perform well. Since the event is essentially an autocross event, which favors cornering over top speed, this means having a chassis that is as light a possible while still maintaining required torsional rigidity, and a suspension system that can maximize the performance of the tires in contact with the road.
- Develop the assembly which is sustainable in all situation of impact accidents having good aesthetics and low weight.
- Develop the assembly with low maintenance and also with less complication and cost effective.
- Develop the effective braking system in which all 4 wheels are lock at a time. And car should stop in shortest possible time and distance.
- The objective of the system is to convert the kinetic energy of the vehicle into thermal energy, allowing the vehicle to decelerate optimally and safely.
- Aspects of ergonomics, safety, ease of manufacture, and reliability are incorporated into the design specifications.

Fig 2 Actual wheel assembly

4. Methodology

- Wheel assembly of formula one car is different than the wheel assembly of passenger vehicle. Passenger vehicle car is mainly focus on the performance of vehicle but F-1 car is focus on the efficient braking of 4 wheel at a time and also good acceleration and speed. So, the components which are involve in F-1 car are designed accordingly.
- For doing this activity first we understand the basic parameters which are responsible for the designing such as caster angle, camber angle, king -pin, toe angle etc.
- Then by doing research on above parameters and also some new parameters we come to conclusion of their actual value which we have to implement while designing because depends on these parameters our car reliability is dependent
- After this we have done the design using SOLIDWORK software of modelling and afterword we done the simulation on major components using simulation software called as ANSYS.
- Before doing simulation, we have done some calculation which is required for simulation. Calculation include CG height, various load transfer in dynamic condition, etc.
- Also, the market survey is done which include cost, availability and strength then come to the conclusion of perfect material for each component which we are planning for manufacturing.

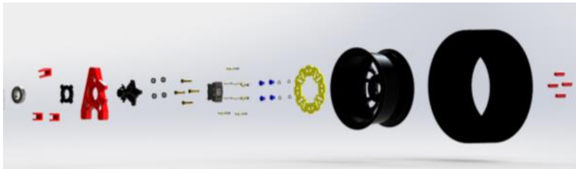

Fig 3 Exploded view of assembly

- Once the design is completed then we looked for manufacturing process with material and other components such as nut bolts, bearing etc.
- Upright, hub and brackets are manufactured with the help of CNC turning, VMC and WATER JET machining to get the precision and accuracy.

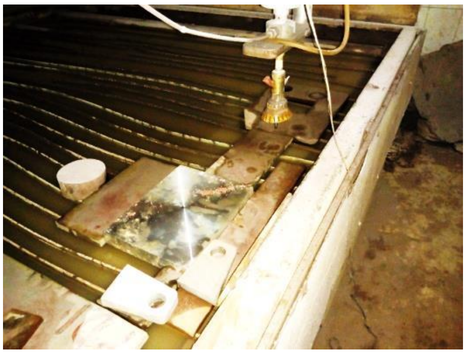

Fig 4 Water jet machining

- Bearing lock and floater button are manufactured by using conventional lathe and milling machine.
- Then assembly of components are done with the help of all these manufactured and OEM parts.
- Before assembly all parts which are manufactured are powder coated to avoid the corrosion and other environmental effect.
- Finally, after completion of car running test is done for testing purpose.

Fig 5 Components of assembly

5. Dynamics and Common Terms

The uprights are critical to locating different suspension geometries including the outer most control arm points as well as the tie rod connections. The location of these points relative to the mounting locations on the chassis determines caster, camber, toe, and Ackerman.

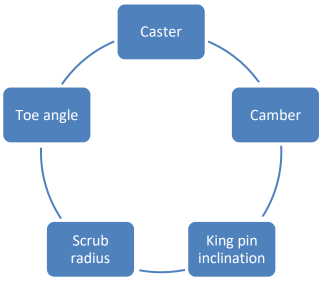

Fig 6 Common parameters

5.1 Caster

Definition: Caster is the angle to which the steering pivot axis is tilted forward or rearward from vertical, as viewed from the side.

Caster angle can be seen from left or right side of vehicle.

Range of caster angle – 3 to 6 degrees.

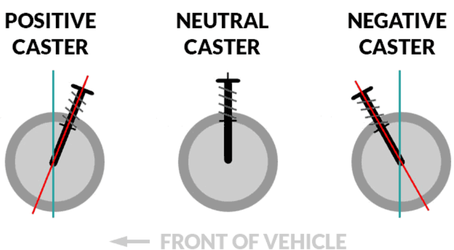

Fig 7 Types of caster angle

Types

- Positive caster

If the pivot axis is tilted backward (that is, the top pivot is positioned farther rearward than the bottom pivot), then the caster is positive.

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Advantage of positive caster

- Caster causes a restorative force felt by the driver which returns the steering wheel to center
- The forks point forward at the bottom and slope backward at the top. This rearward slope causes the front tire to remain stable when riding straight ahead and tilt towards the inside of the corner when turned.
- Caster angle settings allow the vehicle manufacturer to balance steering effort, high speed stability and front-end cornering effectiveness.
- Increasing the amount of positive caster will increase steering effort and straight-line tracking, as well as improve high speed stability and cornering effectiveness.
- Positive caster also increases tire lean when cornering (almost like having more negative camber) as the steering angle is increased.
- Negative caster

If pivot axis is tilted forward, then the caster is negative.

Example - castering front wheels of a shopping cart.

The steering axis of a shopping cart wheel is set forward of where the wheel contacts the ground. As the cart is pushed forward, the steering axis pulls the wheel along, and since the wheel drags along the ground, it falls directly in line behind the steering axis. The force that causes the wheel to follow the steering axis is proportional to the distance between the steering axis and the wheel to ground contact patch the greater the distance, the greater the force.

5.2 Camber

Definition: Camber is the relative angle of the wheel with respect to vertical.

Camber angle will be seen from front view of car

Range – 2 to 4 degrees.

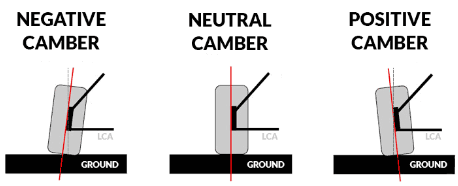

Fig 8 Types of camber angle

Types

- Negative camber

Negative camber is defined as the configuration when the top of the wheel is leaned in towards the center of the car.

Advantage of negative camber

- Benefits of increased grip during heavy cornering with negative camber.
- A negative camber improves the cornering stability of the vehicle, so that when the vehicle is describing a turn, the wheels make complete contact with the road surface and hence the driver feels more confident while driving.
- A negative camber is better for the cornering performance of the vehicle, while a 'zero' camber will help the vehicle to accelerate, as the front wheels of the vehicle will get more surface of the road to interact.

- Positive camber

Positive camber is the opposite in which the top of the wheel is leaned away from center.

Positive camber is also useful for the vehicles in which there is loading on the wheels, either in the front or rear side. It is assumed that the wheels will get the surface of road when they are loaded and if the camber were zero the wheels would have got negative camber after loading.

The best way to determine the proper camber for competition is to measure the temperature profile across the tire tread immediately after completing some hot laps. In general, it's desirable to have the inboard edge of the tire slightly hotter than the outboard edge. However, it's far more important to ensure that the tire is up to its proper operating temperature than it is to have an "ideal" temperature profile. Thus, it may be advantageous to run extra negative camber to work the tires up to temperature.

5.3 King pin inclination

Definition: Kingpin angle was obtained by 2 lines out of which one line is tire center line and another is line joining the upper and lower arm of suspension pivots points.

KPI also seen from front of the vehicle.
Greater the king pin less is the scrub radius so king pin should not be so much greater.

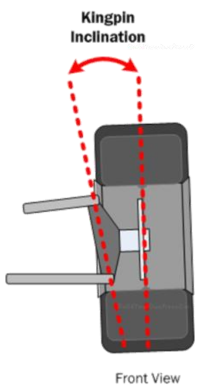

Fig 9 King pin inclination

Effects of king pin: The King Pin Inclination of an axis of turn provides weight stabilization and steady movement of the automobile on a straight line and in turns

5.4 Scrub radius

Definition: The scrub radius is the distance in front view between the king pin axis and the center of the contact patch of the wheel, where both would theoretically touch the road.

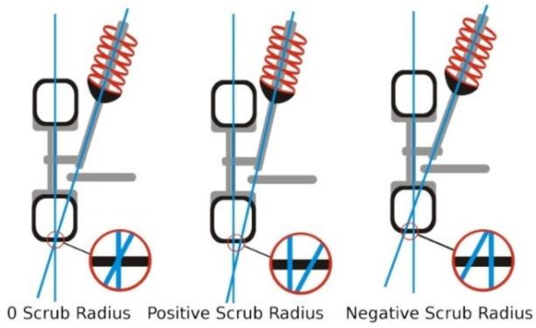

Fig 10 Types of scrub radius

Zero Scrub Radius, the car steers easily and will have little or no kickback from bumps. At the same time there will be virtually no road feel or feedback and there will be a feeling of directional instability while cornering due to the tendency of the tires to squirm.

Positive Scrub Radius will increase steering effort, torque steer (Making the car harder to control during turning and cornering) and kickback on bumps to a considerable degree. At the same time, a blowout or a failure of one front brake could yank the wheel hard enough to pull it out of your hands. The advantage is that there is much greater road feel and feedback so that you can feel when the front tires start to break loose in a corner. Consequently, this is often the set-up of choice on race cars. And you had different braking in the two wheels; it would jerk the wheel out of a straight line, making the car unstable. Usually used for RWD vehicles or Sport Cars.

Negative Scrub Radius will also increase steering effort, torque steer and kickback but to a noticeably lesser degree than the positive kind. Additionally, front tire blowouts and single brake failures will act with less force on the steering wheel. Finally, there will be less road feel and feedback and less ability to feel when the front tires are about to break loose as compared with the positive state. In general, front-wheel drive cars are set-up with negative scrub radius.

5.5 Toe

Toe is the angle created between the front tires when pointed directly forward. There are two possible toe conditions, toe in and toe out.

Definition

Wheel toe-in is an angle formed by the Centre line of the wheel and the longitudinal axis of the vehicle, looking at the vehicle from above, the sum of the toe values for each wheel gives the total toe value,

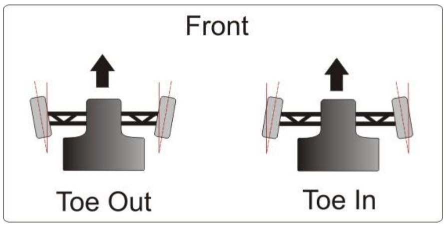

Fig 11 Types of toe angle

When the extensions of the wheel Centre lines tend to meet in front of the direction of travel of the vehicle, this is known as toe-in, If, however the lines tend to meet behind the direction of travel of the vehicle, this is known as toe-out.

The toe adjustment is most commonly made by shortening or lengthening the tie rod links. The less toe you have (in or out) is likely going to give you the least tire wear in most cases. Here are some examples of how toe will affect your race car.

- Front toe-out will introduce a bit of over steer (looser)
- Front toe-in will produce the opposite - under steer (tighter)
- Steering response will be improved with front toe-out

Straight-line stability will be improved with front toe-in

5.6 Types of Load Transfers

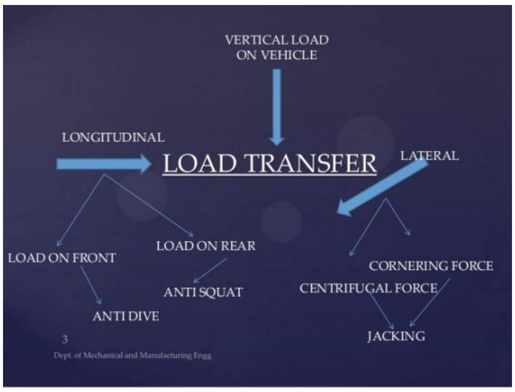

Fig 12 Load transfer

- Longitudinal Load Transfer

Longitudinal load transfer is the result of the car mass accelerating from the front of the vehicle to the back or the back to the front under accelerating or deaccelerating (Braking) respectively. It is important to mention that "The total weight of the vehicle does not change, Load is merely transferred from the wheels at one end of the car to the wheels at other end."

Note: Weight is defined as the weight that rests on the wheel set that is being analyzed i.e. front or back and wheelbase is the distance between center contact patch of the front tire to the Centre of contact of tire of the rear tires.

- Lateral Load Transfer: In essence the lateral load transfer experienced by the vehicle is the same principle as the longitudinal transfer only just rotated to 90 degrees such that load is either transferred from the right to left under left hand corner and from the left to the right in right hand corner

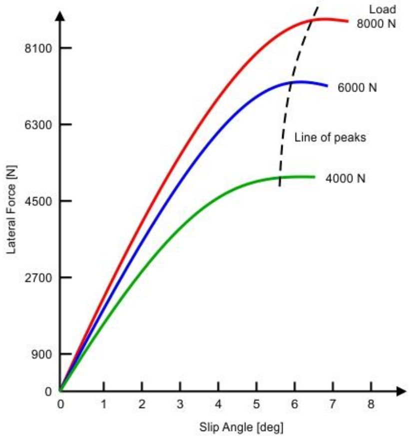

Fig 13 Lateral load vs slip angle

Note: Weight is defined as the weight that rests on the wheel set that is being analyzed i.e. front or back and track width is the distance between the Centre of the contact patch of the right and left tires.

- Vertical Load Transfer: Vertical load transfer is nothing but opposite reaction of vehicle load on wheels and also fluctuating loads occurs during bump.

6. Components of wheel assembly.

Table 1 Components of wheel assembly

1	Upright or knuckle	8	E clips
2	Hub	9	Floater button
3	Bearing lock.	10	Wheel studs lug nut
4	Brackets	11	NY lock nut
5	Tires	12	Tripod
6	Rims	13	Allen bolts and nuts
7	Bearing		

Table 2 Distinguish between the different material

Properties	Mild Steel	Aluminum 6061 T6	Aluminum 7075 T6
Thermal Conductivity	51.9 W/m-K	167 W/m-K	130 W/m-K

Density		7.87g/cc 2.70 g/cc	2.81 g/cc
Hardness, Brinell	131	95	150
Tensile Strength, Ultimate	450 MPa	310 MPa	572 MPa
Tensile Strength, Yield	310 MPa	276 MPa	503 MPa
Poisson Ratio		0.290.33	0.33

Table 3 Comparison of different material

Sr.no	Mild steel	Aluminum 6061 T6	Aluminum 7075 T6
1	Easily available in local market	Available in big market	Available only in big markets
2	Low cost around 60Rs/kg	Average cost around 350RS/kg	High cost around 850Rs/kg
3	Low machining cost	High machining cost like waterjet, etc.	High machining cost like waterjet, etc.
4	Easily available in required sizes	Less Available in required sizes	Not available in required sizes
5	Strength is high	Strength is comparatively less for same dimensions	Strength is more than 6061 for same dimensions.
6	Heavy material	Light material	Light material than 6061

6.1 Upright

The uprights are the central component for the suspension of a Formula SAE race car. All suspension components including 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. All forces that the car will encounter will go through the uprights. The uprights must be sufficiently strong in order to withstand many of these forces occurring simultaneously, as well as any forces that may happen as a result of a crash or other kind of emergency without failure. Any failure of the uprights would render the car un-drivable. The hub's main duty is to prevent excessive deflections, particularly in bending. In torsion, which is mainly experienced during braking, excessive deflections can cause excessive fatigue in the hub as well as other components. Deflection during bending can affect overall camber angle during bump and cornering and can affect toe during braking and cornering. Small toe changes can negatively affect the car's handling, as this will effectively steer the car if the displacement is high enough. Small changes in camber do not necessarily cause immediate change in car behavior, but will change the tire's contact with the road, which can affect overall cornering ability and also cause poor fatigue situations for the hub itself. The hub must connect the inner face of the wheel, the rotor and the upright. As mentioned before the hub and upright are typically connected by a steel spindle. This design includes the hub and an effective spindle in once piece.

- Goal Statement and Task Specifications for the Uprights and Hubs
- Reduce unsprung mass
- must be the sole component in resisting forces in torsion and bending
- must withstand braking force
- must allow minimal compliance during cornering and bump loading
- must fixture a full floating rotor with an effective diameter of 8.75 inches
- must locate the wheel at the proper track width of 1250 mm
- Decision making – by comparing all the parameters we selected aluminum 6061 T6 for upright as per design considerations and also budget concern. [refer table 2 and 3]
- Design of front upright
- First, we have to select the caster angle. We have 4-degree caster angle
- Accordingly, we had given it in upright front view. Then we have select the proper length of upright so the suspension point should get packed within it
- Boring size is according to bearing outside diameter as bearing has to press fit inside that bore.
- Boring thickness is depending on 3 parameter one is bearing thickness and others are bearing lock and step inside for bearing stop.
- Bearing lock and step on opposite side prevent that bearing and whole assembly to move axially.
- Next is to mount caliper position according to proper dimension caliper mounting has to be decide
- Then mount the steering point according to point given by steering head.
- Make hole for bolting of upper and lower suspension A arm bracket and also for bearing lock also counter boring from opposite side
- Proper fillet to avoid the sharp edges
- And weight reduction as per the simulation.

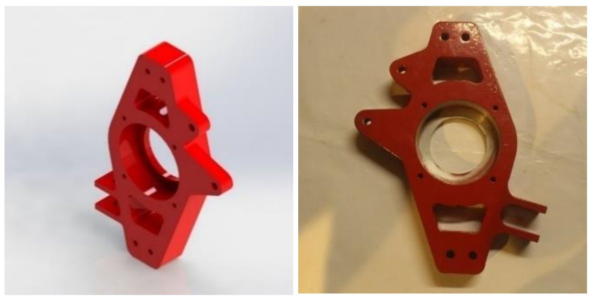

Fig. 14 Front Upright Cad model and actual

Design of rear upright

- Rear upright design is different than front upright design, because the rear upright is not having steering point as in case of front instead rear upright having toe rod.

- Toe rod is used to stabilize the rear wheel as rear wheel only follow the path of front wheel.
- Arrangement of toe rod bracket is similar to front lower bracket.
- Because of this reason the rear upright design is much bulkier than front one
- Other procedure is same as front upright.

Interfacing Systems

The uprights must directly interface with upper and lower control arms, tie rods, brakes, hubs, and wheel speed sensors.

- A-arm geometry is provided by the suspension lead in a vehicle specifications spreadsheet. The a-arms can capture a spherical bearing in the upright OR the upright can capture aspherical bearing in the a-arm.
- Tie rod points connect the steering system the uprights, allow for steering input by the driver, and allow for the adjustment of toe. These are also given in the vehicle specifications spreadsheet.
- The brakes must be mounted to the uprights. Their location relative to the upright is determined by the team member in charge of the braking system

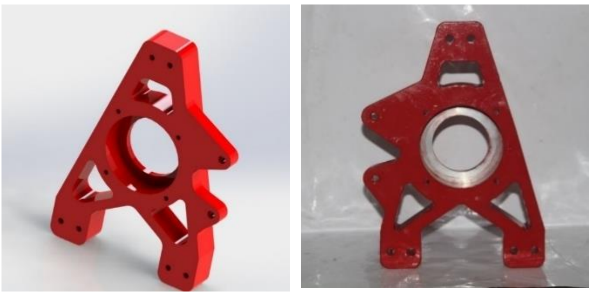

Fig 15 Rear Upright CAD model and actual

Below is a list of some of the design features that were determined while modeling the main body of the front uprights.

- Large pockets were utilized to reduce the weight of the system
- The side ribs of the top and bottom pocket lie on a line that is coincident with the a-arm mounting point and the wheel center. This should increase the stiffness of the upright.
- Careful consideration of manufacturing details had to be taken into account in order to keep the number of machine setups to a minimum and to make the manufacturing process easier. Below is a list of some of the features that were implemented:
- All sharp internal corners were eliminated and were modeled with a 10mm radius to allow for a mill to be used. Attempting to cut the deep pockets with anything smaller could result in chatter or broken tools.
- All bearing bores were modeled at as manufactured dimensions so there is no confusion when machinist was making the part. These dimensions must be double checked before manufacturing so as not to scrap the part.
- Tool clearance was considered for operations such as using a wheel cutter for brake caliper slots. This required modeling of the wheel cutter and cut path to ensure that this operation would be possible given the conditions of the part. It would also be possible to use an end mill to cut these slots as well.

6.2 HUB

The hub's main duty is to prevent excessive deflections, particularly in bending. In torsion, which is mainly experienced during braking, excessive deflections can cause excessive fatigue in the hub as well as other components. Deflection during bending can affect overall camber angle during bump and cornering and can affect toe during braking and cornering. Small toe changes can negatively affect the car's handling, as this will effectively steer the car if the displacement is high enough. Small changes in camber do not necessarily cause immediate change in car behavior, but will change the tire's contact with the road, which can affect overall cornering ability and also cause poor fatigue situations for the hub itself. The hub must connect the inner face of the wheel, the rotor and the upright. As mentioned before the hub and upright are typically connected by a steel spindle. This design includes the hub and an effective spindle in once piece. The hub was made from MS for BOTH

The wheel hub has to be strong enough to withstand the forces acting on it. During a race, there are four main forces acting on the wheel hub

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

The design of the hub was bounded by various conditions such as track width, wheel bearing size, caliper dimensions, and the bolt pattern on the wheel. However, the uprights and hubs make up for the remaining distance. Since it was decided that the hub and spindle would be one component the effective spindle dimensions was driven by the selected bearing. The distance between where the caliper bolts onto the upright and the center of the brake pads drove the placement of the rotor support. Finally, the bolt pattern on the wheels was 4 bolts on a 4-inch circle. Since the upright design was also bounded by many of the stated conditions the two components were designed simultaneously. Some of the hub dimensions could not be finalized until the upright's dimensions was finalized. The initial design determined the rough geometry of the hub to address the task specifications. The front face of the hub matches the bolt pattern of the wheel. As seen in the side view of the assembly the portion of the hub on the right is the effective spindle. The portion directly to the left of the effective spindle will rest against the bearing. The combination of a hub and effective spindle causes large stress concentrations which can lead to premature failure of the component. In cases like these there is no way to avoid stress concentrations; however, there are many options to reduce stress. The hub is essentially a stepped shaft that decreases in size at the point in which it is pressed into the bearing. The most obvious method to reduce the stress concentrations at the step is to design a corner with a radius, however the hub needs to have a flat face to locate the bearing axially and radially on the spindle of the hub.

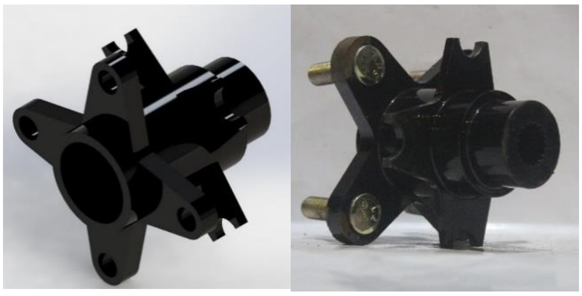

Fig 16 Hub CAD Model and actual

To reduce the stress concentration material was removed directly behind the step to improve the force streamlines.

- Decision making – by comparing all the parameters we selected mild steel for hub as per design considerations and also budget concern. Also, hub is undergoing heavy torque so to compensate that parameter we selected mild steel as our hub material. [refer table 2 and 3]

Design procedure

- First, we have to measure the boss diameter of the rim. oz aluminum has boss diameter of 50 mm so in order to avoid the vovality we take the hub boss diameter of 50 mm.
- Then according to PCD and stud diameter of rim we draw the first flange of hub which is get attached to the rim from inside.
- For weight reduction purpose we made the petal type shape to that flange.
- Then for disc mounting we decide the float for floating disc having PCD 100 mm and according to floater button design and disc design we made hub second flange.
- Then it is essential to keep the disc in Centre of 2 pads of caliper so accordingly that step is made.
- And finally, the last step of 40 mm diameter which is according to bearing inner race for bearing mounting on hub was made.
- Then hole from bearing side was made for insertion of tripod.
- Then slotting is done accordingly the tripod spline in hub.
- Proper fillet to avoid the stress concentration.

6.3 Bearing lock

Decision making – by comparing all the parameters we selected mils steel for bearing lock as per design considerations and also budget concern. Also, as less material is required for bearing lock so the cost of aluminum is more for such small size hence we selected Mild steel for bearing lock [refer table 2 and 3]

Fig 17- Bearing lock CAD model and actual component

Design and manufacturing process

- Bearing lock is used to retain the bearing in their proper axial position.
- So, its diameter was kept according to boring diameter of upright which is 80 mm and with proper tolerance for easy mounting and dismounting in that bore.
- 4 holes at PCD of 95 mm for bolt locking purpose.

6.4 Brackets for an arm mounting

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

Decision making – by comparing all the parameters we selected mils steel for bearing lock as per design considerations and also budget concern as bushings are of mild steel we selected the mild steel as bracket material to reduce the chances of wear. [refer table 4.2 and 4.3]

Upper bracket

- Upper bracket design is slightly angled towards the ground, the reason behind this to compensate the length of upright with respect to chassis suspension mounting point.
- Wafer of A arm should not get touch any of the flange of bracket, so accordingly the angle is given.

Fig 18 Upper Bracket CAD model and actual Component

Lower bracket

- Lower bracket design is less difficult compared to the upper. Because lower A arm is perfectly parallel to ground so bracket is perfectly C shape while seeing from side view.

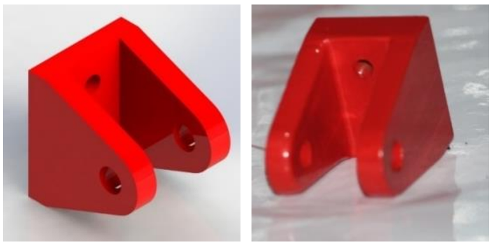

Fig 19 Lower bracket CAD model and actual component

6.5 Tires

A tire is a ring-shaped component that surrounds a wheel's rim to transfer a vehicle's load from the axle through the wheel to the ground and to provide traction on the surface traveled over. Most tires, such as those for automobiles and bicycles, are pneumatically inflated structures, which also provide a flexible cushion that absorbs shock as the tire rolls over rough features on the surface. Tires provide a footprint that is designed to match the weight of the vehicle with the bearing strength of the surface that it rolls over by providing a bearing pressure that will not deform the surface excessively.

The materials of modern pneumatic tires are synthetic rubber, natural rubber, fabric and wire, along with carbon black and other chemical compounds. They consist of a tread and a body. The tread provides traction while the body provides containment for a quantity of compressed air. Before rubber was developed, the first versions of tires were simply bands of metal fitted around wooden wheels to prevent wear and tear.

- Types

Fig 20 Types of racing tires

- Super Softs - These tires have the maximum amount of oil in them and provide excellent grip. Hence, they are the fastest set of tires. They deteriorate really quickly and don't last long.
- Softs - A bit harder than the Super Softs, these set of tires are a little slower but also last a bit longer.
- Mediums - These are the optimum tires to complete majority of the race on. They provide the perfect balance between speed and durability.
- Hard - These tires provide very less grip but last really long. Typically used for 1 stop strategies.
- Intermediates - A variant between hard and full wet tires. To be used when the track is slightly wet. Faster than full wets.
- Full Wets - These tires have full treads and are only to be used when it is raining or the track is really wet. Will deteriorate really fast if used in dry conditions

Dry – (slick)
Brand – Hoosier 20.5 * 7.0 R13
Model number – 43163
Car weight is 230 kg > 200 kg so R13 tires are use
Of car weight is < 200 kg then only we can use R10 wheels
R13 generate more force so more cornering stiffness and self-aligning moment is same as that of R10.

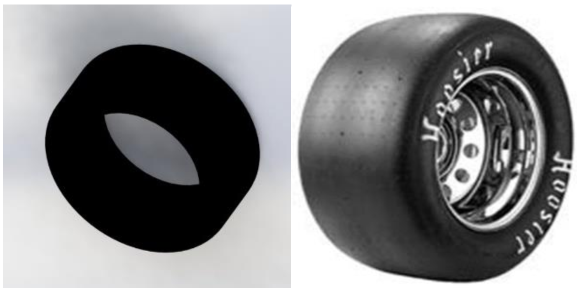

Fig 21 Tyres CAD model and actual

CIRCUIT RACING SLICKS - BIAS

ITEM NUMBER	TIRE SIZE	APPLICATION	TREAD WIDTH	APPROX. DIA.	APPROX. CIRC.	RECOM. RIM WIDTH	MEASURED RIM WIDTH	SECTION WIDTH	COMPOUND COMPARISON SOFT TO FIRM
43101	18.0 x 6.0-10 C2000	F500	6.2"	18.1"	57.0"	6"	6"	8.1"	R25B, R35A
43105	18.0 x 7.5-10	F500	7.5"	18.3"	57.5"	7-8"	8"	9.5"	R25B, R35A
43110	19.5 x 6.5-10	F500	6.5"	19.4"	61.0"	6"	6"	8.2"	R25B
43120	19.5 x 7.5-10	F500	7.5"	19.5"	61.2"	7-8"	8"	9.5"	R25B
43127	20.5 x 6.0-13 A2500	FF	6.0"	20.9"	65.7"	5.5"	5.5"	7.0"	R25B
43130	20.0 x 6.0-13	CF	6.0"	20.7"	65.0"	5.5-6.5"	5.5"	7.0"	R60A
43163	20.5 x 7.0-13 A2000	FB, FC, FM, SR	6.75"	20.8"	66.2"	6-8"	6"	8.0"	R25B, R35B, R45B
43168	20.0 x 7.5-13 A2500	SR	8.0"	20.6"	65.0"	6-9"	8"	9.4"	R25B, R35A

Fig 22 Specification of tire

Fig 23 Tire compounds

Road racing tires: (Includes catalog numbers beginning with 43, 44, 45, 46) In most cases, Hoosier tires used in Road Race applications should be

mounted with the serial code toward the center of the vehicle. Once a tire has been run in the proper orientation it is acceptable to remount the tire in the opposite direction to even out the wear.
13 - diameter of the wheel rim in inches.

R - R means it is radial construction.

Radial, which means the layers run radially across the tire.

These production race slicks provide excellent grip superior to other race slicks, require minimal warming to allow you to drive from the drop of the flag, and are very consistent through their long lifespan. Testing has proven these radial slicks to improve every aspect of vehicle dynamics from more grips,

It is unsafe to operate any Hoosier Racing Tire, including D.O.T. tires, on public roads.

The prohibited use of Hoosier Racing Tires on public roadways may result in loss of traction, unexpected loss of vehicle control, or sudden loss of tire pressure, resulting in a vehicle crash and possible injury or death. And like all Dry Racetrack tires they are not intended to be driven in near-freezing temperatures, through snow or on ice.

It's also essential these tires be stored indoors at temperatures maintained above 32 degrees F.

- better cornering speeds,
- improved tread wear,
- less rolling resistance
- Faster lap times.

Tire compound R25B, R35A, and R45B

R25B – SOFT – operate between temperature 50 – 80 degrees Celsius

R35B – MEDIUM

R45B - HARD

Radial, which means the layers run radially across the tire.

The hard compound tire has a greater number of cross-links between the long rubber molecules (introduced by Vulcanization). This restricts the length of the rubber molecule that can interact with the track surface. Less interaction between the rubber molecules and the surface means that the tire has fewer grips on the track.

The situation is exactly opposite for the soft compound tire. It has far fewer cross-links and therefore a greater length of rubber molecule that can interact with the surface. Greater interaction means greater grip.

Over 150 components are mixed to form the tire material mainly are mechanical oil, carbon and rubber by varying their integration can produced tire with different characteristics.

Components are often described as hardness and softness; it does not mean actually how hard the tire upon touch but describes how the tires are behaved on race track. Harder compound tires are more durable then softer compounds tires. Means driver can complete longer distance while running before the tire can lose their performance. However, the harder compound will provide fewer grips than softer one meaning that driver will not able to go as fast as they want. The choice of tire is therefore compromise between performance and grip. Choose soft and get more grips while running.

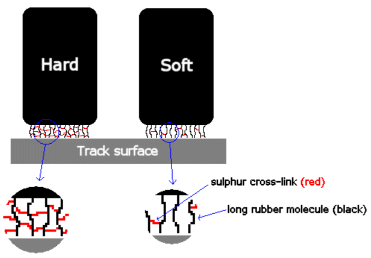

Fig 24 Hard vs soft tire material

Nitrogen vs. air

- Most car tires are filled with air to ensure they are road ready.
- However, some racing cars and other commercial vehicles pump their tires with nitrogen gas.
- This is a colorless and odorless non-toxic gas used to reduce the temperature of the tire and increase its lifespan.

Fig 25 Advantage of nitrogen

6.6 Rims

Alloy wheels are wheels that are made from an alloy of aluminum or magnesium. Alloys are mixtures of a metal and other elements. They generally provide greater strength over pure metals, which are usually much softer and more ductile. Alloys of aluminum or magnesium are typically lighter for the same strength, provide better heat conduction, and often produce improved cosmetic appearance over steel wheels. Although steel, the most common material used in wheel production, is an alloy of iron and carbon, the term "alloy wheel" is usually reserved for wheels made from nonferrous alloys.

- Strength to weight ratio. Aluminum is typically not as strong as steel, but it is also almost one third of the weight. This is the main reason why aircraft are made from Aluminum.
- Corrosion. Stainless steel is made up of iron, chromium, nickel, manganese and copper. The chromium is added as an agent to provide corrosion resistance. Also, because it is non-porous the resistance to corrosion is increased. Aluminum has a high oxidation and corrosion resistance mainly due to its passivation layer. When aluminum is oxidized, its surface will turn white and will sometimes pit. In some extreme acidic or base environments, Aluminum may corrode rapidly with catastrophic results.
- Thermal Conductivity. Aluminum has a much better thermal conductivity (conductor of heat) than stainless steel. One of the main reasons it is used for car radiators and air conditioning units.
- Cost. Aluminum is typically cheaper than stainless steel.
- Workability. Aluminum is fairly soft and easier to cut and form. Due to its resistance to wear and abrasion, Stainless can be difficult to work with. Stainless steels are harder and are especially harder to form than aluminum.
- Welding. Stainless is relatively easy to weld, while Aluminum can be difficult.
- Thermal properties. Stainless can be used at much higher temperatures than Aluminum which can become very soft above about 400 degrees.
- Electrical Conductivity. Stainless steel is a really poor conductor compared to most metals. Aluminum is a very good conductor of electricity. Due to its high conductance, light weight, and corrosion resistance, high-voltage overhead power lines are generally made of aluminum.
- Strength. Stainless steel is stronger than Aluminum (provided weight is not a consideration).
- Effect on Foods. Stainless steel is less reactive with foods. Aluminum can react to foods which may affect color and flavor.

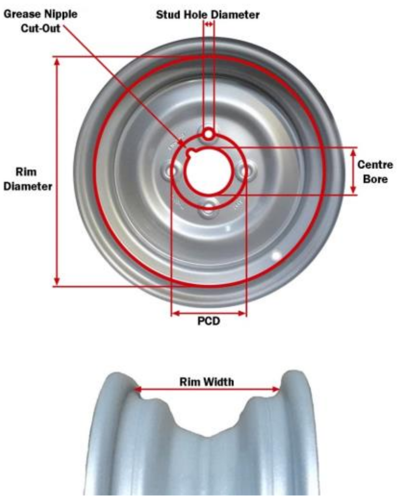

Fig 26 Rim constructional view

Oz aluminum vs. oz magnesium

- Easily available
- They are strong enough without suffering significant damage
- Aluminum is heavier than magnesium but more corrosion resistant.
- Once the magnesium gets bend it cannot be straightened as in case of aluminum
- Magnesium is highly reactive and less stable than aluminum
- Magnesium is expensive than aluminum

Fig 27 Rim CAD model and actual component

13" Aluminum wheel for slick tires

- standard dimension: 7x13
- offset: 22
- indicative weight: 3.4 Kg
- standard bolt pattern: 100x4 and center hole diameter 50 mm for wheels to be fixed by bolts or 100x12 and center hole diameter 72 mm for wheels with central lock
- finishing: black with OZ RACING white letterings

6.7 Bearing

A bearing is a machine element that constrains relative motion to only the desired motion, and reduces friction between moving parts. The design of the bearing may, for example, provide for free linear movement of the moving part or for free rotation around a fixed axis; or, it may prevent a motion by controlling the vectors of normal forces that bear on the moving parts. Most bearings facilitate the desired motion by minimizing friction. Bearings are classified broadly according to the type of operation, the motions allowed, or to the directions of the loads (forces) applied to the parts.

Rotary bearings hold rotating components such as shafts or axles within mechanical systems, and transfer axial and radial loads from the source of the load to the structure supporting it. The simplest form of bearing, the plain bearing, consists of a shaft rotating in a hole. Lubrication is often used to reduce friction. In the ball bearing and roller bearing, to prevent sliding friction, rolling elements such as rollers or balls with a circular cross-section are located between the races or journals of the bearing assembly. A wide variety of bearing designs exists to allow the demands of the application to be correctly met for maximum efficiency, reliability, durability and performance

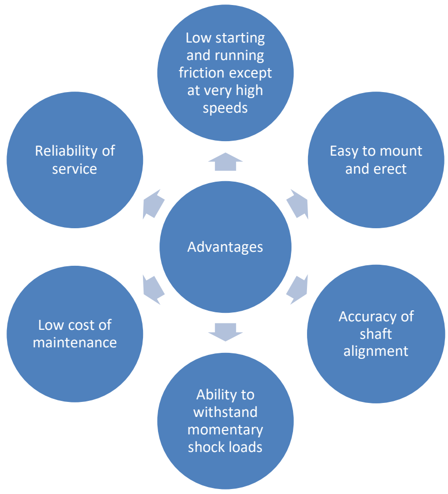

Fig 28 Advantages of Rolling Contact Bearings Over Sliding Contact Bearings

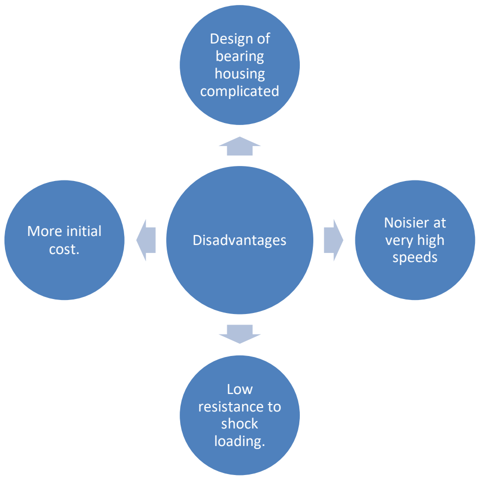

Fig 29 Disadvantages of Rolling Contact Bearings Over Sliding Contact Bearings

Ball Bearings

Ball bearings are most common type of bearing and can handle both radial and thrust loads. Ball bearings are also known as deep-groove single-row or Conrad bearings. The inner ring is typically fastened to the rotating shaft and the groove on the outer diameter provides a circular ball raceway. The outer ring is mounted onto the bearing housing. The ball bearings are housed in a race and when the load is applied, it is transmitted from the outer race to the ball and from the ball to the inner race. The raceway grooves have typical curvature radii of 51.5% to 53% of the ball diameter. Smaller curvature raceways can cause high rolling friction due to the tight conformity of the balls and raceways. Higher curvature raceways can shorten fatigue life from increased stress in the smaller ball-race contract area.

Advantages of Ball Bearing

1) The bearing uses grease with higher dripping point (195 degree)
2) Large operating range temperature (-40 ~ 180 degree)
3) Better sealing shield to prevent leaking of lubricant and avoid foreign particles entering the casing
4) Easy bearing replacement
5) Increase motor performance (less motor friction)
6) Bearing is easy available in the market
7) Less precaution during assembly
8) Cheaper cost for replacement

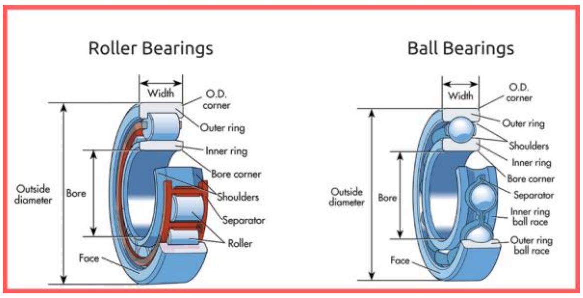

Fig 30 Constructional view of bearings

What are the benefits of deep groove ball bearings?

- **Versatility –** because they can carry radial and axial loads, they have a wider range of applications for many industries
- **Cost savings –** deep groove ball bearings create less friction torque. This lowers operating temperature (which extends the life of the bearing) and reduces energy cost of running equipment (such as conveyor belts).
- **Less upkeep –** because of their simple design, low operating temperature, and low friction, deep groove ball bearings have a longer expected shelf life than other bearings. They do not require additional lubrication after installation, which also means less maintenance downtime.

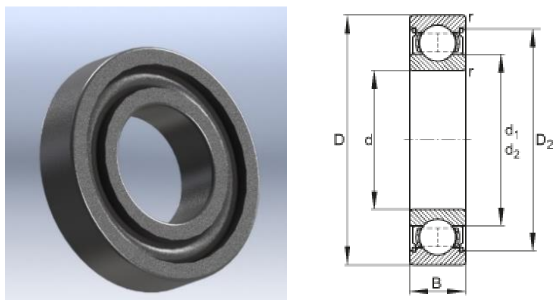

Fig 31 Deep groove ball bearings 6208-2Z

Table 4 Bearing specification

Bearing number	6208-2Z
Size (mm)	40x80x18
Brand	FAG
Bore Diameter (mm)	40
Outer Diameter (mm)	80
Width (mm)	18

Table 5 Details of 6208 bearing

d	40 mm
D	80 mm
B	18 mm
D_2	70,4 mm
$D_{a\ max}$	73 mm
d_1	53 mm
m	0,382 kg / Weight
$d_{a\ min}$	47 mm
$r_{a\ max}$	1 mm
r_{min}	1,1 mm
C_r	31000 N / Dynamic load rating (radial)
C_{0r}	17800 N / Static load rating (radial)
n_G	10400 /min / Limiting speed
C_{ur}	820 N / Fatigue limit load, radial

6.8 Circlip

Definition: A circlip is a type of retaining ring or fastener that takes the form of a flexible, open-ended ring, made from metal. They may also be known as retaining rings, snap rings, c-clips, or Jesus clips.

- Most applications which need a part to pivot, spin, or turn, where a bearing is used, will need a circlip or lock ring fastening.
- Circlips are commonly used in motors, turbines and pistons Circlips fit into a groove on the inside of a bore or the outside of a shaft.
- They work as a load-bearing shoulder which positions and holds mechanical parts. They provide continuous radial force and are secure against high rotational speeds because they are retained within the groove.

The two basic designs circlips come in are external and internal. External circlips fit around a shaft, pressing on it, whilst internal circlips are fitted inside a cylindrical bore, or housing, and push outwards.

E-clips

Fig 32 - E-clip CAD model and Actual component

E-clips are fitted radially, rather than axially, onto shafts. Similar to standard circlips, they fit in a groove. However, they do not have grip holes so can't be used with circlip pliers. They can be installed by being pushed onto a shaft with a special tool, or pair of standard pliers. As e-clips can be fitted radially, they can be installed or removed

6.9 Floater button

Decision making – by comparing all the parameters we selected mils steel for floater button as per design considerations and also budget concern. Also, as less material is required for floater button so the cost of aluminum is more for such small size hence we selected Mild steel for floater button [refer table 2 and 3]

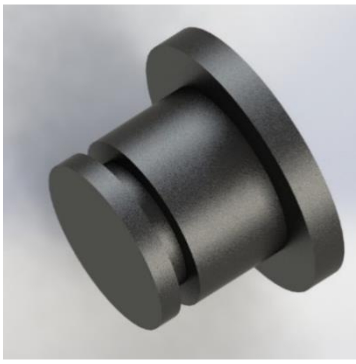

Fig 33 Flooter button CAD model

Design and manufacturing procedure

- Diameter of 24 mm having thickness 3 mm is made.
- Then next diameter of 16 mm which is according to the brake disc and hub geometry for the length 9 mm which include disc thickness and 2 washer arrangements.
- Groove for e clip arrangement according to e clip diameter and thickness.

6.10 Studs and Lug nut for rim mounting

Wheel studs are the threaded fasteners that hold on the wheels of many automobiles. They are semi-permanently mounted directly to the vehicle hub, usually through the brake drum or brake disk. Lug nuts are fastened onto the wheel stud to secure the wheel

Wheel studs are replaceable and come in two basic kinds: press-in and screw-in. Welded-in studs are possible but unlikely to be encountered.

- Screw-in

Screw-in studs simply screw into the existing threaded bolt hole in the hub. The end that screws into the hub is usually either threaded with a higher tolerance fit or installed with a chemical thread-locking fluid to keep it from backing out from the hub when the lug nut is removed.

- Press-in

Press-in studs are installed from the back side of the

disk or drum hub and may require removal of the hub from the vehicle for installation or removal. They consist of a threaded portion and a larger diameter section, called the knurl, that is splined to prevent rotation. The diameter of the knurl is larger than the hole in the hub requiring a press fit to seat the stud. The stud is prevented from being pulled through the hub by a larger diameter stop on the end.

Most press-in studs are designed and recommended to be installed with a mechanical or hydraulic press to ensure proper seating without damage

Fig 34- Studs CAD model and actual

Table 6 Wheel stud specification

Knurl Length (in.)	0.275
Under head Length (in.)	2.6
Wheel Stud Style	Press-in
Wheel Stud Thread Size (mm)	12 x 1.50 RH

A lug nut or wheel nut is a fastener, specifically a nut, used to secure a wheel on a vehicle. Typically, lug nuts are found on automobiles, trucks (lorries), and other large vehicles using rubber tires

A Corrosion resistant coating protects against moisture, salt spray & oxidization, 60-degree taper to suit most alloy wheels.

Fig 35 Lug Nut CAD model and actual

The conical lug's taper is normally 60 degrees and is designed to center the wheel accurately on the axle, and to reduce the tendency for the nut to loosen, due to fretting induced precession, as the car is driven.

Table 7 Lug nut specification

Conical Seat Type	Standard
Lug Nut Finish	Chrome
Lug Nut Head Style	Internal hex
Lug Nut Material	Steel
Lug Nut Seat Style	Conical seat - 60 degree
Lug Nut Thread Size	12mm x 1.50 RH
Wrench Size Required (mm)	19mm

7. Calculation

Table 8 Design parameters

The parameters such as king pin inclination, caster, camber, track width is the essential factor which is to be consider before designing

King Pine Inclination	8 degrees
Caster Angle	4 degrees
Tie Rod Angle	6 degrees
Track Width	1200 mm rear, 1250 mm front
Wheel Diameter	520.4 mm
Total length of Knuckle	240 mm
Upper A-arm angle	(- 3) degrees
Lower A-arm angle	0

7.1 Design of Spindle

Firstly, the spindle is designed on which other components such as knuckle; bearings and hub will be fitted.

Material: - The material for manufacturing the spindle is taken to be mild steel. There will be parts which will be press fitted on the spindle. Besides the yield strength in tension of mild steel is also high. [refer table 4.2 and 4.3]

Syt = 370 N/mm^2
Endurance Limit = 220 N/mm^2

The forces acting on the spindle are as follows

A. Weight of the vehicle

During static and dynamic conditions, a constant force of the self-weight is acting on the spindle at the part inside the knuckle. Even if it is considered as the more than half the weight of the car is acting at the front portion of the car during braking, the weight on the one wheel is

Weight in the front portion = 175 kg
Weight on one tire = 175/2=87.5 kg
Force due to weight of the vehicle = 87.5 × 9.81 = 858.375 N
weight to be 1000 N.

B. Bump force on the tire

At the time of a bump in the surface a force will act on the portion of the spindle which is inside the spindle. This is because the hub is bolted directly to the wheel. This force is obtained from the wheel rate. For design purpose the wheel rate is kept as $27/mm^2$. Also, it is considered that there will be no bump more than 30mm as the track is extremely flat.

Bump Force = Wheel rate × Travel due to bump = 27× 30 =810 N

Bump Force = 810 N

C. Torque on the spindle.
Torque = mass on the spindle × g × radius of the wheel × coefficient of friction
Mass on spindle = 75 kg
g= 9.81
radius of wheel = 0.26 m
coefficient of friction = 0.65

Therefore, **torque on spindle of hub = 125 KN**.

7.2 Shear Failure of the Spindle

The spindle is likely to fail in shear because of the bump force. The spindle is a critical part and it is not at all desirable to fail any condition, hence the factor of safety is taken to be 3.

The allowable shear stresses

$$\tau = \frac{Syt \times 0.5}{Fs}$$
$$= \frac{370 \times 0.5}{3}$$
$$= 61.66 \text{ N / mm}^2$$

τ = 62 N / mm²

Now,

$$\tau = \frac{\text{Force}}{\text{Resisting Area}}$$
$$62 = \frac{810}{\frac{\pi}{4} d^2}$$

d = 4.07 mm
Thus, from the shear failure the diameter of spindle- **'d' comes out to be 5 mm**

7.3 Bending and Torsion Failure of the Spindle

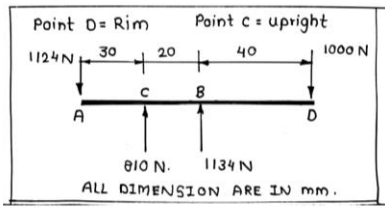

Fig 36 FBD of hub and upright

To find out the Maximum Bending Moment, the SFD and BMD are to be drawn. Referring to the force diagram of the spindle
Equating forces along Y-axis to 0
$R_a + R_b$ =1000-810
$R_a + R_b$ = 190 N

Equating the moments about point A =0
- (810 × 30) – (R_b × 50) + (1000 × 90) = 0
R_b = 1314 N
Ra = (-1124) N
After reconsidering the force diagram comes out be as shown in figure
Now,
Moment about D= 0
Moment about B= (1000 × 40) = 40000 N-mm
Moment about C= (1000 × 60) – (1314 × 20) = 33720 N-mm
Moment about A= (1000 × 90) – (1314 × 50) – (810 × 30) = 0 N-mm
From above it is clearly seen that the maximum bending moment is at point B, thus Mb= 40000 N-mm

The Torque acting on spindle can be directly taken
M_t = 125 N-m = 125000 N-mm
As mild steel is a ductile material, using maximum shear stress theory to find out the diameter of the spindle

$$d^3 = \frac{16}{\pi \times \tau} \times \sqrt{(Mb^2 + Mt^2)}$$
$$d^3 = \frac{16}{\pi \times 62} \times \sqrt{(40000^2 \times 125000^2)}$$

d = 22.09 mm =23 mm
d = 23 mm
Thus, from the bending and torsional failure the diameter of spindle- 'd' comes out to be 23 mm
And we take it as 40 mm
d = 40 mm

7.4 Design of Knuckle / uptight

Knuckle is that part of the wheel assembly which is press fitted on the spindle and the A-arms are also mounted on the Knuckle. Besides the knuckle also serves the function of providing mounting to the Brake Caliper. The Steering Arm which is used to connect the wheel assembly and the tie rod is also mounted on the knuckle. Thus, due to all these mountings, there are a lot of forces acting on the knuckle. The Knuckle as such is subjected to completely reversed types of stress while turning from one turn to the other and also during braking and accelerating. Thus, a brittle material is not at suitable for this application. Thus, taking a tensile material called Aluminum 6061 T6. It has medium strength to weight ratio. Thus, with much lower weight one can produce strong knuckles.

7.5 Longitudinal Forces during Braking

While Braking, the weight of the rear side tends to come in the front side of the vehicle so there is a load transfer that is taking place form rear to front. It interns affects the knuckle as these forces act on the A-arm mounting points through the A-arms.

Considering Maximum acceleration of 1g = 9.81 m/s2

Force at the front side = mass at the rear side of the vehicle × acceleration

Let the mass at the rear side of the vehicle be 0.6 times the total weight

Mass at the rear side of the vehicle=0.6 × 300 =180 kg
Force = 180 × 9.81
Force = 1765.8 N
Now force on 1 wheel =1765.8/2 =882.9 N
Longitudinal Force =882.9 N

7.6 Lateral Forces

Lateral forces are because of two reasons – centrifugal force and lateral load transfer from outside to inside while turning. The centrifugal force is considered as follows
Let the vehicle take a turn of 6m turning radius and at a speed of 30kmph = 8.33 m/s.
r = turning radius =6m

1 km = 1000 m; 1 hr. = 3600 sec.

$$1\ km/hr = \frac{1000}{3600}\ m/sec = \frac{5}{18}\ m/sec$$

To convert km/hr. into m/sec, multiply the number by 5 and then divide it by 18.

$$\text{Centrifugal Force} = \frac{m \times v^2}{r} = \frac{0.4 \times 300 \times 8.3333^2}{6}$$

Centrifugal Force = 1388.77 N

Now consider if all the weigh at the front side comes on the wheel assembly the force will be Force due to lateral load transfer = 0.4 × 300 × 9.81 =1175.5

Lateral load transfer = 1175.5 N

7.7 Force on the Steering Arm

According to the steering effort, the force on steering arm was found out to be 1165.52 at an angle of 8.5560. After resolving these forces.

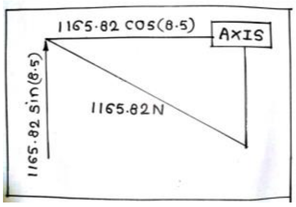

Fig 37 Steering force FBD

Force 1 = 1165.82 × cos (8.556) = 1152.84 N
Force 2 = 1165.82 × sin (8.556) = 173.44 N
But the force on steering arm =1165.52 N

- Forces on the caliper mounting points due to torque

The radius for the upper and lower caliper mount points are 89 mm.

$$\text{Force} = \frac{\text{Torque}}{\text{radius}} = \frac{125000}{89} = 1404.49\ \text{N}$$

7.8 Selection of Knuckle Upper and Lower Bracket Bolt

The bolts are standard parts and have a defined value of yield strength. All bolts used in the Wheel Assembly are made up of a minimum of 8.8 Grade.
S_{yt} = 640 N/mm^2
Factor of Safety = 2

Material = carbon steel – quench and tempered.
Shear Force acting on these bolts $= \sqrt{442^2 + 695^2} =$ 823.64 N

Shear Stress on the bolts =

$$\tau = \frac{\text{Syt}\times 0.5}{\text{Factor of Safety}} = \frac{640\times 0.5}{2} = 160\ \text{N/mm}^2$$

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 1690 = \frac{823.64}{2 \times \left(\frac{\pi}{4} d_c\, 2\right)}$$

$$\therefore d_c = 1.81\ \text{mm}$$

$$\therefore d = \frac{d_c}{0.8}$$

d = 2.5 mm
Value is too small for practical use, so
bracket bolt size is M8

7.9 Selection of Caliper Mounting Bolt

$$\text{Shear Stress on the bolts} = \tau = \frac{\text{Syt} \times 0.5}{\text{Factor of Safety}}$$

$$= \frac{640 \times 0.5}{2}$$

=160 N/mm^2

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 160 = \frac{1404.49}{2 \times \left(\frac{\pi}{4} d_c\, 2\right)}$$

$$\therefore d_c = 3.34\ \text{mm}$$

$$\therefore d = \frac{d_c}{0.8}$$

d = 4.17 mm

Thus, **selecting the caliper bolt size as M8**

7.10 Selection of Steering Arm Bolt

Shear Stress on the bolts =

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor of Safety}} = \frac{640 \times 0.5}{2}$$

= 160 N/mm^2

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 160 = \frac{1165.49}{2 \times \left(\frac{\pi}{4} \mathrm{d_c}\, 2\right)}$$

$$\therefore \mathrm{d_c} = 2.09 \text{ mm} \therefore \mathrm{d} = \frac{\mathrm{d_c}}{0.8}$$

d = 2.61 mm

Thus, **selecting the Steering arm bolt size as M8**

Longitudinal Shear Failure of the Knuckle Bracket

Allowable Stress in the knuckle in shear =

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor ofSafety}} = \frac{503 \times 0.5}{2} = 125.75 \text{ N/mm}^2$$

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 125.75 = \frac{442}{2 \times (\mathrm{t} \times \mathrm{b})}$$

$$\mathrm{t} * \mathrm{b} = 1.7574$$

Where,

t= thickness of one plate of bracket

b= distance between the hole and the end of bracket

If t = 6mm b= 0.3 mm

Thus, for practical reasons the **width of bracket in longitudinal direction is taken to be 12 mm**
The thickness of the bracket is taken as 6 mm

7.11 Lateral Shear Failure of the Knuckle Bracket

Allowable Stress in the knuckle in shear =

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor ofSafety}} = \frac{503 \times 0.5}{2}$$

$$= 125.75 \text{ N/mm}^2$$

Lateral force acting on bracket = 695 N

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 125.75 = \frac{695}{2 \times (\mathrm{t} \times \mathrm{b})}$$

t × b = 2.7634

Where,

t= thickness of one plate of bracket

b= distance between the hole and the end of bracket

If t = 6mm b= 0.46 mm

Thus, for practical reasons **the width of bracket in lateral direction is taken to be 10 mm, The thickness of the bracket is taken as 6 mm**

7.12 Bending Failure of Knuckle

Bending due to Longitudinal Force

This bending is due to the force of 883 N. The longest part of knuckle is 120 mm away from the center and the knuckle is almost symmetric. Thus, bending moment in such cases is taken to be

Mb = 883 × 0.5 × 120

Mb = 52980 N-mm

Now,

By Flexural Equation

$$\frac{\mathrm{M_b}}{\mathrm{I}} = \frac{\sigma_\mathrm{b}}{\mathrm{y}}$$

$$\sigma_\mathrm{b} = \frac{\text{Syt}}{\text{Factor of Safety}} = \frac{276}{2} = 138 \text{ N/mm}^2$$

$$\mathrm{y} = \mathrm{b}/2$$

$$\mathrm{I} = \frac{1}{12}\mathrm{tb}^3$$

$$\frac{52980}{\frac{1}{12}\mathrm{tb}^3} = \frac{138}{\mathrm{b}/2}$$

$$\mathrm{t} \times \mathrm{b}^2 = 2048.18$$

If t= 20mm, b = 10 mm

But here it is also important to understand that the bearing will be fitting in the knuckle thus for this reason the width of knuckle is taken as 40mm at the upper side and 100 mm at centre where bearing get fitted and would then decrease to 40 mm till the end.
Thus, the width of knuckle is 40mm.
The thickness of knuckle is 30 mm.

Bending due to Centrifugal Force

This bending is due to the force of 1388.77 N. The longest part of knuckle is 120 mm away from the center and the knuckle is almost symmetric. Thus, bending moment in such cases is taken to be

Mb = 1388.77 × 0.5 × 120

Mb = 83326.2 N-mm

Now,

By Flexural Equation,

$$\frac{\mathrm{M_b}}{\mathrm{I}} = \frac{\sigma_\mathrm{b}}{\mathrm{y}}$$

$$\sigma_\mathrm{b} = \frac{\text{Syt}}{\text{Factor of Safety}} = \frac{276}{2} = 138 \text{ N/mm}^2$$

$$\mathrm{y} = \mathrm{t}/2$$

$$\mathrm{I} = \frac{1}{12}\mathrm{bt}^3$$

$$\frac{83326.2}{\frac{1}{12}\mathrm{bt}^3} = \frac{138}{\mathrm{b}/2}$$

$$\mathrm{b} \times \mathrm{t}^2 = 3622.87$$

If t= 20mm, b = 9.05 mm

But here it is also important to understand that the bearing will be fitting in the knuckle thus for this reason the width of knuckle is taken as 40mm at the upper side and 100 mm at Centre where bearing get fitted and would then decrease to 40 mm till the end.
Thus, the width of knuckle is 40mm.
The thickness of knuckle is 30 mm.

7.13 Shear Failure of the Steering Arm

Allowable Stress in the knuckle in shear =

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor ofSafety}} = \frac{503 \times 0.5}{2} = 125.75\ \text{N/mm}^2$$

Force acting on steering arm as obtained from result is 1165 N

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 125.75 = \frac{1165}{2 \times (t \times b)}$$

t × b = 4.6322
Where,
t= thickness of one plate of steering arm
b= distance between the hole and the end of bracket
If t = 6mm b= 0.77 mm
Thus, for practical reasons the **width of bracket in lateral direction is taken to be 10 mm**
The thickness of the bracket is taken as 6 mm

7.14 Bending Failure of Steering Arm

This bending is due to the force of 1165 N. The longest part of steering arm is 50 mm away from the center and the knuckle Thus bending moment in such cases is taken to be
Mb = 1165 × 50
Mb = 58250 N-mm
Now,
By Flexural Equation

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{\text{Syt}}{\text{Factor of Safety}} = \frac{503}{2} = 251.5\ \text{N/mm}^2$$

$$y = t/2$$

$$I = \frac{1}{12} bt^3$$

Where,
t= thickness of steering arm in contact with knuckle
b= width of steering arm in contact with knuckle

$$\frac{58250}{\frac{1}{12} bt^3} = \frac{251.5}{t/2}$$

$$t \times b^2 = 1389.66$$

If t= 8, b = 5.579 mm

Thus, the width of steering arm is 10 mm.
The thickness of steering arm is 6 mm.

7.15 Shear Failure of the Caliper Mounting

Allowable Stress in the knuckle in shear =

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor ofSafety}} = \frac{503 \times 0.5}{2} = 125.75\ \text{N/mm}^2$$

Force acting on caliper mounting is 2158.28 N

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 125.75 = \frac{2158.28}{(t \times b)}$$

t × b = 17.163
Where,
t= thickness of one caliper mount
b= distance between the hole and the end of mount
If t = 6mm b= 2.86 mm

Thus, the **width of the caliper mount is taken to be 12 mm**
And for safety purpose **The thickness of the bracket is taken as 6 mm.**

7.17 Design of Hub

Hub is the part of wheel assembly on which the wheel and disk are mounted. Both the Wheel as well as the disk is mounted on the hub with the help of bolts. As discussed earlier the outer race of the bearing is press fitted in the hub and hence provision is made in the hub to enclose the bearing. The Hub itself is made of 2 Petal parts. One of the wheel and the other of the brake disk.

The following Forces are acting on the Hub.
Torque on the Brake Disk Petal: -

A torque of 125 Nm is acting on the Brake Disk Petal.

$$\text{The Force acting on each hole} = \frac{\text{Moment} \div \text{Radius}}{\text{No. of holes}}$$

$$= \frac{125000 \div {}^{100}/_{2}}{4}$$

= **625 N**

Torque on the Wheel Petal

In order to sustain this braking effect, the wheel must also provide and equal and opposite torque. Thus, the magnitude of torque is same but the direction is opposite.

$$\frac{125000 \div {}^{100}/_{2}}{4}$$

= **625 N**

Force due to Side Impact

If the vehicle is banged by other vehicle from side or if the vehicle has a collision with the fencing from side, there are chances that the petals might bend. Hence this side impact force must also be considered.

Here the side Impact force is taken to be 2G = 2 × g × vehicle mass

Impact force = 2 × 9.81 × 300 = 5886 N

Impact force on 1 petal = 8829/4 = 1471.5 N

Selection of Wheel Bolt

Shear Stress on the bolts =

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor of Safety}} = \frac{640 \times 0.5}{2} = 160\ \text{N/ mm}^2$$

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 160 = \frac{625}{\left(\frac{\pi}{4} d_c\, 2\right)}$$

d = 2.91 mm

also, in the **rim there is provision of M12 bolt. Hence selecting bolt of M12.**

Selection of Brake Disk Bolt

Shear Stress on the bolts

$$t = \frac{\text{Syt}\times 0.5}{\text{Factor of Safety}} = \frac{215\times 0.5}{2} = 53.75\ \text{N/ mm}^2$$

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 53.75 = \frac{625}{\left(\frac{\pi}{4} d_c\, 2\right)} = d_c = 2.99\text{mm}$$

$\therefore$ d = $\frac{d_c}{0.8}$ d = 3.75 mm

Thus, we select the **floater button of diameter 16 mm**

Shear Failure of Petal

Allowable stress in the Hub in shear =

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor of Safety}} = \frac{370 \times 0.5}{2} = 92.5\ \text{N/ mm}^2$$

Force acting on Petal is 625 N

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 92.5 = \frac{625}{(2 * (t * b))}$$

$t * b = 3.37\text{mm}$

Where,

t= thickness of Wheel Petal

b= distance between the hole and the end of petal

If t = 8mm b= 0.475 mm

Thus, the width of the petal is taken to be 12mm. The thickness of the petal is taken as 10 mm.

Bending of Wheel Petal

This bending is due to the force of 625 N. The radius of effective bending is 50 mm

Mb = 625 × 50

Mb = 31250 N-mm

Now,

By Flexural Equation,

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{\text{Syt}}{\text{Factor of Safety}} = \frac{370}{2} = 185\ \text{N/mm}^2$$

y = 2b/2

2b=d

$$I = \frac{1}{12} td^3$$

Where,

t= thickness of knuckle

b= width of knuckle

$$\frac{31250}{\frac{1}{12} td^3} = \frac{185}{d/2}$$

$t \times d^2 = 1013.51$

If t= 8mm d = 11.36 mm

Thus, the width of knuckle is 10mm.

The thickness of knuckle is 10.11 mm

Shear Failure of Petal

Allowable stress in the Hub in shear

$$\tau = \frac{\text{Syt} \times 0.5}{\text{Factor of Safety}} = \frac{370 \times 0.5}{2} = 92.5\ \text{N/ mm}^2$$

$$\frac{M_b}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{\text{Syt}}{\text{Factor of Safety}} = \frac{370}{2} = 185\ \text{N/mm}^2$$

y = t/2

$$I = \frac{1}{12} bt^3$$

Where,

t= thickness of petal

b= width of petal

Force acting on Petal is 625 N

$$\tau = \frac{\text{Force}}{\text{Area}}$$

$$\therefore 92.5 = \frac{625}{(2 * (t * b))}$$

$t * b = 3.37\text{mm}$

Where,

t= thickness of Wheel Petal

b= distance between the hole and the end of petal

If t = 5mm b= 0.05 mm

Thus, the width of the petal is taken to be 15 mm

The thickness of the petal is taken as 5 mm.

7.18 Selection of Bearing

W_R = Radial Load = longitudinal load = 882.9 N

W_A = Axial load = lateral load = 1175.5 N

N = 8500 rpm

Since Avg. life of Bearing is 5 years at 10hrs/day

Therefore, life of Bearing in hrs.

$L_H = 5 \times 300 \times 10$

$= 15000$ hrs. … (300 working day/year)

Life of bearing in revolution

$L = 60\ N\ \times L_H$

$= 60 \text{ X } 8500 \text{ X } 15000$

$= 7.65 \times 10^3 \text{rev}$

Basic dynamic equivalent radial load

$W = XVW_R + YW_A$

To determine,

Radial load factor (X)

Axial load Factor (Y)

We require $\frac{W_A}{W_R}$ and $\frac{W_A}{C_O}$

C_o is not known – Basic static load capacity

$\therefore$ let us take $\frac{W_A}{C_O} = 0.5$ $\qquad \frac{W_A}{W_R} = \frac{1176}{83} = 1.33$

From table

e=0.44- Most greater value

as $\frac{W_A}{W_R} > e \quad 1.33 > 0.44$

X=0.56, Y=1.0

Since rational factor (V) for bearing is 1

Basic dynamic equivalent load radial

$W = 0.56 \times 1 \times 883 + 1 \times 1176$

$= 1670.48$

$= 1671$ N

Service factor for Ball Bearing (KS) is 1

Table 9 Load factor for bearing

Type of bearing	Specifications	$\frac{W_A}{W_R} \le e$		$\frac{W_A}{W_R} > e$		e
		X	Y	X	Y	
Deep groove ball bearing	$\frac{W_A}{C_0} = 0.025$				2.0	0.22
	= 0.04				1.8	0.24
	= 0.07				1.6	0.27
	= 0.13	1	0	0.56	1.4	0.31
	= 0.25				1.2	0.37
	= 0.50				1.0	0.44
Angular contact ball bearings	Single row		0	0.35	0.57	1.14
	Two rows in tandem		0	0.35	0.57	1.14
	Two rows back to back	1	0.55	0.57	0.93	1.14
	Double row		0.73	0.62	1.17	0.86

Basic dynamic load rating

$$C = W\left(\frac{L}{10^6}\right)^{1/k}$$

$$= 1671\left(\frac{7.65 \times 10^3}{10^6}\right)^{1/3}$$

$= 32925.33$ N $= 32.92$ KN … k=3 (Ball Bearing)

$C = 32$, FOR 308

$C_O = 22$

$\frac{W_A}{C_O} = \frac{1176}{22000} = 0.0534$

X=0.56 Y=2.0

$W = 0.56 \times 1 \times 883 + 2 \times 1176$

$= 2846.48$

Basic load Rating

$$C = 2847\left(\frac{7.65 \times 10^9}{10^6}\right)^{1/3}$$

$= 56097.21$ N

$= 56.09$ kN

We have 208,308,408

$\therefore$ **We select 208**

$\mathbf{I_D = 40\ ,\ O_D = 80}$

Table 10 Dynamic load coefficient of bearing

(1)	Static (C_0) (2)	Dynamic (C) (3)	Static (C_0) (4)	Dynamic (C) (5)	Static (C_0) (6)	Dynamic (C) (7)	Static (C_0) (8)	Dynamic (C) (9)
200	2.24	4	—	—	4.55	7.35	1.80	5.70
300	3.60	6.3	—	—	—	—	—	—
201	3	5.4	—	—	5.6	8.3	2.0	5.85
301	4.3	7.65	—	—	—	—	3.0	9.15
202	3.55	6.10	3.75	6.30	5.6	8.3	2.16	6
302	5.20	8.80	—	—	9.3	14	3.35	9.3
203	4.4	7.5	4.75	7.8	8.15	11.6	2.8	7.65
303	6.3	10.6	7.2	11.6	12.9	19.3	4.15	11.2
403	11	18	—	—	—	—	—	—
204	6.55	10	6.55	10.4	11	16	3.9	9.8
304	7.65	12.5	8.3	13.7	14	19.3	5.5	14
404	15.6	24	—	—	—	—	—	—
205	7.1	11	7.8	11.6	13.7	17.3	4.25	9.8
305	10.4	16.6	12.5	19.3	20	26.5	7.65	19
405	19	28	—	—	—	—	—	—
206	10	15.3	11.2	16	20.4	25	5.6	12
306	14.6	22	17	24.5	27.5	35.5	10.2	24.5
406	23.2	33.5	—	—	—	—	—	—
207	13.7	20	15.3	21.2	28	34	8	17
307	17.6	26	20.4	28.5	36	45	13.2	30.5
407	30.5	43	—	—	—	—	—	—
208	16	22.8	19	25	32.5	39	9.15	17.6
308	22	32	25.5	35.5	45.5	55	16	35.5
408	37.5	50	—	—	—	—	—	—
209	18.3	25.5	21.6	28	37.5	41.5	10.2	18
309	30	41.5	34	45.5	56	67	19.6	42.5

8. Result

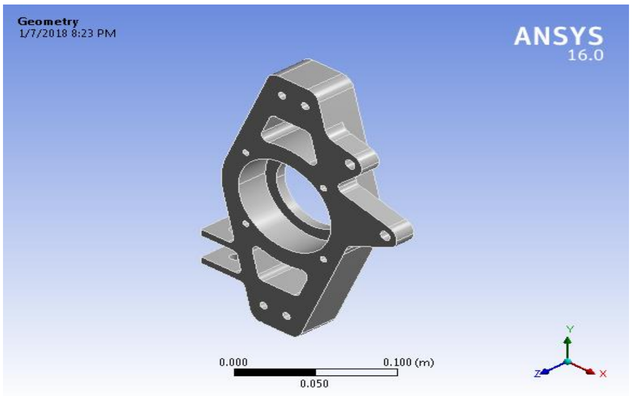

Fig 3 Geometry of front upright

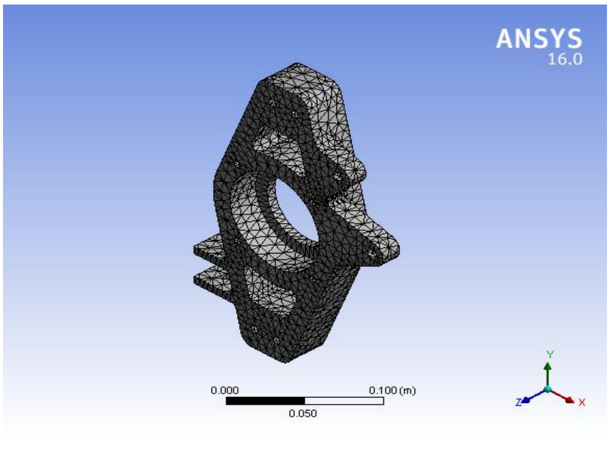

Fig.39 Meshing of Front Upright

8.1 Longitudinal force on upright

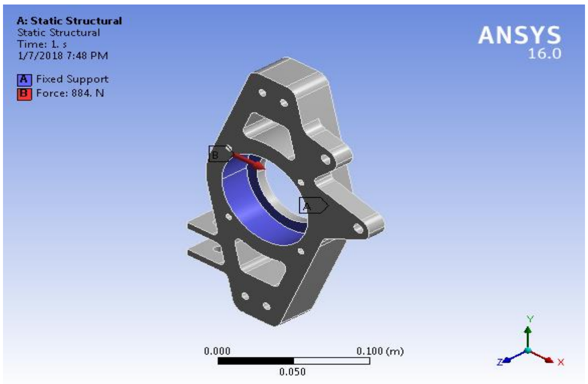

Fig 40 Longitudinal -fix support and force

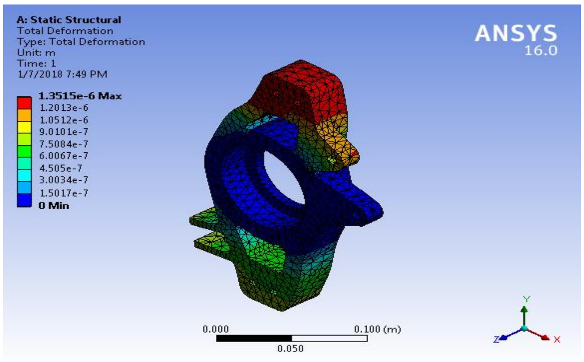

Fig 41 Longitudinal - Total deformation

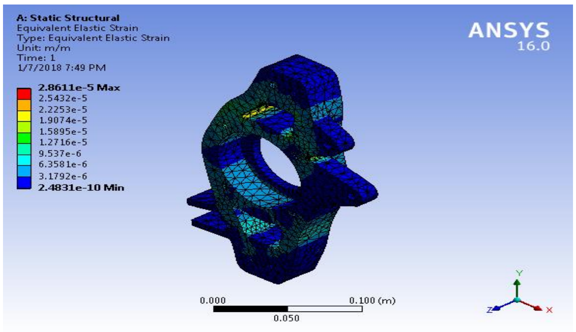

Fig 42 Longitudinal **-** Equivalent elastic strain

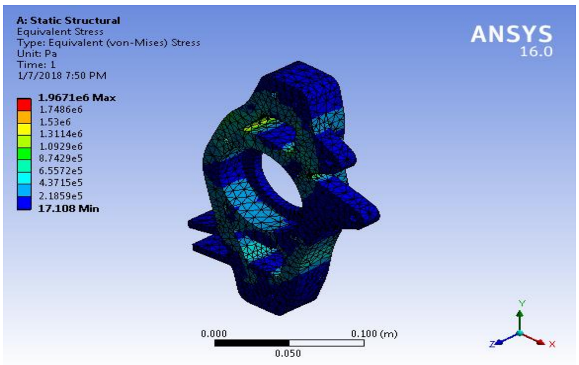

Fig 43 Longitudinal - Equivalent stress

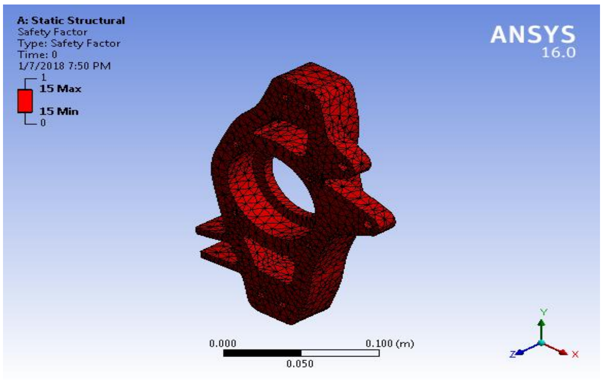

Fig 44 Longitudinal – Safety factor

8.2 Lateral force on upright

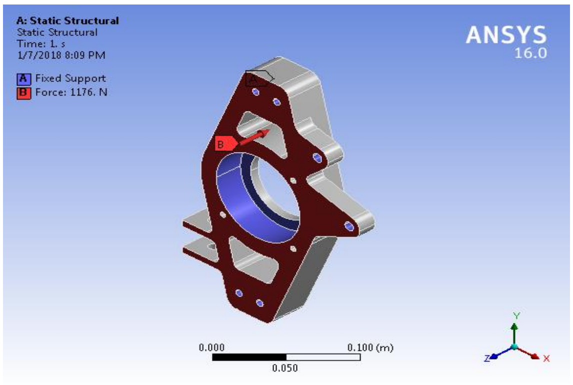

Fig 45 Lateral – fix support and force

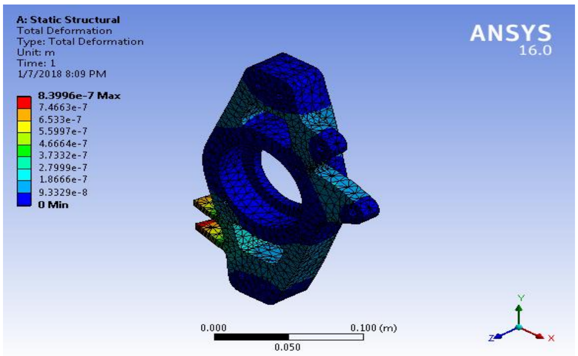

Fig 46 Lateral - Total deformation

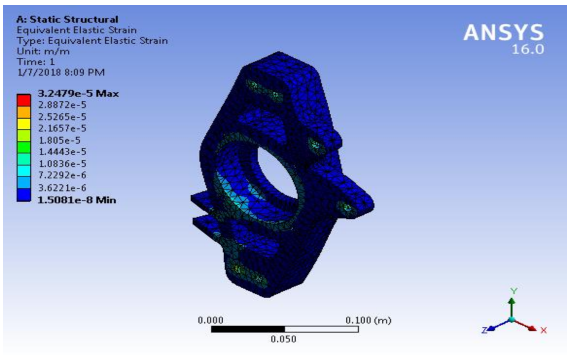

Fig 47 Lateral - Equivalent elastic strain

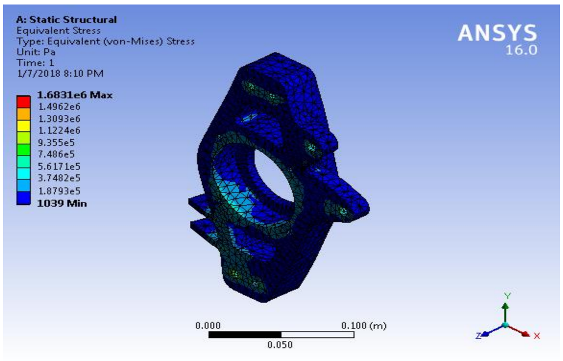

Fig 48 Lateral - Equivalent stress

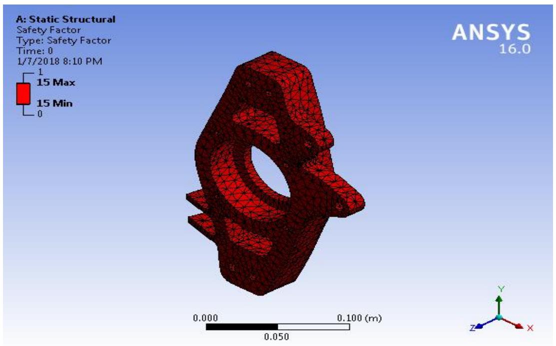

Fig 49 Lateral - Safety factor

8.3 Centrifugal force on upright

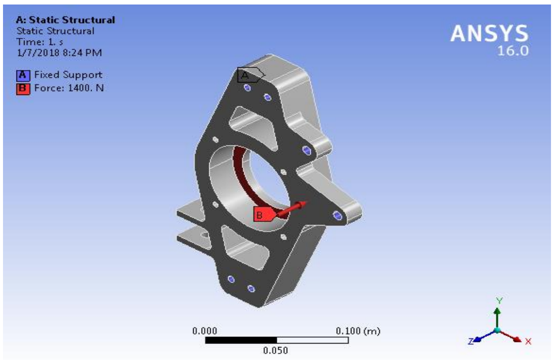

Fig 50 centrifugal – fix support and force

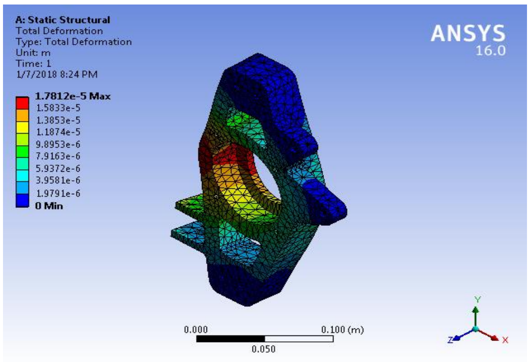

Fig 51 Centrifugal - Total deformation

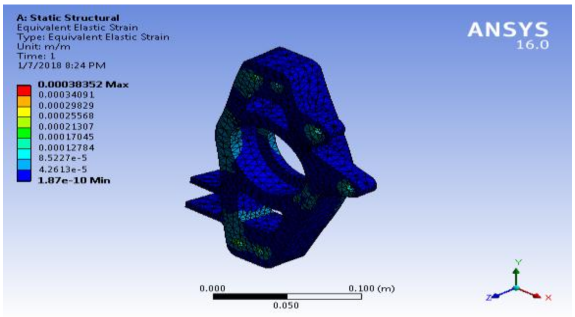

Fig 52 Centrifugal Equivalent elastic strain

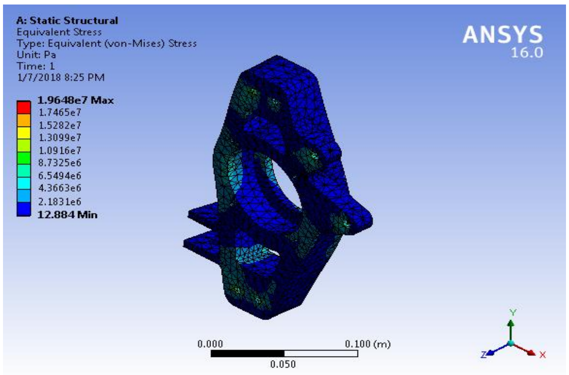

Fig 53 Centrifugal - Equivalent stress

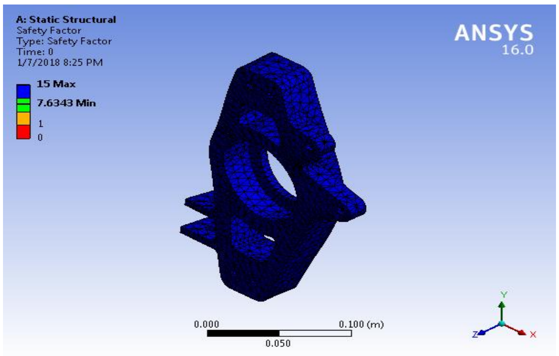

Fig 54 Centrifugal - Safety factor

8.4 Steering force on steering point

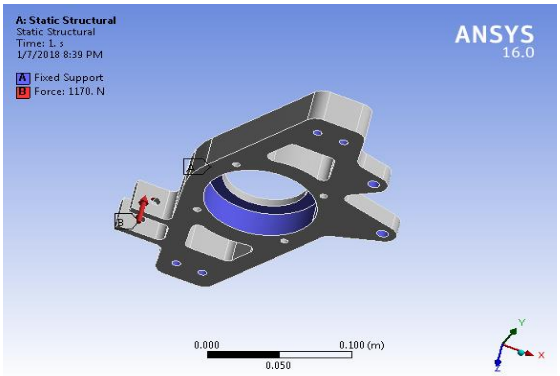

Fig 55 Steering force – fix support and force

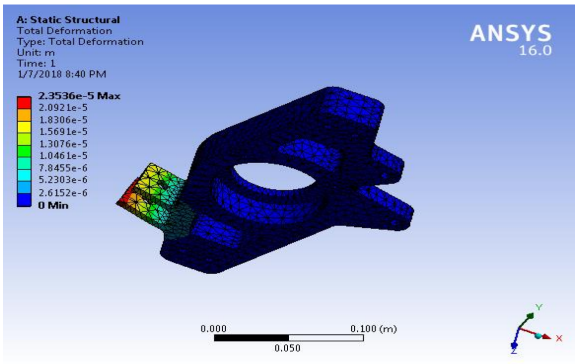

Fig 56 Steering force - Total deformation

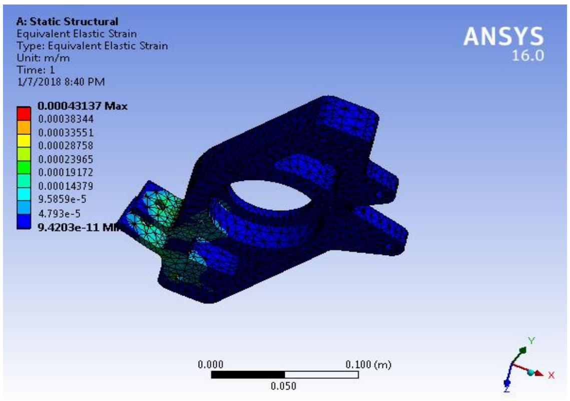

Fig 57 Steering force - Equivalent elastic strain

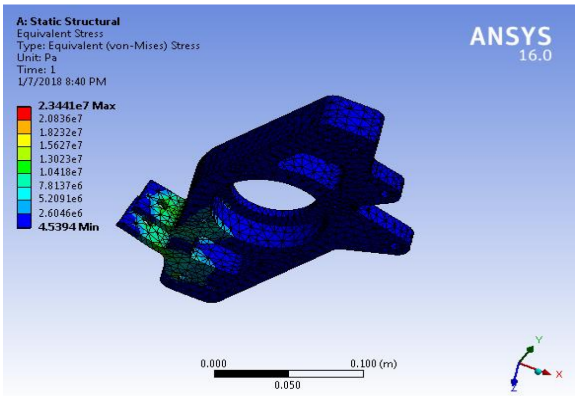

Fig 58 Steering force - Equivalent stress

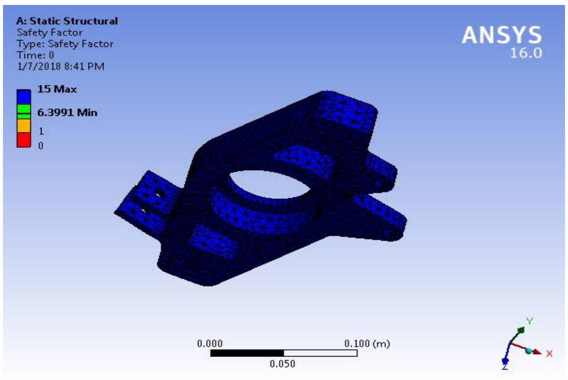

Fig 59 Steering force - Safety factor

8.5 Force on caliper mounting

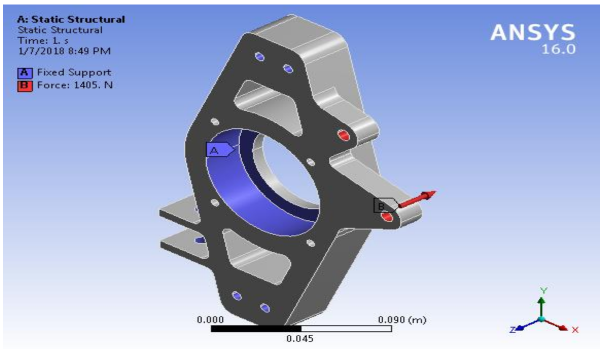

Fig 60 Force on caliper mounting – fix support and force

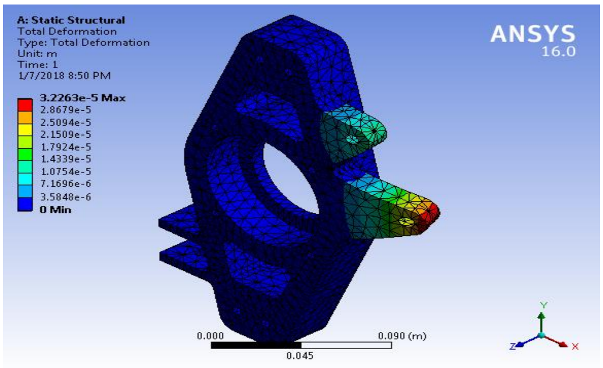

Fig 61 Force on caliper mounting - Total deformation

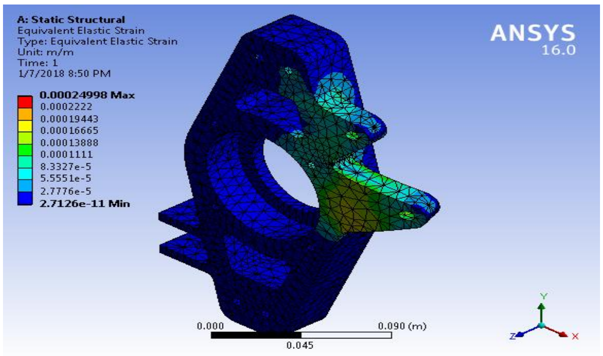

Fig 62 Force on caliper mounting - Equivalent elastic strain

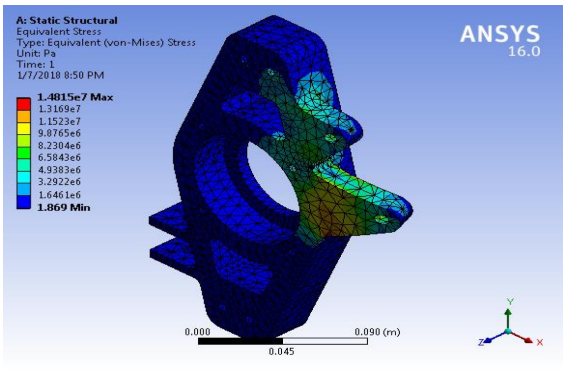

Fig 63 Force on caliper mounting - Equivalent stress

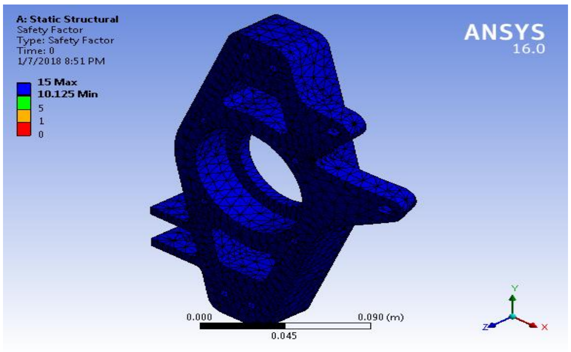

Fig 64 Force on caliper mounting - Safety factor

8.6 Torque on caliper mounting

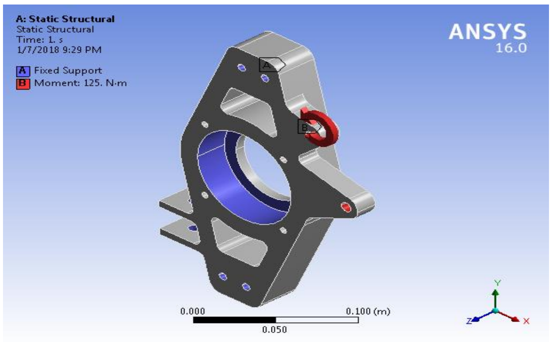

Fig 65 Torque on caliper mounting – fix support and Torque

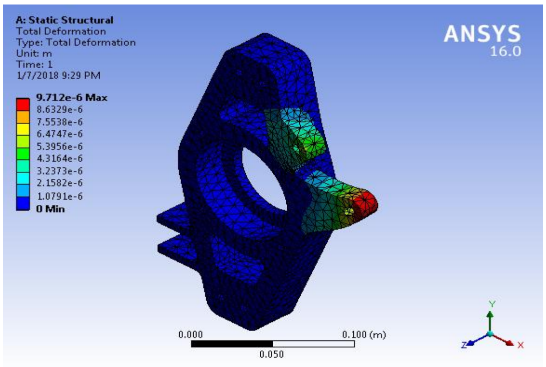

Fig 66 Torque on caliper mounting - Total deformation

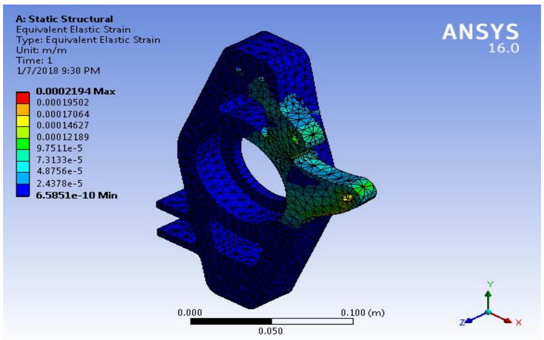

Fig 67 Torque on caliper mounting - Equivalent elastic strain

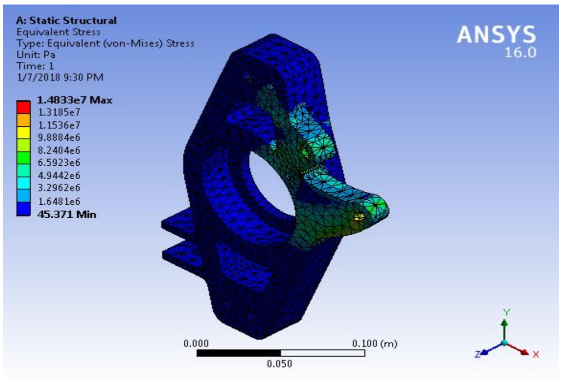

Fig 68 Torque on caliper mounting - Equivalent stress

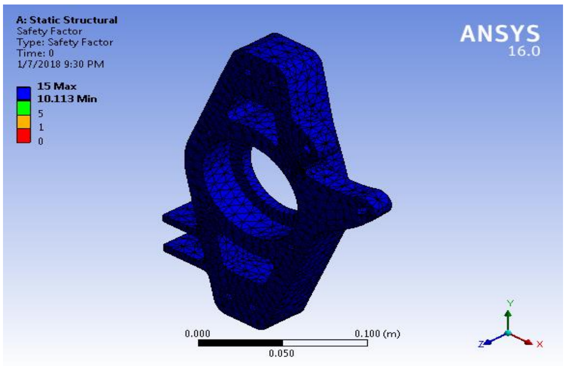

Fig 69 Torque on caliper mounting - Safety factor

8.7 Torque on hub petal

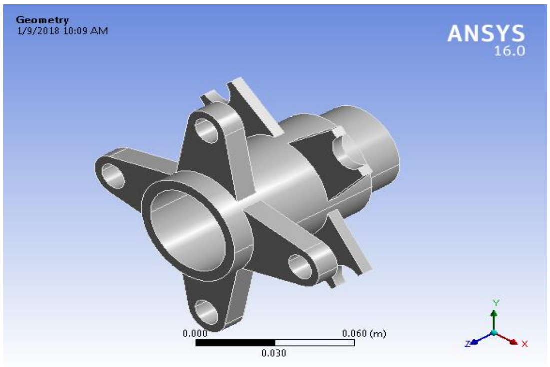

Fig 70 Combine torque on hub – geometry

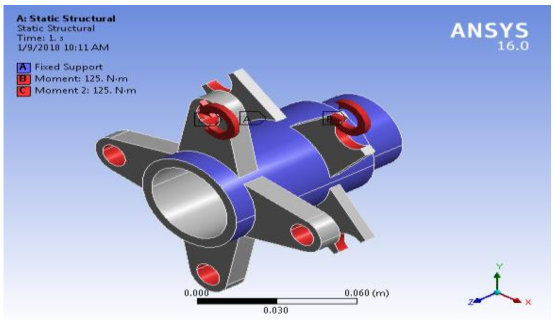

Fig 71 Combine torque on hub – fix support and torque

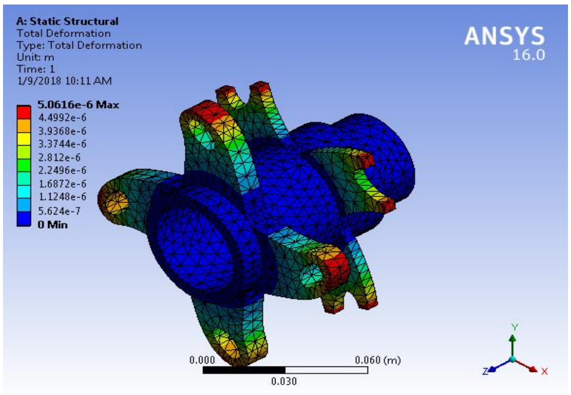

Fig 72 Combine torque on hub - Total deformation

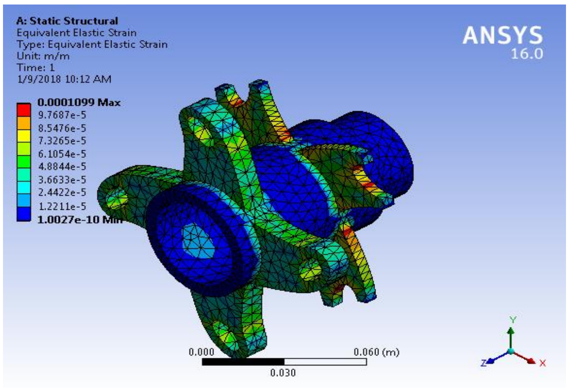

Fig 73 combine torque on hub - Equivalent elastic strain

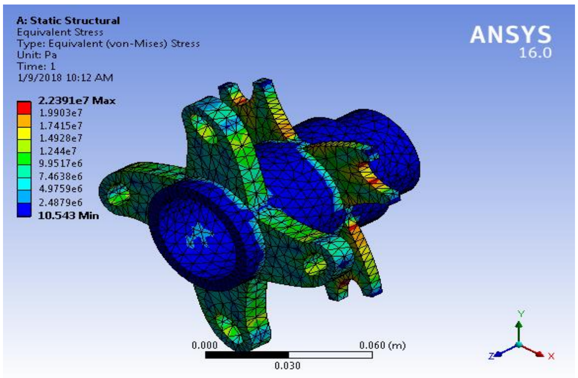

Fig 74 Combine torque on hub - Equivalent stress

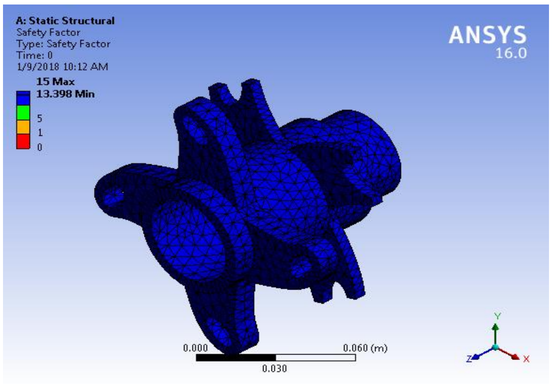

Fig 75 Combine torque on hub - Safety factor

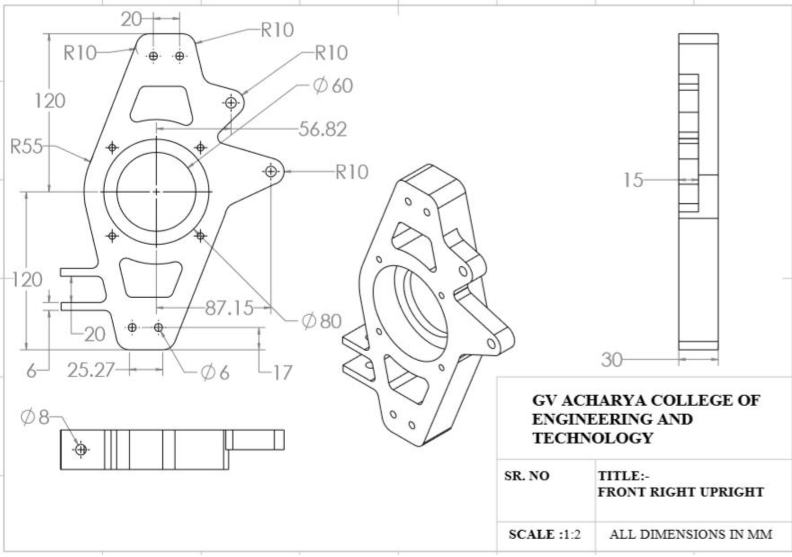

Fig. 76 Front Right upright CAD Drawing

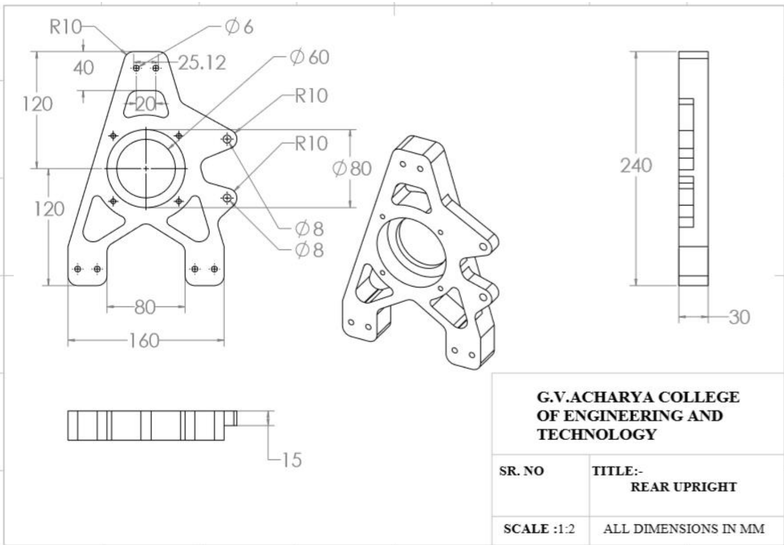

Fig. 77 Rear Right upright CAD Drawing

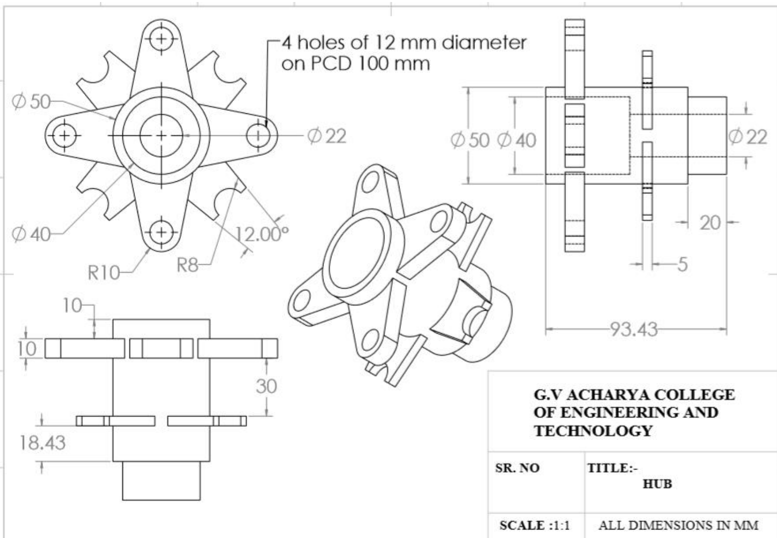

Fig. 78 Hub CAD drawing

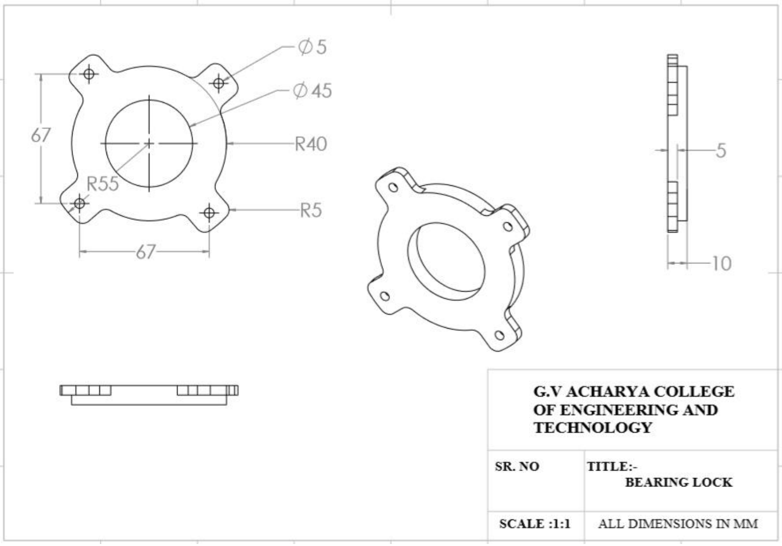

Fig. 79 Bearing lock CAD drawing

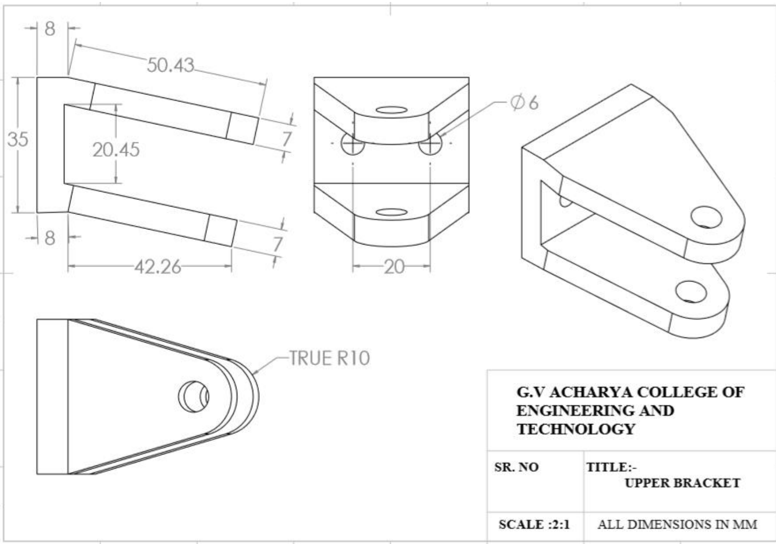

Fig. 80 Upper Bracket CAD Drawing

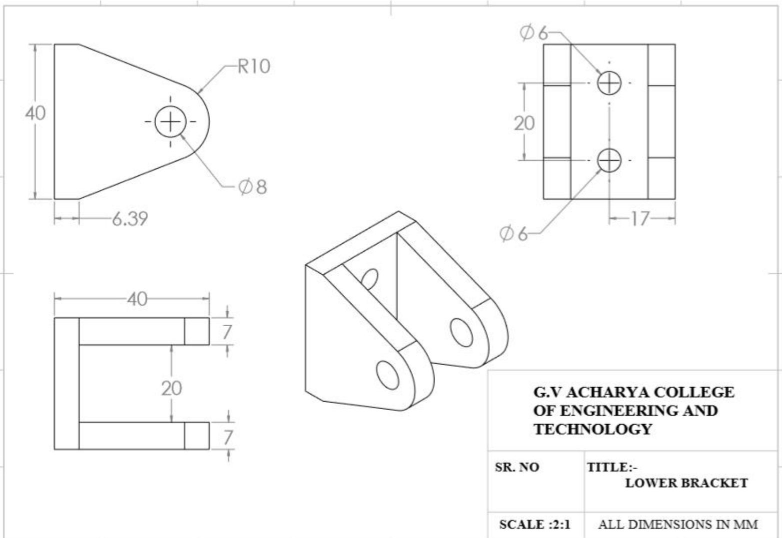

Fig 81 Lower Bracket CAD Drawing

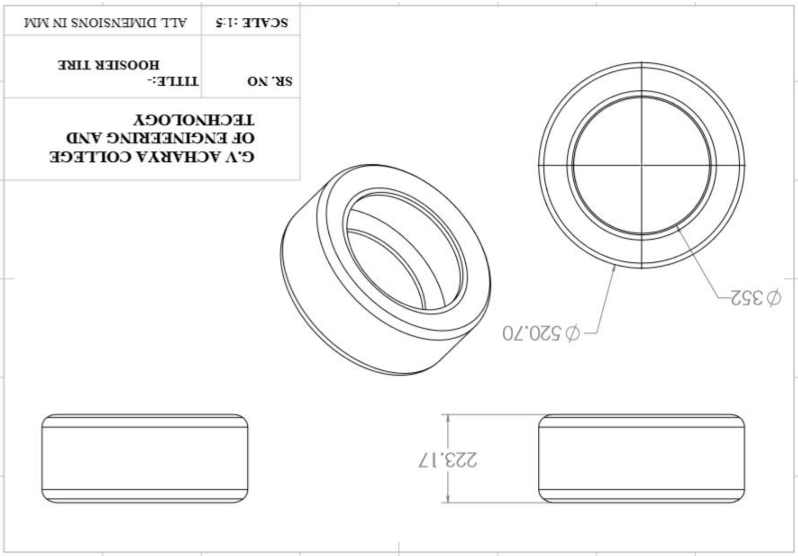

Fig. 82 Tire CAD Drawing

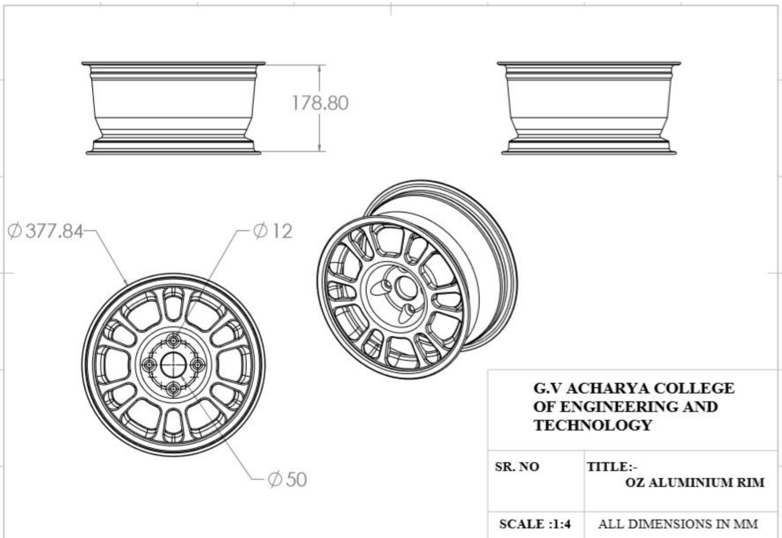

Fig. 83 Rim CAD Drawing

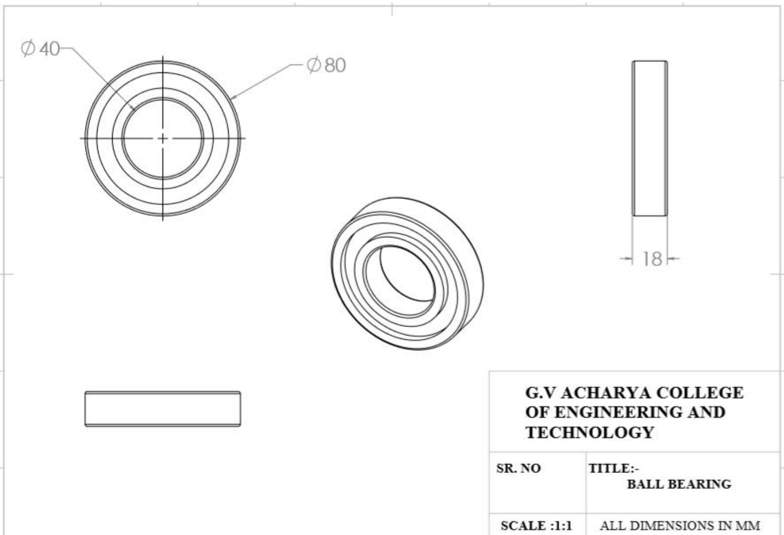

Fig. 84 Ball bearing CAD Drawing

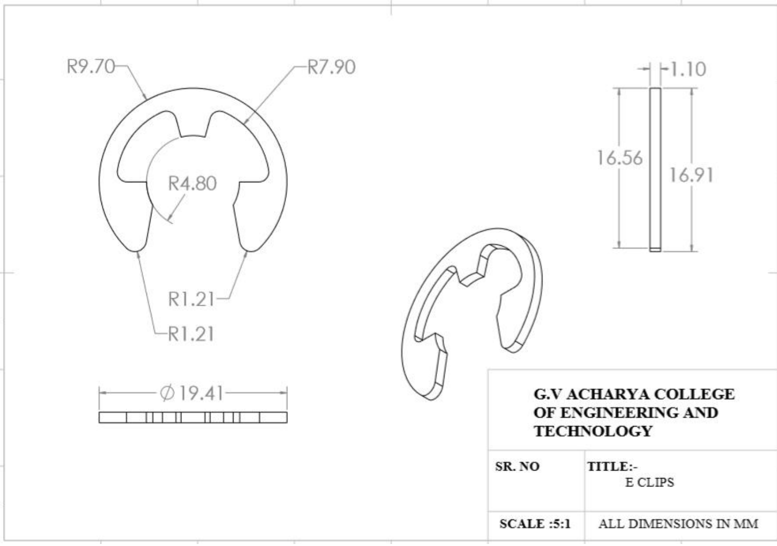

Fig. 85 E-Clip CAD Drawing

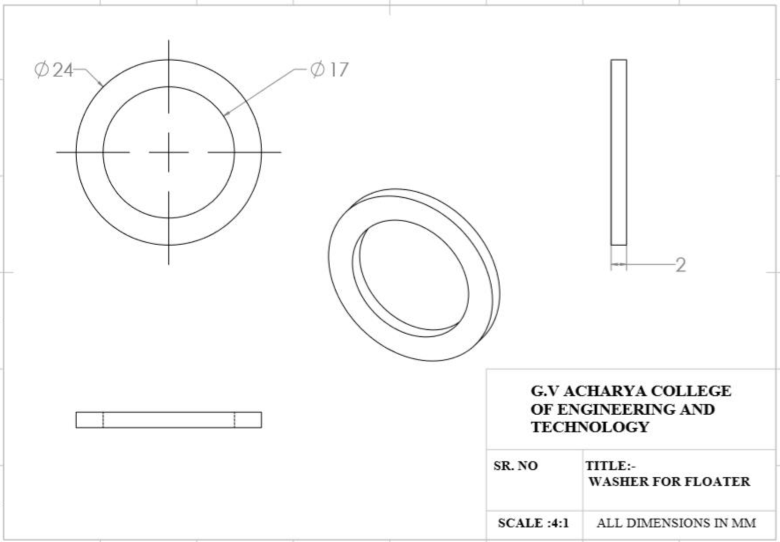

Fig.86 Floater Washer CAD Drawing

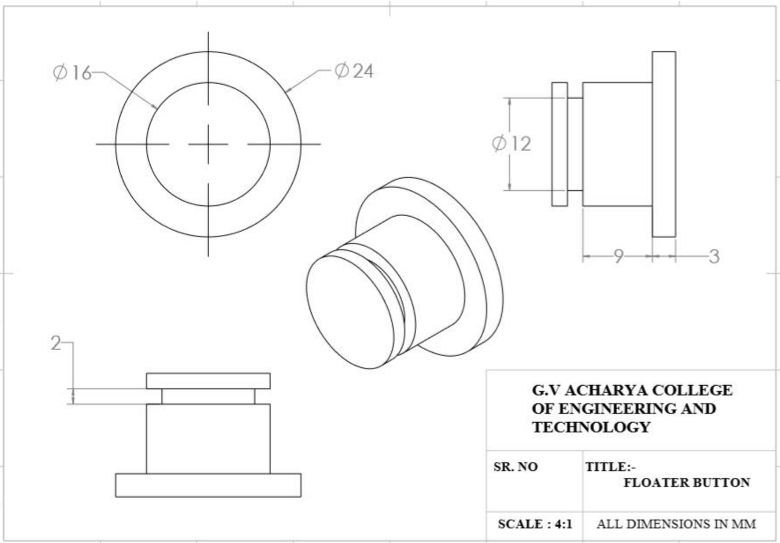

Fig.87 Floater Buttons CAD Drawing

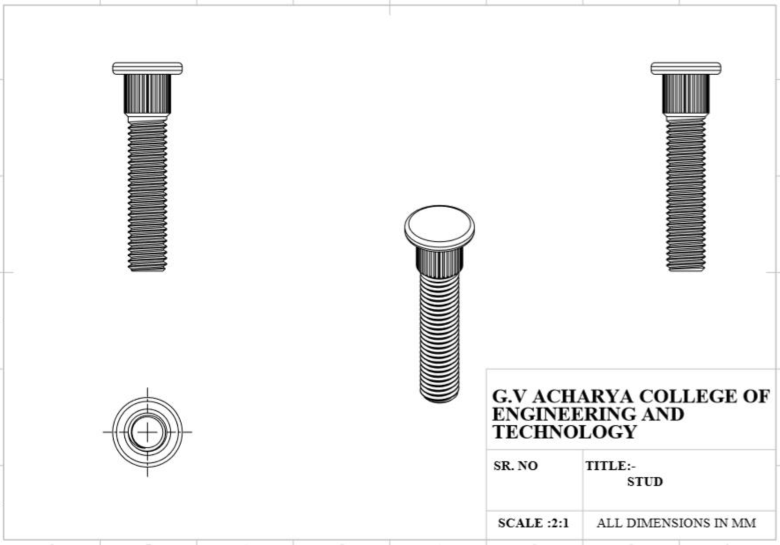

Fig.88 Stud CAD Drawing

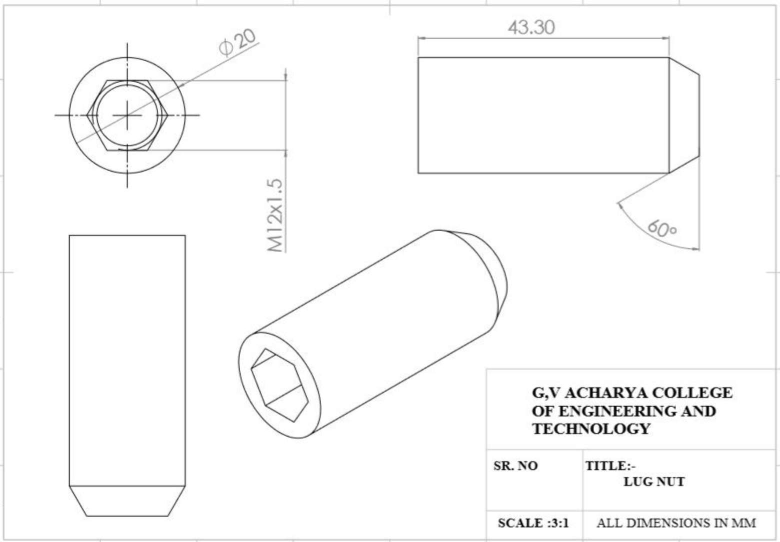

Fig. 89 Lug Nut CAD Drawing

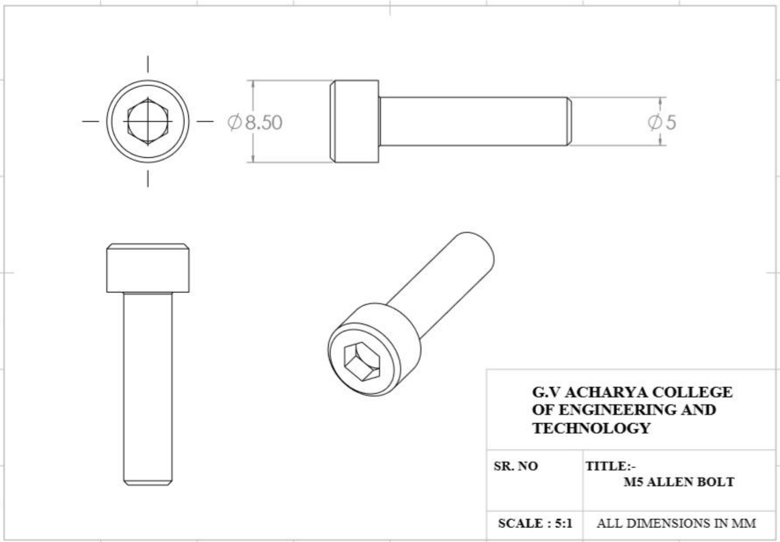

Fig. 90 Allen Bolt CAD Drawing

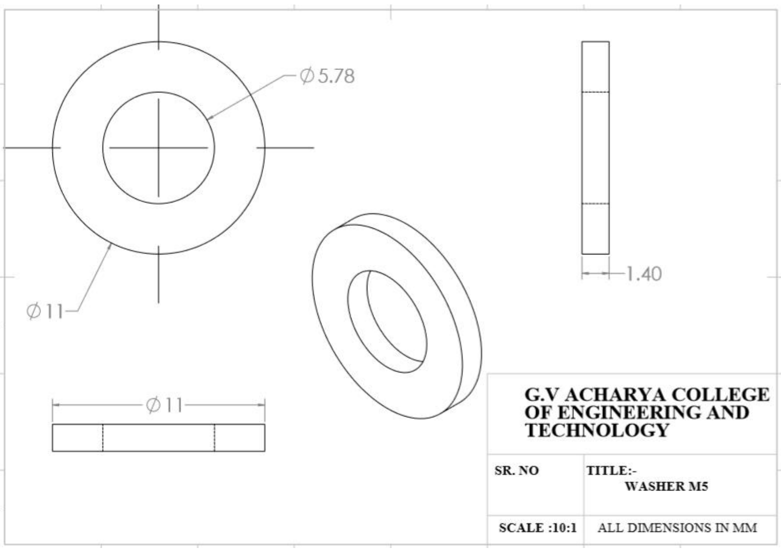

Fig. 91 Washer CAD Drawing

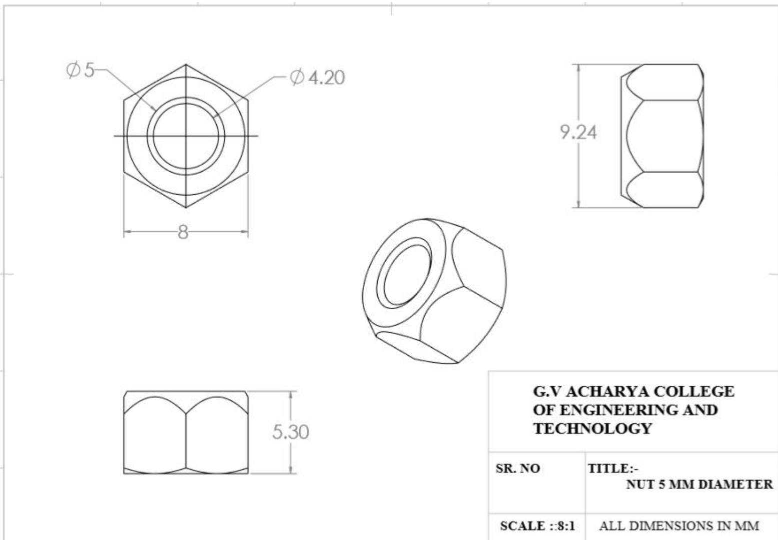

Fig. 92 Nut CAD Drawing

Future Work

This report covers a broad range of parts with different approaches. Some parts have more complex alternatives that would need further inspection and analysis. Implementation was weighted against the attainable results due to the time constraints imposed.
• One of the biggest obstacles was the central locking system. A further analysis of the central locking system is needed that weren't implemented in this thesis. Manufacture ideas as well as minor tweaks to the system will be further developed in the following months.

• A part of participating in Formula Student is to bring a fully produced prototype. In order to bring a functional prototype, the vehicle needs to be manufactured. With that being said, it is a high priority to purchase the parts and material needed for building every part described in this thesis.

• A big impact in this thesis was the involvement of the brake system. The brake system is one of key safety and handling feature in any race car. In this thesis the coverage of the brake system was only a scratch on the surface and further development to the hydraulics of the overall system will need further analysis. From the caliper itself to the brake pedal of the driver.
• Some thoughts for future development: integrating each part together, conduct analysis of implementing the brake caliper into the upright itself and check the advantages of using a bolted housed wheel bearing between the hubs and uprights

Conclusion

The purpose of this project is not only to design and manufacture the upright assemblies for the car, but also to provide an in-depth study in the process taken to arrive at the final design. With the overall design being carefully considered beforehand, the manufacturing process being controlled closely, and that many design features have been proven effective within the performance requirement of the vehicle. The FEA result indicates that the upright assembly is able to perform safely in real track condition as per performance requirement.

References

P. Vishwakarma, M. Kanungoo, (2014), Finite Element Analysis of Chervolet Front Hub with The Help of Inventer, International Journal in IT and Engineering, vol.2, Issue 2, ISSN: 2321-1776.

G. Fisher,V. V. Graubisic, (1998), Design consideration and durability approval of wheel hub, SAE international,11-16-1998.

S.dhar, (1988), Fracturer analysis of wheel hub fabricated from pressure die aluminium assembly, Theoretical and applied fracturer mechanics, vol 09- 02-1988.

S. Li, Z. Lindu, (2009), Research on Optimization of Hub-And-Spoke Logistics Network Design with Impedance Effect, POMS 20th Annual Conference Orlando, Florida, U.S.A, 011-0370.

V.B. Bhandari, (1999), Design of Machine Elements, 8th edition, Wiley.

Kutz M., (1986), Mechanical engineering handbook, New York, Wiley

Alejandro Diaz, Osvaldo Fernandez, Ricardo Gonzalez and Christian Ramos, November 24, 2014, “FSAE 2015 Chassis and Suspension Final Report”, EML 4905, pp – 3

Abhijeet Das, October 2014 - March 2015, “Virtual Prototype of Upright Assembly of A Race Car for SUPRA SAEINDIA” Competition, ISSN 2348-7593, pp – 79 and 80.

Josh Carrol, Lloyd Outten, Joseph Perry, Taylor Watkins, “FormulaSAE Team Final Report”, Old Dominion University, April 2013

“Design and Analysis of Hub and Knuckle of FSAE Race Car” by Mr. Sangram K. Pisat, Mr. Aditya S. Phule, Mr. Rohan R. Shinde, Prof. Arun V. Javir Department of Mechanical Engineering, Rajendra Mane College of Engineering and Technology, Ambav, Mumbai University, 01/05/2016, pp – 764 to 766

International Journal of Mechanical Engineering and Technology (IJMET) Volume 7, Issue 2, March-April 2016, pp. 168–176, Article ID: IJMET_07_02_018, “Force Calculation in Upright of a Fsae Race Car” Anshul Dhakar and Rishav Ranjan Department of Mechanical Engineering, RV College of Engineering, Bangalore, India

“Sesign and optimization of a formula sae racecar” by William Davis. Billygolfer Faculty of worcester polytechnic institute, Project Number: MQP-DCP2012-A11-D12