Research Article

Design, Optimization and Manufacturing of Wheel Assembly System of Formula Society of Automotive Engineers (FSAE) Car

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Abstract

A wheel assembly system joins the chassis and the wheels through the suspension system. Any failure in the Wheel assembly is catastrophic to human life, which is why it is vital to develop a safe design. The primary purpose of this project was to design, optimize, and manufacture the wheel assembly system, focused primarily on improving the previous design and reducing the un-sprung weight of the vehicle. The design was to be analyzed and implemented, taking into consideration the various forces acting on the entire wheel assembly system under different conditions such as braking, acceleration, and cornering. Designing and optimization were carried out by using SolidWorks. This paper illustrates in its entirety all the processes, including design, optimization, and manufacturing undertaken by us.

Keywords: Uprights, Clevis, Hub, Formula SAE, Wheel Assembly, Un-sprung, Brake caliper

1. Introduction

The primary goal while building a race car is to achieve the best performance to weight ratio. The reduction of weight in any area will lead to an overall increase in the performance of the car. From Newtonian Physics, we know force = mass*acceleration, thus by reducing mass, with a given amount of force capable of being exerted from the vehicle, acceleration can be maximized. Also, in order to exploit the finite amount of available grip and achieve the primary aim of maximum acceleration from the tire it is essential to reduce the vehicle weight. In conclusion, weight is inevitably a key constraint in designing any component in the race-car. The two types of weight in an automobile are sprung and un-sprung weights. Sprung mass can be defined as the mass of the vehicle that is damped by the spring. As the spring does not damp the wheel assembly mass, it comes under the un-sprung mass category. Essentially un-sprung mass is the mass that is not supported by the shock-absorbers like the hub, wheel and uprights. It is crucial to reduce un-sprung mass in order to increase acceleration. The higher the un-sprung mass, slower the acceleration. The goal is to produce a lighter and performance-oriented design of upright assembly in comparison with that of season 2016-2017 car and thereby contributing to making the car of season 2018-2019 better than its predecessor. Use of a conventional

upright assembly will increase the overall weight of the race car. So, for this season, it was decided to use live spindle assembly to reduce the components. The goal of a lighter upright assembly is achieved by less sophisticated design and proper material selection. Also, proper stiffness and reliability can be attained by accurate analysis of the design of the upright assembly. Some of the advantages of live spindle are given below:

- Wheel assembly gets stiffer, so there is no camber change.
- Easy installation
- Weight reduction
- Modularity: Breaking of the system in smaller parts
- The brake disc is in the inboard side of upright which reduces space between the uprights and wheel center hence reducing bending load.
- CV housing is completely inside the uprights, which reduces space between the uprights which in turn reduces driveshaft angle.
- Compact component packing
- Higher stiffness to system mass which results in fewer vibrations.

The objectives of the project are as follows:

1) Lightweight to maintain excellent performance to weight ratio of the race car.
2) Optimum stiffness to ensure the reliability of upright assembly and to maintain designed geometry of suspension system.

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3) Ease of maintenance for enhancing serviceability and setup repeatability.

The front upright connects the steering arm, which allows the driver to change the direction of the vehicle. There lies a bearing between the hub and the upright which allows the upright to be stationary relative to the chassis. The hub rotates with the wheel by the power transmitted from the transmission assembly. Typically, the hub and the bearing is press-fitted inside the upright. In addition, the caliper is also supported by the upright. Any failure in the wheel assembly can result in severe consequences for the driver. Thus, the wheel assembly system must be designed and analyzed carefully.

2. Methodology

- In the research phase, we first understood basic parameters like camber, caster, kingpin inclination and toe angle.
- Designs of various other teams and the various values used by them were evaluated.
- At the same time we began learning softwares such as Solidworks and Ansys which helped us complete the design phase with ease.
- The design phase was completed along with some calculations which would help us keep the overall stresses low. These calculations included various load transfers in dynamic conditions keeping in mind various FSAE rules.
- All of the design was later completed on Solidworks and the simulations were done.
- During the manufacturing phase, various materials were taken into consideration along with the market survey for availability and cost of the materials.
- Taking all of these factors into consideration a suitable material was chosen.
- Once the design was completed, then we studied the various manufacturing process available and other components such as nut, bolts, bearings and other essential items.
- Upright, hub and brackets were manufactured with the help of CNC turning, VMC and WATER JET machining to get the necessary accuracy.

3. Design and Analysis

3.1 Upright Design

Upright is an integral part of the wheel assembly which is press-fitted on the hub and on which the A-arms are mounted. Besides, the upright also serves the function of providing mounting to the brake caliper. The Steering Arm which connects the wheel assembly and the tie rod is also mounted on the upright. Thus due to all these mountings, many forces act on the upright. Upright is also subjected to completely reversed types of stresses while turning and also during braking and

acceleration. Thus a brittle material is not suitable for this application. Consequently, it was decided to take a tensile material called Aluminium-6061 T6. It has a high strength to weight ratio. Thus with a much lower weight, one could produce sturdy uprights.

The Endurance limit of this material is much more than that of other aluminum series. The material properties are as follows:

- S_{yt} = 276MPa
- Endurance Limit =96.5MPa
- Density = 2700 kg/m3

First, the caster angle was selected. It was decided to have a 7-degree caster angle. Then, the proper length of the upright was selected, so the suspension points lie within it. Subsequently, the central boring size was decided which is dependent on the bearing outside diameter as bearing has to be press-fit inside that bore. The boring thickness depends on bearing thickness, and a step was provided inside to restrict the movement of the bearing. Two bearings have been press-fitted on the opposite side of the step. Internal circlips have been added to prevent movement of the bearings in the axial direction. Next caliper position was mounted according to the dimensions received from the brakes department. Then the steering point was mounted according to points given by the steering department. Proper fillet and weight reduction were given.
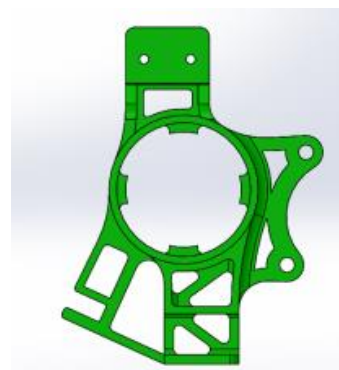

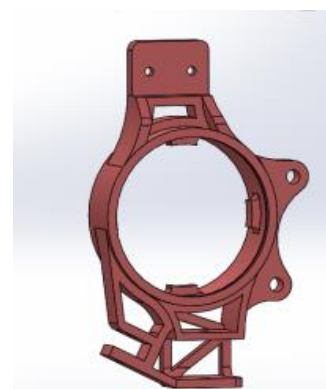**Fig 01** – Front Upright in SolidWorks

Fig 02 – Rear Upright in SolidWorks

3.2 Clevis (Camber shims)

Clevis is the part which joins the upright to the A-Arms and is also useful to vary the camber angle. These were manufactured from the same material as the upright that is Aluminium-6061 T6 due to its various advantages as mentioned above.
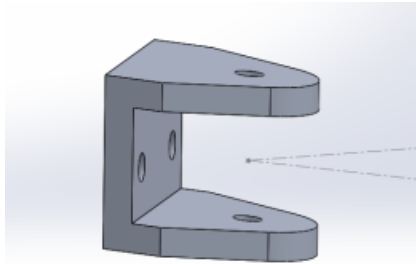

Fig 03 – Clevis in SolidWorks

3.3 – Hub Design

Hub is that part of the wheel assembly on which the wheel and brake 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 inside the hub, so provision is made in it to enclose the bearing. The hub itself is made of two petal parts- one of the wheel and the other of the brake disk. Since it was decided to design a live spindle; hence the length of the hub for the rear was also dependent upon the CV joint.

The four primary forces acting on the hub are:

1) Force due to acceleration or deceleration
2) Cornering
3) Wheel travel or bump
4) Brake torque or torque from the axles

The design of the hub was bounded by various conditions such as track width, bearing size, rotor dimensions, and the bolt pattern on the wheel. First, the boss diameter of the rim was measured which was found to be 63.5 mm. So, in order to avoid wobbling, the hub boss diameter was taken as 63.5 mm. Then according to the PCD and stud diameter of the rim, the first flange of the hub was drawn which attaches to the rim from inside.

For weight reduction purpose, petal-type shape was decided for the flange. Later for disc mounting, the float for floating disc was designed having PCD 100 mm and according to floater button design and disc design, the second flange of the hub was made. Also, it was essential to keep the disc in the centre of the 2 pads of calliper, so accordingly a step was made. Furthermore, the last step of 40 mm diameter, which is according to bearing inner race for bearing mounting on the hub was made. Then hole from bearing side was made for

insertion of the tripod. Subsequently, slotting is done according to the tripod spline in the hub. Also, proper fillet was put in to avoid stress concentration.
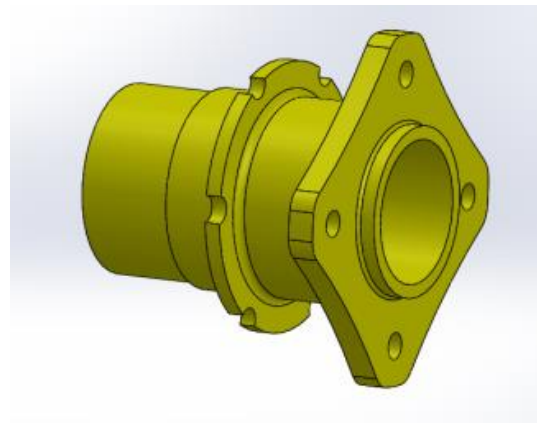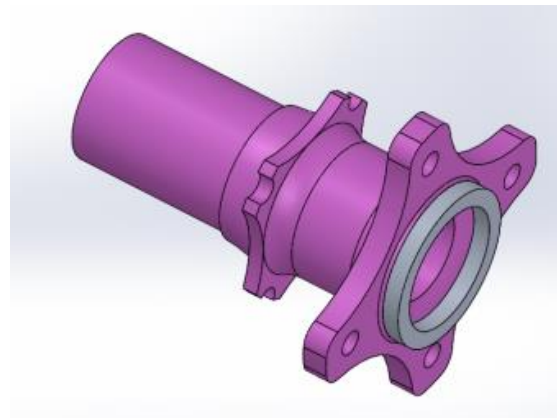

Fig 04 – Rear Hub in SolidWorks

Fig 05 - Front Hub in SolidWorks

3.4 Bearing Calculations

In order to ensure free rotation of the hub, it was necessary to select an appropriate bearing by conducting the necessary research. Thus it was decided to use deep groove ball bearings between the upright and the hub due to the following reasons:

- They can carry radial and axial loads.
- They create less friction torque which lowers operating temperature and extends the life of the bearing.
- 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.

Given ahead are the calculations for bearings and the forces acting on them.

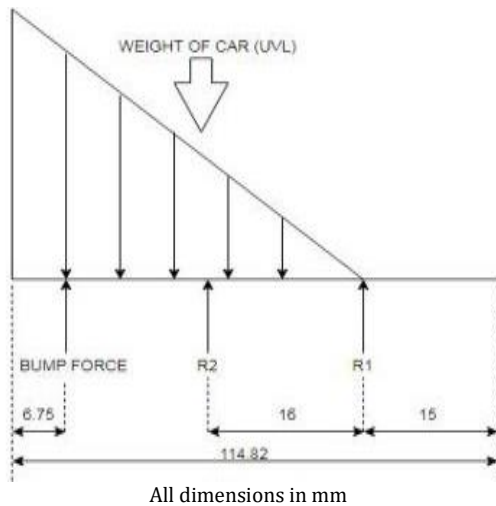Forces Acting on Bearings in Upright

Rear Bearing Calculations

All dimensions in mm

Fig 06 - F.B.D. of rear bearings

Forces acting due to uniform varying load
=05*(250*9.81) *(141.07-37)/141.07
=904.62…. (I)
Also
R1 + R2 + 2538=904.62
R1 + R2= -1633.38… (II)
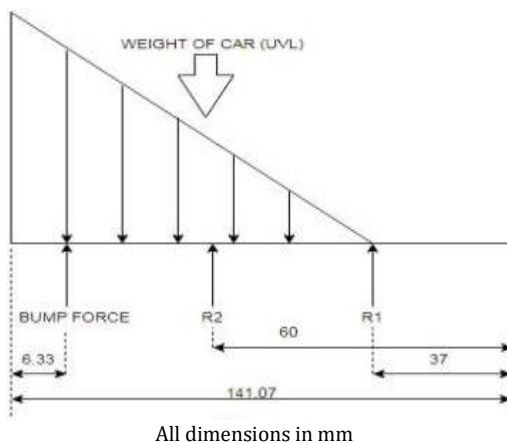R1=-9192.39 N
R2=7569.01 N

Front Bearing Calculations

All dimensions in mm

Fig 07 - F.B.D. of front bearings

R1 + R2 + 2538=265.62
R1 + R2 = -2272.38N…. (I)

Also

88.82*R1 + 103.8*R2 = -2538*6.73+265.62*34.6
R1=-15201 N
R2=1292.5 N

Based on above calculations bearing numbers (61910-2RS) for front and (61814-2RS) for rear were selected.

Checking if the selected bearings are safe or not:

P = 4000N = 4kN
L = 1000 hrs
N = 408 rpm
{Engine Power=2πNT/60000
For KTM 390, Max power = 43 HP at 9000rpm
Therefore,
32 = 2*π*(N)*750/60000
N=408r rpm}
Step 1: Life in mR
Life in mR = Lh*60*N/10^6
=1000*60*408/10^
=24.5mR

Step 2: Selecting and checking bearing from standard catalogue

For SKF bearing 61910-2RS

Outer diameter =72mm
Inner diameter =50mm
Dynamic capacity=14.5kN
Checking for capacity
C=(Life mR)^1/k*P
{k=3 for DGBB}
= (24.5)^1/3* 4000
= 11.53kN
C = 11.53 kN < 14.5kN

Therefore, Bearing is safe as design dynamic capacity on the bearing is less than the available max dynamic capacity of the bearing.

For SKF bearing 61814-2RS

Outer diameter =90mm
Inner diameter =70mm
Dynamic capacity=12.5kN
Checking for capacity
C=(Life mR)^1/k*P
{k=3 for DGBB}
= (24.5)^1/3* 4000
= 11.53kN
C = 11.53 kN < 12.5kN

Therefore, Bearing is safe as design dynamic capacity on the bearing is less than the available max dynamic capacity of the bearing.

3.5 Upright Analysis

3.5.1 Longitudinal

Forces during Braking

While braking, the weight of the rear side tends to transfer to the front side of the vehicle so there is a load transfer that is taking place from the rear to the

front. It in turn affects the upright 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 0.61 times the total weight
Mass at the rear side of the vehicle=0.6 × 250 =150 kg
Force = 150 × 9.81
Force = 1471.5 N
Now force on 1 wheel =1471.5/2 =735.75 N
Thus Longitudinal Force =735.75 N

3.5.2 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 4.5m turning radius and at a speed of 30kmph
r = turning radius =4.5m
v= 30kmph = =8.3333 m/s
Centrifugal Force =1503.42N
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 =0.39 × 250 × 9.81 =956.4765
Thus Lateral Force = 956.4765 N

3.5.2 Force on the Steering Arm

According to the steering effort, the force on steering arm was found out to be 689N
Force on steering arm = 689 N

3.5.3 SolidWorks analysis of uprights

From Lotus software, the magnitude and directions of the forces on the upper ball joint and lower ball joint were obtained. Accordingly, the forces were applied to get the satisfactory factory of safety. Redundant weight was removed in order to reduce the overall weight of the car which helped in increasing acceleration timing. Various material like Aluminum 6 series and 7 series were analyzed and have selected the one which had given us optimized results even after machining. The analysis of the front and rear uprights using SolidWorks software is given in the figures below.
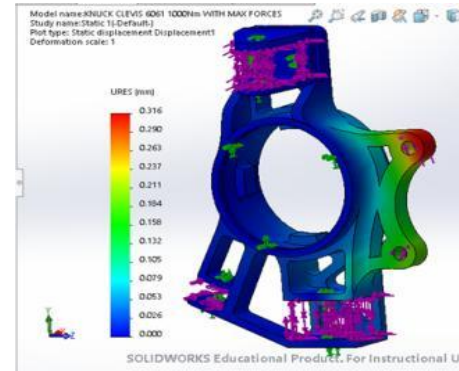

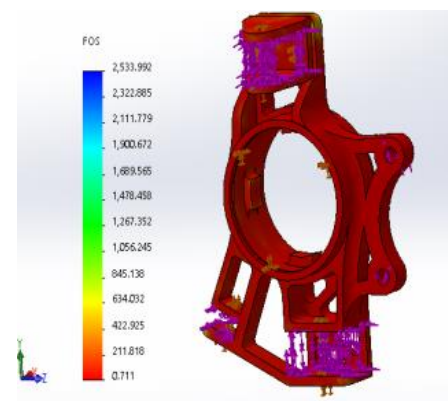**Fig 09 -** Displacement Analysis of front upright
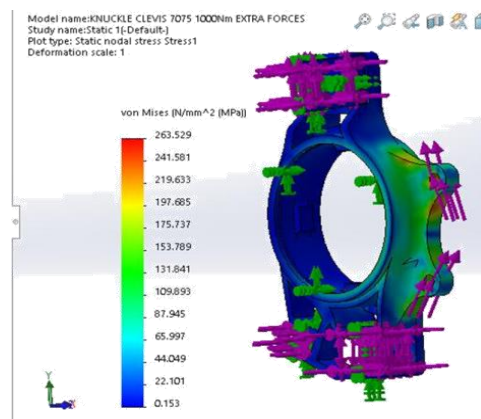

Fig 10 Factor of safety of front upright

Fig 11 Stress analysis of rear upright
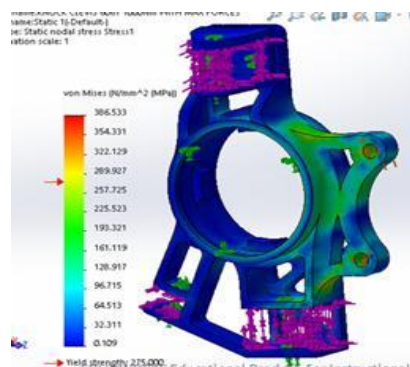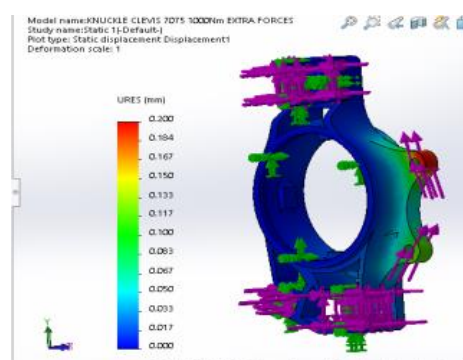

Fig 08 - Stress analysis of front upright

Fig 12 Displacement Analysis of rear upright

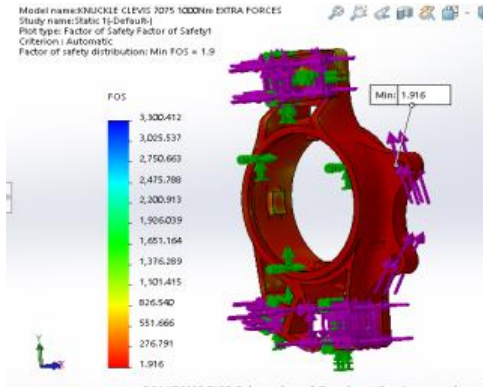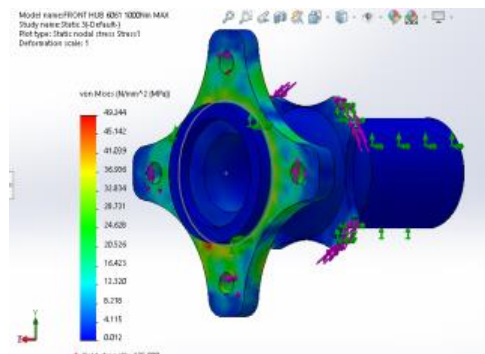**Fig 13 -** Factor of safety of rear upright

3.6 Hub Analysis

3.6.1 Determining the forces acting on the Front Hub: -

The following Forces are acting on the Hub.

a. Torque on the Brake Disk Petal: - 350 Nm
A torque of 350 Nm is acting on the Brake Disk Petal.
The Force acting on each hole = 350 / (4 x 0.04) = 2812.5 N

Thus force acting on each hole of front hub due to brake disc = 2812.5 N

b. 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.
The Force acting on each hole = 350 / (4 x 0.05) = 1750 N
Thus force acting on each hole of front hub due to wheel = 1750 N

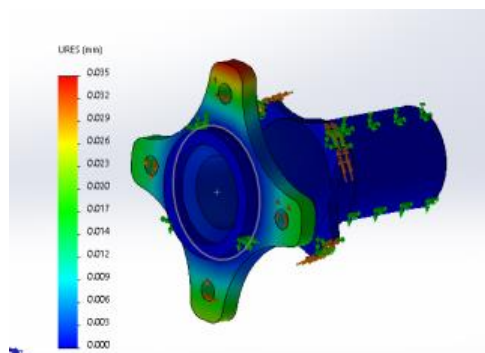*3.6.2 Determining the forces acting on the Rear Hub*

The following Forces are acting on the Hub.

a. Torque on the Rear Brake Disk Petal is taken as that of front: - 350 Nm
A torque of 350 Nm is acting on the Brake Disk Petal.
The Force acting on each hole = 350 / (6 x 0.05) = 1166.667 N
Thus force acting on each hole of rear hub due to brake disc = 1166.667 N

b. 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.
The force acting on each hole = 350 /(4 x 0.05) = 1750 N
Thus force acting on each hole of rear hub due to wheel = 1750 N

Fig 14 - Stress Analysis of front hub(braking torque applied)

Fig 15 Displacement analysis of front hub (braking torque applied)
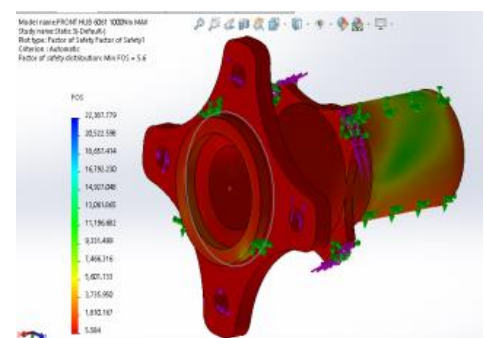

Fig 16 Factor of safety of front hub (braking torque applied)
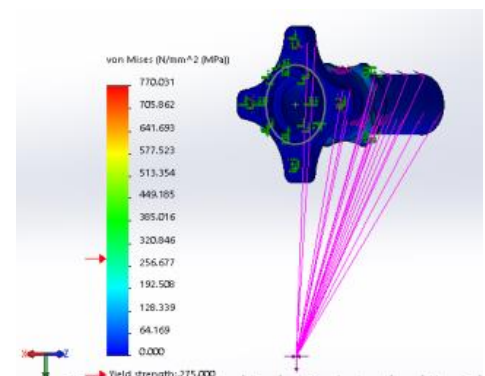

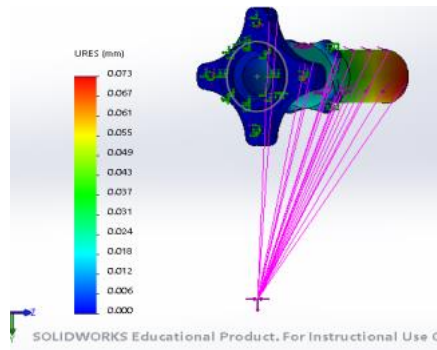**Fig 17** Stress Analysis of front hub (applying ground reactions)

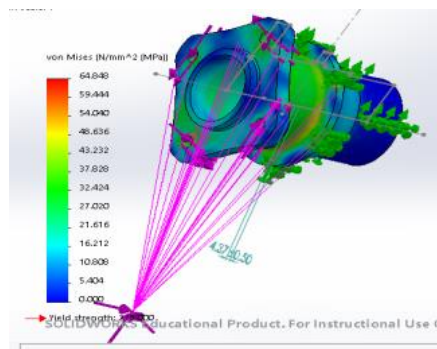**Fig 18** Displacement Analysis of front Hub (applying ground reactions)

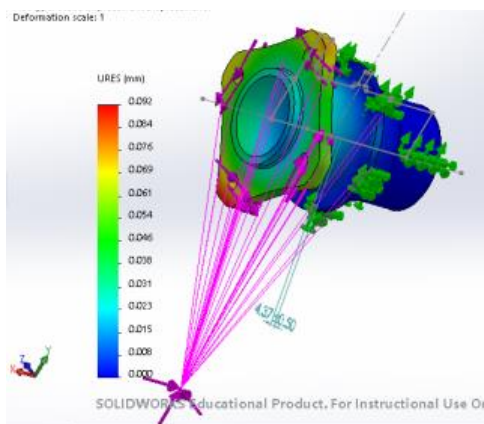**Fig 19** Stress Analysis of rear Hub (applying ground reactions and braking torque)

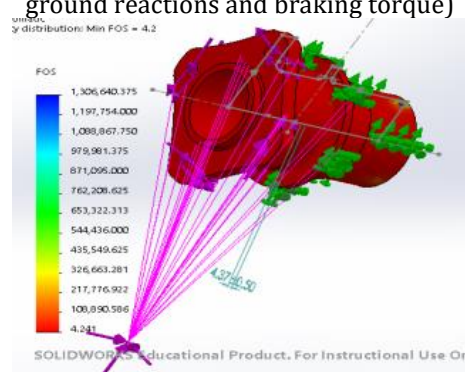**Fig 20** Displacement Analysis of rear Hub (applying ground reactions and braking torque)

Fig 21 - Factor of safety of rear Hub (applying ground reactions and braking torque)

3.7 Manufacturing of Wheel Assembly

After the analysis of the wheel assembly, it was time to manufacture it. Since the parts were too intricate, it was decided to manufacture it using computer numeric control (CNC). Because of the feasibility that CNC machines give, to design of the components were possible without any restrictions. As mention earlier selected aluminium T6 6061 was selected for both upright and hub. So the block of aluminium was purchased having size a little larger than the original component. Then each was machined on the vertical machining centre (VMC) having an excellent finishing. For the hub, first, it was machined on a lathe to give the profile so it would be easily machined on the VMC and would save time. Clevis was also made on VMC, and it was a two setting job which had taken up to 4 hours each.

Before machining, 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 more straightforward. All sharp internal corners were eliminated and were modelled with a 5mm 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 standard dimensions, so there would be no confusion when machinist was making the part. These dimensions must be double-checked before manufacturing so as not to make the part obsolete. Tool clearance was considered for operations such as using a wheel cutter for brake caliper slots. This required modelling 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. Bearing lock or a step was given in the upright and on the hub which was used to retain the bearing in their proper axial position. It will also restrict it from coming out of its place when running. So, the bearing diameter was kept according to boring diameter of upright which was 50mm for the front and 90mm at the rear and with proper tolerance for easy mounting and dismounting. 4 holes having PCD of 100 mm were put up for wire locking the bolts on the hub cap.
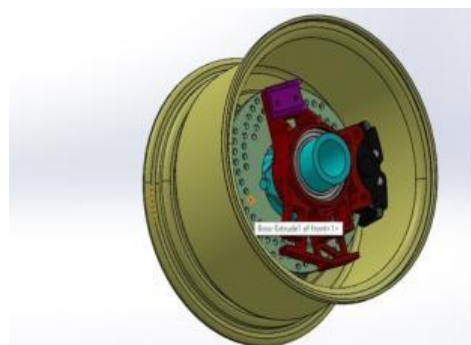

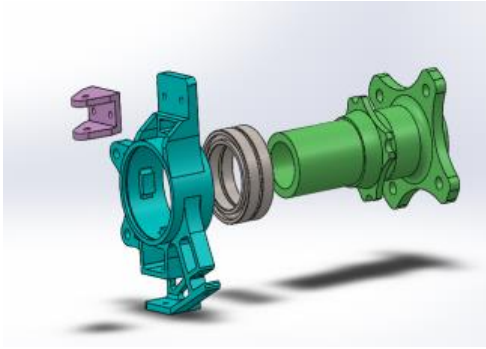**Fig 22** Front Wheel assembly in SolidWorks (including rims and caliper)
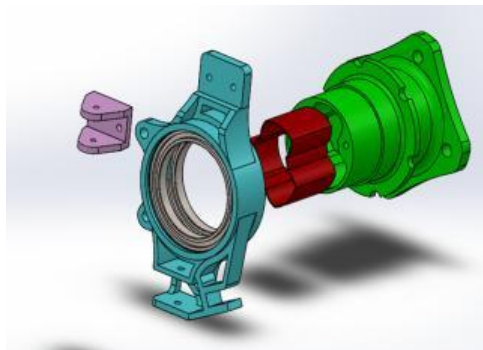
Fig 23 Front Wheel assembly in SolidWorks

Fig 24 Rear Wheel assembly in SolidWorks

Fig 25 Machining of upright on a VMC machine

Fig 26 Finished front upright

Fig 27 Finished rear upright

3.8 Floater or Bobbin

Being a floating design, a floater or bobbin needs to be made for the installation of the brake disc. The floater minimizes rattling noise and lets the brake disc stay in place while allowing it to expand when its temperature rises for the floater. By comparing all the parameters, mild steel and aluminum were selected for floater button as per design considerations and also the budget concern. They were manufactured by using CNC lathe.

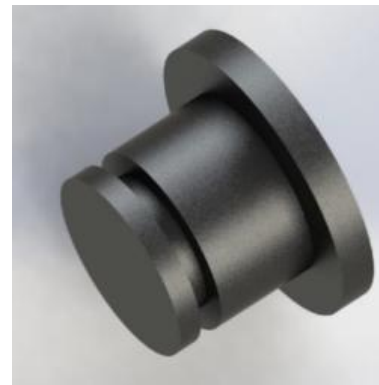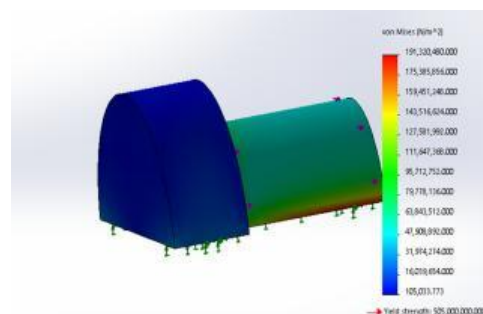**Fig 28** Floater or bobbin

Fig 29 Analysis of floater or bobbin

3.9 Tripod Sleeve

In the hub, the tripod bearing would have caused a lot of wear due to friction. Hence a sleeve was made by using wire cutting machinery. This sleeve was fitted

into the recces given for tripod bearing and is of 1 mm thickness and made up of steel. This sleeve was machined by wire cutting process.
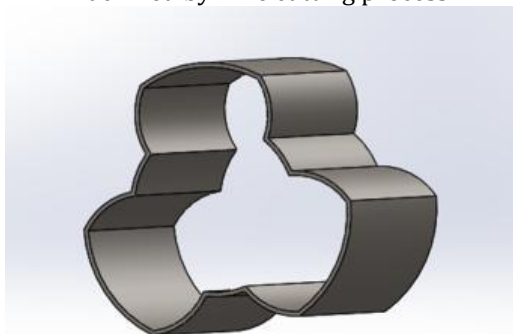

Fig 30 Tripod Sleeve in SolidWorks

3.10 Hub cap

In order to keep the upright confined on the hub its motion was restricted by a step on one side and by cap on the other. The cap was manufactured to be screwed at the end of the hub. This was made from commercial Aluminum which is locally available. This was machined on the lathe machine along with threading.
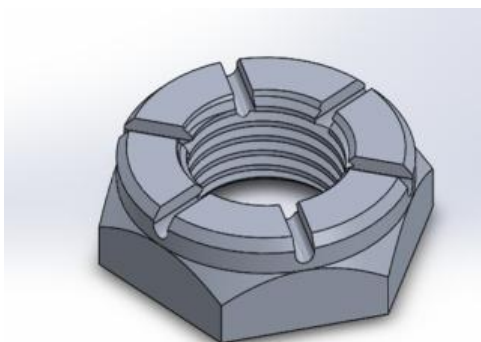

Fig 31 Hub Cap in SolidWorks

4. Final assembly

After completing the individual manufacturing, it was now time for final assembly. The pictures are shown below.

Fig 32 Actual rear assembly

Fig 33 Actual front assembly

Conclusion and result

Finally after the design, analysis, manufacturing and validation phase it is time to put the result in the table in order to compare it with previous design shown in the table below. The assembly was much lighter in weight and was able to sustain the forces while accelerating, cornering and braking when tested for 250kms, meeting the design expectations. On validating the results are satisfactory.

Table 1

SR38 (2016-17)			
	Weight (grams)		Weight (grams)
Front Hub	523.09	Rear Upright	463.35
Front Upright	715.14	Rear Toe Link Arm	46.13
Front Clevis	291.41	Rear LBJ Clevis	85.62
Spindle	327	Rear UBJ Clevis	111.06
Rear Hub	500		

Total Rear Wheel Assembly Weight = **1.45 kg**
Total Front Wheel Assembly Weight = **1.856 kg**
Overall weight **= 3.306 kg**

Table 2

Current Car			
	Weight (grams)		Weight (grams)
Front Hub	621	Clevis	36
Front Upright	430	Cap	122
Rear Hub	800	Tripod Sleeve	80
Front Upright	435		

Total Rear Wheel Assembly Weight = **1.315 kg**
Total Front Wheel Assembly Weight = **1.1 kg**
Overall weight **= 2.415 kg**

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Race car vehicle dynamics -- William F Milliken & Douglas L. Milliken.

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