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Design, Analysis, and Topology Optimization of Front Upright for Electric Solar Vehicle

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Abstract – *This Paper consist of calculation of dynamic forces acting on Upright of an Electric Solar vehicle. Proper calculations are done to find out the value of these forces. After getting the value of Forces, Analysis and Optimization are done to Reduce the unsprung mass and thereby increasing the performance of Vehicle. Upright are one of the important unsprung mass. The main aim for Suspension system Design of the vehicle is to keep unsprung mass as low as possible. Hence, various Optimizations are carried out to get the least mass for the components, which can sustain the forces acting on the Vehicle. Upright transfers force from chassis to Ground and also absorbs forces that are caused due to motion of the vehicle.*

Key Words: Front Upright, Electric Solar Vehicle, A-Arm, Dynamic Forces, Altair Inspire, Knuckle, Vehicle Dynamic Analysis, Topology Optimization

1. INTRODUCTION

Upright is the Mechanical component that connects A-Arm to the Wheels/Tyres. In our Electric Solar Vehicle Project, we have designed an upright connecting the A-Arm, which in turn is connected to Chassis. Upright also has a brake rotor mounted on it. The Steering Arm connects to it at the end, which need to satisfy minimum bump steer condition. The brake rotor is mounted in such a way that it is clamped at two points on the Upright and Brake Disc are in between the pads of the rotor. The Full Assembly Details are shown in Fig. 1.

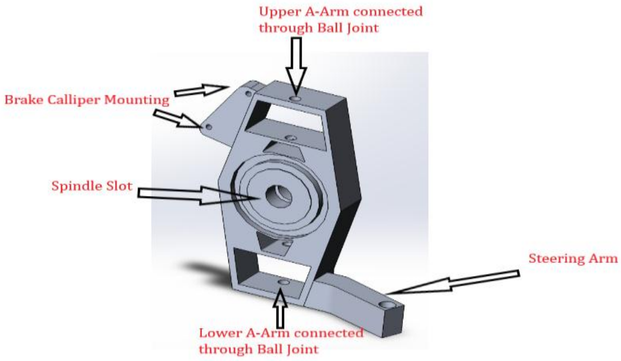

Fig. 1: Front Upright

1.1 Vehicle Specifications

The Vehicle is Electric Solar Vehicle, Designed to Participate in Electric Solar Vehicle Championship. The Designing of the components are done according to rule specified in the rulebook.[1]

Table -1: Vehicle Specifications

Dimension	**Front**	**Rear**
Gross Vehicle Mass (Kg)	450 Kg.	
Kerb Vehicle Mass (Kg)	350 Kg.	
Wheelbase (L)	2.18 m	
Track width (T)	1.75 m	1.75 m
Static Weight Distribution	55	45
Sprung mass (Kg)	380 Kg.	
Unsprung mass(kg)	70 Kg.	
Turning Radius (m)	3 m.	

1.2 Knuckle and Tire Specifications

Upright, also called as knuckle is an interchangeable word used mainly in off-road vehicles. Knuckles are designed in such a way that the rim of the wheel fits inside it with brake calliper mounted.

Table -2: Knuckle Specification

Scrub Radius (mm)	15.467
Length of Steering Arm (mm)	138
Distance of Hub Centre to Lower A-Arm point (a) (mm)	141.026

Distance of Hub Centre to Upper A-Arm point (b) (mm)	131.080
Pitch Circle Radius of caliper Bolt(r)(mm)	88.42
Tire Model	155/65 R13
Tire Radius (R) (mm)	266

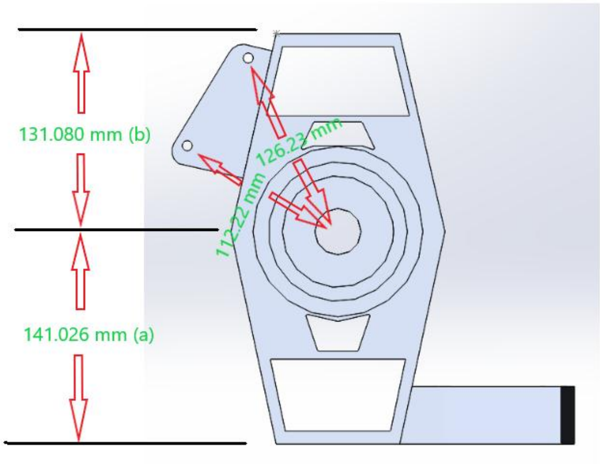

Fig. 2: Upright Dimension

2. DYNAMIC FORCE CALCULATION FOR ANALYSIS

Analysis and Optimization are very important parts for production of any mechanical component. Today, Automotive Companies are focusing on Analysis and Optimization of the Components through various CAE software before physical testing, as this will reduce the production price of the component without compromising its performance. To do this, Precise Estimations of Dynamic Force are required, for which we have carried out the force calculation by considering the following forces:

1. Braking force and torque acting on Caliper mounting
2. Lateral Force acting due to Cornering of the vehicle
3. Bump Force acting due to Suspension geometry
4. Force due to push and pull of tie rod

2.1 Braking force and torque acting on the Caliper Mounting

Total mass on the front axle= mf =247.5 kg
Braking Distance= d =4.95 m
Rear Wheelbase= Lr= 0.981 m
Height of CoG from ground= H =0.58293 m
Maximum velocity in a straight line= Vb= 16.67 m/s
Maximum deceleration = ad= -v^2/(2*d) = 15.6425 m/s^2
Coefficient of friction = v = 0.6

Dynamic force on the front axle = Fd= (m*g*Lr + m*ad*H)/L

Fd= (247.5*9.81*0.981+247.5*15.6425*0.58293)/2.18
= 1649.91 N
Vertical load due to unsprung mass = Fvu = (Unsprung mass on one wheel * g)
Fvu = 9.81 * 15.75 = 154.35 N

Total vertical load on one wheel = Fvu + Fd = 1649.91 + 154.35
Fvu = 1804.26 N

Frictional force on one wheel = f = v *Fvu
f= 0.6 * 1804.26 N = 1082. 56 N

Braking torque= T = f * Radius of Tyre = 1082.56 * 0.266 N-m
T = 287.96 N-m

Force exerted in the caliper mounting= Fc= T/r
Fc= 287.96/(0.08842)= 3245.72 N

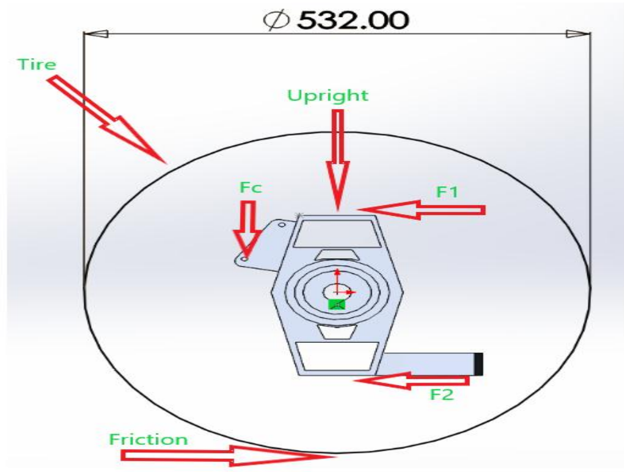

Fig. 3: FBD from side view

Balancing the moment of forces due to braking forces and force on the caliper mounting about center of spindle we get,
F1*b - F2*a + Fc*r=0 -(1)
Also, the net horizontal force acting is zero,
F1 + F2 = f -(2)
From eqn (1) and (2), we get
F1= -1057.68 N (towards right)
F2= 2140.24 N (towards left)

2.2 Lateral force acting due to cornering of the vehicle

Sprung mass =Ms = 380 kg
Unsprung mass= Mus = 70 kg
Total Mass =m = 450 kg
Maximum velocity during cornering = Vc = 9.72 m/s
Height of CoG from ground = h= 0.58293 m
Cornering Radius = Rc =3 m
Trackwidth = tw =1.75m
Mass of each wheel = Mw = 8 kg

Vertical load acting on each wheel = Fa = ((Ms +Mus)/4)*g

Fa= ((380 + 70)/4)*9.81
= 1103.625 N

Vertical load due to centrifugal couple:
Fcc= (m*Vc^2*h)/(2*Rc*t)
= (450 * 9.72^2 *0.58293)/(2*3*1.75)
= 2361.41 N

Vertical force due to gyroscopic effort:
Fg= (4* Mw * R^2/2)*(Vc^2/(R*r))
= (4*8* 0.266^2/2)* (9.72^2/(0.266*0.08872))
= 134.0946 N

Net vertical load acting on wheel:
Fv= Fa+Fg+Fcc = 3599.1296 N
Lateral force acting on the wheels:
Fl= v*Fv= 0.6* 3599.1296= 2159.48 N

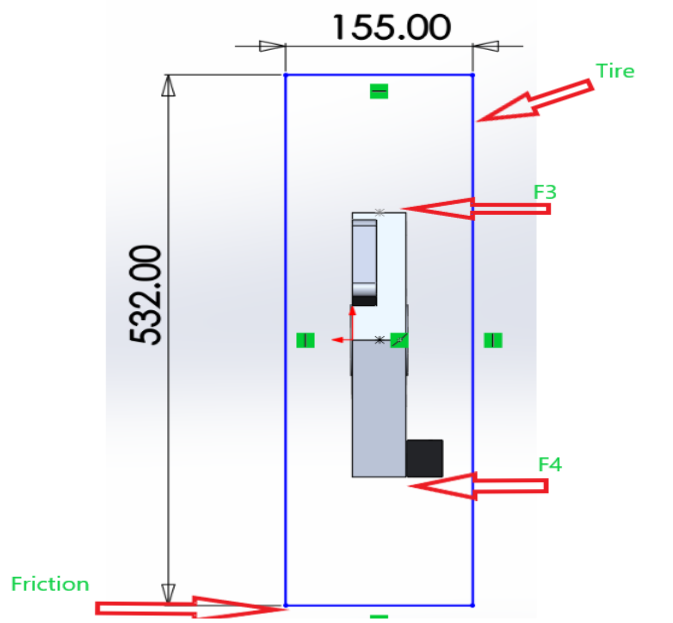

Fig. 4: FBD from Front view

Since the net horizontal force is zero, we have:
F4=Fl+F3 -(3)

Balancing moment of forces about the point of action of F4, we have:
Fl*(R-a) =F3*(a+b) -(4)

From equations (3) and (4), we get
F3= 1000.71 N (towards right)
F4= 3160.19 N (towards left)

2.3 Bump force acting due to the suspension geometry

Stiffness of the spring =k= 18.928 N/mm
Maximum compression in the spring =x= 40 mm
Angle in spring is mounted from vertical =theta= 30.2 degree

Length of suspension arm = L2 = 0.8793 m
Perpendicular distance from spring mounting on lower control arm to chassis = L1 = 0.690 m

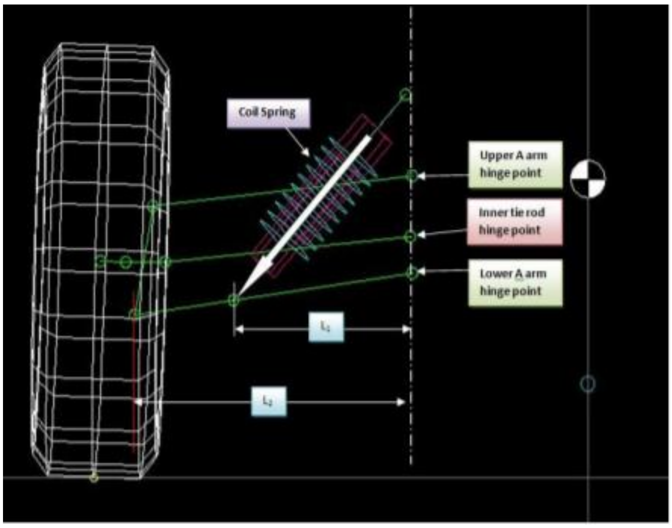

Fig. 5: Suspension Geometry

Force applied by coil spring = F = k*x
F= 18.928*40 = 757.12 N

Vertical load on wheels due to spring force= Fv =F*sin(theta)
Fv= 757.12*sin (30.2) = 380.85 N

Torque about point of contact of spring with chassis:
T= Fv*L1 = 380.85 *0.69

Bump force acting on the wheel:
Fb= T/ L2 = 380.85*0.69/0.8793= 294.4634

2.4 Force due to push and pull of the tie rod

Scrub Radius= Rs= 15.467 mm
Length of steering arm = l= 138 mm

Vertical load acting on each wheel = Fa = ((Ms +Mus)/4)*g
Fa= ((380 + 70)/4)*9.81
= 1103.625 N

Frictional force acting on the wheel = f= v*Fa
f= 0.6*1103.625 = 662.175 N

Torque acting about the Kingpin Inclination Axis:
T= f*Rs= 662.175*15.467= 10241.85 N-mm

Force acting on the tie rod= Ft= T/l= 10241.85/138
Ft= 74.22N

Table 3: Force Values and Direction

FORCE/TORQUE	VALUE (N or N-mm)
F1	1057.68 (towards right)
F2	2140.24 (towards left)
F3	1000.71 (forward) (as per fv towards right)
F4	3160.19 (backwards) (as per fv towards left)
F5	74.22 (backwards)
BUMP FORCE	294.46 (Upwards)
BRAKING TORQUE	287.96 (CCW)

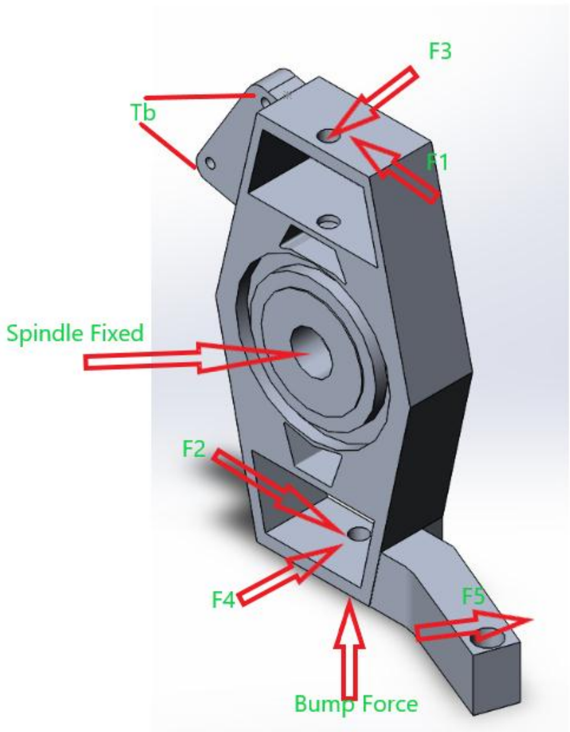

Fig. 6: Force Directions

3. MATERIAL SELECTION AND ANALYSIS

Two materials are selected for comparison. They are grey cast Iron and Al 6061-T6. Their chemical specification can be checked from the reference[2]. We will only be using some properties to compare them.

3.1 Material Selection

These properties are given in the table below, we will finalize a material for further analysis and optimization of the Upright.
The comparisons are given in the table below:

Table -4: Material Properties Comparison

Properties	Grey Cast Iron	AL 6061-T6
Density	7060 kg/m^3	2700 kg/m^3
Price/kg*	Rs. 60	Rs. 275
Ultimate Tensile Strength	276 MPa	310 MPa
Youngs Modulus	124 GPa	69 GPa
Yield Tensile strength	65.5 MPa	160 MPa
Poisson Ratio	0.255	0.33

From the Table, we can see that **AL 6061-T6** is the better Option and we will be using it for All Analysis and Optimization**.**

3.2 Analysis of Upright

The Value of Forces are listed as Below for Analysis.

Table 5: Forces value for Analysis

Forces/Torque	Value	FOS	FOS	Force1	Force 2
F1 (N)	1057.68	1.75	2.00	1850.94	2115.36
F2 (N)	2140.24	1.75	2.00	3745.42	4280.48
F3 (N)	1000.71	1.75	2.00	1751.24	2001.42
F4 (N)	3160.19	1.75	2.00	5530.33	6320.38
F5 (N)	74.22	1.75	2.00	129.885	148.44
Bump Force(N)	294.46	1.75	2.00	515.305	588.92
Braking Torque (Nm)	287.96	1.75	2.00	503.93	575.92

Analysis and Optimization was carried out on Altair Inspire 2020. The type of Optimization done is Topology Optimization

We have decided to take two Factor of Safety, to account for uncertainties in the strength and the uncertainties in load with the guideline of Machine Design book mentioned in Reference [3]. After Analysis, Topology Optimization was carried out with Load cases from the Analysis Part.

Load cases are very important in Analysis. The forces that are acting on upright are not acting altogether, they are acting in different conditions. Hence, load cases are created for proper analysis.

Table 6: Load Cases for Analysis

Force/Torque	**Load case 1**	**Load case 2**	**Load case 3**	**Load case 4**	**Load case 5**
F1	✓				✓
F2	✓				✓
F3		✓			✓
F4		✓			✓
F5			✓		✓
Bump Force				✓	✓
Braking torque	✓				✓

Load Cases are considered as per above table for both Factor of Safety. Result envelope from these Load cases will be considered. Result Envelope contains overall Analysis Results.

3.3 Analysis Result

Analysis Result of Upright was Observed for Total Deformation, Von-Mises stress, and Factor of Safety.
The Results are shown in below table:

Table 7- Analysis Result (for FOS = 1.75)

Parameter	**Value**
Factor of Safety	2.310 (minimum)
Von-Mises Stress (MPa)	93.08 (maximum)
Total deformation (mm)	0.01261 (maximum)

Table 8- Analysis Result (for FOS = 2)

Parameter	**Value**
Factor of Safety	2.021 (minimum)
Von-Mises Stress (MPa)	106.4 (maximum)
Total deformation (mm)	0.01441 (maximum)

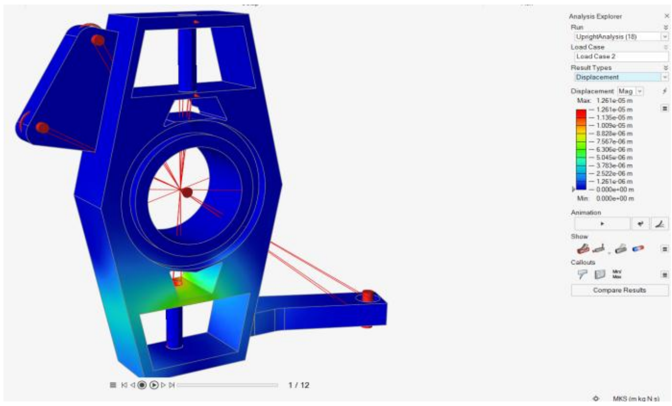

Fig. 7: Total Displacement (FOS= 1.75)

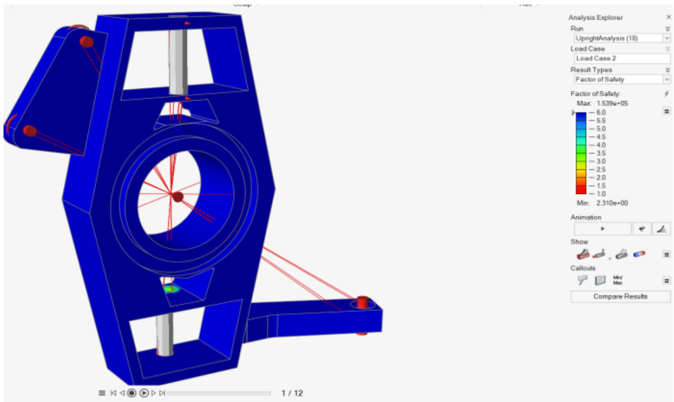

Fig. 8: Factor of Safety (FOS = 1.75)

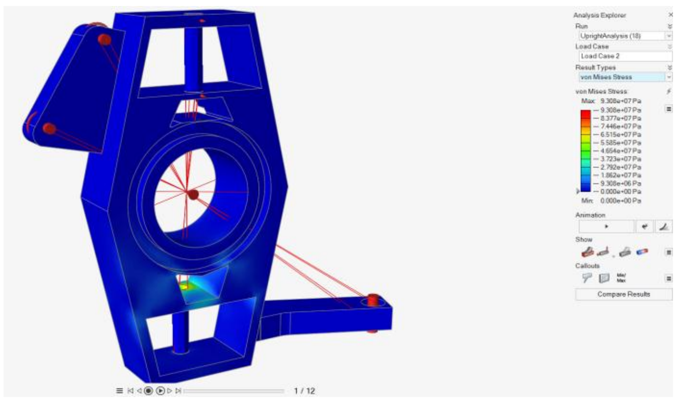

Fig. 9: Von-Mises Stress (FOS = 1.75)

The Analysis result shown above are for FOS = 1.75. figures below indicate the result for FOS =2.

The Result contain same for Total Displacement, Von-Mises stress and Factor of Safety.

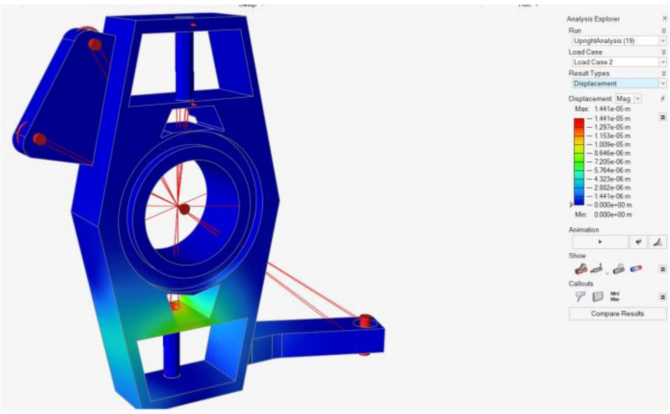

Fig. 10: Total Displacement (FOS = 2)

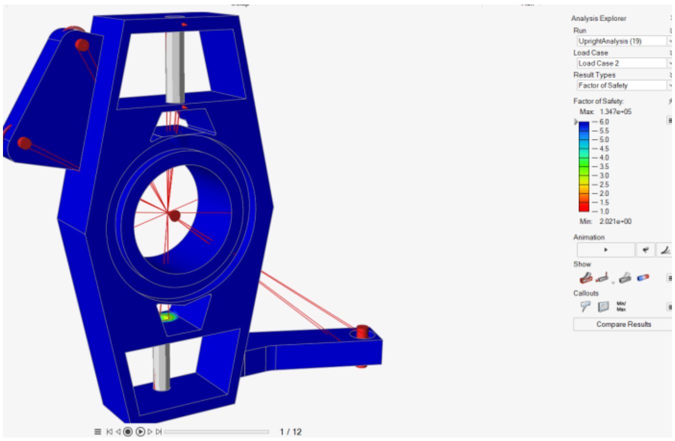

Fig. 11: Factor of Safety (FOS = 2)

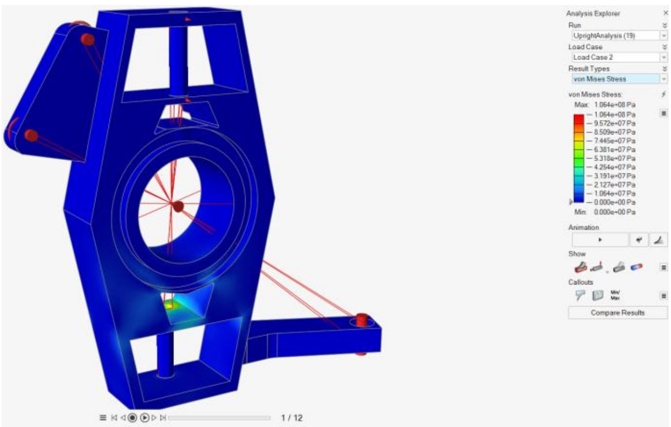

Fig. 12: Von-Mises Stress (FOS = 2)

Overall Factor of Safety:

It is sometimes convenient to define two factors of safety to obtain best results.

1. ns = Accounts for uncertainties in the strength.
2. nl = Accounts for the uncertainties with regards to the load

Overall FoS = ns*nl

(1). For FOS = 1.75

X = 1.75* 2.310 = 4.043

(2). For FOS = 2

Y = 2*2.021 = 4.042

Overall FOS = min (X, Y) = **4.042**

3.4 Topology Optimization

Topology Optimization is used to optimize the distribution of material within a desired boundary known as the design space for a given set of load cases with an aim of maximizing the performance along with minimizing the mass thus reducing the cost for manufacturing.

After the analysis, we proceed for topology optimization by combining both the factor of safety and their load cases. Weight reduction is very important in unsprung components of the vehicle. Iterative Analysis and Optimization were carried out to get an optimum result.

Topology Optimization was carried out in the same software as before i.e. Altair Inspire 2020. The results are mentioned in below table. Topology Optimization have range of results, we will be taking that result which suits our result for further analysis. Targeted weight reduction was at least 20% of the total mass of the upright. Figure below shows one of the solutions from the range of topology optimization.

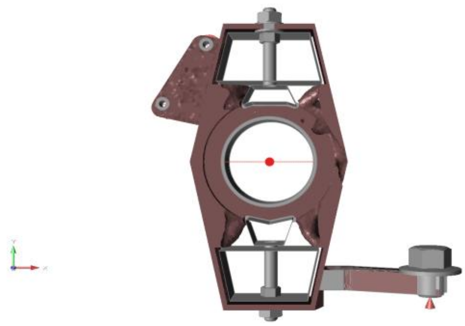

Fig. 13: Topology Optimization result

Figure below represent the upright after the Optimization. This upright was used for further analysis and finalizing the dimensions of the upright.

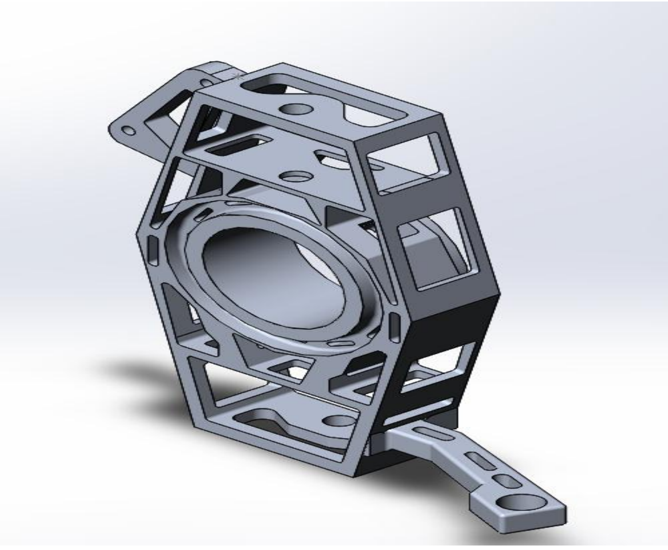

Fig. 14: Final Upright

Reduction in Weight = (Change in weight/Initial weight)
= 0.7/2
= 0.35
Percentage Reduction in Weight = 0.35*100
= 35 %

Table 9- Optimization Result

Parameter	Value
Initial Weight	2.0 Kg
Final Weight	1.3 Kg.
Percentage Reduction in Weight	35 %
Overall FOS	4.042

4. CONCLUSION

The aim of the paper was to design and manufacture upright for the Application in Electric Solar Vehicle. In this paper, we have Analyzed and Optimized the Front Upright using Altair Inspire Software. Iterative process was used for Analysis and Optimization for obtaining the Final Upright. After Optimization, we have carried out further analysis to get the final results. The results were satisfactory and we will start manufacturing the upright and testing will be carried out to check for its failure.

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