Topology Optimization of ATV Components for Electric Baja Vehicle

Rishi Gupta[1]

[1]*Student, Department of Mechanical Engineering, Acropolis Institute of Technology and Research, Indore, Madhya Pradesh, India*

---***---

Abstract -. *This Paper consist of calculation of dynamic forces acting on Upright, Front Hub, Rear Hub spacer and Differential Mounting of an Electric Baja Vehicle. Proper calculations are done to seek out the worth of those forces. After getting the value of Forces, Analysis and Optimization are done to Reduce the Weight and thereby increasing the performance of Vehicle. The main aim for Upright is to transmit all the forces and moments between the wheel and rod control arm. Hence varied Optimizations are performed to induce the smallest amount mass of the parts, which may sustain the forces performing on the vehicle. Upright transfers force from chassis to Ground and also absorbs forces that are caused thanks to motion*

Key Words: Front Upright, Front Hub, Rear Hub, Differential mounting, Vehicle Dynamic analysis, Dynamic forces, Topology Optimization

1.INTRODUCTION

ATV means an all-terrain vehicle which is specially designed for an off-road driving. ATV is meant for very rough terrain, jumps, maneuverability and endurance. The design process of this single-person vehicle is iterative and supported several engineering and reverse engineering processes.

1.1 Vehicle Specification

The Vehicle is Electric Vehicle, Designed to Participate in SAE BAJA INDIA. The Designing of the components are done according to rule specified in the rulebook.

Overall weight	270 kg
Wheelbase	1300 mm
Track-width	1200 mm
Kerb Weight	195 kg
Ground Clearance	304.8 mm
Transmission	Rear wheel drive
Motor type	BLDC 125/6D-4.5
Battery type	LiNMC (48V/110Ah)
Tire Size	23X7-R10
Max. Velocity	41 kmph
Acceleration	4.75 m/s^2

Table.1: Vehicle Specification

1.2 Front Upright

The Front upright is one of the Key components of a racing suspension assembly, which connects the suspension link control arm, tie roads and axle. The main function of the front front upright is to transmit all the forces and moments between the while and the connecting rod control arm.

Fig. 1: Front Upright

1.3 Front Hub

The Hub is employed to attach the wheel to the vehicle. On rear-wheel drive vehicle this is the front wheels and on a front wheel drive vehicle it is the rear wheels. The hub assembly is mounted on a stub axle, which is connected to the suspension of the vehicle.

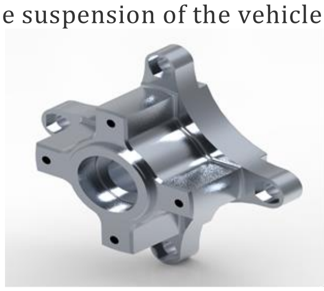

Fig. 2: Front Hub

2. Rear Hub spacer

The Front Hub is simply designed to enable the wheel to spin, while the rear hub is a little more complex as it also forms part of the bike's transmission- the cassette or sprocket which drives the rear wheel is attached to the rear hub, which on most bikes also features mechanism to allow you to coast or freewheel.

Fig.3: Rear Hub

2.1. Differential mounting

In rear-wheel drive automobiles the central drive shaft (or prop shaft) engages the differential through a hypoid gear (ring and pinion). The ring gear is mounted on the carrier of the planetary chain that forms the differential

Fig.4: Differential Mounting

3. DYNAMIC FORCE CALCULATION FOR ANALYSIS

Analysis and Optimization are vital parts for production of any mechanical component. Today, Automotive Companies are that specialize in Analysis and Optimization of the Components through various CAE software before physical testing, as this may reduce the assembly price of the component without compromising its performance.

1.Bump force and lateral force acting on front upright.
2. Lateral Force acting on Front and Rear Hub.
3. Centrifugal Force acting on battery.

3.1 Bump Force and Lateral force acting on front Upright

Total mass = 270 Kg
Mass distribution ratio = 35:65
Then analysis of the Front upright considering Aluminium T6-7075

of shown dimensions was done and we got the safety factor more than 2, which justified the selection.

Total mass on front axle (sprung + unsprang) = 94.5 Kg

Assuming Braking distance = 4.5 m

Distance of cg from front axle = 845 mm
Distance of cg from rear axle = 455 mm
Max velocity of vehicle = 45 kmph = 12.5m/.
Using equation of motion, a=-1.77g
Total vertical load = 1247.94+147.15
=1422.09 N
Frictional force = = 853.25 N
Force on caliper mounting $=\frac{(249.15)}{0.C050}$ = 4983 N
Lateral Load Transfer
Vertical load on each wheel = (Sprung+unsprung mass)/2
= (210+60) +35%)/2
= 462.0 N
Bump force = Spring stiffness * Maximum compression
= 20 N/mm * 140mm
= 2800 sin (45)
= 1979 N
(After applying moment equation)

Bump force (F5) = 1212 N

3.2 Lateral Force acting on Front and Rear Hub

The impact velocity of the vehicle is from kinematic relation for rectilinear motion

V= $\sqrt{2gh-u^2}$ = $\sqrt{2\times 9.81\times 1-0^2}$ = 4.42 m/s
From Work Energy Principle

Charge in K.E of the Object = Work Done on the object

$= \frac{m\times(v^2-u^2)}{2} = F\times d$

$= \frac{270\times(4.42^2-0^2)}{2} = F\times 0.17$

=11428.78 N
Bump Force = 11428.78 N
Breaking Torque = Friction Force generate due to torque on wheel from contact patch
Frictional Force = Coefficient of Friction × Total vertical Load = 0.6 × 1422.09
= 853.25 N

Breaking Torque (T_b) = Frictional Force × Radius of Tire
= 853.25× 0.292
= 249.15 N-m
Lateral Forces=

$$\frac{\text{Sprung Mass at Front or Rear} + (\text{Vehicle Velocity})^2}{\text{Turning Radius of Vehicle}}$$

$$= \frac{94.5 + (12.5)^2}{2.7}$$

= 5962.5 N
=6000 N

3.3. Centrifugal Force acting on battery.

Cornering or centrifugal force CF = MV^2/r
Fc = m*(2.314r/t) ^2/r^2
Fc = 23*4*3.14*2.7
Fc = 289.02N
Bump Force for differential mount
=mgh
F=4512.6N
Force Calculation For battery mounting bump force for battery
Mounting = Mgh
F= 5886N
Centrifugal Force For battery
Cf= m*4*3.14*R
Cf=508.93N

4. MATERIAL SELECTION AND ANALYSIS

4.1 Material Selection

The goal is to style and build a prototype of a rugged, single seat, off-road RV intended purchasable to the non-professional weekend off-road enthusiast. The vehicle must be safe, easily transported, easily maintained and fun to drive. It should be ready to negotiate rough terrain without damage. To make an effort in reducing air pollution caused by automobiles. To make vehicle, this can run easily and fast to perform rescue operation in habitable land. Sub- components were main part of an ATV to mount on chassis. The main objective is to optimize the sub-system of an ATV to reduce the weight of overall ATV and Increase the acceleration.

Table.2: Material Properties

Material Properties			
Material (Aluminium)	EN 8	Al 6061-T6	**Al 7075-T6**
Tensile Strength (Ultimate)	800 MPa	310MPa	**570MPa**
Tensile Strength (Yield)	550 MPa	270MPa	**480MPa**
Poisson's Ratio	0.30	0.33	**0.300**
Elongation % at break	18%	8%	**11%**

Al 7078-T6 is the best suitable material so following it we selected it over Al 6061-T6 because It has excellent mechanical properties and exhibits good ductility, high strength, toughness, and good resistance to fatigue. It is more susceptible to embrittlement than many other aluminum alloys because of micro segregation, but has significantly better corrosion resistance than the alloys from the 2000 series.

4.1.1 Front Upright Analysis Result

The Material for Front Upright analysis was Material- Al 7075-t6. The Boundary condition for analysis is as follow:

1.Fixed - central part of Knuckle.

2.Force applied - Brake caliper Mountings, Suspension

Table 3- Analysis Result

Parameter	Value
Factor of Safety	1.48(minimum)
Equivalent Stress(Mpa)	323.19 (maximum)
Total deformation (mm)	0.46 (maximum)

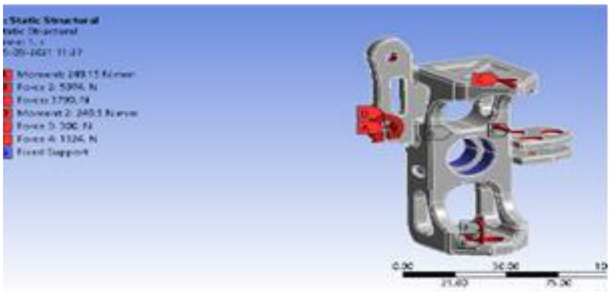

Fig.5: Forces and constraint

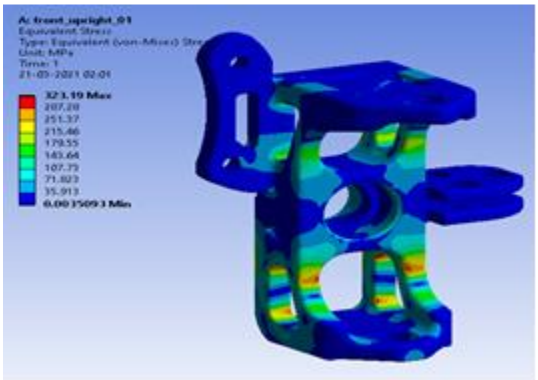

Fig.6: Equivalent stress

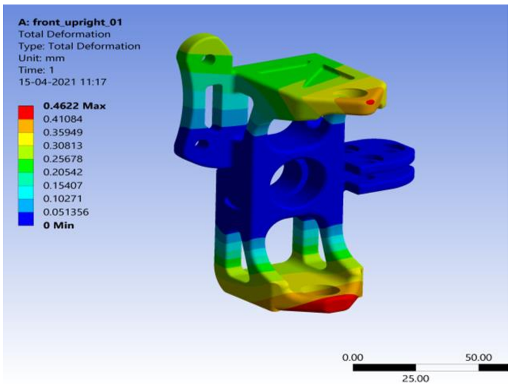

Fig.7: Total Deformation

4.1.2 Front Hub Analysis Result

The Material for Front Upright analysis was Material- Al 7075-t6. The Boundary condition for analysis is as follow:

1. Fixed - central part.

2. Force applied - Cornering and Bump force, Braking torque over bolted points.

Parameter	Value
Factor of Safety	4.28 (minimum)
Equivalent Stress (Mpa)	112 (maximum)
Total deformation (mm)	0.038 (maximum)

Table 4- Analysis Result

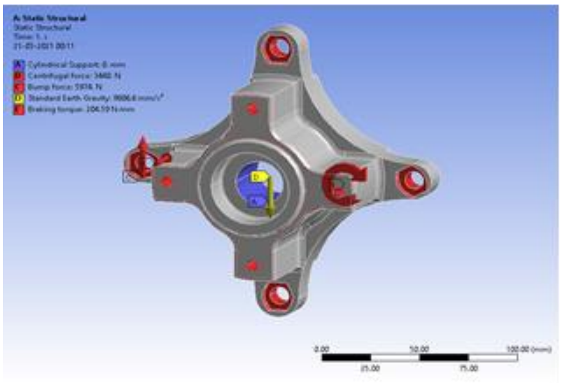

Fig.8: Forces and constraint

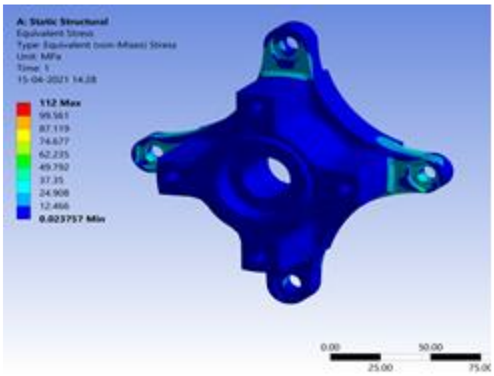

Fig.9: Equivalent stress

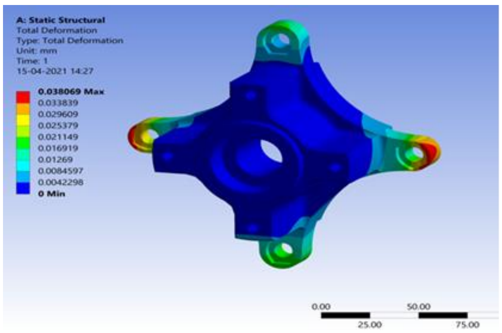

Fig.10: Total Deformation

4.1.3 Rear Hub Spacer Analysis Result

The Material for Front Upright analysis was Material- Al 7075-t6.

Parameter	Value
Factor of Safety	5.96 (minimum)
Equivalent Stress (Mpa)	80.42 (maximum)
Total deformation (mm)	0.190 (maximum)

Table 5- Analysis Result

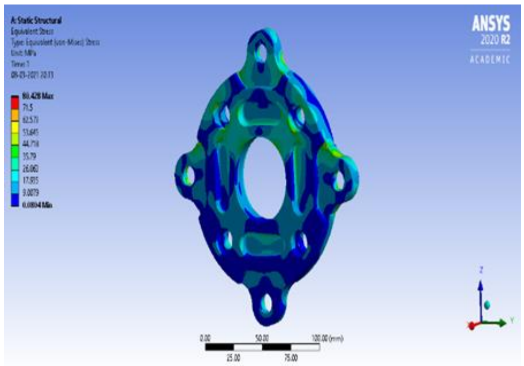

Fig.11: Equivalent stress

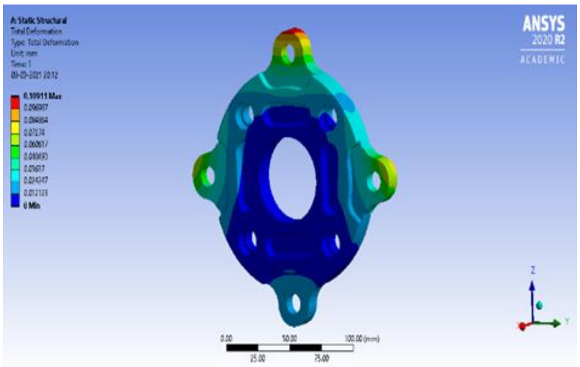

Fig.12: Total Deformation

4.1.4 Differential Mounting

The Material for Front Upright analysis was Material- Al 7075-t6. The Boundary condition for analysis is as follow:

1.Fixed - Bolted points.

2.Force applied - Centrifugal and Bump force, driving shaft torque.

Parameter	**Value**
Factor of Safety	4.1 (minimum)
Equivalent Stress (Mpa)	116.95 (maximum)
Total deformation (mm)	0.70(maximum)

Table 6- Analysis Result

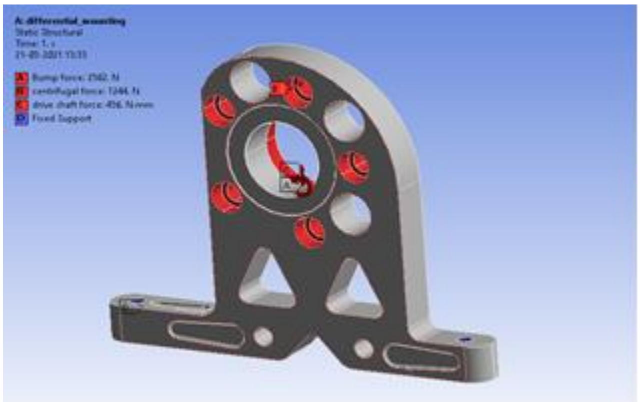

Fig.13: Forces and constraint

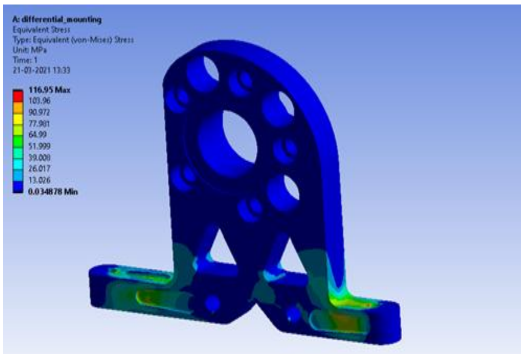

Fig.14: Equivalent stress

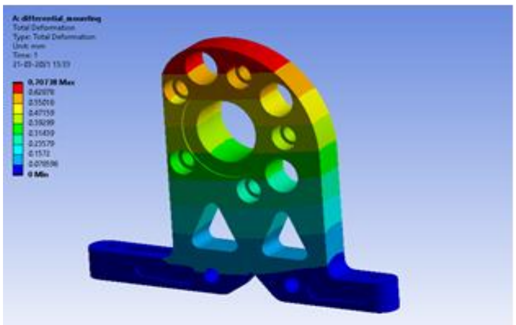

Fig.15: Total Deformation

5. TOPOLOGY OPTIMIZATION

Topology Optimization is employed to optimize the distribution of fabric within a desired boundary referred to as the planning 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 administered to urge an optimum result. Topology Optimization was carried out in the same software i.e., Ansys R2. The results are mentioned in below table. Topology Optimization have range of results, we'll be taking that result which suits our result for further analysis. Figure below shows one of the solutions from the range of topology optimization.

5.1 Front Upright Topology Optimization

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

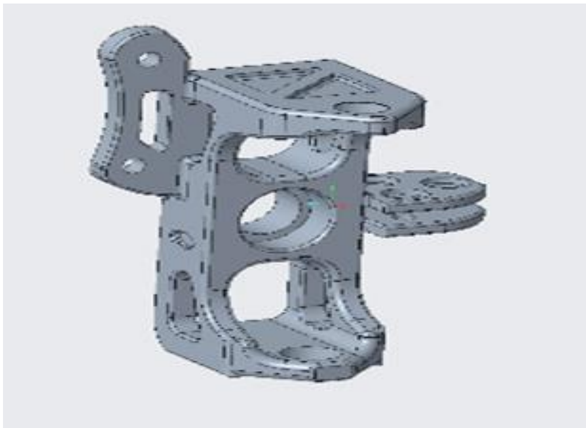

Fig. 16: Final Upright

Parameter	**Value**
Initial Weight	0.902 Kg
Final Weight	0.307 Kg.
Percentage Reduction in Weight	66 %

Table 7 - Optimization Result

5.2 Front Hub Topology Optimization

Figure below represent the Front Hub after the Optimization. This Hub was used for further analysis and finalizing the Size of the Front Hub.

Fig. 17: Final Front Hub

Parameter	**Value**
Initial Weight	1.9 Kg
Final Weight	1Kg.
Percentage Reduction in Weight	47 %

Table 8 - Optimization Result

5.3 Rear Hub Spacer Topology Optimization

Figure below represent the Rear Hub Spacer after the Optimization. This Hub was used for further analysis and finalizing the Size of the Rear Hub spacer.

Fig. 18: Final Rear Hub Spacer

Parameter	**Value**
Initial Weight	1.2 Kg
Final Weight	0.610 Kg.
Percentage Reduction in Weight	49 %

Table 9 - Optimization Result

5.4 Differential Mounting Topology Optimization

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

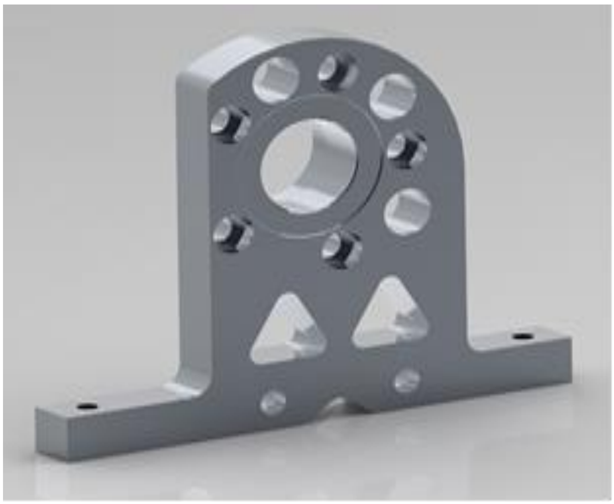

Fig.19: Final Differential Mounting

Parameter	**Value**
Initial Weight	1.2 Kg
Final Weight	0.610 Kg.
Percentage Reduction in Weight	49 %

Table 10 - Optimization Result

6. CONCLUSIONS

The objective of designing one passenger off road race vehicle with high safety and low production costs seems to be accomplished.

The design is first conceptualized supported personal experiences and intuition. Engineering principles and style processes are then wont to verify and make a vehicle with optimal performance, safety, manufacturability and ergonomics. The design process included using PTC Creo, Solid works and ANSYS software packages to model, simulate and assist in the analysis of the completed vehicle. After initial testing we have gone through various methods of optimization without affecting the durability and performance of the vehicle.

REFERENCES

1. Rahul Sharma Design and Weight optimization of wheel assembly Volume 2 – No.6, October 2015

2.IJSRD - International Journal for Scientific Research & Development| Vol. 7, Issue 07, 2019| ISSN (online): 2321-0613 Design, Analysis and Fabrication of Suspension System of an All- Terrain Vehicle Mr. Vijay Zarkar1

3. Race car vehicle dynamics by William F. Milliken and Douglas L. Milliken, 1995,

4. Thomas D. Gillespie, Fundamental of vehicle dynamics (Society of automotive engineers).

5. Carroll Smith, "Tune to Win", Aero Publishers. 1978