Design and Analysis of Upright of an FIA Regulated Cruiser Class Solar Electric Vehicle

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***Abstract*— Knuckle, also called upright in some cases is one of the most critical components in an automobile. It is the part that is connected with suspension arms (depends on the type of suspension system, in this project double wishbone type of suspension is used), the hub, brake calliper mounting and also the steering tie rod. All the loads reacted by the wheels are transmitted to the chassis through the A arms indirectly from the knuckle. Generally the nature of the loads is bending. This component comes under unsprung mass, to improve the dynamics of the vehicle it is required to reduce the weight of the upright (unsprung mass) and at the same time it has to be strong enough to take all the loads acting on it. Since the upright is a critical component, shear concentration is to be taken in designing and analyzing it. Different materials are used for manufacturing these components like Aluminum alloys like 7075-T6 for light weight applications or 6061-T6 for moderate weight and moderate cost or Steel in case of low cost applications.**

First, the upright is designed in CAD (CATIA V5) with respect to the suspension mounting points. The model is meshed using HyperMesh and analyzed based on the loads calculated using analytical method (Free Body Diagram) using formulae for various maneuvers possible. The Al upright is optimized based on various modules in Optistruct software and steel counterpart is based on design intuition (otherwise the yield would be similar to the Al counterpart since the forces are same and stresses would be same for the initial blank so Optistruct would yield same result).

***Keywords*— knuckle, upright, optimize, FEA, CAD, Stress Contour, Solar Vehicle, SolarMobil Manipal**

I. INTRODUCTION

A passenger solar car is designed and manufactured by the SolarMobil Manipal team, an official student project of MIT. The team designs and manufactures solar electric vehicles for international competitions such as Sasol Solar Challenge, South Africa and World Solar Challenge, Australia. MIT also have a FSAE and BAJA teams which also manufactures their uprights using aluminium for weight reduction and easy design. Since some steels have a strength to weight ratio similar to aluminium, why not manufacture upright using steel, with which the cost reduces drastically, minimum material wastage for manufacturing and easy post manufacturing. So, this paper would put that thought to a test where an aluminium upright and a steel upright are designed and compared for strength and weight which are serving the same purpose.

II. BASIC CALCULATIONS

A. *Specification of the passenger solar car*

TABLE I. GENERAL SPECIFICATION OF THE CAR

Length	4400 mm
Width	1750 mm
Height	1200 mm
CG height	450 mm
Weight distribution (Front : Rear)	45:55
Wheel Base	2700 mm
Track Width	1650 mm
Tire specification	5.2- 14
Ground Clearance	150 mm

B. *Various Maneuvres under Consideration*

- The car can stop at a deceleration of 1g i.e., when it is running with a speed of 100Kmph can stop in under 40m and in 3.5 sec.
- The maximum speed the car can attain when it is travelling in a circle of 70m radius is 60Kmph. So, the lateral acceleration is 4m/s^2.
- Acceleration of the car is 4m/s2 i.e., it reaches 60kmph from start in 4s

C. *Formulae for the Calculations*

$$\text{Longitudinal weight transfer} = \Delta W_l = \frac{acceleration \times CGheight \times Weightofthecar}{wheelbase \times 2} \text{ Kg} \quad (1)$$

$$\text{Transverse load transfer} = \Delta W_t = \frac{acceleration \times CGheight \times Weightofthecar \times \%weight}{Trackwidth} \text{ Kg} \quad (2)$$

$$\text{Acceleration over a bump} = \frac{v^2}{r} \text{ m/s}^2 \quad (3)$$

(v= velocity of the car
r= radius of the bump)

III. FREE BODY DIAGRAMS

The free body diagrams of the upright in various maneuvers of the car including the calculations of the forces at the points required are shown in the Fig. 1, Fig. 2 and Fig. 3. The format as followed for every FBD is, after very diagram a sequence of steps are given showing the calculations in deriving the forces using the formulae mentioned in (1), (2) and (3) i.e., first the calculation of longitudinal or transverse load transfer is calculated based on the maneuver the car is performing, then the dynamic vertical load is calculated on each wheel and the reaction forces are calculated on the component under consideration.

The triangle represents the constraints, the arrow marks represent the direction of the forces and a label is provided for every force indicating the magnitude of force.

A. Braking Maneuvre

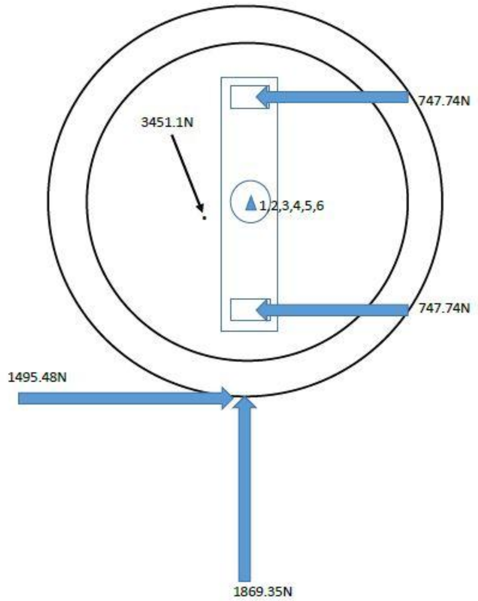

Fig. 1. Front upright, Braking loads when car is decelerating at 1g

- Longitudinal Weight Transfer, $\Delta W_l = \dfrac{1 \times 0.5 \times 600}{2.7 \times 2}$ =55.55kg
- Vertical load on the wheel= (135+55.55) $\times$9.81= 1869.35N
- Frictional Force= 1869.35 $\times$0.8= 1495.48N
- Reaction Forces at the upright= $\dfrac{1495.48}{2}$ = 747.74N

B. Cornering Maneuvre

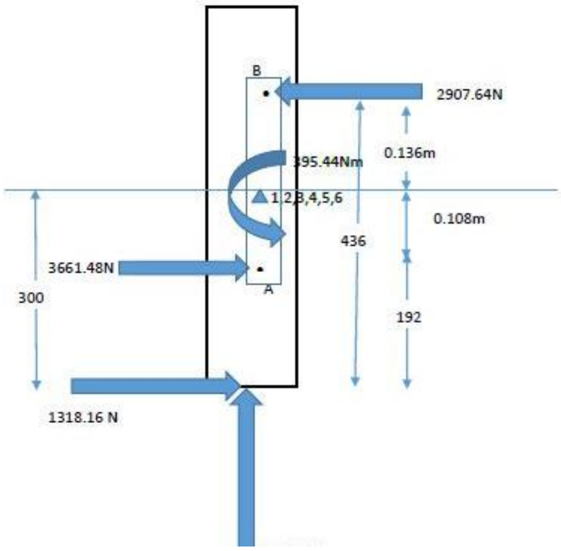

Fig. 2. Front upright, cornering loads, when car is turning at a speed of 60Kmph in a 70m circle-FBD

- Lateral Load transfer, $\Delta W_t = \dfrac{4 \times 0.5 \times 600 \times 0.45}{9.81 \times 1.67}$ = 32.96kg
- Vertical load on the wheel= (135+32.96) $\times$9.81= 1647.7N
- Frictional Force= 1869.35 $\times$0.8= 1318.16N
- Moment due to friction force about the hub= 1318.16 $\times$0.3= 395.44Nm
- Force reacted at point A= $\dfrac{395.44}{0.108}$ = 3661.48N
- Force reacted at point B= $\dfrac{395.44}{0.136}$ = 2907.64N

C. Bump Maneuvre

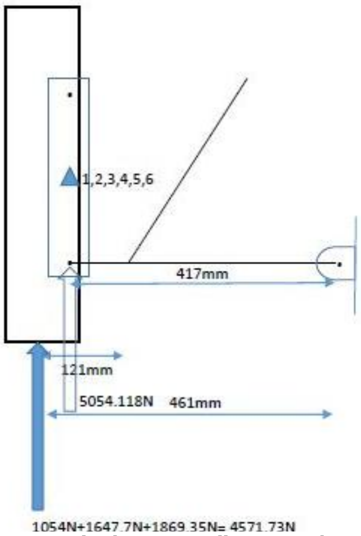

Fig. 3. Front Upright in Bump loads- Car travelling over a bump of 20m at 45Kmph

- Centrifugal force on the bump= $\frac{135\times12.5^2}{20}=$ 1054.68N
- Vertical load from braking= 1869.35
- Vertical load from cornering= 1647.7
- Total load in the worst case scenario= 4571.73N
- Force at A= $\frac{4571.73\times0.461}{0.417}$= 5054.118N

IV. DESIGN AND ANALYSIS OF THE UPRIGHT

A. Aluminium Upright

A raw blank is generated which satisfies the geometrical parameters using software CATIA. The geometry is imported and meshed with an optimization software (Optistruct), a raw analysis is done to see the stress contours. Several topological optimization techniques are used for optimizing the raw blank into the required form. The geometry was again reconstructed with required modifications from optimization results in CATIA and was again imported to Radioss for static analysis.

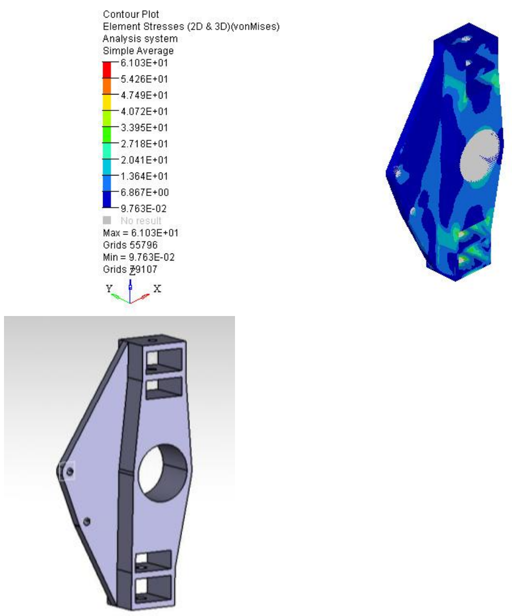

Fig. 4. Raw blank (CAD model on the right) for analysis; max stress= 61MPa; weight= 1.569kg

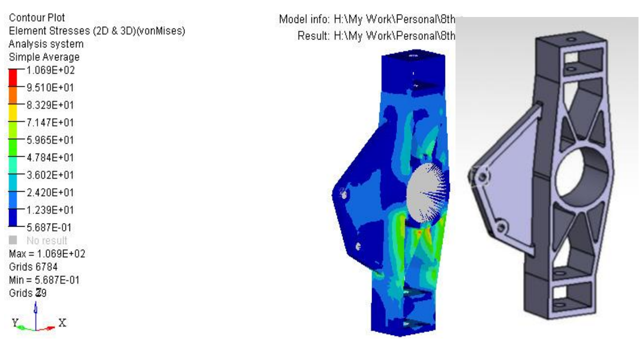

Fig. 5. After optimization (CAD model on the right); max stress 106MPA Weight= 0.854kg

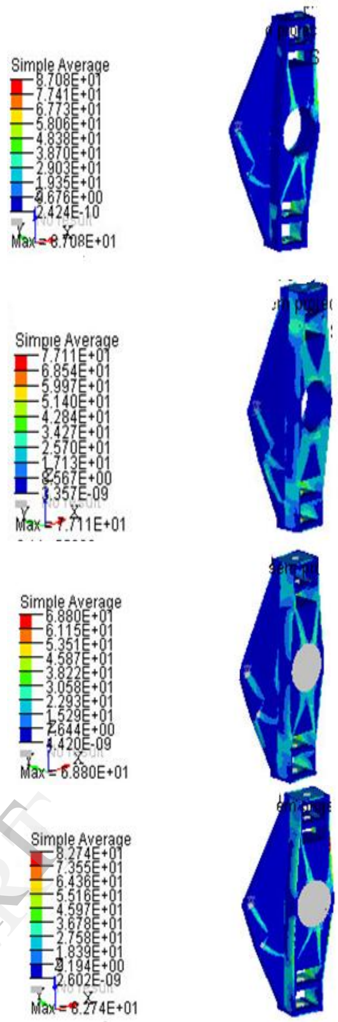

Fig. 6. Several iterations using various topological optimization techniques before arriving at the final iteration

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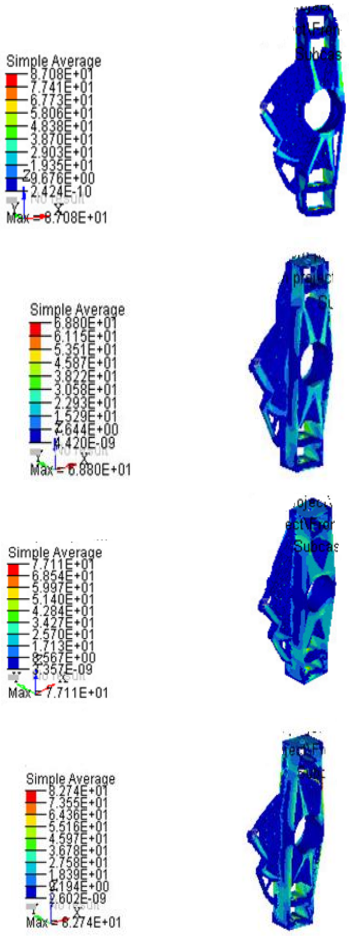

Fig. 7. The elemental densities where the material can be removed is shown through the process of optimization at various iterations

B. Steel Upright

The raw blank of steel was taken satisfying the geometry requirements which has very high weight. A raw run of analysis is carried out to see the stress contours. Several iterations are carried out and based on the iterations the designing is done by providing material where required. The philosophy is to use sheet metal of thickness 2mm to make the upright instead of machining from raw blank of steel. Again analyses are carried out to satisfy the stress contours and decrease of stress if any.

Optimization is done on the reminder to get the final required form. The final form is reconstructed in the CATIA software to generate the final CAD model.

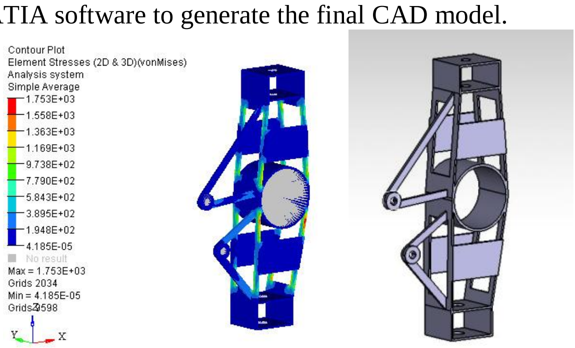

Fig. 8. Steel 1st iteration (CAD model on the right); max stress= 1753MPa weight 0.72kg

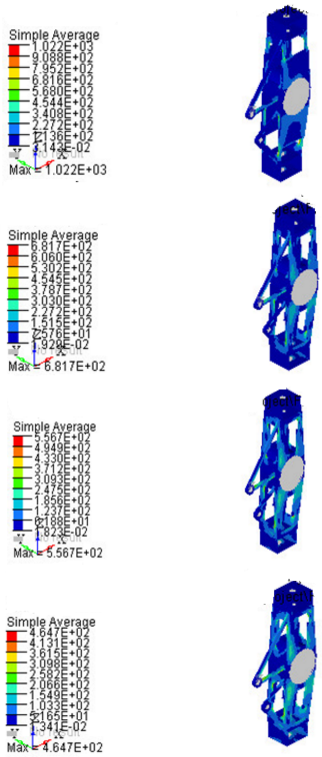

Fig. 9. Series of manual optimization iterations done with decreasing max stress value and increase in weight

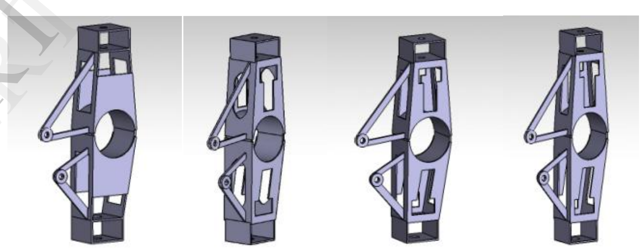

Fig. 10. Steel Upright 2nd, 3rd, 4th, 5th, 6th iterations CAD models respectively with weights as follows 0.817kg, 0.851kg, 0,851kg, and 0.865Kg

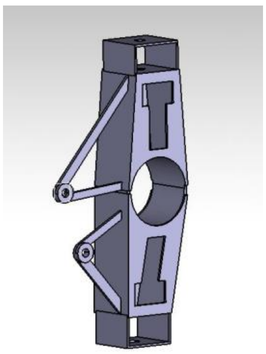

Fig. 11. The last manual iteration of the steel optimization Max. stress 464MPa weight= 0.957kg

V. CONCLUSION

TABLE II. COMPARISION OF ALUMINIUM AND STEEL UPRIGHTS

	Aluminium (6061 T6)	Steel
Max Stress/ Yield Stress (MPa)	164/ 276	464/ (560-700)
Weight (Kg)	0.854	0.957

From the analysis it can be concluded that the steel upright can definitely replace aluminium one in student projects without effecting the performance and making the cars very economical. If proper fixtures are made, the steel upright would resemble close tolerances as the aluminium one. The aluminium alloy (Al 6061 T6), though is easy to find costs more than steel. Lot of material is wasted during the CNC machining which cannot be recycled to the same alloy, so a lot of energy is wasted indirectly (the energy required to produce the alloy). By still refining the steel upright it can be made with better strength to weight ratio.

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