Optimization of chassis for a solar powered vehicle

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Abstract. This study aims at designing and analysis of roll cage for an electric solar vehicle fabricated for Imperial Society for Innovative Engineers ESVC (Electric Solar Vehicle Competition) 2019. The vehicle will be powered by electric motor and charging will be done by solar panels which makes the design eco-friendly. This study focuses on the development of an optimized roll cage structure which will include the compactness of frame design, light weight and driver’s safety considering all the design ergonomics and the driver’s safety measures. Optimization is based on reducing weight, strength considerations and ergonomics consideration. Variation of materials, pipe diameters and 3-D structure is implemented during analysis. Analysis of roll cage is conducted using SolidWorks and loading conditions are taken as per standards. Three-wheel tadpole configuration is used for analysis purpose. Results are presented in form of graphs and SolidWorks analysis outputs.

1. Introduction
1.1. Literature Review
The objective of the study is to design, develop and fabricate the roll cage of an electric solar vehicle for ESVC 2019 in accordance with the rules specified by the ISIE. It is designed to compensate the swelling demand of the conventionally powered vehicles. The roll cage designed is light weight, in accordance with the driver’s safety and adds to the aesthetics of the car. The vehicle should a stable and a strong frame for which various materials are taken into consideration out of which chromoly (AISI 4130) is found to be suitable. The Finite Element Analysis (FEA) of model is done in Solid works 2018. Later, various loading conditions are applied on the design for static simulation to test all modes of failure. Based on the results obtained from these tests, the design is modified accordingly. The FEA of the roll cage needs to be done because of the failure of structural members which can lead to accidents. The roll cage should act as the central mounting structure for all the components. Conventionally, roll cage is a truss welded at joints. The chassis should withstand high energy transfer waves while acceleration and deceleration. This makes the chassis design more compatible and driver safe against accidental situations [1]. For designing a roll cage all the loads acting on it need to be considered. The main loads acting on an automobile’s roll cage are given by C.D. Naiju, Anamalai K. et. al. in [2] as: (i) Longitudinal torsion (ii) Vertical bending (iii) Lateral bending (iv) Horizontal lozenging. Torsional load acts on the opposite corners of the roll cage. Suspension loads acts on the two ends of the car considering the frame to be a torsional spring. Torsion study has been demonstrated with experimental proofs by W.B. Riley and A. R. George in [3]. Deflections decreases with increase in stiffness when the back of the chassis was loaded according to a study on static analysis by A. Ryan in [4]. FEA has highly improved both the standards of engineering designs and the methodology of the design process in many industrial applications. Jason Denny et.al. in [5] researched on the development of a monocoque chassis.
through an iterative FEA process to develop a solar car. The methodology includes an iterative process considering various geometries and changes, with the aim of achieving a single sustainable frame. The suspension mounting points are also analyzed to ensure their rigidity under concentrated loading. A. K. Das et.al. [6] studied emerging growth of population and need to find and alternative source of energy in automobile industry. He worked on the design and fabrication of solar powered car and its performance evaluation. Nader A. Nader et. al. [7] also worked on design and building of Solar Car considering lightweight, aerodynamics and efficiency as major components while designing. P.S. Sreeragh et. al. [8] have worked on the designing of a chassis to fulfill rules and guidelines as indicated by ISIE for ESVC. Suraj Aru et. al. [9] have worked on the design of tubular space frame also called as the roll cage of a vehicle using FEA.

1.2. Problem Statement
The weight of roll cage is an important factor in designing. The overweight of the roll cage might result in reduced performance of the vehicle. Also, keeping in mind the rules as established by ISIE, the size of the vehicle has been reduced. Reduction in the size of the vehicle and procuring maximum power from the solar cells considering less weight of the roll cage along with driver’s safety needs to be addressed. For reducing weight of the vehicle suitable material has been used without hampering the strength to weight ratio. For this purpose, AISI 4130 has been used with minimized dimensions. Also, to increase the power output from the solar cells optimization in the roll cage has been done by increasing the area for the solar cells. The frontal area has been increased by giving an inclination of 17° to the frontal surface of the roll cage with respect to the top plane.

2. Objective
- To design the roll cage structure considering the driver’s safety.
- To design roll cage in accordance with the rulebook of ISIE ESVC 2019.
- To reduce weight of roll cage.
- To obtain maximum power output from the solar cells by optimizing the roll cage.
- To optimize the roll cage by selecting suitable material.
- To use the pipe with minimized dimensions.
- To design a vehicle for driver’s ergonomics which considers driver’s comfort, optimal placement of steering wheel, pedal positions, spacious cockpit.
- To avoid the failure in the front collision, rear collision, side collision, rolling and torsion in rough conditions.

3. Material selection
Various materials are taken into consideration for FEA of roll cage. The parameters considered while material selection are their weight, strength, reliability, availability and manufacturability. Initially, AISI 4130, AISI 6063, AISI 6061 and AISI 1020 are taken. Every material has different properties and advantages. Properties of various materials are detailed in table 1.

| Properties            | AISI1020   | AISI4130   | AA6063 T6   | AA6061 T6   |
|-----------------------|------------|------------|-------------|-------------|
| Yield Strength        | 351.57 MPa | 460 MPa    | 214 MPa     | 276 MPa     |
| Ultimate Strength     | 420 MPa    | 670 MPa    | 241 MPa     | 310 MPa     |
| Density               | 7900 kg/m³ | 7850 kg/m³ | 2700 kg/m³ | 2700 kg/m³ |
| Young’s Modulus       | 205 GPa    | 205 GPa    | 68.9 GPa    | 70 GPa      |
| Moment of Inertia     | 2.027*10⁸ m⁴ | 2.027*10⁸ m⁴ | 1.263*10⁷ m⁴ | 1.263*10⁷ m⁴ |
Frontal impact loading condition is considered while selecting the material. Various pipes with different properties are selected for analysis and based on the calculations and their results the final dimensions and material is finalized. According to the constraints in the rulebook the vehicle can be driven at a speed of not more than 60kmph. The following calculation are included by Abhinav Sharma et. al. in [10].

\[ W_{net} = Net\ Work\ Done,\ F = Force,\ d = Distance\ Travelled \]

Now,

\[ W_{net} = \frac{1}{2} m v_{final}^2 - \frac{1}{2} m v_{initial}^2 \quad (1) \]

\[ W_{net} = | - \frac{1}{2} m v_{initial}^2 | \quad (2) \]

But,

\[ W_{net} = Impact\ force \times d \quad (3) \]

For static analysis, vehicle is considered to come in rest 0.1 seconds next to the collision. If a vehicle moves at a speed of 60kmph, it will travel a distance of 1.66m after the impact by Oturkar Sania et.al. in [11]. From the above equations, it is found to be:

\[ Impact\ force = \frac{1}{2} \times 235 \times (16.66)^2 \times \frac{1}{1.66} \]

\[ Impact\ force = 19632.852\ N \]

Therefore, the maximum force to be applied is 19632.852 N. Assuming this force to be 20KN. Thus, material for roll cage should has max. yield point than the impact force. Frame has been analysed for rollover and collision. The design has been made in such a way that there are no redundant links in the frame for reduced weight without reducing the strength of the frame. Above discussed materials with all dimensions are taken for FEA and several results are obtained. Material is selected after analysing different materials with different dimensions for the values of FOS and various stresses acting on the frame. Below is the tabular representation of the results (all results are based on static studies on SOLIDWORKS-2018) obtained:

**Table 2. Results of simulation done on AISI 4130.**

| O.D. | Thickness  | 1mm | 1.25mm | 1.5mm |
|------|------------|-----|--------|-------|
| 1 inch | FOS: 1.2 | Stress: 1.2×10^2 N/m² | Stress: 1.23×10^2 N/m² | Stress: 1.5 | Weight: 31.08 kg | Weight: 37.60 kg | Weight: 43.98 kg |
| 1.5 inch | FOS: 2.6 | Stress: 7.54×10^4 N/m² | Stress: 7.48×10^4 N/m² | Stress: 3.2 | Weight: 44.71 kg | Weight: 54.65 kg | Weight: 54.45 kg |
| 2 inch | FOS: 4.6 | Stress: 1.009×10^8 N/m² | Stress: 8.17×10^7 N/m² | Stress: 7.3 | Weight: 57.88 kg | Weight: 71.11 kg | Weight: 84.17 kg |

**Table 3. Results of simulation done on AISI 1020.**

| O.D. | Thickness  | 1mm | 1.25mm | 1.5mm |
|------|------------|-----|--------|-------|
| 1 inch | FOS: 0.88 | Stress: 9.36×10^3 N/m² | Stress: 1.65×10^3 N/m² | Stress: 1.2 | Weight: 31.27 kg | Weight: 37.84 kg | Weight: 44.26 kg |
| 1.5 inch | FOS: 2 | Stress: 7.23×10^4 N/m² | Stress: 8.56×10^4 N/m² | Stress: 2.5 | Weight: 44.99 kg | Weight: 55 kg | Weight: 64.86 kg |
Table 4. Results of simulation done on AA 6063-T6.

| Thickness | 1mm        | 1.25mm     | 1.5mm      |
|-----------|------------|------------|------------|
| 1 inch    | FOS: 0.54  | FOS: 0.66  | FOS: 0.76  |
|           | Stress: 1.27×10^2 N/m^2 | Stress: 1.22×10^2 N/m^2 | Stress: 7.51×10^3 N/m^2 |
|           | Weight: 10.69 kg | Weight: 12.93 kg | Weight: 15.13 kg |
| 1.5 inch  | FOS: 1.2   | FOS: 1.5   | FOS: 1.8   |
|           | Stress: 7.05×10^4 N/m^2 | Stress: 9.14×10^4 N/m^2 | Stress: 7.43×10^4 N/m^2 |
|           | Weight: 15.38 kg | Weight: 18.80 kg | Weight: 22.17 kg |
| 2 inch    | FOS: 2.1   | FOS: 2.5   | FOS: 3     |
|           | Stress: 2.56×10^4 N/m^2 | Stress: 2.6×10^4 N/m^2 | Stress: 1.51×10^4 N/m^2 |
|           | Weight: 19.91 kg | Weight: 24.46 kg | Weight: 28.95 kg |

Table 5. Results of simulation done on AA-6061-T6.

| Thickness | 1mm        | 1.25mm     | 1.5mm      |
|-----------|------------|------------|------------|
| 1 inch    | FOS: 0.69  | FOS: 0.85  | FOS: 0.98  |
|           | Stress: 1.14×10^2 N/m^2 | Stress: 1.07×10^2 N/m^2 | Stress: 9.28×10^3 N/m^2 |
|           | Weight: 10.69 kg | Weight: 12.93 kg | Weight: 15.13 kg |
| 1.5 inch  | FOS: 1.6   | FOS: 2     | FOS: 2.3   |
|           | Stress: 8.28×10^4 N/m^2 | Stress: 1.11×10^3 N/m^2 | Stress: 4.24×10^4 N/m^2 |
|           | Weight: 15.38 kg | Weight: 18.80 kg | Weight: 22.17 kg |
| 2 inch    | FOS: 2.7   | FOS: 3.2   | FOS: 3.9   |
|           | Stress: 2.95×10^4 N/m^2 | Stress: 2.44×10^4 N/m^2 | Stress: 1.84×10^4 N/m^2 |
|           | Weight: 19.91 kg | Weight: 24.46 kg | Weight: 28.95 kg |

From the above inferences, it can be said that a vehicle with a maximum moving velocity of 60kmph can withstand an impact load of 20KN when material is selected appropriately considering the above results, AISI 4130 with dimensions of 1inch outer diameter and 1.25mm thickness is best suited for designing the roll cage.

4. Design Procedure

The design procedure is combined process of synthesizing older designs and developing new ideas. Initially, a basic prototype is considered for FEA and further development on the model is done based on the results of FEA. The structure is then tested to obtain some useful evidence to finalize the frame after synthesis. The members in the roll cage are used to distribute load across the frame under any impact and to transfer energy from the source. Also, the cage provides drivers safety in case of roll over. Roll Cage is modelled using Solid Works-2018 software (tab.6). The model is then subjected to various loading conditions by FEA. Based on the results derived from FEA, the roll cage is finalized. Appropriate material is selected to epitomize the roll cage. Various data regarding the material properties is entered.

Table 6. Different views of rolledge.
5. Finite Element Analysis

Using solid modelling and FEA in SolidWorks 2018, an optimized structure with maximum strength, maximum solar output and minimized weight is designed. Finite element of the cage is subjected to analysis which is of two main types:

5.1. Static Analysis
Under this analysis, chassis is subjected to various loads and impact tests i.e. front impact test, rear impact test, roll over test, torsional tests etc. for static conditions with suitable constraints as per different conditions. In static analysis, vehicle is subjected to maximum impact loads under worst conditions.

5.2. Dynamic Analysis
Under this analysis, vehicle is analysed under moving conditions. It is carried out for head-on collision as well as front impact, rear impact, side impact and roll over. Vehicle is tested for maximum moving velocity.

6. Calculation of various impact forces under static analysis

6.1. Front Impact
For front impact analysis, it is assumed that the car hits the wall at a moving velocity of 60kmph and comes to rest in 0.1 seconds by Oturkar Sania et.al. in [11] and Bharat Kumar Sati et. al. [12].

Using Work-Energy Theorem:
From eq. (1) and (2),
\[
W_{\text{net}} = \frac{1}{2}mv_{\text{initial}}^2
\]
\[
W_{\text{net}} = \frac{1}{2} \times 235 \times (16.66)^2
\]
\[
W_{\text{net}} = 32612.783 \text{ N.m}
\]

Now,
\[
W_{\text{net}} = F_{\text{front}} \times d
\]

5
\[ d = \text{impact time} \times v_{\text{initial}} \]  \hspace{1cm} (7)
\[ d = 0.1 \times 16.66 \]  \hspace{1cm} (7)
\[ d = 1.66 \, \text{m} \]
\[ F_{\text{front}} = \frac{W_{\text{net}}}{d} \]  \hspace{1cm} (8)
\[ F_{\text{front}} = 19575.49 \, \text{N} \]

6.2. Rear Impact
For rear impact analysis, it is assumed that a solar vehicle is hit by another moving solar vehicle. The vehicle comes to rest after 0.2 seconds by Bharat Kumar Sati et. al. [7]. Using Work-Energy Theorem in eq. (1) and (2),
\[ W_{\text{net}} = \frac{1}{2}mv_{\text{initial}}^2 \]
\[ W_{\text{net}} = \frac{1}{2} \times 235 \times (16.66)^2 \]
\[ W_{\text{net}} = 32612.783 \, \text{N.m} \]
Now,
\[ W_{\text{net}} = F_{\text{rear}} \times d \]  \hspace{1cm} (9)
Also,
From eq. (7),
\[ d = 0.2 \times 16.66 \]  \hspace{1cm} (7)
\[ d = 3.332 \, \text{m} \]
\[ F_{\text{rear}} = \frac{W_{\text{net}}}{d} \]  \hspace{1cm} (10)
\[ F_{\text{rear}} = 9787.75 \, \text{N} \]

6.3. Side Impact
For side impact analysis, it is assumed that another vehicle hits the vehicle from sides. The vehicle comes to rest after 0.2 seconds by Bharat Kumar Sati et. al. [8].
From Work-Energy Theorem:
From eq. (5),
\[ W_{\text{net}} = \frac{1}{2} \times 235 \times (16.66)^2 \]
\[ W_{\text{net}} = 32612.783 \, \text{N.m} \]
Now,
\[ W_{\text{net}} = F_{\text{side}} \times d \]  \hspace{1cm} (11)
Also,
From eq. (7),
\[ d = 0.2 \times 16.66 \]  \hspace{1cm} (7)
\[ d = 3.333 \, \text{m} \]
\[ F_{\text{side}} = \frac{W_{\text{net}}}{d} \]  \hspace{1cm} (12)
\[ F_{\text{side}} = 19575.49 \, \text{N} \]

6.4. Roll Over
For roll over analysis, car is considered to hit to the ground from a height of 3 meters. The roll over is tested for the strength of roll hoop. The vehicle comes to rest in 0.1 seconds by Bharat Kumar Sati et.al.[8].
During fall,
Potential Energy of the vehicle = Kinetic Energy of the vehicle
\[
\frac{1}{2} m v^2 = m \cdot g \cdot h
\]  \hspace{1cm} (13)

On solving above equation,
\[
v = \sqrt{\frac{2 \cdot g \cdot h}{m}}
\]  \hspace{1cm} (14)

\[
v = \sqrt{2 \times 9.81 \times 3}
\]
\[
v = 7.67 \text{ m/sec}
\]
\[
d = t \times v(15)
\]
\[
d = 0.1 \times 7.67
\]
\[
d = 0.76 \text{ m}
\]

Now,

By Work-Energy Theorem,
From eq. (3)
\[
W_{net} = \frac{1}{2} \times 235 \times (7.67)^2
\]

6.5. Force under torsion

For torsional analysis, acceleration is taken to be 2 times of the gravitational force (a=2.g).

Using Impulse-Momentum Theorem;
\[
F_{torsional} = m \cdot a
\]  \hspace{1cm} (16)

\[
F_{torsional} = 235 \times (2 \times 9.81)
\]
\[
F_{torsional} = 4610.70 \text{ N}
\]

7. Analysis of Chassis

7.1. Front Impact Analysis

For frontal impact we have fixed the geometry at the A-arm and swing arm mounting points. Load of 20KN is applied at the frontal crash zone of the vehicle. All the analysis done in SOLIDWORKS have been shown below (fig. 5-7):

The simulation shows below mention results:

Results for Frontal Impact Test

| Metric            | Value   |
|-------------------|---------|
| Min. FOS          | 1.4     |
| Von Mises Stress  | $1.38 \times 10^5$ N/mm$^2$ |
| Max. Displacement | 2.14 mm |
7.2. Side Impact Analysis
For side impact we have constrained the opposite side of crash zone. Load is applied and the simulation done on SOLIDWORKS are shown as under (fig. 8-10):

The results of simulation are shown as under:

Results for Side Impact Test
Min. FOS 1.2
Von Mises Stress $1.308 \times 10^5$ N/mm$^2$
Max. Displacement mm

7.3. Roll Over Analysis
In this case the vehicle is assumed to be front rolling and all mounting points are fixed. Also, after rolling the vehicle tends to slide therefore a frictional force acting on the point of contact of the chassis and the ground is considered. Simulation done on SOLIDWORKS are shown as under (fig. 11-13):
7.4. Torsional Analysis
For torsional analysis, rear part of the chassis is fixed and load is applied conjugate on either side. The simulation done on SOLIDWORKS are shown as under:

- **Figure 14.** FOS (Torsional Analysis).
- **Figure 15.** Displacement (Torsional Analysis)
- **Figure 16.** Stress (Torsional Analysis).

The results of the simulation are shown as under:

**Results for Torsional Test**
- Min. FOS: 1.2
- Von Mises Stress: $1.308 \times 10^5$ N/mm²
- Max. Displacement: mm

8. Results and Conclusion
After analysing the results, following inferences are derived:
- FOS is an important factor while deciding material for designing the roll cage.
• The pipe for manufacturing cage should be such that it can bear load under critical condition.
• The dimensions should be chosen appropriately such that it should not increase weight of the frame.

Considering FOS and pipe dimensions, we have concluded the use of AISI 4130 (chromoly) for manufacturing chassis with optimized dimensions with maximum strength and considerable weight. After simulation the FOS for the wireframe is observed to be 1.5 which is nominal and optimized. The dimension of the pipe is finalized with an iterative process where the chassis is analyzed under various pipe dimensions out of which pipe with 1 inch outer diameter and 1.25 mm thickness is selected. Also, the cost of chromoly is less as compared to other considered materials.

9. References
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