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## Research Article

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# The design process of the chassis of a prototype vehicle for Shell Eco-marathon

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## Abstract

This paper presents the design, modelling, and fabrication procedures for the chassis of a prototype car with the aim of achieving the objectives: rigid and high strength chassis, reduced vehicle weight, driver safety, and an energy efficient vehicle. The novelty of this work is that it demonstrates how aluminium alloy could be used to construct non-integrated chassis for super mileage vehicles through a load-stress calculation model. Furthermore, a method of Finite Element Analysis (FEA) was presented which achieves the same result as the analytical calculation. The work also presents a method of fabricating aluminium square tube for the chassis by joining two angle bars. The results show that the FEA approach agrees with the analytical design model thereby reducing the time consumed in conceptual design process. The fabricated chassis was found to show no cracks and it was able to resist bending and shear from the loads acting on the vehicle in line with the design data. The energy efficiency recorded from the first test-run of the vehicle was 250 km/L of gasoline. The methods presented could be characterized as accurate and reliable for manufacturers of super mileage racing cars for Shell Eco-marathon events.

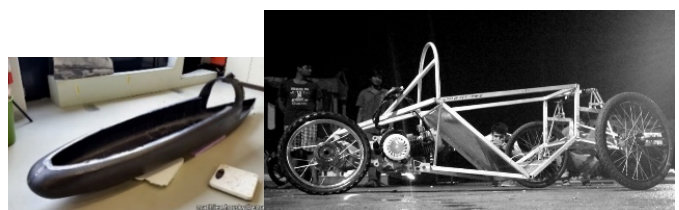
**Keywords:** aluminium chassis; chassis design and fabrication; finite element analysis; prototype car; Shell Eco-marathon

## 1. Introduction

The chassis of a vehicle is the overall load bearing component of the vehicle. It is important that the chassis be designed to have the strength required to withstand the different types of loads that act on the vehicle such as the compressive, tensile, torsional, and bending loads. The chassis should also be designed bearing in mind the safety of the driver in case of any impact on the vehicle.

Several materials have been used in the design and fabrication of the chassis of prototype racing cars. However, due to the energy efficiency objective of most super mileage competitions, such as the *Shell Eco-marathon (SEM)* [1], design engineers have explored several options in their choice of materials for the chassis. Also, there are two major types of chassis used by racing teams in the manufacture of their racing cars for SEM competitions, namely the integrated or monocoque chassis, and the non-integrated chassis as shown in Figure 1. Tomar et al. [2] worked on the efficient design of a super mileage low-cost vehicle chassis using layered laminate natural bamboo composite material. They employed the finite element method to ascertain the crash impact of the vehicle. Simoes [3] presented a paper where a prototype vehicle chassis was manufactured by integrating the chassis and the cockpit structure using advanced sandwich composite materials moulded with CAD/CAM technology. In his work, a sandwich structure consisting of a foam core material combined with high strength skins was the solution for the manufacture of the bodywork of the vehicle. There are also evidences to show that

steel material has been used by various Nigerian teams in the past for the manufacture of their chassis as in [4,5]. It should be noted, however, that the type of chassis in [4,5] was non-integrated. For non-integrated chassis type such as the one designed in this work, aluminium has greater advantage than steel in terms of weight reduction, strength, and rigidity, as shall be discussed in the next section. But there is no work in literature to demonstrate the design and use of aluminium alloy for the manufacture of non-integrated chassis for prototype super mileage cars; hence, this paper will help to fill this gap.



**Figure 1.** Integrated or monocoque chassis (left) [6], and non-integrated chassis (right) [7].

On the other hand, there has been a need for lower conceptual design costs within a shorter time frame, and there has also been a need for new modeling techniques, which has led automotive manufacturers to find better ways to develop vehicles. Simulation through finite element analysis (FEA) is a cost and time efficient way to develop vehicle chassis while utilizing the saved time for innovative product development [8]. The modeling methodology of chassis is another important advantage associated with using FEA as it helps in studying the

structure of the chassis in details.<sup>9</sup> The works done by [8, 10-12] using FEA have proved that their chassis meet the strength requirements under standard deformation modes, without having to spend time calculating the exact loads which cause this deformation.

There are several stages involved in the design and construction of the various components of the prototype vehicle of our team. However, this paper only presents the design, analysis, fabrication, and testing of a major component of the vehicle—the chassis. In other words, the purpose of this paper is to present accurate design and modelling procedure for the chassis of a Shell Eco-marathon prototype car made from aluminium, and to achieve the following objectives: Rigid and high strength chassis, reduced vehicle weight in line with the SEM rules, driver safety, and an energy efficient vehicle.

The novelty of this work is that it first demonstrates how aluminium alloy can be used to construct non-integrated chassis for super mileage (or SEM) vehicles by outlining a load/stress calculation model based on an analytical design approach. It then shows how FEA method can be used to achieve the same design thereby reducing the time consumed in conceptual design process. Lastly, this work introduces a method of joining two angle cross-sections of aluminium bars to form a square pipe by electric arc welding technique—this will help in situations where aluminium square pipes are not readily available for SEM teams in Sub-Saharan Africa.

## 2. Material Selection

The material for the chassis construction is selected while bearing in mind the major objective of the Shell Eco-marathon (SEM) competition—energy efficiency. Hence, one of the key considerations for choosing an energy efficient material is its strength-weight ratio. Other factors that are considered in the design of an SEM prototype vehicle chassis, based on their importance to the overall vehicle performance, are safety, durability, cost, maintainability, and ease of assembly.

With the combination of favourable properties found in aluminium, modern day automobile manufacturers are increasingly turning to aluminium for use in the fabrication of automobile chassis. Most importantly, with high strength aluminium alloys, a lightweight chassis with an equal strength when compared to its steel counterpart can be obtained, noting that the density of aluminium is about one-third of that of steel [13]. Also, aluminium has good corrosion resistance property, and it is environmentally friendly. Hence, for this prototype vehicle chassis design and construction, the Aluminium Alloy 6063 was used. Its mechanical properties as seen in Table 1 show that its strength is high enough to satisfy the torsion and stiffness requirements of the chassis [14]. For this design, aluminium alloy 6063-T6, which has the highest tensile and yield strengths, is selected.

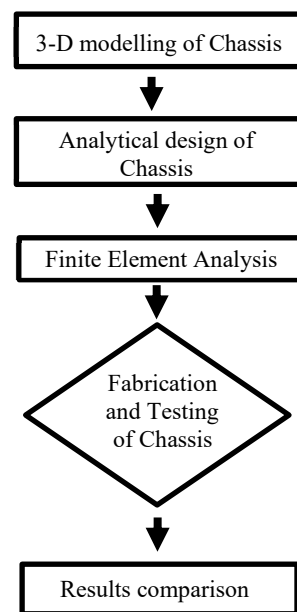
**Table 1.** Mechanical Properties of Aluminium Alloy 6063 [15]

Alloy & Temper	Diameter or Thickness (mm)	Tensile Strength (MPa)	Yield Strength 0.2% Proof (MPa) min.	Elongation (%) ( $GL = 50 \text{ mm or } 5.65\sqrt{A}$ ) min.
6063-O	All	130 max	-	16
6063-T1	Up to 12.0	115 min	60	12
	12.0 – 25.00	110 min	55	10
6063-T5	Up to 12.0	150 min	110	8
	12.0 – 25.00	145 min	105	6
6063-T6	Up to 25.00	240 max	215 max	8

*GL* = gauge length; *O* = un-heat treated alloy, *T* = tempered alloy series.

## 3. Chassis Design Methodology

The chassis was designed while bearing in mind some technical design rules contained in the 2019 Shell Eco-marathon Global Rules Chapter I [16]. These rules only provide acceptable limits for the dimensions of the vehicle. However, it is the sole responsibility of the design engineer to develop his/her own design approach for the success of the project. Our design approach is as described in the flowchart of Figure 2.



**Figure 2.** Flowchart of the design methodology

### 3.1. 3-D Modelling of the Chassis

A 3-D model was developed using CAD software as shown in Figure 3. The chassis was modelled with the weldment feature—structural member using the ISO standard square tube, with dimensions 25mm x 25mm x 4mm (4mm being the thickness of the material). In this model, the front section of the chassis (from the side positions of the driver’s seat) was

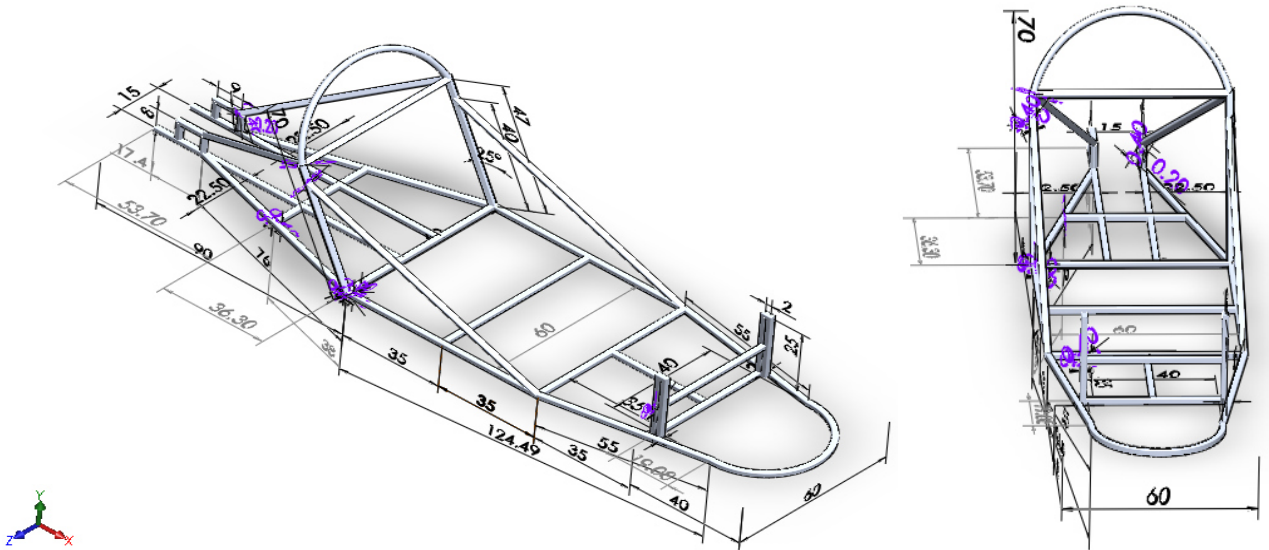


Figure 3. 3-D model of the prototype chassis (dimensions in cm)

designed such that it narrows down to the fore front in order to accommodate the aerodynamic structure of the vehicle's body. The front wheels and steering rod slots are also incorporated in the front design of the chassis to accommodate the corresponding systems. The rear section of the chassis also narrows down backwards from the driver's seating position to also accommodate the body's design and shape. The chassis has an inclined roll bar and provisions are made to accommodate the single rear wheel as required by the SEM design rules. The engine position which lies behind the inclined roll bar (just after the driver's seating position) has extra reinforcements and beams to provide rigid support for the 15 kg engine and electrical components. The two front wheels are located outside the body while every other component for the vehicle propulsion and control are located inside the body. However, the scope of this paper doesn't include the design of the vehicle's body and other systems, such as the brakes and steering, engine and transmission, electrical and electronics, aerodynamic body systems, etc.

### 3.2. Analytical Design of the Chassis

#### 3.2.1. Bending stress of chassis

The bending stress of a member is the resistance offered by its internal stresses to resist bending, while the shear stress of a member is the resistance offered by its internal stresses to resist shear. Figure 4 depicts the cross-section of the aluminium square tube used for the construction of the chassis. The tube is 4 mm thick; for design purpose, the base of the chassis is considered as a simply supported beam. Let the length of the external cross-section of the beam be  $b$ , the length of the internal cross-section be  $h$ , and the moment of inertia of the cross-section about the neutral axis of the chassis beam be  $I$  [17], then

$$I = \frac{b^4 - h^4}{12} \quad (1)$$

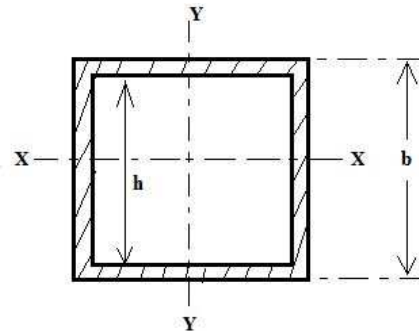


Figure 4. Cross-section of the Aluminium square tube

Next, the bending moment  $M$  acting on the cross-section of the base of the chassis [denoted by equation (2)] is calculated by assuming the chassis to be a simply supported beam at its rear and front wheel axles, denoted respectively as  $X$  and  $Y$ , and having a total length  $L$ , with a total load  $F$  acting on it, as shown in Figure 5 and detailed in Table 2.

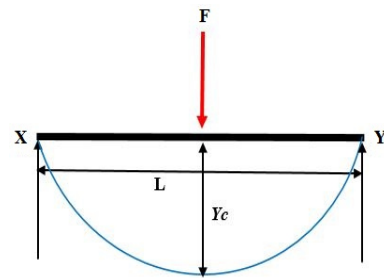


Figure 5. Beam pivoted at both ends with a central load

**Table 2.** Loads distribution across the chassis

Loads	Value
Load of engine and rear compartment components	400 N
Load of the driver	550 N
Load of the vehicle body	150 N
Load of steering assembly	150 N
Total Load $F$	1250 N

The total load  $F$  acting on the chassis consists of the weight of the engine and other rear components (400 N), the weight of the driver (550 N), the weight of the vehicle body (150 N), and the total weight of the steering assembly (150 N). Therefore,  $F = 1250$  N; and  $L$  is the total length of the chassis, which was measured to be 2.35m.

$$M = \frac{FL}{4} \quad (2)$$

The bending stress of the chassis can be calculated analytically using equation (3)

$$\frac{M}{I} = \frac{\sigma}{y} = \frac{E}{R} \quad (3)$$

where  $\sigma$  = bending stress;  $I$  = moment of inertia of the cross-section about the neutral axis;  $y$  = distance from the neutral axis to the extreme fibre (where  $y = \frac{b}{2}$ );  $E$  = young's modulus of the chassis material (aluminium); and  $R$  = radius of curvature of the chassis beam.

### 3.2.2. Shear stress of chassis

The shear stress acting on the chassis is set up by the torsion in the rotating members of the vehicle. This is known as torsional shear stress. It is zero at the centroidal axis of the chassis and maximum at the outer surface [18]. Knowing that the maximum torque from the engine is  $T = 4.10$  Nm, the maximum torsional shear stress on the chassis was calculated using equation (4)

$$\tau = \frac{T \times y}{J} \quad (4)$$

where  $\tau$  = torsional shear stress induced at the outer surface of the chassis beam or maximum shear stress;  $y$  = distance from the neutral axis to the extreme fibre;  $J$  = polar moment of inertia (where  $J = I_{XX} + I_{YY}$ ). For a square tube,  $I_{XX} = I_{YY} = I = 73.37 \times 10^{-9} \text{ m}^4$ .

As aluminium alloy is a ductile material, the von-Mises equivalent stress criterion used as a failure determinant and is given as

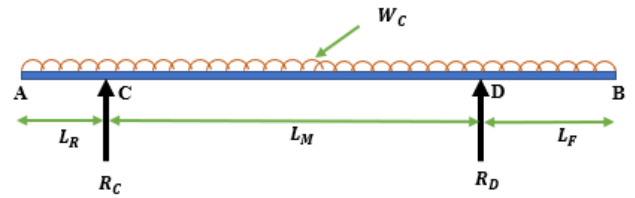
$$\sigma_v = \sqrt{\sigma^2 + 3\tau^2} \quad (5)$$

### 3.2.3. Deflection of chassis

Deflection is an important factor that should be considered in load bearing structures such as beams and cantilevers. Whenever a beam or a cantilever is loaded, it is observed to deflect from its original position. The deflection of beams and cantilevers is dependent on two important design criteria, namely strength and stiffness. The strength and stiffness of the beam should be adequate in order to not deflect to an unsafe extent under the action of the load [17].

As already known from Table 2, the vehicle's chassis is subjected to loads from the engine and rear compartment, the steering assembly, the vehicle body, and the weight of the driver. The resulting total load on the chassis is assumed to be uniformly distributed across the total span,  $(L_R + L_M + L_F)$ , of the chassis as shown in Figure 6; where the reactions  $C$  and  $D$  denote the positions of the rear and front wheel axles, respectively. The maximum downward deflection of the chassis with the uniformly distributed load  $W_C$  is given by equation (6).  $W_C$  is obtained by dividing the total load on the chassis by the total chassis span  $(L_R + L_M + L_F)$ .

$$\delta = \frac{5W_C(L_R+L_M+L_F)^4}{384EI} \quad (6)$$

**Figure 6.** Loaded Chassis schematic

The 3D model of the chassis (from Figure 3) and Table 3 provide the data used for these calculations.

**Table 3.** Data used in calculating the chassis deflection

Definition	Value
Total distributed load $W_C$	531.91 N/m
Length of chassis before the rear axle $L_R$	0.17 m
Length of chassis between the rear and front axles $L_M$	1.78 m
Length of chassis before the front axle $L_F$	0.40 m

### 3.2.4. Shear force and bending moment calculations

1. Determination of reaction forces on the wheel axles:

From Figure 6, the chassis is considered as a beam, simply supported at points  $C$  and  $D$ , which connote the rear and front wheel axles, respectively. Let  $R_C$  and  $R_D$  represent the reactions

on the rear and front wheels, respectively. At equilibrium condition,

$$R_C(L_R) + R_D(L_M + L_R) = W_C(L_R + L_M + L_F) \cdot \frac{(L_R + L_M + L_F)}{2} \quad (7)$$

$$R_C = W_C(L_R + L_M + L_F) - R_D \quad (8)$$

Substituting for  $R_C$  in equation (7) gives the reaction at the front wheel axle,  $R_D$ , from which we calculate the reaction at the rear wheel axle,  $R_C$ , by substituting back in equation (8).

## 2. Shear Force Calculations:

$$\text{Shear Force just to the left of } C: F_{C1} = -(W_C \times L_R) \quad (9)$$

$$\text{Shear Force just to the right of } C: F_{C2} = +(R_C - F_{C1}) \quad (10)$$

$$\text{Shear Force just to the right of } D: F_{D1} = +(W_C \times L_F) \quad (11)$$

$$\text{Shear Force just to the left of } D: F_{D2} = -(R_D - F_{D1}) \quad (12)$$

The Shear Force is zero at midpoint between  $C$  and  $D$ .

## 3. Bending moment calculations:

$$\text{Bending Moment at } C: M_C = -\frac{W_C(L_R)^2}{2} \quad (13)$$

$$\text{Bending Moment at } D: M_D = -\frac{W_C(L_F)^2}{2} \quad (14)$$

The Bending Moment is maximum at the point of zero shear, i.e., at midpoint between  $C$  and  $D$ , and is,

$$M_{max} = R_C \left( \frac{R_C}{2W_C} - L_R \right) \quad (15)$$

Figure 7 presents the resulting reaction forces, shear force and bending moment diagrams of the loaded chassis.

### 3.3. Finite Element Analysis (FEA) of the Chassis

It is necessary to perform static analysis on the vehicle's chassis in order to ascertain the structural safety and strength of the chassis. The static analysis was done using the finite element method as it is an effective and efficient approach. The finite element analysis as described in the following subsections.

#### 3.3.1. Meshing

The type of mesh applied was the Beam Mesh. The maximum and the minimum dimensions of the mesh were set at 8 and 2 mm respectively, so as to include a great number of elements inside the model. Meshing was created in every single part separately. Regions which were determined to experience high changes in stress require a higher finite element mesh density than those which were determined to experience little or no stress variation. Also, points of interest such as holes,

corners or fillets were meshed carefully. A volume mesh was also created by employing the already created surface mesh as a base. Total meshed elements and nodes were 604 and 644 respectively.

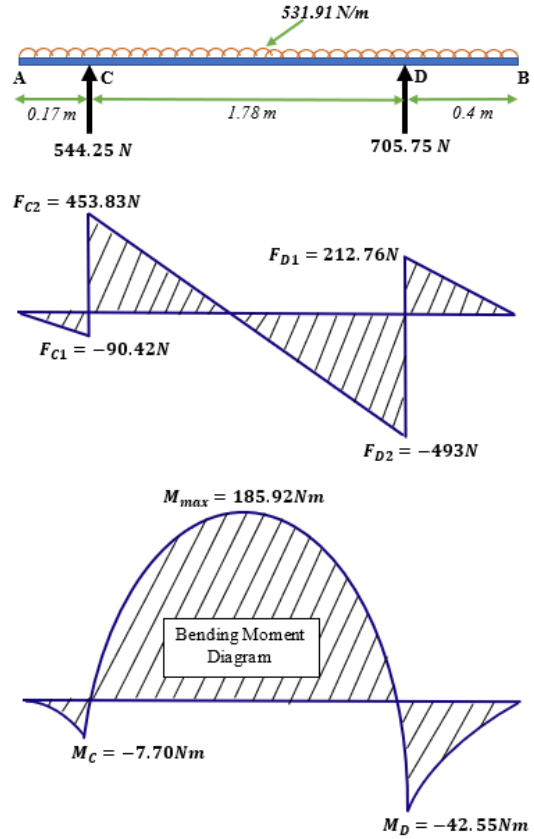


Figure 7. Shear Force and Bending Moment Diagrams of the loaded Chassis

#### 3.3.2. Boundary Conditions and Loading

A fixed geometry was applied during the static analysis, which was represented by 8 joints at supports of the chassis roll bar and at the rear. A total of 34 beams representing the entire chassis structure was subjected to the uniformly distributed load of magnitude 531.91 N/m as shown in Figure 8. The resultant forces and moments obtained are as shown in Table 4 and Table 5 respectively.

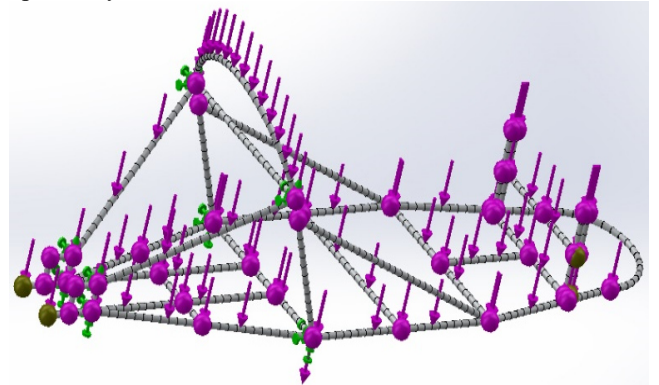


Figure 8. Boundary conditions and loading

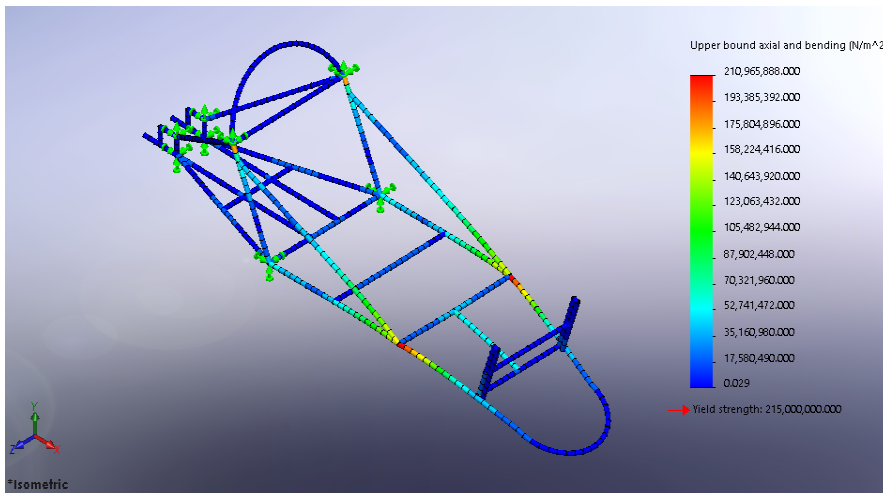


Figure 9. Stress plot

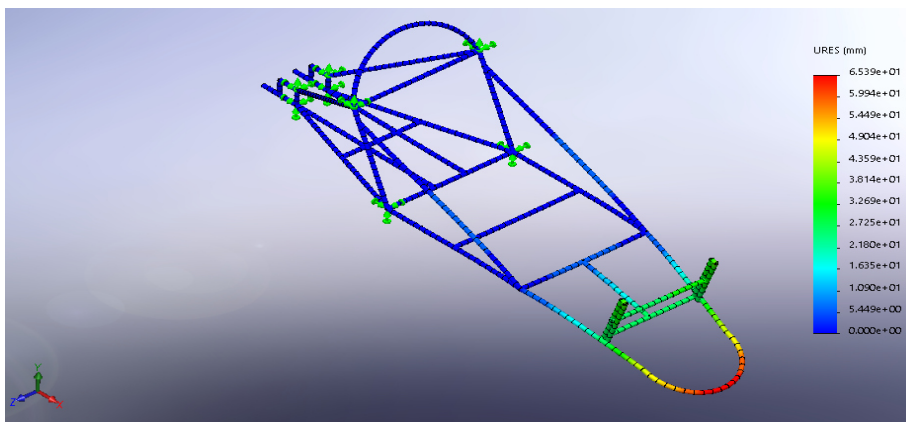


Figure 10. Resultant displacement of chassis

Table 4. Reaction forces

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N	0.001	3429.55	$3.8 \times 10^{-6}$	3429.55

Table 5. Reaction moments

Selection set	Units	Sum X	Sum Y	Sum Z	Resultant
Entire Model	N.m	-0.015	-0.07	-180.63	180.63

### 3.4. Results of the FEA

The linear static analysis was performed by the CAD solver. The equivalent upper bound axial and bending stress is as shown in Figure 9, while Figure 10 shows the resultant displacement of the FEA. The results present a maximum stress value of 210 MPa and displacements up to 65 mm, both being considered acceptable. Specifically, the maximum stress value

of 210 MPa occurs at the base sides of the chassis at midpoint between the driver's seat and steering bracket while the maximum displacement value of 65 mm occurs at the front of the chassis base. These areas were reinforced during the welding stage to cushion the effect. Recall from Table 1 that the yield strength of the chassis material (aluminium alloy 6063-T6) is 215 MPa and, fortunately, the maximum design stress value of 210 MPa falls below the yield strength of the material—which indicates that the design is within safe limits.

### 3.5. Fabrication and Testing

Due to the challenges encountered with getting aluminium square tubes of required specifications in Nigeria, the team devised an alternative and effective plan of fabricating the aluminium square tube used for the chassis. This was achieved by joining two angle cross-sections of the aluminium material via the electric arc welding process as shown in Figure 11. The welding was achieved with the use of aluminium electrodes as shown in Figure 12, and at an average voltage of 90 V. After completion of the chassis fabrication, the welds were further reinforced to ensure firm welded joints. The weight of the

chassis was found to be 15 kg, which confirms the good weight/strength property of aluminium material over popular materials like steel..

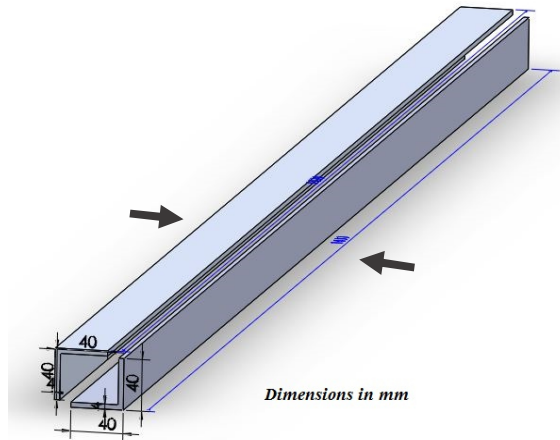


Figure 11. Joining of two aluminium angle sections.



Figure 12. Electric arc welding of the chassis (left); the installed engine components (right).



Figure 13. Test-running of the prototype racing vehicle

Afterwards, holes were drilled at specific locations on the chassis for the insertion of bolts and other vehicle components and all components were installed accordingly as in Figure 12 (right). The chassis was found to withstand the loads of the driver and vehicle components in line with the design data. The completed prototype vehicle was test-run to further reaffirm the strength of the chassis. The vehicle was test-run on a 1.2 km road network in the university (Figure 13), which is an equivalent of the standard SEM race circuit.

#### 4. Final Results and Discussions

The driver completed 8 laps non-stop in successive test runs. The chassis was found to show no cracks as it was able to resist bending and shear from the overall weight of the vehicle and torque from the rotating members. The energy efficiency recorded from the first test run was 250 km/L of gasoline. This could be improved upon by fine-tuning the engine in line with the SEM Global rules.

**Table 6.** Results comparison

Material	Aluminium		
		FEA results	Analytical results
von-Mises stress (MPa)	Max	210	200
	Min	0	0
Torsional shear stress (kPa)	Max	--	559
	Min	0	0
Deflection (mm)	Max	65	42
	Min	0	0

Table 6 shows the final results from the FEA and analytical designs. The results from the analytical design reveal that the von-Mises stress approximately equals that of the FEA design. Also, the position of maximum deflection in the analytical design agrees with that of the FEA. However, both analyses differ in magnitude as the maximum deflection from the FEA was 65 mm while that of the analytical analysis was 42 mm. The results differences are due to the chassis being considered as a simply supported 2-dimensional beam with uniformly distributed load in the analytical design, whereas in the FEA, the load is distributed across the entire surface of the 3-D model. The results further confirm that FEA method is a faster way of achieving the chassis design without spending much time on analytical calculations.

On the other hand, the design achieves the aim of driver safety as the chassis was wide and rigid enough to provide a safe and spacious interior for the driver's comfort. The bulkhead of the engine compartment was electrically isolated (using wire harness and smart circuit design techniques) and thermally isolated (using fire retardant material) from the driver's seating area in line with the SEM safety rules. In addition, the driver's seat was equipped with a portable fire extinguisher as well as with standard racing car seatbelts. It was always ensured that the driver puts on the safety helmet before the engine is started.

## Conclusion

The work has presented an effective method of the design and fabrication of prototype vehicle chassis for Shell Eco-marathon competitions. The design achieved the objectives of rigid and high strength chassis, reduced vehicle weight in line with SEM rules, driver safety, and an energy efficient vehicle. The work demonstrated how aluminium alloy could be used to construct non-integrated chassis for super mileage vehicles by outlining a load/stress calculation model based on an analytical design approach. The fabrication employed the method of joining two angle bar aluminium materials to form the desired square tube via electric arc welding. The chassis had maximum bending stress of  $210 \times 10^6 \text{N/m}^2$  and deflection of 65 mm which are within safe limits of the material. The FEA design approach gives similar results as the analytical approach thereby reducing

the efforts spent on the latter approach. After fabrication, the chassis was found to show no cracks as it was able to resist bending and shear from the overall weight of the vehicle and torque from the rotating members. The energy efficiency recorded from the first test run of the vehicle was 250 km/L of gasoline. Finally, the paper shows that a lightweight, energy efficient non-integrated chassis can be fabricated from welded aluminium angle bars for prototype super mileage vehicles.

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## Declaration of conflicting interests

The authors declare no conflict of interest.

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