Design and Modelling Methodologies of an Efficient and Lightweight Carbon-fiber Reinforced Epoxy Monocoque Chassis, Suitable for an Electric Car

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Abstract

The design of an electric urban car's chassis for the “Shell Eco Marathon” competition takes into account the usage and the type of the vehicle. The most critical factors of designing the new chassis are: the reduction of the weight, improvement of strength and stiffness and reduction of material and manufacturing cost. Towards this direction, a new design approach for a lightweight carbon-fiber reinforced epoxy (CFRE) monocoque chassis, is proposed, which conforms to structural, ergonomic and aesthetic requirements. For the development of this innovative approach, the parametric design method was chosen, in order for the design to be modified easily. The chassis efficiency, in terms of high strength in low mass, was obtained by following appropriate design steps and rules which conform to the vehicle structural and dynamical constraints and by choosing the composite material CFRE. Additionally, a method that calculates the mechanical properties of the composite material CFRE is presented. Furthermore, a model has been created, which calculates automatically the total loads applied on the vehicle's chassis. Worst case stress scenario was chosen and the model's output was evaluated for the new chassis design.

Keywords: Parametric design, Lightweighting, Chassis, Composites, Carbon-fiber, CFRE, Monocoque, CAD, CAE, Vehicle dynamics, FEM, Modelling, Electric car, Racing, COG, Shell eco marathon, Efficiency, Chassis design, Stress scenario.

Introduction

The main goal of this publication is to demonstrate a strategy plan concerning the designing process and guidelines, the materials, the worst case stress scenario and the loads, for the maximization of the car structure efficiency, in terms of high strength and performance in low mass [1]. By applying the new strategy plan, the chassis of an electric car can have less weight and become more durable. Historically, the studied prototype of the electric car has been employing an aluminium space frame and has already won four trophies in six years, in the Shell Eco Marathon, being placed among the best cars in this European competition.

However, using aluminium as structural material, additional aluminium was required to meet stiffness and strength demands. Furthermore, with the use of space frame as chassis type, there was not enough space to install mechanical and electrical parts. Accordingly, it is necessary to create extra housings on the chassis in order to fit in the mechanical and electrical parts. Moreover, space frame is difficult to be manufactured because it is made out of many parts that are assembled together.

Consequently, according to the strategy plan, a CFRE monocoque chassis design, is proposed, that offers great design freedom and is lighter, stiffer, stronger, easier to manufacture and more spacious than the previous one. To achieve a high quality design, the design specifications were compromised with the team’s targets, the ergonomic and safety issues were evaluated, the structural possibilities and limitations regarding the available materials were taken into account, the structural engineering constraints regarding a lightweight, stiff, strong and easy to manufacture design were investigated and the loads that act on the axles were analyzed and calculated. Simulation and manufacturing procedures were outside the scope of this publication. For the three-dimensional design, the ProEngineer Wildfire 5 software was used.

Ergonomics

The driver can be aided in his performance by ensuring that all controls can be easily reached, he/she has a comfortable seating position and that visibility over the front of the chassis is sufficient [2]. The variables for a good seating position are the vertical and horizontal position of the steering wheel, the horizontal position and angle of the seat with respect to the horizontal, the horizontal and vertical position of the pedal assembly, the height and horizontal position of the dashboard and front roll hoop. Besides being comfortable, the driver must be safe at all times [3]. This mainly involves that many rules are followed in order to design a safe car. Some major regulations of Shell Eco Marathon are the existence of a roll bar that withstands 700 N (applied in all directions) and extends 5 cm around driver’s helmet, a bulkhead that secures the driver, a wide and long enough chassis design to protect the driver's body and dimensional demands for the chassis to allow for quick driver egress in case of accidents or fire [4]. These regulations are often with respect to a so called 95th percentile male [5] as shown in Figure 1.
where df is the density of the fiber, dm is the density of the resin, where Ef is the elastic modulus of the fiber, and Em is the elastic modulus of the epoxy.

Assuming an anisotropic thin composite lamina with the fibers aligned in the x₁ direction, transverse to the x₂ direction and vertically to the x₃ direction, Young’s modulus E, shear modulus G and Poisson ratios ν, in all three axes, are required for its characterization [10].

\[
E_x = E_f V_f + E_m (1-V_f) \quad \text{(4)}
\]
\[
E_y = E_f V_f / (E_m V_f + E_f (1-V_f)) \quad \text{(5)}
\]
\[
E_z = E_y \quad \text{(6)}
\]
\[
u_{xy} = v_{xy} V_f + v_m (1-V_f) \quad \text{(7)}
\]
\[
u_{yx} = (v_{xy} E_x) / E_c \quad \text{(8)}
\]
\[
u_{zx} = v_{xy} \quad \text{(9)}
\]
\[
G_{xy} = G_m G_f / (G_m V_f + G_f (1-V_f)) \quad \text{(10)}
\]
\[
G_{yz} = E_y / (2(1-v_{yx})) \quad \text{(11)}
\]
\[
G_{xz} = G_{xy} \quad \text{(12)}
\]

Respectively, the longitudinal tensile strength, the transverse tensile strength and the compression strength on the composite are listed.

\[
\sigma_x = \sigma_f V_f + \sigma_m (1-V_f) \quad \text{(13)}
\]
\[
\sigma_y = \sigma_m (1-\sqrt{4V_f / \pi}) \quad \text{(14)}
\]
\[
\sigma_{comp} = G_m (1-V_f) \quad \text{(15)}
\]

where σₙ is the fibers stress levels, and σₗ is the resin stress levels.

For the multi-ply laminates, the tensile modulus, the shear modulus and the Poisson ratio of a random continuous-fiber composite can be calculated by:

\[
E = (3/8) E_1 + (5/8) E_2 \quad \text{(16)}
\]
\[
G = (1/8) E_1 + (1/4) E_2 \quad \text{(17)}
\]
\[
\nu = (E - 2G) / 2G \quad \text{(18)}
\]

where \(E_1\) is the longitudinal modulus, and \(E_2\) is the transverse modulus for a unidirectional lamina.

The Krenchel model is utilized for the approximation of the strengths of multi-ply laminates. The efficiency factor, \(n_θ\), is used in a mixture-rule calculation [11]:

\[
n_θ = \Sigma a_n \cos^4 \theta \quad \text{(19)}
\]
\[
\sigma_i = n_θ \sigma_f V_f + \sigma_m (1-V_f) \quad \text{(20)}
\]

Table 1 and Table 2 provides the properties of unidirectional CFRE and multi-ply laminates.

<table>
<thead>
<tr>
<th>Property</th>
<th>VALUE</th>
<th>UNIT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic modulus Ex</td>
<td>380.100</td>
<td>Gpa</td>
</tr>
<tr>
<td>Elastic modulus Ey</td>
<td>28.269</td>
<td>Gpa</td>
</tr>
<tr>
<td>Elastic modulus Ez</td>
<td>28.269</td>
<td>Gpa</td>
</tr>
<tr>
<td>Poisson ratio vxy</td>
<td>0.336</td>
<td></td>
</tr>
<tr>
<td>Poisson ratio vyz</td>
<td>0.025</td>
<td></td>
</tr>
<tr>
<td>Poisson ratio vxz</td>
<td>0.336</td>
<td></td>
</tr>
<tr>
<td>Shear modulus Gxy</td>
<td>4.213</td>
<td>Gpa</td>
</tr>
<tr>
<td>Shear modulus Gyz</td>
<td>13.790</td>
<td>Gpa</td>
</tr>
<tr>
<td>Shear modulus Gxz</td>
<td>4.213</td>
<td>Gpa</td>
</tr>
<tr>
<td>Tensile strength ox</td>
<td>2539.400</td>
<td>Mpa</td>
</tr>
<tr>
<td>Tensile strength oy</td>
<td>8.251</td>
<td>Mpa</td>
</tr>
<tr>
<td>Compressive strength ocompx</td>
<td>4.722</td>
<td>Gpa</td>
</tr>
</tbody>
</table>
Braking, are as good as Fiat’s braking control system. Thus, the
s2. In our case it is also supposed, that the driver’s reflexes during
and 30 km/h, the brakes apply a maximum deceleration of 6 m/
how it behaves in braking tests [20]. At speeds between 20 km/h, it encounters a stationary preceding vehicle and decelerates immediately (6 m/s²), to avoid the accident.

Emergency Braking” (AEB) test of Euro NCAP is chosen.

Determine the worst case stress scenario
The loads of a chassis structure are divided into crash, ride,
towing, aerodynamic, cornering, braking and tractive loads. Crash
cases are often the most difficult and critical to design.

Cornering loads are maximized when the vehicle’s speed is
at maximum speed and its turning radius is minimized. At
the Shell Eco Marathon’s track, in Rotterdam, there are four
counterclockwise and one clockwise turns with approximately
the same angle (90°). Regarding to the driving strategy the vehicle
runs at high speed (30km/h) on first turn [18]. Therefore, the first
turn has been investigated while the chosen racing line depends
on the characteristics of the car, the cornering strategies and the
conditions around. In the apex point of the corner, the maximum
speed and stress is reached. So, the apex of the first turn is the
point where there is the cornering worst case stress scenario.

Braking loads cause larger loads than tractive loads [19]. Thus, a real situation needs to be considered when the chassis is
overloaded during braking. Supposing that while the vehicle
moving on the track,with its maximum speed, 30 km per hour,
the preceding vehicle suddenly brakes. Therefore, the driver
is forced to brake immediately, to avoid the collision. At this point,
it is needed to find a realistic “deceleration scenario” for urban
cars, to determine the deceleration value. The “Autonomous
Emergency Braking” (AEB) test of Euro NCAP is chosen. Randomly, the Fiat’s braking control system is selected to see how it behaves in braking tests [20]. At speeds between 20 km/h and 30 km/h, the brakes apply a maximum deceleration of 6 m/
s². In our case it is also supposed, that the driver’s reflexes during
braking, are as good as Fiat’s braking control system. Thus, the
vehicle will be subjected to brake with a deceleration equal to 6 m/s² from 30 km/h to 0 km/h [21-23].

In order to demonstrate the strength of the chassis, it only has
to be shown that it withstands the total load worst case stress
scenario that is the combination of cornering and braking worst
case stress scenario. Thus, it is needed to study the scenario
where the vehicle is turning in the 1st corner and while is
positioned in the apex with 30km/h, it encounters a stationary preceding vehicle and decelerates immediately (6 m/s²), to avoid the accident.

Dynamic axle loads
Presuming that the vehicle sits statically on level ground, the
vertical loads can be calculated [17].

\[ W_x = M_g (c / L) \]  \hspace{1cm} (27)

\[ W_y = M_g (b / L) \]  \hspace{1cm} (28)

where \( M \) is the vehicle mass, \( g \) is the gravity acceleration, \( b \) is
the distance from the front axle to the CG, and \( c \) is the distance from
the rear axle to the CG.

According to the lateral dynamics, the two front wheels can be
represented by one wheel at a steer angle \( \delta \), with a cornering
force equivalent to both wheels. The same assumption is made
for the rear wheels[17].

\[ F_y = F_{yt} + F_{yr} = (MV^2) / R \]  \hspace{1cm} (29)

where \( V \) is the forward velocity.

During cornering, a dynamic load transfer from the inside to the
outside wheels occurs (the second mechanism for this study is
zero, because the chassis has not springs) [24].

\[ F_{in} - F_{oi} = (2F_y h) / t + (2K_p \varphi) / t \]  \hspace{1cm} (30)

where, \( h \) is the roll center height, \( K_p \) is the roll stiffness of the
suspension, and \( \varphi \) is the roll angle of the body.

The torque generated by the rotor, for each wheel brace, as well
as the total braking force is defined [25-27].

\[ F_{bp} = F_{t} (I_2 / L_1) \]  \Rightarrow

\[ P_{mc} = F_{bp} / A_{mc} \Rightarrow \]

\[ P_{cal} = P_{mc} \Rightarrow \]

\[ F_{cal} = P_{cal} A_{cal} \Rightarrow \]

\[ F_{clamp} = 2F_{cal} \Rightarrow \]

\[ F_{friction} = F_{clamp} \mu_{bp} \Rightarrow \]

\[ T_{y} = F_{friction} R_{bp} \Rightarrow \]

\[ \tau = T_{w} / \tau \Rightarrow \]

\[ F_{total} = \sum F_{(tou/LEO/LR/RR)} \]  \hspace{1cm} (31)

where \( F_{bp} \) is the force output of the brake pedal, \( F_{t} \) is the force
applied to the pedal pad by the driver, \( L_1 \) is the distance from
the brake pedal arm pivot to the output rod clevis attachment, \( L_2 \) is
the distance from the brake pedal arm pivot to the brake pedal
pad, \( P_{mc} \) is the hydraulic pressure by the master cylinder, \( A_{mc} \) is
the effective area of the master cylinder hydraulic piston, \( P_{cal} \) is
the hydraulic pressure to the calliper, \( F_{cal} \) is the linear mechanical
force by the calliper, \( A_{cal} \) is the effective area of the calliper hy-
draulic piston, \( F_{clamp} \) is the clamp force by the calliper, \( F_{friction} \) is the

Table 2: Properties of multi-ply laminates

<table>
<thead>
<tr>
<th>PROPERTIES</th>
<th>VALUE</th>
<th>UNIT</th>
</tr>
</thead>
<tbody>
<tr>
<td>Elastic modulus ( E )</td>
<td>160.206</td>
<td>Gpa</td>
</tr>
<tr>
<td>Shear modulus ( G )</td>
<td>54.580</td>
<td>Gpa</td>
</tr>
<tr>
<td>Tensile strength ( \sigma )</td>
<td>971.400</td>
<td>Mpa</td>
</tr>
<tr>
<td>Poisson ratio ( v )</td>
<td>0.468</td>
<td></td>
</tr>
</tbody>
</table>

Table 3: New vehicle’s center of gravity

<table>
<thead>
<tr>
<th>COGx (mm)</th>
<th>COGy (mm)</th>
<th>COGz (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>867.900</td>
<td>391.178</td>
<td>419.625</td>
</tr>
</tbody>
</table>
frictional force by the brake pads, $\mu_{bp}$ is the coefficient of friction between the brake pad and the rotor, $R_{\text{eff}}$ is the effective radius of the rotor, $T_r$ is the torque in the tire, $T_w$ is the torque in the wheel, $F_{\text{tire}}$ is the force in the tire, and $R_t$ is the effective rolling radius of the loaded tire.

During braking, a dynamic load transfer from the rear to the front axle occurs [25,28].

$$WT = \left(\frac{a_v}{g}\right) \times \left(\frac{h_{cg}}{L}\right) \times (M \times g)$$  \hspace{1cm} (32)

where $a_v$ is the deceleration, and $h_{cg}$ is the vertical distance from the CG to ground.

**Chassis load calculator (CLC) model**

The values of the forces acting on a vehicle structure change depending on the characteristics of the structure.

Therefore, for academic and research purposes, a model created which automatically calculates the magnitude and the direction of the loads acting on each vehicle, by importing the characteristics of the vehicle. In order to validate the derived model, data such as track width, wheelbase, center of gravity, mass, et cetera were used as inputs. The breakthrough in this research work is the overcoming of the time consuming process to calculate the corresponding forces of different design structures of the car with the use of CLC model. The equations (27) to (32) that were utilized to implement this model are derived from the theory of Vehicle Dynamics. According to the chosen stress scenario and the imported characteristics of the new chassis, the applied loads are calculated.

First of all, the vertical loads are identified as shown in Table 4 and Table 5.

**Table 4:** The data used for the calculation of vertical dynamics

| Distance from the front axle to the CG | 602.900 mm |
| Distance from the rear axle to the CG | 692.100 mm |
| Distance from the left side of the chassis to the CG | 391.178 mm |
| Distance from the right side of the chassis to the CG | 338.822 mm |

**Table 5:** Vertical dynamics

- Static load on the left front wheel: 372.264 N
- Static load on the right front wheel: 429.788 N
- Static load on the left rear wheel: 324.286 N
- Static load on the right rear wheel: 374.396 N

Then, the lateral loads are calculated according to the cornering worst case stress scenario as presented in Table 6 and Table 7.

**Table 6:** The data used for the calculation of lateral dynamics

| Initial forward velocity | 8333 mm/s |
| Turn radius | 10000 mm |
| Gravity acceleration | 9810 mm/s$^2$ |
| Roll center height | 280 mm |
| Track width | 910 mm |
| Distance of chassis on y axis | 730 mm |
| Track width (-) Distance of chassis on y axis | 180 mm |
| Track width (-) Distance of chassis on y axis (from one side) | 90 mm |

**Table 7:** Lateral dynamics

- Cornering force: 1062.276 N
- Load transfer on the right: 653.708 N
- Load transfer on the left: -653.708 N

During cornering the mass distribution changes, as well as the center of gravity [15,16,29]. Assuming that there is no mass transfer in the z axis, since the car has not shock absorbers as well as the fact that if there is a mass transfer in the x axis, it will be negligible, then the new COG shown in Table 8.

**Table 8:** New center of gravity during cornering

| COGx (mm) | 867.900 |
| COGy (mm) | 704.022 |
| COGz (mm) | 419.625 |

**Table 9:** Mass distribution on left and right wheels

| Distribution of mass on the right side | 96.441 % |
| Distribution of mass on the left side | 3.559 % |

The braking loads are calculated according to the braking worst case stress scenario as presented in Table 10, Table 11, Table 12, Table 13, Table 14, Table 15 and Table 16.

**Table 10:** Data used for the calculation of braking loads

| Final forward velocity | 0 mm/s |
| Absolute value of velocity change | 8333 mm/s |
| Braking time | 10 s |
| Braking distance | 83330 mm |
| Maximum deceleration | 833.300 mm/s$^2$ |
| Wheelbase | 1295 mm |
| Front area of front axle | 265 mm |
| Tyre coefficient of friction | 0.0025 |

**Table 11:** Data for the brake system dimensions

| Distance from the brake pedal arm pivot to the output rod clevis attachment | L1 |
| Distance from the brake pedal arm pivot to the brake pedal pad | L2 |
| Wheel radius | 280 mm | 280 mm |
| Master cylinder diameter | 12.7 mm | 12.7 mm |
| Distance-pushrod to balance bar pivot | 30 mm | 40 mm |
| The effective area of the calliper hydraulic piston found on one half of the calliper body | 800 mm$^2$ | 800 mm$^2$ |
| Pad coefficient of friction | 0.35 | 0.35 |
| Disc diameter | 160 mm | 160 mm |
| Pad depth | 3 mm | 3 mm |
| Gap between top of pad and disc | 1 mm | 1 mm |
Resulting all the above, a 6.19 m/s² deceleration was achieved, whose value is greater than the maximum value of the Fiat’s deceleration (6 m/s²), which was first set as a goal shown in Table 15.

In the case of both braking and turning loads, the mass distribution changes, as well as the center of gravity of the vehicle [15,16,29]. Assuming that there is no mass transfer in the z axis, since the car has no shock absorbers, as well as the fact that if there is a mass transfer in the y axis, it will be negligible, then the new COG shown in Table 17.

Table 17: New center of gravity during braking and cornering

<table>
<thead>
<tr>
<th>COGx</th>
<th>COGy</th>
<th>COGz</th>
</tr>
</thead>
<tbody>
<tr>
<td>599.260</td>
<td>704.022</td>
<td>419.630</td>
</tr>
</tbody>
</table>

This is the center of mass that the vehicle has during braking and cornering coexistence. It is observed that after such a sudden stop in a ¼ turn the mass distribution changes presented in Table 18.

Table 18: Mass distribution on front and rear axles

| Distribution of mass on front axle | 74.188 % |
| Distribution of mass on rear axle | 25.812 % |

With the new center of mass, the cornering force on each wheel can be found in Table 19.

Table 19: The cornering force on each wheel with the new center of gravity

| Cornering force (front) | 788.086 N |
| Cornering force (rear)  | 274.190 N |
| Cornering force on the left front wheel | 28.045 N |
| Cornering force on the right front wheel | 760.041 N |
| Cornering force on the left rear wheel | 9.757 N |
| Cornering force on the right rear wheel | 264.433 N |

Summarizing, the loads that act on each semi-axle of the chassis, in the z axis, are presented in Table 20.

Table 20: Deceleration and stopping distance

| Total force (4 wheels) | 960.757 N |
| Deceleration a | 6194.436 mm/s² |
| Stopping distance | 5604.940 mm |
The vertical distance of $F_z$ from the center of the axle is calculated as well as the vertical distance of $F_z$ from the center of the axle presented in Table 24.

**Table 24: Vertical distances**

<table>
<thead>
<tr>
<th>x = (cos45)*0.0075 + 0.0075</th>
<th>y = (sin45)*0.0075</th>
</tr>
</thead>
<tbody>
<tr>
<td>12.803 mm</td>
<td>5.303 mm</td>
</tr>
</tbody>
</table>

$F_x$, $F_y$ are transferred to the axle. The equivalent system will consist of the braking forces ($F_x$, $F_y$) plus the moments ($M_{x1}$, $M_{x2}$) that are created from the braking forces as shown in Table 25.

**Table 25: Braking forces $F_x$, $F_y$ and moments $M_{x1}$, $M_{x2}$ from disc effective radius to the axle**

<table>
<thead>
<tr>
<th>Braking forces $F_x$, $F_y$</th>
<th>$M_{x1}$</th>
<th>$M_{x2}$</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Left front axle</strong></td>
<td>701.273 N</td>
<td>8978.604 Nmm</td>
</tr>
<tr>
<td><strong>Right front axle</strong></td>
<td>701.273 N</td>
<td>8978.604 Nmm</td>
</tr>
<tr>
<td><strong>Left rear axle</strong></td>
<td>525.954 N</td>
<td>6733.953 Nmm</td>
</tr>
<tr>
<td><strong>Right rear axle</strong></td>
<td>525.954 N</td>
<td>6733.953 Nmm</td>
</tr>
</tbody>
</table>

**Conclusion**

The new chassis is extremely light, only 5.38 kg and consequently less energy is consumed to move it, comparing to the previous one that weights 10.85 kg. This energy decrease is significantly high taking into account that the previous one was the lightest chassis of the competition. Furthermore, the ergonomics and the aesthetic acceptance of the new chassis is better than the previous one.

Consequently, the breakthrough in this research work was not only the achievement of the lightest chassis in the Shell Eco Marathon competition that combines ergonomics, aesthetic and strength demands but also the overcoming of the time consuming process to calculate the corresponding forces of different design structures of the car with the creation and use of the CLC model.

In the future, a FEM model will be developed and used in order to demonstrate the resistance of the new chassis design under the aforementioned extreme stress scenario.

**Acknowledgements**

Open access fees were covered by the Municipality of Agrinio, Western Greece. Authors are grateful to the Mayor of Agrinio, Mr. George Papanastasiou.

References


