Design and analysis of a Cross Car frame. Proposal of a three-level appraisal methodology

Journal:	*Part P: Journal of Sports Engineering and Technology*
Manuscript ID	JSET-21-0007.R3
Manuscript Type:	Original Article
Date Submitted by the Author:	n/a

Complete List of Authors:	Luque, Pablo; Universidad de Oviedo, Transportation Gutierrez, Alex; Universidad de Oviedo, Transportation Rodriguez, Eduardo ; Universidad de Oviedo A. Mantaras, Daniel; Univesity of Oviedo

Keywords:	Cross Car, 3D design, Multibody dynamics, Finite element analysis, Frame

Abstract:	This paper analyses the structural design process of a Cross Car frame. Existing international and national technical regulations were analysed. The analysis of the regulatory guidelines led to the conclusion that the design only includes geometric or material restrictions at the international level. There are other more demanding national regulations (Spanish, for instance) that include structural verifications through static calculations for unidirectional actions (vertical, longitudinal or lateral). The main contribution of this paper is a novel three-level appraisal and a proposed redesign methodology. At the first level, a geometric and material verification is carried out. The second level involves the verification under unidirectional static actions. The third level entails a dynamic verification of three-dimensional combined actions. The load case is obtained from the computer multibody dynamics simulation of the full vehicle assembly in the worst case of driving conditions on circuits. This methodology is a far more detailed tool than traditional design processes. The use of this methodology allows for design optimization, including all the effects of powertrain, brakes, suspension, steering and driver.

Design and analysis of a Cross Car frame. Proposal of a three-level appraisal methodology

Luque, Pablo (*); Gutierrez, Alex (*); Rodríguez, Eduardo (**); Mántaras, Daniel A. (*)

(*) Transportation Area – University of Oviedo – Campus de Gijón, 33203 Gijón
(**) Mechanical Engineering Area - University of Oviedo – Campus de Gijón, 33203 Gijón

Corresponding author:
Pablo Luque
Transportation Area
C/ Pedro Puig Adam 5
Gijón– 33203
Spain
Phone: +34 985 182059
E-mail: luque@uniovi.es

Abstract- This paper analyses the structural design process of a Cross Car frame. Existing international and national technical regulations were analysed. The analysis of the regulatory guidelines led to the conclusion that the design only includes geometric or material restrictions at the international level. There are other more demanding national regulations (Spanish, for instance) that include structural verifications through static calculations for unidirectional actions (vertical, longitudinal or lateral). The main contribution of this paper is a novel three-level appraisal and a proposed redesign methodology. At the first level, a geometric and material verification is carried out. The second level involves the verification under unidirectional static actions. The third level entails a dynamic verification of three-dimensional combined actions. The load case is obtained from the computer multibody dynamics simulation of the full vehicle assembly in the worst case of driving conditions on circuits. This methodology is a far more detailed tool than traditional design processes. The use of this methodology allows for design optimization, including all the effects of powertrain, brakes, suspension, steering and driver.

Keywords: Cross Car, 3D design, multibody dynamics, finite element analysis

Introduction

Cross Cars are a type of competition vehicle that participate in an automobilism modality called Autocross. These vehicles are formed by a tubular frame and rear wheel drive. The powertrain is derived from a motorbike engine. Their light weight and 600cc engine make them very fast and a low-cost sport vehicle. The design of Cross Cars must follow the International Automobile Federation (FIA) guidelines [1], and other additional rules imposed by national or regional regulations.

This paper analyses the state of the art of the Cross Car design process. Various technical regulations were analysed. In the design process, the restrictions of the Technical Regulations must be considered. The Federation Internationale de l'Automobile establishes the conditions for Cross Car (XC). This regulations define a Cross Car as follows: “Rear engine 4-wheeled single-seater land vehicles with a multi-tubular space frame chassis which must have a safety cage as an integral part of the chassis, as defined in Article 10. The propelling device and steering are controlled by a driver on board the vehicle. The vehicles must be 2-wheel rear drive.” The regulation includes conditioning factors and design restrictions in relation to geometric, topological or materials parameters to be used.

There are additional geometric regulations (topology, positioning, bending, angles, etc.). Once the base construction is defined, it must be completed with compulsory members and reinforcements, to which other optional members and reinforcements may be added. Only tubes with a circular section are authorized. The regulation limits the minimum material specifications, minimum tensile strength, and minimum section dimensions. No tests (real or virtual) are specified in the regulations.

This topological, geometric and material verifications have been adopted in regulations by different countries with national competitions. On the international scene, there are other more restrictive regulations for this type of vehicle. For example, the Royal Spanish Automobile Federation (RFEAS) [2] proposes the calculation of the structure using the finite element technique. The testing of virtual prototypes under various static load cases is proposed. The response of the structure to vertical, longitudinal (front and rear) and lateral actions is studied. In each case, the value of the load is established, and it must be checked that the strain or stress (Von Misses, for example) does not exceed an established value. These values are related to the rider's safety in case of an impact.

Lately, with the aim of making vehicles lighter, new techniques are being implemented in the computer-aided design phase. Nowadays, land vehicle design uses computer-aided engineering (CAE) tools [1].

CAE computer software includes computer-aided design (CAD), finite element analysis (FEA), computational fluid dynamics (CFD), multibody dynamics (MBD), durability and optimization. CAE tools are used to analyse the robustness and performance of components and assemblies. These tools include simulation, validation, and optimisation tasks.

Vehicle dynamic computer simulation tools are used frequently to analyse the response to several inputs [3-10]. Finite element method (FEM) has been employed in different structural vehicle designs [11-14]. The traditional methodology is implemented to obtain model stresses and strains when the model is subjected to certain loads, which are usually considered as static or quasi-static conditions. The load definition depends on the operative vehicle conditions. Those studies are often combined with the chassis system design (dumping, steering, braking, powertrain, etc.) [15-18] to improve vehicle performance. Instrumentation and test techniques in real circulation vehicle conditions [19-21] are employed in order to improve the fidelity of those studies, as well as to understand circulation and safety conditions [22-24]. A cooperation study between simulation and data model has been analysed by Kim and Kim [25]. Those techniques have been employed to optimize competition vehicles, from sophisticated to simple categories [26]. In the dynamic simulation of vehicles by computer, it is necessary to evaluate fundamental issues, such as the interaction with the rolling surface. Grip estimation obtained with diverse methods [27-28] is used to characterise this performance. Simulation techniques are now used to perform virtual testing [29] in order to minimise the cost and duration of design evaluation.

Currently, dynamic vehicle simulation tools are combined with FEM [30-32] to research the dynamic response of the vehicles, including structural strains effects.

Within these techniques, the use of multibody dynamic computer simulation allows for the analysis of various dynamic scenarios, representing multiple manoeuvres and driving situations, to obtain sets of forces and torques at the hardpoints of the frame. That is why the structural response can be calculated in the face of innumerable dynamic load cases.

In this paper, a methodology for the appraisal and the redesign of a Cross Car is proposed. This methodology consists of three levels. At the first level, an initial design is evaluated to ensure that the geometric, topological, and material conditions established in international regulations are fulfilled. Once the first level has been verified, the structural behaviour in the static condition is assessed in the second level. For this purpose, the load cases established in a technical regulation are adopted. These loads are unidirectional and are applied in very specific areas with the main goal of preserving a pilot´s survival in case of impact. At the third level, after having verified that the chassis meets the static requirements, the design is completed with the rest of the vehicle's elements and systems (powertrain, suspension, steering, brakes, wheels and tires, cooling, seat and pilot, etc.). With the full equipment of the vehicle (vehicle´s mass in running order, which includes the weight of the driver), a circuit is simulated that includes manoeuvres at different speeds, with acceleration and braking, passing through different curves (right and left), as well as irregularities on the road and bumps. The worst case is established in relation to actions on hardpoints. These conditions are the basis for the definition of the dynamic load case applied to the frame. This three-dimensional load case is applied at all interface points between the frame and vehicle systems to ensure that the loads do not exceed the values established in the vehicle's specification booklet.

2. Methodology

According to the Federation Internationale de l'Automobile, a Cross Car (XC) is a rear engine 4-wheeled single-seater land vehicles with a multi-tubular space frame chassis which must have a safety cage as an integral part of the chassis.

The tubular frame is the structure that provides safety to the driver in case of an accident. The frame also operates like a support for different parts of the vehicle like the seat, powertrain, suspension, steering and braking elements. The frame, which is formed by welded tubes, can have various cross sections.

The design should consider important factors, such as ergonomics, safety, manufacturing, durability, weight, as well as other component position and cost.

The design of a vehicle must consider the final use and established regulations. A Cross Car vehicle is internationally regulated by the FIA [1], which details the set of geometric conditions that must be met.

These cars must comply with different dimensional constraints including maximum car dimensions of overall length (2600 mm), overall width (1600 mm, excluding mudguards) and height (1400 mm excluding engine water radiator air intake). The wheelbase and tracks are free, within the limit of the above. The minimum weight of the vehicle, including the driver wearing his full racing apparel and fluids being full, must be 425 kg. All measurements must be done while the car is stationary on a flat horizontal surface.

From a topological point of view, the multi-tubular space frame structure, which is formed by the compulsory base construction of the safety cage, reinforcement tubes and any other tubular structure or elements welded together for the functioning of the car, is considered “the chassis”.

The regulations provide the following definitions:

The safety cage is defined as a multi-tubular structure installed and welded to the chassis been an integral part of it. Its function is to reduce the deformation of the cockpit in case of an impact.

The safety cage must be welded onto the structure to which the suspension loads are transmitted. If necessary, additional reinforcement at the joint between the chassis and the foot of the rollbar should be made. The mounting points of the front, lateral half and main rollbars must be fitted at least at the level of the cockpit floor. The base construction must be composed of:

- One main rollbar
- One front rollbar + two longitudinal members joining the upper part of the main and front rollbars or two lateral half-rollbars + one transverse member joining the upper part of the lateral half-rollbars
- Two backstays with two near-vertical extensions (maximum angle ± 10° to the vertical) of the same section and quality, going down to the floor level and to the rear end of the car
- Two longitudinal side members joining the vertical extensions to the backstays, main rollbar and front rollbar, ending in front of the pedal box frame
- Four transverse members connecting the vertical extensions to the backstays, main rollbar, front rollbar and front ends of the two lower longitudinal side members
- Two transverse members connecting each side of the main rollbar, one at the height of the door bars and a second one for the safety harnesses
- Pedal box frame
- Door bars
- Diagonal member
- Windscreen pillar reinforcement
- Transverse member on the front rollbar

According to the aforementioned regulation, there are restrictions on the materials that can be used. The use of the following materials is forbidden, unless they correspond exactly to the material of the original part or of a powertrain homologated part:

- Titanium alloy
- Magnesium alloy (< 3 mm thick)
- Ceramics
- Composite or fibre-reinforced material

Titanium alloy is permitted for quick release connectors of the braking circuit. The use of composite material is authorized only for some elements, but not for the main frame.

Once these aspects related to geometry, topology and materials have been verified, other national regulations that include the need for virtual tests are followed, such as the RFEAS [2]. A set of virtual tests is proposed and carried out through structural verification. Various load cases are established and the maximum deformation and maximum stress on the frame are limited. Load cases include the following:

- Vertical load: 7.5 * P [daN]
- Longitudinal load (front and rear): 6 * P [daN]
- Lateral load: 3.5 * P [daN]

where P is equal to the total weight + 80 kg.

If the design meets the strain and stress constraints (static calculation), this verification is accepted. In the use of these vehicles, the structure is subjected to a multitude of actions that can endanger the structural integrity of very specific elements or parts. These points are characterized by concentrating stresses that can exceed acceptable values. In order to identify these effects, an extended methodology is proposed, in which the structural response to dynamic movement actions is analysed.

This methodology proposes a third level of verification once the regulated levels have been reached. In other words, the dynamic verification would only be launched once the geometric and static verification have been passed.

To perform dynamic verification, a set of "worst case" actions must be identified. A test with a virtual prototype should be carried out on a circuit that combines the most severe driving manoeuvres that can occur in this type of automobile competition. This virtual test is carried out with a multibody model of the vehicle, which includes both the designed frame and the rest of the elements (powertrain, suspension, steering, tires, brakes, cooling, seat and pilot). The simulated circuit must include both longitudinal (traction and braking), lateral (varied left and right curves) and vertical (bumps) actions.

The flowchart of this methodology is depicted in Fig. 1.

3. Results

The design process involves various factors, such as material selection, cross section selection, 3D frame design, multi dynamics studio and finite element analysis. The details of each step are summarized below.

According to RFEAS rulebook [2], the material chosen must be below the maximum 0.3% of carbon content and satisfy a minimum tensile strength of 355 MPa, while also having high yield strength to resist all the loads and possible impacts without being deformed. Steel SAE-AISI 1524 is a low-alloy steel containing silicon and manganese as strengthening agents. This steel has good strength, toughness, weldability and machinability. Table 1 outlines the SAE-AISI 1524 steel properties.

Table 1. Properties of SAE-AISI 1524 steel

	Property	Value
1	Density	7800 kg/m^3
2	Ultimate tensile strength	650 MPa
3	Yield strength	540 MPa
4	Young's modulus	190 GPa
5	Poisson's ratio	0.29

In terms of cross-section selection, according to the guidelines [1], the primary elements should be circular tubes with a minimum outside diameter of 40 mm (1.575 inches) and a minimum wall thickness of 2 mm (0.079 inches). The guidelines also establish a constraint for non-primary tube selection. Non-primary tubes must be circular with a minimum outside diameter of 35 mm (1.378 in) and a minimum wall thickness of 1.5 mm (0.059 in). For the initial design (Fig. 2), the minimum size of the tubes is used because the weight of the frame must be as low as possible.

Once the frame is in place, the rest of the system and components are conceptualized and designed (Fig. 3), including the powertrain, brakes, front and rear suspension, steering, wheels, tires, seat, etc. The hardpoints and joints are also designed.

3.1. Level 1 validation - geometry, topology and materials

A first check of the weight and geometry was carried out according to international regulations. The total weight of the first design was 318.13 kg. This weight only took into consideration the weight of the represented elements on Fig. 4. An extra estimated weight of 120 kg was added to account for the weight of the pilot, motor liquids, car body and other vehicle elements. The final weight of 438.13 kg complied with the minimum required by the regulations.

The maximum width was 1598.69 mm, the length was 2392.96 mm, the wheelbase was 1702.37 mm and the height was 1324.92 mm (Fig. 4). These measurements met the design requirements.

3.2. Level 2 validation - static evaluation

For the static validation of the regulatory load cases, a mathematical model was developed based on the Finite Element Method (FEM) and implemented in ANSYS (R) software. The structure was discretised with beam-type elements. From the drawing in CAD software, the model was meshed. A sensitivity analysis was performed and the optimal mesh size was calculated. The meshed model was loaded under ruled cases. The vertical load case in initial design produced a stress state higher than the allowable values (Fig. 5).

After a redesign process, some reinforcements were included to make the tension admissible. In particular, the effect of the two added vertical tubes was significant. This topological configuration was adequate to support the vertical loads. The deformations produced were much smaller and did not exceed the maximum in the technical standards (Fig. 6).

With the redesigned model, the response to longitudinal loads was analysed. The load had to be applied to both the front part (Fig. 7) and the rear part (Fig. 8) of the frame.

The results show that the maximum stresses, both at the front part and rear part, were lower than the elastic limit of the material, so it can be stated that the chassis fulfils the longitudinal load test. Regarding the results of the deformations, they are less than a millimetre, which is a very favourable value.

The lateral load test involved the application of the load on the side protection bars, towards the passenger compartment. The test was carried out on one of the sides (Fig. 9) since the design is symmetrical.

By performing the test at a maximum stress lower than the elastic limit, the tensional results showed the strength of the frame, providing another point necessary for approval.

The lateral deformations were greater than one millimetre, but they still were not large enough to pose any risk to the pilot.

After this set of checks, the design met the specifications for static loads established in the regulations. According to the proposed methodology, the dynamic validation process was then launched.

3.3. Level 3 validation - dynamic evaluation

To perform dynamic validation, a set of loads that characterize the worst case must be defined. A full model was implemented in MSC-ADAMS. The multibody model implemented was complete and included all systems and the inertial effect of the pilot's mass (Fig. 10). The model includes the suspension, steering and wheel systems in detail. All the information on springs, shock absorbers, joints, tires, etc. is included. This allowed the dynamic response of the vehicle to be characterised from the solution of a system of algebraic-differential equations. The inputs included the driver's actions on the powertrain, brakes and steering. The forces at each singular point, for each simulation instant, were calculated as a function of the driving manoeuvres. This made it possible to identify the load states in the simulated circuits.

This full vehicle assembly was subjected to various driving manoeuvres through various circuits (Fig. 11). These circuits represent the racing conditions of these vehicles. They included both acceleration and braking manoeuvres, circulation at different speeds on straights and curves, as well as irregularities in the road, such as bumps.

The load cases were analysed in terms of the combined forces generated in the tires. These actions are influenced by both the driving actions and the configuration of the entire dynamic system that makes up the vehicle as a whole. For the worst-case scenario, the loads at the frame's hardpoints were obtained (Fig. 12). These points were the anchors of the suspension and steering elements. The calculated values are included in Tables 2 and 3.

Table 2. Location and forces of the hardpoints on the right side of the frame

Hard point	x (m)	y (m)	z (m)	F_x (N)	F_y (N)	F_z (N)
A	0.0705	-0.1753	0	24	4171	3327
B	0.2545	-0.3330	0.6318	7	3178	8763
C	0.6030	-0.2278	0	16	927	-1503
D	1.7110	-0.2519	0.1612	6	1179	-90
E	1.7560	-0.2427	0	9	314	-106
F	0.1968	-0.2939	0.4026	1	239	1176
G	2.0400	-0.2243	0	4	2572	985

H	2.0686	-0.2207	0.1612	3	876	117

Table 3. Location and forces of the hardpoints on the left side of the frame

Hard point	**x (m)**	**y (m)**	**z (m)**	**F_x (N)**	**F_y (N)**	**F_z (N)**
A	0.0705	0.1753	0	27	-5315	2580
B	0.2545	0.3330	0.6318	9	-3125	8540
C	0.6030	0.2278	0	15	-843	-1608
D	1.7110	0.2519	0.1612	8	-997	-110
E	1.7560	0.2427	0	11	-438	-183
F	0.1968	0.2939	0.4026	2	-246	1235
G	2.0400	0.2243	0	3	-2466	756
H	2.0686	0.2207	0.1612	5	-1104	102

By applying the loads to the frame, the tension state was obtained, as shown in the Fig. 13(a). The stress analysis verified that they results did not exceed the maximum value allowed by the material. In the same way, the strains (Fig. 13(b)) did not exceed the established limit.

As shown in Fig. 14, the design met the three established levels. Firstly, the design fulfilled the first level corresponding to geometric, topological and material conditions. Secondly, the design complied with the static conditions for vertical, longitudinal and lateral loads. Finally, the design complied with the stress and strain level based on the dynamic worst case obtained from the virtual study with a multibody model.

4. Conclusions

The international regulations governing the technical conditions for the design of the frame of a Cross Car were analysed. This study concluded that designs that meet these conditions do not have to meet other demanding conditions. There are other more demanding technical regulations that pose structural verification by means of virtual tests using the finite element method. With this in mind, a three-tiered sequential methodology has been proposed.

The first level corresponds to the geometric, topological and material verification in accordance with international regulations. The second level corresponds to the frame verification under static loads in the vertical, longitudinal and vertical directions, which is in line with advanced regulations, such as those that exist in some national competitions.

The third level of verification corresponds to a dynamic verification. This structural verification takes the loads that come from a dynamic calculation of the behaviour of the complete vehicle under different driving conditions. The worst case is selected and then the actions on the hardpoints are determined. This load case allows for the analysis of the structural response of the vehicle assembly to a combination of different forces and torques. This third level provides a structural check not considered in the previous levels. This third level includes the forces from the powertrain, brakes, suspension, steering and the rest of the elements included in the vehicle, such as the weight of the pilot. It allows a detailed analysis of, for example, anchor lugs, motor mounts, welded areas, etc. These detailed analyses are not included in any international or national regulations. This third level, through detailed analysis, makes it possible to advance in a more precise structural design. Therefore, it allows optimizing the design process with respect to other methodologies that do not include these effects.

REFERENCES

[1] Appendix J 2020 - Article 279B - Technical Regulations for Cross Cars - WMSC 04.12.2019 - Updated 06.11.2020Published on 06.11.20 | Federation Internationale de l'Automobile (fia.com)

[2] http://www.rfeda.es/documents/20185/776060/Normativa+Homologacion+ES

[3] Raphael B.; Smith I.F.C. (2013). *Fundamentals of computer aided engineering*, 2nd edition, John Wiley & Sons Inc, Hoboken, NJ (USA)

[4] Garcia de Jalon, J.; Bayo, E. (1994). *Kinematic and Dynamic of Multibody Systems. The Real-Time Challenge*, 1st edition. Springer-Verlag, Mechanical Engineering Series, New York

[5] Blundell, M.; Harty, D. (2014). *Multi-Body Systems Approach to Vehicle Dynamics*, 2nd edition Elsevier Butterworth-Heinemann, doi: 10.1016/C2012-0-07000-7

[6] Karnopp, D.C., Margolis, D.L.; Rosenberg, R.C (2006). *System Dyamics. Modelling and simulation of mechatronic systems*, 4th edition, John Wiley & Sons Inc, Hoboken, NJ (USA).

[7] Callejo, A.; García De Jalón, J.; Luque, P.; Mantaras, D.A. (2015). Sensitivity-Based, Multi-Objective Design of Vehicle Suspension Systems, *Journal of Computational and Nonlinear Dynamics*, Vol 10, No. 3, doi: doi.org/10.1115/1.4028858.

[8] Cuesta, C; Luque, P.; Mantaras, D.A. (2016). State estimation applied to non-explicit multibody models, *Nonlinear Dynamics*, Vol. 86, 1673-1686, doi: 10.1007/s11071-016-2985-9

[9] Kanchwala, H.; Luque, P.; Mántaras, D.A.; Wideberg, J.; Bendre, S. (2017). Obtaining Desired Vehicle Dynamics Characteristics with Independently Controlled In-Wheel Motors: State of Art Review. *SAE International Journal of Passenger Cars - Mechanical Systems*, Vol. 10, No. 2, 413-425, doi: 10.4271/2017-01-9680

[10] Henning, K. U.; Sawodny, O. (2016). Vehicle dynamics modelling and validation for online applications and controller synthesis, Mechatronics, Vol. 39, 113–126, doi: 10.1016/j.mechatronics.2016.08.010

[11] Carrera, M.; Herrer, L.; Litago, H.; Valladares, D.; Lafuente, C., Jalón, J. Luque, P.; Mántaras, D.A.; Alcalá, E.; Martín, A. (2016). Fatigue life analysis of semitrailers. Fatigue testing bench and numerical simulation. *Dyna*, Vol. 91, 406-412, doi: http://dx.doi.org/10.6036/7886

[12] Tello, L.; Castejón, L.; Malón, H.; Valladares, D.; Luque, P.; Mántaras, D.A.; Ranz, D.; Cuartero, J. (2020). Development of a fatigue life prediction methodology for welded steel semi-trailer components based on a new criterion. *Engineering Failure Analysis,* Vol. 108, 1-26, doi: https://doi.org/10.1016/j.engfailanal.2019.104268

[13] Luque, P.; Mántaras, D.A.; Pello, A. (2013). Racing car chassis optimization using the finite element method, multi-body dynamic simulation and data acquisition. *Proceedings of the Institution of Mechanical Engineers, Part P: Journal of Sports Engineering and Technology*, Vol. 227, No. 1, 3-11, doi: 10.1177/1754337112444517

[14] Mantaras D. A.; Luque P. (2015). Assessing Motorcyclist Protection Systems Using Finite Element Simulations, *Int. Journal of Simulation Modelling*, Vol. 14, No. 1, 110-120, doi:10.2507/IJSIMM14(1)10.294

[15] Mántaras, D.A., Luque, P.; Vera, C. (2003). A three-dimensional kinematic model for compound crank axle suspension mechanism. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, Vol. 217, No. 2, 115-123, doi: 10.1177/095440700321700205

[16] Mántaras, D.A.; Luque, P. (2006). Ride comfort performance of different active suspension systems. *International Journal of Vehicle Design*, Vol. 40, 106-125, doi: 10.1504/IJVD.2006.008456

[17] Luque, P.; Mántaras, D.A.; Maradona, A.; Roces, J.; Sánchez, L.; Castejón, L.; Malón, H. (2020). Multi-Objective Evolutionary Design of an Electric Vehicle Chassis. Sensors, Vol. 20, No. 13, 1-21, doi: 10.3390/s20133633

[18] Mantaras, D.A.; Luque, P.; Vera, C. (2004). Development and validation of a three-dimensional kinematic model for the McPherson steering and suspension mechanisms, *Mechanism and Machine Theory*, Vol. 39, No. 6, 603-619, doi: 10.1016/j.mechmachtheory.2003.12.006

[19] Luque, P.; Mántaras, D.A.; Rodríguez, A.; Malón, H.; Castejón, L.; Jalón, J.; López, J.; Martín, A. (2016). Low-Cost Monitoring System of Sensors for Evaluating Dynamic Solicitations of

Semitrailer Structure. *Journal of Sensors*, Vol. 2016, Article ID 1740854, 15 pages, doi: https://doi.org/10.1155/2016/1740854

[20] Luque, P.; Mántaras, D.A.; Wideberg, J. (2011). Evaluation of the road network: Use of a monitoring vehicle for the record of potentially dangerous events. *Dyna*, Vol. 86, 431-437, doi: http://dx.doi.org/10.6036/3990

[21] Martínez, A.; Mántaras, D.A.; Luque, P. (2013). Reducing posted speed and perceptual countermeasures to improve safety in road stretches with a high concentration of accidents. *Safety Science*. Vol. 60, 160-168, doi: 10.1016/j.ssci.2013.07.003

[22] Alonso, M.; Mántaras, D.A.; Luque, P. (2019). Toward a methodology to assess safety of a vehicle. *Safety Science*, Vol. 119, 133-140, doi: 10.1016/j.ssci.2019.01.012

[23] Alonso, M.; Mántaras, D.A.; Luque, P. (2018) Methodology for determining real time safety margin in a road vehicle. *Transportation Research Procedia*, Vol. 33, 331-338, doi: 10.1016/j.trpro.2018.10.110

[24] Luque, P.; Mántaras, D.A.; Orueta, A. (2013). Passive safety assessment in rally vehicle using virtual crash test. *Dyna*, Vol. 88, 453-461, doi: 10.6036/5443

[25] Kim, B. S.; Kim, T. G. (2019). Cooperation of Simulation and Data Model for Performance Analysis of Complex Systems. *Int. Journal of Simulation Modelling*, Vol. 18, No. 4, 608- 619, doi: 10.2507/IJSIMM18(4)491

[26] Allonca, D.; Mantaras, D.A; Luque, P.; Alonso, M. (2019). A new methodology to optimize a race car for inertial sports. *Proceedings of the Institution of Mechanical Engineers, Part P: Journal of Sports Engineering and Technology*, Vol. 233, No.2, 312-323, doi: 10.1177/1754337118823971

[27] Luque, P.; Mantaras, D.A.; Fidalgo, E.; Alvarez, J.; Riva, P.; Giron, P.; Compadre, D.; Ferran, J. (2013). Tyre-road grip coefficient assessment - Part II: Online estimation using instrumented vehicle, extended Kalman filter, and neural network, *Vehicle System Dynamics*, Vol. 51, No. 12, 1872-1893, doi: 10.1080/00423114.2013.841963

[28] Mantaras, D.A.; Luque, P.; Nava, J.A.; Riva, P.; Giron, P.; Compadre, D.; Ferran, J. (2013). Tyre-road grip coefficient assessment. Part 1: Off-line methodology using multibody dynamic simulation and genetic algorithms, *Vehicle System Dynamics*, Vol. 51, No. 10, 1603-1618, doi: 10.1080/00423114.2013.818157

[29] Mantaras, D.A.; Luque, P. (2012). Virtual test rig to improve the design and optimisation process of the vehicle steering and suspension systems, *Vehicle System Dynamics*, Vol. 50, No. 10, 1563-1584, doi: 10.1080/00423114.2012.680472

[30] Zeng, X. T.; Meng, G. Y. & Zhou, J. H. (2018) Analysis on the Pose and Dynamic Response of Hydraulic Support under Dual Impact Loads, *Int. Journal of Simulation Modelling*, Vol. 17, No. 1, 69-80, doi: 10.2507/IJSIMM17(1)412

[31] Zeng, X. T.; Meng, G. Y. & Zheng, K (2019) Force Transmission Analysis of Sliding Block-Type Hydraulic Support under Impact Loads, *Int. Journal of Simulation Modelling*, Vol. 18, No. 1, 100-111, doi: 10.2507/IJSIMM18(1)466

[32] Cuesta E.; Mantaras D. A.; Luque P.; Alvarez B. J.; Muina D. (2015). Dynamic Deformations in Coordinate Measuring Arms Using Virtual Simulation. *Int. Journal of Simulation Modelling*, Vol. 14, No. 4, 609-620, doi: 10.2507/IJSIMM14(4)4.311

[33] https://www.fiakarting.com/sites/default/files/2020-03/Reglement%20d%27Homologation%202020_0.pdf

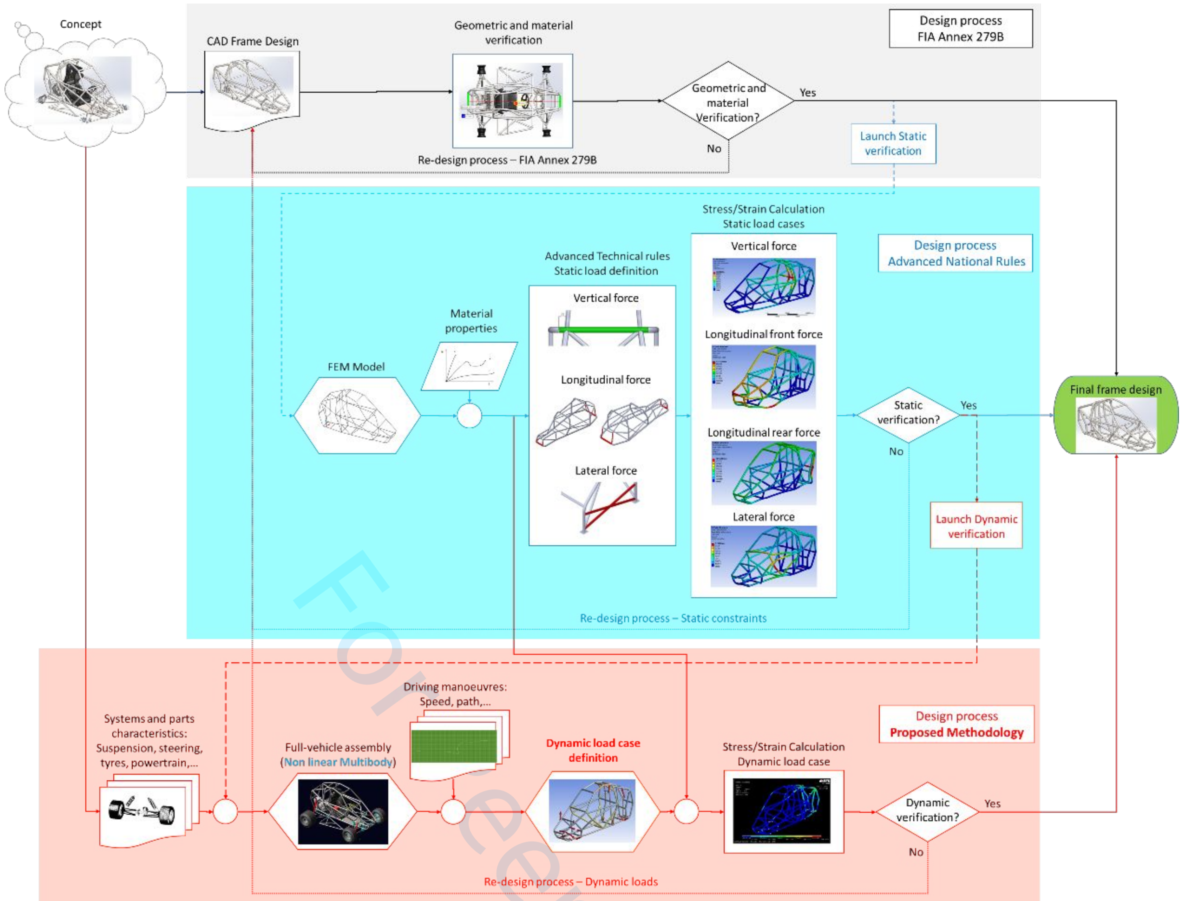

Figure 1. Flowchart of the proposed methodology

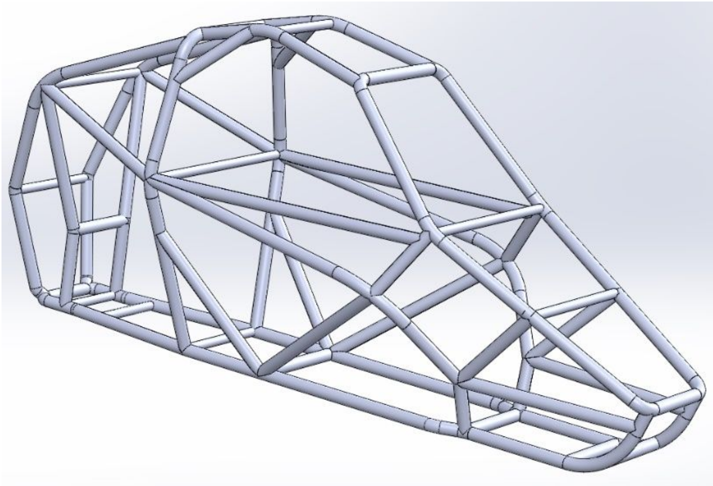

Figure 2. 3D Frame initial design

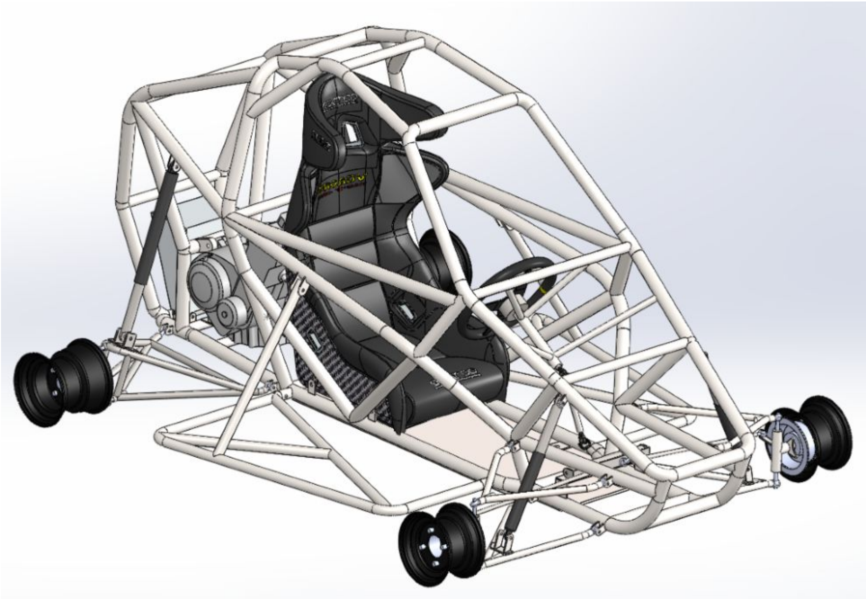

Figure 3. Initial complete conceptual design

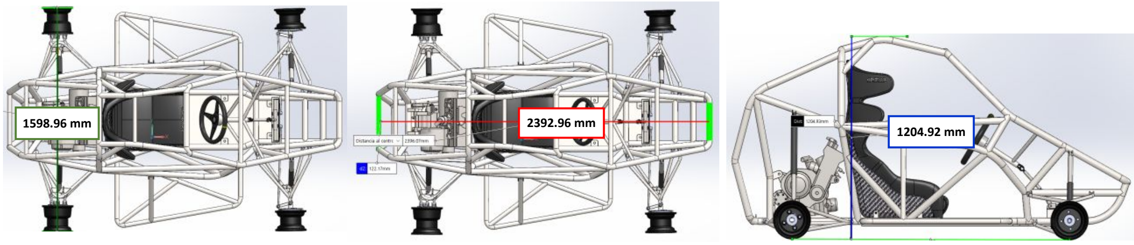

Figure 4. Dimensions check (length, width, and height)

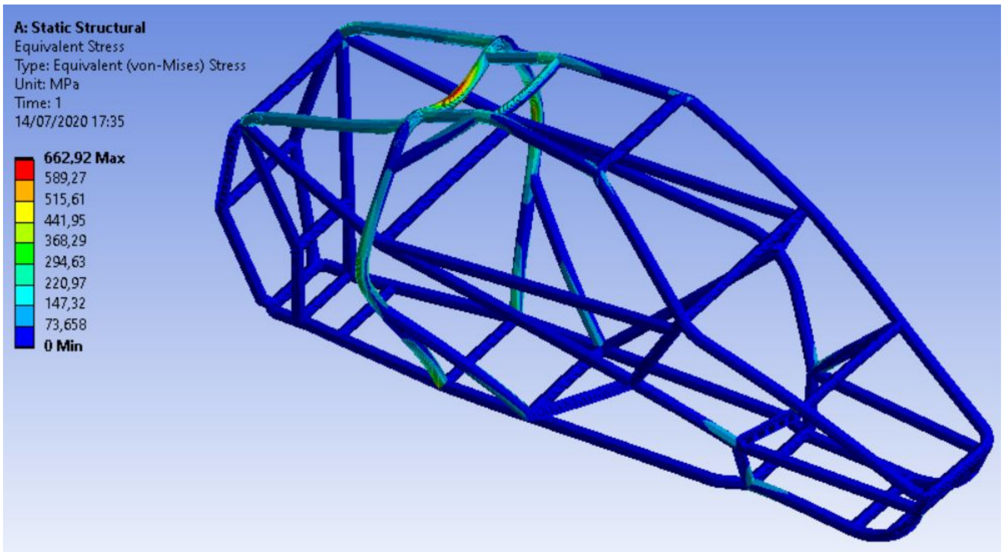

Figure 5. Vertical load case - Initial design

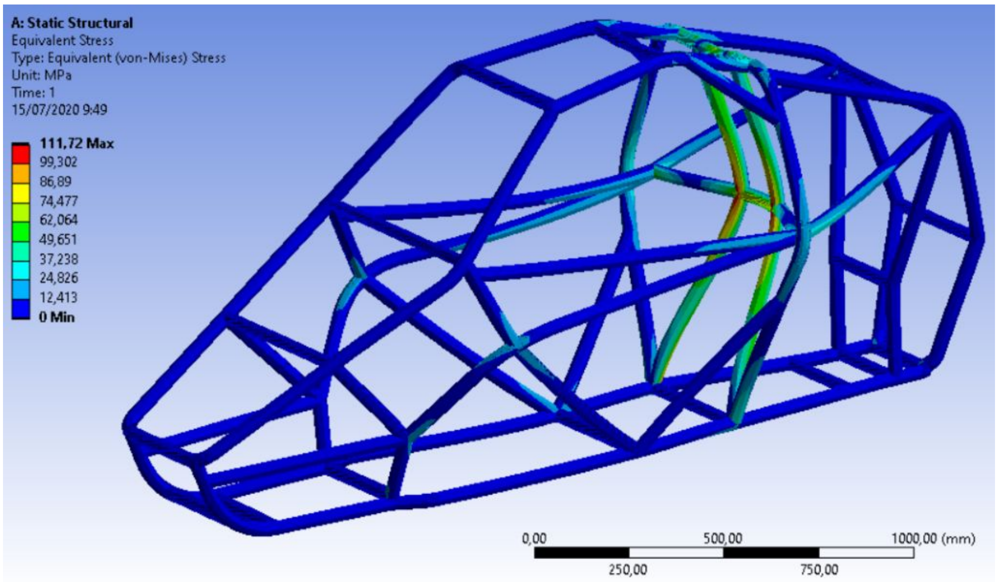

Figure 6. Vertical load case - Reinforced redesign of the frame

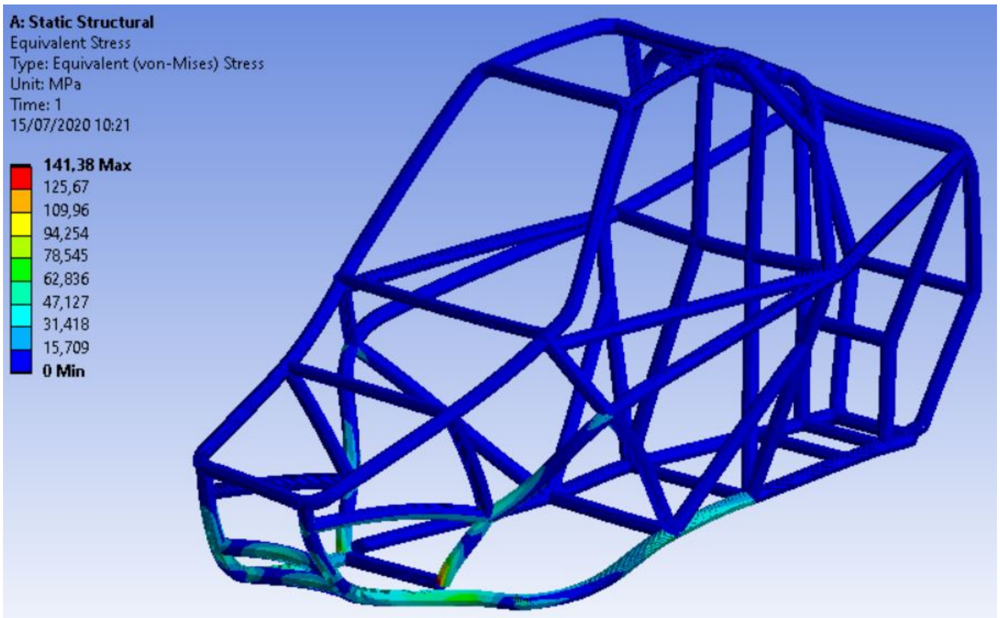

Figure 7. Stresses of the longitudinal load test of the front part

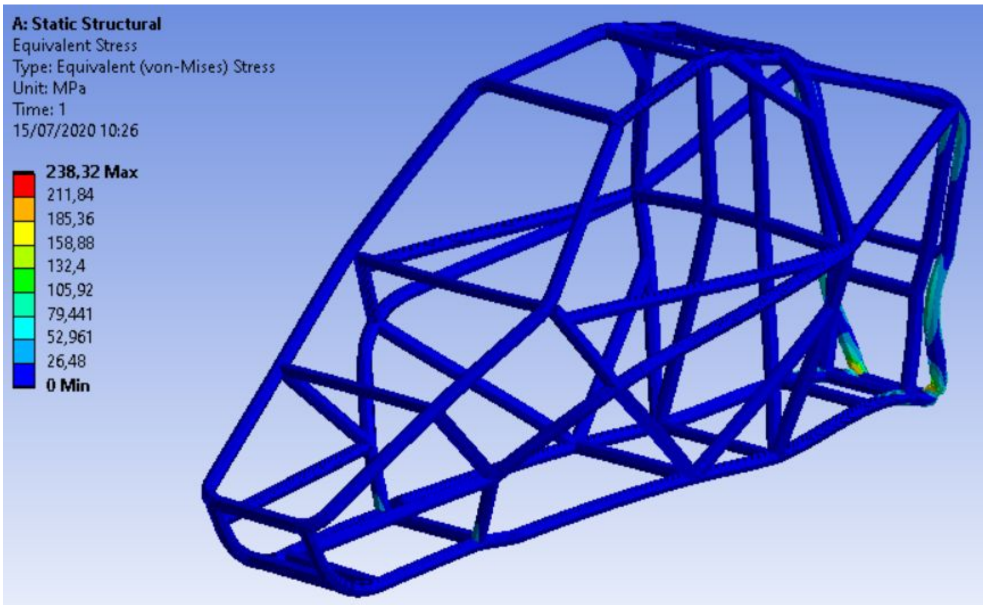

Figure 8. Stresses of the longitudinal load test of the rear part

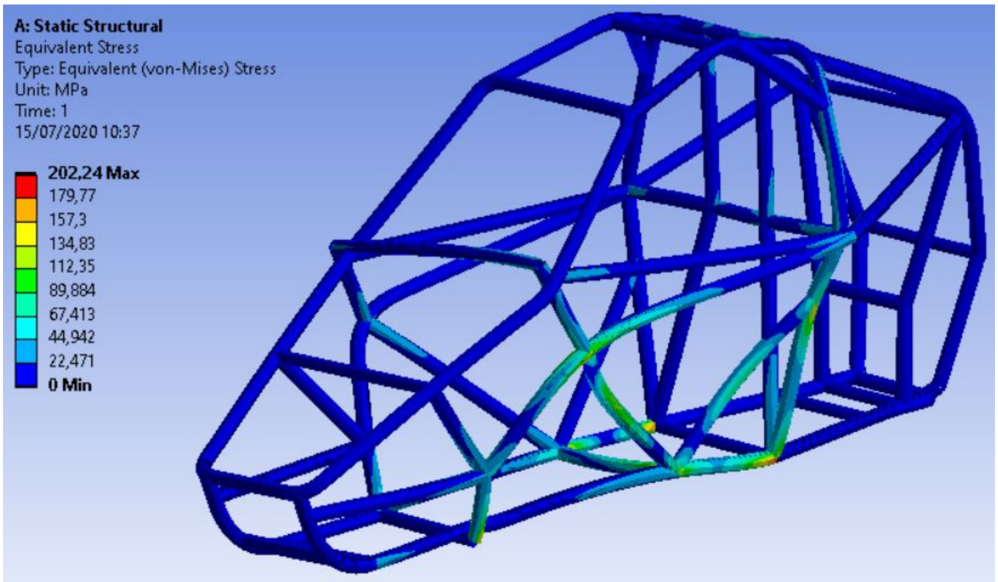

Figure 9. Stresses of the lateral load test

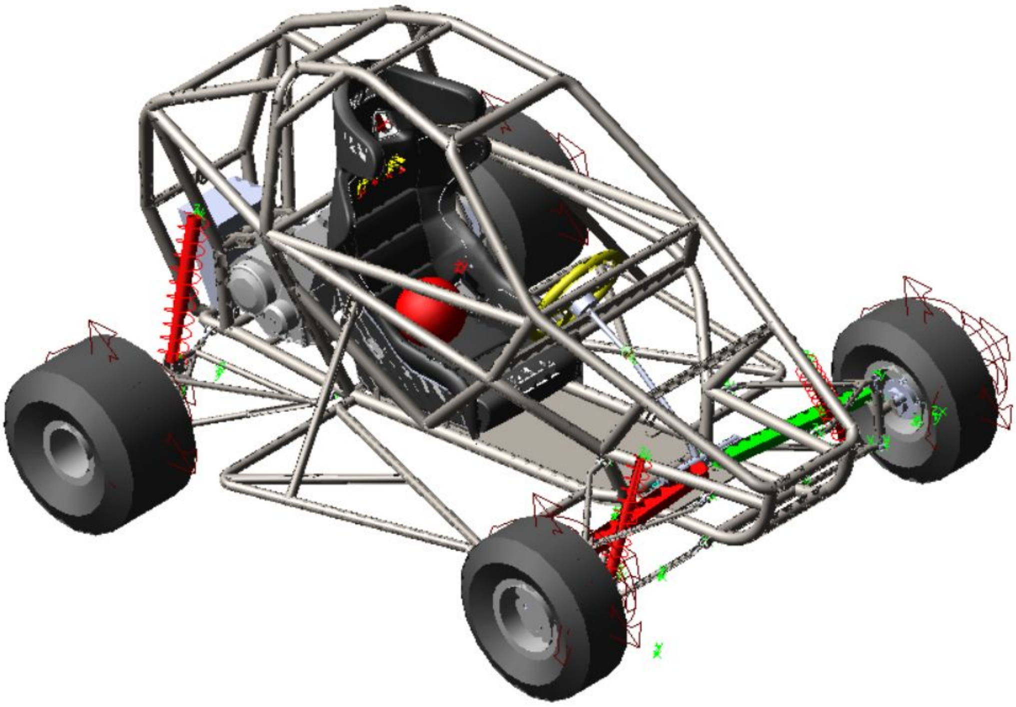

Figure 10. Multibody model of the designed Cross-Car

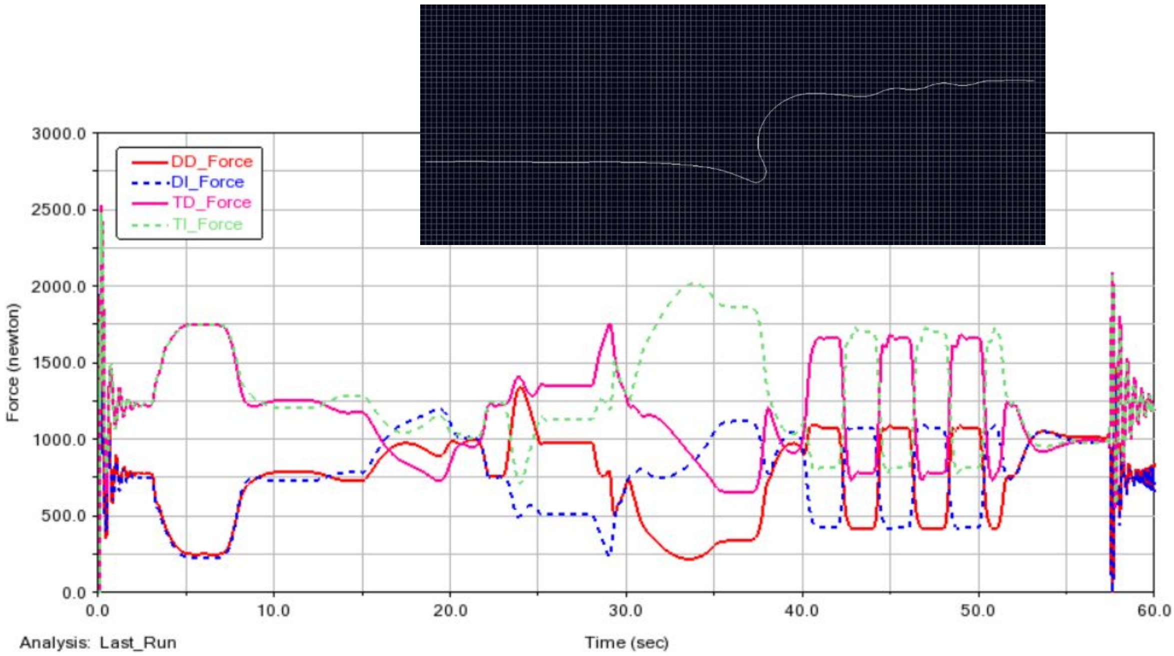

Figure 11. Example of tire loads in a virtual test circuit

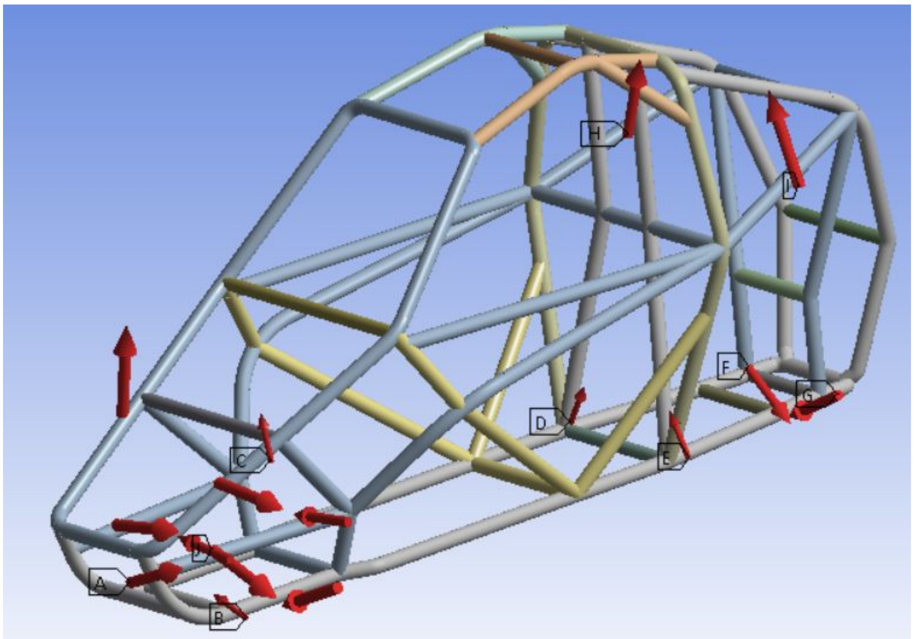

Figure 12. Suspension and steering hardpoints and dynamic load status

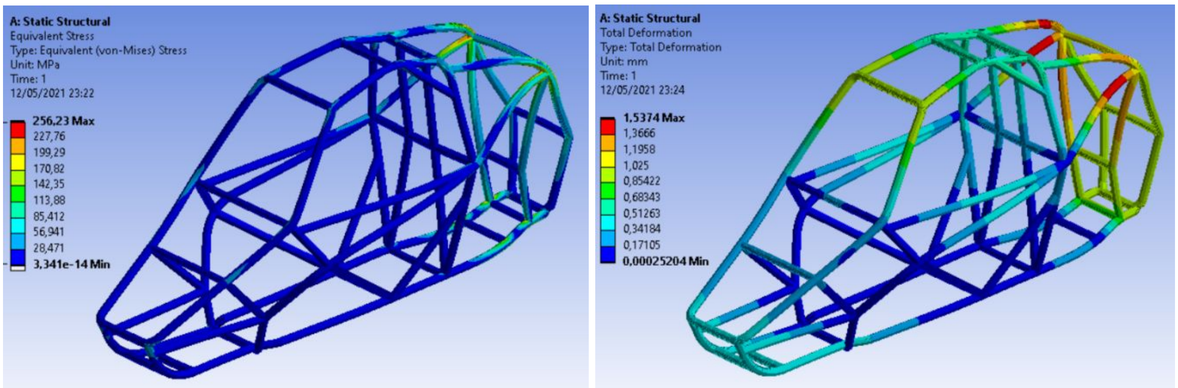

Figure 13. (a) Von Misses stresses and (b) strains when applying the dynamic state of loads

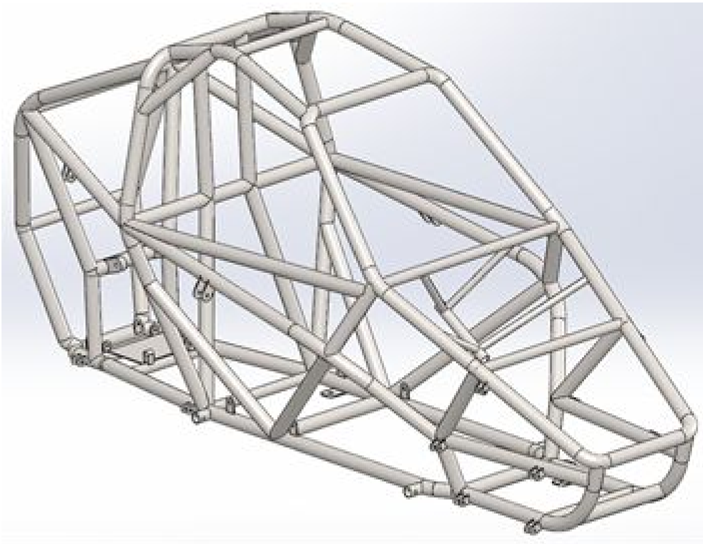

Figure 14. Final design of the frame that meets the three levels