Team #27

CETYS Universidad Baja SAE Design Report

Juan Antonio Figueroa Marrufo
Team captain

Alejandro Arias, Rodrigo Valdez, Leonardo Rodriguez, Josue Romero

Team members

ABSTRACT

ZR-2017 is the 15th competition vehicle from the Cetys University Baja SAE Team. The vehicle is a 4130 steel tube frame with 23 in tires powered with a 10 hp Briggs and Stratton engine running on gasoline. The team is focused on analyzing the strengths and weaknesses of the previous version, as to work on those things that made us fail in the competition.

INTRODUCTION

This report summarizes the work developed by the team members during the conceptual design, mechanical design and manufacturing processes of the vehicle, as well as the redesign and manufacturing processes done on the vehicle afterwards. The start point was the drive train system; making a research and comparison with past competitions, the initial parameters were taken for the car to be able to climb a 50° hill, 140 ft long in 25 seconds. The chassis is lean-designed to provide sufficient safety and stiffness. The front suspension is an unequal parallel double A-arm with 9.75 in of travel, while the rear suspension is a trailing arm with 13 in of travel. The steering system has a fast ratio rack and pinion with 4.5 in lock to lock travel. 3D modeling, finite element and kinematic analyses were done with SOLIDWORKS® 2016, Patran® 2011 and Adams® 2015.

TEAM OVERVIEW

Z Racing Team's way of work consists in 6 systems: 1)Drive train, 2)Chassis, 3)Front suspension and Steering, 4)Rear suspension, 5)Braking, and 6)Electric. Each system developed its design and analysis separately, but integration of all the systems was carefully taken in consideration to guarantee the fittings between them. Several critical scenarios were taken in consideration for the structural analyses; these are referred to as *load cases.*

DRIVETRAIN

OBJECTIVE – The main objective is to optimize the torque transmission performance from the engine to the wheels. The drivetrain is designed to provide enough torque for a 50° hill climb without forfeiting speed. The design assures a maximum axle torque of 653 lb-ft (885 N-m) at 2600 rpm and a top speed of 27 mph (43.5 km/h) at 3600 rpm.

DESIGN – The drive train is composed by a Briggs & Stratton 10 hp engine, a QDS® 780 Series CVT and a DANA® Spicer H-12 FNR C Independent Suspension Transaxle. In order to select these components a study case with a 50 degree hill climb was evaluated with the loads acting on the car, see Figure 1 at the appendix.

The Briggs & Stratton engine provides a maximum torque of 13.75 lb-ft (18.6 N-m) at 2600 rpm while top speed is governed at 3600 rpm. The QDS® 780 Series CVT Pulley System produces a 0.69:1 high ratio and 3.71:1 low ratio reduction. The DANA® Spicer H-12 FNR C Transaxles maintains a 13.25:1 forward ratio and a 14.36:1 reverse ratio reduction which encloses a forward-neutral-reverse helical gearing with a limited slip differential. Table 1 shows the total drivetrain reduction at CVT's high and low ratios.

Table 1: Drivetrain Total Reduction

Component	High Ratio	Low Ratio
QDS CVT 780 Series	0.69	3.71
DANA Spicer H-12 FNR C	13.25	13.25
Total Drivetrain Reduction	***9.14***	***49.15***

In order to improve the torque performance of the CVT pulley, a belt tensioner is implemented. This mechanism is mounted on the engine below the CVT drive pulley. It is actioned by a pedal located on the left side of the brake one. This allows the CVT to perform at its highest torque capability for any velocity. A CAD model can be found in Figure 2 in the appendix section. No significant changes were made to the Drive Train.

CHASSIS

OBJECTIVE – The chassis goal is to keep all the systems working integrally while assuring the driver's safety. Its design allows the frame to endure the most critical load case, a rollover scenario with a 3.1Gs impact on the Rear Roll Hoop. A 5% weight increase compared to the last vehicle's frame was achieved; from 87 lb to 92 lb.

DESIGN – The frame is made of 4130 steel of different diameters and thicknesses that comply with the Baja SAE 2017 Rules (Figure 3). Table 2 shows the comparison between 1018 steel and 4130 steel from which the team based its material justification.

Table 2: 1018 vs 4130 Steel Comparison

Material / Property	**Primary Members**		**Secondary Members**	
	1018 Steel	**4130 Steel**	**1018 Steel**	**4130 Steel**
Outer Diameter in (mm)	1.00 (25.40)	1.25 (31.75)	1.00 (25.40)	1.00 (25.40)
Wall Thickness in (mm)	0.120 (3.048)	0.065 (1.651)	0.035 (0.889)	0.035 (0.889)
Bending Strength (S_yI/c) lb-ft (N-m)	288.61 (391.30)	569.53 (772.18)	109.120 (147.947)	206.669 (280.205)
Bending Stiffness (EI) lb-ft² (N-m²)	6746.60 (2791.13)	8786.72 (3635.15)	2550.79 (1055.28)	2550.79 (1055.28)
Weight per Unit of Length [illegible] ([illegible])	1.131 (1.683)	0.822 (1.223)	0.362 (0.538)	0.360 (0.536)

MIG welding with ER70-6 filler was used to assemble the tubes that form the frame.

ANALYSIS – In order to analyze the chassis structural strength, Patran 2011® was used to model and simulate all the previously defined load cases. See Figure 4.

An energetically-derived inertial loads method was used taking in consideration that the frame would absorb all the potential and/or kinetic energy resulting from the impact. After reviewing the results of every critical load case, the thickness of the elements that failed was increased and re-analyzed until a safe result was acquired. The analysis of secondary members lead to an increase in thickness of some of them from 0.035 in (0.889 mm) to 0.049 in (1.244 mm) or to 0.065 in (1.651 mm) depending of the case.

ERGONOMICS – The roll cage is designed to occupy the least space possible without risking the pilot's comfort and security. This means that in case of an impact, the pilot is at a safe global distance of at least 3 in (76.2 mm) from the SIM's and 6 in (152.4 mm) from the Roll Hoop Overhead and Front Bracing members.

The steering wheel is placed at a safe distance from the pilot's legs of at least 6 in (152.4 mm) and has an inclination angle of 25° with respect to the floor, allowing the pilot to quickly exit from the vehicle during an emergency and eases driving.

The pedals are placed 22 in (457.2 mm) in front of the seat, allowing the pilot to reach them without completely stretching his legs, which represents a sense of comfort.

BRAKING SYSTEM

OBJECTIVE – The main objective of the braking system is to make the car completely stop from its top speed in a 10 ft (3 m) distance interval by locking all the four wheels at the same time.

DESIGN – The system is composed with a pair of VW® 0.748 in (19 mm) master cylinders actioned by a single pedal, a pair of Yamaha YFX450V calipers in the front and a set of Can-Am Maverick 1000 XDS calipers in the rear.

The Yamaha calipers fit with the selected steering knuckle and hubs, while the Can-Am calipers fit with the designed rear knuckles for the rear suspension system.

ANALYSIS & ERGONOMICS – The brake pedal is designed for the pilot to apply the minimum amount of force through the foot. This was achieved by making a proper mechanism that matched the calculations developed. This aids the pilot by decreasing fatigue due to braking. Ease of manufacturing and light weight were taken in consideration for the design.

FRONT SUSPENSION

OBJECTIVE – The suspension system is designed to provide ride comfort and stability when managing obstacles. In addition, it aids the steering and braking systems in turning and braking maneuvers while reducing the possibility of a rollover.

DESIGN – Suspension design considers basic parameters of the car (CG height, Wheelbase, Track, Ride Height) defined with the previously-developed conceptual design.

The front suspension is an unequal parallel double A-arm. The upper arm is shorter than the lower arm, providing ease to the desired camber angles. It is equipped with FOX Float 3 EVOL R shocks. The spring rate of the shocks is variable with respect to the travel; in average, it is equivalent to 231 lb/in (N/mm). The front suspension has a total wheel travel of 9.75 in (248 mm), with jounce equivalent to 5.5 in (140 mm) and rebound of 4.25 in (108 mm), which is suitable for regular off road conditions.

KINEMATIC ANALYSIS - The kinematic analysis of the front suspension was developed along with the steering system. The ideal position of the shock was defined by a two-dimensional iterative analysis that allowed optimization of its position. The designed parallel A-arms provide a 7.4 in (188 mm) Roll Center above the ground and a 13 in (330 mm) Ride height. (Figure 5).

The mechanism was modeled using SOLIDWORKS® and Adams® to study the variation of its parameters during the travel of both the shock and the rack and pinion. The main parameters taken in consideration were the following: vertical travel, camber angle, toe angle, turning radius and Ackerman percentage.

STRUCTURAL ANALYSIS - The front suspension was modeled together with the steering knuckle using Patran® and SOLIDWORKS® (Figure 6). Different load cases were run and the critical reactions were checked to be safe according to the materials properties. No major modifications were made.

STEERING

OBJECTIVE – The steering system is in charge of the vehicle's control. The design's main goal is to reduce the turning radius and improve the steering stability, while maintaining a comfortable interaction with the driver.

DESIGN - The steering system works with a Stiletto Rack and Pinion 6.4:1 Ratio connected to a 10 in Steering wheel via universal joint. The rack travels 7/8 turns from lock to lock and travels 4.5 in (114.5 mm). This rack and pinion is light and provides a fast ratio, reason for which it was selected. The rack is connected to the 4130 steel 0.75 in (19 mm) OD tie rods on its both ends with a C-shape designed joint. The steering and suspension systems work together to provide a caster angle of 2.11°.

Human factor was taken in consideration for the steering wheel inclination, position and diameter.

An iterative geometric analysis performed with SOLIDWORKS® and Adams® to find the ideal position of the rack and pinion and tie rod lengths (see figure 7). Turning radius and Ackerman percentage were critical parameters for a suitable configuration. The final position of the rack and pinion yields an outside turning radius of 9 ft (2.74 m), which is enough to provide control and maneuverability (see table 3). Table 3: % Ackerman and Outside turning Radius

Steer travel (in)	% Ackerman	Outside Turning Radius (ft)
0	-103%	
0.45	161%	44.2
0.9	102%	21.6
1.35	77%	14.7
1.8	61%	11.2
2.25	51%	9.0

No major modifications were made.

REAR SUSPENSION

The rear suspension of an off road vehicle is the connection of the unsprung mass to the sprung mass by a mechanism and a damper or shock absorber. In this case, the rear suspension carries 52% of the ZR total weight.

OBJECTIVE – Its main purpose is to absorb and dissipate the energy due to control forces produced by the tires: longitudinal (acceleration and braking) forces, lateral (cornering) forces, braking and driving torques, and the vibrations caused by the uneven terrain. This energy is converted to kinetic energy (suspension travel) and energy absorbed and dissipated by the shocks. It also aids to prevent roll of the chassis and provides vertical compliance by maintaining the tires in contact with the road while keeping the vehicle stable and the driver safe and comfortable.

DESIGN – The rear suspension is a trailing arm type with 2 equal parallel control arms; all joint to a 7071 aluminum knuckle, a designed and single CNC manufactured part (see Figure 8).

In order to achieve this edition primary goal of a 30% weight reduction, the design of the suspension components was focused to use the least material possible with sufficient resistance, while complying with the required wheel travel. The knuckle design allows it to join the trailing arm with the use of plugs that also restrain the degrees of freedom in order to achieve a controlled motion during the suspension travel. This design also allows to fasten the brake caliper to it, and the equal control arms by the use of ball joints (rod ends) as shown in Figure 9. Both the trailing arm and the control arms are made out of 4130 steel 1 in (25.4 mm) OD tube with a wall thickness of 0.065 in (1.651 mm).

The total wheel travel of the rear suspension is 13 in (330 mm), being 9.5 in (241 mm) in jounce and 3.5 in (89 mm) in rebound, this values were obtained with the kinematic analysis.

The shock absorber used for this system is the same as in the front suspension: FOX Float 3 EVOL R model with a spring rate of 231.2 lb/in (40.66 N/mm) in average, since it is variable over the travel.

ANALYSIS – The structural analysis of the whole rear suspension assembly was made according to the load cases in SOLIDWORKS®, as seen in Figure 10. All the loads (forces, moments and torques) were applied in the area where the bearing is mounted and the fixtures were made at the ends of the tabs that join the chassis.

TIRES AND WHEELS

OBJECTIVE – Wheels and tires goal is to convert the drive train torque into push force to accelerate the car, while proportioning traction and maneuverability to the vehicle.

DESIGN – ZR is designed to drive with four 23 in (584 mm) tires. Front tires are selected to help steering and maneuverability while rear tires to enforce traction with the road. Wheels are made from roll forged, 6061 heat treated aircraft grade aluminum manufactured by Douglas Wheels, selected by their simplicity and low weight, while providing strength to endure rough terrain. Table 4 summarizes the selected wheels and tires.

Table 4: Tires and Wheels

Tires		
	Front	***Rear***
Size	23x7x10	23x8x12

Model	Maxxis RAZR	Carlisle 489 A/T
Wheels		
	Front	***Rear***
Size	10x5 2N + 3B 4/156	12x8 3N + 5B 4/136
Model	Douglas .190	Douglas .190

ELECTRIC SYSTEM

The electric system is composed by the brake and reverse lights, a reverse alarm and two kill switches. The switches are mounted in two key locations in the car, the first one is a couple of inches behind the steering wheel on the element that connects it with the rack & pinion; providing the pilot an easy and quick way to shut down the engine. The second switch is mounted on the top right body panel behind the firewall allowing anyone besides the pilot to shut down the engine if necessary.

The electric circuit is designed to use the least wiring possible without sacrificing continuity and current conduction in every component, while being powered by two 6V batteries connected in series to achieve a total voltage of 12V.

CONCLUSION

The process of designing a vehicle is quite a challenge; as a matter of fact it takes a lot of effort from all members of the team to achieve a successful design and manufacture.

The final prototype is the result of a collaborative multidisciplinary design team. The goal of the project is to design and fabricate an off-road vehicle that met or exceed the regulations for safety, durability and maintenance for our client which is SAE, as well as to achieve a vehicle performance, aesthetics and comfort that would have mass market appeal for the off-road enthusiast for what we consider as a successful final product.

The selection of components were made using engineering knowledge achieved through with off-road enthusiast and engineering advisors, taking as parameters first of all, safety, performance, weight, reliability and cost. Computational design became one of the most important part of the process;

The design is an iterative process in which all the parameters involved are relevant are equally important since the effect of one simple of them can cause severe changes in other parameters. The manufacturing process has to be precise and accurate in order to achieve the design pristine conditions. This represents a heavier challenge in the development of the automobile prototype.

The primary objective of 30% weight reduction was successfully achieved. This is due to the followed and mentioned methodology.

LIST OF TABLES

LIST OF FIGURES

REFERENCES

1. Dixon, J. (1996). *Tires, suspension and handling.* USA: SAE.
2. Gillespie, T. (1992). *Fundamentals of vehicle dynamics.* USA: SAE.
3. Juvinall, R. (1967). *Stress, Strain and Strength.* USA: McGraw Hill.
4. Milliken, W. & Milliken, D. (1994). *Race car vehicle dynamics*. USA: SAE.
5. Shigley, J. & Mischke, C. (1989). *Mechanical Engineering Design.* USA: McGraw Hill.
6. Spotts, M. (1978). *Design of machine elements.* USA: Prentice-Hall.

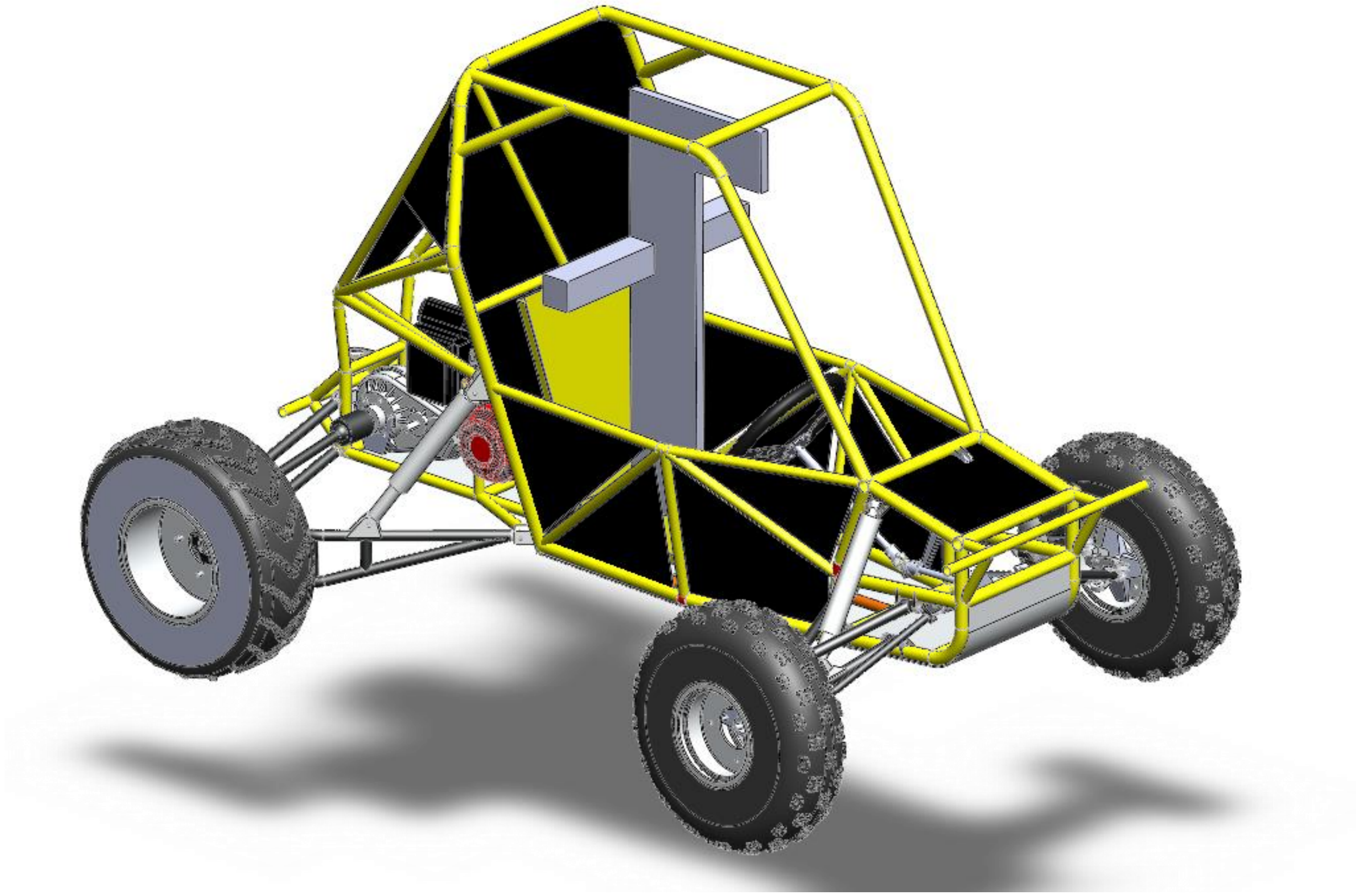
CONTACT

Juan Antonio Figueroa Marrufo – Team captain
juanantonio.figuero@cetys.edu.mx

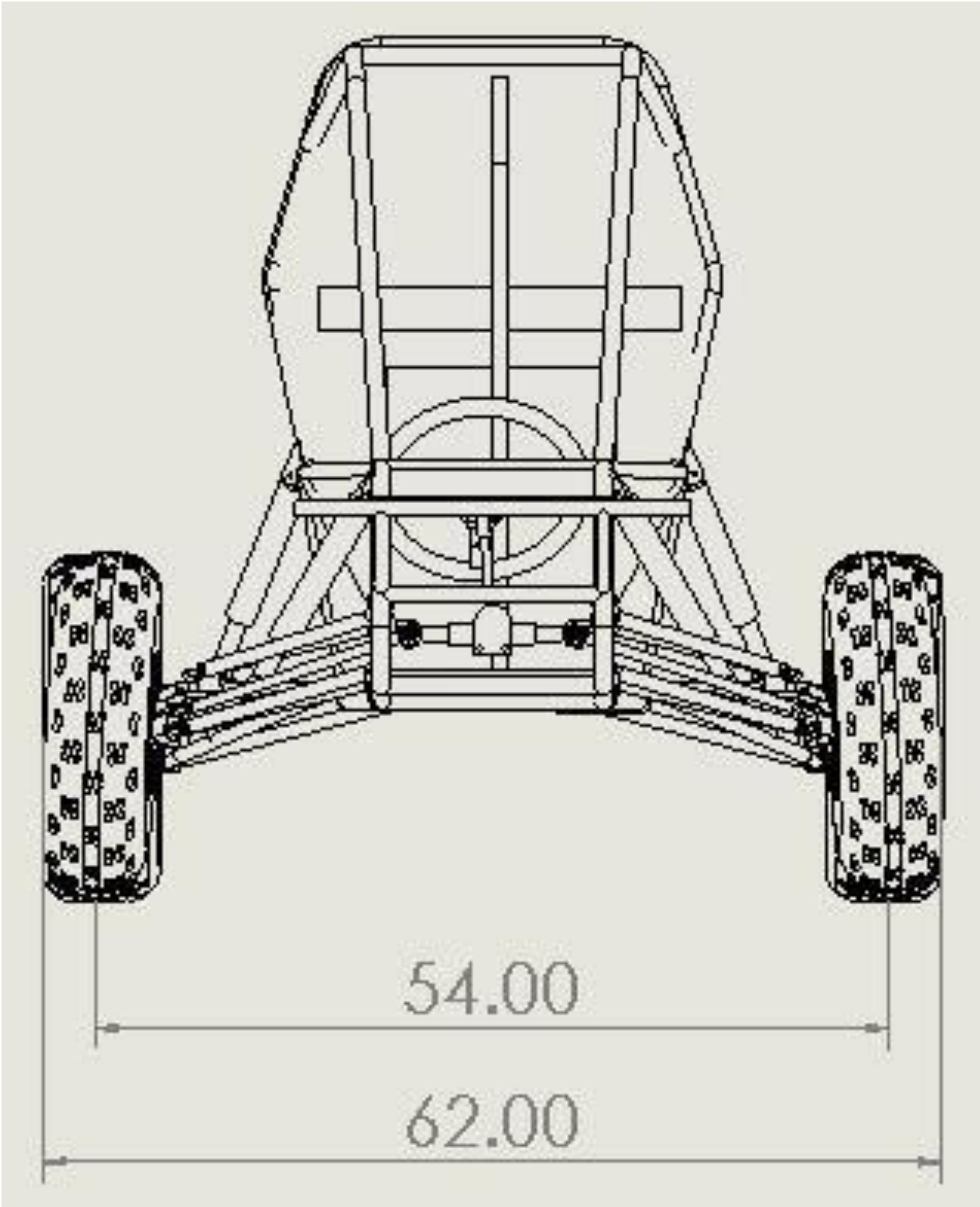
ADDITIONAL SOURCES

- Dassault Sytems SOLIDWORKS ® 2015.
- MSC Patran ® 2011.
- MSC Adams ® 2015.
- CES Edupack ® 2015.

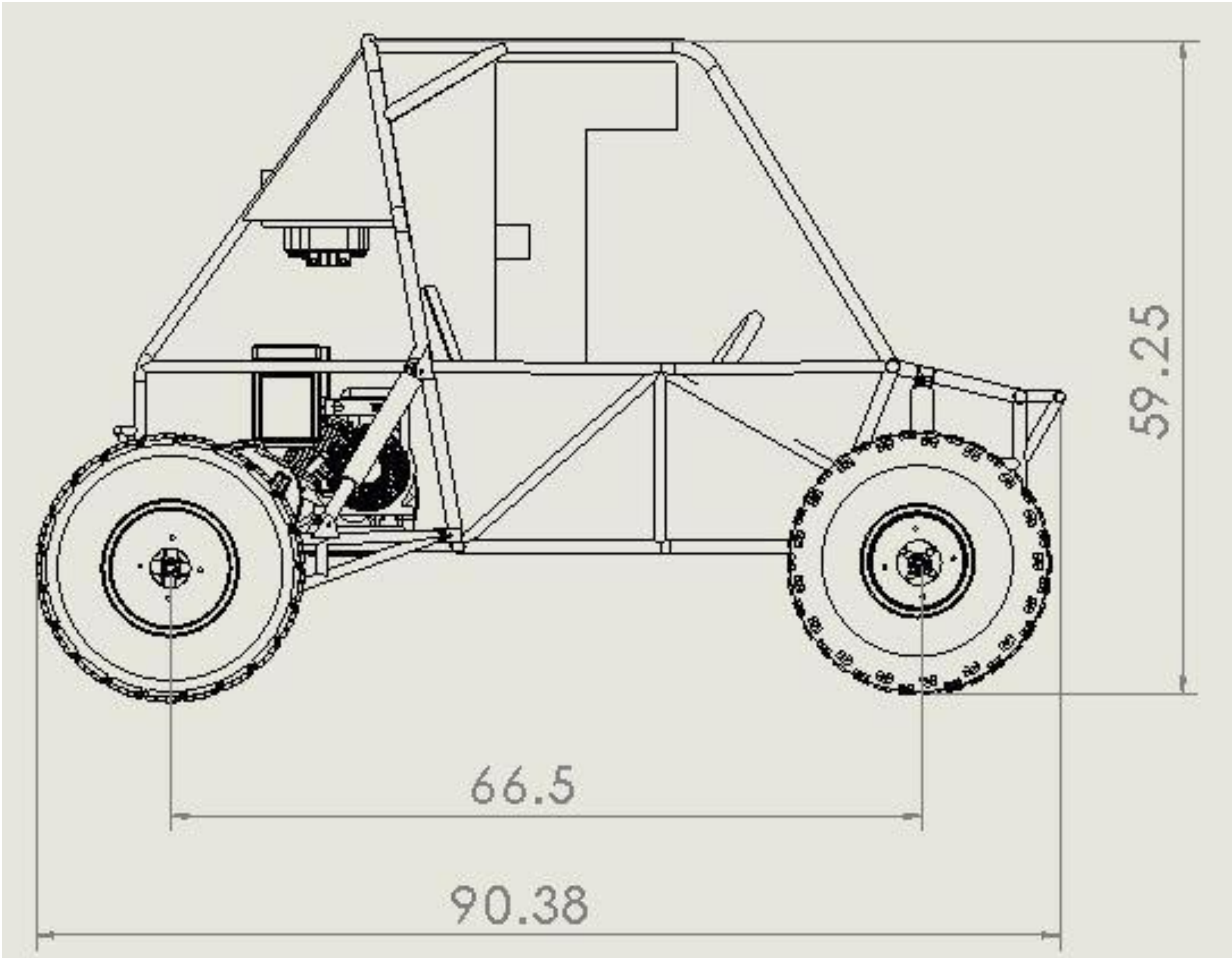
DRAWINGS



Isometric View



Front View



Side View

APPENDIX

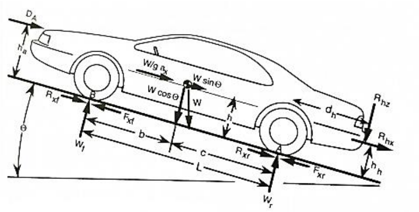


Figure 1 Hill Climb Study Case (Gillespie, 1992)

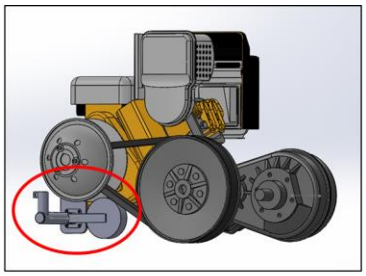


Figure 2 Belt Tensioner Assembly CAD Model

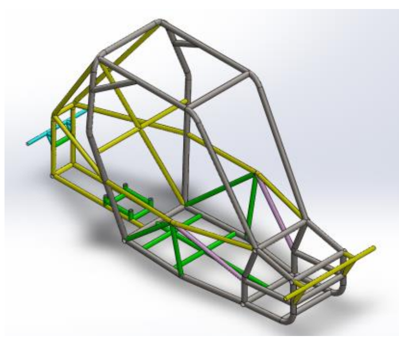


Figure 3 Chassis frame SOLIDWORKS Model

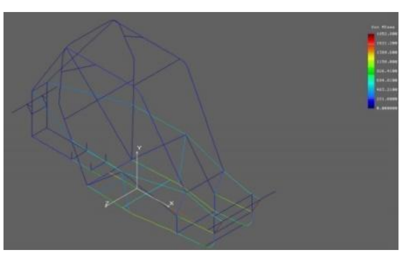


Figure 4 Chassis

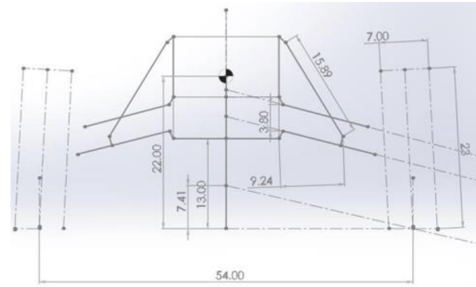


Figure 5 Suspension Basic Parameters

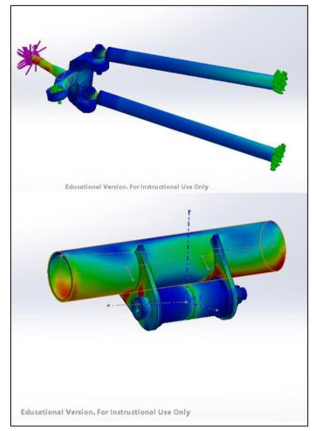


Figure 6 Front Suspension Structural Analysis

Figure 7 Iterative Geometric Analysis with Adams Car

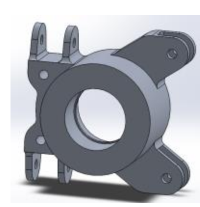


Figure 8 Rear Knuckle CAD Model

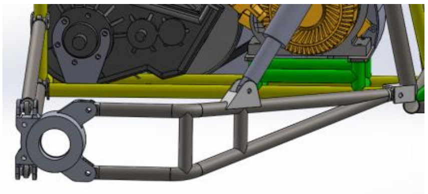


Figure 9 Rear Suspension assembly CAD Model

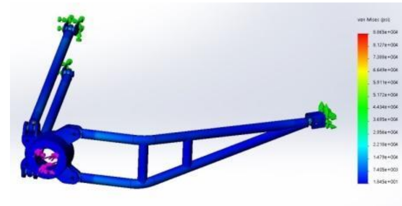


Figure 10 Rear Suspension FEA using SW Simulation