Car No: E-30

Design Report for Team Sovereign 2020

Mann Shah
Team Captain
Raj Chokshi
Design Head

1. Abstract

This report reviews the methodologies used to design and fabricate the ATV. Various automobile concepts and past year experiences were used in order to design the vehicle. The report also throws light on some improvements made from previous edition. The designing prospects include the ergonomics, reliability, Strength, ruggedness, safety, cost and feasibility.

2. Introduction

The main objective of BAJA competition is to simulate real world engineering design projects and their related challenges by designing and fabricating a single seater electric all terrain vehicle that can be easily transported, safe to use and can be driven on rugged terrains.

This report describes the steps and procedures adopted by Team Sovereign to design, manufacture and test the electric all terrain vehicle that will compete in E-BAJA 2020.

3. Design

The design process is iterative and based upon various engineering principles and manufacturing processes.

The design of the vehicle is based on the rulebook specifications provided by BAJA SAE INDIA. The rules define the basic specifications such as the length of the different members, the driver clearances, the minimum width, length and height of the vehicle. These all parameters are kept in mind while designing of the vehicle in order to make it sustain the rugged terrains and conditions. Multiple iterations were carried out to obtain an optimized design considering various factors such as over designing, cost effectiveness, weight etc.

Alongside it a major concern is given towards the driver safety, comfort while driving the vehicle. It included the designing of the safety belt releasing mechanism for the easy entry and exit of driver.

The designing of vehicle was carried out in Solidworks and analysis was performed in ANSYS, Lotus, Hypermesh.

3.1. Frame Design

The chassis is the main component among all other components and subsystems and provides firm support to them. It also deals with driver safety and therefore has to be prepared to withstand the impacts created in any crash and roll over conditions. A 3-D model of the frame was created using Solidworks keeping the rulebook specifications in mind and was further analyzed using ANSYS 17.1. It comprised of both circular as well as square pipes.

The material used is AISI 4130 for the frame. The material selection was carried out by considering various physical properties such as Yield Strength, Ultimate Tensile Strength, Modulus of elasticity, percentage elongation and chemical properties such as Carbon content, Silicon content etc.

The dimensions of pipes used in frame are as follows:

Primary Pipes:	29.2mm X 1.65mm
Secondary Pipes:	25.4mm X 1mm

Table 1. Material Properties of AISI 4130

Physical Properties	
Density	7.47 g/cc
Mechanical Properties	
Tensile Strength	722 MPa
Yield Strength	761 MPa
Elongation at Break	20.67%
Reduction of Area	58%
Modulus of Elasticity	205 GPa
Bulk Modulus	140 GPa
Poisson's Ratio	0.29
Machinability	70%
Shear Modulus	80 GPa

Table 2. Chemical Properties

Chemical Properties	
Element	**Content (%)**
Carbon	0.29
Silicon	0.23
Manganese	0.495
Phosphorus	0.015
Sulphur	0.004
Chromium	0.910

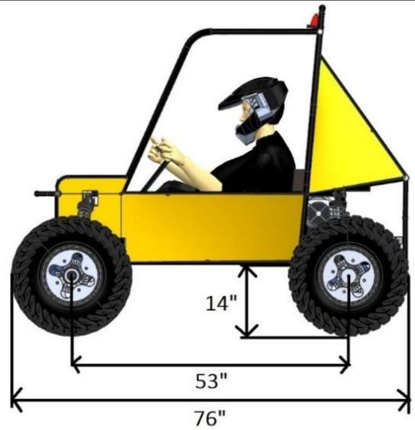


Figure 1. Side View

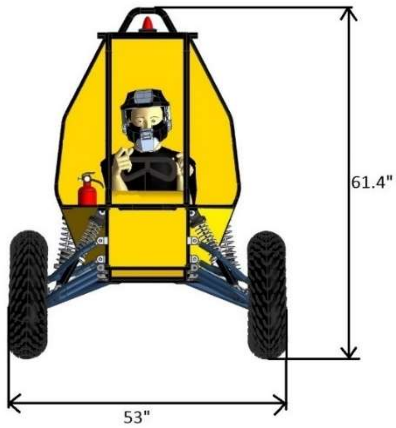


Figure 2. Front View

3.2. Design Optimization

Certain parameters were taken into consideration for the optimization of design.

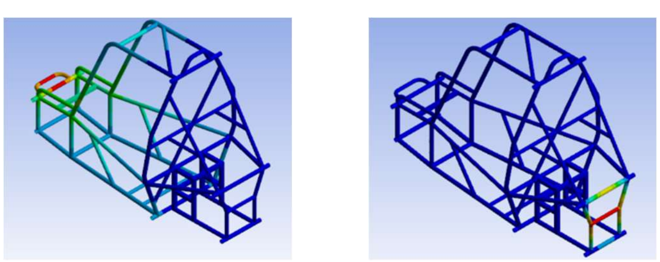
- The Rear Roll Hoop and cradle member of frame are manufactured using welding process instead of bending in order to provide strength as welds have more strength than bends.
- The Overall Size of the vehicle was reduced by 14% as compared to past year. A weight reduction of 22% is achieved this year resulting to weight of 180kg.
- The position of Center of gravity of the vehicle is shifted down from the past year due to changed locations of various components in the vehicle.
- The Motor Mounting points are elevated from the ground level in order to avoid any damage from the obstacles in the track.

3.3. Finite Element Analysis

The most effective method to validate any design is to apply loads on different regions in real time conditions in order to check its durability and strength. All the components were validated using FEA method in ANSYS 17.1 by applying real time loads. A standard procedure was followed while designing the components and analyzing it so that the critical areas can be eliminated and an optimized design can be obtained.

Table 3. Analysis and results

Type of Analysis	Load Applied	Max. Deformation	Von-Mises Stress	FOS
Front Impact	13100 N (5.34G)	0.403 mm	193.28 Nm	2.46
Rear Impact	13100 N (5.34G)	0.351 mm	177.08 Nm	2.67
Side Impact	13100 N (5.34G)	1.004 mm	165.24 Nm	2.88
Roll Over	13100 N (5.34G)	1.579 mm	346.12 Nm	1.37


Front Impact & Rear Impact

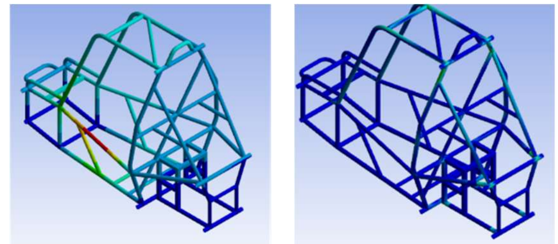
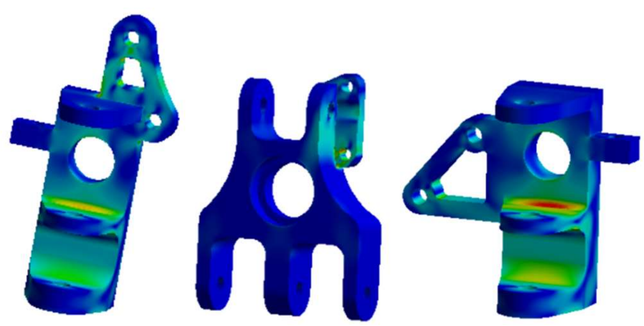

Side Impact & Roll Over
Figure 3. FEA on roll cage considering static forces

Table 4. Analysis and results

Components		Considered Forces	FOS
Hub	Front	Bump Forces	1.48
		Cornering forces	
	Rear	Braking force	3.78
		Bump Forces	
		Cornering forces	
		Braking force	
Knuckles	Front Right	Bump Forces	2.75
		Cornering forces	
		Steering forces	
		Brake forces	
	Front Left	Bump Forces	2.13
		Cornering forces	
		Steering forces	
		Brake forces	
	Rear	Bump Forces	1.81
		Cornering forces	
		Brake forces	
A - Arms		Cornering forces	3.79
		Bump Forces	
		Spring Forces	
H - Arms		Bump Forces	4.47
		Spring Forces	
Hinges		Bump Forces	1.50
		Cornering forces	


Front Left, Rear & Front Right Knuckles

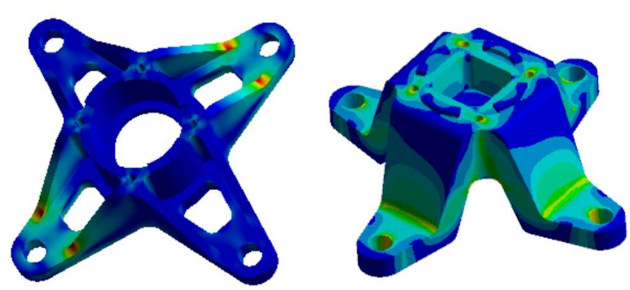
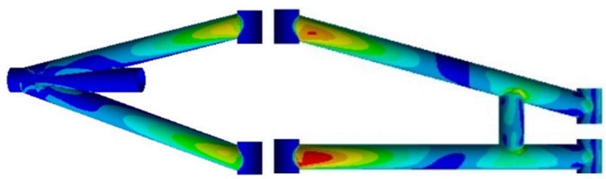

Front & Rear Hubs

Figure 4. Static analysis of wheel assembly sub components



Front (A-arm) & Rear (H-arm)
Figure 5. FEA stress analysis of wishbones

3.4. Suspension

Objective:

The objective of a suspension system is to isolate the chassis of the vehicle as much as possible from the dynamic forces provided by the irregularities on the track. We achieved this objective by designing the optimal geometry which sufficed the desired characteristics of the suspension system.

Calculated camber and caster angles were provided. Rigorous planning and research were done in order to successfully complete dynamic analysis of the system.

Design:

The design of the suspension geometry was done considering the minimum change in camber, caster and toe angles while bump, steering and roll analysis and also by taking care of the parameters like anti dive, motion ratio, ground clearance (14”) etc. We started designing our suspension geometry on lotus shark suspension analysis and by obtaining optimum suspension design factors.

We then continued designing of suspension geometry on Altair HyperWorks, where we added actual shape of the model of suspension components and iterated upon it. We performed analysis such as Kinematic & Compliance (K&C) analysis, bump analysis & roll analysis and obtained several graphs validating our calculations such as vertical force, ride rate, camber change, etc. We further used MATLAB for performing spring calculations after which we designed our suspension struts.

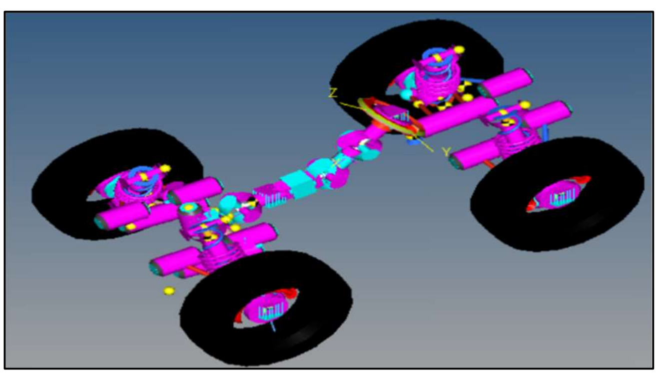


Figure 6. Suspension geometry in Altair HyperWorks

Front suspension

We have used short long double A-arms wishbones as our front suspension system because it gives high stability during rough impacts from the track and is easy to install with minimum camber, caster, toe change during dynamic application.

The front upper wishbones are made of 25.4mm dia. and the lower wishbone is made of 29.2mm dia. AISI 4130 pipes.

We have installed a static negative camber of 2 degrees, which gives better stability of the vehicle during cornering. The maximum camber change during bump encounter is -3.5 to -0.5 degrees. A positive caster of 11.26 degrees and kingpin of 8.96 degrees has been given to improve line stability and greater aligning torque. A static 30% of Anti-Dive mechanism has been installed to further reduce the magnitude of forces acting on suspension components and on the vehicle chassis during braking and when the vehicle comes over any obstacle. The customized suspension strut is installed on the lower wishbone with a motion ratio of 0.6 for better distribution of forces.

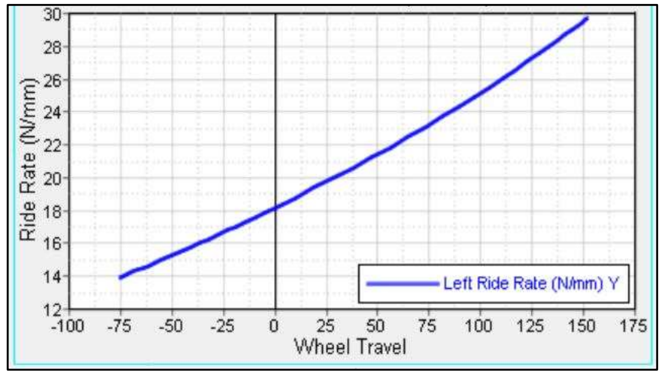


Figure 7. Ride rate vs wheel travel

Rear Suspension

The rear suspension comprises of struts with high spring stiffness due to 63 % of the weight distributed on the rear of the vehicle and the motion ratio fixed is 0.65. Considering ease pf installation, we chose H-arms to form the rear geometry. H-arms do not allow toe change, camber change and also does not possess any caster and hence cannot be used in front suspension. Due to these reasons rear geometry is simpler than front geometry. Two hinge points are provided at the knuckle to prevent wobbling of tires. The rear wishbones are made of the same material as that of the front ones.

Table 5. Specifications of Suspension

	Front	**Rear**
Suspension Type	Double A-arms	H-Arms
Mass Distribution	96 Kg	224 Kg
Sprung Mass	185 kg	
Un-sprung Mass	65 kg	
Camber Angle	-2°	0°
Caster Angle	11.26°	0°
KPI	6.64°	0°
Scrub Radius	41.17 mm	-
Motion Ratio	0.6	0.65
Ground Clearance	14”	
Roll Centre Height	6.95”	7.24”
Shocks	Spring Damper	
Roll Stiffness (N/mm)	3.095	9.578
Suspension Travel	Jounce = 6”, Rebound = 3”	
Spring Rate	22	30
CG Height	21”	
Desired Natural Frequency	1.55 Hz	1.76 Hz
Damping Ratio	0.7	0.7
Damping Value@0.3m/S	43.18	65.80

Figure 8. Angles vs wheel travel

Shock Absorbers

The spring-shock system was designed and customized by considering the dynamic load factor as 2.5 on the suspension system. Coiled spring was used due to its profound advantages such as its light weight, high sustainability, economic benefits, better technical characteristics and that it is replaceable.

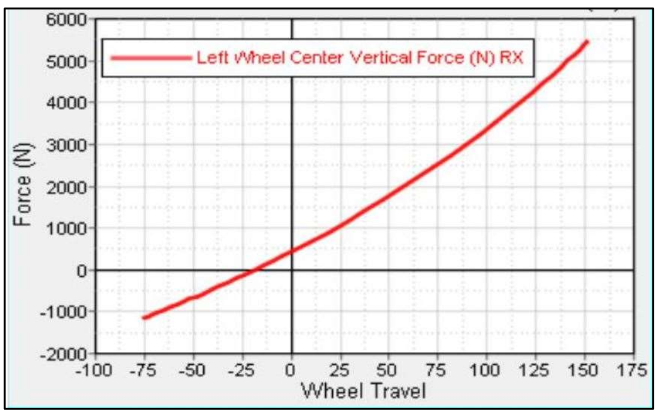


Figure 9. Vertical force vs wheel travel

3.5. Brakes

It is used to inhibit motion by absorbing energy from a moving system and either make the vehicle slow or stop.

Objective:

The main objective is to obtain better braking force than previous year, to make sure simultaneously locking of all four wheels and also to stop the vehicle within a maximum distance of 15 feet as per E-BAJA rulebook 2020.

Design:

Our hydraulic braking system consists of a single pedal in line with two separate single master cylinders. We have used front-rear split braking system. Force obtained from the pedal is biased according to the necessity with the help of balance bar. The master cylinders generate pressure on the brake fluid which is then passed through the metallic (non- flexible) brake lines to the brake caliper. This pressure moves pistons of the brake caliper and thus it clamp brake pads to the disc. The dual master cylinder system is used so that the bias ratio can be adjusted as necessary.

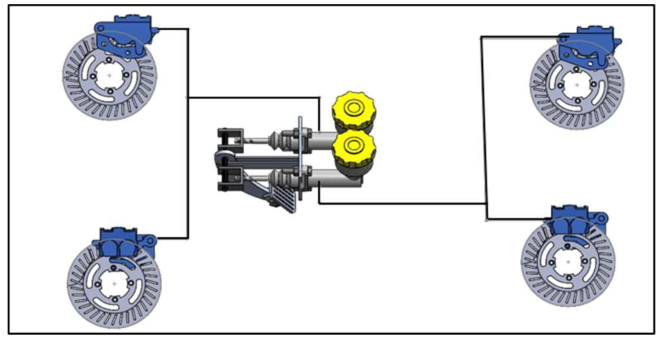


Figure 10. Brake System

- The pedal length is reduced than the previous year so as to decrease the pedal travel and also to reduce the spongy feeling while braking.
- The brake lines used are metallic to sustain high pressure.
- Calipers are placed such that it provides more pad area on the brake disc and provide more braking.
- Brake discs used are customized according to the calculations and its radius have been increased to 82.5 mm to obtain more braking torque.
- Thermal and stress analysis have been done on the disc rotor using the ANSYS software and the maximum temperature recorded was 241.7° Celsius and the deformation to be 0.02mm
- Outboard braking system is used as it is easy to work & provides better cooling.

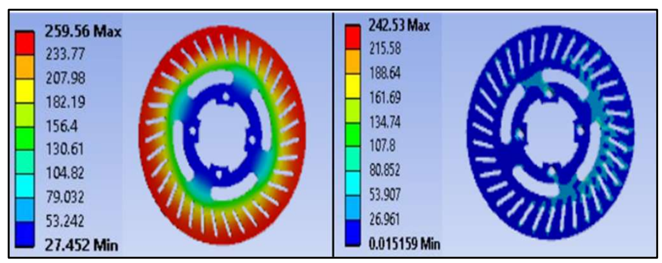


Figure 11. Thermal & Stress analysis of Disc Rotor

Table 6. Specifications of brakes:

Brake Type	Disc
Pedal Force	400N
Master Cylinder (Bore Diameter x Stroke)	19.05 x 25 mm
Brake Hose	Flexible
Brake Caliper Used	SUZUKI GS150R
Caliper Piston Area	791.73mm^2
Pedal Ratio	8: 1
Bias Ratio	62: 38
Brake Fluid	DOT 3
Effective Radius	75 mm
Disc Material	SS304
Front Torque Required (single tire)	244.65 Nm
Rear Torque Required (single tire)	149.35 Nm
Front Torque Generated (single tire)	335.30 Nm
Rear Torque Generated (single tire)	204.69 Nm
Stopping Distance	8.98 m
Friction Coefficient of road	0.7
Friction Coefficient between Brake Pad and Brake Disc	0.45
Dynamic Load Transfer	771.44 N
Static Roll Radius (Tyre)	11.5”

3.6. Steering

Objective:

The steering plays an important role in vehicle dynamics. The main objective of the subsystem is to provide good maneuverability with smooth recovery from turns. Reduction in bump steer and roll steer improves stability of the vehicle. The subsystem is designed to get proper feedback of aligning torque to the steering wheel. The subsystem was designed such that the steering effort for the driver at steering wheel would be minimal.

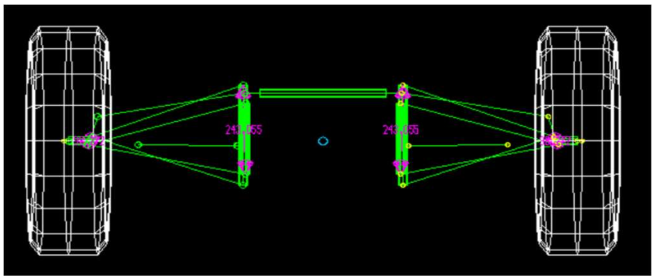


Figure 12. Steering geometry in LOTUS Shark

Design:

The steering subsystem consist of steering geometry, steering gears, tie rod, steering link and steering wheel. We have worked on all the above mention components to optimize the steering geometry as per our aimed objectives. To make the system convenient, we have used OEM rack and pinion steering system with 17” rack length, which provides us a total rack travel of 64mm. The steering ratio that we have achieved is 49 mm/rev. In order to reduce slip angle and to achieve sharp turns, we preferred ackerman steering geometry. For this year’s design we’ve achieved 70 - 80% ackerman and a turning radius of 1.7m. The graph of ackerman percentage vs rack travel is shown in fig 14. The angles set for inner and outer tires are 48° and 27° respectively.

We have optimized caster angle by varying caster trail keeping in mind the minimum steering effort for the driver. Angle and length of steering link are the important aspect that affects steering effort, ackerman percentage, and turning radius. Hence it is necessary to have proper length and angle in order to achieve optimum outputs. We’ve used CNC machined steering link in our subsystem. Tie rod is the vital part of the steering system as it affects the bump steer and roll steer characteristics. Hence many optimizations are performed in its position and length. The material chosen for tie rods is AISI 4130 with the length of 12.1” with an outer diameter of 25.4mm.

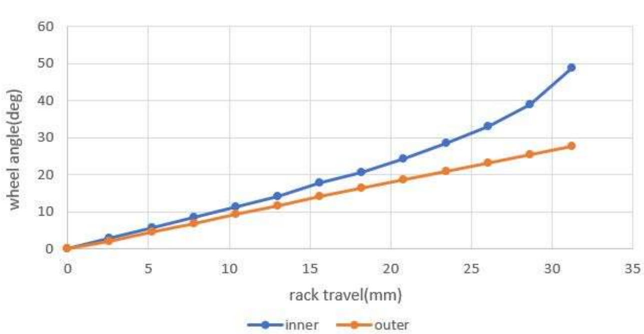


Figure 13. Wheel angle vs rack travel

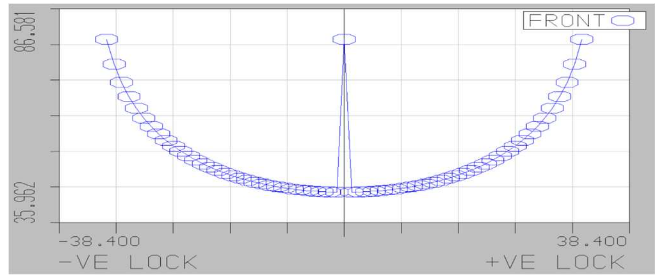


Figure 14. Ackerman percentage vs rack travel

Table 7. Specifications for steering

Steering System	Oversteer
Steering Geometry	Ackerman: 70 - 80 %
Inner/Outer Wheel Angle	48° / 27°(deg)
Turning Radius	1.7m
Steering Gear	Rack and Pinion
End to End Travel	64 mm
Turns Lock to Lock	1.3 turns
Steering Ratio	49 mm/revolution
Column Type	Single column
Drive Type	CHD
IBJ/OBJ Centre Distance	235 /569 mm
Tie Rod Length	12,1 in
Steering Wheel Diameter	250 mm
Steering Wheel Torque	2.4 - 4.1 Nm

3.7. Transmission

Objective:

The main objective of providing a transmission is supplying the torque generated at the motor to the wheels with increment in the torque through a gearbox. We have a constant gearbox that gives us a 11:1 gear ratio.

Motor:

The motor in use is a BLDC Motor, provided by the BAJA SAEINDIA, manufactured by Compage Automation Systems. It gives out a constant torque of 10.3 Nm at 94 A and a peak torque of 38 Nm at start. The rated speed of the motor is 4500 rpm. We have directly coupled motor with the customized gearbox via shaft with external splines on it.

Battery:

This year Lithium-ion NMC Battery is used having high specific charge density and cycle life compared to Lithium-ion LFP Battery. Its specifications are - 48V/110Ah with a nominal voltage of 50V, cut-off voltage 46V and fault output/potential free provided by Jascon energy. This battery supplies current to the motor controller that in turn is connected to the motor. The additional benefit of using this battery is its less space requirement and light weight.

Gearbox:

This year we have customized a lighter and compact gearbox as per our requirements with a fixed gear ratio of 11:1 meshed by compound gear system having 3 gears. Power from the gearbox is transmitted to the wheels through axle. This axle is connected with gearbox via CV joints having six balls bearing to ensure capability of high angle motion of the wheels which makes it compatible for all terrains.

Speed, Acceleration and Gradeability Calculations:

1. Speed calculation:

$$Velocity_{max} = \frac{2\pi R \times N}{60 \times Gear\ Ratio}$$

Velocity of the vehicle according to the formula came out to be 12.5 m/s which is equal to 45.02 kmph.

The values for the above variables:
N, speed of the motor = 4500 rpm
R, radius of the wheel = 0.2921 m
Gear Ratio = 11:1

2. Acceleration calculation:

Air Resistance, $R_a = K_a \times A \times v^2$,

where,
K_a, air drag coefficient, = 0.0609
A, projected area = 0.88 m^2
V, velocity (in kmph) = 45.02 km/hr

Thus, R_a = 109.054 N

Rolling Resistance, $R_r = C \times m \times g$

where,
C, coefficient of rolling resistance, = 0.05
m, mass of the vehicle (with driver), = 250 kg
g, acceleration due to gravity = 9.81 m/s^2

Thus, R_r = 122.625 N

Road Resistance, $R = R_a + R_r$
= 114.792 N

$$Traction = \frac{T \times Gear\ Ratio \times T.E.}{r}$$

where,
T, pear torque = 38 N-m
Gear Ratio = 11:1
T.E., transmission efficiency = 85%

Thus, Traction, F = 1,216.36 N

$$Acceleration = \frac{F - R}{M}$$

Thus, Acceleration = 3.9406 m/s^2

3. Gradeability calculation:

R_r = 122.625 N

Gradient force, $R_g = m\,g \sin\varphi$

Ø = 20° (selected on the basis of the maximum angle the vehicle will be expected to climb)

Thus, R_g = 838.80 N

Accelerating force = ma = 1,101.5 N

Total tractive effort = $R_r + R_g$ + ma
= 2062.92 N

$$Gradeability = (\frac{TTE}{mg} - C) \times 100$$

Thus, gradeability = 79.18%

Table 8. Specifications of Power Train

Rated Motor RPM	4500 rpm
Peak Torque (@200A)	38 N-m
Rated Torque (@94A)	10.3 N-m
Gear Ratio	11:1

3.8. *Electrical Systems*

The electrical system of the car consists of a BLDC motor, 48 V Li-ion battery, Motor controller, brake light, reverse light, reverse alarm and tractive system active light (TSAL). Apart from that, it consists of sensors for the measurement of battery's current, voltage as well as the state of charge. Also, a display will be mounted in the cockpit for the driver to know the condition of battery as well as the distance travelled by the vehicle and the percentage of battery capacity.

The connections of the motor controller with the battery and the motor as provided by BAJA SAEINDIA have been taken into consideration.

Circuits have been developed for the easy mounting and connection of the auxiliaries like brake light and the reverse light. The rule for starting the motor while pressing the brake pedal and keying has been implemented. The circuit for the same has been developed. Further connections have been provided for auxiliaries other than the aforementioned.

The ATV as a whole has been properly insulated to ensure the safety of the driver as well as the vehicle itself. The insulation provided is strictly in accordance with the standards specified by the rule book.

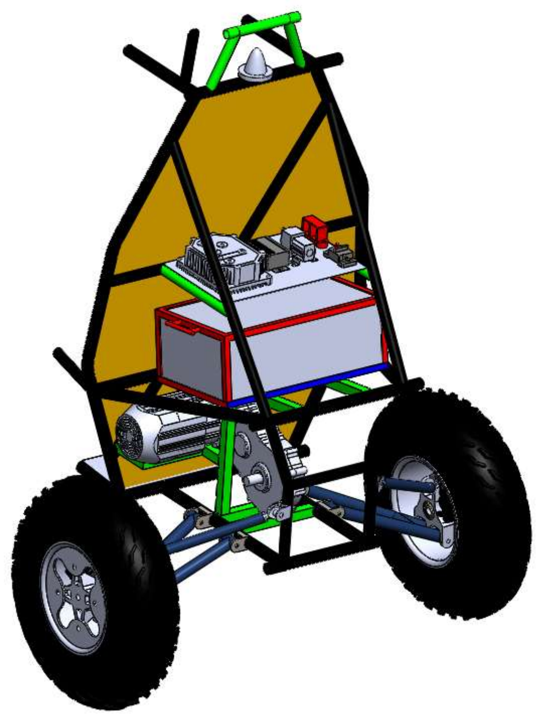


Figure 14. Position of electrical equipment on the roll cage

4. Conclusion

As per the required rules and standards declared by the BAJA committee, we have designed and manufactured a fully functional and robust electric ATV delivering the required performance.

5. References

Books:

1. Thomas D. Gillespie, “Fundamentals of Vehicle Dynamics”, Society of Automotive Engineers, July 976.
2. Dr. Kirplal Singh, “Automobiles Engineering Vol-1”, Standard Publishers, Seventh Edition, 1997, New Delhi.
3. V. B. Bhandari, “Design of Machine Elements”, McGraw Hill Education, Third Edition, 2010, New Delhi, ISBN-0-07-068179-1
4. Fred Puhn, “Brake Handbook”. HP Books, ISBN-0- 89586-232-8
5. David Hutton, “Fundamentals of Finite Element Analysis”, Tata McGraw Hill Education Private Limited, 2005, ISBN-0-07-239536-2
6. Race car vehicle dynamics by William F. Milliken and Douglas L. Milliken
7. BRAKE HANDBOOK by FRED PUHN. ISBN 0-89586-232-8

Websites:

1. www.bajasaeindia.org
2. www.bajasaeindiaforum.com
3. www.nptel.ac.in
4. www.allelectricals.com
5. www.bajatutor.net

6. Contact Information

Mann Shah
6th Semester, Mechanical Engineering,
Pandit Deendyal Petroleum University
Contact No.: +91 9106712831
Email ID.: mann.smc17@sot.pdpu.ac.in

Raj Chokshi
6th Semester, Mechanical Engineering,
Pandit Deendyal Petroleum University
Contact No.: +91 8160423357
Email ID.: raj.cmc17@sot.pdpu.ac.in

7. Acknowledgments

We would like to extend our regard to the administration of Pandit Deendayal Petroleum University for providing us with the workshop and electrical machine lab facilities and much needed support. The report would not have been possible without the knowledgeable inputs of the following team members and faculty advisors of Team Sovereign.

- **Faculty Advisors**:
  Mr. Rahul Deharkar,
  Dr. Alok Jain

- **Design and Analysis Department:**
  Raj, Ishaan, Neel, Aarya, Dhanraj

- **Power Train Department:**
  Ankit, Arun, Vaikunth, Vedant, Karthik

- **Brakes Department:**
  Mohit, Rudra

- **Vehicle Dynamics Department:**
  Mann Shah, Pritesh, Nikhil, Viraj, Yug, Taranjyot, Mann Patel, Swarup

- **Electrical and Electronics Department:**
  Parth, Saptarishi, Sarvesh, Nidhi, Raghav