
CENTER FOR TRANSPORTATION INFRASTRUCTURE AND SAFETY

# Missouri S&T Formula Electric Racing

by

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A National University Transportation Center
at Missouri University of Science and Technology

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**Technical Report Documentation Page**

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| 16. Abstract<br>The Formula Electric racing team will promote Missouri S&T's engineering excellence by successfully competing against other top engineering universities in the US and around the world. Students on the team will have the opportunity to reinforce their classroom education through practical application of modern, and future, automotive technologies. | | | |
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# NUTC Project #00042849 Final Report

## Missouri S&T Formula Electric Racing

**PI: Ryan S. Hutcheson, PhD**
**5/15/2015**

## Contents

## Overview

A proposal from Missouri S&T's Formula Electric racing team requesting $37,362 with $24,825 in direct costs with a one-for-one match was submitted to NUTC in May 2013. Of this request, $22,417 with $14,895 in direct costs were awarded. The team, which is tasked with designing and building an electrically powered formula-style (open-cockpit, open-wheel) race car for the Formula SAE collegiate design competition, has successfully raised matching funds for the grant and completed expenditures of the awarded and matching funds and fulfilled the obligations included in the project proposal. The remainder of this report details the matching fund sources, sponsor and matching expenditures and a description of the fulfillment of required activities and deliverables.

## Matching Fund Sources

Matching funds for the grant came primarily from team fundraising activities. These activities include a summer camp for high school students, proposals submitted to professional organizations and donations from individuals and businesses. Additional matching funding was provided by the Student Design and Experiential Learning Center (SDELC) at Missouri S&T. A detailed listing of matching funds is included in Table 1.

**Table 1 - Matching Fund Sources**

| Source | Type | Amount |
|---|---|---|
| Formula Electric Camp, 2013 | Fundraiser | $15,345.59 |
| AMAE | Prof. org. donation | $5000 (2013), $4000 (2014) |
| Gregory Construction | Business donation | $250 |
| SDELC | University | Balance of match |

## Final Project Budget

An overview of the final budget for the project including sponsor and matching funds is included in Table 2. The sponsor funds were used primarily for expendables (e.g. tires), measurement tools for the shop (e.g. vehicle corner scales), driver uniforms, competition expenses and other supplies and consumables for the shop. Matching funds were used primarily for vehicle construction related expenses including components and raw materials. Total funds from the sponsor and match were $44,896. For the 2014 competition year, the team will spend just under $60,000. The balance of the funds spent by the team until competition in June 2014 will come from sponsorship from individuals and businesses and from the SDELC at Missouri S&T.

Table 2 - Final Project Budget

| Matching Budget<br>PI: Ryan S. Hutcheson | 1/1/2013 - 5/15/2014 | | | |
|---|---|---|---|---|
| | Sponsor | Match | | |
| | Sponsor | Cash Match | In-Kind Match | Sum of Match |
| Department Operating Expenses | | | | |
| Travel | $0 | $0 | $0 | $0 |
| Supplies (730000) | $1,099 | $6,394 | $0 | $6,394 |
| Postage/Shipping & Delivery (723000) | $671 | $0 | $0 | $0 |
| Uniforms (730800) | $2,632 | $174 | $0 | $174 |
| Expendables (731900) | $4,236 | $14,512 | $0 | $14,512 |
| Dues/memberships (738000) | $2,100 | $0 | $0 | $0 |
| Laboratory – Non Capital (740500) | $4,157 | $0 | $0 | $0 |
| Business mtg exp-food catering (721700) | $0 | $280 | $0 | $280 |
| Shop supplies (731600) | $0 | $1,038 | $0 | $1,038 |
| Computing expense (739000) | $0 | $81 | $0 | $81 |
| Total Departmental Operating | $14,895 | $22,479 | $0 | $22,479 |
| Total Expenses Subject to F&A Charges | $14,895 | $0 | $0 | $22,479 |
| Total F/A Cost 50.5% | $7,522 | $0 | $0 | $0 |
| Budget Total | $22,417 | $0 | $0 | $22,479 |

## Project deliverables

The primary deliverable for this project was the 2014 Formula SAE electric competition vehicle (Figure 1). This vehicle will be competing in the 2014 Formula SAE Electric competition in Lincoln, Nebraska during the week of June 18, 2014. The secondary deliverable was the design report documenting the production of the vehicle. This report is included in the following section.

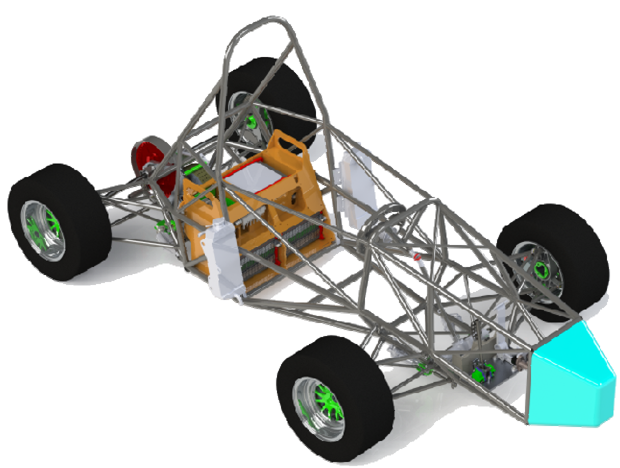

Figure 1 - Solid Model Rendering of Vehicle

## Technical report

2014 Missouri University FSAE Electric Design Report, Ethan Winberg – Chief Engineer

### Introduction

For the past two years Missouri S&T Formula Electric has worked on designing and manufacturing a car to compete in the Formula Electric SAE design competition. Using their time and resources, a car was designed to minimize weight while achieving acceleration within traction limits. Having been established and developed independently and with no prior affiliations to the combustion team on campus, a vast amount of effort was placed into designing an original drivetrain to take advantage of a fully electrical system as opposed to adopting one similar to those used by combustion teams. This additional effort is justified, in that the drivetrain is required to perform reliably and viewed as one of the most critical design components. The extra year of refinements during the car's development period will hopefully allow several parts of the design to carry through to successive cars and thus diverting future focus to redesigning systems that do not perform satisfactorily.  Analysis techniques used during the design process include:

- SolidWorks Finite Element Analysis, Fluid, and Thermal Simulation
- Team Developed Powertrain Simulation
- Electric Motor Dynamometer
- MSC Adams/Car

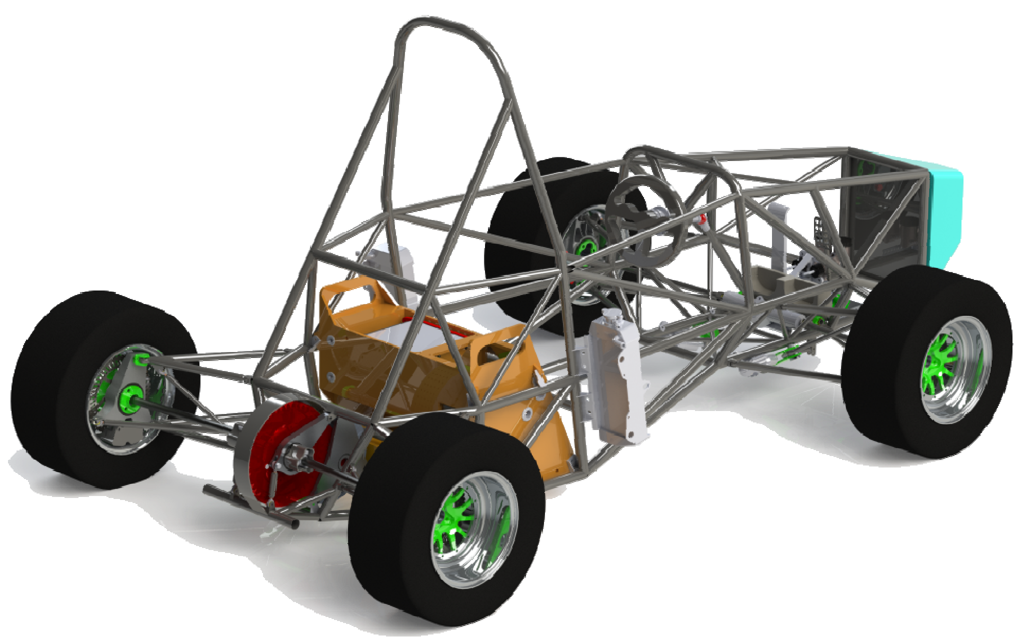

Figure 2 - Rear View of Complete Solid Model

### Motor and Controller

Initially, driving the rear wheels using two separate motors was considered, as it would remove the need for a differential and allow for a torque vectoring capability. However, it would lend to further complications to an already complicated electrical design. For this reason, a more traditional setup with a single motor connected to a differential was chosen. The search began with motors in the 60kW to 100kW range and was narrowed to the motors typically used in electric vehicles. Unfortunately the majority of AC induction motors were air-cooled and anything near the target 85kW range weighed greater than 70kg and had a maximum efficiency of 89 percent. A motor with this weight and low efficiency did not meet the design constraints of high performance and low weight. During the search, a company named Enstroj was discovered, responsible for developing the EMRAX 228, a 100kW axial flux motor weighing a mere 12.3kg and intended for use in an electrically assisted glider. The motor operates at a maximum output speed of 5000rpm, as typically it would directly drive a propeller. This allowed the drivetrain to achieve speed reduction using a simple sprocket and chain. Enstroj also conveniently repackages a Unitek Motor Controller reducing the size and the weight to 4.3kg. This motor controller meets the specifications for voltage and current and both the motor and controller can be water-cooled on a single circuit. After some discussion and experimentation the conclusion was made that the motor controller would be more sensitive to heat than the motor and as such it would be first on the cooling circuit. Overall, this motor and controller were found to have the best power to weight ratio and were reasonably priced compared to other alternatives.

### Power Simulations

A simulation was developed to help predict the power requirements of the accumulator as adding unnecessary capacity would mean significant increases in cost and weight. The simulation utilizes telemetry data from a high performing combustion Formula SAE team, including the speed vs. distance achieved throughout an endurance lap in a previous competition. Accounting for the efficiency of the motor and controller, estimated drag, and weight of car, the simulation attempts to match the acceleration of the combustion car. By setting an upper power limit it was revealed through iterative runs how manipulating this variable affects power usage and lap times. A peak power of 50kW provided a good balance of minimizing lap times and maximizing accumulator capacity. By limiting power consumption to 50kW the car could ideally complete an endurance run with a 5kWh accumulator. This is consistent with the designs of several Formula Electric SAE teams in the Formula Student Germany competition who use accumulators ranging from 4.3kWh to 7kWh.

### Accumulator Cells

In order to select the accumulator cells, a list of potential cells was compiled on a spreadsheet that included all relevant data: weight, capacity, discharge rate, cost, nominal voltage, and maximum voltage. The motor and controller have the capability to operate on 360V, however the rules state the accumulator voltage must not exceed 300V. Using the data, the spreadsheet constructs an ideal cell configuration to meet peak power capacity and voltage requirements. A123's AMP20 (Lifepo4) and EIG's CO20 (Li-NMC) produced the overall lightest configurations and were compared in more depth. The A123 cells had higher discharge rates and safer chemistries, whereas EIG cells were lighter weight, smaller in size, required fewer cells, cheaper, and had a lower voltage drop. The EIG configuration weighed 30.8kg, 9.4kg lighter than the A123 configuration, and consumed roughly 30 percent less space. The 68kW peak discharge of the A123's would yield quicker acceleration, but it was believed the 51kW peak discharge of EIG and reduced weight would perform better in the endurance event worth four times the points of the acceleration one. Overall the EIG cells came out as the best, currently available option.

## Accumulator Cooling

To prevent the AMS from throttling performance when cell temperatures approach 55ºC, an effective cooling system is critical in retaining performance during hot weather. Each cell individually generates an average of 24W waste heat discharge due to internal resistances and regeneration on the track. Initially, an air-cooling setup was considered due to its low weight and perceived simplicity. However, with an ambient temperature of 32ºC, a minimum of 74 liters per second of air would be required to cool the batteries. Because this task wasn't considered feasible, it was deemed necessary to implement a water-cooling setup, as a flow rate of 0.019 liters per second would sufficiently cool the pack. In addition, the cooling system can easily be scaled as necessary, utilizing additional pumps and radiators. Each cell is wrapped with an aluminum plate intended to draw heat toward a water block on the cell's edge. The water blocks also serve as structural members holding the modules together. Because temperature affects the cells discharge rate, an unevenly cooled accumulator effectively reduces capacity in addition to the fact that the entire accumulator shuts down upon depleting a single cell. To ensure even cooling, thermal fluid analysis in SolidWorks was used in designing the water blocks.

## Wheels

The decision to use 10in diameter wheels over 13in was critical in achieving a lightweight competitive car, lacking substantial down force. The smaller wheels effectively remove 6.5kg of rotational mass from the tires and factoring in the moment of inertia, this is equivalent to removing roughly 13kg of static mass from the car, greatly improving acceleration. Even though the 13in tires are typically capable of higher lateral G's, the lightweight design and limited aero-surfaces cause the larger tires to take longer to reach optimal temperatures. The smaller tires are also expected to improve autocross and endurance times and reduce the necessary gear reduction. This decreased gear reduction permits the use of smaller sprockets and makes packaging the differential along the wheel axels easier. Overall the differential, halfshafts, tripods, and their respective housings experience reduced stresses, increasing reliability. The smaller rims added complexity to the suspension geometry as close attention had to be paid to the control arms and the tie rod clearances with the rim. As a result, the control arms were made as wide as possible to reduce the forces transferred into the frame mounts and retain stiffness while braking. A sophisticated parametric kinematic model of the corners was created in SolidWorks to ensure interference would be a nonissue throughout the entire range of suspension and steering angles.

## Suspension

The front suspension was initially designed using SolidWorks sketches, while the kingpin inclination and caster angle were derived from a previous Formula SAE car. In addition, the scrub radius was increased 0.5in in order to prevent the upright from interfering with the rim and free space necessary for the control arms. Although this increases the steering feedback, it is not expected to significantly impact the driving experience. These SolidWorks sketches helped in designing a geometry having logical camber control and a roll center slightly above the ground that minimizes movement with respect to the chassis. The rear suspension control arms are angled backward and as such the camber rates and roll center locations couldn't be modeled with a simple sketch. Instead, MSC Adams/Car was employed to design a rear suspension having similar characteristics to the front.

## Uprights and Hubs

A machined aluminum manufacturing process for the uprights was chosen over the traditional welded steel method as the former holds higher dimensional accuracy and retains more uniform material properties. In addition, a welded upright would also require a manufactured jig, post machining, and heat treatment. Both the front and rear uprights are based on a rotating axle design, with the hub, axle, and tripod housing being one solid aluminum part, thus reducing the number of components in the assembly while saving weight. The hubs and centers are identical front and back, which streamlines the manufacturing process and in the event that the tripods become loose the front and back hubs can easily be swapped. Although the bearing surface of the tripods is aluminum, the halfshaft is angled only a few degrees offset from the axis of the hub's rotation, so limited wear is to be expected. The low car weight and lack of down force permits the use of angular contact bearings capable of withstanding specified lateral loads as opposed to tapered roller bearings that further complicate the design, require preloading, and weigh more. The front rotating axle upright design provides an opportunity to position the brake rotor on the inboard side of the upright; however, in this case the brake rotor would need to be 150mm in diameter to fit between the control arms. In addition, this configuration would make it difficult to get the calipers close enough to the center axis. Even then they would require a large hydraulic advantage to lock the wheels in place. To avoid these issues a 195mm rotor was fitted on the outer side of the upright, and the wheel center had to bow out to provide space for the caliper. For the reasons listed above the extra machining done on the centers was well worth it. Lightweight, narrow calipers were chosen during the selection process to reduce the necessary center offset. The individual weight of the front and rear upright assemblies is roughly 2kg, meeting the light weight design considerations.

### Brake Rotors

Due to the upright design a brake hat was unnecessary, as it would not improve the brake caliper clearances and the lack of distance between the brake surface and hub would cause the weight reduction to be minimal. These facts lead to a single piece rotor design because of its simplicity in design and manufacturing. Steel was used instead of cast iron as there were no major performance benefits that outweighed the additional cost. To compensate for the absence of floating rotors the front calipers are floating and the rear calipers have opposing pistons. A large concern in designing rotors is thermal expansion, which causes a stress cracks. To simulate realistic thermal stresses, a temperature distribution was modeled, with the brake surface elevated to 1000ºC, the thermal limit of the BP-10 brake pad compound, and the central mount elevated to 150ºC. The temperature distribution was then included into the overall stress simulation.

### Frame

To simplify construction, a steel space frame was chosen as it facilitates easy repairs and modifications. Compared to composite monocoque frames, a steel space frame does not require the use of molds, and as such makes more sense for small production runs. Several variations of the frame ran through extensive simulations to determine torsional rigidity and strength to weight ratio. Locating the accumulator in side pods worked well for weight distribution and ease of access, but the additional structure required to support the accumulator added an additional 4.5kg compared to frames of similar stiffness with the accumulator positioned behind the driver firewall. The original target for torsional rigidity was 2170N-m per degree, but increasing to 2740N-m per degree only required 1kg of additional weight. A 26 percent increase in stiffness was well worth 3 percent increase in weight. In addition, careful attention was paid to placing frame nodes on or near the control arm mounts.

# Appendix A – Vehicle Schematics

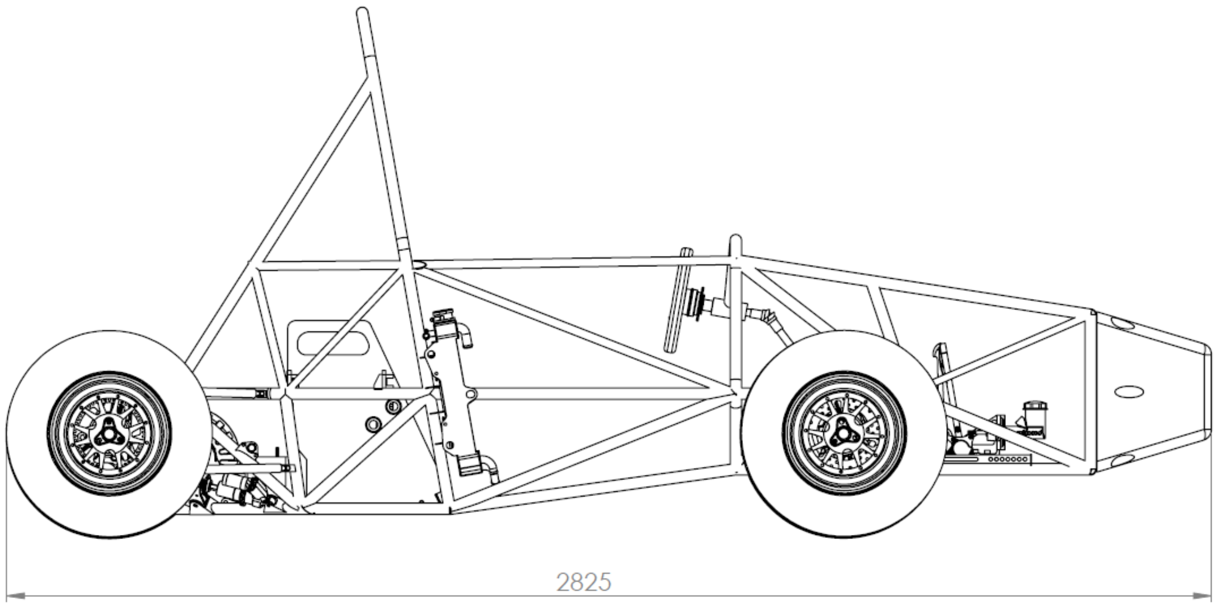


Figure 3 - Side View

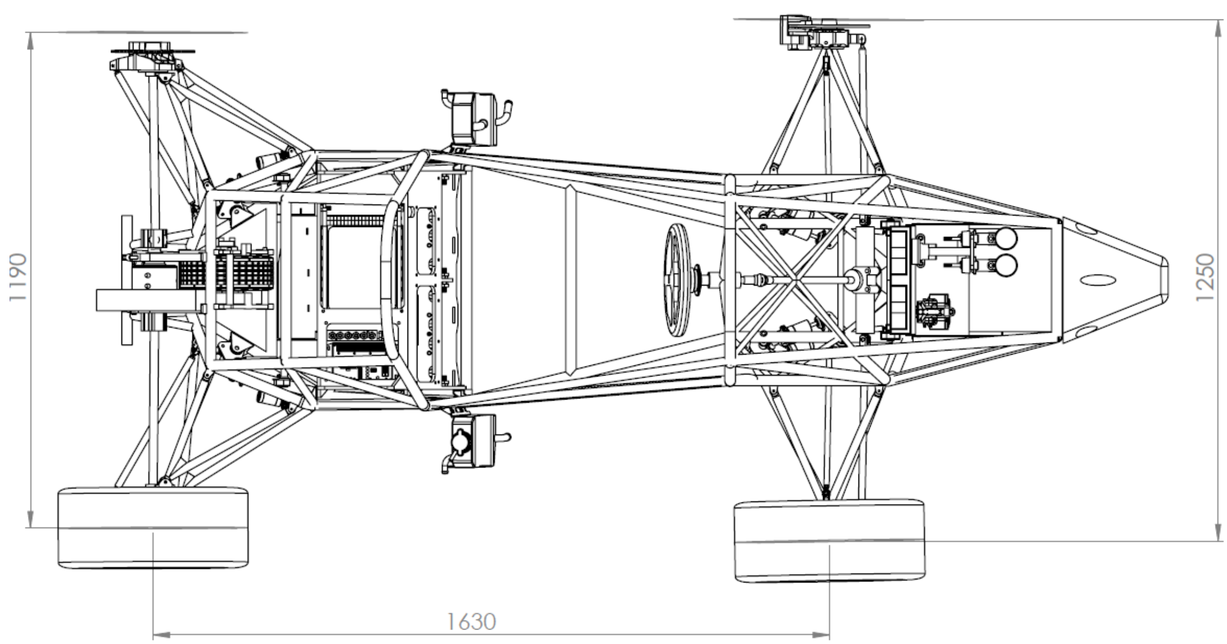


Figure 4 - Top View

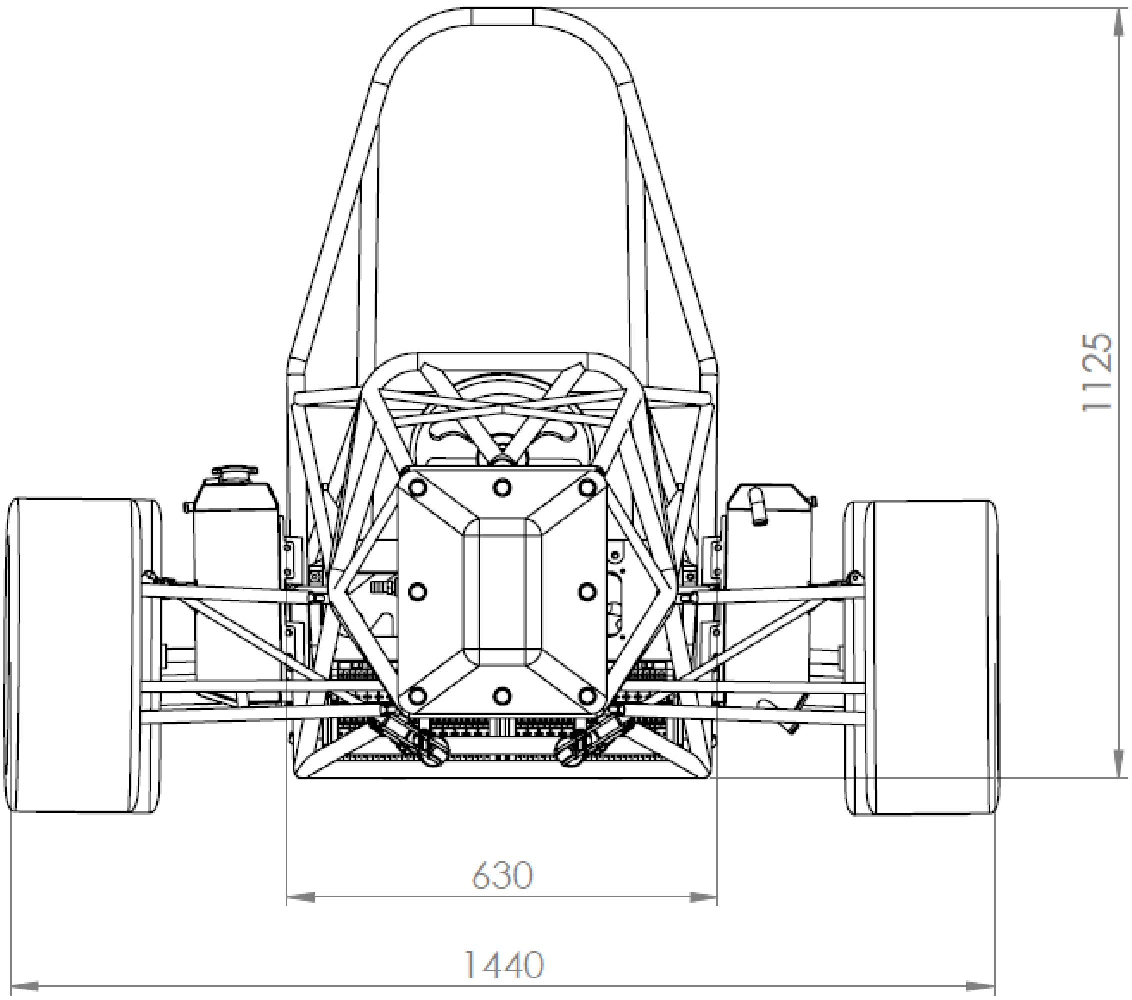


Figure 5 - Front View

# Appendix B – Vehicle Specification Sheet

| Car No. | 219 |
|---|---|
| School | Missouri University Science and Technology |

| Dimensions | Front | Rear |
|---|---|---|
| Overall Length, Width, Height | 2825 mm long, 1440 mm wide, 1175mm high | |
| Wheelbase | 1630 mm | |
| Track Width | 1250 mm | 1190 mm |
| Mass with 68kg driver seated | 117.7 kg | 132.3 kg |

| Suspension Parameters | Front | Rear |
|---|---|---|
| Suspension Type | Double unequal length A-Arm, Pull rod Actuated spring and damper, adjustable sway bar | Double unequal length A-Arm, Pull rod Actuated spring and damper, adjustable sway bar |
| Tire Size, Compound and Make | 18x6-10 R25A Hoosier | 18x6-10 R25A Hoosier |
| Wheels (width, construction) | 6" width, zero offset, 6061 T6, stamped | 6" width, -1.0" offset, 6061 T6, stamped |
| Center of Gravity Design Height | 226 mm | |
| Suspension design travel | 32 mm jounce/ 32 mm rebound | 27 mm jounce/ 27 mm rebound |
| Wheel rate (chassis to wheel center) | 24.27 N/mm | 35.09 N/mm |
| Motion ratio / type | .89/ linear | 1.07 / linear |
| Static camber and adjustment method | -1.0 deg, adjustable via shim plates on upright | -1.0 deg, adjustable via shim plates on upright |
| Front Caster and Kinematic Trail | 4.5 degrees, non-adjustable, 18 mm trail | |
| Front Kingpin Axis Inclination and Offset | 1 degrees non-adjustable, 47 mm offset | |
| Static Ackermann and adjustment method | -20% non-adjustable | |
| Anti dive / Anti Squat | 0 | 0 |
| Steer location, Gear ratio, Steer Arm Length | Front steer, 130 mm c-factor, 76.2 mm steer arm | |

| Brake System / Hub & Axle | Front | Rear |
|---|---|---|
| Rotors | Hub mounted, 4130, Fixed | Hub mounted, 4130, Fixed |
| Master Cylinder | Wilwood Compact, Front (.7in), Rear (5/8in), Bias bar on pedal | |
| Calipers | Wilwood, 1.75" piston, floating caliper | Wilwood, GP200, 1.25" opposed pistons |
| Pedal Force and Line Pressure @ 1g decel | 0.116 kN, 37.4 bar | 0.116 kN, 46.7 bar |
| Hub Bearings | 6813-zz, Shielded single row angular contact, 65x85x10mm | 6813-zz, Shielded single row angular contact, 65x85x10mm |
| Axle type, size, and material | Rotating axle with tripod profile inside, Al 7050 T7451, OD 65mm | Rotating axle with tripod profile inside, Al 7050 T7451, OD 65mm |

| Ergonomics | |
|---|---|
| Driver Size Adjustments | Pedals 3.25" adjustable, replaceable foam seat insert. |
| Shift Actuator (type, location) | N/a |
| Clutch Actuator (type, location) | N/a |
| Instrumentation | State of Charge, error readout |

| **Electrical Power/Control/Systems Management** | |
| --- | --- |
| Power Distribution Management / Control | 250amp Main HV fuse, 2amp LV fuses, Motor Controller/BMS current regulated |
| Wiring / Loom / ECM mounting | LV - 20 gauge single conductor shielded copper, HV - 00 gauge single conductor shielded copper |
| Battery / Charging System | Li-NMC 20Ah, Regenerative Braking up to 20Amps@300V |
| Grounding | Chassis Copper Trace, Rubber Isolation Mounts |
| Driver Assist Systems | N/a |
| Logging / Telemetry | N/a |
| Special Sensing Technology | Current Sensing/Voltage On-Off Sensing/Cell Temperature Sensing |

| **Frame** | |
| --- | --- |
| Frame Construction | Space frame |
| Material | 4130 steel normalized |
| Joining method and material | Tig weld with ER70S-2 |
| Bare frame mass with brackets and paint | 31.75 kg |
| Impact Attenuator material | Dow Impaxx 700 |
| Impact Attenuator dimensions | Width = 305 mm, Height = 356 mm, Depth = 254 mm |
| Impact Attenuator energy capacity | Standard Impact attenuator, Type 14 |

| **Tractive System** | Front | Rear |
| --- | --- | --- |
| Motor Manufacturer / Model / Type | | Enstroj/Emrax 228/Medium Voltage |
| Number of Motors / Location(s) | | One/ Rear |
| Motor Driven Wheels (location) | | Rear wheel drive differentiated |
| Maximum RPM (1/min) / Maximum Torque (Nm) | | 5000rpm/240NM |
| Maximum Torque until xx RPM | | 4000rpm |
| Maximum Power (per motor) | | 100kw |
| Type of Motor Controller(s) | | One, Bamo D3, Unitek |
| Motor Speed Sensors | | Resolver, directly contected to output of motor |
| Nominal Motor Voltage | | 3 phase, 350 Volts |
| Accumulator Cell Manufacturer / Type | | Eig / C020B |
| Nominal Cell Voltage / Capacity | | 3.65 Volts / 20000 mAh |
| Accumulator Cell Technology | | Li-NMC |
| Accumulator Cell Configuration | | 72 cells in series |
| Accumulator Voltage (fully charged) | | 299 Volts |
| Combined Accumulator Capacity | | 5.3 kWh |
| Coolant System and Radiator Location | | Two radiators located in side pods |

| **Drivetrain** | | |
| --- | --- | --- |
| Drive Type | 520 Chain | |
| Differential Type (if used) | Torsen (012000) | |
| Final Drive Ratio | 3.75 : 1 | |
| Vehicle Speed @ max power (design) rpm | | |
| 1st gear, 2nd gear | 115 kph | xx kph |
| Half shaft size and material | Hollow, 20mm OD 12.6mm ID, 4340 Rc53/55 | |