The Design and Construction of the Minnesota State University, Mankato 2008 FSAE Racecar

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ABSTRACT

This paper describes the research and development of the 2008 Minnesota State University, Mankato Formula SAE® racecar. The overall design goal was to create a high performance, open wheel formula-style racecar. Using the 2007 FSAE racecar as a benchmark, heavy emphasis was placed on improving past designs.

INTRODUCTION

The design of the chassis focused mainly on reducing overall weight, improving the structural rigidity of the frame and the car's handling capabilities. The specific goals of the project were to decrease the overall weight of the car to 390 pounds, create a car with neutral handling and more predictable in turns, fabricate a durable light-weight driveline, improve stopping power and reduce brake component weight. The engine design goals were to create a reliable, responsive, high-revving engine with a flat torque curve.

CHASSIS

Frame– The frame was developed through sequential modifications made to the design of the 2007 FSAE racecar. The initial design process was done by assembling balsa wood frame members with hot melt glue to create a one-eighth scale replica of the 2007 frame. This frame, weighing 10.44 grams, was then fixed at the left front, left rear, and right rear suspension mounting points. Then a specific weight of 100.8 grams was applied to a fixed length lever arm (6.25 inches) which acted upon the frame along the right front suspension mounting point. The deflection of the balsa wood frame was then measured in reference to the ground.

According to the appropriate calculations, the final result for the 2007 frame was 31.138 in-g/degree. After completing the baseline tests, a sequential series of modifications and improvements were made to the balsa wood frame. Every effort was made to triangulate or relocate frame members to obtain a more rigid, light-weight structure. After five revisions, the model had a mass of 10.69 grams. The torsional rigidity results were 77.770 in-g/degree for the final balsa wood model - an increase of almost 250 percent over the 2007 model. The actual frame when physically tested required 5825.25 lbs to deflect 1.00".

Tubing– The tubular space frame was chosen over the monocoque alternative as the general frame structure. This was primarily due to the lower cost and less complexity of the space frame. Calculations were made to optimize the materials used to construct the frame. Mild steel (1020 DOM) was the tubing of choice primarily for its impact resistance, or durability, for its ability to be easily welded and for the much lower cost per foot. Alternative high-cost alloy steel choices, like cromoly (4130), contain a high amount of carbon, making them brittle and susceptible to cracking, especially at welded joints. In addition to its high cost and brittleness, cromoly tubing offers no weight benefits without decreasing wall thickness as per SAE rules. When comparing the strength to weight-per-foot of available mild steel tubing sizes, it was clear that there was room for improvement over the 2007 frame. The main hoops and bulkhead tubing sizes specified by SAE® were found to be the lightest material that would meet the safety requirements. However, the bulk of the 2007 frame was made of 1.000" X 0.065", weighing 0.649 lbs/ft and having a bending modulus of 733.79 Lb in^2. The 2008 frame was constructed primarily of 1.125" X 0.049" tubing, weighing 0.563 lbs/ft with a bending modulus of 840.79 lb in^2, making it lighter and stronger.

Welding–- Tungsten inert gas welding (TIG or GTAW) was chosen primarily because of stronger welds generated by a reduced heat affected zone. TIG welds are often lighter than other types of welds and easily allows different thicknesses of material to be welded together. In addition to the process used, the filler rod is critical to the strength and quality of the welds. After researching the subject, the filler known as ER-70S-2 was chosen. The silicon and manganese ratio allows the weld joint to remain as strong - if not stronger - than the surrounding heat affected zone.

Cockpit– Driver comfort is an important aspect of a racecar's overall performance and drivability. The drivers of the 2008 racecar were asked to assume a comfortable driving position in order to take several ergonomic measurements. These included the angle of the seat back, the distance from the soles of the feet to the juncture of the seat back and seat bottom, steering wheel height and the distance of the steering wheel to the driver's shoulder. These measurements, along with the measurements of the 95% male, were applied to the cockpit design in order to obtain a comfortable seated position and compliment driver performance.

Finite Element Analysis- Pro| Engineer® Wildfire 3.0 was used to model the chassis, engine and many sub-components to fine-tune the positions of various frame members, brake calipers, sprockets, hubs and uprights. It was also used to simulate forces various components would encounter while in operation to ensure failures would not occur.

SUSPENSION

The suspension of the 2008 Formula SAE® car for Minnesota State University, Mankato has undergone a nearly complete redesign from its 2007 counterpart. The 2007 suspension was lacking in many critical characteristics, making the vehicle difficult to predict, drive and tune. These are all things that have been deemed severely important to a Formula SAE® racecar considering the dynamic events at competition are primarily handling-based. To achieve these characteristics, we are using a pushrod type front suspension with unequal-length A-Arms measuring 14" on top with a 47 degree spread and 20" on the bottom with a 36 degree spread.

Wheels and Tires- 18x6.0-10 R25A Hoosier® tires were used for both front and rear, along with Keizer® 2pc aluminum, +2.00" offset rims. This combination will lower the center of gravity and rotational inertia. This tire choice is also the narrowest available. This will allow for the most heat to build in the tires, ultimately maximizing traction.

Control Arms- Two types of tubing were tested for the control arms. The first was 1/2" square tubing and the other was 5/8" round tubing. The test welded a 12"-length tube to a stand held horizontally. A notch was made and 83 pounds were hung from the tube. The square tubing deflected .939", while the round tubing deflected only .19". After testing, the decision was made to use 5/8" round tubing. Furthermore the constructed A-arms were tested for lateral force. A weight of 1250 pounds was applied to the arms with no deformation

Suspension Rates– The 2008 ride and roll rates were designed to improve transient handling while also improving high-load, steady-state handling. A 1.75 Hz front and 2.1hz rear ride frequency were chosen to give the vehicle a much lower frequency than 2007's 4 Hz front and 3.8 Hz rear ride rates. This equates to a 40 pound per inch and a 63 pound per inch wheel rate, respectively. Bump rubbers were chosen to provide adequate ride rate in excessive bump situations and prevent chassis bottoming. A roll gradient of 1.5 degrees/g was chosen as a base and is adjustable with anti-roll bars in the front and rear. The 2007 car lacked weight transfer because of steering. Therefore, the 2008 car was designed with +6 degrees of caster to de-wedge the chassis during cornering. This will effectively wedge the chassis as the wheels are straightened, making it easier to exit corners.

Roll Center Location/Migration- Roll center location/ migration was of high interest for the 2008 suspension. The roll axis was lowered and brought closer to ground level to reduce jacking forces and increase the efficiency of the inner tires. Lateral motion was of high concern as well, and was kept within a .2"/degree roll box. These attributes were implemented to make the vehicle extremely predictable in bump and roll.

Scrub Radius– Scrub Radius was another key area of concern because of the increased mechanical caster trail in the 2008 design. Since the magnitude of force required by the driver is a function of scrub radius, mechanical caster trail and other attributes of the steering system, the scrub radius was reduced to 1" to keep positive wheel feedback and allow brake caliper clearance.

Steering– The elimination of both U-Joints provided more direct steering with less friction, while reducing cost and weight. The rack is a Stiletto® 6.4:1 ratio; this was the best rack due to its performance, quality, "Quick" gear ratio and aluminum components.

Weight– Weight-saving gains were made by slotting the frame instead of using tabs to attach the control arms. Also, a significant amount of material was removed from the uprights because of our brake caliper design change, with the brake caliper mounts centralized rather than being outboard floating mounts. The final area of weight savings was the choice of spherical bearings used for the suspension mounting points. Through load transfer calculations and wheel loading, it was found that 1/4" bearings had no adverse effects on the vehicle and reduced weight.

Adjustability– The addition of an adjustable front and rear sway bar greatly increased the tunability of the suspension. Four-way adjustable cane creek dampeners simplify dampener adjustments reducing parts required for tuning the vehicle.

BRAKES

The 2008 brake system components consist of three Wilwood® PS-1 2-piston fixed caliper brakes, two in the front, one in the rear, and two Tilton® 77 series 5/8" bore master cylinders. These components are light in weight, compact in size and easy to obtain. The opposing piston caliper was chosen to shrink the size and weight of the system. One caliper with pads weighed 1.21 pounds for a combined caliper weight of 3.63 pounds. The master cylinders weighed 0.3 pounds each. Combined weight of the master cylinder and the calipers is 4.23 pounds. The combined components weighed less than the 2007 car's combined caliper weight of 4.56 pounds.

Brake Forces– The brakes provide adequate stopping force for a combined car and driver weight of up to 644 pounds with an applied force of 85 pounds utilizing a custom steel brake pedal with a mechanical advantage of 6:1. The required combined front and rear line pressure needed to lock up the brakes is 1535.97 *psi*; we have 1662.34 *psi* available in our system. With these pressures, we achieve a clamping force of 900 pounds per wheel in the front and 737 pounds of clamping force in the rear.

Rotors– Steel rotors were utilized rather than aluminum because of their higher coefficient of friction, cost-effectiveness and wear characteristics. The rotors were lightened as much as possible without affecting structural stability and finite element analysis was done.

Pads– A high friction metallic pad was used because it is the only one made for this caliper application. At operating temperature the coefficient of friction of the brake pad is 0.42.

DRIVETRAIN

The driveline on the 2007 car was simple and lightweight but at the same time had reliability problems. This is attributed mostly to the 350 degree temperatures the "sprotor" experienced during braking. The temperature increase caused the aluminum to lose much of its strength, often rolling the teeth of the sprocket over and causing chain slippage. The 2008 car has diverted from the "sprotor" setup and returned to a sprocket and rotor setup. This eliminated heat in the sprocket and allowed the brakes to be mounted anywhere. The sprocket was also made lighter by removing material where the brake contact area was on last year's car. A separate rotor was added, causing a small amount of excess weight, but improved braking and eliminated the problem of the sprocket deforming because of overheating.

Differentials– The Torsen® differential was chosen this year because of its low weight, compact size, availability and our familiarity with the differential. It is a Zexel/Gleason differential with a torque bias ratio of about 2.6 to 1 after it is broken in. The custom housing is made out of 7073 aluminum, and by using a custom differential housing the weight is from 8.625 pounds stock to 4.59 pounds, greatly reducing the rotating mass.

Axles– TRE® drive axles from Taylor-Race were chosen because of their weight, strength and availability. These are also being used because of their compatibility with the differential and hub selection. These axles are capable of handling 1300 lb-ft of torque which was more than adequate for our engine and gearing. The axle setup for the 2008 car is 1% lighter than the previous year's. The drive angles of the axles were minimized to reduce the frictional power losses that come with large angles. The axles have a drive angle of 4.4 degrees on the left and 3.2 degree on the right. Driveline angles were kept low by aligning the centerline of the sprocket with the centerline of the wheel, and placing the centerline of the sprocket only 1" behind that of the wheel.

Gearing– The final gear ratio this year was 3.73:1. This was done to utilize the power in the lower RPM range to get out of the corners better. The use of an air shifter made it easier and faster to shift gears. Based on the size of the course, a high top-end speed is not needed.

ENGINE

The Honda® CBR™ 600 F4i engine was chosen because of its performance characteristics, reliability, and availability.

Exhaust– The exhaust was designed to maximize scavenging through a specified RPM range, increase horsepower and flatten the torque curve.

To achieve design goals, the exhaust was designed to have equal primary runner lengths to avoid exhaust gasses colliding in the collector. The equal runner length allowed the exhaust gasses from one cylinder to reach the low pressure zone between pulses from the other cylinders.

Based on computer engine simulation results the projected peak power RPM range is 9000 to 10,000. Given this range, the calculated runner length was 18". The primary tube diameter also affected the RPM where peak torque is created. A smaller diameter primary creates a higher flow velocity and increased low end torque. This smaller diameter primary would effectively lower the usable RPM band, but a larger diameter primary would provide higher gas flow volume, raising the peak torque RPM. Using calculations and engine simulation software (Dynomation®) the primary diameter was chosen based on the targeted RPM range. The ideal outer diameter for the exhaust system is 1.25." The entire system is fabricated from mandrel bent 304 stainless steel.

To complement the primary tubing, a 4:1 style collector was chosen. At the engine's operating RPM range, a 4:1 will create the most flow and scavenging compared to a 4:2:1 or Y collector. A relatively short collector length was chosen because of the relatively high RPM the engine operates at. The collector utilizes a transition angle of 13 degrees from 2.00“ to a 2.25“ diameter. This design is commonly referred to as a “high velocity merge collector.” The diameter transition increases exhaust gas velocity, effectively decreasing backpressure and improving the efficiency of the exhaust system. The collector was slip-fitted and held to the rest of the exhaust with springs allowing the headers to expand when operating at temperature. The springs also allow for easy removal and installation of the entire system.

Electronics– The engine controller was chosen from a company called Performance Electronics. The Performance Electronics unit is a completely adjustable engine control system. It controls both the fuel injection and ignition tables and requires only a couple of inputs to function properly. Crank position and an indication of engine load are required for a running engine. Engine load can come from one of two sources: Manifold Absolute Pressure (MAP), or throttle position. We also have the ability to data log to a PC. The engine utilized the following sensors to monitor engine operating parameters:

• General Motors© Intake Air Temperature
• General Motors© Manifold Absolute Pressure
• General Motors© Coolant Tempter

The Performance Electronics® ECU Monitor software was used to tune the engine. The program allows adjustments to be made to the fuel and ignition tables. These changes include advancing or retarding ignition timing and increasing or decreasing injector on-time.

Cooling System– The radiator for the car is a lightweight, aluminum unit manufactured by Polaris®. To move air through the radiator, a lightweight 6.5” electric puller fan was implemented. The fan draws 6.3 amps and moves 330 CFM.

Shifter– To improve drivability and shift time, an air shifter replaced a manual shifter. One actuator is used for shifting, and another to engage and disengage the clutch. The system is a patented design that allows the driver to use a lever to override the system for slow-speed maneuvering and emergency stops. Carbon dioxide or nitrogen is supplied by a refillable 9oz bottle that is fitted with an on/off valve actuated by the push of a button.

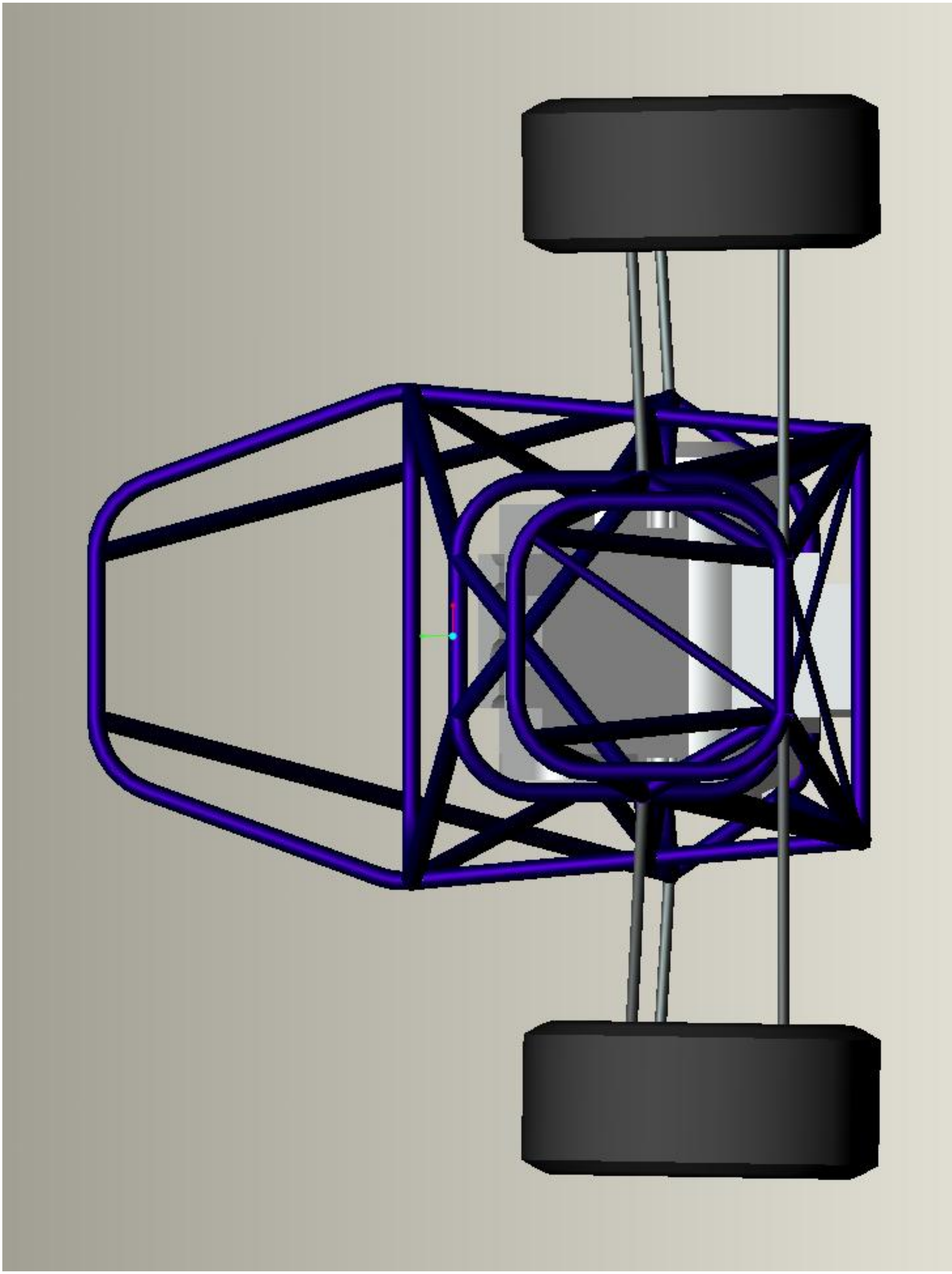
Intake Manifold– The intake manifold was designed to perform well throughout the engines power band. A large displacement plenum with four equal-length runners was chosen. To meet packaging and design restrictions, the intake plenum was placed above the engine’s transmission. The plenum was constructed of 0.065” wall 6061 aluminum sheet. To form the plenum, the sheet is cut, rolled and welded to the desired shape and volume. The runners were made of aluminum mandrel bends and attached to the head via silicon couplers. To complement the plenum, a 19mm restrictor and 35mm throttle body was mounted to the side of the intake manifold to distribute air within the plenum. Research and engine simulation software (Dynomation®) showed that a 4000 cubic centimeter (cc) displacement plenum would make the most average power from 6,000 to 10,000 RPM on a restricted 600cc engine. To complement the plenum, velocity stacks were implemented into the base of the plenum. By design, velocity stacks help pull air in from around the runner opening. The runner diameter remained the same as stock (1.370 inches ID). The runners extend straight out of the head and immediately bend to position the plenum accordingly. Complete intake tract testing was conducted to determine final placement of the restrictor, optimal length of the velocity stacks inside the plenum, and also to ensure that the volumetric flow rate of each runner was equal. A fixture was fabricated to flow the entire intake system simultaneously on a SuperFlow SF1020 flow bench. Valve lift on all 8 intake valves was set to .290” to simulate maximum port velocity. Runner velocities were measured at each runner using a pitot tube and flow rate calculated. Table 1 shows the initial testing at wide open throttle, before optimization.

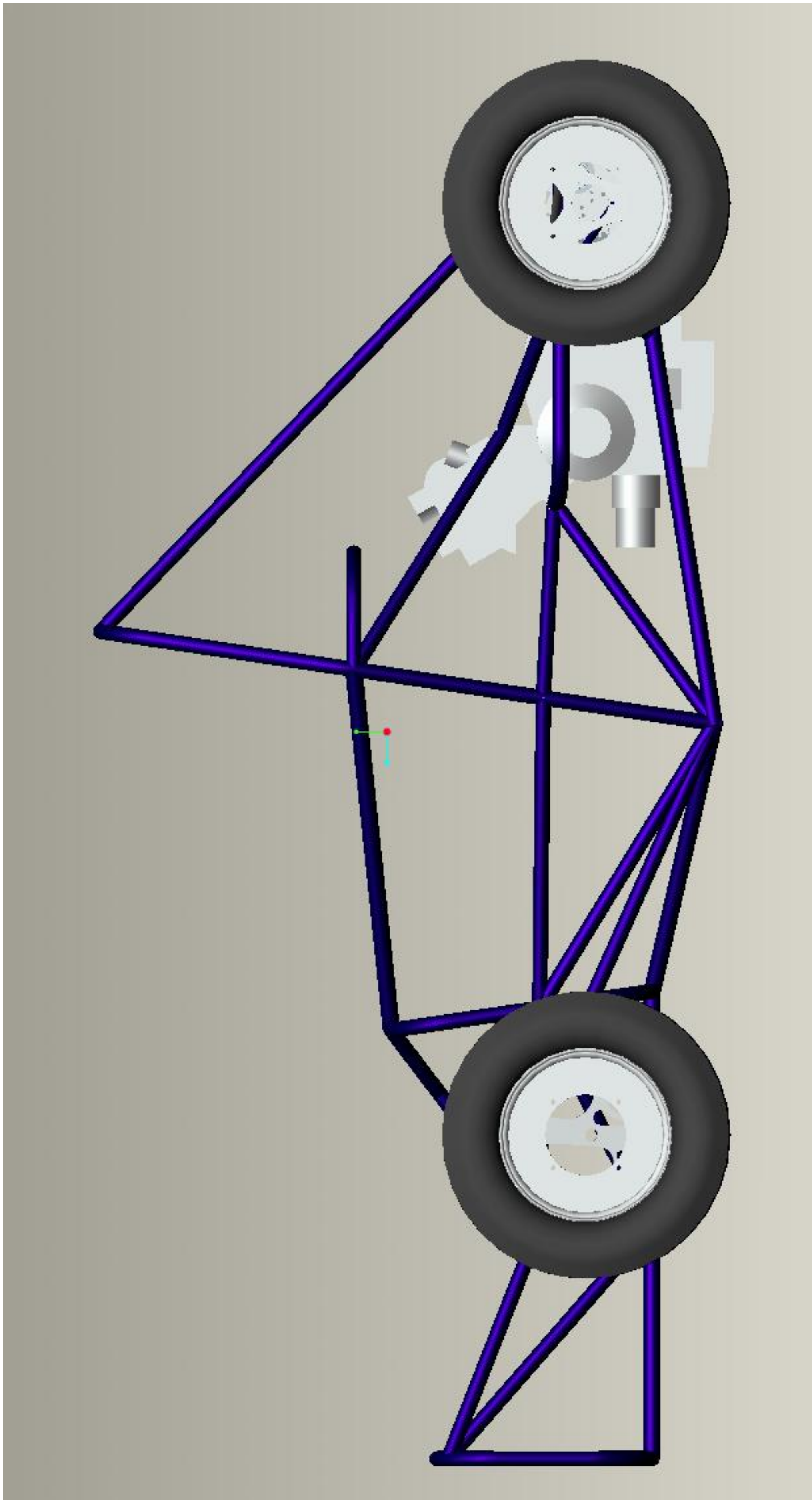
Intake Runner	Velocity (ft/sec) @ WOT	Volumetric Flow Rate (cfm)
1	46.57	28.62
2	44.26	27.18
3	43.06	26.46
4	47.08	28.92

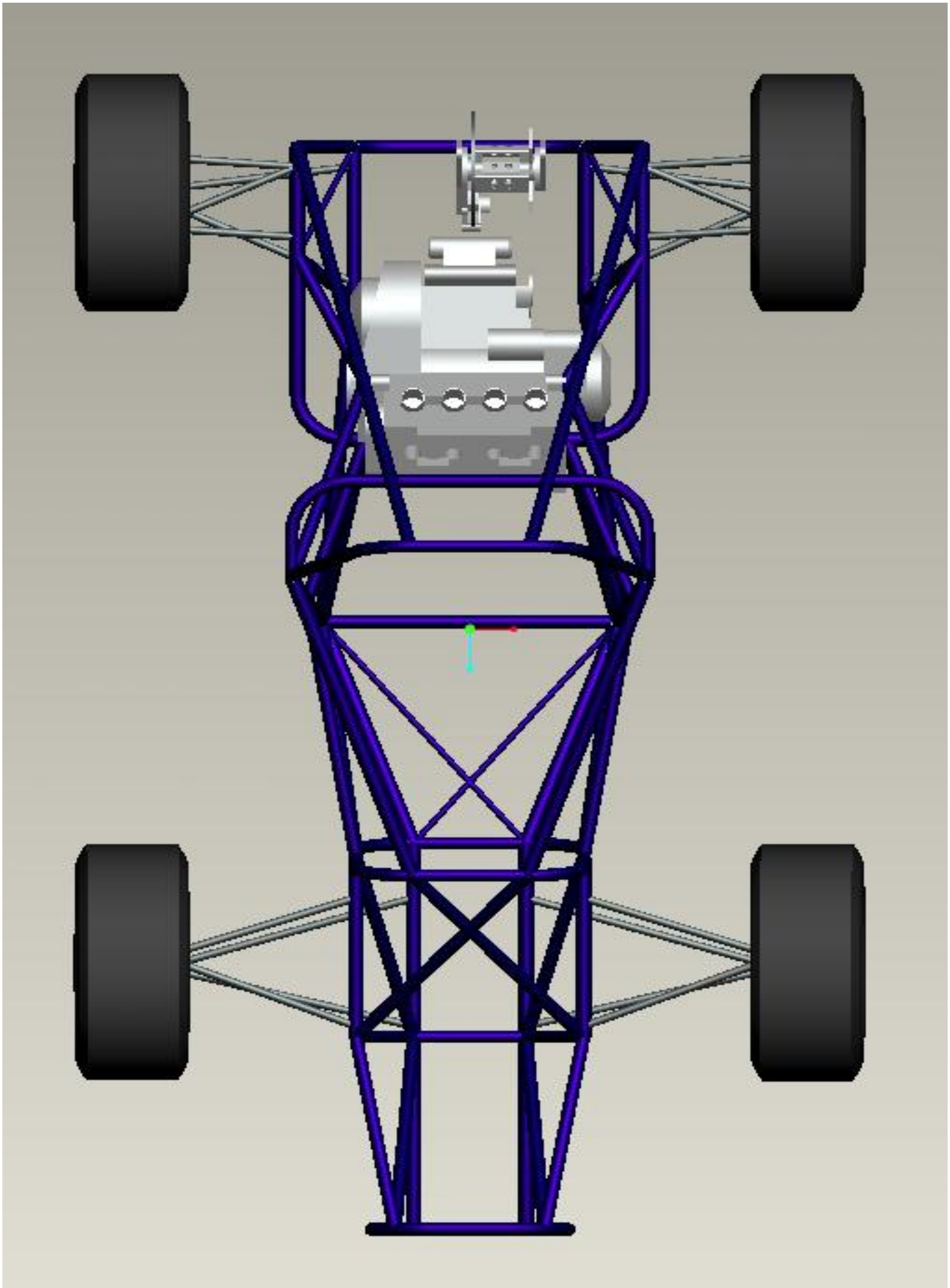
Pro| Engineer® Wildfire 3.0 was used to develop the required 19mm restrictor. The restrictors were drafted, built on a rapid prototyping machine and then flow tested. The restrictor that performed the best flowed 106 cubic feet per minute (cfm) at 25” of water. The inlet and exit angles were chosen after multiple angles were tested. The final restrictor was turned on a lathe from a piece of 6061 aluminum. The restrictor utilized a 23-degree inlet angle and a 7-degree exit angle. A Jenvey® 35 mm throttle body was chosen because of its simplicity, low cost and light weight. The throttle body was ported to provide maximum flow.

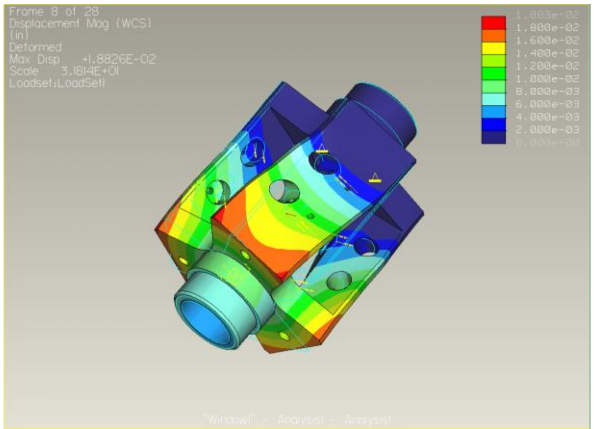
CONCLUSION

The frame team has created a rigid structure weighing close to last year’s car. Suspension team has developed an improved setup that will greatly increase handling and drivability. The braking and driveline systems have improved on weight, cost, durability and reliability. The engine goals dictated decisions on the parts used and the dimensions chosen to build the intake and exhaust systems. After testing and fabrication, the team is confident that the 2008 Minnesota State University, Mankato FSAE car will meet and exceed the goals and expectations developed by both the engine and chassis teams.

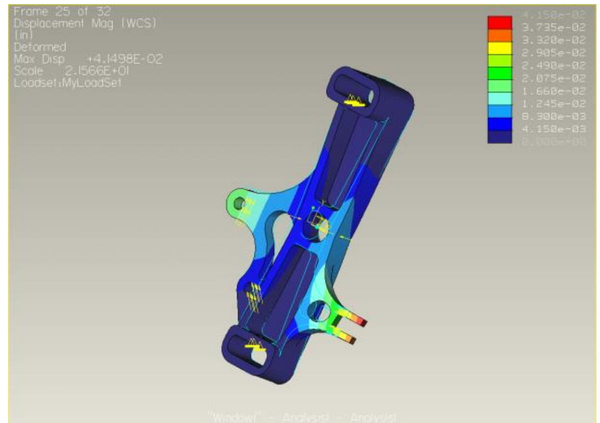




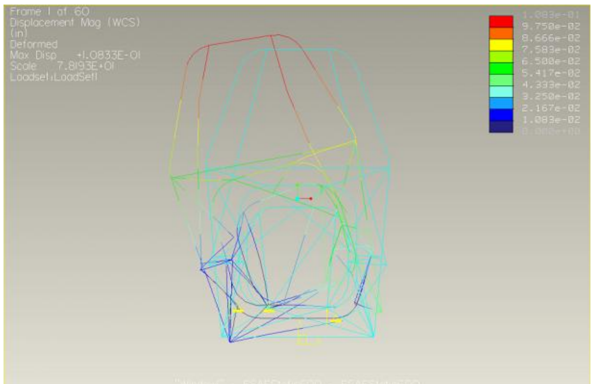


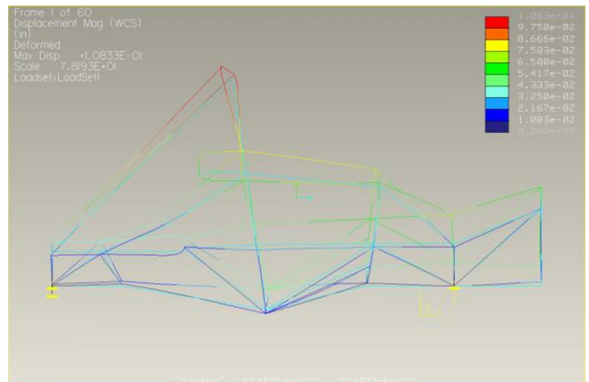


Torsen differential case at bolt shear load (6000 LBF).



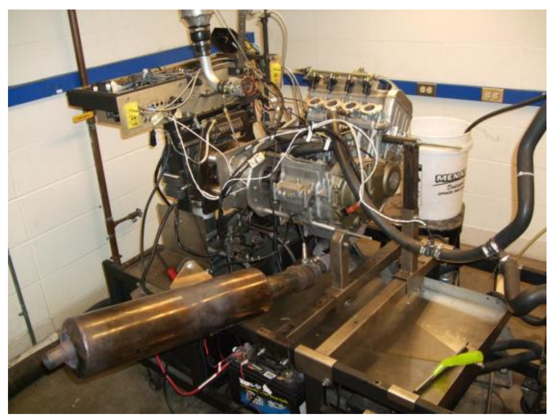
Front upright twisted at the spindle (600 lbf).





Front and side views of the 2008 space frame. 600 lbf input force on the suspension mounting point.

Intake runner flow and velocity testing.



Engine run stand and dyno setup.

Physical torsion testing. 0.103” @ 600 lbf.

Before and after frame deflection.