

University of Alberta – Design Report

INTRODUCTION

The University of Alberta has been a competitor in the Formula SAE competition since 1999. Those years of experience have provided the team with many lessons in the art and science of race car design, from which many successful vehicles have been produced. This design report will serve to discuss the design philosophy, design goals, and technical details of the University of Alberta's 2008 entry in the Formula SAE West competition.

DESIGN PHILOSOPHY

In order to begin conceptual design for the 2008 vehicle, it was necessary for the team to establish a design philosophy which would be used to guide the design process. This design philosophy was structured in the following manner:

1) SIMPLICITY

Many teams have failed to complete dynamic events due to unforeseen problems that have arisen from designs that are unnecessarily complex. By designing a vehicle that contains fewer complex features, the probability of failure is decreased, and the time required to investigate and determine the cause of a failure, should one occur, is also decreased.

2) ANALYSIS OF PREVIOUS DESIGNS

The advantages and drawbacks of previous designs were analyzed to determine whether they could be improved upon, or whether it was necessary to employ different methods to achieve a given requirement. Thus, it was possible to generate areas of focus for the design so that time and resources could be utilized most effectively to produce the largest design improvements.

3) GOAL AND CONSTRAINT SETTING

After the analysis of previous designs had been completed, goals and constraints were set for the 2008 design. These included constraints on overall vehicle characteristics such as wheelbase, track, and center of gravity location, as well as specific constraints for each subsystem of the vehicle.

TECHNICAL DETAILS

SUSPENSION

Un-equal length, non-parallel A-arms were utilized for both the front and rear suspension geometries. This suspension configuration minimized the introduction of positive camber to the un-laden wheel during roll. Varying the inclination of the upper A-arm allowed the static roll center to be located at 3.4mm above the ground plane in the front, and 17.8mm above the ground plane in the rear. These roll centers produce a roll axis approximately parallel to the mass centroid axis. Keeping the roll center just above the ground plane was a compromise between minimizing the roll moment of the geometry, and avoiding excessive jacking forces that arise from high roll centers.

By varying the length of the upper and lower A-arms, negative camber gain coefficients of 0.344°/in bump in the front, and 0.439°/in bump in the rear were achieved. The deficit of camber coefficient in the front suspension was intended to maintain maximum tire coefficient of friction in braking. See Figure 1 for a graph of camber vs. roll for a laden wheel. Anti-dive was not incorporated into the front suspension system to avoid suspension binding under braking. Anti-squat was not deemed necessary due to the relatively low torque produced by the engine, but compensation was provided by progressive rate rear spring geometry.

The front suspension has 4.8° of positive caster which produces 1.2° of negative camber gain on the outside wheel at 20° steer. The steering rack is located in the front steer orientation, connected to the steering wheel via a single U-joint, and is situated to produce zero bump steer. The steering links connect to the front uprights in a manner that produces 0% Ackerman steer.

The front suspension utilizes 250lb/in linear springs. This value correlates to a wheel rate of 91.5lb/in and a resulting natural frequency of approximately 2.44Hz. Mono-tube dampers with floating pistons allow for the jounce damping to be set between 66% and 113% of critical damping at all expected damper velocities.

Rebound damping is adjustable between 60% and 124% of critical damping. Damping coefficients were measured on a single cycle shock dynamometer, dictating that the damping fluid temperature remained close to ambient. It is expected that in service, the dampers will obtain a higher fluid temperature (lower viscosity); therefore the actual damping coefficient will be lower. Figure 2 illustrates the results of damper testing.

The rear suspension utilizes 185lb/in linear springs. Rear rocker geometry progressively increases spring rate in bump, but remains mostly linear in droop. This progressive nature allows for good tire compliance under average operating conditions, but resistance to extreme bump, roll, and squat. The geometry produces an approximate natural frequency of 2.52Hz at 0.5" jounce travel, and 2.71Hz at 1.0" jounce travel. Rear damping can be set between 64% and 110% of critical damping in jounce, and 59% and 120% of critical damping in rebound. Again, these values are larger than expected in operation.

A full mechanical analysis of the suspension system was completed, and all systems were optimized to be as light as possible. In the most severely loaded A-arm (rear upper) a safety factor of 1.1 was present in the improbable situation of combined maximum bump, lateral acceleration, and longitudinal acceleration. The upper rear A-arms were designed to minimize weight while maximizing strength in the required axis by capping the top and bottom of the A-arms, instead of increasing the entire cross section thickness.

WHEEL ASSEMBLIES

The wheel assembly consists of both mild steel uprights manufactured from sheet metal, and wheel hubs made from 6061-T6 aluminum. Both systems were designed to achieve maximum stiffness while minimizing weight. The weights of these systems were more critical than other areas as they contribute to unsprung weight. Hoosier 18.0x6.0-10 tires were chosen to reduce the moment of inertia, thermal inertia, and save weight.

The front and rear uprights are different designs, although symmetric left and right to reduce manufacturing and jigging time. The system was designed using steel sheet metal, as opposed to solid aluminum, to provide a stronger and stiffer system while reducing mass, as well as material and machining costs. A lower profile with single shear mounting maximized the distance between ball joints improving performance and simplifying maintenance. Safety washers were used on all single shear mounts to eliminate the possibility of disconnecting from other systems.

The front and rear wheel hubs are separate designs, although they share many similar features. A 6061-T6 aluminum one-piece hollow design reduced weight and increased stiffness. The use of thin-section angular ball bearings allowed hub diameters to be maximized, further increasing stiffness. Other features include a radius ring to allow for a large radius on the hub flange, intermediate spacers and preload rings to refine bearing preload, and a center locking nut to increase wheel balance and reduce weight. The hub also incorporates several safety features such as a cotter pin to lock the nut on, steel drive pins that can withstand the entire torque load, and a locking secondary preload ring to ensure correct preload on the bearings.

CHASSIS

A steel tubular space frame chassis was chosen for its simplicity in design, analysis, and manufacturing resources. A goal of 75 lbs overall weight (including mounts and welds) was set based on analysis of previous designs, and a torsional rigidity goal of 1000 ft-lbs/° was set based on a parallel analysis with suspension characteristics.

4130 steel tubing was selected for its higher strain energy absorption (when properly heat treated) over more traditional low-carbon steels such as 1018. Also, this tubing material was available in thinner wall thicknesses (0.035") than the low-carbon steel, providing an opportunity to reduce weight while maintaining adequate rigidity.

The chassis structure features triangulated side-pods around the driver cockpit to maintain torsional rigidity while providing the driver with

greater side impact protection. This added space also offers the driver ample room for vehicle operation and ease of egress should the driver need to exit quickly in an emergency.

The chassis was analyzed in Pro/MECHANICA as a beam model (see Figure 3). This model also contained the vehicle's suspension to which the loads and constraints were applied, since this was the closest representation to the physical testing apparatus used to test the torsional rigidity of the chassis once built. Pro/MECHANICA estimates the torsional rigidity of the chassis to be 1200 ft-lbs/°, meeting the required goal of 1000 ft-lbs/°. The results of physical testing will be available at the time of competition.

ENGINE/POWERTRAIN

The 2008 vehicle utilizes a 2003 Honda CBR600 F4i engine. Engine internals have been left stock to provide reliability and ease of maintenance.

GT Power was used to assist in designing the intake and exhaust systems. The goal was to flatten and broaden the torque curve as much as possible, while still achieving a maximum power of approximately 80hp. GT Power was also used to calculate theoretical pressure losses throughout the intake and exhaust systems, and ensure they were equal between all cylinders.

Steady state tuning of the engine was accomplished using a water brake dynamometer coupled to the engine via chain and sprockets. 3D maps for fuel and ignition were generated to create the highest torque and power output at large throttle openings, while maintaining fuel economy and drivability at smaller throttle values. The results of testing show a peak torque of 41lb-ft @ 7700RPM (see Figure 4) and peak power of 81hp @ 11700RPM. 80% of maximum torque occurs below 3500RPM, and power continuously rises until approximately 12000RPM.

The intake system is constructed of 6061-T6 aluminum. $\varnothing 1\frac{1}{2}$ " tubes are used for the runners, while a $\varnothing 4$ " tube is used for the plenum. This allowed the intake to be manufactured with common materials and reduced the amount of machining required. The runner lengths are 236mm and plenum volume is 2775cm³. The venturi was CNC machined from 6061-T6

aluminum and designed with an inlet angle of 6 degrees and an outlet angle of 14 degrees. These values provided optimum performance while maintaining a reasonable size.

The stock Honda fuel rail and injectors were used as they were the most cost effective solution and appropriately sized for the application. The stock fuel regulator was replaced with an aftermarket regulator to provide constant fuel pressure. The stock in-tank fuel pump was replaced with an external high pressure pump as it was much easier to install and replace. A custom aluminum fuel tank was used with fuel foam inserted to reduce slosh. The tank was mounted underneath the seat to provide the lowest possible center of gravity.

The exhaust system is a 4-2-1 design made of 1018 mild steel, with diameters of 2.25" for the primaries, 2.5" for the secondaries, and 3" for the outlet. The primaries and secondaries are equal length, measuring 510 mm and 300 mm respectively. A Yoshimura RS-3 muffler is used to dampen the exhaust noise while minimizing back pressure. The exhaust system also features bungs for thermocouples and an oxygen sensor. The thermocouples are used to ensure reasonable exhaust temperatures, as well as compare temperature between cylinders. During tuning, the exhaust temperatures reached a maximum of 920°C, and did not vary more than 25°C between cylinders. The exhaust system is ceramic coated to reduce external heat transfer to surrounding components.

Autronic's SM4 engine control unit is used to control fuel delivery, ignition timing, and other components such as the radiator fan, tachometer, and automatic shifting. The system utilizes fully sequential injection for improved starting and fuel economy. A single 4-channel igniter is used in conjunction with the stock coil-on-plugs to provide the ignition. A maximum of 50° of advance is used at 11700RPM while 35° of advance is used at 7700RPM.

The drive train consists of a 14 tooth front sprocket, 52 tooth rear sprocket, and 520 series X-ring chain. The rear sprocket is connected to a Torsen T-1 differential, housed in a custom aluminum case, and modified for a torque bias of 4:1. The differential transmits torque through

Taylor Race driveshafts and CV joints to a custom aluminum hub. This setup provides a reliable and simple method of transferring power to the rear wheels while minimizing losses. Other systems considered such as CVT's and direct driveshafts have either greater losses or added weight.

Shifting is performed using pneumatic actuators on both shifter and clutch linkages, controlled via steering wheel mounted pushbuttons. Using an ignition kill of 50ms, the system is able to achieve upshifts in less than 80ms. Smooth downshifts are performed without the need for manual clutching. The system also has a launch control feature that linearly releases the clutch over a one second period. Combined with controlling the engine speed ramp rate, a near perfect launch can be tuned easily. Wheel slip ratio is measured via two Hall Effect sensors measuring front and rear wheel speed. A slip ratio of approximately 10% is the goal to ensure maximum traction from the tires.

PEDALS/BRAKES

The pedal and brake subsystems were designed in parallel, with a main focus on providing the driver with a smooth, predictable braking response, while remaining lightweight, cost effective, and easy to manufacture.

Each pedal lever is constructed from 6061-T6 1" square aluminum tubing to reduce weight while maintaining adequate stiffness. At the driver/pedal lever interface, a lightweight block is sandwiched between the grip plate and the lever. By removing this block and replacing it with another of different thickness, the pedal engagement position can be adjusted to suit different drivers. A traditional clutch pedal has been replaced by a cockpit mounted lever to alleviate any driver confusion when engaging the pedals. The throttle pedal has been designed for 16.6° of travel to provide a smooth and predictable response at the throttle body.

The brake pedal connects to AP Racing pull-type master cylinders through a 4:1 pedal ratio, which provides the driver with firm pedal feel and allows full brake lock-up at 80lbs of pedal force. A cockpit mounted front/rear balance adjuster allows on-the-fly control of the brake balance,

providing the flexibility to respond to changing track conditions.

Outboard brakes are utilized on the front wheels. The front rotors are machined out of 1018 steel, and are acted on by cast aluminum Wilwood PS-1 two piston calipers. Rotor diameter ($\text{\O}7\frac{1}{4}$ ") has been maximized to aid heat dissipation and minimize the chance of brake fade due to thermal degradation of the pad material's coefficient of friction.

A $\text{\O}10$ " steel Wilwood brake rotor is mounted to the differential and acted on by a Wilwood Dynalite Single Floater caliper to provide rear braking. This configuration reduces both unsprung mass at the rear wheels and overall mass of the braking system.

ELECTRONICS

The driver's feedback system includes a 14 LED tachometer, a single digit display for gear, and warning lights for oil pressure, water temperature and neutral. High visibility LEDs are used to ensure they can be seen even in direct sunlight. The brake light consists of two, 1W high power LED's. These LED's will ensure the brake light can be seen by following drivers in any conditions.

An in-house designed and built stand-alone data logger collects data from the driver's inputs as well as the vehicle to analyze performance and provide feedback. Engine parameters are logged using the Autronic SM4 ECU. These values allow the engine tuner to optimize transient response of the engine after it has been tuned on the dynamometer.

CONCLUSION

The University of Alberta has taken the experience gained through years of competition and focused it into the design of the 2008 entry. Following a specific design philosophy and setting specific goals and constraints has led to the creation of a race car that embodies the engineering detail and passion that went into the design.

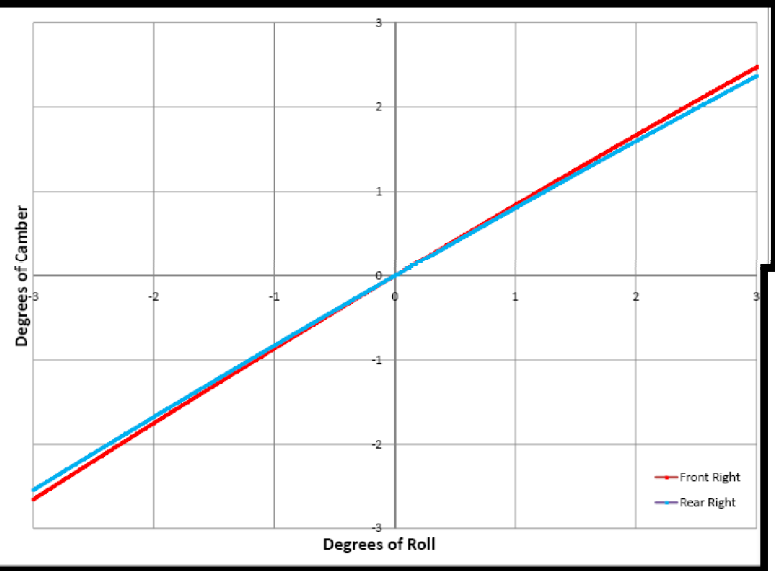


Figure 1: Graph of Camber vs. Roll for Laden Wheels

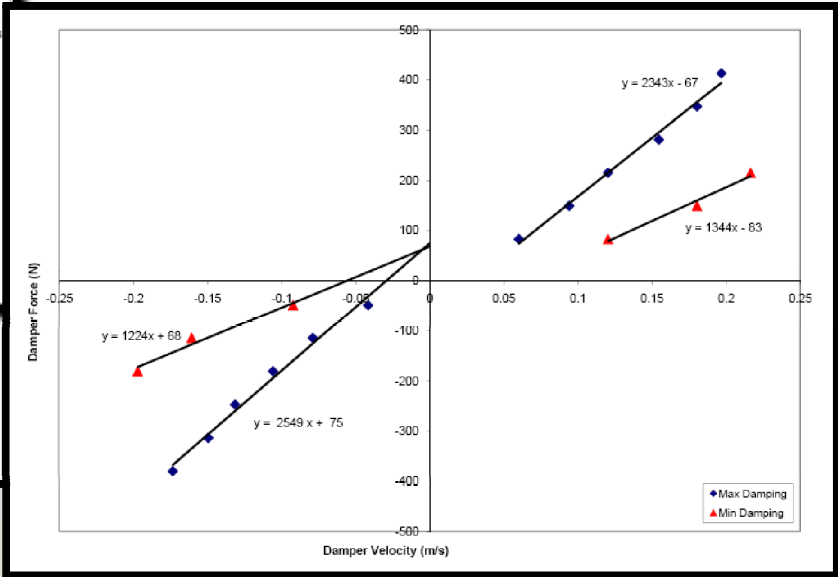


Figure 2: Graph of Damper Force vs. Velocity

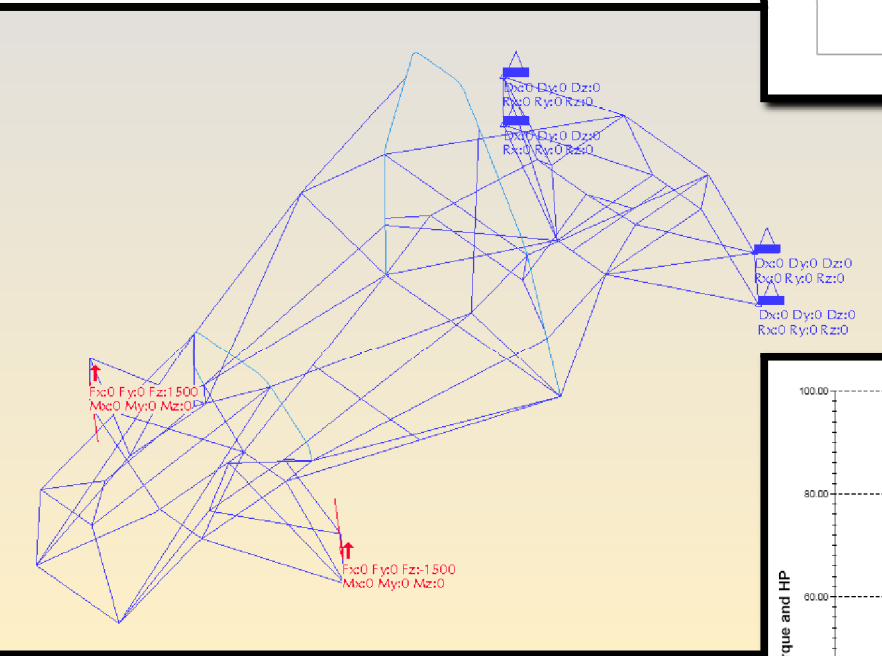


Figure 3: Pro/MECHANICA Chassis FEA Model Setup

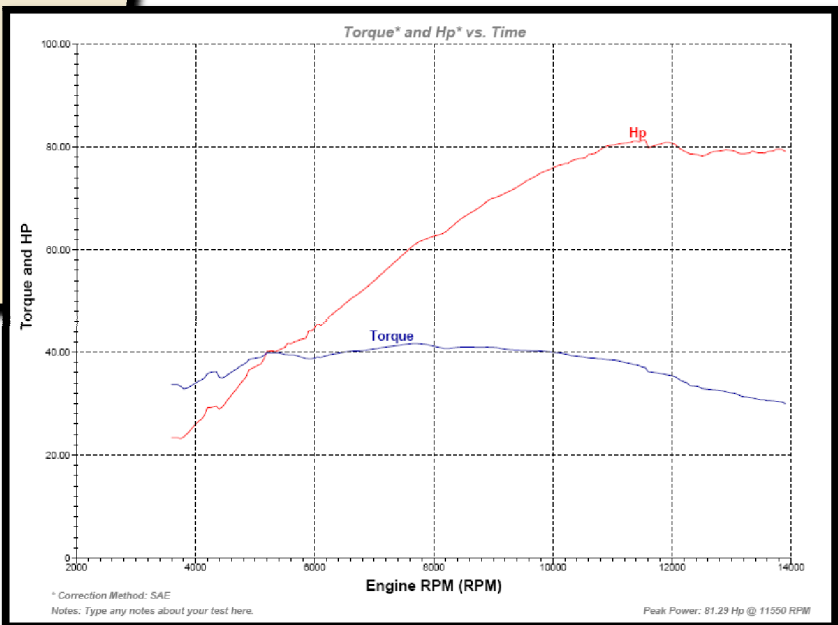
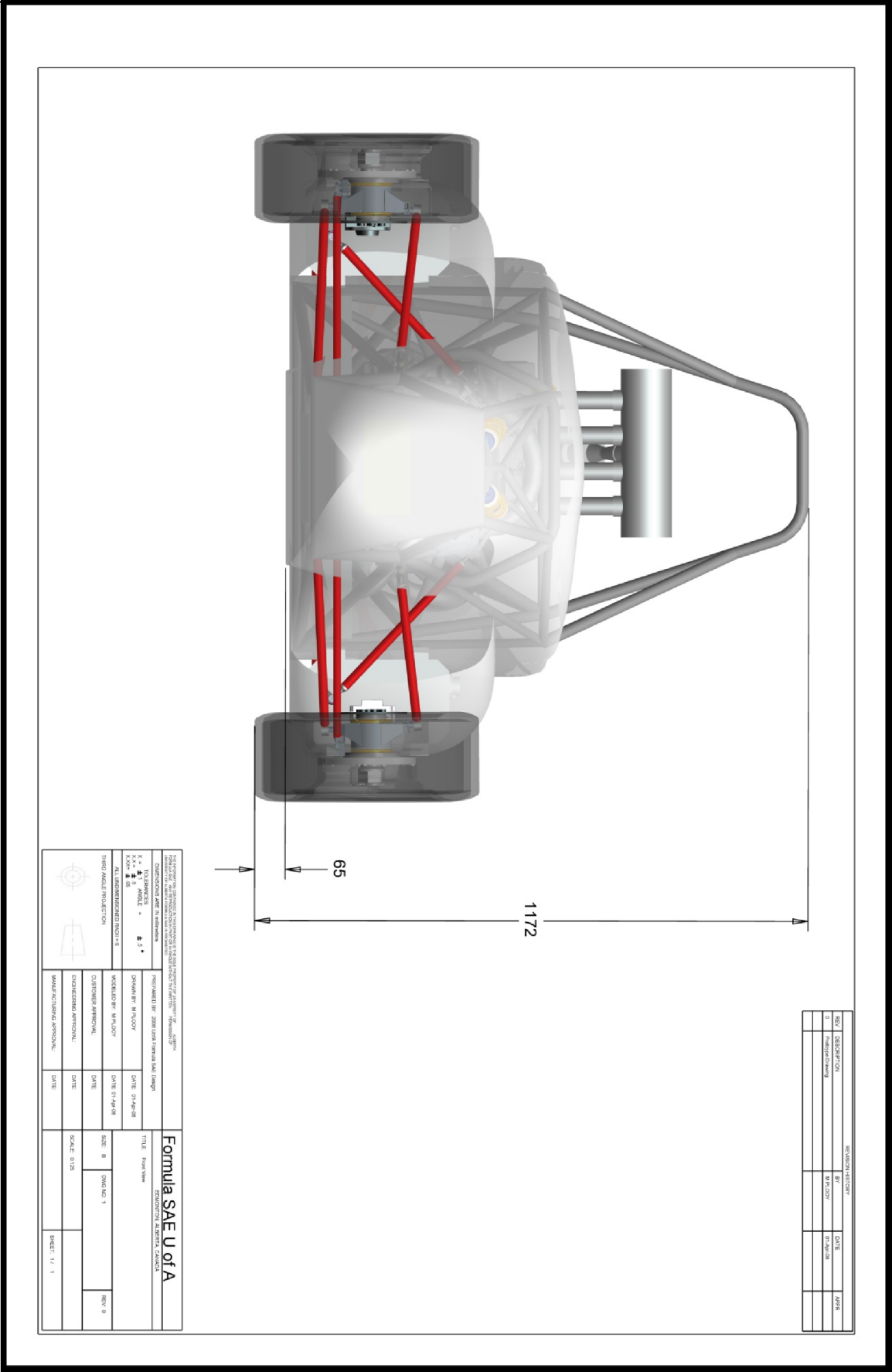
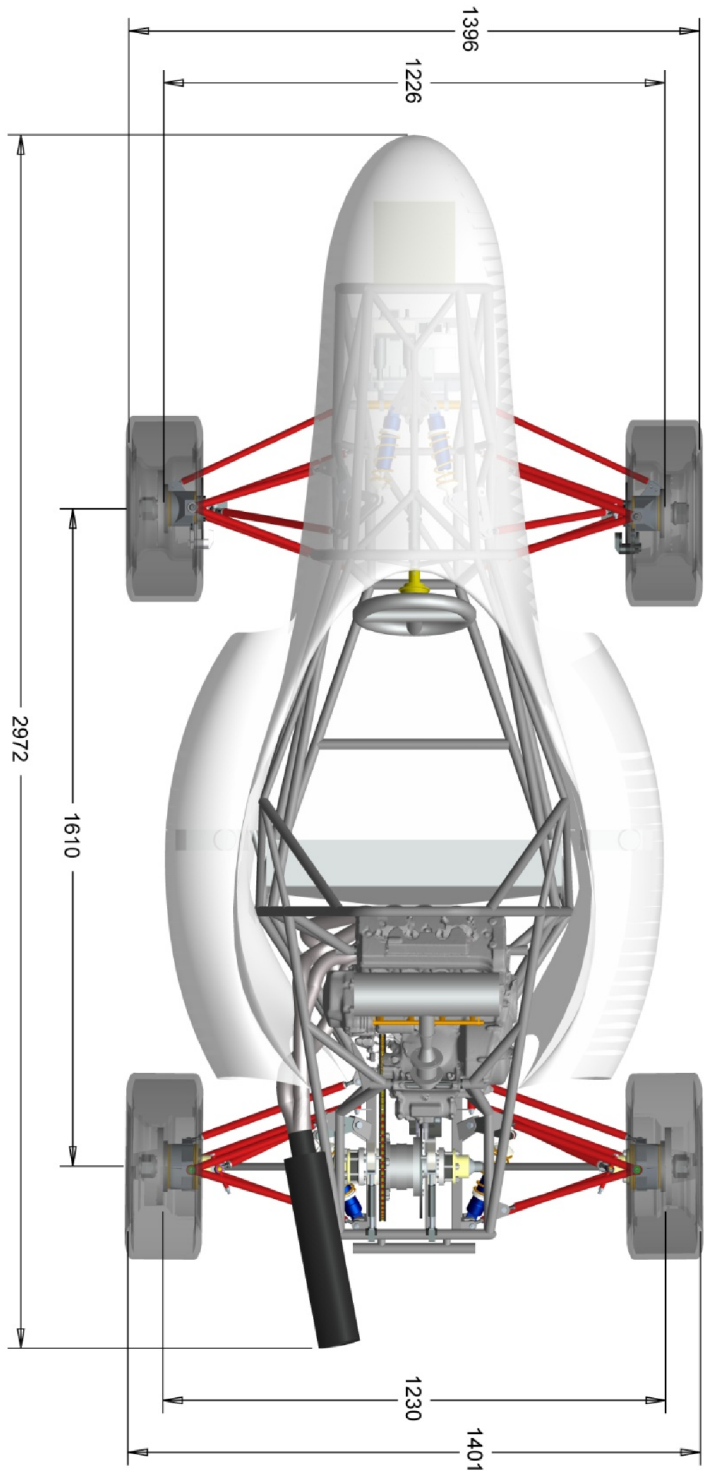


Figure 4: Graph of Dynamometer Tested Engine Torque and Power



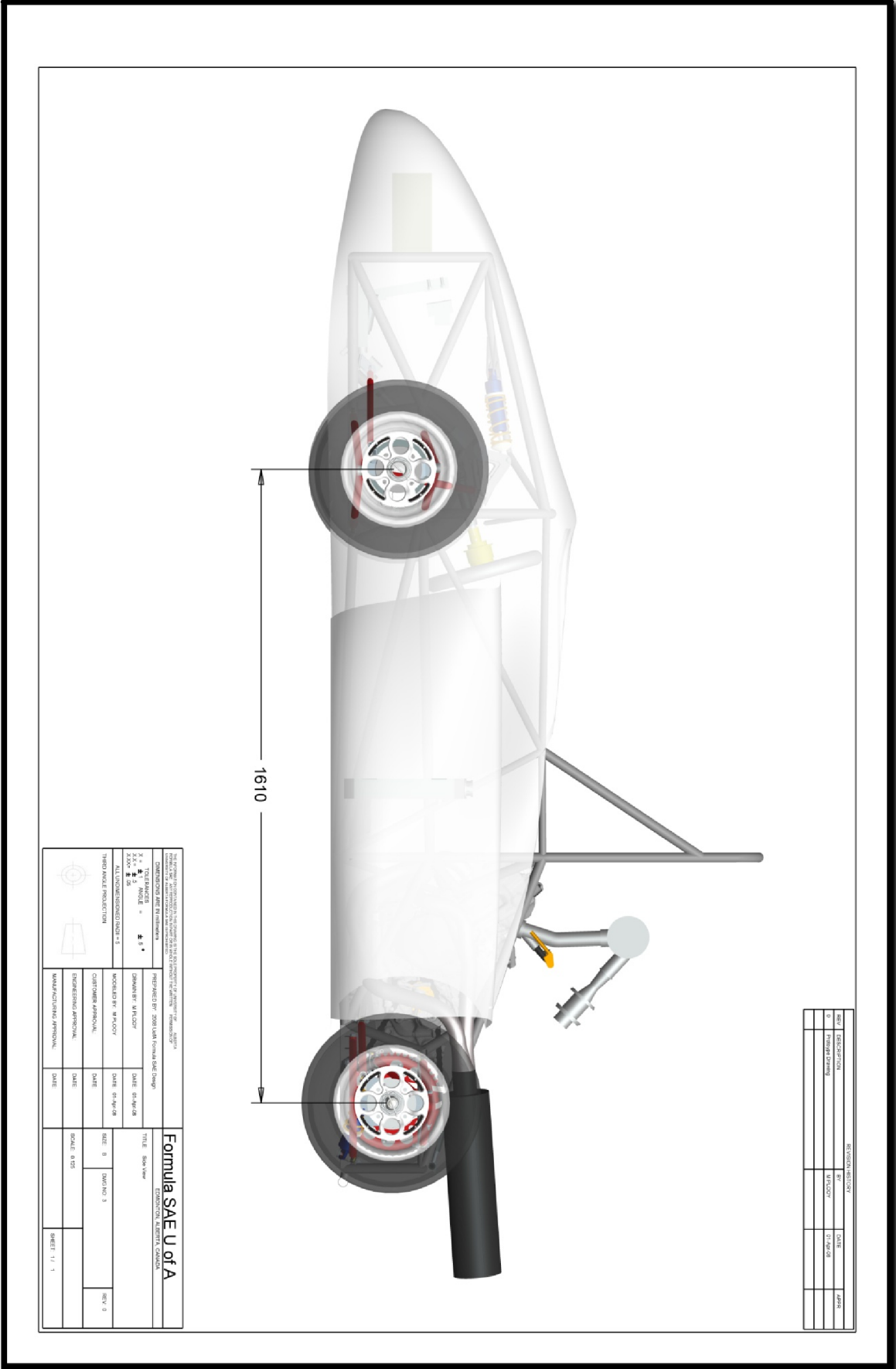
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