# HYBRID FSAE VEHICLE REALIZATION

**Final Report**

**KLK757**

**N10-08**

## National Institute for Advanced Transportation Technology

## University of Idaho

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16. Abstract

The goal of this multi-year project is to create a fully functional University of Idaho entry in the hybrid FSAE competition. Vehicle integration is underway as part of a variety of 2010-11 senior design projects. This leverages a variety of analytical and empirical studies undertaken in FY 2010 that demonstrate feasibility of the electromechanical powertrain, the energy management system, and the chassis layout. This project has engaged a broad population of graduate and undergraduate students in implementing sustainable transportation technology as well as creation of knowledge management systems that will benefit current and future competition teams. Approximately 1/3 of the allowed 20 MJ for the 22 mile endurance event will be supplied in the initial charge state of the battery pack. The remaining 2/3 is stored in the vehicle's gas tank. Accomplishments to date include a road load energy model, performance testing of the electric motor/controller, solid modeling of a reconfigured YZ250F motorcycle engine, suspension modeling, frame design, and a preliminary electronic archive designed to serve as a resource for just-in-time learning of hybrid terminology, subsystem descriptions, analysis methods, and performance tests. Specifications are given for all major subsystems and components.

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## EXECUTIVE SUMMARY

The goal of this multi-year project is to create a fully functional University of Idaho entry into the hybrid FSAE competition. Vehicle integration, pictured in Figure 1, is underway as part of a variety of 2010-11 senior design projects. Supporting technical work on powertrain design, suspension optimization, vehicle solid modeling, and energy management is underway in four different Master's Theses, three in Mechanical Engineering and one in Electrical Engineering. Accomplishments to date include a road load energy model, performance testing of the electric motor/controller, solid modeling of a reconfigured YZ250F motorcycle engine, suspension modeling, frame design, and a preliminary electronic archive designed to serve as a resource for just-in-time learning of hybrid terminology, subsystem descriptions, analysis methods, and performance tests. Specifications and subsystem descriptions are given for all major subsystems and components.

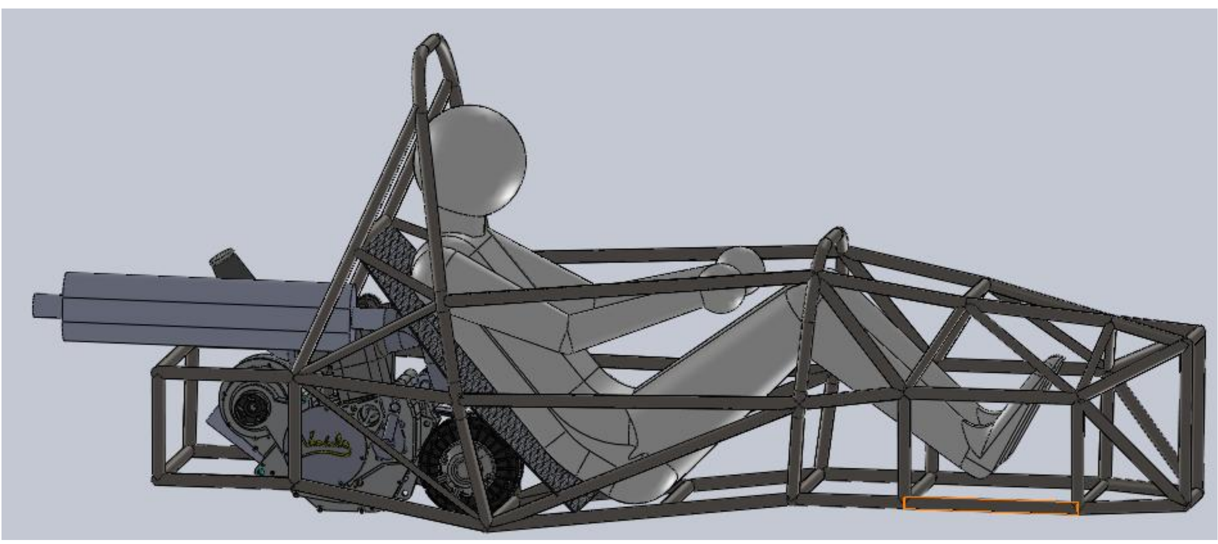

**Figure 1: Latest vehicle packaging w/driver.**

## DESCRIPTION OF PROBLEM

The word ‘hybrid’ describes the combination of attributes of two separate entities working to achieve one desired end goal. This concept has existed for many years in biology, Greek mythology, music, culture, and transportation. Manufacturers in several countries across the world now build highly efficient hybrid locomotives. Gantry cranes that lift rail cars on and off ships now utilize a generator to recover energy while the load is lowered. The Boeing Company is investigating hybrid packages for Auxiliary Power Units in their next generation 737. The military is conducting research on parallel hybrids to support stealth operations and rapid acceleration in their Humvees. The most significant achievement in mass produced hybrid technology is perhaps the Toyota Prius. Since the Toyota Prius hit the automobile market in 1997, its hybrid vehicles have continued to set record sales year in and year out. Students at the University of Idaho can learn much about hybrid propulsion systems by analyzing and emulating design features found in vehicles such as the Prius.

Since the inception of the Formula Society of Automotive Engineers (SAE) collegiate design competition in 1981, the rules forbid the entry of any hybrid vehicles. This has been challenged in the last decade by automotive market trends and public enthusiasm for green design. In 2007, the SAE rules committee created the hybrid only Formula Hybrid competition which is held at the New London Raceway in southern New Hampshire. This project was undertaken to establish a Formula Hybrid SAE (FHSAE) team at the University of Idaho.

The FHSAE rules are very similar to the FSAE competition rules with the major differences focused on the powertrain size and configuration. The displacement of the internal combustion engine is limited to 250cc’s and the use of any number of electric motors is allowed. The competition is broken into two categories. The first portion consists of “Static” events to evaluate the student’s engineering, manufacturing, and marketing knowledge and account for 450 of the allotted 1000 points available in the competition. The second part is comprised of “Dynamic” events to test the robustness, drivability, and overall handling of the car and account for 550 of the allotted 1000 points available in the competition. Subsystem design goals for the UI FHSAE vehicle are given in Tables 1-4.

**Table 1: Design Goals for Repackaged Powertrain**

| **Subsystem** | **Objective** |
|---|---|
| Engine Repackaging | - Connect electric motor to countershaft in YZ250F transmission<br>- Integrate with planetary differential to minimize mounting hardware<br>- Maximize reuse of stock YZ250F hardware<br>- Tilt engine head for tighter packaging behind driver<br>- Integrate WR250 starter in place of kick start |
| Intake & Exhaust | - Optimized for this application using Ricardo Wave<br>- Meet 110 dB sound level limit<br>- Thermally isolated from other components in rear box |
| EFI System | - Convert YZ250F from carbureted to fuel injected operation<br>- Customized fuel mapping for hot starts, cold starts, and idling |
| Teststand | - Monitor performance at any throttle setting with eddy current dyno<br>- Monitor fuel consumption<br>- Monitor air/fuel ratio<br>- Simulate engine tilting |
| Shifter | - Ergonomic, quick, and compatible with EFI system<br>- Provides feedback to driver on current gear and need for shifting |

**Table 2: Design Goals for Frame**

| **Subsystem** | **Objective** |
|---|---|
| Front Box | - Nodal mounting points for all suspension members<br>- Transmit loads from front bulkhead to main roll hoop |
| Driver Box | - Ergonomic sitting position while satisfying FSAE driver templates |
| Rear Box | - Rigidly attach engine and electric motor while protecting these components against all credible accident scenarios |
| Overall Frame | - Rigid (>2400 ft-lb/deg)<br>- Lightweight (<60 lbs)<br>- Compliant with FSAE rules |

**Table 3: Design Goals for Electrical System**

| **Subsystem** | **Objective** |
|---|---|
| Electric Motor | - Assist engine across a broad speed range |
| Controller | - Optimal motor speed control with adequate torque control and cooling for all conceivable operating conditions |
| Batteries | - High efficiency, light-weight energy storage configuration capable of holding approximately 1/3 of the allowed on-board energy |
| Wiring | - Color coded, labeled, and documented wiring system<br>- Common power and ground source block with correct fusing<br>- Sealed for inclement weather operation<br>- Quick disconnects for easy installation and removal<br>- Appropriate shielding for all wires |
| Data Logging | - Equip car with sensors needed to recreate driving conditions<br>- Display selected data to driver<br>- Log all performance data in a single file for future analysis<br>- Export data for further analysis |

**Table 4: Design Goals for Suspension, Steering, and Brakes**

| **Subsystem** | **Objective** |
|---|---|
| Suspension | - Maintain slight understeer suitable for novice drivers<br>- Keep rear wheels in contact with the ground at all times<br>- Tunable for different track conditions<br>- Absorb credible loads under lateral acceleration of 1g<br>- Couple wheel response from left-hand and right-hand side |
| Uprights | - Double sheer, thoroughly analyzed design that matches or exceeds strength, weight, rotational resistance, and torsional rigidity found in previous UI vehicles |
| Steering | - Ergonomic, variable response steering<br>- Adjustability between rack and pinion<br>- Limit play in steering<br>- Implement variable response steering<br>- Configure to reduce driver fatigue |
| Brakes and Pedals | - Capable of locking up all four wheels simultaneously at 20 mph<br>- Rising rate braking<br>- Ergonomic pedal design<br>- Minimize deflection with maximum braking force in FHSAE rules |

# APPROACH AND METHODOLOGY

## Road Load Model

Development of a road load model in TK Solver, and engineering equation solver, was one of the first tasks undertaken in the UI FSAE Hybrid vehicle project. The purpose of the model outlined in Figure 2 was to allow the optimization of gear ratios and motor selection in order to provide the best performing vehicle possible. As components and gear ratios became specified, the model evolved from an optimization tool to prediction tool. In this process, the basic operation of the model did not change significantly; the model has been expanded to include more operational parameters including fuel consumption, a crucial part of the endurance race.

The TK Solver model allows the user to specify shifting points, final drive gear ratios, drag coefficients, cornering capabilities, and many others. By varying the input variables, the user can compare different design modifications and control strategies to find the optimal setup for a given track. The model also allows the user to quickly modify features of the electric motor and engine by exchanging the lookup tables that the model references. Figure 2 is a diagram of the current TK Solver model structure.

The first portion of the model sets up track parameters, vehicle performance, and environment variables. Once these parameters are specified, the loop portion of the program begins. The loop is broken into one-tenth second intervals where the program takes the previous time, position, and velocity of the vehicle and, depending on the upcoming track geometry, determines if the vehicle is accelerating, decelerating, or cruising. This in turn governs the powertrain response, resulting in an acceleration or deceleration of the vehicle. The loop repeats for each time increment until the total distance traveled equals the length of the track. When, the loop ends, the program totals the time steps required and outputs various performance markers such as total time, fuel consumption, and various tables and charts describing vehicle performance at key locations in the course.

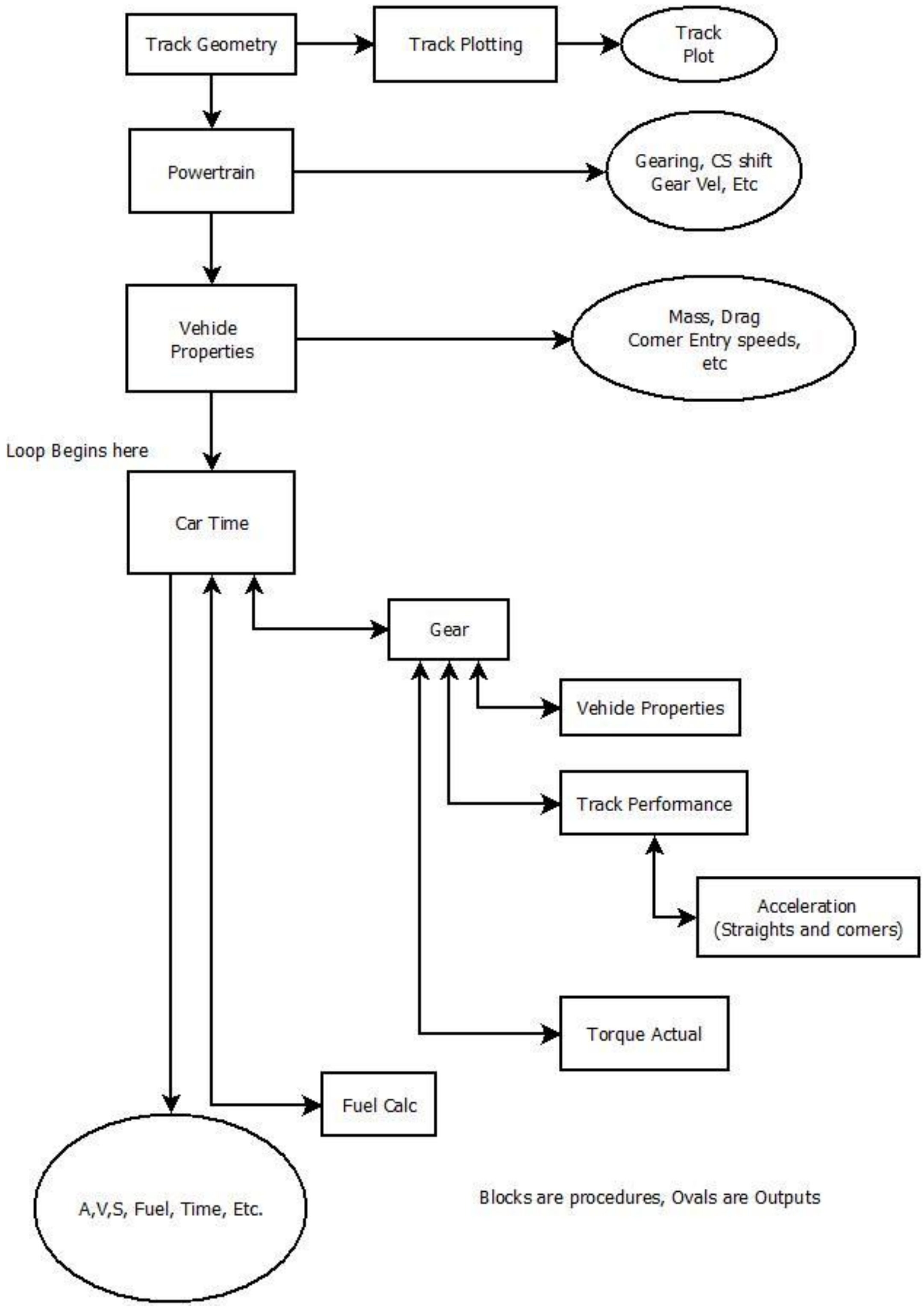


**Figure 2: Organization of TK Solver road load model.**

The initial setup of the program requires user input of various vehicle and environment variables to properly define the endurance event. The track must be defined in terms of starting position, straightaway locations and lengths, corner direction and radii, and finish line position. Once all of these data are specified, a track plotting subroutine plots the track features in the X-Y plane, allowing the user to visually check the track model. In addition to the visual check, the subroutine calculates the total length of user input sections and the total length input by the user to validate the track definition. A worst case track as specified by the FSAE Hybrid rules is shown in Figure 3.

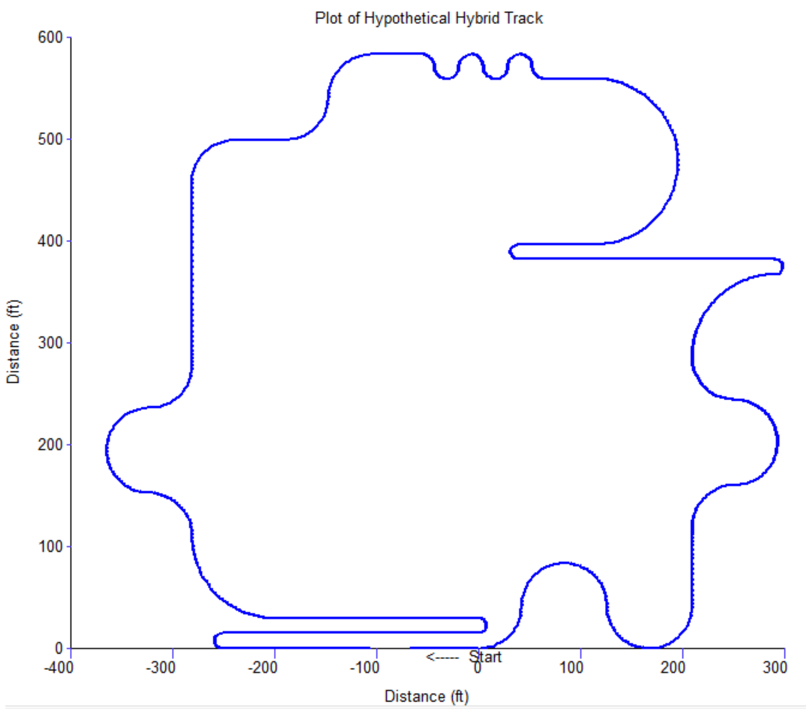


**Figure 3: Worst case track design under the 2011 FSAE Hybrid rules.**

Once the track has been defined, the user must establish the properties of the vehicle. The user may alter various parameters of the vehicle to allow optimization of both the drivetrain performance and also the control strategy. The parameters that can be adjusted as well as the parameters that are fixed within the subroutine are indicated in Table 5. Note that the values here are listed as “Known.” In this case, this means that they are either measured, such as in the case of tire diameter or mandated by rules or manufacture specifications. Other known values are selected by the user based on other knowledge or previous work during the design of the powertrain. These parameters are summarized in Table 5.

**Table 5: Powertrain Parameters**

| Parameter | User Selected/Fixed | Known/Assumption |
|---|---|---|
| Max Vehicle Speed | User | Known |
| Tire Diameter | User | Known |
| Final Drive Ratio | User | Known |
| Max Engine RPM | User | Known |
| Shift RPM | User | Known |
| Primary Reduction | Fixed | Known |
| Transmission Gear Ratios | Fixed | Known |

The *Powertrain subroutine* uses the input values to calculate various performance aspects of the vehicle such as the maximum velocity in each gear as well as ratios for various components of the powertrain such as the crank to countershaft and internal transmission gearing. The subroutine then outputs these values to the main variables section of the TK program for use in other subroutines. The *Powertrain subroutine* allows the user to vary items such as gearing and examine the effect they have on the performance of the vehicle over the course of the track. Additionally, the *Powertrain subroutine* keeps the code organized and contains all of the powertrain properties in one location.

The *Vehicle Properties subroutine* takes user adjustable inputs along with fixed values to calculate the physical properties of the vehicle as a whole. The subroutine calculates properties such as vehicle mass and rolling resistance as well as calculating the maximum velocity that the vehicle can maintain in the various radius corners expected on the endurance track. Vehicle properties parameters are summarized in Table 6.

**Table 6: Vehicle Properties Parameters**

| Parameter | User Selected/Fixed | Known/Assumption |
|---|---|---|
| Vehicle weight | User | Assumption |
| Coefficient Rolling Resistance | User | Assumption |
| Coefficient of Drag | User | Assumption |
| Frontal Area | User | Assumption |
| Corner Radii | User | Known |
| Cornering capacity | User | Assumption |
| Gravity, Air Density | Fixed | Known |

Note that many of the variables used in this subroutine are listed as "Assumptions." These values are estimates based upon the current design of the vehicle. Once the vehicle is assembled, the various parameters will be updated to reflect the actual vehicle which will increase the accuracy of the model and tailor it specifically for the UI FSAE Hybrid vehicle.

Once the basic track, powertrain and vehicle properties have been established, the analysis of track performance can begin. The operation of the *Car Time Loop* is a simple time step analysis, broken into one tenth second intervals. The loop begins by establishing blank lists for position, time, velocity, acceleration, and so forth. The lists all begin with a starting value of zero, except for the velocity which starts with an assumed value based upon the maximum velocity of the corner prior to the start line. This non-zero value is used simply to prevent solving errors at the beginning of the program. Once the initial values are established, the *Car Time Loop* routine uses a series of IF and THEN statements to determine which gear is currently selected (based upon vehicle velocity). The gear information along with the current values of velocity and position are passed to the various subroutines of the *Car Time Loop*.

The subroutines in the *Car Time Loop* analyze the information passed down and return a value of acceleration, cruise, or deceleration for the given time step. This acceleration value is then added to the current velocity to determine what the new velocity is after the time step. This, in turn, determines the distance traveled during the time step. These values are then logged in the blank lists and then recorded as the new current values to be used in the next loop. In addition to the position, velocity, and acceleration, the *Car Time Loop* also uses the user input fuel consumption value to estimate the fuel used during the time interval, which is also logged in a blank list. Various other values are recorded for plotting purposes including engine torque, motor torque, RPM, and so forth.

Once the loop reaches this point, it repeats from the gear selection point. This loop continues until the elapsed distance value equals the length of the track at which point the loop ends and several final equations are used to collect and organize performance data. The time steps are totaled to find the lap time for the vehicle. The fuel usage for each step is also totaled to find the fuel used in one lap, as well as multiplied by the race length to show the total fuel consumption. Average speed as well as fuel economy is also calculated for user reference. At this time, the program only calculates the fuel usage of the Internal Combustion Engine (ICE) and does not account for the energy used by the electric motor to obtain the indicated performance.

The *Gear subroutine* uses the current condition information passed down from the *Car Time Loop* to determine the power available from the powertrain to the wheels based upon performance tables and gearing. Various subroutines are called up to determine the possible and desired acceleration along with the actual ICE output for that time step, all of which are then passed back to the *Car Time Loop*. The *Gear subroutine* first calculates the speed of the countershaft based upon the vehicle velocity and gear currently selected. Using the ratios of the ICE and electric motor to the countershaft, the subroutine uses lookup tables to determine the approximate maximum output of each machine in terms of torque. The maximum torque available to the wheels is then found using the known gearing and is passed on to the *Track Performance subroutine*. This routine then returns an acceleration value which is passed back up to the *Car Time Loop*. This value is also used to find the actual torque used which is also returned to the *Car Time Loop* for use in calculating fuel consumption. Note that in this subroutine the *Vehicle Properties subroutine* is also called up, in this case, to find the drag present for the given velocity.

The *Track Performance subroutine* uses the current position and velocity to determine if the vehicle should be accelerating at maximum effort, cruising, or decelerating at maximum to find the acceleration value that it will pass back up to the *Gear subroutine*.

To make this decision, the current position of the vehicle is compared to the start position of each section (relative to the starting line) using another series of IF/THEN statements. Once the section of the track in use has been identified, a subroutine is activated, either for a straight or corner depending on the current section geometry. The same position and velocity data is passed to the straight/corner subroutine in exchange for the acceleration value, in addition the next type of geometry is also passed along including corner radius if needed. The *Vehicle Properties subroutine* is again called up to find drag information as well as the maximum entry speed of the next section of track geometry. The *Straight subroutine* compares the current position of the vehicle to the distance required to slow down (given the current velocity) to the entry speed for the next corner. If the position of the vehicle equals or exceeds the required braking distance, then the acceleration selected is maximum braking effort. If the position has not reached that point but the maximum velocity of the vehicle has been reached, the acceleration is zero and the vehicle is in a cruise mode. If, however, the maximum velocity of the vehicle has not been reached, the subroutine assumes maximum acceleration is desired. At this point, the acceleration is determined by using the available torque, mass, inertia of the vehicle, and drag to find the resultant net acceleration.

The *Curve subroutine* works in a similar manner to the *Straight subroutine*. If the next section of geometry requires a slower speed than the current curve (i.e., heading into a decreasing radius corner), a braking distance is computed and the vehicle decelerates at the maximum possible for a corner once it reaches the correct position. However, if the next section of geometry does not require a slower speed, it is assumed that the vehicle is already traveling at its maximum possible velocity, and therefore the acceleration is zero. In both previous cases, a value of zero acceleration indicates that the vehicle is “cruising.” This, in turn, calculates the actual torque and fuel consumption by using the drag characteristics of the vehicle. If the acceleration is greater than zero, a similar calculation is performed with the program assuming that maximum engine output at the indicated RPM. A third case involves an acceleration value of less than zero. This means that the vehicle is decelerating, and, therefore, no engine output or fuel consumption is required.

Once the actual torque and fuel consumption calculations have been completed, the loop checks the current position against the total lap distance (using IF/THEN statements). If the current position is less than the total, the loop repeats; otherwise the loop ends and the various lists are totaled and output to a variables sheet for user examination. Table 7 shows the various performance outputs of the program and their intended purpose.

**Table 7: Output Variables**

| Variable | Type | Purpose |
|---|---|---|
| Time | List and Value | List used for plotting; value allows performance rating |
| Position | List | Also used for plotting purposes |
| Velocity | List | Used for plotting |
| Acceleration | List | Used for plotting and error checking |
| Motor RPM | List | Combined with motor torque allows prediction of energy usage |
| Motor Torque | List | Used with motor RPM for energy usage prediction |
| Engine RPM | List | Used with engine torque to predict fuel consumption |
| Engine Torque | List | Used with engine RPM to predict fuel consumption |
| Fuel Consumption | Value | Total fuel used, performance indicator |

Using the various lists within TK Solver, multiple plots can be generated that provide insight about vehicle performance during a single lap. Figure 4 shows the RPM of the ICE over a lap. This allows the user to see the effect of gearing on the RPM range. The engine was tested using an eddy current dynamometer and found to be most efficient when the RPM is greater than 8000 RPM. The electric motor RPM is also plotted vs. time in Figure 5. This allows the user to examine gearing effects, providing insights how to keep the electric motor at its highest efficiency.

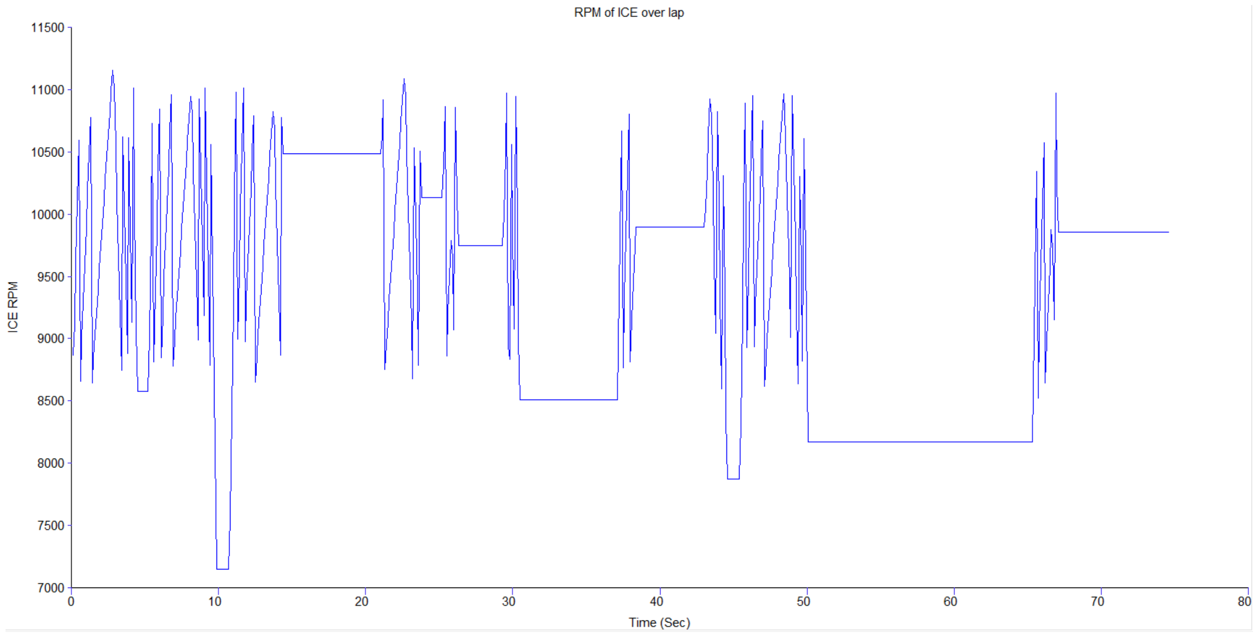


**Figure 4: Engine RPM vs. time.**

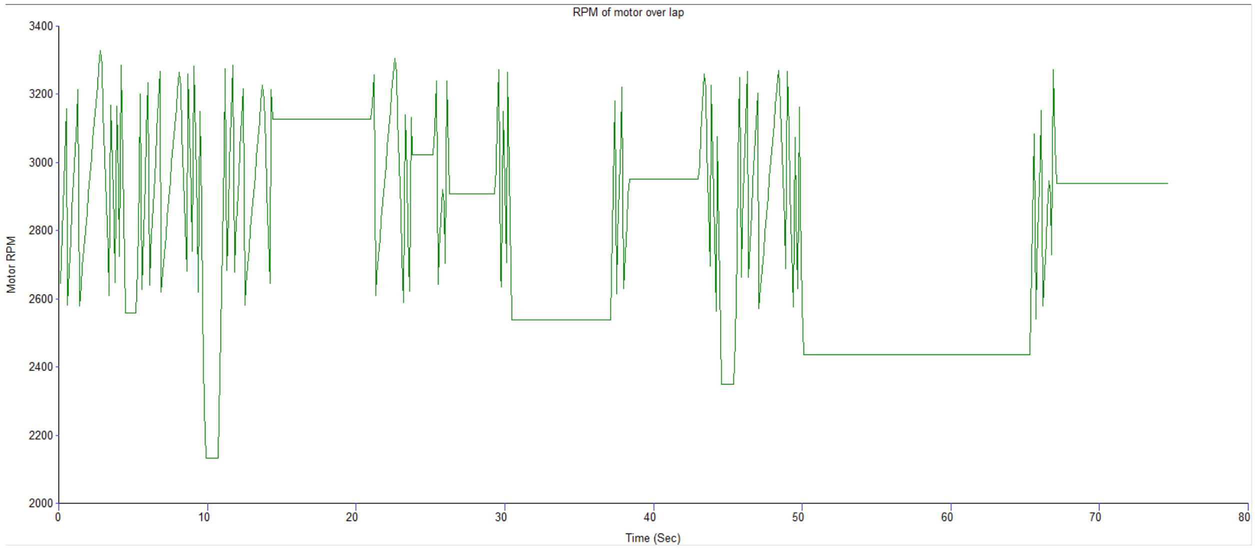


**Figure 5: Motor RPM vs. time.**

## Energy Management

A Lynch 46hp permanent magnet motor was chosen by reviewing the most frequently used motors in past FHSAE competitions and examining their compatibility with the YZ250F engine. Specifications taken into consideration were efficiency, combined torque of the motor and the gasoline engine that did not exceed torque limitations of the counter shaft, maximum power given these torque limitations, and the likelihood that the selected motor would be reused in future FHSAE vehicles.

Circuitry capable of interpreting a proportional signal and adjusting the motor armature voltage is required for vehicle speed adjustments. Since batteries will be used as the energy storage system, a constant supply voltage can be expected. A Kelly PM Motor speed controller was suggested by a Lynch Motor Company engineer as a good match for their motor when supplied by a constant source. Further analysis of Kelly controller specification sheet verified it was capable of operating a permanent magnet motor and rated for 400 Amps over ten second intervals. The operational voltage of the controller is compatible with a 48 Volt (V) configuration for the 2011 car and up to a 96V configuration in future vehicles. Other features include regenerative braking, internal temperature monitoring, and user-friendly light emitting diode [LED] fault codes.

Competition rules state a maximum total energy storage capacity of 20 megajoules (MJ) or 5556 watt hours (Wh). This equates to 0.61 gallons (gal) of regular unleaded gasoline using equation (1).

$$E_{AllGas} = \frac{E_{total}}{E_{GasDensity}} \qquad (1)$$

where

$E_{total} = 5556Wh$

and

$E_{GasDensity} = \frac{9138Wh}{gal}$.

Therefore, the use of any electrical energy storage will result in less than 0.61gal of gas available for the ICE. A 70:30, mechanical -to- electrical, energy ratio was arbitrarily selected as a reasonable starting point. Since electrical energy would be stored in batteries to maintain a more reliable and simplistic design, a greater amount of energy storage will be in the form of gasoline to maintain a lightweight solution. A 30% electrical allotment translates to 1667Wh by:

$$E_{electric} = E_{total} * 0.3. \qquad (2)$$

The minimum requirement for the battery pack was the ability to complete the electric only, 75 Meter [m], sprint in less than ten seconds but to design for a five second finish. A five second finish meant a competitive time compared to the 2010 FHSAE results.

Four battery specifications of importance included: nominal voltage (translates to maximum velocity), discharge rate (translates to maximum acceleration), operational temperature (function of discharge rate), and capacity (ability to travel the entire distance). Since the battery capacity has been predetermined by means of the TK Solver road model and temperature can be managed through packaging applications, the battery selection only consisted of nominal voltage and current parameters that met the desired capacity.

Initial battery research showed batteries are not intended for more than a ten times capacity (10C) burst discharge rate to avoid significantly decreasing the life of the battery. Recalling the motor's maximum armature current and the Kelly's rated current as 400A, a 40Ah configuration became the starting point for iterative analysis. The first step was to calculate the nominal battery voltage from equation (3),

$$V_{nom} = \frac{E_{electric}}{C_{Ah} * 0.8}, \qquad (3)$$

where 0.8 is defined by the Formula Hybrid Rules Committee (FHRC) as the approximate energy conversion efficiency of batteries and $C_{Ah} = 40Ah$. Therefore, the nominal voltage equals 52V.

Other configurations were considered trading off the importance of acceleration time and maximum speed. Also, factors beyond engineering design such as manufacturing lead time, customer support, and physical restrictions (i.e. dimensions and weight) had to be considered. The cell specifications for the 2011 system are 48V and 40Ah.

## Vehicle Integration

### *Engine Tuning and Shifting*

Since the YZ250F's inception in 2001, the engine has strictly been a carbureted one, The largest problem with carburetion is the use of jets. The jet has a fixed diameter and will inject a specific amount of fuel when a vacuum is caused by the change in RPM. A carbureted engine is tuned for a specific altitude and RPM range for maximum power. All other operation points and altitudes have inferior performance. The use of fuel injection allows the engine's ECU, in conjunction with the injector, to deliver the exact amount of fuel that the engine needs for maximum power throughout the entire RPM band. This includes built in altitude, temperature, and humidity compensation. In many cases, the injection angle can also be manipulated for emissions and economy purposes. It has also been well documented that carburetors are not built for lateral accelerations and fuel pickup issues have been observed by FSAE teams that have used carburetors. As shown in Figure 6, we have decided to fuel inject the 2005 Yamaha YZ250F with the use of the MBE EFI Conversion Kit with the MegaTune engine tuning package because of its ease of use and low cost.

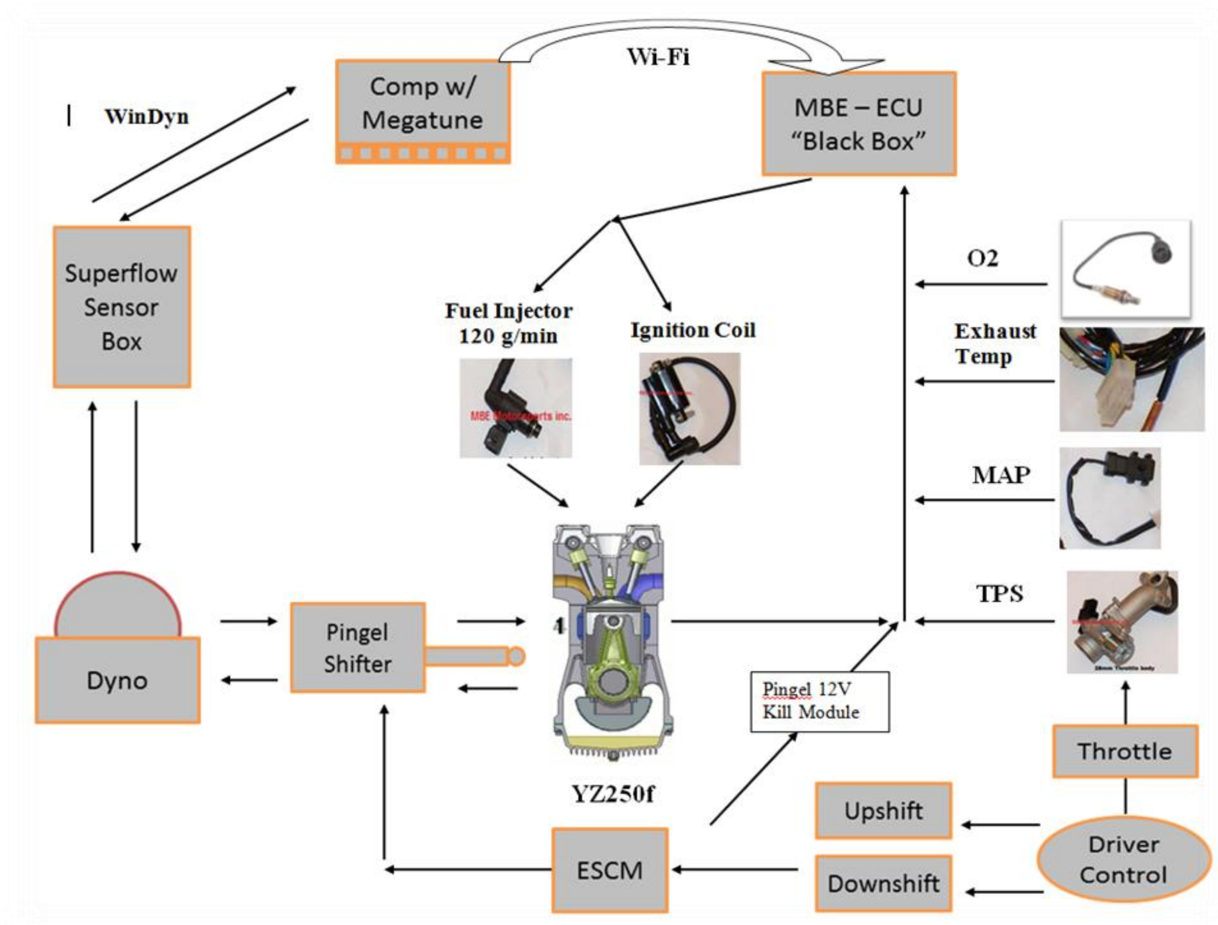


**Figure 6: Engine control system.**

### *Engine Repackaging*

The custom-built case design will act as a structural member and will accommodate components within and attached to the drivetrain that would have been unable to package with the original case. The case can also be designed to act as a load-transferring member between the suspension and frame of the vehicle. In this way, the suspension pickup points can be located off the backside of the differential, ultimately saving weight and increasing handling by lengthening the a-arms. The new case shown in Figure 7 also allows the location of the center of gravity (CG) to be easily controlled, allowing the benefit of keeping it low and next to the driver's CG. Along with the CG, there is an advantage of being able to lean the motor back 37° in such a way that the driver will be able to sit lower, resulting in a shorter car. The weight and cost have yet to be determined.

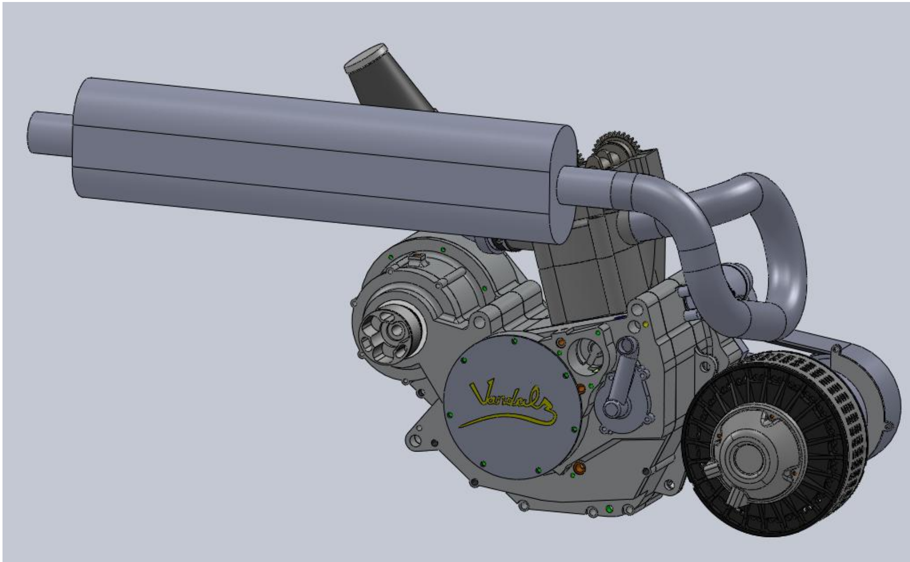


**Figure 7: Engine repackaging with Lynch motor.**

*Suspension and Steering*

Because this prototype racecar is being developed for the amateur autocross racer, predictability in the handling of the vehicle is of utmost importance. The design will use suspension parameters in Figure 8 that are conservative and adjustable.

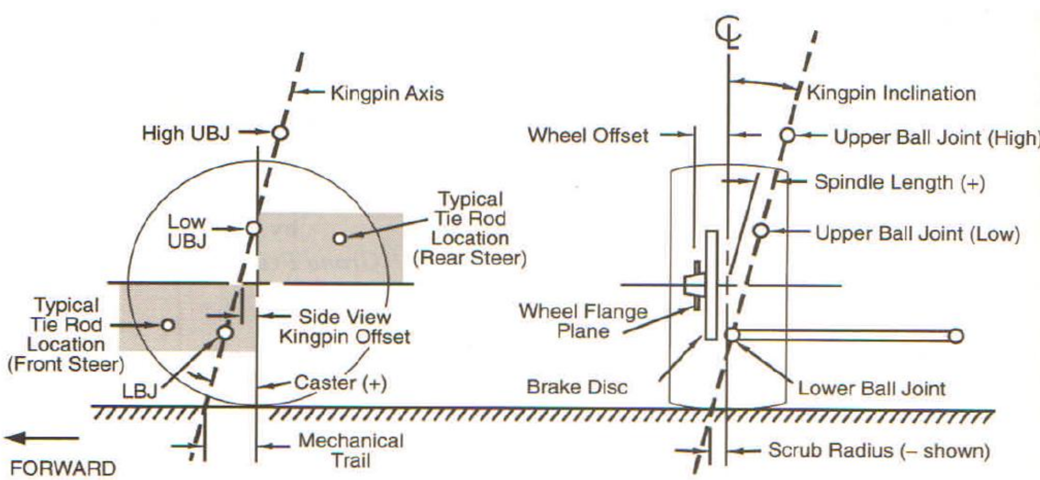


**Figure 8: Suspension parameters.**

The packaging of the front suspension was the first step in the suspension design. Target values for each of the parameters seen in the figure above had to be determined to give a guideline for the packaging. The target values selected are displayed in Table 8 and an explanation of why the values were selected follows.

**Table 8: Front Outboard Suspension Packaging Targets**

| | |
|---|---|
| **Kingpin Inclination** | 0º - 3º |
| **Scrub Radius** | - |
| **Mechanical Trail** | .3 in |
| **Caster** | 3º - 5º |
| **Tie Rod Location** | In shaded areas of figure |

The kingpin angle, spindle length, and scrub radius were determined by first placing the lower ball joint into the wheel as far and low as possible. The issues of the kingpin angle, scrub radius, and spindle length are interrelated. The larger the kingpin angle and spindle length, the greater the lift in the front during steering. The raising of the front end aids in the centering of the steering, and the more kingpin angle, the more positive camber gained in turns. The more spindle length, the more kickback in the steering as the wheel rolls over bumpy terrain. Scrub radius adds feel of the road and reduces static steering effort and many race cars operate more or less ok with large amounts of scrub. Scrub radius is not as big of a concern as it is on FWD vehicles.

Low kingpin angles subtract from the camber gain due to caster and are therefore desirable. Obtaining the lowest kingpin angle possible is recommended in various literatures and is the source of the target. It has been decided to minimize spindle length to reduce front end lift and kickback. The scrub radius that forms as a result from these decisions will be accepted because of the minimal effect it has on this particular application.

Steer-camber is a result of positive caster. Unlike kingpin angle, this steer-camber is favorable with the outside tire moving towards negative camber and the inside tire moving towards positive camber. While too much caster will make the steering very heavy and adds to the front-end lift, it also promotes straight-line stability and gives the steering that comforting self-centering capability. Race cars should not need a lot of self-centering capability. Following the recommendation of multiple experienced authors, we have set a target value of 3° - 5° positive caster.

Mechanical trail goes hand-in-hand with caster. Although you can have one without the other, mechanical trail is determined by the caster angle and the position of the kingpin axis and it's relation to the wheel center. Mechanical trail acts in concert with pneumatic trail, with the pneumatic trail adding to the mechanical trail. A balance needs to be reached between the two to give the driver some warning that the front wheels are nearing the limit, while maintaining reasonable steering torque. If pneumatic trail is much greater than mechanical trail, a fall off of aligning torque would be pronounced before the peak of lateral force. This would result in loose steering right before the tires break into high slip angles. If mechanical trail is much greater than pneumatic trail, then the tires will break loose without giving the driver any warning and the aligning torque will cause premature driver fatigue. Target values were selected based on various tire data and literatures.

The tie rod location depends upon whether a top mounted or bottom mounted rack and pinion is wanted. During cornering, any lateral displacement of the ball joints in relation to the tie rod will cause a steer angle. A resulting toe-out or toe-in due to the lateral force is unavoidable. It is better to have toe-out under-steer than toe-in over-steer. By placing low mounted racks in front of the wheel centerline and high mounted racks rear of the wheel centerline, it would ensure this. For packaging reasons, we are currently planning on placing the rack and pinion low and forward of the wheel centerline.

After packaging the outer suspension, we focused attention on inboard suspension packaging. Selected targets are shown in Table 9. Because suspension design is an iterative process and built upon compromise, target values had to be determined to give the process a focus.

**Table 9: Front Inboard Suspension Packaging Targets**

| | |
|---|---|
| **Swing Arm Length** | 100 - 150 in |
| **Roll Center** | -1 – 2 in |
| **Camber Gain** | .5 - 1 deg/in |
| **Track** | 50 in |
| **Ackermann** | **≈ 100%** |

The unequal length between the upper and lower a-arm was selected to reduce the change in track (associated equal length a-arms) that cause the car to dart when running on rough surfaces. The non-parallel a-arm design was chosen to dial in the roll centers to our target location and target the camber gain. This design also reduces part of the weight transfer in cornering.

The swing arm length is the distance from the chassis centerline to the instant center (the point of intersection of upper and lower a-arms if they were extended infinitely). Determining a desired swing arm length requires compromise. The length of 100 to 150 inches was chosen for the compromise between going with a short and ultra-long design. The advantages of this length include low roll centers, minimal scrub (track variation), and only small camber change going into bump and droop. They also produce a braking grip advantage over shorter swing arm lengths. The disadvantages include less control over roll center sideways migration and positive camber gain. This swing arm length is obtained by instigating a non-parallel design of the a-arms.

The roll center is the theoretical point at which the body will roll. High roll centers offer a smaller chassis roll moment while low roll centers have larger rolling moments, which must be resisted by the springs. If the roll center is above ground, the moment generated by the lateral force on the tire and the instantaneous center cause the wheel to move down and the sprung mass to move up. This is called jacking. On the other hand, if the roll center is located below ground, the sprung mass will be forced down. The target value of –1 to 2 inches was selected to minimize jacking while at the same time keeping low rolling angles.

With the selection of an unequal length a-arm design, negative camber gain in jounce and roll are created. The amount of gain is dependent upon the ratio of length between the upper and lower a-arm. A certain amount of negative camber gain is advantageous in cornering; as it counters the positive induced camber due to tire deformation and kingpin inclination. The desired amount of camber gain is prompted by the tire used, inflation, lateral loads, and other contributing suspension geometries. A target of .5 - 1 deg/in was selected by consulting with experienced drivers and has been validated with computer simulation.

Wide tracks offer a vehicle that is more sensitive and responsive on a tight course, but more twitchy in the straight-line. Also, with wider tracks there is less weight transfer to the laden wheel in corning. Given that the car is being manufactured for the autocross circuit a wider car is more reasonable. A 50 in front track was selected based on the performance of the best car and the top ten competitor’s cars.

Any bumpsteer greater than 0.1 deg through full suspension travel will produce a wondering effect in the front end of the car over a bumpy track. The tie rods must be placed in a geometry that eliminates this effect. After the hard points are decided upon, the tie rod joint can be determined.

The rear suspension targets are less of a concern than the front because the rear uprights are fixed. Selected values are summarized in Table 10. The most important concern with the rear setup is the adjustability of toe. Toe adjustments will counter the effects of slip angles in tight corners and add to the effective tractive effort of the tires.

**Table 10: Rear Inboard Suspension Packaging Targets**

| | |
|---|---|
| **Swing Arm Length** | 80 - 100 in |
| **Roll Center** | -1 – 2 in |
| **Camber Gain** | 1 – 1.5 deg/in |
| **Track** | 48 in |

Swing arm lengths on the rear are considerably shorter than that on the front because of the effects of caster steer in the front. Positive caster adds to negative camber gain in steer. The shorter swing arm lengths induce a greater camber gain to match that of the front.

Although caster, trail, and kingpin values only apply to wheels that can be steered, the plane passing through the three outboard pivot points should pass through the center of the contact patch to make the toe stiffness high. This minimizes the moment arm and reduces acceleration and braking forces acting through the suspension.

The rear roll centers should be in the same range as the front roll centers for the same reasons. The rear roll centers should be higher than the front roll centers to exaggerate the weight transfer across the rear tires and help to reduce under-steer.

The suspension hard points were determined based on our accepted target values, computer simulation, and rule and geometric constraints. Again, the goal is to produce a car that has predictable handling characteristics and maintains maximum grip on the track at all times.

The race tire is arguably the most important part of the suspension. Selecting the appropriate tire and designing the suspension to maximize the effectiveness of that tire is the key to a winning vehicle. We have selected the Hoosier 6 by 13 by 20.5 race slick for the competition tire. The Hoosier compound has the highest friction coefficient at the lowest loads compared with Goodyear and Avon. During cornering experiments, the Hoosier has the most predictable reactions to longitudinal forces and has a higher peak cornering stiffness than Goodyear. These characteristics make the Hoosier desirable over the others.

Now that a tire has been selected, a design that will maximize the ability of the tire is needed. Hoosier tires perform the best at small negative camber angles. During computer simulations using WinGeo, the hard points could be adjusted, leaving a camber of -.5 degrees in a 1.3 g hair-pin turn. This should give maximum grip from the tires on our tightest expected turns.

### *Pedals and Brakes*

The design flow of the brake system utilizes a “wheel to foot” direction. By analyzing the required forces at the brake rotor and caliper, the hydraulic components may then be selected such that the required driver input force is within the ergonomic spectrum. Previous Idaho FSAE teams have had difficulties with braking consistency, especially at cold temperatures. To ensure, consistent and reliable braking at all temperatures (rotors cold and at operating temperatures), a brake temperature analysis is in progress. Thermal sensitive paint is applied directly to the brake rotors. The car is then driven under typical race conditions. Based on the operating temperatures of the rotors, the thermal sensitive paint will change colors, indicating the operating temperature. Knowing the operating temperature will allow for the optimal brake pad selection and vastly improve braking consistency.

The braking system shown in Figure 9 provides a rising rate mechanical linkage. The kinematic prototyping of the linkage was performed in SAM 6.1, a newly acquired kinematic software package. The rising rate linkage allows for a progressive braking system. Additionally, due to the increase of mechanical advantage throughout the pedal stroke, the required driver input force is minimized, thus decreasing driver fatigue. The brake system prototype will be machined from aluminum and tested on the 2008-2009 FSAE competition car. Driver feedback on the rising rate system was acquired to validate the theoretical effectiveness of the system. The brake pedal, linkages, and pedal rails will be optimized with the aid of powerful genetic algorithms in modeFRONTIER.

SolidWorks has been used to start the initial modeling with force analysis being performed on the three concept pedal designs to simulate worst-case scenarios. As the process continues the Genesis program will be utilized in maximizing structural rigidity, while minimizing weight. To ensure ergonomic parameters are maintained for a variety of driver statures, adjustability will include approximately 3” of vertical adjustment and approximately 6” of fore/aft adjustment. The pedal box must also fit inside the front bulkhead and must keep the drivers feet inside the front bulkhead at all times.

The throttle pedal will use a variable throttle linkage, which will adjust the pedal stroke to throttle body rotation ratio. This will maximize performance, by allowing the driver to tune throttle response to his or her preference. To prevent overstressing the throttle system, a mechanical throttle stop will be present at the pedal to limit travel. At the throttle body, a minimum of two return springs will be present.

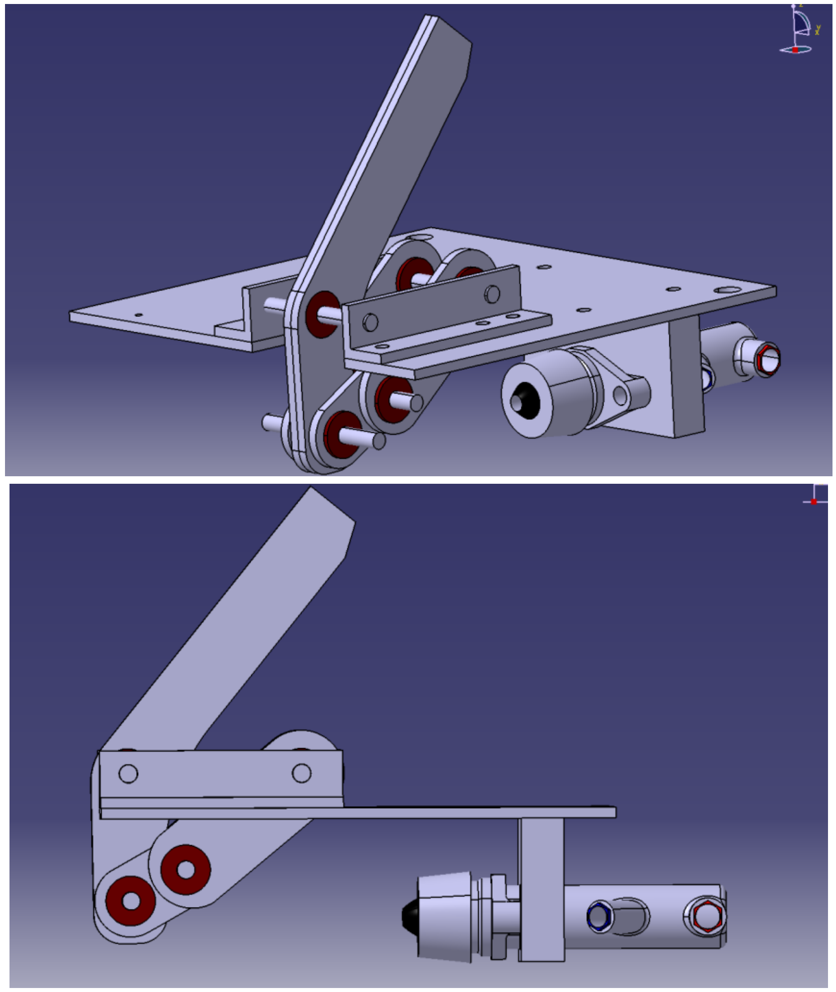

Figure 9: Pedal and brake design.

### *Frame and Chassis*

The switch from Formula to Formula Hybrid requires the design of a new frame. The frame shown in Figure 10 integrates a smaller hybrid power plant and incorporates the addition of a battery pack. The frame must comply with the rules of the competition and integrate the suspension and the placement of the new powertrain. It will also need to take into account the various components such as pedal placement, steering rack, cooling systems, etc. It must also be designed to meet our target values of a minimum torsional rigidity of 2400 lbf-ft/deg and bare weight of 60 lbs.

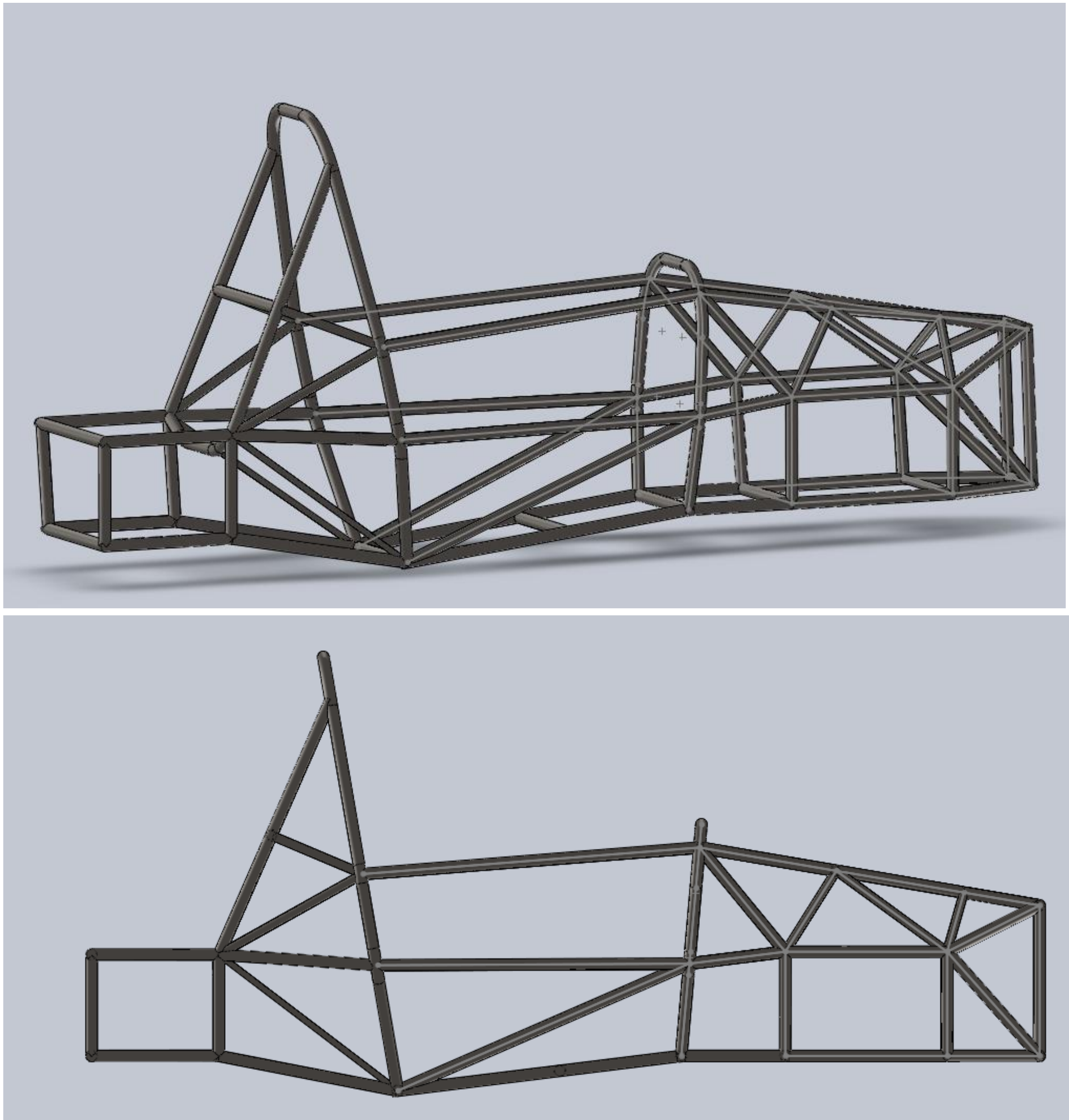

**Figure 10: Frame design.**

The design of the cockpit is mostly defined by the rules of the Hybrid competition. In addition to these rules it has been decided that the cockpit will sit at the minimum height of one inch from the ground when integrated with the suspension. The Hybrid competition has yet to implement the templates used in the Formula competition, but it is a strong belief from the Hybrid community that these templates will be implemented for the 2012 Hybrid competition. In addition to the templates fitting, it is required for a $95^{th}$ percentile male (6’2” and 220 lbs) to fit into the cockpit. The $95^{th}$ percentile male will also need to sit as comfortably as possible to prolong driver fatigue during the endurance race. This position was determined by an ergonomics study that can be found in the appendices.

The rear chassis will utilize the custom case being built for the YZ250F and electric motor as a structural member. By working closely with the powertrain team, optimized locations for the frame to directly connect into the engine case were selected. By doing this, weight was cut and the engine was placed as close to the back of the driver as possible. The placement of the suspension mounting points will also be based off of this, and so the frame will need to take this into consideration.

The front chassis is determined by the template that will run horizontally through the frame and by the placement of the engine. The rules require a minimum of a 60 inch wheelbase. The wheelbase will adhere to the 60 inch rule and the front a-arm mounting points will be determined by the placement of the differential. Unlike previous years, the small packaging of the powertrain allows to easily place the front suspension wherever we would like and chose our desired wheelbase of 60 inches.

As the frame draws closer to a fixed geometry it must be tested for torsional rigidity and adjusted to adhere to a minimum of 2200 lbf-ft/deg. This will be accomplished with either the FEA tools in CATIA or Msc Adams software, but before this can happen, the back end of the car will need to be closer to completion. Once initial testing has been done, the frame can be adjusted to maximize the rigidity while keeping the weight below 60 lbs. Testing and adjustments would then continue until both goals are met, ideally maximizing the rigidity and minimizing the weight.

### *Uprights*

The design process began by researching the upright designs used on other FSAE formula cars, formula one cars, passenger cars, atvs, and other similar vehicles. Formula FSAE uprights are of two basic types. One being machined out of a single piece of bulk material, generally aluminum, and the second being welded out of sheet metal, generally steel. As part of the conceptual design process, an upright of each type was designed.

Each of these designs were evaluated using the finite elemental analysis capabilities of SolidWorks. Using the previous year's upright as a standard, it was determined that the aluminum upright did not provide sufficient torsional rigidity. This was consistent with the experience past University of Idaho teams have had when using aluminum uprights, so we decided to continue with a sheet steel design.

Improving upon the previous year's sheet steel design proved to be very challenging. At this point, it was determined that if the design were going to changed, it would be a good time to consider other modifications like possibly finding lighter, lower resistance bearings.

With the help of Tyler Thornton at Timken Bearing, bearings were found that are half the weight of the bearings on last year's formula car and have 50 percent less friction and heat loss than the old bearings. In order to be able to use these bearings, the bearing housing bore on the uprights needed to be reduced. The bore reduction was performed on the solid mode of the old front upright and a FEA analysis was performed to determine the effects of the change. In this model, the safety factor dropped from 1.92 to 1.78. In one of the other conceptual design upright models, the change in bore diameter reduced the safety factor for 2.30 to 2.28. In both cases, the reduction in safety factor is reasonable, so there is no problem creating uprights that will work with the desired bearing changes. The remaining issue is to determine if the hubs can be successfully redesigned to match the new bearings. Currently ideas are being developed for repackaging the assembly to fit the desired bearing dimensions. The redesigned hubs will need to be analyzed to ensure they provide adequate strength. Ideas for repackaging will be presented, and upon recommendation, there will be a complete detailed design of the upright, hubs, and other parts in this assembly.

In order to find the geometry of the bell crank, a solid works model with the a-arms, push rod, bell crank, and points of attachments for the shock were created. With the model, it was possible to adjust the length of the push rod and each side of the bell crank. Using this model, the suspension was taken through a range of motions and calculated the wheel rates at every ½". It was assumed the suspension was using a spring rate of 170 lb/in. Figure 11 shows the results of this analysis. The horizontal lines are the forces of the car when braking, cornering, and combined all at 1.6 G.

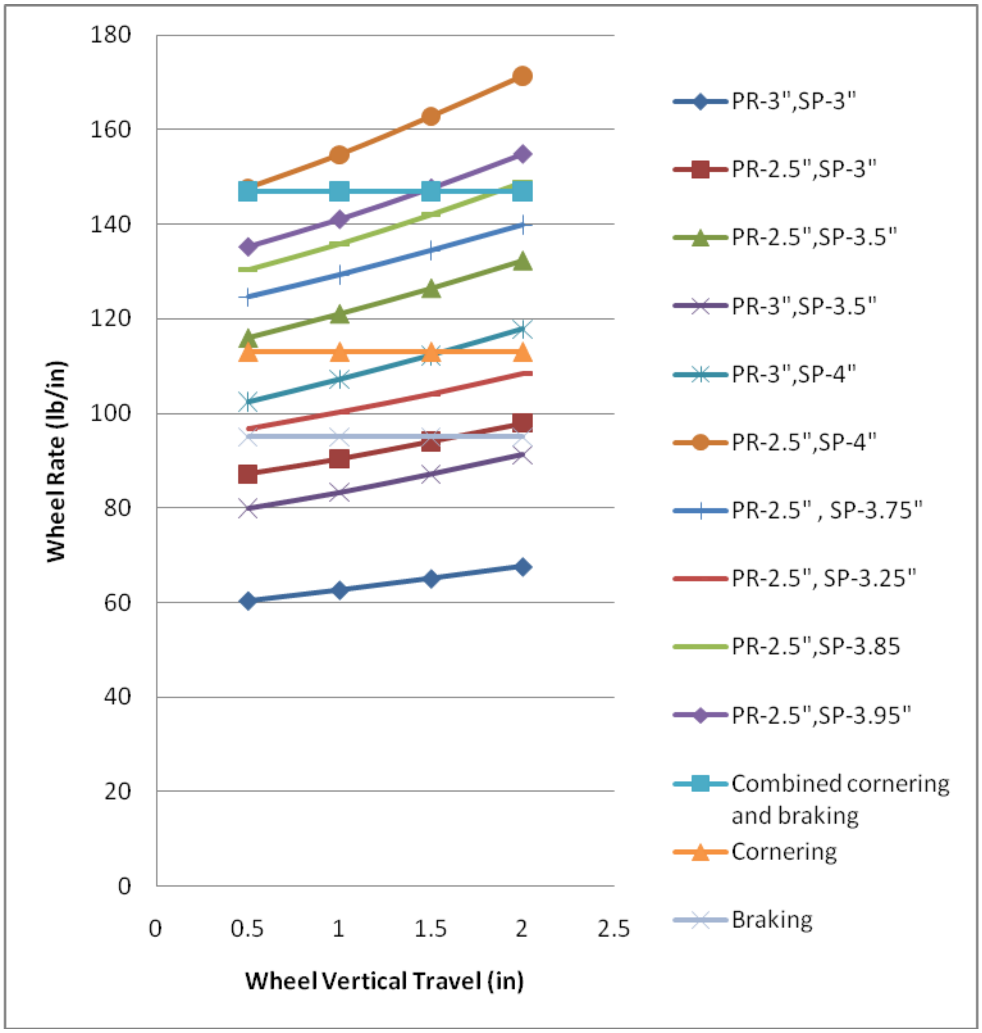


**Figure 11: Wheel rate vs. vertical travel.**

Only two styles of bell cranks were considered - the typical triangular shape used on previous cars and the L-shape. Initially it seemed that the triangular shape is better; however once looking into the loads that go into the bell crank, the triangular design is heavier than the L-shape.

The L- shaped bell crank that will be used in the front suspension of the FHSAE car utilizes an adjustment capability on the shock side of the bell crank, and there is a placement for the torsion bar. This design will be lightened when the program Genesis is used giving the part a webbed structure.

Figure 12 is an exploded view of the bell crank. The long rod will be welded though the frame member then the bell crank and bushing will be slid onto it. A ¼" screw and washers will be fastened into the rod holding the entire bell crank together.

**Figure 12: Bell crank design.**

### *Sensors and Controls*

Building upon the basic drive-by-wire architecture, it was ultimately intended to construct and test the performance of an electronic data acquisition and acceleration control system, also known as a 'throttle-by-wire' system. Electro-mechanical sensors will be placed on the accelerator pedal to capture driver commands. Other sensors will also be placed throughout the car to measure performance variables like speed, RPM, electric motor output, and acceleration. The control unit in this scheme is an electronic micro-controller, a computerized system that can be programmed to read the sensors and achieve the desired control. Figure 13 is a top-level system sketch that also shows potential future drive-by-wire systems, including brake-by-wire, steer-by-wire, and shift-by-wire.

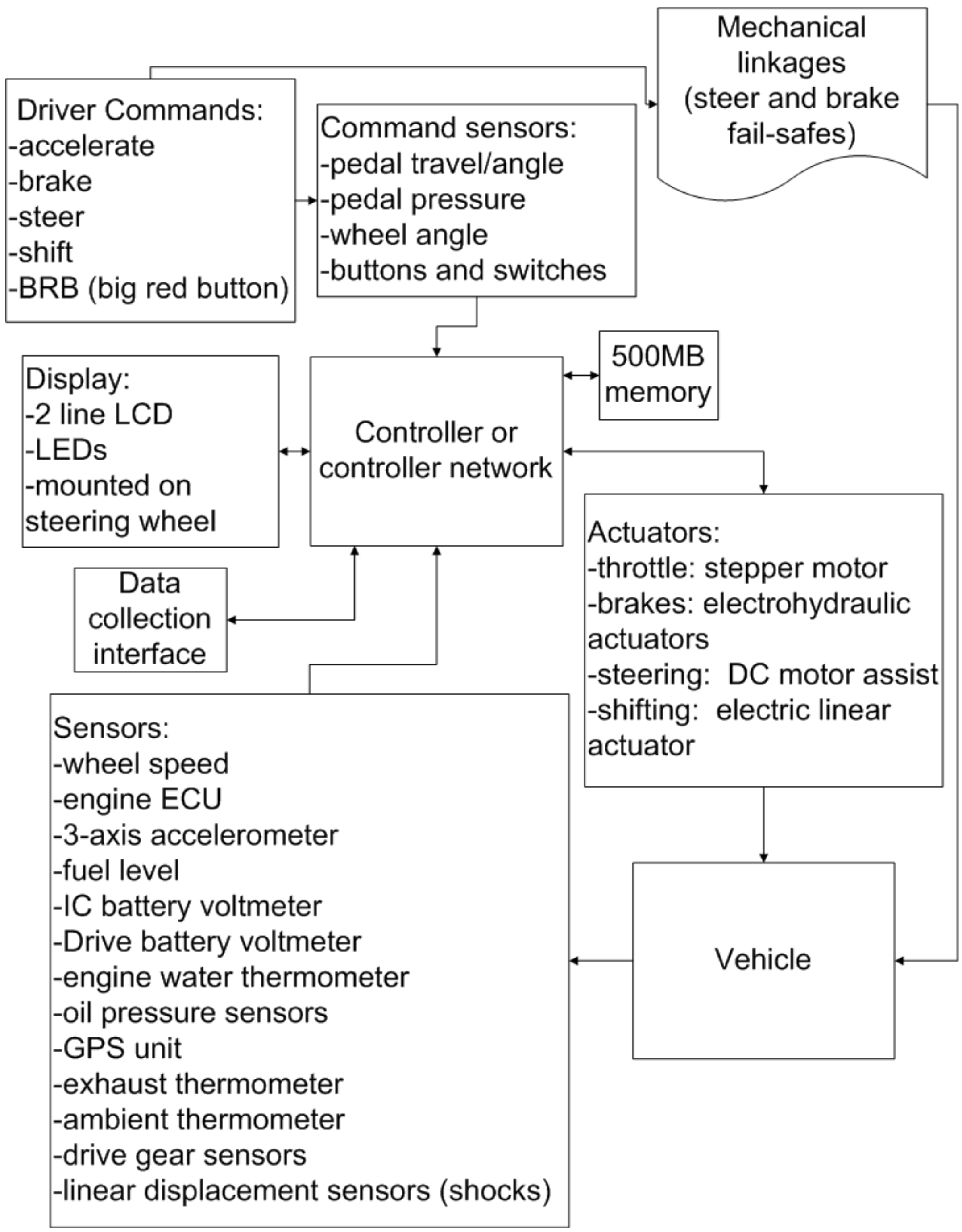


**Figure 13: Proposed sensor and controller network.**

# FINDINGS; CONCLUSIONS; RECOMMENDATIONS

## Road Load Model

At this time, the road load model is a functional model that will be continually updated and revised to improve the usefulness as well as the accuracy of the resulting data. Many of the values used in the calculations have been assumptions based upon previous FSAE knowledge. Initially these assumptions were placeholders to help the development of the model and to verify proper operation. As the model evolves, the values that were assumed or estimated are constantly being updated to measured and known values, which improve the accuracy of the results as well as providing a verification of model precision. As the values used in model computation are updated, the program itself will also be updated and expanded to include vehicle parameters and assumptions derived from roll-down testing as well as energy consumption forecasts that involve power electronics and batteries.

## Energy Management

In order to improve the model and to optimize future vehicles, a series of sensors and data logging equipment will be mounted on the 2011 vehicle. Information such as speed, position, memory, energy consumption, and torque will be stored with timestamps by a microprocessor. Besides the sensory equipment and related hardware, a display will be incorporated to inform the driver of the current linear speed of the vehicle, the amount of gasoline still available in the gas tank, and the approximate state of charge of the batteries.

Time permitting, a communication system may be included that transmits real time data to the pit crew for immediate review on a handheld device.

## Vehicle Integration

The following table contains the major components and design types selected for the UI Hybrid FSAE vehicle. The vehicle design is based upon standard rear wheel drive with front steering. Fabrication is underway and the latest state of subsystem realization can be seen on the capstone design website http://seniordesign.engr.uidaho.edu/2010-2011/hybridformula/.

Table 11: 2011 UI Formula Hybrid Vehicle Overview

| | |
|---|---|
| **Chassis** | **Welded Steel Frame** |
| Suspension | Fully independent unequal length A-Arm |
| Steering | Rack and pinion |
| Body | Carbon fiber composite |
| Tires | Hoosier racing |
| Wheels | Aluminum rim with aluminum centers |
| Brakes | Brembo calipers and custom steel rotors |
| **Gasoline Engine** | **Yamaha YZ250F** |
| Starting system | Yamaha WR250F starter assembly |
| Fuel delivery | Throttle body fuel injection |
| Peak power output | 33 hp @ 12000 rpm |
| Peak torque | 18 ft-lb @9000 rpm |
| **Transmission** | **4 speed constant mesh w/electronic shifting** |
| Clutch | Rekluse Motorsports Z-Start Pro |
| Gearing | Stock WR250F gear ratios |
| Final drive | Planetary limited slip differential |
| **Electric Motor** | **Lynch LEM200-135 RAG** |
| Peak Power output | 46 hp |
| Peak torque | 61 ft-lb |
| Continuous torque | 29 ft-lb |
| Motor controller | Kelly KDH09401A |
| **Electrical System** | **Parallel hybrid configuration** |
| Batteries | Impact |
| Nominal voltage | 48 Volts |
| Maximum current | 400 Amps |
| Capacity | 40 Amp-hours |

## Design Infrastructure for Next Generation Vehicles

A program is under development that combines the evolutionary design capabilities of ESOP (Evolutionary Structural Optimization Program) with the geometric analysis functions of WinGeo, developed by Bill Mitchell. This new program, VSOP (Vehicle Suspension

Optimization Program), optimizes the node locations of a double a-arm suspension system to give the best fit to a set of desired suspension characteristics. Desired values of camber, roll center height, roll center width, caster, and VSAL (virtual swing arm length) are input by the user, as are constant parameters such as wheelbase and track length. The program then finds the suspension configuration that comes closest to satisfying all parameters. Because suspension parameters often offer conflicting performance curves, a Pareto surface is developed, over which the global optimum, or Pareto point, is located. This optimum is then output as the program reaches convergence. This program is being used along with Msc Adams/car dynamic simulation software to ensure that the FHSAE vehicle exhibits superior suspension performance for given track specifications.

To create a complete, stable, and functional assembly model of the FHSAE vehicle, it is necessary to use a high-end modeling tool such as CATIA. Crafting machine elements such as fully-involute gears in the 3D modeling environment is a very difficult task because there are many parameters and geometrical constructions involved that are unique to gear manufacturing. Unfortunately, there is currently no software package available that can take standardized gear parameters as inputs and have CATIA make an accurate model. Such an application is under development using Visual Basic 6. Five different gear models will be supported by this effort: spur, helical, bevel, worm, and internal. A CATIA-produced macro that contains all of the information necessary to develop the specific gear has been created and is currently being manipulated to fit into the Visual Basic scripting language. Finally, a graphical user interface is being made to allow end-users to input their available parameters and the software will ascertain whether or not sufficient information exists to make the part in CATIA. Provided there are no input errors, the program communicates with CATIA and executes the macro, creating the true-to-life part that can then be used in assembly modeling, simulation, and manufacturing.

# APPENDIX A – FHSAE ERGONOMIC ANALYSIS

## Brake Pedal Design

A. 95% Male

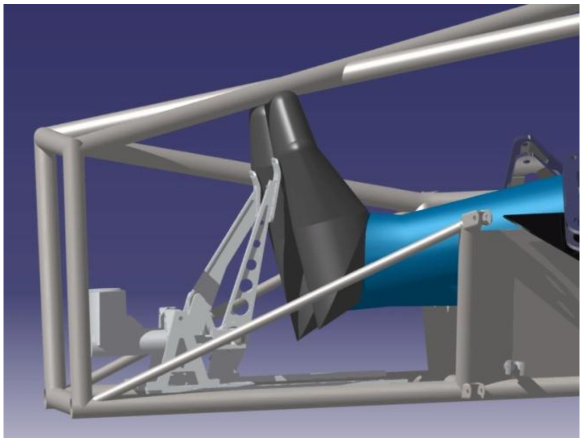

Figure 1A: This shows the 95% male manikin with his feet on the pedals, with the pedals in a neutral position.

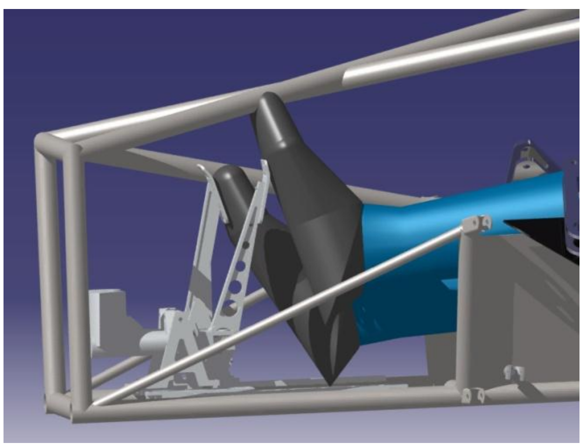

Figure 2A: This shows the 95% male manikin with his feet on the pedals, with the pedals fully depressed.

As you can see in Figure 1A, for the driver to have his feet in a position to maximize force, a foot rest needs to be added to the floor of the car. Also, for this driver to have his feet in an optimum position, the upper roll-cage bars should be raised to allow for clearance between the roll-cage and his feet, this is very apparent in Figure 2A where the driver has the pedals fully depressed. Another change that could be helpful would be to shorten the throw of the gas pedal to allow the driver to apply pressure to the pedal closer to the ball of his foot, rather than his toes.

B. 5% Female

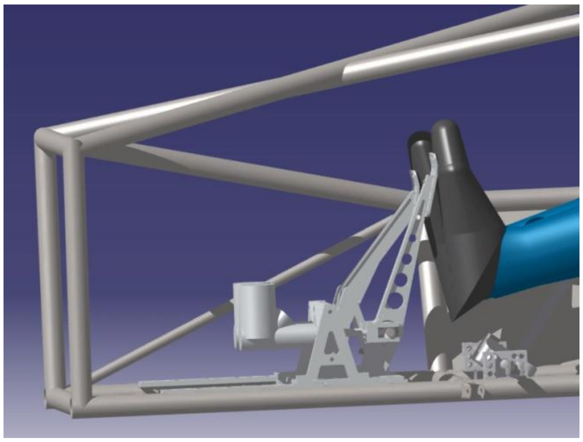

Figure 3A: This shows the 5% female manikin with her feet on the pedals, with the pedals in a neutral position.

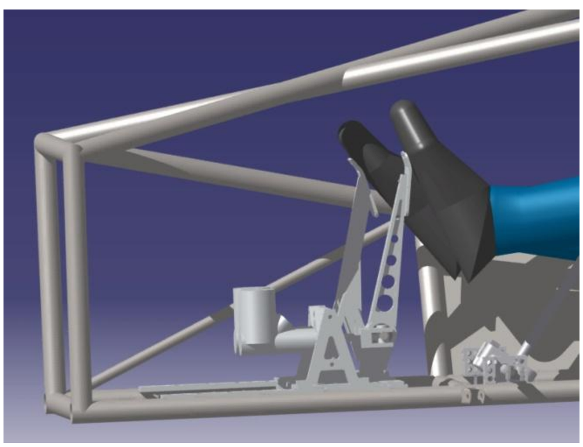

Figure 4A: This shows the 5% female manikin with her feet on the pedals, with the pedals fully depressed.

For a 5% female driver, a large foot rest needs to be added to the floor, because as seen in Figure 4A, there is a great deal of space between her feet and the floor of the car. Figure 4A is also another good example for reducing the throw of the gas pedal, as the 5% female driver can barely reach to depress the pedal all the way. Another recommendation for fitting this small driver in the car would be increasing the length of the pedal adjustment track, in order to bring the pedals closer to the driver.

C. 50% Male

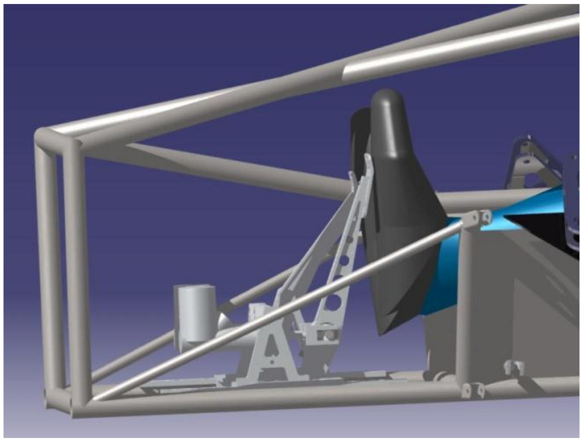

Figure 5A: This shows the 50% male manikin with his feet on the pedals, with the pedals in a neutral position.

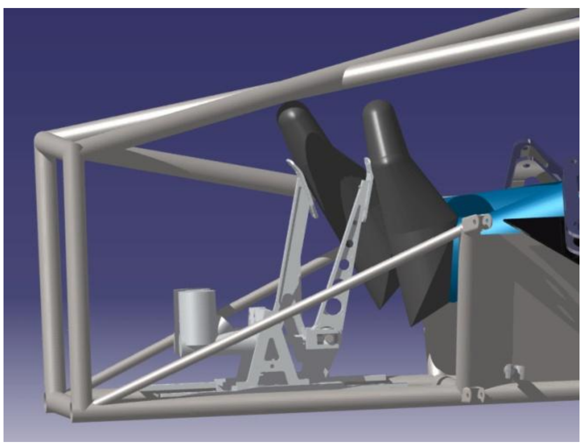

Figure 6A: This shows the 50% male manikin with his feet on the pedals, with the pedals fully depressed.

This driver also would require a foot rest to raise his feet to an optimal position, as well as provide leverage when pressing on the pedals. The clearance between this driver’s right foot and the top of the roll-cage is minimal at best, and it would benefit the driver if the roll-cage height was increased, or the length of the pedals were decreased.

## Steering Design

A. 95% Male

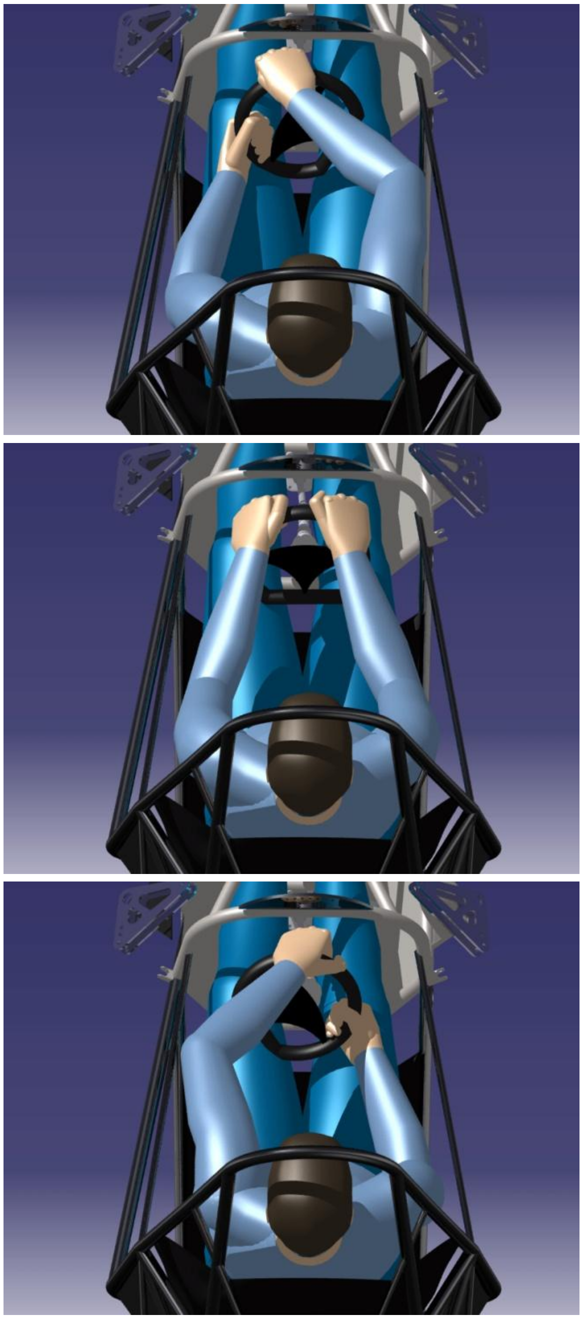

Figure 7A: This shows the 95% male starting at a 60° left turn and moving to a 60° right turn.

No interference was detected while the 95% male was turning the wheel, but the fit of the driver in the roll-cage appears cramped. For a more comfortable fit, the roll-cage should be widened slightly.

B. 5% Female

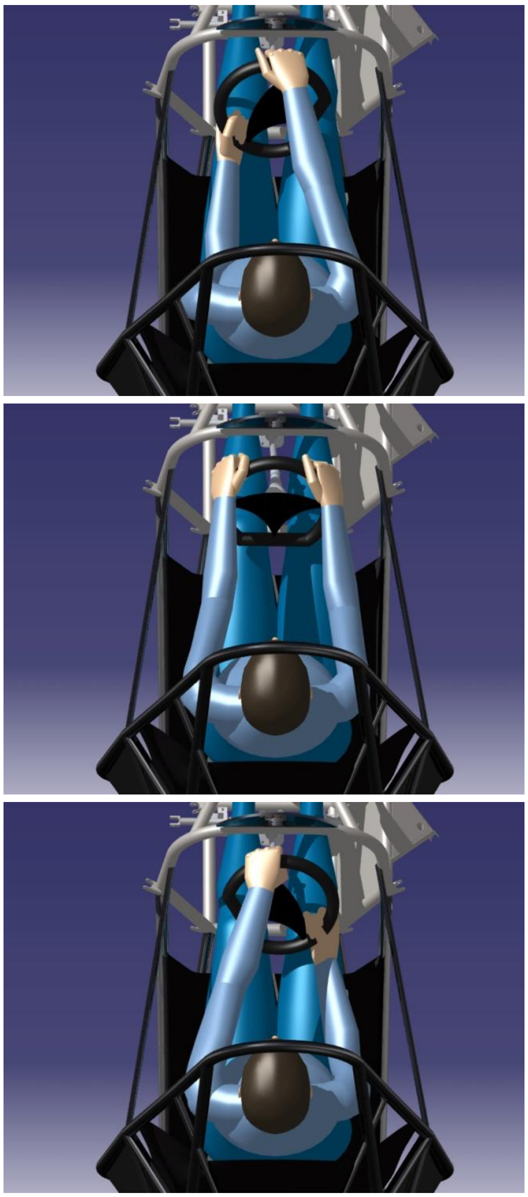

Figure 8A: This shows the 5% female starting at a 60° left turn and moving to a 60° right turn.

No interference is detected between the roll-cage and this driver. The only concern with this driver's ability to steer the car is her reach to the steering wheel. She needs to be forward from the seat approximately 3.5" in order to reach the wheel or pedals, although she can reach the wheel, her arms are at nearly full extension and she would have little power to turn the car. We recommend that she be moved forward an additional 4” in order to maximize turning power, and also put her in a better position to work the pedals. We suggest creating a foam pad to fill this gap.

C. 50% Male

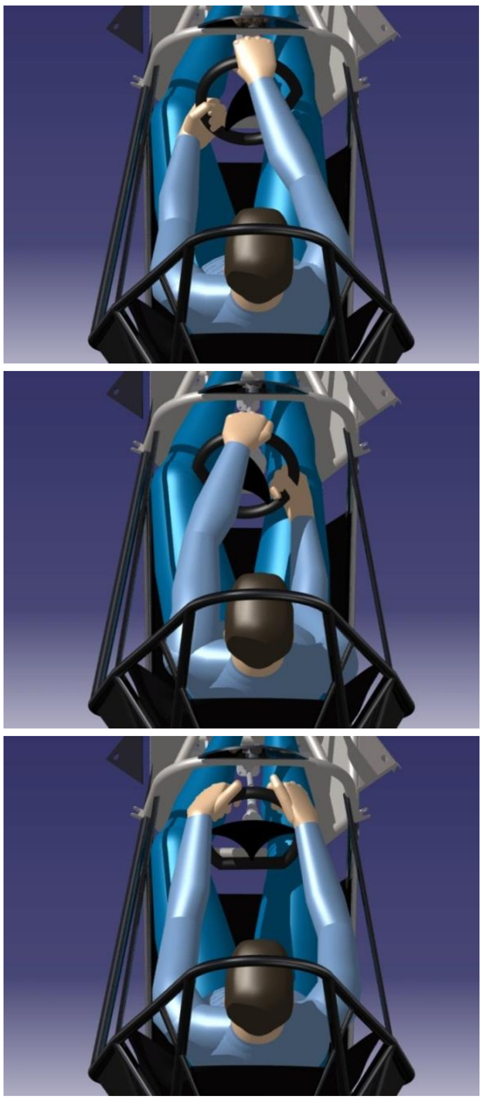

Figure 9A: This shows the 50% male starting at a 60$^{\circ}$ left turn and moving to a 60$^{\circ}$ right turn.

This driver fits very well into the car, the only suggestion that we have to make is possibly creating a foam pad, similar to the 5% female’s that would move him off of the current seat-back 2”-3”.