MEMO

TO: Andy Dethlefs
FROM: NAU Efficient Vehicle
SUBJECT: Final Design Report
DATE: 2, May 2008

Our team would like to thank you, Andy Dethlefs for participating in our 2007-2008 NAU senior design capstone project. The document enclosed is our final design report that explains all of the aspects of the Northern Arizona University Efficient Vehicle. Please review the document and return to Dr. Tester via email.

It was a pleasure to have you attend the capstone conference and we hope our performance was a good showing. We were happy to present the final product to you and have your first response approval. As discussed at the capstone conference, our team will be completing a number of design alterations before the SAE Supermileage competition. The drive train will be completely changed and the components of this have already been purchased from Mcmaster and Comet Kart Components. These parts should arrive within the next week or so. Once the parts arrive the team will be working in the shop to install them and begin testing.

This year long experience has been very rewarding for our team. We have learned how the practical applications of engineering can be applied to automotive energy solutions to obtain better fuel economy. Many of the ideas and design our team has implemented onto the vehicle could be used in the automotive industry to increase the fuel economy of passenger vehicles.

NAU Efficient Vehicle Team

Northern Arizona University Efficient Vehicle

Final Design Report Rev 3

ME 486C-Spring 2008

Dr. John Tester

May 2, 2008

Mike Barton
Karl Busalacchi
Tanya Gallagher
Sun Mahasuverachai
Shane Stoterau

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Executive Summary

The Society of Automotive Engineers (SAE) holds a competition, known as the SAE Supermileage, once a year to produce a high efficiency vehicle. Our senior design team has chosen to compete in the SAE competition as our Senior Capstone Project. The goal for the 2008 seniors is to compete in the SAE Supermileage competition and in the Shell Eco-Marathon Americas. The stipulated goal of this project is to produce a high efficient vehicle that can compete in two distinct competitions vehicle. Shell Eco-Marathon Americas will be held on April 10-13, 2008 at the California Motor Speedway in Fontana, California. The SAE Supermileage competition will be held on June 5-6, 2008 in Marshall, Michigan. Mike Barotn, Karl Busalacchi, Shane Stoterau, Tanya Gallagher and Sun Mahasuverchai, will design and construct a complete, competition ready vehicle. Most of the competition rules are very comparable making one vehicle feasible for both events. Senior Capstone is a two-part course lasting for one academic year making time scheduling imperative to the construction of a competitive vehicle for both events. During the Fall Semester the vehicle was completely designed and the initial phases of fabrication began. The Spring Semester was primarily used to complete the fabrication of the vehicle.

Scope of Problem

The scope of the problem is to design, analyze, and construct a single-person, highly fuel-efficient vehicle using a single cylinder, four-cycle engine, while abiding by the safety rules set forth by SAE Supermileage and Shell Eco-Marathon Americas. The vehicle will be built with a very limited budget provided by SAE@NAU and acquired sponsors and must be completed by April 9, 2008 in order to compete in Shell Eco-Marathon and June 5, 2008 to compete in SAE Supermileage. A Faculty Advisor must also agree to attend each competition in order for our team to register at the competitions and compete. To accomplish remarkable fuel-efficiency, the vehicle will be as light-weight as possible, have very low aerodynamic drag, low rolling resistance, increased engine efficiency, low friction drive train components, and will maintain all specified safety standards. The following items and systems have been designed, analyzed, and built prior to the Shell competition:

- Frame
- Fuel management system
- Fairing
- Drive train
- Engine modifications
- Driver seating
- Steering system
- Braking system
- Electrical system

State-of-the-Art Research

Fuel Management

Research began by identifying the benefits and shortcomings of a fuel injection system compared to that of a carburetion system to monitor the air-fuel ratio. The optimal air-fuel ratio for an Otto Cycle is approximately 14.7:1. Both fuel management systems are capable of monitoring and adjusting the air-fuel ratio.

- Fuel Injection
  - Advantages
    - The fuel injection system adjusts in real-time.
    - Air fuel ratios will stay very close to ideal.
  - Disadvantages
    - These fuel injection systems had many parts to assemble, increasing time to assemble as well as reducing reliability of the system. (Tripod: Jbabs Fuel Injection 18).
    - This fuel injection system also required that we have another power source to energize and control the air-fuel mixture, thus increasing our overall vehicle weight and taking away from engine output.
    - Another drawback to the fuel injection system is that the complexity of the change from carburetion to fuel injection is another huge commitment by the team and could be considered one of the major adjustments from last year's vehicle (Tripod: Jbabs Fuel Injection 18). This could possibly be an adjustment for teams to consider for future NAU SAE Supermileage and Shell Eco-Marathon Americas competitions.

- Carburetion
  - Advantages
    - The research conducted discovered that it is possible to adapt a carburetor from another smaller engine and provide the optimal air-fuel ratio while significantly reducing the amount of fuel consumed in the process.
    - If the process is controlled and monitored by a wide band oxygen sensor it will allow the group to still maintain the optimal air-fuel ratio which will reduce fuel consumption.
    - Constant velocity carburetors are more efficient than compared to slide carburetor. The varying inlet diameter and fuel passages maintain a precise air fuel ratio at all engine speeds. (http://www3.telus.net/ 20)
  - Disadvantages
    - The carburetion system needs to be either adjusted by loosening or tightening bleed valves and adjustment screws that affect the amount of air and fuel allowed into the cylinder.

Frame

The team did considerable research looking for any space-age, lightweight materials that could possibly be used for the frame of our vehicle to reduce the overall weight. The material we sought must have a significant reduction in weight while still possessing high rigidity and functionality. The material must be also capable of being attached to the wheel supports, engine mounts, etc. as well as being capable of accepting and sustaining dynamic vibration displacements. For these reasons, the team decided to explore composite sandwich structures, methods of fixturing to the panel, and if they would be suitable to satisfy our weight reduction requirements.

- Composite sandwich structures
    - Advantages
        - Researched was performed with several different companies that provided technical data for composite sandwich structures (http://www.mcgillcorp.com/ 10). Aluminum honeycomb cores provided the group with a significant weight reduction and only a slight reduction in the overall vehicle frame rigidity compared with the alternative.
    - Disadvantages
        - Mounting to these panels is rather difficult, for aluminum welding could destroy and not adhere to the face sheets. This lead to researching mounting methods.
        -
- Adhesives
    - Advantages
        - Adhesives have developed significantly within the last decade and are drastically stronger with an associated increased reliability. There is a significant weight reduction when comparing the aluminum adhesives to conventional aluminum welds. Another benefit to using the adhesives is that there exists a leftover amount to use for the threaded inserts needed to mount to the frame, decreasing costs and time committed to this sub-system.
    - Disadvantages
        - Curing times for certain adhesives is rather extensive, increasing fabrication times.

- Potted-In Inserts
    - Advantages
        - Based on the information provided by Shur-Lok Corporation, we found data and charts pertaining to the shear strength (www.shurlok.com).
        - The information stated that under our initial load considerations, the mounting methods would work.
    - Disadvantages

- Avialable inserts do not secure to bottom face sheeting on the honeycomb panel. This could lead to a failure of the upper face sheet.

Fairing Design and Fabrication

- Outsourcing possible fairing production
  - Advantages
    - Through the production of a foam male mandrel through Foam Plastic Specialties Inc. in Tempe, AZ, we were planning on finding a company willing to form a polycarbonate set of sheets to create our body. Since the outsourcing of the fairing was not capable, we set up sponsorship with Quintus Inc. in Camp Verde, AZ for they are willing to make and donate a fairing of a hybrid fiberglass/graphite composite as long as the team provides them with a female model of our fairing (http://www.Quintusinc.com/ 12).
  - Disadvantages
    - Many of the companies contacted were capable of plastic forming using methods varying from thermoforming, vacuum forming, vacuum bagging, foam lay-up, injection molding, and etc. yet none of the them were capable of creating our body given our massive dimensions of the model.
- Three part mold making process
  - Much of our research was dedicated to learning how a positive to negative to positive composite part was to be made. The team worked with Quintus Inc. to figure out much of how this process takes place. The first step is to create a male mandrel using some sort of foam with a top layer of fiberglass and non-foam-dissolving resin with subsequent layers of lightweight body filler that is formed by hand manipulation and sanding. From this male mandrel, a female mandrel is created using a releasing agent and successive layers of fiberglass and resin. Using the female mandrel with a vacuum bagging process, fiberglass and resin layers, and squeegees, a final part can be constructed (12).
  - Disadvantages
    - The creation of a male mandrel, female mandrel, and part is significantly time consuming; due to prior group experience.

Engine Modifications

The supplied 3.5 HP Briggs & Stratton Engine is not very efficient. The motor is designed for long life and longevity and not necessarily designed with performance in mind. Some of the following factors causing the inefficiency are due to: relatively low compression ratio of the engine, non-existence monitoring of the air-fuel ratio injected into the engine during intake, internal friction loses within the load bearing surfaces, and many other sources of mechanical losses. The focus hinges on several modifications and whether or not they can potentially help in reducing engine losses.

- Overhead valve conversion.

- Advantages
  - For the valve train on the engine there were many parts that needed to be researched and located. For the overhead head valve (OHV) conversion from the initial L-head configuration, the group needed a new lifter, pushrod, valve, spring assembly and most important a new cylinder head.
  - The stock L-style system expels gases through side ports of the chamber, while the OHV system expels the gases directly above the combustion chamber, reducing the internal friction inherent in the L-style system. Changing the head will also reduce the amount of surface area in the combustion chamber creating less heat loss from the combustion chamber.
  - From the equation$\eta_{th} = \frac{W_{out}}{Q_{in}}$. By changing the location of intake and exhaust valve it would allow for a more direct air flow path of the gasses. This would allow for increased volumetric efficiency. Volumetric efficiency is the ratio of gasses entering the combustion chamber compared to the displacement of the engine. (Fundamentals of Engineering Thermodynamics Fifth Edition 11)
- Disadvantages
  - This conversion will take numerous parts and massive amounts of custom fabrication to ensure that everything is in synchronization.

- Overhead cam conversion
  - Advantages
    - The overhead valve conversion provides the team the opportunity of relocating the camshaft to the overhead position. This combination will help reduce mechanical frictional losses in the engine.
  - Disadvantages
    - This conversion will take numerous parts and massive amounts of custom fabrication to ensure that everything is in synchronization.

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Requirements and Specifications

Functional Requirements and Specifications

Northern Arizona University's Efficient Vehicle will have to comply with multiple specifications from both the SAE Supermileage competition and the Shell Eco-Marathon competition. These specifications are given in each of the respective competitions rules and are explicitly stated with safety in mind. It is important that the team take all of these rules into account before any actual design takes place. The completed vehicle will be able to pass both design inspections. For specifications that overlap with one another the team will design to accommodate the more stringent design parameter. The section below gives a breakdown of each substem followed by a detailed outline of each requirement. Some quotations and paraphrasing have been used from SAE Supermileage 2008 Rules and Shell Eco-marathon Americas Official Rules.

- Roll Bar and Impact Resistance
- Dimensions and Stability
- Steering and Maneuverability
- Braking
- Engine Modifications
- Visibility
- Electrical Systems
- Fire Wall
- Egress/Ingress
- Minimum and Maximum Speed Requirement

Roll Bar and Impact Resistance

A roll protection device is required and must be made of "substantial" material. The device must extend a minimum of 5cm (2 inches) above the tallest driver's helmet. Also in the driving position, some portion of the driver's helmet must be within 10cm (4 inches) of the device. The roll bar must extend in width beyond the shoulders of the authorized drivers. Requirements are shown in Figure 3. Participants shall ensure that the vehicle shell and/or chassis are structurally solid. This roll bar must be capable of sustaining a 114kg (250 lb) static force applied from all directions without bending/buckling. Moreover, all sides of the cockpit shall be sufficient to protect the driver from possible lateral and frontal shock forces.

A 5cm-thick layer of polyurethane foam with a minimum density of 28kg/m^3 shall be placed on the inside wall of the front of the vehicle body in order to protect the driver's feet in the event of a collision.

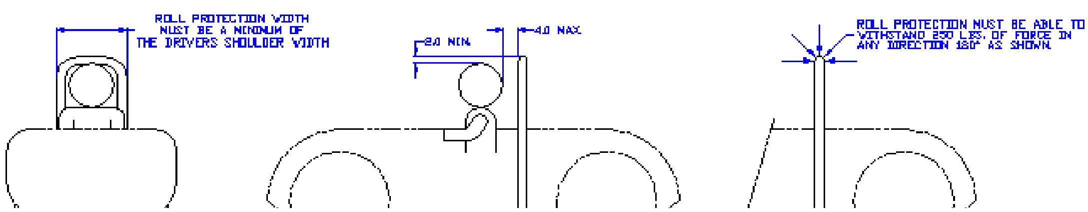


Figure 1: Roll bar requirements

SAE Supermileage 2008 Rules pg. 17

Dimensions and Stability

The maximum height of the vehicle will be measured at the top of the driver's compartment and shall be less than 1.25 times the maximum width of the vehicle between the two outermost wheels. The width between the tires of the vehicle shall be at least 50cm, measured between the midpoints where the tires contact the ground, and at most 110cm. The wheelbase shall be at least 1m. Maximum total vehicle width shall be 130cm, maximum total length shall be 350cm and the maximum vehicle weight, without the driver, shall not exceed 160kg. These dimensions are intended to ensuring sufficient stability, given the circuit layout. Each vehicle will be required to demonstrate its lateral stability. The vehicle, with the qualified driver, must maintain full wheel contact with a ramp of 20 degrees (measured from horizontal) when located statically on the ramp to the following configuration: one front wheel and one rear wheel of the vehicle must contact a horizontal line on the ramp with the vehicle in full right and left turn configuration. No supporting structure or wheel contact is permitted on the ramp below the horizontal line. Vehicle stability will also be evaluated during technical inspection using the slalom part of the maneuverability course described in Figure 4 below.

Steering and Maneuverability

Each vehicle must have steering geometry capable of a 15.2 m (~50 feet) maximum inside turning radius. Vehicle maneuverability will be evaluated during technical inspection using the maneuverability course described in Figure 2. Vehicle must traverse 30.5 meters (100 feet) slalom section in less than 15 seconds. Pylon spacing will be 7.6 meters (~25 feet). Our team will also be including a factor of safety for ease that will make our maximum turning radius 7.6 meters.

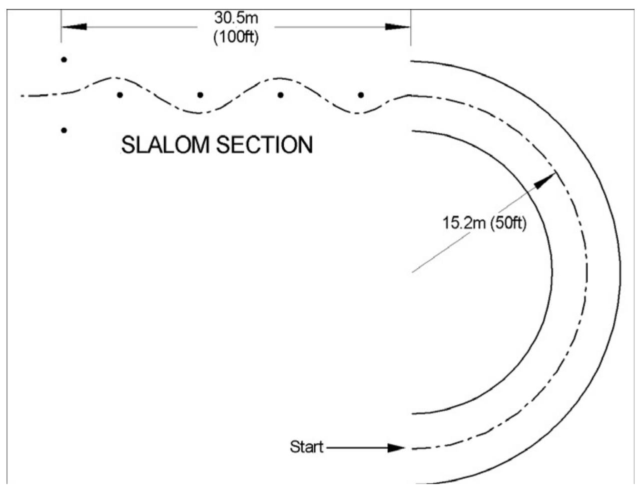


Figure 2: Maximum turning radius and Slalom section

SAE Supermileage 2008 Rules pg. 9

Braking

Vehicles shall be equipped with two independently activated brakes or braking devices, each including command control, command transmission (cable or hydraulic hose) or an activator (caliper or shoe). These two devices may act on one wheel or on one disk. If braking is done on two wheels, the right and left brakes shall be properly balanced. It shall be possible to activate the two systems at the same time without losing control of vehicle steering. The commands shall be perfectly ergonomic (no contortion shall be allowed for activation of such commands). Effectiveness of the two braking devices shall be tested during vehicle inspection. The vehicle shall be placed on an incline with a 20 percent slope. Brakes shall be activated each in turn. In both cases, the vehicle shall remain perfectly immobile. The use of a hydraulically controlled braking system is recommended. If a bicycle-type brake shoe system is used, only the V-Brake system shall be authorized. Mounting of the brake actuator must be on the interior of the vehicle and may not be on anybody panel. The driver must have access to the brake actuator at all times. The brake system must be capable of decelerating the vehicle from 24 kph (15 mph) at a rate greater than 0.25 g's (gravity). Brake system performance will be evaluated at Technical Inspection using the course shown in Figure 3.

a) Acceleration zone: There will be a minimum of 50m (~164 feet) available for the vehicle to accelerate to a minimum speed of 16 kph (10mph).

b) Coast zone: The vehicle must traverse the coast zone in less than 1.5 seconds.

c) Brake zone: The vehicle must come to a complete stop within the brake zone.

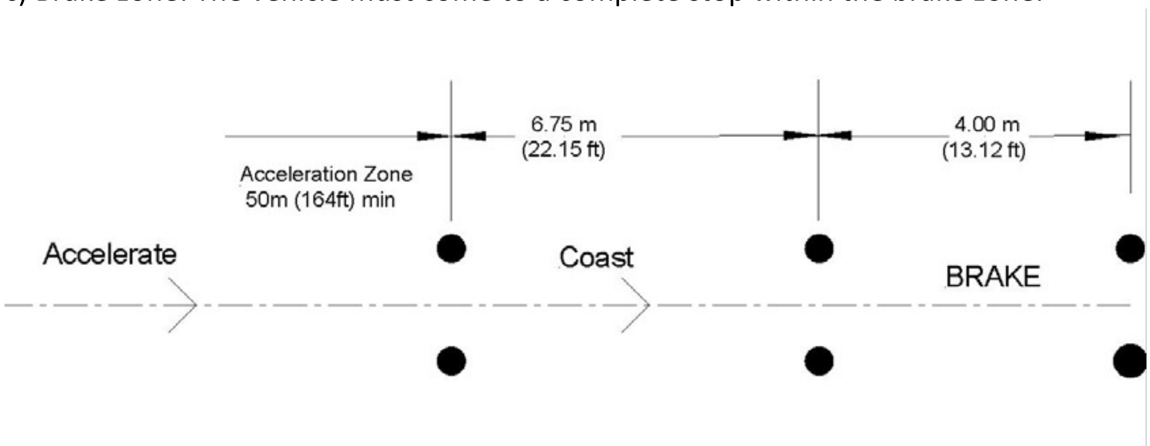


Figure 3: Braking test

SAE Supermileage 2008 Rules pg. 15

Engine Modifications

All vehicles must use the same base engine supplied to each entrant by Briggs & Stratton Corporation (Model 091202 Type1016E1A1001 or similar). The engine is air cooled, four cycle, with a 2.61 kW (3.5 horsepower) rating at 3600 rpm. Changes to the base engine, that may be desired for improved efficiency, are acceptable but must satisfy the requirements stated below. The cylinder and crankcase must be identifiable as components of the base engine supplied by Briggs & Stratton to the entrant. Crankshaft support bearings (journals) may be altered or replaced but must be retained solely by the Briggs & Stratton crankcase. The piston friction surface may be altered or replaced but must be retained solely by the Briggs & Stratton crankcase. Crankcase cannot be ventilated to engine intake air. Engine exhaust must exit the body (if so equipped). Mufflers are not required. Exhaust pipe must be insulated or guarded to reduce the risk of burns. The exhaust pipe must extend a minimum of 25 mm (~1.0 inch) beyond the outer body.

Visibility

The Driver must have adequate direct visibility in front and on each side of the vehicle and be able to turn his or her head 90° on each side of the longitudinal axis of the vehicle. This field of vision shall be achieved without aid of any optical devices such as mirrors, prisms, periscopes, etc. Moreover, the vehicle shall be equipped with a side view mirror on each side of the vehicle, each with a minimum surface area of $25cm^2$. The visibility provided by these mirrors, and their proper attachment, shall be subject to inspection.

Visibility in each of the vehicles shall be checked by an Inspector sitting in the driver's seat in order to assess on-track safety. This Inspector shall check for good visibility with seven 60cm high blocks spread out every 30° in a half-circle, with a 5m radius in front of the vehicle. Note that the Driver must be able to move his/her head in order to see any "blind spots". All the windows should be covered with a safety film on the inside of the windows to prevent sharp splinters from injuring the driver.

Electrical Systems

All other electrical items (fuel pumps, injectors, ignition, instrumentation, etc) must use a 12V battery with a C20 rated capacity no larger than 1.4Ah. An engine driven generator may be required to keep the battery charged if power consumption is high. All electrical connections to any batteries MUST be fused with an appropriately rated fuse. Team communication, stopwatches, bicycle computers, or similar devices that have self -contained battery sources are permissible and are not governed by the above battery restrictions.

Fire Wall

A wall of steel or aluminum 0.813 mm (0.032 inches) minimum thickness must completely separate the operator from the engine compartment. Furthermore, the firewall must not interfere with the operation and use of the on-board fire extinguisher. The firewall must extend to top of driver's helmet. No openings larger than 13mm (0.5 inch) in diameter will be permitted in the firewall. This includes gaps between the firewall and body.

Egress/Ingress

The driver must be able to exit the vehicle within 15 seconds, unassisted, in case of an emergency. A maximum of two support personnel must also be able to quickly extract a driver from a vehicle without assistance from the driver within 20 seconds. Exit ability will be tested during tech inspection.

Minimum and Maximum Speed Requirement

A performance run will consist of each vehicle running six laps around a 2.6 km (1.6 mile) oval test track. The vehicle must achieve a minimum six lap average speed of 24 kph (15 mph). This means each vehicle will be required to travel a total distance of 15.5 km (9.6 miles) in a maximum of 38.4 minutes. The vehicle must not exceed a single lap average speed of 40 kph (25 mph). This means a vehicle must take longer than 3 minutes 50 seconds to complete each lap. Vehicles must be capable of ascending a 1 percent grade and descending a 7 percent grade.

Design Features

This project was accomplished by adapting off-the-shelf parts and materials to fit the needs of the vehicle. Almost every part on the vehicle was purchased from a supplier and was used "as is" or was modified to fit the necessary application. Several specialized parts were needed to be manufactured by the team because they are essentially impossible to procure.

This vehicle was designed around several potential drivers. The frame was constructed using the lightest possible material that can support the required loads. The vehicle will travel on the lightest tires with the lowest rolling resistance. The engine will drive the vehicle by turning one of the wheels using a light weight belt connected to gears on the wheel and the output shaft of the engine. The vehicle brakes are light-weight and have sufficient braking power. Our steering system for this vehicle needs to be extremely reliable, even if it contributes more weight than is necessary. The fairing was designed and built by the team because a custom fairing built by an outside company would significantly impact our funding for the rest of the project. The circuitry was also designed and built by the team because it is a relatively simple system and obtaining outside help would be a frivolous expense.

Nonfunctional Requirements and Specifications

In addition to the rules set forth by both competitions, the team has created guild lines and solutions to solve them. The guild lines are limited by: fabrication feasibility, ease of fabrication, facilities, scheduling, and funding. The general goals the team aims to meet in order to be competitive at the competitions and to complete the vehicle are listed below. These goals will be completed using the ideas and concepts that we have found in our State-of-the-Art research.

- Lightweight chassis
- Aerodynamic body
- Low rolling resistance
- Engine management
- Increase engine efficiency
- Simple fabrication
- Pass technical inspections

Design Features

Engine

Using the supplied engine from SAE, a Briggs and Stratton 3.5 hp 4 stroke engine that is shown in figure 4. Changing the head could also change the volume of the combustion chamber; as a result the compression ratio would change. For the competition the vehicle is required to use 100-octane gas. This requirement is the limiting factor on the highest compression ratio we can use. Our group has decided to run at 9.5:1 compression ratio to reduce the chance of pre-ignition/engine knock that would damage our engine and dramatically decrease efficiency. Below in figure 5 is a curve that shows a ratio of combustion ratio to fuel efficiency.

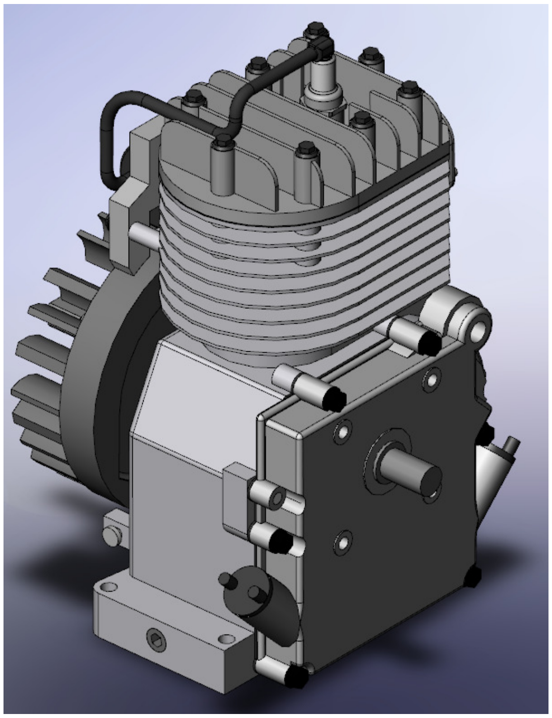


Figure 4: Stock Briggs and Stratton motor

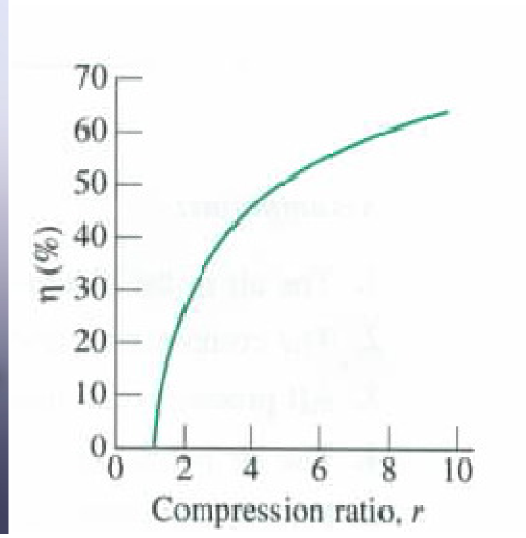


Figure 5: Compression ratio to thermal efficiency (from Thermo Book) Pg 403

The valve position in an overhead valve setup will allow for an increased volumetric efficiency. “Volumetric efficiency” is a ratio (or percentage) of the volume of fuel and air actually entering the cylinder during induction to the actual capacity of the cylinder under static conditions (Engine Science).” Because of the efficient, direct path to the combustion chamber when utilizing an OHV, this will greatly increase our volumetric efficiency. It was decided to use an existing cylinder head from another Briggs and Stratton motor. The cylinder head was purchased from http://www.dynocamstore.com. A picture of the head is shown in figure 6. The team modified the bolt-hole pattern on the existing cylinder to allow for the new head to be attached. The problem we encountered here is the mounting holes extend over the existing cylinder, so an adapter plate was fabricated to accommodate this modification. A CAD model of this adapter plate is shown in Figure 2.

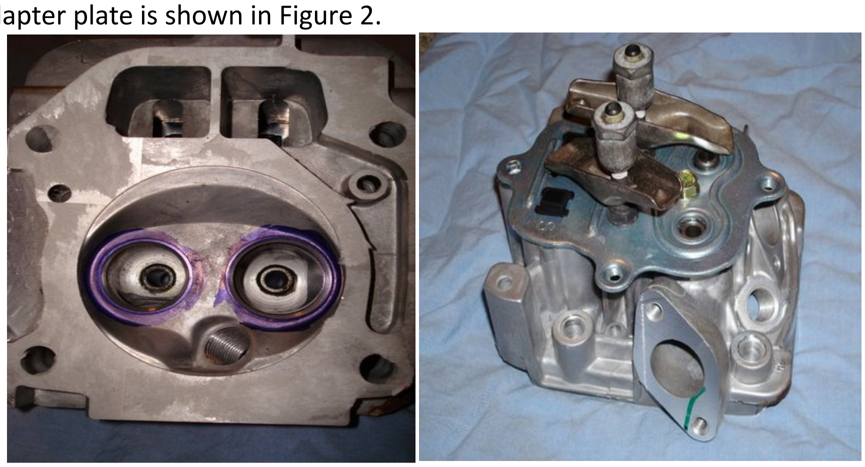


Figure 6: Overhead-valve conversion cylinder head

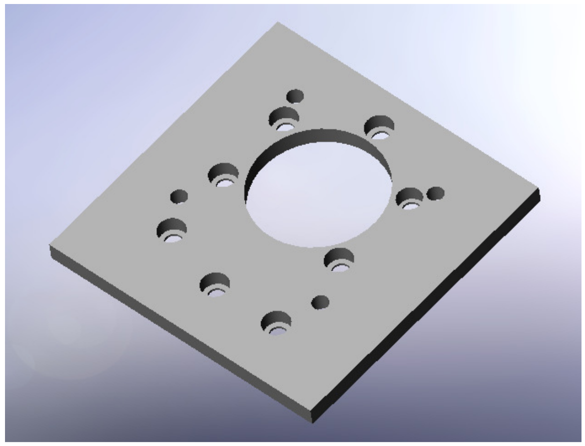


Figure 7: Engine head adapter plate

The top cooling fins will be machined off of the cylinder and the plate will be fitted around the cylinder providing us extra surface on which to bolt the cylinder head. Since we were using a new head we needed to use a new camshaft, so the center distance between the valves would be the same as the center distance between the cam lobes. Since the cam needed to be moved, it needed to be modified to fit the dimension restraints of the valve train. To be able to move the camshaft to the overhead position we began by adding a support bracket. This bracket fastened to the top of the head. The bracket created a good surface to be able to fasten our camshaft support brackets.

A new rocker arm needed to be fabricated to allow for direct contact with the cam. Bearings were also utilized to help reduce friction on the pivot point of the rocker. On the valve side of the rocker we threaded a hole into the rocker to allow us to adjust the valve backlash. The rocker arms were made from 6061-T6 and the pivot pin was made from case hardened pre-ground Thompson shaft. For the adjustment nuts we used O-1, this was to make a hard ware surface between the rocker and the valve stem. Figure 8 shows the rocker arm assembly.

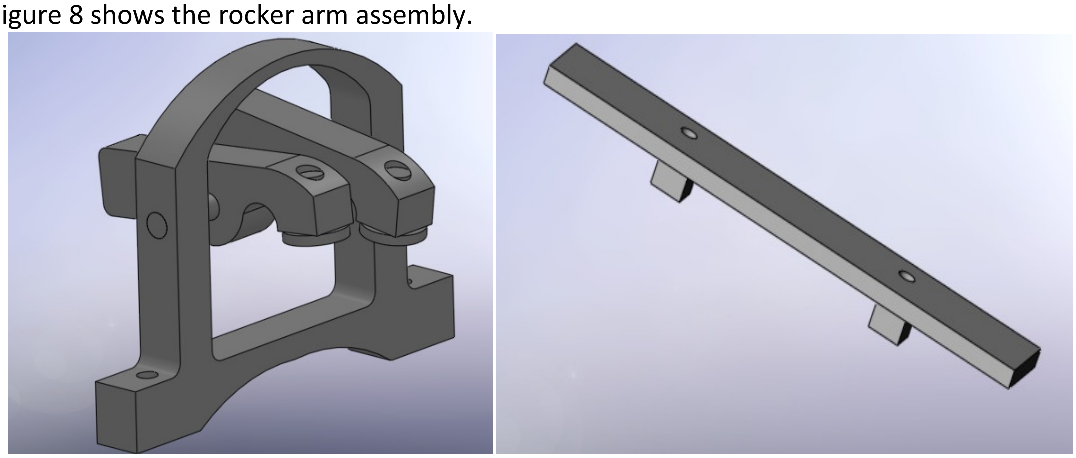


Figure 8: Rocker arm assembly (left) Cam shaft support (right)

The other main design change of the Briggs & Stratton engine is to reduce the power output which will decrease the amount of fuel consumed. This can be done by reducing the displacement of the engine. One way is to reduce the piston diameter and sleeve the existing cylinder bore to accommodate the new smaller piston. We chose a piston from a 3 HP Briggs engine. Choosing this piston allowed us to use the existing crankshaft and connecting rod. The nominal piston diameter of the new piston is 2.375 inches.

The sleeve that we fabricated had an outside diameter of 2.5635 and an inside diameter of 2.377 from 304SS. The part was held into place utilizing an interference fit.

Mechanical losses will be minimized utilizing bearings. The 3.5 HP Briggs & Stratton does not utilize bearings to support the crankshaft-- it uses bushings. If we switch the bushings to bearings, there will be less mechanical loss in our system raising the overall efficiency of the engine. We pressed needle bearings into the two supports. Another bearing was added to the cam shaft this was also a needle bearing.

We also needed to transmit power to the camshaft from the crankshaft. To do this we purchased two Timing pulleys from Mcmaster.com. We enlarged the inside diameter to .5 inch on one and .75 inch on the other. The bottom pulley was connected to a camshaft extension rod that extended out from the stock cam shaft. The extension rod was made from 17-4PH and hardened to 38 Rockwell C. The stock cam shaft was drilled to a diameter of 5/16ths of an inch by 1.1 inches deep. The extension rod was then slipped into the camshaft and pinned in place using a shear pin. A hole was drilled into the crank case to allow the extension piece to pass through and out of the engine compartment. Since we utilized the camshaft, the extension shaft will be turning the appropriate speed and direction that the cam shaft needs to travel at. The 0.75in inner diameter pulley was keyed and set screwed into place on the cam shaft extension rod. The upper pulley was keyed and set screwed into place on to the overhead cam. A timing belt was purchased from Mcmaster.com to link the two pulleys. A tensioner was installed using an idler pulley to keep the belt tight. Shown below is a picture of the Stock Briggs and Stratton engine rendered in solid works. This is the supplied engine from SAE without the gas tank, air fuel delivery system, starting system and also manifolds. From hear we were able to modify the motor to suet our needs.

Basic Vehicle Configuration

The vehicle will be a three wheel design, one wheel in the rear and two in front. This layout will provide planar stability, possess aerodynamic efficiency, and allow for all of the components of the vehicle to be placed properly. This design will pass the technical inspection guidelines for both SAE Supermileage and Shell Eco-Marathon. The team chose an aluminum honeycomb core with aluminum face sheets as the main load-bearing frame. Aluminum honeycomb core with aluminum facing has the greatest strength/weight ratio of most common materials. The shape of the fairing will be designed to accommodate our drivers as well as all of the components attachments. The size of the frame is 236cm x70cm (93 inches x 28 inches) (l x w). The shape and dimensions are shown in figures 9 and 10. The shape of the frame was designed to coincide directly with the aerodynamic fairing. The tear drop shape will allow sufficient room in the front for the driver, steering components and wheels. The portion of the frame behind the driver tapers to 15cm (6 inches) at the rear of the vehicle. This shape provides enough space for the engine, rear wheel and many types of drive train components. The overall height of vehicle is shown in figure 11.

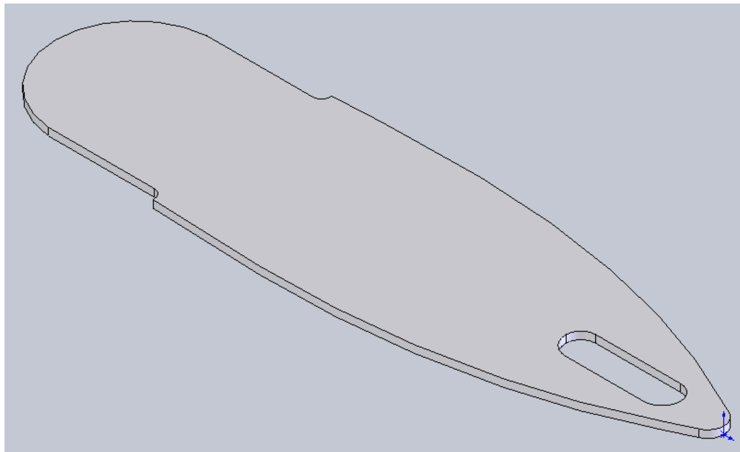


Figure 9: Isometric view of aluminum honeycomb frame

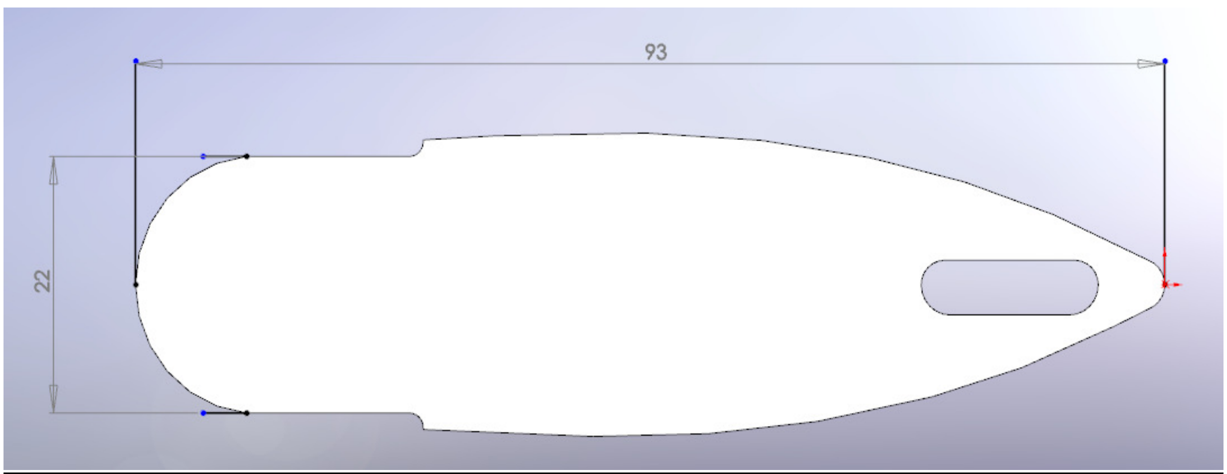


Figure 10: Top view of aluminum honeycomb frame with dimensions

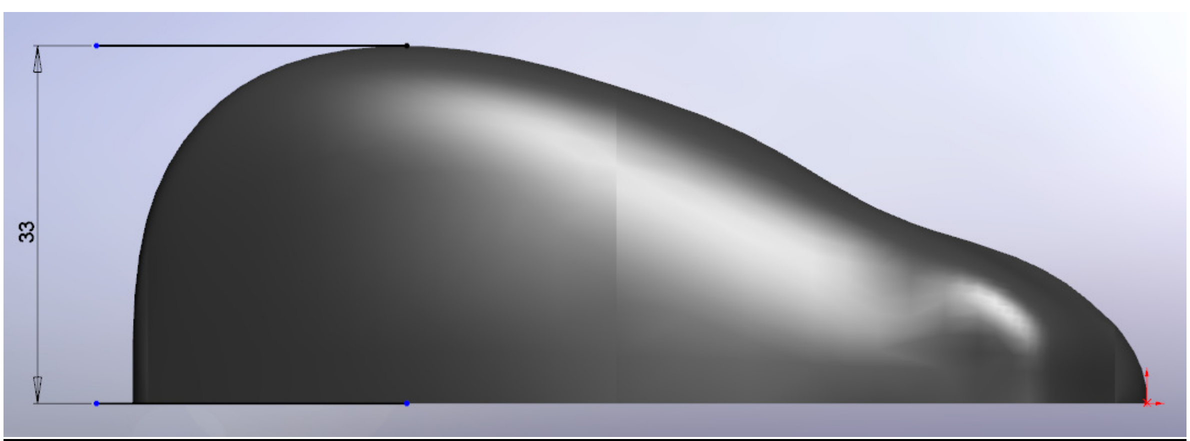


Figure 11: Side view of vehicle

The next aspect of the frame design to be considered was mounting components to the aluminum honeycomb. We decided to use high strength adhesives to fasten threaded aluminum inserts to the frame. The team did a great deal of research to decide what type of adhesive was best suited for our application. Henkel-Loctite is an industrial supplier of commercially available adhesives. A spreadsheet shown in Appendix B describes different adhesive types and there specifications. High strength adhesives were used throughout the frame and components for fastening purposes. Also, to insure everything was securely fastened and able to operate properly; mechanical fasteners were used as well as for extra safety. There are a number of different techniques to secure components to honeycomb. Our team decided to use threaded inserts provided by Shur-lok Corporation. From our initial State-of-the-Art research it was shown that this type of fastening technique is used throughout industry and in aerospace (Shur-lok Corporation) The company sent us a blind threaded type insert which is shown in figure 12.

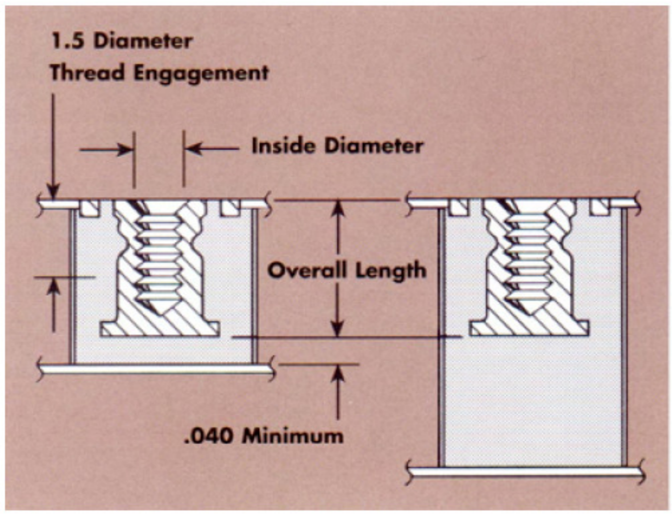


Figure 12: Insert used to secure components to frame

In order to ensure adequate support for the given loads, destructive testing was completed on multiple insert/adhesive samples. The overall results are shown in table 1. It was decided to use the Hysol type epoxy for all major load bearing supports. The acrylic without fibers was used where high loads would not be seen.

Table 1: Destructive tension testing results

Adhesion Technique	Failure Load (LBS)	Failure Mode
Acrylic with out fibers	750	Crack propagation in face sheet
Acrylic with Fibers	450	Shear in epoxy
Hysol without fibers	900	Crack propagation in face sheet
Punched w/ Hysol	940	Crack propagation in face sheet

Power Train Configuration

Another crucial system of the Supermileage vehicle is the drive train. The design of this system includes the necessary calculations that coincide with the design as well as several other factors. The designed system consists of a Briggs & Stratton 3.5 HP engine that will transfer power to the rear drive wheel via a centrifugal clutch, v-belt system consisting of lightweight, aluminum sheaves and a composite link v-belt, and adaptors fabricated to create the best mode of fixture to both of the sheaves. The freewheeling hub provides for a single stage reduction that reduces friction in our drive train. Since the links could be added or removed from the belt due to damage or wear, the composite link v-belt allowed for the adjustment of the total belt length. The tires incorporated on the rear wheel hub have minimal tread, resulting in lower-rolling resistance for the vehicle. In order to adapt the rear wheel sheave to the freewheeling hub, a unique spline had to be modeled in SolidWorks and fabricated using a wire EDM to cut the spline into the aluminum sheave. This allowed for the sheave to be mounted directly to our freewheeling rear hub. Figure 13 depicts a picture of the final spline fabricated into the rear wheel sheave. A unique clutch adaptor needed to be fabricated in order to mate the circular boss of the centrifugal clutch to the drive aluminum sheave. This was done using a turning lathe to create the circular boss for the sheaves to mount to and the circular extrude cut to accept the centrifugal clutch. The adaptor was broached to accept a keyway stock of 3/16” X 3/16”. The adaptor was then welded into place with the cover of the centrifugal clutch. Figure 14 shows a SolidWorks rendering of the drive train assembly.

Figure 13: Spline fabricated into rear wheel sheave

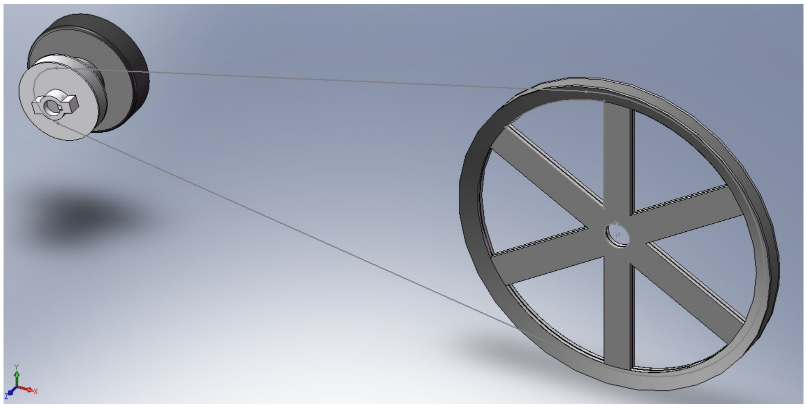


Figure 14: Drive Train Assembly

Brake System

For the design of the braking system our Client required that we use two different sets of brakes actuated independently of each other, one on the front wheels and one on the rear wheel. To satisfy this requirement we decided to use a combination of a rim brake and two disc brakes. To stop the rear wheel we used a bicycle rim brake since the drive train would not allow for mounting of a rotor. To mount this brake we bolted it directly to the frame of the vehicle directly behind the rear tire. To stop the front two wheels bicycle disc brakes were mounted on each the front driven wheels because they were easier to mount to wheels that steer than rim brakes would have been. To mount the front disc brakes so they would move with the wheels while steering we mounted the brake shoes from an extension to the wheels axles and the rotors were bolted directly to the tires hubs. To activate the brakes we decided to use mechanically actuated systems rather than hydraulic systems because they weigh less and are easier to fix if something were to malfunction at competition. The actuator for the front brakes was mounted to the steering column for ease of reach and the rear brake actuator was mounted to the side of the driver on the frame to keep it out of the way of the driver but still accessible if needed. To actuate both of the front brakes at the same time to not affect the steering of the vehicle while braking, a mechanical brake line splitter was used. To determine the exact style and brand of brakes that were installed we first determined the stopping power required by the SAE braking test. To do this we used a number of conversions and calculations shown below in figure 15. Using the outcomes of these calculations and looking at such considerations as cost and ease of adjustability we chose to use Avid BB7 mountain bike disc brakes on the front wheels as well as Tektro rim V-brakes on the rear wheel.

Decelerate from 10 miles per hour to 0 miles per hour in 4 meters

$$V = 10\frac{miles}{hour} = 4.47\frac{meters}{\sec ond}$$

$$t = \frac{d}{V} = \frac{4meters}{4.47\frac{meters}{\sec ond}} = 0.9\sec onds$$

$$KE = \frac{1}{2}mV^2 = \frac{1}{2}(136.1kg)(4.47\frac{m}{s})^2 = 1359.7\frac{kg \cdot m^2}{s^2}$$

$$F_{fric} = C_{rr}xN_f = 0.0025(136.1kg)(9.81\frac{m}{s^2}) = 33.4N$$

$$F_{decel} = \frac{KE}{d} - F_{fric} = (\frac{1359.7\frac{kg \cdot m^2}{s^2}}{4m}) - 33.4N = 306.5N$$

$Where:$

$V = StoppingVelocity$

$t = StoppingTime$

$KE = KineticEnergy$

$m = VehicleMass$

$F_{fric} = RollingFrictionForce$

$C_{rr} = RollingCoefficentOfFriction$

$N_f = VehicleWeight$

$F_{decel} = DecelerationForce$

$d = StoppingDis\tan ce$

Figure 15: Calculations used to determine required braking force.

Suspension and Running Gear

Wheels and Tires

For the front wheels we chose 16" rims and tires with extremely small traction grooves. We made this decision by using a study that was done by an ASME human powered vehicle from our university. This study concluded that smaller diameter wheels had lower rolling resistance and that this style of tires has low rolling resistance because we can fill them to a higher tire pressure than normal. For the front tires we also chose to use Canonndale Lefty hubs because they require an axle that is only supported from one side of the wheel. This type of hub helped minimize the frontal area of the vehicle while still maintaining a wheel track that was sufficient for required stability. These hubs also contributed to the fuel efficiency of the vehicle because have roller bearings in them that reduce the friction that occurs from the tires supporting the weight of the car. This minimization of rolling friction allows the vehicle to travel a farther distance without the input of excess power.

Axels and Front Wheel Supports

Because of the decision to use Lefty Hubs and the weight, size, and cost of purchasing the accompanying axles we deemed it necessary to fabricate our own axles and knuckles. To design these components we researched go-kart design because the running gear set-ups were similar to what could be done with our design. Using Limestone Media's "How To Build A Kart: Go Cart Steering Plans" as a platform to build our design off of we came up with a very capable design which is pictured below in Figure 16 (http://www.limestonemedia.com/how-to-plans/go-kart-steering-plans.htm. The axle was dimensioned to seat perfectly inside of the hub and to support the hub bearings without any excess pre-loading. The axle was then welded to a hollow cylinder which was machined to house two needle roller bearings that help support the king pin and help ease the forces necessary to turn the wheels by reducing friction. Between the inner edges of the knuckle and the axle assembly are two thrust bearings. These bearings were chosen because they are rated to support the maximum load and will reduce more of the friction required by turning the wheels. To support the rear wheel of the vehicle, aluminum box tubing was used to support machined pillow blocks that are shown in figure 17. All of these components were first modeled in SolidWorks and a Finite Element Analysis was completed on them to check for failure. A picture of one of these analyses is shown in the physical modeling section.

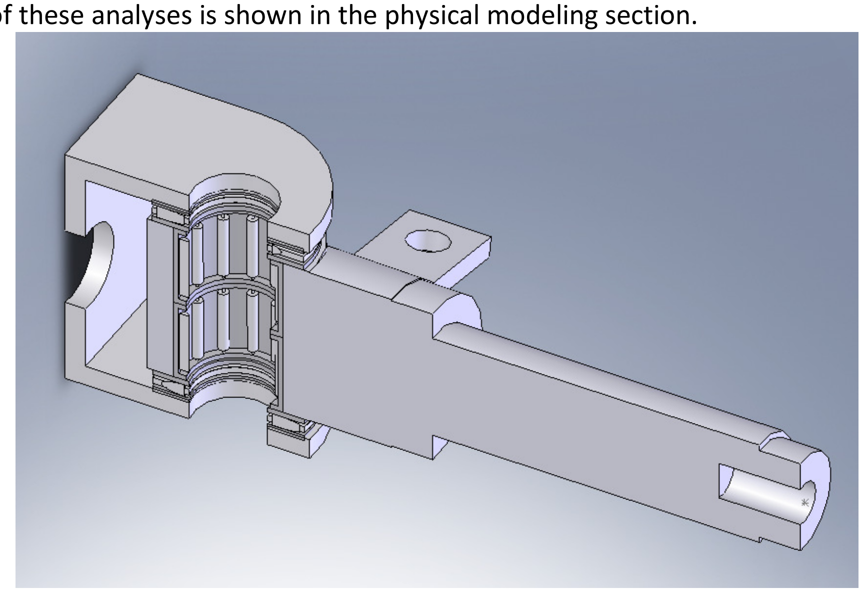


Figure 16: Cross section of the front wheel axle and supporting knuckle.

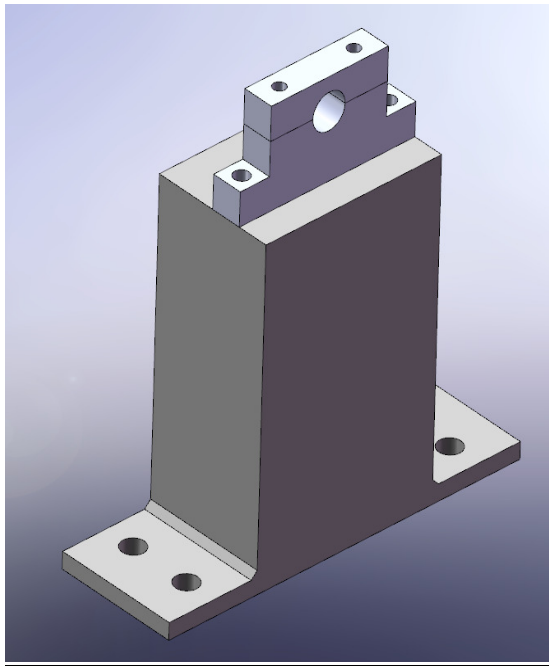


Figure 17: Rear wheel supports assembly

Steering

To steer the vehicle the group decided to use a rack and pinion based steering system. The driver turns a steering wheel that turns the pinion gear through a u-joint. The rack then pushes/pulls on tie rods connected to the control arms which are welded directly to the axle. The axle then rotates about the king pin. The point where the tie rods meet the control arms lays on a line front the center of the front axle to the center of the rear axle; this design uses the Ackerman steering theory to reduce the amount of tire scrub while turning (http://www.auto-ware.com/setup/ack_rac.htm. Using the calculations in Figure 18, we determined the angle each of the front wheels needed to turn to turn the vehicle around the required steering obstacle course which was about 12 degrees. Using that value we determined the length and angle the tie rods needed to be using a kinematics and sketches in SolidWorks. It was decided that the vehicle would help self-straighten after a turn so a caster angle of negative 10 degrees was implemented into the front wheel design.

$$R = \frac{t}{2} + \frac{w_b}{\sin \alpha}$$

Where:

R = Radius = 50 ft

t = track width = 24 in

w_b = wheel base = 72 in

Solving for α:

α = turning angle = 7.034 degrees

Figure 18: Calculations solving for required turning angle of wheels.

Fairing

The overall body force on the vehicle, P_r, is determined according to the following equation and is a function of the indicated variables:

$$P_r = \left(C_R M_v g + \frac{1}{2}\rho_a C_d A_v S_v^2\right) S_v$$ (Internal Combustion Engine Fundamentals)

where: C_R = coefficient of rolling resistance

M_v = mass of vehicle

g = acceleration due to gravity

ρ_a = ambient air density

C_d = drag coefficient

A_v = frontal area

S_v = vehicle speed

This equation shows how body forces related to vehicle mass and frontal area produce the total force on the vehicle. To gain the desired effect, the frontal area and drag coefficient were both minimized within the decided vehicle dimensions. Because a driver and several components must reside within the vehicle, the reduction of the frontal area is greatly constrained. The drag coefficient is a parameter that will have a significant effect on the overall body force. These calculations are shown in Appendix C Our group aimed to minimize flow separation due to low pressure areas over the vehicle. The shape of the vehicle was designed using intuition, previous knowledge of fluid flow, and computational fluid dynamics (CFD). The fairing was modeled in Solidworks™ and the shape was created using a series of cross sections that are used to create a surface loft. The surface was then thickened to 0.125 in. A first revision rendering of the fairing is shown in figure 19.

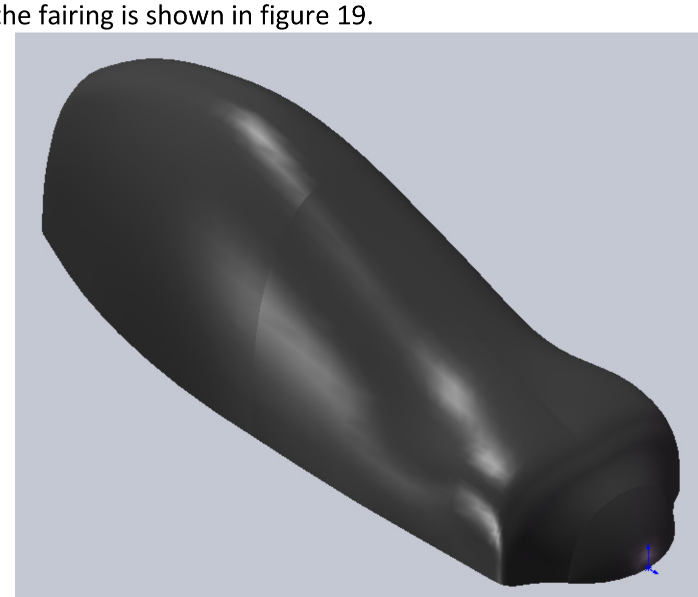


Figure 19: 1st Revision Fairing Rendering

Once the shape of the fairing was rendered in Solidworks™, the fluid flow was analyzed using Cosmos FloWorks. It was determined from this analysis that the shape could be altered slightly in some areas to reduce the coefficient of drag. In particular, the area over the drivers head sloped to too quickly and was creating a low pressure zone. To reduce the low pressure the top was kept at nearly horizontal until the very edge of the rear of the car. After multiple iterations of the shape, a final design was modeled again. It was determined that the overall drag force on the vehicle was less than about 3 lbs at 25 mph.

It was decided that the fairing would be fabricated using a lightweight fiberglass composite material. In order to create a unique shape of our desired design, a three stage molding process was implemented. The process involved making a positive (male) mold, a negative (female) mold, and finally the finished positive fairing. By creating the final part as a male component, the outside surface was smooth and free of disruptions in the contour.

To create the first male mold, the solid model was mocked together using 95 individual Styrofoam cross sections which were cut out using a CNC router. In order to locate each piece, three half inch circles were cut into each cross section and aluminum tubing was inserted into each piece to form the whole part. The sanded, assembled foam is shown in figure 20. Fiberglass was then cured over the top of the foam to form a hardened exterior. Non polyester lightweight body filler was then applied to the exterior and the sanded to obtain the desired shape. Once the team was satisfy with the shape of the male mold it was painted and sanded to prepare for the next stage. A release wax was used to reduce any sticking of the female mold. Next, approximately six layers of fiberglass were laid up onto the male plug to form a rigid part. This part was removed and sanded to create the final female mold. Blue on gold colored fiberglass was used as the exterior layer and two layers of 7075 fine glass were used on the interior to provide extra strength. Once the part was cured it was released from the mold and trimmed. The team is still in the process of making the final touches on the aerodynamic shell, this includes: fixturing, window placement, and body work. The final part is shown in figure 21.

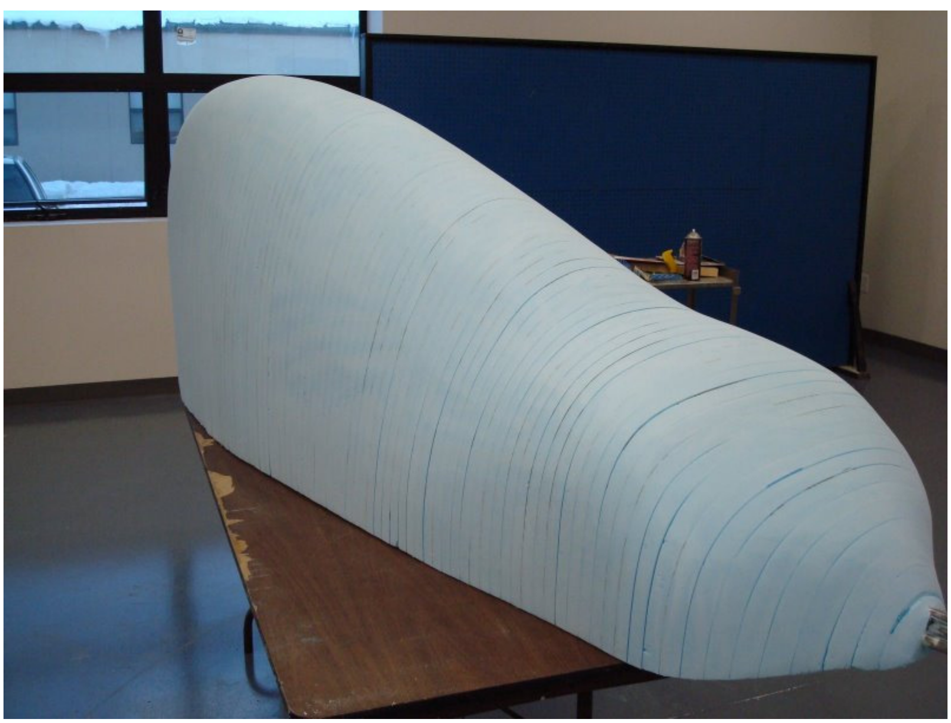


Figure 20: Completed Foam Model

Figure 21: Completed Fairing

Performance

Calculations have been made concerning the velocity of the car using certain experimental values such for the gear ratio and the input rpm of the shaft. Aluminum sheaves were purchased based on which sheave combinations produced the desired speed at the maximum power band at the greatest rate of acceleration. This resulted in a significant gear reduction system with a smaller drive gear and massive driven gear. These Microsoft Excel Spreadsheet calculations are shown in the Appendix C (8). The vehicle will have a maximum speed of 42 mph at the angular velocity of 3000 rpm with a drive gear outer diameter of 2.75 in. and a driven gear diameter of 11.75 in. The sheaves add only five pounds to the total weight of the car, which is significantly less than what they could weigh if they were manufactured from cast iron. In order to achieve the optimal fuel efficiency, the driver will power the vehicle up to speed with the engine, coast using the freewheeling hub by stopping the engine, and reengage the engine to power back up to speed. This manipulation will allow us to accelerate as fast as possible to our maximum speed, allowing for less fuel consumption. No calculations were proposed to analyze the predicted fuel consumption of the vehicle since the engine had not been completed by the deadline for this report; thus, zero road testing or dynamometer engine testing has occurred.

Tension Test

In order to decide if the honeycomb inserts are satisfactory for our vehicles design, we decided to complete destructive testing on multiple samples the test procedure was completed as follows:

- The tension test was conducted with the specimen supported in a box frame fixture by a 2.5 inch diameter steel ring with a 0.250 inch thick wall.
- The fastener was centered in the middle of the ring.
- The bolt used was of sufficient strength to fail the specimen.

- Care was taken to assure that the direction of loading is perpendicular to the surface of the specimen.
- A dial indicator or automatic recorder shall be used to measure deflection.
- The tension test specimen was a 4.0 inch square section.
- The fastener shall be installed in the center of the specimen.
- A minimum of three specimens shall be tested.
- For molded-in fasteners a full cure of the compound shall be obtained prior to testing.

The test setup is shown in figure 22

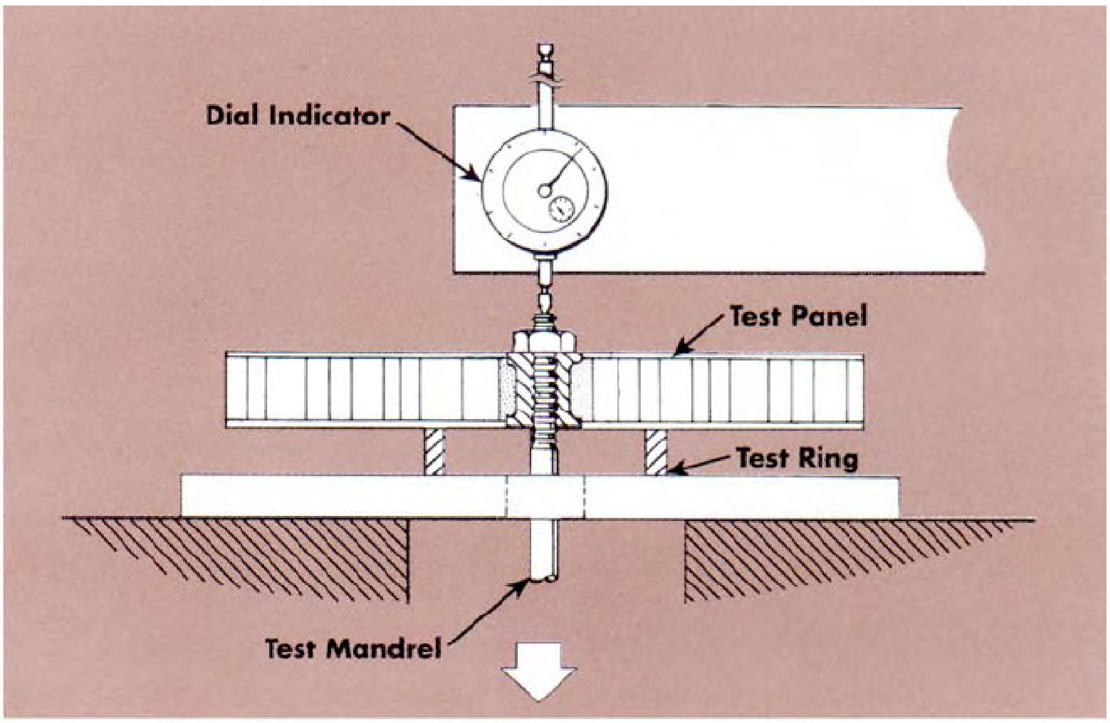


Figure 22: Tension test of honeycomb inserts

Modeling

Supporting Mathematical Models

Most of the initial calculations, located in the Appendix, focused around the feasibility of adapting most of the researched off-the-shelf-parts to the vehicle such as the aluminum sheave selection given the maximum power output of the engine. Since the peak power band occurs around 3500 RPM, numerous iterations were conducted to receive the best acceleration from the selected sheaves given this peak power value. From these calculations, several aluminum sheaves were purchased and implemented into the car's assembly as well as an accompanying belt of calculated length. A beam deflection analysis of the composite sandwich panel was also used in order to choose the best panel for the frame construction. An Excel spreadsheet was created to analyze the maximum deflection for the given loads applied to the frame. A composite panel was then selected based upon the calculations conducted that predicted that the panel would not fail with the given loads present on the vehicle. Volumetric efficiency calculations were performed to investigate the effects of temperature and pressure on the efficiency of the engine and how to combat these affects. A spreadsheet was created to address the situation with the engine testing with the values received from the dynamometer. The spreadsheet's goal is to determine reduce the specific fuel consumption by adjusting certain parameters individually. Reynold's number analysis was performed on the fairing to determine the overall drag force applied to the vehicle. With this analysis, the team came to determine that the fairing significantly reduced the overall drag force applied to the vehicle. The road load calculations were to determine the overall load applied by the wind to the car given our frontal area. These calculations helped to determine that with a smaller frontal area the team could achieve a low wind load applied to the vehicle. Calculations were also conducted on the stress inherent within the timing shaft. Since the shaft was made of high strength steel, these calculations validated that the shaft would not fail under the given stresses.

The aluminum honeycomb panel that will best fit our application with dimensions: 48''x96''x1.04'', core size of 3/16", with a density of 5.7 pounds/cubic foot. Based on calculations from M.C. Gill Corporation, these features were selected. There were many considerations to consider including; facing, core materials, weight, cost, strength, and corrosion resistance. The purpose of the calculations is to double check the overall factor of safety to determine whether chosen panel meet the criteria desired. The following was taken from www.mcgillcorp.com (10):

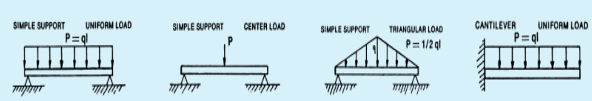
Table 2 – Used symbols and variables shown in the calculations

NOMENCLATURE RELATIVE TO CALCULATIONS ON PAGES 4 & 5

Δ	Deflection, inches.	K_b	Coefficient for panel bending. (See Fig.1)
λ	Safety factor (usually 1.5-2.0).	K_s	Coefficient for core shear.
a	Span, length, inches.	K_f	Coefficient to correct for flexural modulus of facing. (See Table 2).
b	Span, width, inches.		
c	Core thickness, inches.	K_c	Coefficient to correct for core type: 2.2 for foam, 1.0 for honeycomb, 0.7 for plywood.
C_s	Coefficient for core shear stress, from Figure 1.		
C_b	Coefficient for facing stress, from Figure 1.	K_1	Bending constant, sandwich panel loaded as a plate* (See Figure 3).
d	Total panel thickness, inches.	K_2	Constant for facing stress for panel loaded as a plate* (See Table 1).
D	Panel rigidity	K_3	Constant for core shear stress.
E_f,	Flexural modulus of either top or bottom facing, psi. (See Table 2).	P	Load applied to panel, lbs./inch width.
G_c	Shear modulus of the core, psi. (See Table 3).	q	Uniform load, psi (P = qa).
		S	Core shear stress, in psi.
h	$(t-t_f)$, thickness of panel between centroids of facings	t_f	Thickness of facing, inches.
		t	Total panel thickness, inches.

**Assumes constant properties in thickness direction.*

The panel was modeled as a simply supported beam with a load applied at the driver's seating position. Figure 23 shows an example of this type of loading. Table 2 describes the specific variable used in the deflection calculations.



BEAM TYPE	Simple support, uniform load	Simple support, center load	Simple support, triangular load	Cantilever, uniform load
Bending Deflection Constant K_b	.013	1/48	1/60	1/8
C_b	1/8	1/4	1/6	1/2
C_s	1/2	1/2	1/2	1

Figure 23: Typical loading cases for, simply supported is the vehicle's loading condition

Kb = Bending constant found in Figure 1
Pc = KcP
P = Load, in lbs. per inch of width (total load □b or qa where q = load per unit area)
Kc = Coefficient to correct for foam weakness
Kc = 2.2 for foam cores, 1 for honeycomb and balsa cores, 0.7 for plywood
a = Unsupported span in inches
□= Safety factor, usually 1.5-2.0
□= Maximum allowable deflection
Kf = Flexural coefficient for facing

To determine the stiffness of the panel, the rigidity was calculated using the following equation:

$$D = \mu\ Kf\ Kb\ Pc\ (a^3/\delta)$$

Where:
μ = 1.2 (Estimated Approximation)
Kf = 1.0 (For all Aluminum)
Kb = 1/48 (Figure 24)
Kc = 1.0 (For Honeycomb)
P = 160 lb (Driver's weight)
Pc = P*Kc = 160 lb
a= 120 in
$\delta = 9*10^4$ in (Flexural Rigidity Curve for Aluminum Facing, (Figure 24))

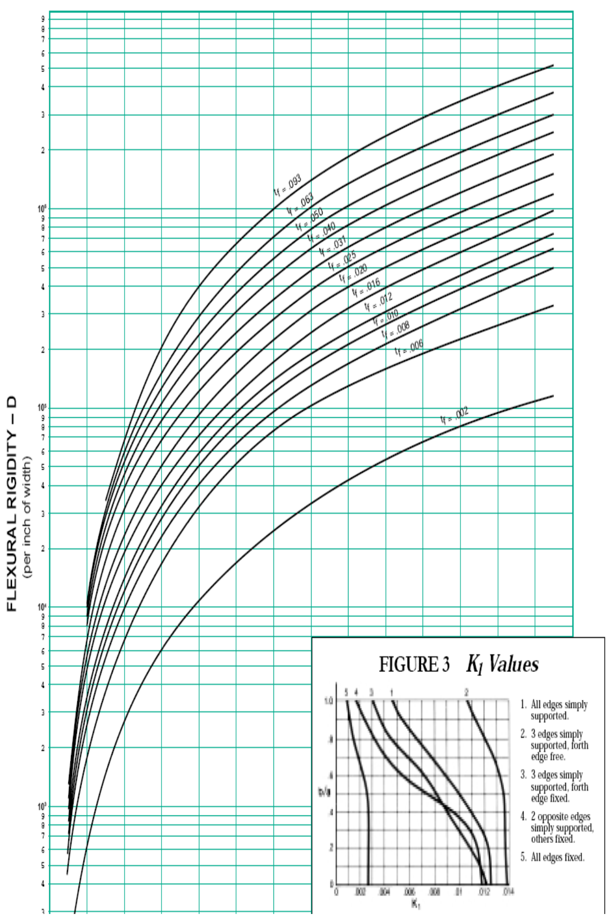


Figure 24: Flexural rigidity of honeycomb panels

To determine the core sizing for the applied load, the following equation was used:

S = (CsP/h)

Where:
$h = T_f + T_c$
T_f = 0.02 in
T_c = 1 in
H = 1.02 in
Cs = ½ (Figure 24)
P = 160 lb (Driver's weight)
To determine the face sheet sizing for the applied load, the following equation was used:

$$Fs = (Cb*Pc*a/ h*T_f)$$

Cb = ¼ (Figure 24)
Pc = 160 lb
A = 120 in
H = 1.02 in
T_f = 0.02 in

Table 3: Total stresses seen in loading condition

Variables	Values			
μ	1.2			
Kf	1	D = μ*Kf* Kb* Pc *(a3/δ)	0.000767	Rigidity
Kb	0.0208			
Pc	160	S = (CsP/h)	78.43137	Core Stress lb/in^2
a	120			
δ	9000000000	Fs = (Cb*Pc*a/ h*tf)	94.11765	Facing Stress lb/in^2
Cs	0.5			
P	160			
h	1.02			
Cb	0.25			
tf	0.02			
tc	1			

Using the data acquired in this study, the actual panel thickness, core material, and core sizing was chosen based on the data in figure 26.

Core	Cell Size in.	Foil Th. in.	Density pcf	Shear Str. psi L/W Dir.*	Shear Mod. ksi L/W Dir.*	Stabilized Compress. Strength psi	Heat Transfer U = BTU/ hr/ft²/°F
Aluminum Honeycomb	1/8	.0007	3.1	155/90	45/22	215	0.85
	1/8	.001	4.5	285/168	70/31	405	0.95
	1/8	.002	8.1	670/400	135/54	1100	0.95
	3/16	.002	5.7	410/244	90/38.5	600	0.95
	1/4	.001	2.3	100/57	32/16.2	130	1.00
	1/4	.002	4.3	265/155	66/29.8	370	1.00
	1/4	.003	6.0	445/265	96/40.5	660	1.00
	3/8	.003	4.2	255/150	65/29	355	1.00

Facing Material	Yield Strength f_f (psi x 10^3)	Modulus of Elasticity E_f (psi x 10^6)	Wt. per Mil Thickness (lb/ft²)	K_f	Comments
Aluminum-2024-T3	42	10	0.014	1.0	Good strength, moderate cost
Aluminum-3003-H16	20	10	0.014	1.0	Moderate strength, good weathering when Alclad
Aluminum-5052-H32	23	10	0.014	NA	Coated for corrosion resistance
Aluminum-6061-T6	21	10	0.014	1.0	Workable, corrosion resistant
Aluminum-7075-T6	60	10	0.014	1.0	High tensile strength and dent resistant
Cold rolled carbon steel- 1.5% carbon content	50	28	0.040	.35	Low cost, high weight, hard to cut with hand tools
Stainless steel-316	60	29	0.040	.33	Heavy, expensive, hard to bond and fabricate with hand tools; high rigidity and strength

Figure 25: Choices of honeycomb panels from M.C. Gill

Physical Modeling

A substantial amount of 3-dimensional modeling has been conducted for the fabrication of our vehicle. Since most of the parts used in the vehicle's construction were off-the-shelf parts, dimensioning and tolerance issues needed to be addressed with the 3-D modeling of these parts. The main software product used consisted of SolidWorks and several of its add-ins such as FloWorks and CosmosWorks.
Figure 26 below depicts a FloWorks analysis of the fairing in a controlled wind environment. The areas in blue denote areas of stagnation which induce a higher coefficient of drag. CosmosWorks allowed the team to analyze certain aspects of the vehicle that would be cumbersome with hand calculations. An analysis of the Al 6061 steering knuckles was conducted, and plots of the factor of safety were created. Neither the roll hoop nor the steering knuckles would fail under the applied loads the team is designing for. Figure 27 depicts a picture of the analysis done on the Al knuckles. There exists a plan for a CosmosWorks analysis of the welds, keyway, and individual drive train parts under the torque produced by the engine. The individual parts have been modeled and depicted in Figure 28, but the welds still need to be applied and the analysis run by CosmosWorks. Also, the electrical schematic for the vehicle is shown in figure 29.

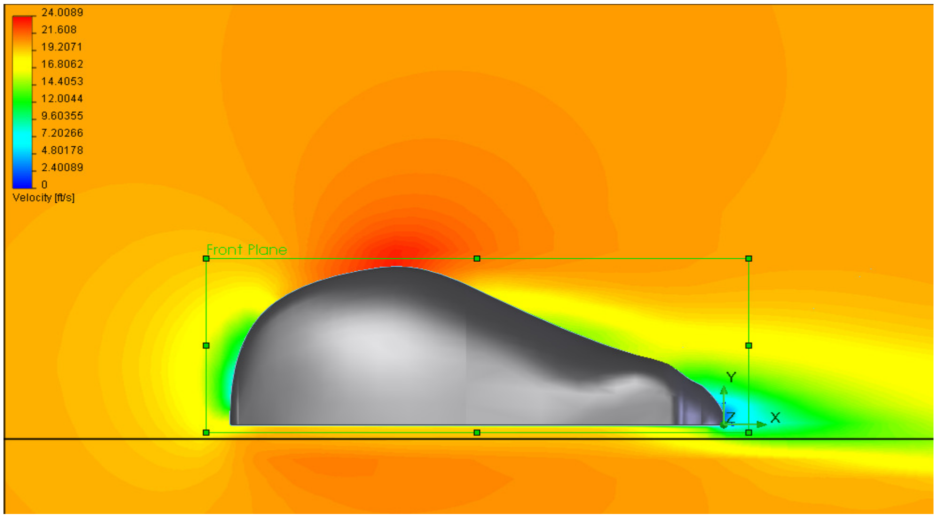


Figure 26: FloWorks analysis of fairing

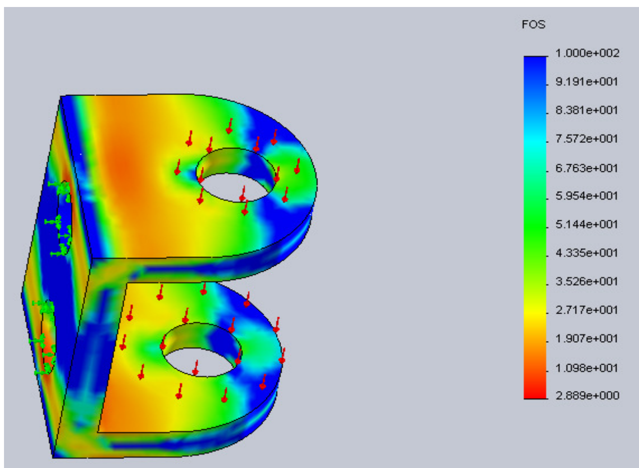


Figure 27: CosmosWorks Design analysis of the Al steering knuckles showing a Safety Factor of 2.9.

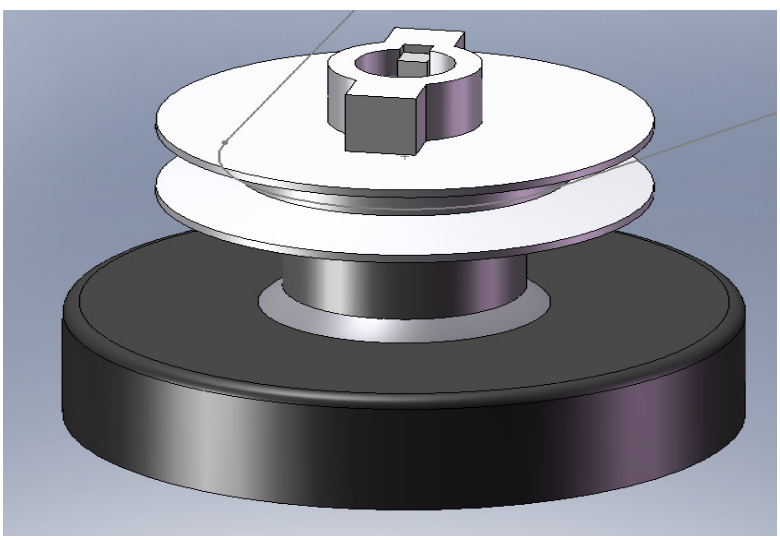


Figure 28: Drive train assembly with welds

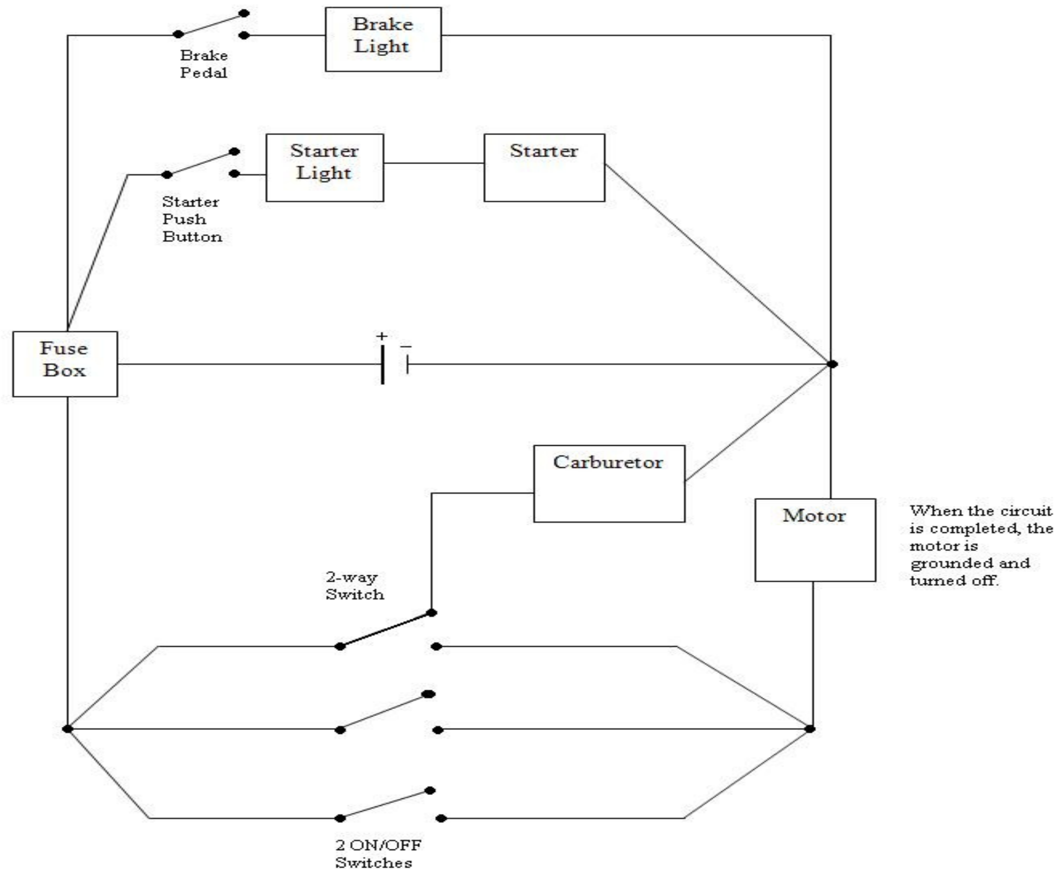


Figure 29: Complete Electrical Diagram

Driver Safety Features

The top priority in the designing of this vehicle has been to comply with all of the 2008 SAE Supermileage Rules. The safety specific rules and their respective design solutions that have not already been discussed are discussed below.

Kill Switches

A total of three kill switches are mounted to the vehicle. The two external switches are mounted flush with the body on both the right and left sides of the vehicle. The placement of these switches is 20" above the ground just behind the firewall, in a visible and accessible location. The third kill switch is mounted to the steering wheel for ease of access from inside the vehicle. As required, all three switches are labeled with the words "run" and "kill". Labels are highly visible which details to any person where to kill the engine in case of an emergency.

Guards and Shields

The frame acts as a body pan to separate the driver from any contact with the road. The firewall protects the driver against exposure from all fuel carrying components and any potential debris caught by the rear wheel. The front two tires are separated from the driver by bulkheads made of fiberglass, three ply's thick. Any electrical components in the cockpit are guarded using spiral wrap and heat shrink to prevent any driver contact.

Helmets/Clothing

The driver will wear an SA2000 helmet. This helmet is DOT rated and is supplied with an impact resistant face shield. Both interior and exterior surfaces of the face shield have been treated with an anti-fogging agent. The driver will also wear a fire resistant racing suit and durable closed toe shoes.

Fuel and Lube System

The fuel bottle will be mechanically fastened with fuel clamps to ensure minimal leaking of the fuel into the engine bay. The SAE 30R11 rated fuel line will be clamped using worm drive type hose clamps. A quick-connect fuel line adaptor will allow for a simple removal of the fuel bottle from the fuel line. The carburetor chosen for the vehicle has a bleed valve with removable bolt that allows for the fuel removal from the carburetor itself. A mixture of a dry lubricant, boron nitride, and liquid lubricant, conventional 5W-30 oil, will be manually placed onto the contacts of the rocker arms and valve stems as well as the contacts between the cam lobes and rocker arms to reduce friction that could potentially cause a fire. The team has developed a mechanical system and checklist for applying the lubrication to the rocker arms and their contact surfaces before each trial the vehicle undergoes.

Fire Extinguisher

The 2lb fire extinguisher is mounted vertically along the firewall in the engine compartment to the left of the engine. Tubing of .5" diameter with a .035" wall thickness is used to direct the extinguisher toward the engine and is remote actuated by the driver.

Exhaust System

The exhaust system extends approximately 1" from the body of the vehicle to expel any toxic fumes from the vehicle. The exhaust pipe is made from 1.25" aluminum round tubing that has 0.125" wall thickness and is welded directly to the exhaust manifold. The exhaust manifold is bolted to the exhaust port on the engine and is sealed with a gasket. The whole exhaust system is wrapped in exhaust insulation to help prevent burn when the engine has been running for an extended period of time.

Fire Wall

The fire wall is made of aluminum of .060" thickness. The fire wall is sealed within the fairing, separating the engine compartment from the cockpit. There is approximately 2" of additional height of the fire wall beyond the top of the driver's helmet.

Exit ability

The fairing is attached to the frame with Velcro and can be removed with little effort. Once the fairing is removed the driver has adequate space to maneuver him/herself out of the vehicle

Visibility

Front and side windows have been cut from the fairing in order to aid visibility of the driver. These cutouts were positioned so every driver of the vehicle is able to see 90˚ to either side of straight forward and +/- 80˚ longitudinally of straight forward. These window cutouts were replaced with a wind shield made of 0.022" thick polycarbonate plastic. The outside of the wind shields have been treated with a water beading agent and the inside of the wind shields have been treated with an anti-fogging agent. The vehicle is also equipped with two circular mirrors, one on either side of the driver to allow for visibility behind the vehicle. These mirrors have an approximate surface area of 2 in^2 and are domed to ensure sufficient visibility to the sides of the vehicle as well as directly behind it.

Roll and frame hoop rules
The roll cage is made of aluminum tubing that has an outer diameter of 1.25" and a wall thickness of 0.125". This material was selected based on a Finite Element Analysis done on the shape of the roll hoop. The roll cage is approximately 3" above the top of the tallest driver's head and only 1" behind the driver's head. The Solidworks rendering of the vehicle's roll hoop is shown in figure 30.

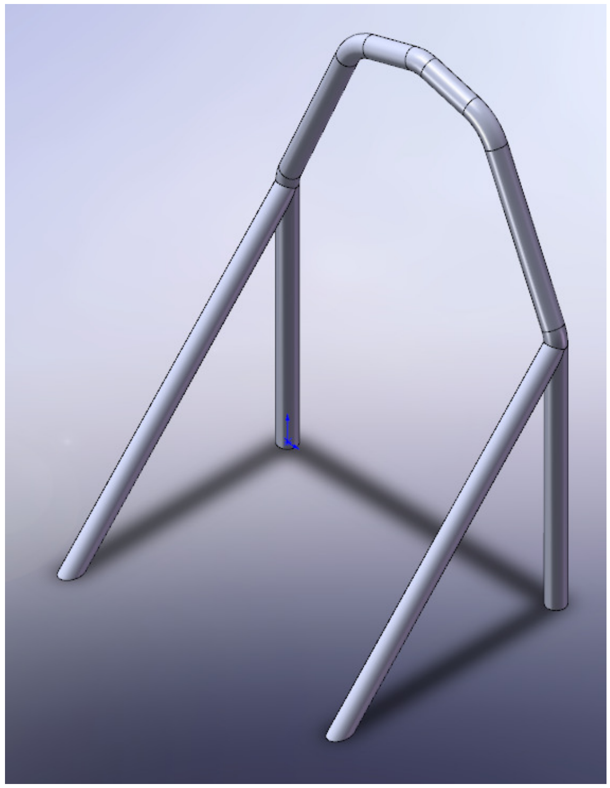


Figure 30: Final roll over protection design

Driver restraint
A 5-point safety harness restrains the driver to the frame of the vehicle. It accomplishes this with two over-the-shoulder straps, a lap belt, and one belt that attaches between the drivers legs.

Cost Estimate and Manufacturing Methods

The Northern Arizona Efficient Vehicle was funded from two separate sources. SAE at NAU has donated half of our funds in the amount of $3000. The second source of funding came from the technical advisor for the project, Dr. M.R. Mitchell in the amount of $3000. Below, in Table 4, is a Categorized Budget showing a summary of how much was spent on each vehicle subsystem and other parts of the project. These values were obtained using the Bill of Materials in Appendix C. Many of the materials and parts used to build the vehicle were donated by various Sponsors and some of the parts were reused from last years' vehicle. The clutch, starter, flywheel, wheels, one of the brakes, the helmet, and the battery were all purchased by last years' team and are being reused this year. The material used to fabricate the frame was donated to this project by M.C. Gill Corporation located in El Monte, CA and most of the materials used to make the fairing were donated by Quintus Incorporated located in Camp Verde, AZ. The stock material used was donated to this vehicle by Northern Arizona University's Machine Shop and the engine was donated by Briggs and Stratton Motorsports upon registering for SAE Supermileage 2008. Also, all of the vehicles hardware was donated by Copper State Bolt and Nut Company in Flagstaff, AZ. All the manufacturing done for the vehicle was completed by the team members. Approximately 1000 hours of fabrication were completed for this vehicle. It is estimated that $15 an hour would have been charged had these parts been outsourced for fabrication. Table 5 shows a fair market value for all of the parts.

Table 4: Categorized Budget

Categorized Budget	
Wheels	$121
Engine	$950
Steering	$306
Drive Train	$499
Fairing	$1038
Brakes	$309
Frame	$622
Fuel Delivery	$63
Safety	$367
Competitions	$2278
Misc.	$182
Total	**$6735**

Table 5: Donation and manufacturing estimates

Donation Cost Estimates	
Clutch	$92.45
Starter	$85.95
Flywheel	$67.35
Wheels	$647.60
Brakes	$50.00
Helmet	$120.50
Battery	$75.67
Frame	$1,250.00
Fairing	$2,000.00
Stock material	$100.00
Engine	$298.00
Hardware	$300.00
Manufacturing	$15,000.00
Total	**$20,087.52**

Bibliography

1. A2 Cold Work Tool Steel, Buffalo Precision Products Inc.
    a. http://www.buffaloprecision.com/data_sheets/DSA2TSbpp.pdf 10/2/2007

2. Anderson, John D. Fundamentals of Aerodynamics, Fourth Edition. McGraw-Hill 2007, McGraw-Hill Companies, Inc.

3. Brecoflex Co., L.L.C. http://brecoflex.web.aplus.net/ 10/8/2007

4. Budynas, Richard and Nisbett, Keith. Shigley's Mechanical Engineering Design Eight Edition, McGraw Hill 2008, McGraw-Hill Companies, Inc.

5. Comp Cams. http://www.compgoparts.com/ 11/23/2007

6. Groover, Mikell. Fundamentals of Modern Manufacturing Second Edition, 2004, John Wiley & Sons.

7. Heywood, John. Internal Combustion Engine Fundamentals, McGraw Hill 1988, McGraw-Hill, Inc.

8. Hibbeler, R.C. Engineering Mechanics: Dynamics Tenth Edition, 2004, Pearson Education, Inc. John Wiley & Sons, 2004.

9. Lakshminarasimhan, V, Ramasamy, Ramachandra. 4 Stroke Gasoline Engine Performance Optimization Using Statistical Techniques, SAE Technical Paper Series 2001-01-1800/4221, 2001 ATA, SAE International and SAE of Japan.

10. M. C. Gill Corporation. 4056 Easy Street, El Monte, CA 91731-1087. (626) 443-4022. http://www.mcgillcorp.com/

11. Moran, Michael and Shapiro, Howard, Fundamentals of Engineering Thermodynamics Fifth Edition

12. Quintus Inc. 684 Industrial Dr, Camp Verde, AZ 86322. (928) 567-3833. http://www.Quintusinc.com/

13. SAE International, Supermileage Overview. http://students.sae.org/competitions/supermileage/ 12/4/07

14. SCI Electronics. Repair FAQ. Gasoline FAQ. http://www.repairfaq.org/filipg/AUTO/F_Gasoline7.html 10/25/2007

15. Serway, Faughn. College Physics Fifth Edition. 1995

16. SI Combustion and Direct Injection SI Engine Technology. SAE International SP-2016. 2006

17. Tech Edge Pty. Ltd. 2002. NTK L1H1 Sensor Information. http://techedge.com.au/vehicle/wbo2/wbntk.htm 10/20/2007

18. Tripod: Jbabs Fuel Injection Section. http://members.tripod.com/~jbabs714/index.htm 10/5/2007

19. http://auto.howstuffworks.com/question377.htm, April 2008

20. http://www3.telus.net/dougsimpson/CVcarb.html, Jan 2008

21. Hexcel http://www.hexcel.com/Products/Downloads/HexWeb+Honeycomb+Datasheets.htm, April 2008

22. Engine Science http://www.auto-ware.com/combust_bytes/eng_sci.htm, April 2008

23. Ackerman Steering and Racing Oval Tracks http://www.auto-ware.com/setup/ack_rac.htm , April 2008

24. http://microcarproject.tripod.com/html/tuned_exhaust_system.htm, March 2008

Appendix A

Purchases

Donations:	$6,666.00
Our total budget:	$6,332.70
What we have left:	$ (360.09)

Team Member	Description of Part	Price	Tax	Total	Date Purchased	Category
Mike	Velocity Aero heat wheel 16x1 3/8	$51.95	$30.65	$82.60	11/19/07	Wheels
Mike	Oxygen Sensor NTK I1H1	$195.00	$25.00	$220.00	11/19/07	Engine
Tanya	Steering rack 8"	$89.00	-	~~$0.00~~	11/19/07	Steering
	Steering U-joint (chrome)	$28.00	$11.13	$0.00		Steering
	Steering rack adaptor	$11.00	$14.26	$153.39		Steering
Mike	Tire Pump	$31.95	$6.95	$38.90	11/19/07	Wheels
Karl	Piston	$16.10	$8.95	$25.05		Engine
Mike	Sheave	$6.02		$0.00	12/07/07	Drive Train
	Sheave	$6.96		$0.00		Drive Train
	Sheave	$8.42	$9.69	$0.00		Drive Train
	Sheave	$37.41	$3.83	$72.33		Drive Train
Mike	15 Styrofoam Boards	$312.00		$0.00	12/14/07	Fairing
	3 Al tubes	$65.97	$31.47	$409.44		Fairing
Tanya	Rasp	$17.99		$0.00	12/15/07	Fairing
	Sand paper 40 grit	$3.29		$0.00		Fairing
	Sandpaper 100 grit	$3.29	$2.04	$26.61		Fairing
Tanya	4 5 min Epoxies	$17.96		$0.00	12/16/07	Fairing
	Sand paper 40 grit	$3.29	$1.77	$23.02		Fairing
Tanya	3 5 min Epoxies	$13.47	$1.12	$14.59	12/16/07	Fairing
Mike	Respirators	$13.49		$0.00	12/20/07	Fairing
	Gloves	$12.99		$0.00		Fairing
	3 Spray paints	$10.47		$0.00		Fairing
	2 putty knives	$3.58		$0.00		Fairing
	Taping knife	$7.49		$0.00		Fairing
	Sand paper 220 grit	$4.49		$0.00		Fairing
	Sand paper assorted	$4.49		$0.00		Fairing
	2 SGL Cut Keys	$2.98	$4.99	$64.97		Fairing

Tanya	Squeegees	$4.99		$0.00	12/29/07	Fairing
	Body Filler 1 gal	$24.09		$0.00		Fairing
	Body Filler 1 gal	$45.87		$0.00		Fairing
	Squeegees	$3.39	$6.52	$84.86		Fairing
Tanya	Fiber glass	$18.85		$0.00	12/30/07	Fairing
	Body Filler 1 gal	$26.29		$0.00		Fairing
	Body Filler 1 qt	$19.38	$5.37	$69.89		Fairing
Tanya	2 60 sec Epoxies	$7.98	$0.66	$8.64	12/30/07	Fairing
Tanya	SAE Supermileage Registration	$350.00		$350.00	11/16/07	Competition
Perry	Honeycomb Panels Shipping	$525.00		$525.00	12/17/07	Frame
Mike	Body Filler 1 gal	$33.09	$2.76	$35.85	12/27/07	Fairing
Mike	Gas	$50.77		$50.77	12/19/07	Fairing
Mike	Spreaders	$3.97		$0.00	01/07/08	Fairing
	Bondo	$22.97		$0.00		Fairing
	Spray Paint	$9.36	$3.02	$39.32		Fairing
Mike	4 5 min epoxies	$17.96			12/15/07	Fairing
	40 grit sand paper	$3.29	$1.77	$23.02		Fairing
Mike	4 Needle Roller Thrust Bearings	$10.12			01/16/08	Steering
	10 Washers	$8.50				Steering
	4 Needle Roller Bearings	$35.64				Steering
	Polyurethane Foam	$9.25				Safety
	Quick Disconnect Socket	$11.77				FuelDelivery
	5 ft Nylon Tubing	$11.15				FuelDelivery
	Quick Disconnect Plug	$8.78				FuelDelivery
	Tube 90 degree Elbow	$6.84	$5.50	$107.55		FuelDelivery
Karl	Animal Head	$132.12			10/22/07	Engine
	Spark Plug	$11.17				Engine
	2 Tappet valves	$6.80				Engine
	2 Animal ball bearings	$38.90				Engine
	2 intek animal retainer valves	$12.90				Engine
	boot plug to kill	$2.65				Engine
	cover rocker	$9.50				Engine
	gaket rocker	$3.05				Engine

	gasket - cylinder head plate	$2.30				Engine
	plate - cylinder head	$7.60				Engine
	2 guides - push rod	$2.80				Engine
	2 rod - push	$5.70				Engine
	intek animal screw out	$0.00				Engine
	intake gasket	$2.45				Engine
	animal billet lifter	$23.50				Engine
	animal exhaust gasket	$2.80				Engine
	hot coil 3 hp	$70.00				Engine
	animal gasket copper	$23.80	$11.85	$369.89		Engine
Karl	2 Piston Assemblies	$28.88	$8.95	$37.83	12/11/07	Engine
Karl	Animal Cam	$90.59	$9.50	$100.09	12/14/07	Engine
Karl	Gas for travel	$30.07		$30.07	12/10/07	Travel
Tanya	Steering Wheel	$92.23	$6.17	$98.40	01/24/08	Steering
Tanya	Brake Light	$16.00	$10.00	$26.00	01/29/08	Brakes
Tanya	Rental car for Super	$284.38		$284.38	02/19/08	Travel
Mike	Throttle pedal	$6.49				Engine
	Brake Pedal	$6.49				Brakes
	2 Throttle Cables 90"	$11.50				Engine
	4 Tapered Cable Anchors	$3.52				Brakes
	2 Wire Swivels	$1.36				Engine
	Throttle control rod kit	$8.39				Engine
	Brake control rod kit	$6.79				Brakes
	Rod/control coupler	$1.98	$6.99	$53.51	01/28/08	Brakes
Mike	Water jet cutting	$50.00		$50.00	01/16/08	Frame
Mike	Applicator gun	$69.40				Frame
	Speed Bonder	$24.20	$8.58			Frame
	Mixing nozzels	$9.46	$10.80	$122.44	01/09/08	Frame
Mike	Bystarter Assy. Auto	$36.85				Engine
	Jet, Main	$5.12	$17.73	$59.70	03/05/08	Engine
Karl	Timing belt	$4.03	$10.25	$14.28	03/04/08	Engine
Tanya	spray paint					Fairing
	paper roll			$11.35		Fairing
Tanya	Aluminum angle					Fairing

	5 min epoxy					Fairing
	spray paint			$40.56		Fairing
Tanya	rear axel	$11.95	$9.10	$21.05	03/10/08	Drive Train
Mike	disc brakes			$270.82		brakes
Karl	gas for travel			$40.01		travel
Tanya	eco hotel			$190.00		travel
Tanya	eco hotel			$269.62		travel
Mike	Styrofoam and gloves					
	Epoxy and great stuff			$52.60		Fairing
Tanya	seatbelt					
	racing suit			$177.13		safety
Karl	6 things			$41.98		engine
Tanya	checker			$7.99		safety
Tanya	radioshack			$118.53		safety
Mike	v-belt			$46.18		drive train
Mike	fedex stuff to eco			$8.05		admin
Mike	plywood			$18.28		fairing
Mike	gas			$82.19		travel
Karl	Gaskets			$45.93		engine
Mike	SAE Report Shipping			$44.02		admin
Mike	SAE Report Printing			$23.43		admin
Mike	Painting Supplies			$13.18		fairing
Mike	Lights, oil, etc			$54.10		safety
Mike	Bolts, Tape, Rivets, etc			$33.88		frame
Mike	Bolts, etc			$22.20		frame
Mike	Painting Supplies			$51.40		fairing
Mike	Carburetor Jets			$19.33		fuel delivery
Mike	Tensioner, chain, sprocket			$115.59		drive train
Mike	Clutch			$244.00		drive train
Tanya	SAE hotel			$225.00		travel
Tanya	rental car			$300.00		travel
Tanya	cargo van			$250.00		travel
Shane	reimbursement			$182.00		other
			Total	$6,692.79		

Completed bill of materials

Appendix B

Product Description	Component	Cure	Application	Performance of cured materials
H8600	One part (no mixing)	Room temp	Bonding	Aluminum (Etched) 23.2 KPA
H3292	Two parts(mixing)	Room temp	Bonding	Aluminum (grit blasted) 0.5 KPA to 1 KPA
Red Explosive Charge Bonder	One part(no mixing)	Room temp	Bonding	Aluminum to Aluminum 40 PA
H3000	Two parts	Room temp	Bonding	Aluminum 25 to 30 KPA
H3101	Two parts	Room temp	Bonding	N/A
H3151	Two parts	Room temp	Bonding	Aluminum 3600 psi
LOCTITE® Speedbonder™ H4720	Two parts	Room temp	Bonding	Aluminum 2000 psi
Speedbonder® Adhesive H4800	Two parts	Room temp	Bonding	Aluminum 1810 psi
Speedbonder® Adhesive H4840	Two parts	Room temp	Bonding	Aluminum 2400 psi to 2600 psi
Speedbonder® Adhesive H4850	Two parts	Room temp	Bonding	Aluminum 3000 psi

Produced by our team with references to Henkel Loctite™

Appendix C

Drivetrain Design

Diameter of input gear	Di	3	inches
Diameter of output gear	Do	10	inches
Diameter of rear wheel	Drw	20	inches
Overall gear ratio	GR	0.3	
RPM of engine shaft	RPMshaft	3000	rpm
frequency of oscillation input	Wi	314.1593	rad/sec
frequency of oscillation output	Wo	94.24778	rad/sec
velocity of the input gear	Vi	471.2389	in/sec
velocity of the output gear	Vo	471.2389	in/sec
velocity of the rear wheel	Vrw	53.54987	mph

Constant gear ratio calculations while varying RPM

1.5	1.5	1.5	1.5	1.5	1.5	1.5
6	6	6	6	6	6	6
20	20	20	20	20	20	20
0.25	0.25	0.25	0.25	0.25	0.25	0.25
3000	2750	2500	2250	2000	1750	1500
314.1593	287.9793	261.7994	235.6194	209.4395	183.2596	157.0796
78.53982	71.99483	65.44985	58.90486	52.35988	45.81489	39.26991
235.6194	215.9845	196.3495	176.7146	157.0796	137.4447	117.8097
235.6194	215.9845	196.3495	176.7146	157.0796	137.4447	117.8097
44.6249	40.90615	37.18741	33.46867	29.74993	26.03119	22.31245

Constant RPM calculations while varying gear ratio

1.5	1.5	1.5	1.5	1.5	1.5	1.5
6	7	8	9	10	11	12
20	20	20	20	20	20	20
0.25	0.214286	0.1875	0.166667	0.15	0.136364	0.125
3000	3000	3000	3000	3000	3000	3000
314.1593	314.1593	314.1593	314.1593	314.1593	314.1593	314.1593
78.53982	67.31984	58.90486	52.35988	47.12389	42.8399	39.26991
235.6194	235.6194	235.6194	235.6194	235.6194	235.6194	235.6194
235.6194	235.6194	235.6194	235.6194	235.6194	235.6194	235.6194
44.6249	38.24991	33.46867	29.74993	26.77494	24.34085	22.31245

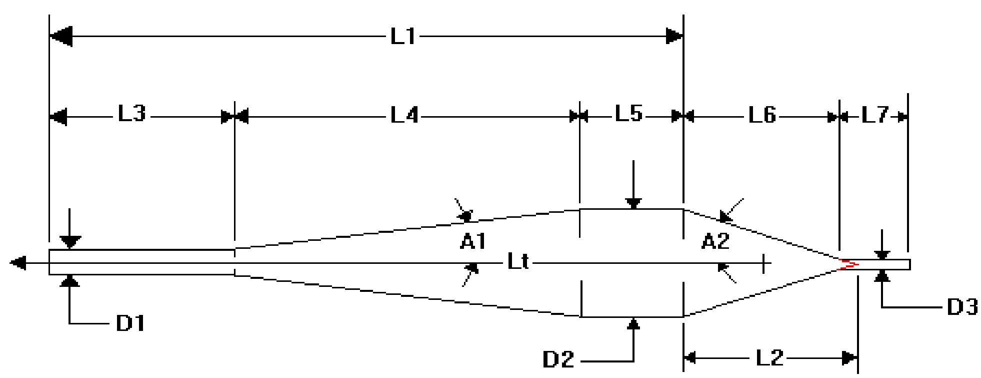
1.5	1.5	1.5	1.5	1.5	1.5	1.5
7	7	7	7	7	7	7
20	20	20	20	20	20	20
0.214286	0.214286	0.214286	0.214286	0.214286	0.214286	0.214286
3000	2750	2500	2250	2000	1750	1500
314.1593	287.9793	261.7994	235.6194	209.4395	183.2596	157.0796
67.31984	61.70986	56.09987	50.48988	44.8799	39.26991	33.65992
235.6194	215.9845	196.3495	176.7146	157.0796	137.4447	117.8097
235.6194	215.9845	196.3495	176.7146	157.0796	137.4447	117.8097
38.24991	35.06242	31.87493	28.68743	25.49994	22.31245	19.12496

1.5	1.5	1.5	1.5	1.5	1.5	1.5
6	7	8	9	10	11	12
20	20	20	20	20	20	20
0.25	0.214286	0.1875	0.166667	0.15	0.136364	0.125
2750	2750	2750	2750	2750	2750	2750
287.9793	287.9793	287.9793	287.9793	287.9793	287.9793	287.9793
71.99483	61.70986	53.99612	47.99655	43.1969	39.26991	35.99742
215.9845	215.9845	215.9845	215.9845	215.9845	215.9845	215.9845
215.9845	215.9845	215.9845	215.9845	215.9845	215.9845	215.9845

1.5	1.5	1.5	1.5	1.5	1.5	1.5
8	8	8	8	8	8	8
20	20	20	20	20	20	20
0.1875	0.1875	0.1875	0.1875	0.1875	0.1875	0.1875
3000	2750	2500	2250	2000	1750	1500
314.1593	287.9793	261.7994	235.6194	209.4395	183.2596	157.0796
58.90486	53.99612	49.08739	44.17865	39.26991	34.36117	29.45243
235.6194	215.9845	196.3495	176.7146	157.0796	137.4447	117.8097
235.6194	215.9845	196.3495	176.7146	157.0796	137.4447	117.8097

Drive train calculations Shigley's Mechanical Engineering Design Eighth Edition

Tuned Exhaust		
Tuned Length		
Exhasut open period (°)	Exhaust temperature K	Wave speed (m/s)
3.801327111	922	608.6974334
Crankshaft speed (rpm)	Tuned Length (m) **Lt**	Tuned Length (in) **Lt**
2500	0.925543222	36.43863666
Inlet pipe length (in) L3		
10		
Outlet pipe length (in) L7	**Outlet pipe diameter (in) D3**	
7.44	0.62	
Diffuser Portion		
Exhaust port diameter (in) **D1**	Contrast ratio	Diffuser diameter (in) **D2**
1	6.25	2.5
Diverge taper (rad) **A1**	Diverge taper (°) **A1**	
0.13962634	8	
Baffle Cone		
Diffuser diameter (in) **D2**	Angle of convergence (rad) **A2**	Angle of convergence (°) **A2**
2.5	0.27925268	16
Inlet to center of cone (in)	length of cone	
2.179634027	5.336527292	



Theoretical tuned exhaust system for the Briggs Engine

Coefficient of Rolling Resistance	**Mass of vehicle (lbm)**	**Drag Coefficient**	**Area of vehicle (ft^2)**
0.0025	200	0.015	4.528
	210	0.016	4.528
	220	0.017	4.528
	230	0.018	4.528
	240	0.019	4.528
	250	0.02	4.528
	260	0.021	4.528
	270	0.022	4.528
	280	0.023	4.528
	290	0.024	4.528
	300	0.025	4.528
	310	0.026	4.528
	320	0.027	4.528
	330	0.028	4.528
	340	0.029	4.528
	350	0.03	4.528
Vehicle Speed (ft/sc)	**Air Density (lb/ft^3) (STP)**	**Total Load on Vehicle (hp)**	**Total Load on Vehicle (Watt)**
10	0.073	0.013	9.976
11		0.015	11.527
12		0.018	13.179
13		0.020	14.933
14		0.023	16.788
15		0.025	18.744
16		0.028	20.802
17		0.031	22.962
18		0.034	25.223
19		0.037	27.587
20		0.040	30.053
21		0.044	32.621
22		0.047	35.292
23		0.051	38.066
24		0.055	40.942
25		0.059	43.921

Total Road load on vehicle Internal Combustion Engine Fundamentals

Volumetric efficiency

volumetric efficiency is a ratio (or percentage) of what volume of fuel and air actually enters the cylinder during induction to the actual capacity of the cylinder under static conditions. Therefore, those engines that can create higher induction manifold pressures - above ambient - will have efficiencies greater than 100%. Volumetric efficiencies can be improved in a number of ways, but most notably the size of the valve openings compared to the volume of the cylinder and streamlining the ports. Engines with higher volumetric efficiency will generally be able to run at higher RPM and produce more overall power due to less parasitic power loss moving air in and out of the engine.

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Volumetric flow rate = Q

Q=AV

V = Velocity

A = Area

Test proposal of volumetric of efficiency

1.) Calculate theretical flow rates of air fuel mixture at a given RPM

2.) Measure ehast flow rate using Anemometer

3.) Calculate volumetric flow rate using Q=A*V

4.) Compare actual Vs. theoretical to get volumetric efficiency

5.) Perform this test when engine modifications are made.

148 CC displacment (theroretical)

RPM	Effective stroke	air flow (CC)/S	air flow in^3/S	area of exhaust (in^2)	V(in/S)
1800	7.5	1110.00	67.71	2	33.855
2200	9.17	1356.67	82.75666667	2	41.37833333
2600	10.83	1603.33	97.80333333	2	48.90166667
3000	12.50	1850.00	112.85	2	56.425
3400	14.17	2096.67	127.8966667	2	63.94833333
3800	15.83	2343.33	142.9433333	2	71.47166667
4200	17.50	2590.00	157.99	2	78.995
4600	19.17	2836.67	173.0366667	2	86.51833333
5000	20.83	3083.33	188.0833333	2	94.04166667

148 CC displacment (actual)

RPM	measured V(in/S)	area (in^2)	air flow in^3/S (Q)	actual /thero	volumetric efficiency %
1800	32	2	64	0.945207503	94.52
2200	40	2	80	0.966689491	96.67
2600	4	2	8	0.081796803	8.18
3000	50	2	100	0.886132034	88.61
3400	55	2	110	0.860069327	86.01
3800	68	2	136	0.951425973	95.14
4200	72	2	144	0.911450092	91.15
4600	81	2	162	0.936217757	93.62
5000	90	2	180	0.957022596	95.70

Volumetric efficiency Internal Combustion Engine Fundamentals