Supermileage Team - Urban Concept Competition Vehicle
Chassis Design Report

Conceptual Chassis Designs

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Abstract

This research and design project was proposed by the Cal Poly Supermileage team. They have requested for our team to design an Urban Concept car, a new category of vehicles to compete for fuel efficiency in the Shell Eco-marathon in 2010. The primary focus of our team is in the design and construction of a chassis the Supermileage team can use for the 2010 competition in Houston, TX. This design must meet dimensional and functional requirements set by Shell while being designed to maximize efficiency in the competition.

A carbon fiber backbone chassis was selected to be the best in minimizing weight while having sufficient strength with a safety factor of over 40, and minimal deflection from various loads and torsional testing of less than 0.1" in our expected worst case scenario. This design also allowed for the chassis to be easily modified by future students. After construction, testing of the main beam showed that our calculated deflection showed 22% more than our experimental, .088" and .068" respectively.

Chapter 1 Introduction

1.1 Sponsor Background

The Cal Poly Supermileage team was revived in 2005, after a 12-year hiatus, by seven students as a senior project. They designed and built a fuel efficient vehicle to compete in the 2006 SAE Supermileage Competition in Marshall, Michigan where they achieved a fuel efficiency of 861 mpg. This placed them in sixth place out of a field of twenty, which is respectable for their first competition. In 2007, the energy company Shell held its Eco-marathon competition in Fontana, California where the team competed. They earned a first place victory with 1907 mpg, more than doubling their achievement the prior year. In 2008, the team competed again in the Shell Eco-marathon and earned a second place victory with 2752 mpg. Today, even though most of the original members have graduated and left the team, the team has expanded to approximately 15 members. Figure 1 shows Cal Poly's 2008 Supermileage team and their prototype car.

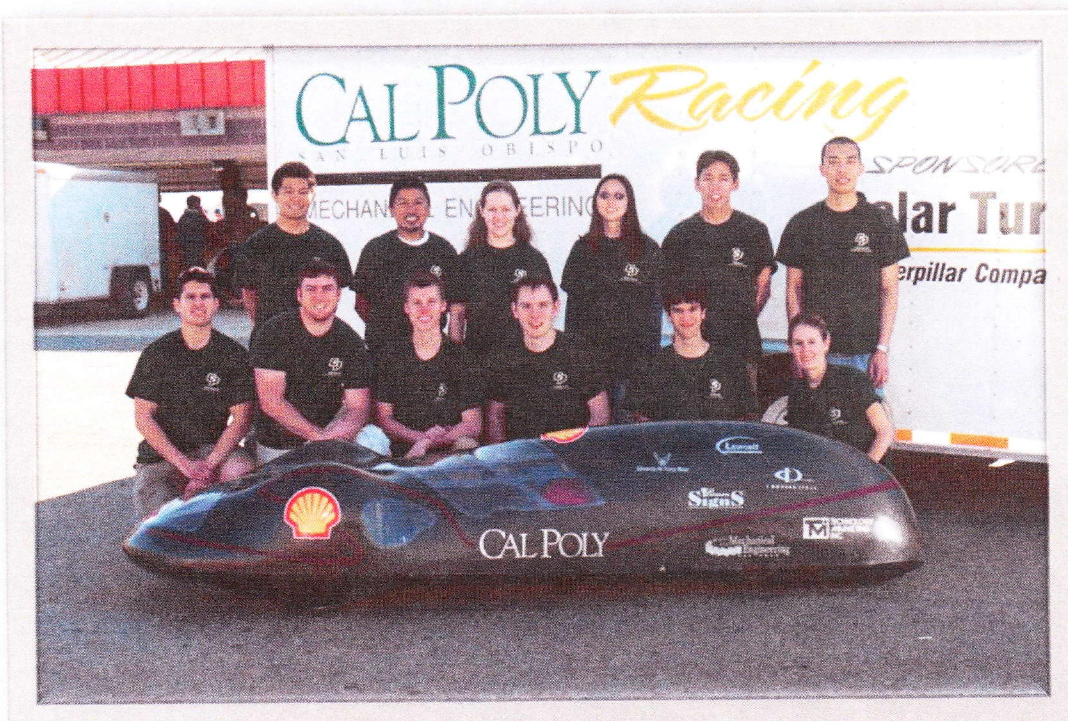


Figure 1 - 2008 Supermileage Team and Prototype

1.2 Problem Definition

A vehicle needs to be built to compete in the 2010 Shell Eco-marathon Urban Concept group. For 2009, the Supermileage team has proposed two senior projects to the mechanical engineering department that involve the design of an Urban Concept car, a new category of vehicles to compete for fuel efficiency in the Shell Eco-marathon. The two projects are the selection of an engine for the Urban Concept car and the construction of a chassis for it. The design and construction of the chassis was assigned to five Cal Poly mechanical engineering seniors: Andrew Allport, Kevin Braico, Kevin Charles, Wei Kyi, and William Lai.

Shell is hosting the Urban Concept group for the Eco-marathon competition for the first time in America in 2009. This group differs from the original Prototype group in that the vehicles are more practical. The Urban Concept vehicles, an example of which is shown in Figure 2, must meet dimensional and functional requirements that make them more acceptable for road use. This includes having front and rear lights, bumpers, side mirrors, and a windshield that allows the driver a full range of visibility from the cockpit amongst many other criteria. Appendix A contains the specifications the car needs to meet.

Figure 2 - The M-112 Urban Concept Vehicle

1.3 Objective & Specification Development

The Supermileage team intends to have a vehicle to compete in the Urban Concept group at the 2010 Shell Eco-marathon. They have stated that their primary goal is to have a vehicle completely ready for the competition. Since it will be their first time competing in the Urban Concept group, they are using this as a learning experience to prepare them to be a future leader in the group. They want to ensure a working vehicle first, and will refine it in the coming years to positively influence fuel efficiency.

It is the primary objective of Conceptual Chassis Designs to design and construct a chassis for the Supermileage team that will be used in the 2010 Shell Eco-marathon. Through careful material selection, structural design, and strength/stiffness to weight optimization a chassis will be delivered that will allow the team to be top competitors at the race. The design of this project will be regulated by the 2009 Shell Eco-marathon rulebook. We will collaborate closely with the Supermileage and engine design teams to ensure a cohesive design of the overall vehicle that may be modifiable for possible rule changes in the future.

Shell provides a comprehensive rulebook to all teams developing vehicles for that year's competition. While some constraints are very specific, Shell states that it purposely leaves other areas vague to influence creativity in design. We will meet all specifications outlined in the currently published 2010 rulebook pertaining to the chassis and body while also considering integration with all other subsystems. While it is not required by the current rules, the Supermileage club has expressed an interest in integrating a suspension system for the 2010 car. The chassis and body will be designed to function with the suspension system designed by the Supermileage club.

Attached in Appendix A is a specification chart developed from the Shell rulebook that pertain to the chassis and body design. These rules specifically regulate dimensions including length, width, and height. They also impose a maximum weight, a minimum wheelbase, and minimum track widths. There are also regulations regarding driver safety. The Supermileage team has given us freedom to develop within these requirements until they impose further design requirements within the rule's specifications. Also in this list are some specifications that have already been developed by the team including suspension integration requirements. Some of these specifications are subject to change as the club continues to develop their designs. This point again emphasizes the importance of close collaboration with the club throughout the project, developing and modifying designs to meet any targets that they choose to establish as

the process progresses. Throughout our design process, the club will be developing the crucial subsystems, and system integration will be a focus of all involved in the project.

Beyond meeting the rules, our chassis will need to meet strength and stiffness requirements. The chassis will need to be strong enough to carry the loads of the driver and subsystems, and tolerate the extra forces imparted by handling dynamics. The chassis will need to be stiff enough to act as a stable operational platform, and not deflect so much as to damage itself or any other components. The chassis needs to be stiff longitudinally to decrease deflection when the driver enters the vehicle. The frame also needs to be torsionally stiff so that it does not overly deflect under driving loads.

Chapter 2 Background

2.1 Existing Products

Shell has hosted the Eco-marathon since 1985 in Europe and 2007 in America. The Urban Concept group began in 2003 in Europe and is new to America for 2009. While many of these cars have been raced in Europe, there is not a lot of available information about them. However, the results of the 2008 competition are known.

<u>2008 Urban Concept Results:</u>

Haagse Hogeschool (NL / 1994 mpg)
NTNU (NOR / 1714 mpg)
University of Sakraya (TUR / 1065 mpg)
Lulea University (SE / 703 mpg)
FEUP Porto (PT / 684 mpg)

For the 2008 Shell Eco-marathon competition in Europe, the winners of the Urban Concept class were the Dutch *"HydroCruisers"*. Their vehicle, shown in Figure 3, was powered by hydrogen fuel cells and created a new world record of 1994 miles per gallon, beating the old record by over 600 mpg. The most economic internal combustion engine was from the Swedish *"Baldos"* team. Using their IC engine, they reached 703 mpg of Shell petrol. Iran sent a team using a lawn mower engine and a custom computer. They achieved 402 mpg, placing them at 90th overall. A French team created a concept vehicle primarily made from bamboo. This allowed them to have a light yet strong frame, a common goal among teams. Unfortunately due to their heavy 125 cc engine running on Liquefied petroleum gas (LPG), this design only achieved 174 mpg.

Figure 3 - Dutch "HydroCruisers" at European 2008 Eco-marathon

2.2 Current State of the Art

While the Eco-marathon competition is for purpose built vehicles only, a main goal is to see innovations used on the track deployed on the roads in passenger vehicles. On the market today, one of the best known fuel efficient vehicles is the Toyota Prius. With a combined EPA rating of 46 miles per gallon (mpg), few cars can beat the Prius in terms of fuel economy while retaining as many features and passenger capacity. A relatively new company, Loremo, has plans to introduce a series of fuel efficient cars in the US by 2010. One of their prototypes, the Loremo LS, is shown in Figure 4. The company mainly focuses on the use of lightweight materials and to create an aerodynamic body to reduce drag. Power will come from either a 700cc or 850cc Otto engine for its compact dimensions and low weight. The car is aiming to get over 100 mpg with a seating capacity of two adults and two children.

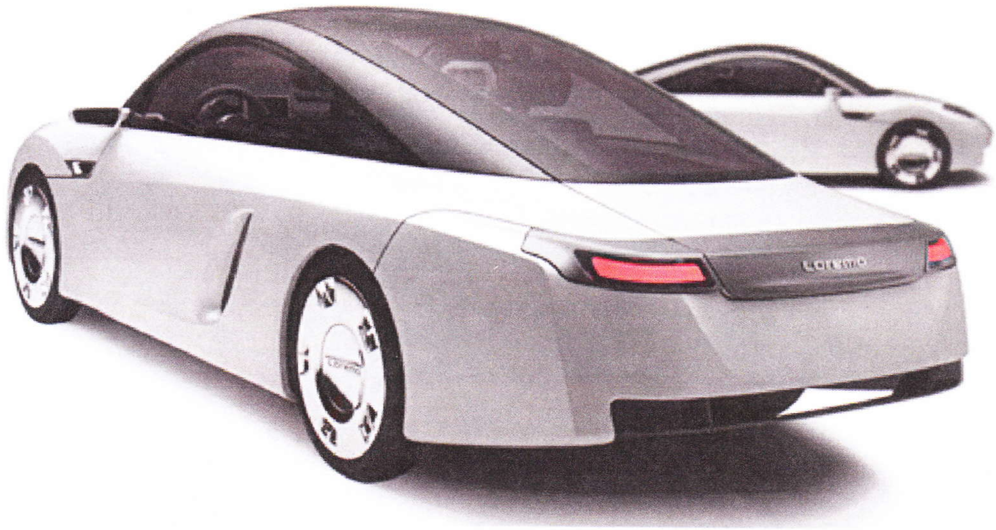


Figure 4 - Loremo LS Prototype

While these vehicles represent current developments in fuel efficient transportation, their relatively low fuel consumption is not ground breaking. Other vehicles that have been on the roads for years have used a low tech, low feature, and low weight approach to achieve outstanding fuel economy. The 1991 Geo Metro XFI using a 1 liter, 3 cylinder motor has been rated at the same 46 combined mpg as the 2009 Toyota Prius at a fraction of the initial purchase cost for the consumer. While technology and simplicity lead to lower fuel consumption, it is the goal of the Shell Eco-marathon to push this concept even further with the Urban Concept group offering a view of what may soon be incorporated in road going vehicles.

2.3 List of Applicable Standards

Though our design is not regulated by any codes or standards and there is no design review, we do need to design according to the Eco-marathon rules. Articles from the 2009 rulebook are attached in Appendix A.

2.4 Possible Frame Types

There are many automotive chassis designs for us to investigate to give us ideas for our own design. The main types we have been considering are a composite monocoque, a skeleton space frame, and a backbone chassis. All three can provide the required structural integrity while remaining lightweight. However, they each have their advantages and disadvantages.

The monocoque's advantages come from the fact that it is a continuous shell. It is the safest option because the driver is enclosed in the structural shell. A benefit of the monocoque is that it can incorporate the body and frame into one shell. It can be designed so that weight is saved by eliminating the need for a fairing or fairing sections. There are also problems associated with the monocoque that aren't encountered with the other designs. Composite materials do not handle point loads well, such as those found where suspension parts mount to the frame. Hard point inserts are used to deal with mounting problems, but these increase part count and weight. Fatigue is another downside to composites, where a crack can propagate and cause failure. The worst part is if any failure occurs, additional layers of material must be added to do a repair layup, which adds weight. Some of the other drawbacks are high cost for materials and difficult manufacturing. An example of a monocoque chassis is shown in Figure 5.

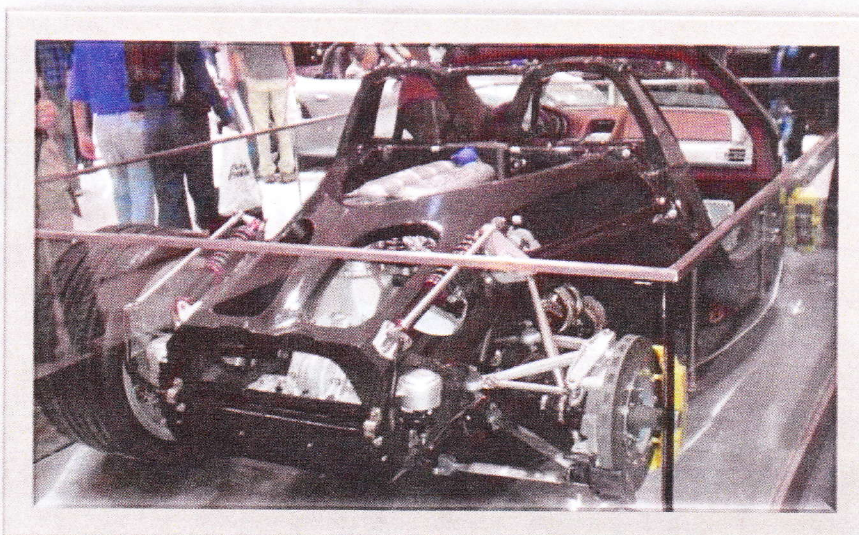


Figure 5 - Porsche Carrera GT Carbon Fiber Monocoque

The space frame design is composed of a truss structure with a body covering it for aerodynamic and appearance purposes. Many of the weaknesses of the monocoque are not present with the space frame. Accessibility is increased because it is possible to reach through the frame. Repairs are easier because tubes can be removed and reattached without increasing the weight. This also allows later revisions to the design if necessary, while it is too late to change the monocoque once the mold is made. Figure 6 shows an example of a space frame chassis.

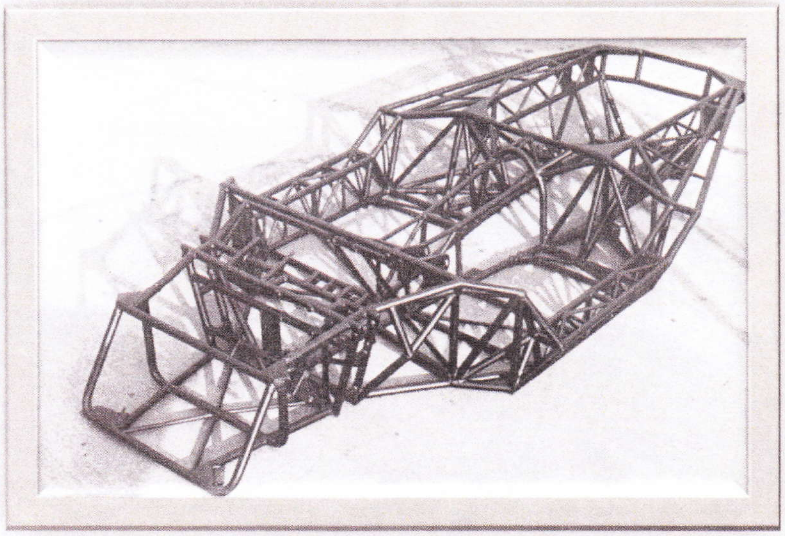


Figure 6 - Lamborghini Countach Space Frame

The last type of frame we have looked at is the backbone design. It is a beam running down the middle of the car upon which all other components are attached. It has many positive attributes due to its simplicity. It would be the easiest to manufacture because it has fewer parts than the space frame design. The accessibility of the car would be greater with a backbone frame because there is no truss in the way. Figure 7 shows an example of a backbone chassis.

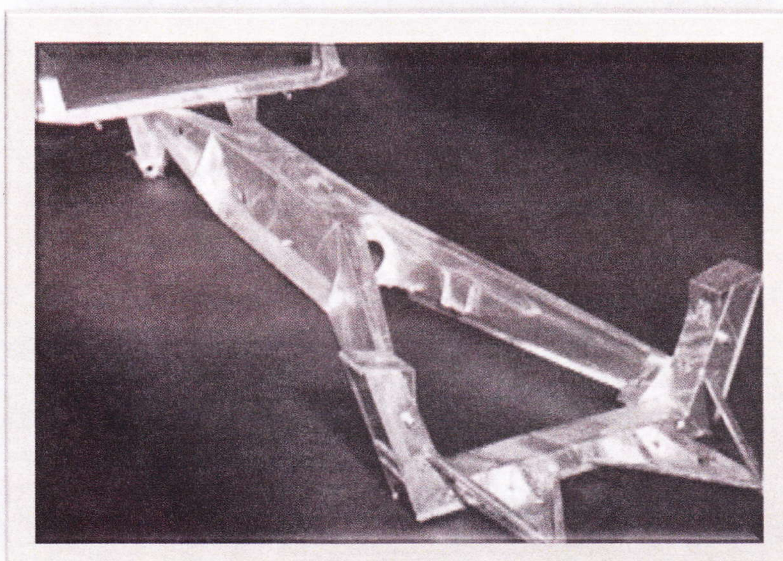


Figure 7 – Lotus Elan Backbone Frame

The advantages and disadvantages of each chassis type are summarized in Table 1. From this table, we found that all three designs are feasible choices for us to consider as our final design. In the following chapters, we will discuss the advantages and disadvantages in greater detail and decide which design is best suited for this project.

Table 1 - Chassis Comparison

	Advantages	Disadvantages
Monocoque	• Safety • Few parts • High Stiffness to weight • Can combine body and frame	• Repairs • Point loads • Fatigue • Cost • Analysis • Accessibility • Manufacturing
Space Frame	• Accessibility • Reparability • Cost • Analysis	• Manufacturing • Many parts
Backbone	• Manufacturing • Reparability • Cost • Analysis • Few parts • Accessibility	• Side impact • Handling dynamics

2.5 Stiffness and Strength

All automotive chassis must be designed adequately for stiffness and strength. Generally, stiffness is the driving consideration when creating a chassis. This is because the structure must be sufficiently stiffer than the suspension to provide adequate handling. Adequate handling can be attained by making the chassis stiff enough so that roll stiffness between sprung and unsprung masses are due almost entirely to the suspension. If a chassis is not stiff enough, it will deflect and absorb energy unpredictably which may make the car's handling less precise. Also, a chassis must be sufficiently stiffer than the suspension for suspension tuning to work. An example of where this can be a problem is if you build a frame out of titanium; it can provide the same amount of strength as a steel frame at less weight

because titanium's specific strength is higher. The titanium frame can be built with less material without breaking due to its higher specific strength; however, the specific modulus of elasticity of titanium is equal to steel, so the titanium frame would not meet the same stiffness requirements. Strength must also be considered especially at joints and welds where failure is more of a concern than deflection.

2.6 Resistance Concerns

One major area that affects fuel mileage is overcoming resistance to motion. There are four major components that fit into this category; inertia, driveline friction, tire rolling resistance, and air drag. At low speed stop-and-go city driving, inertia and driveline friction accounts for up to 80% of the vehicle's total resistance. Our focus will be on minimizing these resistances as much as possible.

Inertia is an object's resistance to change its state of motion. This includes getting an object at rest to move and getting a moving object to stop. When a vehicle is at rest, it must overcome friction forces to begin moving. When a vehicle is moving, it must overcome friction and drag forces to keep it moving. Since static friction is greater than dynamic friction, at low speeds it takes more force to move an object at rest than it does to keep it moving at a constant rate. And because our vehicle will primarily be driven at low speeds, resistance forces due to inertia, which are proportional to weight, will be a large concern. We will aim to lower these resistances by reducing the weight of the vehicle.

Driveline friction comes from the friction losses incurred from the engine and transmission parts including bearings, shafts, gears, etc. These losses come directly from the design of the engine and transmission, and we will depend on the Supermileage engine team to pick an appropriate and optimal engine for the vehicle.

Tire rolling resistance is the force required to move the tire forward. This force is directly proportional to the product of the rolling resistance coefficient and load on the tire. The load on the tire depends on the weight of the vehicle, which again demonstrates the importance of weight for this project. The tires used on the Cal Poly Supermileage Prototype vehicle had a tire rolling resistance coefficient of only 0.001. This coefficient for new consumer vehicles is in the range of 0.007 to 0.014. One method of reducing rolling resistance is to increase the pressure of the tire. This reduces the contact patch between the tire and the road which causes less rolling resistance. The disadvantages to this are that the vehicle has less grip and stability, which has a negative impact on safety. It also causes an increase in tire wear. But

in low speed driving conditions, vehicle handling and dynamics are less significant in terms of safety, and may be sacrificed for better mileage. Figure 8 shows the effects of weight on the vehicle's rolling resistance.

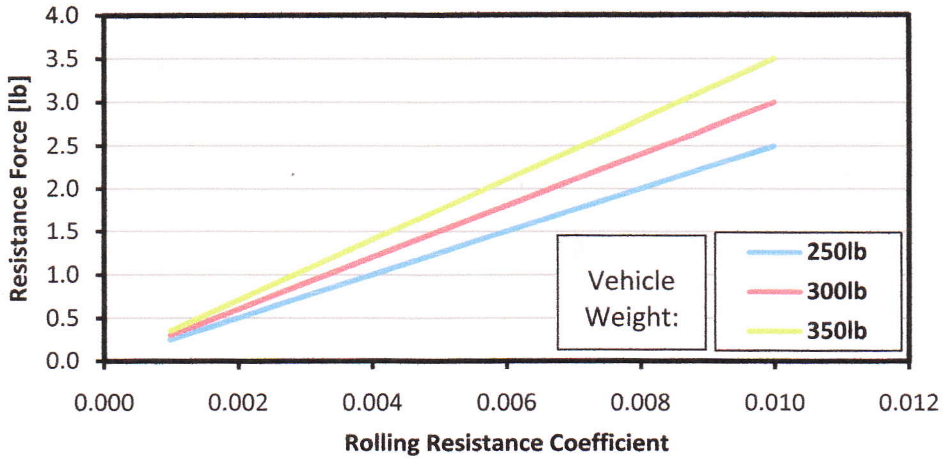


Figure 8 - Rolling Resistance Effects

2.7 Material Consideration

There are many types of structural materials to choose from when designing the chassis. The two most important factors in our chassis material selection will be to minimize weight yet maximize strength. Cost will also be a driving factor in choosing what will be used for this initial design.

One of the possible materials being considered in the design of the chassis is wood. Wood has one of the lowest densities of structural materials being considered, but also has some of the lowest strength properties as well. This may still be a viable choice since the loads on the chassis will be very low, approximately 300lbs with driver, depending on final chassis design. An aluminum alloy may be selected as another chassis material. Aluminum alloys have up to five times the density of wood but can also yield strength up to ten times stronger. Figure 10 shows a chart of the density of some materials and their strength. Table 2 shows a comparison of the properties of several different materials.

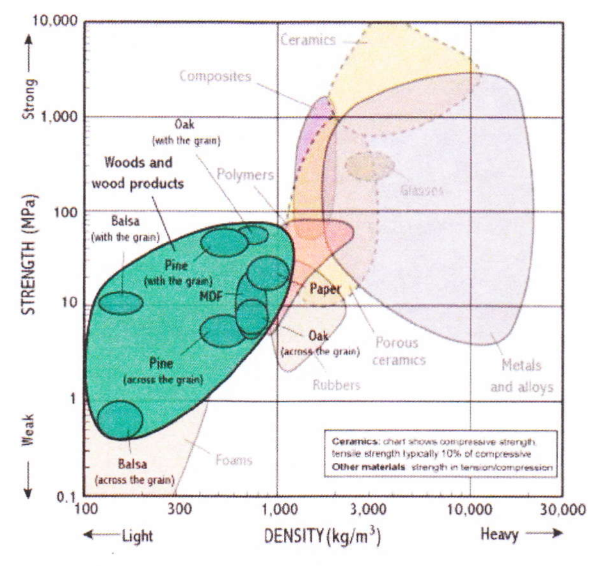


Figure 10 - Density of Materials Compared to Strength

Table 2 - Material Properties

Material	Yield Stress	Ultimate Stress	Specific Strength	Modulus of Elasticity	Specific Stiffness	Density	Cost
	(MPa)	(MPa)	(kN·m/kg)	(Gpa)	(Gpa-m3/kg)	(×1000 kg/m^3)	($)per 6' Lengths
Aluminum Alloy (6061)	276	100 - 550	222	69	24.5-26	2.64 - 2.8	6'x1" Round bar $29.96
Carbon Fiber	X	**4600-5650**	**2457**	142	**811**	**0.175**	Carbon Tube .750"OD x .540"ID x 6' $148.50
4130 Steel Alloy	345	670	254	200	25.47	7.85	Steel Round Tube 1.000" OD, .250" Wall Thickness, 6' Length $86.33
Fiberglass	X	675	1307	25	100	0.25	X
General Steel	**280 – 1600**	340 - 1900	231-284	**205**	26.11	7.85	X

The most likely material to be used for the chassis will be carbon fiber. It has less tensile strength than Kevlar but a higher tensile strength than fiberglass. Carbon fiber is on average approximately three times stiffer than either fiberglass or Kevlar, depending on the type. It is more expensive than fiberglass but less expensive than Kevlar.

Chapter 3 Design Development

3.1 Concept Generation

After studying the available information on the vehicles currently being raced in the Shell Eco-marathon Urban Concept group and other types of chassis designs, we developed a list of possible chassis types and materials to be considered for each one. These chassis types were tabulated in a decision matrix against criteria relevant to the project. Some of the more important criteria included chassis weight, safety, ease of subsystem integration, and ease of modification. These criteria were then weighted on their relative importance to the overall design. The decision matrix can be seen in Appendix B.

After comparing the chassis selections through the decision matrix, we found many of the possible designs were still ranked relatively close to each other; however, the matrix was good for eliminating designs that were relatively weak. We were able to eliminate the concepts that used wood for the material based on the relative strength and stiffness of the material compared to chassis that used metal and composite materials. This process also helped in weeding out other designs that scored much lower than our primary top designs such as the metal pan or flat frame and the sheet metal monocoque. The pan frame did not offer many advantages when compared to the leading designs and the sheet metal monocoque offered few advantages when compared to the effort it would take to construct.

While the composite monocoque did not do poorly on the decision matrix, we were concerned with the adaptability of the design. If there was a need for a major redesign of any subsystem or component, we thought that it may be too difficult to modify a monocoque structure fabricated primarily as a single unit. If any major geometry needed to be changed, we didn't want the club's only option to be cutting into major portions of the frame, compromising the structure and adding unnecessary weight. Or even worse, we didn't want them throwing it away and starting from scratch. After working the composite monocoque through the decision matrix, we were left to rethink the composite chassis option. Due to composite properties, specifically carbon fiber's high strength and stiffness to weight ratio, we wanted to keep it as a viable option. A conceptual chassis design we developed was to make a modular composite frame with simple primary components that would leave more options for subsystem integration and redesign.

After considering an original list of ten chassis designs, we have narrowed the selection to three. The top three choices we came up with are a space frame, a backbone, and a modular

composite chassis. We are also considering the use of a hybrid chassis that incorporates positive design attributes from the above concepts.

3.2 Conceptual Designs

The Space Frame

The main advantages of the space frame chassis are that it is rigid and lightweight. The rigidity comes from the structure being composed of trusses. Since the chassis will be made out of tubular metals or composites, there will be weight savings as opposed to using entire sheets of metal. Because these two criteria are top priorities in the design of the vehicle, the space frame is a viable approach. Other advantages of the space frame are its ease of modification, simplicity to repair, and easy subsystem integration. The main disadvantage to the space frame is that it has more complex joints than other chassis designs to analyze. This makes the design process more demanding than other chassis. Also, the construction of the space frame is more time consuming, and sometimes more costly than other chassis due to these joints. The following two figures are concept models of the space frame design.

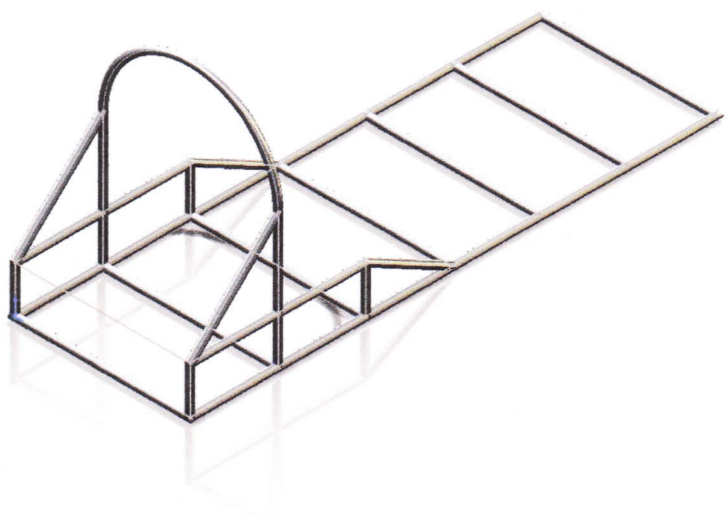


Figure 10 - Simple Concept Space Frame

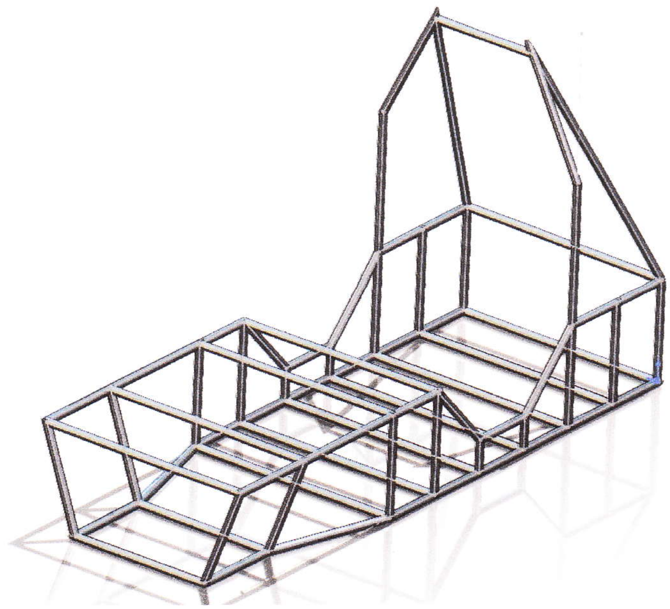


Figure 11 - More Complex Concept Space Frame

The Backbone Frame

The backbone chassis' main advantage is its simple shape. This makes it easier to design, build, and modify. Typically, vehicles with a backbone chassis weigh more and are less safe than monocoques and space frames because cars must withstand high driving loads. In the Shell Eco-marathon, the vehicle will not have to support very high dynamic loads and our target weight for the car with the driver is only 350 pounds, so a backbone chassis will be able to handle these conditions well. Since the chassis does not have to be extremely strong or stiff for this competition, the backbone design may be just as low weight as other options while offering the most simplicity. Safety is a big concern, so there will be a roll bar which mounts to the side of the car to protect the driver from any possible collisions. A model of the backbone concept is shown in Figure 13.

Figure 13 - Backbone Concept Model

The Modular Composite Frame

The third conceptual design, which was developed after doing the initial decision matrix, is the modular composite chassis. We came about this concept while trying to create a composite structure that was easier to modify and adapt than a composite monocoque. A traditional composite monocoque uses few, large molded components (often only one or two pieces) as the primary structure of the vehicle. While this approach typically offers a stiff and lightweight structure, it is hard to change major geometries in the design. Our modular composite structure would be constructed from multiple simple segments that would be joined together. These segments might include a floor pan, main side beams that would take up the primary bending load, a roll bar, and a front dash panel. These pieces would be designed so that they could easily be reconfigured or modified to allow for subsystem integration changes. Also, the piecewise construction could be disassembled and replacement pieces could be fabricated and installed if necessary. This would allow for the reuse of certain components is a major design change was necessary whereas a whole new chassis would need to be constructed when using a tradition monocoque. Also, the large, flat areas in this design would accommodate subsystem integration and mounting well. Figure 14 is a sketch of the modular composite frame concept.

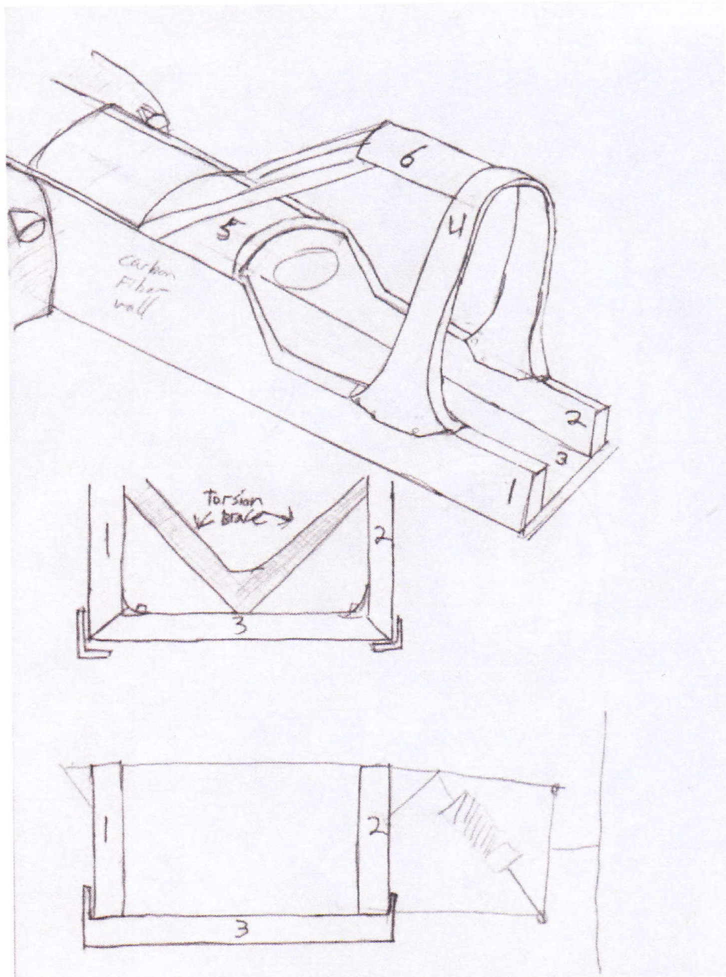


Figure 14 - Modular Composite Frame Concept Sketch

3.3 Concept Elimination

Modular Composite Frame

After detailed analysis was performed on the modular composite frame, we were able to eliminate it by comparing the torsional and bending stiffness with other chassis designs. Seen in Figure 14, the modular composite design has an open U-shape section by the driver's entrance and exit region. According to Shigley's *Mechanical Engineering Design* textbook, by Budynas and Nisbett, open thin-walled sections in torsion should be avoided in design because the shear stress and angle of twist are inversely proportional to wall thickness. For these areas, the stress and angle of twist can become substantially large.

For the purpose of eliminating concept designs, we used carbon fiber for our calculations. The in-plane shear modulus of rigidity for carbon fiber is 4.19E6 psi, which was

calculated using equations shown in Appendix D from the text called *Analysis and Performance of Fiber Composites* by Agarwal. Based on the results, the open section U-shape geometry has a shear stress of 1.36E6 lb_f/in^2 with an angle of twist of 400 degrees while the rectangular hollow shape geometry has a shear stress of 172.3 lb_f/in^2 with an angle of twist of 0.06 degrees. This confirms with Shigley's *Mechanical Engineering Design* text that we should avoid open sections such as those found in our modular composite chassis design, where the open section area around the driver is the weakest for torsion. More material can be added on this weak section to reduce the shear stress and large angle of twist, but this will add unwanted weight to the chassis. Thus, the modular composite design is eliminated after comparing the stress with the rectangular hollow backbone beam chassis design.

Space Frame

We decided not to pursue a tubular space frame design after initial analysis indicated it would be heavier than the backbone design. The frame was drawn in SolidWorks as a 3D sketch, with emphasis on layout packaging and triangulation. Next, the space frame was analyzed using a wireframe beam finite element model in ABAQUS CAE, shown in Figure 15. The purpose of the test was to see if the created frame designs would meet our target stiffness requirements. Our target chassis stiffness requirement was determined to be 427 ft-lbs/deg, which is an order of magnitude higher than the suspension stiffness.

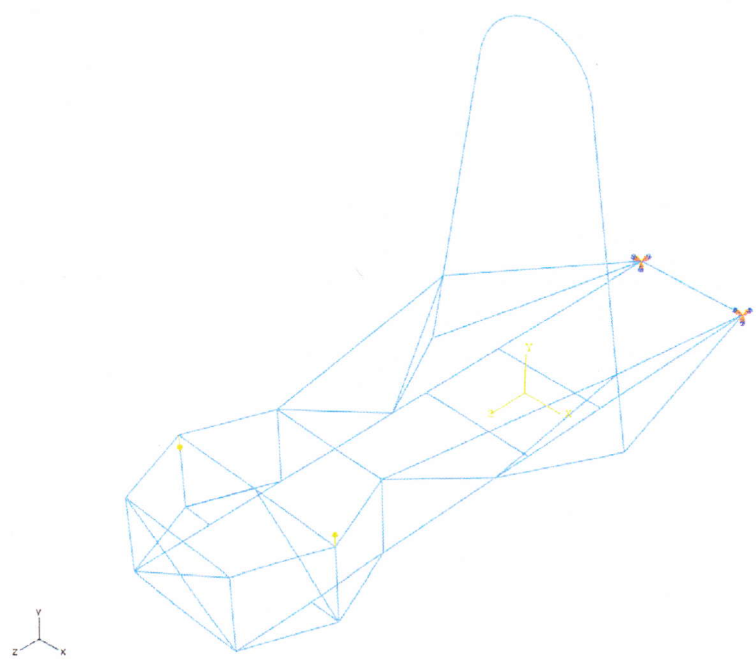


Figure 15 - Space Frame Model in ABAQUS

To measure this in ABAQUS, the rear suspension mount points were fixed and a force couple was applied to the front suspension mount points. Each force is 213.3 pounds and positioned 12 inches from the centerline of the chassis. This creates a 427 ft-lb torque around the center line. The chassis must twist less than 1 degree under this load to meet our specification. We measured this twist by looking at the vertical displacement, U2, at the front suspension mounting points. These points rotate about the center on 12 inch long lever arms, so the angle of twist is $\varphi = \sin^{-1}\frac{U2}{12\ in}$. A schematic of this is shown in Figure 16.

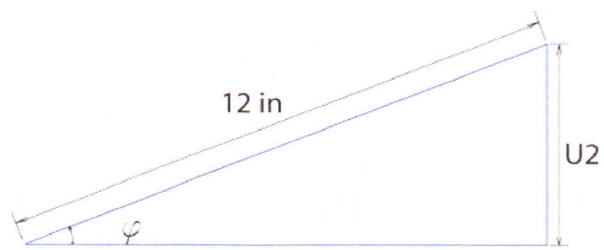


Figure 16 - Suspension Mount Displacement

To verify the accuracy of the FEA results, we modeled a simply supported beam using the same modeling methods used to make the space frame. A 100 pound load was applied to the center of the beam, and the resulting displacement was only 0.6% off from hand calculations using the equation derived using Euler-Bernoulli beam theory. This confirmed that modeling wireframe beam structures using our methods will produce accurate solutions. Figure 17 shows the magnitude of displacement in the vertical direction when loaded with the force couple described above. Red indicates a high positive displacement, blue indicates a high negative displacement, and green means there was not much vertical displacement.

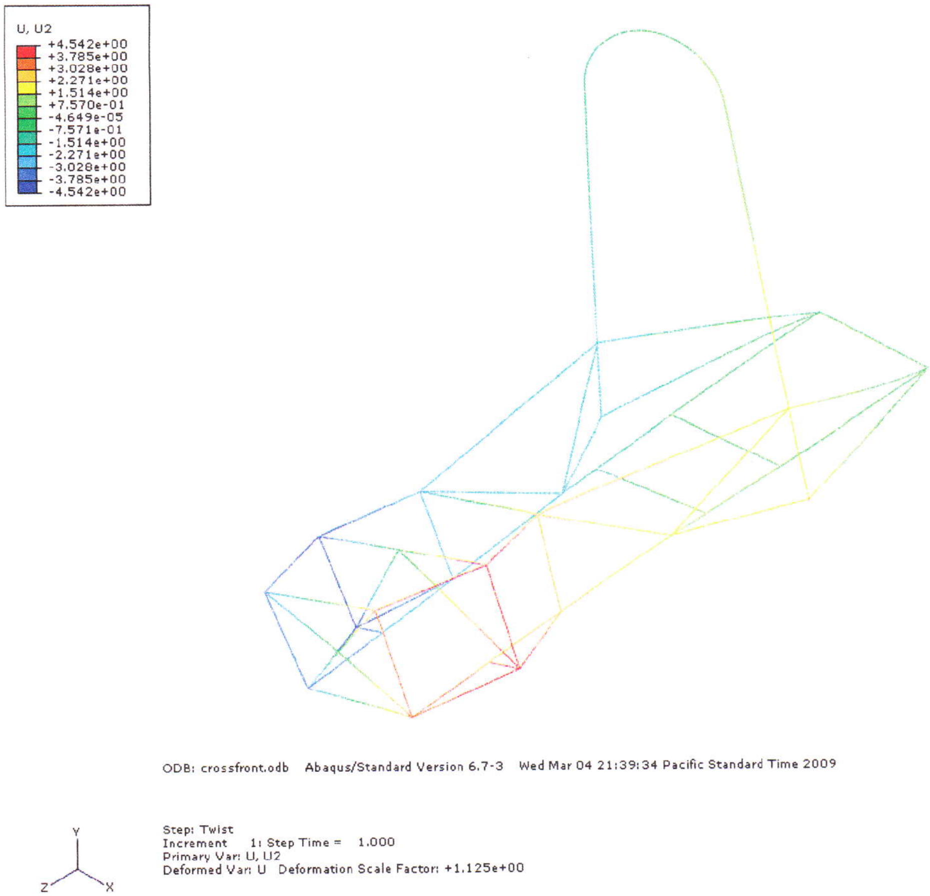


Figure 17 - ABAQUS U2 Deflection Results

Various cross sections were tested to find designs which twisted less than 1 degree. Cross sections with larger outer diameters and smaller inner diameters were found to have the highest specific stiffness. This is because putting an equivalent amount of mass as far away from the center of a tube as possible will provide a higher moment of inertia. A survey of results can be found below in Table 3. The results show it is difficult to get an adequately stiff chassis using tubes with diameters less than 2 inches. There are many variations of the space frame which meet our initial chassis weight target of 50 pounds; however, when we started analyzing the backbone, we predicted it to weigh less than 20 pounds. Therefore, we changed our target to 20 pounds for the space frame as well. The only cross section that comes close to this target weight has a 2.5 inch outer diameter and a 0.035 inch wall thickness. The t/D ratio is so small it raises concerns about the walls buckling.

Table 3 - FEA Analysis

Outer Diameter D [in]	Wall Thickness t [in]	Weight [lb]	Vertical Displacement U2 [in]	Torsional Stiffness [ft-lb/deg]
1.00	0.250	51.7	0.930	96.07
1.50	0.065	25.7	0.500	178.81
1.50	0.250	86.2	0.216	414.01
1.75	0.065	30.2	0.336	266.13
2.00	0.065	34.7	0.226	395.69
2.00	0.075	39.8	0.200	447.13
2.25	0.049	29.7	0.210	425.84
2.25	0.065	39.2	0.160	558.93
2.50	0.035	23.8	0.209	427.88

3.4 Preliminary Analysis

As stated before, the final chassis will need to be designed to handle both longitudinal bending and torsional deflection. Designing the chassis to be an order of magnitude stiffer than the effective chassis stiffness was achieve by designing the chassis to allow an additional 1/10th of an inch of wheel deflection for each inch deflection by the suspension. Designing to these specifications should yield a chassis that will handle predictably and respond well to suspension adjustments. Hand calculations for this analysis can be seen in Appendix D.

Chapter 4 Final Design

4.1 Overall Layout

As shown before, there were many different chassis options that would possibly work for the Urban Concept vehicle. We decided upon the final design with the option that best met the design specifications while remaining as simple as possible with regards to design, analysis, and manufacturability. A frame designed around a primary load carrying backbone was chosen for its relatively predictable reactions to loading conditions, simple design and manufacturing geometries, and efficient use of material. The overall layout of the chassis is shown in Figure 18. The detailed assembly drawing of this backbone design is shown in Appendix C.

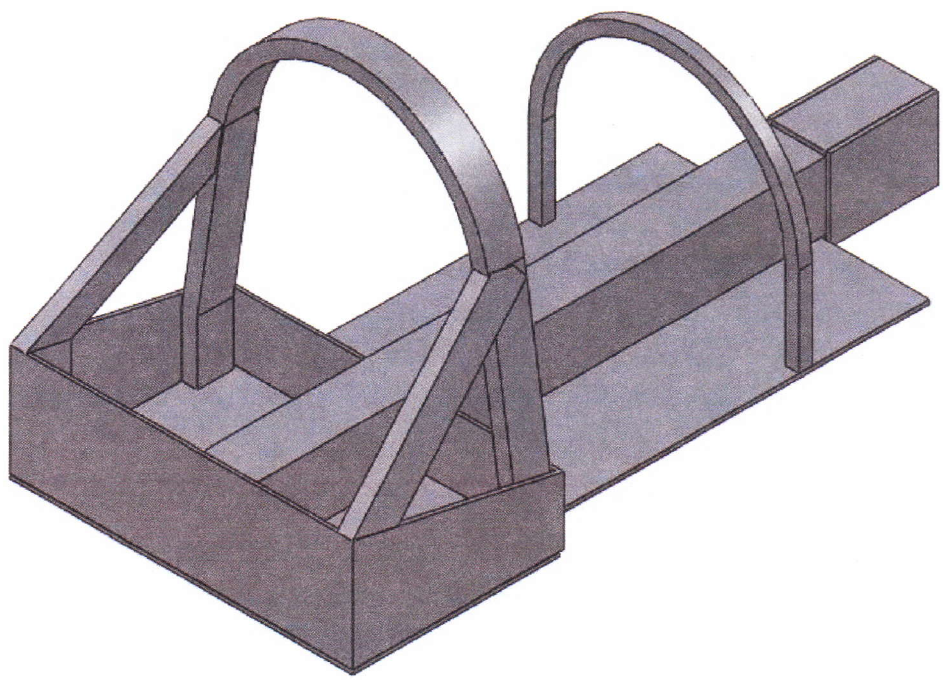


Figure 18 - Overall Backbone Chassis Layout

While using the backbone to support the primary load, auxiliary components were integrated for functionality. The rear compartment sub-frame allows for engine mounting and a luggage space while incorporating suspension component mounting locations. The compartment frame also incorporates mounting locations for the roll bar. The front suspension frame allows for mounting of upper and lower A-arms. The driver is seated on the main beam with a structural floor throughout the driver's compartment. The floor is designed to take point loads from the driver and also act as a structural mounting area for the dash bar. The floor also ties the front and rear sub-frames together to help support driving loads.

4.2 Detailed Design Description

With the backbone acting as the primary load carrying member, careful attention had to be paid to its design to ensure appropriate capacity, longitudinal and torsional deflection under loading, and geometry to allow for integration of all other necessary components. Due to its relatively simple shape, simple beam equations could be used for most of the design work.

Stiffness was a major consideration when designing the backbone frame. Because a suspension is to be used on the Urban Concept vehicle, frame design was driven largely by suspension requirements. Frames of ground vehicles need to be stiff enough to allow the suspension system to function correctly. A carefully designed suspension system is worthless when mounted to a frame that is too flexible and deflects to a point that does not allow for predictable wheel location and ground contact. Before designing any frame components, attention was paid to basic suspension design. Using spring frequency and sprung vehicle weight, we were able to conservatively estimate the vehicle's effective spring rate at the wheel. With this information we were able to design a backbone with stiffness an order of magnitude higher than that of the suspension, which would allow for more predictable chassis and suspension dynamics. More specifics of this analysis and the results can be seen in the following section.

After determining the required torsional and longitudinal stiffness, we were able to incorporate these values with estimated chassis loads and locations. With this data we were able to finalize overall bending limitations. Once these specifications were finalized, the backbone geometries could be determined to meet all specifications. The main beam will be constructed for both 6k plain weave fabric and 6k unidirectional tape. Five 0.007" thick layers will be used as a torsional wrap at a 45 degree angle around the entire tube. Spar caps will be placed along the top and bottom of the beam to support bending loads. These caps will consist of 6 layers of 6k unidirectional tape. Analysis and design specifics can be found in the following section.

Once the backbone size was determined, the front and rear sub-frames could be designed to meet other requirements and rule imposed geometries. The rear compartment frame was designed to incorporate the volume of the luggage space. A 20"X16" (L X W) by more than 10" tall space was designed to be slightly oversized to allow the luggage item to be easily stored to the left of the main beam to the rear of the driver's compartment. For symmetry, the engine compartment was designed to the same specifications. Since the engines being considered by the engine senior project team take up roughly one cubic foot of volume, the 20"X16" floor space was considered to be more than adequate. The rulebook requires that the

engine be completely separated from the driver by a firewall. For our design, we used separators to isolate the engine and act as a firewall. These separators allow for a 14" tall engine compartment behind and to the right of the driver. A major benefit in making both halves of the rear compartment symmetrical is that the luggage and engine sides can be switched without any chassis redesign.

The rear sub-frame is also designed to act as a mounting location for the rear suspension and roll bar. With an overall width of 40 inches, the rear sub-frame allows for attachment of a roll bar that is well within the design requirements of the rules. Roll bar rearward supports act to both stiffen the roll safety structure and act as an upper mounting location for the rear suspension components. The rear of this sub-frame offers a large area for integrating the pivot/mounting locations for rear suspension trailing arms. The roll bar will span the width of the rear sub-frame and extend to a height above the driver's head within the rule dimensions.

4.3 Analysis Results

From the analysis of the main beam a combination of carbon fiber materials in different orientations was found to be appropriate for the loading conditions and stiffness requirements for the frame. Five layers of 6k 0.007" plain weave cloth applied at 45 degrees to the main beam were found to be sufficient for torsional loading. It was also found that six layers of 6k 6" wide unidirectional tape at both the top and bottom of the beam acting to resist longitudinal bending would be required. Details and calculations can be found in Appendix D.

Roll Bar Analysis

The Shell Eco-marathon competition rules require our car to have, "an effective roll bar that extends in width beyond the shoulders of both authorized drivers. The roll bar must be included in the body/chassis and also extend 5 cm above the top of the driver's helmet in the normal driving position with the safety belt properly fastened. This roll bar must be capable of withstanding a 70kg static load applied to its center without bending." In addition, the cross bar supporting the seatbelt mounts must be able to withstand 1.5 times the driver's weight.

We are assuming a 24 inch shoulder width which will accommodate a small driver for the competition and it will also have space for an average person. The bottom of the roll bar needs to be 44 inches above the bottom of the main beam to provide the 5 cm clearance needed for an average driver. The beam design must fit these general dimensions and also

provide enough stiffness to resist bending under a 70kg load (155 pounds). We interpret this to mean the amount of deflection is invisible to the naked eye. Structural engineers use a rule of thumb that says the maximum deflection of a beam must be smaller than Length/180 for the deflection to be indiscernible to the naked eye. It is common to use a maximum deflection of the beams length/360 to ensure the deflection is invisible to the naked eye. This criterion is usually used for simply supported beams, and it is conservative when used for an arch shape.

To find the deflection of a curved beam, Castigliano's theorem must be used. His theorem states that a deflection caused by a certain load, in the same direction of the load, is equal to the partial derivative of the total strain energy with respect to that load. This is expressed by equation 1,

$$\delta_i = \frac{\partial U}{\partial F_i} \tag{1}$$

Strain energy is the potential energy stored in an elastic member from the work done to deform it. It is similar to the potential energy stored in a spring. The strain energy for a curved beam in bending is expressed by equation 2,

$$U = \int \frac{M^2 d\theta}{EI} \tag{2}$$

The bending moment for a curved beam is defined by equation 3,

$$M = \frac{F}{2} R(1-\cos\theta) - \frac{F}{\pi} R\sin\theta \tag{3}$$

To find the displacement at the top of the arch, the partial derivative of the strain energy must be taken, and the strain energy integral must be evaluated from 0 to 90 degrees. This becomes equation 4,

$$\delta = \frac{FR^3}{EI} \int_0^{\pi/2} \left(\frac{1}{4}(1-\cos\theta)^2 - \frac{1}{\pi}\sin\theta(1-\cos\theta) + \frac{1}{\pi^2}\sin^2\theta \right) d\theta \tag{4}$$

The final solution for displacement is found with equation 5,

$$\delta = \frac{FR^2}{EI}\left(\frac{3\pi}{8} - 1 - \frac{1}{2\pi}\right) \tag{5}$$

There are many different shapes and cross sections that will fulfill our deflection requirements, so we decided to use a shape that allows for flexible mounting for the fairing and doors. We chose a box cross section which is 1inche tall and 4 inches wide. The tall height

provides stiffness by making the moment of inertia, I, large. The 4 inch width provides a large area for mounting. The roll bar design is shown in Figure 19.

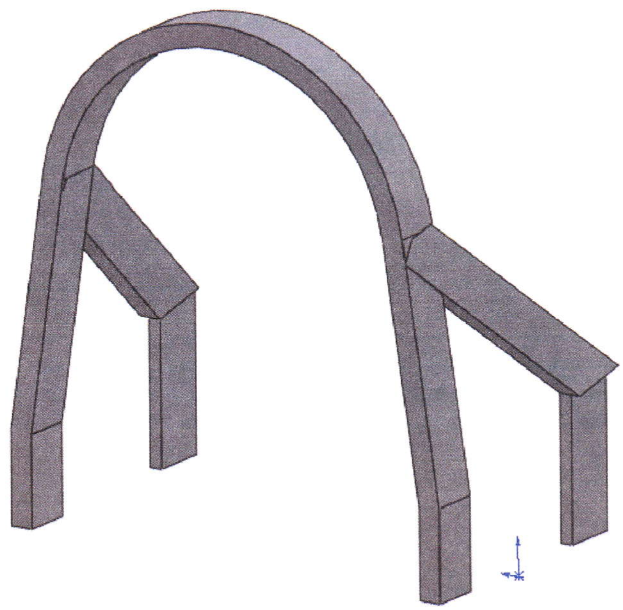


Figure 19 - Roll Bar

There are two primary load cases: 150 pounds down on the top of the roll bar, and 225 pounds forwards on the cross bar to hold the seatbelt in case of a crash. In the first case, the top and bottom faces of the roll bar are in tension and compression while the front and back faces are in torsion. In the second case, it's the opposite. The layup schedule will be unidirectional fiber in the direction of the tension and compression loads, and 45º plain weave layers to take the shear stress. If we use 2 layers of unidirectional fiber and 2 layers of torsion wrap, the roll bar will only deflect 0.005 inches when the vertical load is placed on it, and this exceeds our stiffness criteria.

4.4 Cost Analysis

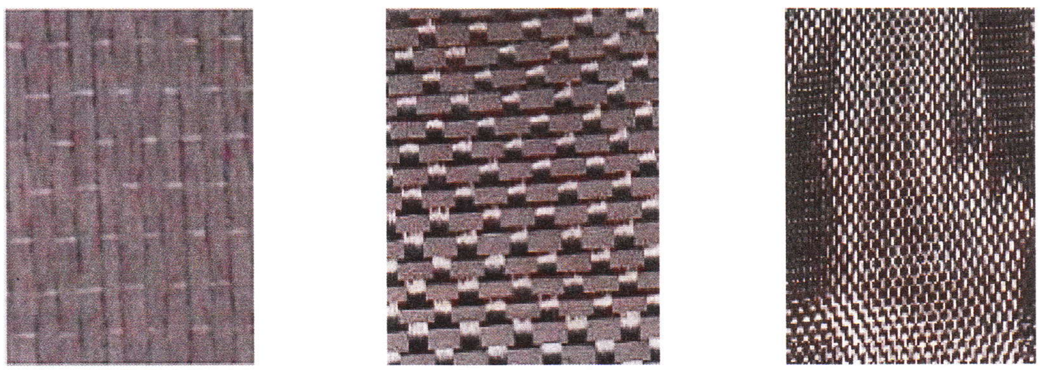
With the help of Dr. Joseph Mello, we have calculated the number of fabric layers required based on bending strength, shear stress, and torsion with a factor of safety of 1.5. The calculations are shown in Appendix D. Based on the results, a minimum of five layers of carbon fiber layup on all four sides are needed to meet the target strength and stiffness in both bending and torsion.

A bill of materials has been constructed and is located within Appendix E.

4.5 Material, Geometry, and Component Selection

Main Beam

The 6KPL50 fabric is selected as the material for the main support beam. This fabric offers good workability because the fiber fabric stays straight when doing the lay-up and it does not rotate out of alignment. 6k fabric was chosen over 3k because it requires half the amount of layers, which makes it easier to do the lay-up. 6k fabric is also more cost effective and readily available than 3k fabric. Figure 20 shows a comparison of three different carbon fiber weaves.



Unidirectional Fabric *6KPL50* *3KPW50*

Figure 20 - Carbon Fiber Weaves

The 6KPL50 is a dry carbon fiber fabric that requires a hand lay-up to impregnate the resins into the fibers using rollers or brushes. A hand lay-up on the main beam will be done because it is designed as a hollow rectangular member, and the structure will collapse easily when it is vacuumed under pressure. West Systems resin 105 and hardener 209 were selected for this hand lay-up process. Soller Composites, the company selling the 6KPL50 fabric, and Dr. Joseph Mello recommended this type of resin for its easy workability, pot life, and price. The physical properties of this West Systems resin are shown in Appendix E.

Carbon fiber fabric selection was developed with advice and resources from Soller Composites, George Leone, and Dr. Joseph Mello. Both Mr. Leone and Dr. Mello have a good amount of background knowledge and experience in composites.

The selection of resins and hardeners are as important as selecting fabric. In order to stiffen the carbon fiber fabric, a resin and hardener mixture is required. The primary functions of the resins are to hold the fibers together as an adhesive and to transmit stresses before

reinforcement fibers. For our application, we will be using thermosetting resins because these types of resins do not have the ability to convert back to the original liquid form once cured. Once these resins are cured, they will soften under heat and are unable to reshape afterward.

Hardener or curing agent is also needed to cure the epoxy resins into a much harder and rigid form. It is very important to have a proper ratio of mixture between the resins and hardeners because too much hardener will dry the resin really fast and not enough hardener will take a long time for the resin to dry. Also, at a higher temperature, the resin will dry a lot quicker than at a lower temperature.

Floor Panel

The material for the construction of the floor panel will be Toho Tenax HTS 12k prepreg fabric because a company called TenCate donated a prepreg roll of approximately 600 square feet to the Supermileage team. This prepreg fabric has a tensile strength of 666 ksi, a tensile modulus of 34.7 Msi, and a density of 1.77 grams per cubic centimeter with up to 265°F of oven cure temperature. This prepreg was chosen for the floor panel even though it has a small thickness of 0.005 in. because it is free and the floor panel is not the main structural support of the chassis. According to the calculations in Appendix D, a total of 10 quasi-isotropic layers of this prepreg are needed to handle a point load of 150 pounds at the edge of the floor panel with 0.13 in. of deflection as shown in Figure 21.

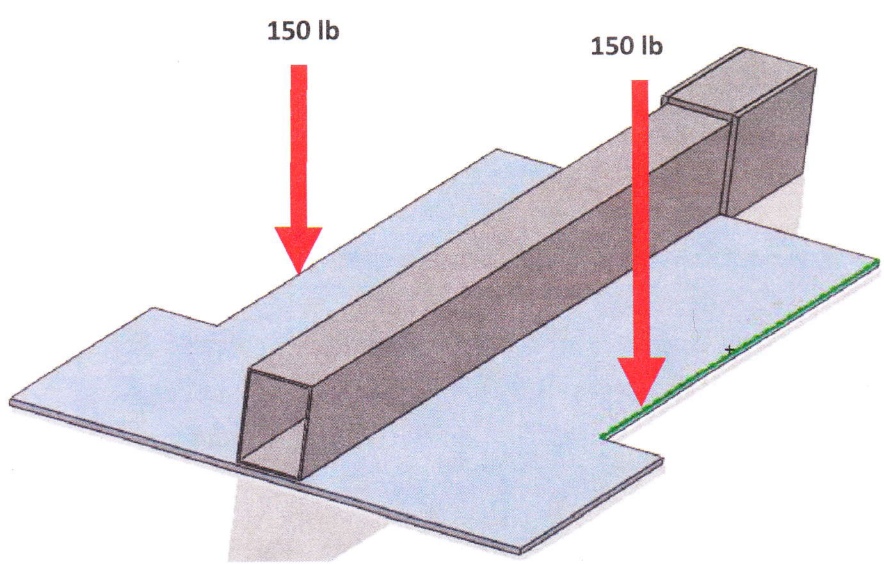


Figure 21 – Example of Expected Loads on the Floor Panel

Since the floor panel may be subjected to a high compressive point load when a person steps onto it with one foot, it will be reinforced with a honeycomb core to provide additional strength and stiffness. For a solid sheet metal of thickness t, adding a honeycomb of the same thickness t in a sandwich type construction increases the strength of the original metal by 350 percent, increases the stiffness by 700 percent, yet only increases weight by 3 percent. This is shown in Figure 22. The material for the honeycomb core was chosen to be Nomex, a lightweight fiber for its high strength at low densities, formability, and ability to provide a good bonding surface. An aluminum core has the highest strength and rigidity to weight ratios of the cores considered, but is more difficult to bond to carbon fibers and is therefore taken out of consideration. A cell size of 1/8 inch, density of 3 pounds per cubic foot (pcf), and thickness of ½ inch were chosen which provides a compressive strength of approximately 300 psi. A smaller density of 1.8 pcf may have worked, but the compressive strength would be approximately 100 psi. In this case, it would be beneficial to increase the weight a little to increase the strength a lot for structural and safety reasons. Also, a larger cell size could have been chosen to lower costs, but it may cause "dimples" in the outer surface of the sandwich and may also reduce bonding area for the core and surface materials. Compressive strength decreases about 5% when choosing a ¼ inch cell size. The standard 1/8 inch cell size was chosen to provide a balance of cost, appearance, and strength.

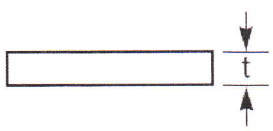
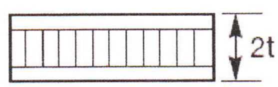
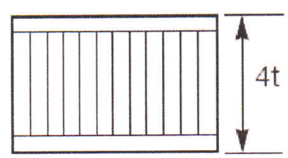
	Solid Metal Sheet 	Sandwich Construction 	Thicker Sandwich 
Relative Stiffness	100	700 7 times more rigid	3700 37 times more rigid!
Relative Strength	100	350 3.5 times as strong	925 9.25 times as strong!
Relative Weight	100	103 3% increase in weight	106 6% increase in weight

Figure 22 - Properties of Honeycomb Sandwich Construction

Geometry decisions for our frame were based primarily on two main factors with an overall goal to keep things small to reduce material and thus weight. We were required to design within the rules so that the dimensions of the vehicle would comply with required size limitations. We also had to develop our frame around the driver, ensuring proper dimensions for comfort and ease of entry. Since the club has not selected this vehicle's driver(s), we used Mannequin software and its integral human size database to design the driver's compartment. Small to average male models in the US were placed in the program in driving and seating positions and their critical dimensions were measured. These models can be seen in Appendix C. With this information, we were able to design the vehicle to accommodate a driver between 5' 5" and 5' 6" with some room for adjustment and space allowances made for sub-system integration.

Components for our frame were designed to reduce the overall weight of the vehicle by having multiple functions and reducing part count. For example, the rear sub-frame is designed as both the engine and luggage compartment and also to serve as a mounting location for the roll bar and supports. Another example is the roll bar supports which function to help prevent the roll bar from collapsing and also as upper mounting locations for rear suspension components.

4.6 Manufacturing

Components

The chassis will be constructed of five primary components including: the main beam, the floor, the rear compartment, the roll hoop, and the dash hoop. Also included in the chassis will be small support and reinforcing components. Basic fabrication information and considerations for each component follows. Lay-up schedules for all parts can be found in Appendix E.

The Main Beam

The main beam was constructed around a foam core that was supplied by the Supermileage team. A picture of the foam used is shown in Figure 23. The foam for the beam was hand-cut to dimensions of 60"L x 6"W x 7"H.

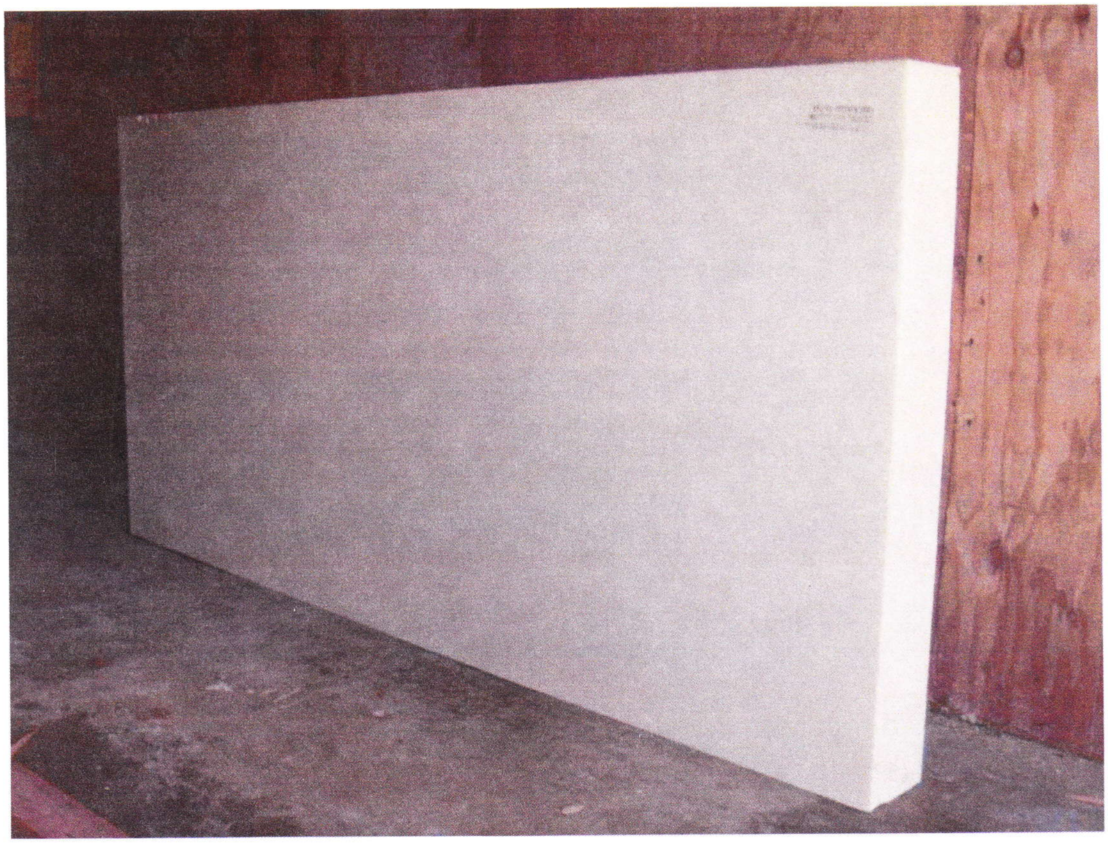


Figure 23 – Foam Used to Make the Core

Since the beam was not hollow, the vacuum infusion process (VIP) was used for its advantages over a typical vacuum bag lay-up. A picture of the beam during the VIP is shown in Figure 24. The main advantage of the VIP is that it achieves a better fiber-to-resin ratio. This has important benefits because any extra resin in the part will weaken it and increase its weight, both of which are unfavorable in achieving our goal.

Figure 24 - Main Beam During Vacuum Infusion Process

The West Systems 105 resin with the 209 slow hardener was used to coat the carbon fiber for the beam. The slow hardener was selected to give us a longer pot life and workable time. For the main beam, the 6k fabric and unidirectional tape layers were laid in the pattern, $[((0/90)/\pm45/0)_s]_3$. Since the 6k fabric needs to be used on a 0/90 as well as ±45 degree angle, we developed a cut pattern for the fabric to reduce wasted material. This pattern was designed to yield strips of 45 degree material to wrap the perimeter of the tube cross section and allow the longest amount of tube length available per yard of fabric. This pattern can be seen in Appendix C. The 26" wide strips allow for 2" of overlap around the beam and when the strips are combined with a 2"overlap, they will yield over 360" of tube length which allows for more than 4 layers of torsion wrap from 6.5 yards of material. Also used for the main beam was 6" wide unidirectional tape. 40 linear feet are required for 3 layers on both the top and bottom of

the beam. The completed beam, shown in Figure 25, weighs approximately 22 pounds including the foam core. The final dimensions of the beam are 60"L x 6.5"W x 8"H.

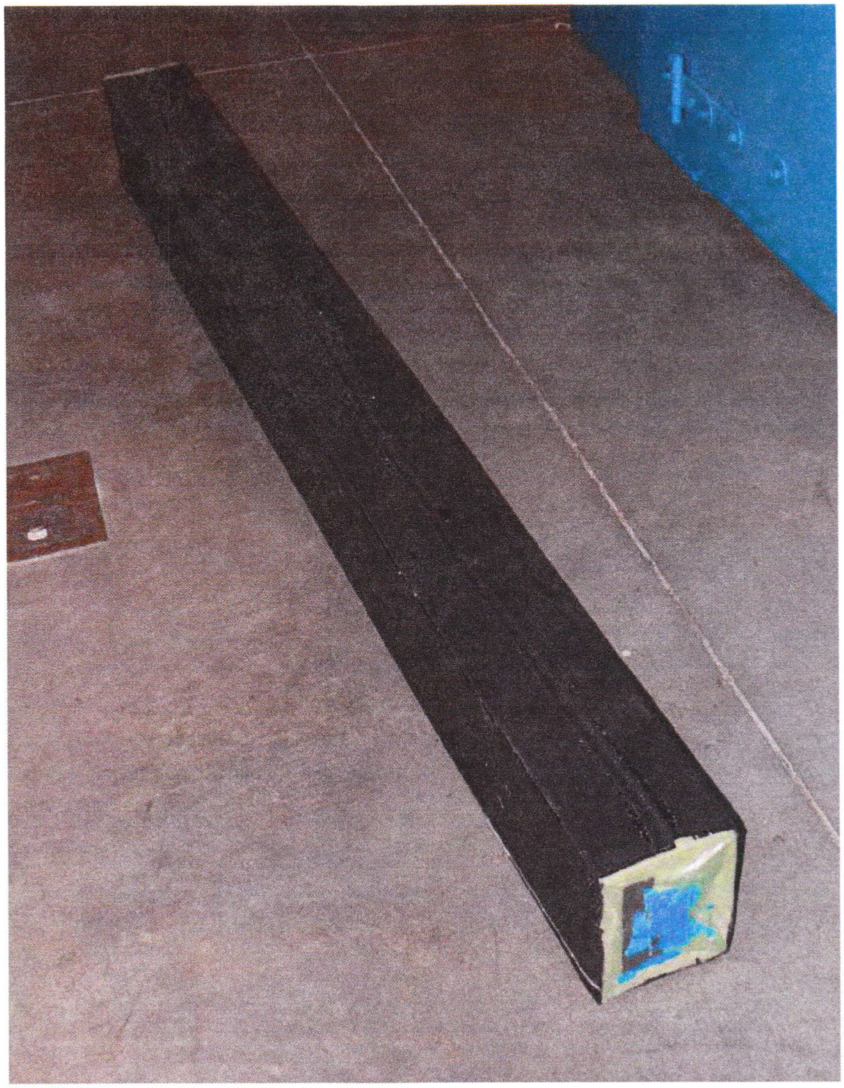


Figure 25 - Main Beam Cured

The Floor Panel

The floor panel will be constructed out of a sandwich composite structure which consists of a woven prepreg carbon fiber skin for the outside layer and a honeycomb core for the inside. The construction of the sandwich structure should be done carefully to save material. The honeycomb material comes in sheets that measure 4' x 8'. Although the chassis will not be using all of this material, sections of the sheet have been pre-allocated for use in each respective part in the chassis. The honeycomb cut pattern can be found in Appendix C.

The area has been optimized to use the honeycomb as efficiently as possible to save on costs. Another half-sheet of honeycomb may be used for testing purposes.

The construction of the sandwich composite for the floor panel was done by bonding the prepreg carbon fiber skin to the Nomex honeycomb core. This can be done in a variety of ways, but the basic procedure is the same. The first step is to set up a working area with a clean surface. The surface can be a plastic or a metal with mold release sprayed onto it to prevent the epoxy resin from bonding to the metal. Several layers of carbon fiber were placed in a manner to create a quasi-isotropic laminate then cut into the size of the sheets needed for the floor. There were two pairs of these carbon fiber layers; one pair for the front floor panel and the other pair for the rear floor panel. The honeycomb was placed between the two skins as shown in Figure 26.

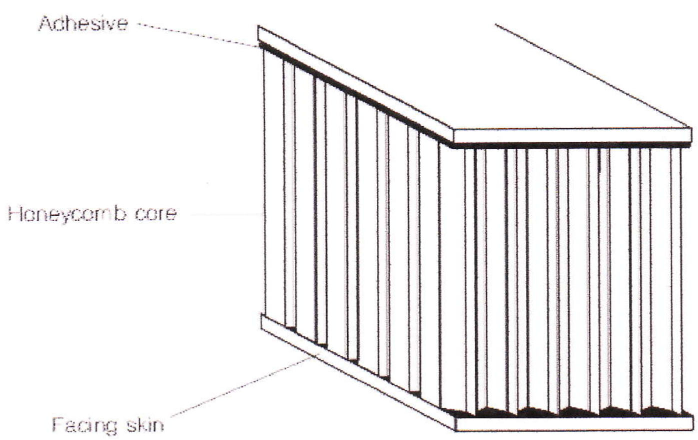


Figure 25 - Sandwich Panel

After the sandwich panel was formed, it was vacuum bagged. The vacuum procedure is done to optimize the fiber to resin ratio in the composite. It also helps to achieve a uniformly distributed compression around the surface of the layup, which helps prevent bubbles or wrinkles from forming on the skin. A typical vacuum bag lay-up requires a number of items including the composite laminate, release coatings, peel plies, release films, bleeder plies, breather plies, vacuum bags, sealant tape, and damming material. They were applied in the order as shown in Figure 27. Since the quality of the finished part highly depends on the quality of work done, practice pieces will be done to gain experience in the lay-up procedure. After the basics of the lay-up process were familiarized, vacuum bagging was done on the beam and dash hoop.

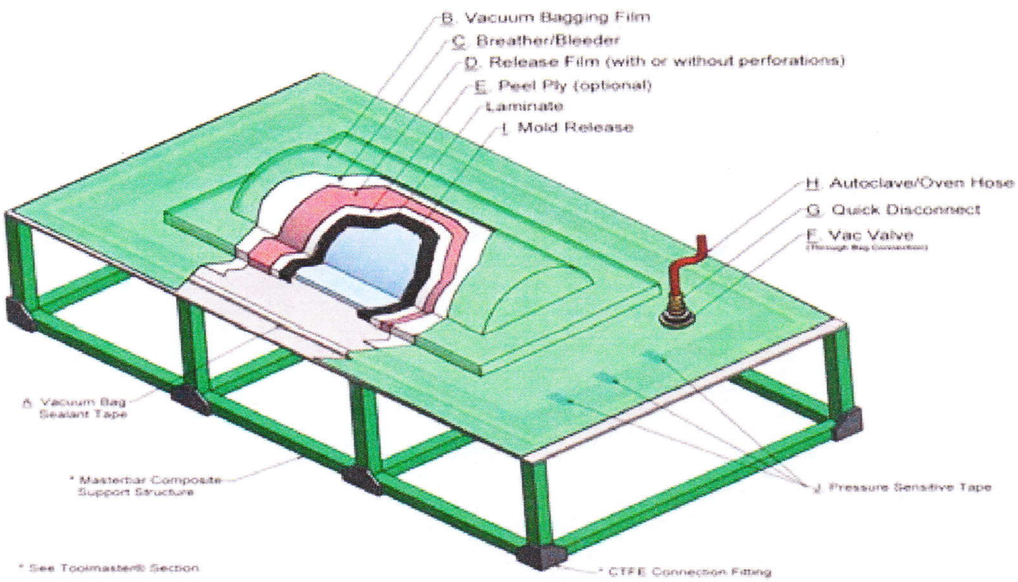


Figure 27 - Typical Vacuum Bagging Lay-up

In Figure 27, the first step is to apply a layer of release agent called peel ply to the noncombustible mold plate or caul plate. This is to prevent the carbon fiber laminate from bonding to the caul plate. Sealant tape will then be applied around the edges of the plate with the release paper side kept on until the bag is ready to be placed. Release films are added, and used to separate the bleeder and breather plies from the laminate. They are porous enough to allow excess resin to flow through and be absorbed by the bleeder and breather plies. Peel plies are used to protect the molded part from contamination and to prevent the prepreg fiber from sticking to the mold plate and other bagging material. They are also used to give the laminate a better surface finish for better bonding properties. Breather plies are porous and allow for the removal of air and volatiles during curing. The last item to be placed is the bag, usually a thick plastic. The bag should be pressed onto the sealing tape once the release paper has been removed. Care must be taken to ensure that no wrinkles in the bag are formed and that the bag is completely sealed, otherwise it will leak which defeats the purpose of vacuum bagging. Finally, a vacuum port must be attached before closing the bag. This will provide the port to draw air out of the lay-up.

Figure 28 shows the carbon fiber sandwich panel was prepared for vacuum bagging. In Figure 29, the sandwich panel was ready to be baked in the oven with specific cure profile. It means this carbon fiber prepreg must go through thermosetting condition to obtain a specified material property. By going through thermosetting stage, the epoxy resin trapped within the fibers would harden and strengthen the stiffness of the fibers.

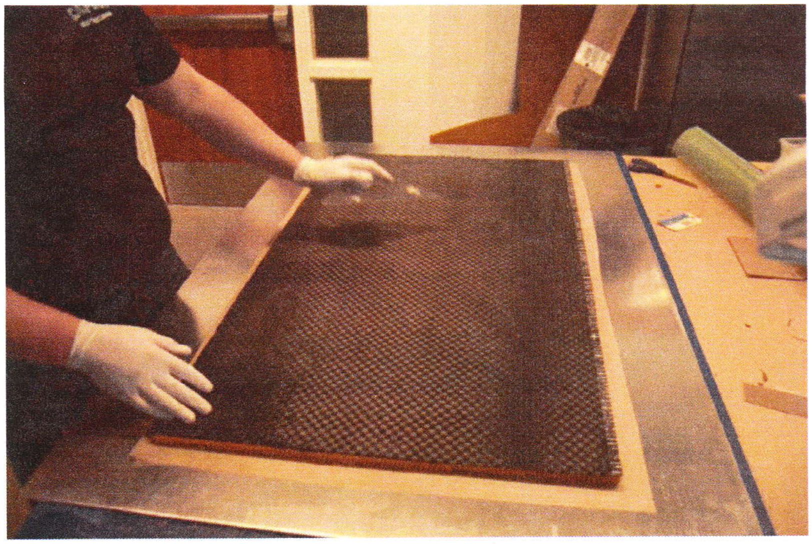


Figure 28 - Preparation of the Prepreg Sandwich Panel

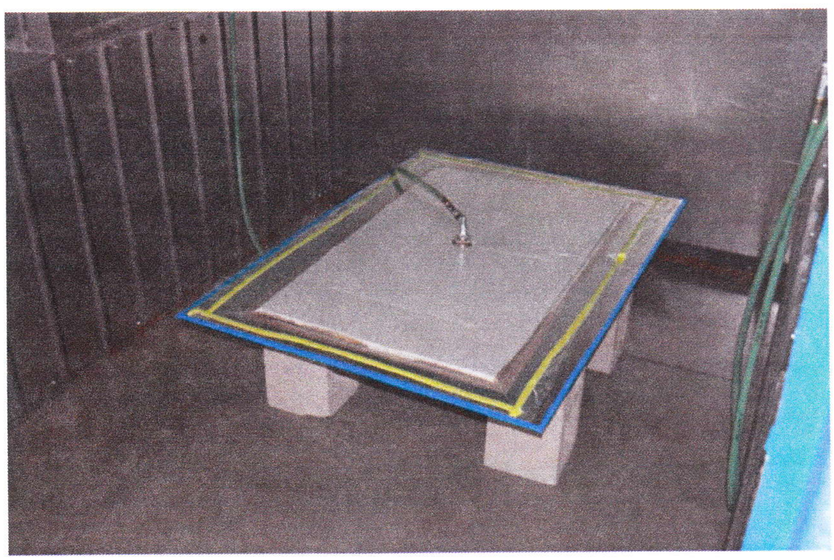


Figure 29 - Prepared Sandwich Panel for Curing

The Rear Compartment

For the rear compartment, we will use the same Nomex honeycomb that was used for the floor to create the core that can take compressive and local point loads while supplying a rigid structure to attach other components. A panel for the components of the rear section will be cut from the same 4'W x 8'L sheet as the floor for more efficient use of material. This panel will be covered with carbon using the same method that was used to make the floor panel. We

plan on using a 2 layers per side (3 layers per side for the rear panel), quasi-isotropic, 0/90° ±45° alternating lay-up. Since the rear section will be loaded from multiple directions with imposed moments we believe this carbon construction will offer the most resilient structure.

Once the panel is fully cured, the front, rear, and side sections of the rear compartment can be cut out. These sections consist of a 38"W x 12"H section for the rear, a 38"W x 7"H section for the front, and two 22"L sections with an angled cut top to join the 12"H rear and 7"H front sections. These sections will be bonded together with West Systems G/Flex 2 part Epoxy (properties in Appendix E) to form the rear compartment with the side sections inside of the front and rear sections resulting in a 38"W x 23"L compartment. With the rear compartment sections glued together, we will use strips of the carbon fabric to reinforce the seams and corners. At the inside and outside of each corner, 4" wide carbon strips can be wrapped across the seam, tying the edges together with fabric in shear and strengthening the joint, as seen in Figure 30.

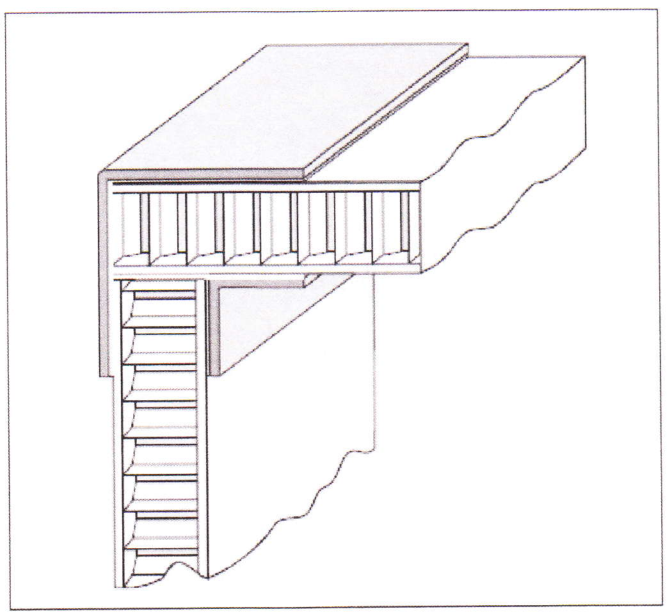


Figure 30 - Rear Compartment Construction Example

Attached to the rear compartment will be additional support honeycomb panels (similar to the support panels on the front of the main beam) that will aid in the support and mounting of the roll bar.

Dash Hoop

The dash hoop is a non structural part that can be used for mounting gauges and controls. We constructed this piece by shaping a piece of foam and using the vacuum resin infusion method, shown in Figure 31, to lay up carbon fiber over it. Its lay-up schedule is [±45/(0/90)]. We decided to use the vacuum resin infusion method to achieve a good fiber to resin ratio; however, it was very difficult to wrap the part with dry fabric. The fabric was not wrapped tightly, so when we sealed it in a vacuum bag, all of the excess material was compresses into creases. This made the dash hoop look unappealing and its strength was decreased because the fibers are not aligned in a straight line. The dash hoop turned out acceptable because it is not a structural component of the car.

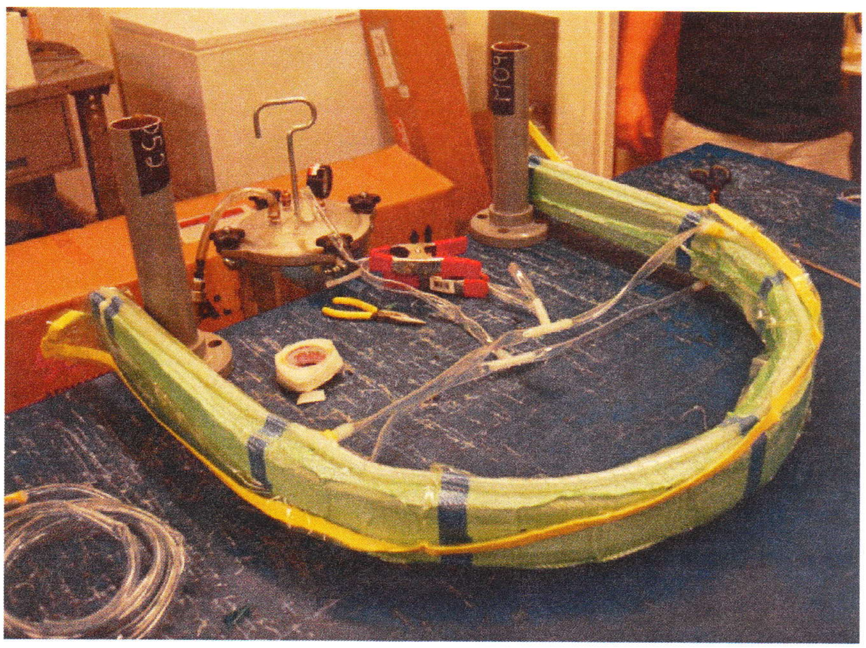


Figure 31 – Resin Infusion on Dash Hoop

Roll Bar

The roll bar was made by doing a carbon fiber wet layup around a foam core. The carbon fiber lay-up schedule for the roll bar is [±45/0/±45/0/(0/90)]. The foam core was cut out and sanded to shape by hand. After the difficulty we had manufacturing the dash hoop, we decided to do a wet lay-up for the roll bar. The wet lay-up fibers stay in place better than dry fabric because it is already sticky from the resin. The roll bar is an important part of the cars structure so it needed to turn out strong.

The roll bar was made in two separate pieces: the semi circle and the straight support members. Wrapping carbon fiber around the individual smaller parts is easier than wrapping around one larger part. Wrapping the curved arch is difficult to do using large sheets of fabric because it bunches up around the bends. Also, the ±45 layers fall apart when they are folded too much. We laid up the arch by placing strips of unidirectional fiber along the top and bottom surface of the arch. We also cut out 2" strips of ±45 and 0/90 layers and wrapped them around the entire circumference of the arch cross section. This method of application allows for the fabric to be tightly wrapped. Figure 32 illustrates how we wrapped the ±45 and 0/90 layers onto the beam.

Figure 32 - Wrapping Roll Bar

The straight members were constructed by a wet layup and it was wrapped easily due to its rectangular shape. They were cut and mitered so the ends lined up with the arch and the car. Everything was glued into position using 2 part epoxy. Figure 33 shows us applying glue to the joints to hold the beams in position. Then the joints were bonded by laying up 4 inch carbon fiber strips over the connections. The joint layup schedule exceeds the parts they are connecting to ensure the joints are not the weakest links in the part.

Figure 33 - Gluing Joints

Joints

To join individual parts, adhesive joints must be made. These bonds are complicated to analyze because there is a discontinuity in the material. The force is transferred between parts through shear in the bonding layers. Figure 34 shows how the load is transferred.

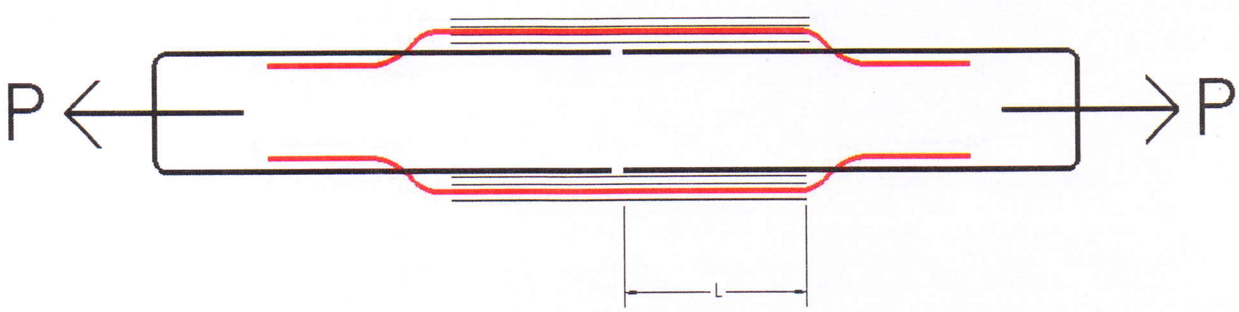


Figure 34 - Joint Load Path

The red lines indicate the load paths, the thick black lines outline the parts being connected and the thin black lines are the bonding layers used to join the two pieces together. The load starts out in each part and then goes through the bonding layers to the other part.

The load is transferred from each part to the bonding layers by shear, so the surface area of the bonding layers must be large enough to ensure a low shear stress. The average shear stress in the joint can be approximated by the following equation where P is the load, L is the length of overlap, and w is the width of the bonding layers going into the page,

$$\tau_{AVG} = \frac{P}{wL}$$

This approximation does not take into account the complex details inherent in bonded joints, but if we use a high safety factor, it will produce conservative bonds that will be sufficiently strong. The maximum allowable shear stress criteria we are using for joints is 500 to 1000 psi. This will be easy to accomplish because our structure sees small loads.

The joint between the main beam and the floor sections is critical to the support of the driver and many of the vehicle's sub-systems. These components will be joined using the same West Systems G/Flex Epoxy discussed in the rear compartment construction section. The joint surface is as wide as the beam and as long as the floor section that is being mounted to it. For example, the driver's compartment adhesion area is 5" wide and 45" long (225 square inches). This surface area combined with the epoxy's tensile strength will be more than strong enough to support the loads of our vehicle.

While the epoxy will be strong enough to form a secure joint we also checked the possibility of the carbon layers delaminating at the joint do to a failure of the resin. It can easily be seen in Appendix E that the resin we will be using has more than twice the tensile strength of the G/Flex epoxy. It would be highly unlikely for the carbon structure itself to delaminate before the G/Flex epoxy joint fails.

An additional consideration was made to the possibility of the joint tearing from the edge when the driver loads the floor away from the center of the car. A simple moment calculation shows that if this load was only supported by a one foot section of the floor, a strip of glue less than two hundredths of an inch wide along the edge of the beam would provide sufficient support.

Support Panels

There are two places on the chassis where we will attach panels to provide extra thickness for mounting. Honeycomb sandwich panels will be added to the front of the main beam to provide a thick area for suspension mounts as shown in Figure 35. We will use potted

inserts as hard points so these plates will provide the required thickness for the inserts. We made and cut out the mounting panels but we have not attached them yet because the Supermilage team has not decided where they want to mount suspension. Everything is prepared to be glued on and bonded once the location is finalized.

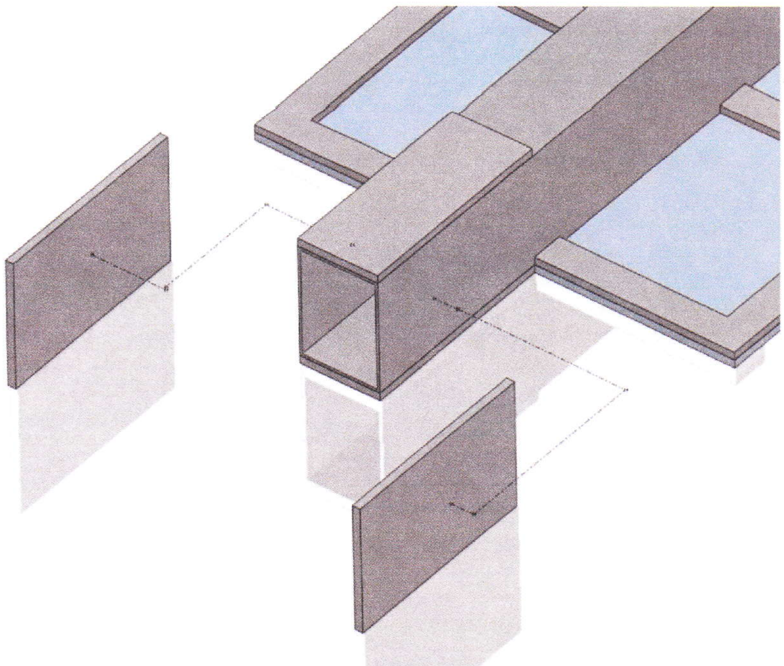


Figure 35 - Front End of Beam Honeycomb Sandwich Panels

Support beams will also be added where the roll bar attaches to the rear box as shown in Figure 36. These beams have foam cores with a wet carbon fiber lay-up. They provide a base to place the roll bar and a large area to bond with the roll bar. The beams were attached to the inside of the rear box.

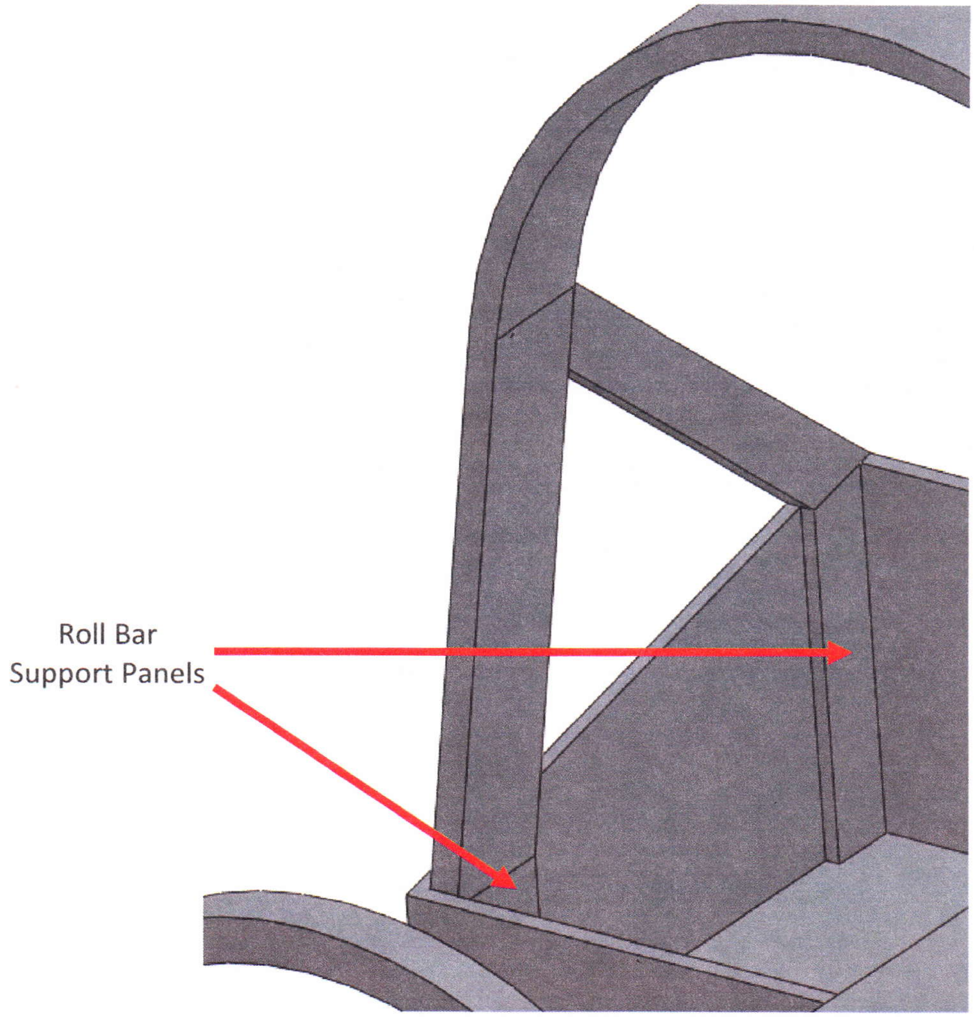


Figure 36 - Roll Bar Supports Attached to Rear Box

4.7 Safety Considerations

Since the chassis is being designed for use in a competition where it will reach speeds of over 25 miles per hour, safety must be a priority. Though this speed does not seem high compared to speeds that can be achieved by typical passenger cars, serious injuries occur at 25 miles per hour in automotive collisions. In the competition, safety considerations are multifaceted. The driver must wear protective equipment and the vehicles must meet certain safety regulations outlines in the rulebook. With regards to the chassis and fairing, these include: a roll bar requirement, a side impact safety requirement, a front bumper, and a visibility requirement. Shell also requires the driver be retained by a five point safety harness.

As stated in the rules, the vehicle must be fitted with a roll bar that does not bend with a 70kg static load applied top center. This roll bar must extend in width beyond the driver's

shoulders and in height 5cm above the driver's helmet. Using the Mannequin model and geometries of other components, we designed the roll bar to incorporate the driver as specified in the rules and mount to the rear sub-frame. While ply count and wall thickness have not been determined, the overall size and geometries will be sufficient to meet the specifications.

The rules state that the driver must be sufficiently protected from side impacts. We believe that the sides of the roll bar, dash bar, and floor will offer sufficient protection. The frame has been designed with ample room to mount a front foam bumper. The seating position and locations of other components should allow for adequate visibility with windows correctly positioned in the fairing.

Adequate locations have also been incorporated on the chassis for harness mounts. The lap belt can be anchored at either side of the driver to hard points integrated into the rear sub-frame. Shoulder belts can be mounted to the roll bar or cross bar between the sides of the roll bar. Finally, the fifth belt can be mounted between the driver's legs to the main beam.

4.8 Maintenance and Repair Considerations

For most designs, maintenance is a vital procedure that must be followed to ensure the proper operation of components, equipment, and systems. Although with composites there is not much direct maintenance that can be done, careful inspections of the composite structure, especially at critical load bearing locations, should be done often. Proper and careful use should be an important concern and maintenance of all systems should be followed to avoid negative impacts on the chassis. With inspections, damage such as fatigue cracks and delimitation can be repaired before they worsen and compromise the structure.

The proper repairing of composites such as carbon fiber is a tedious task that should be avoided if possible. In the case where a repair is needed to be done in a short amount of time, a patch kit containing epoxy, extra fabric, and a brush is all that is needed. Simply applying the epoxy and fabric to connect the broken part will provide a quick fix to the problem. However, the original characteristics of the composite are greatly compromised in this method. To restore the composite near its full potential, a more tedious procedure is required for the repair. First, the damaged area is identified and the work area is taped off. The area is smoothed out and any broken pieces are removed and replaced with a new piece by use of glue. Layers of carbon fiber are mixed with resin and applied over the damaged area in the same weave pattern of the pre-existing layers which can then be heated and bonded to the part. In some cases, resin can also be injected into the failure to re-bond the composite.

Chapter 5 Design Verification Plan

5.1 Test Descriptions

Testing our frame will consist mostly of verifying our designed stiffness requirements, testing the roll bar for adequate stiffness based on the rules, measuring for geometrical compliance with the rules, testing driver fitment, and qualifying the chassis for subsystem integration with the Supermileage club.

The chassis will need to be tested for stiffness for two main criteria: longitudinal bending and roll bar stiffness. Since we are designing the main beam to support all our loading requirements without any auxiliary components, we will first test the beam itself as a bare structure. Basic testing procedures for these stiffness requirements are as follows:

Longitudinal Bending

To test the beam for longitudinal bending, we measured the vertical displacement of the beam at a critical location under loads. The basic setup for this test can be found in Appendix C. The beam was placed on supports at the front and rear axle centerline locations. Supports were metal structures capable of withstanding substantial load. A dial indicator was used at approximately the center of the beam, as shown in Figure 37, to measure the deflection there under applied loads. The final load of 480 lb is equivalent to more than 3g acceleration. With this load, we observed the maximum deflection of 0.101". The indicators were zeroed for an unloaded reference. The beam was then loaded at the center. Once the beam was loaded, readings from the dial indicator were taken. Simple geometry can be used to calculate the chassis deflection at any location. The result of this testing is shown in Figure 38. The inverse of the slope of the graph represents the stiffness of the beam, which is approximately 7200 lb/in.

Figure 37 - Beam Test

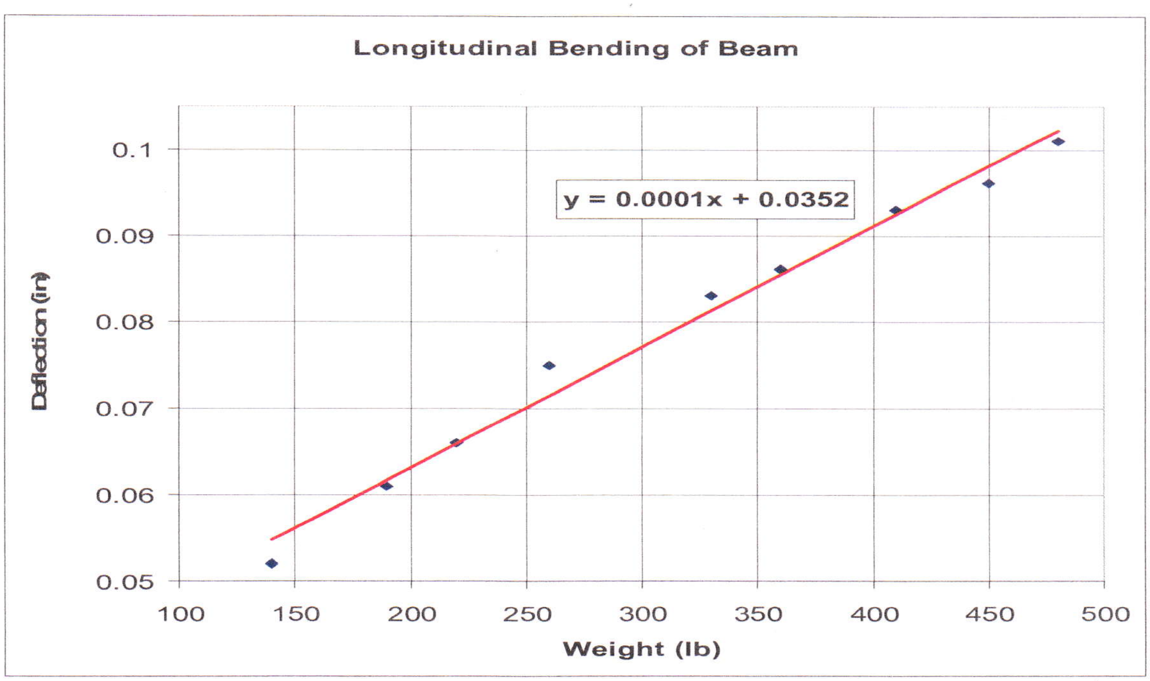


Figure 38 - Beam Longitudinal Bending Results

Roll Bar

The roll bar is a component that must meet stiffness requirements in the rules. After it was built, it was loaded with approximately 160 pounds of weight as shown in Figure 39. A visual inspection revealed no deflection under the load. A similar test method will be used during the technical inspection at the Eco-marathon competition. The rule book requires that there be no visual deflection under a 70 kg (155 lb) load.

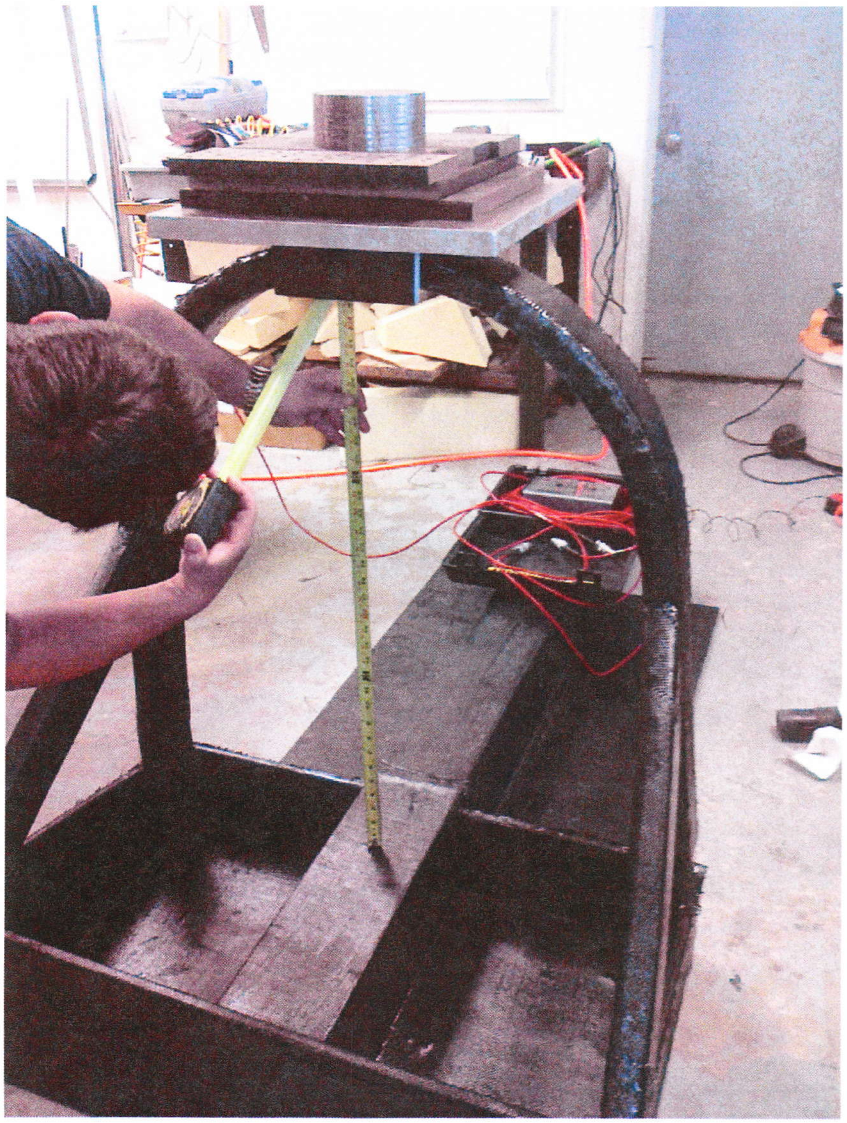


Figure 39 - Roll Bar Under 160 lb Load

Other

Other dimensional testing was done such as ensuring that a driver's head would clear under the roll bar as required by rules. A potential driver was place in a proposed seat. There was more than enough clearance above the driver's head with a helmet on. All other geometries are within the allowable tolerances required in the rules and allow for integration of other vehicle components as necessary. These tests were done to verify the car's eligibility to enter the 2010 Shell competition. The chassis met these requirements so it is able to compete.

5.2 Specification Verification Checklist or DVPR

After the completion of the carbon fiber lay-ups, various tests were conducted on the chassis to verify calculations and design dimensions. The first to be conducted is a simple dimension verification to ensure the chassis is within Shell's Official Rules guidelines. After that, we conducted various static loads tests on the chassis to verify its strength and measure the deflection that occurs due to the loads.

Chapter 6 Project Management Plan

Our project's primary goal was to have a completed chassis, ready to deliver to the team by December. The primary concern with the chassis was developing the geometry needed to withstand any expected loads while minimizing weight. A Gantt chart was used to plan for deadlines and future schedules of the project and its progress. This chart included important deadlines such as the deadline for the Supermileage team to deliver their final designs and component choices so that we can finish our final dimensions of the chassis by build time. The chassis build began in May through November at which point we changed focus to testing the bare chassis.

One thing we learned that should be considered in future projects is the time it takes to make composite parts. There is a lot of extra time required to make carbon fiber parts that we did not expect, such as downtime for the parts to cure in the oven and between wet layups. Also, during assembly, we could only attach a few parts per day, because we had to wait for those to dry overnight. During scheduling, it is critical to account for these unexpected delays, and provide more time for tasks than what is anticipated.

Chapter 7 Conclusions and Recommendations

The conclusion from the weight, strength, deflection, and tension calculations showed that a carbon fiber backbone chassis was the best design for the 2010 Shell Eco-marathon Urban Concept competition. Throughout the design and construction of the chassis, many problems and challenges arose, but we were able to meet our original goal of building a light, sturdy, and adaptable chassis for the Supermileage team. One of the big challenges was the construction process. Because none of us had prior practical experience working with carbon fiber, we ran into some troubles where we had to build our part differently than originally planned. An example of this would be in constructing the roll bar, and a more detailed description of the problem is described in section 4.6.

Another problem arose early in the final quarter of the project when the Supermileage team proposed some new ideas that asked us to change our design. We made minor modifications to accommodate their new ideas. When the chassis was fully assembled we had to spend a fair amount of time cleaning up sharp edges where fiber had hardened and was sharp enough to cut someone. After grinding down sharp fibers and recoating them with extra resin, a large majority of these edges were taken care of. In the end we met all of our desired tests and weight goal. Our plan was for a less than 50 lb chassis and we ended up with a 48 lb chassis with roll bars included.

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Appendix A – Rules and Specifications

A.1 Shell Eco-marathon Rules

A.2 Specification Chart

The effectiveness of the two braking devices will be tested during vehicle inspection. The vehicle will be placed on an incline with a 20 percent slope. The brakes will be activated each in turn. Each system alone must keep the vehicle immobile.
The use of a hydraulically controlled braking system is recommended. Cable operated systems are allowed, and if a bicycle-type brake shoe system is used, only the V-Brake system is authorised.

Article 41: Exhaust System

The exhaust gases must be evacuated outside the vehicle body.
Exhaust pipes must not extend beyond the rear of the vehicle body.

Article 42: Sound Level

The sound level for a Prototype vehicle must not exceed 90dB when measured 4 metres away from the vehicle.

Article 43: Emergency Shut-down mechanism

An emergency shutdown mechanism, accessible from the exterior, must be installed on all vehicles. A red arrow at least 10cm long and 3cm wide at the widest point must be positioned on the vehicle body to indicate clearly the position of this emergency shutdown mechanism from the exterior. This system must stop the engine and isolate the battery.

Article 44: Additional Inspections

After passing the technical inspection, the replacement of major engine or vehicle part will be subject to re-approval from Race Inspectors.
After any significant incident on the track the vehicle will be subject to a re-inspection.

At any time, the Organisers may perform unannounced inspections on the vehicles.

3B - UrbanConcept Group

Article 45: Definition

Under the name "UrbanConcept", Shell offers an opportunity to design and build fuel-economy vehicles that are closer in appearance to road-going cars than prototypes. UrbanConcept vehicles must comply with the specific rule of the Shell Eco-marathon for this group. One particular feature of this group is that vehicles competing in this group will require "stop & go" driving.

Article 46: Energies

All authorised types of energy for prototypes are also permitted for UrbanConcept vehicles.
In addition, the use of hybrid technology is also allowed for the UrbanConcept Group. Hybrid technology means the combined use of internal combustion engine and electric motors in vehicles supported by an electric power accumulation system. Solar panels are not allowed for hybrid vehicles. Regenerative energy braking systems are allowed in this group.

It is not permitted to preheat the engine after commencement of the fuelling operations for the attempt.

Article 47: *Vehicle Design*

During vehicle design/construction and competition planning, competitors must pay particular attention to all aspects of safety, i.e. Driver safety and the safety of other participants and spectators.

UrbanConcept vehicles must have four wheels, which under normal running conditions must be all in continuous contact with the road. Aerodynamic appendages, which adjust or are prone to changing shape due to wind whilst the vehicle is in motion, are forbidden (e.g. no shrink wrap allowed).

Vehicle bodies must not include any external appendages that might be dangerous to other participants. The vehicle interior must not contain any objects that might injure the Driver during a collision.

Article 48: *Dimensions*

- The total vehicle height must be between 100cm and 130cm.
- The total vehicle width must be between 120cm and 130 cm.
- The total vehicle length must be between 220cm and 350cm.
- The track width must be at least 100cm for the front axle and 80cm for the rear axle.
- The wheelbase must be at least 120cm.
- The Driver's compartment must have a minimum height of 88cm and a minimum width of 70cm at the Driver's shoulders.
- The ground clearance must be at least 10cm.
- The maximum vehicle weight (excluding the Driver) must be 160kg.

Article 49: *Vehicle Body*

The body must cover all mechanical parts, whether the vehicle is viewed from the front, the rear, the sides or from above. When seen from above, the body must cover the wheels. When seen from front, the body must cover the wheels down to the ground clearance of the vehicle. Wings/fenders must be an integral part of the body and not only attached to the wheel axle.

- It is prohibited to use a commercial vehicle body (e.g. mini-car).
- The vehicle must be equipped with a side door enabling easy access. This door must be easy to open from both the inside and the outside of the vehicle. The side door opening must extend from a maximum height of 10cm above ground clearance to a minimum height of 10cm below the total vehicle height
- The vehicle must have a roof covering the Driver's compartment.
- A windscreen is mandatory.
- Luggage space must be available for a suitcase-like object with dimensions of 50 x 40 x 20cm (LxHxW). This space must be easily accessible from the outside and must include a floor and sidewalls to hold the luggage in place when the vehicle is moving.
- The vehicle must not have any sharp edges on its exterior.
- A towing hook or ring is mandatory on the front of the vehicle, so that it can be towed with a cable by another vehicle. This hook or ring must resist a traction force of 2000N.

Article 50: *Body/Chassis Solidity*

Teams must ensure that the vehicle shell and/or chassis are solid. The cockpit must be equipped with an effective roll bar that extends in width beyond the shoulders of both authorised Drivers. The roll bar must be included in the body / chassis and also extend 5 cm above the top of the Driver's helmet in the normal driving position with the safety belt properly fastened. This roll bar must be capable of withstanding a 70kg static load applied to its centre without bending.

Moreover, all sides of the compartment must be sufficient to protect the Driver from possible lateral and frontal shocks. Any vehicle not equipped with the above safety features will be subject to disqualification.
A 5cm-thick layer of polyurethane foam with a minimum density of $28kg/m^3$ must 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 frontal collision.

Article 51: ***Engine and Fuel System Isolation from the Driver***

A permanent, rigid, fire resistant bulkhead must be mounted between the engine compartment and the cockpit, thus preventing any manual access to the engine compartment by the Driver.
The whole fuel system, from the tank to the engine, must be placed behind this bulkhead or in a compartment completely separated from the cockpit.

Article 52: ***Fire Extinguisher***

Each vehicle must be fitted with a fire extinguisher (ABC or BC type). All Drivers must be trained in the use of said fire extinguisher. This extinguisher must have a minimum capacity of 1kg (2lb unit for US application), be full and must have a certificate of validity bearing the manufacturer's number, the date of manufacture, and the expiry date.

Plumbed-in extinguishers may be located in the engine compartment and must discharge into the engine compartment. Triggering systems must be located within the cockpit and be operable by the Driver in his normal driving position.

Hand held extinguishers must be located within the cockpit and be accessible to the Driver once they have vacated the vehicle. In the event of a fire, Drivers should first exit the vehicle and then if possible, remove the extinguisher and attempt to extinguish the fire if safe to do so.

Article 53: ***Visibility***

The Driver must have *access to a* direct *arc of* visibility (*ahead, and to*) 90° on each side of the longitudinal axis of the vehicle. This field of vision must be achieved without aid of any optical *(or electronic)* devices such as mirrors, prisms, periscopes, etc. *Movement of the Driver's head within the confines of the vehicle body to achieve a complete arc of vision is allowed.*

The vehicle must be equipped with a rear-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, will be subject to inspection. An electronic device must not replace rear-view mirror.

An Inspector will check visibility in each of the vehicles in order to assess on-track safety. This Inspector will check good visibility with seven 60cm high blocks spread out every 30° in a half-circle, with a 5m radius in front of the vehicle.

Article 54: ***Safety Belts***

The Driver's seat must be fitted with an effective safety belt having at least five mounting points to maintain the Driver in his/her seat. **The fifth point must be designed and fitted to prevent the Driver to from slipping** forward in case of frontal accident. The 5 independent belts must be firmly attached to the vehicle's main structure and be fitted into a single buckle, specifically designed for this purpose. Safety belt buckles and attachments must be made of metal. The safety belt must be worn and fastened at all times when the vehicle is in motion. The fitness for purpose of **the belt and its fitting will be evaluated during technical inspection by raising the vehicle with the Driver on board**

using the safety harness for suspension. The safety belt must withstand a force of at least 1.5 times the Driver's weight.

Article 55: ***Vehicle Access***

It is imperative for Drivers to be able to vacate their vehicles at any time without assistance in less than 10 seconds.
The door opening must be covered by means of a hinged or sliding doors. The release mechanism must be easily operable from the inside. The method of opening from the outside must be clearly marked by a red arrow and must not require any tools.
It is forbidden to attach or to reinforce the door with adhesive tape.

Article 56: ***Steering***

Vehicle steering must be achieved by means of a steering wheel. It must be precise, with no extra play. The turning diameter must be less than 12m.

Article 57: ***Wheels***

The rims must be 16 or 17 inches in diameter.
The wheels located inside the vehicle body must be made inaccessible to the Driver by a bulkhead. Any handling or manipulation of the wheels is forbidden from the moment the vehicle arrives at the starting line until it crosses the finish line.
Teams must take into account the fact that bicycle and motorcycle wheels are not generally designed to support substantial lateral cornering forces, such as may be found in Shell Eco-marathon vehicles at certain speeds. Furthermore, such axles are usually not appropriate for cantilever type load distribution.
Therefore, bicycle wheels are not permitted and all wheels and axles must be of a size appropriate for the application.

Article 58: ***Tyres***

All tyre types are allowed as long as they are fitted on the type and size of rims recommended by their manufacturers. The tyre / rim assembly must have a minimum width of 90mm, measured from sidewall to sidewall. The width is measured with the tyre fitted on its rim at its rated pressure. Caution: the manufacturer's size indications should not be taken as measure, as the width of the rim directly impacts the width of the rim/tyre assembly.

Article 59: ***Lighting***

The vehicle must have a lighting system in proper working order for on- road use, including:

- Two front headlights
- Two front turn indicators
- Two amber rear turn indicators
- Two red brake indicators lights in the rear
- Two red rear lights (may be combined with the brake lights)
- The centre of each headlight beam must be located at least 30cm to each side of the longitudinal axis of the vehicle.
- The mandatory red indicator light for the self starter operation must be separate from any of the above (Article 78)

Article 60: ***Horn***

Each vehicle must be equipped with the authorised horn that can be purchased on the Shell Eco-marathon Website's e-shop centre.

Article 61: Vehicle Handling and Driver Position

A vehicle handling course may be set up in order to verify the following when the vehicle is in motion: turning radius, steering precision and the Driver's position inside the vehicle. In particular, Inspectors will verify that steering is precise, with no extra play,

Article 62: Braking

The vehicle must be equipped with a four-disc hydraulic brake system, with a brake pedal, which has a minimum surface area of 5 x 5cm.

The brakes must operate independently on the front and rear axles or in an X pattern (i.e. right front wheel with left rear wheel, and left front wheel with right rear wheel).

A single master cylinder may be used, provided that it has a dual circuit (two pistons and dual tank).

The effectiveness of the braking system will be tested during vehicle inspection for both Drivers. The vehicle must remain immobile when it is placed on a 20 percent incline with the main brake in place. Moreover, a dynamic inspection may be performed on the vehicle-handling course.

Race Inspectors may check the brakes again just prior to the start.

Article 63: Clutch and Transmission

Vehicles with internal combustion engines must be equipped with a clutch system, so that they can be immobilised on the starting line without any outside assistance.

The fitting of chain guard(s) is mandatory.

Article 64: Exhaust System

The exhaust gases must be evacuated outside the vehicle body

Exhaust pipes must not extend beyond the rear of the vehicle body

Article 65: Sound Level

The sound level for an Urban Concept vehicle must not exceed 90dB when measured 4 metres away from the vehicle.

Article 66: Emergency Shut-down mechanism

An emergency shutdown mechanism, accessible from the exterior, must be installed on all vehicles. A red arrow at least 10cm long and 3cm wide at the widest point must be positioned on the vehicle body to indicate clearly the position of this emergency shutdown mechanism from the exterior. This system must stop the engine and isolate the battery.

Article 67: Additional Inspections

After passing the technical inspection, the replacement of major engine or vehicle part will be subject to re-approval from Race Inspectors.

After any significant incident on the track the vehicle will be subject to a re-inspection.

At any time, the Organisers may perform unannounced inspections on the vehicles.

Spec. #	Category	Parameter Decription	Requirement or Target (units)	Tolerance	Risk or Mandated	Compliance	Notes
							M- Measured, I- Inspected, T- Test, A- Analysis
A	**Critical Dimensions**						
1A.		Total Height	100-130 (cm)	Range	Mandated	M	
2A.		Total Width	120-130 (cm)	Range	Mandated	M	
3A.		Total Length	220-350 (cm)	Range	Mandated	M	
4A.		Front Track Width	100 (cm)	Min.	Mandated	M	
5A.		Rear Track Width	80 (cm)	Min.	Mandated	M	
6A.		Wheelbase	120 (cm)	Min.	Mandated	M	
7A.		Driver's Compartment Height	88 (cm)	Min.	Mandated	M	
8A.		Driver's Compartment Width	70 (cm)	Min. at Driver's Shoulders	Mandated	M	
9A.		Ground Clearance	10 (cm)	Min.	Mandated	M	
10A.		Vehicle Weight	160 (kg)	Max.	Mandated	M	
		Wheels	4	Total	Mandated	I	
B	**Vehicle Body**						Must cover all mechanical Parts
1B.		No view of wheels from above			Mandated	I	
2B.		No view of wheels from from front above ground clearance			Mandated	I	
3B.		Side Door	10 (cm) max. above ground clearance to 10 (cm) min. below height		Mandated	I, M	Must open from inside/outside
4B.		Roof			Mandated	I	Must cover Driver's Compartment
5B.		Windscreen			Mandated	I	
6B.		Luggage Space	50x40x20 (cm LxHxW)		Mandated	I	Must have floor, sidewalls, be easily accessable from outside
7B.		Towing Hook	Hold 200 N	Min.	Mandated	I	Located on front
C	**Safety**						
1C.		Roll Bar:					
		Demensions	Extend to 5 (cm) above driver's head	Min.	Mandated	M	Extend in width beyond driver's shoulders
		Load	70 (kg)	Min.			Static loading with no roll bar deflection
2C.		Driver's Compartment			Mandated	I, T	Reasonable protection from side impacts
3C.		Front Bumper:					Placed on inside front wall
		Thickness	5 (cm)	Min.	Mandated	M	
		Density	28 (kg/m^3)	Min.	Mandated	A	

4C.		Location for Fire Extinguisher	2 (lb) capacity model		Mandated	I	Located in cockpit and accessable before and after driver exits
5C.		Firewall			Mandated	I	Permanent, rigid, fire-resistant between engine compartment and cockpit
6C.		Safety Harness	5 Mounting Points on Chassis		Mandated	I	One point to keep driver from slipping forward. Must withstand force of 1.5 times driver's weight.
D	**Visibility**						
1D.		Frontal Visibility	90 (degrees) from each side of longitudinal center line	Min.	Mandated	I, M	Checked with array of 7 60 cm high pylons spread every 30 degrees 5 meter radius in front
2D.		Rear View Mirrors	1 each side, 25 (cm^2)		Mandated	I, M	
E	**Lighting**						Accomodations for listed lights only
1E.		2 Front Headlights	30 (cm) from center to midline of vehicle	Min.	Mandated	M, I	
2E.		2 Front Turn Indicators			Mandated	I	
3E.		2 Rear Amber Turn Indicators			Mandated	I	
4E.		2 Red Read Brake Lights			Mandated	I	
5E.		Red Indicator Self Starter Light			Mandated	I	Indicates when vehicle starter is running
F	**Other**						
1F.		Wheel Bulkheads			Mandated	I	All wheels must be made inaccessable to driver
2F.		Sub-systems Accomadations					Chassis/Body to allow integration of all other vehicle sub-systems
G.	**Developed Specs.**						
1G.		Chassis Deflection	Chassis May not deflect below minimum ground clearance		M	I,T	When fully loaded and at ride height, chassis my not deflect below the 10 (cm) ground clearance
2G.		Suspension	2 (in) travel	Max.	M	A,I,T,M	Integrate with Club designed suspension
		Front	S.M. Club design				Possible A-arm suspension
		Rear	S.M. Club design				Possible trailing arm or flexible member

Appendix B – Decision Matrix

B.1 Decision Matrix

#	Concept / Criteria	WEIGHT FACTOR	COMPOSITE MONOCOQUE		COMPOSITE SPACE FRAME		STEEL SPACE FRAME		ALUMINUM SPACE FRAME		COMPOSITE BACKBONE		ALUMINUM BACKBONE		STEEL BACKBONE	
1	Weight	0.23	5	1.15	4	0.92	2	0.46	3	0.69	5	1.15	4	0.92	3	0.69
2	Safety	0.1	4	0.4	3	0.3	4	0.4	4	0.4	3	0.3	3	0.3	3	0.3
3	Subsystem Integration	0.09	3	0.27	3	0.27	4	0.36	4	0.36	3	0.27	4	0.36	4	0.36
4	Ease of Modification	0.09	2	0.18	2	0.18	4	0.36	4	0.36	2	0.18	4	0.36	4	0.36
5	Overall Strength	0.08	4	0.32	3	0.24	5	0.4	4	0.32	3	0.24	3	0.24	4	0.32
6	Frame Stiffness	0.08	5	0.4	4	0.32	5	0.4	5	0.4	4	0.32	4	0.32	4	0.32
7	Repairability	0.07	2	0.14	3	0.21	4	0.28	4	0.28	3	0.21	4	0.28	4	0.28
8	Testing and Analysis	0.06	3	0.18	3	0.18	4	0.24	4	0.24	4	0.24	4	0.24	4	0.24
9	Lower Cost	0.05	2	0.1	1	0.05	4	0.2	3	0.15	3	0.15	3	0.15	3	0.15
10	Design Modeling	0.03	3	0.09	3	0.09	4	0.12	4	0.12	4	0.12	4	0.12	4	0.12
11	Ease of Construction	0.03	3	0.09	3	0.09	3	0.09	3	0.09	3	0.09	3	0.09	3	0.09
12	Minimal Build Time	0.03	3	0.09	3	0.09	3	0.09	3	0.09	3	0.09	4	0.12	4	0.12
13	Ability to Prototype	0.03	2	0.06	3	0.09	4	0.12	4	0.12	3	0.09	4	0.12	4	0.12
15	Material Availability	0.03	2	0.06	2	0.06	4	0.12	3	0.09	2	0.06	3	0.09	4	0.12
		1		3.53		3.09		3.64		3.71		3.51		3.71		3.59

Appendix C – Drawings

C.1 Mannequin Drawings

C.2 Bending Test Setup

C.3 Isometric View

C.4 Chassis Assembly

C.5 Chassis Exploded View

C.6 Main Backbone Beam

C.7 Floor Panel

C.8 Dash Hoop

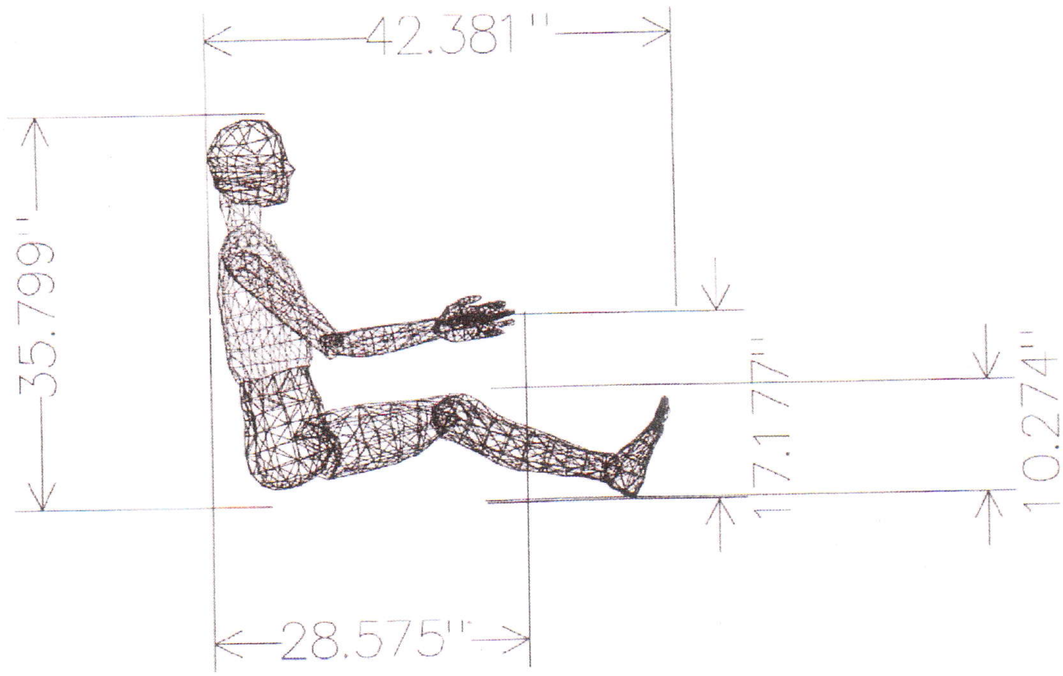
C.9 Roll Bar

C.10 Rear Compartment

C.11 Braces

C.12 Honeycomb Cut Pattern

C.13 Main Beam Cut Pattern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7.177"
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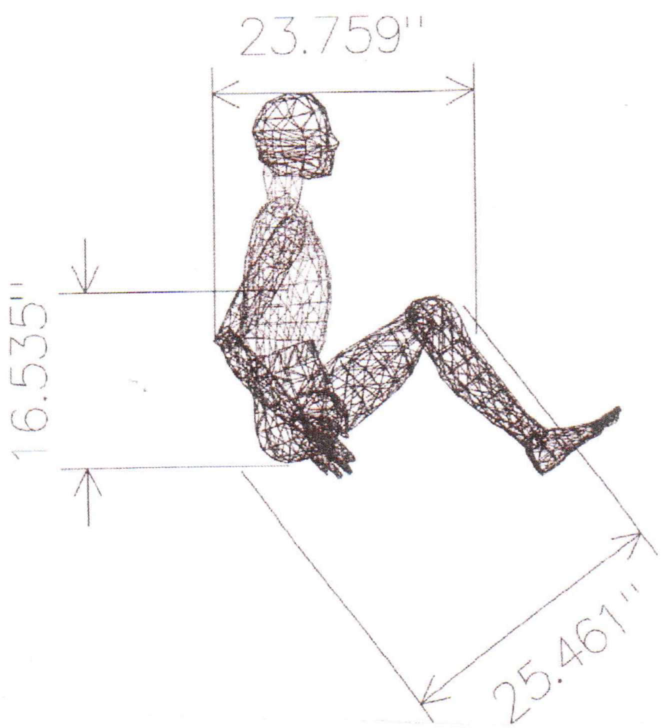
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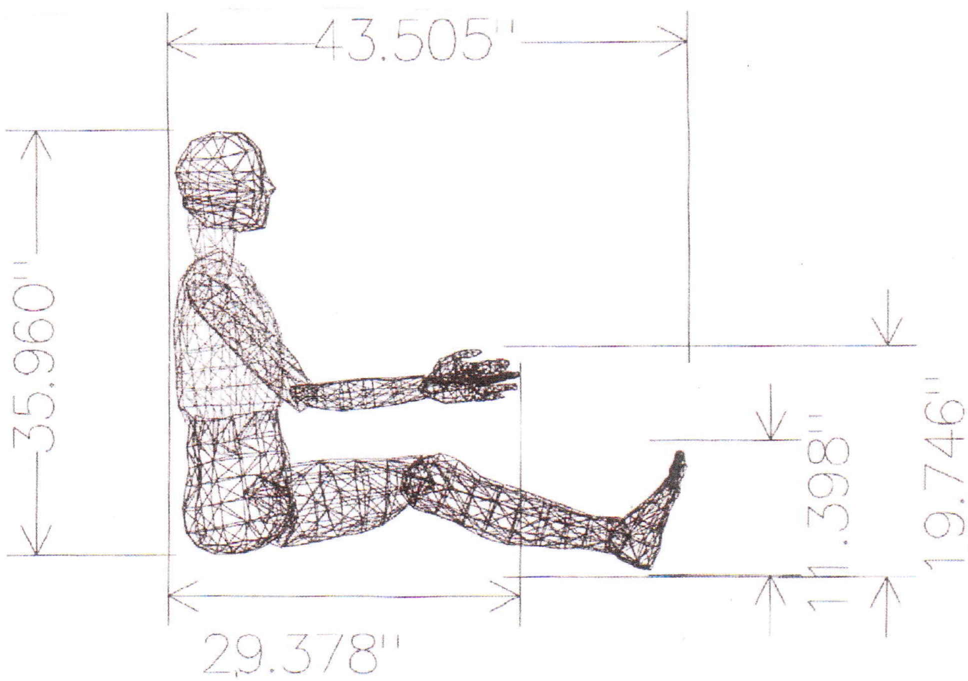
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11.398"
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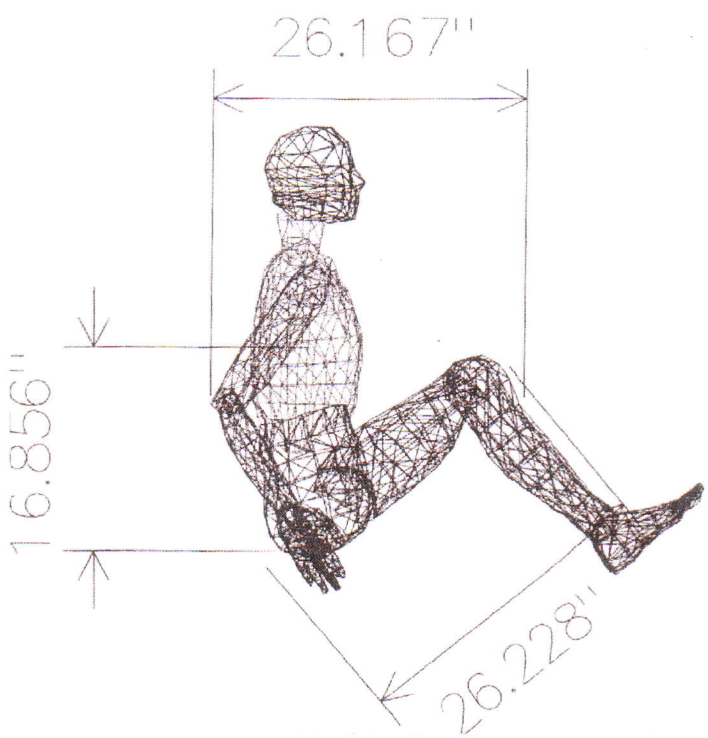
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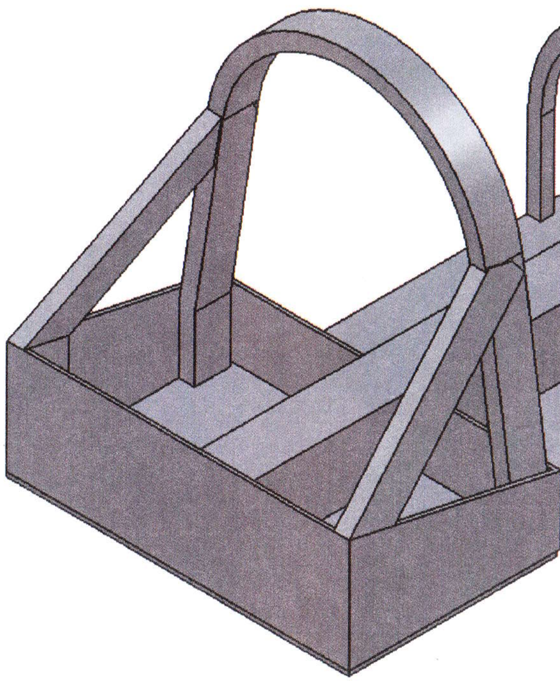
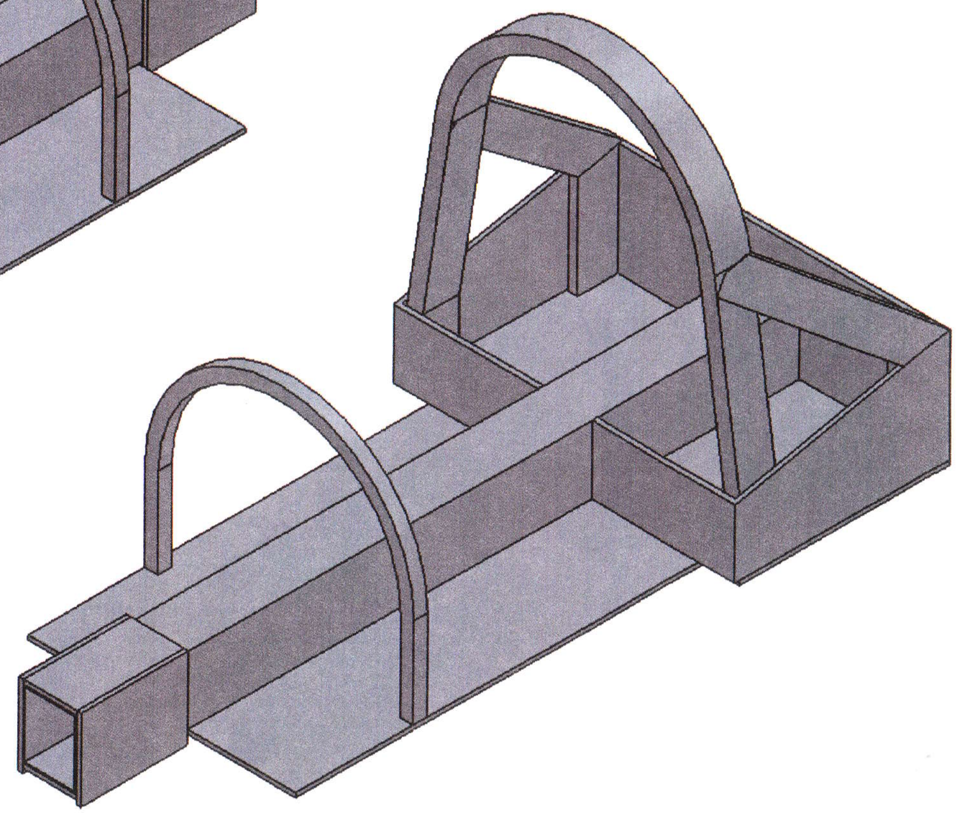
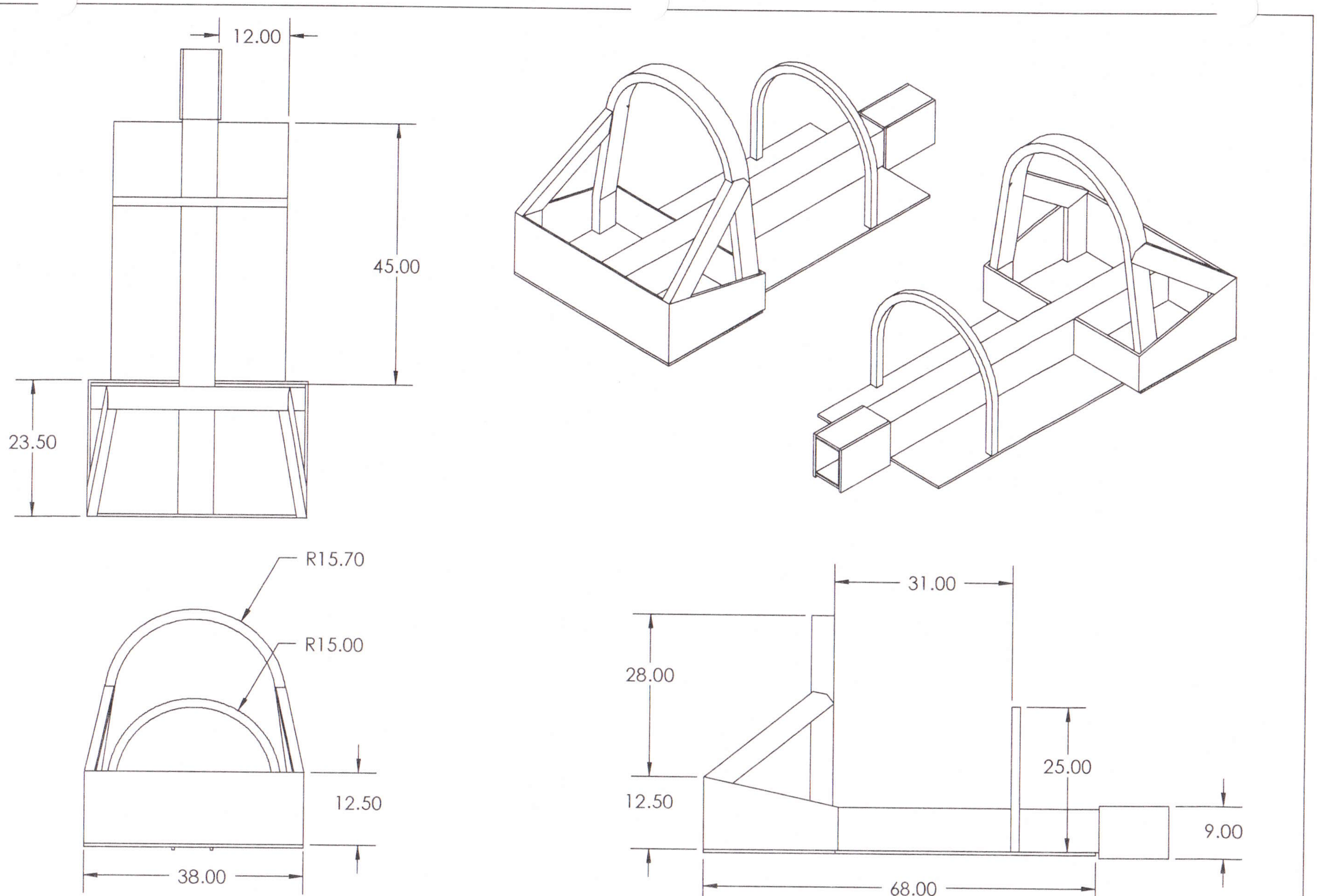
26.228"



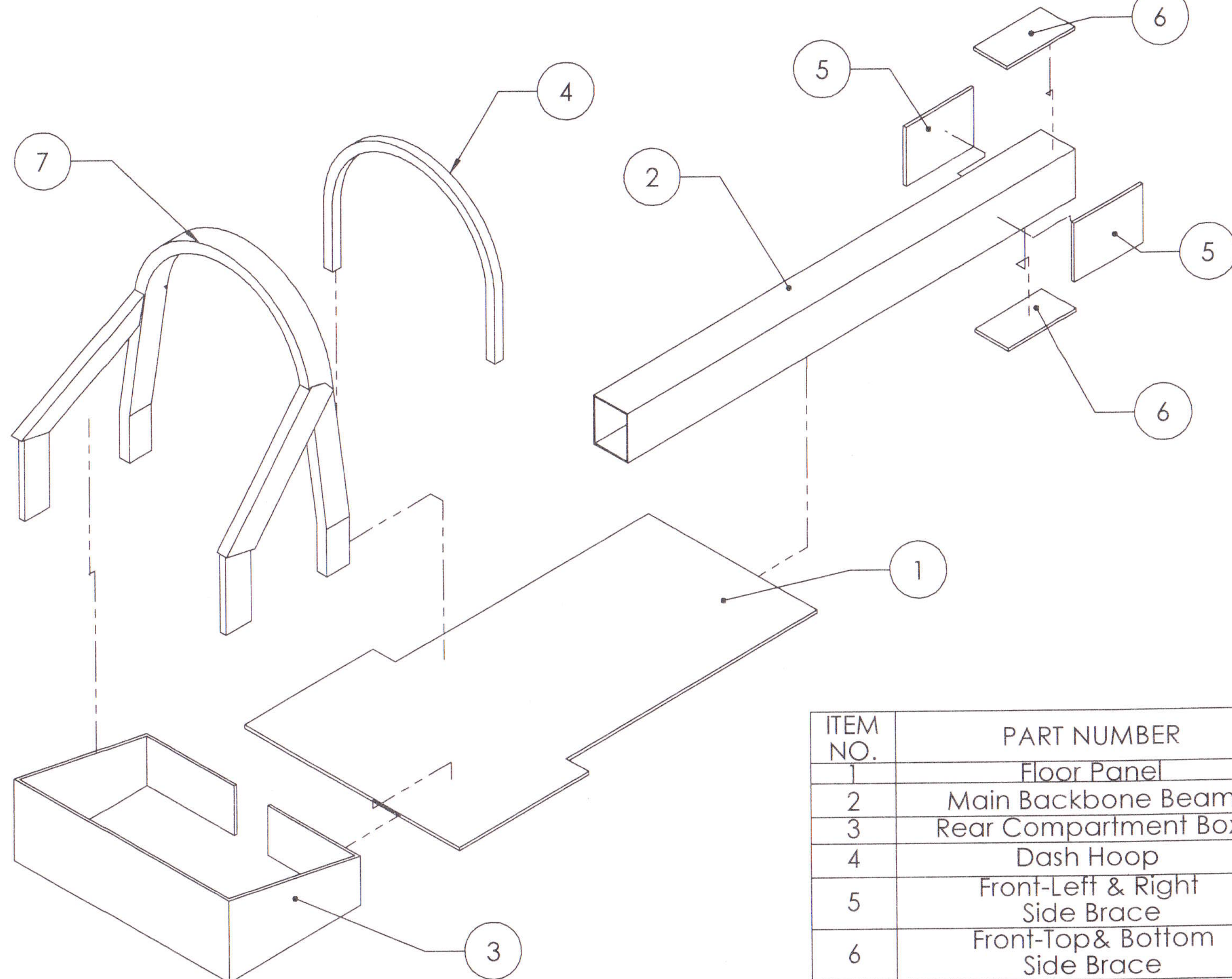


Conceptual Chassis Designs	Ckd by: **Kevin Charles**	Init:	Drawn By: **Wei Kyi**	Init:
	Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
	Tolerance: ± 0.125	Scale: **1:22**	Material: **6KPI50 & 12k Prepreg Carbon Fiber**	
		Drawing #: **CCD-ASSY1**	Title: **Chassis Isometric View**	



Conceptual Chassis Designs

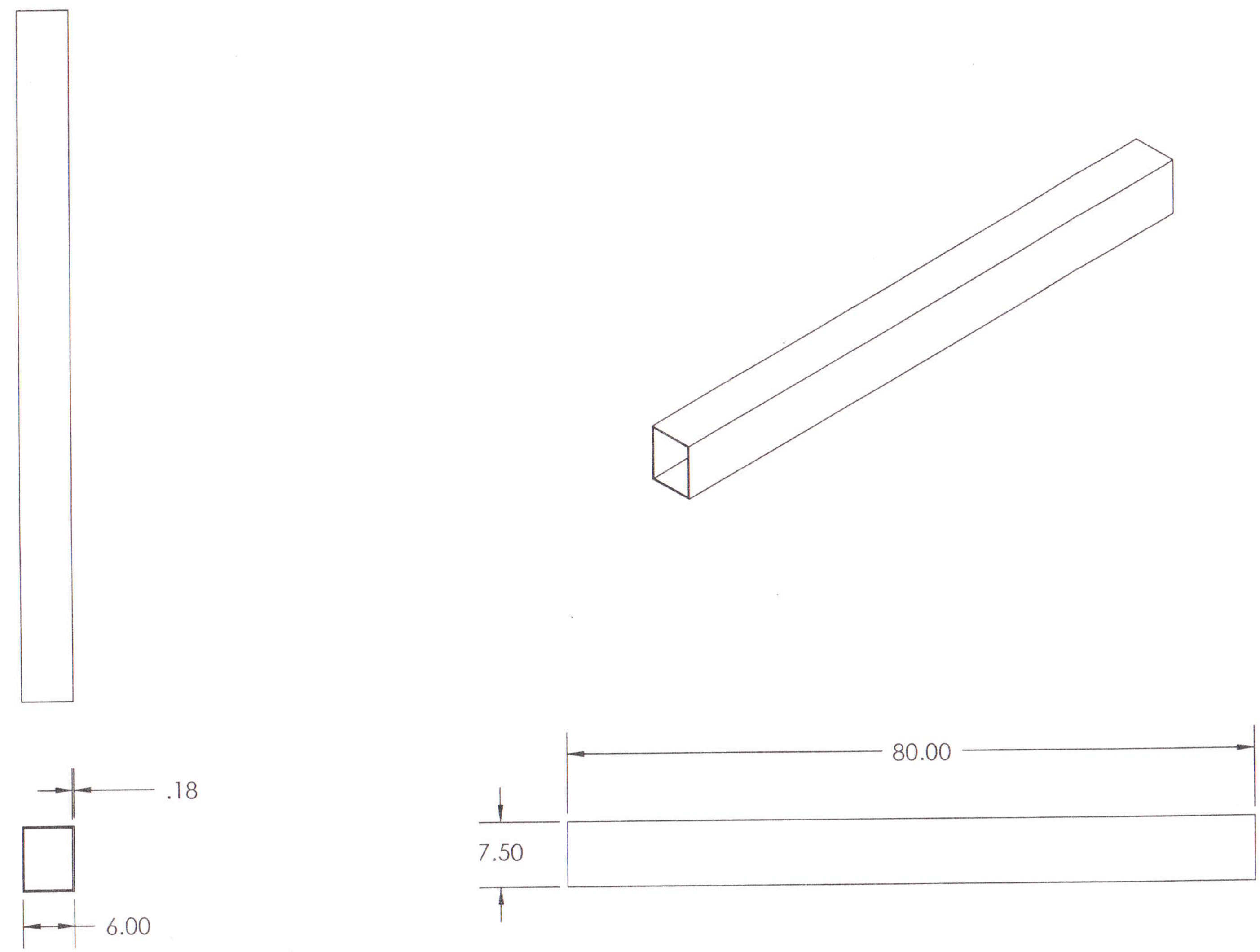
Ckd by: **Kevin Charles**	Init:	Drawn By: **Wei Kyi**	Init:
Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
Tolerance: ± 0.125	Scale: **1:24**	Material: **6kPL50 & 12k Prepreg Carbon Fiber**	
	Drawing #: **CCD-ASSY1**	Title: **Chassis Assembly**	



ITEM NO.	PART NUMBER	QTY.
1	Floor Panel	1
2	Main Backbone Beam	1
3	Rear Compartment Box	1
4	Dash Hoop	1
5	Front-Left & Right Side Brace	2
6	Front-Top& Bottom Side Brace	2
7	Roll Bar	1

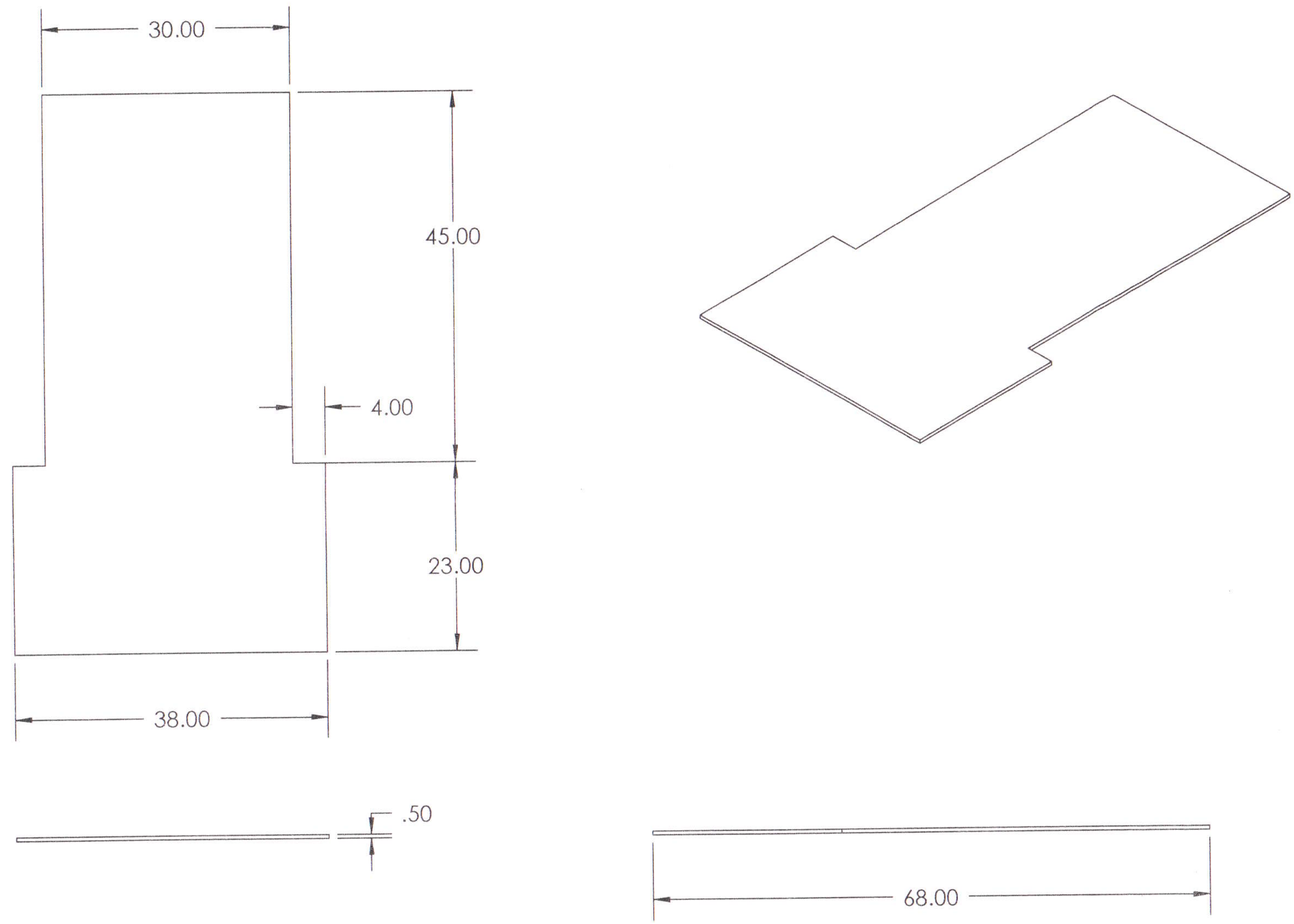
Conceptual Chassis Designs

Ckd by: **Kevin Charles**	Init:	Drawn By: **Wei Kyi**	Init:
Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
Tolerance: ± 0.125	Scale: **1:22**	Material: **6KPI50 & 12k Prepreg Carbon Fiber**	
	Drawing #: **CCD-ASSY2**	Title: **Chassis Exploded View**	



Conceptual Chassis Designs

Ckd by: **Wei Kyi**	Init:	Drawn By: **Andrew Allport**	Init:
Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
Tolerance: ± 0.125	Scale: **1:20**	Material: **6kPL50 Carbon Fiber**	
	Drawing #: **CCD-MBB1**	Title: **Main Backbone Beam**	



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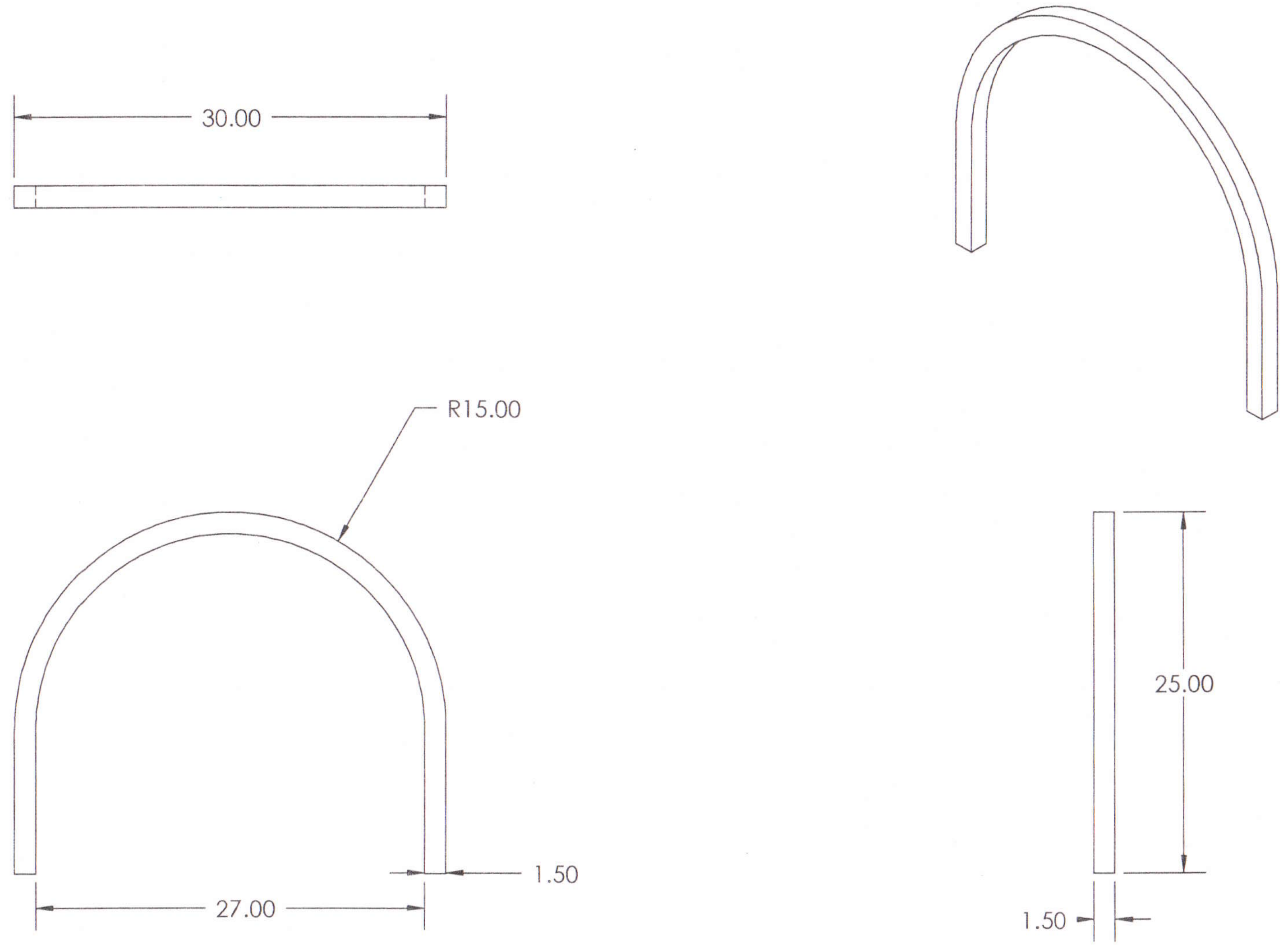
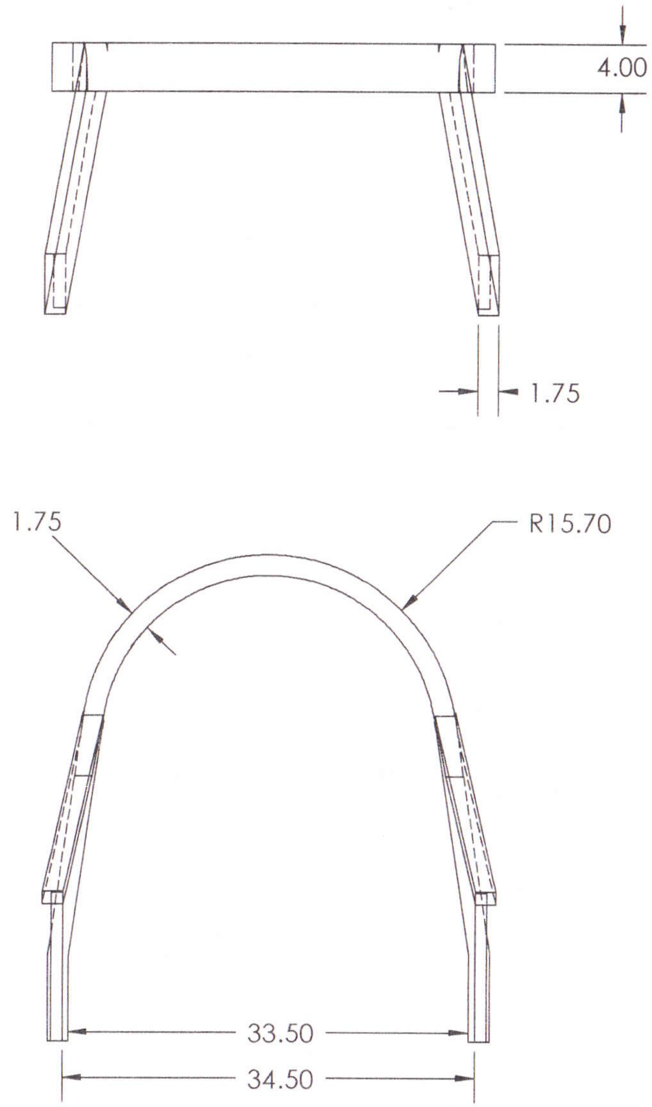
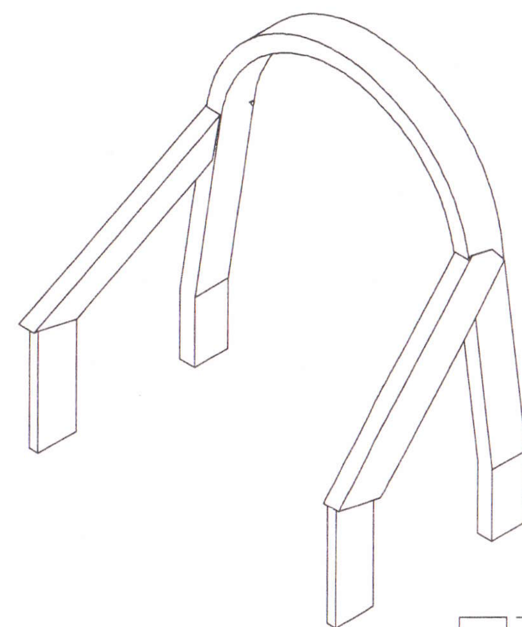
Date: **04/15/2009**	Units: **inches**		Group: **Conceptual Chassis Designs**	
Tolerance: ± 0.125	Scale: **1:20**		Material: **12k Prepreg Carbon Fiber**	
	Drawing #: **CCD-FP1**		Title: **Floor Panel**	

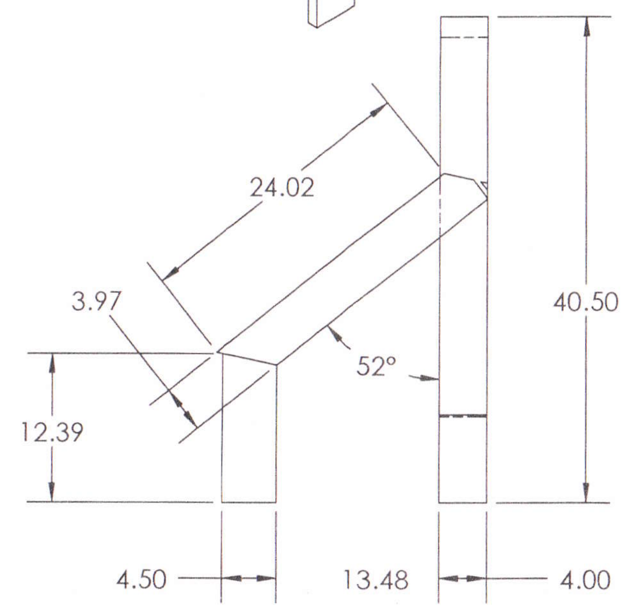
Conceptual Chassis Designs



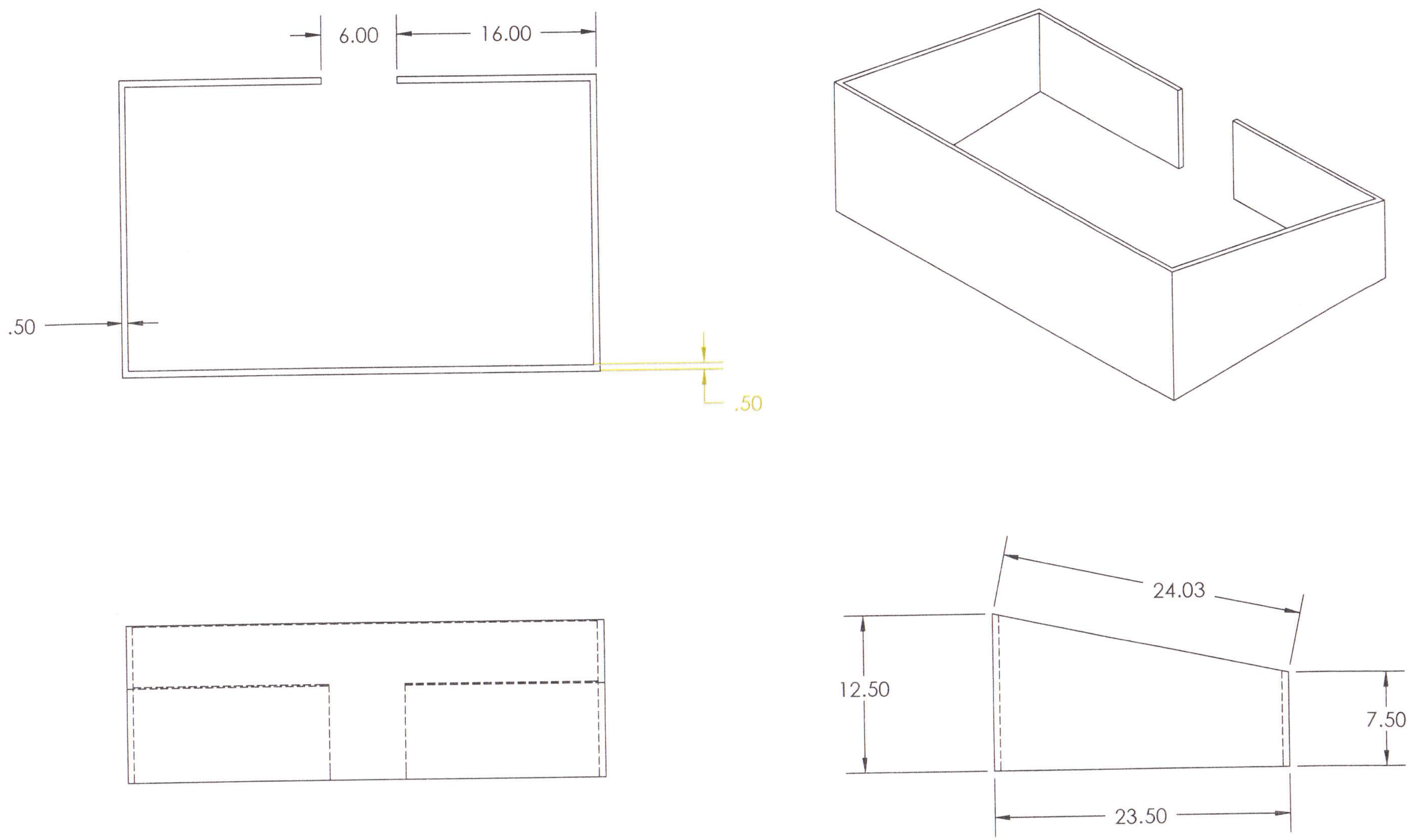
Conceptual Chassis Designs	Ckd by: **William Lai**	Init:	Drawn By: **Andrew Allport**	Init:
	Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
	Tolerance: ± 0.125	Scale: **1:12**	Material: **6kPL50 Carbon Fiber**	
		Drawing #: **CCD-DH1**	Title: **Dash Hoop**	





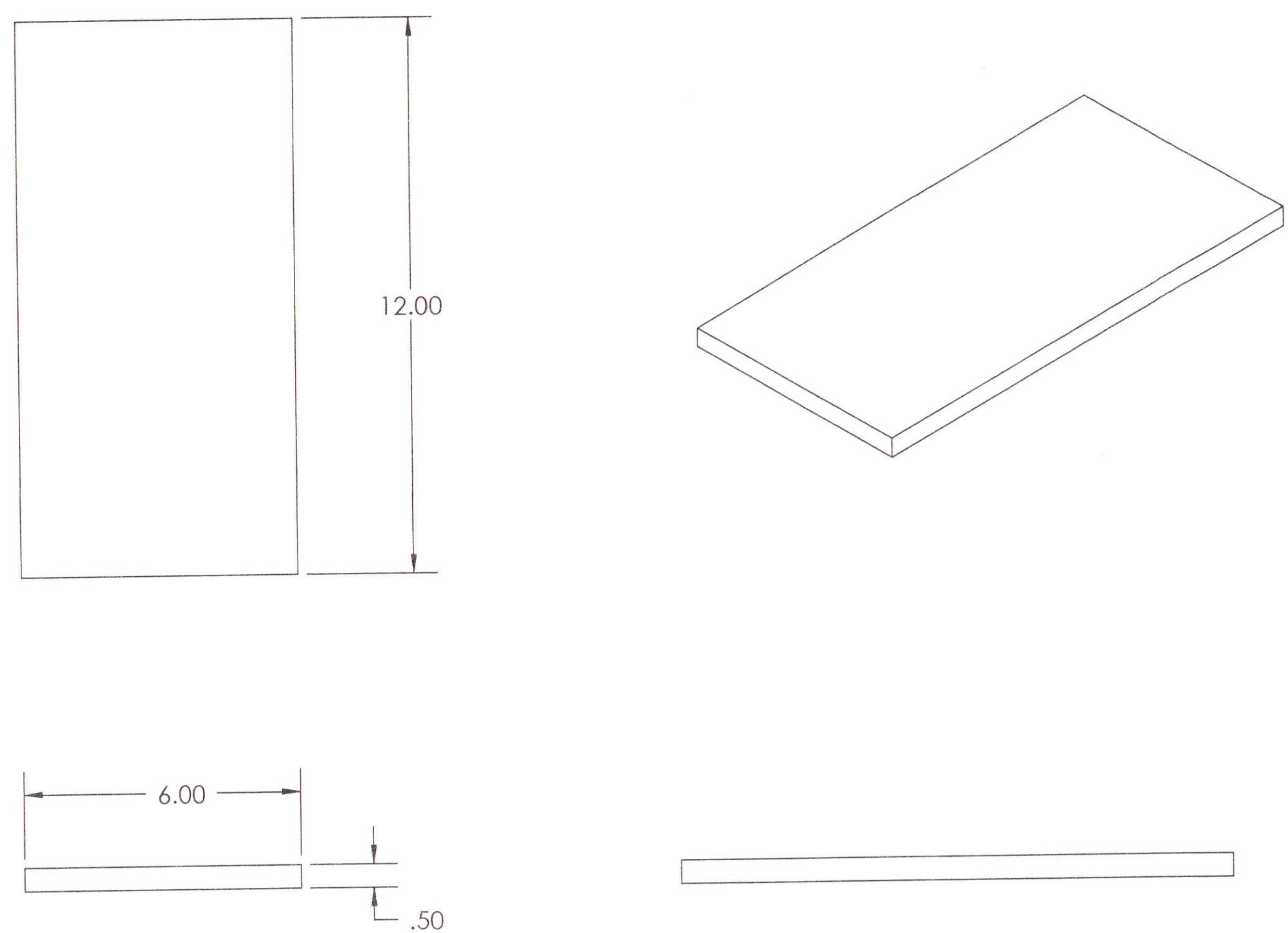


Conceptual Chassis Designs	Ckd by: **Wei Kyi**	Init:	Drawn By: **Andrew Allport**	Init:
	Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
	Tolerance: ± 0.125	Scale: **1:16**	Material: **6kPL50 Carbon Fiber**	
		Drawing #: **CCD-RB1**	Title: **Roll Bar**	

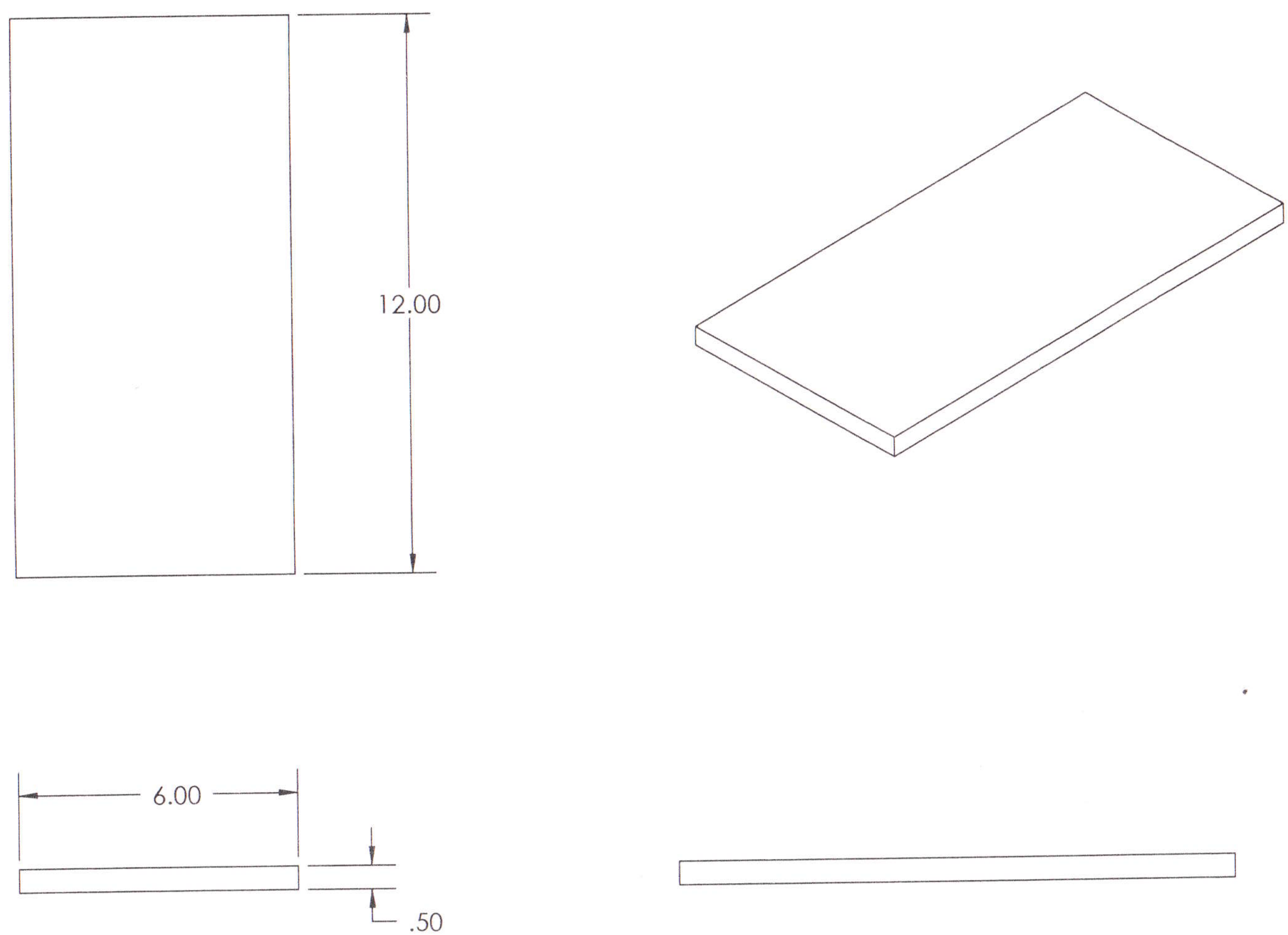


Ckd by: **Wei Kyi**	Init:	Drawn By: **Andrew Allport**	Init:
Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
Tolerance: ± 0.125	Scale: **1:12**	Material: **12k Prepreg Carbon Fiber**	
	Drawing #: **CCD-RCB1**	Title: **Rear Compartment Box**	

Conceptual Chassis Designs

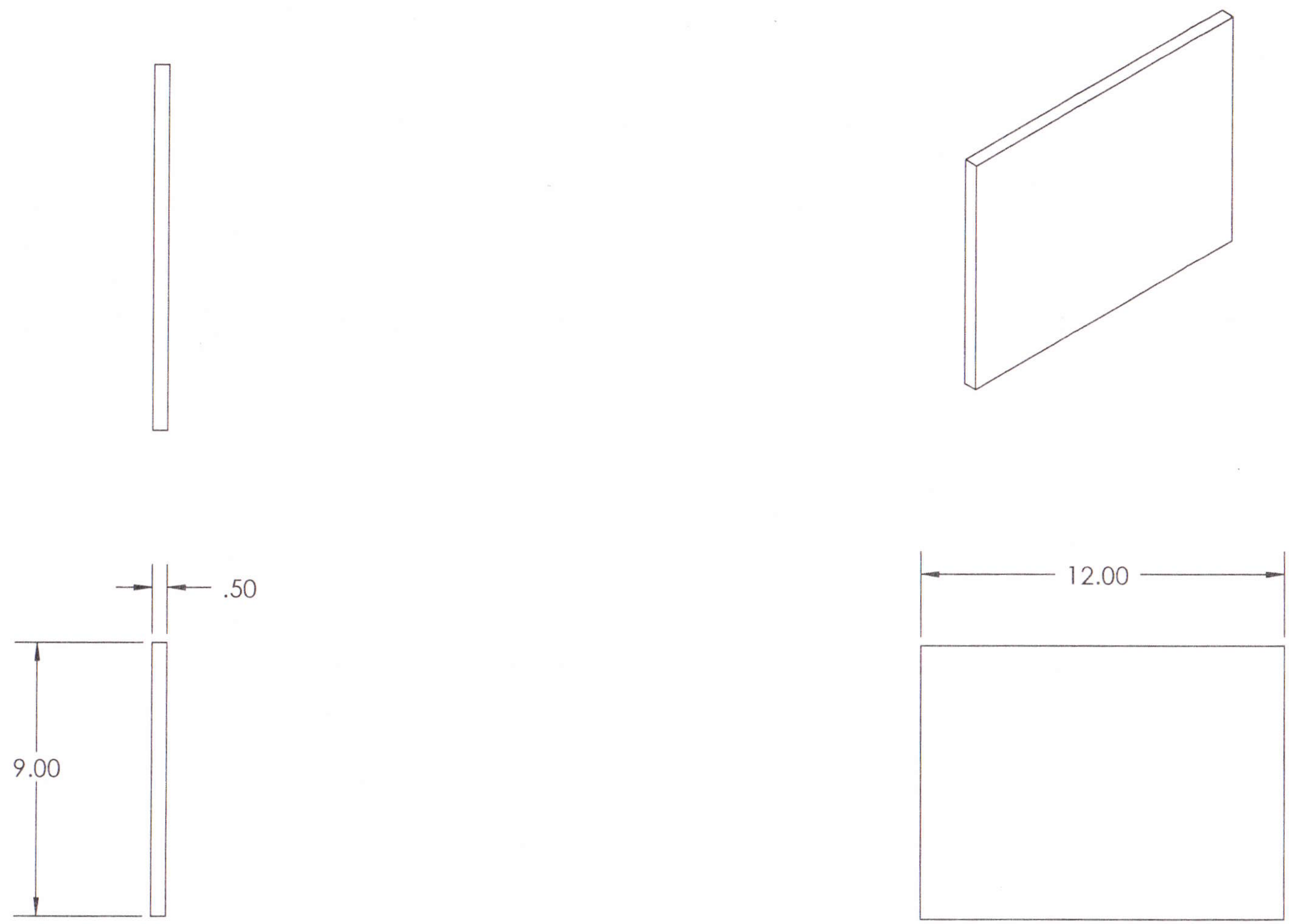


Conceptual Chassis Designs	Ckd by: **Wei Kyi** Init:		Drawn By: **Andrew Allport** Init:
	Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**
	Tolerance: ± 0.125	Scale: **1:6**	Material: **12k Prepreg Carbon Fiber**
		Drawing #: **CCD-FTSB1**	Title: **Front-Top Side Brace**



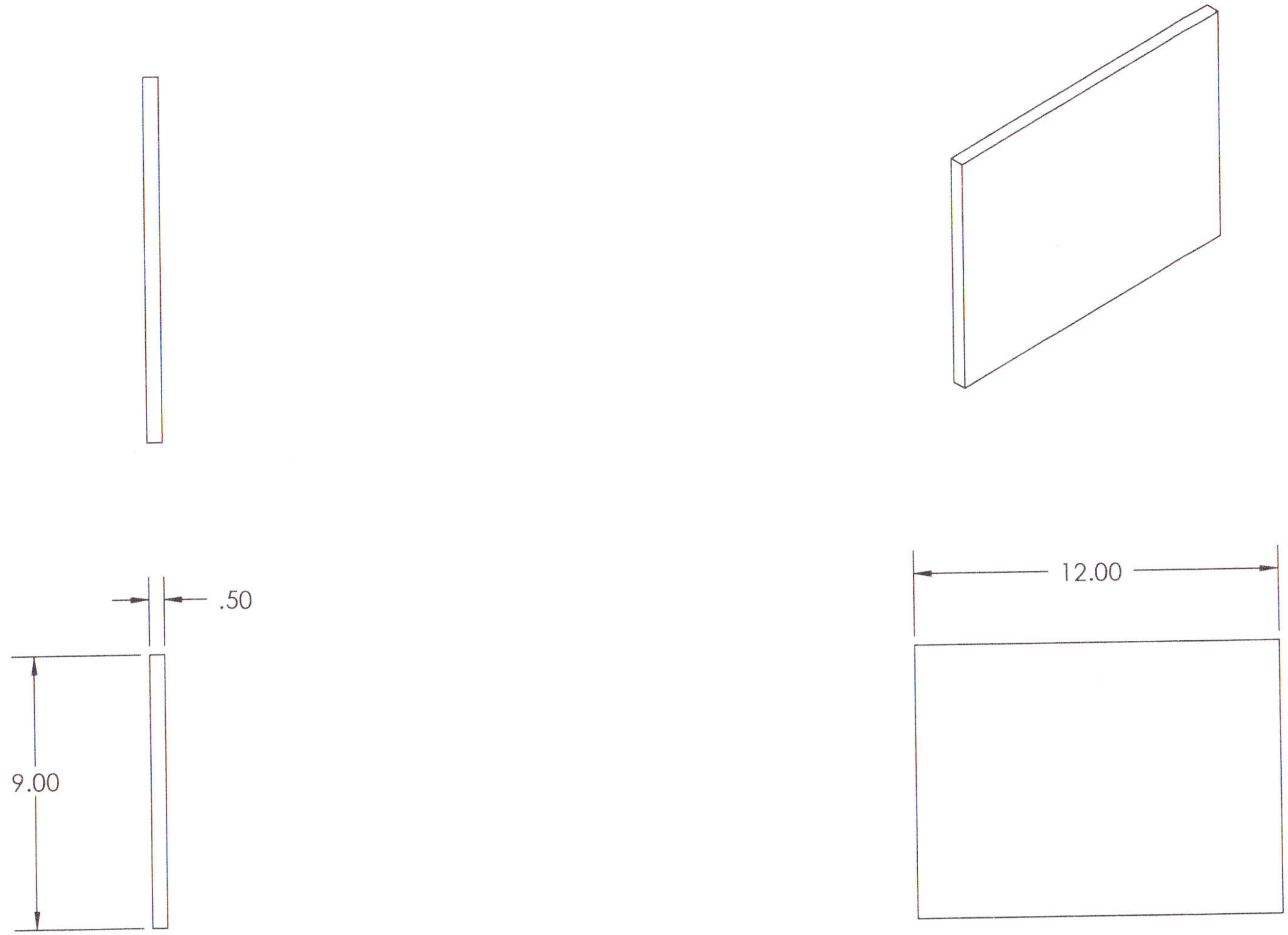
Conceptual Chassis Designs

Ckd by: **Wei Kyi**	Init:	Drawn By: **Andrew Allport**	Init:
Date: **04/15/2009**	Units: **inches**	Group: **Conceptual Chassis Designs**	
Tolerance: ± 0.125	Scale: **1:6**	Material: **12k Prepreg Carbon Fiber**	
	Drawing #: **CCD-FBSB1**	Title: **Front-Bottom Side Brace**	



Conceptual Chassis Designs

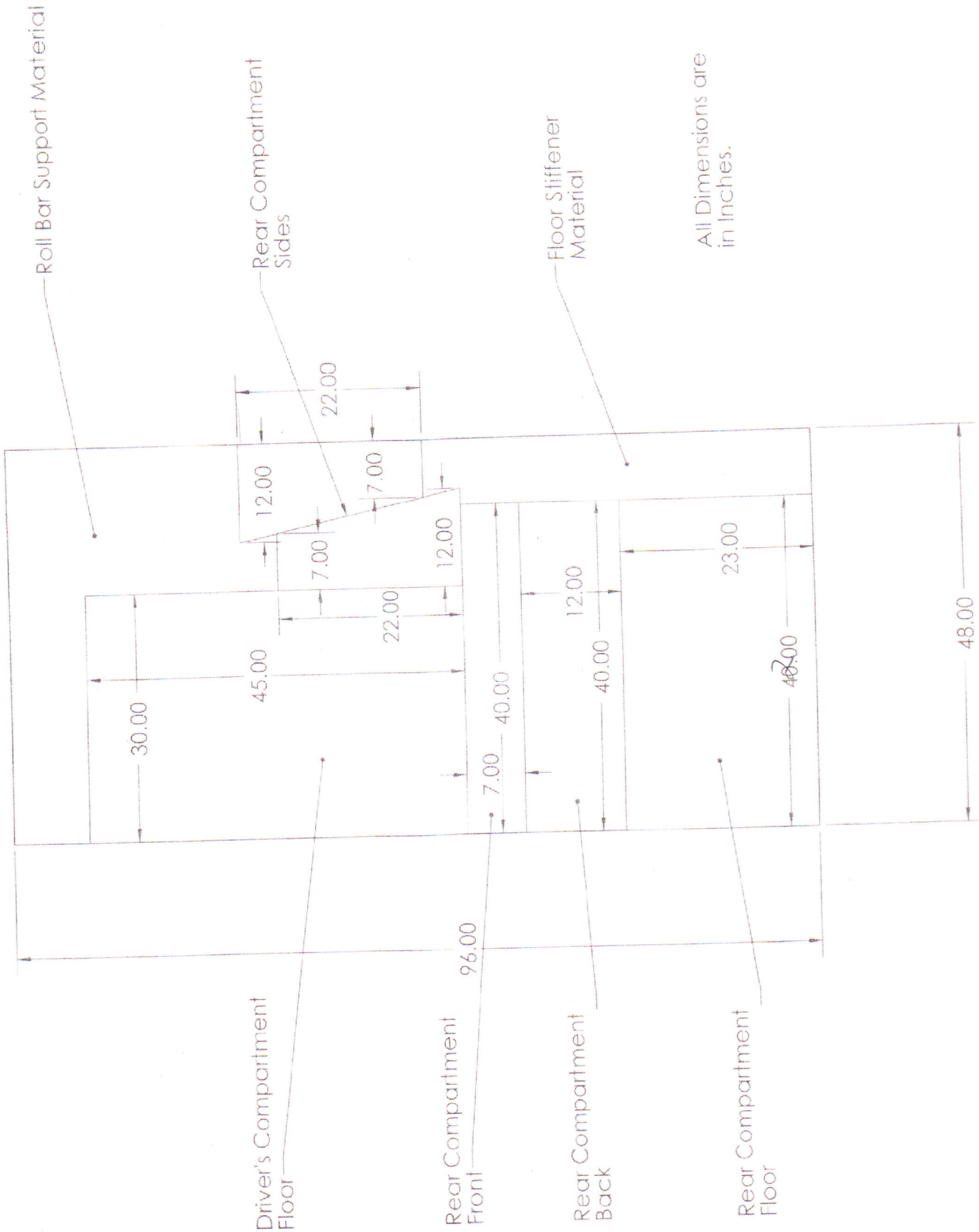
Ckd by: **Wei Kyi**		Init:	Drawn By: **Andrew Allport**	Init:
Date: **04/15/2009**	Units: **inches**		Group: **Conceptual Chassis Designs**	
Tolerance: ± 0.125	Scale: **1:6**		Material: **12k Prepreg Carbon Fiber**	
	Drawing #: **CCD-FLSB1**		Title: **Front-Left SideBrace**	



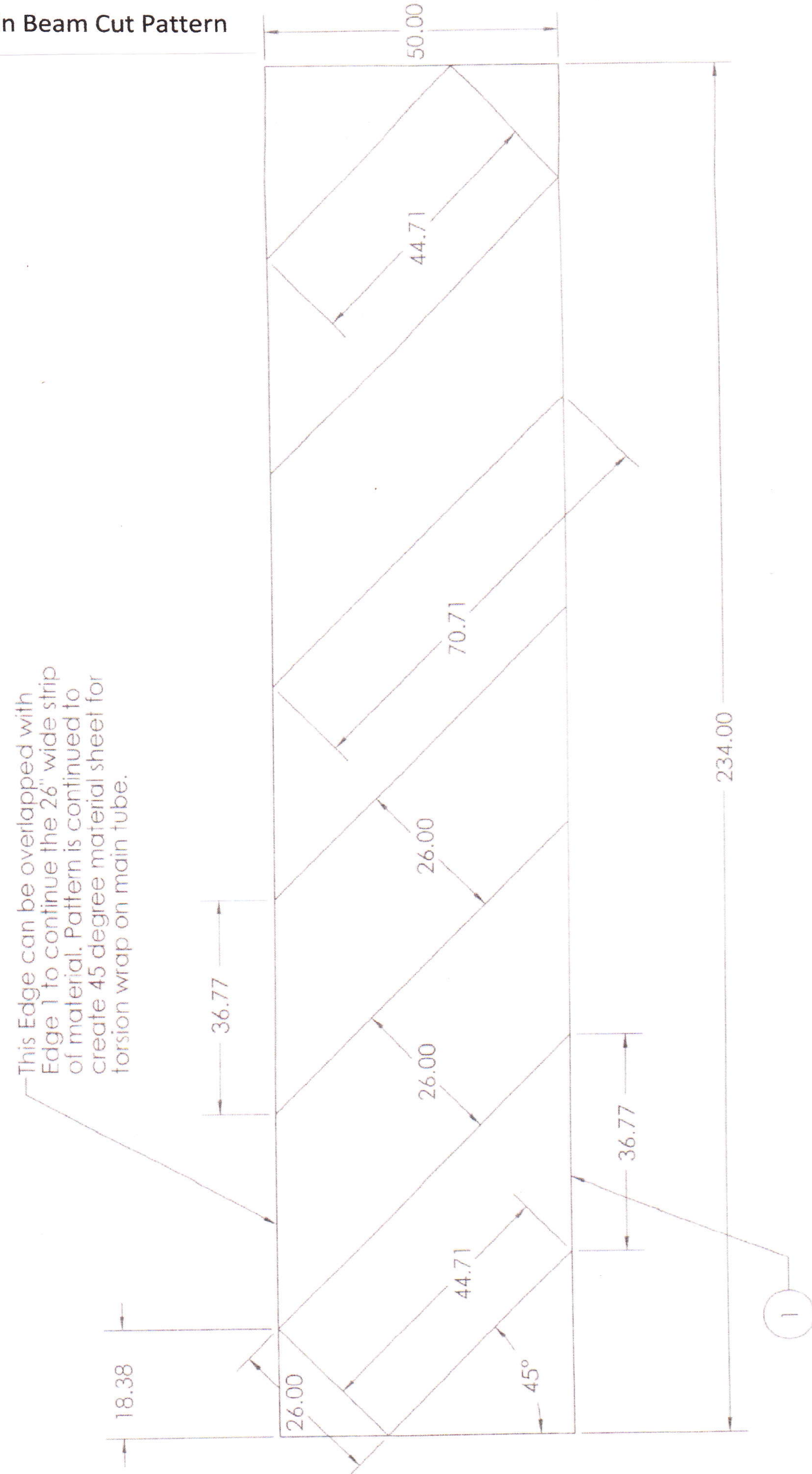
Conceptual Chassis Designs

Ckd by: **Wei Kyi**		Init:	Drawn By: **Andrew Allport**		Init:
Date: **04/15/2009**	Units: **inches**		Group: **Conceptual Chassis Designs**		
Tolerance: ± 0.125	Scale: **1:6**		Material: **12k Prepreg Carbon Fiber**		
	Drawing #: **CCD-FRSB1**		Title: **Front-Right SideBrace**		

Honeycomb Cut Pattern



Main Beam Cut Pattern



Appendix D – Detailed Supporting Analysis

D.1 Design Calculations

D.2 Floor Calculations

D.3 Dr. Mello's Composite Design Notes

Backbone Design Analysis

Assumptions

$W_{driver} := 150lb$

$W_{engine} := 30lb$

$W_{body} := .875\frac{lb}{in}$ *Distributed Load of Chassis and Various Components*

Length of Chassis

$L := 80in$

Distance from Rear to Front Suspension

$L_{Rb} := 75in$

$E := 8\cdot10^{6}\frac{lb}{in^{2}}$

Dimensions of Beam

$n := 3$ Number of Plies

$t_{ply} := .014in$ Thickness of Ply

$w := 5in$ Width of Beam

$B_o := 5in$

$B_i := B_o - (n\cdot.007\cdot2)in$

$d := 7in$ Height of Beam

$H_o := 7in$

$H_i := H_o - (n\cdot.007\cdot2)in$

$I := \frac{\left(n\cdot t_{ply}\cdot w\cdot d^{2}\right)}{2}$

$A := B_o\cdot H_o - B_i\cdot H_i = 0.502in^{2}$

Solving for Modulus of Rigidity for Carbon Fiber

$G_{LT} := 535200\frac{lb}{in^{2}}$ $\nu := .045$ $\theta_t := 45deg$

$S_{22} := \frac{1}{E} = 1.25\times10^{-7}\frac{in^{2}}{lb}$ $S_{66} := \frac{1}{G_{LT}} = 1.868\times10^{-6}\frac{in^{2}}{lb}$

$S_{11} := \frac{1}{E} = 1.25\times10^{-7}\frac{in^{2}}{lb}$ $S_{12} := \frac{-\nu}{E} = -5.625\times10^{-9}\frac{in^{2}}{lb}$

$S_{66bar} := 2(2S_{11} + 2\cdot S_{22} - 4\cdot S_{12} - S_{66})\cdot\sin^{2}(45deg)\cdot\cos^{2}(45deg) + S_{66}\left[\cos^{4}(45deg) + \sin^{4}(45deg)\right]$

$S_{66bar} := 2.3875\cdot10^{-7}\frac{in^{2}}{lb}$

$G_{star} := \frac{1}{S_{66bar}} = 4.188\times10^{6}\frac{lb}{in^{2}}$ $G := G_{star}$

Summing Equations

Moment From Rear

$$\Sigma M_a := 0$$

$$R_b := \frac{\left(W_{engine} \cdot 12in + W_{driver} \cdot 30in + W_{body} \cdot L \cdot \frac{L}{2}\right)}{\left(L_{Rb}\right)}$$

$R_b = 102.133\ lb$ Front

Moment From Front Suspension

$$\Sigma M_b := 0$$

$$R_a := \frac{\left[-R_b \cdot \left(L - L_{Rb}\right) + W_{engine} \cdot 68in - W_{driver} \cdot 50in + W_{body} \cdot L \cdot \frac{L}{2}\right]}{L}$$

$R_a = 147.867\ lb$ Rear

$$\Sigma F_y := 0$$

$R_a + R_b - W_{engine} - W_{driver} - W_{body} \cdot L = 2.35 \times 10^{-14}\ lb$ *Aprox 0 off due to rounding of lengths*

Bending Stress

Driver to Rear *(Driver seat being the location of our max moment)*

$$M_c := -R_a \cdot 30in + W_{engine} \cdot 18in + W_{body} \cdot 30in \cdot 15in$$

$$M_c = -3.502 \times 10^3\ in \cdot lb$$

Driver to Front

$$M_{c1} := R_b \cdot 50in - W_{body} \cdot 50in \cdot 25in$$

$$M_{c1} = 4.013 \times 10^3\ in \cdot lb$$

$$c := \frac{H_o}{2}$$

$$\sigma_b := M_c \cdot \frac{c}{I}$$

$$\sigma_b = -2.382 \times 10^3 \frac{lb}{in^2}$$

$$\sigma_{ultimate} := 100 \cdot 10^3 \frac{lb}{in^2}$$

$$SF := \frac{\sigma_{ultimate}}{\left|\sigma_b\right|} \qquad SF = 41.973$$

Loaded Deflection

Location of Deflection Observation X=0 @ Rear

$x := 30in$

Deflection for Weight of Backbone

$$\delta_b := \frac{(W_{body} \cdot x) \cdot \left(2 \cdot L_{Rb} \cdot x^2 - x^3 - L_{Rb}^{\;3}\right)}{24 \cdot E \cdot I}$$

$\delta_b = -8.341 \times 10^{-3}\,in$

Deflection due to Engine Weight

$$\delta_e := \frac{\left[W_{engine} \cdot 12in\left(L_{Rb} - x\right)\right] \cdot \left[x^2 + (12in)^2 - 2 \cdot L_{Rb} \cdot x\right]}{6 \cdot E \cdot I \cdot L_{Rb}}$$

$\delta_e = -3.023 \times 10^{-3}\,in$

Deflection due to Driver

$$\delta_d := \frac{\left[W_{driver} \cdot 30in\left(L_{Rb} - x\right)\right] \cdot \left[x^2 + (30in)^2 - 2 \cdot L_{Rb} \cdot x\right]}{6 \cdot E \cdot I \cdot L_{Rb}}$$

$\delta_d = -0.03\,in$

$\delta := \left(\delta_b + \delta_e + \delta_d\right)$

Total Vertical Deflection from Static Loads

$\delta = -0.041\,in$

Scenario With 1" Suspension Deflection Acting on Chassis

Spring Rate

$$K := (2Hz \cdot 2 \cdot \pi)^2 \cdot \frac{\left(\frac{R_b}{2}\right)}{32.2\,\frac{ft}{s^2}}$$

$K = 20.87\,\frac{lb}{in}$

Torsion

$n_t := 4$ *Number of Layers for Torsion (45deg)*

$D := 7in$

$d := 6.8in$

$T := K \cdot 20in \cdot 1in$

$T = 417.398\ in \cdot lb$

$G = 4.188 \times 10^{6} \frac{lb}{in^2}$ G_{star}

$L_m := 24in$

$A_m := 35in^2$

$t := n_t \cdot .011in$

$t = 0.044in$

$$\theta := \frac{(T \cdot L_m) \cdot L_{Rb}}{\left(4 \cdot G \cdot A_m{}^2 \cdot t\right)}$$

$\theta = 8.32 \times 10^{-4}$ radians

$$\theta_{deg} := \theta \cdot \frac{360}{2 \cdot \pi}$$

$\theta_{deg} = 0.048$

$\delta_{tor} := -20in \cdot \sin(\theta)$

$\delta_{tor} = -0.017in$

Bending Stiffness 1" Suspension Displacement

$P := 2 \cdot K \cdot 1in$

$$\delta_s := \frac{\left[-P \cdot \left(L_{Rb} - x\right)^3\right]}{3 \cdot E \cdot I}$$

$\delta_s = -0.031\ in$

Sum of all Deflections

$\delta_{max} := \delta + \delta_{tor} + \delta_s$

$\delta_{max} = -0.088\ in$

Studying an Open Thin-Walled Section (Modular Design)

Shear Stress

$$\tau := \frac{(3\cdot T)}{L_{Rb}\cdot(t)^2}$$

$$\tau = 8.624\times10^{3}\,\frac{lb}{in^2}$$

Angle of Twist

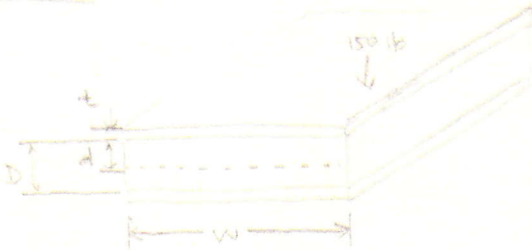
$$\theta_{modular} := \frac{(\tau\cdot L_{Rb})}{G\cdot(t)}$$

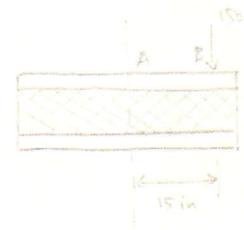
$\theta_{modular} = 3.51$ Radians

$$\theta_{modular\_deg} := \theta_{modular}\cdot\frac{360}{2\pi}$$

$\theta_{modular\_deg} = 201.085\ deg$ = FAILURE

FLOOR PANEL





$M_{AB} = F(x-a)$
$= 150\text{ lb}(15\text{ in})$
$= 2250\text{ lb}\cdot\text{in}$

$I = 2Ad^2$
$A = wt \quad \rightarrow w = 12.5\text{ in}$
$d = D/2 \quad \rightarrow D = 0.5\text{ in}$
$I = 2(wt)(D/2)^2$

1 LAYER

$t = 0.005\text{ in}$
$I = 0.0078125\text{ in}^4$

$$y_{max} = \frac{FL^3}{3EI} = \frac{150\text{ lb}(15\text{ in})^3}{3(34\times10^6\text{ lb/in}^2)(0.0078125\text{ in}^4)} = \boxed{0.035\text{ in}}$$

4 LAYERS

$t = 0.02\text{ in}$
$I = 0.03125\text{ in}^4$

$$y_{max} = \frac{(150\text{ lb})(15\text{ in})^3}{3(34\times10^6\text{ lb/in}^2)(0.03125\text{ in}^4)} = \boxed{0.24\text{ in}}$$

5 LAYERS

$t = 0.025\text{ in}$
$I = 0.0391\text{ in}^4$

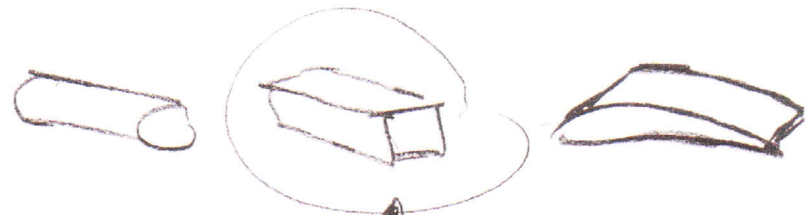
$$y_{max} = \frac{(150\text{ lb})(15\text{ in})^3}{3(34\times10^6\text{ lb/in}^2)(0.0391\text{ in}^4)} = \boxed{0.13\text{ in}}$$

Design of Composites

Apply beam / bar or bending / torsion mechanics of materials solution to composites design.

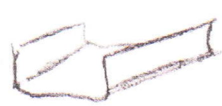
Geometry

- Tubes
  Thin-wall

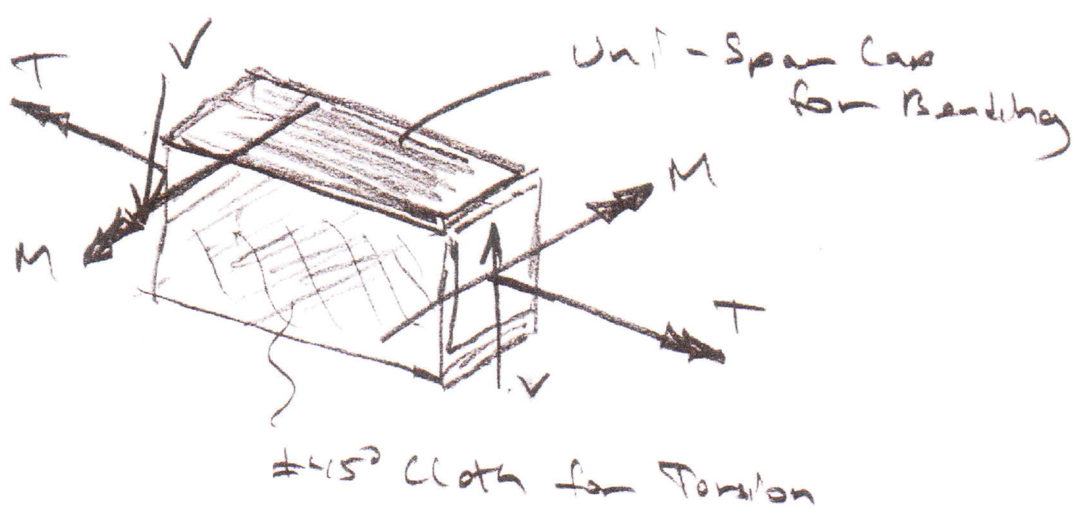


Focus on RECTANGULAR Tube

- Open sections



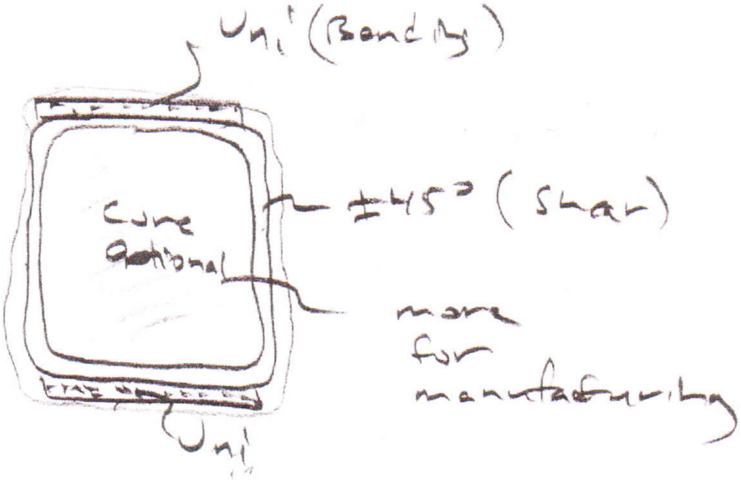
Focus on rectangular tube for now



Assumed loading from V, M, T diagrams

Each Section can be sized

Proposed Composite Design



Local Laminate Coords X runs along the span

Preliminary Design

- Ignore ply interaction in laminate
- Put the fiber in the direction of the load
- Look at each load independently and then superimpose results

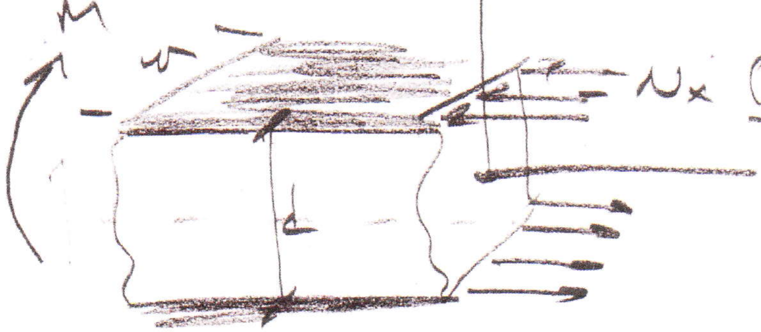
Look at these seperately

Bending

Torsion

Direct Shear

*Bending



Bending carried by Force Couple ($N_x\ w$)

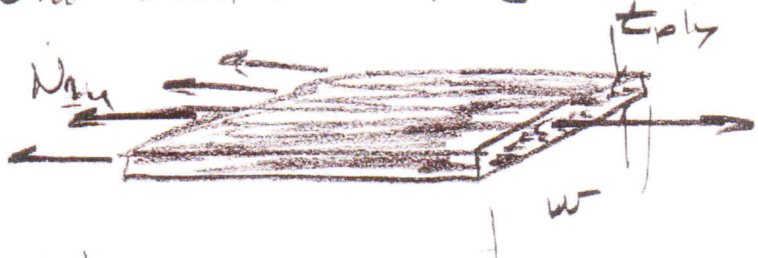
(SPAR CAPS)

$\circlearrowleft + \Sigma M = 0$

$$N_x (w)\, d - M = 0$$

$$\underline{N_x = \frac{M}{w\, d}}$$ applied line load

Uni Lamina strength



$\underline{\sigma_{1u}} \approx 200\text{–}250$ ksi fiber tensile strength (note σ'_{1u} compression strength is lower

N_{1u} ← allowable line load

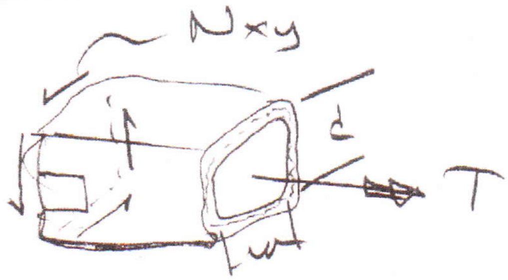
$$N_{1u} \cdot \cancel{w} - \sigma_{1u}\, \cancel{w}\, t = 0$$

$$\underline{\underline{N_{1u} = \sigma_{1u}\, t_{ply}}}$$

\# of plies needed spar cap uni

$$\#\text{plies} = \frac{N_x}{\underbrace{N_{1u}(SF)}_{\text{safety factor}}}$$

* Torsion



w, d are mean dimensions to middle of laminate

N_{xy} (shear flow) another line load

From Mechanics of Materials Solution

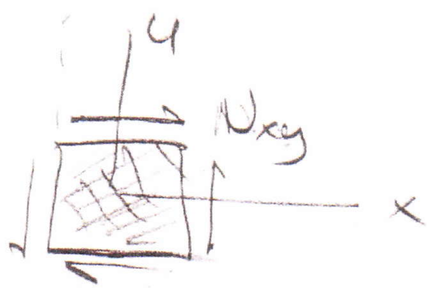
$$N_{xy} = \frac{T}{2A_m}$$

thin closed shell in torsion

$$A_m = wd$$

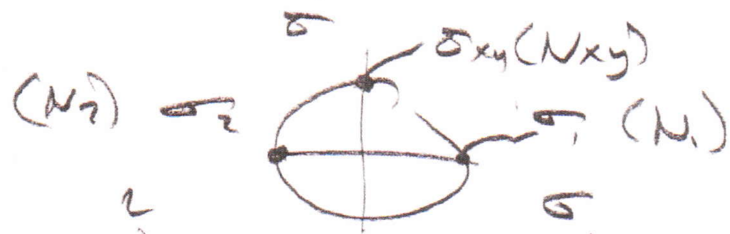
mean enclosed area

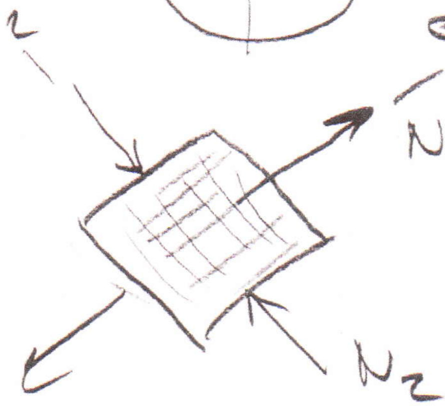
$$N_{xy} = \frac{T}{2wd}$$



Nxy transforms just like stress

recall pure shear $\qquad \sigma_{xy} = N_{xy}/t$





$$\underline{N_1 = -N_2 = N_{xy}}$$

Transformed Line Loads

Cloth is ideal for this.

Balanced weave has strengths approx

$$\sigma_{1u} = \sigma_{2u} \approx 100 \text{ ksi}$$

compressive is less

$$\sigma'_{2u} = \sigma'_{1u} \approx 80 \text{ ksi (check)}$$

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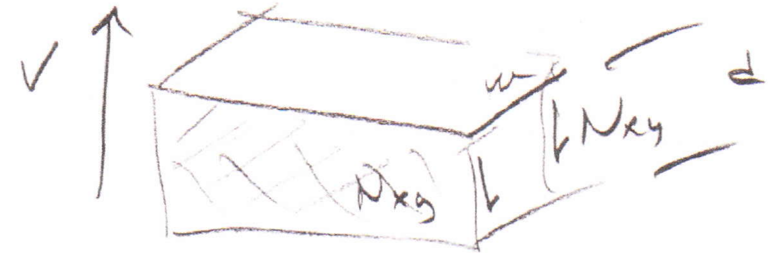
— use compressive

$N_{2u} = N_{1u} = \sigma_{1u}' \, t_{cloth}$

$$\#\ \text{cloth plies} = \frac{N_1}{N_{1u}'(SF)}$$

* Direct Shear

In mechanics of materials an "I" beams webs carry nearly all the shear.



For the composite design assume the two side webs of cloth carry the shear they are analogous to the I web.

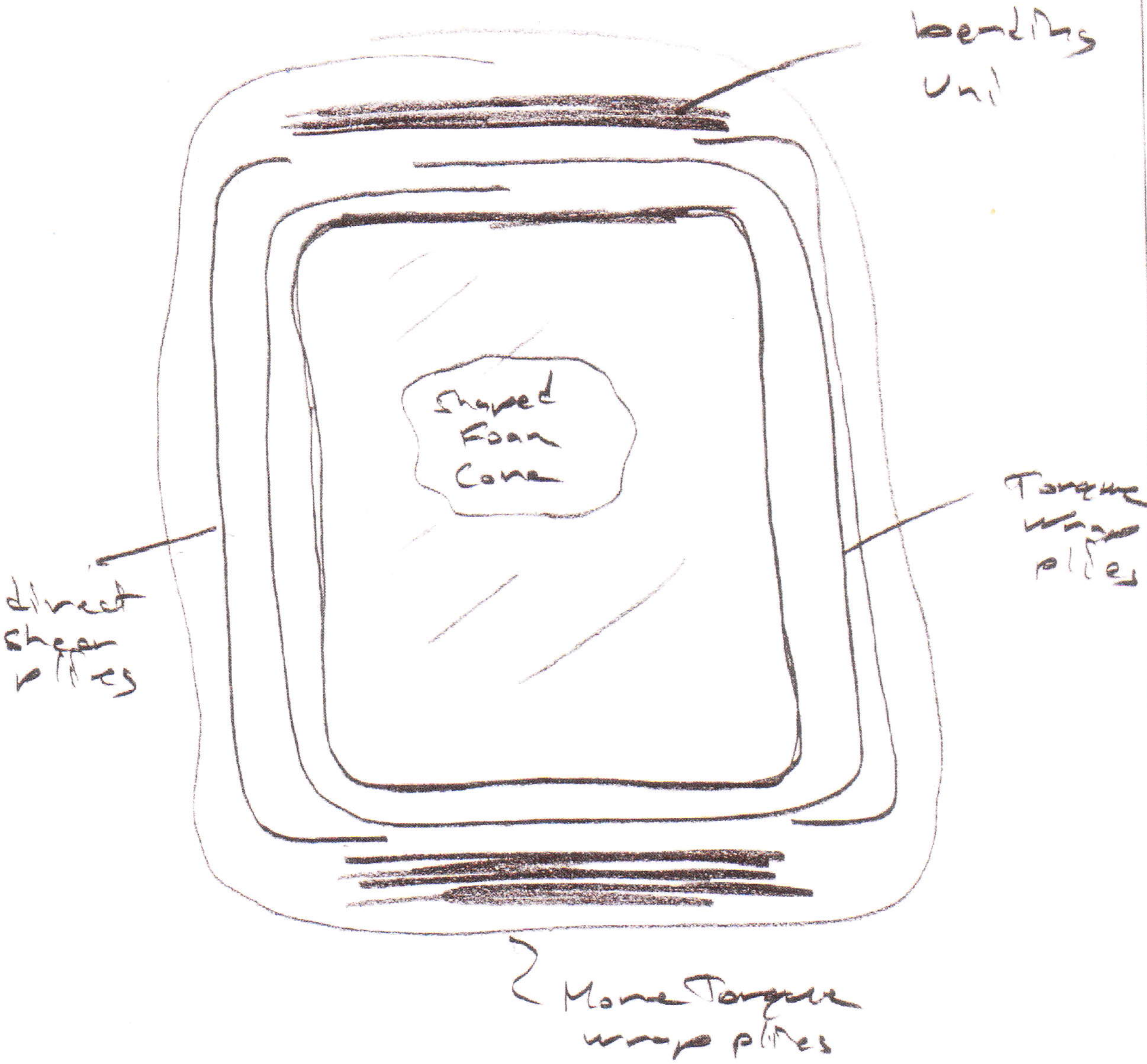
$\Sigma F_v = 0 \qquad V - 2N_{xy}d = 0$

$$\underline{N_{xy} = \frac{V}{2d}}$$

This might mean a few more cloth plies on the sides is all.

Transform to $N_1 = N_2 = N_{xy}$ and use allowable as with Torsion shear

So the design may look like this



This is a Strength based design

Stiffness

Note all this was sized based on strength.

Stiffness drives the designs usually.

δ_{bend} : $\frac{EI}{L^2}$ E, for uni $I = AD^2$

θ : $\frac{KG}{L}$ Effective G or Gxy for 45° cloth

depends on section

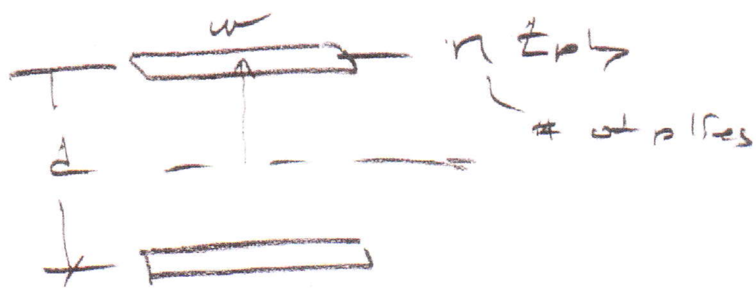
δ_s : $\frac{GA}{L}$ Shear Defo

Bending

EI

$\underline{E = E_1}$ fiber direction modulus

$E_1 \approx 17 - 20$ msi for T700 fiber



// axis Thm

$$I = 2A\left(\frac{d}{2}\right)^2, \quad A = w(n\, t_{ply})$$

$$I = \frac{2Ad^2}{4}$$

$$I = \frac{Ad^2}{2}$$

$$I = \frac{n\, t_{ply}\, w\, d^2}{2}$$

$$\boxed{EI = \frac{E_1 (n\, t_{ply}\, w)\, d^2}{2}}$$

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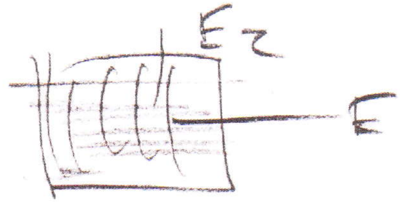
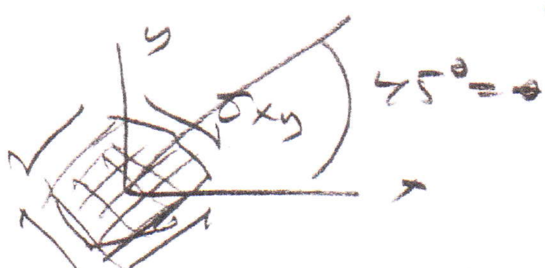
Torsion, Direct Shear

Need G_{xy} for fiberglass cloth
in this case at 45°



$E_1 = E_2 = \sim$ 8 – 10 msi
for cloth



for pure shear — transformed compliance

$$\begin{Bmatrix} \epsilon_x \\ \epsilon_y \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} \text{etc.} & & \\ & & \\ & & \bar{S}_{66} \end{bmatrix} \begin{Bmatrix} \cancel{\sigma_x} \\ \cancel{\sigma_y} \\ \tau_{xy} \end{Bmatrix} \begin{matrix} 0 \\ 0 \\ \\ \end{matrix}$$

equation 5.96 ABC et al
pg 192

$$\gamma_{xy} = \bar{S}_{66}\,\tau_{xy}$$

$$\tau_{xy} = \gamma_{xy} / \bar{S}_{66}$$

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35500

$$\tau_{xy} = G_* \gamma_{xy}$$

G_* is effective shear modulus

$$G_* = 1/\bar{S}_{66}$$

$\bar{S}_{66} = ($ 4th Eqns – – $)$ 5.97 pg 192

need S_{11}, S_{22}, S_{66}

$S_{11} = 1/E_1$ 5.77 pg 185

$S_{22} = 1/E_2$

$S_{66} = 1/G_{LT}$

$\frac{1}{G_{LT}} = \frac{V_f}{G_f} + \frac{V_m}{G_m}$ (3.51) Pg 93

Appendix E – Component Specifications

E.1 Cal Poly Composite Lab Test Results

E.2 Published Properties for Tested Materials

E.3 West Systems Resin

E.4 Bill of Materials

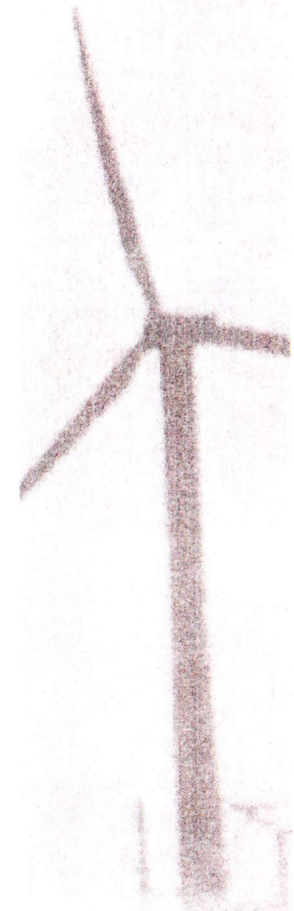
E.5 Carbon Fiber Lay-up Schedule

Results from Tests

Material Properties

	6k Carbon		E-Glass	
	Avg	S.D.	Avg	S.D.
Tangent Modulus of Elasticity, Fs (ksi) [short beam test]	93.186	0.258	23786.	5.6
Apparent shear strength, S_H (psi) [ILSS test]	6929	396	4670	450
Ultimate shear strength, τ_{12} (psi) [45 tensile test]	9673	**	10500	**
Ultimate Tensile Force, F_{1t} (lb)	4820	67	1042	137
Ultimate tensile strength, σ_{tu}(ksi)	104.6	3.1	40.5	5.5
Modulus of Elasticity, E_t(Msi) [tensile test]	10.0	**	3.006	0.117
Poisson's Ratio, ν_{12}	*		0.153	0.0156
Max Shear Stress (psi) [tensile test]	27120	287	9040	287
Shear Modulus, G_{12}	535200	**	521000	**

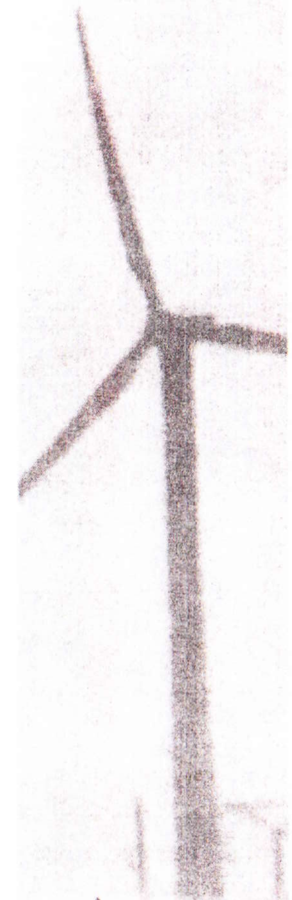
* - In the process of testing for these properties.
** - not enough specimens have been tested yet for the standard deviation to be calculated.



Published Material Properties

	AS4 6k Carbon Weave	E-Glass EA 9396 8-HS Weave
	Mean Values	
Ultimate Tensile Strength, σ_{tu} (ksi)	114	48.3
Tensile Modulus, E_1 (Msi)	9.61	3.39
Ultimate Compressive Strength, σ_{cu} (ksi)	104	46.4
Compressive Modulus, E_c (Msi)	8.49	3.45
Ultimate Shear Strength, τ_{12} (ksi)	12.6	11.5
Shear Modulus, G_{12} (Msi)	0.514	0.758
Poisson's Ratio, ν_{12}		0.115

- Values in this table were found in Mil-Handbook 17, "The Composites Handbook"
- These are pre-preg values



West Systems Resin- Physical Properties

Test specimens cured at room temperature for two weeks, unless otherwise noted. Typical values; not to be construed as specifications. Neat epoxy specimens *(i.e.*, containing no fillers or fiber reinforcements) were used for testing.

Property	105/205	105/206	105/207	105/209	Six10	G/flex
Mix Ratio by weight*	5.07:1	5.0:1	3.4:1	3.56:1		1.2:1
Mix Viscosity @ 72°F (cPs)	975	725	775	650		15,000
Pot Life of 100 g @ 72°F (min.)	12	21.5	26.4	62	42	45
Specific Gravity of Cured Resin	1.180	1.180	1.16	1.16		
Hardness @ 1 day (Shore D)	80	80	78	77		70
Hardness @ 2 weeks (Shore D)	83	83	82	82	80.6	75
Compression Yield @ 1 day (PSI)	10,120	7,990	6,014	3,835		
Compression Yield @ 2 weeks (PSI)	11,418	11,500	10,838	11,960	9,693	5,268
Tensile Strength (PSI)	7,846	7,320	7,509	7,280	6,438	3,440
Tensile Elongation (%)	3.4	4.5	3.4	3.6		32.7
Tensile Modulus (PSI)	4.08E+05	4.60E+05	4.10E+05	3.98E+05	3.71E+05	1.44E+05
Flexural Strength (PSI)	14,112	11,810	13,016	12,459	11,320	5,192
Flexural Modulus (PSI)	4.61E+05	4.50E+05	5.14E+05	3.97E+05	3.51E+05	1.56E+05
Heat Deflection Temperature (°F)	118	123	118	117		127
Onset of Tg by DSC (°F)	129	126	123	122		138
Ultimate Tg by DSC (°F)	142	139	137	130		154
Izod Impact, notched (ft-lbs/in)	0.93	0.54	1.27	1.33		1.28

* Actual Ratio Dispensed by Calibrated WEST SYSTEM® Minipumps.

BILL OF MATERIALS

Component	Sub-Component	Material Type	Dimensions	Count	Cost	Source	Weight
Main Beam	Balsa Core						
		Balsa Sheet	48"x8"x1/8" (not all used)	8	$50	specializedbalsa.com	< 3 lb
		Balsa Strip	1/2"x1/2"x48"	8	$8	specializedbalsa.com	< 2/3 lb
		Gorilla Glue		1- 8oz.	$14	Home Depot	
	Torsion Tube	6k Plain Weave Carbon Fabric	50" x 10 yard roll	< 6 yards	< $400	Soller Composites	8.9oz/sq. yd. 3.5-4lb
	Spar Caps	9 oz. 6" Uni Tape	40' x 6"	1- 50' roll	< $100	Soller Composites	9 oz./sq. yd. 1.25 lb
	Resin	West Systems Epoxy/Hardener		2 Gallon	$200	West Marine	Approx 50% Weight 4lb
Beam Totals					**$772***		**Approx. 13lb**
				*Reflects extra Material that will be used in other components			
Floor	Honeycomb Core	Nomex 3lb/ft^3 1/8" cell	1/2" x 4' x 8' (use 40"x23" and 30"x45" sections)	1 sheet	$315	avtcomposites.com	2/3 lb
	Carbon Venere	12k Spread weave 193gsm Prepreg	1.76 sq. yd. per layer 5 layer per side	17.6 sq. yd. (14.72 m^2)	Donation	Tencate	< 6.3 lb
Floor Stiffeners	Honeycomb Core	same as above	approx 8"x46" total	From above sheet	Included above		.1lb
	Carbon Venere	same as above	.284 sq. yd/layer 5 layer per side	2.84 sq. yd. (2.36 m^2)	Donation	Tencate	1lb
Floor Totals					**$315***		**8.1lb**

Component	Sub-Component	Material Type	Dimensions	Count	Cost	Source	Weight
Rear Compartment	Honeycomb Core	same as above	Rear: 40"x12" Front: 40"x7" Sides: 19"x22"	From above Sheet	Included above		1/3lb
	Carbon Fabric	6k Plain Weave Carbon Fabric	50" x 10 yard roll	750 in^2 per layer 4 layer per side quasi-isotropic 3.3 yards	<$400	Soller Composites	8.9oz/sq. yd. 2.6lb
	Resin	West Systems Epoxy/Hardener				West Marine	Approx 50% Weight 2.6 lb
Rear Comp. Totals					**$400***		**5.6 lb**
Roll Bar	Hoop Core	Last-a-Foam 4lb/ft^3	96"x24"x4" sheet	use 4"x1.5"x84.2"	$240	fiberglasssupply.com	1.2lb
	Rear Stay Core	same	From same sheet	2[1.5"x3"x20"]	Included Above	fiberglasssupply.com	.4lb
	Carbon Fabric	Use same 6k fabric cut into 4" + 5" tape	5"x84.2" and [4"x20" X 2]	6 layers/side 2.2 yards	Included Above	Soller Composites	8.9 oz./sq. yd. 1.75 lb
	Resin	West Systems Epoxy/Hardener			Included above		50% Weight 1.75 lb
Roll Bar Totals					**$240***		**5.1 lb**
Dash Bar	Hoop Core	Last-a-Foam 4lb/ft^3	Included from above sheet	2"x1.5"x68"	Included above	fiberglasssupply.com	.5lb
	Carbon Fabric	Use same 6k fabric cut into 3" tape	3"x68"	6 layers/side .75 yd	Included above	Soller Composites	.6 lb
	Resin	West Systems Epoxy/Hardener			Included above		50% Weight .6 lb
Dash Bar Totals					**$0***		**1.7 lb**
Total Cost					**<$1730**		**33.5 lb**

Lay-up Schedules

Part	Lay-up Schedule
Beam	$[\,(\,(0/90)\,/\,\pm 45\,/\,0\,)_s\,]_3$
Roll Bar	$[\,(\,0\,/\,\pm 45\,/\,(0/90)\,)_s\,]_2$
Back Wall	$[\,(\,(0/90)\,/\,\pm 45\,)_3\,/\,\text{honeycomb}\,]_s$
Floor	$[\,(\,(0/90)\,/\,\pm 45\,)_2\,/\,\text{honeycomb}\,]_s$

Appendix F - Testing

F.1 DVP&R

F.2 Main Beam Testing

ME428/ME481/471 DVP&R Format

Report 12/1/2009 | Sponsor: Supermileage | Component/Assembly | REPORTING ENGINEER: K. BRAICO

TEST PLAN / TEST REPORT

Item No	Specification or Clause Reference	Test Description	Acceptance Criteria	Test Responsibility	Test Stage	SAMPLES Quantity	SAMPLES Type	TIMING Start date	TIMING Finish date	TEST RESULTS Test Result	TEST RESULTS Quantity Pass	TEST RESULTS Quantity Fail	NOTES
1	Dimension Verification	Measure various locations of chassis to confirm the vehicle fits withing tolerances	Meets Shell Rule Book	AA/KC	CV	10	A	9/14/2009	11/15/2009	Pass			
2	Load Strength	Load the bare chassis to verify it will withstand required loads	Withstand loads up to 500lbs	KB	DV	3	B	9/14/2009	11/15/2009	Passed			Loaded with 700lbs
3	Deflection	Apply loads to bar chassis while measuring the vertical deflection along the chassis	>0.1"	KB	DV	6	B	9/14/2009	11/22/2009	Passed			Loaded up to 480lb for a 0.1" Deflection
4	Roll Bar	Apply loads to the top of the roll bar and check for visual deflection	No visual deflection	AA/KB	DV	1	B		11/26/2009	Passed			Loaded with 160lb and no visual deflection using a tape measurer

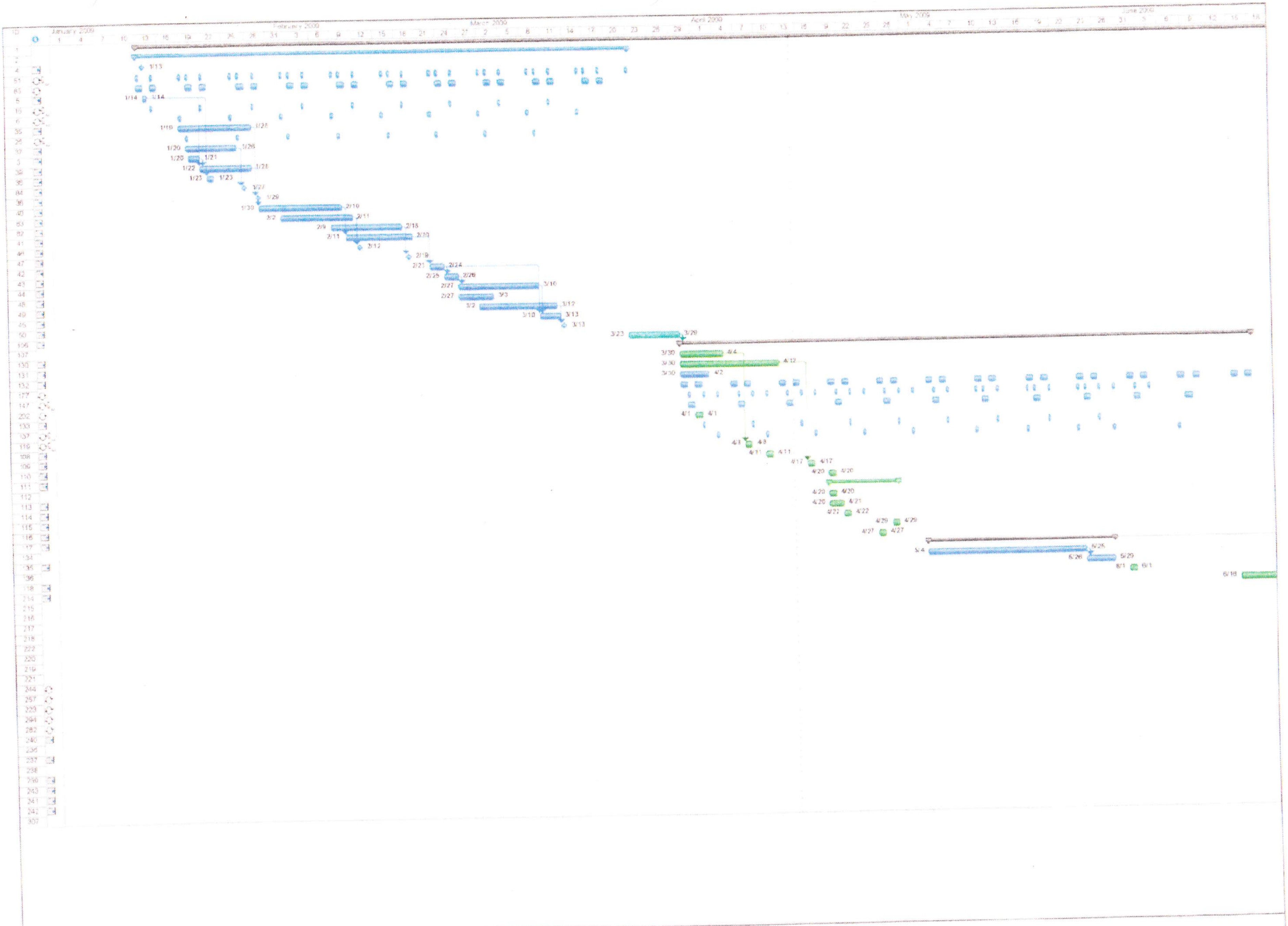
Vertical Deflection of Main Beam Testing

Weights were loaded in the driver position. A caliper was used to measure any deflection of the beam.

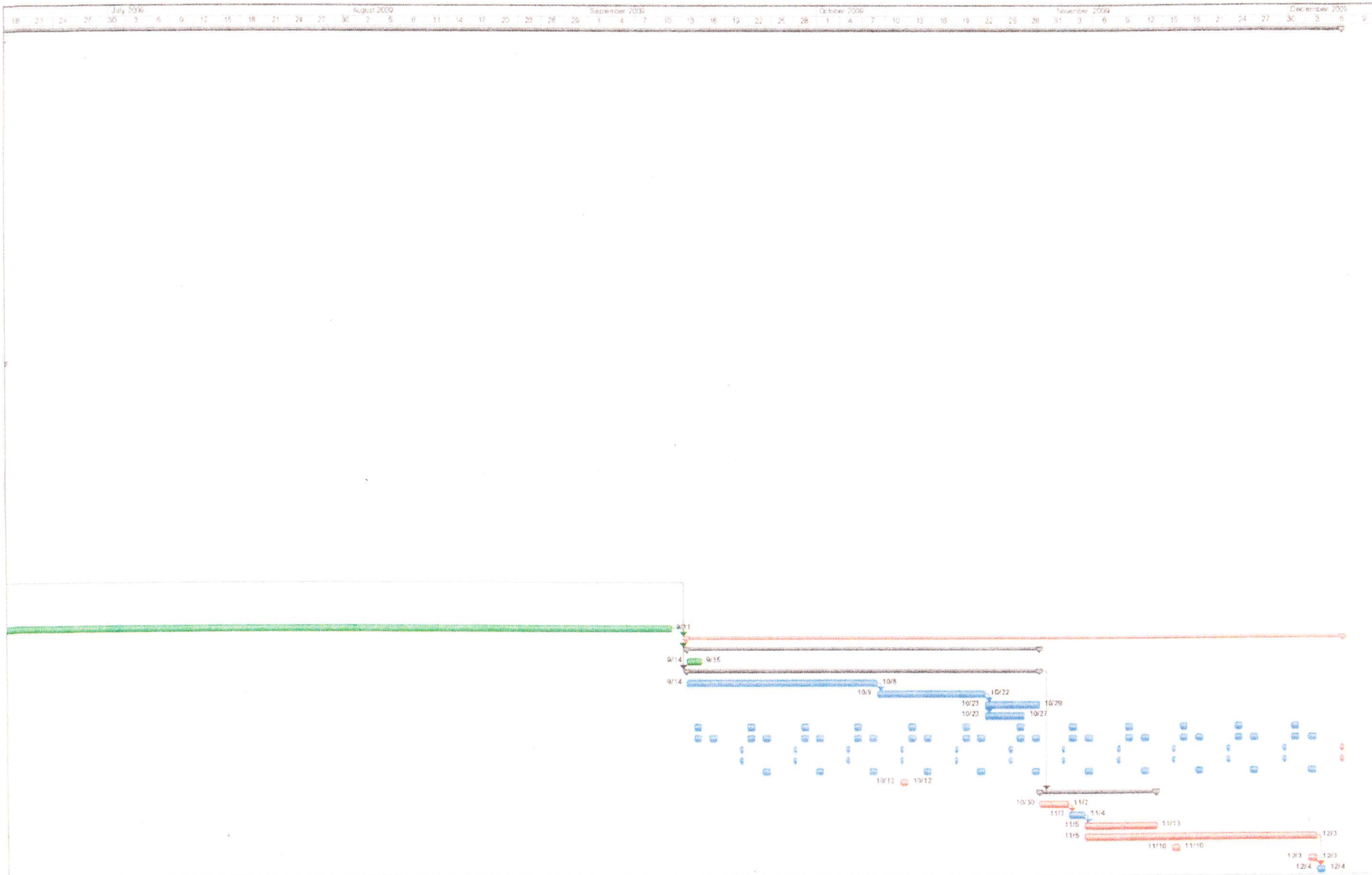
Appendix G – Gantt Chart

G.1 Gantt Chart

ID	Task Name	Man Hours	Duration	Start	Finish	Predecessors
1	Urban Concept Design	937.95 hrs	242.25 days	Tue 1/13/09	Mon 12/7/09	
2	WINTER QUARTER	358.97 hrs	49.13 days	Tue 1/13/09	Mon 3/23/09	
4	Project Introduction Letter to Sponsor	2 hrs	1 day	Tue 1/13/09	Tue 1/13/09	
51	Log Book Entry	30 hrs	49.13 days	Tue 1/13/09	Mon 3/23/09	
65	Lab Meetings	60 hrs	48 days	Tue 1/13/09	Thu 3/19/09	
5	Read over Shell Technical Rules	6 hrs	0.63 days	Wed 1/14/09	Wed 1/14/09	
16	Meeting with Professor Fabijanic	2.97 hrs	40.04 days	Thu 1/15/09	Thu 3/12/09	
6	Saturday Meetings with Urban Concept Team	27 hrs	40.38 days	Mon 1/19/09	Mon 3/16/09	
35	Work on Project Proposal	22 hrs	8 days	Mon 1/19/09	Wed 1/28/09	
26	Weekly Supermilage Meetings	8 hrs	35.13 days	Tue 1/20/09	Tue 3/10/09	
37	Project Schedule	9 hrs	5 days	Tue 1/20/09	Mon 1/26/09	
3	Background Research	24 hrs	0.35 wks	Tue 1/20/09	Wed 1/21/09	
39	Brainstorm Design Solutions	12 hrs	1 wk	Thu 1/22/09	Wed 1/28/09	3,5
36	Email Shell	2 hrs	1 day	Fri 1/23/09	Fri 1/23/09	5
54	Project Schedule Due	0 hrs	1 day	Tue 1/27/09	Tue 1/27/09	37
38	Project Proposal Due	0 hrs	1 day	Thu 1/29/09	Thu 1/29/09	35
40	Begin Solidworks Modeling	10 hrs	1.5 wks	Fri 1/30/09	Tue 2/10/09	39
83	Decision Matrix	8 hrs	8 days	Mon 2/2/09	Wed 2/11/09	
82	Interim Design Report	22 hrs	8 days	Mon 2/9/09	Wed 2/18/09	
41	Simple Hand Calcs on Top Few Chassis	16 hrs	1.5 wks	Wed 2/11/09	Fri 2/20/09	40
46	Decision Matrix Due	0 hrs	1 day	Thu 2/12/09	Thu 2/12/09	83
47	Interim Design Report Due	0 hrs	1 day	Thu 2/19/09	Thu 2/19/09	82
42	FEA on the Multiple Chassis Designs	10 hrs	2 days	Mon 2/23/09	Tue 2/24/09	41
43	Select a chassis	6 hrs	2 days	Wed 2/25/09	Thu 2/26/09	42
44	Detailed hand calculations on selected Chassis	21 hrs	1.5 wks	Fri 2/27/09	Tue 3/10/09	43
48	Stress Analysis on Chassis	15 hrs	3 days	Fri 2/27/09	Tue 3/3/09	
49	Draft Design Report	28 hrs	9 days	Mon 3/2/09	Thu 3/12/09	
45	Material Considerations for the Chassis	9 hrs	3 days	Tue 3/10/09	Fri 3/13/09	42,44
50	Draft Design Report Due	0 hrs	1 day	Fri 3/13/09	Fri 3/13/09	49
106	SPRING BREAK	0 hrs	5 days	Mon 3/23/09	Sun 3/29/09	
107	SPRING QUARTER	250.47 hrs	64 days	Mon 3/30/09	Wed 6/17/09	106
130	Prepare for Design Presentation	20 hrs	6 days	Mon 3/30/09	Sat 4/4/09	
131	Final Design Report	26 hrs	12 days	Mon 3/30/09	Sun 4/12/09	
132	Contact Possible Donors of Materials	4 hrs	4 days	Mon 3/30/09	Thu 4/2/09	
177	Lab Meetings	72 hrs	64 days	Mon 3/30/09	Wed 6/17/09	
147	Log Book Entry	29 hrs	53.13 days	Tue 3/31/09	Thu 6/4/09	
202	Supermileage Meeting	11 hrs	57 days	Tue 3/31/09	Tue 6/9/09	
133	Reserve Paint Booth For Lay ups	1 hr	1 day	Wed 4/1/09	Wed 4/1/09	
137	Team Meeting with Advisor	2.97 hrs	45.04 days	Thu 4/2/09	Thu 5/28/09	
119	Entire Urban Concept Team Meeting	30 hrs	51.25 days	Sat 4/4/09	Mon 6/8/09	
108	Design Presentation at Cal Poly	4 hrs	1 day	Wed 4/8/09	Wed 4/8/09	130
109	Deadline for Supermilage to submit dimensions	0 hrs	1 day	Sat 4/11/09	Sat 4/11/09	
110	Final Design Report Due	0 hrs	1 day	Fri 4/17/09	Fri 4/17/09	131
111	Critical Design Report with Sponsor	1 hr	1 day	Mon 4/20/09	Mon 4/20/09	
112	Chassis Materials	12.5 hrs	9 days	Mon 4/20/09	Wed 4/29/09	
113	Compile List of Materials to be ordered	3 hrs	1 day	Mon 4/20/09	Mon 4/20/09	
114	Compare supplier costs	6 hrs	2 days	Mon 4/20/09	Tue 4/21/09	
115	Submit approval for funding of orders	1.5 hrs	1 day	Wed 4/22/09	Wed 4/22/09	
116	Order Materials	2 hrs	1 day	Wed 4/29/09	Wed 4/29/09	
117	Student Presentations - Ethics	2 hrs	1 day	Mon 4/27/09	Mon 4/27/09	
134	Testing: Verify Materials Specs (Scaled Model Testing)	27 hrs	20 days	Mon 5/4/09	Fri 5/29/09	
135	Build Scaled Beam	24 hrs	16 days	Mon 5/4/09	Mon 5/25/09	
136	Test Beam and Compare to Calcs	3 hrs	4 days	Tue 5/26/09	Fri 5/29/09	135
118	Project Update Report to Sponsor	8 hrs	1 day	Mon 6/1/09	Mon 6/1/09	
214	SUMMER BREAK / MAKE-UP TIME	0 hrs	66 days	Tue 6/16/09	Fri 9/11/09	
215	FALL QUARTER	328.5 hrs	60.25 days	Mon 9/14/09	Mon 12/7/09	214
216	Build: Chassis	112 hrs	34 days	Mon 9/14/09	Thu 10/29/09	214
217	Set up Forms	16 hrs	2 days	Mon 9/14/09	Tue 9/15/09	
218	Lay ups	96 hrs	34 days	Mon 9/14/09	Thu 10/29/09	134
222	Backbone + Floor	40 hrs	19 days	Mon 9/14/09	Thu 10/8/09	
220	Structural Supports	30 hrs	10 days	Fri 10/9/09	Thu 10/22/09	222
219	Roll Bar	14 hrs	5 days	Fri 10/23/09	Thu 10/29/09	220
221	Misc	12 hrs	3 days	Fri 10/23/09	Tue 10/27/09	220
244	Meeting with Supermileage and Urban Concept	12 hrs	56 days	Tue 9/15/09	Tue 12/1/09	
257	Lab Meetings	72 hrs	58 days	Tue 9/15/09	Thu 12/3/09	
223	Meeting with Entire Urban Concept Team	24 hrs	55.25 days	Mon 9/21/09	Mon 12/7/09	
294	Saturday Meetings	24 hrs	55.25 days	Mon 9/21/09	Mon 12/7/09	
282	Meeting with Advisor	5.5 hrs	51 days	Thu 9/24/09	Thu 12/3/09	
240	Senior Exit Exam	4 hrs	1 day	Mon 10/12/09	Mon 10/12/09	
236	Test Bare Chassis	31 hrs	11 days	Fri 10/30/09	Fri 11/13/09	218
237	Load and Deflection Testing	9 hrs	2 days	Fri 10/30/09	Mon 11/2/09	
238	Torsion Testing	8 hrs	2 days	Tue 11/3/09	Wed 11/4/09	237
239	Possible Modifications to Weak Points	14 hrs	7 days	Thu 11/5/09	Fri 11/13/09	238
243	Final Project Report	30 hrs	21 days	Thu 11/5/09	Thu 12/3/09	
241	Complete Senior Survey	2 hrs	1 day	Mon 11/16/09	Mon 11/16/09	
242	Senior Design EXPO IV	12 hrs	1 day	Thu 12/3/09	Thu 12/3/09	
307	Final Report Due	0 hrs	1 day	Fri 12/4/09	Fri 12/4/09	243



Project: simple example
Date: Wed 4/15/09



Project: simple example
Date: Wed 4/15/09