University of Central Florida

2010 Human Powered Vehicle

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Abstract

Walking and running are the most basic forms of human powered transportation. As humans became more intelligent, they've found that a vehicle allows them to travel more efficiently, allowing humans to cover longer distances with less effort. The bicycle is the most common of these vehicles. Bicycles in varying forms have been around since the nineteenth century and since that time there have been never ending efforts to improve the design. To reduce the air drag of the vehicle the rider is placed in the supine position which is referred to as a recumbent bicycle. To further reduce the drag the addition of an aerodynamic enclosure, known as a fairing, can be added. This design, a recumbent bicycle with a fairing, is one of the more common forms of an HPV. Although machined powered vehicles such as cars, airplanes, boats, and trains are more commonly used, the demand for fossil fuels will eventually deplete the world’s supply, making human powered vehicles the main source of transportation.

The American Society of Mechanical Engineers sponsors the Human Powered Vehicle Challenge (HPVC) as a way to promote the use of human powered vehicles. The HPVC tests Mechanical Engineering students to design, manufacture, and man a HPV that is to be judged in categories such as design, safety, and performance. Ultimately, the HPV will be driven in both a time trial and endurance race against entries from universities across the United States and around the world. For senior engineering students at the University of Central Florida (UCF) attending the HPVC has become a regular annual event. UCF has participated in the HPVC since 2002. At the East Coast HPV Challenge, the UCF HPV team has been a strong competitor. In the design process the 2010 HPV team has built off of previous designs and manufactured a HPV that is a combination of many ideas and designs. Design and engineering innovations were utilized from concepts attained in the classroom. The frame of the 2010 UCF HPV employs some features from the 2007 UCF frame such as a three wheeled configuration and other features from the 2006 UCF frame design such as the steering mechanism and u-joint.

Some key differences of the of the 2010 design are the use of the micro cassette on the rear wheel, an adjustable seating mechanism, and an intricate steering design, among other innovations. The most important change in the 2010 frame is the reduced weight and overall stability achieved with a wider 3 wheel design. Due to the increased width of the frame, the team decided to design a fairing that would enclose the middle of the vehicle only, allowing for a much narrower fairing than in previous years. This feature greatly enhances the aerodynamics of the vehicle, ultimately reducing the drag of the vehicle and allowing it to reach speeds greater than a standalone HPV frame. In general, the 2010 UCF HPV team designed a vehicle that builds on the success of the past HPVs. It incorporates successful designs observed, while at the same time bringing a new and innovative design to the table. The report presented in the next several pages covers the design and manufacturing processes that the team has carried out for the production of the 2010 UCF HPV. The report is broken down into description, analysis, testing, safety, and practicality sections that cover frame and fairing considerations in order to provide you with a perspective on the approach taken towards the engineering of the 2010 UCF HPV.
Table of Contents

1. Introductory Material.........................................................5

2. Discussion of Developmental phase.................................6

3. Discussion of Parametric and Detailed Design.................7
   A. Frame Design.................................................7-11
   B. Fairing Design...........................................12-18
   C. Steering Design.........................................19-27

5. Concluding Discussion..................................................28

6. Project Schedule
   a. Gantt Chart...................................................29
   b. Individual assignments/Responsibilities................31

7. Resources........................................................................32
Design Description

The American Society of Mechanical Engineers sponsors the annual Human Powered Vehicle Competition with the intent of finding an ideal design to be used for everyday activities ranging from commuting to and from work or going to the store. Although some Human powered vehicles have topped 80 mph, the ASME HPVC is more focused on applying engineering and ingenuity to the design of the vehicle. The goals for the University of Central Florida HPV team are to design a vehicle that is lightweight, aerodynamic, safe, comfortable and enjoyable to ride in the 2010 East Coast Human Powered Vehicle Challenge. In order to completely describe our engineering design problem, a Quality Function Deployment was completed. Also, in order to understand the requirements of the project customers, a House of Quality was created. The requirements for our vehicle were as follows: braking within 20 ft at 15 mph, 100-ft straight steering, 25-ft turning radius, rollover protection, helmet and restraint, ergonomic, accessible components, strength to support rider, lightweight, and cost effective, in order to satisfy our $5000 budget.

It was determined that the design criteria or requirements in the design of our vehicle were for it to be safe, reliable, and stable. Goals or objectives for our design were then chosen taking into account our design requirements. The goals are a rider change-time less than 20 sec., 50 lb total vehicle weight, 40 inch and 45 inch maximum width and height, and a speed of 25mph for the men’s sprint race and 20mph for the women’s sprint race.

1.1 Frame Configuration

Two wheels vs. Three wheels

For the Human Powered Vehicle Challenge speed category, there are only two competitive designs used. There is a recumbent two wheel design, or a recumbent three wheel design, like a tricycle. In order to make the selection between a two and a three wheel design, a comparison of the two was conducted.

Figure 2: Two wheels vs. Three wheels

This selection was based more on thought and reasoning than any mathematical proof of concept. Between the two concepts our group decided to go with a three wheel design for one main reason, stability. The HPVC has different events in the competition as determined from the design requirements portion; these events include a top speed event, and an endurance event. For the
endurance event the course will not be a simple straight line, varying degrees of cornering are expected. On top of this cornering there will be vehicle traffic as well. It has been noted that some teams in the past, including UCF’s from last years competition, have had trouble on sharp, low speed corners with a two wheel design resulting in their vehicle losing stability and falling over (See Figure 2). Also, in all events, the vehicle must be launched un-assisted with respect to getting the vehicle moving. Two wheeled designs can have team mates holding the vehicle up right, but soon as the vehicle starts to move, it cannot be assisted any more. Therefore starting in a two wheel vehicle is more difficult based on stability, which is much less than a standard bicycle due to the fairing as well as unfamiliar recumbent riding position. Also, for the endurance event, rider changes during the race are required, so for a two wheel design, it must be held upright while the rider change takes place which makes the process more difficult, while in a three wheeled design the vehicle is self stable and allows for easier rider changes and standing starts. For these reasons, mainly all attained from research, we chose to go with a three wheel design.

Three Wheeled Configerations

The three wheeled design, can be broken further into two configerations. The first of these has one front wheel and two rear wheels. The second of these has two front wheels and one rear wheel. These configerations can be seen in figure 3 below.

![Figure 3: Three wheeled configurations](image)

The decision was based on the two designs’ turning capabilities. Both configerations are comparably stable at a stand still, in starts, and in a straight line, but the difference comes about when cornering. As shown in Figure 4, when one looks at the basic layout of each design and does a visual vector analysis of forces while cornering, one can see that two wheels in the front is more stable than two wheels in the rear. So combining the forward momentum and the centrifugal force as shown by the vectors in blue, they combine to give the vector in green, which, as one can see, points in the directions of the tires in the case of the three-wheel front design, and points outside of the front tire in the case of the three-wheel rear design. This basic proof of concept shows why two-wheels in the front are a more feasible design choice than two wheels in the rear. It was also noted that handlebars for the first configuration would be conflicting with a narrow fairing design. So for the frame configuration it was decided to do a three wheel design with two wheels in the front.
1.2 Frame materials

The alternative concepts generated for frame materials were aluminum 7005-T6, 4130 chromoly, and some form of generic cold-worked steel. Figure 5 shows a comparison of the modulus of elasticity and density of each choice.

As you can see, aluminum is significantly lighter than both chromoly and cold worked steel based on density. Its modulus of elasticity is lower but its tensile yield strength is comparable. Of the three materials, we chose the aluminum 7005. Another reason for us to choose the aluminum is that one of our sponsors, Catrike has offered us the material at a low cost and will allow us to a facility for bending our pipe and welding the frame together. This 7005 grade aluminum is a rare grade used specifically for cycling purposes, on their commercial tri-cycles. They offered assistance in bending our pipes and welding the frame together. They have a certified welder who TIG welds, and primarily welds aluminum. They also have a furnace to heat treat our frame to normalize the welds, since this is another issue with aluminum which loses a lot of its strength when welded. This eliminates most of the difficulty in selecting aluminum since we have a way to manufacture the frame. There are also major differences in materials between the aluminum and the other two choices due to their modulus of elasticity. Since aluminum is lower, we are plan on compensating by raising its moment of inertia. This is done by using larger diameter/wall thickness piping with respect to the other two materials, but since the density is so low on the aluminum it will still be lighter than the other materials. For these reasons we have selected Aluminum 7005 for be our frame material.
1.3 Length adjust

The last sub-function that was created was how to adjust the seating position from the rider. The alternative solutions for this are a boom adjustment, or a seat adjustment. The boom adjust was found be used commercially by our sponsor Catrike, however after visiting the factory and trying the adjustment out for ourselves, the boom adjust was sort of tedious, and was in the fore of the vehicle, which is planned to be faired in, so access to a boom adjust may be restricted. For this reason we have selected to go with some form of seat adjustment. We haven’t decided on the exact solution yet because different possibilities are still being researched and we haven’t come across a solution that we like yet. However, the seat adjustment will have to be, by our requirements, relatively quick and easy, and is a requirement because our tallest team member is in the upper six foot range and our smallest in the mid five foot range.

1.4 Wheel selection

The alternative solutions for our tire/wheel selection were all road style wheels, all mountain bike style wheels, or a combination of wheels. The main difference between road style and mountain bike style is tire/wheel diameter and width. The reason we might be interested in mountain bike tires is because they might provide more traction because they have more tread, but they would also provide more rolling resistance. Road style tires are narrow and larger, so we could attain a higher top speed and have less rolling resistance. The way we went about selecting our tire sizes were based more on research and need. We decided to use a 700c road style rear tire/wheel while using two 20inch tires up front. The reason we chose a 700c road rear tire was because it was be advantageous to our gearing selection. We went with a gearing combination that would be benefited by using the diameter size of a 700c tire. Also, since the tire is narrow it offers less rolling resistance than a mountain bike tire, so it was an obvious choice for the rear. However, the front tire and wheel selection was a bit more involved. The issue with the front wheels in the mounting of them. Most wheel hubs use 5mm skewers to hold them onto dropouts, like the rear tire will. But these hubs are mounted in double shear. The way we are going to mount the front wheels is cantilevered like a common car wheel is, that is, it is mounted on the side and the axle is in single shear with the uprights. For this reason a 5mm skewer may not be strong enough for the expected loads and would interfere with the braking system. Figure 6 shows a proof of concept for the hub selection.
An alternative to a 5mm skewer is a 20mm thru axle. These 20mm thru axle hubs are common on downhill racing and free-ride mountain bikes. So as seen in Figure 6, with a material of generic steel and a load of around 70 lbs the area moment of inertia was calculated for a 5mm solid axle and a 20mm thru axle with .049 wall thickness. Then these values were plugged into the bending formula for a cantilever beam with uniform weight distribution. From the results one can see that the 5mm skewer deflects to much, this shows that it isn’t very strong for static loads let alone any dynamic loads. It also would affect the braking since we plan on using disc brakes this deflection would cause the rotor to hit the pad. In comparison, the 20mm thru axle only deflects by .02mm and therefore is our selection for hubs. Figure 7 shows a example of the type of hub that will be used for the front wheels.

This hub will have a 20mm thru axle, 32 spoke count and 6-bolt disc compatibility. Once the hub was selected then the tire diameter had to be chosen. The options were that of a 700c road style rim of a 20 inch rim that was found for a 1 1/8 tire. Between the two the 20 inch rim was selected. This was chosen on the premises that the rolling resistance from the tread would be the same as a road tire since the widths of compatible treads would be the same. The advantage of the 20 inch rim is the fact that the spokes can handle the lateral loads better, but having stronger wheels in the front. So our final selection for the tires and wheels are a 700c road tire/wheel with 5mm skewer for the rear, and two 20 inch wheels with 20mm thru axle hubs on the front.
1.5 Braking system

The alternative solutions for the braking systems included all disc brakes; all rim brakes, or a mix of disc and rims brakes. Upon the initial assessment was determined for sure that we wanted disc brakes on the front tires, but the option of having a rim brake on the rear was still uncertain, so this comparison of options was later put into the KT DA. However, the reason for selecting disc brakes for the fronts will be discussed now. Since the front wheels were going to be mounted cantilevered, the logical option for braking was to use disc brakes. This is so because if a rim brake was to be used it would have to be mounted on either side of the tire, and also would give as reliable a braking force as disc brakes would. For this reason, and the fact that the disc brakes would provide greater brake pressures, we selected disc brakes for the fronts. The only option left then would be to use either hydraulic or cable actuated disc brakes. Between the two cable actuated was selected. This was done because the installation and adjustment of hydraulic brakes was researched to be more tedious and less adjustable than cable actuated brakes. After further research we decided to select a SRAM BB-5 brake calipers with Avid disc. This product was chosen because the BB-5 caliper is cable actuated, easy to install, and is ball bearing actuated so it was minute pad adjustment so we can further tune our brakes. The compatible rotors for this braking system come in 160mm and 210mm sizes so we can select the correct rotor size when we come to an estimation of what the overall weight of the vehicle will be. Figure 8 shows pictures of these products.

![Figure 8](image)

1.6 Steering system

The alternative concepts of steering systems include wheel, under-seat steering, rack and pinion, or handlebars. The options that weren’t chosen are the under-seat steering and rack and pinion. The reason that under-seat steering wasn’t chosen was because the width of the vehicle may be to wide since applications of the under-seat steering are on commercially available, generally narrower, tri-cycles. The reason the rack and pinion wasn’t selected was because, after research, the smallest rack and pinion units founds were used on dune buggy and formula karts, and weight about 2lbs. From this we concluded that the rack and pinion unit would be an over design and weight to much. From this we decided to choose a v-type linkage with steering wheel. This incorporates a center bracket with two tie rods that would connect to the spindles using rod ends. This type of steering is easier to design and set handling characteristics such as the Ackermann for. A figure showing a sketch of what the steering system might look like can be seen in Figure 9.
1.7 Drivetrain components

The alternative concepts researched for the drivetrain components include road components, mountain bike components, 2 chainring front 9-speed back, or single-speed front 2 10-speed back. Between road components and mountain bike components we have already decided on road components. This was done because in general, road components have higher gearing ratios and are meant to work with a 700c wheel, which we previously selected for our rear wheel. Now in the road components category, our two main options were a 2 speed front and 9-speed back or a single speed front and two 10-speed back. The reason that a single-speed front and two rear cassettes was thought of as a concept was because, if we used standard gearing, we would gain a mechanical advantage by using the second cassette. However, the added cassette would make the system more complicated, shifting more tedious, and they system more susceptible to failure. Also, after component research, we found a 2 chainring front, and a 9-speed rear that would produce comparable top speeds, but give a better medium range of gears and significantly easier shifting, installation, and use. For the rear cassette we selected a Shimano Capreo 9-26 tooth. This cassette has significantly lower gearing compared to standard cassettes, which commonly have 12-23 teeth, which will allow us to obtain higher top speeds. For the front crank we selected to use a SRAM rival 53-39 crankset. This crankset has relatively higher gearing with a possibility to change the upper chainring to a 55 tooth if it is later deemed necessary. Figure 10 shows the selected component and a theoretical top speed calculation.
As one can see a theoretical speed, based purely on gearing ratios and possible rider cadence, of 47.3 mph might be obtainable as long as a cadence of 100rpm can be reached in top gear. This speed could be increased in the cadence can be increased as determined by the rider or if the upper chainring is switched to the 55 tooth. This system is also beneficial since it is compatible with 9-speed flat-bar gear shifters which will make shifting and rider use easier.

1.8 Fairing Design

The fairing is an aerodynamic shell built around the frame. It is used to improve air flow and also protects the rider. There are a several different design possibilities. To begin we wanted to design the fairing to be as safe and aerodynamic as possible. The main constraints limiting our design were those set by the rules of the ASME HPV competition and the frame configuration. Since the fairing is being built around the frame, the final dimensions of the fairing depend on those of the frame. One of the difficulties in designing the vehicle has been that we are dealing with a large range of heights for our riders. Through many calculations a frame with a rollbar for protection was designed to accommodate the riders. These dimensions set up limitations. The fairing must be designed to fit around the rollbar and give enough room for the rider to pedal without hitting the sides. After taking the dimensions of the frame into account, an aerodynamic shape can be designed for the fairing based off the minimum dimensions from the frame. A lot of research was done on UCF’s past designs as well as other competitions. An ideal shape would be like the one shown in Figure 30. It would begin with a round nose and end with a tapered edge. This helps to maintain laminar flow over the entire shape. When the air flow does not hug the surface of the fairing this causes turbulent flow and increases drag.
However we must account for the increased width at the rollbar which changes the geometry. Below is an example of a streamlined vehicle designed to go over 60 mph. By comparing several aerodynamic fairings like the one shown in Figure 31, we were able to adjust the outlines of the fairing to have a low surface area while maintaining the required dimensions and an aerodynamic shape.

The final CAD configuration of the fairing is shown below in Figure 32. The final dimensions include a height of 39.5 inches and a width of 25.4 inches. It is good to note that the same drawing has been used repeatedly through the manufacturing and decision making process. For one it was used in manufacturing to determine the correct sized cross-sections that could be cut out of the foam to make the mold. Also this drawing was generally referred to when determining the positions of components such as the windscreen, naca-ducts, and cutouts.
The two main areas of concern had been the rollbar and the pedals. Sketches were created around these areas to ensure they were fully enclosed in the fairing. The desired values for the width are represented below in a satisfaction diagram (Figure 34). Through the rules of the competition we earn points in design for remaining under certain dimensions. We would have been 100% satisfied if the width of the vehicle stayed under .75 meters. This would have given us full points for that category. And while the fairing is .65 m, the wheel base of the frame is over .75 m. This was due to our decision to use three wheels as opposed to two. We decided it was an important safety factor and chose stability over points. It is still less than 1.25 m (0% satisfaction) which would mean no points in the design portion, and also a larger area for air to flow over.
Another constraint is weight. We need the vehicle to be as light as possible. We would be 100% satisfied with a vehicle under 40 lbs. The heavier the vehicle means more force will be required of the rider to move the vehicle. We want to encourage optimal performance from the rider; therefore we must improve the design so the rider does not become exhausted too quickly. (Figure 35)

![Weight Satisfaction Graph](image)

**Figure 35**

**Fairing Components**

There are also several features and components that must be built into the fairing after it is manufactured. For one there must be a way for the rider to get in and out of the fairing quickly and safely. Many competitions and vehicles are designed to race with one rider. We will be participating in an endurance event where we will be switching riders throughout the race. Therefore the way in and out of the vehicle must be simple and quick to accomplish. Some schools have made the screen the removable part however when watching videos of past races it seems that the screen alone would be too flimsy and more difficult to place back on the fairing. Other teams have cut the fairing almost in half, allowing the top portion to be removed and giving more room for the rider to settle into the vehicle. We will be employing this method however with some added features to make placement of the top easier and quicker. One method is to add v-shaped pieces on the bottom half to help guide the top half into place. A hinge of sorts will also be employed to keep the hatch on while the vehicle is in motion, however it should be easily removable in the case that the race day is too hot to completely enclose the rider so as to avoid a rider overheating.
Another component that needs to be addressed would be the screen. It is required that the rider can see more than 180° around them. This requirement is preferred anyways as a safety precaution. We do not want any unnecessary crashes due to a blind side. Several screens will be employed such as in the vehicle seen in figure XXXXX. This will allow the hatch to retain some stiffness while giving the rider 180° viewing. It also simplifies the manufacturing process. If one large screen was made, a separate mold would need to be made that would withstand large forces to heat treat plastic over the complex shape of the fairing. Instead the plastic screens are being arranged so that they can be shaped by hand. The other components that will be implemented in the fairing are naca-ducts. An example of one is displayed in the drawing below in Figure 37. The naca-ducts are strategically placed around the fairing to provide air flow to the rider. This would be a safety precaution as well as a comfort feature. Enclosing the rider alone is going to trap body heat. When the rider is pedaling the heat in the fairing environment is going increase rapidly. The hotter the rider gets the more exhausted they become and the power output from the pedaling decreases. By placing naca-ducts around the fairing we plan to have sufficient airflow to the rider to cool them down.

The configuration of these components was an issue of great discussion. The difficulty was making the correct cuts and as few as possible to allow the frame to be inserted into the fairing as well as for the wheel components to come out of the fairing and a hatch to be created. Also the correct locations of visors and naca-ducts must be determined. Figure XXXXXX below is a very crude picture of the planned layup for these components. The areas of white indicate windscreens. All other markings indicate the other cuts that must be made. Currently there are five naca-ducts planned for the fairing, three to allow air to flow in and two allowing air to flow out the back. These are the last components to be placed based on the areas that have already been taken by windscreens and detachments from the fairing.
2.3 Materials Selection

We chose to use a variety of materials to achieve a preferred combination of light weight and strength in the fairing. The main material of the fairing will be a style 7781 8.9 oz fiberglass E-cloth. By comparing this style with many others, it proved to have the better weight-to-strength ratio. It has a thickness of only .009”. The thicker a material is the more resin is retains therefore the heavier it becomes. It was also chosen for its ease of use. Since our shape has compound curves it was imperative to choose a material that would form to the shape of our mold therefore reducing the chance of leaving air pockets and points of weakness. Coremat® will then be utilized to add strength in areas where the frame will be connected to the fairing and also to add irregularities in the body itself. By laying fiberglass over an irregular surface it adds stiffness to the body of the fairing. Originally carbon fiber was being considered for strengthening; however carbon fiber is about 10X more expensive.

The next main material used in the manufacturing of the fairing is resin. Two different resins were used. For the main body a epoxy resin with 3 to 1 hardener was utilized. This allowed a reasonable curing time when working over such a large area. Also since foam is still part of this process it was important that the resin used did not eat away any of the foam. Once the main body is cured, it is removed from the mold and there is no more foam. All other additions to the body are also laid in small strips of fabric and therefore do not require a lot of time to lay everything out. For this a 435 standard polyester layup resin is used. While using the epoxy the fiberglass would take a few days to completely harden, the polyester takes only hours. Therefore, when adding the last few pieces of fabric, the time spent between applications is largely reduced. The downfall to using the polyester is that it would eat away at the foam therefore it cannot be used any earlier on in the manufacturing process.
Fairing Manufacturing Process

The manufacturing for the fairing will require several steps. First a two molds (right-side and left-side) are made using Styrofoam. Cross sections of the foam are cut out and adhesive is used to build the pieces on top of each other. The outsides are then sanded down and a compound is applied to the outside to create a smooth surface. A couple layers of fiberglass are added to the mold and sanded down to produce an even smoother solid surface.

From the mold, a combination of Partall Paste Wax and PVA is applied as a mold release agent. Then four layers of fiberglass and resin are laid out across surface and left to cure. This provides the body of the fairing which is sanded down and later painted. When the body is removed from the mold the Coremat® is applied in a leaf spring pattern for strength. The two halves are then combined using fiberglass and sections are cut out for adding the screens and naca-ducts to finalize the fairing construction.

Analysis

2.1 Frame Design
The primary design variables of the frame are the strength, dimensions and weight. The strength of the frame is completely reliant on the size and material of the tubes used. In the Conceptual Design stage of the project Aluminum 6061-T6 was selected to use for the frame design due to its light weight and excellent strength. Since that time the choice was reevaluated and Al 7005-T6 was chosen for its slight increase in strength over Al6061 and its better heat treating properties. In determining the tube sizes, there were two driving requirements issued by the ASME competition rules. First, the frame must be able to withstand a load of 600lbs applied to the top of our roll bar at an angle of 12 degrees toward the rear of the vehicle and have a deflection of no greater than 2 inches. Second, there must be a side roll bar able to withstand a horizontal load of 300lbs with a deflection of no greater than 1.5 inches. Having selected the material, the necessary diameters and wall thicknesses were calculated using beam deflection formulas.

\[
P := 600 \\
E := 1040000 \\
d_{\text{outer}} := 2 \\
d_{\text{inner}} := 1.834 \\
I := \pi \left( \frac{d_{\text{outer}}^4 - d_{\text{inner}}^4}{64} \right) \\
\text{MaxDeflection} := \frac{P L^3}{48 E I} \\
\text{Deflection} := \frac{P a^2 b^2}{3 E I L} \\
\]

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<th>Outer Tube Diameter</th>
<th>Wall Thickness</th>
<th>Deflection</th>
</tr>
</thead>
<tbody>
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<td>0.083</td>
<td>0.504</td>
</tr>
<tr>
<td>1.75</td>
<td>0.083</td>
<td>0.766</td>
</tr>
<tr>
<td>1.5</td>
<td>0.083</td>
<td>1.247</td>
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</table>

Using a safety factor of 2 and selecting from three possible sizes available, both the 1.75 and 2 inch tubes would be satisfactory, however, for additional rigidity the 2 inch tube was selected for the main frame. Different tube sizes were selected for other sections of the frame. For the front cross members, the 1.75 inch tube was selected while 1.25 inch tube was selected for the rear chain stays. The roll bar sections will use 1 inch tube with a wall thickness of .065in. Since the track width and the wheelbase are determined, the ground clearance of the frame was considered next. For performance, it is necessary to keep the center of mass as low to the ground as possible while still accounting for the fairing thickness and allowing for clearance of obstacles. The lowest point of the frame was chosen to be 3 inches. With a maximum fairing thickness of .5 inches, this leaves 2.5 inches of ground clearance. The rider geometry is the next variable that was considered. The seating position of the rider was determined through research of the efficiency of different seating angles and rider comfort. The seating angle was chosen to be 45 degrees. The seat that was chosen was the one used in previous team designs. Based on research and from the dimensions and position of the seat, an ideal elevation of the pedaling position was determined. This placed the bottom bracket of the crank assembly at 16 inches above the ground. The leg length of the tallest rider was used to find the necessary length of the boom which was found to be 36 inches.
In designing the chain stays, their length was determined by the radius of the wheel. Based on the desired speed of the vehicle, a 28 inch wheel was chosen. This required a minimum clearance of 14 inches for the tire. The chain stays will be placed 5.14 inches apart to accommodate the length of the axle. A prefabricated dropout was selected which accounts for 2.3 inches of the necessary length of the chain stay. As a result, the length of tube section of the chain stays was determined to be 15.77 inches which includes a 70.5° bend of a 4 inch radius which will be welded to the main frame 14 inches above the ground (Figure 13).

The next consideration was the rollbar. In order to provide protection for the driver, the rollbar must extend at least 2 inches above the helmet of the rider. From the measurements of the tallest rider, and allowing 2 inches for the thickness of the helmet material, the height of the roll bar was placed at 41.5 inches. From the base of the chain stays, the roll bar was angled outward by 22.76 degrees to reduce the length necessary for the side roll bar sections, thus providing additional support. With only one available bending radius for the one inch tubing, three bends were used to complete the section. For added rigidity a cross member was added to the rear roll bar (Figure 14).
The side roll bars are required to give 2 inches of clearance to the rider. Given a rider shoulder width of 20 inches the roll bar width was set at 24 inches. The trajectory of the side roll bar extends from the rear roll bar, outward 12 inches from the centerline of the frame, then angles inward and joins the vertical section of the main frame just below the chain stays. Again, one inch tubing is used for the side bar and uses 4 inch radius bends (Figure 15).

The next section of the frame considered is the front cross member. The variables for the design of the cross member are track width, wheel size and the center-point steering angle. The desired track width is 40 inches, the front wheels are 20 inches in diameter and the center-point steering angle is 21.5 degrees. Using the dimensions of the kingpin and these variables the cross members were angled upward by 18.6 degrees and extended outward 17 inches from the centerline of the frame. A 4 inch long tube for the kingpin mount was added to the end of each of the cross members at an outward angle of 21.5 degrees and a rearward angle (caster) of 13 degrees (Figure 16).
The seat position was the next variable considered. For the vehicle to accommodate riders of different sizes the seat will be made adjustable. However, with the limiting visibility of the fairing and the raised pedal position, the shorter riders would need to be elevated as well as moved forward. For this, an angled mounting plate was designed to raise the rider by 3 inches when moved forward by 8 inches. The seat mount consists of a slotted base bracket and a flat plate mounted to the base of the seat which will slide along the slotted base (Figure 17).
2.4 Steering Design

Wheelbase and track-width design

The initial design variables for the HPV project was first to determine the wheelbase and track-width of the vehicle. Most other sub-system designs are directly dependent on these values. A schematic of what the wheelbase and track-width is defined as can be seen in Figure 20. As one can see, the wheelbase is defined as the distance from the rear axle to the front axles of the vehicle.

The track-width is defined as the centerline distance from one front tire to the other. A couple of techniques were used to determine our final values. First, a generic wheelbase was determined from rider geometry and based on researched conducted in the first two design sections of this project. From this aspect we started at a wheelbase of about 45 inches. Once we had this the other technique we used was that of making and utilizing a stability diagram of our vehicle. This technique was recommended by our technical advisor Paulo at Catrike. The idea behind this technique is to make a CAD drawing of the axle locations of the three wheels, and then connecting those three points with a line. This forms a stability triangle. From this one calculated the desired centrifugal force generated by our heaviest expected rider, and the normal force of that rider at a determined center of mass. Then, the two vectors are added to get a resultant vector. This resultant vector is then placed at the COM of the rider and if the vector falls within the boundaries of the triangle, then the vehicle will be stable up to the desired cornering forces. To begin this process we first needed the weight of our heaviest expected rider, who happens to be our team member Mike. He weighs about 200 lbs, from this we
calculated our desired centrifugal force of the rider. Our desired centrifugal force was 1g. This was determined because our researched showed that the cornering forces generated by the rubber compounds that we plan on purchasing don’t exceed this value. From which, our resultant vector was simply at a 45 degree angle from vertical. The COM of the rider was determined base on a couple of factors. First was the geometry. The size of the rear tire and frame work surrounding the tire dictated the location of our seat. From this, our technical advisor said that we can estimate the COM of the rider to be around the belly button in an average recumbent seating position. From these two factors we calculated that the COM of our largest rider would be about 34 inches from the rear axle and 16 inches from ground level. These values along with the radius of our tires, and the initial wheelbase values were used to model a stability diagram, of which can be seen in Figure 21. This model was made in Pro-e, the top picture in Figure 19 shows the relative height of the radius of the rear tire, 14 inches, to the height of the radius of the front tires, 10 inches. Also, the vertical line in-between the front and rear tires is the side view of the generated vector. What this shows is how we tuned the wheelbase of our vehicle. What we did was adjust the wheelbase until the weight distribution from the rider was at about 67% front, 33% rear. The reason we did this is because there is only one tire in the rear, and two tires in the front, of which the contact patch of the tires are all equal.

In turn, this weight distribution will be split evenly between the three tires. Once this was determined the adjustment of the track width was made. As can be seen in the bottom picture, the distance between the two front tires was adjusted until the resultant vector safely fell inside the stability triangle. As can be seen, the track-width could have been made smaller, but wasn’t due to the fact that this only takes in consideration the effects of the rider, we felt that some room would provide us a safety factor for when we added other masses that may have a COM above that of the riders such as
the fairings. Using this method we came to a final wheelbase value of 46.6 inches, and a track width of 40 inches. This track-width met our previous requirement of being less than 1.25 m as determined from the initial design report, based on the competitions rules for maximum width of vehicles. Once the wheelbase and track-width was determined, sub-systems of the overall design such as the frame and steering design could then take place.

Steering design

The needed outputs for the steering design included the caster, camber, center-point steering angle, control arm lengths, and Ackermann. Figure 22 tries to schematically describe what each of these design components are. The caster is the angle difference between the steering axis of rotation of the tire to vertical.

![Figure 22](image)

The camber, if looking at the wheel assembly from the front, is the angle the tire makes with vertical. Center-point steering refers to the angle that the kingpin, the unit that connects the tire to the frame, makes with the tire itself. So if the center-point steering angle is zero, the kingpin is vertical/parallel to the tire. Control arm lengths refer to the distance from the axle of the tire to the point where the tie rod connects to the control arm. This distance affects the rate and angle at which the steering system can rotate. Lastly, Ackermann is an offset of the control arm from zero. What the Ackermann does is create a phase difference between the left and right steering tires by offsetting the control arms so that the inside and outside tires can turn at different rates so that a turn can be made properly, another term used to describe the Ackermann is the scrub radius, because if the Ackermann angle is not tuned correctly one tire will be tracking correctly and the other will be scrubbing while it turns, thus wasted energy and slowing the vehicle down, which we would like to avoid.

The first steering design component determined was the caster. For this we made a satisfaction diagram of the acceptable ranges for caster that can be used, as seen in Figure 23. From research, we found that most tricycle applications use between 10-14 degrees of caster, thus for the diagram that would give us a satisfaction of 1.
Less caster can be used but is less desirable because the camber gained by using caster is less, and thus the efficiency of the front tires in turning is lost. If too much caster is used, the tires will lean to much while cornering and possibly create stability issues. Our advisor at Catrike generally uses around 12 degrees of caster, but for our application we decided on 13 degrees. This was determined by our track width, since the width of our vehicle is greater, as well as our desired cornering speeds, the increase in camber of the tires from the caster is necessary, and since our vehicle is wider than common commercial applications, there won’t be any stability issues with the caster.

Next we had to determine the center-point steering angle, the satisfaction diagram of which can be seen in Figure 23 as well, but on the ride hand side of the figure. Since the center-point steering angle is determined by the kingpin, we were limited in our selection because of the fact that we plan to purchase commercially available kingpins that suit our applications. From this one can see that between 21.5 and 23.5 degrees we would be satisfied in our selection. However and angle exceeding 23.5 is not desirable because it can create brake-steer and bump-steer. What the center-point steering does is attempt to direct forces impacting the tire directly along the axis of the kingpin. From this no net torque is created that would translate back as torque steer on the wheel. As can be seen in Figure 22 for the center-point steering figure, if a bump on the ground is hit, or the brakes applied, and the center-point angle is such that the axis of the kingpin directly hits the center of the contact patch of the tire, those external forces will be absorbed by the frame, and not translated back to the driver, making the vehicle more drivable and stable. However, as can be seen in the satisfaction diagram, if the angle is a little less that can sometimes be acceptable because the little forces that could be translated back in those instances can be felt as driver feedback, or give the driver more ‘feel’ of the road. However, since we are going to purchase our kingpins, we selected 21.5 degrees. This angle would match the kingpin with our 20” diameter from tires, making the axis of the kingpin directly hit the center of the contact patch of the tire.

For static camber, we decided that we would use 0 degrees. This means that when the wheels are pointing straight, there is no camber in the wheel, and it is vertical. The reason this was done was because our frame is rigid, with no suspension, there is no need for static camber. Camber is generally designed for suspension application because the rate of camber changes when the suspension moves. Instead, the only camber we will gain is from the caster when the wheels are turned. Since the axis of steering rotation has been made 13 degrees back from vertical, when the wheels are turned they will gain camber, as seen from the front, when the vehicle turns.

The next steering design component to be determined are the control arm lengths. To start this process, the maximum angle made by the wheels had to be first determined. Since it is a competition requirement that the vehicle has to be able to complete a 25 foot turning radius is was essentially to make sure that it could. Figure 24 shows the Mathcad spreadsheet used to calculate the control arm lengths. So, a minimum design turning radius of 20 feet was first selected. Then the inboard and
outboard angles were calculated. From this a max angle of 25 degrees was calculated for the vehicle to complete a 20 foot turning radius. 26 degrees was then used, to allow room for error, to calculate the clearance needed for the tires to complete that turn. From this it was determined that the tires would need at a minimum of 5 inches clearance from the centerline of the wheel to make the turn, this is important for work on the fairing.

Figure 24
Determining Wheel angle for desired turning radius

\[ w := 46.6 \quad r := 120 \]
\[ t := 40 \]

\[
inboard := \arctan \left( \frac{w}{r - \frac{t}{2}} \right) \quad \text{outboard} := \arctan \left( \frac{w}{r + \frac{t}{2}} \right)
\]

\[ \text{inboard} := 24.986 \quad \text{outboard} := 18.41 \]

Max angle is 24.98, so will use 26 deg.
To find wheel clearance of 20 inch wheel
\[ 10 \cdot \sin(26 \text{deg}) = 4.384 \]
So will use 5 inches as wheel clearance to take in consideration width of wheel
Now will determine length of center v-link and kingpin links to allow desired movement of steering wheel and wheels

Since using tie rods with an OD of .750 we need clearance of 2 inches from center of steerer tube, and 20 degree from centerline offset for tie rod clearance(see design drawing of v-link)

With 2 inches length, and 20 degree offset x translational distance of v-link is:
\[ 2 \sin(70 \text{deg}) = 1.879 \]
So needed length of kingpin connection to obtain desired wheel turning radius of 26 degrees is:
\[ \frac{1.879}{\sin(26 \text{deg})} = 4.286 \]

This will allow for steering wheel max rotation of 70 degrees and max wheel rotation of 26 degrees to allow desired turning radius and comfortable steering range input.

Then the center control arm needed to be designed. Figure 23 shows the detailed drawing of the center v-link/control arm. This design was dictated by the size of the steering tube, and the size of the rod ends to be used. The steering tube was selected to be 1” OD, and the rod ends .25”, with a .75” shell OD. From this a 2 inch control arm length with a 20 degree offset was determined based on the clearances needed for the 1” steering tube and .75” shell OD of the rod ends. The angles and clearances needed for the rod ends while turning were also taken into consideration. Once this was made the x-translation of the center control arm was calculated as shown in Figure 22, from which the wheel control arm lengths was determined from the x-translation of the center control arm, and the designed 26 degree angle needed. From this a wheel control arm length of 4.286” was calculated. The detailed drawing of the system can be seen at the bottom of Figure 6. These angles are going to give a turning angle of 140 degrees lock to lock for the steering wheel, and maximum of 26 degrees to the wheels to be able to successfully make our designed radius, while also giving a comfortable steering input range.
A modeled version of the steering system can be seen in Figure 27 respectively. Once the control arms were done work could then begin on the Ackermann. The Ackermann is the offset of the wheel control arms which allow the wheels to turn at different rates to compensate for the fact that the wheels are turning at different respective radii. Figure 9 shows a crude drawing that tries to show how the Ackermann works. As can be seen in the figure without Ackermann, the tires turn and the same angles, which means one or both of the tires are going to scrub through the corner.
The second figure (Figure 28) shows control arm with an Ackermann offset, which shows that when turned the tires turn at different rates compensated for the angle difference. To help calculate the Ackermann a excel spreadsheet was recommended to use by our Catrike advisor. This spreadsheet was written by Peter Eland and is available online. What this spreadsheet does is takes all the inputs that we’ve just calculated including the wheelbase, track-width, tire geometry, control arm lengths, steering wheel offsets, and other dimensions and then calculates the ‘real’ tire angles achieved while turning through different radii and compares them to ‘exact’ or ‘ideal’ tire angles. Then a % error per radius level is given. This spreadsheet can be seen in Figure 29. First, as shown on the top of Figure 26, is the ideal Ackermann determined for our application. The highlighted box gives us an Ackermann of 117 degrees, which is 27 degrees from ‘straight’. 0 degrees for this spreadsheet is pointing directly to the center of the frame. The picture on the top left of the spreadsheet shows a schematic of the steering control arms and tie rods, while the picture on the right shows the steering system on the frame, for both figure the ‘front’ of the vehicle is pointing down. To obtain this ideal Ackermann, the angle was adjusted until, looking at the cells on the right under ‘this geometry’, the errors were reduced as low as they can for the radii shown next to the ideal Ackermann for those radii. However, after calculated the distance from the control arms ‘straight’ position, we found that this ideal value would have clearance issues with the wheel itself. So a compromise angle was found to be 100 degrees, 10 degrees from straight, which would translate to a .75 inch offset from straight, and would not have any clearance issues, the % errors are of course greater, but this is the best solution provided for the clearances we have. Once this was complete, all the components on the steering design were found, and the steering design complete.
Figure 29

Ideal

2.2 Fairing Design
A major influence in the design concepts for the fairing is the drag produced by the shape. Based on research a tear drop is a desired shape as it allows the air current to hug the surface of the fairing and reduce turbulent flow. However the width of the fairing is limited by the pedals and the rollbar since the frame was designed to fit riders with varying heights. Therefore while the final design still maintains the general tear drop the frontal area and height of the vehicle is larger than that of stream-lined vehicles built with little room to move and adjust once inside. To ensure that the fairing design still maintained laminar flow over the body, a CFD analysis was performed using SolidWorks Floworks. Computational Fluid Dynamics (CFD) is a method used in the field of fluid mechanics by numerically solving the Navier-Stokes equations by computer. It allows designers to view the flow of a gas or liquid over an object as well as to find the force experienced in order to calculate the drag coefficient. Shown below is a general shape of the desired fairing created in Solidworks. Using the analysis, air was simulated to flow towards the front of the fairing at a velocity of 30 mi/hr.

As it can be seen, very little if any turbulent flow was experienced. The most pressure is experienced at the nose of the vehicle where the air flow makes first contact with the vehicle. Using the equation below the data collected from this analysis can be used to find the estimated drag coefficient.

\[ F_D = \frac{1}{2} C_d \rho AV^2 \]

The flow resulted in a maximum force of 10.23 N and with the maximum cross-sectional area being .51 m², the estimated drag coefficient is .18. It must also be noted that this analysis is done on the fairing only and does not include the wheels that sit outside which will increase drag slightly, though at the speeds being considered this effect is almost negligible. However if the fairing was designed to cover the wheels the frontal area would be much larger which would increase the drag significantly.

**Economic Summary:**
A budget of $5000 was given by the university to go towards materials, components and manufacturing of the vehicle. Table 1 below gives a summary of funding spent. Overall this project was largely under budget.

### Table 1: Summarized cost of one unit

<table>
<thead>
<tr>
<th>Frame</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>steering</td>
<td>238.64</td>
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<tr>
<td>frame materials and labor</td>
<td>932.45</td>
</tr>
<tr>
<td>tires and tubes</td>
<td>176.89</td>
</tr>
<tr>
<td>seat</td>
<td>83.02</td>
</tr>
<tr>
<td>brake system</td>
<td>138.96</td>
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<tr>
<td>drive train</td>
<td>655.34</td>
</tr>
<tr>
<td>other</td>
<td>495.43</td>
</tr>
<tr>
<td><strong>Frame Total</strong></td>
<td><strong>2720.73</strong></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Fairing</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Foam</td>
<td>343.19</td>
</tr>
<tr>
<td>Resins</td>
<td>292.31</td>
</tr>
<tr>
<td>Fabric</td>
<td>421.42</td>
</tr>
<tr>
<td>Sandpaper</td>
<td>62.56</td>
</tr>
<tr>
<td>Misc tooling</td>
<td>453.77</td>
</tr>
<tr>
<td><strong>Fairing Total</strong></td>
<td><strong>1573.25</strong></td>
</tr>
</tbody>
</table>

**Grand Total** | **4293.98**

For the building of the frame most of the cost goes towards the price of components and therefore remains relatively the same. It would be expected for that set price to go down when components are ordered in bulk. Also the building of the frame structure itself involves considerable labor in bending and welding the pipes. A jig must be made to hold the pipes together for welding. Once this is made though, the same jig can be used for each unit. It also takes time to set up machines to do the specific bends required. Were the frame be put into mass production, the machines would most likely have a set setting to account for these bends and reduce time.

When building the fairing most of the funding goes towards fabric and resin cost. When bought in bulk the price of the materials cost less. Also the mold requires about $600 of the manufacturing cost. Currently it is designed as a single-use mold, however if a couple precautions were made the mold could last for a few different layups, significantly lowering costs. Also an investment could be made into better power tools that could perform the same job in a lot less time with a lot less effort. There would still be a large amount that would need to be spent for labor as the fairing is extremely time consuming. After taking into account all of these factors Table 2 below shows the estimated production costs over one month, one year, and six years.

### Table 2: Projected production costs

<table>
<thead>
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<th>Time Period</th>
<th># of units</th>
<th>Cost per unit for:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Frame</td>
</tr>
<tr>
<td>1 month</td>
<td>10</td>
<td>2457.67</td>
</tr>
<tr>
<td>1 year</td>
<td>120</td>
<td>2104.58</td>
</tr>
<tr>
<td>6 years</td>
<td>720</td>
<td>1809.6</td>
</tr>
</tbody>
</table>
The general shape was modeled to analyze the flow. Wide frontal area. While reduced some by placing wheels outside of the fairing is still large to account for the pedal rotation represented by the square.

**Testing**

3.1 Frame Loading

Roll over strength testing was done on the frame to ensure the safety of the rider in the event of a rollover. Load testing was done also to validate the results from the FEA and provide real hands-on test results. The first case tested was top loading in which the frame was constrained at the dropouts and the front wheels/axle as shown in Figure 40. A 400-lbf load was applied at the top of the roll bar in the downward direction, and this was mounted as shown in the figure. There was no significant deflection noted. This concurs with the FEA analysis which shows that the roll bar would only deflect 0.23 inches under a load of 600 lbf. With the results from the testing, it can be conjectured that under another 200lbs, the roll bar would not deflect much. Further testing will verify this and will be discussed during the oral presentation of the design event.

The second test was side loading in which the frame was constrained at the front axle and the bottom of the frame. A 200-lbf load was applied to the side of the frame at the position where the rider’s head would be, as shown in Figure 40. No significant deflection was noted in the roll bar, which agrees the FEA which showed that under a 350-lbf load the deflection was ___ inches. The test results determined that with another 150 lbs, the roll bar would deflect very little.

![Figure 40: Roll bar Load Testing](image)

**Fairing**

Fairing Manufacturing

During the manufacturing of the fairing a test piece was used to try out the effect of different chemicals on the foam as well as to practice applying them. The first chemical applied was the 3m77 spray adhesive. Through testing we found that too much in one place resulted in the foam being eaten away. This caused fewer surfaces to be in contact when stuck together and therefore reduced the
effectiveness of the adhesive. By practicing on the test piece we reduced the chances of ruining the foam. The next step was marking the outlines of the shapes so that when shaping it was recognized not to shave past a certain point. Originally spray paint was suggested. Unfortunately the wrong type was purchased. Thankfully it was tested first and as can be seen in Figure XXXX below, the paint did a considerable amount of damage to the foam. Instead, a permanent marker was used to mark up the foam.

![Figure XXXX: Test piece](image)

The test piece was later filled in and sanded down to be used to practice techniques for applying fiberglass. This was especially helpful since no one had any experience working with the chemicals and materials. It helped establish the methods used and job designations to mixing and applying the resin.

Cooling

When the vehicle is completed each naca-duct will be tested for its effectiveness in cooling the rider. To do this certain ducts will be taped off in sections and the rider will operate the vehicle. From past years we know that if there are cutouts for the wheels most of the air will escape through those and the ducts will not do much. However this vehicle is designed with the front wheels on the outside and the only opening for the back wheel is on the bottom where it makes contact with the ground. Therefore the addition of naca-ducts in strategic areas should be more effective in rider cooling.

Safety

4.1 Frame Safety

For the UCF team, the safety of our riders and all other riders in the competition has been a requirement since day one. For this reason, the team integrated rollover protection, a safety harness and a stable 3 wheel design. The material of the frame is 7005 aluminum which is a very safe and strong material. The roll bar is made of 1 inch outer diameter, ____ inch thick aluminum which provides excellent rollover and side protection for the rider. From the testing it was concluded that the rollbar will provide excellent protection for the rider. The safety harness has been securely placed in the vehicle. This harness will keep the rider inside the vehicle in the case of a collision. The three wheel design that the team chose is a very safe alternative to the 2 wheeled recumbent designs. With a 40 inch wheelbase, it will be almost impossible for the vehicle to tip. Clip pedals ensure that the rider’s feet do not slip and get caught under the frame. Without these clip-in pedals and at high speeds, the rider is at risk of severe injury should his or her foot make contact with the road surface.

4.2 Fairing Safety
In the rare event of a roll over where the vehicle slides on its side, the pilot is protected by several mechanisms. Initial impact protection is provided by the frame’s roll bar. However, the sliding motion typical of such accidents is managed by the outer skin of the fairing four layers of fiberglass in addition to layers of Coremat® in areas the rider is likely to come in contact with (shoulders, arms, hips, etc.). This abrasion protection will prevent painful road rash injuries that are common of this type of crash. The fairing hatch may be opened from the outside to allow a team member to assist any sustained injury. Also all sharp edges and burrs are sanded down and covered inside and outside of the vehicle so that should anyone slip up or crash they will be protected from scratches and cuts. Also there are additional attachments to fasten the fairing to the frame. This reduces the likelihood of the fairing wobbling during movement and causing instabilities. Screens are being placed strategically to allow the rider the best range of viewing to help avoid obstacles such as other vehicles. Naca-ducts will be placed to provide the rider the best air circulation to avoid overheating inside the vehicle.
Resources


