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ASME Human Powered Vehicle Competition Frame Design

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ASME Human Powered Vehicle Competition

Frame Design

By

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Abstract

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Each year, the American Society of Mechanical Engineers (ASME) sponsors the Human Powered Vehicle Challenge. The purpose of the competition is to provide a continued source of research into human powered vehicles. Generally, the vehicles for this competition consist of faired recumbent tricycles and bicycles, capable of reaching high speeds in an efficient manner.

The subject of this report is the design and analysis of a new vehicle frame for Union's 2014 competition entry. The frame is the structural skeleton of the vehicle, supporting the rider and all vehicle components. The frame is what withstands the majority of the loads applied to the vehicle during riding, and needs to be able to distribute them safely around the vehicle without failure. In addition, the frame is meant to act as a safety device. The frame is required to include a roll protection system (RPS), which prevents any injury to the rider during a rollover scenario.

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Introduction

A human powered vehicle is defined as a vehicle which uses the energy provided by a human operator as the only source of power. The defining feature of these vehicles is that no external energy sources are used in their operation, making it the world's most prevalent form of transportation. This is especially true in the underdeveloped or impoverished areas of the world, where access to energy sources is limited. In order to have ongoing research in this transportation area, the American Society of Mechanical Engineers (ASME) hosts a yearly competition between schools from across the globe. The purpose of this competition is to create a human powered vehicle (HPV) which has high efficiency, and can be used as a form of transportation in a daily commute.

Vehicle Components

The design of a successful vehicle for the ASME competition revolves around three major components, the drivetrain, an aerodynamic fairing, and the frame. The drivetrain will be designed by Marjorie Chee, and contains all of the components which propel and control the bicycle. For this project, it includes a set of chain-rings connected by a chain which control the power output of the vehicle. The drivetrain also includes the steering and breaking mechanisms, providing control for the vehicle. Another required aspect of the design is the addition of an aerodynamic fairing, which will be constructed by John Lombardi. When a vehicle begins to gain speed, the resistance it feels due to the drag force also increases. With the addition of aerodynamic devices, this force is reduced, and the vehicle is able to reach greater velocities.

The final component of the vehicle is the frame, which will be described in detail throughout this report. The frame is the underlying skeleton of the vehicle, and is what

connects all of the other components together. In addition, the frame is what distributes any of the loading which occurs while riding. During operation, there are many scenarios in which a load is applied outside of just the rider's weight. For example, when encountering a speed bump or pot hole, the force on the vehicle is a factor greater than the weight of the rider and vehicle combined. The goal of the frame is to be able to withstand such loads, as well as provide a smooth and comfortable experience for the rider.

Competition Requirements

Since the competition will take place with many different vehicles on the track at once, there are safety requirements in place to ensure the rider's safety. Mainly, this is the inclusion of a rollover protection system in the design of the frame. This rollover protection system (RPS) is in place should the vehicle roll over after a crash, protecting the rider from any harm. The RPS must be designed such that it prevents any part of the rider from coming into contact with the ground, both when the vehicle rests on its side, and when the vehicle has flipped entirely over. In addition, the RPS must be able to withstand a variety of loading scenarios without deforming plastically or coming in contact with the rider. These scenarios can be seen in the figure below.

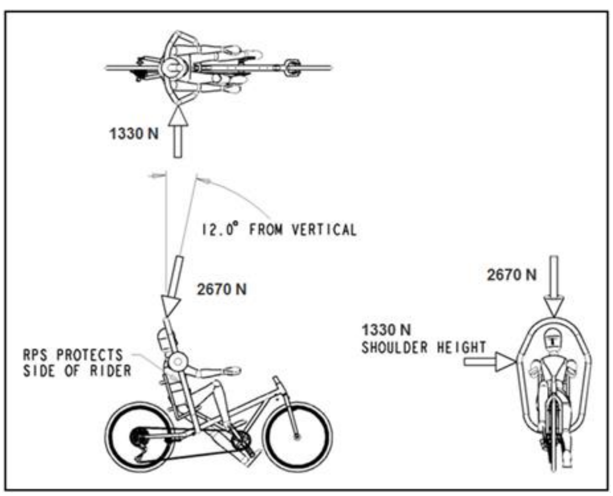

Figure 1: The loading scenarios which the frame must be subject to for competition certification. (ASME)

The first loading scenario involves the application of a load of 1330 N (300 lbf) to the RPS at shoulder height. The second loading scenario involves applying a load of 2670 N (600 lbf) to the top of the RPS, oriented 12° from the vertical towards the rear of the vehicle. Each scenario is tested separately, and the RPS need not withstand both loads simultaneously. Should the RPS fail these requirements, the vehicle will not be able to compete in the competition.

Background Research

As with any design project, there needs to be a starting point. For the design of this frame, the first step was to research frame designs, and find out what makes them successful. As a team, this involved the selection of a vehicle style, which was a recumbent tadpole style tricycle. Since this is the same style of vehicle which was used by last year's team, it was possible to deconstruct the frame design, and find out what features proved to be a success or failure. Afterwards, research was done involving a variety of pre-existing designs. The major source of this research came from past competition teams. By comparing what top-finishing teams did in relation to our past design, it was possible to find a direction to go in.

Wheel Configurations

One of the most influential factors in the design of a HPV frame is the wheel configuration which is chosen. While there are some vehicles with four wheels, arranged at the four corners of the vehicle, most teams opt to use two or three wheels to reduce the rolling resistance.

Figure 2: The three major wheel configuration types: Tadpole Tricycle[1] (A), Recumbent Bicycle[2] (B), and Delta Tricycle[3] (C).

Each of the designs has its benefits and flaws. The tadpole tricycle offers the most stability during turning, as the centrally located dual wheels provide the most support when needed. However, they cause the vehicle to have a large frontal area, which is a large factor in the drag coefficient. A recumbent bicycle generally offers the most efficient option, as the profile is smaller and there is less rolling resistance. However, as a small team, these are the most difficult vehicles to produce. The numerous stops on the course require a stabilizing device to be implemented, especially on vehicles which are fully faired. The delta wing tricycle offers a low frontal area similar to the bicycle, and offers the support of a tadpole tricycle. However, having the dual rear wheels causes the bike to be unstable during turning, and it has a tendency to roll over.

After considering the pros and cons of each of the potential design choices, it was ultimately decided that a tadpole tricycle would be constructed. The tadpole tricycle offers the greatest stability for the rider during all potential scenarios. In addition, the tadpole cycle is one of the easier styles to construct, with the previous frames designed by Union being of this style. Since the school also has been using this design for the past few years, it gives us a better understanding of what does and does not work with them.

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[1] http://trikeasylum.files.wordpress.com/2010/05/actionbent-t1x-suspended-tadpole-trike.jpeg
[2] http://www.linearrecumbent.com/images/Garys-Linear-Limo-3-md.jpg
[3] http://www.tandems-recumbents.com/wp/blog-images/trizard_3352.JPG

Analysis of the Previous Frame

Since the style of tricycle which will be used is identical to last year, the first place to look for ideas was at the old frame.

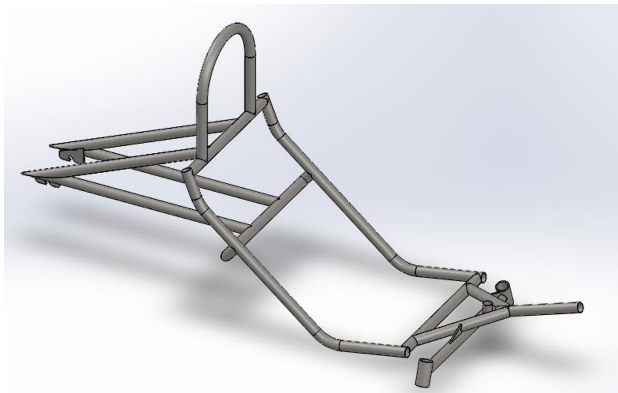

Figure 3: The current HPV frame, designed by Ian Mason in 2011.

After talking to the previous team from Union College, it was clear what the main drawbacks of the frame were. For one, the vehicle was excessively heavy, having a final weight over 100 lbs. A SolidWorks study on the weight of the frame places it at 23.6 lbs., nearly a quarter of the overall weight of the vehicle, without including any tabs or attachments. The team decided to build the frame using 1.25’’ OD 0.065’’ WT 4130 steel (commonly referred to as Chromoly) for the frame, which is overdesigned for the loads which the frame will be subjected to. The frame was able to withstand well over the competition required loads with little to no deformation, but this increase in strength added a large amount of weight to the frame.

In addition to the weight issue, there were also problems with the length of the frame. When the frame was designed in 2011, there was the intention of placing an energy recovery system next to the rear wheel, which created a large gap between the rear wheel and the seat. Last year, where no such system was to be in place, the length of the frame still remained the same. The main problem with the length of the frame is that it dramatically impacted the

turning radius of the vehicle. Because the wheel base is so long, the vehicle was unable to navigate sharp corners, and only barely met the competition required turning radius.

In order to improve on this year's vehicle, the plan is to look into both of these issues. During a previous class project, it was found that the size of steel used in the frame was excessive, and that the amount used could have been reduced dramatically. An optimal size of material will be chosen, which will reduce the overall weight of the frame, while still maintaining the required strength. In addition, the length of the frame should be shortened significantly. By removing the gap between the rear wheel and the rest of the frame, the wheel base can be shortened significantly, which should drastically improve the turning radius and overall stability of the vehicle.

Investigating Working Designs

With the goals of having a shorter and lighter frame in mind, potential frame geometries were investigated by looking at operating tadpole tricycles. The first method of research was looking through pictures from the 2013 ASME competition. When browsing through the pictures, there was one vehicle seemed to meet this description.

Figure 4: The design of the Southern Illinois University Edwardsville HPVC team.[4]

[4] Photo Credit: Rachel Brown'13

The Southern Illinois University Edwardsville team had a style of vehicle similar to what we were looking for. The vehicle has a short wheelbase, and appears to be made out of a far lighter material than 1.25'' chromoly. Looking at the results of the team, they finished 4th overall in the endurance competition, and 5th in the speed trials. It was decided to use the design of this frame as a base point, and see what ways we could come up with to modify or improve on the design.

The most immediate way which was discovered was the use of only a single support spar for the rider. The previous frame for Union used a dual spar design, with crossbars serving as the supports for the seat. This added extra material to the frame, which dramatically increased the weight.

Figure 5: A commercially available vehicle which features a single support spar.[5]

As seen in the figure above, by using only a single beam, there is a very small amount of material used in the vehicle. If a similar support structure is used, along with a shape and size similar to the SIUE design, it should be possibly to drastically improve the performance of this our vehicle.

Material Selection

With the style of vehicle being determined, the next step was to decide on a material which the frame would be constructed from. The most commonly used bicycle frame

[5] http://trikeasylum.files.wordpress.com/2010/05/actionbent-t1a-recumbent-tadpole-trike.jpeg

materials are Chromoly Steel, Aluminum 6061, and a Titanium Alloy. Each material has its strengths and weaknesses, which can be seen in the table below.

Table 1: A comparison of the properties for the three desired frame materials.

Quality	4130 Steel	Aluminum 6061	Titanium 6Al-4V
Density (lbf/in^3)[6]	0.284	0.0975	0.16
Strength (ksi)[6]	63.1	40.0	128
Price ($/ft.)[7]	$11	$12	$14
Strength-Weight Ratio	222.2	410.3	800

The first material which was considered, and the material which we intended to use in the manufacturing of the frame, was Aluminum 6061. The major benefit of this grade of Aluminum is its high strength to weight ratio. While the strength of Aluminum is a fraction of steel, the density of the material is dramatically reduced. This means that while more material needs to be used to reach the same level of strength as a steel tube, the weight of an Aluminum tube will still generally be lower. However, these ideal properties come with some disadvantages, it is much more difficult to machine Aluminum, and the total predicted cost is over double that of steel. In order to weld Aluminum, a specialized method needs to be used which increases the production time. The material must be heat treated after welding to remain at the desired temper. In addition, it is extremely difficult to put a bend into a piece of Aluminum. Due to the brittle nature of the material, it kinks when bending, and there are very few ways to prevent this. Ultimately, since these restrictions were in place, other options were investigated.

[6] "Online Materials Information Resource." *MatWeb*. N.p., n.d. Web. 08 Nov. 2013.

[7] "McMaster-Carr - Tubing Pricing." *McMaster-Carr*. N.p., n.d. Web. 08 Nov. 2013.

The next material which was looked at was a Titanium alloy. Titanium is one of the stronger materials available for HPV construction, meaning that a very small amount of material needs to be used. In addition, the density of a Titanium alloy is much lower than that of steel, further reducing the weight. Overall, this material produces some of the lightest and strongest frames available. However, coupled with these supreme properties is an extremely high cost, almost quadruple that of a similar quantity of steel when accounting for the entire material bill for the frame. In addition, it is very difficult to machine titanium, and the machine shop does not work with the material regularly. Overall, the cost factor of the material was just too great to justify its use, especially with the budget the team is granted.

The material which was ultimately selected for use in the frame was 4130 (Chromoly) steel. Chromoly is not only a strong and ductile variety of steel, but it also is the most machinable of the group. After talks with the machine shop, it was found that most projects generally use this material, and they are very familiar with manufacturing it. Since it is so ductile, it is able to be bent into the shapes which are desired, and it also is easy to weld together. However, at a cost to all of these benefits is the high density of the material. When a large amount of tubing is used in the design of the frame, the weight of this material begins to add up very quickly. To counter this, the minimum amount of material needs to be used in the design of the frame.

Material Testing

Once the material which would be used in the frame was selected, testing needed to be done to determine the size and thickness of all components. In this step, both a static testing method, as well as a dynamic testing method was used on potential tubing choices. In the static method, gym weights were applied to the tubing and the deformation was recorded.

For the dynamic testing, a group of volunteers jumped on the tubing from an elevated height, producing a load greater than the equivalent static load of the volunteer. From the results of both of these tests, a minimum tubing size and wall thickness could be determined which would be used in the design of the vehicle frame.

Static Testing

The first method of testing which was used was the basic static test. The materials being investigated in this analysis were two different thicknesses of 1'' OD Chromoly tubing. One tube had a wall thickness of 0.049'', and the other a wall thickness of 0.058''. The goal of this experiment was to determine whether either of the tubing options could withstand the required 2670 N (600 lbf) force without yielding. The set-up of this experiment can be seen in the figure below.

Figure 6: The experimental setup used for the static loading scenario.

The tubing was supported by a set of benches located in the Union College gymnasium, which were located 29.7 inches apart. A weight carrying rod was attached as close to the center of the span as possible, oriented perpendicular to the test beam. Weight was added to the rod in the form of gym weights, which were added in 90 lbf increments.

After each load was applied, a tape measure was used to measure the height of the bottom of the tubing at the center of the span. The difference between the original height and the loaded height of the tubing represented the deflection.

In order to verify the results of this experiment, two separate methods of analytical analysis were used. The first involved using empirical formulas to calculate the maximum deflection which the beam should have. This was done using the equation:

Equation 1

$$y_{max} = \frac{PL^3}{48EI}$$

In this formula, L represents the distance between each bench, E is the modulus of elasticity of the tubing, I is the area moment of inertia of the tube, and P is the load applied. In a final analysis technique, each rod was recreated in ABAQUS, having the same length as the gap between the gym benches in the experiment. The same loads were applied to the sample in ABAQUS as in the experiment. The comparison of all three analysis methods can be seen in the figure below.

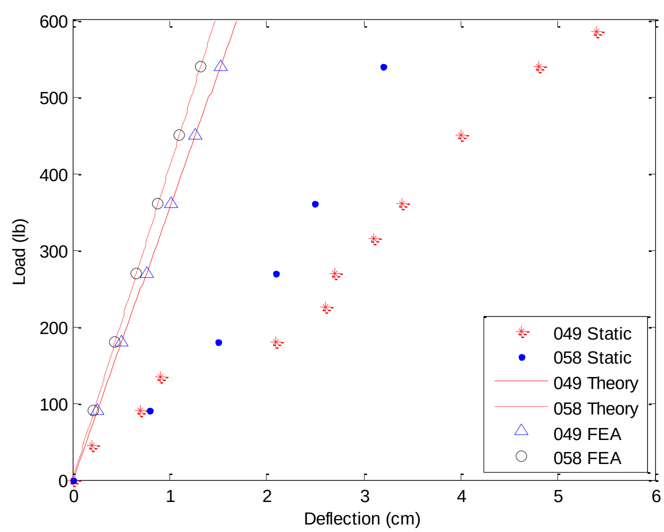

Figure 7: A plot of the deflection against the applied load for a set of tube sizes. The plot contains both experimental and theoretical results. The theoretical analysis was done both using empirical formulas, and ABAQUS.

Unfortunately, the results from the theoretical analyses are far different than the measured results. After careful inspection of the test methodology, it was possible to come up with some theories as to why. As seen in Figure 6, when the load is applied to the sample, it also applies the same amount of force distributed over the two supports. However, the supports used, a set of gym benches, were topped with a layer of foam padding. When the load was applied to the foam padding, it deformed quite significantly. It appears that this deformation is the main source of error in the experiment. Additionally, since only a tape measure was available to measure the deflection, the measurements takes were likely not very accurate. With the weights in the way of the beam, it was difficult to get a consistent reading. While this is a smaller form of error than the foam padding, it still introduces experimental error in the results.

Without the inclusion of the experimental data, the two theoretical results seem to agree. The comparison between the theoretical and ABAQUS analysis can be seen in the figure below.

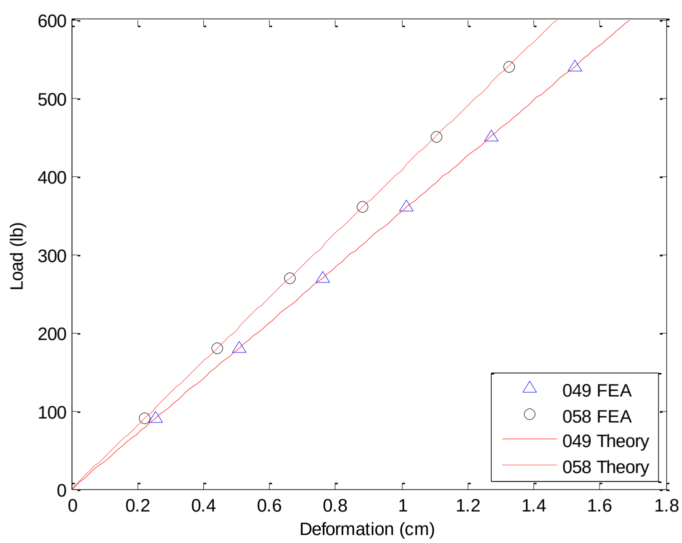

Figure 8: A plot of the deformation against the applied load for the set of tube sizes found from empirical formulas, and ABAQUS.

Both methods are done using a linear geometry approach, which assumes that the length of the pipe does not change due to the bending. While this is not necessarily true, it is a close enough approximation in cases where the deflection is far less than the length. Since there is some error in the measurement of the deflection experimentally, this analytical method can be used to get a relative idea of the deformation in each piece, without the deformation of the foam bench lining.

Thankfully, even with the skewed experimental results, a meaningful result was still obtained. The main objective of this test was to determine whether either of these tubes would be able to withstand this load without deforming elastically. While both of these tubes were able to withstand the load, the 0.049’’ WT tubing deformed elastically. After removing the load, there was still a highly noticeable amount of deformation in the tubing. When the 0.058’’ WT was unloaded, there was little to no deformation visible in the tubing. From these results, it was possible to immediately eliminate the use of 0.049’’ WT tubing, and recognize the potential of 0.058’’ WT as a possible frame material.

Dynamic Testing

In addition to the applying a static load to the tubing, dynamic testing was done using two different material sizes. Since a dynamic load will exert more force on the material than a static load of the same weight, it was necessary to know how the material reacted to such forces. For this experiment, tubes with a 1’’ OD and wall thicknesses of 0.058’’, and 0.065’’ were used. The setup of this experiment can be seen in the figure below.

Figure 9: The set up for the dynamic load testing experiments.

A group of three different subjects were used, roughly covering the size range of the riders. One test subject weighed 120 lb., one 160 lb., and the heaviest 200 lb. Each person stood on a small platform 12 inches above the ground, and jumped onto a tube which was suspended between two supports. Using high-speed imaging, the deflection of the tubing under each load was recorded. As a scale, a ruler was attached to the platform, which allowed the deformation of the tubing to be measured.

The first tubing to be tested was the 0.058'' WT tubing. The results from the heaviest rider landing on the tube as captured by the high-speed imaging can be seen in the figure below.

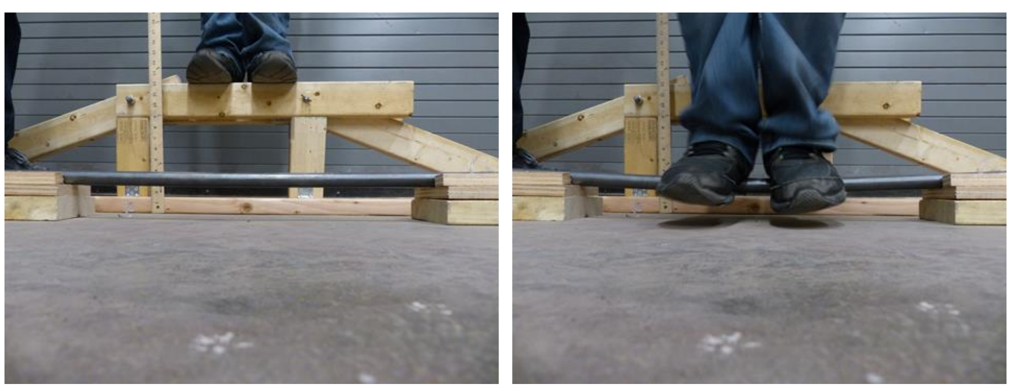

Figure 10: The experiment involving the 0.058'' WT tubing with a 200 lb. test subject. Pictured is the tubing before loading (left) and at the maximum deformation (right).

The results of this test showed a maximum deflection of approximately 0.75 inches from the 200 lb. rider. After the testing was completed, it was noted that there was significant permanent deformation to the piece. There is a high chance that this was due to the fatigue of the material under repeat loadings, as the material was not only tested six times in this experiment, but was also used in the static loading tests. However, even with this knowledge, the material is simply not strong enough to continually withstand an impact load. It may be possible to use this material size for non-impact members or support tubes, but there is potential for failure if a sudden high load is applied to it.

The second tubing to be tested was the 0.065'' WT tubing. The results of the experiment can be seen in the figure below, which shows the deformation captured by the high-speed imaging under the heaviest rider's weight.

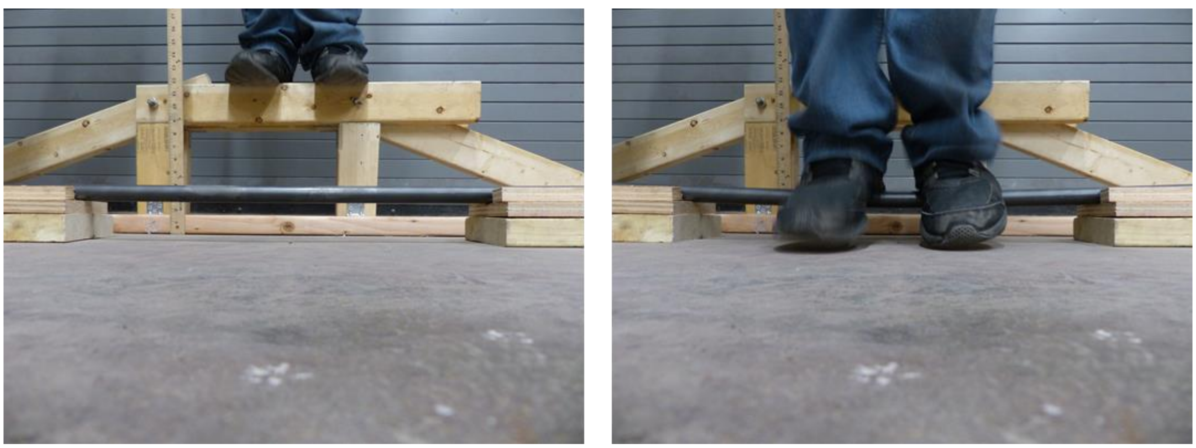

Figure 11: The experiment involving the 0.065'' WT tubing with a 200 lb. test subject. Pictured is the tubing before loading (left) and at the maximum deformation (right).

The results of this test showed that this piece of tubing had a maximum deformation of approximately 0.4 inches under the weight of the heaviest rider. After testing the sample numerous times, it was found that was no visible permanent deformation in the tubing. The loading scenario was applied to the sample at a minimum of six times, so there was an

adequate amount of fatigue in the material. Upon the results of this test, it appears that the 0.065'' WT tubing is the minimum wall thickness which can be used in any impact prone supports. This is especially true for the required roll protection system, which is meant to withstand the impact loading from a rollover accident.

Frame Design

Manufacturing Limitations

With the material for the frame set, and the general idea of the shape in mind, it was possible to start designing the frame. However, even before that could be done, the limitations of the machine shop had to be taken into account. The manufacture of the frame will be completed by the college's machine shop staff. During numerous projects in the past, designs have been submitted to the shop without any prior communication, and with impossible to manufacture components. To counter this, talks with the machine shops began early in the first term to discover what limitations would be placed on the frame. The most prominent of these was the available bending radii.

Table 2: The bending radii available for use at the Union College machine shop, and the minimum wall thickness associated with each.

Tube Size	1'' OD	1.25'' OD
Bend Radius	3 inches	4.5 inches
Minimum WT	16 gauge	16 gauge
Bend Radius	6 inches	6 inches
Minimum WT	18 gauge	16 gauge

Each tubing size had only a few options available, and there are limits on the thicknesses of tubes which can be bent. This put limits on the types of geometries which can be created with the pieces of steel. Other than these limitations, the only other issue involved the material being used. Originally, the plan was to use an Aluminum alloy as the material; however talks with the shop proved that it could not be done. These bend radii are only applicable to steel tubing, which is one of the defining factors of the choice of material.

Other Considerations

The first major consideration with the design of the vehicle has to do with the handling. In order to properly align a suspension, the kingpin angle needs to be properly aligned with the wheel. Since the bearing holding the kingpin is directly attached to the frame, this needed to be considered while designing the front wheel mount. This piece should be angled in such a way that the line going through the kingpin intersects the point where the wheel touches the ground.

Figure 12: An example of the kingpin angle needed to control wheel camber and caster.[8]

[8] http://scottishcarties.org.uk/files/images/kingpin1.png

This is done to prevent any camber or caster in the wheel, which introduces instability. If camber or castor exists in the wheel, there is a permanent tilt in the wheel under static conditions. While some amount of either can help in certain applications, it also makes the design of a steering assembly far more complicated.

Another important aspect involving the drivetrain is the Ackerman compensation angle. The steering linkage is generally attached to the front wheels at the kingpin. The attachment point is offset some angle from the wheel, called the Ackerman angle. To produce the optimal steering, lines extending from these mounts should intersect at the center of the rear wheel.

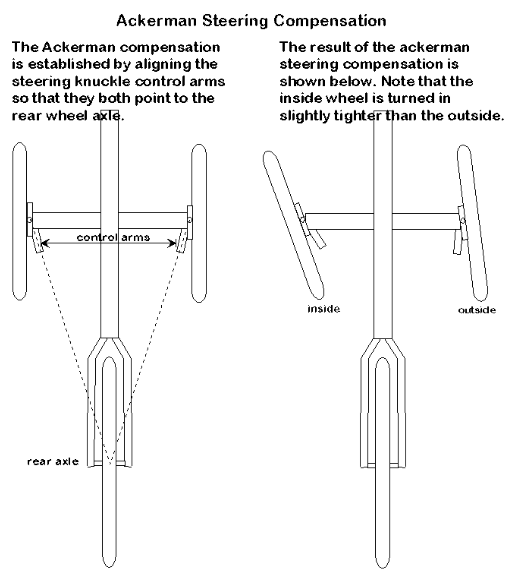

Figure 13: An example of an Ackerman steering geometry[9].

Great care had to be put into the locations of both the front and rear wheels to meet this requirement. The Ackerman angle is an important part of the steering geometry, and can greatly affect how the vehicle will handle. Since the front wheels and spindles of the previous vehicle were being reused, the angle could be easily accounted for in the design stage.

[9] http://www.ihpva.org/Projects/PracticalInnovations/ACKERMAN.GIF

Finally, the last consideration which needed to be made involved the riders themselves. Each rider for this vehicle has different body dimensions, which vary greatly among the group. To compensate for this, the vehicle had to be designed to accommodate all such riders. Anywhere where there may be contact between the rider and the ground during a rollover was designed with the largest dimensioned riders in mind. The more difficult variable is the pedal length, which is entirely dependent on height. To mitigate this issue, an adjustable seat mount was designed for the vehicle, which allows for seat adjustment based on the rider's height and preference.

Initial Design

The first design which was created was heavily influenced by the frame design of the Southern Illinois vehicle pictured in Figure 4.

Figure 14: The first iteration of the frame, based heavily off the SIUE frame design.

The frame features a very similar geometry to that of Southern Illinois, including two RPS pieces which stick off of each side of the frame at the location of the shoulder. It also includes a high peaked top roll bar, which is angled sharply back down to the frame. The

unique features are the attachments to the rear wheel, which were intended to minimize the distance between the wheel and the frame. Another unique addition is the front wheel mount, which is sharply angled to allow for a steering assembly to fit under the main support. The main problems with this frame design were the weight, and the interference with the steering mechanism. As designed, the frame is not much lighter than last year's design, only shaving off approximately six pounds. In addition, the designed steering mechanism intersects with the main spar of the vehicle. While many changes to the geometry of the frame were made to attempt to fit it in, it was necessary to seek an alternative design.

Final Design

The final design was based off of the initial frame idea. The major changes involved the roll protection system, and the front steering mount.

Figure 15: The final design of the 2014 Union College HPV frame.

After carefully rereading the rules, the decision was made to move the shoulder supports of the roll protection system to the rear of the frame. The rules only specify that the RPS needs to be able to withstand a load applied at shoulder height, and that the rider must not touch the

ground when the vehicle is turned on its side. Originally, it was interpreted as the RPS needs to cover the shoulder directly, which was incorrect. The switching of this design feature reduced the amount of the material used; dropping the weight by another three pounds.

The second update involved changing the geometry of the front wheel mount to allow the use of the designed steering linkage.

Figure 16: An assembly of the final frame of the vehicle, with the designed steering linkages and drivetrain components.

To compensate for the steering linkage, the ends of the mount were moved down so that the linkage no longer intersected with the main spar. This was done by changing the overall geometry used in the support. Instead of instantly rising from the main beam, the mount stays horizontal for part of the length before rising to the appropriate angle and height.

The material used for the frame was as minimal as allowed by the limitations of the design. The main spar is designed using 1.25'' OD 0.065'' WT 4130 steel tubing, which is meant to provide extra support for the most loaded section of the frame. The remaining elements of the frame are made using 1'' OD 0.065'' WT 4130 steel tubing, with the

exception of two pieces. With the top tube for the roll protection system, there were difficulties getting the 0.065'' wall thickness tube to bend to the desired angle. The wall thickness needed to be raised to 0.083'', which will not only allow the tube to bend more easily, but will bolster the strength of the most important piece of the RPS. The 0.083'' wall thickness is also designated for use in the front wheel mount. There is only a small gap of 1'' between the main spar and the steering linkages underneath. If it were to be loaded, it may break, as it is not intended to be a load bearing device. To prevent any excess deformation which may break this linkage, the thicker wall tubing will be used, which will increase the strength of the piece.

Frame FEA Analysis

While the material testing gives a good indication of the overall strength of the material, certain geometries may cause parts of the frame to fail under lower loads than expected. Elements such as bends, which are prevalent in the frame design, represent a major form of stress concentration. In order to predict how the material may react in non-standard geometries, a finite element analysis was done using the ABAQUS software. Since it would be very difficult to build and test each component individually, this software allows for the measurement of both deformation and stresses on critical pieces of the frame. Before construction of the frame can begin, these simulations need to be done to not only verify the materials which were selected, but also ensure the safety of the rider.

Main Spar

The main spar is one of the most important elements of the frame. While its main purpose is to provide support to the rider, it also serves as a means of distributing any loads throughout the frame. The loading scenarios which were investigated for this section of the

frame primarily involved forces created by the rider. The rider is the heaviest component of the vehicle, and will be the source of many of the loads on the frame.

The first loading scenario investigated was the weight of the rider alone on the spar. To simulate a loading scenario, the approximate locations of the seat attachments were found. Since we will be reusing the seat from last year, the SolidWorks model of it was placed on the main spar, and the locations were marked. These served as the site of the load application, since this is the area in which the rider's weight will be transferred to the frame. The weight of the heaviest rider (~200 lb.) was applied to the frame, distributed over these two locations. As the boundary conditions, the ends of the tube were held fixed. In reality, the fixtures for this tube would lie closer to the central bend, creating less of a moment arm, and reducing the stress on the beam.

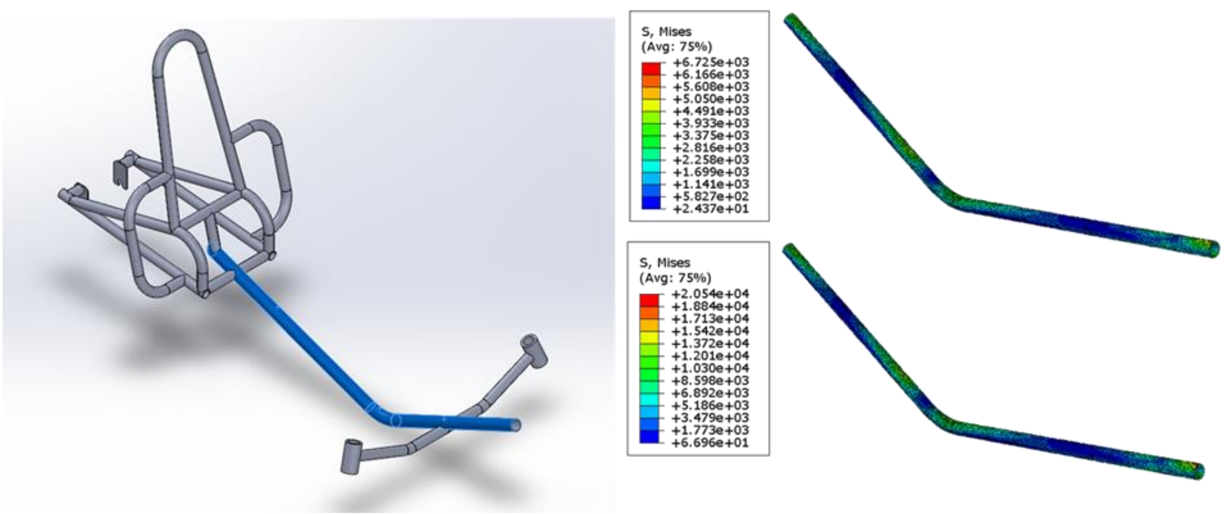

Figure 17: An ABAQUS analysis of the main spar of the vehicle highlighted in blue on the frame (left). The spar was tested under rider weight (top-right), and under an impact scenario (bottom-right) to determine if it would yield.

As a result, the maximum stress acting on the spar is 6.73 ksi. With 4130 steel having a yield strength of 63.1 ksi, this piece has a factor of safety of 9.37. Since this stress is likely higher

than the stress which will actually be applied, this geometry should have no problem supporting the rider.

To investigate the actions of the main spar under impact conditions, a force of three times the heaviest rider's weight (600 lbf) was distributed over the two attachment points. The analysis indicates that the maximum stress applied to the spar due to this loading is 20.54 ksi. Using the yield stress of chromoly (63.1 ksi), a factor of safety of 3.07 could be found. From these results, there is great confidence that this main spar will be able to withstand any potential loading scenarios. Since this is the most critical part of the frame, it was intentionally overdesigned so that it would prove to be the strongest member. With this piece serving as the backbone of the frame, the overall strength of the frame should be adequate enough to withstand most if not all of the loads which may be applied to it.

Wheel Mount

The biggest concern with the wheel mount of the vehicle was the deflection due to loading. Underneath the wheel mount is where the steering linkage will be located. Measurements of the vehicle put the gap between the main spar and this steering linkage at just under one inch. If the wheel mount was to be loaded in situations greater than the rider's weight, the mount may flex such that the steering linkage comes into contact with the main spar. Since the steering linkage is not intended to be load bearing, it may fracture if there is contact. Due to the geometry of the wheel mount, there was potential for enough stress to cause this level of deformation. To investigate this issue, two trials were run, one concerned with only the weight of the rider, and another impact scenario.

The first loading scenario which was investigated was under the weight of a rider alone. While there is not expected to be very much deformation under the weight of the rider,

especially since the load will be distributed among both the front and back wheels, it served as a comparison for a loaded example. In this scenario, the steering mount was held fixed in all directions at the location which it would be attached to the main spar. On each end of the mount, a 100 lbf load was applied to the bottom face, representing a worst case scenario where all of the heaviest rider's weight is distributed over this section of the frame.

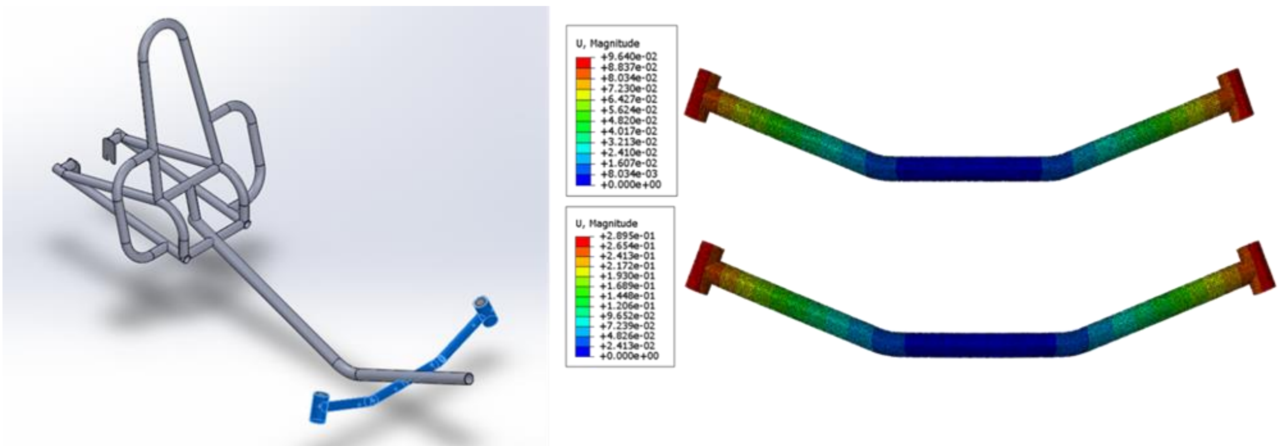

Figure 18: An ABAQUS analysis of the wheel mounts of the vehicle highlighted in blue on the frame (left). The mount was tested under rider weight (top-right) and under an impact scenario (bottom-right) to test linkage spacing.

As a result of this loading scenario, the frame did not deform enough to damage the steering assembly. The steering mount deflected by 0.0964 inches, not enough to cause any interference with the linkages. In addition, it is seen that the stress is not highest at the bends, which was expected since they introduce a stress concentration. Instead, the highest stresses were located at the attachment point to the main spar. Since the main spar will also distribute the load across its length, and the welds are assumed to be stronger than the steel, this should not prove to be an issue.

To test a case where there may be an impact loading, such as when the vehicle travels over a speed bump, a load of three times the rider's weight was applied to the piece. Once

again, the boundary condition was set such that the attachment point to the main spar was held fixed, and loads of 300 lb. were applied over the bottom face of the mount.

Once more, the piece does not deform enough to interfere with the steering linkages. Even with this extreme loading scenario, the maximum deflection of the piece is only 0.2895 inches, less than half of the gap which exists. As in the previous simulation, the highest stresses lie not at the bends in the piece, but at boundary conditions. Even in this extreme scenario, the applied stress is still less than the yield strength of the material and it can potentially be distributed through the length of the main spar. These results agree with the material testing, which showed that even a smaller thickness piece of steel will not permanently deform under this magnitude of a load. With all of these results, there is high confidence in the strength and durability of the piece.

Roll Protection System Side Load

As per the competition requirements, a loading analysis needed to be done on the roll protection system. The first scenario which needed to be analyzed was a side loading condition. A 1330 N (~300 lbf) load was applied to the roll protection system at shoulder height. The boundary conditions are set so that the front wheel attachment faces are held fixed, and the rear wheel acts as a simple support.

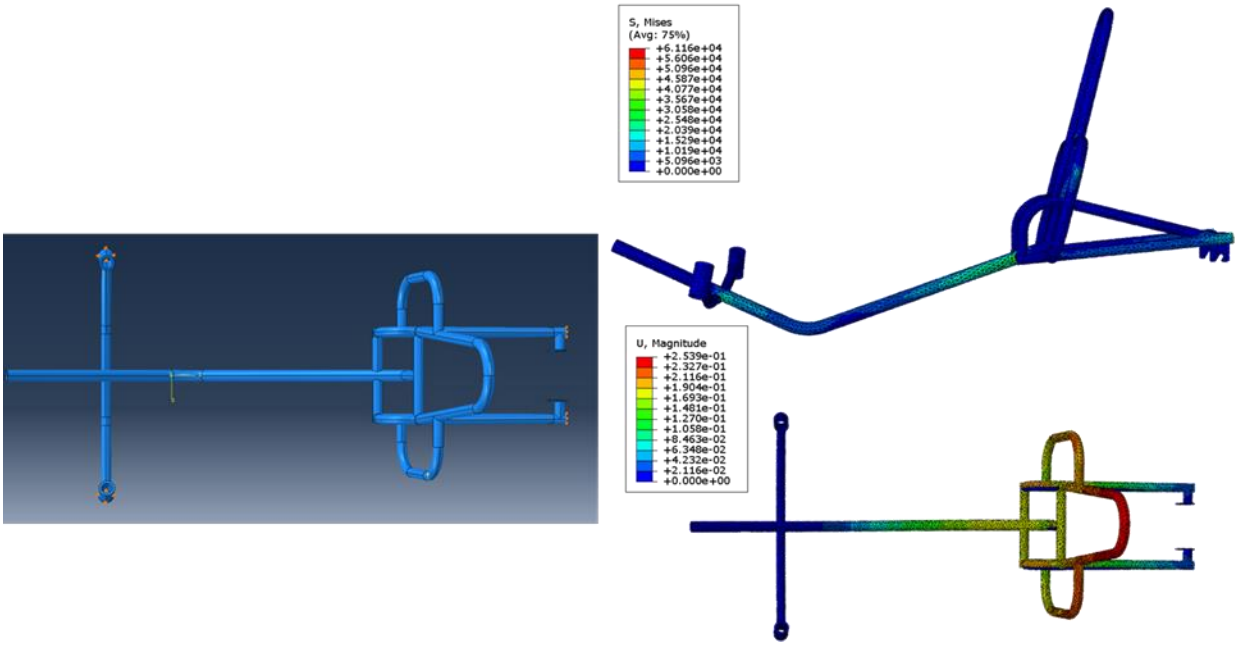

Figure 19: An ABAQUS stress analysis for the side RPS loading (300 lbf). A point load is applied to the side of the RPS at approximately shoulder height. The boundary conditions are placed on the front wheel bearings, and the rear of the frame. Included is a von Mises stress plot and deformation analysis.

As expected, the stresses felt by the RPS are lower than the yield stress of the steel. In order to pass a safety inspection, the RPS must not deform elastically. The results of this simulation show that the maximum stress imposed on the RPS is 61.16 ksi, which is lower than the yield strength of 63.1 ksi. From this, it is possible to calculate a factor of safety for this piece, which is 1.03. While this factor of safety may be low, it should be noted that the stress concentrations were located at the boundary conditions, not at the location of the applied load.

In addition to the not yielding, the deformation of the piece is not large enough to come into contact with the rider inside the RPS. The side bar of the RPS was designed with a total width of 21 inches, an inch wider than the maximum shoulder width of any of the riders. Since the material only deflects 4.18×10^{-3} in, there is next to no chance of it coming in contact with the rider.

Roll Protection System Top Load

The final simulation which was required for the design of the frame was the top load applied to the RPS. According to the competition rules, a load of 2670 N (~600 lbf) is to be applied to the roll bar, offset an angle of 12° from the vertical. The frame is fixed at the front wheel mount bottom faces, and is simply supported at the rear wheel mount.

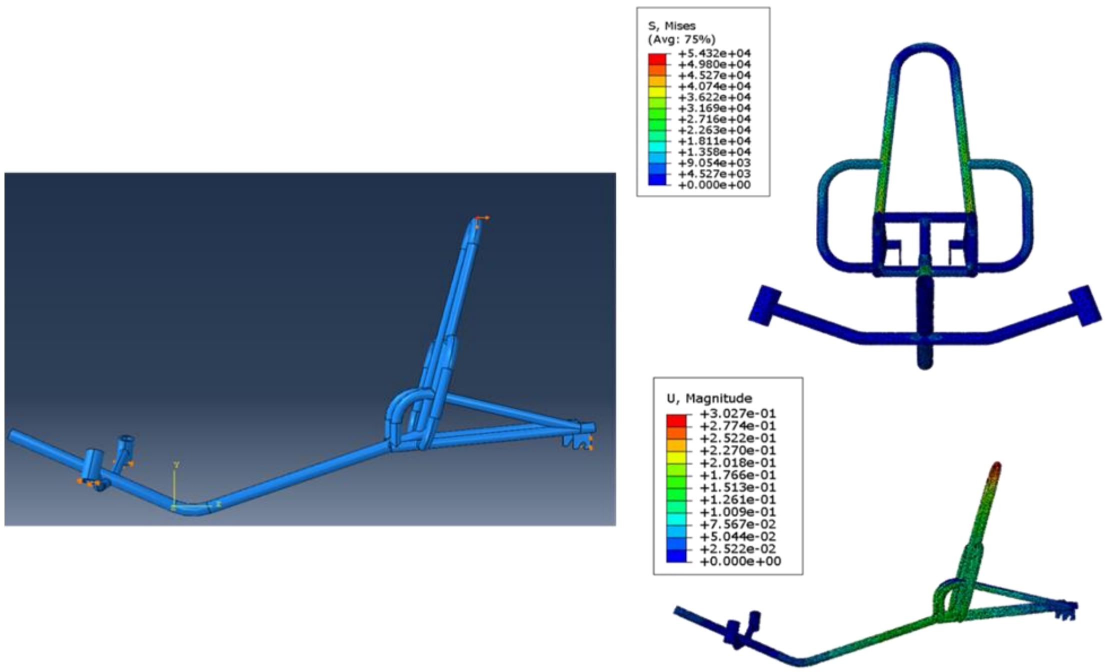

Figure 20: An ABAQUS stress analysis for the top RPS loading (600 lbf). A point load is applied to the top of the RPS, distributed to represent a 12° offset. The boundary conditions are placed on the front wheel bearings, and the rear wheel tab. Included is a von Mises stress plot and a deformation analysis.

According to this analysis, the maximum stress which the RPS is subjected to with this loading scenario is 54.32 ksi. Since the yield stress of chromoly steel is 63.1 ksi, this leaves this piece with a factor of safety of 1.16. With this factor of safety, the RPS should be able to safely handle the competition required loading without yielding.

Since the frame was constructed with the largest dimensions of all riders in mind, there is intended to be a gap of at least 1 inch between the top of the rider’s helmet and the

bottom of the RPS. Under loading, the frame only deflects by 0.303 inches, which is far less than this distance. In addition, due to the angle which the load is applied at, the deformation is primarily away from the rider, toward the rear of the vehicle. Due to this, there should be no conflict with the riders should a rollover occur. From these results, there is great confidence that the RPS will be able to meet all necessary requirements for this competition.

RPS Load Tests

Safety is one of the key factors in designing a human powered vehicle, especially one travelling at high speeds. A crucial component to rider safety is having a roll protection system which protects the rider from any rollover scenario. While theoretical analysis has proven that the frame should be able to withstand expected loading scenarios, it is necessary to compare these results to experimental data, ensuring that the frame is performing as expected.

After construction of the frame was complete, the same load testing which was simulated using ABAQUS was repeated using an experimental method. In the first test method, the vehicle was placed on its side, and a load of 300-lbf (1334 N) was incrementally applied to the frame, and the deflection at each load was measured.

Figure 21: The testing method used to measure deflection in a side impact scenario. A 45-lbf gym bar was chained to the RPS at shoulder height (left). Weights were added to the bar until it was loaded to 300-lbf (right).

In order to test the top loading scenario, differing support heights were used to position the frame such that it would be tilted approximately 12° from the vertical. The required load of 600-lbf (2670 N) was incrementally applied, and the deflection of the frame was measured.

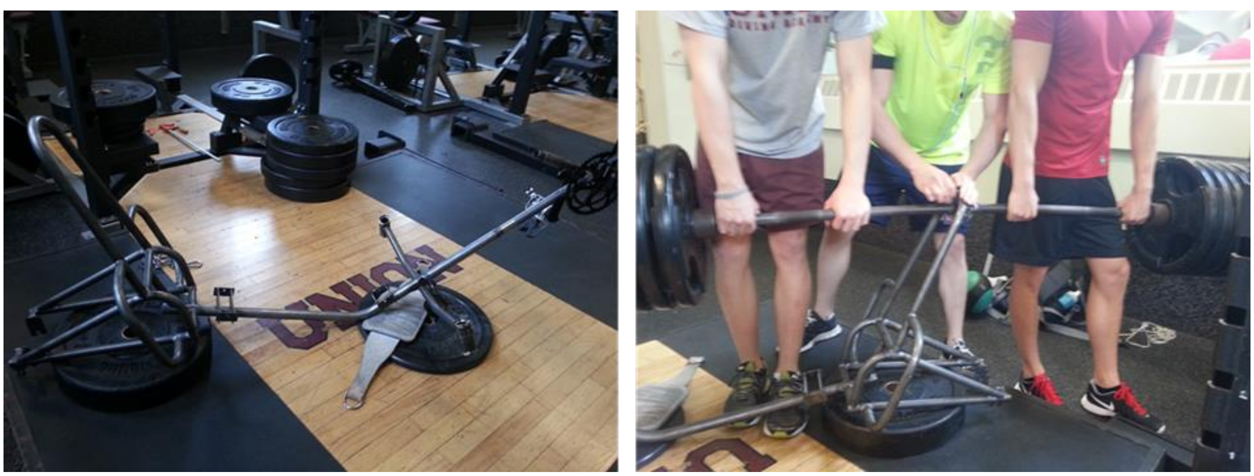

Figure 22: The testing method used to measure deflection in a rollover scenario. The frame was angled to approximately 12° from the vertical via the use of different support heights (left). A gym bar was chained to the top of the RPS and loaded to 600-lbf (right), with assistants ensuring the apparatus did not slide off the top of the RPS.

Compared to ABAQUS simulations, the frame appeared to perform even better than what was originally expected. As seen in the table below, in both cases, the measured deflection of the RPS was less than the analytical estimates.

Table 3: The deflection of the RPS in each load case, including analytical simulations and experimental testing.

Load Case	ABAQUS Deflection	Experimental Deflection
RPS Side Load	0.2539 in.	0.0787 in. (0.2 cm)
RPS Top Load	0.3027 in.	0.2362 in (0.6 cm)

In the side loading scenario, the deflection of the frame was limited by the boundary conditions set during the experiment. In order to safely hold the frame in place during testing, the frame was fixed further up the main spar, decreasing the bending seen in this piece. Since most of the deflection in ABAQUS was attributed to this piece bending, the limited motion seems like a reasonable result. In addition, the deflection of the RPS due to top load could only be measured in the vertical direction. Since the majority of the deflection was towards the rear of the frame, some of the total deflection could not be accounted for. However, the most important result from these tests is that no part of the RPS yielded. From this result alone, it is possible to conclude that the rider will have a high amount of protection in any impact or rollover scenario.

Frame Performance Testing

To test the overall performance of the frame, the vehicle has been driven in a variety of test scenarios. The first crucial test came with the first ever ride by our advisor, Prof. Ramasubramanian. To test the vehicle, it was brought out to a parking lot, and some basic

functional tests occurred. The main goals of the test were to see if the vehicle could not only withstand a rider load, but be able to carry it while peddling, steering, and braking as well. The first test went off without a hitch, and the vehicle safely made it through its first test ride.

After this point, the vehicle went through more unusual riding scenarios to test the strength and comfort of the frame. This included brining the vehicle up and down a set of hills, driving through potholes, and off the curb of the sidewalk. Going up hills proved to be a difficult task for last year's vehicle, and it was important to see that the low gears in combination with the lighter weight allowed the vehicle to travel up steep grades. The potholes and curb drop were meant to represent conditions which may be encountered during a standard ride. When riding over a pothole, the frame was able to absorb all of the impact, and remain intact without changing course by a significant degree. In addition, when dropping off of a curb, the frame only deflected slightly, backing up evidence seen in the FEA studies.

Gathering input from the riders, there were two main points which could be used in improving the design of the vehicle. First, when going over rough terrain, the vehicle is able to absorb much of the energy from impact, which should result in a smooth ride. However, with the single spar design, the cantilevered section of the vehicle has the highest tendency to deflect. Since this is where the rider is located, they instead feel most of the shock being absorbed. One way this could be improved is my adding more side to side support for the spar. Currently, only the rear of the vehicle offers this kind of added stiffness. If this trend were repeated in the front of the vehicle, it may result in a smoother ride. Secondly, there were issues with all of the riders being able to properly reach the pedals of the vehicle. Due to time constraints, a temporary seat mount was placed on the vehicle, which allows for basic

tests. However, due to these issues, the potential of an adjustable seat mount design was investigated.

Adjustable Seat Mount

One of the most prominent issues with the vehicle determined during testing was the location of the seat. The height difference between the tallest and shortest rider of the team is over six inches. While a temporary seat mount was attached to the vehicle, it was fixed to the frame at a location designated for the two tallest members of the team. While this was sufficient for them, the other three riders on the team had difficulty with this seat location. When the pedals were at their farthest location during the stroke, two riders had difficulty reaching the pedals, while one could not even reach them at all. To ensure that all riders were able to operate the vehicle comfortably, an adjustable seat mount was developed.

The main features of the adjustable seat mount are the ability to quickly adjust the position of the vehicle's carbon fiber seat, without compromising the safety of the rider. The mount is made up of two distinct pieces, supports for both the front and rear of the seat, which can be seen in the figure below.

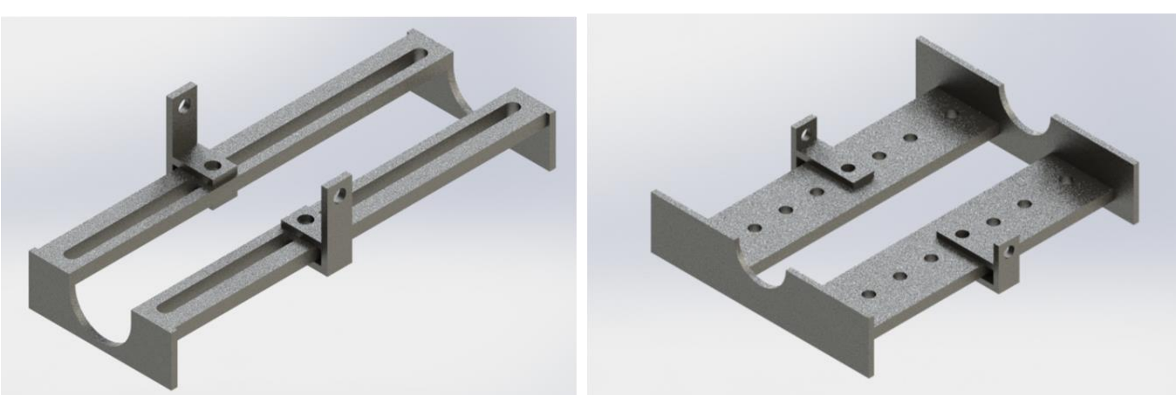

Figure 23: The adjustable seat mount system is made of two pieces, a support for the front of the seat (left), and a support for the rear of the seat (right).

In order to provide adjustability to the system, the seat was constrained in a different method for each support. For the front of the vehicle, the seat is pinned through a set of L-brackets which lie on the support. The L-bracket is then lightly bolted through the slot in the support. This constrains the horizontal movement of the piece, while allowing the brackets to slide forward and backwards when being adjusted. A similar system is used in the rear of the vehicle; however this support also serves to constrain the motion while riding. The seat is once again pinned to a set of L-brackets which rest on the support. Instead of having a slotted design, the rear support features holes spaced one inch apart along the length of the piece. The intention is that a pop-pin, similar to those found on gym equipment, would be used to pin the L-brackets in place. The pin has a spring system which allows it to be disengaged by pulling a small plunger. Since the front of the seat is free to move forwards and backwards, the pin simply needs to be popped and slid over the appropriate hole to adjust the seat position. As a result of this system, the seat is able to be adjusted to bring the rider's legs up to six inches closer to the pedals from rear-most to forward-most position. The motion of the seat through this movement, as well as the location of the seat mount, can be seen in the figure below.

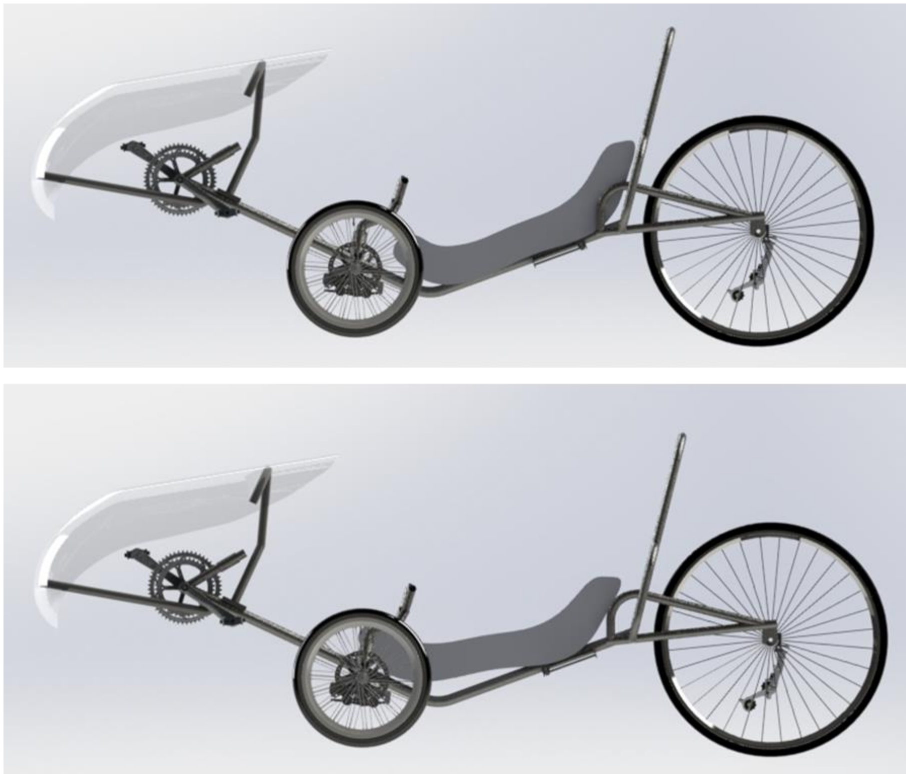

Figure 24: The adjustable seat mount shown in its rear-most (top) and forward-most (bottom) positions.

In order to finalize the design of the adjustable seat mount, FEA analysis needed to be completed to show that the mount will not deform under the weight of the rider, both statically and in an impact scenario. In order to simplify the analysis, the seat mounts will be investigated in a dynamic load scenario of three times the heaviest rider's weight, as this is when the tabs are most likely to fail. Since there are two pieces involved with the seat mount assembly, the rear and front tabs, the load had to be distributed to simulate the distribution of the rider's weight across the seat. As part of this analysis, the load is assumed to be evenly distributed between the front and rear mounting points, with 300 lbf applied to each piece.

The first attachment assembly to be analyzed was the rear mounting point. In the analysis, the adjustable tabs which connect to the seat were fixed in the center pin location in the support span. At this location, previous analysis studies showed the highest level of stresses and deformations. As a boundary condition, the circular cut-outs where the assembly

will attach to the frame were held fixed. As a loading scenario, the 300 lbf load was distributed across the four bolt holes in the attachment’s L-brackets, a total of 75 lbf applied to the lower half of each bolt hole. The results of the analysis can be seen in the figure below.

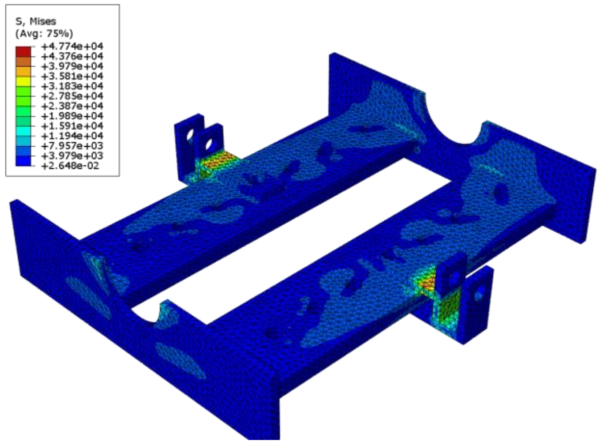

Figure 25: An ABAQUS analysis of the rear attachment assembly of the adjustable seat mount design.

During this loading scenario, the maximum stress experienced in the piece was 47.74 ksi. Since the material being used for this piece was 4130 steel, with a yield strength of 63.1 ksi, the factor of safety could be found, which is 1.32.

The next attachment assembly to be analyzed was the front mounting point. In the analysis, the sliding attachment points were located such that they represented their location when the rear mounting point was in its central position. This allowed for both ends of the seat mount to be investigated at the positions which are expected to produce the highest loads. As a boundary condition, the circular cut-outs where the assembly will attach to the frame were held fixed. As a loading scenario, a 300 lbf load was distributed across the four bolt holes in the attachment’s L-brackets, for a total of 75 lbf applied to the bottom half of each bolt hole. The results of the stress analysis can be found in the figure below.

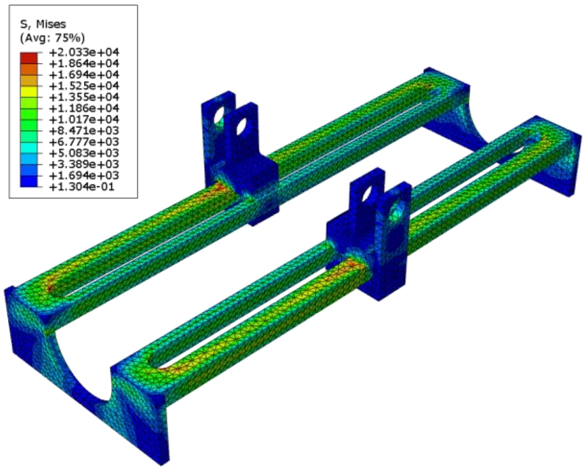

Figure 26: An ABAQUS analysis of the front attachment assembly of the adjustable seat mount design.

In this loading scenario, the maximum stress seen in the piece was 20.33 ksi. Compared to the yield stress of 4130 steel, the construction material, of 63.1 ksi, the assembly was found to have a factor of safety of 3.1.

As seen from the stress analysis results, the adjustable seat mount has adequate strength to support the rider even in the most extreme of loading scenarios. As for the deflection of the piece, it was found to be nearly negligible compared to the scale of the piece being studied. The deflections seen by the pieces are unlikely to interfere with the adjustability of the seat mount as a whole. With this information in hand, it was possible to construct and attach the adjustable seat mounts to the vehicle frame.

Conclusions and Recommendations

In conclusion, the overall results of the project appear to be a success. When beginning the design of this frame, the emphasis on the project was reducing the overall length and weight of the vehicle. Current measurements have put the total weight of this year's human powered vehicle with all components attached at 52 lbf, which is less than half of the previous year's vehicle. While the overall length of the vehicle has only decreased by

one inch, one major improvement was the shortening of the wheelbase. With a wheelbase of only 46 inches, the vehicle has far greater maneuverability and handling, with a turning radius less than half the maximum allowed by ASME. In addition, all of these goals were met without putting the safety of the rider at risk in any way. The roll protection system of the vehicle is able to withstand both loading requirements, and shows no sign of permanent deformation.

At the ASME competition, Chester’s Chariot gave Union its best finish at competition in school history, placing 9th overall out of a field of 36 teams. The highlight of the competition was a 2nd place finish in the endurance race, a four hour race testing the durability of every component of the vehicle. Throughout the competition, the frame appeared to be performing at or above expected levels. It handled the abuse of the endurance course without any mechanical failures, and sustained no damage throughout the series of events. From the performance of the vehicle at competition, it appears that all aspects of the project this year have been a success.

However, even with the successful performance of the vehicle, there are still minor improvements which could be made to increase the vehicle’s performance in the future. First, while a great emphasis was placed on reducing the overall weight of this year’s vehicle, little emphasis was placed on its stiffness. After competition, it was learned that for most sprint vehicles, the frame needs to be very stiff, sometimes at the expense of extra weight, in order to reduce the energy lost during a power stroke. This frame design was not intended to be a stiff design, instead opting for a flexing frame acting as a quasi-suspension system. This potentially harmed us during our sprint event, where we did not meet our predicted top

speed. By adding extra material to the frame to increase the overall stiffness, it may be possible to slightly increase the vehicle's top speed.

In addition to the stiffness of the vehicle, the seat position was still found to be a concern. While the adjustable seat mount was designed to move the rider up to six inches closer to the pedals, in reality, the distance moved was much less. Half of the riding team still had trouble reaching the pedals, and required some improvisation to produce the desired amount of power. To improve on this, the seat mount could be redesigned, but what may be the best option would be to relocate the position of the pedals. By moving the pedals closer to the seat on the vehicle, the adjustability of the seat may become relevant, as it was simply too far away to accommodate the desired range of riders. In addition, this would also decrease the bending moment in the pedal mount, adding some of the desired stiffness to the frame.

Since the frame is intended to be reused in the construction of next year's competition entry, it may be useful to look into these aspects of the frame, and improve on it for next year. Overall, the performance of the vehicle has proven that the frame is a valid, tested, and durable design. With a few minor adjustments, it very well may be possible to improve on this year's results, and continue building a successful team for Union's future.

Works Cited

ASME. *Rules for the 2014 Human Powered Vehicle Challenge*. New York, NY: American Society of Mechanical Engineers, 2013.

Special Thanks to:

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