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Design and Optimization of the Steering Braking and Drivetrain for the ASME Human Powered Vehicle

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Design and Optimization of the Steering, Braking, and Drivetrain

for the ASME Human Powered Vehicle Challenge

By

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DESIGN AND OPTIMIZATION OF THE STEERING, BRAKING AND DRIVETRAIN FOR THE ASME HUMAN POWERED VEHICLE

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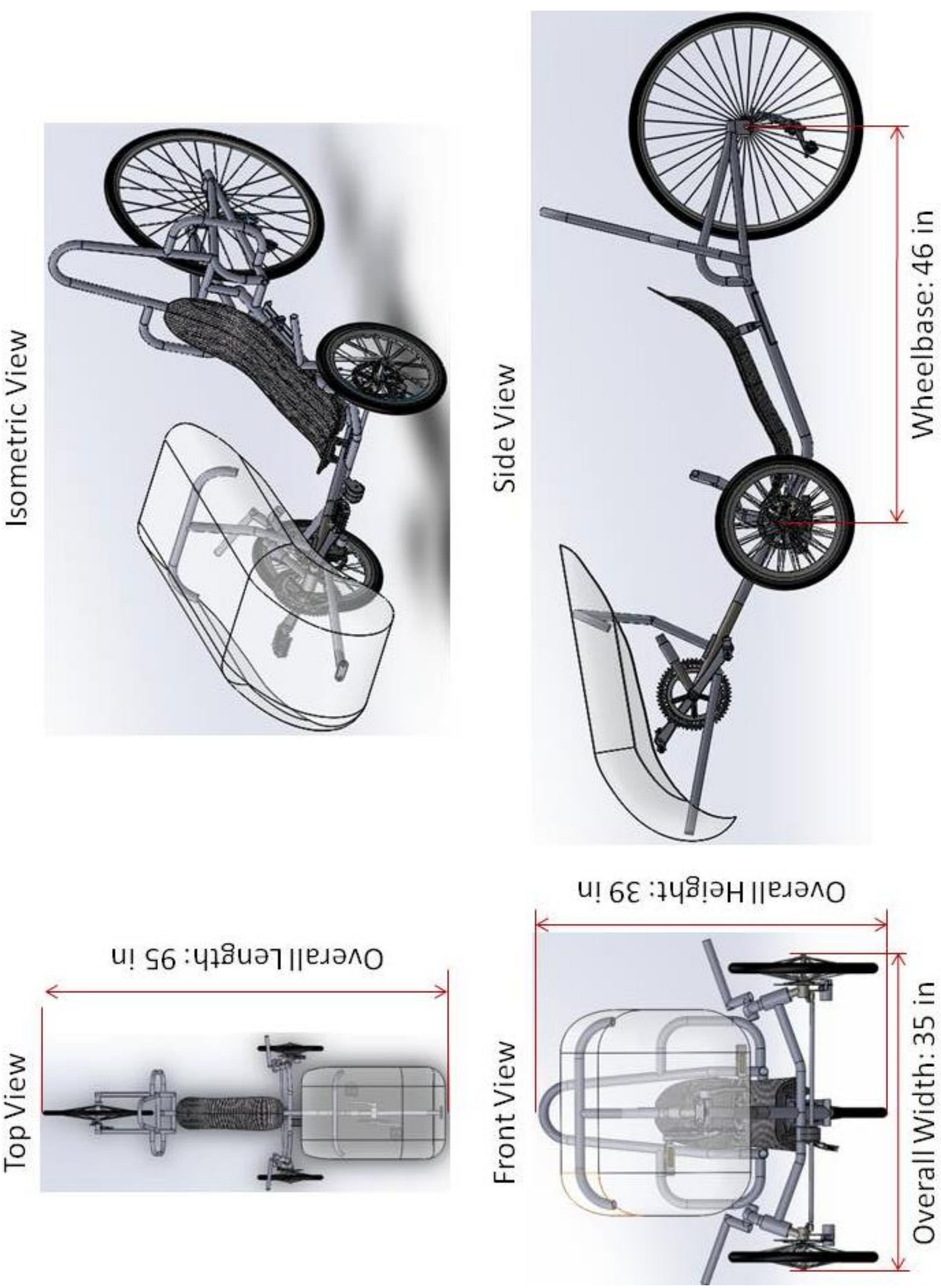
ABSTRACT

The Union College Human Powered Vehicle Team has designed, built and tested Chester's Chariot for the 2014 American Society of Mechanical Engineers Human Powered Vehicle Challenge. The goal of this year's team was produce an efficient, sustainable, and practical human powered vehicle with a high degree of safety. The team conducted gear ratio analysis, beam bending analysis, aerodynamic analysis using computational and analytical methods. In addition, physical testing was completed to demonstrate the safety and performance of Chester's Chariot.

Chester's Chariot has the capability of obtaining higher speeds due to an optimized gear set. Last year's vehicle did not have enough high gears and were not able to reach their desired speeds. This year's vehicle drivetrain is consisted of a 2 chainring crankset of 39 and 53 teeth, and a 9 speed 11-34 teeth cassette. The gears were chosen through optimization and performance testing. The maximum calculated velocity is 35 mph whereas the previous team's was 26 mph. In addition, two disc brakes are used on each side of the vehicle instead of single caliper brake at the rear wheel. The steering component of the vehicle is comprised of a single tie rod that connects the two front wheels and was designed using the Ackermann Steering Geometry. With these changes, Chester's Chariot is capable of making tighter turns, obtaining higher speeds while still maintaining low speeds for uphill climbs and a more reliable braking system.


Isometric View
Side View
Wheelbase: 46 in
Top View
Overall Length: 95 in
Front View
Overall Height: 39 in
Overall Width: 35 in

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INTRODUCTION

Each year, the American Society of Mechanical Engineers (ASME) organizes an international competition for colleges, universities, and organizations to compete in the Human Powered Vehicle Challenge (HPVC). It is a competition that provides an opportunity for students to demonstrate their knowledge and application of engineering design. Within the United States, there are two divisions: the East and West divisions. Union College will be participating in the Eastern division. This year's competition will take place from April 11 to April 13 in Orlando, Florida, which is two weeks earlier than the previous year's. The competition requires a team of students to design and build a vehicle that is powered entirely by human exertions. Each team is not limited to a specific design; in fact, each team can choose any design they believe will provide an advantage in speed and endurance. With an engineered design, the competition requires each team to perform in several events. These events include speed, endurance, innovation and a formal written design report.

The motivation for ASME to hold this competition is to develop sustainable and practical transportation alternatives for underdeveloped parts of the world where automotive vehicles are not an option. The engineered vehicles should be designed for everyday use including commuting to work and carrying goods to markets.[1] A human powered vehicle is the most sustainable form of transportation as it does not produce any environmental pollution. In addition, a human powered vehicle requires low maintenance and in many emerging nations, people already ride bicycles.

COMPETITION DETAILS

The competition this year is judged on four events. The four events are: the design event, the drag event, the endurance event, and the innovation event. The design event is judged on the overall design of the vehicle; whether or not it is safe, follows good engineering practices, is well supported, and has a roll protection system that can withstand 600 pounds of force from above, and 300 pounds from the side. The drag event is judged on how fast the vehicle can travel to cross a set finish line from a standing start. Vehicles compete two at a time in a tournament style and each rider is given over 350 meters of track to gain speed. The endurance event is judged based on how many laps the vehicle and riders can finish in an allotted time (usually 2 hours), with a track size of 1.5 miles. Finally, the innovation event is judged based on any innovations the team has made to their vehicle. The innovation event allows for teams to develop unique solutions to human powered vehicle problems. The overall competition is comprised of separate events, each of which contributes to the overall score. The scoring for each event is shown in Appendix: Overall Scoring.

BACKGROUND

This will be Union's third year attending the competition. The previous team came in 13th out of 31 competing teams. The breakdown of the results for the previous team can be found within the parentheses in Appendix: Overall Scoring. Because this is the third year that Union will be entering the competition, there already exists a wealth of knowledge from previous efforts regarding the design of the vehicle. The vehicle can be divided into three main sections: the frame of the vehicle, the steering and drivetrain, and the fairing surrounding the vehicle. A fairing is an external structure that is placed over the frame and rider to reduce drag caused by

the air flowing around the vehicle. Zacarie Hertel is in charge of the design of this year's frame and John Lombardi will design and construct the fairing of the vehicle. I am responsible for the design and optimization of the steering, drivetrain and braking of the vehicle.

A drivetrain is any system in a motor vehicle that connects the transmission to the drive axel. In this case, the motor is the human, the transmission is the pedals and crank, and the drive axel is the rear wheel. The steering of the vehicle is usually by handlebars and controls the direction.

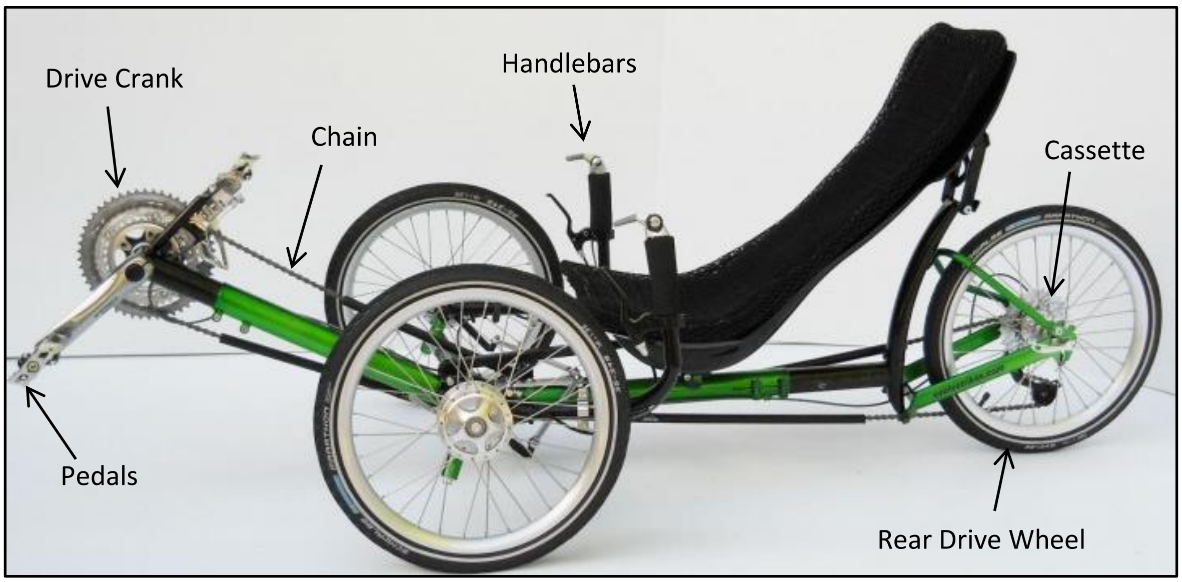

Figure 1: *An image of a tadpole-style tricycle with the drivetrain and steering components identified.*[2]

As shown in Figure 1, unlike a bicycle, the drivetrain extends from the front to the back of the vehicle. The chain is connected to the drive crank and cassette on the rear drive wheel. The drive wheel is in the back of the vehicle because when a person exerts pressure on the pedals, the rear wheel will rotate. The front wheels of the vehicle are connected to the handlebars and are used to steer. The drivetrain and steering have many components that had to be determined in order to build a competitive, well-engineered vehicle.

The previous team used a frame that was designed and constructed two years ago and they focused on the drivetrain and fairing of the vehicle. There were a few problems that the

previous team encountered. The results of the previous team showed that their vehicle was overdesigned for safety. Although the frame was the main support structure of the vehicle, it was designed as if it were for a car. In addition, the length of the vehicle of the previous team was too long. It exceeded 8 feet in length and barely passed one of the obstacle tests. One other significant problem was the weight of the vehicle. The previous team's vehicle was very heavy and weighed over 100 pounds, whereas other vehicles in the competition weighed at least 40 pounds lighter. Another major difficulty that the previous team faced was that there were not enough high gears for both the speed and endurance events. Since they did not have enough high gears, this prevented the vehicle from accelerating even though the rider continued to pedal.

Figure 2: *a) A photograph of the previous team's vehicle with fairing. b) An image of the previous team's vehicle without the fairing.*

As shown in Figure 2, the previous team's vehicle was too long and consisted of a heavy frame and fairing. The steering was also on the left side of the rider, which was not ergonomically placed. Another problem that the previous team faced was their innovation idea. The innovation idea for the previous team was a door on the soft fairing for easy access. The previous team had many points deducted in the innovation event, and this year, the innovation event will be considered into our design.

MOTIVATION

With the problems that the previous team faced in mind, this year the overall goal for the team is to build a vehicle from scratch that would resolve the issues that the previous team encountered. The goals of this year's team were to design and construct a vehicle that is lighter and shorter in length, to optimize the gears to ensure enough high and low gears, to incorporate innovation, and to perform many tests before competing. In addition, many points were deducted from the previous team in testing, and in their design report. These setbacks were considered in this year's team design.

This year, the teams are asked to build the vehicle using sustainable methods, having unique safety features or having a weather protection system as part of the innovation event. Since points were lost from this event last year, this year our innovation idea is to provide the necessary power to fuel a front and tail light using only solar panels. In addition, there are many parts of the vehicle that can be reused and would earn the team bonus points for manufacturing the vehicle using a sustainable method.

Even though the team this year was tasked to build a completely new vehicle, the design of the overall vehicle was kept as the previous year's, a recumbent tadpole-style tricycle. Using the research conducted by the previous team, a recumbent tadpole-style tricycle provides more

stability than a bicycle and has less of a tendency to tip sideways compared to a delta tricycle. In addition, many colleges from the previous competition had the same design, which confirms the research gathered by the previous team.

PREVIOUS DRIVETRAIN AND STEERING DESIGN

The previous drivetrain and steering design was performed by Kevin Skeuse. The drivetrain consisted of a long chain connecting a sprocket that was attached to the crank and an internal gear hub at the rear wheel. Once the rider pedals, the chain turns the rear wheel. The brakes that were used were caliper brakes and the steering design was a linkage-type steering. Caliper brakes are brakes that are applied using friction pads to the rim of a rotating wheel to slow down a vehicle. The problem with using caliper brakes, encountered by the previous team, was that the vehicle was not able to make quick stops. An internal gear hub is a gear ratio changing system that uses planetary gears and is enclosed in a casing. The issue with using an internal gear hub was that the gears are internal and cannot be seen, maintained and repaired easily. The problem that the previous year had using an internal gear hub was that the gears were not properly calculated and were not able to be easily changed.

Figure 3: *An image of the internal gear hub used by the previous team.*

As shown in Figure 3, an internal gear hub was mounted to the center of the drive wheel and could not be easily taken apart. In addition, the shifting of the gears could not be seen and the only way to determine if the gears function properly was by the feel of the rider when he or she used the shifters.

The steering of the vehicle was controlled by a handlebar on the left side of the rider. The handlebar was attached to many rods forming a linkage system that turned the wheel to the left when the handlebar was pushed down and turned the wheel right when the handle bar was pulled. In addition, the shifters were located on the right side of the rider on a stationary handle. The shifter was used to shift gears in the internal gear hub. An image of the steering system of the previous team's vehicle is shown below:

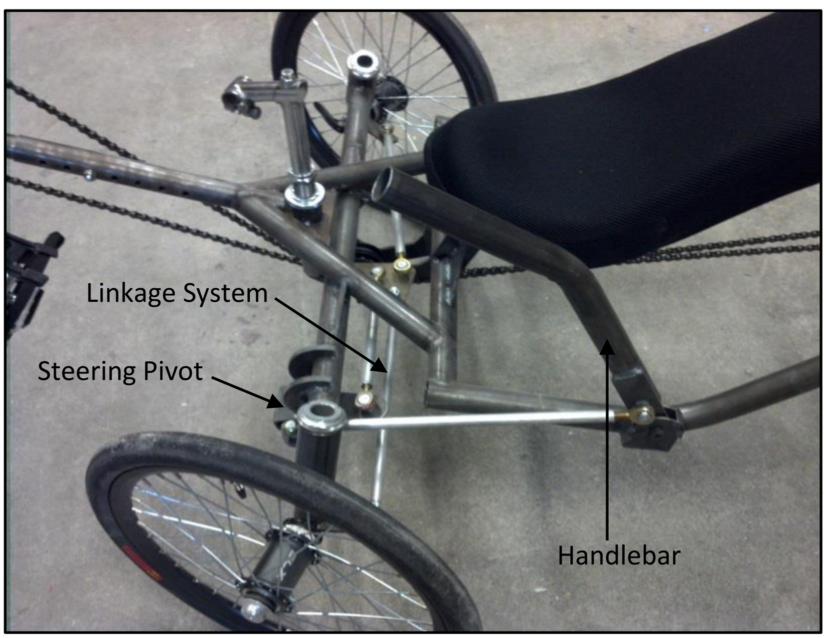

Figure 4: *A photograph of the previous team's steering design where the front wheels turn by pushing and pulling on the steering handle.*

The front wheels turned when a force was applied to the handlebar. When the handlebar was pushed down or pulled up, the connecting rods rotated within a limited degree allowed by the tie rod ends. The steering handle was connected to a rod and that rod was connected to a

steering pivot. The steering pivot translated the forward motion created by the steering handle into a sideways motion that turned the wheels.

CURRENT STEERING DESIGN

The steering design for this year's vehicle is different from the previous team's. Instead of using a complicated linkage steering design that uses many rods, a simple direct steering approach will be used. A direct steering system involves only one connecting rod and connects both front wheels together. There are two handlebars, one on each side of the vehicle, connected to a spindle that is attached to a mount. The mount is where the rod will be placed to connect both front wheels. The direct steering approach was chosen because it is much simpler than a linkage steering design and prevents the possibility of toggle positions.

In order to determine the location of the rear drive wheel, Ackermann Steering Geometry was used. Ackermann Steering Geometry provides the location of the center of the rear wheel by finding the intersection of the two steering mounts.[3] This geometry is shown below in Figure 5:

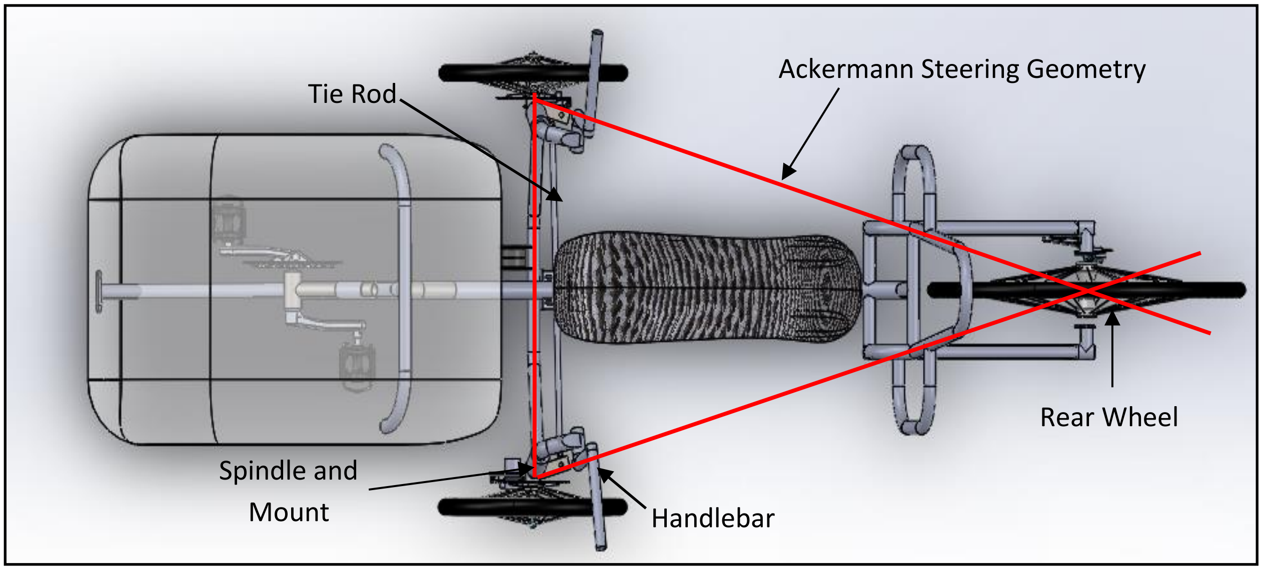

Figure 5: *A SolidWorks model of the direct steering design and Ackermann Steering Geometry for this year's vehicle.*

In Figure 5, the top view of the current vehicle is shown, along with the location of the rear wheel and the direct steering design. The Ackermann Steering Geometry was also used to achieve tighter turns because when the vehicle makes a turn, the inside wheel will turn more than the outside wheel, thus resulting in a smaller turning radius. The Ackermann Steering Geometry also prevents the tires from scrubbing. Tire scrubbing is when the tire slides and rubs against the road, which slows the vehicle down and wastes energy.

The next important aspect to steering was the kingpin and caster angles. The kingpin angle is the angle between the wheel and the steering spindle. The caster angle is the angle from the vertical axis to the steering spindle. The kingpin angle is important for maintaining control when there is contact from bumps and holes in the road. The main function of the caster angle is to provide the front wheels to self center and increase vehicle handling. Using SolidWorks to model the vehicle, the kingpin angle was determined to be 20 degrees in order for the steering mount to be aligned to the patch of tire that touches the ground. The caster angle was determined to be 8 degrees and both angles are shown below:

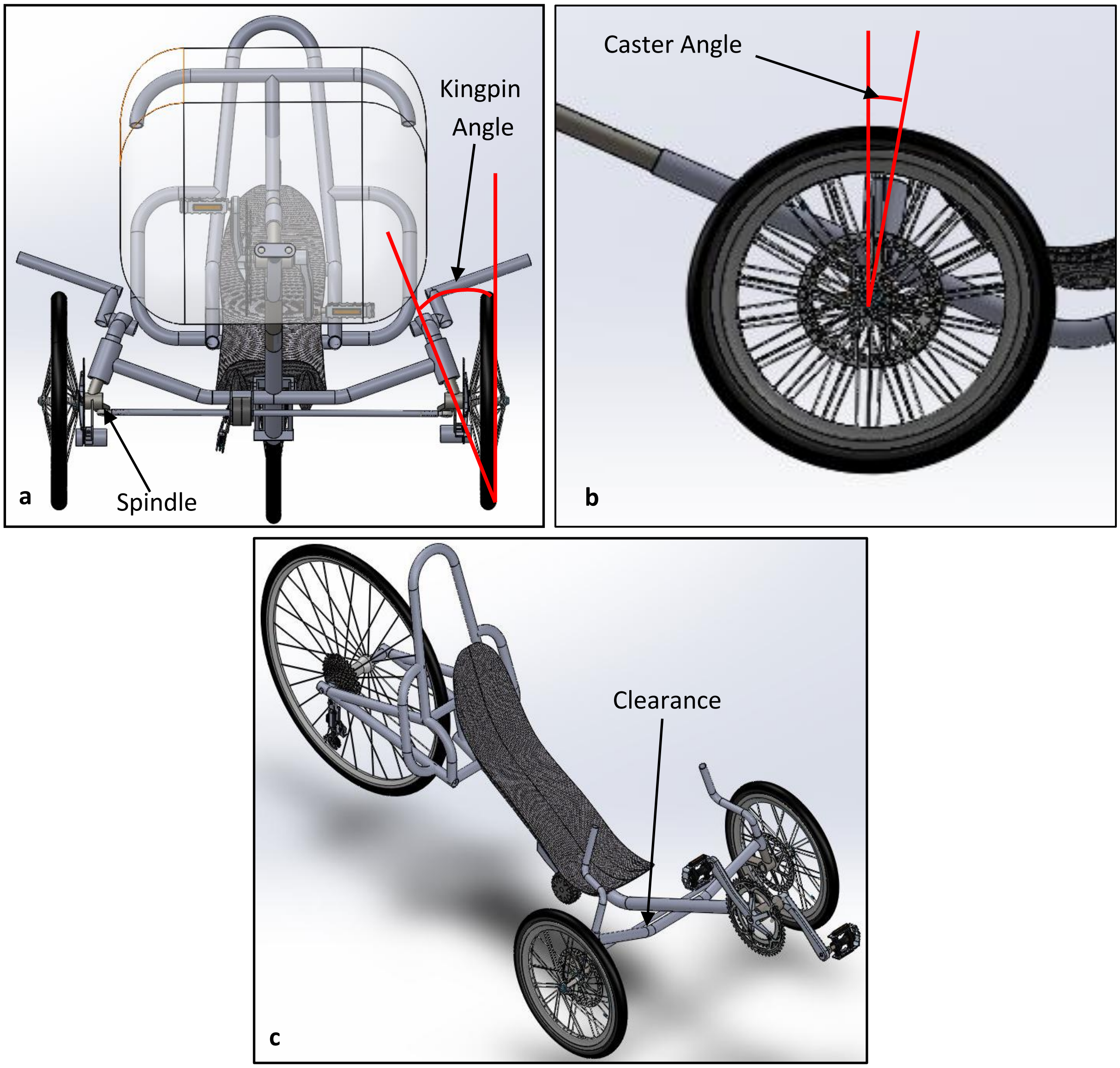

Figure 6: *A SolidWorks model of vehicle showing the a) kingpin angle. b) caster angle. c) clearance between the tie rod and the frame.*

In Figure 6, the kingpin and caster angles are shown. In addition, an isometric view of the vehicle is shown to indicate that there is a clearance of an inch between the tie rod and the frame. In addition to computer modeling, a prototype was also made to model the steering of the vehicle. The prototype in Figure 7 demonstrates the movements of the wheels and the tie rod. Although the prototype showed the movements of the steering system, the movements will be smoother when the steering system is on the vehicle because there will be bearings that will reduce friction between the steering spindle and the frame.

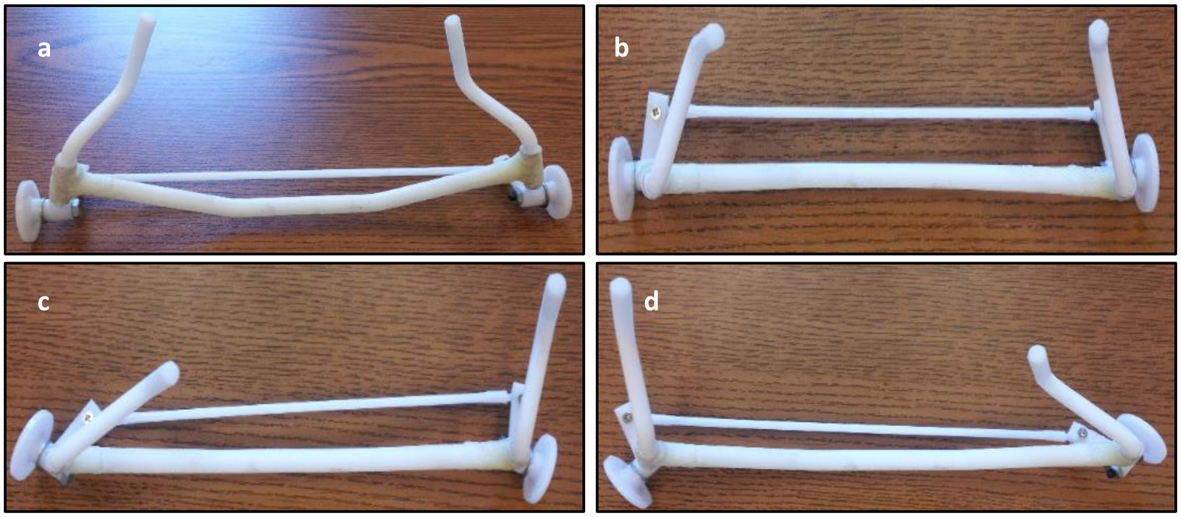

Figure 7: *Prototype of steering system showing different movements: a) front view of steering system b) top view with wheels straight c) top view with wheels turning right c) top view with wheels turning left.*

After the frame was constructed, the handlebars were redesigned to provide a comfortable position for the hands of the rider. The handlebars are connected to a goose-neck stem which is attached to the steering spindle by creating a friction fit along the inside wall through the goose-neck wedge. Figure 8 shows the relaxed position of the wedge, allowing it to be inserted, as well as how the goose-neck stem behaves inside the steering spindle. Inside the steering spindle, the wedge is pulled up and out, creating a very tight fit between the steering spindle and the gooseneck. This provides the direct steering for the rider. When he or she turns the handlebars left, the wheels will also turn left and vice versa.

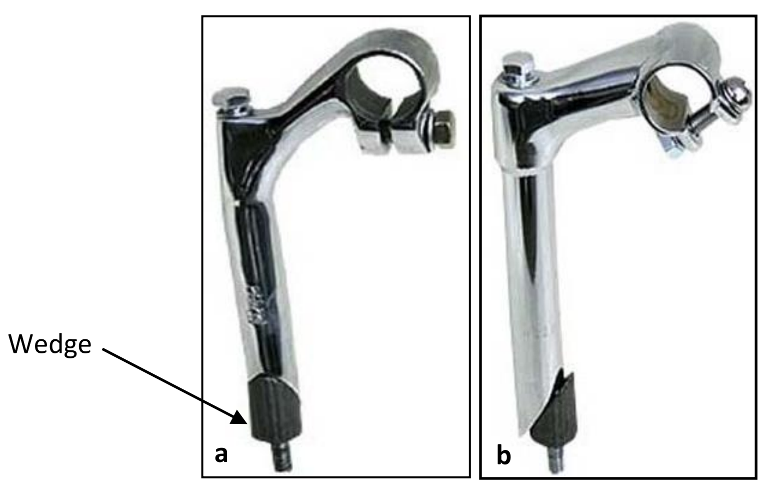

Figure 8: *a) Goose-neck stem with wedge in relaxed position prior to being inserted into steering spindle. b) Goose-neck stem with wedge in tightened position inside steering spindle.*

In addition, the use of the goose-neck stems provide a height and rotational adjustment that can be configured to each rider. An image of the current steering design that was implemented on the vehicle is shown below in Figure 9.

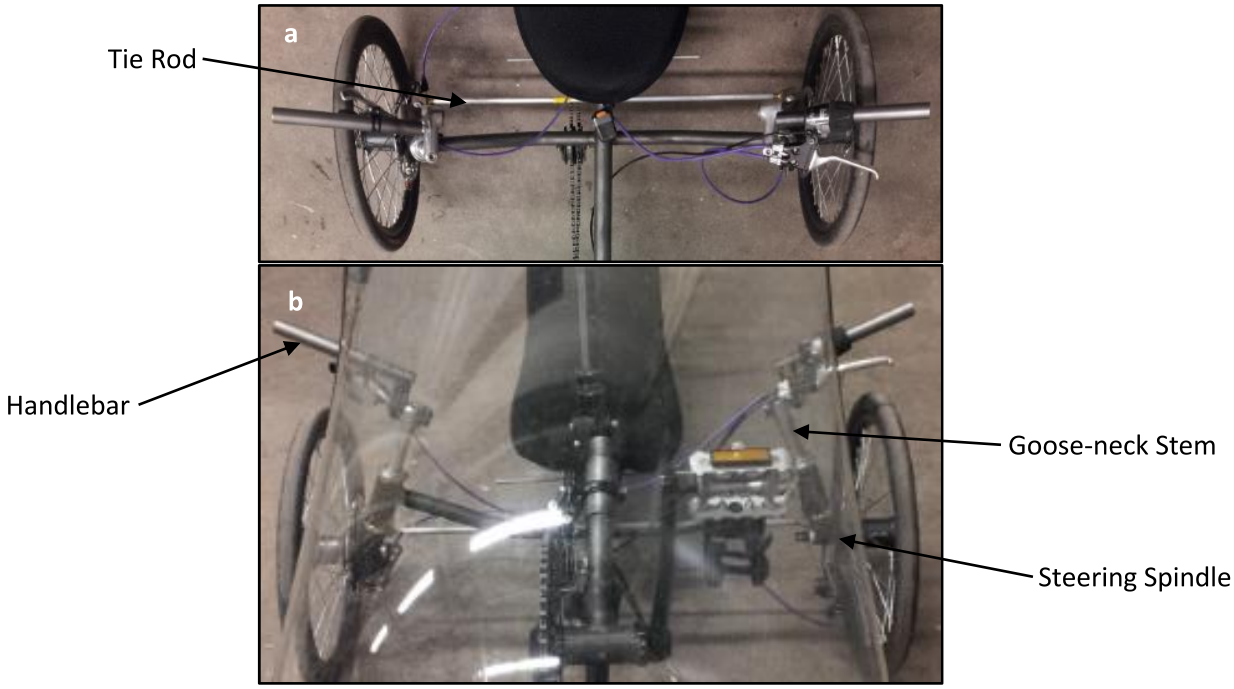

Figure 9: *a) top view of current direct steering system on the vehicle. b) front view of steering system on the vehicle showing handlebars, goose-neck stems, and steering spindle.*

GEAR RATIO OPTIMIZATION

This year, an internal gear hub will not be used; instead a cassette will be used. A cassette is a set of sprockets where each sprocket has a different number of teeth. Cassettes are commercially available, but can also be put together by purchasing individual sprockets. A cassette will be used this year to allow for easy access to the gears for repairs and will easily show if the gears are shifting correctly. In addition, the drive crank will have two sprockets instead of one. Having two sprockets in the drive crank will allow for a greater range of gear ratios.

The chosen gears for this year's vehicle is a cassette that has sprockets with teeth ranges from 11 to 34. The drive crank has a sprocket that has 39 teeth and one that has 53. These two gears were chosen through calculations and physical testing. The drive crank was first chosen because these are commercially available and are commonly used on racing vehicles. The cassettes were chosen through an exhaustive optimization method. The gear ratio for each commercially available 8, 9, and 10 speed cassette was determined using the gear ratio chart in the book, *The Bicycling Guide to Complete Bicycle Maintenance & Repair*.[4] Just the gear ratios alone were meaningless without an idea of what speeds can be obtained. For each set of cassette, the maximum and minimum speeds were calculated at a specific cadence. Cadence is the revolutions per minute that a person can pedal. The average person pedals at approximately 90RPM. For speeding, a person can easily pedal beyond 90RPM and for uphill climbs, a slower cadence of 70RPM is used to determine the speeds. The gear ratios were converted to gear inches to take the size of the tire into account. An example of one of the spreadsheet for the gear inches for each cassette set is shown in Appendix: Gear Inches of Available Cassettes for a 700C Wheel for 70RPM, where the purple represents possible gear cassette for the vehicle and green is the chosen cassette.

After determining the gear inches for each combination of cassettes, calculations were done to determine the minimum gear inch for both speeding and uphill climbing situations. With the help of Professor Keat, the calculations showed that the gear inches were independent of the size of the tire and can be used to compare the gear inches obtained from the cassette combinations. In addition, the calculations performed took the rolling resistance and the drag on the vehicle into account. For the speeding case, the calculations assumed a cadence of 100RPM and a power output of 0.938HP. This assumption was determined from the data gathered from

human power output curves and several journal articles that determined the average output power of a person over 30 seconds.[5] The results from the calculations were able to indicate the velocity of the vehicle to be 35 mph, which was consistent with a table found from another article that focused on human powered vehicle performance.[6] The gear inches found from the calculations were 117.7 inches, and this gear inch was within the gear inches for the combination of cassettes that were found. This gear inch that was found from the calculations also assumed a total weight of 220 pounds: a 150 pound person and a 70 pound vehicle.

For the incline situation, the gear inches were calculated to be 52.2 inches and the velocity obtained for the same vehicle at a cadence of 70RPM with a power output of 0.4HP will be 10.9 mph. These results were again confirmed with the journal articles. The incline situation assumed a 5% grade uphill, which was the maximum according to ASME. The rolling resistance, drag and weight of the vehicle remained the same for the incline situation. These calculations can be found in Appendix: Speeding Gear Optimization and Incline Gear Optimization.

The calculated gear inches were then compared to the available cassettes and the cassettes without the calculated gear inches were eliminated. The objective of testing the calculated gear inches was to determine if the calculations concur with actual cassettes. By a process of elimination, there were 5 possible cassettes that had both the incline gear inch and the speeding gear inch. The final cassette was chosen through testing by one of the riders, Jesse Coull. The gears available during testing were a drive crank that had a 39 teeth sprocket and a 53 teeth sprocket. The available cassette ranged from a 11 teeth sprocket to a 27 teeth sprocket. The low speed was the major factor in choosing a cassette because of the weight of the vehicle, thus the testing was focused on the speed on the uphill climb. For uphill climbing, the teeth used were 39 from the drive crank and 27 from the cassette. With this combination and pedaling at a

cadence of 70RPM up a hill greater than 5% gradient, an average speed of 6mph was obtained using a radar gun. Below is a graph that shows the speeds obtained for five trials.

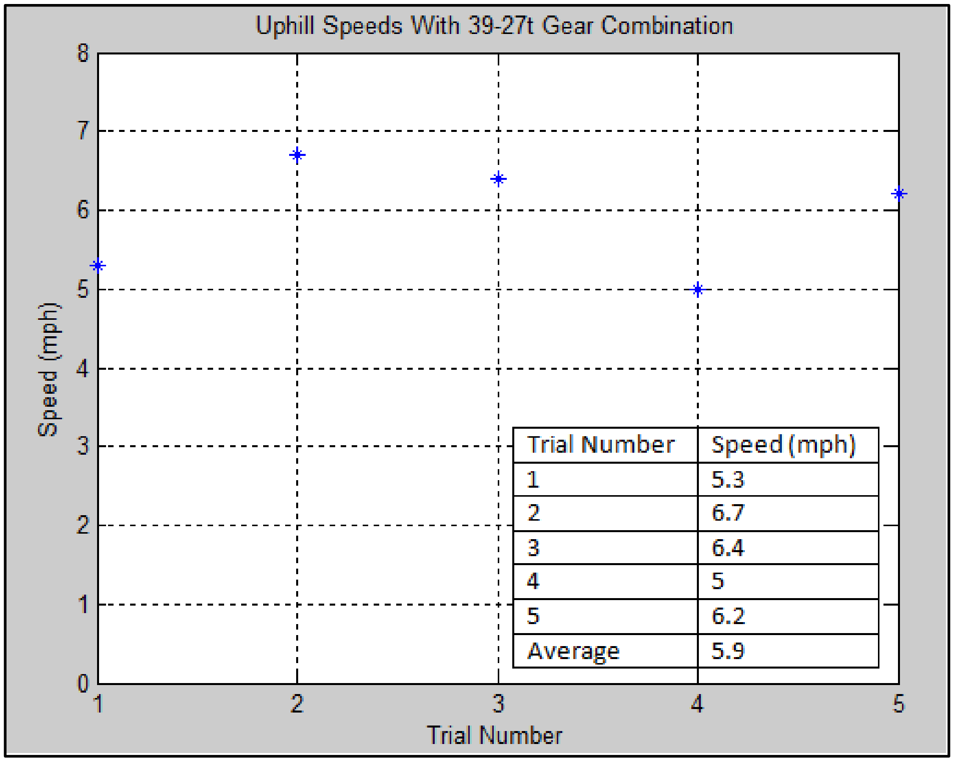

Figure 10: *Uphill Speeds with 39-27t Gear Combination at 70RPM cadence.*

The speeds obtained from the testing were consistent with the gear ratio comparisons. The top speed was not a concern when choosing the cassette because all the possible cassettes had the same gear inch. In addition, it is difficult to reach the highest gear inch, but it was still chosen in our cassette to include the potential of someone reaching it.

The cassette chosen was optimized to provide low enough speeds for traveling uphill and also high enough speeds for speeding. Comparing our top speed with a cadence of 100 RPM to the previous year, by using the cassettes chosen, this year's vehicle will be 9 mph faster than last year's. Below Figure 11 shows a graphical representation of the speeds that can be obtained from this year's gears and the previous year's. As shown, there are a greater variety of speeds for this year. The maximum speed that can be reached at a cadence of 100RPM is 35 mph, whereas the

previous team's maximum speed was 26 mph. This also shows that this year's gears will be able to maintain the previous team's minimum speed and surpass their maximum speed. The previous team's speed was limited by the single sprocket drive crank and the internal gear hub that was used. This was the reason why their gears were maxed out very quickly in each race and were not able to accelerate. The calculations to determine the previous team's top speed is shown in Appendix: Top Speed Calculation of Previous Team Using Internal Gear Hub.

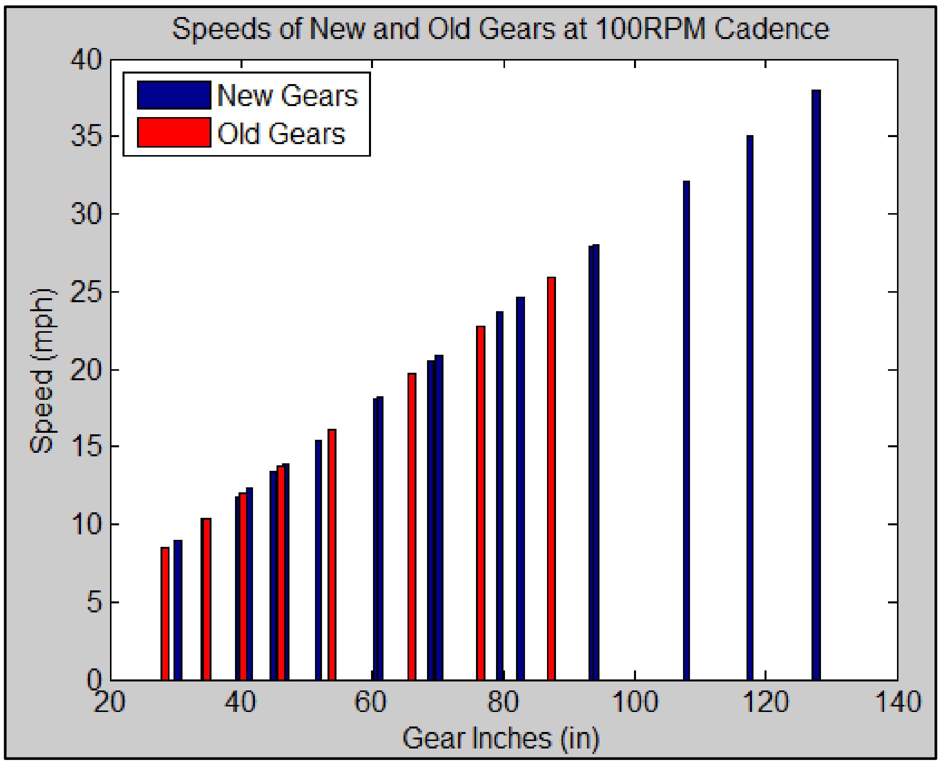

Figure 11: *Speeds of new and old gears at 100RPM cadence.*

CURRENT DRIVETRAIN DESIGN

As mentioned earlier, the optimal gears for this year's vehicle is a front two chainring crankset consisting of 39 and 53 teeth and a 9 speed cassette with sprockets ranging from 11 to 34 teeth. Since this year's design has two sprockets for the front chainring, a front derailleur was required to shift the chain from the 39 teeth sprocket to the 53 teeth sprocket or vise versa. The

front derailleur is clamped onto a front derailleur mount, which was specifically designed to ensure that the chain will not interfere with the derailleur cage when shifting to the 39 teeth chainring. This is shown in Figure 12.

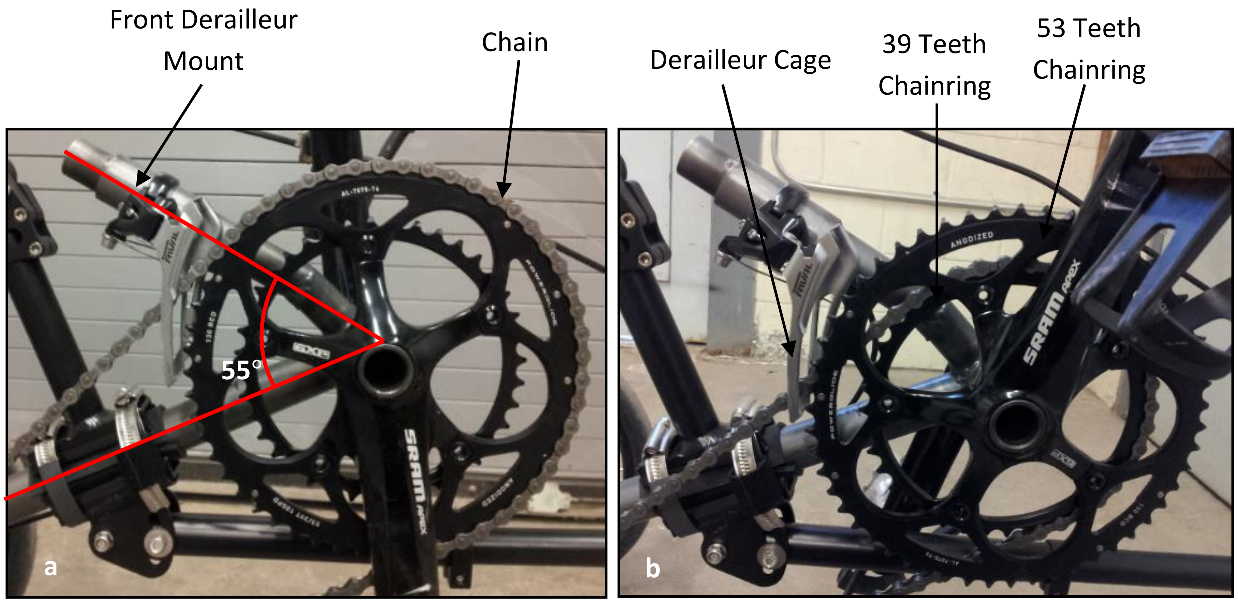

Figure 12: *Chain position on a) 53 teeth chainring. b) 39 teeth chainring. Both cases are free from interfering with derailleur cage.*

The angle between the frame and the derailleur mount was determined to be 55°, whereas standard bicycles range from 66° to 69°. There was a difference between the angle for the front derailleur mount because the chain for a standard bicycle runs horizontally between the front and rear sprockets. The chain in the designed vehicle does not run horizontally to the rear sprockets because the front crankset is positioned in a recumbent style. The chain descends after leaving the front crankset and is guided by an idler to the rear sprockets. The idler is positioned under the frame to provide a path for the chain that eliminates any interference with the rider and any parts of the vehicle. A photograph of the idler on this year's vehicle is shown in Figure 13.

Figure 13: *An image of the idler mounted onto the frame of the vehicle.*

The idler is designed seven inches off the ground and the inner ring of the idler is placed in line with the front crankset. The inner ring of the idler guides the chain to the front crankset from the rear sprockets and the outer ring of the idler guides the chain from the front crankset to the rear derailleur. The idler is mounted in between two tabs and a chain keeper is used to prevent the chain from falling to the ground.

The rear derailleur is directly mounted onto the frame. A rear derailleur tab is designed to hold the rear derailleur in place and is welded to the frame. Housing stops were also made and welded to the frame to keep the cable housing from moving and allowing the rear derailleur cable to be pulled. The rear derailleur with a housing stop is shown in Figure 14.

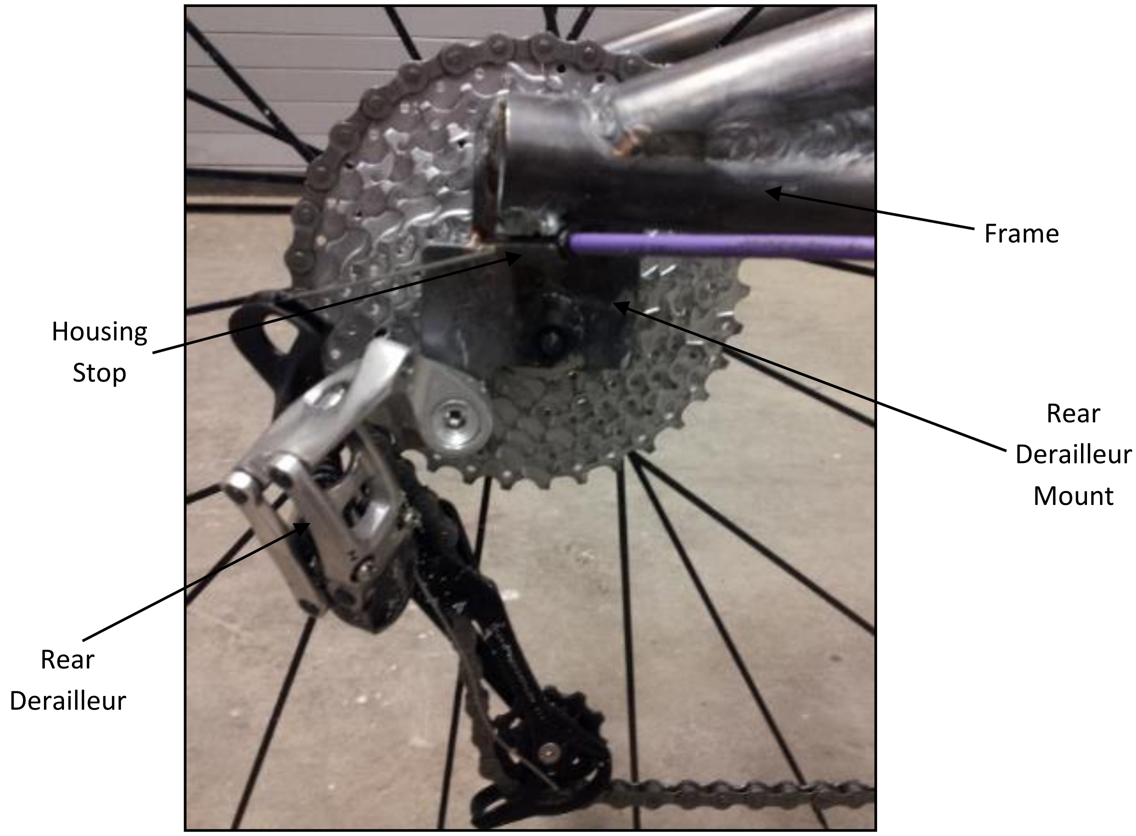

Figure 14: *Rear derailleur with mount and housing stop attached to the vehicle's frame.*

BRAKING DESIGN

This year the braking system is different from the previous team. The previous team used a caliper brake on the rear wheel. This year, disc brakes are used on each of the front wheels. Unlike caliper brakes, the disc brakes provide greater control for braking. Disc brakes will also be used as the braking system because they are able to make quick stops during the competition.

The disc brakes will be located at the front of the vehicle, which will allow the vehicle to stop even if someone is pedaling. In addition, the caliper brake was located very close to the edge of the tire where dirt can accumulate and would be harder for the friction pads to grip the rim. However, the disc brakes are centered at the wheels and if the vehicle rolls over dirt, the disc brakes and friction pads will not be affected.

Figure 15: *Photographs of disc and caliper brake systems. a) top: Wheel with disc brake system. b) left: Clean wheel with caliper braking system. c) right: Dirty wheel with caliper braking system.*[7]

As shown in Figure 15, the caliper brakes are useful when the rim is clean. However, it would not be efficient to continuously clean the wheels. Many tests will be done before the competition so there is a possibility that the rims will be dirty, thus using disc brakes will provide a cleaner surface for the friction pads and will not require maintenance.

After the vehicle was assembled with wheels, custom disc brake mounts were designed and fabricated for each side of the front wheels. Although the vehicle was symmetrically designed and constructed, the brake mounts were not able to be designed to simply reflect the line of symmetry that intersects the front view. Since standard bicycle frames are manufactured to position the disc brake calipers on the left side when viewing from the front view; it was difficult to find a disc brake caliper that is designed to be used on the right side of the bicycle. Therefore, two identical left disc brake calipers were used and the disc brake mounts were

designed to adapt to the disc brake calipers. Unlike a typical bicycle where there is plenty of space for the disc brake caliper, as shown in Figure 15a, the design of this year’s vehicle has limited space for the disc brake caliper and can only be positioned below the steering spindle. A photograph of the left side and right side of the disc brake mounts are shown in Figure 16.

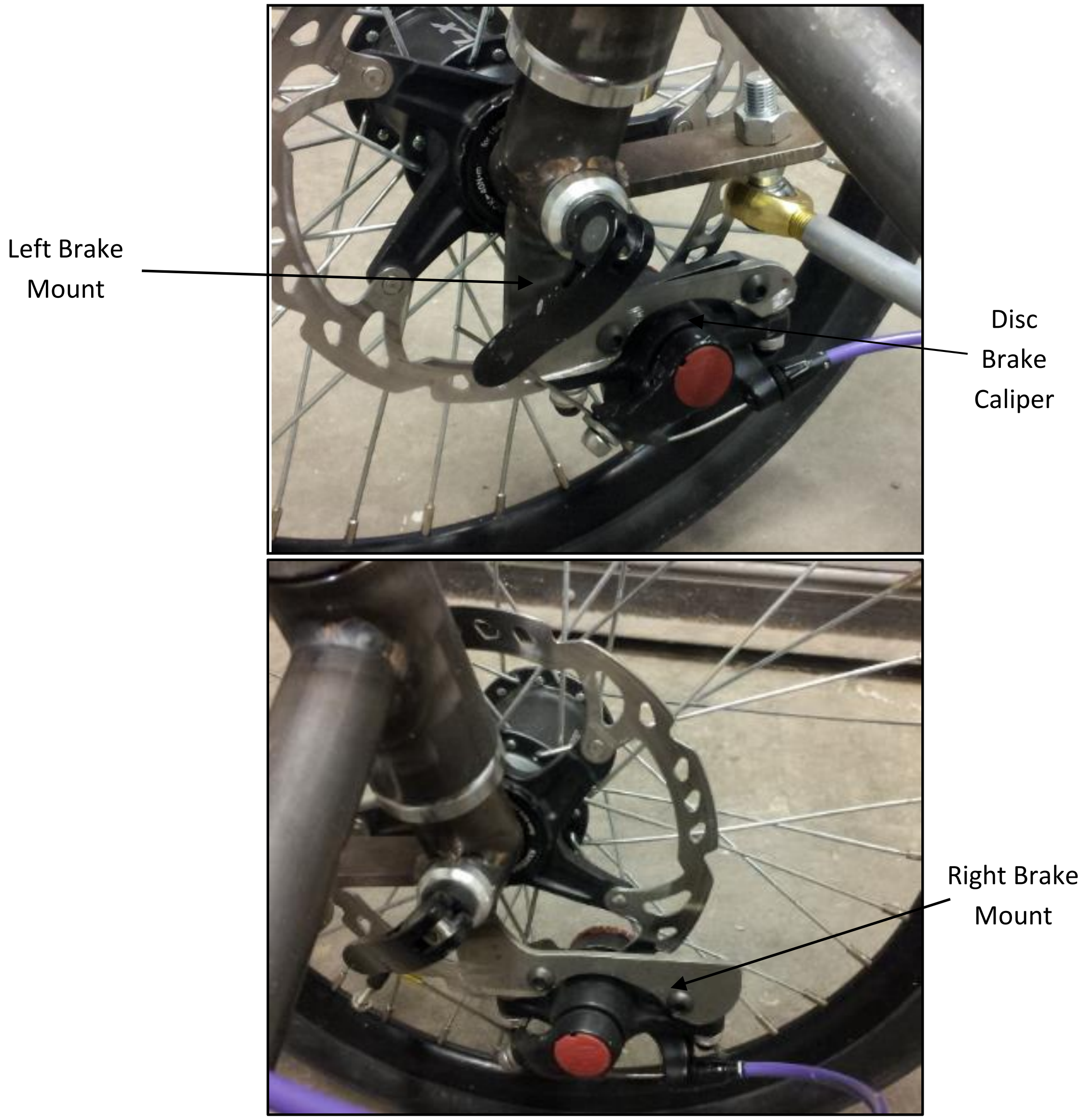

Figure 16: *Disc brake mount for a) left side of vehicle. b) right side of vehicle relative to the front view.*

Since disc brakes are used on the vehicle, mounts had to be fabricated to hold the disc brake calipers in place. The disc brake calipers are placed on each side of the vehicle and are identical. The initial manufactured brake mounts are made from an 1/8 inch 4130 steel plate and

was welded to bottom of the steering spindle. During performance testing, the brake mounts started bending towards the disc rotor due to the lack of support from the small welded attachment surface.

After many performance tests, a new design for the brake mounts were created to prevent the brake mounts from bending. The new design included finite element analysis testing to show that the brake mounts will not yield, then deforming plastically and result in bending. The new design of the brake mounts are shown below in Figure 17, where gussets are added along the top of each side of the brake mount and provide a greater weld contact area between the brake mount and the steering spindle. In addition, a long, thicker gusset is added to the top edge of the brake mount to distribute the stress over a larger surface than the 1/8 inch thick top edge.

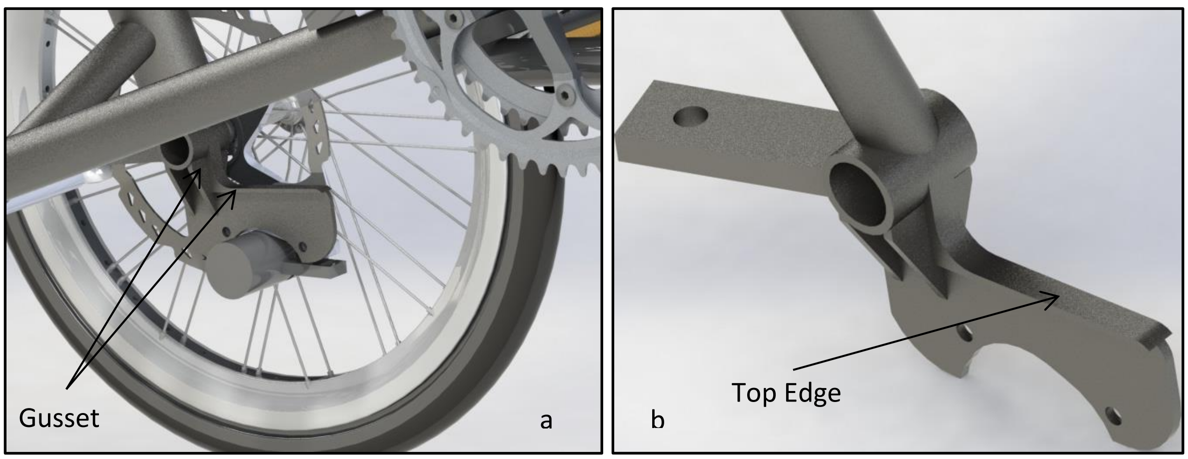

Figure 17: *a) New design of the brake mounts with gussets implemented onto the vehicle. b) Isometric view of mount brake showing a greater thickness than the 1/8 inch top edge of the brake mount.*

The finite element analysis testing was computed in ABAQUS using the same properties as the frame material. A pressure load was applied on the inner surfaces of the two holes on the brake mount. This is the area where the bolts of the brake calipers are attached to the brake mounts. The pressure load was applied on the top half of one hole and the bottom half of the other hole. The inner surface of the steering spindle was fixed because this part of the spindle does not move relative to the disc rotor and brake caliper when the vehicle is in motion. The

finite element analysis was simulated using the worst case scenario, where a force of 400lbs was applied to each hole of the brake mount to create a moment. A moment was inputted into the simulation because the brake caliper is bolted onto the brake mount and attempts to maintain an equilibrium force to keep the brake caliper in place when the rider applies the brakes. The 400lbs is comprised of double the vehicle's 50lb weight and a rider of 150lb weight. This was also the worst case scenario because this assumed that only one brake was functioning properly, but there is actually one brake on each side of the vehicle. The result of the finite element analysis simulation is shown below in Figure 18.

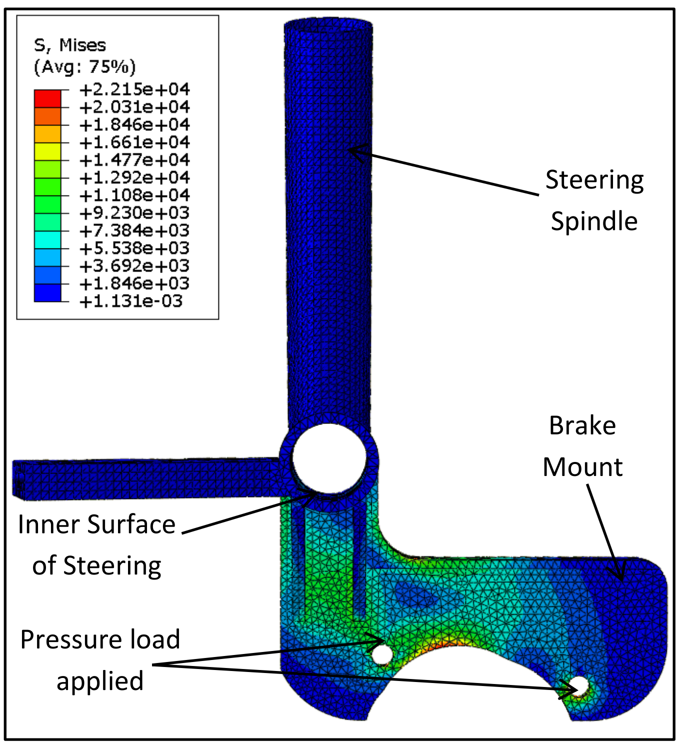

Figure 18: *Result of the finite element analysis of the brake mount using the same material properties as the frame simulated in ABAQUS. The inner surface of the steering spindle was fixed as the boundary conditions and included is a von Mises stress plot.*

According to this analysis, the maximum stress which the brake mount is subjected to in this worst case scenario is 22.15 ksi. Since the yield stress of chromoly steel is 63.1 ksi, this resulted in a factor of safety of 2.85 for the brake mount. Since the maximum stress is less than

the yield stress, the brake mount will not plastically deform and bend. Since the right and left brake calipers, brake mount material and design is identical, this is also applicable to the right brake mount. Performance tests were performed once the new brake mounts were attached onto the vehicle and showed no visible deformation.

Since there are two brakes on the vehicle, a dual brake lever is used to control both front brakes. The dual lever functions similarly to a standard brake lever, except it pulls both the left and right brake cables at once. On a standard bicycle with two brake calipers, there is usually a brake lever on each side of the vehicle and the rider shifts by turning their handlebars. However, for this year’s vehicle, the rear shifter that was compatible with the cassette is a shifter that shifts by clicking it. The dual brake lever is used on the vehicle because the rider has to shift the front and rear derailleur. The dual brake lever is located on the left side of the rider because the rear shifter is located on the right side, which is the dominant side for all the riders this year. The use of the dual brake lever allows the rider to use their dominant hand to shift while still allowing the vehicle to brake with the two front wheels. A picture of this year’s vehicle with the dual brake lever is shown below.

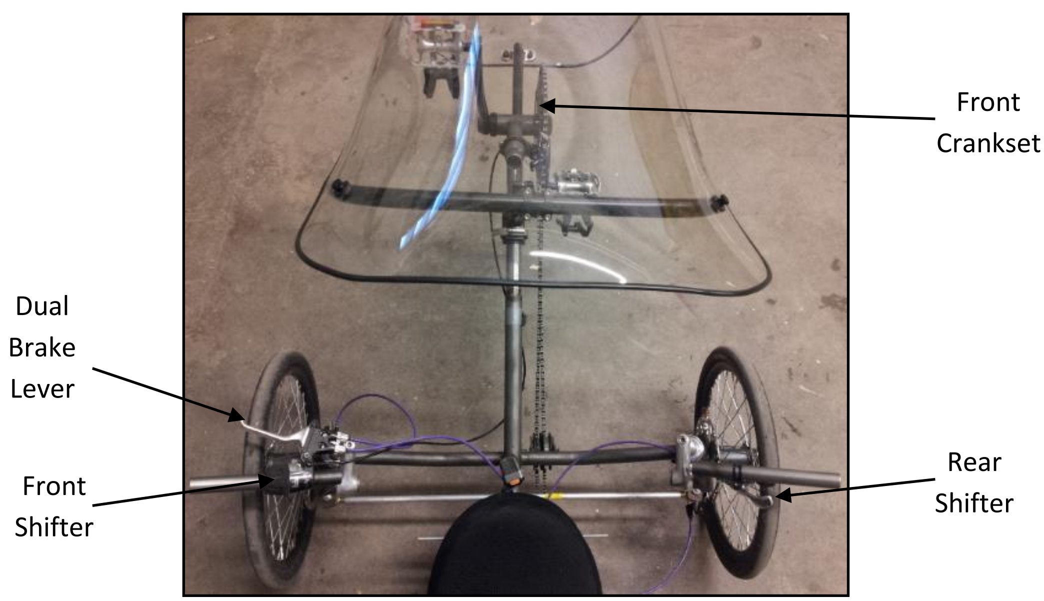

Figure 19: *Dual brake lever and shifting devices as seen from the rider's view.*

FUTURE WORK

The vehicle has to be disassembled to be shipped to the competition site in Orlando, Florida. Once it arrives in Florida, all drivetrain and steering components will have to be reassembled and tested again to ensure that drivetrain and steering pieces function the way they are designed to. All parts that need lubrication will be greased again to eliminate wear on the components. In addition, spare tie rod ends will be purchased because the tie rod ends connect the vehicle’s steering and the vehicle would not be able to function if the tie rod ends break.

CONCLUSION

The goal of Union College’s Human Powered Vehicle team for the 2013-2014 competition is to design a well-engineered vehicle to compete in the Human Powered Vehicle Challenge organized by the American Society of Mechanical Engineers. The vehicle is comprised of a direct steering design driven by a single tie rod, a 39/53 teeth front crankset, a 9 speed 11-34 teeth cassette, two front disc brakes, and an idler that guides the chain. The vehicle will compete in a speed and endurance event, as well as an innovation event. The team will also participate in the design report event. Another goal for this team is to build Union’s reputation in national competitions and bring Union positive publicity. In addition, the team will encourage underclassmen participation and build a solid foundation for future HPVC teams.

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Riders Coming to Competition:

Jesse Coull
Melissa Mansfield

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Student Travel Grant
Engineering Student Competition Funding

APPENDIX

Overall Scoring:

Overall Score Scores from Design Event, Speed Event and Endurance Events scores will be combined to determine the overall standing of the competition.

The formula for combining the scores is:

$$\text{Overall Score} = \sum \text{Event Scores}$$

The maximum event points are:

Competition Event	Maximum Points	
Design Event	30	(20.0)
Male Speed Event	12.5	(6.5)
Female Speed Event	12.5	(7.8)
Innovation Event	20	(8.0)
Endurance Event	25	(15.9)
Total Score	100	(100)

In the case of a tie in the overall point count, the order of finish in the Design Event will determine the overall finish for all vehicles.

Note: The points received from the previous team are shown inside the parentheses.

Gear Inches of Available Cassettes for a 700C Wheel at 70RPM:

700Cx 23mm tire | 53/39 Crankset | with pedaling at | 70 RPM

Rec
11-23 Gear Range 10 Speed Cassette

	39	53
11	94	127.7
12	86.1	117
13	79.5	108
14	73.8	100.3
15	68.9	93.6
16	64.6	87.8
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
23	44.9	61.1
Max,min mph	9.354167	26.60417

11-25 Gear Range 10 Speed Cassette

	39	53
11	94	127.7
12	86.1	117
13	79.5	108
14	73.8	100.3
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
23	44.9	61.1
25	41.3	56.2
Max,min mph	8.604167	26.60417

11-26 Gear Range 10 Speed Cassette

	39	53
11	94	127.7
12	86.1	117
13	79.5	108
14	73.8	100.3
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
23	44.9	61.1
26	39.8	54
Max,min mph	8.291667	26.60417

Rec
11-27 Gear Range 10 Speed Cassette

	39	53
11	94	127.7
12	86.1	117
13	79.5	108
14	73.8	100.3
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
24	43.1	58.5
27	38.3	52
Max,min mph	7.979167	26.60417

Rec
11-28 Gear Range 10 Speed Cassette

	39	53
11	94	127.7
12	86.1	117
13	79.5	108
14	73.8	100.3
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
24	43.1	58.5
28	36.9	50.2
Max,min mph	7.6875	26.60417

Rec
11-30 Gear Range 8 Speed Cassette

	39	53
11	94	127.7
12	86.1	117
14	73.8	100.3
16	64.6	87.8
18	57.4	78
21	49.2	66.9
26	39.8	54
30	34.5	46.8
Max,min mph	7.1875	26.60417

Rec
11-32 Gear Range 8 Speed Cassette

	39	53
11	94	127.7
13	79.5	108
15	68.9	93.6
18	57.4	78
21	49.2	66.9
24	43.1	58.5
28	36.9	50.2
32	32.3	43.9
Max,min mph	6.729167	26.60417

11-34 Gear Range 9 Speed Cassette

	39	53
11	94	127.7
13	79.5	108
15	68.9	93.6
17	60.8	82.6
20	51.7	70.2
23	44.9	61.1
26	39.8	54
30	34.5	46.8
34	30.4	41.3
Max,min mph	6.333333	26.60417

11-36 Gear Range 10 Speed Cassette

	39	53
11	94	127.7
13	79.5	108
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
24	43.1	58.5
28	36.9	50.2
32	32.3	43.9
36	28.7	39
Max,min mph	5.979167	26.60417

MAX SPEED/Most Teeth		56
	11	134.9
	13	114.2
	15	98.9
	17	87.3
	20	74.2
	23	64.5
	26	57.1
	30	49.5
	34	43.6
Max,min mph	9.083333	28.10417

Rec
12-23 Gear Range 8 Speed Cassette

	39	53
12	86.1	117
13	79.5	108
14	73.8	100.3
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
23	44.9	61.1
Max,min mph	9.354167	24.375

Rec
12-26 Gear Range 8 Speed Cassette

	39	53
12	86.1	117
13	79.5	108
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
23	44.9	61.1
26	39.8	54
Max,min mph	8.291667	24.375

12-30 Gear Range 10 Speed Cassette

	39	53
12	86.1	117
13	79.5	108
14	73.8	100.3
15	68.9	93.6
17	60.8	82.6
19	54.4	73.9
21	49.2	66.9
24	43.1	58.5
27	38.3	52
30	34.5	46.8
Max,min mph	7.1875	24.375

TerraTrike

	30	42
11	94	127.7
12	86.1	117
14	73.8	100.3
16	64.6	87.8
18	57.4	78
21	49.2	66.9
24	43.1	58.5
28	36.9	50.2
32	32.3	43.9
Max,min mph	6.729167	26.60417

11-34 Gear Range 9 Speed Cassette

	26	36	48
11	62.6	86.7	115.6
13	53	73.4	97.8
15	45.9	63.6	84.8
17	40.5	56.1	74.8
20	34.5	47.7	63.6
23	30	41.5	55.3
26	26.5	36.7	48.9
30	23	31.8	42.4
34	20.3	28.1	37.4
Max,min mph	4.229167		24.08333

gear inches * cadence / 336 = mph

Note: The purple represents possible cassettes and the green represents the chosen cassette.

Speeding Gear Optimization and Incline Gear Optimization:

Speeding Gear Optimization with 100RPM Cadence, 0.938HP Power Output, Rolling Resistance, and Drag Coefficient

Speeding Gear Calculations

Data Used:

From McCartney Paper: $Power\ Output\ for\ 100RPM\ (P) = 700W\left(\frac{1hp}{746W}\right) = 0.938hp$ (average power over 30s at $\omega_{pedal} = 100RPM$) which corresponds to First Class rider in HPV Performance Chart

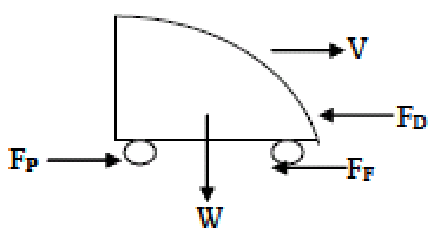
From HPV Performance Chart (for recumbent)
$C_D = 0.77$ (drag coefficient)
$A = 3.8\ ft^2$ (frontal area)
$C_DA = 2.9ft^2$ (effective frontal area)
$C_R = 0.005$ (rolling resistance)

Other useful facts:
For a racing bike using 700C wheels: $D_{tire} = 622mm\ or\ 2.04ft$ (diameter of tire)
Chains are 95.99% efficient → Assume no losses in chain

From ideal gas law: for air
$pv = mRT \rightarrow \frac{p}{RT} = \frac{m}{v} = \rho$ where p is the pressure, v is the specific volume, m is the mass, R is the universal gas constant, T is the temperature, and ρ is the density.
Where for air $= 0.3704\frac{psia\ ft^3}{lbm°R}$, $= 70°F + 460 = 530°R$, $p = 14.7psia$
$\therefore \rho_{air} = \frac{14.7psia}{\left(0.3704\frac{psia\ ft^3}{lbm°R}\right)(530°R)} = 0.0749\frac{lbm}{ft^3}$

Find torque put out by the rider: $T_{in,pedals} = \frac{P}{\omega_{pedals}} = \frac{(0.938hp)\left(516\frac{ft\ lb}{s}\right)}{\left(100\frac{rev}{min}\right)\left(\frac{1\ min}{60s}\right)\left(\frac{2\pi\ rad}{1\ rev}\right)} = 49.3ft\ lb$
Calculate the required torque at the wheel: Assume constant velocity, weight = 220lbs: 70lb for vehicle, 150lb person

Free Body Diagram:



V = velocity
F_D = drag force
F_F = friction force (rolling resistance)
F_P = propulsion force
W = weight

$F_P = F_D + F_F$
$= \frac{1}{2}\rho_{air}AV^2C_D + C_RW$
$= \frac{1}{2}\left(0.0749\frac{lbm}{ft^3}\right)\left(\frac{slug}{32.2lbm}\right)(3.8ft^2)V^2(0.77)\left(\frac{lbf\ s^2}{slug\ ft}\right) + (0.005)(220lbf)$
$F_P = 0.0034V^2 + 1.1\ lb$ units of V: $\frac{ft}{s}$

$\therefore$ Torque at the wheel $(T_{out,rear\ wheel}) = F_P r_{tire} = (0.0034V^2 + 1.1)R_{tire}\ ft\ lb$
units of R_{tire}: ft

Calculate attainable velocity:

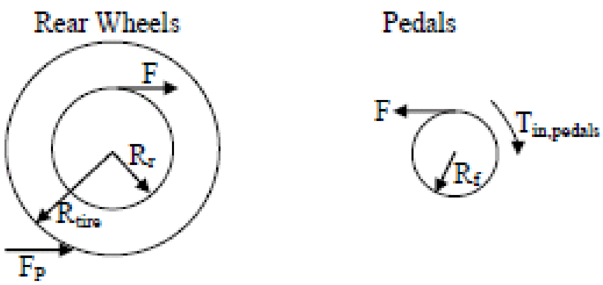
$T_{in,pedals}\omega_{in,pedals} = T_{out,rear\ wheel}\omega_{out,rear\ wheel}$ $\qquad \omega = \frac{rad}{s} = \frac{V}{r_{tire}} = \frac{ft/s}{ft} = \frac{1}{s}$

$$(49.3 ft\ lb)\left(100\frac{rev}{min}\right)\left(\frac{1\ min}{60s}\right)\left(\frac{2\pi\ rad}{rev}\right) = (0.0034V^2 + 1.1)R_{tire}\left(\frac{V}{R_{tire}}\right)$$

$$516.3\frac{ft\ lb}{s} = 0.0034V^3 + 1.1V$$

$$\therefore V = 51.3\frac{ft}{s} = 35mph$$ → Consistent with HPV Performance Chart

Calculate the associated gear ratio:



where F = force in chain
R_r = radius of rear sprocket
R_f = radius of front sprocket

$F_P R_{tire} = F R_r$ $\qquad$ $T_{in,pedals} = F R_f$

$\frac{F_P R_{tire}}{R_r} = F$ $\qquad$ $F = \frac{T_{in,pedals}}{R_f}$

$$\frac{F_P R_{tire}}{T_{in,pedals}} = \frac{R_r}{R_f} = Gear\ Ratio$$

Note that torque at the rear wheel:

$$T_{out,rear\ wheel} = F_P R_{tire} = (0.0034V^2 + 1.1)R_{tire}$$
$$= \left(0.0034\left(51.3\frac{ft}{s}\right)^2 + 1.1\right)R_{tire} = 10.05R_{tire}\ ft\ lb$$
$$\therefore \frac{R_r}{R_f} = \frac{10.05R_{tire}\ ft\ lb}{49.3\ ft\ lb}$$

Finally translating this into bicyclist terms:

$Gear\ Inches = (Diameter\ of\ drive\ wheel)\left(\frac{N_{t,front}}{N_{t,rear}}\right)$ $\qquad$ where N_t = number of teeth in front or rear sprocket

Note: number of teeth is proportional to radii of sprockets

$$\therefore Gear\ Inches = (2R_{tire})\left(\frac{R_f}{R_r}\right) = (2R_{tire})\left(\frac{49.3}{10.05R_{tire}}\right)\left(\frac{12\ in}{1\ ft}\right) = 117.7\ inches$$

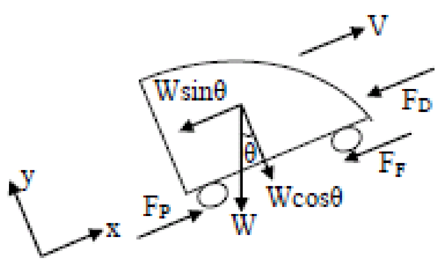
Note: result is independent of radii of the tire.

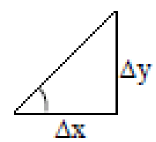
Incline Gear Optimization with 70RPM Cadence, 0.4HP Power Output, Rolling Resistance, and Drag Coefficient

Uphill Gear Calculations:

Using the same data and assumptions from speeding calculations, except the incline is at a 5% gradient and from Human Power Output Curve: $Power\ Output\ for\ 70RPM\ (P) = 0.4hp$

Free body diagram of vehicle at 5% incline:





$$\frac{\Delta y}{\Delta x} = tan\theta = 0.05$$
$$\theta = 2.86°$$

V = velocity
F_D = drag force
F_F = friction force (rolling resistance)
F_P = propulsion force
W = weight
θ = angle at 5% gradient

$$F_P = F_D + Wsin\theta + F_F$$
$$= \rho_{air} A V^2 C_D + Wsin\theta + C_R Wcos\theta$$
$$= \frac{1}{2}\left(0.0749\frac{lbm}{ft^3}\right)\left(\frac{slug}{32.2lbm}\right)(3.8ft^2)V^2(0.77)\left(\frac{lbf\ s^2}{slug\ ft}\right) + (220lb)(\sin 2.86°) + (0.005)(220lbf)(\cos 2.86°)$$
$$F_P = 0.0034V^2 + 12.08\ lb \quad \text{units of V: } \frac{ft}{s}$$

Using the same equations from the speeding calculations to find the torque put out by the rider:

$$T_{in,pedals} = \frac{P}{\omega_{pedals}} = \frac{(0.4hp)\left(516\frac{ft\ lb}{s}\right)}{\left(70\frac{rev}{min}\right)\left(\frac{1\ min}{60s}\right)\left(\frac{2\pi\ rad}{1\ rev}\right)} = 28.2ft\ lb$$

Thus, the attainable velocity is $V = 15.9\frac{ft}{s} = 10.9mph$

Using this result and the relation found from the speeding calculation for the associated gear ratio:

$$\frac{F_P R_{tire}}{T_{in,pedals}} = \frac{12.9\ R_{tire}\ ft\ lb}{28.2\ ft\ lb} = \frac{R_r}{R_f} = Gear\ Ratio$$

The associated gear inches are:

$$Gear\ Inches = (2R_{tire})\left(\frac{R_f}{R_r}\right) = (2R_{tire})\left(\frac{28.2}{12.9R_{tire}}\right)\left(\frac{12\ in}{1\ ft}\right) = 52.2\ inches$$

Note: result is independent of radii of the tire.

Top Speed Calculation of Previous Team Using Internal Gear Hub:

Model Number	SG-8R31
Series	NEXUS
Speeds	8.0
Silent Clutch	yes
Gear Change Support	yes
Shifting Power Modulator	yes
Driving Efficiency	Standard
Shifting Lever	SB/SL/ST-S500)
Coaster Brake	-
Roller Brake Compatible	yes
Dust Cap for V-Brake	-
Disc Brake Rotor Mount	-
Hub Shell Material	aluminum
Hub Shell Finish	painted
Gear Ratio Total Difference	3.07
Gear Ratio 1	0.527
Gear Ratio 2	0.644
Gear Ratio 3	0.748
Gear Ratio 4	0.851
Gear Ratio 5	1.0
Gear Ratio 6	1.223
Gear Ratio 7	1.419
Gear Ratio 8	1.615
Gear Ratio 9	
Gear Ratio 10	
Gear Ratio 11	
Axle Length	132mm

Calcualtions for Previous Team's Gears
Data Used:
Internal gear hub-Shimano Nexus 8 Speeds Model SG-8R31
Maximum gear ratio: 1.615
Number of teeth on front sprocket used: 44
Number of teeth on rear sprocket used: 22
Wheel size used: 27 inches
Assume same speeding cadence of 100RPM

To find gear inches:

$$Gear\ Inches = (Diameter\ of\ drive\ wheel)\left(\frac{N_{t,front}}{N_{t,rear}}\right)$$

$$= (27\ in)\left(\frac{44}{22}\right)(1.615) = 87.2\ inches$$

Thus the associated velocity in miles per hour:

$$Speed\ in\ MPH = (Cadence)(Gear\ Inches)(\pi)$$

$$= \left(100\frac{rev}{min}\right)(87.2\ in)\left(\frac{\pi}{rev}\right)\left(\frac{60\ min}{1\ hr}\right)\left(\frac{1\ mi}{63360\ in}\right) = 25.9\ mph$$

Note: 25.9mph is the maximum attainable speed which corresponded to maximum speed of previous team.