



<http://go.asme.org/HPVC>

Vehicle Description Form

(Form 6)

Updated 12/3/13

Human Powered Vehicle Challenge

Competition Location: Las Vegas, NV

Competition Date: March 17-19, 2017

This required document for all teams is to be incorporated in to your Design Report. Please Observe Your Due Dates; see the ASME HPVC for due dates.

Vehicle Description

School name: Northern Arizona University

Vehicle name: Cerberus

Vehicle number: 4

Vehicle configuration: **Semi-recumbent**

Frame material: **6061-T6 Aluminum**

Fairing material(s): **Carbon Fiber, Gillcore HK Honeycom**

Number of wheels: **3**

Vehicle Dimensions (*please use in, in³, lbf*)

Length: **94.76 in** _____ Width: **43.08 in**

Height: **40.50 in** Wheelbase: **54.21 in**

Weight Distribution Front **45.34 lbs** Rear **26.93 lbs** Total Weight: **72.27 lbs** _____

Wheel Size Front **20 in** _____ Rear **27.56 in** _____

Frontal area **850 in²**

Steering **Front**

Braking **Front**

Estimated Cd: **.130**

Vehicle history (e.g., has it competed before? where? when?) _____ N/A _____

**Northern Arizona University
2017 ASME HPVC West**



*College of
Engineering, Forestry
& Natural Sciences*

Presents

Cerberus

Vehicle #4

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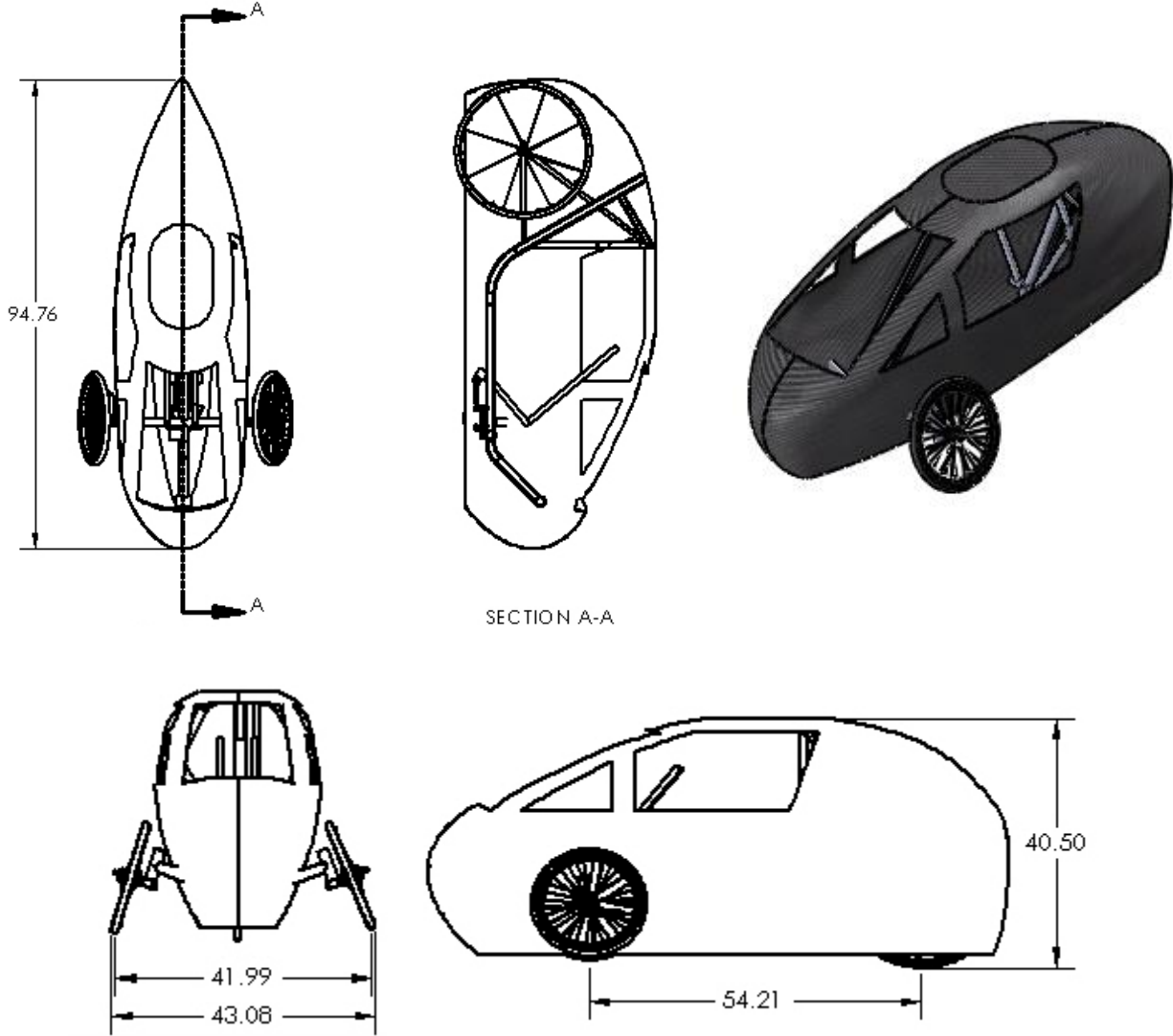
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Four-View Drawing



Abstract

The 2016-2017 Northern Arizona University's Human Powered Vehicle (HPV) capstone team set out to design and build a sustainable, innovative, and competition-worthy vehicle to compete at the 2017 HPV Competition. The vehicle created, Cerberus, is a cost-effective, aerodynamic, and rider-friendly alternative to typical automobiles. Cerberus was designed and built to abide by the ASME established competition requirements as well as the team's own goals meant to improve upon past Northern Arizona University HPV teams.

The project began at the beginning of the 2016-2017 academic year and spanned until the competition date in March of 2017. All design and manufacturing took place in this period, as well as all analyses, preliminary testing, and the development of final testing procedures. The types of analyses completed during this time include structural, aerodynamic, CO2 life cycle, cost, and deformations of the Rollover Protection System. These analyses confirmed the team's design choices and so Cerberus was built accordingly. To further check the vehicle's final designs, physical testing based on the analyses will be performed in order to ensure Cerberus will compete well at the competition.

Cerberus' final design consists of a 6061-T6 Aluminum frame that is partially curved, reducing welding joints and thus increasing the structural integrity. Three wheels will attach to the frame where the wheelbase and track for these wheels are 54.21 and 41.99 inches, respectfully. Using a dual actuated bell crank to steer, a chain driven system and an optimized seat angle of 120°, Cerberus is expected to ride efficiently. The full fairing made of Carbon Fiber and Gillcore HK Honeycomb streamlined airfoil shaped design will provide an aerodynamically smooth ride. As a final safety measure, Cerberus will contain a removable electrochromic windshield to block UV rays and also acts as a theft deterrent.

Table of Contents

1. Design Methodology	1	2.3.2 Alternatives Evaluated	16
1.1 Design Objective	1	2.4 Cost Analysis	17
1.2 Background Research.....	1	2.5 Product Energy/CO2 Life Cycle Analysis	18
1.3 Prior Work.....	2	2.6 Other Analyses.....	19
1.4 Organizational Timeline	2	2.6.1 Steering.....	19
1.5 Design Specifications	2	2.6.2 Drivetrain	20
1.6 Concept Development and Selections Methods.....	4	3. Testing	21
1.6.1 Frame	4	3.1 Rollover/Side Protection System	21
1.6.2 Steering	5	3.1.1 Top Load Testing Setup.....	21
1.6.3 Drivetrain	5	3.1.2 Top Load Testing Results	22
1.6.4 Fairing.....	6	3.1.3 Side Load Testing Setup	22
1.7 Structured Design Methods	6	3.1.4 Side Load Testing Results.....	22
1.8 Description	7	3.2 Developmental Testing.....	22
1.8.1 Frame Description (Seat Description).....	7	3.2.1 Frame.....	22
1.8.2 Steering Description.....	7	3.3 Performance Testing.....	23
1.8.3 Drivetrain Description	8	4. Safety	23
1.8.4 Fairing.....	9	4.1 Rollover/Side Protection System	23
1.9 Innovation	10	4.2 Seat Belt.....	23
2. Analysis	11	4.3 Steering System	23
2.1 Rollover/Side Protection System	11	4.4 Braking System	24
2.1.1 Top Load Modeling	11	4.5 Sharp Edges, Protrusions, Pinch Points .	24
2.1.2 Top Load Results	11	4.6 Manufacturing Safety	24
2.1.3 Side Load Modeling.....	12	4.7 Rider Field of View	24
2.1.4 Side Load Results	12	4.8 Safety Accessories	25
2.2 Structural Analytical Calculations	13	4.9 Additional Safety Features.....	25
2.2.1 Frame	13	5. Conclusion.....	25
2.2.2 Steering	14	5.1 Comparison.....	25
2.3 Aerodynamics	16	5.2 Evaluation	26
2.3.1 Aero Device Incorporated	16	5.3 Recommendations.....	26
		6. References	27
		7. Appendices.....	29

Appendix A29
Appendix B30
Appendix C32

1. Design Methodology

1.1 Design Objective

Northern Arizona University's (NAU) Human Powered Vehicle Team designed and built Cerberus throughout the 2016-2017 academic year. The team decided to assign each member to be the lead of a subsection of the vehicle: Frame, Fairing, Ergonomics, Drivetrain, Steering, and Innovation. The team's mission statement followed throughout the design and manufacturing of Cerberus is as follows:

“Design and build a sustainable vehicle that will compete at the 2017 ASME Human Powered Vehicle Competition. The vehicle will be able to promote an inclusive learning environment and encourage participation in future competitions from younger students.”

1.2 Background Research

To ensure the success of the project, research was conducted in the following manner: through observation at last year's HPVC, online searches of informative rules, and speculation of past team's design reports.

Past competition design reports for vehicles from The University of Akron, Rose-Hulman, Utah State, University of Southern California, and NAU's "Pulaski" were read over by the team and broken down.

Several of NAU's previous HPVs are still found on campus so preliminary research was performed. Team members practiced steering, pedaling, and braking using these vehicles. Through this research, the team was able to determine where improvements could be made such as seat positioning, welding locations, and other manufacturing changes. This opportunity helped guide the team in creating a more rider friendly vehicle while minimizing manufacturing mistakes.

The team met with the "Pulaski" team members multiple times allowing the team a better understanding of the project and timeline to be set. These meetings proved to be invaluable as any question pertaining to the competition such as the presentation, report, or races were answered immediately.

More in-depth research was performed once designs were finalized within each subsection which is discussed in the final design description section of this report.

1.3 Prior Work

All work was performed by the team during the 2016-2017 academic school year. Prior vehicles production methods and design methods were evaluated; however, all designs, data, and analysis are original to the vehicle. Some testing fixtures and manufacturing fixtures from past years were reused, however the data collected was different and served to meet different goals.

1.4 Organizational Timeline

Below illustrates the team schedule used to maintain completion on time (Figure 1). The design portion incorporates concept generation, concept selection, and analytical analysis of the concepts selected.

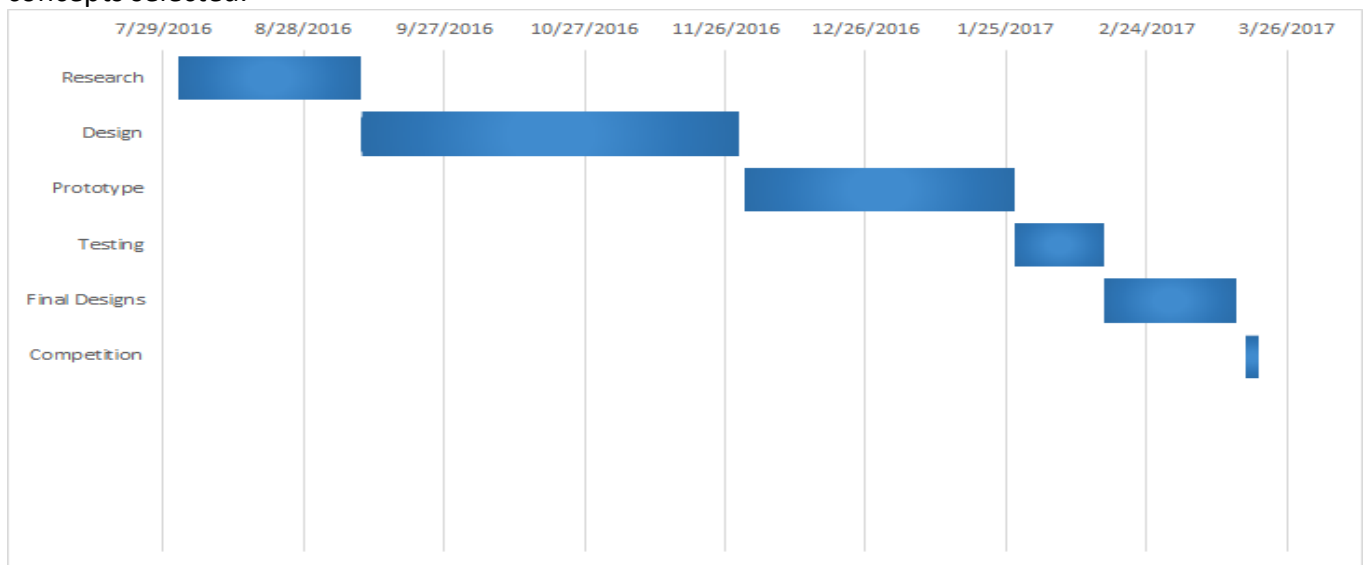


Figure 1: Gantt Chart

1.5 Design Specifications

To compete, the ASME constraints were followed strictly while the team also created requirements based on past competition vehicles, especially those from NAU that could be closely examined during the vehicle’s design phase.

Table 1: ASME 2017 HPVC Constraints

ASME HPVC Constraints	
Performance	<ul style="list-style-type: none"> • Stop from a speed of 15 mph (25 km/hr) in a distance of 20 ft (6.0 m) • Turn within a 26 ft (8.0 m) radius • Demonstrates stability by traveling for 100 ft (30 m) in a straight line at a speed of 3-5 mph (5 to 8 km/hr)

Safety	<ul style="list-style-type: none"> • Braking system with properly designed brakes on the front most wheel of vehicle at a minimum • Include a Rollover Protection System (RPS) that must: <ul style="list-style-type: none"> ○ Protect all drivers in the vehicle in the event of an accident ○ Absorb sufficient energy in a severe accident to minimize risk of injury ○ Prevent significant body contact with the ground in the event of a fall or rollover ○ Provide adequate abrasion resistance to protect against sliding ○ Deform less than 2 in (5.1 cm) under a 600 lb (2670 N) top load ○ Deform less than 1.5 in (3.8 cm) under a 300 lb (1330 N) side load • Provide rider with a 180 field of view • A safety harness attached to the RPS • A functioning bell/horn • 300 lumen or greater white headlight • 10 lumen or greater red taillight • Side reflectors • Rear View Mirror(s) • An additional safety feature
Rider	<ul style="list-style-type: none"> • Minimum of 30 minutes of riding experience in vehicle prior to competition

Table 2: 2016-2017 NAU Team's Constraints

NAU Cerberus Requirements	
Frame	<ul style="list-style-type: none"> • Overall length of less than 65 inches • Width is equal to or less than 36 inches • Fit rider height (from shortest to tallest) of 65-72 inches
Overall Vehicle	<ul style="list-style-type: none"> • Vehicle will have a 10-foot turning radius • Achieve a top speed of 40 mph • Weight is less than 90 lbs
Cost	<ul style="list-style-type: none"> • Maintain a budget of \$7,500 or less

The team's self-imposed frame constraints were developed to ensure that all team members would be able to operate the vehicle. The overall vehicles constraints pertain to the goal of improving upon previous NAU HPVs. Finally, the single cost constraint is a goal based off budgeting research performed using data from the 2016 HPV competition's most successful vehicles.

1.6 Concept Development and Selections Methods

1.6.1 Frame

Multiple components of the frame were considered for the final decision of the frame. The primary considerations consisted of material selection, vehicle layout, and tubing cross sectional geometry.

The material selection for the frame consisted of material that could be provided by the team's material sponsor, Industrial Metal Supply. For tubing, Aluminum 6061 - T6, 4130 Seamless Steel, and a hybrid of Aluminum 6061 - T6 with carbon fiber wrapped around critical locations of the frame were considered. NAU's 2015-2016 HPV used 4130 Seamless Steel as the primary material and it featured a front-end weight disadvantage that caused the vehicle to lean downwards. Thus, we decided to choose between bare Aluminum or a hybrid with carbon fiber. While both bare Aluminum 6061 - T6 and a hybrid of Aluminum 6061 - T6 wrapped in carbon fiber provide high strength to weight ratios, the hybrid does not provide an advantage to the critical locations. Therefore, Aluminum 6061 - T6 is used as the primary material for the frame.

Three different designs for the main support beam were considered: straight, partially curved, and fully curved (Figure 2a-c). The curved beams were considered with the thought of reducing the number of joints and the strength increase due to strain hardening. Since being able to manufacture the frame at the machine shop provided by the NAU campus was a main factor, the partially curved beam was chosen because it only required two bends at different locations. Also, a circular cross section was selected for the tubing to ease the process of bending for the support beam and roll cage.

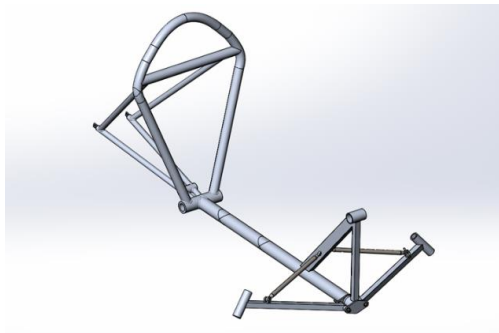


Figure 2a: Straight Frame

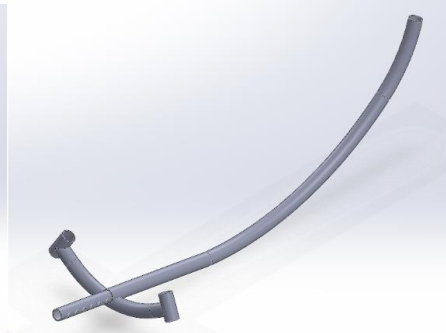


Figure 3b: Completely Curved Frame

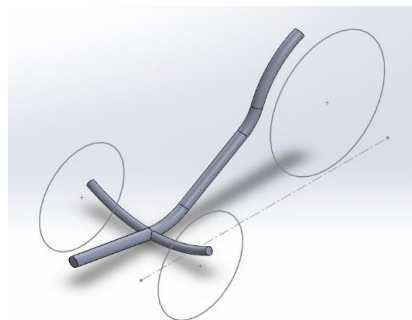


Figure 4c: Partially Curved Frame

1.6.2 Steering

While the team chose to adopt the NAU 2014 HPV team's design for steering, concepts were evaluated based on those that are successful in current cars and trucks. The team considered a cam and sector, rack and pinion, and L-shaped crank actuated steering systems. These concepts can be seen in Figures 3a-c. Many issues arose with manufacturing rack and pinion systems as well as cam and sector systems in house. These issues included finding ways to manufacture gears and produce sufficient housings. Similarly, buying existing rack and pinion systems and modifying it for use in the vehicle would have proved to be difficult to mount and keep stable.

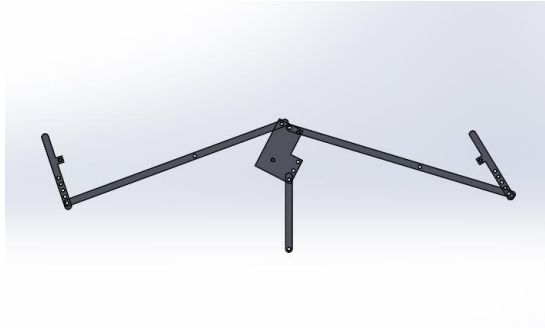


Figure 3a: L-Shaped System

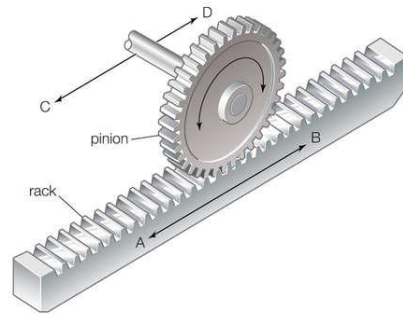


Figure 3b: Rack and Pinion

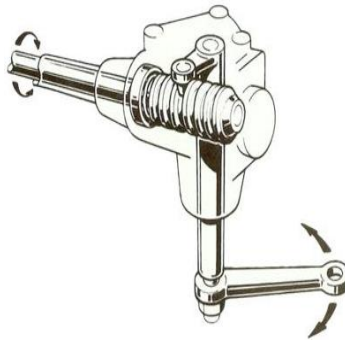


Figure3c: Cam Sector [1]

1.6.3 Drivetrain

The team decided upon three possible systems that would provide durability, simplicity, and innovation to the vehicle; each system excelling in one of these fields. Figure 4a depicts a simple chain driven system. This system allowed for high durability along with being easily maintainable, but a downside to this system is a greater chance in chain drop and the chain slipping into a different gear. This is primarily due to tensioning issues with a long chain. Figure 4b displays a basic belt driven system which proved to be beneficial in chain drop due to the ease of tensioning a belt. The main drawback to this system was the manufacturability being very difficult and time consuming. The third design, which can be seen in Figure 4c, is the front driven system. Due to the nature of this system being very complex, the team decided to choose a different approach, even though it was the lightest and most innovative option.

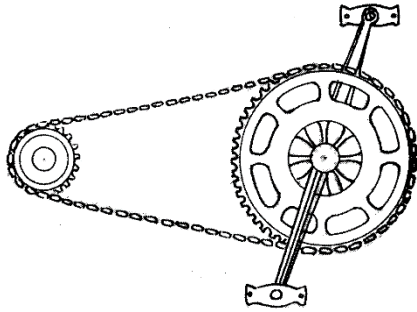


Figure 4a: Chain Drive



Figure 4b: Belt Drive



Figure 4c: Front Drive

1.6.4 Fairing

The full fairing design, seen in Figure 5a encases the vehicle completely. The fairing will be made from carbon fiber to make it light weight. The advantage of a full fairing design is that it has the lowest drag coefficient while the disadvantage of this design is that it is costlier as well as adds weight and has a longer manufacturing time. A front fairing design such as the one seen in Figure 5b uses a fairing just over the front of the vehicle. This has some drag reduction but much lower than that of a full fairing. The advantages of this fairing type are it costs less to manufacture due to requiring less material and therefore, adds less weight to the vehicle. A rear fairing design incorporates a fairing over only the back section of the vehicle as seen in Figure 5c. advantages for this design is the cost is less than that of a full fairing. The time to manufacture is also less. The disadvantages are that is has little drag reduction and adds weight with no help to improve the vehicle.



Figure 5a: Full Fairing

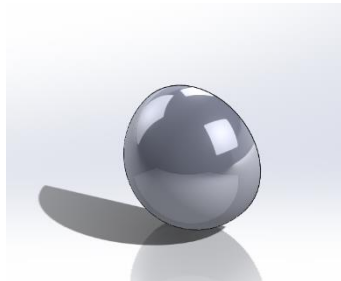


Figure 6b: Front Fairing



Figure 7c: Rear Fairing

1.7 Structured Design Methods

Once concepts were created for each vehicle subsection, they were input into a decision matrix (Appendix A) to determine which concept was the most viable option. The concepts were rated on a 0-5 scale in seven categories: cost, functionality, simplicity, manufacturability, safety, resilience, and reparability. The ratings were then averaged and the two highest ratings of the concepts were highlighted where the highest is denoted in yellow and the second highest is orange. Further research was performed on these two concepts to ensure the best option was chosen. This research is explained later within the description section of this report.

1.8 Description

1.8.1 Frame Description (Seat Description)

The final frame is semi - recumbent with a three-wheeled tadpole configuration. Aluminum 6061 - T6 is used as the primary material of the support beam and Rollover Protection System (RPS). Since the primary member consisted of a 1 ¾-inch outer diameter with a wall thickness of 0.125 inches and needed to be bent, the bent areas had to be annealed because of the T6 stiffness properties of the aluminum. An Oxy - Acetylene torch with multiple Tempilstiks that gradually increased to a temperature of 860 °F were used to anneal the Aluminum. After letting the metal cool overnight with insulation and fiberglass wrapped around it, the material could be bent. Furthermore, the RPS is built from 1 ¼- inch tubing with a wall thickness of 0.125 inches. Due to availability of material from the supplier for a 7/8-inch outer diameter with a thin wall thickness, Aluminum 6063 - T832 is used for the rear triangle. To increase the stiffness of the overall layout of the rear triangle and reduce the stress due to cornering forces, two gusset beams are attached to the upper and bottom portions of the rear triangle.

Once the frame is tested, the team will determine if additional gusset plates need to be added to the frame. Furthermore, vertical dropouts will be used in the final design to allow the rider to easily install the rear tire. The final frame design is shown in Figure 6.

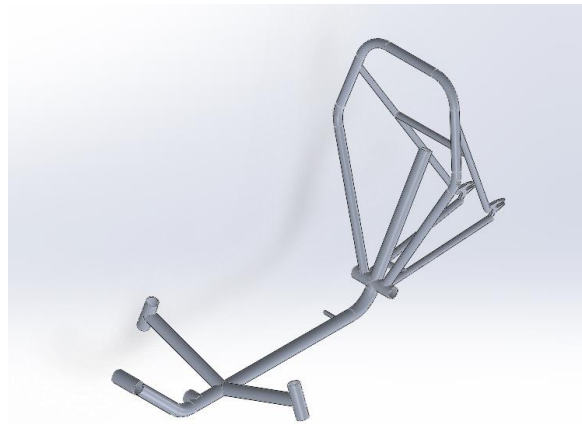


Figure 6: Final Frame Design

The current design of the seat consists of orientating the rider at a 60° angle from horizontal. This angle was determined to be the most ergonomic for the team based on testing that was done a power tap. The seat will be constructed with fiberglass, but the material is subject to change to carbon fiber if allocations allow for it.

1.8.2 Steering Description

The final steering system design is a dual actuated bell crank (Figure 7). The steering knuckle will be manufactured of 1018 Steel while all other components will be made of 6061 Aluminum and heat treated to T-6.

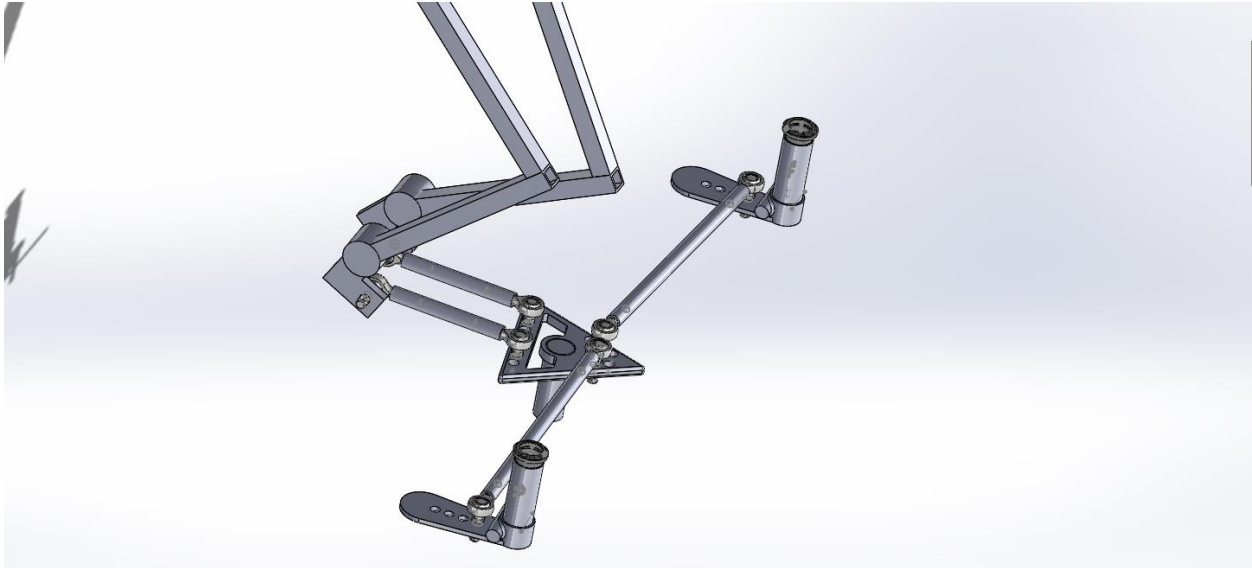


Figure 7: Final Steering Design

This system can accurately provide Ackerman steering angles if all linkages are fully adjustable. The equations to find these steering angles are provided in Figure 8. To make this system fully adjustable, all components have multiple attachment points. With multiple attachment points, this system should be able to achieve turn radii from 10 to 30 ft.

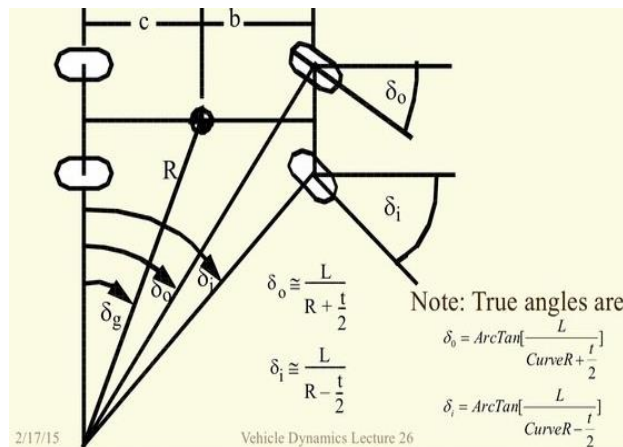


Figure 8: Ackerman Steering Equations

Caster angle was designed into this system to allow the steering to come back to center and remain stable. Likewise, a steep kingpin angle ensures that the steering is responsive and can come back to center quickly. Finally, a slight camber angle was introduced to increase the tire's contact patch while travelling through a corner.

1.8.3 Drivetrain Description

At first, the team selected three systems that each provided crucial aspects that were detrimental to the design of the vehicle: durability, simplicity, and innovation. After weighing these systems against one another, the team decided to approach this challenge with a simple chain driven

system. Due to how common a simple chain system is, the team wanted to attempt electronic shifting. Many beneficial characteristics would come from this, such as eliminating a cable driven rear derailleur and lowering the vehicle weight. Shifting was made simple by implementing only two buttons instead of a mechanically driven shifting apparatus. Overall, maintenance was reduced as well. The shifter the team utilized was SRAM Red Blip controls (Figure 9) along with the corresponding rear derailleur, SRAM Red Rear Derailleur (Figure 10), and a SRAM Junction Box (Figure 11). A major drawback with implementing this system is the overall cost of these products.

The team's next step was to design a path for the chain to travel, which worked around the frame and was safe for the rider to operate the vehicle without hindering his/her performance. To do this the team developed a path beginning at the rider's feet, extending down to an idler gear and then along the bottom of the frame to the rear cassette to power the vehicle. The chain used on the vehicle was a SRAM Red chain specific for an eleven-speed cassette. The cassette used was a XG-1190 Cassette (Figure 12) which incorporates 11 gears.



Figure 9: Blip Controls

Figure 10: Rear Derailleur

Figure 11: Junction Box

Figure 12: XG-1190

1.8.4 Fairing

The vehicle will include a completely enclosed fairing constructed from 3k plain and twill weave carbon fiber along with Gillcore HK kevlar stiffeners. Stiffeners will be located on the front pillars and side walls. A door will be included on the left side of the vehicle for quick and independent egress along with a reinforcement layer to support the panels weight. Front lights will be located on the apex of the vehicle nose along with LED tail lights and adequate reflectors to ensure safety. Aerodynamic properties of the fairing were developed to match a symmetrical airfoil from the topical view of the vehicle in Figure 13. Visibility properties were developed to allow integration of the electrochromic windscreen. Windscreens will be removable and replaceable using Velcro.

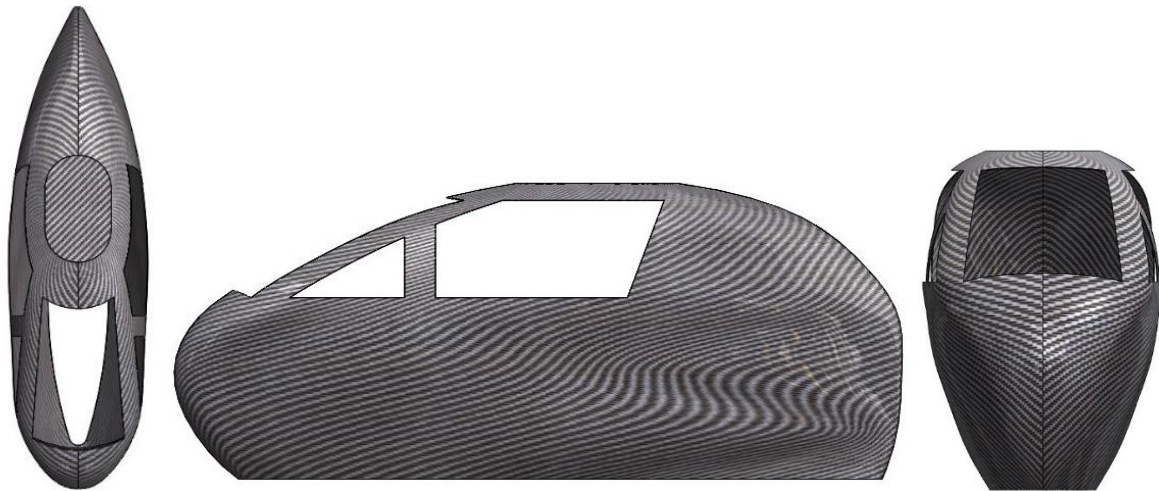


Figure 13: Final Fairing Design

1.9 Innovation

Table 5 demonstrates the different design ideas that were evaluated for the Innovation subsection.

Table 3: Innovation Concepts

Innovation Concepts	
Concept	Description
Onboard Computer	Display that shows power output, RPM, MPH, etc.
Electric Drag Reduction	Fin behind fairing that is powered by electric switch
Hydraulic Braking System	Utilizes hydraulic fluid for braking power
Phone Application for Bicycle	Phone application for bicycle that signals for help and displays rider outputs
Honeycomb System	Honeycomb system applied to the top surface of the back fairing to reduce turbulent flow
Adaptive Material Implementation	Composite capacitor that is actuated by vibrations of tire
Electrochromic Windshield	Electrochromic plastic that turns clear to opaque with electrical current

The electrochromic windshield will be implemented into the vehicle as a safety factor. When the power switch is off, the windshield is opaque and any outside viewers cannot see within the

vehicle. Once the power switch is on, the windshield turns clear and the rider can begin to maneuver the vehicle safely. The electrochromic material is currently being tested and additional details are explained within NAU’s Innovation Report.

2. Analysis

2.1 Rollover/Side Protection System

Table 4: Rollover Protection System Analysis Summary

RPS Loading Condition	Applied Force (lbf)	Allowed Deformation (in)	Actual Deformation (in)
Top	600	2.00	0.68
Side	300	1.50	0.26

2.1.1 Top Load Modeling

To ensure that the Rollover Protection System met the constraints set by ASME, Finite Element Analysis (FEA) was applied with the software ANSYS. First, the frame was suppressed into only a portion of the support beam, the Roll Protection System, RPS brackets, and the rear triangle because these are the components needed to correctly support the RPS in the analysis. Therefore, the hubs for the drivetrain, steering arms, king-pins, and a portion of the support beam were suppressed.

The FEA model consisted of applying fixed supports to the support beam and RPS brackets. A load of 600 lbf (2674 N) at a 12° angle from vertical was placed at the center of the top face of the RPS. The model of the analysis for the frame is shown in Figure 14. After applying these factors, the deformation was calculated for.

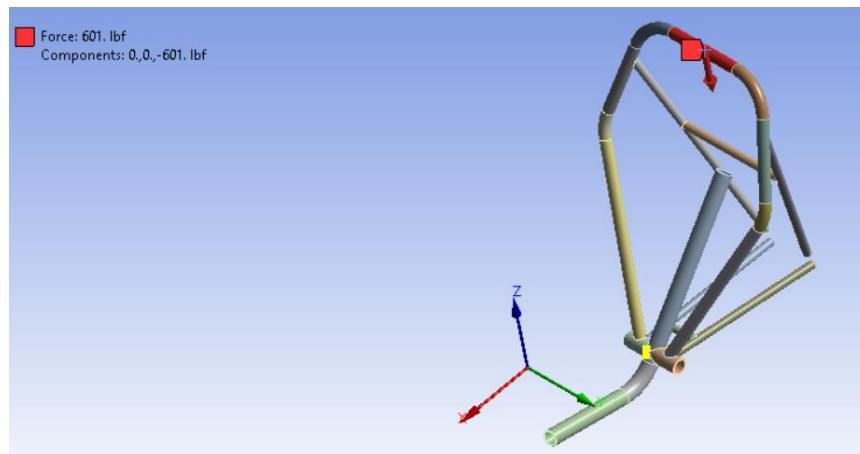


Figure 14: Top Load Model

2.1.2 Top Load Results

The maximum deflection that the RPS can experience per ASME is 2 inches. After running the analysis, the max deflection that the frame experienced was 0.68 inches. This result (Figure 15)

meets the ASME constraint and will be verified through physically testing a prototype of the frame.

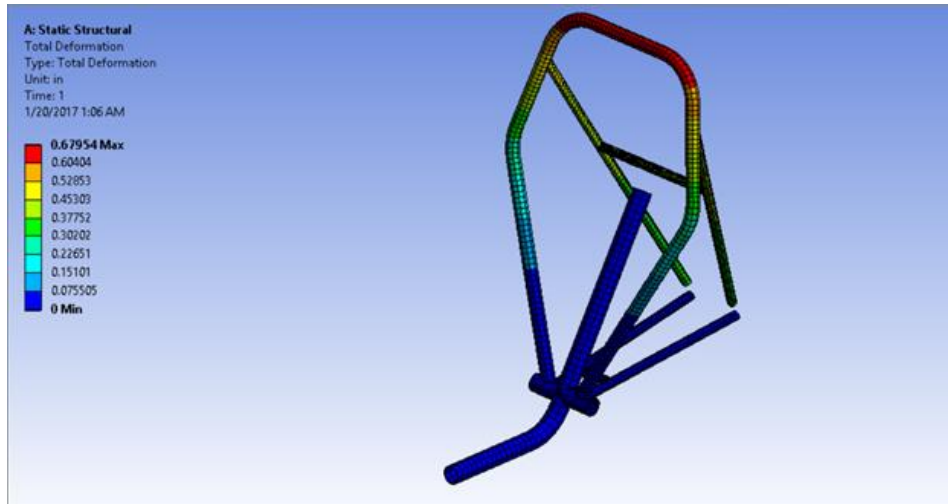


Figure 15: Top Load Deflection Results

2.1.3 Side Load Modeling

The side modeling of the frame consisted of the same setup as the top load model, but a load of 300 lbf (1334 N) was applied to the RPS at shoulder length. The side load setup is demonstrated in Figure 16.

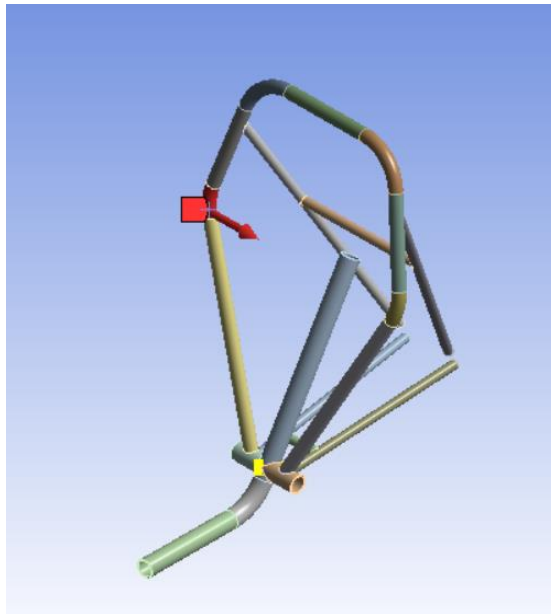


Figure 16: Side Load Model

2.1.4 Side Load Results

The maximum deflection caused by a side load of 1330 N that is allowed by ASME is 1.5 inches. Based on the analysis done, a maximum deflection of 0.26 inches is expected on the RPS (Figure 17). The RPS meets the constraint per the analysis, but the deflection due to the appropriate load will be measured with a proper test setup.

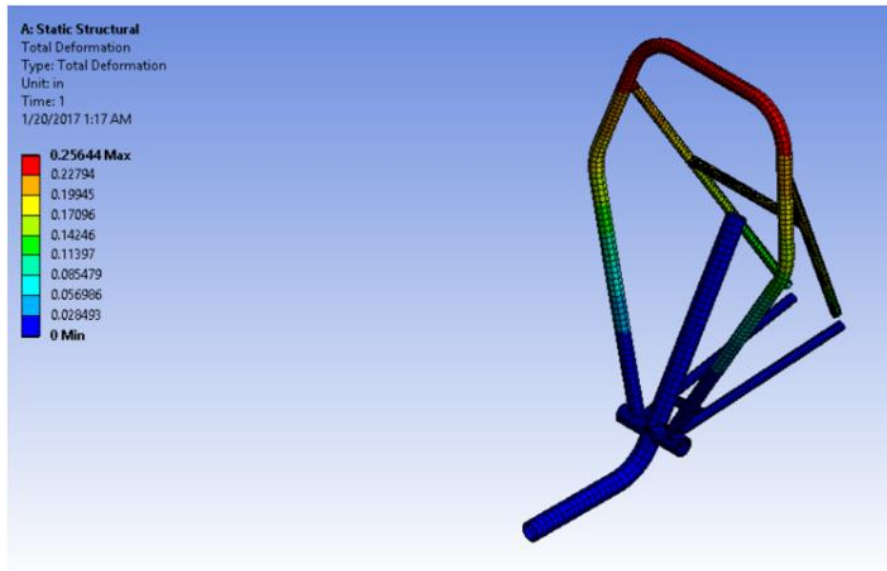


Figure 17: Side Load Analysis Results

2.2 Structural Analytical Calculations

Table 5: Structural Analysis Calculations Summary

Structural Analysis Performed	Testing Performed	Load Applied (lbf)	Max Stress Result (psi)
Frame	ANSYS: Rider Weight	200	945
Linkage	ANSYS: Cornering	2.70	1450
Bell Crank	ANSYS: Cornering	2.70	14503

2.2.1 Frame

Structural analysis was done on the first frame design to ensure that it would not fail due to the certain external loads applied to it. The constraints applied to the model included fixed supports at the king-pins and after applying the constraints, external loads were applied to the frame. The rider's maximum weight (200 lbs.) was applied to the center of mass of the frame and a bending moment was applied to the front section of the support beam due to the force applied against the pedals while pedaling. The maximum stress that the frame experienced was 945 psi (Figure 18).

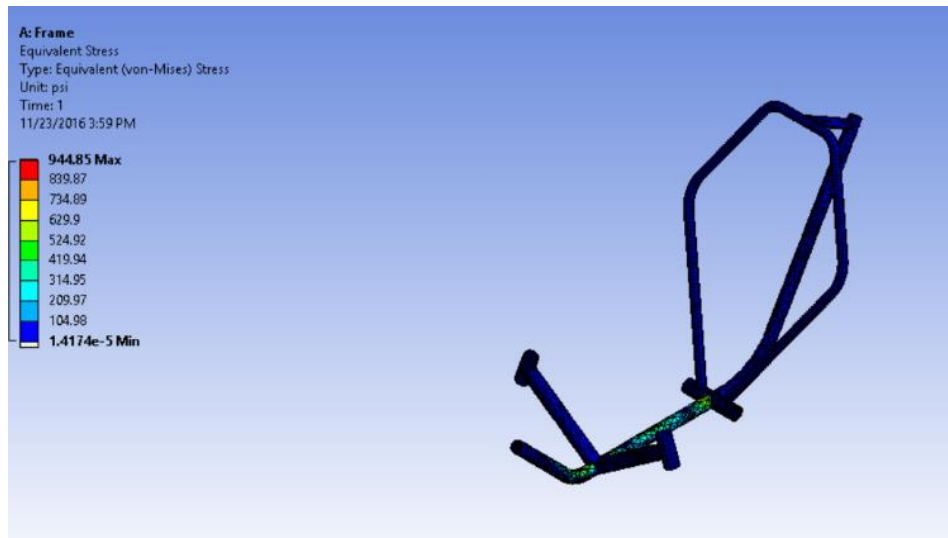


Figure 18: Stress Analysis of First Frame Design

2.2.2 Steering

For the steering system, components were evaluated to determine if certain components would fail during heavy cornering. By looking at Liril D Silvi's lecture slides on steering forces (Figure 19), the team can determine if these forces are too much for certain components [2]. The main force considered was at the steering knuckle. In a fast corner, the wheels would feel the cornering force but it will act at the radius of the tire on the ground attempting to bend the wheel around the steering knuckle. To determine forces in other linkages in the steering system, the team had to determine the self-righting torque acting through a corner resisting the driver. This was found by using pneumatic trail of the wheel and the cornering force.

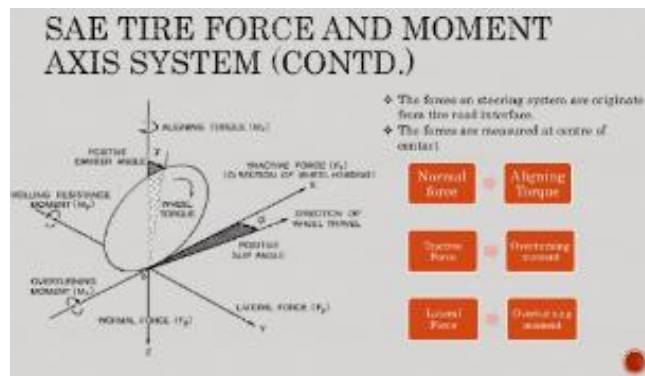


Figure 19: Forces on Tires

To begin modeling, the cornering force was computed using the equation: $F_c = mv^2/R$ which is simply the centripetal acceleration multiplied by the mass of the vehicle. For the steering knuckle test, the team decided that a turn of a ten-foot radius at five mph was considered a hard corner. From this calculation, a total of 12 Newtons acted on the bottom of the wheel, ten inches away from the knuckle. The fixed point for this evaluation was made to be the inside of the spindle where the headset would be installed. Similarly, the forces in the linkages will be the same

cornering force of 12 Newtons but they will act .0267 meters from the contact point due to the flex in the tire.

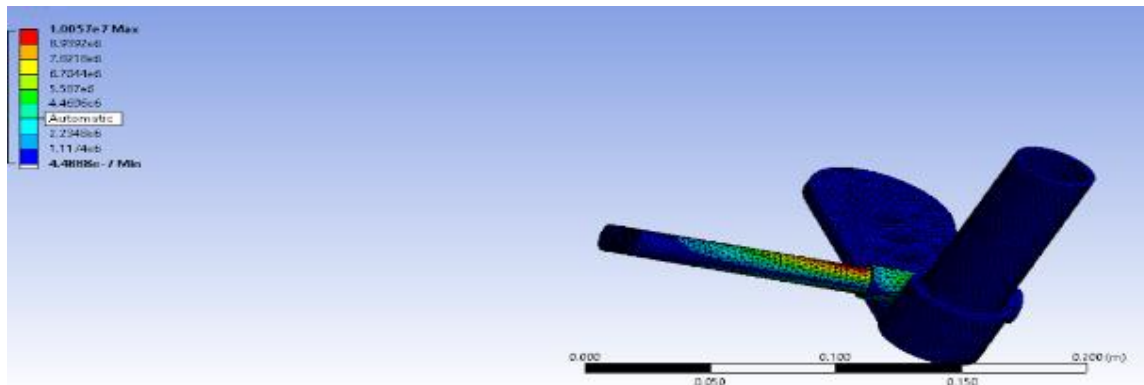


Figure 20: Steering Linkage Analysis Results

From Figure 20 the team can see that the max stress is just over 10 MPa which provides a large factor of safety for one of the most complicated pieces of the steering system. This figure also shows that the max stress is seen on the top and bottom of the axle shaft.

Similarly, from Figure 21 the bell crank only sees just over 100 MPa which gives a factor of safety over 2.5. The team can see from this analysis that the places with highest stresses are near the pivot point for the bell crank.

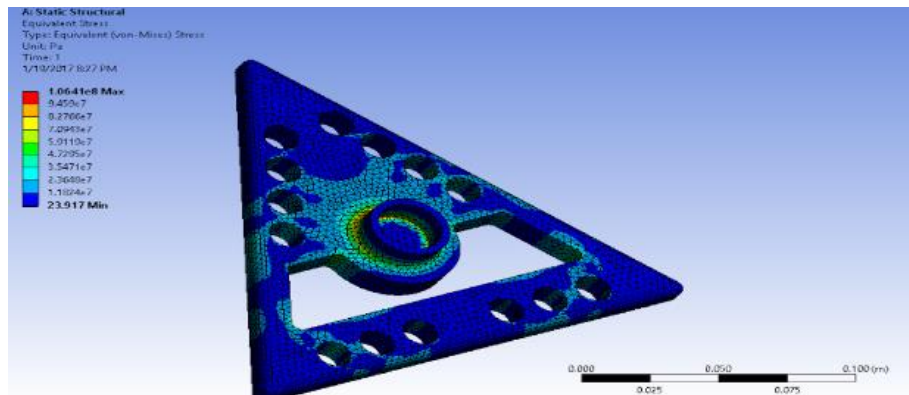


Figure 21: Steering Bell Crank Analysis Results

By employing the use of ANSYS Mechanical, the team could determine where the points of maximum stress would be. Now, the team can take special care in the construction of the steering system to ensure welds at critical stress areas are without flaw.

2.3 Aerodynamics

2.3.1 Aero Device Incorporated

A drag reducing fairing was designed for the vehicle. The fully enclosed composite fairing includes a streamlined airfoil shape as shown in Figure 22 below. The shape is symmetrically balanced and feasible when analyzing its capabilities to fit the shape of the chassis.

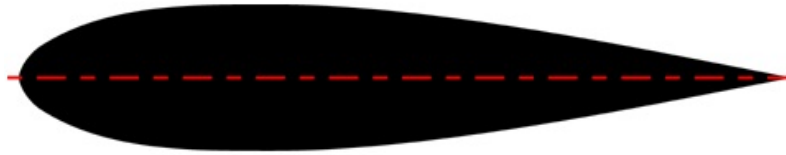


Figure 22: Symmetrical Airfoil [3]

2.3.2 Alternatives Evaluated

Upon the design decision of a full fairing, two aerodynamic shapes were analyzed for their ability to encapsulate the vehicle allowing both drag reduction and security to the rider. Other factors such as visibility, ease of access, weatherproofing, and innovation integration were also considered. Design 1, below, was a bullet shaped fairing with a risen section for rider visibility and was analyzed using ANSYS Fluent due to a geometry incompatibility with SolidWorks Flow Simulation. Design 2 is a streamline airfoil shape at each horizontal cross-section. The results of the analyses are presented in Table 6. Assumptions made for all analyses included air density and pressures in Flagstaff, Arizona. This was controlled to accurately compare with physical wind tunnel data to be completed before the competition.

Table 6: ANSYS Fluent Results

Design #	Coefficient of Drag	Weight (lbs)	Stagnation Pressure (psi)
1	.0175	11.68	1.211e-002
2	.130	17.18	14.74

The original Design 1 analysis yielded impractical drag coefficients, however more accurately represented was the pressure gradient on the vehicle. Design 1 saw a large stagnation pressure at the tip that constituted to a low-pressure section which following the vehicle causing a turbulent wake. The inability to analyze in SolidWorks along with the significantly lower pressure values issued a red flag on the geometry.

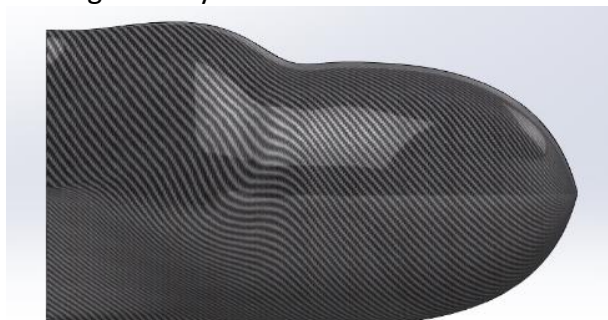


Figure 23: Fairing Design 1

Design 2 as seen in Figure 24 below consisted of a streamline shape modeling a symmetric airfoil. Symmetric airfoils are not only balanced but can change in cross sectional shape to fit nearly any RPS.



Figure 24: Fairing Design 2

Design 2 was analyzed using SolidWorks Flow Analysis. The stagnation pressure shown in Figure 25 shows a much more consistent flow with a pressure gradient that has a range of nearly .25 psi outperforming the analysis of Design 1 as seen below.

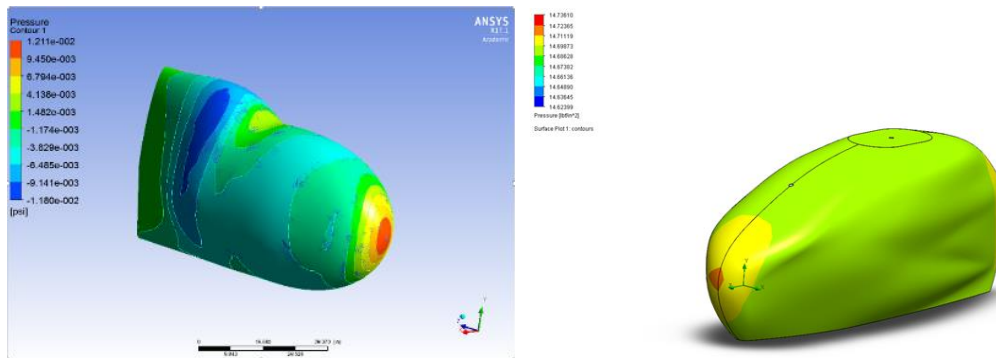


Figure 25: Flow Analysis Results

In conclusion, the Design 2 analysis fir the vehicle criteria more appropriately and yielded more consistent data in comparison to predictions.

2.4 Cost Analysis

After gathering an initial budget of \$7500, creating estimates for each subsection, receiving sponsorships, and beginning construction, a cost analysis was performed to assess how much of each projected budget was remaining. From Table 5 the team can see that steering is over budget while every other subsection is under budget at this point.

Table 7: Cost Analysis

Section	Allocated	Actual Expense
Frame	\$1200	\$116.85
Fairing	\$2100	\$1070.15
Steering	\$550	\$610.80
Innovation	\$650	\$88.64
Drivetrain	\$700	\$159.49
Total	\$5200	\$2045.93

2.5 Product Energy/CO2 Life Cycle Analysis

Currently, keeping the CO2 emissions to a minimum is very important. Already, the vehicle does not directly emit any CO2. However, this does not mean that producing it is carbon neutral. To manufacture the aluminum in the frame, power was required to heat, extrude and cut to finish. Since only 11% of electricity is gathered from sources other than fossil fuels, the team can determine the CO2 equivalent (CO2e) that the vehicle emits strictly from buying and using certain products [5]. Furthermore, machining this material and disposing of certain materials like foam for making the fairing must be considered since decomposition or incineration of waste produces greenhouse gases like CO2 and Methane. Because of this, the team has been dedicated to using scrap student stock at the machine shop whenever possible. By recycling, the team can reduce further impact from buying new material. This is the most effective way to reduce the team's carbon footprint. Besides recycling, the team intends on using proper machining techniques to save both time and electricity. After construction, emissions are still released due to buying replacement parts, food for energy, and the construction of bike lanes.

Since the material costs for this design requires two complete frames, three sets of three wheels, an estimate of 140 kilograms (kg) of aluminum required, 10 kg of rubber for tires, and 50 kg of fiberglass. The team estimates the total miles traveled will be 20,000 miles throughout its lifetime. Through the construction of bike lanes, food, and upkeep, the team will emit .07 kg of CO2 per mile traveled. By referencing Table 6, the total CO2 emissions can be determined for the construction of Cerberus.

Table 8: CO2 Emissions

Product	CO2e	Product Used	CO2 Production (g)	Product Life Cycle Energy Consumption (MJ/kg)	Product Life Cycle Energy Consumption (J)
Aluminum	9.2 kg per kg of material [6]	140kg	1,288,000	46.8 [7]	6552,000,000 [7]
Tires	4.4 kg per kg of material [6]	10kg	44,000	12.9 [8]	129,000,000 [8]
Fiberglass	7.88MMBtu per ton [6]	50kg	37,140 [9]	NA	415,670,000 [9]
Bicycle	.07 kg per mile [6]	20,000 miles	1,400,000	NA	NA
Machining	.155 kg per kg of material [6]	50kg	7,750	2.015 [8]	100,750,00 [8]
Foamular 250	60.8kg per sq. meter [10]	17kg	1,626,400	80.7 MJ per sq. meter	2,158,725,000
Welding	62 kg per kg of material [8]	25kg	1,550,000	920 [8]	23,000,000,000 [8]

2.6 Other Analyses

2.6.1 Steering

Using SolidWorks Motion Analysis, the team could determine steering angles at various displacements put into the system by the driver. The major objectives that the steering system should accomplish are to achieve a minimum turn radius of 10 feet, minimize toe-in angle, and can adjust the steering system to respond to a range of small and large actuation from the driver. To be able to determine whether these objectives are met, criteria will be established to define success or failure.

Turn radius calculation can be determined using steering angle of either the inside or outside wheel and plug into the Ackermann equations. As a true measurement, however, the team can measure one wheel's starting point, propel the car 180 degrees, and measure the distance between the starting and finishing point which gives the true turning radius. If the vehicle achieves a turn radius of more than 10 feet, the team will have failed to achieve the self-imposed objective, but as long as it is below 26 feet, the ASME requirement will be met. Similarly, toe-in can be measured easily by taking steering angle measurements and comparing them to the straight angle of the vehicle's frame. A toe-in angle of more than 5 degrees will be considered a failure for this objective. Adjustability of the steering system cannot be measured directly but rather will be measured by having the ability to maintain stability at high speed while the driver

is making vigorous movements but also can make adjustments to achieve a turn radius not more than 10 ft.

Table 9: Steering Turn Radius Analysis Summary

Objective	Method	Result
Structural Strength	ANSYS	Components Passed
Steering Articulation	SolidWorks Motion Analysis	Turn Radius Passed

2.6.2 Drivetrain

The drivetrain utilized a 38-tooth asymmetric chain ring. This chain ring alone improved power endurance by 10% at a cadence estimated around 90 rpms. This can be seen in Figure 26 along with the overall percentage of power the rider gained.

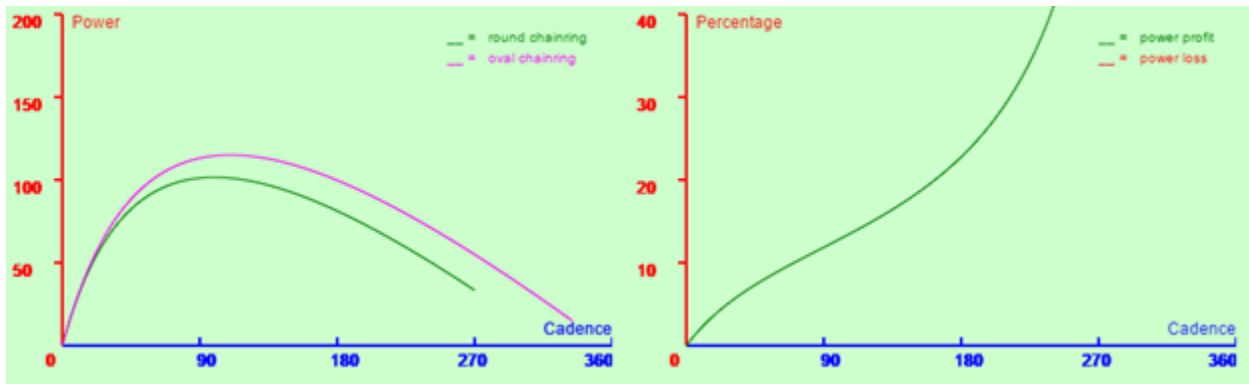


Figure 26: Efficiency of 38T Q-ring [12]

This 38-tooth asymmetric chain ring also helps with muscle fatigue. In a normal pedal stroke the rider loses power on the upstroke due to the muscles not being able to produce the same amount of power as the down stroke. In Figure 28, the muscle groups that are activated along a normal pedal stroke can be seen. The asymmetric ring allows for these muscles in the upstroke to effectively provide the same amount of power as the muscles in the down stroke.

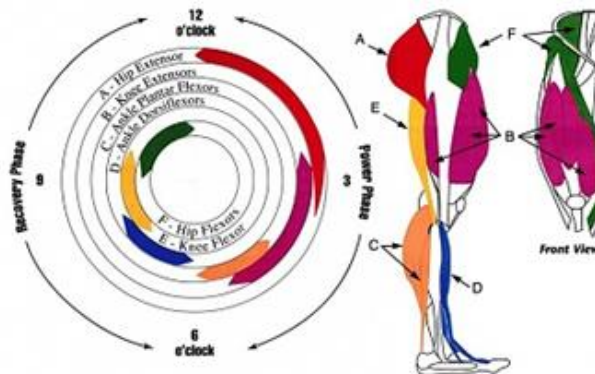


Figure 27: Pedal Stroke [13]

Along with the asymmetric chain ring, the hubs that house the chain ring were also optimized. The hub (Figure 28) itself was housed within a larger hub, however the smaller hub was offset within the larger one. This was done to develop and implement a tensioning system within the hub that houses the chain ring. The larger hub rests inside the shell which is in turn welded to our frame. This larger hub will be free to rotate within this shell, and due to the offset small hub, when rotated, the chain ring will either tighten the chain link system or loosen it.

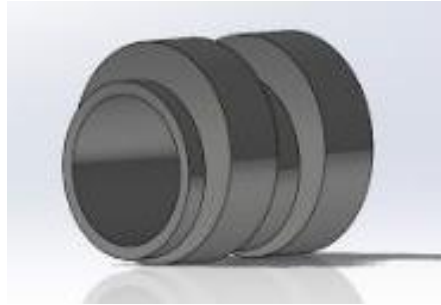


Figure 28: Asymmetrical Hub

To achieve the correct speed set out by the team, specific calculations for speed, cadence, and gear ratios were conducted with a set crank length of 170mm. The full table of values and equations used can be viewed in Appendices B and C.

Table 10: Drivetrain Crucial Gear Ratio Summary

Criteria	38x11 ratio	42x11 ratio	54x11 ratio
Max Speed achieved	34.09	38.85	49.94
Avg. Cadence 90 rpm	24.29	26.90	34.57

3. Testing

3.1 Rollover/Side Protection System

3.1.1 Top Load Testing Setup

Currently, the team is working on setting up a top load testing setup. This setup will possibly consist of utilizing a steel structural frame as a reactant force. The frame will be bolted to a steel plate that will be angled 12° counterclockwise to achieve the correct orientation. A hydraulic ram that is bolted to the steel structural frame and driven by a hydraulic pump will be lowered towards the top of the frame. A seven-foot-long bar will be placed on top of the RPS and the end of the bar will be placed one foot past the RPS. The hydraulic ram will be located directly above the contact point between the RPS and the bar. At the other end of the bar, 100 lbs of weights will be added. At the contact point, a bending moment of 600 lbf-ft will be applied as a reactant force against the ram. After applying the load, a ruler or laser measure will be used to receive an estimate of deformation.

3.1.2 Top Load Testing Results

The top load testing setup is currently being designed and the RPS will be tested after this report submission.

3.1.3 Side Load Testing Setup

A similar setup to the top load test will possibly be used for the side load testing setup. The hydraulic ram will be placed on the side of the steel structural frame and the furthest edge of the RPS will be placed against the ram. Once all the components are properly placed, a load of 300 lbf will be applied to the edge of the RPS. With the ram pressing the load against the RPS, the maximum deflection will be measured with a ruler laser measure.

3.1.4 Side Load Testing Results

The side load testing setup is currently being designed and the RPS will be tested after this report submission.

3.2 Developmental Testing

3.2.1 Frame

When choosing the geometry of the frame, the cross-sectional geometry was taken into consideration to ensure that the tubes could bend. To ensure this, the team purchased 3-foot-long tubes of 1 ¾ and 1 ¼ inch outer diameter with a wall thickness of 0.125 inches. A wall thickness of 0.125 inches was chosen because if the tubing is too thin, it will crinkle and not be structurally stable. The tube bender provided by NAU's machine shop was used to bend the tubing to its extremities. Before bending the tube, the die and bender were lubricated with grease to reduce wear on the dies.

Both tubes were able to be bent, but the 1 ¾ outer diameter experienced brittle failure. This was due to the material being heat treated to T6 from the metal supplier. The tube was only able to bend at 10 degrees and then would catastrophically fail. At first, the team tried looking for another material selection, such as Aluminum 6061 - T0, to be able to bend it and then heat treat it to a T6 specification, but most vendors either supply 6061 - T6 or 6063 Aluminum. Instead, the tubing was annealed with an Oxy-Acetylene torch at the bend locations and then cooled overnight. This allowed for the tubing to bend. The 1 ¼ outer diameter tubing did not fail; it was able to bend at an angle of about 85°. Figure 29 displays both test specimens.



Figure 29: Bending Test Specimens

After bending both specimens and realizing they would have to have heat added to the bend locations to allow for ease of bending, the team decided to keep the same cross sectional geometry and material selection.

Due to the testing, the 1 ¼ tube will be annealed and air cooled to change the properties to a T0 specification and allow the team to bend it.

3.3 Performance Testing

As the final vehicle is still under construction at this time, all performance testing will be discussed during the presentation in March.

4. Safety

4.1 Rollover/Side Protection System

The HPV will have a RPS to ensure that the rider will not make contact with the ground if the vehicle were to roll over. The RPS is wide and long enough to ensure that the biggest and smallest rider will not experience harm in the event of a rollover occurring. Also, the RPS satisfies the ASME deflection requirements through analysis and is expected to pass in the physical test setup as well.

4.2 Seat Belt

The vehicle will be equipped with a retractable 3-point safety harness. The harness will mount to the roll cage as well as the primary frame members.

4.3 Steering System

To ensure the safety of all individuals that come in contact with the vehicle, the steering system must guarantee accurate Ackermann angles and ensure minimum toe-in and toe-out. If the steering angles are not consistent, excess stress will be transmitted through the steering linkages and could potentially lead to the failure of the steering system. Additionally, if there is any toe-in or toe-out between the two wheels, the front tire tread will wear out extremely fast. If the tire

tread becomes compromised and a tire fails at high speed the vehicle, driver, and spectators can be put at risk. Since all linkages will be adjustable, nuts will be installed to stop components threading together when they are not supposed to. Furthermore, all linkages will be adjusted to the point where no looseness is present between components. Finally, stability will be assessed by the driver being able to sprint while not influencing the vehicle's trajectory. The steering input will be adjusted to cater to stability.

4.4 Braking System

By choosing to implement a hydraulic braking system, the team is able to achieve a stopping distance as well as an increase in stability both in a dead stop and within the turns. The braking system uses TRP HY\RD mechanical-to-hydraulic brakes, which sits on 160mm rotors. The mounting brackets are welded directly to the steering knuckles to ensure the correct spacing and optimized brake position. The hydraulic piston is actuated by a mechanical cable tied into a single lever at the controls. The team has implemented two of these hydraulic brakes, where one of each is placed onto the two front wheels. Instead of having two separate levers controlling each brake independently, the team has designed a device that joins the two cables into one single lever.

4.5 Sharp Edges, Protrusions, Pinch Points

All sharp edges, protrusions, and pinch points will be covered or made safe to the rider. Primary concerns include open tube ends, steering components, as well as various other moving drivetrain parts. Tubes will be capped to ensure no cuts or protrusions are a liability. Fairing edges will be sanded and finished to limit possibilities of cuts. Steering components are located beneath the frame to protect the rider.

4.6 Manufacturing Safety

Manufacturing precautions were taken throughout construction of the vehicle. The team followed NAU's manufacturing facilities rules in accordance to MSDS and equipment user manuals. Special precautions were taken in experimental wiring and testing of devices which included consulting professionals and increasing initial background research.

4.7 Rider Field of View

The vehicle will include 3 major windows. The center window will accommodate up to 30 degrees of frontal rider visibility. Four side windows will provide an additional 70 degrees of rider visibility. Although there is a 5-degree blind spot located on the A-Pillar of the vehicle as shown in Figure 30 below, slight head movement will alter the field of view and allow for the driver a full 180-degree range of view. Rear facing mirrors will be mounted on the fairing allowing an additional rear view of up to 20 degrees.

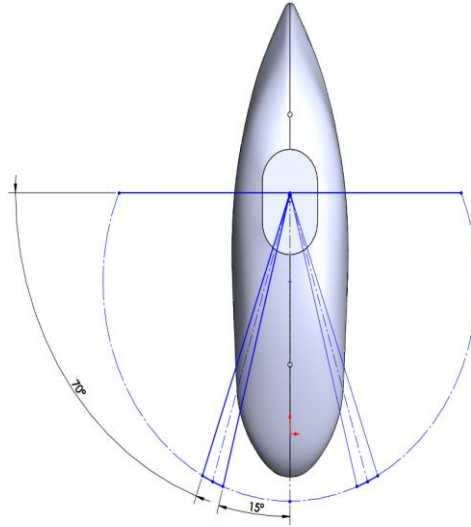


Figure 30: Rider's Field of View

4.8 Safety Accessories

Multiple safety accessories were included in the vehicle design. These accessories include a bell/horn which will be mounted inside of the vehicle. A white headlight will be mounted within the front of the vehicles fairing along with a rear LED brake light. Both will be toggled by a switch located within the rider cockpit as well as meet the distance requirements of 150 meters set by ASME. Rear and side reflectors will be installed on the final vehicle to meet competition requirements.

4.9 Additional Safety Features

The innovation for the vehicle acts as the team's additional safety feature. The tint is polarized which is beneficial to the rider's eyes against the UV rays from the sun. The electrochromic windshield also adds vehicle security for the owner.

5. Conclusion

5.1 Comparison

Table 11: Comparison of Target Goals vs Results

	PDS	Target	Actual
Frame	Overall length of less than 65 inches	< 65"	61.25 in
	Width is equal to or less than 36 inches	≤ 36"	31.60 in
	Fit shortest and tallest rider (65-72 inches)	Yes	Yes

	RPS top load deflection equal to or less than requirement	2"	0.68"
	RPS side load deflection equal to or less than requirement	1.5"	0.26"
Cost	Maintain a budget of \$7500 or less	≤ \$7500	On track
Overall	10-foot turning radius	10'	TBD
	Achieve top speed of 40 mph	40 mph	TBD
	Weight is less than 90 pounds	< 90 lb	TBD
	Stop from speed of 15 mph in 20 ft distance	20'	TBD
	Demonstrate stability at 3-5 mph	100'	TBD

5.2 Evaluation

As it can be noted from the above table, many design goals have yet to be determined if they have been achieved or not. Those that have been completed are well within the desired range. For those that are still to be determined, preliminary analyses have led the team to believe the majority, if not all, of the design goals will be accomplished. The team was also successful in respect to the analyses performed as they reinforced the team's design choices. Although performance testing is incomplete, the team is expecting to learn what designs may need to be altered in order to achieve the rest of the design goals.

5.3 Recommendations

Machines needed to improve the vehicle would be an oven in which the Aluminum 6061-T6 could be annealed for ease of bending. A large-scale CNC would be helpful in cutting out accurate profiles for the fairing when making the plug. Taking the drivetrain into account there are many areas that can be improved: implementing a shortened chain as well as designing a tensioning system will improve chain drop and misalignment issues. Routing the chain internally will aid in safety and reduce chain noise. Ideally the vehicle should have a direct drive chain system instead of using idler gears to make up the distance the chain need to cover. As the final vehicle is still being built, more recommendations may be developed based on the performance testing. If so, this will be discussed during the presentation.

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7. Appendices

Appendix A

Table 12: Decision Matrix

Scale	0-5
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	Cost	Functionality	Simplicity	Manufacturability	Safety	Resilience	Repairability	Average
Frame								
Straight	3	4	5	4	4	3	3	3.71
Cont Bend	2	2	1	1	4	4	1	2.14
3 Point Curve	4	4	3	3	4	4	3	3.57
Steering								
Joystick (Bell Crank)	3	2	2	3	2	2	1	2.14
Single Arm (Bell Crank)	3	3	4	3	3	2	3	3
Double Arm (Bell Crank)	3	4	4	3	4	4	3	3.57
Rack and Pinion (Wheel)	2	1	2	4	4	4	1	2.57
Drivetrain								
Belt	2	3	3	2	4	3	3	2.86
Chain	3	4	5	5	4	4	4	4.14
Front Drive	2	3	0	1	4	2	1	1.86
Internal Transmission	1	3	1	1	5	1	0	1.71
Fairing								
Full	2	5	3	3	5	4	3	3.57
Front	4	3	5	4	2	4	3	3.57
Rear	3	3	4	3	1	4	3	3
Soft Fairing	4	3	5	4	2	2	4	3.43

Appendix B

Table 13: Speed at Specific Cadences

Cadence Gear	30	40	50	60	70	80	90	100	110	120	130	Cadence Gear
38x11	8.10	10.80	13.50	16.20	18.90	21.59	24.29	26.99	29.69	32.39	35.09	38x11
38x12	7.44	9.92	12.40	14.88	17.36	19.84	22.32	24.80	27.28	29.76	32.24	38x12
38x13	6.85	9.14	11.42	13.71	15.99	18.28	20.56	22.85	25.13	27.42	29.70	38x13
38x14	6.36	8.48	10.60	12.72	14.84	16.96	19.08	21.20	23.32	25.44	27.56	38x14
38x26	3.43	4.57	5.71	6.85	8.00	9.14	10.28	11.42	12.57	13.71	14.85	38x26
38x28	3.19	4.26	5.32	6.38	7.45	8.51	9.58	10.64	11.70	12.77	13.83	38x28
42x11	8.97	11.96	14.94	17.93	20.92	23.91	26.90	29.89	32.88	35.87	38.85	42x11
42x12	8.22	10.95	13.69	16.43	19.17	21.91	24.65	27.38	30.12	32.86	35.60	42x12
42x13	7.58	10.11	12.64	15.16	17.69	20.22	22.74	25.27	27.80	30.33	32.85	42x13
42x14	7.04	9.39	11.74	14.08	16.43	18.78	21.12	23.47	25.82	28.17	30.51	42x14
42x26	3.80	5.07	6.34	7.60	8.87	10.14	11.41	12.67	13.94	15.21	16.48	42x26
42x28	3.52	4.69	5.87	7.04	8.22	9.39	10.56	11.74	12.91	14.08	15.26	42x28
52x11	11.10	14.80	18.50	22.20	25.91	29.61	33.31	37.01	40.71	44.41	48.11	52x11
52x12	10.16	13.55	16.94	20.33	23.71	27.10	30.49	33.88	37.27	40.65	44.04	52x12
52x13	9.39	12.52	15.65	18.78	21.91	25.04	28.17	31.30	34.43	37.56	40.68	52x13
52x14	8.71	11.61	14.51	17.42	20.32	23.22	26.12	29.03	31.93	34.83	37.74	52x14
52x26	4.69	6.26	7.82	9.39	10.95	12.52	14.08	15.65	17.21	18.78	20.34	52x26
52x28	4.37	5.82	7.28	8.73	10.19	11.64	13.10	14.55	16.01	17.46	18.92	52x28
54x11	11.52	15.37	19.21	23.05	26.89	30.73	34.57	38.42	42.26	46.10	49.94	54x11
54x12	10.56	14.08	17.60	21.12	24.65	28.17	31.69	35.21	38.73	42.25	45.77	54x12
54x13	9.74	12.99	16.23	19.48	22.73	25.98	29.22	32.47	35.72	38.96	42.21	54x13
54x14	9.06	12.08	15.10	18.12	21.14	24.16	27.18	30.20	33.22	36.24	39.26	54x14
54x26	4.88	6.51	8.14	9.76	11.39	13.02	14.65	16.27	17.90	19.53	21.16	54x26
54x28	4.53	6.04	7.55	9.06	10.57	12.08	13.59	15.10	16.61	18.12	19.63	54x28

Table 14: Gear Ratios

Ring Cog	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54
11	3.5	3.6	3.6	3.7	3.8	3.9	4.0	4.1	4.2	4.3	4.4	4.5	4.6	4.6	4.7	4.8	4.9
12	3.2	3.3	3.3	3.4	3.5	3.6	3.7	3.8	3.8	3.9	4.0	4.1	4.2	4.3	4.3	4.4	4.5
13	2.9	3.0	3.1	3.2	3.2	3.3	3.4	3.5	3.5	3.6	3.7	3.8	3.9	3.9	4.0	4.1	4.2
14	2.7	2.8	2.9	2.9	3.0	3.1	3.1	3.2	3.3	3.4	3.4	3.5	3.6	3.6	3.7	3.8	3.9
15	2.5	2.6	2.7	2.7	2.8	2.9	2.9	3.0	3.1	3.1	3.2	3.3	3.3	3.4	3.5	3.5	3.6
16	2.4	2.4	2.5	2.6	2.6	2.7	2.8	2.8	2.9	2.9	3.0	3.1	3.1	3.2	3.3	3.3	3.4
17	2.2	2.3	2.4	2.4	2.5	2.5	2.6	2.7	2.7	2.8	2.8	2.9	2.9	3.0	3.1	3.1	3.2
18	2.1	2.2	2.2	2.3	2.3	2.4	2.4	2.5	2.6	2.6	2.7	2.7	2.8	2.8	2.9	2.9	3.0
19	2.0	2.1	2.1	2.2	2.2	2.3	2.3	2.4	2.4	2.5	2.5	2.6	2.6	2.7	2.7	2.8	2.8
20	1.9	2.0	2.0	2.1	2.1	2.2	2.2	2.3	2.3	2.4	2.4	2.5	2.5	2.6	2.6	2.7	2.7
21	1.8	1.9	1.9	2.0	2.0	2.1	2.1	2.1	2.2	2.2	2.3	2.3	2.4	2.4	2.5	2.5	2.6
22	1.7	1.8	1.8	1.9	1.9	2.0	2.0	2.1	2.1	2.1	2.2	2.2	2.3	2.3	2.4	2.4	2.5
23	1.7	1.7	1.7	1.8	1.8	1.9	1.9	2.0	2.0	2.0	2.1	2.1	2.2	2.2	2.3	2.3	2.4
24	1.6	1.6	1.7	1.7	1.8	1.8	1.8	1.9	1.9	2.0	2.0	2.0	2.1	2.1	2.2	2.2	2.3
25	1.5	1.6	1.6	1.6	1.7	1.7	1.8	1.8	1.8	1.9	1.9	2.0	2.0	2.0	2.1	2.1	2.2
26	1.5	1.5	1.5	1.6	1.6	1.7	1.7	1.7	1.8	1.8	1.9	1.9	1.9	2.0	2.0	2.0	2.1
27	1.4	1.4	1.5	1.5	1.6	1.6	1.6	1.7	1.7	1.7	1.8	1.8	1.9	1.9	1.9	2.0	2.0
28	1.4	1.4	1.4	1.5	1.5	1.5	1.6	1.6	1.6	1.7	1.7	1.8	1.8	1.8	1.9	1.9	1.9

Table 15: Cadence at Specific Speeds

Speed Gear	5.0	10.0	15.0	20.0	25.0	30.0	35.0	40.0
38x11	19	37	56	74	93	111	130	148
38x12	20	40	60	81	101	121	141	161
38x14	24	47	71	94	118	141	165	189
38x26	44	88	131	175	219	263	306	350
38x28	47	94	141	188	235	282	329	376
38x32	54	107	161	215	269	322	376	430
42x11	17	33	50	67	84	100	117	134
42x12	18	37	55	73	91	110	128	146
42x14	21	43	64	85	107	128	149	170
42x26	39	79	118	158	197	237	276	316
42x28	43	85	128	170	213	256	298	341
42x32	49	98	146	195	244	293	341	390
54x11	13	26	39	52	65	78	91	104
54x12	14	28	43	57	71	85	99	114
54x14	17	33	50	66	83	99	116	132
54x26	31	61	92	123	154	184	215	246
54x28	33	66	99	132	166	199	232	265
54x32	38	76	113	151	189	227	265	303

Appendix C

Equation 1

$$Speed = (\pi * D_w) * \left(\frac{G1}{G2}\right) * C$$

D_w = Diameter of Wheel

$G1$ = # of gears on ring 1

$G2$ = # of gears on ring 2

C = Cadence

Equation 2

$$G_r = \left(\frac{G1}{G2}\right)$$

G_r = Gear ratio of chainrings

$G1$ = # of gears on ring 1 (Drive Ring)

$G2$ = # of gears on ring 2 (Driven Ring)

Equation 3

$$C = \frac{Speed}{(\pi * D_w) * G_r}$$

C = Cadence

D_w = Diameter of Wheel

G_r = Gear ratio of chainrings

Speed = MPH of bike