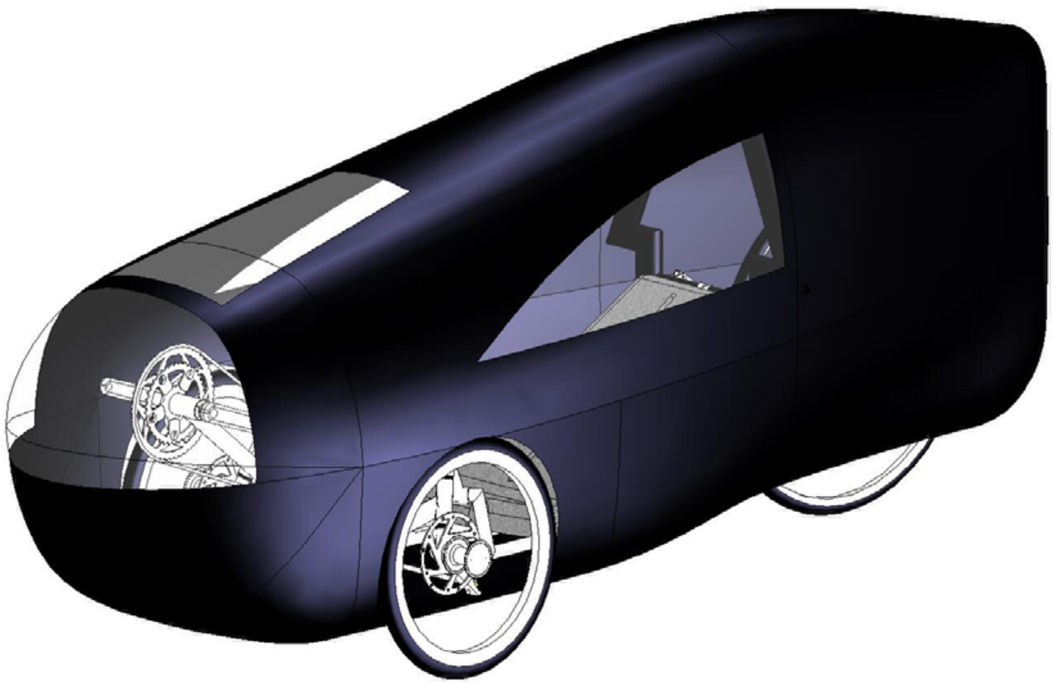
June 7, 2006

Portland State University
Human-Powered Vehicle Team



D'Alembert's **Vike Trike**

Human Powered Vehicle Design Report

ME 493 Final Report – Year 2006

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Executive Summary

The 2006 Portland State University human-powered vehicle is a fully-faired, recumbent tricycle, designed and built to win the overall single rider category at the 2006 ASME West Coast HPV Challenge. Every aspect of this vehicle's design is original to this year's team.

The vehicle has been named D'Alembert's Vike Trike, to highlight the attention that has been paid to aerodynamics in its' concept and fabrication. In an effort to realize D'Alembert's paradox, the fairing shape and surface texture were designed and built as to best achieve steady, uniform flow with minimal drag.

Beneath the carbon fiber composite sits a 4130 steel monotube frame, with two 20 inch wheels in the front, and a single 700C wheel in the rear. Power is delivered through a Shimano racing transmission, with the addition of a single Terracycle idler gear to direct the chain path. The cockpit consists of an adjustable, custom-made, carbon fiber composite seat, protected by a 6061–T6 Aluminum roll bar.

The main subsystems of the vehicle are the frame, fairing, the mechanical integration, drivetrain, and the rider protection systems. Extensive research, analytical modeling, and computer-aided design have been performed on multiple aspects of the vehicle. Wherever possible, the results of these analyses have been verified or compared to controlled testing, the details of which follow in the body of this report.

The performance of the Vike Trike in competition has served as the ultimate test of its' design and functionality. Our team represented Portland State University at the 2006 ASME West Coast Human Powered Vehicle Challenge and proudly pedaled our way to a third place finish at over 40mph. With the success of this first prototype and the knowledge gained from testing it, we believe that it will serve as an excellent platform for future HPV research and development at Portland State.

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[1] Introduction

Each spring the American Society of Mechanical Engineers (ASME) sponsors a Human Powered Vehicle (HPV) competition for colligate engineering teams from across the country. A truly engineering inspired competition, the three vehicle classes and four events are designed to focus on vehicle design, innovation, and performance rather than athletic ability. Such a competition structure demands a successful design to be superior in multiple mechanical disciplines including fluid mechanics, machine design, heat transfer, and material science.

To test the versatility of the HPVs each competition event is designed to asses a separate vehicle discipline. The sprint event is designed to test the top speed of the vehicles by timing them through a 100 meter time trap following a 500 meter run-up. Vehicle endurance and maneuverability are tested during a 65 kilometer road course in which teams must switch riders multiple times. The ability of a vehicle to handle utility tasks is evaluated on an obstacle course on which competitors are required to transport packages from station to station. Finally, the vehicle design event asses the quality of the vehicle's design based on a design report and presentation. After visiting the 2005 competition and finding the design, endurance and sprint events to be the most competitive, our team set a goal to win these events at the 2006 competition.

To accomplish this, our team set performance targets based on exceeding the performance of last years winners (see Appendix 8.1 for a summary of 2005 results). We determined that the vehicle must achieve a maximum velocity greater than or equal to 45mph, and an average endurance speed greater than or equal to 20 mph. We began the design process with these initial benchmarks.

[2] Mission Statement

Our mission is to design and produce a competitive, innovative, and safe human powered vehicle for entry in the speed and endurance events of the 2006 ASME West Coast Challenge in April of 2006. We aim to win the overall single rider category by producing the most efficient vehicle possible, and to fulfill our Portland State University mechanical engineering senior capstone design sequence requirements.

[3] Major Design Specifications

The two main customers of the HPV are the race team members who are the end users of the product, and the ASME judges who determine vehicle scoring at the competition and inspect the vehicle for compliance with competition rules. Using the expectations of these two groups as design requirements, the following list of major design specifications was developed (see Appendix 8.2 for the complete PDS document).

A)The vehicle must be compliant with all competition rules. A number of rules have been set by the ASME for all vehicles entering the competition, a summary of which is presented in Appendix 8.3. To be allowed to enter the competition, the vehicle must meet or exceed these requirements.

B) The vehicle must be light. In order to accelerate and corner faster than our competition, the weight of the vehicle must be kept as low as possible. With a maximum weight benchmark set by the 2005 endurance winner of 58lbs, a competitive target vehicle weight was set at 50lbs.

C) The vehicle must be aerodynamically efficient. At the velocity required to place first in the ASME sprint event, aerodynamic drag is the largest force resisting vehicle motion [see Appendix 8.6.1]. Therefore, to achieve the design goal of 45mph, the power required to overcome aerodynamic drag at this velocity must not exceed the estimated rider power output of 0.5 hp [Ref. Wilson pg. 44].

D) The vehicle must be safe. Because the vehicle will be operated in a dynamic racecourse environment, it must be designed such that the riders are protected from bodily harm regardless of vehicle motion or orientation relative to the road surface.

E) The vehicle must be built within budget. Total vehicle production costs must not exceed $5,000.

[4] Top Level Design Alternatives

During the design phase we conducted extensive internal and external searches to develop a list of design options for each system of the vehicle. Condensed versions of the internal and external search documents appear in Appendix 8.4. Using a concept scoring matrix (see Appendix 8.5) we made design decisions based on each options ability to satisfy the requirements of the PDS. Of these design options, three of the most important top level design alternatives are presented in detail here. They are: the fairing size, number of wheels and frame style.

Competition rules require a fairing covering 1/3 of the vehicle's frontal area as shown in figure 4.1, however greater vehicle efficiencies can be achieved by enclosing the vehicle in a full aerodynamic shell. The drawback to this increase in efficiency is higher overall weight and cost of the vehicle. A lower weight fairing also reduces the rolling resistance, but the increase in rolling drag is far outweighed by the reduction in aerodynamic drag at velocities greater than 35mph (see Appendix 8.6.1). After researching vehicle aerodynamics and comparing theoretical models of various fairing sizes, we determined that a full fairing was necessary to obtain the aerodynamic efficiency goal set by the PDS.

In selecting the number of wheels the vehicle should have, our team considered two, three and four wheels. Increasing the number of wheels can increase the stability of the vehicle. However, decreasing the number of wheels in contact with the ground decreases the rolling resistance of the vehicle, and decreases the rotating mass. The number and orientation of wheels on the vehicle determines its' overall size and shape, which has significant impacts on aerodynamic efficiency [Ref. Tamai, pg 174]. At the 2005 ASME competition, we saw that two wheeled vehicles were often unstable which caused many of them to crash during cornering. In addition, the two wheeled vehicles with full fairings experienced difficulties starting and stopping due to their inability to place their feet on the ground. We determined that two 20in wheels in front and a single 700c wheel in the back (known as a 'tadpole trike'), would give us the best balance between aerodynamic efficiency and stability.

Figure 4.1. Example of vehicle with 1/3 frontal area coverage.

The selection of frame style was also a major design consideration. For this decision three major alternatives were considered. The first was a monocoque tub-frame design in which the bottom shell of the fairing is designed and built using advanced composite construction techniques. This design reduces overall vehicle weight by combining two separate parts into a single multipurpose part. While stiff and light, we determined that a monocoque design was unacceptable due to the cost of manufacturing several molds which were required for a successful design. The second option was a tubular space frame design in which small diameter tubing is assembled into a rigid structure by using multiple triangulated sections. This design requires a frame to be spatially large in order to achieve the required stiffness, and is therefore unacceptable from an aerodynamic perspective. The third option, a monotube design uses one main frame member and can be designed to fit into the bottom of a fairing and consume very little space. These aerodynamic benefits, as well as ease of manufacture, led to the decision to use a monotube frame in the vehicle.

[5] Final Design and Evaluations

The 2006 Portland State University HPV is an assembly of multiple subsystems, all of which are unique in function and design. For presentation clarity the vehicle design has been divided into five sections: fairing, frame, drive train, mechanical integration, and safety (see figure 5.1). Design summaries for each of these subsystems are presented in sections 5.1-5.5.

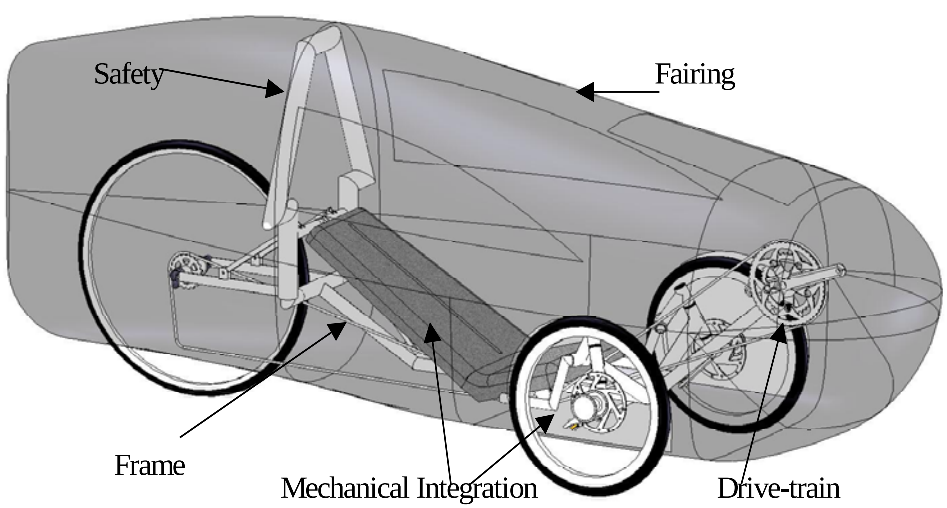


Figure 5.1: Overview of design and subsections

A summary of the various product design specifications, targets, and evaluation results are presented in Table 5.1. The durability of the prototype was evaluated during pre-competition road testing and the two competitions it entered. During the course of the competitions, the assembled vehicle was ridden through potholes and irrigation channels at 25mph, rolled onto its side, and crashed into course barriers. The entire

Metric	**Target**	**Produced**	**Target Met?**
Turning Radius	<= 25 ft.	15 ft	Yes
Stopping Distance	<20 ft from 15mph	8 ft from 15mph	Yes
Straight line stability	0˚/100ft	0˚/100ft	Yes
Vehicle Identification	Yes	Yes	Yes
Frame Weight	<= 30 lb	32 lb	No
Fairing Weight	<= 20 lb	18 lb	Yes
Aero Drag Power	<= 0.5 hp	0.48 hp	Yes
Production Cost	<= $5,000.	$5,290	No
Free of sharp edges	Yes	Yes	Yes
Roll-over protection	Yes	Yes	Yes
Rider Restraint	Yes	Yes	Yes
Horizontal Visibility	> 90˚	184°	Yes
Vertical Visibility	> 50°	80°	Yes
Max Velocity	> 45 mph	43.3 mph	No
Endurance Velocity	> 20 mph	26 mph	Yes
Life in service	April 30, 2006	May 30, 2006	Yes
Static SF	>= 5	5	Yes
Fatigue SF	>= 2	2.31	Yes
Internal Temp.	<10˚ above ambient	6˚	Yes
Roll velocity	10 mph / 20 ft rad	15 mph /20 ft rad	Yes
Pre-comp Maintenance	<= 1 hour	20 minutes	Yes
Comp maintenance	0 minutes	0 minutes	Yes
Shoulder room	>= 20 in	20.98 in	Yes
Max X-seam	>= 45 in	45 in	Yes
Min X-seam	<= 39 in	38 in	Yes
Rider exchange	<= 10 sec	9 sec avg.	Yes

Table 5.1: Summary of PDS Targets and prototype statistics.

vehicle was shown to be durable as it continued to function as designed with no components showing signs of deformation or failure. Detailed evaluations for each subsystem are included in the following subsections.

[5.1] The Fairing

[5.1.1]Overview

As stated above, a minimum of 33% frontal coverage is required by the competition, and a full fairing was determined to be necessary to meet the PDS requirements. Using the PDS requirements for aerodynamics, safety, and weight as the critical design parameters, the fairing was designed based on the theoretical and experimental information described below.

[5.1.2]Basic Geometry

Studies of submerged body flow and general aerodynamics indicate that the two main sources of drag as outlined by most classical fluid dynamics texts are those due to pressure and viscous effects [Ref. Munson]. While these are the largest contributors to drag for general submerged flows, in the study of aerodynamics for streamlined vehicles, Tamai identifies interference and induced drag as two additional sources. To produce the most efficient design possible, the team considered all four of these sources of drag and made design decisions based on the greatest overall aerodynamic benefit.

For general submerged flow problems, pressure drag due to high-pressure zones at the leading surface of the body and low-pressure zones on downstream surfaces is the largest contributor to drag. Years of research in the fields of fluid dynamics and aerospace have produced many geometries which successfully address this problem and achieve almost complete pressure recovery. This nearly eliminates pressure drag.

The National Advisory Committee for Aeronautics (NACA) spent years developing airfoil geometries which have since been published in the public domain. These databases were accessed using John Dreese's Design FOIL software and sized to fit around the Vike Trike, as detailed in section 5.1.3. The result is a fairing constrained in plan view by a NACA 4-series airfoil, and a nose constrained on the top and bottom using curves derived from NACA 6-series airfoils. The resulting bulk geometry, shown in plan and side views, are presented in Figures 5.2 and 5.3, respectively.

The second largest source of drag, is viscous drag. Viscous drag is due to the shearing of fluid along its interface with the solid as constrained by the no slip condition. While this cannot be eliminated, the design team attempted to reduce it by two methods. First, the overall wetted area of the fairing was kept to the minimum possible size by reducing interior geometrical clearances to the minimum acceptable for comfort and safety of all riders.

Second, attempts were made in the design to control the state of the boundary layer along the length of the fairing. As in the highly studied case of the flat plate, there are three possibilities for the state of the boundary layer. Arranging these cases in order of increasing drag as stated by Tamai: laminar, turbulent, and separated, the ideal case for design is clear. While details of boundary layer flows are outside the scope of this paper [see: Acheson ch.8, Tamai ch.2.2], theory predicts and experiments show that producing a favorable pressure gradient (-dp/dx along the length of the fairing) extends the length of the laminar boundary layer. Bernoulli's equation indicates that this may be accomplished by increasing the fluid velocity along the length of the surface.

In the design of the PSU Vike Trike, attempts to accomplish laminar boundary layer flow were made by designing what Tamai calls 'gentle' contours to thin the boundary layer on the nose and other up-stream surfaces, examples of which may be seen in Figure 5.4. For contours downstream of the maximum width where a favorable pressure gradient is not possible, we used a maximum body convergence angle of 17 degrees from free stream flow as suggested by Tamai (see Figure 5.5).

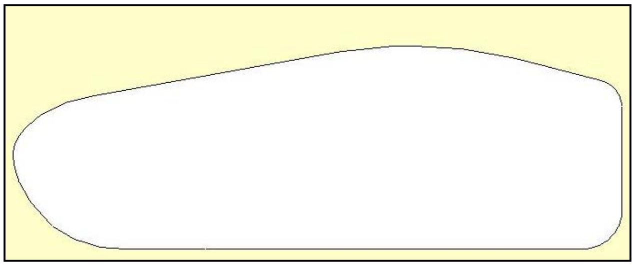


Figure 5.2: Side view of bulk geometry

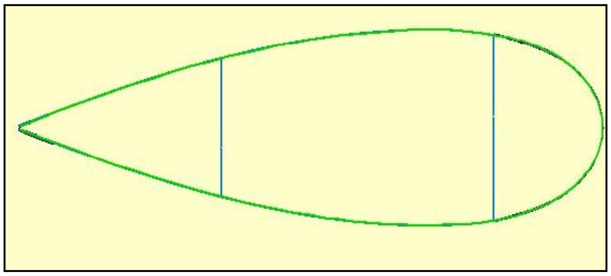


Figure 5.3: Plan view of bulk geometry

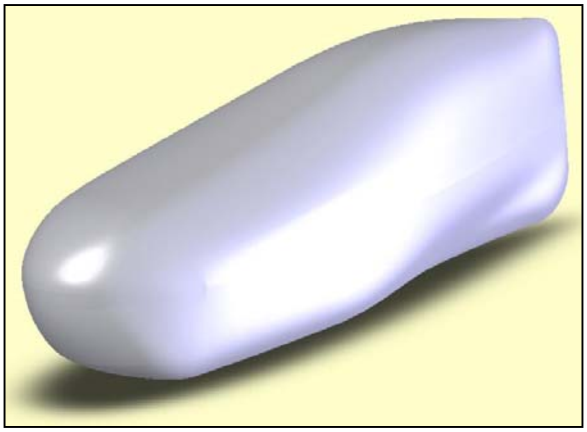


Figure 5.4: View of curves used on nose and sides.

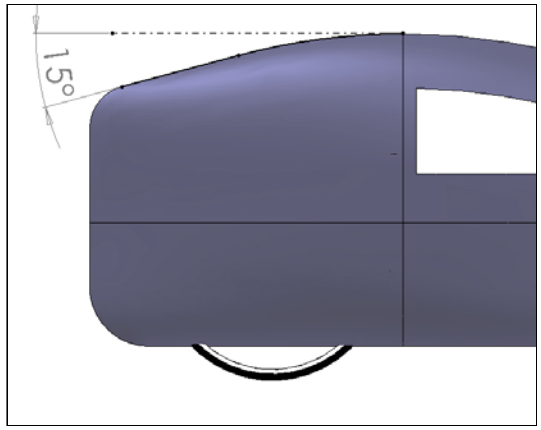


Figure 5.5: View of curves used on tail sections.

Once the main contributions to drag were mitigated, steps were taken to reduce the other sources of drag as defined by Tamai. Induced drag, that which is inherent in the production of lift or down force, was eliminated by designing the fairing with zero angle of attack.

Interference drag, due to surface roughness, body seams, etc. was reduced using fabrication techniques to reduce surface abnormalities.

[5.1.3] Scale Optimization

Once the general airfoil curves had been chosen as detailed in section 5.1.2, the scaling of the fairing was optimized. From data plotted for symmetric airfoils [Ref. Munson figure 9.16], minimum drag coefficients occur at Reynolds numbers on the order of three million. Using a design speed of 45 mph, and fluid properties from Munson corresponding to climate data for San Luis Obispo [see Appendix 8.7], we determined that Reynolds numbers on the order of three million could be achieved with fairing lengths in the 100in range.

Additional research found data from Hoerner, plotted by Wilson [Ref. Wilson figure 5.9], which suggests that the lowest drag on a streamlined body occurs at a length to thickness ratio of approximately 3.7. Combining these results, along with minimum interference dimensions, the bulk geometry was constrained to a NACA 4-series airfoil with a length of 106 inches and a maximum width of 30 inches (see figure 5.6).

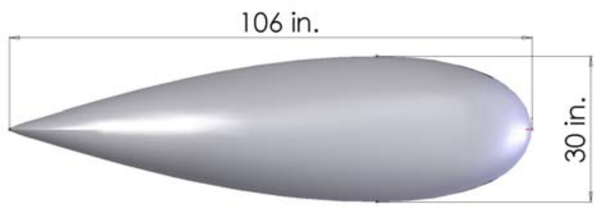


Figure 5.6: Optimized length and width dimensions of the fairing.

[5.1.4] Determination of Ground Clearance

The development of internal flow, causing high levels of drag between the road and fairing bottom was also a design concern. Because the product design specifications require the vehicle to be fast as well as agile, the vehicle is required to have a center of gravity as low as possible without sacrificing aerodynamic integrity. Again studies presented by Tamai detail that for a Torpedo style shape with a flat bottom, the ratio of ground clearance to body length is optimized at .03-.05 [Ref. Tamai 3.3.2]. For a 106.3 inch length, the minimum ground clearance was determined to be 3.2 inches.

[5.1.5] Material Selection

We selected a composite structure of carbon and aramid fibers in an epoxy matrix for fairing construction. These materials were selected because of their formability to complex geometries, lightweight construction, resistance to abrasion, and low surface roughness. While strength predictions for the material are difficult to determine due to the inconsistencies in the hand lay-up process, testing of initial material samples consistently showed the final construction using three layers of carbon and a single layer of aramid to be durable enough for the predicted loadings.

[5.1.6] Fairing Evaluation

The design was analyzed using computational fluid dynamics software to determine the theoretical aerodynamic efficiency achieved with our geometry. The results of this analysis, the details of which are presented in Appendix 8.8.1, show that for our frontal area of 886 in^2 the drag coefficient is 0.11. Using this drag coefficient, the power required to overcome aerodynamic drag at a vehicle velocity of 45mph is calculated to be 0.49hp, which satisfies the PDS requirement for aerodynamic efficiency.

The weight specification for the fairing requires that it have a weight of no more than twenty pounds. This requirement was evaluated by placing each fairing piece on a scale and then summing the measured results. Using this technique, the fairing was found to meet the weight requirements of the PDS with a total weight of 18.6 lbs.

[5.2] The Frame

[5.2.1] Overview

The frame design was determined to be a monotube recumbent tadpole trike. With this basic geometry determined, the final design was produced by creating a frame geometry which places the rider in the optimum power producing position while fitting within the fairing as detailed above, and allowing agile maneuvering.

[5.2.2] Frame Geometry

Once the basic configuration had been determined, the frame design was focused on optimizing rider power output. In a study at Colorado State University [Ref. Reiser] on the effect of backrest angles on recumbent cycling power, it was determined that backrest angles (BA in figure 5.7) of 30 degrees and 40 degrees produced the greatest power output for each rider in their study.

In this study the hip orientation (HO in figure 5.7) was held constant at 15 degrees because a previous study had held the backrest angle constant while varying hip orientation, and had determined that a hip orientation of 15 degrees produced maximum power output [Ref. Reiser]. The combination of these two results formed the foundation for the rider position. A backrest angle of 35 degrees was chosen to optimize the aerodynamic benefit of a small frontal area, while keeping the bike as short as possible, with a hip orientation of 15 degrees.

Once rider position had been decided, the industry standard for sizing recumbent bikes was used to determine the proper distance between the seat and the bottom bracket. Commercially available recumbent bicycles are matched to riders based on their x-seam measurement [Ref. Coventry], so each of the riders' x-seams were measured. This resulted in a team x-seam variation of 6 inches with median x-seam being 40.5 inches.

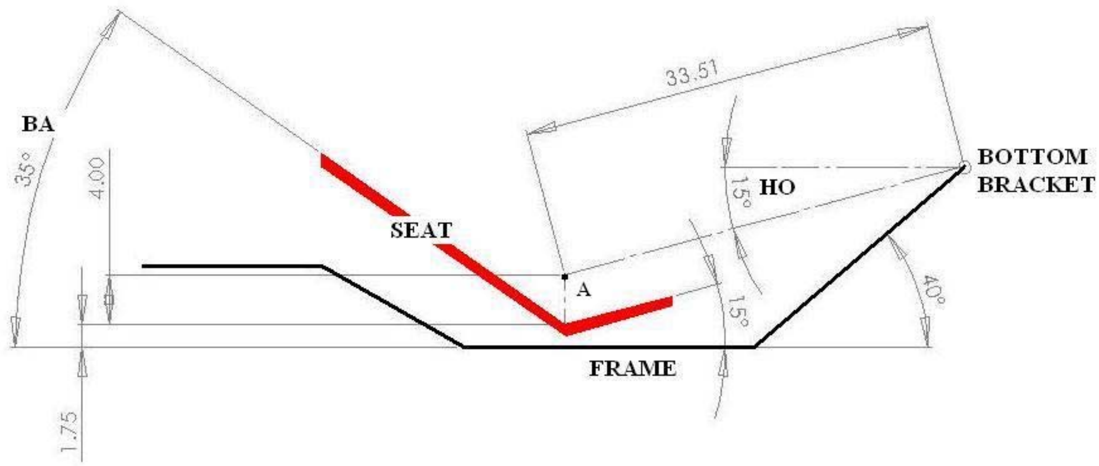


Figure 5.7: Relation of frame geometry to critical rider angles.

The frame was then optimized for the median rider, using the following parameters: 40.5 in. x-seam, 4 in. seat depth (distance between seat and point A) and 172.5 mm cranks.

With these requirements for rider ergonomics, and the requirement to fit inside the fairing, the final frame geometry was determined. The front wheel track was determined based on a combination of maneuverability requirements. Though widening the track increases stability while cornering, setting it at 29in meets the PDS requirement for cornering stability (see Appendix 8.6.4 for details) and maximizes the aerodynamic efficiency by placing the wheels inline with the fairing sides. These geometrical requirements, when combined, resulted in the frame geometry as shown in figure 5.8.

[5.2.3] Material Selection and Testing

The main options identified for frame tubing were chrome-moly steel, aluminum, and titanium. After scoring each option, 4130 chrome-moly steel was selected, based primarily on its cost, weldability, and strength.

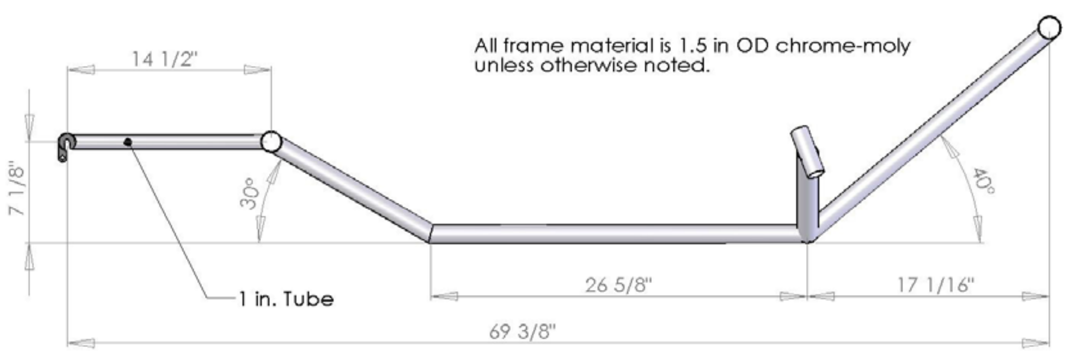


Figure 5.8: Detail of monotube design.

Industry standards and local recumbent builders were consulted to validate the theoretical analysis regarding the appropriate tube diameter and wall thickness for the main frame members. These considerations resulted in the selection of 1.5 in. outside diameter, 0.049 in. thick tube for the prototype design.

To ensure vehicle strength a specimen of the steel tube used for the base frame was sent to Koon-Hall-Adrian Metallurgical for testing. The strength of a test weld was also determined using a crystal micrograph analysis, performed by Dr. Jack Devletian of Portland State University. This testing determined that the strength of the welds far exceeds the strength of the parent material, and thus the steel tubing yield strength of 58.7 ksi was used as the governing static strength value in all calculations.

[5.2.4] Frame Testing and Evaluation

Analysis based testing of the frame was used to determine adherence of the design to PDS requirements for strength safety factors. Finite element analysis of the frame was conducted to determine the safety factors for each member. The results presented in figure 5.9 show that the lowest safety factor is 5.8, meeting the PDS minimum requirement of 5. The area with this minimum safety factor is highlighted in red in the figure. To determine the factor of safety against failure due to the oscillating pedal forces, we performed a laboratory experiment to determine the resulting stresses in the frame (see Appendix 8.8.2 for experimental details). This experiment showed that the minimum factor of safety in fatigue is 2.3 for 2,000 hours of cycling at a 60 rpm cadence. This again exceeds the PDS target of 2.

Evaluation of the frames' compliance to the weight requirement of the PDS was completed by weighing the welded frame on a scale. The as built rolling frame weighs in at 32.4 lbs, which is slightly above the PDS requirement which states that it must weigh less than 30 lbs.

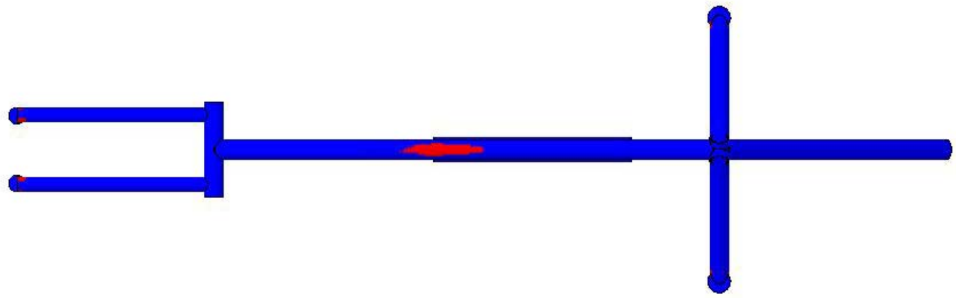


Figure 5.9: Results of FEA showing the areas with highest stress and lowest safety factor

[5.3] Mechanical Integration

[5.3.1] Overview

The vehicle mechanical integration encompasses the seat and the seat adjustment mechanism, the steering system, the braking system, and vehicle controls. The design goals of this section include the major design specifications as defined in section 3, as well as user interface ergonomics, and adjustability. The design methodology for meeting these criteria, as well as appropriate evaluations are presented here.

[5.3.2] Seat and Seat Adjustability

Rider support is handled by a carbon composite seat which was hand shaped according to the requirements of the race team. Integrated into the seat base is a steel bracket which rides on two rails welded to the main tube of the frame. The bracket was TIG welded for strength and punched with a series of holes to reduce weight (see figure 5.10). This allows the seat to slide forward to a minimum X-seam of 38in and back to a maximum of 45in.

[5.3.3] Steering and Maneuverability

Steering angles were developed using force balance techniques for each planar angle: camber, caster and toe. Each steering angle has an advantage and a disadvantage, the design process involved balancing the advantages with the disadvantages. We achieved this by balancing forces and moments applied to each wheel at its' contact patch (see Appendix 8.6.6).

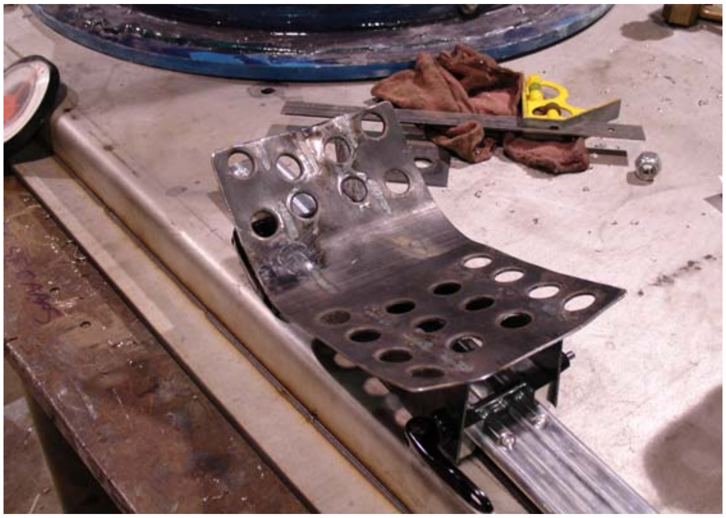


Figure 5.10: Seat bracket before integration into seat back.

Camber angles were set by optimizing the resultant force vector for a variety of cornering speeds and setting the camber angle such that the resultant cornering force would cross the centroid of the wheel, thereby reducing the thrust loads on the bearings and bending moment in the steerer tubes. These camber angles were then balanced with the needed turning radius to achieve a resulting camber angle of 3 degrees from vertical. Determining the caster angles involved balancing the wheel restoring force, steerer tube bending moment and turning radius by summing forces. This resulted in a 15 degree caster angle. We also determined the optimal toe angle theoretically, and confirmed it empirically to be 1degree in. Centerpoint steering was used to reduce wheel scrubbing by forcing the contact patch to remain stationary during rotation of the wheel, rather than traveling in an arc. This is achieved by bringing the contact patch in line with the steering axis as shown in figure 5.11.

[5.3.4] Braking

We selected left and right front Avid Ball-Bearing 5® mechanical disc brakes for their quick adjustment capability and excellent stopping power. Braking analysis was completed by calculating the stopping distance limited by interfacial friction. A second analysis was also performed to determine the forward tipping tendency during deceleration using a sum of moments. The stopping distance was found to be limited by

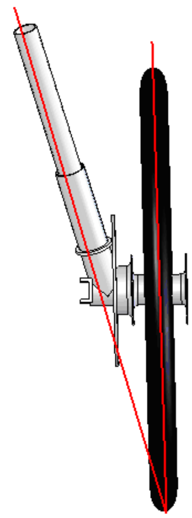


Figure 5.11: Illustration of intersection of steering axis with contact patch

the interfacial friction and not the tipping potential with a value of 8 ft from 15 mph, which meets the PDS requirements (see Appendix 8.6.5 for details).

[5.3.5] Vehicle Controls

A direct under seat steering interface design was selected as being the lightest, simplest and most adaptable method of steering the trike. Design of the interface was geometrically constrained by rider, frame, seat, and fairing dimensions. Using an under seat steering method allows for riders to quickly enter and exit the vehicle, sweep the handle bars under the seat, and maintain a comfortable and ergonomic position while racing. SRAM Rocket-Shorty twist shifters and brake levers placed at the handle bar position allowed racers to easily access the controls at all times while riding the vehicle.

[5.3.6] Mechanical Integration Evaluations

The PDS requirements for vehicle turning radius, stopping distance, and stability were all set by the minimum requirements of the ASME competition rules (see Appendix 8.3). Official evaluation of these requirements were conducted by ASME judges during the vehicle safety inspection conducted before competition. The inspection confirmed that the vehicle met and exceeded all of these requirements (see table 5.1 for details).

Requirements for the seat and adjustment system state that all team members must be able to fit in the vehicle and reach the vehicle controls with the seat adjusted for their

X-seam. Adherence to this article of the PDS was tested by having each rider move the seat into their riding position to ensure that it was comfortable for them. This test confirmed that the six inches of seat adjustment matches the six inch range of rider X-seams.

[5.4] Drive-Train

[5.4.1] Overview

The drive-train consists of the mechanical components used to transfer rider power out-put from the pedals to the road surface. A standard bicycle chain drive system was selected after initial research showing the efficiencies of such systems to be up to 98% compared to 94% for shaft drive and 95% for belt drives [Ref. Burrows]. Due to the amount of research and development companies such as Shimano and Campagnolo have conducted in this area of vehicle and design, as well as the economies of scale associated with their mass production facilities, it was deemed both impractical and uneconomical to develop custom components. Therefore, this area of vehicle design involved the selection of off-the-shelf components from various manufactures. The main PDS requirements governing component selection were weight, cost, and durability. Appropriate selections were made as described below.

[5.4.2] Component Selection

After reviewing technical specifications and costs from various manufactures, our team decided that Shimano components were both the most economical and available of the competing brands. Furthermore, personal experience has shown that all Shimano components function equally as well as higher cost components when new, with increases in cost affecting the long-term durability only. With the relatively short period of time the vehicle was to be in service, component selection was based solely on the cost and weight of each piece. Further cost incentives for a number of components were made available by team sponsors The Bike Gallery, and Chris King. Through these sponsorships a number of components were made available at discounted or no cost, and when the available components met the functionality and weight requirements as described above, they were selected for use.

Due to the unique design of the vehicle, a number of components were specially ordered from manufacturers. One such item is an under-under chain idler system from team sponsor Terracycle (see figure 5.12). While the industry standard idler consists of a stationary piece of polyethylene which the chain runs over causing large amounts of friction, the Terracycle unit uses a geared idler which rotates with the chain on sealed bearings. This system increases the efficiency of the drive-train and the service life. Custom cantilevered disk brake hubs were also ordered from team sponsor Phil Wood & Co. The use of cantilevered front hubs allows for single sided steering knuckles which significantly reduces the weight of the vehicle. A summary of the components selected for the vehicle is presented in Table 5.2.

Figure 5.12. Terracycle geared idler selected to improve drivetrain efficiency [Courtesy Robert Johnson]

Component	**Model**	**Quantity**	**Cost ea.**
Rear Derailleur	Shimano 105	1	$0.00
Bottom Bracket	Shimano 105 Octalink	1	$0.00
Cranks	Shimano Dura Ace	1	$0.00
Chain	Shimano Dura Ace	3	$28.99
Pedals	Shimano SPD	1 (set)	$0.00
Cassette	Shimano Ultegra	1	$0.00
Rear Wheel/Hub	Mavic/Shimano Ultegra	1	$190.00
Front Wheel/Hub	Mavic/Phil Wood	2	$220.00

Table 5.2: Summary of drive-train components

[5.5] Safety Systems

[5.5.1] Overview

Customer requirements and competition rules mandate a number of safety systems be designed for the vehicle. While an inclusive list of safety requirements are detailed in the PDS and table 5.1, the three most important components are detailed here.

[5.5.2] Roll-bar

Competition rules require all vehicles to have a roll-bar equivalent in strength to 1.5in OD 4130 chrome-moly tubing with a 0.049in wall thickness. Our design fulfills this requirement with the 6061-T6 roll-bar design. In addition to fulfilling the minimum strength requirement, the aluminum design is 2.1lbs lighter and 12% stronger than a similar chrome-moly design (see Appendix 8.6.2 for details).

[5.5.3] Rider Restraint

Competition rules state that all vehicles must have a rider restraint system including both lap and shoulder restraints. This requirement is met using a four-point automotive racing harness from Andover automotive. The selected harness is designed for quick length adjustments and a single lever action buckle to speed rider exchanges. The purchased unit was modified to reduce weight by converting the bolt on shoulder straps to a loop on system. The lap belts are attached to the frame using a grade eight automotive fastener.

[5.5.4] Visibility

Seeing clearly was a major concern of the race team, so we paid attention to providing ample forward and peripheral visibility. To view all areas of the racecourse a four window system covers the full range of forward and side visibility. The combined window system creates a total of 184 degrees of horizontal view and 80 degrees of vertical view (see figure 5.13). Windows were constructed of 1/32in polycarbonate for its' excellent optical properties, low weight, and impact strength 250x that of glass [Ref. Matweb].

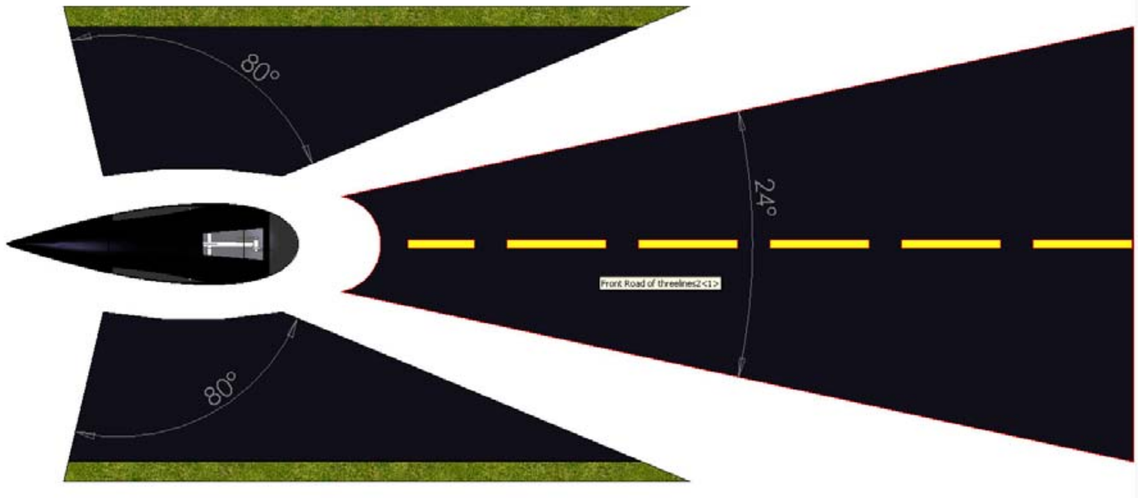


Figure 5.13: Plan View of vehicle showing distribution of horizontal view.

<u>[5.5.5] Safety Systems Evaluation</u>

The compliance of the safety systems were evaluated and deemed satisfactory by the judges during the safety inspection. The function of each system was evaluated during multiple competition incidences in which no riders were injured.

[6] Future Design Considerations

After testing the prototype in competition the design team concluded that several design modifications would improve performance. The weight of the frame could be reduced by constructing it using a lighter material such as aluminum. The length of pit stops could be shortened by designing a new hatch system with a hinge and two simple latches which could be operated from inside the vehicle. Vehicle stability could be increased by implementing a headset integrated steering damper to reduce road force inputs to the rider interface. Finally, wet weather visibility would be improved by implementing an exterior wiping system and interior resistance heater to the top window. Though the vehicle performs well as designed, these minor modifications would make the vehicle even more competitive.

[7] Conclusion

Constructing and testing the prototype has verified many aspects of our design and shown a few areas for improvement. Of the 26 PDS targets presented in Table 5.1, 23 of them were met or exceeded. By reusing the same fairing molds and constructing a new light weight frame it is believed that these speed, cost, and weight targets may be simultaneously met. The vehicle functioned well throughout competition and was able to obtain a third place finish in the 2006 ASME West Coast competition. More importantly, the prototype will provide a dynamic laboratory in which data can be collected and new ideas tested. While not perfect, the 2006 PSU VikeTrike proved to be very competitive with all other vehicles currently in production.

[8] Appendices

Appendix [8.1] : Summary of 2005 ASME West Coast Results

The 2005 ASME West Coast Challenge was held in Fresno, California in April. The following is a summary of the sprint and endurance results printed from www.asme.org/hpv.

Endurance Event Standings

VEH NO.	CL	TEAM/ ENTRY	LAPS COMPL.	ORDER OF FINISH	PENALTY LAPS	NET LAPS COMPL.	ENDURO EVENT FINISH	POINTS
1	S	Cal Poly San Luis Obispo	45		0	45	2	28.3
2	S	Montana Tech	29		0	29	13	10.0
3	S	San Jose State University	40		0	40	6	21.7
4	S	Central Washington University	38		0	38	7	20.0
9	S	California State University, Fresno	31		0	31	12	11.7
11	S	Oregon State University	43		0	43	3	26.7
12	S	University of Nevada, Reno	41		0	41	5	23.3
13	S	Colorado State University	0		0	0		0.0
14	S	University of Hawaii-Manoa	29		0	29	14	8.3
16	S	California State University, Chico	45		0	45	1	30.0
19	S	South Dakota School of Mines and Technology	27		0	27	16	5.0
21	S	San Jose State University	29		0	29	15	6.7
22	S	San Francisco State University	33		0	33	10	15.0
24	S	University of Missouri-Rolla	42		0	42	4	25.0
26	S	California State University, Long Beach	25		0	25	17	3.3
29	S	UCLA	34		0	34	8	18.3
55	S	University of Colorado Denver	32		1	31	11	13.3
88	S	Seatle University	33		0	33	9	16.7

2005 ASME HPV West Coast Standings in the Sprint Event

SINGLE RIDER SPRINT STANDINGS

VEH NO.	CL	TEAM/ ENTRY	Men's MPH	MEN'S FINISH	Women's MPH	WOMEN'S FINISH	AVERAGE FINISH	PLACE	POINTS
1	S	Cal Poly San Luis Obispo	43.4	1	34.7	1	1.0	1	30.0
2	S	Montana Tech	26.5	13	23.7	10	11.5	10	15.0
3	S	San Jose State University	32.4	7	26.0	8	7.5	7	20.0
4	S	Central Washington University	28.8	11	21.9	12	11.5	11	13.3
9	S	California State University, Fresno	33.1	6	26.7	6	6.0	6	21.7
11	S	Oregon State University	34.6	5	29.2	5	5.0	5	23.3
12	S	University of Nevada, Reno	38.6	3	32.4	3	3.0	3	26.7
13	S	Colorado State University	23.3	18	18.1	17	17.5	17	3.3
14	S	University of Hawaii-Manoa	24.6	16	20.3	13	14.5	15	6.7
16	S	California State University, Chico	36.8	4	30.4	4	4.0	4	25.0
19	S	South Dakota School of Mines and Technology	32.3	8	25.2	9	8.5	9	16.7
21	S	San Jose State University	28.1	12	18.8	16	14.0	14	8.3
22	S	San Francisco State University	31.7	9	26.5	7	8.0	8	18.3
24	S	University of Missouri-Rolla	43.3	2	34.3	2	2.0	2	28.3
26	S	California State University, Long Beach	29.1	10	19.2	15	12.5	13	10.0
29	S	UCLA	23.9	17	18.0	18	17.5	18	1.7
55	S	University of Colorado Denver	24.9	14	22.9	11	12.5	12	11.7
88	S	Seatle University	24.8	15	19.2	14	14.5	16	5.0

Appendix [8.2]: Product Design Specifications

Scope

This Product Design Specification (PDS) clearly defines the following for the PSU-HPV:

- The design constraints (metrics and targets)
- The priority of constraints

Customer Identification

External Customers	Internal Customers
Racing Team, ASME Judges	The Race Support Crew
ASME Student Section,	The Fabrication Crew
MME Department, Sponsors	

The primary customer of the HPV Project is the 2006 Race Team, whose members are the End Users of the product. The ASME Judges are also considered a primary external customer, as it is their judgment of the design's compliance, performance, and safety that determines the vehicle's ranking at the competition.

The ASME Student Section benefits from any and all progress which this year's team makes towards establishing an HPV team here at Portland State. This year's completed product will serve as a foundation for future teams to work from, and the knowledge gained will provide future design teams a valuable resource from which to draw. The Department of Mechanical Engineering benefits from the prestige and positive public relations of fielding a competitive team.

The Fabrication and Race Support Crews are also key internal customers. The Fabrication Crew has customer needs that include consideration of labor time and effort, as well as availability of technology required. The Race Support Crew has needs that pertain to the service and operation of the vehicle during competition.

Customer Feedback

Customer feedback for this project comes primarily from discussions with the PSU-HPV Race Team, as well as consultations with industry experts.

Product Design Specifications

High Priority

Criterion	Compliance	
Requirement	Must be legal to enter into competition	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Turning radius	Feet	<= 25
Stopping distance	Mph, Feet	From a speed of 15 mph to 0 mph in 20 feet or less
Straight line stability	Degrees per foot	0°/100’
Vehicle identification	Yes/No	Vehicle must be properly labeled
**also see safety criteria		
Target Basis	2006 Published Rules (See Appendix 8.3)	
Verification Method	Direct Comparison to rule book, judging at competition.	

Criterion	Performance	
Requirement	Vehicle must be light	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Frame weight	lbs	<= 30
Fairing weight	lbs	<= 20
Target Basis	Power Availability, vehicle weight of 2005 competition winners	
Verification Method	Measurement with scale	

Criterion	Performance	
Requirement	Aerodynamic efficiency in forward motion	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Power to overcome aero drag	hp	<= .5
Target Basis	Theoretical Research of available power [ref. Wilson pg. 44]	
Verification Method	Determination of Drag Coefficient using CFD	

Criterion	Cost	
Requirement	Must be affordable to produce	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Total Fabrication Cost	$	< 5000
Target Basis	Total Available Funds	
Verification Method	Measurement, Documentation	

Criterion	Safety	
Requirement	Rider Safety	
Primary Customer	PSU-HPV Race Team	
Metrics & Targets	**Metric**	**Target**
Safe Operating Conditions	Yes/No	Free of sharp edges and pinch points
Roll-over protection	Yes/No	Rider must not touch the ground in case of roll-over
Rider Restraint	Yes/No	Harness system must hold rider in vehicle during collision
Visibility*	Degrees of horizontal and vertical view	Horizontal >90 degrees Vertical >50 degrees
Target Basis	Competition Rules, *Survey of rider preferences	
Verification Method	Inspection, Measurement	

Medium Priority

Criterion	Performance	
Requirement	Velocity	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Exceed top speed of last year's winner	Miles per hour	> 45
Exceed average speed of last years endurance winner	Miles per hour	> 20
Target Basis	2005 Race Results	
Verification Method	Measurement by time trial	

Criterion	Life in Service	
Requirement	Needs to last through testing, training, competition	
Primary Customer	PSU-HPV 2006 Race Team, Future PSU-HPV Race Teams	
Metrics & Targets	**Metric**	**Target**
	End of service date	April 30, 2006
Target Basis	Budget constraint, purpose of this HPV	
Verification Method	Performance at competition, post race inspection	

Criterion	Documentation	
Requirement	ASME/ Senior Capstone Papers	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
		Documentation of engineering process
Target Basis	ME 492/3 Course requirements	
Verification Method	Measurement	

Criterion	Materials	
Requirement	Adequate Strength against failure	
Primary Customer	PSU-HPV Race Team	
Metrics & Targets	**Metric**	**Target**
Static safety factors	Safety Factor (S_y/σ_y)	>=5
Fatigue safety factor	SF for 2,000 hours at 60rpm	>= 2
Target Basis	Industry standard, Engineering Analysis	
Verification Method	Deflection and Strain Testing	

Criterion	Performance	
Requirement	Stability	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Vehicle does not flip under normal turning conditions	Mph / ft radius	10 / 20
Target Basis	Performance of 2005 competitors	
Verification Method	Empirical Analysis, Vehicle Testing	

Criterion	Aesthetics	
Requirement	Visual Appeal	
Primary Customer	PSU-HPV 2006 Race Team, PSU-HPV Sponsors	
Metrics & Targets	**Metric**	**Target**
Fairing Appearance	Unquantifiable – Subject to judges interpretation	Clean lines, Smooth surface of uniform school colors
Frame Appearance	Unquantifiable – Subject to judges interpretation	Frame to be powder coated
Target Basis	Competition research	
Verification Method	Competition results	

Low Priority

Criterion	Documentation	
Requirement	Beginning of a legacy project	
Primary Customer	PSU-HPV 2006 Race Team, Future PSU-HPV Race Teams	
Metrics & Targets	**Metric**	**Target**
Level of documentation	Yes/No	Documentation of engineering process for future PSU-HPV Teams
Target Basis	Increase PSU MME programs awareness	
Verification Method	Response from 2007 team	

Criterion	Maintenance	
Requirement	Minimal maintenance	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Before competition	Hours	< 1
During competition	Minutes	0
Target Basis	Comparison to racing vehicles, goals set as ideal	
Verification Method	Measurement	

Criterion	Rider Comfort	
Requirement	Rider ergonomics	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Minimum width of shoulder box	Inches	>= 20
Maximum x-seam adjustability	Inches	>= 45
Minimum x-seam adjustability	Inches	<= 39
Cockpit temperature	˚F above atmospheric	<= 10˚
Target Basis	Race team measurements	
Verification Method	Measurement	

Criterion	Performance	
Requirement	Rider exchange time	
Primary Customer	PSU-HPV 2006 Race Team	
Metrics & Targets	**Metric**	**Target**
Time to enter and exit	seconds	<= 10
Target Basis	Research in competitors designs	
Verification Method	Measurement	

Appendix [8.3]: Summary of 2006 ASME Competition Rules

Printed from:

http://www.asme.org/hpv/summaryofrules.html

Complete rules are available at:

http://files.asme.org/asmeorg/Events/Contests/HPV/4781.pdf

Sponsored by the American Society of Mechanical Engineers

ASME sponsors the Human Powered Vehicle Competition in hopes of finding a design that can be used for everyday activities ranging from commuting to and from work to going to the grocery store. Senior engineering students can use this competition for their capstone project and with their efforts design and construct a fast, sleek, and safe vehicle capable of road use.

The competition includes three classes of vehicles.

- Single Rider - operated and powered by a single individual
- Multi-rider - operated and powered by two or more individuals
- Utility - vehicle designed for every-day transportation for such activities as commuting to work or school, shopping trips, and general transportation

Single Rider and Multi-rider vehicles will participate in three events: Design, Sprint, and Endurance. Utility vehicles will participate in two events: Design and Utility Endurance.

Fairing

All vehicles in all classes of competition are required to have a full or partial aerodynamic fairing. This fairing must cover 1/3 of the frontal area of the vehicle and be built such that it clearly shows the provided number assigned to the vehicle and ASME logo. The number and logo must be displayed on every fairing in front of the rider and must be visible from both sides of the vehicle.

Safety

All vehicles and teams in all classes must abide by all the safety requirements.

1. Make a complete stop in a distance of 20 feet or less from a speed of 15 miles per hour
2. Travel is a straight line for 100 feet
3. Negotiate a turn within a 25-foot radius
4. Provide rollover protection for riders and stokers, equivalent to chrome-molybdenum steel tubing with an outer diameter of 1.5 inches and a wall thickness of no less than 0.049 inches
5. Wear helmets that meet given standards
6. Wear seat belts or shoulder harnesses, in accordance to the rulebook
7. Show that all surfaces of the vehicle, both exterior and interior region of the rider(s), are free from sharp edges and protrusions

Vehicles found unsafe during inspection or anytime of the competition will be removed from the competition until the problem has been resolved.

Energy Storage
The use of energy storage devices by non-utility vehicles is prohibited. Normal operating components involved in the drive train are specifically permitted in as much as their design is not primarily influenced by energy storage considerations. Utility vehicles will be allowed to store regenerative energy. Prior to every event, they must show that their energy-storing device has no initial energy stored. All of the energy stored by the device must be a result of the vehicle being in motion.

Design
The design event will include vehicles from all three classes. Judges will consider both the formal written report and the oral presentation when reviewing vehicle designs. There will be an emphasis on originality and the soundness of the design. The focus will be the new work that has been completed in the last year.

Sprint
The Sprint event will include Single Rider and Multi-rider vehicles. Approximately four hours of competition will be ran on a single track such that everyone will be capable of obtaining a sprint time. The timed portion of the course is a 100 meter straight a way. There will be a preceding distance of 300 to 400 meters for vehicles to gain speed before entering the timed portion, as well as a minimum of 200 meters at the end for the vehicles to slow down.

Endurance
The Endurance event will involve all three categories. Single Rider and Multi-rider vehicles will compete in grand prix style road races of approximately 65 kilometers (40 miles). Vehicles must start the event with female rider(s) who must complete at least 5 kilometers. No individual can compete in the vehicle for more than 20 kilometers, and all laps by any individual must be consecutive. When the lead vehicle crosses the finish line, each team will be allowed to finish the lap it is on to end the competition.

The Utility Endurance event includes Utility vehicles only. The course will be a distance of approximately 10 kilometers and will include obstacles such as a driveway entry ramp, speed bumps, stop signs, and "head in" parking. Along with these obstacles, the rider will be required to dismount his/her vehicle to pick up parcels or packages (29.2 cm x 17.2 cm x 39.3 cm) as well as drop them off. The event is over when all vehicles have completed the course.

The specifications for each event, including the mandatory use of female riders, can be found in the rulebook. How the scores are tallied for each event and vehicle can also be found there. Forms for registration, certifications, and eligibility, along with others are all included in the appendix of the rulebook. To avoid disqualification competing teams are strongly encouraged to become familiar with all the rules and regulations.

Appendix [8.4]: Internal and External Search Summaries

Internal Search

The internal search process consisted of two months of weekly brainstorming sessions during which design concepts were generated and refined. Presented here is an example of the alternative designs generated for the top level decisions of rider position, fairing type, and wheel configuration.

A delta trike option using rear wheel steering is shown in figure 8.4.1. With a single front and dual rear wheels, long wheel base delta trikes have increased high speed stability but are at a significant disadvantage when cornering. This front wheel drive concept also has increased drive-train efficiency due to its' short chain path but reduced aerodynamic efficiency due to its' maximum width occurring at the tail end.

With a more conventional rider position figure 8.4.2 shows a forward leaning upright concept. Using only a small front fairing and rear tail-box the design has the potential to be very lightweight and attractive to riders already comfortable with standard bicycles. Much of the efficiency of this design is lost however, with the integration of the required roll-bar and harness system.

Using radical rider positioning for increased road visibility, the prone design concept of figure 8.4.3 was proposed for its potential to be both efficient and simple. With the rider's hands positioned at the front wheel and feet around the rear wheel, this concept has the potential to use direct drive and steering systems. It was ultimately rejected due to stability concerns and the required rider support system placing pressure on the chest and restricting breathing.

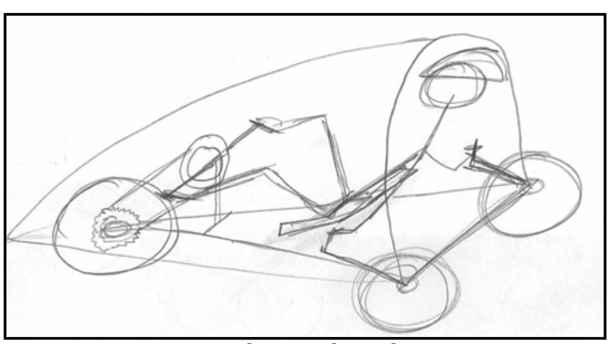


Figure 8.4.1. Recumbent Delta trike concept

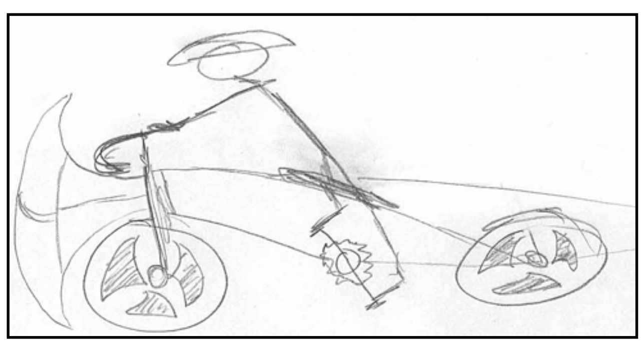


Figure 8.4.2. Upright partially faired concept

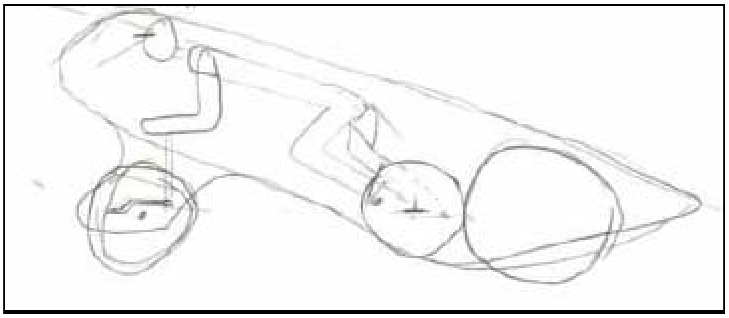


Figure 8.4.3 Two wheeled prone concept

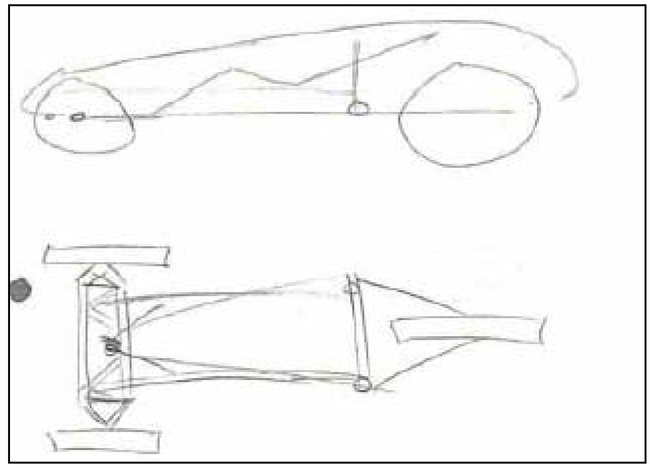


Figure 8.4.4. Tadpole trike concept used in design

Chosen as the final design concept, figure 8.4.4 shows a delta trike which places the rider in a recumbent position. With two wheels in front and a single in the rear, vehicle stability while cornering is improved over the alternative delta design. The dual front wheel system is also conducive to aerodynamic design. While not as stable at high speeds as a delta design, proper steering geometry and rider experience is likely to make up for this disadvantage.

External Search

As part of the pre-design brainstorming process, the design team conducted an external search of existing HPVs and similar applicable technologies. Each member heavily researched technologies in specific components of the project, as well HPVs as a whole. By conducting a comprehensive search we were able to identify the successful components of existing designs, as well as possible niches in the market where other designs have failed to succeed. In order to ensure that our search encompassed all available technologies, including those outside of the ASME HPV competition, the design team also researched production bike models and HPVs built by the international HPV racing community. By including theses three sectors of the market and surveying all available aspects of the designs, it is believed that we were able to successfully build off of, and excel past, the performance of existing vehicles. Included below is a sample of the top vehicles of each of the three categories surveyed.

2005 ASME West Coast Competition

Champions of the ASME West Coast Challenge three years running, Cal-Poly's vehicles are highly refined through years of trial an error. Figure 8.4.5 shows their 2005 entry which features a light and strong chassis via a carbon fiber monocoque frame integrated into its' full aerodynamic shell. Additional unique features include a front wheel drive system using an internally geared front hub and a chain path which requires the chain to flex during turning.

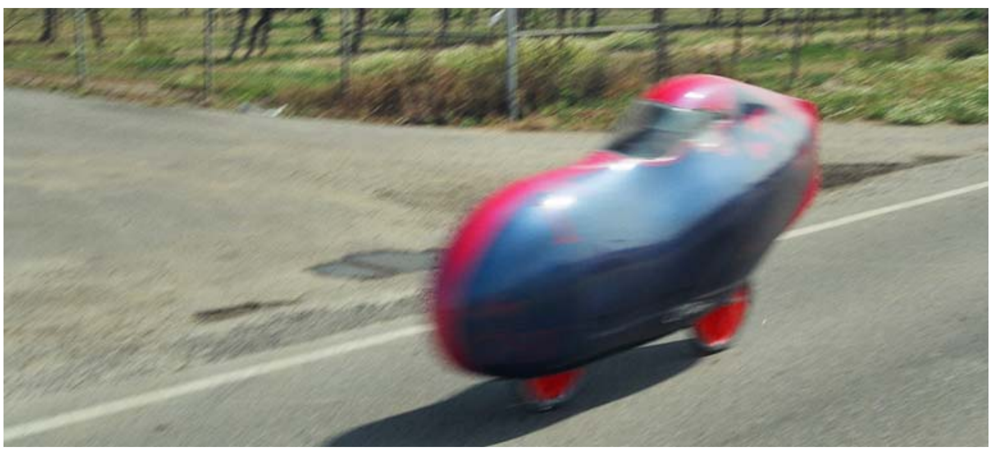


Figure 8.4.5: Cal-Poly, three year ASME West Coast Champion.

Second place at the 2005 west coast competition and multiyear champions of the east coast competitions, the University of Missouri at Rolla 2005 entry is shown in figure 8.4.6. Using carbon fiber wheel disks to complement the full fairing, this vehicle's focus on aerodynamic design led it to first and second place finishes in the 2005 east and west coast competitions respectively. Also note that the top of the fairing is removable to preserve rider comfort during hot conditions.

The 2005 west coast endurance event was won by a trike produced by Chico State (see figure 8.4.7). With a vehicle weight of 52lbs and an average speed during the endurance event of 22mph, this vehicle appeared to excel due its ability to hold speed through corners. Their excellent visibility also seemed to add to rider confidence.

Figure 8.4.6: UM Rolla, second place overall in 2005 ASME west coast comp.

Figure 8.4.7: Chico states' 2005 West Coast entry, The MochaChico

International Racers

Developed and raced in Australia, the TriSled (see figure 8.4.8) is a tadpole trike built specifically for racing. With a combination fiberglass and fabric fairing, this vehicles' aerodynamic efficiency is likely improvable by adding wheel disks and a hard top fairing. Of note is the unique fairing attachment system which uses a system of chrome-moly hoops to produce a rigid attachment and reduce fairing vibration. Holder of multiple world records including the top speed record, the Varna shown in figure 8.4.9 is the current bench mark in HPV speed design. With a weight over 80lbs this design places top priority on stiffness and aerodynamics. Power is transferred

Figure 8.4.8: The TriSled, and Australia tadpole trike [Ref. www.recumbents.com]

through a front wheel drive system attached to a steel frame. Lack of adjustability to rider geometry is a significant drawback to this design.

Production Models:

A highly refined model, the Go-One shown in figure 8.4.10 was developed with commuting in mind. With a sealed cabin, cargo storage, and blikers, this vehicle is the current standard in velomobiles. An integrated tub-frame adds to the stiffness and light weight of this vehicle.

The aluminum space frame of the Catrike Road shown in figure 8.4.11 adjusts to multiple rider sizes with an adjustable boom to extend the x-seam. Three 20in wheels produce a compact package and center point steering with Ackerman compensation produce a stable rider platform.

Figure 8.4.9: The Varna, current world record holder for speed [Ref: Varnahandcycles.com].

Figure 8.4.10: The Go-One sets the current standard for velomobiles [ref: www.recumbents.com]

Figure 8.4.11: The Catrike Road uses an adjustable boom.

Appendix [8.5]: Concept Scoring Matrix

Following is the concept scoring matrix used in evaluation and comparison of various design alternatives. Those options receiving the highest scores were selected for use in the 2006 PSU HPV and are highlighted below in yellow.

Scoring System:	5: amazing 4: good 3: average 2: poor 1: terrible 0: not applicable	Compliance	Weight	Aerodynamic Benefit	Cost	Ease of Fabrication	Low C.O.G.	Comfort	Power Output/ Efficiency	Stability	Ease of Operation	Total Score
Subsystem	Priority:	0.2	0.18	0.175	0.15	0.15	0.1	0.05	0.05	0.1	0.05	
Number of Riders												0
	single	5	5	4	4	4	3	0	3	4	4	4.825
	tandem	5	2	2	2	3	3	0	4	3	3	3.4
Frame Material												0
	Aluminum	3	4	0	3	3	0	0	3	0	0	2.35
	Steel	4	3	0	5	4	0	0	4	0	0	2.875
	Titanium	4	5	0	2	2	0	0	4	0	0	2.475
	Composite	3	5	0	2	2	0	0	3	0	0	2.225
Fairing Material												0
	Carbon Fiber	3	4	4	3	2	0	3	4	3	3	3.55
	Fiberglass	3	3	3	3	3	0	3	3	3	3	3.3
	Corrugated Plastic	3	3	3	3	2	0	3	3	3	3	3.15
	Frame and Fabri	3	4	3	3	3	0	4	2	3	3	3.475
Drive wheel												0
	Rear-Wheel	3	3	3	4	4	3	4	3	0	3	3.65
	Front-Wheel	3	3	3	3	3	3	3	4	0	3	3.35
Drivetrain												0
	Chain	3	4	4	4	5	3	0	5	0	5	4.15
	Belt	3	3	3	2	2	3	0	4	0	4	2.95
	Shaft	3	2	4	2	2	4	0	3	0	3	2.95
	Direct	3	4	3	2	3	2	0	3	0	2	3.025
Roll Bar												0
	None	1	5	0	5	5	0	0	0	0	0	2.575
	CrMo	5	3	0	5	5	3	0	0	0	0	3.325
	Composite	4	5	0	4	3	4	0	0	0	0	3.125
	Aluminum	3	4	0	3	3	3	0	0	0	0	2.5

Appendix [8.6]: Analysis Based Decision Examples

Index of Solutions:

[8.6.1] Percent Fairing Coverage Analysis

Summary

The objective of this analysis is to determine the optimum fairing coverage for the 2006 PSU HPV. The competition rules state that a minimum of 33% frontal area coverage be used (see figure 8.6.1 for an example), but no maximum allowable is set. While it is known that aerodynamic efficiency increases with increased fairing coverage [Ref. Wilson pg. 188], rolling resistance also increases with fairing coverage due to the increased weight.

Upon analyzing the relationship between the two drag forces by comparing two vehicle models with known properties, it is determined that the increased weight of the fairing pays off in overall efficiency at all velocities over 35 mph. It is therefore suggested that for a target velocity of 45mph a full fairing should be used (see figure 8.6.2 for an example).

Figure 8.6.1: Example of vehicle with partial fairing coverage [Courtesy Don Mueller]

Figure 8.6.2: Example of vehicle with full fairing coverage [Ref: www.recumbents.com]

Given:

A vehicle fairing is to be designed to cover the PSU HPV. A minimum coverage of 1/3 of the frontal area is set by the competition. The team may however, increase the size of the fairing to complete coverage is desired. The following data is given by Wilson for vehicles with 1/3 coverage and full coverage.

Coverage	Frontal Area (m^2)	Vehicle Weight (kg)	$C_{d,fa}$	C_r
1/3	0.40	85	0.52	0.00435
Full	0.42	105	0.11	0.00435

Find:

-The percentage of power used to overcome aerodynamic drag for each configuration at 45mph.

-The difference in power requirement for each vehicle at 45mph.

-Based on these results, recommend a percentage coverage for use on the PSU HPV.

Assumptions:

-The density of air is 1.225 kg/m^3 (see appendix 8.7)

Solution:

-The force produced by aerodynamic drag, as stated by Wilson, is given by:

$$AeroDrag = (C_d A)(\frac{1}{2}\rho V^2) \qquad \textit{equation 1}$$

-Where C_d is the drag coefficient, A is the appropriate area, ρ is the fluid density, and V is the velocity of the fluid relative to the vehicle. Similarly the force produced by rolling resistance is given by Wilson as:

$$RollingDrag = \left[C_{rr1} + \left(\frac{n}{3}\right)\left(6.6 * 10^{-5}\frac{1}{mph}\right)(V))\right](W) \qquad \textit{equation 2}$$

-Where C_{rr1} is the zero speed rolling resistance coefficient, n is the number of wheels, and W is the vehicle weight. Multiplying these equations for drag force by the vehicle velocity we obtain expressions for the power required to overcome each component.

$$AeroPower\,\mathrm{Re}\,quired = (C_d A)(\frac{1}{2}\rho V^3) \qquad \textit{equation 3}$$

$$RollingPower\,\mathrm{Re}\,quired = \left[C_{rr} + \left(\frac{1}{n}\right)\left(6.6*10^{-5}\frac{1}{mph}\right)(V)\right](W)(V) \quad \textit{equation 4}$$

-The given values may be plugged into equations 3 and 4 above, with units converted properly, and solved for a variety of velocities. This was done using spreadsheet software and is presented in figures 8.6.3 and 8.6.4 below.

		1/3 faired				
		Drag Coef.	0.52			
		Frontal area (m^2)	0.4			
		weight (N)	833			
		C_{R^*}	0.00435			
		Density (kg/m^3)	1.225			
Velocity						
(mph)	**(m/s)**	Aero drag (N)	Rolling drag (N)	Total Drag (N)	Percent Aero Drag	Total Drag (W)
4.5	2	0.5	3.8	4.3	11.9	8.594
8.9	4	2.0	4.0	6.0	34.0	23.959
13.4	6	4.6	4.1	8.7	52.7	52.211
17.9	8	8.2	4.3	12.4	65.6	99.463
22.4	10	12.7	4.4	17.2	74.1	171.832
26.8	12	18.3	4.6	23.0	79.9	275.433
31.3	14	25.0	4.8	29.7	84.0	416.381
35.8	16	32.6	4.9	37.5	86.9	600.791
40.3	18	41.3	5.1	46.4	89.0	834.778
44.7	20	51.0	5.3	56.2	**90.6**	1124.458
49.2	22	61.7	5.4	67.1	91.9	1475.945
53.7	24	73.4	5.6	79.0	92.9	1895.356
58.2	26	86.1	5.8	91.9	93.7	2388.805
62.6	28	99.9	5.9	105.8	94.4	2962.406

Figure 8.6.3. Rolling and aerodynamic drag calculations for a 1/3 faired vehicle

		Fully faired*				
		Drag Coef.	0.11			
		Frontal area (m^2)	0.42			
		weight (N)	1029			
		C_{R^*}	0.00435			
		Density (kg/m^3)	1.225			
Velocity						
(mph)	**(m/s)**	Aero drag (N)	Rolling drag (N)	Total Drag (N)	Percent Aero Drag	Total Drag (W)
4.5	2	0.1	3.8	3.9	2.9	7.862
8.9	4	0.5	4.0	4.4	10.2	17.756
13.4	6	1.0	4.3	5.3	19.3	31.658
17.9	8	1.8	4.6	6.4	28.1	51.545
22.4	10	2.8	5.1	7.9	35.6	79.396
26.8	12	4.1	5.7	9.8	41.7	117.187
31.3	14	5.5	6.4	11.9	46.5	166.895
35.8	16	7.2	7.2	14.4	50.3	230.497
40.3	18	9.2	8.1	17.2	53.2	309.971
44.7	20	11.3	9.0	20.4	**55.6**	407.294
49.2	22	13.7	10.1	23.8	57.5	524.443
53.7	24	16.3	11.3	27.6	59.0	663.394
58.2	26	19.1	12.6	31.8	60.2	826.126
62.6	28	22.2	14.1	36.2	61.2	1014.616

Figure 8.6.4. Rolling and aerodynamic drag calculations for a fully faired vehicle

Percent aerodynamic drag at 45mph for full fairing = 55.6%

Percent aerodynamic drag at 45mph for 1/3 fairing = 90.6%

Total drag for full fairing at 45mph = 407.3 W

Total drag for 1/3 fairing at 45mph = 1124.5 W

Reduction in power requirement using a full fairing = 717.2 W

Conclusion:

The performance of the PSU HPV would be greatly improved by the use of a full fairing as the reduction in aerodynamic drag more than offsets the increase in rolling resistance.

[8.6.2] Roll Bar Material Selection Analysis

Summary

The objective of this analysis is to determine the optimum material for use in construction of the 2006 PSU HPV rollbar. The competition rules state that a roll bar with strength equivalent to that of 4130 chrome-moly tubing with an outside diameter of 1.5in and a wall thickness of 0.049in (see figure 8.6.5). In an effort to reduce the weight of this component, an analysis of the weight of equivalent strength aluminum tubing was completed (see figure 8.6.6).

After analytically determining the weight of chrome-moly and aluminum tubing of equivalent strength, aluminum tubing is recommended for vehicle construction. The use of this alternative material is predicted to reduce the weight of the roll bar by 2.1lbs.

Figure 8.6.5: Example of vehicle with a Chrome-Moly roll bar

Figure 8.6.6: Example of vehicle with an equivalent strength aluminum roll bar

Given:

Material for the roll-bar of the 2006 PSU HPV is to be sized. Competition rules require a minimum material strength equivalent to that of 4130 chrome-moly tubing having an outside diameter of 1.5in and a wall thickness of 0.049in. The current roll-bar design requires a total of 84in of tubing with multiple mitered and welded joints. Current material options include: 4130 chrome-moly, 6061-T6 Aluminum, and 6061-T6 Aluminum with post weld heat treating.

Find:

-Determine the weight of each option and make a recommendation for material selection.

Assumptions:

-The primary loading mode is bending.

Solution:

-The material properties of the various options, as are [Ref. AlcoTechnics]:

Material	Density (lb/in^3)	Yield Strength (ksi)	Tensile Strength (ksi)
4130	0.283	60.5	95
6061-T6	0.0975	19	30
6061-T6 W/PWHT	0.0975	40	45

-Using these material properties, the failure load of the chrome-moly must first be determined. For strength comparisons, the simple three point bending model will be used as shown in figure 8.6.7.

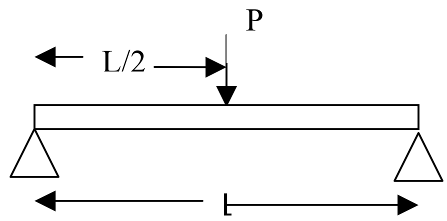


Figure 8.6.7: Three point bend schematic.

-For the three point bend, the maximum stress is calculated using beam theory as given in equation 1.

$$\sigma_{\max,bending} = \frac{My}{I} \qquad \text{(Ref. Gere)} \quad \textit{equation 1}$$

-Where the maximum moment is given by:

$$M_{\max} = \left(\frac{P}{2}\right)\left(\frac{L}{2}\right) = \frac{PL}{4} \qquad \text{(Ref. Gere)} \quad \textit{equation 2}$$

- Then for a cylindrical tube we can calculate the geometrical properties of the chrome-moly using:

$$A = \pi(r_o^2 - r_i^2) \qquad \textit{equation 3}$$

$$I_x = I_y = \frac{\pi}{64}(d_o^4 - d_i^4) \qquad \text{(Ref Gere)} \quad \textit{equation 4}$$

- And combining equations 1 and 2 we find the magnitude of the maximum stress in the three point bend test to be:

$$\sigma_{\max,bending} = \frac{My}{I} = \frac{PLy}{4I} \qquad \textit{equation 5}$$

-Substituting the geometrical and material properties for the chrome-moly into equation 5 and solving for the loads at which yielding and failure occurs:

$P_{max,yielding}$ = 1584lbs **$P_{max,failure}$ = 2484lbs**

-Plugging these values for the maximum yielding and failure loads back into equation 5, along with the material properties of each of the two aluminum options, the minimum moments of inertia required to avoid failure of each material may be determined. These moments of inertia may then be converted to required wall thicknesses based on commonly available outside tube diameters. This process was completed using spreadsheet software and the results are presented in figures 8.6.8 and 8.6.9.

AL as welded		OD (in)	Req ID (in)	Wall (in)	ID (in)	Wall (in)	Weight (lbf)
UTS (psi)	30000	1.5	1.06	0.22	1	0.25	8.0405137
Density (lb_f/in^3)	0.0975	1.625	1.30	0.16	na	na	na
Load (lbf)	2484.8	1.75	1.49	0.13	1.25	0.25	9.6486164
YS (psi)	19000	2	1.82	0.09	1.75	0.125	6.0303853
		2.25	2.11	0.07	2	0.125	6.8344366
		2.5	**2.39**	**0.05**	**2.37**	**0.065**	**4.0723594**
		2.75	2.66	0.04	2.62	0.065	4.4904661
		3	2.93	0.04	2.87	0.065	4.9085728

Figure 8.6.8: Acceptable options for aluminum tubing in the as welded condition.

AL after HT							
UTS (psi)	45000	OD (in)	Req ID (in)	wall (in)	ID (in)	Wall (in)	Weight (lbf)
Density (lb_f/in^3)	0.0975	1.5	1.26	0.12	1.25	0.125	4.4222825
Load (lbf)	2484.8	1.625	1.43	0.10	1.375	0.125	4.8243082
YS (psi)	40000	1.75	1.59	0.08	1.5	0.125	5.2263339
		2	**1.88**	**0.06**	**1.87**	**0.065**	**3.236146**
		2.25	2.16	0.04	2.12	0.065	3.6542527
		2.5	2.43	0.04	2.37	0.065	4.0723594
		2.75	2.69	0.03	2.62	0.065	4.4904661
		3	2.95	0.02	2.87	0.065	4.9085728

Figure 8.6.9: Acceptable options for aluminum tubing with post weld heat-treatment to T-6 condition.

Weight of chrome-moly option: 5.3lb

Weight of lightest aluminum option: 3.2lb

Associated dimensions of aluminum: 2in OD. with 0.065in wall.

Conclusion:

The weight of the roll-bar may be reduced by 2.1lb by using aluminum with post weld heat-treatment. In addition the component would achieve a 12% increase in failure strength.

[8.6.3] Tie-Rod Optimization

Summary:

The objective of this analysis is to minimize the weight of the tie-rod. The tie-rod is a simple cylindrical member with threaded ends as shown in figure 8.6.10 below. The results of this optimization analysis will be manufacturing specifications for the lightest possible tie-rod meeting the strength requirement. These specifications will include everything required to manufacture the tie-rod including the outside diameter, wall thickness, and material composition.

The completed analysis shows that the geometry which is both the lightest and the cheapest to manufacture is a 6061-T6 aluminum cylinder with an outside diameter of 0.375in and wall thickness of 0.08in. This geometry meets the strength requirements and may be directly tapped to accept the specified tie-rod ends. In addition, this design is stronger and lighter than a number of steel tie-rods available in consumer products of similar function.

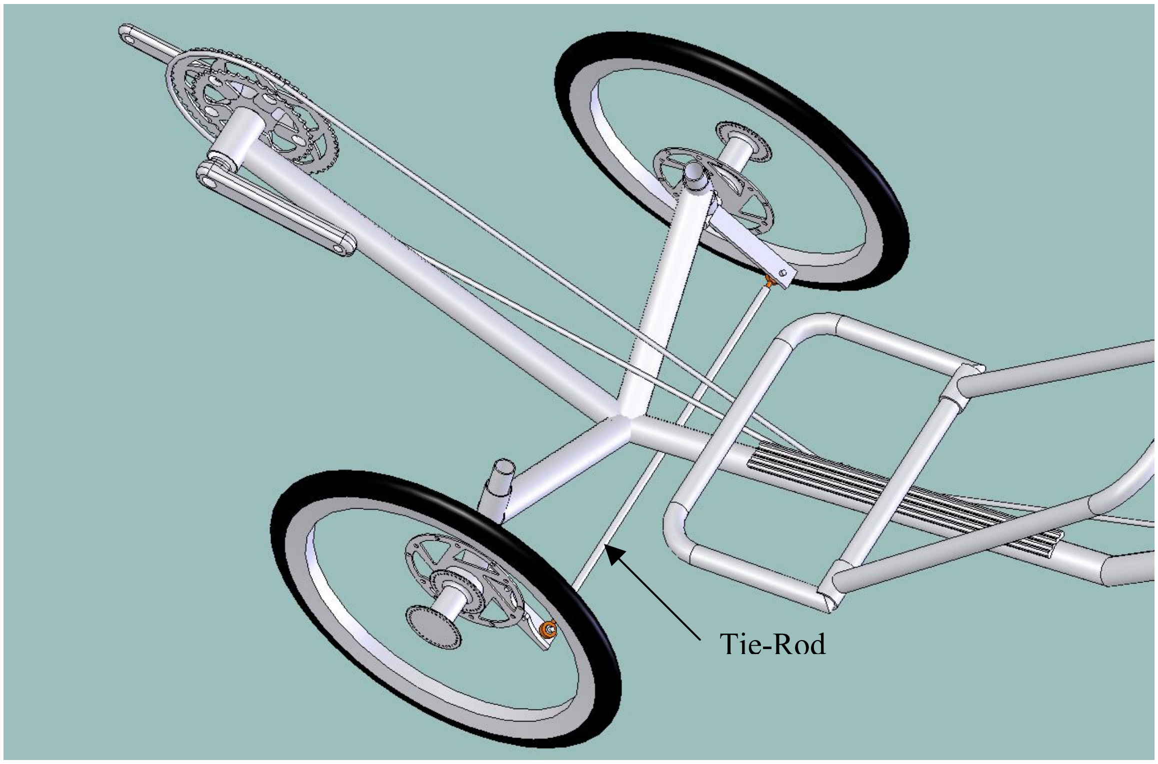


Figure 8.6.10. Detail of tie-rod function and location on 2006 PSU HPV

Given:

A tie rod for the 2006 PSU HPV is to be optimized to be as light as possible while still withstanding a critical compression loading of 200lbs and a critical tensile loading of 200lbs. The tie rod is 20.65in long and must have ¼-20 female threads on each end to accept tie-rod ends. The current project budget makes aluminum and steel the only material options.

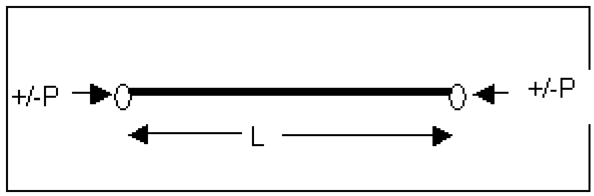


Figure 8.6.11 Tie-rod loading schematic

Find:

Determine the lightest possible cross section for the cylindrical tie rod based on the strength requirements stated above.

Assumptions:

-The member will be loaded only in uni-axial tension and compression.

-The tie rod ends will allow the ends of the member to rotate in any direction and may be approximated as pinned supports.

-The member must be able to be directly threaded to accept ¼-20 female threads, or must have hardware added to it which weights approximately 0.15lb

Solution:

The member has two possible failure modes, tensile failure due to the tensile loading at the pinned ends, and buckling failure due to compressive loading on the pinned ends. The defining equation for the critical tensile loading as defined by Juvinall is given as:

$$\sigma_{\max} = \frac{P_{cr}}{A_{cs}} \qquad \textit{equation 1}$$

Where σ_{max} is the maximum allowable stress, P_{cr} is the associated maximum tensile load, and A_{cs} is the cross-sectional area. Solving the above equation for the cross-section gives the minimum allowable member cross-section as:

$$A_{cs} = \frac{P_{cr}}{\sigma_{\max}} \qquad \textit{equation 2}$$

The defining equation for critical buckling loading as defined by Juvinall is given as:

$$P_{cr,compression} = \frac{\pi^2 E I_{cs}}{L_e^2} \qquad \textit{equation 3}$$

Where $P_{cr,compression}$ is the maximum allowable compressive loading, E is the modulus of elasticity of the material, I_{cs} is the moment of inertia of the cross-section, and L_e is the equivalent length of the member given it's supports and eccentricity. For the case of two pinned ends Juvinall defines L_e as being equal to the member length L, solving equation 3 with this simplification for the moment of inertia of the cross-section gives:

$$I_{cs} = \frac{P_{cr,compression} L^2}{\pi^2 E} \qquad \textit{equation 4}$$

For a design to be acceptable, the cross-section must satisfy both equations 2 and 4, given the properties of the material being analyzed. Material properties for the two acceptable materials, 6061-T6 aluminum and 1018 Steel are given in figure 8.6.12 below.

Material Property	6061-T6 Aluminum	1018 Steel
σ_t (ksi)	45	50
σ_y (ksi)	40	39.9
E (ksi)	10000	29000
ρ (lb/in^3)	0.0975	0.284

Figure 8.6.12. Material properties (Ref. Matweb)

Using the material properties given in Figure 8.6.12. and geometrical requirements given by equations 2 and 4, the required cross-section and moment of inertia of each material option was calculated for the maximum load of 200lb. The results are presented below in Figure 8.6.13.

Required geometry of Aluminum Cross-Section	
Tensile case (in^2)	0.005
Buckling case (in^4)	0.000864
Required properties of Steel Cross-Section	
Tensile case (in^2)	0.005013
Buckling case (in^4)	0.000298

Figure 8.6.13. Cross-section requirements.

Using the information in Table 2, and a list of commonly available outside dimensions for cylindrical rod stock, a list of acceptable specimens for each material was created using spreadsheet software. In addition to the acceptable geometric dimensions, the weight of each option was calculated in order to determine the lightest possible option. The resulting spreadsheets are presented in Figures 8.6.14 and 8.6.15 below.

Aluminum					
d_o (in)	d_i (in)	t (in)	I (in^4)	A (in^2)	weight (lb)
0.25	NA	NA	NA	NA	NA
0.375	0.216	0.080	0.0009	0.074	0.1487
0.5	0.460	0.020	0.0009	0.030	0.0603
0.75	0.739	0.005	0.0009	0.012	0.0251
1	0.996	0.002	0.0009	0.007	0.0140
1.25	*1.247*	0.001	*0.0010*	*0.005*	0.0101
1.5	1.498	0.001	0.0014	0.005	0.0101
1.75	1.748	0.001	0.0019	0.005	0.0101
2	1.998	0.001	0.0025	0.005	0.0101
2.25	2.249	0.001	0.0032	0.005	0.0101
2.5	2.499	0.001	0.0039	0.005	0.0101
2.75	2.749	0.001	0.0047	0.005	0.0101
3	2.999	0.001	0.0056	0.005	0.0101

Figure 8.6.14. Acceptable aluminum cross-sections

Steel					
d_o (in)	d_i (in)	t (in)	I (in^4)	A (in^2)	weight (lb)
0.25	NA	NA	NA	NA	NA
0.5	0.487	0.006	0.0003	0.010	0.0573
0.75	*0.746*	0.002	*0.0004*	*0.005*	0.0294
1	0.997	0.002	0.0006	0.005	0.0294
1.25	*1.247*	0.001	0.0010	0.005	0.0294
1.5	*1.498*	0.001	0.0014	0.005	0.0294
1.75	*1.748*	0.001	0.0019	0.005	0.0294
2	*1.998*	0.001	0.0025	0.005	0.0294
2.25	*2.249*	0.001	0.0032	0.005	0.0294
2.5	*2.499*	0.001	0.0039	0.005	0.0294
2.75	*2.749*	0.001	0.0047	0.005	0.0294
3	*2.999*	0.001	0.0056	0.005	0.0294

Figure 8.6.15. Acceptable steel cross-sections

The above tables show that the lightest possible option is an aluminum tie-rod with outside diameter of 1.25in and wall thickness of 0.001in. However, the hardware required to allow this geometry to accept the tie-rod ends has a weight of 0.15 lb, making it considerably heavier. The lightest option is therefore an aluminum cylinder with outside diameter of 0.375in and wall thickness of 0.08in which may be directly threaded to accept the tie-rod ends with no additional hardware.

Conclusion:

Optimized tie-rod geometry

Length: 20.65in
Outside Diameter: 0.375in
Wall thickness: 0.08in
Material Specification: 6061-T6 Aluminum

[8.6.4]: Determination of Roll-Over Velocity

Summary

The objective of this analysis is to determine the speed at which the 2006 Portland State University Human Powered Vehicle will roll over in a turn of a given radius. There are multiple factors that will affect the roll-over velocity of the HPV, the height of the center of gravity of the vehicle, the width of the two front wheels (track), and the radius of the turn. This analysis presents the roll-over velocity (V) of the HPV as a function of increasing turning radius (ρ) for given track and height of center of gravity.

The results show that the vehicle is able to take a 20ft radius turn at over 15mph without tipping. This result meets the PDS requirement for vehicle stability.

ρ (ft)	V(mph)
10	10.62
11	11.14
12	11.64
13	12.11
14	12.57
15	13.01
16	13.44
17	13.85
18	14.25
19	14.64
20	***15.03***
21	15.40
22	15.76
23	16.11
24	16.46
25	16.80

From the 2006 HPV Challenge (the race) these results are reasonable; the HPV rolled twice and went up on two of the three wheels once. All three instances were at speeds in excess of 10 mph and in turns greater than 10 ft in radius.

Given:

The 2006 Portland State University Human Powered Vehicle has a mass of 230lbs with a center of gravity located 20in above the ground and 34in forward of the rear axel. The front wheel track of the vehicle is 29in and the overall wheel base is 34in.

Find:

Determine the speed at which the 2006 Portland State University Human Powered Vehicle will roll over in a 20ft radius turn.

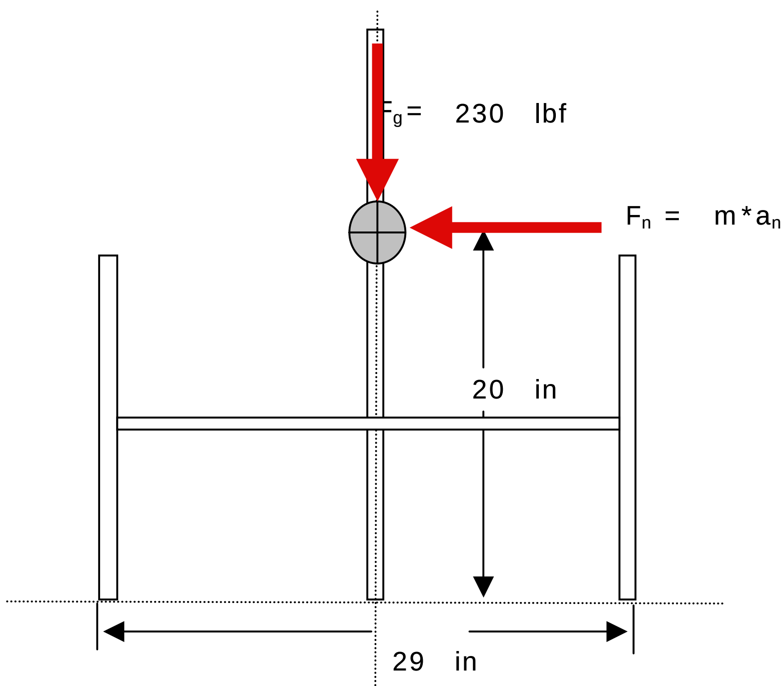
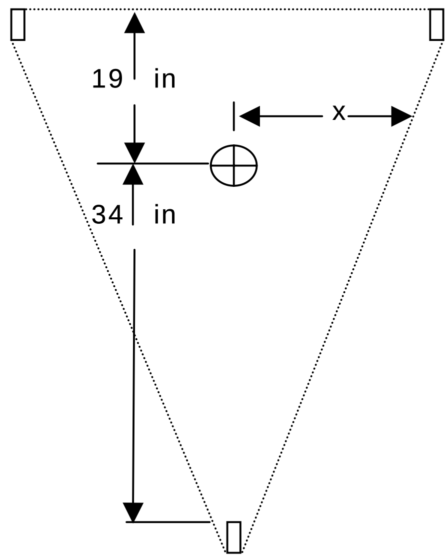


Figure 8.6.16: Location of vehicle center of gravity.

Solution: From the given information, the location of the center of gravity of the vehicle can be determined as shown in Figure 8.6.17:



We can find, from basic trig, the distance from the CG out to the wheel edge line as:
x = 9.30 in

Figure 8.6.17: Location of vehicle center of gravity in plan view.

So from the first schematic given we find the variable in the analysis is the normal force (f_n) from the acceleration of a particle in circular motion, which is dependant on the radius of the turn and the speed at which the turn is taken.

From a sum of moments, and with the distance x as the moment arm, we can develop the roll-over velocity from the following equations [Ref. Meriam]:

$$\boxed{\sum M = F_n h = mgx} \qquad \textit{equation 1}$$

Where:

$$\boxed{F_n = m * a_n = m * \frac{V^2 h}{\rho}} \qquad \textit{equation 2}$$

Solving for the velocity we find:

$$\boxed{V = \sqrt{\frac{xg\rho}{h}}}$$

Thus the roll-over velocity as a function of the turning radius for turning radii of 10 through 25 ft is:

Results:

ρ (ft)	V(mph)
10	10.62
11	11.14
12	11.64
13	12.11
14	12.57
15	13.01
16	13.44
17	13.85
18	14.25
19	14.64
20	**15.03**
21	15.40
22	15.76
23	16.11
24	16.46
25	16.80

The results show that a 29in wheel track is wide enough to make a 20ft radius turn at 15mph which meets the requirements of the PDS.

[8.6.5]: Analysis of minimum stopping distance

Summary:

The objective of this analysis is to determine the stopping distance of the 2006 PSU HPV. Competition rules state that the vehicle must be able to come to a complete stop from a speed of 15mph in 20 feet or less. To determine the theoretical stopping distance of the HPV, two models were built. One model determines the stopping distance as limited by interfacial friction between the tires and road, and the other model determines the stopping distance based on forward tipping potential. A picture of the disk braking system being used is shown in Figure 8.6.18.

The analysis shows that the vehicle stopping distance is limited by interfacial friction in all cases. For the predicted road conditions the model shows that the vehicle will be able to come to a complete stop in approximately 8 feet. After constructing the prototype and testing its' braking ability, we found this analysis to be accurate to within a foot for various conditions.

Figure 8.6.18: Disk braking system used in the 2006 PSU HPV

Given:

The 2006 PSU HPV is required to stop in less than 20ft from an initial velocity of 15 mph. The center of mass of the vehicle and rider is 20in above the ground and 19in behind the front wheel axis. The total vehicle and rider weight is 230lb.

Find:

-Determine the minimum stopping distance of the vehicle from 15mph.

Assumptions:

-The disk brake system can apply enough force to lock the front wheels.

Solution:

-Two criteria limit the stopping distance of the HPV. The interfacial friction between the tires and the road, and the tendency for the vehicle to tip forward during hard braking.

-The minimum braking distance based on friction is given by Wilson as:

$$S = \frac{V^2}{20 * C_{\text{int}}} \qquad \textit{equation 1}$$

Where V is the initial velocity in meters per second, and C_{int} is the interfacial coefficient of friction between the road and tire. A plot of stopping distances, based on various initial velocities and coefficients of friction, is presented in Figure 4.5. For an initial velocity of 15 mph, the stopping distance for this model was determined to be 8 feet.

- To determine the stability of the vehicle during hard braking, a second analysis was performed. A balance of moments based due to gravity and deceleration forces is given by equation 2:

$$\sum M_{cp} = mgA - ma_{\max} h_{cg} = 0 \qquad \textit{equation 2}$$

Where: m = combined mass of vehicle and rider, A=horizontal distance between front contact patch axis and center of gravity, and h_{cg} = height of center of gravity.

-The results of both of these calculations are presented in figure 8.6.19.

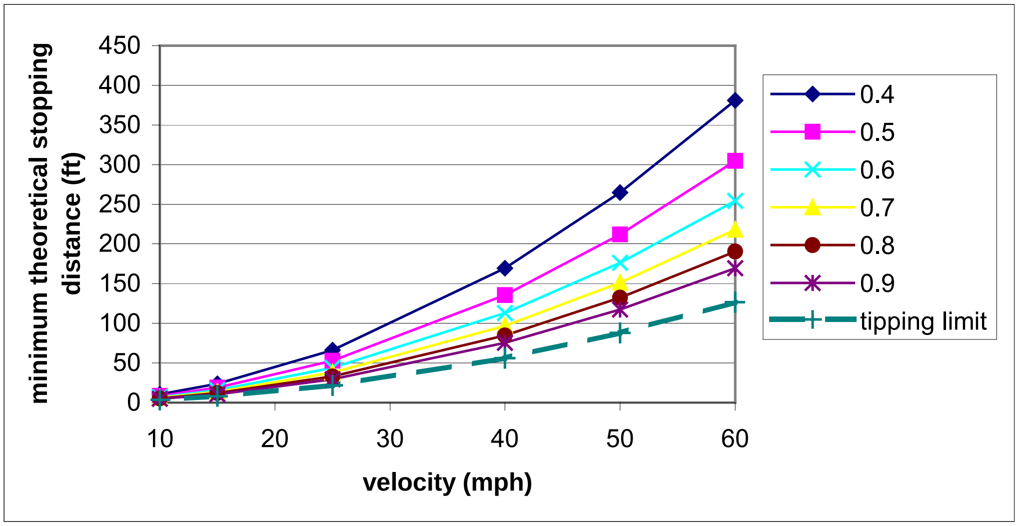


Figure 8.6.19: Plot of Stopping distances based on friction and tipping limits.

Conclusion:

-Figure 8.6.19 shows that the stopping distance is limited by the interfacial friction and not the tipping potential. Further, for predicted competition road conditions of a standard dry asphalt surface, the theoretical stopping distance is 8ft which meets the PDS requirements.

[8.6.6]: Steering Angle Comparison and Selection

Steering geometry is composed of three planar wheel angles commonly referred to as camber, caster and toe. These three angles dictate road and friction force transmission and therefore, greatly affect the steering characteristics of the bike.

Camber

Definition:

Camber is the angle between wheels as defined by the difference in the distance between the tops and bottoms of parallel wheels. A zero degree camber angle indicates that the wheels are totally parallel. A negative angle indicates that the distance from the tops of the tires (length A in the figure below) is smaller than the distance between the bottoms of the tires.

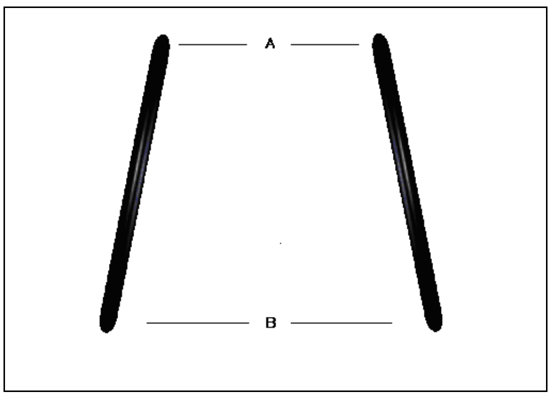


Figure 8.6.20: Camber angles as seen from the front of the vehicle. Negative camber indicates that distance A (top of the tires) is less than distance B (bottom).

Benefits:

Greater camber angles can provide turning stability by increasing the wheel track and redirecting normal acceleration loads through the centroid of the wheel.

Drawbacks:

Possibility of damage to the wheel. Increased camber causes premature tire wear by focusing the contact patch to a smaller portion of the tire. Extreme increases in camber can reduce turning radius by causing out of plane turning. Camber may also increase wear on wheel bearings by placing constant torque on them as well as placing an excessive moment on the wheel hubs.

Selection method:

Camber angles were chosen as a balance of planar wheel turning radius reduction and resultant force vector transmission through the centroid of the wheel. Tire and bearing wear were taken into consideration, as well as minimal increases in wheel track, but were found to be too small to make a quantified assessment for. Three spreadsheets were created, one to calculate camber based on the resultant force angle (centripetal force due to turning and normal force due to gravity). The resultant angle is used to influence camber angle. By matching these angles, force is transmitted through the wheel in a compressive manner without generating a moment about the hub of the wheel while turning. another spreadsheet is used to determine the reduction in steering angle as a function of an increase in camber angle. A third spreadsheet is used to determine the moment load on the hub of the wheel and the associated factor of safety as a function of the camber angle.

V (mph)	V (fps)	a (ft/s^2)	Force (xdir)	Force (y dir)	Resultant	Camber angle
5	7.33	2.15	15.37	230.00	230.51	3.82
10	14.67	8.60	61.46	230.00	238.07	14.96
15	22.00	19.36	138.29	230.00	268.37	31.02
20	29.33	34.42	245.84	230.00	336.66	46.91
25	36.67	53.78	384.13	230.00	447.72	59.09

Figure 8.6.21: Camber selection based on resultant vector forces

Steering view	Camber angle	Percent view reduction
1.74	0.00	0.00
1.74	1.00	0.02
1.74	2.00	0.05
1.73	3.00	0.08
1.73	5.00	0.24
1.71	10.00	1.14
1.68	15.00	1.92
1.50	30.00	10.34
1.23	45.00	18.35
0.00	90.00	100.00

Figure 8.6.22: Reduction in turning ability as a function of camber angle.

Camber angle	Static moment	Impact moment	Safety factor
1	2.91	29.09	12.03
2	6.689070201	66.89	5.23
3	8.72	87.23	4.01
4	13.3699908	133.70	2.62
5	14.53	145.26	2.41
10	33.28256739	332.83	1.05
90	166.67	1666.67	0.21

Figure 8.6.23: Moment loading on wheel hub as a function of camber angle.

Caster

Definition:

Also known as the kingpin angle, caster is the measured angle of the “fork” where positive caster projects the bottom of the fork forward, in front of the bike.

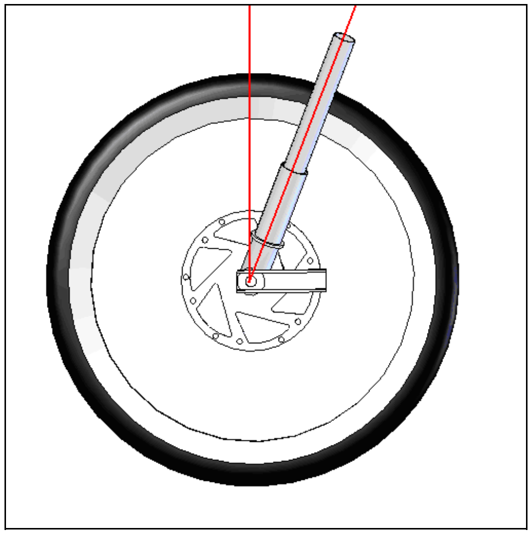


Figure 8.6.24: Caster angle as measured from steerer tube to vertical axis.

Benefits:

Increasing caster can increase “wheel return” or the tendency for the bike to travel in a straight line. Caster also helps to transmit road force in to the frame, rather than the steering interface.

Drawbacks:

Excessive caster can cause understeer, making the vehicle feel unresponsive. Too much caster can increase bending moment in the steerer tube as well as create excessive wear in the headset. Caster creates loading in the tie rod and ,similar to the camber angle, extreme caster angles can reduce the effective turning angle of the wheel.

Selection method:

The method of caster angle selection is dictated by the balance between the wheel return force and the bending stress in the steerer tube as well as an attempt to minimize the reduction in realized steering angle.

Caster angle	Normal force (lb)	Wheel return force (ft-lb)
10	76.67	1.00
12	76.67	1.20
15	76.67	1.49
18	76.67	1.78
20	76.67	1.97

Figure 8.6.25: Wheel return force as a function of caster angle.

Caster angle	Static force	Bending stress	Compressiv e stress	Safety factor
10	39.94	782.34	28.66	24.66
12	47.82	936.71	41.09	20.45
15	59.53	1166.06	63.67	16.26
18	71.07	1392.22	90.76	13.49
20	78.66	1540.91	111.19	12.11

Figure 8.6.26: Bending stress as a function of caster angle

Steering view	Caster angle	Percent view reduction
1.74	0.00	0.00
1.74	1.00	0.02
1.74	2.00	0.05
1.73	3.00	0.08
1.73	5.00	0.24
1.71	10.00	1.14
1.68	15.00	1.92
1.50	30.00	10.34
1.23	45.00	18.35
0.00	90.00	100.00

Figure 8.6.27: Reduction in planar turning as a function of caster

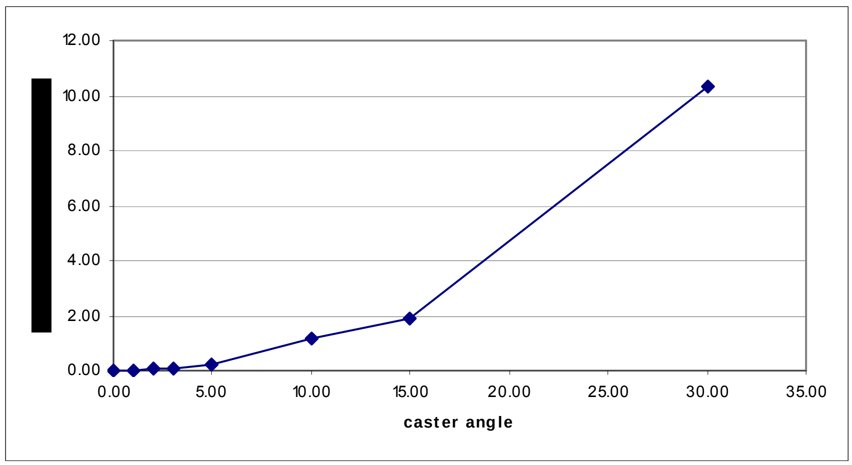


Figure 8.6.28: Percent effective steering reduction due to caster angle.

Toe

Definition:

Toe is the difference in the distances between the front and the rear of the front tires, as shown in figure 8.6.29.

Benefits:

Increased toe can reduce the scrubbing effect on tires at specific velocities and also increase tracking or the tendency to travel in a straight line. Toe can reduce the road impact felt by the rider and can also increase turning radius.

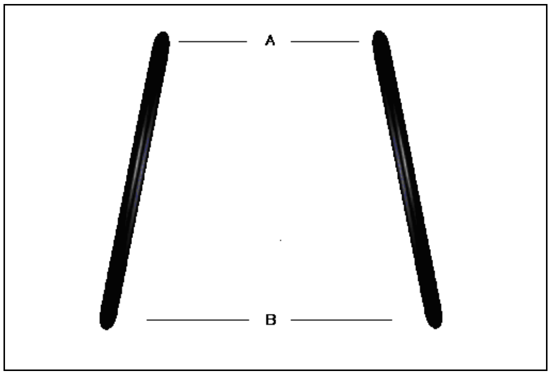


Figure 8.6.29: Toe angles as seen from the top of the vehicle where "toe in" is defined as distance A (front of the vehicle) being greater than distance B.

Drawbacks:

Incorrectly adjusted toe can increase tire scrubbing, understeer, oversteer, and dramatically reduce tire life. Creates loading in the tie rod.

Steering Angle Decision Matrix

Prioritization of steering parameters was needed in addition to quantified results to compare the results in terms of importance and compliance to competition rules, safety and performance benefit. A decision matrix was created to narrow these parameters. In the table below, each decision criteria was given a weight to quantify its importance. The criteria are scrubbing (corresponds directly to vehicle speed), complexity (translates to cost and weight), stability (rider safety and rule compliance), turning radius and safety. Each steering angle was then rated -2…2 with -2 having extreme negative impact, 0 having no impact, and 2 having substantial positive impact. The high and low measures for each angle were then compared to one another to determine the appropriate solution.

	Scrubbing	Complexity	Stability	Turn radius	Safety	Total
Weight	**0.25**	**0.1**	**0.2**	**0.2**	**0.4**	**1.15**
high camber(10+)	-1	-1	2	1	-1	-0.15
low camber (0-5)	0	0	1	1	1	0.8
high caster (10-20)	1	-1	2	1	0	0.75
low caster (0-10)	1	0	1	1	0	0.65
toe in	-1	0	1	0	1	0.35
toe out	-1	0	0	2	0	0.15

Figure 8.6.30: Steering angle decision matrix

With these results the steering angles are selected to be: Camber = 3° , Caster = 15° and Toe = 1°

Appendix [8.7]: Climate and Geographic Data for San Luis Obispo

--Climate Data

--Weather station Morro bay fire dept, San Luis Obispo county is at about 35.36°N 120.85°W. Height is 35m / 114 feet above sea level.
(ref. Worldclimate.com: http://www.worldclimate.com/cgi-bin/data.pl?ref=N35W120+1300+045866C)

--April Averages:

Max Temp: 17.4 C 63.3 F

Min Temp: 7.1 C 44.8 F

24hr Average: 12.2 C 54 F

--Weather station San Luis Obispo is at about 38.29°N 120.70°W. Height about 91m / 298 feet above sea level.
(ref. Worldclimate.com: http://www.worldclimate.com/cgi-bin/data.pl?ref=N38W120+2100+7249204G1)

--April Average:

Monthly Rainfall: 41.2mm 1.6 in

--Campus Geography:

Elevation: 300-1185 ft Average = 742.5
(Ref Calpoly.edu: http://polyland.calpoly.edu/overview/Archives/nieto/study.html)

--Atmospheric Properties corresponding to 24hr average temps:

For an altitude of 0 ft and temp of 59 F (15C) the properties of the US standard atmosphere are: (ref. Fundamentals of Fluid Mechanics, M.Y.O)

Pressure:	14.696 lb/in^2 (abs)	101300 N/m^2 (abs)
Density:	.002377 $slugs/ft^3$	1.225 kg/m^3
Dynamic Viscosity:	3.737 E –7 $lb*s/ft^2$	1.789 E-5 $N*s/m^2$

Appendix [8.8]: Design Analysis and Testing Details

Index of Analysis:

[8.8.1]: Overview of CFD analysis procedure

For validation of the vehicle fairing design a computational fluid dynamics (CFD) analysis was performed on the geometry. Two separate models were built, one to determine the properties of the flow under standard riding conditions, and a second to determine the effects of crosswinds on vehicle stability.

All models were built and meshed in STAR-Design and then exported to STAR CCM+ for solving and post processing. All solver runs were allowed to iterate until a minimum convergence level of 0.001 was achieved for all parameters. All presented solutions include the Reynolds-Averaged Navier-Stokes turbulence model, approximated using the Jones and Launder κ-ε model [For details see: Ferziger ch 9].

The first model involved cutting the fairing down its' vertical center plane such that a symmetry boundary condition could be implemented to conserve computational resources. A mesh with approximately 250,000 cells (Figure 8.8.1) was then built and the flow was solved for a variety of free-stream inlet velocities covering the range of design values.

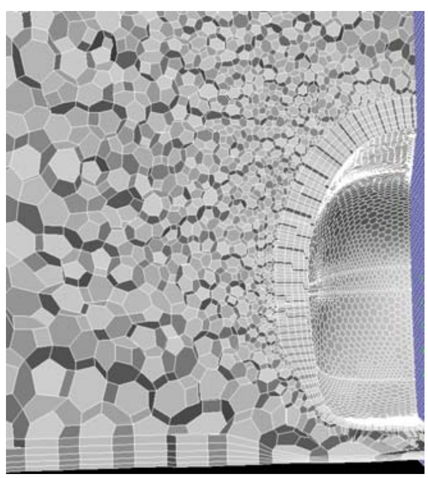


Figure 8.8.1: Cut view of mesh used in analysis

For each solver run, velocity fields were plotted to visually inspect for flow abnormalities, pressure and shear force data were used to calculate drag coefficients and were plotted on the fairing surface (Figures 8.8.2 and 8.8.3). Based on these simulations, and a model frontal area of 886 square inches, the drag coefficient at 22 mph was determined to be 0.11.

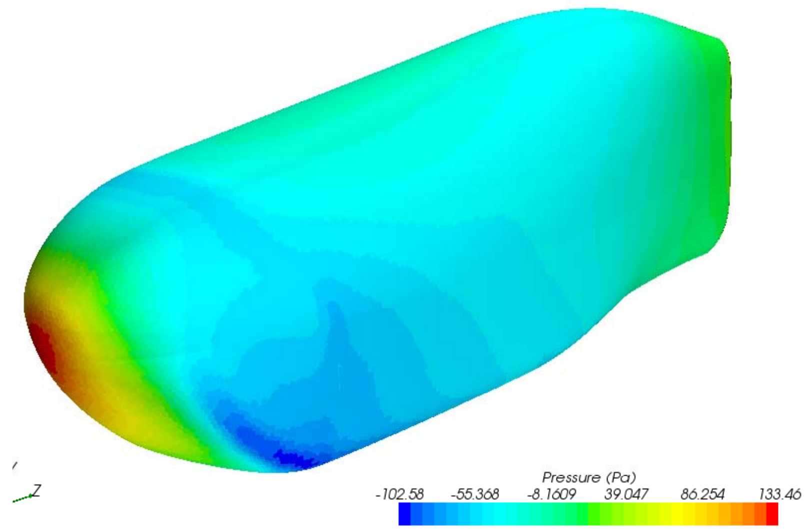


Figure 8.8.2: Pressure distribution on surface for 34mph free stream velocity

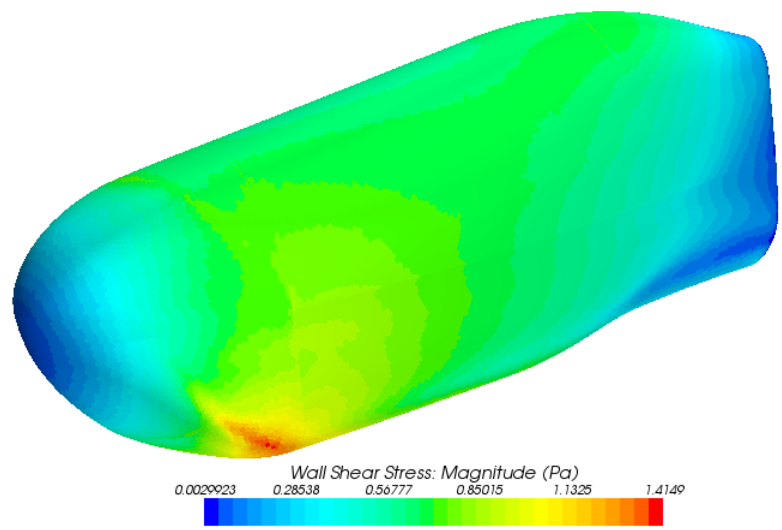


Figure 8.8.3: Shear stress on surface for 34mph free stream velocity

A second model was built to simulate the crosswind condition of a vehicle traveling at 22mph relative to the ground, with a 22 mph wind perpendicular to the direction of travel (Figure 8.8.4). This model achieved mesh convergence at approximately 450,000 cells, and was able to produce information on the center of pressure of the vehicle given the boundary conditions, as well as a total force perpendicular to the direction of travel due to fluid forces of 53 lbf.

The model predicts that the center of pressure is located approximately 4in behind and 1in above the center of gravity. Studies cited by Wilson suggest that bicycle stability is increased by locating the center of pressure in front of the center of gravity, however for vehicles having more than two wheels, Tamai reports that vehicle stability is enhanced by the opposite condition. Thus the model predicts the Vike Trike fairing design is likely to mitigate some of the effects of crosswinds on vehicle stability.

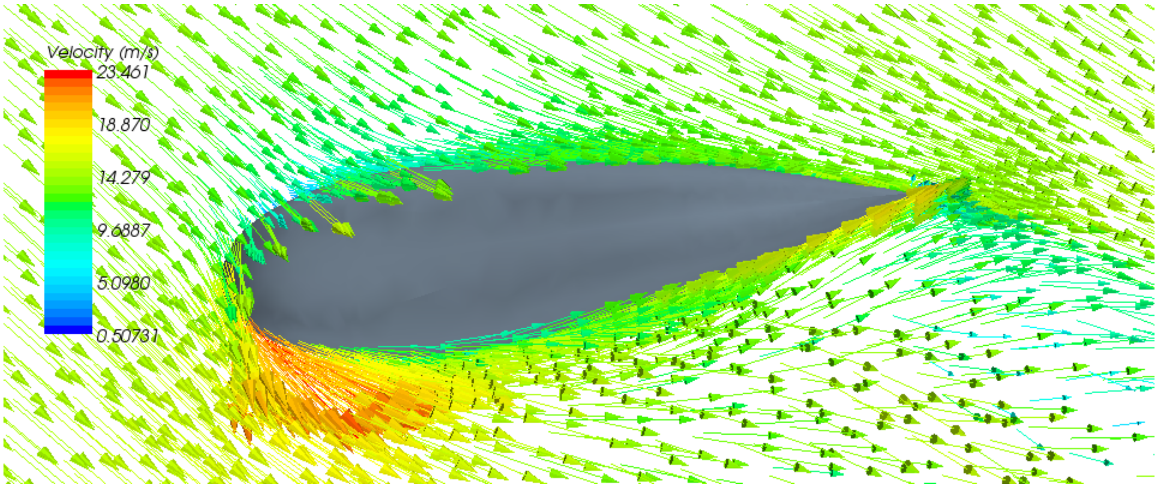


Figure 8.8.4: Velocity vector field for model of 22mph vehicle travel in 22mph cross wind.

[8.8.2] Overview of Strain Gage Testing and Results

The vehicle was outfitted with a series of strain gages, and was operated on a rear-wheel treadmill in a lab. For the dynamic testing, the transmission was set at the largest possible gear ratio, using a 53-tooth drive gear with a 12-tooth rear sprocket for all trials. Data for each trial was collected at 50 Hz for a total of 20 seconds. Multiple riders were used in the live pedaling trials, and the pedaling rate was ranged from of 60 to 138 RPM.

The strain gauges were mounted to the vehicle frame as shown in figure 8.8.5. During testing, it became clear that the mounting system for the seat rail was less than perfectly rigid. This caused some deviation from the FEA model, which does not allow for any slippage between the materials.

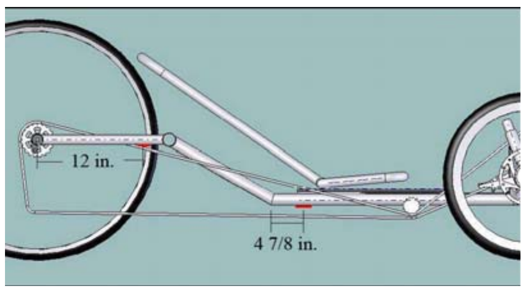


Figure 8.8.5: Location of Chainstay and Main Tube Strain Gauges

The modified endurance limit (S_e) for the material was calculated using the appropriate correction factors, and was used in the Goodman Fatigue Criterion (Table 2.4), along with the alternating stress components (σ_a) and mean stress components (σ_m), determined from the test data. The inputs for fatigue calculations are listed in Figure 8.8.6.

Ultimate Tensile Strength				UTS=100 ksi
Endurance Limit				S_e'=50 ksi
Loading Factor in bending				$k_c = 1$
Size Factor	d < 2in.		$k_b = \left(\frac{d}{.3}\right)^{-.1133}$	K_b=.807
Surface Finish Factor (hot-rolled steel)	a=14.4 ksi	b=-.718 ksi	$k_a = aS_{ut}^{\ b}$	K_a=0.528
Modified Endurance Limit				S_e=21.3 ksi
Goodman Fatigue Criterion (for n=10^7 cycles)	(approx. 2,800 hours cycling at 60 RPM)			$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n_f}$

Figure 8.8.6: Fatigue Equation Inputs and Correction Factors [Ref. Shigley]

Although the vehicle is not necessarily expected to see cyclic loading in excess of 2,000 hours of riding, a fatigue factor of safety of 2 has been set as the target. Figure 2.9 shows the difference between mean and alternating stress in the chainstay and main tube sensor locations. The test rate of 90 RPM is a high rate for cycling a long distance, but is realistic for short bursts and sprint courses.

The results of the fatigue analysis are presented in figure 8.8.8. The resulting minimum factor of safety in fatigue of 2.31 meets the PDS requirement of 2.

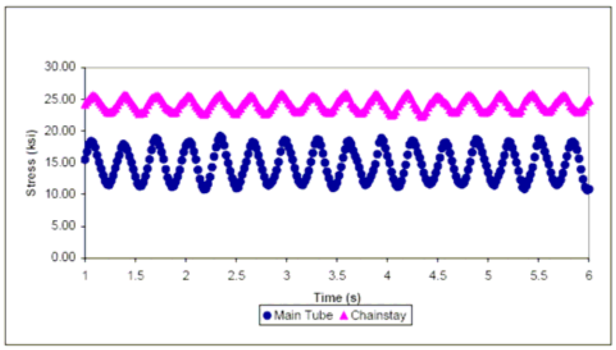


Figure 8.8.7: Combined Plot of Fluctuating Stresses for 165-lb rider at 90 RPM

Rider Load	165 lb	140 lb
Gage Location	Main Tube	Main Tube
RPM	90	138
Static Load (lbf)	166	140
Static Stress (ksi)	16.60	12.56
Max Stress (ksi)	19.23	18.97
Min Stress (ksi)	10.51	5.82
Alternating Stress (ksi)	4.36	6.58
Mean Stress (ksi)	14.87	12.39
Fatigue Safety Factor, n_f	**2.83**	**2.31**

Figure 8.8.8: Max Dynamic Load Data

Appendix [8.9]: Vehicle Maintenance Schedule

Every ride check:

- True of wheels
    - Spin each wheel and watch the rim. If the rim wobbles, up and down or from side to side, repair before riding.
- Tire inflation
    - Inflate the tire to the pressure recommended on the tire label. Also inspect the tire for any cuts or abrasions to the contact surfaces or sidewalls.
- Brakes
    - Squeeze each brake lever toward the handlebar to make sure the brake moves freely and stops the bike. If the brake lever can be pulled to the handlebar, the brake is too loose. The brake pads should be 0.25 to 0.75 mm away from the disc when the brakes are not applied. If the pads are too close, the brake is too tight, or misaligned.
    - Make sure rotors are free of foreign substances and oils.
- Check chain tension.
    - With no force on the pedals the return side of the chain should have enough tension to cause the chain to be pulled against the bottom of the return side idler.
- Frame
    - Carefully inspect your frame for signs of fatigue:
        - Scratches
        - Cracks
        - Dents
        - Deformation
        - Discoloration
    - If any part shows signs of damage or fatigue, repair the frame before riding.

Weekly Check:

- For loose spokes
    - Make sure there are no loose or damaged spokes.
- Fairing and seat.
    - Unlike metal parts, carbon composite parts that have been damaged may not bend, bulge, or deform. After any high force load, like a crash or other impact to your HPV, thoroughly inspect all the parts, and use the following procedures to inspect carbon composite parts
        - Check for scratches, gouges, or other surface problems.
        - Check the part for loss of rigidity.
        - Check the part for delamination.

Monthly Check:

- Attachment of steerer tubes
- The security of the handle bars by attempting to rotate them in the stems. If the handlebars rotate in the stems don't ride the HPV
- The cassette and chain
    - Check that the chain and cassette are clean, free of rust, and properly oiled. All links of the chain should pivot smoothly and without squeaking, and no links of the chain should be deformed.
- Cables. Inspect and lube shifter and brake cables
    - SRAM recommends "jonnisnot" or some type of plastic to metal lubricant
- Check brake pads
    - A pad should be replaced when its total thickness is less than 3mm. A pad wear indicator is at the center of each inboard and outboard red adjusting knobs. As the knob is turned in, the indicator will retract deeper into the knob giving a visual indication of approximately how much the pads have worn.[ref Avid]
- Check wheel bearings
    - Check that all hub bearings are properly greased and adjusted. Lift a wheel off the ground with one hand and attempt to move the rim left to

right. Look, feel, and listen for any looseness in the hub bearings. Spin the wheel, and listen for any grinding or other unusual noises. If the hub feels loose or makes any noise, the hub needs an adjustment. Repeat these procedures for the other two wheels.

Every 3 months check:

- That the cassette is tight.
    - Attempt to move the largest rear cog from side to side. If there is any movement, tighten to the torque specifications.
- Check your chain for wear with a chain wear gauge or a ruler.
    - Each new chain link measures 1in. If the chain stretched such that 12 links measure more than 12 1/8 inches the chain should be replaced.

Appendix [8.10]: Vehicle Detailed Design

This section includes a bill of materials and the complete set of drawings required to construct the vehicle in a well equipped shop. The assembly process involves no special tools other than a TIG welding machine and an operator with proper training.

[8.10.1] Bill of Materials

Stock Components

Part	Manufacturer	Quantity	Unit
32 Spoke 700C Rear Wheel	Mavic rim with Shimano hub	1	ea
28 Spoke 20" Front Wheel	Mavic rim with Phil Woods hub	2	ea
700C Racing Tire	Bontragger	1	ea
700C Tube	Giant	1	ea
20" Racing Tire	Continential Grand Prix	2	ea
20" Tube	Specialized	2	ea
9-speed Casette	Shimano Ultegra	1	ea
Cranks with Triple Chainring	Shimano Dura Ace	1	ea
Bottom Bracket	Shimano	1	ea
Chain	Shimano Dura Ace	3	ea
Pedals	Shimano SPD	1	set
Rear Wheel Dropouts	Vanilla Cycles	1	set
Chain Idler	Terracycle	1	ea
Front Disc Brakes	Avid	2	ea
Brake Levers	Shimano XTR	1	set
Stem	Profile Design BOA	2	ea
Headset	Chris King	2	ea
Shifters	SRAM Rocket Shifts	1	ea
Shifter Cable	Shimano	5	ft
Shifter Housing	Shimano	5	ft
Brake Cable	Shimano	2	ft
Brake Housing	Shimano	2	ft
Seatbelt (w/ hardware)	Andover Auto	1	ea

Tubing

Material	ID / Thickness	Length	Unit
4130 Chro-molly	1.5" / 0.049"	9.5	ft
4130 Chro-molly	1.0" / 0.065"	3	ft
A36 Steel	1/16" flatstock	80	in^2
6061 T-6 Aluminum	2.0" / 0.065"	5.5	ft
6061 T-6 Aluminum	1.0" / 0.065"	3	ft
6061 T-6 Aluminum	0.0375" solid rod	2	ft
6061 T-6 Aluminum	1/16" flatstock	6	in^2
6061 T-6 Aluminum	1 1/4" solid rod	2	ft
6061 T-6 Aluminum	2" x 2", 1/4" angle stock	1	ft
6061 T-6 Aluminum	1", 1/8" thick	1	ft

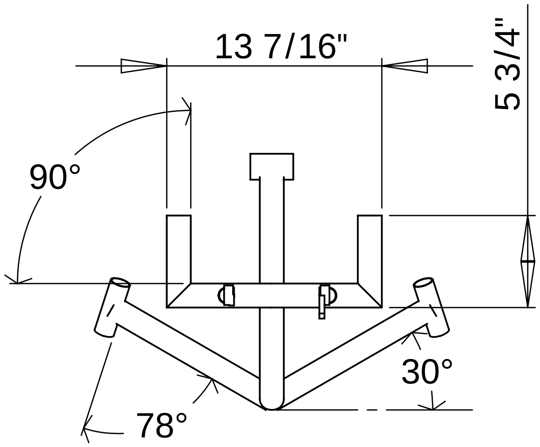
Hardware

Item	Description	Quantity	Unit
Allen Bolts	1/4" - 20, 1 1/2" long	8	ea
Nylock Nuts	1/4" - 20	8	ea
Nylon Spacers	1/4" bore x 1/4" tall	2	ea
Grade 8 Bolts	3/8", 2 1/2" long	2	ea
Grade 8 Bolts	3/8", 1/2" long	1	ea
Rivets	3/16", 1/2" long	1	box
Hose Clamps	3" diameter	6	ea
Marine Grade Velcro	1" x 3"	4	ea
Tie Rod End	1/4" - 28 right thread	1	ea
Tie Rod End	1/4" - 28 left thread	1	ea
Hex Nuts	1/4" - 28	2	ea
Pipe Insulation	2" diameter	5	ft
Pink Foam Insulation	2' x 8'	1	sheet
Zip Ties	6" long	10	ea

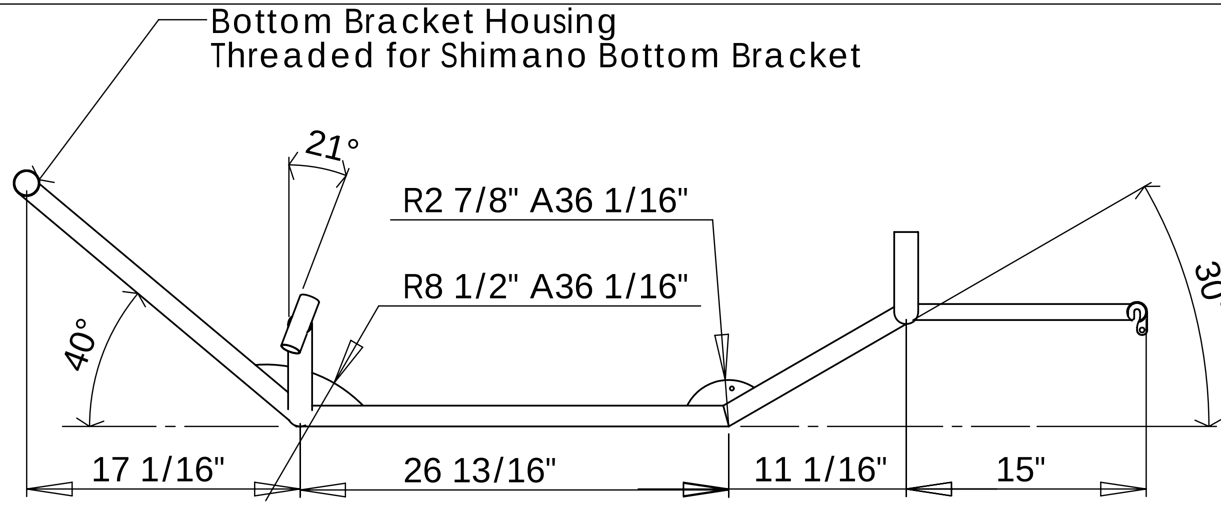
Fairing Material

Item	Description	Quantity	Unit
Carbon Fiber Tape	2" wide	20	yd
Carbon Fiber Cloth	10 oz, 12k x 12k weave	30	yd^2
Kevlar	5 oz	10	yd^2
Resin	Marine epoxy A-side resin 314	2	gallon
Hardener	Slow B-side hardener 109	2	64 oz

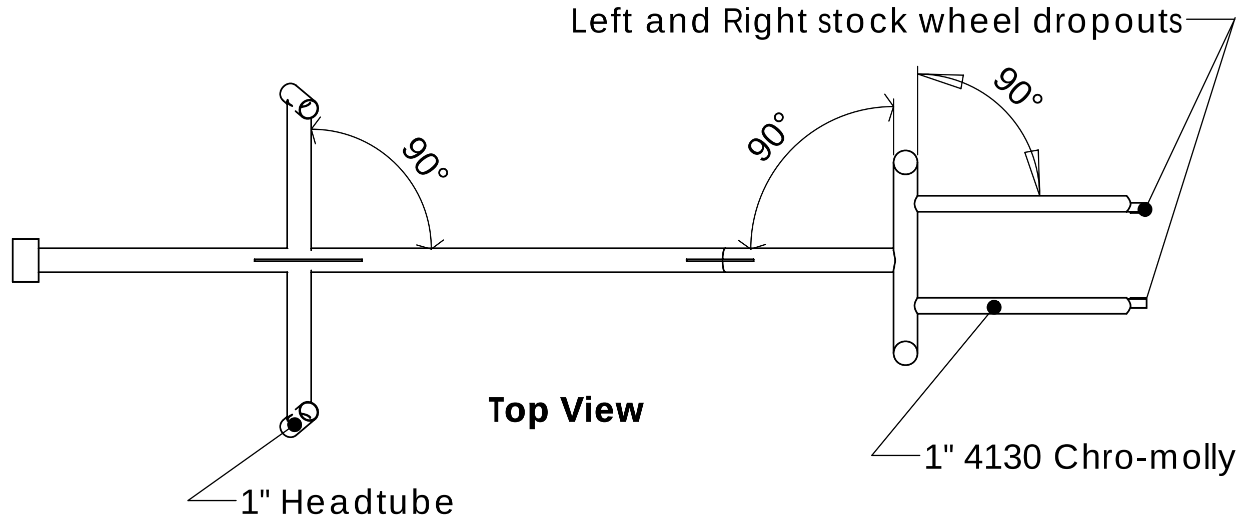
[8.10.2] Detailed Drawings



Rear View



Side View



Top View

Notes:

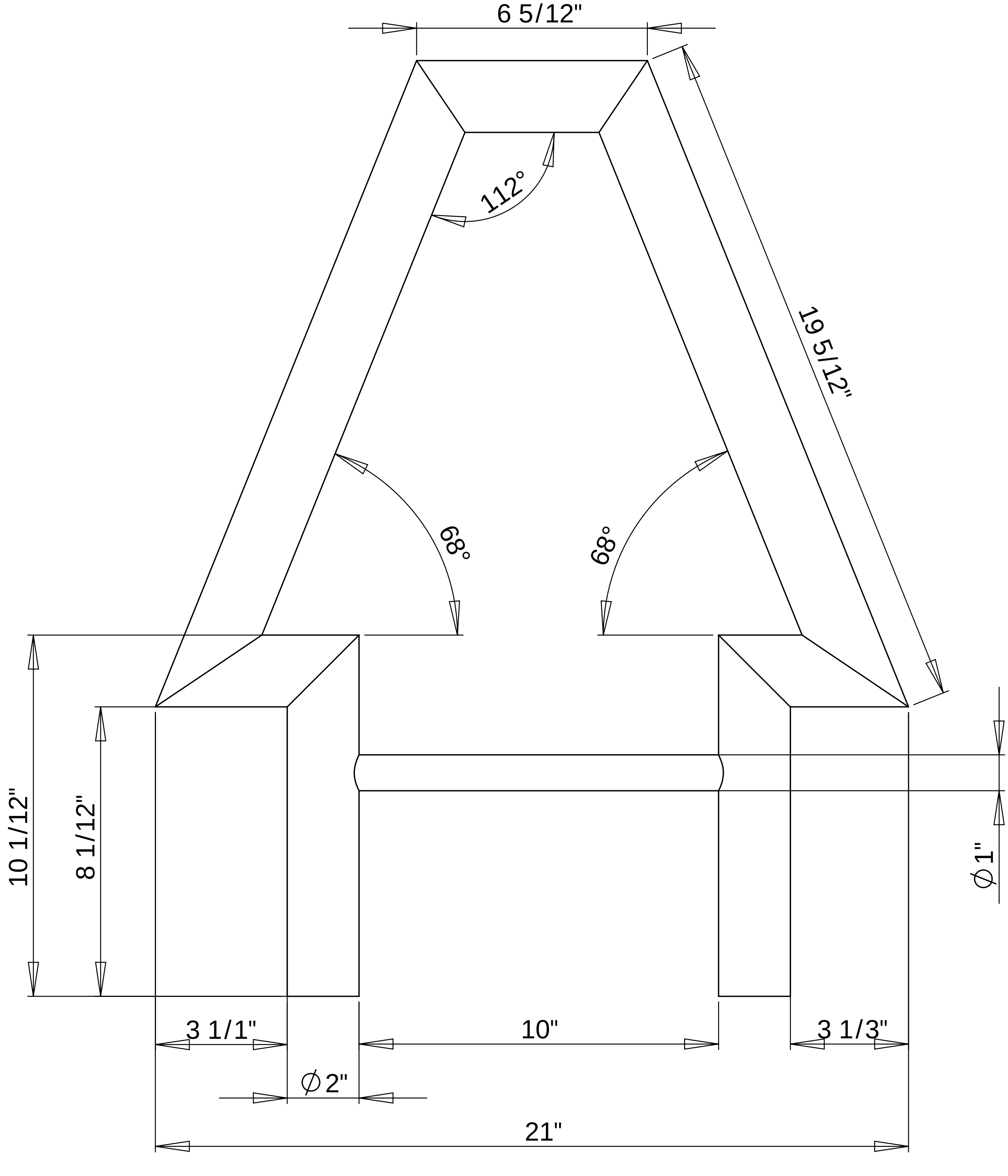
All tubing 1.5" 4130 Chrome-moly unless otherwise stated.

All joints are welded throughout by GTAW

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			Frame
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:20 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1

2" 6061-T6 tubing unless otherwise stated
All joints are welded throughout



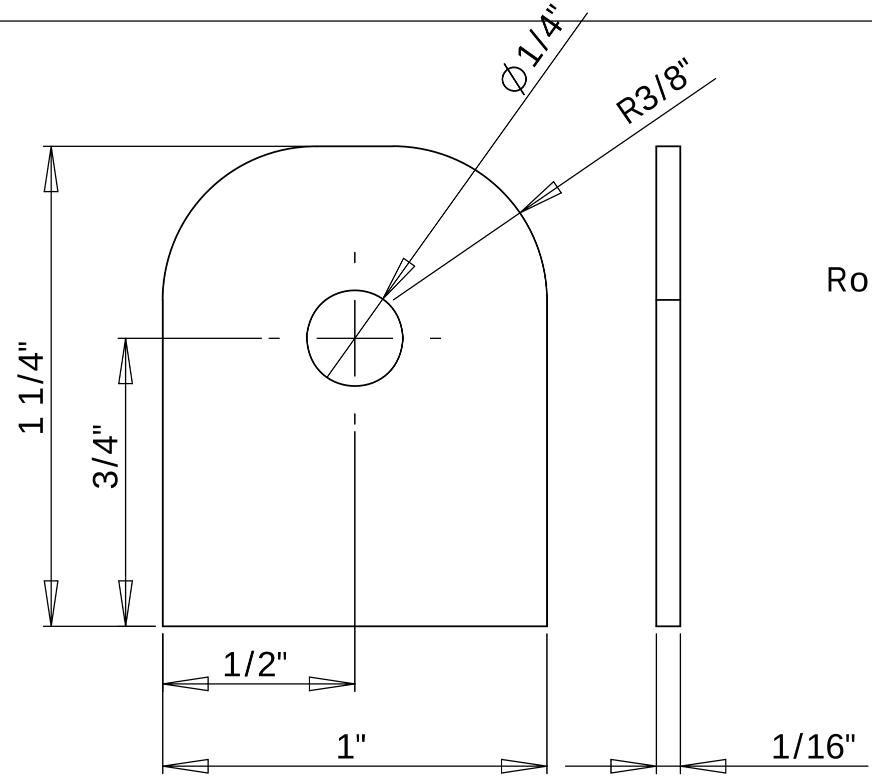
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE
	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
MATERIAL	Q.A.		
FINISH	COMMENTS:		
DO NOT SCALE DRAWING			

2006 PSU HPV TEAM

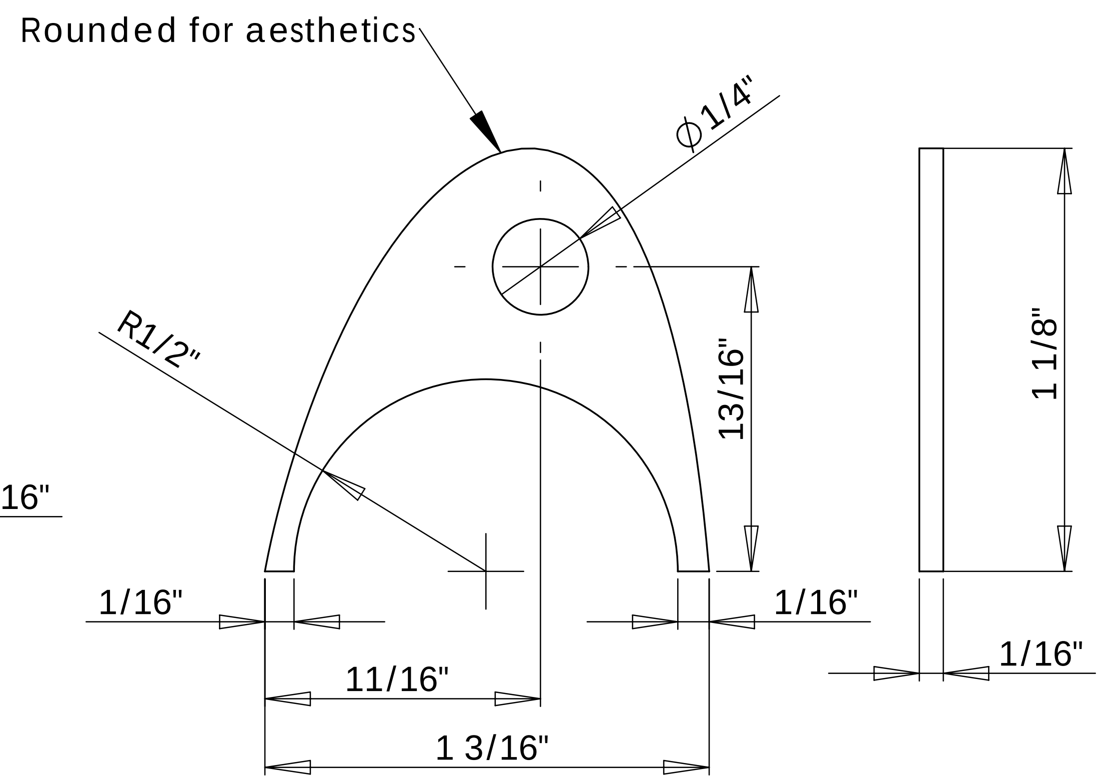
Rollbar

SIZE	DWG. NO.	REV.
A		

SCALE:1:10	WEIGHT:	SHEET 1 OF 1



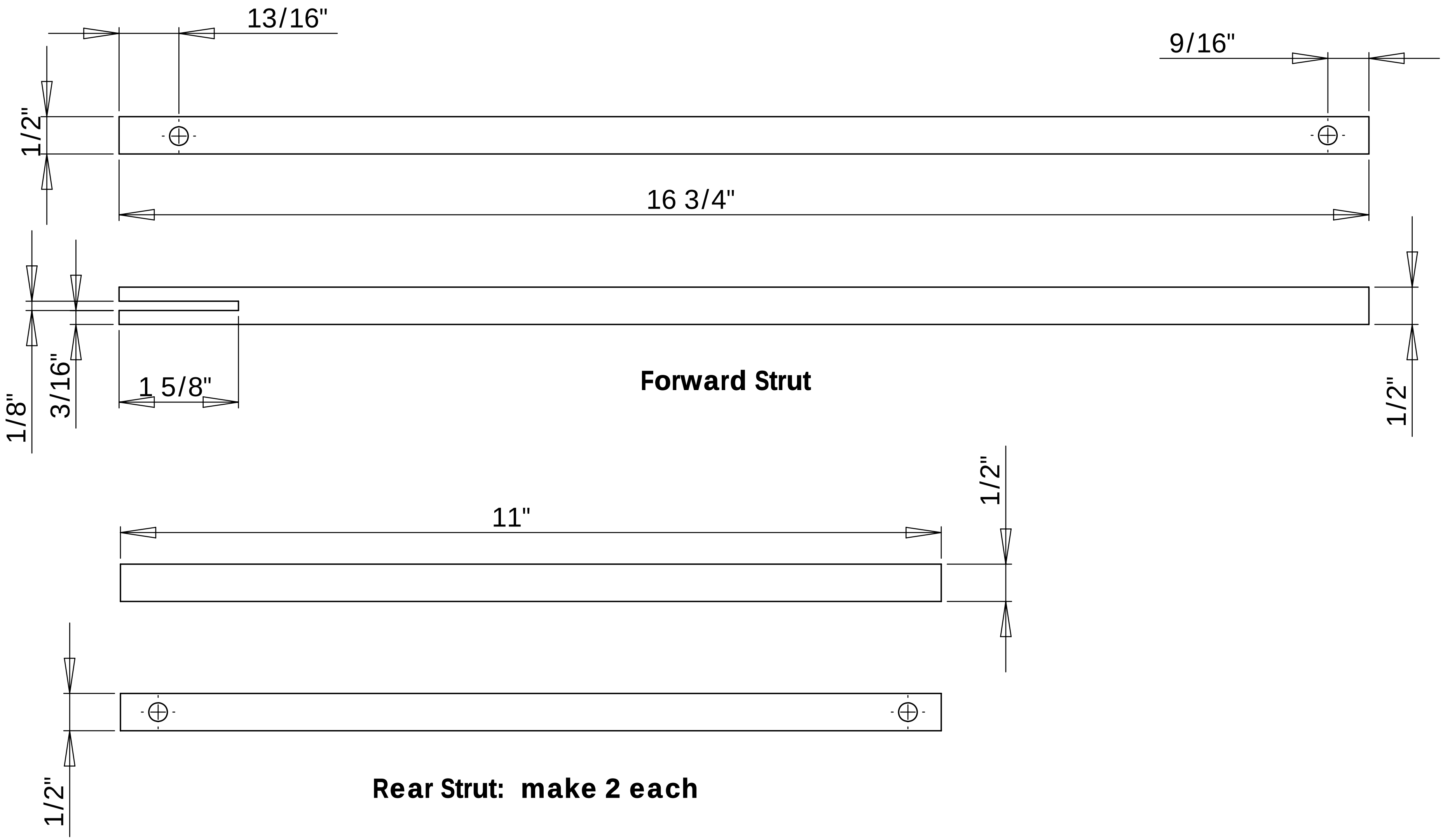
6 ea. to be made from 6061-T6 flatstock



4 ea. to be made from A36 flatstock

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			Rear Strut Brackets
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 2:1 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



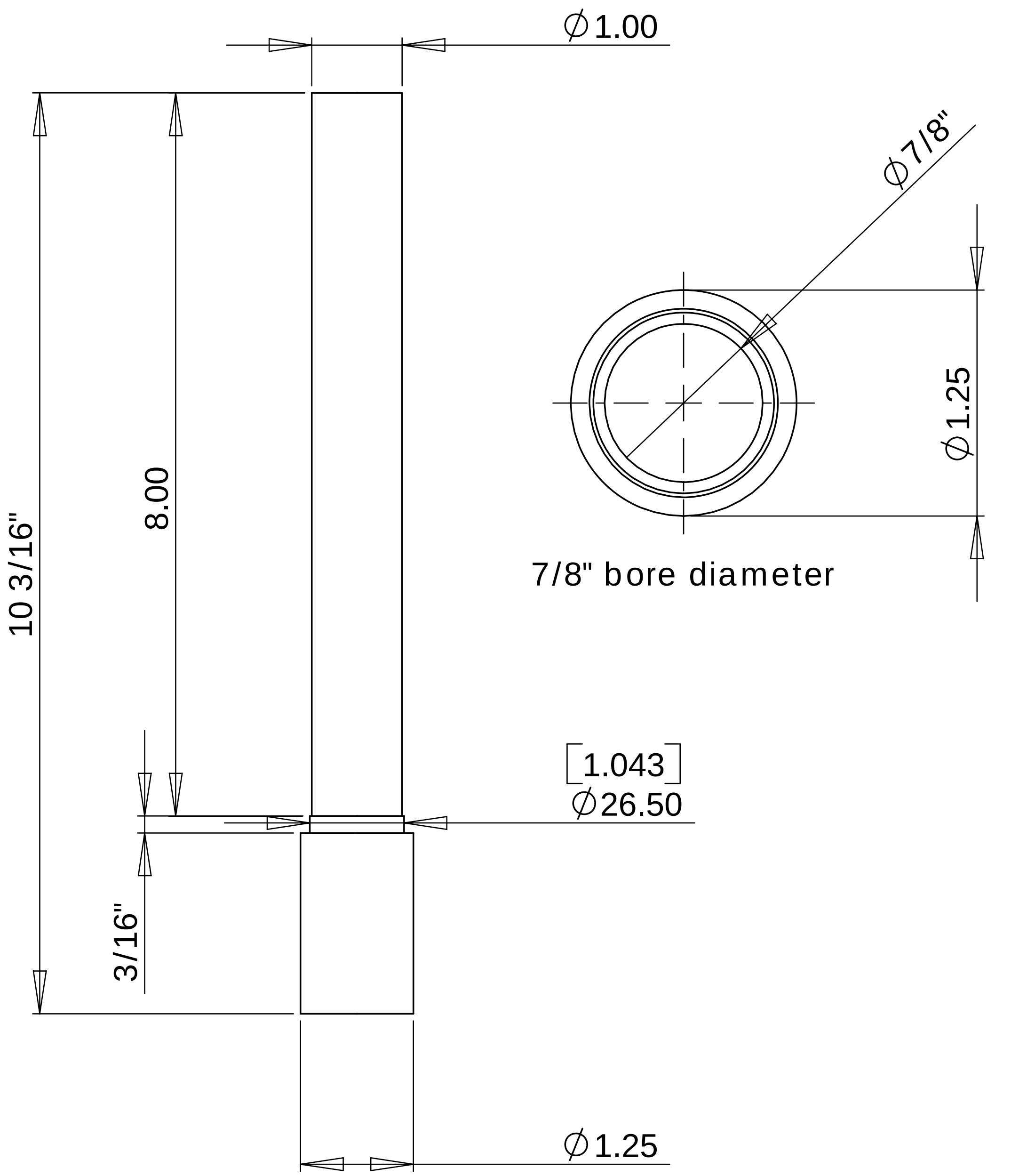
Notes:

1/2" square 6061-T6 tubing
1/16" wall thickness

All holes are 1/4" diameter
unless otherwise stated.

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			
	ENG APPR.			
	MFG APPR.			
	Q.A.			
INTERPRET GEOMETRIC TOLERANCING PER:	COMMENTS:			
MATERIAL				SIZE **A** | DWG. NO. | REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:5 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



1 1/4" 6061-T6 roundstock
to be turned in lathe to final dimensions

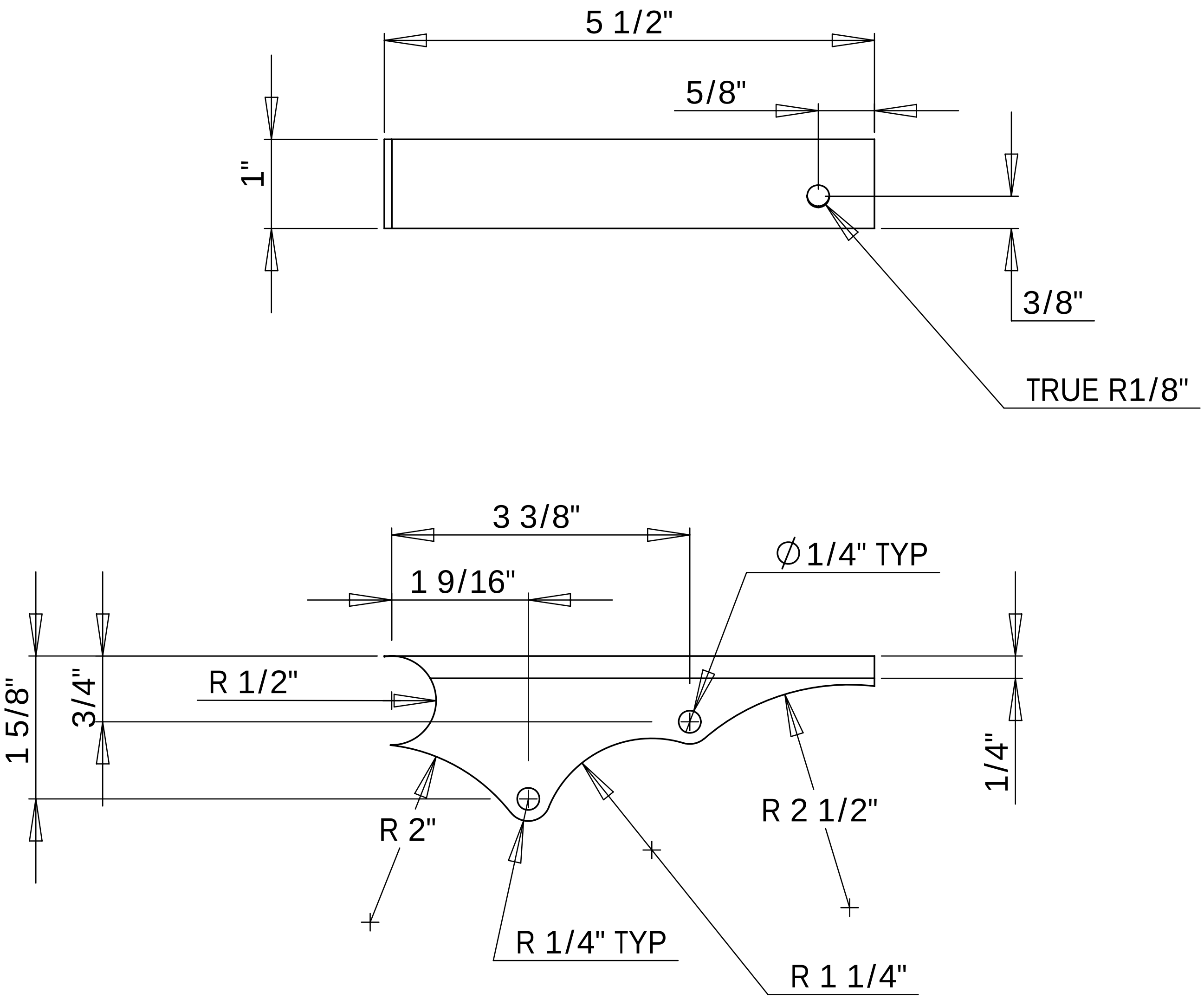
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE	
			DRAWN			
			CHECKED			
			ENG APPR.			
			MFG APPR.			
		MATERIAL	Q.A.			
		FINISH	COMMENTS:			
NEXT ASSY	USED ON					SIZE A DWG. NO. Steer Tube REV.
APPLICATION		DO NOT SCALE DRAWING				SCALE:1:5 WEIGHT: SHEET 1 OF 1

PROPRIETARY AND CONFIDENTIAL

THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.

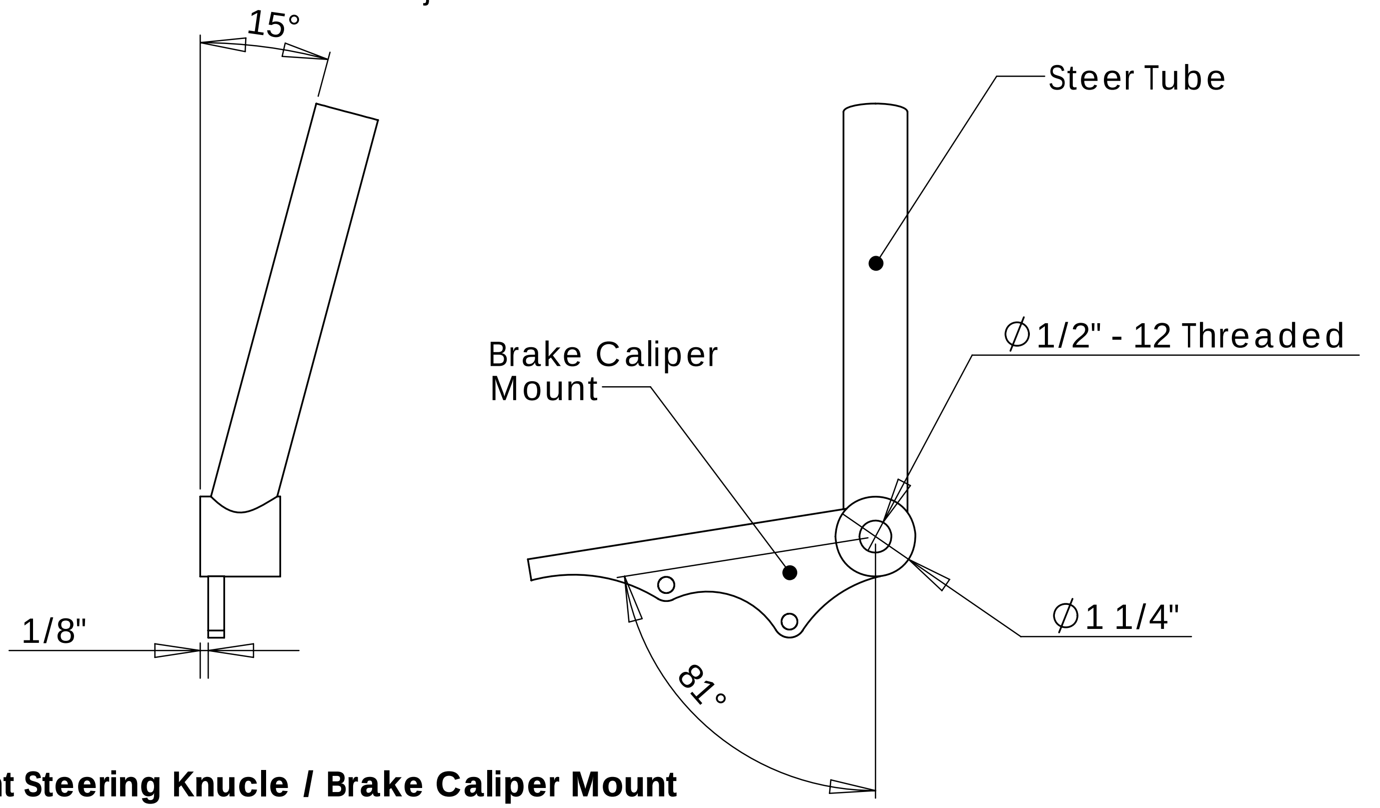
Notes:

Make 2 ea.
1/4" 6061-T6 angle stock

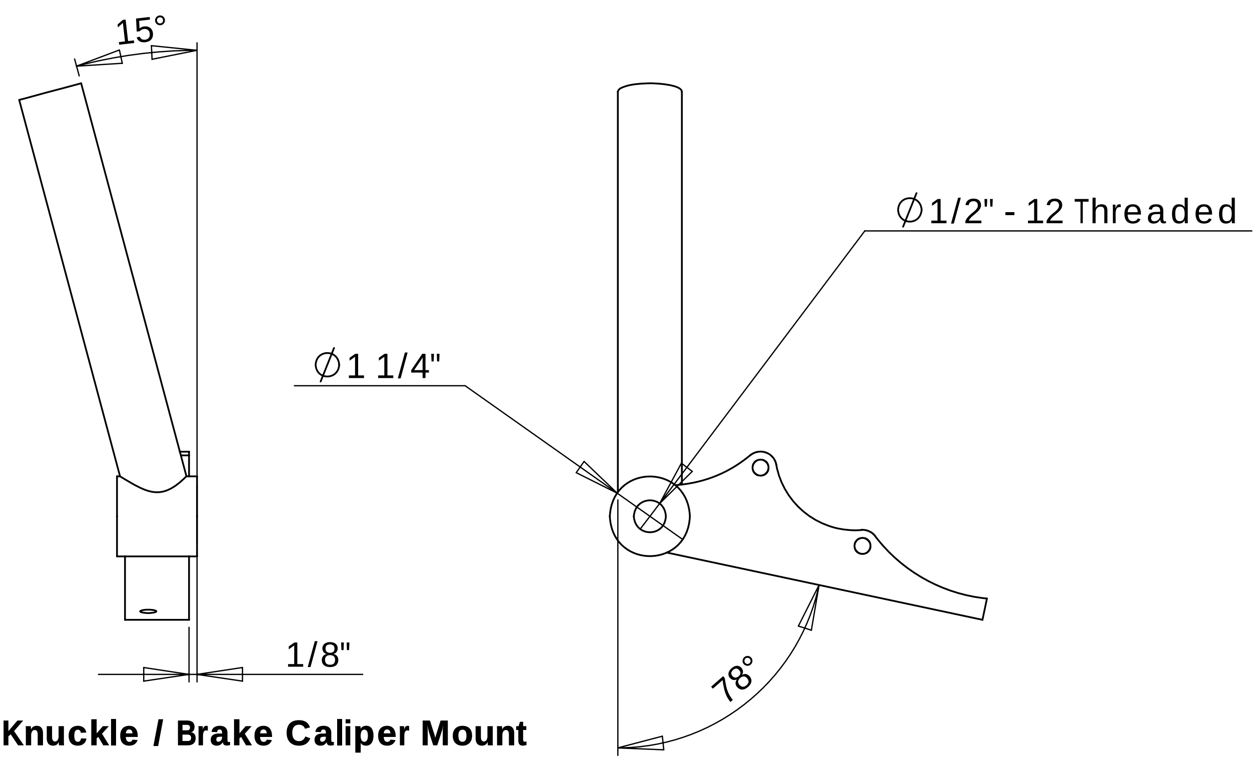


DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE	2006 PSU HPV TEAM
	DRAWN			
	CHECKED			Brake Caliper Mounts
	ENG APPR.			
	MFG APPR.			
MATERIAL	Q.A.			
FINISH	COMMENTS:			SIZE A · DWG. NO. · REV.
DO NOT SCALE DRAWING				SCALE:1:5 · WEIGHT: · SHEET 1 OF 1

All joints are to be welded



Right Steering Knucle / Brake Caliper Mount



Left Steering Knuckle / Brake Caliper Mount

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

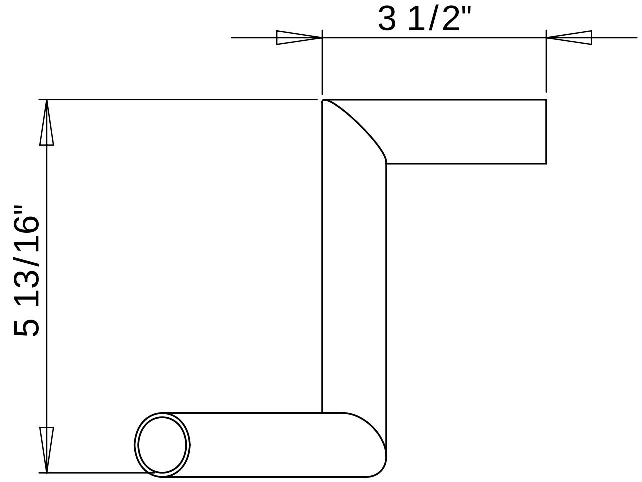
COMMENTS:

2006 PSU HPV TEAM

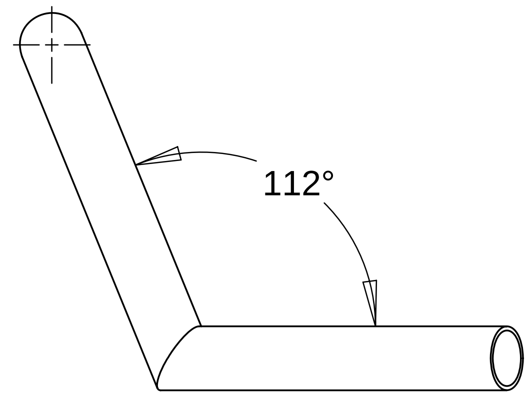
Steering Knucle and
Brake Caliper Mount

SIZE A	DWG. NO.	REV.

SCALE:1:5	WEIGHT:	SHEET 1 OF 1



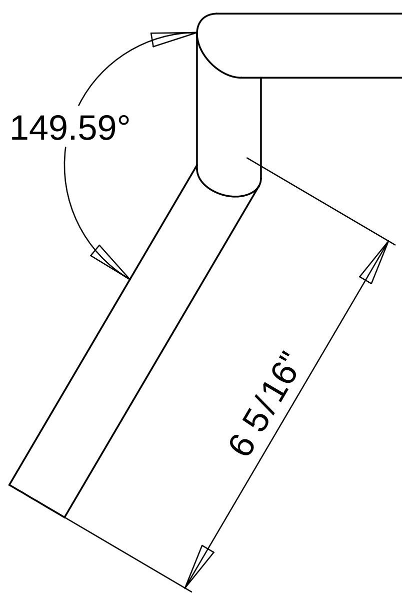
Front View



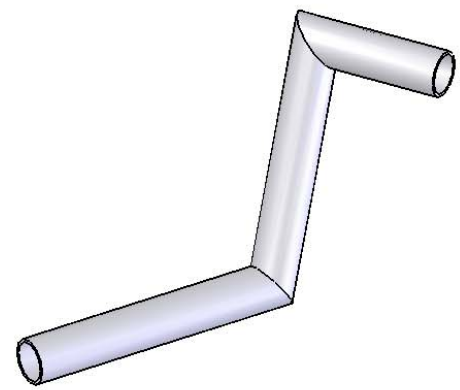
Left Side View

All parts are 6061-T6 Aluminum
1" OD, 0.065 wall thickness.

All joints to be welded.

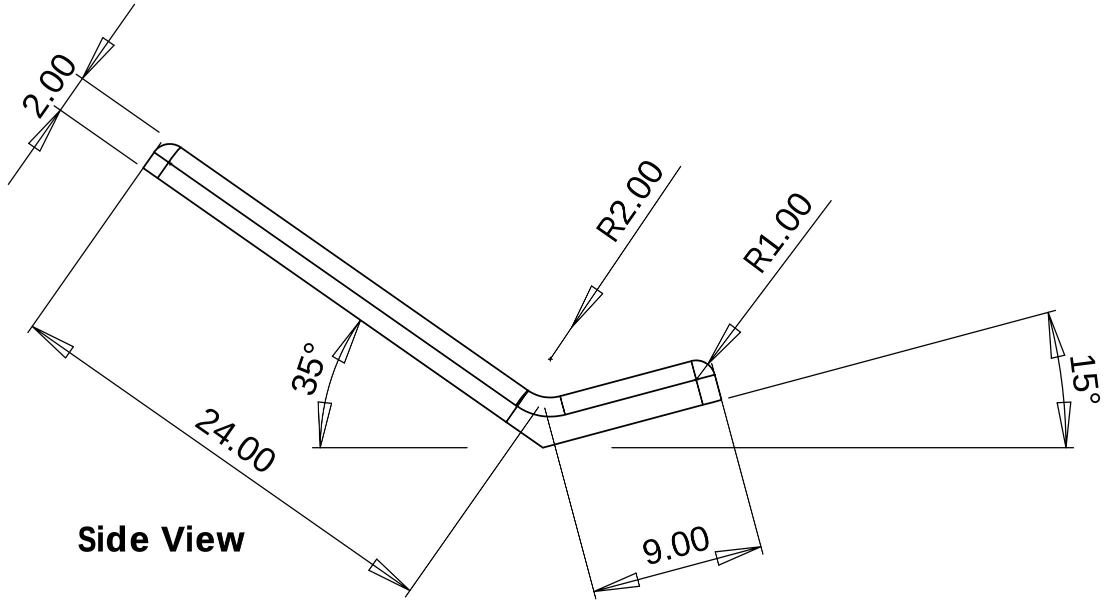


Top View

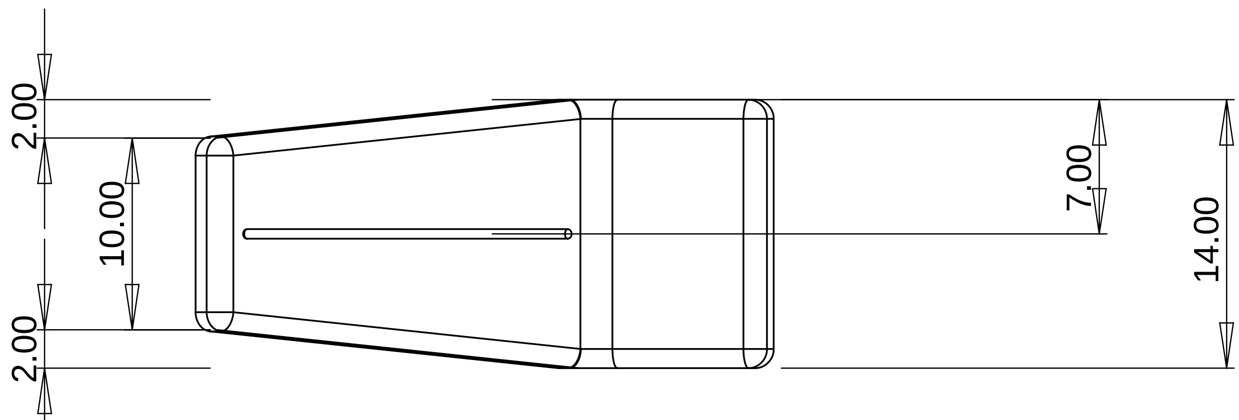


UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			Rider Interface
	ENG APPR.			
	MFG APPR.			
	Q.A.			
INTERPRET GEOMETRIC TOLERANCING PER:	COMMENTS:			SIZE A / DWG. NO. / REV
MATERIAL				
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:2 / WEIGHT: / SHEET 1 OF 1

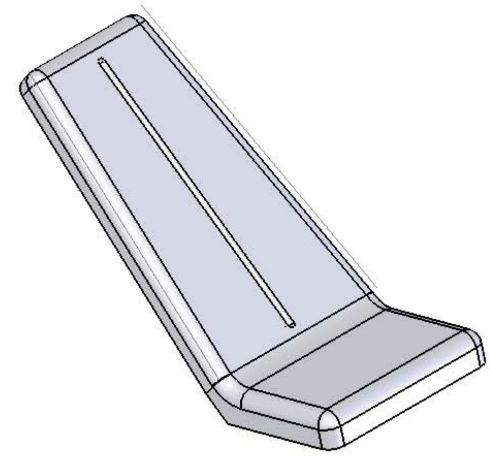
5 4 3 2 1



Side View



Top View



Notes:

Seat is to be fabribated from
5 layers of 10 oz carbon fiber cloth
in an epoxy-resin wet lay-up process.

Final product is to be 60% fiber, 40% resin.

		NAME	DATE
UNLESS OTHERWISE SPECIFIED:			
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

2006 PSU HPV TEAM

TITLE:

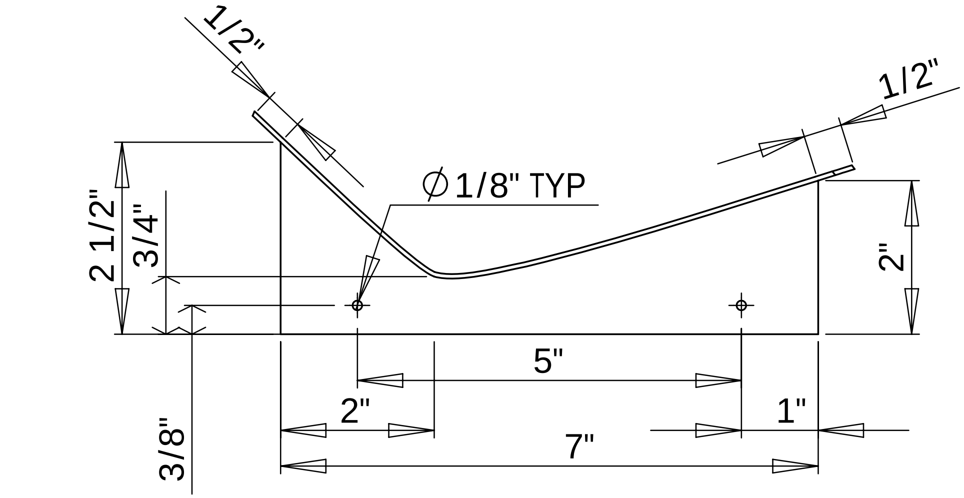
Carbon Fiber Seat

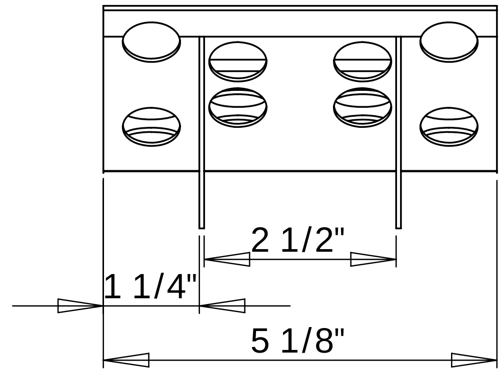
SIZE	DWG. NO.	REV
A		

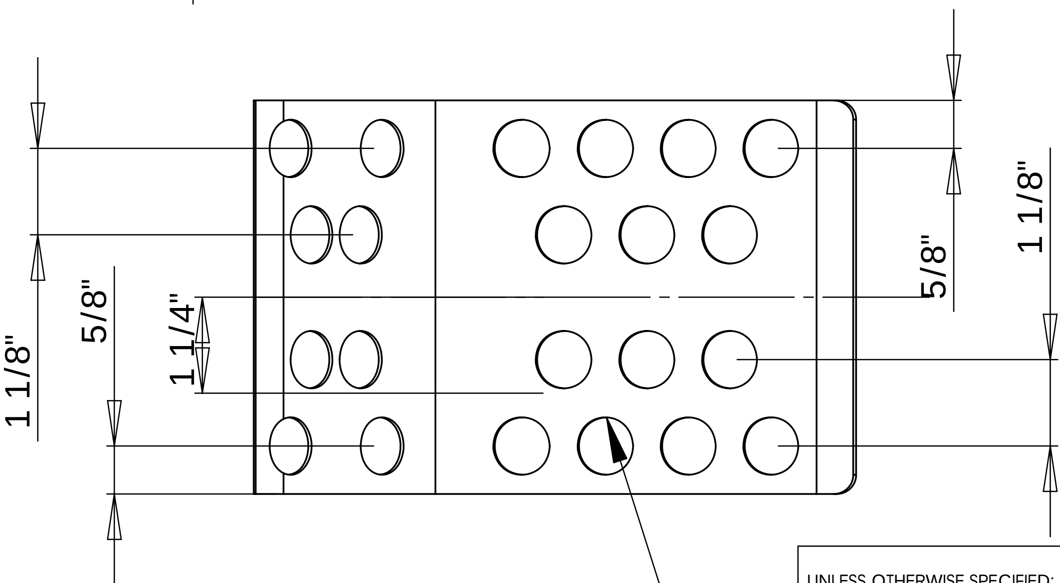
SCALE: 1:10 WEIGHT: SHEET 1 OF 1

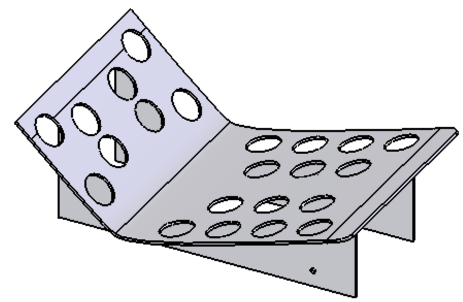
5 4 3 2 1

All joints to be welded throughout
1/16" A36 flatstock







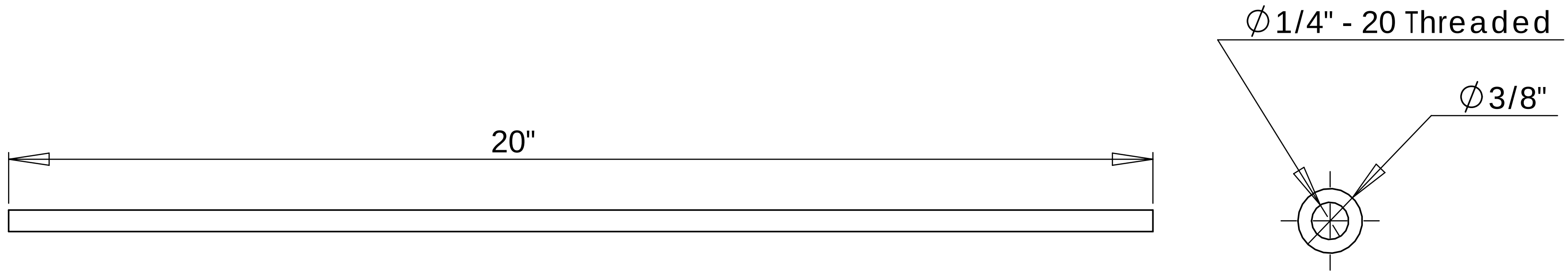


Holes to be added in general configuration for lightening. Dimensional accuracy not required.

Diameter = 3/4"

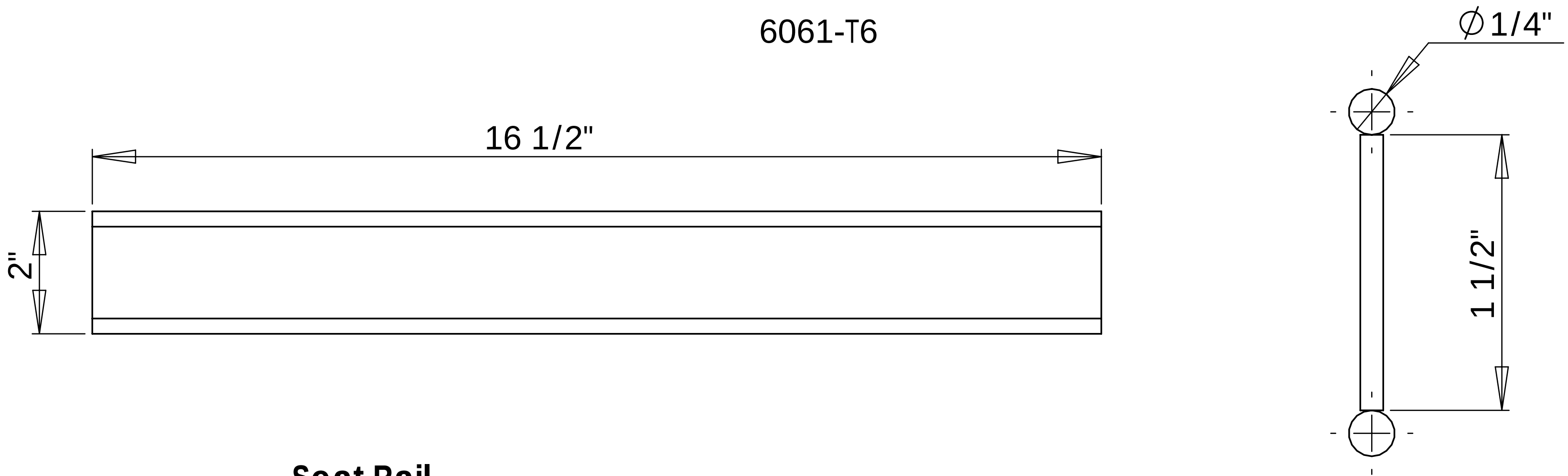
UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			
	ENG APPR.			Seat Bracket
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A	DWG. NO.	REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:2	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1



Tie Rod

6061-T6



Seat Rail

A36 flatstock unless otherwise noted
All joints are to be welded

UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

2006 PSU HPV TEAM

TITLE:

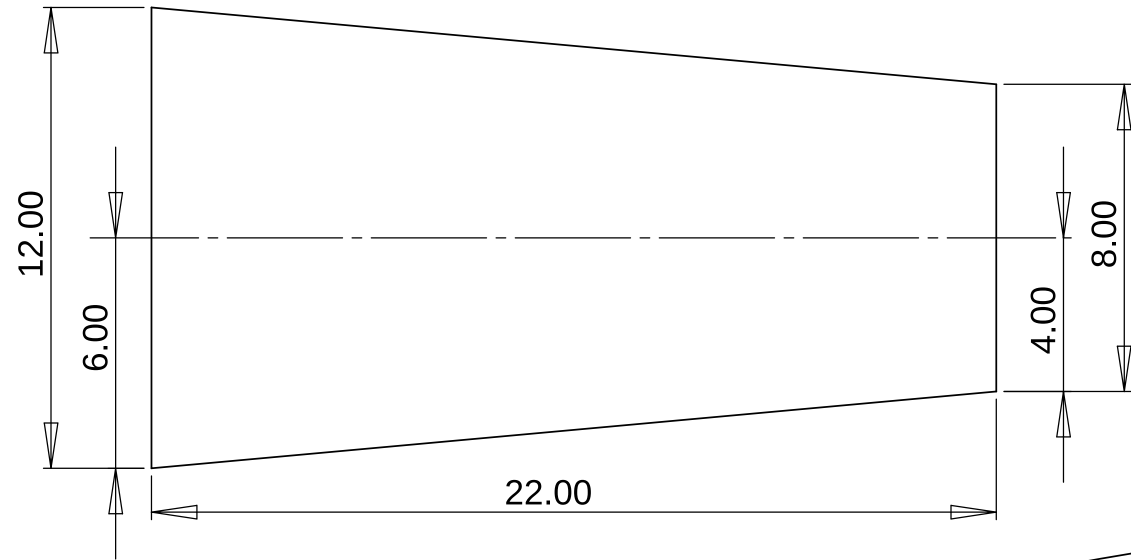
Tie Rod and Seat Rail

SIZE	DWG. NO.	REV
A		

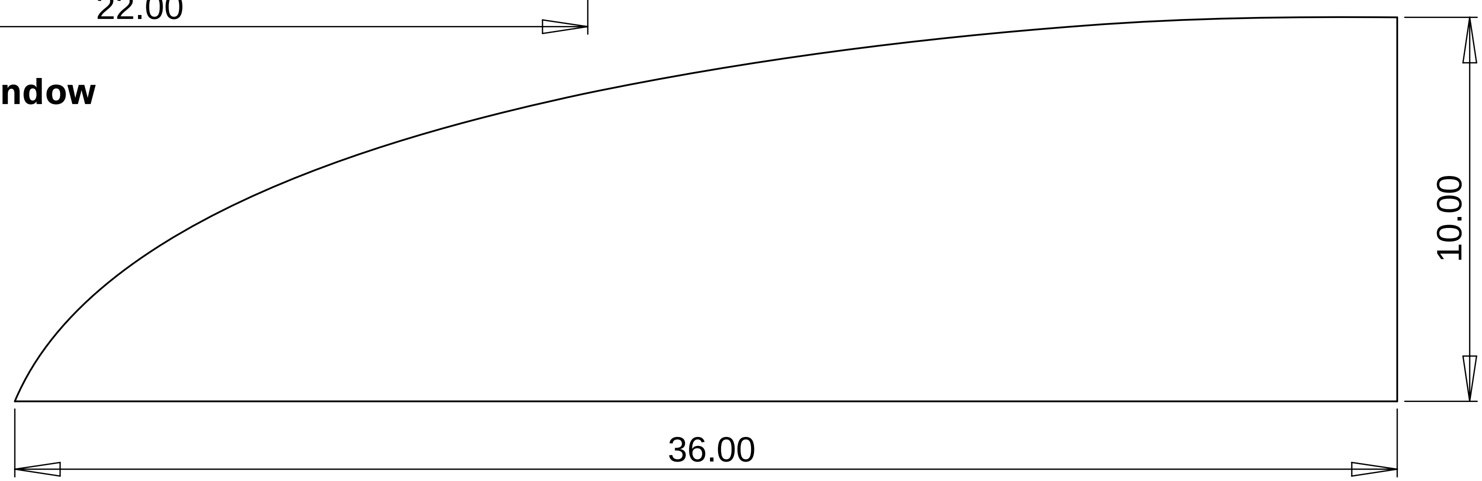
SCALE: 1:5	WEIGHT:	SHEET 1 OF 1

5 4 3 2 1

Windows cut from 1/32" polycarbonate.



Top Window

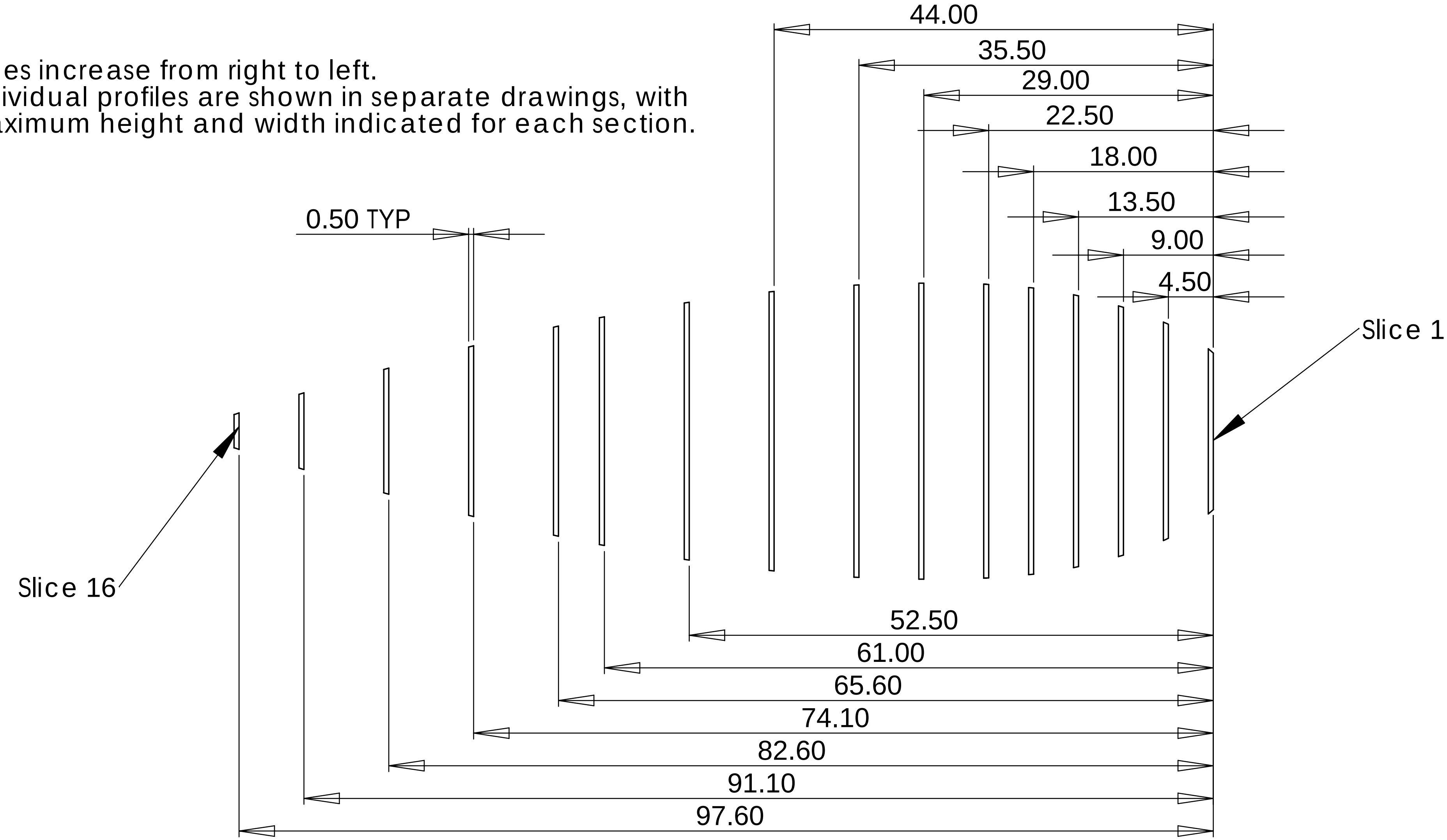


Side Window: Cut 2 ea.

UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			Windows
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A / DWG. NO. / REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:5 / WEIGHT: / SHEET 1 OF 1

5 4 3 2 1

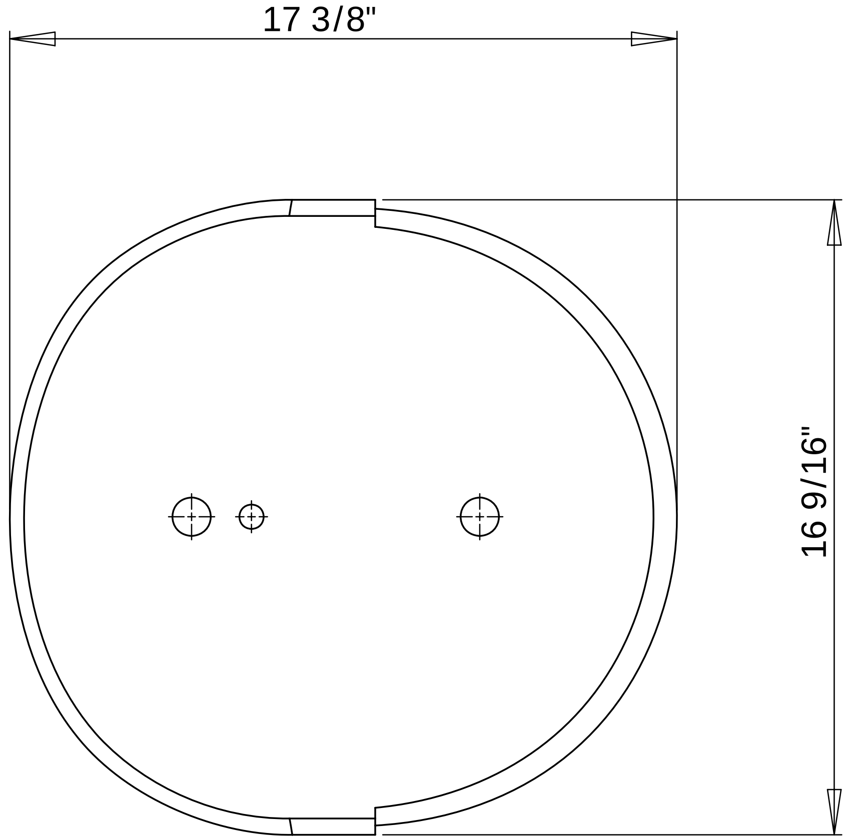
Slices increase from right to left.
Individual profiles are shown in separate drawings, with maximum height and width indicated for each section.



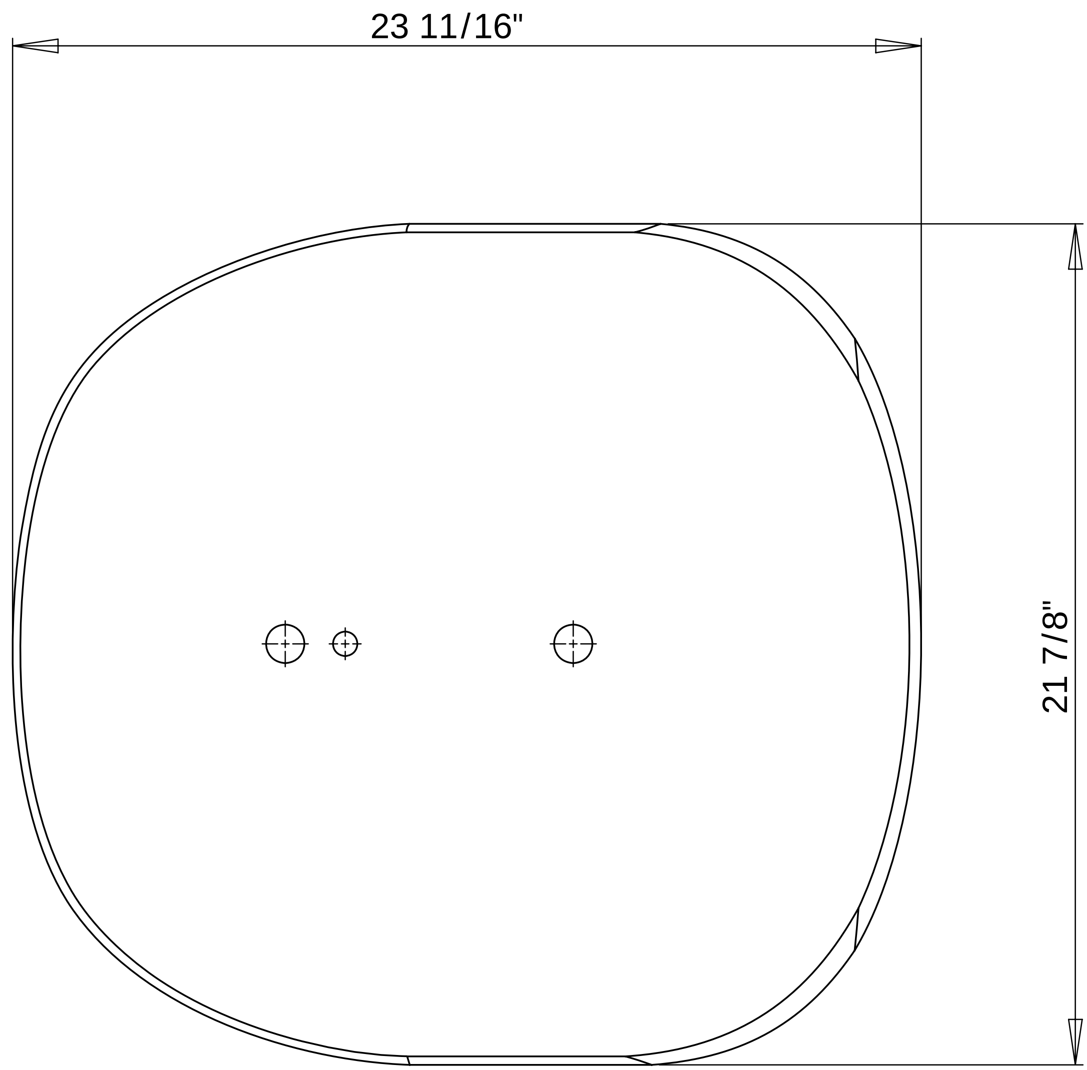
UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			Slice Layout
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:50 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1

For fabrication, all slices should allign so that holes match up.

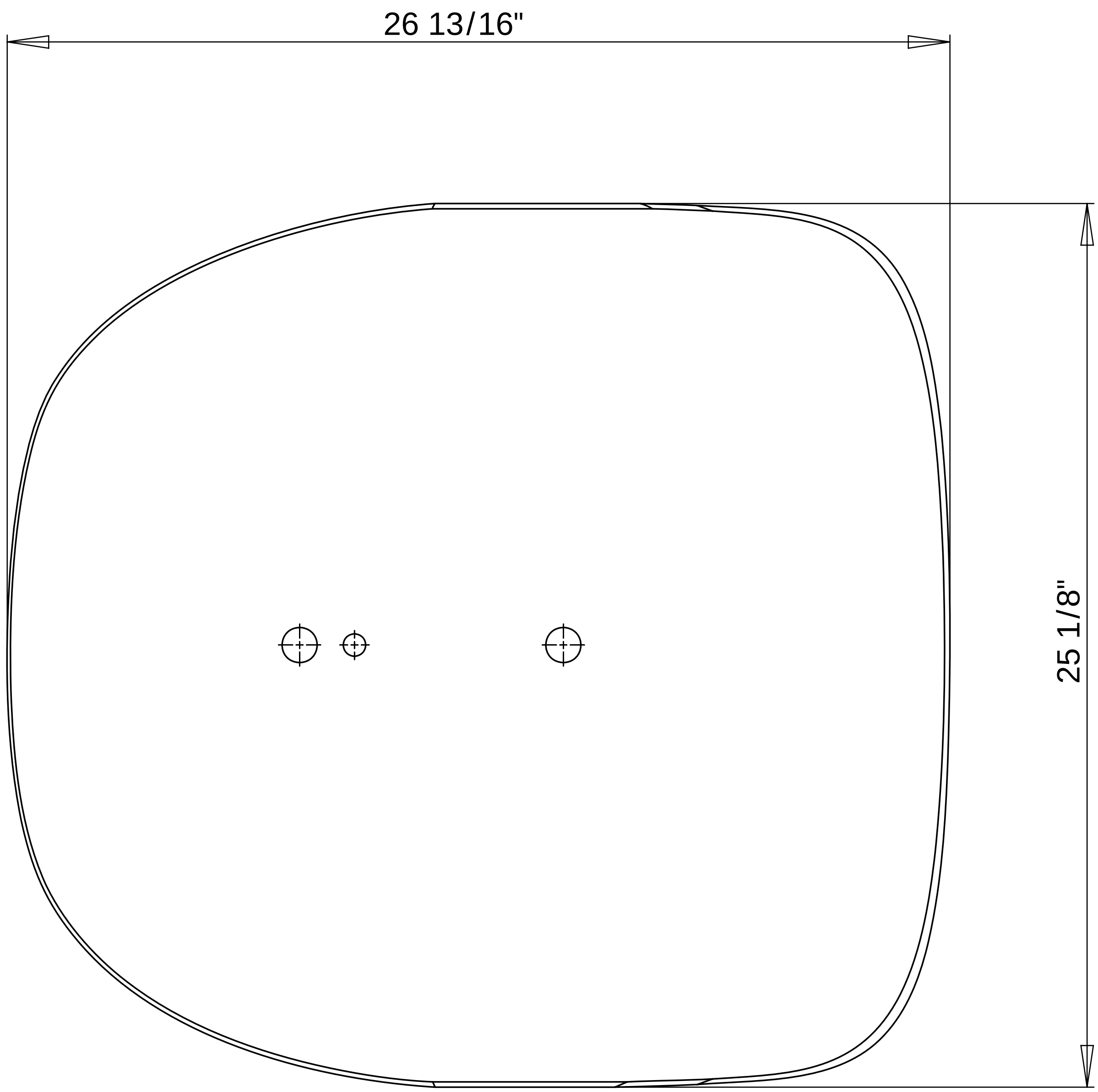


DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE	2006 PSU HPV TEAM
	DRAWN			
	CHECKED			Slice 1
	ENG APPR.			
	MFG APPR.			
MATERIAL	Q.A.			
FINISH	COMMENTS:			SIZE A — DWG. NO. — REV.
DO NOT SCALE DRAWING				SCALE:1:5 — WEIGHT: — SHEET 1 OF 1



DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE
	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
MATERIAL	Q.A.		
FINISH	COMMENTS:		
DO NOT SCALE DRAWING			

2006 PSU HPV TEAM		
Slice 2		
SIZE A	DWG. NO.	REV.
SCALE:1:10	WEIGHT:	SHEET 1 OF 1



DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

MATERIAL

FINISH

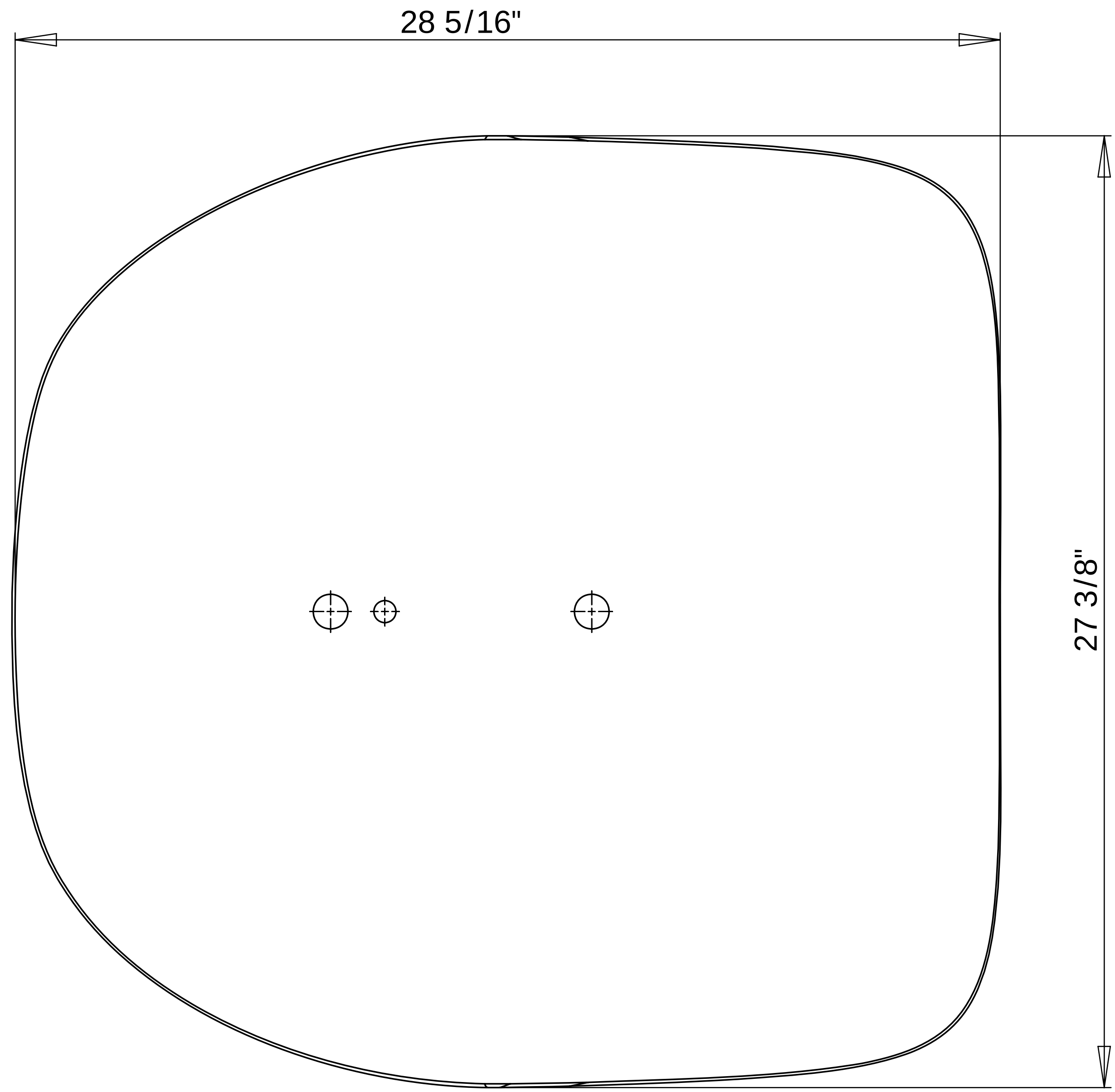
DO NOT SCALE DRAWING

2006 PSU HPV TEAM

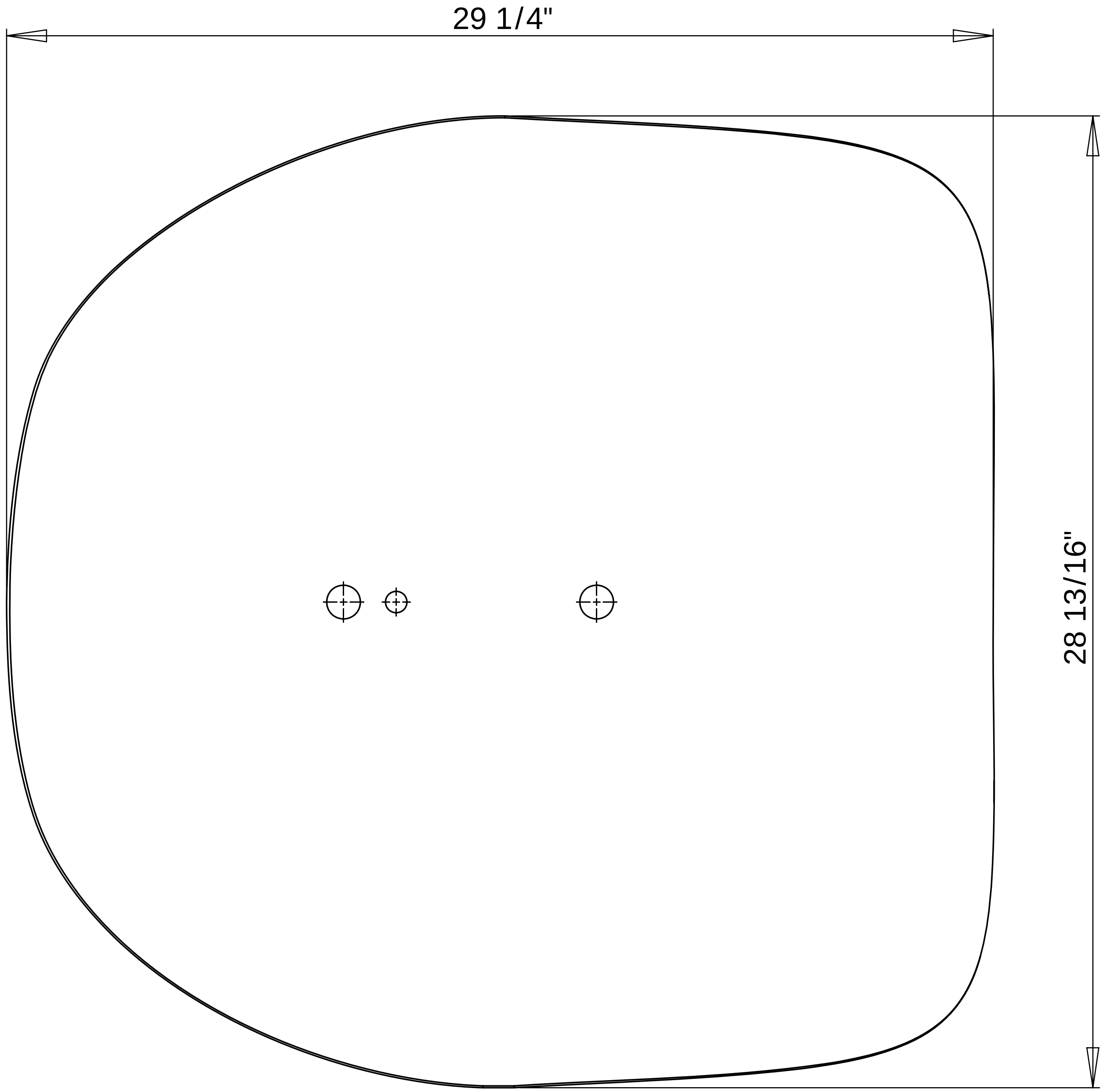
Slice 3

SIZE A | DWG. NO. | REV.

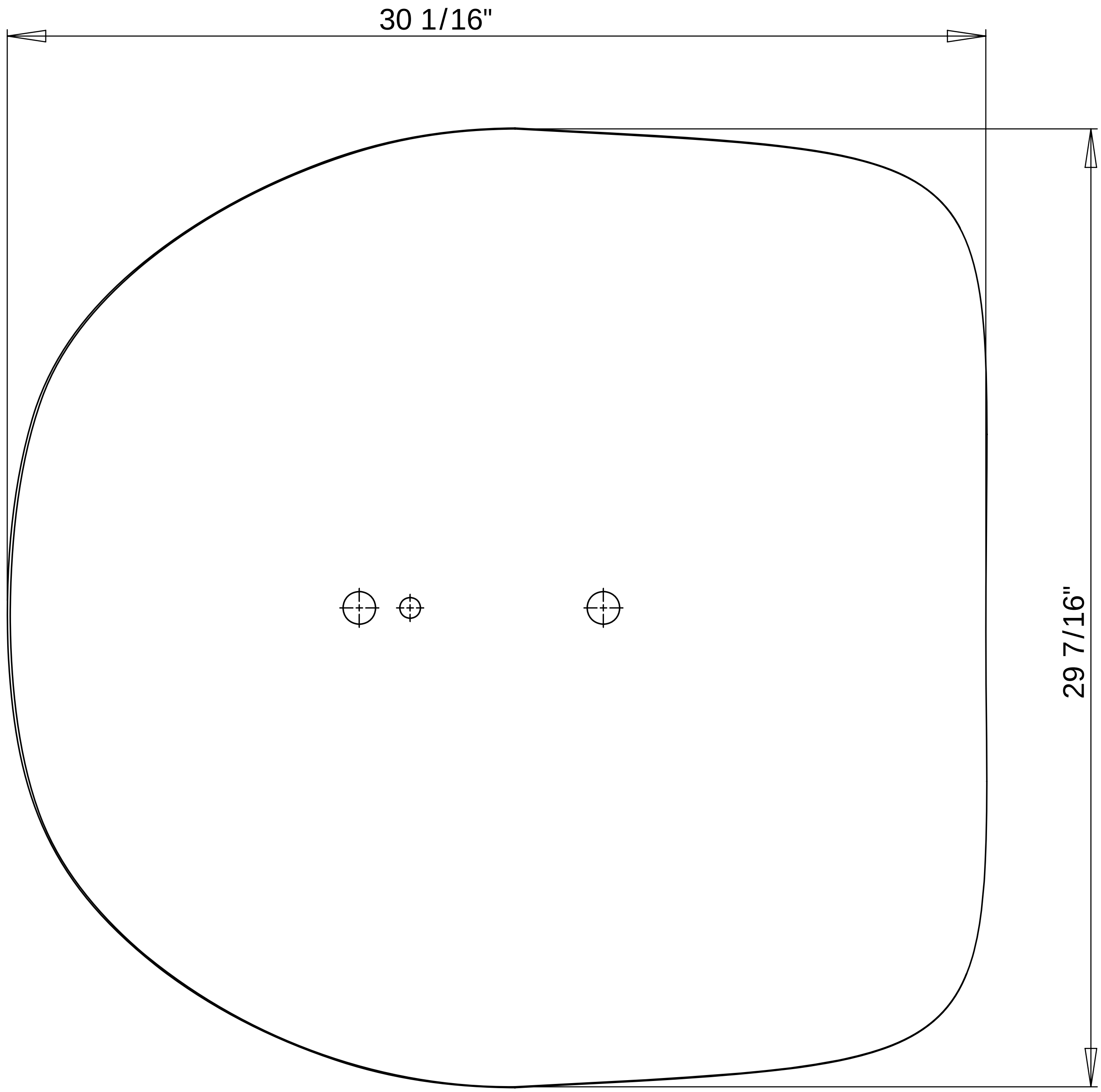
SCALE:1:10 | WEIGHT: | SHEET 1 OF 1



DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE	2006 PSU HPV TEAM
	DRAWN			Slice 4
	CHECKED			
	ENG APPR.			
	MFG APPR.			
MATERIAL	Q.A.			
FINISH	COMMENTS:			SIZE A · DWG. NO. · REV.
DO NOT SCALE DRAWING				SCALE:1:10 · WEIGHT: · SHEET 1 OF 1



DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE	2006 PSU HPV TEAM
	DRAWN			
	CHECKED			Slice 5
	ENG APPR.			
	MFG APPR.			
	Q.A.			
MATERIAL	COMMENTS:			
FINISH				SIZE A · DWG. NO. · REV.
DO NOT SCALE DRAWING				SCALE:1:10 · WEIGHT: · SHEET 1 OF 1



DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

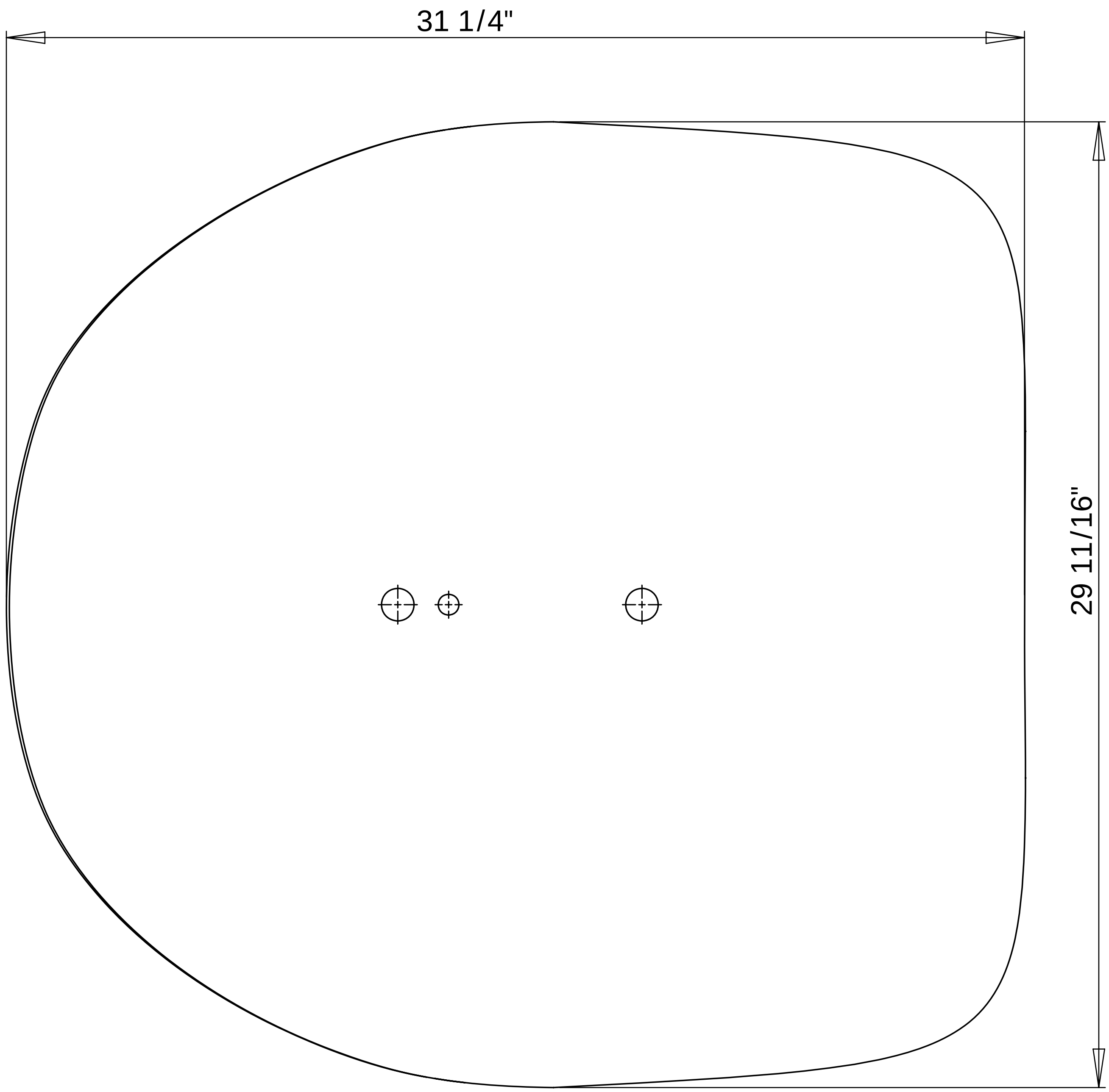
COMMENTS:

2006 PSU HPV TEAM

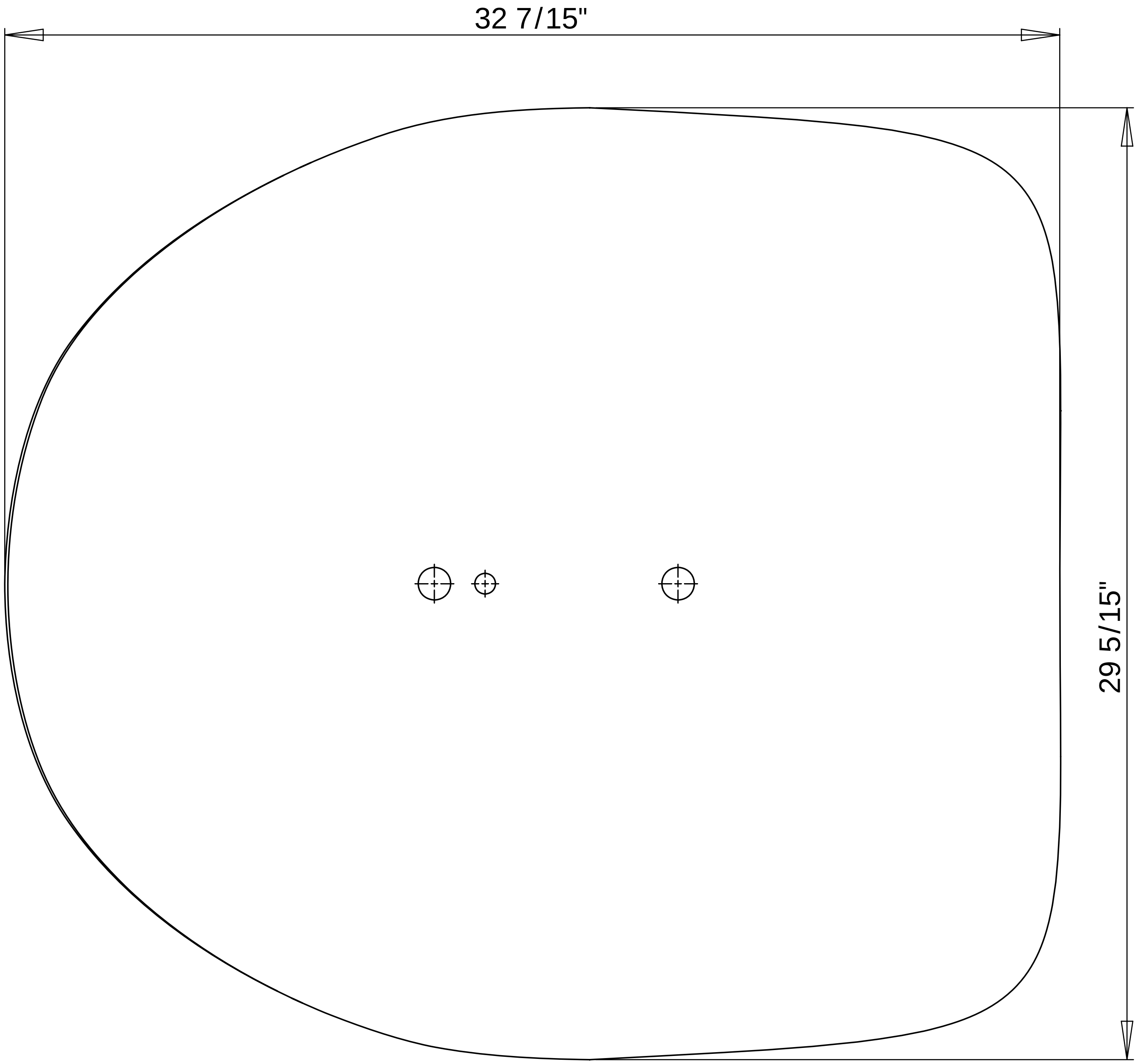
Slice 6

SIZE A | DWG. NO. | REV.

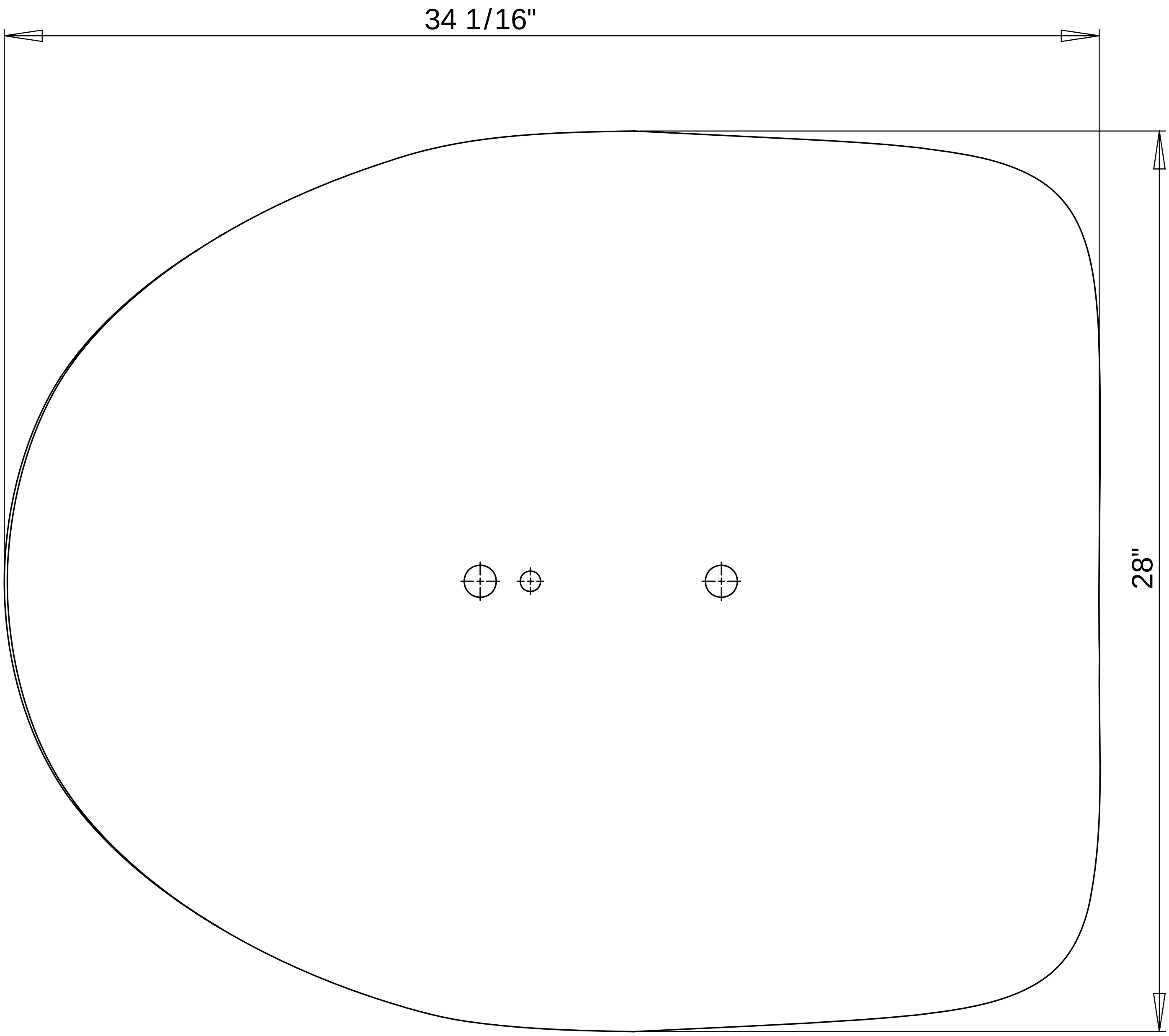
SCALE:1:10 | WEIGHT: | SHEET 1 OF 1



DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE	2006 PSU HPV TEAM
	DRAWN			
	CHECKED			Slice 7
	ENG APPR.			
	MFG APPR.			
MATERIAL	Q.A.			
FINISH	COMMENTS:			SIZE A / DWG. NO. / REV.
DO NOT SCALE DRAWING				SCALE:1:10 / WEIGHT: / SHEET 1 OF 1



		NAME	DATE	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			2006 PSU HPV TEAM
	CHECKED			Slice 8
	ENG APPR.			
	MFG APPR.			
MATERIAL	Q.A.			
FINISH	COMMENTS:			SIZE A — DWG. NO. — REV.
DO NOT SCALE DRAWING				SCALE:1:10 — WEIGHT: — SHEET 1 OF 1



DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

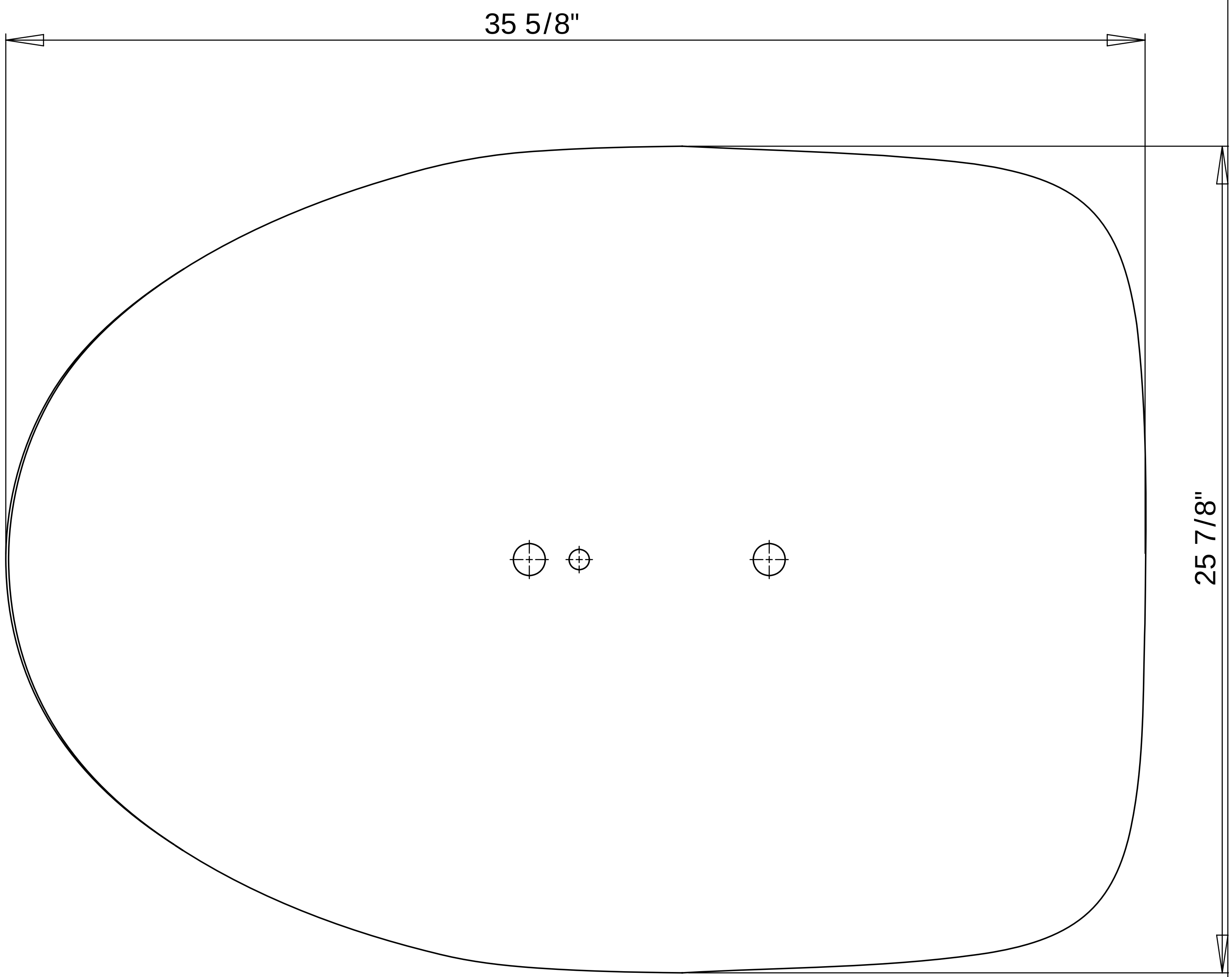
COMMENTS:

2006 PSU HPV TEAM

Slice 9

SIZE	DWG. NO.	REV.
A		

SCALE:1:10	WEIGHT:	SHEET 1 OF 1

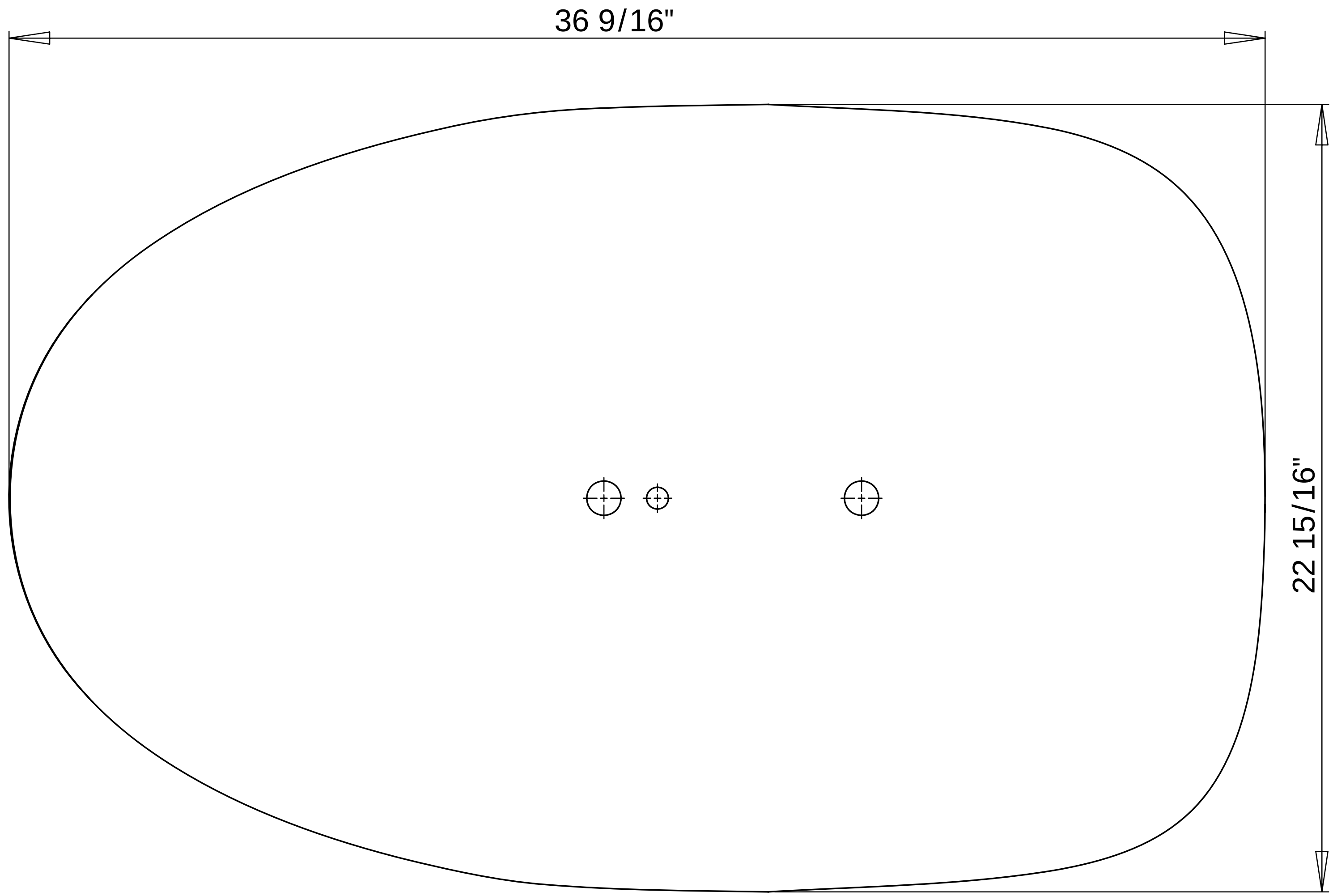


DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±		NAME	DATE
	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
MATERIAL	Q.A.		
FINISH	COMMENTS:		
DO NOT SCALE DRAWING			

2006 PSU HPV TEAM

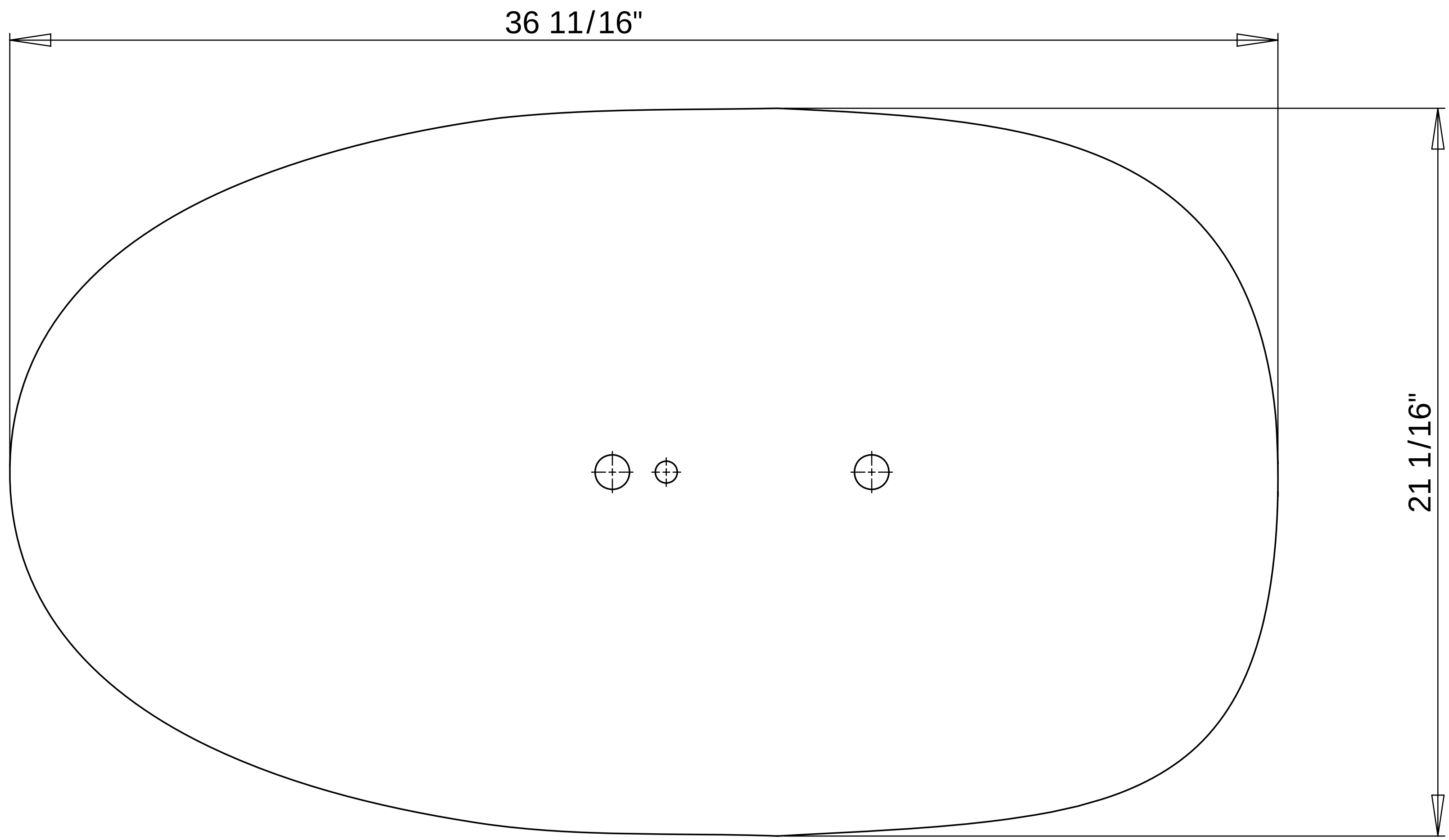
Slice 10

SIZE	DWG. NO.	REV.
A		
SCALE:1:10	WEIGHT:	SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			Slice 11
	ENG APPR.			
	MFG APPR.			
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
MATERIAL	COMMENTS:			SIZE A · DWG. NO. · REV
FINISH				
DO NOT SCALE DRAWING				SCALE: 1:10 · WEIGHT: · SHEET 1 OF 1

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL± ANGULAR: MACH± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN		
	CHECKED		
	ENG APPR.		
	MFG APPR.		
INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
MATERIAL	COMMENTS:		
FINISH			
DO NOT SCALE DRAWING			

2006 PSU HPV TEAM

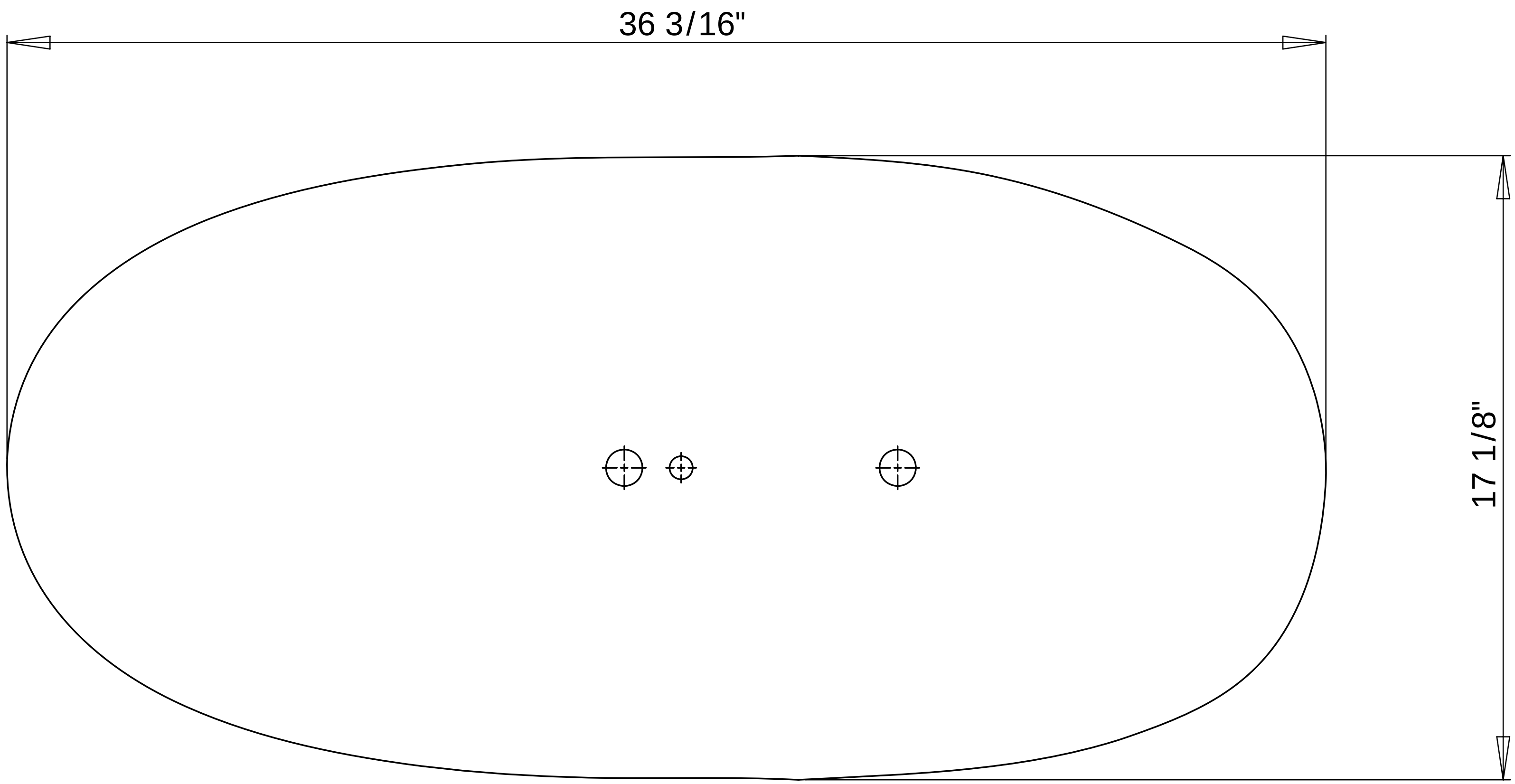
TITLE:

Slice 12

SIZE	DWG. NO.	REV
A		

SCALE: 1:10 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



UNLESS OTHERWISE SPECIFIED:		NAME	DATE	2006 PSU HPV TEAM
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±	DRAWN			TITLE:
	CHECKED			Slice 13
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	MFG APPR.			
	Q.A.			
INTERPRET GEOMETRIC TOLERANCING PER:	COMMENTS:			SIZE A
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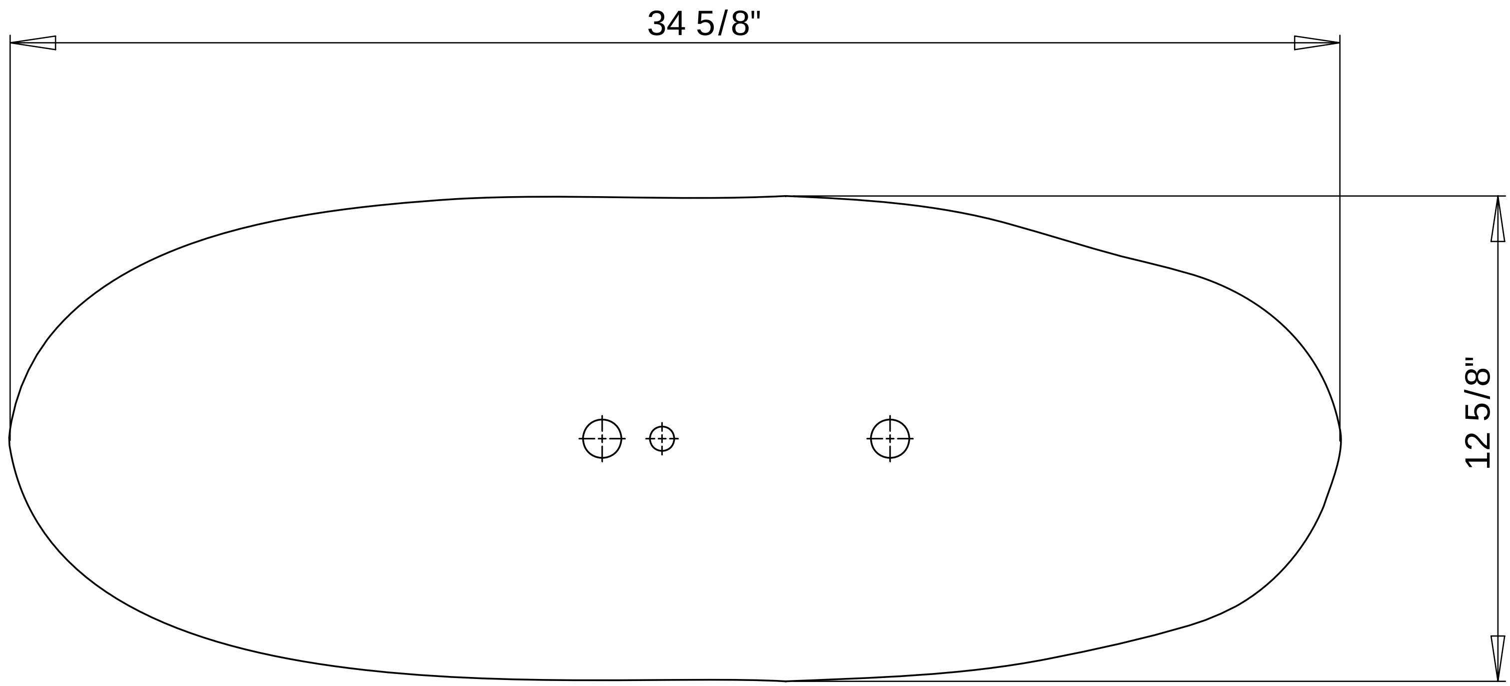
FINISH				REV
DO NOT SCALE DRAWING				SCALE: 1:10
				WEIGHT:
				SHEET 1 OF 1

5 4 3 2 1



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FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
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2006 PSU HPV TEAM

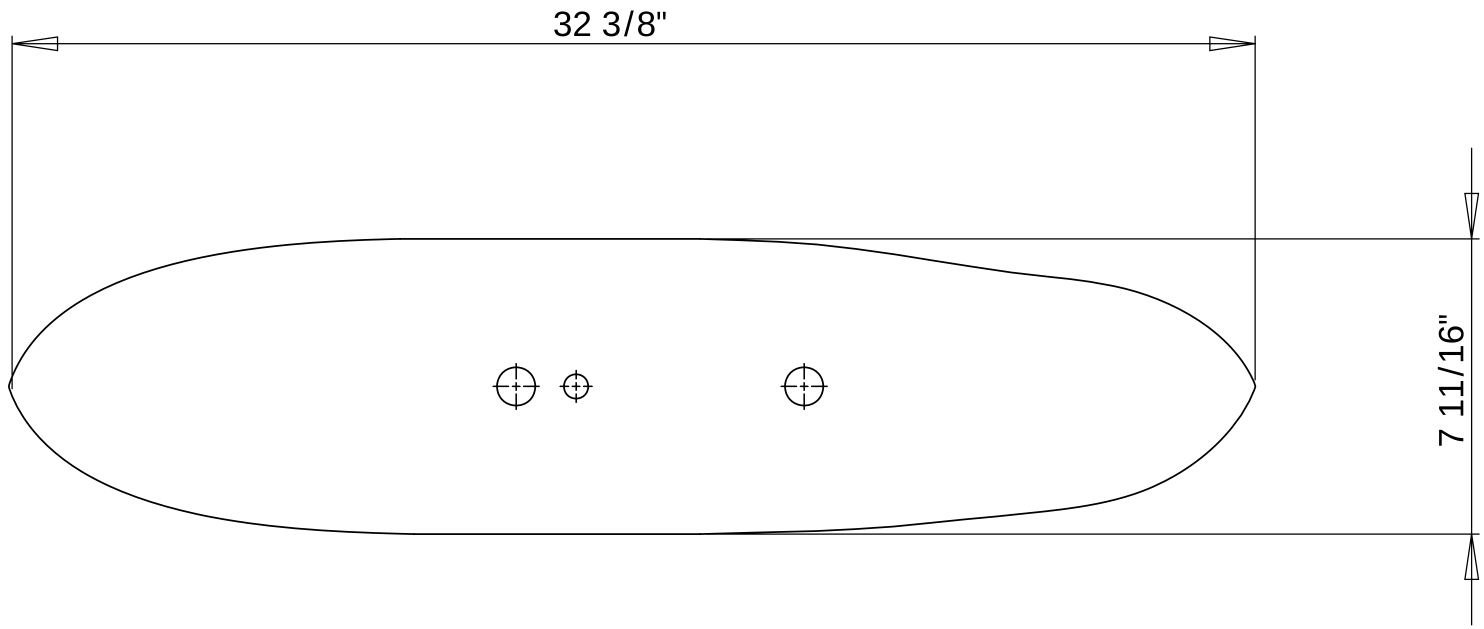
TITLE:

Slice 14

SIZE **A** DWG. NO. REV

SCALE: 1:5 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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TWO PLACE DECIMAL ±
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INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN		
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ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

2006 PSU HPV TEAM

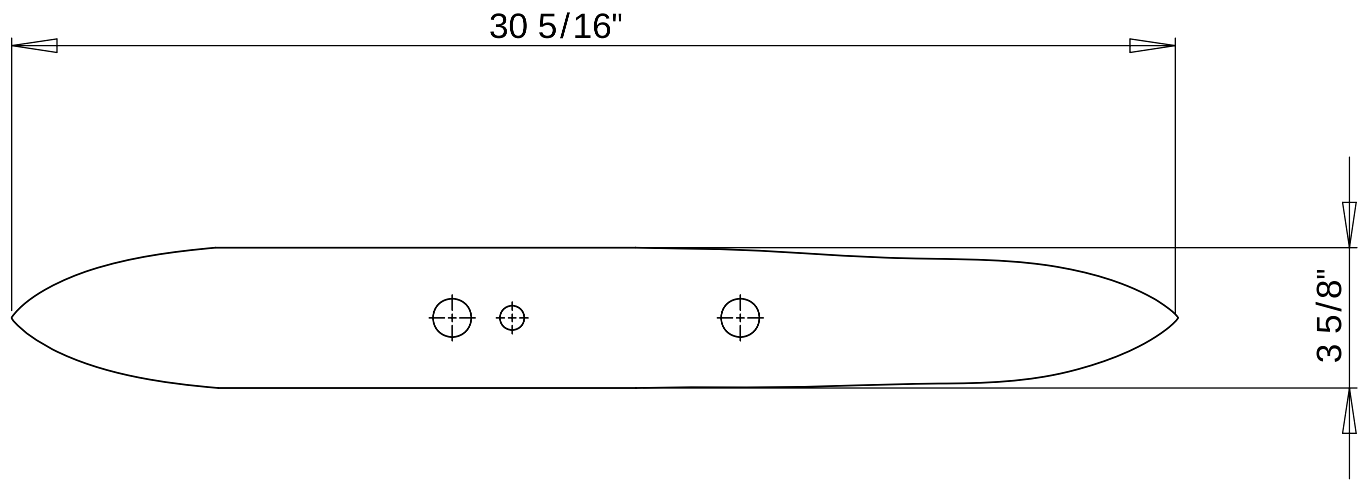
TITLE:

Slice 15

SIZE **A** DWG. NO. REV

SCALE: 1:5 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.			
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FINISH				
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5 4 3 2 1

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[10] Acknowledgements

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