Design and Analysis of the BRAE Tractor Pull Tow Vehicle

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ABSTRACT

This senior project discusses the design and analysis of a towing vehicle for the tractor pulling team and the BRAE department. This report examines different types of existing hand operated towing vehicles and their strengths and weaknesses and how they can be modified to design a custom tow vehicle for the BRAE department. The purpose of this report was to make transporting and handling of the modified pulling tractors in the BRAE department safer for the operator and bystanders.

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INTRODUCTION

The Cal Poly tractor pull team has two full sized modified tractors that are used to compete across California. Each year, the team competes in various tractor pull competitions throughout the season. The students involved with the team are responsible for moving each 3 – 4 ton machine to and from the pulls. Because of this, the tractors are handled frequently and are constantly being moved around the BRAE facilities and in the pits at a competition. The current method that is used to move the pulling tractors in and out of shop five of the BRAE department is with a forklift and tow strap. The forklift fork attachment has a hook on the end that the tow strap latches on to. The strap is then wrapped around the front of the frame and the front of the tractor is lifted off the ground. This allows the tractor to pivot on the back tires which is required in order to maneuver them into their current storage space. This method is not only cumbersome, it is inefficient and relatively unsafe.

The objective of this project was to design a hand operated vehicle that would replace the forklift assembly that would be much more maneuverable, safe, and efficient. This vehicle would be taken to each event and be used to move the tractors for display and pit purposes. This project is rewarding because there is a possibility of it being made in a future BRAE class or a future senior project.

LITERATURE REVIEW

Hand Operated Tow Vehicles. The most common type of hand operated towing vehicle is an aircraft tug. These hand trucks work by pulling the front wheel of the aircraft up onto the vehicle. This puts the weight of the front of the aircraft onto the hand truck giving the hand truck more traction upon which the torque acts. This is the alternative to hitching to the front wheel of the aircraft and then pulling the aircraft without any of the weight on the hand truck. The problem with that option is that the wheels will tend to slip.

Figure 1: Image of an aircraft tug.

Figure 1 shows an aircraft tug made by Lindbergh Aircraft Tug Company. The winch on the top of the tug by the handle attaches to the front wheel of the aircraft and pulls the aircraft onto the tug. The challenge that this vehicle poses and the reason this will not work in moving the tractors is because this vehicle is designed for only one wheel applications. This would render itself useless with respect to the tractors.

The second type of powered hand trucks are equipment movers. Figure 2 shows an electrically powered hand truck that is used for moving heavy equipment. This vehicle achieves the weight transfer differently than the aircraft tug. There are two ways that this can be done. The first option is by adding steel weights to the inside of the vehicle. This counteracts the moment that is applied by the equipment when the lifting arm raises the equipment.

Figure 2: Powered hand truck for equipment.

The second way that this hand truck achieves traction is by lifting part of the equipment up and then pushing the equipment forward. Therefore, this hand truck can be used to push and pull equipment, or do both simultaneously.

Also, there are many types of trailer movers. Trailer movers are used in essentially the same way as the aircraft tugs. The trailer movers accomplish the weight transfer to the machine by lowering the trailer onto the trailer ball on the machine. The trailer supports are then lifted up and the trailer is then free to move. The weight of the trailer is put directly over the wheels which is ideal for this application because this gives the machine a lot of traction and also helps stabilize the vehicle. This is a very stable design because the weight of the object being towed is directly over the axles of the vehicle which eliminates the use of weights to offset the moments generated when the load is applied some distance from the axles. Figure 3 shows a hand operated trailer mover by Power Caster, Inc. that uses these principles.

Figure 3: Hand operated trailer mover by Power Caster, Inc.

Hydraulic Systems and Components. A hydraulic system has many different components. There are reservoirs, fluid, fluid lines, actuators, pumps, motors, flow dividers, and pressure gauges. However, the main three design components are the motor, actuator, and pump. There are many different hydraulic motors, actuators, and pumps used in the industry today. The pump, motor, and actuator work together in series to cause useful work to take place. Perhaps the most iconic component of a hydraulic system is the hydraulic cylinder. Hydraulic cylinders are known as rams, jacks, or actuators, but they all refer to the cylinder that is doing work for the user. The hydraulic cylinder is the business end of a hydraulic system as this is where all of the useful work is taking place. Hydraulic cylinders come in many different sizes and designs based on the needed application. There are many different components to a hydraulic cylinder that allow the cylinder to work. Figure 4 shows a cutaway of a cylinder with all of the necessary components.

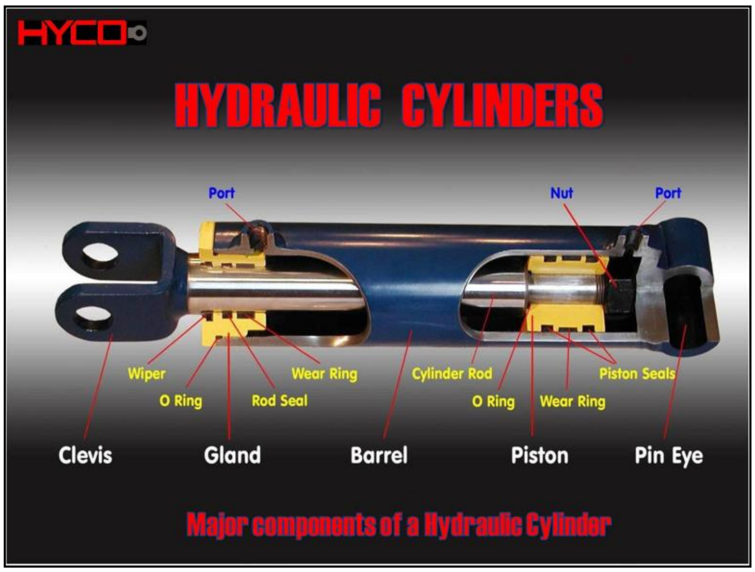

Figure 4: Cutaway of a hydraulic cylinder.

There is one major equation for hydraulic cylinders that helps designers know the required pressure of the system. Equation 1 shows the relationship between required pressure of the hydraulic cylinder and the force needed to do work (Indiana Fluid Power).

$$P = \frac{F}{A} \tag{1}$$

The pressure required is directly related to the force. This means that as the force increases on the cylinder, the required pressure also increases linearly. Conversely, the required pressure is inversely related to the area of the piston. As the bore diameter of the cylinder decreases, the effective area of the piston also decreases, meaning there must be a greater pressure in order to counteract the load on the cylinder. The units for this equation are generally in the form of psi, or pounds force per square inch.

At the heart of the hydraulic system is the pump that supplies the pressure to the actuator. Pumps also come in different sizes and designs (gear, vane, and piston), but are all guided by the same principles. There is an ideal revolutions per minute (rpm) for every hydraulic system. Usually, this is driven by a hydraulic motor's requirements, but in some cases (as in the case with hydraulic cylinders) the required flow rate of the system comes down to the desired speed of the actuator. When a hydraulic cylinder extends, there is an associated volumetric displacement that occurs.

A designer will generally want this extension to happen in a set amount of time. If the associated volumetric displacement was 50 cubic inches and the time desired for the extension of the cylinder was 5 seconds, this would yield a desired flow rate of 10 cubic inches per second. However, pumps are marketed as the cubic inch displacement per revolution of the pump shaft. Therefore, in order to change the units from cubic inches per revolution to cubic inches per second, one must know the rpm's of the electric motor driving the pump. Once the rpm's are known, this number can be multiplied by the cubic inches per revolution of the pump to give cubic inches per minute. Dividing this number by 60 will give the final desired units of cubic inches per second.

The last major component of some hydraulic systems is the motor. Hydraulic motors are essentially hydraulic pumps that do not have an electric motor linked to them. In the case of pumps, the motor will have an electric motor linked to the hydraulic motor to cause displacement of the hydraulic fluid which effectively makes the motor a pump. However, in the case of a motor, there is not electric motor mounted to the hydraulic motor. Fluid flows through the motor at a specific flow rate which causes the shaft to turn at a certain rpm relative to the flow rate of the fluid and the cubic inches per revolution of the motor.

PROCEDURES AND METHODS

Design Procedure

A final design of the tow vehicle was completed after many hours of talking with Virgil Threlkle, Dr. Mark Zohns, Dr. Andrew Holtz, and various knowledgeable students in the BRAE department from the tractor pull team and quarter scale design team. All of the drawings were made using the 3D modeling software SolidWorks.

Design Specifications

The project had a few different constraints that it had to meet. First, the vehicle needed to be able to fit on the beaver tail of the tractor pull team's trailer. The dimensions of the trailer's beaver tail are 8 feet wide by 4 feet deep. The tow vehicle needed to fit on the beaver tail of the trailers because the vehicle would be used at events across the state that the team goes to. Another constraint that had to be met was that the tow vehicle needed to be able to go across concrete as well as be able to do moderate off-road activities at various tractor pulls throughout the season. Because of this, the vehicle needed to have adequate clearance from the ground and have the necessary traction to still move the vehicle while in loose soil conditions.

Another constraint that the tow vehicle had to overcome was being able to lift the necessary 2000 pounds of the front of the tractors. This was a necessary design specification because of the requirement that the vehicle be able to turn in a very tight radius. This leads into the last design specification which was that the tow vehicle is able to make very tight turns while in the BRAE shops. This was to ensure that the vehicle would completely replace the need for the forklifts. This would also ensure that the vehicle would be able to actually maneuver the shops while the tractor was being towed.

Initial Designs

There were a few different initial designs that were evaluated. The first design that was considered was modeled after the airplane tug mentioned in the literature review. The vehicle was going to work in much of the same way as the tug where the tractor was going to be pulled up onto the tow vehicle via a wench system. The front wheels were going to be towed up onto a platform that would hold both wheels. The wheels would then be locked in attaching the tow vehicle to the tractor. This would have allowed for the needed traction of the tow vehicle, but would not allow for the necessary clearance of the vehicle. In order for the platform idea to work, the vehicle had to have a very low profile to allow the tractors wheels to roll up onto it. To remedy this issue, the platform was then reevaluated and it was decided that the platform would have to raise up into the air to allow the tow vehicle to not have such a low profile. The problem with this was that the platform would have to lock into place when the tractor wheels were being pulled up onto the vehicle. Then, the platform would have had to be able to swivel once the vehicle was in the air to allow for the necessary tight turns. This was immediately discarded as this was an obvious endangerment to the operator and bystanders.

The next design consideration included the use of a scissor jack that would still lift the vehicle in the air and would attach to the hitch on the front of the tractor. After much consideration, this design was discarded as well because the scissor jack would be inadequate for continuous use and would also have to be extremely overdesigned in order to handle the large loads of the tractor.

Final Design

Final Design Overview. The final design took into consideration the trailer movers and the aircraft tugs mentioned in the literature review and took the best components of each. The final design included the use of hydraulics and the use of the front hitch receiver on the tractors. The final design can be seen below in Figure 5.

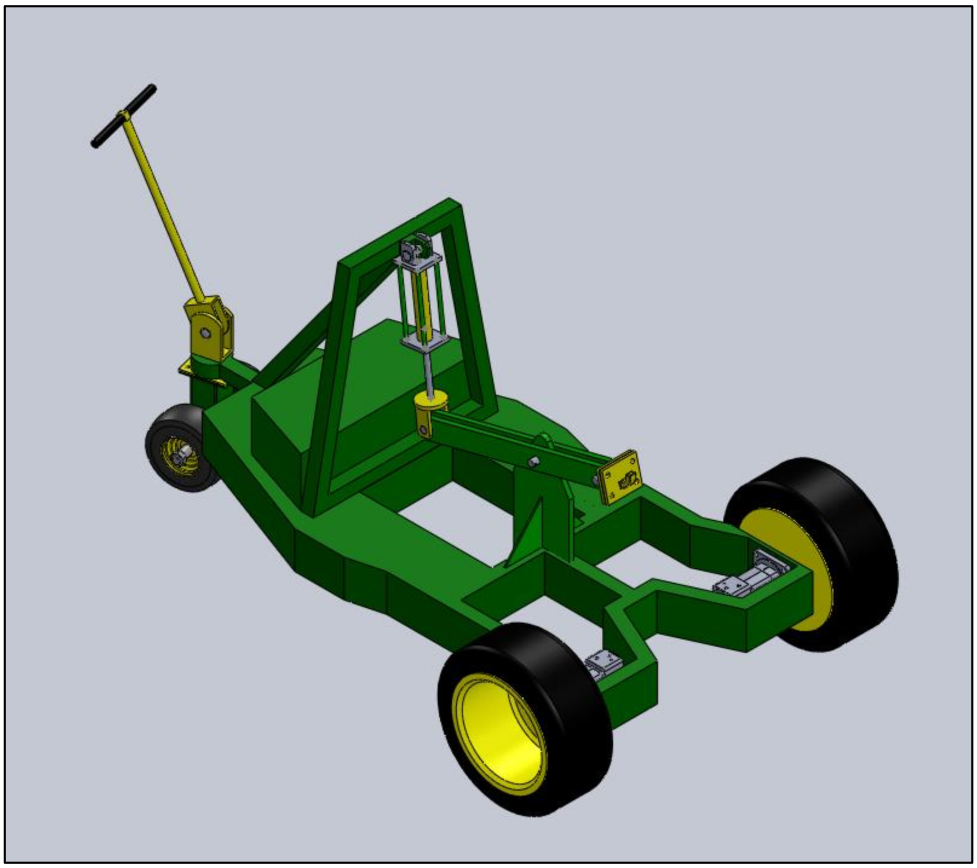

Figure 5: Isometric view of the final design.

<u>Frame.</u> The frame was designed in order to handle the stresses of the tractor load, have enough space for the hydraulic reservoir, and be large enough hold the hydraulic linkage system and the hydraulic wheel drive system. Figure 6 shows the final frame design with the sheet metal installed. Figure 7 shows the frame without the sheet metal installed.

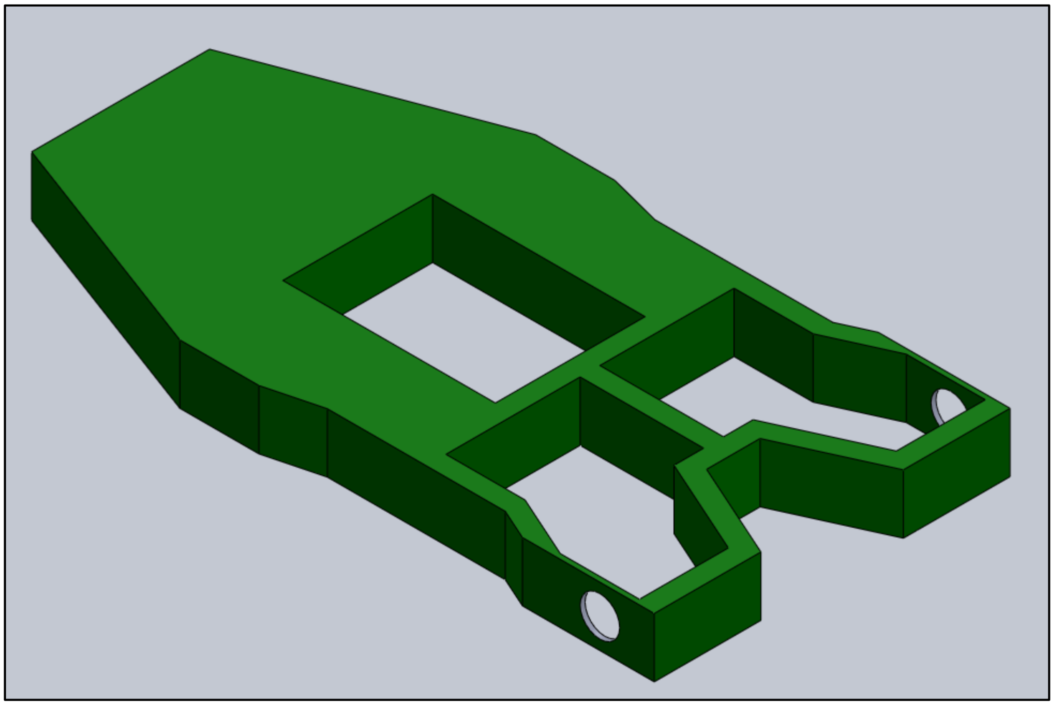

Figure 6: Final frame design with sheet metal installed.

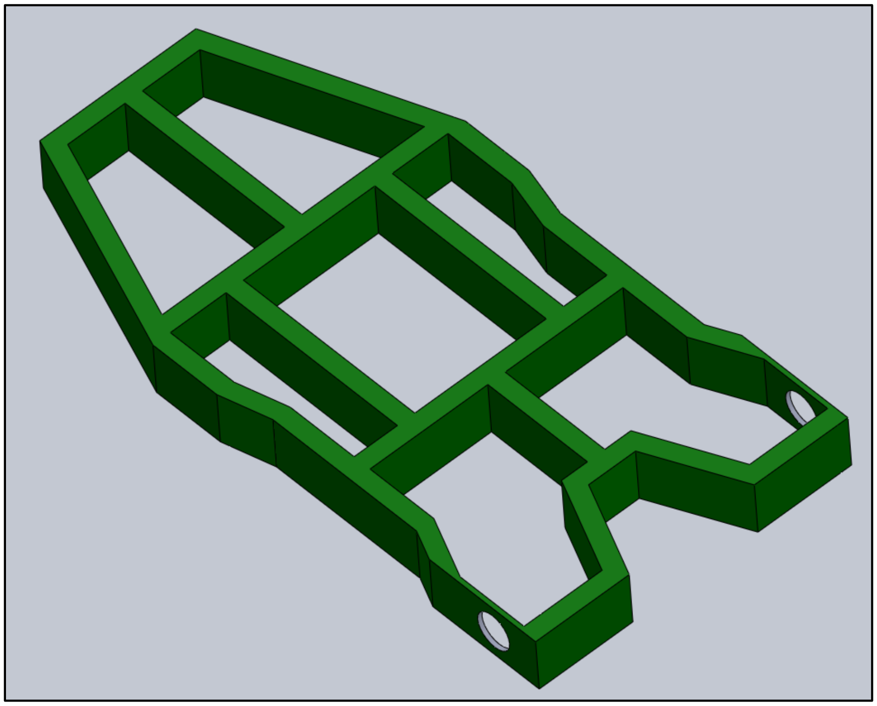

Figure 7- Frame without sheet metal.

The frame was designed with 6" x 2" x ¼" rectangular tubing. There were a couple of reasons for this. The first reason was, as stated before, the frame needed to withstand the large loads of the tractor being applied. The yield strength of the steel tubing is approximately 46,000 pounds per square inch (psi). Using values from the American Institute of Steel Construction (AISC) manual, and using the maximum bending stress equation of:

$$\sigma = \frac{Mc}{I} \tag{2}$$

The maximum bending stress was 1,431 psi (Appendix G). When designing for structural steel applications, the maximum bending stress is designed at 60% of the yield strength, or 0.6(S_y) which is equal to 27,600 psi. The maximum bending stress is well below this. There is ½ inch sheet metal where the hydraulic drive motors are mounted.

The large flat area in the front of the frame was designed to allow for the needed space of the reservoir. The area in the middle of the frame was left open to allow for adequate movement of the hydraulic linkage system. The two flat areas to the left and right of this opening was designed to allow adequate space for any hardware, pumps, and batteries. The two open areas at the end of the frame were left open to reduce weight and cost. Calculations and design can be seen in Appendix G.

Hydraulic Lifting System. The lifting mechanism of the towing vehicle had a couple of requirements that needed to be met. The system needed to be able to handle a 2,000 pound load vertically and the total weight of the tractor during the towing process. The original idea for this, as mentioned before, was to lift the vehicle directly underneath in much of the same way that scissor jacks do.

However, this would have required a very low profile for the vehicle because of the low clearance of the front of each tractor. The final design does not need to lift the vehicle from directly underneath because it relies on the before mentioned hitch (Appendix F). The new system, as seen in Figure 8, used a linkage system.

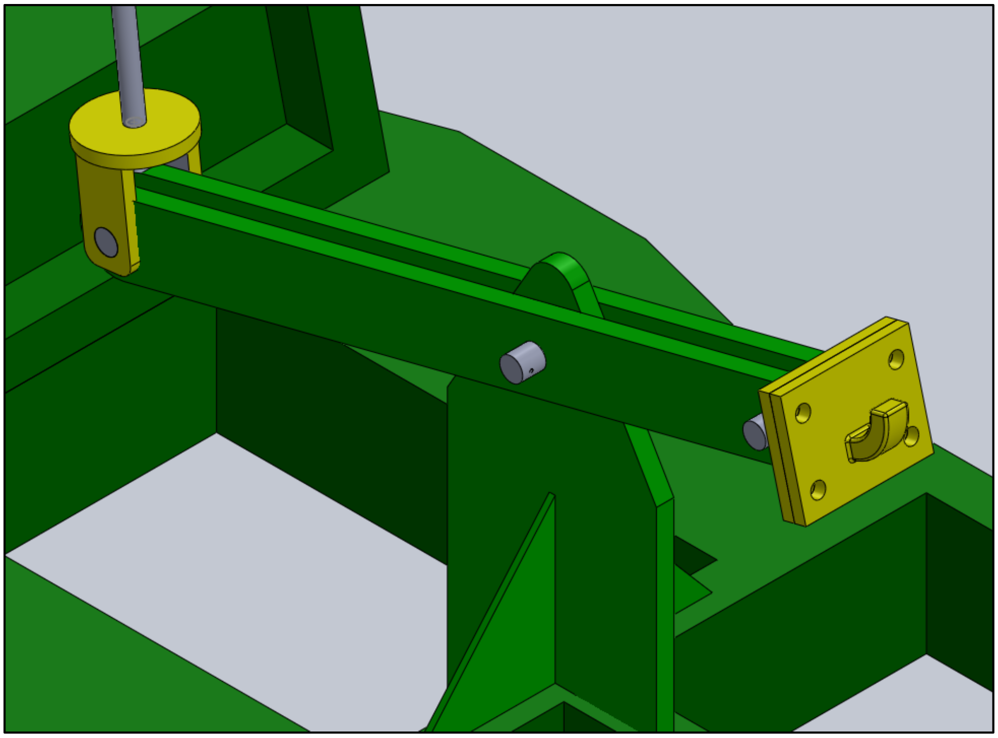

Figure 8: Hydraulic linkage lifting system.

The lifting system design calculations can be seen in Appendix B.

Hydraulic Drive System. The driving wheels incorporates two hydraulically driven wheel motors. Each motor was designed using the White Hydraulics configurator. The website can be referenced in the references section. The motor size for both wheels was determined to be 30.3 cubic inches per revolution. The speed of both motors was designed to be 1 ft/sec at 15 rpm with a required pressure drop of 1,750 psi. The flow rate was determined to be 1.9 gallons per minute. More information on the design aspects of the motors can be seen in Appendix C. Below in Figure 9 is an example of how they would be mounted and used.

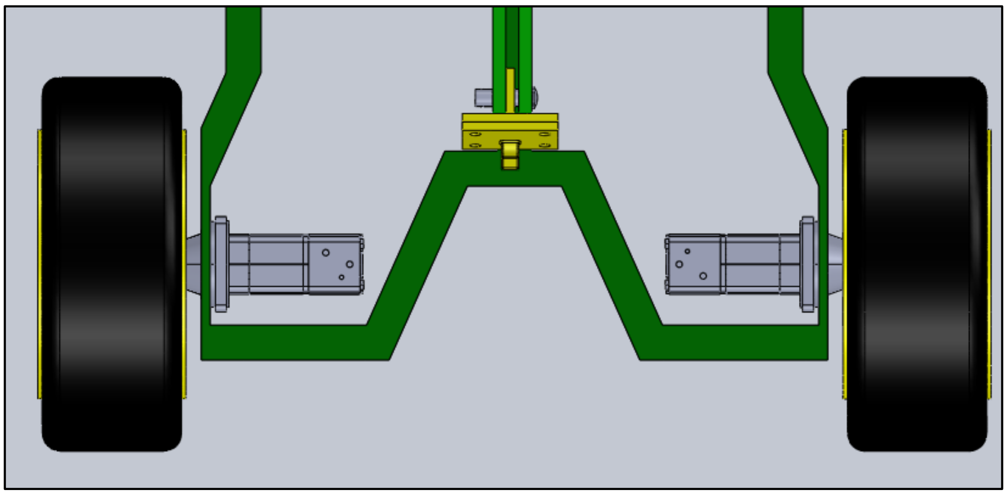

Figure 9: Hydraulic drive system.

Pumps and Batteries. There was two different applications that needed a hydraulic pump. The first application was for the two hydraulic wheel drive motors. Both motors required 1.9 gpm flow rate with a combined flow rate of 3.9 gpm. Using the information from Appendix E, the hydraulic pump required for this was found to be 0.5 cubic inches per revolution at 1750 rpm. The second application that needed a hydraulic pump was the lifting cylinder which would run on the same pump as the two wheel drive motors. Both systems were designed to operate off of the same pump. The hydraulic schematic can be found in Appendix H.

The vehicle needed to have a mobile power source since the vehicle was going to be operated in very remote environments. Because of this, two 12v batteries were chosen to be the source of the power needed over an internal combustion engine. To power the pump for the hydraulic wheel drives, the electric motor needed for the pump was found to be 4 horsepower (Seen in Appendix E. Use 5hp as exactly 4hp is not available). The 5 horsepower Baldor electric motor needs 230 volts AC at maximum load. Because of this, two 12 volt batteries were selected with one DC to AC power inverter to acquire the needed 230 volts AC. The inverter needed to be able to handle 3,220 Watts (230v multiplied by the 14 amps needed by the electric motor). Therefore, the inverter selected was the AIMS Power 6000W (lower watt models did not output 230v AC), 24v DC to 230v AC power inverter. The 230v AC allows the electric battery to be wired in three phase, which was needed by the electric motor. The batteries that were selected were two, 12v, 70 amp-hour deep cycle batteries. The deep cycle batteries were selected as they are a better battery for this application that needed less amps over a longer period of time. Also, because the Baldor 5hp electric motor has a demand of 14 amps, the batteries would be expected to last approximately 5 hours before needing to be recharged (70 amp-hours divided by 14 amps). The use of the two batteries and power inverter is completely feasible as the power inverters are designed to be mobile. The dimensions of the inverter (23.5” x 8.59” x 7.05”) allows this inverter to fit onto any one of the open mounting areas on the frame.

RESULTS

The final design implemented a few different aspects of already designed hand powered towing vehicles. The entire design was built around the idea that the vehicle was to be hydraulic in nature. Therefore, the lifting system and the hydraulic drive system were both designed using hydraulic components. This was especially important as there was to be no electrical sources out in the field during competitions.

The hydraulic drive system implements two White Drive Products WS355 wheel drive motors. Both motors are 30.3 cubic inches per revolution and both motors run at 1.9 gpm. Therefore, the hydraulic system was designed to meet the 3.9 gpm peak demand. To meet this demand, a 0.5 cubic inches per revolution pump was selected to operate at 1750 rpm with a 5 horsepower three phase electric motor. This allowed the vehicle to operate in the remote locations of pit areas without having to have direct access to an electrical power source.

The hydraulic lifting system was designed to lift the 2000 pound load of the front end of the tractors and also be able to then clamp down onto the hitch/tow strap via a drawbar coupler and coupler eye. The hydraulic lifting system was also designed to allow the tractors to rotate freely to reduce the amount of torque on the lifting system. This was important as minimal bracing reduced the weight and cost of the entire vehicle.

The frame was designed with 6” x 2” x ¼” rectangular tubing as the frame had to carry significant loads. More details about calculations for all of the hydraulic components, frame, and stresses can be found in the following appendices.

Cost Estimate.

Table 1. Cost estimation of the towing vehicle.

Material	Total length/units of material needed	Units	Cost per unit	Total Cost
6"x2"x1/4" Rectangular Steel Tubing (Metals Depot)	30.2	FT	13.49	$ 406.96
2"x2" Solid Steel Square Bar (Metals Depot)	30.0	Inches	2.13	$ 63.90
3/4" Steel Plate (McMaster-Carr)	3.0	SQ. FT	0.82	$ 2.46
1/2" Steel Plate (McMaster-Carr)	1.0	SQ. FT	1	$ 1.00
1/4" Steel Plate (McMaster-Carr)	50	SQ. FT	2.09	$ 104.50
White Drive Products Hydraulic Drive Motors (average value found online)	2	EA	350	$ 700.00
1-14" Tapered Shaft to 5-Hole Wheel Hub Adaptor (Surplus Center)	2	EA	43.95	$ 87.90
2"x2"x11 GA. Square Tubing (McMaster-Carr)	12	FT	2.87	$ 34.44
10" 5-Hole Wheels (Gempler's)	2	EA	43.05	$ 86.10
10" Tires (Nebraska Tire)	2	EA	145	$ 290.00
Drawbar Coupler (McMaster-Carr)	1	EA	215.54	$ 215.54
1 Inch Dia. Clevis Pin (McMaster-Carr)	8	EA	2.25	$ 18.00
Pneumatic Caster (McMaster-Carr)	1	EA	400	$ 400.00

1-1/2" Bore Hydraulic Cylinder (McMaster-Carr)	1	EA	415.48	$ 415.48
Hydraulic Oil (McMaster-Carr)	11	Gallon	10.36	$ 113.96
Thrust Bearing for Steering Mechanism (McMaster-Carr)	1	EA	3	$ 3.00
1"x0.12" Steel Tubing for Steering (McMaster-Carr)	6	FT	3.38	$ 20.28
1/2" Diameter Hydraulic Hose (McMaster-Carr)	35	FT	7.25	$ 253.75
5 hp Baldor Electric Motor (Baldor)	1	EA	1237	$ 1,237.00
Hydraulic Pump (McMaster-Carr)	1	EA	195.48	$ 195.48
12v,70 Amp-h Deep Cycle Battery (McMaster-Carr)	2	EA	123.57	$ 247.14
6000W, 24v DC to 230v AC Inverter by AIMS Power	1	EA	1248	$ 1,248.00
Labor (Assuming benefits)	100	Hours	80	$ 8,000.00
			Material Cost:	$ 6,144.89
			Labor Cost:	$ 8,000.00
			Total:	**$ 14,144.89**

DISCUSSION

The first couple iterations of the project involved different existing ideas from other functioning towing and lifting equipment. The first iteration involved the use of a power screw in much of the same way a scissor jack works for lifting one side a car off of the ground when changing a tire. The electric motor, still powered by two 12v DC batteries with an inverter, would be directly mounted to the power screw of the scissor jack. The motor would have to be geared down significantly to reduce the revolutions per minute and increase the torque. The load of the tractor would be placed on the lifting platform of the scissor jack. In order for this iteration to have worked, a few different safety features would have had to been implemented. For instance, there needed to be a way to lock the lifting platform to the frame to make the system safe. The other problem with this iteration is that there was possible frame damage that could occur. Because the cons were too great, this iteration lead to the second iteration.

The second iteration attempted to take away the concept that the lifting mechanism needed to connect to the frame and instead looked for a way to lift the tractors with the existing components. To completely reverse the first idea, the second iteration was to not connect to the tractors at all, but rather, it was to lift the tractors from under the tires as the aircraft tugs do. However, the aircraft tugs that are currently being used in industry do not lift the aircrafts into the air. This was not adequate as there was a definite need to have the tractors lifted into the air to provide sufficient ground clearance when operating in adverse soil conditions and to allow for a very tight turning radius. The safety problem with this iteration is that the tractors would have the tendency to twist off of the lifting platform because there was no apparent way to lock them down or to have the platform rotate freely. Also, the other design flaw with this iteration was that the towing vehicle needed to have a very low profile when pulling the tractors up onto the platform. After the tractors were on the platform, the towing vehicle needed to be raised up into the air to achieve the ground clearance needed. Again, there were too many cons that led to the last and final iteration of this report.

The final iteration is by far the safest and most efficient towing vehicle that was considered, which was the objective of this project. The towing vehicle was designed with the correct ground clearance already achieved. The other design consideration of needing to attach the towing vehicle to the tractors was achieved by the use of the drawbar coupler with the locking top clamp and hitch. This helped to ensure that the tractors would not bounce off of the towing vehicle while driving over rough terrain. The drawbar coupler also helped make the final iteration a much safer process. The use of pneumatic wheels and a pneumatic caster in the front of the vehicle allowed for operation in rough soil conditions while still having enough traction on concrete and asphalt. Finally, the use of hydraulics was ideal for this design as this allows for the large loads to be lifted by relatively small components which helped reduce weight and size of the vehicle. The only disadvantage to this was that the hydraulic system needs significant space for the hydraulic oil. However, this became another design feature as the large amounts of hydraulic oil in the front of the vehicle helped prevent the vehicle from flipping backwards from the large moment generated while lifting and towing the tractors. Also, the use of hydraulic drive wheels was ideal because the motors are not free turning, which helped to prevent the towing vehicle from continuing to drive once the operator stops the flow of hydraulic oil. Overall, the final design was the safest and most efficient design.

RECOMMENDATIONS

If this project is to be built and utilized in the future, it will be very important to analyze the type of welds that will be used. Gas Tungsten Arc Welding (GTAW) would be the preferred method if this project were to be built. Though more time consuming and tedious, GTAW can lead to higher quality welds and a much stronger weld throughout. Also, it would be recommended to perform as little welding as possible because as soon as one welds a piece of metal to another, their strength characteristics can change drastically. The more one welds the same piece of metal, the integrity of that piece of metal can be significantly reduced. Because of this, it would be advised to investigate using hardware where at all possible.

Although the final hitch design is more than sufficient for this application, it would be recommended to revisit the design and to look at other ways of achieving the necessary height while reducing deformation in the hitch.

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APPENDIX A: BRAE Requirements

How Project Meets Requirements for the BRAE Major This project meets the requirements for the BRAE major by encompassing aspects from each fundamental course of engineering. Some of those fundamentals include: physics, calculus, strength of materials, dynamics, statics, hydraulics, AutoCAD, Solid Works, and electrical engineering and steel construction and design.

Major Design Experience - The project must incorporate a major design experience. Design is the process of devising a system, component, or process to meet specific needs. The design process typically includes the following fundamental elements. Explain how this project will address these issues. (Insert N/A for any item not applicable to this project.)

Establishment of objectives and criteria	The objectives and criteria for this project were to design and analyze a towing vehicle for the BRAE tractor pull team. This vehicle was to be safer, effective, and efficient at towing the modified pulling tractors for the tractor pull team.
Synthesis and analysis	Synthesis and analysis of the design was accomplished via Solidworks, a 3D modeling software, and Microsoft Excel.
Construction, testing and evaluation	There is a heavy emphasis on construction, testing, and evaluation with this project as the success and implementation of the project relies heavily on all three. Extensive testing was done through Solidworks and various engineering calculations.
Incorporation of applicable engineering standards	Steel standards regarding design and construction were taken from the Steel Construction Manual published by the American Institute of Steel Construction.

Capstone Design Experience - The engineering design project must be based on the knowledge and skills acquired in earlier coursework (Major, Support and/or GE courses).

Incorporates knowledge/skills from earlier coursework	Strength of materials, physics, calculus, hydraulics, power transmission, electrical engineering, dynamics, steel design and construction, economics, and statics are the main courses that were the basis for all design and calculations. This project would not be possible without knowledge from each course.

Design Parameters and Constraints - The project should address a significant number of the categories of constraints listed below. (Insert N/A for any area not applicable to this project.)

Physical	The physical constraint was the profile of the vehicle as the vehicle needed to have a certain amount of ground clearance in order to operate in adverse soil conditions.
Economic	The project was to be designed with as little material as possible to reduce the cost.
Environmental	The design of this vehicle does not incorporate any hazardous waste as defined by Subtitle C and D of the EPA regulations and standards.

Sustainability	The design of this vehicle was designed with as minimal material as possible
Manufacturability	Shop time and use will be a big factor in getting the design into a working version. Shop availability will be the biggest constraint.

Health and Safety	The project took into consideration the safety of the operator and bystanders. As mentioned in the report, the final design was selected for the vast amounts of safety built into the design itself.
Ethical	N/A
Social	N/A
Political	N/A
Aesthetic	The finished design includes use of powder coating all finished surfaces to make the vehicle pleasing to the eye.
Other	N/A

APPENDIX B: Hydraulic Lifting System Design and Calculations

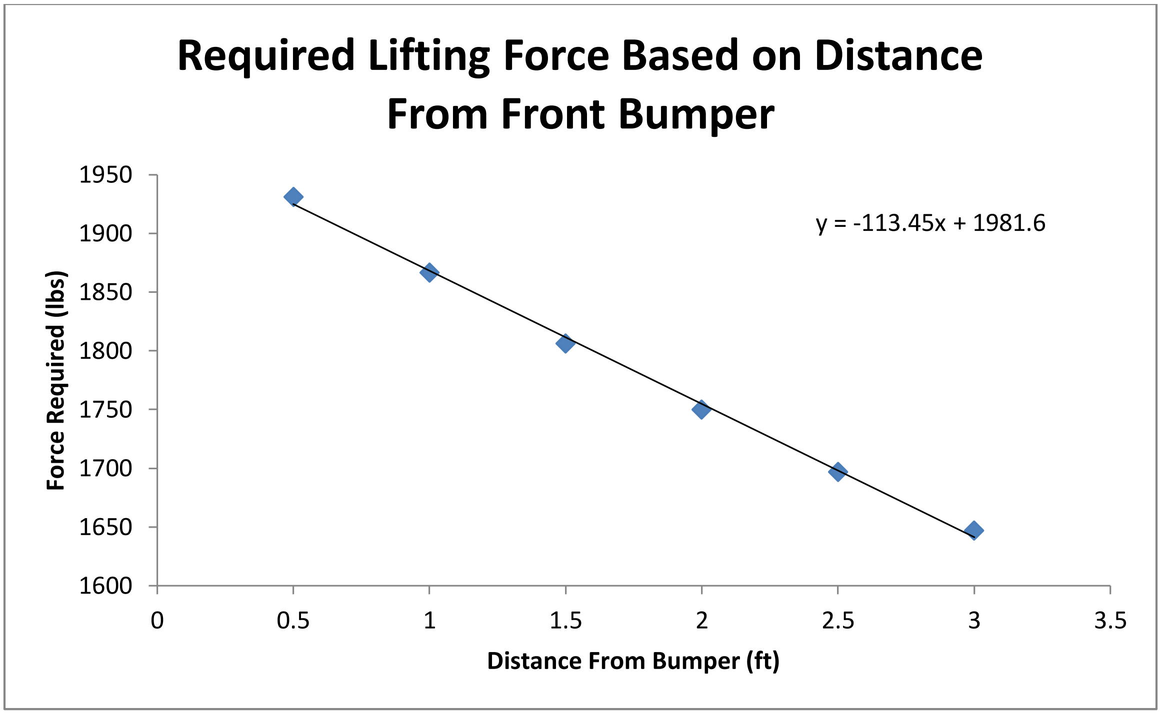
The first step in designing the hydraulic lifting system was to first determine how far away from the hitch receiver the lifting end of the linkage system was going to be. In doing so, the required lifting force could be determined.

Tractor W:	2000	lbs
Wheel base:	14	ft

Dist from front bumper (ft)	Required Lifting Force (lbs)
0.5	1931
1	1867
1.5	1806
2	1750
2.5	1697
3	1647

The required lifting force was calculated using the sum of moments about the back wheels of the tractor. The tractor wheel base was 14 feet as seen above. The weight of the front of the tractor is approximately 2000 pounds.

As one can see, the farther away the lifting end of the hydraulic system is from the front bumper of the tractor, the lower the required lifting force becomes. The reason for this is because the mechanical advantage from the back wheels (fulcrum) becomes greater as the distance increases. The final design of the hitch was designed using the 2 feet length as boxed above. The relationship between the distance from the front bumper and the lifting end of the hydraulic system can be seen graphically below:



Now that the required lifting force is known, the bore diameter of the lifting cylinder must be determined. To do this, a selection table is made that shows the bore size and required pressure based on the area of the cylinder and the lifting force required:

Bore Diameter (in)	Required PSI	Stroke Length (inches)	Volume to full stroke (cu. In)
1.5	**990**	**8**	**56.5**
2	557	8	100.5
2.5	357	8	157.1
3	248	8	226.2
3.5	182	8	307.9
4	139	8	402.1

The 1.5 inch diameter bore was selected due to the required pressure. The design pressure of the system was 1000 psi. Therefore, the 1.5 diameter bore cylinder meets this requirement while keeping cost to a minimum because of the smaller size of the cylinder.

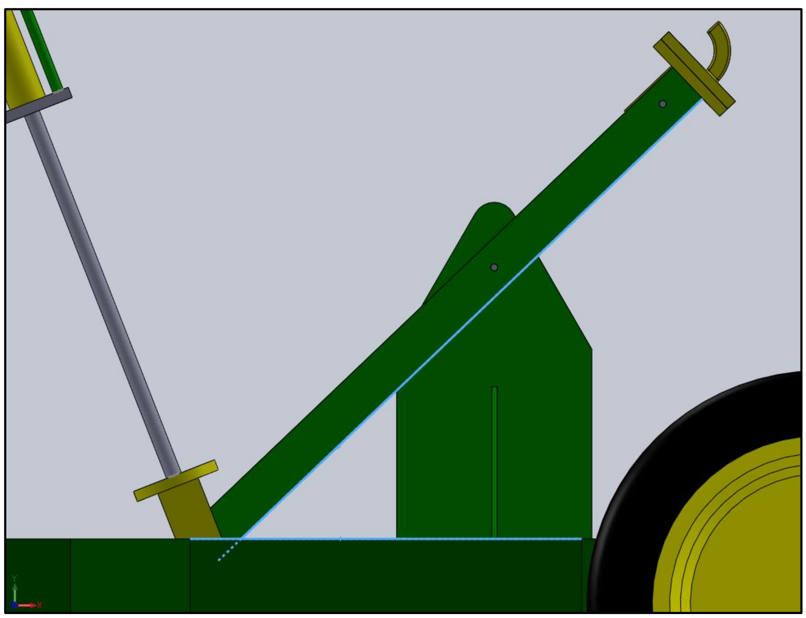
Also, the linkage design was calculated in the following way:

<u>Height AT 45 degrees:</u>

Degrees:	45	Degrees at maximum height
Radians:	0.79	radians
y-distance:	8.0	Inches at max height

22.6 inches	<-----	This value is if the rod moment is 2/3rds the way down the linkage
33.9 inches	<-----	This is the total linkage length with the moment 2/3rds of the way down the linkage

The new total length of the linkage arm is 31.9 inches		
This means that the short length is 1/3 of 31.9 =	10.6	inches
And the long length is =	21.2	inches



Pivot arm at full design height, or 45 degrees.

The total length to date is 29.94 inches. The long length is 18.54 inches and the short length is 11.4 inches. that means that if the 2000 pounds is pushing down on the short side, the total needed force on the long side is:

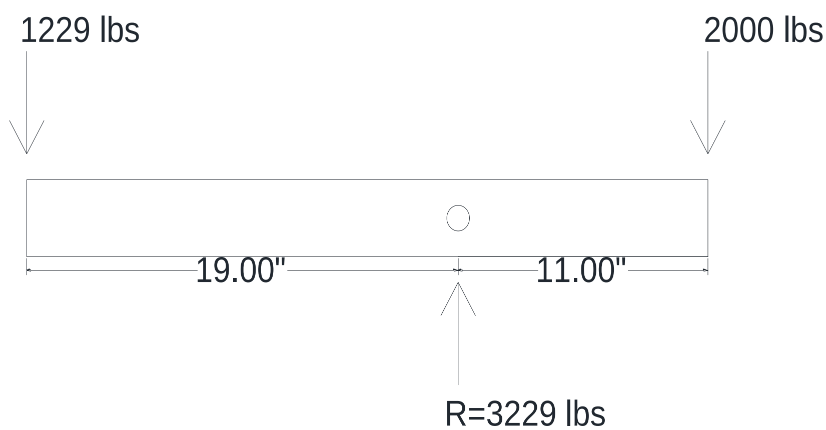
∑Fy=0		
length of small side:	11.4	inches

length of long side:	18.54	inches
force on small side:	2000	pounds
force req'd on long:	**1229.773**	pounds required by the hydraulic cylinder at the end of the longer side of the linkage.

The required force on the hydraulic cylinder side to lift the tractors load of 2000 pounds was found to be 1229 pounds of force. This is smaller than the design lifting force of 1750 pounds, so the design is adequate.

The size of the lifting arms was determined by setting a thickness of 0.75 inches and then solving for the height by using the bending stress equation.

The height of the lifting arms needed to be determined. Assuming a width of 0.75 inches, the sizing and design of the lifting arms was as follows:



To find the height needed for the pivot arm, the bending moment equation was utilized. A value of 46,000psi was used for the yield strength:

$$\sigma = \frac{Mc}{I}$$

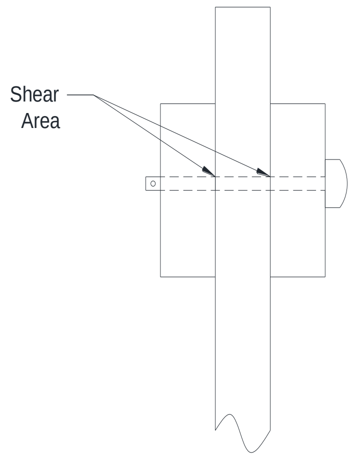
$$\sigma = \frac{(2{,}000lbs)(11in)(0.5h)}{(2\ bars)(\frac{1}{12})(0.75in)(h^3)} \leq 0.6S_y$$

$$\frac{(22{,}000lb - in)(0.5)}{3{,}450psi} \leq h^2$$

$$1.79\ \leq h$$

Use 2 inches.

To size the pins needed to hold the pivot arm:



End View of Pivot Arm
Assembly

$$\tau = \frac{P}{A} \leq 0.577(S_y)$$

$$\frac{3{,}229lbs}{(2\ shear\ areas)(\frac{\pi D^2}{4})} \leq 0.577(46{,}000psi)$$

$$D \geq \sqrt{\frac{12{,}916lbs}{(2)(\pi)(26{,}542psi)}}$$

$$D \geq 0.28\ inches$$

Use 5/16 inch diameter.

APPENDIX C: Hydraulic Wheel Motor Design and Calculations

From Dr. Zohns:

-Assume 800 lbs rolling resistance (from knowing that it takes 4 guys to push the tractor around in typical soil conditions. Assume each guy can push roughly 150-200lbs force.)

-Assume 18” wheel diameter (9” radius)

Therefore:

The required torque is equal to the rolling resistance multiplied by the wheel radius:

$$(800\ lb) \times (9\ in.) = 7{,}200\ lb - in$$

Go to the White Drive Hydraulics website:

-Choose the WS355 motor and the 30.3 cubic inches per revolution with ΔP required of 1,750 psi to give 7,205 lb-in of torque at 8 gpm. This also gives a speed of 50 revolutions per minute. However, in this application, the flow rate was determined to be 3.9 gpm (Appendix E). Therefore, the motor speeds are approximately:

$$\frac{3.9\ gallons}{minute} \times \frac{231\ in^3}{gallon} \times \frac{rev.}{30.3\ in^3} = 29.7\ rpm\ (when\ in\ series)$$

However, when the motors are in parallel, the flow is divided amongst the two motors, so the speed is reduced by 2, to approximately 14.9 rpm.

The motor that was selected was the WS355 series wheel drive motor with 30.3 cubic inches per revolution.

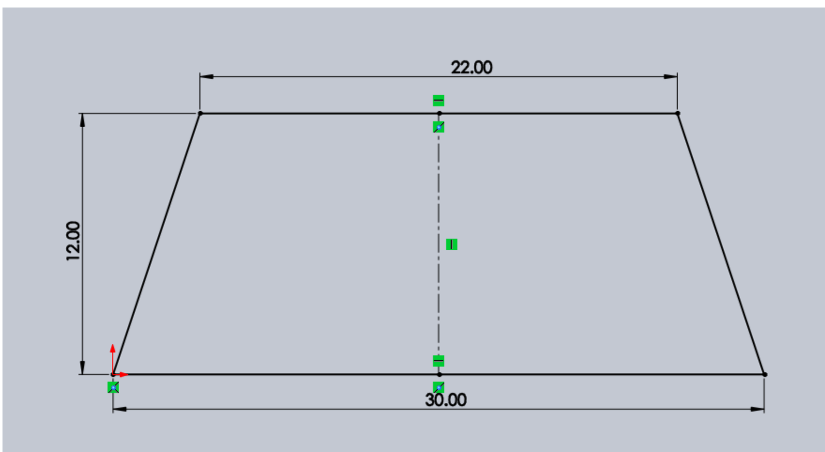
APPENDIX D: Reservoir Sizing Calculations

To size the reservoir, the generally accepted rule is that the reservoir is to be three times larger than the required gallons per minute of the system. The system had a requirement of 3.9 gpm because of the two wheels which require 1.95 gpm each. Therefore, the accepted size of the reservoir needed to be 12 gallons. However, because of space requirements, that could not be achieved. The following table describes the design process and selection of the final size:

Designing the reservoir:
The area of the flat surface on which the reservoir would reside is composed of two of the above triangles and a rectangle that measures 22" by 12".
Therefore, the area of the shape above is:

Area of two triangles =	48	square inches
Area of rectangle =	264	square inches
Total area =	312	square inches
Needed height of reservoir=	8	inches
Total volume =	2496	cubic inches
There are 231 cubic inches in a gallon		
Therefore there are:	10.8	gallons

Because of limitations in the available space of the vehicle, we are going to go with 2 times the size of the rated output of the pump instead of the traditional 3 times. Therefore, the final size of the reservoir will be 10.8 gallons.



APPENDIX E: Pump Design and Calculations

The horsepower requirements of the pump were calculated using the equation:

$$HP = \frac{PSI\ x\ GPM}{1714}$$

Using this equation, and adding 20 psi for losses to the required pressure drop across the motors of 1,750 psi, we can then solve for the hp:

$$HP = \frac{(1750psi + 20psi)(3.9GPM)}{1714}$$

$$HP = 4.03hp$$

Using the desired speed of 1 ft/sec for the drive wheels, the flow rate of the pump must be determined. We must first determine the wheel rpms:

$$\frac{1\ ft}{sec} \times \frac{12\ in}{ft} \times \frac{1\ rev\ of\ tire}{\pi(18\ in)} \times \frac{60\ seconds}{minute} = 12.7\ rpm\ \ (13\ rpm)$$

Now that the wheel rpm's have been determined, the required pump flow rate can be calculated:

$$\frac{30.3\ in^3}{rev\ of\ motor} \times \frac{12.7\ rev}{min} = 385\ cubic\ inches\ per\ minute$$

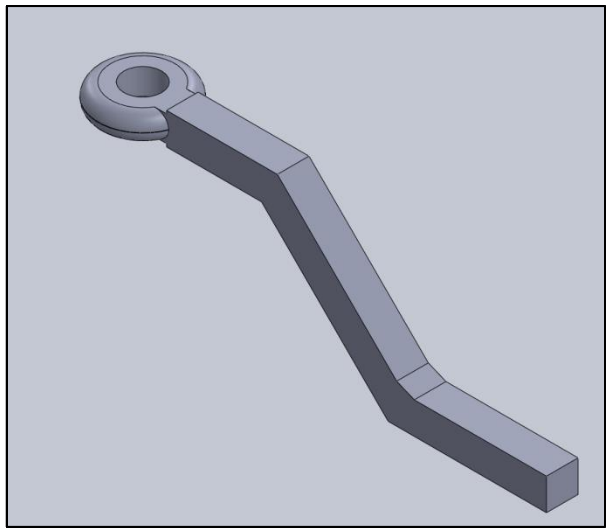
Because there are two motors with the same flow rate, the flow rate must be multiplied by 2. Therefore, the pump must have a flow rate of 770 cubic inches per minute. To determine the size pump to be used, the desired flow rate is divided by the electric motor rotation speed to yield the correct size of the pump:

$$\frac{770\ \frac{in^3}{min}}{1750\ \frac{rev}{min}} = 0.44\ \frac{in^3}{rev} \sim 0.5\frac{in^3}{rev}$$

When the size of the pump is increased slightly to the 0.5 in^3/rev, the flow rate is increased slightly to 900 in^3/min (3.89 gpm). This will yield a total wheel rotational rate of approximately 15 rpm (Appendix C).

APPENDIX F: Hitch Design

A new hitch had to be designed in order to allow the vehicle to have a higher ground clearance while also being able to carry the large loads. The figure below shows the final design that was decided on for the hitch. The hitch was designed to be made out of 2 inch by 2 inch 1018 carbon steel square bar.



Final new hitch design.

The lower right-hand portion of the above figure is the end of the hitch that would be inserted into the hitch receiver on the tractors. The upward bend in the hitch was specifically designed that way to allow for the hitch height to be increased to whichever height is necessary. This design allowed for an increased height of 8 inches from the original height of 11 ¼ inches on Mustang Fever and 9 15/16 inches on Poly Thunder from the ground to the existing hitch receiver. This enabled the tow vehicle to have the necessary ground clearance for off-road conditions.

Also, the hitch featured a weld on drawbar eye that would connect to the drawbar coupler that can be seen in Figure 5 as well as in Figure 8. The drawbar eye served a couple of different purposes. The first, and most obvious, is that drawbar eyes are rated to excessive loads, in this case being upwards of 20,000 pounds force. Second, the eye's design allows the two vehicles to roll from left-to-right independent of each other. This was an important design feature of the hitch because as the vehicle goes into off-road conditions, the vehicle will pitch in different directions relative to the tractor. This was also important because this would take a lot of torque away from the linkage system.

The hitch was designed with the previous information listed below:

Tractor W:	2000	lbs
Wheel base:	14	ft

Dist from front bumper (ft)	Required Lifting Force (lbs)
0.5	1931
1	1867
1.5	1806
2	1750
2.5	1697
3	1647

Because there was a need to be able to turn the vehicle tightly when operating in the BRAE building or in the pit areas, 2 feet was selected. The other reason that 2 feet was selected for the length of the hitch was to minimize deflection. As the hitch length would increase, the deflection would also increase. Also, there was a requirement for the height of the hitch to be increased from the original height of 11 ¼" on Mustang Fever and 9 15/16" on Poly Thunder. This is why the hitch has a "Z" like appearance.

To keep deflection at a minimum, the hitch was designed with 2" x 2" solid square steel bar. This was also the only way that the stresses would not exceed the yield strength of the steel.

The hitch needed to be as rigid as possible in order to withstand the load of the tractors along with a low displacement. The maximum stress was found to be 36,000 psi. This can be shown by the following bending stress calculation:

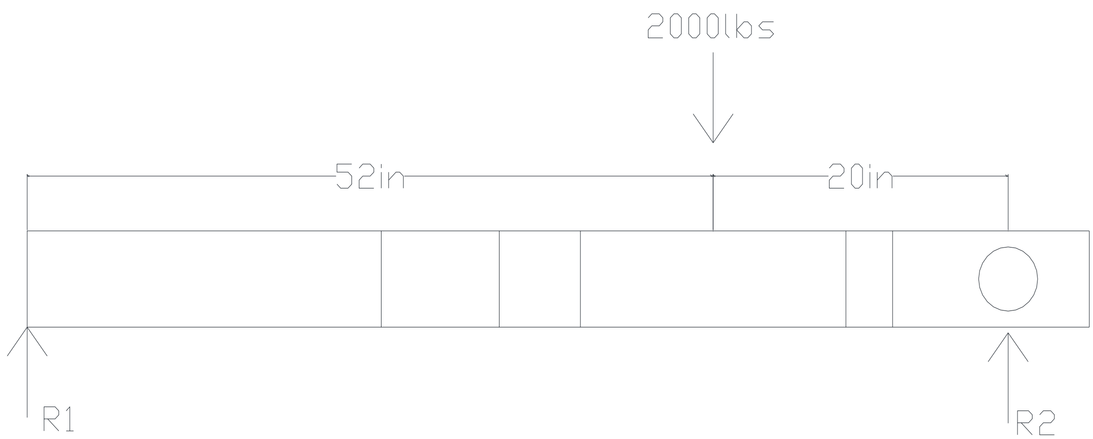
$$\sigma = \frac{(2000lb)(24in)(1\text{in})}{\frac{(2in)(2in)^3}{12}}$$

$$\sigma = 36{,}000\ psi$$

However, it was ultimately determined that this hitch design would not work with the current setup on the tractors. Currently, the front receiver tube is mounted only to the front cross member. In order for the new hitch design to work, the distance from the front of the tractor was designed to be 24 inches. The receiver tube is significantly shorter. Because of this, whatever load is placed on the 24 inch moment arm is magnified and would place the front member in significant torsion. The member would have to resist a moment of 4000 ft-lbs (2000lbs * 2ft). Therefore, the front hitch receiver on the tractors would need to be reinforced enough to resist the moment generated of 4000 ft-lbs. This reinforcing would allow the new hitch design to be utilized. There would need to be a supporting cross member at the rear and front of the hitch receiver. The support at the end of the receiver helps resist the moment, which alleviates much of the torsion placed on the front cross member.

APPENDIX G: Frame Design

The frame needed to hold the 2000 lb load from the tractor without bending significantly. The frame also needed sufficient space for all of the hydraulic components (i.e. pump, motors, reservoir, valves, etc.). The frame also needed to have a footprint smaller than 8' by 4' in order to fit on the back of the trailers that are currently used to get to tractor pulls. The size of the frame was chosen in the dimensions show in Appendix I for a couple reasons. The first reason is that in order to have enough space/room for all of the required hardware and hydraulics, the vehicle needed to have a large footprint for the hardware, hydraulics, etc. The dimensions were close to the 8' by 4' limit, but did not use the entire area. The next reason was to allow for a large enough moment from the reservoir and any counter weights that are needed on the front to counteract the moment from lifting and towing the tractors. The following is the loading, shear, and moment diagrams for the frame:



The calculations are as follows:

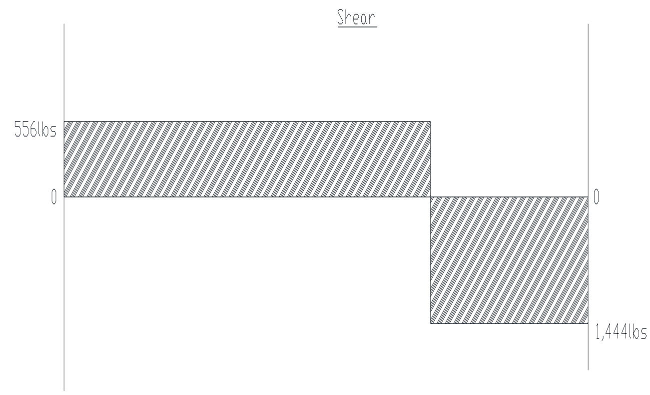
$$\sum M_{R1} = 0$$

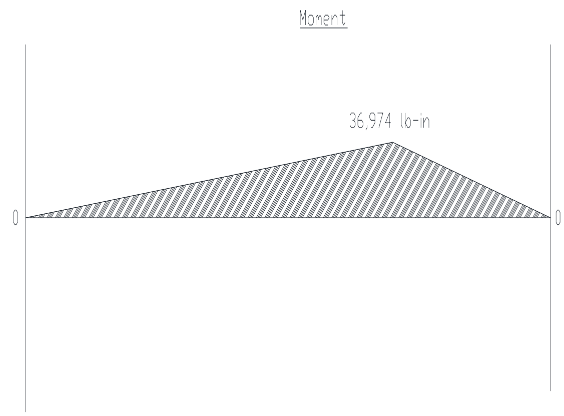
$$(72in)(R_2) - (52in)(2{,}000lbs) = 0$$

$$R_2 = \frac{(52in)(2{,}000lbs)}{72in}$$

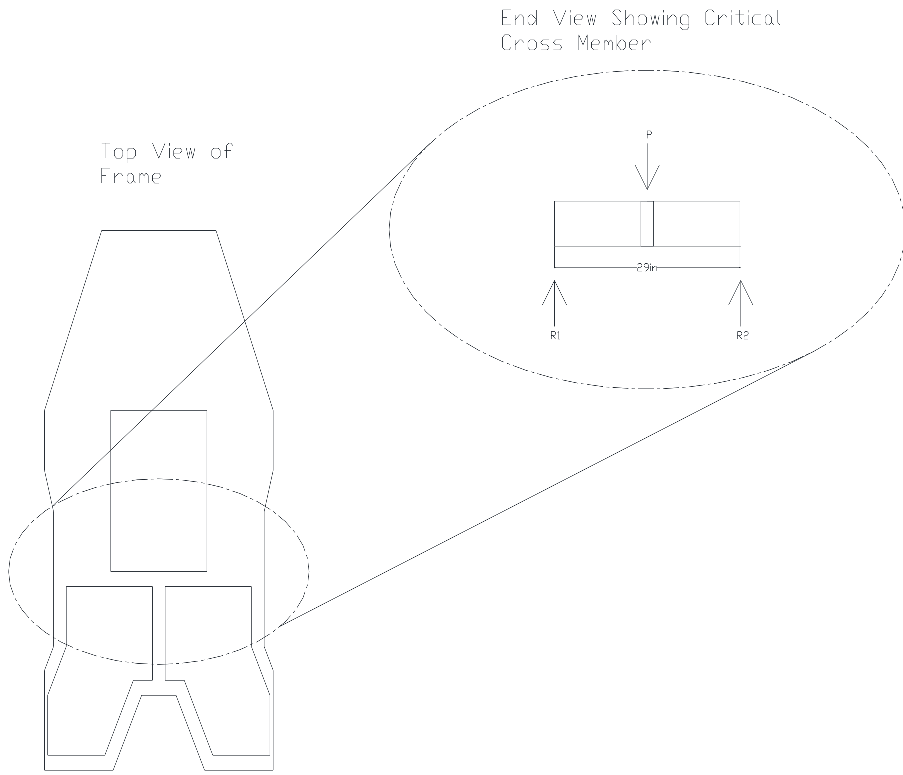
$$R_2 = 1{,}444lbs$$

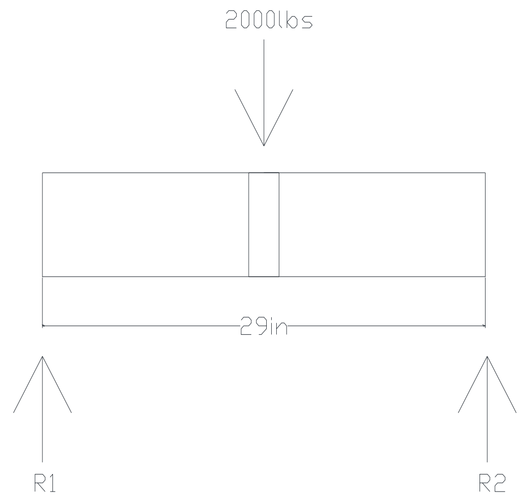
$$\therefore R_1 = 556lbs$$


Shear
556lbs
0
0
1,444lbs


Moment
36,974 lb-in
0
0

The next calculations are for the cross member that is holding the base for the pivot arms. This member, although supported in the middle, was evaluated in a worst-case scenario where the middle brace member is non-existent. This middle cross member bears all of the weight of the tractor and is therefore the most important member to evaluate.





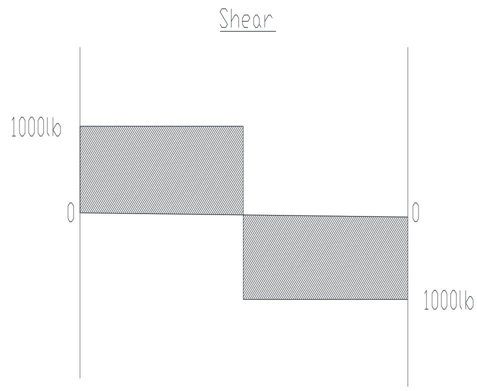
$$\sum M_{R1} = 0$$

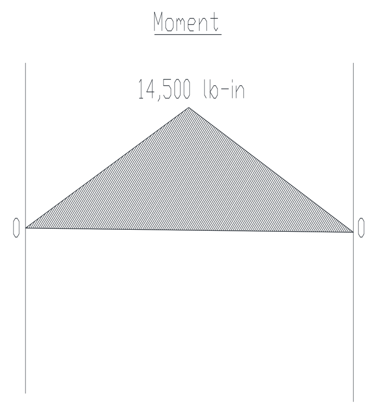
$$(29in)(R_2) - (14.5in)(2{,}000lbs) = 0$$

$$R_2 = \frac{(14.5in)(2{,}000lbs)}{29in}$$

$$R_2 = 1{,}000lbs$$

$$\therefore R_1 = 1000lbs$$





In both cases, the maximum moment generated yields a bending stress well below 60% of the yield strength of the steel. The load crosses into both the left and right sections of the cross member. To show this, consider the moment generated of 14,500 lb-in:

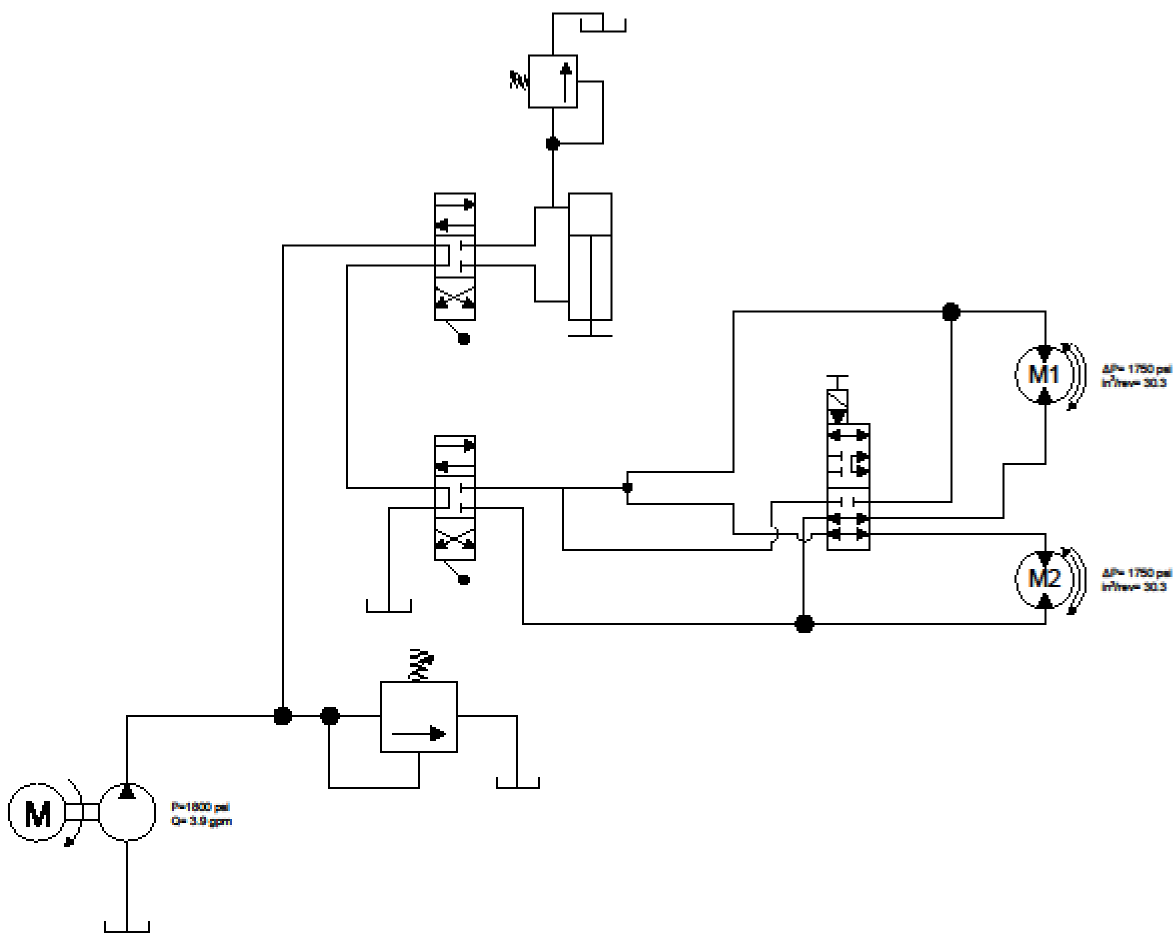
$$\sigma = \frac{Mc}{I}$$

$$\sigma = \frac{(14{,}500lb - in)(3in)}{(2\ sections)\left(\frac{1}{12}\right)[(2in)(6in)^3 - (1.5in)(5.5in)^3]}$$

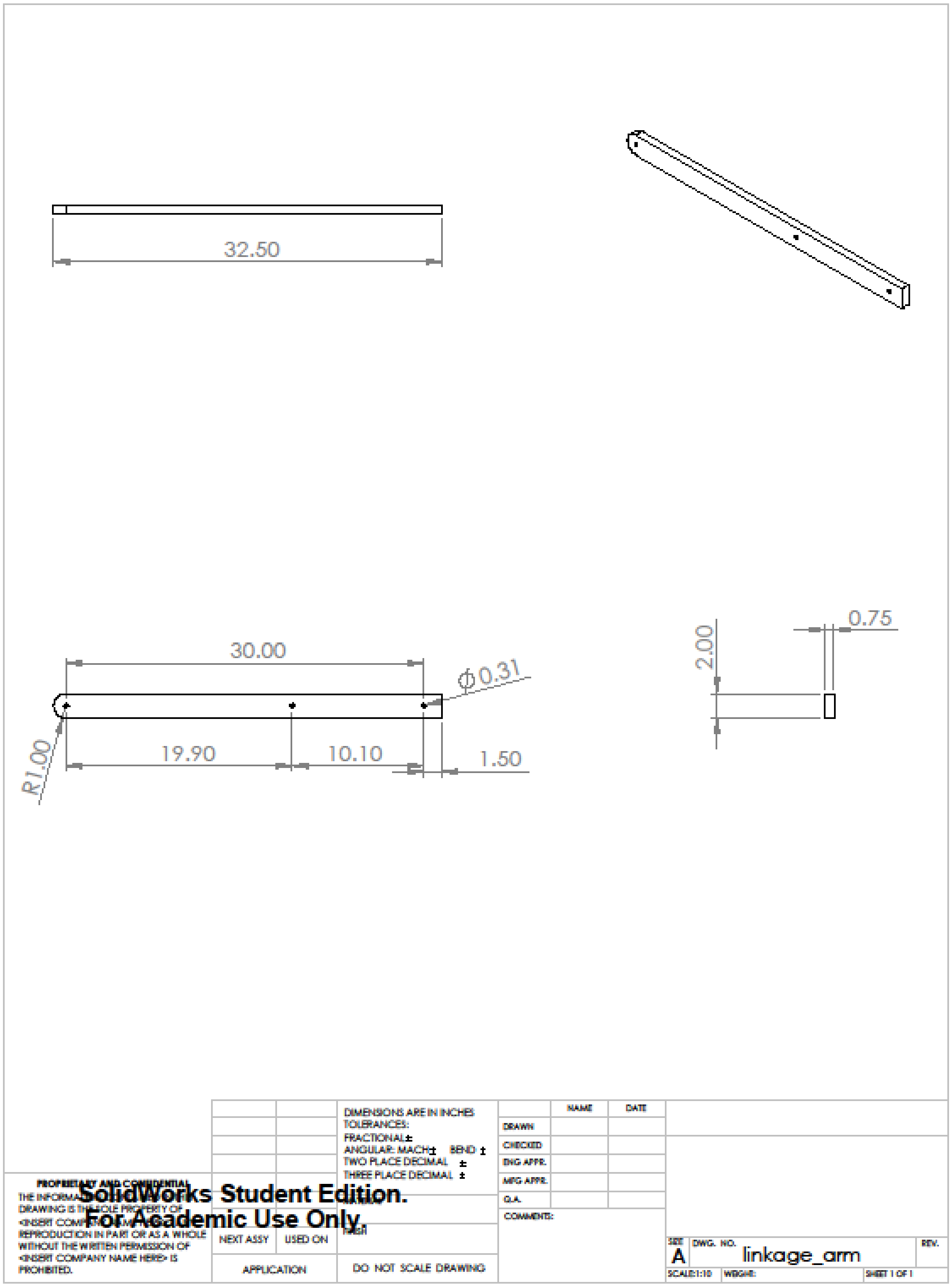
$$\sigma = 1{,}431\ \frac{psi}{section}$$

APPENDIX H: Hydraulic Schematic

Hydraulic Schematic



APPENDIX I: Drawings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⌀0.31
R1.00
19.90
10.10
1.50
2.00
0.75
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ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±
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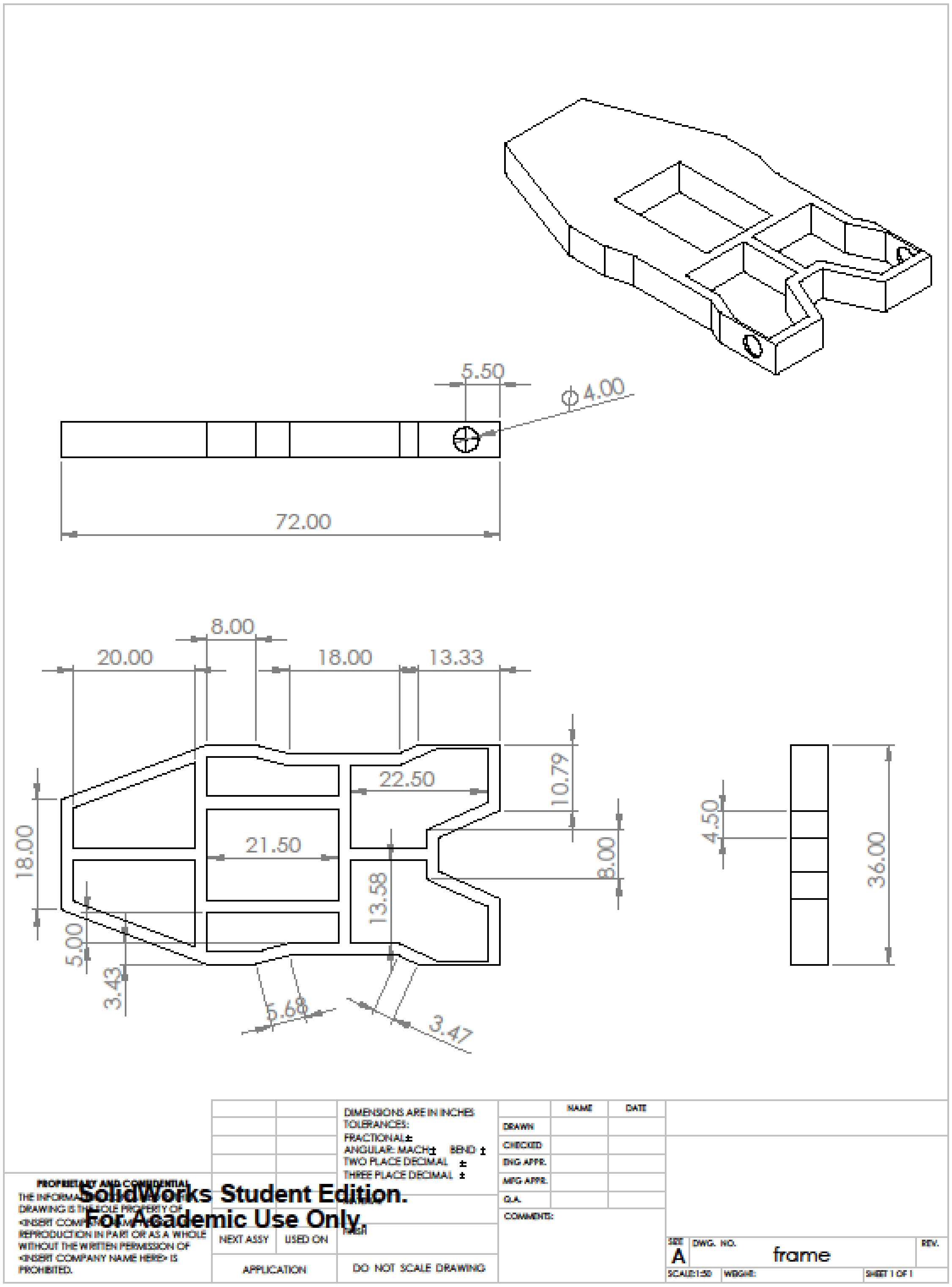
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Φ4.00
72.00
8.00
20.00
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13.33
22.50
10.79
21.50
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4.50
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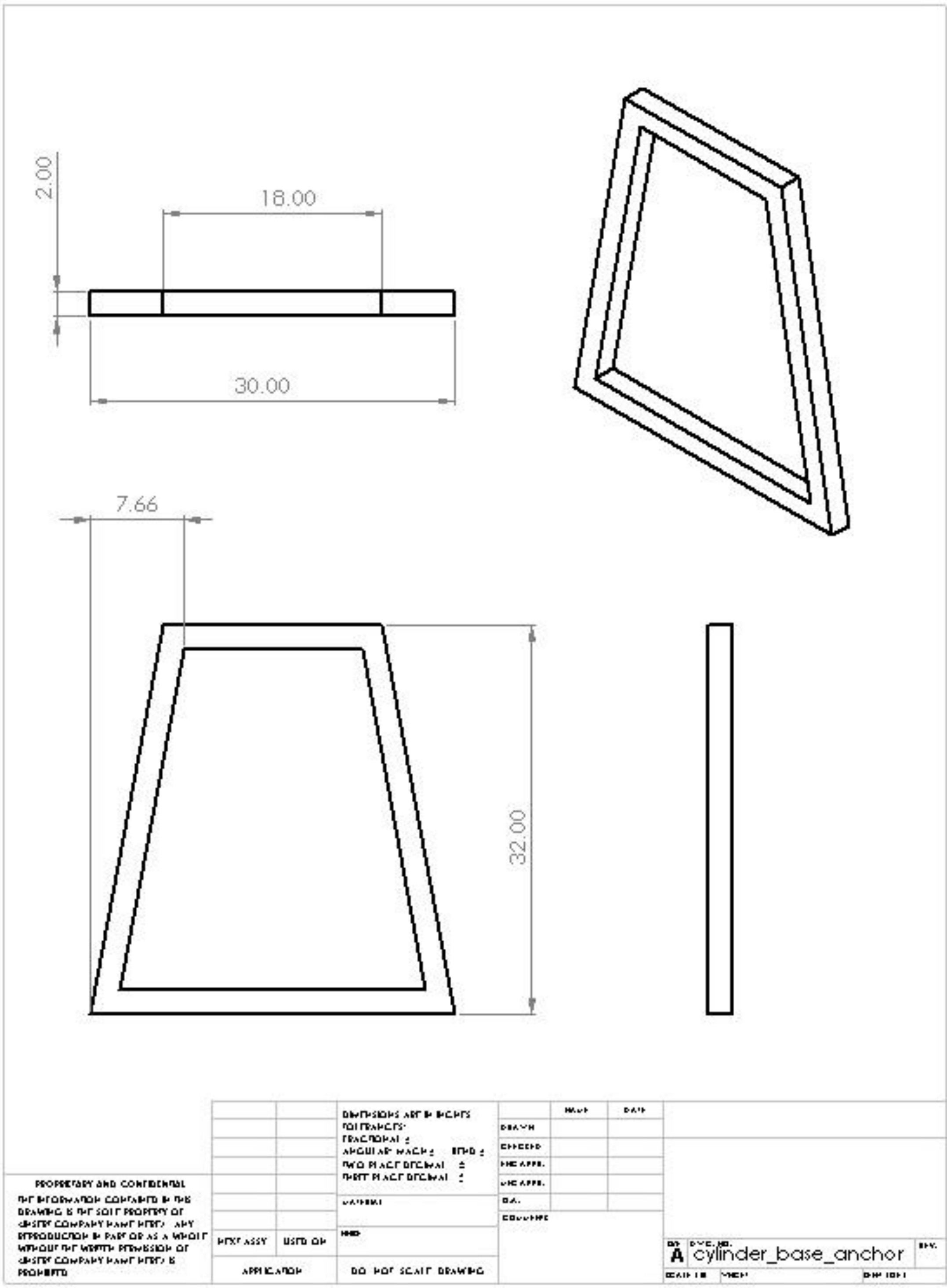
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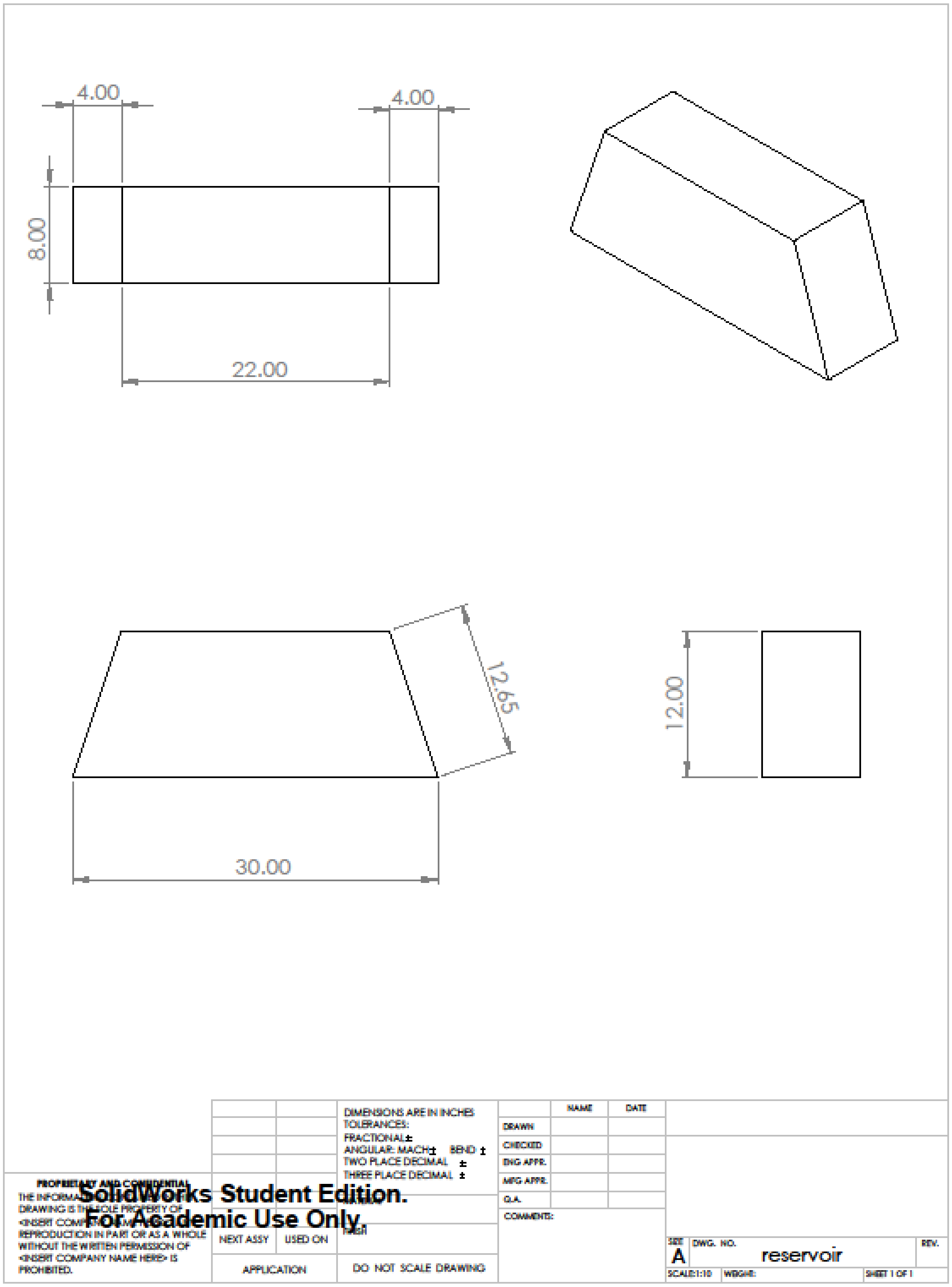
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TWO PLACE DECIMAL ±
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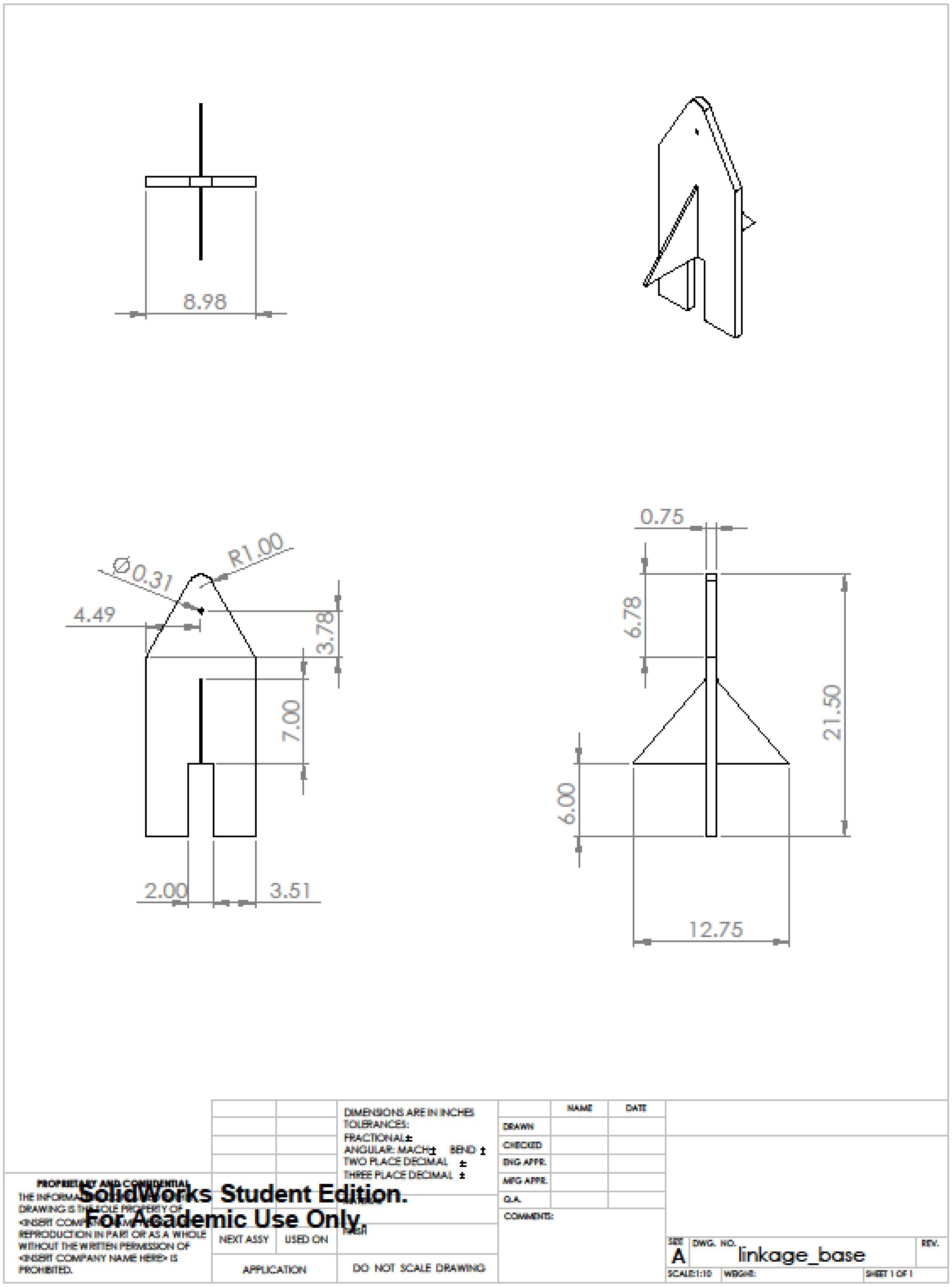
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4.49
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7.00
2.00
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0.75
6.78
21.50
6.00
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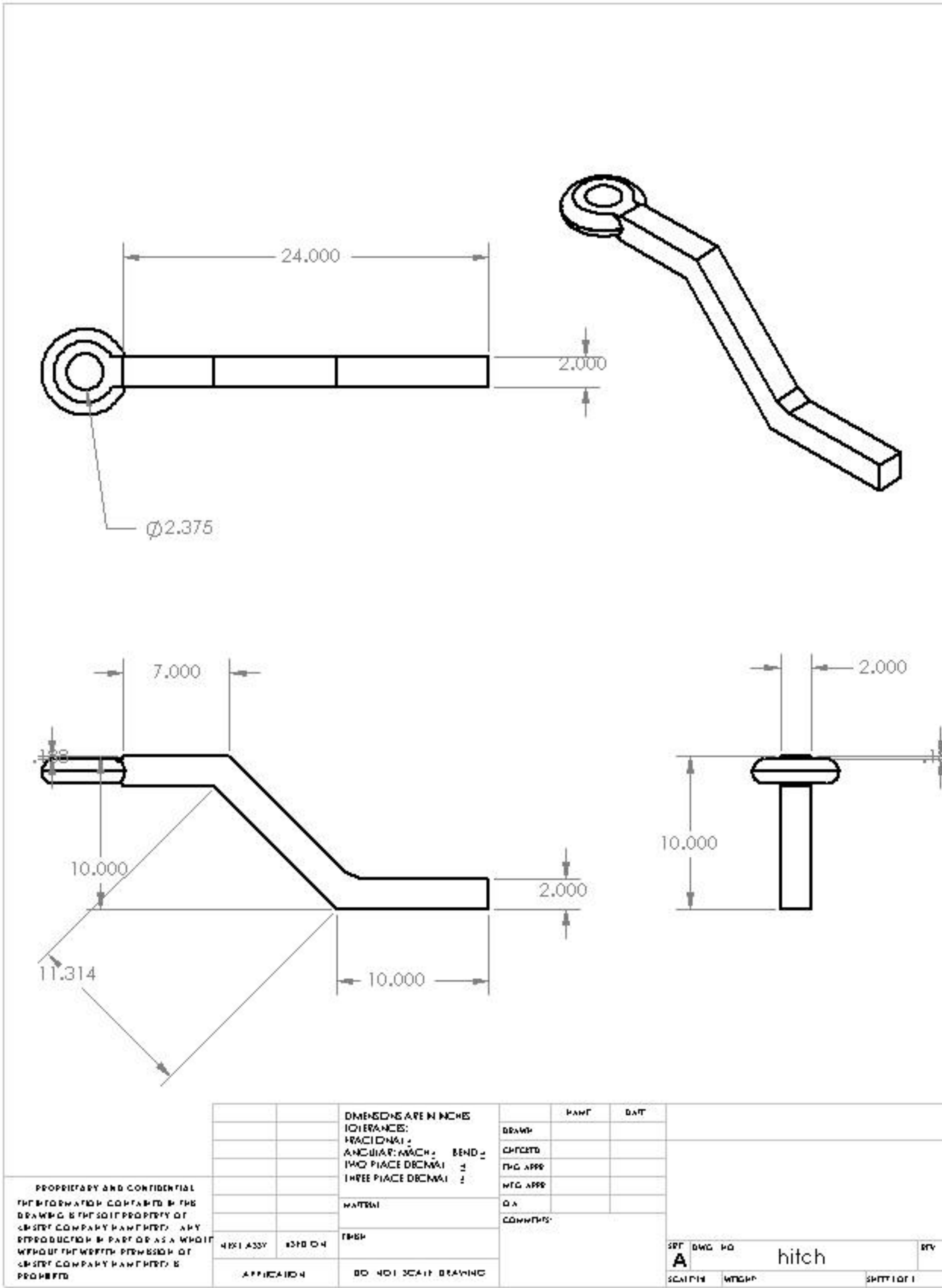
24.000
2.000
Ø2.375
7.000
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11.314
2.000
10.000
2.000
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TOLERANCES:
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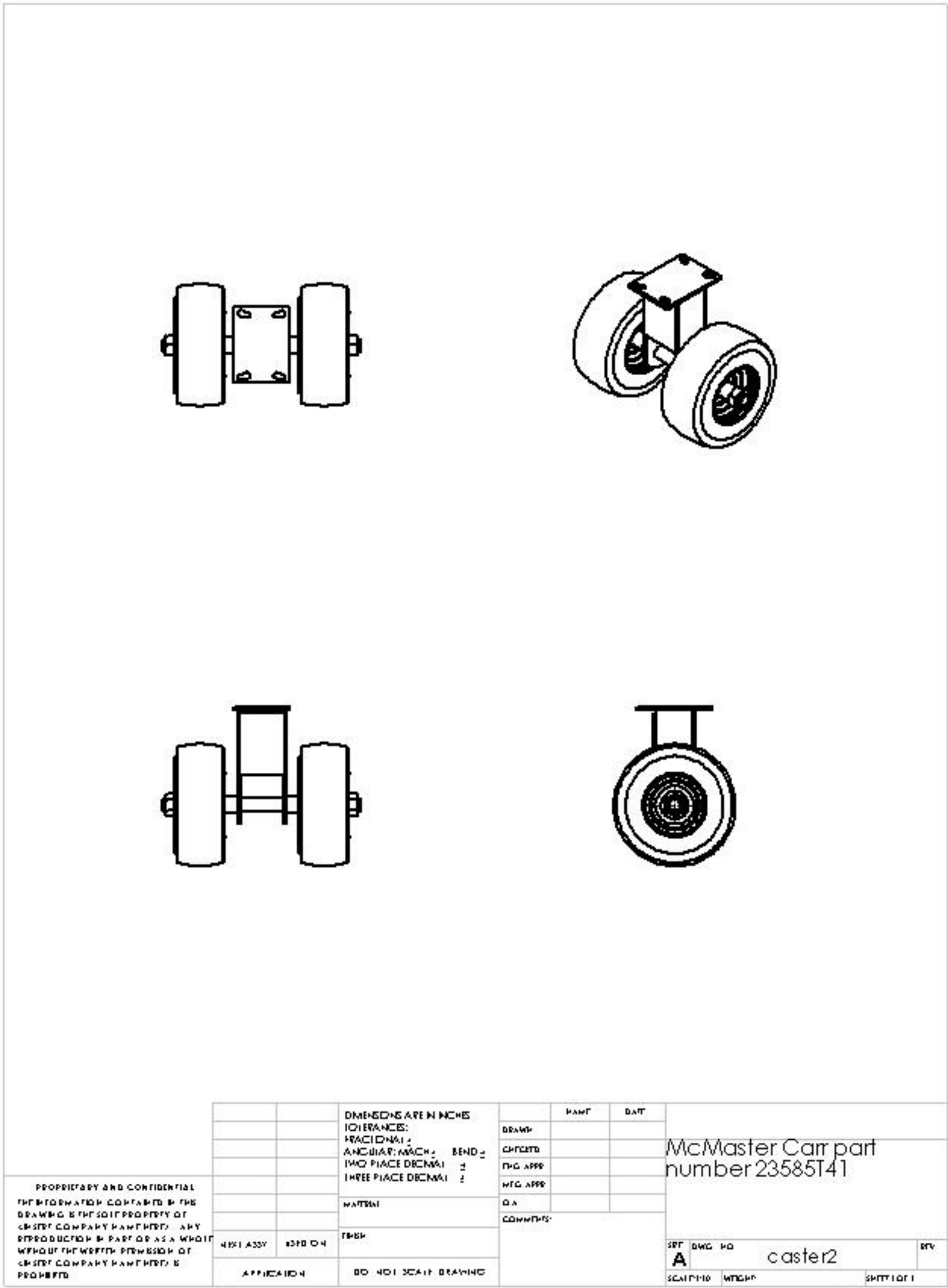
McMaster Carr part number 23585T41
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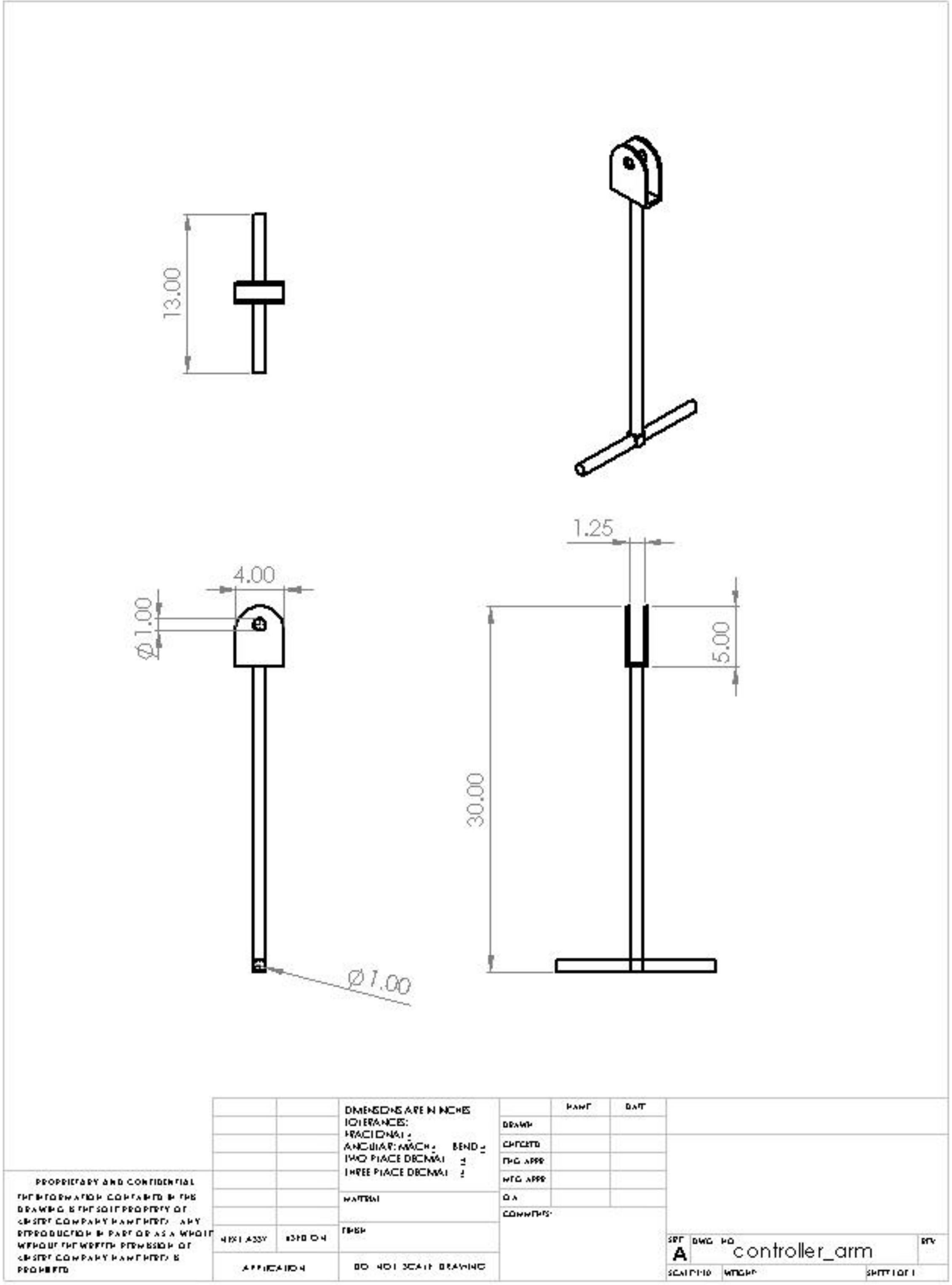
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4.00
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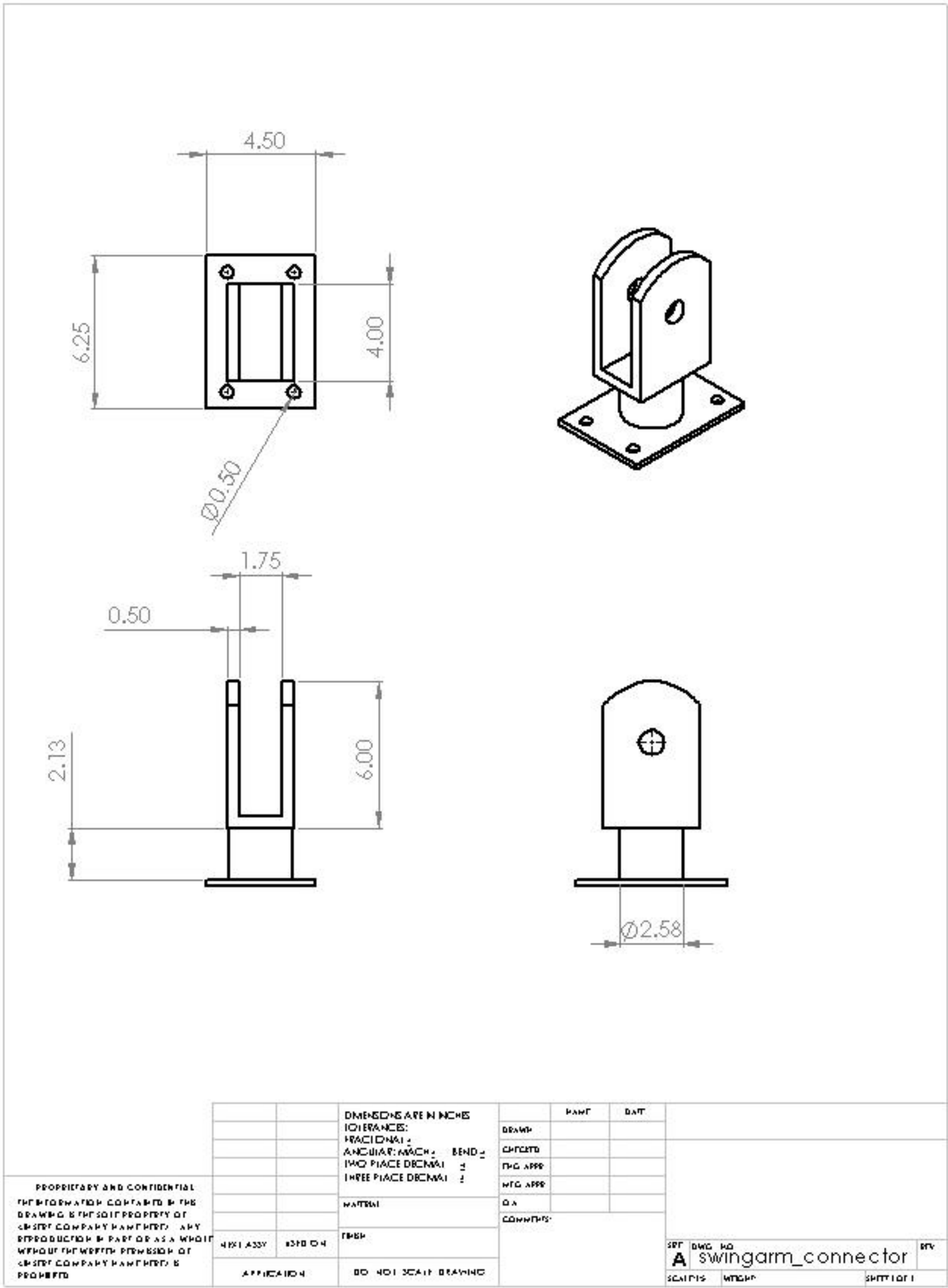
controller_arm


4.50
6.25
4.00
Ø0.50
1.75
0.50
2.13
6.00
Ø2.58
DIMENSIONS ARE IN INCHES
TOLERANCES:
PROPRIETARY AND CONFIDENTIAL
DO NOT SCALE DRAWING
A
swingarm_connector