MECH 4860

ENGINEERING DESIGN

TEAM 3 – TRIPLE E RV

COMPOSITE MATERIAL UNCOILER

Final Design Report

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Submitted to:

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Submitted on: December 6, 2017

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December 6, 2017

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Dear Dr. Paul Labossiere,

Please find the enclosed Final Design Report for the Triple E RV Composite Material Uncoiler project.

The report begins by outlining Triple E RV's issues with the composite material which are to be addressed, as well as constraints imposed on the design. A thorough needs analysis was then performed to determine design targets and marginal values. The report then presents the iterative process of brainstorming concepts to address individual issues, and evaluating the performance of these concepts. Lastly, the report explains the final design developed by the team in detail, and summarizes the overall project, goals achieved, and recommendations for the client moving forward. Several Appendices are then attached containing all technical engineering drawings developed by the team, the in-depth concept generation and selection process, analytical calculations regarding design specifications and safety, a thorough stress analysis of all components of the design including finite element analysis, weld sizing, and bolt bearing stresses, and lastly, an in-depth cost analysis of the final design.

Thank you for your time, and please contact us with any concerns or feedback you may have.

Sincerely,

__ ______________
Matt Froese

______ ______________
Kuankuan Lu

__ ____________
Brenden Scott

_____ _____________
Mannan Thakur

MECH 4860 – Engineering Design, Team 3
Department of Mechanical Engineering
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Executive Summary

Triple E RV, manufacturer of recreational vehicles, currently faces an issue regarding the manufacturing of floors and walls of their vehicles. The current system being used to handle 5000 lbs rolls of composite material, called Cosmolite contains no provision to maintain the tightly wrapped roll once the packaging straps are removed. Additionally, the current system is difficult to maneuver due to its large size and weight. To reduce operator strain and increase efficiency, Triple E RV has requested that a new system be designed to handle the rolls of Cosmolite material.

By examining the requests of the client, the team has identified the most important needs of the project which include supporting the weight of the roll of composite material, constraining the composite material in a tightly wrapped roll during use, providing a better method of uncoiling and recoiling the composite material, positively retaining the roll of composite material, and improving the maneuverability of the roll of material.

A total of 56 feasible concepts to achieve the client's requirements were generated in the team's concept generation and selection phase. These concepts were evaluated using screening and scoring matrices based on their ability to meet the needs of the client. The results of the screening and scoring process, along with input from the client, helped the team identify the concept which would result in the optimal final design.

Numerous analyses were performed during the optimization included in the final design phase of the project. Analysis of the composite material's stiffness was determined to evaluate the force required to maintain the composite material in a tightly wrapped roll. Safety and kinematic analysis was performed on the composite roll of material to determine the force required to move the design with a full roll of material, as well as the torque required to rotate the roll at an appropriate rate for uncoiling and recoiling. Stress analysis, including preliminary finite element analysis, welding sizing analysis, and bearing stress analysis were performed on all load bearing components to determine the required component sizes necessary to maintain a safe design.

The foundation of the design is the Bottom Frame, constructed of structural steel tubing. The frame is supported by four Walking Axles, each with two steel swivel casters. The Walking Axles pivot such that all eight caster wheels maintain contact with the floor, reducing bearing and rolling friction, and thus reducing the amount of force required to maneuver the design. Four steel rollers are mounted to the Bottom Frame with pillow block bearing housings.

These four rollers form a cradle in which the roll of composite material rests. Using the four-roller system eliminates the need for a shaft to support the roll of composite material, thus greatly reducing the time required during the composite roll changeover process.

The composite material is constrained in a tightly wrapped roll by the Top Roller subassembly. Four steel rollers, supported by flanged bearing housings rest firmly on top of the roll of composite material. The 285 lbs weight of the Top Roller subassembly provides sufficient force to maintain the curvature of the composite material, eliminating unwanted expansion of the material.

The Top Roller subassembly is supported by a horizontal Top Swinging Arm, which is in turn supported by a vertical Side Supporting Arm extending from the Bottom Frame. A hydraulic cylinder with an integrated manual pump provides the ability to raise the Top Roller subassembly during the composite material roll exchange process.

Two Retaining Shafts are inserted into both ends of the roll of composite material. These subassemblies secure to the composite roll to the Bottom Frame, thus positively retain the roll of composite material from escaping the design should the composite material roll be bumped or the design overturned.

Uncoiling and recoiling of the composite material is by achieved using one of two methods. Large Bus Wheels attached to the Bottom Rollers on which the composite material rests provide leverage to manually rotate the composite material in either direction. A liquid rubber coating applied to all steel rollers provides a high friction surface for the effective transfer of torque to the roll of composite material. Alternatively, an electric motor and reduction gearbox can be used to rotate the Bottom Rollers via a chain drive system. The electric drive system can reduce operator strain and increases the speed at which the composite material can be uncoiled and recoiled.

Lastly, through an in-depth cost analysis, the team determined that the total cost of producing the final design is $5,072.23 USD.

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1. Introduction and Project Background

The following sections include background information on Triple E RV, introduce roles and responsibilities of all the project stakeholders, and describe the full project as stated by the client.

1.1. Triple E RV

The Triple E RV has been producing recreational vehicles, trailers, and leisure travel vans since 1965 [1]. The production process combines many processes including welding, molding, painting, laser cutting, cabinet manufacturing, walls and ceiling manufacturing, and many more [2]. Using these processes, Triple E RV can produce a wide array of recreational vehicles. Triple E RV has three different production lines including the Serenity, Unity, and Wonder models [1].

The walls and ceiling manufacturing process is one of the most important processes at Triple E RV. It starts with manually uncoiling a 5000 lbs composite sheet material (Cosmolite), which is then used in other processes such as gluing, pressing and cutting. Due to the heavy weight of the coiled Cosmolite roll, four to six people are required every time to uncoil a sheet of composite material. The objective of the project is to make the process of uncoiling the composite easier and manageable, ideally by only two individuals.

1.2. The Team

The team consists of four mechanical engineering students in their graduating year: Matthew Froese, Mannan Thakur, Brenden Scott, and Kuankuan Lu. Table I lists the individual roles and responsibilities of each team member. The team observed, studied, and analyzed the Cosmolite uncoiling process completed by Triple E RV employees such that the process is completely understood. Once the team understood the process, a robust design capable of supporting and easing the use of the composite material was be developed. By developing a new design, Triple E RV will be able to manage the Cosmolite material, thereby saving time and decreasing risk of injuries. The stakeholders for this project consist of Triple E RV management and employees who work within the current process, the team's academic advisor, and the mechanical engineering student team members. Kevin Peters and Cornie Fehr are the company contacts from Triple E RV. Their responsibilities include project guidance and approval, and accepting, and implementing the final proposed design delivered by the mechanical engineering student team at the end of the project. The academic project advisor is Dr. Paul Labossiere, P. Eng., who will be guiding and advising the team on the execution of this project. The advisor is also responsible for reviewing the team reports and clearing up any questions the team might have about the project.

TABLE 1-I: TEAM MEMBER ROLES AND RESPONSIBILITIES

Team Member	Role	Responsibility
Matthew Froese	Team Manager	• Communicating with the client.
		• Planning and leading team meetings.
		• Driving the team to the client's site.
Mannan Thakur	Team Secretary	• Preparing agenda and meeting minutes.
		• Noting actions and decisions made during meetings.
Brenden Scott	Team Member	• Keeping track of project progress and updating the Gantt chart.
		• Compiling and formating all reports.
Kuankuan Lu	Team Member	• Backing up all documents and work done.
		• Researching existing technology related to the project.

1.3. Problem Statement

Uncoiling of the Cosmolite material is a serious problem Triple E RV is facing during their manufacturing process. The composite sheet material used for flooring and walls of a recreation vehicle is supported by a shaft and framed cart. On the current frame, the composite material will automatically uncoil after the packaging and straps are removed. This causes trouble for operators during the manufacturing process. The uncoiled composite sheet material will expand and touch the rolling support frame, which makes excess sheet material difficult to recoil. Currently, four to six people are required to work together when excess sheet material must be recoiled. This recoiling issue makes the manufacturing process time-consuming, laborious and inefficient. Therefore, solving the uncoiling problem is imperative as it will save manpower, time, reduce the risk of operator injury, and make the manufacturing process more efficient.

1.4. Design Objectives

Through examining the University of Manitoba IDEA Program application form submitted by Triple E RV, as well as conversations with the client, the team identified several design objectives to be satisfied [2], [3]. This section presents these objectives as expressed by the client on the IDEA application form.

Objective 1: The frame assembly must be capable of supporting 5000 lbs.

This first objective states that the design, assumed to be a frame, must support the weight of the roll of Cosmolite material. A single roll of the material weighs approximately 4200 to 4300 lbs. The weight of a roll plus any part of the design fixed to the roll which must also be supported was approximated as 5000 lbs.

Objective 2: The frame assembly must be on wheels that swivel and be movable by two individuals.

The second objective involves the transportation of the roll of Cosmolite. The client's current production process involves the use of two different thicknesses of Cosmolite for the construction of walls and floors, with both thicknesses of material being used at the same cutting and gluing table. Because two different thicknesses are used, the rolls of material must be moved to the work station when in use, and to a temporary storage area when not in use. The design must facilitate this movement, with the client's preference that no more than two individuals be required to move the rolls of material. This would be an improvement on the current process, in which four to six individuals are currently required to move the rolls of material.

Objective 3: There must be a mechanism in place to prevent the spool of material from uncoiling.

The rolls of Cosmolite material used by the client tend to unroll themselves and expand once the retaining straps used during transportation are removed, as can be seen in Figure 1-1. This loosely rolled material poses a problem for the client, as the material cannot easily be recoiled once pulled onto the cutting and gluing table. During the team's site visit, the recoiling process with the current system required four individuals. Two workers operate a crank on each side of the frame to turn the roll, and two workers attempting to contain the loosely wrapped material, pushing the material into a tighter roll. This process can be seen in Figure 1-1.

Figure 1-1: Material recoiling process requiring four workers [4].

The design should include a method of constraining the material in a tightly wrapped coil after the retaining straps have been removed, which would simplify the material recoiling process.

Objective 4: The design must be able to recoil excess sheet material.

This objective relates to the process described in objective three. Each time that material is uncoiled from the roll and pulled onto the cutting and gluing table, approximately 8-10 feet of material must be recoiled. The design must provide a means of retrieving this extra material.

Objective 5: The eight-foot roll should be simple to exchange with a forklift.

Due to the heavy weight of a roll of Cosmolite material, rolls must be placed on the current system using a forklift. The client wishes to continue using forklifts to position new rolls of material. Therefore, the design should be easily capable of accepting rolls via a forklift.

Objective 6: The shaft through the coil must lock in place for safety.

The current system used by the client utilizes a shaft running through the center of the roll of material. Each end of the shaft is lowered into vertical slots, which restrain the roll of material horizontally. The weight of the roll is relied on to retain the roll vertically. The current method of securing the shaft is shown in Figure 1-2.

Figure 1-2: Vertical slots used to secure shaft [5].

The client has requested that the design provide a means of locking this shaft (should the design use a shaft) in place, both horizontally and vertically. The roll should be secured in place for safety reasons; should the design be overturned or bumped, the roll of material should be prevented from escaping and causing damage to equipment or workers.

Objective 7: Powered uncoiling could be considered.

The objective of providing a powered method of uncoiling was given by the client as an optional feature which could be considered. Due to the weight of the roll and the loosely wrapped state of the material using the current system, the uncoiling and recoiling process can be difficult. Providing a means of uncoiling and recoiling the material using external power would simplify these processes and could result in increased productivity and reducing strenuous labor.

The objectives listed in this section are stated as described by the client. A thorough needs analysis was performed and presented later in this report, which will determine more specific specifications from the objectives listed here, as well as target values for each specification.

1.5. Design Constraints and Limitations

Through conversations with the client during site tours, the team identified four constraints that must be considered for the design. This section outlines these constraints as described by the client.

Constraint 1: Size.

The first constraint that must be considered is the overall size of the design, specifically, the footprint. The current system used by the client measures approximately nine feet long by four feet wide. The client would prefer that the design not exceed these dimensions, as doing so would reduce the free space available at the work station [2].

Constraint 2: Existing floor tracks.

The second constraint that must be considered is the design's points of contact with the floor. The client's current system utilizes angle iron floor tracks, fit to the system's current swivel caster wheels. These floor tracks locate, secure, and align the roll of Cosmolite with the cutting and gluing table. The client prefers that the design also utilize these floor tracks to position the roll of material [2]. Dimensions of the floor tracks are shown in Figure 1-3.

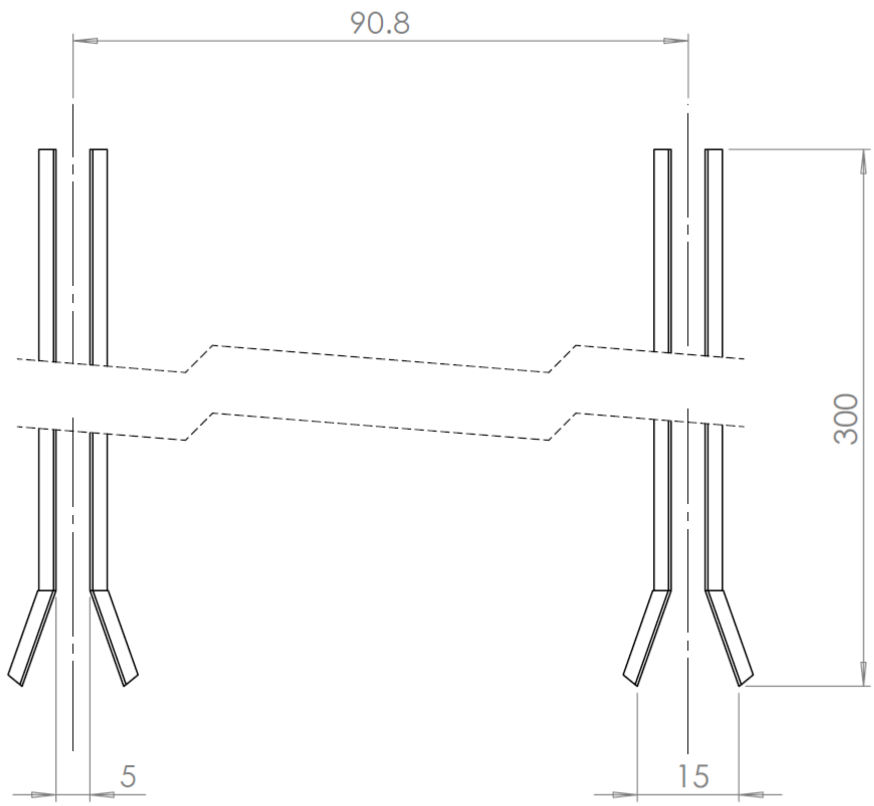

Figure 1-3: Floor track dimensions [cm] [6].

Constraint 3: Manufacturability.

The client has expressed their intentions to manufacture the design in-house, rather than sourcing the manufacturing out to a third party. This imposes the constraint that the design be manufacturable using methods available to the client. These methods include steel tube cutting, welding, laser cutting, and drilling [2], [7].

Constraint 4: Budget.

The client has stated that they are not particularly concerned with cost, so long as the cost was reasonable [7]. This subject will require revisiting later in the design optimization process, once a realistic cost of production is more clearly understood to more accurately determine what a reasonable budget limit is.

Again, it is important note that these constraints are presented as described by the client. These constraints are analyzed more thoroughly in the Needs Analysis section of this report, and are subject to change throughout the design phase as more information becomes available.

1.6. Design Scope

This project is to develop a robust rolling fixture. With this fixture, a 5000 lbs roll of composite sheet material can be safely supported and revolved. The design will be proven to exceed the current system in terms of ease of use, maneuverability, and safety. A preliminary engineering technical drawing package and all CAD models will be provided to the client as a final deliverable. The design will be evaluated via stress analysis, kinematic analysis, and cost analysis with list of suggested purchased items. Additionally, the team has determined the following are within the scope of the project: the composite coil changeover process, any maintenance of the design, and evaluation of an automated alternatives. In contrast, prototyping the design, integration of the design into Triple E RV's manufacturing process, and design of the electric motor control system are outside of the scope of the project.

2. Needs Analysis

The needs analysis process is important to guide the direction of the design process and establish a means of determining the success of the design. The analysis process began with the design objectives described previously in the Introduction and Project Background section. For each objective, the team determined a more general need which must be met to satisfy the design objective. For each need, measurable specifications or qualities were determined, which could be used to determine how well the need is met by the design. These specifications and qualities are shown in the "Measured Quality" column of Table 2-I. Each specification and quality were then assigned a number to indicate its importance to the perceived success of the design. The triangle, open circle, and dotted-circle system was used to assign importance, with a dotted-circle being assigned to items of high importance, open circle being assigned to items of moderate importance, and triangle being assigned to items of low importance. These symbol weightings were also used during the concept selection process to scale design performance based on the importance of each category. Next, each measured quality was given a metric depicting how the functional specification will be measured. For example, for the design objective stating that the design must support the weight of the roll of material, a measured specification of stress levels in the structure was chosen. The stress level was chosen to be evaluated using a factor of safety with respect to the chosen material's yield stress.

Not all measured qualities were assigned a quantitative metric; for the design objective of "the shaft through the coil must lock in place for safety," a need of "the design secures the roll of material" was determined. This need was chosen to be measured qualitatively by how difficult the roll of material is to remove from the design with the retention method engaged. This specification will be evaluated on a scale of one to five, with five being very difficult or impossible, and one being very easy.

Each metric was then assigned a target value. This value represents the best-case scenario, or the maximum realistically achievable performance of the design. Meeting these values should not necessarily be mandatory, as they may exceed the team's and client's expectations. Acceptable values were also assigned for each metric. These values represent the minimum acceptable performance of the design, and should be met or exceeded wherever possible.

The results of the needs analysis process are summarized in Table 2-I, which shows all identified needs, along with their measured quality, metric, target value, and acceptable value. Figure 2-1 also displays a graphical representation of which need relates or contributes to the fulfillment of each of the original design objectives.

TABLE 2-I: LIST OF ALL NEEDS, METRICS, AND TARGETS

Definition of Need	Importance Rating*	Measured Quality	Metric	Ideal Target	Acceptable Target
Physical Requirements					
The design supports the mass of the roll of material		Stress levels not to exceed yield limits	Factor of safety	2.5-3	2.5-5
The material is constrained from unwanted uncoiling		Ratio actual diameter to tightly wrapped diameter	-	1	<1.25
The design secures the roll of material		Difficulty to remove roll with restraint engaged	Qualitative (1-5)	5	3
The design utilizes existing floor tracks		Utilization of floor tracks is possible	Binary	1	1
The design fits within the existing work station		Length	Feet	8	<9.5
		Width	Feet	4	<5
	○	Height	Feet	5	<6
The design aligns the roll of material with the next work station	○	Maximum allowable angle of misalignment	Degrees	<1°	<5°
Manufacturing and Cost Related Needs					
The design is produced using manufacturing methods available to the client	○	Percent of outsourced manufacturing	%	0	<20%
The design requires minimal purchased parts	○	Percentage of total cost	%	<10%	<25%
The design requires minimal labor cost to manufacture		Percentage of total cost	%	~50%	<65%
The design uses minimal raw material		Percentage of total cost	%	~40%	<50%
Process and Design Use Needs					
The design is maneuverable		Force required to maneuver	Newtons	<2X single person push force	<3X single person push force
The design is able to recoil material		Yes or No	Binary	1	1
The material is able to be removed and installed with a forklift		Yes or No	Binary	1	1
The design is stable after transportation		Ability to fix design in place	Binary	1	0
The design is able to be moved by minimal amount of people	○	Number of people required	People	<2	<=3
The roll of material requires minimal manpower to exchange		Number of people required	People	1	2
The material can be exchanged in a timely manner		Time required	Minutes	<30 minutes	<1.5 hours
Product Lifecycle Needs					
The design requires little maintenance over its life	○	Maintenance period	Years	<5	<2
The design provides easy access to servicable items	○	Disassembly time	Minutes	<10 minutes	<20 minutes
The design lasts a long time		Functional life	Years	>10	>5
Safety Needs					
The design adheres to relevant safety standard		Yes or No	Binary	1	1
Automation Needs					
The design uses an external energy source to uncoil and recoil the material	○	Yes or No	Binary	1	0
The design can operate using either manpower or external power	○	Cost to implement external power	$	250 (~cost of motor)	500 (~2x motor)
	○	Time to switch between power methods (after installed)	Minutes	<10	<30

***Note: = weak, ○ = medium, = strong**

Need \ Design Objective	Frame must be cacpable of supporting 5000 lbs	Frame assembly must be on wheels that swivel and movable by 2 individuals	Must be a mechanism in place to prevent spool from uncoiling	Must be able to recoil material	8' roll should be simple to exchange with forklift	Shaft through coil must lock in place for safety	Powered uncoiling could be considered	Cost must be reasonable	The size of the frame must work with existing processes	Frame must be manufacturable by Triple E RV	Frame must be durable	Frame must be maintainable
The design supports the mass of the roll of material	•											
The design aligns the roll of material with the next work station		•										
The design is maneuverable		•										
The design is stable during and after transportation		•										
The design is able to be moved with a maximum of two individuals		•										
The material is constrained from unwanted uncoiling			•									
The design is able to recoil material				•								
The material is able to be removed and installed with a forklift					•							
The material can be exchanged in a timely manner					•							
The material requires minimal manpower to exchange					•							
The design secures the roll of material						•						
The design uses an external energy source to uncoil and recoil the material							•					
The design can operate using either manpower or external power							•					
The design requires minimal purchased parts								•				
The design requires minimal labour cost to manufacture								•				
The design uses minimal raw material								•				
The design fits within the existing work station									•			
The design utilizes existing floor tracks									•			
The design is produced using manufacturing methods available to the client										•		
The design lasts a long time, subject to regular use											•	
The design adheres to relevant safety standard												•
The design requires little maintenance over its life												•
The design provides easy access to servicable items												•

Figure 2-1: Original design objectives verse design specifications.

3. Concept Generation and Selection

After defining the project specifications and having a clear understanding of the problem statement, each of the team members were asked to come up with at least five design ideas individually. Overall, the team was able to generate a total of 37 feasible concepts. The team screened all concepts into four main categorizes: ability to support the composite material roll, ability to constrain expansion of the composite material, ability to reduce the pushing and pulling effort for operators, and ability to reduce coiling and uncoiling effort. By screening each concept based on their relative individual performance within each category, the best concepts were selected to move forward with. Computer aided design (CAD) of the chosen models were created to display the ideas to the client.

A secondary concept generation was done after the client evaluation. Based on the feedback from the client, the team received two primary points of inspiration: First, the client particularly liked shaftless concepts, where two supporting rollers would support the composite material roll. The client requested the team to move forward with the shaftless concept. Second, the Triple E RV employees mentioned some of their concerns and ideas while moving forward on said shaftless concept such as ensuring the center of mass stays in the middle of the cart as the composite material roll is used such that the cart cannot flip due to its own weight. Thus, ensuring that there was another external means of fixing the composite material roll to the cart was important. These points were further emphasized with the design moving forward.

Based on the suggestions from client, the team separated the remaining components of the cart to be selected into four main categories: ability to constrain expansion of composite material, ability to positively retain the composite roll, ability to reduce movement effort, and ability to reduce coiling and uncoiling effort. By taking a more structured approach than with the initial concept generation, the team developed 19 new concepts that were named and identified by a new letter coding system. With these 19 concepts, an in-depth concept scoring process was completed to select the best design components moving forward. From the scoring process, the team determined the main functional components to be included in the final design consisted of: Weighted Top Rollers to constrain composite roll expansion, a support mechanism consisting of a Top Swinging Arm, Side Supporting Arm, hydraulic lifting cylinder, Bottom Framing, Bottom Rollers, Walking Axles, Handle Bars, Retaining Shafts, and manual or automatic drive systems, Bus Wheels or electric motor, respectively.

The entire concept generation process, from initial research to the chosen conceptual components of the final design, are outlined in further detail in Appendix B: Concept Generation and Selection Process.

4. Final Design Details

This section gives a final overview of the composite material uncoiler design, including the details of individual subassemblies and their integration of design features. Lastly, the suggested operating procedures of the design are also presented.

4.1. Final Design Details

The foundation of the composite material uncoiler is the Bottom Frame, consisting of six rectangular structural steel tubing. Two long sections of tube run the length of the cart, with four shorter sections of tubing spanning between the two longer sections. The shorter sections of tubing are arranged such that one tube spans between each set of ends of the longer tube sections, with the remaining two tubes spaced inwards 28". The Bottom Frame can be seen in Figure 4-1.

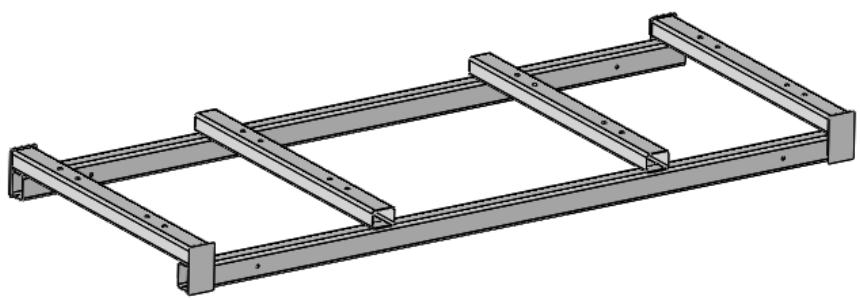

Figure 4-1: Bottom Frame tubing sections [8].

The Bottom Frame rest on four sets of Walking Axles. Each Walking Axle consists of a bent sheet metal base which pivots on a bolt extending through the Long Frame Supports, with two swivel caster wheels mounted to each base. The wheels are constructed of steel to reduce rolling friction between the wheels and the ground surface, reducing the force required to move the cart. The pivoting action of the Walking Axles ensures that all eight swivel casters are in contact with the ground at all times, further reducing rolling friction and force required to move the cart.

Between each outer set of Frame Cross Supports spans a pair of steel rollers. The rollers are constructed of cut sections of round structural tubing with steel end plates welded to each end. A solid steel shaft runs through the roller and is welded to each end plate. The rollers are supported by pillow block bearing housings which are bolted to the top surface of the Frame Cross Supports. These four rollers form a cradle in which the roll of composite material rests. The use of four rollers supporting the composite material from beneath eliminates the need for a shaft running through the roll of material, as is currently used to support the composite material. Eliminating the long shaft through the roll greatly simplifies the roll exchange process, since the roll can simply be placed on the four rollers using a forklift, rather than requiring the removal of the shaft from the old roll and installation into the new roll. A liquid rubber coating is applied to the surface of each roller, serving two purposes: protection of the composite material from damage, and the transfer of torque from the rollers to the roll of material necessary for the uncoiling and recoiling processes. The four Bottom Rollers, as well as the Walking Axles, can be seen in Figure 4-2.

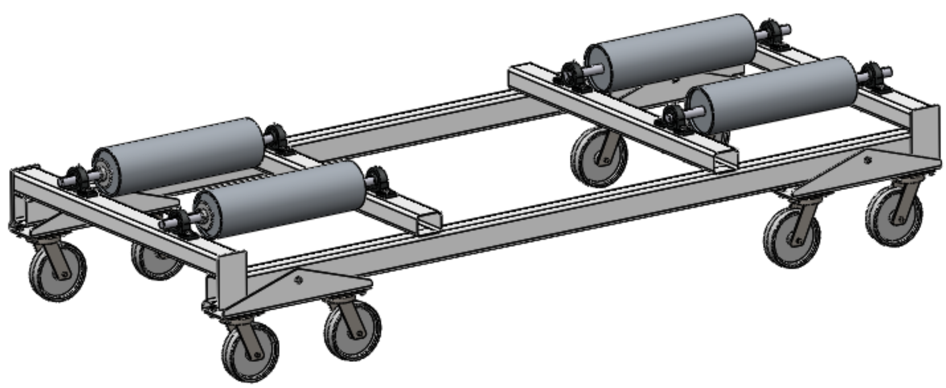

Figure 4-2: Bottom Frame with Bottom Rollers and Walking Axles [8].

Unwanted expansion of the composite material is prevented using the Top Roller assembly. The Top Roller assembly consists of two sets of rollers which rest on top of the roll of composite material. These rollers are again constructed of round structural tubing with sheet steel end plates. One roller in each set is welded to a solid steel shaft running through the roller, as with the Bottom Rollers. However, the shaft is not welded to the second roller in the set. The protruding end of the solid shaft passes through a set of flanged bearings in the Top Roller subassembly before passing through the second roller. A clamping shaft collar is then slid onto the end of the shaft and locked in place. This shaft collar prevents the set of rollers from separating, and allows for simple disassembly of the Top Roller Assembly, should the Top Roller bearings need to be replaced or inspected.

The Top Roller hanger consists of two rectangular steel tubes welded into the shape of an inverted “V”, with a steel plate extending upwards from the point of intersection. Two flange bearing housings are mounted to either end of the “V”, which support each set of Top Rollers. The Top Rollers and hanger rest on top of the roll of composite material, with their weight constraining the expansion of the material. The Top Roller and Hanger assembly, with one roller in the exploded state, is shown in Figure 4-3.

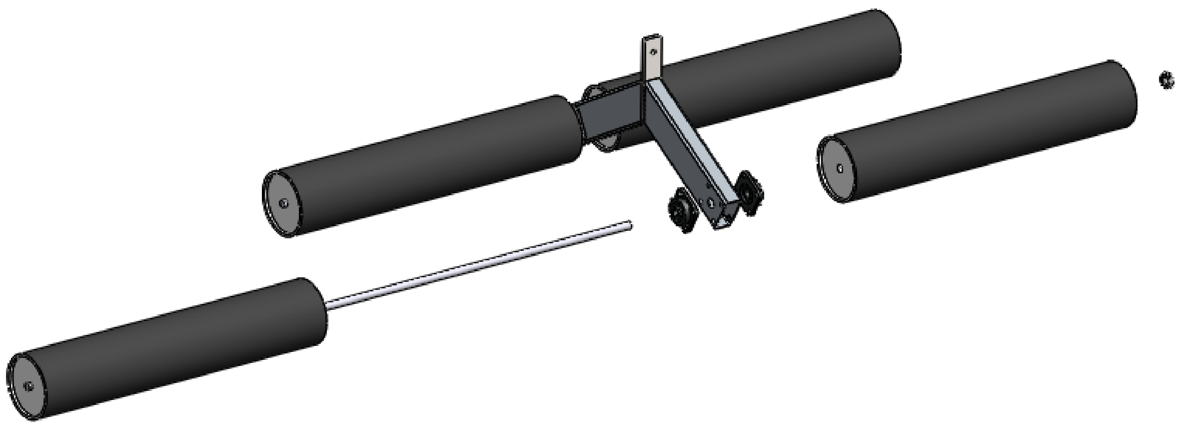

Figure 4-3: Top Roller assembly [8].

The Top Roller subassembly is supported by the Swinging Arm, which extends horizontally from one end of the cart. The Swinging Arm is constructed of rectangular structural steel tubing, with a section of the tubing wall cut open for the Hanging Plate of the Top Rollers to be inserted. The Hanging Plate is supported with a single bolt which allows the Top Roller assembly to pivot, adjusting orientation to rest evenly on the composite material roll. The other end of the Swinging Arm is supported by a vertical Side Arm, which extends from the Bottom Frame. A single bolt connects the Swinging Arm and the Side Arm, allowing the Swinging Arm to pivot and adjust the height of the Top Roller Assembly. A manual pump hydraulic cylinder extends between steel brackets welded to each of the Swinging Arm and the Side Arm. During the composite roll exchange process, this hydraulic cylinder can be used to raise the Top Roller assembly to such a height that a new roll of composite material can be safely placed on the Bottom Frame using a forklift. After the roll of material has been exchanged, the valve in the hydraulic cylinder can be opened, causing the full weight of the Top Roller subassembly to be placed on the composite roll. In addition to the hydraulic cylinder supporting the Top Rollers during the exchange process, Safety Brackets are welded to the top of the Side Arm. With the Swinging Arm in the raised position, a safety pin can be passed through holes in the Safety

Bracket and Swinging Arm Bracket. This safety pin will support the weight of the Top Roller subassembly should the hydraulic cylinder develop a leak. The Side Arm and Swinging Arm are shown supporting the Top Roller subassembly in Figure 4-4.

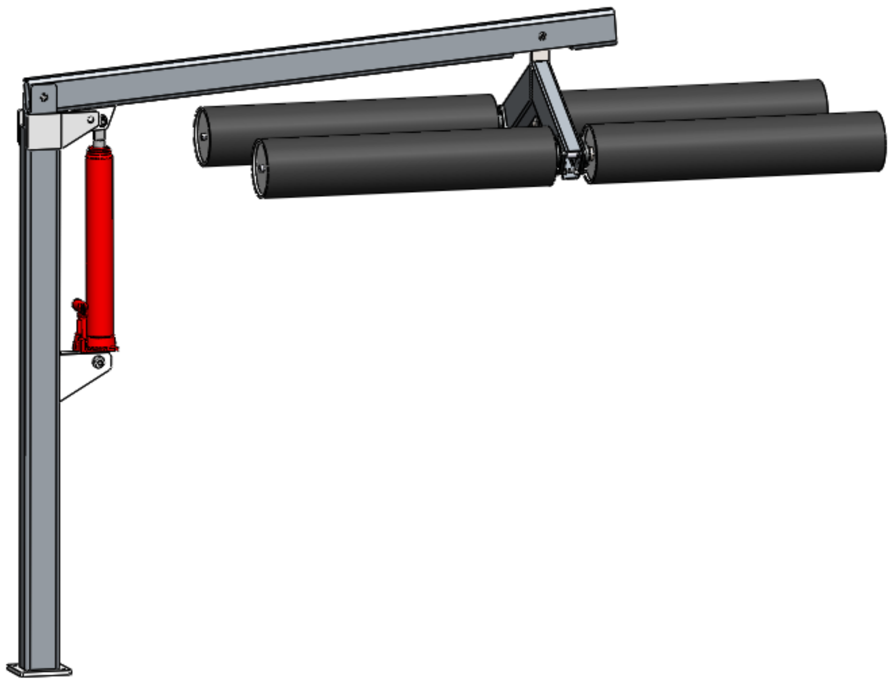

Figure 4-4: Top Roller assembly support structure [8].

Because the roll of composite material rests on the four Bottom Rollers with only its own weight, positive retention of the roll is not attained. Additional retainment of the roll of material is achieved using the retaining shaft assemblies. The retaining shafts consist of two short sections of square steel tubing spaced to fit to the inner diameter of the roll of material. These sections of tubing are welded to a steel Backing Plate. Gussets constructed of bent sheet metal provide structural rigidity to the Backing Plate, and are welded between the Square Shafts and Backing Plate. The Gussets form a 3” gap between each other, which fit around the Side Arm at one end of the cart and a vertical steel tubing section welded to the opposite end of the cart. Additional brackets are welded to the backing plate below the Gussets, which also fit around the Side Arm and provide a constraint against rotational motion of the Retaining Shaft subassembly. Flat sheet metal plates known as the Locking Plates are bolted to the Gussets and brackets, securing the Retaining Shafts to the Side Arm. The Retaining Shafts prevent the roll of composite material from escaping the cart should the roll be bumped, or the cart overturned. The Retaining Shafts are not truly shafts as they do not extend through the entire length of the roll of material, and do not support any load under normal conditions. Instead, the Retaining

Shafts can be thought of as emergency retention devices. Figure 4-5 shows the Retaining Shaft assembly.

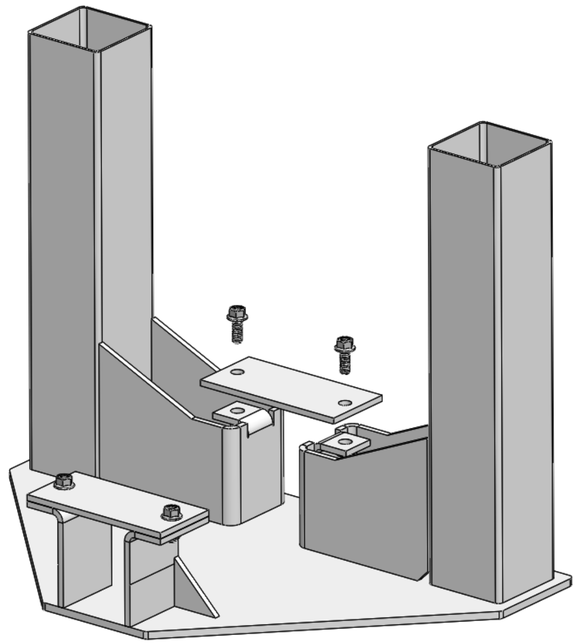

Figure 4-5: Retaining Shaft assembly [8].

Uncoiling and recoiling of the composite material is accomplished via one of two methods. The first method utilizes manual labor to turn the Bottom Rollers. Round steel tubing bent into a ring is welded to four straight sections of tubing, meeting in the center of the ring at a steel shaft coupling. This welded subassembly, known as the Bus Wheel, is attached to one Bottom Roller at each end of the cart via the shaft coupling. The Bus Wheel can be turned by hand to rotate the Bottom Rollers, which in turn rotate the roll of composite material. The rubber coating on the Bottom Rollers help to provide a high-friction surface to ensure that sufficient torque can be transferred between the Bottom Roller and the composite material. Roller-chain runs between equal sized sprockets on the shafts of the Bottom Rollers at each end of the cart. These chains cause the rollers to rotate in unison, providing additional tractive force to rotate the roll of material. Chain Guards constructed of bent sheet metal cover the chains at each end of the cart for increased safety.

The second method of uncoiling and recoiling the composite material is through the use of an electric motor. A 3 hp, 115 V, single phase motor is mounted to a gearbox, providing a 15:1 speed reduction via a worm gear. The gearbox is mounted to the Bottom Frame using two welded brackets. The gearbox is connected to one Bottom Roller via a chain drive with a 1:1 ratio. Since this Bottom Roller is connected to another Bottom Roller with the chain drive previously described, two rollers are effectively driven by the electric motor. The electric motor can be operated in both the forward and reverse directions, providing uncoiling and recoiling abilities with minimal operator input and the elimination of operator strain. The Bus Wheel and electric motor drive systems are shown in Figure 4-6.

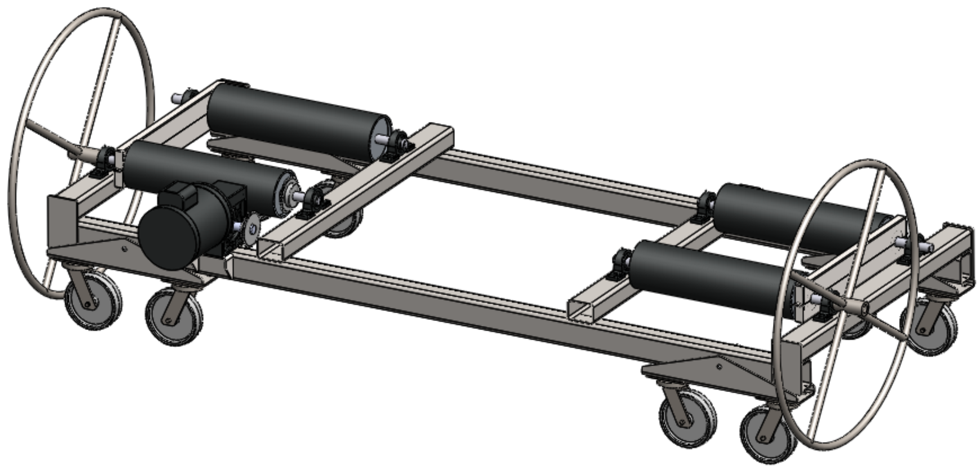

Figure 4-6: Bus Wheel and electric drive systems [8].

The Side Arm assembly is secured to the Bottom Frame assembly using four bolts and a steel plate, known as the Mounting Plate. The use of bolts rather than welding provides easier disassembly for transportation through tight spaces, and elimination of the Top Roller subassembly should it no longer be required. The bolt and Mounting Plate configuration also transfers the load of the Top Roller subassembly to the Bottom Frame more effectively than a weld.

A section of round steel tubing is welded horizontally to the Side Arm and the vertical tubing section on which the Retaining Shaft mounts. This tube serves two purposes. Firstly, the tube provides a handle for operators to securely grab and apply force to move the cart, improving safety by increasing ergonomics. Secondly, the tubing provides a stop for the Retaining Shafts should the cart be overturned.

A complete model of the final design in shown in Figure 4-7, including a full roll of composite material.

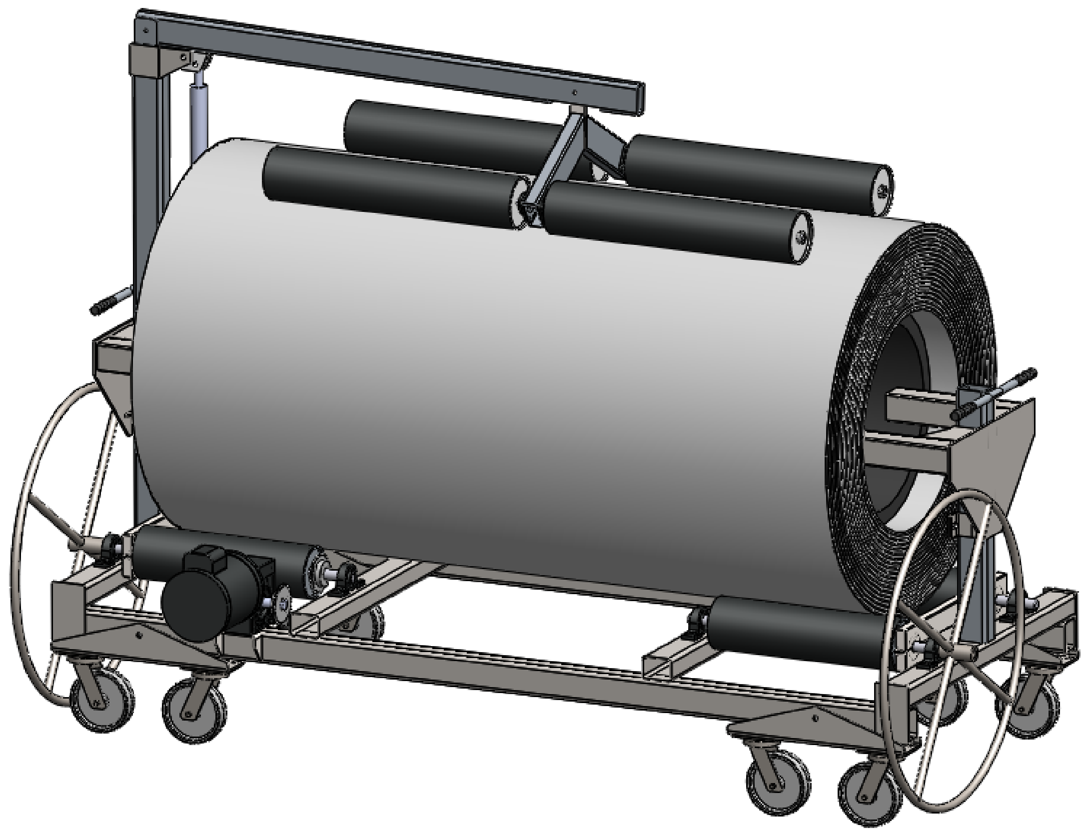

Figure 4-7: Composite Material Uncoiler final design [8].

4.2. Roll Exchange Procedure

This section provides a step-by-step procedure for preforming exchanges of the rolls of composite material, with the assumption that the procedure begins with an empty roll of material on the cart. The process outlined here represents the recommendations of the designers, and should be followed as close as possible.

1. Place the cart in an open area, away from any obstacles or low-hanging obstructions.
2. Lock the caster wheels using the integrated brakes or using wheel chocks.
3. Ensure all workers are clear of the cart.
4. Close the hydraulic cylinder valve and raise the Top Roller subassembly using the hand pump on the hydraulic cylinder.
5. Continue raising the Top Roller subassembly until the holes in the Safety Bracket are aligned with the holes in the Swinging Arm bracket.

6. Insert the safety pin through Safety Bracket and Swinging Arm bracket. Ensure the pin is through all brackets, and insert cotter pin through safety pin to lock safety pin in place.
7. While one individual supports the weight of the Retaining Shaft assembly, have another individual remove the four 3/8" bolts securing the Locking Plates to the Retaining Shaft.
8. Remove Retaining Shaft from cart and set aside.
9. Repeat steps 7 & 8 for second Retaining Shaft.
10. Using two individuals, remove cardboard roll core and set aside.
11. Using a forklift, pick up full roll of composite material.
12. Using one individual for guidance, carefully position roll of composite material above Bottom Rollers using forklift.
13. Gently lower roll of composite material onto Bottom Rollers, staying clear of cart while roll is being lowered.
14. Do not move the forklift once the roll of material has been lowered onto cart.
15. Position Retaining Shaft on Side Arm tubing with shafts inserted into roll of material.
16. Have second individual install Locking Plates using four 3/8" bolts. Tighten bolts to 30 ±5 ft-lbs.
17. Repeat steps 15 & 16 for second Retaining Shaft.
18. Remove the cotter pin from the safety pin, and remove pin from the Safety bracket.
19. Slowly open hydraulic cylinder valve until Top Rollers contact roll of composite material.
20. Slowly reverse forklift, ensuring forks do not contact composite material or cart.

4.3. Final Design Details of Subassemblies

Each primary functional subassembly is further outlined in detail below.

4.3.1. Top Rollers Subassembly

The top roller subassembly is designed to apply force via weight to the top of the composite roll such that roll expansion is constricted. As determined in Appendix C: Composite

Roll Expansion Force, the approximate force to constrain the composite material roll expansion is 32 lbs per layer. The resulting weight of the top roller subassembly of 285 lbs, exceeds the said prediction. Thus, the design should can hold approximately eight layers of material tight, if the roll is accidently let to expand.

The subassembly consists of four long rolling elements which are divided between on two long shafts. The outside tubing shells of each roller consists of centering plates which are welded together. The plates then act as the centering point of the long shafts which runs between all parts. On the back side of the assembly, the roller shell and plating are welded to the shaft directly, whereas on the other side, the roller shell and plates are retained by a retaining clamp. The shafts run concentric through two flanged bearing mounts each. The bearing housings are then mounted between two rectangular tube arms via fasteners. This assembly allows the rollers to easily be disassembled and reassembled in the case where a bearing may need to be replaced. From the shafts, the tubing arms are angled upwards and attached together via welding to an intermediate mounting plate. The mounting plate extends upward and contains a single through hole which pins the top roller subassembly to the swinging arm tube via a bushing. Figure 4-8 and Figure 4-9 display the front side and back side of the Top Roller subassembly, respectively. The specific sizing and dimensioning of the subassembly and individual parts can be found in Appendix A: Final Design Technical Engineering Drawings.

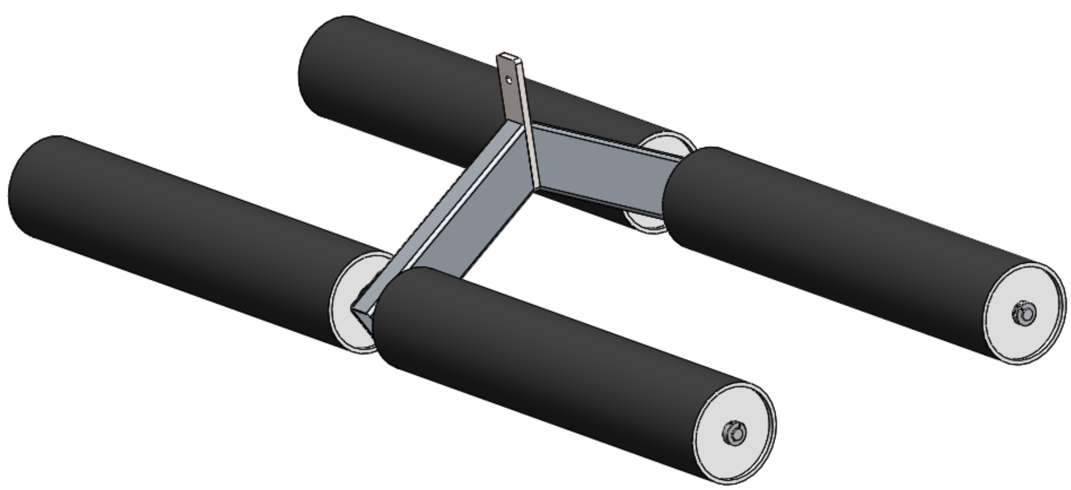

Figure 4-8: Front side isometric view of the top roller subassembly [9].

Figure 4-9: Back side isometric view of the top roller subassembly [9].

The round tubing selected for the roller shells are HSS (hollow structural sections) consisting of a 6.625” outside diameter with 0.28” thickness. The outside diameter of the tubing was selected to be as large as possible from the clients suggested metal supplier, Russel Metals, such that when coiling or uncoiling the composite material, the friction due to the deflection of the rollers is minimized. Through in-depth stress analysis for each part and welds between parts found in Appendix E: Stress Analysis and geometrical limitations, the dimensions of all components were determined. The thickness of the plates is set to 0.25” such that they approximately match the thickness of the roller tubing shells and proper weld penetration can be achieved. The diameter of the shaft was determined to be 0.75”, resulting in a stress factor of safety of 2.72. Thus, the longest shaft of this diameter possible, 72” was sourced from McMaster-Carr to cover the full length of the composite material roll as much as possible. The distance between each roller was then determined to be 23.19”, as shown in Figure 4-10. This separation provides the maximum spacing the rollers can be while maintaining approximately 10% interference on both sides of the composite material roll in its smallest state. Thus, the composite roll of material shall stay permanently centered between the roller throughout the entire use of a material roll.

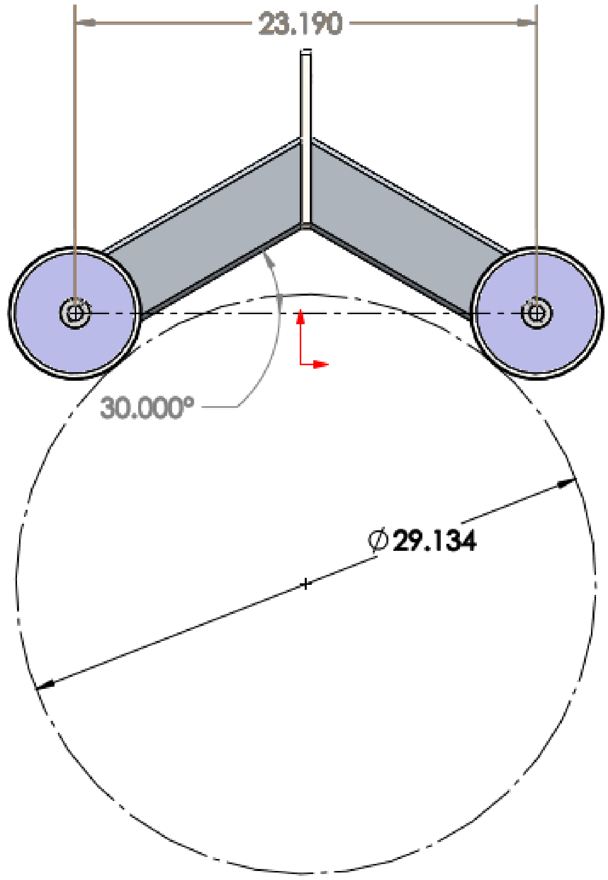

Figure 4-10: Spacing between top rollers [9].

To ensure the top rollers do not slip relative to the composite material roll during use, the team has specified that each roller should be rubber coated. Although the team was able to find commercial rubber coating products available on McMaster-Carr, these products should first be tested to ensure they will provide enough rolling friction and withstand the torsional forces being transferred between parts. If it is determined that this product is unsuitable, outsourcing of industrial level rubber coating must be done.

Next, flange bearing housing sourced from Fastenal (due to being approximately five times less expensive than McMaster-Carr) are used to fit the 0.75” shaft. Each bearing’s internal ring contains a set screw, to further ensure the shaft does not slip relative to motion of the composite material roll. Each bearing is rated for a static load of 1494 lbs, which far exceeds the load applied to them by the weight of the rollers and shaft of approximately 66 lbs each [10]. Small spacers, with an internal chamfer are fitted between the rollers and inner bearing ring. Thus, a small axial load can be applied from the roller centering plates to the bearing, without uneven interference due to weld radius on the shaft. Due to the size and spacing of the flange bearing, the minimum possible rectangular tubing available from Russel Metals of 4” x 2” x 0.125” thickness was chosen to mount each bearing housing. Additional holes are cut in the tubing arms such that the shaft may fit freely through the center, and mounting holes are aligned with the bearing housings. For each bearing, four sets of 7/16” bolts, washers, and nuts are used to mount the housing to the tube.

As seen in Figure 4-10, from the horizontal line made between both shaft, the tubing arms extend at 30° until they meet the hanging plate. The hanging plate is sized at 2” x 9” x 0.5” thickness. On the bottom four corners of the plate, edges are beveled to allow better welding penetration. At the top of plate, a through hole of 0.515” was used such that a bronze bushing with 0.5” outside diameter can be closely fit into the hole. Although the thickness chosen for this plate far exceeds to the strength requirements of a factor of safety of 3, it was chosen such that a reasonably strong weld may be placed between the plate and tubing and so angular play between this plate and the bushing it sits on is kept relatively low. Ultimately, the play between these parts may affect the overall centering of the top roller on the composite material roll, and thus additional spacers must also be used to axially locate the plate in the center of the swinging arm tubing.

4.3.2. Swinging Arm, Side Arm, and Hydraulic Lift Subassembly

The Top Swinging Arm applies constant pressure to the top of the composite roll using the Top Roller subassembly. The hydraulic lift assists in moving the top arm out of the way during changeovers, and allows the weight to be easily placed back down on the composite roll afterwards. The Side Arm supports both the hydraulic cylinder and the Top Swinging Arm.

This design was selected due to its ability to retain the material in a roll, ease of manufacturing, and mechanism safety compared to other designs. Another reason this design was selected was because it fits within the footprint of the design space. The implementation of this concept will aid in maintaining the material in a tightly wrapped roll, as well as help to improve the efficiency of the coiling and uncoiling process.

The team came up with several mechanisms for the support assembly. After a thorough discussion of ideas and research, the design that all members agreed upon was selected based on simplicity, manufacturability, cost, footprint of the design, and capability of supporting the Top Rollers subassembly. The selected subassembly consists of a Top Swinging Arm, Side Supporting Arm, and a hydraulic cylinder lift. The purpose of the entire top assembly of the model was to keep the composite material roll from falling off the bottom rollers, and to prevent roll expansion. Figure 4-11 shows the supporting mechanism subassembly, as the Swinging Arm rotates through the stages of the design. All tubing and fasteners are sized via the stress analysis outlined in Appendix E: Stress Analysis. Full details of each specific part can be found in the series of technical drawings in Appendix A: Final Design Technical Engineering Drawings.

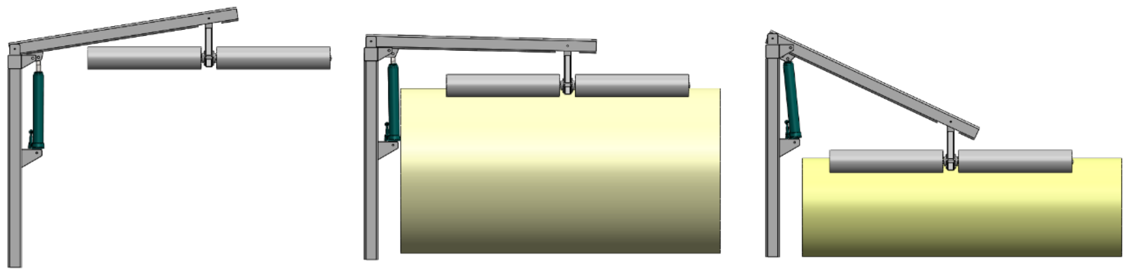

Figure 4-11: Front view and resolution of supporting mechanism [11].

Starting from the bottom of the subassembly, the side arm is a 66" long, 4" x 3" x 0.25" thickness structural tubing placed vertically on the mounting plate. Side Arm Brackets are welded to the Side Arm at a height of 29.5" from the bottom of the tubing. These brackets are pinned via a 5/8" bolt, nut, and washers to the clevis base of the hydraulic cylinder. The top edge of the Side Arm is connected to the Top Swinging Arm by a pin support, consisting of oil sleeve bushings mounted by a series of spacers, 3/8" bolt, nut, and washers. As the name suggests, the Top Swinging Arm is free to rotate up and down depending on the diameter of the roll. The Top Swinging Arm is made of a 4" x 2" x 0.125" thickness structural tubing, and is 68" long. The Top Roller subassembly is pinned by a similar 3/8" bolt and oil sleeve bushing assembly at the front of the Top Swinging Arm at a distance of 58" from the pin connection between the Top Swinging Arm and Side Arm. Additionally, a safety bracket is welded on each side of the top of the Side Arm, such that a safety pin can be placed during composite roll changeovers. After using the hydraulic cylinder to lift the Top Swinging Arm during a changeover, the pin provides an extra safety support for the Top Swinging Arm to rest upon is the unlikely scenario of a hydraulic fluid leak. This subassembly also consists of a Top Arm Bracket for the connection of the top of the hydraulic cylinder to the Top Swinging Arm. Both hydraulic supporting brackets on the assembly are made of 0.5" thick steel plating to safely resist the stress due to the hydraulic arm. The distance between the pin connection of the Side Arm and Top Swinging Arm to the hydraulic cylinder was set to 9". As further explained in Appendix E: Stress Analysis, this results in a hydraulic lift force of 3000 lbs, or half of the rated capacity of the sourced hydraulic lift from Northern Tool, while keeping the hydraulic cylinder as close to the Side Arm and as horizontal as possible. The exploded view of the supporting mechanism subassembly is shown in Figure 4-12.

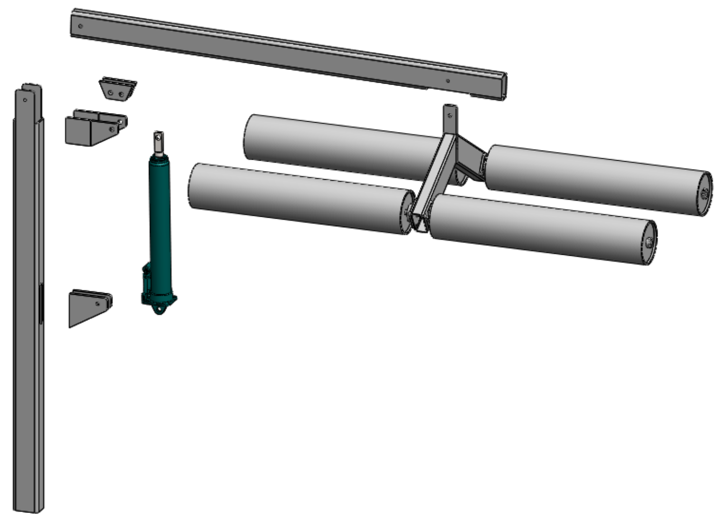

Figure 4-12: Exploded view of the supporting mechanism for the Top Roller Subassembly [11].

4.3.3. Bottom Frame Subassembly

The selected Bottom Frame subassembly consists of four Cross Supports placed on top of two Long Supports, as shown in Figure 4-13. The Long Supports are 116” long, whereas the Cross Supports are 48” long. Both Long Supports and Cross Supports are made with 4” x 2” x 0.125” thickness structural tubing. A gap of 44” between the two center Cross Supports allows for easy roll exchange using a forklift. The Long Supports are welded to Cross Supports at every connection using a typical weld pattern; the stress analysis of all the welds can be seen in Appendix E: Stress Analysis. Each cross support has two bolted pillow block bearing housings to support the Bottom Rollers which will be further discussed in the next section. The bearing housings are source from Fastenal, and are 6.50” in length, 3.65” in height, and are produced for 1.25” shaft diameters. This design is selected due to its ability to support the weight of the roll and all parts of the design fixed to the roll, safety, ease of roll exchange using a forklift, and manufacturability. Furthermore, the simplicity of the design allows easy assembly and maintenance. Details of all subassembly components are outlined in the technical drawings presented in Appendix A: Final Design Technical Engineering Drawings.

The design process of the bottom frame was based on its load capacity, size of the footprint, ease of roll exchange using forklift, and manufacturability. At first, the team decided to place all the Cross Supports in between the Long Supports, thus putting them all at the same height. However, this design interfered with the roll exchange process of using a forklift, as the Bottom Rollers were not high enough for a new roll of composite material to be placed back on the cart without the forklift hitting the Long Supports. Therefore, the team decided to place the Cross Supports on top of the Long Supports. This alteration in the bottom frame design provided an extra 3” of clearance for the forklift which was judged as acceptable, after placing the composite material on the Bottom Rollers.

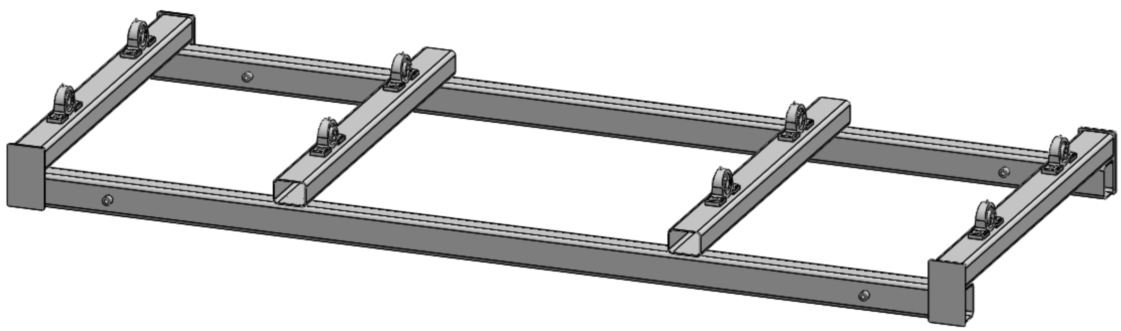

Figure 4-13: Isometric view of the Bottom Frame [11].

4.3.4. Bottom Rollers Subassembly

The idea of placing rollers underneath the composite roll was particularly liked by the client. Therefore, the team came up with various ideas of placing rollers on the bottom frame. In the design process, numerous designs were brainstormed which included rollers extending the length of the roll supported by bearing housings, spring supported rollers extending the length of the roll, and multiple roller designs leaving space at the center to allow for roll exchange using forklift. There was not much room for experimentation in the rollers length, since the location of cross supports was fixed from the bottom frame design. Therefore, the majority of the time in designing the bottom rollers was spent on determining the desirable diameter of the bottom rollers.

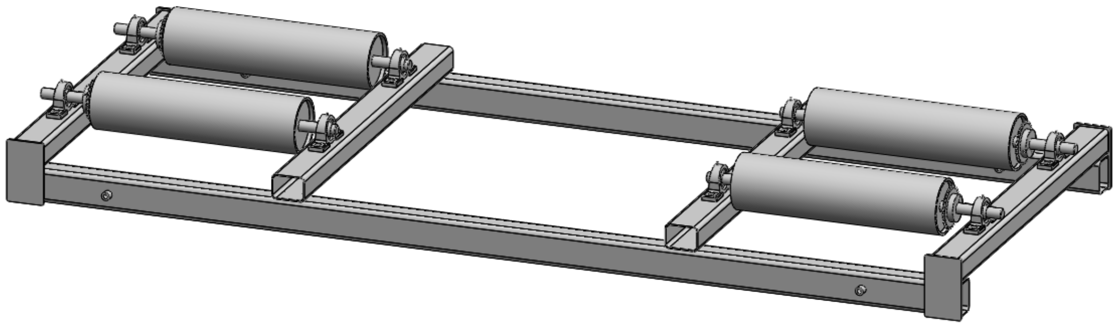

Figure 4-14: Isometric View of the Bottom Frame and Bottom Rollers subassembly [11].

After a thorough discussion, the team decided to use a four-roller design supported by pillow block bearing housings, as shown in Figure 4-14. The Bottom Rollers are primarily based off the same welded assembly of the Top Rollers, but with different sizing of each element. The rollers sit on their respective shafts which are supported by bearing fixtures located on the Cross Supports. Each roller is 25” long and 6.625” in diameter. Additionally, each shaft is 36” long and 1.25” in diameter which were determined via stress analysis outlined in Appendix E: Stress Analysis. As with the Top Rollers, Bottom Rollers are rubber coated to allow for easier coiling and uncoiling processes. The adjacent rollers are 23.67” apart, center to center such that the minimum roll diameter size of composite material will not fall through the framing as it is used.

This design was selected mainly due to its ability to support the roll and to reduce the coiling and uncoiling effort. The implementation of bottom rollers helps eliminate the need for a supporting mechanism through the center of the composite roll, since the rollers can support the weight of the composite material roll. In addition, the subassembly also provides easy roll exchange using a forklift, prevents roll expansion, and gives a solid base to the overall design.

4.3.5. Walking Axles Subassembly

First the team selected eight high capacity vulcanized steel swivel casters with diameters of 8” from McMaster-Carr due to their supporting load of 1400 lbs per caster, and 2” wheel width. Next, the design of the Walking Axles began by considering its construction. Sheet metal, bent into the shape of a c-channel, was chosen over a welded assembly for its simpler construction. Next, the swivel caster mounting plate was considered. The selected caster wheels mount using four 1/2” bolts in a rectangular pattern, 5-1/4” by 3-3/8”. Given that 1/2” bolts have a 3/4” head, the minimum spacing between the inside faces of the flanges is 4.125“. Since the required width of the Walking Axle is wider than the long support member to which it will be mounted, spacers are required to reduce compliance in the bolted joint. Low density polyethylene (LDPE) spacers were sourced from McMaster-Carr, with a length of 3/4”. LDPE spacers are adequate in this situation, as the spacers do not take any load, they are simply required to center the Walking Axle on the long support member. The addition of the LDPE spacers resulted in a spacing between the flanges of 4.5”, providing adequate space for bolt heads.

Next, the length of the Walking Axle was considered. Sufficient spacing between the two casters is required, such that the wheels do not contact if they are both oriented towards each other. Given that the casters have a swivel radius of 6-1/4”, the minimum spacing between swivel axes is 12-1/2”. It was decided that the caster wheels should have a minimum spacing of 4-1/2” at any orientation, bringing the swivel axis spacing to 17”.

The height of the mounting hole was chosen based on having a minimum of 3/4” spacing between the bolt head and the bottom of the long support frame under operating conditions. This spacing would give the Walking Axle an adequate range of motion to ensure that weight is distributed between all wheels at all times.

With the exterior dimensions for the Walking Axle determined, the final step was to determine an appropriate mounting bolt size. It is known that the bolt must support one quarter of the weight of the frame and roll of material, 5500 lbs, in double shear. Since the Walking Axle pivots directly on the bolt, wear on the bolt is expected. For this reason, a factor of safety of 10 was desired. Using a shear strength of 65 ksi for a grade 8 bolt, a diameter of 1/2” was selected. FEA was conducted on the final design, indicating a factor of safety of 5.26, well above the target value of 3.0. A full explanation of the analysis can be found in Appendix E: Stress Analysis.

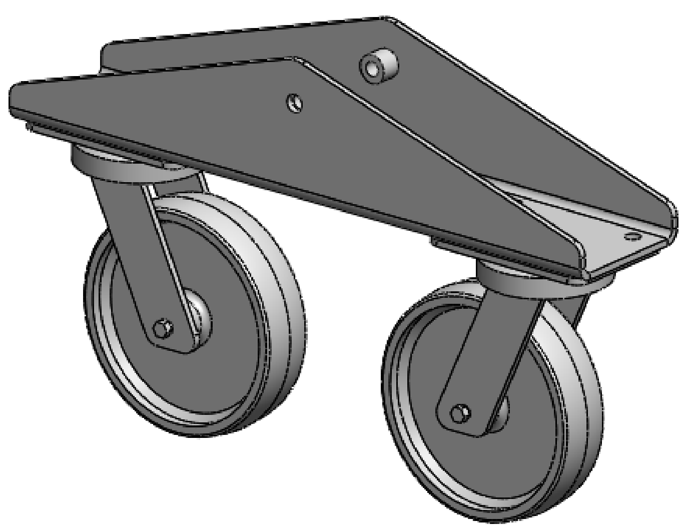

Figure 4-15: Isometric View of the Walking Axles subassembly [9].

4.3.6. Retaining Shaft Subassembly

The design of the Retaining Shafts began with selecting steel tubes to be used as the two shafts. From the dimensions of the frame, it was known that roll of composite material would contact the shafts 12” from the backing plate. Assuming that the backing plate is rigid, the shaft can be analyzed as a cantilever beam. With a load equal to 2150 lbs (half of the weight of the roll of material) applied at 12”, a maximum bending moment of 25800 lbs-in. An excel spreadsheet was created using available circular tube sizes from Russel Metals, calculating the moment of inertia of each size, and thus the maximum stress in the tube. Using this spreadsheet, a tube which attained a factor of safety of three was selected for the first iteration of the Retaining Shaft design. A 3D model of the backing plate and shafts was then created for FEA, and can be seen in Figure 4-16.

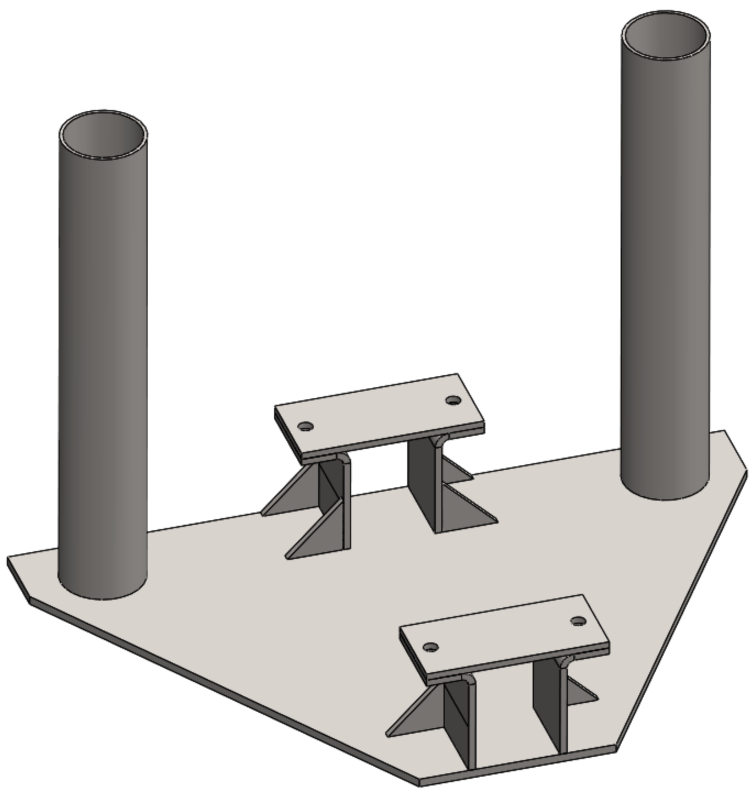

Figure 4-16: Initial Retaining Shaft assembly [8].

Upon analysis, it was found that the backing plate would fail under the expected loading scenario. It was clear that gussets between the shaft and backing plate were required to provide stiffness and strength to the backing plate.

To provide a flat surface to weld the gussets to the shaft, the shaft was changed to a square tube. An excel spreadsheet was once again used to select a tube which achieved a minimum factor of safety of 3.0. The selected tube has external dimensions of 3.5” x 3.5” x 0.125” thickness, resulting in a factor of safety of 3.25 based on the cantilever beam calculation. Gussets were then designed from 0.25” bend sheet metal, with integrated mounting to the vertical frame. The final assembly can be seen in Figure 4-17.

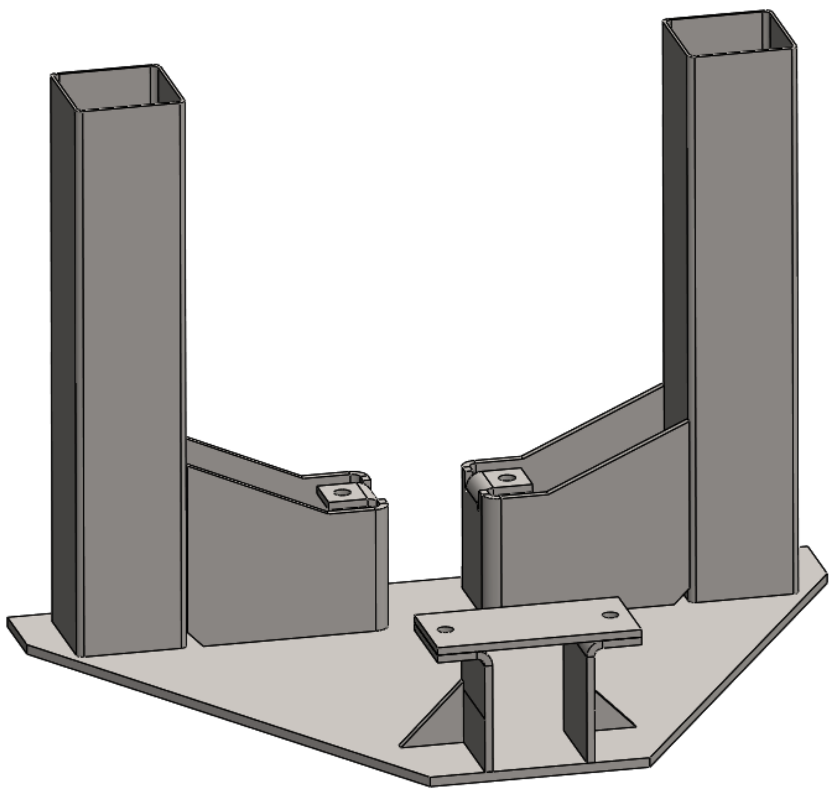

Figure 4-17: Final Retaining Shaft assembly [8].

By integrating the gussets with the mounting to the vertical frame member, the force from the shaft can be transferred to the vertical frame member. This minimizes the bending moment induced on the backing plate, which had a relatively low geometric stiffness. The dimensions of the gussets were determined through iterative design, finite element analysis, and adjustment.

To increase the stability of the Retaining Shaft assembly, a second mounting location to the vertical frame member was created, 10” from the shaft gussets. This second mount was also constructed of 0.25” bent sheet metal, welded to the backing plate. Dimensions of all final Retaining Shaft parts can be found in Appendix A: Final Design Technical Engineering Drawings.

With all dimensions of the Retaining Shaft assembly determined, a solid bodied FEA model was created with identical geometry. This model was used for a more thorough finite element analysis than was used when determining geometry. The results of the analysis indicated a maximum stress of 19.34 ksi, resulting in a factor of safety of 3.0.

4.3.7. Drive Systems

The different drive systems including the final design's Handle Bar design, Bus Wheel design, and Electric motor system design are outlined below.

Handle Bar Design

During the team's site visit with the client, the team found the current composite material cart is extremely difficult to be moved. One reason for this is because the current cart has no ideal position for an operator to push or pull on. Currently, the operators can only move the composite material cart by pushing or pulling on a vertical tubing frame support or on the composite roll directly. Pushing or pulling on the cart in these unstable positions may easily cause more strain than necessary, and injure the operators over time.

As safety is a primary concern of the design, the team decided to design a handle bar on both sides of the cart, such that moving the cart is more safe and convenient. Theoretically, by setting a horizontal handle bar at the average men's elbow height from the ground, strain on the operator is minimized as outlined further in Appendix D: Final Design Safety and Kinematics. Based on the maximum height location of the retaining shaft, the position of the handle bar must sit higher than 46.465" from the ground. Therefore, two handle bars were decided to be welded at the 50" from the ground on the Side Arm Tubing Support and Welded Side Arm Tubing. From the teams maximum pushing force test, also outlined in Appendix D: Final Design Safety and Kinematics, the rough maximum pushing force of an individual male is around 125 lbs. Thus, this maximum force was used to evaluate the strength of the Handle Bars. The final design for the Handle Bars are made of ASTM A-500 Gr. B cylindrical tube. The length of the handle bar is 22", which is the approximate shoulder width of the average man, such that the operators may easily push, pull, and rotate the cart. The outer diameter of the handle bar is 1" with a thickness of 0.133", allowing the average man to easily grip the sourced Handle Bar Grips from McMaster-Carr which fit over it. By stress analysis outlined in Appendix E: Stress Analysis, the maximum stress on the handle bar is 6.3 ksi, which results in a large factor of safety of 6.57. Figure 4-18 and Figure 4-19 show the handle bars on both sides of the cart.

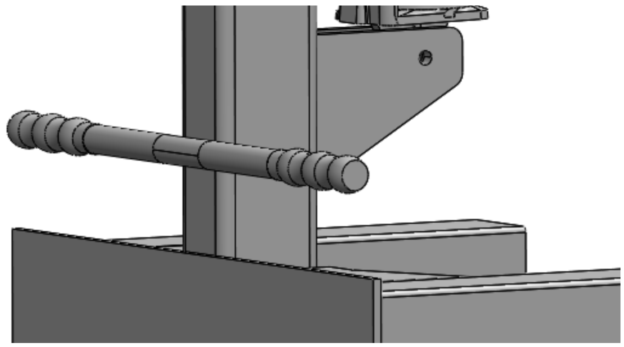

Figure 4-18: Handle Bar on Side Arm Support tubing [12].

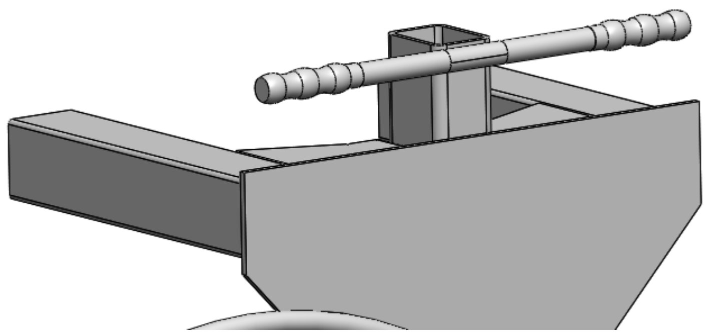

Figure 4-19: Handle Bar on Welded Side Arm tubing [12].

Bus Wheel Design

To manually drive the composite material roll, the bottom roller must be driven by an external force, such as a crank or large “Bus Wheel” design. Comparing these two items, the team found the Bus Wheel will likely perform better and safer. With a Bus Wheel, the operator can rotate the wheel by pulling from a consistent position while standing perpendicular to the Bus Wheel. This motion produces much less strain on an operator instead of moving up and down cyclically, which would be necessary if using a crank, and thus could reduce the relative back injuries of employees. The Bus Wheel design developed is shown in Figure 4-20.

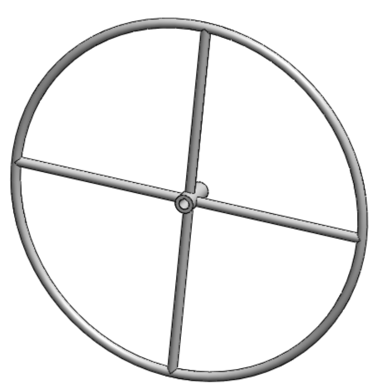

Figure 4-20: Bus Wheel design [12].

From the video the team took during the site visit at Triple E RV, the operating speed of the composite material roll was approximately one rotation over three seconds, or around 20 rpm. Due to the Bottom Roller's diameter previously chosen to be 6.625", the no-slip speed ratio to the composite roll can be calculated as 7.31. Thus, the resultant bottom roller speed is 146.2 rpm, which is not feasible by only rotating the bus wheel alone. However, operators may extract the composite material sheets by using the gripped sheet material pulling tools that are currently being used. Once the sheet is cut and processing of the sheet begins, the operators can use the downtime to recoil the material at the slower and safer rate produced by turning the bus wheel. The space between Bottom Roller's shaft and the ground is 20.266". Thus, the maximum outer radius of the Bus Wheel that can be implemented is 17", accounting for additional clearance of the caster floor tracks mounted to the ground. The team decided to use the full 17" as the outer radius of Bus Wheel, since the larger radius provides a mechanical advantage requiring less force to rotate. The Bus Wheel is made by welding one shaft coupler, four connector tubes and a rolled ring tube together. All formed tubing is made from the same 1" outside diameter, and 0.133" thickness stock size and materials as was used on the Handle Bars. A shaft coupler sourced from McMaster-Carr was selected with a 1/2" keyway used for connecting the Bus Wheel to the Bottom Roller shafts. Using the 75 lbs safe pulling force per person determined in Appendix D: Final Design Safety and Kinematics, assuming two people operating two Bus Wheels on each side of cart together, the rotating speed of the Bottom Roller will achieve to 30 rpm within 0.3 seconds. This results in an output rotating speed of the composite material of 4.06 rpm.

Electric Drive System Design and Specifications

The design process for the electric drive system began with a kinematic analysis of the uncoiling process. By observing the current process, it was found that the roll is uncoiled at approximately rate of one revolution every three seconds. This equates to an uncoiling speed of 20 rpm, or 2.09 rad/s. It is known that a full roll of composite material weight approximately 4300 lbs, and has an outer diameter of 1.24 m, an inner diameter of 0.7 m, and a length of 2.4 m. Using these values, the density of the composite material can be approximated as 972.2 kg/m^3. The torque required to angularly accelerate the roll of material is defined by its polar mass moment of inertia and can be calculated as follows.

$$J_0 = \frac{\rho \pi L({D_o}^4 - {D_i}^4)}{32} = \frac{972.2 * \pi * 2.4 * (1.24^4 - 0.7^4)}{32} = 486.6\ kg * m^2 \qquad (4\text{-}1)$$

Where J_0 is the polar mass moment of inertia of the roll about its axis, ρ is the density, and L is the length of the composite roll. This value does not account for any friction in the system, or any energy required to overcome deformation of the material. Since these values are extremely difficult to quantify, they are accounted for by multiplying the polar mass moment of inertia by three, resulting in an equivalent moment of inertia of 1459.7 kg·m^2.

It was determined that the drive system should be capable of bringing the roll of material to its full speed within three seconds. This results in a desired acceleration of 0.697 rad/s^2. The torque and power required to achieve this acceleration is calculated as follows.

$$T = J_0 * \alpha = 1459.7 * 0.697 = 1016.9\ Nm \tag{4-2}$$

$$P = T * \omega = 1016.9 * 2.09 = 2125.4\ W = 2.84\ hp \tag{4-3}$$

Where T is the required torque, α is the angular acceleration, and ω is the desired angular velocity.

With this information, an electric motor could be selected. Since the desired uncoiling speed is relatively low, selecting a motor with a low operating speed is beneficial, as it would require a lower gear reduction and possibly reduce gearbox costs. Electric motors are generally offered at 1750 rpm and 3450 rpm operating speeds. A 3 hp, 1750 rpm, 115 V electric motor was selected for use in the electric drive system. The selected electric motor is manufactured by Leeson, part number 131544, and is distributed via Electric Motor Warehouse.

The next step in the design was to achieve the desired rate of uncoiling. Given that the driving rollers and the roll of composite material have different diameters, a reduction in rotational speed will be present between the two, equal to the ratio of their diameters. The selected rollers have a diameter of 6.625", and the roll of material has an average diameter of 38". This produces a speed reduction of 5.74. If this reduction alone were to be used, the roll of material would have an average rotational speed of 1750/5.74=305 rpm. Clearly this is much higher than the desired uncoiling speed. The additional required speed reduction can be calculated by dividing the previously calculated rotational speed by the desired rotational speed.

$$r = \frac{\omega_{calculated}}{\omega_{desired}} = \frac{305}{20} = 15.25 \tag{4-4}$$

Using this value of desired gear reduction, a gearbox could be selected. Due to the high reduction required, a worm gear was selected as the method of reduction. The selected gearbox is manufactured by Superior Gearbox Company, part number BMQ075-180TC.

With the desired output rotational speed attained, a method of transferring torque to the bottom rollers needed to be designed. Since the worm gear box has a high torque-low rpm output, a chain drive to the Bottom Rollers was determined to be optimal. Chain and sprocket sizes were selected using the maximum working load values provided by Oriental Chain Mfg. These values state the maximum tension that various sizes of roller chain should be subjected to under working conditions. Many chain manufacturing companies provide horsepower tables for selecting appropriate chain sizes based on an expected life of approximately 15000 hours. Since the drive system will be operated for short periods of time, few times per day, basing the chain size on such a lifespan is not an appropriate method. Therefore, the maximum working load method was used instead.

To begin, an excel spreadsheet was created, calculating the minimum radius of drive sprocket required to transfer the torque output by the gearbox based on a variety of chain sizes and their respective maximum working loads. The required radius of sprocket was calculated by dividing the torque output of the gearbox by the safe working load of the chain, equal to the maximum working load divided by a factor of safety of two. Given that the bottom rollers have an outer diameter of 6.625", the diameter of the sprocket must be less than this value to avoid the roll of composite material being damaged by the chain and sprocket. It was found that an ANSI 40 chain, which has a maximum working load of 860 lbs, required a minimum diameter of 4.67" to transfer the torque output by the gearbox with a factor of safety of two [13]. Based on this diameter, an ANSI 40 sprocket with 30 teeth was selected, having a pitch diameter of 4.78". These sprockets, with the appropriate size chain, will be used to transfer torque both from the gearbox to the bottom roller, and for the synchronization system between the bottom rollers at either end of the frame.

4.4. Final Design Summary

The overall specifications of the team's composite material uncoiler final design are summarized in the following sections.

4.4.1. Fully Integrated Design Specifications

The composite roll fixture design satisfies the project specifications provided by the Triple E RV. Every component of the final design went through a rigorous design process followed by a series of analysis. By doing a thorough stress, safety, kinematics, and cost analysis, the team was able to determine the desired specifications of every component of the final design. The final design consists of several sub-assemblies which include: Top Rollers, a support mechanism consisting of a Top Swinging Arm, Side Supporting Arm, hydraulic lifting cylinder, Bottom Framing, Bottom Rollers, Walking Axles, Handle Bars, Retaining Shafts, and manual or automatic drive systems, Bus Wheels or electric motor, respectively. The selection of all the design components were based on key design and operation factors such as the manufacturability, safety, ease of use, cost, and stress analysis.

To examine the safety and strength of the design components, a thorough stress analysis was performed on all the components of the design. This was done through SolidWorks Simulation FEA and was also verified through analytical analysis wherever possible. The stress levels of all components exceeded the initial team goal of a factor of safety of 2.5 or higher. The stress analysis of all the components can be seen in Appendix E: Stress Analysis.

Throughout the design process, the team aimed at minimizing the weight and cost of the overall design. The weight of the entire assembly without the roll of composite material is approximately 1200 lbs, or 5500 lbs with the full roll of composite material. The overall cost of the design was determined by the sum of the design's cost breakdown into sourced parts, raw materials, and approximated production costs as explained in Appendix F: Final Design Cost Analysis. The total cost of the design including sourced parts, raw materials, and approximated production were determined to be $5072.23 USD.

4.4.2. Final Design Comparison with Project Goals

This section compares the final design specifications to the ideal and acceptable targets of the initial goals set by the team through an in-depth Needs Analysis. The results of the design specifications can be seen in Table 4-I.

Table 4-I: Final Design Specifications Compared to Initial Needs Analysis

Definition of Need	Importance Rating*	Measured Quality	Metric	Ideal Target	Acceptable Target	Final Design Specification**	Final Design Specification Notes
Physical Requirements							
The design supports the mass of the roll of material	⊙	Stress levels not to exceed yield limits	Factor of safety	2.5-3	2.5-5	min = 2.72 max > 5	All parts meet or far exceed stress factor of safety requirements.
The material is constrained from unwanted uncoiling	⊙	Ratio actual diameter to tightly wrapped diameter	-	1	<1.25	1	The composite material is fully constrained from expansion.
The design secures the roll of material	⊙	Difficulty to remove roll with restraint engaged	Qualitative (1-5)	5	3	5	The composite material roll is positively retained, even in the case of a flipped cart.
The design utilizes existing floor tracks	⊙	Utilization of floor tracks is possible	Binary	1	1	1	The design fits within the footprint of the floor tracks.
The design fits within the existing work station	⊙	Length	Feet	8	<9.5	9.667	Although the footprint of the design is slightly outside the initial specifications, it will still easily function within the work space provided. The increased size was necessary to meet all other design goals.
	⊙	Width	Feet	4	<5	4	
	○	Height	Feet	5	<6	7.25	
The design aligns the roll of material with the next work station	○	Maximum allowable angle of misalignment	Degrees	<1°	<5°	<1°	Due to the shaftless assembly, the composite material roll self aligns between rolls and material is automatically square to the production table.
Manufacturing and Cost Related Needs							
The design is produced using manufacturing methods available to the client	○	Percent of outsourced manufacturing	%	0	<20%	0	No parts of the design need to be outsourced for manufacturing.
The design requires minimal purchased parts	○	Percentage of total cost	%	<10%	<25%	51%	Although the cost percentages do not agree with the initial goal, overall they are acceptable. This is because for the purchased parts category, the worst case scenario of sourcing all standard parts from McMaster-Carr and the inclusion of all automation costs. The total cost of the sourced parts may be cut up to 40% by strategically purchasing parts, as outlined in Appendix F: Final Design Cost Analysis.
The design requires minimal labor cost to manufacture	Δ	Percentage of total cost	%	~50%	<65%	31%	
The design uses minimal raw material	Δ	Percentage of total cost	%	~40%	<50%	18%	
Process and Design Use Needs							
The design is maneuverable	⊙	Force required to maneuver	Newtons	<2X single person push force	<3X single person push force	2X	The design is capable of being maneuvered by two persons.
The design is able to recoil material	⊙	Yes or No	Binary	1	1	1	The design is capable of recoiling material.
The material is able to be removed and installed with a forklift	⊙	Yes or No	Binary	1	1	1	The composite material can easily be installed by forklift.
The design is stable after transportation	⊙	Ability to fix design in place	Binary	1	0	1	The design consists of two swivel casters with locks such that the cart may be fixed in place when not being moved.
The design is able to be moved by minimal amount of people	○	Number of people required	People	<2	<=3	2	The design is capable of being moved by two people.
The roll of material requires minimal manpower to exchange	Δ	Number of people required	People	1	2	2	The design requires one person to use a forklift, and another person to help align the roll onto the cart.
The material can be exchanged in a timely manner	Δ	Time required	Minutes	<30 minutes	<1.5 hours	<30	The shaftless design changeover process is greatly simplify, and should easily take less than 30 minutes,
Product Lifecycle Needs							
The design requires little maintenance over its life	○	Maintenance period	Years	<5	<2	<5	Barring faulty parts such as bearings, bearings and bushes should easily last more than 5 years.
The design provides easy access to serviceable items	○	Disassembly time	Minutes	<10 minutes	<20 minutes	<10 minutes	All items that may need to be serviced are easily accessible, requiring one or two bolts to be removed at most.
The design lasts a long time	Δ	Functional life	Years	>10	>5	>10	Given preventive maintenance is performed, the design shall not prematurely fail.
Safety Needs							
The design adheres to relevant safety standard	⊙	Yes or No	Binary	1	1	1	The design adheres to all mechanical safety standards. Safe working forces may need to be exceeded when the composite material roll is full, without using the motorized design.
Automation Needs							
The design uses an external energy source to uncoil and recoil the material	○	Yes or No	Binary	1	0	1/0	The design can use both manual labor and electrically outlet power to operate.
The design can operate using either manpower or external power	○	Cost to implement external power	$	450 (cost of motor)	900 (2x motor)	$700	The total cost to implement external power is the approximate cost of the electric motor and gearbox required.
	○	Time to switch between power methods (after installed)	Minutes	<10	<30	<10	The motor or bus wheel may easily be installed due to the use of chain drives or keyways, respectively.

Notes: *Δ = weak, ○ = medium, ⊙ = strong. **Green = Ideal Achieved, Yellow = Acceptable Achieved, Red = Outside Initial Specifications

To determine the success of the final design, each need was evaluated against its corresponding final design specification. This was done to provide a clear view of the design specifications that were satisfied by the final design, and any initial targets that the final design failed to satisfy. The results of the comparison can be seen in Table 4-I.

A total of 26 design needs were developed during the initial phase of the project. Throughout the design process, the team attempted various alternative approaches to meet all 26 established needs. Out of all the design needs evaluated, 18 were satisfied ideally (in green), 6 were satisfied with reasonable target values (in yellow), and 3 were slightly outside the set targets (in red). Thus, resulting in an 89% success rate in satisfying the initial needs.

Two of design specifications that fall outside the initial specification limits are the length and height of the composite roller fixture. The length of the cart design is 0.167 ft over the acceptable target limit of 9.5 ft. The failure of not meeting the length specification of the cart should not hamper with the other functions of the cart such as coiling, uncoiling, stability during use, and safety. The height of the cart design is 1.25 ft over the acceptable target limit of 6 ft. Again, the minor height difference should not hinder with the cart's own functions and other processes in the facility. In addition, there is ample space available to allow easy roll exchanges even when the hydraulic cylinder is fully extended. Therefore, both the length and height extension of the cart will not create any problems, and will allow the cart to function at its full potential. It should be noted that the team deemed it necessary to violated these requirements, as it allowed the majority of the other, more important requirements to be achieved.

The last design specification that fell outside the initial specification limits is the relative cost of sourced parts compared to the total cost of the entire design. This is due to the presented sourced parts cost is taken as a worst-case scenario. For the majority of standard off the shelf parts such as bolts, nuts, and washers, McMaster-Carr was used as the vendor. It should be noted that the team assumed that Triple E RV would need to purchase a full package of a standard part, even when the full package would not be used. Additionally, the prices listed by McMaster-Carr are often significantly more than what actual costs may be, assuming Triple E RV has an on-going account set up with them to achieve better prices. It should also be noted that the automated drive system parts are included in the sourced parts list. Although the team strongly recommends the use of the automated drive system, these parts account for approximately $700 USD in total, and thus if omitted, sourced parts costs could be cut by approximately 30%. Thus, the percentage of sourced parts cost, relative to the final design costs can significantly be reduced from the presented value.

5. Project Summary and Recommendations

The following sections present the summary of all work done by the team during the project, as well as the recommendations the team made for the client to move forward into production of the final design.

5.1. Project Summary

Triple E RV requires a composite roll fixture for walls and floor manufacturing process, that is more user-friendly than the current frame assembly being used. The current process is viable, but it is very time-consuming, laborious, and inefficient. The purpose of the composite roll fixture is to support a weight of over 5000 lbs, allowing easy coiling and uncoiling of the composite material. The new design should also be maneuverable, durable, safe to use, and within other customer specification limits. In addition, the design should also retain the material horizontally, allow easy changeovers of the roll using a forklift, and constrain the composite material from expanding to reduce coiling and uncoiling effort.

To design the composite roll fixture, the team followed a three-phase design process taught in the Engineering Design course at the University of Manitoba. The design process included Project Definition as phase I, Concept Design as phase II, and the Final Design as phase III. Each phase of the process was carried out in a detailed manner in accordance with the client's specifications and requirements.

During the initial phase of the project, the team defined the problem statement and scope of the project. To get a clear understanding of the problem, the team observed and analyzed the composite uncoiling process completed by Triple E RV. Through the IDEA program application and client feedback, the team identified seven objectives and four constraints of the project. Each objective was then analyzed and specified measurable specifications or qualities using a needs analysis methodology. This was done to establish a means of determining the success of the design by creating general needs using project objectives. Furthermore, the team also identified the project deliverables, and created a project breakdown which included project milestones, work breakdown structure, and a Gantt chart.

The team began the secondary phase by doing research on material coiling machines, roll retaining, and safety mechanisms to help generate ideas for the project. After researching on relatively similar concepts required for the rolling fixture design, the team brainstormed a total of 37 ideas based on the desired specifications of the project. These ideas were then screened based on ability to support the composite roll, ability to constrain expansion of the roll, ability to reduce movement effort, and ability to reduce coiling and uncoiling effort. The selected concepts form the screening process were showcased and explained to the client. Through client's recommendations and concerns over the designs portrayed, the team generated 19 more alternatives in secondary concept generation by combining previously chosen concepts and new ideas together. The newly generated ideas were based on four primary categories which include ability to constrain expansion of composite material, ability to positively retain the composite roll, ability to reduce movement effort, and ability to reduce coiling and uncoiling effort. The generated designs in each of the four categories went through a scoring process to determine the optimal concept for each aspect. Following the concept scoring process, the selected concepts were then integrated to determine the final conceptual design.

The team began the third phase by making minor changes to the selected conceptual design. After finalizing the selected design, the team did a thorough stress, safety, kinematic, and cost analysis to ensure that the final design meets the desired requirement. The components of the design were analyzed using finite element analysis (FEA) with SolidWorks Simulation, which were verified using analytical solutions wherever possible. Following the analysis, the final engineering technical drawings, bill of materials, and a list of recommendations were developed for the client.

To conclude, the project was completed successfully as the final design meets the desired specifications and requirements. The total cost of the entire design including the sourced parts, raw materials, and approximated production costs came out to be $5072.23 USD. The final design specifications comply with the initial project needs set by the team during phase I. The cart is designed in a way that it will constrain the composite material, retain the roll horizontally, and allow easy coiling and uncoiling of the composite roll. In addition, the development of the new design will help workers avoid injuries and improve efficiencies regarding roll changeover and the material extraction process. The final design of the composite material uncoiler developed by the team is shown in Figure 5-1.

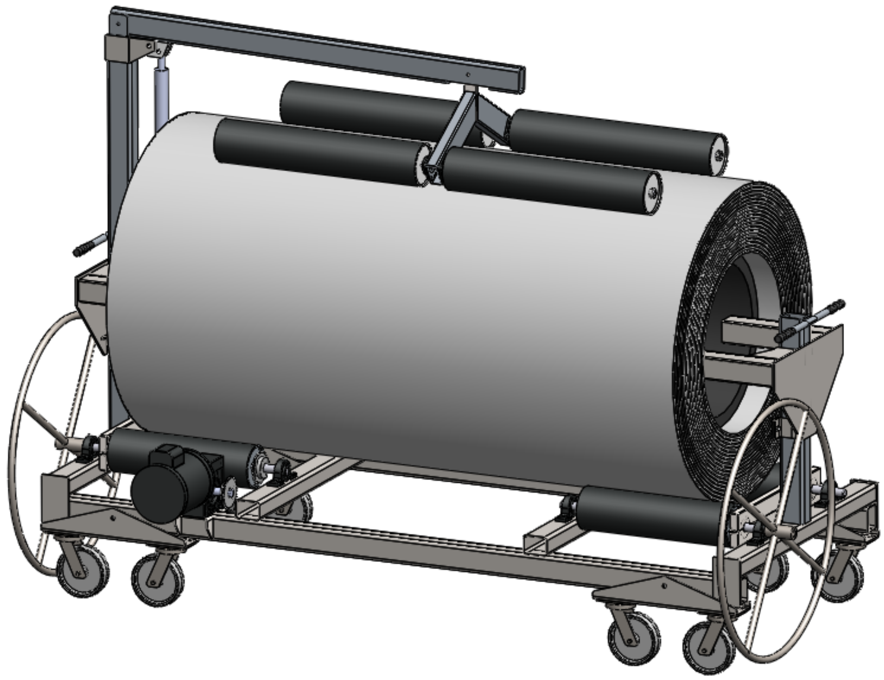

Figure 5-1: Composite Material Uncoiler final design [8].

5.2. Project Recommendations

Moving forward with the validation and production of the team's design, some key aspects identified by the team in order of importance are outlined below. The list of recommendations includes the verification of results obtained, safety precautions, procedures, and uncertainties about the design will be discussed.

1. All technical drawings and analyses presented throughout this report are considered preliminary and should be reviewed by a certified Professional Engineer, before moving forward into production of the design.
2. During the initial tests of the design, the team recommends first prototyping the design without the weight of the Top Roller subassembly. The composite roll should be placed on the Bottom Rollers to see if the weight of the composite roll itself will constrain the expansion. The team reckons that the weight of the roll could be enough to constrain the material until the roll is just about empty. Depending on the behaviour of the composite roll, adjusting the weighted mechanism arm with an appropriate weight could reduce the force required to push and pull the cart.
3. The preliminary calculations for the weight of the Top Rollers subassembly necessary to constrain the expansion of the composite roll are rough estimates at best. A Professional Engineer should vary these calculations by performing trial-and-error

experiments of the actual force required to completely constrain the expansion of the roll to further justify or adjust the designed weight of the Top Rollers.

4. During roll exchanges, the safety pin should be placed in the safety bracket to prevent the Top Rollers subassembly from falling, in case the hydraulic cylinder fails. The placement of the new roll using forklift should be centered as best as possible.
5. The team recommends cleaning the floor around the cart before transportation to allow for easy maneuverability of the cart. Additionally, note that the cart should only be operated on flat surfaces, as the cart is not designed for inclinations.
6. The team expects the rubber coated rollers to have enough friction to rotate the composite material roll, based on the amount of weight being applied. However, If the coating slips to much or wears off, the client may need to outsource the rubber coating to an industrial rubber coating provider.
7. As a safety precaution, the inspection of ball bearings and welds for fatigue failure and cracking should be incorporated into company's preventative maintenance program. This is suggested to prevent any easily detectable but potentially dangerous failure of the design which can be easily avoided through visual checks.
8. The selected 3/8" diameter bolts used to pin the rotating parts of the design, based on bearing stresses calculated in Appendix E: Stress Analysis already have a hefty safety factor. As an option, even bigger sized bolts could be used depending on the client's preference.

6. References

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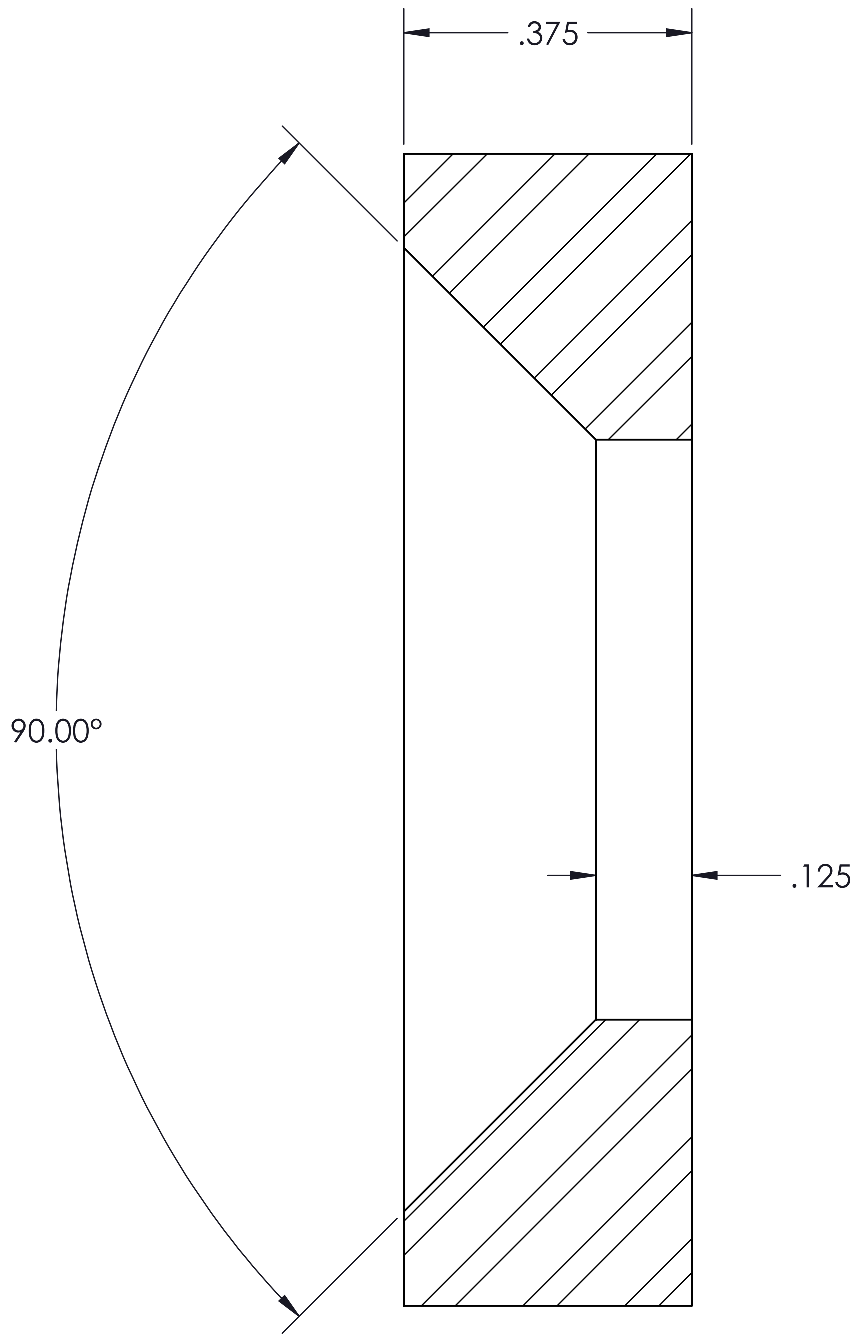
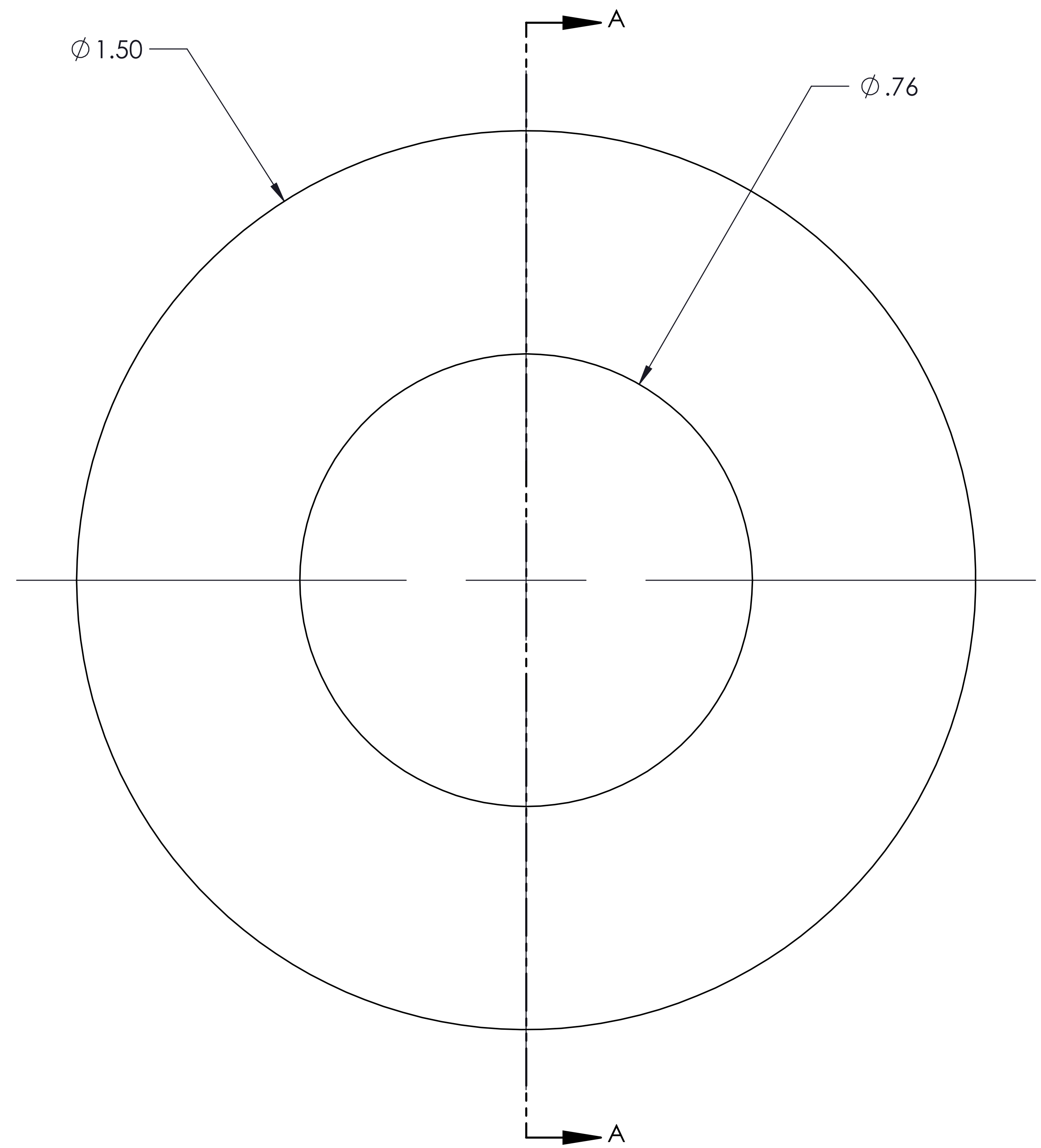
Appendix A: Final Design Technical Engineering Drawings

The following pages contain the technical engineering drawings of the team's final design. As per the client's request, drawings are numbered sequentially upward using the scheme of "M#####". Part numbers M00001 to M00005 are parts used in the Top Rollers subassembly, M00006 to M00010 are parts used in the hydraulic lift mechanism subassembly, M00011 to M00016 are parts used in the retaining shaft subassembly, and M00017 to M00030 are parts used in the bottom frame, rollers, walking axles, and drive system subassemblies. Additionally, M00031 to M00040 include drawings of intermediate subassemblies and M00076 to M00083 include drawings of the high-level subassemblies outlined in detail throughout the report. Finally, M00084 includes the final assembly drawing of the team's design. Note, all drawings provided to the client are consider preliminary, and thus, should be checked and signed off by a Professional Engineer, before preceding to production.



SECTION A-A



TRIPLE E RECREATIONAL VEHICLES

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LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°		TITLE: Top Roller Spacer UNIT		
MANUFACTURED BY: -				
MATERIAL: AISI 4130 Steel, annealed at 865C		DRAWING NO: M00001	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

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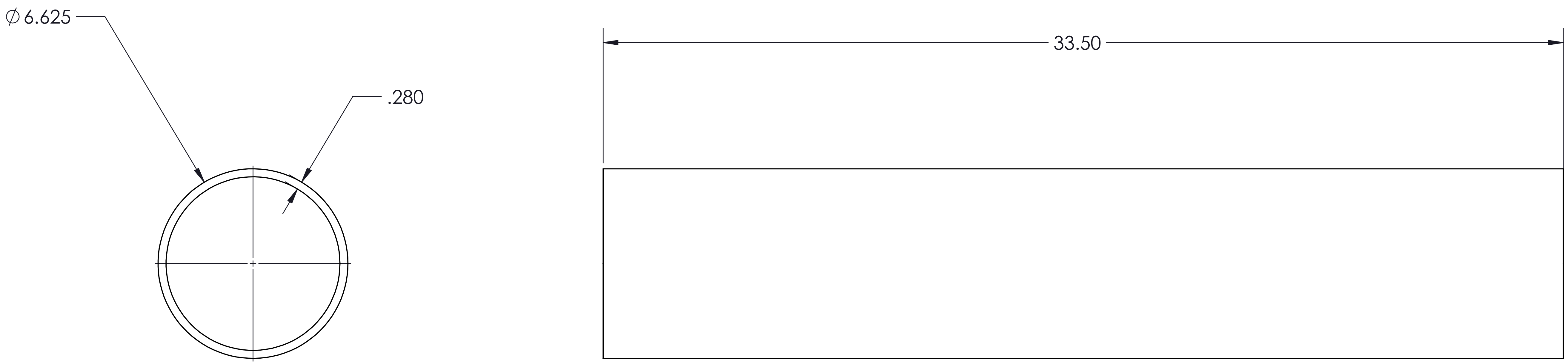

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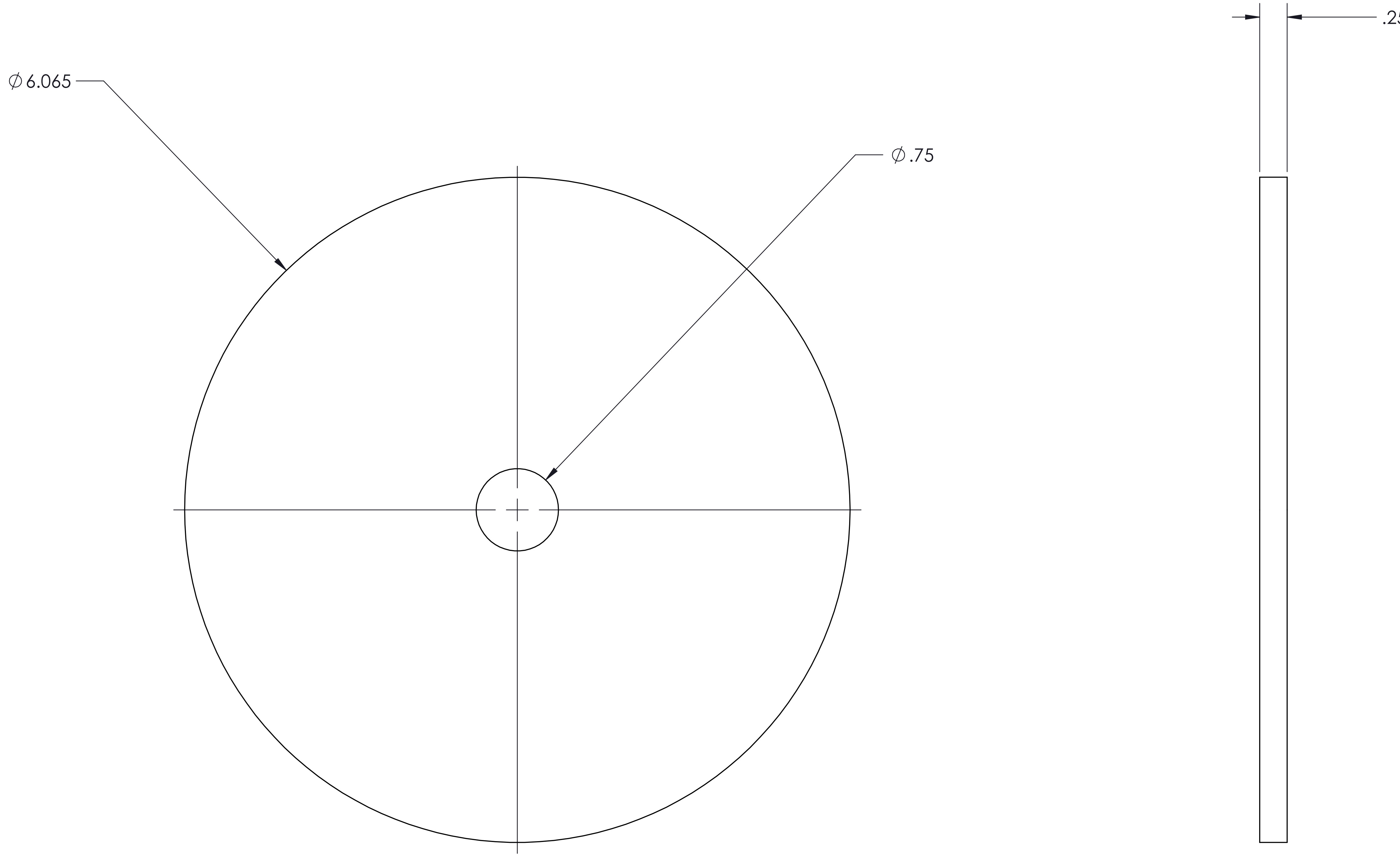
LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Top Roller Tube		
MANUFACTURED BY: -		UNIT		
MATERIAL: ASTM A-500 Gr. B		DRAWING NO:	SHEET:	REV:
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	M00002	1 OF 1	-
START: -	REPLACED: -			



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LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Top Roller Plate UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A36 Steel Plate		DRAWING NO: M00003	SHEET: 1 OF 1	REV: -
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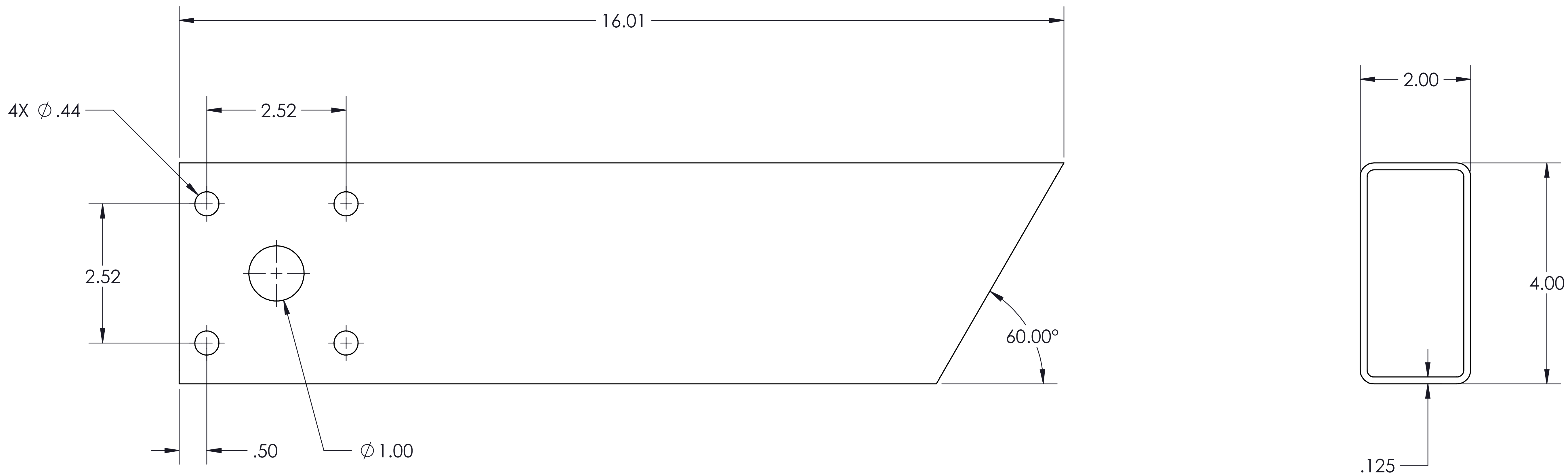

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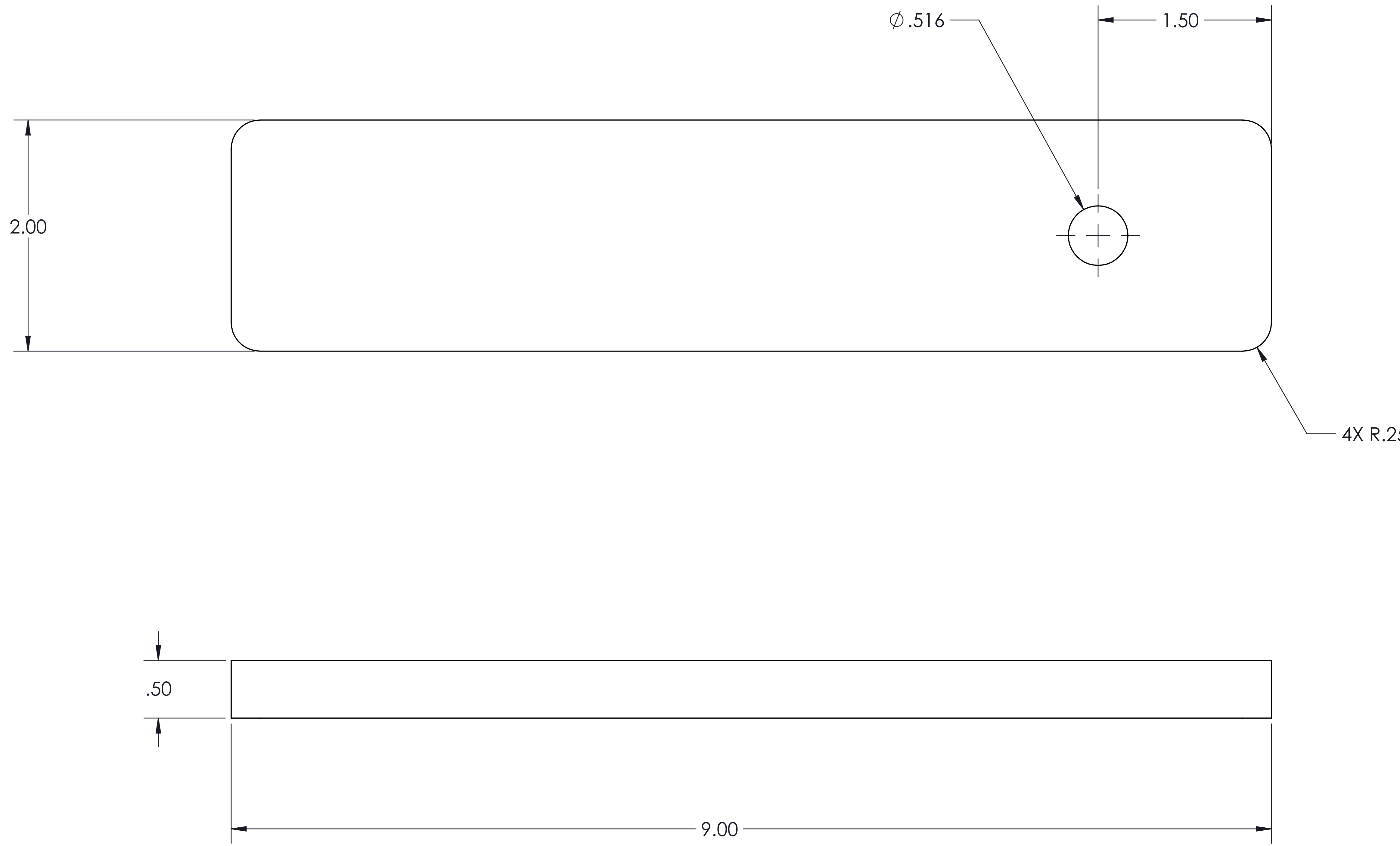
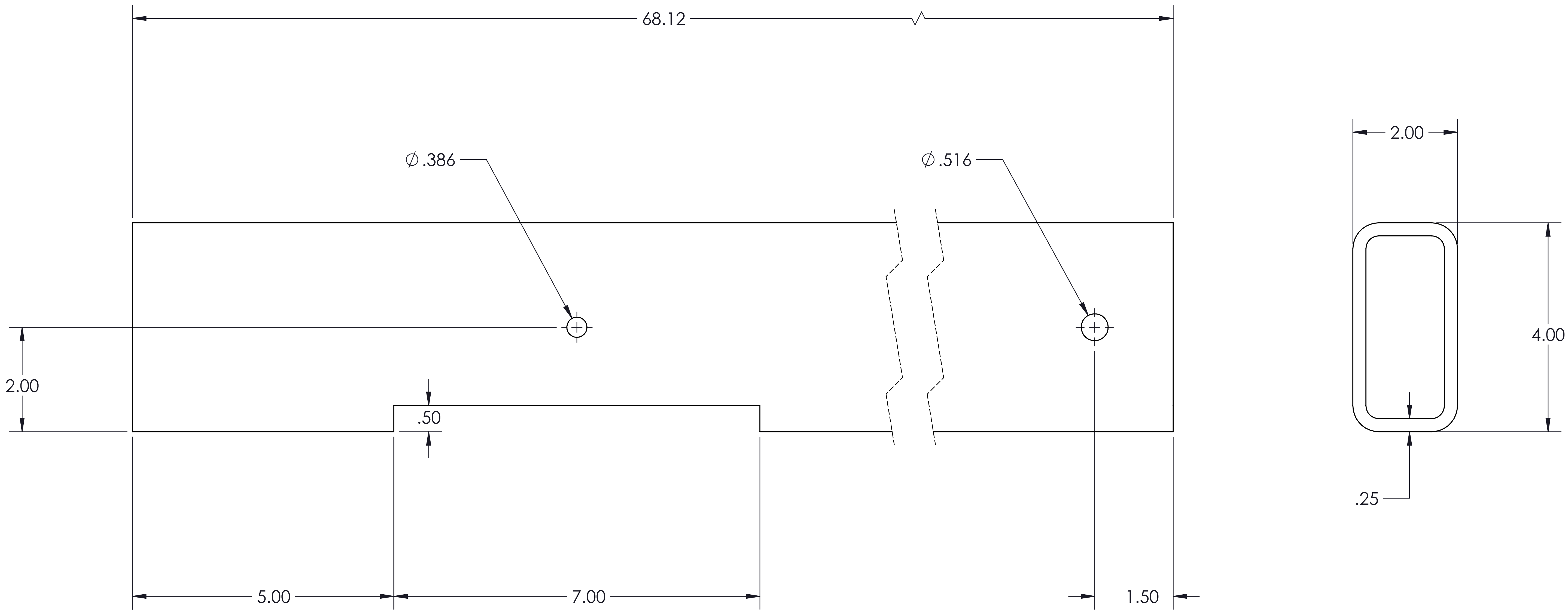
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Tube Arm
MANUFACTURED BY: -	UNIT
MATERIAL: ASTM A-500 Gr. B	DRAWING NO: M00004
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - / REPLACED: -	REV: -

M00004 November 29, 2017 9:19:46 PM Formula



TRIPLE E RECREATIONAL VEHICLES	THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.	LEISURE TRAVEL VANS	
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Hanging Plate UNIT		
MANUFACTURED BY: -			
MATERIAL: Plain Carbon Steel	DRAWING NO: M00005	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE DRAW DATE: 29/11/2017			
START: - REPLACED: -			

M00005 November 29, 2017 9:24:33 PM Formula



TRIPLE E RECREATIONAL VEHICLES

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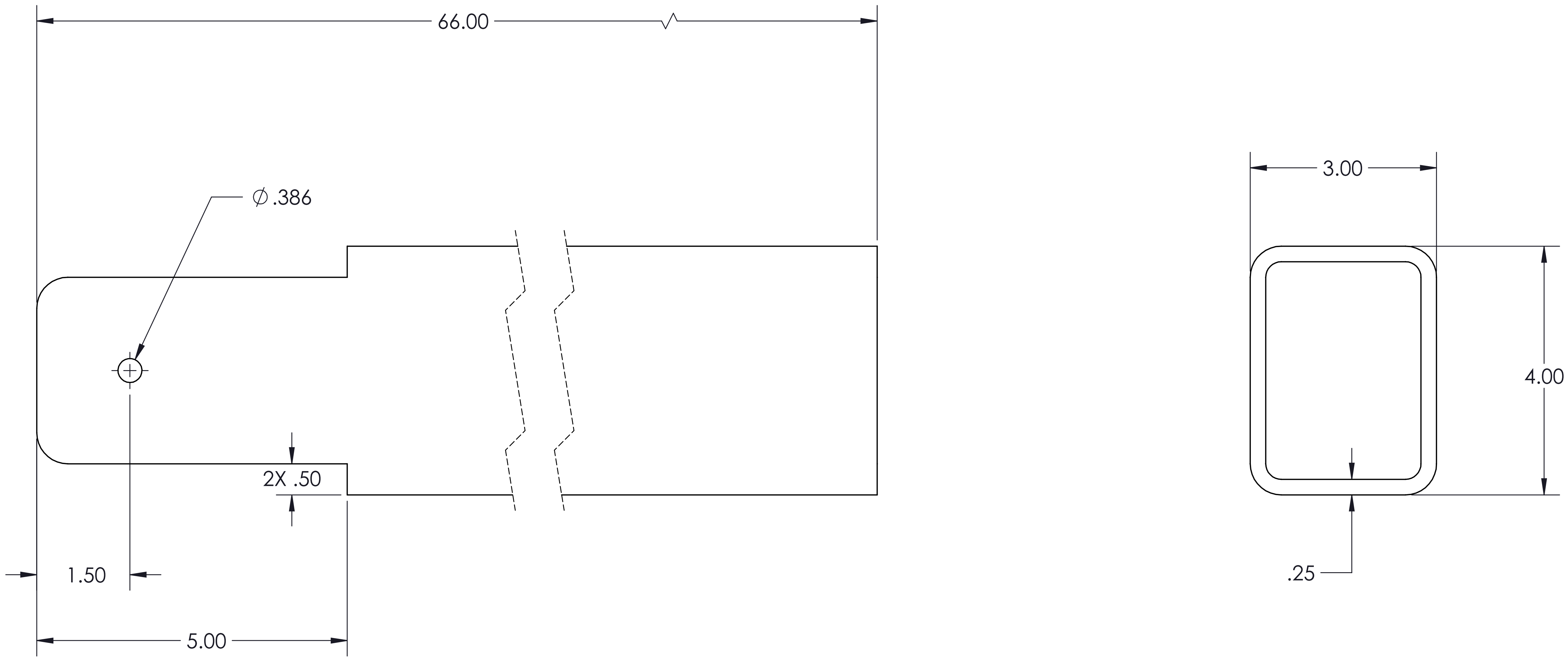
LEISURE TRAVEL VANS

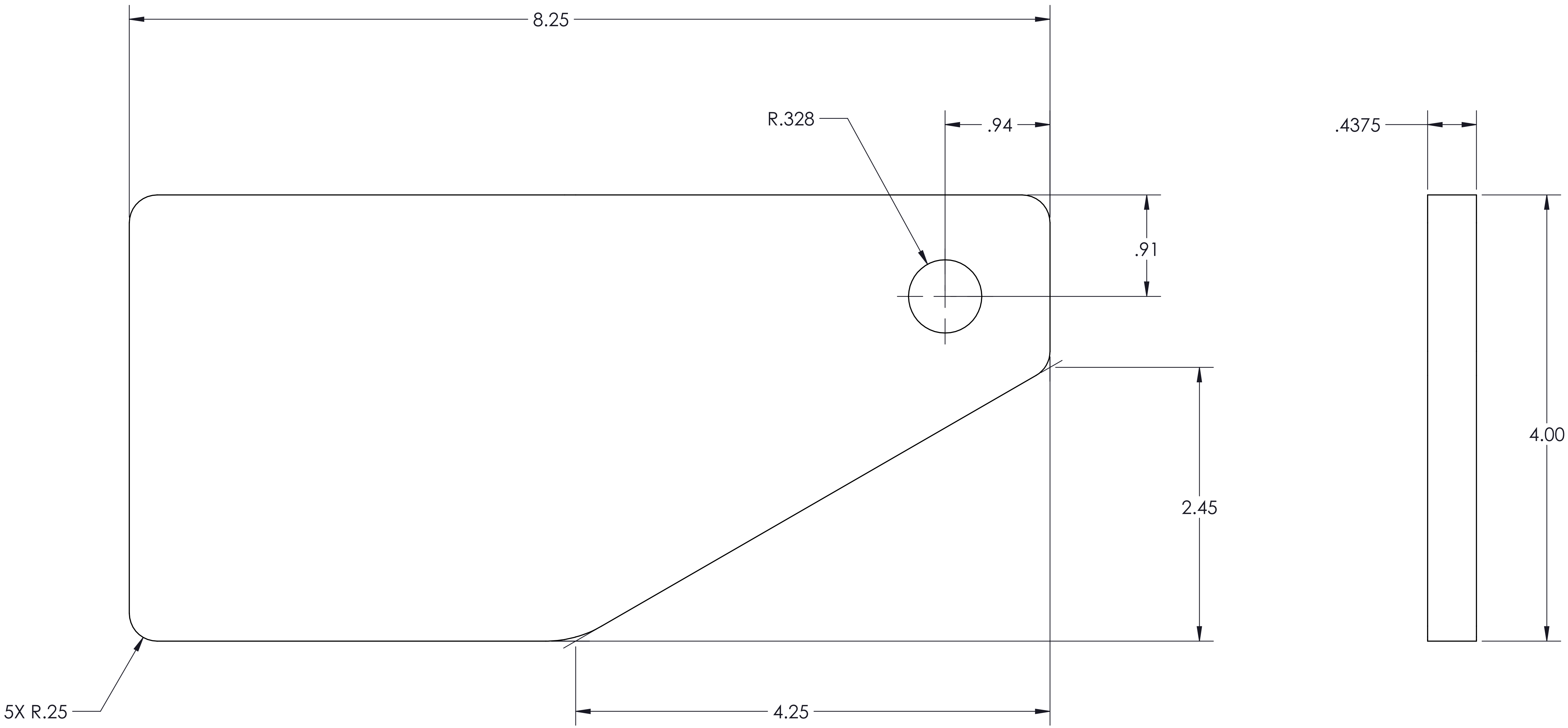
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Swinging Arm UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B		DRAWING NO: M00006	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

M00006 November 29, 2017 9:30:40 PM Formula



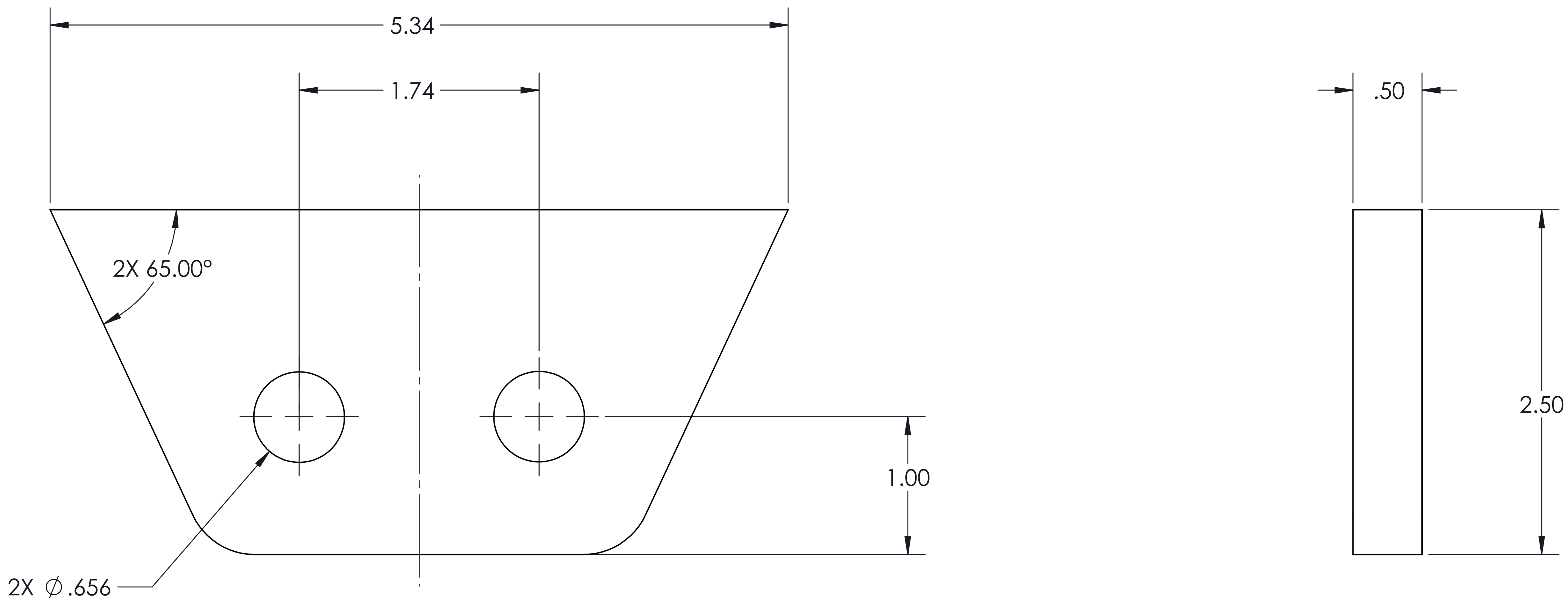
TRIPLE E RECREATIONAL VEHICLES	THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.	LEISURE TRAVEL VANS
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Side Support Arm	
MANUFACTURED BY: -	UNIT	
MATERIAL: ASTM A-500 Gr. B	DRAWING NO: M00007	SHEET: 1 OF 1
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017		REV: -
START: - / REPLACED: -		

M00007 November 29, 2017 9:35:49 PM Formula



TRIPLE E RECREATIONAL VEHICLES	LEISURE TRAVEL VANS
THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.	
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Safety Bracket
MANUFACTURED BY: -	UNIT
MATERIAL: ASTM A36	DRAWING NO: M00008
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - / REPLACED: -	REV: -

M00008 November 29, 2017 9:45:38 PM Formula



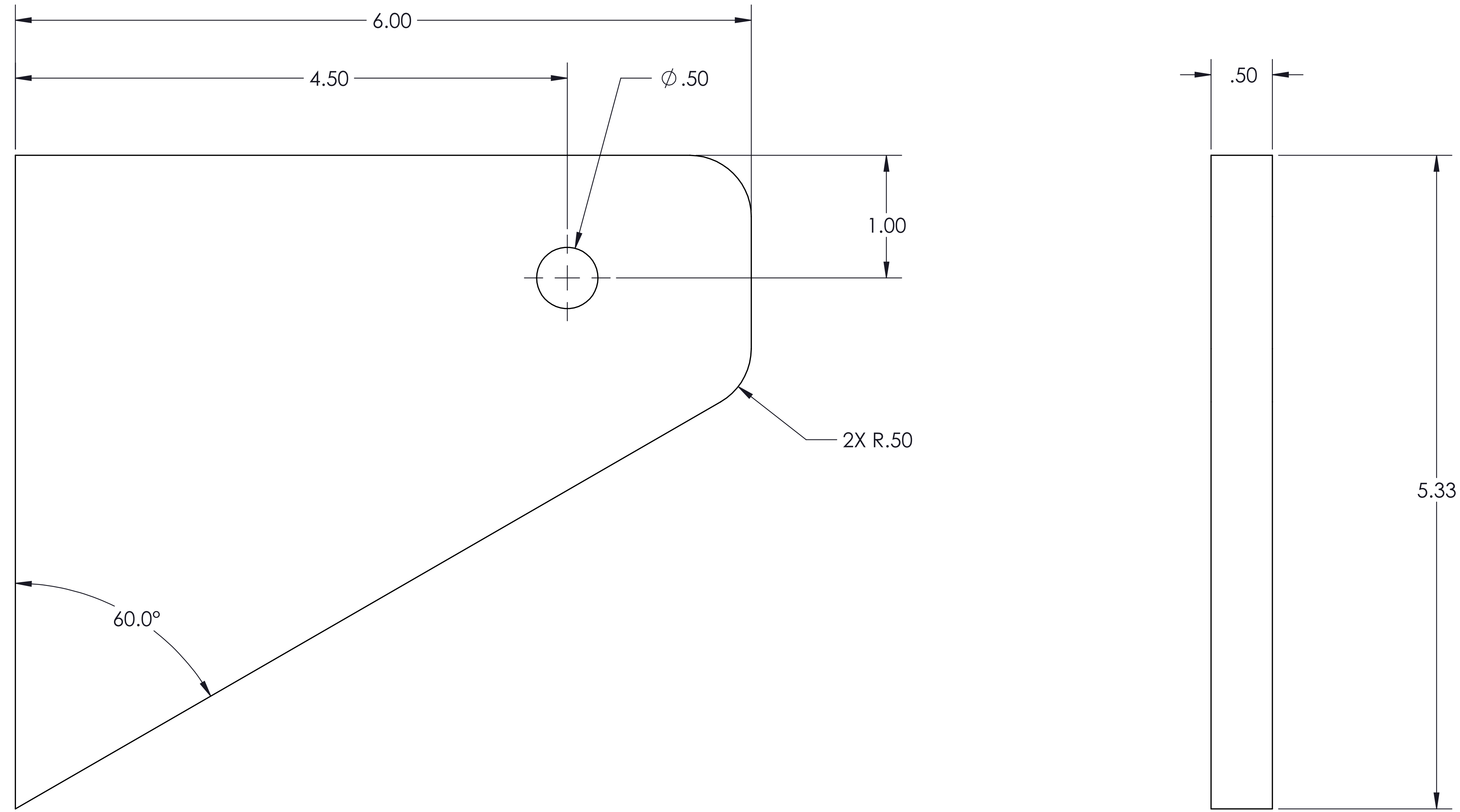
TRIPLE E RECREATIONAL VEHICLES

THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Bracket-Top Arm UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A36 Steel Plate		DRAWING NO: M00009	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

M00009 November 29, 2017 9:56:45 PM Formula

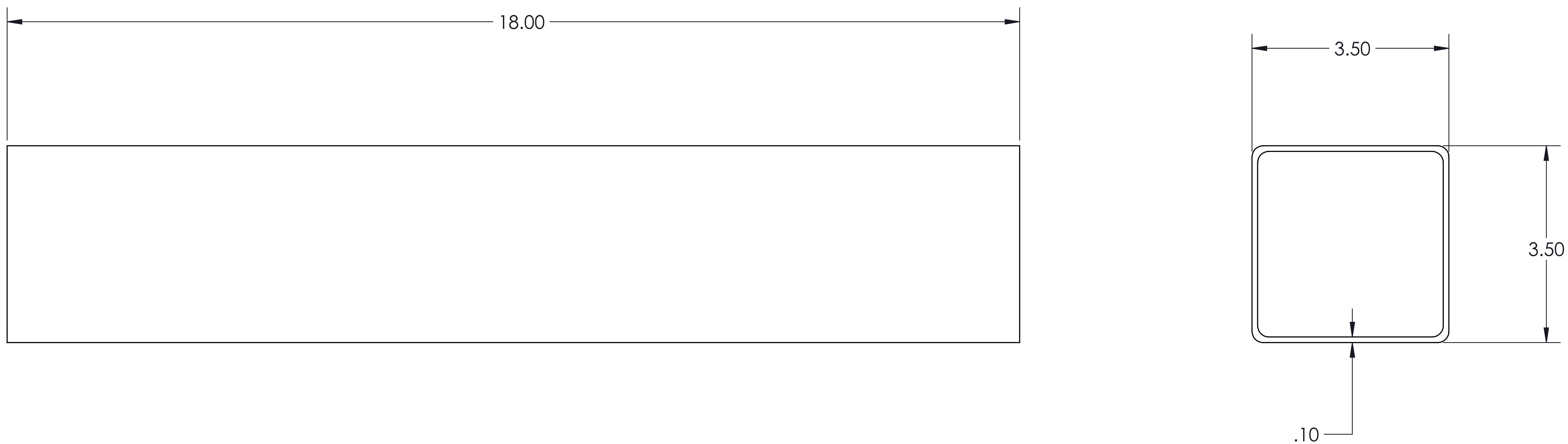


TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Bracket-Side Arm
MANUFACTURED BY: -	UNIT
MATERIAL: ASTM A36 Steel Plate	DRAWING NO: M00010
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - / REPLACED: -	REV: -

M00010 November 29, 2017 10:01:22 PM Formula

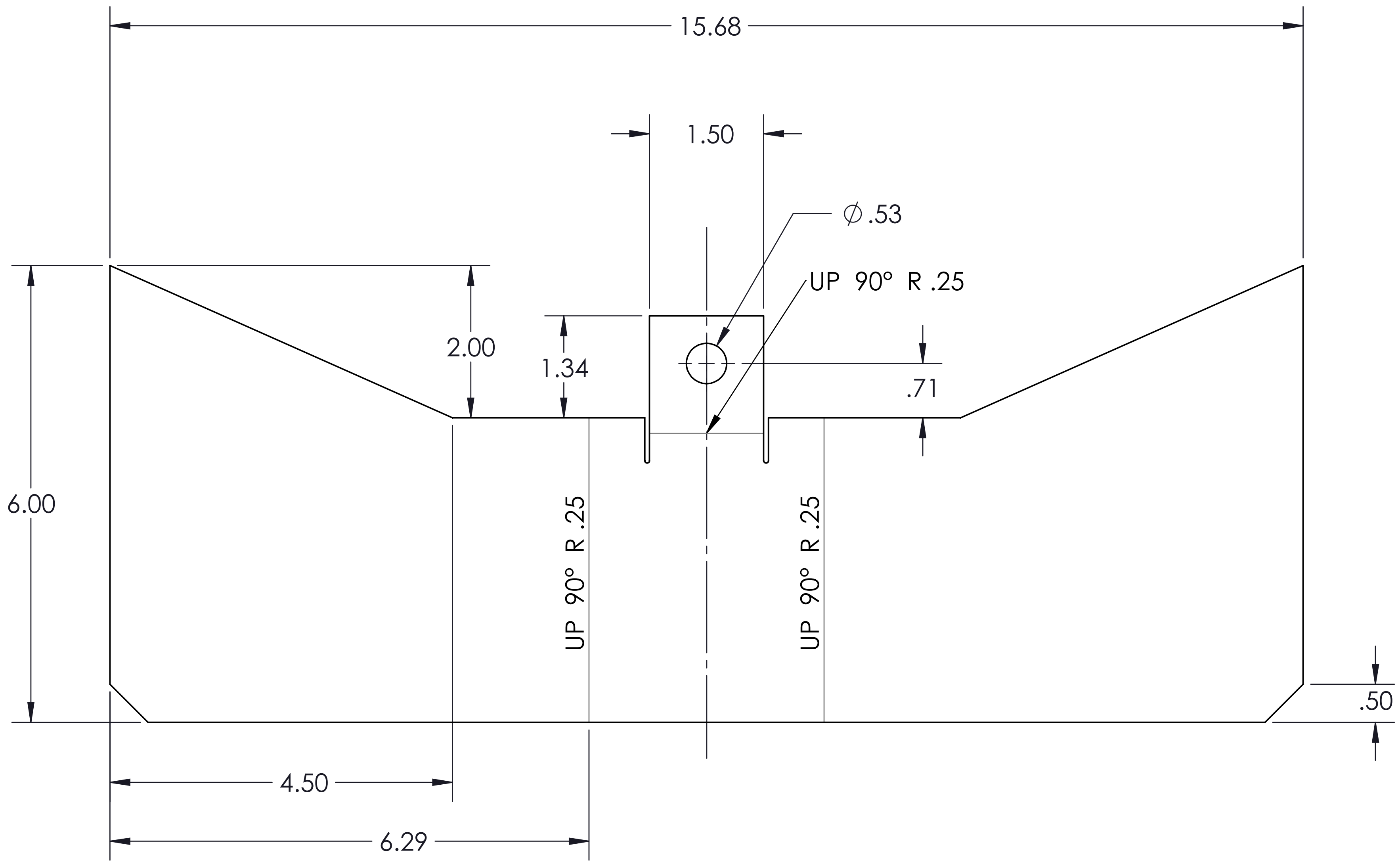


M00011 November 29, 2017 10:07:17 PM Formula

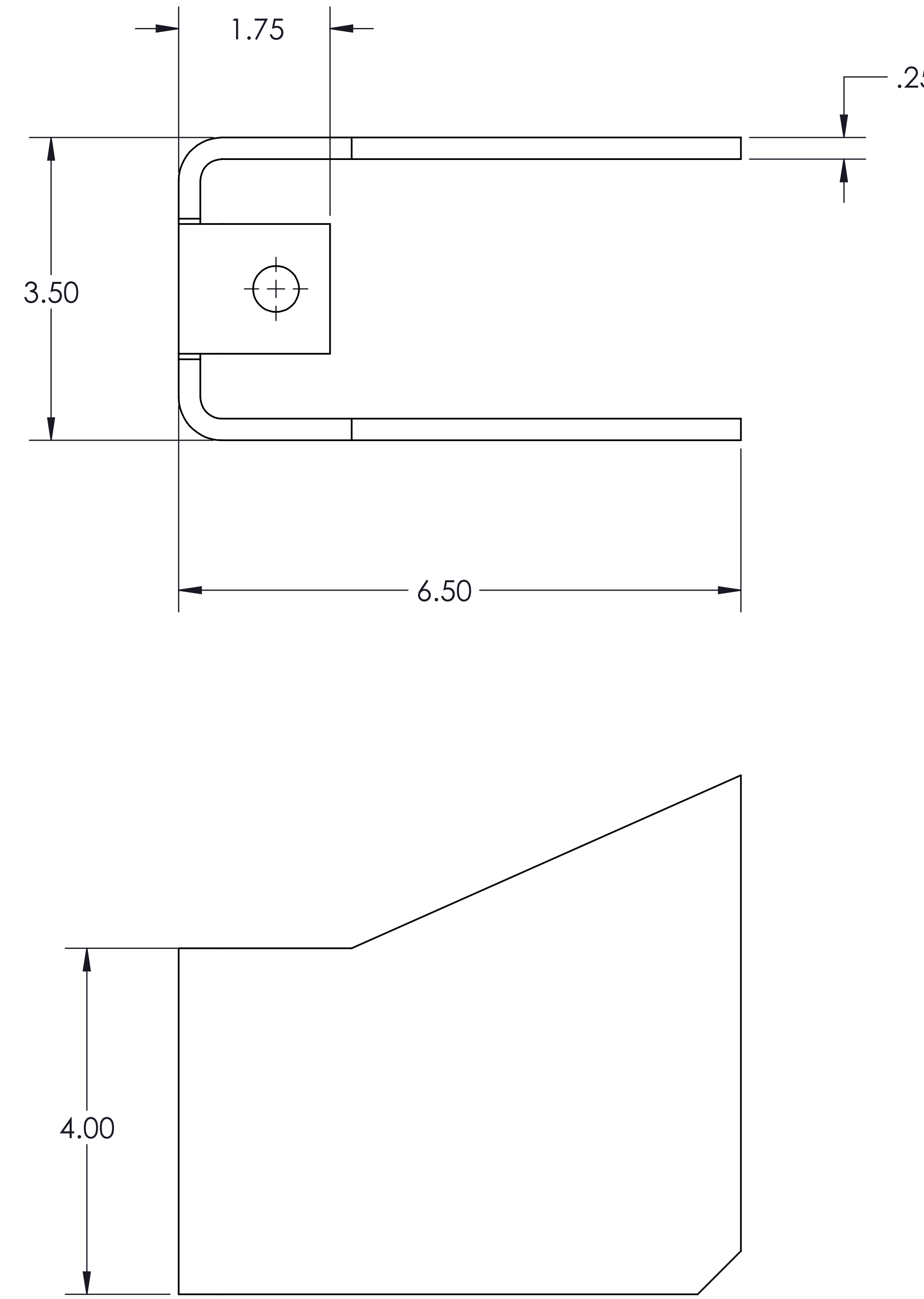
TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Square Shaft UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B		DRAWING NO: M00011	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			



FLAT PATTERN



FORMED PART

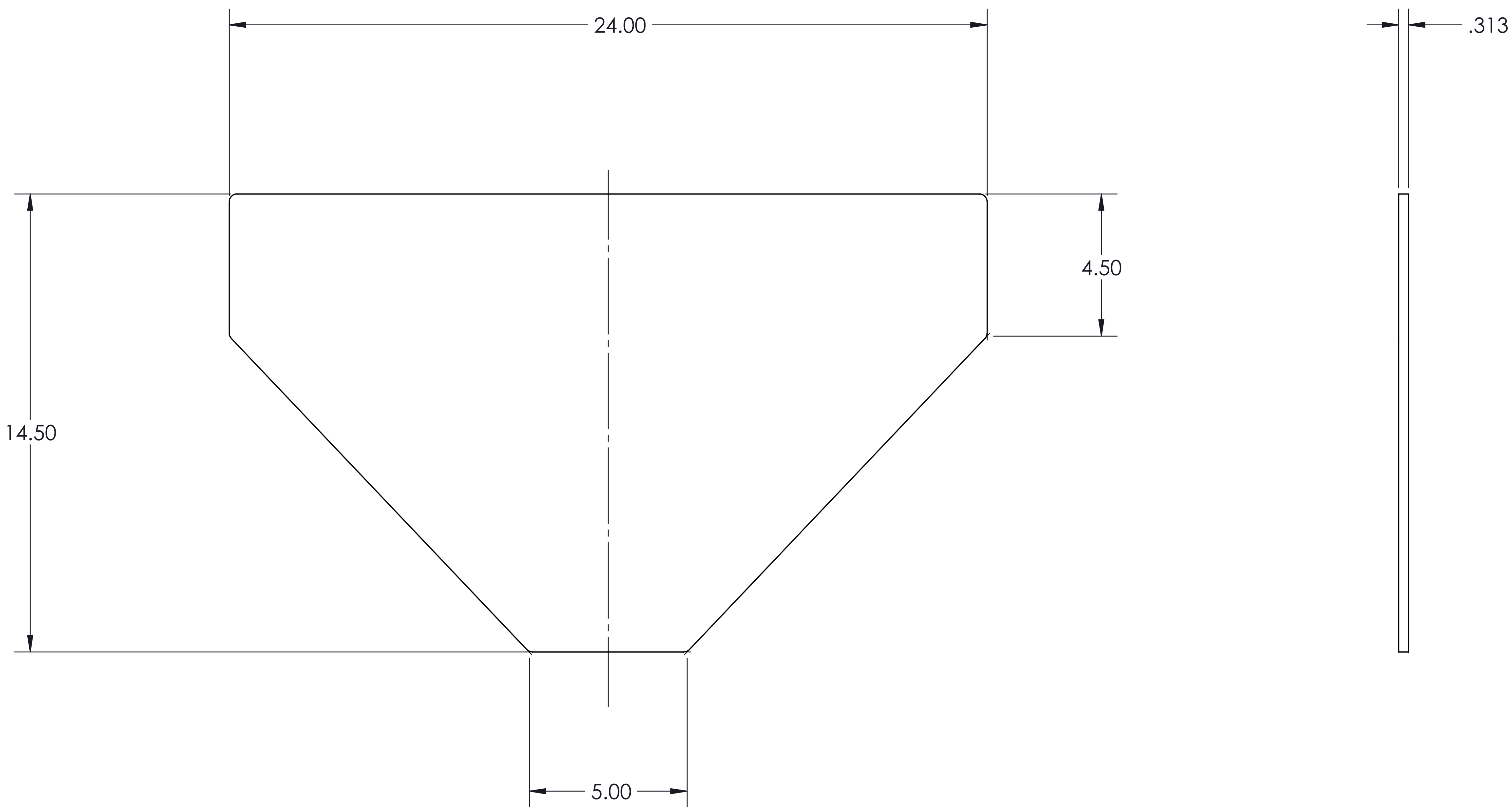
NOTE: FORMED PART DIMENSIONS SUPERCEDE FLAT PATTERN

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Gusset
MANUFACTURED BY: -	UNIT
MATERIAL: ASTM A36 Steel Plate	DRAWING NO: M00012
DRAWN BY: MFROESE — DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - — REPLACED: -	REV: -

M00012 November 29, 2017 10:27:00 PM Formula

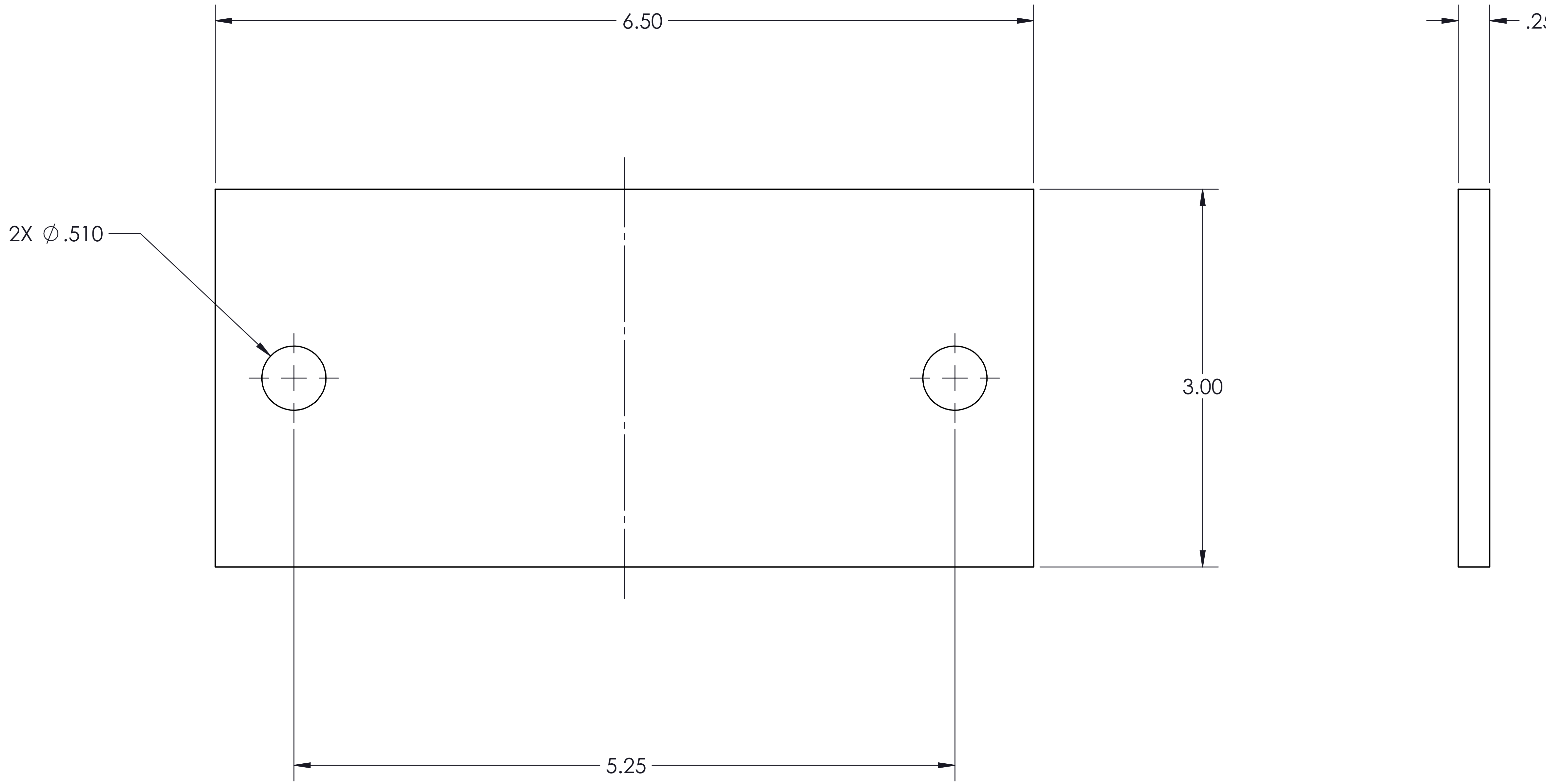


TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Backing Plate UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A36 Steel Plate		DRAWING NO: M00013	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

M00014 November 29, 2017 10:37:27 PM Formula

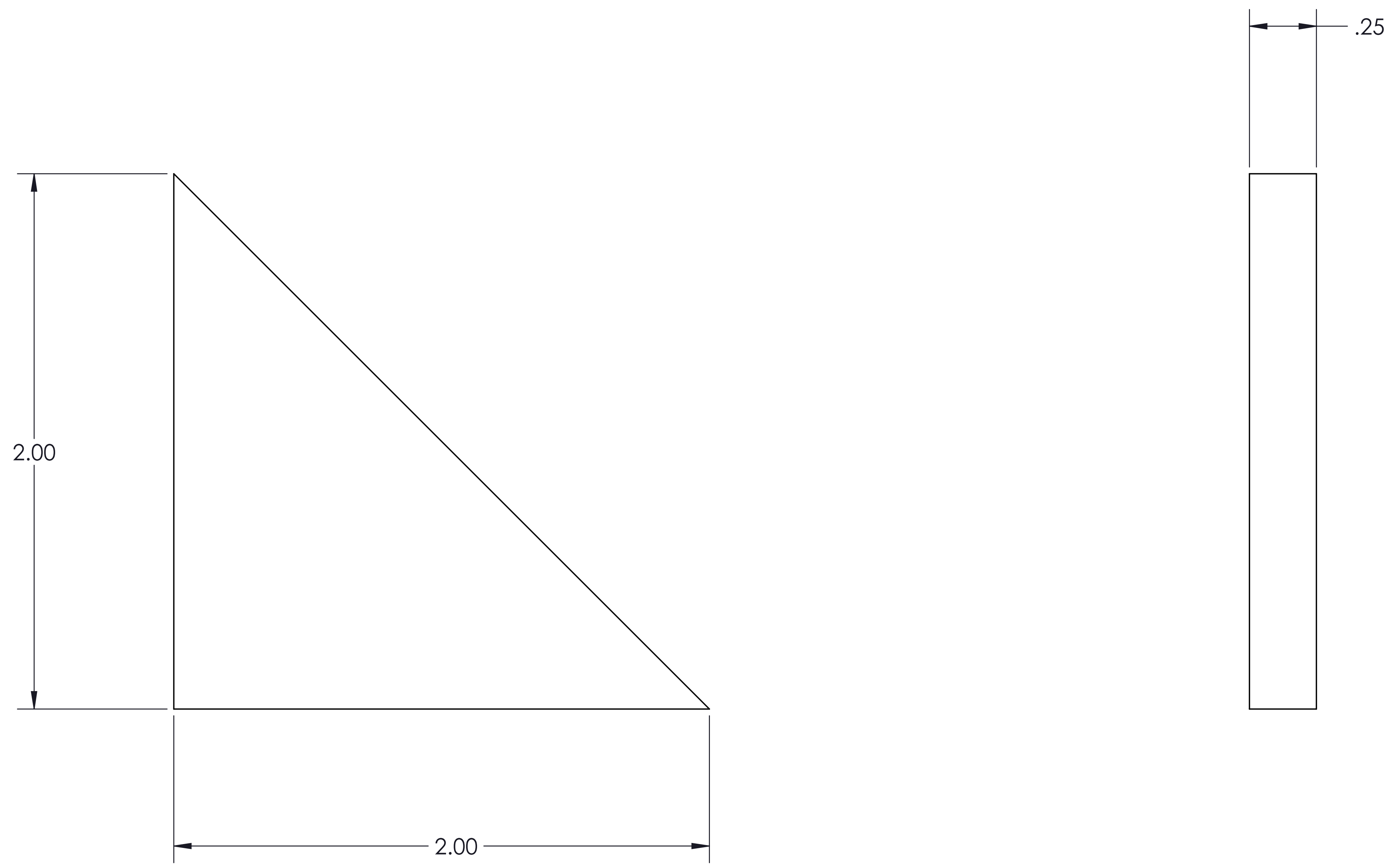


NOTE
FINISH: POWDER COAT BLUE

TRIPLE E
RECREATIONAL VEHICLES

LEISURE
TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Locking Plate UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A36 Steel Plate		DRAWING NO: M00014	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

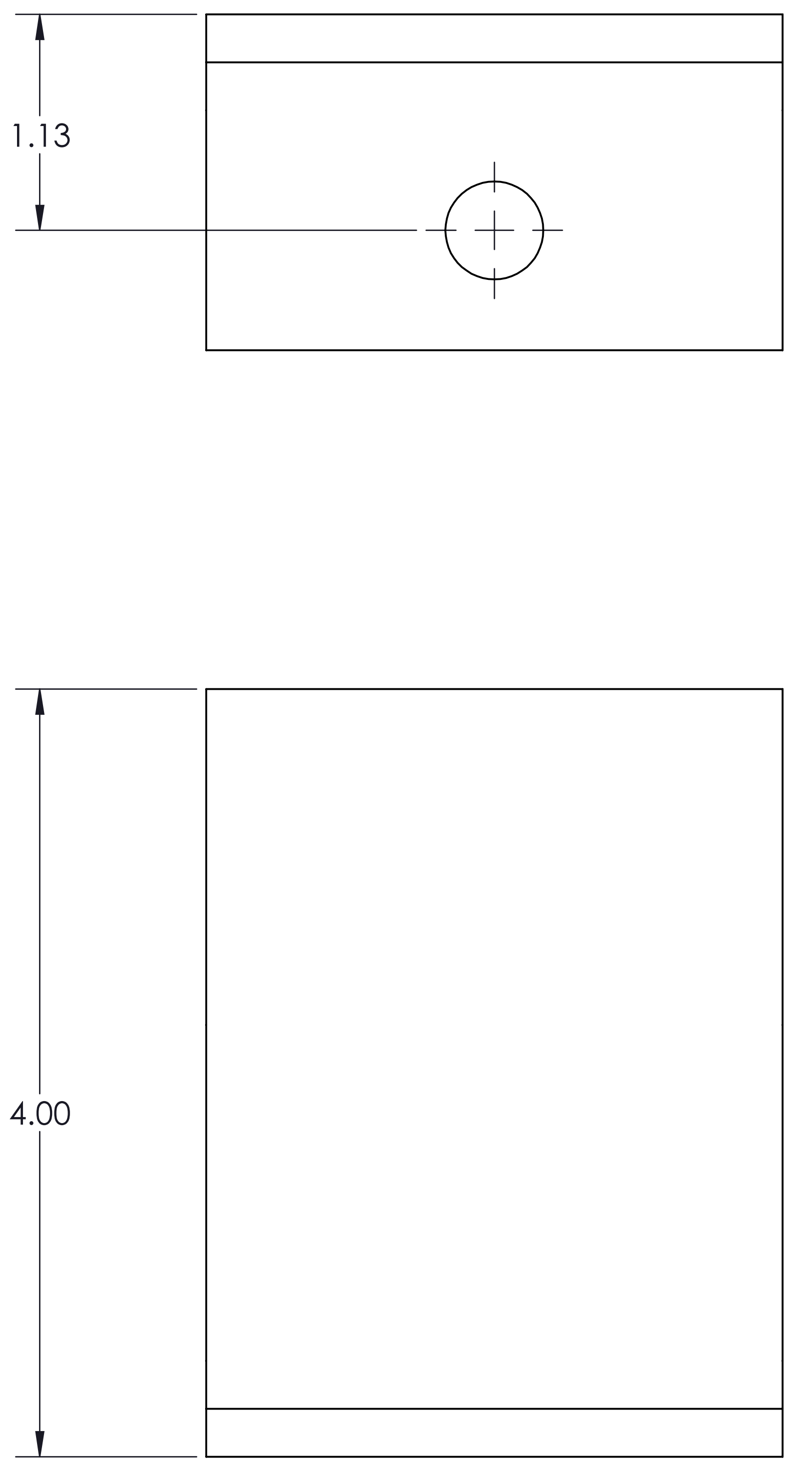


TRIPLE E RECREATIONAL VEHICLES

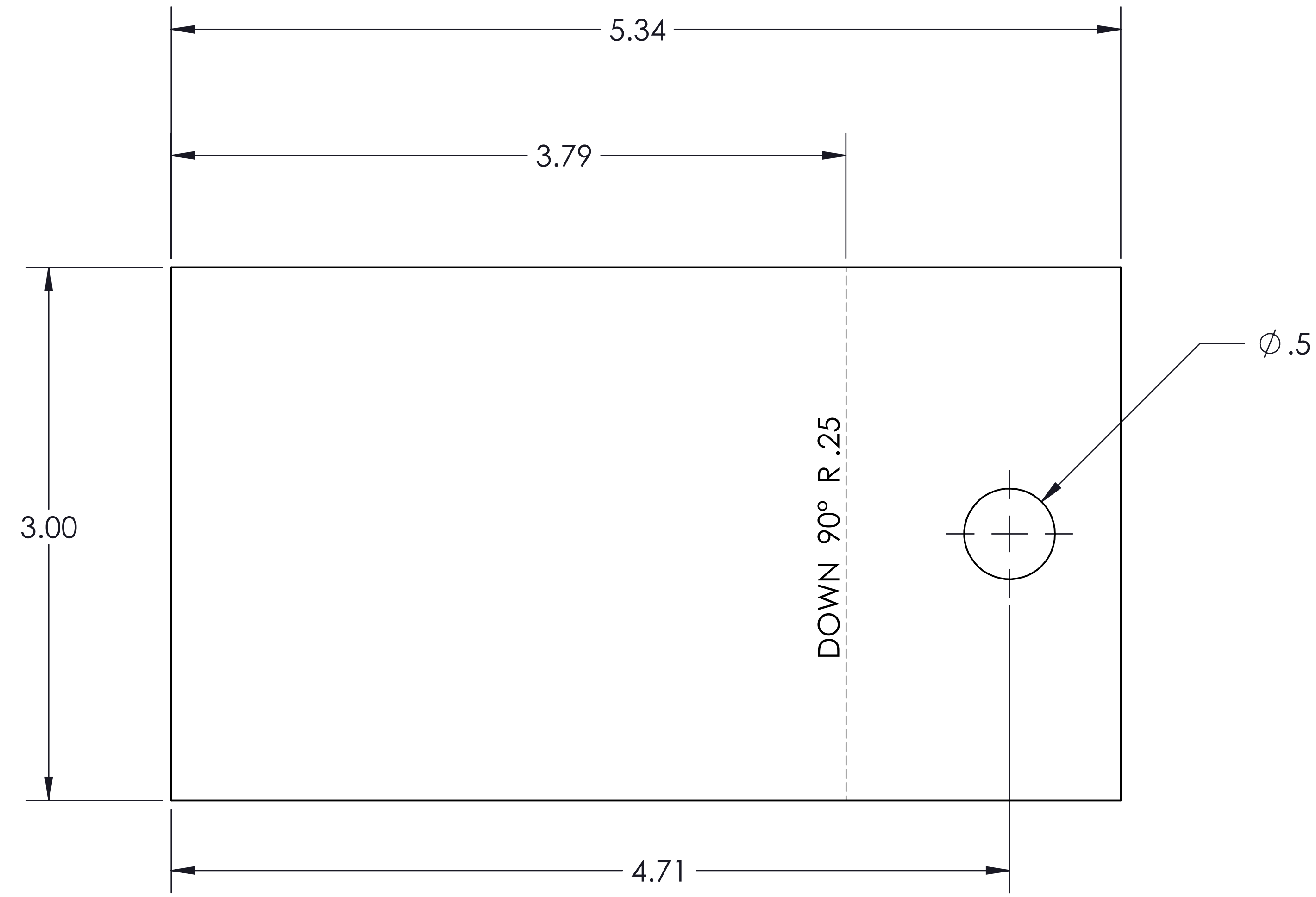
LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Lower Gusset UNIT
MANUFACTURED BY: -	
MATERIAL: ASTM A36 Steel Plate	DRAWING NO: M00015
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - / REPLACED: -	REV: -

M00015 November 29, 2017 10:41:01 PM Formula



FORMED PART



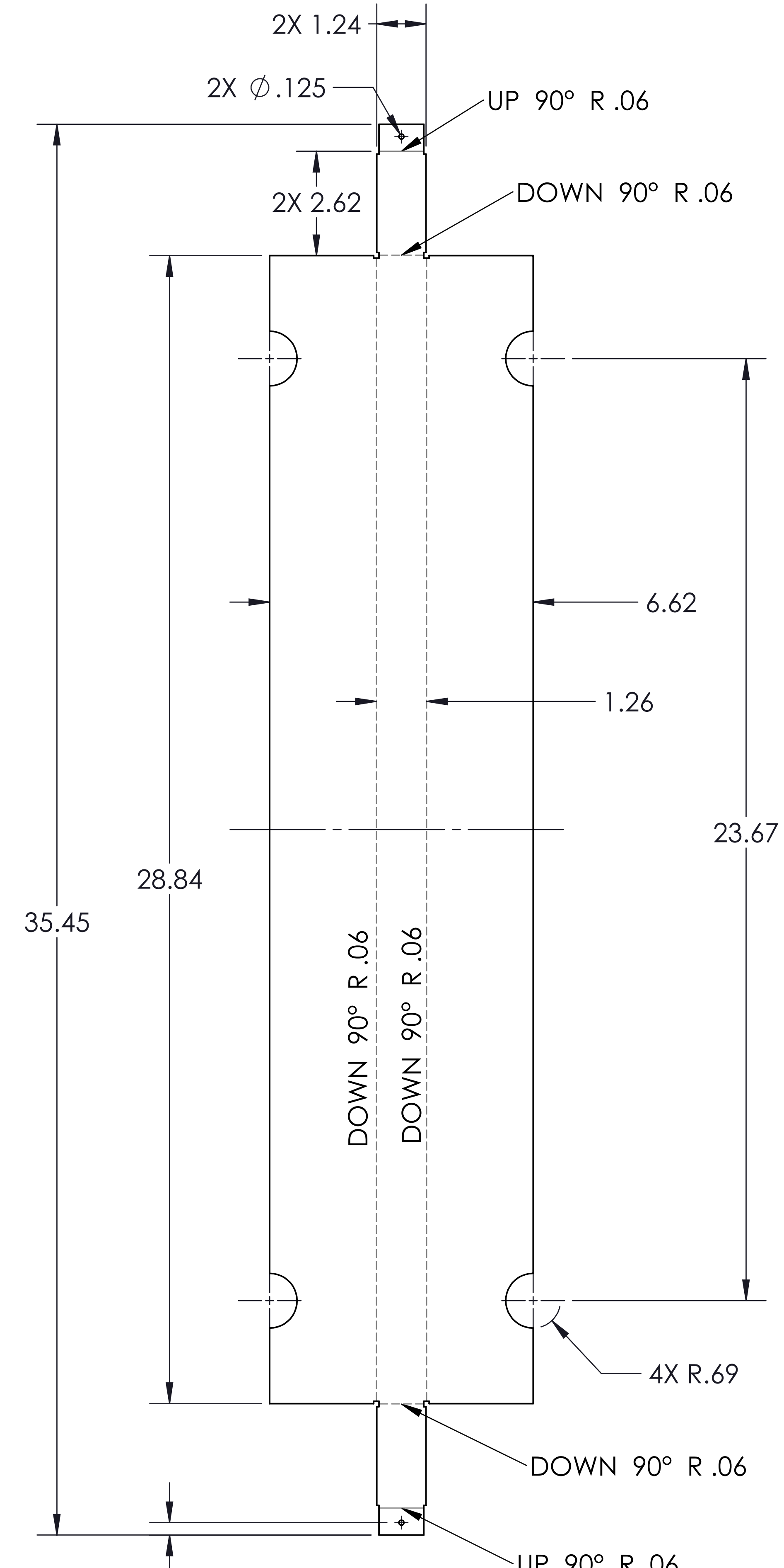
FLAT PATTERN

NOTE:
FORMED PART DIMENSIONS SUPERSEDE FLAT PATTERN

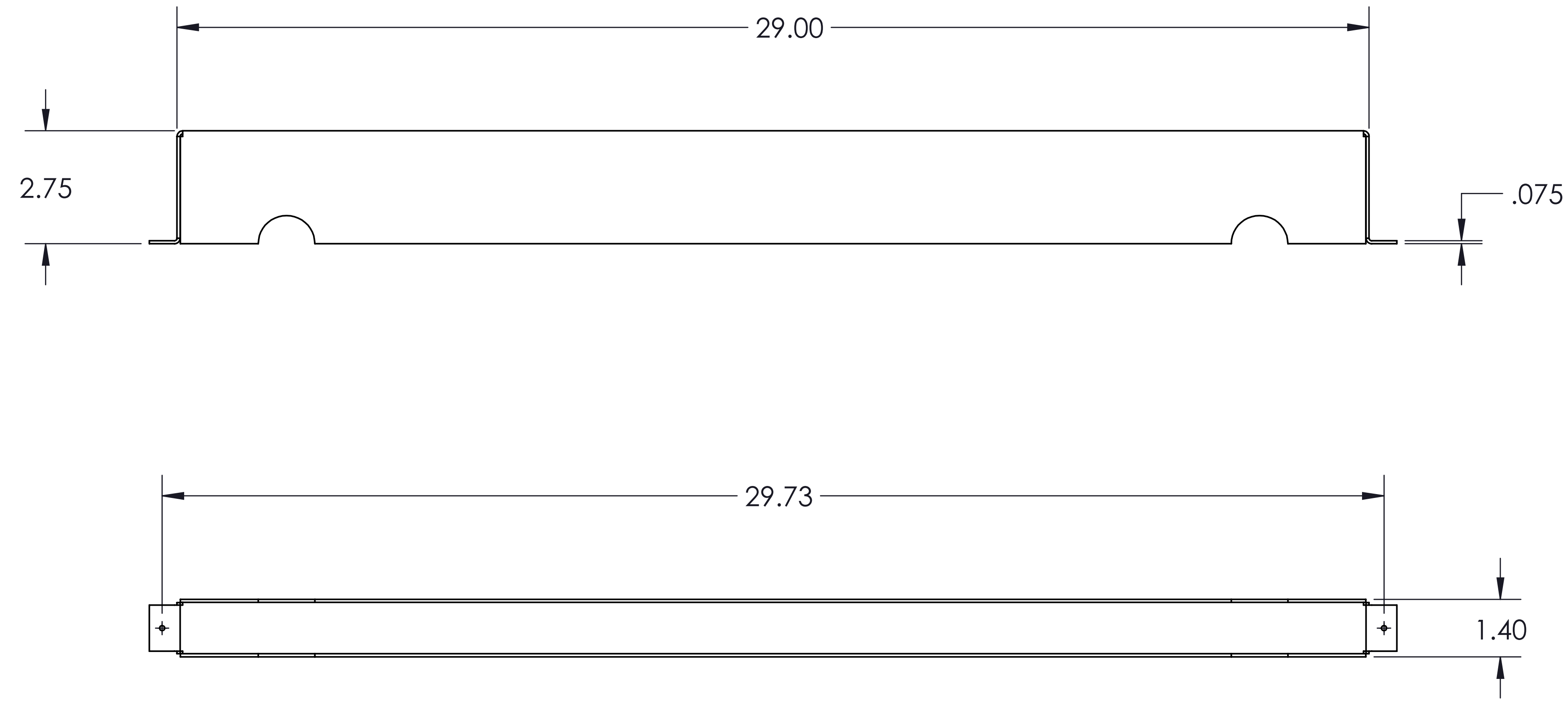
TRIPLE E RECREATIONAL VEHICLES	THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.	LEISURE TRAVEL VANS
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Retaining Shaft Bracket	
MANUFACTURED BY: -	UNIT	
MATERIAL: ASTM A36 Steel Plate	DRAWING NO: M00016	SHEET: 1 OF 1 / REV: -
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017		
START: - / REPLACED: -		

M00016 November 29, 2017 10:47:37 PM Formula

M00017 November 29, 2017 11:04:03 PM Formula



FLAT PATTERN



FORMED PART

NOTE:
FORMED PART DIMENSIONS SUPERSEDE FLAT PATTERN
FINISH: POWDER COAT BLUE

TRIPLE E RECREATIONAL VEHICLES

THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL: ASTM A36 Steel Plate

DRAWN BY: MFROESE

DRAW DATE: 29/11/2017

START: -

REPLACED: -

TITLE: Chain Cover

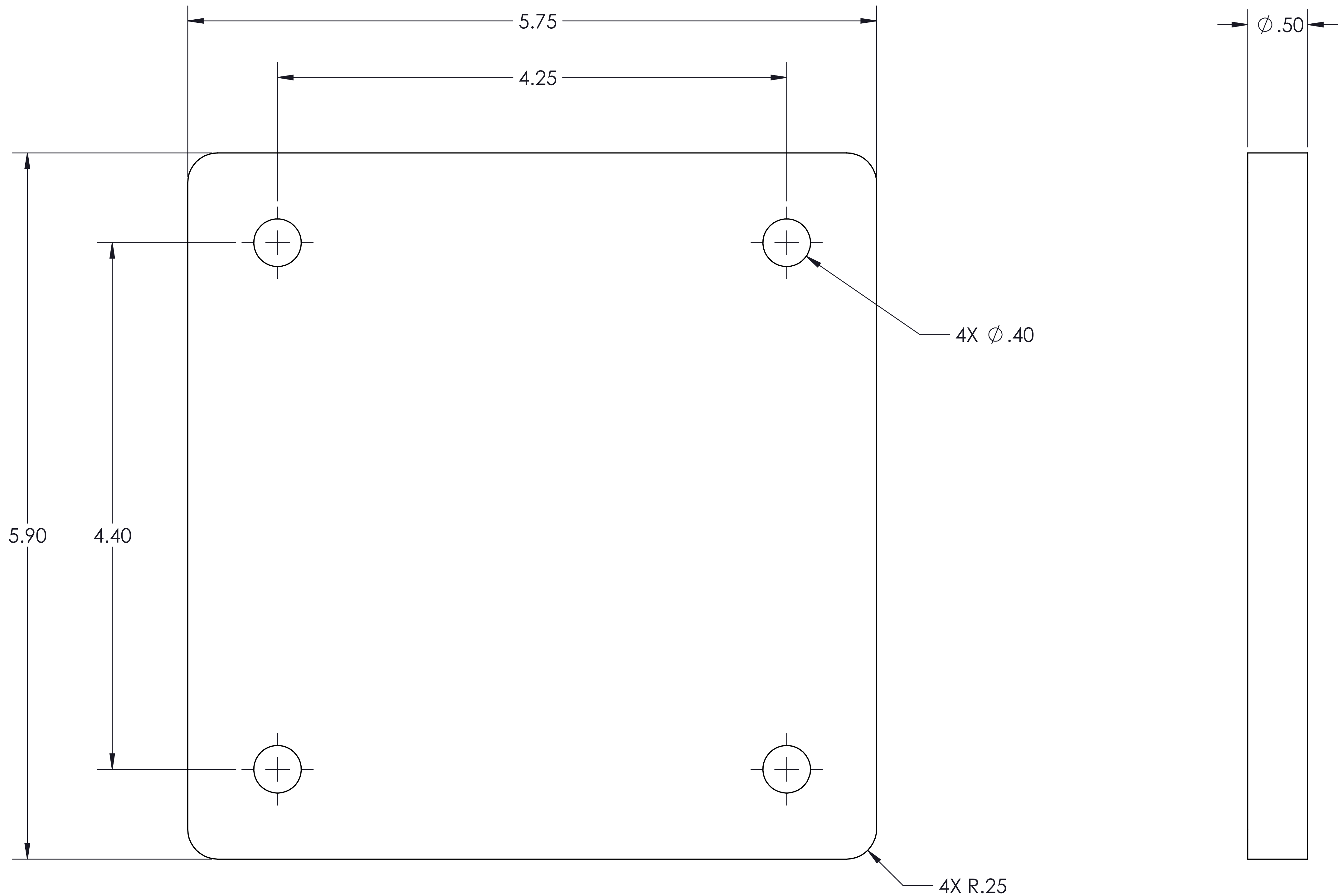
UNIT

DRAWING NO: M00017

SHEET: 1 OF 1

REV: -

M00018 November 30, 2017 10:27:10 AM Formula

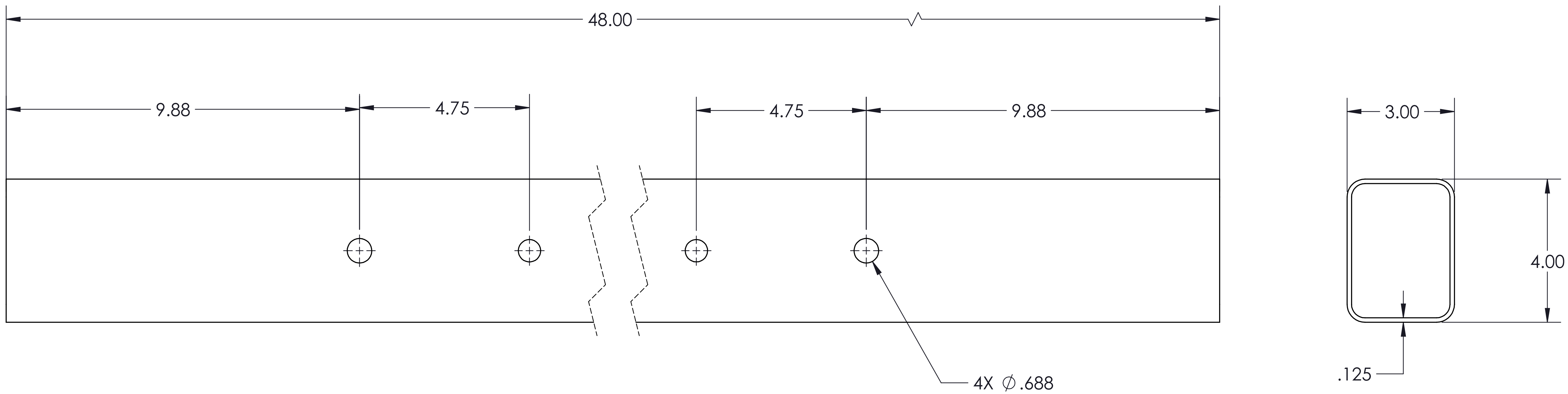


TRIPLE E RECREATIONAL VEHICLES

THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.

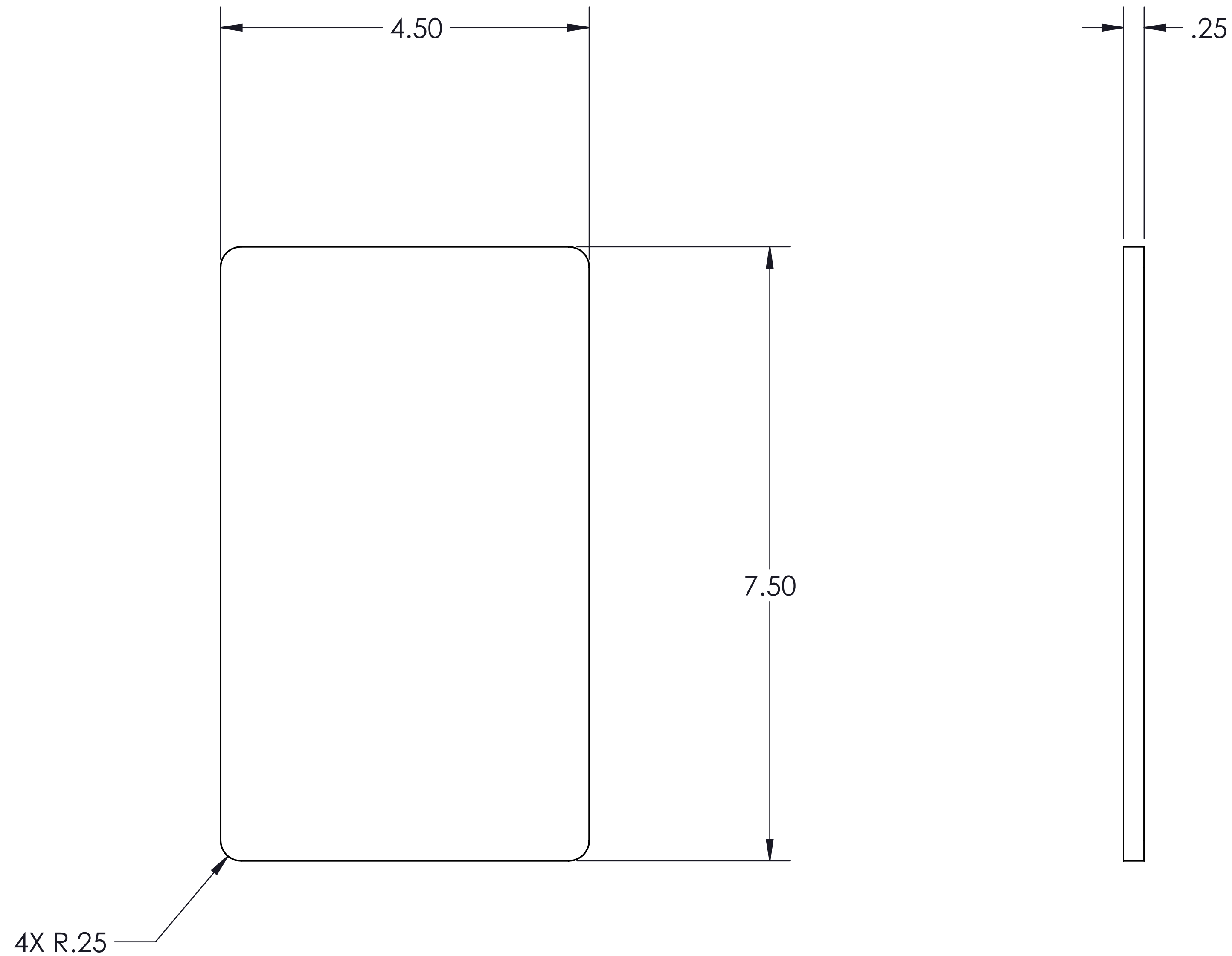
LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Mounting Plate UNIT
MANUFACTURED BY: -		
MATERIAL: ASTM A36 Steel Plate		DRAWING NO: M00018
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: -	REPLACED: -	REV: -



M00019 November 30, 2017 10:36:32 AM Formula

TRIPLE E RECREATIONAL VEHICLES	THIS DRAWING CONTAINS PROPRIETARY INFORMATION. ITS CONTENTS MUST NOT BE DISCLOSED IN WHOLE OR IN PART WITHOUT THE PRIOR WRITTEN CONSENT OF TRIPLE E CANADA LTD.	LEISURE TRAVEL VANS		
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Frame Cross Support UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B				
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	DRAWING NO: M00019	SHEET: 1 OF 1	REV: -
START: -	REPLACED: -			



TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Cross Brace Support UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A36 Steel Plate		DRAWING NO: M00020	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
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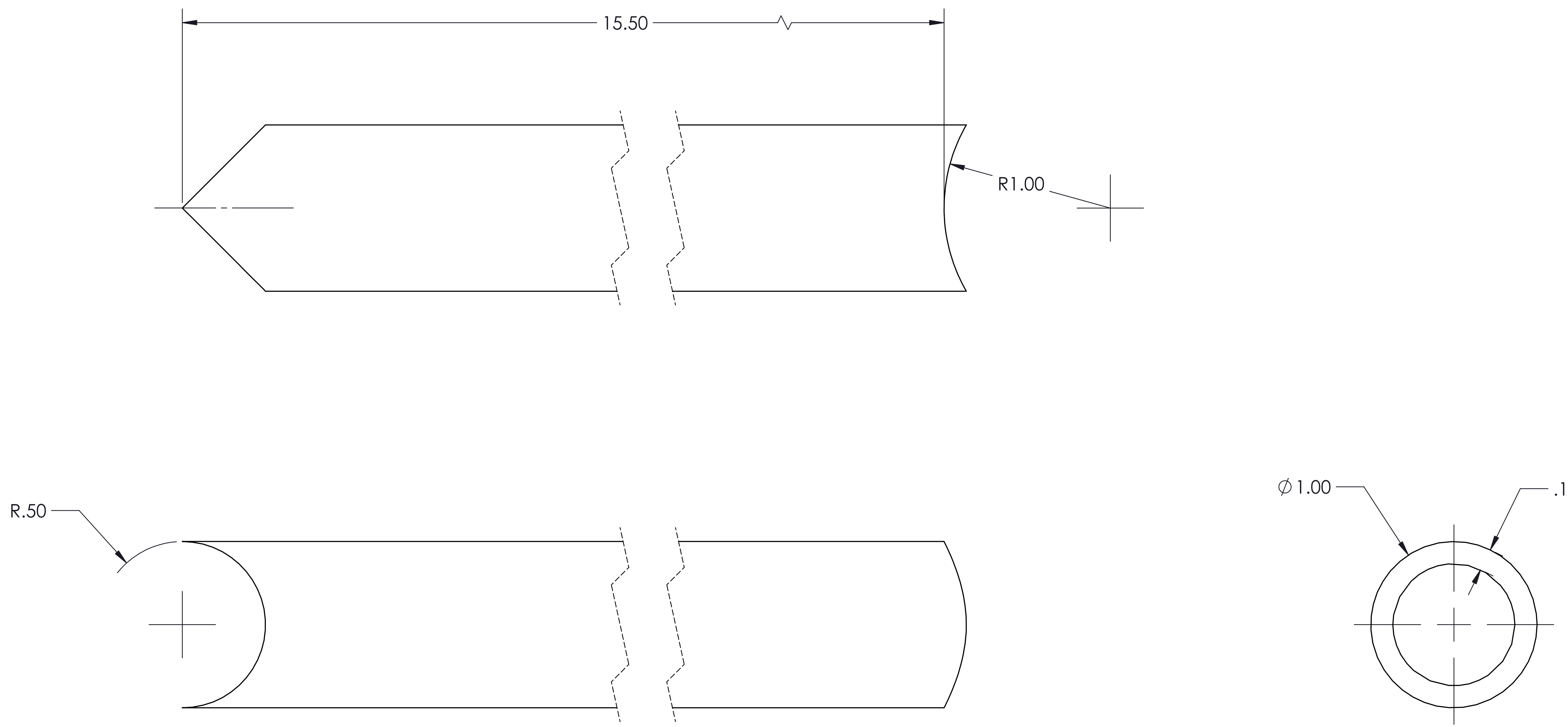
M00020 November 30, 2017 10:39:56 AM Formula



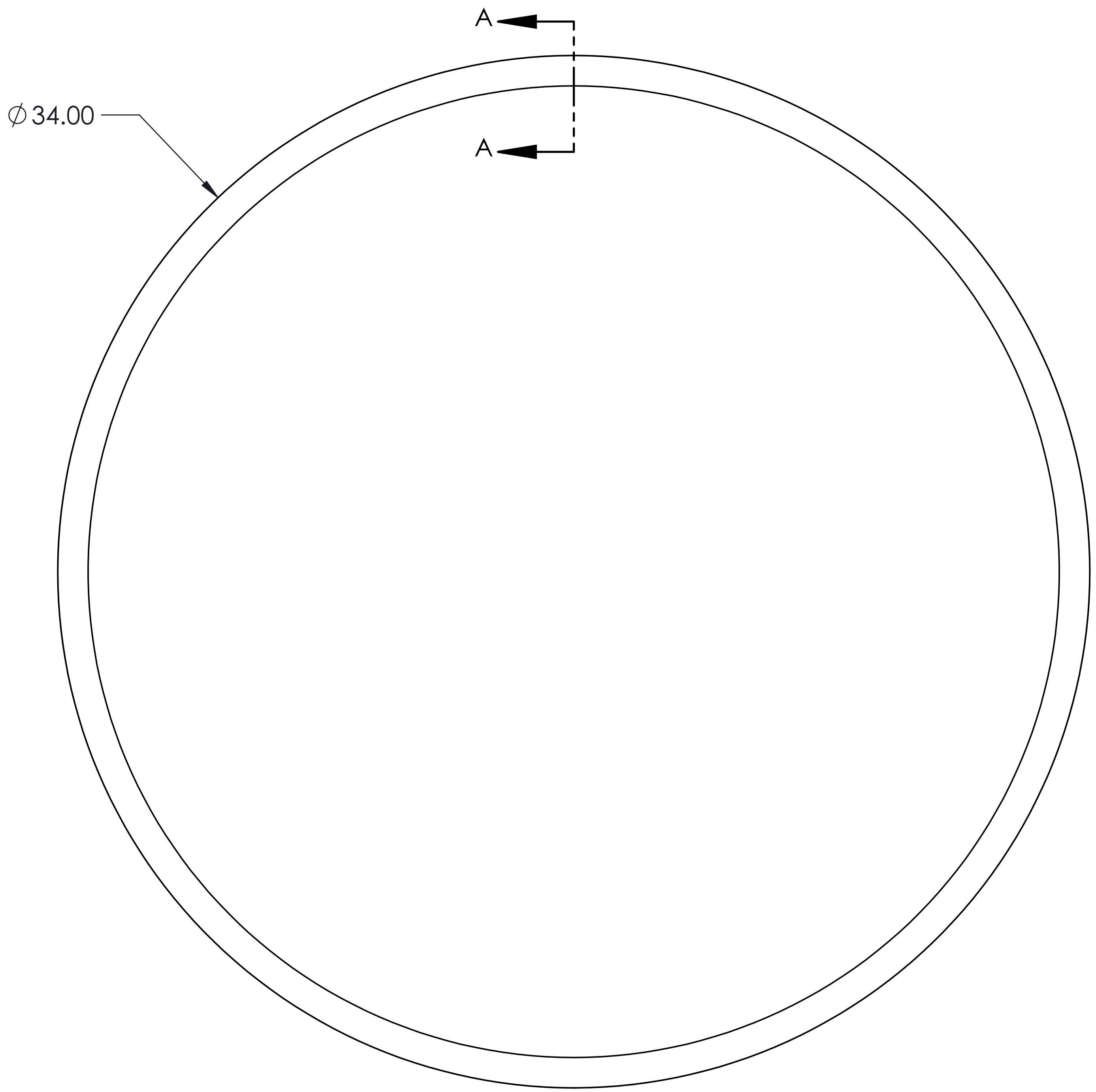
TRIPLE E RECREATIONAL VEHICLES

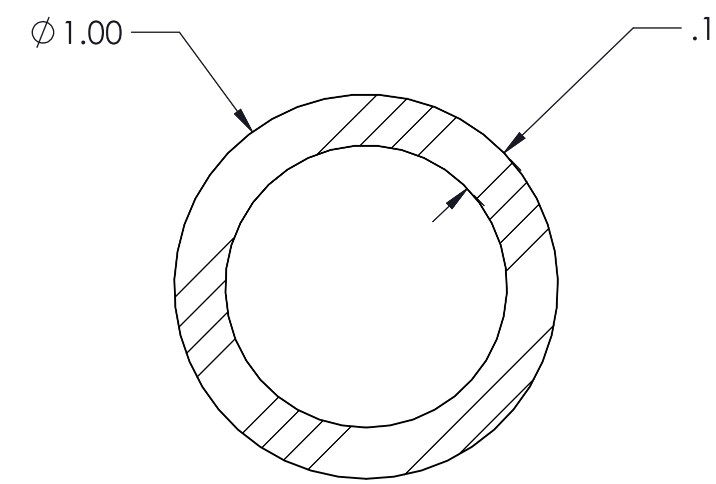
LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in + ANGLES 1° +		TITLE: Bus Wheel Bar UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B		DRAWING NO:	SHEET:	REV:
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	M00021	1 OF 1	-
START: -	REPLACED: -			

M00021 November 30, 2017 11:27:15 AM froesem5

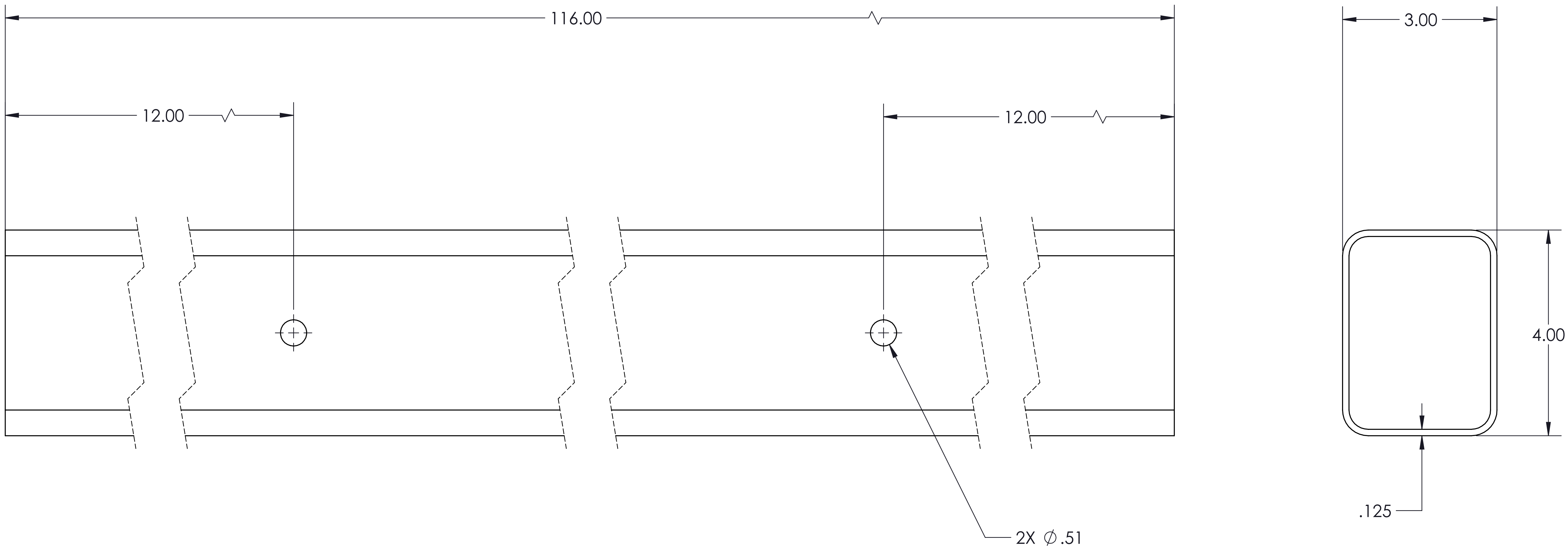
M00022 November 30, 2017 11:30:54 AM froesem5





SECTION A-A
SCALE 2 : 1

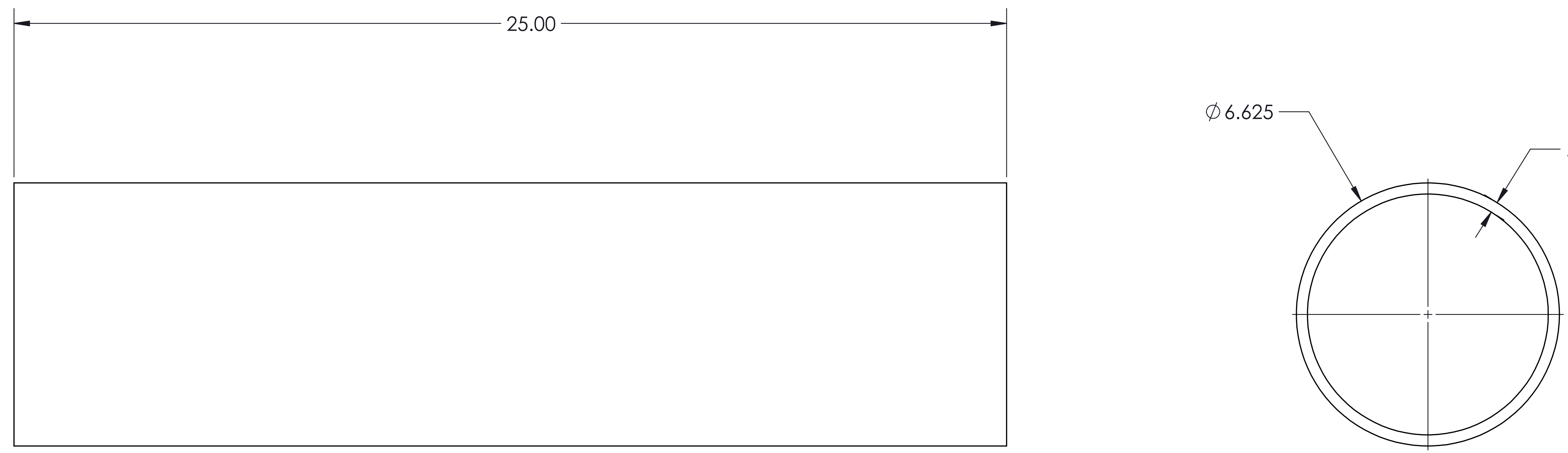
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°		TITLE: Bus Wheel Ring UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B		DRAWING NO:	SHEET:	REV:
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	M00022	1 OF 1	-
START: -	REPLACED: -			



TRIPLE E RECREATIONAL VEHICLES — LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°		TITLE: Long Frame Support UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B		DRAWING NO: M00023	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

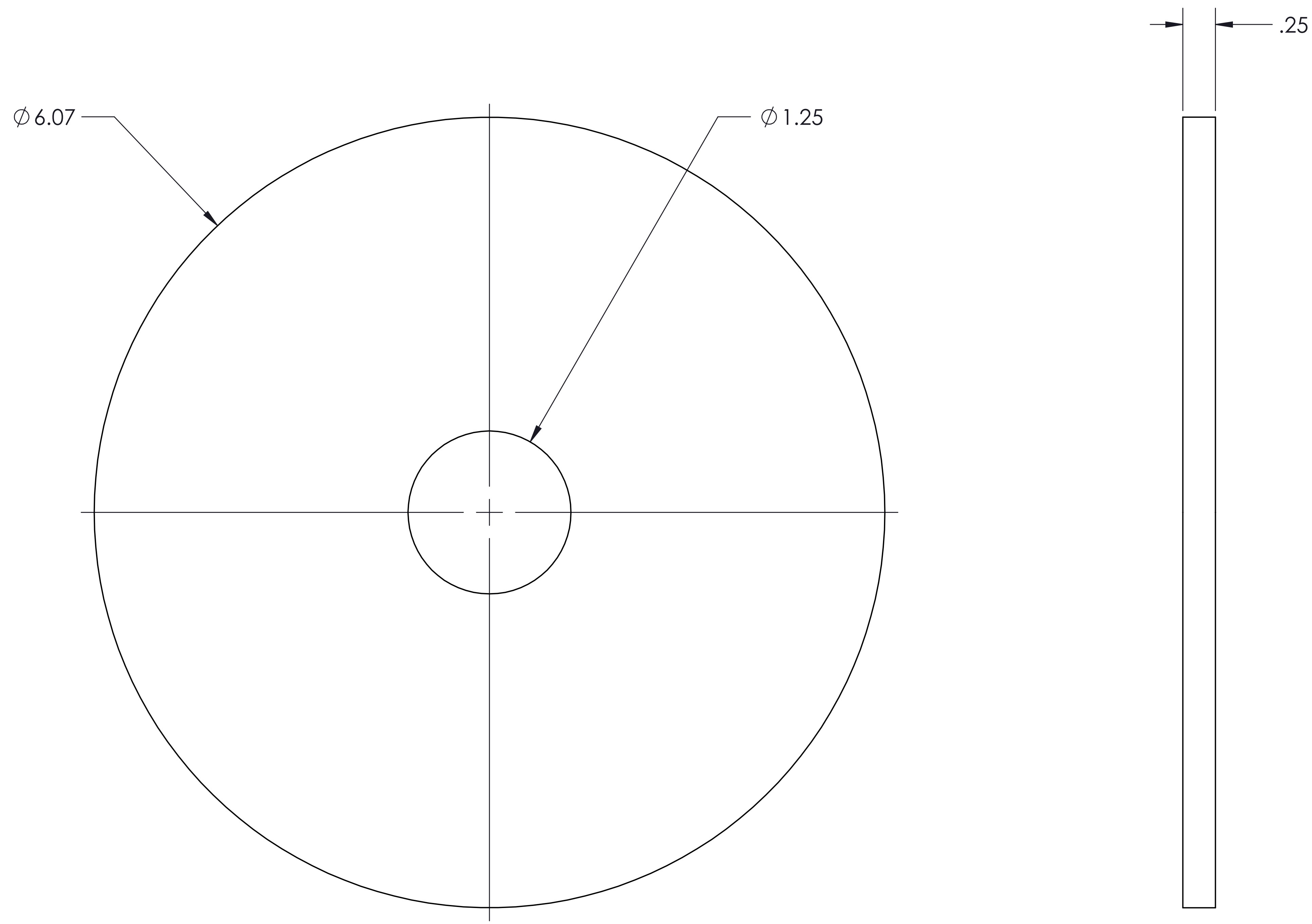
M00023 November 30, 2017 11:44:57 AM froesem5



TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°	TITLE: Bottom Roller Tube UNIT
MANUFACTURED BY: -	
MATERIAL: ASTM A-500 Gr. B	DRAWING NO: M00024
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - / REPLACED: -	REV: -

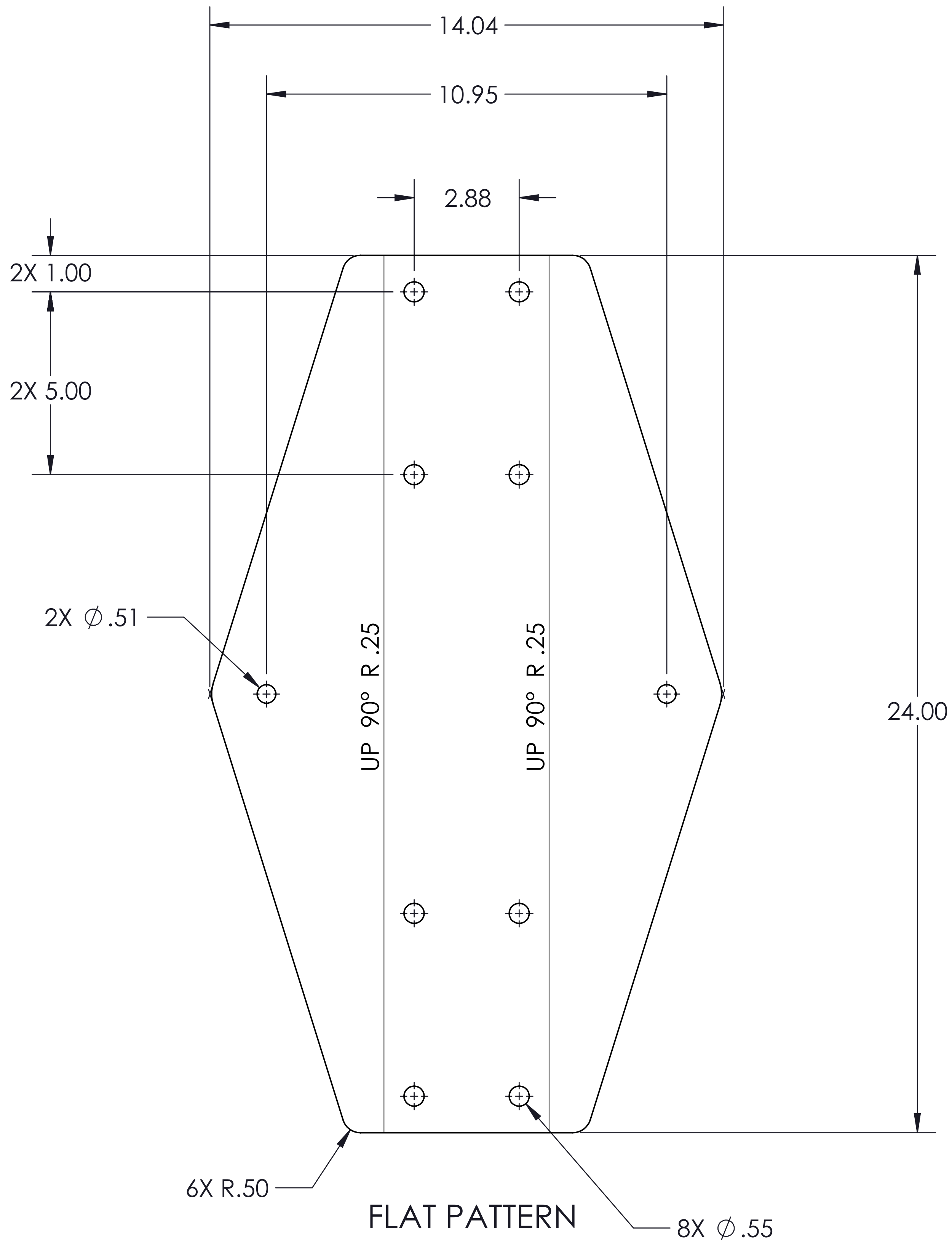


M00025 November 30, 2017 11:50:16 AM froesem5

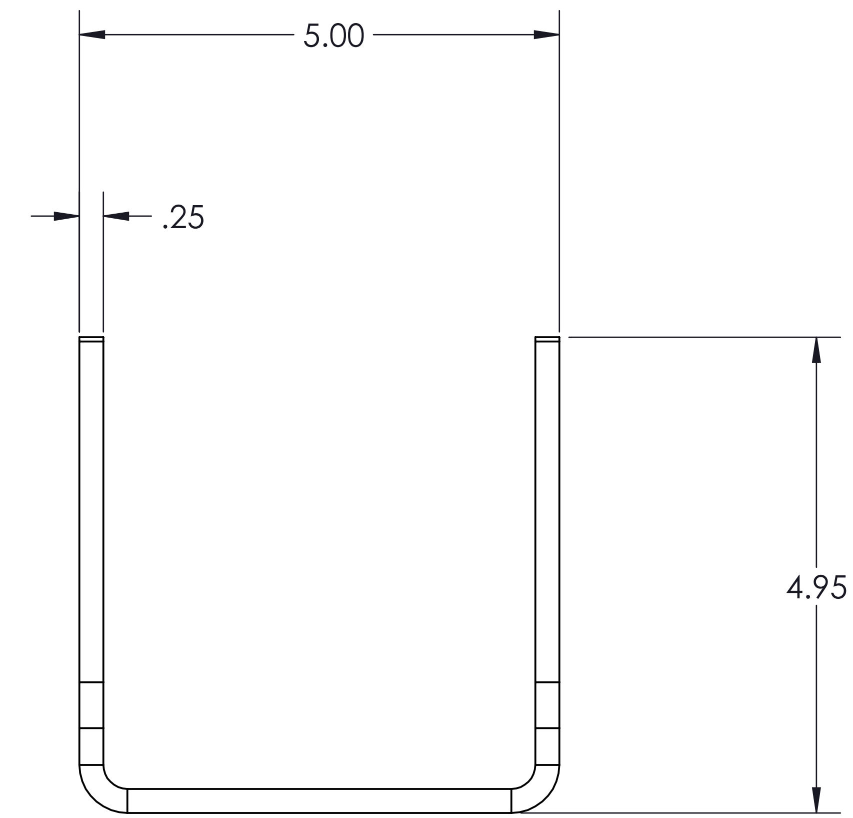
TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°		TITLE: Bottom Roller Plate		
MANUFACTURED BY: -		UNIT		
MATERIAL: ASTM A36 Steel Plate		DRAWING NO:	SHEET:	REV:
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	M00025	1 OF 1	-
START: -	REPLACED: -			



FLAT PATTERN



FORMED PART

NOTE:
FORMED PART DIMENSIONS SUPERSEDE FLAT PATTERN
FINISH: POWDER COAT BLUE

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED):
DIMENSIONS 0.0625 in ANGLES 1°

MANUFACTURED BY: -

MATERIAL: ASTM A36 Steel Plate

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

TITLE: Walking Axle Base

UNIT

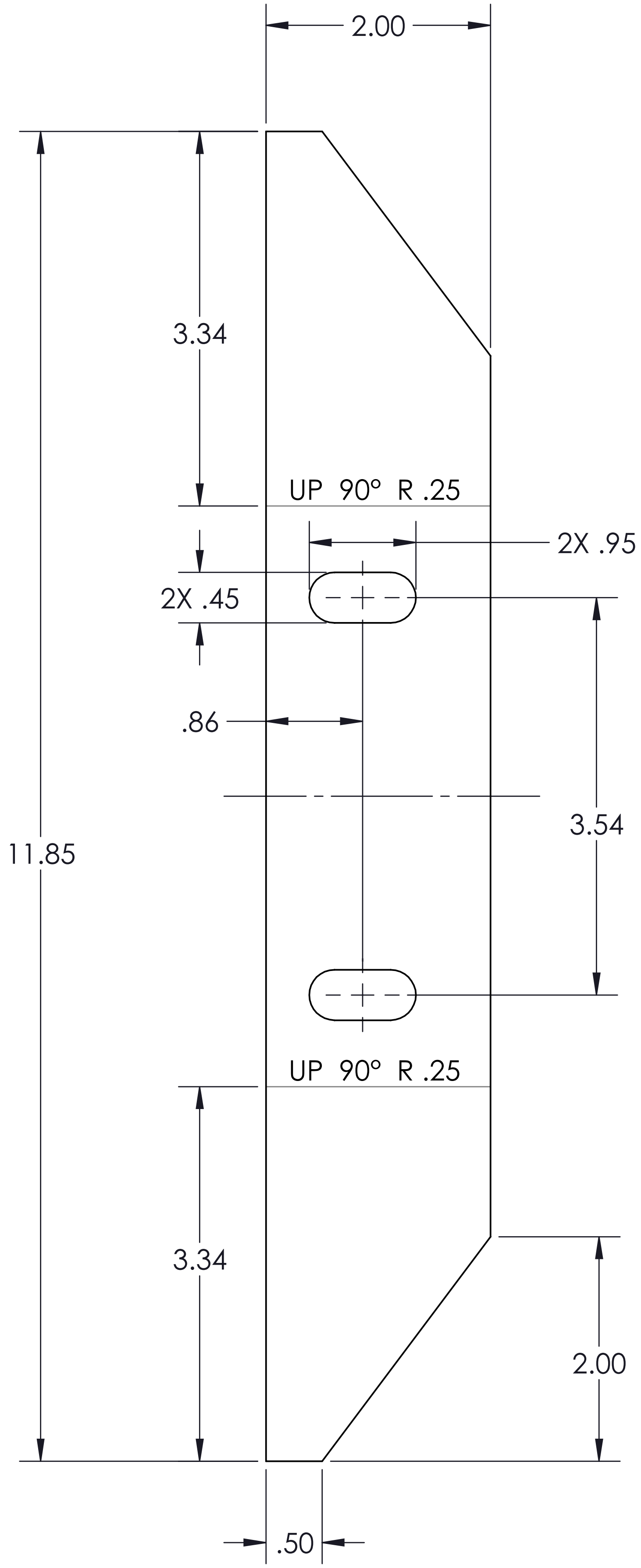
DRAWING NO: M00026

SHEET: 1 OF 1

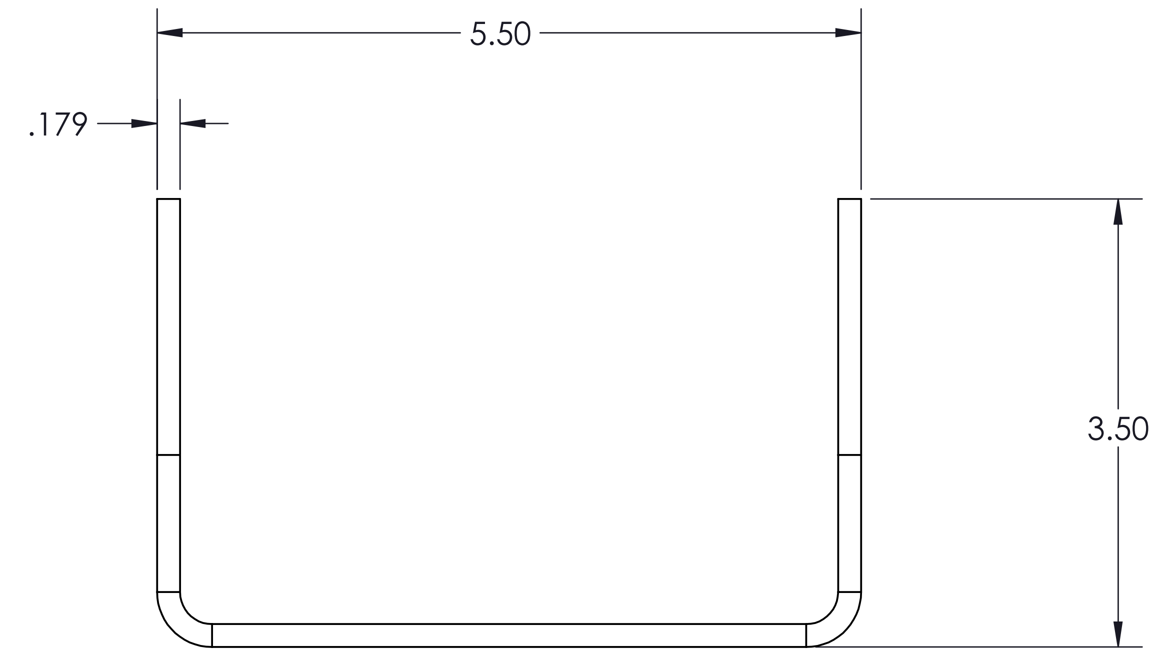
REV: -

M00026 November 30, 2017 11:58:49 AM froesem5

M00027 November 30, 2017 12:10:33 PM froesem5



FLAT PATTERN



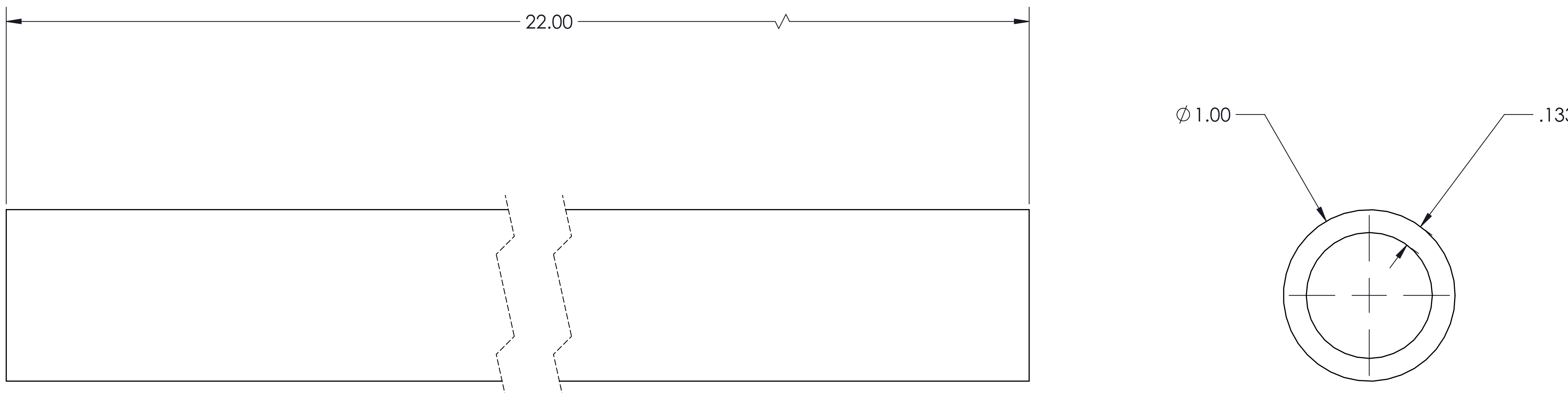
FORMED PART

NOTE:
FORMED PART DIMENSIONS SUPERSEDE FLAT PATTERN

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°		TITLE: Gearbox Support UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A36 Steel Plate		DRAWING NO: M00027	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

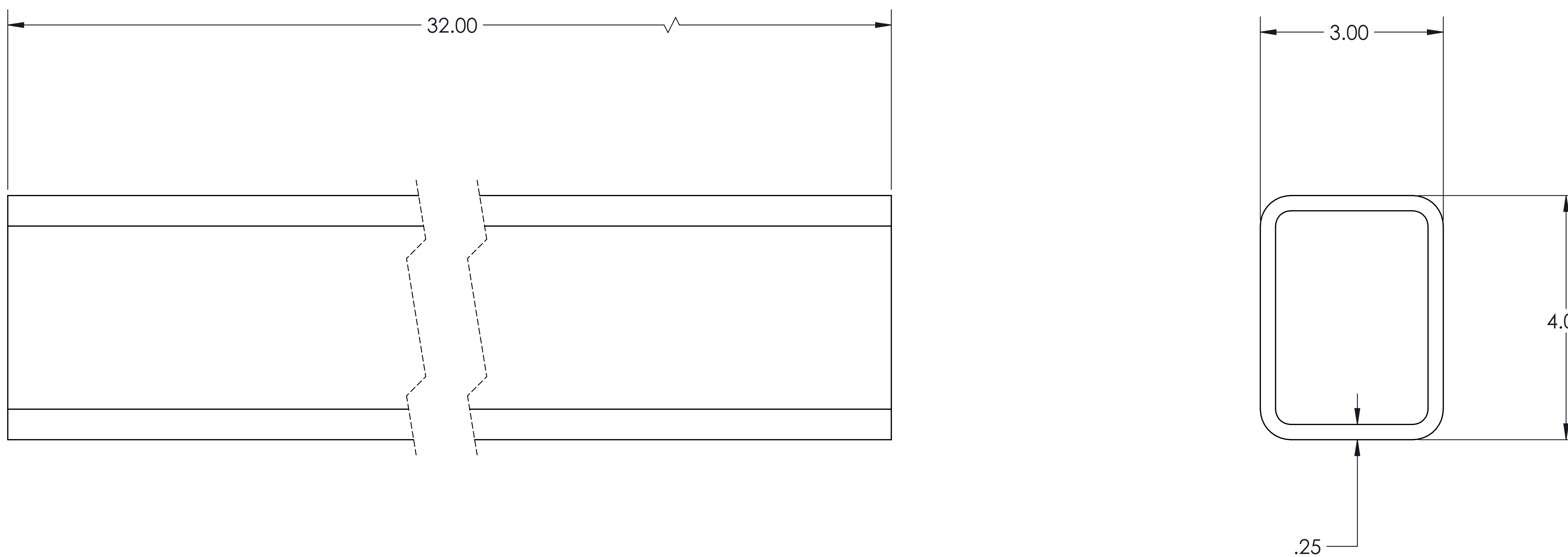


TRIPLE E RECREATIONAL VEHICLES — LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°	TITLE: Handle UNIT
MANUFACTURED BY: -	
MATERIAL: ASTM A-500 Gr. B	DRAWING NO: M00028
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - / REPLACED: -	REV: -

M00028 November 30, 2017 12:13:53 PM froesem5

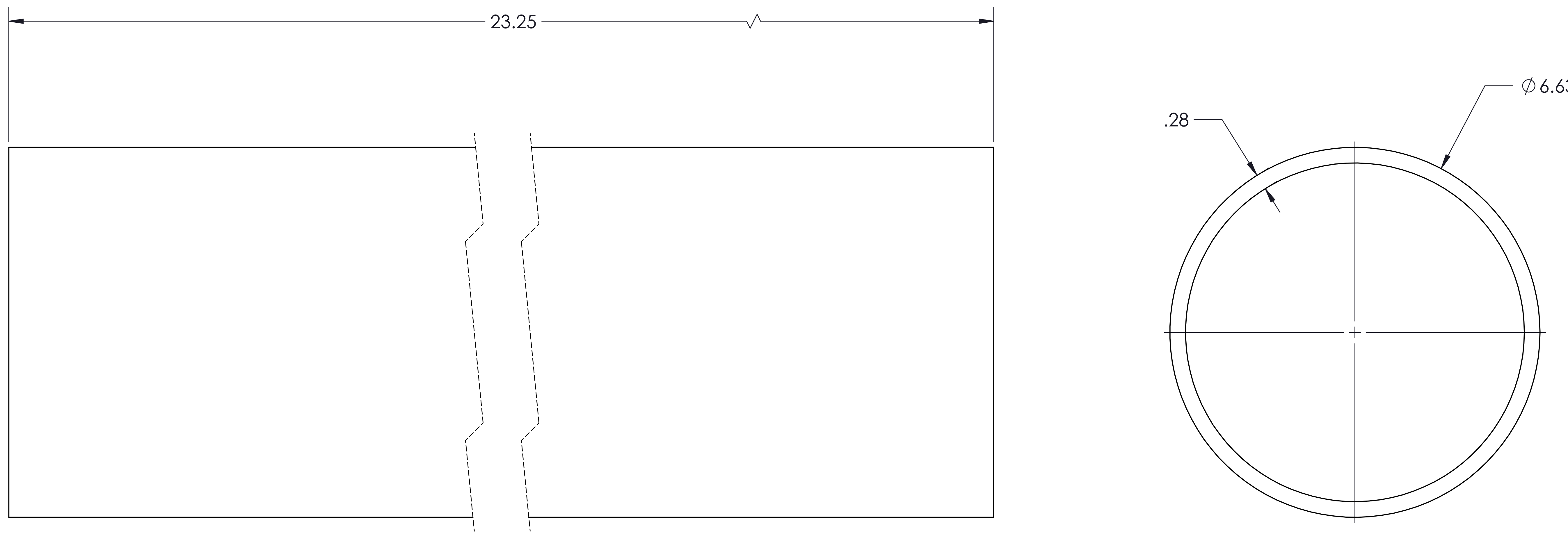
M00029 November 30, 2017 12:15:58 PM froesem5



TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Welded Side Arm UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B		DRAWING NO: M00029	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			



M00030 November 30, 2017 12:18:38 PM froesem5

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°		TITLE: Bottom Roller Driven Tube UNIT		
MANUFACTURED BY: -				
MATERIAL: ASTM A-500 Gr. B		DRAWING NO: M00030	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			

SECTION A-A

NOTE:
COAT ITEM 2 WITH PART NUMBER M00042 (RUBBER COATING) AS REQUIRED

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	M00003	Top Roller Plate	2
2	M00002	Top Roller Tube	1

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED):
DIMENSIONS 0.0625 in ANGLES 1°

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

TITLE: Top Roller WLDMT

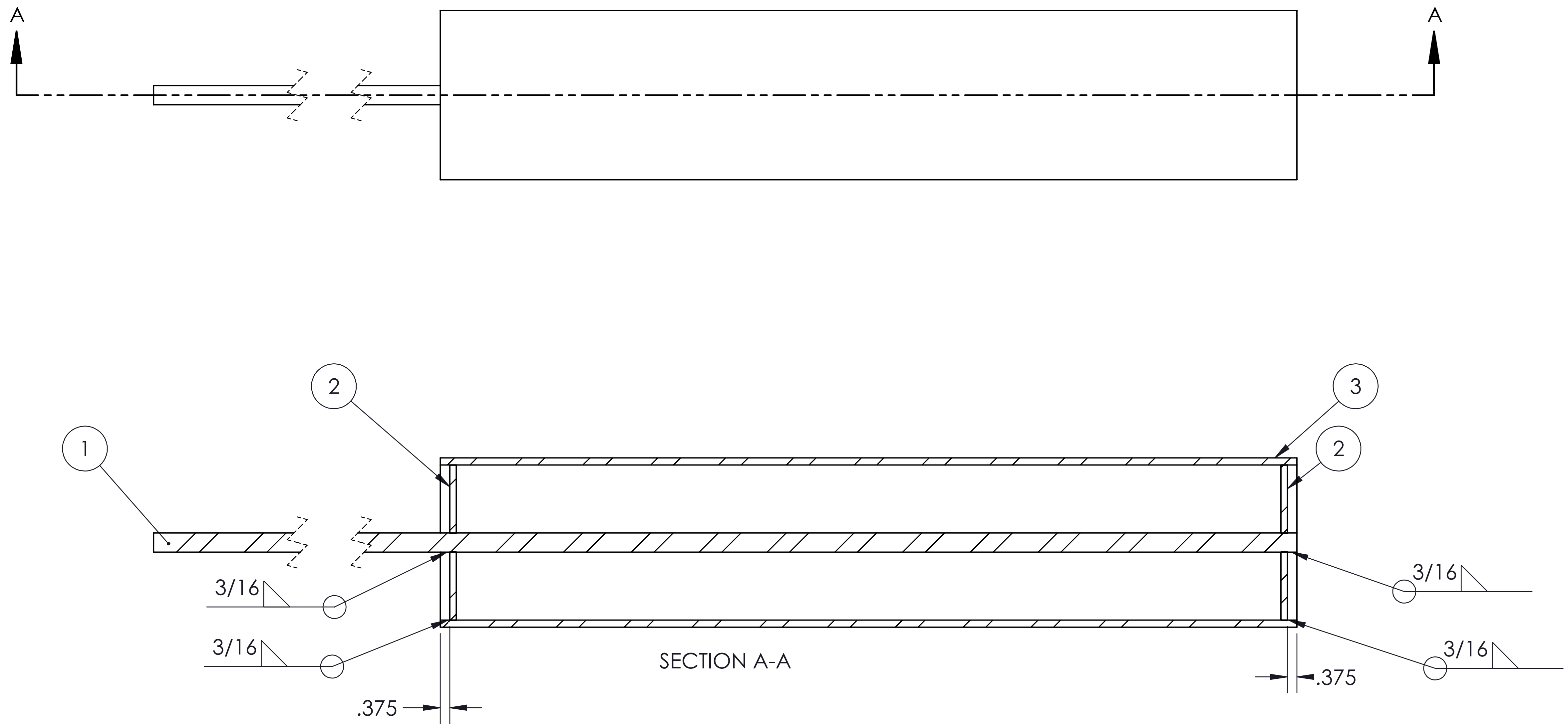
UNIT

DRAWING NO: M00031

SHEET: 1 OF 1

REV: -

M00031 November 30, 2017 12:55:30 PM froesem5



NOTE:
FINISH ITEM 3 WITH PART NUMBER M00042 (RUBBER COATING) AS REQUIRED

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	1346K36	McMaster Carr	Top Roller Shaft	1
2	M00003		Top Roller Plate	2
3	M00002		Top Roller Tube	1

TRIPLE E RECREATIONAL VEHICLES

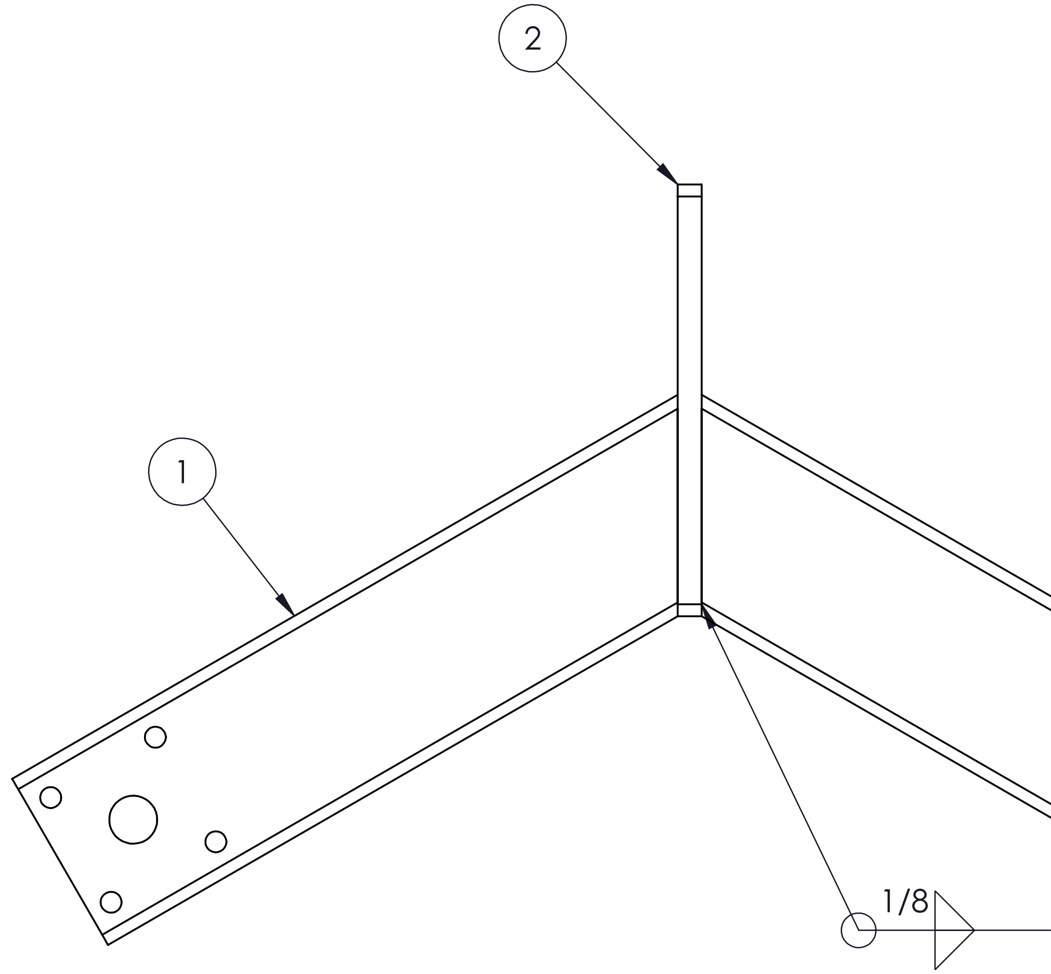
LEISURE TRAVEL VANS

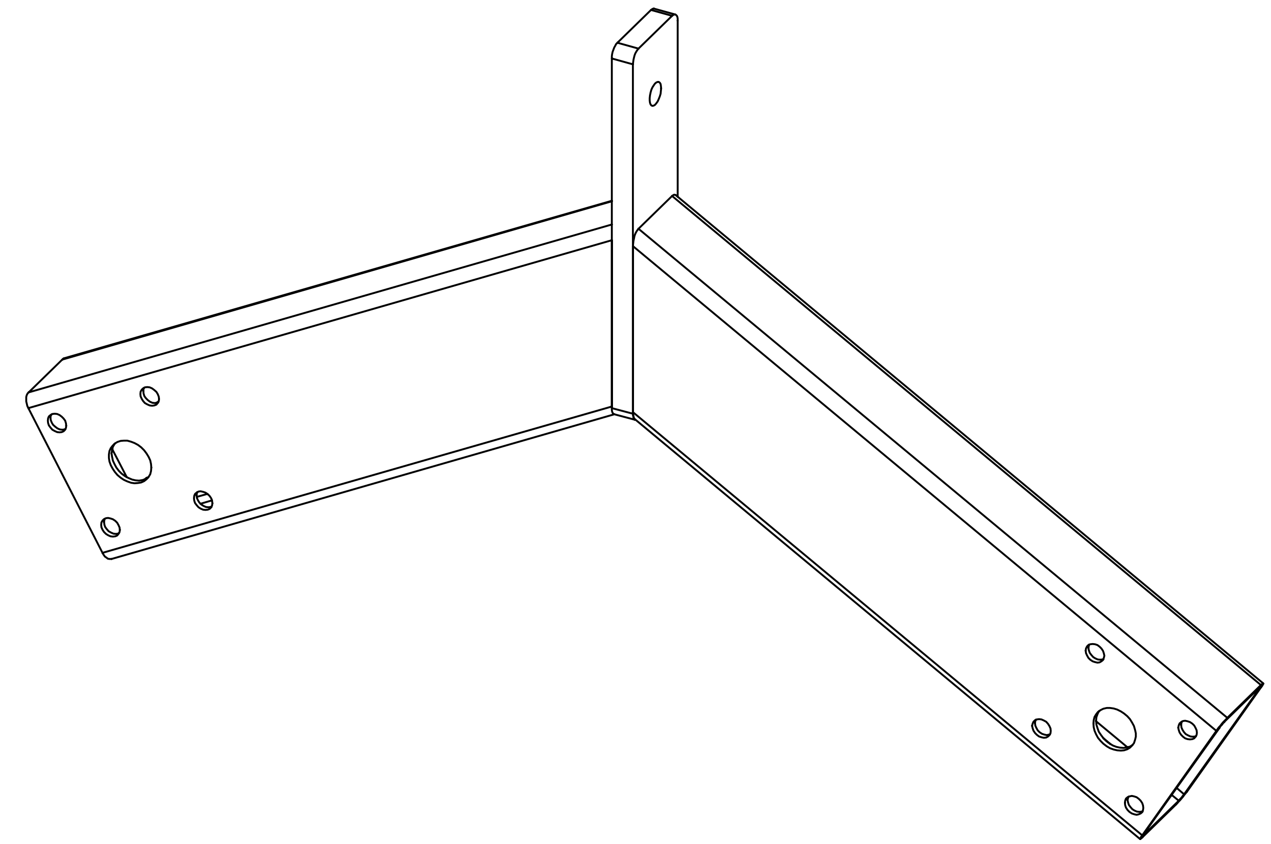
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±
MANUFACTURED BY: -
MATERIAL:
DRAWN BY: MFROESE | DRAW DATE: 29/11/2017
START: - | REPLACED: -

TITLE: Top Roller W/ Shaft
UNIT
DRAWING NO: M00032
SHEET: 1 OF 1
REV: -

M00032 Friday, December 1, 2017 7:07:26 PM Matt

M00033 November 30, 2017 1:02:34 PM froesem5



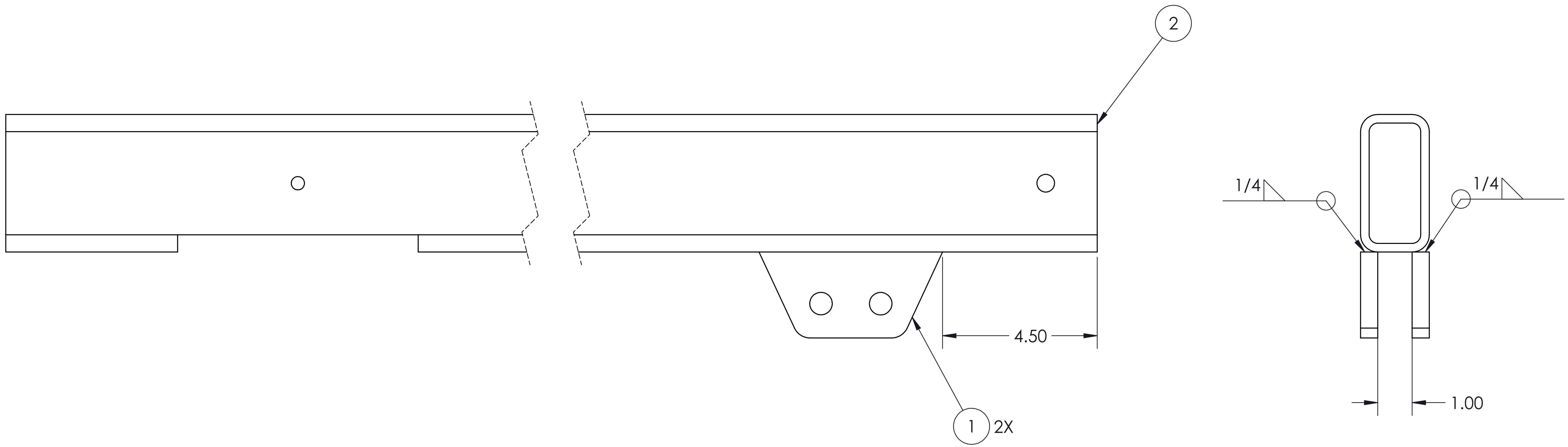


ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	M00004	Tube Arm	2
2	M00005	Hanging Plate	1

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°		TITLE: Top Roller Hanger WLDMT UNIT		
MANUFACTURED BY: -				
MATERIAL:		DRAWING NO: M00033	SHEET: 1 OF 1	REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017			
START: -	REPLACED: -			



FINISH: POWDER COAT BLUE

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	M00009	Bracket-Top Arm	2
2	M00006	Swinging Arm	1

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°

TITLE: Swinging Arm WLDMT

UNIT

MANUFACTURED BY: -

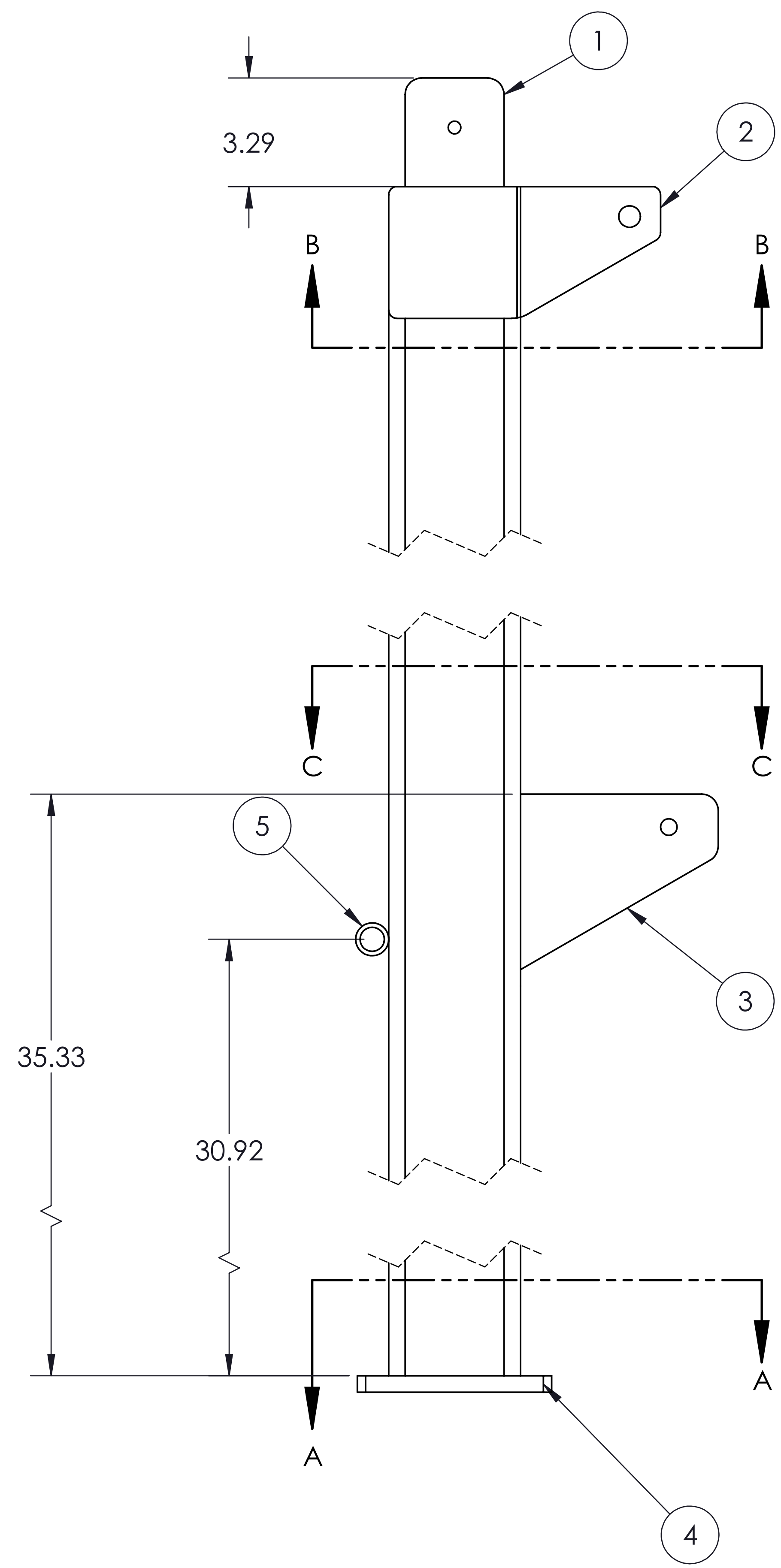
MATERIAL:

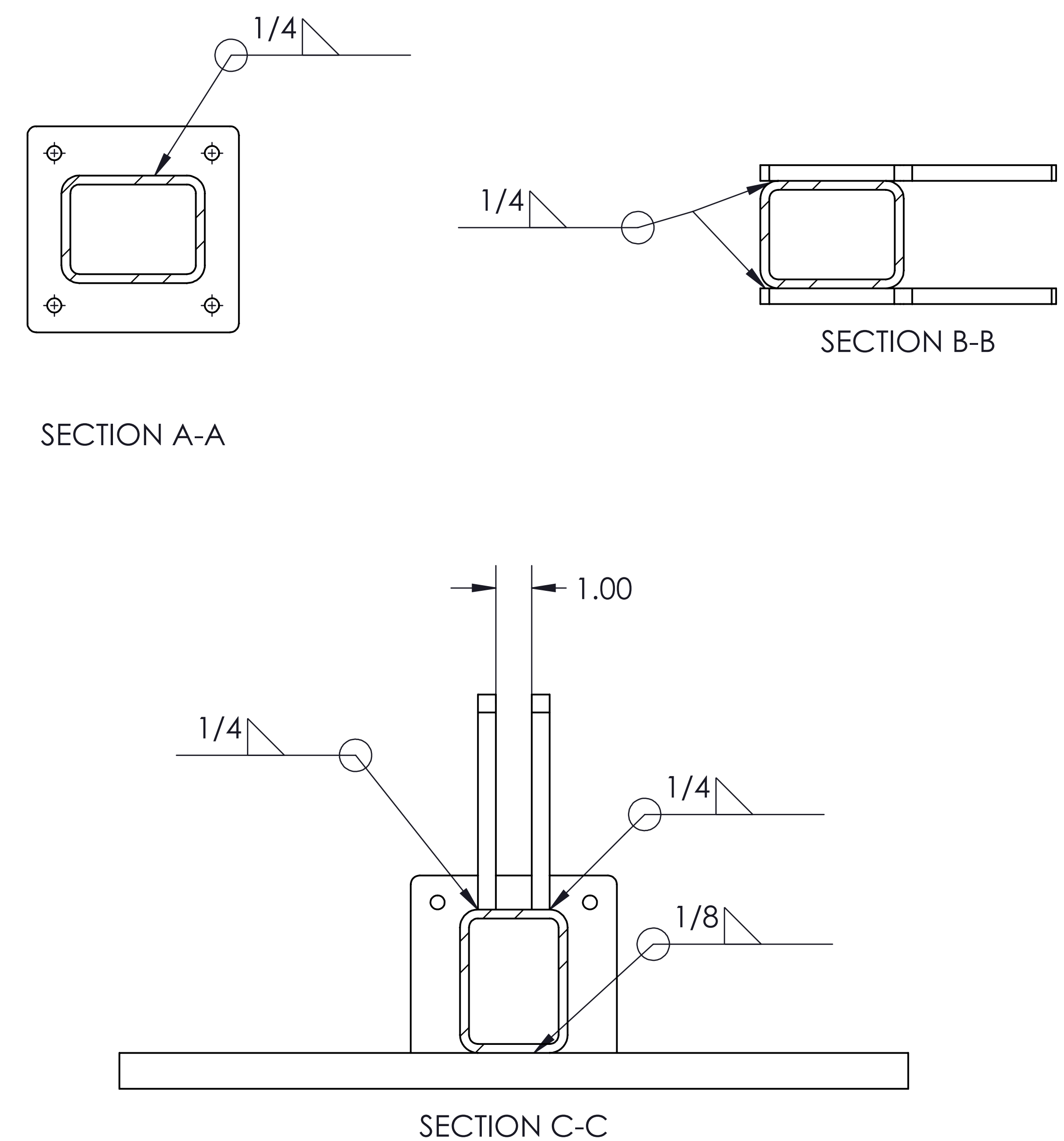
DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

DRAWING NO: M00034 | SHEET: 1 OF 1 | REV: -

M00034 November 30, 2017 1:12:41 PM froesem5





NOTE:
FINISH: POWDER COAT BLUE

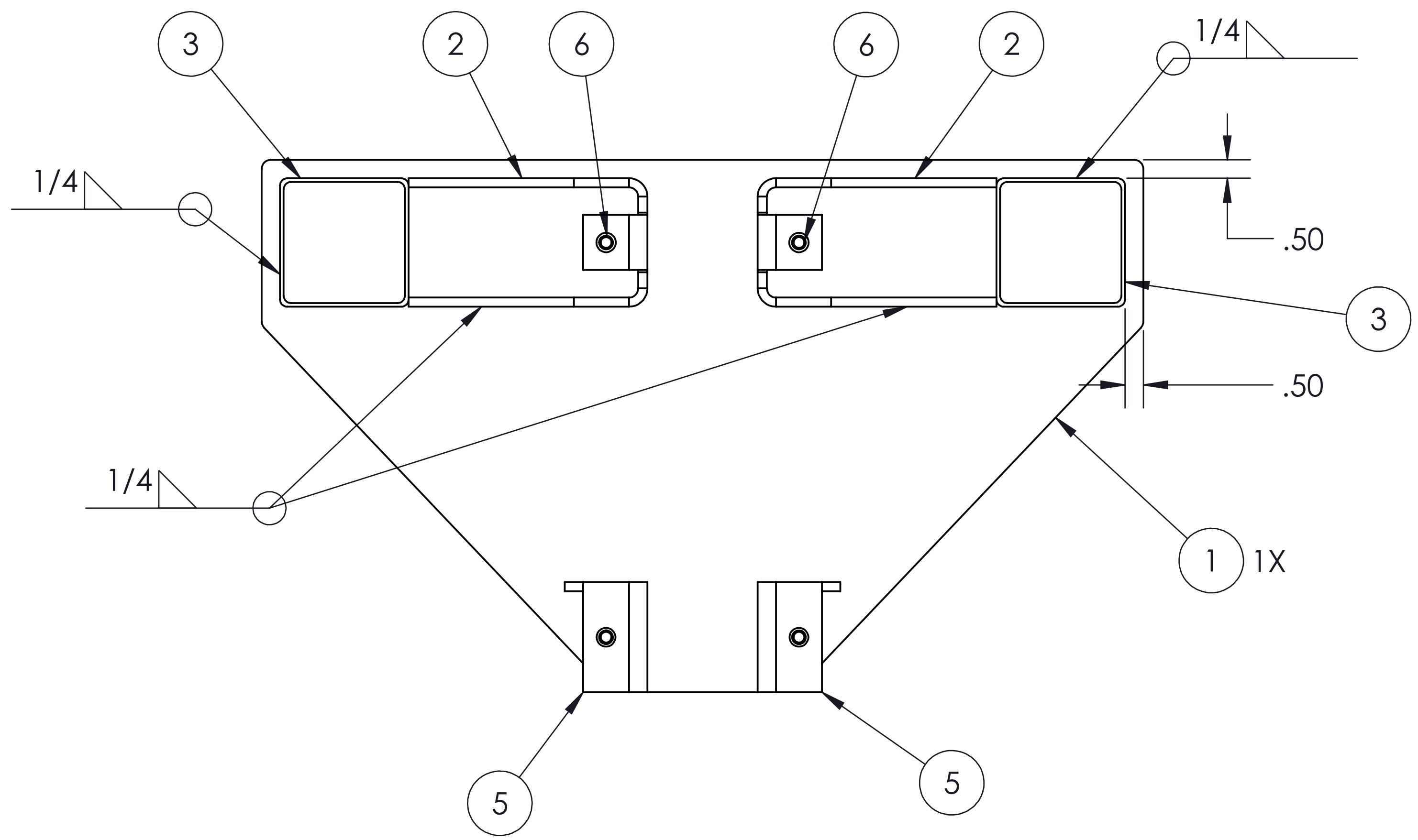
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	M00007	Side Support Arm	1
2	M00008	Safety Bracket	2
3	M00010	Bracket-Side Arm	2
4	M00018	Mounting Plate	1
5	M00028	Handle	1

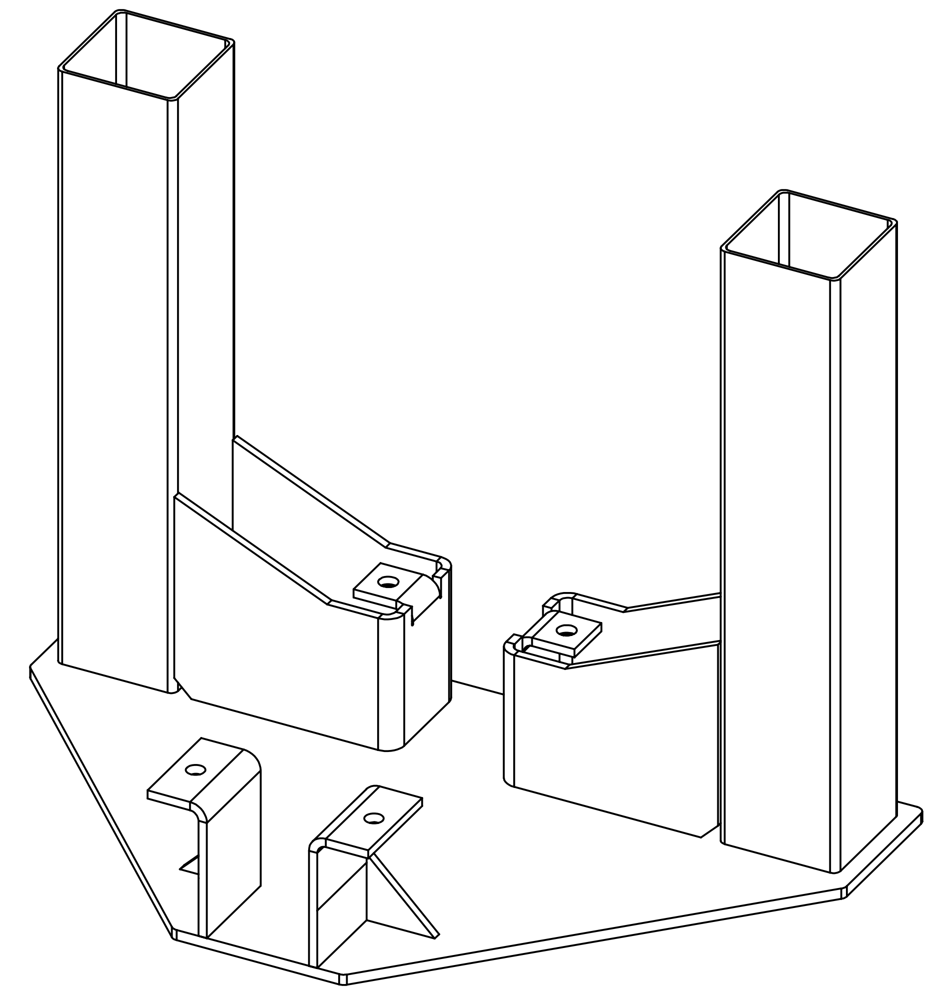
TRIPLE E RECREATIONAL VEHICLES

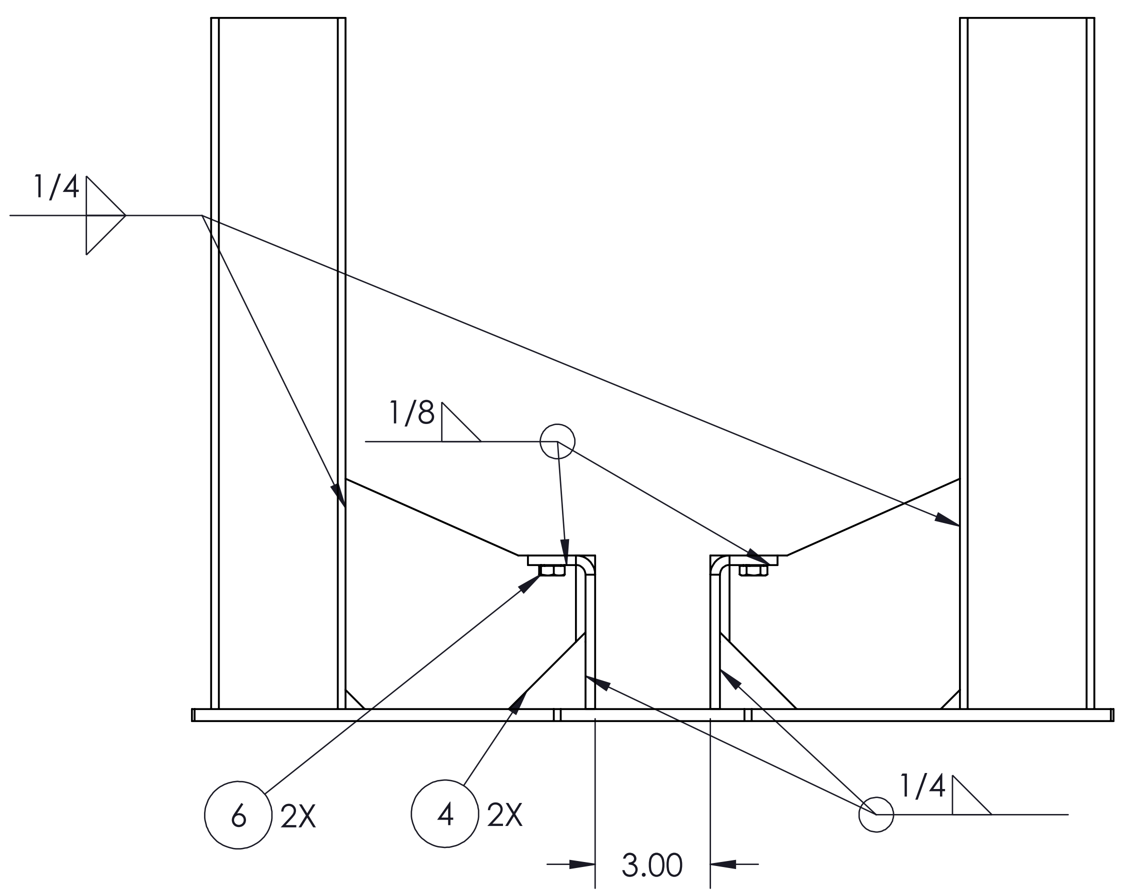
LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ANGLES 1°	TITLE: Side Arm WLDMT UNIT
MANUFACTURED BY: -	
MATERIAL:	DRAWING NO: M00035
DRAWN BY: MFROESE / DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - / REPLACED: -	REV: -

M00035 November 30, 2017 1:33:30 PM froesem5







NOTE:
FINISH: POWDER COAT BLUE
MASK ALL THREADS PRIOR TO POWDER COATING

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00013		Backing Plate	1
2	M00012		Gusset	2
3	M00011		Square Shaft	2
4	M00015		Lower Gusset	2
5	M00016		Retaining Shaft Bracket	2
6	93560A160	McMaster Carr	3/8" Weld nut	4

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED):
DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

TITLE: Retaining Shaft WLDMT

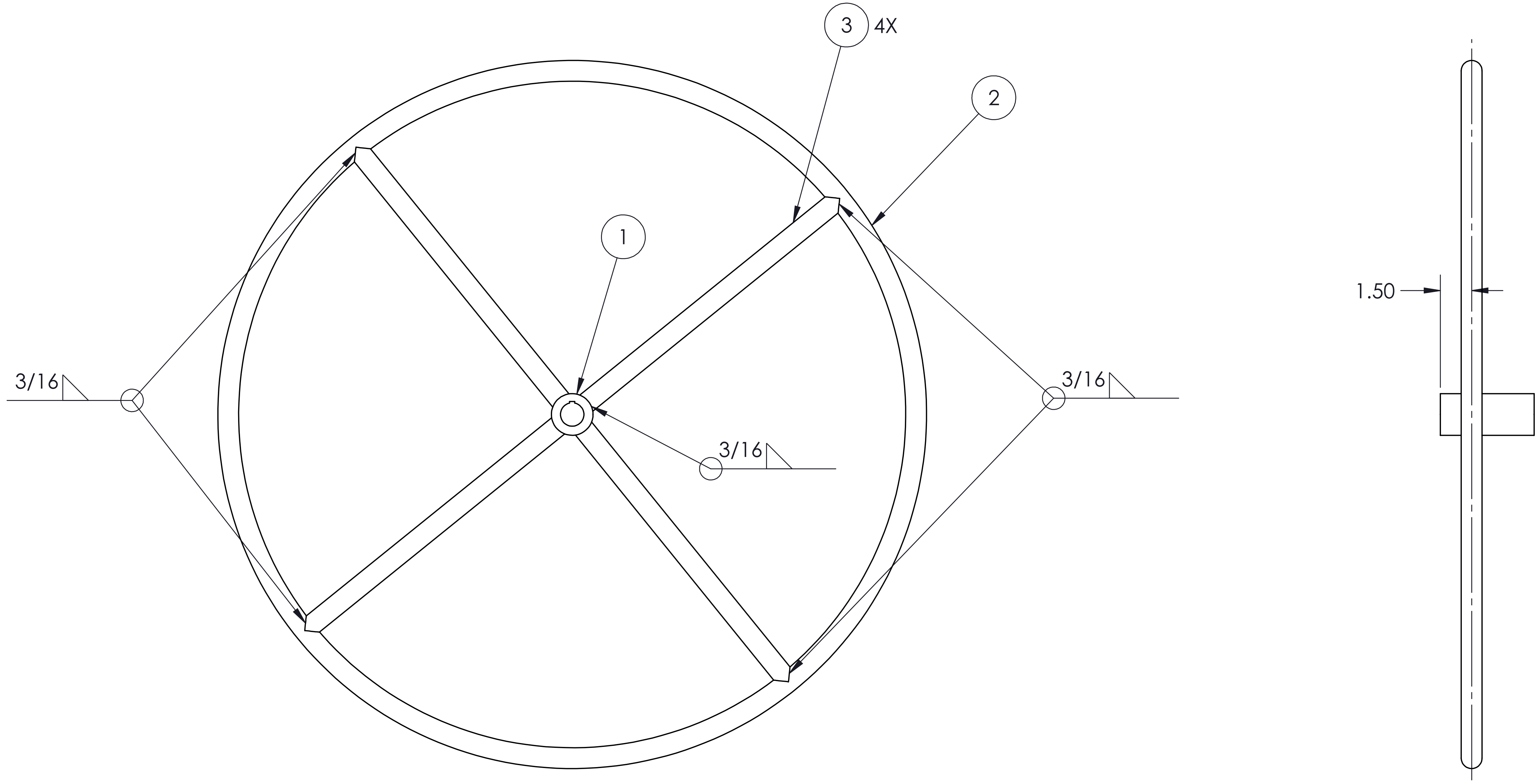
UNIT

DRAWING NO: M00036

SHEET: 1 OF 1

REV: -

M00036 Friday, December 1, 2017 7:13:04 PM Matt



NOTE:
FINISH: POWDER COAT BLUE

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	6412K47	McMaster Carr	1.25" Shaft Coupling	1
2	M00022		Bus Wheel Ring	1
3	M00021		Bus Wheel Bar	4

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED):
DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

TITLE: Bus Wheel WLDMT

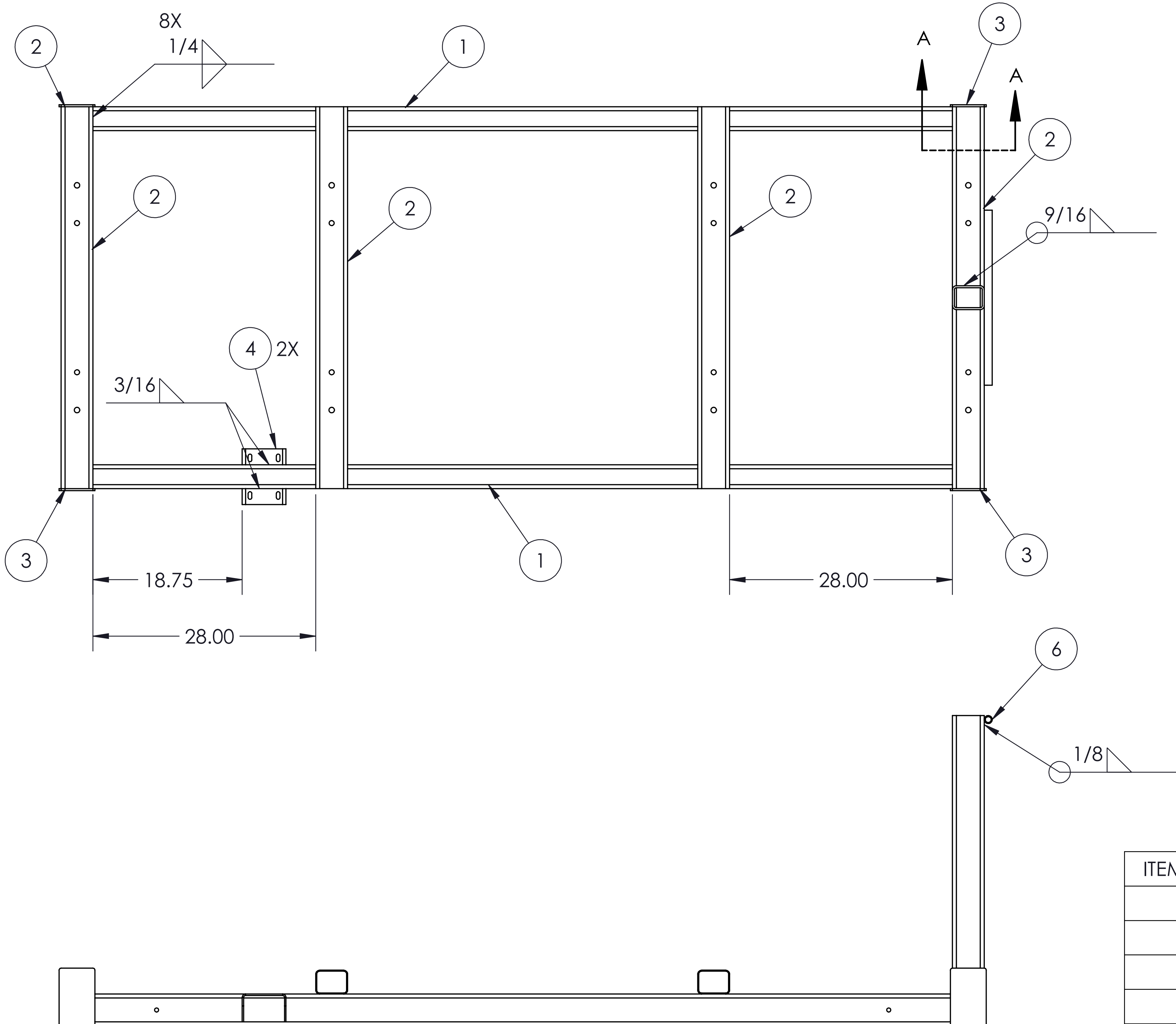
UNIT

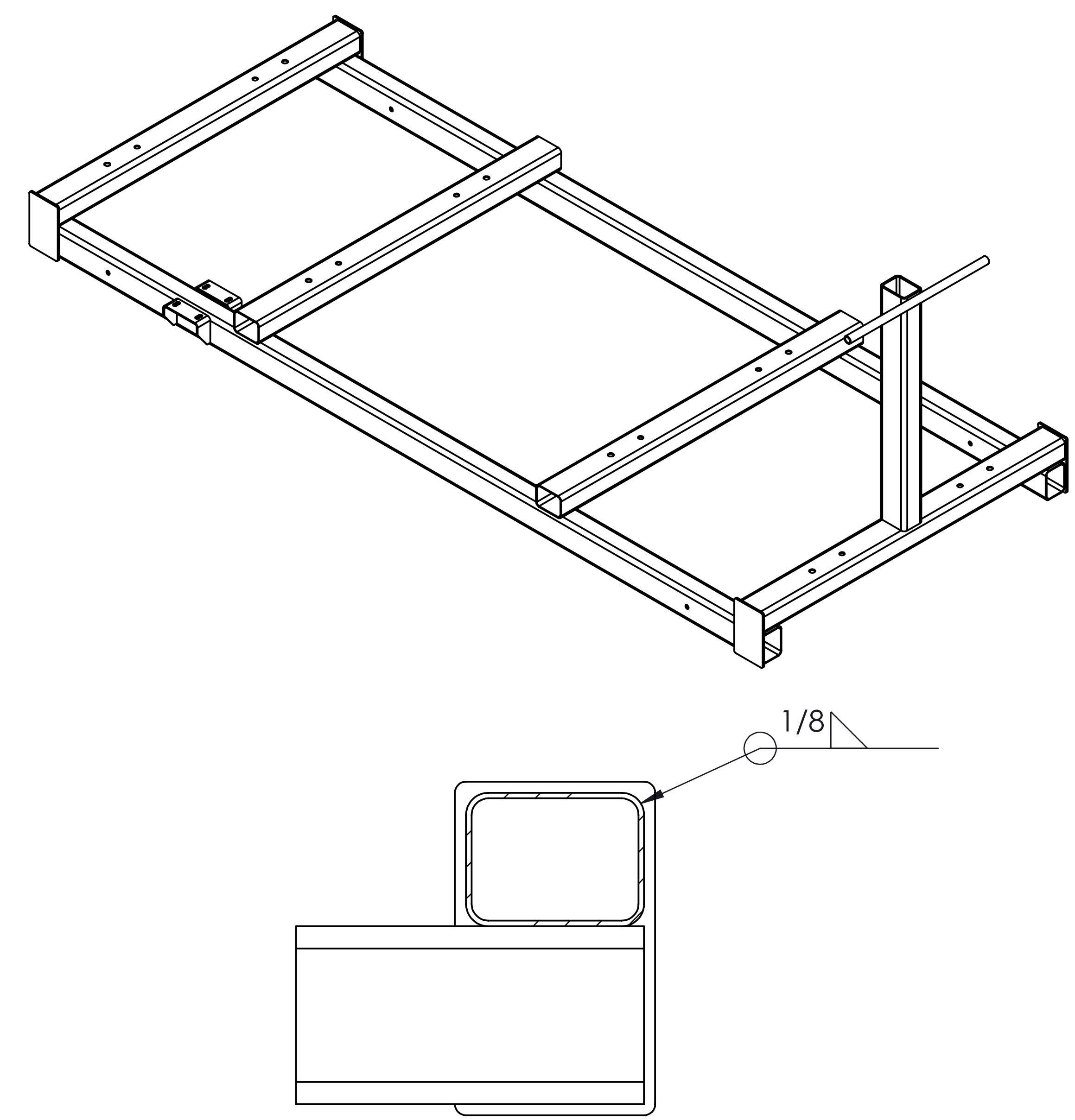
DRAWING NO: M00037

SHEET: 1 OF 1

REV: -

M00037 Friday, December 1, 2017 7:16:31 PM Matt





SECTION A-A
SCALE 1 : 4

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	M00023	Long Frame Support	2
2	M00019	Frame Cross Support	4
3	M00020	Cross Brace Support	4
4	M00027	Gearbox Support	2
5	M00029	Welded Side Arm	1
6	M00028	Handle	1

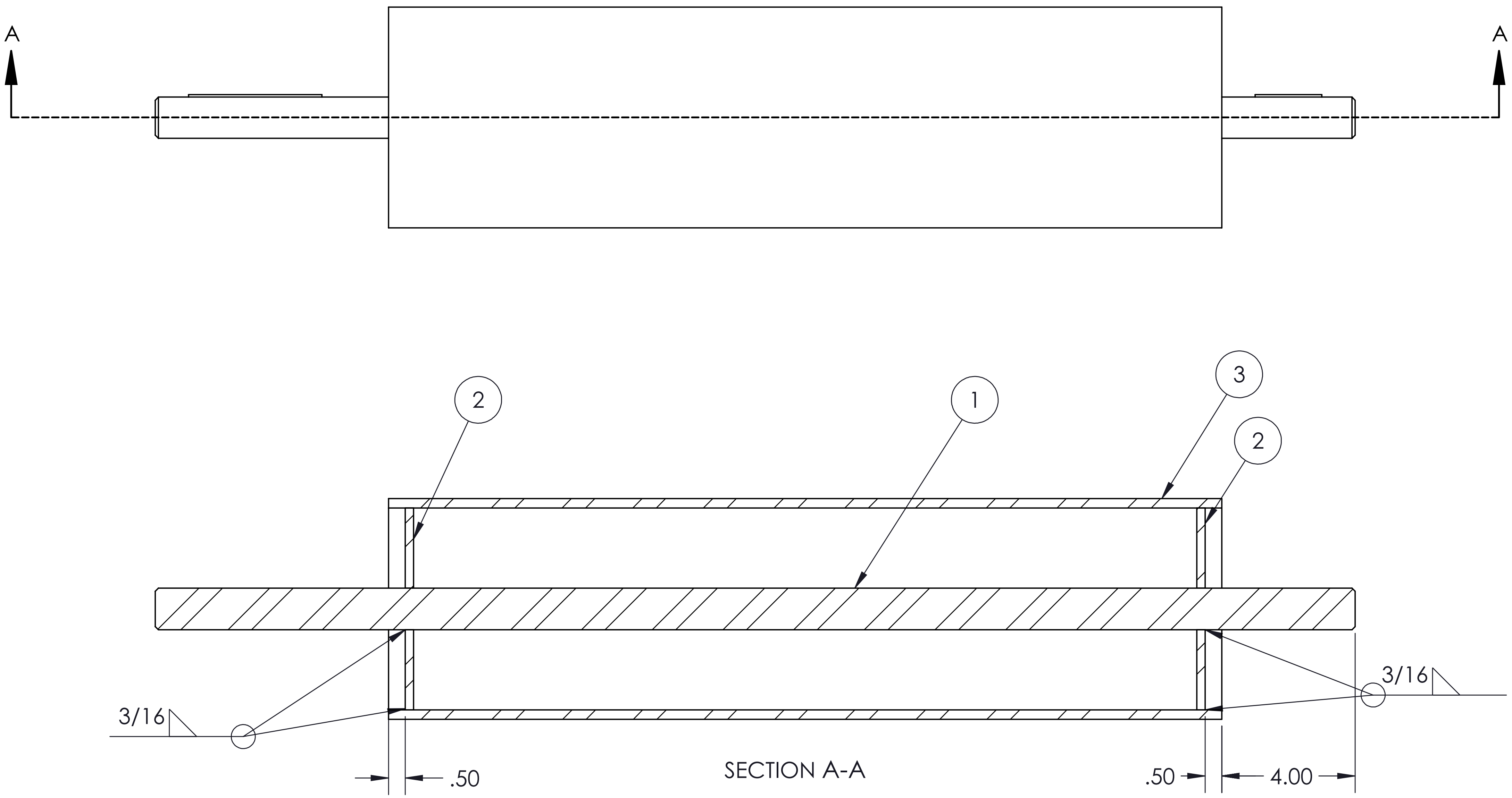
NOTE:
FINISH: POWDER COAT BLUE

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Bottom Frame WLDMT		
MANUFACTURED BY: -		UNIT		
MATERIAL:		DRAWING NO:	SHEET:	REV:
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	M00038	1 OF 1	-
START: -	REPLACED: -			

M00038 Friday, December 1, 2017 5:05:20 PM Matt



NOTE:
COAT ITEM 3 WITH PART NUMBER M00042 (RUBBER COATING) AS REQUIRED

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	6117K59	McMaster Carr	1-1/4" Shaft, 36" Lng	1
2	M00025		Bottom Roller Plate	2
3	M00024		Bottom Roller Tube	1

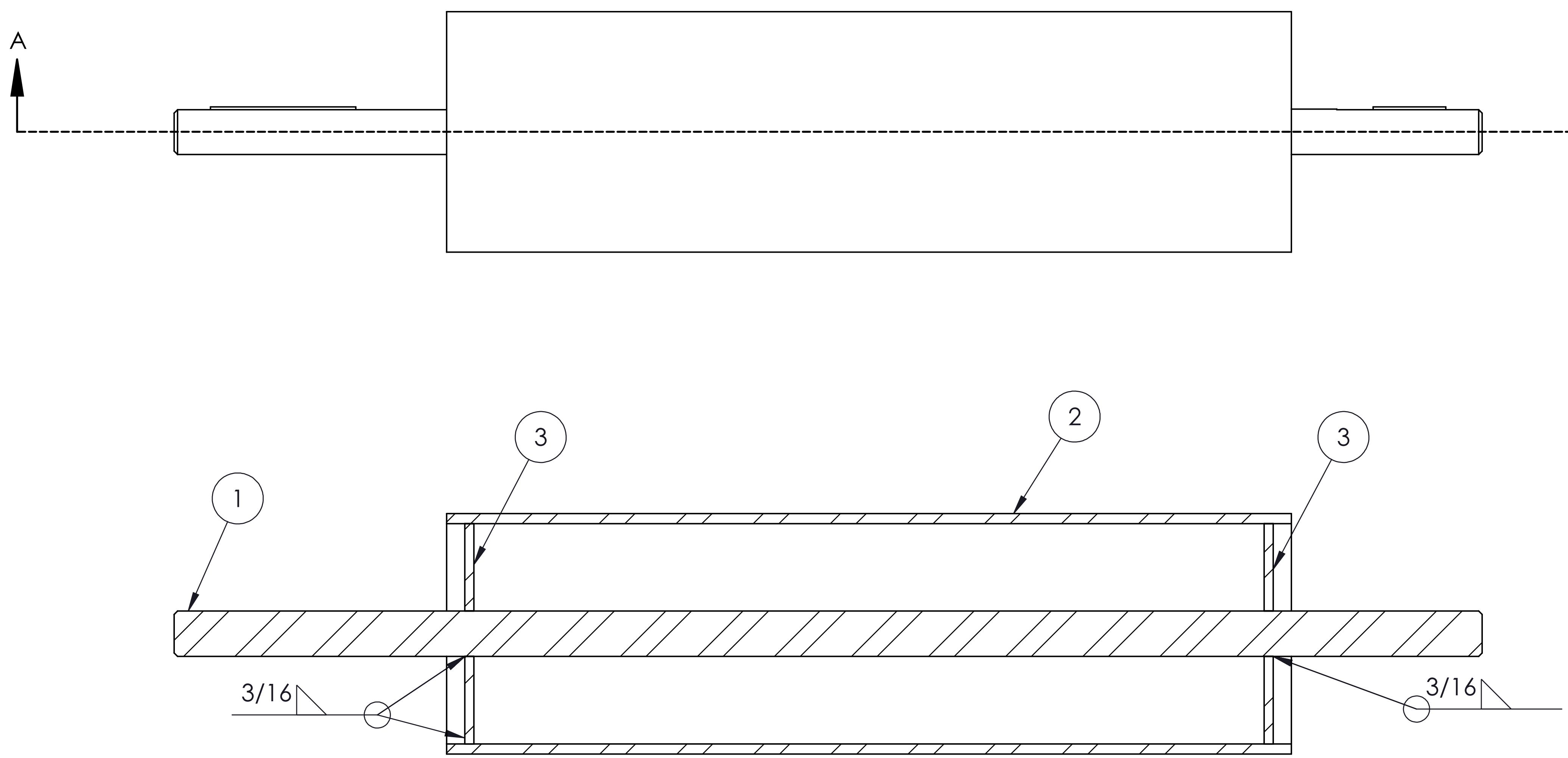
TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±
MANUFACTURED BY: -
MATERIAL:
DRAWN BY: MFROESE | DRAW DATE: 29/11/2017
START: - | REPLACED: -

TITLE: Bottom Roller WLDMT
UNIT
DRAWING NO: M00039
SHEET: 1 OF 1
REV: -

M00039 Friday, December 1, 2017 7:21:36 PM Matt



SECTION A-A

NOTE:
COAT ITEM 2 WITH PART NUMBER M00043 (RUBBER COATING) AS REQUIRED

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	6117K59	McMaster Carr	1-1/4" Shaft, 36" Lng	1
2	M00030		Bottom Roller Driven Tube	1
3	M00025		Bottom Roller Plate	2

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED):
DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

TITLE: Driven Roller WLDMT

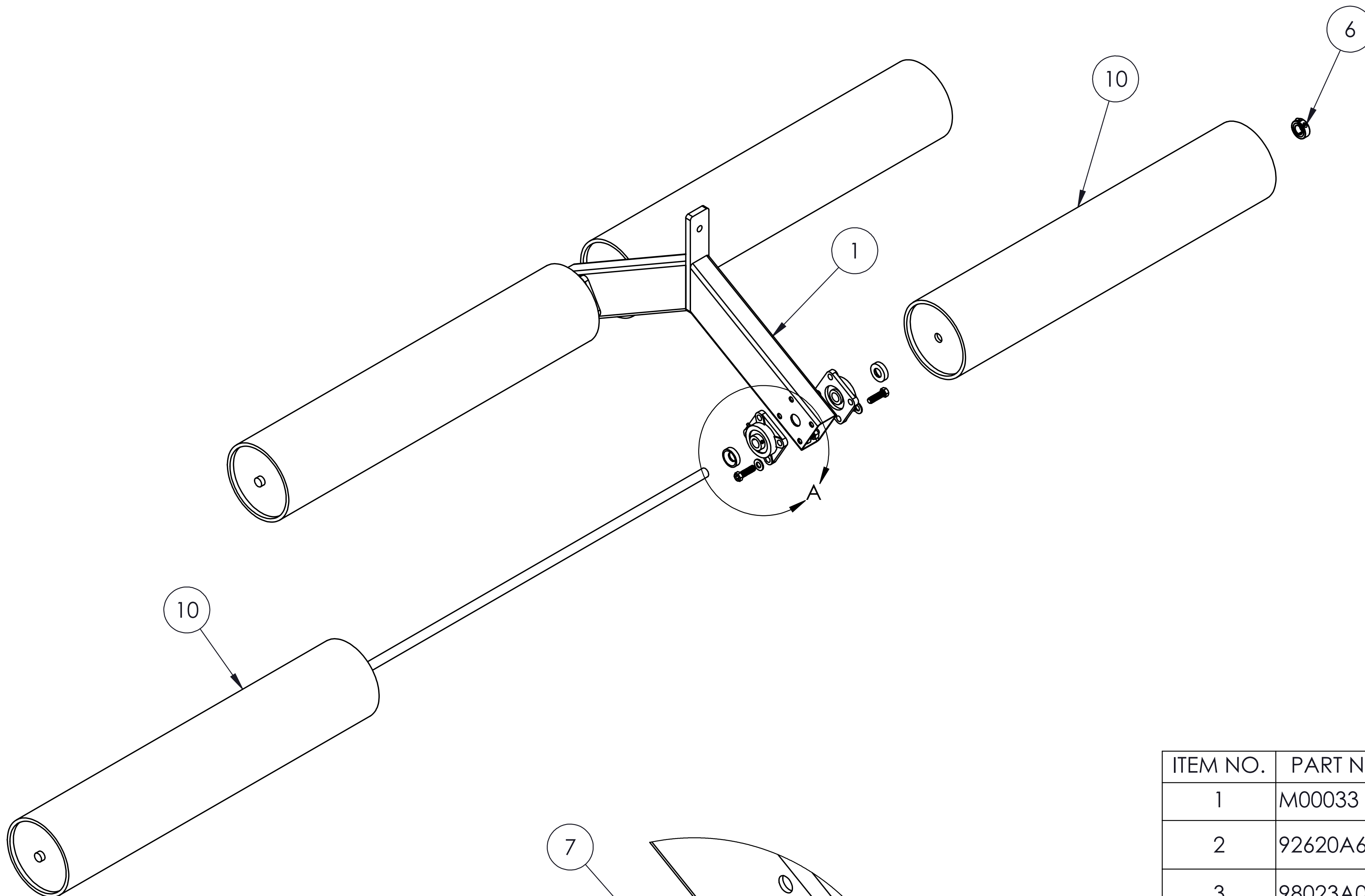
UNIT

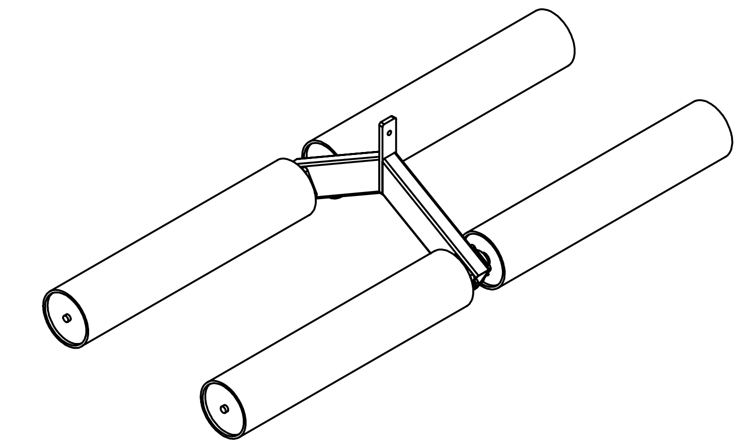
DRAWING NO: M00040

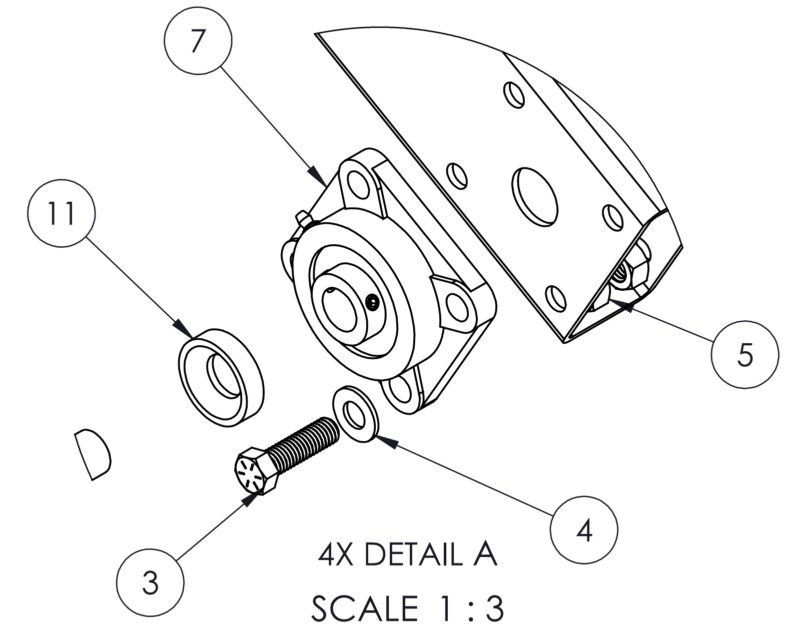
SHEET: 1 OF 1

REV: -

M00040 Friday, December 1, 2017 7:23:19 PM Matt







4X DETAIL A
SCALE 1 : 3

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00033		Top Roller Hanger WLDMT	1
2	92620A673	McMaster Carr	7/16" x 1-1/2" Bolt	16
3	98023A032	McMaster Carr	7/16" Flat Washer	16
4	94895A817	McMaster Carr	7/16" Hex Nut	16
5	6435K16	McMaster Carr	3/4" Shaft Collar	2
6	0474550	Fastenal	3/4" Flange Bearing Housing	4
7	M00032		Top Roller W/ Shaft	2
8	M00031		Top Roller WLDMT	2
9	M00001		Top Roller Spacer	4

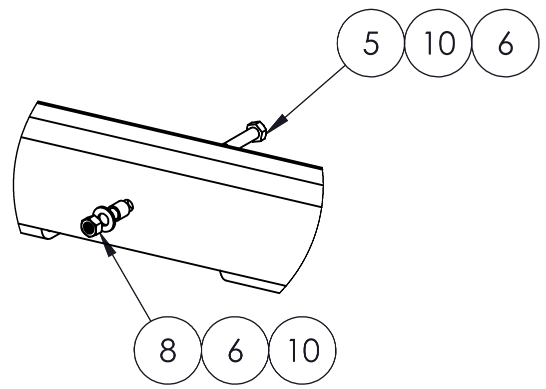
TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

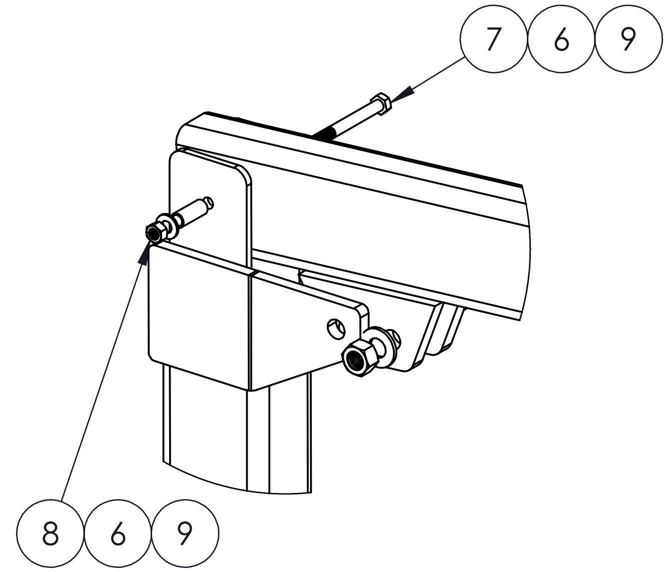
TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±
MANUFACTURED BY: -
MATERIAL:
DRAWN BY: MFROESE | DRAW DATE: 29/11/2017
START: - | REPLACED: -

TITLE: Top Roller Hanger Assy
UNIT
DRAWING NO: M00076
SHEET: 1 OF 1
REV: -

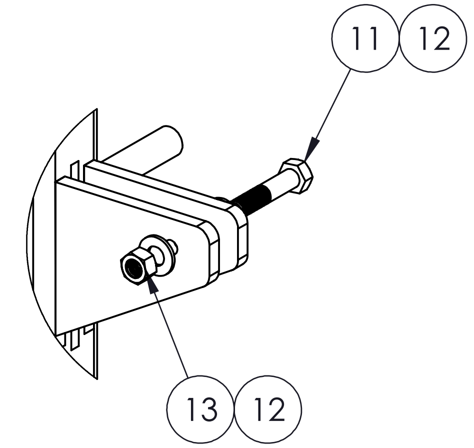
M00076 Friday, December 1, 2017 2:48:16 PM Matt



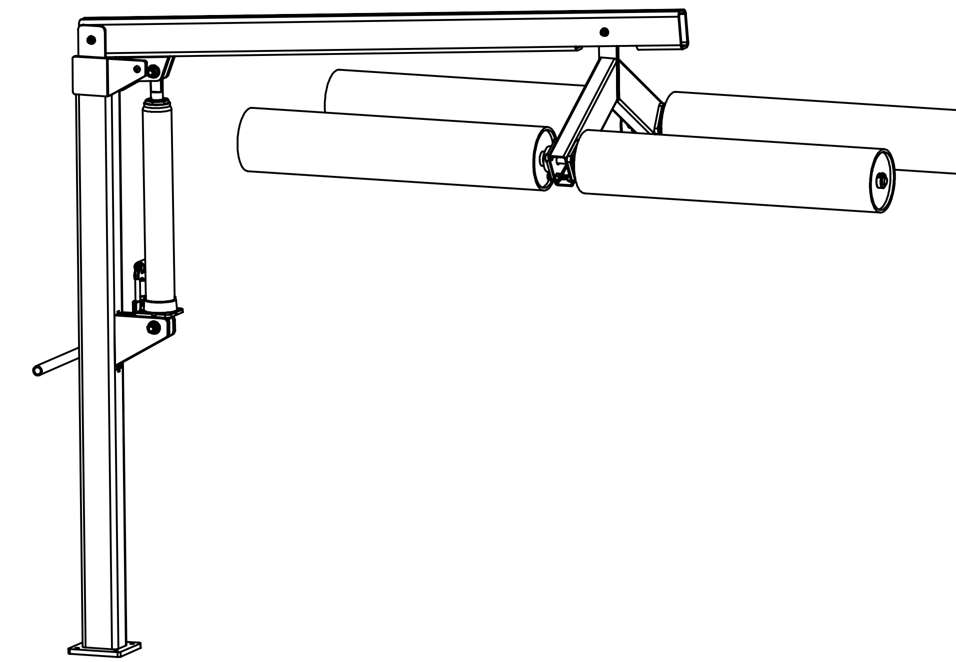
DETAIL A
SCALE 1 : 6

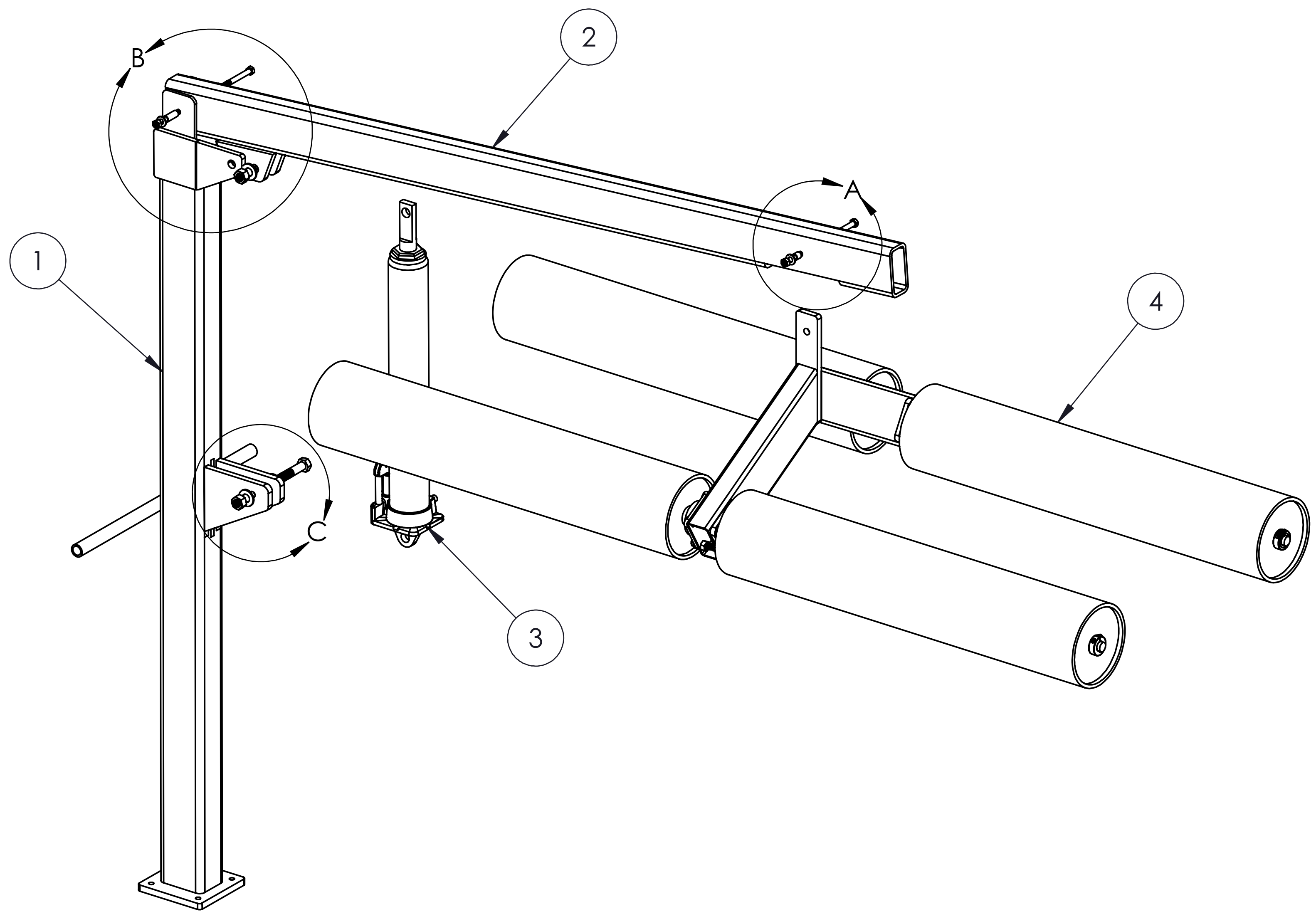


DETAIL B
SCALE 1 : 6



DETAIL C
SCALE 1 : 6





ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00035		Side Arm WLDMT	1
2	M00034		Swinging Arm WLDMT	1
3	46214	Strongway	Hydraulic Lift Cylinder	1
4	M00076		Top Roller Hanger Assy	1
5	91257A634	McMaster Carr	3/8" x 2.5" Bolt	1
6	98023A031	McMaster Carr	3/8" Flat Washer	4
7	91257A638	McMaster Carr	3/8" x 3.5" Bolt	1
8	94895A031	McMaster Carr	3/8" Hex Nut	2
9	6391K179	McMaster Carr	3/8" x 1-1/4" Sleeve Bearing	2
10	6391K173	McMaster Carr	3/8" x 7/8" Sleeve Bearing	2
11	91257A806	McMaster Carr	5/8" x 3" Bolt	2
12	98023A035	McMaster Carr	5/8" Flat Washer	4
13	94895A035	McMaster Carr	5/8" Hex Nut	2

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

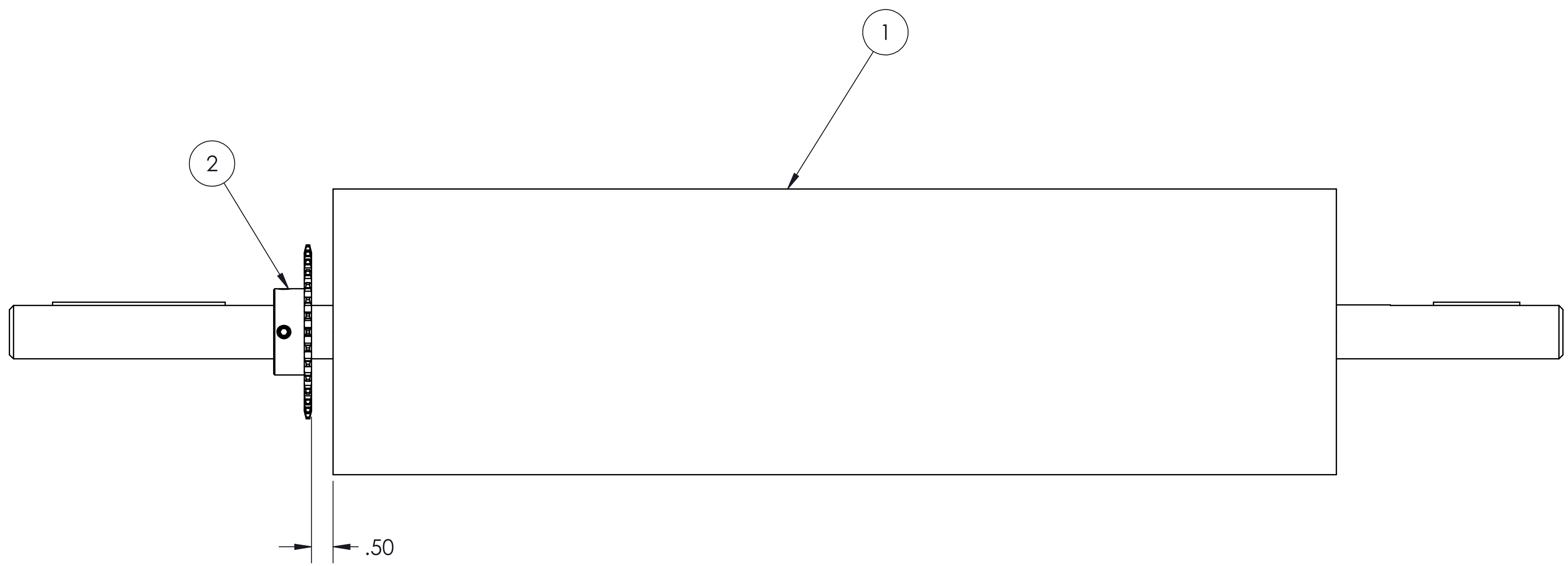
TITLE: Side Arm Assembly

UNIT

DRAWING NO: M00077 | SHEET: 1 OF 1 | REV: -

M00077 Friday, December 1, 2017 4:26:46 PM Matt

M00078 Friday, December 1, 2017 4:53:55 PM Matt

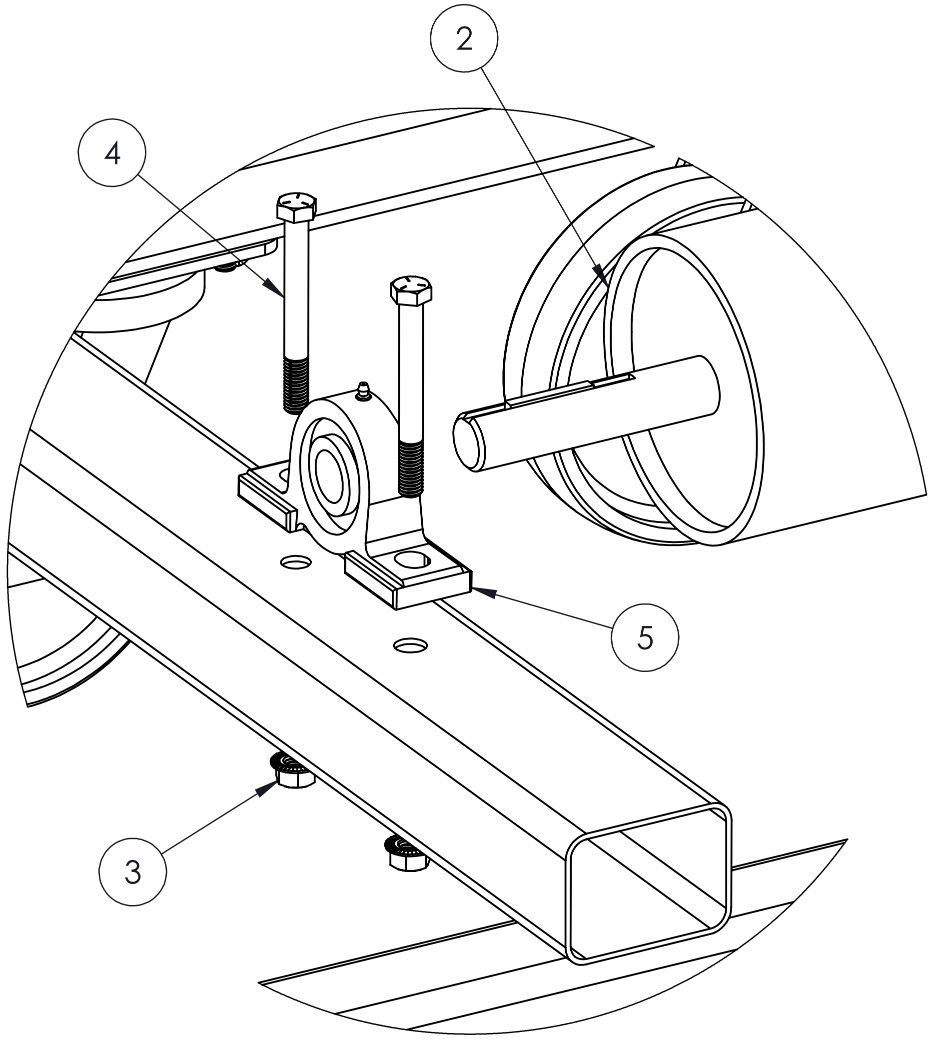


ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00040		Driven Roller WLDMT	1
2	6236K123	McMaster Carr	ANSI 40-30T Sprocket	1

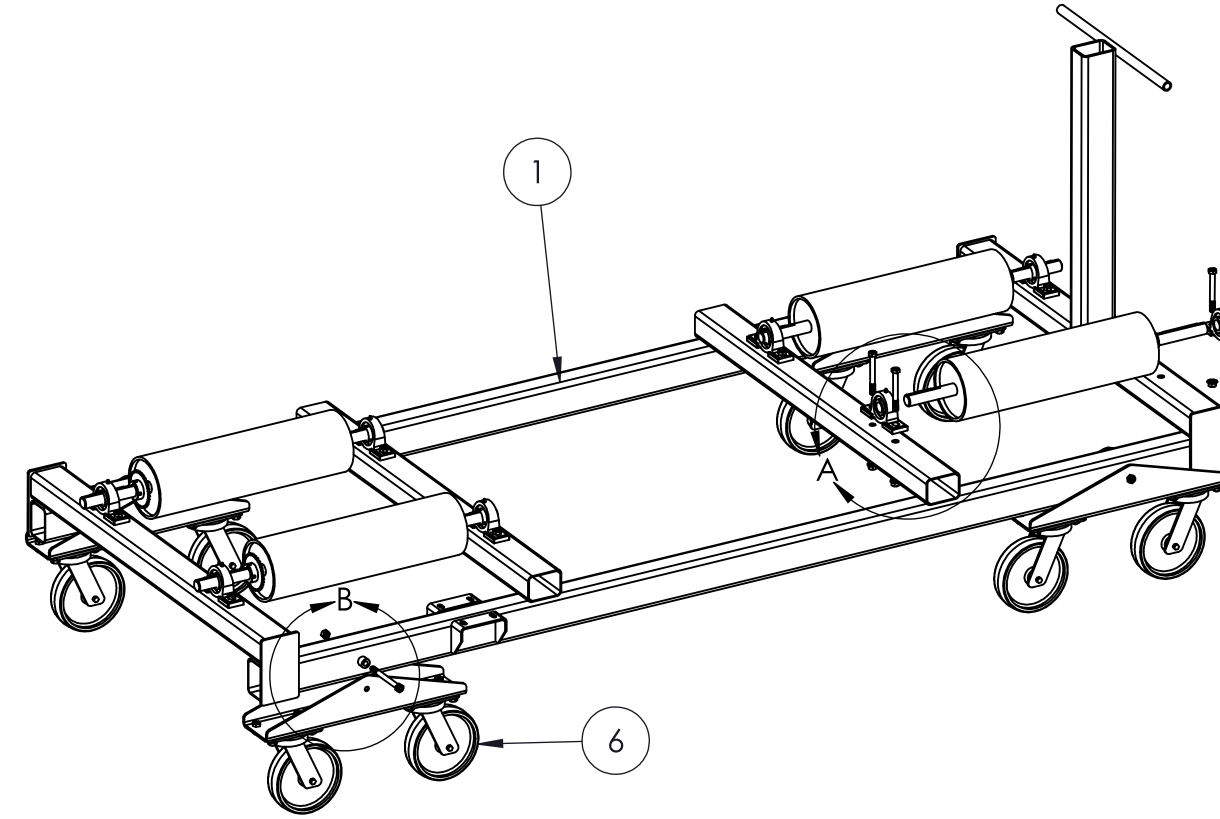
TRIPLE E RECREATIONAL VEHICLES

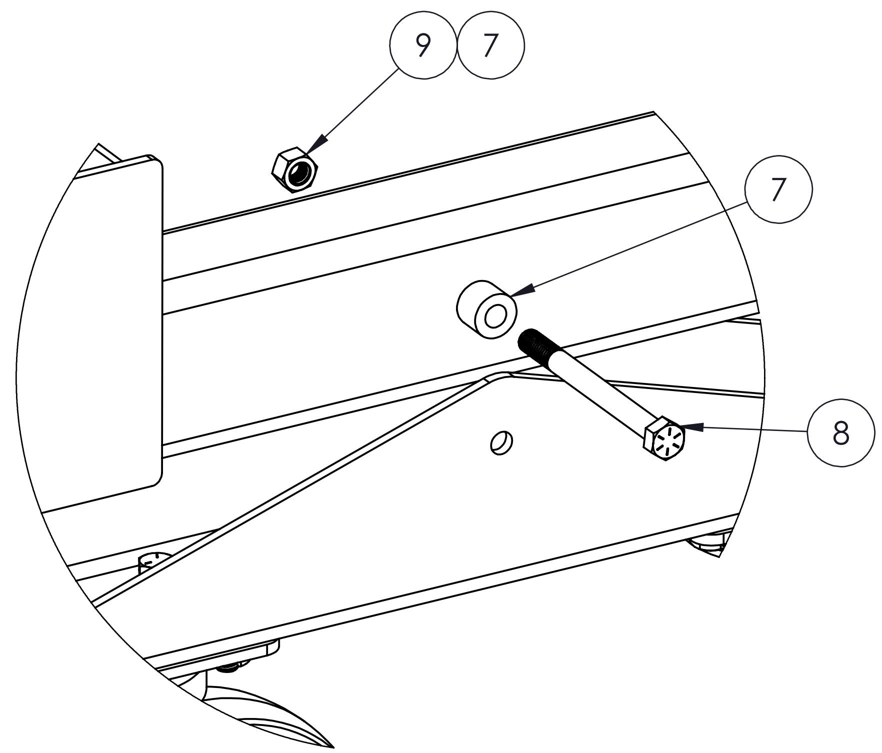
LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±	TITLE: Bottom Roller Assembly UNIT
MANUFACTURED BY: -	
MATERIAL:	DRAWING NO: M00078
DRAWN BY: MFROESE DRAW DATE: 29/11/2017	SHEET: 1 OF 1
START: - REPLACED: -	REV: -



DETAIL A
SCALE 1 : 4





DETAIL B
SCALE 1 : 4

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00038		Bottom Frame WLDMT	1
2	M00078		Bottom Roller Assembly	4
3	99904A104	McMaster Carr	1/2" Serrated Face Nut	16
4	91247A730	McMaster Carr	1/2" x 4.5" Bolt	16
5	0474562	Fastenal	1.25" Pillow Block	8
6	M00081		Walking Axle Assembly	4
7	92825A338	McMaster Carr	LDPE Spacer	8
8	91257A734	McMaster Carr	1/2" x 5.5" Bolt	4
9	90636A060	McMaster Carr	1/2" Centerlock Nut	4

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED):
DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

TITLE: Bottom Frame Assembly

UNIT

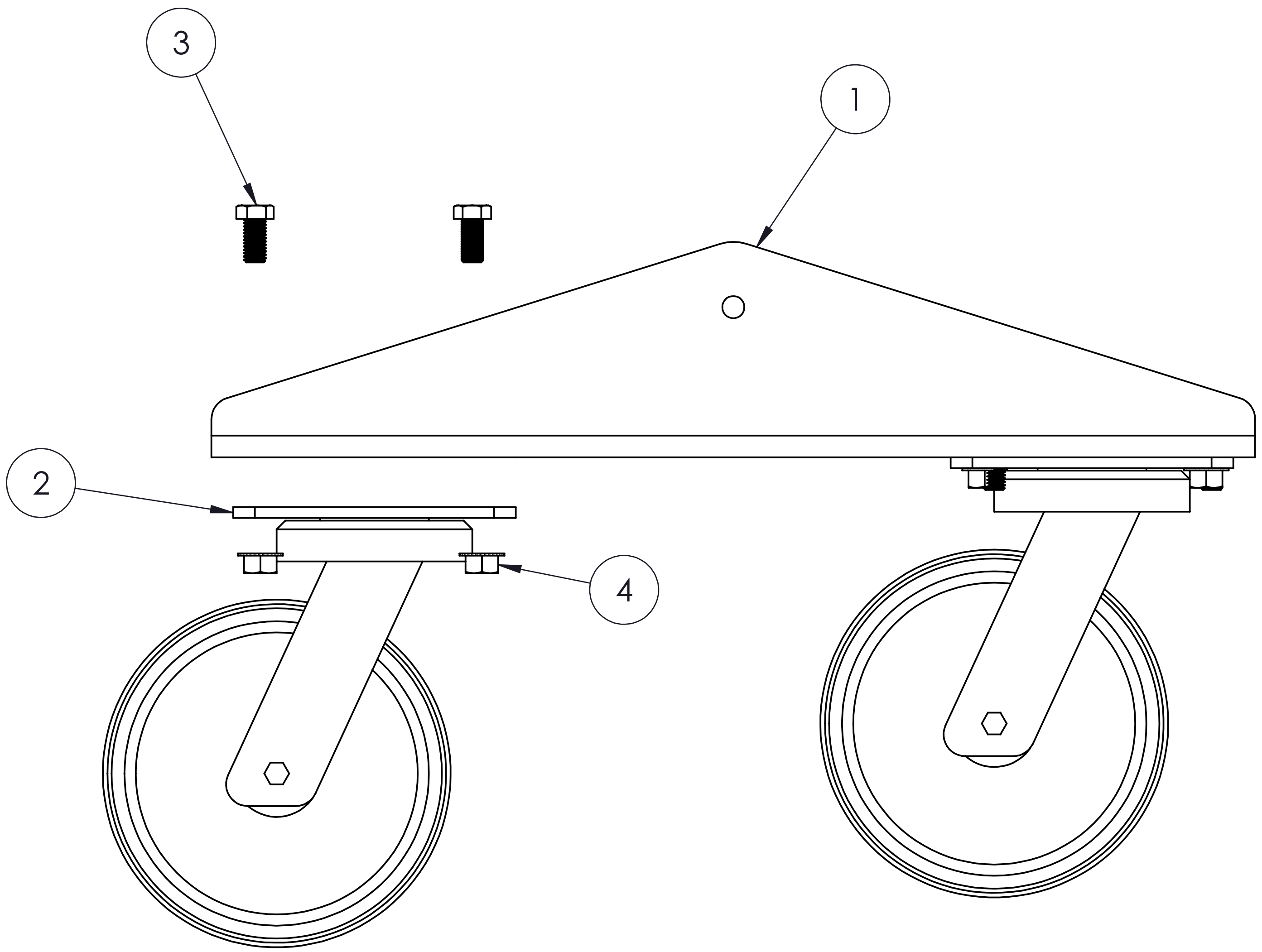
DRAWING NO: M00080

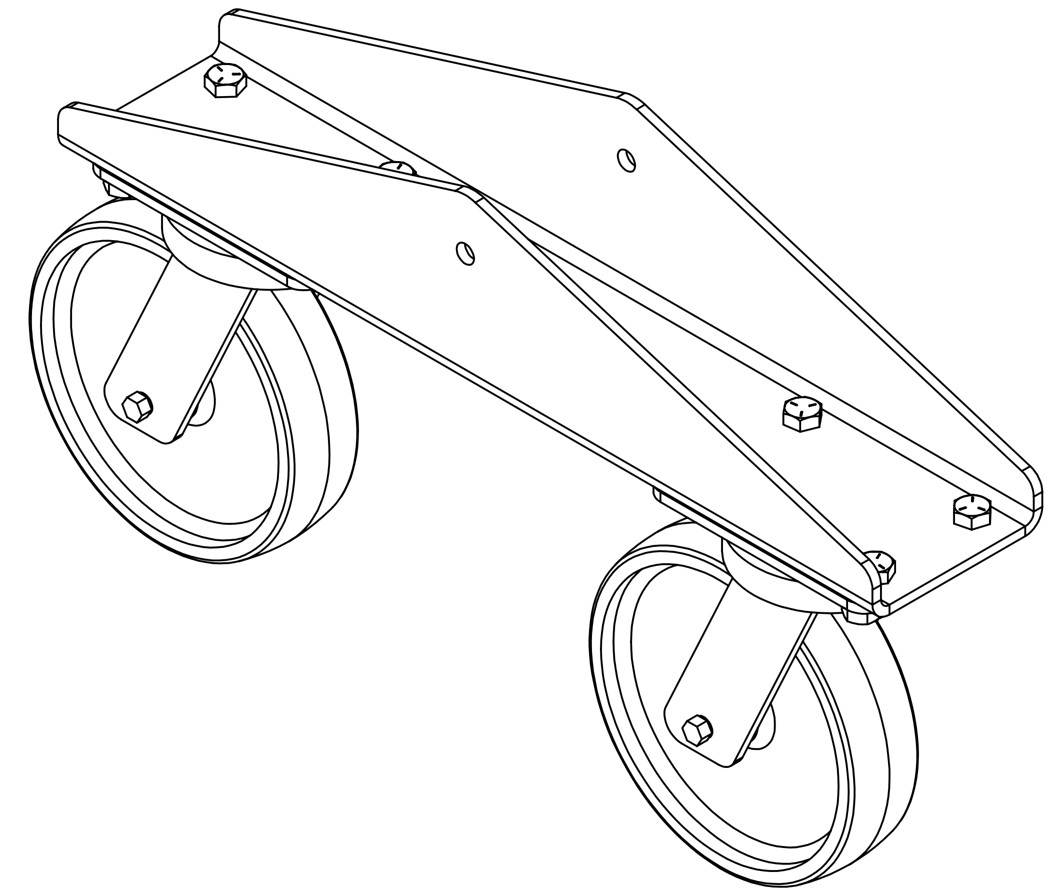
SHEET: 1 OF 1

REV: -

M00080 Friday, December 1, 2017 6:07:54 PM Matt

M00081 Friday, December 1, 2017 5:28:51 PM Matt



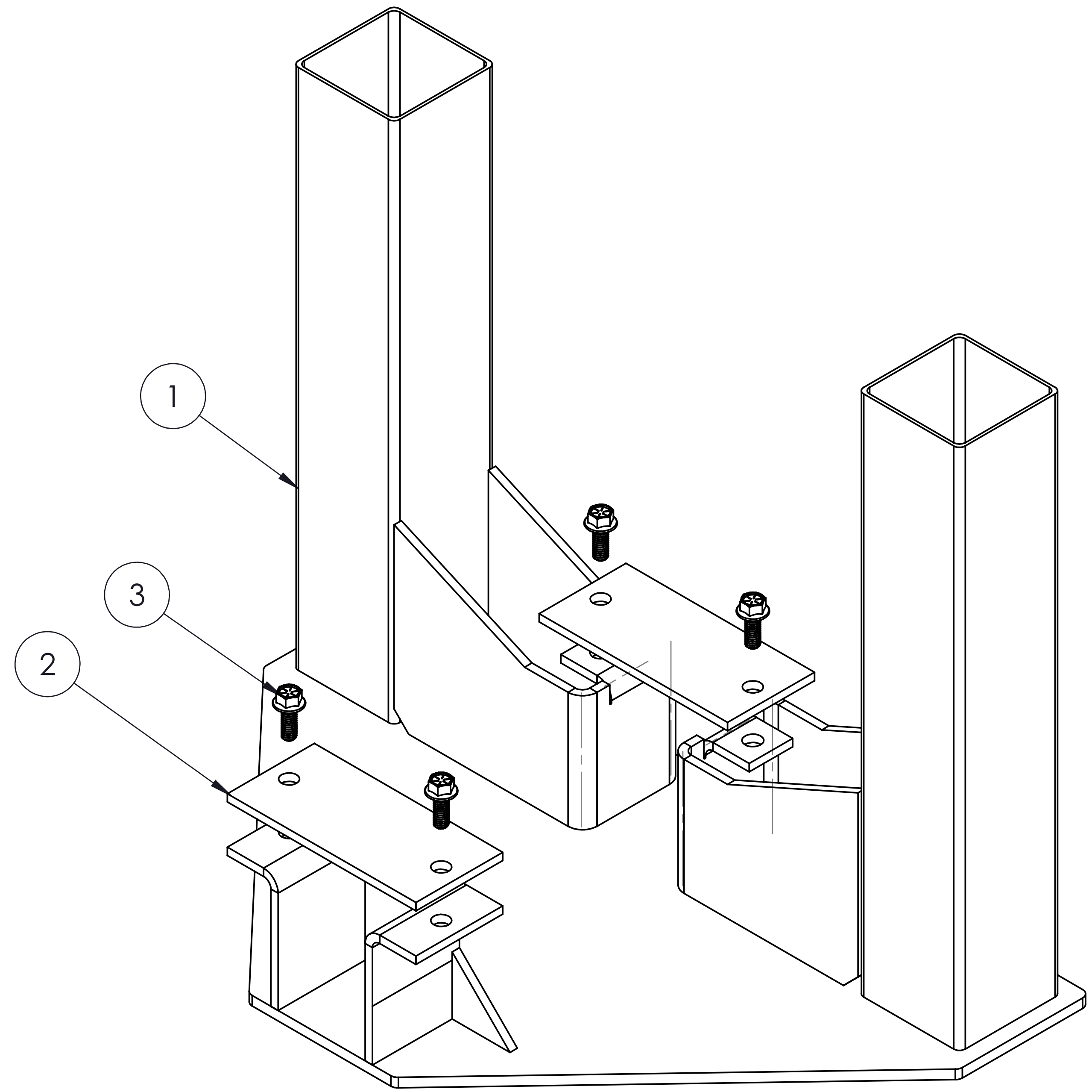


ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00026		Walking Axle Base	1
2	22665T74	McMaster Carr	Swivel Caster	2
3	92865A712	McMaster Carr	1/2" x 1" Bolt	8
4	99904A104	McMaster Carr	1/2" Serrated Face Nut	8

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±		TITLE: Walking Axle Assembly UNIT
MANUFACTURED BY: -		
MATERIAL:		DRAWING NO: M00081 — SHEET: 1 OF 1 — REV: -
DRAWN BY: MFROESE	DRAW DATE: 29/11/2017	
START: -	REPLACED: -	



ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00036		Retaining Shaft WLDMT	1
2	M00014		Locking Plate	2
3	92316A624	McMaster Carr	3/8" x 1" Bolt	4

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

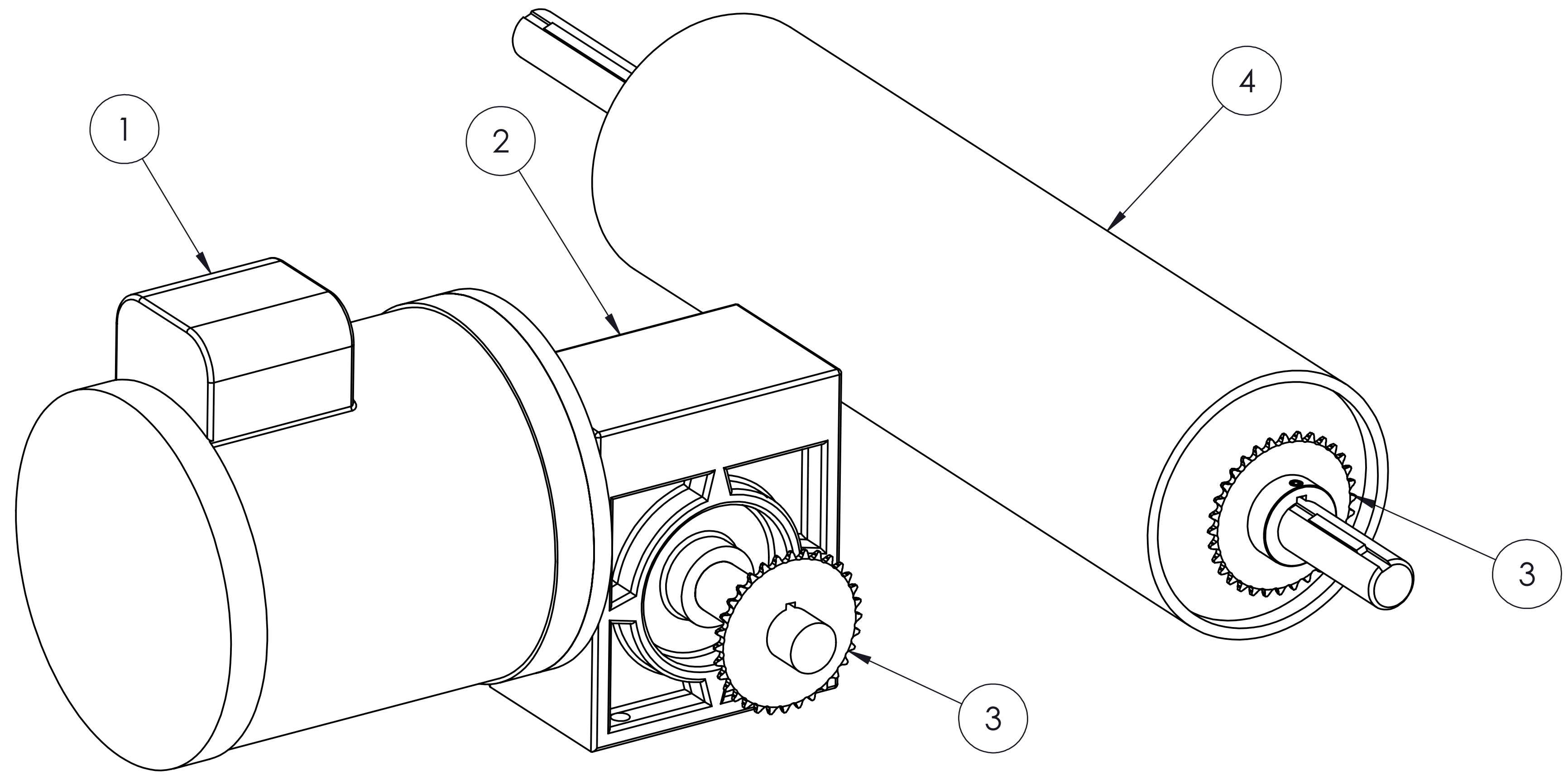
START: - | REPLACED: -

TITLE: Retaining Shaft Assembly

UNIT

DRAWING NO: M00082 | SHEET: 1 OF 1 | REV: -

M00082 Friday, December 1, 2017 5:39:27 PM Matt



ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	131544	Leeson	3 hp Electric Motor	1
2	BMQ075-180TC	Superior Gearbox	Worm Gear Reduction	1
3	6236K123	McMaster Carr	ANSI 40-30T Sprocket	2
4	M00040		Driven Roller WLDMT	1

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED): DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

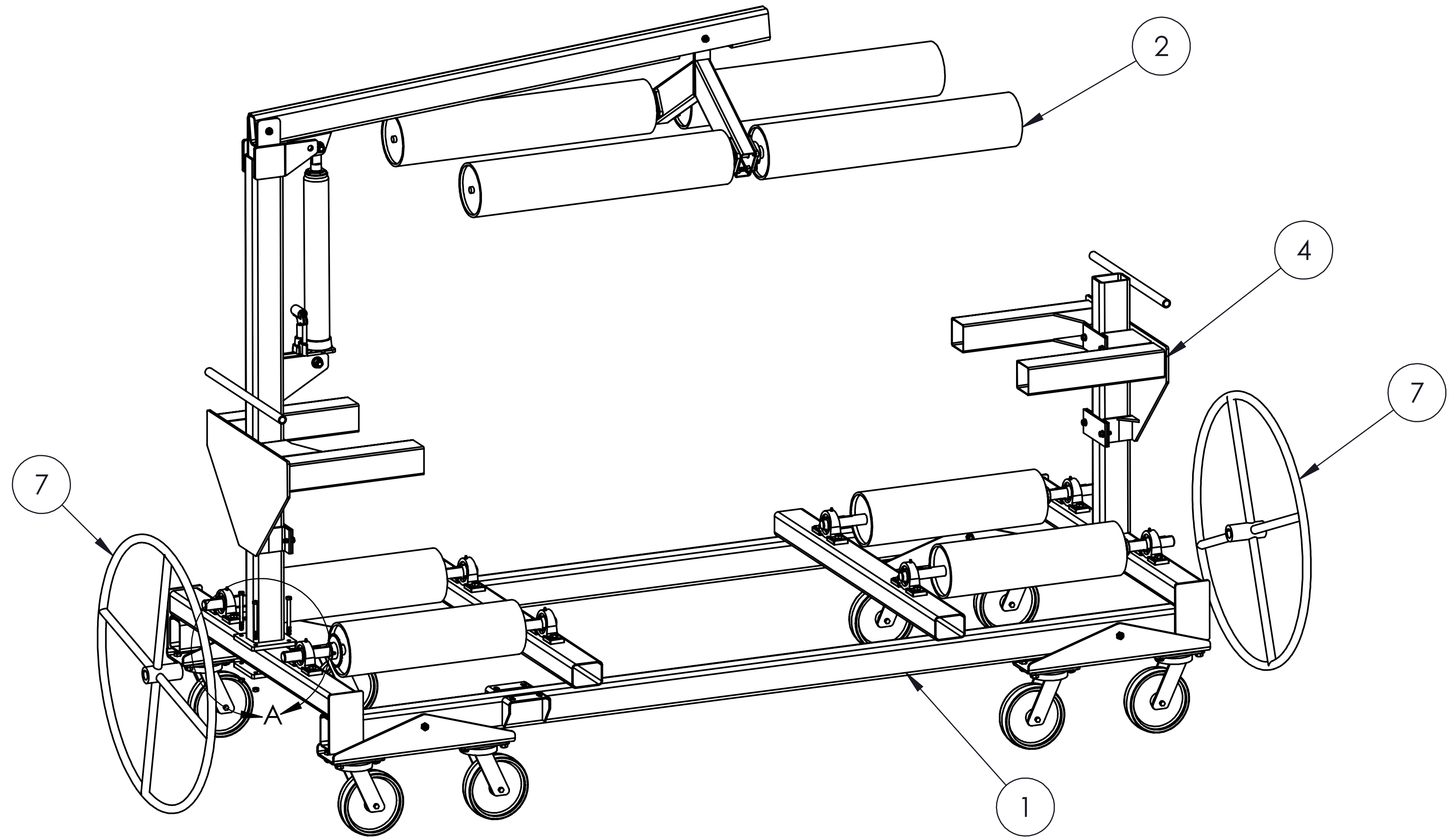
START: - | REPLACED: -

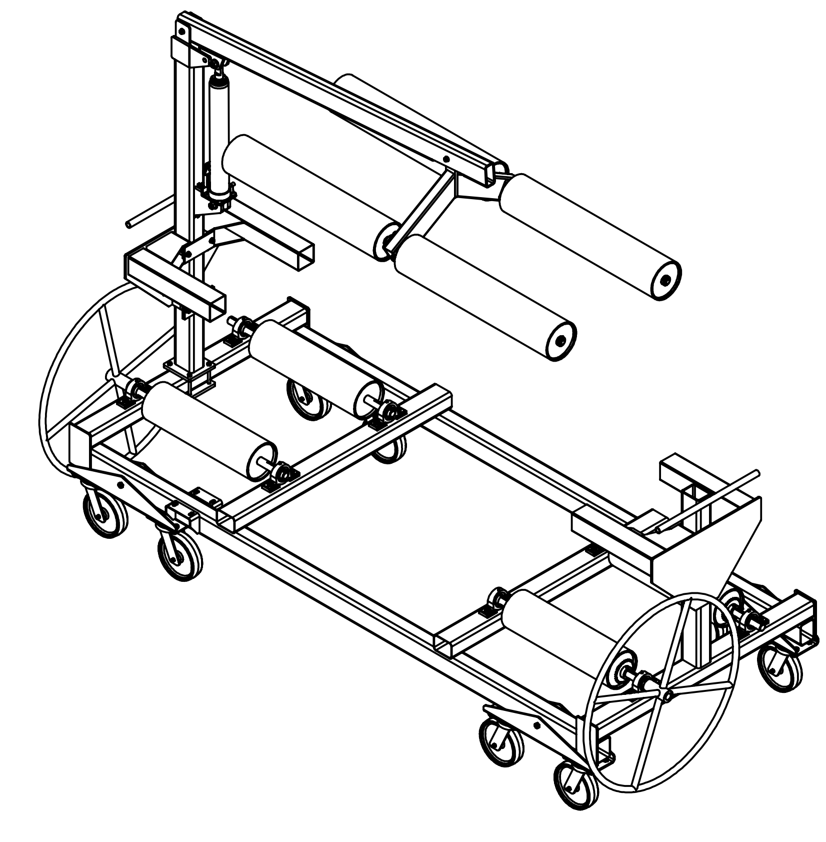
TITLE: Electric Drive Kit

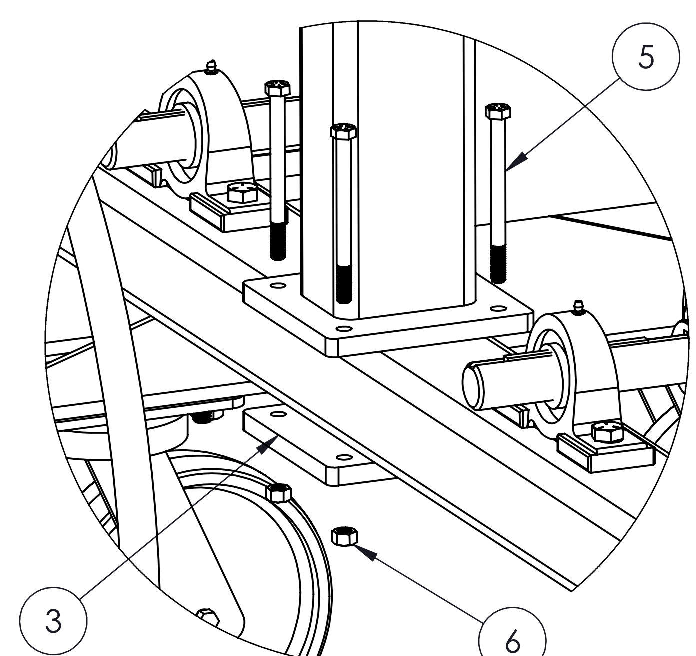
UNIT

DRAWING NO: M00083 | SHEET: 1 OF 1 | REV: -

M00083 Saturday, December 2, 2017 11:35:15 PM Matt







DETAIL A
SCALE 1 : 5

ITEM NO.	PART NUMBER	Supplier	DESCRIPTION	QTY.
1	M00080		Bottom Frame Assembly	1
2	M00077		Side Arm Assembly	1
3	M00018		Mounting Plate	1
4	M00082		Retaining Shaft Assembly	2
5	91257A642	McMaster Carr	3/8" x 4.5" Bolt	4
6	94895A031	McMaster Carr	3/8" Hex Nut	4
7	M00037		Bus Wheel WLDMT	2

TRIPLE E RECREATIONAL VEHICLES

LEISURE TRAVEL VANS

TOLERANCES(UNLESS SPECIFIED):
DIMENSIONS 0.0625 in ± ANGLES 1° ±

MANUFACTURED BY: -

MATERIAL:

DRAWN BY: MFROESE | DRAW DATE: 29/11/2017

START: - | REPLACED: -

TITLE: Composite Material Uncoiler
UNIT

DRAWING NO: M00084

SHEET: 1 OF 1

REV: -

M00084 Friday, December 1, 2017 6:58:27 PM Matt

Appendix B: Concept Generation and Selection Process

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1. Research and Supporting Materials

The following sections outline the in-depth concept generation and selection process of the best possible design to develop and optimize, such that all project objectives can be met. Before the team started generating possible design solutions, all members began individually researching material coiling machines and carts with different performance attributes to help brainstorm for the project. The research primarily focused on roll retaining methods, material spring back constraining, footprints of other concepts, and mechanical safety.

1.1. Roll Retaining Mechanisms

The current frame for the composite roll uses a shaft with two tires on both sides to support the composite material. When expansion of the composite material occurs, the inner and outer diameters of the composite material become larger. Therefore, one concern the team wanted to address is the redesign of the shaft such that it may adjusted to the inner diameter of the roll itself. An example of this technique is shown in Figure B-1.

Figure B-1: New Madison manual material uncoiler [1].

Figure B-1 shows a manual material uncoiler designed for cantilever loading with an expanding toggle action mandrel. Since the capacity of the team's design should achieve a supporting weight of 5000 lbs, the cantilever loading mode is not particularly appropriate nor safe. Therefore, the team may consider expanding the toggle action mandrel design such that it can be supported by both sides, and other aspects of design could be found later via stress analysis.

Considering the difficulty of manufacturing a cart that provides easy composite roll changeovers via forklift, the team decided to research ways to support the composite roll without the use of a shaft. The composite material roll could be supported by two horizontal and parallel cylindrical rollers, such that the rollers would rotate with the composite roll to complete the uncoiling and recoiling processes. This design could be easier to manufacture, and changeover of composite rolls could easily be performed by a forklift in significantly less time than the current process. Such design also may have the potential to reduce the overall footprint. Figure B-2 below illustrates an example of how a roll of material may be supported without a shaft.

[redacted]

Figure B-2: Support from without shaft [2], [3].

In Figure B-2, the maximum capacity of this particular uncoiler is 300 kg [2]. It consists of welded frames, two swivel casters, and supporting rollers on ball-bearings that are made of galvanized steel [2]. The uncoiler is optionally available with a braking device that acts on one of the supporting rollers [2]. However, this kind of support frame is typically used for supporting light roll materials, and would not immediately be appropriate for a 5000 lbs roll material. Therefore, the team still needs to consider how the concept could be applied in a way such that capability and safety are increased. Additionally, the team would need to make sure the composite material roll will not fall off the frame during uncoiling and recoiling processes.

1.2. Constraining Roll Expansion

With Triple E RV's current cart, the composite material roll will expand once the packaging straps are removed. This causes the uncoiling and recoiling processes of the material to be more difficult. Therefore, the roll material should be constrained by a specific mechanism during both uncoiling and recoiling processes, such that the process will be more stable and require less effort. Figure B-3 illustrates how such a mechanism may work.

[redacted]

Figure B-3: Roll constrainer [4].

As shown in Figure B-3, the roll constrainers in the left scenario can keep the material flush to the roll all the times. For the current design with a shaft, there may need to be multiple constrainers on both the bottom and top sides of the material to stop expansion. For a design without a shaft, it may only need a top rolling constrainer, since the cylindrical rollers at the bottom can be used to constrain the material by using its own weight.

1.3. Mechanical Safety of Design

Looking at increasing the overall safety of the design, the team looked at adding handles to the current cart. Since the composite material is heavy, it is extremely hard to move the current supporting frame as is. Thus, the addition of a handle could reduce the difficulty to move the material and help eliminate back injuries of the operating staff. A typical cart that utilizes a handle is shown in Figure B-4.

[redacted]

Figure B-4: Cart with handle [5].

With the same safety considerations in mind, the diameter of the researched cylindrical rollers are small in the shaftless design. It would be hard to use the crank concept on the client's current cart to rotate the composite material during uncoiling and recoiling. Therefore, the team considered using a large wheel, or “bus wheel” concept in place of the crank. A bus wheel concept can be seen in Figure B-5.

Figure B-5: Bus wheel concept [6].

Potentially, a bus wheel could make the rotating process easier by giving the operator a mechanical advantage, which can also reduce the likelihood of injuries. An operator can rotate the bus wheel while pulling from one side with a stable posture instead of the substantial body movement required when using a crank.

2. Concept Generation

After defining the project specifications and having a clear understanding of the problem statement, the team moved into the concept generation phase for the composite roll fixture to be designed. The concept generation phase involves a thorough exploration of alternatives to meet the desired specifications of the project. Design specifications including supporting the roll, moving the roll by two individuals, easy exchange of the roll using a forklift, roll retainability, constraining roll expansion, and powered mechanisms were all considered during the brainstorming process.

To begin the concept generation phase, everyone on the team was asked to come up with a minimum of five designs individually. Each team member presented their results to the rest of the team, where additional concepts were formed as a group. The outcome of the concept generation phase resulted in a total of 37 conceptual designs. Some concepts were designed based on individual project specifications while some satisfied multiple project specifications. All 37 concepts are described below. An additional pictorial master list of all concepts, as well as which pages their descriptions can be found within the Appendix is contained in section Concept Generation Master List, on page B-58 for convenience of the reader.

Concept #1

In Concept #1, the weight of the roll of material is supported by rollers running the length of the roll, rather than a shaft running through the roll. The roll of composite material rests on these rollers, allowing the material to spin in place. This concept eliminates the need for a shaft and supporting structure.

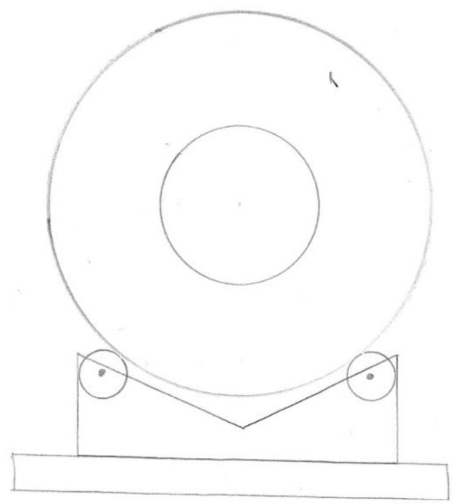


Figure B-6: Concept #1 sketch [7].

Concept #2

Concept #2 aims to replace the tires which are currently used to support the roll of material. Two cone shaped wedges consisting of rolled sheet metal and supporting gussets are allowed to slide along a shaft. These wedges would be slid into both ends of the roll of material, wedging into the inner tube. This would center the roll of material on the shaft and support the material. As with the current method used, the shaft would need to be supported with a frame.

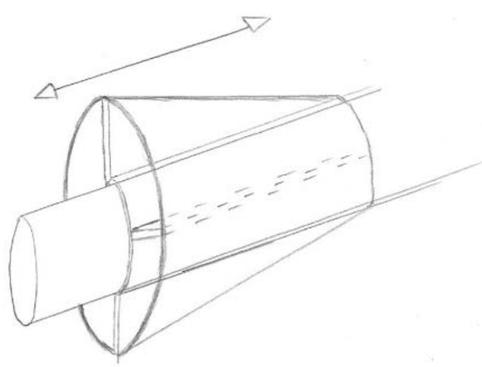


Figure B-7: Concept #2 sketch [7].

Concept #3

Concept #3 is also intended to replace the tires used to support the roll of material on a shaft. A four-lobe cab is centered on a shaft, with a solid block mating to each lobe. As the cam rotates, the blocks are forced outwards away from the shaft, eventually pressing against the inner tube of the roll of material. One cam assembly would be used at each end of the shaft. As with the currently used design, the shaft would need to be supported by a frame.

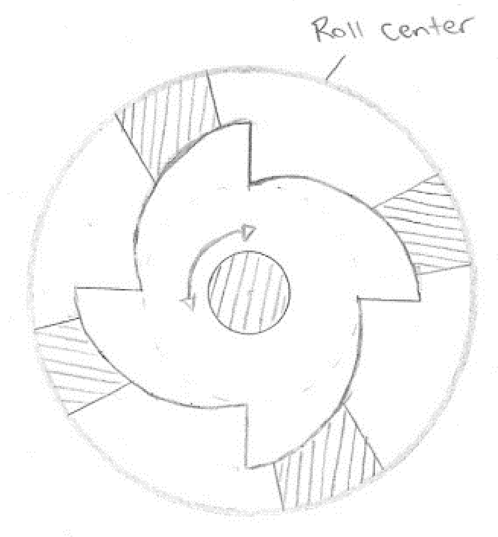


Figure B-8: Concept #3 sketch [7].

Concept #4

Concept #4 is also intended to replace the tires used to support the roll of material on a shaft. Triangular shaped sheet metal pieces are welded to a shaft in a radial pattern. Smaller, similarly shaped triangle sheet metal pieces are allowed to slide along the angled faces of the welded metal triangles. As the small triangles slide up the face of the welded triangles, they press against the inner tube of the roll of material, centering the shaft. As with the currently used design, the shaft would need to be supported with a frame.

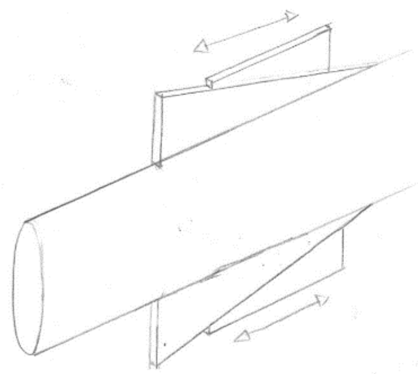


Figure B-9: Concept #4 sketch [7].

Concept #5

Concept #5 aims to reduce the amount of effort required to move the roll of composite material. The design contains four sets of walking axles, one at each corner of the support structure. Each walking axle uses two caster wheels. By increasing the number of wheels, and thus decreasing the load carried by each wheel, rolling friction in the wheel bearings is reduced, theoretically reducing the effort required to move the material.

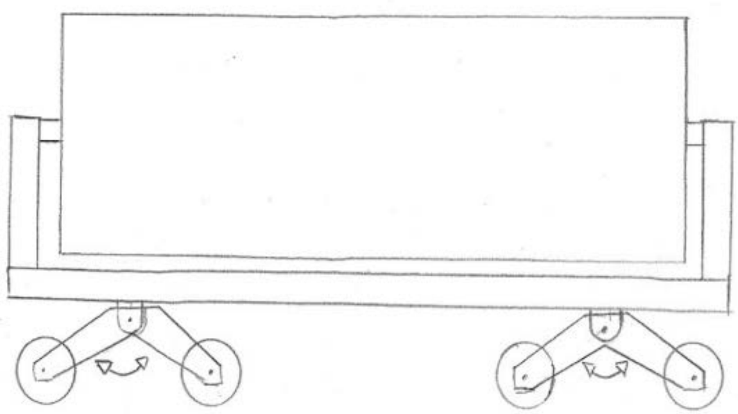


Figure B-10: Concept #5 sketch [7].

Concept #6

Concept #6 is also aimed at making the material easier to move. The team noted that the design currently used by the client used did not contain a handle. Adding a handle to the design provides a convenient and safe location from which to push and pull the roll of material.

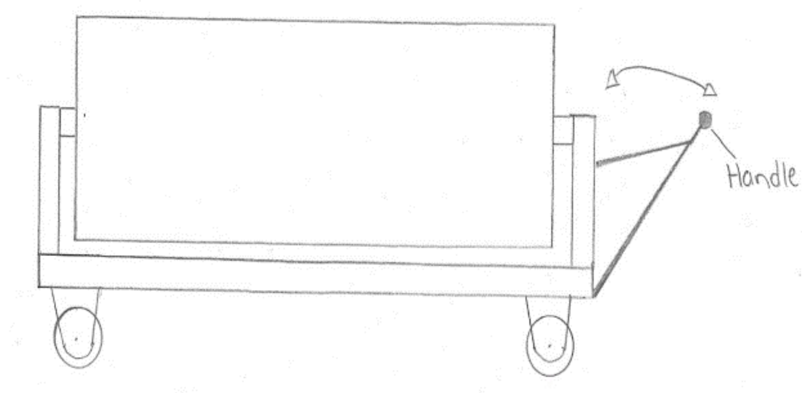


Figure B-11: Concept #6 sketch [7].

Concept #7

Concept #7 is also intended to reduce the effort required to move the roll of composite material. An additional wheel is added to the rear of the support structure. This wheel is powered using an electric motor, and is attached to the cart on a swinging arm, so as to be engaged and disengaged with the floor.

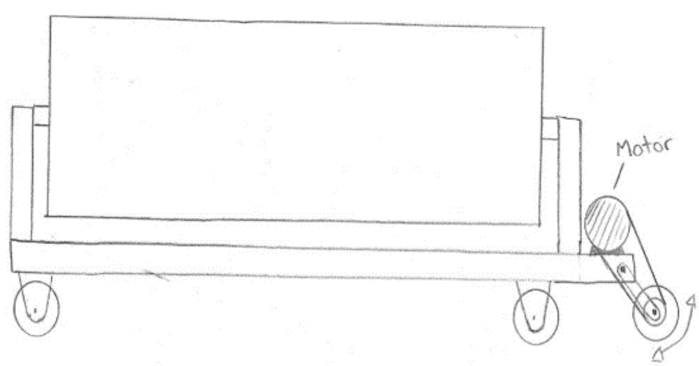


Figure B-12: Concept #7 sketch [7].

Concept #8

Concept #8 aims at reducing the effort required to uncoil and recoil the composite material. An electric motor is attached directly to the shaft supporting the roll of material. This motor would assist in rotating the roll in both directions.

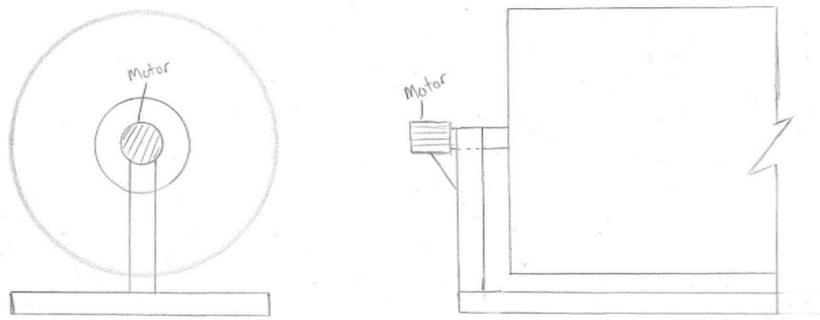


Figure B-13: Concept #8 sketch [7].

Concept #9

Concept #9 also intends to reduce the effort required to uncoil and recoil the composite material. An electric motor is attached to the shaft supporting the material by means of a chain. By using a chain, the input and output sprocket sizes can be chosen to provide increased torque to the roll or provide an optimal rotation speed.

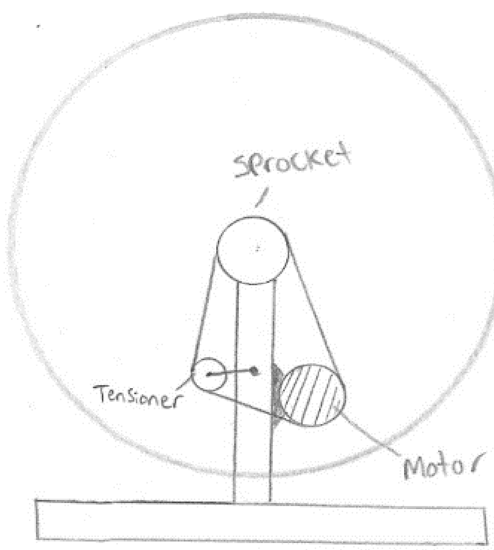


Figure B-14: Concept #9 sketch [7].

Concept #10

Concept #10 also aims at reducing the effort required to uncoil and recoil the composite material. A roller runs the length of the roll of material, and is attached to the support structure by a swinging arm. This roll is driven, either directly or by means of a chain, by an electric motor. The roller is held against the roll of material using a spring, and can be rotated in both directions using the electric motor.

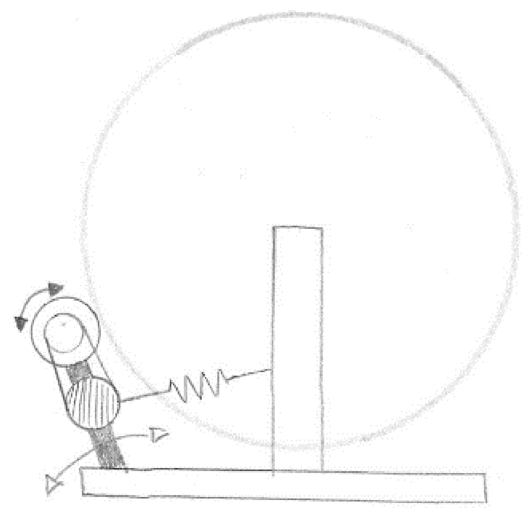


Figure B-15: Concept #10 sketch [7].

Concept #11

Concept #11 aims to reduce the effort required to uncoil the composite material. Material leaves the roll and passes through two rollers which are held against each other using spring tension. These two rollers are powered using an electric motor, which can be used to pull material from the roll.

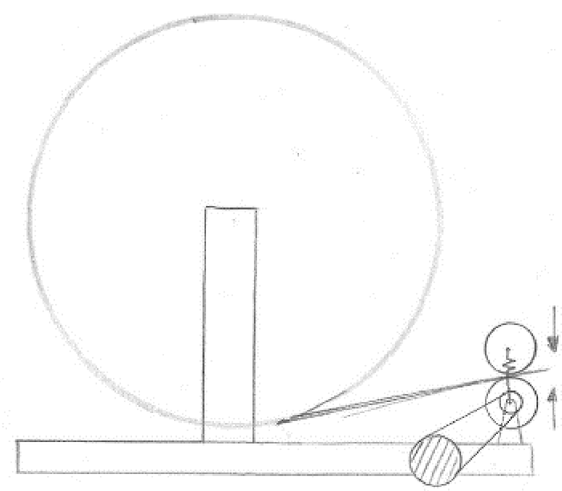


Figure B-16: Concept #11 sketch [7].

Concept #12

Concept #12 aims at maintaining the composite material in a tightly wrapped roll. Material leaves the roll and passes through two rollers, which are pressed against each other using spring tension. These rollers can be locked from rotating, preventing the composite material from recoiling. At the same time, a ratchet mechanism on the composite roll prevents the roll from rotating and loosening the wrap of material.

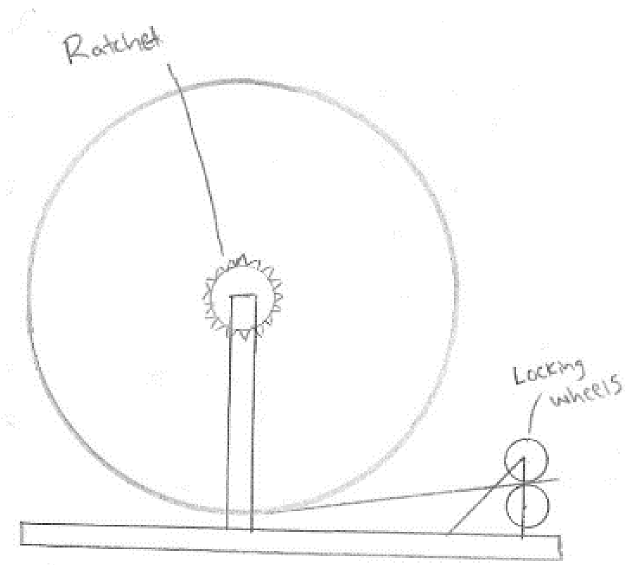


Figure B-17: Concept #12 sketch [7].

Concept #13

Concept #13 aims to maintain the material in a tightly wrapped roll. Four rollers run the length of the roll of composite material. These rollers are held against the roll of material using spring tension. By applying a compressive force to the roll of material, the roll is prevented from expanding.

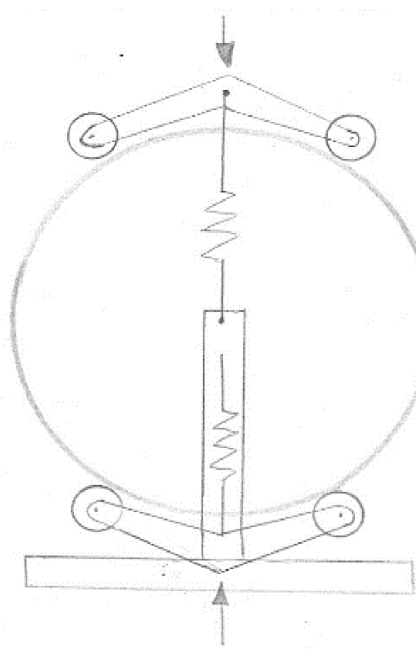


Figure B-18: Concept #13 sketch [7].

Concept #14

Concept #14 also intends to maintain the material in a tightly wrapped roll. Straps wrap around the exterior of the roll of material, with one end of the strap being fixed, and the other end being held in tension using a spring. Applying this compressive force to the roll of material prevents the roll from expanding.

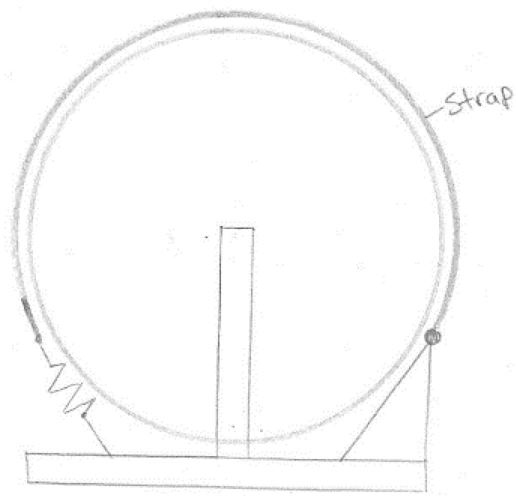


Figure B-19: Concept #14 sketch [7].

Concept #15

Concept #15's method also aims to maintain the material in a tightly wrapped roll. Four rollers press against the roll of composite material, with two rollers on the left side of the roll and two rollers on the right side of the roll. Each pair of rollers is connected by an intermediate arm, which is connected to the support structure using swinging arms. The rollers are held against the roll of material using spring tension.

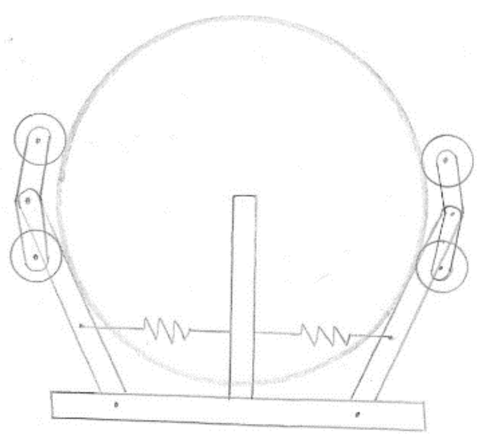


Figure B-20: Concept #15 sketch [7].

Concept #16

Concept #16 aims to support the roll of material as well as maintain the material in a tightly wrapped roll. A horizontal beam, pinned at one end, supports two rollers which the roll of composite material rests on. The free end of the beam is supported by a cable extending from a horizontal beam located above the roll of material. This beam is also pinned at one end and applies a downward vertical force to the roll of material via two rollers on the other end of the beam. This downward force, generated by the weight of the roll of material, prevents the roll from expanding.

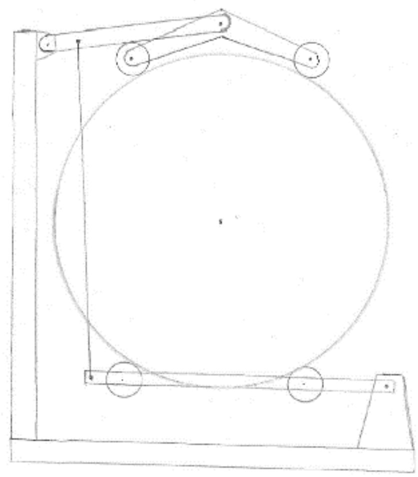


Figure B-21: Concept #16 sketch [7].

Concept #17

Concept #17 utilizes the current cart's shaft design to center the roll. A no-slip roller or ideal frictionless rigid retainer is pushed up into the bottom of the composite roll via springs. The forced applied by the springs would be significant, such that the coiled composite material shall stay confined to the minimum diameter possible.

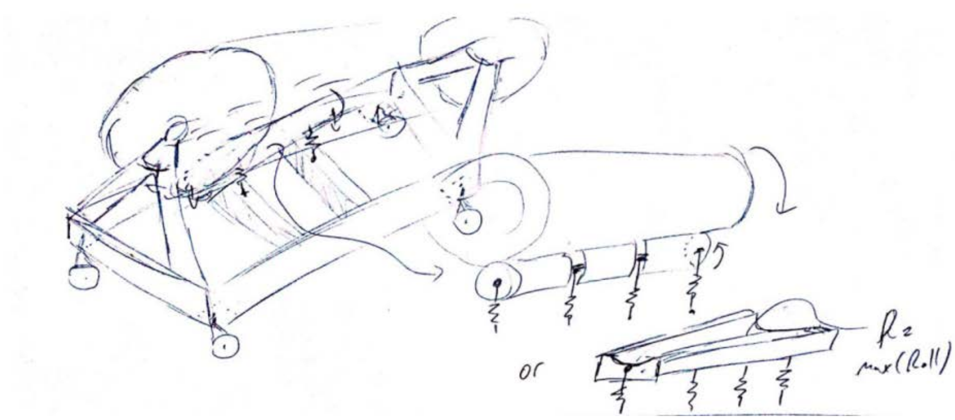


Figure B-22: Concept #17 sketch [8].

Concept #18

Concept #18 utilizes two supporting rollers such that the coiled composite material roll will firmly sit in-between them. A third roll is then fixed on top such that the composite material is retained to no more than initial size of the roll diameter. Assuming all rollers will not slip relative to the material roll, a singular supporting roll could be driven by using a large bus wheel to gain mechanical advantage.

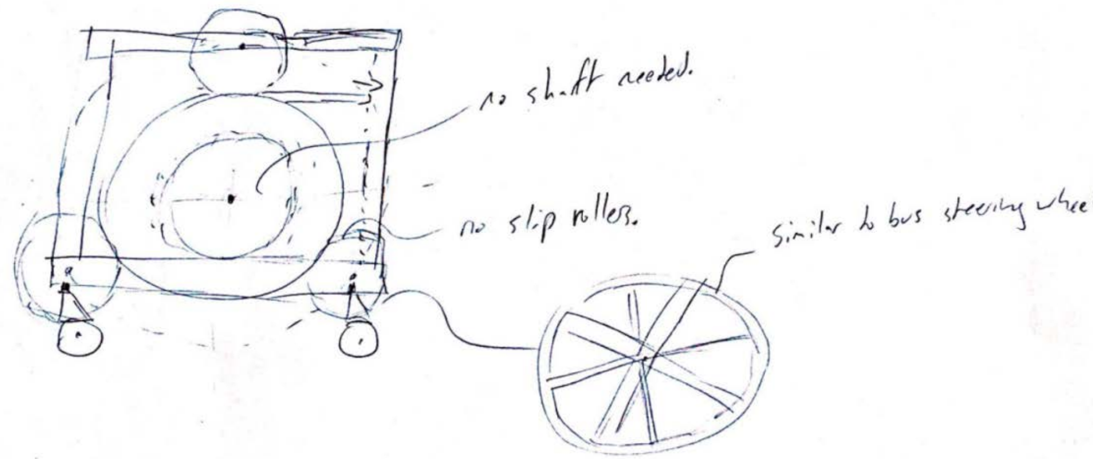


Figure B-23: Concept #18 sketch [8].

Concept #19

Concept #19 is based off a cantilever design where the roll would freely insert into a shaft. A chucking mechanism would be used from the fixed end of the cantilever, such that as the locking spindle turns, the clamping hands move towards the center of the shaft and apply pressure to the roll. Thus, this can keep the diameter of the roll minimized. Notably, in this concept, the chuck would need to be continually tightened as the material is used.

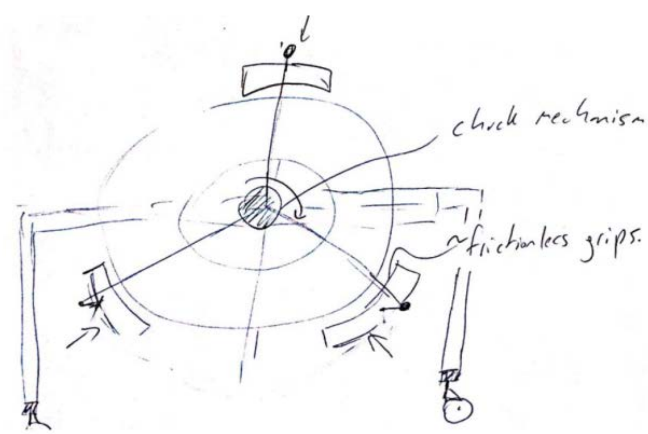


Figure B-24: Concept #19 sketch [8].

Concept #20

Concept #20 is based on reducing the effort to move the composite material current cart. The rolling friction is reduced by the addition of a large, rigid wheel placed on a center axle. This wheel, assuming it stayed rigid enough, would allow the cart to move with less force than the current cart. However, rotation of the cart would be confined to only when the cart is stopped, and about the center of gravity of the large wheel. Additionally, a handle bar is added to the cart, approximately set to the average center of mass of the cart, such that pushing and pulling effort is reduced. Overall, the concept is less strainful on the operator.

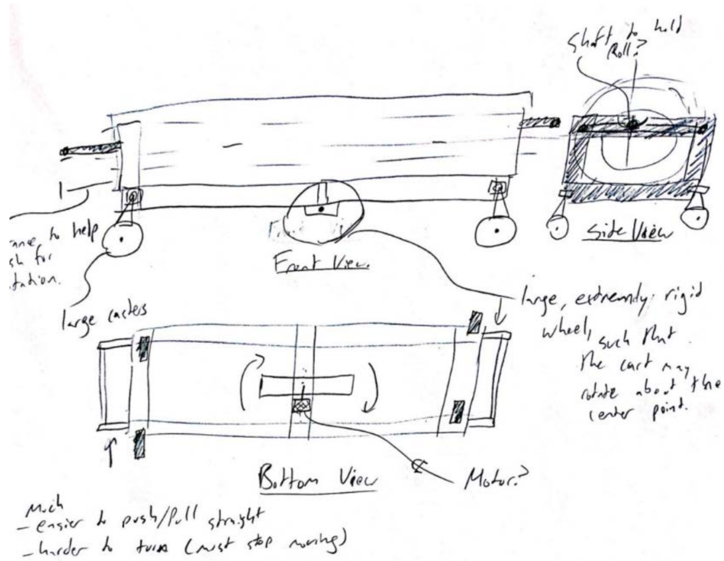


Figure B-25: Concept #20 sketch [8].

Concept #21

Concept #21 utilizes a shaftless configuration with two supporting rollers to hold the composite material roll. Additionally, a third roller which is freely allowed to move up and down through a slot would apply pressure to the top of the composite material via the tops rolls weight. Thus, gravity is used to constrain composite material roll expansion. Additionally, a lever mechanism could be used to assist lifting the top roller to a position such that a pin support could safely hold it in place during roll changeovers.

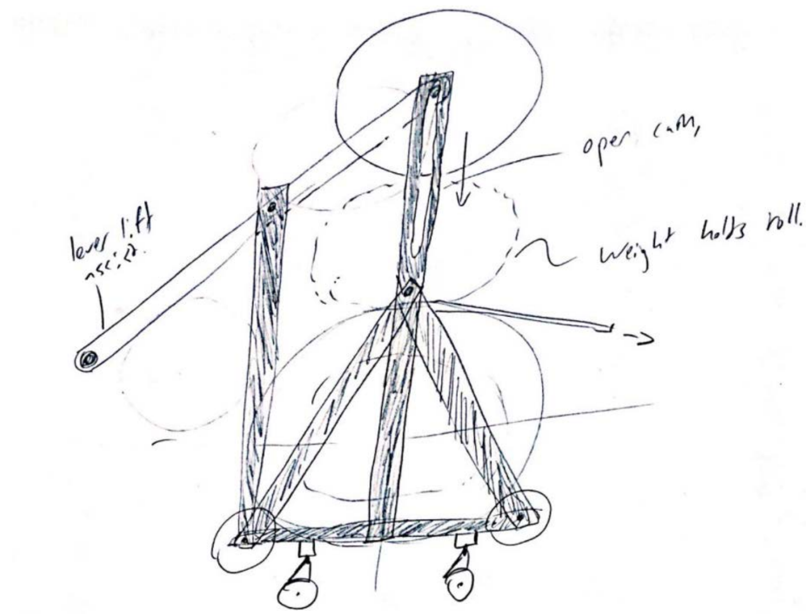


Figure B-26: Concept #21 sketch [8].

Concept #22

Concept #22 is centered around reducing the amount of force it takes to push and pull the cart by creating some sort of mechanism that automatically adjusts a handle bar so that it always lines up with the center of mass of the cart. As the roll of composite material's diameter shrinks during regular use, the change in diameter size would somehow determine and adjust the height of the handle bar accordingly via a complex mechanism that would need to be designed.

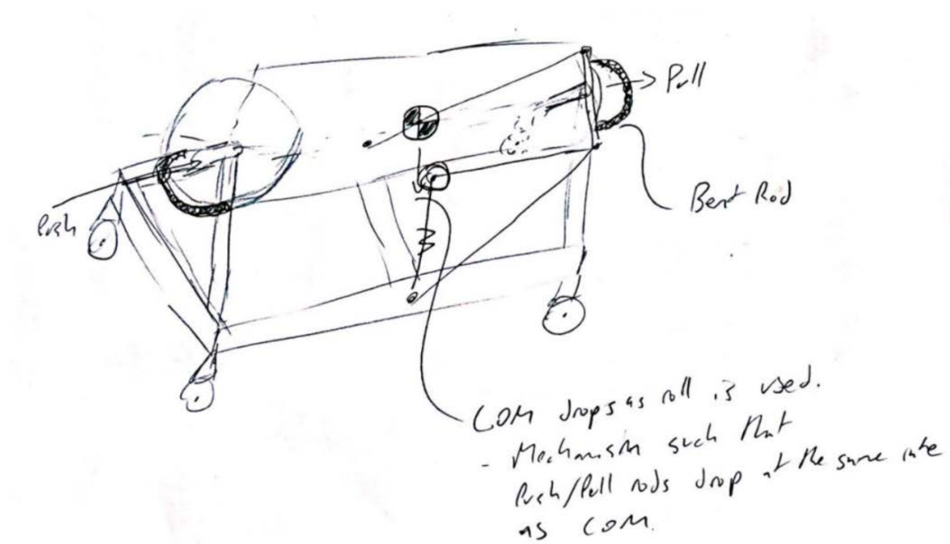


Figure B-27: Concept #22 sketch [8].

Concept #23

Concept #23 utilizes the idea of supporting the composite material roll via a shaft in a cantilever configuration. This way, the roll could be lifted to the proper height via forklift, and the cart's shaft could be rolled into the center of the material. Next, using an internal chucking mechanism, a screw could be rotated to expand the internal grips until they supply sufficient force to hold the material. After the mechanism is set, some means of locking it would be required.

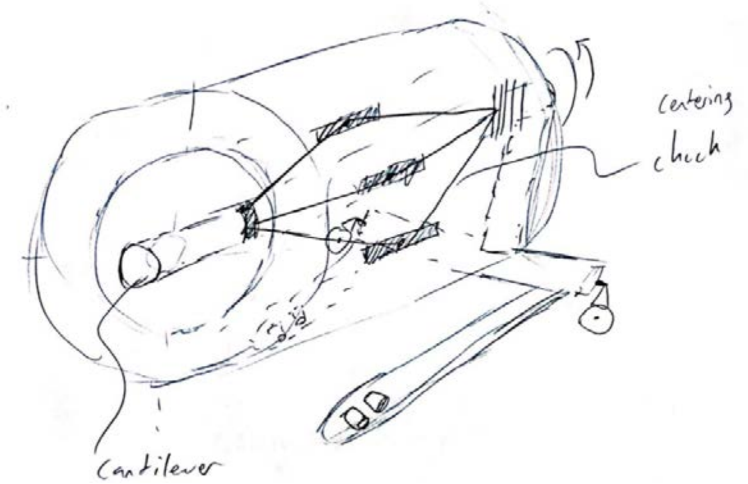


Figure B-28: Concept #23 sketch [8].

Concept #24

Concept #24 utilizes a shaftless configuration where two supporting rollers hold the composite material roll. An additional roller which applies force to the top of the material is used to help constrain the expansion of composite material. The third roll is applied by rotating it via a perpendicular pivoting rod such that the roller is free to rotate out of the way during changeovers, and back down to match diameter of composite material as it is used.

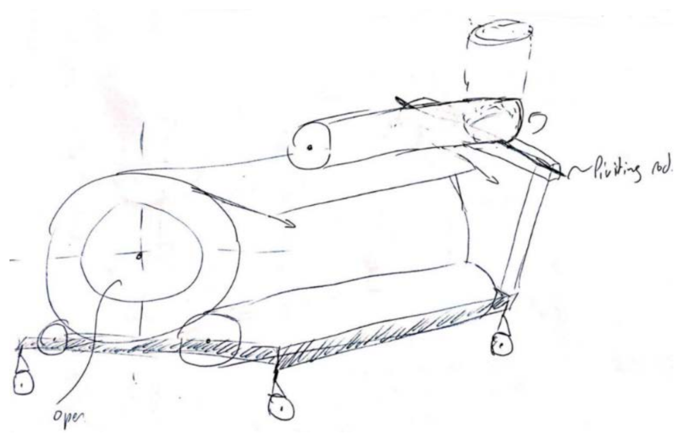


Figure B-29: Concept #24 sketch [8].

Concept #25

Concept #25 works by placing the composite material roll into a cage or shell via the use of a centered shaft. After being inserted into the cage, the top side would be closed such that the composite material would be permanently locked inside. A slot approximately equal to material thickness would be cut into the cage such that the material could be uncoiled. Additionally, the cage would restrict the roll from expanding to more than the designed diameter.

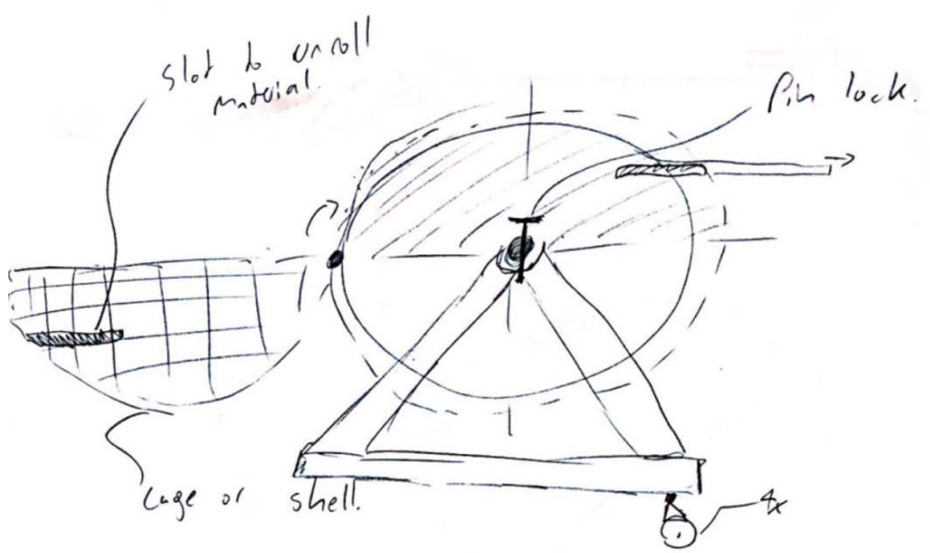


Figure B-30: Concept #25 sketch [8].

Concept #26

Concept #26 uses a large, capsule like frame to retain the composite material roll. The material would be inserted into the frame from the side, and would rest between three supporting rolls. These rolls would use a chucking system, which push off the fixed capsule wall. As a lead screw is turned, each roller applies pressure against the composite material. This pressure would be the primary means of eliminating composite material expansion. Additionally, due to the nature of a capsuled design, coiling, uncoiling, and chucking mechanisms could readily be automated via motors without worrying about additional safety concerns.

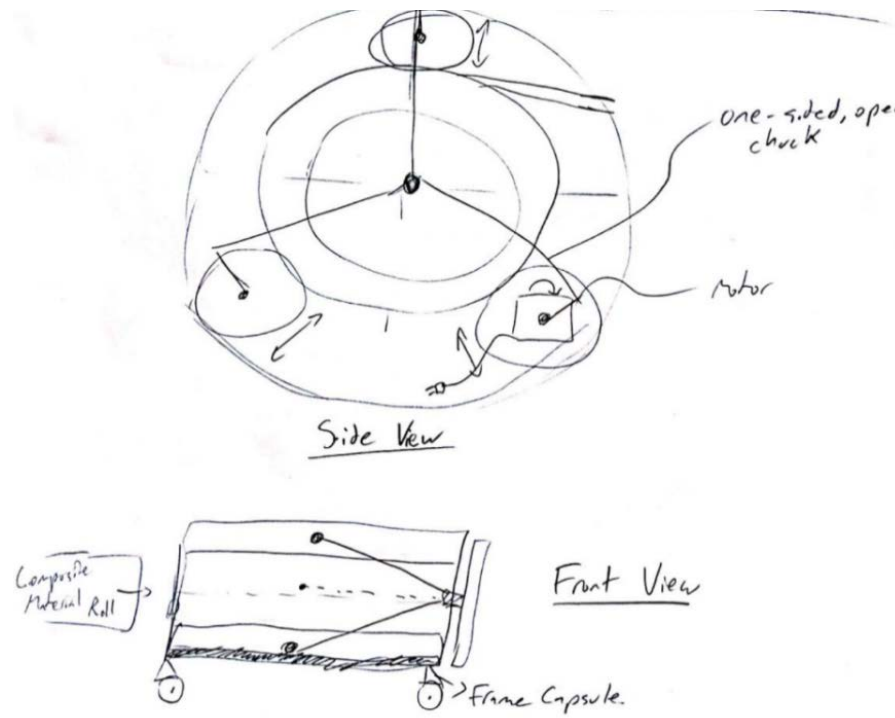


Figure B-31: Concept #26 sketch [8].

Concept #27

Concept #27 utilizes the idea of a shaftless configuration. With the addition of supporting the composite material roll via two rollers, a third roller on top would be used to constrain the expansion of the composite roll. This third roller would be attached to a frame mechanism and slot that would lock the roll to the top region of the composite material when in use. During change overs, some sort of pin release would allow the roller to completely move out of the way, allowing adequate space for a forklift.

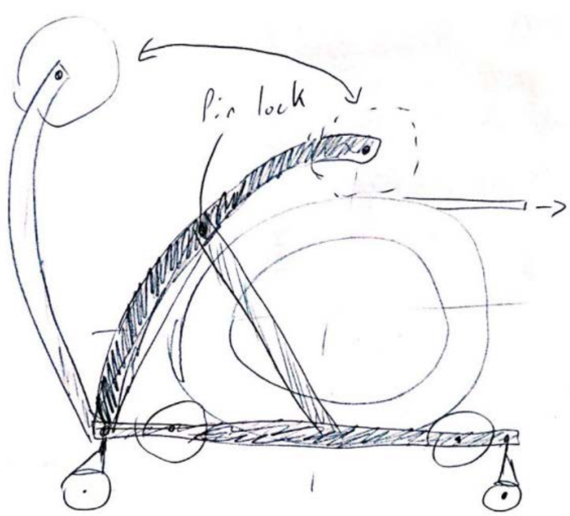


Figure B-32: Concept #27 sketch [8].

Concept #28

Concept #28 works by placing the composite material roll onto two supporting rollers, which are further supported by a series of springs. Ideally, the springs under the rollers would produce extra pressure to the outside of the composite roll such that expansion could be fully retained with only two rollers. Additionally, a crank mechanism could be used to power one of the no-slip rollers such that the composite roll may be uncoiled or recoiled.

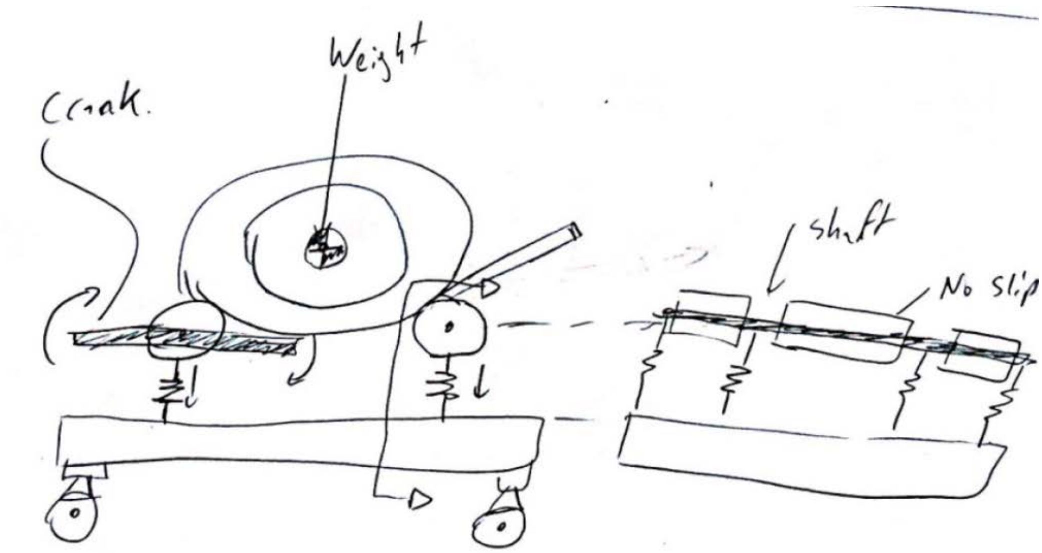


Figure B-33: Concept #28 sketch [8].

Concept #29

Concept #29's design focuses on the mobility of the cart, supporting the weight of the roll, retaining the roll and making the process of coiling and uncoiling easier. The roll of the material sits on two shafts which have four wheels placed equally across the shaft. It also consists of a handle for easy mobility and a locking mechanism to secure the material in place. The composite material roll passes through the locking mechanism which is made up of two rubber rollers. The center shaft has a mechanism that tightly fits with the inner space of the composite roll.

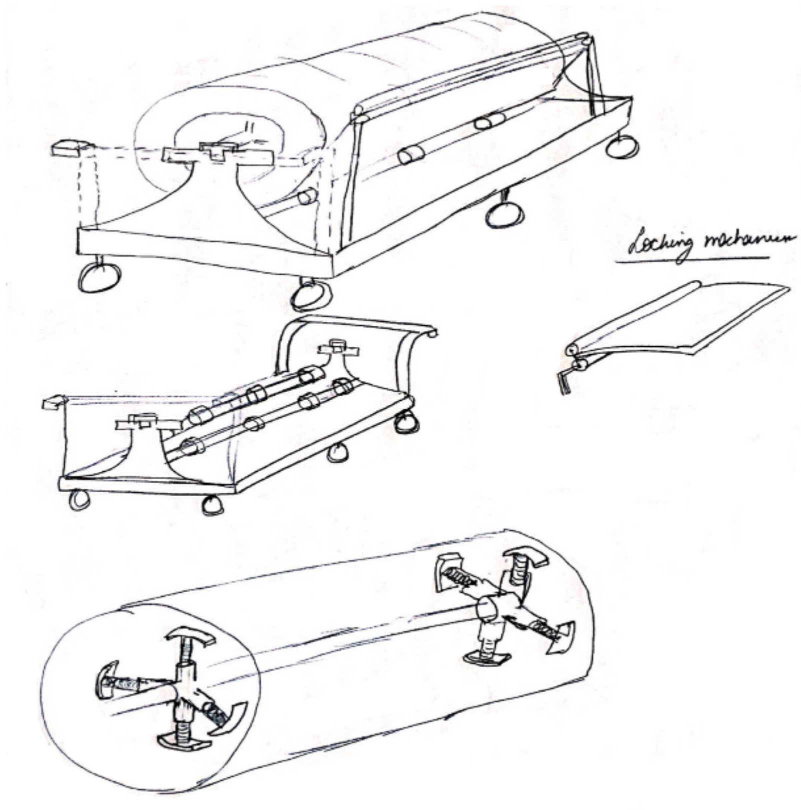


Figure B-34: Concept #29 sketch [9].

Concept #30

Concept #30 is a cage design which focuses on retaining and supporting the roll. It has a locking mechanism made up of two rubber rollers and a center shaft with supports at each end. The roll sits on two shafts with rubber wheels and passes through the locking mechanism. This design also consists of shafts with wheels on top of the roll as the cage door is locked. This is to help with retaining and uncoiling the composite material.

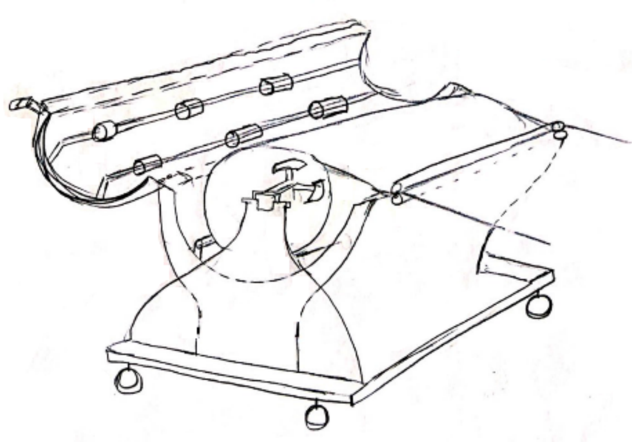


Figure B-35: Concept #30 sketch [9].

Concept #31

Concept #31 focuses on the ease of replacing the roll and reducing the amount of effort required for coiling and uncoiling processes. This design consists of a fixed side that is attached to a shaft. One side of the shaft is fixed while the other rests on a slot similar to Concept #30. The rest of the sides can be lowered down for ease of replacing the roll. Two sides have two shafts with rollers to reduce coiling and uncoiling effort.

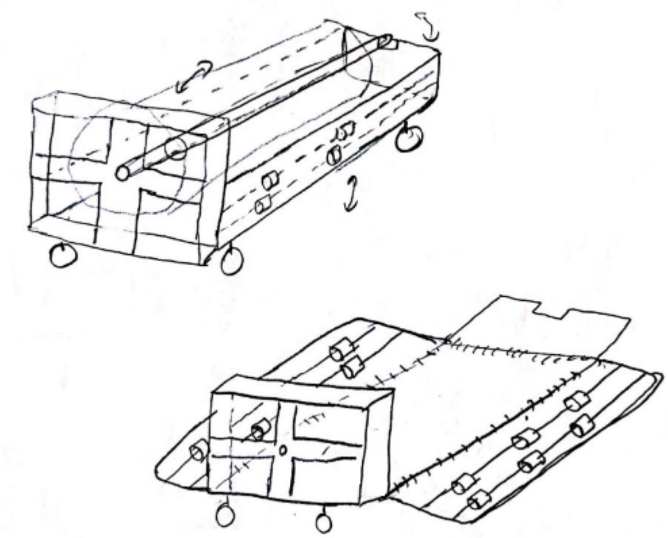


Figure B-36: Concept #31 sketch [9].

Concept #32

Concept #32 is concentrated of a different approach to meet the design specifications. Instead of putting the roll horizontally, the roll was placed vertically on a shaft with supports on each end. The design has four walls with shafts and wheels for supporting the roll of material. These walls can be moved out of the way for ease of replacing the roll. The design also has a handle for ease of mobility.

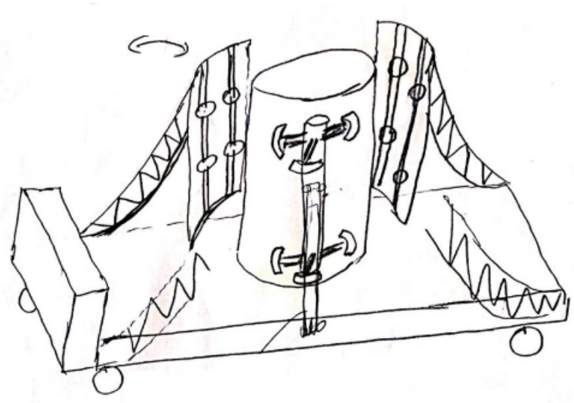


Figure B-37: Concept #32 sketch [9].

Concept #33

Concept #33 consists of a vertical shaft that supports the weight of the roll. The design has two shafts placed on the sides of the roll for retaining the roll. Both supporting shafts can be moved and locked in place depending on the thickness of the roll. Rollers are added for ease of coiling and uncoiling processes. This design also consists of a handle for easy mobility.

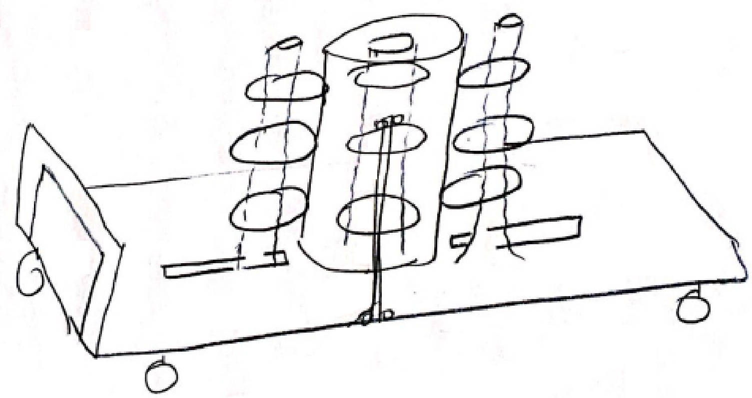


Figure B-38: Concept #33 sketch [9].

Concept #34

Concept #34's design is aimed at making the process of coiling and uncoiling easy. The roll of the material rests on a shaft with supporting mechanism on both ends of the roll. The material uncoiling is carried out with the help of two rollers, one placed at the bottom and the other on top.

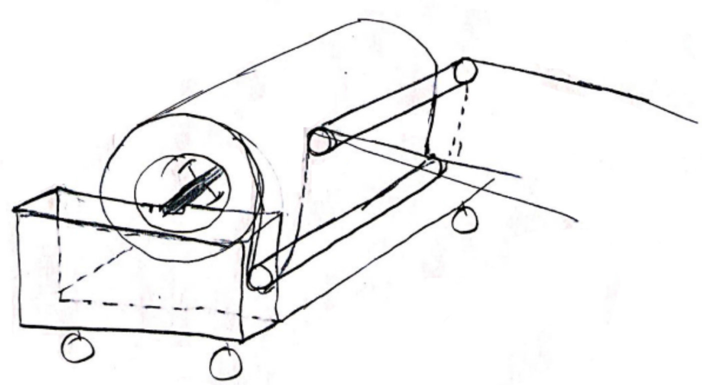


Figure B-39: Concept #34 sketch [9].

Concept #35

Concept #35 contains three main devices which include a self-centering shaft, roll expansion constrainer and roll expansion constrainer adjuster, shown in Figure B-40, Figure B-41, and Figure B-42, respectively.

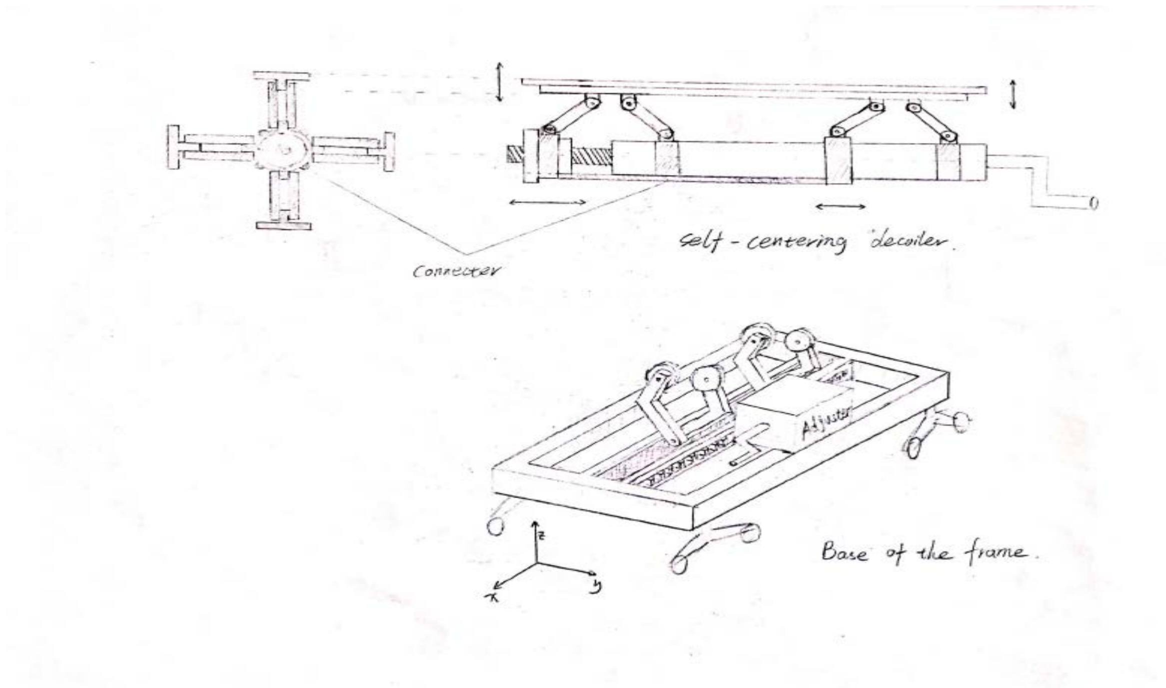


Figure B-40: Concept #35 sketch 1 of 3 [10].

In Figure B-40, from the side view, the self-centering shaft contains four stents. These stents will support the composite material on the inside diameter. The working principle of the self-centering shaft is shown in the front view. When rotating the rotor from left side, the rotor will move horizontally on the screw associated with the connecter. Thus, the maximum diameter of the shaft can be adjusted to match the inner diameter of the composite roll.

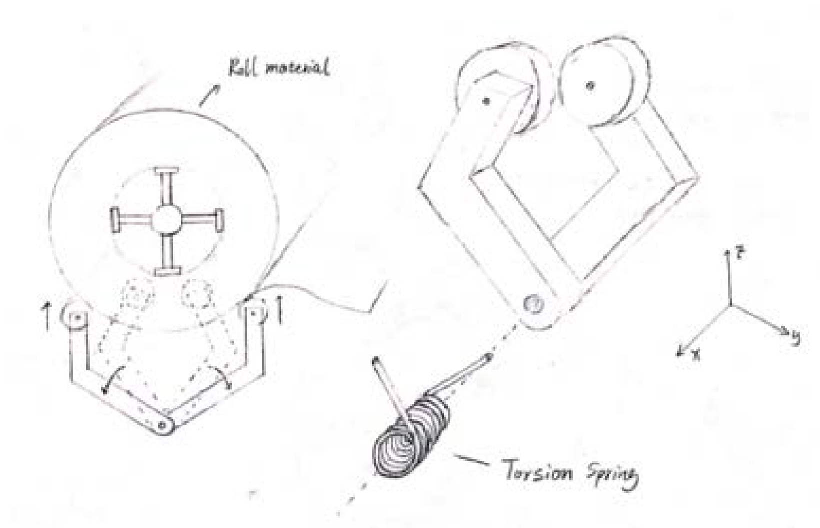


Figure B-41: Concept #35 sketch 2 of 3 [10].

In Figure B-41, there are two roll expansion constrainers set on the middle of the base to prevent expansion. This device uses torsion springs to apply a pressure to the bottom side of the composite material roll. This is ideal because the expansion of the composite material will always occur from the bottom side, due to gravity.

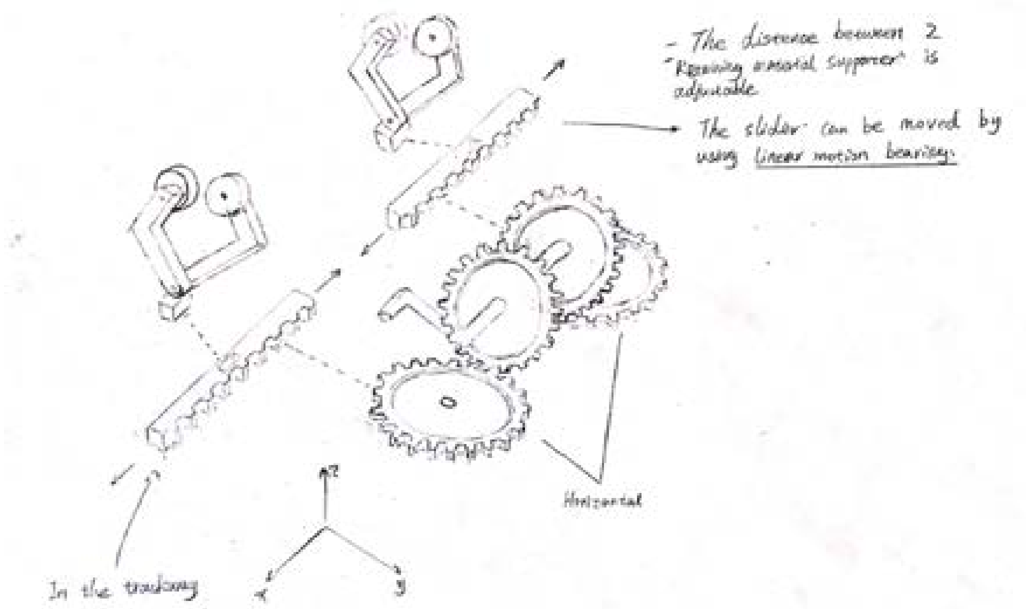


Figure B-42: Concept #35 sketch 3 of 3 [10].

In Figure B-42, the distance between the two roll expansion constrainers can be equally adjusted by the mechanism shown. The two roll expansion constrainers are separately set on two sliders. Each slider runs horizontally and is installed perpendicular to the base plates. The distance between the two sliders can be adjusted by rotating the gear set. This device can be used for different sizes of roll material, which is not necessarily within the project scope.

Concept #36

Concept #36 uses two push spring roll expansion constrainers which apply force to the bottom side of the composite material. The constrainers will keep applying an upward pressure on the bottom side due to a spring buffer. The spring buffer could be either inside or outside of a chamber depending on safety or maintenance needs.

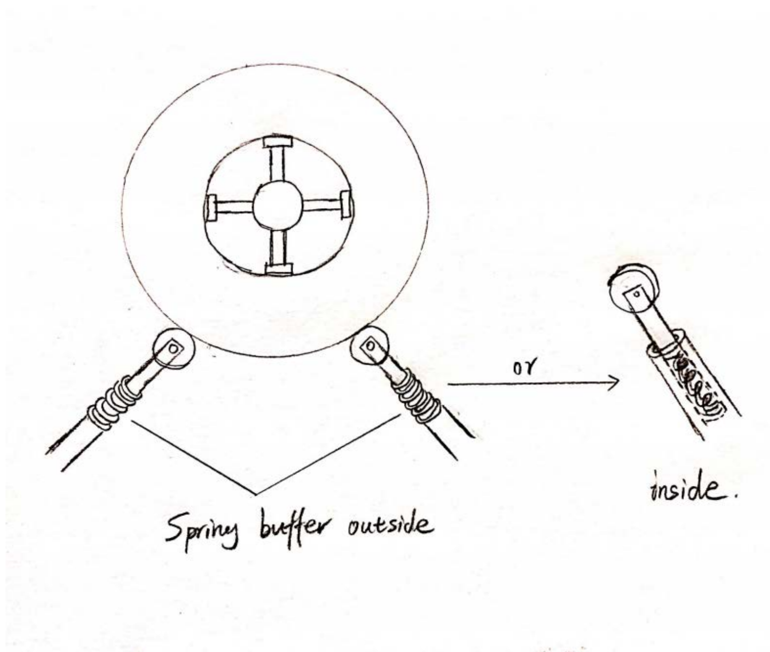


Figure B-43: Concept #36 sketch [10].

Concept #37

Concept #37 is based on a roll expansion constrainer and roll supporting method without a shaft. The spring system will be extracted when the roll of composite material sits on it. The spring on both sides will expand such that the rollers will support the material, aiding with the coiling and uncoiling processes.

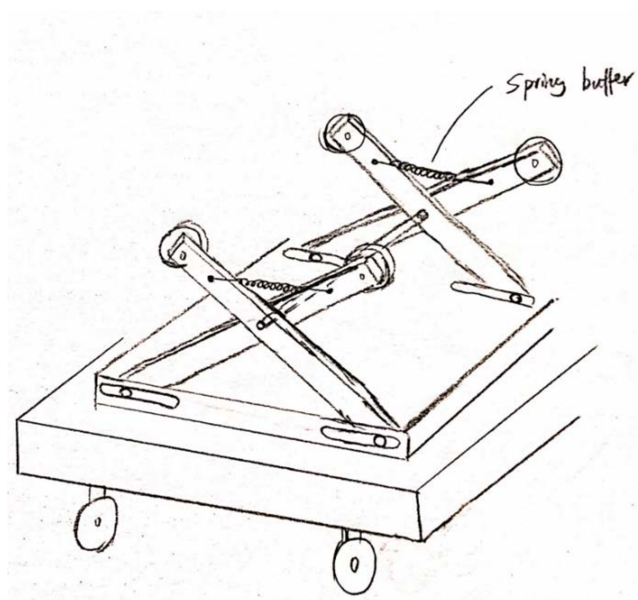


Figure B-44: Concept #37 sketch [10].

3. Concept Screening

With the large number of concepts developed, the team decided to first screen all 37 by dividing them into four categories: ability to support the composite roll, ability to constrain expansion of the composite material, ability to reduce movement effort, and ability to reduce coiling and uncoiling effort. Each category was given its own set of selection criteria which personally relate to that category for evaluation. Due to the nature of concepts generated, and each concept not necessarily falling into specific singular categories, the team did a quick visual scan of all concepts and mutually decided if each concept should be included in the screening process of each particular category.

To compare each concept against each other, each was rated against the current cart being used, (referred to as Concept #0 in all screening tables) as a reference point for each selection criteria being screened. If a concept was better than current cart, with respect to a particular selection criterion, it was given a plus. If a concept was worse than the current cart, it was given a minus. Additionally, if a concept was approximately the same as the current cart, that selection criteria rating was left blank. In extreme cases of ranking concepts better or worse than the current cart being used, double-plus or double-minus was given to show this severity. All cases where the team deemed this necessary are further explained along with each selection criteria in all four categories.

To compare all selected concepts for each category, a net score for each concept was calculated by adding all pluses, and subtracting all minuses for that concept. Concepts were than ranked, highest to lowest based on their total net score. Depending on the ranking results in each category, a number of concepts were selected to generate preliminary CAD models which could be used to further communicate the ideas to the client. These concepts chosen are shown in their respective categories in the following sub-sections.

3.1. Ability to Support the Composite Roll

When concept screening for the ability to support the composite roll category, it should be noted that only aspects of the sketched concepts that relate to supporting the roll were considered. Thus, within the category, the team decided to include five different selection criteria based on the needs analysis results: safety, roll exchange, uncoiling effort, manufacturability, and minimize footprint. The meaning of each criterion is explained below.

Safety - This refers to the overall safeness of the proposed concept.

Roll Exchange – How easy or long it would take to changeover a new roll of composite material using by using a forklift.

Uncoiling Effort – How the concept may either help or hinder the ability to coil the material.

Manufacturability – How difficult it may be to manufacture the proposed concept.

Minimize Footprint – How small or big the concept may be compared to the current footprint of the cart.

The concepts selected for this category and screening results are shown in Table B-I.

Table B-I: Concept Screening of Ability to Support the Composite Roll

Concept Screening	Concept Numbers																
Selection Criteria	0	1	2	3	4	16	18	21	23	24	27	28	29	32	33	35	37
Safety		-	+	+	+	-			+	+	+	-	+			+	-
Roll Exchange		+				-	-							-	-		+
Uncoiling Effort		+				+	+	+		+	+	+	+	+	+		+
Manufacturability		+		-		-			-			-	-	-	-	-	
Minimize Footprint																	
Sum of +'s	0	3	1	1	1	1	1	1	1	2	2	1	2	1	1	1	2
Sum of 0's	5	1	4	3	4	1	3	4	3	3	3	2	2	2	2	3	2
Sum of -'s	0	1	0	1	0	3	1	0	1	0	0	2	1	2	2	1	1
Net Score	0	2	1	0	1	-2	0	1	0	2	2	-1	1	-1	-1	0	1
Rank	9	1	4	9	4	17	9	4	9	1	1	14	4	14	14	9	4
Continue?	N	Y	N	N	N	N	N	N	N	Y	Y	N	N	N	N	N	N

In Table B-I, Concepts #1, #24, and #27 were chosen as viable ways to better support the roll compared to the current cart. It should be noted that all three of these top choices focus on a shaftless design. This selection was further reinforced based on the client's feedback when discussing possible concepts, as described later in the Secondary Concept Generation section.

3.2. Ability to Constrain Expansion of the Composite Material

When concept screening for the ability to constrain expansion of the composite material category, only mechanisms which provided additional material expansion constraining were considered. For concepts in the category, the team decided to include six different selection criteria based on the needs analysis: performance, manufacturability, minimize footprint, roll exchange, safety, and recoiling. The meaning of each criteria is explained below.

Performance – How well the proposed concept will actually constrain the expansion of the material. Note, some concepts for this criterion were given a double-plus were performance was expected to well exceed some of the average concepts, due the high importance of this need from the client.

Manufacturability – How hard the concept may be to manufacture.

Minimize footprint – How well the concept may fit into the proposed design space.

Roll Exchange – How difficult a new roll may be to exchange into the concept via forklift.

Safety – How safe the constraining mechanisms are.

Recoiling – How the constraining mechanism may impede or help recoiling of the composite material.

The concepts selected for this category and screening results are shown in Table B-II.

TABLE B-II: CONCEPT SCREENING OF ABILITY TO CONSTRAIN EXPANSION OF THE COMPOSITE MATERIAL

Concept Screening	Concept Numbers																										
Selection Criteria	0	1	10	11	12	13	14	15	16	17	18	19	21	24	25	26	27	28	29	30	31	32	33	34	35	36	37
Performance		+	+			++	+	++	+	+	+	++	++	+	+	++	++	+		+	+	++	++	+	+	+	+
Manufacturability			-	-		-		-	-	-		-			-	-		-	-	-	-	-	-		-	-	
Minimize Footprint		+				+		-	-	+	+	-		+		-	-					-	-	-	+		+
Roll Exchange		+				-					-	-		+	-	-		+		-	-	-	-				+
Safety		-	+	+	+	+	+	+	-	+		+			+	+		-	+	+	+	-	-	+	+	+	-
Recoiling			+	+	+	+		+		+		-							+			-	-		+	+	
Sum of +'s	0	3	3	2	2	5	2	4	1	4	2	3	2	3	2	3	2	2	2	2	2	2	2	2	4	3	3
Sum of 0's	6	2	2	3	4	0	4	1	2	1	3	0	5	3	2	1	4	2	3	2	2	0	0	3	1	2	2
Sum of -'s	0	1	1	1	0	2	0	2	3	1	1	4	0	0	2	3	1	2	1	2	2	5	5	1	1	1	1
Net Score	0	2	2	1	2	3	2	2	-2	3	1	-1	2	3	0	0	1	0	1	0	0	-3	-3	1	3	2	2
Rank	18	5	5	13	5	1	5	5	25	1	13	24	5	1	18	18	13	18	13	18	18	26	26	13	1	5	5
Continue?	N	N	N	N	N	Y	N	N	N	Y	N	N	N	Y	N	N	N	N	N	N	N	N	N	N	Y	N	N

In Table B-II, Concepts #13, #17, #24, and #35 all tied for the top rank. Thus, these concepts were continued forward in the selection process.

3.3. Ability to Reduce Movement Effort

Similar to the previous categories, only concepts were directly related to reducing the amount of effort it takes to move and maneuver the composite roll are considered. Thus, the team decided to choose five selection criteria to evaluate concepts: Floor compatibility, effort to move, manufacturability, safety, and number of people required.

Floor compatibility – The compatibility of the concept with the current floor tracks being used in the process.

Effort to Move – The required force to reasonable accelerate or maneuvered the concept. Note, in cases where the team felt effort to move a concept was greatly reduced, a double-plus was given to express the importance of said need.

Manufacturability – How hard the concept may be to manufacture.

Safety – How safe the concept would be to move.

Number of People Required – How many people, at minimum, would be required to move the concept.

The concepts selected for this category, and screening results, are shown in Table B-III. Note, Concept #20, as seen in Table B-III, was evaluated both with and without a driving motor to differentiate the pros and cons of both configurations.

TABLE B-III: CONCEPT SCREENING OF ABILITY TO REDUCE MOVEMENT EFFORT

Ability to Reduce Movement Effort							
Concept Screening	Concept Numbers						
Selection Criteria	0	5	6	7	20*	20**	22
Floor Compatibility							
Effort to Move		+	+	++	+	++	+
Manufacturability				-			-
Safety			+	-		-	+
# of People Required				+		+	
Sum of +'s	0	1	2	1	1	1	2
Sum of 0's	5	4	3	1	4	2	2
Sum of -'s	0	0	0	2	0	1	1
Net Score	0	1	2	-1	1	0	1
Rank	5	2	1	7	2	5	2
Continue?	N	Y	Y	N	Y	N	Y

Note: * - implies no motor is used, ** - implies a motor is used

In Table B-III, Concept #6 was ranked the highest, whereas Concepts #5, #20 without a motor, and #22 tied for second. Thus, these concepts were chosen to move forward with. Note, at this stage the team identified that concepts utilizing different wheel configurations could be combined with concepts utilizing handles to further reduce the movement effort of the design. Using both would also be beneficial in terms of safety for all operators who must use the design.

3.4. Ability to Reduce Coiling and Uncoiling Effort

In the ability to reduce coiling and uncoiling effort category, only components of the selected concepts which were meant to specifically address this issue were considered. Thus, the team decided to select a total of seven selection criteria: Manufacturability, number of people required, effort to uncoil, effort to recoil, cost, safety, and roll exchange.

Manufacturability – How hard the coiling mechanism is to manufacture.

Number of People Required – The number of people required during the uncoiling or recoiling process. Note, when rating concepts relative to the current cart, it was always the case where the process improved due to the requirement that the design must be operated by two individuals maximum. However, in the case where the team believed only one person was needed to operate the concept, such as where a motor was used, the team assigned a double-plus to show the vast improvement.

Effort to Uncoil – The amount of force and time required to uncoil the composite material.

Effort to Recoil – The amount of force and time required to recoil the composite material.

Cost – The cost to implement the concept's solution. Although cost is not the primary concern of this project, any automation techniques at this stage will be considered as an extra expense.

Safety – How safe the coiling mechanism is.

Roll Exchange – How hard the changeover process via forklift is due to the coiling mechanism. Note, in cases where the team thought the exchange of the material was significantly harder due to the coiling mechanism, a double-minus was given to said concepts.

The concepts selected for this category and screening results are shown in Table B-IV.

TABLE B-IV: CONCEPT SCREENING OF ABILITY TO REDUCE COILING AND UNCOILING EFFORT

Concept Screening	Concept Numbers							
Selection Criteria	0	8	9	10	11	18	28	29
Manufacturbility			-	-	-		-	-
# of People Required		++	++	++	++	+	+	+
Effort to Uncoil		+	+	+	+			-
Effort to Recoil		+	+	+	+	+		-
Cost		-	-	-	-			
Saftey		-	-	-	-	+		
Roll Exchange		--						
Sum of +'s	0	4	4	4	4	3	1	1
Sum of 0's	6	1	0	0	0	3	4	2
Sum of -'s	0	4	3	3	3	0	1	3
Net Score	0	0	1	1	1	3	0	-2
Rank	5	5	2	2	2	1	5	8
Continue?	N	N	Y	Y	Y	Y	N	N

In Table B-IV, Concept #18 was ranked highest, whereas Concepts #9, #10, and #11 were tied for second. Thus, these concepts were chosen to move forward with.

4. Secondary Concept Generation

After screening all concepts that were generated by the team, preliminary CAD models were developed of each concept selected to move forward with. These models had two primary purposes: One, to better understand the limitations and details of each concept that would further need to be worked out. Two, to be used to communicate the team's ideas with the client. Shortly after concept screening, the team setup a meeting with the client to showcase and explain all concepts and ideas the team had generated, particularly focusing on the selection process and results of concept screening. From the client, the team received feedback that in general, all shaftless concepts were particularly liked, and the client asked the team to move forward with the idea. Additionally, Triple E RV employees mentioned some of their concerns and ideas while moving forward on a shaftless design. Some suggestions such as ensuring the center of mass stays in the middle of the cart as the composite material roll is used, (so the cart cannot flip due to its own weight) and ensuring that there is some other external means of fixing the composite material roll to the cart in the event the composite roll was bumped or cart were overturned by accident. Emphasis was given to these points moving forward.

Based on the meeting with the client, the team decided to break up the remaining components of the cart to be selected into four primary categories: ability to constrain expansion of composite material, ability to positively retain the composite roll, ability to reduce movement effort, and ability to reduce coiling and uncoiling effort. Taking a more structured approach than with concept generation, the team also decided to combine some of the previously chosen concepts together, as well as include new ideas developed during the meeting with the client in the concept scoring phase. All concepts from this point on were particularly designed to cover only their respective functional category, and thus are divided into individual subsections below. Additionally, at this point the team felt it was important to name each concept as well as change the numbering system from numerals to letters, such that concepts may be referred to more easily.

4.1. Ability to Constrain Expansion of Composite Material

The following concepts are based on constraining expansion of composite material of the design. Original and new concepts within this category are further explained below.

Concept a. – Top Spring Assembly

Concept a, Top Spring Assembly is based on combining the ideas of a shaftless, two roller support design with Concept #13's double roller spring tension material constraining design. Between all four rollers on which the roll of material rests, and spring tension to apply compression to the top of the composite material roll, the roll is constrained from expanding.

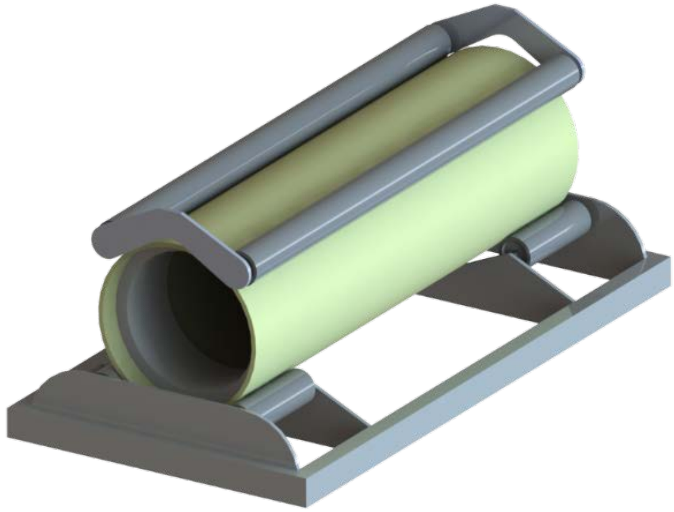


Figure B-45: Concept a. – Top Spring Assembly [11].

Concept b. – Back Roller Arm

Concept b, Back Roller Arm further advances the idea of applying a top roller to the composite material roll developed in Concept #27. As seen in the model, the two supporting rollers were divided into three separate rolls, thus allowing space for forklift arm to fit through. This allows a forklift to easily deposit a roll of composite material between the supporting rollers, while allowing the forklift to easily back out afterwards.

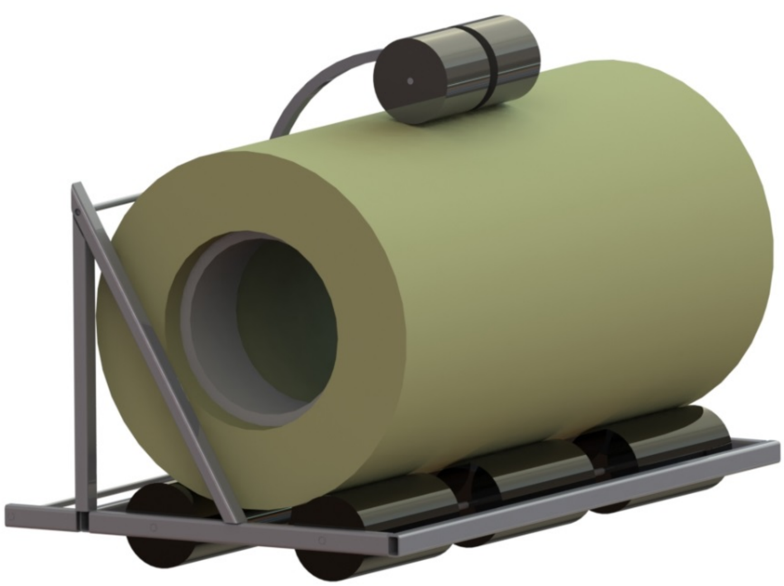


Figure B-46: Concept b. - Back Roller Arm [11].

Concept c. – Side Roller Arm

Concept c, Side Roller Arm, combines some of the features within Concepts #5 and #24. The primary concern with Concept #24 was that when the top roller was tilted onto the composite roll, only one point of the roller would be held down due to the geometry. By replacing the roller with a rotating arm, and applying another rotating bracket to a horizontal hanging roller, the revised concept would allow the entire length of the top roll to be applying pressure to the top of the composite material roll at all times.

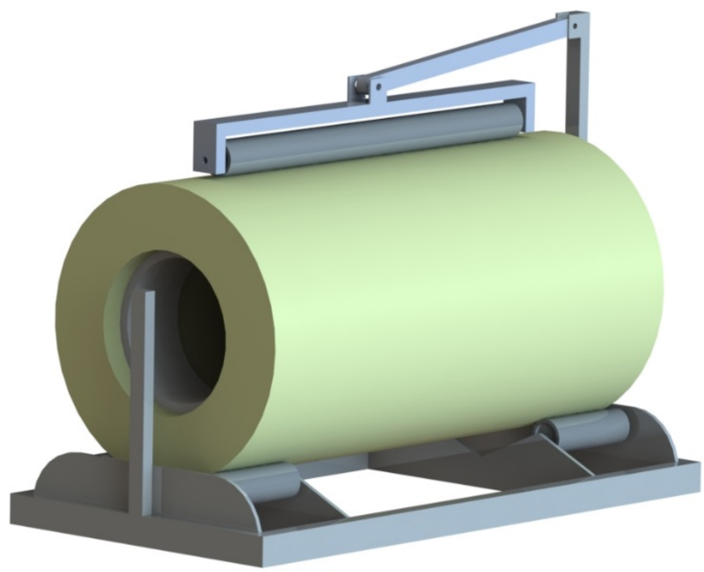


Figure B-47: Concept c. – Side Roller Arm [11].

Concept d. – Linkage Mechanism

Concept d, Linkage Mechanism, addresses the issue with Concept b in that the top roller may not contact the roll of composite material directly on top, depending on the outer diameter of the roll at the time. By using a linkage mechanism to support the overhead roller, the relative lengths and placement of the links can be chosen such that the overhead roller remains as close to the center line of the roll of material as possible through the range of outer diameters seen through the life of the roll.

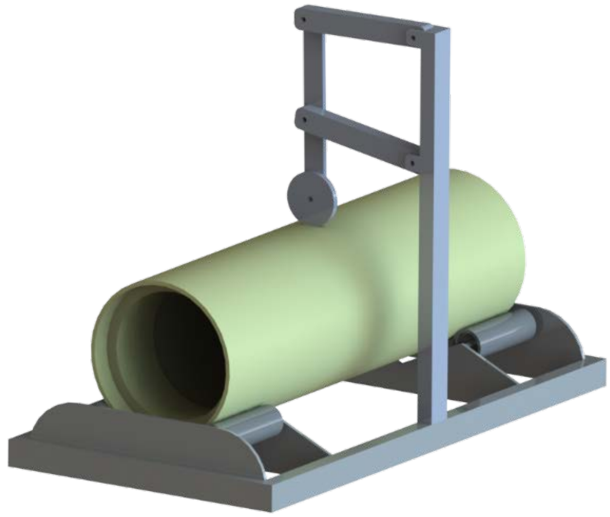


Figure B-48: Concept d. – Linkage Mechanism [11].

4.2. Ability to Positively Retain the Composite Roll

The following concepts are based on retaining the composite material roll within the design space in the event where the design may be overturned. Original and new concepts within this category are further explained below.

Concept e. – No Additional Roll Constraining

Concept e, No Additional Roll Constraining, refers to using the selected concept from ability to constrain expansion of composite material category, combined with the shaftless, two roller supports solely to lock the material within the design. These previously chosen series of three-rollers are present on all concepts in the “ability to constrain expansion of composite material” category, and thus no additional locking mechanism would be necessary.

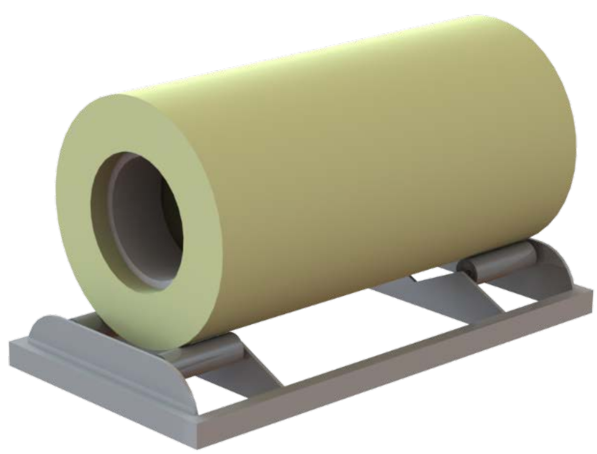


Figure B-49: Concept e. – No Additional Constraining [11].

Concept f. – Shaft

Concept f, Shaft, is based on inserting two short "shafts" into each side of the composite roll, after dropping it into the cart. These shafts would use a horizontal bar to freely move up and down the sides of the frame of the cart, forcing the roll to remain perfectly centered. The shafts would be positively locked via some series mechanisms such that after inserted into the composite roll, such that the roll must physically stay fixed within the cart until the locks are released.

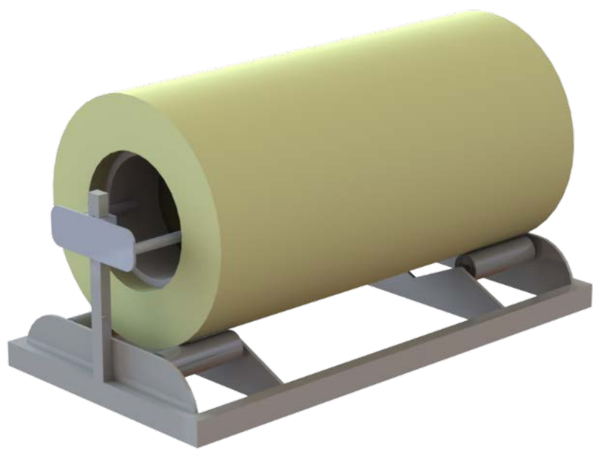


Figure B-50: Concept f. – Shaft [11].

Concept g. – Chain

Concept g, Chain, refers to the simplistic idea of running a chain through the inside diameter of the composite material after being dropped on to the cart. The chain would be tightened and locked such that the composite roll would be permanently fixed to the bottom of the cart. During changeovers, the chain would need to be unlocked, removed, replaced and relocked for safety.

Figure B-51: Concept g. – Chain [11].

Concept h. – Vertical Posts

Concept h, Vertical Posts, uses again a simplistic idea of running higher vertical posts on each side of the composite roll, such that if the composite roll were to roll off a supporting roller it would be caught by the post before coming off the cart. Additionally, another locking and unlocking bar could be run on top of the vertical posts such that the composite roll would always be constrained to the footprint of the cart.

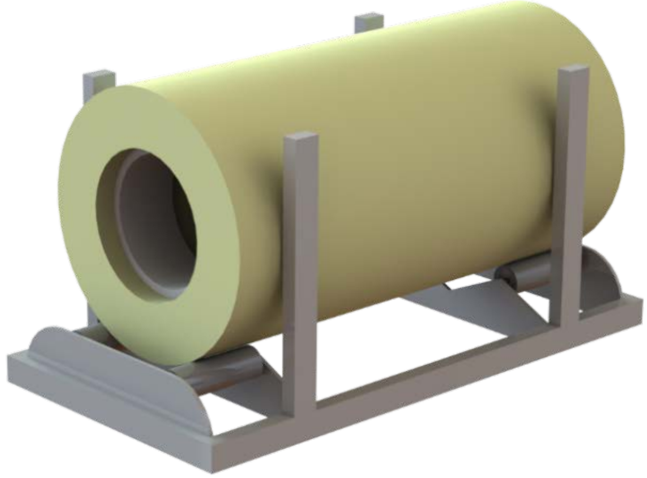


Figure B-52: Concept h. – Vertical Posts [11].

Concept i. – Additional Rollers

Concept i, Additional Rollers, utilizes a similar idea as Vertical Posts by physically adding stops to the outside of the cart to prevent the composite roll from falling off. However, in the case of Additional Rollers, four outside rollers would be fixed such that if the composite material were to roll into the extra rollers during use, the additional rollers would apply more rolling friction such that the composite material keeps coiling and uncoiling properly. Additionally, these additional rollers could be specified based on the current rollers which must be designed to support the composite roll, and thus could reduce manufacturing variability.

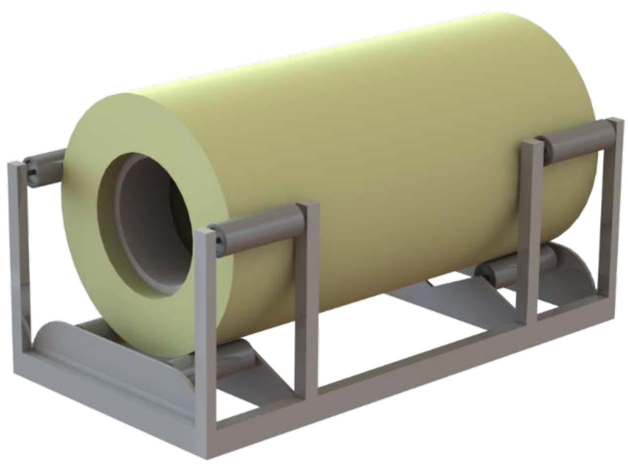


Figure B-53: Concept i. – Additional Rollers [11].

4.3. Ability to Reduce Movement Effort

The following concepts are based on the reducing movement effort of the design. Original and new concepts within this category are further explained below.

Concept j. – Walking Axle Casters

Concept j, Walking Axle Casters, is directly based off Concept #5. By effectively modelling the concept, the team found careful attention will need to be given to bearing stresses between the walking axles, frame, and casters for the design to effectively work. However, if successful, the added casters should all carry the same weight due to symmetry about the frame of the cart. These additional casters will further reduce the rolling and ball-bearing frictions of the cart, thus allowing it to maneuver more easily.

Figure B-54: Concept j. – Walking Casters [11].

Concept k. – Four Casters

Concept K, Four Casters, utilities the existing configuration when simply four casters are fixed to the four corners of the cart frame.

Figure B-55: Concept k. – Four Casters [11].

Concept l. – Double Wheelbarrow

Concept l, Double Wheelbarrow, is a design idea presented to the team based on the client's other in-house developed carts. In this concept, two large, non-swiveling wheels are fixed on a shaft along the middle of the cart. Additionally, at each end of the cart a single swivel caster is used. The wheels in the middle of the cart are slightly bigger such that depending on the weight or other kinetic forces on the cart, one of the swivel wheels touches the ground while the other does not. This creates enables greater motion and maneuverability of the cart like a wheelbarrow. However, due to lesser wheels supporting the carts load, this may significantly increase rolling friction of the cart.

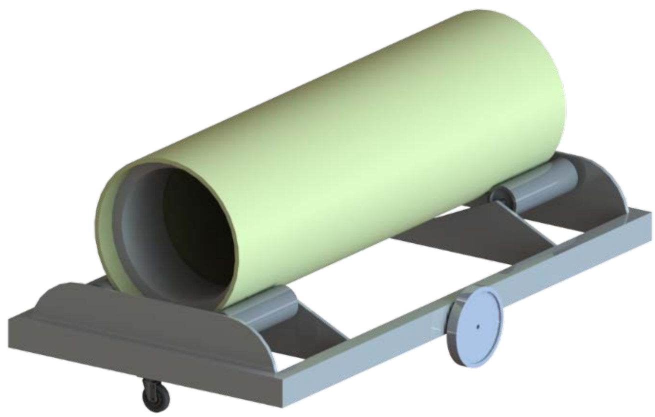


Figure B-56: Concept l. – Double Wheelbarrow [11].

Concept m. – Large Center Wheel

Concept m, Large Center Wheel, is based on the non-motorized version of Concept #20. As previously mentioned, this wheel will restrict rotation of the cart to when the cart is not moving, but also lower overall rolling friction. After modelling said concept, it should be noted that the large wheel forces the height of the cart to increase due to the fact that the composite material roll may not interfere with the wheel.

Figure B-57: Concept m. – Large Center Wheel [11].

Concept n. – Handle

Concept n, Handle, is based off the idea of the handles proposed in Concepts #6 and #20. Assuming the team decides to integrate handles on each side of the cart, more information as per the best location and configuration of the handles is needed.

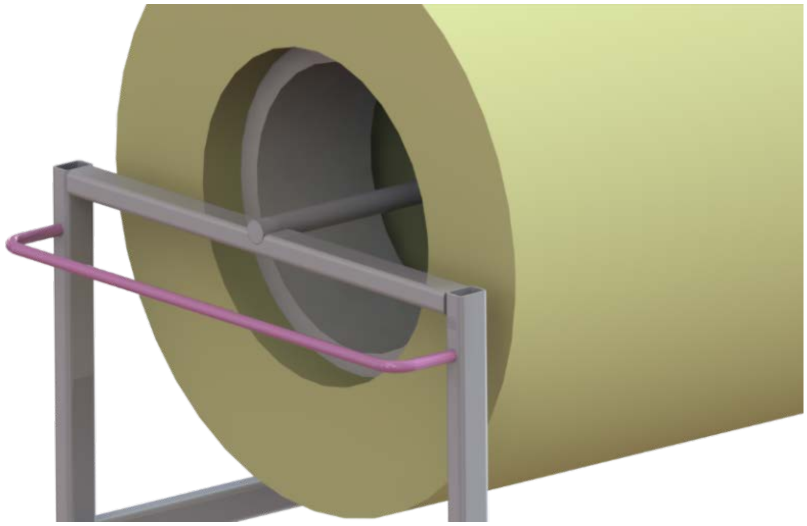


Figure B-58: Concept n. – Handle [11].

4.4. Ability to Reduce Coiling and Uncoiling Effort

The following concepts are based on reducing the effort required to coil and uncoil the composite material roll. Original and new concepts within this category are further explained to below.

Concept p. – Tensioned Roller

Concept p, Tensioned Roller, is based off the idea of powering a motor such that it turns a non-slip roller held tangent to the composite material via spring force. By modeling the concept, the team noticed that the footprint of the cart would need to be increased such that the powered roller may firmed rest on the outside of the composite material roll.

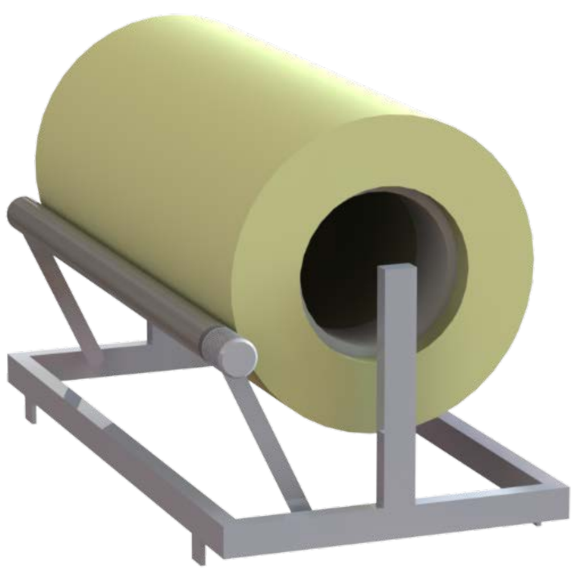


Figure B-59: Concept p. – Tensioned Roller [11].

Concept q. – Bus Wheel

Concept q, Bus Wheel, is based off the "bus wheel" design used to turn a single roller in Concept #18. After modelling the concept, the team determined that it may be difficult to try and manufacture a safe and easy to use bus wheel in-house, and sourcing the part may be the better option.

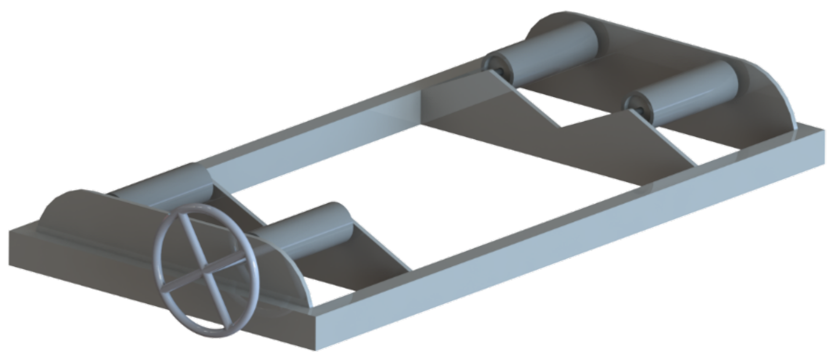


Figure B-60: Concept q. – Bus Wheel [11].

Concept r. – Powered

Concept r, Powered, refers to powering a single external supporting roller, instead of turning said roller by hand, to coil and uncoil composite material. By powering a roller, the amount of physical strain placed on the process operators can be greatly reduced. However, the cost of implementing said motor would be significant.

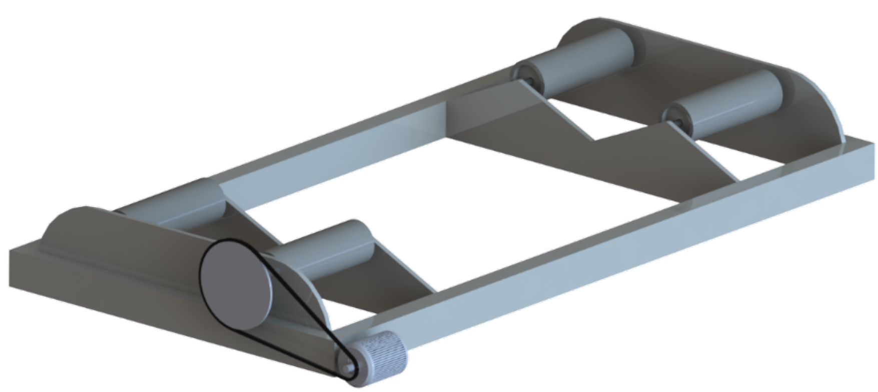


Figure B-61: Concept r. – Powered [11].

Concept s. – Synchronized Bus Wheel

Concept s, Synchronized Bus Wheel, is similar to the design of Concept q. In this concept, the "bus wheel" is used to turn both supporting roller at the same time by linking both rollers via a chain or belt configuration. Ideally, by moving both rolls at the same time, the overall slippage of the supporting rollers and the composite material rolls shall be greatly reduced.

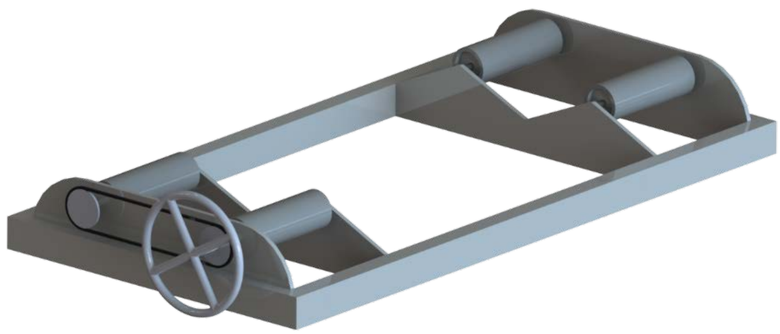


Figure B-62: Concept s. – Synchronized Bus Wheel [11].

Concept t. – Synchronized Powered

Concept t, Synchronized Powered, is based on combining the Concepts r and s. Both supporting rolls would be powered via an external motor which are linked via a chain or belt drive. Again, by moving both rolls at the same time, the overall slippage of the supporting rollers and the composite material rolls shall be greatly reduced.

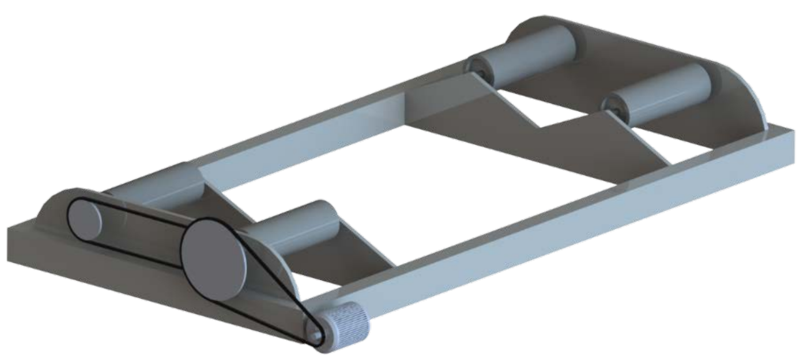


Figure B-63: Concept t. – Synchronized Powered [11].

5. Concept Scoring

With the method of supporting the composite material roll selected based on advice from the client, concepts for the remaining aspects of the design must be chosen. The remaining aspects are a method of maintaining the material in a tightly wrapped roll, a method of securing the roll of material in place, a method of providing an easy uncoiling and recoiling process, and a method of reducing the effort required to move the roll of material. This section will outline the process used to select the optimal concept for each of these aspects.

5.1. Ability to Constrain Expansion of Composite Material

The process for determining the optimal method of retaining the composite material in a tightly wrapped roll began with determining the criteria with which the team would evaluate the performance of each concept. Seven evaluation criteria were selected, and are described below.

Manufacturability – This refers to the difficulty in producing the design, taking into consideration the manufacturing methods available to the client, and any outsourcing that would be required. Amount of required material is also considered.

Constraining Material – This refers to the ability of the design to maintain the composite material in a tightly wrapped roll.

Retaining Roll – This refers to the ability of the design to maintain the roll of material within the footprint of the design, should the roll be bumped, or the design overturned.

Footprint – This refers to any increase in size to the overall design that would result from implementing the concept.

Purchased Items – This refers to the amount of required purchased parts, such as bearings, rollers, and springs.

Forklift Exchange – This refers to the compatibility of the design with the process of exchanging rolls of material with a forklift. Considerations include any time added to the exchange process, any additional steps or procedures that must be taken to exchange rolls of material, and any additional personnel that are required to exchange rolls of material.

Safety – This refers to any safety concerns associated with the design, including exposed springs, pinch points, and large suspended masses.

To determine the relative importance of each criteria, a weighting matrix was used. Each criterion appears along the left side and top of the matrix, with the intersection of the rows and columns representing the two individual criteria being directly compared in terms of importance. The criteria number of the more important evaluation criteria is placed in the intersection cell. After all criteria have been evaluated against each other, the relative importance, or weights, of each criterion can be determined by counting the number of times the criteria's number appears in the matrix.

The results of the weighting process are shown in Figure B-64.

		1	2	3	4	5	6	7
		Manufacturability	Constraining Mateial	Retaining Roll	Footprint	Purchased Items	Forklift Exchange	Saftey
1	Manufacturability		2	3	1	1	6	7
2	Constraining Mateial			2	2	2	2	2
3	Retaining Roll				3	3	3	3
4	Footprint					4	6	7
5	Purchased Items						5	7
6	Forklift Exchange							7
7	Saftey							
	Criteia	1	2	3	4	5	6	7
	Weight	9.5%	28.6%	23.8%	4.8%	4.8%	9.5%	19.0%

Figure B-64: Ability to constrain expansion of the composite material weighting.

With the weights of the evaluation criteria determined, the overall performance of each concept can be evaluated. To do this, a scoring matrix was used. For this matrix, a scale of one to five was used to represent the performance of the concepts. The team used a comparative method of evaluating concepts, in which one of the concepts was chosen as a baseline, and given a score of three in each criterion. All other concepts were then evaluated against this baseline in each criterion, with a score of one representing much worse performance, and a score of five representing much better performance. The total score of each concept was determined by multiplying each score by the weight of that criteria, and summing the results.

The results of the scoring process are shown in Table B-V.

TABLE B-V: ABILITY TO CONSTRAIN EXPANSION OF THE COMPOSITE MATERIAL SCORING MATRIX

Selection Criteria	Concepts				
		Top Spring Assembly	Back Roller Arm	Side Roller Arm	Linkage Mechanism
	Weight	**a**	**b**	**c**	**d**
Manufacturability	9.5%	2	3	2	1
Constraining Mateial	28.6%	2	3	3	2
Retaining Roll	23.8%	4	3	4	4
Footprint	4.8%	4	3	3	2
Purchased Items	4.8%	2	3	3	2
Forklift Exchange	9.5%	1	3	3	4
Saftey	19.0%	3	3	3	2
Total	100%	18	21	21	17
Total Weighted		2.67	3.00	3.14	2.57
Rank		3	2	1	4
Continue?		**N**	**N**	**Y**	**N**

Concept c, Side Roller Arm, received the highest score of 3.14. To validate the scoring process, a sensitivity study was performed. In this study, the weights of each criteria were adjusted and the scores of each concept re-evaluated. The results of this study showed Concept c remaining with the highest score, validating the scoring process.

5.2. Ability to Positively Retain the Composite Roll

As with material retention, the process for selecting the optimal method of retaining the roll of material began with determining the criteria for a successful design. The criteria selected to evaluate concept performance are outlined below.

Manufacturability – This refers to the difficulty in producing the design, taking into consideration the manufacturing methods available to the client, and any outsourcing that would be required. Amount of material is also considered in this category.

Safety – This refers to how well the design retains the roll of material, as well as any safety concerns regarding operating or using the design.

Forklift Exchange – This refers to the compatibility of the design with the process of exchanging rolls of material with a forklift. Considerations include any time added to the exchange process, any additional steps or procedures that must be taken to exchange rolls of material, and any additional personnel that are required to exchange rolls of material.

Maintenance – This refers to any additional maintenance that must be performed on the design, including the inspection or replacement of any bearings or wear surfaces.

The relative importance of these criteria was evaluated against each other using the weighting matrix described earlier. The results of this process are shown in Figure B-65.

		1	2	3	4
		Manufacturability	Safety	Forklift Exchange	Maintenance
1	Manufacturability		2	3	1
2	Safety			2	2
3	Forklift Exchange				3
4	Maintenance				
	Criteria	1	2	3	4
	Weight	17%	50%	33%	0%
	Adjusted Weight	15%	50%	30%	5%

Figure B-65: Ability to retain roll of composite weighting matrix.

The initial weighting process resulted in maintenance having a weight of zero. Since the team felt that maintenance should be represented in the selection process, the weights were adjusted to give maintenance a weight of 5%. This shift in weights is shown in Figure B-65.

With scoring criteria determined, the concepts for roll retention could be evaluated using the concept scoring matrix described earlier. For this scoring process, Concept e, No Additional Roll Constraining, was chosen as the baseline, with all other concepts being relatively evaluated. The results of the scoring process are shown in Table B-VI.

TABLE B-VI: ABILITY TO RETAIN ROLL OF COMPOSITE MATERIAL SCORING MATRIX

Selection Criteria	**Concepts**					
	Weight	**No Additional Roll Constraining**	**Shaft**	**Chain**	**Vertical Posts**	**Additional Rollers**
		e	**f**	**g**	**h**	**i**
Manufacturability	15%	3	2	2	2	2
Safety	50%	3	5	4	4	4
Forklift Exchange	30%	3	2	2	3	3
Maintenance	5%	3	2	3	3	2
Total	100%	12	11	11	12	11
Total Weighted		3.00	3.50	3.05	3.35	3.30
Rank		5	1	4	2	3
Continue?		N	Y	N	N	N

Concept f, Shaft, received the highest score of 3.50. To validate this result, a sensitivity analysis was performed. The weights of the safety and forklift exchange criteria were lowered, shifting the weight to manufacturability and maintenance. With the modified criteria weights, Concept f again received the highest score, validating the selection process.

5.3. Ability to Reduce Movement Effort

The criteria chosen to evaluate concepts for reducing the effort required to move the roll of material are outlined below.

Manufacturability – This refers to the difficulty in producing the design, taking into consideration the manufacturing methods available to the client. Amount of material required, and any manufacturing outsourcing are also considered.

Maneuverability – This refers to the flexibility of movement of the design, for example, can the design be moved in any direction.

Force to Move – This refers to the force required to initiate movement of the roll of material, as well as the force required to maintain movement.

Safety – This refers to the stability of the design, both while stationary and during motion.

Compatibility – This refers to the concept's ability to be fitted to a variety of designs without interference or impedance of the design's function.

The importance of these criterion were evaluated against each other using a weighting matrix, with the results shown in Figure B-66. The initial results of the weighting process yielded a weight of zero for manufacturability. The team adjusted the weights to give manufacturability a weight of 10%, equal to the lowest weighted criteria.

		1	2	3	4	5
		Manufacturability	Maneuverability	Force to Move	Safety	Compatability
1	Manufacturability		2	3	4	5
2	Maneuverability			3	4	2
3	Force to Move				4	3
4	Safety					4
5	Compatability					
Criteria		1	2	3	4	5
Weight		0%	20%	30%	40%	10%
Adjusted Weights		10%	17%	27%	36%	10%

Figure B-66: Reducing movement effort weighting matrix.

For the scoring process, Concept k, Four Casters, was chosen as the baseline since it is the method currently being used by the client. Each concept was then evaluated compared to Concept k, with the results shown in Table B-VII.

TABLE B-VII: REDUCING MOVEMENT EFFORT SCORING MATRIX

Selection Criteria	Concepts					
	Weight	Walking Axle Casters	Four Casters	Double Wheelbarrow	Large Center Wheel	Handle
		j	k	l	m	n
Manufacturability	10%	2	3	3	1	4
Maneuverability	17%	3	3	2	2	3
Force to Move	27%	5	3	4	5	4
Safety	36%	3	3	1	2	4
Compatibility	10%	3	3	3	2	5
Total	100%	16	15	13	12	20
Total Weighted		3.44	3.00	2.38	2.71	3.93
Rank		2	3	5	4	1
Continue?		Y	N	N	N	Y

Concept n, Handle, received the highest score of 3.93, with Concept j, Walking Axle Casters receiving the second highest score of 3.44. A sensitivity analysis was performed by reducing the weights of Force to Move and Safety, with the weight being shifted to the remaining three criteria. With these adjusted weights, Concept n received the highest score of 3.95, and Concept j received the second highest score of 3.25. These results validated the selection process. It is important to note that Concept n is not mutually exclusive with any other concept, meaning that it can be implemented in addition to any of the presented concepts. For this reason, the top two concepts noted were selected moving forward.

5.4. Ability to Reduce Coiling and Uncoiling Effort

The method of selecting the optimal method of uncoiling and recoiling the rolls of material followed a similar process to material retention and roll retention, beginning with the generation of concept evaluation criteria. The chosen criteria are outlined below.

Uncoils Material – This refers to the ability of the design to uncoil material, taking into account the effort required to do so.

Recoils Material – This refers to the ability of the design to recoil excess material, taking into account the effort required to do so.

Manufacturability – This refers to the difficulty in producing the design, taking into consideration the manufacturing methods available to the client. Amount of material required, and any manufacturing outsourcing are also considered.

Footprint – This refers to any increase in size to the overall design that would result from implementing the concept.

Roll Exchangeability – This refers to the compatibility of the concept with the process of exchanging the rolls of material with a forklift, taking into consideration any additional time, processes, or personnel required.

Maintenance – This refers to any additional maintenance required by implementing the concept.

Safety – This refers to any safety concerns introduced by the implementation of the concept.

These criteria were evaluated against each other using a weighting matrix, with the results shown in Figure B-67. The initial weighting process resulted in footprint/size receiving a weight of zero. The team felt that this criterion should be represented, thus the weights were adjusted to give footprint/size a weight of 5%, equal to the lowest weighted category.

		1	2	3	4	5	6	7
		Uncoils Material	Recoils Material	Manufacturability	Footprint	Roll Exchangability	Maintenance	Safety
1	Uncoils Material		2	1	1	1	1	7
2	Recoils Material			2	2	2	2	7
3	Manufacturability				3	5	3	7
4	Footprint					5	6	7
5	Roll Exchangability						5	7
6	Maintenance							7
7	Safety							
Criteria		1	2	3	4	5	6	7
Weights		19%	24%	10%	0%	14%	5%	29%
Adjusted Weights		18%	23%	9%	5%	13%	5%	27%

Figure B-67: Ability to uncoil composite material weighting matrix.

Using the adjusted criteria weightings, the methods of uncoiling and recoiling the rolls of material were evaluated, with the results shown in Table B-VIII. For this process, Concept q, Bus Wheel, was chosen as the baseline, and given scores of three for each criterion.

TABLE B-VIII: ABILITY TO UNCOIL COMPOSITE MATERIAL SCORING MATRIX

Selection Criteria	Weight	Concepts				
		Tensioned Roller	Bus Wheel	Powered	Synchronized Bus Wheel	Synchronized Powered
		p	q	r	s	t
Uncoils Material	18%	4	3	4	4	5
Recoils Material	23%	2	3	4	4	5
Manufacturability	9%	2	3	2	2	1
Footprint/size	5%	2	3	3	3	3
Roll Exchangeability	13%	2	3	3	3	3
Maintenance	5%	2	3	2	2	1
Safety	27%	3	3	2	3	2
Total	100%	17	21	20	21	20
Total Weighted		2.63	3.00	3.00	3.27	3.27
Rank		5	4	3	1	2
Continue?		N	N	N	Y	Y*

As can be seen, Concept s, Synchronized Bus Wheel, and Concept t, Synchronized Powered, received the highest scores with 3.27 each. A sensitivity analysis was performed by lowering the weights of the uncoils material, recoils material, and safety criteria, shifting the weight to maintenance, footprint, and manufacturability. With these adjusted weights, Concept s received the highest score with 3.15, and Concept t received the second highest score with 3.08. These results proved to validate the selection process, since the top two concepts received the highest scores for both weighting schemes. Both Concept s and t were selected moving forward.

6. Concept Selection and Integration

Following the concept scoring in each of the four design categories, numerous concepts were selected and combined for the final design. In this section, a detailed explanation of the reasons behind the selection of design concepts, as well as a description of the design aspects, will be discussed.

To control the expansion of the composite roll, Concept c, Side Roller Arm, was chosen. This concept uses a swinging arm, located at the side of the composite roll, to apply a force to the top of the composite roll via a pivoting roller located at the end of the arm. Figure B-68 shows a preliminary render of this concept. This design was selected due to its ability to retain the material in a roll, ease of manufacturing, and mechanism safety compared to other designs. Another reason this design was selected was because it fits within the footprint of the design space. Currently, Triple E RV does not have a mechanism in place to control the roll expansion. Implementation of this concept will aid in maintaining the material in a tightly wrapped roll, as well as helping to improve the efficiency of the coiling and uncoiling process.

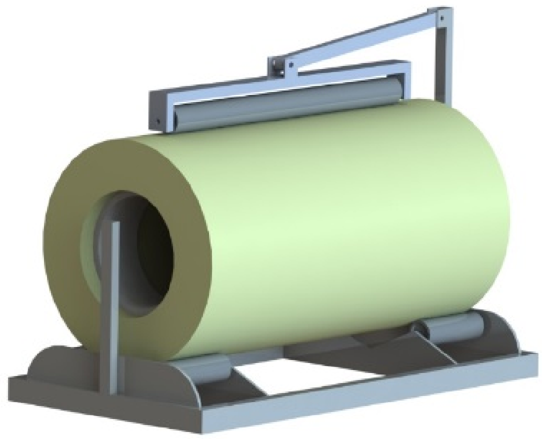


Figure B-68: Selected method of constraining material expansion [11].

To positively retain the composite roll, Concept f was chosen. Concept f consists of removable securing brackets on either end of the composite roll. These brackets feature two short shafts, spaced such that they form a tight fit with the inner diameter of the roll, as shown in Figure B-69. The two shafts retain the roll of material within the frame, and are allowed to slide along a vertical shaft as the height of the center of the roll changes.

Many designs for retaining the roll of material during use were evaluated. Out of all the designs examined, a two-shaft bracket was selected due to its high level of safety. This design is also relatively easy to manufacture, easy to maintain, and simple to remove and install during roll exchanges. Compared to the current design which includes a shaft going through the entire roll with wheel supports, this design is relatively simple to manufacture, maintain and has an improved level of safety.

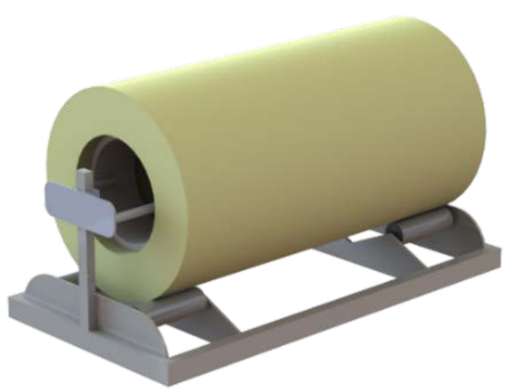


Figure B-69: Selected method of retaining roll of composite material [11].

The biggest complication in the current design is the manual labor required to coil and uncoil the thick sheet of composite material. To reduce the coiling and uncoiling effort, the team chose two concepts to further develop. These concepts are Concept s, Synchronized Bus Wheel, and Concept t, Synchronized Powered. As can be seen in Figure B-70, both alternatives can be used to rotate the rollers at the bottom of the cart, providing a method of uncoiling and recoiling the composite material. Compared to other designs that were evaluated in the same category, these designs were selected due to their ability to both recoil and uncoil the material effectively.

Concept s, Synchronized Bus wheel, will be further developed as the standard method of coiling and uncoiling the material. Concept t, Synchronized Powered, will also be further developed as a feature which can be easily added to the final design, should the client choose to do so.

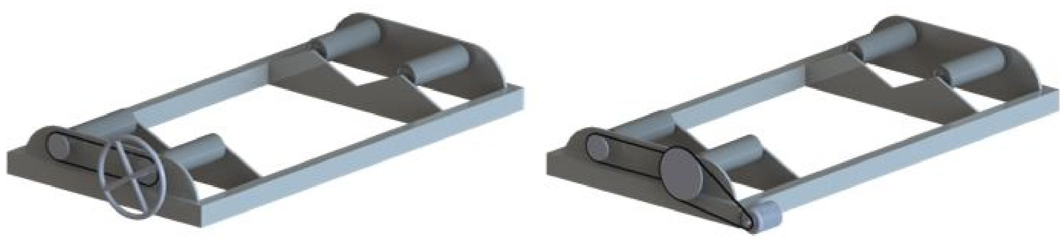


Figure B-70: Selected methods of coiling and uncoiling the composite material [11].

Another complication with the current design is the amount of effort required to move the cart supporting the composite roll. Several designs were evaluated to improve the maneuverability of the design. Out of all the designs evaluated for this category. Concept j, Walking Axle Casters, and Concept n, Handle, were chosen to move forward with.

Concept j was selected due to its reduction of bearing friction and rolling friction while maintaining stability of the frame. Concept n was chosen because it adds a convenient and ergonomic location to apply force to the frame while maintaining a low cost.

Shown in Figure B-71, the selected design for this project consists of a cart with four rollers on which the roll of composite material rests. These rollers can be rotated using a large wheel which is attached to the shaft of one roller, with a synchronizing chain turning a second roller. The weight of the material, as well as force provided by an additional overhead roller, prevent the material from expanding. The roll of composite material has retaining shafts inserted into its center from either end, which secures the roll in place. The frame assembly has integrated handles which provide a safe and stable location from which to push the cart.

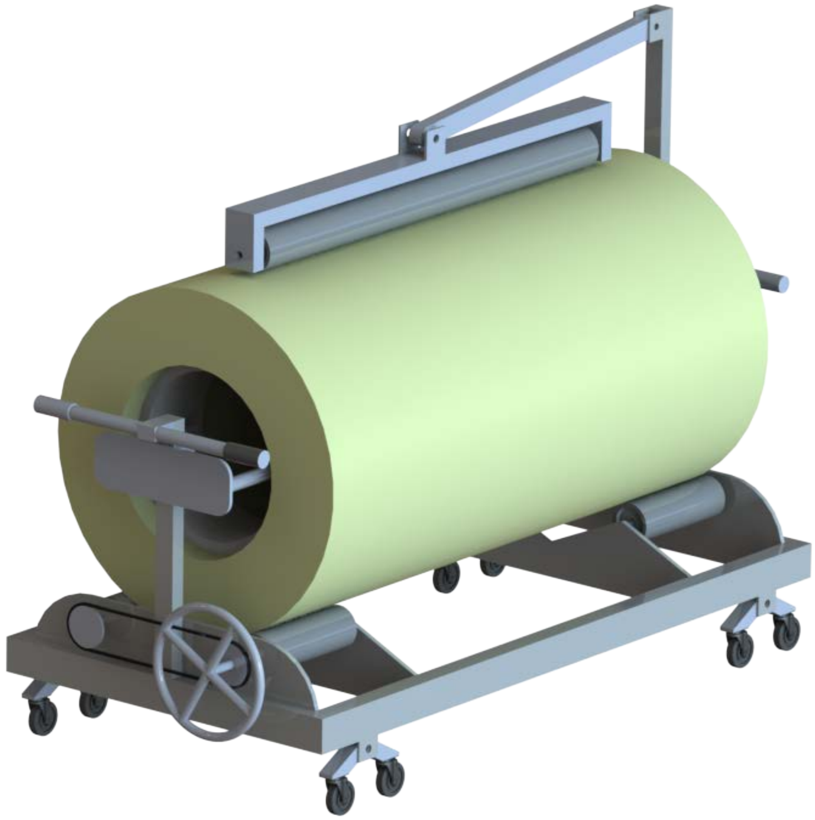
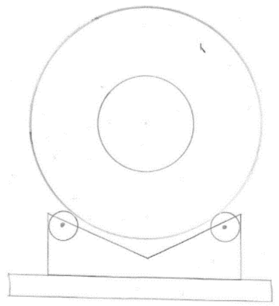
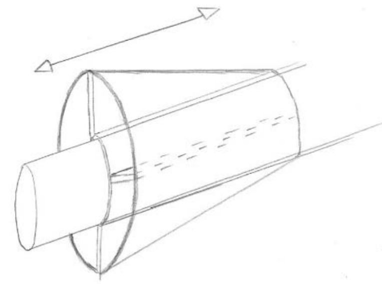


Figure B-71: Integration of selected concepts into preliminary final design [11].

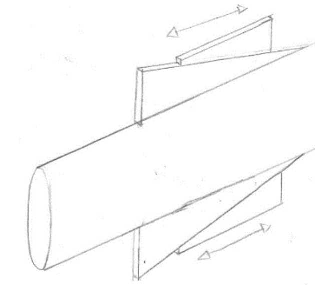
7. Concept Generation Master List

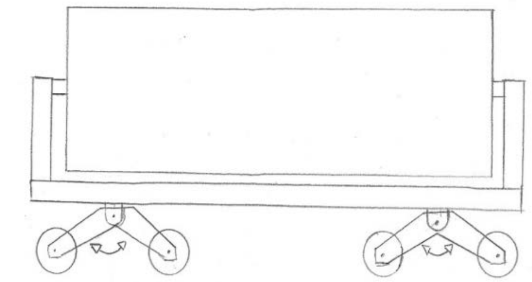
All concepts generated by the team are presented below. This document is primarily to aid the reader by being quickly to view a concept and determine which page contains that concepts description throughout the report.

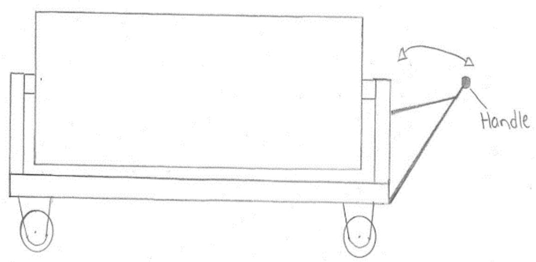

Concept #1, Page: B-8


Concept #2, Page: B-8

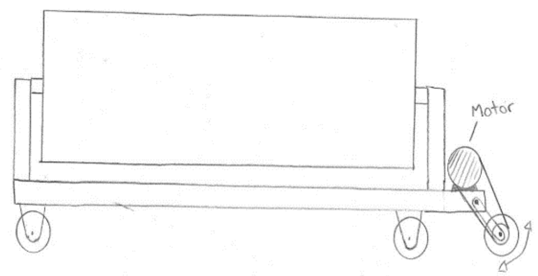
Concept #3, Page: B-9


Concept #4, Page: B-10

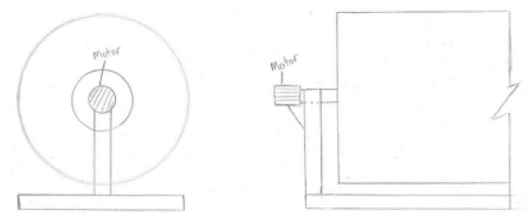

Concept #5, Page: B-10



Concept #6, Page: B-11



Concept #7, Page: B-11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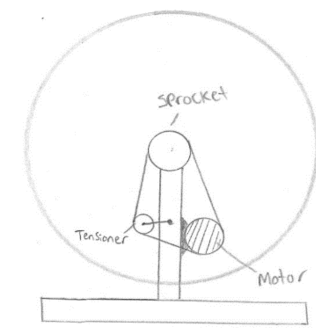

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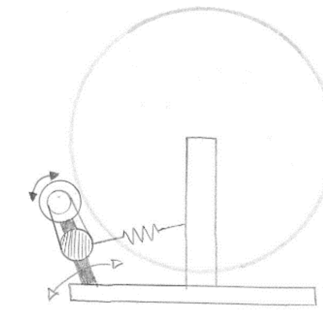
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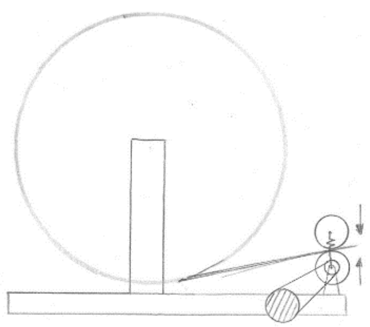
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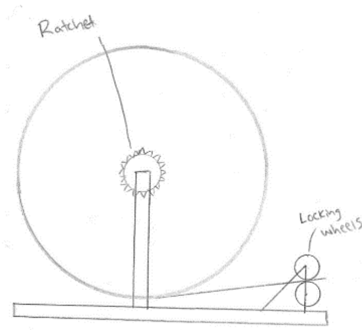

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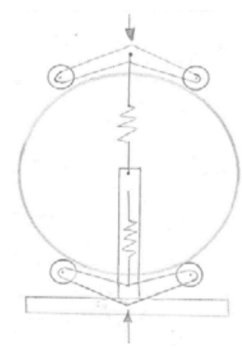
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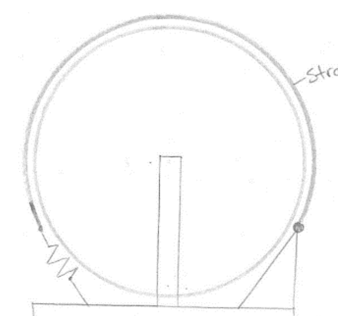

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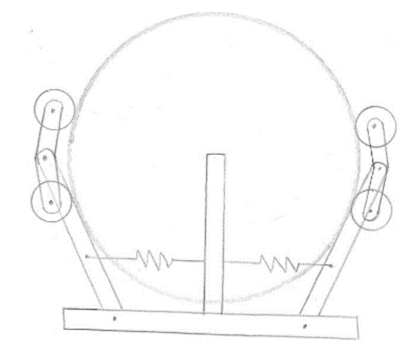
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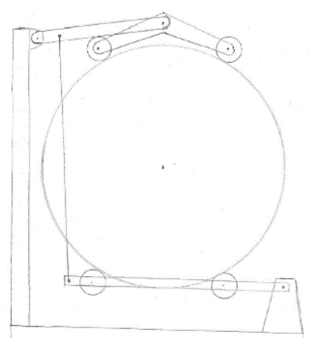
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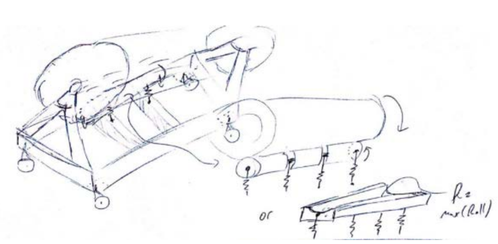
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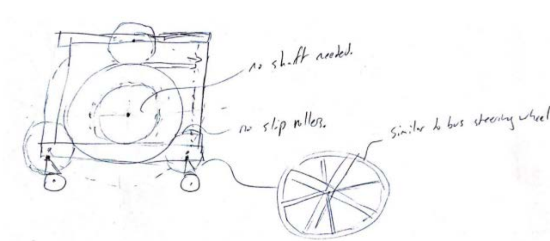
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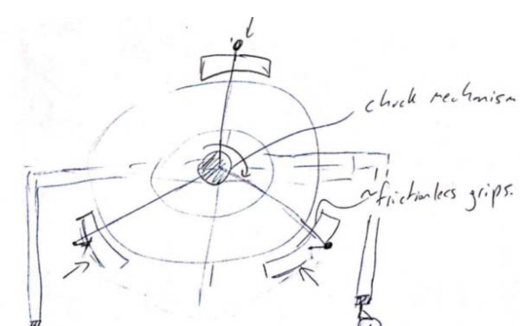
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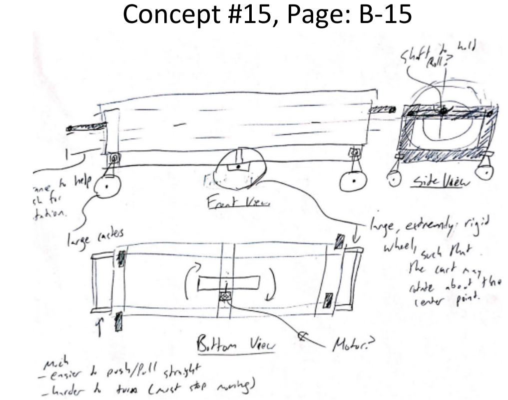
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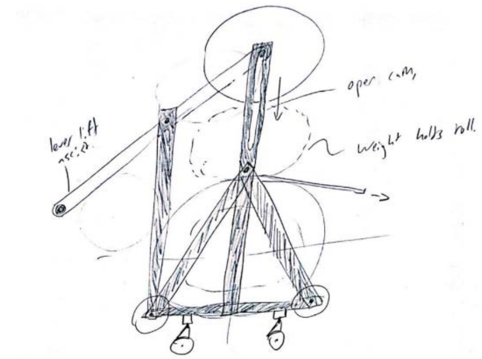

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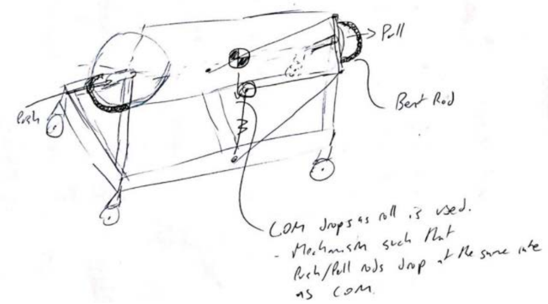

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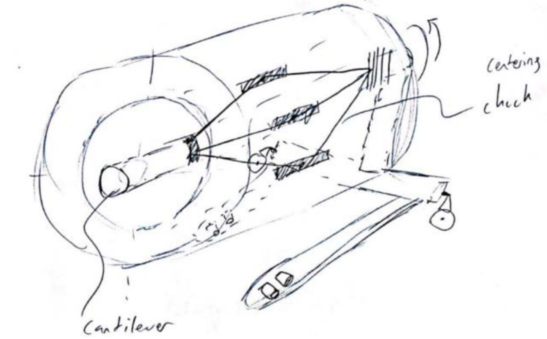

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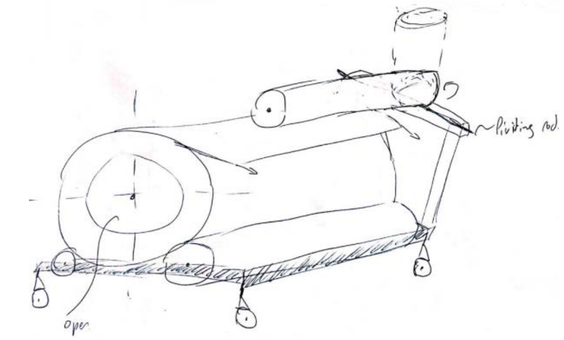

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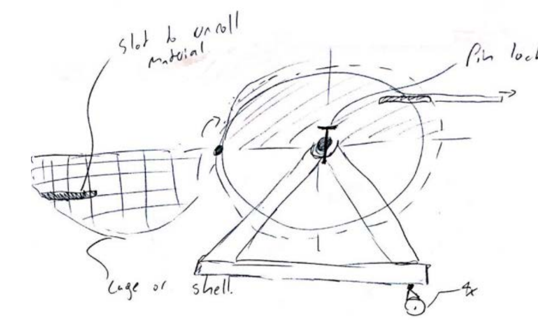

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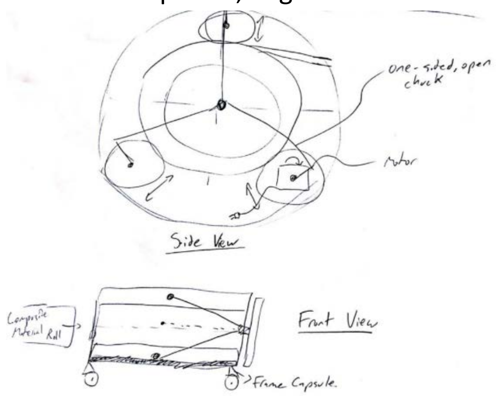

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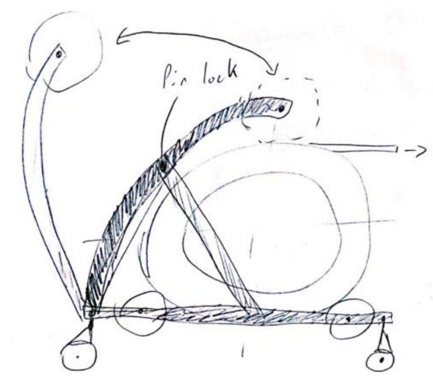

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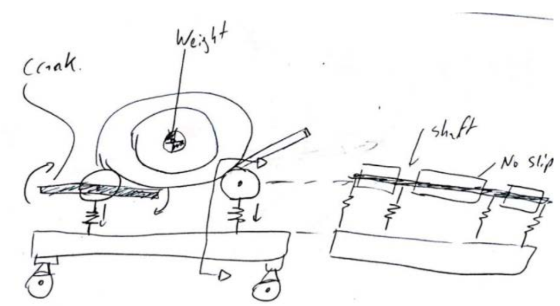

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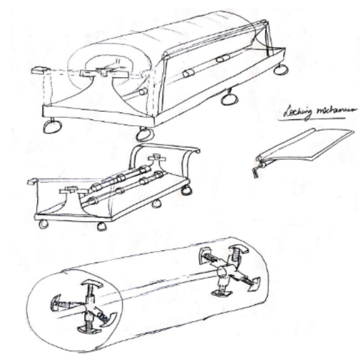

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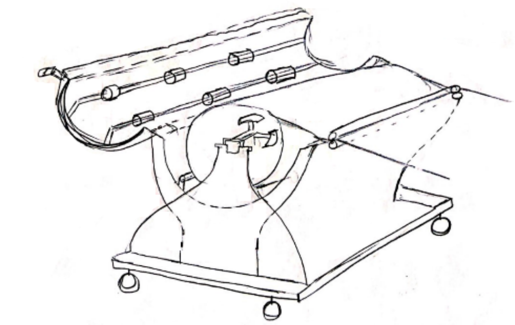

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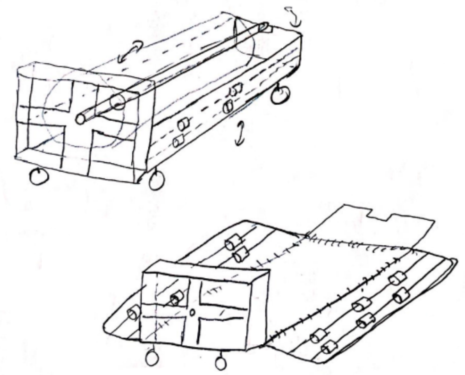

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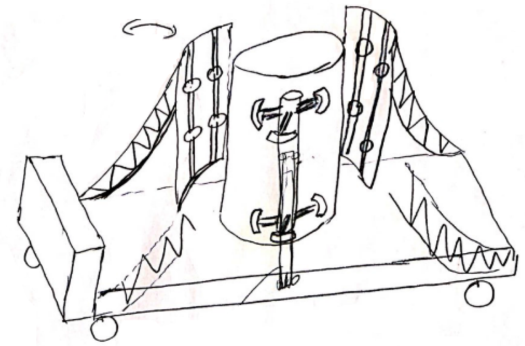

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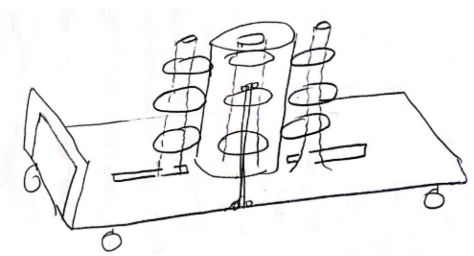

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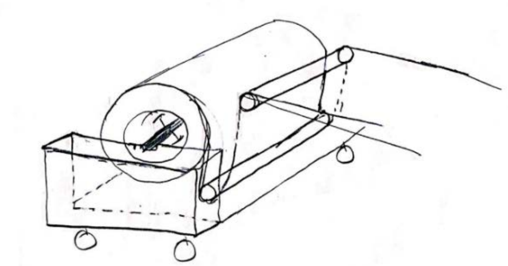

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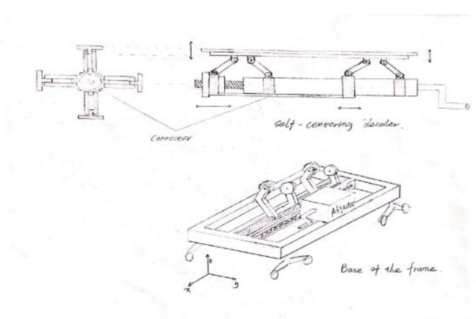

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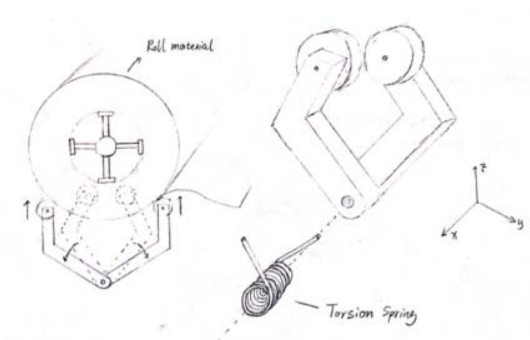

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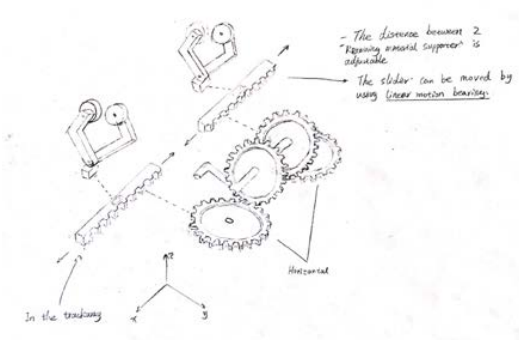

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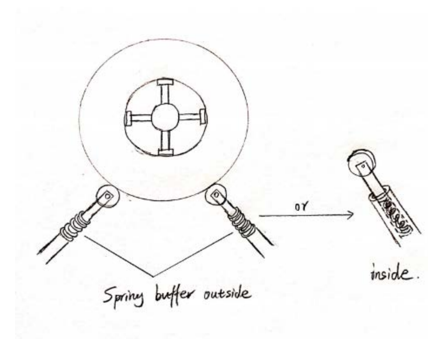

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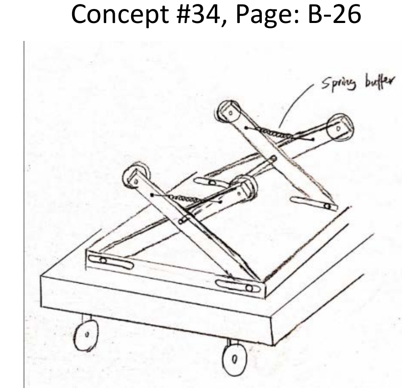

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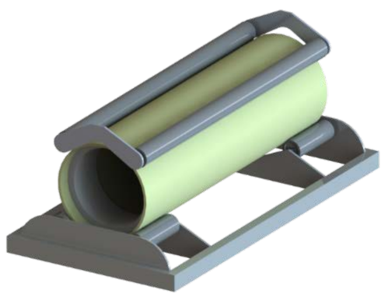

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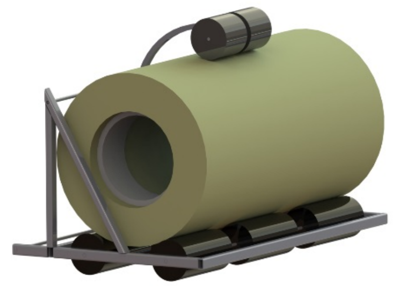

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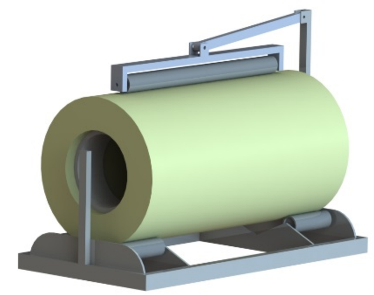

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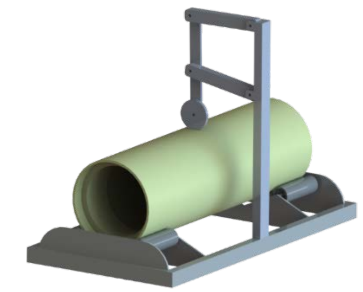

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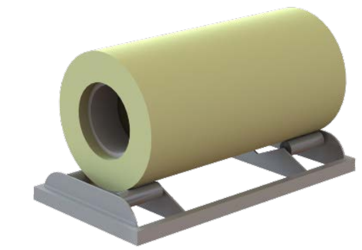

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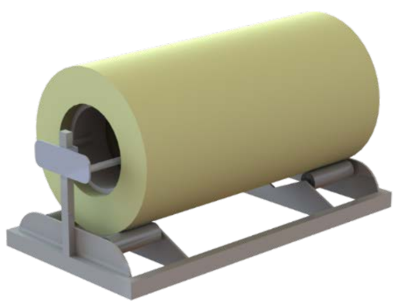

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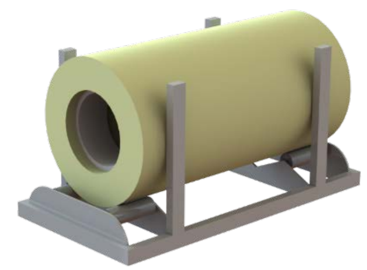

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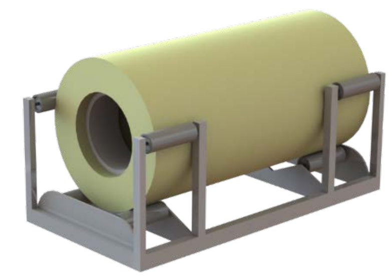

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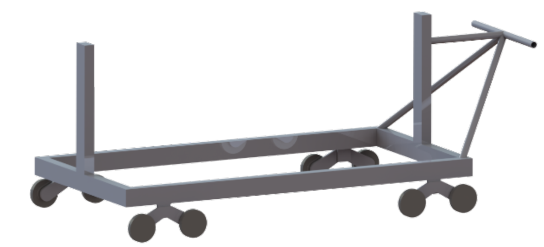

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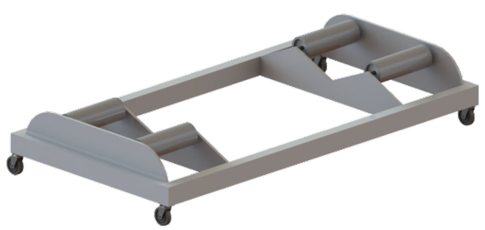

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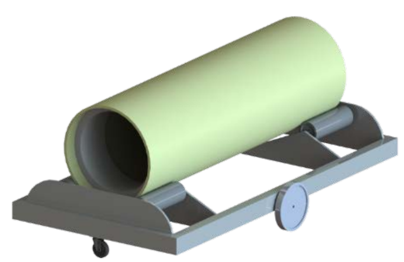

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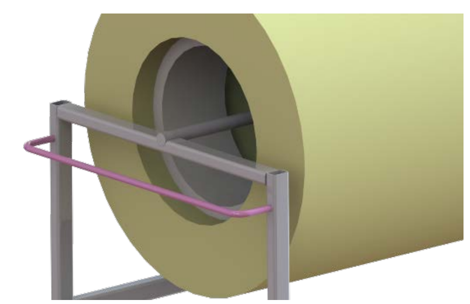

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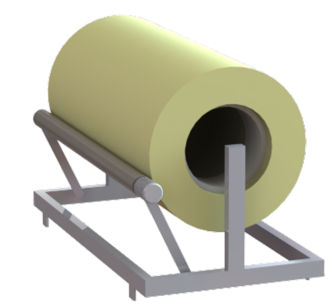

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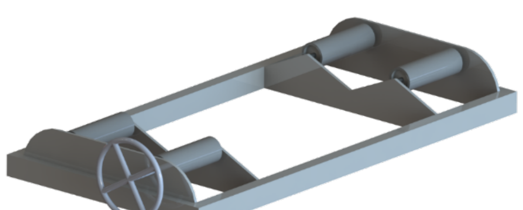

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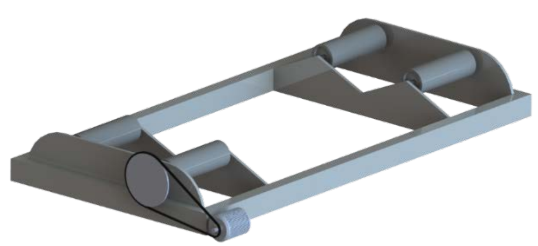

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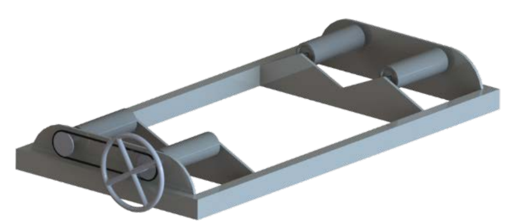

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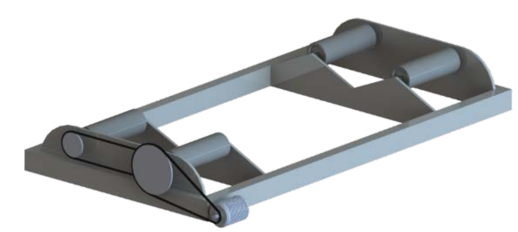

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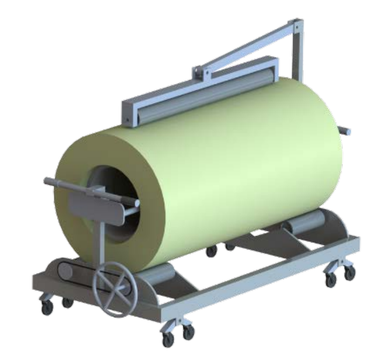

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Appendix C: Composite Roll Expansion Force

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1. Roll Expansion Constraining Force

After finalizing the conceptual design selection, the first and foremost problem the team decided to tackle was force required to counteract the expansion of the composite material roll. To determine an approximation of the force required, the team followed the following approach: First, the stiffness or modulus of elasticity of the composite material was determined the by a series of tensile tests. Second, by using the stiffness determined, the moment, and subsequently force that would be required to bend the composite material such that its radius of curvature matches that of the ideal radius of the roll was calculated.

1.1. Stiffness of Cosmolite

Initially, the team attempted to research the properties of the Cosmolite composite material. However, with little luck the team was forced to test the material internally. To test the material, the team received a 12" x 10" sample from Triple E RV. The sample was cut into 10 sample strips of approximately 12" x 1" each. Each sample was numbered and measured for initial dimensions: length, base, and thickness. Three measurements per dimension, per sample were taken with an electric caliper, and the average was recorded and shown in Table C-I. Next, using the resources at the University, the team acquired access to a MTS tensile testing machine for a day.

Table C-I: Raw Tensile Test Specimen Data

Specimen Number	Full Length (mm)	Width (mm)	Thickness (mm)	Testing Strain Rate (mm/min)
0	304.8	26.23	3.50	10
1	304.8	24.65	3.59	5
2	304.8	26.52	3.63	2.5
3	304.8	25.59	3.52	10
4	304.8	25.34	3.47	10
5	304.8	26.43	3.48	10
6	304.8	26.92	3.74	10
7	304.8	25.45	3.45	10
8	304.8	25.04	3.38	10
9	304.8	26.5	3.44	10

Using the MTS tensile testing machine, an initial length between clamps for all specimens was set at 205 mm as a strain reference point. Each specimen was tested under a uniform strain rate of 10 mm/min, with the exception of specimens 1 and 2, which were tested at 5 and 2.5 mm/min, respectively. Specimens 1 and 2 were tested at a different rates to see if the effect of strain rate changed the overall stiffness of the material. However, no noticeable different was made between the faster and slower testing of the strain rates, thus the team moved back to a faster rate of 10 mm/min for the remaining test.

After testing, for each specimen a separate raw data text file was extracted. This text data file contained numerous rows of recorded data, at one row per 0.1 seconds of testing. In each row data recorded contained the value of force measured by a load cell, time in second, and specimen extension in mm. For each test, this data was sorted via an excel spreadsheet, such that graphically a modulus of elasticity could be determined. The following calculations were performed on each line of data to achieve proper measures of stress and strain.

$$Stress = \frac{Force}{Area} = \frac{Load\ Cell\ Force\ Measured}{(Specimen\ Thickness)(Specimen\ Width)} \tag{C-1}$$

$$Strain = \frac{\Delta Length}{Length} = \frac{Extension\ of\ Specimen}{Initial\ Reference\ Length\ of\ Specimen} \tag{C-2}$$

Using these simple relationships, stress and strain was calculated for all rows of data for each specimen. Using the calculated data, the results were plotted. By introducing a linear line of best fit, and displaying the equation of said line, the slope of the line or modulus of elasticity could be determined. An example stress-strain curve for specimen 0 is shown in Figure C-1. As seen on the plot, the slope of the best fit line is 3471.8 MPa, and thus the modulus of elasticity of specimen 0 was determined to be 3.47 GPa.

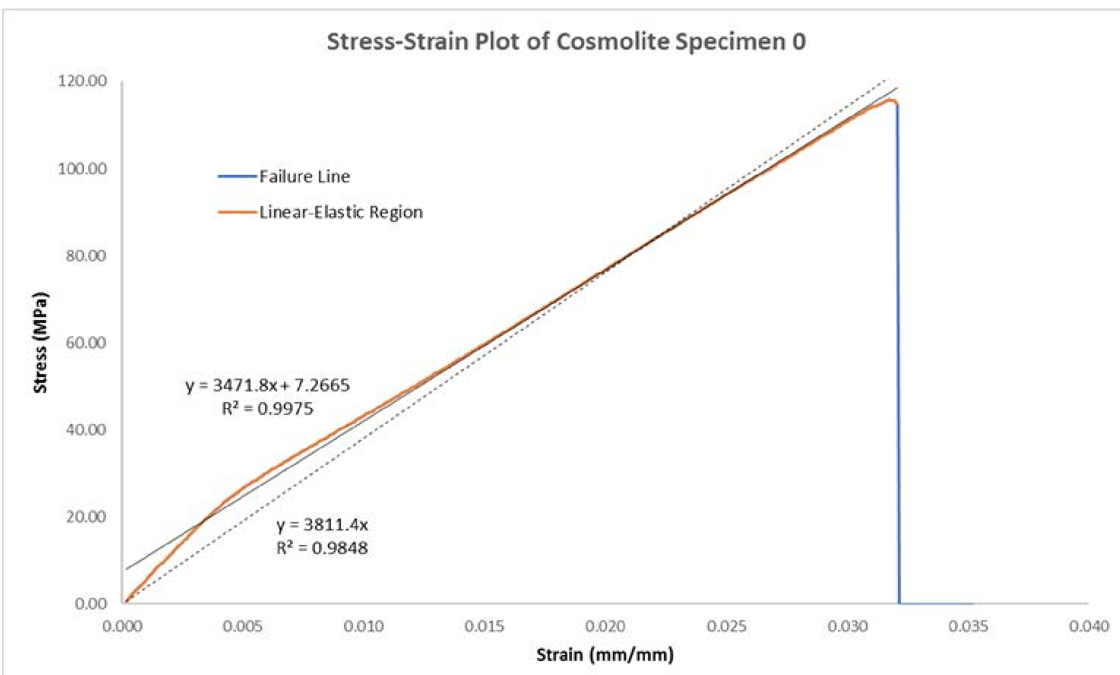

Figure C-1: Stress-Strain Curve for Cosmolite Specimen 0

The same process was repeated for the other nine specimens, and the results of their modulus of elasticities are shown in Table C-II. As seen in the Table, the average modulus of elasticity for Cosmolite was determined to by 3.33 GPa, with a standard deviation of 0.38 GPa.

TABLE C-II: MODULUS OF ELASTICITY RESULTS FOR COSMOLITE SPECIMENS

Specimen and Test ID	Modulus of Elasticity, E results (GPa)
0	3.47
1	3.55
2	3.81
3	3.71
4	3.37
5	2.87
6	2.73
7	2.87
8	3.34
9	3.56
Average	**3.33**
St. Dev.	**0.38**

1.2. Pure Bending of Composite Roll

For any beam in pure bending, the radius of curvature, ρ can be calculated as shown in Eq. (C-1).

$$\frac{1}{\rho} = \frac{M}{EI} \tag{C-1}$$

Where *M* is the moment about the beam, *E* is the modulus of elasticity, and *I* is the moment of inertia of the beam. To determine the amount of force to keep a layer of composite material from expanding off the roll, the moment must be solved for. After the moment required is determined, using the spacing between the Top Rollers, *d* the force required to hold the roll in position, *F* can be determined using Eq. (C-2).

$$M = Fd \rightarrow F = \frac{M}{d} \tag{C-2}$$

Looking at the composite material roll, the radius of curvature ranges from the radius of a full roll to the radius of a nearly empty roll. Given a full roll of composite material is 1.24 m in diameter, and an empty roll is 0.70 m in diameter, the maximum and minimum values of radius curvature are equal to 0.62 m and 0.35 m, respectively. The moment of inertia of the cross-section of the roll can be determined by Eq. (C-3).

$$I = \frac{1}{12} bh^3 \tag{C-3}$$

Where *b* is the full length of the composite material equal to 2.4 m, and *h* is the thickness of the roll equal to 3.5 mm. Thus, the moment of inertia is:

$$I = \frac{1}{12}(0.24)(3.5e^{-3}) = 8.932\ x\ 10^{-9}\ m^4$$

Using the average modulus of elasticity previously determined of 3.33 GPa, all values can be substituted back into Eq. (C-2) to solve for moment.

$$M_{max} = \frac{EI}{\rho} = \frac{(3.33e^{9})(8.932e^{-9})}{0.35} = 84\ Nm$$

$$M_{min} = \frac{EI}{\rho} = \frac{(3.33e^{9})(8.932e^{-9})}{0.62} = 47.5\ Nm$$

Next, given the distance between Top Rollers is 0.589 m, these values can be substituted into Eq. (C-3) to solve for the minimum and maximum forces required to hold the composite material tight.:

$$F_{max} = \frac{M_{max}}{d} = \frac{84}{0.589} = 142.6\ N = \mathbf{32\ lbs/layer}$$

$$F_{min} = \frac{M_{min}}{d} = \frac{47.5}{0.589} = 80.6\ N = \mathbf{18.1\ lbs/layer}$$

In theory, if the composite roll is held tight, only the top layer of the roll will be able to expand at any given time. Given the design of the top rollers naturally sits at 285 lbs, the team determined this weight alone would be more than enough to keep the composite material rolled as tight as possible. If the composite roll were mismanaged and the roll was accidently let to expand more than the first layer of material, in theory the weight of the Top Roller assembly should be enough to push back approximately eight or more layers of the composite material at a time. Thus, if the expanded composite material is recoiled with the weight applied, it should reasonably tighten back to its ideal, minimized diameter possible.

Appendix D: Final Design Safety and Kinematics

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1. Safety Standards

One requirement the client laid out for the team at the beginning of the project was to ensure the final proposed design was safe to be used by their employees. After some researching, the team was able to find a highly regarded statistical based safety standards regarding maximum pushing and pulling forces for men and women, published in 1983 by Ciriello and Snook (also known as the Liberty Mutual Tables) [1]. From the extensive tables published, the team was able to extract the relevant data for maximum initial and sustained pushing and pulling forces, which apply to the project. The data extracted include the maximum allowable initial pushing and pulling forces, as well as sustained pushing and pulling force for single-trip, 25 ft travel distance per day allowable for 75% if the population of men and women. The data is summarized in Table D-1.

TABLE D-1: MAXIMUM ACCEPTABLE FORCE PUSHING AND PULLING [1]

Maximum Acceptable Forces of Push (Acceptable for 75% workers)		
Females		
Height (in) (Ground to Hands)	**Initial Force (lbs)**	**Sustained Force (lbs)**
53	58	32
35	58	36
22	46	32
Males		
Height (in) (Ground to Hands)	**Initial Force (lbs)**	**Sustained Force (lbs)**
57	88	55
37	91	50
25	85	80
Maximum Acceptable Forces of Pull (Acceptable for 75% workers)		
Females		
Height (in) (Ground to Hands)	**Initial Force (lbs)**	**Sustained Force (lbs)**
53	55	36
35	58	36
22	61	32
Males		
Height (in) (Ground to Hands)	**Initial Force (lbs)**	**Sustained Force (lbs)**
57	61	40
37	85	50
25	94	55

Within the Liberty Mutual Tables, the suggested goal is to design processes such that they are safe for 75% of the female population [1]. However, due to the large amount of weight required to be pushed in this project, (4300 lbs roll of composite material plus the weight of the cart itself) it is unlikely that the force suggested for 75% of the female popular will be enough to handle the design. Thus, the team decided to take the less conservative, but more realistic approach and design towards safe forces for 75% of the population of men. Taking a reasonable average throughout the forces rated for men, the team determined the general appropriate acceptable force per person to be 75 lbs. Additionally, the team performed a series of tests where each member pushed on an industrial scale at a moderate force and hard as possible. The team found a pushing force of approximately 65 to 75 lbs felt reasonably safe for working, and a maximum of pushing force of approximately 125 lbs was the upper limit achieved. Using these forces, the team was able to evaluate the safety of the design based on kinematics.

2. Kinematics of Moving the Design

Moving forward, the team decided to estimate the amount of force necessarily to move the cart assuming the final weight of the roll and cart combined was approximately 5500 lbs (2500 kg). By observing the video the team took of the composite material handling process, the cart can be seen to move approximately 10 ft (3.048 m) over 17 seconds. Thus, the average speed, *v*, of the cart is estimated as 0.1793 m/s. Neglecting friction, the force required to accelerate the cart to average speed, given a certain time of applied force, can be calculated as:

$$F = ma = m\frac{v}{t} \quad (1)$$

Where *m* is the mass of the cart, *F* is the applied force, *a* is the acceleration of the cart, *v* is the desired average velocity, and *t* is the duration of applied force.

Due to bearing friction and rolling friction being very difficult to accurately calculate, these frictions were accounted for by applying a factor of three to the theoretical value of force required to accelerate the cart. Using this, the following modifications are made to Eq. (1):

$$F = 3 * ma = 3 * m\frac{v}{t} \quad (2)$$

By using the determined safe working force of 150 lbs, or 668.9 N (75 lbs combined for 2 workers moving the cart), Eq. (2) can be solved for the time required to achieve the desired velocity.

$$t = 3 * m\frac{v}{F} = 3 * 2500\frac{0.1793}{668.9} = 2.01\ seconds$$

However, the team reasoned it is unlikely that individuals will control their own force exerted on the cart to a safe level. In fact, due to the weight of the cart, it is more likely that they would push and pull as hard as they physically can to get the cart to start moving. Thus, in the actual design of the cart, to reduce the required force as much as possible the team considered numerous ideas, such as determining the size, material (hardness), and number of swivel casters fully contacting the floor required. As the size (diameter) of the swivel casters used increases, the related rolling friction decreases. Thus, the largest diameter wheels that fit within the process's floor tracks were selected. Additionally, due to a wheel's tendency to "flatten" overtime due to the large weight supported, the initial forces to move the cart may significantly increase. The team decided a very hard steel material would be best suited to resist this phenomenon. However, this comes with additional challenges, due to the inability of steel wheels from being able to roll over any sort obstruction. Thus, the team asks that the client adds additional precaution to keep the ground around the cart free of debris, such that strain on the employees is minimized. Lastly, by increasing the number of wheels in contact with the ground, each wheel's bearings support less load and thus reduce frictional forces. To ensure that all eight wheels consistently make good contact with the ground, the developed Walking Axle swivel caster assembly is used.

Another safety concern is addressed by the design with the addition of Handle Bars. For applications which require both pulling and pushing forces, the ideal location of applied the load is approximately the elbow level, or around 40" from the ground, of the average man [1]. Thus, the team located the handle bars at 50" from the ground, which was as close as possible to this height without interfering with the other critical components of the design. Lastly, the Handle Bars were sized 22" in length, such that the spacing between grips is approximately equal to the shoulder distance of the average man. This allows for the easiest maneuverability of the cart, reducing risks of employee injury during pulling, pushing, and rotation operations.

3. Kinematics of Rotating the Composite Material

Two separate analyses were performed on the uncoiling and recoiling processes; one on the manual process using the Bus Wheel design, and one using the electric drive system. These analyses are outlined in this section.

3.1. Bus Wheel Kinematic Analysis

In order to determine the rotational speed of the composite material that can be attained using the manual Bus Wheel, a kinematic analysis was performed. From the Liberty Mutual tables described earlier, it was found that a safe sustained pulling force was approximately 75 lbs [1]. Given that two Bus Wheels are present on the design, and can be used simultaneously, an equivalent force of 150 lbs can be applied at the radius of the Bus Wheel

(17” from the location of rotation). Using these values, as well as the dimensions of the bottom rollers and composite roll, the torque applied to the roll of material can be calculated as follows.

$$T = F * r_b * \frac{D_r}{D_b} = 668.9 * 0.4318 * \frac{1.24}{0.168} = 2131.8\,Nm \quad (3)$$

Where *F* is the applied force in Newtons, r_b is the radius of the Bus Wheel in meters, D_r is the diameter of the composite roll in meters, and D_b is the diameter of the Bottom Roller in meters. Given that the effective polar mass moment of inertia, J_0, of a full roll of material after accounting for friction and material deformation is 1459.7 kg-m^2 (calculated in the Drive Systems section in the main body of the report), the acceleration of the roll can be calculated as follows:

$$\alpha = \frac{T}{J_0} = \frac{2131.8}{1459.7} = 1.46\,rad/s^2 \quad (4)$$

If it were possible to attain the desired roll speed of 20 rpm (2.09 rad/s), full speed could be attained in 2.09/1.46=1.43 seconds. However, due to the speed ratio between the composite roll and the Bottom Roller (and thus the Bus Wheel), achieving a composite roll speed of 20 rpm is not realistic. The team decided that a rotational speed of 30 rpm (3.14 rad/s) at the Bus Wheel was realistically attainable, resulting in a composite roll uncoiling speed of 4.06 rpm (0.425 rad/s). Using this, the maximum uncoiling speed can be attained in 0.425/1.46 = 0.291 seconds.

3.2. Electric Drive System Analysis

Following a similar procedure as was used in the previous section, an analysis of the electric drive system was performed to determine the performance capabilities of the system. The motor selected for the drive system is rated at 3 hp output at 1740 rpm. Given this information, the motor output torque can be calculated as:

$$T_m = \frac{P_m}{\omega_m} = \frac{2238}{183.26} = 12.21\,Nm \quad (5)$$

Where P_m is the motor power in watts, and ω_m is the motor rated speed in rad/s. The torque applied to the composite roll can then be calculated as:

$$T_r = T_m * r_g * \frac{D_r}{D_b} = 12.21 * 15 * \frac{48}{6.625} = 1327.21\,Nm \quad (6)$$

Where r_g is the gearbox reduction ratio, D_r is the diameter of a full roll of material in inches, and D_b is the diameter of the Bottom Roller in inches. Using the same multipliers, the rotational speed of the roll of material is calculated as:

$$\omega_r = \omega_m * \frac{1}{r_g} * \frac{D_b}{D_r} = 183.26 * \frac{1}{15} * \frac{6.625}{48} = 1.686\, rad/s \quad (7)$$

Given that the effective polar mass moment of inertia of a full roll of material is 1459.7 kg-m^2, the acceleration of the roll can be calculated as:

$$\alpha = \frac{T_r}{J_0} = \frac{1327.21}{1459.7} = 0.909\, rad/s^2 \quad (8)$$

Finally, the time required to achieve the full uncoiling speed can be calculated as:

$$t = \frac{\omega_r}{\alpha} = \frac{1.686}{0.909} = 1.855\, seconds \quad (9)$$

A number of assumptions were made in the calculations performed in this section. Firstly, all calculations were performed for a full roll of composite material. As the roll empties, its polar mass moment of inertia decreases and the performance of the electric drive system is expected to increase. Secondly, it was assumed that the electric motor's torque output remains constant across all operating speeds. This assumption was made necessary by the lack of a torque-curve available for the chosen motor. In reality, it is unlikely that the torque output of the motor remains constant during the acceleration of the roll of material. As such, the values presented in these calculations will likely have some inaccuracies that should be further tested once the motor is purchased.

4. References

[1] Ciriello. et al. (1983). *Liberty Mutual Manual Materials Handling Tables* [Online]. Available: https://libertymmhtables.libertymutual.com/CM_LMTablesWeb/taskSelection.do (December 2, 2017).

Appendix E: Stress Analysis

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1. Stress Analysis of Design

To verify the safety and strength of the design, a rigorous stress analysis was performed on all structural and load bearing parts. This section outlines the process taken to analyze the stresses in each component by analytical and finite element analysis (FEA), stresses in welds, and bearing stresses applied between fasteners.

1.1. Stress Analysis of Components

The primary method of evaluating stresses in the design was using finite element analysis (FEA). For all analyses, the Simulation package in SolidWorks was used. SolidWorks Simulation's default element, ten-node tetrahedral, was used for all analyses shown in this report. This element is considered a "general purpose element" as it provides high accuracy under a variety of loading scenarios and part geometries.

Wherever possible, the results of the FEA were validated with analytical calculations. If the two types of analyses provided similar results of stresses, the results were taken to be sufficiently accurate. Where complex part geometries or loading scenarios prohibited analytical calculations, convergence studies were used to show the validity of the finite element analyses. Convergence studies monitor the results of certain critical values of the analysis, such as maximum stress or deflection, over many consecutive mesh refinements. If the monitored value converges to a steady value, the analysis has converged, and the results can be taken as accurate.

In all cases, mesh refinement was performed via h-adaptive studies in SolidWorks. This method iteratively refines the mesh in areas of high stress gradients. H-adaptive studies use a strain energy convergence criterion. The energy of deformation is compared between consecutive mesh refinements, with the percent difference taken as the strain-energy error. Convergence is attained when the strain-energy error satisfies the target accuracy. For example, if the target accuracy is set to 98%, convergence is attainted when the difference in strain-energy between two consecutive studies is less than 2%. If this criterion is not met, the mesh is further refined, and the study runs again.

Four types of material were used in the analysis of components; ASTM A-500 Grade B was used for all structural tubing components, ASTM A36 was used for all components made from steel plating or sheet metal, AISI 1117 was used for Bottom Roller steel shafts, and AISI 1566 was used for Top Roller steel shafts. The properties of these materials are summarized in Table E-I.

TABLE E-I: MATERIAL PROPERTIES FOR MATERIALS USED IN STRESS ANALYSIS STUDIES [1]

Material	Use	Yield Strength (ksi)	Ultimate Strength (ksi)
ASTM A-500 Gr. B	Structural Tubing	46	58
ASTM A-36	Plate and Sheet Steel	36.3	58
AISI 1117	Bottom Steel Shafts	58	68.9
AISI 1566	Top Steel Shafts	75	-

All stress plots shown in this section indicate von Mises stresses, with units of ksi (10^3 psi). The indicated stress is typically compared to the material's yield strength (except in the case of the retaining shafts) to determine a factor of safety.

It is important to note that the analyses presented in this report are preliminary, and should not be considered an exact representation of the actual design. Further analysis is required to verify the results shown in this report.

1.1.1. Hanging Top Rollers

A single side of the top rollers subassembly analyzed to primarily ensure the chosen shaft diameter of assembly was appropriate. A solid model of the multi-part assembly was created. Fillets of 0.3125" radius were placed in all locations where the parts were to be welded to help simulate the stress concentrations in those areas. On the shaft, in the location that the shaft was to run through the specified ball bearings, the faces were split. Bearing supports with constricted rolling were placed on these inside diameter faces to simulate the self-alignment of shaft between the bearings. Gravity was the only applied force as the shaft need only support the weight of the assembly. Figure E-1 displays the setup of the static study.

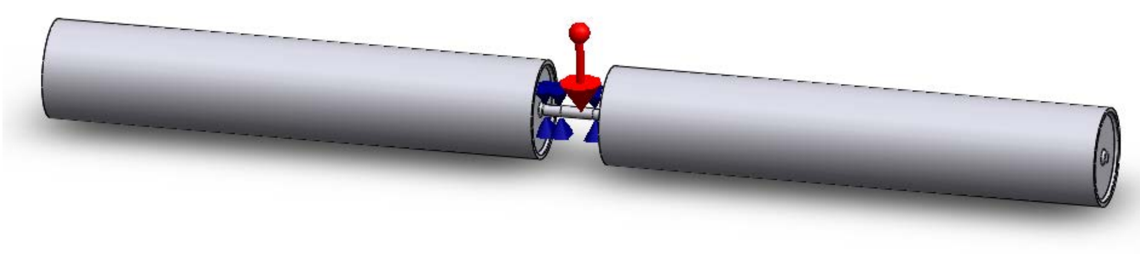

Figure E-1: Hanging Top Rollers FEA boundary conditions [2].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 82746 total elements, with a minimum element size of 0.299". As can be seen in Figure E-2 and Figure E-3, the static study converged to 7.4% total relative strain energy with the majority of elements being located around the bearing support boundaries and weld simulating fillets.

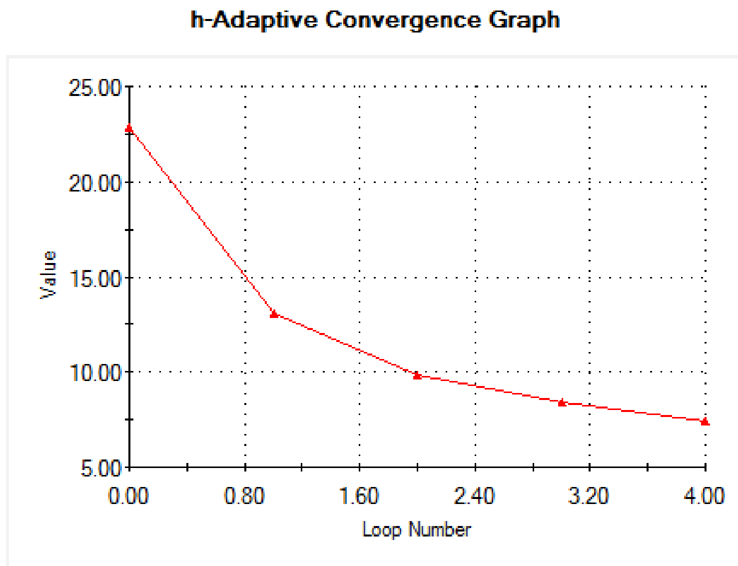

Figure E-2: Hanging Top Rollers h-adaptive convergence plot [2].

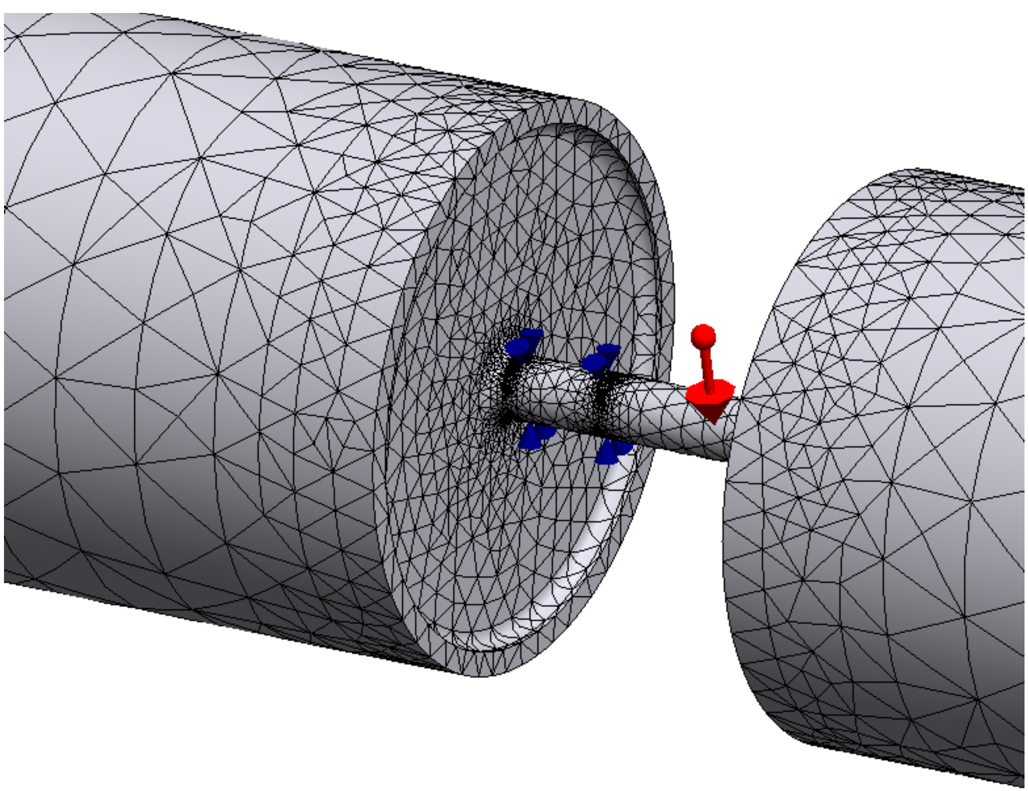

Figure E-3: Hanging Top Rollers meshing [2].

As seen in Figure E-4, the stress throughout the assembly is nominally close to zero aside from the areas between both rollers on the shaft, as expected. Within this area, a maximum stress of 131.89 ksi occurs. Looking closer in this area with Figure E-5, this stress is highly localized near the boundary of the bearing support condition. Thus, this stress is a diverging concentration and should be ignored.

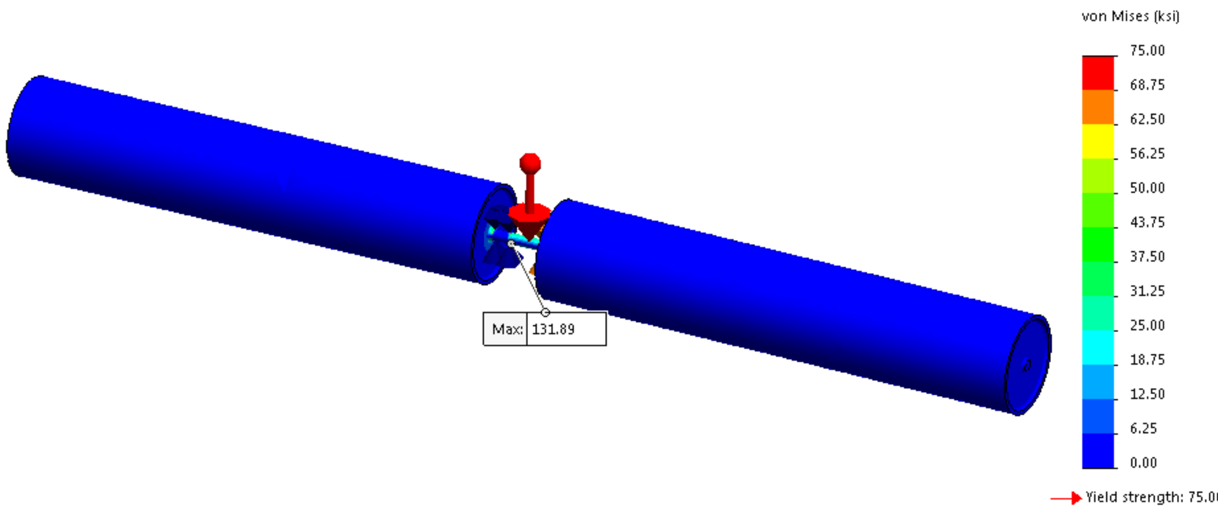

Figure E-4: Hanging Top Rollers stress plot [2].

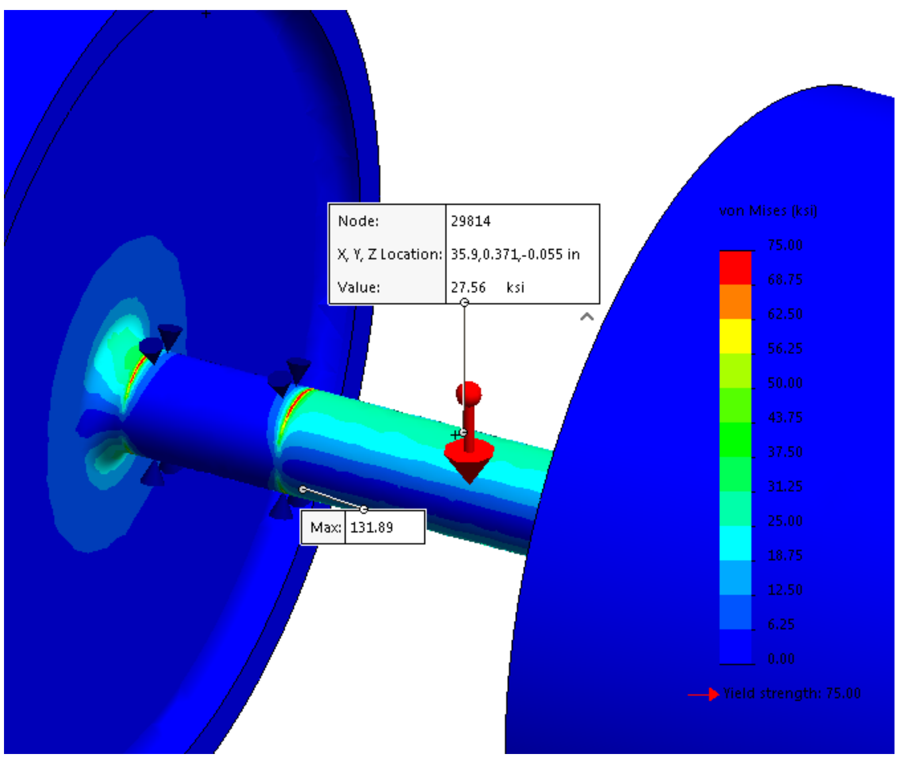

Figure E-5: Hanging Top Rollers stress concentration plot [2].

Looking at the other high stress areas of the shaft, maximum stresses of approximately 27.56 ksi exists on the very top of the middle of the shaft and fillets. This intuitively makes sense due to matching the stress patterns that are due to bending, which is the primarily source of stress present in this study. Comparing the maximum stress to the yield strength of the shaft, it can be found that the factor of safety of the shaft is 2.72. Additionally, to ensure the shaft is truly experiencing primarily bending stresses significant only the middle of the shaft, a section plot of stress through the middle of the model is shown in Figure E-6. Notably, the stress patterns do match as expected with the applied loading conditions.

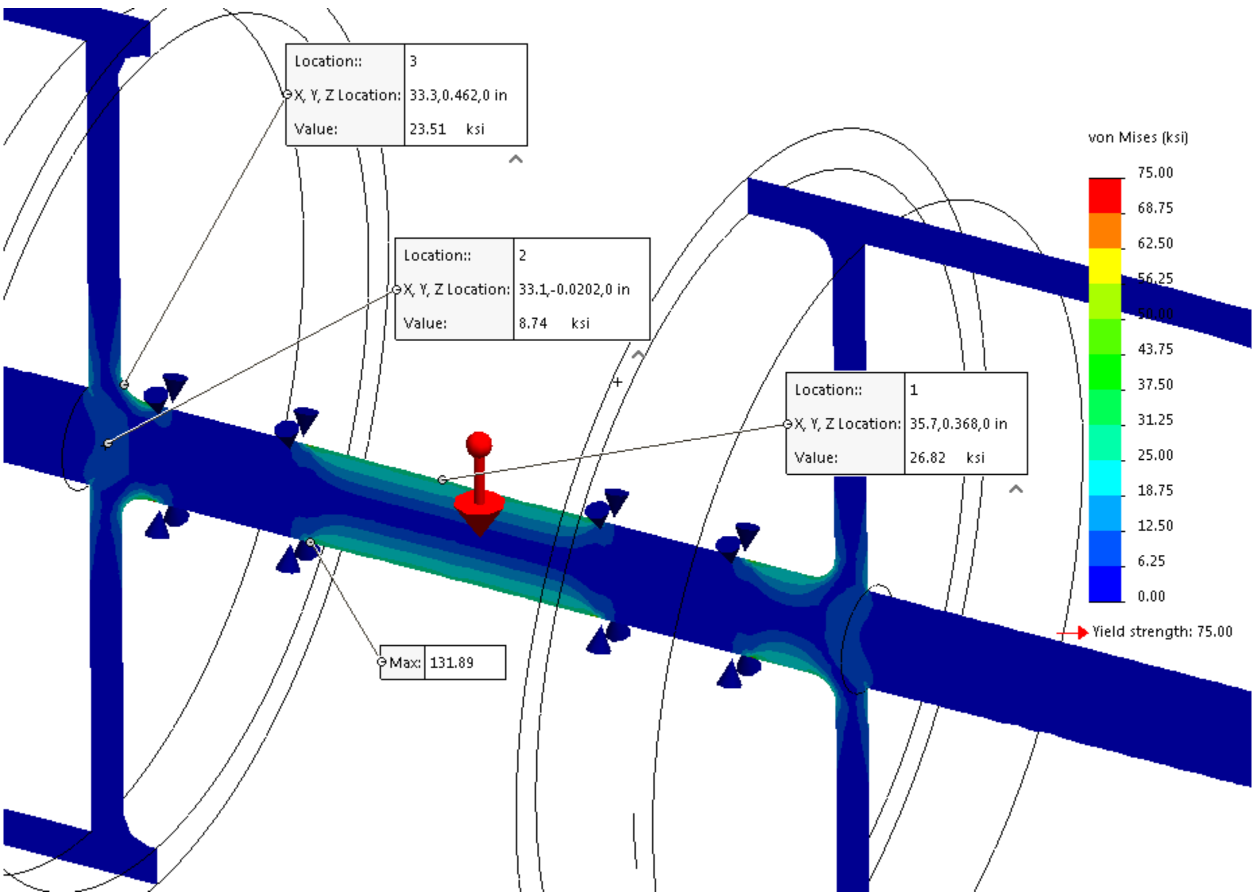

Figure E-6: Hanging Top Rollers internal stress plot [2].

1.1.2. Hanging Plate Assembly

The hanging plate assembly was setup by modeling both top roller tube arms and hanging plate as a solid body. Between parts, small fillets of 0.125" radius were applied to better predict any stress concentrations in the welds. Due to the expectation that the single bolt hole on the hanging plate was going to be a larger concern in terms of stress compared to the 16 larger bolts holes which mount the bearing housings, the study was setup upside-down. Fixed hinge boundary conditions were placed on each bearing housing's 7/16" bolt holes. Additionally, a bearing force of 250 lbs directed upward on the bolt hole of the hanging plate was applied. This upward force is the equal and opposite reaction force of the hanging top rollers weight that would be applied to the part. The study setup is shown in Figure E-7.

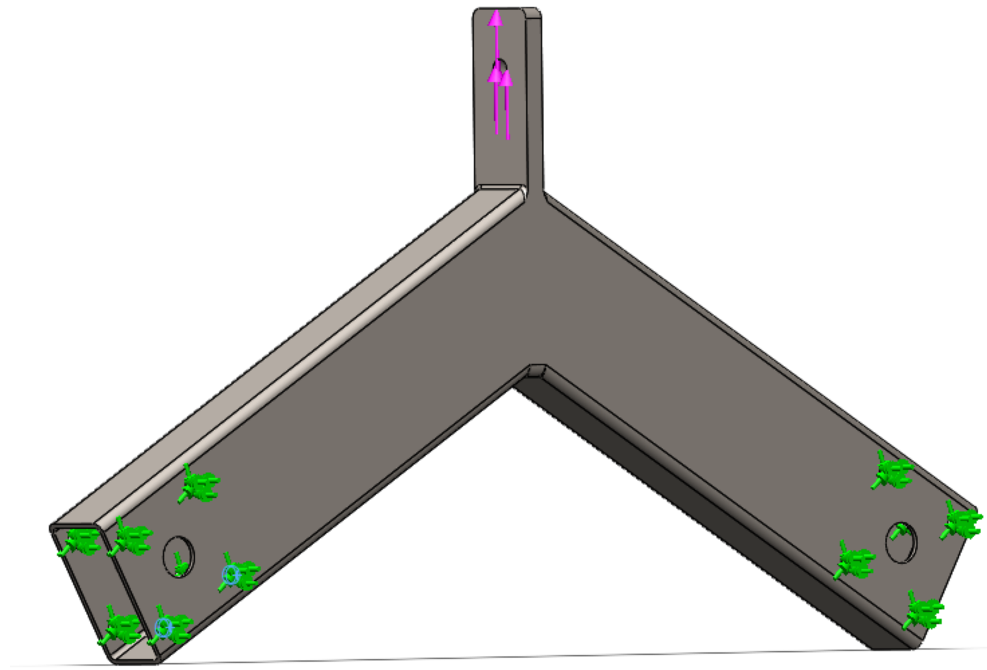

Figure E-7: Hanging Plate FEA boundary conditions [2].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 69116 total elements, with a minimum element size of 0.361". As can be seen in Figure E-8 and Figure E-9, the static study converged to 2.0% total relative strain energy with the majority of elements being located around the fillets and hanging plate bolt hole.

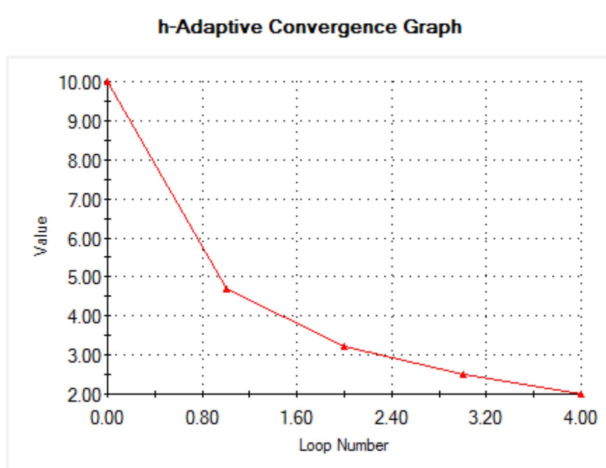

Figure E-8: Hanging Plate Assembly h-adaptive convergence plot [2].

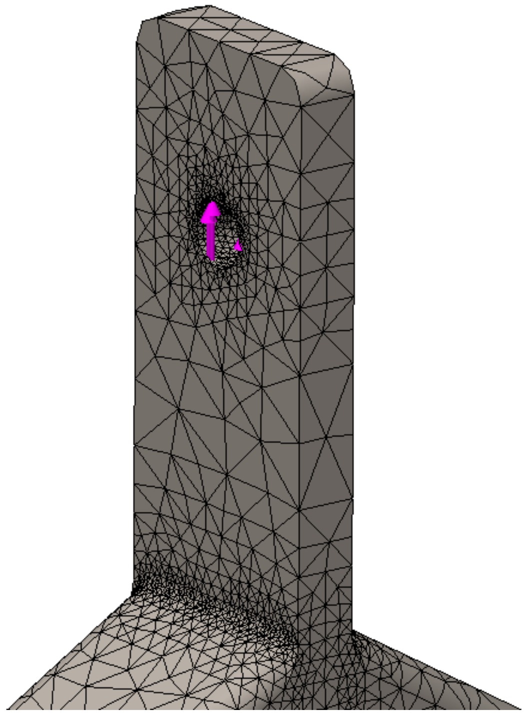

Figure E-9: Hanging Plate assembly meshing [2].

Figure E-10 and Figure E-11 show stress plots of the hanging plate assembly, and bearing stress in the hanging plate bolt hole, respectively. The maximum stress occurs in the bolt hole at 1.71 ksi. This stress is significantly lower than the plates yield strength of 36.3 ksi, resulting in a factor of safety of 21.22. To verify the overall low stress is correct, the normal stress, σ through the cross-sectional area, *A* of the hanging plate can be checked analytically by Eq. (E-1) where *F* is the applied force.

$$\sigma = \frac{F}{A} = \frac{250}{(0.5)(2)} = 250\,psi = \mathbf{0.25\,ksi} \qquad \text{(E-1)}$$

Overall, the analytical result does reasonably match that of the FEA model.

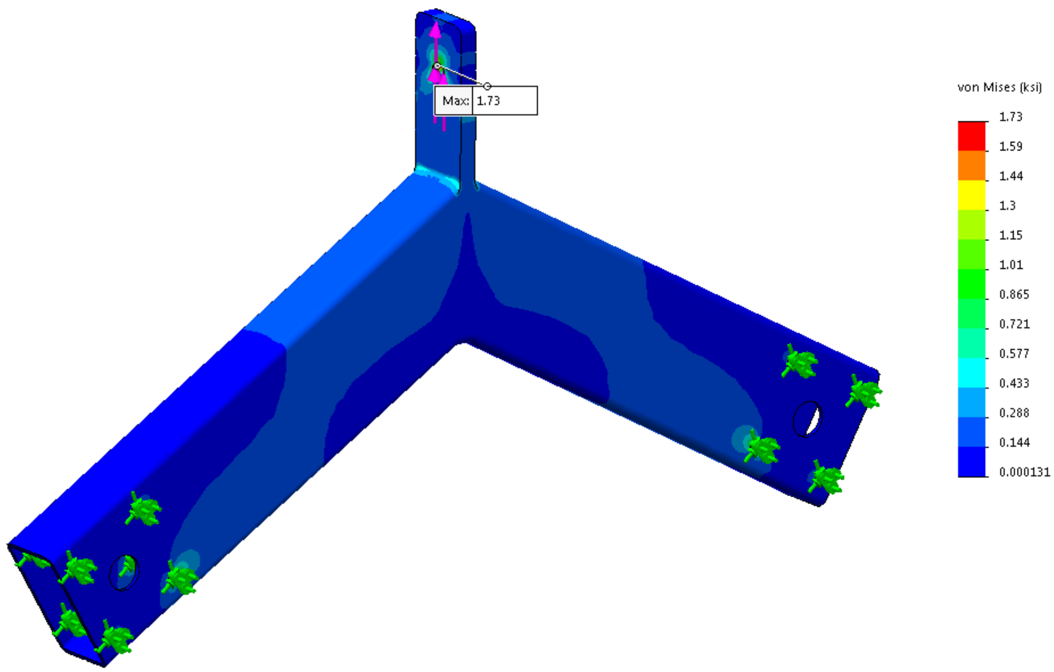

Figure E-10: Hanging Plate assembly stress plot [2].

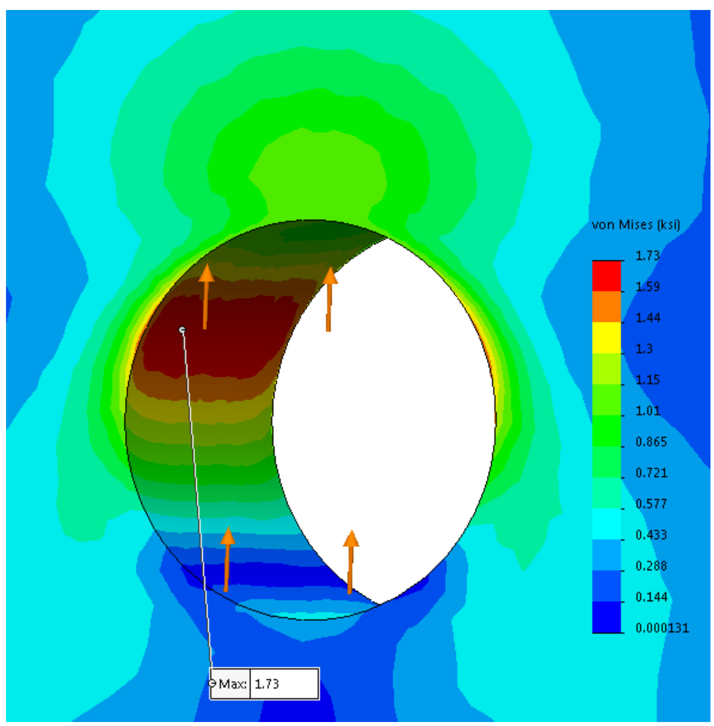

Figure E-11: Hanging Plate Assembly bearing stress plot [2].

1.1.3. Swinging Arm Tubing

The actual swinging arm model was used directly in the FEA stress analysis presented below. An external bearing load was applied to the front side bearing bolt hole of 285 lbs, simulating the total weight the top roller assembly hanging from it. Fixed geometry was applied to split faces located at the dedicated position of swinging arm top brackets, simulating the applied load of 3000 lbs by the hydraulic arm. The back-side bolt hole was applied a fixed hinge, simulating the hinged bushing. Finally, gravity was applied downwards such that stress introduced by the arm's own weight were accounted for. Figure E-12 displays the study setup.

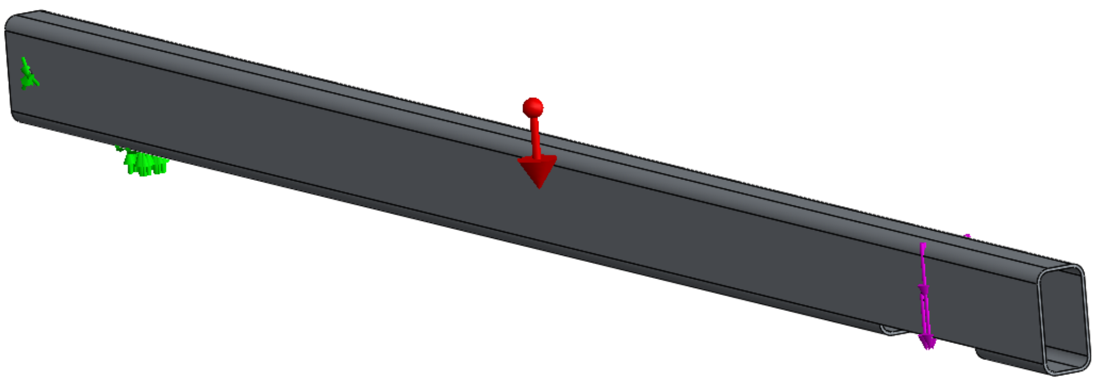

Figure E-12: Swinging Arm Tubing FEA boundary conditions [2].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 53600 total elements, with a minimum element size of 0.448". As can be seen in Figure E-13 and Figure E-14, the static study converged to 4.0% total relative strain energy with the majority of elements being located around the fixed boundary conditions.

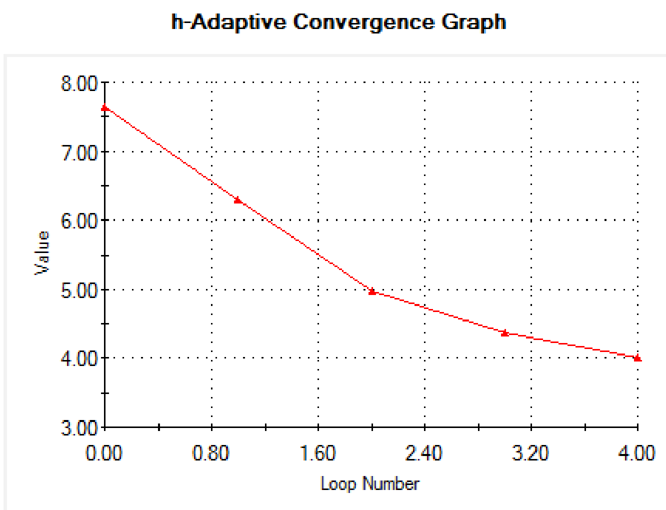

Figure E-13: Swinging Arm Tubing h-Adaptive convergence plot [2].

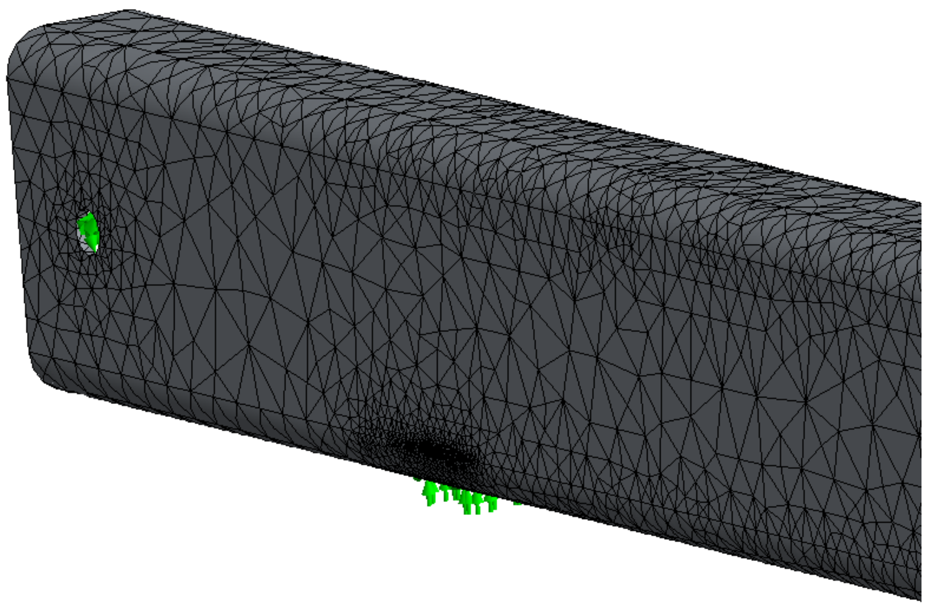

Figure E-14: Swinging Arm tubing meshing [2].

Figure E-15 and Figure E-16 show the stress plot of the entire tube and the stress concentration near the fixed geometry boundary condition, respectively. The maximum stress in the model is shown as 322.15 ksi. However, this stress is an induced divergent concentration due to the applied fixed boundary condition, and thus is ignored. By probing the locations of highest stress outside this small concentration region, 12.50 ksi is found, and thus the factor of safety of the tube is 3.68.

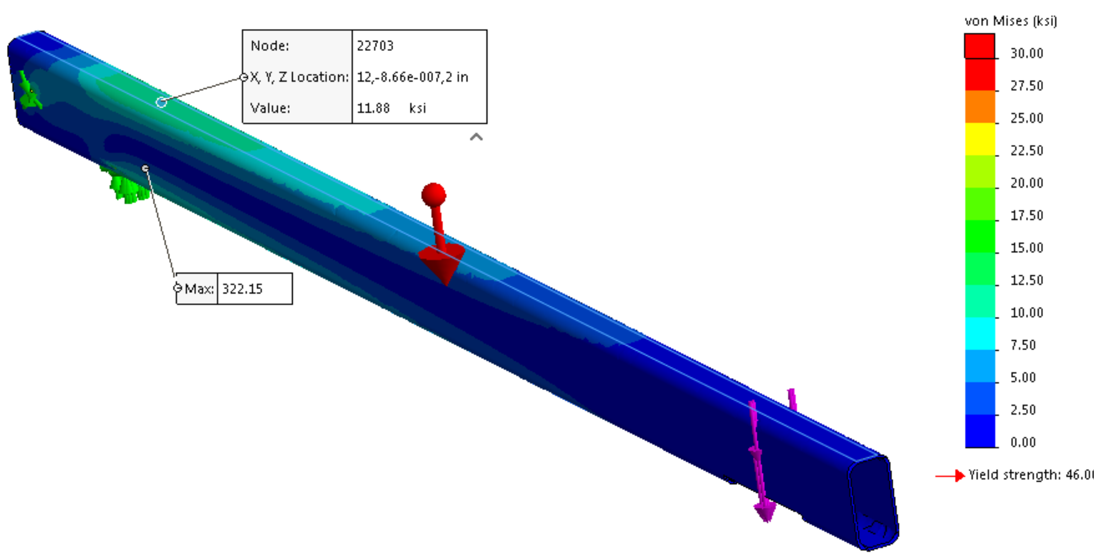

Figure E-15: Swinging Arm Tubing stress plot [2].

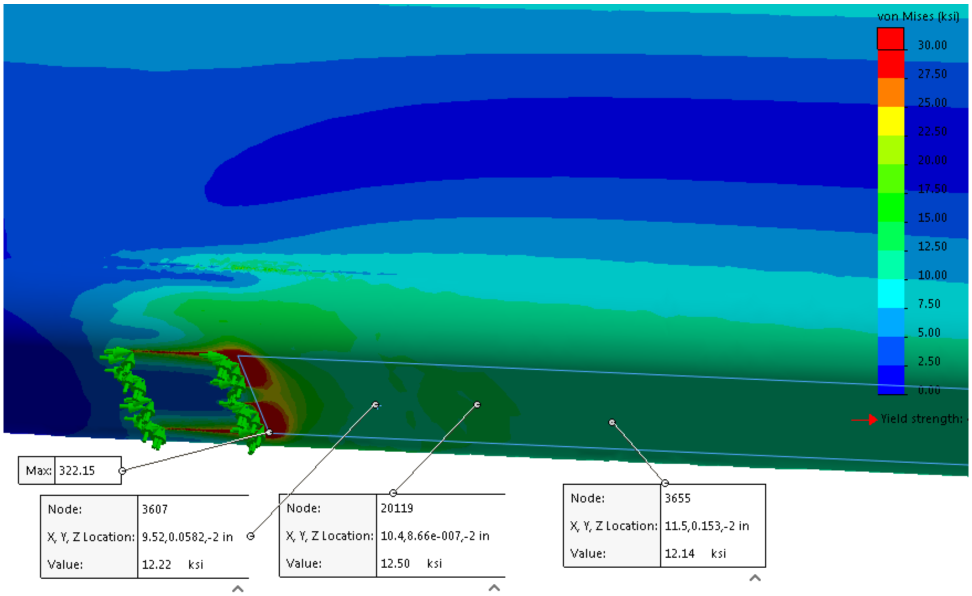

Figure E-16: Swinging Arm Tubbing stress concentration plot [2].

To verify this stress, the tube can be represented by a simply supported beam. Figure E-17 shows the loading scenario, shear, and moment diagram of the beam.

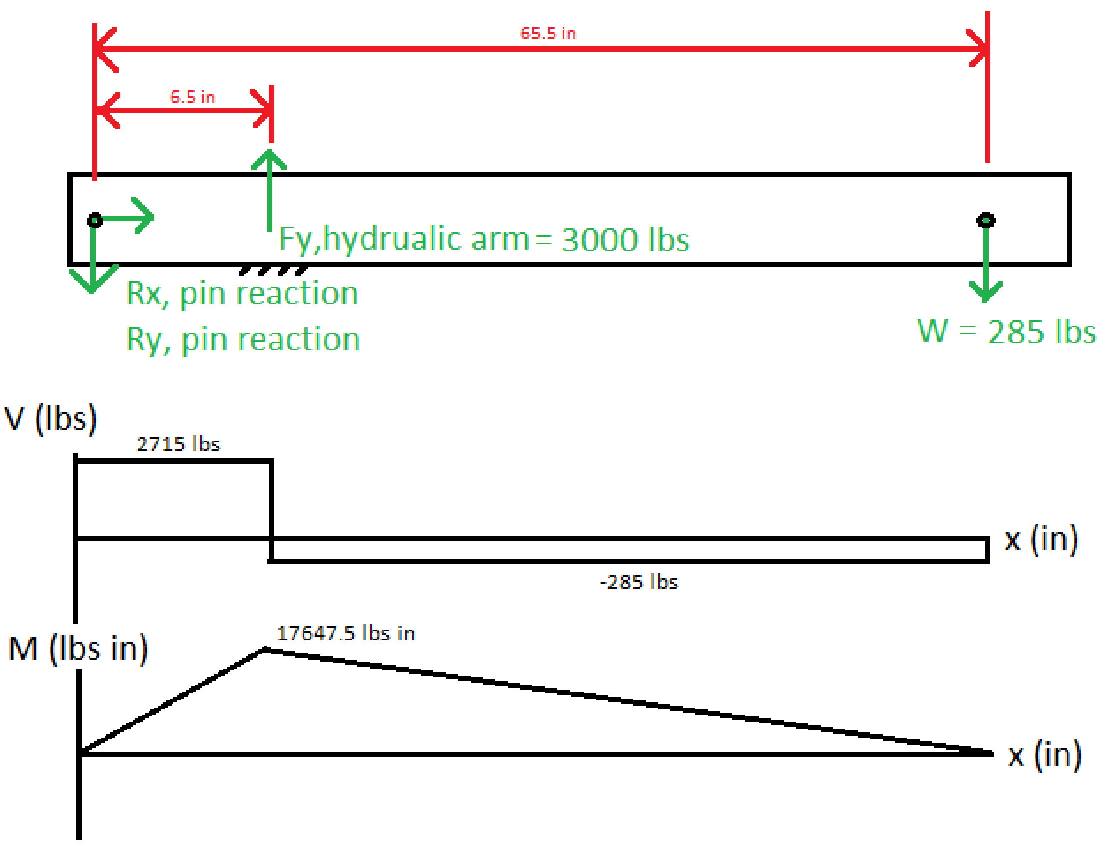

Figure E-17: Shear and moment diagrams of Swinging Arm Tubing [2].

As seen in the Figure, the largest moment occurs at the applied force of the hydraulic arm (or fixed geometry in the FEA model) as expected. Using this moment, a maximum normal bending stress can be analytically calculated using Eq. (E-4). The dimensions of this tube are 4" x 2" x 0.125" thickness, thus:

$$b = 2", h = 4", t = 0.125"$$

$$M = 17647.5\ lbs * in$$

$$c = h/2 = 2"$$

$$I = \frac{1}{12} bh^3 \ - \frac{1}{12}(b - 2t)(h - 2t)^3 = 2.976\ in^4$$

$$\sigma_{bending} = \frac{(17647.5)(2)}{2976} = 11859\ psi = \mathbf{11.9\ ksi}$$

Thus, the relative error between analytical and FEA results is 4.8%.

1.1.4. Side Arm Tubing

The side arm tubing support FEA stress analysis was setup by combing both side arm brackets with the side arm support into a solid model. This was done to generate a better overall stress reading in the region of the side bracket, which is expected to be critical in the design. Additionally, large fillets of 0.25" radius were added on the top, bottom, and outside of each bracket to help simulate the stress that would be experienced by welds in those locations. Additionally, although not a perfect representation of the actual parts, a smaller radius of 0.03" radius was placed on the inside of each bracket and the side arm tube such that the simulation would prevent these locations from diverging to infinite stresses. A fixed geometry was applied to the bottom face of the tube, simulating where the tube will be fixed via welds and bolts to the bottom frame. Bearing loads of 2715 lbs and 3000 lbs were applied to the top bolt holes on the tube, and to the bolts holes on the side arm brackets, respectively. These forces were determined by a simple static moment balance on found in Figure E-17 of the Swinging Arm Tube analysis. Figure E-18 displays the setup of the static study.

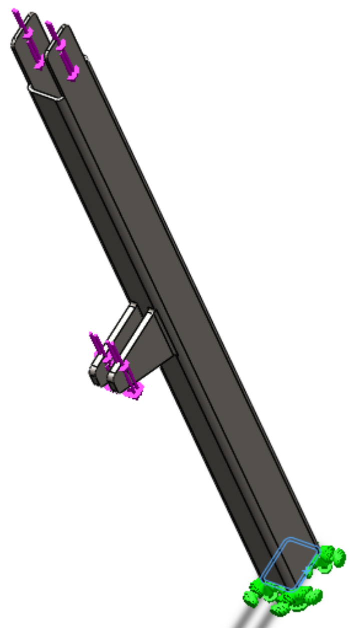

Figure E-18: Side Arm Tubing FEA boundary conditions [2].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 93365 total elements, with a minimum element size of 0.556”. As can be seen in Figure E-19 and Figure E-20, the static study converged to 1.9% total relative strain energy with the majority of elements being located around the fillets between the tubing and brackets, and top tubing bolt holes.

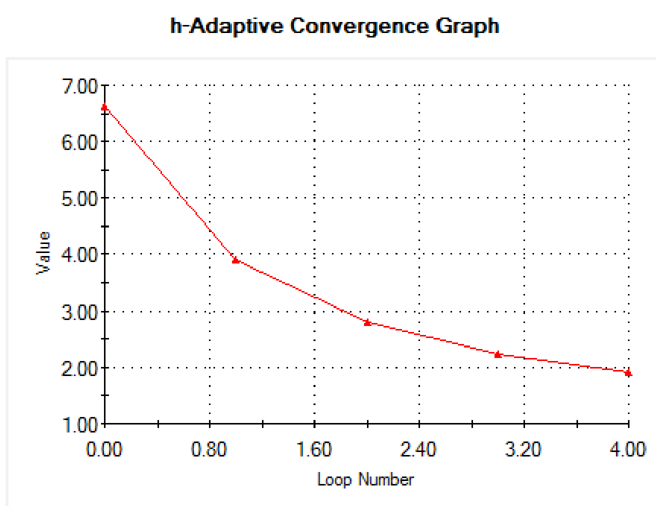

Figure E-19: Side Arm Tubing h-adaptive convergence plot [2].

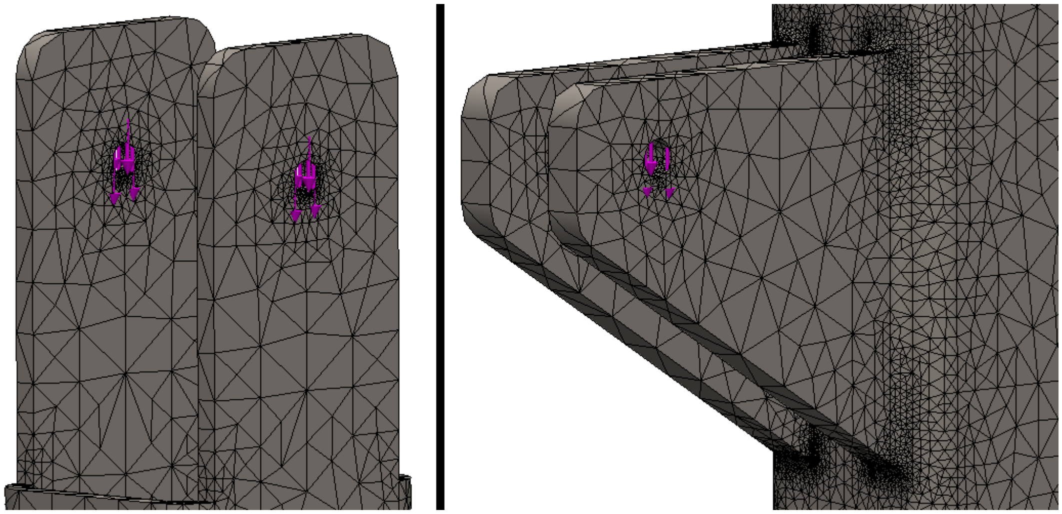

Figure E-20: Side Arm Tubing meshing [2].

Figure E-22 and Figure E-21 show the stress plots of the entire tube and the stress plot of the stress concentrations seen near where the brackets and tube are connected. The maximum stress experienced in the model is 36.65 ksi. However, as seen in Figure E-21, this stress lies directly on the sharp edge of the fillet, and thus is a diverging stress concentration. Ignoring this concentration, and probing areas maximum stress areas further away from the point, a maximum stress throughout the tube can be found to be approximately 10.48 ksi. Additionally, a maximum stress along the bottom fillet can be found to be 26.70 ksi. However, this stress on the fillet is less of a concern in this study due to the actual size and stress in the weld being determined analytical in the Weld Sizing and Stresses section. Focusing on the maximum 10.48 ksi stress, it can be found that the factor of safety of the tube is 4.38.

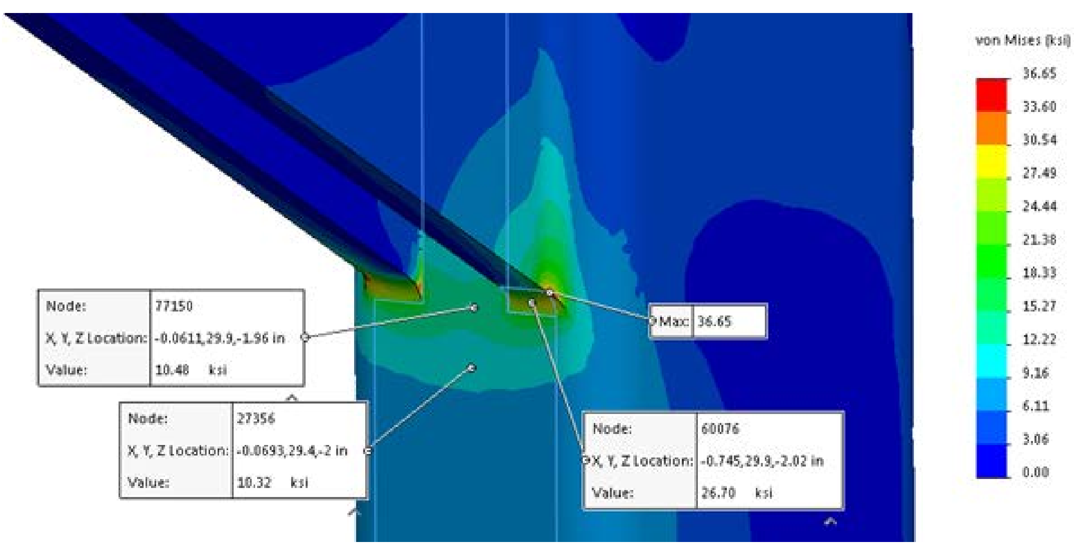

Figure E-21: Side Arm Tubing stress concentration plot [2].

To further validate the results of the study, the front face of the tube was found to be approximately 7.66 ksi. Since this face is experiencing a combination of compressive normal stress and compressive bending stress due an equivalent the moment, analytically we can solve for this stress by combining Eq.'s (E-4) and (E-1), as seen below.

$$\sigma_{max,compression} = \sigma_{bending} + \sigma_{normal} = \frac{Mc}{I} + \frac{F}{A} \quad \text{(E-2)}$$

Where *M* is the equivalent moment about the tubing, *c* is the maximum distance of the face from the neutral axis of the tube, *I* is the moment of inertia of the tube, *F* is the total compressive force acting on the tube, and *A* is the cross-sectional area of the tube. Given the tube is constructed of 4" x 3" x 0.25" thickness tubing, these variables can be solved accordingly.

$$b = 3", h = 4", t = 0.25"$$

$$A = (bh) - (b - 2t)(h - 2t) = 3.25\ in\text{^}2$$

$$F\ = 3000 + 2715 = \ 5715\ lbs$$

$$\sigma_{normal} = 5715/3.25 = 1758\ psi$$

$$M = Force * distance = (3000)(4.5 + 2) = 19500\ lbs * in$$

$$c = h/2 = 2"$$

$$I = \frac{1}{12}bh^3 - \frac{1}{12}(b - 2t)(h - 2t)^3 = 7.068\ in^4$$

$$\sigma_{bending} = (19500)(2)/(7.068) = 5518\ psi$$

$$\sigma_{max,compression} = (5518) + (1758) = 7277\ psi = \mathbf{7.28\ ksi}$$

Thus, the error between analytical results and FEA results are approximately 4.97%.

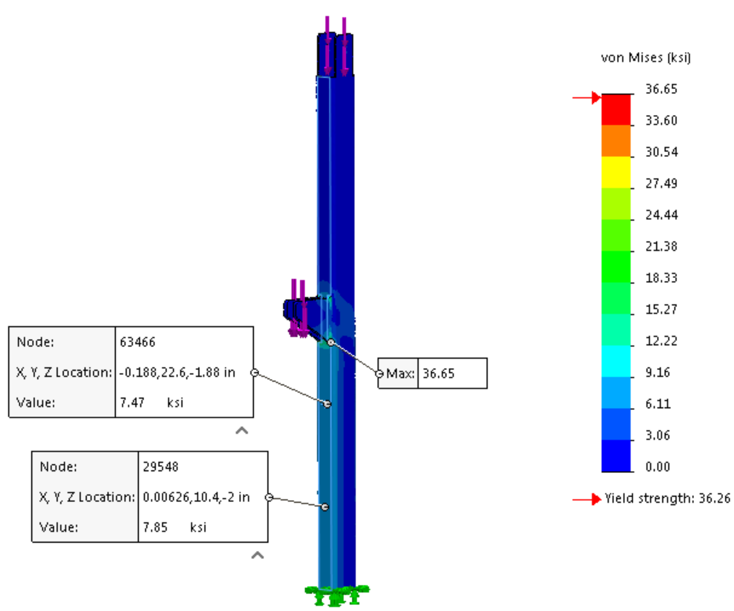

Figure E-22: Side Arm Tubing stress plot [2].

1.1.5. Swinging Arm Top Bracket

As with the side arm bracket, the swinging arm top bracket is also individually checked to ensure it can support the load of the hydraulic arm. A fixed geometry boundary condition is applied to the top face of the bracket where the bracket is to be welded to the swinging arm tube. A 1500 lbs bearing load is applied upward to the right bolt hole to simulate half of the hydraulic cylinders load being applied where they are connected. The setup of the study is displayed in Figure E-23.

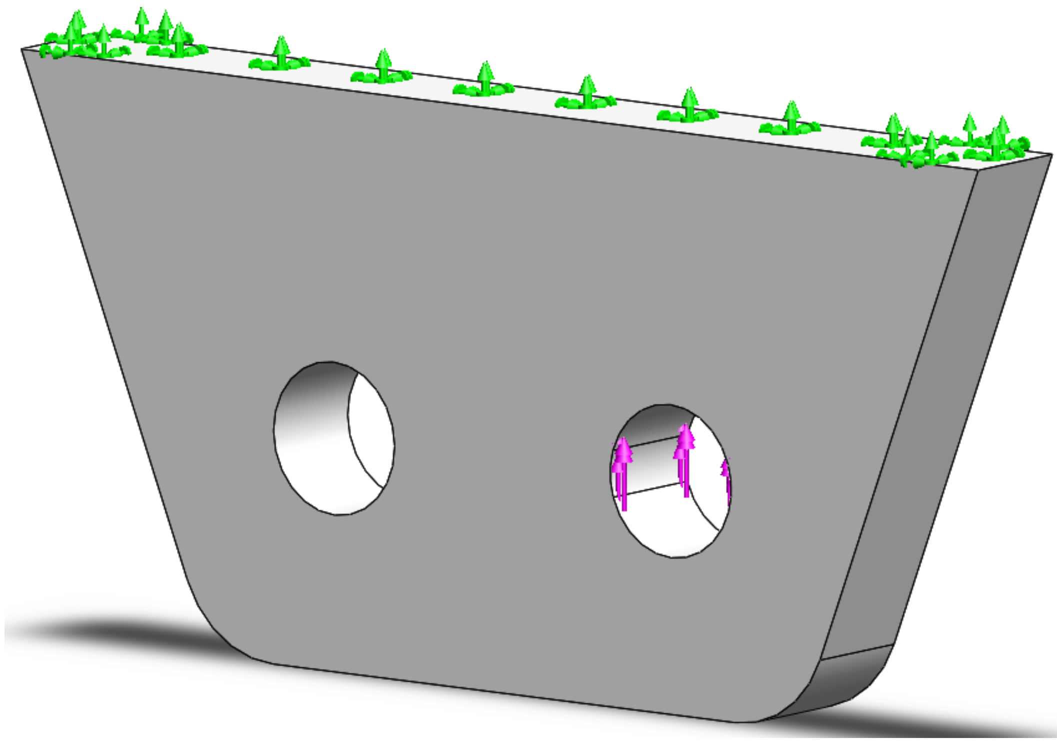

Figure E-23: Swinging Arm Top Bracket FEA boundary conditions [2].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 21066 total elements, with a minimum element size of 0.141". As can be seen in Figure E-24 and Figure E-25, the static study converged to 1.3% total relative strain energy with the majority of elements being located around the fixed boundary condition above the bolt hole and the bolt hole itself.

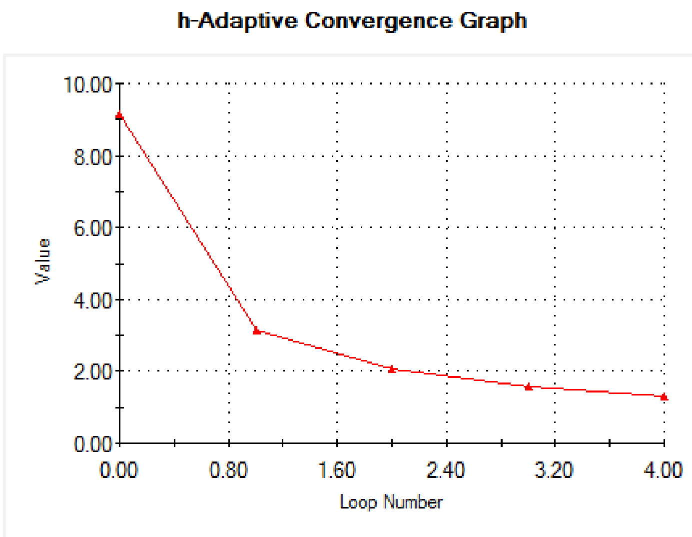

Figure E-24: Swinging Arm Top Bracket h-adaptive convergence plot [2].

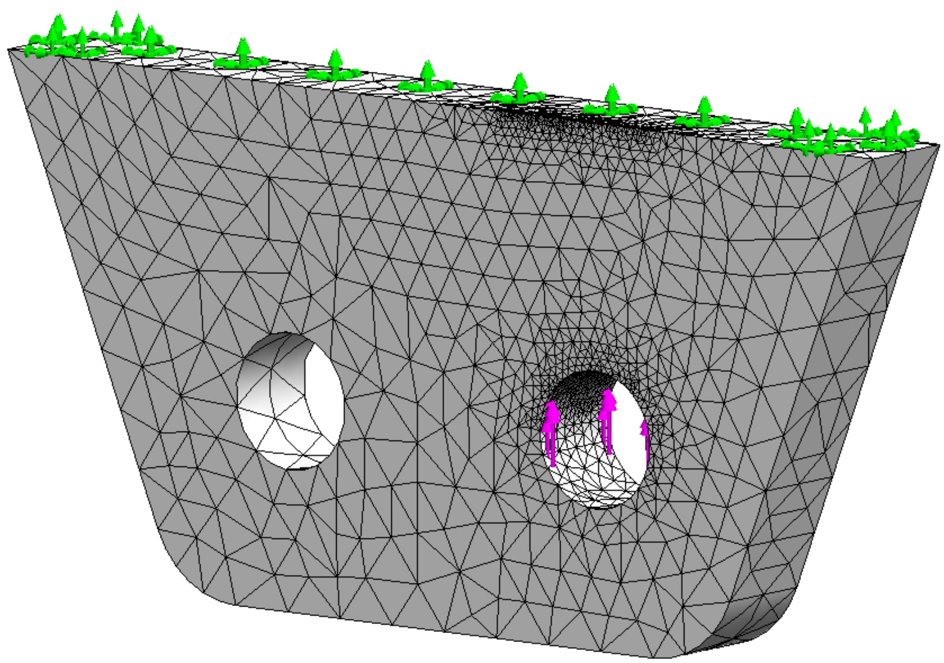

Figure E-25: Swinging Arm Top Bracket meshing [2].

As seen in Figure E-26, the maximum stress of 11.53 ksi in the model occurs on the top side of the bolt hole. This is as expected, and results in a factor of safety of the bracket of 3.15. Figure E-27 displays a closer view of the stress concentrations in this area.

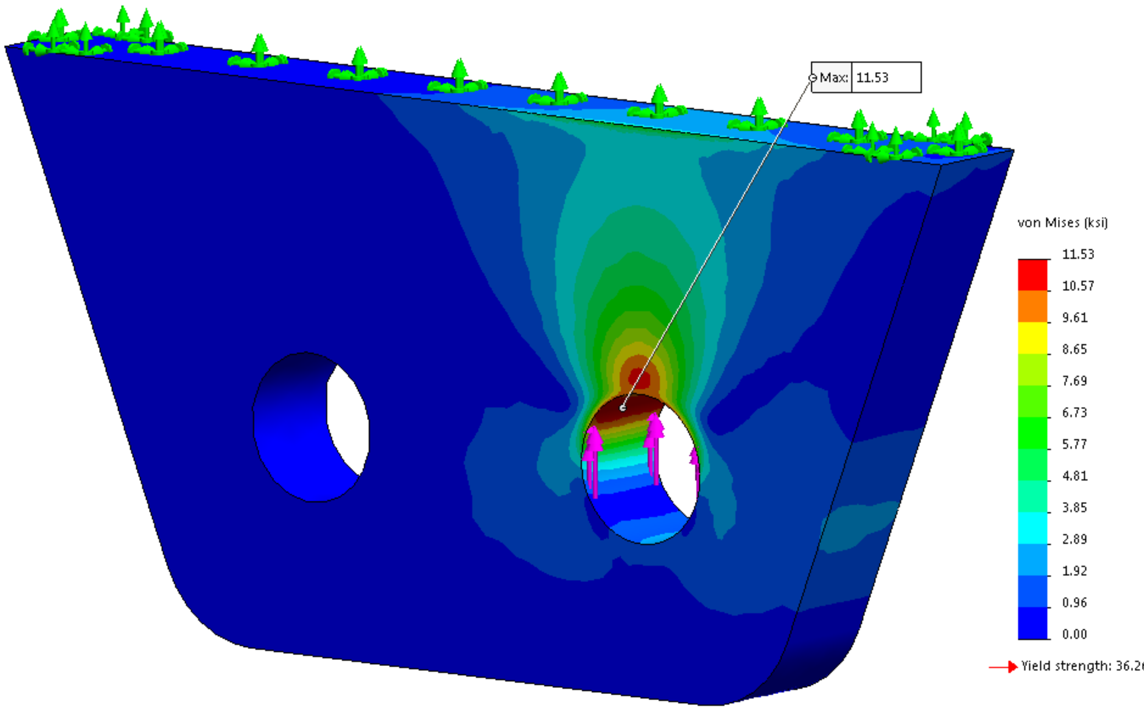

Figure E-26: Swinging Arm Top Bracket stress plot [2].

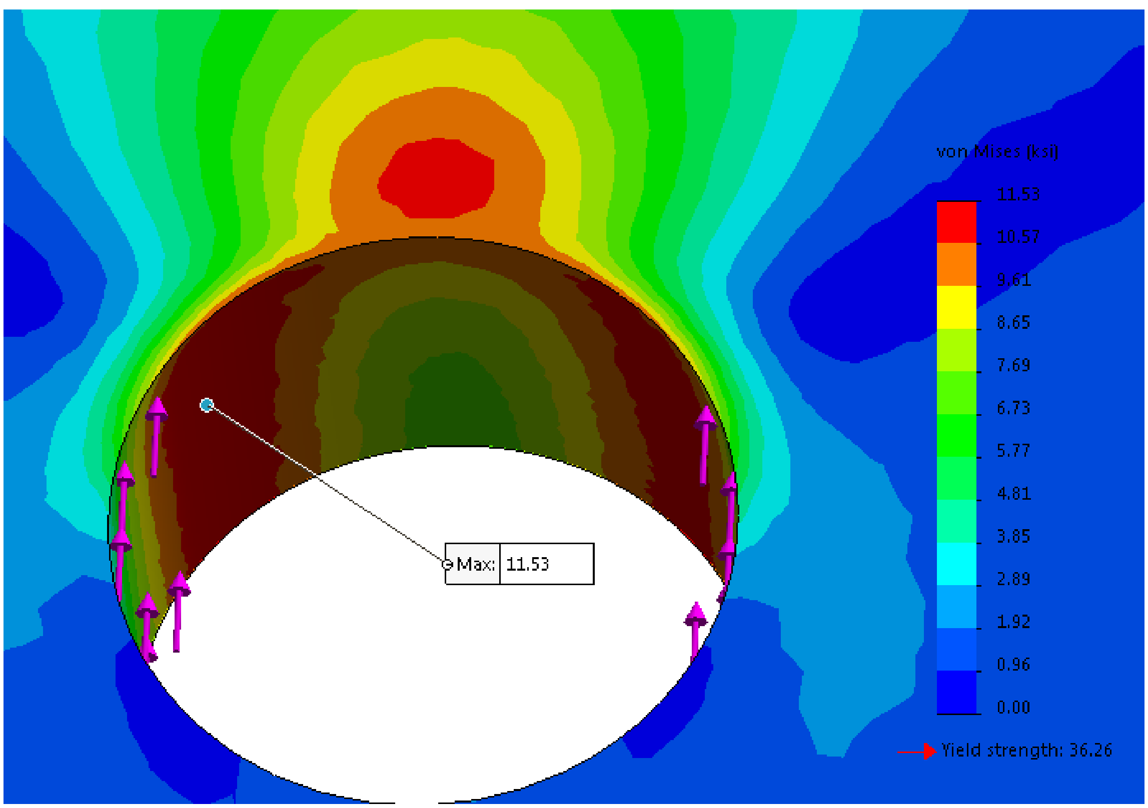

Figure E-27: Swinging Arm Top Bracket bearing stress plot [2].

1.1.6. Side Arm Bracket

An additional stress analysis was done solely on the side arm bracket to ensure it could hold the applied load by the hydraulic cylinder. The study was setup by applying a fixed geometry boundary condition on the backside of the bracket which is welded to the side arm tubing. A bearing load of 1500 lbs, or half the applied load of the hydraulic cylinder (due to having two symmetrical brackets) was applied downward to the brackets bolt hole. The setup of the study can be seen in Figure E-28.

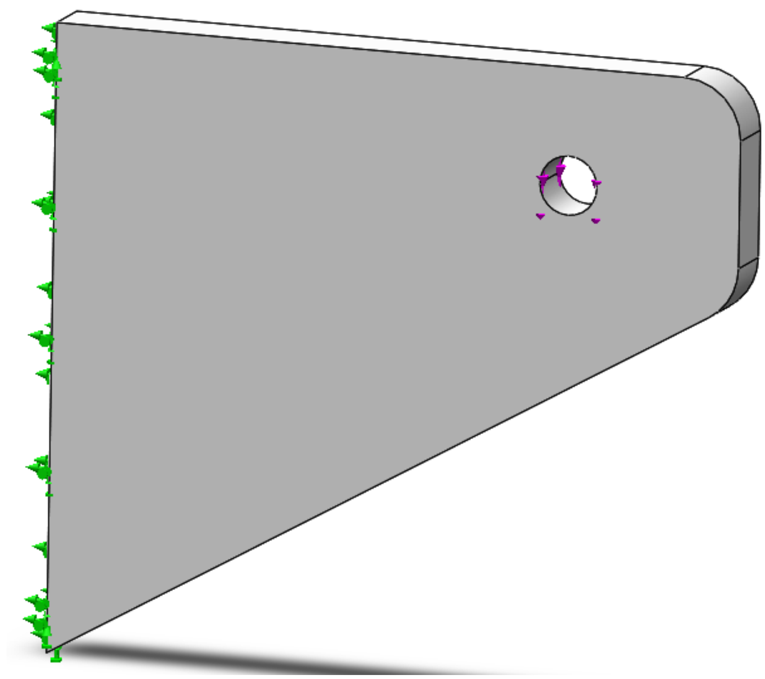

Figure E-28: Side Arm Bracket FEA boundary conditions [2].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 16751 total elements, with a minimum element size of 0.107”. As can be seen in Figure E-29 and Figure E-30, the static study converged to 1.1% total relative strain energy with the majority of elements being located around the fixed boundary condition and bolt hole.

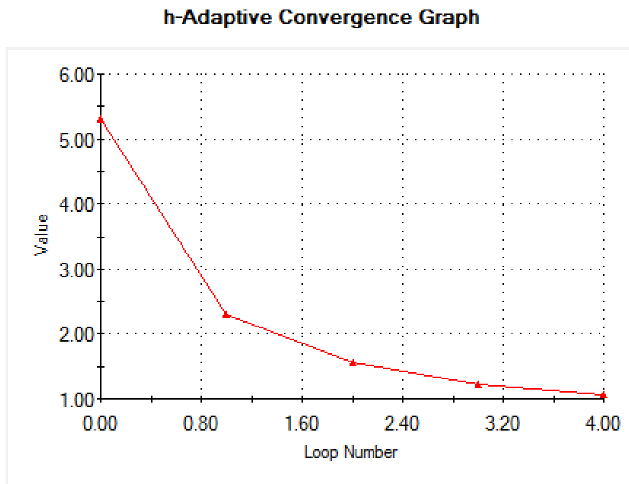

Figure E-29: Side Arm Bracket h-adaptive convergence plot [2].

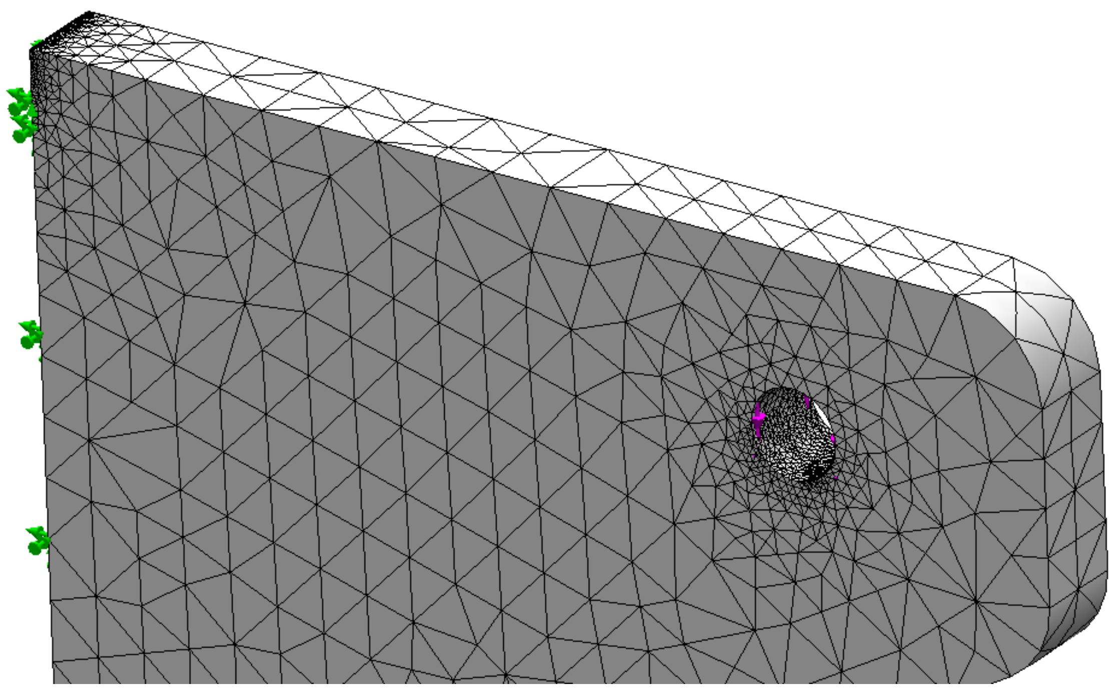

Figure E-30: Side Arm Bracket meshing [2].

As seen in Figure E-31, the maximum stress of 14.54 ksi occurs at the back-corner edge of the bracket. This stress is a diverging stress concentration due to the boundary conditions and thus is ignored. The actual highest stress can be found inside the bolt hole and is 10.47 ksi, giving the safety bracket a factor of safety of 3.46. Figure E-32 displays a better look at the concentrated stresses in this area.

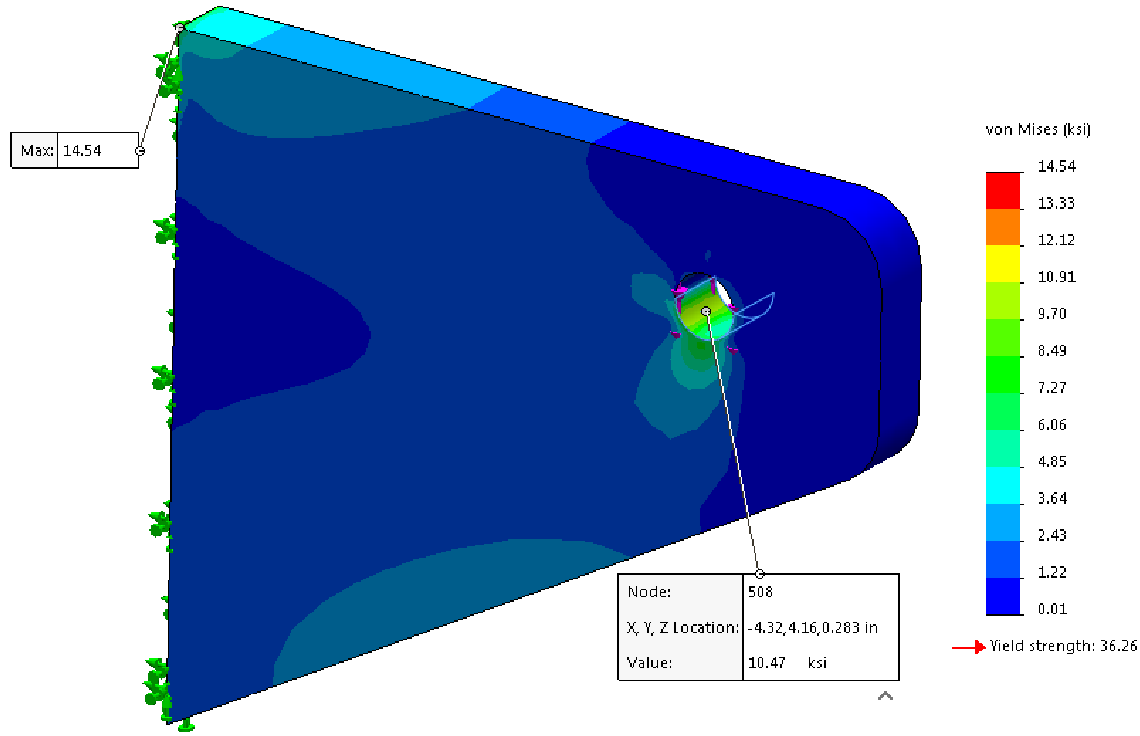

Figure E-31: Side Arm Bracket stress plot [2].

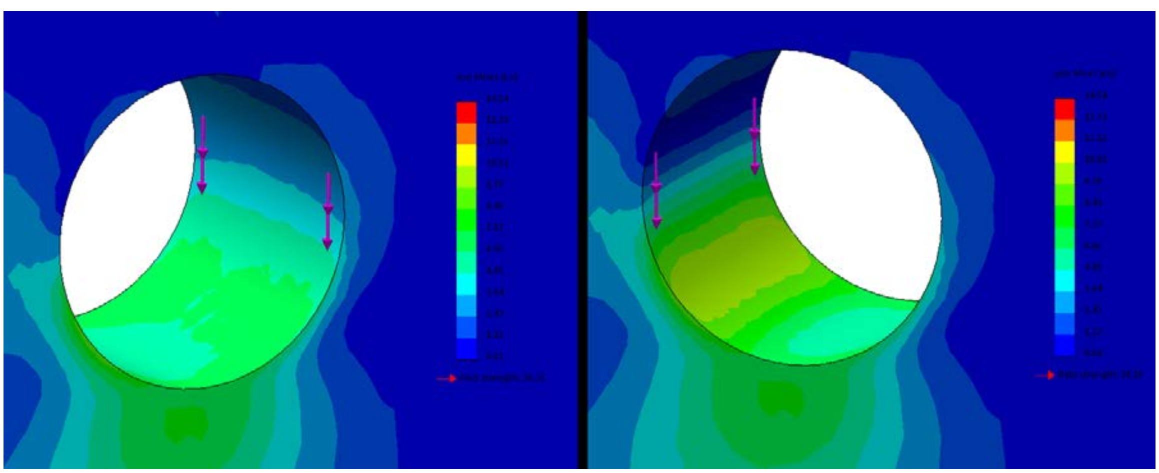

Figure E-32: Side Arm Bracket bearing stress plot [2].

1.1.7. Safety Bracket

For the analysis of the safety bracket, fixed geometry was used on the back and bottom faces, as well as a split face applied at the location that the safety bracket is welded to the Side Arm tubing. A vertical load of 3530 lbs was applied to the lower half of the safety pin hole. Figure E-33 shows the fixtures and loads applied to the safety bracket.

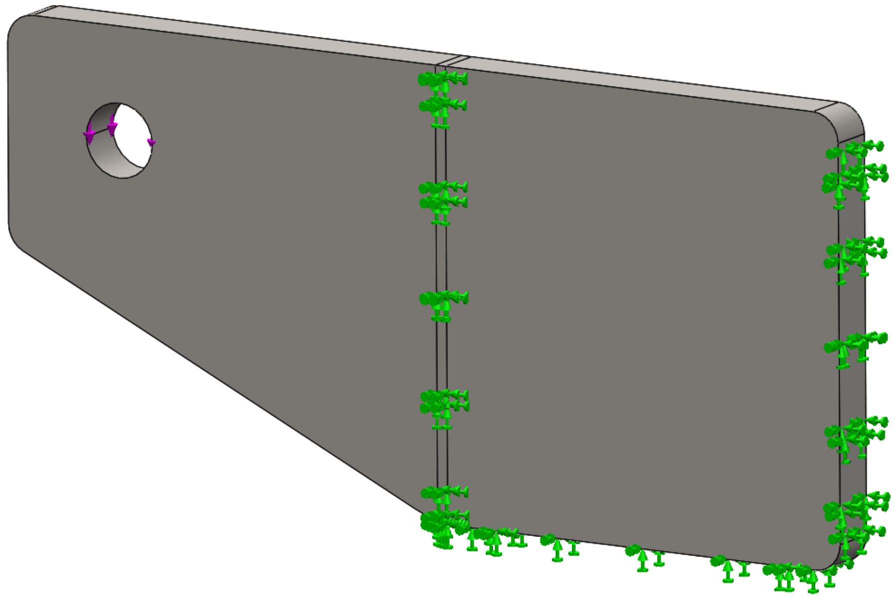

Figure E-33: Safety bracket FEA boundary conditions [3].

After four iterations of mesh refinement, the study converged with a global accuracy of approximately 97.4%. Figure E-34 shows the study convergence plot.

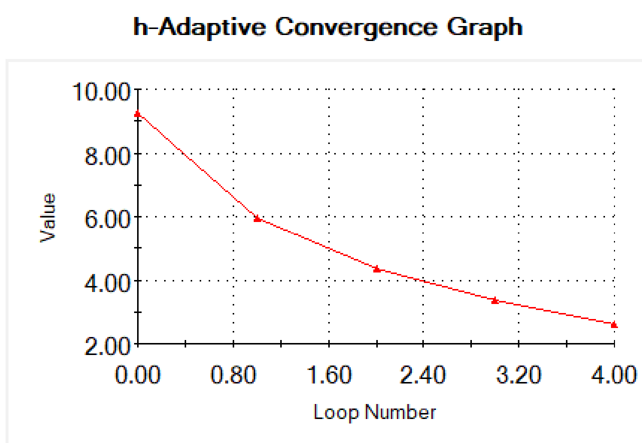

Figure E-34: Safety bracket h-adaptive convergence plot [3].

The final mesh consisted of 149319 elements, with a minimum element size of 0.138”. Figure E-35 shows the final mesh. The elements are concentrated in the area of the fixtures.

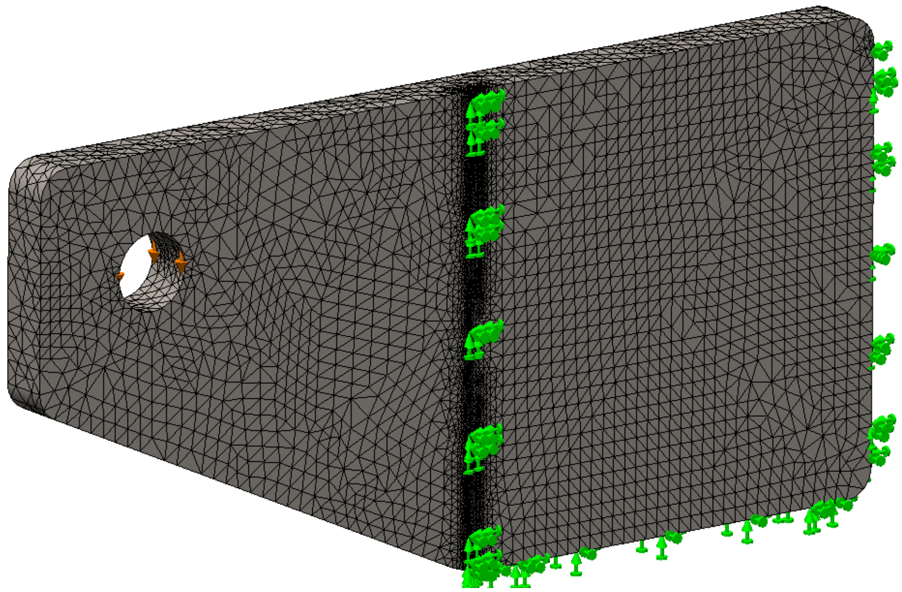

Figure E-35: Safety bracket meshing [3].

Figure E-36 shows the resulting stress plot, with a maximum stress value of 287 ksi occurring along the welded split face. Since this stress concentration occurs at a fixture, it is disregarded. High stress values were also found on the face of the safety pin hole, and are ignored for the same reasons.

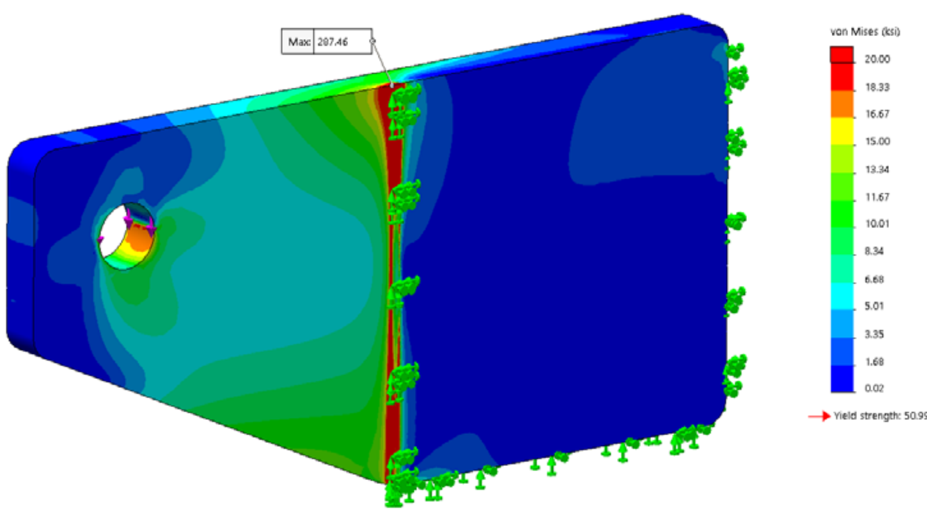

Figure E-36: Safety bracket stress concentration plot [3].

The next highest stress value occurred on the opposite face, at the top of the welded split face. The stress value of 11.95 ksi can be seen in Figure E-37, and results in a factor of safety of 3.01.

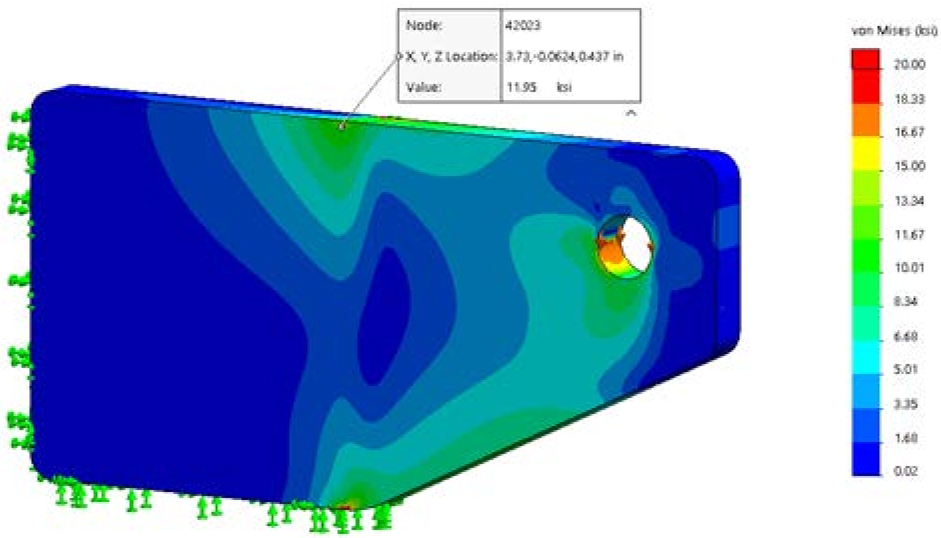

Figure E-37: Safety bracket stress plot [3].

1.1.8. Mounting Plate

For the analysis of the mounting plate, the load applied by the upper roller assembly support member was simulated using a remote load. The load was applied to a split face in the location of the support member, with the load being applied at the location of the upper roller assembly. The magnitude was equal to the weight of the Top Roller subassembly, 285 lbs. Two types of fixtures were used to secure the mounting plate: virtual walls and fixed hinges. Fixed hinges were applied to the inside surface of the bolt holes. This fixture allows rotation about the center axis of the hole, and translation along the axis, but does not permit translation perpendicular to the axis. The second fixture, virtual wall, simulates interactions with other parts. A solid face and a plane are specified, and the face is not allowed to pass through the plane. No resistance to separation of the two faces is provided by this fixture. Virtual walls were used to simulate the cross support to which the mounting plate is bolted, as well as the bolt heads which clamp the two mounting plates around the cross support. Figure E-38 shows the applied loads and fixtures for the finite element study.

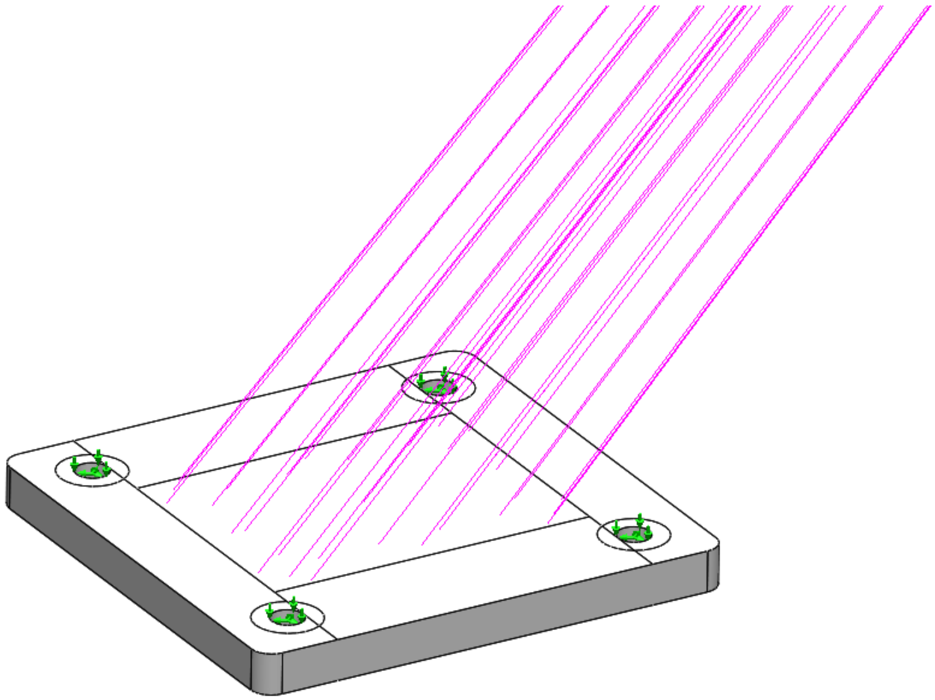

Figure E-38: Mounting plate FEA boundary conditions [3].

After six iterations of mesh refinement, the study converged with a global accuracy of approximately 96%. The convergence plot is shown in Figure E-39.

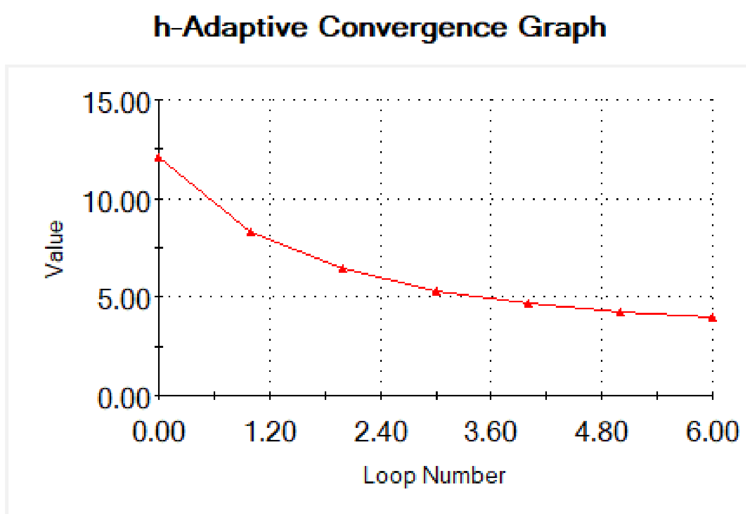

Figure E-39: Mounting plate h-adaptive convergence plot [3].

The final mesh consisted of 77085 elements with a minimum element size of 0.085”. The final mesh is shown in Figure E-40.

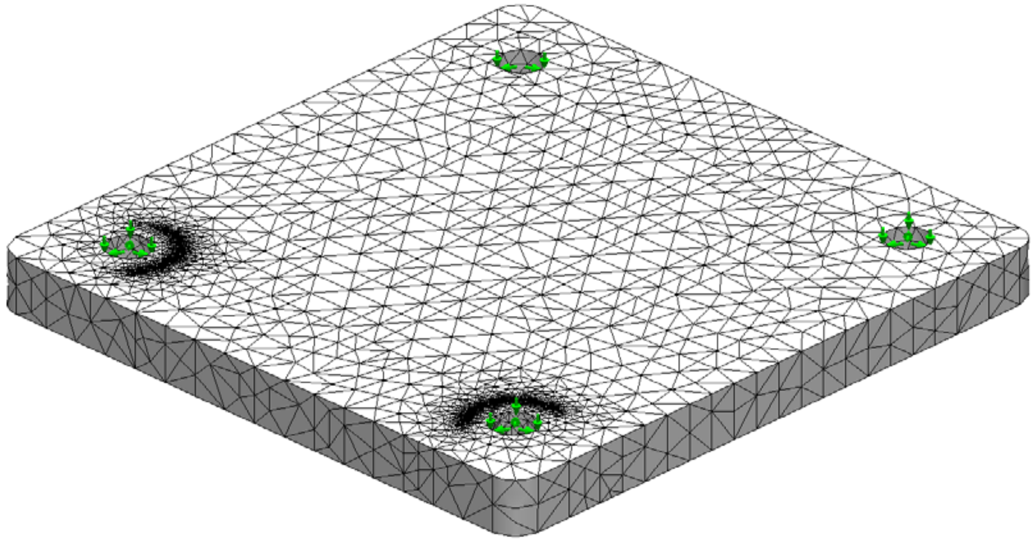

Figure E-40: Mounting plate meshing [3].

As can be seen, the elements are densely concentrated around two of the bolt holes. This area corresponds to the edge of the simulated washer.

Figure E-41 shows the resulting stress plot, with high stress values and gradients near the edge of the simulated washers.

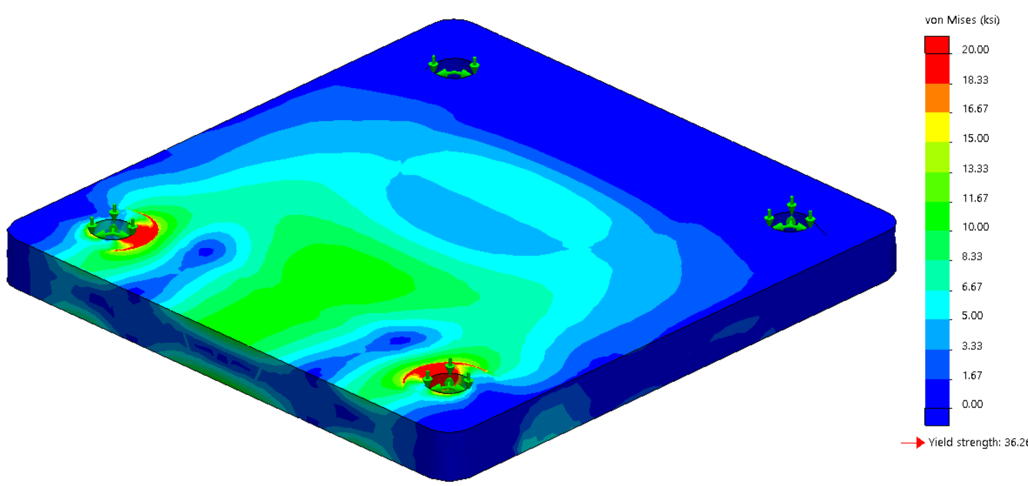

Figure E-41: Mounting plate stress plot [3].

If the stress values near the bolt holes are ignored due to their close proximity to the edge of fixtures, the peak stress of 11 ksi results in a factor of safety of 3.27.

1.1.9. Long Frame Support

For analysis of the long frame support, bearing fixtures were used at the location of the Walking Axle mounting holes. This fixture simulates the sinusoidal distribution of force provided by the bolt running through the support. At the location of each cross-support member, a vertical load of 538 lbs was applied to a split face the size of the cross supports. Additionally, a remote load of 142.5 lbs (half the weight of the upper roller assembly) was applied to one end of the part. The load was applied at the location of the roller when it is in the locked position during a composite roll changeover. The load was applied to the split face on which the vertical load is applied, as well as a split face on one vertical face of the support, simulating the cross-brace reinforcement plate. The fixtures and applied loads are shown in Figure E-42.

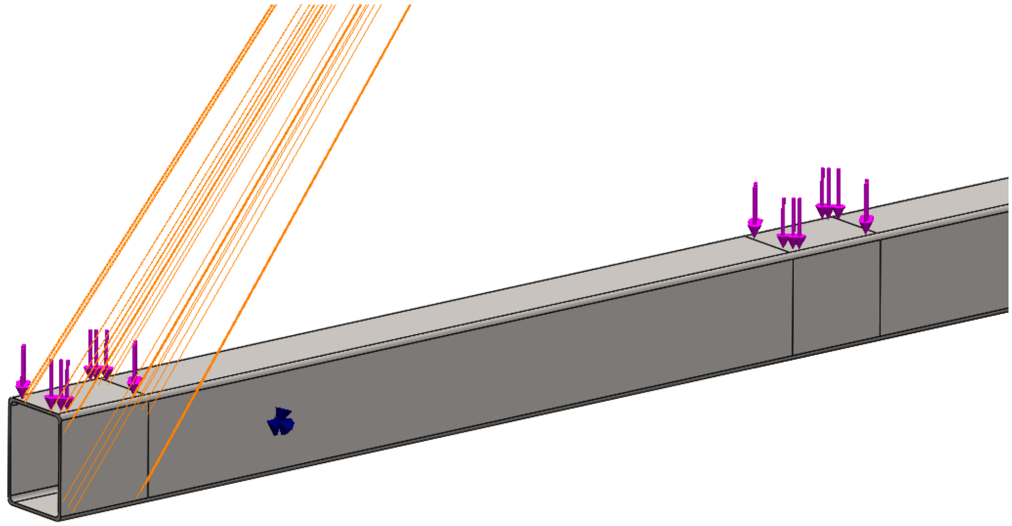

Figure E-42: Long frame support FEA boundary conditions [3].

After four iterations of mesh refinement, the study converged with a global accuracy of approximately 98.1%. The convergence plot is shown in Figure E-43.

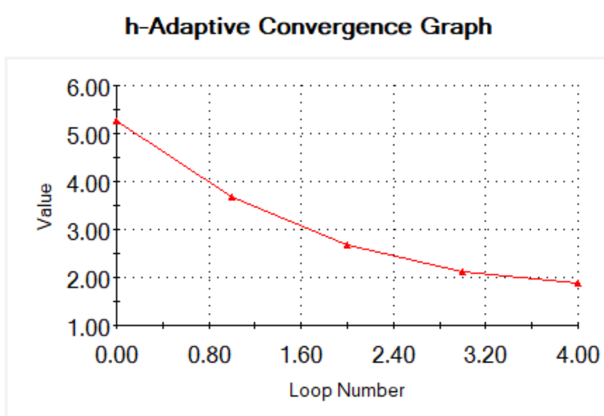

Figure E-43: Long frame support h-adaptive convergence plot [3].

The final mesh consisted of 165273 total elements, with a minimum element size of 0.39”. The final mesh can be seen in Figure E-44. As can be seen, the elements are concentrated around the radiuses of the tube, as well as on the faces where loads and fixtures are applied.

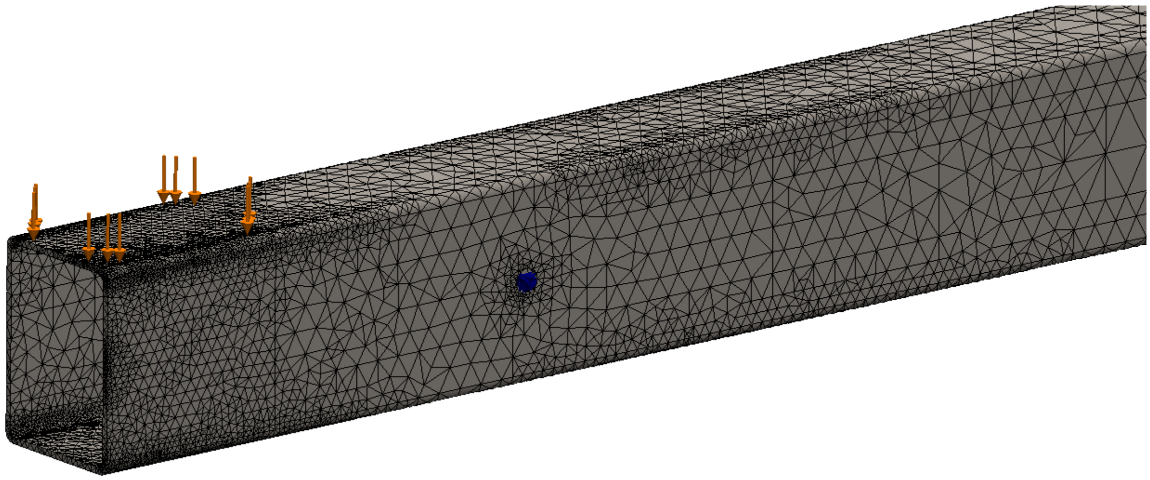

Figure E-44: Long frame support meshing [3].

The resulting stress plot is shown in Figure E-45. A maximum stress value of 21.55 ksi occurred at the edge of the cross-brace reinforcement plate, as can be seen in Figure E-46.

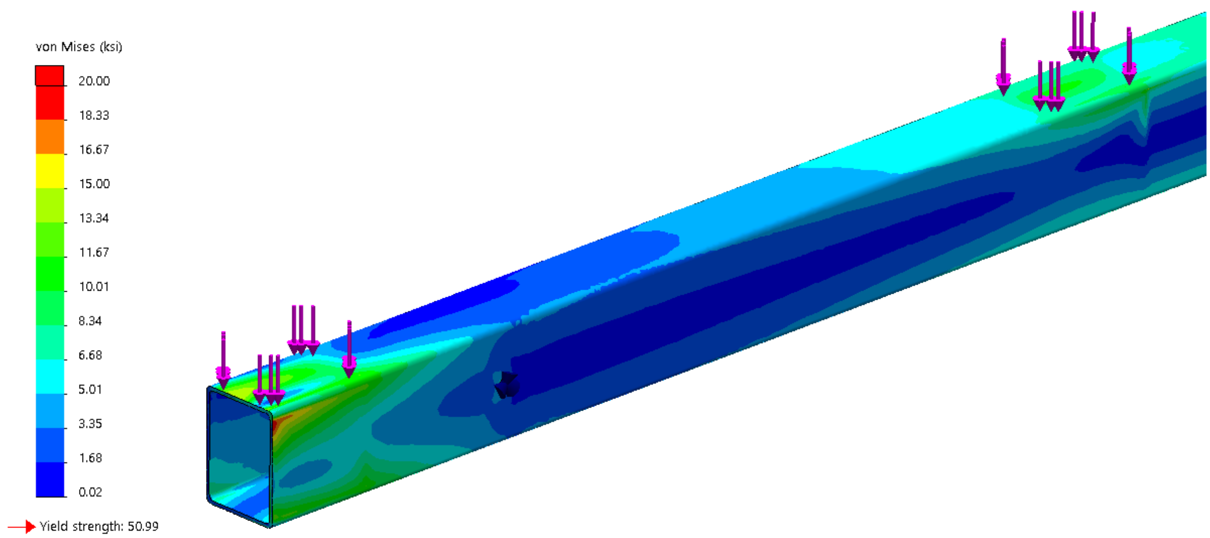

Figure E-45: Long frame support stress plot [3].

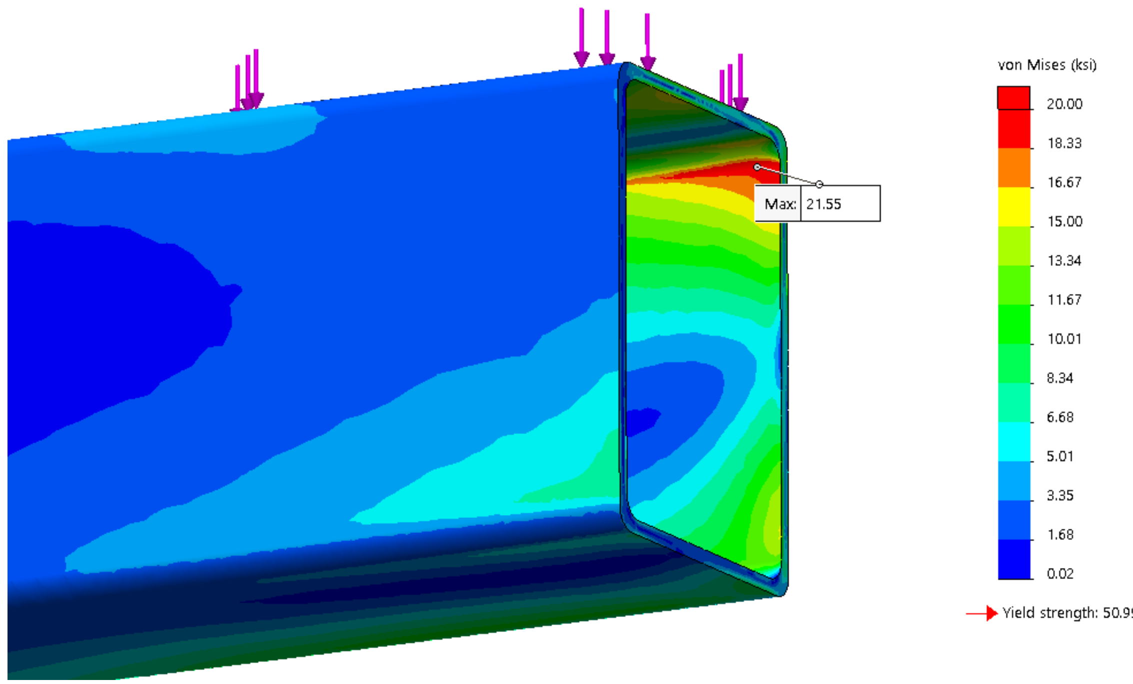

Figure E-46: Long frame support stress concentration plot [3].

As seen in Figure E-46, the maximum stress is localized around the border of an applied fixture, and is therefore disregarded. Ignoring this area, the maximum stress value occurs on the top face of the support at the end which supports the top roller assembly. The stress value of 14.9 ksi results in a factor of safety of 3.09.

1.1.10. Cross Support

When analyzing the stresses in the frame cross supports, two loading scenarios were considered. The first, more simple loading scenario applies to three of the four cross supports, which only support loads from the pillow blocks. The second loading scenario applies to the cross support which also supports the load of the upper roller support member.

For both scenarios, fixed geometry controls were applied to each end, where the support is to be welded to the long frame support member. Also, the vertical loads from the pillow blocks were applied to split faces centered at the location of the pillow block. The load magnitude was set to one eighth of the weight of the roll of material, 538 lbs. These fixtures and loads can be seen in Figure E-47.

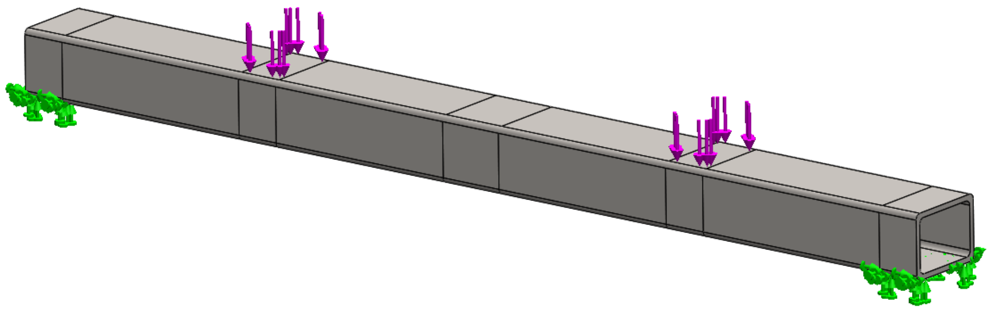

Figure E-47: Cross support FEA boundary conditions [3].

After four iterations of mesh refinement, the h-adaptive study converged with a global accuracy of 95.2%, as shown in Figure E-48.

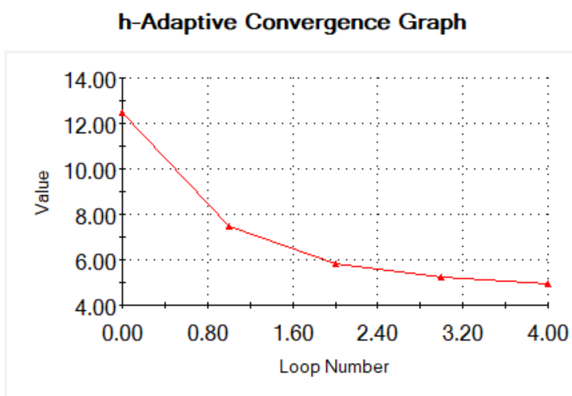

Figure E-48: Cross support h-adaptive convergence plot [3].

The final mesh iteration consisted of 174586 elements, with a minimum element size of 0.231”. As can be seen in Figure E-49, the elements are concentrated around the points of fixturing and load application. This is to be expected, as the part has a relatively simple geometry, and St. Venant’s Principle predicts high stress gradients in the region of the applied energetically equivalent loads.

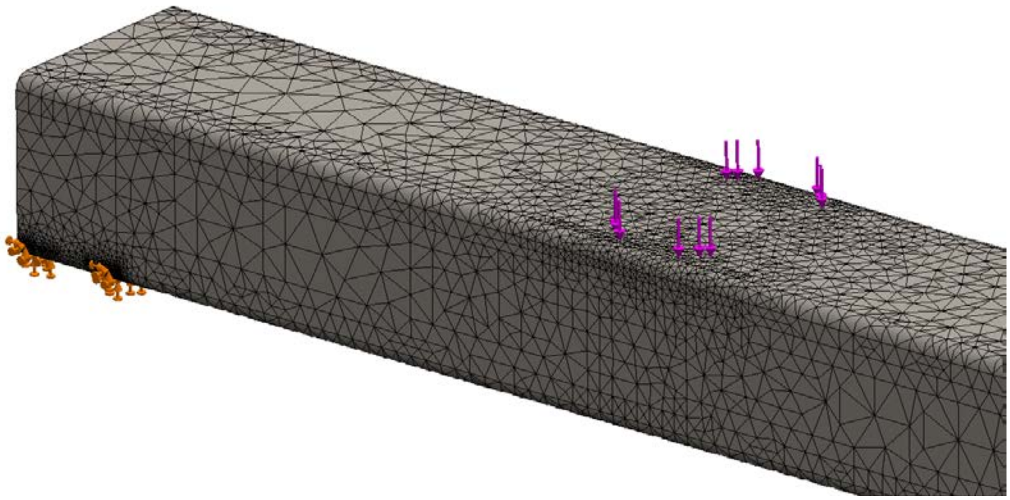

Figure E-49: Cross support meshing [3].

The resulting stress plot can be seen in Figure E-50, as expected, the highest stress values and gradients occurred in the vicinity of the applied loads and fixtures. A highly localized peak stress of 229 ksi occurred along the edge of the fixed geometry, as shown in Figure E-51. This stress concentration was ignored, since FEA does not provide an accurate representation of the loads in the immediate area of the applied loads. Aside from this stress concentration, the peak stress was 13.45 ksi, resulting in a factor of safety of 3.42.

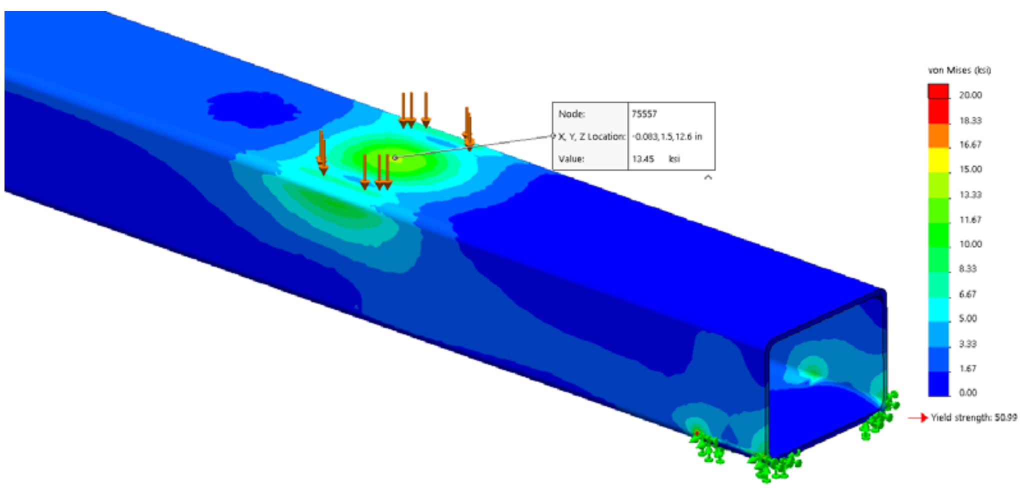

Figure E-50: Cross support stress plot [3].

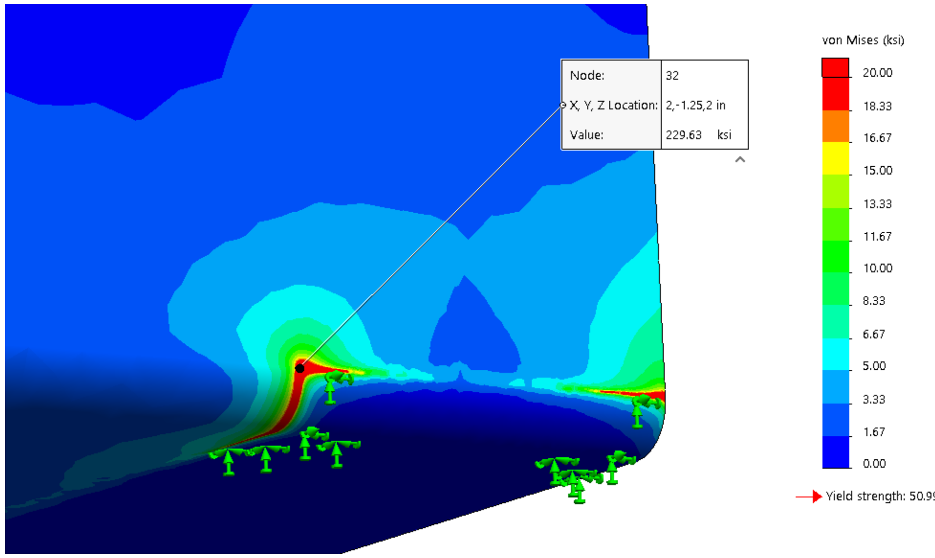

Figure E-51: Cross support stress concentration plot [3].

For the second loading scenario, the load generated by the Top Roller subassembly member was applied via a remote load. The load was applied to a pair of split faces at the location of the mounting plate. Applying the load to both the top and bottom face of the cross brace simulates the load being transferred through both the top and bottom mounting plates. In addition to the fixed geometry used in the first loading scenario, additional fixtures were placed on either end of the cross brace. This fixture simulates the cross-brace reinforcement plate which is welded to the ends of the cross brace. This fixture is shown in Figure E-52.

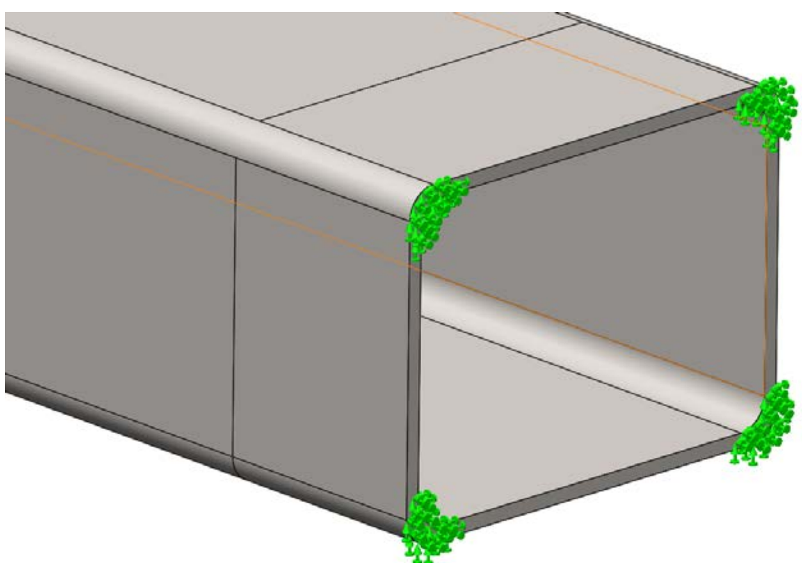

Figure E-52: Cross support scenario two FEA boundary conditions [3].

After six mesh refinement iterations, the study converged with a global accuracy of approximately 96% as can be seen in the convergence plot in Figure E-53.

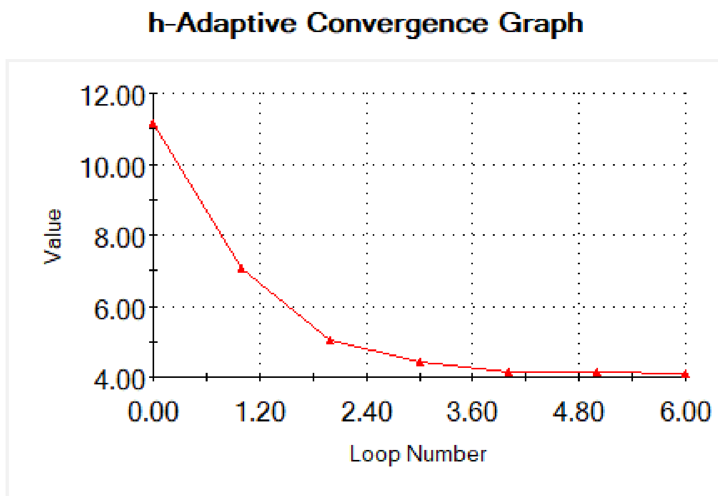

Figure E-53: Cross support scenario two h-adaptive convergence plot [3].

The final mesh consisted of 175850 elements, with a minimum element size of 0.24". The final mesh can be seen in Figure E-54.

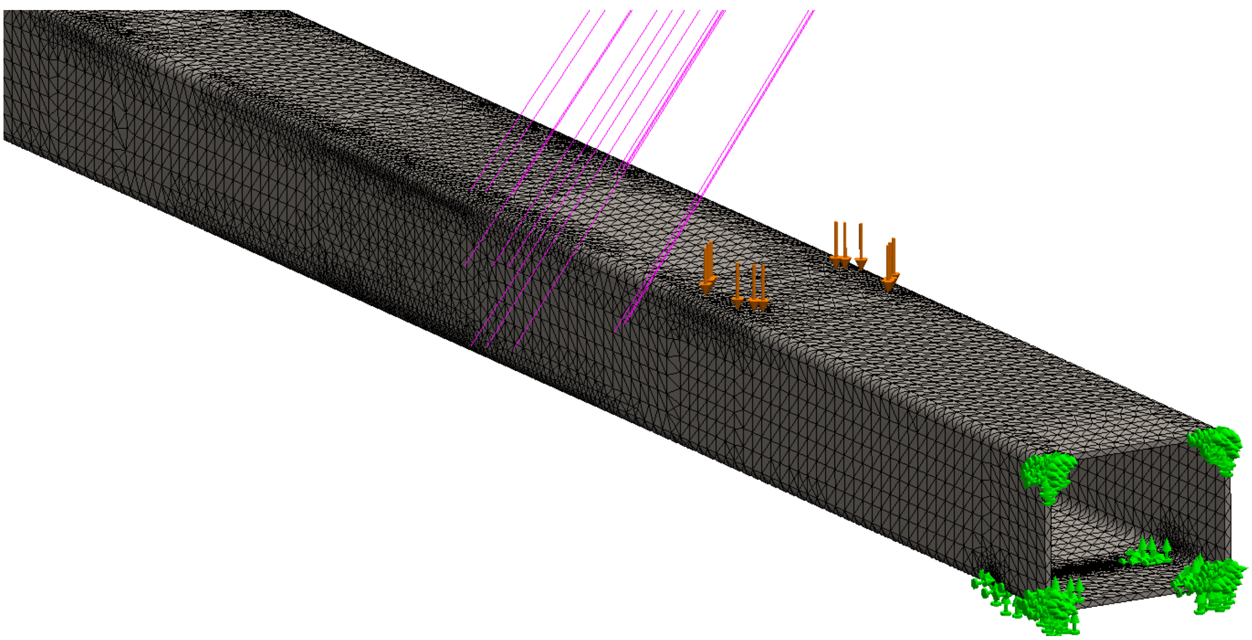

Figure E-54: Cross support scenario two meshing [3].

The elements are focused around the edges of the cross brace in the area of the remote load application. The resulting stress plot is shown in Figure E-55. As with the first loading scenario, a localized stress concentration occurred at the edge of the fixed geometry, with a value of 281 ksi. This value can be disregarded, as is occurs directly along the edge of the applied fixture and is highly localized. This stress concentration is shown in Figure E-56.

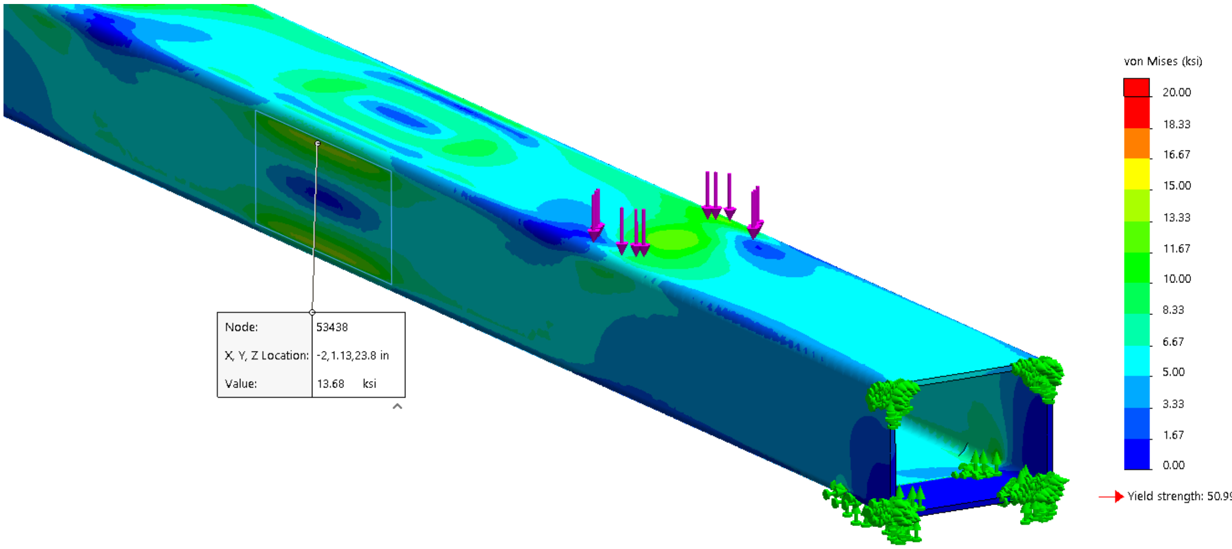

Figure E-55: Cross support scenario two stress plot [3].

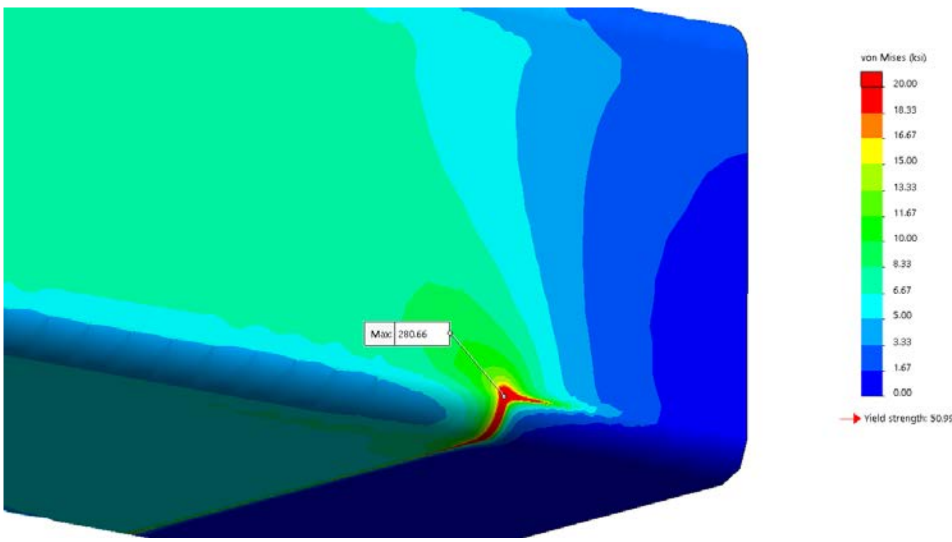

Figure E-56: Cross support scenario two stress concentration plot [3].

Ignoring this highly localized stress concentration, the peak stress value of 13.7 ksi occurred at the location of the upper roller support member, resulting in a factor of safety of 3.36.

1.1.11. Bottom Roller

In order to perform stress analysis on the Bottom Roller assemblies, a finite element model was made. This model combined the shaft, side plate and outer tube features into a single solid body. Creating the roller as a solid body eliminated the need for defining interactions between the components, thus reducing the possibility of error in the analysis.

Bearing fixtures were used to secure the roller, accurately representing the pillow block supports of the design. The fixtures allowed for self alignment of the shaft, meaning that angular misalignment is allowed at the bearing locations. A single bearing was also set to constrain rotation of the roller.

A load representing the weight of the roll of composite material was applied to a portion of the face of the roller using a split face. The magnitude of the load was set to one fourth of the weight of a full roll of material, 1075 lbs. Applying the load over a small area as opposed to a single line avoids infinite stresses at the point of load application and produces more accurate results. The fixtures and applies load are shown in Figure E-57.

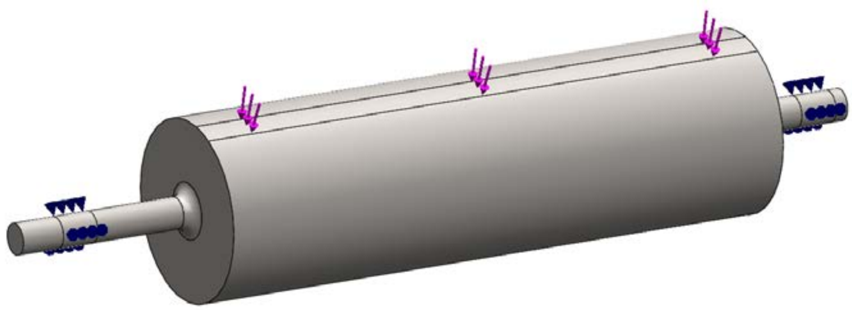

Figure E-57: Bottom roller FEA boundary conditions [3].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 56537 total notes, with the highest concentration of elements in the area of intersection of the shaft and side plate. The final mesh can be seen in Figure E-58, with the final von Mises stress plot shown in Figure E-59.

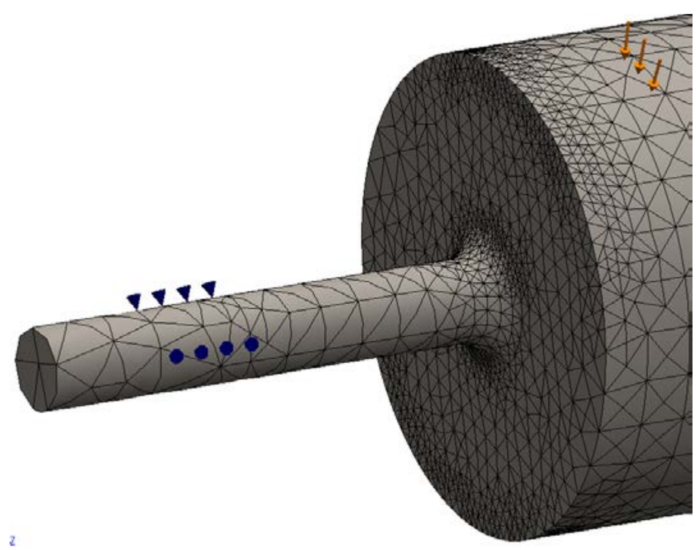

Figure E-58: Bottom roller meshing [3].

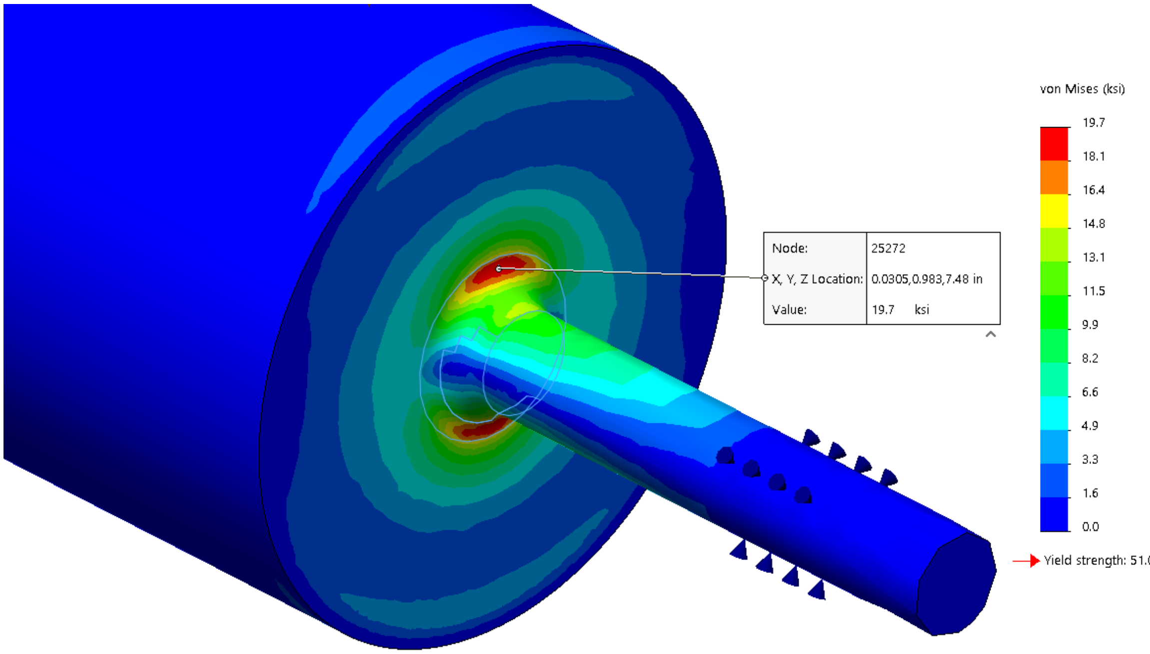

Figure E-59: Bottom roller stress plot [3].

As can be seen, the maximum stress value of 19.7 ksi occurred in the weld fillet between the shaft and side plate. This stress value results in a factor of safety of 2.94.

On the fourth iteration of mesh refinement, the resulting in a global accuracy of 93.5%. This can be seen in the mesh convergence plot, shown in Figure E-60. These results indicate that the results of this study are sufficiently accurate.

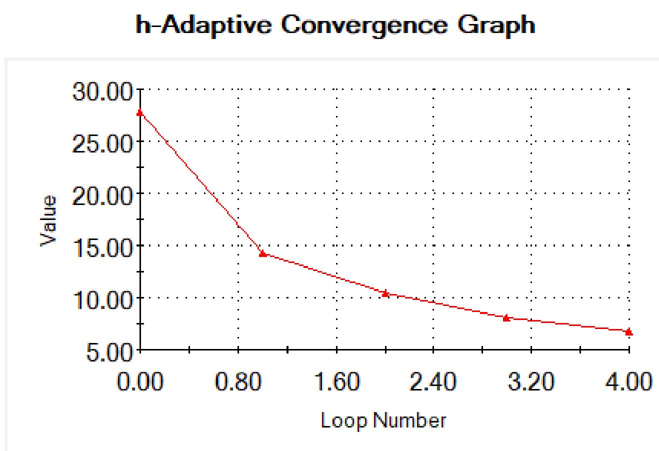

Figure E-60: Bottom roller h-adaptive convergence plot [3].

1.1.12. Walking Axle

For analysis of the Walking Axle, bearing fixtures were used at the location of the 0.5" mounting bolt. Loads were applied to split faces at the location of the caster mounting plate.

The load magnitude was set to 688 lbs per face, or one eighth of the estimated weight of the cart with a full roll of composite material (5500 lbs). The fixtures and applied loads are shown in Figure E-61.

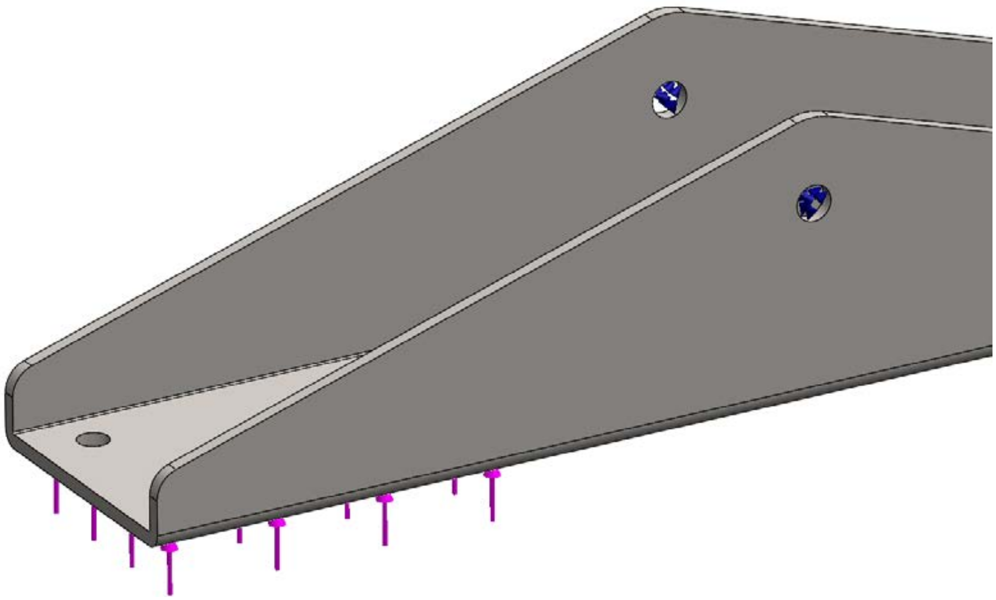

Figure E-61: Walking axle FEA boundary conditions [3].

After three iterations of mesh refinement, the study achieved a global accuracy of approximately 97.4%. The study convergence plot can be seen in Figure E-62.

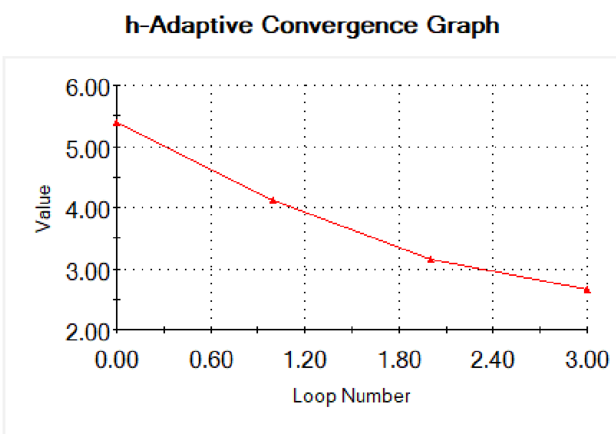

Figure E-62: Walking axle h-adaptive convergence plot [3].

The final mesh consisted of 141194 elements, with a minimum size of 0.177”. the final mesh is shown in Figure E-63 and Figure E-64. As can be seen, elements are concentrated around the bend radius on the two flanges, as well as around the holes for mounting the casters and the walking axle to the frame.

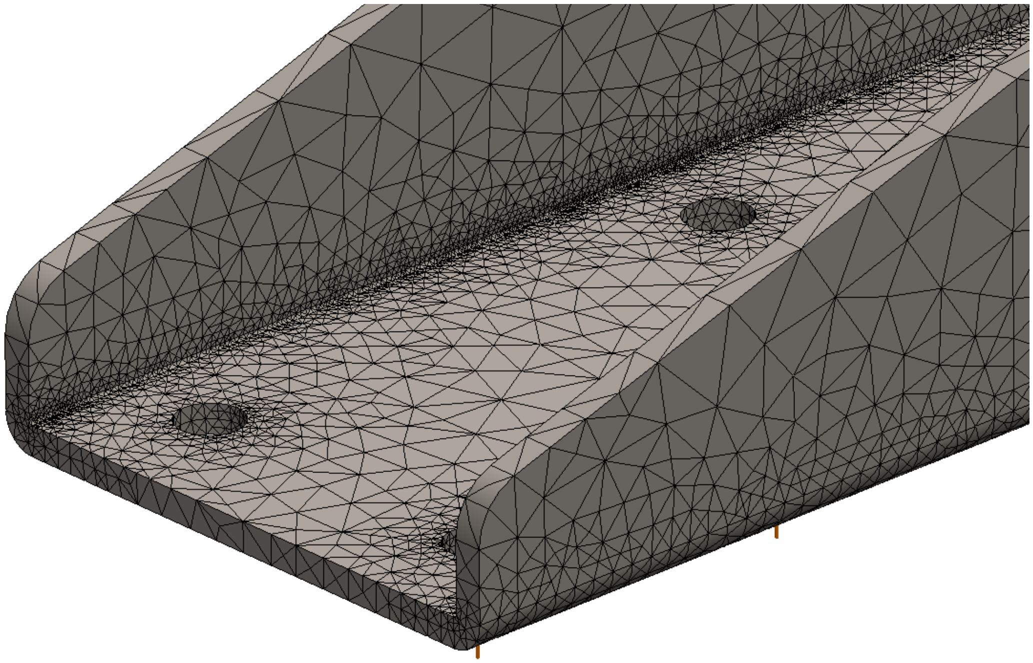

Figure E-63: Walking axle meshing (caster mount) [3].

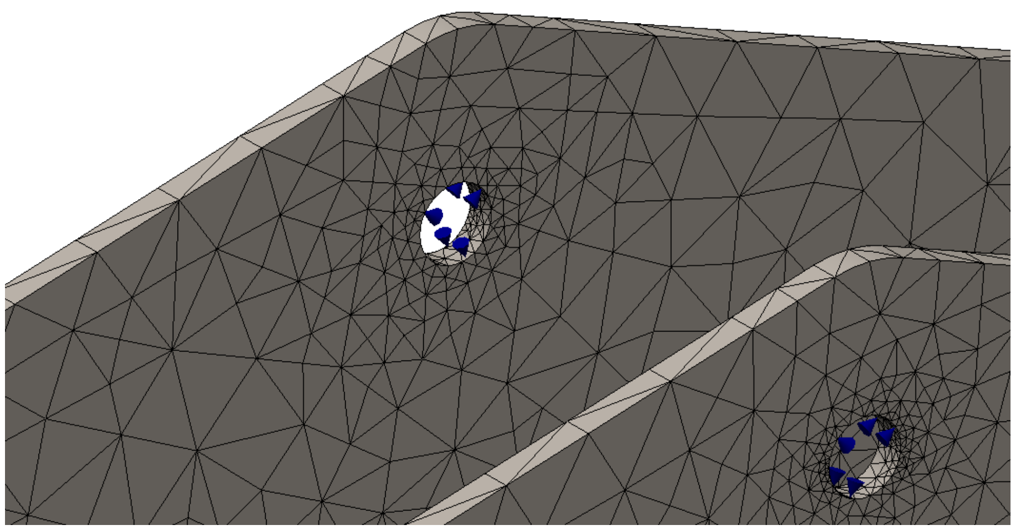

Figure E-64: Walking axle meshing (mounting bolt) [3].

The resulting stress plot is shown in Figure E-65. As can be seen, a maximum stress of 6.85 ksi occurred in the area of the caster mounting holes. It is expected that the highest stress would occur at a mounting hole, as a load is being applied at this location, and the hole causes a stress concentration. This peak stress value results in a factor of safety of 5.26.

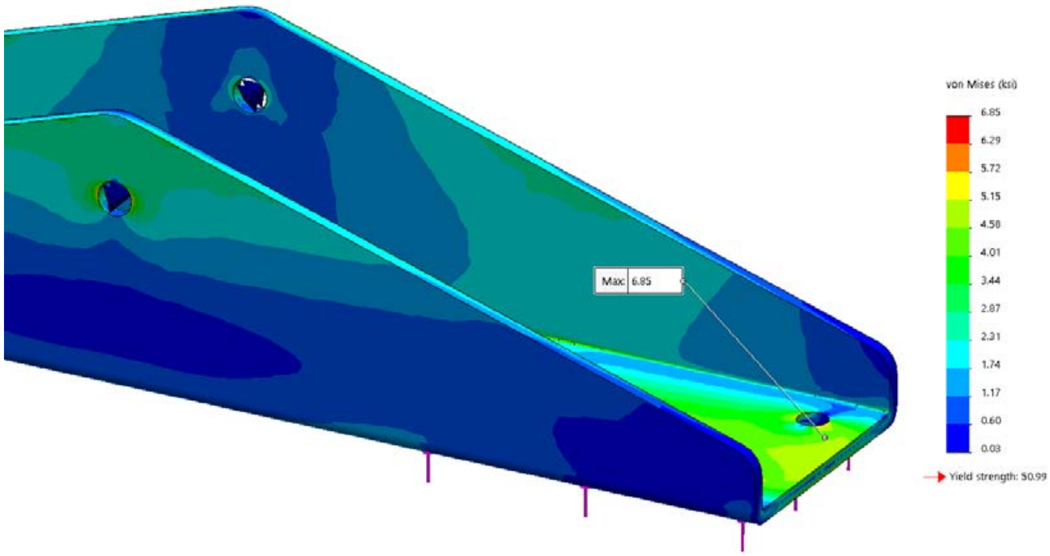

Figure E-65: Walking axle stress plot [3].

1.1.13. Retaining Shafts

For analysis of the retaining shaft assembly, a finite element model was made. This model incorporated the shafts, backing plate, gussets, and locking plate into a single solid body in order to eliminate the need for contact sets and to maximize the accuracy of the results.

A load was applied to a split face near the end of one shaft, in the direction of the second shaft. The magnitude of the load was set to one half the weight of a full roll of material (2150 lbs). Figure E-66 shows the load application.

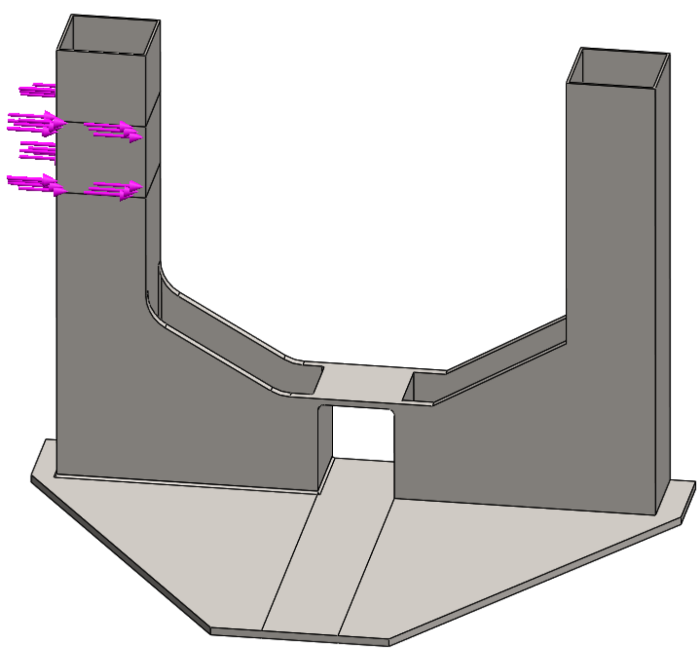

Figure E-66: Retaining shafts FEA loading scenario [3].

Virtual walls were used to simulate the presence of the Side Arm tubing support member, with the fixtures applied to the gussets, backing plate, and locking plate. Figure E-67 shows the faces on which virtual wall fixtures were applied (highlighted in blue).

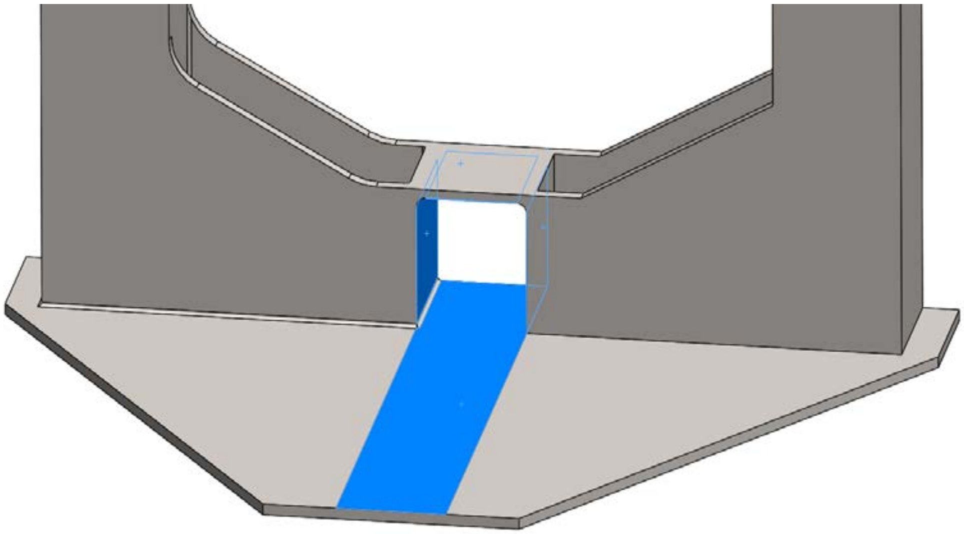

Figure E-67: Retaining shafts FEA boundary conditions [3].

After three iterations of mesh refinement, the study reached a global accuracy of approximately 90%. The study convergence plot is shown in Figure E-68.

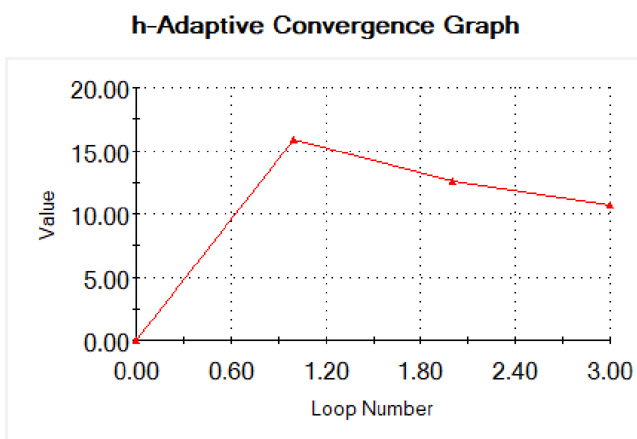

Figure E-68: Retaining shafts h-adaptive convergence plot [3].

The final mesh is shown in Figure E-69. As can be seen, the elements are concentrated at the intersection of the gussets and the shaft, as well as at the intersection of the gussets and the locking plate. The final mesh consisted of 33186 elements, with a minimum element size of 0.34”.

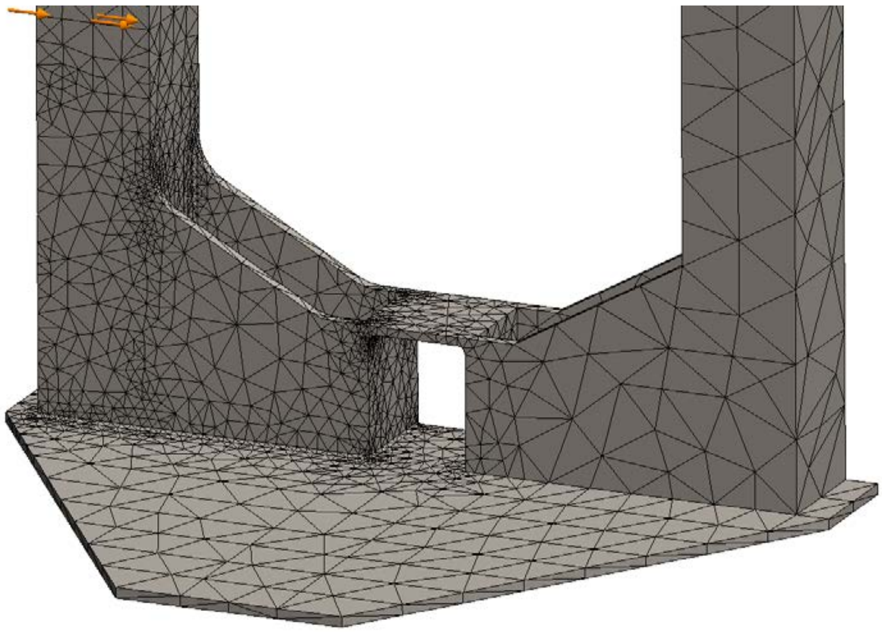

Figure E-69: Retaining shafts meshing [3].

Figure E-70 and Figure E-71 show the resulting stress plot, with a maximum stress of 32.9 ksi occurring at the edge of the virtual wall fixture on a gusset. Since this stress concentration is in the immediate vicinity of a fixture, it is ignored. The next highest stress value of 19.34 ksi occurred at the intersection of the gusset and the shaft, and results in a factor of safety of 3.0.

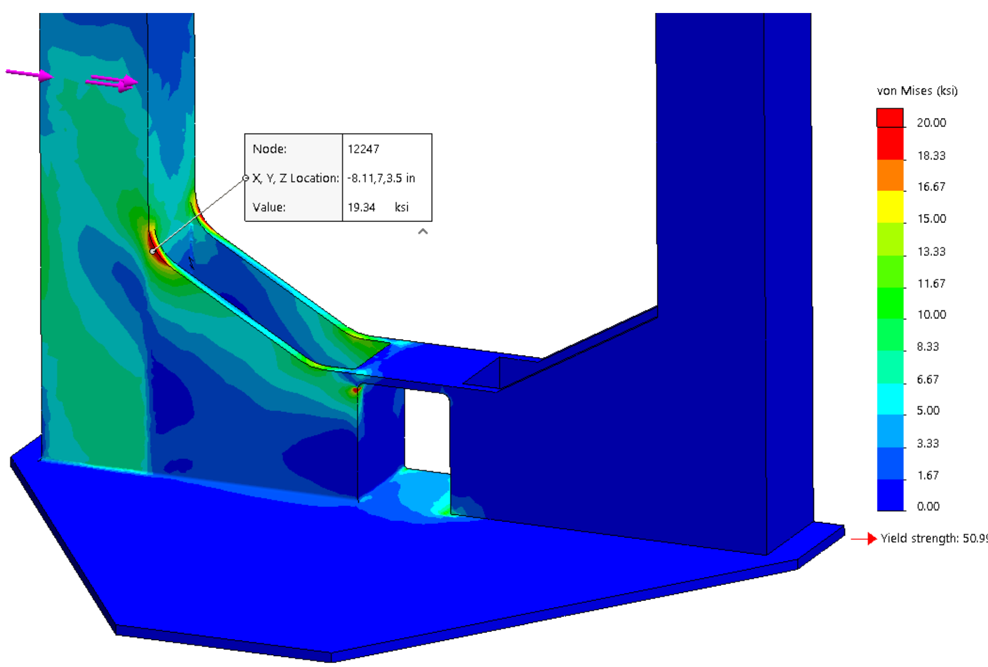

Figure E-70: Retaining shafts stress plot [3].

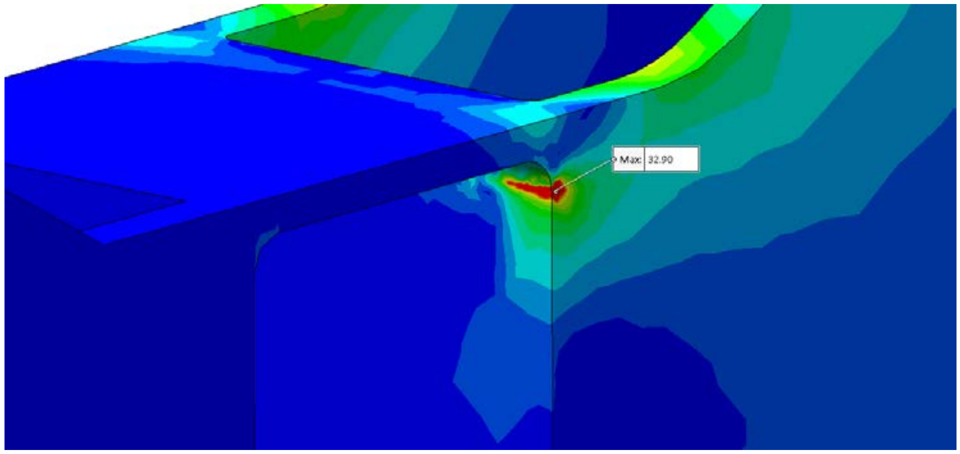

Figure E-71: Retaining shafts stress concentration plot [3].

It is important to note that the factor of safety of 3.0 was calculated using the ultimate tensile stress of the material. This calculation is valid due to the fact that the retaining shaft is treated as a one-use feature, meaning that the retaining shaft should be replaced in the event that the cart should be overturned, since the shaft is required to support the weight of the entire roll of material.

1.1.14. Handle Bars

The handle bar model was simply prepared for an FEA stress analysis by first splitting faces on each end of the rod by 4” and splitting a face on the middle of the rod by 3”. The 4” ends represent the approximate size of an average person’s hand, and the 3” split represents the area that will be welded to the side arm support tube. A total 125 lbs was applied to each outside end, representing a reasonable maximum force someone would apply while pushing the cart. One side of the 3” split face in the middle of the rod was applied fixed geometry fixture to represent that piece being welded. The applied forces and fixture are shown in Figure E-72.

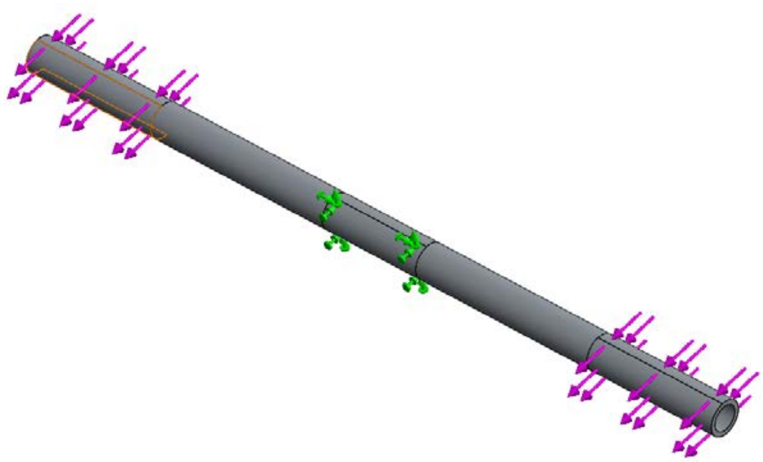

Figure E-72: Handle Bars FEA boundary conditions [2].

After five iterations of mesh refinement using the h-adaptive solver, the final mesh consisted of 16329 total elements, with a minimum element size of 0.215”. As can be seen in Figure E-73 and Figure E-74, the static study converged to 4.5% total relative strain energy with the majority of elements being located around area of the fixed geometry boundary condition.

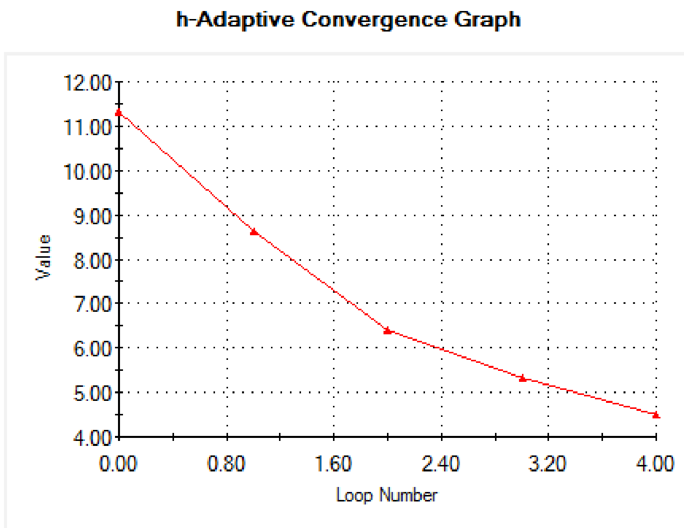

Figure E-73: Handle Bars h-adaptive convergence plot [2].

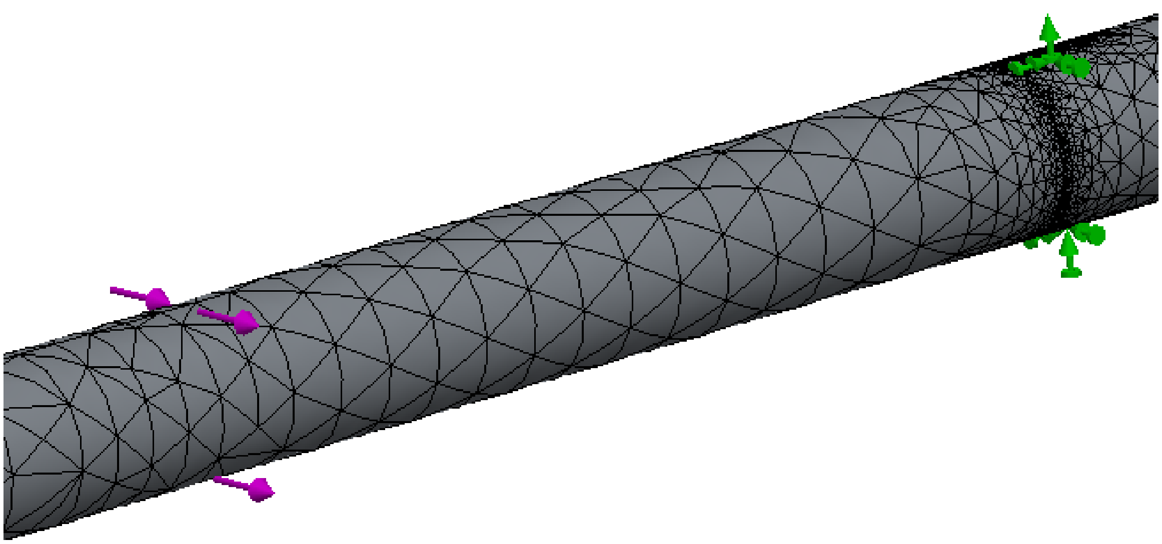

Figure E-74: Handle Bars meshing [2].

As seen in Figure E-76, the maximum stress of 30.63 ksi occurs in the model along the boundary of the fixed geometry. This stress is an unavoidable diverging stress concentration due to the setup of the study. By ignore the concentration, Figure E-75 shows that a more realistic maximum stress within the model. This stress occurs on the front face of the tubing, and is approximately 7 ksi.

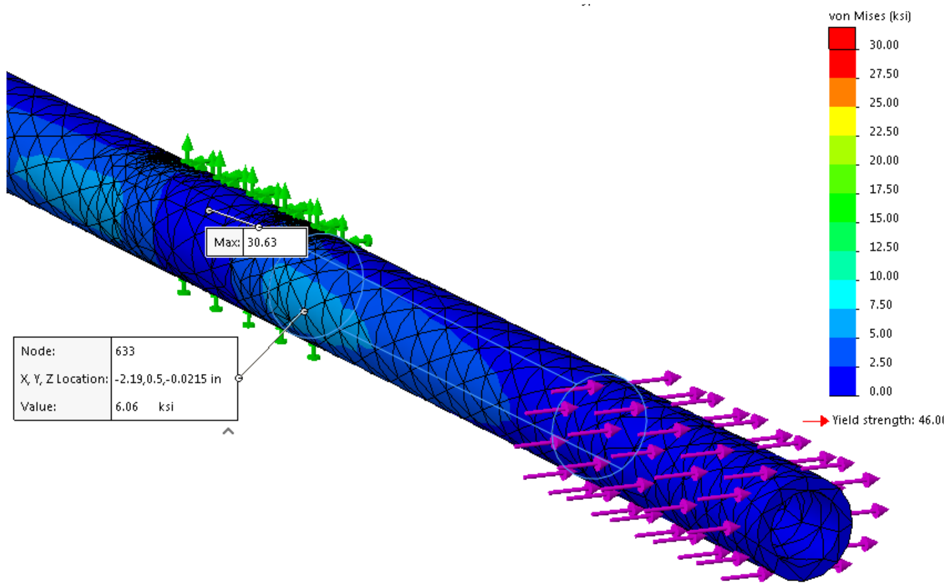

Figure E-75: Handle Bars stress plot [2].

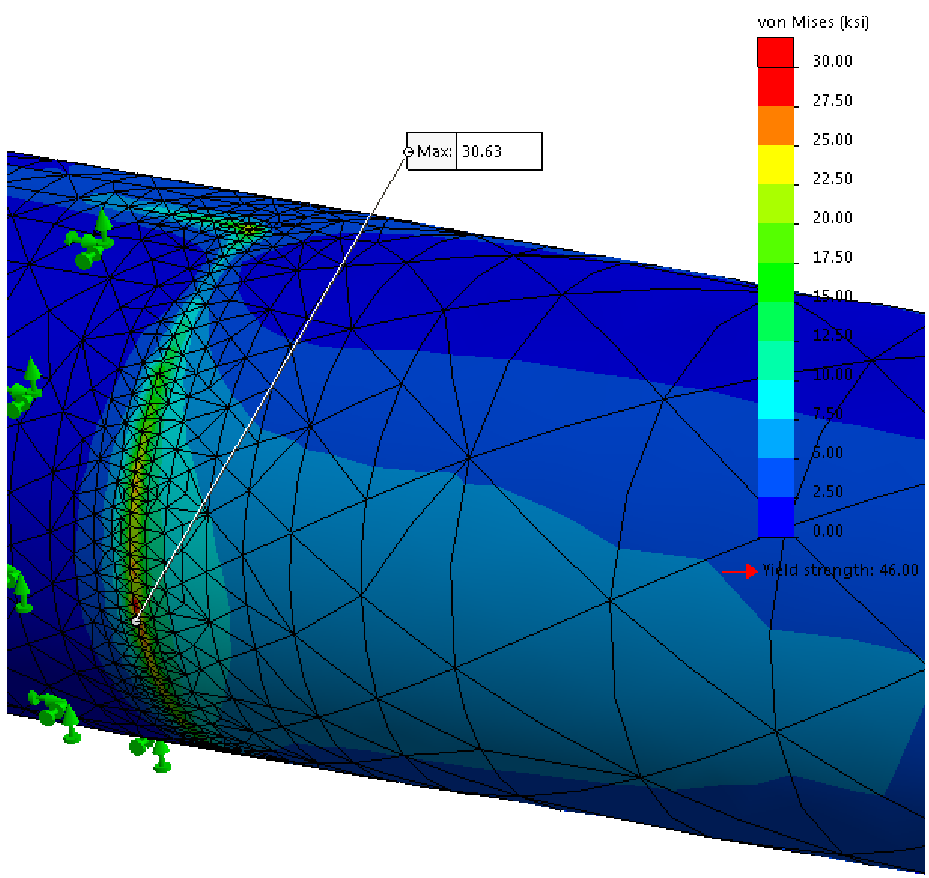

Figure E-76: Handle Bars stress concentration plot [2].

To verify the maximum stress of 7 ksi, the handle bar can be thought of as a cantilever beam if cut at the end of the fixed geometry. Using the applied load of on only one half of the rod multiplied by the distance to the fixed geometer, a maximum moment can be found. By applying and solving Eq.'s (E-3) and (E-4) below, a maximum stress due to bending can be found.

$$Max\ Moment = Force * Distance = \left(\frac{125}{2}\right)(7.5) = \ 437.5\ lbs\ in \tag{E-3}$$

$$\sigma = \frac{Mr_0}{I} \tag{E-4}$$

Where σ is the maximum normal stress in the bar, M is the maximum moment on the bar, r_0 is the outside radius of the bar, and I is the moment of inertia of the bar. For a round hollow bar, the moment of inertia I can be calculated as seen in Eq. (E-5), where t is the thickness of the hollow bar.

$$I_{hollow\ circle} = \frac{\pi}{2}(r_o^4 - (r_o - t)^4) = \frac{\pi}{2}(0.5^4 - (0.5 - 0.133)^4) = 0.035in^4 \tag{E-5}$$

$$\sigma = \frac{(437.5)(0.5)}{0.035} = 6278\,psi = \boldsymbol{6.3\ ksi} \tag{E-6}$$

Thus, resulting in approximately 10% relative error with the FEA results and validating the study. Therefore, the stress study results in a factor of safety of 6.57.

1.1.15. Bus Wheel Connector Tube

The stress through the Bus Wheel due to applied torsion was analyzed. Due to the sourced shaft coupler specified already with a large factor of safety (maximum rated load of 6310 lbs-in), and the outer-ring of the Bus Wheel theoretically carrying close to zero stress, only the connector tube was considered.

To analyze the connector tube, the torque applied on the outside of the connector tube needed to be simulated. By assuming the maximum pulling force of the operator to be 125 lbs, and dividing it by four (due to four connector tubes equally sharing the applied torque) the resultant force to apply is calculated as 31.25 lbs. Thus, the force was applied to the end of the connector tube that is to be welded to the outer-ring of the Bus Wheel. A fixed geometry was set on the opposite face of the connector tube, thus simulating the portion of the tubing welded to the shaft coupler. The boundary conditions are shown in Figure E-77.

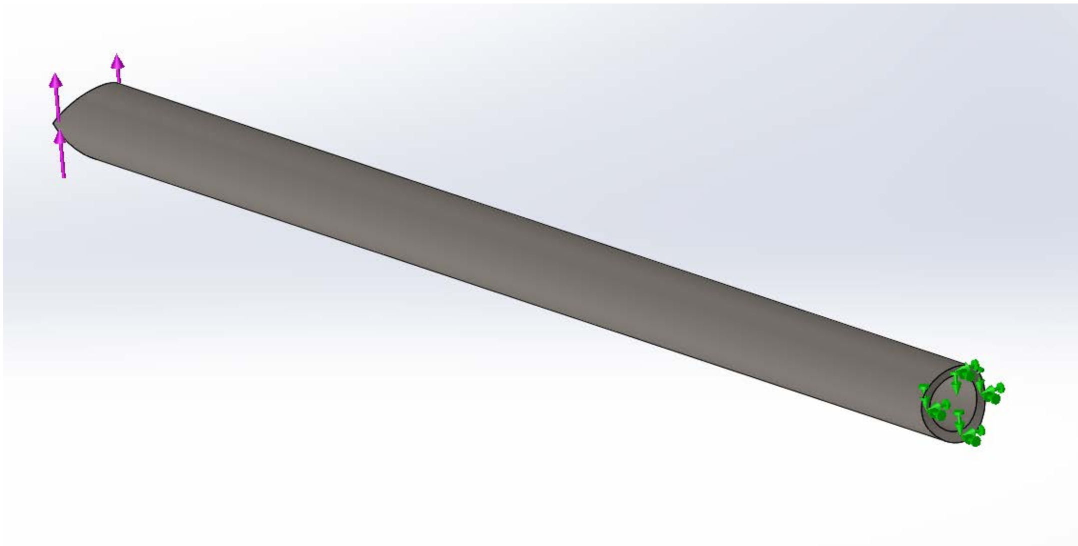

Figure E-77: Bus Wheel connector tube FEA boundary conditions [4].

After five iterations of mesh refinement using the h-adaptive solver, the static study converged to total relative strain energy less than 0.91% with the majority of elements being located around area of the fixed geometry boundary condition. The convergence plot is shown in Figure E-78. The final mesh produced is shown in Figure E-79.

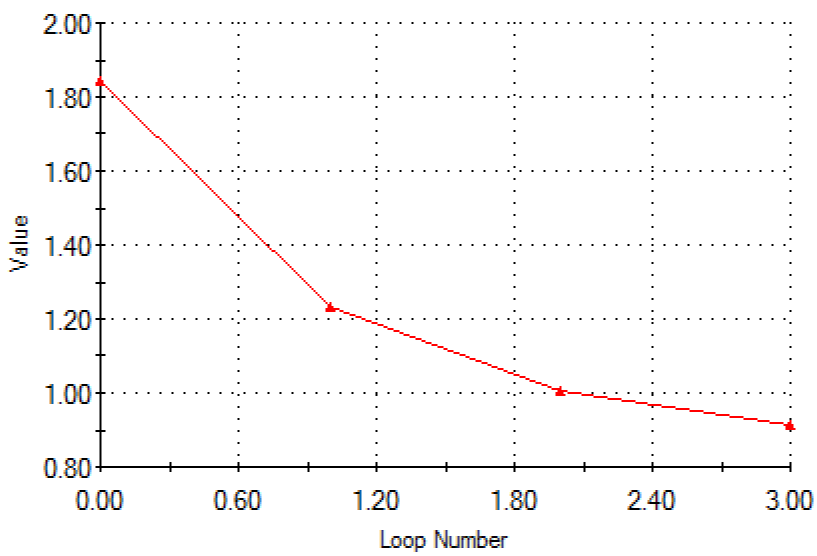

Figure E-78: Bus Wheel connector tube h-adaptive convergence plot [4].

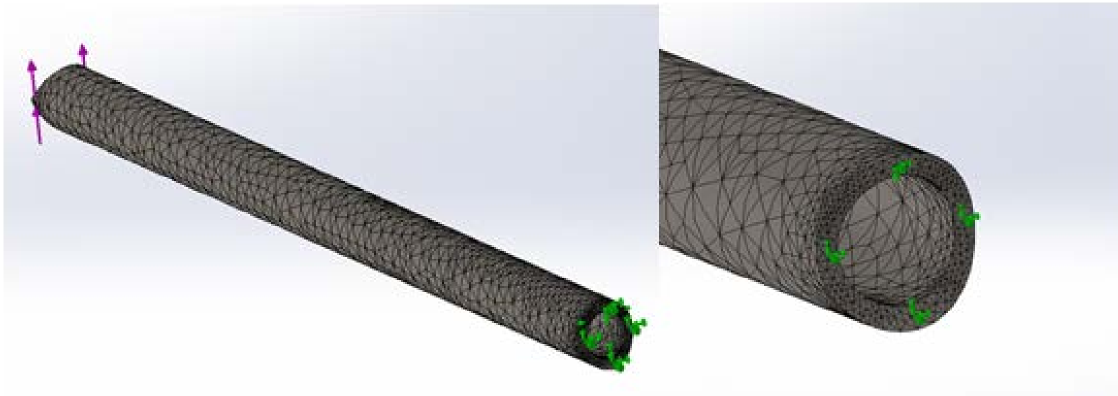

Figure E-79: Bus Wheel connector tube meshing [4].

From the result of the study, the maximum stress of 17.28 ksi occurs in the model along the boundary of the fixed geometry. This stress is an unavoidable diverging stress concentration due to the fixed boundary condition applied in the study. Ignore the concentration, a more realistic maximum stress within the model happens along the top and bottom face at the fixed end of the tube of approximately 15.7 ksi, as shown in Figure E-80 and Figure E-81. This results in a resultant factor of safety of 2.93.

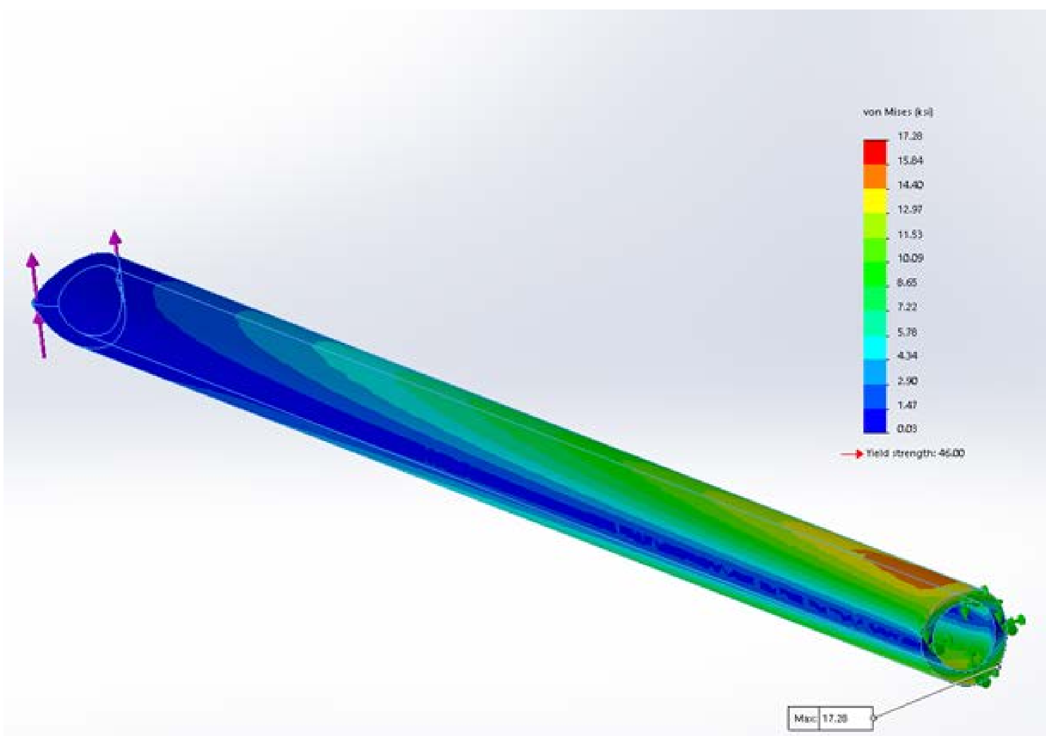

Figure E-80: Bus Wheel connector tube stress plot [4].

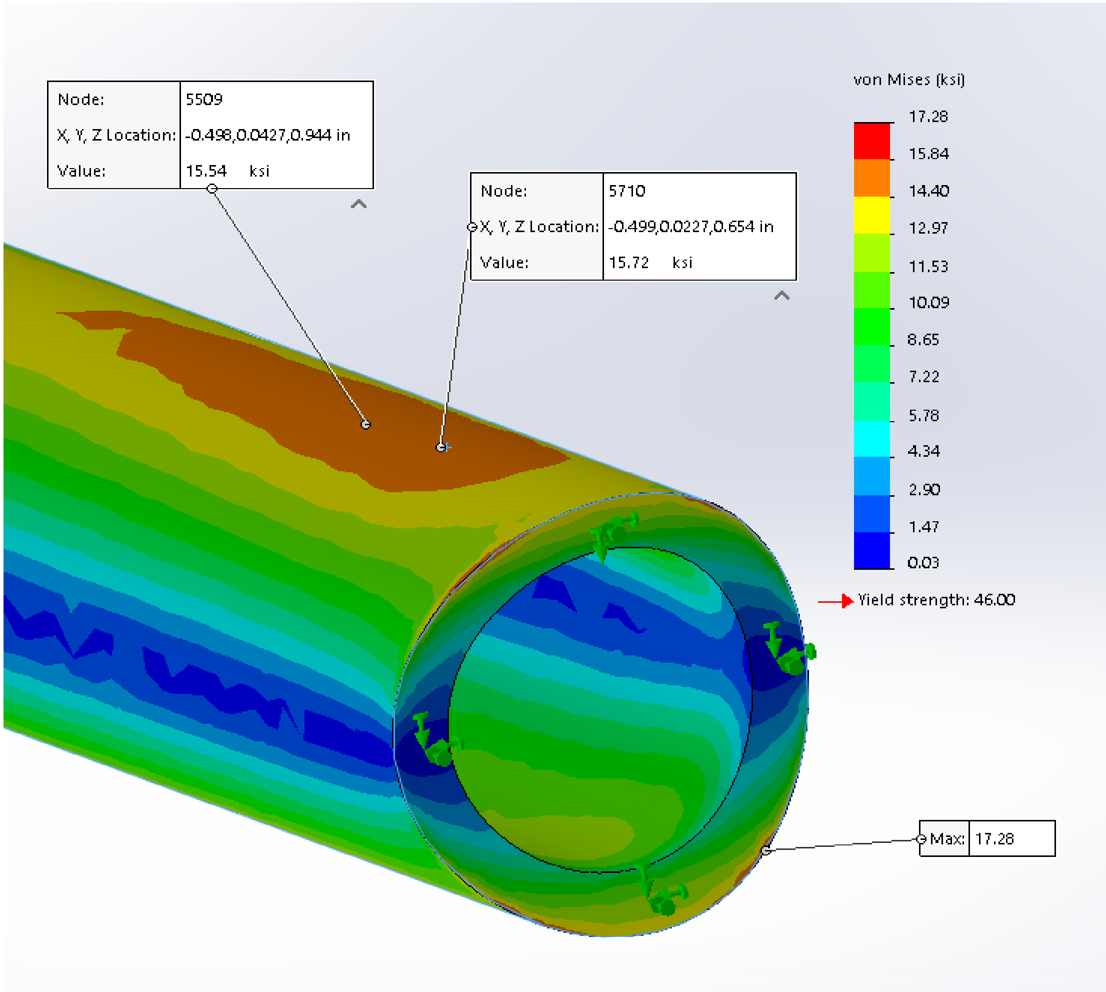

Figure E-81: Bus Wheel connector tube stress concentration plot [4].

1.2. Weld Sizing and Stresses

Due to the large amount of welded parts within the team's design, it is necessary to perform an in-depth stress analysis of each welded part to determine proper weld sizing such that parts are safe and while still minimizing the required input labor of welding. The team focused on two different approaches for sizing welds: First, using a simplified analytical approach for simple weld patterns in bending conditions. Second, using FEA to determine minimum fillet sizes for complex loading and weld patterns that cannot easily be determined analytically. For all welds through analytical analysis, a E60 strength class electrode was selected due to its rated yield strength of 50 ksi being the closest to that of the metal parts used throughout the design. Additionally, E60 is considered the bare minimum strength of a steel based electrode that is generally considered acceptable [5]. For all weld calculations, a minimum factor of safety of 3 was used, correlating to a maximum allowable normal stress 16.67 ksi or approximately 10 ksi for pure shearing modes. In cases where the calculated weld sizes were small, they were replaced by the minimum allowable weld size outlined by the AWS D1.1 standard [6]. As with other stress analyzes, it is important to note that the results presented in this report are preliminary, and should not be considered an exact representation of the actual design. Further analysis is required to verify the results shown in this report.

Weld Stress in Bending

A fillet weld is considered in the state of bending when all parts of the weld are aligned along the same plane, perpendicular to a force or moment applied at some constant distance away. By treating the part welded as a cantilever, a resultant shear force and moment reaction can be found at the fixed side of the beam. These forces can be used to estimate the shear stresses within each weld. Figure E-82 displays an example of the loading conditions to treat the welds in bending.

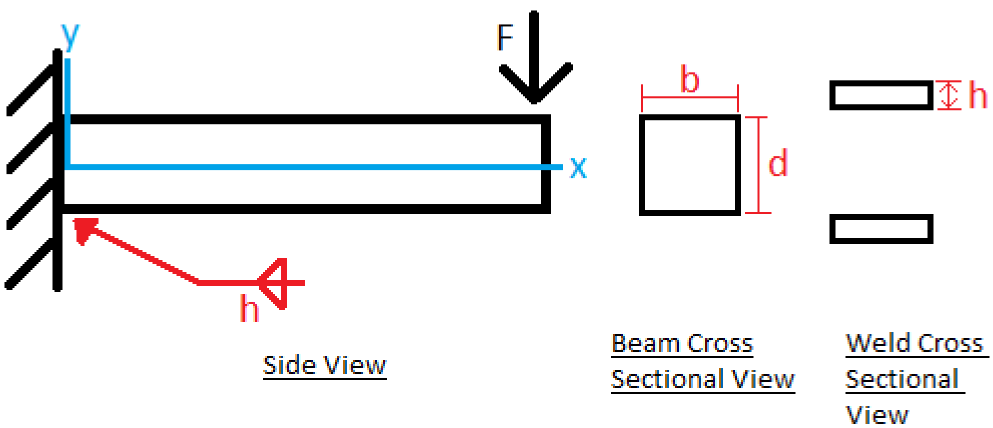

Figure E-82: Welds in bending example setup, redrawn from [5], [7].

When welds are loaded in bending, a simple approach can be used to determine the stresses in the weld, similar to simple bending stresses of a beam. Shear stresses in welds consist of two separate loads: shear force and bending moment that can be calculated by Eq.'s (E-7) and (E-8) below [5].

$$\tau' = F/A \tag{E-7}$$

$$\tau'' = \frac{Mc}{I} \tag{E-8}$$

Where τ' is the shear stress due to shear force, τ'' is the shear stress due to bending moment, *F* is an applied shear force, *A* is the throat area of a weld, *M* is the bending moment about the weld, *c* is distance from the center of gravity of the weld to the highest or lowest of the actual weld, and *I* is the moment of inertia of the weld pattern. The area of the throat of a weld can be calculated by 0.707*h multiplied by the total length of welds, where h is the leg size of that weld. The moment of inertia of a weld pattern can be calculated by 0.707*h multiplied by a unit moment of inertia of a weld pattern. Unit moment of inertias for welds are readily calculated and published by many sources such that they may be directly applied. For the team's design, only four different weld patterns were used: rectangular all-around welds, circular all-around welds, double-line (top and bottom) welds and C-channel (top, bottom, and one side) welds. The unit moment of inertia for these patterns are listed below [5]:

$$I_{u,C-Channel} = \frac{d^2}{12}(6b + d) \tag{E-9}$$

$$I_{u,rectangle} = \frac{d^2}{6}(3b + d) \tag{E-10}$$

$$I_{u,double-line} = \frac{bd^2}{2} \tag{E-11}$$

$$I_{u,circle} = 2\pi r^3 \tag{E-12}$$

Where *b* is the base length of any weld, *d* is the height length of any weld (or horizonal distance between base length of weld in the double-line configuration) and *r* is the radius of a circular weld. Once both τ' and τ'' for a weld have been calculated, their resultant can be found by Eq. (E-13), and can be directly compared to the allowable shear strength for pure shearing.

$$\tau = \sqrt{\tau'^2 + \tau''^2} \tag{E-13}$$

Welds in Axial Loading

Normal stress for a weld in axial loading can be simply computed by Eq. (E-14) [5].

$$\sigma = F_z/A \tag{E-14}$$

Where σ is the normal stress in the weld, F_z is an applied axial load, and *A* is the throat area of the total weld. Normal weld stress may be directly checked via allowable yield stress of a particular weld electrode under pure axial loading.

Combination of Shear and Normal Weld Stress

In scenarios where welds are subject to a combination of both shear and normal stresses, both must be considered. Thus, a failure criterion such as von Mises is appropriate to use due to the ductile nature of most welds. The resultant stress in a weld experiencing both shear and normal stress, as presented above can be combined using Eq. (E-15) [5].

$$\sigma_{equivalent} = \sqrt{\sigma^2 + 3\tau^2} \tag{E-15}$$

This equivalent stress may be directly compared to allowable normal stress.

1.2.1. Analytical Weld Stress and Sizing Results

For the purposes of all parts designed with welds, a spreadsheet solver was developed such that specific weld parameters could be inputted quickly, and the required weld size could be automatically solved for. The following sections list the inputs, displays a figure of the part being welded with weld length dimensions outlined in red, and display the results of minimum weld size needed. The Excel spreadsheet used to solve for weld sizes will also be provided to the client to review and use moving forward with the project.

1.2.2. Top Rollers Shell

Inputs:

Shear Force: 63 lbs

Bending Moment: 1055.25 lbs-in

Weld Pattern: Circular

Weld radius, r: 3.0325"

Results:

Estimated Weld Size, h: 0.01", therefore use: **3/16"** (minimum as per standard)

$\tau_{resultant}$ = 2625 psi

1.2.3. Top Rollers Shaft

Inputs:

Shear Force: 63 lbs

Bending Moment: 1055.25 lbs-in

Weld Pattern: Circular

Weld radius, r: 0.375"

Results:

Estimated Weld Size, h: 0.17", therefore use: **3/16"** (minimum as per standard)

$\tau_{resultant}$ = 9939 psi

Figure E-83: Top Roller Shell and Shaft weld area [7].

1.2.4. Hanging Plate

Inputs:

Shear Force: 125 lbs

Bending Moment: 1417.5 lbs-in

Weld Pattern: Double-line

Weld height, d: 4.62”

Weld width, b: 2“

Results:

Estimated Weld Size, h: 0.02”, therefore use: **1/8”** (minimum as per standard)

$\tau_{resultant}$ = 6166 psi

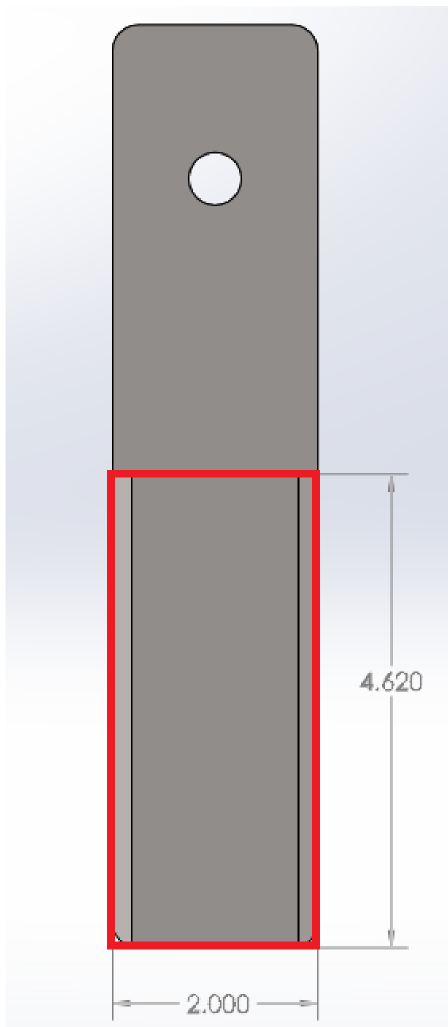

Figure E-84: Hanging Plate weld area [7].

1.2.5. Side Arm Bracket

Inputs:

Shear Force: 1500 lbs

Bending Moment: 6750 lbs-in

Weld Pattern: C-channel

Weld height, d: 5.33”

Weld width, b: 0.5“

Results:

Estimated Weld Size, h: 0.14”, therefore use: **1/4”** (minimum as per standard)

$\tau_{resultant}$ = 9522 psi

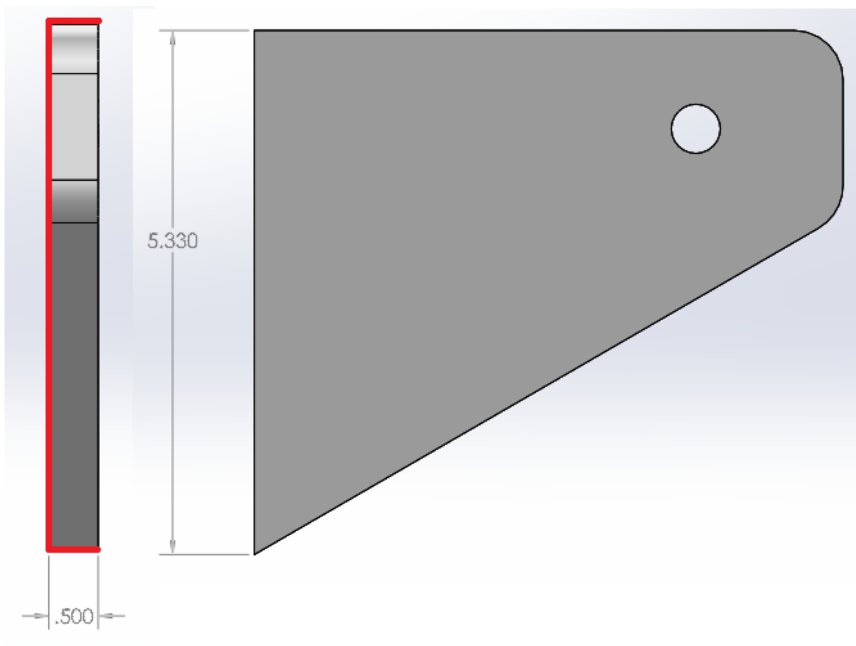

Figure E-85: Side Arm Bracket weld area [7].

1.2.6. Swinging Arm Top Bracket

Inputs:

Axial Force: 1500 lbs

Weld Pattern: C-channel

Weld height, d: 0.5”

Weld width, b: 5.209“

Results:

Estimated Weld Size, h: 0.03”, therefore use: **3/16”** (minimum as per standard)

σ_{normal} = 11390 psi

Figure E-86: Swinging Arm Top Bracket weld area [7].

1.2.7. Side Arm Tubing

Inputs:

Shear Force: 0 lbs

Bending Moment: 18750 lbs-in

Axial Force: 5715 lbs

Weld Pattern: Rectangle

Weld height, d: 4”

Weld width, b: 3“

Results:

Estimated Weld Size, h: 0.17”, therefore use: **1/4”**

$\tau_{resultant}$ = 9000 psi

σ_{normal} = 3396 psi

$\boldsymbol{\sigma_{equivalent}}$ **= 15954 psi**

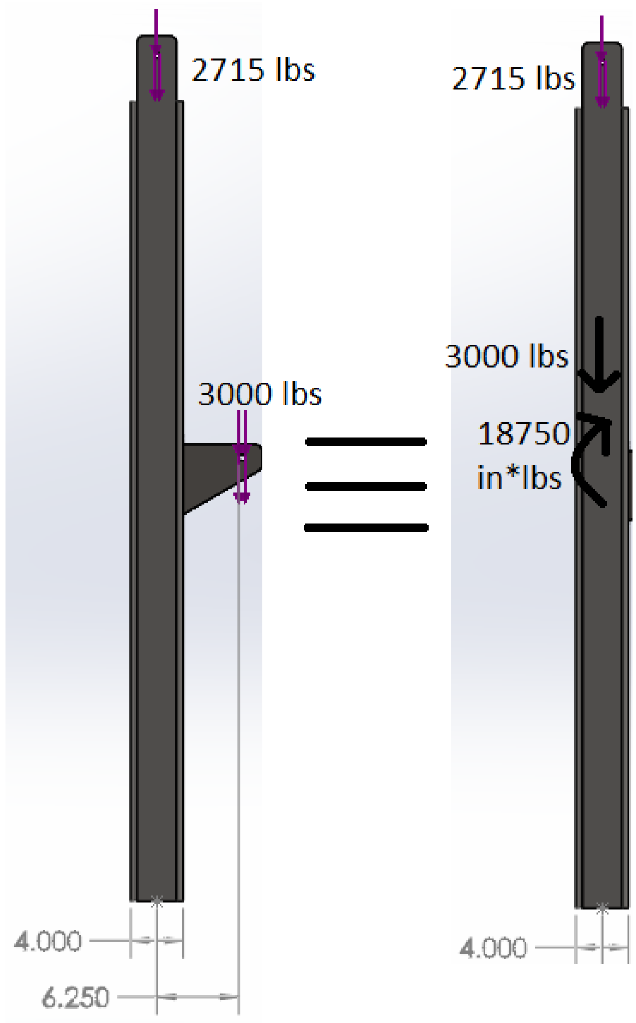

Figure E-87: Side Arm Tubing equivalent forces [7].

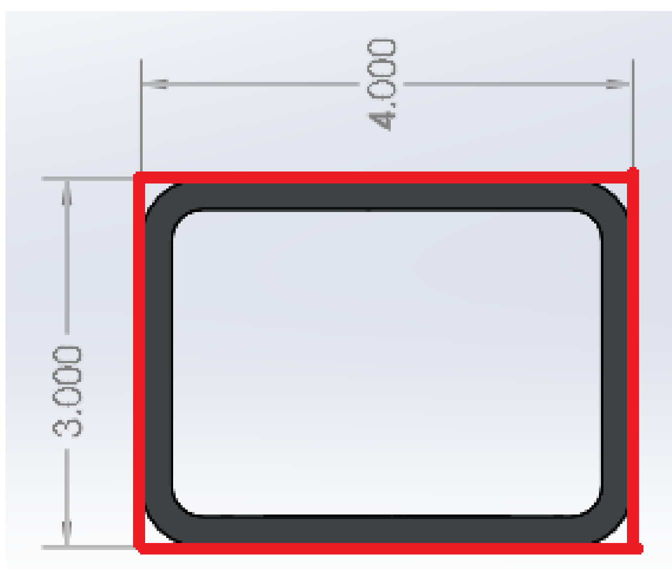

Figure E-88: Side Arm Tubing weld area [7].

1.2.8. Welded Side Arm

Inputs:

Shear Force: 2150 lbs

Bending Moment: 53750 lbs-in

Weld Pattern: Rectangle

Weld height, d: 3”

Weld width, b: 4“

Results:

Estimated Weld Size, h: 0.51”, therefore use: **9/16”**

$\tau_{resultant}$ = 9947 psi

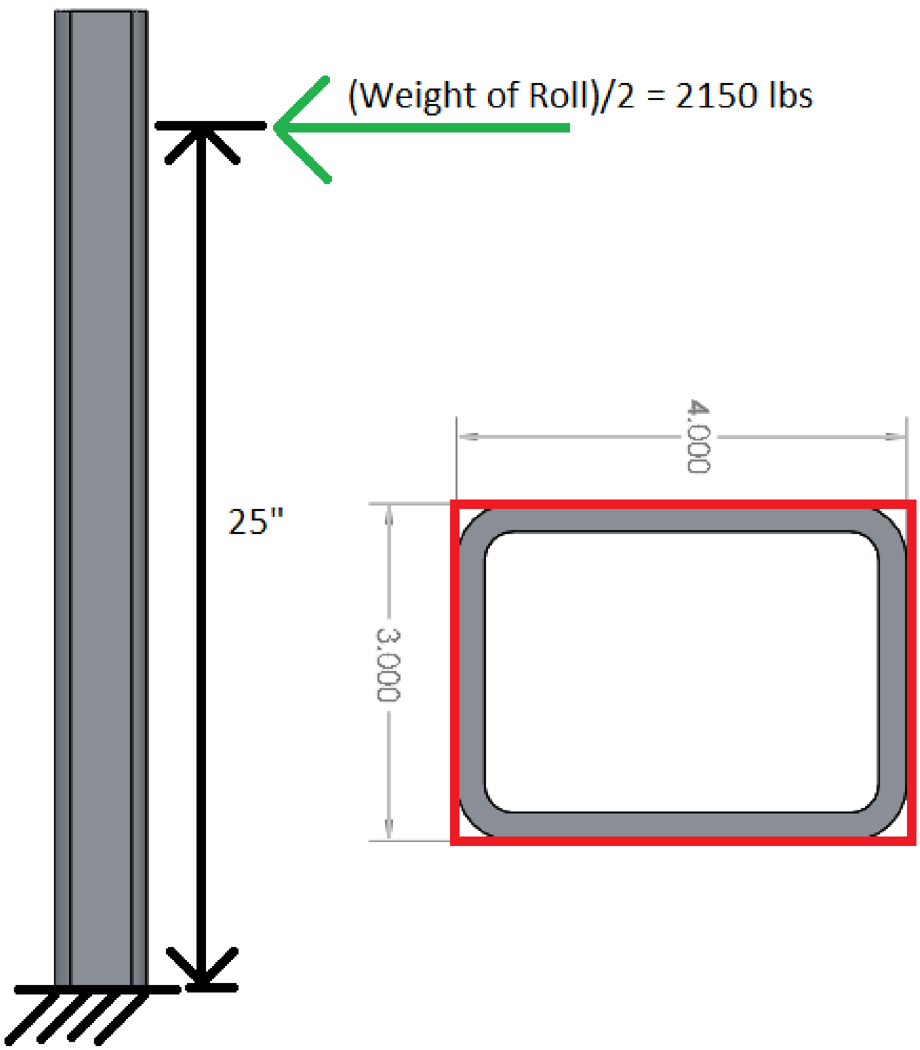

Figure E-89: Welded Side Arm loading conditions and weld area [7].

1.2.9. Bottom Roller Shell

Inputs:

Shear Force: 1146.25 lbs

Bending Moment: 13755 lbs-in

Weld Pattern: Circular

Weld radius, r: 3.0325"

Results:

Estimated Weld Size, h: 0.04", therefore use: **3/16"** (minimum as per standard)

$\tau_{resultant}$ = 8682 psi

1.2.10. Bottom Roller Shaft

Inputs:

Shear Force: 1146.25 lbs

Bending Moment: 13755 lbs-in

Weld Pattern: Circular

Weld radius, r: 0.625"

Results:

Estimated Weld Size, h: 0.80", therefore use: **1"**

$\tau_{resultant}$ = 9922 psi

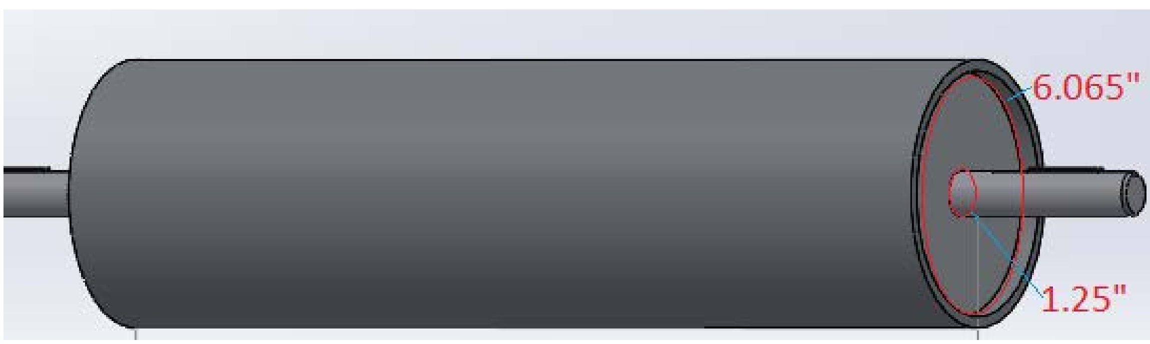

Figure E-90: Bottom Rollers Shell and Shaft weld area [7].

1.2.11. Handle Bar

Inputs:

Shear Force: 62.5 lbs

Bending Moment: 687.5 lbs-in

Weld Pattern: Double-line

Weld height, d: 1"

Weld width, b: 1.5“

Results:

Estimated Weld Size, h: 0.07”, therefore use: **1/8”** (minimum as per standard)

$\tau_{resultant}$ = 9271 psi

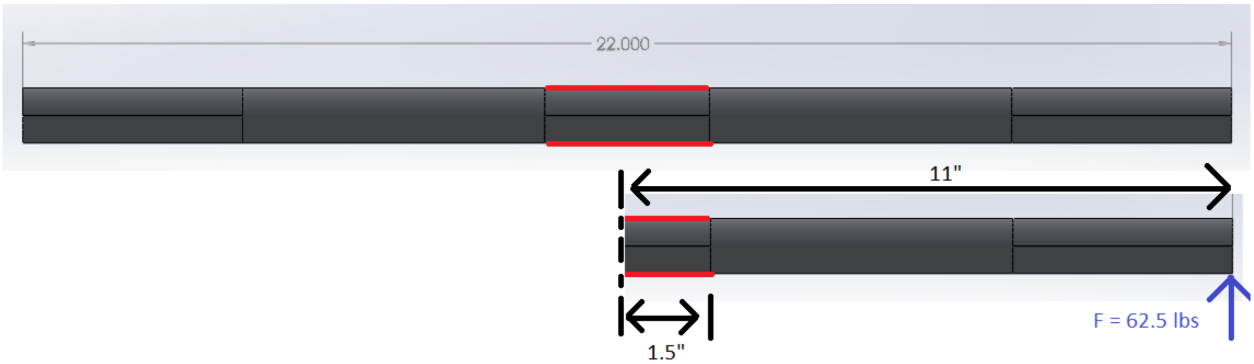

Figure E-91: Handle Bar welding area [7].

1.2.12. Bus Wheel

Inputs:

Shear Force: 31.25 lbs

Bending Moment: 593.75 lbs-in

Weld Pattern: Circular

Weld radius, r: 0.5”

Results:

Estimated Weld Size, h: 0.06”, therefore use: **3/16”** (minimum as per standard)

$\tau_{resultant}$ = 8914 psi

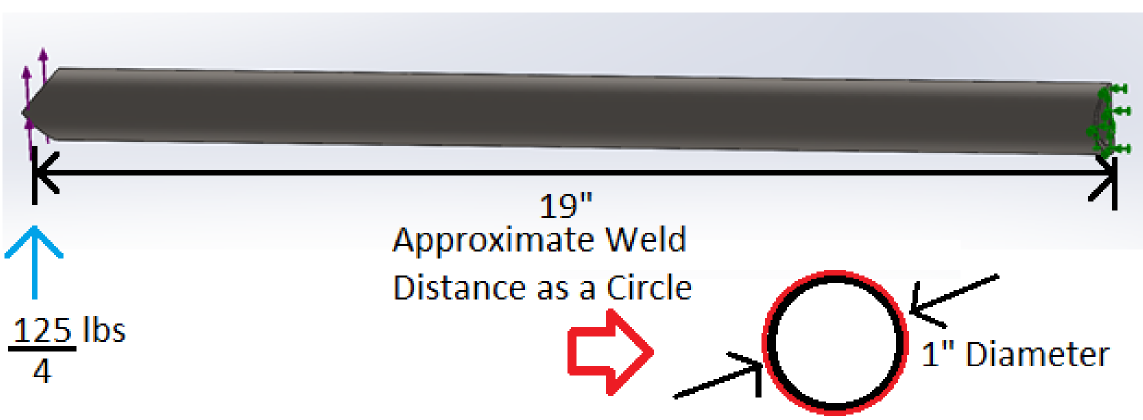

Figure E-92: Bus Wheel welding area [7].

1.3. Numerical Weld Stress and Sizing Results

Due to their complex geometry, the analytical analysis of certain welded joints was deemed not feasible. For such situations, analysis of stress in welds was performed via numerical simulation. All analyses were performed using the simulation package in SolidWorks. Analyses were performed using a finite element model representation of specific welded assemblies, combining the welded bodies into single solid body in order to eliminate contact sets in the analysis. Where necessary, the welded area of the assembly was isolated to reduce necessary computing power and increase the level of accuracy as high as possible. Fillets were used to represent the welds, with radii equal to the desired leg size used. Von Mises stresses were used to evaluate the stress levels in the welded locations, which were used to determine the strength of weld electrode required for the particular assembly. For all analyses, a factor of safety of 3.0 was targeted.

Again, h-adaptive studies were used to refine the mesh used in the weld analysis. Convergence plots were used where possible to prove the validity of the study.

1.3.1. Safety Bracket

The finite element model for the analysis of the welds on the safety bracket consisted of a section of the Side Arm and a single safety bracket. Welds (fillets) were created along the top and bottom edge of the safety bracket, as well as the inside edge (towards the center of the cart). Weld size was set to 0.25", equal to the thickness of the Side Arm. The locations of welds are shown in Figure E-93.

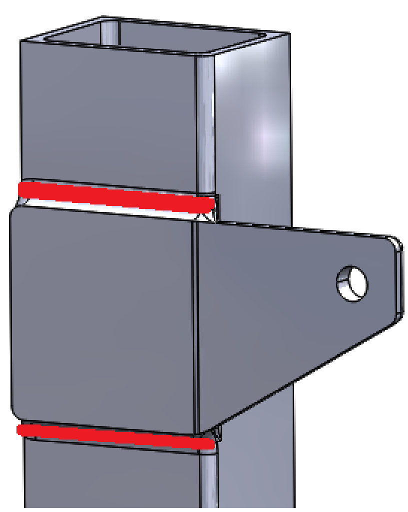

Figure E-93: Safety bracket weld location [3].

The bottom surface of the Side Arm was fixed using a fixed geometry boundary condition. A vertical load of 330 lbs was applied to the safety pin hole in the Safety Bracket, simulating the weight of the Top Roller subassembly resting on the safety pin. Figure E-94 shows the loads and fixtures applied for the study.

Figure E-94: Safety Bracket weld FEA boundary conditions [3].

After three iterations of mesh refinement, the study converged with approximately 98% global accuracy. The final mesh consisted of 66467 elements, with a minimum element size of 0.132”. The h-adaptive graph and final mesh can be seen in Figure E-95 and Figure E-96 respectively.

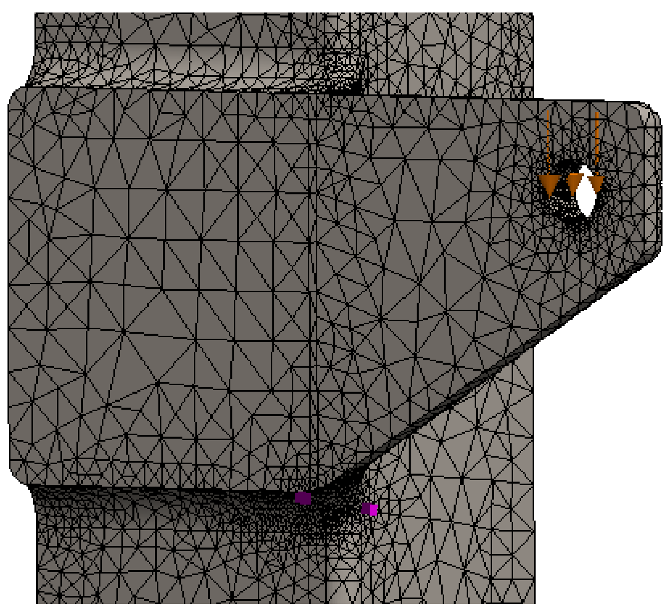

Figure E-95: Safety Bracket weld FEA meshing [3].

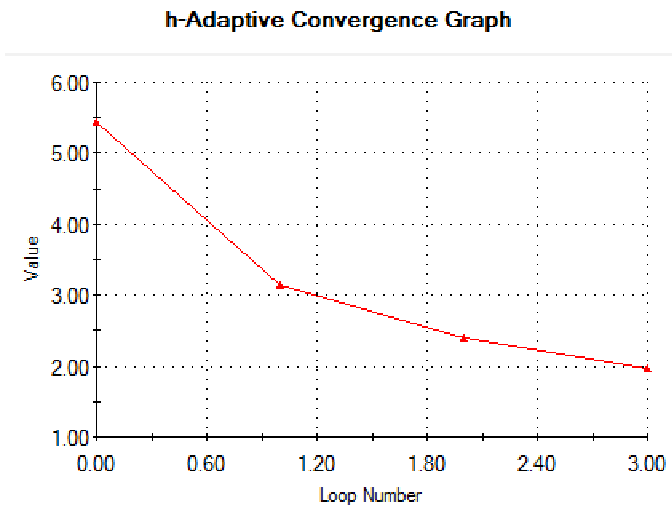

Figure E-96: Safety Bracket weld FEA h-adaptive convergence plot [3].

As can be seen, the elements concentrate around the corners of the weld closest to the applied load. This is to be expected, as these areas will take the most load, and are expected to have the highest stress gradients. The resulting stress plot is shown in Figure E-97.

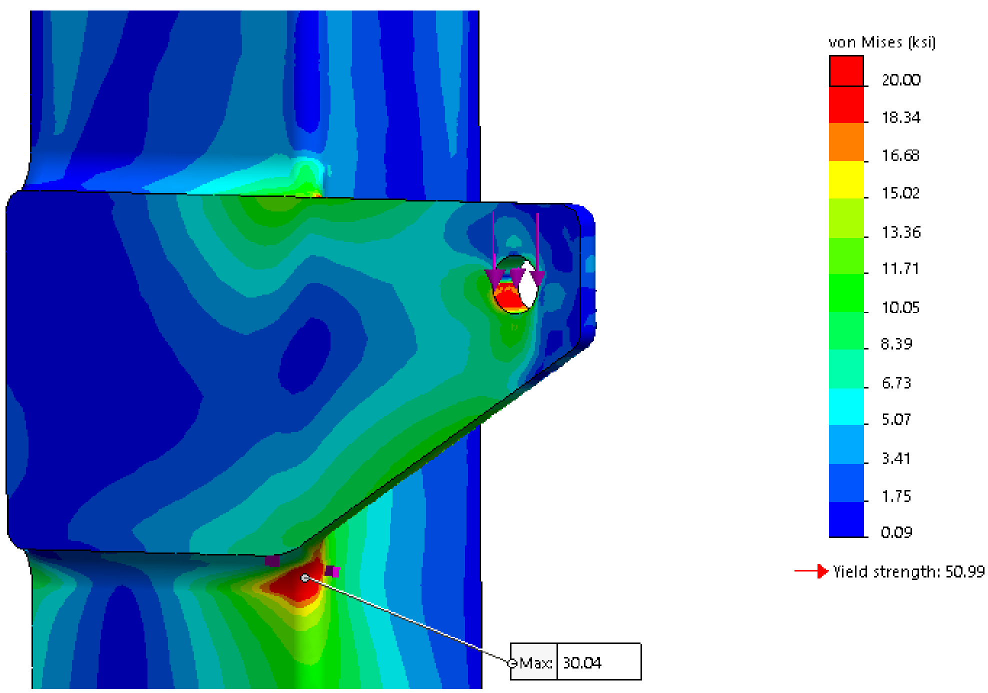

Figure E-97: Safety Bracket weld FEA stress plot [3].

As can be seen, a maximum stress of 30 ksi occurred at the bottom-inner corner of the weld. In order to attain a factor of safety of 3.0, a weld electrode with a minimum yield stress of 90 ksi is needed for this weldment. This requirement corresponds to an E110XX type electrode, which has a minimum yield stress of 95 ksi [8].

1.3.2. Bottom Frame

For the analysis of the bottom frame welds, a finite element model consisting of short sections of the Long Frame Support and Cross-Support, as well as the Cross-Brace Reinforcement Plate was created. Welds of 0.25” radius were modelled between all mating edges of the three parts. The bottom face of the Long Frame Support was fixed using fixed geometry, and two loads were applied to the Cross Support: one representing the load applied by the pillow block, and a remote load representing the portion of the Top Rollers which the weld must support. The fixtures and loads applied for the study are shown in Figure E-98.

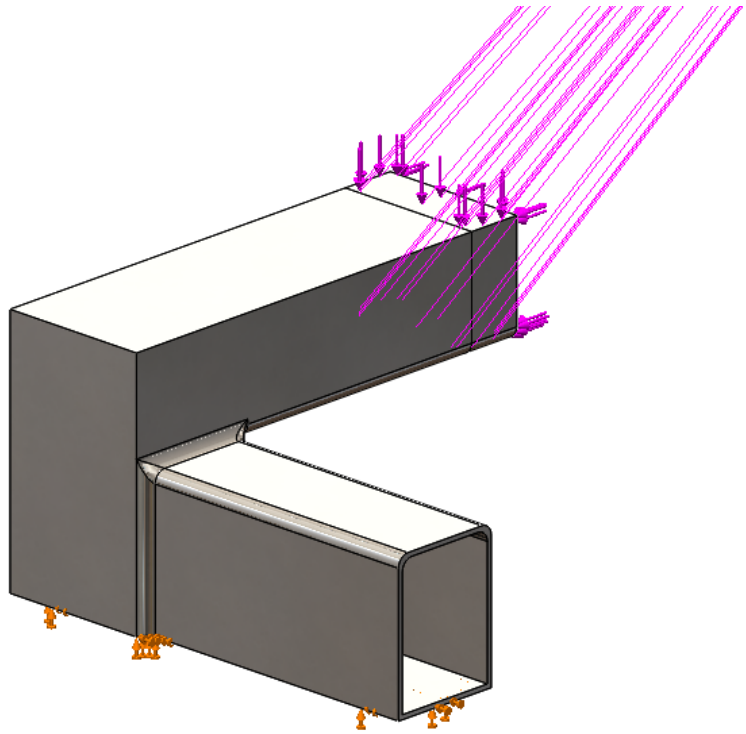

Figure E-98: Bottom Frame weld FEA boundary conditions [3].

After five iterations, the stresses in the welds were found to stabilize at a consistent value. An h-adaptive convergence plot was not used for this study since the stresses in the vicinity of the applied loads did not stabilize after multiple iterations. Since the stresses at the loads is not of concern in this study, monitoring the stresses in the areas of concern for consistency between mesh iterations is sufficient evidence of convergence.

The final mesh consisted of 137214 elements, with a minimum element size of 0.171”. The final mesh is shown in Figure E-99.

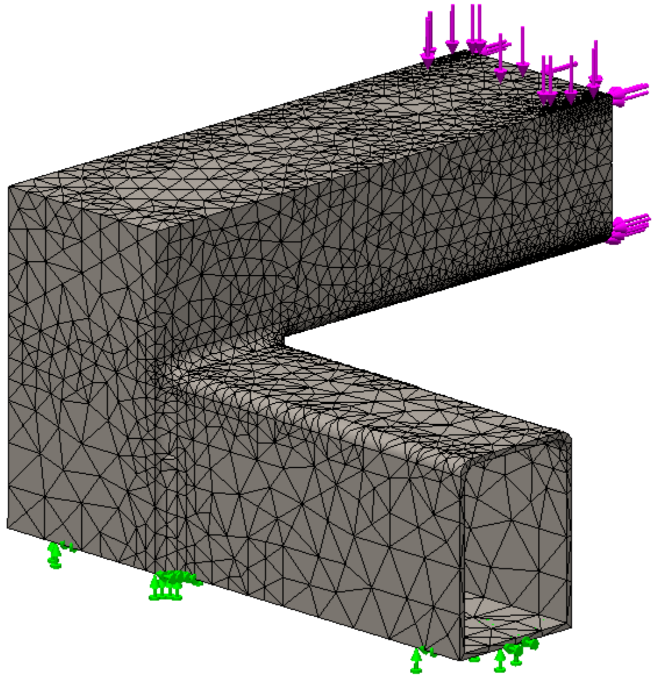

Figure E-99: Bottom Frame weld FEA meshing [3].

The resulting stress plot is shown in Figure E-100. The majority of the welds have a stress value below 10 ksi, with a small area having a maximum of 13.92 ksi, as shown in Figure E-101. This value of maximum stress results in a factor of safety of 3.59 using the E60 electrode described earlier.

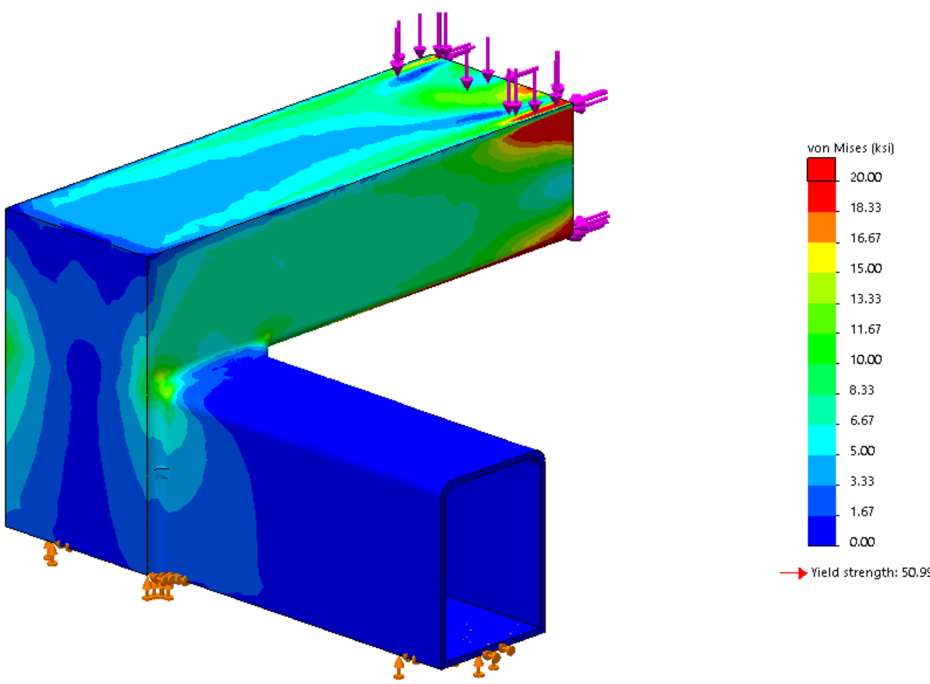

Figure E-100: Lower Frame weld FEA stress plot [3].

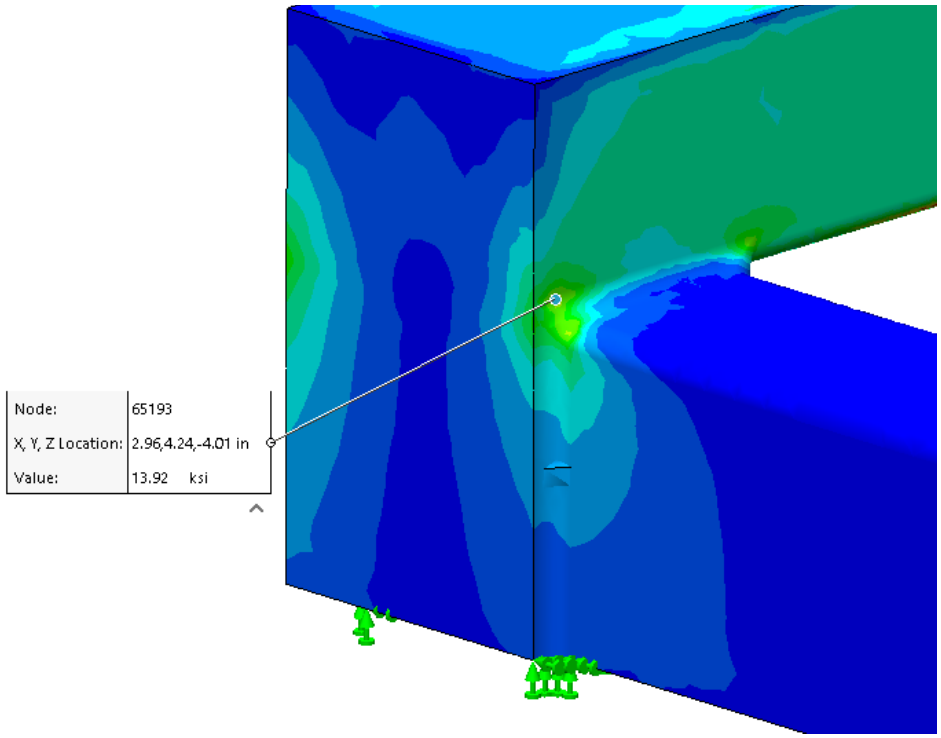

Figure E-101: Lower Frame weld FEA max stress [3].

1.3.3. Retaining Shafts

For the analysis of the welds in the retaining shaft, the same finite element model used for material strength analysis was utilized, this time adding welds of 0.25" radius using fillets. Virtual walls were again used to simulate the presence of the Side Arm, and a load was applied to one Square Shaft as before.

After five iterations of mesh refinement, the study converged with a global accuracy of approximately 94%. The h-adaptive convergence plot can be seen in Figure E-102.

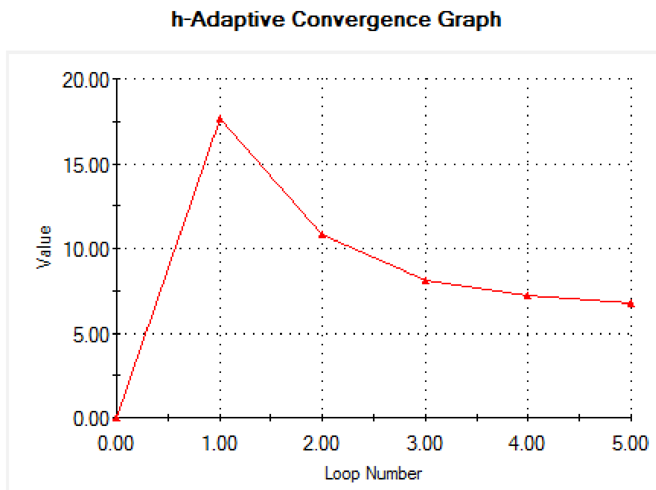

Figure E-102: Retaining Shaft weld FEA h-adaptive convergence plot [3].

The final mesh consisted of 64427 elements, with a minimum element size of 0.283”. The final mesh can be seen in Figure E-103.

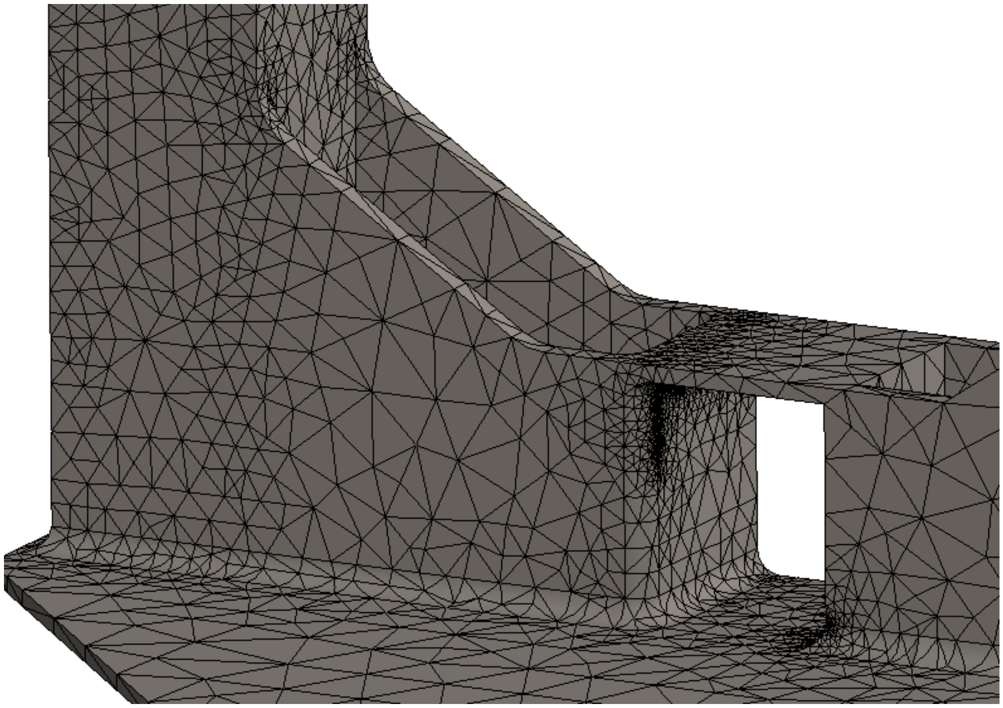

Figure E-103: Retaining Shaft weld FEA meshing [3].

As can be seen, the elements concentrate around the welds between the Gussets and the Backing Plate at the location of the Side Arm. The resulting stress plot is shown in Figure E-104.

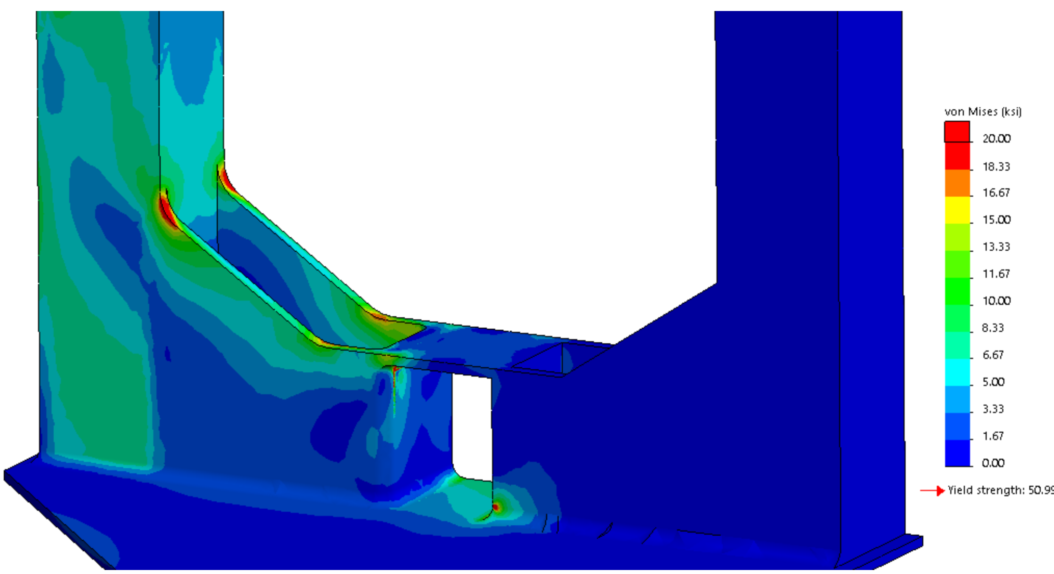

Figure E-104: Retaining Shaft weld FEA stress plot [3].

The highest level of stress in the welds occurred at the top of the weld between the Gusset and the Square Shaft. Since this location is in compression, the weld in this area is not likely to fail. The high stress in this area can therefore be disregarded. The next highest stress value in the welds occurred in the weld between the Gusset and the Backing Plate, with a value of 18.65 ksi. The maximum stress location is shown in Figure E-105.

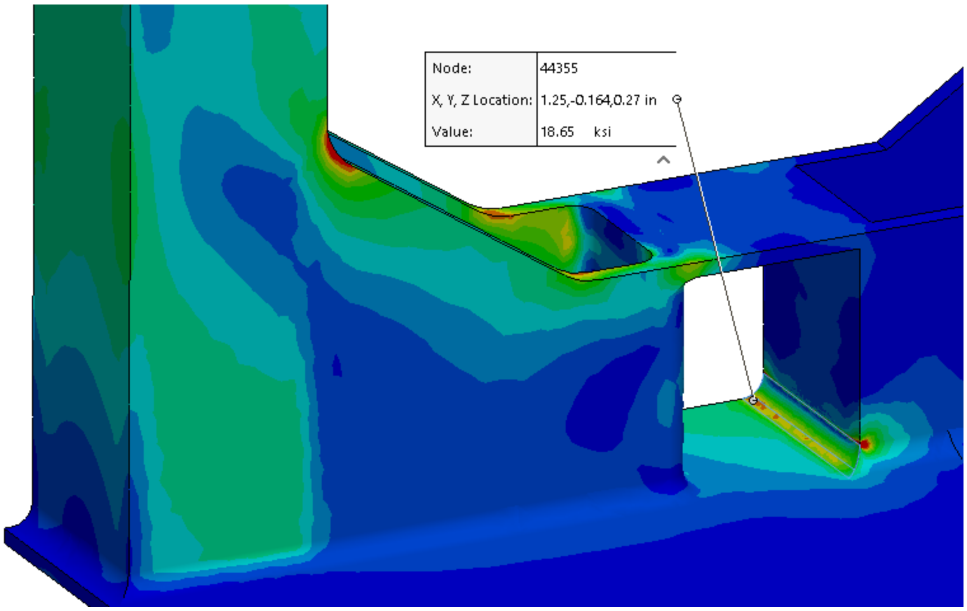

Figure E-105: Retaining Shaft weld FEA max stress [3].

With this maximum stress value, the use of E60 electrodes results in a factor of safety of 3.22. It should be noted that the ultimate strength of the weld electrode was used for the calculation of factor of safety since the Retaining Shaft is intended as a single-use feature.

1.4. Bearing Stress Analysis of Fasteners

In this section, the bearing stresses on various structural members connected using bolts are analyzed. The connections made using bolts or pins create stresses in the members that are connected along the surface of contact. The stress experienced by the contact surface between the connecting members is called bearing stress. For example, consider the bolt and plate A shown in Figure E-106. When the bolt and the plate are connected, the bolt exerts a force, *P* on plate A. The same force, *F*, is exerted by the plate on the bolt. The force exerted by the bolt is the resultant of all the forces distributed on the contact surface. The contact surface in this case being the inside surface the plate with diameter, *d* and length t equal to the thickness of the plate. This overall stress experienced by the surface of contact is called the bearing stress.

[redacted]

Figure E-106: Forces exerted over bearing surfaces [9].

Using a simplified model, bearing stress can be calculated as seen in (E-16).

$$\sigma_b = \frac{F}{A} \tag{E-16}$$

Where, *F* is the force, and *A* is the area of the contact surface. In the final design, the team used zinc coated Grade 8 steel 3/8" diameter bolts for the connection of Top Swinging Arm and Top Rollers subassembly. Similar sized hex head bolts are also used for the connection of Top Swinging Arm and Side Arm. To calculate the bearing stress of each location, the shear stress experienced by the bolt is compared against the ultimate shear strength of the selected bolt to determine the factor of safety. Most of the bolts picked for the final design were well over sized to prevent failure since the equations and parameters used for the bolts sizing are very basic representations of the average bearing stress over the entire contact surfaces. The calculation of the bearing stress of bolts of the connection of the Top Swinging Arm and Top Rollers subassembly is shown below:

$$\tau = \frac{F}{A} = \frac{F}{\frac{\pi}{4}d^2} = \frac{285*4}{\pi*0.375^2} = 2580.4\,psi = \mathbf{2.58\,ksi}$$

Where *F* is the shearing force at the location, and *A* is the area of the bolt. The ultimate tensile strength of the grade 8 steel 3/8" diameter bolt is 150,000 psi and ultimate shear strength of the bolt is approximately 60% of its ultimate tension strength, in pure shearing conditions. The calculation of the ultimate shear strength of the 3/8" bolt is shown below:

$$\tau_{ult} = 0.6\,\sigma_{ult} = 0.6*(150000) = 90000\,psi = \mathbf{90\,ksi}$$

Thus, for the 3/8" bolt, a safety factor of approximately 35 was calculated. Similarly, a 3/8" diameter bolt with a hefty factor of safety for the connection of side arm and top swinging arm was used.

For the connection of hydraulic arm to the side arm and top swinging arm, the team used a grade 8 steel hex head screw with 5/8” diameter, due to the hole sizing on the hydraulic arm. The calculation of the bearing stress of the connection of the hydraulic arm to the top swinging arm is shown below:

$$\tau = \frac{F}{A} = \frac{F}{\frac{\pi}{4}d^2} = \frac{3000 * 4}{\pi * 0.625^2} = 9778.45\, psi = \mathbf{9.78\, ksi}$$

The safety factor for the bolt located at the hydraulic connection was calculated to be 9.1. The same factor of safety applies to the bottom connection of the hydraulic to the side arm bracket.

For the walking axles subassembly, the team used grade 8 steel hex head screw with 1/2” diameter, and 5-1/2” in length. The shear force was calculated based the total weight of the assembly divided into four sections since four walking axles subassemblies are used. The calculation of the bearing stress used on the walking axles is shown below:

$$\tau = \frac{F}{A} = \frac{F}{\frac{\pi}{4}d^2} = \frac{1375 * 4}{\pi * 0.5^2} = 7002.8\, psi = \mathbf{7.0\, ksi}$$

Thus, comparing the calculated shear stress with the ultimate shear strength of the material, a safety factor of 12.8 is calculated.

2. References

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[7] B. Scott. "Welding Images." Winnipeg: Engineering Design, University of Manitoba, Winnipeg, MB, Nov. 26, 2017.

[8] Red-D-Arc. (n.d.). *Welding Electrode Classifications* [Online]. Available: https://www.red-d-arc.com/pdf/Welding%20Electrode%20Classifications.pdf [November 29, 2017].

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Appendix F: Final Design Cost Analysis

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List of Tables

1. Design Cost Breakdown

The follow sections outline the total cost the team's final design. The costs are broken into four different categories: sourced items, raw materials made of structural tubing, raw materials made of plate and sheet metal, and manufacturing processing and assembly costs. Lastly, all costs are summarized in a single table for comparison. The majority of items were sourced from McMaster-Carr and Russel Metals, as per the client's requests. However, for some specialty parts, additional vendors were selected and are presented in the following sections.

1.1. Sourced Items Cost Breakdown

Table F-I breaks down all sourced components used in the final design. Parts are sorted by the respective subassembly they are used in, and the parts name, description, vendor, vendor part number, quantity of said part used in the subassembly, package quantity the part is sold in, the cost per package of the part, and the total cost of that part are displayed. For the majority of standard off the shelf parts, such as bolts, nuts, and washers, McMaster-Carr was used as the vendor. It should be noted that the team assumed the worst-case scenario where Triple E RV would need to purchase a full package of a standard part, even when the full package would not be used. It is recommended that Triple E RV source equivalents of these parts in lower quantity from local suppliers where ever possible. Additionally, most larger companies that deal with McMaster-Carr on a regular basis often have a privately setup account where parts can be purchased at a discounted price. Thus, the team expects if that is the case, the overall sourced part costs could potentially be reduced so as much as 40%.

It should also be noted that the automated drive system parts are included in the sourced parts list. Although the team strongly recommended the use of the automated drive system, these parts account for approximately $700 USD in total, and thus if omitted, sourced parts total cost could be cut by approximately 30%.

Lastly, it should be noted that for the second entry of the 1/2" Serrated Flange Nuts in the Walking Axle subassembly, that the cost is omitted. This is due to the previous entry in the Bottom Frame and Rollers subassembly, where enough extra nuts were already listed.

TABLE F-I: COST BREAKDOWN OF ALL SOURCED PARTS

Sourced Parts Breakdown								
Subassembly (Part Number)	Sourced Part Name	Sourced Part Description	Vendor	Vendor Part Number	Quantity Used in Subassembly	Quantity Sold per Package	Cost per Package ($USD)	Total Cost ($USD)
Top Rollers (M00076)	Bearing Housing	3/4" 4 Bolt Flange Bearing	Fastenal	0474550	4	1	$ 11.39	$ 45.56
	3/4" Rotary Shaft	1566 Carbon Steel, 3/4" Diameter, 72" Long	McMaster-Carr	1346K36	2	1	$ 62.25	$ 124.50
	3/4" Clamping Shaft Collar	Black-Oxide 1215 Carbon Steel	McMaster-Carr	6435K16	2	1	$ 2.16	$ 4.32
	Rubber Coating	14-1/2 oz. Pail	McMaster-Carr	9560T7	3	1	$ 10.20	$ 30.60
	7/16" Hex Nut	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	94895A817	16	50	$ 7.33	$ 7.33
	7/16" Bolt Washer	7/16" Screw Size, 0.469" ID, 0.922" OD	McMaster-Carr	98023A032	32	25	$ 4.69	$ 9.38
	7/16"-1.5" Long Fully Bolt	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	92620A673	16	25	$ 12.21	$ 12.21
Swinging Arm and Hydraulic Lift (M00077)	Oil-Embedded Sleeve Bearing	3/8" Shaft Diameter, 1/2" OD, 7/8" Length	McMaster-Carr	6391K173	2	1	$ 0.76	$ 1.52
	Oil-Embedded Sleeve Bearing	3/8" Shaft Diameter, 1/2" OD, 1-1/4" Length	McMaster-Carr	6391K179	2	1	$ 1.40	$ 2.80
	Hydraulic Cylinder Lift	3 Ton Clevis Long Ram Jack	Northern Tool	46214	1	1	$ 49.99	$ 49.99
	3/8" Bolt Washer	3/8" Screw Size, 0.406" ID, 0.812" OD	McMaster-Carr	98023A031	12	50	$ 5.97	$ 5.97
	3/8" Hex Nut	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	94895A031	6	100	7.46	$ 7.46
	3/8"-2.5" Long, Partial Thread Bolt	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	91257A634	1	25	$ 12.33	$ 12.33
	3/8"-3.5" Long, Partial Thread Bolt	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	91257A638	1	10	$ 8.45	$ 8.45
	3/8"-4.5" Long, Partial Thread Bolt	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	91257A642	4	5	$ 7.23	$ 7.23
	5/8" Hex Nut	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	94895A035	2	25	$ 8.18	$ 8.18
	5/8" Bolt Washer	5/8" Screw Size, 0.656" ID, 1.312" OD	McMaster-Carr	98023A035	4	25	$ 9.58	$ 9.58
	5/8"-4" Long, Partial Thread Bolt	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	91257A810	1	5	$ 11.92	$ 11.92
	5/8"-3" Long, Partial Thread Bolt	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	91257A806	1	5	$ 9.67	$ 9.67
Bottom Frame and Rollers (M00078 and M00080)	Bearing Housing	1.25" Pillow Block	Fastenal	0474562	8	1	15.69	$ 125.52
	1/2"-4.5" Long, Partial Threaded Bolt	Medium-Strength Grade 5 Steel Hex Head Screw	McMaster-Carr	91247A730	16	10	$ 11.17	$ 22.34
	1/2" Serrated Flange Locknut	Grade 5, Zinc-Plated	McMaster-Carr	99904A104	16	50	$ 11.38	$ 11.38
	1-1/4" Partially Keyed Rotary Shaft	1117 Carbon Steel, 1-1/4" Diameter, 36" Long	McMaster-Carr	6117K36	4	1	$ 66.00	$ 264.00
	Rubber Coating	14-1/2 oz. Pail	McMaster-Carr	9560T7	4	1	$ 10.20	$ 40.80
Walking Axles (M00081)	8" Steel Wheel Swivel Casters	High-Capacity Vulcan Caster	McMaster-Carr	22665T74	8	1	$ 62.18	$ 497.44
	8" Diameter Wheel Brake	High-Capacity Vulcan Caster	McMaster-Carr	22665T754	2	1	$ 12.38	$ 24.76
	1/2"-1" Long	Grade 5, Zinc-Plated	McMaster-Carr	92865A712	32	25	$ 9.65	$ 19.30
	1/2" Serrated Flange Locknut	Grade 5, Zinc-Plated	McMaster-Carr	99904A104	32	50	-	-
	LDPE Unthreaded Spacer	1" OD, 3/4" Long, for 1/2" Screw Size	McMaster-Carr	92825A338	8	20	$ 13.00	$ 13.00
	1/2"-5.5" Long, Partial Thread Bolt	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	91257A734	4	5	$ 12.58	$ 12.58
	1/2" Center-Lock Locknut	Grade 8, Zinc Yellow-Chromate Plated	McMaster-Carr	90636A060	4	10	$ 6.00	$ 6.00
Retaining Shaft (M00082)	3/8"-1" Long, Flange	High-Strength Grade 8 Steel	McMaster-Carr	92316A624	8	25	$ 11.00	$ 11.00
	3/8" Weld Nut	Steel Hex	McMaster-Carr	93560A160	8	50	$ 10.15	$ 10.15
Drive Systems (M00083)	Electric Drive Sprocket	ANSI 40, 34T, 1.25 Shaft	McMaster-Carr	9236K182	6	1	$ 56.70	$ 340.20
	Chains	ANSI 40 Chain, xx" Length	McMaster-Carr	6261K173	1	1	$ 64.88	$ 64.88
	Motor	3 HP, 1740 RPM, 115 Volt Electric Motor	Electric Motor Warehouse	131544	1	1	$438.37	$ 438.37
	Gearbox	15:1 Worm Gear Reduction Gearbox	Superior Gearbox	BMQ075-180TC	1	1	$250.00	$ 250.00
	Bus Wheel Keyed Collar	Black-Oxide 1-1/4" Diameter Keyed Shaft	McMaster-Carr	6412K47	1	1	$39.08	$ 39.08
	Slide on Grips	Grooved, with Ribbed Texture, for 1" OD	McMaster-Carr	97045K55	4	6	$ 15.85	$ 15.85
							Total Sourced items Cost:	$2,575.65

1.2. Raw Materials – Structural Tubing Cost Breakdown

Table F-II breaks down the cost of structural tubing raw materials used within the design. Parts are sorted by the respective subassembly they are used in, and the parts name and assigned part number, stock material type, nominal dimensions, weight of the tubing per foot, cost of the tubing per weight, length of the part, number of the parts needed in the design, and total raw material cost of that part are included. All stock sizes are based on what was available in Russel Metals catalog. The cost of $0.91 per pound of steel used was a rough estimate given to the team via Russel Metal's sales associate. For more accurate pricing, a formal quote should be requested.

TABLE F-II: COST BREAKDOWN OF RAW MATERIAL - STRUCTURAL TUBING

Raw Material Costs - Structural Tubing							
Subassembly (Part Number)	**Part Name (Part Number)**	**Stock Type**	**Wt. per Ft. (lbs)**	**Cost per Lbs ($USD)**	**Length of Part (ft)**	**Part Quantity**	**Cost for Part ($USD)**
Top Rollers (M00076)	Top Roller Shell (M00002)	6.625" x 0.280" Round	18.99	$ 0.91	2.792	4	$ 192.97
	Tube Arms (M00004)	4" x 2" x 0.125" Rectangular	4.75	$ 0.91	1.333	2	$ 11.53
Swinging Arm and Hydraulic Lift (M00077)	Swinging Arm (M00006)	4" x 2" x 0.125" Rectangular	4.75	$ 0.91	5.677	1	$ 24.54
	Side Support Arm (M00007)	4" x 3" x 0.25" Rectangular	10.15	$ 0.91	5.500	1	$ 50.80
Bottom Frame and Rollers (M00078 and M00080)	Cross Support (M00019)	4" x 3" x 0.125" Rectangular	5.61	$ 0.91	4.000	4	$ 81.68
	Long Support (M00023)	4" x 3" x 0.125" Rectangular	5.61	$ 0.91	9.667	2	$ 98.70
	Welded Side Arm (M00029)	4" x 3" x 0.25" Rectangular	10.15	$ 0.91	2.667	1	$ 24.63
	Bottom Roller Shell (M00024)	6.625" x 0.280" Round	18.99	$ 0.91	2.083	4	$ 144.01
Retaining Shaft (M00082)	Square Retaining Shaft (M00011)	3.5" x 3.5" x 0.125" Square	4.75	$ 0.91	1.500	4	$ 25.94
Drive Systems	Handle Bar (M00028)	1" x 0.133" Round	1.68	$ 0.91	1.833	2	$ 5.61
	Bus Wheel Ring (M00022)	1" x 0.133" Round	1.68	$ 0.91	10.472	1	$ 16.01
	Bus Wheel Rod (M00021)	1" x 0.133" Round	1.68	$ 0.91	1.667	4	$ 10.19
						Total Structural Tubing Cost:	**$ 686.60**

1.3. Raw Materials – Plate and Sheet Metal Cost Breakdown

Table F-III breaks down the cost of plate and sheet metal raw materials used within the design. Parts are sorted by the respective subassembly they are used in, and the parts name and assigned part number, stock material type and nominal dimensions, approximate density of steel, cost of the steel per weight, blanking volume of said part, number of the parts needed in the design, and total raw material cost of that part are displayed. All stock plates and sheet gage sizes are based on what was available in Russel Metal's catalog. As with the structural tubing estimates, the cost of $0.91 per pound of steel used was a rough estimate given to the team via Russel Metals sales associate. For more accurate pricing, a formal quote should be requested.

TABLE F-III: COST BREAKDOWN OF RAW MATERIAL - PLATE AND SHEET METAL

Raw Material Costs - Plate and Sheet Metal							
Subassembly (Part Number)	Part Name (Part Number)	Stock Type	Density (lbs/in^3)	Cost per Lbs ($USD)	Volume of Part (in^3)	Part Quantity	Cost for Part ($USD)
Top Rollers (M00076)	Centering Plate (M00003)	1/4" Plate	0.284	$ 0.91	1.500	8	$ 3.10
	Hanging Plate (M00005)	1/2" Plate	0.284	$ 0.91	9.000	1	$ 2.33
Swinging Arm and Hydraulic Lift (M00077)	Top Arm Bracket (M00009)	1/2" Plate	0.284	$ 0.91	2.775	2	$ 1.43
	Side Arm Bracket (M00010)	1/2" Plate	0.284	$ 0.91	15.990	2	$ 8.26
	Safety Bracket (M00008)	7/16" Plate	0.284	$ 0.91	14.438	2	$ 7.46
	Mounting Plate (M00018)	1/2" Plate	0.284	$ 0.91	16.531	2	$ 8.54
Bottom Frame and Rollers (M00078 and M00080)	Gear Box Support (M00027)	7 Gage (0.179")	0.284	$ 0.91	4.232	2	$ 2.19
	Centering Plate (M00025)	1/4" Plate	0.284	$ 0.91	1.500	8	$ 3.10
	Cross Brace (M00020)	1/4" Plate	0.284	$ 0.91	16.875	4	$ 17.44
Retaining Shaft (M00082)	Backing Plate (M00013)	5/16" Plate	0.284	$ 0.91	108.750	2	$ 56.21
	Shaft Bent Gusset (M00012)	1/4" Plate	0.284	$ 0.91	23.550	4	$ 24.35
	Top Bracket (M00016)	1/4" Plate	0.284	$ 0.91	4.005	4	$ 4.14
	Gusset (M00015)	1/4" Plate	0.284	$ 0.91	0.500	4	$ 0.52
	Locking Plate (M00014)	1/4" Plate	0.284	$ 0.91	4.875	4	$ 5.04
Walking Axles (M00081)	Axle Base (M00026)	1/4" Plate	0.284	$ 0.91	82.410	4	$ 85.19
Drive Systems	Chain Cover	14 Gage (0.075")	0.284	$ 0.91	17.573	4	$ 18.17
						Total Plate and Sheet Metal Cost:	$ 247.48

1.4. Manufacturing Processing and Assembly Costs

To estimate the manufacturing processing and assembly costs of the design, a rough approach was taken. Due to having no additional information on Triple E RV's operational costs for processes such as blanking plate or sheets, sawing and milling tubing, press braking, welding, general assembly, and material handling costs, an approximate average wage between all operations was estimated at $25.00 USD per hour. Next, due to the design of the cart likely being a produced a total of only one or two times, it is likely that the manufacturing procedure of the cart would not be heavily controlled. Additionally, a manufacturing Engineer would likely be walking the design through processes, such that the team's recommendations and tests during production are accounted for. Given this scenario, the team believes the full design should still be produced over a period of no more than 3 full working days, or 24 hours.

Additionally, variable unknowns such as overheads and equipment wear factors are accounted for by standard suggested multipliers of total labor cost [1]. The results of the estimation analysis are presented in Table F-IV. For a more thorough manufacturing processing and assembly costs, the analysis should be recompleted using the proper factors and labor rates that apply to Triple E RV.

TABLE F-IV: COST BREAKDOWN OF PRODUCTION LABOR, OVERHEAD, AND MACHINE WEAR

Total Production Time (hrs)	24
Labor Rate ($USD/hr)	$ 25.00
Total Labor Cost ($USD)	$ 625.00
Overhead Factor (+%)	100%
Equipment Factor (+%)	50%
Total Production Costs ($USD)	**$ 1,562.50**

2. Summary of Design Cost

The summary of all estimated part sourcing costs, raw material costs, and production costs are displayed in Table F-V. As seen in the Table, the total cost of the team's final design is $5072.23 USD, with 51% being associated with sourced parts, 18% being associated with raw materials, and 31% being associated with production costs.

TABLE F-V: SUMMARY OF FINAL DESIGN TOTAL COST

Cost Breakdown	Cost ($USD)	% of Total Cost
Total Sourcing Parts Cost	$ 2,575.65	51%
Total Raw Material - Structrual Tubing Cost	$ 686.60	18%
Total Raw Material - Plate and Sheet Cost	$ 247.48	
Total Production Cost	$ 1,562.50	31%
Total Final Design Cost	**$ 5,072.23**	**100%**

3. References

[1] G. Kremer. (2016). *Recommended method for determining production costs* [Online]. Available: https://www.ohio.edu/mechanical/design/Resources/CostsforSrD.pdf [November 26, 2017].