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University of Minnesota Duluth Engineering Design Challenge

Bill Pederson
REGENTS OF THE UNIVERSITY OF MINNESOTA MINNEAPOLIS

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14. ABSTRACT

This report discusses the lifting mechanism designed by Bulldog Innovations for the United States Air Force design competition. The goal of the competition is to redesign a solution for rescuing personnel or equipment that may become trapped beneath an overturned vehicle weighing up to 45,000 lbs. The design will be judged on overall weight, size, lifting capacity, and ease of transporting. Many initial designs were considered including fire hose lift bags, hydraulic lifts, a scissor lift, a spring loaded pawl tri-pod, and a pulley lift system. These devices were proven to be undesirable due to many issues including weight, buckling under high impact forces, and cost. Due to the condensed timeline of this single semester project, Bulldog Innovations had to disprove ineffective solutions and confirm the optimal feasible design as soon as possible to complete the project on schedule.

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U.S. Air Force Lift Kit

Bulldog Innovations

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Executive Summary

This report discusses the lifting mechanism designed by Bulldog Innovations for the United States Air Force design competition. The goal of the competition is to redesign a solution for rescuing personnel or equipment that may become trapped beneath an overturned vehicle weighing up to 45,000 lbs. The design will be judged on overall weight, size, lifting capacity, and ease of transporting.

Many initial designs were considered including fire hose lift bags, hydraulic lifts, a scissor lift, a spring loaded pawl tri-pod, and a pulley lift system. These devices were proven to be undesirable due to many issues including weight, buckling under high impact forces, and cost. Due to the condensed timeline of this single semester project, Bulldog Innovations had to disprove ineffective solutions and confirm the optimal feasible design as soon as possible to complete the project on schedule.

Bulldog Innovations final solution is a portable winch system. The winch system is comprised of four main components including the winch, a block and tackle, a motor, and anchors. A synthetic rope, (Amsteel Blue), was used throughout the system for its extremely high strength to weight ratio. A gas powered chainsaw motor is attached directly to the winch for power. Attached to the base of the winch is the main rigging plate supporting the entire load of the system.

Upstream of the winch is a block and tackle system which provides a mechanical advantage of a factor of ten to the winch. The lower plate of the block and tackle is hard mounted to the primary rigging plate and the upper plate is hooked to the vehicle with the high strength synthetic rope. To support the entire system, terrain adaptable anchors are set behind the winch and are attached directly to the main rigging plate.

To ensure that components could withstand the very high loads experiencing this application, Finite Element Analysis (FEA) was completed as well as experimental tensile testing. Despite its brittle behavior, the 7075-T6 was chosen for its high strength to weight ratio. The tensile testing also proved to be very helpful in the design of the components by identifying stress concentrations in the plates. It also was utilized for observing the behavior of the synthetic rope while undergoing high loads. The FEA and tensile testing of all structural components in the lift verified the design to be an effective solution.

It was determined that for this particular design a shoring device will not prove beneficial. The shoring device's function is to provide a failsafe device in case of failure of the main lifting system. Because the greatest load is at the initial point of lift, the most likely failure of the system would occur prior to the vehicle leaving the ground. Once the vehicle has started to move, the vehicle will be capable of being completely overturned in less than two minutes if desired. As a result of the combination of these two factors, it was determined that a shoring solution would likely not be necessary. If a shoring device is necessary for a particular lift, a self-locking pneumatic powered tripod is recommended.

The competition resulted in failure from an insufficient quantity of anchors set in the ground. The remainder of the system performed as designed and would have effectively lifted the vehicle with the proper anchoring system. The final solution from Bulldog Innovations peaked strong interest from the PJs due to the unique application, differing from their current lifting techniques.

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Introduction

Bulldog Innovations is comprised of four engineers that provide valuable skills and experience to this project.

Design Team

Project Manager
Calin Davidson

Calin Davidson will be graduating in May 2014 with a bachelor's degree in Mechanical Engineering. Calin has been employed as an intern with Fraser Shipyards in Superior, WI. Calin is a Staff Sergeant in the Air National Guard based in Duluth, MN. Calin is interested in a career with the Air Force Research Laboratory organization.

Design Engineer
Levi Shrake

Levi Shrake will graduate in May 2014 with a bachelor's degree in Mechanical Engineering. He has recently completed an internship with AMSOIL, Inc., where he gained hands-on experience in design, facility operations, and internal combustion engine laboratory testing. Levi will continue his employment with AMSOIL, Inc. following graduation as a Mechanical Laboratory Engineer.

Design Engineer
Jacob Pearson

Jacob Pearson will graduate in May 2014 with a bachelor's degree in Mechanical Engineering. He has experience in design and manufacturing as an intern at Ardisam. He also gained experience in software analysis as an intern at Halliburton. Jacob will be heading into the energy industry as a field engineer with Halliburton after graduation.

Manufacturing Engineer
Kyle Schroer

Kyle Schroer will be graduating with a dual degree in Mechanical and Industrial Engineering in May 2014. He has completed multiple internships with Cummins Inc. and American Precision Avionics, providing him with valuable experience in component design, production support, and project management. Upon graduation, he will be working for Cummins as a Product Validation Engineer.

Overview

The United States Air Force (U.S.A.F.) is one of the five primary branches of military in the United States. U.S.A.F. personnel are equipped and trained to fight and defend in the air, space, and cyberspace battlefields. Currently there are nearly 700,000 active, civilian, air guard, and reserve personnel in the U.S.A.F. which has renowned itself as the world's largest air force military branch.

Problem Statement

The U.S.A.F. is actively improving and employing new designs to make the jobs of the personnel as efficient and safe as possible. The United States Air Force Special Operations Command's Special Tactics Pararescue Unit (P.J.'s) is consistently involved in recovery and rescue missions, from vehicles and personnel damaged by Improvised Explosive Device (I.E.D.). In many cases these I.E.D.'s overturn the vehicles they strike and trap personnel in or underneath the 45,000 lb. vehicles. The P.J.'s involved in these situations need a device to lift these vehicles and get them out of a dangerous situation as quickly and safely as possible.

Figure 1 displays the lift equipment that is currently used by the U.S.A.F. The kit consists of two air bags each capable of lifting 10,000 lbs, necessary hoses, controller, and air tank. The kit weights nearly 60 lbs and fits into a backpack. To safely support the overturned vehicle in case one of the airbags fails, the rescue team stacks rocks or fills empty sandbags as a safety precaution before they are able to complete their rescue.

Since many armored vehicles weigh two times what these lift kits can handle there is room for improvement. The current kits leave these four man rescue teams vulnerable due to long set-up times. Perhaps the largest downfall is the method for supporting the airbags, by placing rocks or filling empty sandbags. These pitfalls increase the chance of failed missions and loss of life.

Figure 1: Current Airbag Lifting Device

Mission/Objective/Constraints

Mission Statement

Bulldog Innovations will design and manufacture a lifting device capable of lifting a 45,000 lb. vehicle on unknown terrain for the United States Air Force Special Operations Command's Special Tactics Pararescue Unit by April 17th, 2014.

Objective

To produce a high quality product that will surpass our goals, a set of objectives was defined to provide the right path for the project's success. These high level objectives are simple, yet imperative to keep in mind during the duration of the project.

- Complete Lifting Mechanism Design
- Manufacture the Lift
- Test the Product
- Win the Competition

Constraints

The following constraints were considered when designing the lifting apparatus:

- Lifting Capacity of 22,500 lbs. (To lift one side of a 45,000 lb. vehicle).
- Lifting Height of 20 in.
- Total weight less than 30 lbs.
- Portable Dimensions of 12" x 12" x 6"
- Operational in Varying Hot/Cold Temperatures
- Withstand Sloped and Varying Terrain (sand, mud, shale rocks, etc...)
- Operational at High Altitudes
- Easy to Operate
- Portable

Final Solution

The final solution developed by Bulldog Innovations is a winch lifting mechanism. The winch lifting system consists of four main components: the winch, block and tackle, anchors, and motor. The winch will be utilized as the main lifting mechanism of the system. Powering the winch is a 3.62 horsepower chainsaw motor attached directly to the winch gear box. A block and tackle system is placed upstream of the winch to provide a large mechanical advantage in lifting the vehicle. Anchors are attached to the winch and shoring device to support the system as the vehicle is lifted. The anchors are adaptable to the particular terrain required for each application. Figure 2 displays a layout of the winch lifting system application.

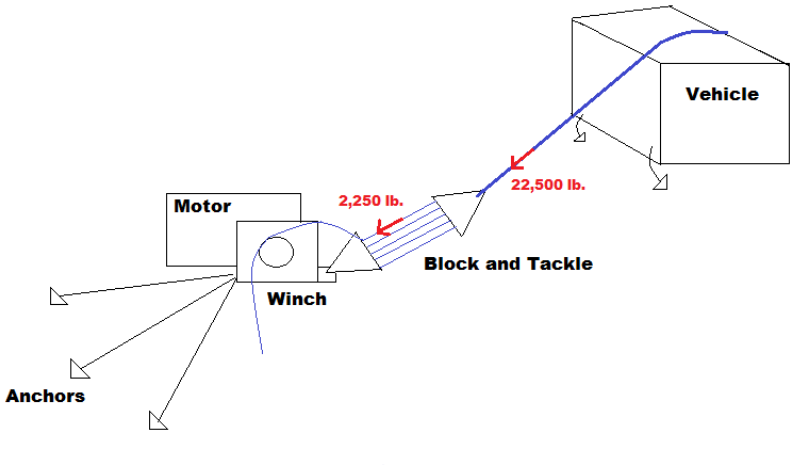


Figure 2: Winch System Layout

Winch System

The winch used in this system is a Capstan Rope Winch shown in Figure 3. The winch attaches to a chainsaw motor via the 3/8" pitch drive sprocket and an adapter on the winch gearbox. The gearbox contains a set of 4 stage spur gears with a 125:1 gear ratio to reduce the high speed of the chainsaw motor and in return developing a higher torque. The Capstan Winch is capable of pulling up to 2,500 lbs., which satisfies the requirements of this system after an overall force reduction associated with the block and tackle discussed in the next section. Per field testing results of the winch system and advice from winch experts at Superior Lidgerwood-Mundy, a 1" longer winch spindle was manufactured to provide four more wraps of the rope resulting in an increased friction resistance on the load.

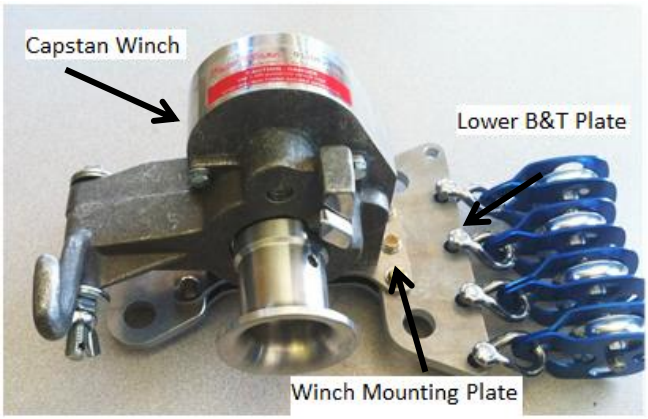


Figure 3: Captsan Rope Winch with a portion of the block and tackle system shown.

Figure 4 displays a mounting base that was developed for attaching the winch to the block and tackle system as well as the anchors. The plates were constructed out of 7075-T6 aluminum to acquire a high strength to weight ratio. Both finite element analysis (FEA) and experimental tensile testing were completed to ensure that the components can surpass the high loads they will endure. Figure 5 shows the FEA results with a 26,000 lb. load. The load is distributed along the block and tackle mounting holes while the anchor holes were kept fixed resulting in a maximum stress of 59,000 psi, which is below the yield stress of 7075-T6 aluminum.

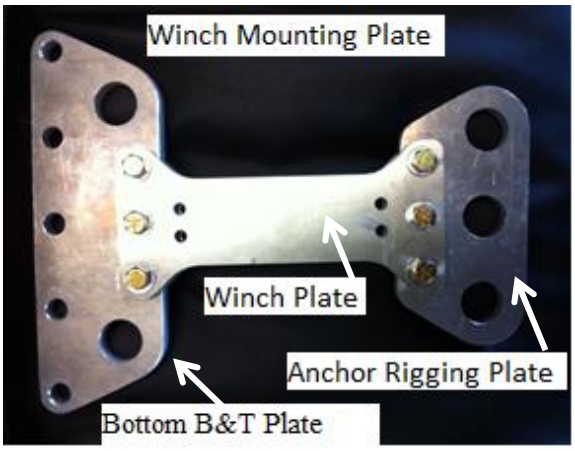


Figure 4: Winch Mounting Plate Assembly

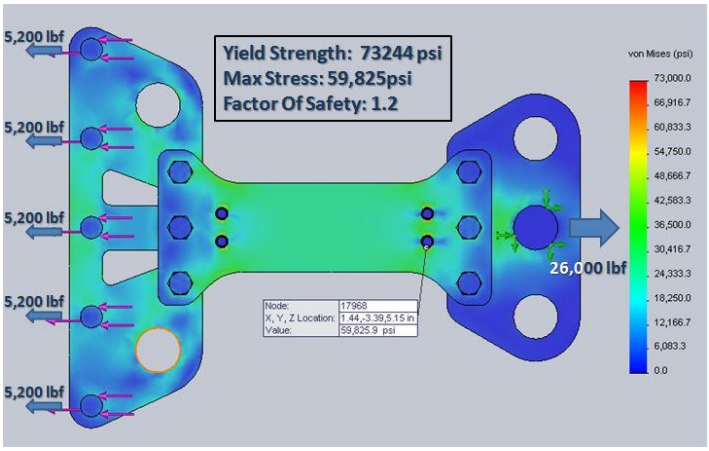


Figure 5: Winch Mounting Plate von-Mises Stress Plot

The two inner plates of the baseplate mounting system were tested in the tensile tester as these were the weakest components based on the FEA results. Figure 45-48 in Appendix A displays the FEA analysis of each component. As expected with 7075-T6 Al, a very brittle break was experienced at the winch mounting holes with very little yielding shown in Figure 6. The results from the tensile test converged with the F.E.A. results previously performed. The first plate that underwent the test reached a maximum load of 29,000 lbs. and the second of 27,200 lbs. In the final assembly both of the plates will be installed providing a safety factor of two for the winch mounting base. Figure 7 illustrates the load results from the experimental tensile test for one of the plates.

Figure 6: Winch Mounting Plate Post Tensile Test

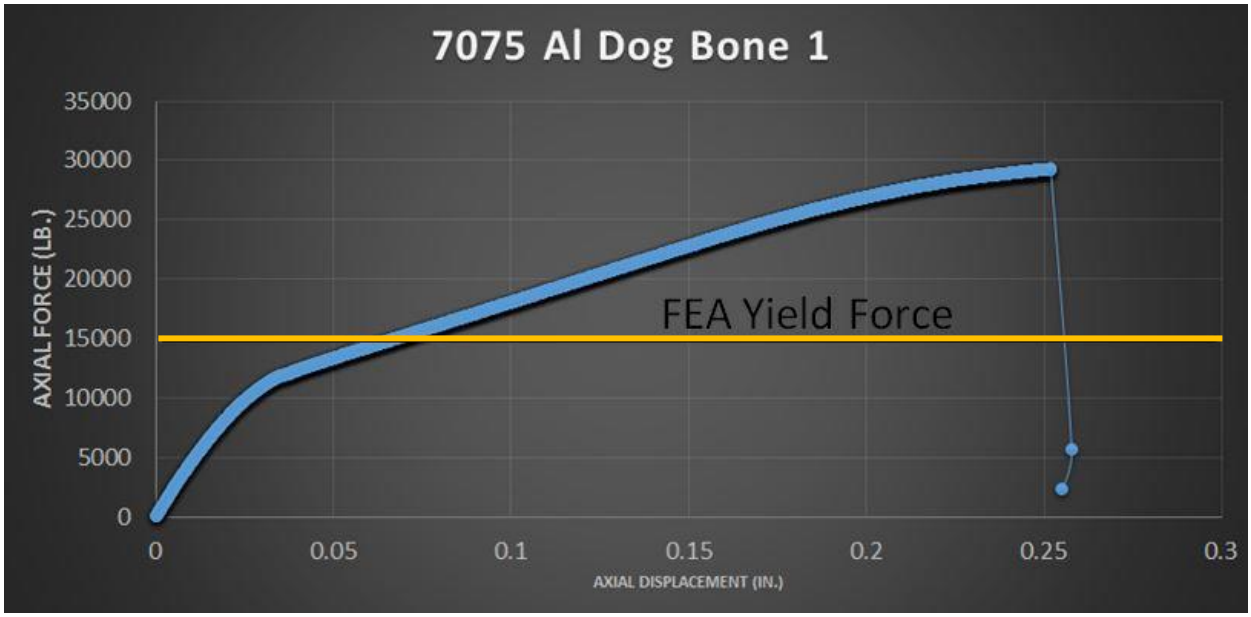


Figure 7: Baseplate Tensile Test Results

Block and Tackle

A block and tackle mechanism was used in this system to aid in the reduction of the force required to lift the vehicle. Without a block and tackle, the winch system and motor would need to be redesigned to support a much higher load resulting in a much heavier system. Figure 8 displays a free body diagram of the block and tackle. The block and tackle consists of two sets of five pulleys, creating 10 separate parts of the rope. This provides a large mechanical advantage as the input force required to move the load is reduced by 10 times, from 22,500 lbf to 2,250 lbf.

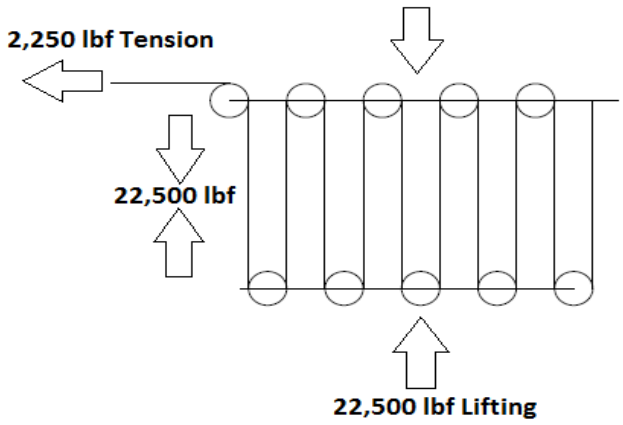


Figure 8: Block and Tackle Free Body Diagram

Figure 9 displays the assembled block and tackle system. Each pulley is rated at a breaking strength of 6,000 lbs., surpassing the 4,500 lbf that they will experience during the lifting operation. The top and bottom plates were constructed out of a ½" 7075-T6 Al plate for its lightweight and high strength properties. To determine where the critical strength areas were on these parts, a destructive tensile test was completed on a prototype block and tackle system. Figure 10 displays the first prototype experimental block and tackle attached to fixtures in the tensile tester.

Figure 10: Assembled Block and Tackle

Figure 9:Block and Tackle Tensile Tester Setup

Figure 11 shows the post tensile test results using ¼" plates for the block and tackle. The test resulted in a maximum force of 11,100 lbs. shown in Figure 12. These results signified that the plates required strengthening in the mounting hole region to be able to handle the full 22,500 lb. load. To strengthen the plate, it was changed from ¼" to ½". Figure 13 displays the FEA analysis of the final ½" block and tackle plate. Using the results from the destructive tensile tests and FEA, the plate was strengthened in the upper mounting hole region. Material was removed from non-critical regions allowing for a 20% weight reduction to the component. Additional data and test results can be found in Appendix A.

Figure 11: Block and Tackle Tensile Test Failure Location

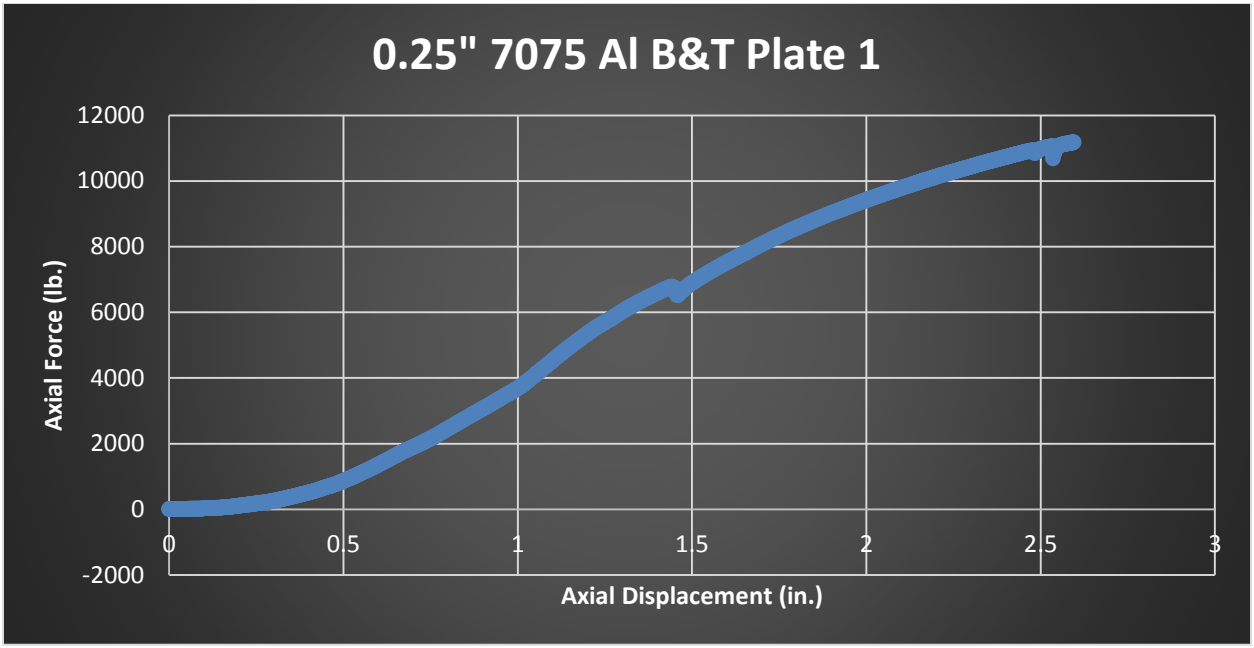


Figure 12: 1/4" Block and Tackle Tensile Test Load Data

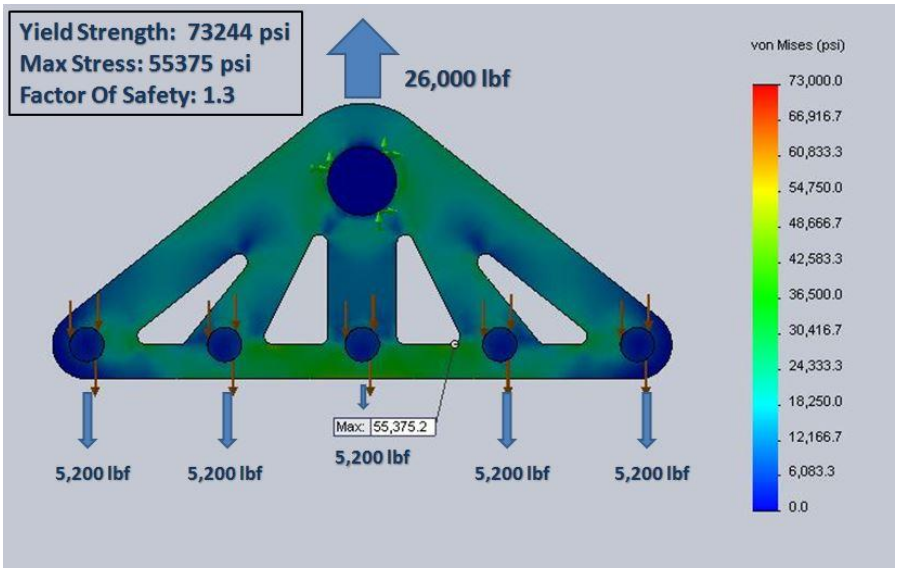


Figure 13: Block and Tackle Upper Plate von-Mises Stress Plot

Synthetic Fiber Rope

The cable routed throughout the winch system is an Amsteel Blue Synthetic Rope. A synthetic fiber rope was chosen for the winch system for its very high strength to weight ratio compared to steel cables. Amsteel Blue synthetic ropes are 85% lighter and have a 30% higher breaking strength than wire cable. It is comprised of Duneema SK-75 synthetic fiber which has no snapback providing a safer working environment around the lift. Two different sizes of the rope were used within the winch system, ¼" and ½".

¼" Rope

The ¼" Amsteel Blue Synthetic rope will be utilized in areas of the system that do not experience the full 26,000 lbf. These locations include the block and tackle pulley system where the mechanical advantage reduces the force to only 2,600 lb. and the anchors where the load will be distributed between multiple anchors. The ¼" Amsteel Blue Synthetic rope is rated at a breaking strength of 7,700 lbs. and a weight of 0.04 lb. per foot.

To confirm the ratings of the ¼" rope, it was tested on a hydraulic tensile tester at the University of Minnesota Duluth, as shown in Figure 14. The purpose of the test was not only to verify the strength of the rope, but also to test a bowline knot considered for attachment points in the system. The rope was secured with a bowline knot to a fixture on each end and the fixtures were clamped into the upper and lower grippers of the tensile tester.

Figure 14: Hydraulic Tensile Tester

Figure 17 displays the resulting load data from the ¼" rope tensile test. The test verified that the bowline knot did hold, however it broke at 3,600 lbs, 47% less than the rated breaking strength of the rope. Figure 16 shows a bowline knot, and Figure 15 displays a different method of tying the rope called a fixed eye loop. This loop consists of feeding the rope back through the hollow center of itself where it will ultimately tension down on itself when a load is applied. Figure 18 shows that the fixed eye loop broke at nearly 8,000 lbs. confirming the rating of 7,700 lb. breaking strength of the rope. This confirmed that a knot will drastically reduce the strength of the rope and a fixed eye loop does not affect the strength of the rope. Therefore a fixed eye loop will be used when possible.

Figure 16: Bowline Knot

Figure 15: Fixed Eye Loop

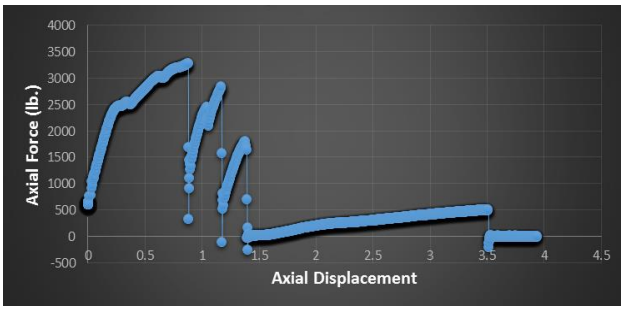


Figure 17: 1/4" Rope Bowline Knot Test Results

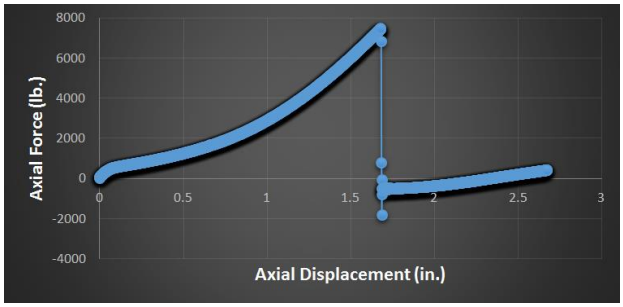


Figure 18: 1/4" Rope Fixed Eye Loop Test Results

½" Rope

The pulley system design requires one attachment point which will carry the full 22,500 lb. load to hoist the vehicle. This area includes the section from the block and tackle to the hard point, or point of attachment to the vehicle. In this application, the ½" Amsteel Blue synthetic rope will be utilized, providing a much stronger rope than the ¼". The ½" Amsteel rope has a breaking strength of 30,600 lbs and only weighs about 0.085 lbs per foot. Figure 19 shows a comparison of the blue ½" to the grey ¼" rope.

Figure 19: 0.5" and 0.25" Amsteel Blue Synthetic Rope

Anchors

In order to support the winch system it will be anchored based on the appropriate terrain at the spot of the vehicle turnover. Figure 20 displays a diagram of the application of the anchoring system in a soil material. As tension is brought to the rope, it displaces the buried anchor upwards until it catches and wedges itself into the soil.

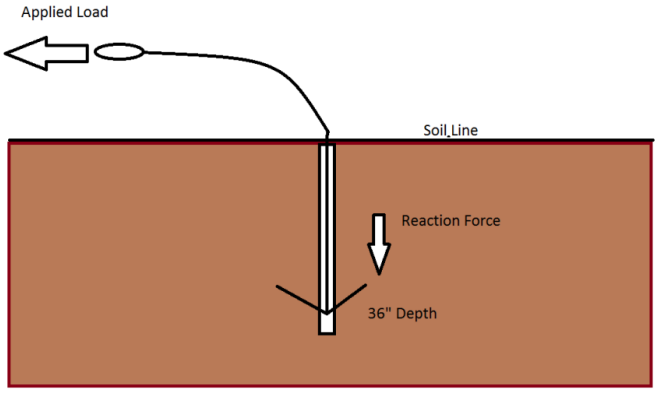


Figure 20: Anchor Application Free Body Diagram

Figure 21 displays an expandable anchor that was designed for sand, dry soil, clay, mud, and loose shale. The anchor is comprised of two square tube sections that are pinned together. A rope will be attached to the center pin between the tubes. The anchor is set into a hole that's pre-drilled by an auger displayed in Figure 22. The auger is attached to the chainsaw with a Lewis Multi-Drill attachment.

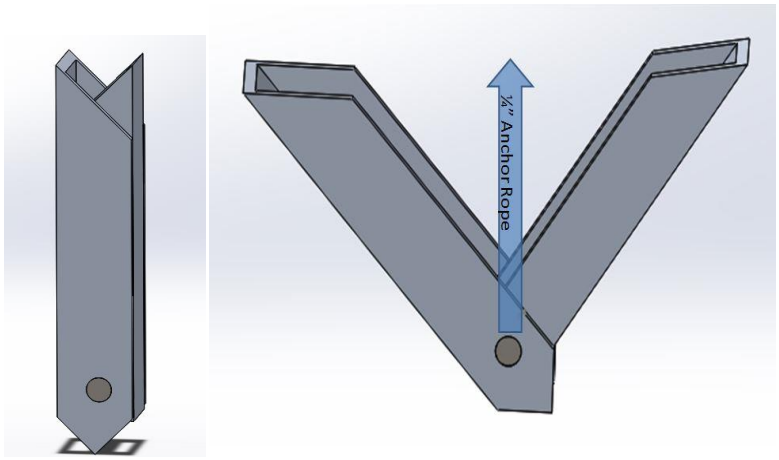


Figure 21: Expanding Anchor

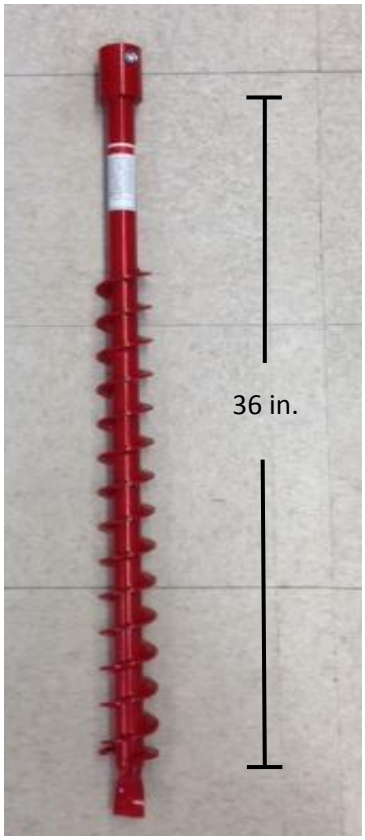


Figure 22: 2" Auger Bit

For loose slate rock applications similar to what is seen in Figure 23, the anchor described above will still be utilized. In a loose slate condition, the auger will still have the ability to penetrate the surface to the required anchor depth. The anchor will be placed in the hole and deployed as tension is applied to the rope. The slanted portion on the top of the anchor tubes will allow the anchor to grab into the layered slate material and support the applied loading. Due to frozen soil the anchors cannot be tested in the terrain that is to be expected during the actual operation of this device.

Figure 23: Shale Rock and Soil- *www.greatsouth.net*

Chainsaw Motor

Figure 24 displays the Husqvarna 460 Rancher chainsaw that was chosen to power both the winch and the auger for its high power to weight ratio. An electric motor was first considered for the winch apparatus for its precision and compactness, but was eventually disregarded due to the heavy weight associated with not only the motor itself but the batteries required to power it. A hand winch was another option, but was also not considered due to the accompanying manual labor and slow process time.

Figure 24: Husqvarna 460 Rancher Chainsaw With Winch Adaptor

The 460 Rancher has a 60.3 cc displacement producing 3.62 hp. At this horsepower, the winch produces its maximum pulling force of 2,500 lb. This resulting force exceeds the 2,250 lbf required after the initial load reduction by the block and tackle. A custom adapter is required to accommodate both the winch and drill attachments for the chainsaw. Figure 25 displays the standard adapter for the winch application.

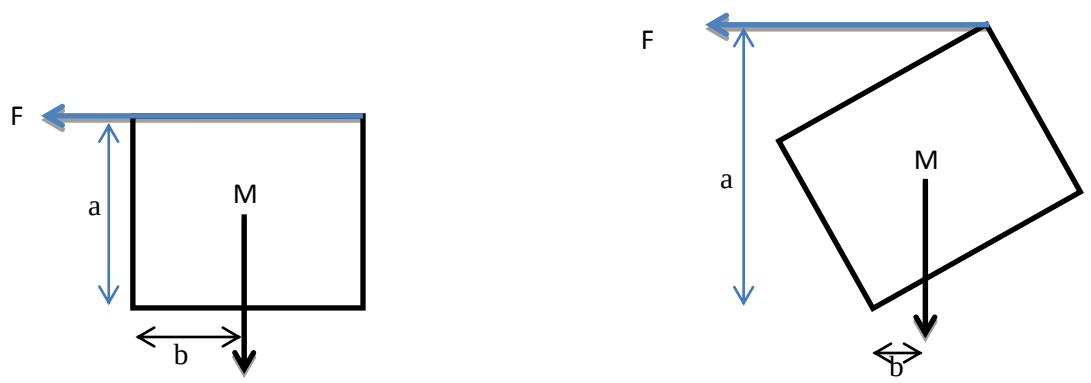
Figure 25: Winch Chainsaw Adapter

Shoring Device

Due to the unique design solution for the United States Air Force, it was determined that a shoring device will not be necessary while implementing this winch design. The shoring device's main function is to prevent the system from collapsing upon failure of the main lifting mechanism. The event of a failure is highly unlikely to occur while the vehicle is elevated because the greatest force will be exerted at the initiation of lift, allowing little to no drop if failure in the system occurs. Figure 26 displays the force analysis of lifting the vehicle.

The winch system will allow enough height for personnel or equipment to be removed from underneath within seconds of starting and will completely flip the vehicle in a few minutes. Shoring the vehicle as it is being raised will slow the time it takes to operate the system causing risk to rescue personnel.

In the event that shoring is required personnel, a pneumatic tripod may be used. The tripod contains three self-locking legs that rise with the vehicle using pressurized air. The tripod will not provide any lifting force but will remain in contact with the vehicle.



$F = Force\ applied\ by\ winch\ system$	$M = Mass\ of\ target\ vehicle$
$a = lever\ arm\ of\ F$	$b = lever\ arm\ of\ M$
$F * a \geq M * b$	

The force (F) is multiplied by the lever arm (a) which must overcome the mass (M) times the lever arm (b) in order to lift the vehicle. As the lever arm (a) is roughly twice the length of (b) the force (F) only has to be half that of mass (M). As the vehicle is lifted arm (b) will become shorter reducing the required force to lift, and the length of arm a increases thus increasing the effect of force (F).

Figure 26: Load Distribution Analysis

Shoring Force Analysis

Figure 27 illustrates an MRAP armored vehicle as an example for this application. The force required was evaluated as if the vehicle was completely overturned and pulled until it was back to its upright position. Table 1 shows the specifications of the MRAP vehicle.

Figure 27: MRAP Armored Vehicle

Table 1: MRAP Vehicle Specifications

MRAP Specs.		
Width	100	in
Height	120	in
Mass	45000	lb

The force required to overturn the vehicle was analyzed as the vehicle goes from a flat position (0 degrees) and rotated up until the required force is 0. Equation 1 illustrates how the static force was analyzed.

$$Force = \frac{Mass * b}{a}$$

Equation 1: Static Force Analysis

Table 2 concludes the results of the force analysis of the vehicle as it is being pulled. Figure 28 displays a plot of the resulting data. The results show that the initial position requires the greatest force, and as its orientation changes as it is being lifted the force gradually reduces until it reaches the most upright position. It is also noted that when the vehicle is at a 90 degree orientation the center of mass is skewed towards the bottom of the vehicle.

Table 2: Vehicle Force Analysis

Force Analysis			
Angle (Deg)	**Force (lbf)**	**Arm a (in)**	**Arm b (in)**
0	22500	100	50
10	15353	119	40.6
20	10000	135	30
30	5525	146.6	18
40	1765	153	6
45	0	155	0

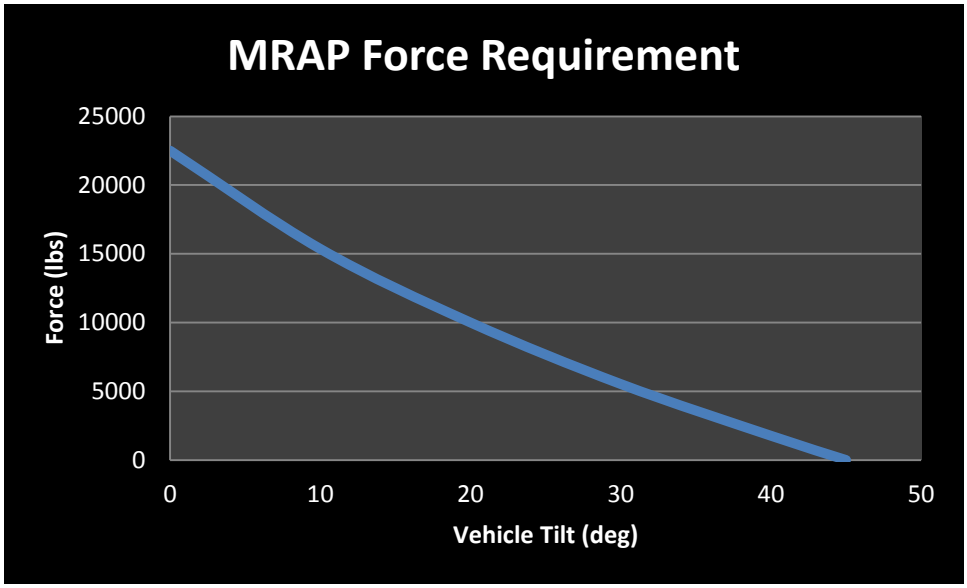


Figure 28: MRAP Force Requirement

Shoring Solution

In the event where a shoring device is required a self-locking automatic tripod has been designed. Figure 29 and 30 illustrates an expanded and collapsed 3-D model. As the vehicle is being lifted this tripod extends along with the vehicle using compressed air. In the event of a failure the three legs have a locking mechanism, similar to a one-way bearing, designed to stop the 22,500 lbf. The tripod incorporates telescoping tubular legs, upper and lower plates, tension cables for the legs, a central fixture plate, and appropriate air hoses. The collapsed tripod fits into a 3 in. opening and weighs approximately 25 lbs. The operator only needs to place the shoring device underneath the vehicle and supply the tubes with a pressure of 5 psi to expand the device.

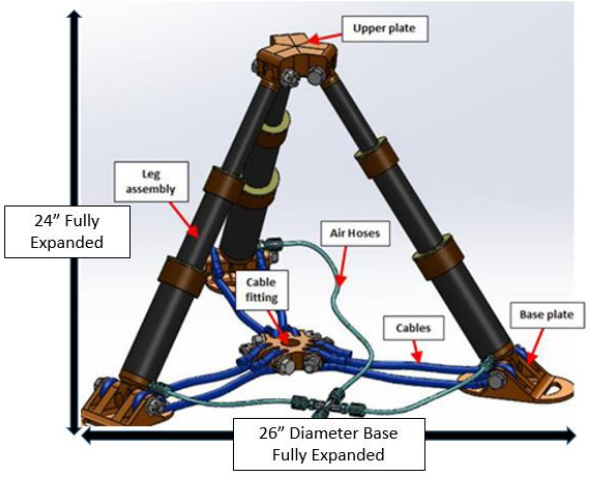


Figure 30: Expanded Tripod Shoring Solution

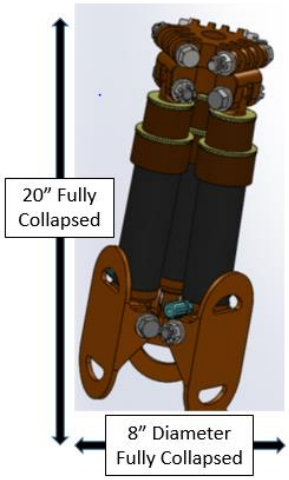


Figure 29: Collapsed Tripod Shoring Solution

The one way locking mechanism consists of bearing balls that become locked onto a steel wedge-shaped race. Implementing this mechanism gives the shoring mechanism virtually no drop in the event of a primary lift failure. A cross sectional view of the locking mechanism is shown in Figure 31. The bearing balls are internally spring loaded by a plate spring. This allows the bearing balls to freely rotate as the tube is extended, and to wedge themselves as it collapses. In order for the operator to disengage the locking mechanism and collapse the tripod, the aluminum cap is twisted, thereby pulling the bearing balls upward and preventing them from being wedged.

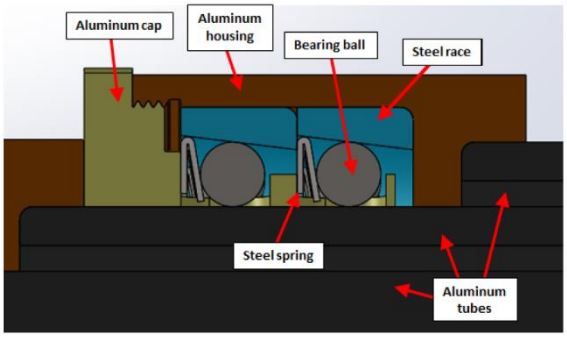


Figure 31: Locking Mechanism Cross Section

An exploded view of the tripod legs can be seen in Figure 32. Each of the legs are constructed from aluminum tubes. There are two bearing locations on each of the legs which include a threaded aluminum cap, two rows of bearing balls, two steel wedge-shaped races, two plate springs, and a steel housing. The upper tube contains a pressed aluminum cap with a hardened steel bushing. The upper tubes are kept from sliding by aluminum ends which are pressed. The lower cap is also pressed in and contains a spherical bearing pressed into the aluminum cap.

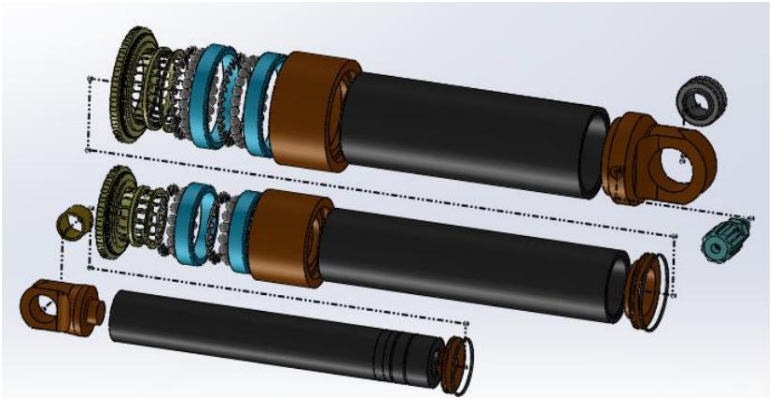


Figure 32: Tripod Tube Exploded View Assembly

The supporting base of each of the tripod legs is seen in Figure 33. It is constructed of 7075-T6 Aluminum and is bolted to the leg with a 5/8" grade 9 bolt. Each of the bases is connected to a centralized fixture plate using ½" Amsteel Blue synthetic rope. This connection creates a tension to prevent the tubes from sliding and collapsing.

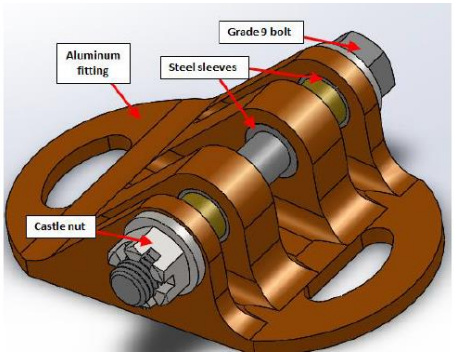


Figure 33: Tripod Leg Base Plate

The centralized fixture plate is manufactured from 7075-T6 Aluminum and is illustrated in Figure 34. The synthetic rope from the base plates are connected to the plate using a 5/8” grade 9 bolt. Figure 35 shows the upper plate which connects all three tripod legs together. The plate is manufactured from 7075-T6 aluminum and is connected to the tubes using a 5/8” grade 9 bolt.

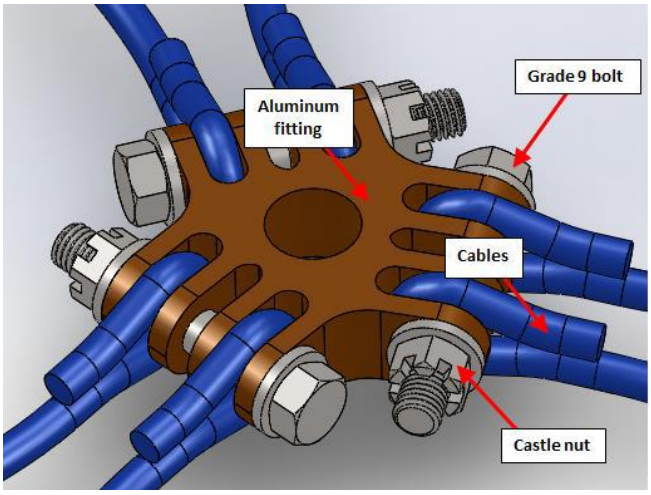


Figure 34: Tripod Central Fixture Plate Assembly

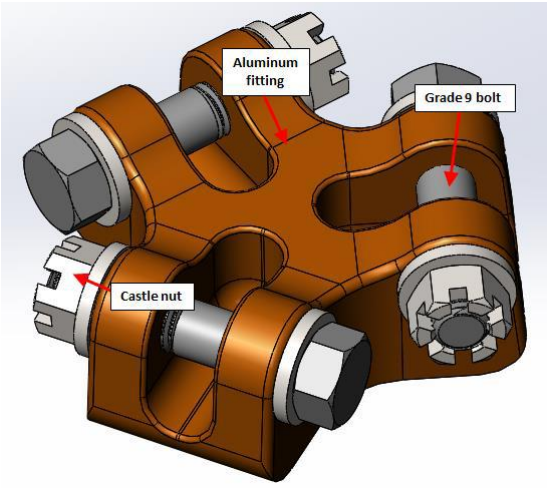


Figure 35: Tripod Upper Plate Assembly

The tripod shoring device must fit under the vehicle into a 3 in. opening and can rise to a 24 in. lift height. The use of telescoping tubes allows for this, and also for easy transportation. When the upper plate has lifted 2 in. the tube is at a 10.8 degree angle to ground, as shown in Figure 36. With the maximum force normal to the upper plate at 22,500 lbf., each leg is taking a maximum vertical load of 7,500 lbf. With the leg at 10.8 degrees the axial load per leg is 40,025 lbf displayed in Equation 2.

$$Axial\ Load\ per\ Leg = \frac{7{,}500\ lbf}{\sin(10.8)} = 40{,}025\ lbf$$

Equation 2: Axial load per Leg

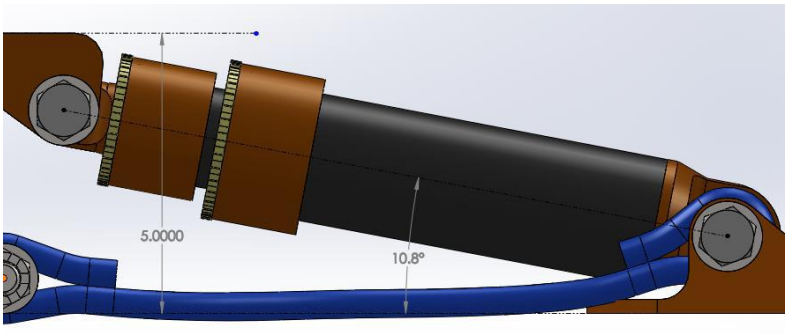


Figure 36: Tripod Leg Angle at 5 in. Height

Further finite element analysis was used to evaluate the stresses on all of the tripod components. The results from the analysis can be found in Appendix B. Due to the extended lead times to manufacture all of the components required for the tripod shoring device it will not be used in the competition. If shoring is required the current process using sand bags will be utilized for the winch lift system.

Operational Procedure

Figure 37 visualizes the steps in operating the winch lifting device.

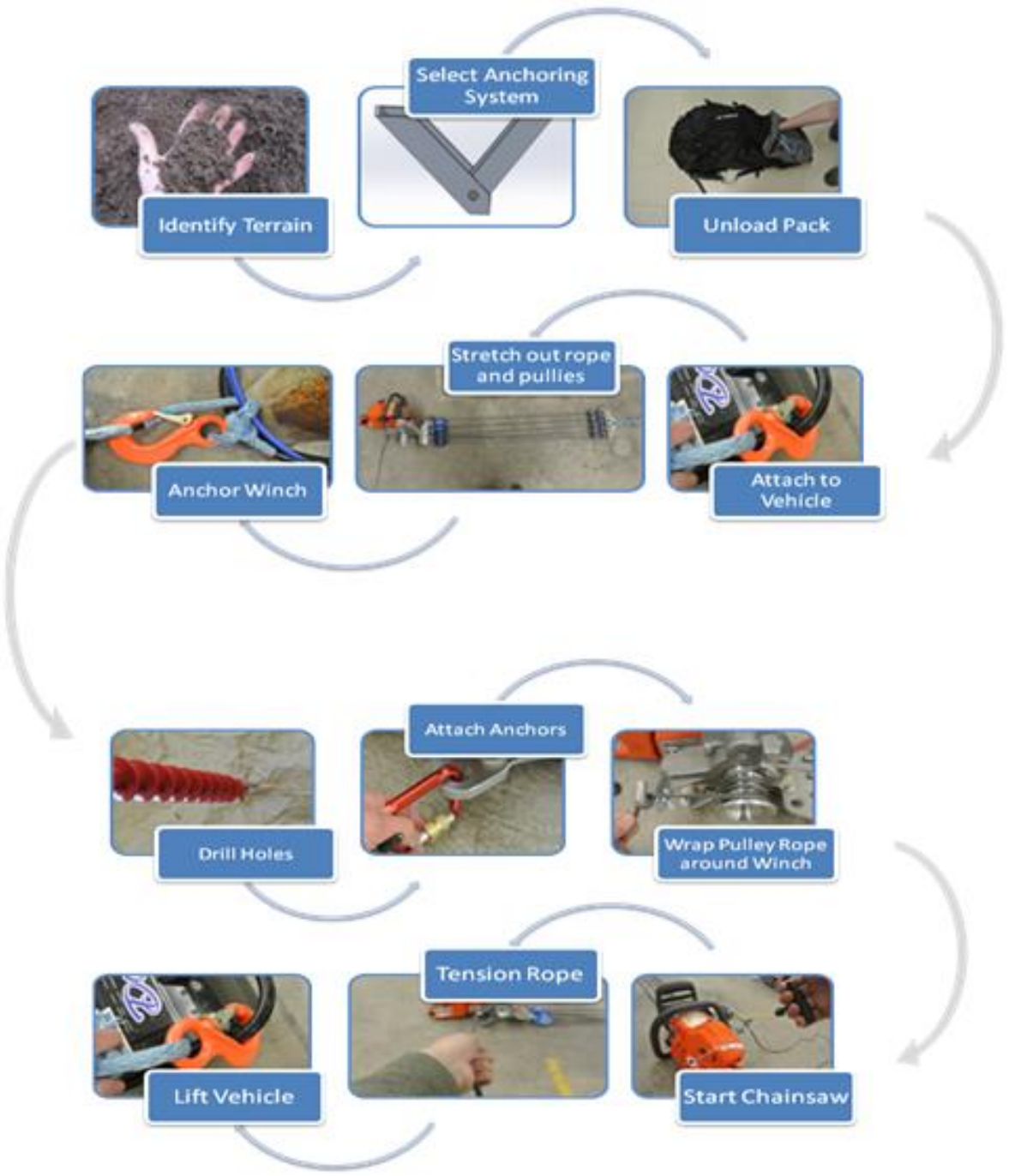


Figure 37: Operational Procedure

The following process will be implemented to lift the vehicle with the winch lifting device.

1. Identify terrain
2. Select proper anchoring system
3. Unload pack
4. Attach ½" rope to hard point on the opposite side of the vehicle
5. Stretch out ½" rope, pulleys and winch away from vehicle
6. Identify ideal location for anchor points
7. If a rigid structure is available, tie off ½" rope to structure (Skip steps 8-10)
8. Drill holes with auger
9. If needed, Drill additional holes next to vehicle to prevent slippage
10. Set anchors in holes
11. Attach anchors to winch and the vehicle
12. Remove chainsaw from auger and attach to winch
13. Wrap ¼" block and tackle rope around winch drum and remove slack from the pulleys
14. Start chainsaw
15. Provide constant tension to the rope wrapped around winch drum
16. Slowly increase the chainsaw to full throttle while maintaining tension on rope
17. Lift until desired height or vehicle is flipped over

Standards

ASME B30.26 – 2010, Rigging Hardware:

Applies to the construction, installation, operation, inspection and maintenance of detachable rigging hardware used for load handing activities including shackles, links, rings, swivels, eyebolts, wire rope clips and rigging blocks [1]. This standard applies to the rigging plate portion of the block and tackle and it applies to the shackles attached to the pulleys. Depending on the application, it also concerns the attachment to the vehicle.

ASME B30.7 – 2011, Winches:

B30.7 includes provisions that apply to the construction, installation, operation, inspection, testing and maintenance of winches arranged for mounting on a foundation or other supporting structure for moving loads. Winches addressed in this volume are those typically used in industrial, construction and maritime applications. The requirements included in this volume apply to winches that are powered by internal combustion engines, electric motors, compressed air or hydraulics and that utilize drums and rope [2]. The construction, installation, operation, inspection and testing sections of the standard directly applies to our system as the final winch was modified.

ISO 2307:2010, Rope Standard:

ISO 2307:2010 standard characterizes the method for determining linear density, lay strength, braid pitch, elongation, and breaking force. The winch lift elongation and breaking force were essential to understand to ensure the safety of the operators.

The elongation corresponds to the measured increase in length of the rope when the tension to which it is subjected is increased from an initial value (reference tension) to a value equal to 50 % of the minimum specified breaking strength of the rope. [3]

The breaking force is the maximum force registered (or reached) during a breaking test on the test piece, carried out on a tensile testing machine with constant rate of traverse of the moving element. The breaking force values given in the tables of rope specifications are only valid when this type of testing machine is used. [3]

System Comparison

The winch lift was also compared to the current heavy lift kit that the Air Force currently uses. Table 3 shows each of the criteria for which they were compared. The winch system was found to be a better option in all categories with an exception to the weight. The greatest enhancements are to the lifting height, required opening, and the minimum operating height. The winch lift is designed to attach to any hard point on the vehicle and room permitting will allow the operator to completely flip the vehicle over. Therefore there is no required opening, only a hard point to attach to on the outside of the vehicle, the total lifting height would be limitless in completely overturning the vehicle, and the minimum operating height is not a factor in this design compared to the airbags.

Table 3: Winch System Comparison to Current Lifting System

	Winch Lift System	Current System	Percent Change	Better/Worse Comparison
Total Weight (lbs)	80	56	-43%	Worse
Lift Capacity (lbs)	25000	20000	25%	Better
Capacity to Weight Ratio	368	357	3%	Better
Required Opening (in^2)	0	576	100%	Better
Lifting Height (in)	No Limit	20	-	Better
Min Operating Height (in)	0	3	100%	Better

Alternative Solutions

The initial design process was broken into two components, the lifting device and the shoring device. Many of these designs brought forth new unexpected complications that allowed the next design to be developed with a more effective solution. The final recommended design takes into consideration all of the weaknesses in the initial designs to develop an optimal solution for completing the objective. Supporting data for the alternative solutions can be found in Appendix B.

Fire Hose

One solution that was considered was a pressurized fire hose assembly to lift the vehicle. This pneumatic system consists of three fire hose air bags inflated by a battery powered air pump as shown in Figure 38. The Fire Hose Lifting Device was tested and failed to meet the required pressure to lift the 45,000lb object. The required theoretical pressure was ~265 psi. Under a ramp test of 10 psi every minute the Fire Hose Lifting Device failed at 120 psi. The failure was due to delamination between the flanges at the top and bottom of the hose due to improper clamping. The bags also did not collapse to 3 in., therefore not satisfying the competition requirements.

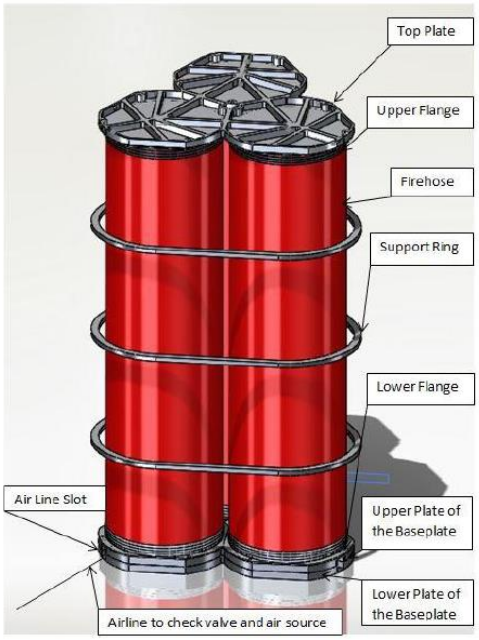


Figure 38: Fire Hose Lifting Bags

Hydraulic Tripod Lift

Hydraulic spreaders are commonly seen with Emergency Medical Technicians (EMTs), and lifting equipment such as forklifts. Figure 39 shows a single telescoping leg of the hydraulic tripod that would have been filled to 4,000 psi with hydraulic fluid by means of an electric pump. The lift would have required the P.J.s to carry approximately 2-3 gallons of hydraulic fluid which weighs up to 21 lbs. The electric pump could have been operated via rechargeable batteries; however the lightest pump capable of this operation weighs approximately 25 lbs.

Figure 39: Hydraulic Telescoping Leg

This lift would have been unable to lift from a near flat condition without a small pancake cylinder centered on the device which adds an additional 10lbs to the kit. Due to the obvious drawback of sheer weight the hydraulic tripod lift was no longer considered as a solution to the current problem. It was followed by the dangers associated with the flammability of hydraulic fluid and unsuccessful resolution to supporting the device in case of leaks or failures.

Scissor Lift

Using an airbag in the center and locking arms on the side; the scissor lift was designed to be the lifting and shoring device in a 12in X 12in X 3in box. Figure 40 and 41 shows the extended and collapsed scissor lift. The airbag placed in the center would act as the main lifting device while the legs of the lift were intended as a shoring device in case the airbag fails. The device would require about 200 psi of pressure provided by two nitrogen air tanks. The device would be capable of lifting 25,000 lbs up to 22 inches.

Two major problems with the design were acquiring an adequate airbag and a working locking mechanism. The required a custom airbag which costs $65,000 to manufacture and test which was not within the budget. For the shoring mechanism, the force due to even a ¼ inch fall could prove catastrophic to the supporting arms as the impact force resulted in much higher than the material breaking strength.

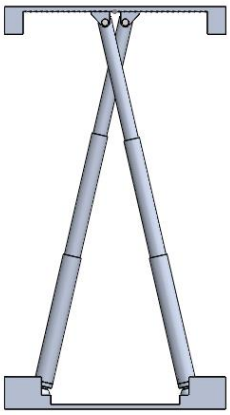


Figure 40: Extended Scissor Lift

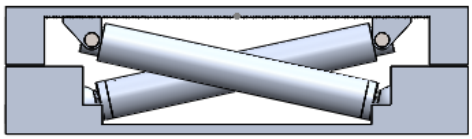


Figure 41: Collapsed Scissor Lift

Tripod Pawl

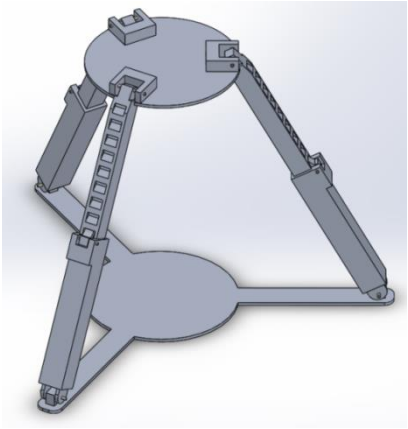


Figure 42: Tripod Pawl Shoring Assembly

This tripod shoring device would have been used in parallel with an airbag lifting system either in the middle of the tripod or externally as a standalone system. The concept is similar to a ratcheting pawl device on a gear system, but transverse in the linear direction. A tripod layout of the spring loaded pawl mechanism with one pawl per leg is shown in Figure 42. Figure 43 displays one option for a torsion spring design of the pawl. The pawl was designed with a curved profile to allow the translation of the force all the way along the entire pawl without risk of shear failure.

The main issue that is of concern with this design was the impact force on the legs if the airbags fail and the vehicle drops onto the shoring device. This design was not considered because the impact force proved to be too high for the legs to handle. At 0.5 inch fall, the impact force was roughly 47,000 lbf on each individual leg creating too much stress for the tripod to handle.

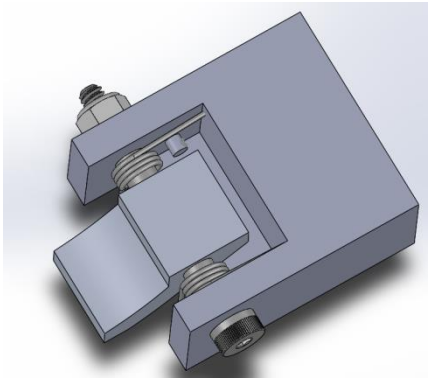


Figure 43: Spring Loaded Pawl Mechanism

Pulley Lift System

The pulley lift consists of a series of ten pulleys routing a cable to a winch device similar to the final solution winch system. This system differed from the final solution as it is all integrated into one lifting unit. Figure 44 displays the initial design of the pulley lifting device. The end effector forks on the bottom of the structure are connected to a rail system to allow for translation in the vertical direction. A set of 5 pulleys are integrated into the end effector and are attached with a rope to a set of fixed pulleys on the top of the structure creating a block and tackle system. The winch is powered by a DC brushless motor connecting to a gear reduction box. Figure 45 displays the application of the lift with the large block representing the vehicle load.

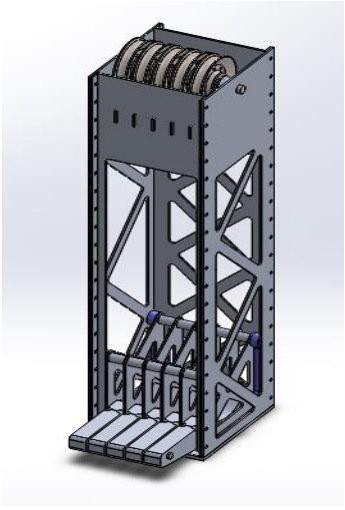


Figure 44: Pulley Lift Assembly

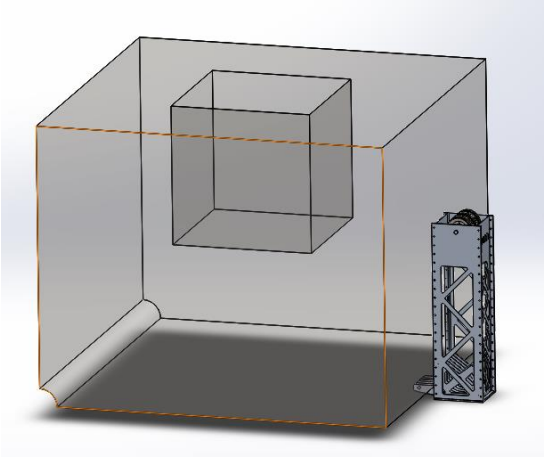


Figure 45: Pulley Lift Application

There were two main concerns that ultimately led to the dismissal of this design. The first concern was while the system was under high stresses during operation it would cause components to flex resulting in the end effector to bind in the track. This situation could avoided be switching to stronger materials, but that led into the second concern of the system being too heavy. Further designs were considered, but with the system already surpassing 50 lbs. using 7075-T6 Al, a simplification of the design was required resulting in the final winch solution.

System Comparison

The winch lift was chosen as the optimal design for the application at hand. To assist in choosing this design over the other alternatives a decision matrix was constructed shown in Table 4. Each of the alternatives that were explored was rated on a scale of 0-5, with 5 having the strongest relationship with the respective characteristic. This design matrix confirmed that the winch lift was the optimal design choices for the problem at hand.

Table 4: Lifting System Decision Matrix

Lift	Functional	Ease of Use	Weight	Compact Size	Cost	Manufacturability	Total
Fall Semester Design	0	2	3	2	1	1	**9**
Ratchet/Airbag	0	2	3	4	1	1	**11**
Hydraulic Lift	3	2	1	4	3	3	**16**
Scissor Lift	0	3	3	4	1	1	**12**
Backpack	2	3	2	2	4	3	**16**
Winch Lift	5	4	2	2	4	5	**22**

Cost Analysis

Table 5 displays a high level overview of the budget for this project. A total of $8,829.99 was spent on the entire project including testing, prototype components, and expenses from the previous semester. The cost associated with building one winch system is $3,917.13. Table 6 in Appendix A shows the budget breakdown of each component in the winch system. Table 7 in Appendix A is a complete expanded budget for the entire project.

Table 5: Winch System Total Project Budget

Category	Lift Assembly	Testing	Total
Raw Material	$226.20	$544.86	$771.06
Block and Tackle	$685.68	$311.25	$996.93
Gas Powered Winch	$1,189.22	$668.95	$1,858.17
Attachment/Anchor	$1,627.51	$87.80	$1,715.31
Miscellaneous	$188.52	$300.00	$488.52
Previous Costs	$-	$-	$2,990.94
TOTAL	$3,917.13	$1,912.86	$8,820.93

Competition Results

On April 17, 2014 Bulldog Innovations traveled to Arnold Air Force Base located in Manchester, Tennessee for the Air Force Research Laboratory Competition. The presentation yielded profound interest in the winch lift design from the PJs and other judges. The pulley winch system offered a unique solution that differed from the current Air Force lift application.

The demonstration portion of the competition displayed the application of our lift system. Failure resulted during the lifting process due to an insufficient quantity of anchors. The remainder of the system performed up to expectations by bringing the load nearly to the lifting point of the vehicle. Future research and design should be focused on the anchoring system as well as soil mechanics.

Future Recommendations

Moving forward, the main recommendation is to create a lighter, smaller version of the lift. By minimizing the material used throughout the system, the weight could be considerably reduced. Combining the motor and winch into one assembly and utilizing a planetary gear reduction would allow the overall size and weight of the system to be condensed. Composite materials could potentially be used for the block and tackled to further reduce weight. To reduce the packing size, the components could be redesigned to fit together inside a compact pack custom made for the direct placement of components into designated spots.

One of the best applications for this lift system is on a vehicle itself. This system could be adapted to mount onto another vehicle for a quicker setup time. An electric motor could be attached to the winch, drawing electricity from the vehicle's power system. The block and tackle portion of the lift could fit in a small box in the back of the vehicle, and could easily be attached if the weight reduction is required. Some anchoring may still be required if the vehicles own weight isn't adequate to support the load.

Summary and Conclusion

The United States Air force requested an improvement in their current heavy lifting device of a series of air bags. The device must be able to lift a 45,000 lb. vehicle up to 20 inches. Bulldog Innovations considered several designs including an extending tripod, specialized airbags, a hydraulic lift, hybrid scissor lift, and a pulley lift system. These systems proved to be unrealistic due to many issues including weight, buckling under high impact forces, and cost.

Bulldog Innovations developed a new alternative for heavy lifting devices for the U.S. Air Force. The new lift system consists of four main components; a winch, block and tackle, gas motor, and anchors. A ½" synthetic rope is secured to a hard point location on the vehicle. The opposite end of the ½" rope is routed to the block and tackle system. The block and tackle consists of ten pulleys to develop a mechanical advantage through the load distribution of ten separate parts of the rope. This allows the initial load of the system to be reduced to only 2,250 lbs.

To generate the required tension on the line to lift the vehicle, the ¼" rope is routed through a Capstan winch powered by a gas chainsaw. The winch utilizes a 125:1 gear reduction to reduce the chainsaw speed and in return provide the required torque. A Husqvarna 460 Rancher supplies the required displacement to operate the winch at the required load. To restrain the winch system, terrain adaptable anchors utilized.

This lifting system design eliminates the need for a shoring device. The shoring device's function is to provide a failsafe device in case of failure to the main lifting system. The winch systems failure point will only be upon the initial lift therefore allowing little to no drop of the vehicle. A shoring device would only be necessary in the event of the vehicle failing to completely overturn.

This system provides advantages to the current lift system in operation. Compared to the current air bag lift system used by the U.S. Air Force, the winch lift offers an increase in increased lifting capacity by 25%, capacity to weight ratio by 3%, required operating surface area by 100%, lifting height by 100%, and the minimum operating height by 100%. Bulldog Innovations recommends the winch system for Air Force rescue applications as it is an efficient, portable, and easily manufacturable option for lifting an overturned military vehicle.

The competition objective was not completed due to an insufficient quantity of anchors. The remainder of the system performed as designed and would have effectively lifted the vehicle with the proper anchoring system. The final solution from Bulldog Innovations peaked strong interest from the PJs due to the unique application from their current lifting techniques.

Acknowledgment

On behalf of the University of Minnesota Duluth, Bulldog Innovations would like to thank the United States Air Force and Air Force Research Laboratory for this opportunity. It has been an honor to partake in a competition such as this and to provide a tool that could potentially be used in the World's Greatest Air Force. This design has given us the experience and tools to take into the workforce to be successful.

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[1] ASME. (2010). *Rigging Hardware: B30.26 - 2010.* Retrieved from www.amse.org: https://www.asme.org/products/codes-standards/b3026-2010-rigging-hardware

[2] ASME. (2011). *Winches: B30.7 - 2011.* Retrieved from www.asme.org: https://www.asme.org/products/codes-standards/b307-2011-winches-formerly-titled-basemounted

[3] ISO Standard. (2010). *www.iso.org.* Retrieved from ISO Standards Web Site: http://www.iso.org/iso/home/store/catalogue_tc/catalogue_detail.htm?csnumber=52347

Appendix A: Final Solution Additional Resources

FEA Analysis

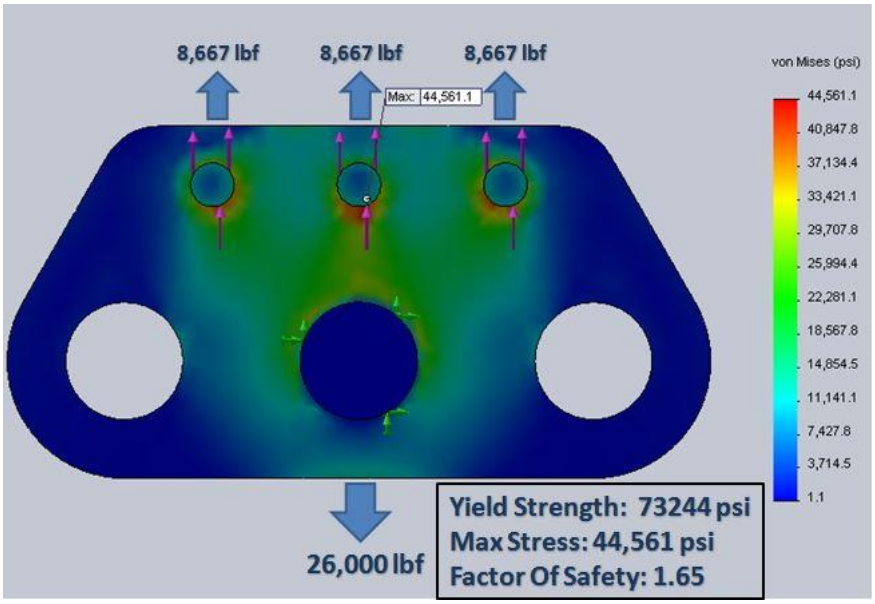


Figure 46: Anchor Rigging Plate von-Mises Stress Plot

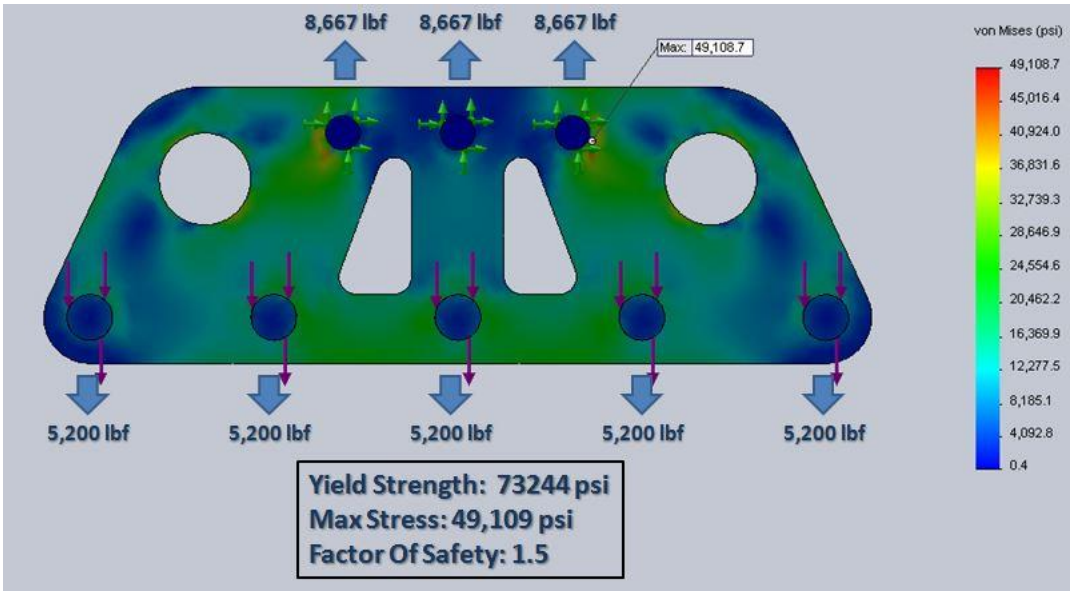


Figure 47: Block and Tackle Bottom Plate von-Mises Stress Plot

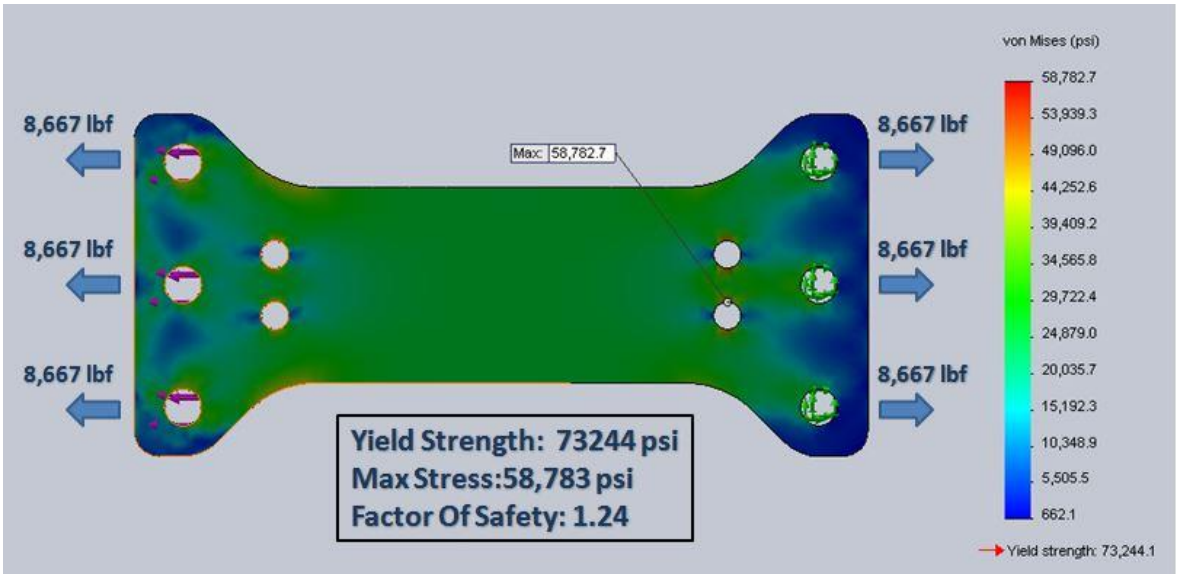


Figure 48: Winch Mounting Plate von-Mises Stress Plot

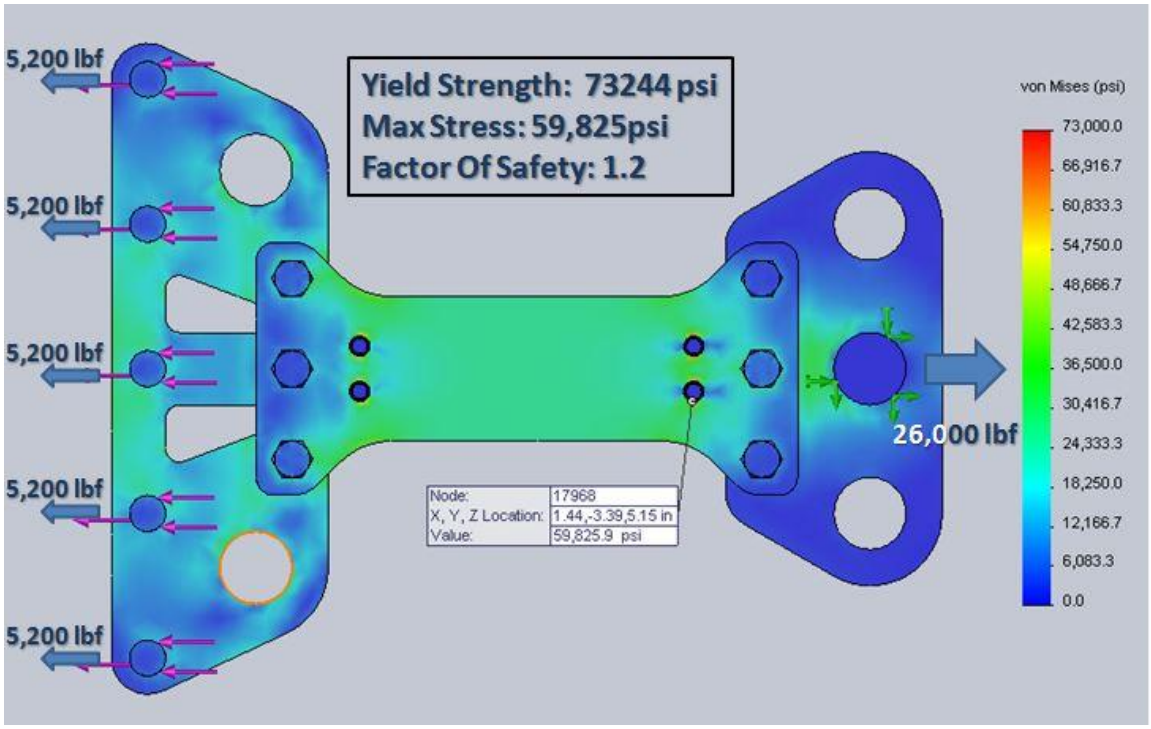


Figure 49: Winch Mounting Assembly von-Mises Stress Plot

Tensile Testing Results

Initial Shoring Designs

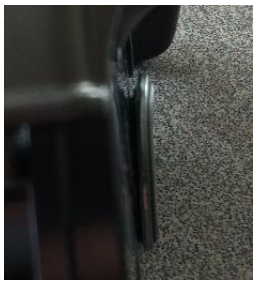


Figure 50: Standard Belay Device Testing

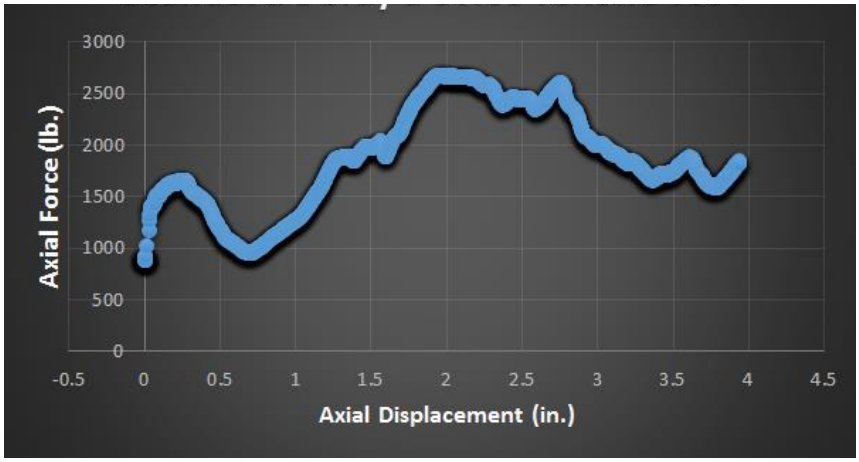


Figure 51: Standard Belay Test Results

Figure 53: Custom Belay Device

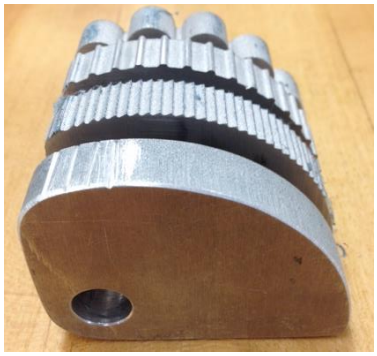


Figure 52: Various Cam Options for Custom Belay Device

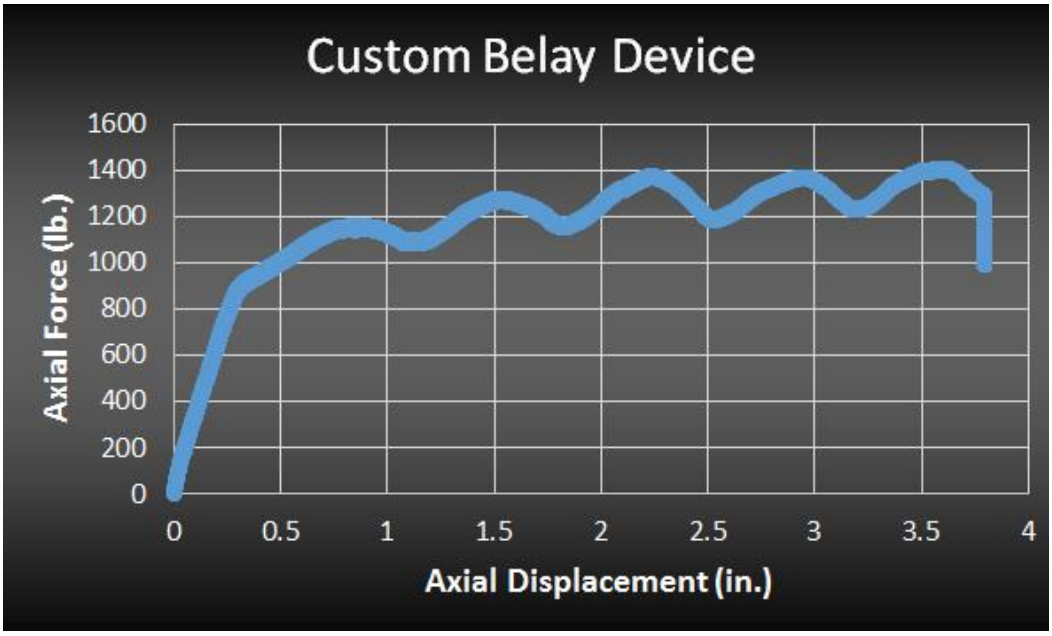
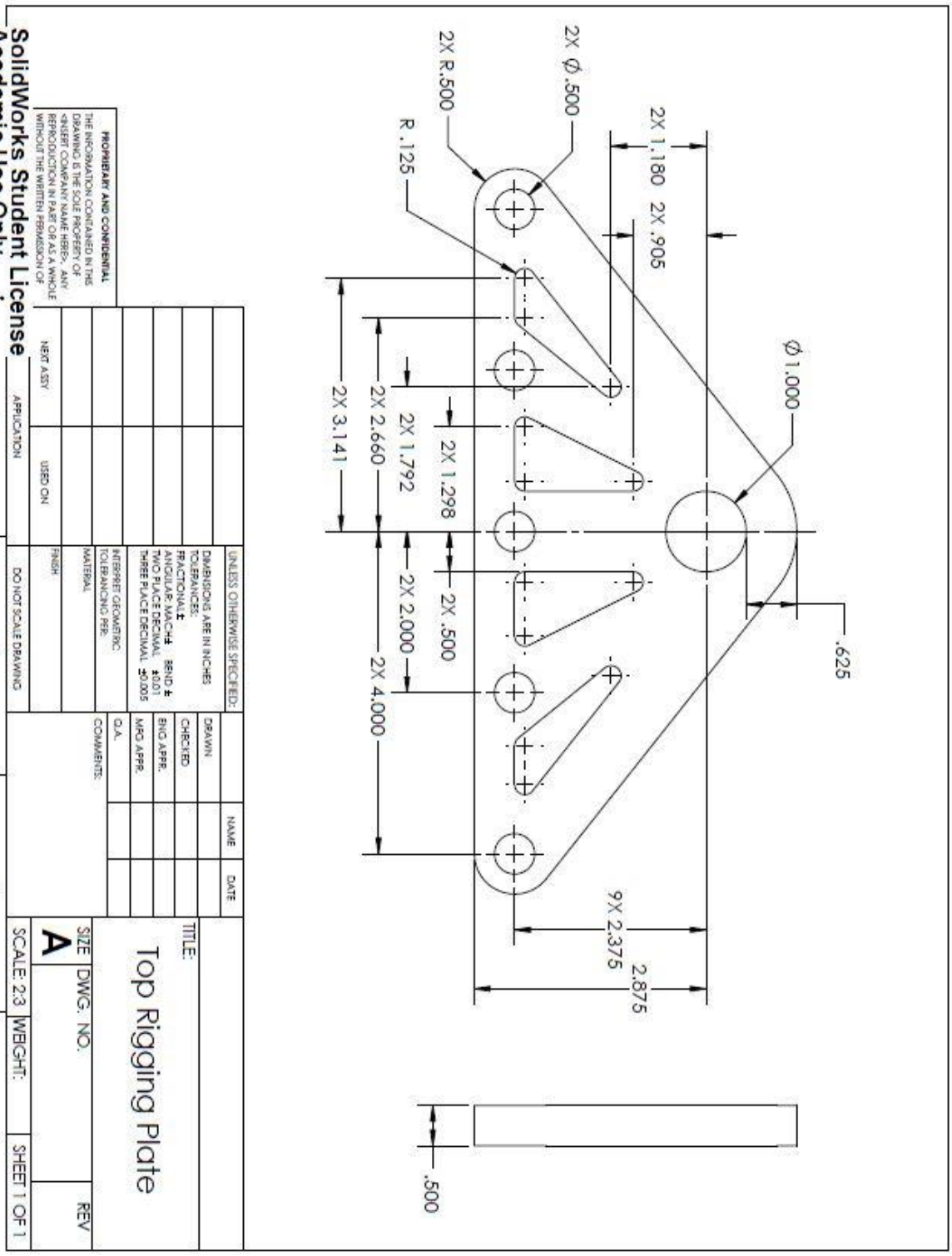
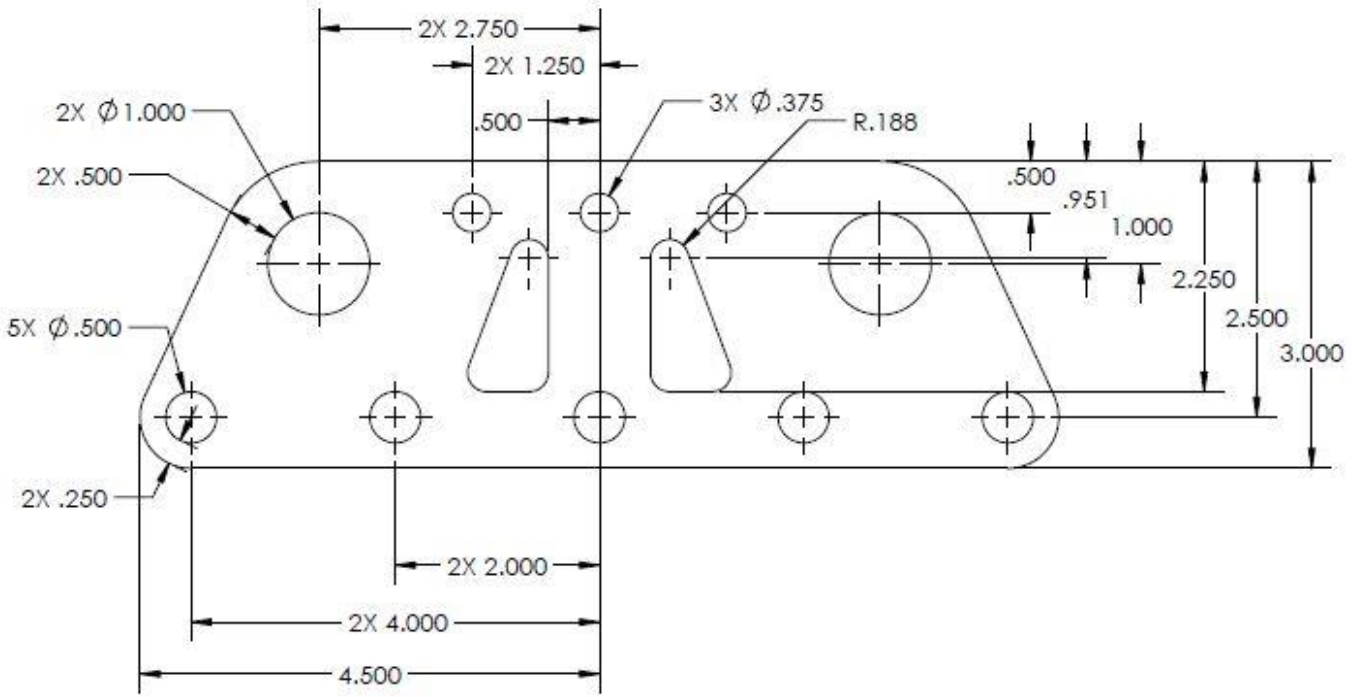


Figure 54: Custom Belay Device Test Results

*Note: Low force results were due to slipping of rope in device not failure of components

Engineering Drawings





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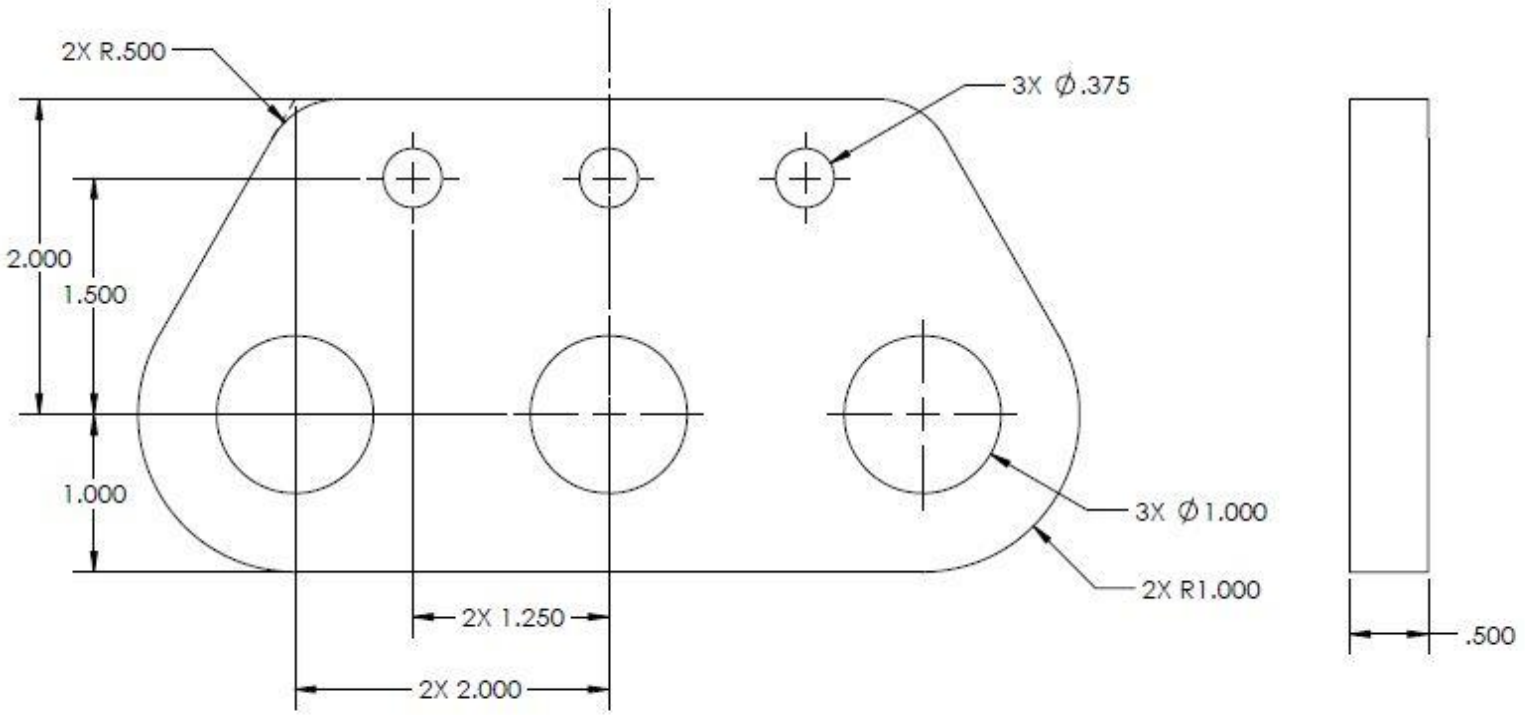
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ±0.01 THREE PLACE DECIMAL ±0.005	DRAWN		
			CHECKED		
			ENG APPR.		
			MFG APPR.		
			Q.A.		
		INTERPRET GEOMETRIC TOLERANCING PER:	COMMENTS:		
		MATERIAL			
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE: Rigging Plate Bottom

SIZE	DWG. NO.	REV
A		

SCALE: 2:3 | WEIGHT: | SHEET 1 OF 1

4 3 2 1

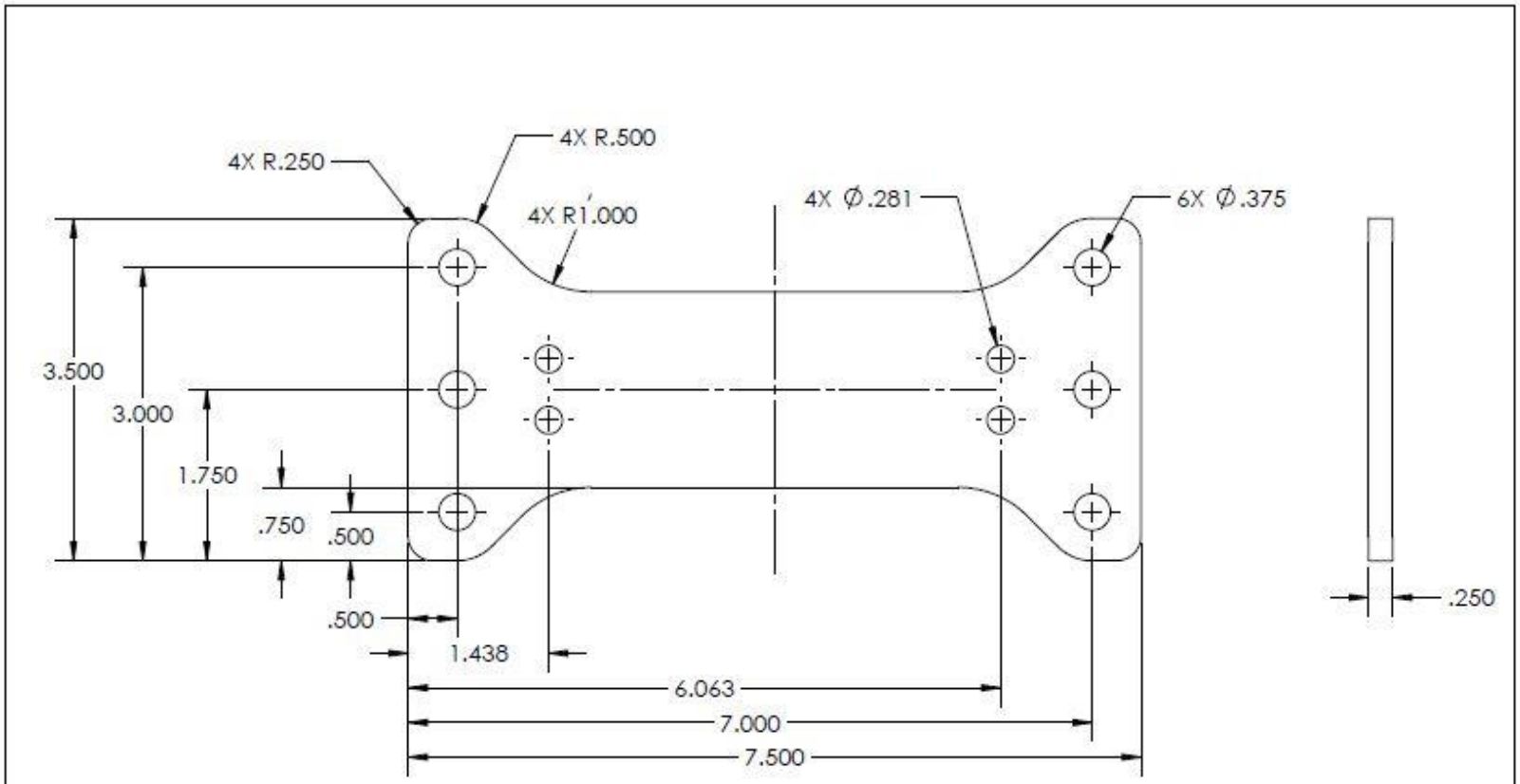


		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ±0.01 THREE PLACE DECIMAL ±0.005	DRAWN		
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE: Anchor Rigging Plate

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

4 3 2 1



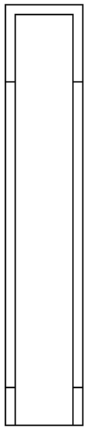
		UNLESS OTHERWISE SPECIFIED:		NAME	DATE
		DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ±0.01 THREE PLACE DECIMAL ±0.005	DRAWN		
			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

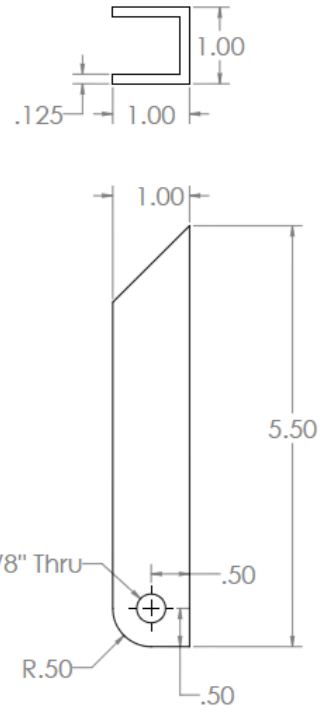
TITLE: Winch Plate

SIZE	DWG. NO.	REV
A		
SCALE: 2:3	WEIGHT:	SHEET 1 OF 1

4 3 2 1

1" Square Stock
Remove interior of one side





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

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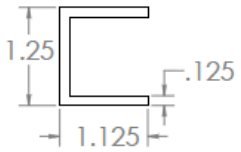
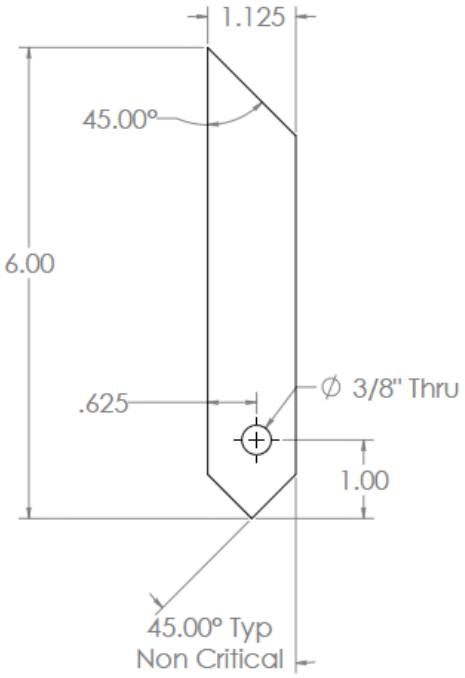
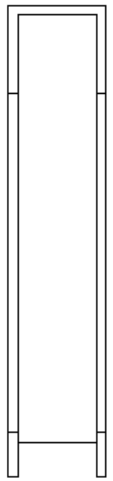
TITLE:

SIZE	DWG. NO.	REV
C	Interior	
SCALE: 1:1	WEIGHT:	SHEET 1 OF 1

1 1/4" Square Stock
Remove one side







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

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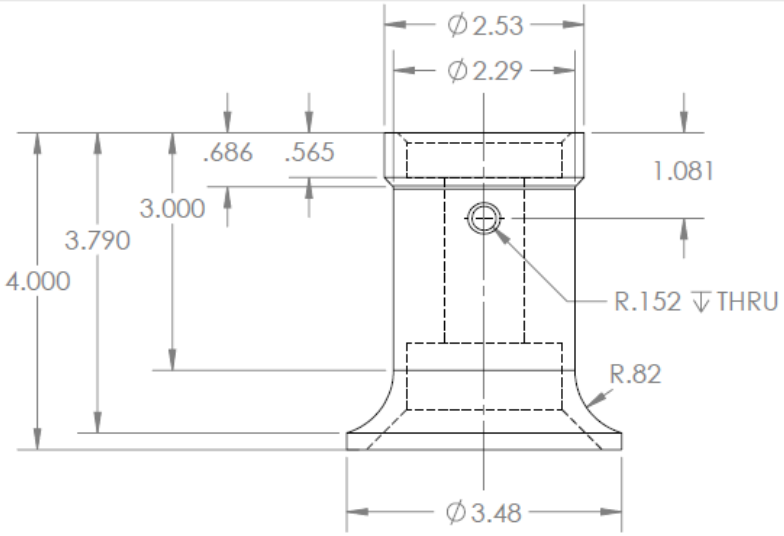
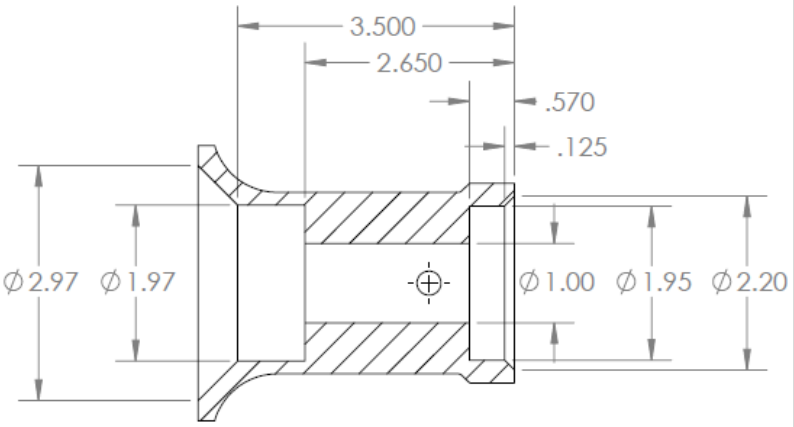
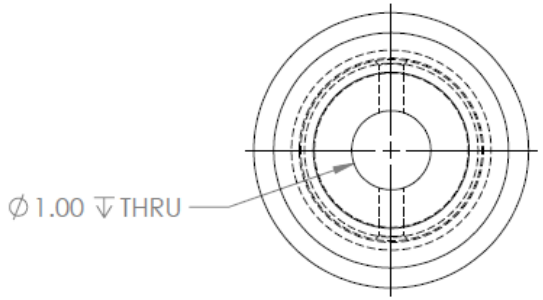
TITLE:

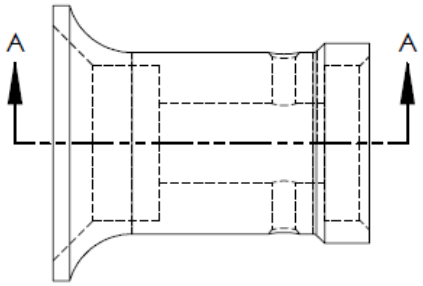
SIZE	DWG. NO.	REV
C	exterior	
SCALE: 1:1	WEIGHT:	SHEET 1 OF 1





SECTION A-A





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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

SIZE | DWG. NO. | REV

A Winch Spindle

SCALE: 1:2 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1

Cost Analysis

Table 6: Single Winch System Budget

Assembly Budget	Description	Cost	Quantity	Shipping	Total
Raw Material					**$ 226.20**
	1/4" x 12" x 12" 7075-T6 Aluminum Plate	$ 41.89	1	$ 16.39	$ 58.28
	0.5" x 4" x 36" 7075-T6 Aluminum Flat Bar	$ 92.77	1	$ 13.86	$ 106.63
	1.25" x 8ft 6061 Aluminum Square Tube	$ 25.87	1	$ 18.05	$ 43.92
	1" x 8ft 6061 Aluminum Square Tube	$ 17.37	1	$ -	$ 17.37
Block and Tackle					**$ 685.68**
	Pulleys	$ 52.25	9	$ 12.00	$ 482.25
	1/4"x 150ft Amsteel Blue Rope	$ 138.00	1	$ -	$ 138.00
	3/8" Twisted Clevis	$ 5.50	9	$ 15.93	$ 65.43
Gas Powered Winch					**$ 1,189.22**
	Winch	$ 668.95	1	$ 20.28	$ 689.23
	Chainsaw, 60.3 CC	$ 499.99	1	$ -	$ 499.99
Attachment/Anchor					**$ 1,627.51**
	2" Earth Auger	$ 115.00	1	$ 12.97	$ 127.97
	Lewis MultiDrill	$ 575.00	1	$ -	$ 575.00
	1/2" x 150 ft Amsteel Blue Rope	$ 662.00	1	$ -	$ 662.00
	1/4"x 150ft Amsteel Blue Rope	$ 138.00	1	$ -	$ 138.00
	Hooks	$ 84.00	1	$ 20.00	$ 104.00
	1/4" Thimbles	$ 0.67	10	$ -	$ 6.70
	1/2" Thimbles	$ 3.46	4	$ -	$ 13.84
Miscellaneous					**$ 188.52**
	Velcro Rope Straps	$ 8.53	1	$ -	$ 8.53
	Backpack	$ 179.99	1	$ -	$ 179.99
TOTAL				**TOTAL**	**$ 3,917.13**

Table 7: Entire Project Budget

Complete Budget	Description	Cost	Quantity	Shipping	Total
Raw Material					**$ 771.06**
	1/4" x 24" x 32" 7075-T6 Aluminum Plate	$ 171.85	1	$ 16.39	$ 188.24
	1" x 4" x 12" 7075-T6 Aluminum Flat Bar	$ 64.81	1	$ -	$ 64.81
	1" x .88" x 8 Ft 2024 Aluminum Tube	$ 121.00	1	$ 32.12	$ 153.12
	1.5" x 1.26" x 8 ft 2024 Aluminum Tube	$ 125.40	1	$ -	$ 125.40
	0.375" x 3ft 4340 Steel Rod	$ 11.84	1	$ -	$ 11.84
	1" x 3ft 4340 Steel Rod	$ 41.34	1	$ -	$ 41.34
	0.5" x 4" x 36" 7075-T6 Aluminum Flat Bar	$ 92.77	1	$ 13.86	$ 106.63
	1.25" x 8ft 6061 Aluminum Square Tube	$ 25.87	1	$ 18.05	$ 43.92
	1" x 8ft 6061 Aluminum Square Tube	$ 17.37	1	$ -	$ 17.37
	3/8" Spring, 6 Pack	$ 6.13	3	$ -	$ 18.39
Block and Tackle					**$ 996.93**
	Pulleys	$ 52.25	12	$ 12.00	$ 639.00
	1/4"x 300ft Amsteel Blue Rope	$ 276.00	1	$ -	$ 276.00
	3/8" Twisted Clevis	$ 5.50	12	$ 15.93	$ 81.93
Gas Powered Winch					**$ 1,858.17**
	Winch	$ 668.95	2	$ 20.28	$ 1,358.18
	Chainsaw, 60.3 CC	$ 499.99	1	$ -	$ 499.99
Attachment/Anchor					**$ 1,715.31**
	2" Earth Auger	$ 115.00	2	$ 12.97	$ 242.97
	Lewis MultiDrill	$ 575.00	1	$ -	$ 575.00
	1/2" x 150 ft Amsteel Blue Rope	$ 662.00	1	$ -	$ 662.00
	Hooks	$ 84.00	2	$ 20.00	$ 188.00
	1/4" Thimbles	$ 0.67	50	$ -	$ 33.50
	1/2" Thimbles	$ 3.46	4	$ -	$ 13.84
Miscellaneous					**$ 488.52**
	Velcro Rope Straps	$ 8.53	1	$ -	$ 8.53
	Backpack	$ 179.99	1	$ -	$ 179.99
	Carabineers	$ 21.41	10	$ -	$ 214.10
	Belay Device	$ 85.90	1	$ -	$ 85.90
TOTAL				**TOTAL**	**$ 5,829.99**

Appendix B: Alternative Solution Additional Analysis

Fire Hose Lift

Figure 55: Fire Hose Sample

Figure 56: Upper and Lower Plates, respectively

Figure 57: Top and Bottom Flange

Figure 58: Fire Hose Assembly

Figure 59: Delaminated Fire Hose Post Test

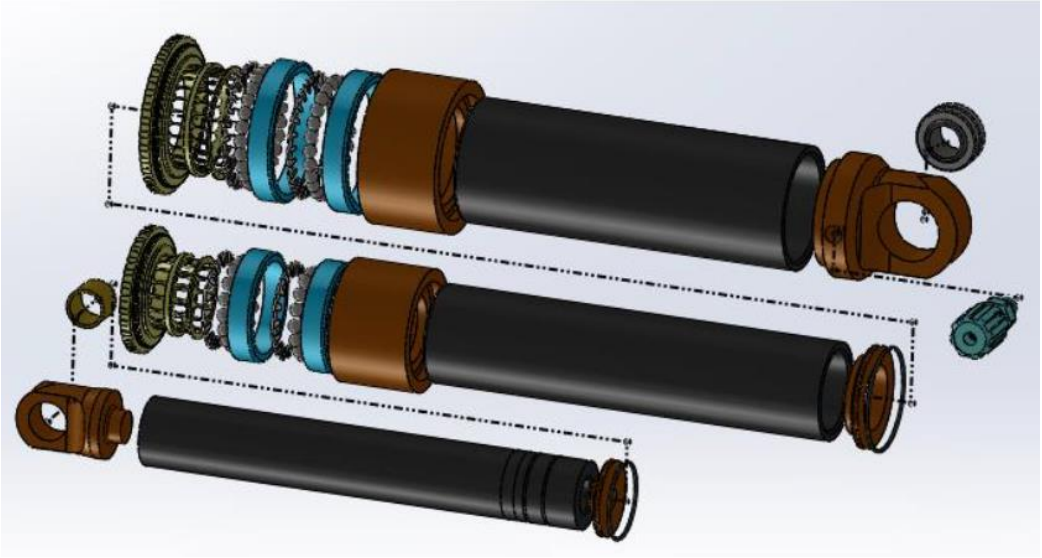

Figure 60: Tripod Self-Locking Leg - Exploded View

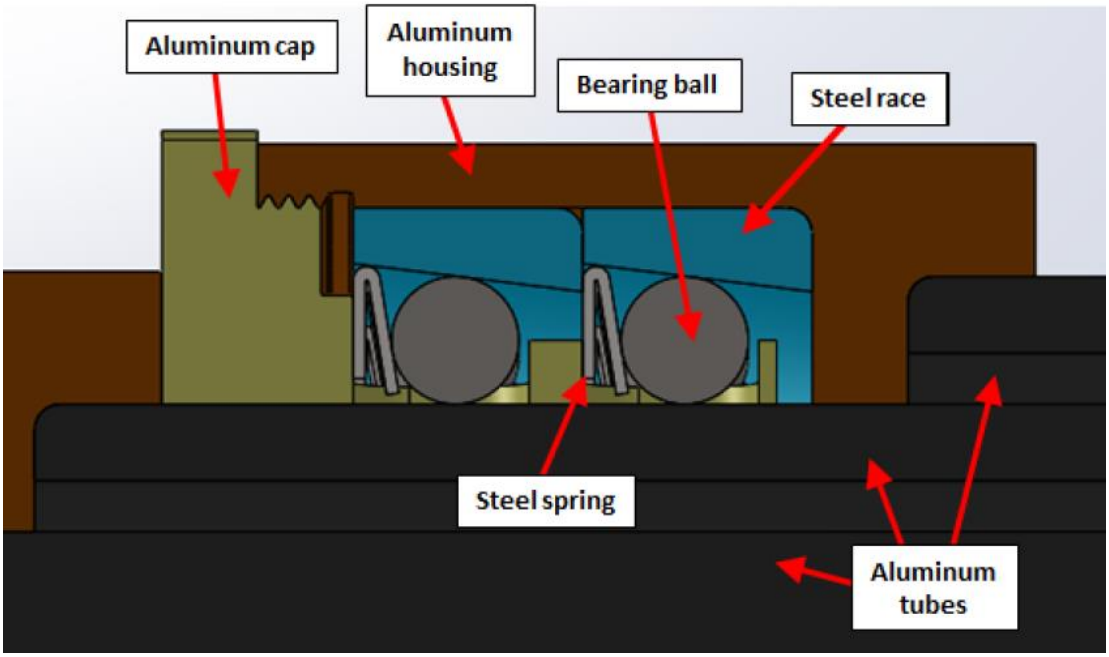

Figure 61: One-way bearing Cross-Section View

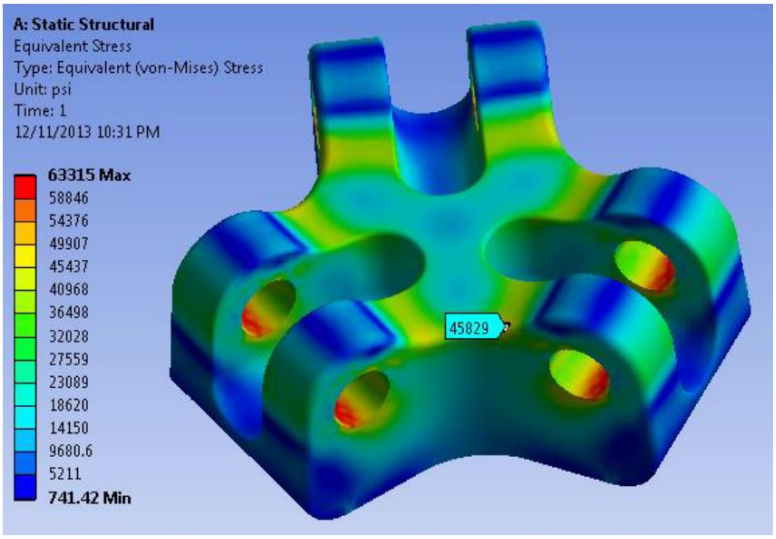

Figure 62: Tripod Top Plate von-Mises Stress Plot

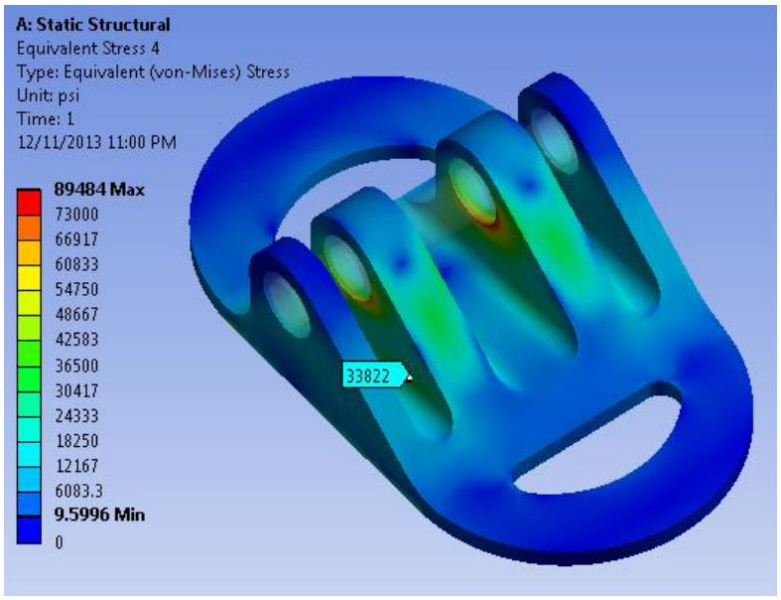

Figure 63: Tripod Lower Plate von-Mises Stress Plot

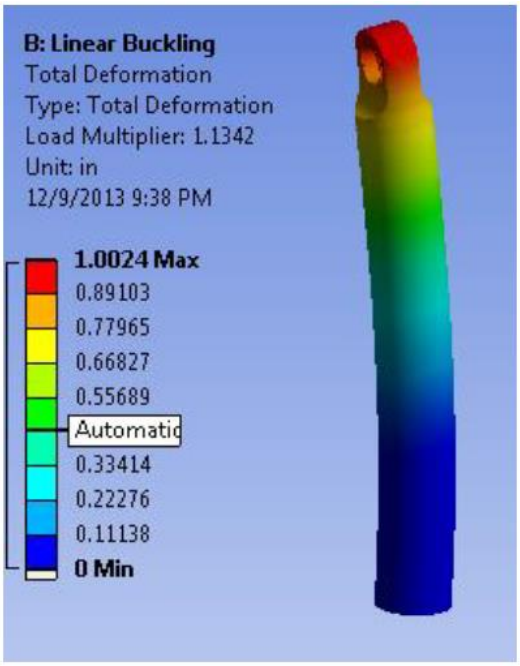

Figure 64: Tripod Upper Tube Assembly Buckling Deformation Plot

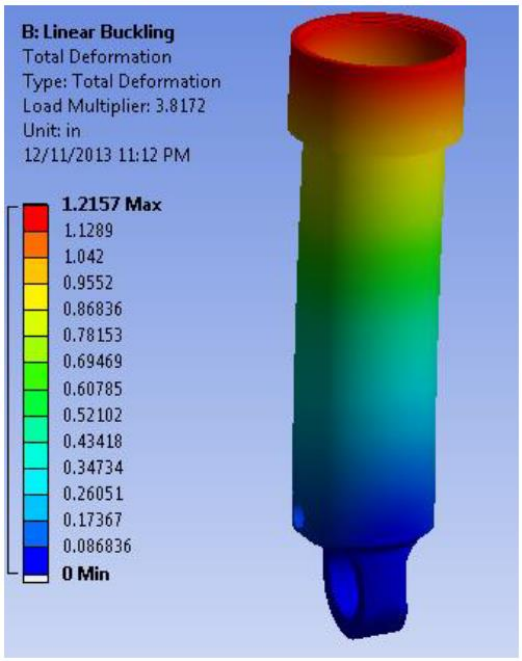

Figure 65: Tripod Lower Tube Assembly Buckling Deformation Plot

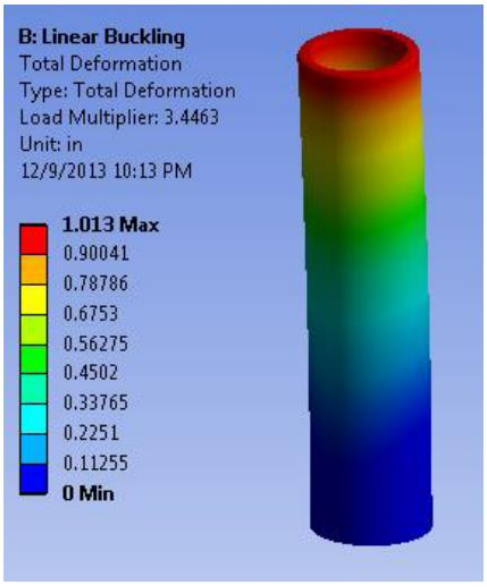

Figure 66: Tripod Middle Tube Buckling Defermation Plot

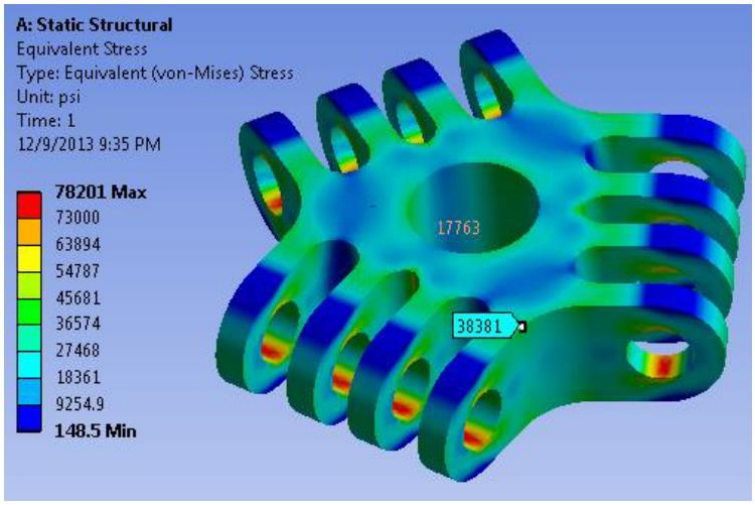

Figure 67: Tripod Center Mount von-Mises Stress Plot

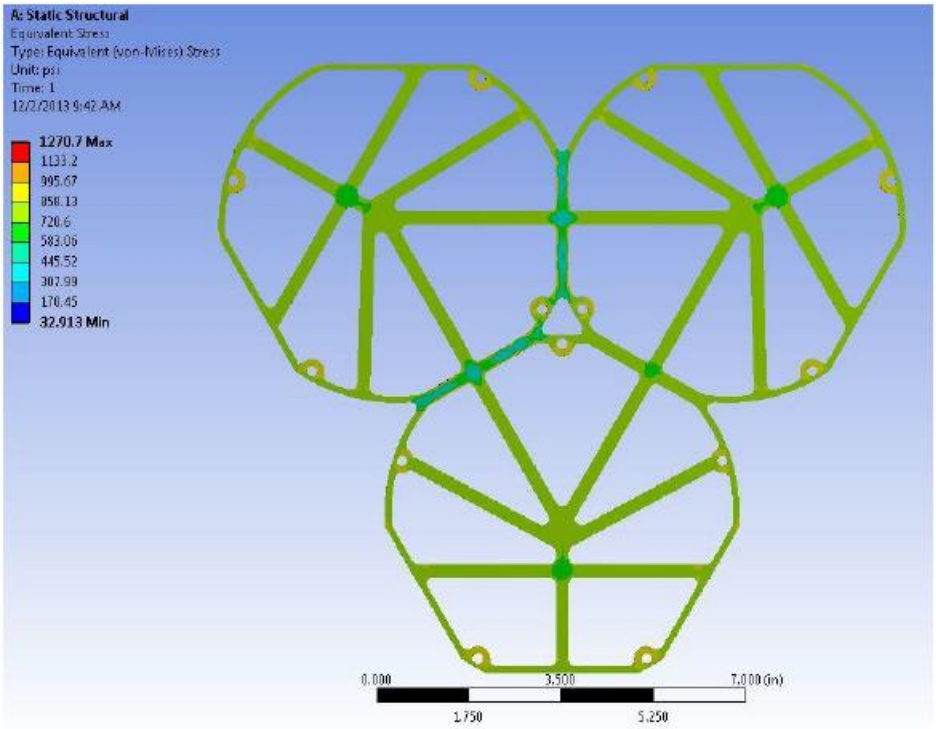

Figure 68: Fire Hose Top Plate von-Mises Stress Plot

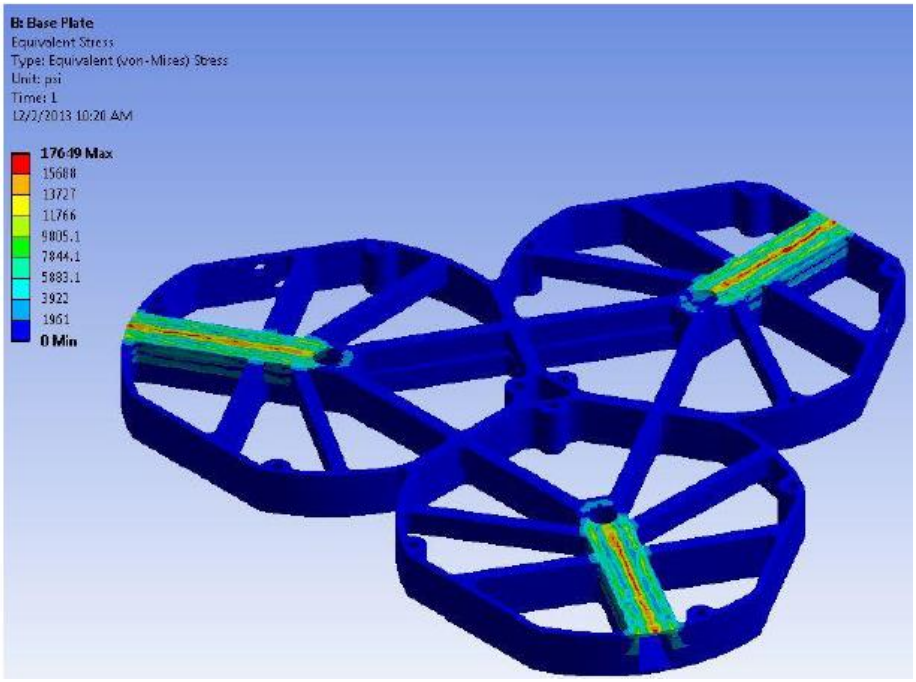

Figure 69: Fire Hose Bottom Plate von-Mises Stress Plot

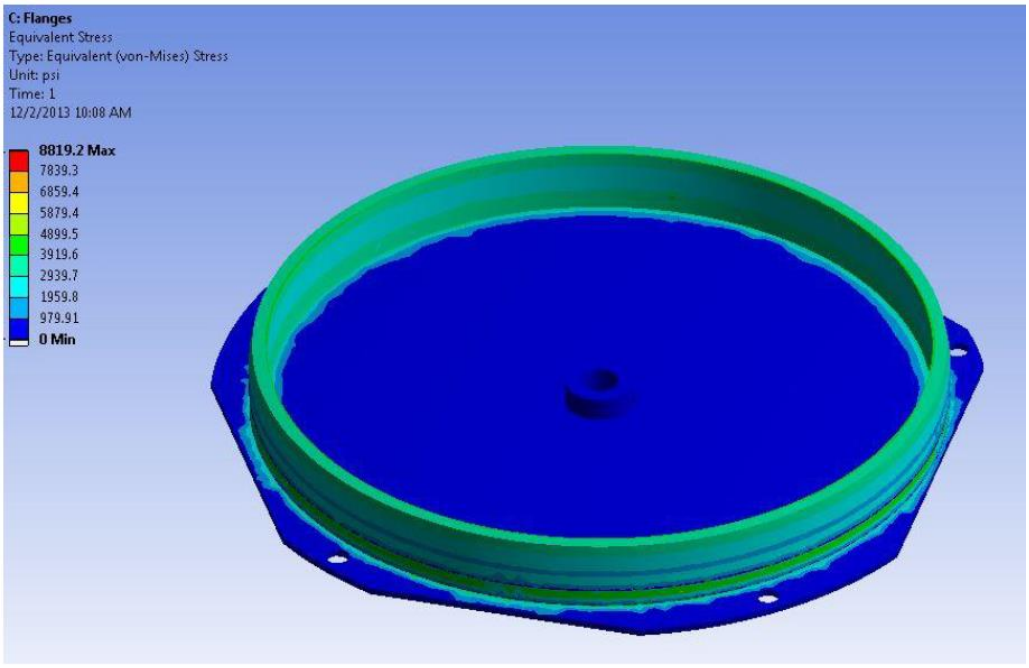

Figure 70: Fire Hose Flange von-Mises Stress Plot

Custom Air Bag Requirements

Air Bag requirments

Given:

$F := \frac{55000}{2} \cdot lbf$ $r := 6\ in$ Max Lifting Weight Weight

$A := \pi \cdot r^2 = 113.097\ in^2$ Max Lifting Area

$h := 20\ in$ Max Lifting Height

Calculations

Required Pressure

$p := \frac{F}{A} = 243.153\ psi$

Required Volume

$V := A \cdot h = \left(2.262 \cdot 10^3\right)\ in^3$

Required Mass of Nitrogen

$MM := .028\ \frac{kg}{mol}$

$R := 8.3144\ \frac{J}{mol \cdot K}$

$T := 300\ K$

$D := \frac{MM \cdot p}{R \cdot T}$

$M := V \cdot D = 1.538\ lbm$

Mass from Tank

$V_t := 90\ in^3$

$p_t := 4500\ psi$

$D_t := \frac{MM \cdot p_t}{R \cdot T}$

$M_t := V_t \cdot D_t = 1.132\ lbm$

Required Number of Tanks

$N := \frac{M}{M_t} = 1.358$

$V_t := \frac{M}{D_t} = 122.222\ in^3$

Required Air Volume

$p_r := 14.696\ psi$

$D_r := \frac{MM \cdot p_r}{R \cdot T} = 0.071\ \frac{lbm}{ft^3}$ $V_r := \frac{M}{D_r} = 21.658\ ft^3$

Spring Loaded Pawl

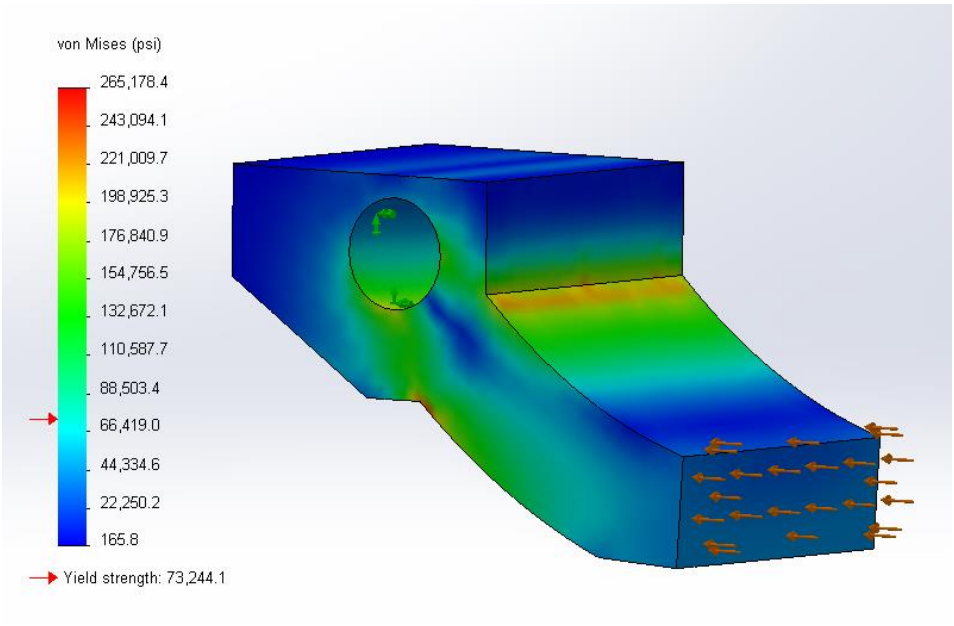

Figure 71: Spring Loaded Pawl von-Mises Stress Plot

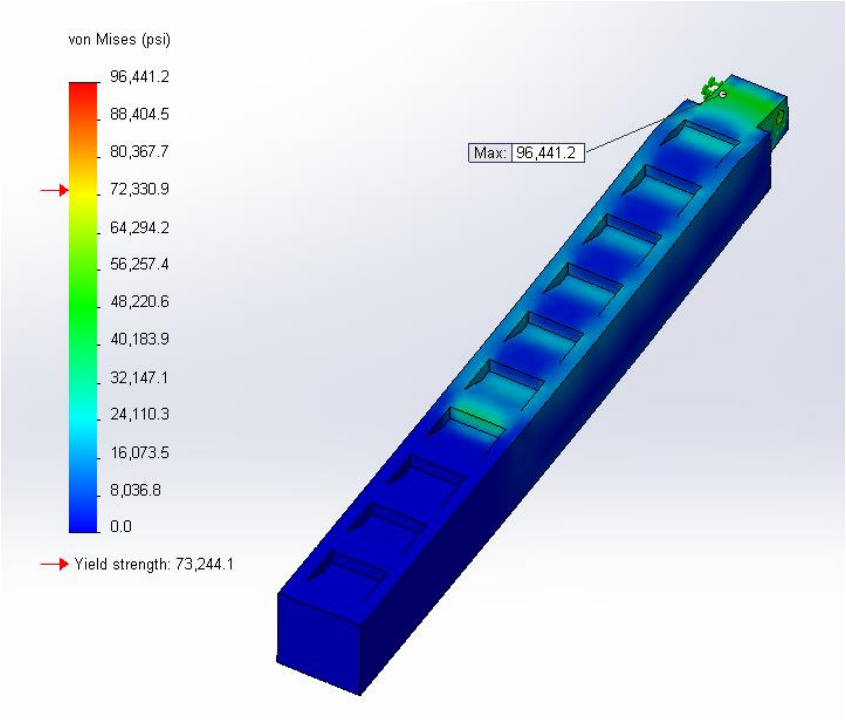

Figure 72: Spring Loaded Pawl Upper Leg von-Mises Stress Plot

Static Force on Each Leg

$n := 3$

$F := 27500\ lbf$

$\theta := 30\ deg$

$F_\theta := \frac{F}{n \cdot \sin(\theta)} = (1.833 \cdot 10^4)\ lbf$

Drop Height

$L_{base} := 9\ in$

$L_{leg} := \frac{L_{base}}{\cos(\theta)} = 10.392\ in$

$h := \tan(\theta) \cdot L_{base} = 5.196\ in$

$\Delta L := .5\ in$

$L_{legdrop} := L_{leg} - \Delta L = 9.892\ in$

$h_{drop} := \sqrt{\left(-L_{base}^{\ 2} + L_{legdrop}^{\ 2}\right)} = 4.106\ in$

$s := .5 \cdot in$

$\Delta h := s + h - h_{drop} = 1.59\ in$

Impact Force

$m := \frac{45000\ lbm}{2} = (2.25 \cdot 10^4)\ lbm$

$F_{fall} := \frac{m \cdot g \cdot \Delta h}{s} = (7.157 \cdot 10^4)\ lbf$

$F_{fallperleg} := \frac{F_{fall}}{n \cdot \sin(\theta)} = (4.771 \cdot 10^4)\ lbf$

Pulley Lift System

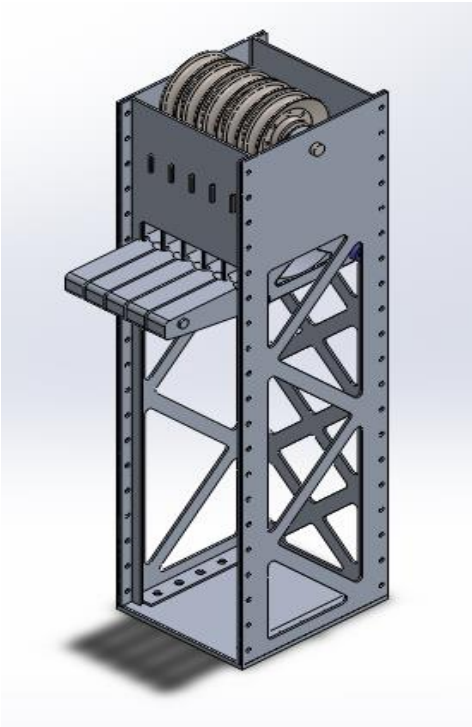

Figure 73: Pulley Lift Assembly

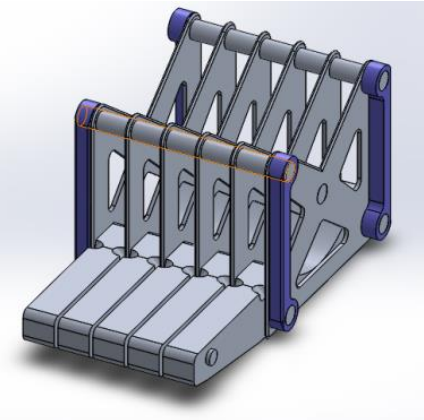

Figure 74: Pulley Lift End Effector

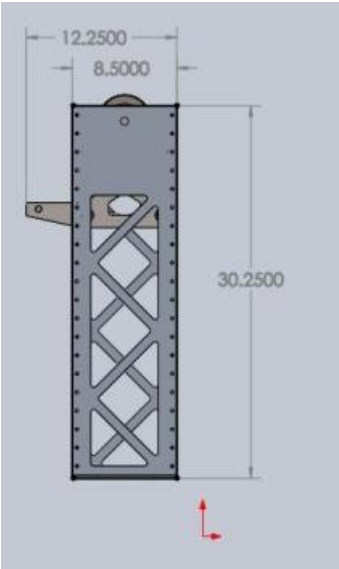

Figure 75: Pulley Lift Side Base Dimensions (in)

Figure 76: Pulley Lift Front Side Base Dimensions (in)

Figure 77: Pulley Lift Backpack Mounting Frame

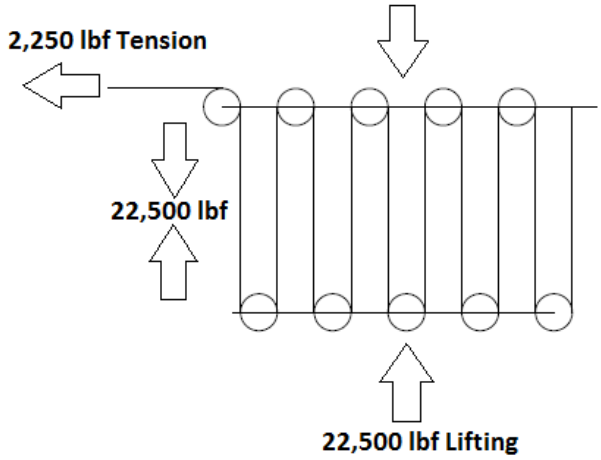

Figure 78: Pulley Lift System Force Diagram

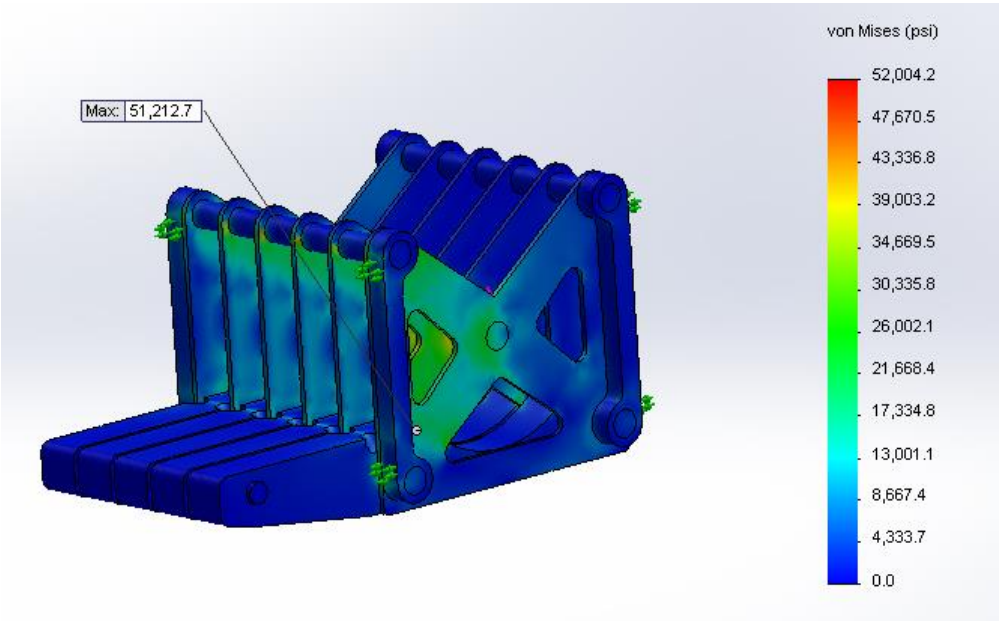

Figure 79: Pulley Lift End Effector von-Mises Stress Plot

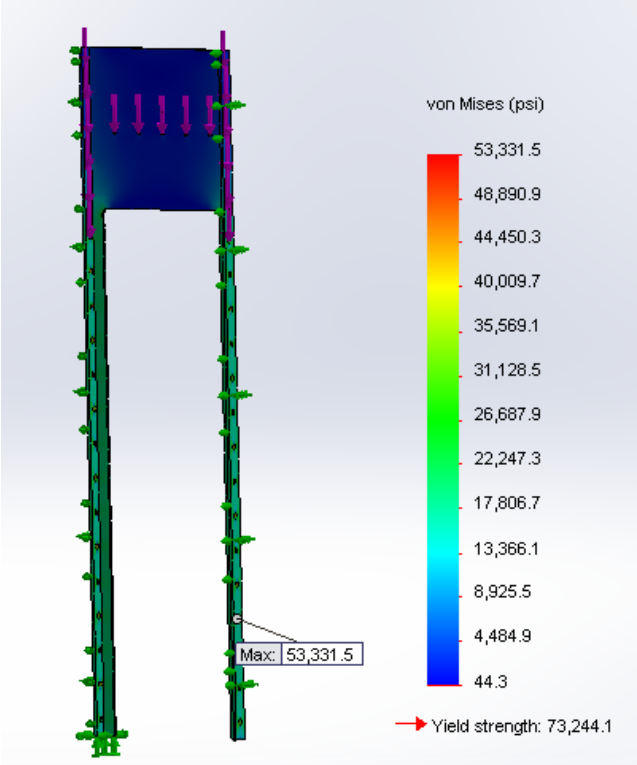

Figure 80: Pulley Lift System Front Plate von-Mises Stress Plot

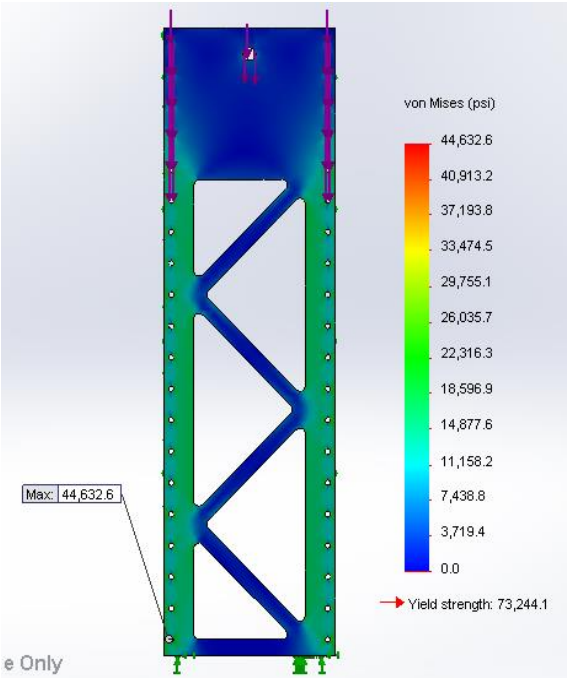

Figure 81: Pulley Lift System Side Plate von-Mises Stress Plot

Electric Motor Power Calculations

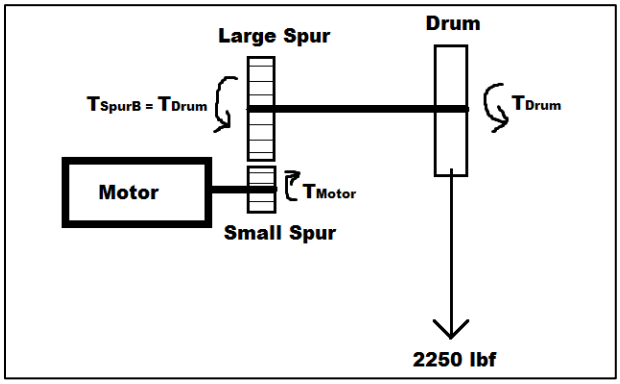

Figure 82: Motor and Gear Train Free Body Diagram

Governing Equation:

$$T = \frac{P}{w}$$

Where: T = Torque
P = Power
w = Angular Velocity

Assumption 0: Assume no bearing losses on rotating parts, and large components turn at a very slow rate so that there are no windage losses.

Assumption 1: Raise 2,250 lb. object a distance of 200 in. in 60 seconds.

Known Variables:

Mass:	Gravity:	Distance:	Time:
$m := 2250lbf$	$g = 32.174 \cdot \frac{ft}{s^2}$	$d := 200in$	$t := 60s$

Average Power:

$$P_{Avg} = \frac{\Delta W}{\Delta t}$$

Where: W = Work
t = Time

$$\Delta W = F \cdot d = m \cdot g \cdot d$$

$$P_{Avg} := \frac{m \cdot g \cdot d}{t}$$

$$P_{Avg} = 1.136 \cdot hp$$

Assumption 2: Lift the 2,250 lb. object with a cable and winch drum of diameter 2 in.

Known Variables:

Drum Radius:

$$r_{Drum} := 1in$$

Torque on Drum Shaft:

$T_{Drum} = F \cdot r$

Where: T_{Drum} = Torque along direction of cable, tangent to drum

$T_{Drum} := m \cdot g \cdot r_{Drum}$

$T_{Drum} = 2250 \cdot in \cdot lbf$

Angular Velocity of Drum

$\omega_{Drum} = \frac{v}{r_{Drum}}$

Where: v= Velocity of the Rope

$v := \frac{d}{t} = 3.333 \cdot \frac{in}{s}$

$\omega_{Drum} := \frac{v}{r_{Drum}}$

$\omega_{Drum} = 31.831 \cdot rpm$

Assumption 3: Large Spur gear is attached to the winch drum

Known Variables:

Spur Gear Pitch Diameter: $r_{pSpurB} := 1.5in$

Number of Teeth (Ration): $N_{SpurB} := 5$

Tangential Force on Spur Gear

$F_{SpurB} := \frac{T_{Drum}}{r_{pSpurB}}$

$F_{SpurB} = 1500 \cdot lbf$

Assumption 4: The force on the large spur gear is transferred to a smaller spur gear to provide a gear reduction.

Known Variables:

Input Gear Pitch Diameter

$r_{pSpurA} := .125in$

Number of Teeth

$N_{SpurA} := 1$

Force on Input Gear:

$F_{SpurA} := F_{SpurB}$

$F_{SpurA} = 1500 \cdot lbf$

Torque on Motor Shaft:

$T_{motor} := T_{Drum} \cdot \frac{N_{SpurA}}{N_{SpurB}}$

$T_{motor} = 450 \cdot in \cdot lbf$

Angular Velocity of Motor Shaft:

$\omega_{motor} := \omega_{Drum} \cdot \frac{N_{SpurB}}{N_{SpurA}}$

$\omega_{motor} = 159.155 \cdot rpm$

Ideal Shaft Power on Motor

$P_{MotorIdeal} := T_{motor} \cdot \omega_{motor}$

$P_{MotorIdeal} = 1.136 \cdot hp$

Actual Power: Including 87% efficient motor and 50% efficient mechanical system on the gear train due to frictional and windage losses.

$\eta_{motor} := 87\%$ $\eta_{gear} := 50\%$

$P_{MotorActual} := \frac{P_{MotorIdeal}}{\eta_{motor} \cdot \eta_{gear}}$

$P_{MotorActual} = 2.612 \cdot hp$

Required Energy

Lifting Mass: $m := \frac{4500}{2}lbm$

Gravity: $g = 32.174\frac{ft}{s^2}$

Height: $h := 20in$

$E := m \cdot g \cdot h$

$E = 3.75 \times 10^3 ft \cdot lbf$

Battery Capacity - MaxAmps.com Lithium Polymer

Battery Voltage Rating: $V_s := 25.9V$

Battery Current Rating: $I := 5450mA \cdot hr$

$E_s := V_s \cdot I$

$E_s = 3.748 \times 10^5 ft \cdot lbf$

Two Batteries in Series

$V_{total} := 2 \cdot V_s$

$V_{total} = 51.8\,V$

$E_{total} := V_{total} \cdot I$

$E_{total} = 7.496 \times 10^5 ft \cdot lbf$

From: MOSES, JULIE J CIV USAF AFMC AFOSR/RTB
To: WILLIAMS, MONIQUE F CIV USAF AFMC AFOSR/ITM
Subject: FW: FW: AFOSR Grant FA9550-11-1-0353 Final Performance Report is extremely late
Date: Wednesday, March 18, 2015 10:38:36 AM
Attachments: U.S. Air Force Final Report - University of Minnesota Duluth.pdf

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Monique,

Please find attached a copy of the final report for FA9550-11-1-0353.

Cheers,
Julie

-----Original Message-----
From: Bill Pedersen [mailto:wpederse@d.umn.edu]
Sent: Tuesday, March 17, 2015 5:39 PM
To: MOSES, JULIE J CIV USAF AFMC AFOSR/RTB
Subject: Re: FW: AFOSR Grant FA9550-11-1-0353 Final Performance Report is extremely late

Sorry my confusion. Thanks, Bill

On Tue, Mar 17, 2015 at 3:12 PM, MOSES, JULIE J CIV USAF AFMC AFOSR/RTB <julie.moses@us.af.mil> wrote:

> Bill,
>
> I would take a copy of the report you delivered to the competition.
>
> Thanks,
> Julie
>
> -----Original Message-----
> From: Bill Pedersen [mailto:wpederse@d.umn.edu]
> Sent: Tuesday, March 17, 2015 12:27 PM
> To: MOSES, JULIE J CIV USAF AFMC AFOSR/RTB
> Subject: Re: FW: AFOSR Grant FA9550-11-1-0353 Final Performance Report is extremely late
>
> Julie,
>
> I guess I am confused what report that you need. Is this a report from my sponsored projects people? What is the content of this report? Every year we submit project reports to the AFRL at the competition, but apparently that is not what you are looking for. Can you please let me know what I am supposed to submit so that I can do so.
>
> Thanks, Bill
>
> On Tue, Mar 17, 2015 at 11:13 AM, MOSES, JULIE J CIV USAF AFMC AFOSR/RTB <julie.moses@us.af.mil> wrote:
>
> Bill,
>
> Did you submit a final report for the University Design Challenge grant to AFOSR? If you did, please resend it to me. If not, we really need the final report. See below.

Thanks,
Julie

Julie J Moses, PhD
AFOSR/RTB
875 N Randolph St
Arlington, VA 22203
(703)696-9586

-----Original Message-----
From: AFRLDL-EBSAFOSRWorkflowManager@wpafb.af.mil [mailto:AFRLDL-EBSAFOSRWorkflowManager@wpafb.af.mil]
Sent: Wednesday, March 04, 2015 2:46 PM
To: KRAUS, ROBERT J Col USAF AFMC AFOSR/RT
Cc: MOSES, JULIE J CIV USAF AFMC AFOSR/RTB; ROACH, WILLIAM P DR-04 USAF AFMC AFOSR/RTB; CURCIC, TATJANA DR-04 USAF AFMC AFOSR/RTB
Subject: AFOSR Grant FA9550-11-1-0353 Final Performance Report is extremely late

********This is an automatically generated email, please do not reply to this email********

Dr. JULIE MOSES,

The Final Performance Report for Grant FA9550-11-1-0353, "University of Minnesota Duluth Engineering Design Challenge," which was due 29 DECEMBER 2014, is now at least two months late.

At this point, you must now generate and submit this report on your PI's behalf to close out the Grant. Please compile and submit a representative report by 12 March 2015.

Other consequences may be concurrently imposed by your respective Department Head until this report is completed and submitted into the Livelink workflow.

Regards,

The AFOSR Business Team

--

Bill Pedersen, PhD, PE

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--

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