Miniature Modular Rack Launcher Combo

EML 4551C – Senior Design – Fall 2011
Final Deliverable

Team # 3

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Executive Summary

The senior design project, *Miniature Modular Rack Launcher Combo*, was a project proposed by Mr. Russell Roberts at the Eglin Air Force Base in Destin, Florida. His goal was to lead a mechanical engineering team to design, and build a miniature bomb rack unit (BRU). This BRU will be housed to the wing of the Tigershark UAV and hold a given payload of 10lb. The BRU must be able to properly and safely house, maintain, and eject the given payload. The BRU must be able to communicate to the user, following proper ARM and FIRE procedures before the release of the payload occurs.

During the concept generation phase we divided the BRU in four major components with multiple designs. These components were analyzed through numerical and CAD analysis, in an effort to output an optimal design. The components of the BRU were divided into the following subsystems:

1. Hook Release
2. Safety Block
3. Ejector Mechanism
4. Sway Brace

Through our concept generation we were able to choose the best subsystems based on, weight, reliability, durability, and overall size. The linear hook release design was selected and will be powered by a pneumatic cylinder. The pneumatic system will be charged on the ground using an air compressor, which connects to a check valve that protrudes outside the BRU. A solenoid valve will be controlled by a microcontroller, which will control when the compressed air will be released into the cylinder. A servomotor will be the main component of the safety subsystem, this will prevent the payload from firing before the ARM signal is received from the user. When the ARM signal is given the servo will raise the safety block out of the path of the linearly traveling hook.

The ejector mechanism is coupled with the linear travel of the hook. As the hook travels horizontally that motion transferred into a vertical motion of the ejection mechanism. The ejection mechanism consists of a steal bar, which will contact the payload with a force provided by the pneumatic cylinder. The calculated exit velocity of the payload is 5.33 ft/s. The sway brace system is designed to stabilize the payload as the UAV performs in flight maneuvers with up to 2Gs of lateral loads. This system is designed with an angled aluminum bracket with a translating bolt assembly. A leveling foot is located at the bottom of the bolt assembly using a ball joint. Since the bolt is able to translate multiple radii of payloads can be used with the BRU.

A MiniDragon microcontroller will be controlling the electrical interface component of the BRU. This includes operation of the safety servomotor, and solenoid valve. The microcontroller prevents the payload from firing before the ARM signal is received from the user. In addition a limit switch will be used to inform the user of the position of the hooks being open or closed. This will output to a LED that is displayed on the control board.

Next semester a prototype will be created and tests will be conducted to confirm the numerical and CAD analyses.

Project Scope

Problem Statement

The emphasis of our project involves the design and fabrication of a launcher for the Tigershark UAV, capable of housing a given weapon system. The launcher design must meet the requirements specified by the AFRL, and must undergo a critical design review before the prototype is implemented. The finished launcher will then be integrated with the UAV in which a fit check will be performed with the selected weapon system.

Justification and Background

Unmanned aerial vehicles (UAVs) have become increasingly common on today's battlefield. Since to the UAV does not need room for a pilot, the aircraft can be much smaller making it difficult to be seen from the ground. In Iraq and Afghanistan, UAVs, such as the Predator and Global Hawk, have assisted ground forces by providing real time video of the battlefield using high resolution cameras. Increasingly common today are UAVs which can carry weapons, such as missiles and bombs. The Tigershark is a small, cheap, autonomous UAV developed by L-3 Unmanned Systems. The Tigershark has a wingspan of 17.5 feet, empty airframe weight of 150 pounds, and a gross takeoff weight of 300 pounds. Currently this UAV is used only as a surveillance drone. Our project will entail weaponizing the Tigershark UAV. This will allow the Tigershark to become more versatile and further assist ground forces on the battlefield.

Objective

As a project team our goal is to create a system that is lightweight and strong. This will be done by researching existing systems used on larger aircraft and essentially shrinking those into a simple lightweight mechanical system. A detailed budget analysis must be included and presented with recommendations for the system.

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Design and Analysis

Constraints

As mentioned earlier, this system is going to be used on the Tigershark UAV; thus requiring the system to be light, five pounds, to be able to be used. The BRU will hold a payload of ten pounds, and is required to withstand a 2G lateral load and 1G landing shock. Our system is to have an operational temperature range of -20°C to 60°C. It is also required to have safety pins that are marked remove before flight for ground control; as well as a mechanical safety lock that is used in flight. Another specification is that the payload must be ejected from the BRU with an ejection velocity of at least 4 ft/s, while not exceeding an ejection energy of 75 ft-lbs. Only 28V will be supplied from the aircraft, and someone also must be able to visually inspect the BRU to see if it is in "armed" mode. The final constraint for the system is that it must be able to mount to a pylon that is only an inch thick with a quarter inch holes that are 11 inches center to center. To achieve these goals, the requirements have been broken into three main components. The first component is the hook system that will be used to secure the payload. The next component is the mechanical safety lock that will hold the hook in place until the system is armed. The last component is the ejection system that will be used to achieve the 4 ft/s velocity, and not exceed the 75 ft-lbs. of energy. These designs are presented below as well as a section on the sway braces.

Latch Systems

The latch system is the first main component that will be analyzed. This system will use a hook to hold the payload with a mechanical release mechanism to swing the hook away during the firing procedure. Several different types of release mechanisms will be considered and are outlined in the following section.

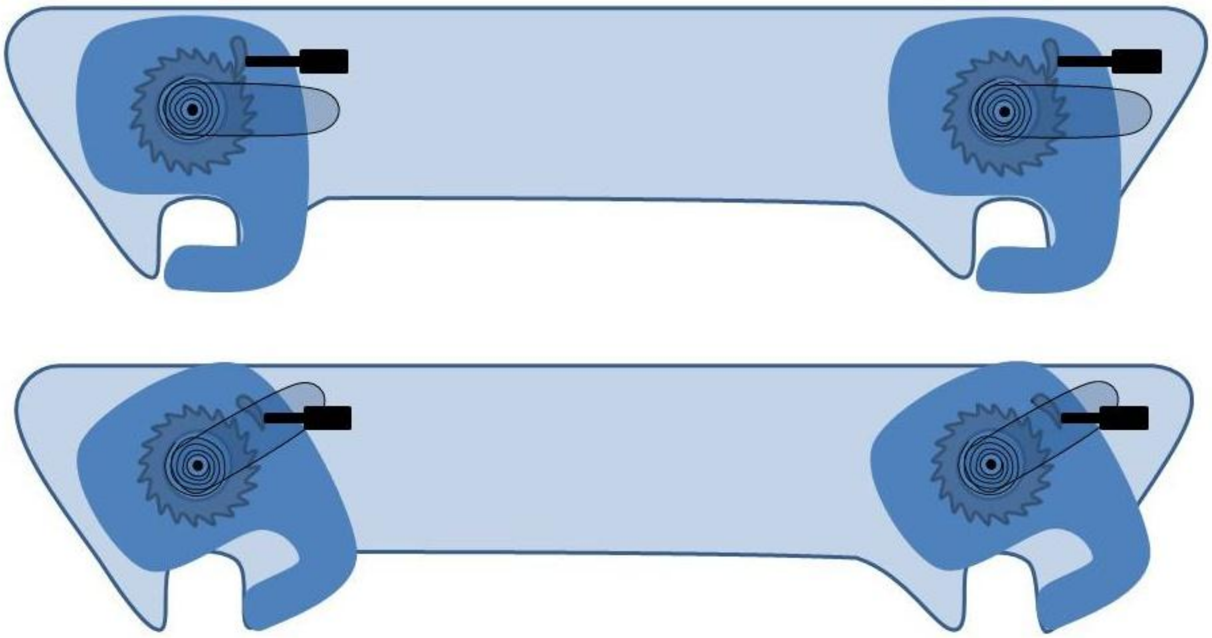

Figure 1- Ratcheting latch design in closed and open position

The first latch system that will be considered is shown above in figure 1, in its closed and open positions, respectively. This system utilizes a torsional spring that holds the latch in the open position, and a ratcheting system to hold the latch in the closed position. During the loading procedure, the hook is ratcheted closed by a lever arm that protrudes through the front of the housing unit. The pawl of the ratchet holds the hook in the closed position against the spring force. The torsional spring stores energy that will allow the hook to spring open quickly to release the payload. During the firing procedure, the pawl on the ratchet will be moved by a linear actuator. This will release the energy in the torsional spring which will rotate the hook and release the payload.

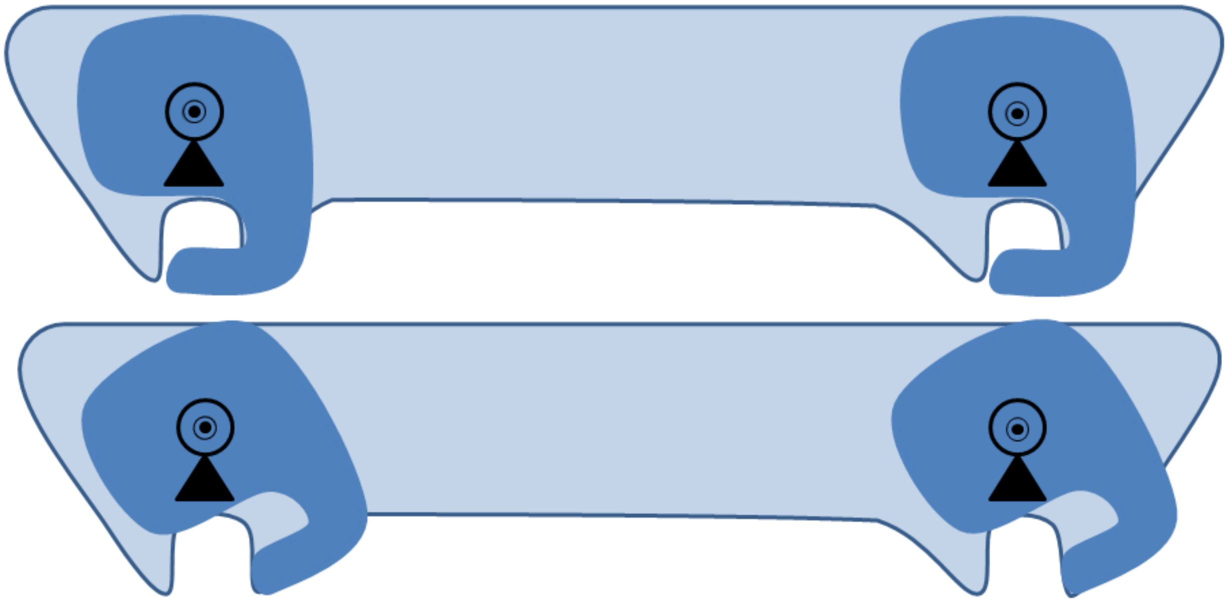

Figure 2- Motorized latch design in closed and open position

The second latch system that will be considered is shown above in figure 2. This system utilizes an electric motor attached at the pivot point of each hook. The hook used here is almost identical to the previous design, however, it does not have any spring connected to it, and there is no ratcheting action. This system uses the rotational work of the motor to hold the hook closed, and when fired, the motor provides the rotational force to spin the hook open and release the payload. This method, depending on the motor used, will not release the payload as quickly as a system designed using the stored energy of a spring to aid in turning the hook.

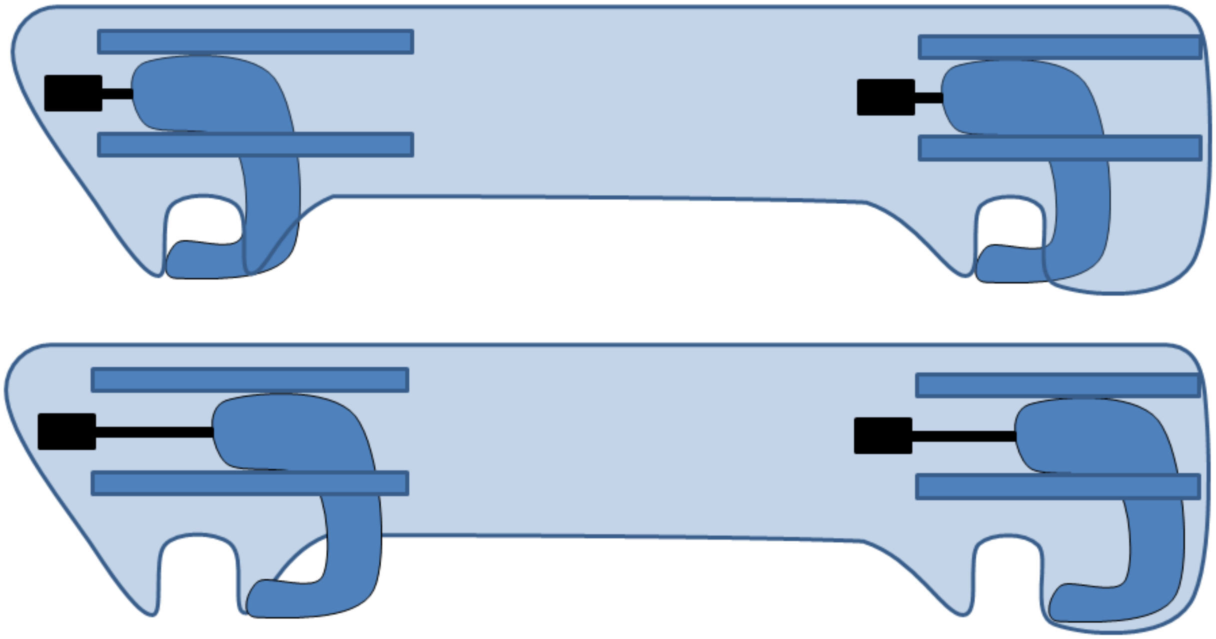

Figure 3- Sliding latch design in closed and open position

The next system that will be considered is shown above in figure 3. This system design uses a sliding hook that is guided by channels inside the main housing unit. The movement of the hook is purely translational; a linear actuator would be used to move the hook along the channel. When the payload is locked, the linear actuator retracts and the hook holds the payload securely. When the fire signal is given, the linear actuator is activated and it slides the hook down the channel, releasing the payload. Depending on the strength of the linear actuator used, this method might also be too slow to release the payload without any drag. There also will be increased friction that would have to be overcome due to the sliding.

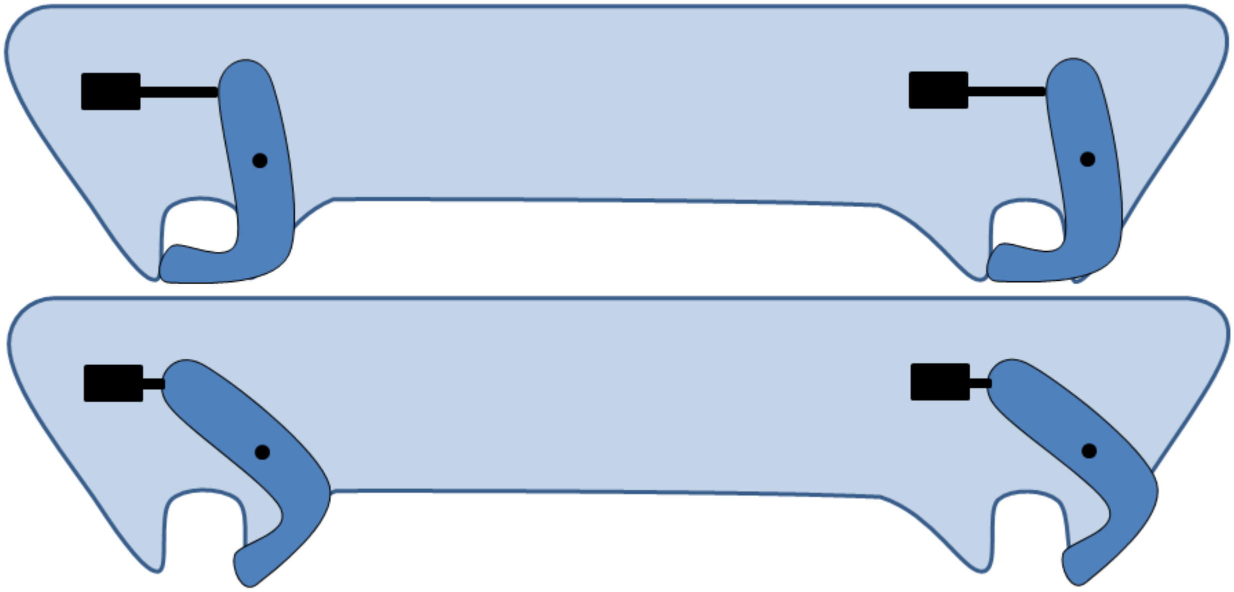

Figure 4- Linear actuator design in closed and open position

The next design, depicted above in figure 4, uses a rotating hook. This design is similar to the previous design that used a motor connected at the pivot point, but a linear actuator would be used that is connected by a pin to a lever arm on the hook. If this method was implemented, it would have to be carefully designed because there would be some induced sideways torque on the linear actuator shaft. This could be eliminated by using a two piece linkage to connect the actuator to the hook. This method will also have trouble opening the hooks fast enough to release the payload with minimal drag.

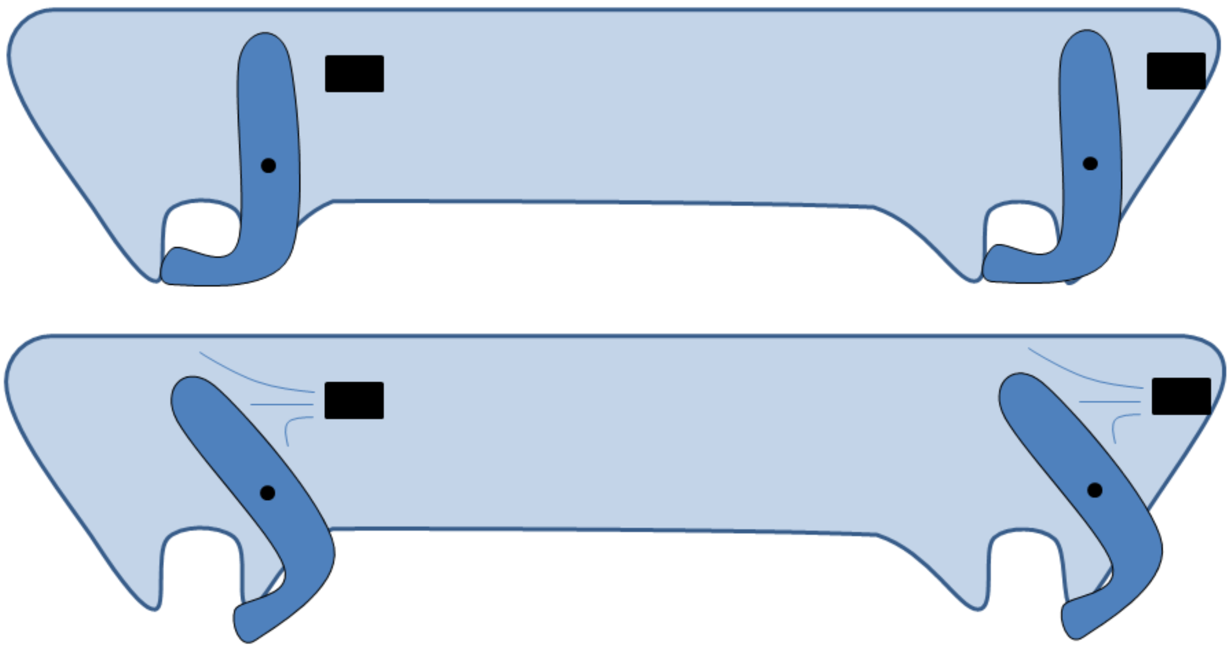

Figure 5- Compressed air latch design in closed and open position

The final latch design that will be considered is shown above in figure 5. This design uses compressed air to provide the energy to open the latch. The hook is virtually identical to the previous design; however the linear actuator is replaced by a compressed air tank. During the firing procedure, the compressed air will be released by a valve and will push the hook into the open position, releasing the payload. This method would provide the quick impulse of energy needed to open the hook quickly so it does not drag on the payload. This system would also consume much less electrical power because the only electrical power needed is for the valve system to open the tank. This system would also be lighter weight because there is no large motor or actuator.

Latch System Decision Matrix

		Designs									
		1		**2**		**3**		**4**		**5**	
Specifications	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight
Compactness	0.1	2	0.2	3	0.3	3	0.3	5	0.5	5	0.5
Weight	0.25	4	1	2	0.5	5	1.25	5	1.25	5	1.25
Strength	0.15	3	0.45	4	0.6	3	0.45	4	0.6	4	0.6
Durability	0.1	3	0.4	4	0.4	2	0.2	4	0.4	4	0.4
Operational Speed	0.4	5	2	3	1.2	2	0.8	3	1.2	5	2
Total		**4.05**		3		3		**3.95**		**4.75**	

Table 1- Latch System Decision Matrix

A decision matrix was used to analyze the different latch systems to determine the top 3 designs that will be subjected to further engineering analysis. The single-most important aspect of the latch is the operational speed. It is very important that the latch opens fast enough to eliminate the possibility of drag while releasing the payload. The weight is also an important deciding factor. From this matrix, the best designs to further analyze are design numbers 1, 4 and 5. Design 1 scored well because of its speed. It utilizes energy from a spring to snap the hook open quickly. Design 4 has a good score because of its light-weight and simplicity, but it lacks the important speed. Design 5 scored the highest because it has a compressed air energy storage system that is very lightweight, has very few moving parts, and has the ability to open the latch very quickly.

Mechanical Safety Systems

As mentioned earlier, our product is required to have a mechanical feature that locks the hooks until the "Arm" command is given. Once the system is armed the mechanical lock will move out of the way to allow the hook to move. In order to move this feature we have decided to use a servomotor. The first design of mechanical safety system is shown below.

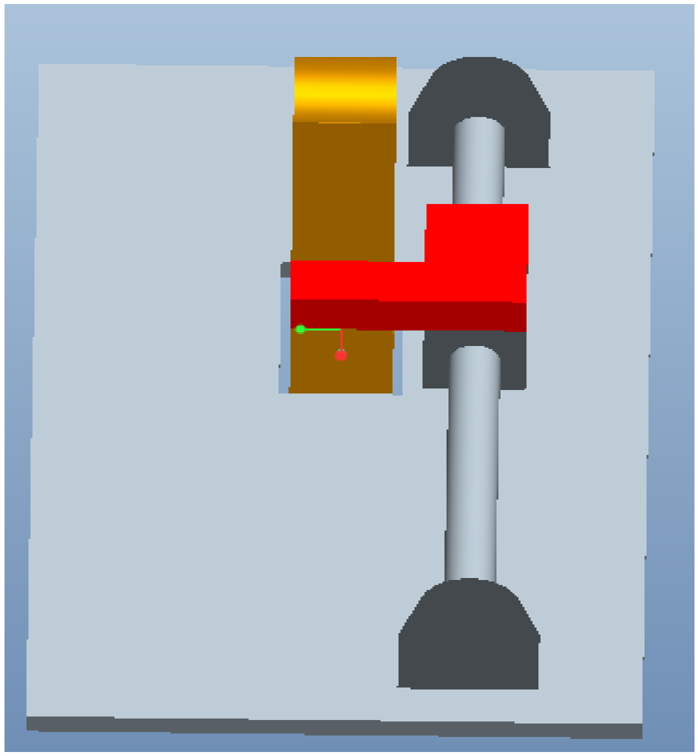

Figure 6- Mechanical Safety Design 1

In this design a single linear servomotor placed on the side of the hook. As it can be seen in figure 6, the safety stop block, colored red, has an L-shape design. This allows the stop block to be attached more rigidly to the servomotor. Once the system is given the "Arm" command, the servomotor moves forward out of the way. One drawback of this system is that when the block is engaged with the hook, a torque will be applied to the servomotor. This puts extra stresses on the servomotor that can lead to system failure. To compensate for this torque, another servomotor can be used on the other side of the hook. This second servomotor is implemented in the next design, figure 7 shown on the next page.

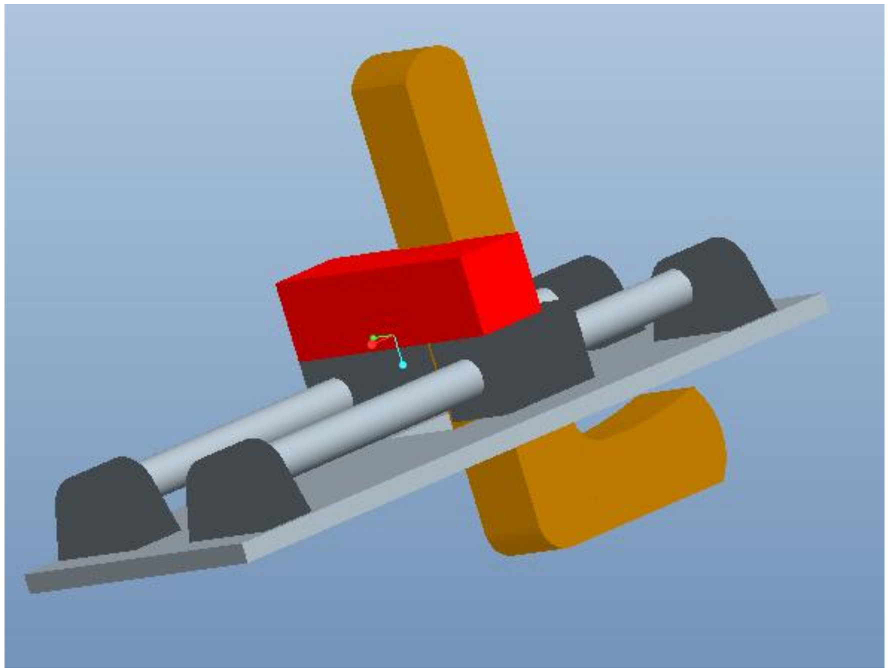

Figure 7- Mechanical Safety Design 2

Along with the added servomotor, the stop block is a rectangular piece that connects to both servomotors. This design removes the torque from the servomotors, as well as adding more force to the safety system. This allows for two smaller servomotors to be used to hold the stop block in place. One of the disadvantages to this system is the extra weight added with the extra servomotor and mounting system. Another disadvantage to this design is the added cost of the extra servomotor and mounting system. When the "Armed" command is given to the system, the servomotors move away from the hook like the first design allowing there to be low friction. A drawback with both of the first two designs is that the stop blocks are mounted on top of the servomotors. This adds a shear stress to the mounts between the stop block and the servomotors. Another drawback of these systems is that the stop blocks do not touch the bottom of the hooks. This could allow a hook that rotates to open prematurely. The next design, shown below, takes away these problems by changing the direction of motion and rotating the motor 90° to have the top of the servomotor facing the hook.

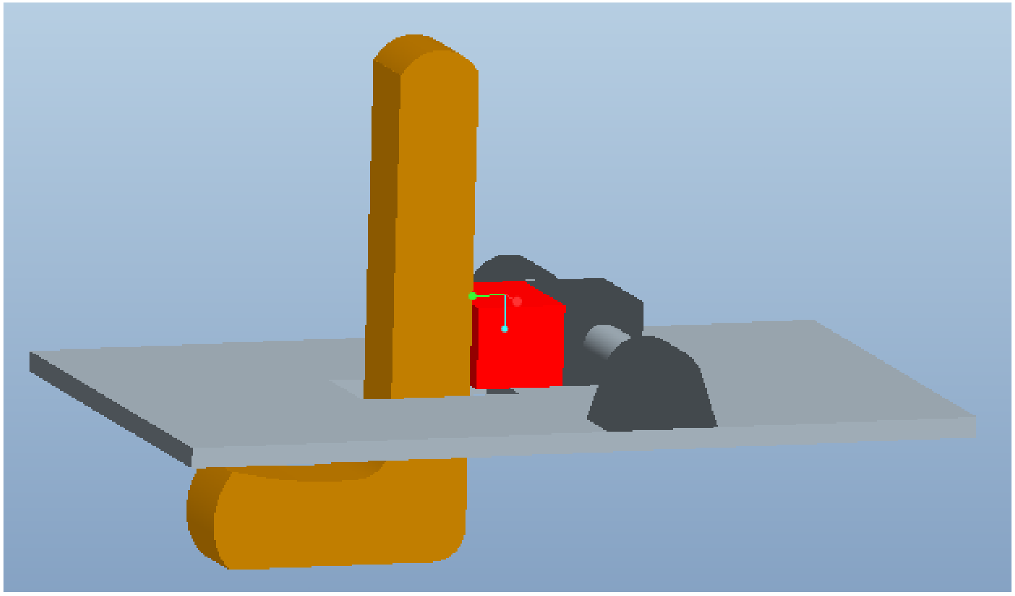

Figure 8- Mechanical Safety Design 3

As it is shown in the above figure, figure 8, the servomotor moves perpendicular to the hook. This system is beneficial because the stop block is smaller than the other design blocks, saving weight on the system. The stop block is moved to the right to allow the hook to freely move and release the payload. A disadvantage to this system is the added friction to the system from the way that the stop block disengages with the hook. The next design changes the direction of motion again by moving vertically. As shown on the next page, this design incorporates a larger stop block and two servomotors.

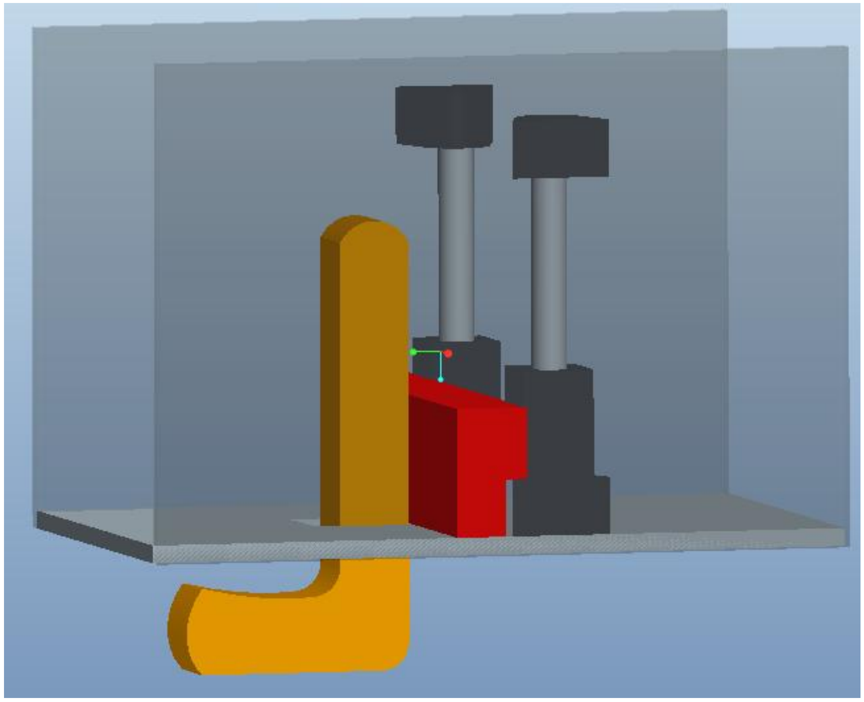

Figure 9- Mechanical Safety Design 4

As it can be seen, this design has the two servomotors placed on opposite sides of the hook. This takes away any torque from the hook. Like the previous design, design 3, this design has the servomotor mounted behind the stop block instead of under it. The block is twice as big as the servomotor to not allow the hook to move at all. This system adds more weight to the BRU by having the stop block larger than the others. Once the “Armed” command is given the stop block is moved high enough to allow for the hook to move freely. As with the previous design, the stop block has more friction on it when disengaged from the hook. The last to designs put more compressive strain on the servomotors if the arming sequence fails; making the designs less durable than the previous designs.

The previous four designs use a servomotor that move linearly in some orientation to the stop block. The next designs of the Mechanical Safety will employ a servomotor that rotates a stop block out of the way instead of using linear motion. On the next page is figure 10 showing the design of this type of system.

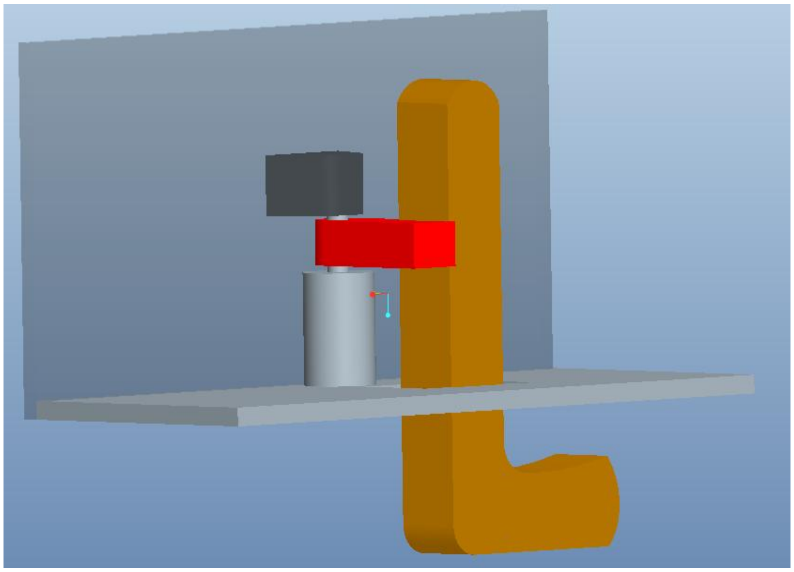
Figure 10-Mechanical Safety Design 5

As figure 10 shows, this system is much more compact than the designs using linear servomotors. This is because a rotational servomotor is mounted in between the two mounting blocks. An advantage to this design is that it does not move linearly. As with the first two designs, this system may allow for a rotating hook to prematurely open since it does not meet near the bottom of the hook. When the "Armed" command is given the stop block will rotated 90° to allow the hook to move freely. This system has a low amount of friction because of the way it is disengaged from the hook. The next design uses this same type of system, but rotates along the vertical instead of the horizontal axis.

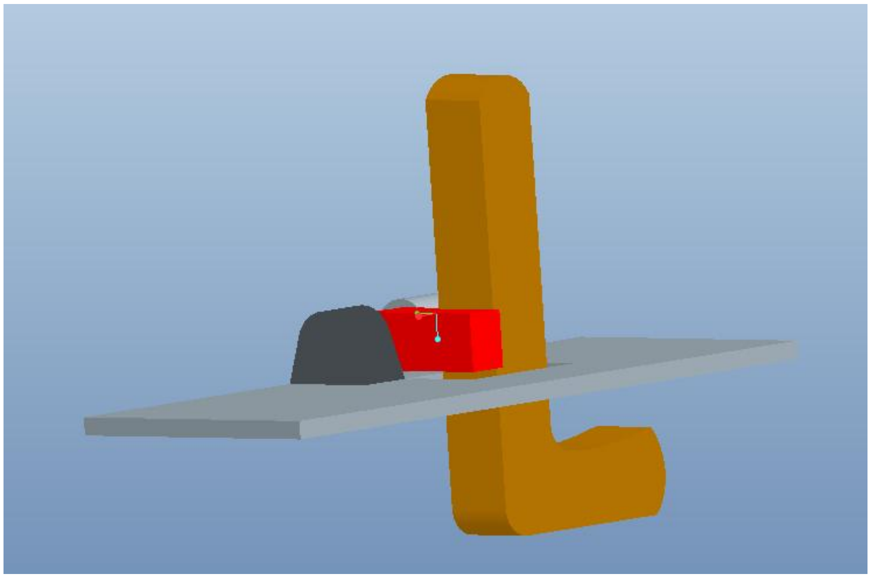

Figure 11- Mechanical Safety Design 6

As the above figure shows, this design is essentially Design 5 rotated 90°. This will take the torque on the motor out its axis of motion. This will cause the servomotor to have a shear stress when engaged with the hook. This will also allow the stop block to make contact at the bottom of the hook, not allowing any motion. A benefit of this system, like the last, is its compactness. The difference is that this design takes up less space when moved into “Armed” mode. Since design #5 moves along the length of the BRU, considerations have to be made to allow for this motion. Once moved into “Armed” mode the hook moves vertically allowing space to be saved that can be used for mounting other systems onto the BRU. As it was discussed earlier, removing the stop block in the vertical direction adds more friction on the system when disengaging from the hook.

Mechanical Safety System Design Decision Matrix

In order to make an accurate decision on which Mechanical Safety will work best for our system. From this decision matrix the top three systems will be selected for further review. One of the reasons for this is that the best Mechanical Safety can only be chosen after the hook system is chosen. This will give the strongest system for that style of hook system, and ultimately making this system the safest it can be. The features that will be analyzed with the decision matrix are the compactness, weight, strength, durability, and operational speed of each design.

		Mechanical Safety System Designs											
		1		2		3		4		5		6	
Specifications	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight
Compactness	0.2	3	0.6	2	0.4	3	0.6	2	0.4	4	0.8	5	1
Weight	0.2	3	0.6	2	0.4	4	0.8	2	0.4	5	1	5	1
Strength	0.3	3	0.9	5	1.5	4	1.2	5	1.5	4	1.2	4	1.2
Durability	0.2	3	0,6	5	1	2	0.4	3	0.6	4	0.8	3	0.6
Operational Speed	0.1	4	0.4	5	0.5	2	0.4	3	0.3	5	0.5	4	0.4
Total		3.1		**3.8**		3.4		3.2		**4.3**		**4.2**	

Table 2- Mechanical Safety Decision Matrix

From the decision matrix the top three designs for the Mechanical Safety System are Design numbers 2, 5, and 6. The next step for these designs is to undergo an analysis to find which one will work best with the type of hook system used or this project.

Sway Bracing

Sway bracing is a critical design feature on the BRU, and is used to prevent the store from moving laterally or vertically during flight. The store could experience lateral forces of up to 2G during a turn. The sway brace must be able to resist this force and keep the store steady. Below are two designs for the sway brace.

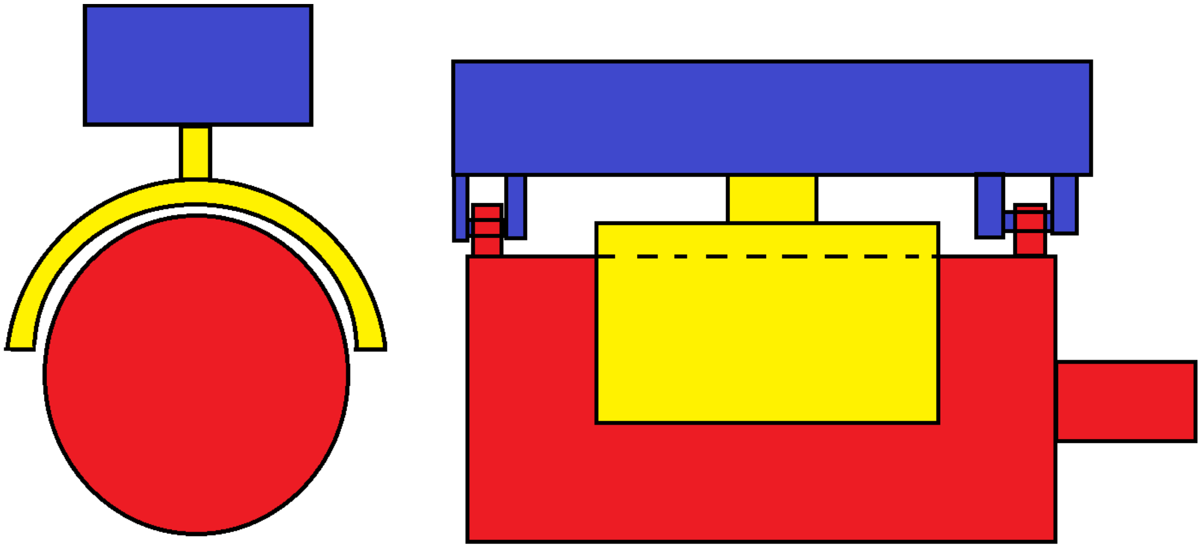

Figure 12- Sway Brace Design 1

Design 1, shown above, illustrates one design for the sway bracing needed to keep the payload steady in flight. This design uses a stationary sway brace (yellow) fixed to the BRU (blue). The store (red) will fit inside the radius of the sway brace. When in a turn, the payload will push up against the sides of the brace preventing lateral motion. Pitching, vertical motion, of the payload will also be limited by this design.

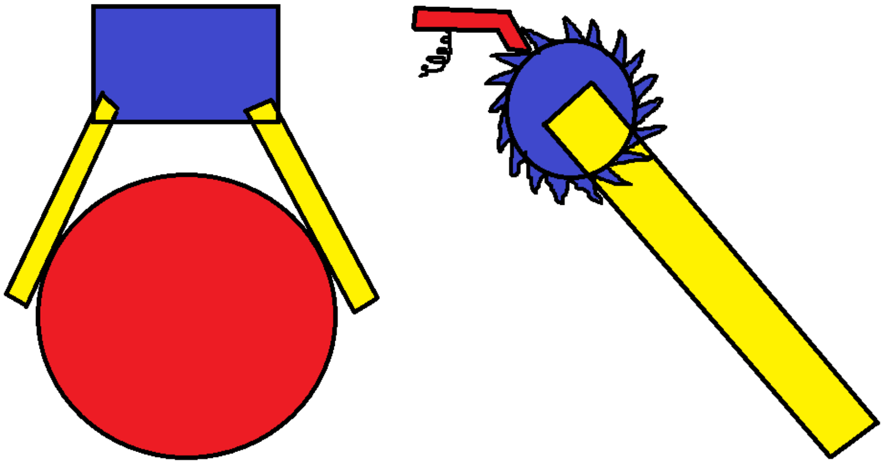

Figure 13- Sway Brace Design 2

The second design for sway bracing is illustrated above. There are four arms located near the corners of the BRU (blue box on left). Once the payload (red on left) is locked into the hooks, the arms will be manually lowered and will self lock with a ratchet-paw system. Each arm (yellow) is attached to a ratchet gear (blue gear on right). The arms will lightly pinch the payload preventing its movement. A manual release will be used to disengage the paw (red on right) so the arm can be raised and reset.

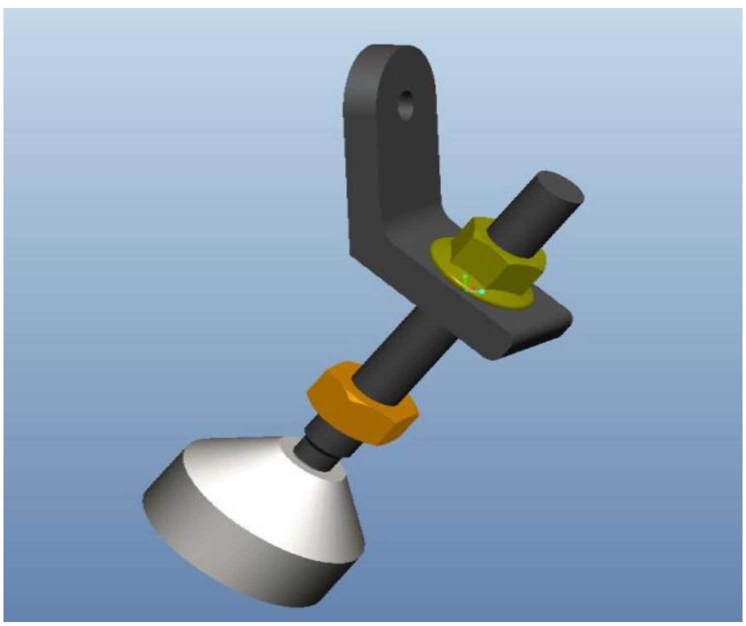

Figure 14- Sway Brace Design 3

The final design for the sway brace is shown above. This design entails using four arms as the previous design did, but it has a screw that can be moved to pinch the payload. This is done by two nuts that are located on each side of the brace arm. Connected to the bottom of the screw is a swivel foot that allows great flexibility in securing the payload. The main advantages of this system are that it is lightweight and extraordinarily simple to use.

Sway Brace Decision Matrix

In order to select which sway brace will be better for the BRU the following decision matrix has been assembled. Six categories with assigned weights are used to aid in the selection process. The most important factor for the sway brace is weight. As the BRU will have lots of important parts, the weight needs to be kept low. The next important deciding factor is the brace's load carrying capability. The sway brace will need to withstand lateral forces of up to 2 Gs as well as keeping the store from moving in flight. Payload Size Flexibility looks at how easily the sway brace can adapt to changes in the size/shape of the Payload. Durability looks at how well the brace can withstand repeated loadings/releases. Ease of Use refers to how easily ground crews can set the sway brace up so it is ready for flight. Finally, Simplicity focuses on how simply the sway brace can be implemented to the BRU.

		Sway Brace Concepts					
		1		2		3	
Specifications	Weight	Score	Weight	Score	Weight	Score	Weight
Weight	0.3	2	0.6	4	1.2	5	1.5
Load carrying	0.3	5	1.5	3	0.9	5	1.5
Payload Size Flexibility	0.15	1	0.15	5	0.75	5	.75
Durability	0.1	4	0.4	2	0.2	5	.5
Ease of Use	0.1	5	0.5	4	0.4	5	.5
Simplicity	0.05	5	0.25	2	0.1	5	0.25
Total		3.4		3.55			**5**

Table 3-Sway Brace Decision Matrix

As it can be seen in the table the third design is by far the best design. This will be the design that will be implemented into our design of the BRU.

Payload Ejection

When the “Release” signal is given from the aircraft, the hooks release the payload. This causes the store to fall entirely due to gravity. This free fall is too slow and an ejection method is required. The payload will need to leave the BRU at a minimum velocity of 4 ft/sec and cannot be forced down with more than 75 ft-lbs of energy. Below are four designs for the payload ejection.

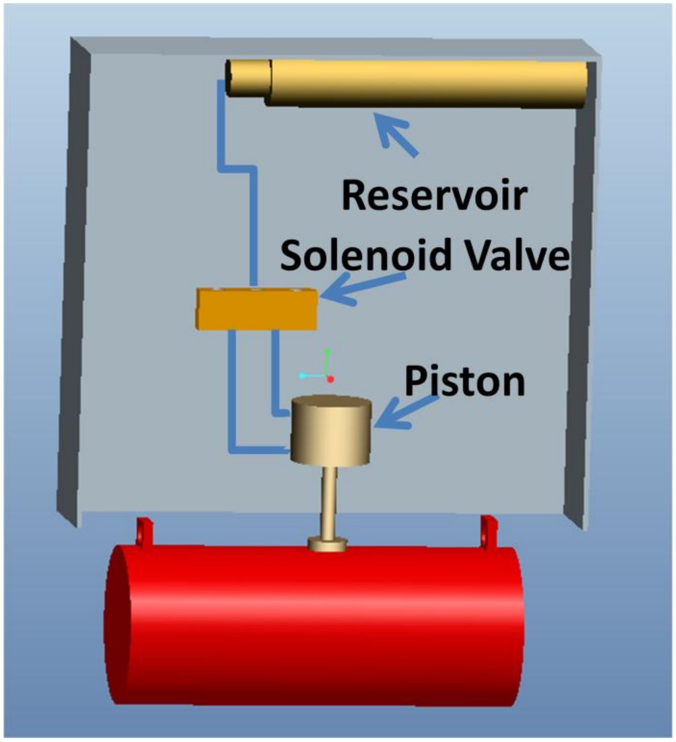

Figure 15- Payload Ejection System Design 1

Design 1 above uses a pneumatic piston (green) to eject the store (red). The piston will push the store down when the “Release” signal is given. A pneumatic canister will be filled preflight on the ground and installed into the BRU (not shown). The fixed sway brace can be used as a “foot” attached to the piston. When the piston fires it forces the sway brace down at a high velocity, ejecting the payload.

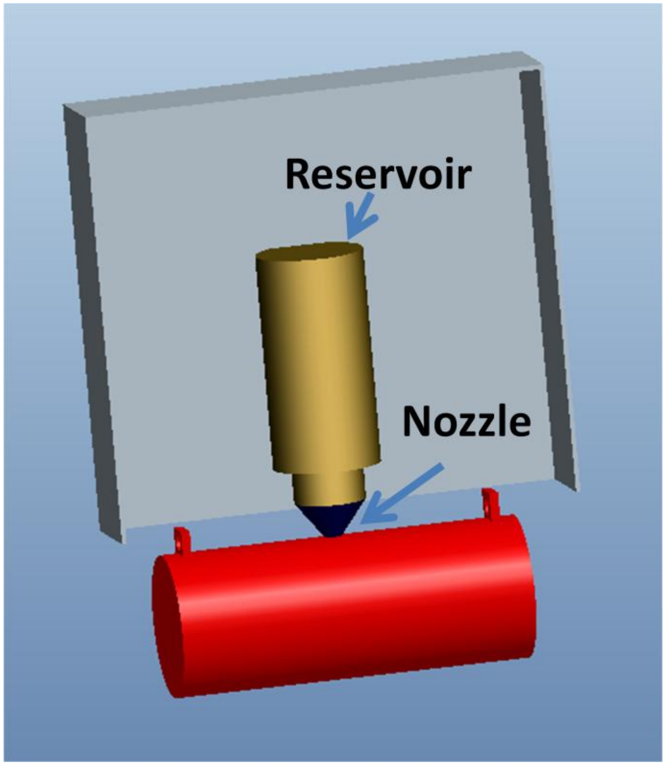

Figure 16- Payload Ejection System 2

Design 2 uses compressed air to eject the store. A compressed air canister (green) will be filled on the ground and inserted into the BRU during ground operations. When the “Release” signal is given the air will be released and forced through a nozzle directed toward the payload pushing it down away from the aircraft.

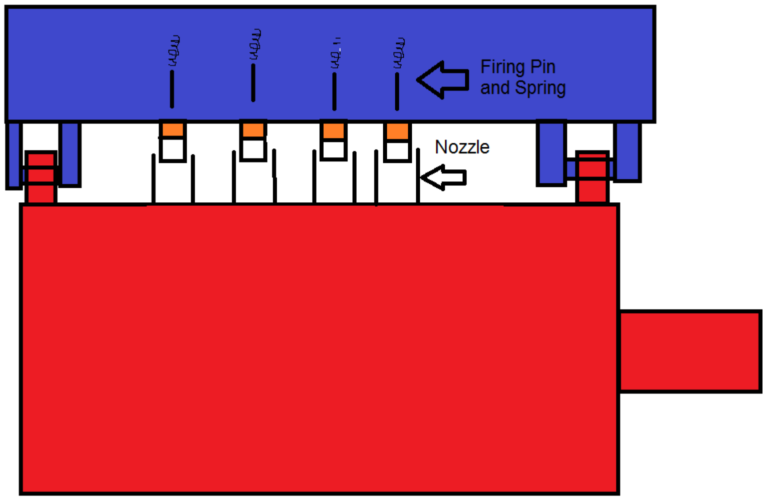

Figure 17- Payload Ejection System Design 3

Design 3 above uses pyrotechnics to eject the payload away from the aircraft. The pyrotechnics involved will use gun powder (orange) from a bullet to create a controlled explosion to force the payload down and away from the aircraft. As shown in the illustration, multiple explosives can be arraigned to create enough ejection force. To set off the explosive, a firing pin is attached to a spring, which will be compressed in flight for safety reasons, when the "Release" signal is given the spring will be released forcing the firing pin into the bullet. The pin hits the primer igniting the propellant creating a controlled explosion which will force the payload down away from the aircraft.

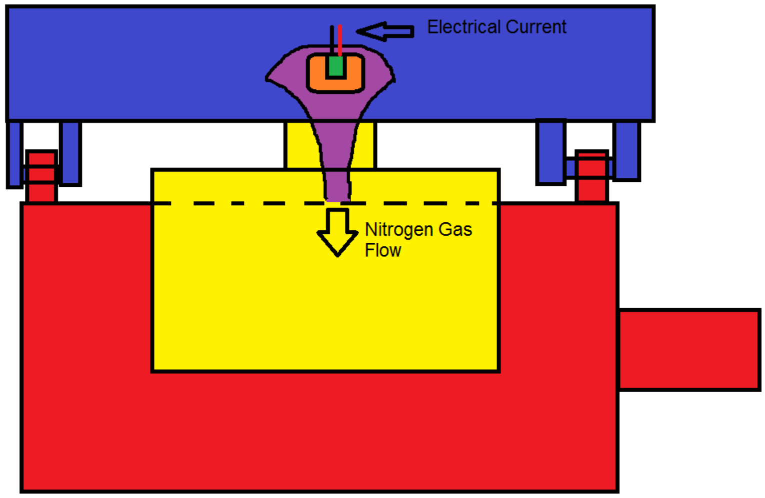

Figure 18- Payload Ejection System Design 4

The final ejection design uses the same technology that is used in air bags in cars. When the car experiences a crash, an electrical signal is sent to the bag and sets off an extremely violent chemical reaction that combines sodium azide (NaN_3) with potassium nitrate (KNO_3). The product of this reaction is nitrogen gas. An air bag can deploy in one-twenty-fifth of a second, faster than a person can blink an eye. As an ejector for the BRU, a nozzle (purple) will funnel the nitrogen gas directly onto the payload. The strong pressure created by the reaction will push the payload away from the aircraft.

Payload Ejector System Decision Matrix

The following decision matrix does a simple comparison of how the designs fare with regard to the design criteria.

		Payload Ejector System Designs							
		1		2		3		4	
Specifications	Weight	Score	Weight	Score	Weight	Score	Weight	Score	Weight
Weight	0.25	2	0.5	3.5	0.875	5	1.25	2.5	0.625
Size	0.15	3	0.45	4	0.6	5	0.75	2	0.3
Cost	0.1	2	0.2	3	0.3	4	0.4	2	0.2
Safety	0.2	4.5	0.9	4	0.8	1	0.2	2	0.4
Ease of Use	0.2	3	0.6	3	0.6	4	0.8	3	0.6
Simplicity	0.1	3	0.3	3	0.3	2	0.2	4	0.4
Total		2.95		**3.475**		**3.6**		2.525	

Table 4- Payload Ejection System Decision Matrix

As you can see, designs 2 and 3 have the greatest score. However further design review and calculations are required to determine the viability of the designs.

Final Design

Pneumatics System

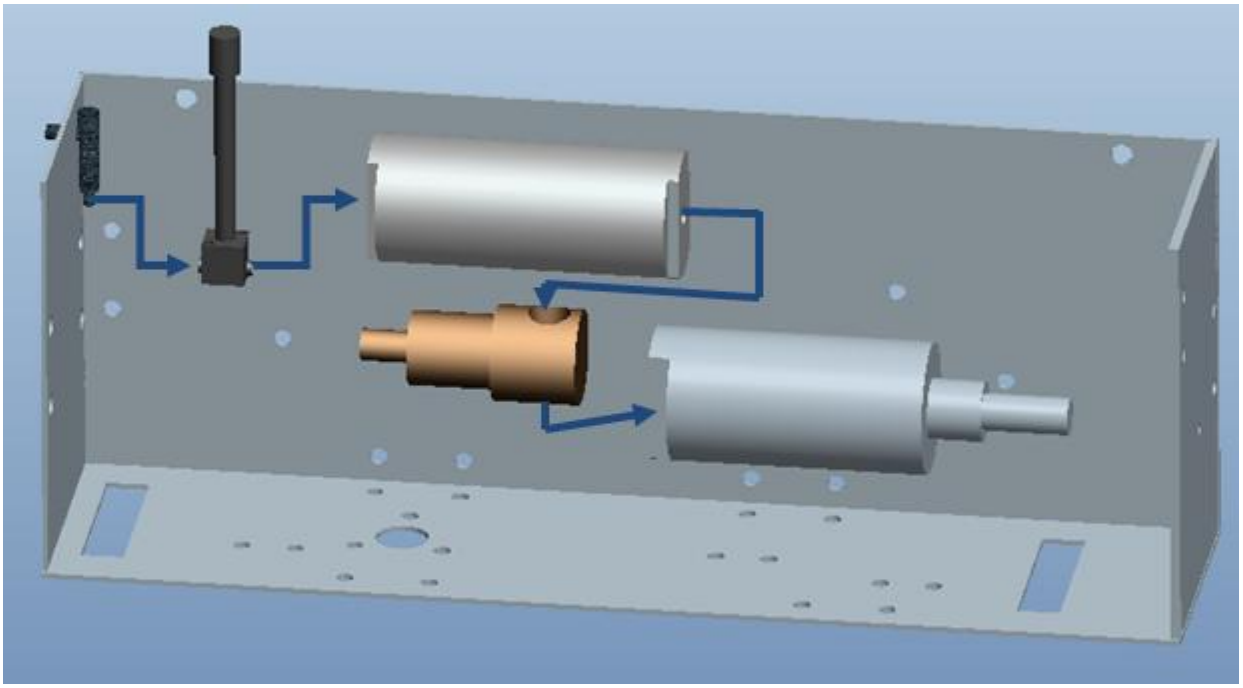

Figure 19. Pneumatic System Components. Left to Right: Check Valve, Back-Pressure Regulator, Air Tank, Solenoid Valve, Air Cylinder

The pneumatics system is vital to the operation of the BRU. This system is in charge of disengaging the hooks to allow the payload to be fired. The primary component is the pneumatic cylinder, which when supplied with sufficient air pressure will extend a piston. Based on calculations for the ejection system, it was determined that a force of about 150 lbf would be needed to be placed on the hook-bar to create sufficient ejection velocity. Based on this, a pneumatic cylinder was chosen with a 1 ½ in. bore with a 1 in. stroke; this gives an approximate volume of 1.7 in^3. With a safety factor of 2 to account for any possible losses through the tubing and connections, a 4 in^3 air tank was selected to house the air needed to fire the cylinder.

To keep the pneumatic system at a constant and safe operating pressure, a back-pressure regulator is required. The regulator chosen is adjustable for pressures between 0.5 and 90psi. For the operation of the BRU, the regulator will be set to its max pressure of 90psi. When 90psi is reached in the system the excess air will be released through the cap of the regulator. The cap will be positioned above the top of the BRU so ground crews can physically tell the BRU is full of air and to stop filling. The regulator will be located ahead of the air tank in the pneumatic system chain. Ahead of the regulator is a check valve. This will allow air to be added to the system when the aircraft is on the ground and will refill the air tank. The check valve has a push-to-connect feature which will allow a hose connected to an air compressor to quickly connect to the BRU's pneumatic system. When the compressor hose is removed, air will not be allowed to escape through the check valve. The valve is designed to prevent backflow of air so air can only be added, not lost.

Following the air tank, the next component of the pneumatics system is a solenoid valve. This valve acts as a switch in a circuit. When signaled, the valve will open sending the air stored in the tank to the air cylinder. The solenoid uses 12 VDC which will be supplied by the aircraft. When the “Fire” command is given, current will be sent to the solenoid, switching the position of the valve allowing air to be sent to the cylinder.

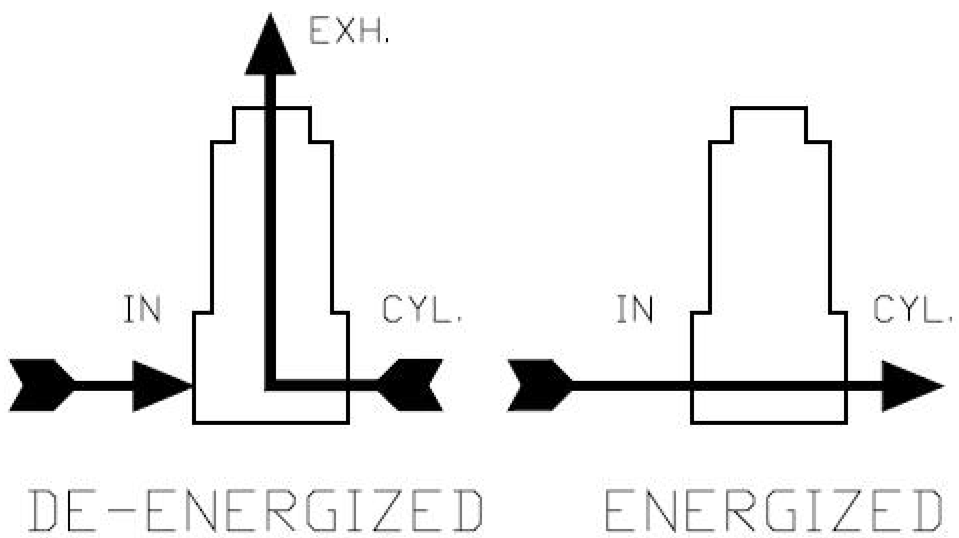

Figure 10. Solenoid Valve Operation Diagram. Courtesy of cylval.thomasnet-navigator.com

After ejection, the solenoid valve will return to its de-energized state and the air inside the air cylinder will flow through an exhaust port inside the solenoid valve. The hooks will once again be closed inside the BRU.

Hook Analysis

For the Hooks an FEM analysis had to be done in Pro/Engineer Mechanica to show that it could withstand the force of the payload and the pneumatic piston. The first sets of figures, shown below, show the force of the payload on the hooks when in the disarmed or armed mode. These stresses were analyzed by assuming a 30 lbf load on both hooks from the payload.

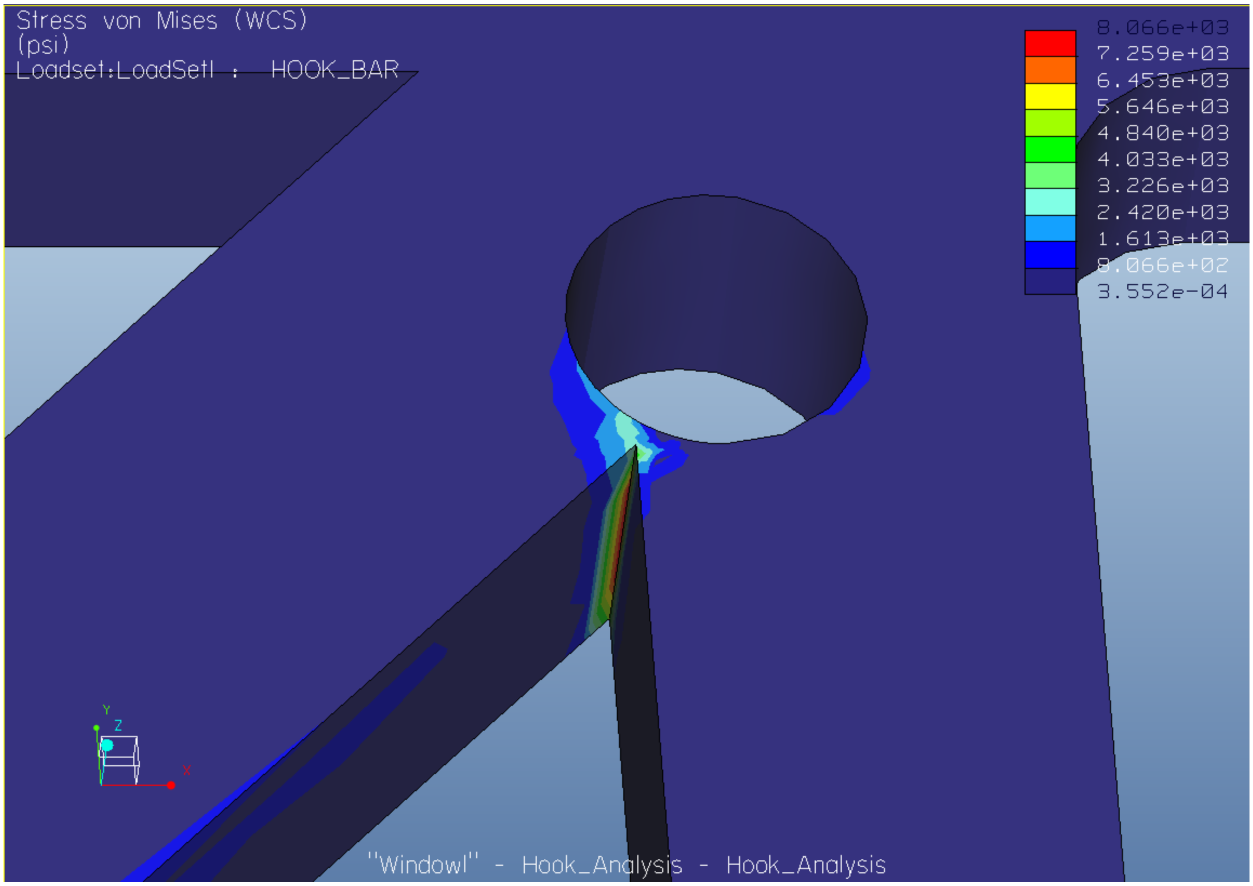

Figure 21- von Mises Stress. Concentating on Upper Portion of Ejector Ramp

In order to optimize the shape of features on the hook, stresses were analized and stress concentrations were located. Shown above in Figure 21 is a major stress concentration peeking at about 8066psi. This is greater than the 8000psi yeild stress of aluminum. A fillet will need to be added to remove the stress concentration.

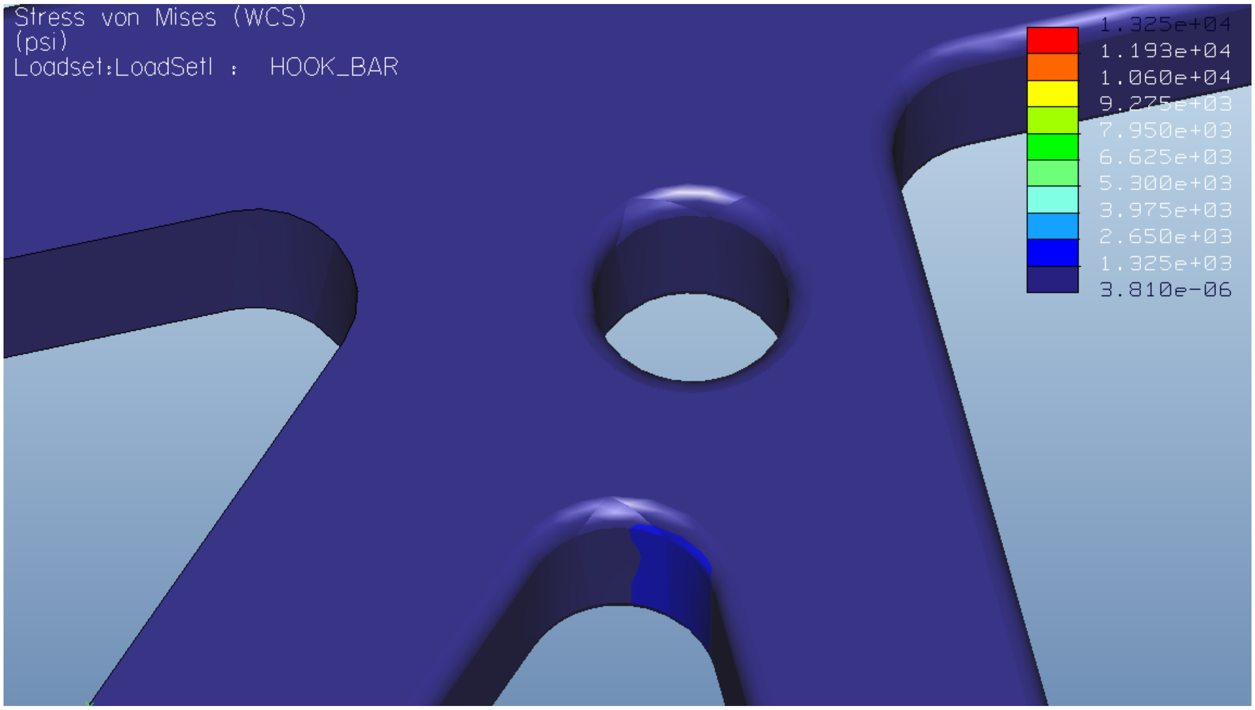

Figure 22- von Mises Stress. Concentating on Upper Portion of Ejector Ramp. Fillets added to reduce stress.

Figure 22 above shows the stresses seen after the fillets were added. It can be obviousily seen that the stress concentration has been greatly reduced. Max stress show at this location is around 2000psi, this is a drastic reduction compared to the unrounded stress of 8066psi.

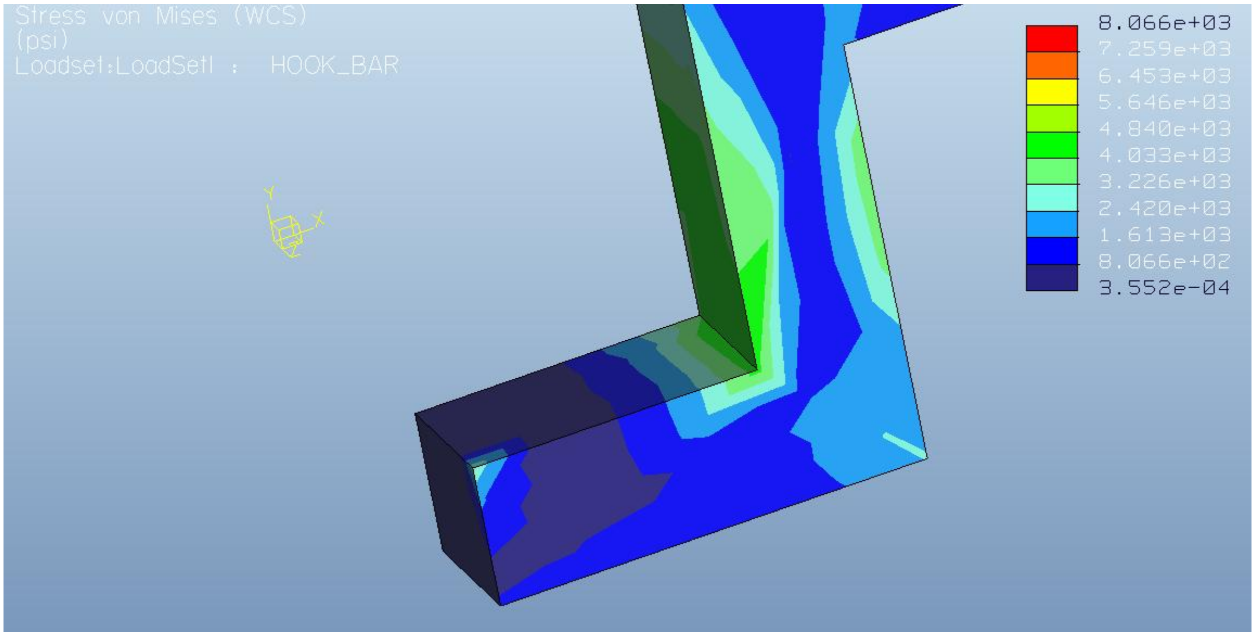

Figure 23- von Mises Stress of the Hook.

Similarly, the stresses were analized in the hook portion of the part, Figure 23. Stresses of about 4000psi were seen. While this provides a sufficient safety factor of about 2, a fillet at this location would be benifical.

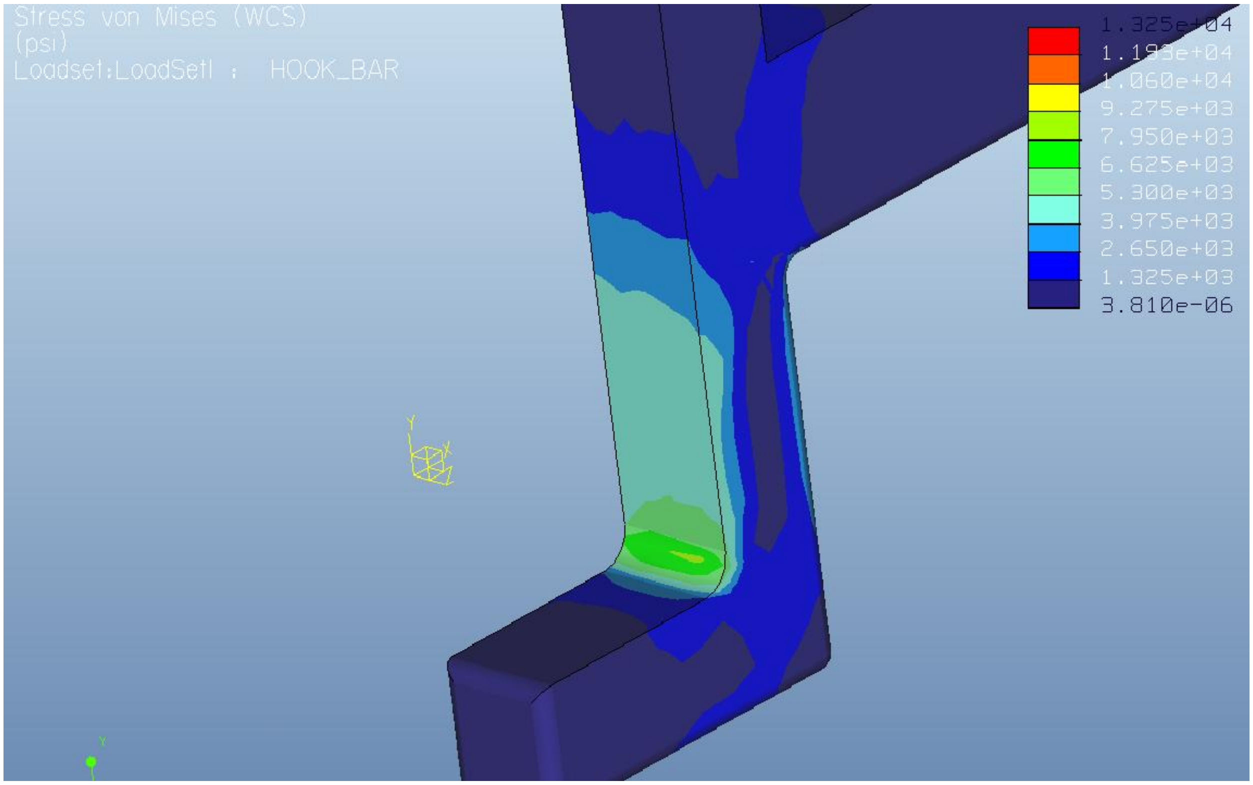

Figure 24- von Mises Stress of the Hook. Rounds added to reduce stress concentration.

Once again fillets were added to the hook, which reduces the stress concentrations. While it appears that stress seen at the curve is slightly greater, the overall stresses throughout the shape of the hook have been greatly reduced. It is concluded that rounding the hook is a benifical alteration.

Bearing Analysis

Bearings will be used to assist in providing the translation movement of the hook with minimal friction. External roller bearings were chosen and will be mounted underneath and above the hook assembly. Delrin channels will be fabricated and will provide the top and bottom channels in which the hook will slide. Delrin will be used because of its light weight, durability and its low friction coefficient. A bearing analysis was performed to determine not only the total bearing loads but also the ideal bearing locations to evenly split the forces between the bearings. Figure 25 below shows a bearing system mounted above and below the hook. There are similar bearing blocks on the other end of the hook.

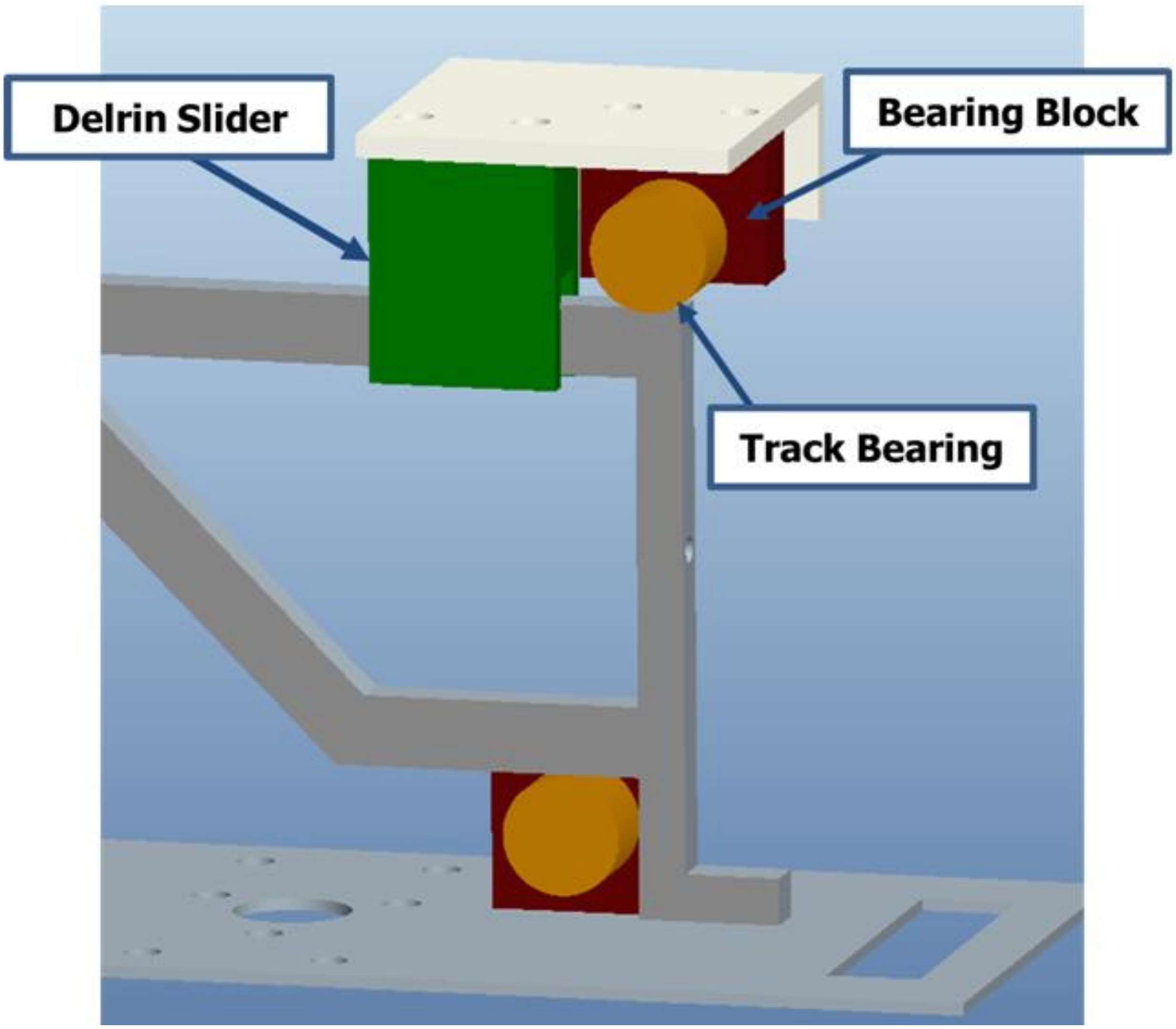

Figure 25- Bearing Assembly

The first step of the bearing analysis was to determine the weight that will be supported by each hook. This was done by first analyzing the payload. The center of gravity of the conceptual payload was specified; a torque analysis was performed using this location and the overall weight of the payload to determine the weight at each payload hook. These are the specific weights that must be supported by the two hooks on the BRU.

Once the forces on each hook are known, the next step is to perform a torque balancing analysis on the hook itself. The first case analyzed was the hook in the closed position. In this position the entire weight of the payload is supported by the bottom two bearings; the forces in the upper bearings are zero. In the open position the hook exerts a downward force on the piston which in turn exerts an upward force on the hook. In this position the forces in the upper bearings were calculated.

The equations for the reactions forces in the bearings were entered into MathCAD; iterations were performed to locate the optimized bearing locations in which the bearing reaction forces are as close to equal as possible. The hook has an over length of 12 inches; the ideal lower bearing locations, limited by the hook geometry, are 1.75-inches and 8 inches from the front end of the hook. Due to space limitations and other components within the BRU housing, the ideal locations could not be used, but they are as close as the constraints will allow. The ideal locations for the upper bearings are 2-inches and 9-inches. The track bearing shown below in figure 26 will be used.

Figure 26- Track Bearing

Mounting Tab Analysis

The mounting tabs are the critical pieces that attach the BRU to the wing of the aircraft. The entire weight of the BRU structure as well as the weight of the payload must be supported by the mounting tabs. These tabs must hold the BRU in place during aircraft lateral aircraft maneuvers of up to 2-G as well as a 1-G landing shock. These are the values that were discussed earlier in the specifications and requirements section. These acceleration forces form a combined loading situation on the mounting tabs that must be determined to find the max stress within the tabs. The minimum thickness of the tabs was found to keep the applied stress within the allowable stress of the material. The mounting tab is shown below in figure 27.

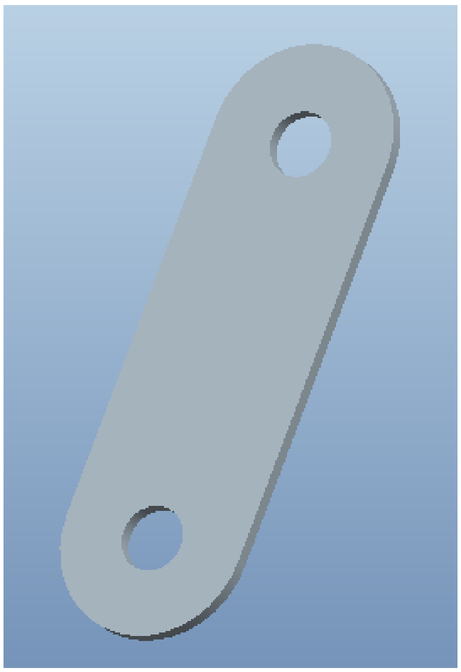

Figure 27- Mounting Tab

The BRU to be designed was specified to have a maximum weight of five pounds. This maximum weight was used in the analysis as part of the force that is to be supported by the mounting tab. The payload was specified to have a weight of ten pounds. This means that fifteen pounds is the static force that is supported by the mounting tabs. In addition to gravity there is an additional 3-G total force that needs to be accounted for in the analysis. This is the case when the maximum lateral force and the landing force occur in the same direction, simultaneously. In this case there is a total downward force of 60 lbf. This value is used in multiple analyses.

The bolts that connect the mounting tab to the pylon will be in double shear. The maximum force that could be on them is 60 lbf, in the maximum downward 4-G case. The mounting hole is specified to have a quarter inch diameter, so quarter inch bolts will be used. The average shear stress in the bolt was calculated to be 611 psi. Grade 1 fasteners have an allowable shear stress of 36 ksi; the lowest grade fasteners can be used to connect the mounting tabs to the pylon and to connect the BRU to the mounting tab.

The possibility of mounting tab bolt-hole tear out was considered. 60 lbf downward was the force used in the analysis. The minimum plate thickness was determined to be 0.016 inch to resist tear out. This thickness will keep the shear stress within allowable limits including a safety factor of 1.5. It will be hard to find a standard plate thickness so the smallest available aluminum plate will be used, or it could be machined down further to save weight.

Now a combined loading case will be considered. This is the case when the 2-G force is acting laterally while another 2-G force is acting downward. This will create various stressed that can be summed by superposition to find the total maximum stress. The point of maximum stress will be at the bottom of the mounting tab bolt hole. This is the point at which the stresses were calculated. This combined loading creates the greatest stresses within the material so this loading analysis will be used to design the thickness of the tab. The following stresses were calculated based off the optimized thickness of 3/32-inches. This is the smallest standard plate thickness that will keep the safety factor above 1.5.

The 2-G force from the combined weight and landing shock creates normal and shear stresses within the mounting tab. This normal stress was calculated to be 427 psi. The shear stress was calculated to be 1.28 ksi. The lateral force creates a bending stress about the bolt hole. This bending stress was calculated to be 35.4 ksi. There is a separate bending stress that was also considered from the force of wind acting on the payload. This bending stress was calculated to be 0.6 psi. This stress was so small it probably could have been neglected. Using superposition to sum these stresses, the total stress at the mounting tab hole is 37.1 ksi. This is the maximum stress if the forces were placed on one mounting tab. However, this design specifies four mounting tabs; this will divide this stress by four. The total max stress per tab then becomes 9.3 ksi.

Ejection System Analysis

In the following section the method of calculating the velocity of the payload will be discussed. This calculation involved a multi-step force and momentum balance. The system was modeled using a free-body diagram from which a system of equations was formed and solved. In addition a model of the system was created within Working Model and a simulation was performed. The values from the simulation were then compared to the numerical solution. The goal of this analysis was to provide a baseline to show the approximate amount of force that needs to be supplied by the air cylinder to give the payload an initial velocity of at least 4 m/s.

The equations modeling the system were formed using a free-body diagram including the hook, ejection piston, and the payload. The air cylinder was modeled as a point force acting on the hook; no other components of the system were included. This analysis was done assuming that all frictional forces are negligible.

The first thing to realize is that the hook and piston will assume equal velocities because they are coupled together at 45-degree angles. Their combined mass related to the force provided by the air cylinder is what will determine their acceleration. The system is designed so that the hook and piston begin to move before the piston contacts the payload. This gives the piston time to gain speed and momentum before contacting the payload. The equations were transformed into state-space form and loaded into MathCAD. Utilizing MathCAD's equation solving capabilities, this distance was optimized and it was determined that a spacing of 0.14 inches between the cylinder and piston will give the highest initial velocity. Once the acceleration of the piston is known and the vertical travel distance is known, the velocity of the piston upon contacting the payload can be calculated. This is the end of the first step of the momentum analysis. Although 0.14-inches gives the highest velocity, the hook design will not work will such a small spacing. The hook has to travel long enough for the hooks to slide out of the way before the piston contacts the payload. Because of this design constraint, the spacing that will be used is 0.32". This spacing is shown on the next page in figure 28.

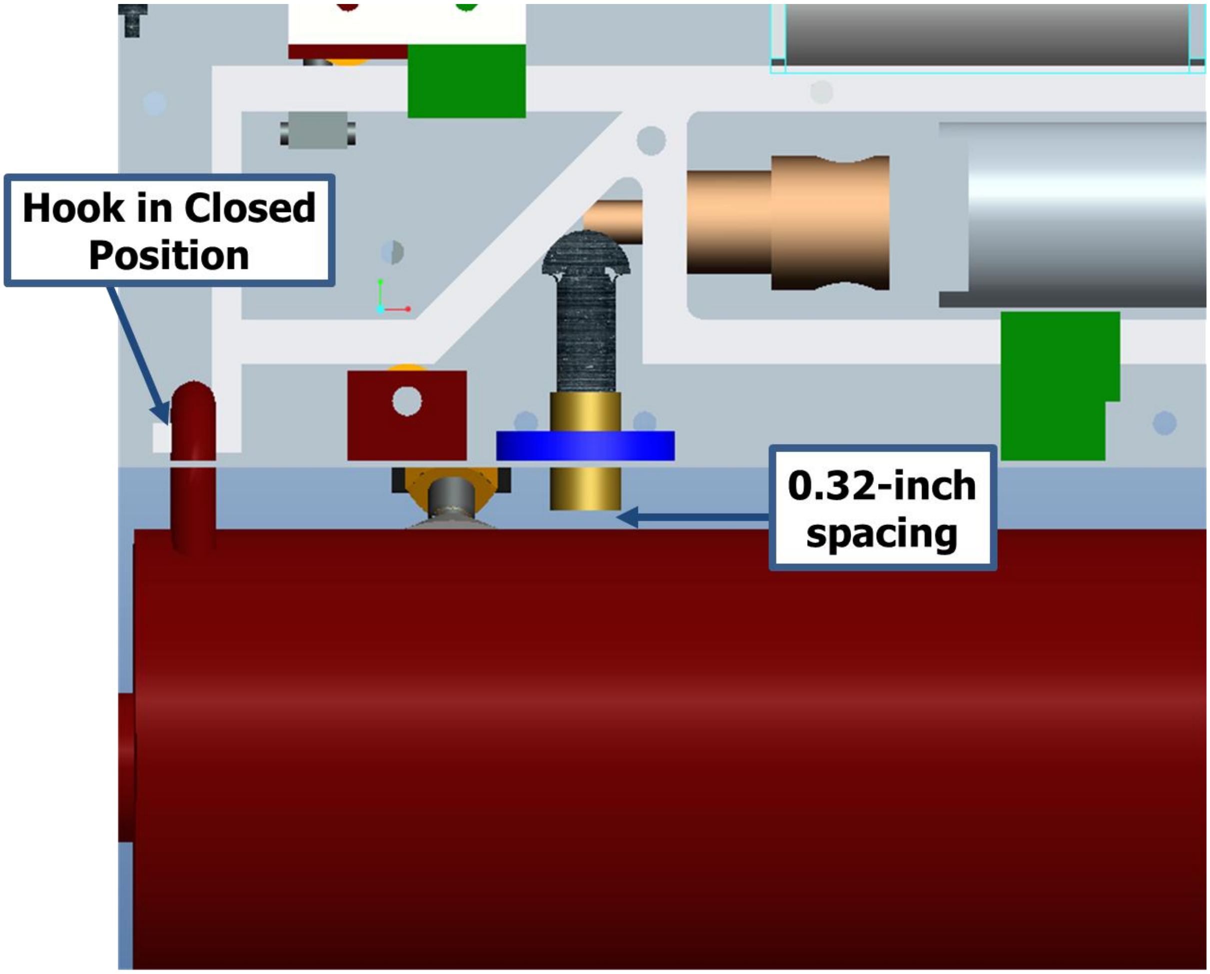

Figure 28- Ejector piston in retracted position

The second step of the momentum transfer analysis involves an assumed perfectly inelastic collision between the piston and the payload. During this step of analysis, the mass and velocity of the hook and piston right upon impact are related to the mass of all three components and a new common velocity. This second step is shown in figure 29 on the next page; the piston has contacted the payload and will not begin to accelerate the payload away from the BRU.

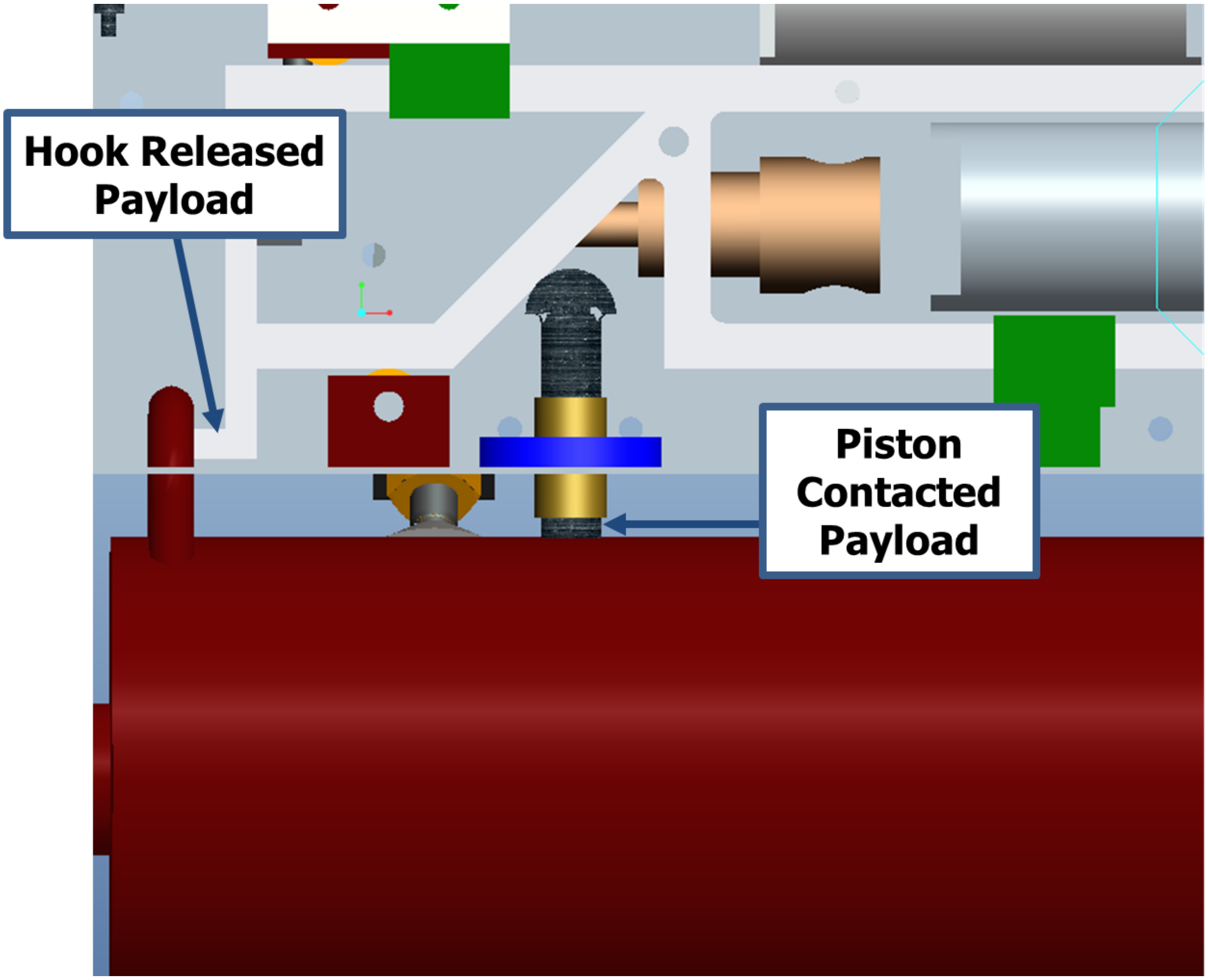

Figure 29- Ejector Piston Contacted Payload

The third step of analysis involves the continued force transfer from the air cylinder which is now acting upon the hook, piston and the payload for the remainder of the hook and piston travel. The total piston/hook travel is one-inch, so the air cylinder continues adding force to the system for 0.86-inches. At the beginning of this step, the payload had already assumed a velocity from the momentum transfer and continues to accelerate during the force transfer. At the end of the piston travel, the payload will now have its total initial velocity gained from the BRU and will continue traveling in free-fall. This final stage of the ejector process is shown on the next page in figure 30.

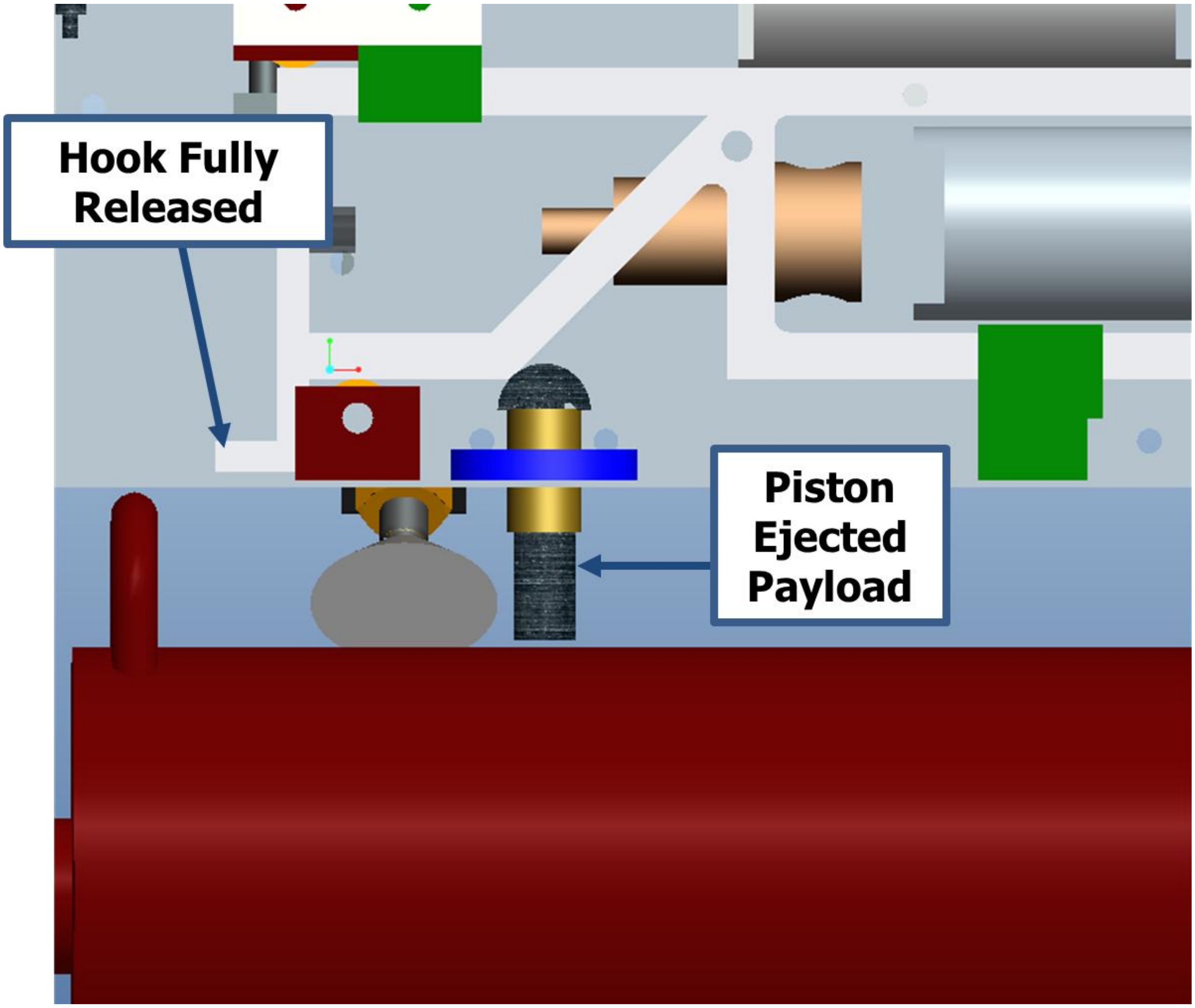

Figure 30- Ejector Piston Contacted Payload

The details of the Working Model simulation will now be presented. Figure 31 on the next page shows the model built within Working Model. The model was built with a 2 blocks connected at 45-degree angles. These blocks represent the sliding hook and the piston. They are connected to ground in the model by keyed slots. The force from the air cylinder is modeled as a point force on the hook. The payload is modeled as a rectangle box position at 0.14-inches below the piston. The weights of each piece are entered into Working Model and the simulation is run.

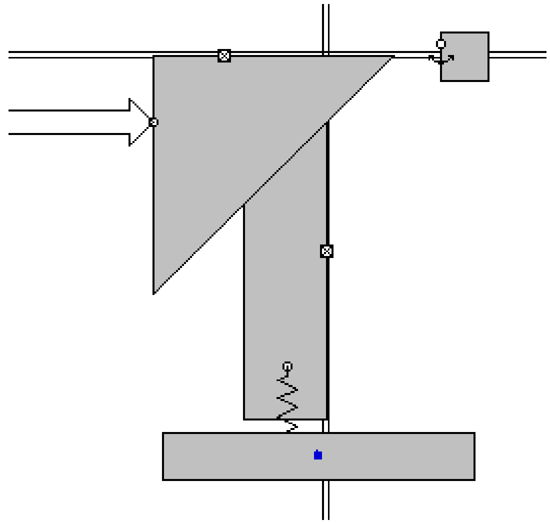

Figure 31- Velocity Model Within Working Model

The Working Model simulation solutions were compared to the numerical solutions provided by MathCAD. Figure 32 below is a graphical representation of the simulation results; payload velocity in inches/second is graphed versus time in seconds. There are two definite slopes on the velocity curve. The first slope is much steeper and corresponds to the time that the piston is in contact with the payload. This is the time that passes as the hooks are releasing and the payload is being ejected from the BRU. Around the time 0.05s is when the slope backs off and this corresponds to the velocity of the payload as it is free falling. At this point the payload velocity is 5.33 ft/s so this is the expected initial velocity of the payload, based on the simulation.

The numerical solution for the initial velocity was calculated to be 8.47 ft/s. These values will be compared with the actual values once the prototype is built and testing begins.

Ejector Mechanism Simulation

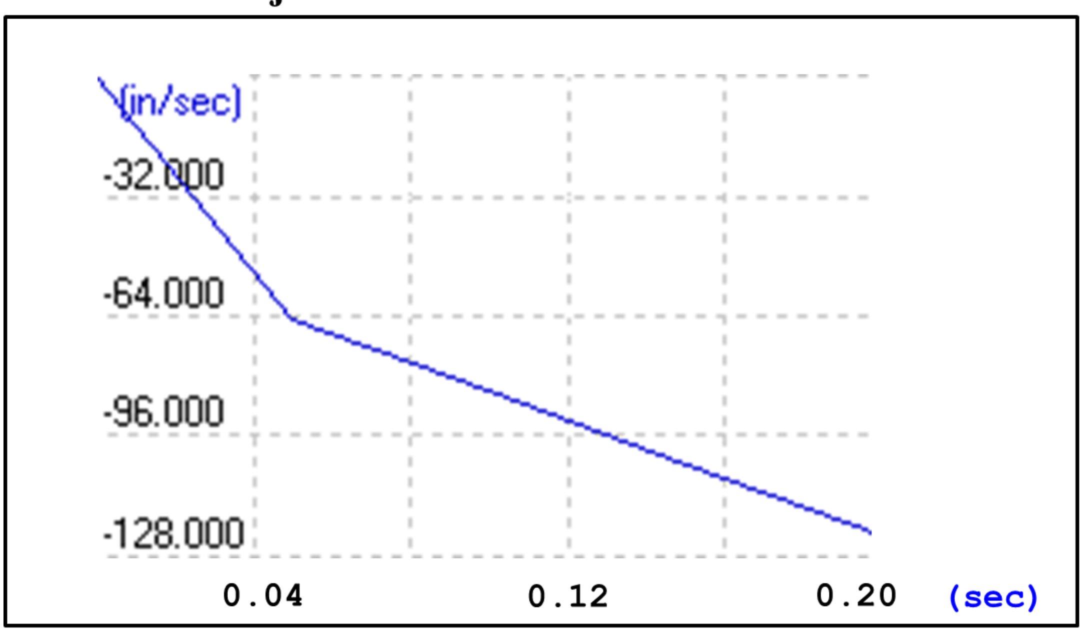

Figure 32- Ejector Mechanism Simulation

Mechanical Safety Analysis

For the final design we decided to use design 6 for the Mechanical Safety because of its compactness and lightweight. Shown below is the final drawing for this system before it was placed in the BRU assembly.

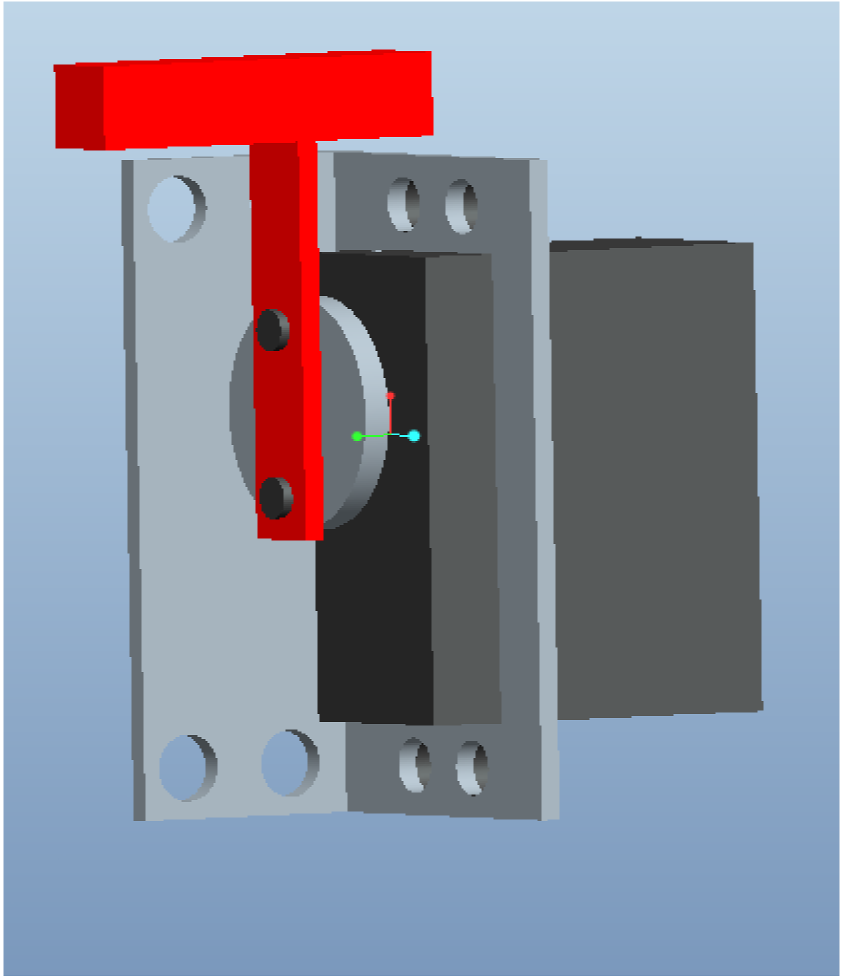

Figure 33 - Final Mechanical Safety Drawing

In the figure are the servo motor, bracket, and the stop block. The stop block is colored red and in the shape of a T. This was done to minimize the weight of the block, putting less strain on the servomotor when it has to rotate the stop block out of the way. The top of the T will rotate in between the front of the hook and the front wall of the BRU. Doing this takes all the stress that is placed on the stop block when the "Release" command is given before being "Armed" off of the servo motor.

Since the servo motor only has to be able to rotate the stop block out of the way the only calculation done on it was to make sure the motor has enough power to rotate the block out of the way. The servo motor is rated with a torque of 4.75 lb-in at 4.8 volts, which is much greater than the 0.0066 lb-in needed to rotate the stop block out of the way. The torque rating of the servo motor was converted to lb-in from the 76 oz-in given off the web page. The torque required to rotate the stop block by first, assuming that the point where the overall torque being applied is half way between the mounting holes. Next a Pro/Engineer analysis gave the mass and the center of gravity location for stop block. To get the distance from the center of gravity to the midpoint in between the mounting holes the midpoint was subtracted from the center of gravity, which gives 0.766 in. That distance was then multiplied by the total mass given by Pro/Engineer to get the 0.0066 lb-in reported. The torque created on the servo motor by the stop block being off centered was not calculated; it was assumed that this value would be extremely small and pose no threat to the servo motor.

A finite element analysis was done on the stop block to make sure it could withstand the stresses felt from the hook when improperly used. The pneumatic piston being used is rated at a 155 lb force at 100 psi of pressure. However, we only plan to run our system at 90 psi, but an analysis was still done at 155 lbf to ensure that the block will easily stand up to the force of the piston. Below are some figures of the Pro/Engineer Mechanica analysis done on the stop block.

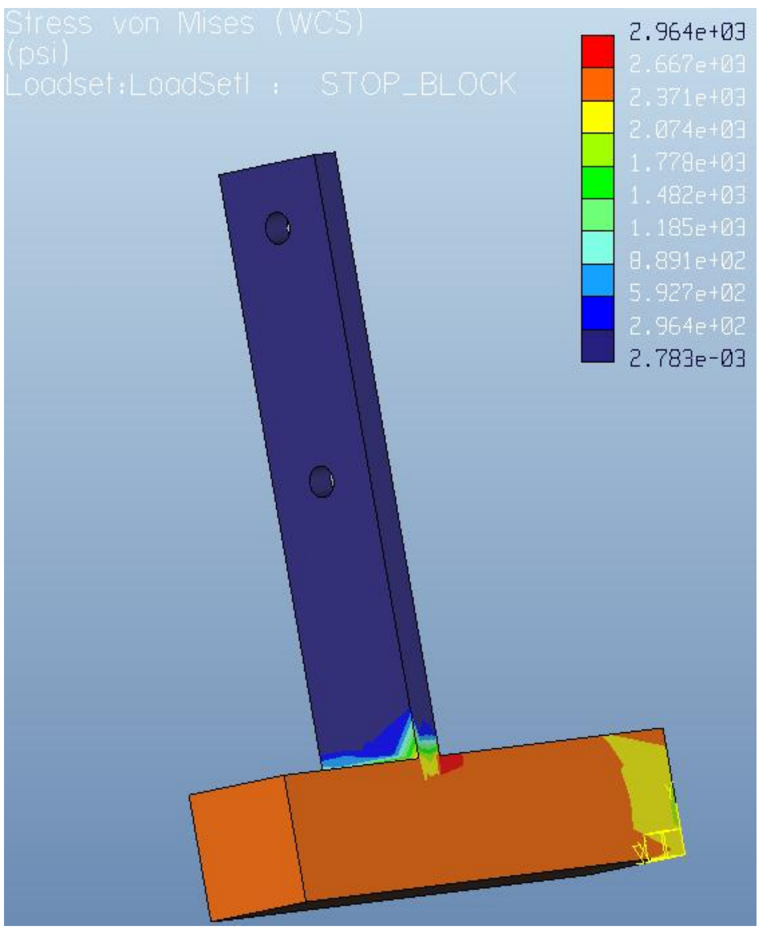

Figure 34- von Mises Stress Analysis of Stop Block

In the figure above, figure __, is the von Mises stress analysis of the stop block. To obtain this analysis a force was placed on the top end of the T that is facing outward. This force is compressive and went towards the other end of the T. An all degrees of freedom constraint was placed on the end of the t top that is facing inward. The maximum overall stress felt by the stop block is 2964 psi, as the color scale shows in the top corner of the figure. This stress is well inside the elastic region of Aluminum 6061, which has a yield strength around 8000 psi, giving a factor of safety of 2.7. This ensures that the part will not fail under any load presented by the hooks. The next figure shows an analysis of the strain the stop block undergoes when stressed.

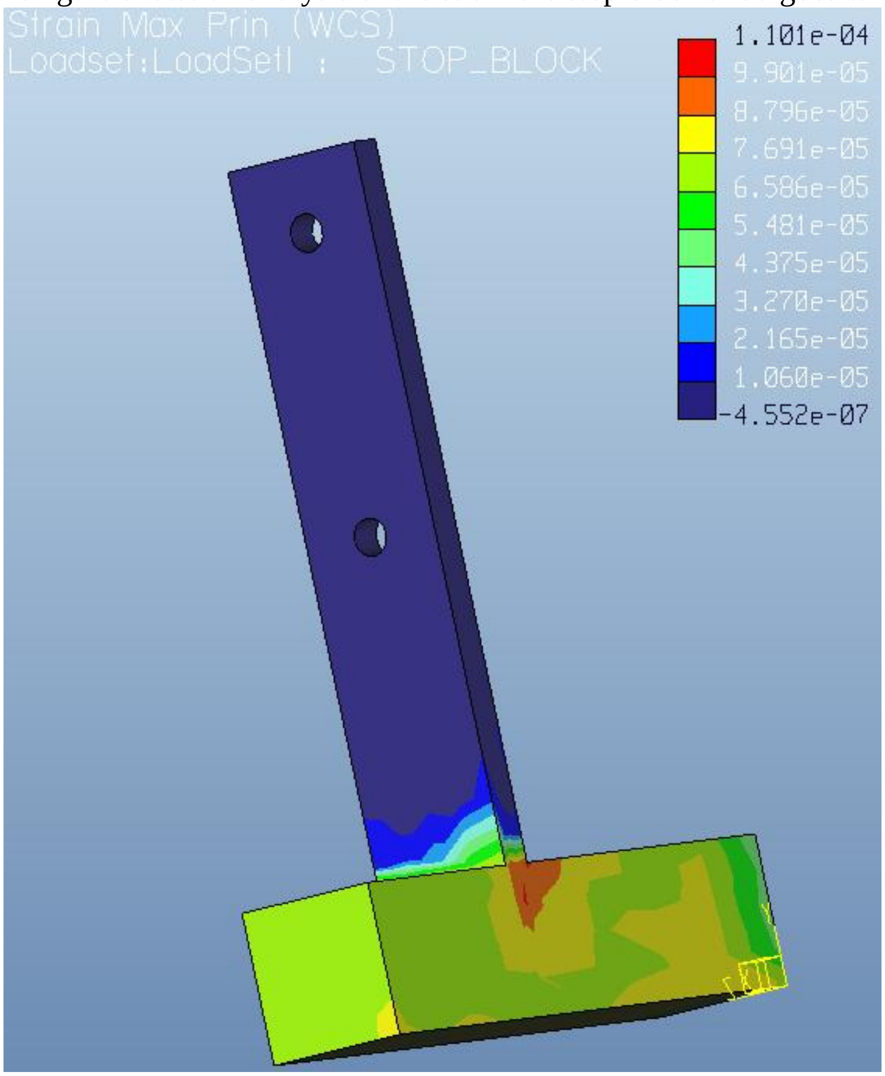

Figure 35- Strain Analysis of the Stop Block

In this figure the force of the hook was placed in the same place as the von Mises Stress Analysis shown previously. As the scale in the top right corner indicates, the stop block undergoes a max strain of 0.0001101. Since the stop block undergoes stress that is much lower than the yield stress, and the strain is very low it can be stated that this deformation is elastic. This means that under this load the strain, or deformation of the stop block is not permanent and the block will return to its original shape after the load has been removed.

Since the stop block will rest against the side plate directly in front of it, an analysis of this part was also necessary to ensure that failure would not occur. Below are a couple of figures showing the analysis done in Pro/Engineer Mechanica.

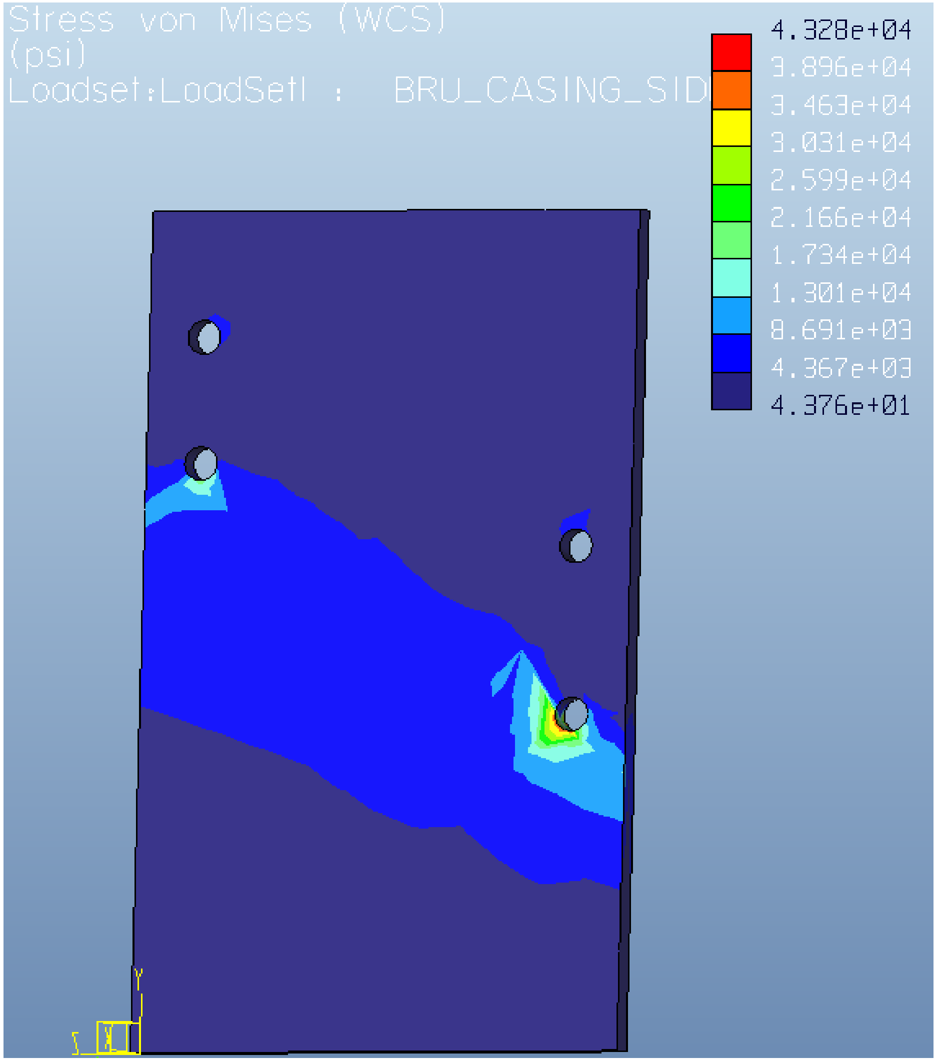

Figure 36- von Mises Stress Analysis of BRU Side Plate

The first thing to notice in this figure is the scale in the top corner that shows a max stress of 43,280 psi. As the figure shows this stress is over a very small region by the mounting holes of the plate. However, since the plate will be mounted with screws and brackets this stress concentration will go done tremendously when the system is built. Therefore the highest stress felt in the system at the 155 lbf tested will be 4367 psi, which is the royal blue color shown going across the middle of the part. Remembering that the yield strength of Aluminum 6061 is 8000 psi, this gives a factor of safety of 1.8. This again ensures that the part will not fail due to stress. The final figure in the analysis of the mechanical safety system is the strain analysis shown below.

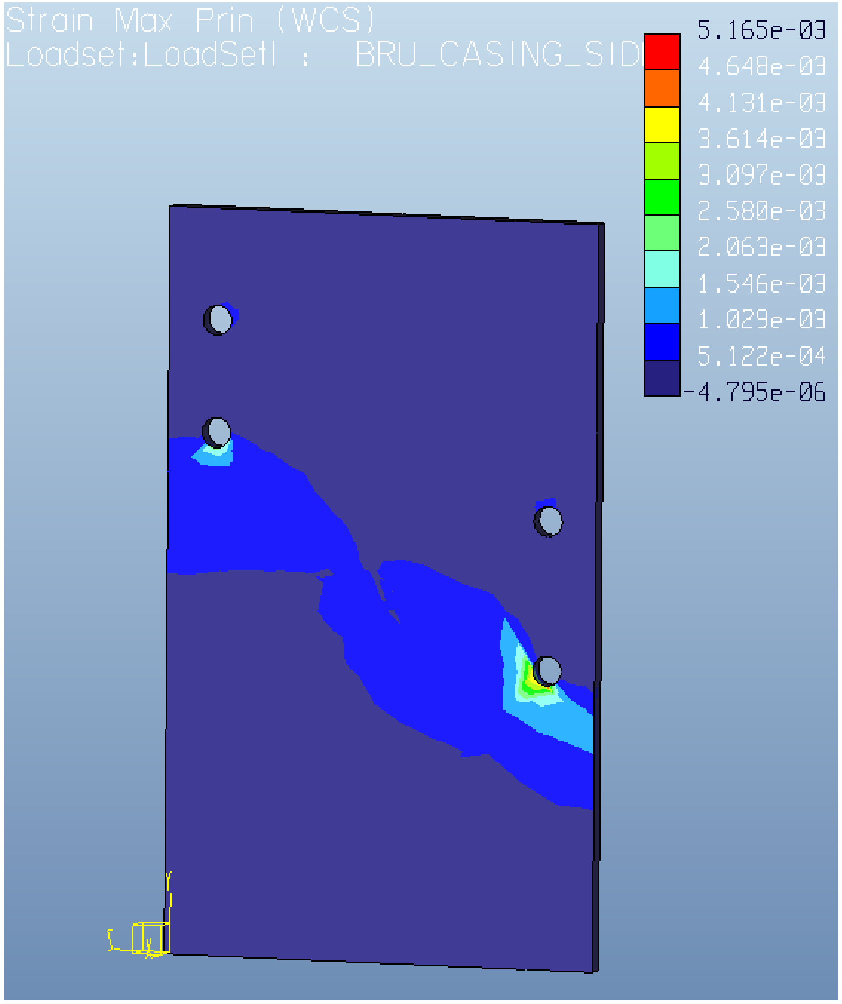

Figure 37– Strain Analysis of BRU Side Plate

Again, remembering that the area of interest is the sections in royal blue, the max strain of the system is approximately 0.0005122. This then assures that the system will be deforming in the elastic region and that the part will return back to its original shape. This also shows that the system as a whole undergoes a very low strain, giving a very low displacement when loaded and ensuring that the system will not be able to misfire when it is not armed.

Sway Brace Analysis

Assumptions

- Sway brace brackets takes only compressive loads.
- Pads absorb both normal and shear loads.
- Sway brace is assumed to be rigidly connected to the BRU which can be considered a high strength structure.

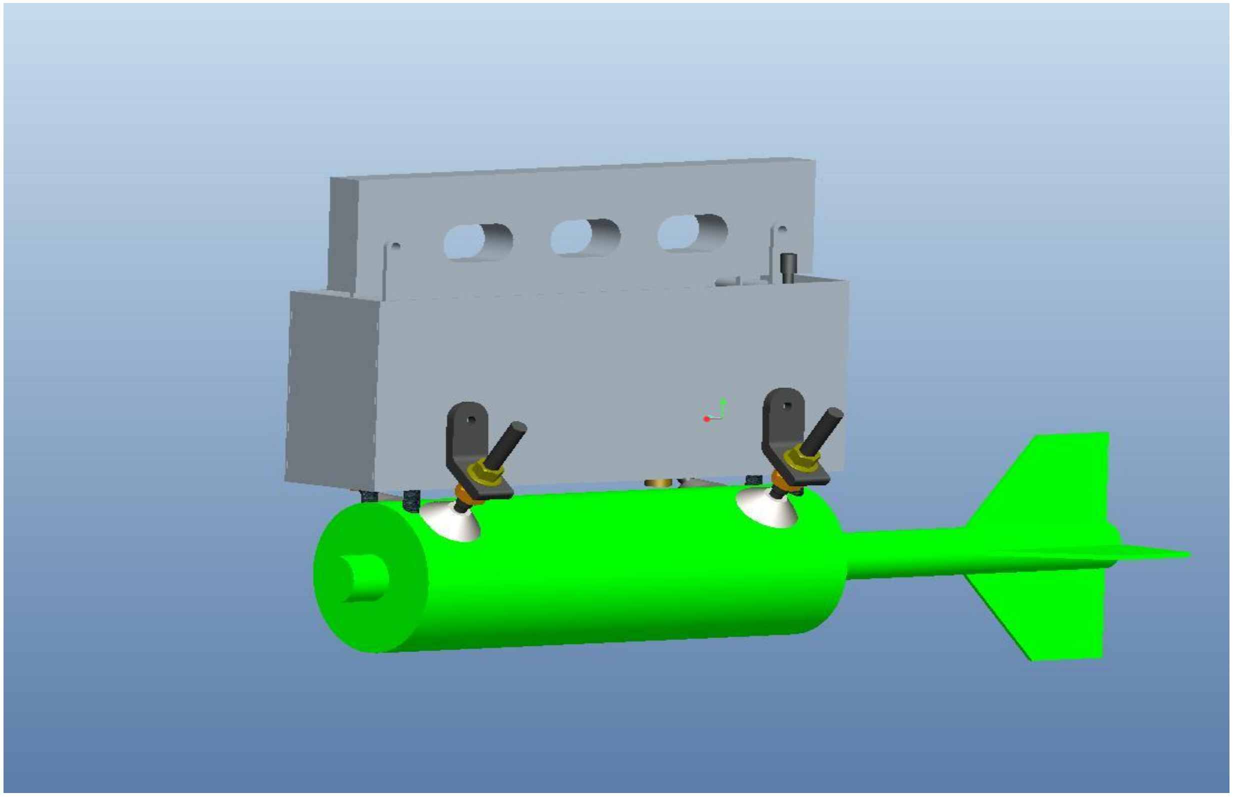

Figure 38- Sway Brace Mounted to BRU with Payload

The sway braces are designed to withstand the lateral and vertical loads during in flight operation of the BRU. During payload release the sway brace keeps the payload stable allowing proper ejection. The sway brace must be able to retain the payload during aircraft maneuvers up to 2GS of lateral load and 1G of landing shock. The analysis of the sway brace can be divided into two major components; the sway brace mounting bracket as well as the pad that come into contact with the payload.

The mounting bracket will be machined out of Aluminum 6061, which has a maximum yield strength of 8000 psi. This is the maximum stress before the bracket will start to experience plastic deformation. Using the Pro/Engineer Mechanica application, the maximum stress and the von Mises stresses can be located on the bracket. The part can then be redesigned to lower these concentrations, which will increase reliability of the part. Below is the analysis done by Mechanica on the sway brace brackets.

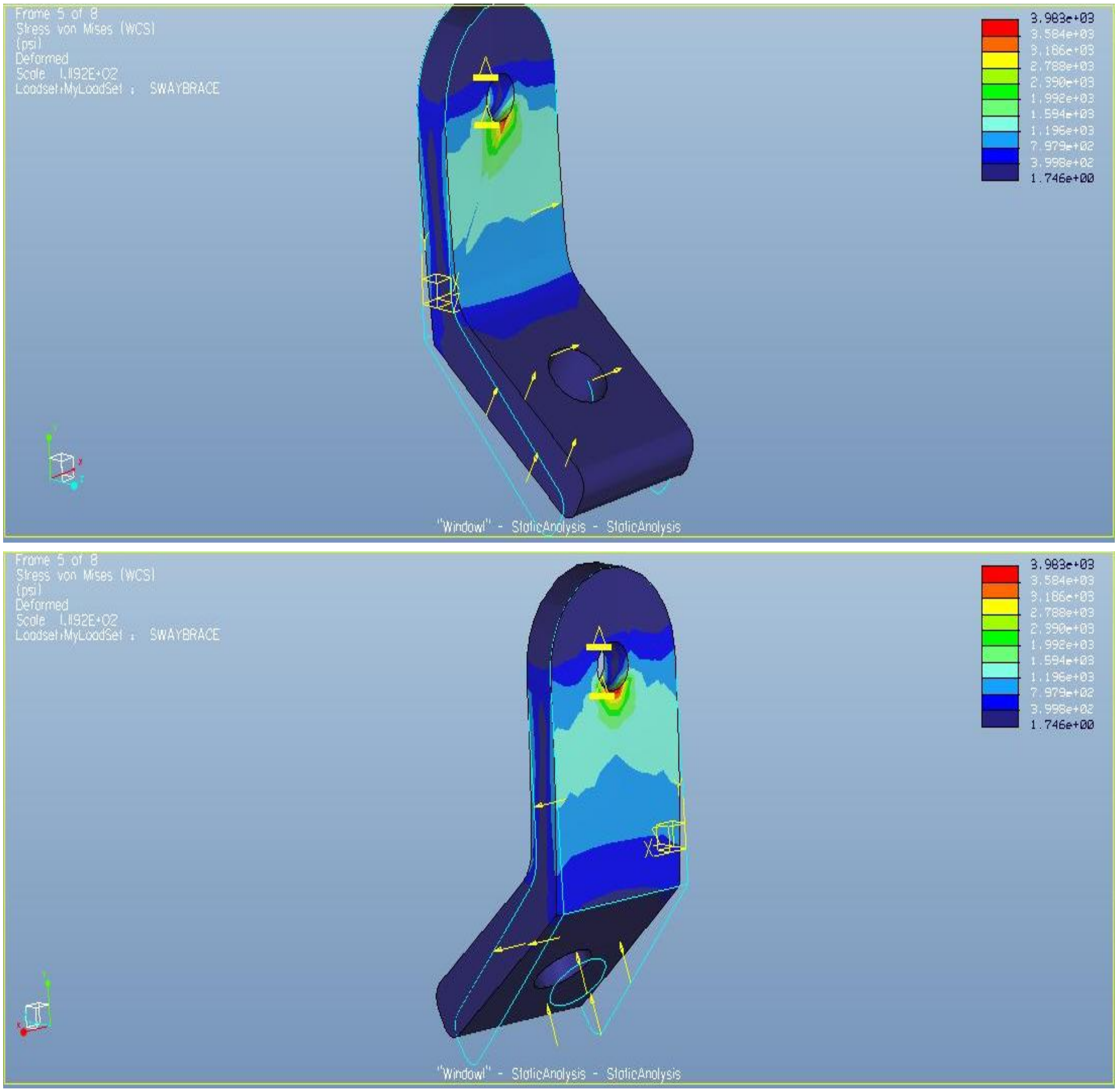

Figure 39- Bracket Von Mises Stress (Front and Back)

The part is constrained by the bolt hole on the top face of the bracket. Using a safety factor of 1.5 the bracket experiences a 15lbf compressive force normal to bottom bolt face. In flight the bracket experiences wind force; this was estimated using 75mph across the face of the BRU. Using these loads in Mechanica, maximum stress concentration occurs at the top bolt face of the bracket. The highest von Mises stress recorded occurred at the bottom of the bolt hole recording a stress concentration of 3983 psi. This load is well below the 8000psi yield strength of Aluminum 6061, thus keeping the bracket in the elastic region. Using the formula below the force of the wind across the side of the bracket can be calculated.

This formula outputs a wind force across the side of the bracket . This will also change as the UAV turns producing varying wind loads on various sway braces. A safety factor of 1.5 was incorporated into the design to reduce any chance of failure.

The next component of the brace that needs to be analyzed is the steel mounting pads that come into contact with the payload. These steels pads absorb both the normal and shear forces from the payload, keeping it stable as the UAV performs various in-flight maneuvers. Using Mechanica and applying both the normal and shear forces, the stress concentrations of the part are revealed. Normal force of 30lbf was applied to the face of the steel pad with a shear force of 6lbf. When applying these loads a safety factor of 1.5 is used to prevent any type of failure. Below is the output from the Mechanica finite element analysis.

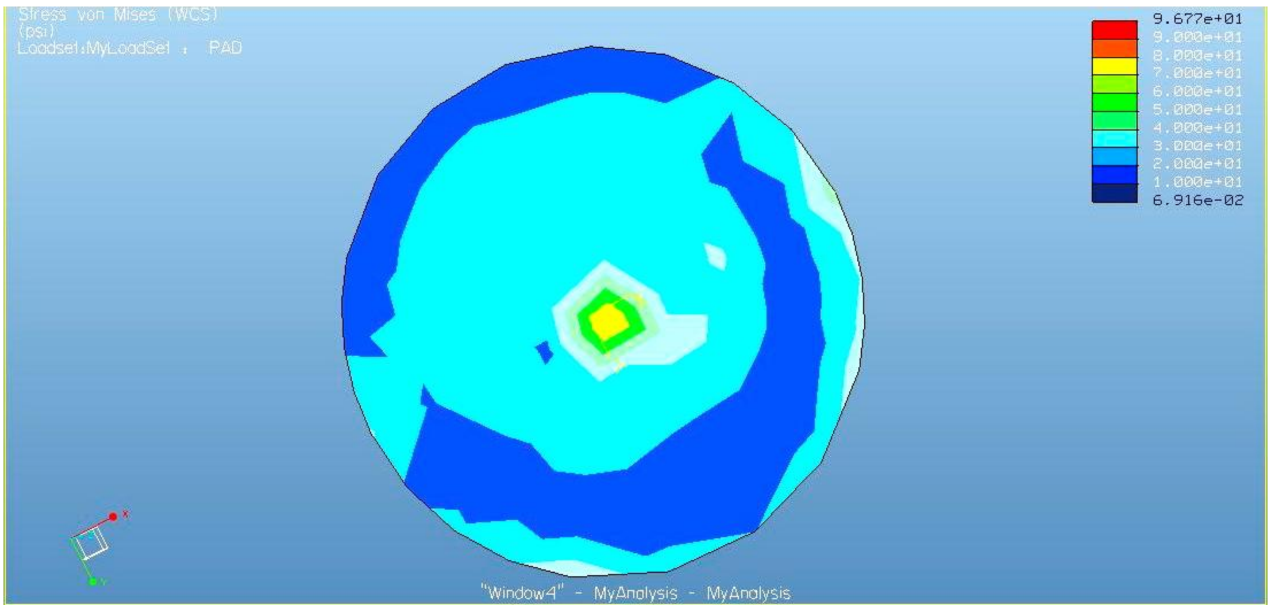

Figure 40- Steel Pad Von Mises Stress (Bottom Face)

Mechanica reveals a maximum stress concentration of 60psi, which is well within the elastic region of steel. There will be no plastic deformation that occurs in flight. The maximum stress concentration occurs at the center of the pad, which is where the bolt is mounted into the pad. This will experience upward normal force which creates a stress concentration along the center face of the steel pad.

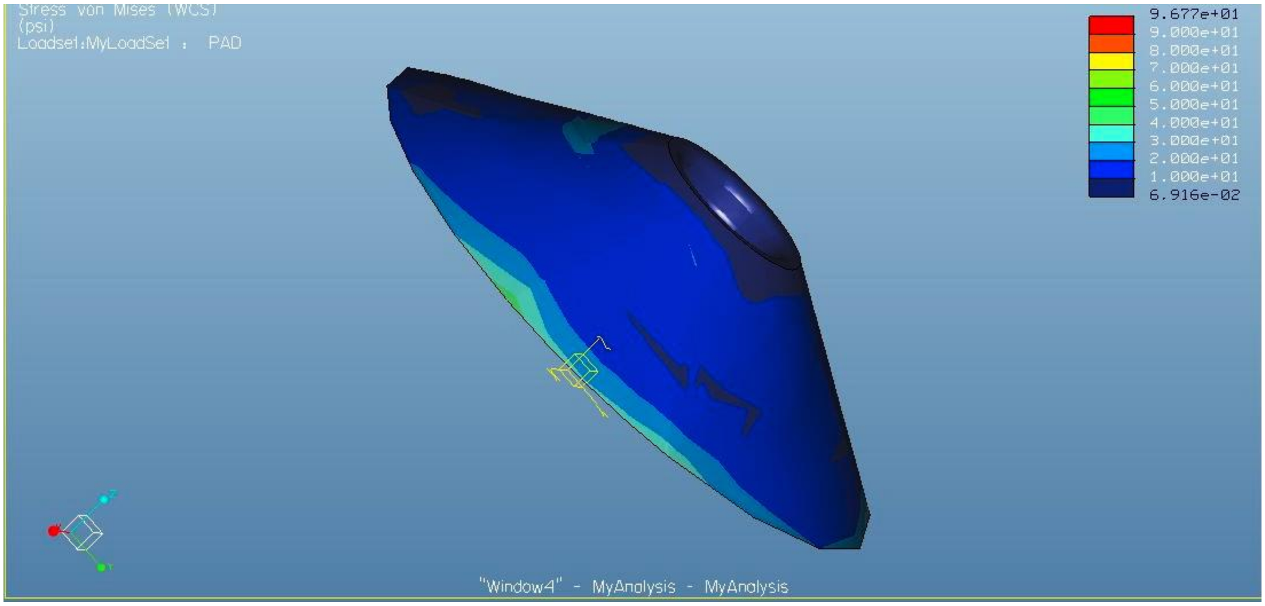

Figure 41- Steel Pad Von Mises Stress (Side Face)

The side face of the pad experiences little to no major stress concentrations. The majority of the stress is located on the bottom/edge of the steel pad. The stem of the sway brace is connected to the steel pad via ball joint, which allows the stress to be distributed along the face of the part. This also allows the pad to swivel if necessary under various UAV airplane maneuvers.

Overall this sway brace design is commonly used in much larger aircrafts, due to the reliability and the ability to house various sizes of payload. With the stem being able to translate, allowing the BRU to hold multiple types of payloads.

Electrical Interface

A micro controller will be the main control between the user in arming and firing the payload. The controller will be expected to send and receive signals to the solenoid valve, servo motor, and contact switches. The MicroDragon will be the microcontroller that will be used to input the program, as well as a motor driver will be used to output the high voltage needed to open and close the solenoid air valve. The MicroDragon comes pre-installed with Codewarrior which is the program that is used to write code for the BRU. This microcontroller uses a 5V input to run, and with the Motor driver is capable of outputting up to 30V. Below is a basic flow chart of the program that will control both safety servo motor, as well as opening the solenoid for the firing operation.

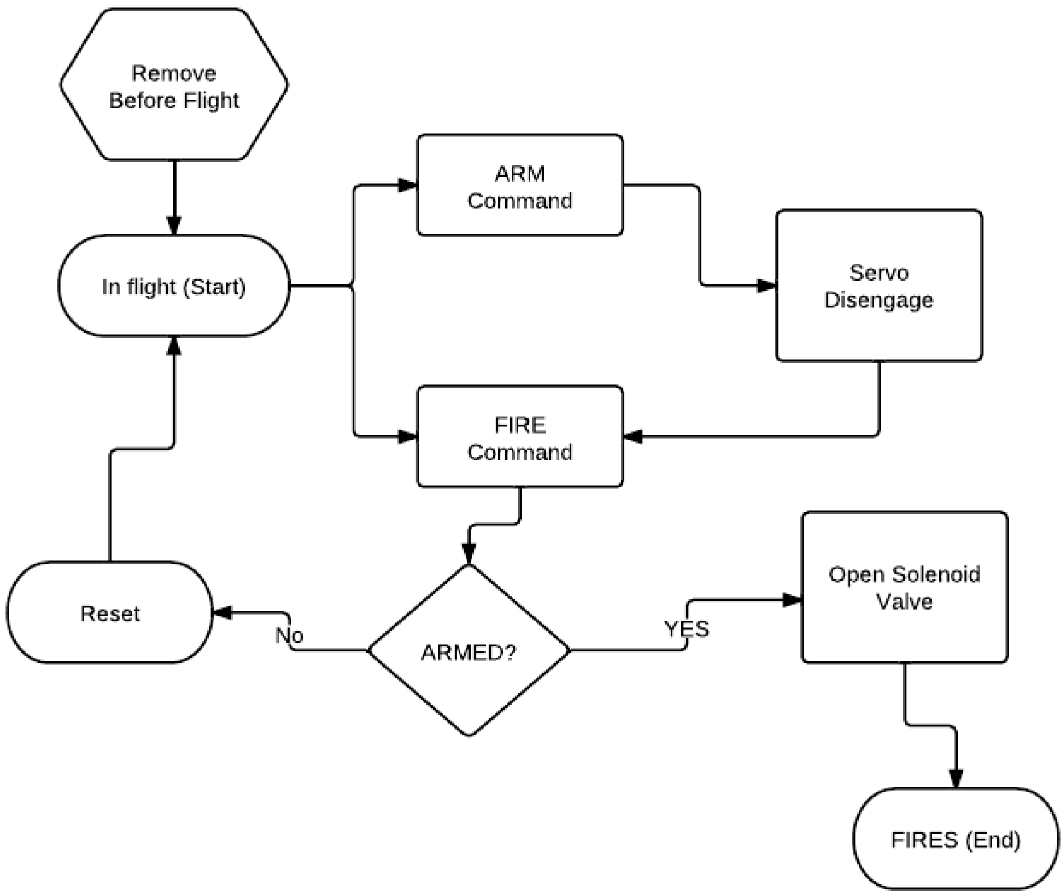

Figure 42- Flow Chart for proper arming and ejection program

The process starts for the BRU when the UAV is on the ground. The first step is to remove the "Remove Before Flight" pins, which goes through the BRU and prevents the system from arming or firing. Once this is removed the UAV is cleared for flight with payload attached. Once the UAV is in flight the program begins giving the user two options ARM or FIRE. If the FIRE command is selected the program will confirm if the system was armed, if not program resets to in flight status. If the user selects ARM the servo motor releases the safety block which will allow the hooks to translate. Then the user has the option to select FIRE. The program in this case will check if the system is armed, which in this case it would be. Current is then sent to the solenoid which opens the valve that releases the compressed air. This will send air to the cylinder which starts the release of the payload.

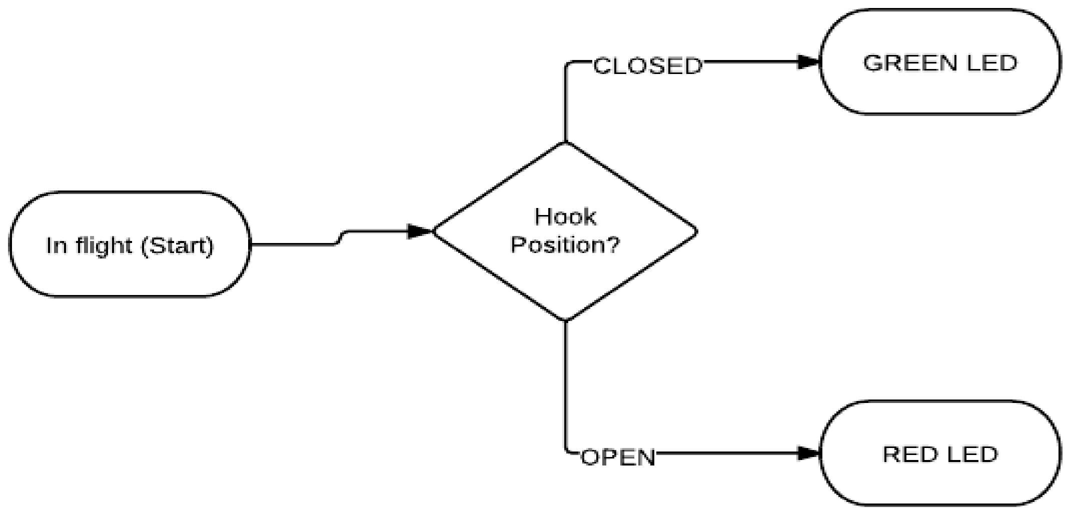

Figure 43- Flow chart of the hook position program

The second program that is run by the microcontroller, tells the user if the hook is either in the open or closed position. A contact switch will be placed in the center of hook assembly which if in the closed position a green LED will be lit. This will tell the user the payload is stable and in the closed position. Once the hooks translate for the ejection process the hook assembly will come into contact with the switch will output a red led, thus showing the hooks are in the open position.

Specifications for the MicroDragon and the motor driver are located in the appendix.

Environmental and Safety Concerns

For this design there were no environmental concerns with the construction of the BRU. However, safety is a big factor in the design of this system, especially since there is a possibility of people's lives being at stake. The biggest concern was with the safety system of the BRU, this being the system that stops the payload from being released unintentionally. The smallest factor of safety between the stop block and the side wall that are being used to stop the hook was 1.8. This means that the system will be strong enough to hold under the force of the hooks. The system also will have holes cut into it to allow the operators to easily identify if the system is in the armed position. This will ensure that the operators will be able to quickly take the appropriate measures to disarming the system if need be.

Conclusion

The first semester of our design project has been successful. As this report shows, we have meet the constraints given by our mentor Russell Roberts from AFRL at Eglin Air Force Base. Pro/Engineer aided in stress analyzation, and provided a visual model of the system. MathCAD also assisted in conducting numerical analysis. The design was successfully optimized based on weight, reliability, durability, and size. The design cost of $671.91 is well under the $2000 budget, giving leeway for next semester if more parts are needed.

Our next step is ordering the parts required to assemble the prototype. During the next semester a prototype will be constructed and tested against the constraints. Close attention will be paid to the weight of the system to ensure that the total weight of the system stays under the five pound constraint given. If possible a fit test will be done with the Tigershark UAV to complete full integration.

References

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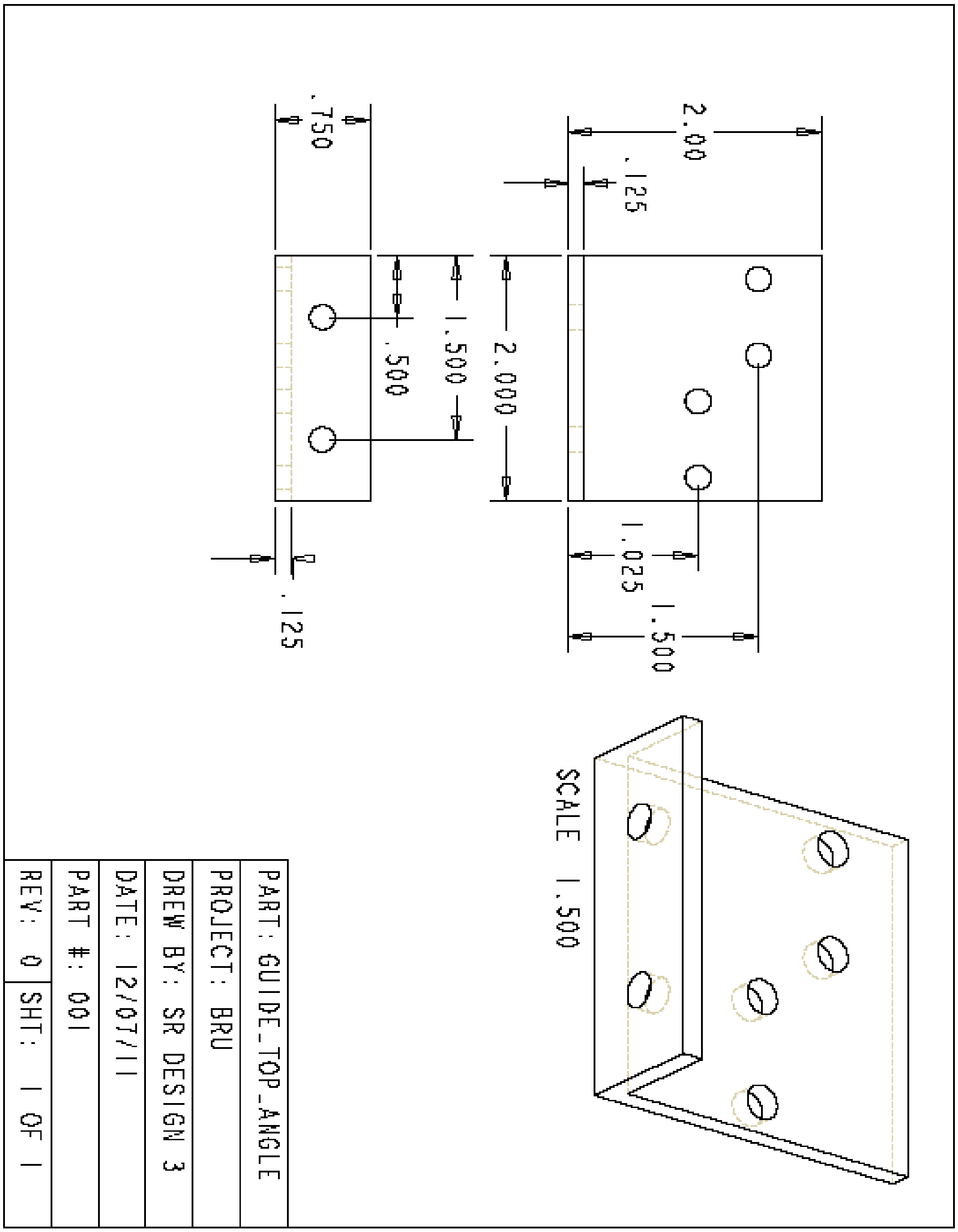
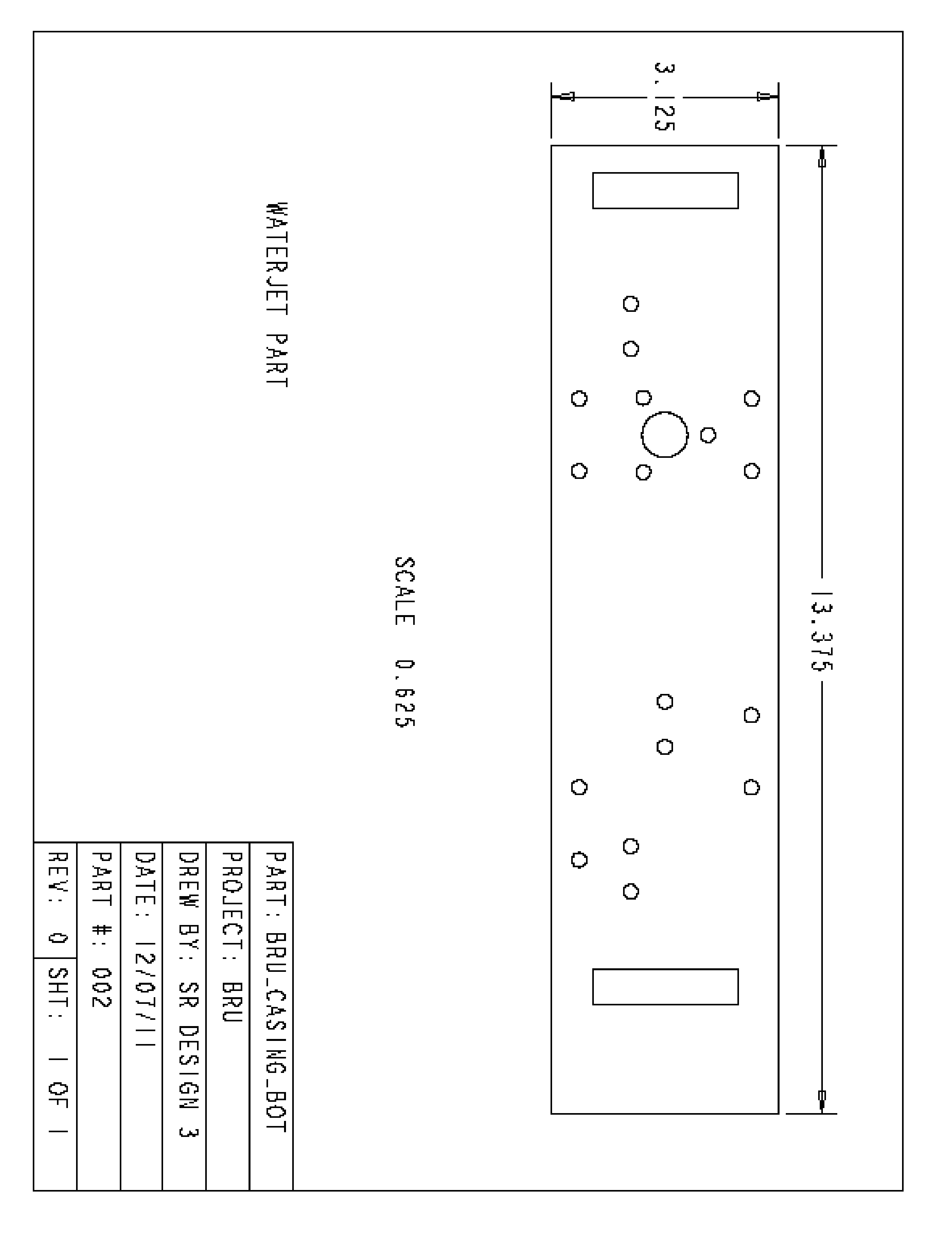
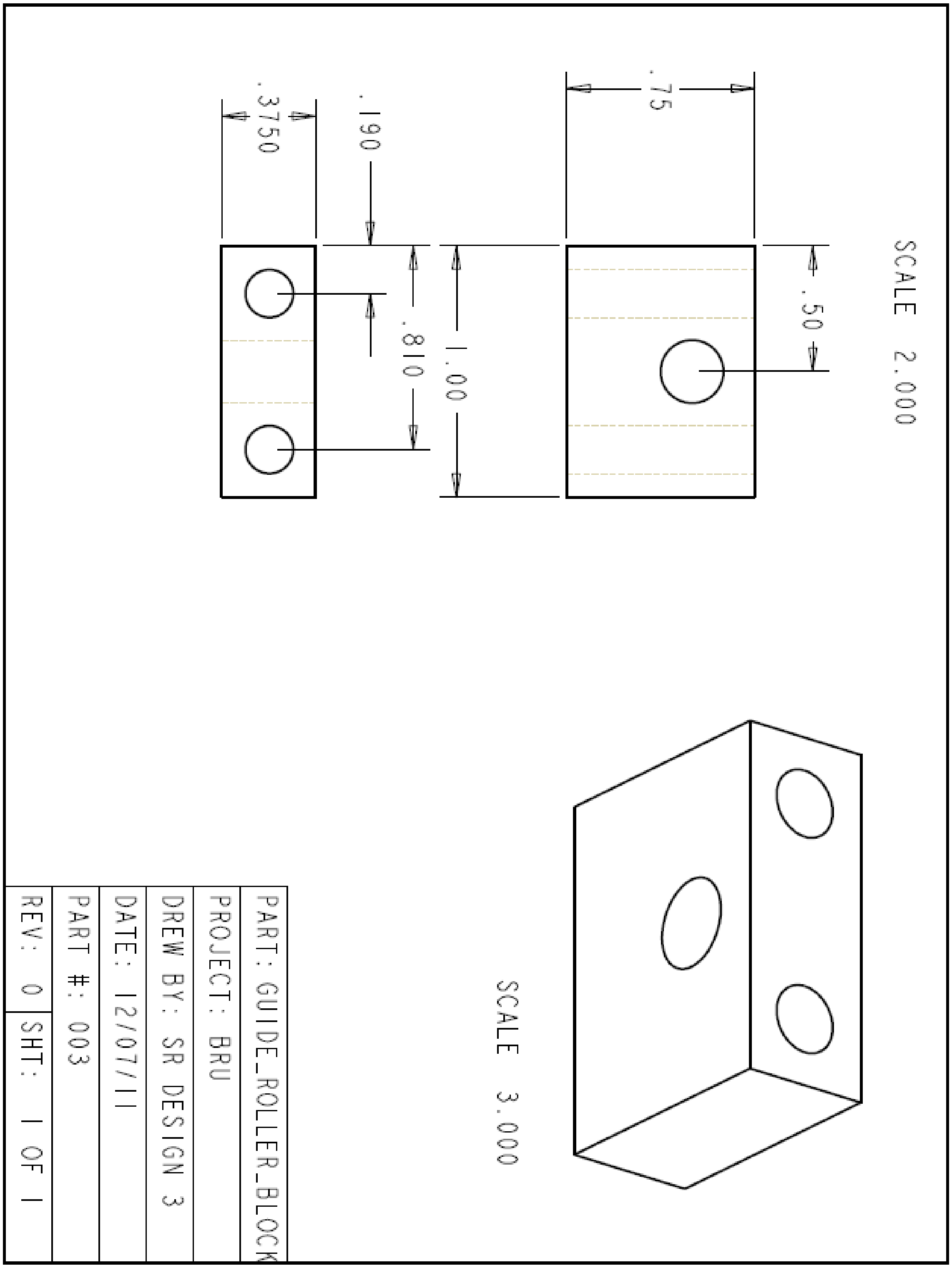
Appendix

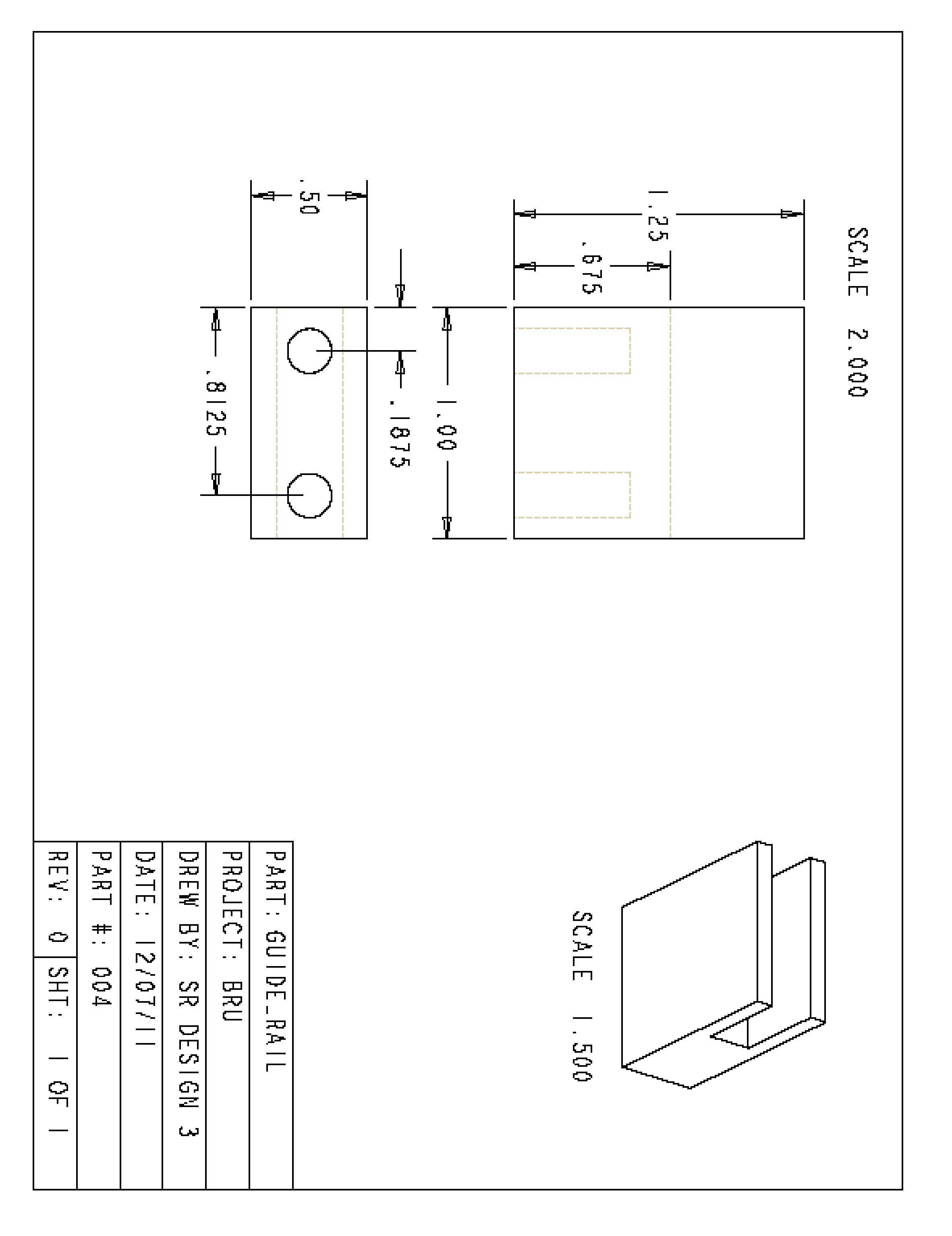
Part Drawings

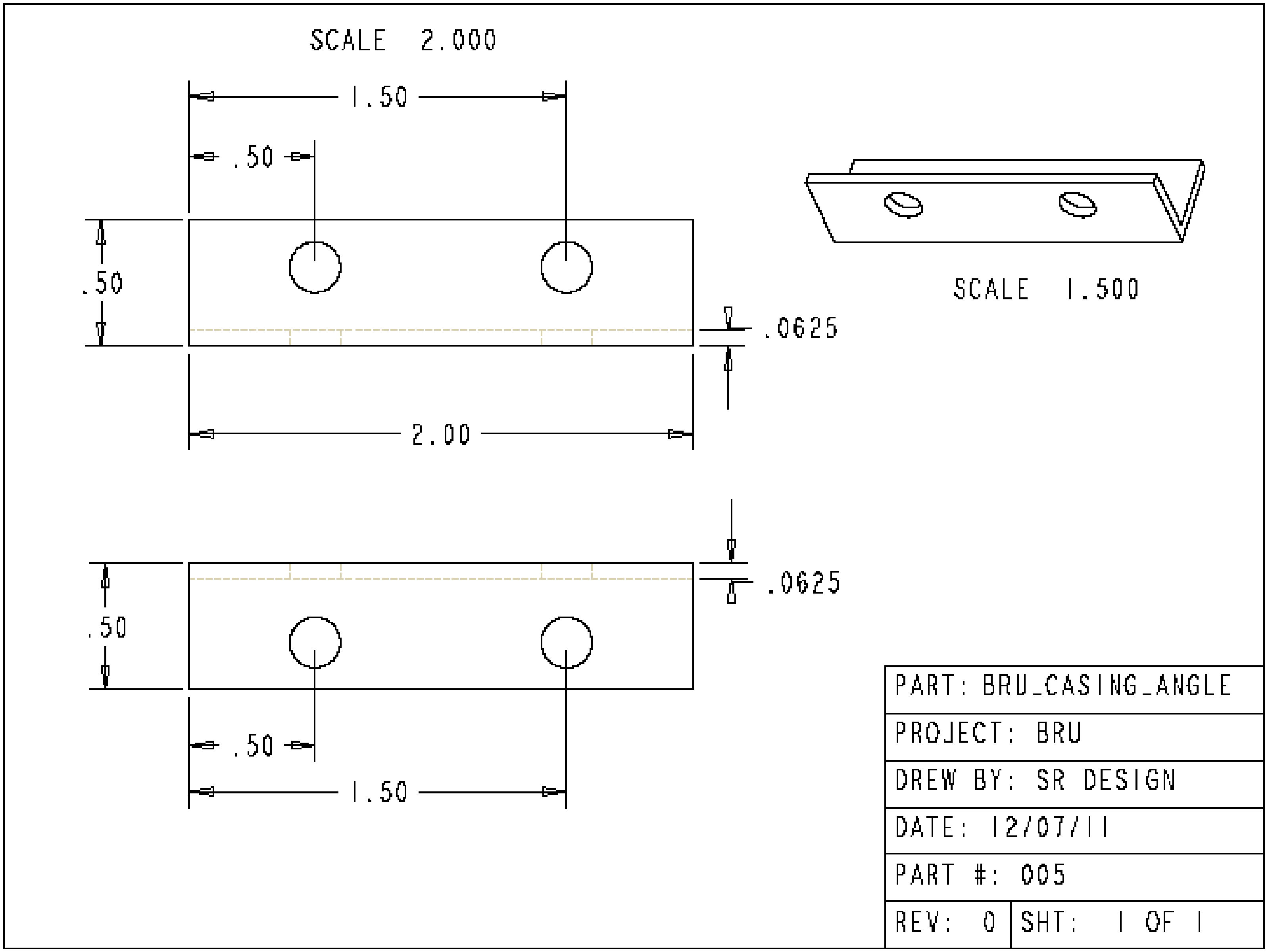


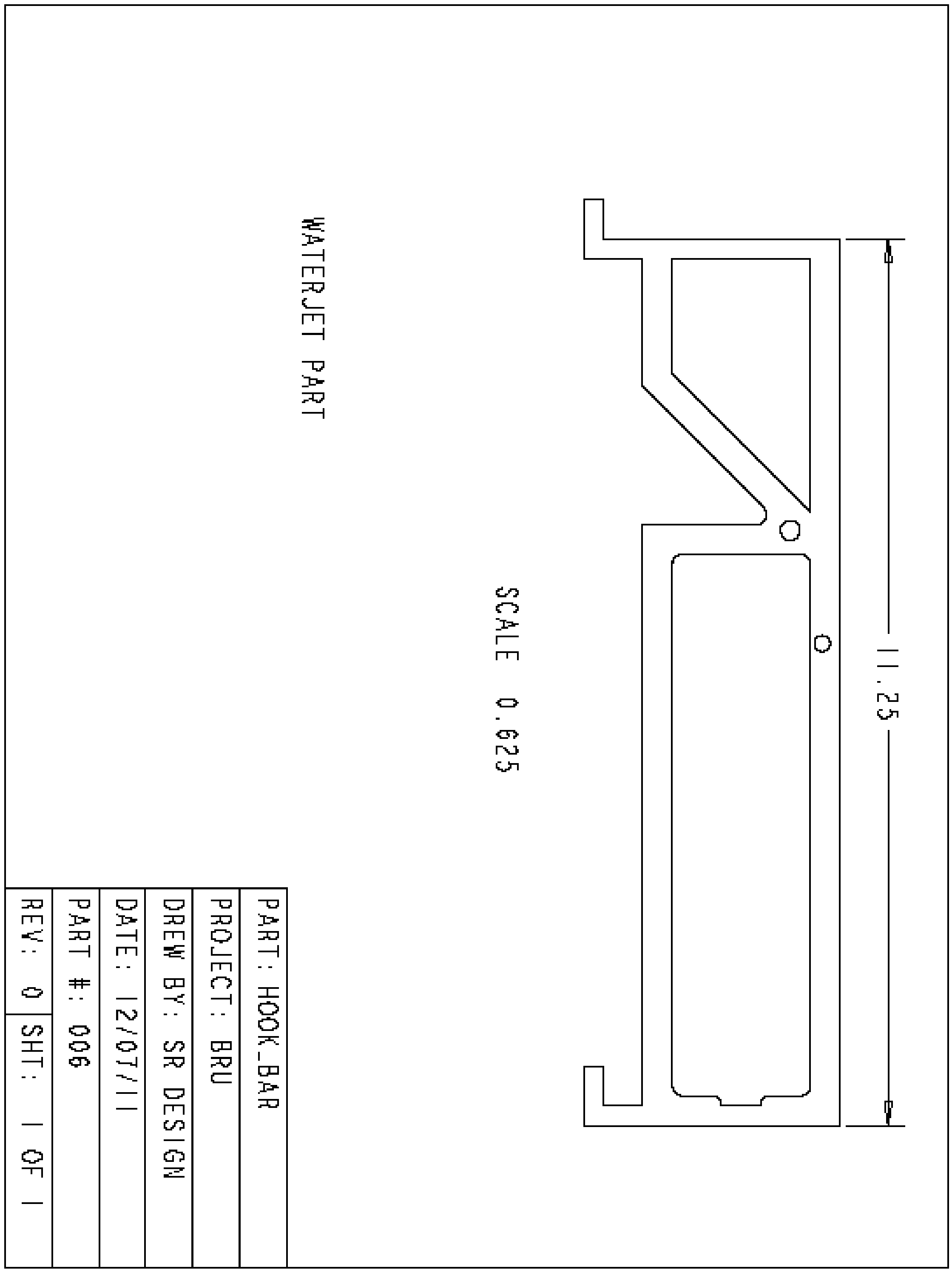


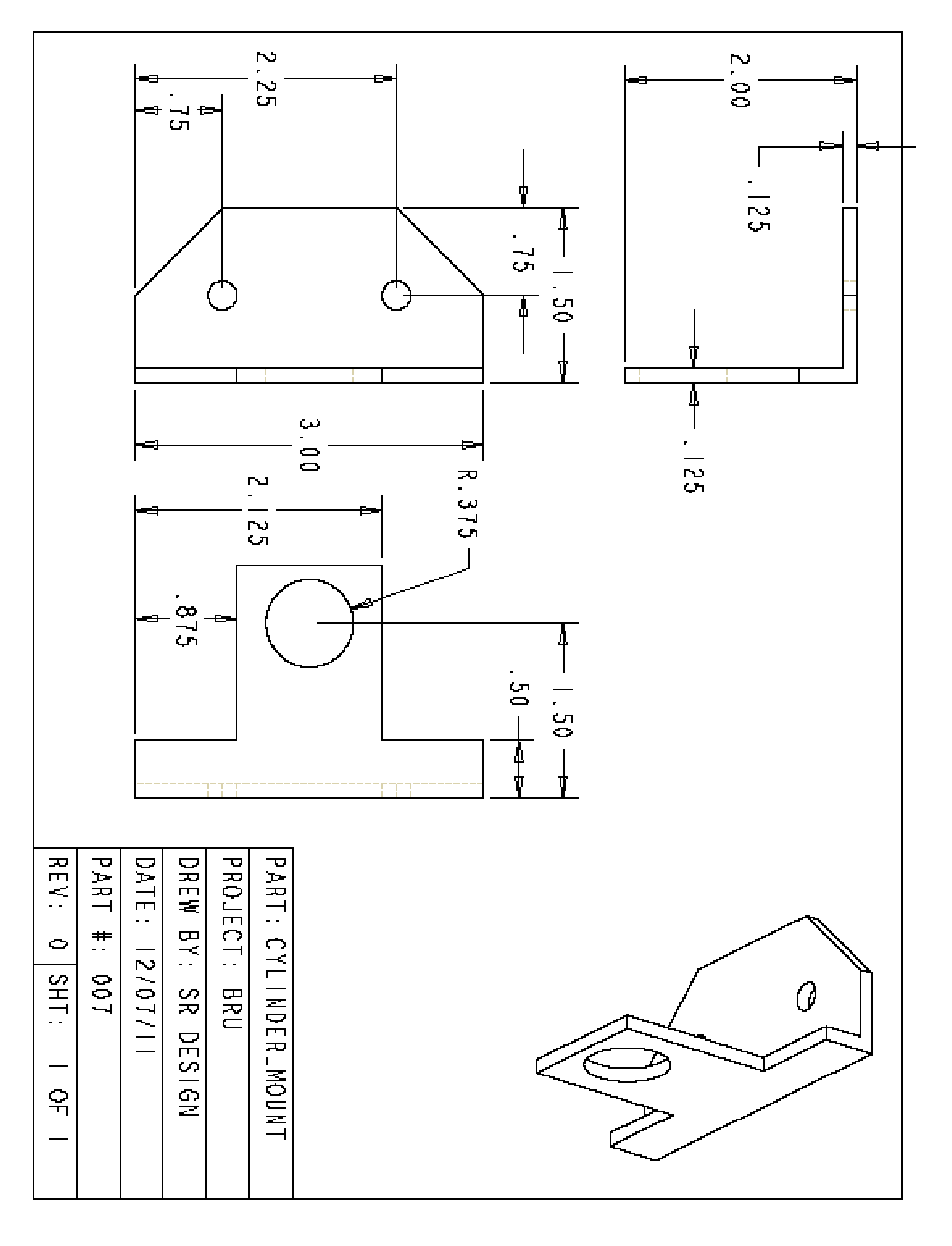
3.125
13.375
WATERJET PART
SCALE 0.625
PART: BRU_CASING_BOT
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 002
REV: 0
SHT: 1 OF 1


SCALE 2.000
.75
.50
.190
.3750
.810
1.00
SCALE 3.000
PART: GUIDE_ROLLER_BLOCK
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 003
REV: 0
SHT: 1 OF 1

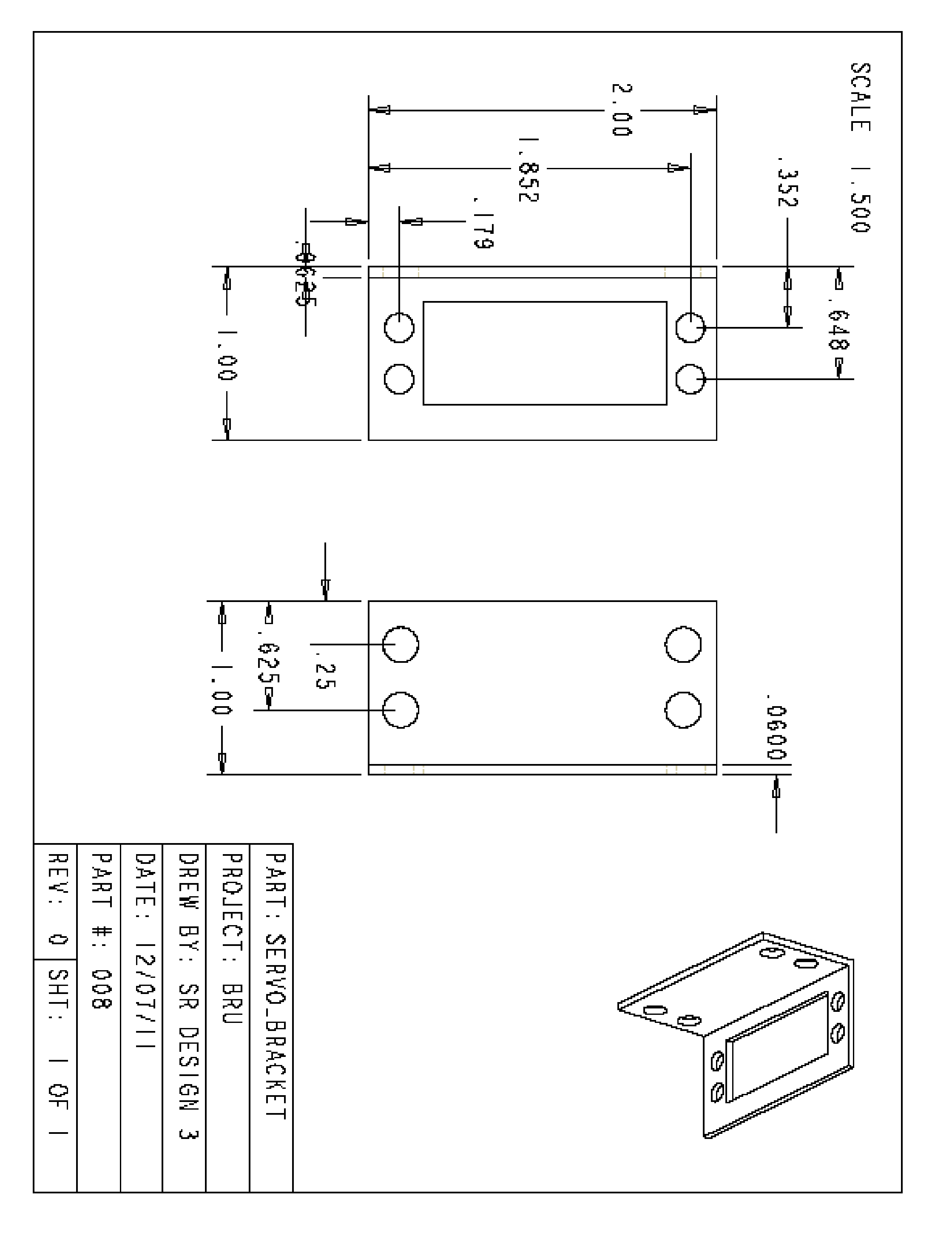

SCALE 2.000
1.25
.675
.50
.8125
.1875
1.00
SCALE 1.500
PART: GUIDE_RAIL
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 004
REV: 0
SHT: 1 OF 1

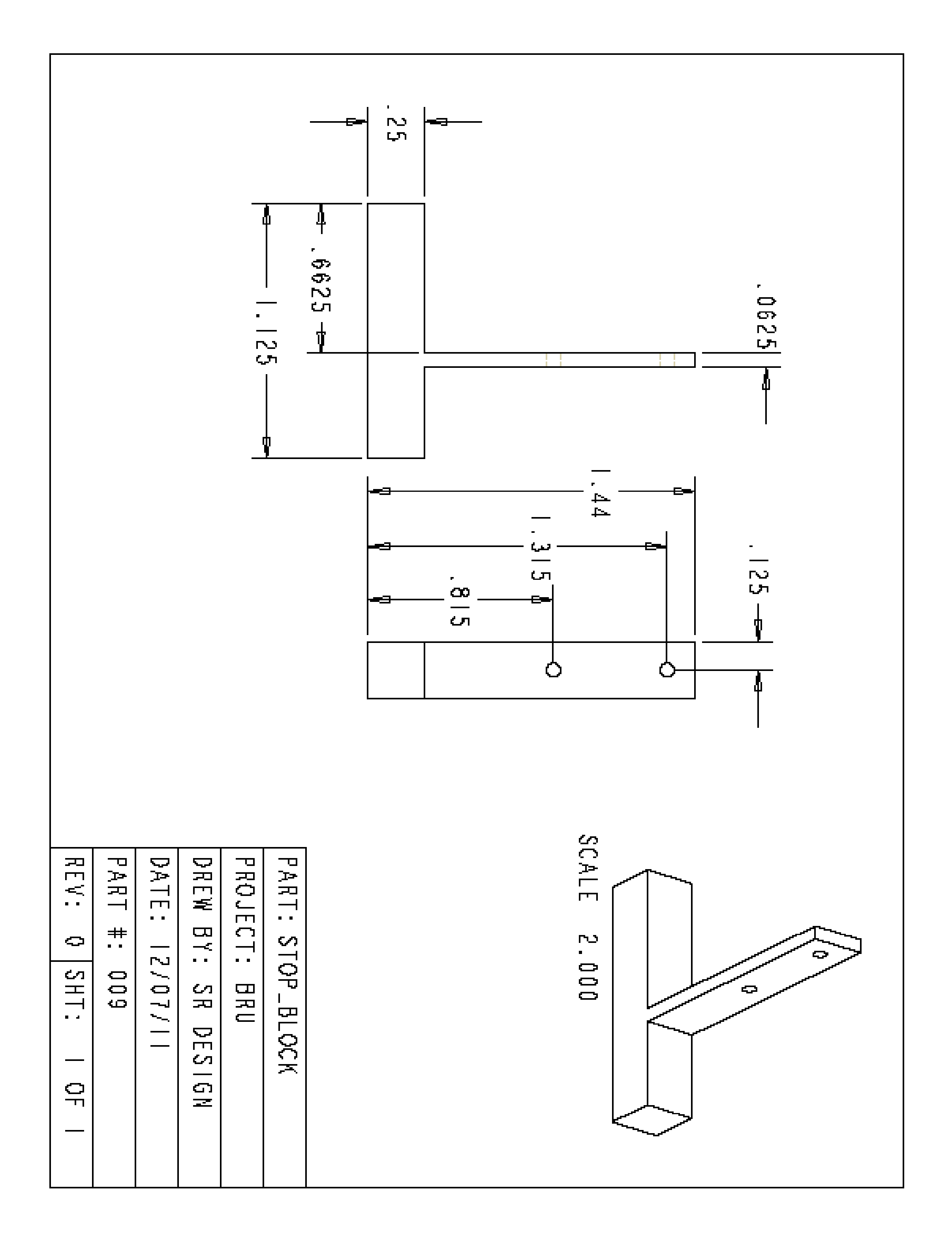

SCALE 2.000
1.50
.50
.50
.0625
2.00
.0625
.50
.50
1.50
SCALE 1.500
PART: BRU_CASING_ANGLE
PROJECT: BRU
DREW BY: SR DESIGN
DATE: 12/07/11
PART #: 005
REV: 0
SHT: 1 OF 1

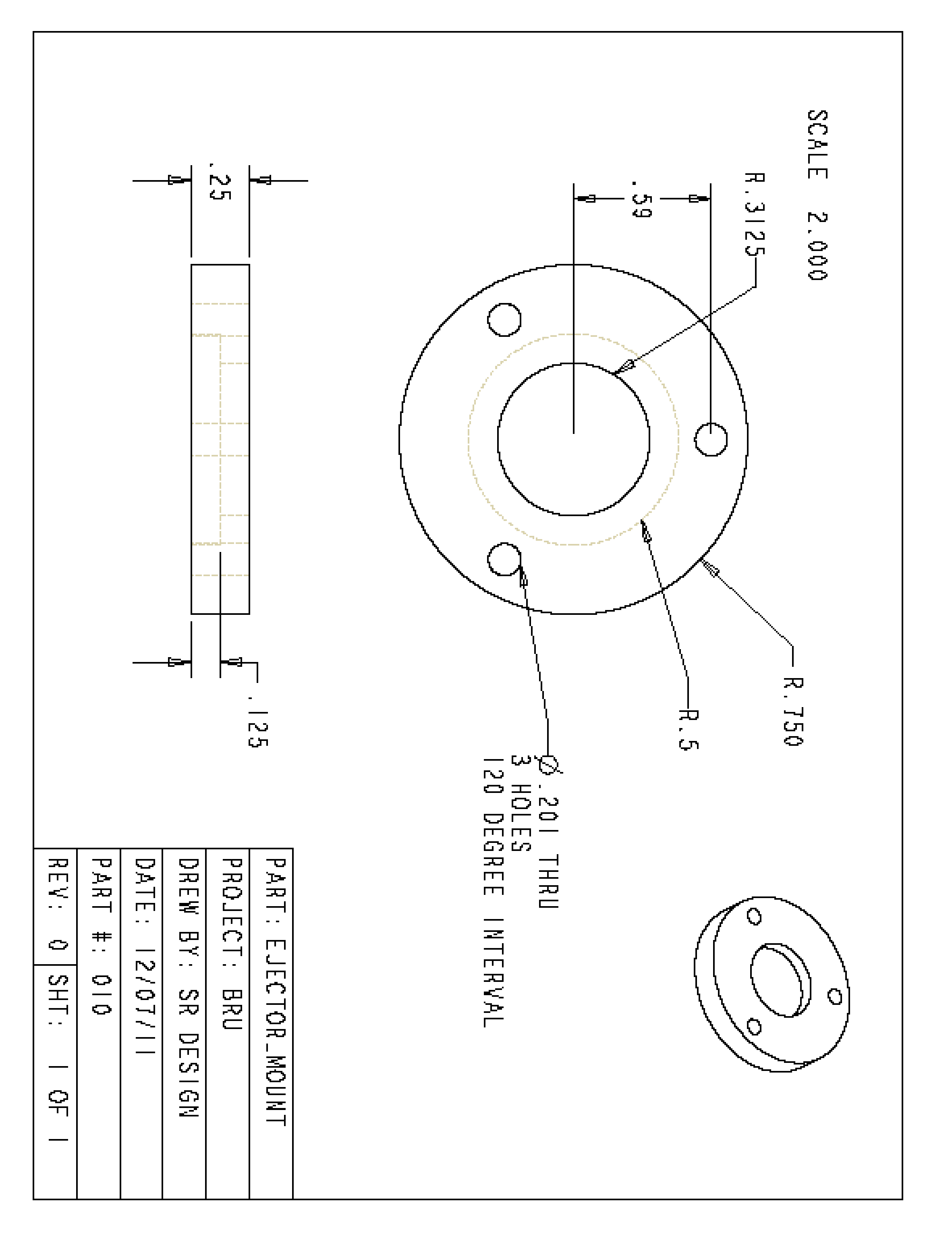

WATERJET PART
SCALE 0.625
11.25
PART: HOOK_BAR
PROJECT: BRU
DREW BY: SR DESIGN
DATE: 12/07/11
PART #: 006
REV: 0
SHT: 1 OF 1

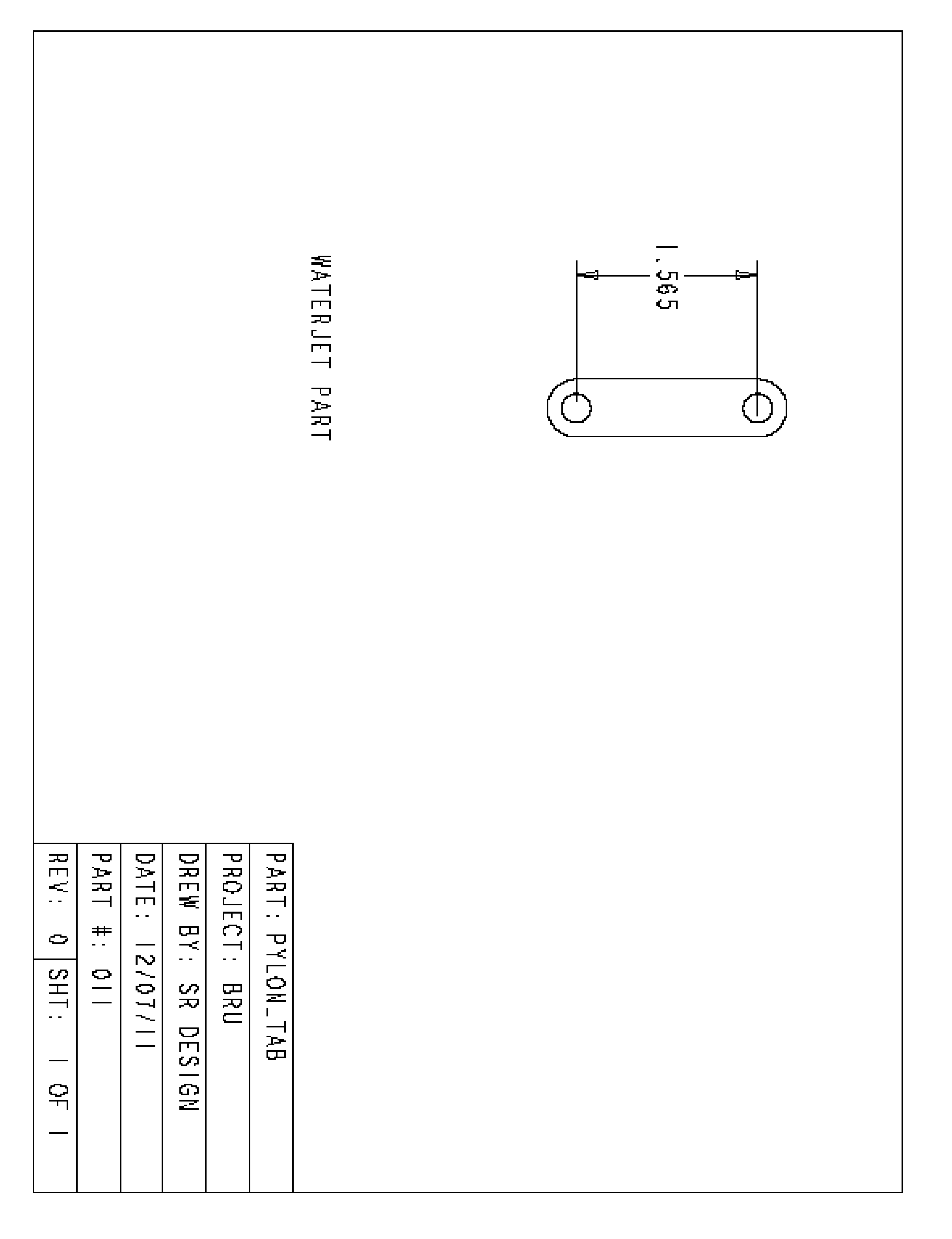


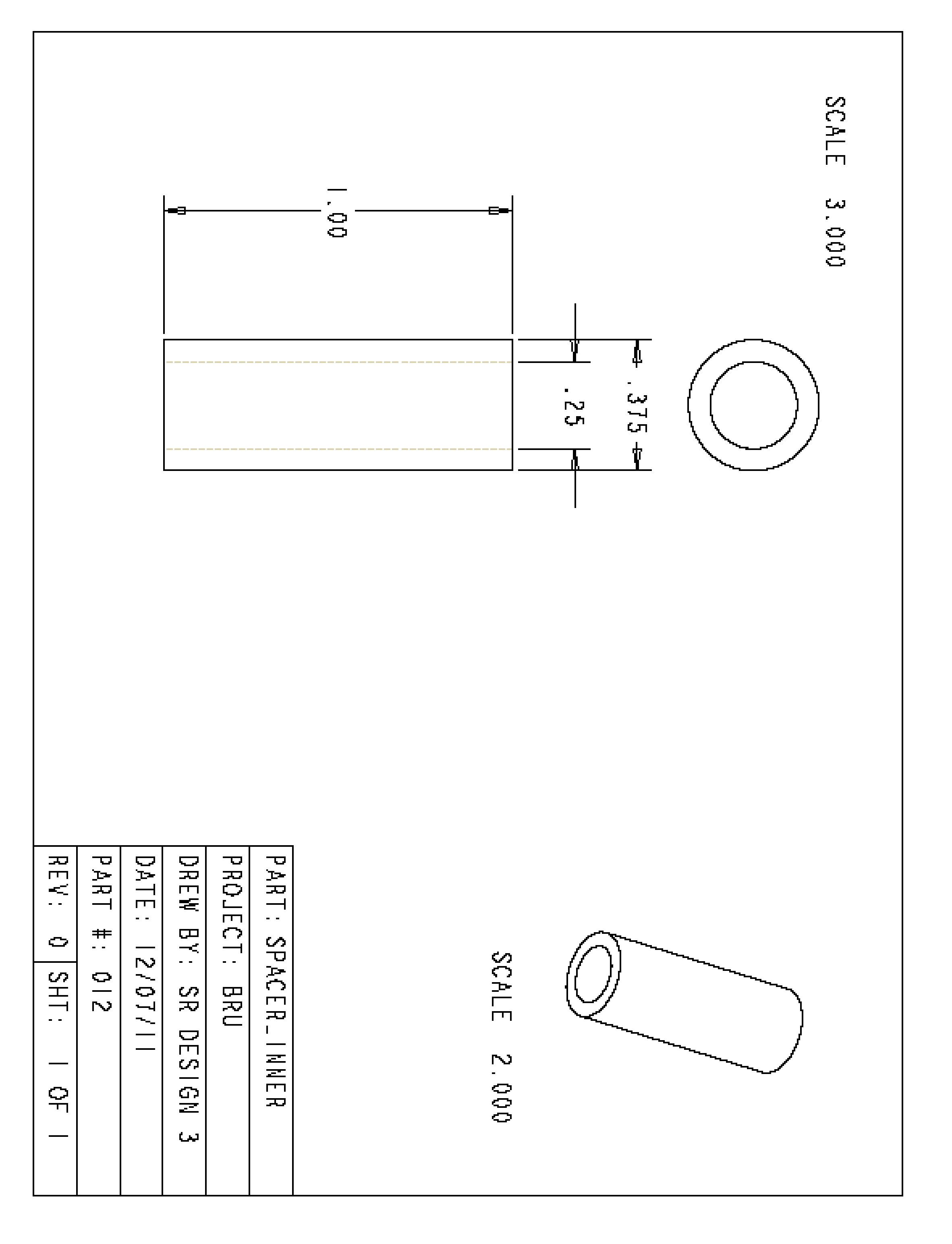
PART: CYLINDER_MOUNT
PROJECT: BRU
DREW BY: SR DESIGN
DATE: 12/07/11
PART #: 007
REV: 0
SHT: 1 OF 1
2.25
.75
1.50
2.00
.125
.125
3.00
R.375
2.125
.875
.50
1.50

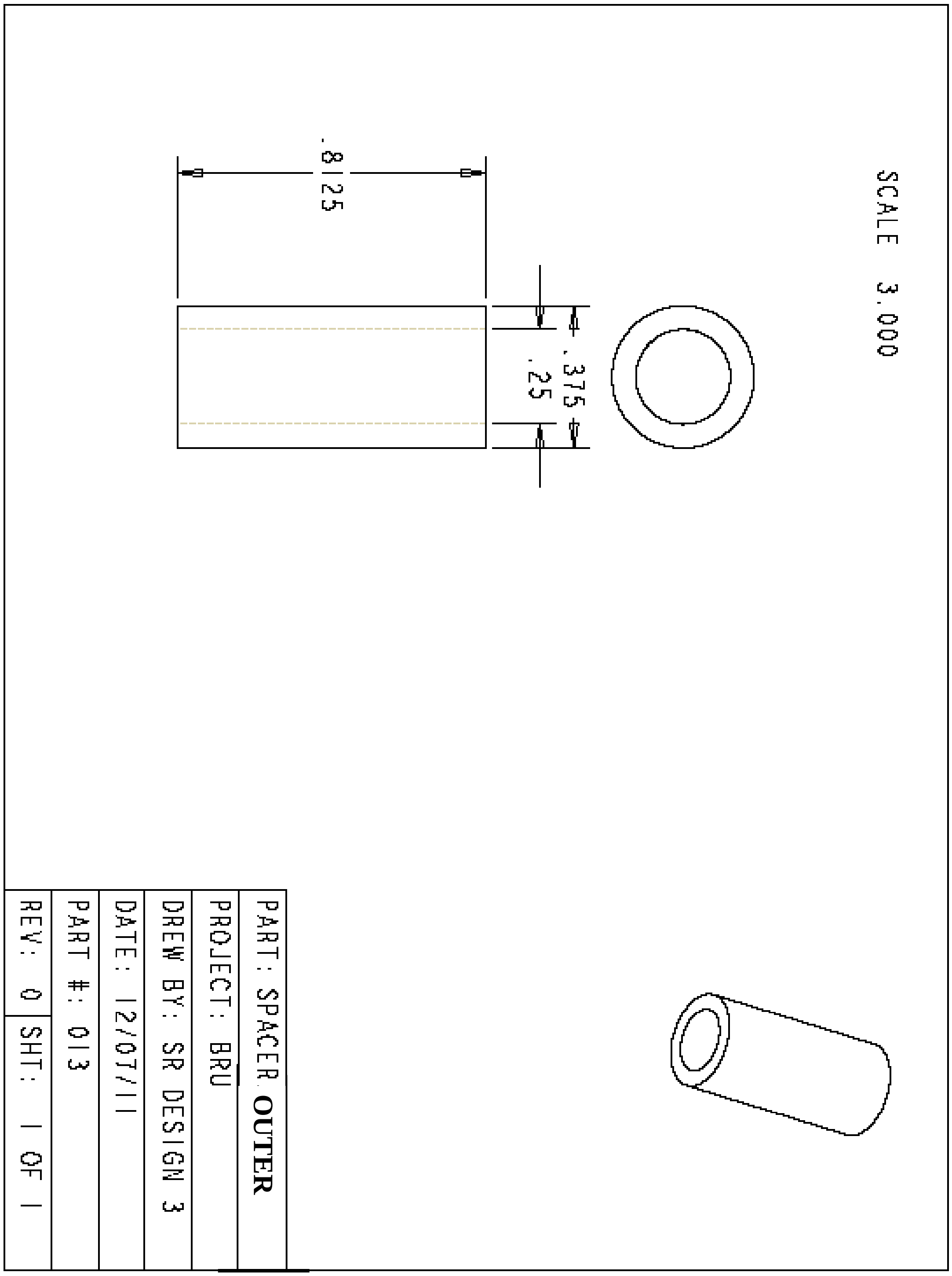

SCALE 1.500
2.00
1.852
.352
.179
.0625
.648
1.00
.625
.25
1.00
.0600
PART: SERVO_BRACKET
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 008
REV: 0
SHT: 1 OF 1

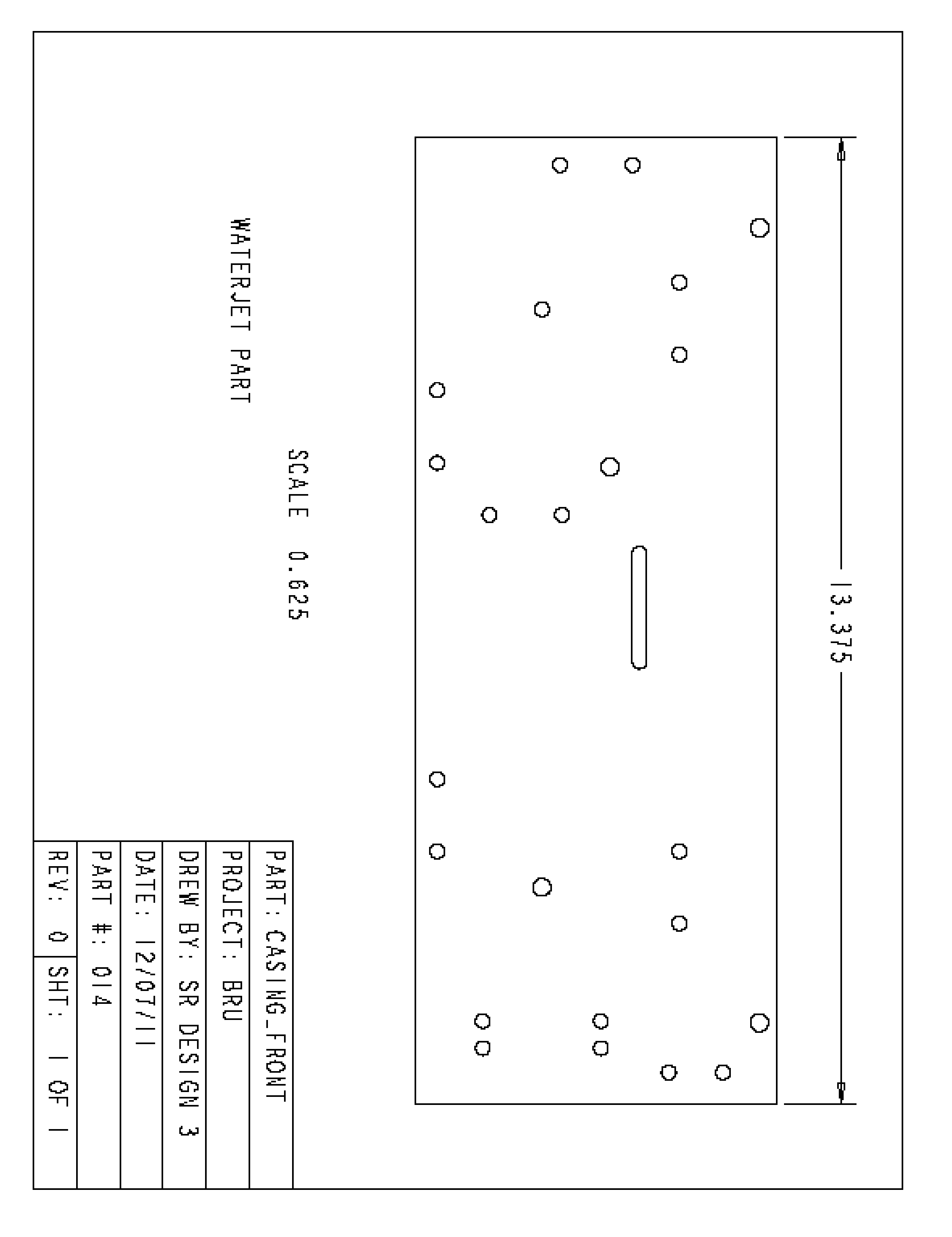

.25
.6625
1.125
.0625
1.44
1.315
.815
.125
SCALE 2.000
PART: STOP_BLOCK
PROJECT: BRU
DREW BY: SR DESIGN
DATE: 12/07/11
PART #: 009
REV: 0
SHT: 1 OF 1

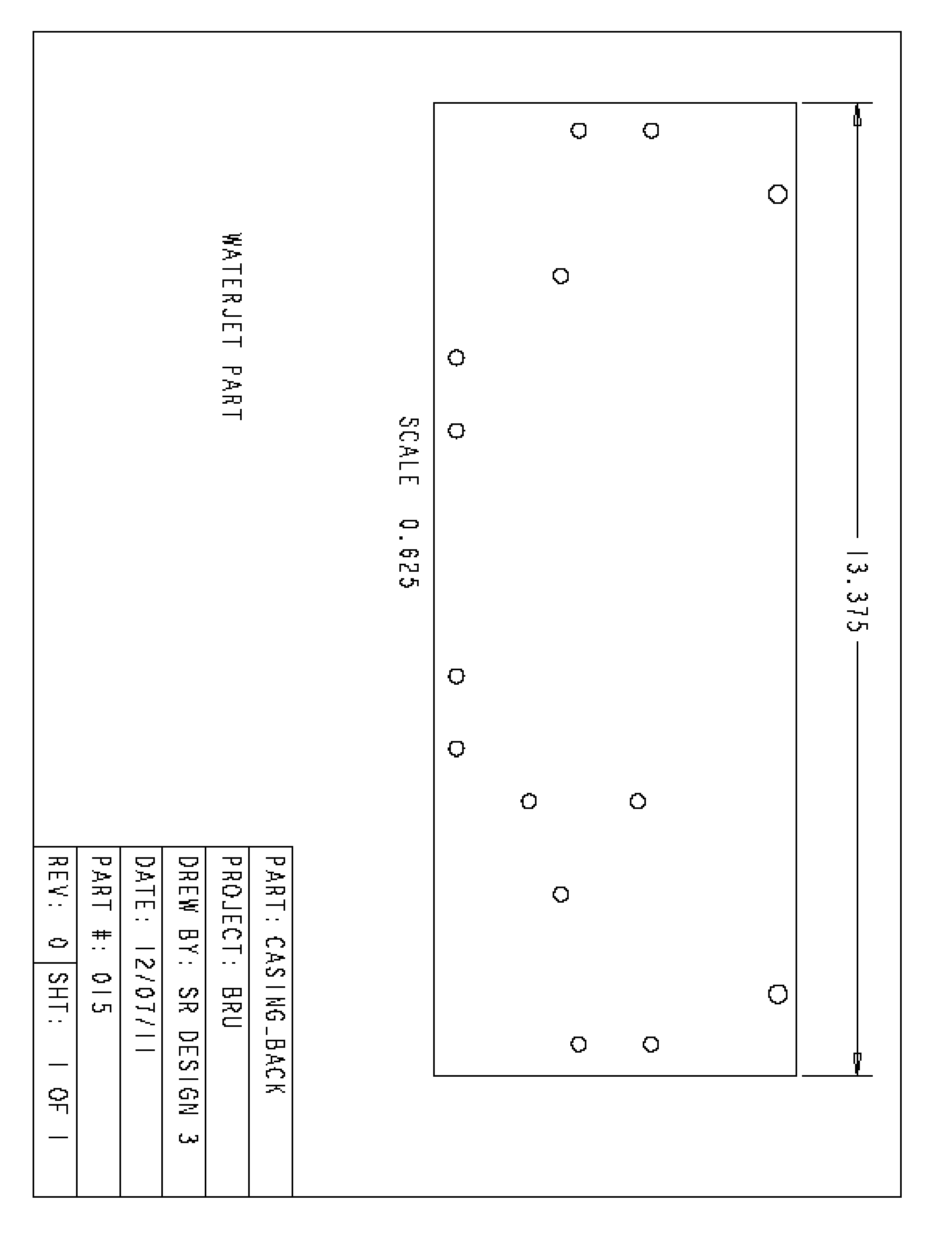

SCALE 2.000
R.3125
.59
.25
R.750
R.5
.125
Ø.201 THRU
3 HOLES
120 DEGREE INTERVAL
PART: EJECTOR_MOUNT
PROJECT: BRU
DREW BY: SR DESIGN
DATE: 12/07/11
PART #: 010
REV: 0
SHT: 1 OF 1

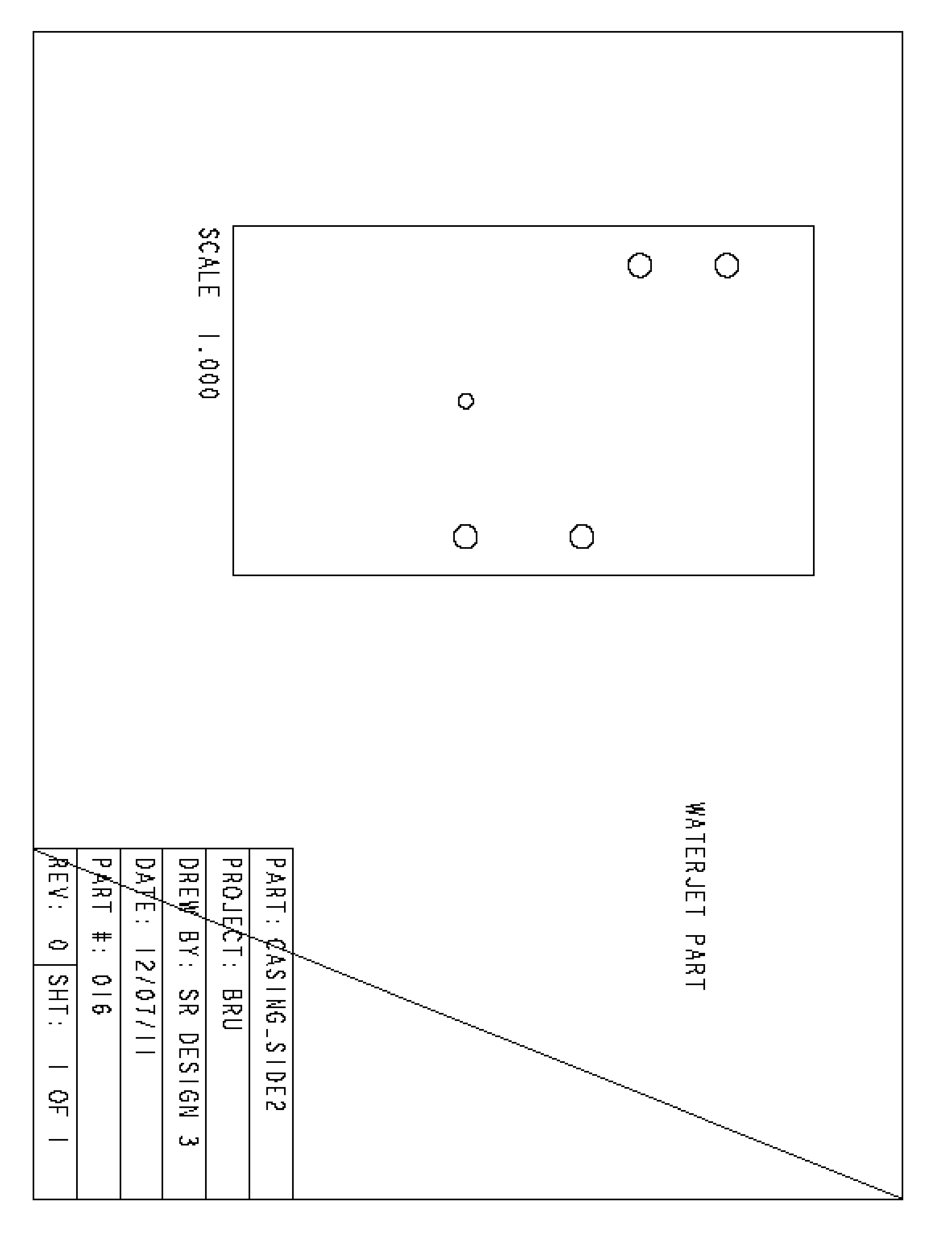

1.565
WATERJET PART
PART: PYLON_TAB
PROJECT: BRU
DREW BY: SR DESIGN
DATE: 12/07/11
PART #: 011
REV: 0
SHT: 1 OF 1


SCALE 3.000
1.00
.375
.25
SCALE 2.000
PART: SPACER_INNER
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 012
REV: 0
SHT: 1 OF 1

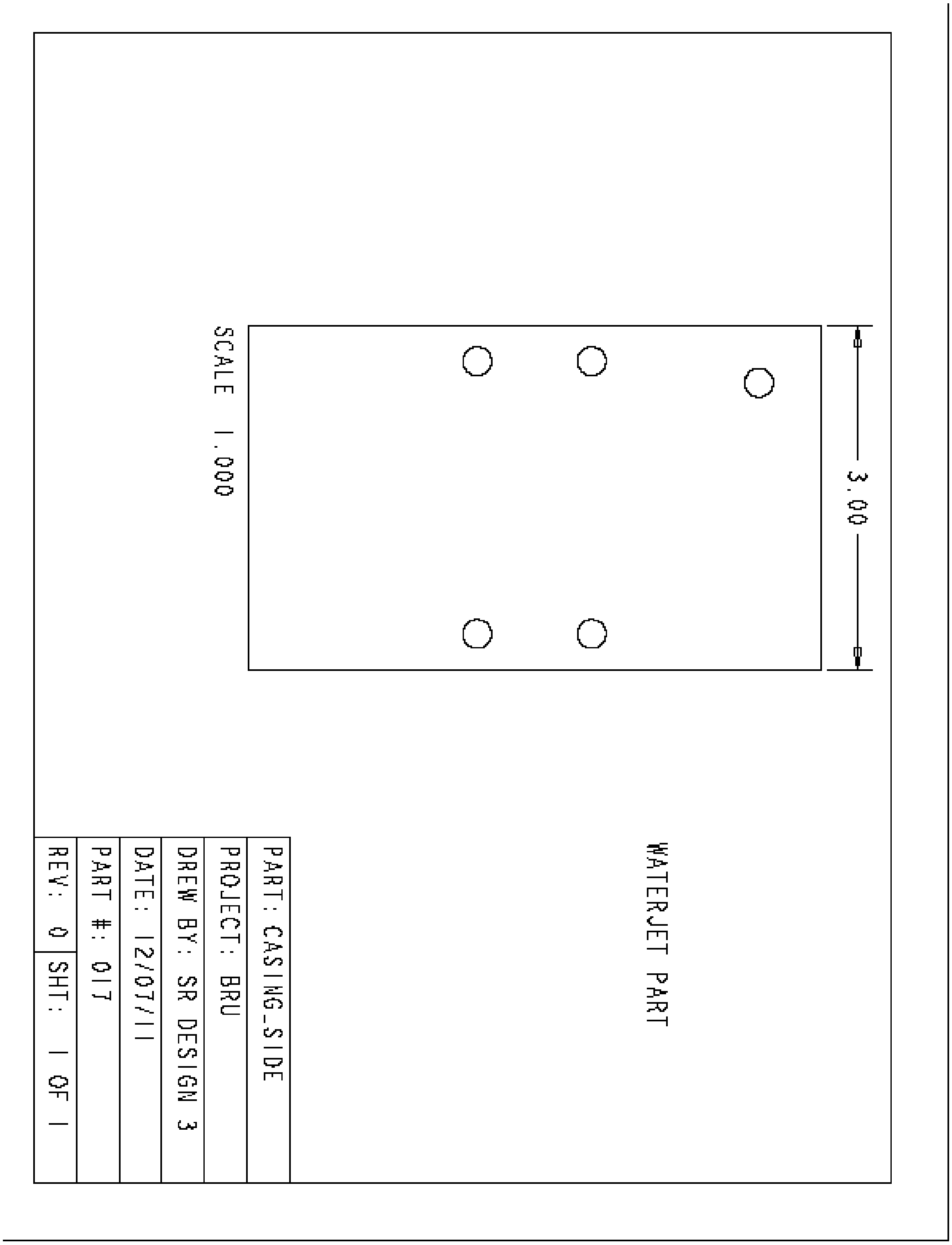

SCALE 3.000
.8125
.375
.25
PART: SPACER OUTER
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 013
REV: 0
SHT: 1 OF 1

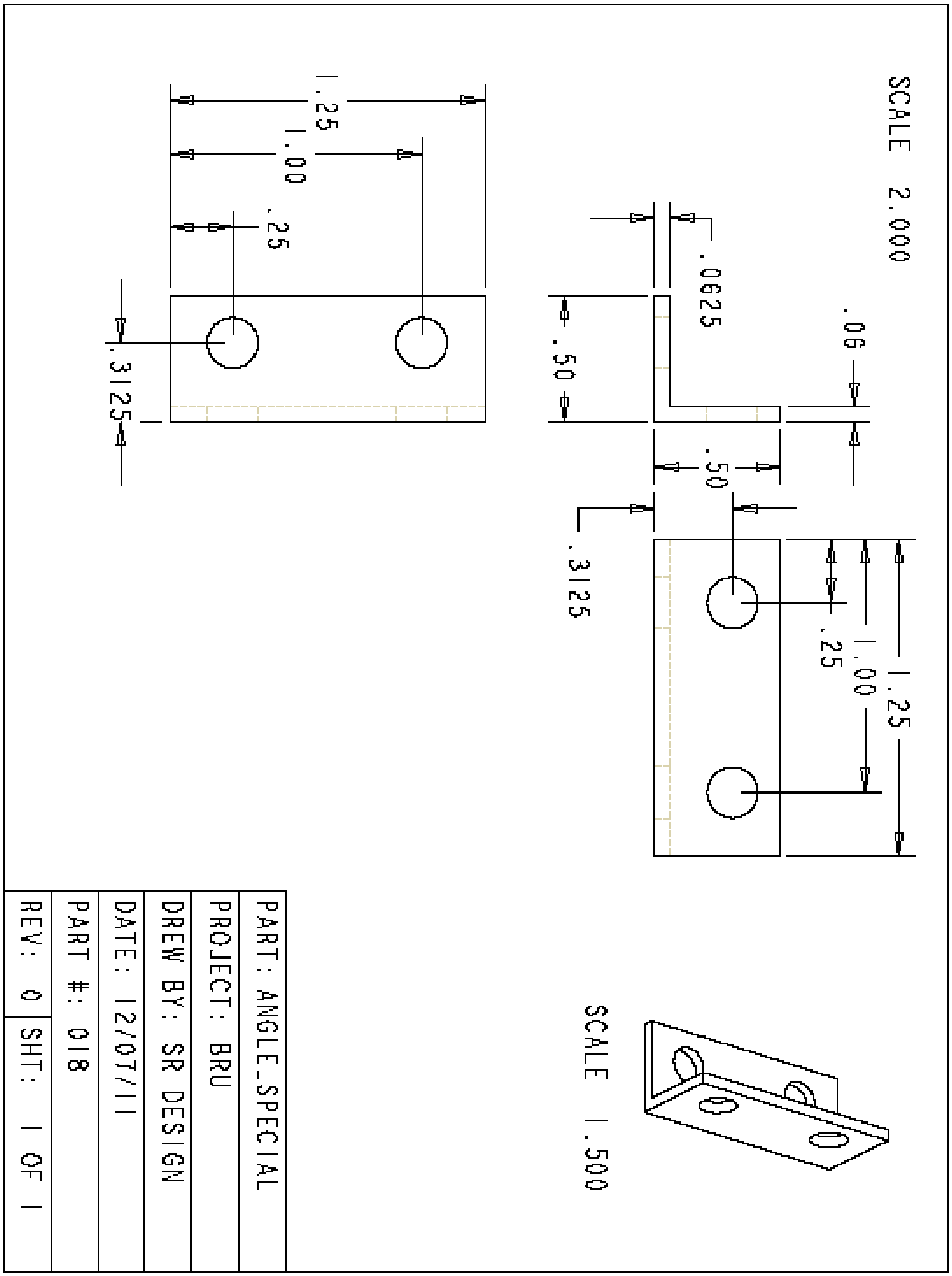

WATERJET PART
SCALE 0.625
13.375
PART: CASING_FRONT
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 014
REV: 0
SHT: 1 OF 1

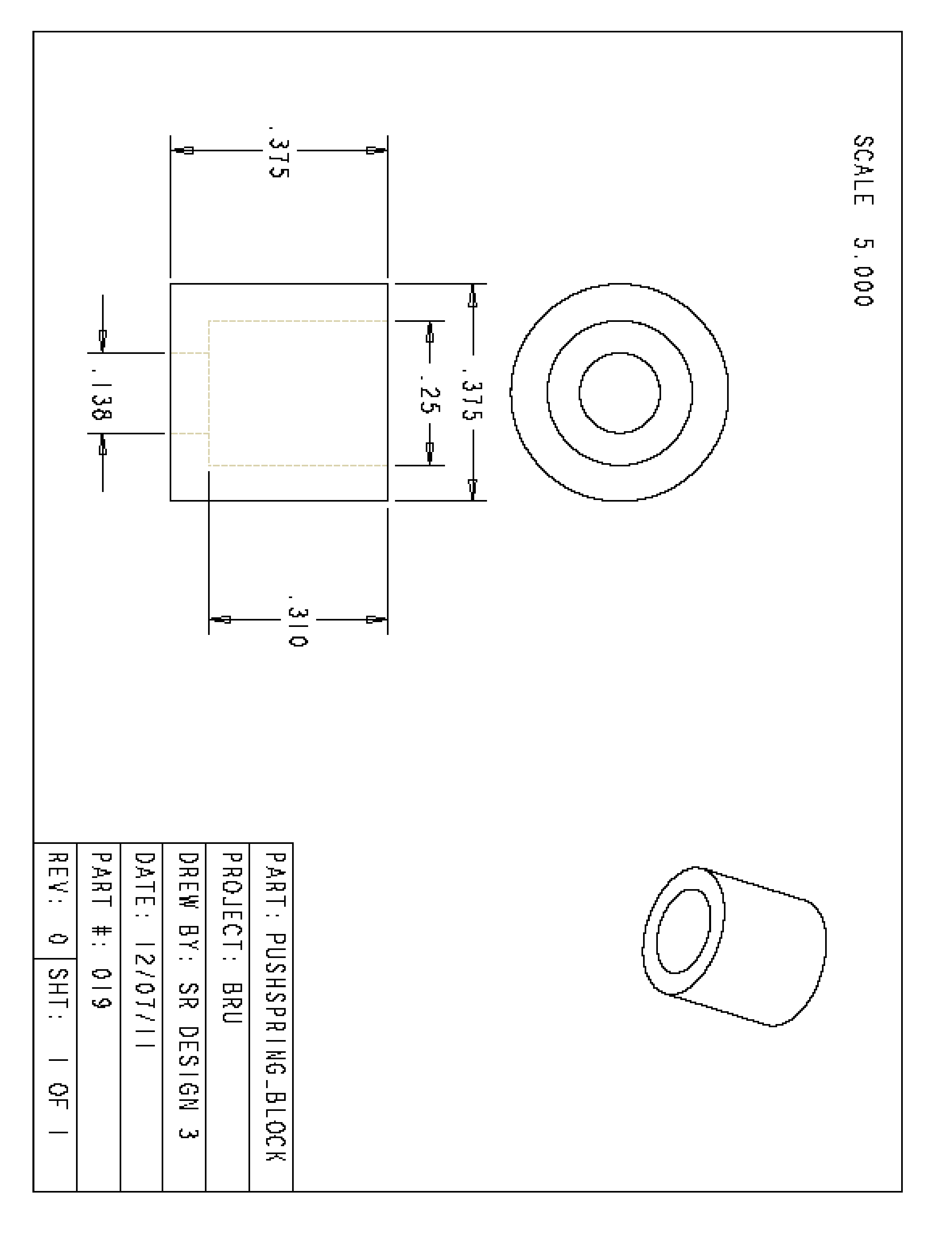

13.375
SCALE 0.625
WATERJET PART
PART: CASING_BACK
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 015
REV: 0
SHT: 1 OF 1



SCALE 1.000
WATERJET PART
PART: CASING_SIDE2
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 016
REV: 0
SHT: 1 OF 1


SCALE 1.000
3.00
WATERJET PART
PART: CASING_SIDE
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 017
REV: 0
SHT: 1 OF 1


SCALE 2.000
.06
.0625
.50
.50
.3125
.25
1.00
1.25
1.25
1.00
.25
.3125
SCALE 1.500
PART: ANGLE_SPECIAL
PROJECT: BRU
DREW BY: SR DESIGN
DATE: 12/07/11
PART #: 018
REV: 0
SHT: 1 OF 1


SCALE 5.000
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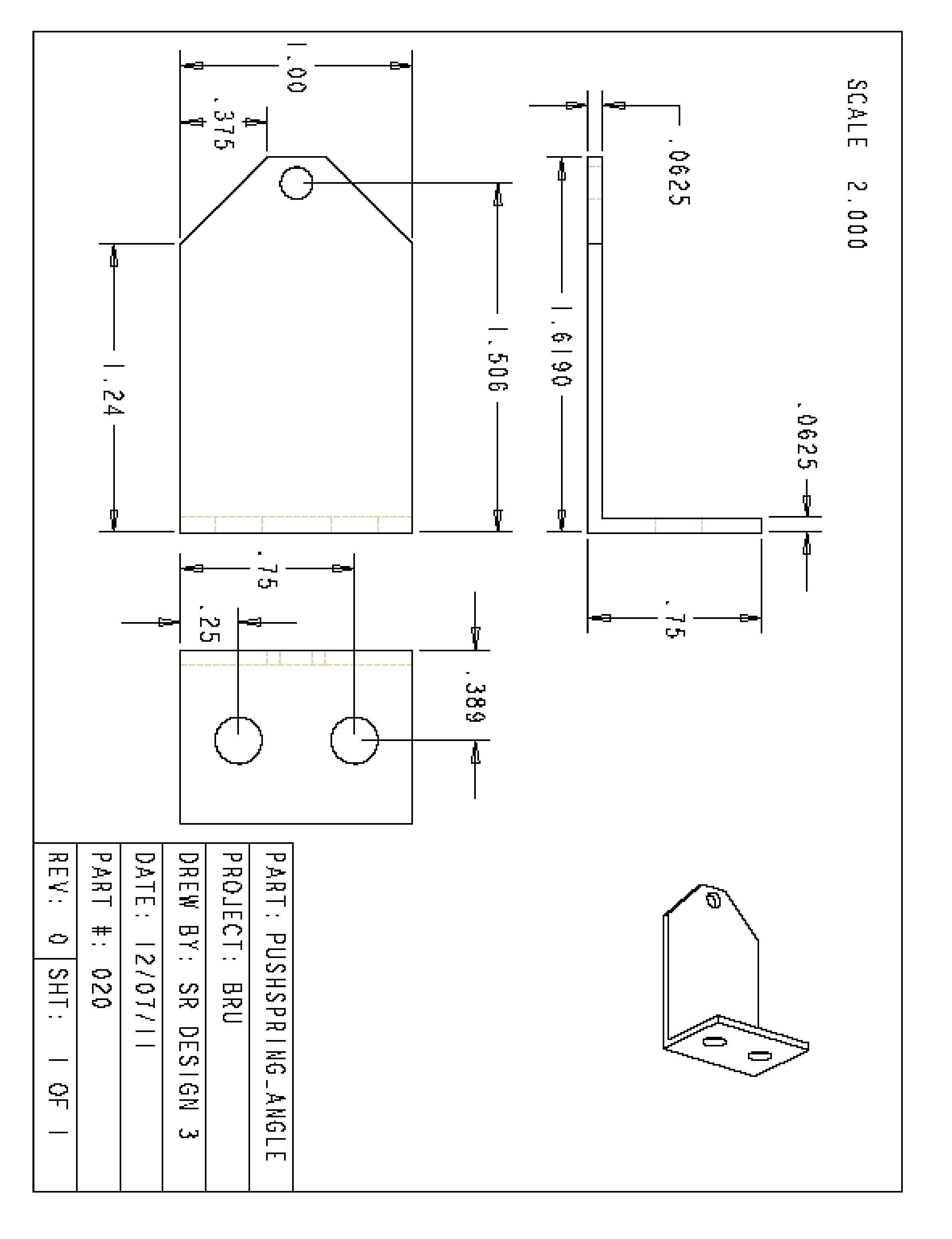
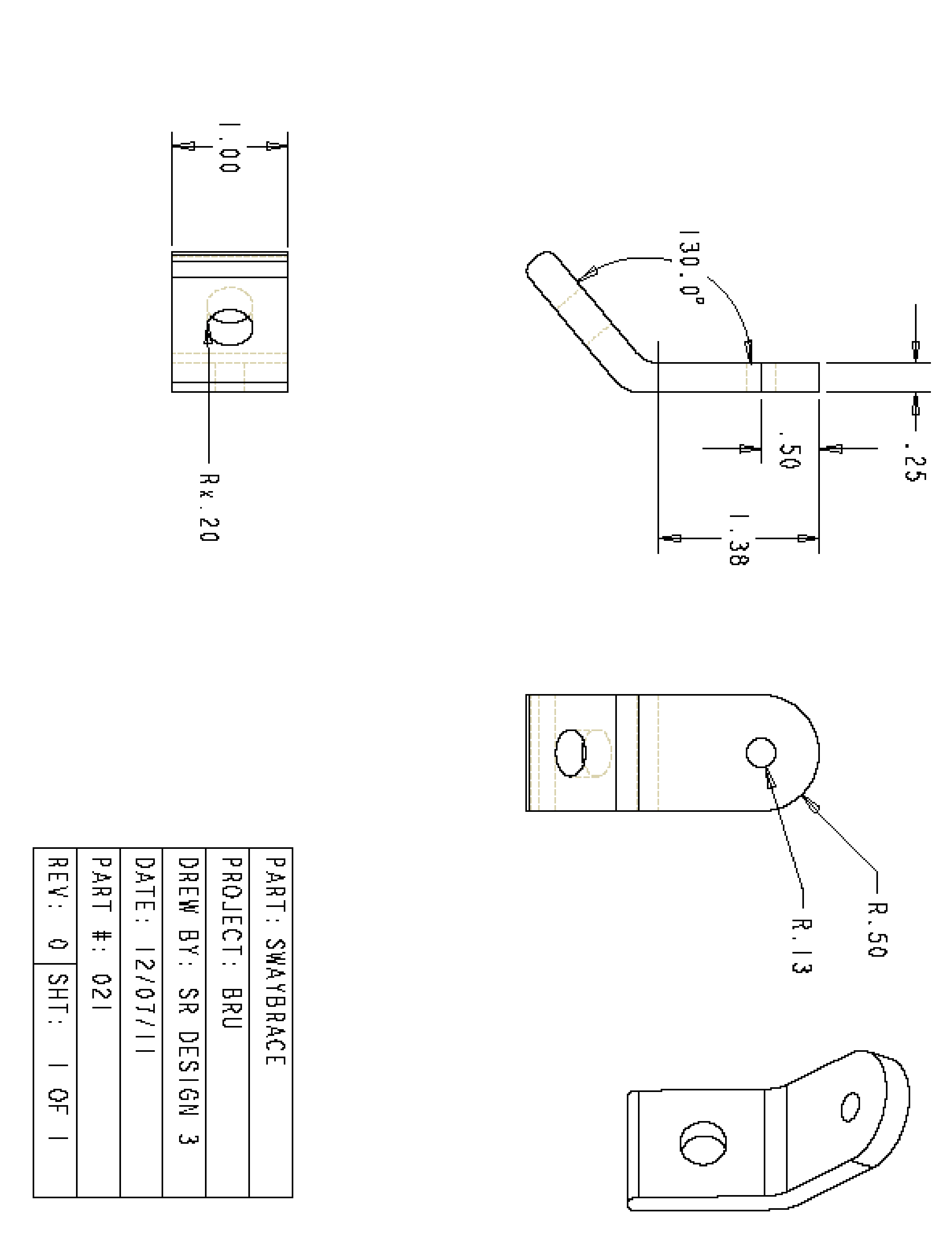
.375
.25
.138
.310
PART: PUSHSPRING_BLOCK
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 019
REV: 0
SHT: 1 OF 1


SCALE 2.000
1.00
.375
.0625
1.506
1.6190
1.24
.0625
.75
.75
.25
.389
PART: PUSHSPRING_ANGLE
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 020
REV: 0
SHT: 1 OF 1


1.00
Rx.20
130.0°
.25
.50
1.38
R.50
R.13
PART: SWAYBRACE
PROJECT: BRU
DREW BY: SR DESIGN 3
DATE: 12/07/11
PART #: 021
REV: 0
SHT: 1 OF 1

Bill of Materials

Purchase Items

Part	Vender	Part Number	Price	Quantity	Total Price
Air Cylinder	McMaster	6498K211	$33.42	1	$33.42
Air Tank	Clippard	AVT-24-4	$16.82	1	$16.82
Check Valve	McMaster	3208K22	$14.74	1	$14.74
Regulator	McMaster	99045K48	$34.80	1	$34.80
Solenoid	cylval	SA31NC	$43.80	1	$43.80
Guide_roller	Grainger	1ZGT7	$48.25	4	$193.00
Ejector_bushing	McMaster	6377K114	$20.27	2	$40.54
Ejector_spring	Grainger	1NCT2	$7.69	1	$7.69
MicroDragon	EVBplus		$55	1	$55.00
Motor Driver	Pololu	1455	$59.95	1	$59.95
UTB cord	EVBplus		$14.00	1	$14.00
Pushspring	McMaster	9657K48	$6.29	1	$6.29
Servo	Futaba	FUTM0513		1	$0.00
Limit Switch	McMaster	7090K37	$7.91	1	$7.91
RBF PIN	McMaster	90293A139	$17.98	1	$17.98
Raw Material	Various				$125.97
Total					**$671.91**

Material Weight

Assembly	Part	Material	Quanitity	Machine Required	Weight (lb) each	Weight (lb) total
Hook Eject	Guide_top_angle	Al	2	Mill	0.0619	0.1238
	BRU_casing_bot	Al	1	Waterjet	0.1884	0.1884
	Guide_roller_block	Plastic	4	Mill	0.0095	0.038
	Guide_roller	Varied	4	Purchase	0.01	0.04
	Guide_rail	Plastic	3	Mill	0.0188	0.0564
	BRU_casing_angle	Al	4	Mill	0.0106	0.0424
	Hook_bar	Al	1	Waterjet	0.2718	0.2718
	Cylinder_mount	Al	1	Mill	0.0783	0.0783
	Air Cylinder	Steel	1	Purchase	0.18	0.18
Mech Safety	Servo	Plastic	1	Purchase	0.0586	0.0586
	Servo_bracket	Al	1	Mill	0.0171	0.0171
	Stop_block	Al	1	Waterjet&Mill	0.0087	0.0087
Ejector	Ejector_mount	Plastic	1	Mill	0.0134	0.0134
	Ejector_bushing	Plastic	2	Purchase	0.0025	0.005
	Ejector_piston.pt1	Steel	1	Band Saw	0.1172	0.1172
	Ejector_piston.pt2	Steel	1	Mill&Weld		
	Ejector_spring	Steel	1	Purchase	0.005	0.005
Pylon Attach	Pylon_tab	Al	4	Waterjet	0.0108	0.0432
	Spacer_outer	Plastic	4	Lathe	0.0022	0.0088
	Spacer_inner	Plastic	2	Lathe	0.0027	0.0054
BRU Casing	Casing_back	Al	1	Waterjet	0.3563	0.3563
	Casing_front	Al	1	Waterjet	0.352	0.352
	Casing_side	Al	1	Waterjet	0.0703	0.0703
	Casing_side2	Al	1	Waterjet	0.142	0.142
	Casing_angle	Al	3	Mill	0.0082	0.0246
	Casing_angle_special	Al	1	Mill	0.0063	0.0063
Push Spring	Pushspring_block	Plastic	2	Lathe	0.0011	0.0022
	Pushspring_angle	Al	1	Mill	0.0084	0.0084
	Pushspring	Steel	1	Purchase	0.005	0.005
Pneumatic System	Solenoid Valve	Varied	1	Purchase	0.05	0.05
	Air Tank	Steel	1	Purchase	0.25	0.25
	Check Valve	Varied	1	Purchase	0.01	0.01
	Pressure Regulator	Varied	1	Purchase	0.01	0.01
	Tubing&Connectors	Varied	1	Purchase	0.1	0.1
Sway Brace	Swaybrace	Al	4	Mill	0.07	0.28
	Stem	Steel	4	Purchase	0.1	0.4
	Pad	Plastic	4	Purchase	0.05	0.2
Other	Limit Switch	Plastic	1	Purchase	0.03	0.03
	Fasteners	Steel	1	Purchase	1.25	1.25
Total						**4.8486**

Raw Materials

	Material	Size	Vendor	Price
Angle	Al6061	2.5x2.5x.125-12in	onlinemetals	$3.28
	Al6063	2x2x.0625<12in	onlinemetals	$3.98
		.75x.75x.0625-24in	onlinemetals	$2.81
		1x1x.0625<12in	onlinemetals	$1.64
Flat Bar	Al6061	.25x5-24in	onlinemetals	$15.17
		.125x4-12in	onlinemetals	$3.16
Plate	Al6061	12x14x.0625 in	onlinemetals	$23.94
	Acetal	12x12x.5 in	onlinemetals	$40.78
Round	Acetal	1.5x12in	onlinemetals	$21.97
		.375x24in	onlinemetals	$3.48
	Steel 1018	.5x<12in	onlinemetals	$2.11
		.75x<12in	onlinemetals	$3.65
Total				**$125.97**

Calculations

Pneumatic Calculations

Volume in Air Cylinder

$Bore := 1.5in$

$stroke := 1in$

$$Volume_{cyl} := \pi \cdot \left(\frac{Bore}{2}\right)^2 \cdot stroke = 1.767 \cdot in^3$$

Air Tank Safety Factor: $SF := 2$

$Volume_{tank} := SF \cdot Volume_{cyl} = 3.534 \cdot in^3$ 4 cubic inch air tank needed

Servo motor Torque

$\tau := 76oz \cdot in = 4.75 \, lb \cdot in$

$m := 0.008667 \, lb$ $cg := 0.27216 \, in$

$L := 1.41in$ $Midpoint := 0.375in$

$L_{torque} := L - cg - Midpoint$ $L_{torque} = 0.763 \, in$

$\tau_{required} := m \cdot L_{torque}$ $\tau_{required} = 6.612 \times 10^{-3} \cdot lb \cdot in$

Unit must withstand 2G lateral and 1G landing shock.
This is potentially a 3G peak shock in the downward direction adds to 1G from gravity.
Assuming the max weight for the BRU: 5lbs

$m_{bru} := \frac{5lbf}{g} = 2.268kg$

$m_{payload} := \frac{10lbf}{g} = 4.536kg$

$m_{total} := m_{bru} + m_{payload} = 6.804kg$

Total downward force mounting tabs must withstand:

$F_{total} := m_{total} \cdot 4 \cdot g = 266.893N$

quarter inch mounting tab bolts specified by the pylon structure

$d := 0.25in$

$A := \frac{\pi}{4} \cdot d^2 = 0.049 in^2$

bolt will be in double shear

$V := \frac{F_{total}}{2} = 133.447N$

Average shear stress within the bolt

$\tau := \frac{V}{A} = 4.214 MPa$

Grade 1 1/4" steel bolt minimum tensile strength:

$U_{min} := 60000psi$

$\tau_{allow} := .6 \cdot U_{min} = 36 ksi$

$\boxed{\tau = 4.214 MPa}$

$\boxed{\tau_{allow} = 248.211MPa}$

Since the allowable shear stress in a grade 1 bolt is 248 MPa, the shear stress of 4.2 MPa will be easily supported by the lowest grade fasteners.

Tab tear out considerations

$A := 0.25in \cdot 0.25in = 0.063 in^2$

$\tau := \frac{V}{A} = 3.309 MPa$

AL-6061 specifications

$\tau_{allow} := 78.5MPa$

$A_{req} := \frac{V}{\tau_{allow}} = 2.635 \times 10^{-3} \cdot in^2$

$t_{req} := \frac{A_{req}}{0.25in} = 0.011 \cdot in$

The plate thickness required to resist the tear-out shear forces is 0.011in.

The smallest available plate thickness could be used.

Combined loading considerations. The tab could potentially be subjected to a 2G lateral load as well as a 2G downward load. A wind force at 75mph will also be considered acting on the front of the BRU.

Parameters to be set by design:

Tab width:

$w := 0.75\,in$

Tab thickness:

$$t := \frac{3}{32}\,in$$

Normal stress from weight and landing load

$F := m_{total} \cdot 2 \cdot g = 133.447\,N$

$A := w \cdot t$

$$\sigma_n := \frac{F}{A} = 2.942\,MPa$$

Shear stress

$V := F = 133.447\,N$

$$Q := \frac{t}{4} \cdot w \cdot \frac{t}{2}$$

$$I := \frac{1}{3} \cdot w \cdot t^3$$

$$\tau := \frac{V \cdot Q}{I \cdot t} = 8.825\,MPa$$

Bending stress from lateral force

$M := m_{total} \cdot 2 \cdot g \cdot 5.19\,in$

$$c := \frac{t}{2}$$

$$\sigma_b := \frac{M \cdot c}{I} = 244.284\,MPa$$

Bending stress from wind force

BRU front dimensions set by design:

$w := 4in$

$h := 8in$

$$r := \frac{h}{2} + 1.19\,in = 5.19\,in$$

$a := w \cdot h = 0.222\,ft^2$

$$a_2 := \frac{\pi}{4} \cdot (4in)^2 = 8.107 \times 10^{-3}\,m^2$$

$Q := 0.00256(75)^2 = 14.4$

$p := 14.4\,psf$

$F_{wind} := a \cdot p \cdot 2 = 28.469\,N$

$F_{windpay} := a_2 \cdot p \cdot 2 = 2.513\,lbf$

$M := F_{wind} \cdot r$

$$c := \frac{t}{2}$$

$$I := \frac{1}{3} \cdot t \cdot w^3 = 2 \cdot in^4$$

$$\sigma_w := \frac{M \cdot c}{I} = 5.368\,kPa$$

Using superposition the total stress is found by summing the individual stresses.

$$\sigma_{combined} := \sigma_n + \tau + \sigma_b + \sigma_w = 256.056\,MPa$$

The maximum combined loading for 1 tab is 427 MPa. Four tabs will be used so this value is divided by 4.

$$\sigma_{perTab} := \frac{\sigma_{combined}}{4} = 64.014\,MPa$$

AL-6061 specifications

$$\sigma_{ult} := 117\,MPa$$

$$\sigma_{perTab} = 64.014\,MPa$$

$$\sigma_{eighthInch} := 60.242\,MPa$$

$$117\,MPa = 16.969\,ksi$$

$$FOS := \frac{\sigma_{ult}}{\sigma_{eighthInch}} = 1.942$$

$$FOS := \frac{\sigma_{ult}}{\sigma_{perTab}} = 1.828$$

By optimization it is found that a 3/4" X 3/32" aluminum tab is the smallest standard plate thickness that will keep the maximum combined stress within the ultimate stress limits of aluminum.

Also by optimization it is found that a 3/4" X 1/8" aluminum tab is the smallest standard plate thickness that will allow for a factor of safety of at least 1.5.

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Mass of hook: m1

$m1 := .27lb$

Mass of piston: m2

$m2 := .104lb$

Mass of payload: mpayload

$m_{payload} := 10lb$

Force from pneumatic cylinder: Fair

$F_{air} := 120lbf = 533.787N$

$K1 := 50\frac{N}{m}$

Hook spring rate: K1
Hook travel distance: X1
Spring preload distance: X2

$X1 := 1in$

$X2 := 0$

$FS1 := K1 \cdot (X2 + X1) = 0.286\,lbf$

Piston spring rate: K2
Spring preload distance: X3

$K2 := 450\frac{N}{m}$

$m2 \cdot g = 0.463N$

$X3 := 0$

$FS2 := K2 \cdot (X1 + X3) = 11.43\,N$

$$A := \begin{pmatrix} m1 & \frac{1}{\sqrt{2}}kg \\ m2 & \frac{-1}{\sqrt{2}}kg \end{pmatrix}$$

$$Y := \begin{pmatrix} F_{air} - FS1 \\ m2 \cdot g - FS2 \end{pmatrix}$$

$$Y = \begin{pmatrix} 532.517 \\ -10.967 \end{pmatrix} N$$

$$X := A^{-1} \cdot Y = \begin{pmatrix} 1.009 \times 10^{4} \\ 723.797 \end{pmatrix} \cdot \frac{ft}{s^2}$$

$$\boxed{a := X_0 = 1.009 \times 10^{4} \cdot \frac{ft}{s^2}}$$

$$\boxed{FR := X_1 \cdot kg = 49.596lbf}$$

Time the piston takes to reach the payload

Distance between the piston and payload: X4

$X4 := .14in$

$$t := \sqrt{\frac{2 \cdot X4}{a}} = 1.521 \times 10^{-3} s$$

Velocity of the piston when it contacts the payload:

$V_{piston} := a \cdot t = 15.341 \cdot \frac{ft}{s}$

Principle of linear impulse and momentum to calculate initial velocity of payload:

$$V_{payload} := \frac{F_{air} \cdot t + (m1 + m2) \cdot V_{piston}}{m_{payload}} = 1.161 \cdot \frac{ft}{s}$$

$X5 := 1in - X4 = 0.86 in$

$X5 = 0.022m$

$$a2 := \frac{F_{air}}{m1 + m2 + m_{payload}} = 372.169 \frac{ft}{s^2}$$

$$t2 := \sqrt{\frac{2 \cdot X5}{a2}} = 0.02 s$$

$$V_{payload2} := V_{payload} + a2 \cdot t2 = 8.465 \frac{ft}{s}$$