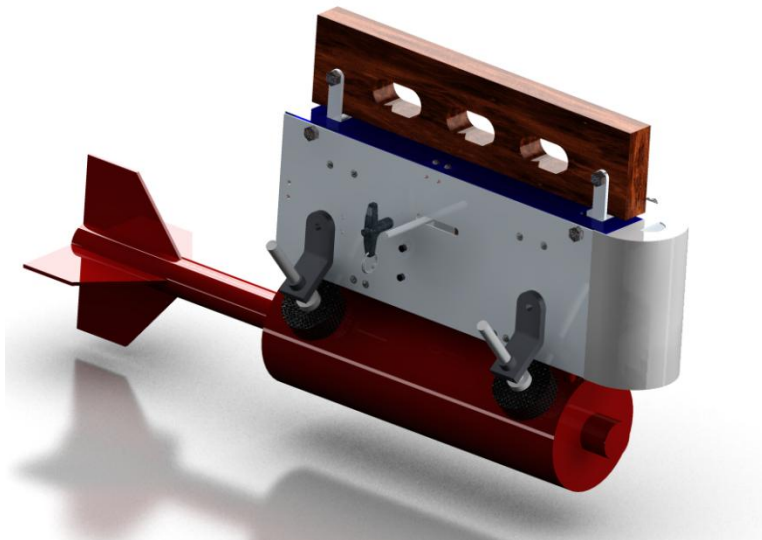# Miniature Bomb Rack Unit

Senior Design Final Report - April 2012

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Abstract

The senior design project, *Miniature Bomb Rack Unit*, was a project proposed by Mr. Russell Roberts at the Eglin Air Force Base in Ft. Walton Beach, 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 control when a shot of compressed air is released to the pneumatic 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 allowing the system to be fired.

The ejector mechanism is coupled with the linear travel of the hook. As the hook travels horizontally that motion is transferred into the vertical motion of the ejection mechanism. The ejection mechanism consists of a steel 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.

The servomotor will be controlled using an r/c controller for testing. The solenoid valve and limit switch will be wired directly to a test control panel. This 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.

In the spring semester of our project, the BRU was constructed and tested. The parts for the BRU were ordered and raw materials were machined at the FAMU/FSU College of Engineering machine shop. Once all parts were received the assembly of the prototype began. All parts of the BRU came together smoothly, with slight modifications due to tight tolerances. The pneumatic system was then installed to complete the integration of the hook assembly and ejection mechanism.

The first test attempted failed, due to the hook assembly jamming with the ejector piston. The friction between the piston head and sliding hook was too great, which prevented any lateral motion of the hook assembly. The hook assembly was then redesigned with the ejector slot having a 30-degree angle versus the previous one of 45-degrees, which increased the downward force exerted on the piston. The slot and piston head were sanded with a 400 grit sand paper to reduce friction. With the new hook assembly design implemented, the pneumatic system was able to fire correctly moving the hooks and at the same time firing the ejection piston.

Once the prototype was built a testing wing structure was constructed. The pylon, which is used to mount the BRU to the aircraft, was mounted to the wing. This allowed for a stable platform to mount the BRU and payload to during testing. Before testing began the payload, which was made of PVC pipe, had to be filled with soil to reach the 10 lb. specification. Videos were taken to test the ejection velocity and an average of 4.31ft/s was measured. This meets the constraint given of at least 4 ft/s ejection velocity. The weight of the BRU was measured at 5.4 lb, that is 8% overweight than the 5lb limit. In the attempt to keep the design low cost, more expensive composites such as carbon fiber were not used. It is suggested that a slight increase in budget to allow for said composites could easily allow for a BRU to weigh less than the required 5 pound limit. The 1G landing shock and 2G lateral load were tested, but empirical data could not be recorded; however, we do feel that our BRU and payload meet these design specifications.

Overall the project was a success, meeting the overall requirements provided by the customer. A great deal was learned about the design process, and we were able to apply the knowledge learned from previous classes to design and build a working prototype of a Miniature Bomb Rack Unit.

Project Scope

Problem Statement

The emphasis of our project involves the design and fabrication of a Bomb Rack Unit (BRU) for the Tigershark UAV, capable of housing and deploying a given payload. The launcher design must meet the design requirements specified by the Air Force Research Lab (AFRL). After successfully completing the design phase, a prototype will be manufactured. This prototype will then undergo a series of tests to ensure the BRU meets the requirements set by the AFRL.

Justification and Background

Unmanned aerial vehicles (UAVs) have become increasingly common on today's battlefield. Since 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 entails 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.

Design and Analysis

Constraints

As mentioned earlier, this system is going to be used on the Tigershark UAV; thus requiring the system must 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. 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 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 mount to a pylon that is 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 four main components: the latch system that will be used to secure the payload, the mechanical safety lock that will hold the hook in place during flight, sway braces which prevent lateral and vertical motion in flight, and the ejection system that will propel the payload away from the aircraft.

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 slide the hook away during the firing procedure. Several different types of release mechanisms will be considered and are outlined in the following section.
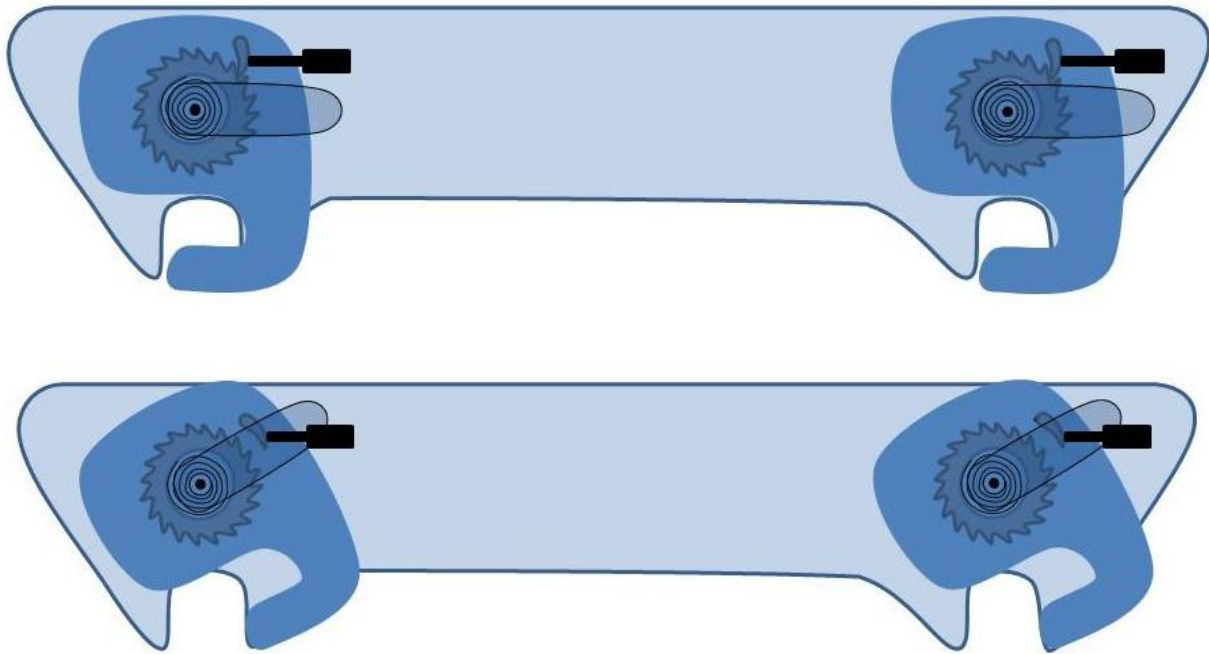
Figure 1- Ratcheting latch design in closed and open position

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 torsion 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 torsion 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 torsion spring which will rotate the hook and release the payload.
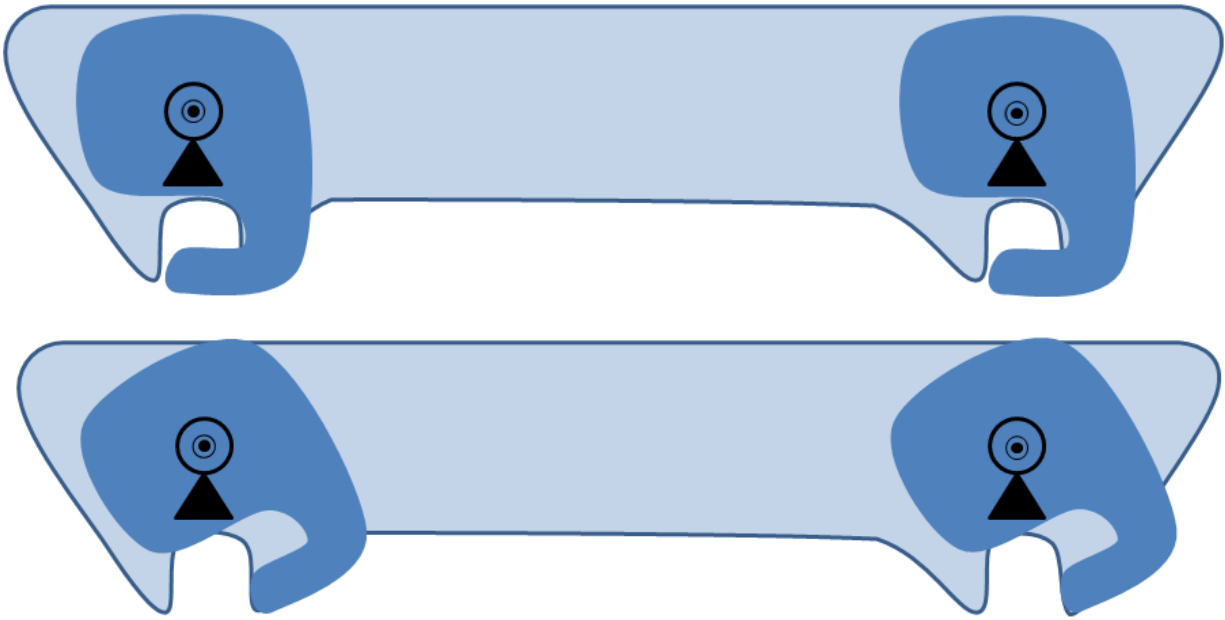
Figure 2- Motorized latch design in closed and open position

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.
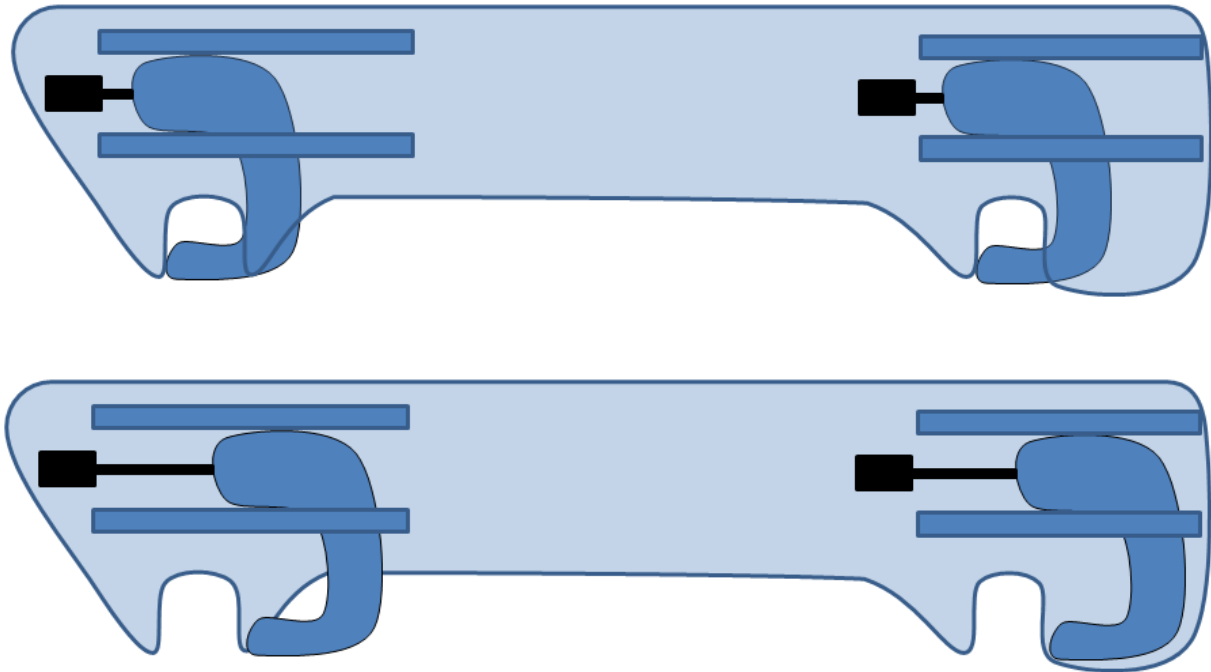
Figure 3- Sliding latch design in closed and open position

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.
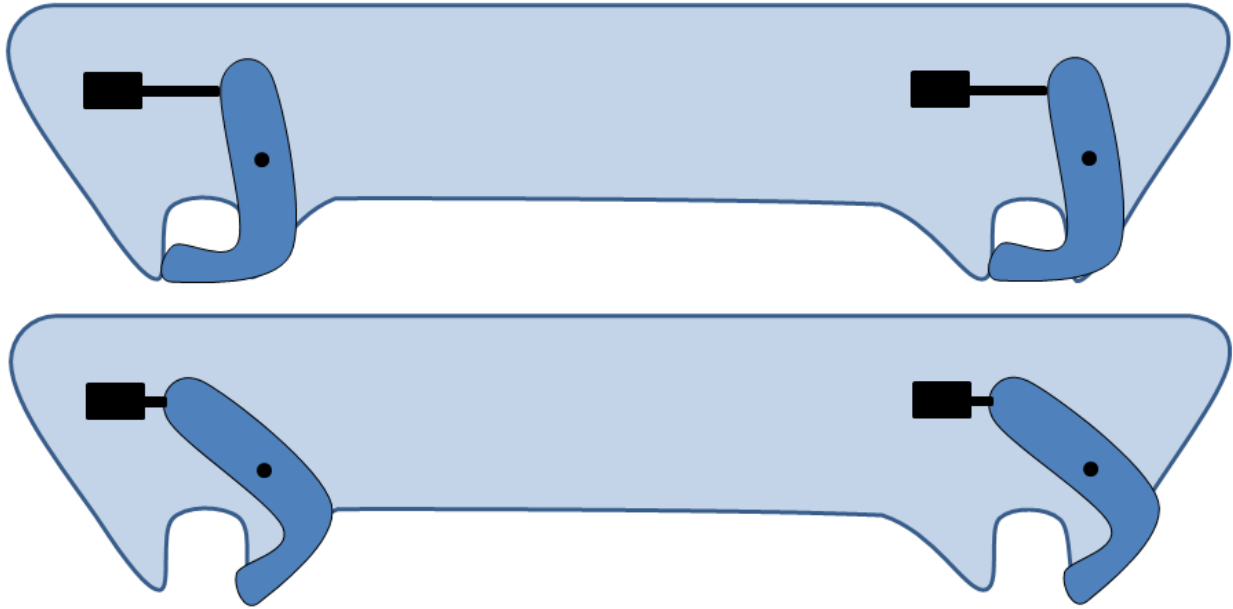
Figure 4- Linear actuator design in closed and open position

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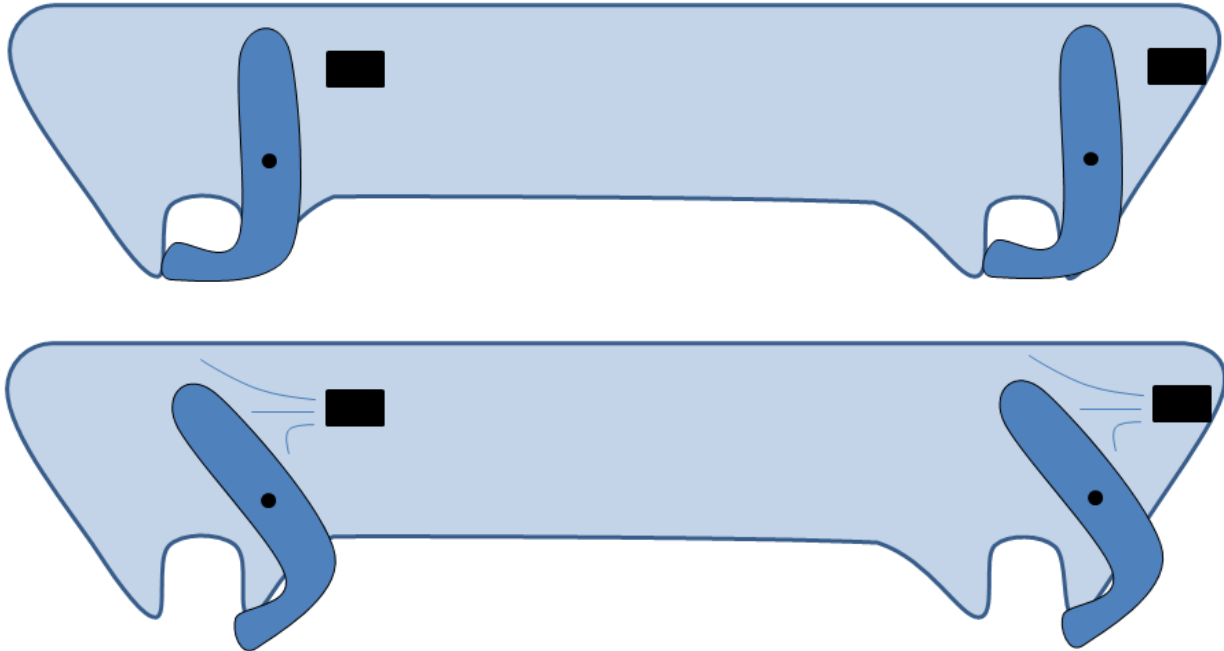
Figure 5- Compressed air latch design in closed and open position

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.

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.
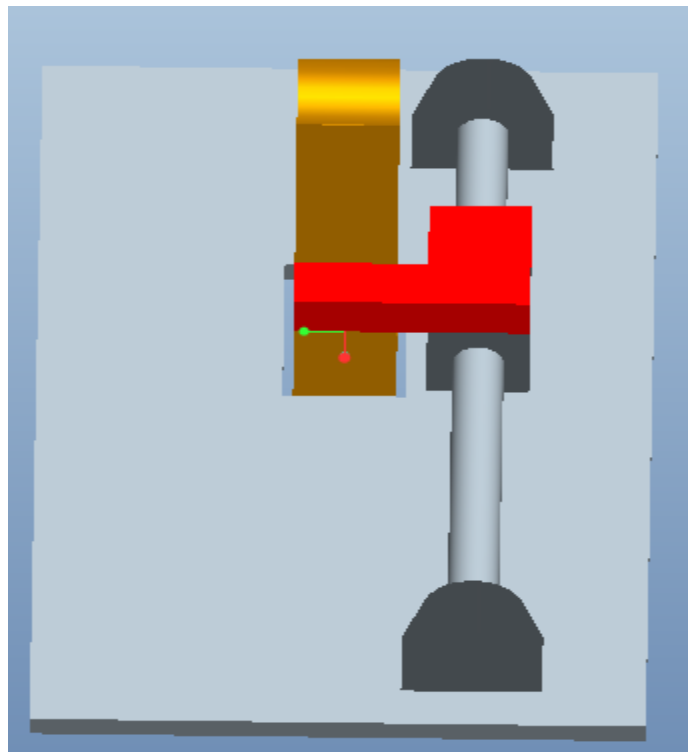
Figure 6- Mechanical Safety Design 1

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.
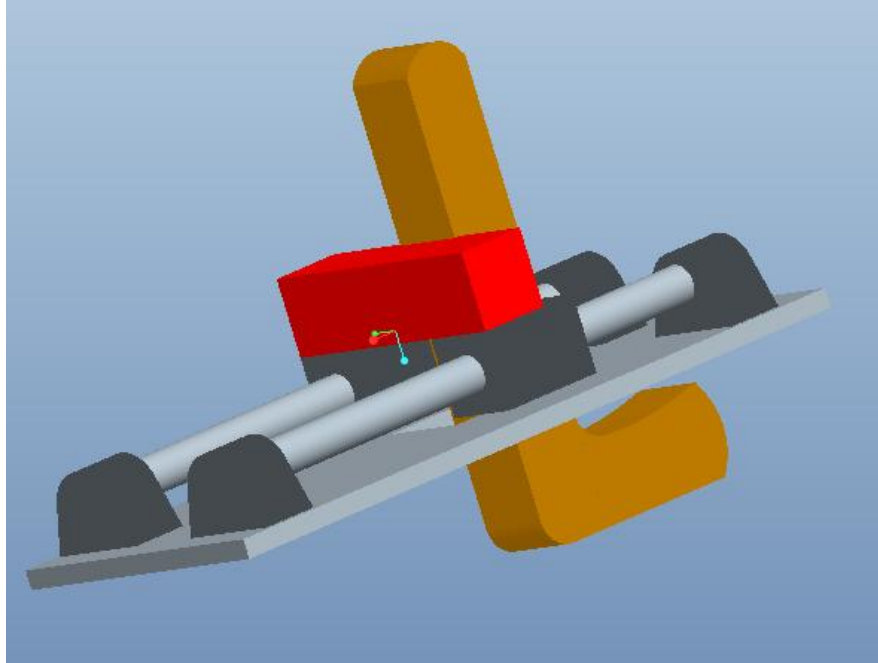
Figure 7- Mechanical Safety Design 2

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.
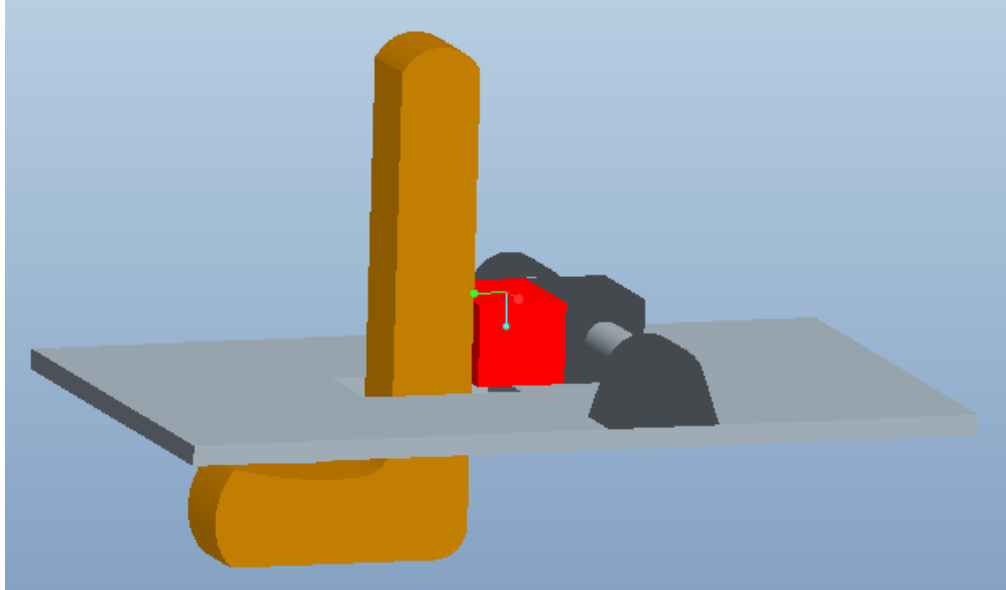
Figure 8- Mechanical Safety Design 3

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.
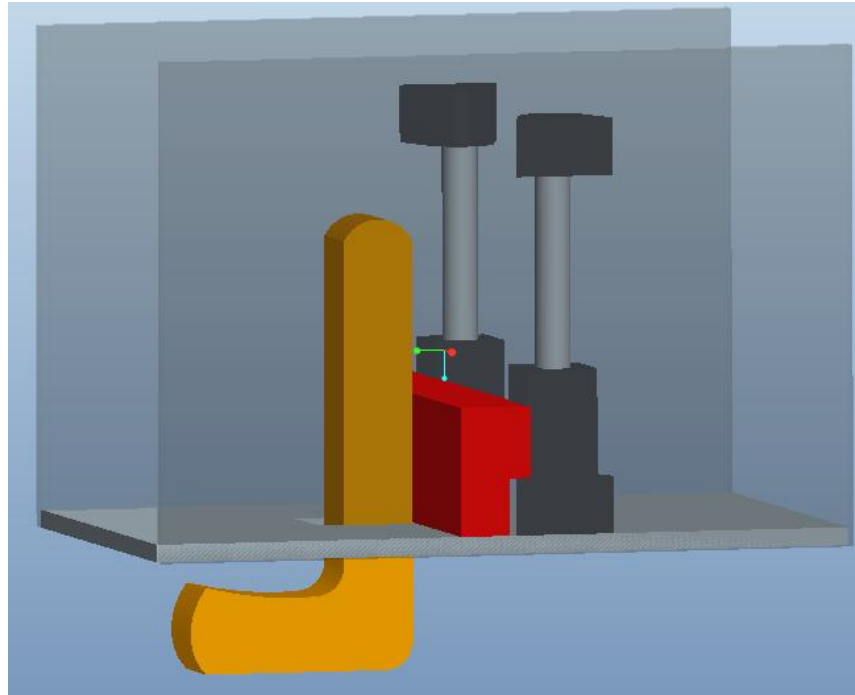
Figure 9- Mechanical Safety Design 4

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 two 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, figure 10 shows the design of this type of system.
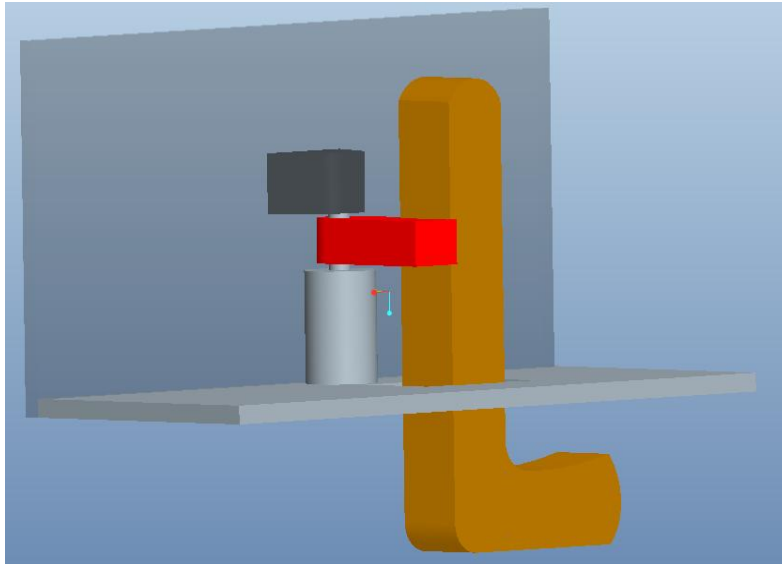
Figure 10- Mechanical Safety Design 5

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.
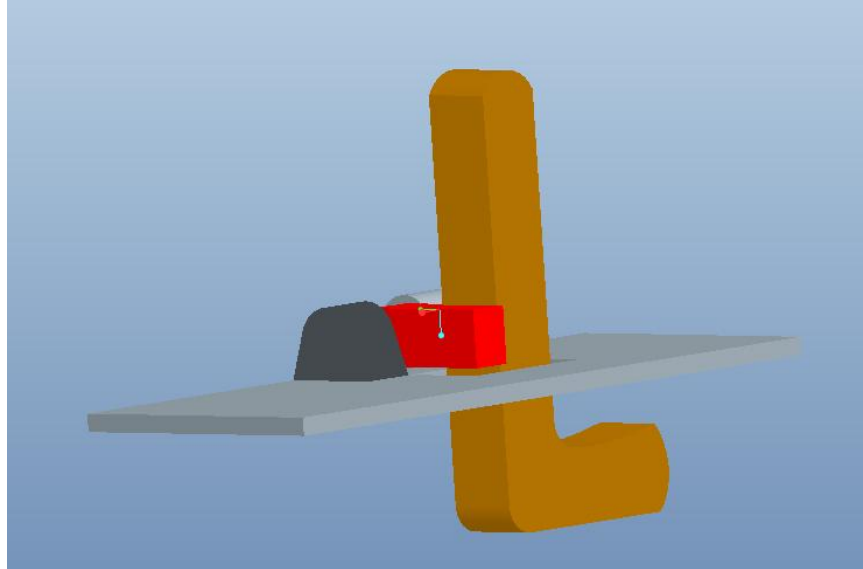
Figure 11- Mechanical Safety Design 6

As the above figure illustrates, the 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.

Specifications	Weight	Mechanical Safety System Designs											
		1		2		3		4		5		6	
		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

The sway brace is a critical design feature on the BRU, and is used to prevent the payload from moving laterally or vertically during flight. The payload could experience lateral forces of up to 2G during a turn. The sway brace must be able to resist this force and keep the payload steady. Below are two designs for the sway brace.
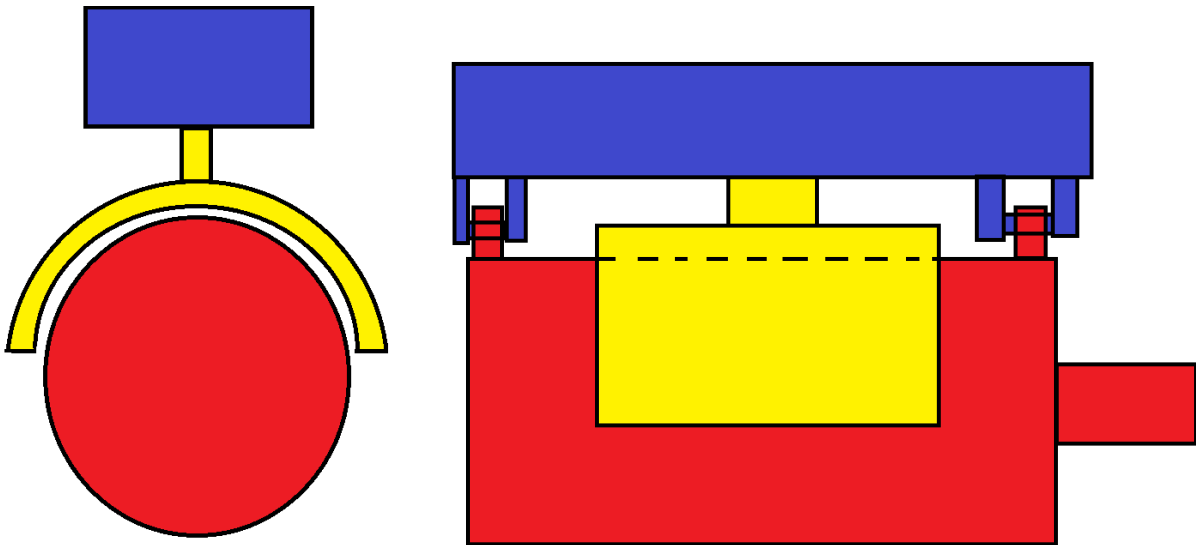
Figure 12- Sway Brace Design 1

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 payload (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.
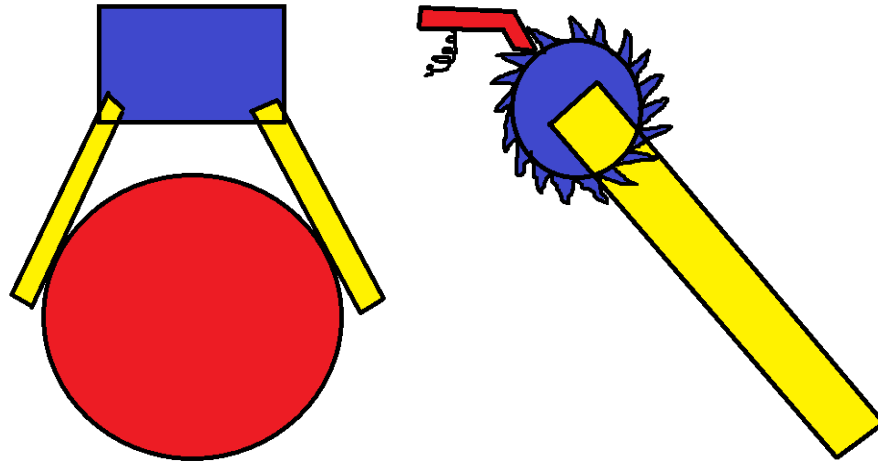
Figure 13- Sway Brace Design 2

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 2G as well as keeping the payload 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 payload 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.
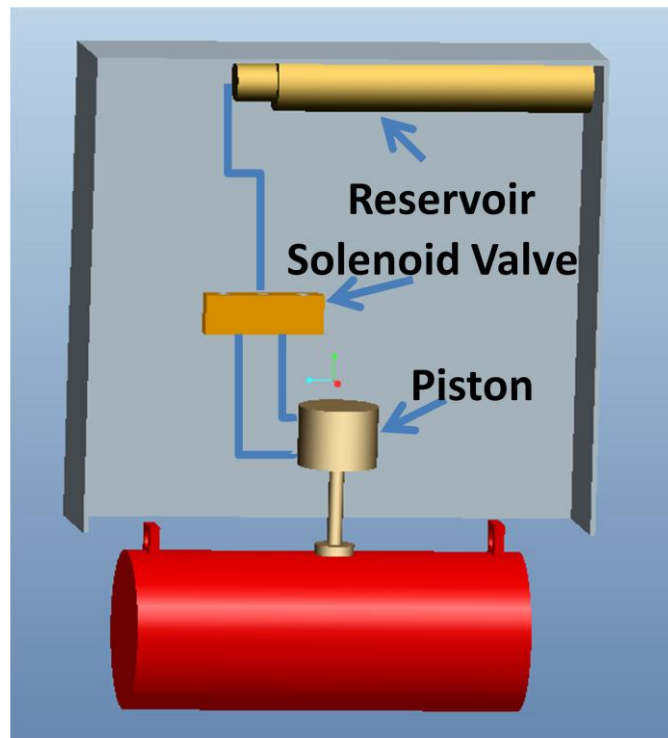
Figure 15- Payload Ejection System Design 1

Design 1 above uses a pneumatic piston (silver) to eject the payload (red). The piston will push the payload 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.
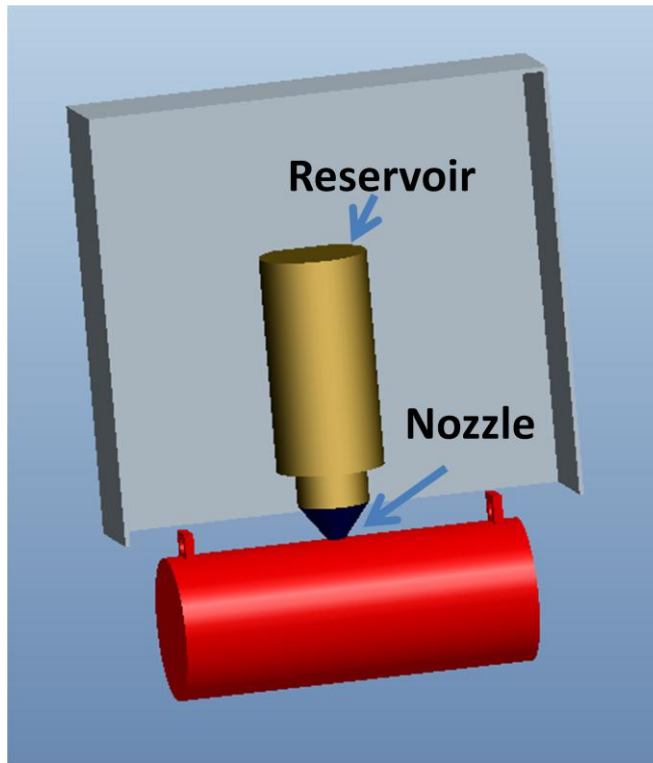
Figure 16- Payload Ejection System 2

Design 2 uses compressed air to eject the payload. A compressed air canister (brown) 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.
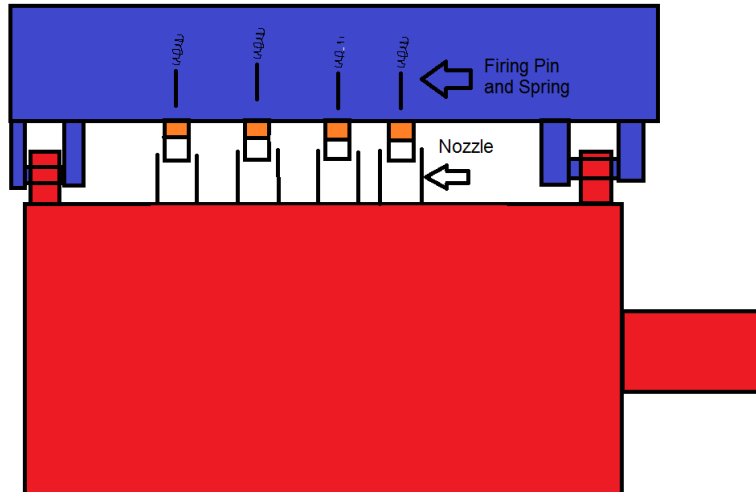
Figure 17- Payload Ejection System Design 3

Design 3 above uses pyrotechnics to eject the payload away from the aircraft. The pyrotechnics involved will use gun powder (orange) to create a controlled explosion to force the payload down and away from the aircraft. As shown in the illustration, multiple explosives can be arranged 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.
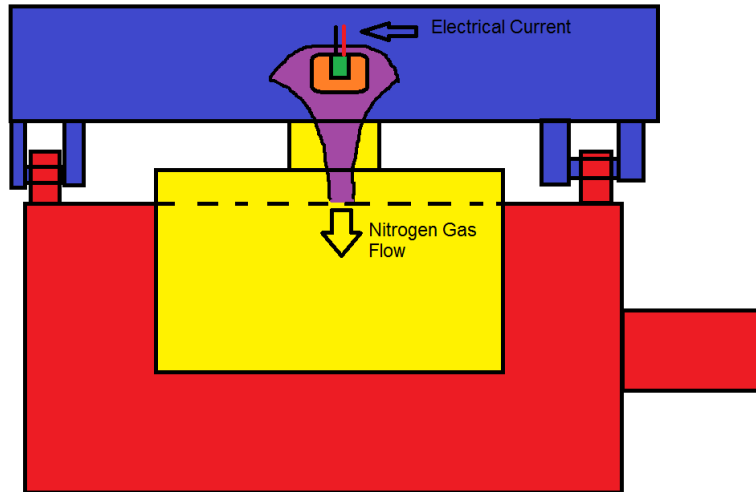
Figure 18- Payload Ejection System Design 4

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

Using the design criteria above, each design idea earned a score based on how well it achieved each specification. These results show that designs 2 and 3 have the greatest score. However further design review and calculations will be required to determine the viability of the designs and what ultimately will be used as an ejection mechanism for the BRU.

Final Design

Overall Design
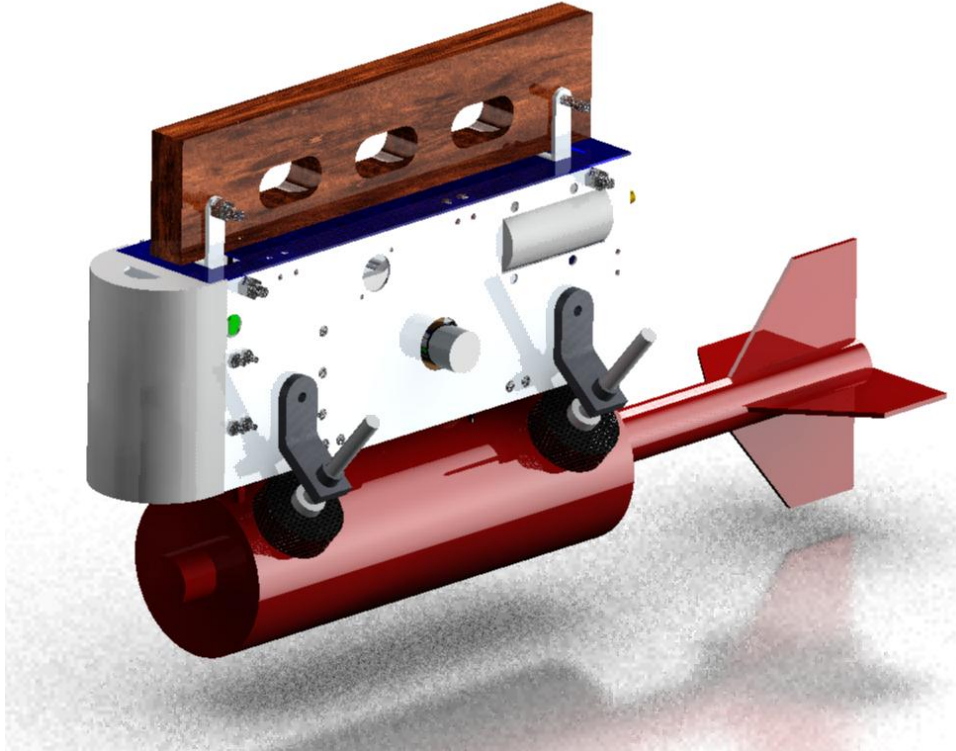
Figure 19- Final Design Render

The final design, shown in Figure 19 above, combines the various subsystems into one cohesive Bomb Rack Unit. A linearly actuated hook design is incorporated to support the payload in flight as well as quickly and easily open to allow payload deployment. This hook design is operated by a pneumatic (air) system which will be pre-charged during ground operations and controlled in flight by a solenoid valve. In order to keep weight low and keep the overall complexity of the system to a minimum, the ejection system has been integrated into the hook motion using the force of the pneumatic cylinder to power the ejector. This reduces the overall weight of the BRU by requiring only one power source. For the safety system a servo is used to rotate a "stop block" into the path of the hook preventing unwanted deployment. Finally, the sway braces incorporate a simple COTS swivel foot which will be screwed down to provide support for the payload. The following sections discuss in more detail how each subsystem operates and the analysis done to ensure optimum results.

Pneumatic System

Figure 20- Pneumatic System Components. Left to Right: Schrader Valve, Air Tank, Solenoid Valve, Air Cylinder

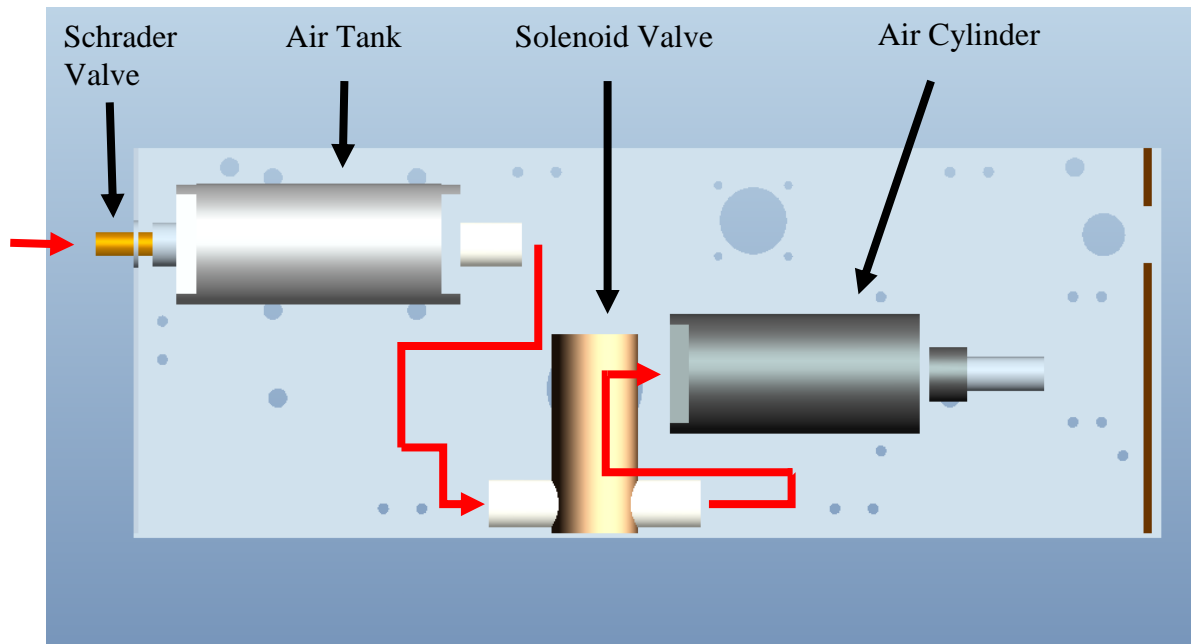The pneumatics system is vital to the operation of the BRU. This system is in charge of disengaging the hooks to allow the payload to be launched as well as provide sufficient force to eject the payload. The primary component is the pneumatic cylinder, which when supplied with sufficient air pressure will extend a one inch long piston. Based on calculations for the ejection system and comparing to available COTS air cylinders, it was determined that a cylinder which will provide a force of about 150 lbs. would be needed to create sufficient ejection velocity. Based on this, a pneumatic cylinder was chosen with a 1 ½ in. bore with a 1 in. stroke. A single acting cylinder, which has an internal spring, was chosen for its natural position being closed and does not require air pressure to hold it in that position. A 4 cubic inch air tank was chosen for its small size and high pressure capacity of 250psi, which is well above design requirements. The tank will store the air needed for payload ejection while in flight.

To allow ground operations personnel to fill the system a Schrader valve is attached to the air tank. This is a one way valve commonly used on vehicle tires. Tools which would be used to connect the Schrader valve to a compressor are common at a military base, so no new tooling is required. A pressure gauge is required to ensure the system is within safe operating

range. The recommended pressure range is between 90 and 110psi. Below this range the payload will not eject at desired velocity and above this range could cause damage to various components.

Following the air tank, the next component of the pneumatics system is a solenoid valve. This valve acts like 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 valve, switching the position of the valve allowing air to be sent to the cylinder. After ejection, the solenoid valve will return to its de-energized state and the air inside the air cylinder will flow out to the atmosphere through an exhaust port inside the solenoid valve. The hooks will once again be closed inside the BRU. Below is an illustration of the solenoid operation.
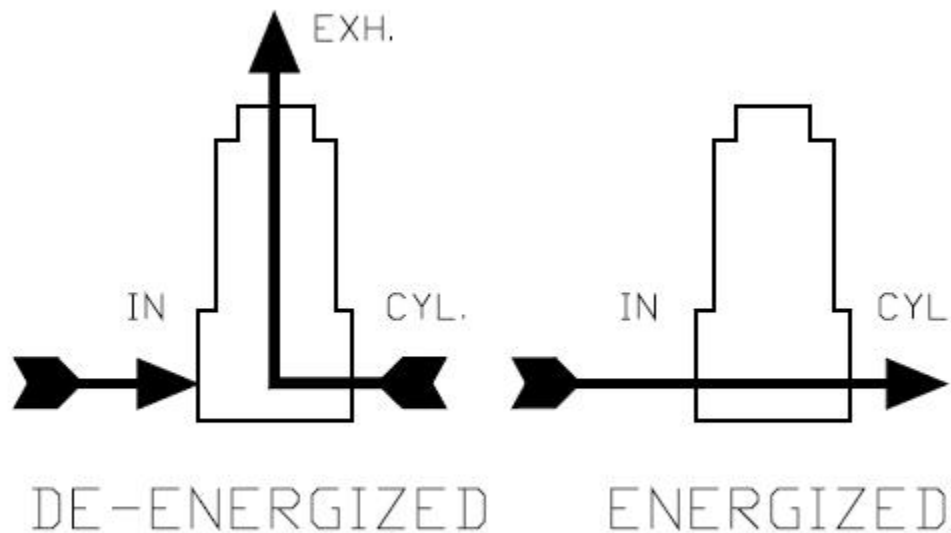
Figure 21- Solenoid Valve Operation Diagram. Courtesy of cylval.thomasnet-navigator.com

Hook Analysis

For the Hooks an FEM, finite element method, 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 lbs. load on both hooks from the payload.
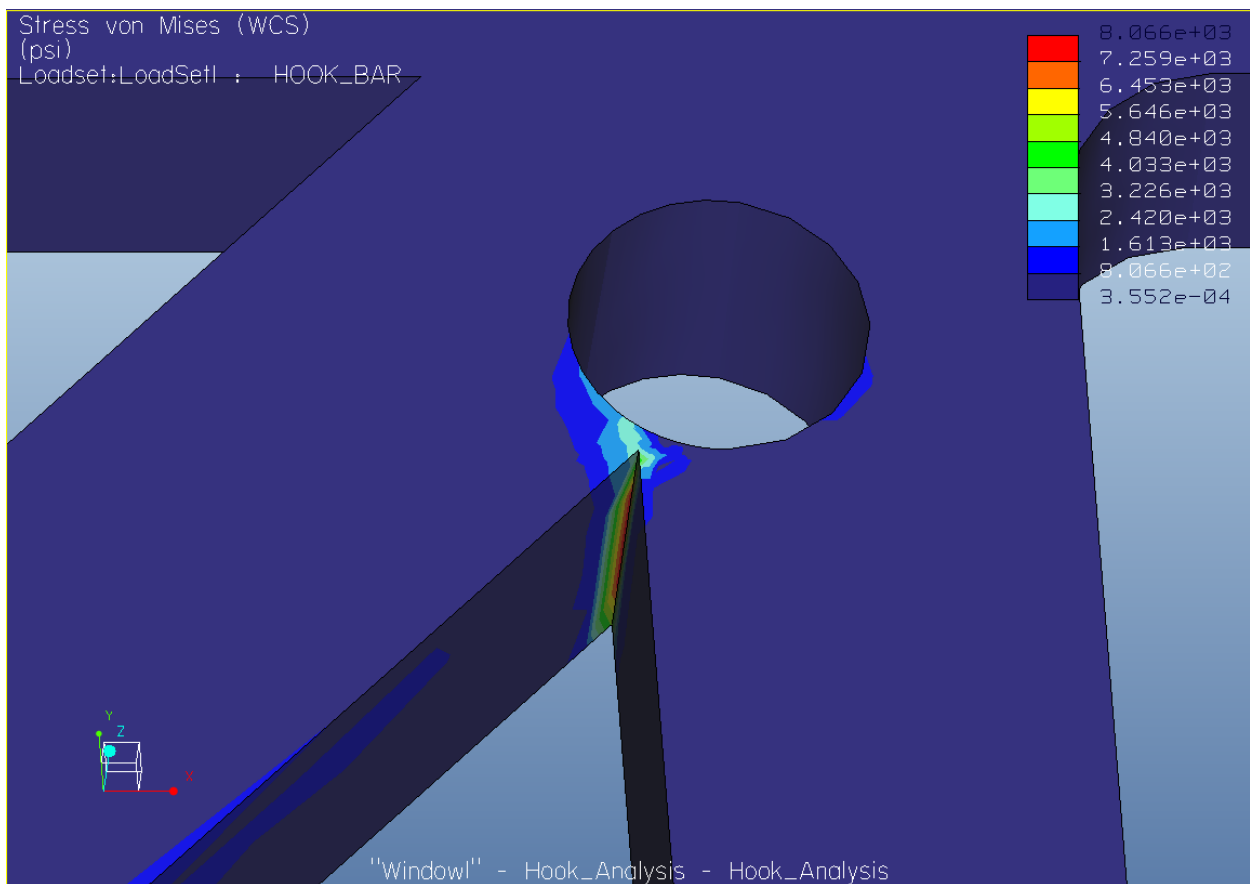
Figure 22- von Mises Stress. Concentrating on Upper Portion of Ejector Ramp

In order to optimize the shape of features on the hook, stresses were analyzed and stress concentrations were located. Shown above in Figure 22 is a major stress concentration peeking at about 8066psi. This is greater than the 8000psi yield stress of aluminum. A fillet will need to be added to remove the stress concentration.
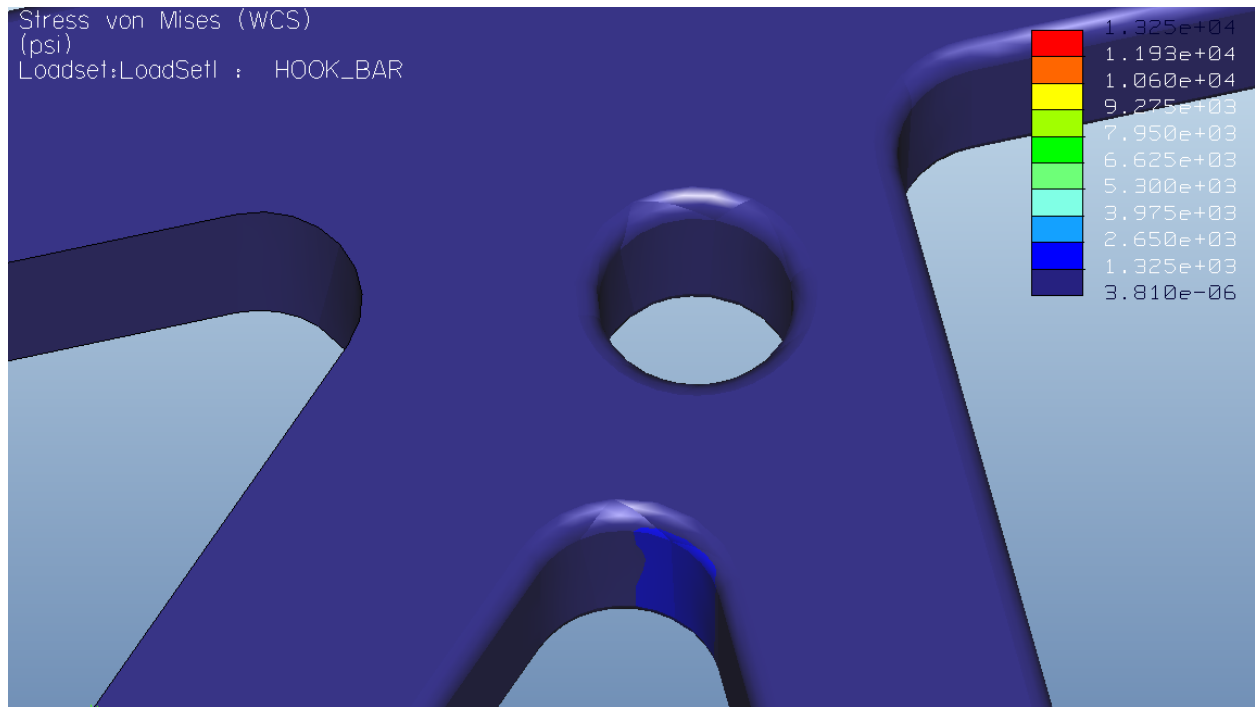
Figure 23- von Mises Stress. Concentrating on Upper Portion of Ejector Ramp.   Fillets added to reduce stress.

Figure 23 above shows the stresses seen after the fillets were added. It is shown that the stress concentration has been greatly reduced.   Max stress shown at this location is around 2000psi, this is a drastic reduction compared to the unrounded stress of 8066psi. As a result of these findings, fillets will be created when machining the hook slot.
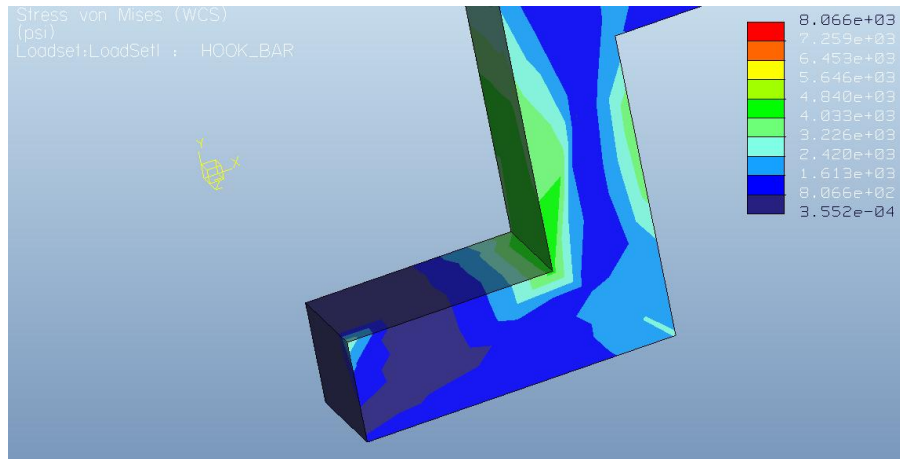
Figure 24- von Mises Stress of the Hook.

Similarly, the stresses were analyzed in the hook portion of the part in Figure 24. Greatest stress occurred along the upper corner, peeking at about 4000psi. While this analysis provides a sufficient safety factor of about 2, a small fillet at this location would be beneficial.
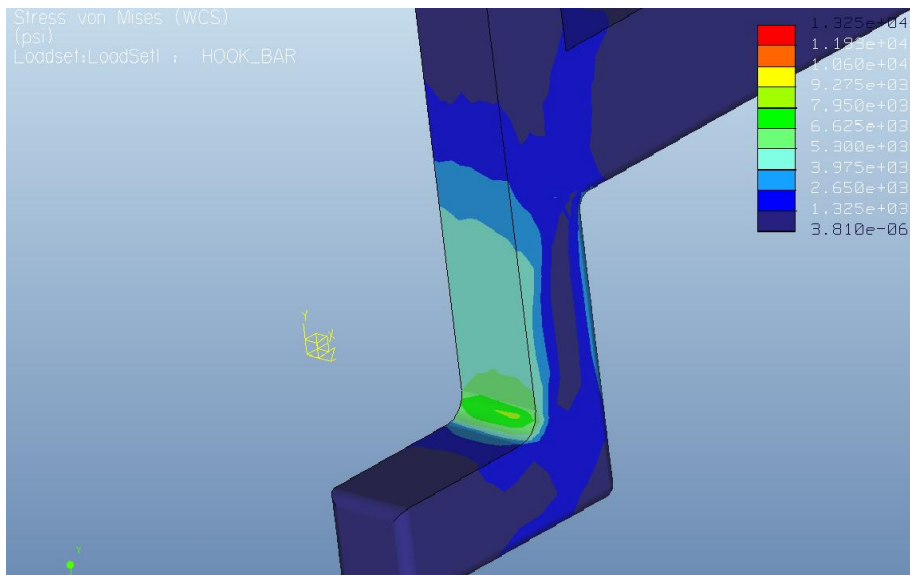
Figure 25- von Mises Stress of the Hook. Rounds added to reduce stress concentration.

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 beneficial 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 lightweight, durability and 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 26 below shows a bearing system mounted above and below the hook. There are similar bearing blocks on the other end of the hook.

Figure 26- 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.
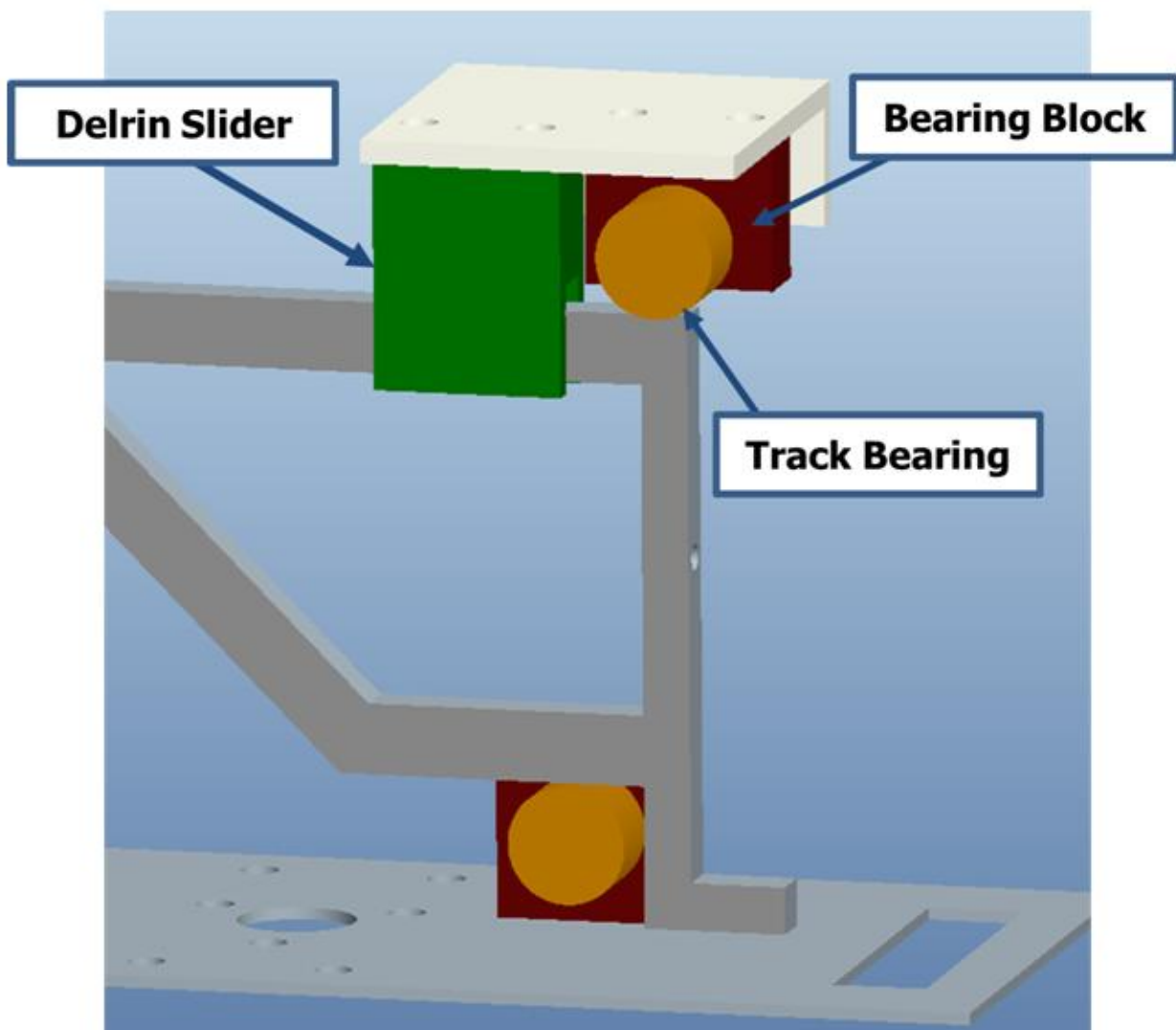
Once the forces on each hook are known, the next step is to perform a torque balancing analysis on the hook itself. The first case analyzed was the hook in the closed position. In this position the entire weight of the payload is supported by the bottom two bearings; the forces in the upper bearings 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 27 will be used.

Figure 27- 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 2G as well as a 1G 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 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 3G 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 lbs. 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 lbs. in the maximum downward 4G 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 lbs. 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 2G force is acting laterally while another 2G force is acting downward. This will create various stresses 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 2G 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.
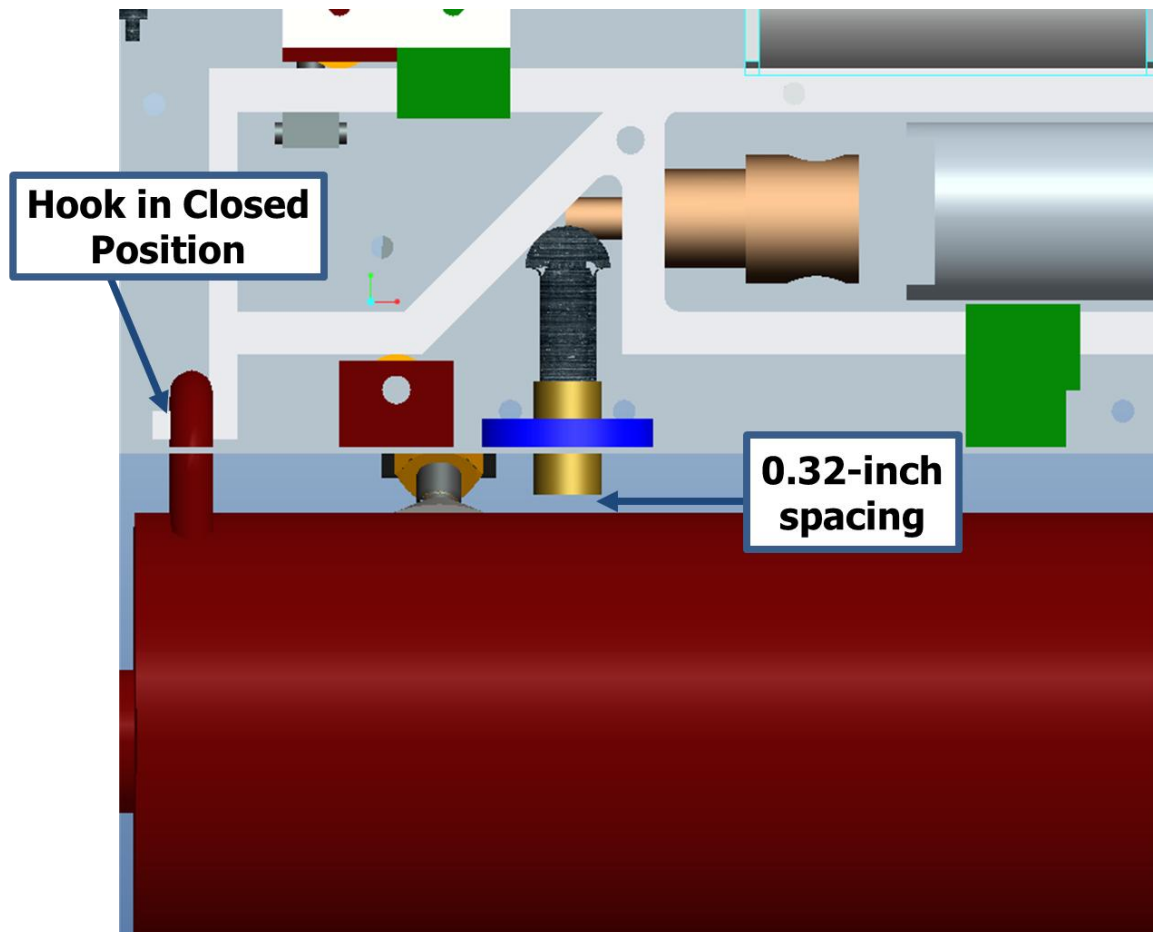
Figure 28- Ejector piston in retracted position

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.
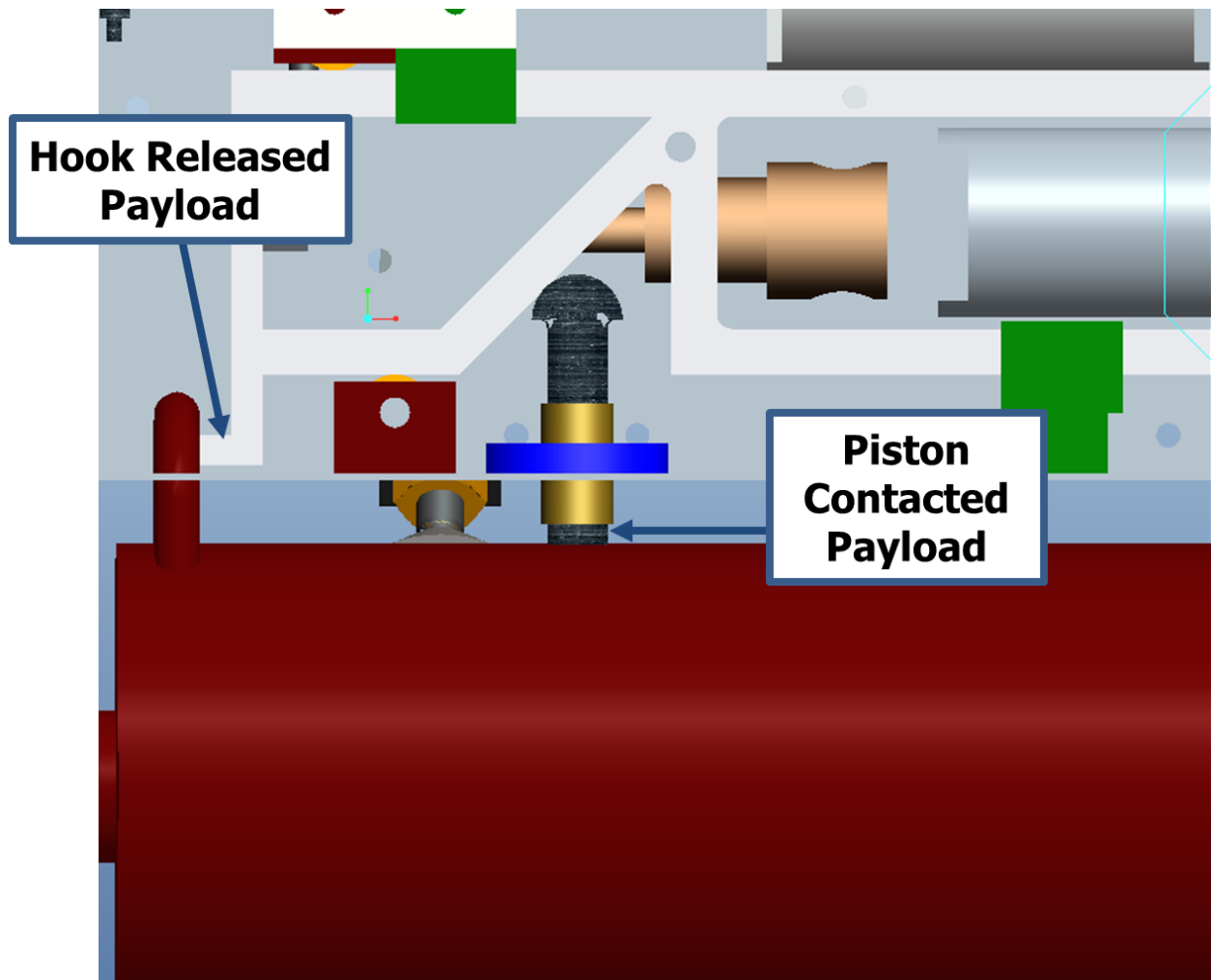
Figure 29- Ejector Piston Contacted Payload

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.
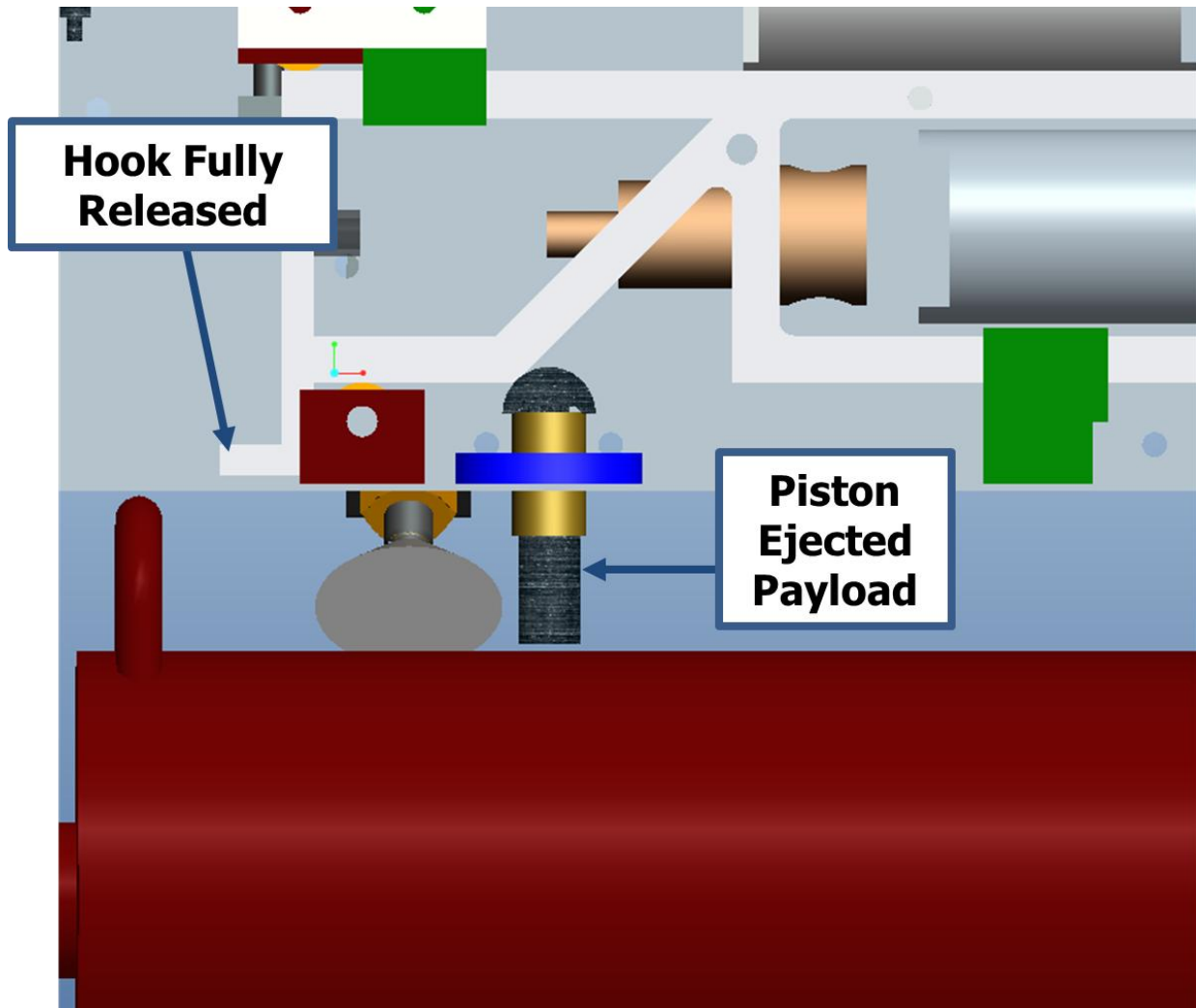
Figure 30- Ejector Piston Contacted Payload

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 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- Working Model Analysis

The Working Model simulation solutions were compared to the numerical solutions provided by MathCAD. Figure 31 above is a graphical representation of the simulation results; payload velocity in inches/second is graphed versus time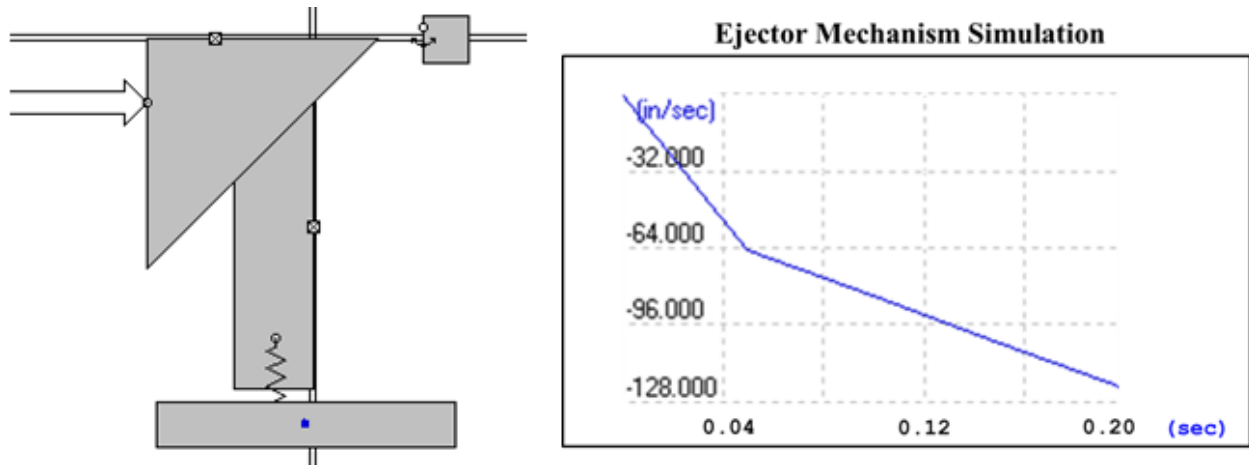 in seconds. There are two definite slopes on the velocity curve. The first slope is much steeper and corresponds to the time that the piston is in contact with the payload. This is the time that passes as the hooks are releasing and the payload is being ejected from the BRU. Around the time 0.05s is 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.

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 is placed in the BRU assembly.
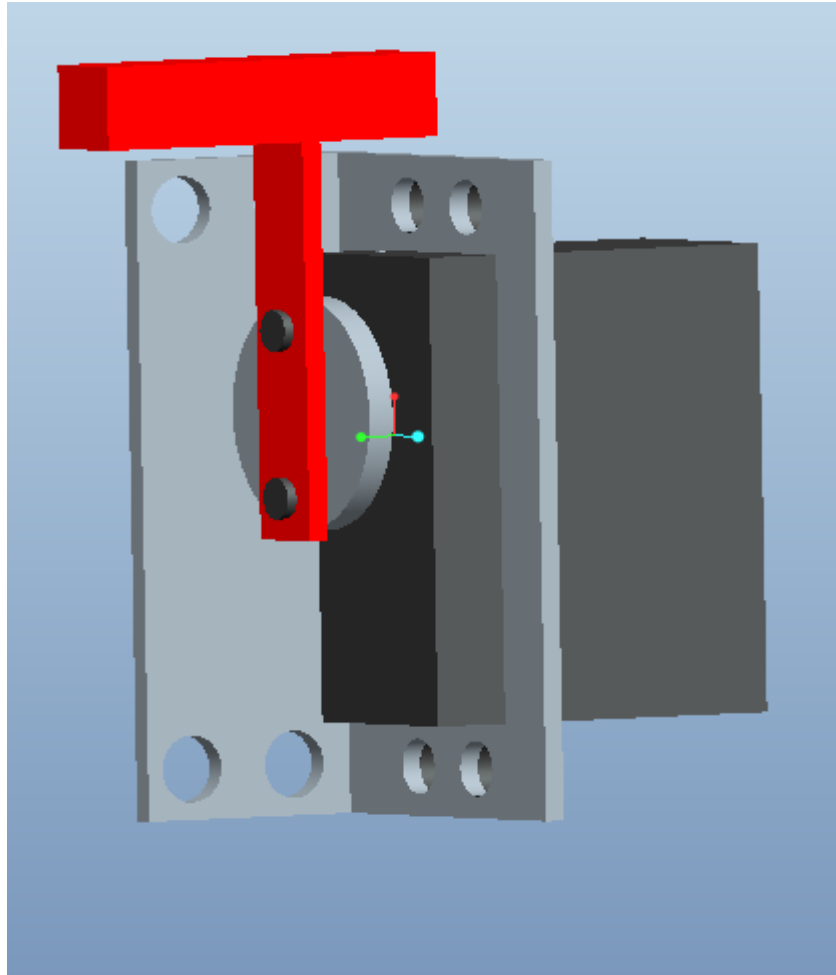
Figure 32- Final Mechanical Safety Drawing

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 manufacturer's specification. The torque required to rotate the stop block is calculated 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 from improper use or a misfire. The pneumatic piston being used is rated at a 150 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 150 lbs. 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.
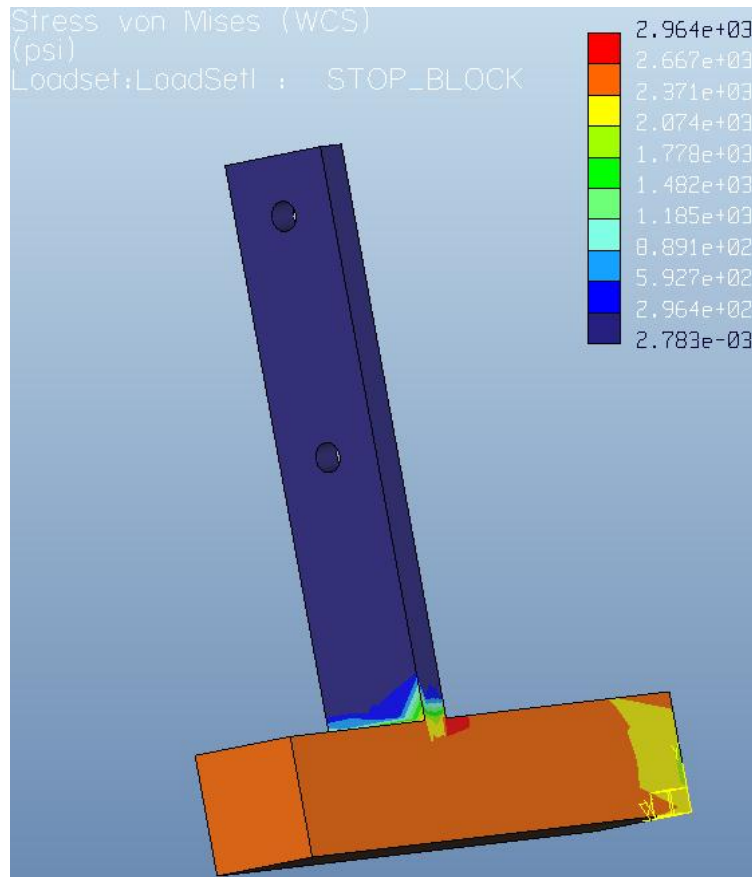
Figure 33- von Mises Stress Analysis of Stop Block

In the figure above, figure 33, 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.
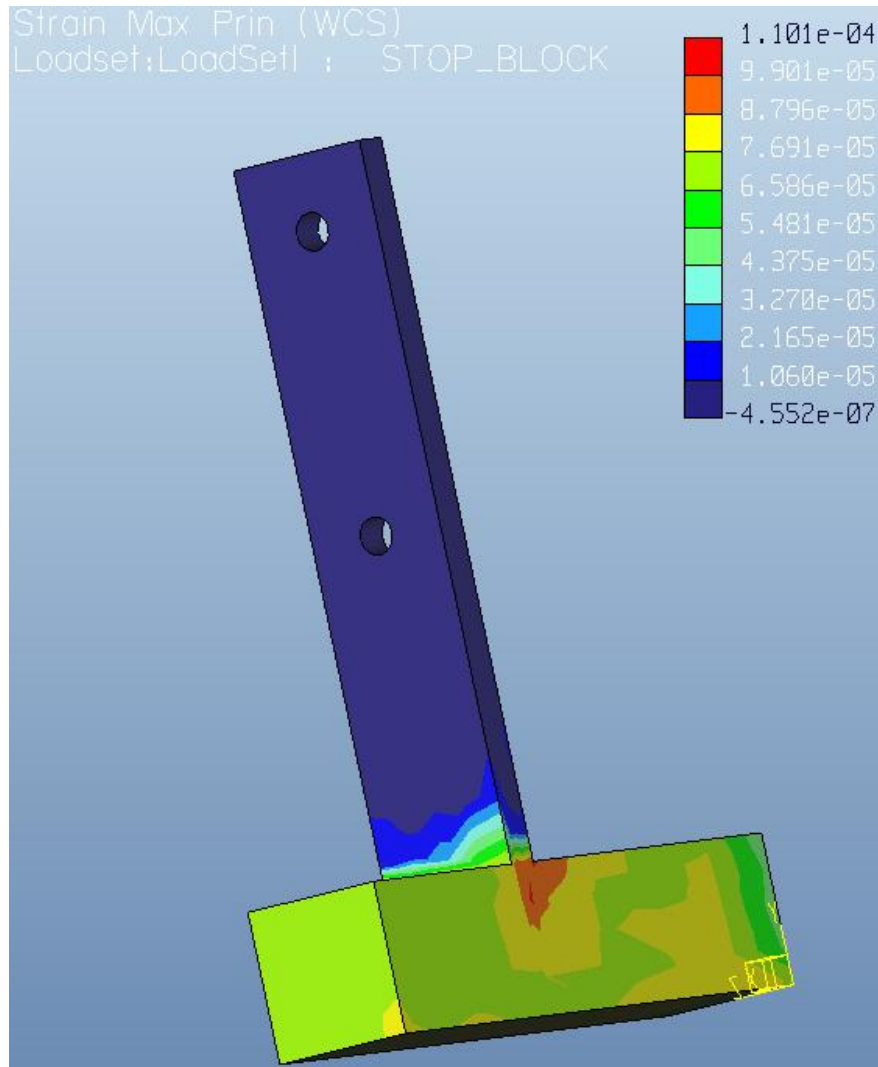
Figure 34- Strain Analysis of the Stop Block

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.
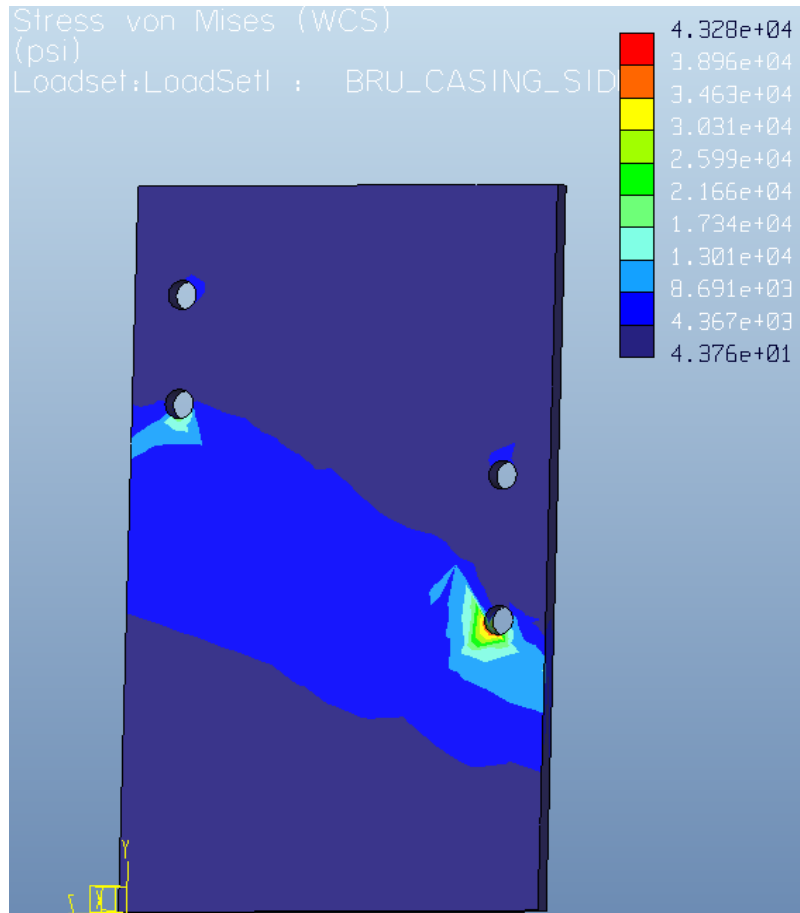
Figure 35- von Mises Stress Analysis of BRU Side Plate

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 down tremendously when the system is built. Therefore the highest stress felt in the system at the 155 lbs. 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.
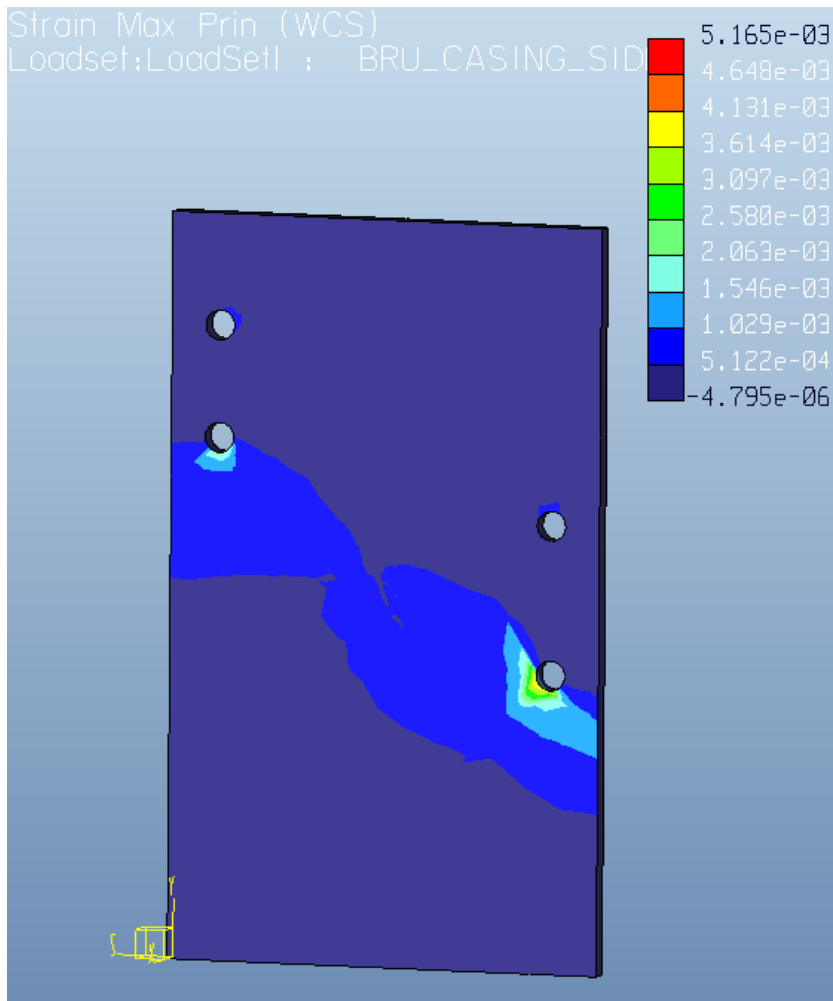
Figure 36– Strain Analysis of BRU Side Plate

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.
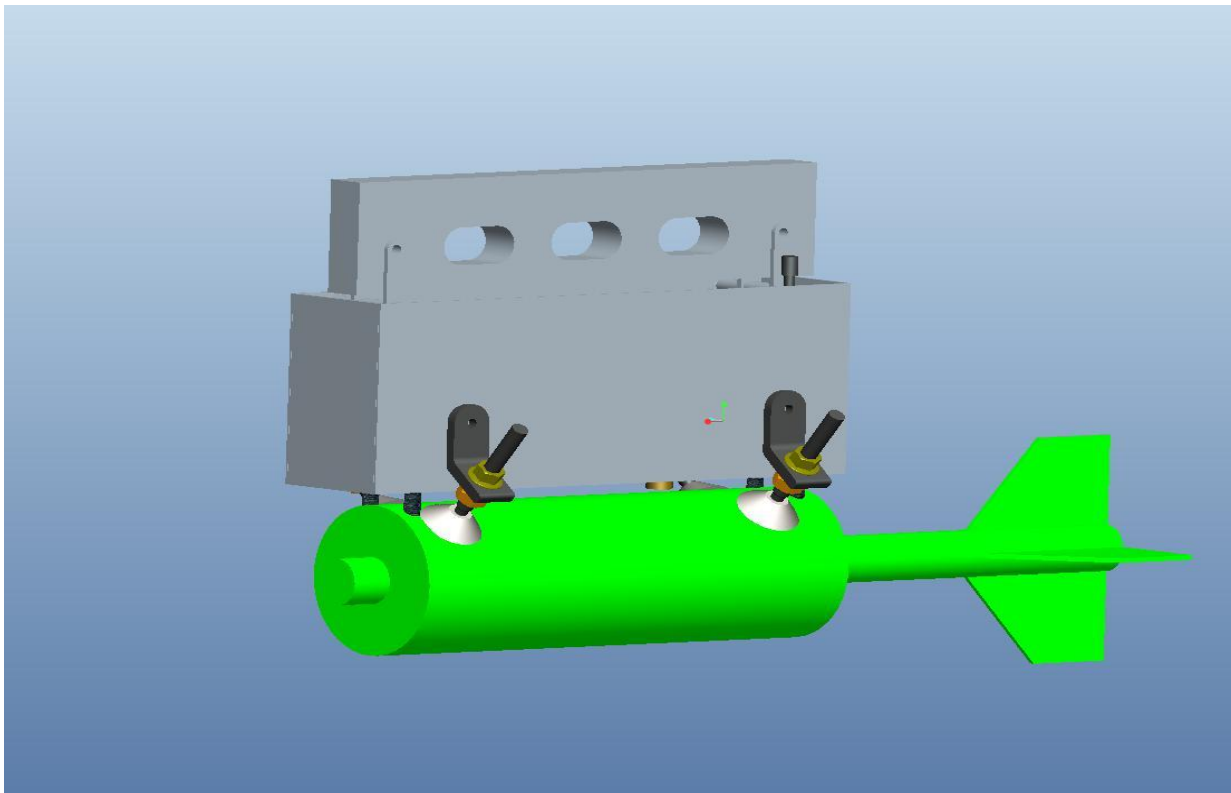
Figure 37- Sway Brace Mounted to BRU with Payload

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 2G 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 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. The next page discusses the analysis done by Mechanica on the sway brace brackets.
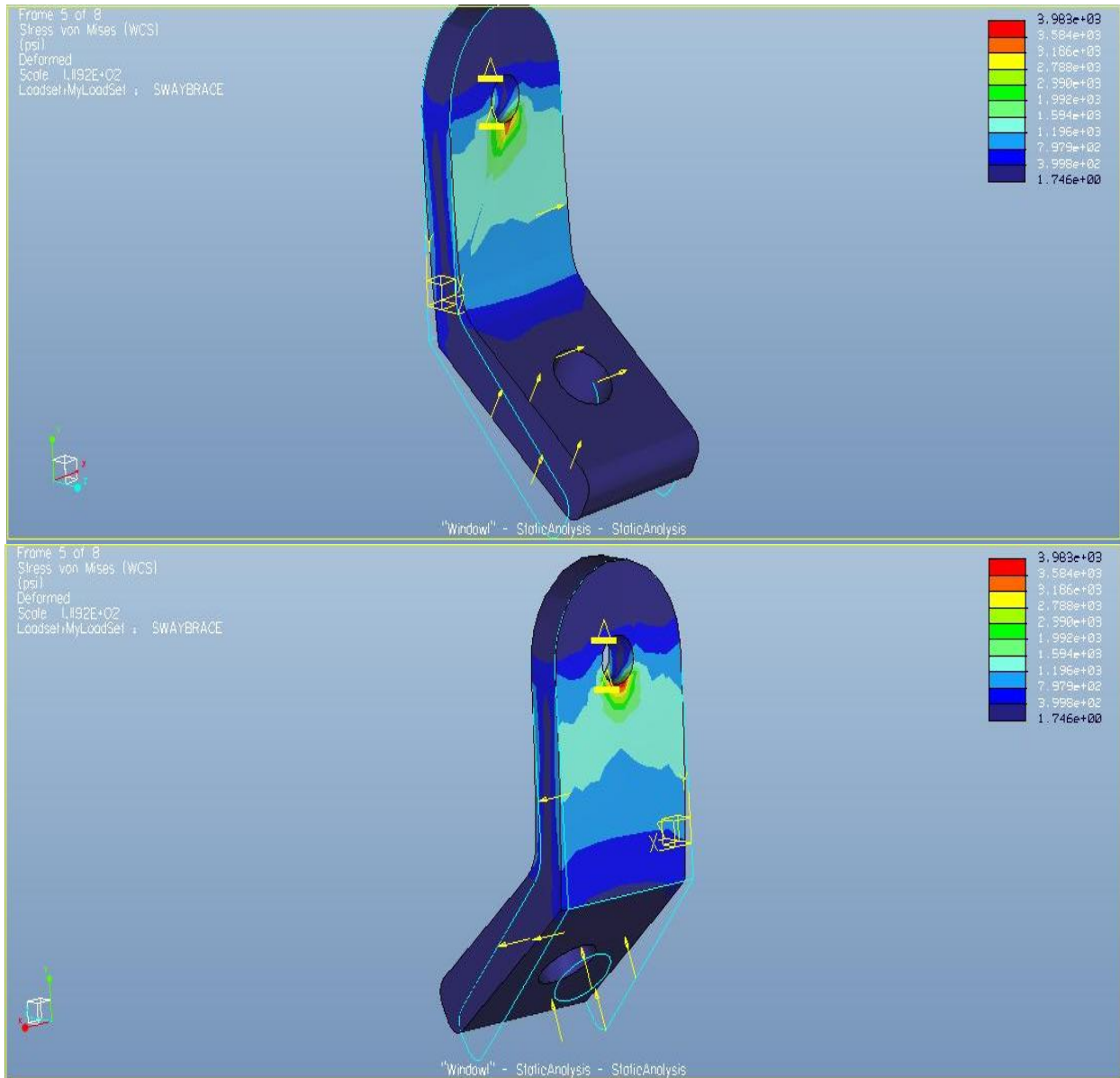

Figure 38- Bracket Von Mises Stress (Front and Back)

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 15lbs 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.

$$F = A*P*C_D$$

$$A = \text{Surface Area of the Bracket}$$

$$P = \text{Wind Pressure, Psf} = .00256 * V$$

$$C_D = \text{Drag Coefficient} \approx 2.0$$

This formula outputs a wind force across the side of the bracket ≈ 6 lbf. 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 30lbs was applied to the face of the steel pad with a shear force of 6lbs. 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.
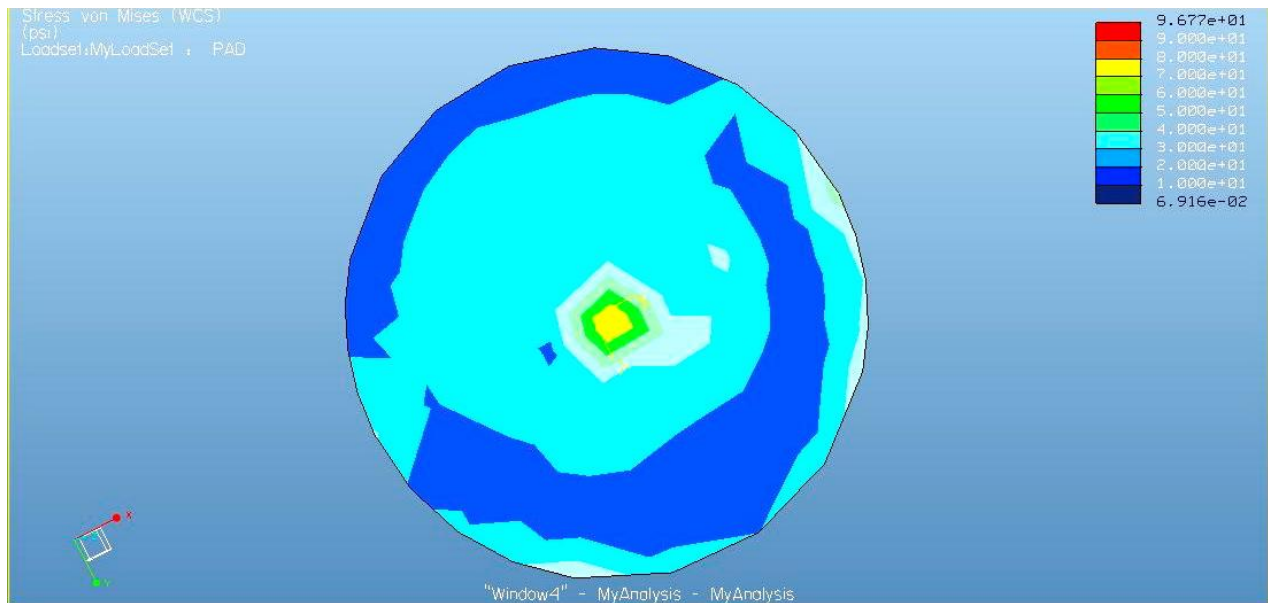
Figure 39- Steel Pad Von Mises Stress (Bottom Face)

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.
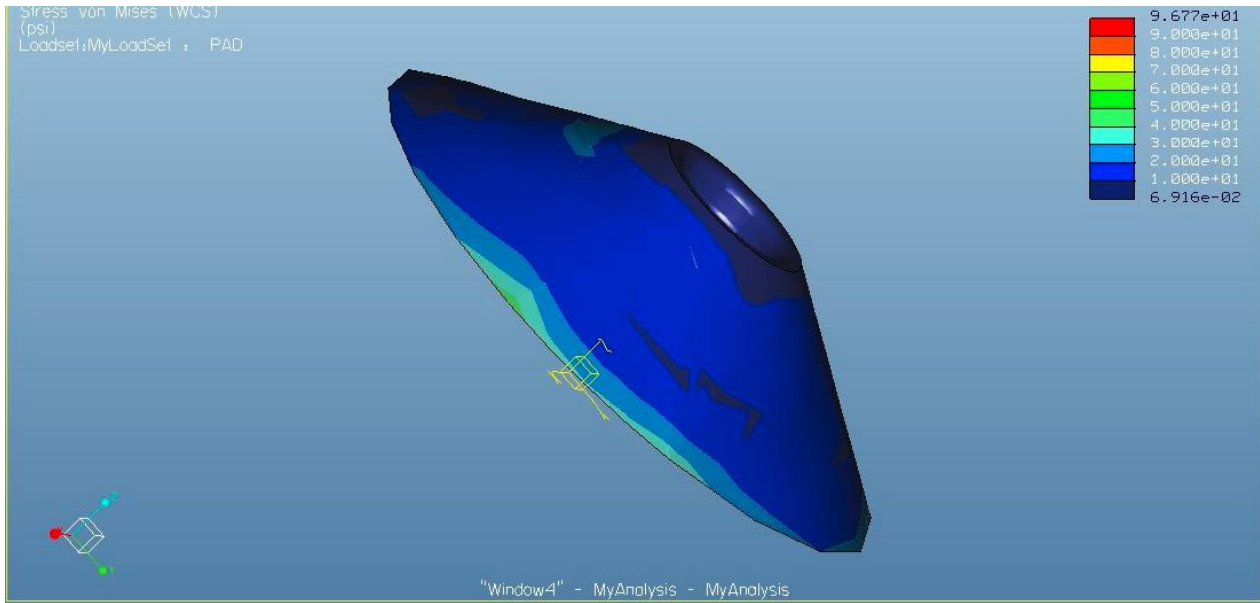
Figure 40- Steel Pad Von Mises Stress (Side Face)

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

Operation of this Bomb Rack Unit requires the use of an electrical interface. A microcontroller is needed to operate the servo used for the mechanical safety system as well as send a signal to the solenoid valve when the "Release" command is given to deploy the payload. The microcontroller and everything needed to operate the BRU will be housed in the forward nose of the BRU. Here the D-shaped connector will be located, allowing connection to the Tigershark's electronics. An electrical team will be in charge of setting up and programming the BRU at a later time. It was decided that for the purposes of this prototype a test control panel is all that is needed. The control panel will include a control devise to operate the servo between its "Safe" and "Armed" positions. There will also be a switch which will be connected to the solenoid valve; when toggled the air cylinder will fire. Finally two LED lights will signal to the operator whether the hook is in the closed (red) or open (green) position. The wires which would normally go to the microcontroller will be connected directly to the control panel instead.
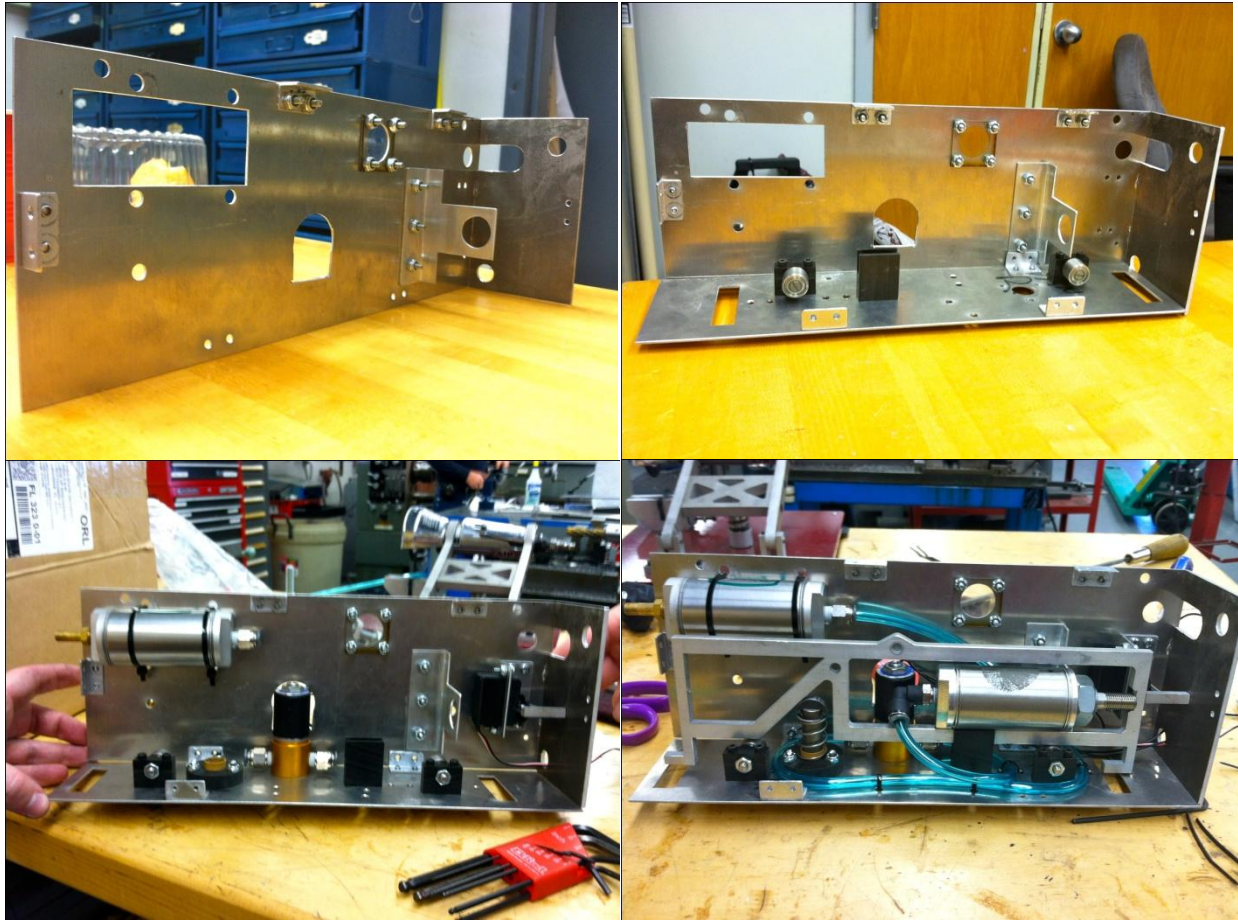
Design Assembly

Figure 41- Assembly of the BRU

The figures above illustrate the assembly process of the Bomb Rack Unit. The port-side panel is welded to the forward panel. Angle brackets are riveted to these panels so additional panes can be mounted. Assembly continues with the addition of the lower bearing blocks and guild which allow the hook to slide linearly and smoothly during ejection. Next the air tank, solenoid valve, and servo safety block assembly are added. All of these components are screwed down except for the air tank. The air tank is secured by zip ties which will hold it tight to the port-side wall. The ejector piston, hook bar, and air cylinder are added next. After ensuring the hook can slide smoothly in the channel, the safety stop block can be manually raised to check the fit. This is critical because if the fit is too tight then the block may not engage and if it's too loose the safety could fail allowing the hooks to open prematurely. The next step is to run the air

hose. The hose must travel from the air tank to the solenoid valve; this means doing a loop of the interior of the BRU before it is able to connect. Leaving the valve, another hose follows a similar path to the air cylinder. The hoses must be pressed all the way into the connectors as to prevent the possibility of leaks. The aft panel can now be mounted and finally the starboard-side panel with the upper bearing blocks and guild rails. These mount to the angle brackets on the interior of the BRU. Finally the roof can be added and the BRU can be mounted to the Pylon and the payload can be attached. The picture below shows the completed assembly. Wires coming from the solenoid valve, servo, and limit switch are run through the forward panel; these would normally go to the microcontroller, however for this prototype they will connect to a test control panel.

Figure 42- Fully Assembled BRU with Payload

Results and Discussion

Stop Block Analysis

Testing of the stop block consisted of making sure the servomotor moved the stop block into position and stop the hook from opening when misfired. To test the servomotor an r/c controller was implemented to move the stop block in and out of the safe position. Power is supplied to the servomotor by four AA batteries. The flaps control is used to control the servomotor in and out of position with high accuracy.

It was found during testing of the servomotor that the stop block is easily moved in and out of position. To save battery life the servomotor control is turned off when moved into the safe and armed positions. A test of the safety block during misfire showed that the stop block easily held up to the force of the pneumatic system. This data is not completely accurate because the tests were performed at approximately 70psi in the pneumatic system. The tests were executed at the lower pressure due to low strength compressor. Given this information we are still confident that the safety block will easily withstand the system at the full 90 psi given in the design.

Weight Analysis

The design constraints for the Bomb Rack Unit state that the BRU must weigh less than 5 pounds, as such the BRU was designed to be light weight while remaining low cost. To achieve this, the top and one side panel were made out of Plexiglas, and a number of other parts were made of Delrin. However, a large portion of the BRU consists of aluminum. The pneumatic system accounts for the most weigh as many of the components consist of stainless steel, a high density material. Upon completion of assembly, the prototype was weighed and it was found to be 5.4 pounds. This exceeds the specified weight limit by 8%. While lightening holes could be drilled into various components to get the overall weight down below 5 pounds, it was decided that this could cause parts to fail and would not be beneficial. It is recommended that to achieve the weight goal, a higher budget would be needed so that composites such as carbon fiber could

be used. These composite materials have the strength needed to provide the same support given by the prototype BRU as well as have a much lower density compared to aluminum.

Ejection Velocity Analysis

Ejection velocity was an essential constraint that was incorporated into the overall design of the BRU. The goal was to achieve an initial velocity of at least 4ft/s of the payload. For in flight tests, the payload must be pushed away from the UAV to prevent any potential damage to the aircraft. The payload in our design was ejected using a piston like rod that integrated with the hook assembly.
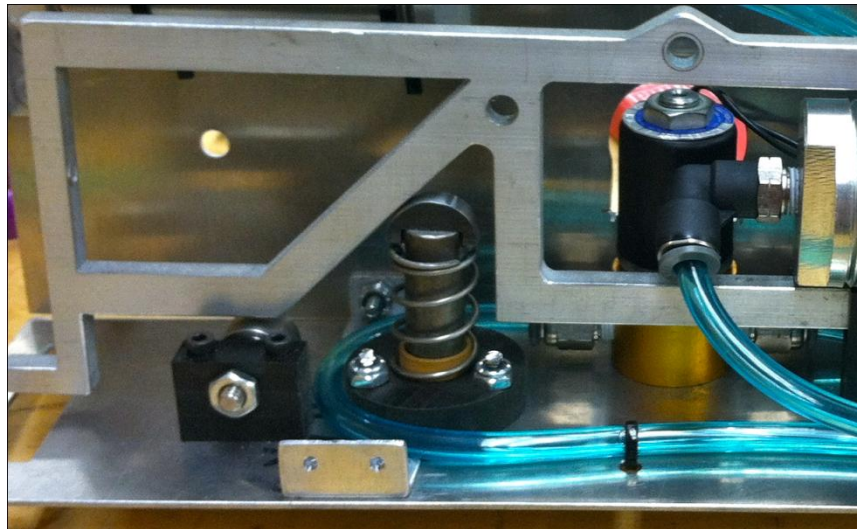
Figure 43- Ejector Piston

Using basic kinematic velocity analysis the initial ejection velocity was estimated. The equation below was used in this analysis. Video of the ejection was taken with the Apple iPhone which records high quality video at 24 frames per second. To break down the video, the program iMovie was used to break the video into time-constrained frames.

$$d = v_i t + \frac{1}{2} a t^2$$

d = distance traveled
v_i – initial velocity
t – time
a - acceleration

Three tests were run with the BRU pneumatic system filled between 60-70 psi for each test. The distance from the bottom of the payload to the bottom of the testing wing was measured to be approximately 30 inches. The time was calculated using iMovie, which in each case was approximately .8 sec from release to the bottom of the testing structure. Below are the calculated velocities for each of the three tests performed.

Velocity Test			
	V.Initial (ft/s)	V.Final (ft/s)	Time (s)
Test 1	3.50	8.33	0.30
Test 2	4.72	8.92	0.28
Test 3	4.72	8.92	0.28
Average	4.31	8.72	0.29

Table 5- Velocity Test Data

The ejection velocity of at least 4 ft/s was achieved using our piston ejector system design. To achieve higher velocities for high-speed applications, pyrotechnic ejection could be used to safely eject the payload. For the application of the Tigershark UAV, the ejection speed of 4ft/s is ideal for safe release of the provided payload.

Landing Shock and Lateral Load Analysis

As a requirement for the completion of the successful completion of this project, the BRU must be able to withstand a 1G landing shock and a 2G lateral load. A simple way to calculate the G forces felt by an object would be to know the distance travelled and the velocity right before the object is stopped. Experimentally finding these values allows an acceleration to be found by:

$$A_{stop} = \frac{V^2}{x}.$$

The value of A_{stop} found is then divided by the acceleration of gravity to give the G force felt by the object.

To implement this simple calculation a test rig would have to be constructed to safely test the BRU, making sure the system does not hit any objects or fall out of the harness during testing. Building this rig however is complex due to the ratio of velocity to distance needed to get the G force needed. This means that multiple tests and calculations have to be conducted to find the right distance for the system to drop using only gravity. Given this issue and problems found in other areas a system was not constructed to test the BRU against these forces.

An arbitrary test was done using human forces was conducted. Since human force was used to test the system, quantitative results of the tests were not found. Although we were not able to produce numerical results the BRU passed the tests without a problem. It was decided from these tests that the BRU would be able to withstand the landing shock and lateral loads.

Cost Analysis

For this design a budget of $2000 was made available to complete the project. This budget was established to purchase all of the raw materials, parts, and equipment needed to build and test the BRU design. At the end of the fall semester the raw materials and the servomotor were ordered. At the start of the spring semester the part ordering was completed and materials for the mock payload were purchased in town. Upon completion of machining, the payload was assembled and mock of the wing constructed. A compressor was then needed in order to fill the pneumatic air system for launch. Upon completion of the project the total cost, including testing equipment, came to $1304.55. This cost is well below the budget provided for the project, and a complete breakdown of the cost is shown in the Appendix on page 90. Based on the parts used strictly on the BRU, not including test equipment or unused components, the cost of the BRU came to only $688.74. This illustrates the lost cost measures taken during the design process. It should be noted that this value does not include the cost of machining, which was done at the FAMU/FSU College of Engineering machine shop. It is felt that this cost would be relatively high because of the high number of parts that had to be machined.

Problems and Solutions

During the initial assembly it was discovered that a small number of unforeseen issues had arisen. The first issue dealt with the pneumatic system. The original design called for a check valve to be used for inputting air. It was believed that the valve would be one way and that a typical air compressor could easily attach. As this was not the case, a Schrader valve was implemented to achieve the desired method of pressurizing the system. This valve is also able to easily release air to the atmosphere when desired; this removes the need for the very heavy exhaust valve. Also, a back pressure regulator was originally placed after the air tank to keep the air pressure at no more than 90psi. As the use of this device was not straight forward and could easily be turned to a lower pressure by accident the part was also scrapped. In order to check the pressure in the air tank now, a pressure gauge is used.

Figure 44- Filling the BRU using Schrader Valve

The largest issue observed upon completion of the assembly involved the ejector piston jamming in the hook when attempting to open. Attempts to solve this problem started with sanding the slider surfaces as smooth as possible in a attempt to reduce friction. This only slightly reduced the jamming. Next, the ejector spring, which returns the piston to its natural position, had a high spring constant. This spring was replaced with a newer spring which has a lower spring constant, lessening the vertical force seen on the hook. However, jamming still

occurred after these changes. The final solution involved a redesign of the hook itself. The original slot was designed at 45 degrees. This was believed to transfer the force input by the air cylinder directly into the ejector. However this did not create enough downward force to overcome the friction seen in the slot. The new hook uses a 30 degree slot. This reduces the length the ejector piston travels as well as the amount of force it will transfer to the payload, however the new slot provides enough downward force to overcome the friction and the piston no longer jams. Below shows the before and after hooks, notice the decreased angle associated with the lower hook.
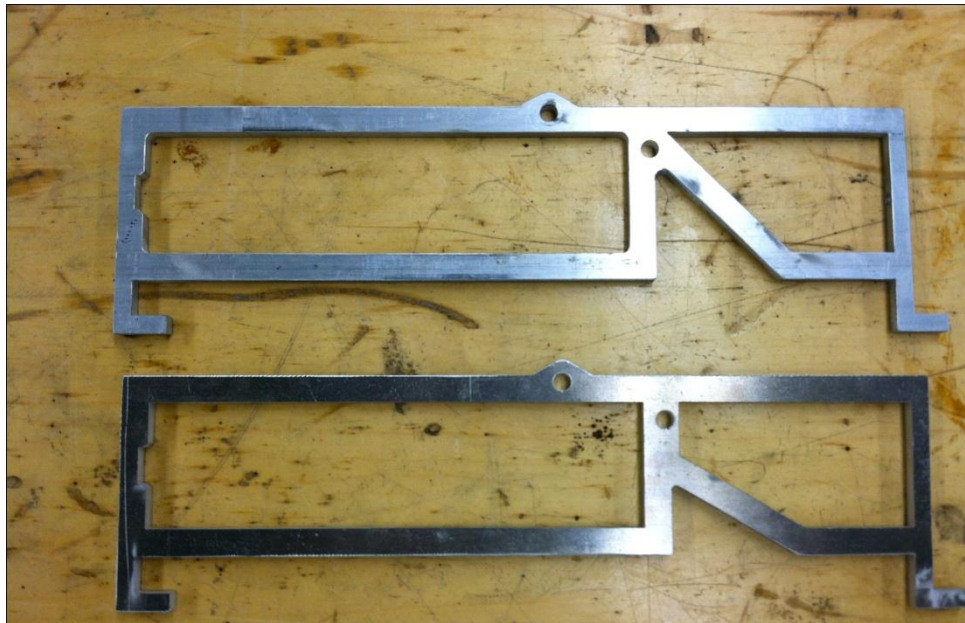

Figure 45- Reduced slot angle eliminates jamming

Another issue occurred with the sway brace pads. The outer diameter of the pads interfered with the hooks on the payload. By simply cutting the pads down to a smaller size provides enough clearance while still supporting the payload in flight.

Some other minor issues came up involving tolerances. Some holes were smaller than desired after going to the waterjet. These were simple to drill out to the correct size. Also the Stop Block had a tight fit, so this part needed to be filed down to remove any binding that occurred.

Finally, during assembly it was noticed that some of the nuts used to secure the side panels of the BRU were difficult to reach and easily get a wrench on. A solution for this would be to have the holes in the angle mounting brackets taped, that way bolts could simply screw the panels on without having to squeeze hands and tools into tight spaces.

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. During test firing, operators must take safety precautions as a heavy object (the payload) will be falling at a high rate of speed. Injury could occur if someone is in the path of the falling object.

Conclusion

The design and fabrication of the Miniature Bomb Rack Unit has been a success. The BRU was optimized based on weight, reliability, durability, size and cost. The assembled system has proven that it can hold onto a payload and eject it when the "Fire" command has been given. To test the prototype, a control panel was created with a switch to control the solenoid valve, LED lights to inform the operator of the position of the hooks, and an r/c controller to control the servo safety block. During testing the BRU was attached to a mock wing and pylon to demonstrate how the BRU would fit on the actual Tigershark UAV.

Fall semester calculations said that the payload would eject at a velocity of 5.33 ft/s. Testing found that the BRU actually ejected the payload at a velocity of 4.31 ft/s, which successfully exceeds the 4 ft/s requirement set by the AFRL. A reason for this decrease for theoretical to experiment is largely due to some issues faced with the ejector piston jamming in the hook slot. This was found to be caused by a large amount of friction between the two surfaces. This jamming issue was solved by redesigning the slot the piston traveled along. A better solution would be to redesign the ejector piston itself so that instead of two metal surfaces sliding against each other, a roller bearing would be used so as to further reduce friction. This was the biggest issue experienced during the assembly of the BRU.

Upon weighing of the BRU, it was found to be overweight by 8% at 5.4 pounds. This is due mainly to the heavy components associated with the pneumatics section as well as the use of aluminum for the majority of the system. A solution to decrease the weight for an actual BRU, not a prototype, it is recommended that the budget be slightly increased to allow for the purchase of composite materials, for example carbon fiber. This would give the BRU the same rigidity the prototype currently has yet reduce the overall weight of the system.

The 1G landing shock and 2G lateral load were tested, but empirical data could not be recorded; however, we do feel that our BRU and payload meet these design specifications. The total spent on this project was $1304.55; however the final assembly of the BRU only cost $688.74. Overall the project was a success, meeting the overall requirements provided by the
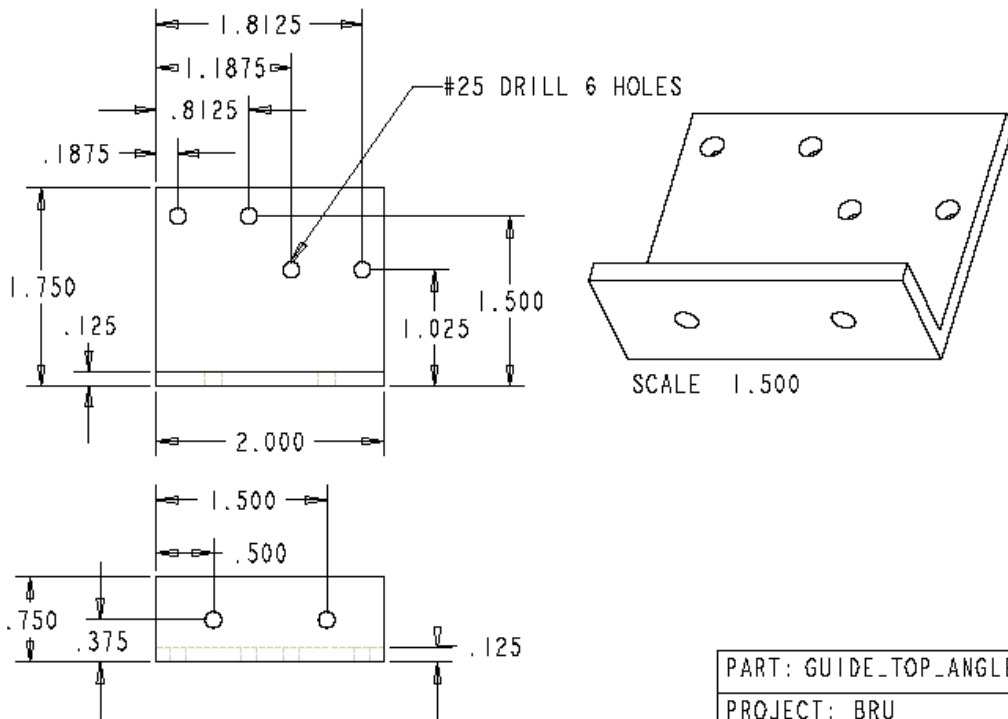
customer. A great deal was learned about the design process, and we were able to apply the knowledge learned from previous classes to design and build a working prototype of a Miniature Bomb Rack Unit.
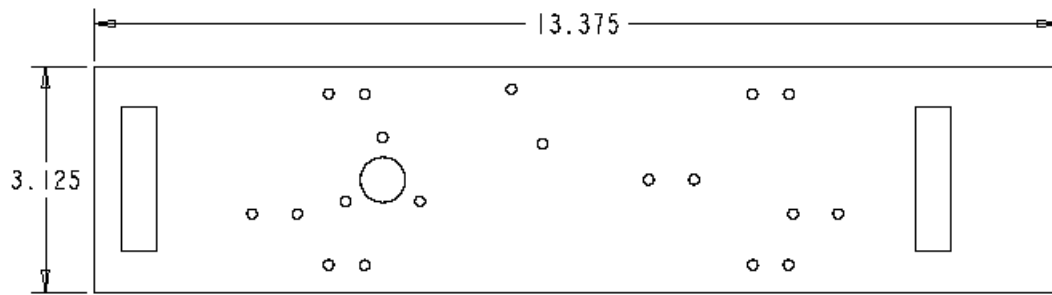
Acknowledgements

We would like to give acknowledgement to Dr. Shih, Dr. Kosaraju, and Dr. Dalban-Canassy for giving guidance on reports and presentations. We would also like to give thanks to Mr. Russell Roberts and Eglin Air Force Research Lab for creating this project and working with us on the design constraints. Finally, we would like to pass acknowledgement to Dr. Clark for helping us when we needed it and allowing us to use some of his equipment. Special thanks is also given to Jeremy Phillips and the machine shop for machining our parts and supplying the tools needed to assemble and make necessary modifications to our design.

Appendix

Part Drawings


```
#25 DRILL 6 HOLES

1.8125
1.1875
.8125
.1875
1.750
.125
1.025
1.500
2.000

1.500
.500
.750
.375
.125

SCALE  1.500
```

PART: GUIDE_TOP_ANGLE	
PROJECT: BRU	
DREW BY: SR DESIGN 3	
QTY 2	
PART #: 001	
REV: 1	SHT: 1 OF 1

13.375

3.125

SCALE 0.625

WATERJET PART
THICKNESS: 1/16 INCH

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

R.125

.50

.75

.50

1.00

.810

.190

.375

#25 DRILL 2 HOLES
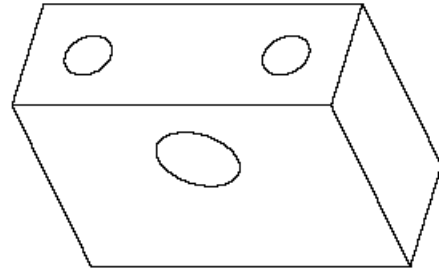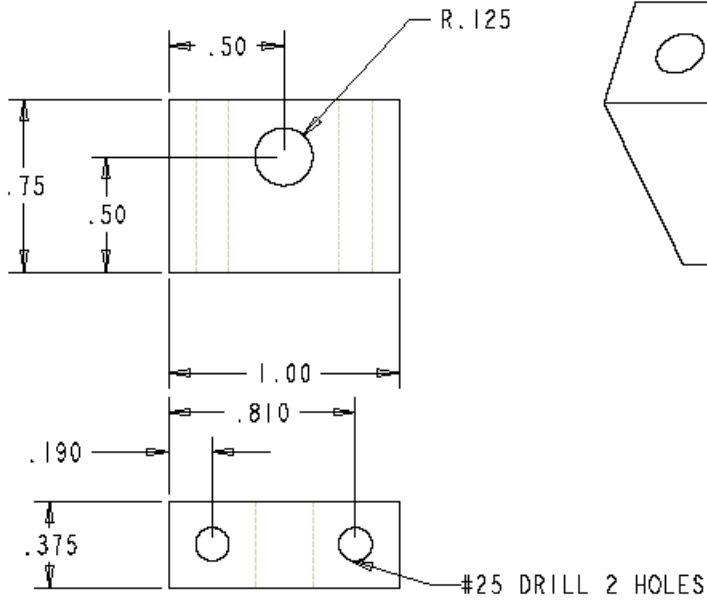
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MATERIAL: DELRIN

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PROJECT: BRU	
DREW BY: SR DESIGN 3	
QTY 4	
PART #: 003	
REV: 1	SHT: 1 OF 1

SCALE 2.000

.109

1.250

.675

.488

1.00

.1875

#25 THRU
Ø0.25
0.1875

.500

.250

.8125
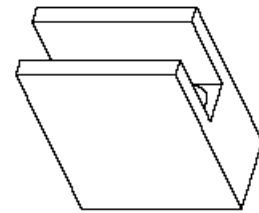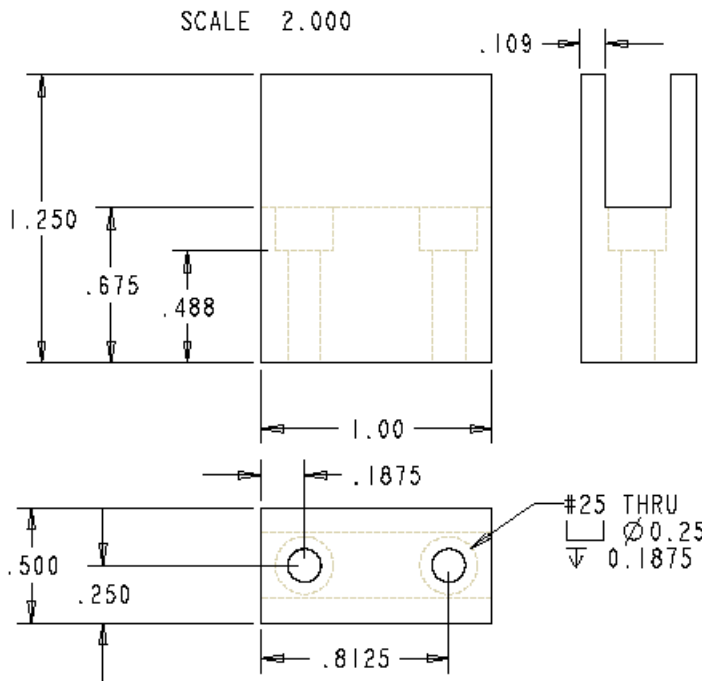
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PART #: 004	
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.50

1.00

.0625

.0625

SCALE 1.500

.50

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.250

.750
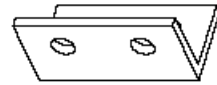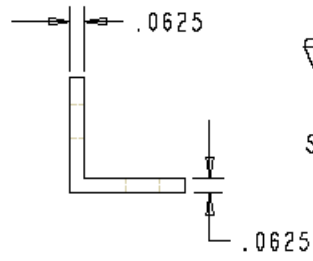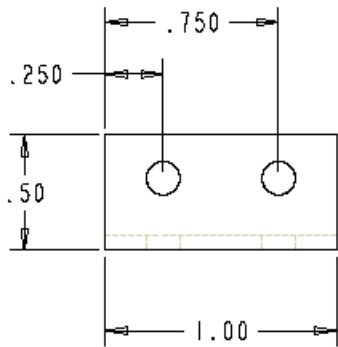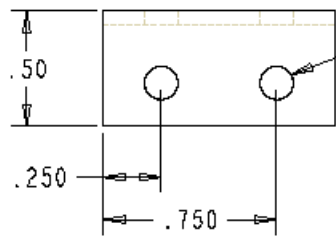
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PART #: 005	
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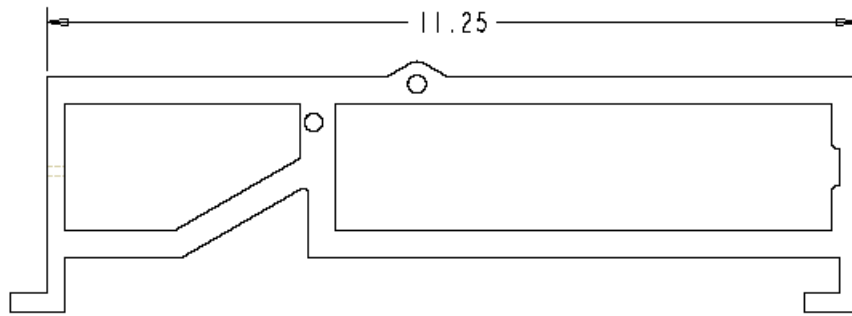
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WATERJET PART

1/4 INCH THICK

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DATE: 12/07/11	
PART #: 006	
REV: 0	SHT: 1 OF 1

.125

.125

1.00

.375

2.50

1.50

.50

3.00

#25 DRILL 3 HOLES

2.00

1.50

.50

2.125

.875

∅0.625

PART: CYLINDER_MOUNT	
PROJECT: BRU	
DREW BY: SR DESIGN	
QTY 1	
PART #: 007	
REV: 1	SHT: 1 OF 1

#25 DRILL 8 HOLES

.80
.648
.352
.20
.0625
00
1.852
1.71
.81
.179
.188
.32
.0625
.250
1.00
.625
1.00

SCALE 1.500

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PROJECT: BRU	
DREW BY: SR DESIGN 3	
QTY I	
PART #: 008	
REV: 0	SHT: I OF I

.0625

.250

.125

.44

1.315

.815

#30 2 HOLES

.25

.6625

1.125
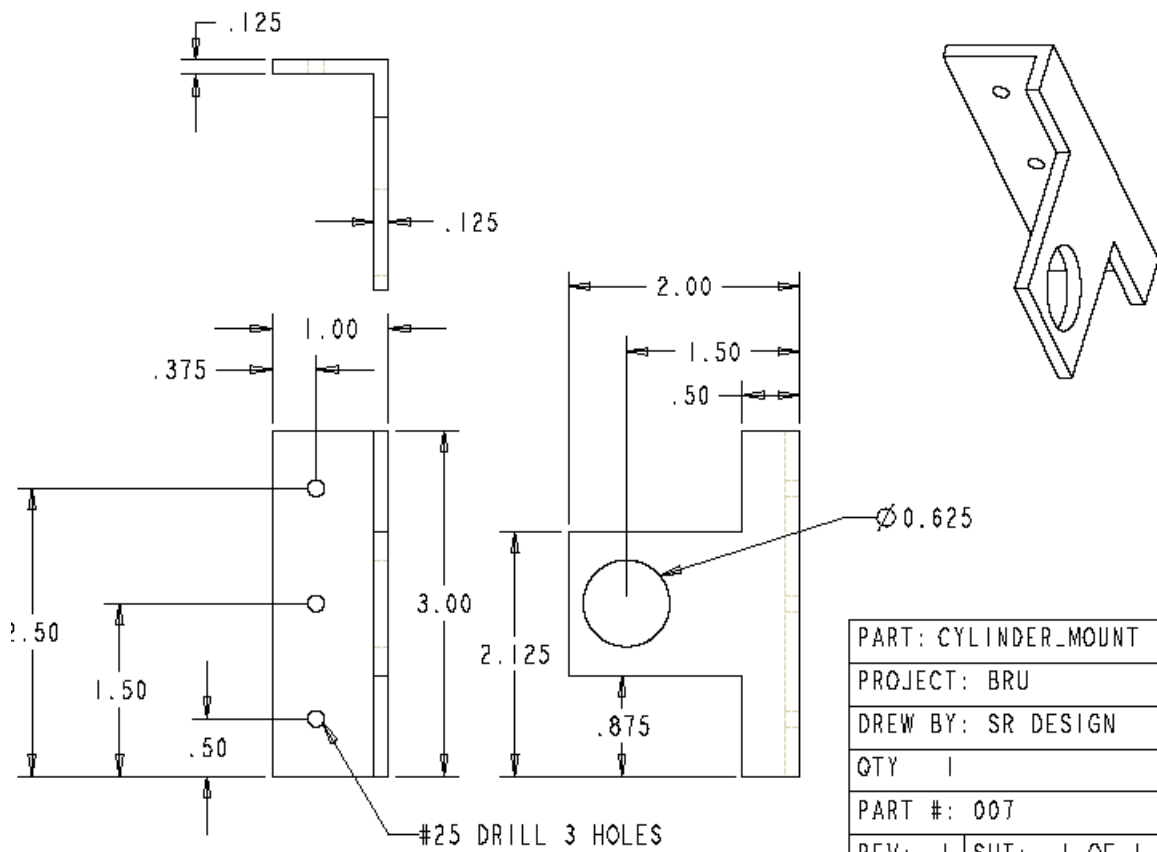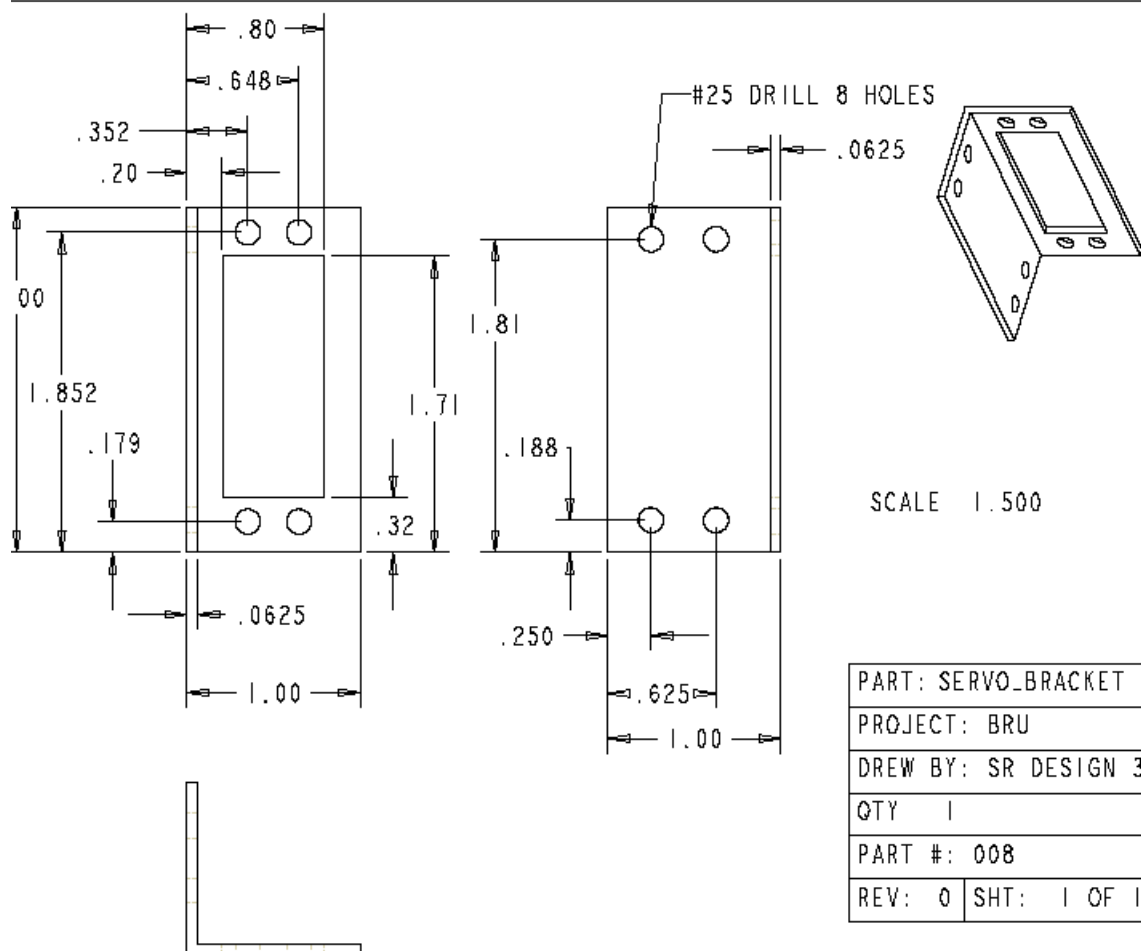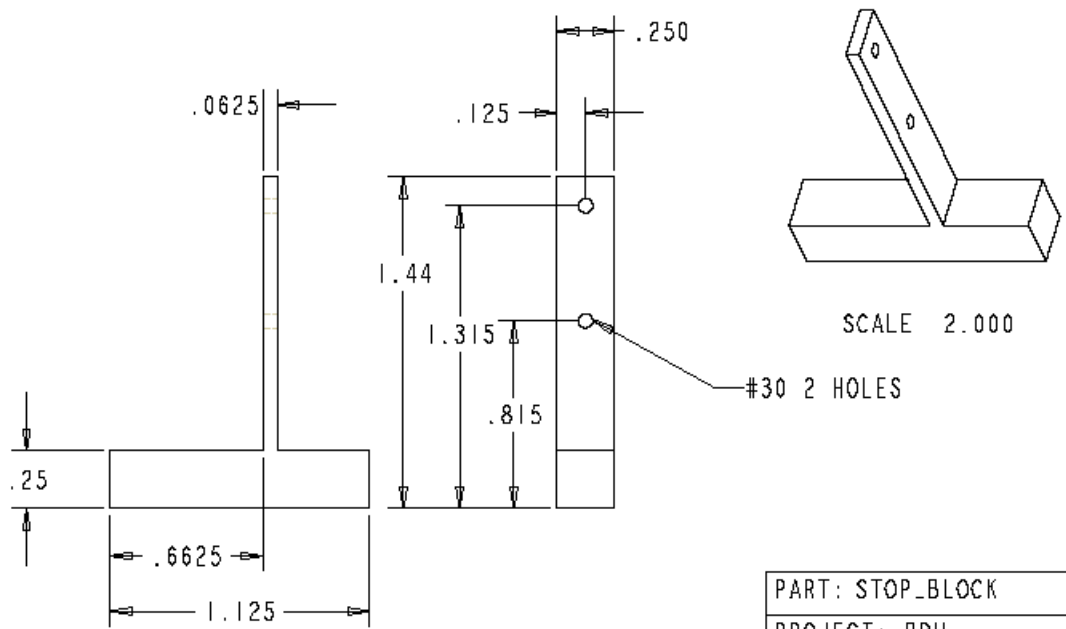
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PROJECT: BRU	
DREW BY: SR DESIGN	
QTY 1	
PART #: 009	
REV: 1	SHT: 1 OF 1

SCALE 2.000

Ø19/32
⌴Ø7/8
▽0.125

R.750

Ø.201 THRU
3 HOLES
120 DEGREE INTERVAL

.59

.251
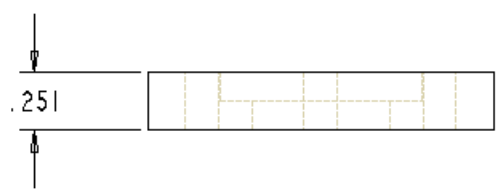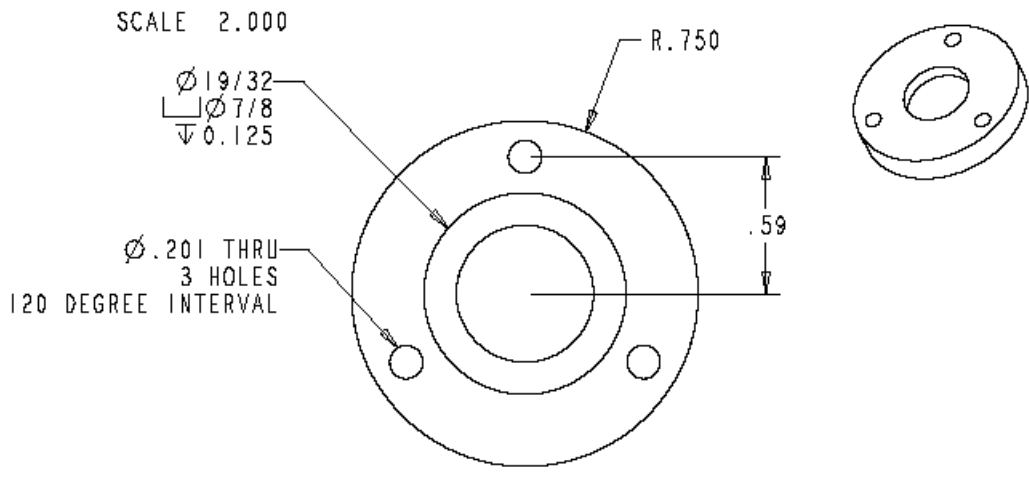
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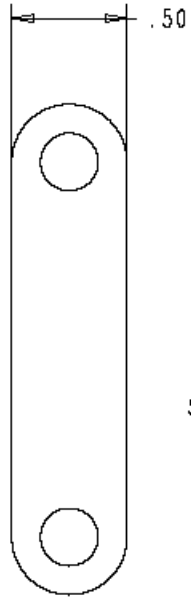
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QUANTITY: 2

SCALE 2.000

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PROJECT: BRU	
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.375

.25
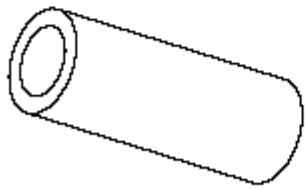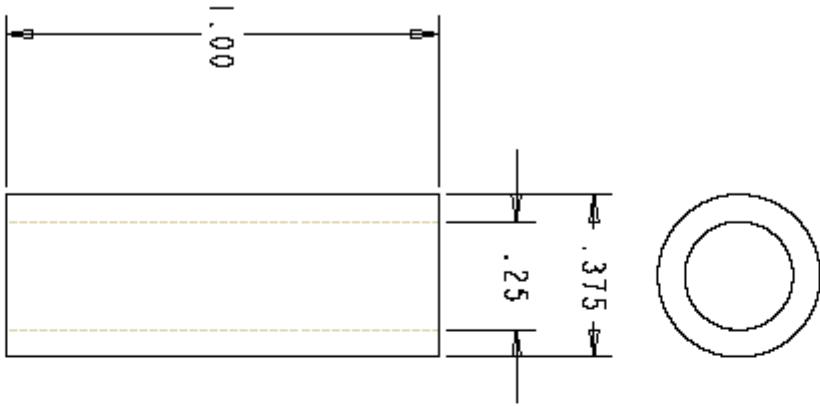
SCALE 2.000

PART: SPACER_INNER

PROJECT: BRU

DREW BY: SR DESIGN 3

DATE: 12/07/11

PART #: 012

REV: 0   SHT: 1 OF 1

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.375

.25

.8750
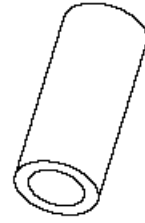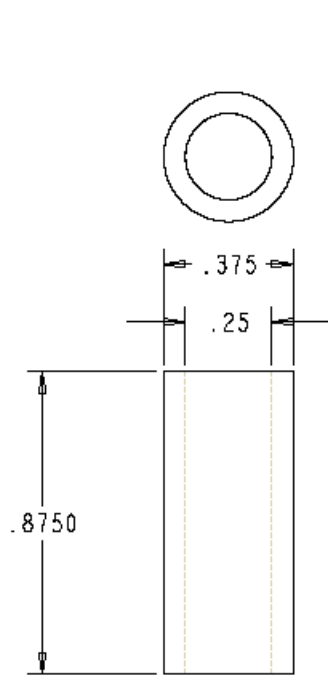
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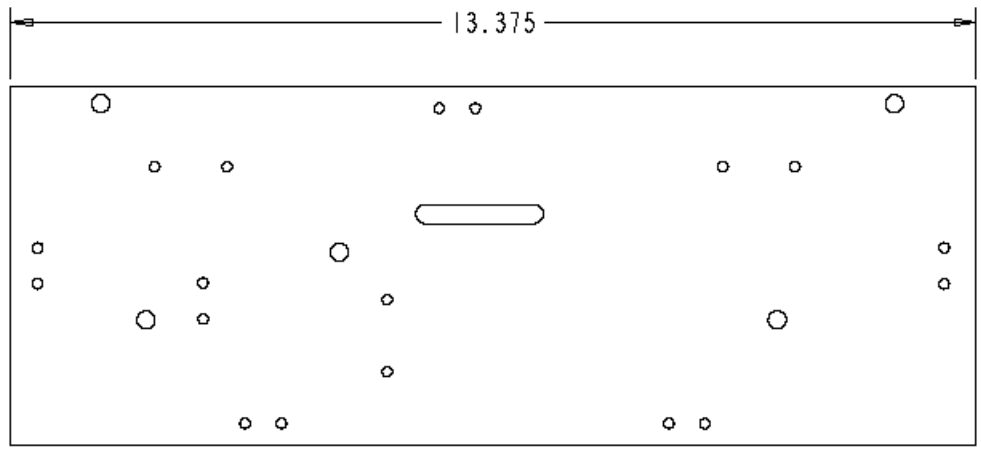
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WATERJET PART
THICKNESS: 1/16 INCH

PART: CASING_FRONT	
PROJECT: BRU	
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PART #: 014	
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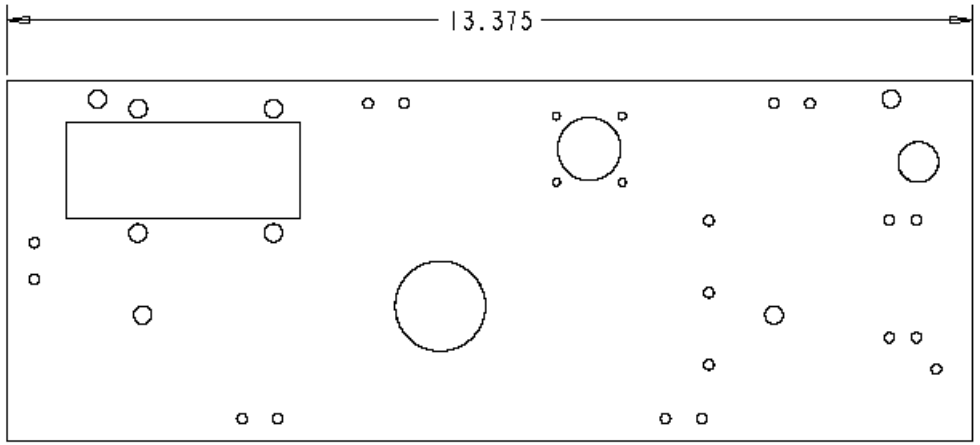
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3.00

5.00
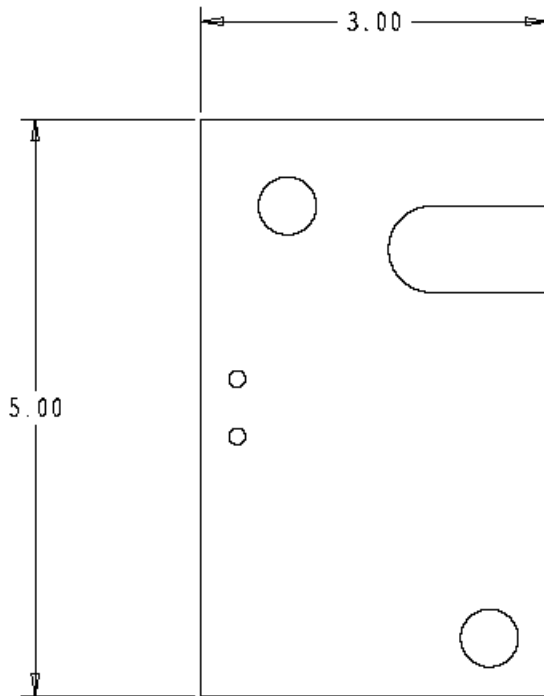
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WATERJET PART

1/8 INCH THICK

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REV: 0	SHT: 1 OF 1

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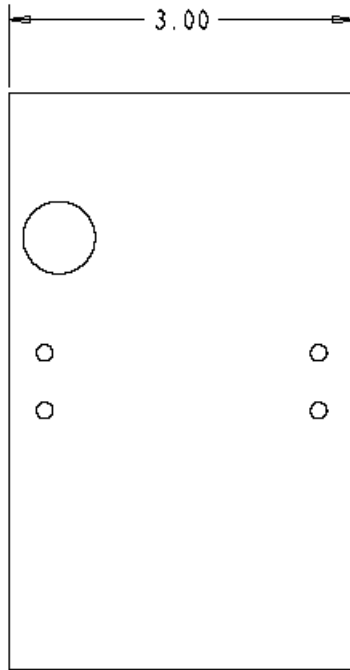
LASER CUT PART

SCALE 1.000

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DATE: 12/07/11	
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.25

.50

1.38

130.0°

R.50

R.13

1.00

Rx.20
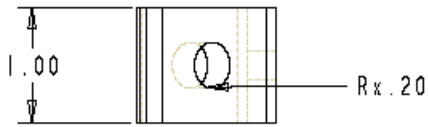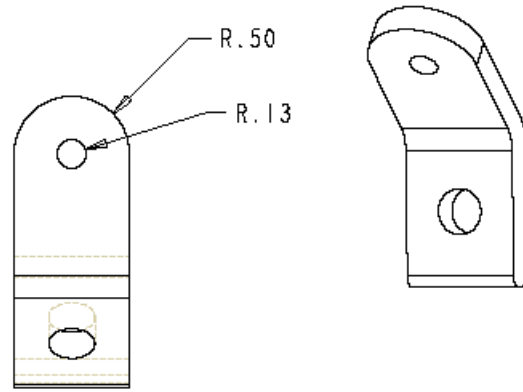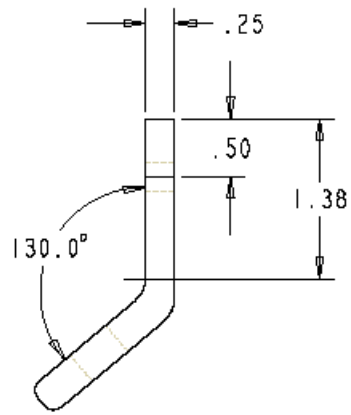
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DATE:	12/07/11
PART #:	018
REV: 0	SHT: 1 OF 1

Bill of Materials

Raw Materials				
	Material	**Size**	**Vendor**	**Price**
Angle	Al6061	2x2x.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	12x24x.0625 in	onlinemetals	$14.90
	Acetal	12x12x.5 in	onlinemetals	$40.78
	Polycarb	24x24x0.06in	onlinemetals	$11.00
Round	Acetal	.375x24in	onlinemetals	$3.48
	Steel 1018	.5x<12in	onlinemetals	$2.34
Total				**$102.54**

BRU Cost Breakdown					
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
Solenoid	cylval	SA31NC	$40.80	1	$40.80
Schrader Valve	McMaster	8063K38	$4.43	1	$4.43
Guide_roller	Grainger	1ZGT7	$48.25	4	$193.00
Ejector_bushing	McMaster	6377K114	$20.27	2	$40.54
Ejector_spring	Home Depot				$5.37
Servo	Futaba	FUTM0513	$125	1	$124.98
Limit Switch	McMaster	7090K37	$7.91	1	$7.91
RBF PIN	McMaster	90293A139	$17.98	1	$17.98
Nuts/Bolts	Various				$20.00
Leveling Foot	McMaster	2531k61	$7.30	4	$29.20
air connector	omega	OM-AIR-C24250418-5PK	$5.50	2	$11.00
air hose	omega	TYUTH95-1418-50-TRBL	$21.00	1	$21.00
female connector	omega	OM-AIR-C24260418-5PK	$11.50	1	$11.50
90deg elbow	omega	OM-AIR-C24470418-5PK	$8.25	1	$8.25
Raw Materials	onlinemetals				$102.54
Total					**$688.74**

Overall Cost Breakdown					
Part	**Vender**	**Part Number**	**Price**	**Quantity**	**Total Price**
Check Valve	McMaster	3208K22	$14.74	1	$14.74
Regulator	McMaster	99045K48	$34.80	1	$34.80
Exhaust Valve	McMaster	4149T41	$35.55	1	$35.55
Pushspring	McMaster	9657K48	$6.29	1	$6.29
Extra Nuts/Bolts	Various				$20.00
Electrical Material	RadioShack				$21.77
Test Rig Materials	Home Depot				$75.00
Remote Control	Hobby Town				$100.00
Air Hose & Tooling	Wal-Mart				$15.37
Air Compressor	McMaster	9965K62	$189.75	1	$189.75
BRU Cost					$688.74
Raw Material	onlinemetals				$102.54
Total					**$1,304.55**

Calculations

Pneumatic Calculations

Volume in Air Cylinder

$\text{Bore} := 1.5 \text{in}$

$\text{stroke} := 1 \text{in}$

$\text{Volume}_{cyl} := \pi \cdot \left(\dfrac{\text{Bore}}{2} \right)^2 \cdot \text{stroke} = 1.767 \text{in}^3$

Air Tank Safety Factor: $\quad SF := 2$

$\text{Volume}_{tank} := SF \cdot \text{Volume}_{cyl} = 3.534 \text{in}^3 \qquad$ 4 cubic inch air tank needed

Servo motor Torque

$\tau := 76 \text{oz} \cdot \text{in} = 4.75 \text{lb} \cdot \text{in}$

$m := 0.0086674 \text{lb} \qquad\qquad cg := 0.27216 \text{in}$

$L := 1.41 \text{in} \qquad\qquad\qquad \text{Midpoint} := 0.375 \text{in}$

$L_{torque} := L - cg - \text{Midpoint}$

$\qquad\qquad\qquad\qquad\qquad\qquad L_{torque} = 0.763 \text{in}$

$\tau_{required} := m \cdot L_{torque}$

$\qquad\qquad\qquad\qquad\qquad\qquad \tau_{required} = 6.612 \times 10^{-3} \cdot \text{lb} \cdot \text{in}$

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

$$d := 0.25in$$

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

Bolt will be in double shear

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

Average shear stress within the bolt

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

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

$$U_{min} := 60000psi$$

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

$$\boxed{\tau = 4.214MPa}$$

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

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

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

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

AL-6061 specifications

$\tau_{allow} := 78.5 MPa$

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

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

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:

$t := \dfrac{3}{32} in$

Normal stress from weight and landing load

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

$A := w \cdot t$

$\sigma_n := \dfrac{F}{A} = 2.942 MPa$

Shear stress

$V := F = 133.447 N$

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

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

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

Bending stress from lateral force

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

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

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

Bending stress from wind force

BRU front dimensions set by design:

w := 4in

h := 8in

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

$$a := w \cdot h = 0.222 \text{ft}^2$$

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

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

$$p := 14.4 \text{psf}$$

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

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

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

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

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

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

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

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

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

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

AL-6061 specifications

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

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

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

$$117\text{MPa} = 16.969\text{ksi}$$

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

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

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.

AL-6061 specifications

$$\sigma_{ult} := 117 MPa \qquad\qquad \sigma_{perTab} = 64.014 MPa$$

$$\sigma_{eighthInch} := 60.242 MPa \qquad\qquad 117 MPa = 16.969 ksi$$

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

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

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.

Mass of hook: m1

$m1 := .27lb$

Mass of piston: m2

$m2 := .104lb$

Mass of payload: mpayload

$m_{payload} := 10lb$

Force from pneumatic cylinder: Fair

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

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

Hook spring rate: K1

Hook travel distance: X1

Spring preload distance: X2

$X1 := 1in$

$X2 := 0$

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

Piston spring rate: K2

Spring preload distance: X3

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

$m2g = 0.463N$

$X3 := 0$

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

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

$$Y := \begin{pmatrix} F_{air} - FS1 \\ m2g - FS2 \end{pmatrix}$$

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

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

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

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

Time the piston takes to reach the payload

Distance between the piston and payload: X4

$$X4 := .14\text{in}$$

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

Velocity of the piston when it contacts the payload:

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

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

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

$$X5 := 1\text{in} - X4 = 0.86 \text{in}$$

$$X5 = 0.022\text{m}$$

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

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

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

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Team Biography

Bryan Rickards

I am a senior mechanical engineering student at Florida State University. I graduated from Cocoa High School as well as Brevard Community College with an Associates of Arts Degree in spring 2008. I became fascinated with engineering by competing in the F.I.R.S.T. Robotics competition during my time at Cocoa High. F.I.R.S.T. teams partner with professional engineers to design, prototype, build, and test a competition robot in just a six week period and compete around the country. The design aspect of the engineering process is my strong suit and I have over seven years experience with 3D CAD modeling software. This senior design project has been a great experience for me and I am excited to take my skills into the next stage of my life by entering the professional engineering world.

Casey Brown

I am a senior in my fifth year of Mechanical Engineering at Florida State University. I was born and grew up here in Tallahassee. I achieved my A.A. degree from TCC in 2009. When I complete my degree I will also have a minor in Math and a specialization in the area of Thermal Fluids. In this last semester I have undertaken research in Computational Fluid Dynamics, CFD, under Dr. Taira. I also have held a job at a local recycling company since the spring of 2008. Upon the completion of my B.S. degree I plan to join the Air Force.

Cyril John

I was born on April 3rd 1989, in Yonkers, NY. I moved to Tampa when I was five, which I spent the next twelve years. I attended Middleton High School, which is a magnet school for engineering. This point of my life I began having a serious interest in pursuing my education in an engineering field. After high school I started my collegiate education at Florida State University pursuing a degree in Mechanical Engineering with a specialization in Thermal Fluids. My career goal is to pursue a career in engine design, preferably in the aeronautical field.