ME 450: Design and Manufacturing III

Design of a Mechanical Impacter for Testing Cable Wave Propagation

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TEAM 13

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ABSTRACT

The United States Navy wants to know the tension in oceanographic cables submerged to depths of up to 10,000 ft. Professor Noel Perkins, with the University of Michigan, has devised a method to estimate cable tension by measuring the speed at which impact-induced waves propagate through a cable. Our objective is to work with our sponsors to design and construct a mechanical impacter prototype to use with the new testing method. Additionally, the impacter must interface with a submersible robot arm. Our design will be tested on an existing cable test bed on the University of Michigan campus.

INTRODUCTION

Oceanographic cables are buried in shallow trenches on the sea floor to protect them during their long service life. A key metric in determining the lifetime of these cables is the tension in the cable. Currently, the US Navy is looking for a way to measure these tensions using an unmanned submersible robot. Professor Perkins' laboratory has devised a system for measuring the tension in these cables by inducing a wave in the cable and recording the speed at which it travels. A mechanical impacter has been designed to induce a consistent wave in the cable, however this design is not suitable for use by a remote manipulator arm. We are tasked with designing and constructing an impacter that can be used by this robotic arm.

To accomplish the task of testing the tension of said oceanographic cables, we will send a submersible robot to the sea bed where the cable is located. The robot will clear a small trench in order to fully expose the cable. It will attach two accelerometers a known distance apart from one another on the cable and finally, it will use the impacter to strike the cable, causing a fundamental wave to propagate. Using the data acquired by the accelerometers, it is possible to calculate the tension in the cable within five percent.

In order to ensure Professor Perkins' calculations for tension in the cables are accurate, we have to provide a propagating fundamental wave through the cable with the correct spectral content. The spectral content refers to the range of frequencies of a set of waves that propagate from an impact.

INFORMATION SEARCH

Our sponsor has provided us with a task as well as many technical constraints to facilitate and guide the design process. Our information search has consisted of research and analysis of previously successful designs and detailed discussion with our sponsors and contacts.

Technical Benchmarks

Past impacter models have included a wooden 4x4 and a purely mechanical impacter designed for use in land base testing. While these designs were effective in a laboratory setting, their use in the field would prove to be highly impractical. The 4x4 imparts inconsistent impact magnitudes with highly consistent spectral content while the mechanical impacter provides very consistent impact magnitudes with poor spectral content. Hyung Min Chae, a graduate student working under Professor Perkins in the past, performed and documented quantitative studies on the forces and impacts qualities that our design must impart.

Contacts

Our task information was obtained from our sponsors, Professor Noel Perkins and graduate student Tom Waisanen, in conjunction with Karen Miller and Steve Karnofski of the US Navy. Information regarding the device's manipulation by the submersible robot was obtained from Greg Cooper, an R.O.V. expert with the Naval Facilities Engineering Service Center (NFESC).

The NFESC has provided us with detailed specifications as well as operating manuals and drawings for the submersible robot. Additionally, we have access to the robotic manipulators that will operate the impacter prototype and more recently, the NFESC has sent us a fully operational robot arm. From these materials we have gained qualitative and quantitative knowledge regarding the robot's limitations. From these samples we have dimensioned the manipulators accurately and have formulated design criteria for our final prototype.

CUSTOMER REQUIREMENTS AND ENGINEERING SPECIFICATIONS

Our sponsors, Professor Noel Perkins and the US Navy, have requested that we design and fabricate a working cable impacter. They have dictated specific design constraints that we have used to compile a list of customer requirements.

Robotic operation. A robotic submersible vehicle will operate our device, so our final design must provide ease of use in addition to functionality. Our contacts at the US Navy have specified that the use of large T-shaped handles will help facilitate operation.

Underwater operation (corrosion, pressure, water). The prototype will be operated in the ocean at depths up to 10,000 feet. As such, the design must be impervious to the pressures, corroding elements, and other factors present in the working environment through physical isolation and material resistance to these factors.

Impact properties. Professor Perkins' testing method dictates that our device must impart a consistent, repeatable mechanical impact. Specifically, testing requires that the impact is of explicit magnitude and induces a fundamental wave in the target cable in a specific frequency range. Further, the nature of the testing procedure necessitates repeatable, identical impacts over the course of a single dive.

Simplicity. Due to the eventual robotic operation of our device, simplicity is imperative. The prototype must be compact and light enough for the robot to operate effectively. Additionally, it must not require complex movements to operate.

Operates on 24 V DC at 2 A. A power supply line will be available for use and if our prototype requires electrical power, it should accept the available voltage, current, and mode without major regulation.

Table 1: Customer requirements and their ranking and weights

Customer Requirements	Ranking	Weight
Can be Operated Robotically	1	17
Operable in Salt Water	3	11
Pressure Resistant to 10,000 ft	2	15
Provides Repeatable Impact	5	9
Resistant to Corrosion	8	6
Provides Specified Impact Quality	5	9
Can Provide Multiple Impacts per Dive	5	9
Provides Specified Impact Magnitude	3	11
Simple Design	11	2
Compact Size	10	3
Lightweight	11	2
Can Operate on 2 A at 24V DC	8	6

In order to evaluate how well our design meets our sponsors' needs, it is imperative to quantify the set requirements with conventional engineering metrics and projected values for said metrics. Using data from successful experiments and the prototype operating conditions, we have established the following engineering specifications.

Past experimentation. Professor Perkins has analyzed his testing method in the past and in doing so, has established ideal impact magnitudes, vibration frequency (spectral content) ranges, and contact times to optimize accuracy. Impact magnitudes of approximately 30 N have been ideal, as this impact is large enough to register accurate data on the accelerometers and small enough such that this reading is not oversaturated. Vibration frequencies from 5 to 25 Hz have been most successful in testing because they excite a fundamental structural wave in the cables which yield the most accurate results. Our device must also meet these specifications up to ten times over the course of a single dive.

Prototype operating conditions. Other engineering specifications are derived from the conditions present in the device's eventual operation. The operating environment is 10,000 feet under seawater, so the design must resist pressure differences up to 35 MPa. Additionally, the capabilities of the robot dictate a maximum external dimension of 500 mm and a maximum effective weight of 15 N. Further, the robot is limited to 5 degrees of freedom and a maximum range of motion of about 1 meter in any given direction.

Table 2: Engineering specifications and their projected values

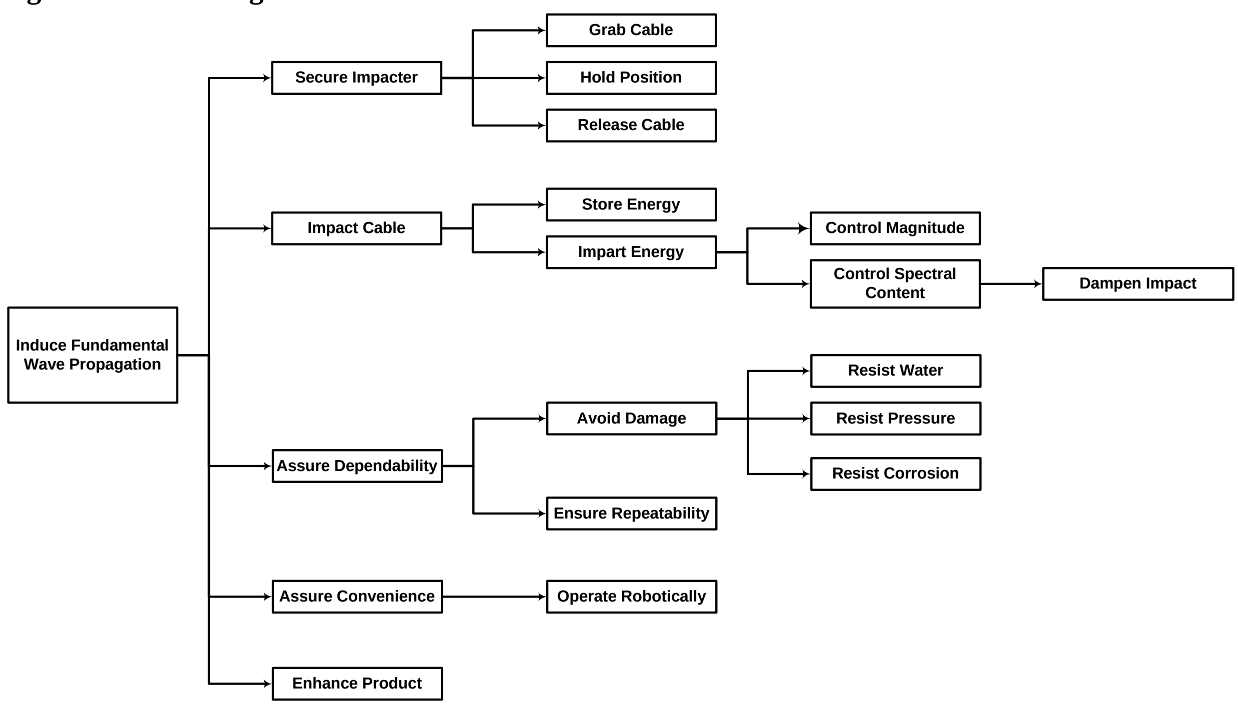
Engineering Specifications	Projected Value	Units
Impact Magnitude	30	N
Vibration Frequency Induced	5—25	Hz
Time of Contact	0—0.01	s
Impacter Outer Dimensions	0.5	m
Impacter Weight	15	N
Maximum External Pressure	35	MPa
Number of Impacts per Dive	10	#
Manipulator Degrees of Freedom	5	#
Manipulator Range of Motion	1	m

Finally, in order to compare important customer requirements with the respective engineering specifications, a Quality Function Deployment was utilized (Appendix A). The most important specifications to meet are induced vibration frequency, impact magnitude, and impacter outer dimensions.

CONCEPT GENERATION

In order to formulate viable conceptual designs, we dissected the required functions of our final design using a Function Analysis System Technique (FAST) Diagram. Our analysis led us to design our impacter in two modular parts: a clamping device and an impact generating device. These two functions are easily isolated because there is little relevant interaction between each module. The basic function, securing the impacter, is handled entirely by the clamp module of our design, and the basic function, impact cable, is handled by the impacter module. Functions related to system robustness include assuring dependability, assuring convenience and enhancing the original product. These functions were addressed in the design of both the clamp and the impacter modules.

Figure 1: FAST Diagram



Clamping Function

The design of our clamping system focused on assuring convenience for the robot operator as well as overall dependability. Our Morphological chart helped us to identify the main functions of the clamp and to design for optimization. Because the gripper can exert far more force than a human hand, but has less ability to articulate itself, we designed with this in mind.

Figure 2: Morphological chart for clamping subfunctions

	"Hook" Carabiner	Regular Carabiner	"Clothespin"	"Squeeze Hook"
Grab Cable	Attaches with Simple Motion	Attaches with Simple Motion	Squeezes with Robotic Arm, Places and Releases	Squeezes with Robotic Arm, Places and Releases
Hold Position	Depends on K of Spring, Cannot Hold Under Large Loads	Depends on Size of Carabiner, Cannot Hold Tightly	Depends on K of Spring, Cannot Hold Under Large Loads	Will Hold Position Properly Regardless of Load
Release Cable	Simple with a Low K	Requires Some Maneuvering From Robot Operator	Requires Little Effort from Robot Operator	Requires Little Effort from Robot Operator

The other important sub-function of the clamp is its ability to transmit an impact from the impacter to the cable. In order to do this, it must have a solid grip on the cable so as not to be pulled off. It also must be able to remain in constant contact with the cable so that rattling does not affect the spectral content of the impact. The clamp may also be utilized to ensure the correct spectral content by coating it with a damping material.

Table 3: Conceptual designs for cable clamp design

Design	Description
Hook Carabiner	This design clips around the wire, creating a solid contact, but placing and removing it is more difficult with the use of stronger springs, which are better for preventing it from moving during impact.
Regular Carabiner	This design easily attaches to the wire, but is prone to “rattling” since it does not tightly grip the wire, but sits around it.
Clothespin	This design is attached like a simple clothespin, but can be pulled off if too much force is exerted.
Squeeze Hook	This design is somewhat more difficult to place on the cable, but is much better suited to handling impacts, as it is nearly impossible to shift or pull off during impact.

Impact Generating Function

Using our Morphological chart, we were able to identify several ways of accomplishing the sub-functions of creating an impact. Because of lessons learned in past experimentation, we were able to address some of the sub-functions (most notably the damping) in ways we know will be successful. As a result there is not much differentiation across the designs for these elements. There was however, a wide range of variation in the area of power generation and conversion to create the impact.

Our concepts for the impacter were generally separated by their power source. Because of the limited resources available, most designs employed electromechanical systems or purely mechanical systems

Figure 3: Morphological chart

	Torsion Spring	Solenoid	Pinball Plunger	Linear Spring with Squeeze Charge	Linear Actuator	Cartridge	Guitar Pick	Flywheel	Gas Powered	Pneumatic	Rotary Hammer	Piezo Electric
Store Energy	Loading the Torsion Spring by the Rotary Motor	Electrical Energy is Drawn Upon for Impact	Charging the Spring by Pulling the Plunger with the Robot	Squeezing the Impacter, Which in Turn Charges the Linear Spring	Electrical Energy is Drawn Upon for Impact	Energy is Stored Chemically in an Explosive Charge	Pulling the Device and Deflecting the Restraining Pick	Speeding up a Flywheel with an Electric Motor	Energy is Stored by Means of a Pressure Differential	Pneumatic Energy is Drawn Upon from the Robot	Electrical Energy is Drawn Upon to Power the Motor	Electrical Energy is Drawn Upon for Impact
Control Magnitude	Changing the Distance the Rotary Motor Travels Before the Clutch Disengages	Changing the Distance the Plunger Travels or Changing the Current or Voltage	Changing the Distance the Robot Arm Pulls the Plunger	Changing the Amount of Charging done by Robot	Changing the Distance the Mass Travels or Changing the Current or Voltage	Altering the Amount of Explosive Used in the Cartridge	Adjusting the Mounting of the Pick, and the Material of the Pick	Altering the Rotational Velocity of the Flywheel	Magnitude is Controlled by the ratio of Pressure Stored to Depth	Applying a Different Amount of Pneumatic Pressure	Adjusting the Speed and Mass of the Hammer	Magnitude is Constant and Based on Size of Piezo Electric Motor
Dampen Impact	Placing a Damping Material at the Point of Impact	Placing a Damping Material at the Point of Impact or Around Grippers	Placing a Damping Material at the Point of Impact or Around Grippers	Placing a Damping Material at the Point of Impact	Placing a Damping Material at the Point of Impact	Placing a Damping Material at the Point of Impact	Changing the Material of the Pick	Placing a Damping Material at the Point of Impact	Placing a Damping Material at the Point of Impact	Placing a Damping Material at the Point of Impact	Placing a Damping Material at the Point of Impact	Placing a Damping Material at the Point of Impact
Resist Water	Placing Device in a Pressure Vessel	Placing Device in a Pressure Vessel	Using Water Reisistant Materials	Using Water Resistant Materials	Placing Device in a Pressure Vessel	Placing Device in a Pressure Vessel	Using Water Resistant Material	Using Water Resistant Materials	Using Water Resistant Materials	Using Water Resistant Materials	Using Water Resistant Materials	Placing Device in a Pressure Vessel
Resist Pressure	Placing Device in a Pressure Vessel	Placing Device in a Pressure Vessel	No Pressure Sensitive Elements	No Pressure Sensitive Elements	Placing Device in a Pressure Vessel	Placing Device in a Pressure Vessel	No Pressure Sensitive Elements	No Pressure Sensitive Elements	Device Itself is a Pressure Vessel	No Pressure Sensitive Elements	No Pressure Sensitive Elements	Placing Device in a Pressure Vessel
Resist Corrosion	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials	Using Corrosion Resistant Materials
Ensure Repeatability	Having the Clutch Disengage at the Same Point Every Time	Keeping a Consistent Electrical Charge, and Distance for Plunger to Travel	Pulling Plunger the Same Distance Each Time Before Release	Charging to the Same Point Before Releasing Trigger Each Time	Having the Clutch Disengage at the Same Point Every Time	Using a consistent amount of explosive in each cartridge	Repeatability Cannot be Ensured	Modulating Power to the Motor to Provide Consistent Speed in Flywheel	Using a Consistent Pressure Differential Between the Charged Cartridge and Water Pressure	Using a Consistent Driving (Pneumatic) Force	Using a Consistent Driving Force on the Hammer	Providing a Consistent Current and Voltage in the Motor
Operate Robotically	The System Can be Charged and Actuated Electronically	The System Can be Actuated Electronically	The System Can be Operated Entirely by the Robotic Arm	The System Can be Operated Entirely by the Robotic Arm	The System Can be Charged and Actuated Electronically	An Electrical Control System Actuates Firing Pin	By Placing the Device Around the Wire and Pulling, the Structural Wave is Generated	Motor and Solenoid (Stopper) are Controlled Electronically by the ROV	Cartridge is Opened by ROV Using Electronic Controls	The Motion is Actuated by Pneumatic Control System On-Board ROV	Motor Driving Hammer is Controlled Electonically by ROV	Motor is Electronically Controlled by ROV

utilizing the robotic arm. Other designs however, used pneumatic systems, (de)pressurized gas, and even chemical charges. A sketch and brief description of each conceptual design is presented in Appendix #.

Electromechanical concepts. One of the more obvious choices for delivering a mechanical impact was a solenoid. Because solenoids operating in the 24VDC range are common, and they change electrical energy to linear kinetic energy, they are ideal for use in our design. Solenoids that are sealed to function under 10,000ft of sea water are not available for our purposes, however. This means that our design would require our solenoid (and impact damping materials) in a sealed chamber which is both water and pressure resistant. Although sealing the damping materials in the chamber makes them difficult to adjust, adjusting the voltage input to the solenoid allows us to easily control the magnitude of the impact.

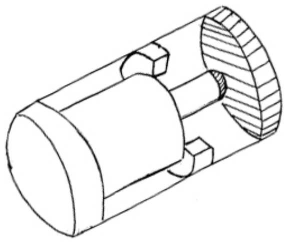
Purely mechanical concepts. By far our simplest category of designs is those which are mechanically actuated, as they require no auxiliary connection to the ROV. The robot simply places and operates the impacter using its robotic arm. Our least complicated design is simply a spring loaded plunger that the arm pulls back and releases. The spring drives a mass into the impact surface, and the device is ready to fire again. Because it has no sensitive parts, it can be exposed to the elements near the sea floor, instead of being sealed in a special chamber. It also requires no special controls on a surface ship, since all interaction can be accomplished using the robotic arm.

Alternatively powered concepts. Since only the mechanical and electrical power are constantly available while submersed, designs using other power sources have a major limiting factor in that they would provide a finite amount of impacts before requiring the ROV to surface again. One such design utilized chemical energy in the form of an explosive. Similar to the action of a nail gun, the device would fire a “blank” charge, propelling a mass to create impact. This design is limited by the amount of ammunition carried on the impacter, and because it requires a signal to fire the charge, it also requires an electronic control system. Its complexity and usability limitations make the design very infeasible.

CONCEPT EVALUATION AND SELECTION

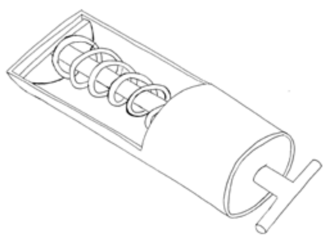
After the initial brainstorming of designs was complete we had to narrow down and combine various aspects of the concepts into viable prototype design ideas. Looking at the concepts any that were beyond our budget financially, not feasible due to complexity, or not economically or ecologically friendly were filtered out. The remaining design concepts can be seen on the next page.

Solenoid (S)



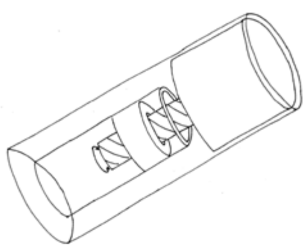
Using a solenoid to create a linear impact was an obvious choice for our impacter. As there are no commercially built, deep ocean solenoids available to us, any solenoid design would require an atmospheric chamber. The solenoid is attached to the impacter casing, and by adjusting the electrical input, the force with which the plunger strikes the impact surface could be controlled. The spectral content is adjusted by coating the plunger and impact surface with a damping material.

Pinball Plunger (PP)



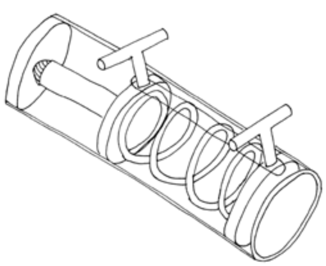
This design is extremely simple. The robot manipulator pulls back on the plunger T-handle, compressing a spring until the handle stops. When the handle is released, the spring drives the plunger mass into the impact surface. The impact force is controlled by placing plunger stops along the impacter casing. The spectral content is adjusted by coating the impact surfaces with damping material.

Linear Actuator (LA)



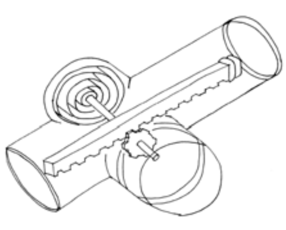
To impart an impact perpendicular to the cable we thought to incorporate a linear actuator into our design. Shown here is a power screw. The motor spins a worm gear a predetermined amount driving a mass into the impact surface. The motor then reverses and pulls the mass away. Due to the motor the device requires an atmospheric chamber. The impact force is adjusted by tuning the motor, while spectral content is controlled with damping materials on the impact surfaces.

Squeeze Charged Plunger (SC)



This impacter is a completely mechanical system. By squeezing the two t-handles together the spring between them is compressed. When the handles are released the stored energy in the spring drives the plunger into the impact surface. By placing stops along the interior of the impacter case the spring compression can be controlled changing the impact force. Spectral content is adjusted by adding damping material to the impact surfaces.

Torsion Spring (TS)



This is the most complicated of our designs. A slip-clutch motor drives the rack backward while tightening a torsion spring. At a predetermined point the clutch disengages and the spring drives the rack into the impact surface. The motor requires an atmospheric chamber to work. The impact force is adjusted by tuning the motor, while the impact quality is controlled by the damping material on the impact surfaces.

The five design concepts have their own merits and limitations, all of which can be seen below.

Table 4: Merits and limitations of top five design concepts

<table>
<tr><th>Design Concept</th><th>Merits</th><th>Limitations</th></tr>
<tr><td>Pinball Plunger</td><td rowspan="2">➢ Simple design
➢ Repeatable
➢ Robotically operable
➢ Impact force and spectral content easily adjusted
➢ Not tethered to ROV by electric power umbilical cord</td><td rowspan="2">➢ No force feedback when spring is fully compressed
➢ Multiple tasks for robot to perform
➢ Alignment on cable
➢ Reaction forces
➢ No data acquisition coordination</td></tr>
<tr><td>Squeeze Charged Plunger</td></tr>
<tr><td>Linear Actuator</td><td rowspan="3">➢ Motors allow for precise tuning and accurate impact force
➢ Repeatable
➢ Robotically operable
➢ Easily coordinated with data acquisition systems</td><td rowspan="3">➢ Requires atmospheric chambers
➢ Difficult to open to adjust motors and damping material
➢ Require programming and electrical systems
➢ Difficult to manufacture</td></tr>
<tr><td>Solenoid</td></tr>
<tr><td>Torsion Spring</td></tr>
</table>

In order to narrow the top five design concepts to one design prototype the concepts had to be evaluated quantitatively according to the engineering specifications outlined for the project. By applying the weights associated with the engineering specifications the concepts are assigned a (+), (-), or 0 in a Pugh chart. The concept with the highest final value was chosen as the final prototype design. The Pugh chart can be seen below with the results of our concept comparison.

Figure 4: Pugh chart

		CONCEPT VARIANTS					
SELECTION CRITERIA	Weight	TS	S	PP	SC	LA	REF
Can be operated robotically	17	+	+	0	-	+	0
Pressure resistant to 10,000 ft.	15	-	-	+	+	-	0
Operable in salt water	11	-	-	+	+	-	0
Provides specified impact magnitude	11	0	-	0	0	0	0
Provides specified impact quality	9	0	0	0	0	0	0
Provides repeatable impact	9	-	+	0	+	+	0
Can provide multiple impacts per dive	9	0	+	+	0	0	0
Resistant to corrosion	6	-	-	0	0	-	0
Can operate on 2A at 24VDC	6	0	0	+	+	0	0
Compact size	3	0	0	0	0	0	0
Simple design	2	-	0	+	-	-	0
Lightweight	2	0	0	0	0	0	0
∑+		17	35	43	41	24	
∑–		43	43	0	19	34	
∑		-26	-8	43	22	-11	
RANK		5	3	1	2	4	
CONTINUE?	**Y/N**	**N**	**Y**	**Y**	**Y**	**N**	

After reviewing our design selection with our sponsor, we decided to combine features of our top two ranked concepts, the pinball plunger and squeeze charged plunger, to create our final prototype.

SELECTED CONCEPT

After analyzing each of our design concepts, we chose to combine the most promising features of the pinball plunger and the squeeze charged plunger in our final design. The prototype design is operated solely via robotic arm manipulation and thus requires no on-board auxiliary power source. Additionally, this design does not require any delicate electronics, thus allowing all components of the device to be in contact with seawater without risking critical failure.

Figure 5: CAD rendering of selected design concept

The impacter attaches to the cable via the clamp opposite the handle end. The gripper squeezes the plunger handle against the end of the impacter casing which pulls the plunger back until the pawls lock into place. The gripper then squeezes the pawls' t-handles together, releasing the plunger, and allowing the compressed spring to drive a mass forward. The mass and its attached damper then impact the closed end of the impacter casing, creating an impact perpendicular to the cable.

The impacter, including spring will be constructed of aluminum with the exception of the damping material and hardware. It will be approximately 11 inches long by 3 inches in diameter, with about 1 inch of travel for the plunger handle. The impacter casing will be constructed from aluminum tubing, the ends will be caps cut and lathed from aluminum plate, and the pawls will be cut from plate aluminum. The spring to power the impacter was specified to be stainless steel since it is the only readily available spring material that is corrosion resistant. The clamping device, like the impacter will be aluminum, milled from stock with special ordered stainless steel torsion springs.

ENGINEERING ANALYSIS

The key variables taken into consideration when designing our prototype were maintaining dimensions that allow the ROV to easily manipulate our device while insuring that the proper impact magnitude and spectral content were produced. Handle geometry and dimensions were decided based on the geometry of the robot manipulator provided to us, while the overall dimensions of the impacter were influenced by the experimental environment it will be used in. No parts on our impacter require overly precise tolerances, since most characteristics of the design can be adjusted once constructed, as a result, lathed parts were made to a tolerance of .01” and others were made to approximately .05”. Measurements taken off the existing impacter provided us with information about spring constant and compression distance that we needed to incorporate into our design. *CES Edupack 2007* was used to determine the optimal material for manufacturing the impacter. 6061 Aluminum was chosen for its resistance to corrosion, ability to be welded and machined, its light weight, durability, and lower cost in comparison to other materials. Stock will be procured from local distributers such as Alro Metals Plus, and McMaster-Carr. These providers were chosen based on their ability to provide a very wide selection of materials on very short notice.

Quantitative Analysis

In completing the final design of the impacter, a number of design issues arose which required a quantitative analysis. Some parts required specifications defined by our constraints, and others were identified as likely points of failure and analyzed to determine safety factors in various modes of failure. All material properties and specifications are courtesy of the product vendors (for off the shelf components) or *CES Edupack 2007* software (for raw materials). Below is a summary of the evaluation of each component. Appendix F contains summaries of the calculations as they are defined below.

Spring. The main spring driving the impacter must fulfill our size, corrosion resistance, and mechanical requirements. Stainless steel springs are readily available and resistant to corrosion. Our mechanical constraints, as defined reports detailing the previous iteration of the impacter, suggest a total impact energy of between 11 and 38 lbs*in.

We were able to determine the magnitudes of impact offered to us by different stainless steel springs that fulfilled our size constraints using the equation,

$$Energy\ Stored\ In\ Spring = \frac{1}{2} * k * x^2$$

One such spring, available through McMaster-Carr (PN 1986K25), provided us with a maximum impact of 50 in*lbs. By adjusting spacers in different locations in our impacter, we are able to adjust the impact magnitude from almost 0 lbs*in to 49.5 lbs*in. It has an overall length of 3”, with a travel of 1.25” and a spring constant of 64 lbs/in.

Flow Holes. Since our design operates under water without a sealed container, water fills the cavity where the mass moves. As a result, it is necessary to "vent" this cavity to allow the water surrounding the mass to move without heavy resistance. Ideally, there would be a single hole which is the same size as the cross-section of the mass. Unfortunately, due to other design constraints, we must use multiple, smaller holes. Since the mass moves at the fastest rate close to the end of its travel (before impact) it is desirable to have a large "open" surface area both in front of, and behind the mass at this point in its travel. Unfortunately, because of the geometry of the impacter, this also corresponds to the point at which the least surface area is available to create vents.

The equation governing the ratio of vent size to mass cross-section is as follows:

$$Ratio = \frac{Area\ of\ Vents}{Cross\ sectional\ area\ of\ mass} = (\#\ of\ vents) * (width) * \frac{Exposed\ Length\ at\ Impact}{\pi r_{mass}^2}$$

The vents must not be large enough to introduce large debris into the mechanics of the device. As a result, a width of .5" was chosen, and 8 vents are included. To allow for room to attach the clamp and impacting surface, the length of the vents exposed at impact will be a minimum of .75". This yields a ratio of only 60% at the worst case scenario, however, if the impacter is adjusted to fire at lower than its maximum magnitude, more of this vent will be exposed, allowing for a ratio of 100% or better.

It is important to realize the effect of multiple, smaller openings on the flow characteristics of the impacter. Larger openings allow less resistance to a flow than multiple smaller openings of the same total size. Unfortunately, we are limited by the geometry of the impacter, and the hostile environment in which it operates. As a result, we are unable to have larger holes, which may introduce debris that would jam the workings of the impacter.

Cap Screws. The design our cap (the surface impacted by the mass) calls for a removable piece so that springs and impact damping material can be changed easily. It is held in place by bolts which run perpendicular to the direction of the impact. Since these bolts may absorb the entire force of the impact in the event that the casing is held in place, we evaluated them for failure in shear.

The equation used is as follows:

$$Shear\ Stress = \frac{Force}{Area} = \frac{\frac{Impact\ Energy}{Impact\ Distance}}{Area\ of\ Bolt}$$

The maximum impact energy that our prototype can impart is 50 lbs*in. Our choices of damping materials suggest that the impact will be absorbed over a minimum distance of .125". This results in a force of 400lbs during impact. The shear stress for 6061-T6511 Aluminum is assumed to be .6*tensile stress. Using these figures and analyzing a design with three (3) .25" diameter Type 316 Stainless Steel bolts, the design has a safety factor of over 5. This is acceptable for our purposes since the maximum loading on the bolts will likely be much lower than the 400lbs calculated. This is because the spring will likely not be fully charged, and the bolts do not have to react the entirety of this force since the majority of it is transferred through the cap and into the clamp.

Connector Rod. The connector rod between the mass and the movable (inner) plate was evaluated for failure in tension. It was assumed that the maximum loading would be 150lbs, which is the maximum force the robotic gripper can exert on the two plates, however, a maximum force of only 80lbs would be required to fully charge the spring.

To calculate tension in the rod the following equation was used:

$$Tension = \frac{Force}{Area} = \frac{150lbs}{\pi * r^2}$$

Using the minimum tensile strength of 6061 T6511 Aluminum (27.99ksi), a minimum diameter of approximately .08" is required. Due to the need to weld the connector rod to the mass and movable plate, a diameter of .5" was chosen for our rod. This also works well with the geometry of our robotic gripper. Using the same equation above, the tension in the .5" dia. rod is 764psi, yielding a safety factor of 37.

It is appropriate to use this safety factor for this application because the "Heat Affected Zone" created by our welds will significantly weaken the material surrounding the welds. No method is available to us to calculate the exact strength of our H.A.Z., however because of the negligible cost in increasing our safety factor from 1 to this excessive value, it is a sound decision.

Stresses on Pawls and Supports. The pawls each react a moment generated by the force of the charged spring at their point of rotation. This creates a bending moment and a shear stress in the pawls, as well as a shear stress in the axle on which the pawls rotate. The shear stress in the pawls was calculated in the same manner as the cap screws above, and yielded a safety factor of over 100. The bending moment on the pawls causes tensile and compressive stresses in the edges of the pawl which are maximized at the point where it has the smallest cross-section.

The tensile/compressive forces in the pawls were calculated using the following equations:

$$\sigma = Moment * \frac{(Distance\ from\ Center\ Axis)}{(Second\ Moment\ of\ Inertia)}$$

$$Second\ Moment\ of\ Inertia\ for\ a\ Rectangular\ Beam = \frac{(Base * Height^3)}{12}$$

At the minimum cross section (which also has the minimum second moment of inertia), the bending moment causes forces of 3800psi. This yields a safety factor of 7.3 when compared to 6061-T6511 Aluminum's tensile strength of 27.99ksi.

Shear in the pin that holds the pawls must also be evaluated. In order to calculate the shear stress in this member, two reaction forces are considered to cause the shear stress, one at each end of the contact between the axle and pawl. This is a reasonable assumption because a loose fit between axle and pawl will result in these two points of contact. The equation used to calculate these forces is as follows:

$$Moment\ on\ Pawl = Moment\ Exerted\ by\ Axle = 2 * .25 * Shear\ Force$$

The shear force is calculated in a similar manner to that of the cap screws above, yielding a safety factor of approximately 4. This is acceptable because the force on the end of the pawl can be accurately calculated. In addition, the ends of the pawls have additional support from the walls of the casing.

Impact Spectral Content. In the previous impacter design, trial and error were used to find an "Impact Absorbing Material" that generated suitable spectral content. According to previous research from Professor Perkins and other students working with him in the past, it is very difficult to predict the best material and geometry for this application. We do know, however, that the accelerometers measuring the impact wave propagating through the cable respond best when a lower frequency impact is generated. As a result of qualitative descriptions of "good impacts" and examination of successful impact absorbing materials, we chose a space of 1" to add between the mass and impacting surface. This allows us the ability to experiment with multiple designs, without knowing for sure which material and geometry will work best. Our sponsor's encouraged us to solve this through trial and error, as was done before, because of the unpredictability of the spectral content in relation to engineering characteristics and geometry of our materials.

Jamming of the Mass. A major design concern was that our mass may jam during its travel along the inside of the impacter casing. As a result, the geometry of the mass, spring, connector rod and movable plate were adjusted such that the mass could not turn enough to jam itself. The corners were also rounded to help prevent jamming. Because the jamming of the mass depends on the coefficient of friction between the mass and the casing, and the coefficient of friction depends heavily on the material surfaces (which must be machined), we chose to machine and test the mass to verify our assumptions. At this point, the mass has been tested, and does not jam, even when turned well beyond the maximum angles it will experience when fully assembled.

Qualitative Analysis

Throughout the design process, different forms of qualitative analysis were necessary to ensure that our design was as cost effective as possible for mass production, as well as safe for the environment and users, all without losing any functionality. We used three forms of qualitative analysis, FMEA, DFE, and DFMA. The FMEA was used to help us determine points of potential failure, and redesign to avoid such failures. The DFE was used to ensure our product is safe for the environment through all stages of production and use. And the DFMA was used to optimize our manufacturing and assembly processes to make them as cost effective as possible.

FMEA. Our FMEA (see Appendix G), or Failure Mode and Effect Analysis, was produced to identify points of potential failure, rate their likelihood of occurrence, as well as severity and ease (or difficulty) of detection. The product of these three values gave us our Risk Priority Number (RPN). The RPN represented the risk involved with each part. The higher the RPN, the more likely a catastrophic failure was to occur at that point.

After our original values for RPN were found for our original design, we augmented our design in any way possible in order to lower the RPN. For example, we found that when charged, the mass would place a very large bending moment on the pawls which would cause instability and could cause failure of the trigger mechanism. In order to prevent this failure, we redesigned the holes that the pawls travel through to sit flush against the pawl, so that the bending moments were reacted by the casing, and the pawls never lost their stability. While utilizing a simple redesign such as adjusting the placements of the pawl holes, we were able to reduce the RPN for this failure mode from 162 to 12. To see the rest of our potential failure modes and the recommended courses of action used to resolve them, please view Appendix G.

DFE. In most cases, the DFE, or Design for the Environment, would be used to ensure that the mass produced product would be environmentally friendly. This is achieved by identifying the environmentally harmful stages of production, use, and disposal of the product. By redesigning these stages, we are able to produce a product that is much better for the environment.

For our particular product, the mechanical impacter, environmental friendliness came naturally. Since the only inputs and outputs to and from our impacter are simple mechanical forces, we know there is no waste that can be harmful to the environment. By using solely recyclable materials in the manufacturing and assembly of our impacter, we can further be sure we are not hurting the environment. Below is our DFE, which outlines our method for determining our products environmental influence. Since we could not find any issues with our impacter that we believed to be harmful to the environment, we chose to keep our design constant in this respect, and to use recyclable metals in our manufacturing.

Table 5: Design for the Environment Chart

Product Undersea Cable Impacter	**Project** Me 450		
Date Fall 2007	**Author** Team 13		
Production (Materials, treatments, transport, and extra energy)			
material or process	*amount*	*indicator*	*result in millipoints*
Aluminum 100% Recycled	4	60	240
Casting	4	72	288
Steel High Alloy	0.25	910	227.5
Gas-Fired Heat (industrial furnace)	40	5.3	212
		Total	**0.97**
Use (Transport, energy and possible auxillary materials)			
process	*amount*	*indicator*	*result*
None	0	0	0
		Total	**0**
Disposal (Disposal processes for each material type)			
material and type of processing	*amount*	*indicator*	*result*
Recycling Aluminum	4	-720	-2880
Recycling Ferro Metals	0.25	-70	-17.5
		Total	**-2897.5**
Total (mPt) All phases			**-2896.53**

DFMA. The DFMA, or Design for Manufacturing and Assembly, is used to optimize product design in terms of cost and time effectiveness. For our purposes, we found that the majority of cost for mass production of our prototype would come in the manufacturing followed by assembly stages. In order to reduce these costs, the machining of raw materials into parts must be drastically reduced, as this is by far the most costly step in production. In order to avoid the machining steps many of our components undergo, we decided to cast parts in aluminum. This would save time from the production of each individual part, as well as the wasted materials resulting from milling and turning, and the materials needed for welding. By eliminating the number of processes and materials necessary to create our product the manufacturing costs of mass production will decrease.

By casting components of the impacter the assembly of the product becomes much easier. To create our prototype 10 various parts were welded and screwed into place. By casting parts we would have 5 components that are simply screwed together. This reduces the cost to assemble the impacter. A graphical representation of the DFMA can be found on the next page.

Table 6: Design For Manufacturing and Assembly Chart

<table>
<tr><th>Part</th><th>Current Design</th><th>High Cost of Current Design</th><th>Redesign</th><th>Design Guidelines</th><th>Specific Guidelines Incorporated</th></tr>
<tr><th colspan="6">Part Manufacturing</th></tr>
<tr><td rowspan="4">Pawls</td><td rowspan="4">Cut, drill, weld, finish</td><td rowspan="5">Man hours for production, wasted metal material, welding supplies</td><td rowspan="9">Cast</td><td rowspan="3">Design for Casting</td><td>DCF-1
Avoid sharp corners</td></tr>
<tr><td>DFC-2
Add draft angles</td></tr>
<tr><td>DFC-3
Keep uniform wall thickness</td></tr>
<tr><td rowspan="4">Design for Machining</td><td>DFMC-1
Preshape by casting</td></tr>
<tr><td>Casing</td><td>Weld, cut, mill, finish</td><td>DFMC-14
Minimize tool changes and setups</td></tr>
<tr><td>Plunger Handle</td><td>Lathe, drill, thread, finish</td><td rowspan="4">Man hours, wasted metal material</td><td>DFMC-9
Place holes away from edges and holes</td></tr>
<tr><td>Plunger Shaft</td><td>Cut, thread</td><td>DFMC-3
Use standard dimensions</td></tr>
<tr><td>Mass</td><td>Cut, lathe, drill, tap</td><td rowspan="2">Design for Part Insertion</td><td>DFPI-2
Add alignment features</td></tr>
<tr><td>Casing Cap</td><td>Cut, drill, finish</td><td>DFPI-1
Add features for easy insertion</td></tr>
<tr><th colspan="6">Assembly</th></tr>
<tr><td rowspan="4">Assembled Impacter</td><td rowspan="4">10 machined parts, welding, and screwing required</td><td rowspan="4">Significant man hours , welding materials,</td><td rowspan="4">5 cast parts screwed together</td><td rowspan="2">Design for Assembly</td><td>DFAS-1
Minimize part counts</td></tr>
<tr><td>DFAS- 5
Standardize to reduce part variety</td></tr>
<tr><td rowspan="2">Design of part handling</td><td>DFPH-1
Maximize Part symmetry</td></tr>
<tr><td>DFPH-2
Add features to facilitate orientation</td></tr>
</table>

FINAL DESIGN

Impacter Casing. The impacter casing is constructed of 6061 aluminum alloy. The component's chief purposes are to contain the impacter's moving components and to react the force necessary to compress the employed spring. The casing is designed to allow the ROV grippers proper clearance for the required manipulating tasks and to allow water and debris to flow freely out of the impacter mechanisms. This design aims to increase functionality by making it simplifying remote use and maintenance via robot manipulators.

Impact Assembly. The impact assembly, or assembly of the top squeeze charge plate, connecting rod, and impact mass, is constructed of 6061 aluminum alloy and is assembled via threaded joints and tungsten inert rod (TIG) welds. The purpose of this assembly is to transmit the forces necessary to charge and discharge the impacter. The component includes a critically dimensioned connector rod and notched mass designed to compress the spring and to lock in place at a critical displacement. Additionally, the device aims to facilitate robotic use by integrating ridges on the top squeeze plate to prevent slipping of the robot manipulators.

Spring. The spring is constructed of Type 302 stainless steel and was purchased from McMaster-Carr (PN 1986K25). Based on previous testing, the spring needed an impact magnitude of between 11 and 38 lbs*in. It also needed to be able to accomplish this in a relatively short travel, so that it could be charged by a single movement of the gripper. The spring chosen fits well within our design at 3" in total length with a 1.25" outer dia. and enough room inside to accommodate our .5" connector rod. It charges to 50 lbs*in with only 1.25" of travel (the travel can be adjusted to allow for impact magnitude between 0 and 50 lbs*in).

Pawls. The pawls are constructed of 6061 aluminum alloy. The devices' chief purposes are to lock the notched impacting mass in place at a critical displacement and to subsequently release the mass on command to provide an optimal impact. The pawls are designed with a one-way locking mechanism, similar to those employed on door locking mechanisms, such that the mass can be compressed past its critical distance with ease, but cannot be released without further manipulation. The pawls' T-handles were designed with robot manipulation in mind and are dimensioned and spaced to simplify the manipulation needed to release the mass toward the impact surface.

Impact Surface. The impact surface will consist of various compliant polymers to provide a damped mechanical impact per the project goals. This material is fixed to the aluminum mass and will deform as it collides with the impacter casing, dissipating the energy that would otherwise provide a undesirable impact spectral content. The specific dimensions and materials will be determined through experimentation with the physical prototype.

Operation. The impacter clamps to the cable, opposite the handle end. To charge the device, the ROV gripper squeezes the device's plunger handle down, reacting the pull force on the near end of the impacter casing. The ROV grippers draw the plunger back until two pawls lock the impact mass into place with the spring compressed to a critical displacement. The gripper then rotates 90 degrees to squeeze the pawls' T-handles together, releasing the plunger and allowing the compressed spring to drive the mass forward. The mass and its attached damper then impact the closed end of the impacter casing, creating an impact perpendicular to the cable.

Clamp. Clamp design is not yet finalized. We are currently working in conjunction with Graduate Student, Tom Waisenen on its final design. Dimensioned drawings, as well as a completed clamp design to follow.

Dimensioned drawings for manufactured parts can be found in Appendix D.
For a complete listing of the materials purchased for the production of the impacter prototype see the bill of materials below.

Table 7: Bill of Materials for the Impacter Prototype

Quantity	Part Description	Purchased From	Part Number	Price (each)		Subtotal
2	2-3/4 OD x 0.125 wall, 6061-T6511	Alro Metals Plus, Ann Arbor	26314585	38.88		77.76
1	2-5/8 RD 6061-T6511	Alro Metals Plus, Ann Arbor	21412855	37.32		37.32
1	1/2 RD 6061-T6511	Alro Metals Plus, Ann Arbor	21410605	7.22		7.22
1	3/8 RD 6061-T6511	Alro Metals Plus, Ann Arbor	21410400	7.56		7.56
1	5/8 OD x 0.065 wall	Alro Metals Plus, Ann Arbor	26301180	27.84		27.84
6	302 Stainless steel 64 lb/in linear spring	McMaster-Carr	1986K25	2.32		13.93
1	302 Stainless steel 9.75 in-lb 90 deg torsion spring	McMaster-Carr	9287K96	6.45		6.45
10	316 Stainless steel 1/4-20 hex head cap screw	McMaster-Carr	92186A548	0.67		6.65
	Aluminum filler material	University of Michigan		0.00		0
	Acetone	University of Michigan		0.00		0
	Argon sheilding gas	University of Michigan		0.00		0
	1/2 inch 6061-T6511 plate	University of Michigan		0.00		0
	1/4 inch 6061-T6511 plate	University of Michigan		0.00		0
	3/8 inch 6061-T6511 plate	University of Michigan		0.00		0
			Total	128.26		184.73

MANUFACTURING AND ASSEMBLY

Finally, provide a paragraph to address what ethical issues will arise if your final design is made available for public use.

The final assembled impacter is composed of many parts, most of which were manufactured on site at the University of Michigan Undergraduate machine shop under the supervision of Bob Coury and Marv Cressey. Fastener hardware springs were purchased.

Manufacturing. Due to the cylindrical nature of many system components, manufacturing was done primarily on a lathe. All cylindrical parts (impacting mass, connector rod, end caps, plunger handle) were turned on a lathe. Pawls and pawl supports were cut roughly using a band saw and filed and sanded to their specified dimensions.

The impacter casing was constructed using a multi-step manufacturing process of cutting, welding, milling and drilling. The pawl supports and bottom end cap were TIG welded to the casing stock before cutting all holes in the casing using a band saw (for rough cuts) and mill (for precise cuts). Next, holes were drilled in the casing and top end cap and tapped to accept the specified hardware. Specific process plan sheets for each component can be found in Appendix E.

Assembly. Following the completion of each component, system assembly was completed using the following steps:

1. Thread the impact mass onto the connecter rod and place the spring around the rod;
2. Using ½-13 hardware, fasten damping material to the impact surface of the mass;
3. Insert the mass-connector rod-spring subassembly into the impacter casing as drawn;
4. Thread the plunger handle onto the connector rod, opposite the impact mass;
5. Attach the right pawl to the appropriate pawl support on impacter casing (as drawn) using ¼-20 hardware;
6. Insert prescribed torsion spring into the installed pawl and fix to impacter casing using ¼-20 hardware as drawn;
7. Attach the left pawl to the appropriate pawl support on impacter casing (as drawn) using ¼-20 hardware taking care to insert the torsion spring leg opposite the right pawl;
8. Place and secure the top cap to the impacter casing as drawn.

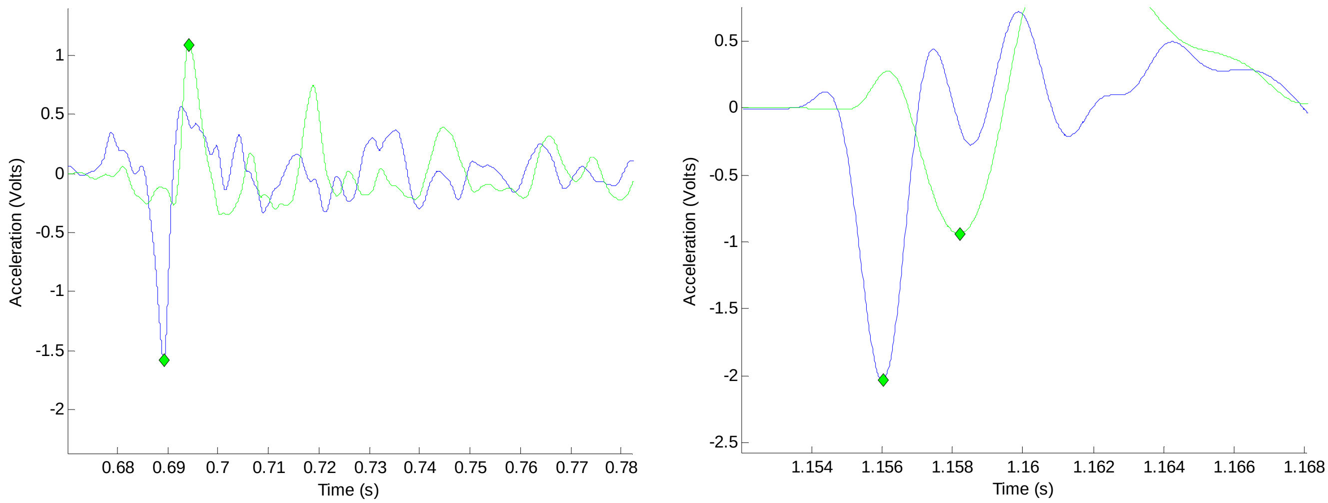
Figure 6: Exploded impacter assembly

TESTING PROCEDURE

Procedure. With the impacter completely assembled, we tested its impact for quantitative and qualitative performance in the wave basin at the University of Michigan Civil Engineering Lab. We attached the impacter to a section of tensioned cable, impacted the cable, and collected acceleration data from near- and far-field accelerometers. Field tests will use a data processing algorithm to reduce the accelerometer outputs and identify the magnitudes and times for acceleration peaks associated with the induced impact. The algorithm then processes the change in time to determine cable tension

Results. Baseline and mechanical impacter responses were recorded, yielding the following acceleration versus time curves.

Figure 7:Impacter (a) and baseline (b) testing results



a. Impacter response **b. Baseline response**

While the impacter response is qualitatively more noisy, the processing algorithm identifies the peak values correctly (denoted with green markers) and results in the same tension measurements as the baseline. To make testing more robust, it will be important in future iterations of this design to damp the impact further to generate more defined peaks for the software to analyze. However, it is clear that the mechanical impacter is effective in providing a usable impact for use with tension testing.

Repeatability was also tested by taking multiple peak-to-peak time measurements for given tensions, tabulated below.

Figure 8: Peak-to-peak times for given tensioning cylinder pressure

Tensioning cylinder Pressure (psi)	Peak-to-peak time (ms)	Tensioning cylinder Pressure (psi)	Peak-to-peak time (ms)
1000	2.25	500	2.16
	2.27		2.24
	2.10		2.08
	2.00		2.08
	2.00		2.00
	2.02		2.00

For a tensioning cylinder pressure of 500 psi, the standard deviation of resultant times is 0.12 ms. For a tensioning cylinder pressure of 1000 psi, the standard deviation of resultant times is 0.09 ms. This consistency is sufficient for the given testing resolution.

Testing has shown that, while tuning the impacter springs and damping materials is desirable for an optimal response in testing, this design is a valid proof of concept for a robotically-operated cable impacter, as it imparts a consistent, valid impact for use in cable tension testing.

RETROSPECTIVE PROJECT PLAN

Our project can be broken down into five main phases: Data Gathering, Preliminary Design, Final Design Preparation, Final Design Execution, and Documentation.

Project Chronology

Data Gathering involved primarily interacting with our sponsors and Professor Perkins. They were able to give us guidance, as well as a considerable amount of information as to the strict requirements for our final design. These weekly meetings were attended as a team along with Professor Perkins and Tom Waisanen. These meetings served as both a status update to our sponsors, as well as a source of information gathering for us.

The Preliminary Design phase, beginning with the conclusion of Design Review 1, involved the creation of multiple, varied concepts. Each team member generated several concepts, then judging one another's concepts to narrow the field down to the best candidates. The design options were evaluated further, and a final design, the pinball plunger, was chosen based on its engineering characteristics. This phase concluded with Design Review 2.

In the third design phase, the Final Design Preparation, we fine tuned our chosen impacter design and advanced it to a point where it could be turned into a functional prototype. This involved verifying the prototype's ability to be actuated robotically, and be pressure, water, and corrosion resistant. Mathematical and engineering analyses were conducted to insure the prototype would operate as desired. Materials such as aluminum stock for the impacter casing and plunger, the internal spring, and damping materials were be ordered. Final adjustments to the design were made to ensure it satisfied all of our sponsor's criteria. This phase concluded with the presentation of our final design during the third Design Review.

During the fourth phase, Final Design Execution, we manufactured our functioning prototype. After machining and assembly, it was tested in the Civil Engineering Lab wave basin at the University of Michigan's North Campus. The testing was to verify our prototype's functionality as well as confirm the correct impact magnitude and spectral content.

The last phase of the project was Documentation. Since this is a multi-year project in progress, we have documented our progress in this, our Final Report. In addition to being an ME450 deliverable, this will allow future teams to easily continue towards this project's goal, using our work as a reference. Our project was also showcased in the Design Expo in the Capitol Building in Lansing. We created a short presentation and exhibit to present our work to the public.

Equipment Required
A project of this nature requires specialized equipment. Fortunately, our sponsors have been gracious enough to provide us with some of the equipment we will need, as well as funding for additional tools we may need. Additionally, since this is a long term project, a lot of groundwork for our project has already been laid.

We have already received from our sponsor a gripper from the robotic arm that will be used on the submersible ROV, a manual describing the robotic arm specifications, and a fully functioning remote controlled robotic manipulator arm. We have also received numerous samples of oceanographic cable for use in our tests.

Technical Assistance
Since the members of our team have only a limited background in vibrations, and none in dealing with undersea robotics, we are fortunate to have the assistance of Professor Perkins and Tom Waisanen, a graduate student assisting him. Our sponsor has also put us in contact with 2 experts on ROVs who will be helping us design an interface for our impacter that will be robot friendly.

DISCUSSION FOR FUTURE IMPROVEMENTS

The build process and testing of our impacter provided us with insight into a number of possible design improvements. Several improvements and tweaks covered the whole device, while others were specific to individual parts. There were also a number of elements of our design which functioned particularly well or better than expected.

Casing. Our casing certainly has some room for improvement. Early designs called for 8 holes around each end of the impacter cavity, covering the entire circumference. Our prototype only has 5 holes since practical constraints limited our ability to fixture the partially completed casing for milling. We did not want to warp our casing when welding, therefore, all welding on the casing was completed before milling. This resulted in the attachment of the pawl supports which prohibited fixturing the casing to allow for milling holes in the underside of the impacter.

It also seems that our casing did not need to be round. It would have simplified some elements of our machining and complicated others to have had a square housing, but it would not have been prohibitively difficult. This casing design, depending on future adjustments made, may be easier to make.
Making the casing larger would allow for not only larger springs, and a heavier mass (discussed below), but would allow for better access to the plunger (for charging the spring).

Mass. The mass could be heavier, although it does not seem to have a negative impact on current performance. Ideally, a heavier material would be used since it would cause the mass to move slower, and therefore lose less energy through friction to the water surrounding it. It will also become necessary to have a larger mass if the spring power is increase (discussed below).

Spring. The spring power seems to be inadequate compared to other impacting methods that have been successful in the past. This will need to be verified through more testing, as the impact generated using the current setup did provide an impact that reached close to the limits of the accelerometers used. If the magnitude of the impact is increased beyond this amount, the data will become saturated, and as a result, unusable.

Pawls/Triggering Mechanism. The current design utilizes a 2 trigger symmetrical system. Because the current design does not include a method to stop the pawl once it has completely released the mass on its side, it is possible for one pawl to release, and travel too far, while allowing the other to stay in place, holding back the mass. This is a catastrophic failure for our device, since there is no way to fire it if this happens, and it must be returned to the surface to be corrected.

A simple remedy for this would be a single pawl release system. By having only one pawl, and a stationary handle to allow the robotic gripper to "squeeze" against, the same effect can be achieved. This eliminates the problem of imbalance, and simplifies the design, and manufacture, or our device.

Pawl Supports. Mounting the pawl supports in place was particularly challenging. Achieving a reasonable tolerance in clamping the piece in place then welding it was difficult. A simple remedy to this would be to "notch" the casing in a mill, leaving a place that the pawl support fits nicely, and will not move in the process of welding.

Fasteners. The use of stainless steel fasteners has so far proved to be successful, however, close attention should be paid to corrosion around the areas where aluminum and steel meet (at the junction of unlike metals, corrosion can increase). Lock nuts, as well as Loctite® or a suitable marine alternative, should be considered to prevent fasteners from loosening over time and with vibration. Another alternative would be the use of pins and cotter pins, which also will not loosen over time.

The use of constant sized bolts throughout the design was very convenient. While some elements of the design may have been able to make use of larger or smaller bolts, the use of constant sized fasteners allowed us to use a single socket wrench to assemble/disassemble the entire impacter. This, combined with parts which can be assembled/disassembled by hand, makes the impacter suitable for use at sea where specialty tools might not be readily available.

Clearances. The choice of leaving large clearances between moving parts worked out very well. We did this to avoid the potential of debris jamming the system. The concern in doing this was that our device would lock up or bind on itself if it traveled off axis. Our dimensions were chosen such that this could not occur, and when tested the device functioned without jamming, and left enough clearance for silt and other debris to move through the device without jamming it.

Damping Material. The choice of damping material and shape was based on trial and error (at the recommendation of Professor Perkins), and observation of damping materials used successfully in the past. It is essential that a solid (not foam/inflated) material is used, as the pressure at 10,000ft would crush it. We used off the shelf rubber chair leg covers cut to the appropriate size based on trial and error. In the future, it would be prudent to use a commodity part for the material, so that the spectral content can be easily replicated.

Behavior of Impact. The impacter currently provides an impact different from that of the wooden post used in terrestrial testing. The wooden post exerts a single impact in a single direction. The impacter however, forces the cable to react the force of acceleration of the mass, and then deceleration of the mass at impact. This seems to distort the readings taken from the accelerometers.
Further testing is needed to verify the effect of this behavior and whether or not it is detrimental to recording the tension of the cables. This information should be available through Professor Perkins or Tom Waisanen, who are performing testing of this impacter design.

Asymmetric Design. The mounting of the casing to the cap and the cap to the clamp were made asymmetrical to guarantee that the device was assembled in the correct orientation. This simplifies assembly, and negates the need for instructions to assemble the device if it is taken apart for cleaning. This is particularly useful since documentation will not be readily available at sea.

Clamp. Our joint clamp design with Tom Waisanen left some room for improvement. The torsion springs used to hold the clamp closed were just enough to prevent this impacter from rotating when mounted on a large cable (where it has more force since the springs are displaced further). This was not tested on smaller cables. Additionally, any increase in impacter weight (from casing size, mass increase, spring replacement, etc.) would necessitate more power from the torsion spring(s) closing the clamp to prevent rotation. The springs used (2) were the most powerful available through McMaster-Carr that were suitable for the task (9.75 in*lbs at full deflection). A number of companies, including Diamond Wire Spring Co. provide custom made torsion springs, which would be more suitable for the task, but carry a longer lead time.

Another possible improvement for the clamp would be the addition of "teeth" along the surfaces which grip the cables. This would help to prevent the impacter from rotating once placed on the cable.

CONCLUSIONS

The United States Navy approached Professor Perkins with the desire to determine the tension in oceanographic cables. Professor Perkins has developed a method to determine the tension in cables by imparting an impact on said cable, and measuring the resulting wave's frequency using accelerometers. After considerable research, we have collectively designed an impacter we believe will fulfill our design requirements. It is our task to complete the design and fabricate a functional version of our impacter prototype. Using our current design, we will be constructing a prototype providing the optimal impact magnitude with the correct spectral content. Our impacter prototype will be used for testing here at the U of M campus at an on-site underwater test bed. Additionally, the U.S. Navy will determine if our prototype concept is suitable for testing, and if so, they will iterate the design further and eventually begin use, testing cables in the field.

ACKNOWLEDGEMENTS

We would like to thank our sponsors, Professor Noel Perkins at the University of Michigan and Chris Nicholson, Karen Miller, Steve Karnoski, and Greg Cooper with the United States Navy for all of their guidance and assistance throughout the design process. We would also like to acknowledge Bob Coury and Marv Cressey for their manufacturing assistance and hours spent supervising the creation of our prototype. In addition, we would like to thank Professors Bogdan Epureanu, Kazuhiro Saitou, Yoram Koren, Katsuo Kurabayashi, Jwo Pan, and Mr. Mohammed Shalaby for their help and encouragement in and out of the classroom. Finally, we would like to thank Tom Waisanen for helping with the conceptual design, manufacture, and testing of our project.

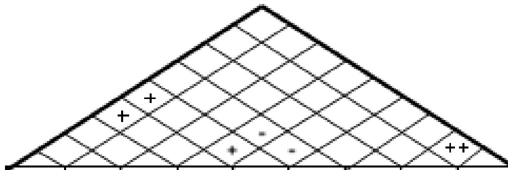
REFERENCES

Chae, H. M. (2006). *Mechanical Impact for Wave Propogation.* Ann Arbor: University of Michigan.

Perkins, N. (2006). *Measuring Tension in Buried Cables, Final Report-Year 1.* Ann Arbor: University of Michigan.

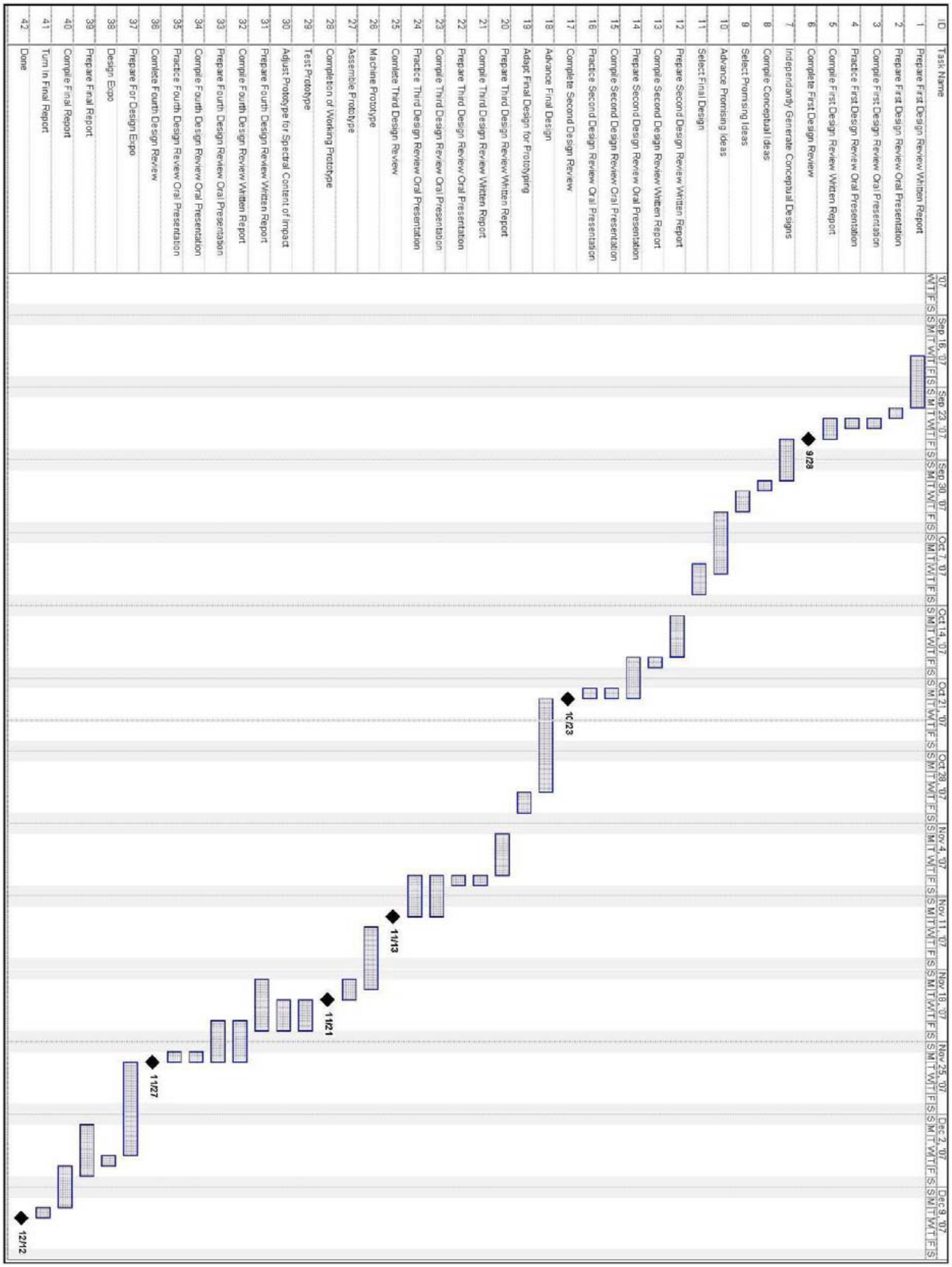
Perkins, N. (2007). *Measuring Tension in Buried Cables, Final Report-Year 2.* Ann Arbor: University of Michigan.

APPENDIX A – QUALITY FUNCTION DEPLOYMENT

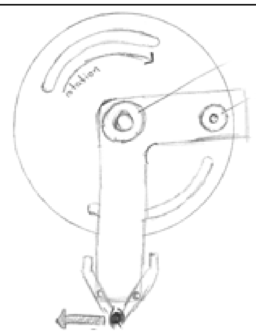
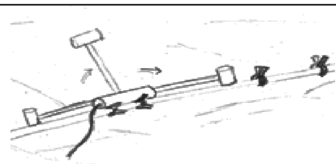
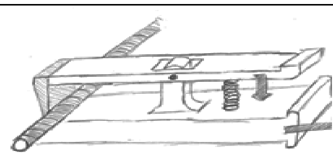
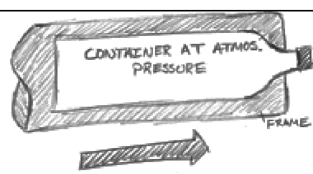
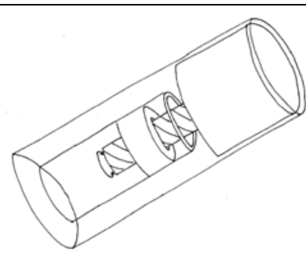
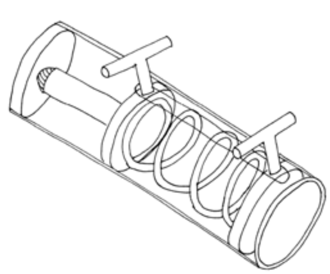


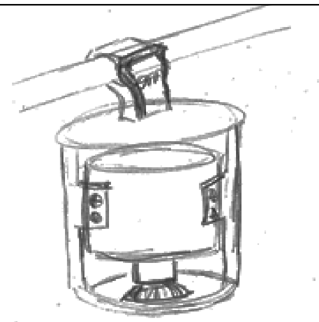
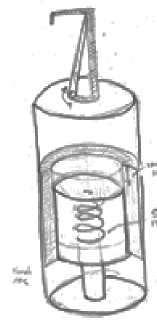
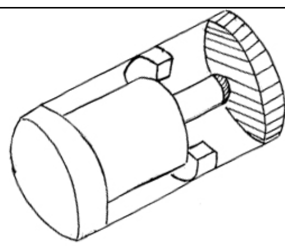
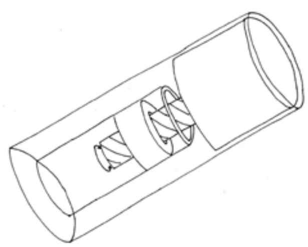
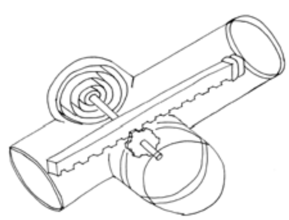
	Weight	Impact Magnitude (N)	Vibration Frequency Induced by Impacter(Hz)	Time of Contact (s)	Impacter Outer Dimensions (m)	Weight (N)	Maximum External Pressure (MPa)	Number of Impacts per Dive (#)	Manipulator Degrees of Freedom (#)	Manipulator Range of Motion (m)	BENCHMARKS	4X4	Cylindrical Spring Powered Impacter
Can be Operated Robotically	17				9	1		3	9	9		1	1
Operable in Salt Water	11	1	1									5	3
Pressure Resistant to 10,000 ft	15						9					-	3
Provides Repeatable Impact	9	3	3	3				3	3	3		-	5
Resistant to Corrosion	6					1						2	3
Provides Specified Impact Quality	9	3	9	1								6	3
Can Provide Multiple Impacts per Dive	9	1						9				-	-
Provides Specified Impact Magnitude	11	9	9	9		1		1				7	3
Simple Design	2				3	3						9	4
Compact Size	3	3		3	9	3						-	3
Lightweight	2	3		3		9						-	2
Can Operate on 24V DC at 2 Amps	6	1	1	1				1				-	-
Measurement Unit		N	Hz	s	m	N	MPa	#	#	m			
Target Value		30	0-25	0-.01	0.5	15	35	10	5	1			
Importance Rating		**2**	**1**	**7**	**3**	**8**	**8**	**4**	**5**	**5**			
Total		194	224	156	186	67	135	176	180	180			
Normalized		0.130	0.150	0.104	0.124	0.045	0.090	0.117	0.120	0.120			

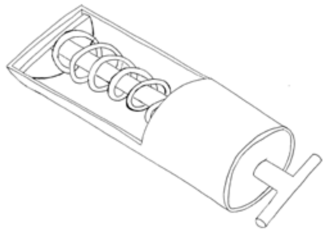
APPENDIX B – GANTT CHART



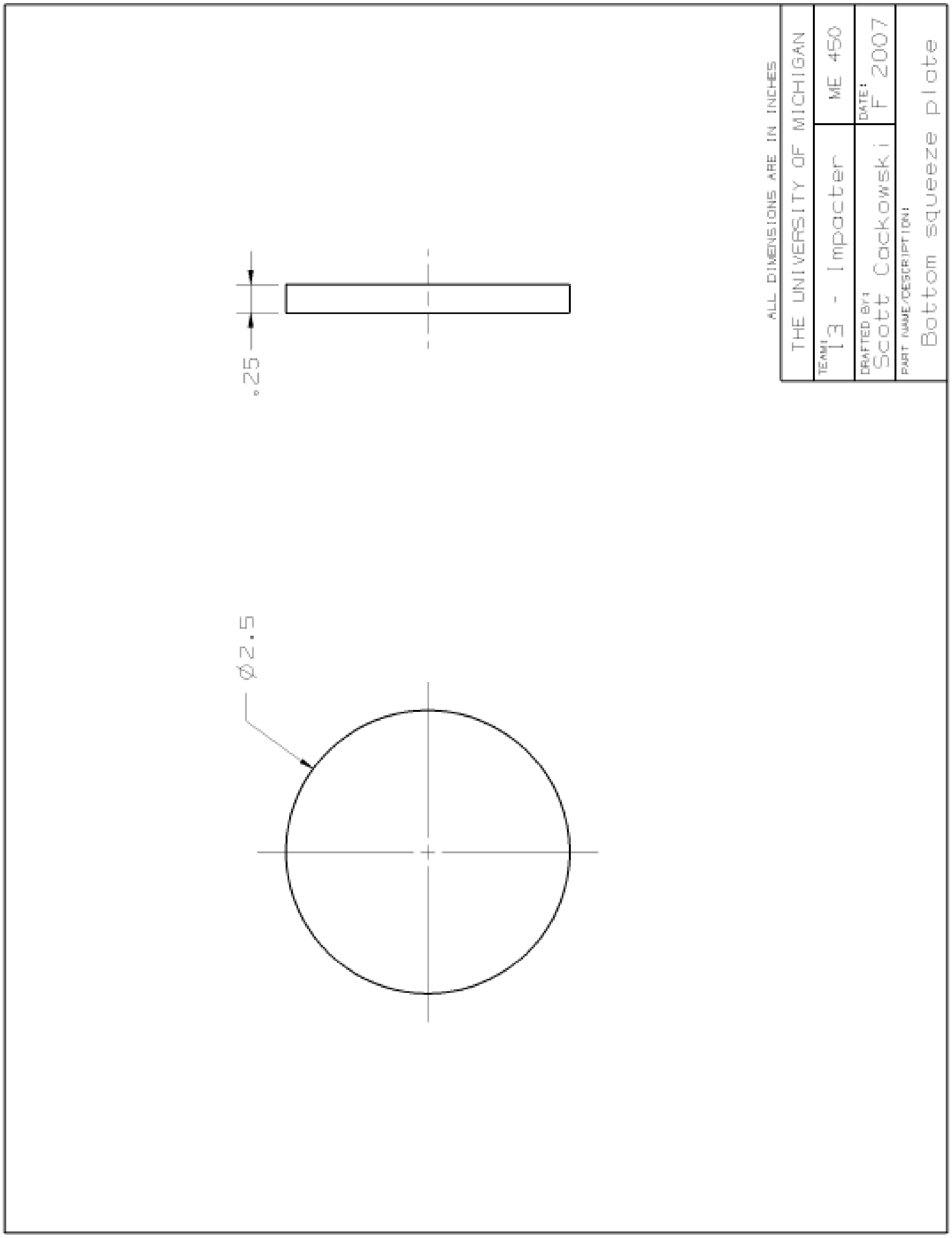
APPENDIX C – CONCEPTUAL IMPACTER DESIGNS

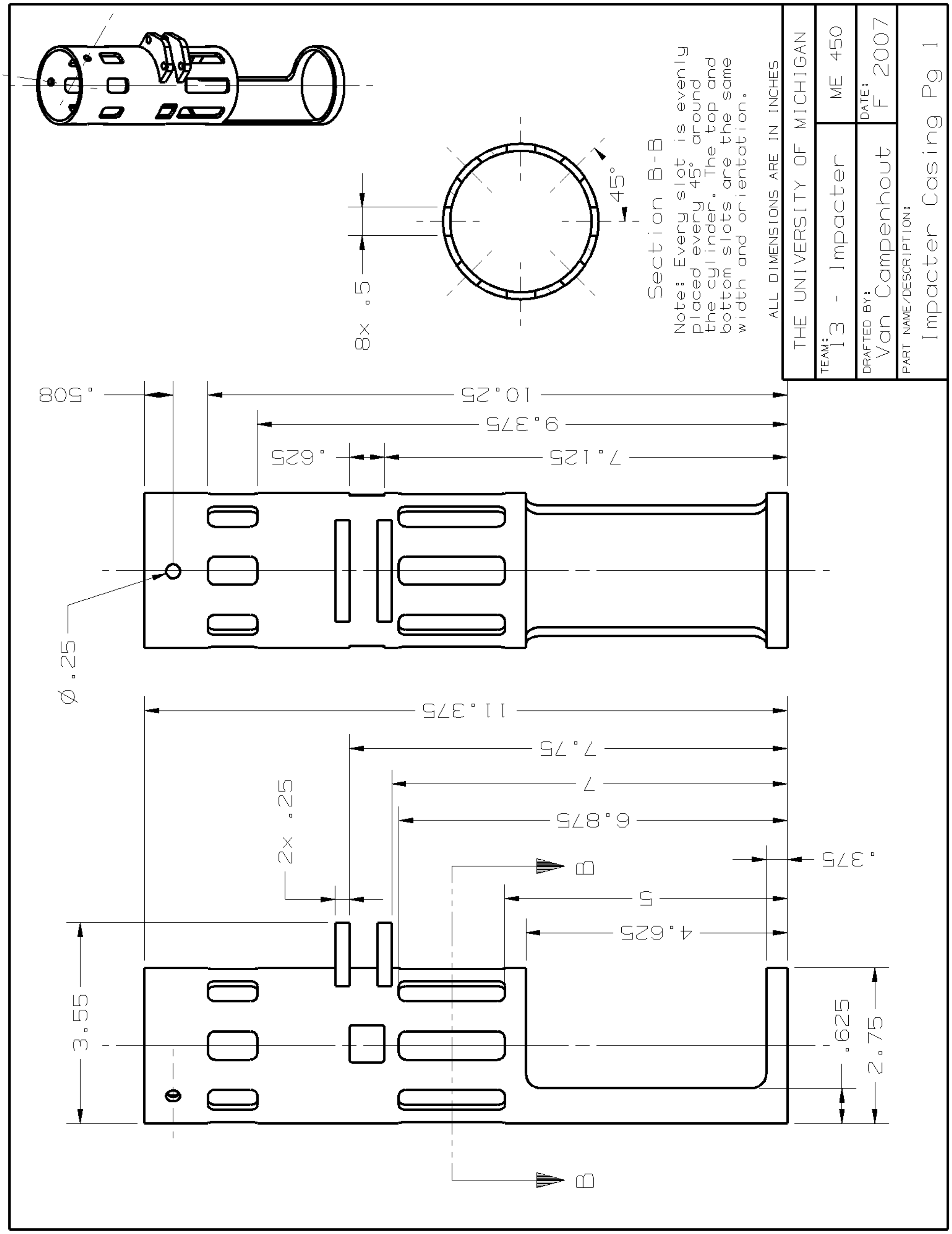
Design	Sketch	Description
Flywheel		By accelerating and then rapidly stopping a flywheel, an impact is generated.
Rotary Hammer		A high torque motor accelerates a hammer, which directly impacts the cable.
Guitar Pick		A pliable material is pulled over the cable inducing a wave similar to a guitar pick and string.
Pressurized Gas		By opening a vessel containing pressurized gas or depressurized gas (atmospheric pressure), a reaction is imparted on the device.
Linear Actuator		An electronic actuator drives a mass towards an impact surface, and then returns it to its starting position.
Squeeze Charge		Using the power of the robotic arm's gripper, a spring is compressed. A separate trigger on the device releases the spring , which drives a mass, creating an impact.

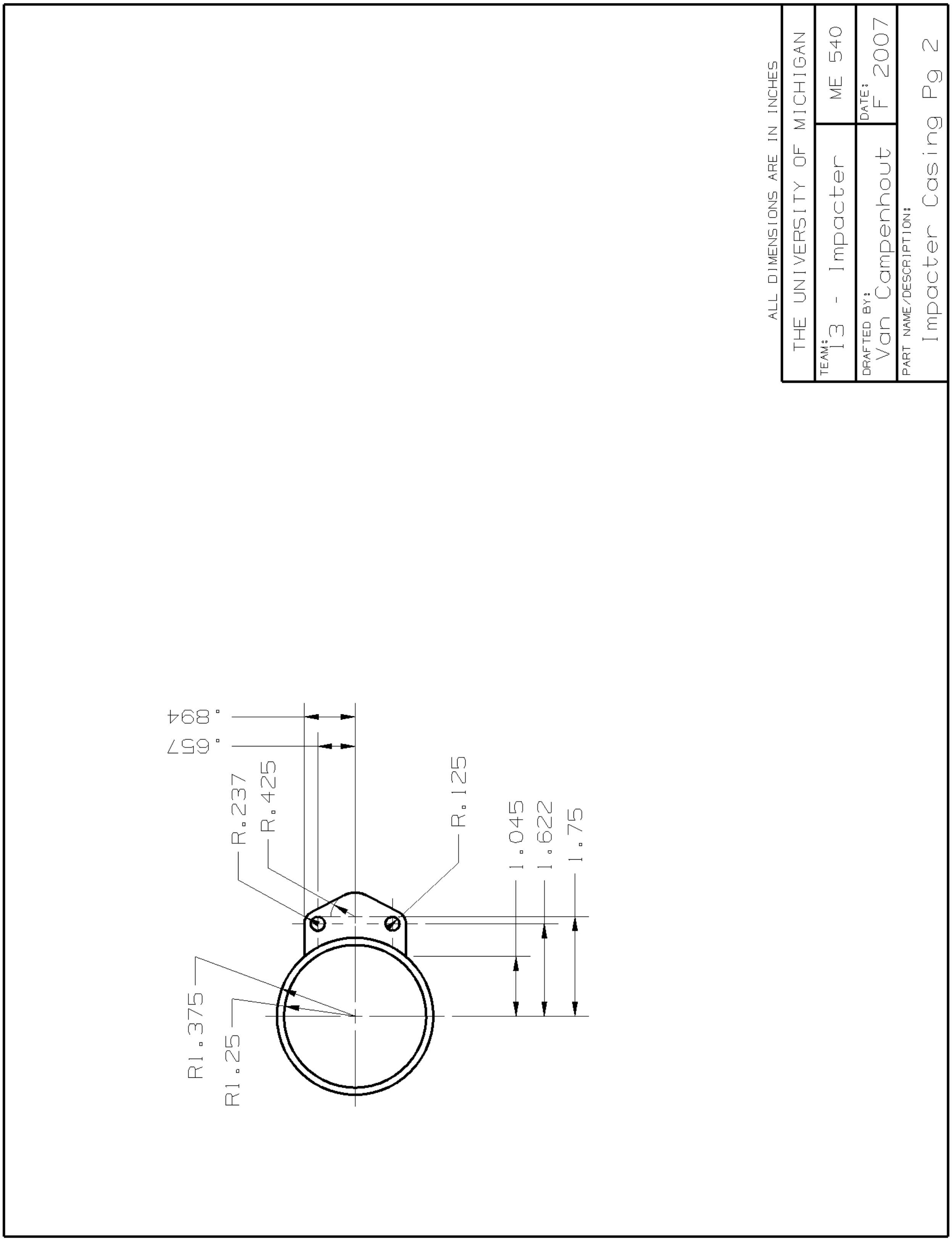
Piezoelectric		Highly controllable linear piezoelectric motors are used to create an impact.
Hydraulic/Pneumatic		The ROV’s onboard hydraulic/pneumatic system is used to drive a linear actuator, creating an impact.
Solenoid		Electric power from the ROV powers a solenoid. The solenoid’s pin then impacts a solid surface, creating an impact.
Cartridge		A chemical charge (similar to that in a bullet) is used to generate an impact in a manner similar to that used in a nail gun.
Torsion Spring		A motor charges a torsion spring, which in turn, drives a rack and pinion system. This allows the ships relatively low power electric system to be used more effectively.

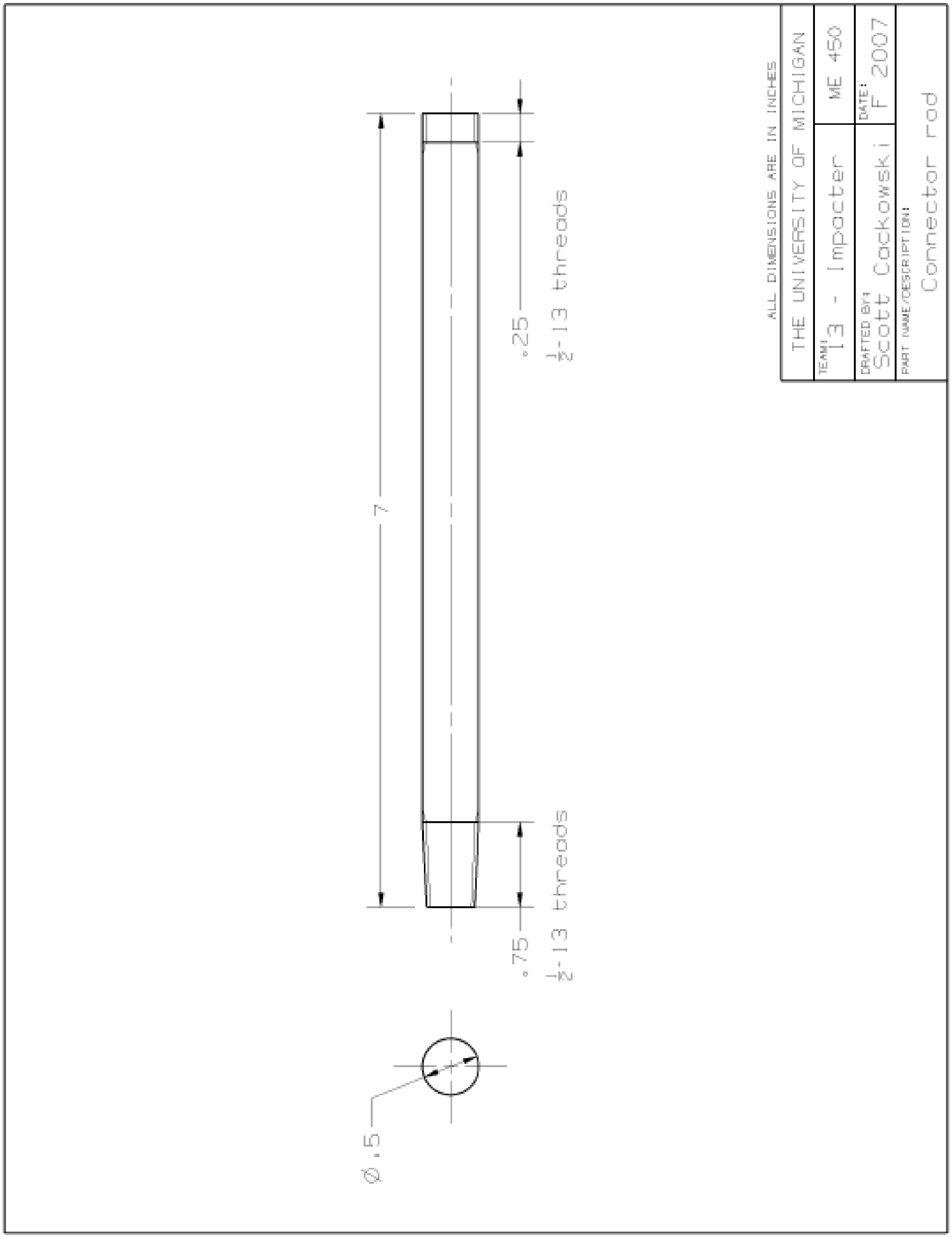
Pinball Plunger		Once attached to the cable, a handle on the impacter is pulled back by the ROV, which charges a spring. When released, the spring returns to its uncompressed state, driving a mass towards an impact surface.

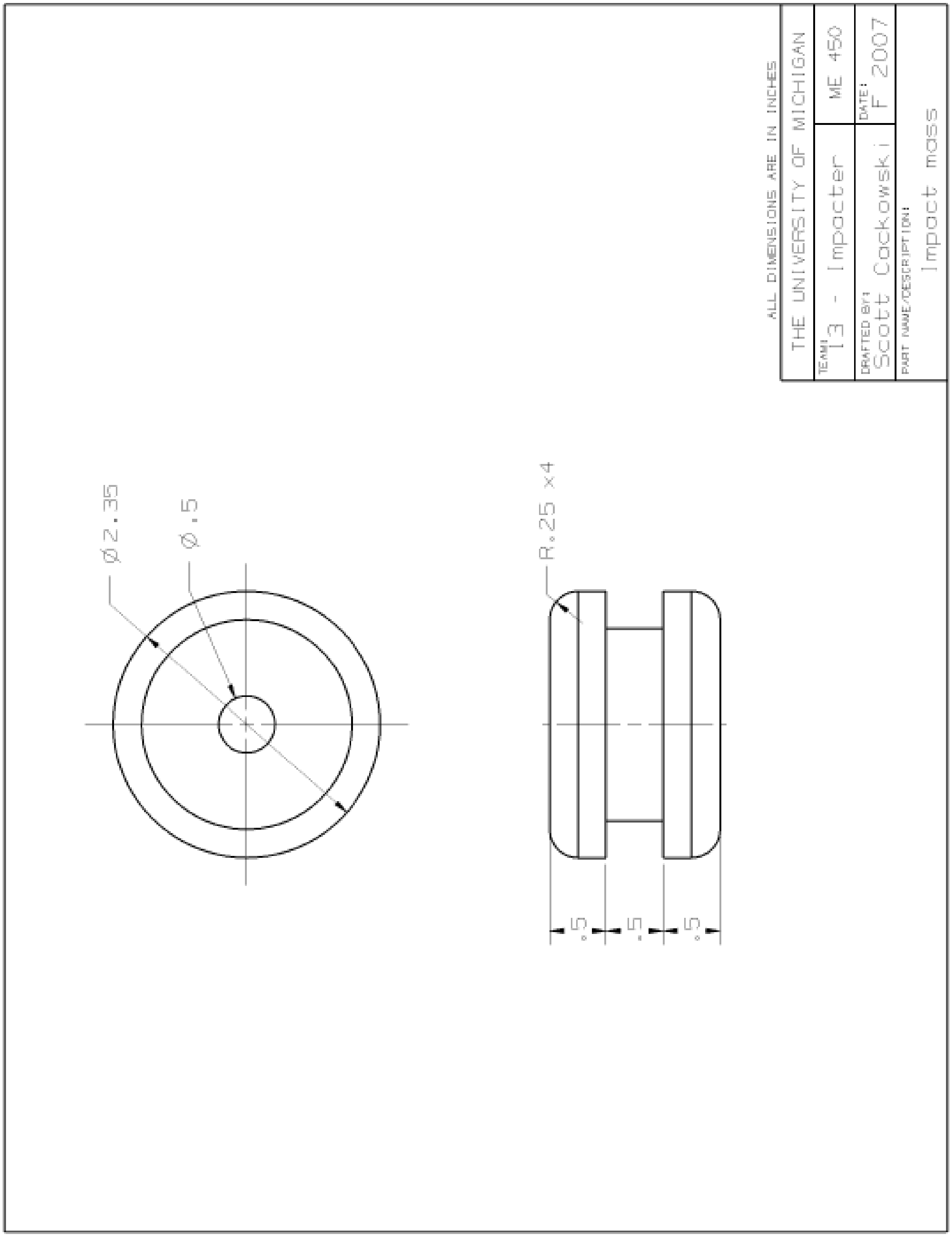
APPENDIX D – SELECTED CONCEPT DIMENSIONED DRAWINGS

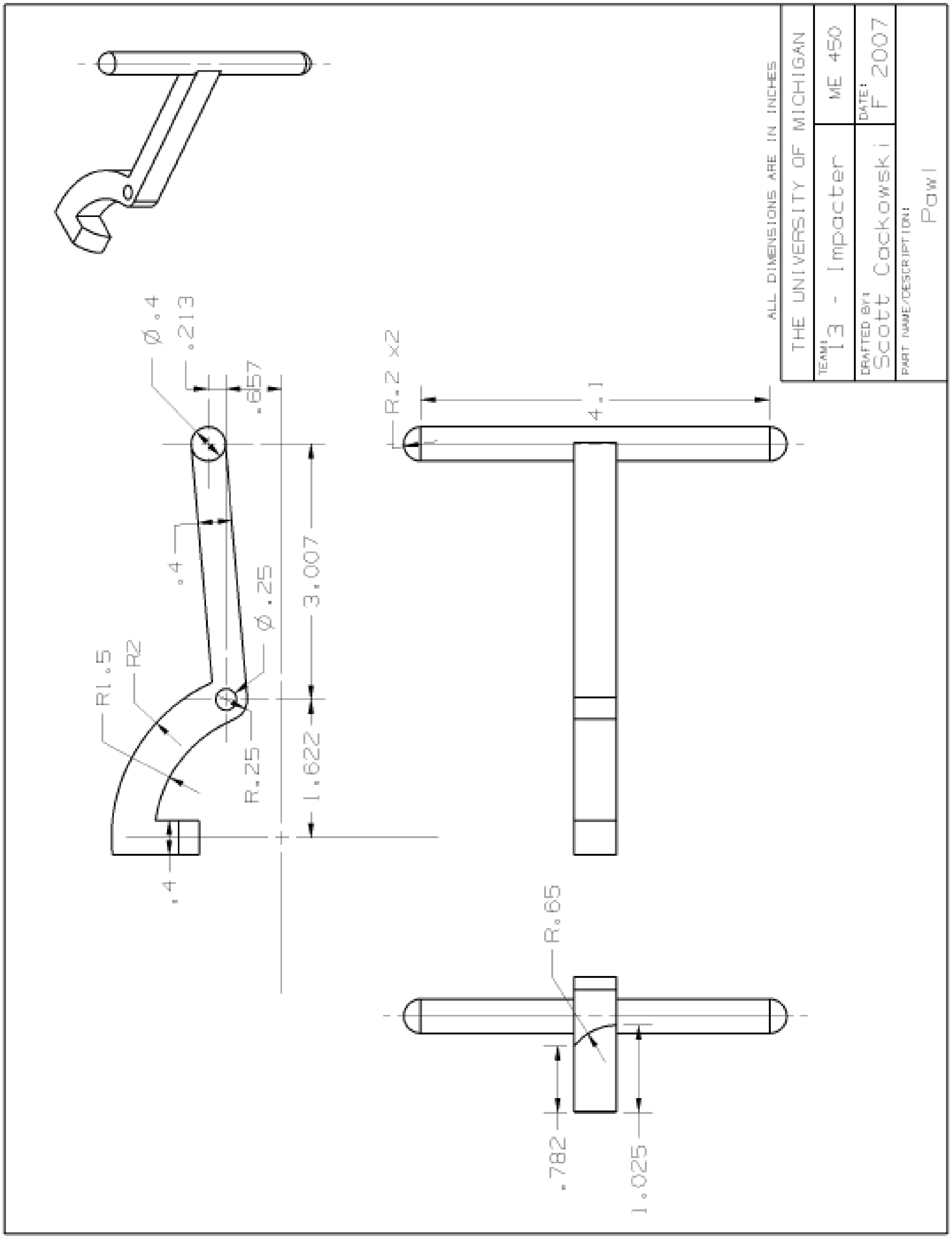


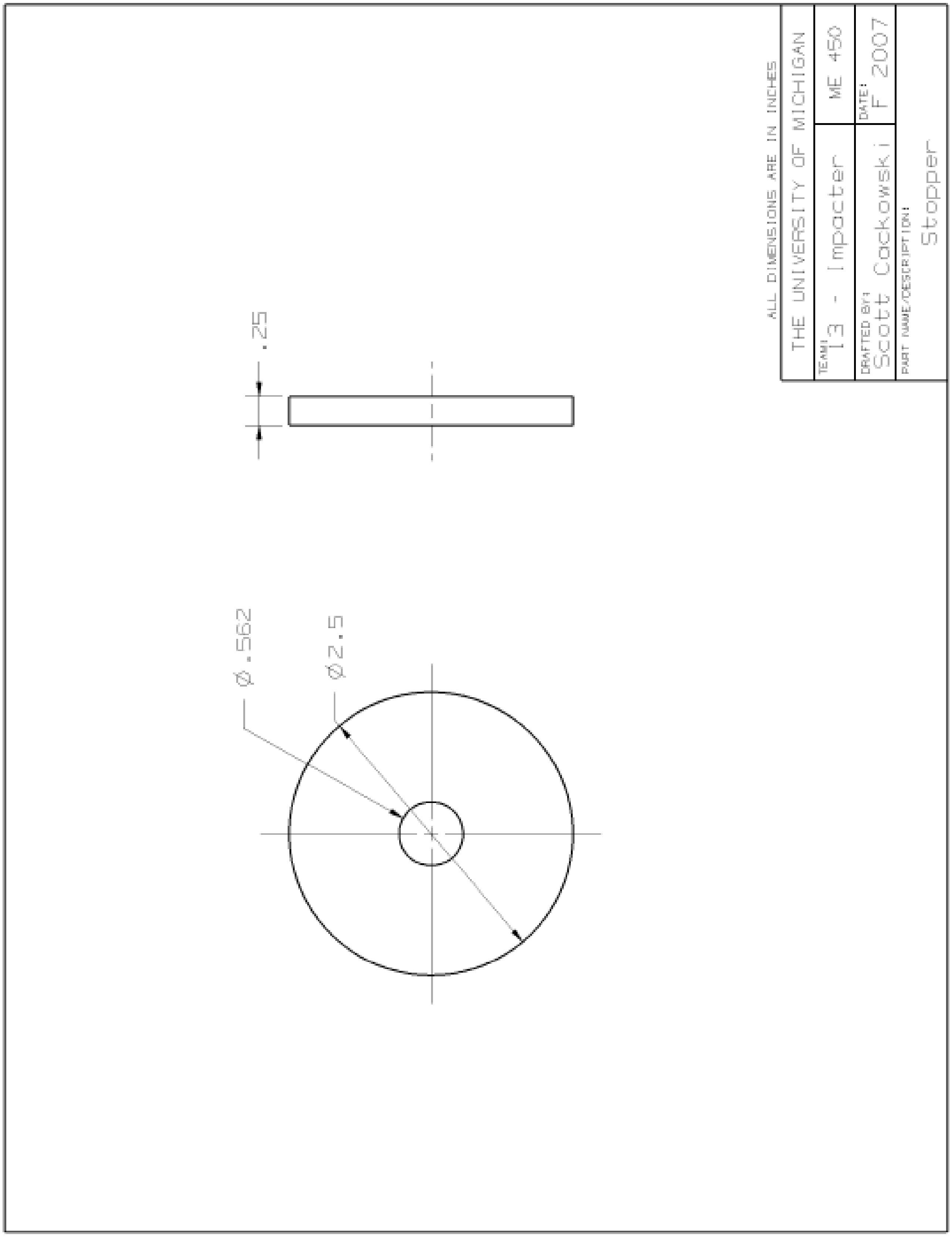

THE UNIVERSITY OF MICHIGAN
TEAM: 13 - Impacter
ME 450
DRAFTED BY: Van Campenhout
DATE: F 2007
PART NAME/DESCRIPTION: Impacter Casing Pg 1
ALL DIMENSIONS ARE IN INCHES
Note: Every slot is evenly placed every 45° around the cylinder. The top and bottom slots are the same width and orientation.
Section B-B
45°
8x .5
.508
10.25
9.375
7.125
.625
Ø .25
11.375
7.75
7
6.875
2x .25
.375
B
B
5
4.625
3.55
.625
2.75

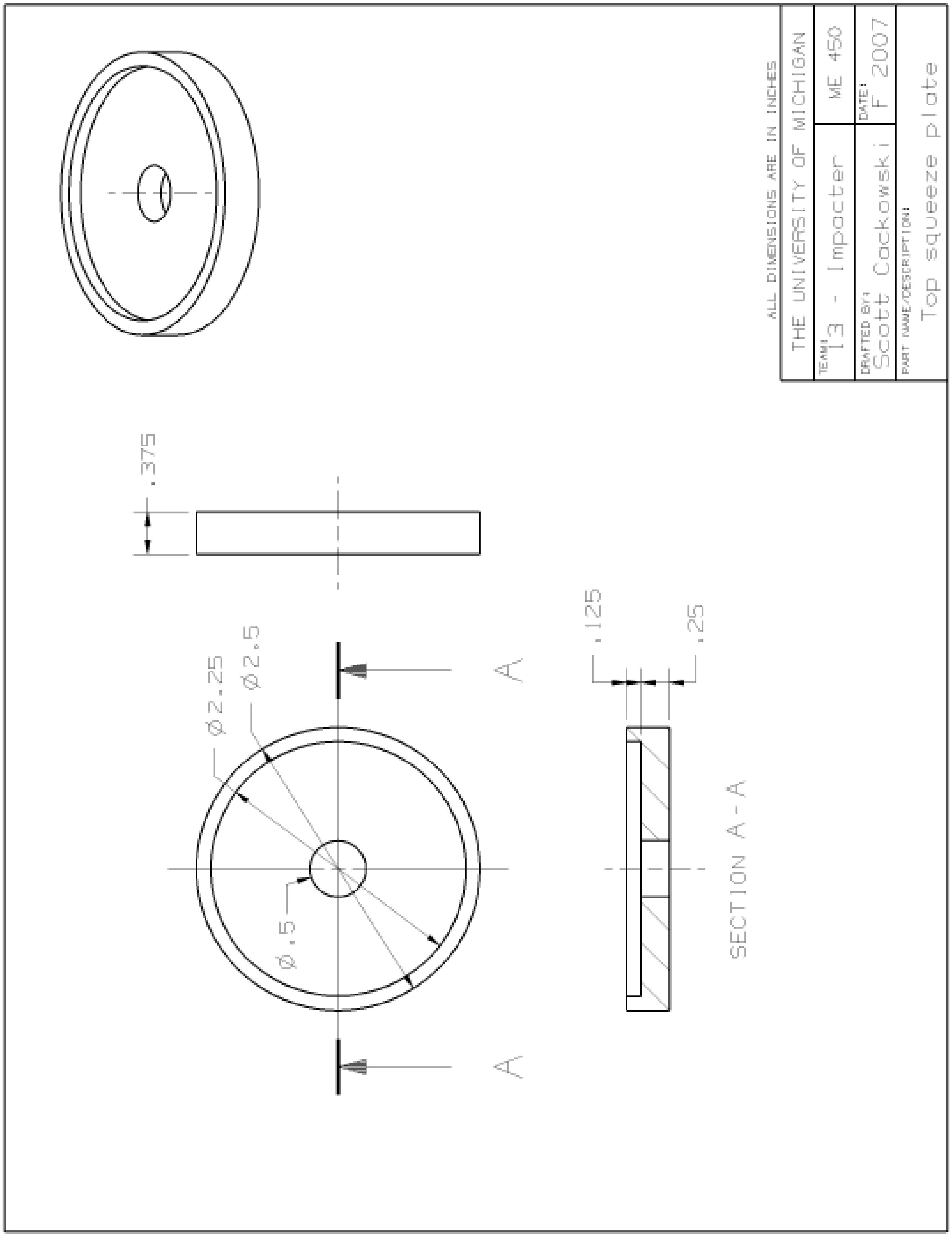

ALL DIMENSIONS ARE IN INCHES
THE UNIVERSITY OF MICHIGAN
TEAM: 13 - Impacter
ME 540
DRAFTED BY: Van Campenhout
DATE: F 2007
PART NAME/DESCRIPTION: Impacter Casing Pg 2
R1.375
R1.25
R.237
R.425
R.125
.894
.657
1.045
1.622
1.75


ALL DIMENSIONS ARE IN INCHES
THE UNIVERSITY OF MICHIGAN
TEAM: 13 - Impacter
ME 450
DRAFTED BY: Scott Cackowski
DATE: F 2007
PART NAME/DESCRIPTION: Connector rod
7
.25
½-13 threads
.75
½-13 threads
Ø.5


ALL DIMENSIONS ARE IN INCHES
THE UNIVERSITY OF MICHIGAN
TEAM:
I3 - Impacter
ME 450
DRAFTED BY:
Scott Cackowski
DATE:
F 2007
PART NAME/DESCRIPTION:
Impact mass
Ø2.35
Ø.5
R.25 x4
.5
.5
.5


ALL DIMENSIONS ARE IN INCHES
THE UNIVERSITY OF MICHIGAN
TEAM: I3 - Impacter
ME 450
DRAFTED BY: Scott Cackowski
DATE: F 2007
PART NAME/DESCRIPTION: Pawl
Ø.4
.213
.657
.4
Ø.25
3.007
R1.5
R2
R.25
1.622
.4
R.2 x2
4.1
R.65
.782
1.025


Ø.562
Ø2.5
.25
ALL DIMENSIONS ARE IN INCHES
THE UNIVERSITY OF MICHIGAN
TEAM:
13 - Impacter
ME 450
DRAFTED BY:
Scott Cackowski
DATE:
F 2007
PART NAME/DESCRIPTION:
Stopper


ALL DIMENSIONS ARE IN INCHES
THE UNIVERSITY OF MICHIGAN
TEAM: 13 - Impacter
ME 450
DRAFTED BY: Scott Cackowski
DATE: F 2007
PART NAME/DESCRIPTION: Top squeeze plate
.375
.125
.25
Ø2.25
Ø2.5
Ø.5
A
A
SECTION A-A

APPENDIX E – PROCESS PLANS

Top squeeze plate				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Trance circles 2.6” in diameter onto 0.375” plate aluminum	0.375” 6061 Plate Aluminum	Compass, Scribe, Ruler	-
2	Center punch center of circle	-	Center Punch	-
3	Center drill center of circle	-	Center Drill	Drill Press 550 RPM
4	Cut circle on band saw leaving a 0.1” edge from scored line	-	-	Band Saw 300 ft/min
5	Completely close chuck, compress circle between chuck and tailstock holding circle in place	-	-	Lathe
6	Remove material, cutting away roughly 0.002” each pass until part is round	-	Tool Post, Cutting Tool	Lathe 650 RPM
7	Once round, turn until 2.45” in diameter		Tool Post, Cutting Tool	Lathe 650 RPM
8	Clamp part in chuck	-	-	Lathe
9	Drill hole through center of part using tailstock	-	27/13 Drill Bit, Drill Chuck, Lubricant	Lathe 650 RPM
10	From front face turn out circle 2.325” in diameter from center 0.125” in depth	-	Tool Post, Cutting Post	Lathe 650 RPM
11	Remove part from chuck	-	-	-
12	Tap center hole	-	0.5/13 Tap with Handle, Vise	-

Stopper				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Trance circles 2.6” in diameter onto 0.25” plate aluminum	0.25” 6061 Plate Aluminum	Compass, Scribe, Ruler	-
2	Center punch center of circle	-	Center Punch	-
3	Center drill center of circle	-	Center Drill	Drill Press 550 RPM
4	Cut circle on band saw leaving a 0.1” edge from scored line	-	-	Band Saw 300 ft/min
5	Completely close chuck, compress circle between chuck and tailstock holding circle in place	-	-	Lathe
6	Remove material, cutting away roughly 0.002” each pass until part is	-	Tool Post, Cutting Tool	Lathe 650 RPM

	round			
7	Once round, turn until 2.45” in diameter	-	Tool Post, Cutting Tool	Lathe 650 RPM
8	Clamp part in chuck	-	-	Lathe
9	Drill hole through center of part using tailstock	-	5/8 Drill Bit, Drill Chuck, Lubricant	Lathe 650 RPM
10	Remove part from chuck	-	-	-

Bottom squeeze plate				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Trance circles 2.6” in diameter onto 0.25” plate aluminum	0.25” 6061 Plate Aluminum	Compass, Scribe, Ruler	-
2	Center punch center of circle	-	Center Punch	-
3	Center drill center of circle	-	Center Drill	Drill Press 550 RPM
4	Cut circle on band saw leaving a 0.1” edge from scored line	-	-	Band Saw 300 ft/min?
5	Completely close chuck, compress circle between chuck and tailstock holding circle in place	-	-	Lathe
6	Remove material, cutting away roughly 0.002” each pass until part is round	-	Tool Post, Cutting Tool	Lathe 650 RPM
7	Once round, turn until 2.45” in diameter		Tool Post, Cutting Tool	Lathe 650 RPM
8	Clamp part in chuck	-	-	Lathe
9	Remove part from chuck	-	-	-

Plunger Mass				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Place 6” long piece of 3” round stock in lathe chuck	6061 Aluminum Round Stock	-	Lathe
2	Face end of stock	-	Tool Post, Cutting Tool	Lathe 650 RPM
3	Turn until outside diameter is 2.35”	-	Tool Post, Cutting Tool	Lathe 650 RPM
4	Begin cutting a slot ___ deep starting 0.75” from faced end of stock, only working radially	-	Tool Post, Parting Tool	Lathe 650 RPM
5	Continue to widen slot until slot is 0.25” wide	-	Tool Post, Cutting Tool	Lathe 650 RPM
6	File stock end to remove a 0.25” radius	-	File	Lathe 650 RPM
7	Remove part from lathe	-	-	-
8	Cut off extra material 2” form finished end	-	-	Band Saw 300 ft/min
9	Place finished end into lathe chuck	-	-	-

10	Face off end of cut stock until part 1.75” long	-	Tool Post, Cutting Tool	Lathe 650 RPM
11	File end to 0.15” radius	-	File	Lathe 650 RPM
12	Drill hole through part using tailstock	-	Drill Chuck, 27/13 Drill Bit	Lathe 650 RPM
13	Remove part from chuck	-	-	-
14	Tap center hole	-	0.5/13 Tap, Tap Handle, Vise	-

Connector rod				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Cut 0.5” round stock to a 7” length	6061 Aluminum	-	Band Saw 300 ft/min
2	Place part in lathe chuck	-	-	Lathe
3	Use tailstock to hold threading die against the end of part. Manually rotate chuck and tail stock in opposite directions until die catches.	-	0.5/13 Threading Die, Die Handle	Lathe
4	Back off tailstock, manually turn chuck until die has completely cut 0.25” of the part	-	0.5/13 Threading Die, Die Handle	Lathe
5	Rotate chuck backwards until die is removed.	-	-	-
6	Remove part from chuck, insert threaded end into chuck and retighten	-	-	Lathe
7	Repeat steps 3-5 except completely thread 1” of part.	-	0.5/13 Threading Die, Die Handle	Lathe
8	Remove part from chuck	-	-	-

Pawl				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Score outline for pawl on material	6061 0.5” Plate Aluminum	Scribe	-
2	Cut out part	-	-	Band Saw 300 ft/min
3	Cut chamfer on locking surface 0.25” radius	-	File	-
4	Drill hole for pins	-	B Drill Bit	Drill Press 550 RPM
5	Cut 0.5” round stock to a 6” length	6061 0.5” Round Stock Aluminum	-	Band Saw 300 ft/min
6	Mill 0.5” diameter indent into handle end of pawl	-	Vice, 0.5” End Mill	Mill
7	Weld handle onto pawl	-	TIG Setup	TIG Welding 100 Amps

Pawl Support				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Score outline for pawl support on material	6061 0.25” Plate Aluminum	Scribe	-
2	Cut out part	-	-	Band Saw 300 ft/min
3	Drill hole for pins	-	D Drill Bit, Vice	Drill Press 550 RPM
4	Ream holes	-	0.25” Ream, Vice	Drill Press 550 RPM

Casing				
Step	**Process**	**Material**	**Tools Required**	**Machine/Setting**
1	Cut 2.75 OD stock to a 11.75” length	6061 2.75” OD, 0.125 wall aluminum tube	Band saw	Band Saw 300 ft/min
2	Weld bottom squeeze plate 0.125” deep into tube stock	Bottom casing surface	TIG welder	TIG Welding 100 Amps
3	Weld pawl supports 5” from bottom of stock, perpendicular to the outer surface	Pawl supports	TIG welder	TIG Welding 100 Amps
4	Score outline for 4” cutout section	-	Scribe	-
5	Secure part in vise			
6	Remove cutout section roughly	-	Band saw	Band Saw 300 ft/min
7	Secure part in vise on mill stage			
8	Remove cutout section precisely	-	Mill	Spindle speed 550 RPM
9	Weld stopper flush with top of cutout cavity	Stopper	TIG welder	TIG Welding 100 Amps
10	Secure part in vise on mill stage	-	-	-
11	Cut top and bottom flow holes	-	Mill	Spindle speed 550 RPM
12	Rotate part 45 degrees in mill vise	-	-	-
13	Repeat steps 11 and 12 for perimeter of part	-	-	-
14	Secure part in vise on mill stage	-	-	-
15	Drill clearance hole for cap screws	-	Mill, Drill press	
16	Rotate part 130 degrees	-	-	-

17	Drill clearance hole for cap screw	-	Mill	-
18	Rotate part -160 degrees	-	-	-
19	Drill clearance hole for cap screw	-	Mill	-

APPENDIX F – QUANTITATIVE ANALYSIS SUMMARIES

Spring Analysis Summary:

Max Original Energy Stored:	38.44	lbs*in
Min Original Energy Stored:	11.31	lbs*in
Spring Constant:	64	lbs/in
Maximum Displacement:	1.25	in
Max Force Required:	80	lbs/in
Max Energy Stored:	50	lbs*in
Max Energy Stored in .125" Pretension:	0.5	lbs*in
Max Energy Stored with with 1.125" Travel:	40.5	lbs*in

Flow Holes Analysis Summary:

Diameter of Mass:	2.5	in
Area of Mass:	4.91	in^2
Number of Holes Per End:	8	-
Hole Width:	0.5	in
Hole Length:	1.5	in
Effective Hole Length:	0.75	in
Hole Area:	6	in^2
Effective Hole Area:	3	in^2
Percentage of Mass Area:	122	%
Effective Percentage of Mass Area:	61	%

The chart above refers to lengths and areas both generically and as “Effective”. The generic dimensions refer to the total size of the holes, and the effective dimensions refer to the size of the holes closest to the clamp end when the mass is completely extended. This is an important dimension because it reflects the point at which the minimum area is provided for water to be ejected from the impacter cavity.

Cap Screws Analysis Summary:

Maximum Energy Absorbed:	50	in*lbs
Minimum Impact Distance:	0.125	in
Maximum Impact Force:	400	lbs
Material:	Stainless 316	-
Minimum Yeild Strength:	24.6	ksi
Maximum Shear Force:	400	lbs
Number of Bolts:	3	-
Maximum Shear Force/Bolt:	133	lbs
Bolt Diameter:	0.25	in
Bolt Area:	0.0491	in^2
Shear Stress:	2716	psi
Safety Factor:	5.43	-

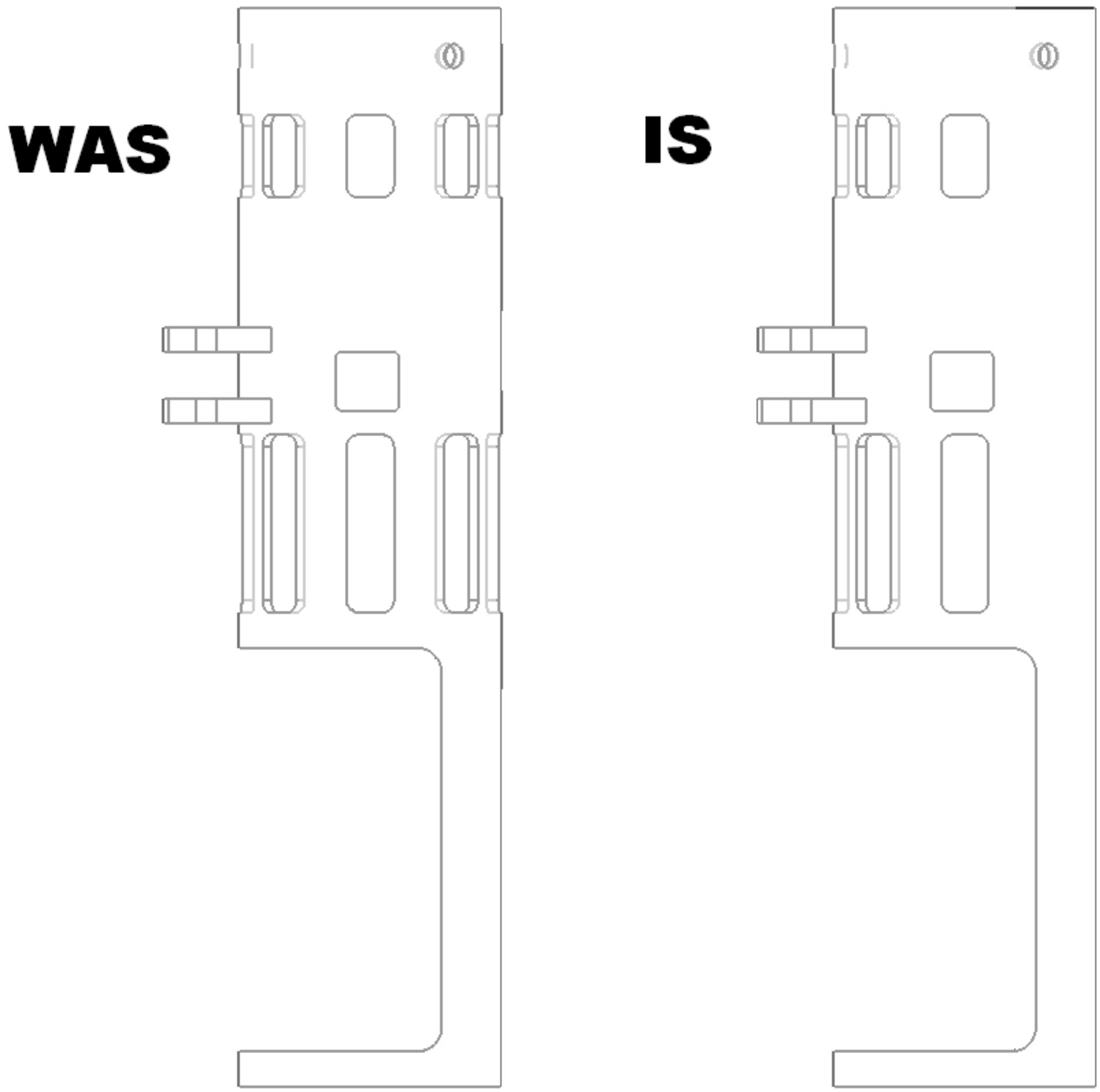
Connector Rod Analysis Summary:

Diameter:	0.5	in
Material:	6061 Aluminum T6511	-
Minimum Yeild Strength:	27.99	ksi
Maximum Tension Force on Rod:	150	lbs
Maximum Tension on Rod:	763.9437268	lbs/in^2
Rod Safety Factor:	36.63882432	-

APPENDIX G – FMEA

Part Name	Part Functions	Potential Failure Mode	Potential Effects of Failure on Impacter	Potential Other Effects of Failure	Severity (S)	Potential Causes/Mechanism(s) of Failure	Occurance (O)	Design/Control Tests	Detection (D)	Recommended Actions	Old RPN	New S	New O	New D	New RPN
Trigger Function	Hold and Release Impacting Mass	Fracture/Material Yield	Trigger mechanism not functional	No effects on property because no forces exceed that which is necessary to damage ROV. No hazards to human users because there are no human users, impacter is used remotely. Negligible impact on environment because the worst case scenario is a lost impacter on the bottom of the ocean.	8	When charged large shear stress on pawls	5	Estimate forces, engineering analysis	3	Make pawls thicker to reduce chance of yield	120	5	1	3	15
		Unstable	Possible failure of trigger mechanism		6	When charged large force on pawls	9	Estimate bending moments	3	Change design to eliminate, minimize moments	162	4	1	3	12
Impacting Mass	Provide Impact for Cable	Seizure	No impact can be delivered		9	When released, if bending will jam	7	Test during manufacturing	3	Add guide, decrese size of mass	189	9	3	2	54
		Unbalanced	Repeatability is negatively affected		5	If not centered, may jam	7	Test during manufacturing	2	Identify problem part(s), remanufacture	70	3	2	2	12
		Loosening	Possible separation of mass and plunger		7	Multiple impacts could loosen threaded connection	4	Repeated testing	7	Weld in place once dampening material is selected	196	4	1	7	28
		Vibrations	Poor spectral content, will affect reading		8	Impact surfaces could cause high frequency vibrations upon impact	9	Test after manufacturing in lab	1	Replace dampening material until desired result is obtained	72	5	1	1	5
Spring	Provide Energy for Impact	Corrosion	Could affect performance		3	Use in salt water	9	Material selection, research	1	Use Aluminum to reduce corrosion	27	1	4	1	4
		Fatigure	Reduced force of impact, eventual failure		4	Repeated use	3	No test is feasible, check with manufacturer	3	Change design so the spring is replaceable	36	1	3	3	9
Plunger	Interface Between Robot and Impacting Mass	Wear	Poor appearance		2	ROV grasping could damage plunger	10	Wear is inevitable	1	No action taken	20	2	10	1	20
		Material Yield	Would make charging more difficult for operator		6	ROV grasping could damage plunger	4	Estimate forces, FBD	3	Change thickness of plunger to minimize chance of yield	72	4	1	3	12
Clamp	Hold Impacter Against Cable	Loosening	Would force re-attaching, could affect reading		8	Impact could cause loosening	7	Test during manufacturing	3	Add torsion springs to increase gripping force	168	8	2	3	48
		Corrosion	Could affect performance		3	Use in salt water	9	Material selection, research	1	Use Aluminum to reduce corrosion	27	2	4	1	8
		Unstable	Could force re-attaching, could affect accelerometer readings		6	Impact could cause loosening	8	Test during manufacturing	3	Add torsion springs to increase gripping force	144	6	2	3	36
Casing	Provide Stability	Corrosion	Could affect performance		5	Use in salt water	9	Material selection, research	1	Use Aluminum to reduce corrosion	45	2	4	1	8
		Leaking	Reduced force of impact		7	If closed casing leaks, pressure difference will reduce force of impact	6	Test after manufacturing	1	Mill slots along sides of casing to promote flow	42	2	1	1	2

APPENDIX H: ENGINEERING CHANGE NOTICE



The number of vent holes in the outer casing of the design had to be altered during the manufacturing stage. After welding the pawl supports into place on the casing, we began to machine the vent holes. We soon realized that we would be unable to safely clamp our casing in place in order to machine the side opposite the supports. After a quick discussion with the group, we determined that the need for vent holes around the circumference of the casing was strictly aesthetic, and thus, it would not affect the performance of the prototype.