NORTHERN ARIZONA UNIVERSITY
Mechanical Engineering

Team 19F18
NG Standoff

To: Dr. Oman and Ulises Fuentes

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Date: March 1, 2020

Subject: Implementation I Memo

The purpose of this memo is to discuss the Northrop Grumman Standoff Capstone Team's implementation of manufacturing the final prototype. Northrop Grumman's Flight Systems Group has requested the team to design, analyze, and build a prototype universal dome standoff bonding tool that can be mounted to attach rings of variations of rockets that will hold standoff brackets in place while the adhesive cures. This memo serves to go over specific major implementation milestones such as methods of manufacturing necessary to complete the product, design changes, future manufacturing plans, and the team's schedule and budget breakdown.

1 Implementation

This section will go into detail on the steps taken to ensure that the device meets the engineering requirements and what types of manufacturing will be implemented to create the final product. The types of manufacturing methods needed to create the final product include the CNC mill, vertical mills, and lathes located in the NAU machine shop, and an outside third-party to apply coatings on the designated surfaces. The design changes made to the device as a result of the information received since the conclusion of last semester have also been justified through calculations and inspection.

1.1 1.1 Manufacturing

This section will focus on the manufacturing implemented for the production of the final product. This will include the machines utilized to fabricate the sub-assemblies as well as the methodologies utilized to ensure the quality of those parts. As of now the team has strictly machined on the vertical mills and lathes that are located in the 98C machine shop. After the preliminary design presentation at the Northrop Grumman headquarters, the senior engineers that work there informed the team that the assembly of the design should minimize the usage of the CNC parts to reduce cost and complexity of the design.

The majority of the design has been machined on the vertical mills as the product was created in Solidworks with simplicity in mind. The design minimizes curved edges as they are almost impossible to create on a vertical mill and would need to be placed in a CNC to accurately mill out of raw material. Straight edges, square bodies, and tight tolerances ensure that the fitments of all the pieces are accurate and move together as the team intended. The current rail cart is composed of a milled C-channel with a flat plate completing the rectangular cart. Tight and accurate tolerances in the parts resulted in a very nice fit to the teams selected rectangular rails that translate the cart across the motor dome. The mill has also helped the team drill counterbores, counter sinks, and cut threads where needed.

The other major machining done on the device has been on the machine shop lathes. The team has constructed the large threaded knobs, axles, and bolts that are critical components of the design on the lathe. The lathe also makes it possible to add chamfers to axles to allow them to fit into slots easier and ensure proper fitments. The other large piece of machinery in the machine shop that hasn't been implemented yet is the CNC mill. This is a computer controlled mill that takes an inputted code and cuts accurate complex shapes into the stock material.

Moving forward with the project, the team will continue to use both the vertical mill and the lathe to machine the majority of the final product. More straight edges and square bodies will be machined as well as pins and thumb screws for the design. However, the CNC will be required as precise curves will need to be cut for the pieces that clamp to the motor ring. The motors that are used by Northrop Grumman have various diameters and the clamps that the team is going to use have to match the diameters exactly to ensure a proper fit and the surface area is maximized. To achieve the proper fitment, the large CNC has to be used and since only one member of the team has CNC experience on the small mills, work orders will have to be placed so the shop managers can construct the parts needed.

Before final parts were created by the team, test samples were created to test fitments between parts of various sizes. For the axles that allow the brackets to rotate, the team began with a twenty thousandths fitment and analyzed the performance. It was clear that the fit was too large and the axle wiggled more than it rotated. The second test was done with a ten thousandths fit which was noticeably better but still allowed for some rocking between the parts. Lastly, a five thousandths fit was created and tested which resulted in a near perfect fit where the axle rotated nicely with extremely minimal rocking. For the rail cart and the rail system, a first test of twenty thousandths resulted in a nearly optimal fit that the team was satisfied with. These machined test fits helped the team explore and understand what fitments to implement for the rest of the produce.

1.2 Design Changes

As discussed previously in this report, Northrop Grumman requested that the capstone team simplify their designs from the PDR presentation to minimize the complexity of the parts while machining and reduce the cost of manufacturing. Along with this, the client has added two project requirements which will be discussed in greater detail in this section. Due to the manufacturing and project requirements that have been made by the client, problems have occurred which required design changes. This section will discuss in detail each design change along with justification. Included in these changes are CAD pictures of each iteration, justification for the changes made, and calculations if necessary to back up the current state of the designs.

1.1.1 1.2.1 Design Iteration 1: Change in rocket motor clamping

Initially, the ring clamp design focused on the ability to interchange clamp jaws, which were built around the geometry of each motor ring, within a streamlined mounting system. This design can be seen in figure (1) below.

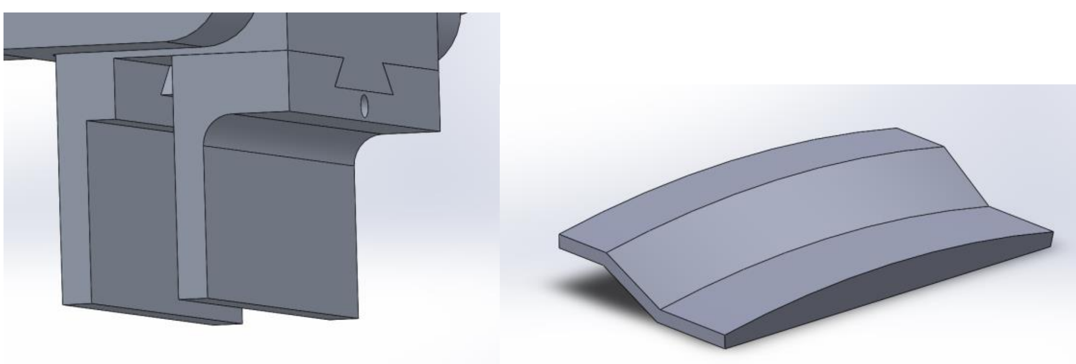


Figure 1: PDR Interchangeable Clamp Jaws

However, this design required extensive CNC machining along with complicated bolted connections and bulky clamping jaws. In order to simplify the operation of the ring clamp, the team decided to integrate the ring geometry into the clamping system itself, which reduced the total machining work while improving the usability of the design. The new design also eliminates the dovetail connection that was initially intended for the system and allows for screws to pass through the existing holes in the motor ring clamp. The slot seen in figure (2) gives the option to position the clamp at any point on the ring while also ensuring that the clamp remains secure during operation.

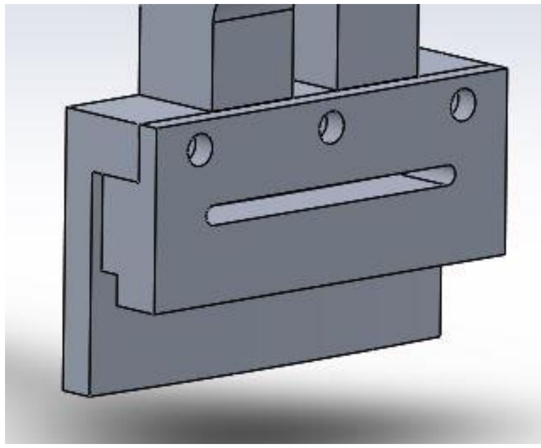


Figure 2: Current Rocket Motor Clamp Design

In order to verify that the clamping design will not cause damage to the rocket motor ring, an FEA was performed to determine the stresses and deflections of the ring when loaded. The FEA seen in Appendix A determines that the factor of safety for this design was 42 which is significantly higher than the 3.0 minimum specified in the project requirements. This analysis included the positioning of the holes located around the rocket motor ring and not just a solid round ring as the holes might be points of failure and stress concentrations.

1.1.2 1.2.2 Design Iteration 2: Change in angling mechanism discussion

The mounting arm must be able to attach to several different rocket motors: Orion 38, Orion 50XL, and Castor 30XL. Due to the curvature of the Castor series rocket motor domes, as seen in figure (3), the device requires an angling mechanism to clear the protrusion of the rocket motor dome.

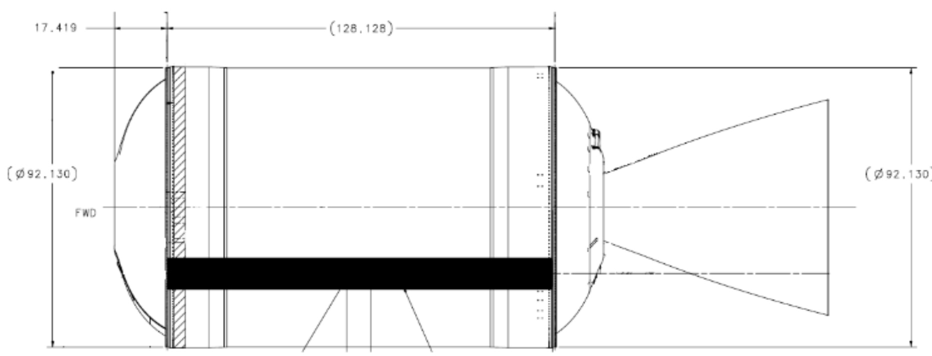


Figure 3: Castor 30 Series Drawing

Originally, the capstone team designed a spring loaded spline shaft mechanism that would allow the hinge section to adjust to multiple angles to conform to the rocket dome profiles. When the splined portion was pulled out, the two hinges would rest on a axle with the outside diameter of the splined shaft to retain hinge alignment. This design can be seen in figure (4) below.

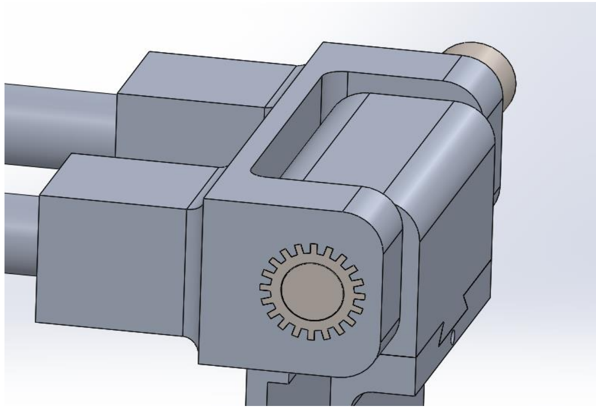


Figure 4: PDR Spline Shaft Mechanism

After presenting the PDR design to Northrop Grumman, one of the main issues they had with the overall design was that the device was complicated to manufacture and should be simplified. Since the spline shaft seen in figure (4) was likely to be outsourced due to the complicated geometry of the design, the team redesigned the angling mechanism and created the current design seen in figure (5).

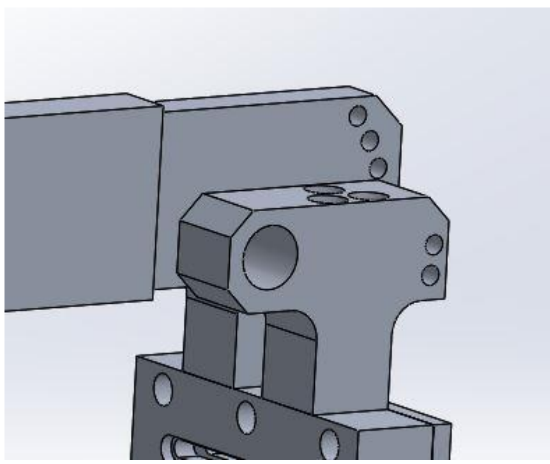


Figure 5: Current Set Pin Angle Mechanism

The current angle mechanism works with six pin slots, located 9 degrees apart from each other that allow the device to increase or decrease its operating angle appropriately. A large pin will serve as the rotating axle that will go through all three of the hinge pieces. The side hinge pieces are attached to the rail system with a set of bolts that allow the device to stay permanently attached to the rocket motor clamps. The center hinge piece, which has five angling holes drilled into the part to allow for the change in angle required of the design is also pinned to the rectangular rail. These quick detach pins will allow the technician to put different length rails onto the design.

This device is less complicated than the previous spine shaft design, however the design does require the use of pins which could result in shear failure. A single pin will be used to resist moment from the entire rail cart lever arm. Since one pin will be used, through a total of three plates, double shear will be imparted to the pin. With a max load of 50lbs resulting in a 360lb internal shear on the pin, a diameter of .207 inches is required for the pin to meet the desired factor of safety of 3.0. Currently the team will use a pin of 0.25 inches, which exceeds the required factor of safety. These calculations can be seen in Appendix B.

1.1.3 1.2.3 Design Iteration 3: Change in rail system discussion

In order to meet the customer requirement of positioning standoff template brackets 4 to 36 inches inboard of the rocket motor ring, the team originally designed a dual rail system, as seen in figure (6).

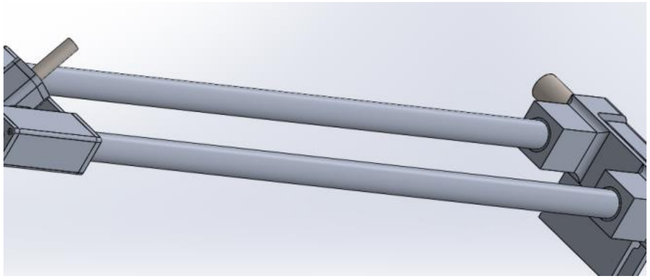


Figure 6: PDR Rail System Design

The original design consisted of two rails that were circular and hollow, with a 1.5” outside diameter and a 1” inside diameter. Instead of similarly sized solid rods, hollow tubing was selected to lower the overall weight of the design while retaining most of the moment of inertia. Cylindrical rails also allowed for the use of readily available linear bearings, which improved the sliding mechanism when repositioning standoff brackets. This design, however, was disregarded due to a change in the customer requirements for the project. As a result of the preliminary design review, the maximum deflection of the rails during operation was requested to be less than 0.1 inches. This was accompanied by a change of the maximum loading condition 50 lbs, which would have occurred during the pull test, to 120 lbs, which provided a 20 lb bonding force for each of up to six standoffs mounted to a bracket. To minimize deflection and handle the newly required maximum load, the intermediate design in figure (7) was introduced.

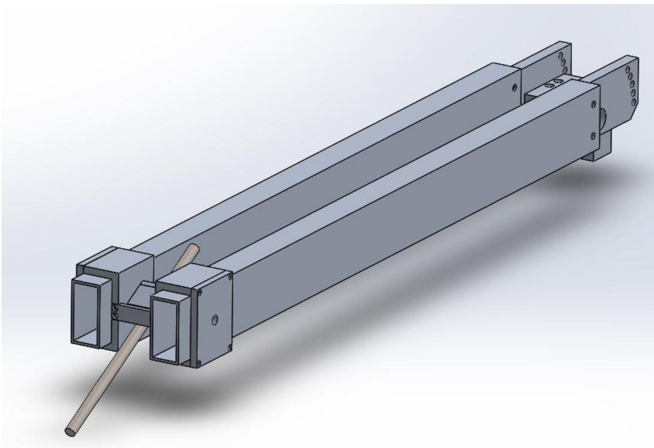


Figure 7: Intermediate Design Change

Due to a recent clarification, however, the design was altered yet again. A clarification in the bonding force, which should have been 20 lbs per entire bracket, allowed for the simplification of the design. With the maximum loading condition now consisting of the 50 lb pull test force, the heavy dual rail system could be reduced to improve operability of the design. However, the rectangular profile of the rail system was maintained to satisfy the requirement for less than 0.1 inches of deflection at the maximum load. The current design, which consists of a single, lightweight rail, is shown in figure (8) below.

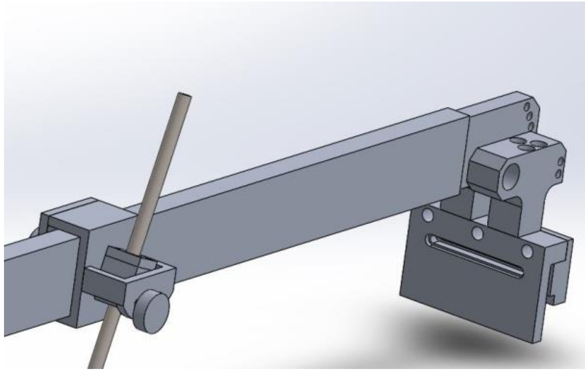


Figure 8: Current Rail System CAD

The singular rail that is to be used in the current design measures 3”x1” with a ⅛” wall thickness. In addition to decreasing the mass of the design, a single rail allows for the device to be more easily operated and reduces the setup time. This rail, in the newly defined maximum loading condition, is also expected to provide a deflection of .082”, less than the 0.1” deflection specified by the customer requirement. The calculations for these deflection values can be seen in Appendix C of this report.

1.1.4 1.2.4 Design Iteration 4: Change in rail cart discussion

Due to the rail system design created by the capstone team, a rail cart was made to allow operators to apply axial forces at set distances inboard of the rocket motor ring and to angle the force applied due to the curvature of the rocket motor domes. The original design for this rail cart can be seen in figure (9) below.

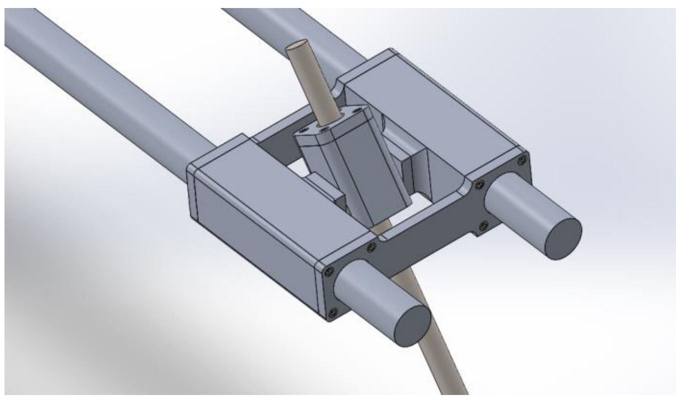


Figure 9: PDR Rail Cart Design

The disadvantages of this rail cart was primarily that the the design did not allow the operators to lock the angle of the lead screw that would be applying the axial forces on the rocket motor dome. Due to this and primarily the rail changes discussed in section 1.2.3, the rail cart displayed in figure (10) was designed.

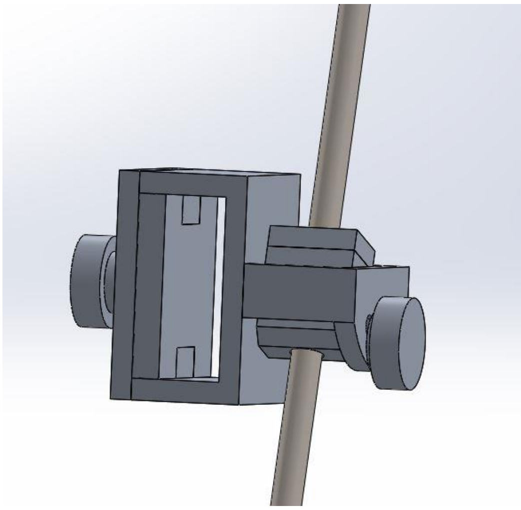


Figure 10: Current Rail System CAD

Since a singular rail is being used in the final design, the rail cart was changed accordingly. The rail cart changed to a rectangular shape to accommodate the 3"x"1 rails. The angling mechanism was moved to the side of the device to still allow operators to angle the lead screw as needed. Two primary changes were made to the rail cart system however that were not made due to the rails. Originally a set screw was going to be used to lock the rail cart so that it would not translate during operation. As seen on the left side of figure (10), a rectangular plate and screw were made to allow the device to press onto the entire side of the rail during operation. The operator would tighten the screw that would press the plate onto the rail which would allow the device to clamp onto the entire side of the rail instead of only tightening a specific spot as it did previously. The second change that was made was an angle locking mechanism seen in figure (11) below.

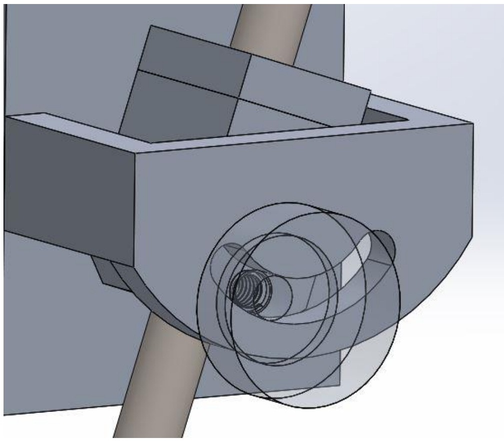


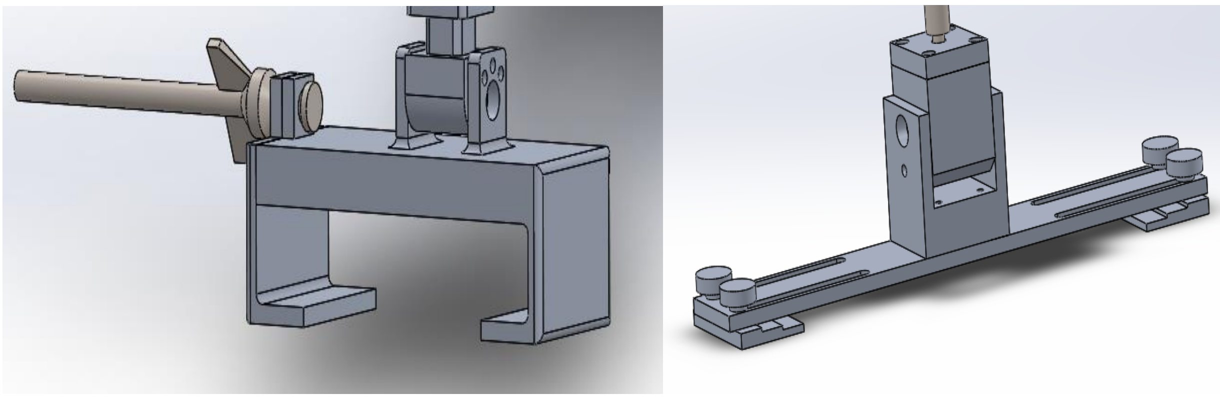
Figure 11: Angle Locking Mechanism

A capped screw was designed to allow the operators to lock the angle of the lead screw during operation. The piece that holds the lead screw housing had a slot milled by a CNC to allow for various locking locations. In the lead screw housing, a threaded hole was made where the screw would tighten into, which would lock the angle of the lead screw.

The only drawback of this design is the existence of an angle of twist that was not previously existent in the PDR design. This is due to the axial force being located to the side of the rail system instead of existing in the center of the design. By performing the calculation for angle of twist on an aluminum 6061 3"x1"⅛" hollow rectangular rail, an angle of twist of .04 degrees was expected. Due to this being approximately 0, the team does not expect this angle of twist to be a problem in the device. The calculation for angle of twist can be further seen in Appendix D.

1.1.5 1.2.5 Design Iteration 5: Change in template holder discussion

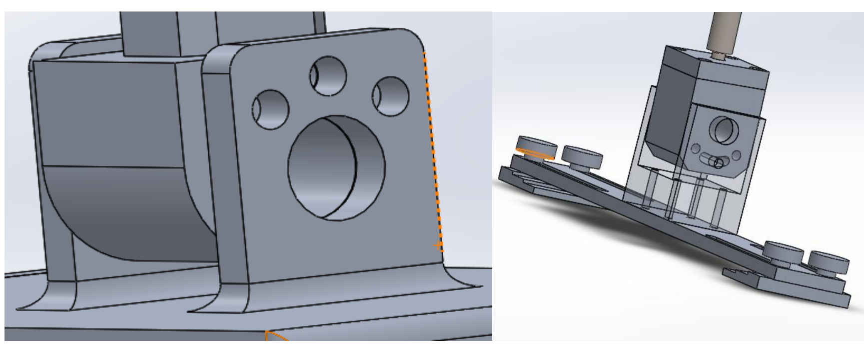
The original design for the template holder used a clamping screw to fix the template to the attachment. This design was inadequate as it did not account for the various sizes of templates that are used by Northrop Grumman. The new design takes into account the various template sizes by using steps integrated into the clamping jaws of the attachment. This will allow a tighter fit around the templates which reduce the chance of the template coming loose during operation. The new and old template holders can be referred to below in figure (12).



(a) **(b)**

Figure 12: Old Template Holder (a) vs New Template Holder (b)

A sub-assembly of the template holder is the positioning mechanism which allows the device to apply force normal to the motor dome. The old design for the positioning mechanism utilized pin holes at normal to and 45 degrees to the neutral axis. This design was changed as it did not allow enough flexibility within the design to achieve angles between those previously stated. The new design allows for an infinite angle between the bounds of 30 degrees on each side. With this added maneuverability of the design, the technicians will be able to find the angle they need and then lock the device in place using a threaded knob (not shown). The modified design of the positioner mechanism can be referred to below in figure (13).



(a) **(b)**

Figure 13: Old Positioning Mechanism (a) vs New Positioning Mechanism (b)

The changes made to the template holder will allow for easier operation of the device, and ensure that any chances for failure due to handler error are mitigated.

1.1.6 1.2.6 Design Iteration 6: Change in spring scale discussion

Northrop Grumman wishes to be able to perform a 20lb push test and 50lb pull test per standoff with the mounting device designed by the capstone team. In order to determine if the axial force performed by the device is accurate, a spring scale was made by the design team during the PDR. This design can be seen in figure (14) below.

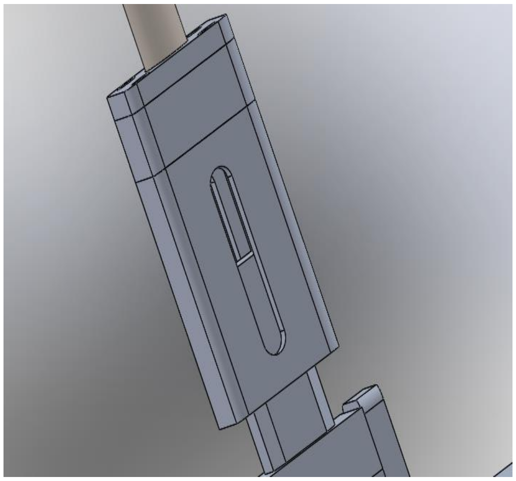


Figure 14: PDR Spring Housing

The spring housing was designed to make a spring display force values with ticks marked on the outside of the housing. This design had some clear existing problems. The spring housing was complicated to manufacture, implement into the design, and required a spring analysis to determine the correct spring to use in the design. In order to simplify the purpose of this housing, a design change was made to remove the housing in exchange for a torque wrench which can be seen in figure (15).

Figure 15: Torque Wrench

A torque wrench allows operators to apply a set force on the lead screw which can be displayed by the tool. This allows for a much similar solution while also reducing the amount of manufacturing for the device.

In order to apply the proper axial loads to the bracket templates it is important to determine the required torque. The torque that is required to raise and lower the loads associated with the power screw were calculated using the torque equations seen below in table (1). These values will become important when creating the handlers manual for our device and what the tolerances on the torque will be to ensure the correct forces are applied.

Table 1: Torque Equations

Parameter	Equation
Torque required to raise the load [T_R] [1,2,3]	$T_R = \frac{F \cdot d_m}{2}(\frac{l + \pi f d_m \sec\alpha}{\pi d_m - fl\sec\alpha}) + T_c$
Torque required to lower the load [T_L][1,2,3]	$T_R = \frac{F \cdot d_m}{2}(\frac{-l + \pi f d_m \sec\alpha}{\pi d_m + fl\sec\alpha}) + T_c$
Frictional torque of the thrust collar [T_c]	$T_c = \frac{F f_c d_c}{2}$
Efficiency during lifting the load [e] [3]	$e = \frac{Fl}{2\pi T_R}$

The equations that were used for the calculations of the lowering and raising torque are located above in table (1). The conditions that are expected during the standard operation of our device include a push and pull force of 20 lbf and 50 lbf respectively. The values that were determined for both of these conditions include a torque required to raise a load consisting of 0.313 lbf-ft and a lowering torque of 0.176 lbf-ft. By using these torque values as indicators on the torque wrench, it will be possible to apply forces accurately to the bracket templates.

2 Future Implementation

This section will discuss the remainder of the manufacturing and design of the system. This will include a schedule of deliverables for the upcoming months which will allow for organization and efficiency. The schedule will be a tool that will act as a reminder and to help delegation of the deliverables. Another aspect of the schedule that will be included within this section is the Gantt Chart which gives an intuitive look at the remainder of the semester and what project milestones are being worked towards. The next topic of this section to be discussed is the budget. The budget will encompass all purchase orders made thus far in the project and what costs are expected to be incurred towards the end of the semester. This will include providing a bill of materials of the upcoming purchases and a discussion on why the purchases are necessary for the completion of the project. Sub-assemblies that are yet to be designed or manufacturing processes that have yet to be completed will also be addressed.

2.1 2.1 Further Manufacturing and Design

Currently the capstone team has completed the manufacturing of the rail cart system discussed in section 1.2.4 excluding the lead screw housing. The rails will be purchased this week along with the other materials that will be required as discussed in section 2.3. Since the design must be completed by March 27th, the team is planning on using most of the time the next few weeks in the machine shop. The team will be working on the rocket motor clamp, angling mechanism, and bracket template holder planning on completing these parts before spring break (March 13th). This will allow the team to use the week after spring break to assemble, make minor changes that are required, and complete the final device. Some of the parts will require the use of the CNC, which will be done with a work order through the NAU

Machine Shop. The parts that are expected to require the use of a CNC can be seen in table (2) below.

Table 2: Expected CNC Parts

Part
Inside Clamp for the Castor 30 XL
Outside Clamp for the Castor 30 XL
Mach Ring Template for Device Demonstration (Holes drilled from 3D printed hole array template)

2.2 *2.2 Schedule Breakdown*

This section will discuss the team's schedule and how it is set up to accommodate the future deliverables of capstone. Appendix E provides a detailed outline of the future deliverables starting from March 1, 2020 to May 6, 2020. The schedule is designed to make the entire team accountable for each individual submission and team deliverable, with the exception of the website checks which make two members accountable for those submissions. Tasks are delegated by Tyler, the project lead, each week on Wednesdays during the team meetings to ensure an equal distribution of workload is spread throughout the team. Sage and Dakota take lead on the team's final design and design changes, inputting extra hours throughout the week to ensure the final design meets the customer and engineering requirements. Elaine and Brandon are in charge of the website, setting up team submittals, and preparation for test procedures. Everyone is responsible for manufacturing components of the team's final product.

2.3 2.3 Budget breakdown

This section will go into detail of the budget as it stands with what is purchased, what remains to be purchased, and what aspects of the system have not been designed or sourced yet. Table 3 provides in-detail of what has already been purchased and the remaining budget. Table 4 is an estimated list of the remaining materials that the team needs to purchase. An order form will be sent out on Monday to get the rest of the needed materials in the team's hands so that the next few weeks can be spent fully finishing the product.

Table 3:. Up-To-Date BOM

Material	Unit Cost	Quantity	Total Cost	Source
6061 Aluminum Block, 4"x4"x12"	100.25	2	248.84	McMaster-Carr
PLA 3D Printing Filament	12.99	1	14.18	Amazon.com
Linear Sleeve Bearing, for 1-1/2" Diameter	141.17	1	175.05	McMaster-Carr
6061 Polished Aluminum Tube, 1/4" wall thickness, 1-1/2" OD	28.95	1	35.90	McMaster-Carr
Acme Lead Screw, 1/2"x10, 2ft long	31.68	1	39.28	Roton.com
Acme Sleeve Nut, 1/2"x10, Bronze	19.09	1	23.67	Roton.com
6061 Aluminum Rect. Tube, 1-1/2"x3"x3/16" thickness, 6 ft long	93.16	2	231.04	McMaster-Carr
Aluminum Socket Head Screws, 8-32, 1/2" long, Blue-Anodize	11.88	5 Packs of 5	73.66	McMaster-Carr
Strain Gauges	52	1 Pack of 8	80.78	Omega.com
		Total Cost to Date	922.40	
		Remaining Budget	9077.60	

Table 4: Future Purchases

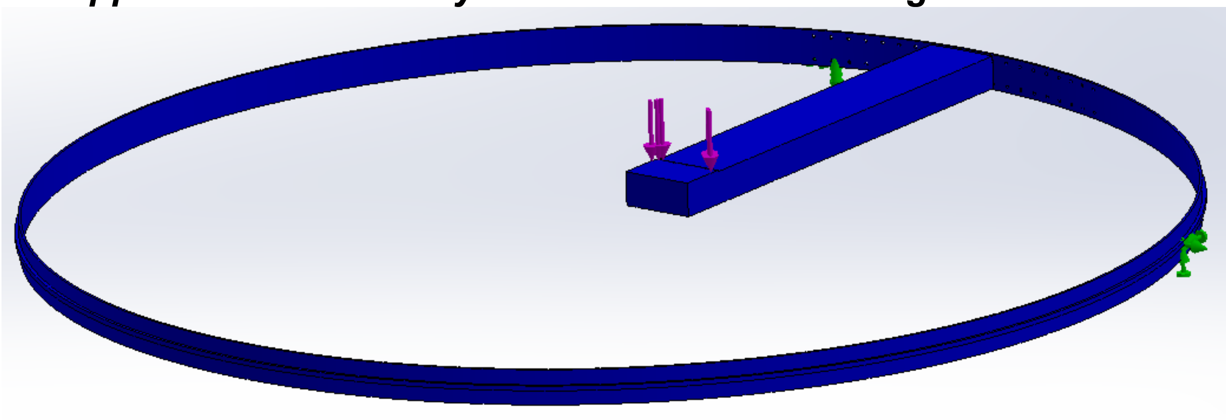
Material	Unit Cost	Quantity	Total Cost	Source
Aluminum 6061 3"x1"x1/8" rectangular tubing 6 ft long	76.68	1	76.68	McMaster-Carr
6061 Aluminum Rod, 2"x1ft	32.31	1	32.31	McMaster-Carr
Stainless Steel Self-Locking Pins, 1/4"x3"	50.83	3	152.49	McMaster-Carr
Stainless Steel Self-Locking Pins, 1"x3"	62.22	1	62.22	McMaster-Carr
Aluminum Plate, 3/8"x2inx1ft	23.53	1	23.53	McMaster-Carr
		Total Cost of Future Materials	347.23	
		Cost of Future Materials + Total Cost to Date	1269.63	
		Expected Remaining Budget	8730.37	

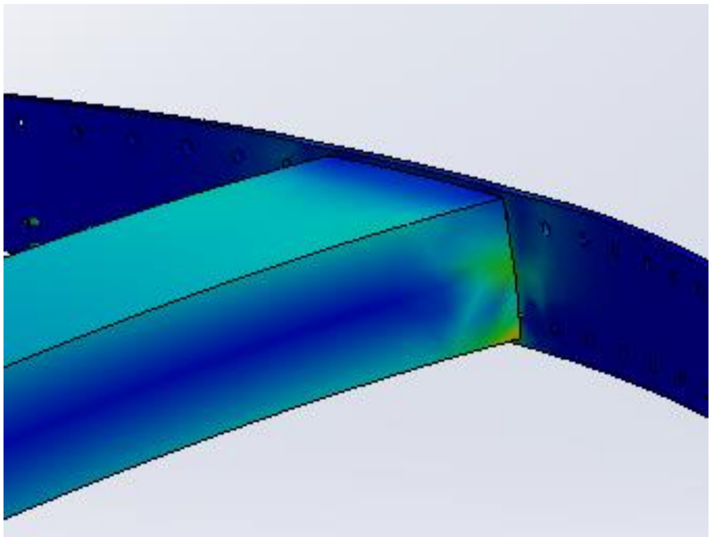
Currently the team still has a piece of raw material that will be machined into some of the smaller components of the design in the next week while the remaining parts are ordered. The entirety of the product is fully designed in Solidworks and no aspects are left undesigned or unsourced. The only concern in the upcoming weeks are the complex CNC parts that are to be work ordered. These work orders with accompanying Solidworks part files will be sent to the machine shop managers within weeks end.

2.4

2.6 3 Appendices

3.1 Appendix A: FEA Analysis for Rocket Motor Ring





3.2 Appendix B: Pin Shear Analysis

Pin Shear Analysis

- Single pin must resist moment from entire rail cart lever arm.
    - One long, single pin going through both sides subjected to double shear.
- Max Load 50 lbs, results in 360lb internal shear on pin.
- Required diameter for desired factor of safety in pins is 0.207 in.

$$\tau_{failure} = 32\ ksi$$
$$F.O.S. = 3$$
$$\tau_{allowable} = 10.67\ ksi$$

$$\tau_{Avg} = \frac{V_{internal}}{A_c}$$

$$D_{required} = \sqrt{\frac{4\ V_{internal}}{\Pi\ \tau_{allowable}}}$$

3.3 Appendix C: Rail Deflection Analysis

Rail System

- Hollow Cylindrical Tube:
  - $I_{xx} = .199 \text{ in}^4$
  - $A_c = .982 \text{ in}^2$
- Hollow Rectangular Tube:
  - $I_{xx} = .95 \text{ in}^4$
  - $A_c = .9375 \text{ in}^2$
- Deflection of Cantilever Beam:
  - $\delta c = .391$ in
  - $\delta r = .082$ in
    - F = 50 lb
    - E = 10000 ksi
    - L = 36 in
- Weight of Rail System:
  - Wc = 3.46 lb
  - Wr = 3.31 lb
    - $\rho = .098 \text{ lb/in}^3$

Hollow Cylindrical Tube:

$$Ixx = \frac{\Pi}{64}(D^4 - d^4)$$

$$A_c = \frac{\Pi}{4}(D^2 - d^2)$$

Hollow Rectangular Tube:

$$Ixx = \frac{1}{12}(BH^3 - bh^3)$$

$$A_c = BH - bh$$

Deflection of Cantilever Beam:

$$\delta = \frac{F L^3}{3 I E}$$

Weight of Rail System:

$$W = \rho A_c L$$

3.4 Appendix D: Angle of Twist Calculations

Angle of Twist

- Length = 36 in
- Torque = 81.625 in-lbs
  - 1.3625" * 50lbs
- Modulus of Rigidity = $3.8*10^6$ psi
- Polar Moment of Inertia = 1.104 in^4
  - Ix = .950 in^4
  - Iy = .153 in^4
- Angle of Twist = .04°

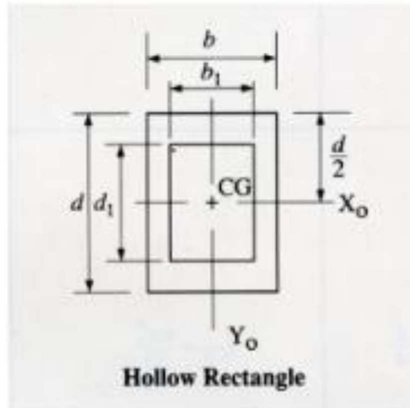


Figure 18. Angle of Twist Dimension Drawing

$$\theta = \frac{TL}{J_{CG}G}$$

$$I_{x_0} = \frac{bd^3 - b_1 d_1^{\,3}}{12}$$

$$I_{y_0} = \frac{db^3 - d_1 b_1}{12}$$

$$J_{CG} = I_{x_0} + I_{y_0}$$

3.4 Appendix E: Gantt Chart

TASK	ASSIGNED TO	PROGRESS	START	END
Midpoint Presentation and Hardware Review				
Midpoint Presentation	Everyone	0%	3/1/20	3/2/20
Hardware Review	Everyone	0%	3/1/20	3/4/20
Individual Analyses	Everyone	0%	3/1/20	3/13/20
Final Product Completion	Everyone	0%	3/1/20	3/27/20
Website Checks				
Website Check 2	Elaine, Brandon	0%	3/1/20	3/27/20
Website Check 3	Elaine, Brandon	0%	3/1/20	5/4/20
UGRADS Preparation				
Proof of UGRADs registration	Everyone	0%	3/1/20	3/7/20
Draft of Poster	Everyone	0%	3/1/20	3/13/20
Final Poster	Everyone	0%	3/1/20	4/3/20
UGRADS (Friday) Presentation	Everyone	0%	3/1/20	4/24/20
Peer Evaluations				
Peer Evaluation 2	Everyone	0%	3/1/20	3/8/20
Peer Evaluation 3	Everyone	0%	3/1/20	4/5/20
Final Peer Evaluation	Everyone	0%	3/1/20	5/5/20
Implementation Memo II	Everyone	0%	3/1/20	4/3/20
Testing Proof				
Testing Procedures	Everyone	0%	3/27/20	4/10/20
Testing Proof Report	Everyone	0%	3/27/20	4/10/20
Final Report	Everyone	0%	3/1/20	5/1/20
Operations/Assembly Manual	Everyone	0%	3/1/20	5/1/20
Final CAD package with BOM	Everyone	0%	3/1/20	5/3/20
Client Handoff	Everyone	0%	3/1/20	5/6/20
NG University Symposium Day	Everyone	0%	3/1/20	5/1/20