Project B8 - Universal Dome Standoff Bonding Tool

Final Proposal

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EXECUTIVE SUMMARY

The Northrop Grumman Standoff Project team was tasked with designing and manufacturing an articulating arm that will effectively hold standoff mounting templates in position for the duration of the adhesive cure process. The teams design looks to replace the current taping method used by Northrop Grumman technicians when mounting avionics electronics to the forward and aft domes of various rocket motors. To achieve a suitable final design to manufacture, the team followed Northern Arizona University Engineering Capstone processes to fully understand, analyze and evaluate all potential concepts. By creating Black Box and Functional Decomposition Models, the team determined the material, energy and signal flows throughout the potential designs to ensure that the final manufactured device met all customer needs and engineering requirements. The sub-functions derived from the two models were used to begin the concept generation and evaluation stage. Numerous potential sub-functions and full concepts were created and analyzed from Pugh Charts, Decision Matrices and general design evaluations conducted by the team. The top concept developed by the team was then created with CAD software to gain a complete visual understanding of the device as a whole and how each sub-function is supposed to work when fully manufactured. This CAD generated assembly is shown in Figure 1 below.

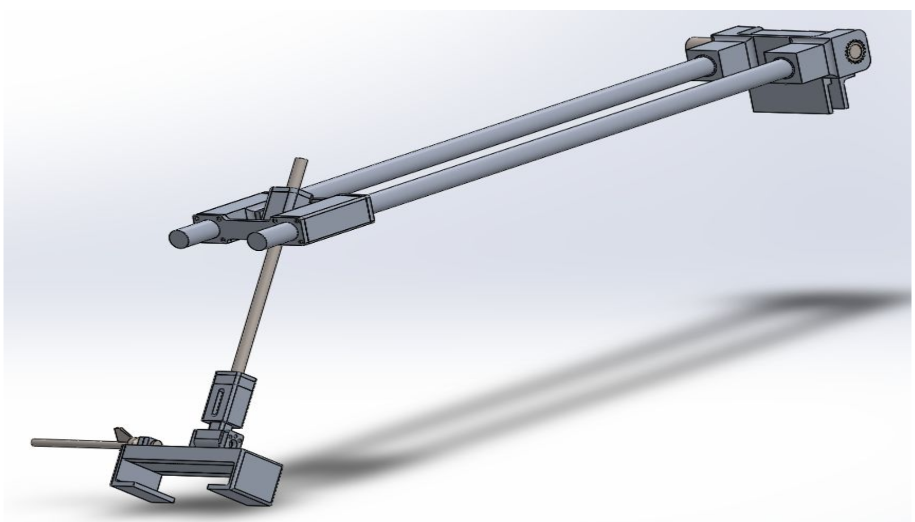

Figure 1: Full CAD Assembly

With a full CAD assembly created, the team created a low fidelity prototype and conducted a Failure Mode Analysis on the design. Moving forward, the team has a preliminary design presentation with critical members at Northrop Grumman in Chandler, AZ. Before beginning material/sub-function testing and manufacturing on the final design, the team must ensure that the current state of the design is satisfactory with the project client.

TABLE OF CONTENTS

1 BACKGROUND

1.1 Introduction

Rocket motor integration activities at Northrop Grumman field sites currently bond standoffs (threaded mounting devices that are used for avionic electrical components) to rocket motor domes using adhesives and tape. The standoffs are mounted to metal brackets, which are taped to the motor dome for between 24 to 72 hours in order for the adhesive to cure. This method is unreliable and fails roughly 5% of the time causing the brackets to either slip or fall off the motor domes. When the taping fails, an increase in man hours is required; this costs time and money when installing these standoffs. For this reason, Northrop Grumman's Flight Systems Group has requested for a team to design, analyze, and build a prototype universal dome standoff bonding tool that can be mounted to the attach rings of several variations of rocket motors (Castor 38, 30XL, and Orion 50XL) seen in Appendix A, that will hold standoff brackets in place while the adhesive cures.

1.2 Project Description

The following is the original project description provided by Northrop Grumman:

> During rocket motor integration activities at Northrop Grumman field sites, many standoffs (threaded mounting devices for avionics) are bonded to motor domes using adhesives. The current method of operations uses a bracket or template, to which the standoffs are mounted. The adhesive is applied, then the bracket is taped to the motor dome to hold the bracket in place for the 24 hour or longer cure period. The tape method is unreliable and occasionally allows the brackets to slip or fall off of the domes. A waste of time and labor hours are incurred when the taping method fails. NGC is requesting that NAU select one team to design, analyze and build a prototype articulating arm that can be mounted to the attach rings of several different motor types that will firmly hold the standoff template brackets in place during adhesive cure.
>
> **Requirements:**
>
> 1. The mounting arm shall be able to support brackets bonded at a range of four inches to 36 inches inboard from the motor circumferential ring.
> 2. The mounting arm, shall have six degrees of freedom to allow the standoff templates to be held in place at the proper location and angle on the motor domes.
> 3. The handling arm shall be mountable to the forward and aft attach rings of several rocket motors (details to be supplied by NGC).
> 4. The handling arm must be ESD (electro static discharge) compliant.
> 5. The handling arm shall be adaptable to several different mounting bracket templates via adapters or another method of re-configuration.
> 6. The handling arm shall be able to hold an adapter and standoffs of total mass up to 10 lbs.

7. The handling arm shall have the ability to be locked into place and apply a force of at least 20 lbs. on the adapter pushing it onto the motor dome.
8. Safety factors for all components must be 3.0 based upon the maximum expected loads. The arm is to be load tested to 125% to demonstrate structural integrity.
9. The handling arm must be easily manipulated by hand.

Additional Information:
For design purposes, the following assumptions may be made:
1. The standoff templates are flat aluminum plates of sizes 6.0” x 6.0” up to 10.0” x 16.0”
2. The arm will be attached to the standoff templates by clamping, not by bolting, bonding, or any other method.
3. The height of the standoffs (distance between motor dome and bottom of template) will be at least 0.5 inches.

Specific interface requirements will be provided upon selection. Other considerations students should take into account are: Life cycle evaluation for service life prediction, service and periodic maintenance, ease of handling and transportation.

Since the beginning of the project, changes to the project requirements have occurred. It is now expected that the mounting arm shall be able to perform a pull test of 50 lbs at 45 degrees from the centerline of the bracket. The pull test is required due to Northrop Grumman’s current process of applying a pull test with a fish scale after the adhesive cures to verify that the mounting bracket will not fall off during flight. The client also now expects to be able to use multiple mounting arms at a time. This is not a requirement for the project, however, this is something the client wants the team to consider for the design. Currently, the technicians secures multiple mounting brackets at a time with the taping method, so this expectation was added to the project description in order to match the efficiency of the current tapping method. The budget for this project is set at $10,000.

2 REQUIREMENTS

In order to fully understand the goal of the project, an in depth analysis of the requirements requested by the customer and requirements that must be met by the design team was performed. These were listed as customer requirements (CRs) and engineering requirements (ERs). After these requirements were listed, a quality function deployment (QFD), which can be seen in Appendix B, was created in order to compare the engineering requirements to the requests of the client and quantify the impact of each on the final design. An in depth description and analysis of these requirements can be read below.

2.1 Customer Requirements (CRs)

As discussed in the project description (section 1.2), the articulating arm must be able to meet the requirements listed by Northrop Grumman. The client requires that the final design be electrostatic discharge (ESD) compliant. If the design were not ESD compliant, the final design could transmit static electricity to the electrical components on the rocket motor dome, which could burn out the circuity. In order to prevent this from happening, ESD standards must be considered during the design process. Along with this material property, the articulating arm must be able to support brackets bonded at a range of four inches to 36 inches inboard from the motor circumferential ring. This will allow standoffs close and far from the rocket motor ring to be bonded to the motor dome. The design should have six degrees of freedom to allow the standoff templates to be held in place at the proper location and angle on the motor dome. This will allow the device to reach all directions to bond the standoffs. The arm should also be mountable to the forward and aft attach rings of several rocket motors. This will allow Northrop Grumman to use the articulating arm on multiple rocket motors instead of creating separate designs for each. The handling arm should be adaptable to several different mounting bracket templates. This is due to there being a large number of standoff templates that are used in these applications; so the design should be able to apply to flat aluminum plates of sizes 6.0” x 6.0” up to 10.0” x 16.0”. Because of these sizes, the weight of each standoff varies. However, the design should be expected to hold an adapter and standoff up to a total mass of 10lbs. To secure the standoffs in place on the rocket motor dome, the design should be able to lock in place and apply a push force of 20 lbs. on the adapter pushing it onto the motor dome. To test if the adhesive has cured, the articulating arm should be able to perform a 50 lb. pull force normal to the rocket motor dome surface, at 45 degrees from the centerline. These axial force tests can be combined into a singular customer requirement that meets both statements discussed by Northrop Grumman. The client also requires that safety factors for all components must be 3.0 based upon the maximum expected loads. The arm is to be load tested to 125% to demonstrate structural integrity. This will verify that the device will be both durable and robust for future use. Along with these requirements specified by the client, the design team is requiring that the device be within the $10,000 budget provided by Northrop Grumman. The handling arm shall also be a reliable design for operators and be safe to use. The client also wants the team to consider having a design that allows for the use of multiple mounting arms at a time. Since Northrop Grumman currently tapes multiple standoffs in place at a time, this was added in order to match the efficiency of the current tapping method. Since the plan is for only one operator to use this device, the final design should also be easy to use and transportable for the technicians. Table 1 below displays the current customer requirements and their weights.

Table 1: Weighted Customer Requirements

	Customer Requirements	Weight
1	ESD Compliance	0.09
2	Apply Axial Forces	0.09
3	Six Degrees of Freedom in Movement	0.09
4	Usable 4”-36” Inboard of Ring	0.09
5	Transportability	0.04
6	Ease of Operation	0.07
7	Durability	0.08
8	Reliability	0.08
9	Adjustable Interfaces	0.09
10	Support 10 lbs. in Locked Position	0.09
11	Minimum 3.0 Factor of Safety	0.06
12	Cost Within Budget	0.03
13	Use of Multiple Mounting Arms at a Time	0.05
14	Safe Operation	0.05

As shown in table 1 above, each customer requirement has a corresponding weight. The weights allow the team to show the significance of each customer requirement related to the project which includes ESD compliance, apply axial forces, six degrees of freedom, usable 4”-36” inboard of the ring, adjustable faces, and support 10 lb in a locked position are equally the highest rated customer requirements due to Northrop Grumman specifically asking these in the original project description. Furthermore, durability and reliability are the next highest customer requirements at a weight of 0.08. If the device is not designed to run effectively multiple times, then it will not meet the expectations of the client. While durability and reliability are important for the overall design, the other customer requirements listed by the client in the project description are ranked higher. If none of the 0.09 customer requirements are met, then the design is inadequate and will not be implemented into their applications. Ease of operation is ranked 0.07 because the client asked this to be considered in the design process. Although this is not a set requirement, it is still ranked highly since it was specifically asked for by the client. Despite being a customer requirement, the minimum factor of safety is ranked as 0.06. Usually, systems used in flights are set to a factor of safety of 1.5. This is because usually the higher the factor of safety, the more weight is added to the rocket. Since the articulating device will not be used in flight, there can be a higher factor of safety that is usually set to 3.0. For this reason, the factor of safety, while important, is ranked lower than the other customer requirements. The use of multiple mounting arms at a time and safe operations are ranked at 0.05. The multiple mounting arm requirements is a late consideration the customer added to the project. While this is a requirement that will be designed around by the team, the client has specified that this is a requirement that should not be a main priority. Safe operation is weighted less than the other requirements due to many of them being directly correlated to safe operation, such as ease of operation, reliability, and the functionality of the device. Since it is not expected to use the entirety of the $10,000 budget, the cost within budget is ranked the lowest at 0.03.

2.2 Engineering Requirements (ERs)

In accordance with section 2.1, verifiable engineering requirements were created to assign measurable parameters or conditions to each customer requirement. This allowed the project team to evaluate if the generated concepts would meet the client's expectations for the final design.

The device should be evaluated if it is electrically conductive (Y or N). This is an essential engineering requirement because the design needs to be ESD compliant to protect the circuitry mounted to the motor dome. For this reason, the material of the device will be evaluated to verify that it will not carry static electricity into any of the electrical components of the rocket motor. The mass (lbm) of the device is another value that will affect the transportability, durability, reliability, factor of safety, usability, and ease of operation. The articulating arm will need to have enough mass from the material thickness to work effectively and reliably, but also have a minimum amount of mass to make sure the device does not damage any of the existing equipment. The device must also be operable by one or two people. The mass will be affected by the principal dimensions (in.) of the device. This will alter the customer requirements associated with mass such as effect if the device is usable 4"-36" inboard of the ring, and determine if the device is usable for adjustable interfaces. These requirements will also be affected by the working length of the device (in.). The working length is one of the most important parts of the articulating arm, because if the device can not reach the standoff location, it is useless. In order to verify if the device can reach anywhere in the rocket motor dome, the working angle of the device (degrees) will be evaluated throughout the concept generation section. The modulus of elasticity is the final engineering requirement that will be directly related to the reliability and durability of the device to verify that it will not break. This will also correlate with the electrically conductive evaluation in order to make sure that the materials that work best for reliability and durability will be ESD compliant. Below is a table of each engineering requirement as well as a design-to value for each.

Table 2: Engineering Requirements with Design-to Values

	Engineering Requirements	Units	Design-to Values
1	Electrically Conductive	Y or N	Yes
2	Mass	lbm	25 ± 5
3	Principal Dimensions	in	8"W x 40"L x 6" H (±2")
4	Working Length	in	32"
5	Working Angle	Degrees	360^0
6	Modulus of Elasticity	lbf/in^2	$< 10.4 \times 10^6$ Psi [2]

As seen in table 2 above, each engineering requirement has a corresponding design-to value as determined by the design team. The device should be electrically conductive so that it can be grounded and carry less of a charge. The mass of the device should be no larger than what one person can carry and operate. For this reason, the design-to value is estimated to be 25 lbs with a ± 5 lb range. The device will clamp on the rocket motor ring with an estimated 8" width along with a 40" length reach out from the rocket ring and an estimated 6" height. These values are detailed further in section 5 of the report. The device is expected to reach 4-36" inboard from the motor circumferential ring, which makes the working length 32". The entirety of the rocket motor dome should be reached with the final device, which means the working angle needs to be 360^0 around the rocket motor dome. It can be assumed at this point in the project that the material used for the final device will be somewhat similar to the material the rocket motor dome ring which is constructed from 7075 Aluminum. 7075 Aluminum has a modulus of elasticity of 10.4×10^6 Psi. As the rocket motor ring cannot be damaged during installation, the team has specified the modulus of elasticity to be less than that of the ring material. This is to ensure that the articulating arm fails before any damage can be caused to the motor ring.

2.3 Functional Decomposition

The functional decomposition serves to provide a visual representation and understanding of the flows and sub-functions of our project. This process includes the functional model as well as the black box model. The black box model represents the expected energy, material, and signal flows into and out of the design as well as the overall function. The flows that are addressed in the black box model include the human hand and aluminum bracket which represent material flow, human power which is an energy flow, and device position which is a signal flow. The overall function of the design is to hold the bracket in place which considers the customer requirements of the 20lb push force and 50lb pull test. The flows represent a material, energy, or signal that is used by or that affects the product. The creation of the functional model followed a reverse engineering and redesign methodology that places an emphasis on what is being accomplished by the design rather than how. Using the ideas and information gathered during the creation of the black box model and evaluation of the customer needs the team could determine the sub-functions required of the design. The sub-functions identified during this process would allow the team to begin the concept generation stage with the creation of a morph matrix. The sub-functions that are identified within the functional model include import bracket, press bracket, transmit M.E., and position bracket which represent operations performed on a flow or multiple flows to transform it from its input to its output. The flows and sub-functions correspond to customer needs and ensures their presence within the model. The functional model and black box model were performed concurrently with the subsystem benchmark which explains the discussion of those topics in this section.

2.3.1 Black Box Model

This section outlines the team's process of creating and finalizing the black box model. The purpose of creating the Black Box Model is to understand the overall function of the product that will be designed and its appropriate inputs and outputs. There are three categories of inputs and outputs, also known as flows, which includes material(s), energy, and signals. These flows provides the team with information on what the product will use and what it will be affected by. The product's overall function was based on the project's requirements which was to "hold standoff in place". Materials input into the design include the human hand and bracket. A bracket will be mounted to/held in the device, utilizing human energy, and positioned in place to push onto the standoffs while the adhesive cures. Human energy is converted into mechanical energy through positioning the device. To know whether the product is pushing the standoffs in place, the product will signal through a click or snap noise. Figure 2, as shown below, shows the team's final Black Box Model.

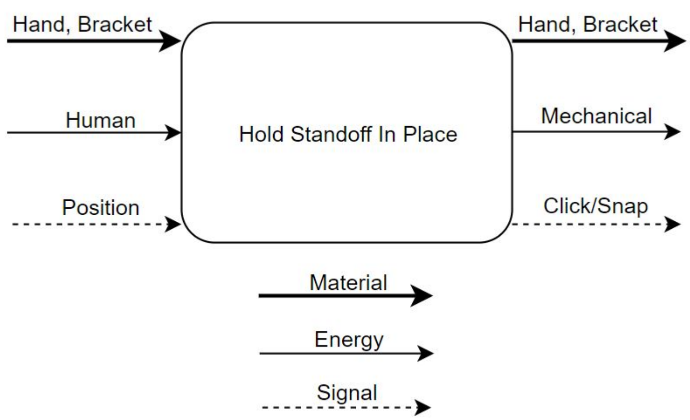

Figure 2: Black Box Model

2.3.2 Functional Model/Work-Process Diagram/Hierarchical Task Analysis

This section covers the functional decomposition model derived from the black box model in Figure 2. The black box model was used to understand the overall function of the proposed design and how it converts inputs to outputs. By taking the material, energy and signal flows that will ultimately be transmitted through the design, and understanding how they are manipulated and used will ensure a deep understanding of the overall functions and working of the design. The functional decomposition model presented in Figure 3 is an expanded view on the black box model above. This model follows each material, energy and signal flow within the design to observe what is happening to each flow throughout the design. The overall purpose of the project design is to orient and secure a bracket in place while the adhesive cures which requires a lot more flow manipulation than it seems.

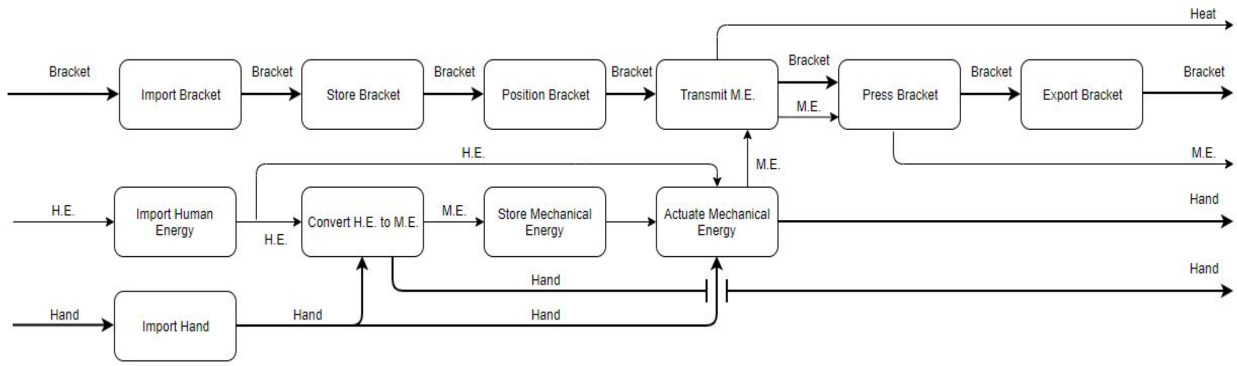

Figure 3: Functional Decomposition Model

In figure 2 above it can be observed where each material, energy and signal flow interacts with the sub-functions of the design. The bracket is imported into the machine, stored, positioned and then pressed into the dome for the curing process before being removed. The human energy, H.E. is imported into the system from the human hand which is then converted to mechanical energy to move the design and bracket into place. This mechanical energy is stored within the system and actuated to transmit the mechanical energy into the pressing of the bracket (i.e. 20lb push test, 50lb pull test). This functional model helped the team understand what each sub-function of the design was supposed to do to achieve the desired outcome. From this model, concept generation for each sub function and subsystem of the design could be proposed to begin creating various concepts that fit the function from the black box model and functional decomposition model. The design being created must import and store a bracket, position that bracket, and apply an axial force to the bracket during the curing process all by means of human power. These derived sub functions of the design are what will be used to ensure the customer needs gathered from Northrop Grumman are satisfied.

2.4 House of Quality (HoQ)

The quality function deployment (QFD) model used to evaluate the customer and engineering requirements for this project can be seen in Figure 4 at the end of this section.The purpose of this QFD was to relate the requirements given by the client to a set of engineering parameters derived by the team. Defined in detail above, the customer requirements outlined the need for a universally positionable handling arm that is capable of mounting to the outer ring of a rocket motor and apply axial forces to the standoff bracket. From these given needs, the team was able to generate a list of engineering requirements, which centered around the ability to service as much area as possible while maintaining ESD compliance and having minimal weight.

The development of the QFD for this design project gave a chance to compare the engineering requirements to the requests of the client and quantify the impact of each on the final design. From these calculations, the team was able to visualize the importance of different aspects of the design given the various effects on customer requirements. The modulus of elasticity of the material ranked the most important, as a failure of the device could damage the expensive components handled by the arm or the dome of the motor itself. At the other end of the spectrum, the strength of components in contact with the motor ring should not exceed that of the ring itself, as the ring should not be damaged in the event of a handling error. While geometry will also factor in to the strength of the part, this is only a starting point given that the final dimensions are currently unknown.

The mass of the ring also stood out to the team as an especially important engineering requirement, as a large mass would add to the stress applied to ring mount while also making the device more cumbersome to use. Given that the current method of standoff application, while prone to failure, requires little handling effort, additional setup time of the teams design should be minimized. The tolerance for the mass of the final design was set to encompass reasonable weights which may be supported by a single operator.

The next two highest weighted engineering requirements, working length and working angle, combine to describe the serviceable area on the rocket motor dome. These relate directly to the customer requirements, as the design must reach predefined inward distances around the entirety of the motor ring. If the final design does not meet these requirements, it will not be usable for the intended purpose.

While weighted as the least important requirements in the QFD, electrical conductivity and limited principle dimensions are still necessary to produce a device that is up to the standard the team would like to achieve. As grounding connections will be accessible when the device is used, and each component can be individually ground, it is not necessary for all parts to be conductive as a single unit. This will factor more into material selection than design choice, but is still an important consideration. Limiting the principle dimensions has a similar weight, as it is not necessary to perform the basic functions required. However, as this handling arm may be used by different operators at multiple facilities, a smaller total size would allow for easier relocation and general use.

Customer Need	Weight	Engineering Requirements	Electrically Conductive (Y or N)	Mass (slugs)	Principal Dimensions (in)	Working Length (in)	Working Angle (Degrees)	Modulus of Elasticity (lbf/in2)
1. ESD compliance	0.10		9	0	0	0	0	0
2. Apply axial forces	0.10		0	1	0	3	3	9
3. Six degrees of freedom in movement	0.10		0	0	0	9	9	0
4. Usable 4" - 36" inboard of ring	0.10		0	1	9	9	3	1
5. Transportability	0.07		0	9	9	3	3	0
6. Ease of operation	0.08		3	9	3	9	9	0
7. Durability	0.09		0	3	0	0	0	9
8. Reliability	0.09		0	3	0	0	0	9
9. Adjustable Interfaces	0.10		0	3	0	3	3	0
10. Support 10lbs in locked position	0.10		0	3	0	3	3	9
11. Minimum 3.0 Factor of Safety	0.07		0	3	0	0	0	9
Absolute Technical Importance (ATI)			1.14	2.9	1.77	3.63	3.03	4.15
Relative Technical Importance (RTI)			0.2746	0.698	0.4265060	0.874	0.730	1
Target ER values			(Reference Table 2)					
Tolerances of ERs								
Testing Procedure (TP#)			3.1	3.2, 3.3, 3.4	3.3, 3.4	3.3, 3.4	3.4	3.2, 3.3, 3.4

Figure 4: House of Quality

2.5 Standards, Codes, and Regulations

The purpose of this section is to discuss the standards and regulations that might be relevant to the project and how they might be applied in industry. The function of standards and regulations within manufacturing and design processes is to ensure safety, reliability, and efficiency. Most standards are promoted and maintained by engineering societies and regulatory agencies such as the Institute of Electrical and Electronics Engineers (IEEE) or the American Society of Mechanical Engineers (ASME) [3]. The codes and standards included within this section were procured from the ASME standards catalog. The standards that were chosen for this section include ASME Y14.5-2009 Dimensioning and Tolerancing, B30.2-2009 Overhead and Gantry, B18.29.1-2010 Helical Coil Thread Inserts, B5.48-1977 Ball Screws, and B1.5-1997 Acme Screw Threads. The codes and standards that were considered for this project can be referred to below in table 3.

Table 3: Standards of Practice as Applied to this Project

Standard Number or Code	**Title of Standard**	**How it applies to Project**
ASME Y14.5-2009	Dimensioning and Tolerancing	This standard could help facilitate the creation of engineering drawings that effectively communicate design intent.
ASME B30.2-2009	Overhead and Gantry	Applies the operation and safety procedures associated with cantilevered gantry cranes which can be related to our product.
ASME B18.29.1 - 2010	Helical Coil Screw Thread Inserts--Free Running and Screw Locking	This standard relates to our project by providing information on threaded hole helical screw inserts which are included within the power screw assembly.
ASME B5.48 - 1977	Ball Screws with Errata	A ball screw will likely be utilized for the power screw assembly. This resource will give information on the definition and classes of ball screws.
ASME B1.5 - 1997	Acme Screw Threads	This resource gives information related to acme screw threads which is the thread form which will be used for the power screw.

As seen in table 3 above, the first standard to be considered for this project was related to the aspect of dimensioning and tolerancing with respect to engineering drawings. The resource from which this standard originated from the AMSE and is denoted as Y.14.5-2009. This standard will help facilitate the creation and review of the engineering drawings in order to ensure their readability and design intent. The project includes the creation of an in depth CAD package which in order to be effectively communicated to the client must be transferred into drawing format. This standard will affect the project by influencing this process and changing the methods of dimensioning and tolerancing of these engineering drawings.

The next standard to be considered will be the ASME B30.2-2009 Overhead and Gantry standard which relates to the cantilevered gantry design of the device. This standard applies to the installation, operation, and maintenance of hand-operated gantry cranes. While the device is not a gantry crane it does share similarities, which include the bracket delivery system and the overhead rail design. This standard also applies to special circumstances such as non-vertical force delivery and guided loads which are consistent with the current design. This standard will help the project by allowing the team to comply with regulations of the industry while maintaining operational efficiency and safety.

The current design utilizes a power screw system to deliver pull and push forces at the appropriate angles to the brackets. This standard is denoted as ASME B18.29.1-2010 within the table above. This assembly will include components such as helical coil screws (lead screws) and screw locking mechanisms. This standard will help with the facilitation of the design of this assembly by providing information on the proper selection of Screw thread insert (STI) taps, installation, and dimensional data. This standard could influence the team’s decision on the power screw that is chosen for the device as it would adhere to standards used in industry.

The integration of the power screw into the device will likely require the implementation of a ball screw which will be mounted to the rail cart system. This ball screw will allow the free rotation of the power screw through it and provide an anchor point to the device. This anchor location will be the point of which the principal forces of the force application of the power screw will originate. This standard will help the team by providing information relevant to the identification of the optimal ball screw for the device. This standard will influence the project due to its information on the standard definitions, classes, and recommended combinations of ball screws. This standard is denoted as ASME B5.48-1977.

The thread form of the power screw is an important consideration to be had for the design of the rail cart assembly. This standard is denoted as ASME B1.5-1997. This standard will give information on the applications and general limits and tolerances of acme threaded screws. Due to the prolific occurrence of acme threaded power screws within force delivery systems, the team will heavily consider its use within the device. The resource will provide information on the three classes of general purpose acme threads as well as their alignment, clearances, major and minor diameters, and appropriate alignment. This will influence the team’s discussion of the correct power screw to be utilized within the design due to the information provided within the standard.

3 Testing Procedures (TPs)

This section outlines the testing procedures for the team's device. Each procedure will outline the overall description of the test and what engineering requirements and customer needs it will satisfy. The objective, resources required, and schedule for each test will also be detailed. Upon conducting these test procedures, the team will prove that the final product meets all engineering requirements as shown in Figure 3 in section 2.4.

3.1 Testing Procedure 1: ESD Compliance

The purpose of conducting an ESD Compliance test of the design is to ensure that the device won't damage any electronic components when mounted to the ring. This test will check that the device is electrically conductive and confirms that it meets the ESD compliance and safe operation customer needs. The procedure includes the use of a multimeter, ESD mat, table, and the design prototype. This test will likely run in the ninth week of the spring semester, when the final product is completed, and is estimated to take 15 minutes.

3.1.1 Testing Procedure 1: Objective

The objective of this test is to confirm that the final prototype is ESD compliant through the utilization of an ESD mat, multimeter, and the device prototype. The experimental set-up includes laying the ESD mat on a sturdy table, clamping the prototype to the edge of the table, ensuring that the prototype is completely on the mat, and then using a multimeter between the prototype and a team member. From this test, it is expected that there will be a reading of 0 Volts from the multimeter when connected to the prototype and user which will confirm ESD compliance. The procedure tests this particular aspect of the project because ESD compliance is such a heavily weighted customer requirement.

3.1.2 Testing Procedure 1: Resources Required

Table 4 below, provides the materials required for this testing procedure. However, additional resources includes the entire NG Team and the 98C classroom. The multimeter will be provided by one of the team members and is necessary to read the voltage between the user and prototype during the testing procedure. Additionally, an ESD Mat will be purchased from Amazon and is needed to ground the prototype. The team will utilize the 98C classroom's table to mount the prototype throughout the duration of the test. The total cost of this testing procedure is expected to be $1,434.55 which is largely due to the cost of the prototype.

Table 4: ESD Compliance BOM

Material	Quantity	Cost	Source
Final Prototype	1	$1391.83	NG Team
Multimeter	1	$0	NG Team
ESD Mat	1	$42.72	https://www.amazon.com/Bertech-Temperature-Rubber-Wrist-Grounding/dp/B01MDO2BGP?ref_=fsclp_pl_dp_4
Table	1	$0	98C

3.1.3 Testing Procedure 1: Schedule

This test procedure is expected to take approximately 15 minutes from the experimental set-up to experimental completion time. It will likely run during the ninth week of school when the final prototype is expected to be completed. Before this test can be run, the team must complete the entire prototype to be able to confirm that all components meet the ESD compliance requirement. Additionally, the team will need to purchase the ESD Mat and schedule a team meeting in advance before this testing procedure.

3.2 Testing Procedure 2: Clamping Force (Ring)

Mounting the device is a critical component of the team's design. The clamp is needed to support the entire device's mass while it functions. The purpose of this clamping force test is to come to a design conclusion of the best suited clamping mechanism to support the device without causing deformation to the motor dome ring. This test is necessary to prove that the final design will meet the following customer needs: Usable 4"-36" inboard of ring, transportability, durability, reliability, minimum 3.0 factor of safety, use of multiple mounting arms at a time, safe operation. The team will utilize pressure sensors, multimeter, Arduino board, wires, soldering kit, aluminum strip, rubber, torque lock vise grips, resistors, laptop, Arduino IDE, and a table. These materials will be used to record the clamping force applied to the aluminum material until deflection. This data will be compared to analytical results and will aid in the final clamping mechanism design.

3.2.1 Testing Procedure 2: Objective

Currently, the team hasn't come to a conclusion for the clamping mechanism that will mount the device to the motor dome ring. Therefore, the objective of this clamping force test is to determine the optimal dimensions of the customized clamp that will support the entire device without causing deformation to the motor dome ring. This objective will be achieved through an analytical analysis and physical experiment results. The analytical analysis will focus on determining the clamp force necessary to support the device, the deflection of the ring, the clamp's materials, and area of applied force that will meet both objectives. The physical experiment is included to determine the functionality of the device's potential clamping mechanism while also collecting data of the applied force. The purpose of this part of the test is to collect data points of the stress applied on the aluminum and the imposed strain.

3.2.2 Testing Procedure 2: Resources Required

Table 5 below shows the required resources for this procedure. The pressure sensors, aluminum strip, rubber pieces, and torque lock vise grips will need to be purchased in advance. The multimeter, Arduino board, wires, soldering kit, resistors, and a laptop with the Arduino IDE will be provided by a team member. Additionally, the team will test the experimental set-up in the ME495L room. The total cost of this test is $70.97.

Table 5: Clamping Force BOM

Material	Quantity	Cost	Source
Pressure Sensor	2	$24.00	https://www.amazon.com/Bolsen-Tech-Pressure-Resistance-Resistor/dp/B07L6LVR7G/ref=sr_1_18?keywords=pressure+sensor&qid=1572637272&sr=8-18
Multimeter	1	$0	NG Team
Arduino Board	1	$0	NG Team
Wires	6	$0	NG Team
Soldering Kit	1	$0	NG Team
Aluminum Strip	1	$29.00	https://www.homedepot.com/p/Everbilt-2-in-x-96-in-Aluminum-Flat-Bar-with-1-8-in-Thickness-802577/204273946
Rubber	2	$5.00	https://www.homedepot.com/p/American-Standard-Rubber-Pad-047161-0070A/205495047
Torque Lock Vise Grips	1	$12.97	https://www.homedepot.com/p/Milwaukee-10-in-Torque-Lock-Straight-Jaw-Locking-Pliers-48-22-3510/205017482
Resistors	3	$0	NG Team
Laptop	1	$0	NG Team
Arduino IDE	1	$0	Arduino Website
Table	1	$0	ME495L Room

3.2.3 Testing Procedure 2: Schedule

Prior to testing, the team will need to program an arduino to accurately read the pressure sensor and purchase the items outlined in table 5. In preparation for testing, length will be added to the pressure sensors by soldering wires to the sensors' wired ends and retested with the arduino to confirm accuracy. The resistors, pressure sensor, and wires will be properly placed on the breadboard and connected to the arduino. The pressure sensor is going to be attached to the aluminum using an adhesive. The rubber pieces will be attached to the torque lock vise grips.

The experimental procedure is expected to take 2 hours and the preparation will take 24 hours. The team is given more preparation time because of the need to program the arduino and ensuring that the set-up is wired correctly. It is expected that the analytical analyses will be done prior to conducting the experiment. This experiment will be conducted on November 15 during the Thermal Sciences Lab, ME495L . During this time, the team will verify that the program is running correctly by applying known masses onto the pressure sensor. Upon verification, the team will proceed to measure and collect the varying clamping forces until the material deforms. This data will be compared with the analytical analyses to confirm the expected behavior. Based on the data comparisons, the team will be able to accurately design the customized clamp that will support the device and not deform the ring.

3.3 Testing Procedure 3: Rail Deformation

The rails mentioned are used to support the cart which fulfills the function of translating the brackets. This component is responsible for ensuring that the cart is accurately placed, therefore making it a critical aspect of the team's design. The purpose of this rail deformation test is to determine the optimal material for the rails and to foresee the functionality of the rails when the product is in use. This test covers engineering requirements such as mass, principal dimensions, working length, and modulus of elasticity. Additionally, it also works to prove that the device meets the following customer needs: apply axial forces, durability, reliability, minimum 3.0 factor of safety, usable 4”-36” inboard of ring, and safe operation. The test will require strain gages, stainless steel rod, aluminum rod, tape measure, 50lbs of weight, c-clamp, and the labview DAQ. This procedure is scheduled to take place on November 21st and the 28th during the ME495L designated time periods in which the metal rods will be tested for deflection and strain. This physical data will be compared to analytical data which will provide the team with information on which material would be best suited for the team's design.

3.3.1 Testing Procedure 3: Objective

The objective of the rail deformation testing is to determine an acceptable material to construct the rails out of and predict the functionality of the rails during implementation of the device. This testing procedure will focus on the translation of the bracket while accounting for the engineering requirements and customer needs listed in section 3.3 above. From this testing different material rails will be subjected to 50 lbs of force at 36” from the mounting point to account for the worst possible situation the device will experience. Physically measuring the deformation at the end of the rail, as well as recording the measured strain from attached strain gages along the top and bottom of the rails, the team will have a strong foundation on determining what material to ultimately build the final design from.

3.3.2 Testing Procedure 3: Resources Required

Table 6 below shows the required materials needed by the team to conduct the rail deformation testing. This includes the Omega strain gages that will go along the top and bottom of the rail being tested. Two materials, Aluminum and Stainless Steel will be utilized for the rails themselves. The required testing resources, the tape measure, 50lbs of weight, c-clamp, table and LabView system are all either personally owned or available on campus. The total expected cost of the testing is approximately $155 for all required materials.

Table 6: Rail Deformation BOM

Material	Quantity	Cost	Source
Omega, 1-axis general strain gage	1 package (10 count)	$52	www.omega.com
Stainless Steel Rod(1”D, 3ft L)	1	$51.75	www.Grainger.com (unless a similar piece can be salvaged)
Aluminum Rod (1”D, 3ft L)	1	$50.50	www.Grainger.com (unless a similar piece can be salvaged)
Tape Measure	1	$0	NG Team
50lbs of weight	1	$0	NG Team
C-Clamp	1	$0	98C
LabView DAQ	1	$0	Room 111

3.3.3 Testing Procedure 3: Schedule

To conduct the experiment outlined above, the team needs a solid fixed table to mount the different rails to. A c-clamp with an adjustable interface will be needed to attach the cylindrical metal rod to the flat table and ensure no movement during testing. With the maximum 50 lbs attached to the end of the rod, at ~36” to account for the worst situation for the device to operate under, the deflection of the rail will be physically measured from the original distance to the floor before the weights were added, and calculating the difference from the measured distance after adding the weight. For both metals, the Omega strain gages will line the top and bottom at equally spaced points to accurately record the strain experienced by the applied weight.

Analyzing the physically measured deflection and the experienced strain in the rods, the expected performance of the rails can be determined to help select the correct material for the rails.

This experiment should take approximately two hours to conduct, including experiment setup and data acquisition. The team expects to conduct this experiment during the time for ME 495 Thermal Sciences Labs on Nov. 21st and Nov. 28th assuming funding will be available in time and the parts can be promptly ordered. Before the lab can be completed, the strain gages and two metal rails must be purchased and delivered whereas all other materials are readily available. With a successful test, the team expects to determine what material to use for the final design rails.

3.4 Testing Procedure 4: Power Screw Effectiveness

The purpose of testing the power screw is to verify the reliability, durability, and safety of the bracket holder, bracket holding component, splined shaft, and the locking mechanisms when the device is functioning. This test is necessary because it allows the team to check that the product passes the pull test while meeting other engineering and customer requirements. The customer requirements that this test will cover are apply axial forces, six degrees of freedom in movement, usable 4”-36” inboard of ring, ease of operation, durability, reliability, adjustable interfaces, support 10lbs in locked position, minimum 3.0 factor of safety, and safe operation. Additionally, the engineering requirements that this test covers are mass, principal dimensions, working length, working angle, and modulus of elasticity. The reason that this test works to verify that the product meets the majority of the customer needs and engineering requirements is because its made to test all device components while applying the axial force. This procedure will require the final prototype, table, bracket, and all team members. Ideally, this test will occur during the fifth or sixth week of the spring semester after the first fully functioning prototype is completed. By doing this, the team will be given enough time to modify the design if necessary.

3.4.1 Testing Procedure 4: Objective

The objective of this procedure is to test the functionalities of the bracket holder, bracket holding component, splined shaft, and the power screw effectiveness while the device applies an axial force. This test focuses on the entire functionality of the device which is why it tests multiple engineering and customer requirements. The purpose of this test is to give the team physical proof of the overall functionality of the design and allow room for further improvements before the final product is due. When the device applies a force, the bracket holding component, cart locking component, and splined shaft will be tested for locking reliability which is a critical aspect to the safety of the team’s design. Additionally, the functionality of the bracket holder will be tested to ensure that its able to reliably hold the bracket in place while the power screw works to impose the axial force.

3.4.2 Testing Procedure 4: Resources Required

Table 7 below shows the materials required. The final prototype, bracket, a sturdy table, and the team are required to successfully conduct the procedure. With that said, the total cost of this procedure is $1391.83. Additionally, the team will utilize the 98C classroom to test the final prototype.

Table 7:. Power Screw Effectiveness BOM

Material	Quantity	Cost	Source
Final Prototype	1	$1391.83	NG Team
Bracket	1	TBD	NG Team
Table	1	$0	98C

3.4.3 Testing Procedure 4: Schedule

Prior to testing, the team will need to build a fully functioning prototype and reserve a space to test. Ideally, the 98C classroom will be used since its expected that there will be enough space, however, the team may reserve another area in advance. This test is planned to take place during either the fifth or sixth week of the spring semester when the first fully functioning prototype is completed.

This test is expected to take approximately 1 hour from set-up to the experimental execution. A team member will secure the bracket to the bracket holder. The device will then be mounted on the edge of the desk and angled 45 degrees normal to the surface of the table. Afterwards, a team member will adjust the bracket component to the angle and apply force using the power screw. The force scale will be read to verify that the device meets the pull test. If the product doesn't meet the test, the team will take the data and make modifications to the design based on the analytical analyses, physical results, and Solidworks FEA.

4 RISK ANALYSIS AND MITIGATION

After determining the appropriate testing required for this project, the design team moved to analyzing potential risks and failures that the final design could experience. This was done to develop a proposed solution to Northrop Grumman based on potential risks for the project. By performing this risk analysis, a plan will be created to mitigate failures within the project. In order to do this, a complete Failure Mode and Effects Analysis (FMEA, as seen in Appendix C) was created to determine critical failures that could be experienced by the final device. A risk and trade-offs analysis was done to compare and contrast the potential critical failures. This allowed the team to determine if mitigating one critical failure would make it harder to mitigate another. After performing these analyses, the final design selected for this semester was made.

An FMEA is designed to determine and quantify top potential failures in a design. This allows for the project team to understand what risks could be present in the design and what recommended actions should be taken to mitigate these risks. In Appendix C, the full FMEA created for the Standoff Project can be seen. In this FMEA, the following seven sub functions were used as the possible function failures for the project: mount to ring, translate the brackets, hold bracket, apply axial force, locking, angle bracket, and ESD compliance. By evaluating the severity of the potential effects for failure, and the occurrence and detection of the potential causes for failure, Table 8 below was made to display the top ten potential failures based on the calculated risk priority number (RPN).

Table 8: Top Ten Potential Failures FMEA

Product Name	Standoff Bonding Tool	Development Team	18F19				FMEA Number			3
							Date			11/14/19
Part #	**Function**	**Potential Failure Mode**	**Potential Effects for Failure**	**Severity** (1-10)	**Potential Cause(s) for Failure**	**Occurance** (1-10)	**Current Design Controls Test**	**Detection** (1-10)	**RPN** (SxOxD)	**Reccomended Action(s)**
2	Mount to ring	Bending of the ring	Rocket Motor Ring will break	10	Overstressing the ring	5	Visual Indicator	8	400	Moment Ring Analytical Analysis
11	Angle Bracket	Bracket Joint Pin Shear Failure	Inability to carry out the pull test	7	Applied axial force	6	Visual Indicator	8	336	Material Selection Analysis
5, 6	Angle Bracket	Spline Mounting Screw Shears	Device will be unsupported	8	Applied axial force	6	Audibe and Visual Indicators	5	240	Spline Shaft Gear Analysis
2	Mount to ring	Clamp slips off	The device will be unsupported	8	Inefficient clamping force	3	Visual Indicator	8	192	Clamp force analysis
4, 13	Locking	Force block slides during axial force tests	Device cannot be applied to a specific location	7	Applied force exceeds lock capacity	5	Audibe and Visual Indicators	5	175	Material Selection Analysis
3	Bracket slips off	Unable to hold standoff bracket	Unable to hold standoff bracket	7	Incefficent clamping force	3	Visual Indicator	8	168	Clamp force analysis
22	Apply Axial Force	Lead Screw Breaks	Unable to apply axial force	8	Force induced deformation of screw	3	Audibe and Visual Indicators	5	120	Lead Screw Stress analysis
1	Translate the brackets	Bending of the rails	Device is not able to translate toward the standoff location	7	Overstressing the pivots	2	Visual Indicator	8	112	Deformation Test
4	Translate the brackets	Force Block does not slide	Device is not able to translate toward the standoff location	7	Ball Bearing Breaks	2	Visual Indicator	8	112	Bearing Analytical Analysis
15	Apply Axial Force	Fish Scale does not read correctly	Force will not display to operator	5	Spring does not function with the device	2	Visual Indicator	8	80	Spring Analysis

The RPN values from each potential failure seen in Appendix C are no larger than 100 other than the RPN values displayed in table 8. The potential failures above allowed the design team to identify the top 10 potential critical failures of the design. These are discussed in further detail below.

4.1 Critical Failures

As shown in table 8, ten critical failures were found based on their RPN values. The furthest right column, "Recommended Action(s)", shows the actions that the team should take to reduce the likelihood of these failures. The following section will describe each critical failure shown in Table 8 and how the failure can be caused, the effect of this failure, and how this failure can be mitigated.

4.1.1 Potential Critical Failure 1: Bending of the Ring

As described in the project description, the standoff mounting arm will be attached to the rocket motor dome in order to hold the standoff brackets in place while the adhesive cures. However as seen in figures 5 and 6, the standoff device will only have .2" x 1.25" to attach to the Castor 30 XL and .205" x 1.375" to clamp onto the Orion 50 rocket motor rings.

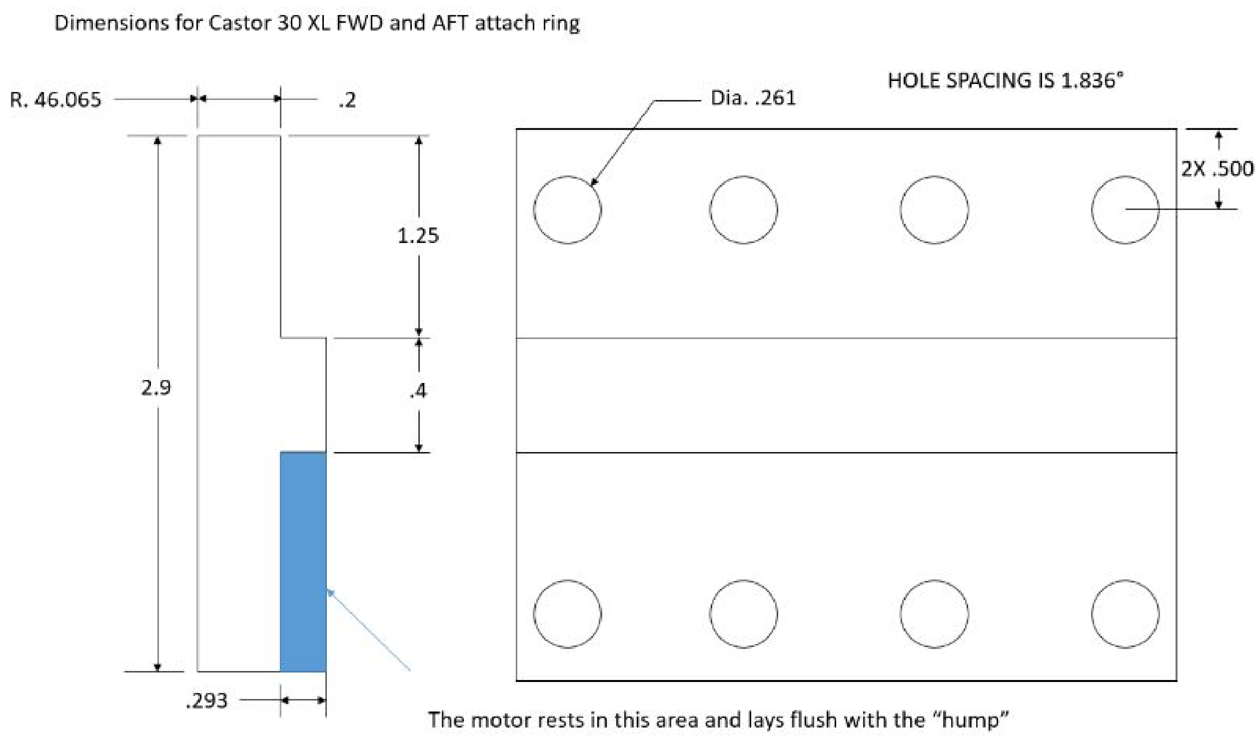

Figure 5: Castor 30XL FWD and AFT attach ring Dimensions

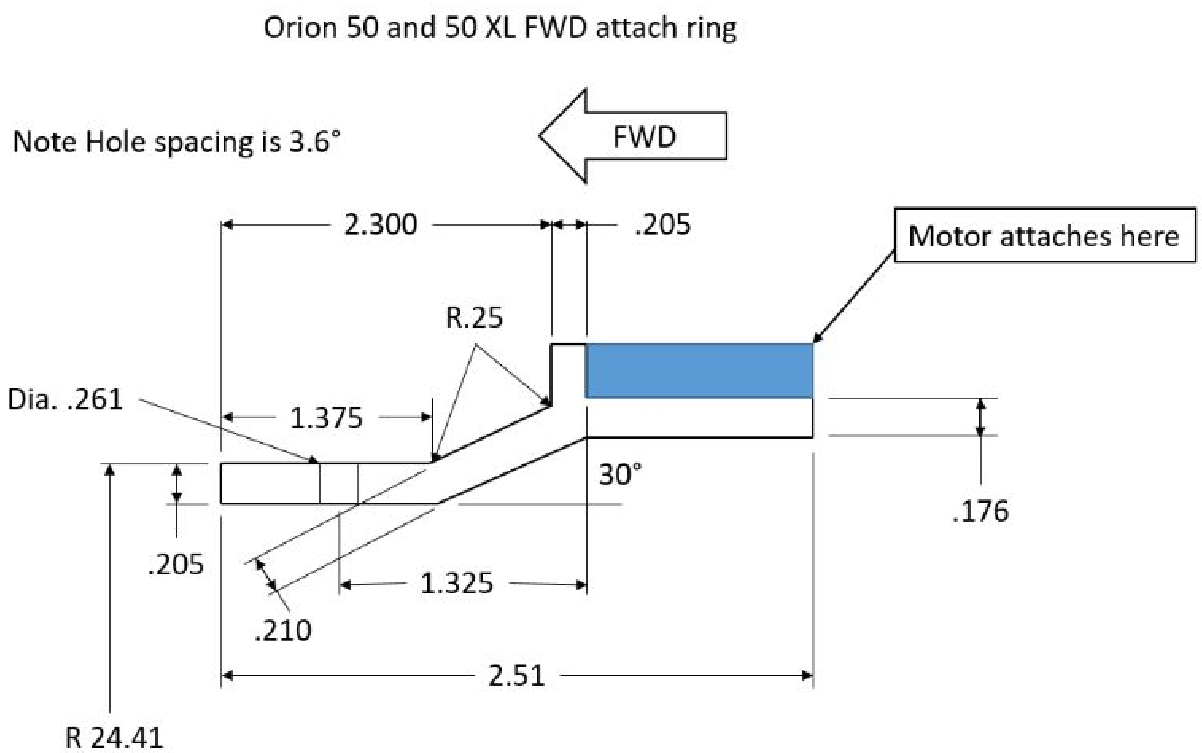

Figure 6: Orion 50 and 50XL FWD attach ring Dimensions

Due to the distance the axial force will be applied from the rocket motor ring (4-36" inboard), a large moment will be applied to the thin rocket motor ring. Despite being comprised of Aluminum 7075, the moment could cause bending of the rocket motor ring. If this were to occur, the rocket designed by Northrop Grumman would be ruined. This is the absolute worst case scenario for the design team and is deemed the most severe of potential failures of the device. That is why this potential failure has the highest RPN values shown in Table 8. To mitigate this failure from occurring, an analysis will be performed which considers the longest moment arm and maximum force to examine the worst case scenario for each ring geometry. The calculations will be performed with Solidworks FEA to visualize the stress concentrations and will be backed with hand calculations. These hand calculations will be used to ensure that the software is providing a reasonable output. The result of this analysis will allow the team to determine the necessary clamp width for load distribution while also ensuring that the maximum moment can be tolerated in all cases.

4.1.2 Potential Critical Failure 2: Bracket Joint Pin Shear Failure

In order to meet the 45 degree pull test requirement specified by Northrop Grumman, the bracket holding component that will mount to the bottom of the force gage will lock in two positions (90° and 45°). In order to lock the bracket in place, a pin will be used to lock between the two different positions. Due to the axial forces that will be performed on the device, a large amount of stress will be applied to the locking pin which could result in shear failure. To mitigate this from occurring, a material selection analysis will be performed to determine the best material and geometry for the pin. In the future, a further analytical analysis may be performed in order to verify that the pin will not suffer a catastrophic failure.

4.1.3 Potential Critical Failure 3: Spline Mounting Screw Shears

As described in the previous preliminary report and section 5 of this report, one of the primary designs that has been considered by the design team is a sliding rail system. The rail system will have two rails that will allow a force block to slide into position to apply the axial forces needed. Originally, the rail system was to move strictly horizontal across the rocket motor dome. However, the Castor 30 series' dome protrudes over the rocket motor ring plane shown in Figure 7 below.

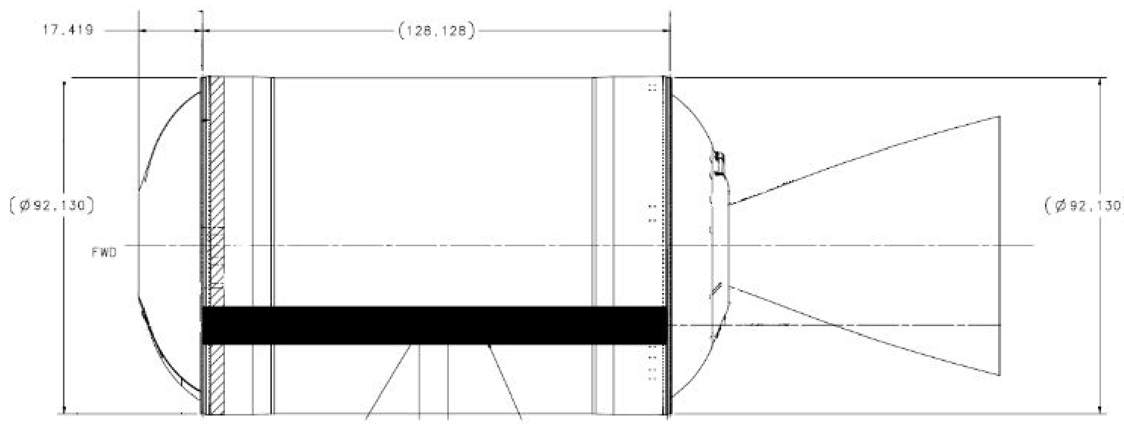

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Figure 7: Castor 30 Series Drawing

Due to the dome protruding over the normal plane of the rocket motor ring, the device must be able to angle vertically to account for the profile of the dome. A splined shaft design was made to allow the device to angle the mounting arm vertically. This spline design can be seen in figure 8 below.

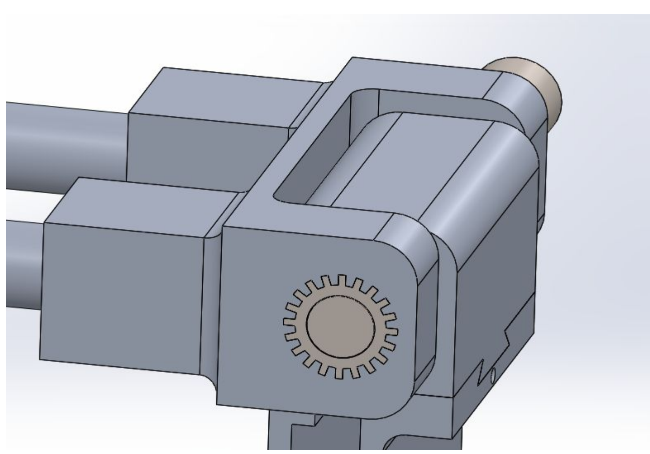

Figure 8: Spline Shaft Design

Due to the teeth of the spline, the axial force could cause damage to the design. This would make the design not lock in a vertical position. In order to prevent this from occurring, an in depth analytical analysis will be performed spring semester. This analysis will focus on mechanics of materials topics including gear teeth, and rotational locations. This analysis will allow the design team to safely select the amount of teeth the spline shaft should have. If the applied axial force is within the 3.0 factor of safety minimum described in the project description section, and whether the spline will be a suitable design for the project team to use. These analyses will be conducted with hand calculations along with an excel sheet that allows the user to change various design variables such as spline teeth and angle.

4.1.4 Potential Critical Failure 4: Rocket Ring Clamp Slips Off

In order to mount onto the rocket motor ring, a clamping device must be designed to secure the device in place. This will allow the mounting arm to hold in a locked position while the axial forces are applied. However, as the axial force is applied, the grip the mounting arm has on the rocket motor ring could loosen and cause the clamp to slip off the locked position. This failure could result in the device, the rocket motor dome, or an operator to be damaged or hurt. In order to prevent this from occurring, a clamping analytical analysis will be made. The primary goal of this analysis, as described in section 3.2, is to solve the exact load distribution along the ring (how much clamp area should be used to disperse the force along the ring) and the necessary clamping force needed to support the design. Solid mechanics hand-calculations, Solidworks FEA, and physical experiments will be conducted by the team. The experiment will be conducted for ME495 lab with the primary purpose of solving the optimal load distribution along the ring which will utilize pressure sensors and strain gauges. Results from the solid mechanics hand-calculations and the experiment will be used to redesign the vise grip to the dimensions that best suit the team's current design along with a Solidworks FEA calculation to prove that the clamping mechanism is feasible.

4.1.5 Potential Critical Failure 5: Force Block Slides due to Axial Force

Described in section 4.1.3, a rail system has been a possible solution for this project. This design will allow the translation of the bracket to be easily performed, while eliminating the use of an articulating arms multiple locking positions. In order to secure the force block in place, a locking mechanism must be designed to secure the location of the axial force. As of now, four rail locking rings will be designed to lock the force block on each rail in each location. With the axial force that will be performed by the device, it is possible for these locking mechanisms to fail. This would cause the force block to slide during the axial force which would make the device not have the ability to be locked into place. In order to mitigate this from occurring, a material analysis will be made on the locking mechanism to verify that the lock will not break and perform as needed. Once this is performed, another FMEA will be made to determine the likelihood of this occurring and if the RPN is still relatively high another analytical analysis of the frictional coefficient and clamping forces will be made to determine if this design should be continued with in the final design or if another option should be made.

4.1.6 Potential Critical Failure 6: Bracket Clamp Slips Off

As the project description states, the standoff mounting device must firmly hold the standoff template brackets in place while the adhesive cures. In order to do this, a clamping device was designed by the project team to hold the brackets in place while the axial force testing is conducted. It is possible that while the axial force is done, the bracket clamp could slip off the bracket template. This would prevent the mounting arm from securing the brackets in place as the customer required. This is similar to the rocket ring clamp slipping off as stated in section 4.1.4, however less likely since a lesser moment will be applied to the bracket clamp. However in order to verify that this will not occur, the testing stated in section 3.2 will be referenced along with a lesser clamping force calculation that will be performed by the design team in the spring semester. An FEA analysis will be done with the designed clamp and bracket templates to verify that the hand calculations that were made were correct.

4.1.7 Potential Critical Failure 7: Lead Screw Breaks

In order to translate the axial forces required for the design, a lead screw is required. This power screw design will allow the operator to apply a force with a drive nut, and then keep that force locked in place by the thread position. Failure could occur from the axial force applied on the device which could cause the lead screw to deform. This would make the device unable to apply an axial force to the bracket templates, which in turn caused the device to be inoperable. In order to mitigate this failure from occurring, a power screw analytical analysis will be performed. This analysis will involve determining the right conditions for the screw to be self-locking as well as its ability to provide adequate push and pull force to meet the client's requirements. The objective of this analysis is to find which elements of the power screw directly benefit the project such as the thread form, pitch, efficiency, cost to procure, cost of maintenance and operability. The project could benefit from this analysis by the discovery of the weight incurred by the screw, as well as its ability to apply axial loads to the bracket holder at multiple angles. The length and width of the power screw will also be considered for this evaluation and help finalize the design of the rail cart sub-system.

4.1.8 Potential Critical Failure 8: Bending of the Rails

As described in section 4.1.5, a rail system has been considered by the design team, which would allow the translation of the bracket to be easily performed, while eliminating the use of an articulating arms multiple locking positions. The draw back of this design however is the fact that a deflection or bend in the rails could occur during testing due to the axial forces being applied onto the rocket motor ring. This could cause the device to be unable to translate the brackets to the appropriate location thus making the device inoperable. In order to mitigate this failure from occurring, a rail deflection calculation will be performed as described in section 3.3. The project description requires a 20 lb push and 50 lb pull test conducted between 4" and 36" inward of the rocket motor ring. At the maximum 36", the 50 lb pull test will generate a large bending moment and shear force within the rails. This analysis will cover the force and bending analysis within the rails and will analyze different materials to predict actual movement and reactions within the rails. These tests will be conducted with hand calculations and backed by MATLAB code and Solidworks FEA analysis to verify the values. These calculations will greatly aid the team in determining what materials are used for the design and how the device will theoretically perform.

4.1.9 Potential Critical Failure 9: Force Block does not Slide

As described in the previous subsection, the rail system considered by the design team will allow operators to slide the force block into position much easier than that of an articulating arm. For this to work appropriately, the force block must be able to slide easily into the correct position. In order to do this, bearings will be installed into the force block to allow operators to easily position the device. As with many other failures described in this section, the axial force performed by the device could result in the bearings to break resulting in the force block not sliding into the correct position. This would cause the device to not meet the translating the bracket sub function. Since the bearings will be at a location where the max force will be applied to the device, it was determined an analysis should be conducted to determine if this can be applied to the final design. This analysis will compare bearing designs to determine: methods to applying bearings to the device, if the bearings would fail under the max load, what materials should be used for the bearings, and if the bearings will cause damage to the rail system under the load of the device. After these analyses it will be determined if bearings could be used on the device, and if so which ones. This analysis will be conducted with hand-calculations as well as MATLAB or an Excel worksheet to verify the values are correct, as well as to create visual aids on the force analysis of the bearings materials.

4.1.10 Potential Critical Failure 10: Fish Scale does not Read Correctly

Currently at Northrop Grumman to test that the brackets are placed in position and will not come off, a fish scale is used to pull the brackets with 50lbs of force to verify that the brackets are in place. The client has asked that the design team implement this into the final design. In order to do this, a fish scale will be placed on the mounting arm with a spring with a given k value. This force scale can be seen in figure 9 below.

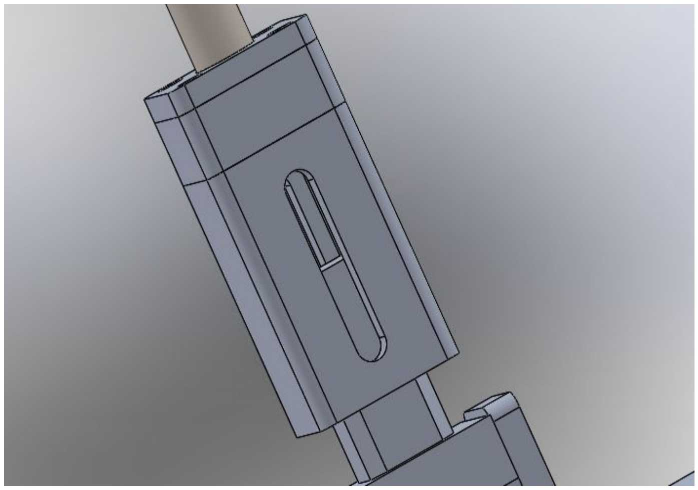

Figure 9: Force Scale Design

Currently the force scale will use the spring implemented inside to allow the reader to see exactly how much force is being applied to the brackets. The problem with this design is that if the spring used becomes deformed due the axial force and regular use, the force reading would become inaccurate. This make the operators possible apply a different force to the rocket dome either making the bracket become not secured onto the rocket motor dome, or damaging the rocket motor dome. In order to verify that this design will not have this failure occur, an analytical analysis of springs that could be used in the design will be performed. This analysis will take various spring lengths, coil counts, and materials placed against one another to see which spring would function the best in the final design. For this analysis, an excel sheet will be made to allow the user to change various design variables to verify which spring is the most applicable to the final design.

4.2 Risks and Trade-offs Analysis

Each critical failure discussed in the previous section has potential solutions that would mitigate the likelihood of that failure from occurring. There are trade-offs to these solutions however as each solution could increase the likelihood of another failure occurring.

To start the best way to mitigate the rocket motor ring from bending is by making the clamp of the device cover as much of the rocket motor ring as it can. At a certain point however, the clamp would have a more difficult time providing the necessary force required to keep the device in place, making it more likely for the rocket ring clamp to slip off. Covering more circumference as well will cause there to be less room for other mounting arms to be placed on the rocket motor dome. Therefore an analysis must be made on how much circumference the clamp can apply to while also being secured on the rocket motor dome safely. The results of the clamp testing and rocket ring moment analysis will allow the design team to select the correct clamping design.

Many potential failures also affect the lead screw design. A likely solution for the bracket clamp from slipping off would be to create more material that the clamp would grab onto. This however would cause more weight to be added to the device which would cause more stress on the lead screw and the rails of the device. The type of spring that the fish scale would use would also affect the lead screw which will need to be taken into consideration. ‘

To prevent deformation of the rails, more material would be the first solution that would come to mind. However, the additional weight would prevent the operators from easily transporting the device. Also due to the mounting position, more weight would cause more deformation toward the rocket motor dome. The solution would likely be selecting an appropriate material the rails could use. This would however affect the force block sliding and not sliding as the frictional coefficient of the rails would change as the material changes. These changes will be determined by the rail deformation test described earlier in this report. Then this will be compared to the bearing analytical analysis to determine if the force block will be able to slide on the material that will deform the least with the highest factor of safety.

The spline shaft solution will rely on the gear analysis that will be performed. The likely solution will be that the material for the spline shaft will need to be strong to verify that the teeth displayed in figure 7 do not break. The material chosen could affect the rocket ring clamp that the spline shaft will be attached to, which will need to be taken into consideration during the gear analysis. Likely what will occur will be after the clamping test is performed, the gear analysis will be done to determine the specifications for the spline hub. This will then be compared to the factor of safety determined in the clamp testing to verify that the selections do not affect the rocket ring clamp factor of safety.

5 DESIGN SELECTED – First Semester

Given the complexities involved with both the engineering requirements and goals of this project, many considerations had to be made to ensure that the overall function was as expected when implementing this design. The following contains the considerations made in the design process, including the final choices that the team will proceed with in manufacturing. Also described is the plan for the physical construction of the device, including processes and material choices. The entire CAD model is shown as well, which serves to provide a visual of the design as a whole and communicate design intent.

5.1 Design Description

The nature of this design project calls for a design that can move and lock in all degrees of freedom, while maintaining a rigid platform for the transmission of a load. This set of requirements created a modular design process in which certain design decisions could be made independently of one another. The final choices that were made for each of the primary subfunctions are discussed in the subsections below, working from the rocket motor inward.

5.1.1 Motor Ring Clamp

The clamping system for the rocket motor ring involves attaching the entire design to a thin piece of aluminum, which will then be used as an anchoring point during operation. This has the potential to create a large moment about the motor ring when the pull test is performed with a long lever arm, which must not damage the aluminum ring.

In order to provide a simple method for engagement of the clamp, the team opted to utilize an aluminum rail with a single attached screw system. This decision allows for a large amount of force to be applied across a section of the ring, while positioning the movable components outside of the motor itself where they can be easily accessed. A simplified version of this system is shown in Figure 10 below, which is subject to change based on the machinability of the components.

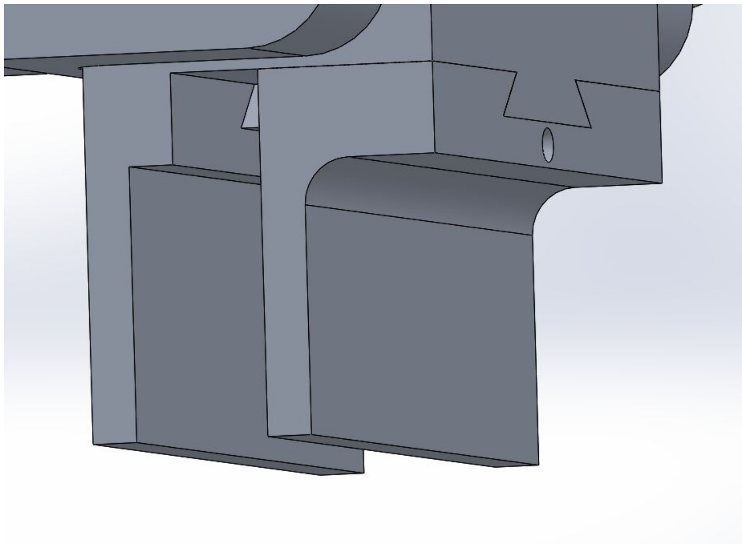

Figure 10: Clamping Mechanism for Attachment to Motor Rings

This design was derived from the large clamping force that may be exerted by a standard vice with little relative input from the user. This is to ensure that the device will not slip from the ring during operation, as such a failure could cause damage to nearby componentry. The width of the clamp was decided with a Solidworks FEA analysis using the large load from the pull test combined with the maximum lever arm; creating the worst case scenario. The resulting factor of safety for the ring in the most vulnerable location was 3.3 with a clamp contact width of 8 inches, which exceeded the 3.0 necessary for the design. The resulting stress distribution is shown in Figure 11 below.

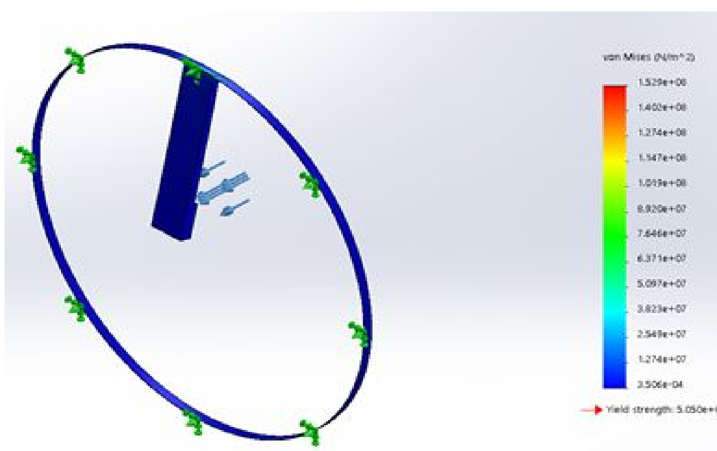

Figure 11: FEA Analysis of Moment Arm on Motor Ring

The only location on the ring which experiences significant stress is near to the fixed edge in the FEA simulation, which is part due to the unrealistic nature of a fully fixed surface. This would be far less in practice, given that the material that the ring attaches to would be allowed to elastically deform slightly. Combined with the positioning of this test at the worst case scenario, the team felt confident in allocating an 8 inch clamp width for the ring attachment.

Because the rotor motor rings have fairly minimal principle dimensions, and given the potential for a large moment about them, it was desirable for the team to maximize the distribution of load across as much of the surface as possible. For this to happen, individual clamp jaws must be utilized for each of the two ring geometries that this device is to be used with. A potential jaw for the Orion 50 motor ring is shown in Figure 12 below.

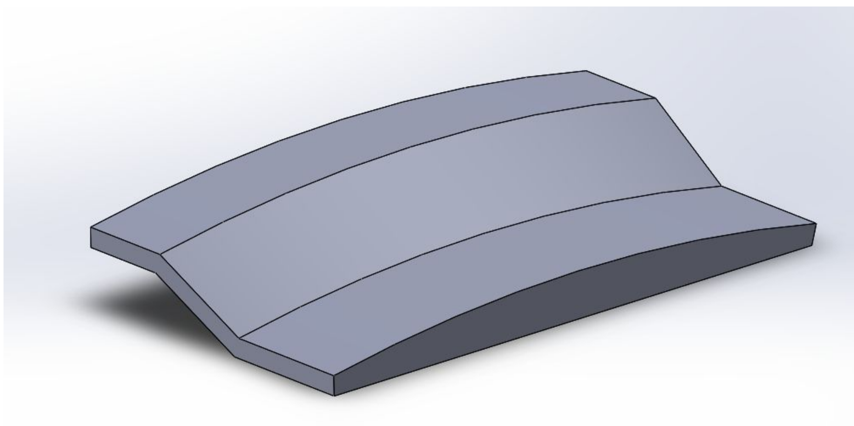

Figure 12: Custom Clamp Jaw for Orion 50 Motor Rings

This clamp jaw was designed to conform to the precise geometry of the Orion 50 motor ring, which allows for a more even stress distribution as well as an increase in clamp area for frictional purposes. While it would not propose an issue for the team, it should be noted that a CNC machine will be required for the manufacturing of these custom clamp jaws.

5.1.2 Spline Shaft for Rail Angle

As shown in Figure 7, the Castor 30 series of motors features a dome which protrudes beyond the mounting ring. This variance will require the ability to angle the device above the raised surface, allowing for installation at any point on the dome. The solution for this angle change needed to be both easily manipulable by hand and provide the necessary strength to withstand a large moment. The team decided to use a spline shaft to accomplish this goal, which would accommodate the change in angle with a set of teeth on both halves of the rotational system. The integration of the spline shaft into the design can be seen in Figure 13 below.

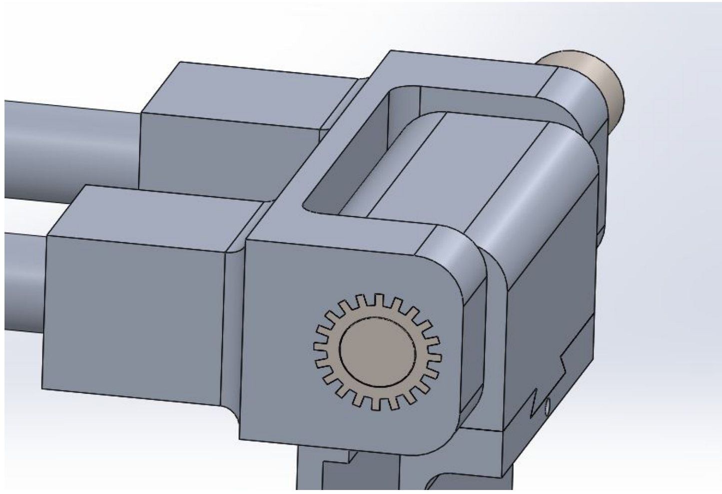

Figure 13: Spline Shaft used to Adjust Rail Angle

Unless complicated tool paths are used to create a spline shaft with a CNC machine, these components will likely be outsourced for production. The depiction in Figure 13 will adjust based on the components purchased, but would optimally contain as many teeth as possible for positioning of the rail system near to the dome surface.

This system is to be accompanied by an alignment feature that will keep the two components together while the angle of the rails is changed. Two options are present to ensure that this can be done, with the final decision depending on what spline shafts and hubs are available from suppliers. One option is to remove the splines from portions of the shaft, allowing for the shaft to be translated for tooth disengagement while the inner diameter keeps the rail and clamp systems together. Another option involves sleeving centralized hubs into the outer portion of the clamp, which would maintain the location of the components when the shaft was removed.

5.1.3 Rail System

The need to translate the standoff brackets to various distances inward from the rocket motor ring required a subsystem that was either extendable or could otherwise position the bracket. Due to low weight and simplicity, as well as the ability to position at any point inward from the ring, a rail system was chosen to serve this purpose. The rail set width was set to distribute the load evenly across the ring clamp while maintaining a small form factor with high maneuverability. The width also allowed for sufficient area to mount the central components, such as the lead screw and nut, between them. A depiction of the rails extending from the ring clamp is shown in Figure 14 below.

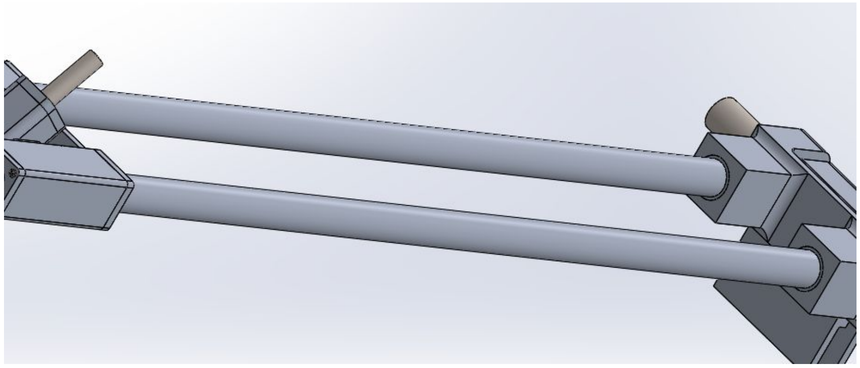

Figure 14: Rail System

As with the ring clamp system, the diameter of the rails was determined based upon the deflection and factor of safety in the worst case scenario for the device. With a 36 inch rail length, the maximum deflection from a 50 pound load can be found using equation (1), where I is the moment of inertia for the rail cross section. To minimize the deflection while maintaining a high factor of safety, low weight and high corrosion resistance, 7075 aluminum was chosen for this application.

$$\delta = \frac{PL^3}{3EI} \qquad [1]$$

Considering an elastic modulus of 10400ksi and an even distribution of the load between the rails, the maximum expected deflection is 0.83 inches with a rail diameter of 0.98 inches. This diameter was chosen to accommodate drilling a 1 inch hole in the cart, while the rail would be made on a lathe and could facilitate a sliding fit [4]. While more calculations will be made in the analytical reports to ensure that this material and geometrical choice was optimal, early FEA provides a factor of safety much larger than the minimum requirement for this project.

5.1.4 Rail Cart and Lead Screw

With the rail system already chosen, the design decisions for the cart revolved around maintaining the required feature set and strength while minimizing weight. As the stresses on this material were lower due to the axial, non-moment-inducing loads, cheaper 6061 aluminum with high machinability, low weight and the same corrosion resistance was selected. This system is shown in Figure 15 below, and was set up to allow for the angling of standoff brackets normal to the rocket motor dome. This was implemented to ensure that the force was evenly distributed across the standoffs and that the pull test could be performed at the specified 45 degrees.

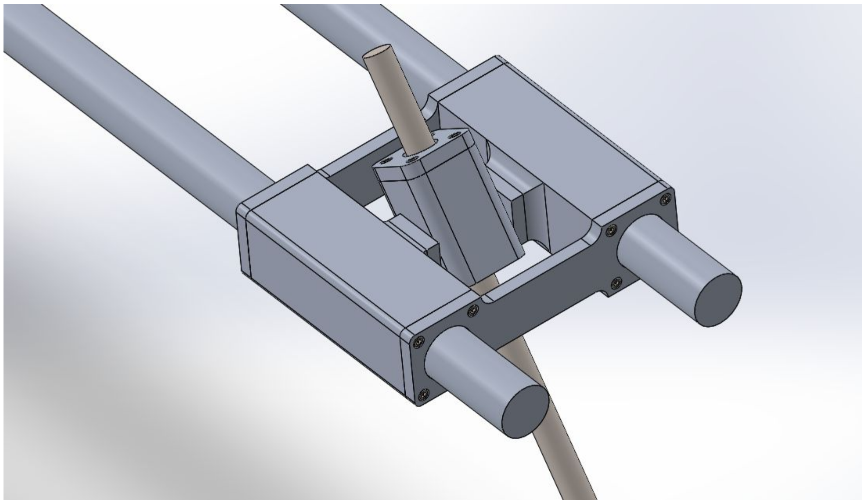

Figure 15: Rail Cart and Angleable Lead Screw

The rail cart itself is braced by plates at the front and rear, which maintains subsystem rigidity when a load is applied through the lead screw. The changes in angle are facilitated through cylindrical rods, which are part of the centerpiece and insert into the sections which lie directly on the rails.

The total weight of the aluminum cart pieces is less than 3 pounds, while the use of a plastic lead screw nut also serves to decrease weight. The total weight of the stainless steel lead screw, which was chosen for corrosion resistance properties, will depend on the length needed for the application. However, less than 1 pound for this component is expected when considering the given rocket motor geometries. More calculations will be performed in a future analytical report to verify the dimensions and locking ability of the lead screw. The addition of a knurled nut at the top of the screw will facilitate extension and force application to occur during the use of the device.

The component strength is far beyond what is required to withstand the load from the 50 pound pull test, but still manages to provide a low total weight that can be easily manipulated by hand. The dimensions of the cart were set to distribute load at the base of the rails, but the maximum principle dimension of 5 inches ensures that it will not be cumbersome in use.

5.1.6 Measuring the Applied Force

The given tolerances of ±2 pounds allow for the force to be measured with low resolution instrumentation. An issue faced by the team, however, is that the force gauge must be integrated into the design and must therefore align axially with the lead screw. The proposed solution to this issue placed a spring usable for extension and compression between the lead screw and standoff bracket. This would allow for the change in length of the spring to be observed and reported as a force, assuming that an accurate spring constant could be determined. The current housing design for this gauge can be seen in Figure 16 below.

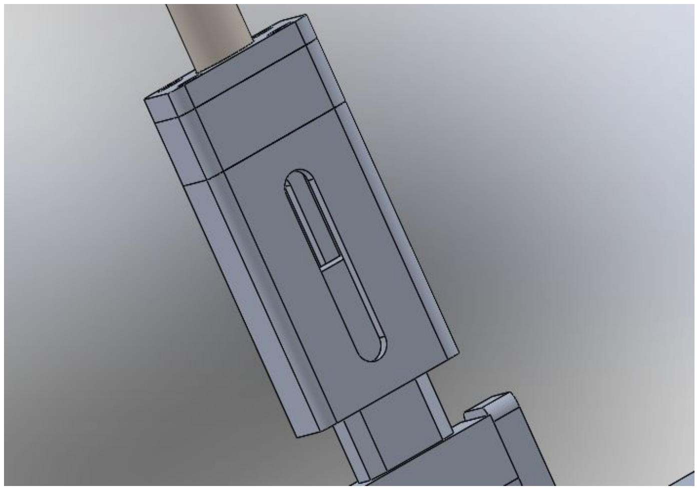

Figure 16: Force Gauge Spring Housing

The optimal spring constant for this device will be determined during testing, which will then allow for marks to be made on the housing with the desired loads applied. This method allows for a lightweight piece of custom instrumentation with the desired accuracy that, aside from the spring itself, can be machined in-house.

5.1.7 Setting the Cure and Pull Test Angles

For this device to operate as intended, the application of a force normal to the surface must be made to cure the adhesive at the base of the standoffs. After this is completed, the angle relative to the bracket must be set to 45 degrees to proceed with the pull test. A joint with pin holes drilled at the necessary locations for these settings is to be positioned at the end of the lead screw assembly. This joint can be seen in Figure 17 below.

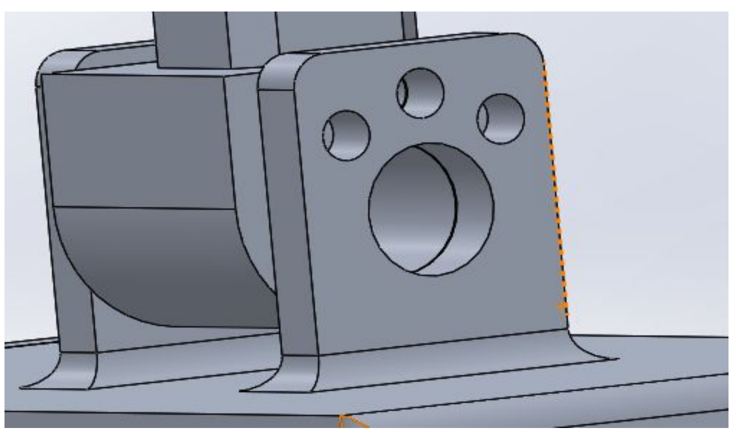

Figure 17: Joint for Setting Angle Relative to the Dome

The larger hole shown in this design carries the bulk of the load created by the lead screw, ensuring that the junction remains rigid. The smaller holes will house a pin that will set the angle of the applied force relative to the surface. This pin is small relative to the center pin, as it is only used to verify that the testing is performed at the correct angle.

5.1.8 Retaining the Bracket

Standoff brackets of varying sizes will be used in conjunction with this device, which will require a subsystem that can accomodate all of them and transmit the load to the surface of the motor dome. The current design for this subsystem is shown in Figure 18 below.

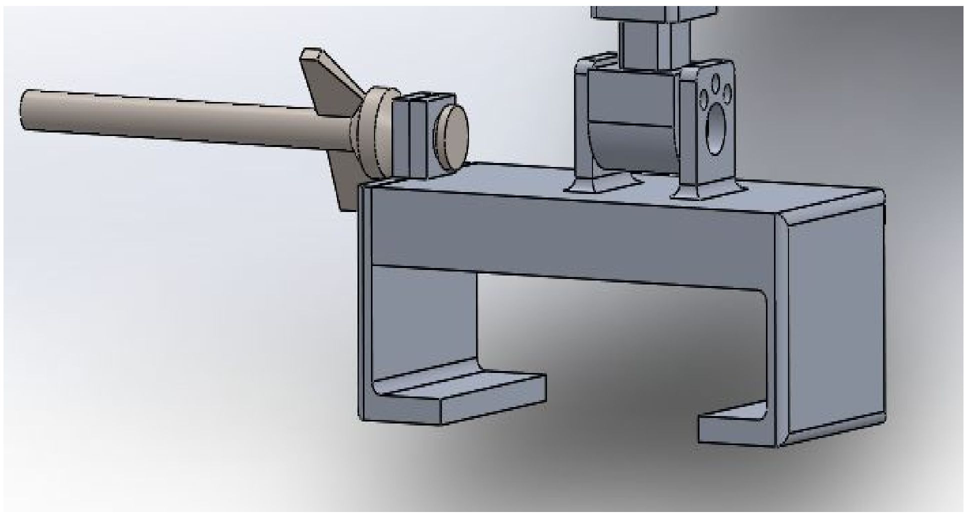

Figure 18: Bracket Retention Subsystem

This design utilizes a rail, similar to those seen in the tool holder of a lathe, to slide two halves of the clamp relative to one another and change the system size. A simple wing nut and stud combination will make using the clamp easy for any operator while ensuring that a force can be applied to keep the standoff bracket in place.

5.1.9 Initial Prototype

Prototyping for this design allowed for physical interaction with the geometry of the design, using different materials of course. The majority was 3D printed, while PVC piping was used to simulate the rail system. This initial design iteration can be seen in Figure 19 below.

Figure 19: Initial Prototype

From interacting with the prototype, the team was able to verify that the bulk of the final design could be easily manipulated by hand, although certain components were not present. While the coefficient of friction for these materials differs from the final materials, this prototype still provided an idea of what the sliding fit on the rails will feel like when in use. The design behind the spline shaft was greatly simplified for this prototype, but still was able to to show the team that a looser fit would allow for better manipulation of the rail angle, which will be a factor if this component is machined in-house. Based on the feel of this design, an end cap may also be added to the end of the rails for better parallel alignment.

5.2 Implementation Plan

The following section of the report details the processes that will be performed in the future to convert this conceptual design into a physical product. This includes how the components will be manufactured or procured, the materials and other resources that will be required, as well as a general schedule of future events and depictions of the final assembly.

5.2.1 Component Machining and Procurement

In order to maximize the learning potential of this project for all members of the team while also positively impacting the budget, as many components as possible will be machined in the NAU machine shop. This will of course increase the required workload during the manufacturing phase, but will undoubtedly benefit all involved in this project.

While the entire team has completed or will soon complete mill and lathe training, some of the more complex parts of this design will require the use of CNC machining. If at all possible, the entire team plans on receiving this training as well so that all aluminum components may be machined without outsourcing. Some parts, however, such as the lead screw, spline shaft, nuts for both of these and all other screws, will be purchased from a vendor due to the complex geometries presented. All components that are expected to be purchased are shown in Appendix D, which contains the current bill of materials for this project.

5.2.2 Expected Materials Overview

The majority of the material used in this design is 6061 aluminum, with 7075 aluminum also included for the construction of the rails. All of this will be purchased as a large piece of aluminum stock, which will then be segmented for the machining of the individual components. This will save the team a significant amount of money while also providing valuable machining experience and the ability to alter finished parts if needed.

The purchased components, such as the lead screw and spline shaft parts, are organized into the bill of materials within Appendix D. A total cost of $994.13 has been allotted for this design, which includes both finished parts and raw materials. When compared to the $10000 maximum budget that is allowed for this project, there is plenty of room remaining for iteration and testing.

5.2.3 Schedule

The schedule of remaining deliverables for this design project is shown in Figure 20 below, with due dates to the right and personal goals for finalizations to the left.

Fall Semester				
Final Report	Everyone	0%	11/6/19	11/15/19
Final BOM/CAD Package	Sage, Dakota	0%	10/14/19	11/20/19
PDR Presentation	Everyone	0%	10/28/19	11/18/19
Individual Analytical Report	Everyone	0%	10/14/19	11/28/19
Final Prototype	Everyone	0%	11/20/19	12/4/19
Website Check 2	Tyler, Brandon, Elaine	0%	11/6/19	12/9/19
Spring Semester				
Post Mortem	Everyone	0%	1/6/20	1/15/20
Self-Learning	Everyone	0%	1/6/20	1/24/20
Hardware Review	Everyone	0%	1/15/20	2/14/20
Website Check 3	Tyler, Brandon, Elaine	0%	2/1/20	2/21/20
Midpoint Presentation	Everyone	0%	2/17/20	3/4/20
Midpoint Report	Everyone	0%	2/17/20	3/6/20
Individual Analysis II	Everyone	0%	2/1/20	3/13/20
Final Product Finished & Device Summary	Everyone	0%	1/15/20	3/25/20
Drafts of Posters	Everyone	0%	3/1/20	4/1/20
Testing Proof	Everyone	0%	3/25/20	4/8/20
Final Poster and Operation Manual	Everyone	0%	4/1/20	4/15/20
Final Presentation (UGRADS)	Everyone	0%	4/1/20	4/24/20
Final Report and CAD Package	Everyone	0%	4/6/20	4/29/20
Website Check 4	Tyler, Brandon, Elaine	0%	4/29/20	5/4/20

Figure 20: Schedule

This schedule is tentative, and will be updated with changes as time progresses. Some events, such as days to be spent machining, will be added during the spring when this is sorted out more accurately.

5.2.4 Final CAD Assembly

Shown in Figure 21 below is the CAD representation of the final design that is being proposed by the team. All subfunctions of this design have been covered at earlier points in the report, but the final model serves both as a visual aide and to communicate design intent as a whole.

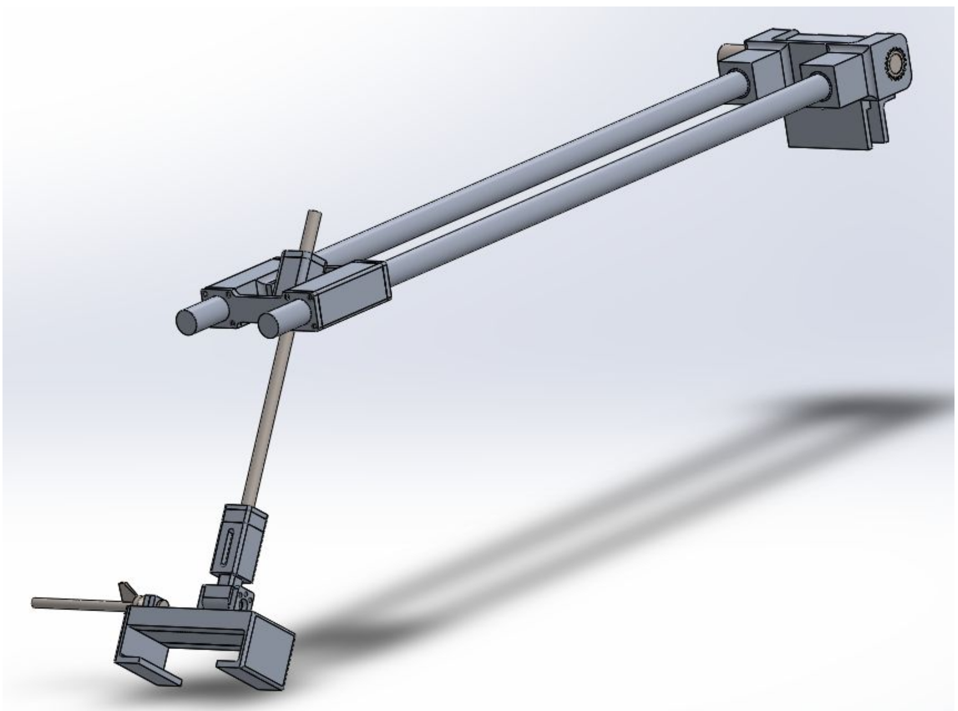

Figure 21: CAD Assembly

To provide a sense for the assembly process, as well as to clarify how the components come together as a whole, an exploded view of the design can be seen in Figure 22 below. The cart portion of the design is greatly clarified by this exploded view, as many different components merge to create this subsystem.

Figure 22: Exploded View

The complexity of this design is more easily communicated through the exploded view, as many components must be put together to successfully meet all of the customer requirements. Machining all of these components will require a lot of planning and preparation from the team, but will also provide a very rewarding experience.

6 CONCLUSIONS

The Northrop Grumman Standoff Project team was tasked with designing and manufacturing an articulating arm that will effectively hold standoff mounting templates in position for the duration of the adhesive cure process. The teams design looks to replace the current taping method used by Northrop Grumman technicians when mounting avionics electronics to the forward and aft domes of various rocket motors. The design created by the team must be durable, reliable, transportable, easy and safe to use, ESD compliant, have 6 degrees of freedom, apply axial forces, within budget, functional within 4”-36” inboard of the ring, and all components must have a factor of safety of 3.0. The project client at Northrop Grumman added that the design should allow for multiple devices to be mounted to the ring at the same time so that multiple standoffs can be applied simultaneously.

This report covers a brief summary of the concept generation and concept evaluation stages completed by the team previously in the semester. The report details codes, regulations, standards and possible failures of the design that the team must account for when moving forward with the project. With a strong understanding of how the design could potentially fail, and how these failures would affect the design and rocket motor, the team can ensure that all failures are minimized. Moving forward with the project, the team plans to implement various tests that focus on critical functions of the design. Testing the mounting mechanism to the motor ring, rail deflection, power screw mechanics and ESD compliance will ensure that the device functions as the team wants.

The final portion of the report details the current state of the teams CAD assembly and the main sub-function components. Each component is not completely finalized and has the potential to be upgraded and improved. From the CAD assembly the team constructed a low-fidelity prototype as seen in Figure 19. This prototype allowed the team to have a hands on feel for how the device should physically function. This report is a summary of the entire semester and all the work the team has done on the project to date.

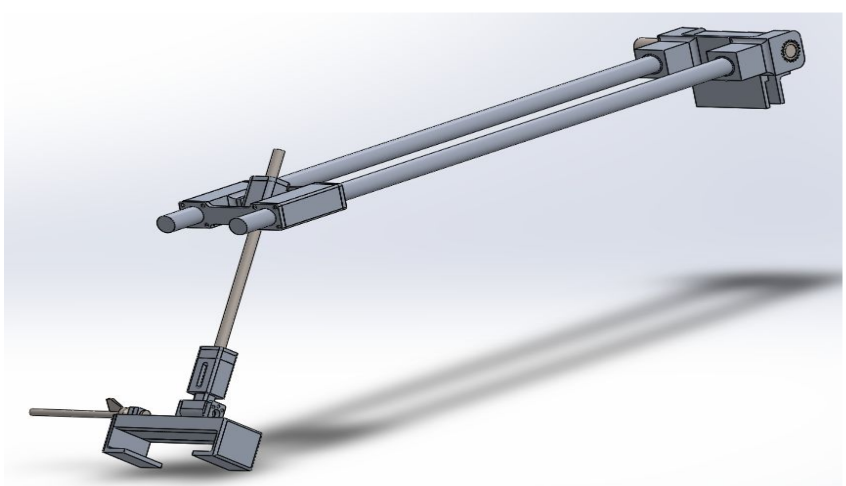

Figure 23: Final CAD Assembly

As seen in Figure 23 above, the team designed a rail system that utilizes a sliding cart to position the standoff brackets where they are needed. The rails are adjustable vertically to account for different motor dome profile between the Orion and Castor models. A power screw located in the center of the cart will provide the axially forces needed to push and pull the standoff brackets as needed. A force gage located at the end of the power screw will display the force being applied by the device. The team feels confident that the design created will meet or exceed all the customer needs and engineering requirements stated in section 2.

Moving forward, the team has a preliminary design presentation in Chandler, AZ with critical members of the Northrop Grumman company. This meeting will ensure the team is on schedule and the client is satisfied with the design before testing and manufacturing can commence. Next semester the team expected to construct a high fidelity prototype that can be extensively tested and analyzed. From this prototype and test results, the team will begin final design construction which will be completed before the Northrop Grumman Symposium and UGRADs at the end of spring semester.

7 REFERENCES

[1] Propulsion Products Catalog, Northrop Grumman, Falls Church, VA, June 2018

[2] "Aluminum Alloys - Mechanical Properties," The Engineering Toolbox, 2008. [Online]. Available: https://www.engineeringtoolbox.com/properties-aluminum-pipe-d_1340.html. [Accessed: 18-Oct-2019]

[3] "List of all Codes and Standards: ASME," *ASME*. [Online]. Available: https://www.asme.org/codes-standards/find-codes-standards. [Accessed: 16-Nov-2019].

[4] "Aluminum 7075-T6; 7075-T651," *ASM Material Data Sheet*. [Online]. Available: http://asm.matweb.com/search/SpecificMaterial.asp?bassnum=MA7075T6. [Accessed: 16-Nov-2019].

8 APPENDICES

8.1 Appendix A: Rocket Motor Catalog

8.1.1 Appendix A.1. Orion 50XL

8.1.2 Appendix A.2. Castor 30XL

8.1.3 Appendix A.3. Castor 38

8.2 Appendix B: QFD

Customer Need	Weight	Engineering Requirement	Electrically Conductive (Y or N)	Mass (slugs)	Principal Dimensions (in)	Working Length (in)	Working Angle (Degrees)	Modulus of Elasticity (lbf/in^2)
1. ESD compliance	0.09		9	0	0	0	0	0
2. Apply axial forces	0.09		0	1	0	3	3	9
3. Six degrees of freedom in movement	0.09		0	0	0	9	9	0
4. Usable 4" - 36" inboard of ring	0.09		0	1	9	9	3	1
5. Transportability	0.04		0	9	9	3	3	0
6. Ease of operation	0.07		3	9	3	9	9	0
7. Durability	0.08		0	3	0	0	0	9
8. Reliability	0.08		0	3	0	0	0	9
9. Adjustable Interfaces	0.09		0	3	0	3	3	0
10. Support 10lbs in locked position	0.09		0	3	0	3	3	9
11. Minimum 3.0 Factor of Safety	0.06		0	3	0	0	0	9
12. Within Budget	0.03		9	9	3	9	3	9
13. Multiple Arms	0.05		0	3	3	3	3	0
14. Safe Operation	0.05		9	9	3	0	0	3
Absolute Technical Importance (ATI)			1.74	3.24	1.77	3.60	2.88	4.11
Relative Technical Importance (RTI)			0.42	0.79	0.43	0.88	0.70	1.00
Target Values			Yes	25	8" (W) x40" (L) x6" (H)	32"	360°	<10.4 *10^6
Target Tolerances			N/A	±5	±2	N/A	N/A	N/A

8.3 Appendix C: FMEA

Product Name	Standoff Bonding Tool	Development Team		FMEA Number	2
			18F19	Date	11/12/19

Part #	Function	Potential Failure Mode	Potential Effects for Failure	Severity (1-10)	Potential Cause(s) for Failure	Occurance (1-10)	Current Design Controls Test	Detection (1-10)	RPN (SxOxD)	Reccomended Action(s)
2	Mount to ring	Bending of the ring	Rocket Motor Ring will break	10	Overstressing the ring	5	Visual Indicator	8	400	Moment Ring Analytical Analysis
		Rocket Motor Ring is damaged			Overstressing the ring	1	Visual Indicator	8	80	
					Stress fractures occur due to regular use	1	Visual Indicator and assembly check	3	30	
		Clamp slips off	The device will be unsupported	8	Inefficient clamping force	3	Visual Indicator	8	192	Clamp Force Analysis
		Clamp Breaks			Overstressing the clamp	1	Visual Indicator	8	64	
					Sliding portion snaps	1	Visual Indicator	8	64	
		Clamp slide attachment is deformed			Overstressing of the clamp	1	Visual Indicator	8	64	
		Clamp attachments fracture			Attachments are maufactured poorly or stress fractures occur over regular use	1	Visual Indicator	8	64	
		Clamp teeth shear			Stress fracture occurs from manufacturing or regular use	1	Visual Indicator	8	64	
					Moment of the mounting arm causes severe stress on the teeth	1	Visual Indicator	8	64	
12	Translate the Brackets	Shear of center pivots in force block	Unable to apply axial force	7	Overstressing the rails	1	Audible and Visual Indicators	6	42	Factor of Safety Analysis
1		Bending of the rails	Device is not able to translate toward the standoff location	7	Overstressing the pivots	2	Visual Indicator	8	112	Deformation Test
4		Force Block does not slide			Friction force is too great	1	Audible and Visual Indicators	5	35	Bearing Analytical Analysis
					Ball Bearing Breaks	2	Visual Indicator	8	112	
1		Rails are damaged			Ball bearing would cause cracks or surface damage	1	Visual Indicator	8	56	
					Sliding would create scratches due to the metal frame of the force block	1	Audible and Visual Indicators	5	35	
					Weight of the rails cause deformation	2	Audible and Visual Indicators	5	70	Deformation Test
13		Force block locks are damaged	Force block is not able to be held in place	7	Formation surface damage of the rails	1	Audible and Visual Indicators	5	35	
					Force Block creates a greater sliding force than the locking force	1	Audible and Visual Indicators	5	35	Bearing Analytical Analysis
					Stress fracture occurs from manufacturing or regular use	1	Audible and Visual Indicators	5	35	Material Selection Analysis
9	Hold Bracket	Clamping screw breaks	Unable to clamp standoff bracket template	8	Stress fracture occurs from manufacturing or regular use	1	Visual Indicator	8	64	Mechanics of Materials Life Cycle Analysis
					Thread shears off	1	Visual Indicator	8	64	Thread Material Properties Analysis
3		Clamp jaw deforms			Stress fracture occurs from manufacturing or regular use	1	Visual Indicator	8	64	Material Selection Analysis
					Clamp shape does not match the bracket template being installed	1	Visual Indicator	8	64	
		Clamp jaws break			Stress fracture occurs from manufacturing or regular use	1	Visual Indicator	8	64	
					Overstressing of the clamp	1	Visual Indicator	8	64	
		Mounting arm cannot be adjusted	Unable to hold standoff bracket	7	Bracket Template mass exceeding mounting arm maximum load	1	Visual Indicator	8	56	
		Bracket slips off			Incefficent clamping force	3	Visual Indicator	8	168	Clamp Force Analysis
		Bracket breaks	Standoffs are not secured in place	8	Clamping force exceeds material max force	1	Visual Indicator	8	64	
					Clamp and bracket contact is minimal causing deformation	1	Visual Indicator	8	64	

22	Apply axial force	Lead Screw breaks	Unable to apply axial force	8	Force induced deformation of screw	3	Audible and Visual Indicators	5	120	Lead Screw Stress analysis
					Moment induced deformation of the screw	1	Audible and Visual Indicators	5	40	
14		Assembly screws break			Overstressing of the device at the screw locations	1	Audible and Visual Indicators	5	40	Material Selection Analysis
					Stress fracture occurs from manufacturing or regular use	1	Audible and Visual Indicators	5	40	
7		Shaft Collar disconnects	Bracket Clamp disconnects from force block	7	Deformation wear of the securing screw	1	Visual Indicator	8	64	Deformation Analysis
		Lead Screw and Fish Scale box attachment breaks due to axial force testing			Attachment breaks due to continous use or manufacturing	1	Visual Indicator	8	64	Material Selection Analysis
15		Fish Scale does not read correctly	Force will not display to operator	5	Spring does not function with the device	2	Visual Indicator	8	80	Spring Analysis
					Spring deforms over continuous use or from manufacturing	1	Visual Indicator	8	40	
					Fish Scale box deforms over continuous testing	1	Visual Indicator	8	40	
4, 13	Locking	Force block slides during axial force tests	Device can not be applied to specific location	7	Applied force exceeds lock capacity	5	Audible and Visual Indicators	5	175	Material Selection Analysis
					Force Block creates a greater sliding force than the locking force	1	Audible and Visual Indicators	8	56	Bearing Analytical Analysis
					Rail deformation is already present	1	Visual Indicator	8	56	
					Frictional Coefficent on rail clamping system is not sufficent for the device	1	Visual Indicator	8	56	
					Stress fracture occurs from manufacturing or regular use	1	Visual Indicator	8	56	Material Selection Analysis
12		Center Block moves rotationally			Applied force exceeds lock capacity	1	Visual Indicator	8	56	
					Side shearing occurs over regular use	1	Visual Indicator	8	56	
4		Center block cart attachments shear			Stress fracture occurs from manufacturing or regular use	1	Visual Indicator	8	56	
					Rotational movement creates deformation	1	Visual Indicator	8	56	
					Applied force exceeds lock capacity	1	Visual Indicator	8	56	
11	Angle Bracket	Bracket Joint Pin Shear Failure	Inability to carry out the pull test	7	Applied axial force	6	Visual Indicator	8	336	Material Selection Analysis
					Stress fracture occurs from manufacturing or regular use	1	Visual Indicator	8	56	
					Moment of the mounting arm	1	Visual Indicator	8	56	
3		Bracket Clamp to Arm attachment shears	Unable to hold standoff bracket	7	Applied axial force	1	Visual Indicator	8	56	
10		Spline Mounting Screw Shears	Device will be unsupported	8	Maufacturing or regular use	1	Audible and Visual Indicators	5	40	Spline Shaft Gear Analysis
					Axial force	6	Audible and Visual Indicators	5	240	
5, 6		Spline Hub breaks			Axial Force is greater than the gear material can experience	1	Audible and Visual Indicators	5	40	
		Spline Shaft breaks			Axial Force is greater than the pin material can experience	1	Audible and Visual Indicators	5	40	
		Spline Shaft Deforms			Axial Force is greater than the pin material can experience	1	Audible and Visual Indicators	5	40	
					Stress fracture occurs from manufacturing or regular use	1	Audible and Visual Indicators	5	40	
-	Does not carry electro static charge	Does not meet ESD Compliance	Damage electronic components	10	Material is non-conductive	1	ESD Test	1	10	Create ESD Testing Procedure
					Device does not have grounding capability	1		1	10	Create ESD Testing Procedure

8.4 Appendix D: Bill of Materials

Bill of Materials Final Design								
Team				Northrop Grumman Standoff Project				
Part #	**Part Name**	**Qty**	**Description**	**Functions**	**Material**	**Dimensions**	**Cost**	**Link to Cost estimate**
1	Rail	2	41L40 Alloy Steel Round Rod 1/2" Diameter x 3' Length	Mode of translation for the Bracket Holder Mechanism (rail cart).	7075 Aluminum	1/2" x 1/2" x 3'	154.71	https://www.onlinemetals.com/en/search/results?q=aluminum+bar%3Aprice-asc%3AMaterial%3AAluminum%3AShape%3ARound%2BBar%3AAlloy%3A7075&checkbox=on&sort=price-asc#
2	36" Clamp	1	Adjustable Clamp 9133 Corner/Splicing Clamp	Secure the device to the motor ring.	6061 Aluminum	1.5" x 7.8" x 6"		
3	Bracket Holder	2	Custom made bracket holder, to be manufactured in machine shop.	Secure the bracket to the device.	6061 Aluminum	16" x 3" x 2"		
4	Rail Cart Rail Connectors	2	Custom made for fabrication at machine shop	Transports bracket holder across rail system.	6061 Aluminum	4" x 3" x 1"		
5	Spline Rail Connector	1	Custom made for fabrication at machine shop	Adjusts the angle of the device	6061 Aluminum	4" x 8"		
6	Spline Clamp Connector	1	Custom made for fabrication at a machine shop	Adjusts the angle of the device	6061 Aluminum	4" x 8"		
7	Shaft Collar	1	Custom made for fabrication at a machine shop	To prevent the bracket clamp from rotating while the handling arm is lowered onto the rocket dome	6061 Aluminum	1/2" x 4"	240.16	https://www.metalsdepot.com/aluminum-products/aluminum-square-bar
8	92 " Clamp	1	Adjustable Clamp 9133 Corner/Splicing Clamp	Secure the device to the motor ring.	6061 Aluminum	1.5" x 7.8" x 6"		
9	Bracket Clamp Screw	1	Custom made for fabrication at a machine shop	Tighten the bracket clamp around the bracket templates	6061 Aluminum	1/2" x 10"		
10	Spline Pin	1	Custom made for fabrication at a machine shop	Secure spline location in place	6061 Aluminum	1/2" x 2"		
11	Universal Joint Pin	1	Small metal rod	Lock joint in place	6061 Aluminum	1/4" x 2"		
12	Rail Cart Rotational Positioner	1	Custom made for fabrication at a machine shop	Allows team to move the location of the axial force on the rail cart	6061 Aluminum	4" x 4" x 8"		
13	Locking Rings	4	Circular Rubber stoppers that can clamp onto the metal rods	Lock the Force Block in place	Rubber	1/2" x 36"	2.99	https://www.rockler.com/non-skid-rubber-bumpers?sid=V9146?utm_source=google&utm_medium=cpc&utm_term=&utm_content=pla&utm_campaign=PL&gclid=EAIaIQobChMIz52AkMjW5QIVbxitBh39DQsrEAQYBSABEgJ_ePD_BwE
14	Design Screws	24	2" long screws	To assemble seperate parts together in the final assembly	Stainless Steel	#4 x 5/8"	8	https://www.amazon.com/Screws-Thread-Stainless-Self-Tapping-Quantity/dp/B01CRE76IK/ref=sr_1_2?keywords=%234+metal+screws&qid=1573076036&refinements=p_n_feature_nine_browse-bin%3A17426584011&rnid=17426558011&s=hi&sr=1-2
15	Lubricant	1	Super Lube 51004 Synthetic Oil with PTFE, High Viscosity, 4 oz Bottle	Reduce friction of the rail system.	Synthetic PTFE Oil	2" x 2" x 6.5"	4.60	https://www.amazon.com/Super-Lube-51004-Synthetic-Viscosity/dp/B000UKUHXK/ref=sr_1_1?keywords=Synthetic+PTFE+Oil&qid=1570591173&s=industrial&sr=1-1
16	Fastener	3	Black Oxide Alloy Steel Socket Head Screw 1/4-20, 4" Long	Works with wing nut to secure bracket to the clamp.	Black Oxide Alloy Steel	3/8" x 4"	9.25	https://www.mcmaster.com/90044a131
17	Wing Nut	1	18-8 Stainless Steel Wing Nut 1/4"-20	Part of clamping mechanism to secure bracket to clamp.	Stainless Steel	31/64" x 1 1/8" x 5/8"	7.32	https://www.mcmaster.com/92001a321
18	Universal Joint	1	Custom made for fabrication at a machine shop	Universal joint to help position the bracket onto the rocket.	Vanadium Steel	6.5" x 5" x 1.1"	11.98	https://www.amazon.com/TEKTON-4964-Impact-Universal-3-Piece/dp/B000NPZ40I/ref=sr_1_21?keywords=Ball+and+Socket+Joint&qid=1570592342&sr=8-21
19	Drive Knob for Cart	1	Genuine Echo KNOB, FASTENER for driveshaft coupler assembly	Transmit force into the push/pull system.	Steel	4" x 1" x 2.5"	10.59	https://www.amazon.com/Echo-V299000160-Fastener-driveshaft-Assembly/dp/B07FDJSSFZ
20	Lead Screw	1	Metal Screw	Drives the bracket toward the rocket dome	Stainless Steel	2' x 1/2"	178.36	https://www.mcmaster.com/lead-screws
			Total Cost Estimate:				994.14	

Calculated Strain:

$E = \frac{\sigma}{\varepsilon}$ (Modulus of Elasticity Definition)

$M_W = \rho\, V$ (Mass of Water)

$m \ = \ M_B + M_W$ (Total Mass)

$F = m\, g$ (Gravitational Force)

$M_x = \ F\ L$ (Bending Moment on Beam)

$\sigma_b = \frac{M_x\, (\frac{d}{2})}{I_C}$ (Bending Stress on Beam)

$I_C = \frac{\pi}{32} d^4$ (Moment of Inertia)

$\varepsilon = \frac{(M_B + \rho\, V)\, g\, L}{\frac{\pi}{16}\, d^3\, E_i}$ (Final Calculated Strain Equation)

Measured Strain :

$(\varepsilon_1 - \ \varepsilon_2 + \varepsilon_4 - \varepsilon_3) \ = \ \frac{\delta\, E_O\, GF}{4\, E_i}$ (Strain Gauge Relation)

$\varepsilon \ = \ \frac{4(E_{02} - E_{01})}{GF\ E_i}$ (Final Measured Strain Equation)

Variables:

$\delta E_O \ = \ Change\ in\ Voltage\ Out$

$GF \ = \ Gauge\ Factor$

$\varepsilon \ = \ Measured\ Strain$

$M_B = \ Mass\ of\ Bucket$

$\rho \ = \ Water\ Density\ Room\ Temperature$

$V \ = \ Volume\ of\ Water$

$g \ = \ Gravitational\ Acceleration$

$\varepsilon \ = \ Strain$

$$L = Length$$
$$d = Beam\ Diameter$$
$$E_i = Input\ Voltage$$

Calculated Strain Uncertainty:

$$\frac{\delta\varepsilon}{\delta M_B} = \frac{16\,L\,g}{\pi\,d^3 E}$$ (Partial in Respect to Mass of Bucket)

$$\frac{\delta\varepsilon}{\delta V} = \frac{16\,L\,\rho\,V}{\pi\,d^3\,E}$$ (Partial in Respect to Water Volume)

$$\frac{\delta\varepsilon}{\delta L} = \frac{16\,g\,(M_B + \rho\,V)}{\pi\,d^3\,E}$$ (Partial in Respect to Length)

$$\frac{\delta\varepsilon}{\delta d} = \frac{-\,48\,L\,g\,(M_B + \rho\,V)}{\pi\,d^4\,E}$$ (Partial in Respect to Diameter)

Measured Strain Uncertainty:

$$\frac{\delta\varepsilon}{\delta E_{02}} = \frac{4}{GF\ E_i}$$ (Partial in Respect to Voltage Out)

$$\frac{\delta\varepsilon}{\delta E_{01}} = \frac{-4}{GF\ E_i}$$ (Partial in Respect to Initial Voltage Out)

$$\frac{\delta\varepsilon}{\delta GF} = \frac{-4(E_{02} - E_{01})}{GF^2\ E_i}$$ (Partial in Respect to Gauge Factor)

$$\frac{\delta\varepsilon}{\delta E_i} = \frac{-4(E_{02} - E_{01})}{GF\ E_i^{\,2}}$$ (Partial in Respect to Voltage In)

Calculated Strain Equation:

$$\varepsilon = \frac{(M_B + \rho\,V)\,g\,L}{\frac{\pi}{16}\,d^3\,E_i}$$