Universal Boom Offloader

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ME 4182 Capstone Design

Section A

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Executive Summary

Mechanisms designed to function in 0g environments require extensive testing prior to launch. During a deployment test, the mechanism is operated using an offload structure, which counteracts its weight to simulate the 0g conditions it would experience inflight. These offload structures are presently designed and built to accommodate each specific test article. The objective of this project is to develop a versatile, reconfigurable deployable boom assembly (DBA) offloader compatible with a variety of mechanisms. The system is to be scalable for a large range of force and motion, and able to interface with different types of booms. The main consideration in solving this problem is to minimize any impact of the offload system on the DBA performance, and minimize any externally induced loads and moments in the DBA. It is also necessary to measure these external effects to ensure that they are within the allowable range. The solution to this design problem is a X-Y Tension Gantry—a gantry system with 2 degrees of freedom (DOF) used to position a tension cable attached to the boom. Machine vision is used to monitor the angle of the tension cable. When it is sensed that there is a deviation from the vertical, the system adjusts the X and Y positioning to correct the offset. If finely tuned, this system can follow the natural path of the DBA with little effect on the DBA's behavior.

Nomenclature

COM	center of mass
DBA	deployable boom assembly
DOF	degrees of freedom
FEA	finite element analysis
UUT	unit under test

1. Introduction

An important aspect of the Earthbound testing of spacecraft is gravity compensation, as Earth's significant surface gravity impacts the performance of the UUT in ways in which it would not experience inflight. The deployment testing of such mechanisms involves their operation from an offload structure designed to simulate 0g conditions. Currently, offload structures are custom built for each mechanism, tailored to its specific size, mass, and deployment range. This project aims to design an adaptable and easily reconfigurable DBA offloader capable of accommodating a large range of force and motion, which can be used to test mechanisms of various characteristics. The offloader is required to be scalable for movements from 2 x 2 ft to 25 x 25 ft and DBAs weighing up to 1000 lb. Additionally, it should be able to support deployment rates of 0.2 to 20 deg/s and deployment ranges of up to 180 deg. The prototype to be developed over the course of this semester aims to support movements from 1 x 1 ft to 6 x 6 ft, weights up to 50 lb, deployment rates of 2 to 15 deg/s, and deployment ranges of up to 150 deg. One of the main challenges is minimizing any impact on DBA performance, and minimizing induced loads and moments in the boom. Other considerations include avoiding single points of failure, the ability to support different temporal deployment profiles, and a method to measure applied forces to the structure.

Sections 2 and 3 of this report detail relevant existing technology and codes and standards that must be followed, respectively. Section 4 introduces the customer requirements and engineering design specifications for this project, and section 5 presents the market research. Section 6 outlines the design concept ideation. Section 7 describes the design concept selection and detailed design, and section 8 describes the industrial design aspect of the project. Section 9 presents the engineering analysis and experiments performed, and section 10 details the final design and the constructed prototype. Section 11 characterizes the manufacturing considerations, and section 12 defines the social, environmental, and sustainability considerations. Section 13 comprises risk assessment and safety, and section 14 discusses the team's plans for patent claims. Finally, Section 15 summarizes the project and provides suggestions for future work.

2. Existing Technology and Prior Art

Gravity compensation is integral to the testing of space structures, as Earth's gravity obstructs the observation of spacecraft performance in several ways. Some examples of issues affected by gravity discrepant with the 0g operating environment of spacecraft include contact forces and friction between components, internal stresses, and the requirement of additional supports [1]. Because of this, a variety of methods have been developed to simulate 0g conditions for the earthbound testing of spacecraft.

2.1 Free Fall

One way to test spacecraft on Earth is via drop tests, where the specimen is elevated to a certain altitude, and then released into free fall. NASA employed this method to test a truss module [2] and a solar array structure [3] in 1981. The main drawback of this method is the difficulty in managing test article launch and catch. Additionally, the amount of time for which the test can be conducted is limited, and effects of air resistance must be considered. Because of these limitations, even the world's tallest vacuum drop test chamber at the NASA Glenn Research Center, which is capable of facilitating a 132 m free fall in 5.18 s, is generally not used to study mechanical phenomena [4].

2.2 Buoyancy in Air

A simple and relatively inexpensive method of gravity compensation involves the use of balloons to counteract an object's weight. The University of Surrey, UK, tested a deorbiting sailcraft boom with balloons in 2014 [5]. However, there are significant limitations to this method, namely spatial constraints and air drag. The spatial occupancy of the balloons can conflict with the spacing of specimen support locations, and must be able to be accommodated by the testing environment. The inertial effect of the air being forced to flow around the balloons impedes their motion, thus limiting this application to test situations where the specimen only moves slowly [1].

2.3 Buoyancy in Water

The buoyant effect of submersion in water can also be used in simulating weightlessness. In 2004, L'Garde, Inc. performed a deployment verification test of a solar sail in a water trough [6]. Similar to the use of air buoyancy, water buoyancy testing is subject to significant drag effects, and therefore can only be used for slow-moving bodies. Additionally, the buoyant stiffness in the vertical direction and wave dynamics in response to specimen motion could affect test performance [1].

2.4 Mechanical Systems

The majority of existing gravity offloading methods fall under the category of mechanical systems, which can offer diverse solutions to the problems presented in spacecraft testing. The simplest mechanical method of gravity offloading is the use of counterweights with pulleys or flybeams to exert lift on the test article, as demonstrated in Patent No. TW200607746A [7]. However, pulley systems include friction, and flybeams limit the range of motion.

Another common mechanical approach to gravity compensation is suspension of the specimen from fixed locations, constraining the vertical position without suppressing lateral motion. This includes stiff suspension, as used by Tennessee State University in the testing of an inflatable/rigidizable hexapod structure in 2002 [8], and compliant suspension with soft springs or elastic cables, as was used for the testing of the PowerSail deployable solar array mechanism in 2003 [9]. Patent No. CN103482089A utilizes a controlled constant tension vertical suspension cable in conjunction with a horizontal follow-up module to carry out horizontal movements and maintain the tension cable in the vertical orientation [10].

Additionally, the specimen can be supported from underneath—such as with sliding supports, air tables, or air bearings—in order to approximate a frictionless surface on which it can slide [1].

3. Codes and Standards

There are a variety of codes and standards that must be followed for this design. All drawings produced must be in accordance with the ASME Y14.100, which sets standards for engineering drawing practices. The relationship between Ball Aerospace and Georgia Tech is subject to the International Traffic in Arms Regulations (ITAR), which restricts and controls the sharing of defense- and military-related technologies in the interest of national security. This requires all members of the team to be U.S. citizens. Though it is not particularly relevant in the early stages of design and prototyping, which constitute the scope of this project, the final product must comply with AS9100, the international quality management system for aviation, space, and defense (ASD). This standard details the safety and reliability requirements for products supplied to the ASD industry. Additionally, the AFSPCMAN 91-710 must be held in consideration. The AFSPCMAN 91-710 establishes safety requirements for airborne vehicles launched from the Air Force Space Command ranges.

4. Customer Requirements and Engineering Design Specifications

4.1 Stakeholder Analysis:

The testing of a deployable boom requires collaboration between various parties within a company, and these parties must be kept in mind when designing a testing apparatus such as an offloader. In order to better understand the needs and desires of theses parties, a stakeholder analysis was performed and the Stakeholder Matrix shown in Table 1 was generated.

Table 1: *Stakeholder Matrix*

Stakeholder	Interests	Impact/Effect	Importance
Ball Test Analysts	Interested in data on impacts of offloader on test results	Requires some method of feedback measurement	Medium
Ball Test Engineers	System must properly offload boom through full range of motion	Drives overall design	High
Ball Facilities	System must be easy to assemble, disassemble and move	Affects overall structure design	Medium
Ball Financial	System must be cost effective	Affects material/component choice	Low
Ball Technicians	System operation should be easy to understand and not prone to user error	Affects user interface design	Medium
Ball Health/Safety	System must meet all required safety codes	Affects overall design	High
Manufacturers	Design must allow for easy manufacturing and assembly	Constrains mechanism design	Low
Material Suppliers	Materials must be easily available and cost effective	Constrains material selection	Low
Government	System must be designed, manufactured and operated following ITAR regulations	Constrains design/manufacturing resources	High
Spacecraft Customers	System must ensure deployment reliability	Drives overall design	High

4.2 User Needs, Functions and Constraints

Based on the various stakeholders identified in the stakeholder analysis, the design team generated general product requirements likely associated with each group's interests. These general requirements are useful for associating product function with the specific engineering requirements supplied by Ball Aerospace. The customer needs are shown in Table 2.

Table 2: *Customer Needs*

Customer Needs	Stakeholder
Precise Offloading	Ball Test Analysts
Scalable for different boom types	Ball Test Engineers
Scalable for different boom sizes	Ball Test Engineers
Easy to Operate	Ball Technicians
Easily cleaned/maintained	Ball Technicians
Safe to Use	Ball Health/Safety
Reliable Data Collected	Ball Test Analysts
Cost Effective	Ball Financial

Additionally, constraints were identified based on the manufacturing, material, and mechanical requirements provided by Ball Aerospace. These constraints, listed in Table 3, will directly drive the design choices made by the team.

Table 3: *Constraints*

Category	Constraint	Comments
Manufacturing	TVAC compatible	Cannot use manufacturing processes that cannot handle thermal vacuum, such as stick welding.
Material	No off-gassing	Cannot use any materials that off-gas in vacuum
	No particulating	Cannot use any materials that particulate
Mechanical	Handle boom weight	Offloader must withstand weight of boom and payload
	Handle deployment speeds	Offloader must operate nominally up to boom deployment speeds

4.3 Specification Sheet

Based on the customer needs for the offloader design, a list of design requirements was provided by Ball Aerospace. With this information, a Specification Sheet was developed, as shown in Table 4. Requirements ranged from fitting sized dimensions to handling UUT behaviors. The offloader must also be safe to use in a clean room, be designed with English units and hardware, and provide a universal system ground. Ball Aerospace also provided desired elements to the offloader, which focused on efficacy and repeatability of the offloader. The offloader should not significantly affect the deployment kinematics and behavior of the UUT, requiring adequate instrumentation in the system.

Table 4: *Specification Sheet*

		Ball Aerospace DBA Offloader Specification Sheet	Issued: 9/17/19
No.	**Changes (date)**	**Requirements**	**Source**
1	9/9/19	Static Envelope (fit through 15' x 20' door)	Requirements Slides provided by Ball
2	9/9/19	Dynamic Envelope (2' x 2' scalable to 25' x25')	
3	9/9/19	Load (handle DBAs 50 lbf scalable to 1000 lbf)	
4	9/9/19	Deployment Speed (handle .2 - 20 deg/sec scalable to 2 - 15 deg/sec)	
5	9/9/19	Deployment Ranges (150 deg scalable to 180 deg)	
6	9/9/19	No use of hydraulic systems	
7	9/9/19	No use of particulating materials	
8	9/9/19	System shall be clean-room compatible (cleanable with IPA wipes, non-volatile, etc.)	
9	9/9/19	Design shall be in English units and hardware	
10	9/9/19	Universal System Ground	
11	9/9/19	If a powered control system is used, fail-safes to protect UUT are necessary	
No.	**Changes (date)**	**Desires**	**Source**
12	9/9/19	TVAC - compatible (no welds, off-gassing materials, etc.)	Goals Slides provided by Ball
13	9/9/19	Minimize impact on DBA behavior (deployment speed, sag, stiffness, etc.)	
14	9/9/19	Minimize loading on DBA (forces and moments)	
15	9/9/19	Modularity (ability to easily modify our assembly to match up with any future boom designs)	
16	9/9/19	Limit single points of failure	
17	9/9/19	Support different deployment profiles (position driven, speed driven, exponential profiles, etc.)	
18	9/9/19	Instrumentation (to measure exact loads applied to UUT by offloader, can be based off camera vision (angles), etc.)	
19	9/17/19	Easy to operate by trained technician	
20	9/17/19	Easily cleaned and maintained	
21	9/17/19	Safe to use	
22	9/17/19	Cost effective	

4.4 House of Quality

Using the customer needs as a starting point, a House of Quality, shown in Figure 1, was created to determine how engineering requirements and their target values will affect those needs. The House of Quality is then used to help rank and justify the design choices. Each of the three concepts is ranked (1-3) on how well they fulfill each need. The design with the lowest sum of rankings best satisfies the customer need, which is considered during the design selection.

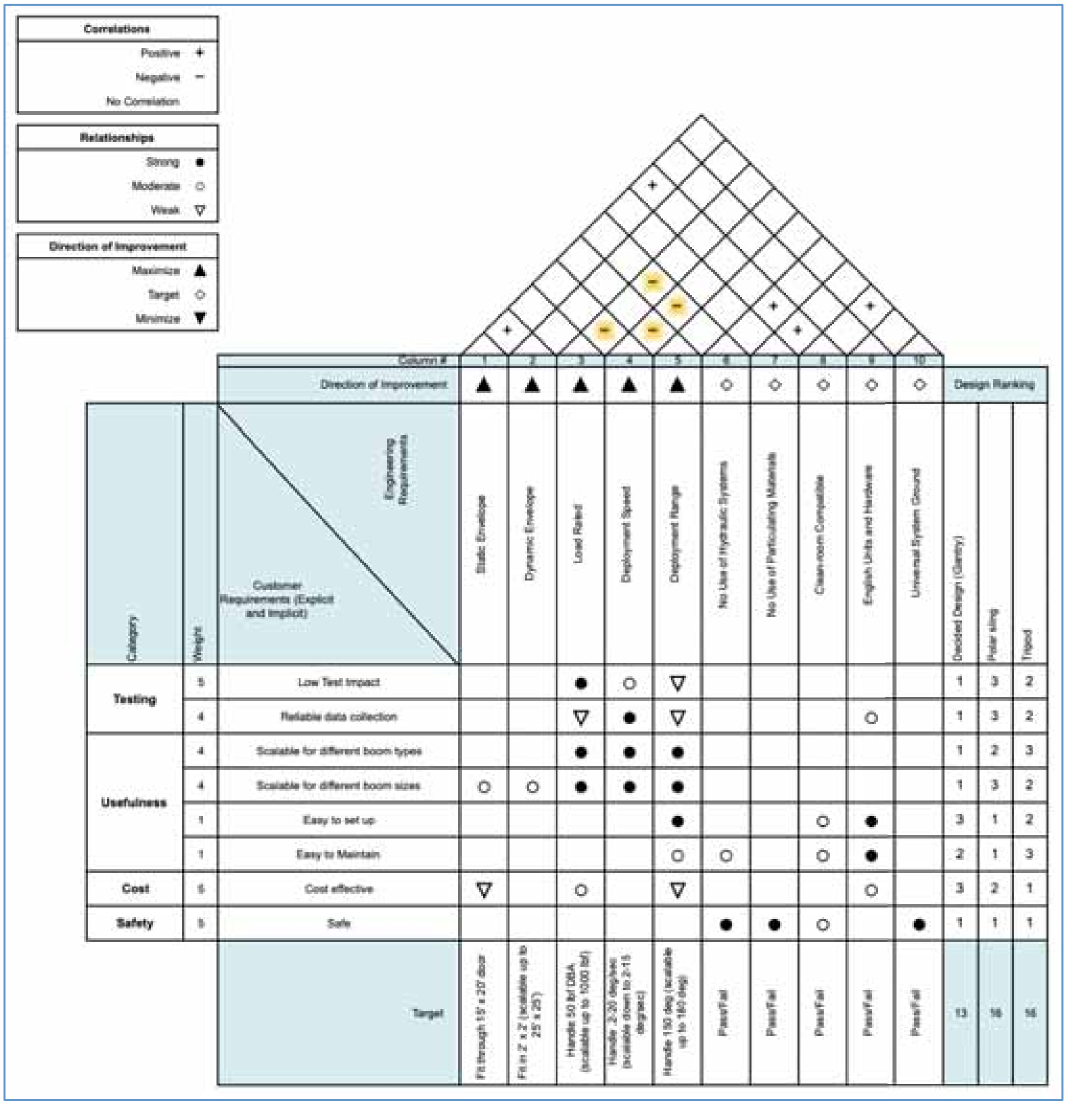

Figure 1: *House of Quality*

5. Market Research

Based on the research of the market, aerospace companies such as Ball, NASA, Northrop-Grumman, Lockheed Martin, and L3Harris make one-use offloaders for each boom size and deployment profile. These companies spend large sums of money to construct and qualify the numerous offloaders. A modular offloader would have cost- and time-saving implications for all of these companies. It is concluded that aerospace companies would be willing to spend upwards of tens of thousands of dollars for a qualified universal offloader.

6. Design Concept Ideation

6.1 Concept Generation

The Universal Boom Offloading Structure serves three main purposes: to accept the Boom, to counteract the weight of the boom, and to minimize the induced interference on the boom's inherent behavior. To further characterize these main functions, sub-functions were determined, as shown in the function tree of Table 5. The challenge of this project comes from the universal aspect of this testing device, which requires extra thought to produce solutions to these sub-functions.

Table 5: *Function Tree*

<table>
<tr><td rowspan="7">Universal Boom Offloading Structure</td><td rowspan="2">Accept Boom</td><td>Interface with Different Boom Types</td></tr>
<tr><td>Interface with Different Boom Sizes</td></tr>
<tr><td rowspan="3">Counteract Self Weight</td><td>Support Across Deployment Path</td></tr>
<tr><td>Point Load</td></tr>
<tr><td>Distributed Load</td></tr>
<tr><td rowspan="2">Minimize Induced Interference</td><td>Maintain Net Vertical Force</td></tr>
<tr><td>Measure Behavior</td></tr>
</table>

Figure 2, the morphological matrix, depicts several creative solutions for the lowest level of the Function Tree. These sub-function solutions were accumulated through the market research, existing technology, prior art searches, as well as other unconventional solutions devised by team members. Many of the solutions may be used in conjunction with others of the same category, while others may be used only in combinations with solutions from other sub functions. The Counteract Self Weight function breaks down into sub functions of supporting across the deployment path by means of point loads or distributed loads. These three sub functions have the largest variety of solutions, which were compiled through research of past works (e.g. Free Fall, Fly Beam Hierarchy, and Balloons) and brainstorming for novel solutions (e.g. Magnets and Air Bag).

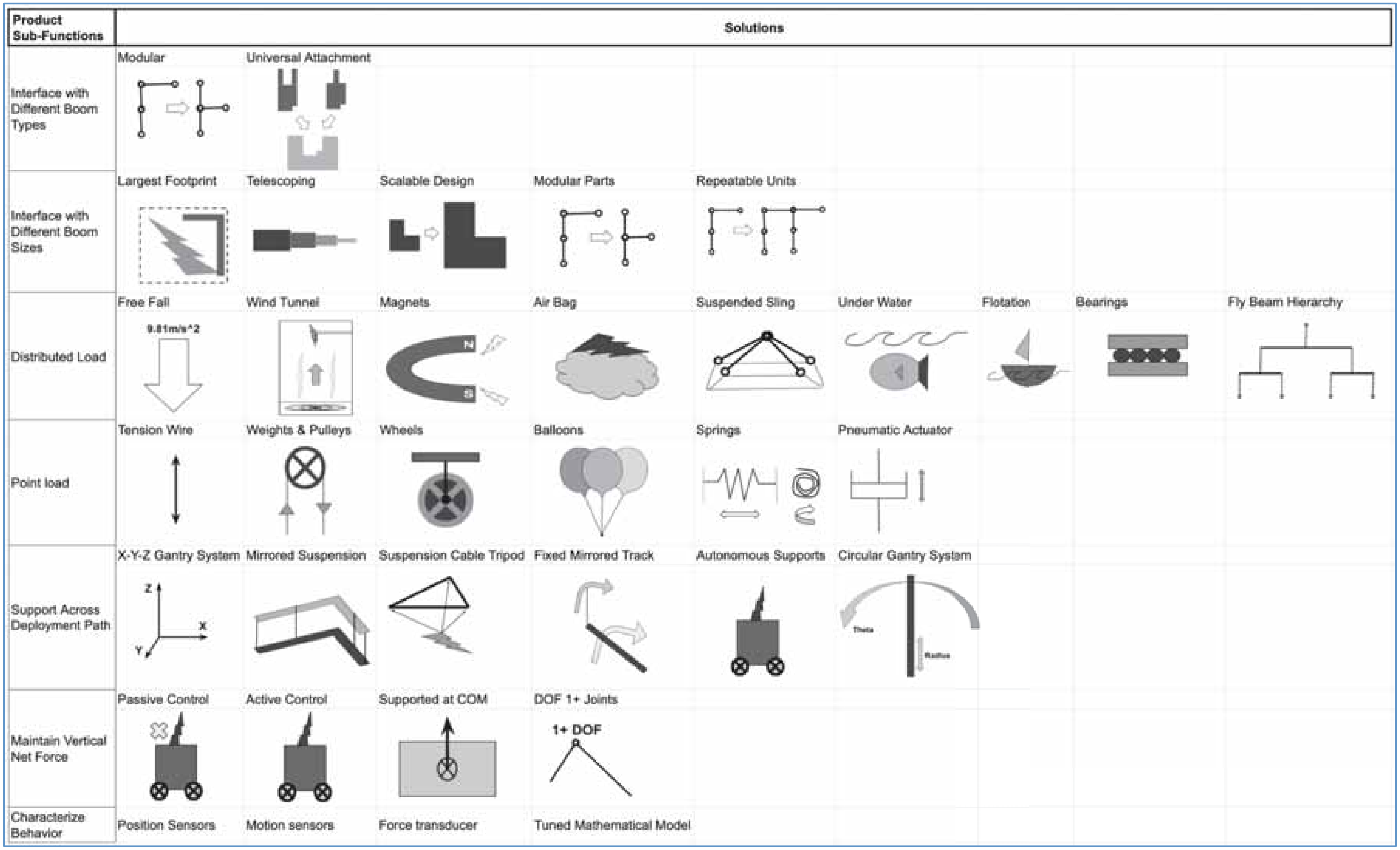

Figure 2: *Morphological Matrix*

6.2 Preliminary Design Concepts

The Morphological Matrix provides an opportunity for the team to take inspiration from novel solutions to individual functions, and develop comprehensive design ideas. The first design, the Tripod Hierarchy, utilizes an actively controlled tripod cable system to guide the path of the support wires, which would attach to a flybeam hierarchy over the COM of the boom, as depicted in Figure 3. The motors of each tensioned cable would follow the projected path of the boom, and the largest common footprint for booms would be built. A second design, the Polar Sling Gantry, is shown in Figure 4. This design utilizes the passive pull of the boom to move the bearings on the polar coordinate gantry system attached to a sling supporting the boom under its COM. This concept would be scaled to match the load and range requirements of each individual boom. The final design, the X-Y Tension Gantry, uses a machine vision correction X-Y crane-style gantry system to minimize the offset on the gantry position and the end of the tensioned cable attached to the boom by a connection with DOF>1. This is shown in Figure 5. The X-Y Tension Gantry design also uses the largest common footprint and would incorporate a load cell in the cable so the tension could be tuned with a turnbuckle.

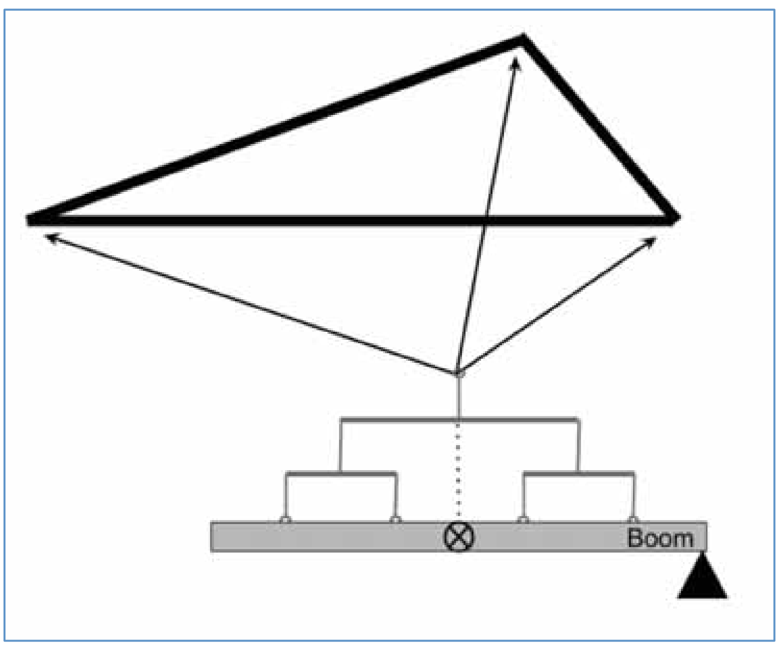

Figure 3: *Tripod Hierarchy*

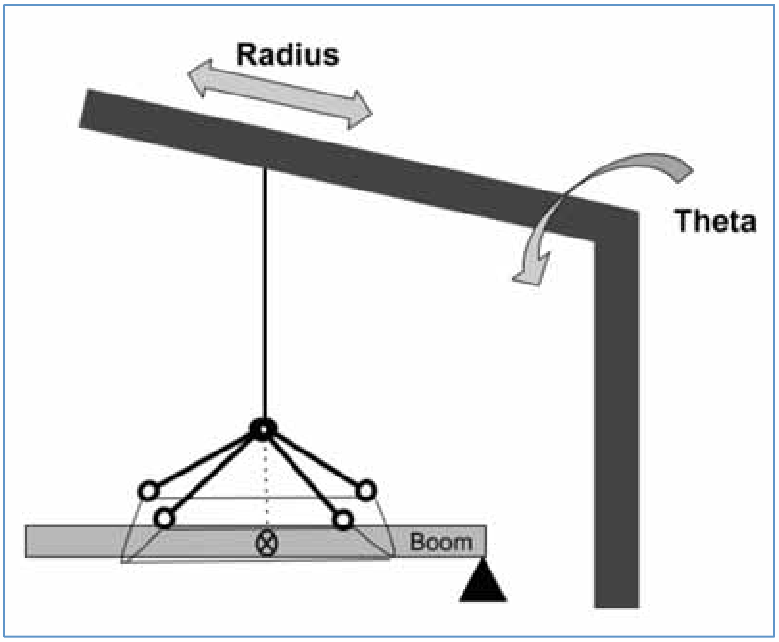

Figure 4: *Polar Sling Gantry*

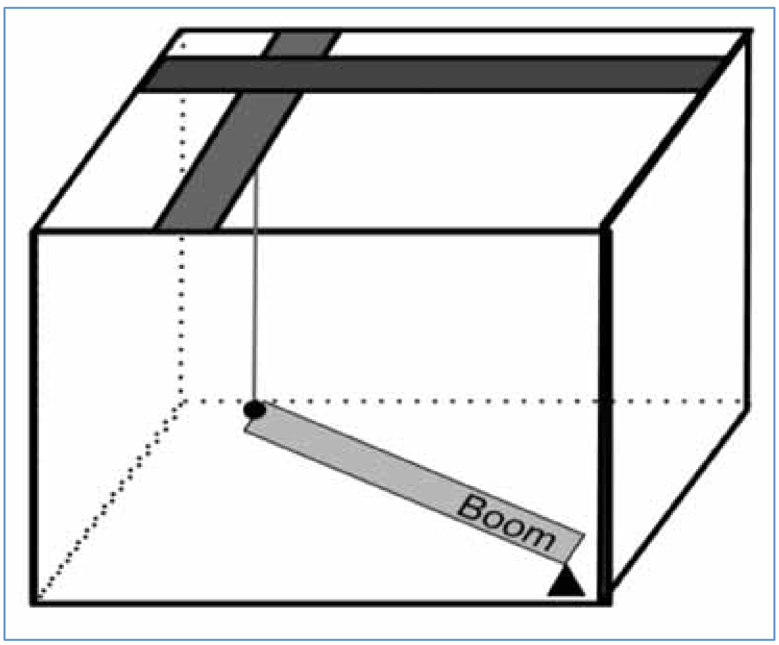

Figure 5: *X-Y Tension Gantry*

7. Concept Selection and Justification

To be able to choose a final design concept, further scrutiny is needed of how well each design fulfills the user needs and executes the desired functions. This section analyzes the advantages and disadvantages of the three design concepts in order to select the optimal one with which to move forward, considers the potential risks characteristic of the selected design, and describes the selected design in detail.

7.1 Concept Evaluation

In this section, the three preliminary design concepts are evaluated. Benefits and drawbacks of each design are considered and analyzed with respect to the project goals.

7.1.1 Tripod Hierarchy

In the Tripod Hierarchy design, the path of the connection point would need to be pre-programed to adjust the cables lengths appropriately as the boom spans an arch or other specified path. Actively driving the position of the support that is counteracting the UUT weight ideally has the support in the correct position, but it runs the risk of driving the deployment of the boom profile as well. Instead of only offsetting gravity, the active driving control may impact the boom by adding axial and transverse loads that inhibit the boom's natural response.

7.1.2 Polar Sling Gantry

The Polar Sling Gantry faces a different problem of passive radial and theta positioning. Even if low friction bearings such as precision bearings or air bearings minimize forces opposing the intended motion, inertial effects of the radial arm will overwhelm the inherent motion of the boom. A rule of thumb provided by Ball Aerospace Engineering staff is that the inertial impacts of the support structure should be less than 5% of the boom.

7.1.3 X-Y Tension Gantry

The X-Y Tension Gantry would have a machine vision system that would sense the angle of the tension cable and its deviation from vertical, indicating that the position of the gantry support is not directly overhead of the boom attachment. The machine vision system would then interpret the angle of offset to drive the respective X and Y belt motors to correct the position in real time. This control scheme follows the natural path of the boom with little lag, instead of dictating where the support needs to be. This will minimize the induced transverse and axial reactions of the boom other than the required vertical support within acceptable levels.

7.2 Design Selection

Based on the above analysis, the team has decided to move forward with the X-Y Tension Gantry design. Actively controlled systems will likely influence the UUT deployment behavior, detracting from the usefulness of tests. As a system dictates a coordinate location, such as in the Tripod Hierarchy setup, the UUT will have no choice but to move to that location. Similarly, a completely passive system will introduce dead weight that the UUT drags along during boom deployment. Acceleration of the boom will be directly affected by the ratio of the mass of the moving offloader segment to that of the test article. Therefore, an active response system is necessary. The X-Y Tension Gantry system incorporates sensors to detect small angle changes in the cable and corrects the offloader head so that the tensioned cable remains as vertical as possible. If tuned, the feedback control system will avoid both passive inertial effects as well as any active driving effects. Quantitative analysis conducted thus far on this design confirm its feasibility and effectiveness. This is documented in detail in Section 9.

7.3 Potential Risks and Countermeasures

At this point in the design process, several potential risks are foreseen. Firstly, the boom could be damaged if there is weight attached but it is not sufficiently offloaded. Two potential solutions to this exist. Either a stand will be designed to support the end of the boom when it is not connected, or an extra backup strap will be used to secure it to the frame. Additionally, precautions will be taken in case the system malfunctions. Both a hardware stop and a software stop will be incorporated for emergency shutdown. There will be a physical kill switch to cut off voltage to the motors if the need arises to abort the operation of the machinery as quickly as possible. Additionally, if the encoders or force transducer read numbers outside the expected range, the system will be programmed to shut itself down in order to prevent continuous overcorrection in the positioning.

7.4 Detailed Design and CAD

Figure 6 shows a comprehensive view of the X-Y Tension Gantry CAD model. The frame and cross bar are made from 80/20 beams, and the position of the cart is adjusted with a motor-driven belt and pulley system. Cameras record the angle of the tension cable with respect to the vertical, and the appropriate positioning corrections are made to follow the motion of the boom.

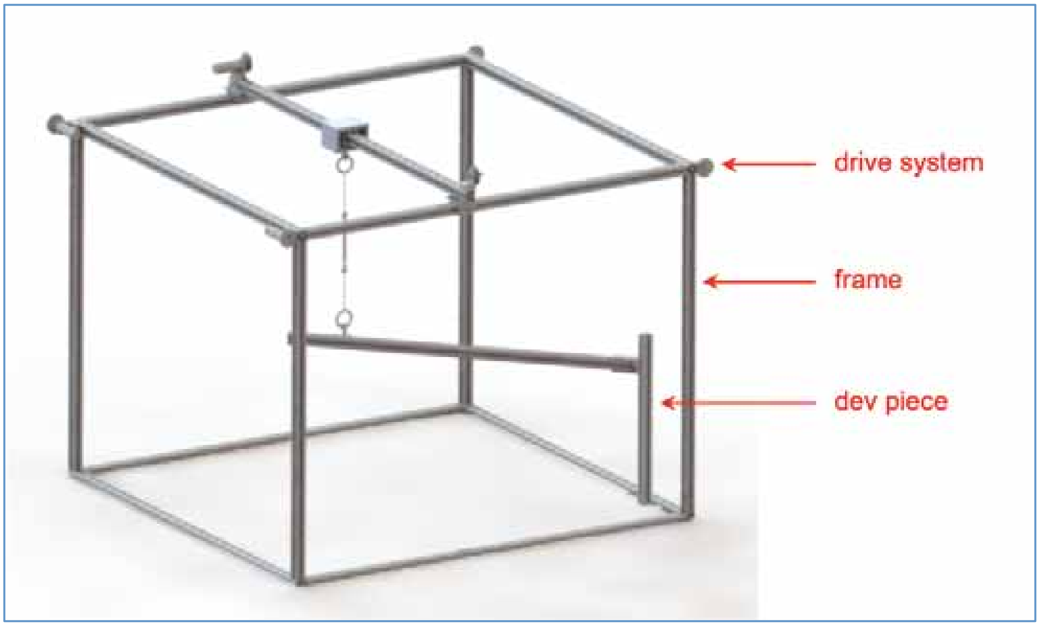

Figure 6: *CAD Model*

Figures 7a and 7b show the pulley and hub arrangement in detail for the driven pulleys and free pulleys, respectively. The hubs are designed to interface with both the motor shafts of the driven pulleys and the free spinning axles of the free pulleys. Additionally, a constant spacing between the mounts and the 80/20 beams is required to maintain belt alignment.

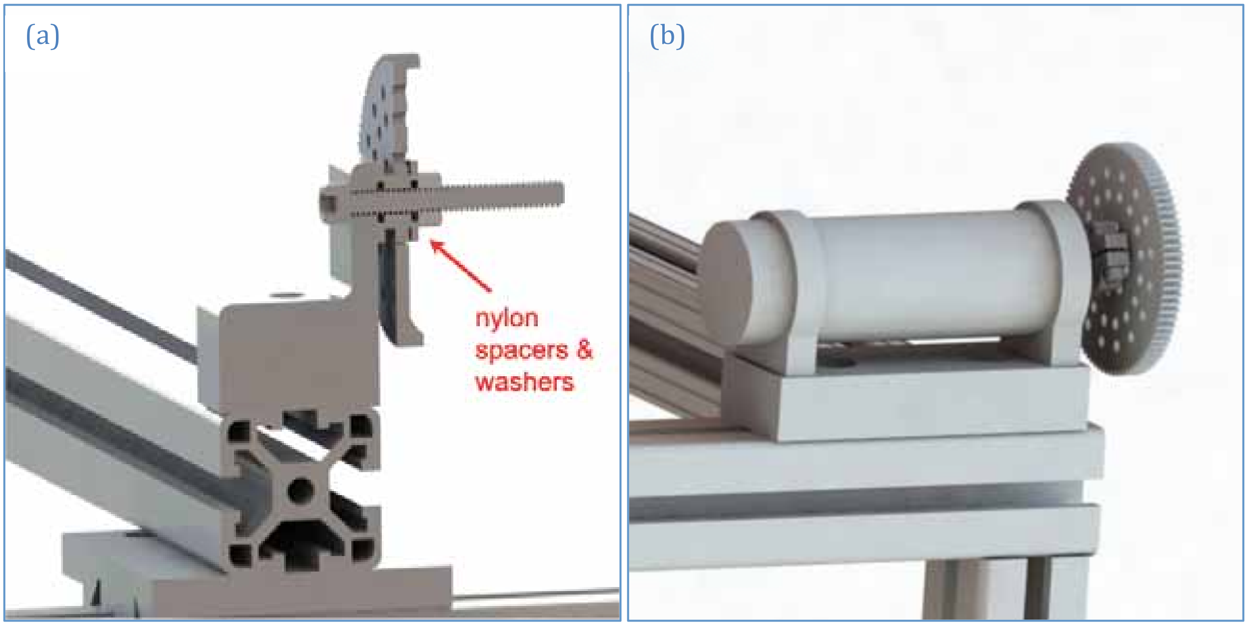

Figure 7: *Pulley and Hub*

A detailed view of the cart and mechatronics housing can be seen in Figure 8. The cart is required to house the cameras and the Raspberry Pi computer. One key consideration in the design of the cart and housing is the routing of the cables, which cannot impede the motion of the components.

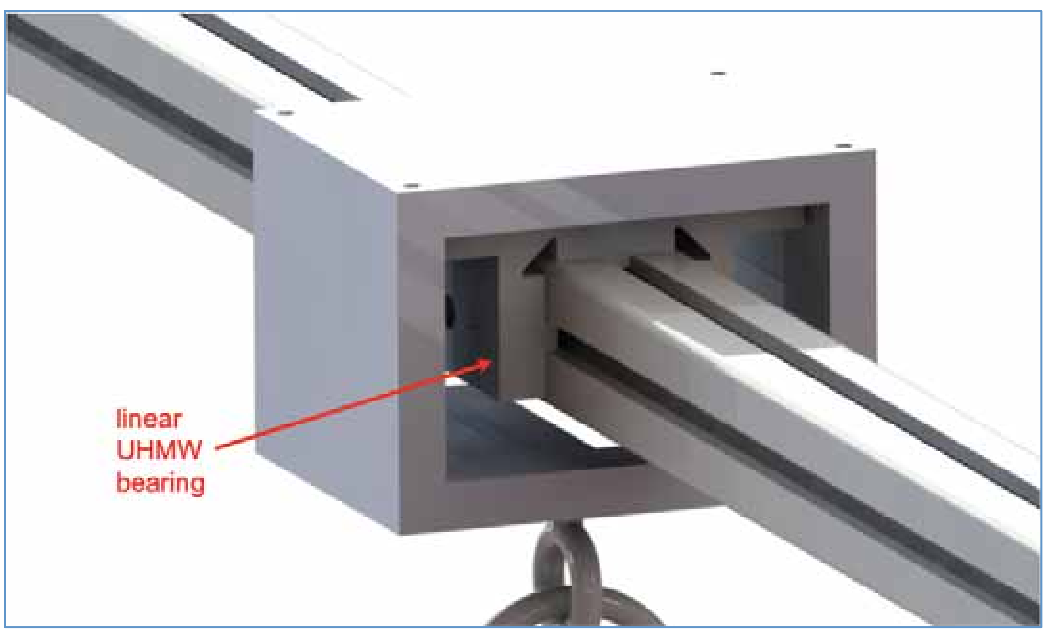

Figure 8: *Cart and Housing*

Figure 9 details the cable connecting the cart to the boom. This includes a turnbuckle to adjust the tension in the cable, and a load cell to measure the force. The attachment at the end of the cable slots into the 80/20 beam comprising the boom.

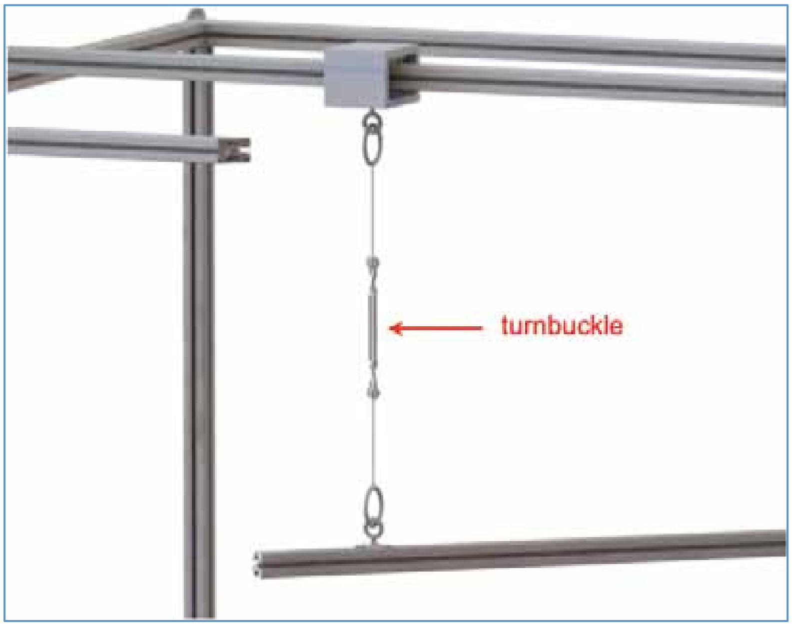

Figure 9: *Connection to Boom*

8. Industrial Design

8.1 Physical, Cognitive, & Organizational Considerations

This project is a highly specialized piece of testing equipment for satellites and space structures. While industrial design considerations of physical, cognitive, and organizational needs are taken into account, they are not of paramount importance to the success of the project.

Physical considerations for our prototype include sizing of the overall footprint. For an average-sized human to test this product, the relevant systems need to be easily accessible, limiting the maximum height of the system to approximately six feet. A height of four feet was decided upon to keep the mechanisms within reach and limit ergonomic aggravations for the engineers. The 80/20 will have plastic end caps to prevent harm to any person assembling, adjusting, or interacting with the prototype. Pinch point analysis will also be completed to add protective coverings to various gears and components that risk jamming potential with a hand or other objects.

There are few cognitive considerations, as the functional requirements are very explicit and the system will only be used by engineers or trained technicians. The operations manual will outline the systematic procedure and necessary steps to the user, using straightforward language for instructions and detailed explanations. Any necessary data will be exported from the controls system to be further analyzed by the engineers. Visual hierarchy considerations would act contrary to a simplified and streamlined design in terms of components locations, so it was not pursued.

Organizational needs are outside the scope of our project, but a suggestion to the customer is that one engineer or technician be the lead resource on the product to direct testing for new satellite or boom systems. The person in the lead role will be responsible for training others with the manual, ensuring the safe operation of the product, and communicating to leadership if any problems arise.

8.2 Visual Considerations

There are multiple software and mechanical failsafes to prevent any malfunctions that would damage the testing or tested hardware. There will also be a large red e-stop button in case the mechanical or software failure is not picked up by the control system in order to prevent further issues. The majority of the machine is industrial silver-metal finished color, inline with similar test fixtures. The large red e-stop button will have great contrast to the silver finish, making it easy to spot in an emergency.

Branding is superficially considered, as the team plans to have a metal plate engraved with approved logos of Ball Aerospace and Georgia Tech, as well as the Boom Brigade team logo for the Capstone Expo competition. To ensure an effective presentation of the prototype, wires will be covered and managed, and 3D-printed casings will be used to house the electronic systems. The metal framing will have plastic end caps to provide an effective finished appearance.

Extraneous visual effects would inhibit the desired function of the prototype and be contrary to the expectations of the customer and thus were not pursued further.

9. Engineering Analysis

9.1 Frame Structural Analysis: Stress, Deflection, and Buckling

To verify the functional feasibility of the design, a structural analysis of the frame was conducted. FEA simulations in ANSYS software were used to help optimize the cross-sectional area of the beams. The material used is Aluminum 6360-T6, and the maximum load to be supported is 50 lb. A multiplier of 1.5 was used to account for the weight of the boom itself, mechatronics, and other various mechanical components. 2 different test cases were considered: (1) a point load of 75 lb in the center of the top plane of the frame, and (2) a point load of 75 lb at the center of one of the sides. These locations are illustrated in Figure 10. For each test case, trials were run with 1 in, 1.5 in, and 3 in cross sections of 80/20. The results for test cases (1) and (2) are shown in Figures 11 and 12, respectively.

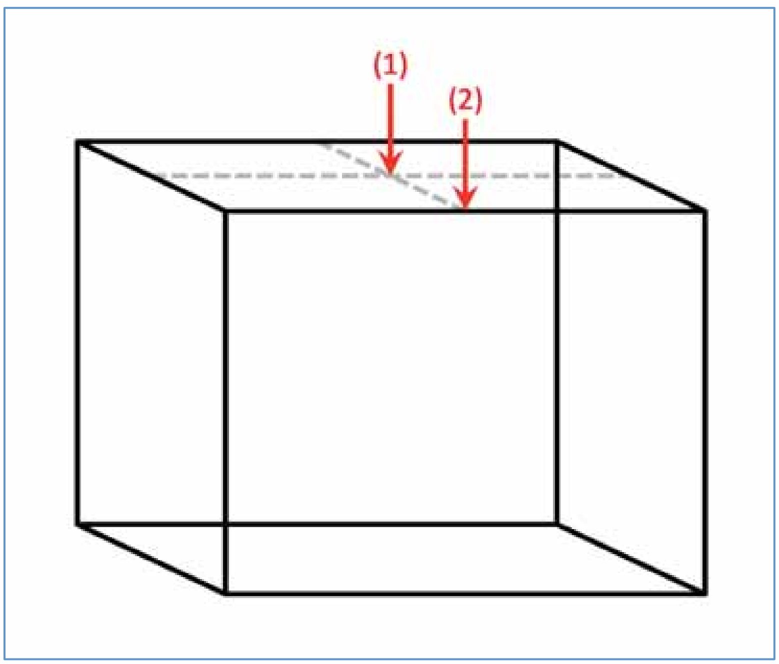

Figure 10: *Frame Structural Analysis Load Application*

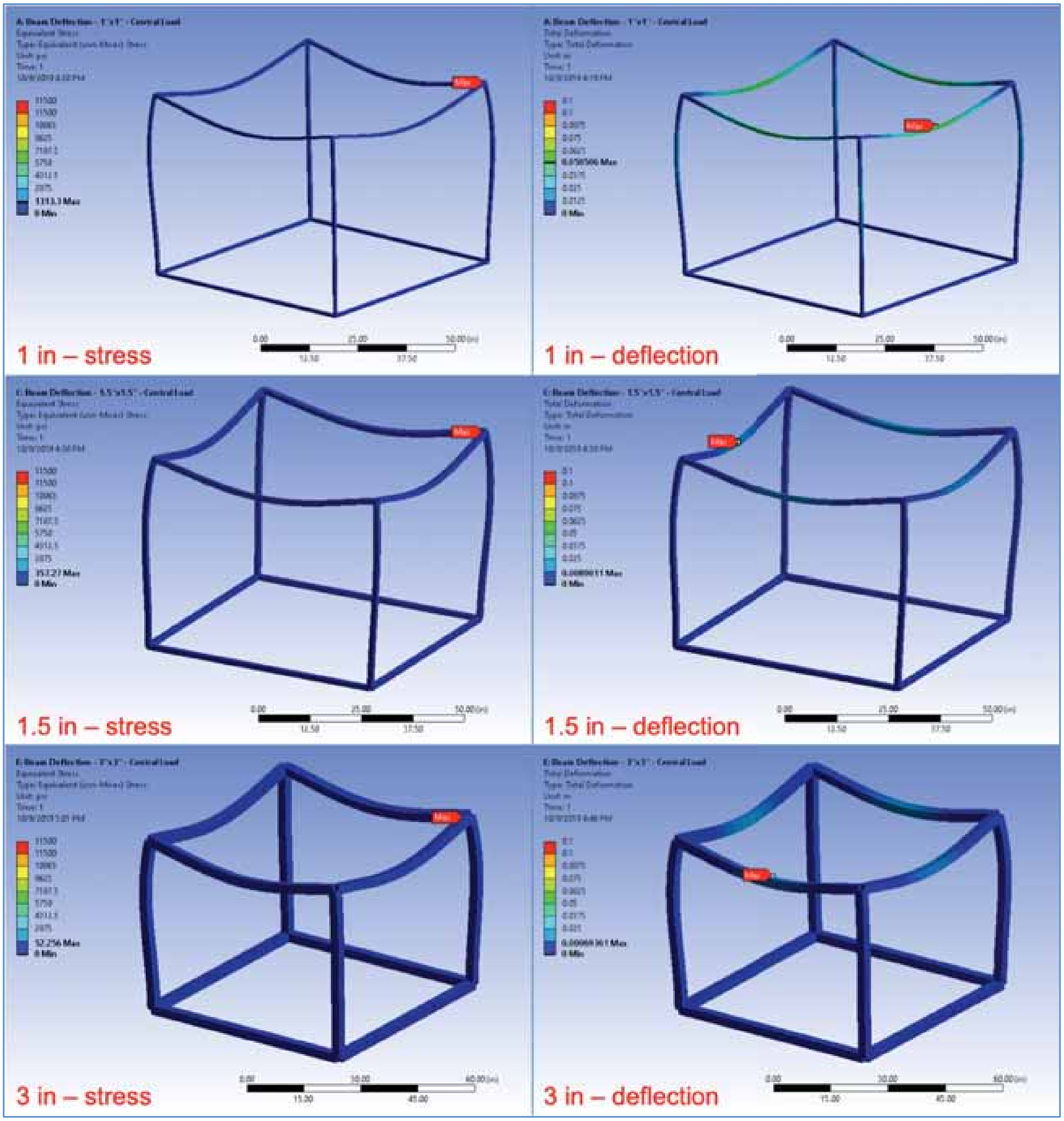

Figure 11: *Frame Test Case (1) ANSYS Results*

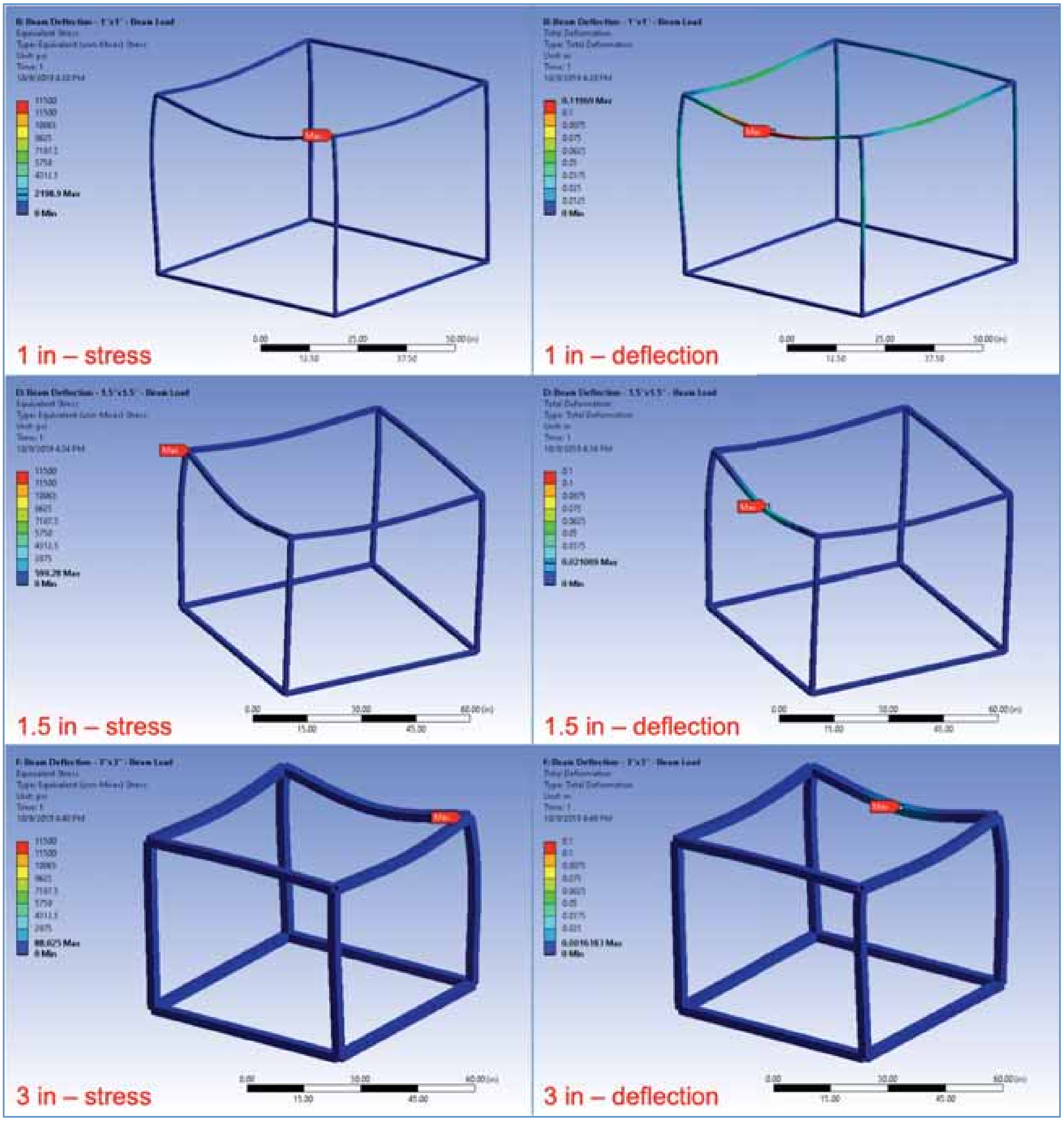

Figure 12: *Frame Test Case (2) ANSYS Results*

Hand calculations for a simplified model were performed to verify the FEA results. For these calculations, only one top beam was analyzed. The beam was considered to be fixed at both ends, with a 37.5 lb point load in the center for test case (1) and a 75 lb point load for test case (2).

The stress in the beam is given by

$$\sigma = \frac{yM}{I} \tag{1}$$

where y is the distance from the neutral axis, M is the moment, and I is the area moment of inertia. Here,

$$M = \frac{FL}{8} \tag{2}$$

defines the moment, where F is the force, and L is the length of the beam [11]. The area moments of inertia for the 80/20 beams used in the calculations were obtained from [12].

Table 6 compares the maximum stress results from the ANSYS model and the hand calculations. As expected, the corresponding values are of the same order of magnitude, but the hand calculations estimate greater stresses than the ANSYS model, as they do not account for the distribution of the load among the other members. Table 7 lists the margins with a safety factor of 2, as requested by Ball Aerospace. Both the ultimate strength of 30000 psi and the yield strength of 23000 psi are considered. This shows that all the stresses are within the allowable range.

Table 6: *Frame Stress Analysis*

	ANSYS model		hand calculations	
beam size	37.5 lb load stress (psi)	75 lb load stress (psi)	37.5 lb load stress (psi)	75 lb load stress (psi)
1010	**1313**	**2199**	**3181**	**6362**
1515	**357**	**599**	**830**	**1660**
3030	**52**	**88**	**123**	**246**

Table 7: *Frame Stress Analysis Margins*

		ANSYS model		hand calculations	
beam size	stress type	37.5 lb load stress margin	75 lb load stress margin	37.5 lb load stress margin	75 lb load stress margin
1010	yield	**7.76**	**4.23**	**2.62**	**0.81**
	ultimate	**10.42**	**5.82**	**3.72**	**1.36**
1515	yield	**31.19**	**18.19**	**12.86**	**5.93**
	ultimate	**40.99**	**24.03**	**17.07**	**8.04**
3030	yield	**219.07**	**129.64**	**92.50**	**45.75**
	ultimate	**286.05**	**169.41**	**120.95**	**59.98**

The maximum deflections in the beams is given by

$$\delta = \frac{FL^3}{192EI} \tag{3}$$

where E is the modulus of elasticity [11]. The deflection results from the ANSYS model and the hand calculations are shown in Table 8, and the margins are shown in Table 9. Similarly, the hand calculations are more conservative than the simulation results, but of the same order of magnitude. Deflections for the 1 in beam are greater than the allowable deflection of 0.1 in. Hence, 1.5 in beams were chosen for our design.

Table 8: *Frame Deflection Analysis*

	ANSYS model		hand calculations	
beam size	37.5 lb load deflection (in)	75 lb load deflection (in)	37.5 lb load deflection (in)	75 lb load deflection (in)
1010	**0.0505**	**0.1197**	**0.2498**	**0.4838**
1515	**0.0089**	**0.0210**	**0.0480**	**0.0886**
3030	**0.0007**	**0.0016**	**0.0047**	**0.0077**

Table 9: *Frame Deflection Analysis Margins*

	ANSYS model		hand calculations	
beam size	37.5 lb load deflection margin	75 lb load deflection margin	37.5 lb load deflection margin	75 lb load deflection margin
1010	0.98	-0.16	-0.60	-0.79
1515	10.23	3.76	1.08	0.13
3030	143.17	60.79	20.28	11.99

Eigenvalue buckling analysis was then performed for the 1.5 in beam with test case (2), which had the highest stresses. The results are shown in Figure 13, and the load factors are listed in Table 10. A load factor of 248 is required to achieve the first mode of buckling, meaning the frame can withstand a load of up to 248 times our maximum load before buckling will occur. Additionally, the nominal stress at this mode is 13623 psi, which is slightly larger than the allowable stress, but not higher than the actual yield and ultimate strength of the frame material. This indicates that it is highly unlikely for the frame to fail due to buckling.

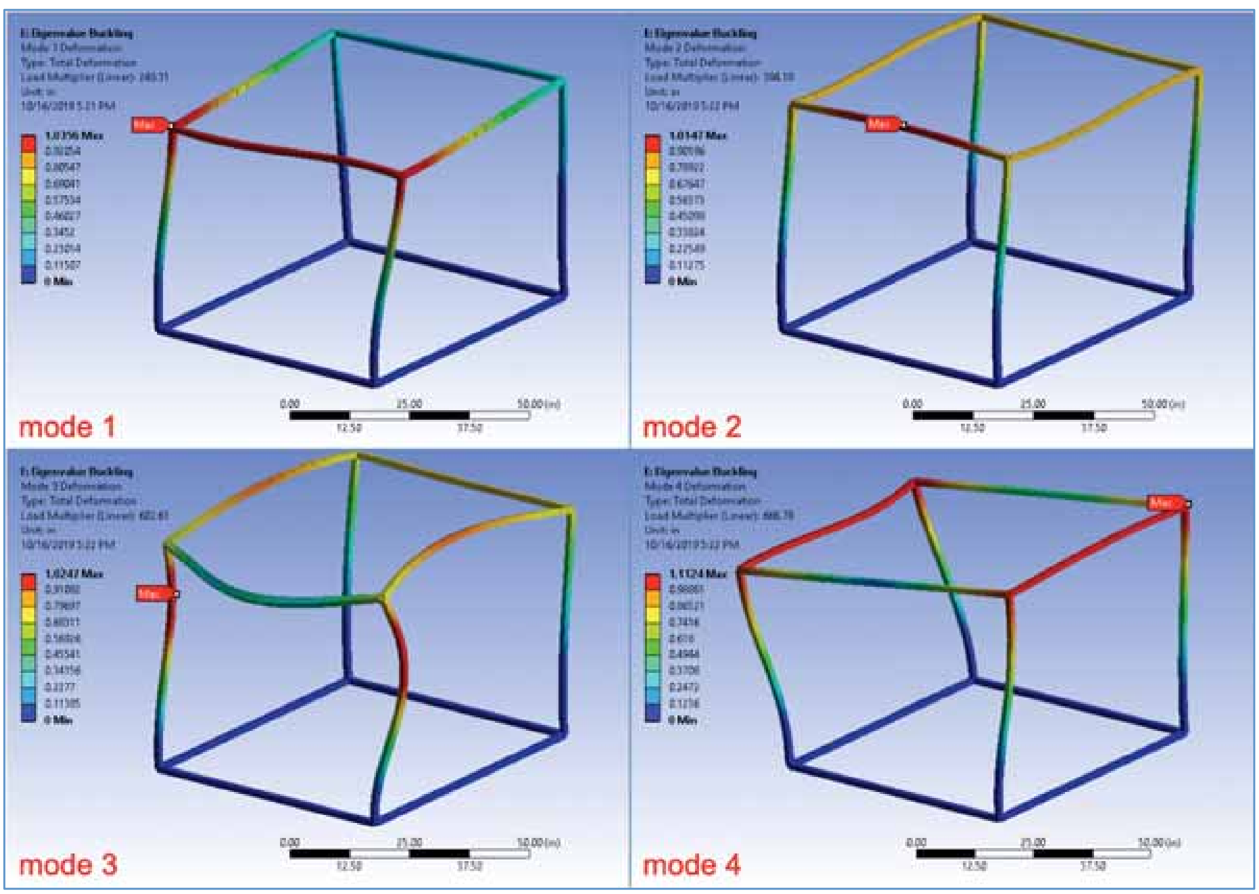

Figure 13: *Frame Buckling ANSYS Results*

Table 10: *Frame Buckling Load Factors*

Mode	Load Factor
1	248.31
2	348.18
3	602.61
4	666.78

9.2 Cart Structural Analysis: Stress

Structural analysis was conducted with the cart to determine the optimal thickness for the bottom plate. The material is Aluminum 6061-T6 with a yield strength of 40000 psi and ultimate strength of 45000 psi. For the actual prototype, the load on the plate will be distributed among four 3/16 in bolts. In the ANSYS simulations, this was approximated as a uniform load across the plate to provide a less conservative estimate of the structural effects. Both a 50 lb load and a 1000 lb load were considered. 50 lb is the maximum load for the prototype, and 1000 lb is the load the design is to be scaled up to. The simulation was run with plate thicknesses of 0.25 in, 0.5 in, and 1 in. The results are shown in Figure 14.

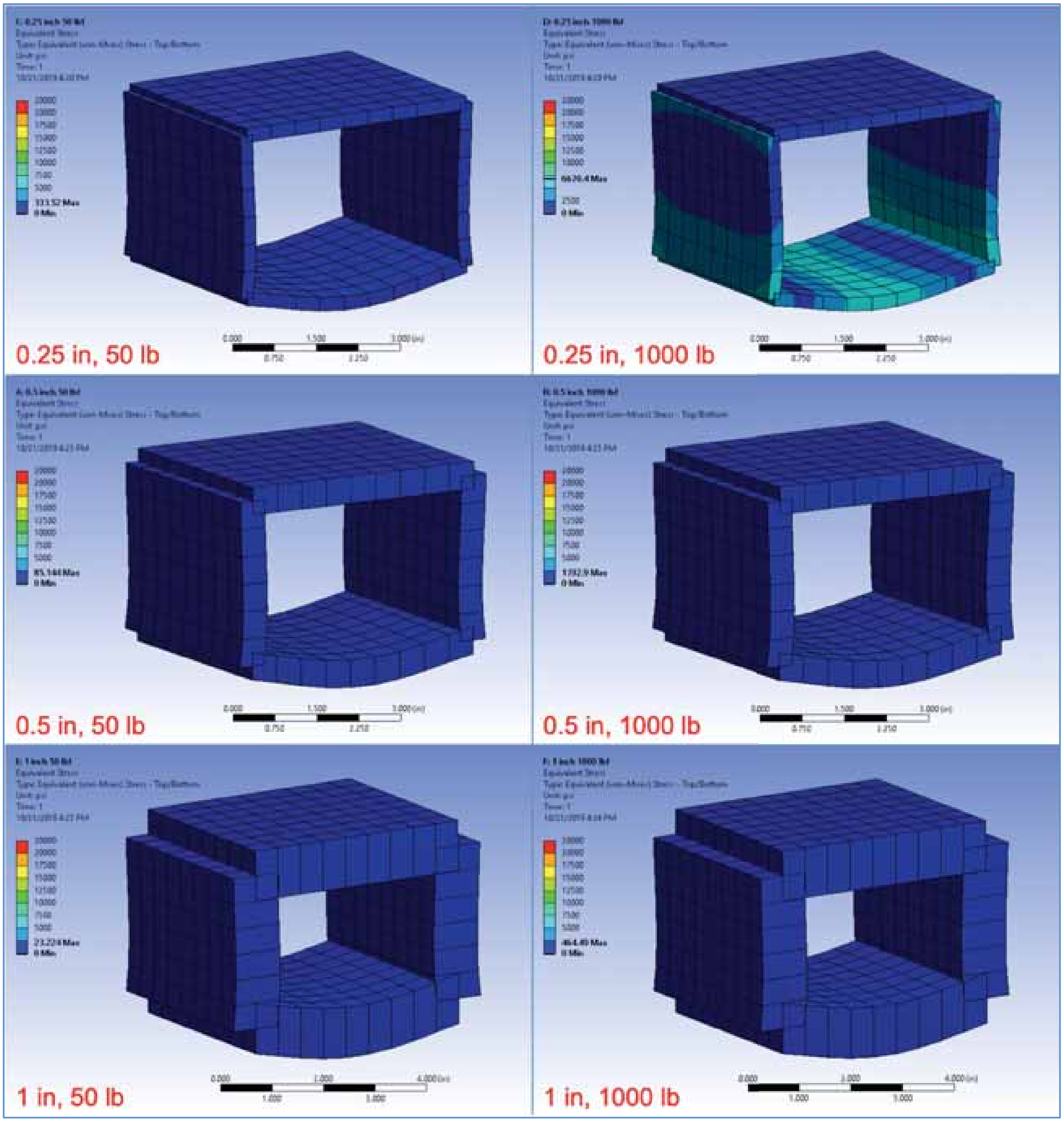

Figure 14: *Cart ANSYS Results*

For the hand calculations, the load was approximated as a single point load in the center of the plate, which provides a more conservative estimate. The stress was calculated using Equation 1. The moment is given by

$$M = \frac{FL}{2} \quad (4)$$

and the area moment of inertia is given by

$$I = \frac{bh^3}{12} \quad (5)$$

where b is the base length and h is the thickness [11]. Table 11 compares the results from the simulation and hand calculations, and Table 12 shows the margins with a safety factor of 2.

Table 11: *Cart Stress Analysis*

	ANSYS model		hand calculations	
bottom plate thickness	50 lb load stress (psi)	1000 lb load stress (psi)	50 lb load stress (psi)	1000 lb load stress (psi)
0.25"	**333.5**	**6670.4**	**2400.0**	**48000.0**
0.5"	**85.1**	**1702.9**	**600.0**	**12000.0**
1"	**23.2**	**464.5**	**150.0**	**3000.0**

Table 12: *Cart Stress Analysis Margins*

		ANSYS model		hand calculations	
bottom plate thickness	stress type	50 lb load stress margin	1000 lb load stress margin	50 lb load stress margin	1000 lb load stress margin
0.25"	yield	**58.97**	**2.00**	**7.33**	**-0.58**
	ultimate	**66.46**	**2.37**	**8.38**	**-0.53**
0.5"	yield	**233.90**	**10.74**	**32.33**	**0.67**
	ultimate	**263.26**	**12.21**	**36.50**	**0.88**
1"	yield	**859.44**	**42.06**	**132.33**	**5.67**
	ultimate	**966.99**	**47.44**	**149.00**	**6.50**

It is desired for the cart to be modular and compatible with the full-scale range of loads. Therefore, since the 0.25 in plate has some negative margins, a plate thickness of 0.5 in was chosen for the design.

9.3 Motor Sizing

Calculations were performed to evaluate the selected motor, which has a maximum angular velocity of 15 rad/s with no load, and a stall torque of 167 lb-ft. A maximum deployment speed of 15 deg/s, boom length of 4 ft, and maximum load of 75 lb were considered. The maximum tangential velocity at the end of the boom was calculated with

$$v = \omega r \qquad (6)$$

where ω is the angular speed and r is the radius. Assuming a coefficient of friction of 0.5, the maximum transverse load was calculated with

$$F = \mu N \qquad (7)$$

where μ is the coefficient of friction and N is the vertical load. The product of the torque and angular velocity was calculated with

$$\tau\omega = Fv \qquad (8)$$

Assuming that this value remains constant, the selected motor should be sufficient to drive the system.

9.4 Experiments

Throughout the prototyping process, many experiments were performed to debug and optimize the system. Most experiments were qualitative and were designed to test the offloader's capability. Initial experiments on the machine vision system served to better tune the acceleration, deceleration, and speed limits on all three motors. These values were determined to be 1000 ticks/s^2, 750 ticks/s^2, and 2200 ticks/s respectively. Additionally, the system would often oscillate back and forth when the cable reading fluctuated about 0º. This sensitivity was mitigated by implementing a software deadzone of +/- 2°, so that if the reading of the cable is within this region the motors will not respond. Oscillation is not representative of the test article's true behavior, so nulling out this phenomenon does not alter the results of any offload tests.

Additionally, the initial spring sizing on the test article was not strong enough to match the response in the offloader system, resulting in random discrete jumping behavior in the drive system as opposed to the expected continuous motor control. To mitigate this, the torsional spring on the test article was increased in stiffness by attaching another spring inside of the original. The effective spring stiffness is now the sum of the two individual springs. Implementing this stiffer spring resulted in repeatable continuous deployment.

10. Final Design, Mockup, and Prototype

10.1 Prototype Construction and Design Changes

Throughout the process of constructing the prototype, a number of problems were identified. As a result of this, several changes in the design were implemented. This section describes the changes from the initial conceptual design presented in section 7.4 in detail.

10.1.1 Pinned Slot Crossbar

It was found that the motion of the crossbar relied heavily on very precise alignment of the linear bearings on each end, which was difficult to ensure. To further facilitate this movement, a pinned slot crossbar design was utilized. This adds an additional DOF to the system, freeing the motion of the crossbar. A schematic of this system is shown in Figure 15.

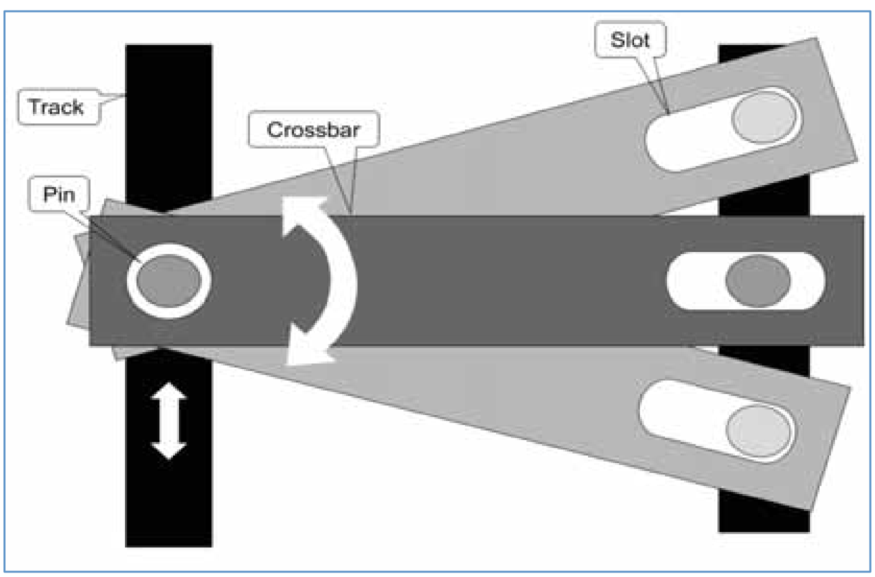

Figure 15: *Pinned Slot Crossbar*

10.1.2 Angle Support Torquing

Eight angle supports were used to provide extra rigidity to the top of the frame, as shown in Figure 16. It was found that torquing these supports too much would deform the 80/20 rails. Due to their small clearance, this would cause the linear bearings to jam. Therefore, the torquing of the angle supports was optimized so that they provided as much support as possible without causing deflection in the top horizontal 80/20 rails.

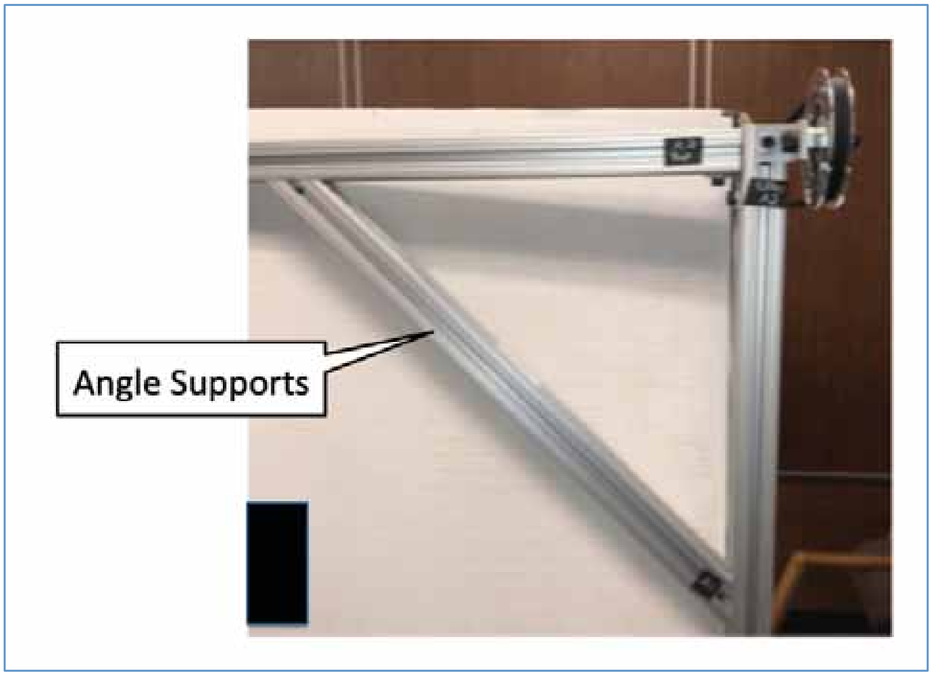

Figure 16: *Angle Support Torquing*

10.1.3 Cart Redesign

Several changes were made to the design of the cart. Firstly, two linear bearings were used instead of one, in order to distribute the load and stabilize the motion. Additionally, elongated vertical plates were attached directly to the sides of the bearings. This saved material and simplified the design. Finally, extension plates were added to the bottom of the cart. This was to allow enough distance for the Pi cameras to adequately focus on the tension cable. The redesigned cart is shown in Figure 17.

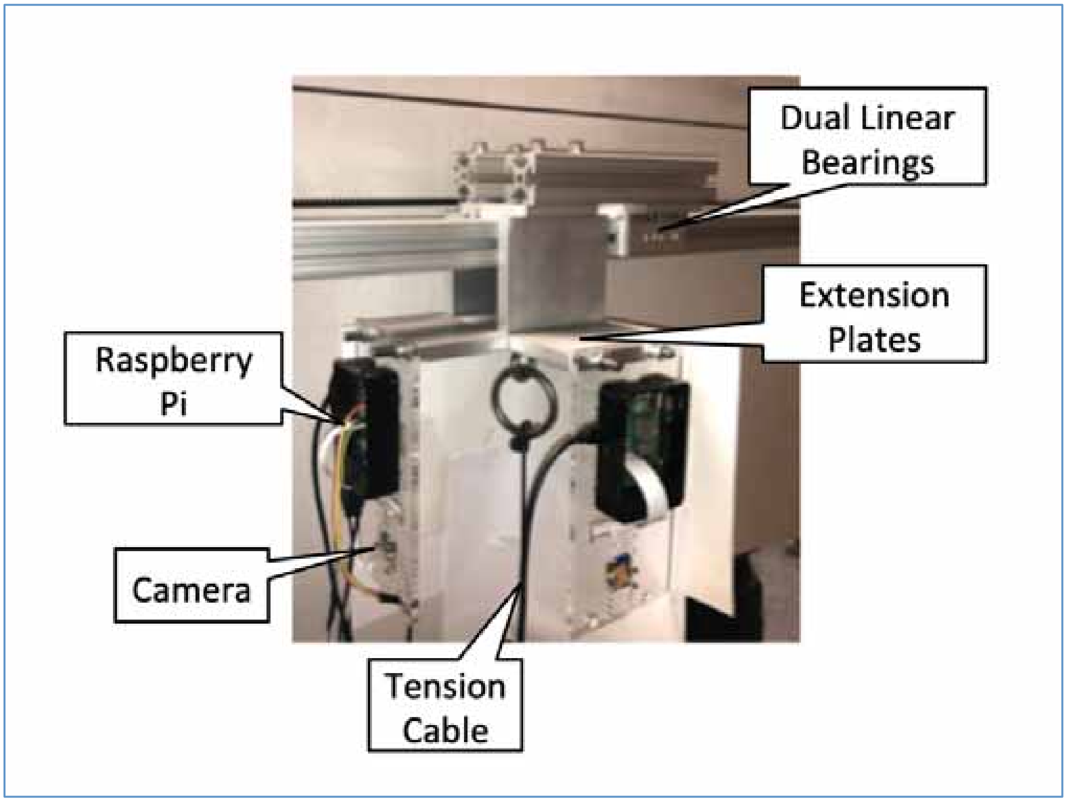

Figure 17: *Cart Redesign*

10.1.4 Camera Assembly

Acrylic plates were cut to house the camera assembly. The Raspberry Pi computers and Pi cameras were mounted on these plates. The plates included holes for cables to pass through, and allowed for adjustment in the vertical positioning of the Pi cameras. This is shown in Figure 18.

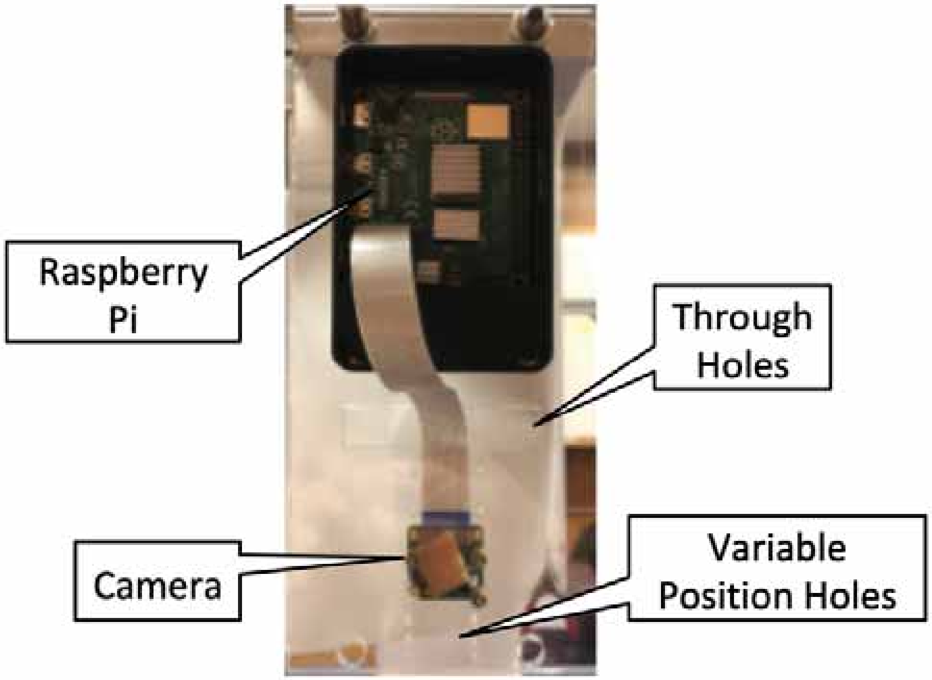

Figure 18: *Camera Assembly*

10.1.5 Motor Controller Plate

The design includes the use of two motor controllers--one to control the two motors that move the crossbar, and one to control the single motor that moves the cart. To house these motor controllers, a plate was cut from insulating acrylic material. This plate has secure attachment points to be mounted on the frame and includes through holes for cable management. The motor controller plate is shown in Figure 19.

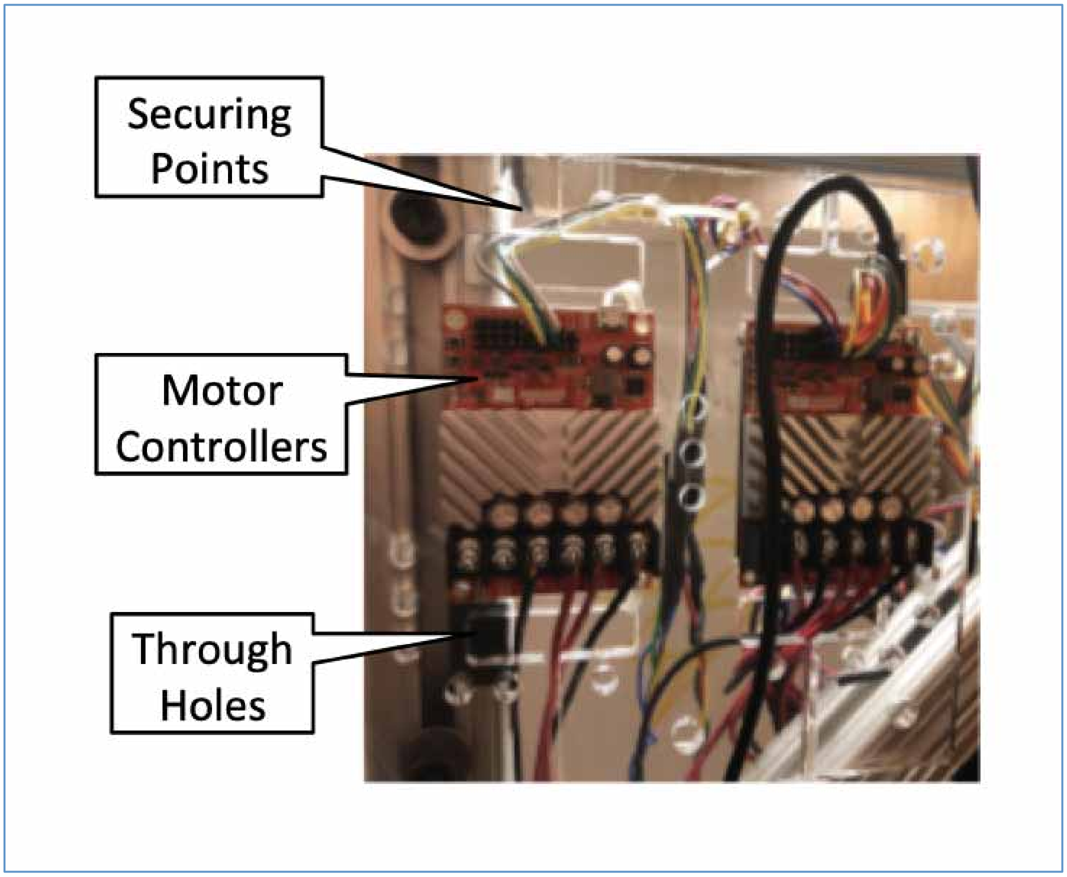

Figure 19: *Motor Controller Plate*

10.2 Prototype Performance and Analysis

As mentioned in section 9.4, many tests were conducted to optimize the performance of the prototype. Buffers in the machine vision software were implemented in order to reduce oscillation when the system is at rest and prevent overshoot or misreads when a high (greater than 10°) angle is detected.

When booting up the system, each individual system is tested in isolation and then integrated methodically. Specifically in one direction, the drive system is tested, the machine vision is validated, and then the systems are integrated. Then, the same steps are followed for the other direction. Finally, both systems are run together.

In testing the drive system, motors are controlled with a varying duty cycle input to ensure that their encoders are reading correctly on the controlling computer. Issues with encoders have been experienced due to signal noise caused due to the motor power cables' proximity to the encoder data cables. Additionally, wires were coiled initially to reduce clutter in the system, but this coiling likely induced a magnetic field affecting the motor encoders' quadrature magnetic systems. Lastly, wires were tucked inside of coated aluminum rails, which essentially serves as an insulated chamber bouncing noise around causing encoder readings to drift endlessly. Once the wires were removed from this insulated chamber the encoder drift started. By limiting power cable proximity, coiling of wires, and trapping encoder wires, the performance of encoders can be better maintained. With encoders reading correctly, position limits can be applied via software to the motor controllers and position control can be tested.

With the drive system confirmed, machine vision is tested to read 0º when the system is at rest and is tested for continuous encoder readings. When the encoders do not provide consistent readings, the system stalls and the resulting motor behavior is not continuous. Therefore, confirming machine vision behavior is necessary in validating the overall system.

With both drive system and machine vision validated, the directional system can be executed and the systems in the other axis can be validated as well. With both directions working, the overall system is live and functional.

The ultimate inhibitors of performance in the offloader derive from the hardware, specifically the electronics. Each system must be isolated and tested for proper functionality before final system integration.

11. Manufacturing

The emphasis on this design was to produce a proof of concept on a short time scale. Because of this, the intended manufacturing processes were restricted to machining and laser cutting. These restrictions, however, are also representative of the manufacturing plan for real world implementation, as the modular design of the offloader is only intended to be manufactured a few times to cover a variety of different scales and deployment profiles. Therefore, it is reasonable to assume that each part of the offloader can be custom machined to ensure accuracy and allow for small-scale production of the system.

During the design process, the manufacturing method was kept in mind when designing individual parts. When possible, parts were designed to have features in a single plane, and were sized with overall dimensions that allowed for the use of a few sizes of common stock. When not possible, the design was made to be symmetrical, allowing for reduced machine programming time on parts that required multiple setups. The material for custom machined parts was kept consistent with 6061 Aluminum in order to allow for easy machining and limit the number of tools required.

Tolerancing and manufacturing specifications are outlined in the provided fabrication package. Due to the implementation of plastic linear bearings and their inherent play, no surfaces are critical enough to necessitate unusually tight tolerances. During final assembly, it must be ensured that the system is level in order for the machine vision to work properly. The play and adjustability in many parts of the system allow for small adjustments to be made throughout the assembly process.

For the produced prototype, the associated manufacturing cost would be limited to the manual machining time used for the manufactured components. An estimate of 15-20 hours was required to produce the associated components, so manufacturing cost would be associated with the hired machine shops rate.

12. Societal, Environmental, and Sustainability Concerns

12.1 Material Regulations

Ball Aerospace is subject to International Traffic in Arms Regulations (ITAR) which limits potential sources for materials to suppliers within the United States for national security purposes. This largely impacted sourcing of electronic components from US suppliers. Limited US suppliers further impacted the overall budget for the project with increased electronics costs.

12.2 Societal, Environmental, and Sustainability Considerations

Societal considerations, while important, are more subject to the sponsoring companies' actions in the space industry and their use of the Universal Offloader. Ball Aerospace is an American manufacturer of spacecraft, components, and instruments for national defense, civil space, and commercial space applications. The Universal Offloader will test satellite booms for the customer to further their business goals, which further American aspirations in science and defense security.

The premise of this project is to create one Universal Offloader to replace multiple specialty built offloaders, which inherently decreases materials consumed and waste. The 80/20 frame is made of aluminum, and while the metal is mined, aluminum is endlessly recyclable even with its anodized coating**.** The rest of the Universal Offloader is composed of fasteners and various electronic pieces, which also can be reused or recycled in a limited capacity. The modular design is sustainable as it can be reconfigured as needs change in the future.

13. Risk Assessment, Safety, and Liability

In addition to the potential risks listed in section 7.5, a number of additional risks have been identified throughout the process of constructing the prototype. This section describes these risks and their countermeasures.

13.1 Mechanics Safety

To protect against pinch points and sharp edges, end caps will be placed on the exposed ends of the 80/20 rails, as shown in Figure 20. Additionally, acrylic gear guards will be used with both the driven and free gears, as shown in Figure 21.

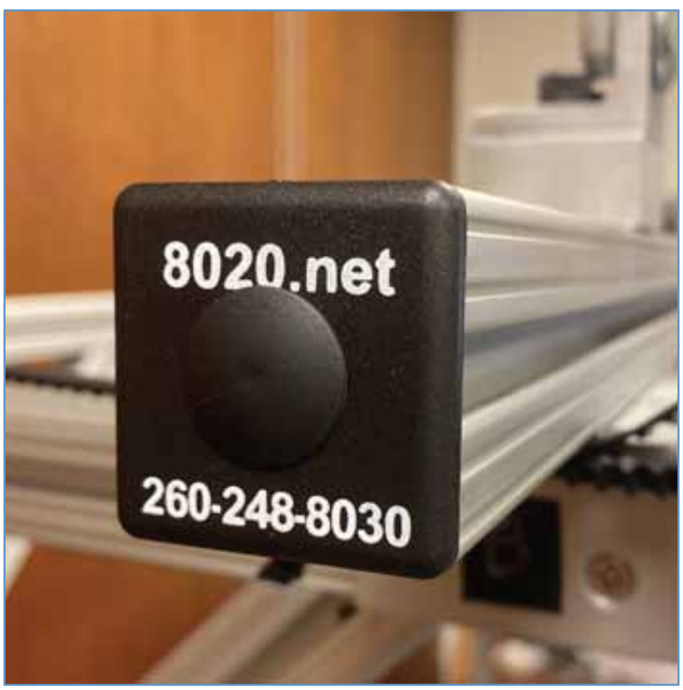

Figure 20: *80/20 End Caps*

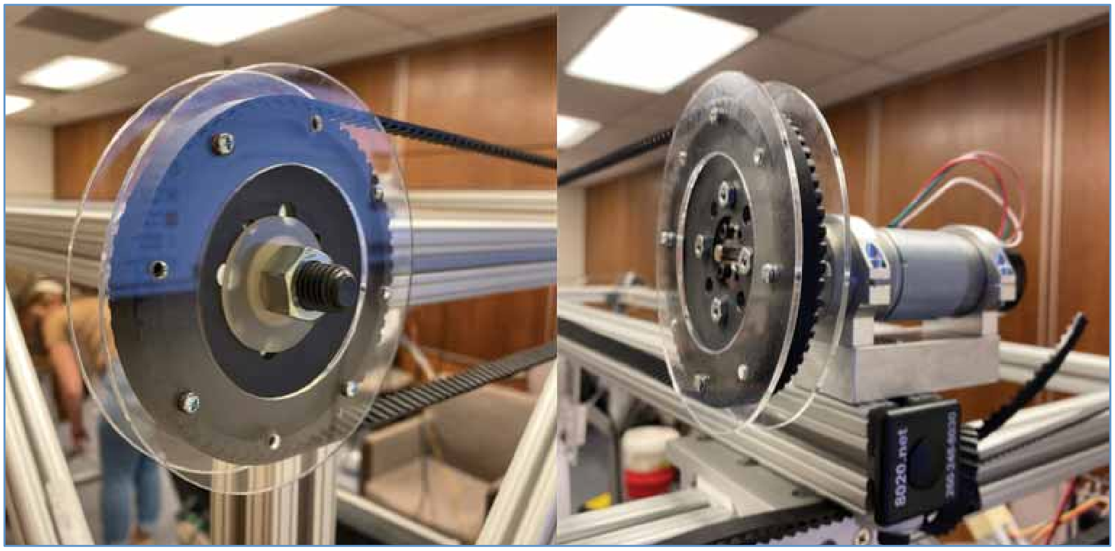

Figure 21: *Gear Guards*

13.2 Electronics Safety

The prototype uses a 600 W power supply (max 50 A at 12 V). To ensure its safe operation, there will be appropriate signage, and its location will be marked off with tape. Additionally, all connections will be sturdy with no exposed wires, and proper cable management will be used. An acrylic plate, shown in Figure 22 is used to mount the motor controllers, and excessive slack in the wires will be taped to the frame.

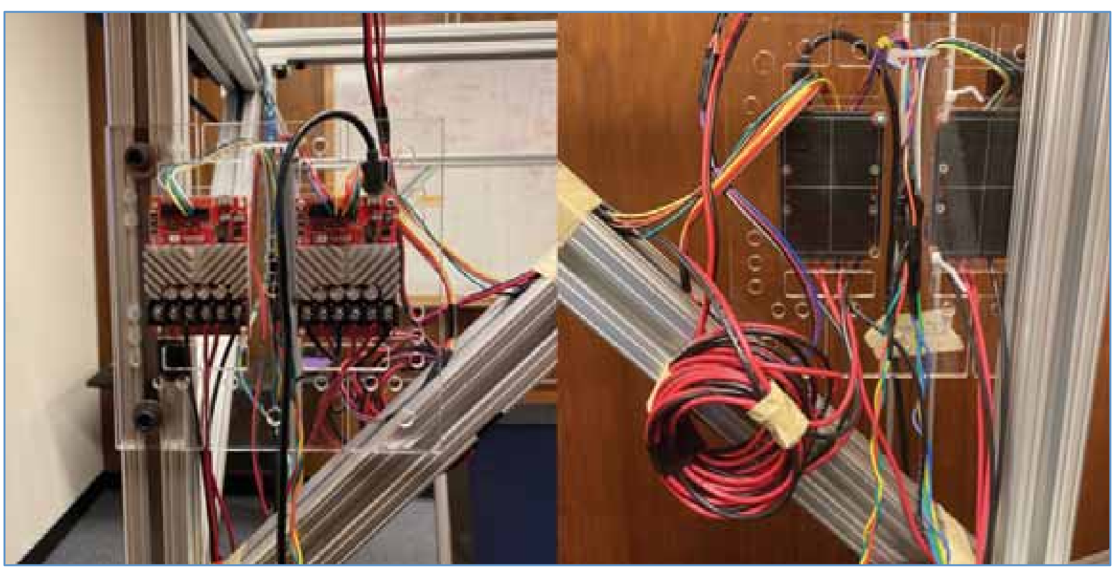

Figure 22: *Motor Controller Plate Cable Management*

13.3 Future Safety Improvements

On a system level, all wired connections will be secured and properly shielded. Inductance will be mitigated by reducing the instances of coiled wires. Additionally, single length wires will be used instead of multiple jumper cables connected in series.

From a software standpoint, position, velocity, and acceleration limits are imposed. Hardware stops can be used in parallel with these software stops to provide redundancy. Namely, limit switches can be mounted to the frame rails such that when triggered, the system halts. A singular toggle switch is used to halt both directional systems, but other toggle switches can be implemented in parallel to handle a single directional system. Additionally, more toggle switches can be installed to halt motor control while maintaining machine vision.

Furthermore from a hardware standpoint, all bolts should be torqued to a specified value depending on their purpose. Even auxiliary shear-loaded bolts used to fasten the gear assemblies should be torqued to a specified value to prevent loosening. With all fasteners torqued and accounted for, the integrated system structure is more reliable and safe.

14. Patent Claims and Commercialization

A patent on this implementation of a Universal Boom Offloader is being explored with the sponsor, Ball Aerospace. While interested in a patent of this design, the Boom Brigade team does not plan to approach additional satellite industry participants to commercialize this product beyond what Ball Aerospace desires.

15. Summary and Future Work

15.1 Project Summary and Timeline

The framework of the project plan is broken down into three phases, as shown in Figure 23, the Team Gantt Chart. For each phase, a report will be written and presented to the advisors and sponsors. Phase 1 is problem understanding and design conception, which was completed on

September 20th. Phase 2, detailed design and engineering analysis, was completed on October 25th. Phase 3 has now been completed. Phase 3 consists of the construction of a prototype, testing, and continuous design improvement based off of lessons learned. Its impact on the test article will also be measured as a means of design justification. The prototype has been built and is ready to be presented at the design expo on December 2nd. Tasks have always been assigned with soft deadlines well before part completion dates. This helped compensate for delays and modifications to tasks.

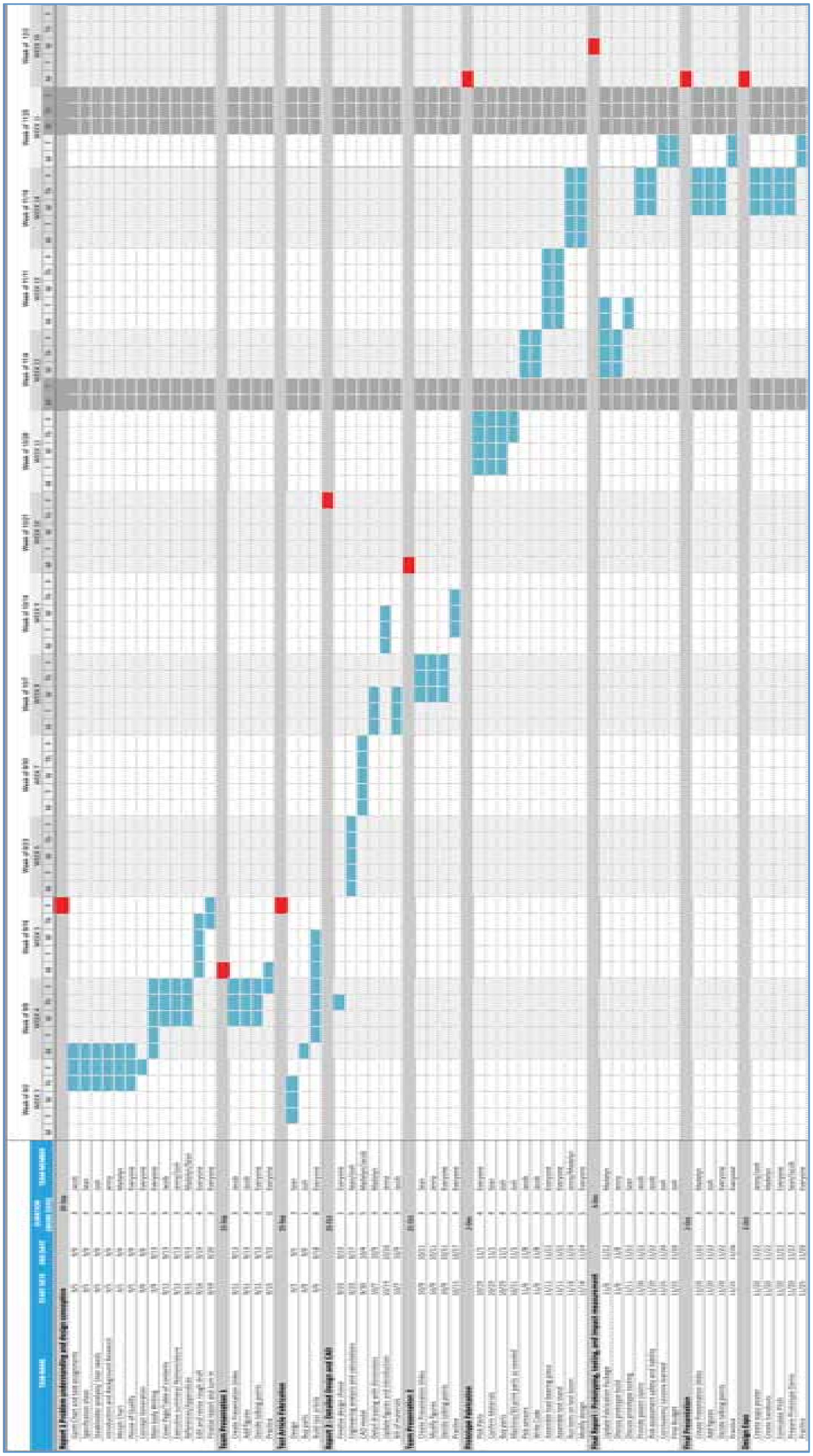

Figure 23: *Team Gantt Chart*

15.2 Future Work

If Ball Aerospace decides to continue work on this project, the team has made plans for future work and design improvements. This section presents the team's suggestions on what to consider when scaling the design up to its full size, as well as some major design changes/additions.

15.2.1 Improvements on Current Prototype and Scaling to Full Size

When the prototype is scaled to actual size, the frame and crossbar will have to be made differently to support a load of 1000 lb across 25 ft. Based on what was learned from constructing this prototype, the team would suggest using a stiffer material such as steel I-beams, since it was found that even minor deflections in the frame will cause jamming at the bearing. Another way to decrease jamming would be to use ball bearings instead of the linear bearings used in this prototype. The final and perhaps most important suggestion would be to use higher quality motors and encoders, which has been the most inconsistent aspect of the prototype.

15.2.2 Major and Functional Additions to the Design

In addition to the improvements mentioned above, the team has the following suggestions for major functional changes/additions to the design:

1) A different boom attachment can be created for a distributed load, such as with a suspended sling or flybeam hierarchy, instead of simply supporting a point load at the boom's COM.
2) The design can be altered to test booms that deploy by spinning in addition to booms that deploy by a lever arm.
3) A system can be developed for attaching different weights to the UUT to simulate different deployment structures.
4) A force transducer can be incorporated into the tension cable boom attachment to monitor the force throughout the deployment and observe its consistency.

The team is confident that with more time and resources, the design detailed in this report can be used to make a full-scale gravity offloader that can be used for any boom deployment profile.

Acknowledgements

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References

[1] Banik, Jeremy A., and Christopher H. M. Jenkins. *Testing Large Ultra-Lightweight Spacecraft*. Reston, VA: American Institute of Aeronautics and Astronautics, 2017.

[2] Herr, R. W., and G. C. Horner. 1981. "Deployment Tests of a 36-Element Tetrahedral Truss Module," http://ntrs.nasa.gov/search.jsp?R=19810010674.

[3] Chung, D. T., and L. E. Young. 1981. "Zero Gravity Testing of Flexible Solar Arrays," http://ntrs.nasa.gov/search.jsp?R=19810013865.

[4] NASA's Glenn Research Center in Cleveland, Ohio, Fact sheet, PS00566141011, 2011, Available at http://facilities.grc.nasa.gov/documents/TOPS/TopZERO.pdf, verified Aug. 22, 2016.

[5] Fernandez, Juan M., Lourens Visagie, Mark Schenk, Olive R. Stohlman, Guglielmo S. Aglietti, Vaios J. Lappas, and Sven Erb. 2014. "Design and Development of a Gossamer Sail System for Deorbiting in Low Earth Orbit." *Acta Astronautica* 103 (October): 204–25.

[6] Lichodziejewski, David, John West, Richard Reinert, Kara Slade, and Keith Belvin. 2004. "Development and Ground Testing of a Compactly Stowed Inflatably Deployed Solar Sail." In *45th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics & Materials Conference*. Structures, Structural Dynamics, and Materials and Co-Located Conferences. American Institute of Aeronautics and Astronautics.

[7] Stokkermans, Joep. 2006. Gravity compensation device. TW:200607746:A. *Patent*, filed May 25, 2005, and issued March 1, 2006. https://patents.google.com/patent/TW200607746A/en?q=gravity&q=compensation&oq=gravity+compensation.

[8] Adetona, Olawale, Lee Keel, Lucas Horta, Dave Cadogan, George Sapna, and Steve Scarborough. 2002. "Description of New Inflatable/Rigidizable Hexapod Structure Testbed for Shape and Vibration Control." In *43rd AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference*. Structures, Structural Dynamics, and Materials and Co-Located Conferences. American Institute of Aeronautics and Astronautics.

[9] Adler, Aaron, Nick Hague, Greg Spanjers, James Goodding, David Murphy, Martin Mikulas, and Brian Engberg. 2003. “PowerSail: The Challenges of Large, Planar, Surface Structures for Space Applications.” In *44th AIAA/ASME/ASCE/AHS/ASC Structures, Structural Dynamics, and Materials Conference*. Structures, Structural Dynamics, and Materials and Co-Located Conferences. American Institute of Aeronautics and Astronautics.

[10] □英民 and 施浩. 2014. “Design Method of Unrestricted Suspension Type Initiative Gravity Compensation System.” *Patent*, January. https://patents.google.com/patent/CN103482089A/en?q=gravity&q=compensation&oq=gravity+compensation.

[11] Budynas, Richard G., and J. Keith Nisbett. Shigley's Mechanical Engineering Design, Ninth Edition. New York, NY: McGraw-Hill, 2008, Table A9.

[12] 80/20 Inc. Deflection Calculator. https://8020.net/deflection-calculator.