

Ground Support Equipment for Northrop Grumman Massive Heat Transfer Experiment

A Senior Project

presented to

the Faculty of the Aerospace Engineering Department  
California Polytechnic State University, San Luis Obispo

In Partial Fulfillment

of the Requirements for the Degree

Bachelor of Science

by

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# Ground Support Equipment for Northrop Grumman Massive Heat Transfer Experiment

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**California Polytechnic State University students designed, built, and certified ground support equipment for the Northrop Grumman Massive Heat Transfer Experiment. The Cal Poly design team built the 10000, 20000, and 30000 assemblies to meet Northrop Grumman requirements. The requirements included interface limitations, design load factors, delivery, and testing specifications. The design process consists of requirements generation, conceptual design, preliminary design, design reviews, manufacturing, and certification. The hardware was successfully completed and is used at the Johnson Space and Kennedy Space Center.**

## Nomenclature

*Cal Poly* = California Polytechnic State University San Luis Obispo  
*CG* = Center of gravity  
*EIDP* = End Item Data Package  
*FBD* = Free Body Diagram  
*FS* = Factor of Safety  
*GSE* = Ground support equipment  
*KSC* = Kennedy Space Center  
*MHTEX* = Massive Heat Transfer Experiment  
*NGAS* = Northrop Grumman Aerospace Systems  
*P* = Load used in analysis or design

## Subscripts

*U* = Ultimate  
*Y* = Yield  
*Proof* = Proof load  
*left* = Left direction  
*right* = Right direction  
*rear* = Aft longitudinal direction

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## **I. Introduction**

Cal-Poly Aerospace engineering students designed, manufactured, and certified ground support equipment for the Northrop Grumman MHTEX. The GSE was subcontracted to Cal Poly in order to reduce operating costs, facilitate practical student education, and establish a relationship that would foster future projects and partnerships. Under the supervision of David Esposto, 7 students conducted all design and manufacturing work from the Cal Poly campus. Bi-weekly teleconferences were conducted between the students and the customers who consisted of Northrop Grumman, project managers, leads, engineers, AFRL STP-H3 Project Manager and Kennedy Space Center Integration Managers. This report discusses the design process, manufacturing, certification, and lessons learned by the students.

### **A. Ground Support Equipment**

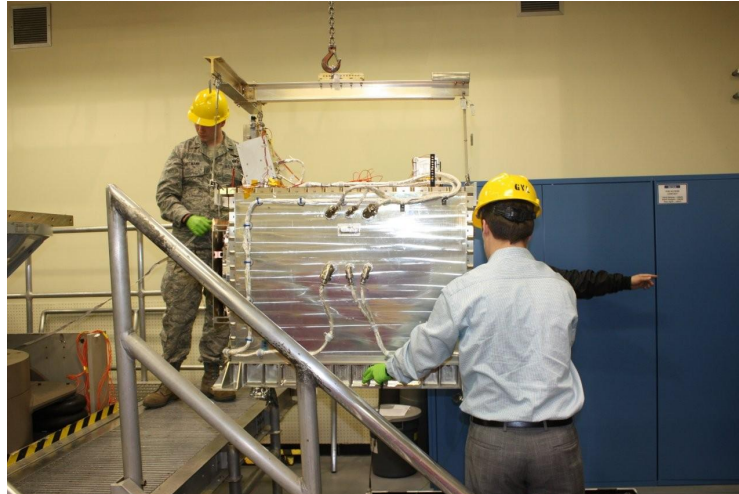
Ground support equipment is specialized equipment that interfaces between flight hardware and testing equipment. Often the flight hardware is not designed for movement with lifting hooks during phases of assembly, integration, or manufacturing. Space flight hardware is not easily secured to testing machines during the testing phase because standardized testing equipment has flat plates and hole patterns that are not spacecraft specific. As a result, specialized lifting slings, jigs, adapters, and bars are machined to interface flight hardware with standard company equipment during stages of assembly, testing, and flight-checkout.

### **B. Proof Testing**

All hardware is certified to a level that will sufficiently support the design loads by a satisfactory margin. The interface loads are increased by specified proof test load factors for assurance that the ground support hardware fixture or sling will not fail in use on space flight hardware. The hardware is then in a manner that is consistent with the designed use. This includes appropriate loading scenarios, exact adapters and fittings, and correct usage times. Each proof test is supervised by company quality assurance personnel that ensure the validity of each test. It is required that all test fixtures be certified to satisfactory levels. All equipment certifications and test results are documented by the quality assurance personnel for future reference

### **C. Northrop Grumman MHTEX**

The MHTEX was a Northrop Grumman experiment that was used to conduct thermal experiments at the International Space Station. It consists of 4 main experiments: Vader, Disk, Canary and Capillary Pump Hoop. Along with these experiments, the MHTEX had multiple configurations with and without the ExPA-pallet. The ExPA-pallet was a large structural plate that provided various tie down points both for ground purposes and flight use. The MHTEX required GSE to support the flight hardware through the manufacturing and testing phases. While being manufactured, the GSE was used to aid in assembling the main exterior structure, internal components, and top cover experiments. The GSE was also used to transport the manufactured experiment to different test stands such as the vibration tests and thermal vacuum tests. The MHTEX was manufactured and tested in Northrop Grumman Redondo Beach and Kennedy Space Center.



**Figure 1. 10000 Assembly being used to move MHTEX**

#### **D. 10000 Assembly: Horizontal Lift Sling**

The horizontal lift sling was designed to interface between the flight MHTEX hardware and standard lifting equipment. During the assembly, testing, and integration stages the horizontal lift sling allows technicians to move the MHTEX enclosure, ExPA-pallet, top plate of the MHTEX with experiments, and the MHTEX with a standard hoisting hook. The sling seen in Fig. 1 was moving the experiment alone. It was required that the horizontal lift sling accommodate the different MHTEX configurations within a predetermined CG range. This CG range also accounts for slight configuration changes early in the MHTEX design and manufacturing phases. It was required that the MHTEX be stable and level in all lifting scenarios. The GSE included adjustments to allow the various components to remain level when lifted. There is +/- 3 inches of horizontal adjustment and +/- 2 inches of lateral adjustment for balancing each different load configuration during the assembly, testing, and integration phases. The sling operates in various climates: from the dry California climate to the moist Florida climate.

The sling components are made from corrosion resistant material per Northrop Grumman requirements. The aluminum components of the lift sling are chromate conversion coated per Kennedy Space Center Requirements. Other non-aluminum components are stainless steel. The lift sling is designed for corrosive environments and will not yield if operated loading situations outside of its design loads.

The horizontal lift sling as a whole was certified at twice the nominal loading capability per Northrop Grumman specifications, but the cable assemblies required higher certification. The cable assemblies were individually proof loaded to higher loads than those of the 10000 assembly. The sling was designed to carry the full MHTEX configuration weight of 1000 lbs. The capability of the lifting sling and the attached cable assemblies were tested and certified by Northrop Grumman personnel.

The sling was comprised of various components. The design of each component was driven by specific requirements. The part specific requirements flowed down from the system requirements. The components of the 10000 assembly differed from each other in analysis and manufacturing. Figure 2 shows the horizontal lift sling and the components listed in the table.

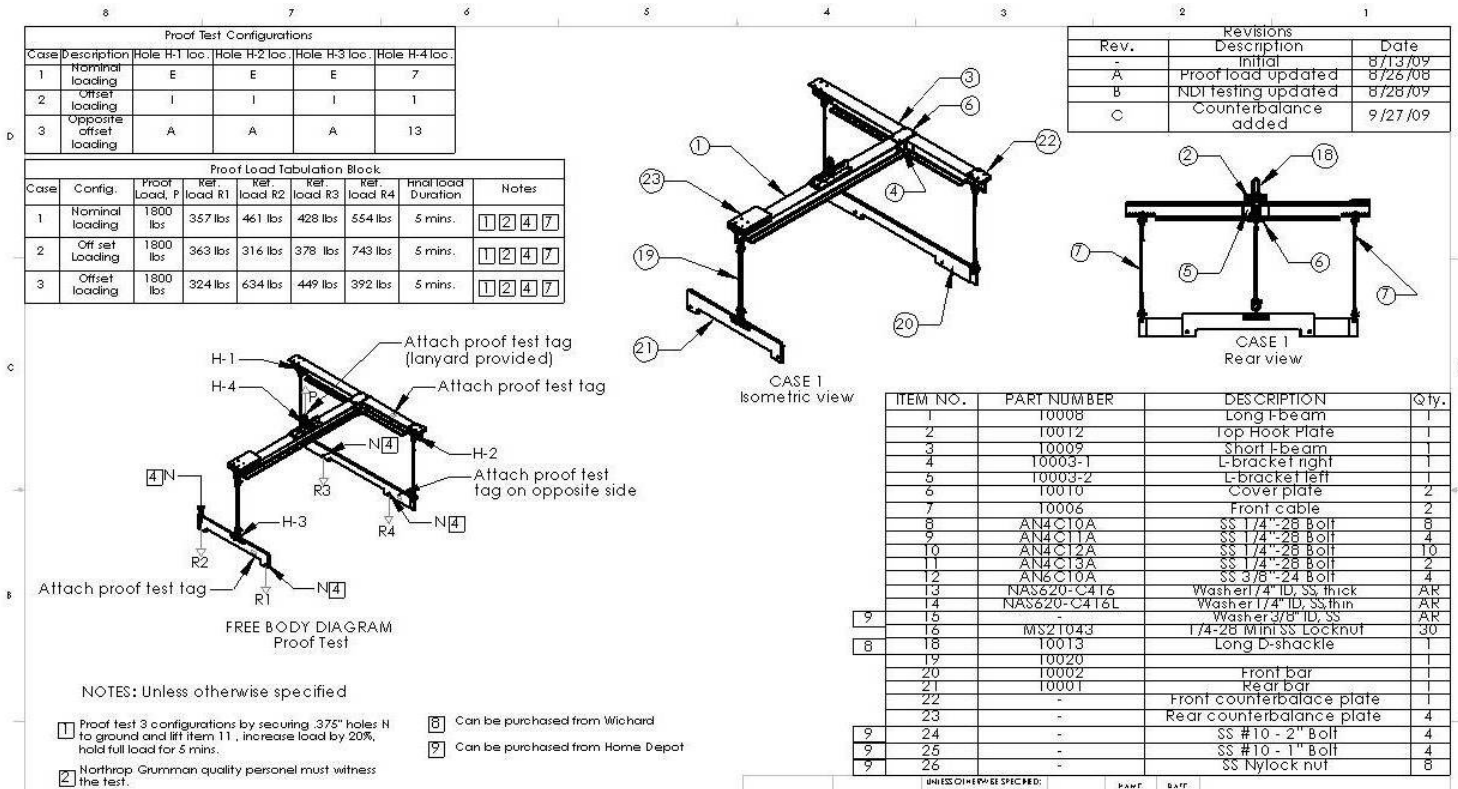


Figure 2. Horizontal Lift Sling and Parts.

### E. 20000 Assembly: Cover Rotation Assembly

The 20000 assembly was used during assembly and integration of the MHTeX experiment. The customer, NGAS and KSC, requested a fixture, which could hold the top cover in place during installation of wiring harnesses within the experiment. Requirements flowed down to the Cal Poly design team for CG offset, load limits, manufacturing and testing procedures, and documentation of the design and analysis.

The parts for the cover rotation assembly were manufactured from non-corrosive 6061-T651 aluminum. Fasteners and other hardware were manufactured from stainless steel. The 20000 assembly was tested as individual parts and loaded to a proof load of a minimum of 50 lbs, twice the expected working load. This was done to ensure the assembly would not fail from over loading or use not specified by the design. The completed assembly is seen in use in figure 3.

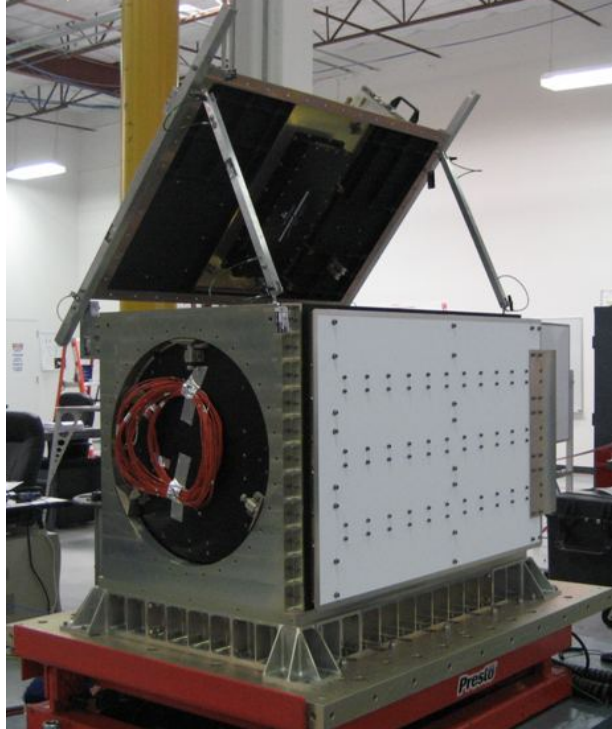
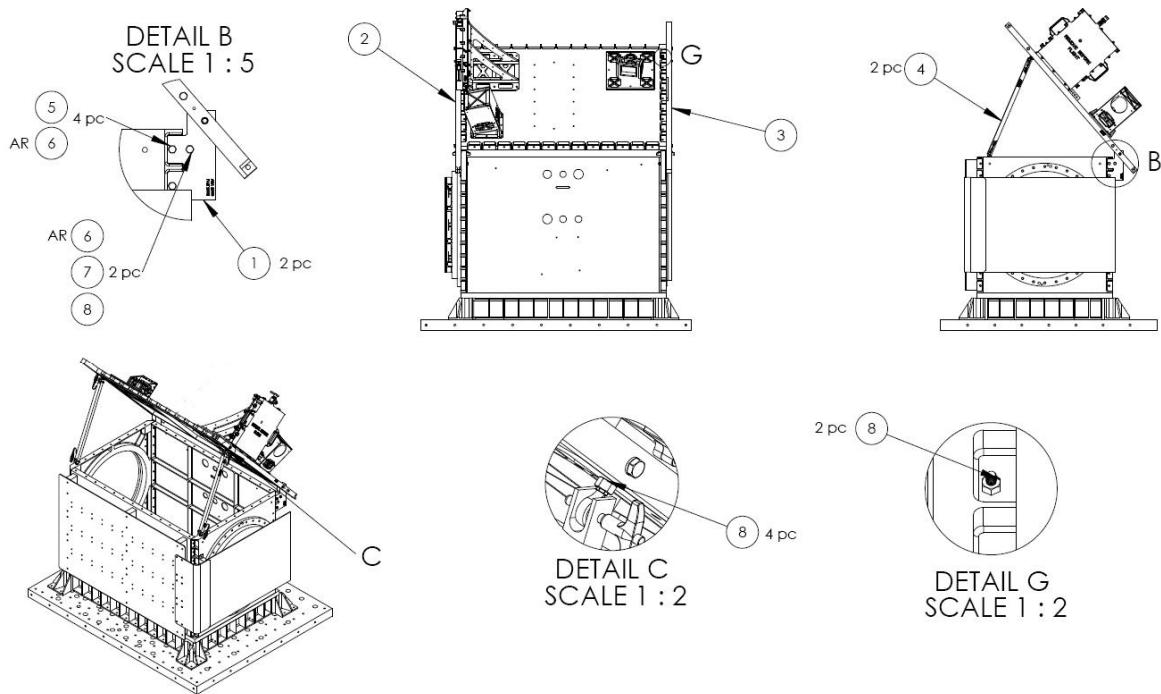


Figure 3. The 20000 assembly being used during assembly and integration of the MHTeX.

The 20000 assembly was comprised of various components. The design of each component was driven by specific requirements. The part specific requirements flowed down from the system requirements. The components of the 20000 assembly differed from each other in analysis and manufacturing. Figure 4 shows the 20000 assembly and the components listed in the table.



Item No.	Part Number	Description	Qty.
1	20001	Swivel Bracket	2
2	20002	Front Lift Bar Assembly	1
3	20003	Rear Lift Bar Assembly	1
4	20005	Support Bar Assembly	2
5	20008	SS 1/4"-28 Swivel Bracket Bolt	4
6	NAS620C416	SS 1/4" Washer	AR
7	AN4C14A	SS 1/4"-28 Bolt	6
8	MS 211043-4	SS 1/4"- Lock Nut	8

Figure 4. 20000 assembly for top cover lifting and rotation configurations and parts.

### F. 30000 Assembly: ExPA-Pallet Lift Assembly

The Cal Poly design team was requested to design and lifting fixture for the ExPA sim plate for use during assembly and integration of the MHTEX experiment. This would enable NGAS and KSC personnel to move the ExPA sim plate in the high bay area without manually lifting the hardware. The requirements for the 30000 assembly flowed down from the customers, NGAS, KSC, and internally from the Cal Poly design team. In addition requirements were also flowed down from the 10000 assembly, which during the design phase of the 30000 assembly was already being manufactured and tested.

The 30000 assembly was comprised of the 10000 assembly without the front and rear lift bars and three rod end bearings. The design of the 30000 assembly was to ensure that the 10000 assembly was capable of lifting the ExPA sim plate with a worst case CG offset. Figure 5 shows the 30000 assembly and the parts list.

Item No.	Part Number	Description	Qty.
1	59915K272	Rod End Bearing	3
2	10000	Lift Sling	1
3	N/A	ExPA sim plate	1

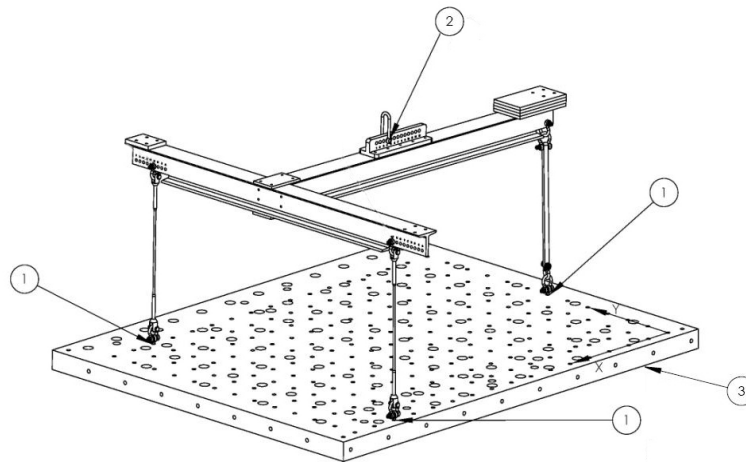


Figure 5. 30000 parts list and assembly for lifting of the ExPA sim plate.

## II. Design Methodology

### A. Project Flow

The MHTEX ground support hardware project began with a request for proposal, RFP. The customer outlines needs and Cal Poly responds with a commitment. After receiving the RFP, clarification of the requirements helps understand specific needs of the customer. The conceptual design is the first idea generated by the particular requirements driving the preliminary design. The concept is presented to the customer as a way of requirement confirmation and shows the customer we understand the requirements.

A preliminary design expands from the conceptual design. The design concept is divided into parts, the parts are given specific dimensions, materials are chosen and supporting analysis show the customer that the preliminary design meets requirements. During the Preliminary Design Review, P.D.R, the customer reviews the preliminary design. The customer accepts or declines the preliminary design. The customer is continually updating the design team with any changes since the original definition of requirements. After the preliminary review, changes are made and more in depth analysis follows.

The CDR is the final design review that approves the assembly for manufacturing. After the design review, technical drawings and specifications are released. The materials and tooling are purchased for the manufacturing. After each component of the system is purchased and manufactured the part is assembled. Aerospace ground support hardware is proof tested before it is used on space flight hardware. After the aerospace company certifies the lifting capability of the lift sling or other assemblies, it is delivered to the facility. A summary of the project flow can be found in Fig. 6.

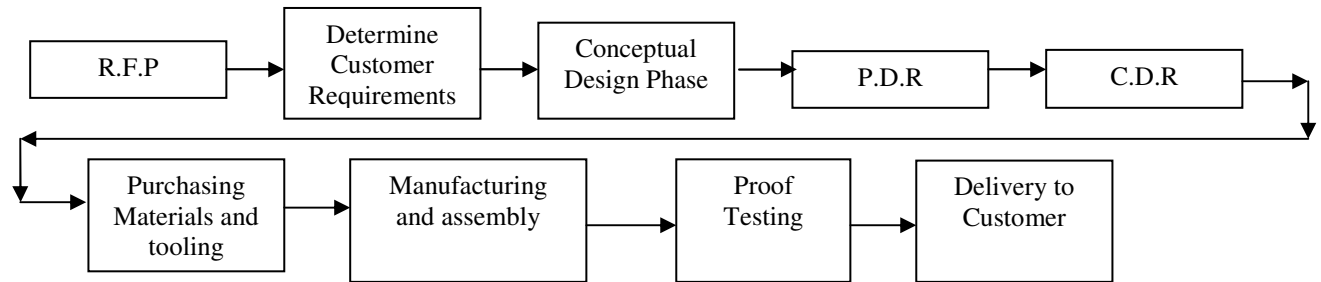


Figure 6. Project Flow Diagram

### B. Design Concept

The preliminary concept of GSE operations was presented to the Cal Poly students as shown in Fig. 7. This document shows early concepts of the various ground support hardware needed to support the MHTEX project during its assembly, testing, and integration procedures. Cal Poly committed to complete the proposed tasks, but it was later determined that not all of the tasks were necessary. It was finalized that required GSE would be required to lift the MHTEX horizontally with and without the ExPA pallet, the ExPA pallet individually, and rotate the top cover assembly.



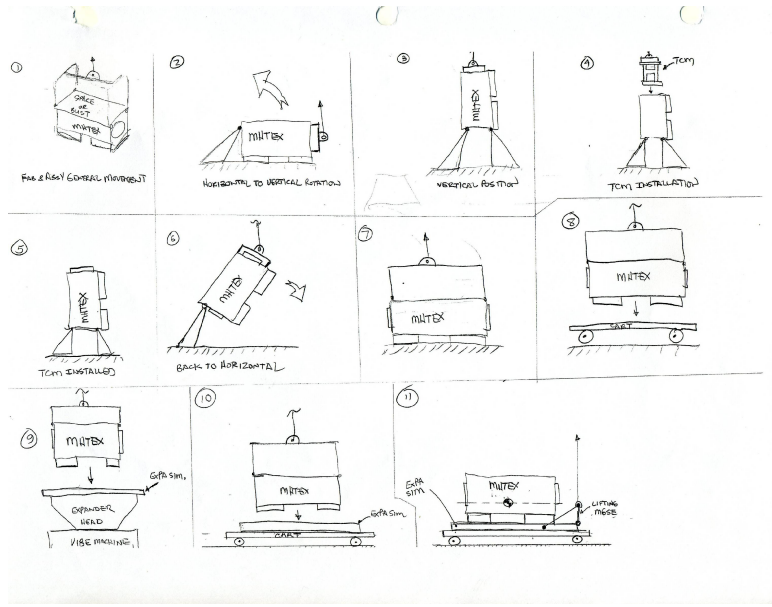


Figure 7. Conceptual Design Sketches from Northrop Grumman

### C. Delivering a Commitment

Initially, Northrop Grumman provided the Cal Poly design team with the rough sketch in Fig. 7. The sketch outlined 6 different pieces of ground support hardware for the MHTEX experiment. From the beginning of the project, it was understood by the design team that we had limited experience in part design, CAD modeling, and manufacturing. In addition, manufacturing facilities at Cal Poly were being cleaned and renovated during the summer; as a result, manufacturing equipment availability was limited. Lastly, the students were not proficient in reading technical drawings or designing for manufacturability. With those limitations in mind, the design team committed to deliver three of the six assemblies.

Professor Espoto took the time to teach the students the basic engineering process and fundamental structural design. The students learned that commitment to projects is thought out and pre planned. In addition to the actual hardware, the finished products include sufficient analysis, CAD models, drawings, and certifications.

### D. Understanding the Customer's Needs

Open communication between the customer and the Cal Poly design team was essential. To fully understand the required hardware, the design team toured the Northrop Grumman facilities where the hardware would be used. The team took measurements of the lift hook height and observed the thermal vacuum chamber and vibration table that the hardware would interface with. Bi-weekly teleconferences with Northrop Grumman were conducted to allow both Cal Poly and Northrop Grumman to update the status of the GSE and MHTEX hardware. The teleconferences were used to ensure that the analysis conducted by the students were all encompassing and adequate for the customers' needs. Northrop Grumman also communicated any changes to the MHTEX that would affect the design of the GSE.

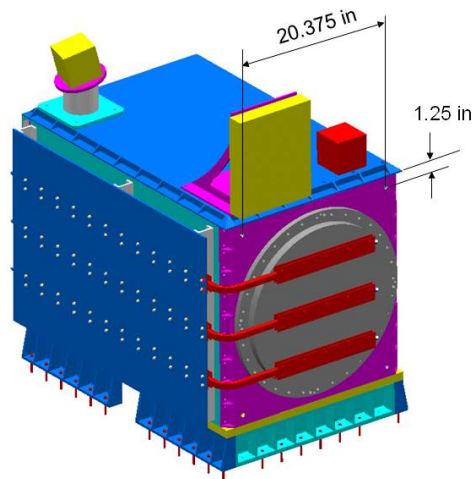
## III. 10000 Assembly: Horizontal Lift Sling

### A. Requirements

## 1. Requirements for the 10000 Assembly

Requirements for the horizontal lift sling were derived from communication from NGAS, KSC, and the design group at Cal Poly. As seen in Fig. 7, in drawings 1,8, 9, and 10, the 10000 assembly was required to lift the MHTEX in the horizontal orientation from the test stand to the vibe table via crane. To do so, requirements were developed to protect the flight hardware and accommodate the various environments the hardware would be used in. Requirements were developed for interfacing with flight hardware, GSE material strength, testing procedures, and transportation issues.

Interfacing with flight hardware entailed protecting the flight hardware from damage and minimizing changes to MHTEX design. To interface with the flight hardware, two interface points were provided identically on the front and rear faces of the MHTEX. These points consist of 3/8''-16 Keenserts manufactured directly into the faces of the MHTEX and were provided by NGAS. These points were determined by Cal Poly and NGAS to provide the most simplistic solution that required minimal changes to the existing design for flight hardware and still provide adequate structural integrity. The interface point locations are shown in Fig. 8 below. To interface with crane hooks, the GSE was required to be compatible with 9/16'' diameter hooks or shackles in NGAS and KSC.



**Figure 8. MHTEX Interface Holes.**

Because the MHTEX was still in the preliminary design phase, the addition of these interface points was not an issue. It was required that the CAD model provided by NGAS in this preliminary design phase would not incur drastic dimensional changes. With this requirement, Cal Poly could safely design the GSE to clear all flight hardware. It was required that the GSE would not physically contact all important flight hardware including the front and rear radiator caps, Vader, Canary, and Disk components. Clearance issues took into account maximum material deflections and the dynamics of sling movement relative to flight hardware. Specific clearance issues will be discussed later in this report. Also, the GSE was required to be compatible with the various configurations of the MHTEX. A preliminary center of gravity (CG) calculation was provided by NGAS to design the horizontal lift sling. It was required that the 10000 Assembly be compatible with  $\pm 3.00''$  CG shift in the longitudinal direction and  $\pm 2.00''$  CG shift in the lateral direction on the MHTEX in 0.5'' increments.

GSE material strength requirements were dictated by the MHTEX/GSE Structural Design Requirements document found in Appendix A. Because the 10000 Assembly was considered a lifting/hoisting device, the required load factors were determined and are listed in Table 1 below. Also shown are the load factors for cables less than 0.188” diameter. For all calculations, it was required that the GSE support a 1000 lb limit load to account for MHTEX and ExPA-pallet weights.

**Table 1. Horizontal Lift Sling Load Factors.**

<b>Load Factors</b>	<b>Lifting/hoisting Devices</b>	<b>Cables &lt; 0.188” diameter</b>
Proof load factor	2.00	2.00
Design yield load factors	3.00	6.00
Design ultimate load factors	5.00	10.00

Besides material strengths, the GSE also had material and coating requirements. Because of high bay usage, the material was prohibited from being galvanized or corrosive materials. Because of this requirement, the 10000 assembly consisted of aluminum and stainless steel materials and fasteners. Also, all aluminum was required to be chromate conversion coated. These requirements protect the clean environment that the GSE would be used in.

For usage in NGAS and KSC, all GSE was required to be certified. To be certified, all GSE was required to satisfy static load testing at proof load factors. Certifications on machinery used were required and documented and NGAS quality assurance personnel were required to be present at each proof test. Testing procedures in terms of correct loading magnitudes, time, and set up were approved by the quality assurance personnel. Upon satisfactory completion of each proof test, NGAS proof test tags were placed on each assembly and were marked valid for their respective weights for a specified amount of time. Besides static load testing, all cables were subjected to non-destructive testing to test for fractures.

All GSE was required to be transported in adequate shipping receptacles. The GSE containers were required to be coated in flame retardant paint. Also, the shipping containers were required to be painted in a white or grey color. The shipping container should provide adequate clearance for fork lift use and should be correctly and clearly labeled. All GSE was also transported with an End Item Data Package (EIDP) that included all analysis, drawings, certifications, and instructions for each assembly.

**2. Requirements for Part 10001: Rear Bar and Part 10002: Front Bar**

The requirements from the 10000 assembly were flown down to the rear and front bars. It was required that the rear and front bars not physically interfere with the radiator cover or any experiments on the top plate during installation or loading scenarios. The rear and front bars were allowed to sit flush with the front and rear faces of the MHTEX and interface using 3/8”-16 bolts. It was also required that the rear and front bars extend the cable attachment points far enough away from the center of the MHTEX so that the cables would not interfere with any flight hardware, especially the Vadar, Disk, and Canary experiments on the top plate. The attachment points were required to provide clearance in all loading scenarios including a minimum 15 degree swing.

**3. Requirements for Part 10003: L-Brackets**

The requirements from the 10000 assembly were flown down to the L-brackets. The L-brackets were required to provide structural integrity to the main I-beam assembly.

#### **4. Requirements for Part 10006: Front Cable Assembly**

Requirements for the front cable assemblies were flown down from the overall lift sling requirements. It was required that each front cable assembly support the maximum load determined at the worst case loading seen at the different CG scenarios. Because the cables were required to support the same load, the front cable assemblies were identical. Also, the shackles included in the front cable assemblies were required to support these loads. The shackles were required to be captive shackles or have pins that were attached via lanyard so that the pins would not be lost or fall onto flight hardware. Because the cable assemblies were removable, it was required that each cable assembly and individual shackle be proof tagged via lanyard. As previously mentioned, the front cables were subjected to a specific set of protoflight, qualification, and proof loads and were required not to contact any flight hardware.

#### **5. Requirements for Part 10007: Rear Cable Assembly**

Requirements for the rear cable assembly flowed down from the overall lift sling requirements. It was required that rear cable assembly support the maximum load determined at the worst case loading seen at the different CG scenarios. Also, the shackles included in the front cable assemblies were required to support these loads. The shackles were required to be captive shackles or have pins that were attached via lanyard so that the pins would not be lost or fall onto flight hardware. Because the cable assemblies were removable, it was required that each cable assembly and individual shackle be proof tagged via lanyard. As previously mentioned, the front cables were subjected to a specific set of protoflight, qualification, and proof loads.

#### **6. Requirements for the Rear Tension Bar**

The rear tension bar was designed to replace the rear cable assembly after the assembly failed NDI testing. The rear tension bar had the same strength and interface requirements of the Rear Cable Assembly. The bar is aluminum and is chromate conversion coated.

#### **7. Requirements for Part 10008: Long I-Beam**

The requirements from the 10000 assembly were flown down to the long I-beam. The long I-beam was required to provide sufficient structural strength to support the MHTEX assembly. The long I-beam was also required to provide adjustment for various MHTEX CG locations. Longitudinal adjustments were required to provide  $\pm 3.00''$  in  $0.5''$  increments.

#### **8. Requirements for Part 10009: Short I-Beam**

The requirements from the 10000 assembly were flown down to the short I-beam. The short I-beam was required to provide sufficient structural strength to support the MHTEX assembly. The short I-beam was also required to provide adjustment for various MHTEX CG locations. Lateral adjustments were required to provide  $\pm 2.00''$  in  $0.5''$  increments.

#### **9. Requirements for Part 10010: Cover Plates**

The requirements from the 10000 assembly were flown down to the cover plates. The L-brackets were required to provide structural integrity to the main I-beam assembly.

## 10. Requirements for Part 10012: Top Hook Plate

The top hook plate allows +/- 3” of lateral adjustment with .5” hole spacing. The Top Hook Plate hole diameter is 5/16” to interface with the Long-D shackle. The Top hook plate part interfaces support the Northrop Grumman load safety factors. Long I-Beam flange and Top Hook plate interface load does not exceed the calculated I-Beam top flange buckling load allowable. Material is corrosion resistant and chromate conversion coated.

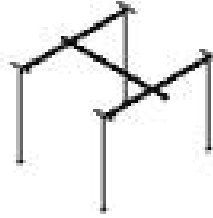
### B. Conceptual Design

The conceptual design of the 10000 assembly began by looking at the requirement that the assembly lift the MHTEX horizontally without interfering with flight hardware and provide adjustability for different CG locations. The initial concept consisted of an all cable lift sling that interfaced directly with the MHTEX without the use of front and rear bars. An example of this assemble can be seen in Fig. 9 courtesy of Pacific Rigging Loft. This assembly had the advantage of being the most simplistic and cheapest option available. The option was soon abandoned because the cables would immediately interfere with the experiments on the top plate of the MHTEX.



Figure 9. All cable lifting sling.

The next concept consisted of a single I-beam with cables attached from the I-beam to the MHTEX. While this provided more clearance for the top experiments, it was determined that the structure would be required to spread the cables wider than initially thought. The resulting design was an “T” shaped structure that provided 4 cable attachment points that ran cables vertically down to the MHTEX attachment points. The cables were spread wide enough as to not interfere with the experiments on the top plate of the MHTEX. Initial discussions with Northrop Grumman concluded that the MHTEX interface points could be machined directly to the sides of the MHTEX top cover. This configuration did not interfere with any flight hardware in a static loading scenario, but it was determined that the cables would impact the experiments in the event of the MHTEX swinging from a bump or incorrect CG adjustment. To provide margin for such a swing, front and rear bars were implemented that spread the cables wider to provide a minimum 15 degree swing angle. This 4 point lift sling satisfied the lifting and swing requirements and an example of which is shown in Fig. 10 courtesy of Rehab Mart.



**Figure 10. Four point lift sling.**

Further analysis, however, showed that the adjustments required for a four point lift sling would be difficult and time consuming to provide stability and equal loading on all cables. To account for the ease of adjustability, a 3 point lift sling was considered. By having a 3 lift point sling as opposed to 4 point sling, user adjustments for MHTEX CG would be easier and ensure more optimal cable load distribution. Because the MHTEX back face did not have experiments near the center of the top plate, a cable was allowed to come directly down from the sling assembly. The resulting design was selected to move forward with and an example of a 3 point lift sling is shown in Fig. 11 courtesy of Rehab Mart.



**Figure 11. Three point lift sling.**

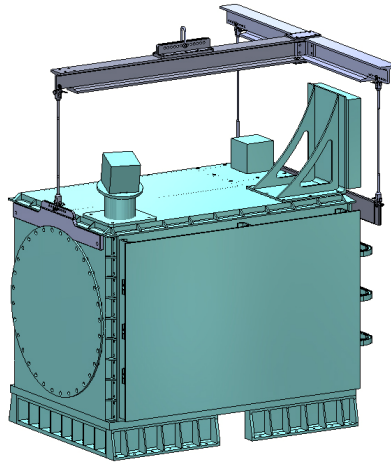
### **C. Preliminary Design**

The preliminary design of the horizontal lift sling consisted of a symmetrical I-beam assembly with cables that attached to the front and rear bars. The I-beam assembly did not include welding because of the inexperience of the students in welding and the cost for a 3<sup>rd</sup> party welder. Because of this, the I-beam assembly included L-brackets and top and bottom cover plates to attach the two I-beams. The assembly provided CG adjustability in the longitudinal direction on the long I-beam and on the top hook plate. CG adjustability was provided on both the long I-beam and top hook plate to assure that the assembly lift the MHTEX over the CG while not swinging. Lateral CG adjustability was provided on both sides of the short I-beam and on the front and rear bars for the same reason.

### **D. Finalized Design**

The final design of the 10000 assembly was very similar to the preliminary design. As the GSE progressed through analysis and design reviews and the MHTEX design became more finalized and a refined CG location became available from NGC. It was determined that the nominal CG location, or the CG location of the MHTEX without the ExPA-pallate, would not be in the center of the MHTEX. In order to account for further possible CG movement and alternate configurations, the nominal hole location, or the center hole at each adjustment point, was designed to correspond to the nominal MHTEX CG location. This led to a non-symmetrical I-beam assembly.

Also, it was desired that the lift sling be level with respect to the horizontal plane as it was being attached to the flight hardware. As the design currently stood, the non-symmetrical sling would be offset and would have been difficult to attach to the MHTEX. To account for this, counter balance plates were designed and attached to the I-beam assembly. The resulting lift sling was horizontally level as technicians attached the GSE to crane hooks and the MHTEX hardware and is shown in Fig. 12.

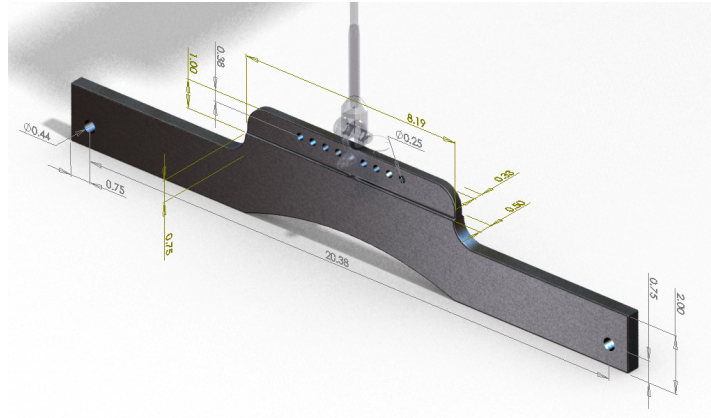


**Figure 12. Final horizontal lift sling design.**

## **E. Component Design, Analysis, and Manufacturing**

### **1. Part 10001: Rear Bar Design**

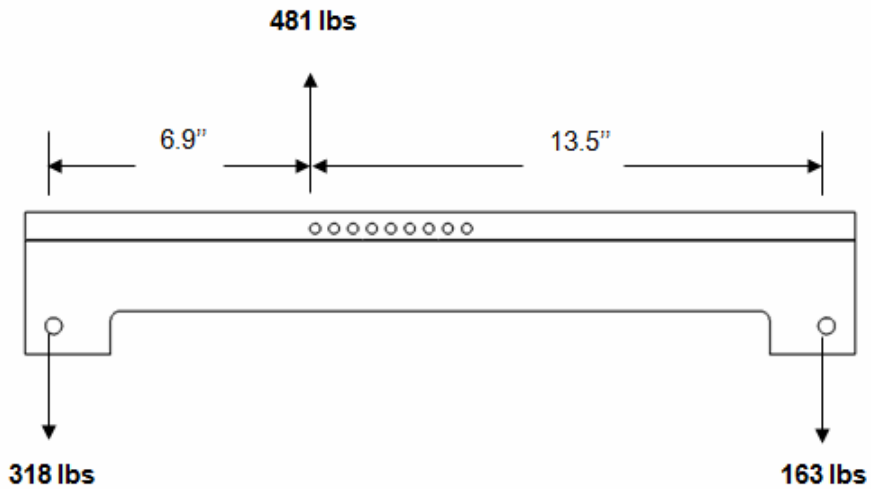
The design of part 10001, the rear lift bar, was influenced by the requirements flow down from the customer, NGAS and KSC, and internally from the Cal Poly design team. The design team originally decided to manufacture the rear lift bar out of 0.500" 6061-T651-aluminum plate. However, because of design constraints on the front lift bar and the used of common materials, 0.500" 7075-T651 aluminum was chosen for manufacturing of the rear lift bar. The rear lift bar was originally designed with rounded edges and a shaped design. For ease of manufacturing the shaped design and rounded edges were removed. A 0.100" deep channel was cut along the backside of the rear lift bar to allow for the cable assembly to interface with the array of 1/4" holes. The 1/4" holes were sized to allow enough margins on the factors of safety under max loading and enough adjustment between the holes locations to adjust for CG offset. The 3/8" holes were sized and located to allow for positional tolerances on the MHTEX and our own parts and to allow for enough margins on the factors of safety under max loading. The rear bar is shown in Fig. 13.



**Figure 13. Rear Bar.**

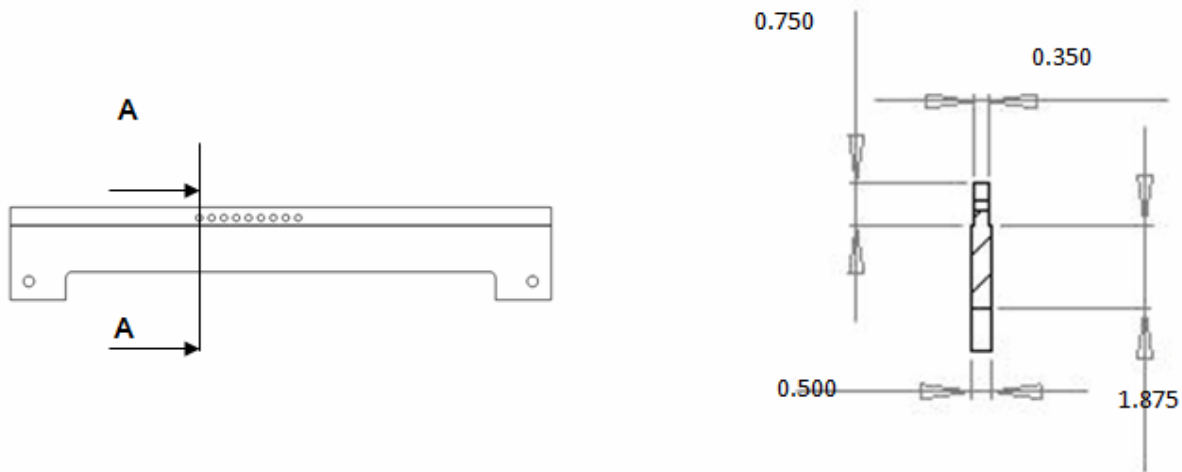
**2. Part 10001: Rear Bar Analysis**

Part 10001 was analyzed for shear, bending, and deflection under a worst case working load of 481 lbs at the minimum cross section of the bar. This applied load resulted in a working shear stress of 318 lbs as seen in the free body diagram in Fig 14, applied to the minimum cross section, as seen in Fig 15.



**Figure 14. Free body diagram of the rear bar**





**Fig 15. The minimum cross section of the rear bar.**

All lifting fixtures require a factor of safety of 5, per NGAS request, resulting in a maximum ultimate shear of 1600 lbs. The ultimate load factor was the driving value for all shear, bending, deflection, and hole calculations. To calculate the margins of safety for the rear bar geometric constants first had to be calculated. First the neutral axis about the Y-axis was calculated using equation 1,

$$Y_n = \frac{\sum AY}{\sum A} \quad (1)$$

where A is the area of the cross section and Y is the distance from that cross sections neutral axis to a common reference axis. This value was then used to calculate the shear moment using equation 2,

$$Q = \sum AY_{bar} \quad (2)$$

where A is the area and  $Y_{bar}$  is the neutral axis in the y direction. The moment of inertia was also calculated using equation 3,

$$I = \sum (I_o + AY_{bar}^2) \quad (3)$$

where  $I_o$  is the moment of inertia of the sub-cross sections, A is the area, and  $Y_{bar}$  is the neutral axis. With the geometric constants calculated the shear was calculated using equation 4,

$$\tau = \frac{VQ}{It} \quad (4)$$

where V is the ultimate shear stress, Q is the shear moment, I is the moment of inertia for the minimum cross section, and t is the thickness of the part.

The worst case applied load of 318 lbs was used at a distance of 6.9in resulting in the maximum working moment of 2200 in-lb. With a factor of safety of 5 the maximum ultimate moment was 11000 in-lb. Using the geometric constants from the shear calculations the bending stress was calculated using equation 5,

$$\sigma = \frac{My}{I} \quad (5)$$

where M is the max ultimate moment, Y is the distance from the neutral axis to the edge of the part and I is the minimum cross section moment of inertia. With the bending and shear calculated the principal stress is calculated using equation 6,

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau^2} \quad (6)$$

The tear out of all holes was calculated for three different modes of failure, shears, bearing, and tension. The shear is calculated using equation 7

$$F_{su} = \frac{P}{2xt} \quad (7)$$

where P is the load, t is the thickness of the part, and x is a geometric constant calculated using equation 8,

$$x = a \left[ \sqrt{1 - \frac{r^2}{a^2} \sin^2 45} - \frac{r}{a} \cos 45 \right] \quad (8)$$

where a is the area of the hole, and r is the radius of the hole. The bearing failure was calculated using equation 9

$$F_{bry} = \frac{4P}{\pi d(t)} \quad (9)$$

where P is the load, d is the diameter of the hole, and t is the thickness of the part. The tension failure was calculated using equation 10

$$F_{ty} = \frac{P}{(w-d)t} \quad (10)$$

where P is the load, w is the distance from edge to edge of the part across the hole diameter, d is the diameter of the hole, and t is the thickness of the part.

All margins are calculated by dividing the ultimate allowable by the expected value and subtracting 1. Table 2 presents the results of the analysis on the rear lift bar.

**Table 2: Ultimate and Yield margins of safety for part 10001: rear lift bar.**

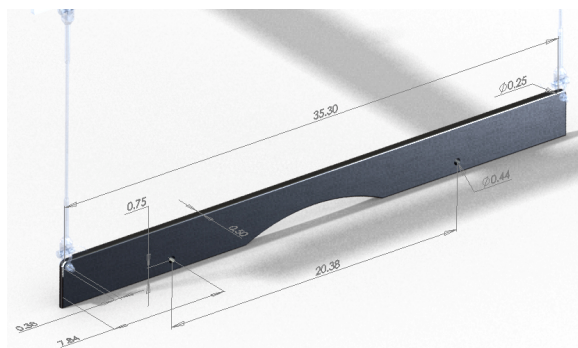
	Ultimate	Yield
Shear Stress	6.1 shear	15.3 shear
Bending stress	1.4 priciple	2.8 principle
Tear out - cable attachment	2.1 bearing	4.1 bearing
Tear out - MHTEX attachment	8.8 bearing	15.4 bearing

### 3. Part 10001: Rear Bar Manufacturing

The rear bar was manufactured from 7075-T6 aluminum. This material was chosen because of its superior material characteristics. A rectangular bar was rough cut from the 0.500” thick plate and then fabricated to the specified design using a hand mill. The holes were placed using the same hand mill.

### 4. Part 10002: Front Bar Design

The front bar was designed to interface the MHTEX with the main I-beam assembly. The GSE interfaced with the MHTEX using the 3/8” fasteners as provided by NGAS requirements. The bottom side of the front bar has a cut to allow sufficient clearance to the radiator cap. It should be noted that the final design had a rectangular cut instead of the circular cut that is shown in Fig.16. The circular cut proved to be difficult to manufacture, so a rectangular cut was used instead to provide clearance to the radiator cap. The front bar’s cable attachment holes were placed far enough away from the center line of the MHTEX to provide clearance for the cables in the event of swinging. The part was constructed out of aluminum 7075-T6 and treated with a corrosion resistance solution.

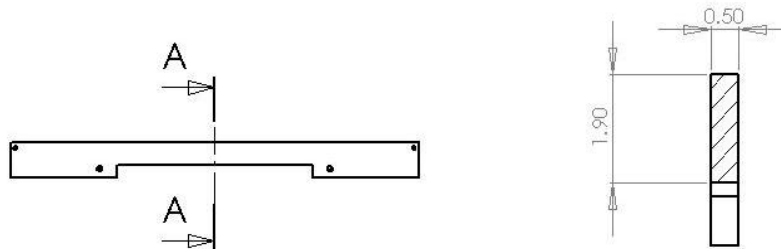


**Figure 16: Front Bar.**

### 5. Part 10002: Front Bar Analysis

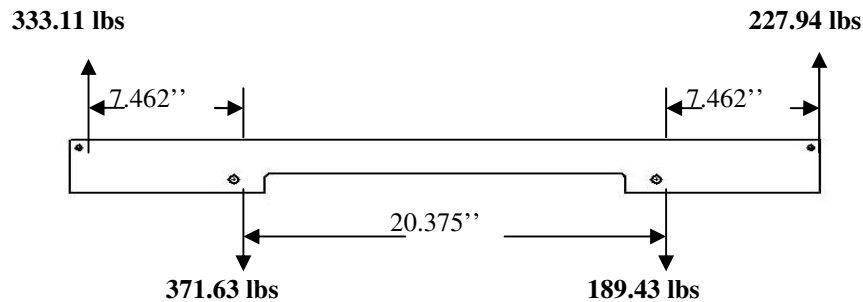
The front bar was analyzed in the same fashion as the previously discussed rear bar. The main difference, however, was that the loading due to the extended cable attachment points creates a larger bending stress in the in the center of the front bar. Because the center of the front bar was the location with minimum area, this became the critical design area. This area needed to support the worst case bending load due to the cables as well as provide enough rigidity to minimize deflection. If the deflection was too great, the center portion would deflect downward and impact the radiator cap. To analyze the bending and deflection at the minimum cross section, it was assumed that the bar had the minimum cross section across the entire length of the bar. This was

a conservative approach since the actual front bar had a larger cross section farther out towards the cable attachment points. The cross section can be seen in Fig. 17.



**Figure 17. Front bar minimum cross section.**

The worst case loading scenario for the front bar was analyzed at the A-1 configuration. In this configuration, the CG was located longitudinally closest to the front of the MHTEX. The loads at this configuration can be seen in the free body diagram in Fig. 18.



**Figure 18. Front bar worst case loading**

Using these loads, the front bar was analyzed for bending and shear stress at the minimum cross section. Also, shear, tension, and bearing tear out at the 0.254" diameter cable attachment hole and the 0.436" diameter MHTEX attachment hole were analyzed. It was determined that the principle stress bending at the minimum cross section became the designing factors for the front bar.

To calculate principle bending stress, shear and bending stress were calculated using the equations below:

$$\tau = \frac{3V}{2bh} \quad (13)$$

$$\sigma = \frac{M_{\max}y}{I} \quad (14)$$

The load of 333 lbs was used in the shear and bending load calculations. Using the calculated shear and bending loads, the principle stress was calculated using the equation below.

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau^2} \quad (15)$$

The principle stress calculation yielded a margin of safety of 0.86, which was the lowest margin of safety of all front bar calculations. Tear out analysis was conducted identically to the rear bar and it was determined that the tear out for both the MHTEX and cables attachment holes were not major designing factors. The summary of margins of safeties can be found in Table 3 below. Full analysis can be found in Appendix A.

**Table 3. Front bar margin of safeties.**

Front Bar Margin of Safety Summary		
	Ultimate	Yield
Principle stress	0.86	1.7
Bending stress	0.86	1.7
Shear stress	15	N/A
0.254" dia. hole - shear tear out	1.6	N/A
0.254" dia. hole - tension tear out	7.4	11
0.254" dia. hole - tension tear out (NASA)	5.7	8.8
0.254" dia. hole - bearing tear out	4	6.2
0.254" dia. hole - bearing tear out (NASA)	4	6.4
0.436" dia. hole - shear tear out	9.1	N/A
0.436" dia. hole - tension tear out	43	61
0.436" dia. hole - tension tear out (NASA)	27	40
0.436" dia. hole - bearing tear out	8	12
0.436" dia. hole - bearing tear out (NASA)	8	12

## 6. Part 10002: Front Bar Manufacturing

The front bar was manufactured out of aluminum 7075-T6 material. The material was chosen to provide enough structural strength required from analysis. The front bar was manufactured from a large 0.5' sheet of aluminum. A rough rectangular shape was cut out of the sheet using a vertical ban saw. This rectangle was then fabricated to specification using a hand mill. The holes were also placed and drilled out using the mill.

## 7. Part 10003: L-Bracket Design

The L-brackets were designed to connect the long and short I-beams. They were designed to fit within the webs of I-beams and were designed to support the shear and moments seen at this junction. The L-brackets are seen in Fig. 19.

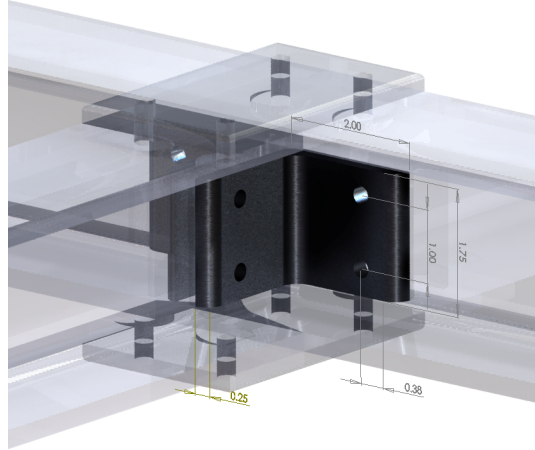


Figure 19. L-bracket design.

## 8. Part 10003: L-Bracket Analysis

A conservative approach was used to analyze the L-brackets. As shown in Fig. 20, the shear observed at each side of the of the I-beam assembly was supported by the single hole closest to the top of the of the L-bracket. This hole was chosen so that the tear out analysis would take into account the smaller effective distance, shown by “a”, as opposed to the bottom hole. The moment created by the cable attachment point was then absorbed by both holes as seen by the free body diagram.

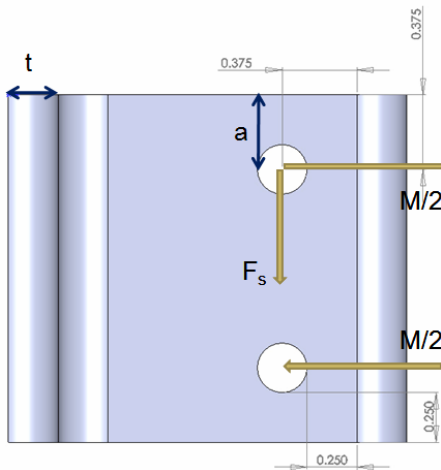
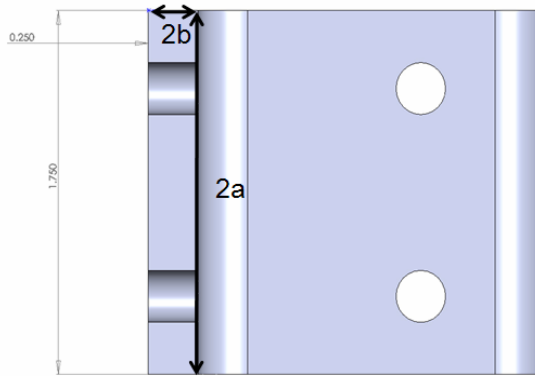


Figure 20. L-Bracket Free Body Diagram.

The L-brackets were then analyzed by bending stress, shear stress, and principle stress. Bolt tear out was also calculated for tension, shear, and bearing tear out. It was determined that

bending stress became the most designing factor. To calculate principle stress, shear stress was calculated using fig. 21 and equation below.



**Figure 21. L-bracket analysis dimensions.**

$$\tau_{\max} = \frac{T(3a + 1.8b)}{8a^2b^2} \quad (16)$$

After shear stress, the bending stress was calculated using the past equ. 5 and principle stress was then calculated using the previously stated equ. 15. The most limiting margin of safety, the principle stress, was determined to be 1.24. Bolt tear out was then analyzed and it was determined that shear tear out was the most limiting factor with a margin of safety of 1.3. Detailed part analysis can be found in Appendix A and a summary can be found in Table 4 below.

**Table 4. Summary of Margins for L-Brackets**

Margin of Safety Summary		
	Ultimate	Yield
Principle stress	1.27	2.742
Bending stress	1.245	2.485
Shear stress	13.52	23.199
0.274" dia. hole - tension tear out	2.258	3.654
0.274" dia. hole - shear tear out	1.292	2.82
0.274" dia. hole - bearing tear out	1.474	2.078
0.274" dia. hole - tension tear out (NASA)	1.575	2.679
0.274" dia. hole - bearing tear out (NASA)	2.803	3.729

## 9. Part 10003: L-Bracket Manufacturing

The L-bracket was manufactured from a pre-cut aluminum 6061-T6 L-bracket. The L-brackets were then cut to specification on a mill and deburred to remove sharp edges. The aluminum piece was then chromate conversion coated as per general requirements. Only 2 holes were drilled into each L-bracket when initially manufactured. This is due to the fact that the main I-beam assembly, L-brackets, and cover plates were match drilled. This process will be discussed further into the report.

## **10. Assembly 10006: Front Cable Assembly**

### **Research and Learning about Cable Assemblies:**

#### **a. Defining a RFP for the Cable Assembly Manufacturer**

A design concept is the first part of the cable assembly preliminary design process. Knowledge about cable assembly types, analysis, and trade studies drives the concept. The internet, David Esposito, and engineers at Northrop Grumman are great resources.

David Esposito, the senior project advisor, explained the types of cable assemblies, challenged the students to brainstorm ideas for their own design concepts, and taught preliminary cable analysis. He personally likes the look and function of swaged cable assemblies with forks on the end. Mr. Esposito spent an hour explaining the analysis, technical terminology, and various materials used in cable manufacturing. The lecture provided a foundation from which the design team could effectively understand literature about cables and manufacturer catalogs.

After a trade study, swaged cable assemblies became the design concept. A trade study with swaged cable assemblies and cable assemblies with locking bolts determined that the swaged cable assembly is a better choice. Trades showed that the swaged cable assembly met the strength, clearance, and looked better than the bolted cable assembly. However, testing requirements did not get traded and the design choice cost Cal Poly.

After the rear cable assembly had been purchased, new testing requirements for the cables were released to Cal Poly. Swaged cable assemblies had to be NDI tested per Kennedy Space Center requirements. The design team learned that the swaging process can cause cable swage fittings to crack microscopically. Unfortunately, one of the three original cable assemblies was found to have a micro surface crack. Even though the cable assembly had passed the proof test, a new cable assembly or bar had to be quickly designed and manufactured.

With a clear understanding of a conceptual cable design, Cal Poly did not have facilities to manufacture cable assemblies. Cal Poly chose a company to build and proof test all of the cable assemblies.

#### **b. Choosing a Manufacturer**

Many companies build cable assemblies for a broad range of customers and Cal Poly is looking for a company that supplies aircraft grade hardware. The manufacturing capabilities of various companies that produce cable assemblies drives the design process.

A design is worthless without the ability to make it a reality. Northrop Grumman and David Esposito suggested various aerospace cable manufacturers. Goggle and Yahoo searches yielded other potential cable manufacturers. The design team contacted each company with the intention of understanding what materials, cable fittings, and machining capabilities are available. From the research, the design team built lists of products easily found and built relationships with manufacturers.

Designing with common parts, and materials decreases manufacturing time. Common parts are kept in stock bins at a manufacturer's facility. It is easier for a manufacturer to make cable assemblies with parts already in their stock. Common materials can easily be purchased or are kept on hand by the manufacturer. When specialized parts are used the manufacturing process takes more time because the manufacturer has to find, purchase, and ship the part to their



location. After calling numerous cable manufacturers, Cableco, had the best response to Cal Poly's RFP.

Cableco serves the aerospace industry and is located close to the customer. Cableco manufactures cable assemblies for the aerospace industry and is familiar with aerospace requirements. Many cable manufacturers do not conduct proof tests. Cableco is equipped for proof tests. Cableco keeps many aerospace specific parts and materials in stock. Cableco is located in Los Angeles near Northrop Grumman. The location benefits the customer because a test engineer did not need to travel long distance to the cable-testing site to supervise.

After choosing Cableco, it is important to understand their product and product specifications. Mr. Baily, a technician at Cableco, brought catalogs of cable rigging hardware, shared design ideas for the lifting cables, and provided hardware specifications. He explained that the loads published in catalogs were working loads. Mr. Baily explained the working load multiplied by a factor of five and three is respectively the breaking and yield strength of the material. However, Craig did not explain shackle safety factors correctly and it is important to double check the wholesaler with the manufacturer specifications.

### **c. Communication Mishap**

Each manufacturer of cable hardware may not use the same ultimate and yield factors that the a cable assembly fabricator reports. It is important to double-check the safety factors of hardware with the specific manufacturer because distributors many not understand the manufacturer correctly. After Northrop Grumman approved the lift sling design for manufacture, Cal Poly purchased QMH shackles from Cableco for the front cable assemblies that used other load safety factors than advertised by Cableco.

Cableco used industry standard load factors and expected that their product manufacturers used the same factors. Some manufactures did use the same load factors, but QMH did not. Cableco provided Cal Poly with information that stated the shackles had an ultimate safety factor,  $FS_U=5$  and a yield factor of safety,  $FS_Y=3$ . During a proof test the shackles failed because the manufacturer did not use the safety factors that Cableco provided. In fact,  $FS_Y=2.5$  and  $FS_U=4$ . Providing the incorrect information cost Cal Poly time and expense. This event taught the design team the importance of concise technical communication and the importance of verifying distributor published load factors with the product manufacturer load factors.

### **d. Cable System Overview:**

The cable system's purpose is to provide an interface between the front and rear bars and the I-beam structure. The front bar (10002) is connected to the short I-beam (10009) with two front Cable assemblies (10006). The rear cable assembly(10007) connects the rear bar (10001) to the long I-Beam (10008). However, the rear cable assembly failed NDI testing and the rear tension (10020) bar was designed in its place. The cable assemblies are composed of components that are driven by requirements.

### **e. Requirements for Cable System:**

The cable system design started with the loads. After the conceptual design was finished a governing free body diagram for the front and rear cables was devised. In Fig. 21 the front cable assembly loads are shown. Table 5 below the free body diagram details the variables in Fig. 21. Figure 22 is the free body diagram for the rear cable load. Table 2 defines the variables in Fig. 2.

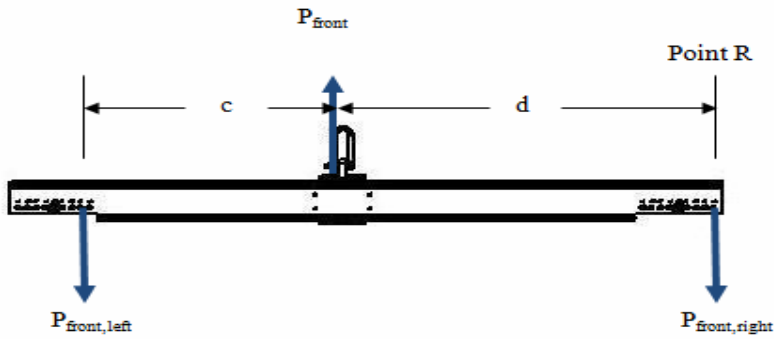


Figure 21. FBD for Front Cable Assembly Loads

Table 5. Variables for Front Cable Assembly Load Free Body Diagram

Variable	Quantity
C	14.342 in
d	20.959 in
$P_{front}$	561.05 lbs
$P_{front,right}$	228 lbs
$P_{front,left}$	332 lbs

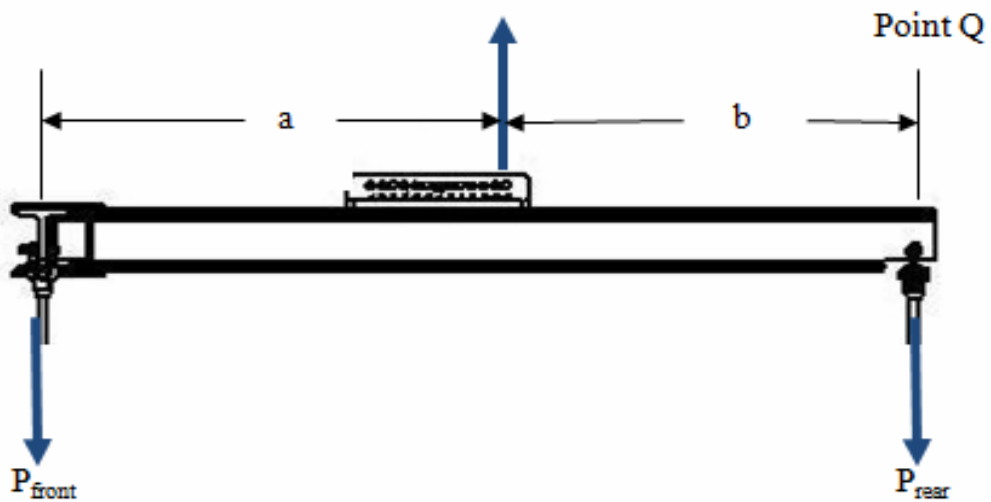


Fig 22. Governing FBD for Rear Cable Loads

**Table 6. Variables for Front Cable Assembly Load Free Body Diagram**

Variable	Quantity
A	20.537 in
b	18.063 in
$P_{top}$	900 lbs
$P_{rear}$	478 lbs
$P_{front}$	423 lbs

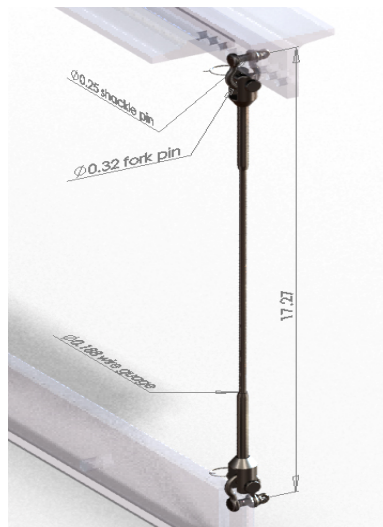
Requirements for the front and rear cable assemblies flow down from the overall lift sling requirements. Requirements include the part design, load factors, material specifications, clearance, and ergonomics.

The maximum design or working load,  $P_{front, left}$ , of a front cable assembly is defined in Table 5. The worst case front cable load is chosen for the design load because the front cables each carry different loads. The highest working from the free body diagram in Fig 21. Front cable loads so that both cable cables can support the higher of the two loads. Both cable assemblies can be interchanged; as a result, it is easier for techs to use. The design load is then modified to meet Northrop Grumman safety factor requirements.

The design load factors per Northrop Grumman require that any lifting cable less than or equal to .188” have  $FS_U=10$ ,  $FS_y=6$ , and each front cable assembly has a  $FS_{proof}=2$  before being used at any of their facilities. The rear cable assembly is different that the front cable assemblies.

The rear cable assembly has different interface and strength requirements. The maximum design or working load,  $P_d$ , of the rear cable assembly is defined in Fig. 22 F.B.D of rear cable loads. From this load the, protoflight and qualification loads of the cable differ from the other assembly because it has a diameter that is greater than .188”. The highest working load from the free body diagram in Fig. 22 specifies the rear cable design load  $P_{rear}$ . The design load factors per Northrop Grumman require that any lifting cable with a  $d>.188$ ” have  $FS_U=5$ , and  $FS_y=3$ . Each cable assembly front cable assembly has a  $FS_{proof}=2$  before being used at any of their facilities.

The cable assemblies are tested separate from the 10000 assembly. The load test factors are different from the overall lifting assembly’s load factors and an NDI test is required for all swaged fittings. The proof test design had to simulate the situation in which the part is used. The test design accounted for the correct loading and loads the part would experience. A Northrop Grumman quality engineer supervised the test and certified the part. The NDI test examines the swaged fittings for micro surface cracking.



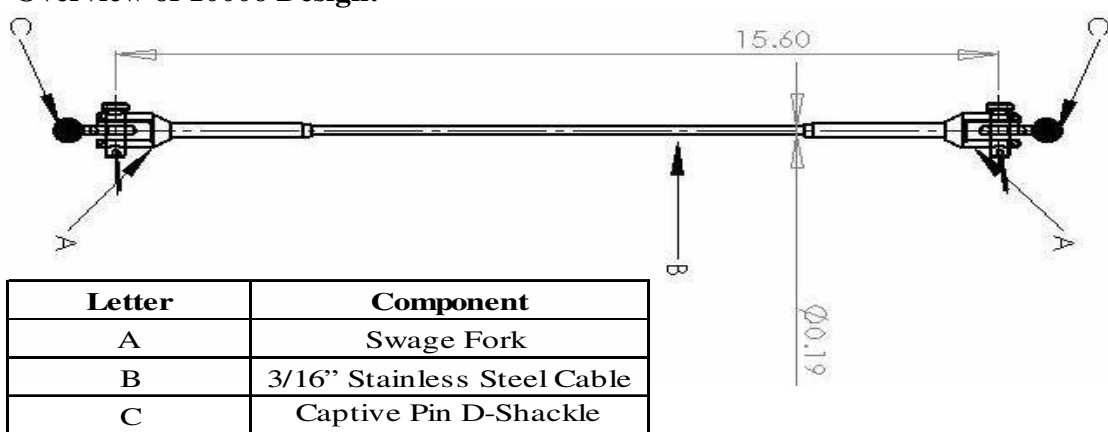
**Fig 23. Front Cable Assembly Concept**

All components of the assembly are corrosion resistant, provide required adjustability and locking features. The front cable assemblies move easily to various adjustment holes with captive pin shackles. The shackles have a threaded end pin that allows them to lock once closed. Stainless steel fittings and cables are used to meet the material requirements. Interface requirements bounded the geometry of the cable assembly.

The forks of the cable assemblies cannot contact the spreader bar or I-beam flange if the assembly sways 15° from the vertical position. Northrop Grumman specified that the hoist may pick up the MHTEX experiment off balance; as a result, the experiment could sway. Damage could be done to the interior of the forks and interference of parts is not professional.

Identification of parts and securing pins used for adjustment are required by Northrop Grumman. Loose parts can fall and damage flight hardware. Lanyards secured all parts like pins that could fall while techs are adjusting the positions of the cables during the lift sling’s use. The parts also were identified with a tag.

**f. Overview of 10006 Design:**



**Fig 24. Drawing of Cable Assembly**

The front cable assemblies are a challenge because they interface between two other parts in the design phase. The lower cap of the I-Beam (10009), seen in Fig. 23, is machined away and

the remaining web has reamed holes. The front bar connects to the MHTEX experiment and has reamed holes for the cable assemblies to connect. The design is driven by available fittings from Cableco, clearance, interfaces of the plate and I-beam, and strength requirements.

The cable and forks selection are driven by material requirements, strength requirements, and availability. Stainless steel 3/16" and stainless steel fittings are in use to meet the corrosion resistant material specification. The material also meets strength requirements outlined in the analysis. However, Cableco did not have any swage forks for 3/16" cable with enough height to connect the forks directly into the I-beam (10009) or front plate (10002). The forks do not permit enough vertical clearance for cable sway of 15°. The calculations show that the cable forks hit the front bar and the I-beam if the MHTEX swayed 15°. An interface between the cable forks, front plate, and short I-beam require a part with more vertical clearance.

The use of anchor shackles in the cable assemblies are driven by interface, rear cable loads, customer, material, and sway requirements. The size of the reamed holes is .274" and the thicknesses of both interfaces were at most .400". The shackle pin diameter cannot exceed the dimension of the reamed holes and fit the .400" thickness with minimum clearance so the fit would be tight. Northrop Grumman requires locking features on the shackle pin; as a result, captive pin shackles are in use because the pins do not come apart from the shackles. The shackles are tethered to the cable forks and will not fall from the assembly when techs move them to different hole locations. The shackle needed to be interchangeable with the front and rear cable assemblies; as a result, the shackle is designed for the higher rear cable load.

The process of design shackle design is primarily finding hardware from a manufacturer that met all of the design drivers.

The design of the cable assemblies is proven to Northrop Grumman by analysis.

#### **g. Analysis for Component A and B From Fig. 24:**

Cables are assumed to be infinitely stiff and support tension only. The safety factors and strength data are detailed below. Parts A and B are rated by Cableco at the same strength. The swage fitting has the same material allowable as the cable. All of the cable allowable information and the load factors are in the attached the analysis in Appendix A. The design load, P, is 333 lbs from fig. 21.

#### **i. Force Calculations:**

$$R_y = (\text{Protoflight})(P) \tag{17}$$

$$R_y = 1998\text{lbs}$$

$$R_U = (\text{Qualification})(P) \tag{18}$$

$$R_U = 3330\text{lbs}$$

P is the worst case working load from the governing free body diagram in Fig. 21. Ry is the proto flight load for the cable assembly. Ru is qualification load for the cable assemblies.

**ii. Margins of Safety for Front Cable Assembly:**

$$MS = \frac{F}{R} - 1 \quad (19)$$

MS is the margin of safety. F is either the yield material allowable or the ultimate material allowable. The load R is the yield or ultimate working load.

**h. Results of Analysis of 10006 Components A and B:**

<b>Table 7. Front Cable Assembly Loads and Margin of Safety Summary</b>		
	<b>Ultimate</b>	<b>Yield</b>
<b>Minimum required load</b>	3320 lbs	1992 lbs
<b>Strength of 3/16" Cable</b>	3700 lbs	2220 lbs
<b>Margin of Safety</b>	.11	.11

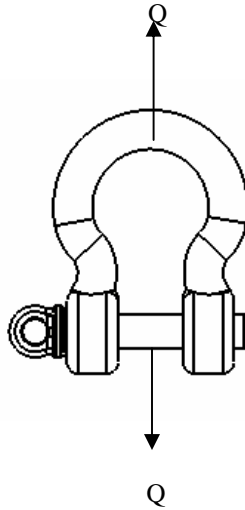
Since the margins of safety for both yield and ultimate are greater than zero, the design is shown to meet strength requirements.

**i. Analysis for Part C-Captive Pin Shackle from Fig. 24**

**i. Design**

The design load, interface, and of the rear cable assembly drove the design of the shackle. The shackle needed to be interchangeable with the front and rear cable assemblies; as a result, the shackle is designed for the higher rear cable load. The material specification drove the use of a marine grade shackle because of the corrosion resistance. The clearance and interface dimensions drove the selection of a Captive pin anchor shackle because it had the needed vertical clearance and dimension

**ii. Analysis:**



**Fig. 25 Anchor Shackle Free Body Diagram**

$$R_y = (\text{Protoflight})(Q)$$

$$R_y = 1437\text{lbs} \tag{20}$$

$$R_u = (\text{Qualification})(Q)$$

$$R_u = 2395\text{lbs} \tag{21}$$

Q is the worst case working load from the governing free body diagram in Fig. 25.  $R_y$  is the proto flight load for the cable assembly.  $R_u$  is qualification load for the cable assemblies.

**j. Margins of Safety for Part C-Anchor Shackle:**

**Table 8. Margins of Safety for the Anchor Shackle**

<b>Load</b>	<b>Margin</b>
$MS_y$	.08
$MS_u$	.93

$MS_y$  and  $MS_u$  are respectively the yield margin of safety and ultimate margin of safety. The results of the analysis are in Table 4. The margins are positive and the design is proven to meet strength requirements by analytical methods. The margins of safety are calculated using Equ. 19 from the front cable assembly analysis.

## 11. Assembly 10007: Rear Cable Assembly

### a. Design

The rear cable assembly interfaced between the rear plate and the long I-Beam. The lower cap of the I-Beam (10008), is machined away and the remaining web has reamed holes similarly to the front I-Beam. The rear bar connects to the MHTEX experiment and has reamed holes for one rear cable assembly to connect. The design is driven by available fittings from Cableco, clearance, the interfaces of the plate and I-beam, and strength requirements.

The cable and forks selection are driven by material requirements, strength requirements, and availability. Stainless steel 1/2" cable and stainless steel swage fittings were selected to meet the corrosion resistant material specification. The cable and fittings also meet strength requirements outlined in the analysis. However, Cableco's swage forks for 1/2" cable had too much fork width and not enough height to connect the forks directly into the I-beam (10009) or front plate (10002). A direct connection between the fork and the I-beam or rear bar did not permit enough vertical clearance for cable sway of 15° and the connection was loose. However, use of anchors shackles to link the cable forks to the I-beam and rear bar provided enough sway clearance shown in Fig. 26.

### b. Rear Cable NDI Test Failure and Learning Design Iteration



Fig 26. Rear Cable Assembly Before NDI Testing

Design is an iterative process and the rear cable assembly is a great example. During the testing of the cable assemblies the rear cable assembly failed an NDI test. The design team, in need of a fast solution, designed a rear tension bar to replace the rear cable assembly.

A customer requirement made the cable assembly testing more stringent. The cable assemblies had to be proof tested and NDI tested. Swaged cable assemblies are prone to fail NDI testing because of the swaging process. If the job was done sloppily then the swage fitting could



appear cracked. Both of the front cable assemblies passed the NDI test, but the rear cable assembly did not. As a result of the failed test, the design team needed a quick fix for the problem.

The design team decided to re-design the rear cable assembly into a bar with captive pin shackles on the ends to interface similarly like the rear cable. The bar design was chosen because of its simplicity. The risk of a new cable assembly failing the NDI test was high because Cableco's manufacturing process was not controlled to guarantee a clean swage. Cableco quoted a manufacturing time for a new assembly that did not fit within the delivery schedule for the 10000 assembly. The design team considered these factors and decided to build a new rear tension bar because the material was available, it could be manufactured and proof tested at Cal-Poly. However, the lack of clarity on the re-designed part led to problems when the 10000 assembly was being assembled at the Johnson Space Center.

The rear tension bar was given the same part number as the failed rear cable assembly and the use of the rear cable assembly's part number caused confusion. The rear tension bar, by mistake, was assigned the same part number as the part that it replaced. The new rear tension bar is part of the 10020 assembly. When our customer was putting together the 10020 assembly, the drawing listed the rear tension bar as 10007. The customer contacted Cal-Poly and requested documentation to prove that the new rear tension bar had not failed NDI and it was labeled incorrectly. The design team had to provide photos to the Kennedy Space center showing the cable assembly that failed the NDI test was not the rear tension bar that was delivered.

## 12. Rear Tension Bar

### a. Introduction to Part and Purpose



**Fig 27. Rear Tension Bar**

The rear tension bar interfaces the MHTeX experiment with the horizontal lift sling assembly using ¼" captive pin shackles. The bar interfaces with the adjustment holes on the Long I-Beam (10009) and the adjustment holes on the rear bar (10001). The rear tension bar shown in Fig. 27 is a quick replacement for the rear cable assembly designed for the same axial load as the Rear Cable Assembly (10006) that failed NDI testing.

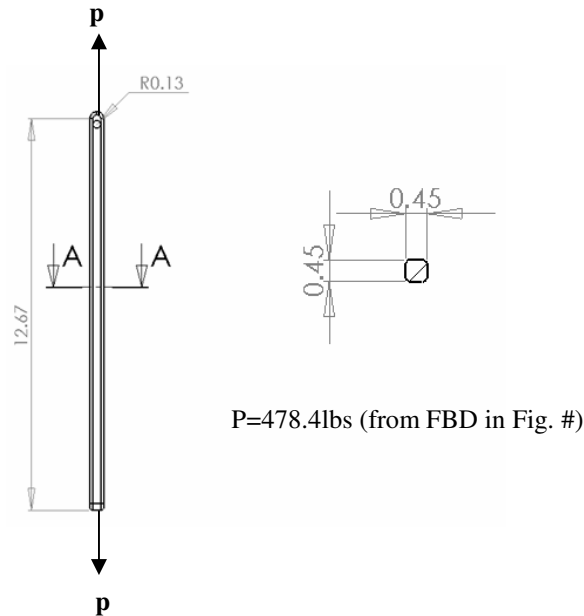
**b. Requirements**

Support the same axial tension load as the rear cable assembly. Material is corrosion resistant and corrosion resistant beam or plate material treated with chromate conversion coat. Rear bar compatible with the shackles which connect to the rear bar (10001) and the Long I-Beam (10009). Distance between the rear bar and long I-beam same as the rear cable assembly Length.

**c. Design**

Interface requirements drove the length of the bar and the hole sizes. The shackle pins are .25” in diameter and the rear tension bar holes are sized to fit the shackle pins. The length of the rear bar is driven by the distance between the rear bar and long I-beam. The bar is designed so the MHTEX will hang level when lifted with the 10000 assembly. The rear bar cross section is driven by the strength requirements. The material choice is driven by the availability and strength requirements. 7075 Aluminum is chosen because the design team had surplus and it has the necessary strength requirements.

**d. Analysis**



**Fig 28. Cross-Section Analyzed for Tension Stress**

**i. Loads from Governing FBD:**

$$P_y = (Pr\ of\ flight)(p) \tag{22}$$

$$P_y = 1436.52\ lbs$$

$$P_U = (Qualification)(p) \tag{23}$$

$$P_U = 2394.1\ lbs$$

The loads used in analysis are  $P_y$  which is the protoflight load,  $P_{ult}$  which is the qualification load, and  $p$  which is the design load from Fig. 22 free body diagram with rear cable load.  $P$  in Fig. 28 is the rear cable load from Fig. 22.

The analysis consists of two parts: axial tension load analysis and hole tear out.

### ii. Ultimate and Yield Axial Tension Analysis:

$$\sigma = \frac{P}{A} \quad (24)$$

$A$  is the area of the cross section in Fig 28,  $P$  is the  $P_{ult}$  from the load calculations. The ultimate tensile stress,  $\sigma$ , is the calculated for comparison with the ultimate tension axial stress in the for the margin of safety,  $MS_{ult}$ , in Equ. 25.

$$MS_{ult} = \frac{F_{TU}}{\sigma} - 1 \quad (25)$$

$MS_{ULT}$  is the ultimate margin of safety .  $F_{TU}$  is the ultimate material stress from the givens and material properties and  $\sigma$  is the calculated tensile stress from the ultimate load. Qualification stresses are compared to ultimate material stresses. Yield material stresses are compared with protoflight stresses.

The yield and ultimate analysis follow the same method, but instead of using the ultimate loads and stresses, one uses the yield loads and stresses.

### iii. Ultimate and Yield Hole Tear Out Analysis:

Hole loading has three failure modes: Tension failure, bearing failure, and shear failure. The tension tear out calculations are done for both yield and ultimate loads. Both analyses follow the same process and the ultimate is shown.  $d$  is the diameter of the hole analyzed,  $w$  is the width of the lug,  $t$  is the thickness of the lug, and  $P_y$  and  $P_{ult}$  are respectively the protoflight and qualification loads. The tension stress on the hole is calculated in Equ. 26.

$$F = \frac{P}{(w - d)t} \quad (26)$$

$F$  is the force from the load  $P$  on the inner surface of the lug that the fastener contacts. The load  $P$  can either be  $P_y$  or  $P_{ult}$  depending on the analysis.

Margins of safety are calculated using  $F$  as the applied force in Equ. 27.

$$MS = \frac{F_{material}}{F} - 1 \quad (27)$$

The bearing out calculations in Equ. 28 are done for ultimate and yield. The process follows similarly in for both cases.

$$F_{bru} = \frac{4P}{\pi d(t)} \quad (28)$$

$P_y$  and  $P_{ult}$  are respectively the protoflight and qualification loads. The diameter of the hole being analyzed is  $d$ , lug thickness is  $t$ . Equ.28 calculates the peak bearing stress at the center of the hole because the stress distribution is non-uniform across the hole. Using material properties and calculated values compute the margin of safety for both yield and ultimate is calculated using Equ. 19 from the cable assembly analysis. Shear failure is calculated for ultimate or qualification loads. The material properties do not list a yield shear stress allowable; therefore, a yield shear stress is pointless because there is not enough information to compute a yield margin of safety. Calculate the position of  $x$  on the hole being analyzed. The position of half the length,  $x$ , of the shear stress area.

$$x = a \left[ \sqrt{1 - \frac{r^2}{a^2} \sin^2 40} - \frac{r}{a} \cos 40 \right] \quad (29)$$

$$F_{su} = \frac{P_u}{2x t} \quad (30)$$

The outer radius on the lug is  $a$ , and  $r$  is the radius of the hole being analyzed. The shear stress calculation then uses the  $x$  value.  $P_u$  is the qualification load from the force calculations,  $t$  is the thickness of the lug, and  $F_{su}$  is the qualification shearing stress. This stress is used in the margin of safety calculation in Equ. 19 from the cable analysis.

### e. Results

The results that are for the formal analysis on the part are below in Table 9. Complete solutions are in Appendix A.

**Table 9. Analysis Margin of Safety Results for Rear Tension Bar**

Margin of Safety Summary		
	Ultimate	Yield
Actual stress	5.5	8.6
shear tear out	0.69	N/A
tension tear out	1.6	2.7

### 13. Part 10008: Long I-beam Design

The long I-beam was designed to span longitudinally across the MHTEX. This I-beam interfaced the top hook plate with the rear tension bar and short I-beam which is connected to the front cable assemblies. An I-beam was chosen to provide structural support with respect to bending and shear loads while providing a structure that can easily interface with another I-beam and fasteners. From a manufacturing perspective, the I-beam was designed to be able to provide the rear bar attachment point and provide attachment to the rest of the main I-beam assembly. The long I-beam is shown in Fig. 29 below. The I-beam cross section was chosen based on readily available I-beams carried by the metal and raw materials supplier.

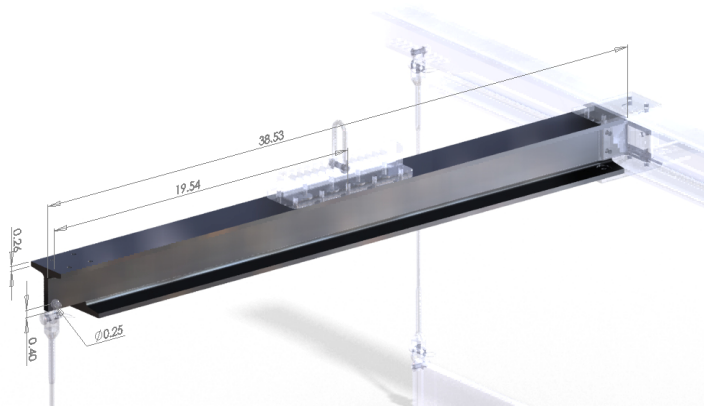


Figure 29. Long I-beam.

### 14. Part 10008: Long I-beam Analysis

The long I-beam was analyzed for bending, shear, and tear out at the worst case loading scenario at the E-13 configuration. The cross section analyzed is shown in Fig. 30 and the free body diagram for this loading is shown in Fig. 30.

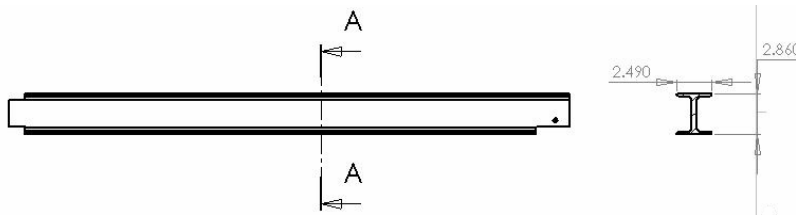


Figure 30. Long I-beam.

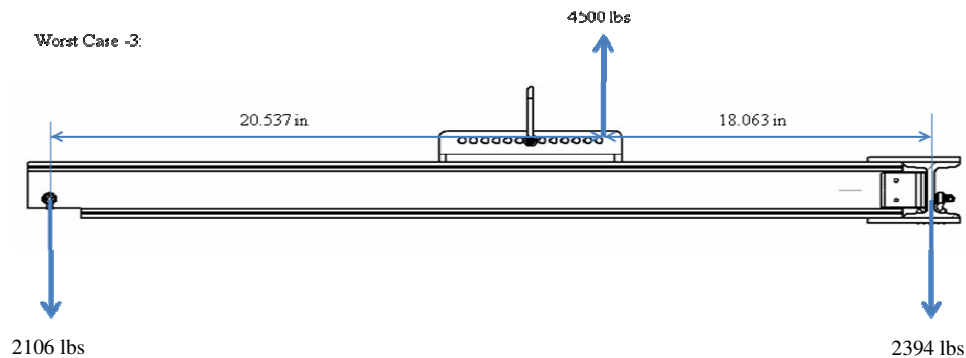


Figure 31. Long I-beam free body diagram.

It was determined that the principle stress and the flange bending created by the top hook plate were the main designing factors. The principle stress was calculated using the method previously stated and equation 2. The margin of safety for the principle stress was found to be 0.285. To calculate the flange bending due to the top hook plate, the free body diagram shown in Fig. 32 was used.

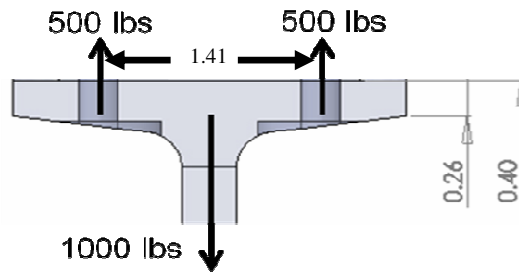


Figure 32. Long I-beam flange bending free body diagram.

The moment of inertia was calculated using a thickness of 0.26'' and a thickness value of 1.75 (the distance between the top bolts interfacing the I-beam with the top hook plate). The deflection was then calculated using the equ. 31 below.

$$\delta = \frac{PL^3}{48EI} \quad (31)$$

Using an L value of 1.41, the deflection was found to be 0.0115'' which was deemed acceptable. A summary of all strength and tear out calculations can be found in Table 10 below.

Table 10. Margin of Safety Summary for Long I-Beam

Margin of Safety Summary			Ultimate	Yield
Principle stress			0.285	1.258
Bending stress			0.298	1.112
Shear stress			6.549	11.573
0.274" dia. hole - tension tear out			6.147	9.21
0.274" dia. hole - shear tear out			0.997	2.329
0.274" dia. hole - bearing tear out			0.469	0.827
0.274" dia. hole - tension tear out (NASA)			3.556	5.508
0.274" dia. hole - bearing tear out (NASA)			3.293	4.339
Maximum Deflection (in)			0.238	0.143

## 15. Part 10008: Long I-beam Manufacturing

The long I-beam was purchased from the supplier. The I-beam was cut to length on a mill and the top and bottom faces were machined to provide a level surface. The rear cable attachment point was placed according to the nearest I-beam end. The top hook plate was match drilled to the long I-beam. The long I-beam was attached to the short I-beam through match drilling and will be discussed later in this report.

**16. Part 10009: Short I-beam Design**

The short I-beam was designed to span laterally across the MHTEX. An I-beam was chosen to provide structural support with respect to bending and shear loads while providing a structure that can easily interface with another I-beam and fasteners. The short I-beam provides lateral CG adjustability for use with the front cable assemblies. The short I-beam was designed to provide enough span to provide clearance from the cables to the MHTEX hardware in the event of the cables swinging. The short I-beam is shown in Fig. 33 below. The I-beam cross section was

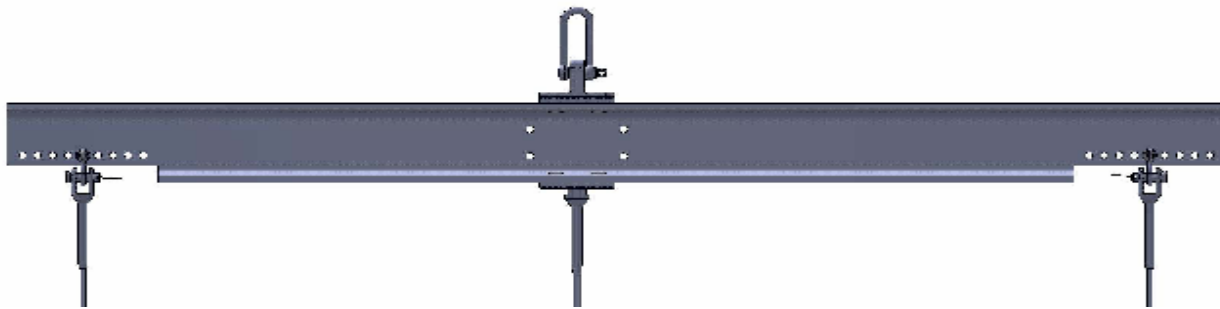


Figure 33. Margin of Safety Summary for Long I-Beam

chosen based on readily available I-beams carried by the metal and raw materials supplier. The short I-beam used to same long I-beam cross section.

**17. Part 10009: Short I-beam Analysis**

The short I-beam was analyzed for bending, shear, and tear out at the worst case loading scenario at the I-1 configuration. The cross section analyzed and loads are shown in Fig. 32.

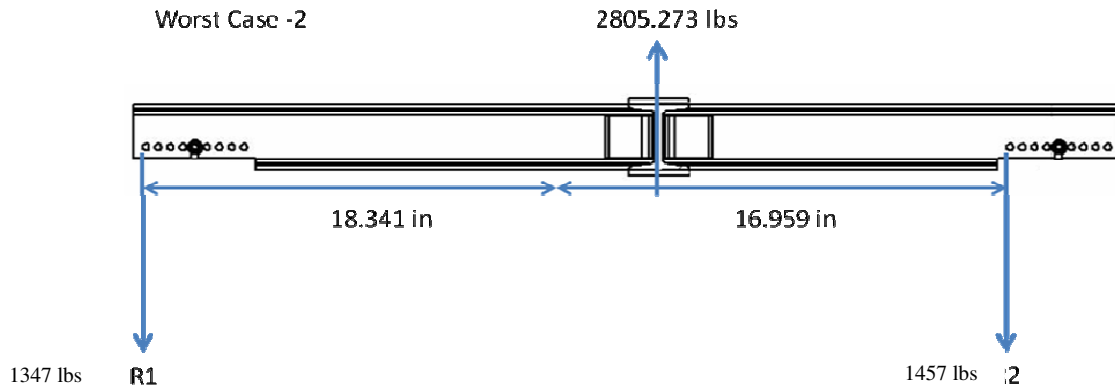


Figure 34. Short I-beam worst case loading free body diagram

Through similar analysis to the long I-beam, it was found that the bending stress was the main design factor for the short I-beam. The bending analysis technique was very similar to previous calculation and used equ. 5 from the rear bar analysis. The summary of the bending, shear, principle stress and tear out calculations can be found in Table 11.

**Table 11. Margin of Safety Summary for Short I-Beam**

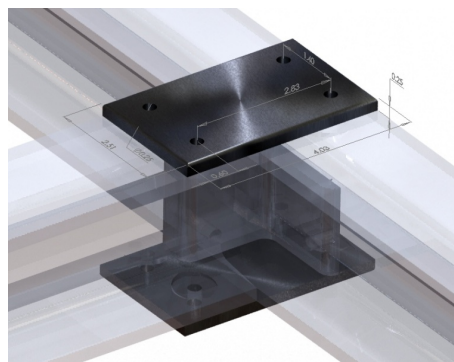
Margin of Safety Summary			Ultimate	Yield
Principle stress			1.27	2.742
Bending stress			1.245	2.485
Shear stress			13.52	23.199
0.274" dia. hole - tension tear out			2.258	3.654
0.274" dia. hole - shear tear out			1.292	2.82
0.274" dia. hole - bearing tear out			1.474	2.078
0.274" dia. hole - tension tear out (NASA)			1.575	2.679
0.274" dia. hole - bearing tear out (NASA)			2.803	3.729
Maximum Deflection (in)			0.123	0.075

**18. Part 10009: Short I-beam Manufacturing**

The short I beam was manufactured from the same I-beam acquired for the long I-beam. Similarly to the long I-beam, the I-beam was cut to length on a mill and the top and bottom faces were machined to provide a level surface. The front cable attachment points were placed according to the nearest I-beam end. The long I-beam was attached to the short I-beam through match drilling and will be discussed later in this report.

**19. Part 10010: Cover Plate Design**

The top and bottom cover plates connect to the short and long I-beams. The top cover plate is design to take the tension load due to bending and the bottom plate is design to take the compression load due to bending. The Cover plates help keep the I-beams together.



**Figure 35. Top and bottom cover plates.**



## 20. Part 10010: Cover Plate Analysis

The cover plates were analyzed to support the bending loads seen at the juncture of the long and short I-beams. The free body diagram of the cover plate is shown in Fig 36.

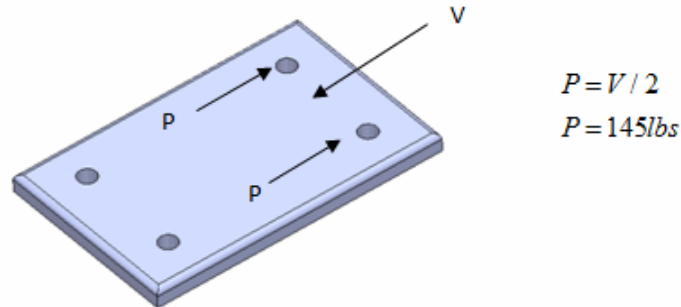


Figure 36. Cover plate free body diagram.

Bolt tear out analysis of the cover plates were conducted and resulted in high margins of safety. The tearout analysis is summarized in the front bar analysis in Equ. 7 through 10. The plate is in shear so shear analysis is done using Equ. 4 and the minimum cross sections. Detailed analysis can be found in Appendix A and a summary can be found in Table 12.

**Table 12. Margin of Safety Summary for Top Plate**

Margin of Safety Summary		
	Ultimate	Yield
0.274" dia. hole - tension tear out	5.034	7.62
0.274" dia. hole - shear tear out	2.78	3.702
0.274" dia. hole - bearing tear out	3.214	6.602
0.274" dia. hole - tension tear out (NASA)	3.77	8.82
0.274" dia. hole - bearing tear out (NASA)	4.81	6.22

## 21. Part 10010: Cover Plate Manufacturing

The manufacturing of the cover plates was relatively simple. Quarter inch aluminum 6061-T651 plate was acquired and cut to size. The edges were then deburred to break all sharp edges and the finished part was chromate conversion coated. Only two holes were drilled on the cover plates because the cover plates would then be match drilled to the main I-beam assembly.

## 22. Assembling the Main I-beam Assembly

With the I-beams, L-brackets, and cover plates individually manufactured, the main I-beam assembly could be assembled. To do so, the long and short I-beams were placed perpendicular to each other at the specified dimensions in the 10000 Assembly drawing. To ensure a perpendicular angle, a square measuring tool was used. With the I-beams correctly mounted, the I-beams were clamped into place as shown in Fig. 37.



**Figure 37. I-beam assembly clamped in place.**

With the beams clamping in place, the top cover plate was fabricated into place. Because only two holes were drilled into the cover plates originally, fasteners were used to drill through these holes into the short I-beam. To secure the assembly, two more holes were drilled directly through the cover plate and through the long I-beam. This procedure was similarly conducted on the bottom plate. The left and right L-brackets were then match drilled to the short I-beam using the pre fabricated holes. Using the last set of prefabricated holes, the right L-bracket was match drilled through the long I-beam and left L-bracket.

## **23. Part 10010: Top Hook Plate**

### **a. Purpose and Introduction**

The top hook plate (10012) provides an interface between the Long D Shackle (Wichard 1124HR) and the Long I-Beam (10008). The top hook plate (10012) is fastened by 10 bolts to the Long I-Beam. The top hook plate has reamed 5/16" holes that provide +/- 3 inches of longitudinal adjustment for the Long D Shackle.



**Figure 38. Top hook plate.**

### **b. Design**

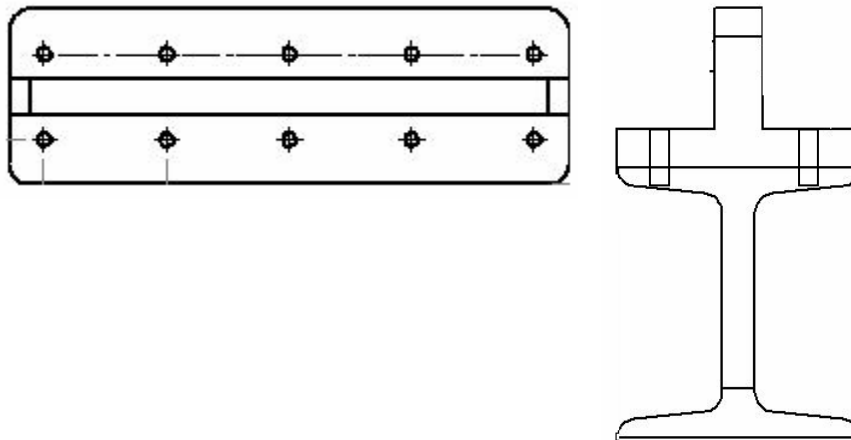
The material selection is driven by the corrosion resistant requirement and the needed strength. The thickness of the Long-D Shackle interface is driven by the shackle width. The

length of the plate is driven by the amount of stress the flange can withstand. The allowable flange stress determines how many fasteners are needed reduce the overall load. The number of fasteners and their spacing drive the length of the Top Hook Plate.

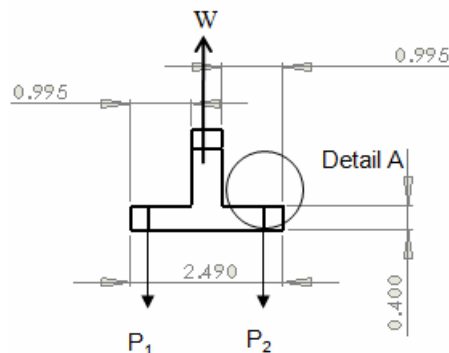
**c. Analysis**

The analysis consists of tear out calculations for the reamed 5/16” holes and computes top flange buckling on the Long I-Beam (10008) allowable. The Top Hook Plate (10012) and I-beam joint consist of two different materials joined by fasteners shown in Fig. 22. The I-Beam is made of 6061 aluminum and the Top Hook Plate is made of 7075 Aluminum. The material properties of each are different and the 6061 Aluminum is weaker than 7075. The analysis models a fastener compression load as a point load on the top cap of the Long I-Beam (10008). The analysis determines the deflection and stress of the 6061 aluminum.

**i. Interface and FBD**



**Fig 39. Top Hook plate and I-Beam Interface**



**Figure 40. Worst Case Loading Free Body Diagram**

The worst case load P is modeled from two fastener holes taking half of the main pickup load W. From Fig. 40 one can see that the design load, P, will be 450 lbs. The effective width of the cross section is calculated to see how much of the I-Beam flange is reacting to the point load P. The portion of the I-beam top cap that is reacting to the fastener load P, is modeled as a guided cantilever beam. The length L of the guided cantilever beam is the effective width of the flange,

e. The effective width  $e$  is shown in Fig. 41. The thickness of the minimum cross section is the thickness of the flange the fastener goes through.

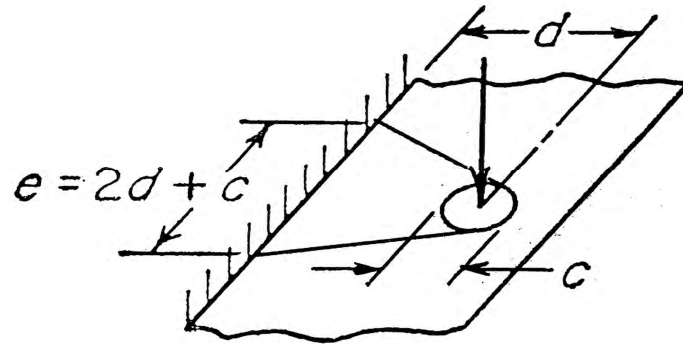


Figure 41. Effective Width

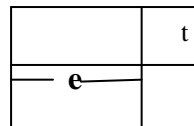


Fig 42. Cross Section of Guided Cantilever Beam.

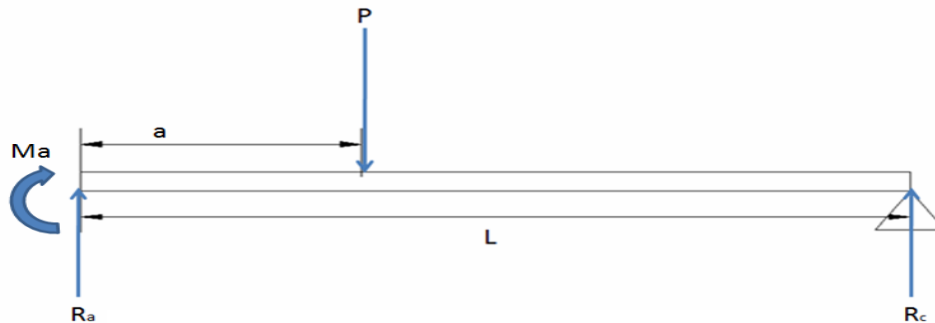
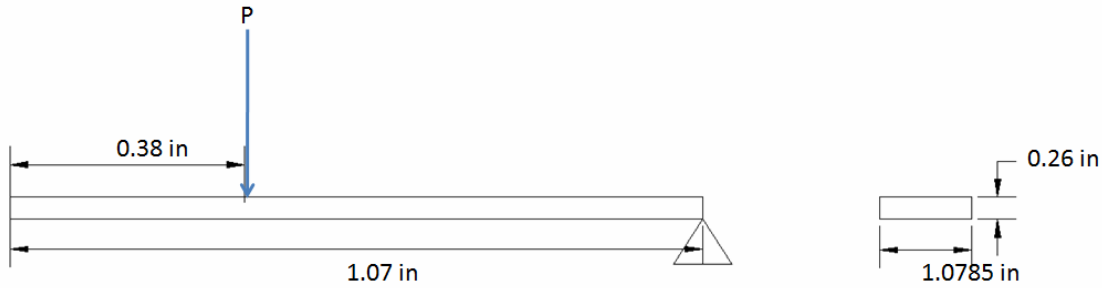


Fig 43. Guided Cantilever Beam and Cross Section of Beam

The effective  $e$ , is used as the width of the cross section in the guided cantilever beam analysis. Fig. 42 is the cross section of the guided cantilever beam. The  $t$ , is the thickness of the I-beam flange.

The general guided cantilever engineering model is in Fig 43. The model represents the deflection of half of the I-Beam top cap because the 7075 material is stiffer than the 6061 so the top cap is not going to deflect at the end. The distance from the center of the top cap to the center of the 5/16" fastener hole is  $a$ , 1 half the width of the base of the top plate, and the fixed end is the centerline along the length of the vertical shackle interface on the top hook plate. Figure 44 shows the guided cantilever model with all of the specific dimensions for the flange analysis.



**Fig 44. Guided Cantilever Beam and Cross Section Used in Formal Part Analysis**

$$R_a = P - \frac{Pa^2}{2L^3}(3L - a) \quad (31)$$

$$R_c = \frac{Pa^2}{2L^3}(3L - a) \quad (32)$$

$$M_a = \frac{Pa^2}{2L^2}(3L - a) - Pa \quad (33)$$

Looking at Fig. 44 this provides all of the specific values for each of the generalized variables in the Engineering Model of the guided cantilever beam.  $M_a$  is the moment at the fixed end of the beam described in Equ. 33. The variable  $a$  is the distance from the fixed end to the load,  $R_a$  is the reaction at the fixed end of the beam computed in Equ. 31.  $R_c$ , computed in equ. 32, is the reaction at the pinned end of the beam,  $L$  is the overall length of the beam, and  $P$  is the load applied at some point on the beam. After finding the moment and reaction forces, the deflection is computed.

$$0 \leq x \leq a$$

$$y(x) = \frac{1}{EI} = \left[ \frac{1}{2}M_a x^2 + \frac{1}{6}R_a x^3 + \frac{1}{6}P(L-a)^3 - \frac{1}{6}R_a L^3 - \frac{1}{2}M_a L \right] \quad (34)$$

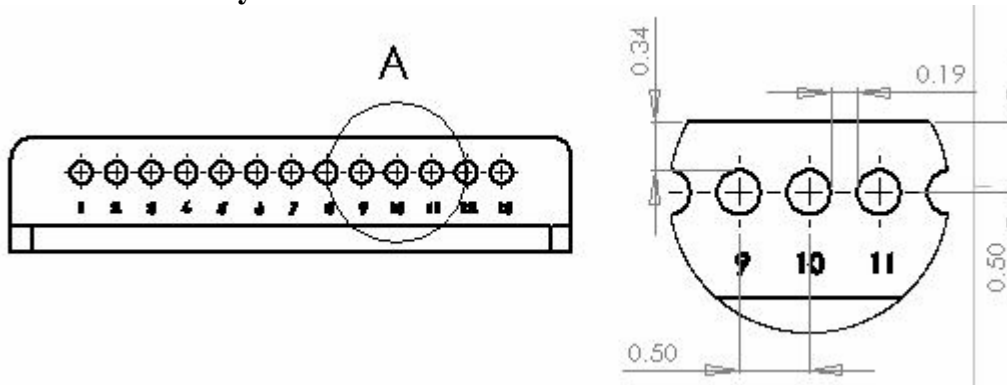
$$a \leq x \leq L$$

$$y(x) = \frac{1}{EI} = \left[ \frac{1}{2}M_a x^2 + \frac{1}{6}R_a x^3 - \frac{1}{6}P(x-a)^3 + \frac{1}{6}P(L-a)^3 + \frac{1}{6}R_a L^3 - \frac{1}{2}M_a L \right] \quad (35)$$

The deflection is  $y$ ,  $E$  is the modulus of elasticity from material properties, and  $I$  is the mass moment of inertia about the centroidal axis, and  $x$  is the position of interest. The analysis is concerned about the maximum deflection. After deflection the normal stress is calculated.

The normal stress,  $\sigma$ , is calculated in Equ. 5.  $M_{\max}$  is the maximum moment,  $c$  is the distance from the centroid to the top edge of the I-beam, and  $I$  is the area moment of inertia. The shear stress is  $\tau$ ,  $V_{\max}$  is the max shear stress,  $Q$  is the first moment of area about the neutral axis,  $t$  is the thickness of the part, and  $I$  is the mass moment of inertia in equ. 4. The principle stress is  $\sigma_p$ ,  $\sigma_x$  is the stress in the  $x$ -direction,  $\sigma_y$  is the stress in the  $y$ -direction,  $\tau$  is the shear stress and equ. 6 is used for the calculation.

**ii. Tear Out Analysis:**



**Fig 45. Top Hook Plate Interface Analyzed for Tear Out**

**1. Tear Out Loads**

$$R_y = (\text{Pr oflight})(P) \tag{36}$$

$$R_y = 3240\text{lbs}$$

$$P_U = (\text{Qualification})(P) \tag{37}$$

$$P_U = 5400\text{lbs}$$

Detail A in Fig. 45 is the cross section analyzed for tearout for the Top Hook Plate. The ultimate load is  $P_u$  and the  $P_y$  are respectively the ultimate and yield loads. These loads are calculated from the load factors and the design load,  $P$ , from the free body diagram in Fig. 40. The tearout analysis follow the same process as prior parts and is detailed in the analysis. Refer to Equ. 7 through Equ. 9 for a brief overview of tearout analysis.

**d. Results**

The results from formal analysis are published in Table 13.

**Table 13. Margin of Safety Summary for Analysis on Top Plate**

Margin of Safety Summary		
	Ultimate	Yield
Principle stress	0	0.66
Bending stress	0.066	0.777
Shear stress	1.653	3.421
0.3125" dia. hole - tension tear out	3.456	5.558
0.3125" dia. hole - shear tear out	1.766	3.61
0.3125" dia. hole - bearing tear out	1.659	2.788
Maximum Deflection (in)	0.0012	0.0007

## 24. Long D Shackle



**Fig 46. Wichard Long Shackle**

### **a. Introduction and Purpose of Part**

The Long D shackle in Fig. 46 (Wichard 1124HR) interfaces between the top hook plate (10012) and the lifting hoist hook at the company facility. The long D shackle is easily removed from the top hook plate and moved to another pin hole on the plate.

### **b. Requirements**

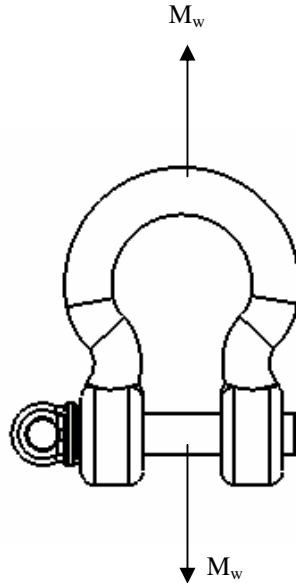
The Long D shackle requirements are defined by strength, interface, and availability. For use with a 1-1/16" hoist hook, the inside diameter of the shackle is greater than 1-1/16" for clearance. The shackle supports the entire lift assembly and MHTeX experiment; as a result, the shackle has an ultimate capability greater than 4500 lbs and a yielding capability less than 2700 lbs. The shackle is made of a corrosion resistant material. Shackle pin is secured to the shackle so it does not fall on any flight hardware and the part is marked with an identification tag.

### **c. Design**

The strength requirement is the primary driver for the shackle. Generally shackles that carried the loads are of a larger variety and already meet interface requirements. A marine

shackle is used for the corrosion resistant properties. The captive pin is standard on most marine shackles so the pin does not fall into the ocean, but this feature also met Northrop Grumman requirements.

**d. Analysis**



**Fig 47. FBD of Long D Shackle**

**i. Load Calculations**

$$P_y = (Pr\ otoflight)(M_w) \tag{38}$$

$$P_y = 3880lbs$$

$$P_U = (Qualification)(M_w) \tag{39}$$

$$P_U = 7760lbs$$

The load  $M_w = 900$  lbs (from the governing free body diagram in Fig 22). The Long D shackle only supports a tension load. In the analysis package in Appendix B, the shackle allowables are compared with the calculated tension loads  $P_u$  and  $P_y$ .  $P_u$  and  $P_y$  are respectively the qualification and protoflight loads used in the margin of safety calculations in equ.40.

$$MS = \frac{F}{R} - 1 \tag{40}$$



$MS_u$  and  $MS_y$  are respectively the ultimate and yield margin of safety.  $F_u$  and  $F_y$  are respectively the ultimate and yield loads from the material properties published in the Whichard 2003 catalog.

**e. Results**

The factors of safety published in the formal documentation in the appendix are in Table 14.

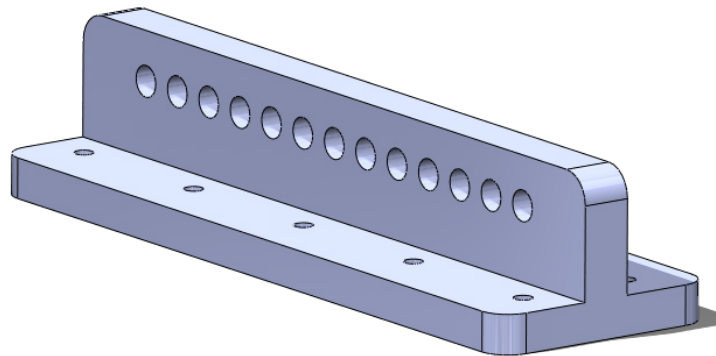
**Table 14. Tabulation of Margins of Safety**

Load	MS
Yield	0.44
Ultimate	0.72

**25. Fastener Analysis**

**a. Top Hook Plate Connection**

**i. Introduction and general purpose**



**Fig 48. Top Hook Plate Fastener Connection**

10-1/4 x 28 AN4C12A fasteners connects the top plate to the I-beam as shown in Fig. 41. These fasteners are in tension. The fasteners transfer the pickup load to the top flange of the long I-Beam assembly. The hole pattern of the fasteners allow no moment within the connection.

**ii. Requirements**

Fasteners reduce load transferred to the flange of the I-Beam to within the flange buckling allowable from the formal Top-Hook plate analysis. Fasteners are also corrosion resistant. Length of bolt shank fits through reamed hole and thread length accommodates a flat washer, lock washer, and nut with a minimum of 2 threads in length left over at the end. Fasteners certified that they meet published material specifications.

### iii. Design:

The use of aircraft and mil-spec hardware is driven by the need for specification control. Aircraft hardware dimensions and capabilities are tabulated for design. Using hardware from Home Depot is useless because the specifications are not controlled or published. Without the specifications the design team is not able to prove the capability of the fasteners to the customer.

The load capability is a requirement that is considered in the diameter and corrosion resistant material of the fastener. However, strength was not a driver. Aircraft hardware is strong and the 10000 assembly design loads are low. The driver for the diameter of the fasteners and material was availability. As with cable assemblies, the design team designed with building the assembly in mind. Fasteners that were easily purchased and quickly delivered drove the fastener diameter and material choices.

Shank and thread length selections are driven by the interface requirements and availability. The length of the shank is driven by the depth of the reamed hole between the Top Hook plate and the I-Beam top cap. The thread length is driven by the needed length to fit a flat washer, lock washer, and locknut with a minimum of two threads left exposed. However, a fastener with long thread could be selected if the exact match could not be located easily or cost effectively.

### iv. Analysis

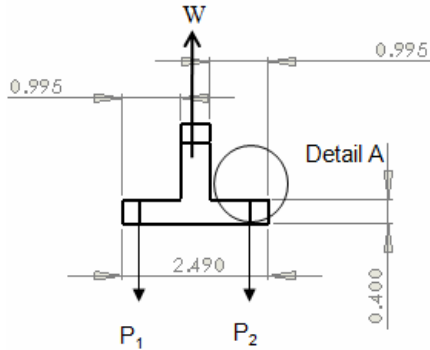


Fig 49. Worst Case Fastener Loading FBD

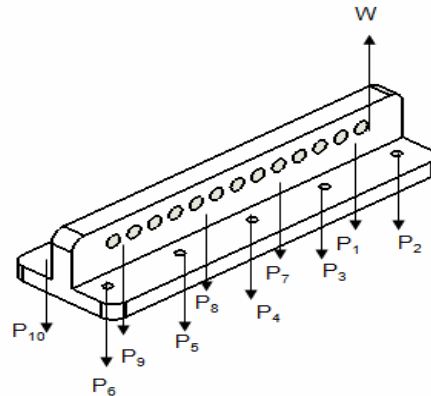


Fig 50. System Free Body Diagram

Fig 50. Shows the FBD of the entire system under load, W.  $P_n$  ( $n=1..10$ ) are reaction loads. The analysis assumes the worst case load situation in Fig. 48. The worst case loading is W distributed between two coplanar fasteners. The statics in equ. 1, 2, and 3 show the load calculations.

$$\sum M_{p_n} = 0 \quad (41)$$

$$\sum P_n = 0 \quad (42)$$

$$\frac{W}{2} = P_1 = P_2 \quad (43)$$

The FBD in Fig. 49, and the statics in equ. 41 and 42 show the total pick-up load, W, divided equally between both reaction loads P<sub>1</sub> and P<sub>2</sub>. There is no moment in the calculation because the reaction loads are symmetric and equal. The analysis for the fastener connection consists of two components: the install load and the tension loading. The install load calculates the normal and shear stresses on the I-beam flange and Top Hook Plate after the fasteners are tightened. The tension load calculations check for gapping between the I-Beam and Top Hook plate during a normal loading cycle. Gapping puts stresses on the fasteners in an unsafe manner and is disqualification criteria.

### a. Installation

Axial install tension stress is calculated in equ. 44.

$$H = \frac{P_I}{A_T} \quad (44)$$

The tension stress, H, is calculated with the tension area, A<sub>T</sub>, and P<sub>I</sub> the install load from the install torque. The A<sub>T</sub> is the tension area and it is calculated from the nominal minor diameter of the thread as cited by the manufacturer. P<sub>I</sub> is the installed load defined in the fastener properties. The shear installed stress is calculated.

$$R = \frac{1}{4} [D_p + D_i] \quad (45)$$

The shear radius R is calculated in equ. 45. D<sub>p</sub> is the pitch diameter and D<sub>i</sub> is the minor diameter. The calculated value of R is put into the shear stress calculation in equ. 46.

$$\tau = \frac{T_R(R)}{J} \quad (46)$$

The installed shear stress is, τ. T<sub>R</sub> is the fastener installation torque, and J is the torsional moment of inertia. The R value is from equ. 45. The shear and normal stresses are combined in a combined stress equ. 47.

$$C = [H^2 + 3(\tau)^2]^{1/2} \quad (47)$$

C is the total combined stress. H is the normal stress and τ is the shear stress. The margin of safety is calculated in equ. 48.

$$MS_I = \frac{F_y}{C} - 1 \quad (48)$$

$MS_I$  Margin of Safety for installation stress,  $F_u$  is the yield stress from the fastener properties, and  $C$  is the combined installation load. The tension load is calculated and the part is checked for gapping.

### b. Tension Analysis:

$$P_y = (\text{Protflight})(M_w) \quad (49)$$

$$P_y = 2250\text{lbs}$$

(From FBD in Fig. 48)

$$P_U = (\text{Qualification})(M_w) \quad (50)$$

$$P_U = 1350\text{lbs}$$

Equ. 49 calculates,  $P_y$ , which is the yield load and Equ. 50 calculates,  $P_u$ , which is the ultimate tension loads being used in the tension analysis. The tension load goes through the stiffest path. The joint stiffness governs the loads in the joint and not the bolt.

$$A_c = (\pi)((d_2 / 2)^2 - (d_o / 2)^2) \quad (51)$$

Joint stiffness is modeled by a frustrum of a cone with half of its apex angle at  $45^\circ$ . The calculation of  $d_2$ , equ. 52, is the width of the joint.  $A_c$  is the annular area that is under the pressure of the fastener clamp load. The equations are listed below. Figure 34 shows the geometry and location of the variables for the install load calculations.

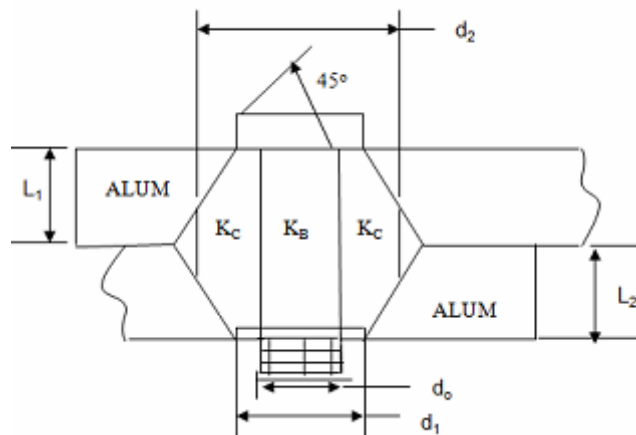


Fig 51. Joint Stiffness Diagram

$$d_2 = d_1 + \frac{1}{2}(L_1 + L_2) \quad (52)$$

The values found for  $d_2$  and  $A_c$  are used in the stiffness calculations in equ. 53 and 54.

$$K_C = \frac{(A_C)(E_C)}{L_C} \quad (53)$$

$$K_B = \frac{(A_B)(E_B)}{L_B} \quad (54)$$

$K_c$  and  $K_b$  are respectively connection stiffness and the stiffness of the bolt. For the  $K_c$  calculation:  $A_c$  is the affected area,  $E_c$  is the modulus of elasticity for aluminum, and  $L_c$  is the length of the connection. For the  $K_B$  calculation:  $A_B$  is the nominal bolt area,  $E_B$  is the modulus of the bolt material, and  $L_B$  is the length of the fastener.  $K_B$  and  $K_C$  are combined in equ. 55.

$$K = \frac{1}{1 + \frac{K_B}{K_C}} \quad (55)$$

$K$  is the fraction of the applied load carried by the connection materials. This is used in the calculation of total bolt stress from the installation and tension load during use of lift sling.

$$T_U = P_I + (1 - K)P_U \quad (56)$$

The variable  $T_U$  is the ultimate load carried by the fastener calculated in Equ. 56. The fastener carries the install load  $P_I$  and a portion of the applied load  $P_U$ . The load is transferred into a stress and combined with the shear stress component from the install load.

$$S = \frac{T_u}{A_T} \quad (57)$$

The total stress on the fastener is calculated in Equ. 57.  $A_T$  is the tension area. The tension stress is used in the combined load function. The combined stress, calculated in Equ. 58, is compared to the ultimate and yield stresses of the bolt.

$$C_U = [S^2 + 3(\tau)^2]^{1/2} \quad (58)$$

The  $MS_U$  is the ultimate margin of safety for the fastener. The stress,  $F_U$ , is the ultimate tensile stress of the bolt. The yield margin of safety is calculated in the same way.

$$MS = \frac{F}{C} - 1 \quad (59)$$

### vi. Analysis Results

**Table15. Table of Analysis Results for Top Hook Plate Fastener Analysis**

<b>Summary-Fastener Analysis: Top Hook Plate to I-Beam</b>			
	Ultimate	Yielding	Installed Stress
Axial and Torque load	-	-	64041 psi
Combined Axial Stress During Loading	64521 psi	64299 psi	-
AN4C12A Fastener Capability	125000 psi	95000 psi	-
Margin of Safety During Loading	.61	.33	-

#### b. Top Plate Connection



**Figure 52. Top plate connection.**

#### i. Introduction and general purpose

The top and bottom cover plates connect to the short (10008) and long I-beams (10009). The top cover plate is designed to take the tension load due to bending and the bottom lower plate is designed to take the compression load due to bending. Eight AN4C10A fasteners secure both the

top and bottom plates to the I-beam connection. The fasteners take the tension and compression loads in shear.

### ii. Requirements

Fasteners are corrosion resistant and meet Northrop Grumman Load safe factors. Fasteners certified that they meet published material specifications.

### iii. Design

The geometry of the fastener is driven by the interface. The interface drives the length of the unthreaded shank. The unthreaded shank is the length top plate and the I-beam flange. A maximum of one thread is allowed inside the reamed fastener hole because the threads create stress concentrations. The diameter of the bolt is driven by manufacturing tools. The loads are small; as a result, a fastener with a smaller diameter could be used. However, a smaller drill and ream bit for smaller AN fasteners are more costly. The fasteners are designed to create a stable connection.

Two fasteners are a stable design. Two fasteners equally take load; as a result, the fastener loads are lower. Lower fastener loads reduce the size of the optimal fastener. Two fasteners take out any moment and there are no torques created by eccentric loading. The connection is designed with manufacturing in mind.

The connection is designed for the fasteners to be installed with the washers and nuts on the topsides of the plates. The nuts and washers are not installed on the flange of the I-beam because it is difficult to get the torque wrench and other tools into the small space. The material specifications are driven by Northrop Grumman requirements.

The material of the bolt is driven by the material requirement. A stainless steel alloy is an option for the AN4 fasteners and is selected for this purpose. The corrosion resistant material

### iv. Analysis

The fasteners react to shear created by the moment and compression bending loads in the plates. The installed stress is the same for each connection. The install stress analysis is in the top hook plate analysis. The Free Body Diagram in Fig. 53.

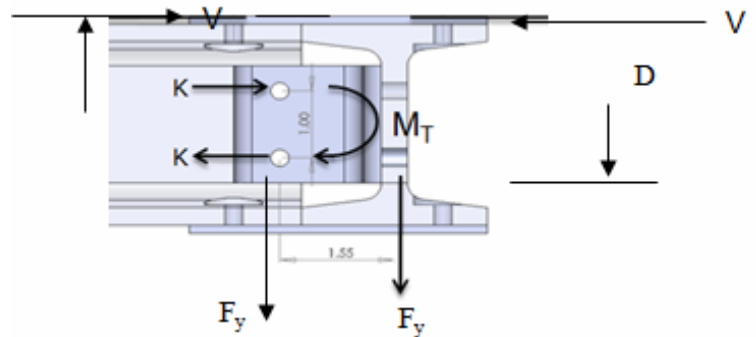
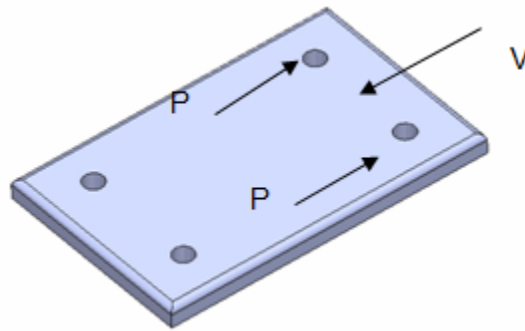


Fig 53. FBD of Top Plate Connection

Table 16. Variables for Top Plate Connection

Variable	Quantity
$F_y$	562 lbs (Worst Case Load From FBD on Pg. 7 of Analysis package in Appendix 1.)
$M_T$	871 in-lbs
$D$	3.00 in



**Figure 54. Free Body Diagram of Top Plate**

In Fig. 54 the overall connection loads are shown.  $F_y$  is the overall downward force on the connection from the centerline of the small I-beam (10008). The fasteners are located 1.55" aft of this point; as a result,  $M_T$  is the moment created. The moment is reacted out as shear in two fasteners on the top plate. Equ. 60

$$V = \frac{M_T}{D} \quad (60)$$

$$V = 290lbs$$

$M_T$  is the moment created from the eccentric loading on the top plates. The moment,  $M_T$ , is divided by  $D$  which is the distance between the two plates. The moment  $M_T$ , is reacted out the plates by the force  $V$ . The force is divided by two because there are two fasteners taking the reaction load,  $V$ . The fastener loads are both equal and referred to as  $P$ .

$$P = V / 2 \quad (61)$$

$$P = 145lbs$$

$$P_y = (Pr oflight)(M_w) \quad (62)$$

$$P_y = 435lbs$$

$$P_u = (Qualification)(M_w) \quad (63)$$

$$P_u = 725lbs$$

Equ. 62 calculates,  $P_y$ , which is the yield load and equ. 63 calculates,  $P_u$ , which is the ultimate tension load used in the tension analysis. The tension load goes through the stiffest path. The joint stiffness governs the loads in the joint and not the bolt. The maximum shear stress is calculated using Equ.4. Shear stress,  $\tau_{max}$ , is the maximum shear stress in a solid round bar.  $A$  is the nominal cross-section area of the bolt. The  $V$  is the shear load.  $P_u$  or  $P_y$  can be substituted in



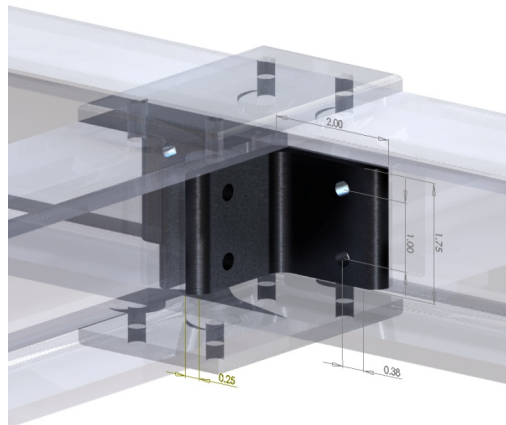
for the shear load in Equ. 4. There is no tension stress. The combined stress calculation is simplified because there is no tension stress and the value for shear stress is put into Equ. 6. The combined stress is compared with ultimate or the yield load margin of safety using Equ. 19.

**v. Analysis Results**

**Table 17. Analysis Summary for Top and Bottom Plates of Connection**

<b>Analysis for Use of AN4C10A in Top and Bottom Plates</b>			
	Ultimate	Yielding	Installed Stress
Combined Axial Stress	25801 psi	15378 psi	64041 psi
Fastener Stress Capability	125000psi	95000 psi	-
Margin of Safety when loaded	3.85	5.24	-

**c. L-brackets**



**Figure 55. L-Bracket Connection**

**i. Introduction and Purpose**

The two L-brackets connect the long I-beam to short I-beam. Each L-bracket is designed to transfer half of the shear load and bending moment caused by the pick up load to the fasteners. Two AN4C13A fasteners secure the connection to the short I-Beam (10009) and the long I-Beam (10008).

**ii. Requirements**

Fastener requirements are set by the interface loads and customer required properties. The fasteners maintain the Northrop Grumman lift sling load safety factors detailed in the analysis appendix of the report. Fasteners are corrosion resistant and do not gaul the hardware.

The interface requirements consist of a hole size of .250” and part thickness.

**iii. Design**

The geometry of the fastener is driven by the interface requirements. The shank length is driven by the thickness of the two L-Brackets and the I-beam web thickness. The holes are reamed so there are no stress concentrations on the shank of the bolt. The bolt shank length is chosen such that there are no threads in the reamed fastener hole on the interface so there are no stress concentrations. The length of the threaded portion of the bolt is driven by the needed length to put a flat washer and locknut on the end with a minimum of one thread protruding after the nut is torqued onto the threads. The diameter of fastener is driven by the hole size.

The material selection is driven by the requirements for strength and material specifications per Northrop Grumman. Aluminum is corrosion resistant, but was not chosen for the fasteners because it could gaul the fastener holes in the 10000 assembly. Stainless steel is chosen because it is corrosion resistant and does not gaul the hole. The calculated stresses are small and stainless steel will provided the required strength.

#### iv. Analysis

The installed stress is the same for each connection. The install stress analysis is in the top hook plate analysis. Only analysis for the applied loads are evaluated.

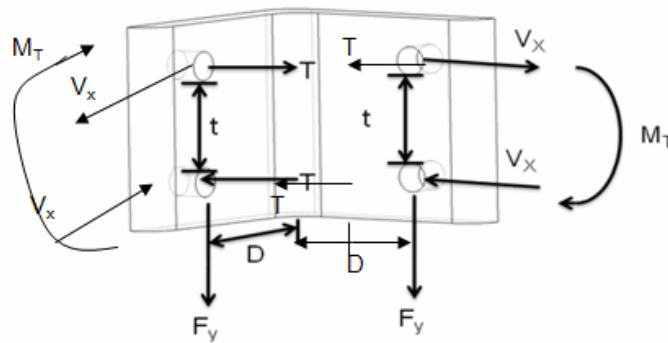


Figure 56. FBD of Fastener Loads

The loads are shown in the FBD in Fig. 56 as  $V_x$ ,  $F_y$ , and  $T$ . The moment,  $M_T$ , from Fig. 56 and is created by the eccentric load  $F_y$  derived from the front cable loads. The load  $V_x$  and  $T$  are respectively the shear and tension loads on each fastener that are reacted out from the moment,  $M_T$ .

Equation 1 shows how the total front pickup load is calculated.  $P_{Front}$  is the total pickup load,  $P_{front, right}$  is the right cable pickup load, and  $P_{front, left}$  is the left cable pickup load. The cable pickup loads are from the governing free body diagram of the 10000 assembly on Pg. 7 of the Analysis package in Appendix A. In Fig. 49. The loads used in analysis are shown in the intermediate steps following Equ. 64.  $P_{Front}$  is divided each fastener supports half of the total load,  $F_y$ , which is the fastener load in Equ. 66. The individual fastener shear is  $F_y$ . The loads and load factors are applied in the following steps. These are the loads used in the analysis.

$$P_{\text{Front}} = P_{\text{front, right}} + P_{\text{ront, left}} \quad (64)$$

$$P_{\text{Front}} = 562\text{lbs} \quad (65)$$

$$F_y = \frac{P_{\text{Front}}}{2} = 281\text{lbs} \quad (66)$$

$$P_y = (\text{Pr ot oflight})(M_y) \quad (67)$$

$$P_y = 843\text{lbs}$$

$$P_U = (\text{Qualification})(F_y) \quad (68)$$

$$P_U = 1405\text{lbs}$$

The load  $P_U$  is the ultimate load and  $P_y$  is the yield or protoflight load. Equation 67 and 68 show how the qualification or ultimate and protoflight or yield loads are calculated. Equation 69 is the calculation of  $M_T$  that is

$$M_T = (D)(F) \quad (69)$$

Equation 69 calculates the moment,  $M_T$ , created by the force,  $F$ , at a distance,  $D$ , from the fasteners. This calculation is done for both the qualification and protoflight load cases. If computing the protoflight one uses  $P_y = F$  and if one is calculating the qualification case one uses  $P_U = F$ . The distance,  $D$ , is specified in the FBD in Fig. 56. A component of shear force is created by the  $M_T$ .

$$V_x = \frac{M_T}{t} \quad (70)$$

The shear force  $V_x$ , is decoupled from the  $M_T$  in equ. 70. The distance between the centers of the fasteners is  $t$ .

$$T = V_x \quad (71)$$

The forces  $T$  and  $V_x$  are equivalent shown in Equ. 71. The shape of the l bracket causes the shear force,  $V_x$ , to be carried by the opposite fastener as the tension force,  $T$ .

$$V_{tot} = \sqrt{V^2 + F^2} \quad (72)$$

$V_{tot}$  is the vector sum of the in plane shear forces in the FBD in Fig. 56.  $V$  and  $F$  are shear forces carried by the fastener. The shear force  $V_{tot}$  is used to calculate the shear stress.

$$\tau = \frac{V_{tot}}{A_s} \quad (73)$$

Equ. 73 calculates the  $\tau$ , shear stress on the fastener. The  $V_{tot}$  is the vector sum of the shear forces and  $A_s$  is the shear area of the fastener. The tension stress,  $\sigma$ , is calculated in the same manner in Equ. 74.

$$\sigma = \frac{T}{A_T} \quad (74)$$

The  $T$  is the tension load on the fastener and  $A_T$  is the tension area of the fastener. The tension area of the fastener is from the fastener properties.

$$C = [\sigma^2 + 3\tau^2]^{1/2} \quad (75)$$

Equation 75 calculates the combined stresses from the normal stress,  $\sigma$ , and the shear stress,  $\tau$ .

$$MS = \frac{F}{C} - 1 \quad (76)$$

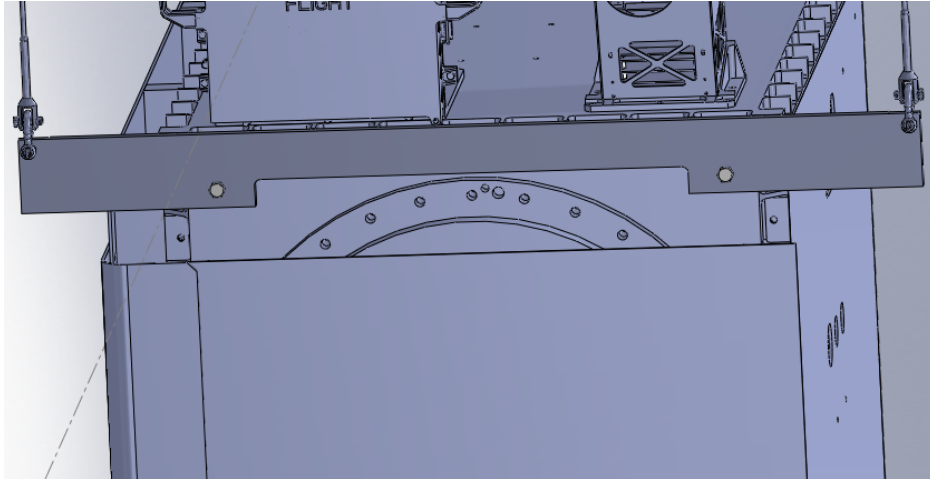
The margin of safety is calculated for tension and shear stresses in equ. 76.

## v. Results

**Table 12. Summary of AN4C13A Fastener Analysis**

<b>Summary of AN4C13A: Fastener Analysis</b>			
	<b>Ultimate</b>	<b>Yielding</b>	<b>Installed Stress</b>
Combined Axial Stress	109215 psi	78591 psi	64041 psi
Fastener Stress Capability	125000 psi	95000 psi	-
Margin of Safety when loaded	0.14	.21	-

## d. MHTEX to Rear and Front Bar Interface



**Figure 57. MHTEX Front Bar Interface**

## **Purpose and Introduction**

Two 3/8"x24 aircraft AN fasteners connect the front and rear plates to the MHTEX experiment. The fasteners thread into two keenserts on the front and rear panels of the MHTEX experiment.

## **Requirements**

Material and interface specifications are the requirements for the fasteners. The interface has a thread depth of 1.00". The thickness of the front and rear bars are .5". Northrop Grumman specifies that the fasteners are corrosion resistant and could not gaul the material. The fasteners support the Northrop Grumman load safety factors published in the analysis.

## **Design**

The interface and strength requirements drove the geometry of the fastener. The thread length of the fastener is driven by the 1.00" depth of the keensert. The length the fastener shank is driven by the thickness of the front and rear bars. The diameter of the fastener is driven by the strength requirements. The loads are small and a fastener diameter of .250" provided a good margin of safety.

The material selected is stainless steel, this choice is driven by the material requirements from Northrop Grumman. It does not gaul the aluminum and is corrosion resistant. The material properties also meet the needed strength requirements.

## Analysis

333.11 lbs

227.94 lbs

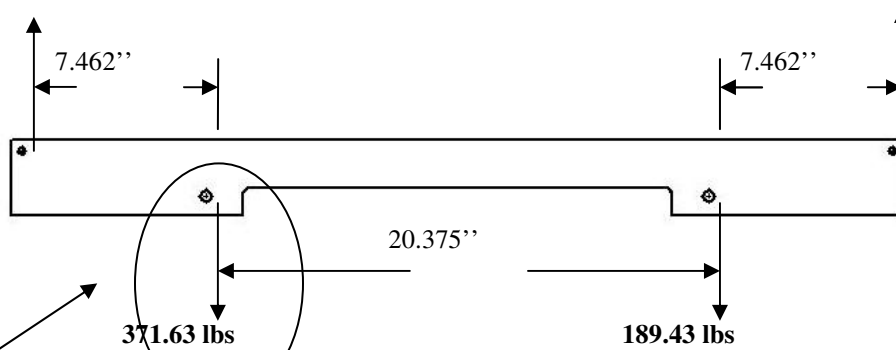


Figure 58. FBD of Worst Case 3/8" Fastener Loading From pg. 138 in Analysis in Appendix A.

Worst Case Load used in AN6 analysis.

The photoflight and ultimate loads are calculated with Equ. 67 and 78 in the L-bracket equations. The fasteners are loaded only in shear. The shear stress photoflight and qualification loads calculations are completed in the analysis using Equ. 62 and 63 from the Top Plate analysis. The install stress is the same as used for the .250" fasteners. The combined stress is calculated with equ. 72 in the top plate analysis. A yield and ultimate margin is computed. The results in the analysis package show the margins are greater than zero.

### e. Locknut



Fig. 59. Locknut FBD

**i. Requirements:**

Nuts have a locking feature to prevent loosening during repeated loading. Fasteners are corrosion resistant. The nut has the same threads and diameter as the bolt. The nut takes the worst case axial tension load.

**ii. Design:**

The decision for a locknut is driven by the locking feature requirement. The design team did not want to use lock washers as a locking feature because the washers damage the pressure surfaces on the back of the nut and on the flat washer.

A mil-spec locknut is easy to find on the central coast. A vender in Paso Robles had large quantities of the nuts in stock and the nut met the required strength.

**iii. Analysis:**

The worst case tension load is found in the L-Bracket analysis. The nut is analyzed using the same tension loads.

$P_y$  and  $P_U$  are respectively the ultimate (qualification) and yield (protoflight) loads from the L-bracket analysis. These loads are compared with the ultimate and yield capabilities of the nut.

$$P_y = (Protoflight)(P) \tag{77}$$

$$P_y = 67587 \text{ psi} \tag{78}$$

$$P_U = (Qualification)(P)$$

$$P_U = 109096 \text{ psi}$$

$$MS = \frac{F}{P} - 1 \tag{79}$$

The function in Equ. 79 is used to calculate the margin of safety for the axial tension in the bolt. F is the yield or ultimate strength of the nut from its specifications. The P can be the  $P_y$  or  $P_U$  values. The cable strength data and calculations are in the appendix A.

**Results:**

**Table 19. Load Safety Factors for Locknut**

Load	Margin of Safety
Qualification	.61
Protoflight	.33

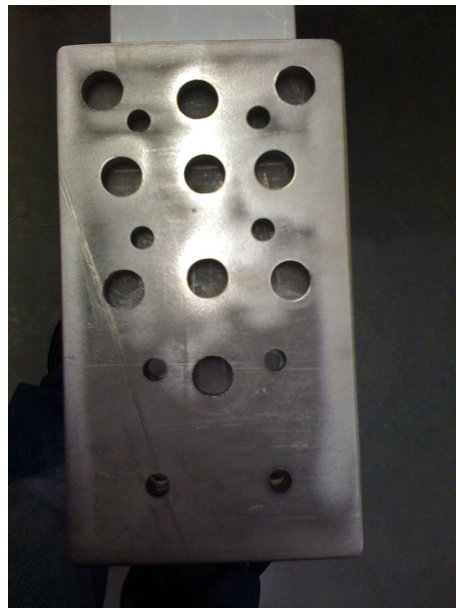
## 26. Counterbalance Plates

To balance the horizontal lift sling while attaching the GSE to flight hardware, the sling was balanced using excess scrap metal. The balancing was necessary because of the asymmetrical lift sling. The unbalanced sling can be seen in Fig. 60.



**Figure 60. Unbalanced horizontal lift sling.**

The necessary counterbalance weights were measured using a fish hook measuring tool. With the weight known, the correct moment arm was calculated from the mounting position of the plates. The necessary counterbalance plate masses were then machined such as in Fig. 61.



**Figure 61. Counterbalance weight.**



## F. Proof Testing

In order to certify the horizontal lift sling, a tensile test was designed to simulate the loading scenarios that the sling would see in operation. Typically, hardware is fully tested to prove yield, ultimate, and full deflection, but due to time constraints, the GSE was only required to satisfy a 2g proof load. The proof test was conducted at Specialty Crane Rigging with all apparatuses certified. The 10000 assembly was certified to a proof load of 2000 lbs. To do so, the lift sling was attached to a heavy dead weight and lifted by fork lift to an equivalent load of 2000 lbs measured by a certified load cell for a set amount of time. The proof test set up is shown in Fig. 62 below.



**Figure 62. Proof load set up.**

Besides load, the test simulated the attachment points on the lift sling as seen from the MHTEX and the crane hook. To simulate the 3/8" bolts securing the front and rear bars to the MHTEX, 3/8" pin shackles were used to interface with the GSE. On the top hook plate, a 9/16" crane hook shackle consistent with the required Northrop Grumman shackle was used. The attachment points can be seen in Fig. 63.



**Figure 63. Proof test attachment points.**

To simulate the different loading scenarios, the horizontal lift sling was tested in different configurations. Three configurations were tested to represent the different worst case loading

scenarios for the various critical points on the hardware. The tested configurations included the nominal loading case where the MHTEX was designed to lift majority of the time (E7), as well as the most offset attachment hole locations to represent the worst case loading for the respective front and rear components (I1, A13). The loading was increased from 0 to 2000 lbs in 20% increments while holding each load for 1 minute. The full load was held for 5 minutes. Upon satisfactory completion of the proof test, proof tags were supplied on the various pieces of the assembly. Because certain pieces could be detached, tags were supplied on the main I-beam assembly, front bar, rear bar, and top hook shackle.

Because the cables had different worst case loading scenarios at different configurations, the cable assemblies were tested separate from the 10000 assembly. The load test factors were different from the overall lifting assembly's load factors and a non destructive investigation, or NDI, was required for all swaged fittings. The proof test design simulated the situation in which each cable was used. The test design accounted for the correct loading the part would experience. A Northrop Grumman quality engineer supervised the test and certified the part. The NDI examined the swaged fittings for micro surface cracking.

### **G. Delivery**

Delivery of the 10000 assembly followed requirements of shipping box construction set by NGAS and KSC. The interior of the box was constructed to securely hold all components without the risk of damage. Zip ties and felt padding were used to secure and protect the GSE. The interior and exterior of the box were painted with a white, flame retardant paint in accordance with KSC requirements. The interior of the shipping container is shown in Fig. 64.



**Figure 64. Interior of the 10000 Assembly shipping container.**

The exterior of the shipping container was constructed in such a way that would be easy to identify and easy to transport. The container was marked “For High Bay Use” and “Cal Poly Horizontal Lift Sling Assembly 10000” to identify it. An image of the markings is shown in Fig.

65. Hinges were used to attach the cover to the box and handles were provided to allow technicians to easily move the container by hand. Large 6x8” beams were placed on the box to provide fork lift access if necessary.



Figure 65. Exterior of the 10000 Assembly shipping container.

#### IV. 20000 Assembly: Top Cover Rotation Assembly

##### A. Requirements

###### 1. Requirements for the 20000 Assembly

The 20000 assembly was the second of three GSE produced. This assembly was responsible for lifting, rotating, and support the MHTEX top cover during various phases of assembly and integration. This was separated into two main features: first, the lifting of the MHTEX top cover; and second, the rotation and support of the MHTEX top cover. The requirements for the 20000 assembly were defined from both the customer, NGAS and KSC, and internally from Cal Poly. Flight hardware interfaces, material strengths, and testing procedures all influenced the definition of these requirements.

Interfacing with the MHTEX top cover required that we did not damage any of the flight hardware and minimized obstruction to the underside of the top cover and its experiments. To interface with the top cover NGAS provided the Cal Poly design team with four 1/4”-28 holes located on the front and rear corners of the MHTEX top cover. The 20000 assembly configured for horizontal lifting is seen in Fig 4. In addition to interfacing with the MHTEX top cover, the 20000 assembly was required to interface with the 10000 assembly provided by the Cal Poly design team, for horizontal lifting, as well as be capable of providing hand holds for personnel to lift and transport the MHTEX top cover.

Interfaces requirements were defined for the 20000 assembly configured for rotation and support of the MHTEX top cover. Again, interfacing with the MHTEX required that we did not damage any of the flight hardware and minimized obstructions. To interface with the MHTEX enclosure NGAS allowed the Cal Poly design team to utilize four 1/4”-28 holes located on the front and rear sides of the MHTEX enclosure. It was required that tolerances be meet on the

interfacing holes to prevent damage to the MHTEX. The 20000 assembly configures for rotation and support of the MHTEX top cover is seen in Fig 4.

In addition to the rotation of the MHTEX top cover the 20000 assembly had to support the top cover during approximately 90 degrees of rotation and prevent any possibility of over rotation. The 20000 assembly was required to minimize loose hardware, including washers, bushings, pins, nuts, bolts, etc. All loose pins were required to be secured to the assembly to prevent loss of parts and damage to the MHTEX.

NGAS provided the Cal Poly design team with proof, yield, and ultimate load factors. A summary of the load factors for the 20000 assembly is listed below in Table 20. It was required that the GSE hold a 50 lb limit load to account for the MHTEX top cover.

**Table 20. Load Factors used for the 20000 assembly**

<b>Load Factors</b>	<b>Lifting/hoisting Devices</b>	<b>Mechanical Rotating Devices</b>
Proof load factor	2.00	2.00
Design yield load factors	3.00	3.00
Design ultimate load factors	5.00	5.00

For use by NGAS and KSC all fixtures and devices were required to be manufactured from non-corrosive materials to protect the facilities and their cleanliness. In addition all fixtures and parts were required to be tested and certified under static loading at proof load factors. NGAS quality assurance personnel were required to be present at each proof test. Testing procedures in terms of correct loading magnitudes, time, and set up were approved by the quality assurance personnel. Upon completion of each test the tested part was tagged with the static load amount and time applied.

The GSE provided to NGAS was required to be shipped in adequate shipping container. All shipping containers were required to be coated with flame retardant paint, in either a white or grey color, and be clearly labeled with a part number. The 20000 assembly was required to be delivered with an EIDP containing all analysis, drawing, certifications, and instruction for use of the particular fixtures.

**2. Requirements for Part 20001: Swivel Bracket**

The requirements from the 20001 assembly were flown down to the swivel bracket. The swivel brackets interfaced the MHTEX experiment using 1/4-28 '' fasteners and to the front and rear lift bar using a 3/8'' pin.

**3. Requirements for Part 20002: Front Lift Bar and Part 20003: Rear Lift Bar**

The requirements from the 20002 assembly were flown down to the front and rear lift bars. The lift bars interfaced the MHTEX experiment using 1/4-28 '' fasteners and to the swivel bracket using a 3/8'' pin.

**4. Requirements for the 20004: Top Cover Support Bar and Part 20005: Cover Support Assembly**

The requirements for the 20004 assembly were flown down to the top cover support bar and cover support assembly. The top cover support bar and cover support assembly supported the MHTEX top plate in a propped up position for ease of access. It lifted the top plate to a minimum of 40° and a maximum of 80° to prevent over rotation.

## **5. Requirements for the 20006: Rod End Bearing**

The requirements for the 20006 assembly were flown down to the Rod End Bearing. The rod end bearing interfaces the T-handle pin and support bar and interfaces the support bar and the MHTEX body on one side and the support bar and the MHTEX top plate on the other. These rod end bearings also are used to interface the 10000 assembly and the ExPA sim plate for horizontal lifting.

## **6. Requirements for part 20008: Swivel Bracket Fasteners**

The requirements from the 20008 assembly were flown down to the swivel bracket fasteners. These fasteners were used to interface the swivel brackets with the MHTE enclosure. In addition these 1/4"-28 bolts were responsible for ensuring the top cover plate did not over rotate past 90 degrees.

## **B. 20000 Assembly Design**

The design of the 20000 assembly was influenced by requirements from the customers, NGAS and KSC, and internally, from the Cal Poly design team. The design of the 20000 assembly served two purposes: first to lift the MHTEX top cover vertically for integration with the MHTEX enclosure; and second to rotate the MHTEX top cover during integration and testing for ease of access to the underside of the MHTEX top cover.

The first configuration of the 20000 assembly was designed for the vertical lifting of the MHTEX top cover. It was originally discussed with NGAS that vertical lifting be achieved by providing hardware that allowed the high bay crane to lift the top cover. The Cal Poly design team decided to create hardware that allowed the 10000 assembly to interface with the MHTEX top cover. There were difficulties early in the design of the vertical lifting configuration with regard to how the Cal Poly hardware would interface with the top cover. It was discussed with NGAS and was resolved with the placement of four 1/4"-28 boss holes in the MHTEX top cover, allowing the Cal Poly design team to easily interface with the top cover. The 20000 assembly was designed in addition to be capable of lifting the top cover with the aid of two or more personnel in the event that the high bay crane could not be used. To allow for hand lifting the front and rear top cover lift bars, part numbers 20002 and 20003 respectively, were given extensions that were shaped to prevent sharp edges.

The second configuration of the 20000 assembly was designed for the rotation of the MHTEX top cover. It was discussed with NGAS that the 20000 assembly be capable of rotating and holding the top cover for access to the underside of the MHTEX top cover. The front and rear top cover lift bars were utilized to support the MHTEX top cover during rotation. The two swivel brackets, part number 20001, interfaced with the MHTEX enclosure by four 1/4"-28 flight hardware holes with one swivel bracket interfacing with two holes on each side of the enclosure. The swivel brackets were fastened to the MHTEX enclosure with four 1/4"-28 fasteners that were then electro polish etched down by 0.004-0.008 in to prevent damage to the flight hardware holes. The swivel bracket was sized to prevent lateral slop between the swivel bracket and the front and rear top cover lift bars. The ball lock pins used to hold the front and rear top cover lift bars to the swivel brackets were attached to their respective lift bars to prevent loss of any loose hardware. To prevent any over rotation of the MHTEX top cover the swivel brackets had additional 1/4"-28 fasteners placed to stop the front and rear top cover lift bars at approximately 90 degrees from the horizontal. The top cover support bars, part number 20005, were designed to hold the MHTEX top cover in position during integration. These supports interfaced with four

1/4"-28 flight hardware holes, two on the MHTEX enclosure and two on the MHTEX top cover. Four 1/4"-28 rod end bearings that were then electro polish etched down by 0.004-0.008 in to prevent damage to the flight hardware holes. T handle pins were chosen to allow for quick installation and removal of the support bars during integration.

The hardware designed for the 20000 assembly was manufactured from 6061-T651 aluminum. This material was chosen for the swivel brackets, the front and rear top cover lift bars, and the top cover support bars because it was relatively easy to machine, its relatively cheap cost compared with other materials, and local availability. It provided the appropriate material characteristics, was non-corrosive, and was capable of maintaining required factors of safety.

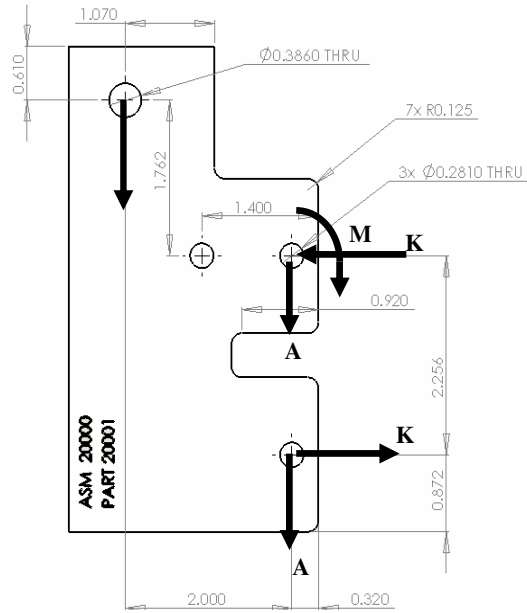
## **C. Component Design, Analysis, and Manufacturing**

### **1. Design for part 20001: Swivel Bracket**

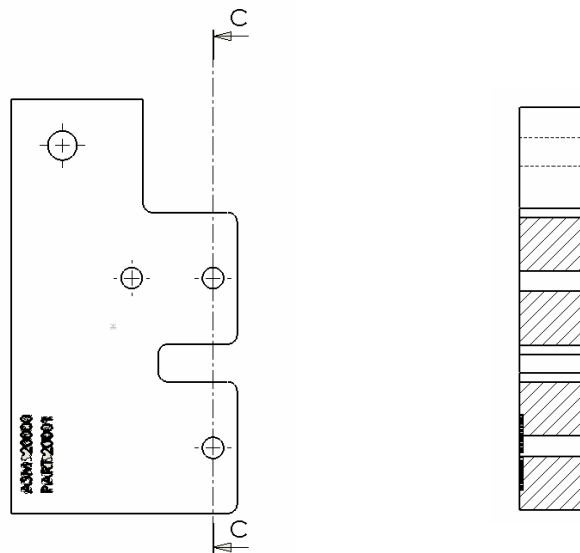
The design of Part 20001, the swivel bracket, was influenced by the requirements flown down from the customers, NGAS and KSC, and internally from the Cal Poly design team. The design team decided to originally manufacture the swivel brackets out of 0.500" 7075 aluminum plate left over from the front and rear bars of the 10000 assembly. However, because of the dimensions of the MHTEX enclosure, and the location of the interface holes along with the requirement that we minimize loose hardware such as washers, bushings, and bolts, the part would have to be significantly thicker than the 0.500" plate. The team considered stacking fastened plates to provide the required thickness but this was deemed too cumbersome on additional hardware. The team concluded that 1.000" thick 6061 aluminum plate would be purchased to manufacture the swivel brackets. This allowed the swivel bracket to interface with the enclosure and the front and rear lift bars with no more than 0.050" of clearance on each side. The swivel bracket would be designed to interface with the MHTEX enclosure at two locations to eliminate the possibility of the part torquing under load. Those holes were oversized 1/4" holes to account for positional tolerances. The hole used to interface with the front and rear lift bars were located to provide no less than 0.050" of clearance between the MHTEX top cover and the MHTEX enclosure to prevent binding of the two surfaces when rotating. The interface holes were oversized 3/8" holes to take into account positional tolerances. An oversized 1/4" hole was placed on the swivel bracket to allow for a 1/4"-28 fastener to be inserted to prevent over rotation of the top cover during integration.

### **2. Analysis for part 20001: Swivel Bracket**

The swivel bracket was analyzed for bending shear and principal stresses along with shear tension and bearing stresses in any holes. The loads were driven from the top cover plate to the swivel bracket, as show in Fig 66. Those loads were analyzed at the minimum cross section show in Fig 67. The top cover applies a load of 25 lbs into the swivel bracket resulting in a load of 111 lbs in the minimum cross section with factor of safety.



**Fig 66. FBD of Swivel Bracket Loads Transferred to Fasteners**



**Fig 67. Minimum cross section of the swivel bracket.**

All margins are calculated by dividing the ultimate allowable by the expected value and subtracting 1. Table 21 presents the results of the analysis on rod end bearings.

**Table 21: Ultimate and yield margins of safety for part 20001: swivel bracket.**

Margin of Safety Summary		
	Ultimate	Yield
Bending stress	267.4	190.7
Shear stress	73.4	N/A
Principle stress	102	65.7
0.281" dia. hole - tension tear out	90.3	128.2
0.281" dia. hole - bearing tear out	87.7	109.3
0.281" dia. hole - shear tear out	63.5	N/A

### **3. Manufacturing of part 20001: Swivel Bracket**

The swivel bracket was manufactured from 6061-T651 aluminum. This material was chosen because of its superior material characteristics. A rectangular plate was rough cut from the 1.000" thick plate and then fabricated to the specified design using a hand mill. The holes were placed using the same hand mill.

### **4. Design for part 20002: Front Lift**

The design of part 20002, the front lift bar, was influenced by the requirements flown down from the customers, NGAS and KSC, and internally from the Cal Poly design team. The design team decided to originally manufacture the front lift bar out of 0.500" 7075 aluminum plate left over from the front and rear bars of the 10000 assembly. However, because of the requirement that the bar have hand holds for personnel the bar became longer than the material we had available. The design team decided to purchase 1.000" square extruded 6061-aluminum to manufacture the front lift bar. This allowed the front lift bar to be long enough to provide handhold that extended beyond the MHTEX enclosure dimension and interface with the 10000 assembly. The front bar had two 1/4" holes located at each end of the bar to provide pick up points for the 10000 assembly. Two oversized 1/4" holes were placed 23.750" apart to interface with the MHTEX top cover and allow for positional tolerances. An oversized 3/8" hole was placed approximately 4-1/4" from the end of the bar to interface with the swivel bracket and to allow for positional tolerances.

### **5. Analysis for part 20002: Front Lift Bar**

The front lift bar was analyzed for bending shear and principal stresses along with shear tension and bearing stresses in any holes. The loads were driven from the top cover plate to the front lift bar, as shown in Fig. 68. Those loads were analyzed at the minimum cross section shown in Fig. 69. The top cover applied a load of 16.7 lbs into the front lift bar resulting in a load of 83.5 lbs in the minimum cross section with factor of safety.



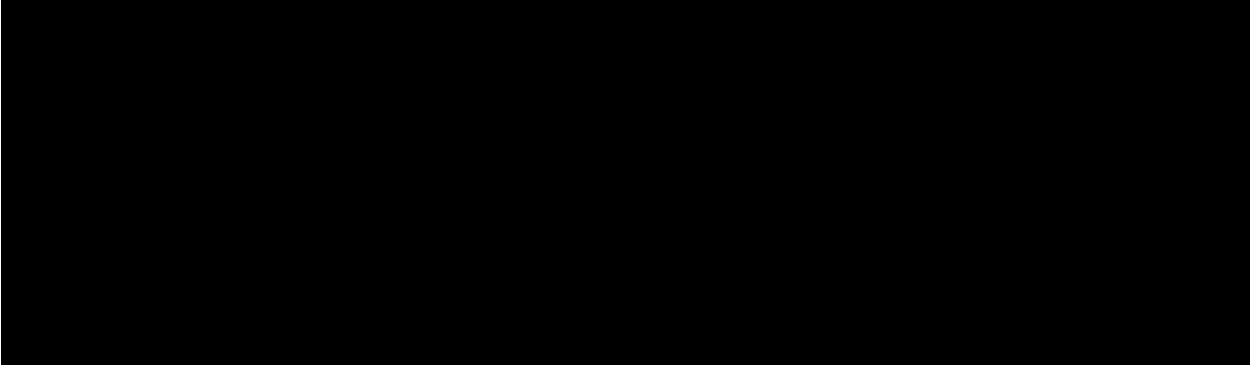


Fig. 68. Free Body diagram for the front lift bar.

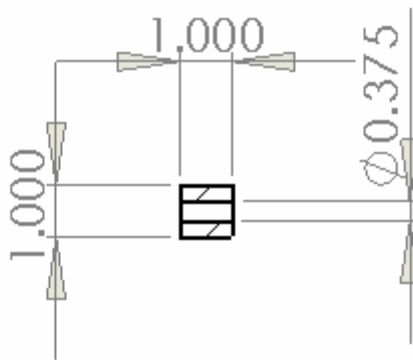


Fig 69. Minimum cross section for the front lift bar.

Table 22 presents the results of the analysis on rod end bearings.

**Table 22: Ultimate and yield margins of safety for part 20002: front lift bar.**

Bending Stress	33.25	56.08
Principle Stress	32	47.68
0.274" dia. hole - tension tear out	94	134
0.274" dia. hole - bearing tear out	67	85
0.274" dia. hole - shear tear out	65	N/A
0.274" dia. hole - bearing tear out (NASA)	104	177
0.274" dia. hole - tension tear out (NASA)	77	129
0.375" dia. hole - tension tear out	455	225
0.375" dia. hole - bearing tear out	130	162
0.375" dia. hole - shear tear out	122	N/A
0.375" dia. hole - bearing tear out (NASA)	170	284
0.375" dia. hole - tension tear out (NASA)	290	284

## 6. Manufacturing of part 20002: Front Lift Bar

The front lift bar was manufactured from 6061-T651 aluminum. This material was chosen because of its superior material characteristics. A rectangular rod was rough cut from the 1.000" square rod and then fabricated to the specified design using a hand mill. The holes were placed using the same hand mill.

## 7. Design for part 20003: Rear Lift Bar

The design of part 20003, the rear lift bar, was influenced by the requirements flow down from the customers, NGAS and KSC, and internally from the Cal Poly design team. The design team decided to originally manufacture the rear lift bar out of 0.500" 7075 aluminum plate left over from the front and rear bars of the 10000 assembly. However, because of the requirement that the bar have hand holds for personnel the bar became longer than the material we had available. The design team decided to purchase 1.000x2.000" extruded 6061-aluminum to manufacture the rear lift bar. This allowed the rear lift bar to be long enough to provide handhold that extended beyond the MHTEX enclosure dimension and interface with the 10000 assembly. The rear bar had an array of 9 oversized 1/4" holes located approximately 17" from the end of the bar to provide pick up points for the 10000 assembly and maintain level lifting with CG variation. Two oversized 1/4" holes were placed 23.750" apart to interface with the MHTEX top cover and allow for positional tolerances. An oversized 3/8" hole was placed approximately 4-1/4" from the end of the bar to interface with the swivel bracket and to allow for positional tolerances.

## 8. Analysis for part 20003: Rear Lift Bar

The rear lift bar was analyzed for bending shear and principal stresses along with shear tension and bearing stresses in any holes. The loads were driven from the top cover plate to the rear lift bar, as show in Fig 70. Those loads were analyzed at the minimum cross section show in Fig 71. The top cover applied a load of 11.2 lbs into the front lift bar resulting in a load of 56 lbs in the minimum cross section with factor of safety.

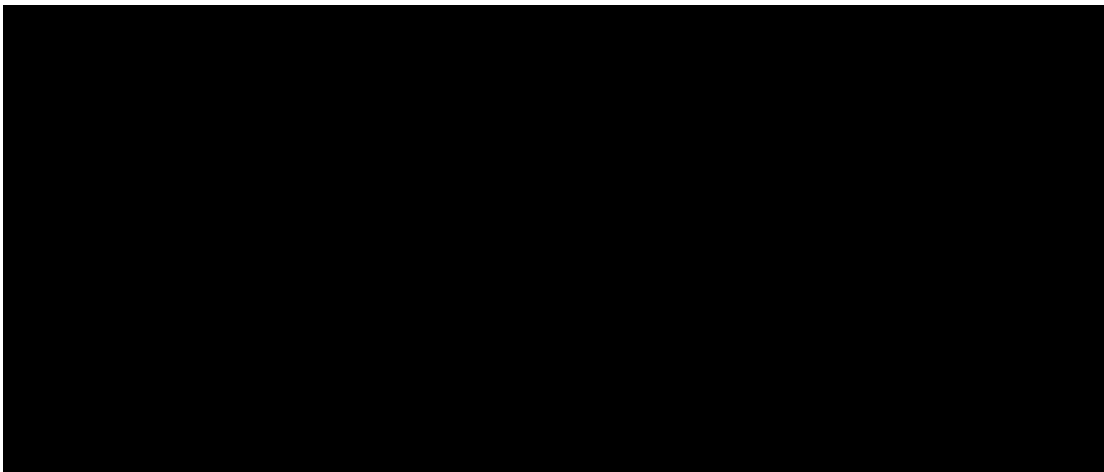


Fig. 70. Free body diagram for the rear lift bar.

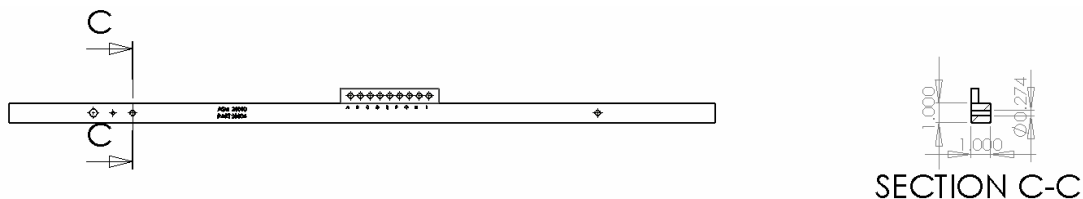


Fig. 71. Minimum cross section for the rear lift bar.

Table 23 presents the results of the analysis on rod end bearings.

**Table 23: Ultimate and yield margins of safety for part 20003: rear lift bar.**

Margin of Safety Summary		
	Ultimate	Yield
Shear stress	232.4	N/A
Bending stress	28.5	41.4
Principle stress	28.5	42.5
0.274" dia. hole - tension tear out	107	153.2
0.274" dia. hole - bearing tear out	69.4	87.6
0.274" dia. hole - shear tear out	48.5	N/A
0.274" dia. hole - bearing tear out (NASA)	70.4	87.6
0.274" dia. hole - tension tear out (NASA)	82.4	117.8

### 9. Manufacturing of part 20003: Rear Lift Bar

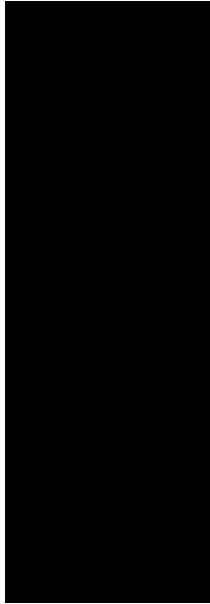
The rear lift bar was manufactured from 6061-T651 aluminum. This material was chosen because of its superior material characteristics. A rectangular rod was rough cut from the 1.000" by 2.000" rod and then fabricated to the specified design using a hand mill. The holes were placed using the same hand mill.

### 10. Design for part 20004: Top Cover Support Bar

The design of part 20004, the top cover support bar, was influenced by the requirements flow down from the customers, NGAS and KSC, and internally from the Cal Poly design team. The design team decided to manufacture the top cover support bars from 3/4" extruded 6061-T651-aluminum. This provided the proper material strengths and eases of manufacturing. The top cover support bar had 0.500" grooves cut in the top and bottom to allow for rod end bearings to be inserted for interfacing with the MHTEX enclosure and MHTEX top cover. Oversized 1/4" holes were placed 0.500" from the top and bottom of the top cover support bar to allow for a quick release T-handle pin to hold the rod end bearings in place. The edge thickness was 0.125" thick to provide proper strengths for the oversized 1/4" holes.

### 11. Analysis for part 20004: Top Cover Support Bar

The top cover support bar was analyzed for bending shear and principal stresses along with shear tension and bearing stresses in any holes. The loads were driven from the top cover plate to the top cover support bar, as show in Fig 72. Those loads were analyzed at the minimum cross section, a 0.750 square. The top cover applied a load of 25 lbs into the cover support bar resulting in a load of 62.5 lbs in the minimum cross section with factor of safety.



**Fig. 72. Free body diagram for the top cover support bar.**

Table 24 presents the results of the analysis on rod end bearings.

**Table 24: Ultimate margin of safety for part 20004: top cover support bar.**

Summary- Buckling Calculations	
	Ultimate
Compression Load	62.5 lbs
Allowable Compression	7627.51 lbs
Margin of safety	122

### **12. Manufacturing of part 20004: Top Cover Support Bar**

The top cover support bar was manufactured from 6061-T651 aluminum. This material was chosen because of its superior material characteristics. A rectangular rod was rough cut from the 0.750” square rod and then fabricated to the specified design using a hand mill. The holes were placed using the same hand mill.

### **13. Design for part 20005: Cover Support Assembly**

The design of part 20005, the top cover support assembly, was influenced by the requirements flown down from the customers, NGAS and KSC, and internally from the Cal Poly design team. The top cover support assembly was designed to allow personnel to set the angle at which the MHTEX top cover sat at. This was achieved by using rod end bearings to allow free rotation of the attachment points. Those rod end bearings were held in place with 1/4” T-handle pins to allow personnel to quickly install and remove the top cover support assembly. The T-handle pins were fastened to the top cover support bars to prevent any loose hardware from damaging the MHTEX flight hardware.

### **14. Analysis for part 20005: Cover Support Assembly**

The cover support assembly was not analyzed as an assembly. Rather, each individual part was analyzed with their loads driven through the system from the MHTEX top cover. This allowed for much simpler analysis of the overall assembly by the Cal Poly design team.

### **15. Manufacturing of part 20005: Cover Support Assembly**

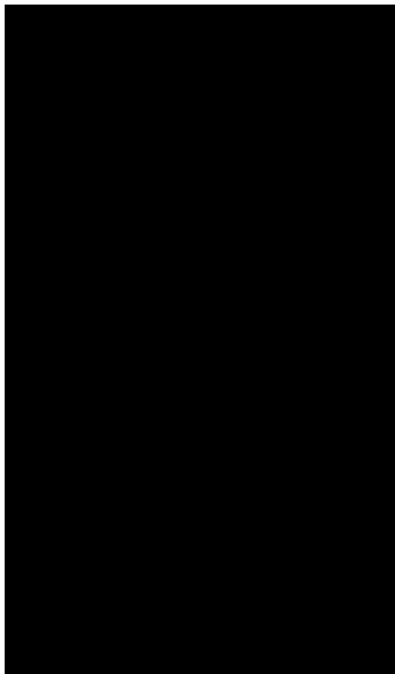
The cover support assembly was an assemblage of the top cover support bar, two rod end bearings and two T-handle pins, one of each on each end.

### **16. Design for part 20006: Rod End Bearings**

The design of part 20006, the rod end bearings, was influenced by requirements flown down from the customer, NGAS and KSC, and internally from the Cal Poly design team. The 1/4"-28 threaded rod end bearings were purchased from McMaster Carr. The rod end bearings used a PTFE fabric instead of a lubricant to prevent contamination of other materials and hardware. The rod end bearings were electro polished to reduce the thread diameter by 0.004-0.008" to ensure the rod end bearings did not damage the flight hardware holes when being installed and removed.

### **17. Analysis for part 20006: Rod End Bearings**

The rod end bearing was analyzed for tension. The loads were driven from the top cover plate to the rod end bearing, as show in Fig 73. Those loads were analyzed at the minimum cross section of the rod end bearing. The top cover applied a load of 25 lbs into the rod end bearing resulting in a load of 125 lbs in the minimum cross section with factor of safety.



**Fig. 73. Free body diagram for the rod end bearing.**

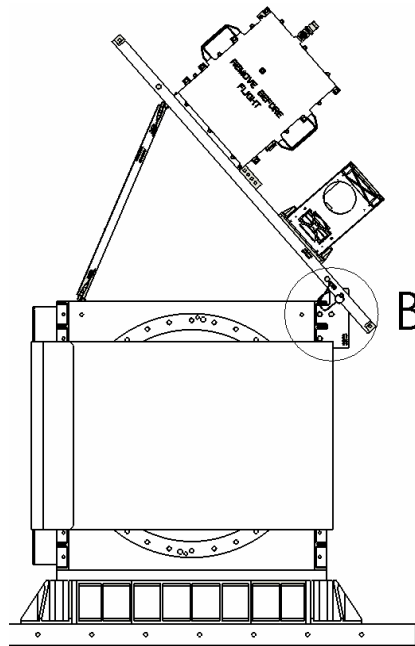
Table 25 presents the results of the analysis on rod end bearings.

**Table 25: Ultimate and Yield margins of safety for part 20006: rod end bearing.**

Margin of Safety Summary		
	Ultimate	Yield
Load Capacity	9.96	17.3

## 18. Manufacturing of part 20006: Rod End Bearings

### a. Swivel Bracket to MHTEX Fasteners



**Fig 74. Drawing of 20000 Assembly**

#### a. Purpose and Introduction

The connection analyzed is detail B in Fig. 74. Each swivel bracket is secured on MHTEX with two AN4C15A non-flight fasteners. These fasteners are electro polished so they do not damage any of the locking features in the MHTEX experiment. The fasteners have an install load and carry loads from the pivoted cover.

#### b. Requirements

The fasteners have specific interface and material requirements. Northrop Grumman used locking keenserts in the front and rear covers of the MHTEX experiment. The fasteners used in the swivel brackets are electropolished to reduce the thread diameter. The non-flight fasteners at nominal diameter would damage the locking features on the front and rear cover. The interface has a thread depth of .5" and the thickness of the L-bracket is .9" thick. The fasteners meet Northrop Grumman provided strength load factors. The material is corrosion resistant for use in Northrop Grumman and Kennedy Space Center facilities.

### c. Design

The loads are small enough that they did not drive the fastener selection. The driving factors for the fastener selections are the customer provided interface and customer material requirements.

Fasteners with a .25” diameter are chosen to fit the MHTEX cover keenserts because the swivel bracket is secured to front cover as shown in detail B in Fig 74. The material is corrosion resistant stainless steel which is driven by the customer’s corrosion resistant requirement.

### d. Analysis

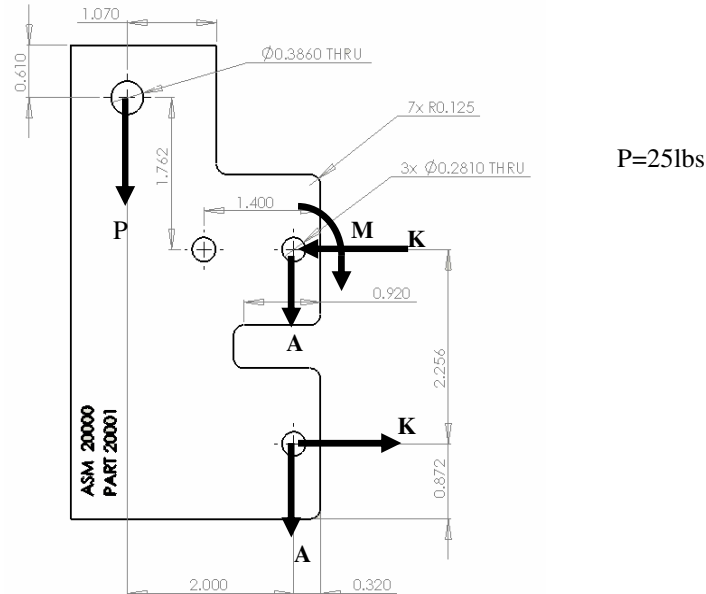


Fig 75. FBD of Swivel Bracket Loads Transferred to Fasteners

### i. Engineering Model

Worst case loading occurs when the cover lid is rotated 90 degrees and the plate’s center of gravity is in line with the center of the .3860” t-handle pin hole. The free body diagram is shown in Fig 75. The load  $P = T$  from page 7 of the 20000 assembly analysis.

### ii. Loads used in Analysis

$$P_y = (\text{Protoflight})(P) \quad (80)$$

$$P_y = 75\text{lbs}$$

$$P_U = (\text{Qualification})(P) \quad (81)$$

$$P_U = 125\text{lbs}$$

The load  $P$ , is load from the FBD in Fig 75. In equations 80 and 81 the bolt loads are calculated from the protoflight and qualification loads.

### iii. Shear Stresses from Loading

The fastener is preloaded, but all of the preloads are the same for each connection in the 10000 and 20000 assemblies. The calculation for the pre-load is in the 10000 assembly's Top Hook Plate fastener analysis. The stresses in the fastener from loading are only considered in the following analysis.

$$A = V/2 \quad (82)$$

$$K = ((V)(2.0\text{in}))/ (2.6\text{in}) \quad (83)$$

The load P causes a moment and a shear that each of the fasteners react out. The load A is a shear load as shown in the FBD in Fig 75. The load K is the shear from the moment that is reacted out by the fasteners. The load P is located 2.0" from the centerline of the fasteners and the fasteners are located 2.6" apart on center shown in Fig 75. Equ. 83 converts the moment created by the eccentric load, V, into a force couple shear system acting on the fasteners.

$$V_{tot} = \sqrt{A^2 + K^2} \quad (84)$$

The fastener shear forces are combined. The combination results in a resultant shear load,  $V_{tot}$ . Equ. 84 is the fastener shear stress calculation. Shear stress,  $\tau_{max}$ , is the maximum shear stress in a solid round bar. A is the nominal cross-section area of the bolt. V is the shear load.

#### Axial Tension due to bending:

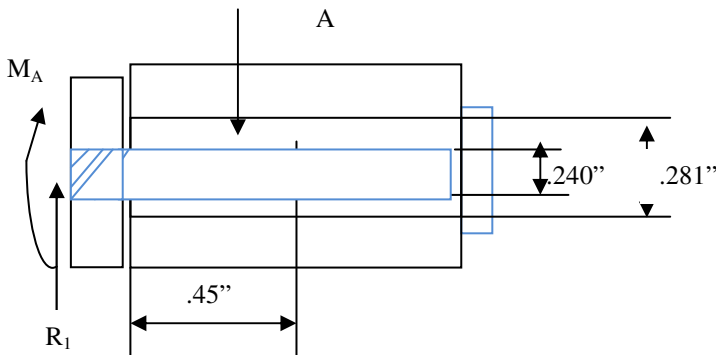


Fig 76. Free Body Diagram of Bolt During Worst Case Swivel Bracket Loading

Treat fastener like a cantilever beam with one end fixed in the nut plate. Assume the applied load A occurs when the center of the swivel bracket contacts the bolt grip.

Please reference page # 67-68 in the 20000 analysis appendix for A load values.

Refer to page # 66 in the 20000 analysis appendix for  $r_m$

The mass moment of inertia is calculated in equ. 85 for the bending calculation.

$$I = (\pi)(r_m)^4/(4) \quad (85)$$

The variable I is the mass moment of inertia,  $r_m$  is the radius of the thread pitch radius from a standard thread chart.

Equ. 86 is the moment calculation.



$$M_A = (.45'')(A) \quad (86)$$

A is the load applied from on the fastener from the cover.  $M_A$  is the moment created by the load.

$$S_y = (M_A)(c)/I \quad (87)$$

$S_y$  is the normal stress calculated from the moment on bolt. The variable c is the distance from the center of gravity or for the case of this analysis, the radius of the bolt. Equ. 88 calculates the combined stress for the fastener loading.

$$y = [S_y^2 + 3(\tau)^2]^{1/2} \quad (88)$$

The value y is the combined stress,  $S_y$  is the normal stress, and  $\tau$  is the shear stress on the fastener.

$$MS_y = \frac{F_y}{C_y} - 1 \quad (89)$$

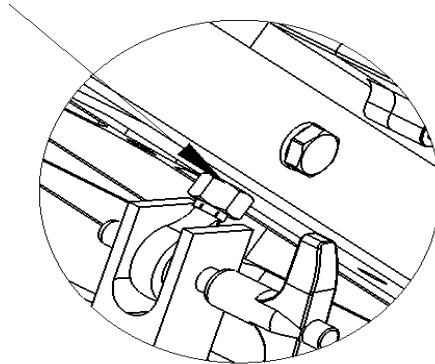
Equ. 89 is the margin of safety calculation for the combined stress. The value  $F_y$  is the ultimate or yield stress from the fastener properties. The variable  $C_y$  is the combined stress from Equ. 88.

**e. Results of analysis:**

**Table 26. Summary of Margins of Safety Swivel Bracket to MHTEX**

Summary-Fastener Analysis: Swivel Bracket to MHTEX			
	Ultimate	Yielding	Installed Stress
Axial and Torque load			64041 psi
Combined Axial Stress During Loading	23340.5 psi	22664.8 psi	
AN4C15A Fastener Capability	125000 psi	95000 psi	
Margin of Safety During Loading	4.35	4.51	

## f. Front and Rear Lift Bar to MHTEX Top Cover Boss Fastener Analysis



**Fig 77. Front and Rear Bar Fastener**

### i. Introduction and Purpose

The front and rear bar is secured on MHTEX top cover with two AN4C14A non-flight fasteners. These fasteners thread into bosses that are provided by Northrop Grumman per Cal-Poly's request. The fasteners are .250" diameter and installed with the same 61 in-lbs of torque as each of the other fasteners.

### ii. Requirements:

The fasteners maintain the Northrop Grumman load safety factors and be corrosion resistant.

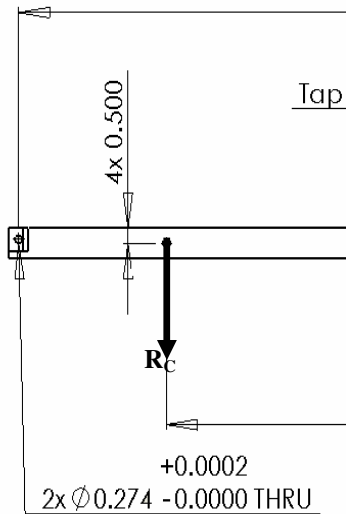
### iii. Design:

The loads are low and strength was not a driving factor in the fastener design. The interface and availability drove the design.

The interface is designed by Cal-Poly and Northrop Grumman interactively. Cal-Poly requested the interface boss holes with .250" thread. The shank length of the fastener is designed by the width of the bar. The bar hole is reamed and no threads are to interface with the reamed hole because of stress concentrations. The analysis shows that the fasteners meet the Northrop Grumman strength requirements and the material choice is driven by the corrosion resistant requirements.

### iv. Analysis:

### a. Free Body Diagram of Loading



#### Engineering Model:

Worst case load located on front bar as shown.

$R_c = 16.71$  lbs (Obtained from FBD on page #5)

Installed stress analysis is the same as the analysis in the 10000 assembly Top Plate.

Fig 78. Free Body Diagram of Worst Case AN4C14A Bolt Load

### b. Loads for Analysis

$$P_y = (\text{Protoflight})(P) \quad (90)$$

$$P_y = 50.1\text{lbs}$$

$$P_u = (\text{Qualification})(P) \quad (100)$$

$$P_u = 83.55\text{lbs}$$

The protoflight or yield load,  $V_y$ , is calculated in equ. #. The qualification or ultimate load,  $V_u$ , is calculated in equ. 90. The value qualification and Protoflight are the load factors from the previous section. The load  $P$ , is load from the FBD in Fig 78.

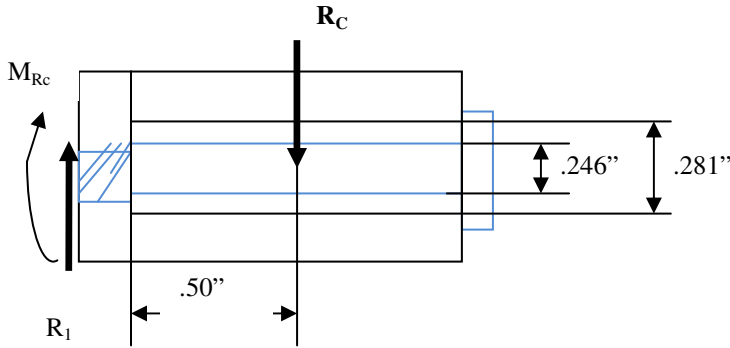
### Shear Stresses from Loading

The fastener is preloaded, but all of the preloads are the same for each connection in the 10000 and 20000 assemblies. The calculation for the pre-load is in the 10000 assembly's Top Hook Plate fastener analysis. The stresses in the fastener from loading are only considered in the following analysis.

$$A = V/2 \quad (92)$$

$$K = ((V)(2.0\text{in}))/ (2.6\text{in})$$

The load P causes a moment and a shear that each of the fasteners react out. The load A is a shear load as shown in the FBD in Fig 78. Shear stress,  $\tau_{\max}$ , is the maximum shear stress in a solid round bar. A is the nominal cross-section area of the bolt. The V is the shear load.  $P_u$  or  $P_y$  can be substituted in for the shear load in the 10000 fastener analysis.



**Treat fastener like a cantilever beam with one end fixed in the nut plate. Assume the applied load A occurs when the center of the swivel bracket contacts the bolt grip. Please reference page # 67-68 for A load values. Refer to page # 66 for  $r_m$**

**Fig 79. FBD of Top Plate and Bar Connection**

The variable I is the mass moment of inertia,  $r_m$  is the radius of the thread pitch radius from a standard thread chart.

Equ. 93 is the moment calculation.

$$M_A = (.45'')(A) \quad (93)$$

A is the load applied from on the fastener from the cover.  $M_A$  is the moment created by the load.

$$S_y = (M_A)(c)/I \quad (94)$$

$S_y$  is the normal stress calculated from the moment on bolt. The variable c is the distance from the center of gravity or for the case of this analysis, the radius of the bolt. Equ. 95 calculates the combined stress for the fastener loading.

$$y = [S_y^2 + 3(\tau)^2]^{1/2} \quad (95)$$

The value y is the combined stress,  $S_y$  is the normal stress, and  $\tau$  is the shear stress on the fastener.

$$MS_y = \frac{F_y}{C_y} - 1 \quad (96)$$

Equ. 96 is the margin of safety calculation for the combined stress. The value  $F_y$  is the ultimate or yield stress from the fastener properties. The variable  $C_y$  is the combined stress from Equ. 96.

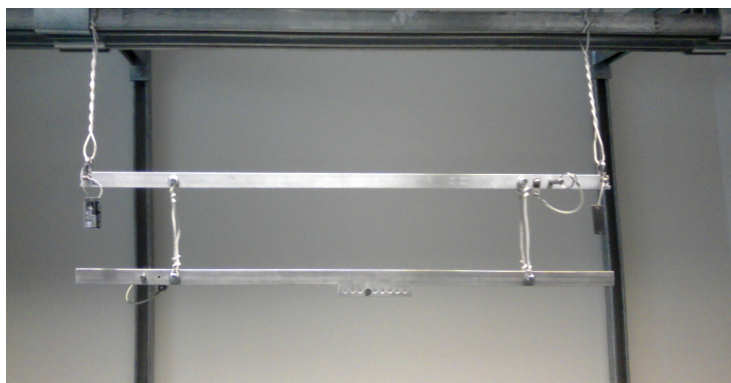
**c. Results:**

**Table 27. Summary of Front Bar to Top Cover Connection Margins**

Summary-Fastener Analysis: Front Bar to Top Cover Connection Margins			
	Ultimate	Yielding	Installed Stress
Axial and Torque load			64041 psi
Combined Axial Stress During Loading	29765 psi	16095 psi	
AN4C15A Fastener Capability	125000 psi	95000 psi	
Margin of Safety During Loading	3.5	5.1	

**D. Proof Testing**

In order to certify the 20000 assembly a tensile test was designed to simulate the loading scenarios that the various parts would see during operation. The proof test was conducted at California Polytechnic State University with all apparatuses certified by NGAS personnel. The front and rear lift bars were certified with a proof load of 50 lbs. To do so the lift sling was attached to a sturdy fixture and loaded with three bags of approximately 25 lbs of shot each and held for two minutes each up to the total approximate weight of 75 lbs and held for five minutes until released. The proof test was set up as shown in Fig. 80 below.



**Fig 80. Proof test setup for part 20002 and 20003, front and rear lift bars.**

The cover support assembly and swivel brackets were tested in the same manner as the front and rear bar with a certified load of 50 lbs. The proof test for the cover support assembly was set up as shown in Fig. 81 below.



**Fig 81. Proof test setup for part 20005, cover support assembly.**

### **E. Delivery**

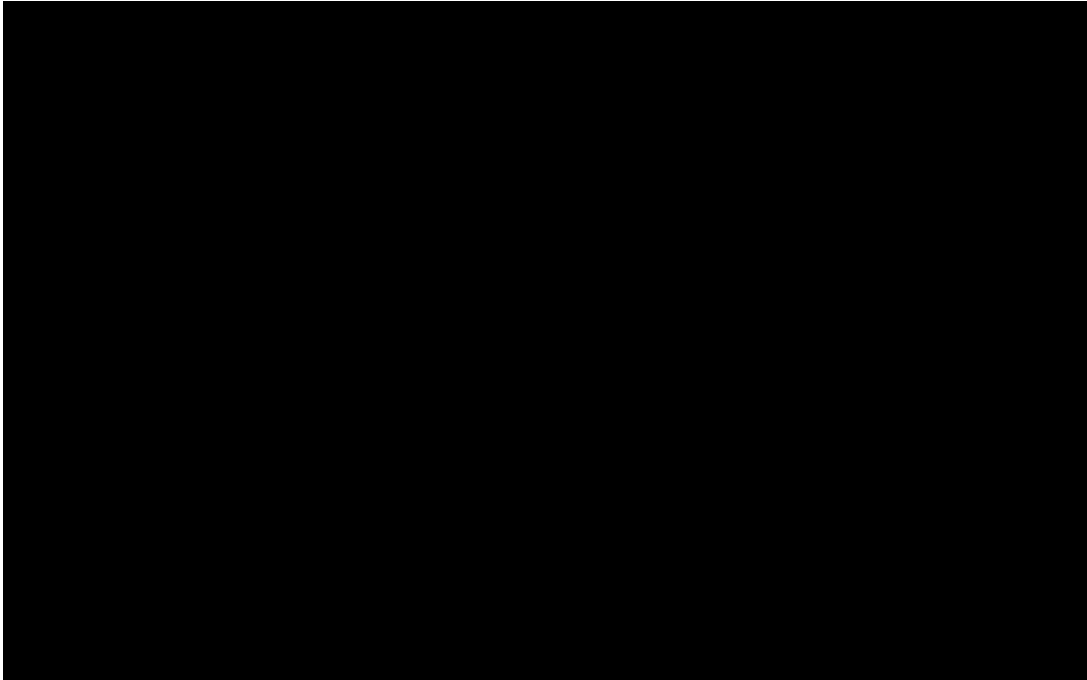
The 20000 assembly was to be shipped in a container designed and built by the Cal Poly design team per NGAS and KSC specification for high bay use. This required the design team to build a shipping container sturdy enough for pick up by a forklift and allow for personnel to handle the crate if necessary. It was also required that the shipping container be painted white or grey with flame retardant paint and be properly labeled with the assembly name and number on all sides and the lid. Zip ties were installed to prevent the parts from moving during shipping and the part holds were lined with felt to prevent damage to the GSE during delivery to NGAS.

## **V. 30000 Assembly: ExPA-Pallet Lift Assembly**

### **1. Requirements for the 30000 Assembly**

The 30000 assembly was the third of three GSE produced. This assembly was responsible for lifting the ExPA sim plate during various phases of assembly and integration. The requirements for the 30000 assembly were defined from both the customer, NGAS, and internally from Cal Poly. Flight hardware interfaces, Material strengths, and testing procedures all influenced the definition of these requirements.

Interfacing with the MHTEX ExPA sim required that we did not damage any of the hardware. To interface with the ExPA sim NGAS provided the Cal Poly design team with models designating the hole sizes and location on the ExPA sim. The 30000 assembly is seen in Fig 82.



**Fig 82. 30000 assembly with 10000 lifting assembly configured to pick up the ExPA sim plate.**

NGAS provided the Cal Poly design team with proof, yield, and ultimate load factors. A summary of the load factors for the 30000 assembly is listed below in Table 28. It was required that the GSE hold a 227 lb limit load to account for the MHTEX ExPA sim.

**Tab. 28. Load Factors used for the 30000 assembly**

<b>Load Factors</b>	<b>Lifting/hoisting Devices</b>
Proof load factor	2.00
Design yield load factors	3.00
Design ultimate load factors	5.00

For use by NGAS all fixtures and devices were required to be manufactured from non-corrosive and out gassing materials to protect the facilities and their cleanliness. In addition all fixtures and parts were required to be tested and certified under static loading at proof load factors. NGAS quality assurance personnel were required to be present at each proof test. Testing procedures in terms of correct loading magnitudes, time, and set up were approved by the quality assurance personnel. Upon completion of each test the tested part was tagged with the static load amount and time applied.

The GSE provided to NGAS was required to be shipped in adequate shipping container. The 30000 assembly was shipped in the 20000 shipping container. All shipping containers were required to be coated with flame retardant paint, in either a white or grey color, and be clearly labeled with a part number. The 30000 assembly was required to be delivered with an EIDP containing all analysis, drawing, certifications, and instruction for use of the particular fixtures.

## **1. Requirements for Part 20006: Rod End Bearing**

The requirements for the 20006 assembly were flown down to the Rod End Bearing. The rod end bearing interfaces the T-handle pin and support bar and interfaces the support bar and the MHTEX body on one side and the support bar and the MHTEX top plate on the other. These rod end bearings also are used to interface the 10000 assembly and the ExPA sim plate for horizontal lifting.

## **B. Conceptual Design**

The design of the 30000 assembly was influenced by requirements from the customers, NGAS and KSC, and internally, from the Cal Poly design team. The design of the 30000 assembly served the purpose of allowing NGAS and KSC the ability to pick up the ExPA sim plate during integration and testing. The conceptual design of the 30000 lifting was driven by the design of the 10000 lifting assembly.

## **C. Preliminary Design**

The Preliminary design of the 30000 assembly was started by checking to ensure that the 10000 assembly had the capability to lift the ExPA sim plate. The mass and CG offset for the ExPA sim plate were given to the Cal Poly design team by NGAS.

## **D. Finalized Design**

To provide the ability to pick up the ExPA sim plate the 30000 assembly was designed to use the 10000 assembly, without the front and rear bars, and provide pick up points for the lifting fixture. To allow the 10000 assembly to lift the ExPA sim plate three 1/4"-28 rod end bearings were to be inserted into the ExPA sim plate in predefined holes. The three holes were chosen to allow the 10000 assembly to lift the ExPA sim plate and allow for a nominal shift in the CG and still be lifted vertically. The 10000 assembly's lifting cables were also configured in particular holes to achieve the proper CG placement for vertical lifting. The rod end bearings were not required to be electro polish etched because the ExPA sim plate was not flight hardware and there was no risk of damaging the holes.

## **E. Component Analysis and Manufacturing**

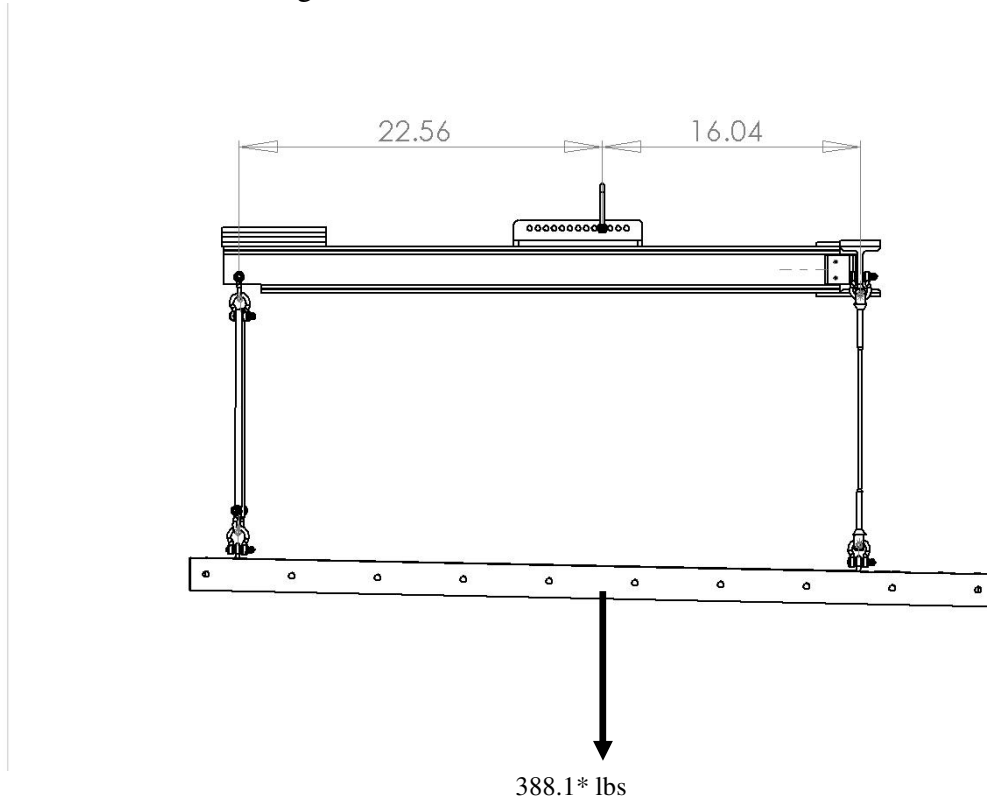
The 10000 assembly was analyzed for the proper CG offset of the ExPA sim plate. Using the proper hole configuration the ExPA sim plate was able to be lifted with limited tilt from horizontal. To first get a rough CG location, holes were chosen on the ExPA sim plate for the rod end bearings to be placed into for the 10000 assembly to interface with. A common axis was placed on a hole pattern of the ExPA sim plate for NGAS personnel to call out during integration to take CG offset into account. The rear cable would attach to the ExPA sim plate at a hole located 6 holes in the y direction and 0 holes in the x direction. One of the rear cables would attach to the ExPA sim plate at a hole located 12 holes in the y direction and 14 holes in the x direction. The other rear cable would attach at a hole located 0 holes in the y direction and 14 holes in the x direction.

The 10000 assembly was to be configured with the top hook plate shackle placed in hole number 4 to take into account the CG offset of the ExPA sim plate as seen on the 30000 assembly drawing in appendix A. The front cables of the 10000 assembly lifting fixture were



placed in holes D and I to take into account CG offset in the y direction as seen in the 30000 assembly drawing.

The loads, seen in the free body diagram for the 30000 assembly show in Figure 83, were analyzed. Under worst case CG offset and loading a rod end bearing was analyzed to see no more than 227 lbs of loading.



**Fig. 83 Free body diagram of for the 30000 assembly**

The rod end bearings were analyzed for ultimate tension and the margin of safety for the rod end bearing was calculated to be 0.70 for ultimate and 1.83 for yield.

### **F. Proof Testing**

The 30000 assembly was not proof tested as an assembly. The 10000 assembly was proof tested to a level higher that was required for the 30000 assembly. The rod end bearings were proof tested with the electropolished red end bearings from the 20000 assembly. These rod end bearings were proofed to a working load of 550 lbs.

### **G. Delivery**

The 30000 assembly was packaged with the 20000 assembly for delivery to NGAS. The three rod end bearings that were used to attach the 10000 assembly lifting sling to the ExPA sim plate were tagged and bagged separately from the rest of the 20000 assembly hardware so that it could later be identified by personnel.

## VI. ABET Requirements

### **(a) an ability to apply knowledge of mathematics, science, and engineering**

Engineering is the use of scientific relationships to solve complex problems with the language of math. The design team used the scientific relationships of stress and strain to show the part materials and designs are able to support the loads.

The analysis of the 10000, 20000, and 30000 assemblies applied statics and mechanics of materials concepts. Free body diagrams, beam theory, torsion theory, and tear out analysis routines are used in part designs. The beam, torsion, and tear out analyses use engineering relationships that prove designs will hold the design loads show in the analysis discussions and appendixes. Mathematics quantifies loads and allows the engineering relationships to be conveyed to Northrop Grumman with calculations. Without mathematics engineering theory would have no medium of communication.

### **(b) an ability to design and conduct experiments, as well as to analyze and interpret data**

Proof test for ground support hardware are designed, implemented, and results provide data for analysis. The proof tests are designed to simulate working conditions of ground support hardware and results are interpreted separately for the 10000, 20000, and 30000 assemblies.

The Proof test design, implementation, and analysis for the 10000 meets requirements for the Kennedy Space Center and Northrop Grumman facilities. The testing requirements consisted of strength and surface properties. Northrop Grumman requires all cable assemblies and the 10000 assembly be loaded to the specified testing loads. The Cal Poly students made a design for the tension tests and communicated them to Cable and Specialty rigging. The proof tests are outsourced because Cal Poly does not have the facilities. The results are analyzed by a quality engineer from Northrop Grumman who looks for surface cracking and verifies the calibration of the testing machinery. Kennedy Space center required NDI testing for shackles or swaged cables to check for micro surface cracking. The conditions of the NDI testing were specified by a specification from Northrop Grumman. The results from that test are interpreted by a materials engineer. The 20000 assembly and 30000 assembly testing design, implementation, and analysis is done at Cal Poly by the design team.

The 20000 and 30000 assembly proof test design is driven, implementation, and analysis is driven by the Northrop Grumman requirements. The proof test loads are low and 25 lb shot bags are used. The parts for each assembly are loaded and a Northrop Grumman quality engineer supervised the test. The analysis of results consists of checking the part for cracks. The loads are low and parts did not show signs of yielding.

### **(c) an ability to design a system, component, or process to meet desired needs within realistic constraints such as economic, environmental, social, political, ethical, health and safety, manufacturability, and sustainability**

Ground support hardware is designed to meet specific interface, strength, and cost requirements. Cal Poly students identified the requirements, designed a feasible product, and efficiently produced the product.

Each assembly is characterized with a specific set of requirements defined by its purpose and Northrop Grumman company standards. Each assembly has a purpose that is characterized with lifting and interface requirements. The interfaces are specified by the MHTEX experiment and the loads are characterized by the weight of the MHTEX or its components. Northrop Grumman requires the ground support hardware meet load safety requirements specified in the appendixes and in part analysis. The customer specified the component testing standards and communicated additional material treatment requirements like chromate conversion coating hardware that went to the Kennedy Space center. The design team designed solutions that met each customer requirement, designed tests that met Northrop Grumman's standards, and effectively communicated the ideas to the customer.

The Design team marketed our solutions to Northrop Grumman. Cal Poly students participated in design reviews that show Northrop Grumman an analytical solution to their hardware needs. The design reviews presented the engineering analysis that proved the strength of the Cal Poly engineered parts. Solid models provided Northrop Grumman with confidence that the Cal Poly lift slings interface correctly with Northrop Grumman hardware. During design reviews engineers at Northrop Grumman manipulated solid models that were sent to them through email. After approval of designs, Cal Poly students purchased and manufactured components for the three assemblies.

The students had to build what they had designed, this drove them to design with manufacturability and practicality in mind. The students designed the parts with manufacturability in mind. Parts are designed to be built with common tools so expensive, specialized tools do not need to be used. Parts are designed to be assembled quickly. Bolts and cables are not hard to install because of space constraints. The tools for manufacturing are common and cheaply purchased. Materials used are easily purchased and available on the central coast.

The design team chose a solution that met customer requirements. The solution is marketed to the customer and the design incorporates feasibility. The students designed with access to materials, tools, and manufacturability in mind.

#### **(d) an ability to function on multidisciplinary teams**

The design team functioned as a multidisciplinary team. Each member of the team committed to various jobs and modalities. The design team purchased, tested, manufactured, and designed hardware.

The design team members liked different aspects of building the ground support hardware and found niches. Cal Poly students working on the MHTEX project committed to various tasks. Some team members liked to manufacture and purchase materials. Other team members liked design of components and proof test design. As the project progressed, team members multitasked. Some students on the MHTEX experiment designed proof tests and components on different parts of different assemblies. This simulated the industry because engineering firms have many engineers working on different projects at different stages of development.

The members communicated in regular meetings and worked cohesively. Each member of the MHTEX team had deadlines and deliverables. The deliverables allowed other students to complete their tasks. Email and meetings were used regularly to ensure that students engaged in different stages of development for a particular component understood current progress.

### **(e) an ability to identify, formulate, and solve engineering problems**

The MHTEX project iterative design process tested the design team's ability to solve problems. A specific example is the shackle proof test failure mentioned in the cable assembly design section of this report. The proof test failure required students to figure out why the part failed the proof test and devise a quick solution to meet deadlines.

After the failure, the students had to gather information and speak with the shackle manufacturer to figure out the problem. The reason the shackled failed was not clear immediately and was not provided by the cable distributor testing the hardware. After a phone call to the shackle manufacturer, the students figured out that incorrect information about the capability of the shackle had been provided to Cal Poly. The manufacturer did not communicate effectively with the distributor and the distributor published incorrect working load data. Cal Poly purchased material from the distributor thinking the incorrect data was correct. The students did not check the capability of the shackle with the manufacturer; as a result, the shackle failed because it was loaded beyond its yielding capability published by Cableco. After the students diagnosed the problem, a quick solution was necessary.

The students had to locate and select another shackle that met requirements to complete the assembly. The students designing the parts that interfaced with the shackle and defined a set of requirements for new shackle. The requirements composed of strength, interface dimensions, and availability. The drivers for the new shackle were defined primarily by strength and availability. The interfaces were not a primary driver because the parts had some flexibility because they were not completely manufactured and could be altered. This is an example of collaborative design to solve problems.

The design team of the entire assembly worked together to adjust to the new shackles. Some members researched new shackles that met the strength requirements. The students that worked on the parts the shackles interfaced with worked out how their interface could change to accommodate the new interface part dimensions. After a few iterations a solutions was devised and the shackles purchased.

The design team's reaction to the shackle is an example of identifying engineering problems and solving them.

### **(f) an understanding of professional and ethical responsibility**

Engineering requires a high degree of commitment and loyalty to one's company. Engineering the ground support hardware taught the Cal Poly design team that ethics are important to insure the success and the interests of those whom use the hardware one designs.

Northrop Grumman had Cal Poly sign an agreement that did not allow the students to share any information with other entities about the MHTEX project. Northrop Grumman taught the students about the importance of working only for one firm. Aerospace components are designed using proprietary processes or specialized materials. These define a firm and allow it to have a competitive edge in the industry. If engineers share work between different companies the tools used for Northrop could be used for another entity. Engineers not only need to be loyal to their company, but also diligent in the accuracy of the information and results they provide to their company.

Students learned that providing incorrect information is unethical because it threatens

safety and causes costly problems in the design process. Providing incorrect information can cause hardware to be used in ways that it is not designed. Hardware that is used improperly can fail resulting in damage to company interests or personnel. Providing incorrect information can cost companies money and put engineers under immense time pressure. The example of the shackle failure shows how incorrect information can cost time and put pressure on engineers. Engineers always need to publish correct information and be diligent in checking their work.

**(g) an ability to communicate effectively**

Working with Northrop Grumman requires the design team to learn clear analysis documentation, how to produce manufacturing drawings, and solid models for engineering communication.

Solid models of the assemblies are produced using an unfamiliar program: Solid Works. The hardware are modeled in solid works because the program is compatible with the customer. The design team easily emailed the components of each assembly for design reviews. Solid models of the components communicate how each function within the Vertical Lift assembly. The solid model allows the customer to directly see the entire assembly and manipulate components. A copy of the drawing parts are used in the analysis package for the part free body diagrams. The drawings provide not only the customer clear information, but the manufacturer clearly understands the what to build from the drawings.

The drawings provide manufacturers with specific details and specifications to build each component of the three assemblies. The drawings provide part numbers of parts not manufactured at Cal Poly and assembly instructions. Specific instructions for proof testing, material requirements, machining tolerances, and important part dimensions are also contained in each of the assembly drawings. The drawings function as a contractual agreement between Northrop and Cal Poly. Northrop agrees that Cal Poly can build what the drawing specifies and test it according to specifications. This shows Northrop Grumman that Cal Poly guarantees the parts will be manufactured to the plans. Clear and concise engineering communication ensure a healthy relationship with the customer

Providing correct information and documentation is essential. The customer's understanding of the function of the assembly determines if the customer will buy the product.

**(h) the broad education necessary to understand the impact of engineering solutions in a global, economic, environmental, and societal context**

The location in which an engineering solution is implemented affects the requirements. The 10000 assembly is not only used at the Northrop Grumman Facility in Redondo Beach, but also at the Johnson and Kennedy Space center. The location added requirements to the design that had economic impacts on the project.

The facilities in which ground support hardware are used have individual requirements for use in their facilities. The Kennedy Space Center required that the aluminum hardware be chromate conversion coated because of the corrosive nature of Cape Canaveral. The cable assemblies had to be NDI tested because of prior experience at Cape Canaveral. Building the ground support hardware to be used at Cape Canaveral incurred extra cost for Northrop Grumman. The additional testing and chromate conversion coating takes more time. The chromate conversion chemicals and testing incur more financial cost to the project.

The design modifications due to location, taught the design team to evaluate the surroundings that the part is being used in. If engineering solutions can be affected by location, they can also be affected by society, health concerns, and many other factors.

**(i) a recognition of the need for, and an ability to engage in life-long learning**

Building ground support hardware for Northrop Grumman showed the design team the importance of individual learning and how to engage in life long learning. The design team recognized the learning involved in completing the project and found resources to provide information.

The aerospace curriculum did not prepare students with the tools to analyze structures or product knowledge; as a result, the students used various resources to learn. Dave Esposto showed the design team how to analyze fasteners, cables, and I-beam flanges. Professor Esposto worked with the team to make us think in terms of structural engineering models. Professor Esposto stressed the importance of making the engineering models have the same boundary conditions as the real physical situation. The students learn structures for design reviews and part analysis. Students went out into industry for materials and product research. The students learned how to get the specifications for components used in the assemblies and how to ask for bids. The students did much learning over the summer, but most importantly they learned the process of life long learning.

The students learned to synthesize new concepts not learned in their classes. The students learned to locate information. After problems are identified the design team learned to research and compare feasible solutions. Some learning involved contacting those in the industry who have experience and asking questions. After the information is located the students synthesize the new information and applied it to design problems. The students quickly learned the art of learning is taking new concepts and solving problems with them.

Building ground support hardware required students to learn by experience and by doing. New concepts were learned and applied to the current problems.

**(j) a knowledge of contemporary issues**

Deadlines, cost effectiveness, and efficient designs are contemporary issues that engineering firms work with each day. The students working on ground support hardware for Northrop Grumman worked completed deliverables, iterated to find the most efficient design for the least amount of money.

The design team planned design reviews and assembly delivery dates with Northrop Grumman. The design team worked under time pressure to meet internal submittal dates and make time sensitive design decisions. The team learned that each phase of the design-build process is linked. If the design is not finished on time then more time pressure is put on the manufacturing or purchasing of components to meet the delivery date to the customer.

Efficient and cost effective design is critical to the satisfaction of the customer and profit. The design team used common tooling on manufactured parts to save money. Each part has similar features so new tooling does not need to be purchased. Purchasing new tools costs extra money in set up time and product. Efficiency in design is characterized by designing with manufacturability and the customer actually using the part in mind. The part designs are inefficient if they cannot be easily built with common tools. The customer does not want a part

that is complicated and hard to use.

The design team worked diligently to ensure the deadlines were met and an efficient design that the customer could easily use was produced.

**(k) an ability to use the techniques, skills, and modern engineering tools necessary for engineering practice.**

The design team acted as purchasing agents, developed schedules, worked with the customer to design proof tests, and with outside vendors to carry out proof tests that could not be done at Cal-Poly. Analysis, solid modeling software, planning software, and technical communication are a few of the modern engineering tools used to complete these various engineering tasks.

The design team acted as purchasing agents to procure hardware and other services that could not be completed at Cal-Poly. Students had to purchase raw materials like stock aluminum, fasteners, and other mechanical parts. It is tough to find components that meet particular specifications when working with companies on the central coast who do not deal with the aerospace industry. Specification sheets with the needed detail were not readily available. The design team had to do extra work to contact the manufacturer and find specification sheets because these were typically not provided by the wholesaler or distributor. Deciding what parts to purchase consisted of design and analysis. These tools were used size specific parts and after the part passed a design review, the students went shopping for the material. However, the students found that there were not only materials to purchase, but services like electro polishing or NDI testing that the customer required, that could not be done at Cal-Poly. The students contacted vendors that did the needed testing. The students used part drawings, printed specifications, and technical documentation to provide the vendor with detailed information so they could provide the correct product or service. The design team learned to work with other companies to ensure that the part or service provided was correct.

The design team created a schedule of proof tests, design reviews, and group deliverables to ensure delivery in a timely manner. Microsoft scheduler was used to build a timeline of the project. Each phase: design, analysis, procurement, manufacturing, and delivery, was given a specific amount of time. Each person in the group was responsible for a portion of the work and understood that they had to meet the deadline. Students also coordinated with Northrop Grumman engineers to schedule design reviews so the project designs could be approved before the scheduled time for manufacturing and part procurement. Working from this timeline, members of the team coordinated proof tests with both quality assurance personnel and the proof testing vendor. Designing proof tests is not only a complicated scheduling issue, but also a complicated communication problem between the customer and outside vendor.

The design team developed proof tests that met customer requirements and organized those proof tests that could not be done at Cal-Poly with outside vendors. The design team interpreted customer testing requirements from the load safety factors provided by Northrop Grumman for specific hardware or other tests like NDI that was specified by the Kennedy Space Center. The proof test designs were reviewed by Northrop Grumman during design reviews. After the approval of a proof test, the test instructions were documented and the vendor contacted by a student. The design team found vendors like: Specialty Crane, Earth Systems, and Cableco who had the equipment to perform the needed proof tests. The design team then provided the vendor with proof test drawings created in Solid Works and written instructions.

Modern engineering tools were used continuously throughout the project to complete various steps of the engineering process. Modern engineering tools were used to schedule events, design proof tests, and purchase materials.

## **H. Acknowledgements**

The success of the Cal Poly team in designing, manufacturing, certification, and delivery of the MHTEX ground support equipment was due to the synergistic efforts of various industry personnel. Special thanks are required for the Northrop Grumman personnel that made the project possible. These personnel include Ted Nye for making this project available to the students, Cecelia Penea, Danny Li, and Derek Saddler for working closely with the students in bi-weekly teleconferences, and Jim Davis for supervising various proof tests. Proof tests were supplied by Specialty Crane, electropolishing was provided by Electromatic, and cable assemblies were supplied by Cableco. Special thanks to David Esposito for making the project possible by interfacing with industry and teaching the students basic engineering principles.