LAWRENCE
LIVERMORE
NATIONAL
LABORATORY

Mirror Sub-Assembly End-Effector Design

Becky Butlin

February 9, 2007
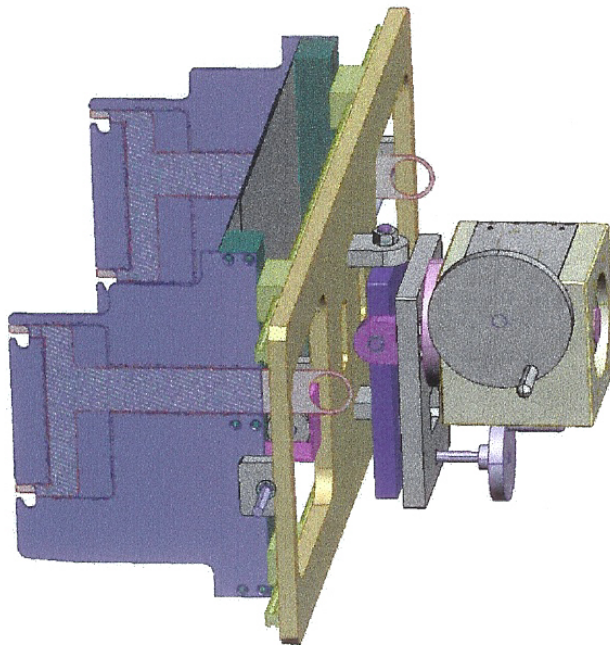
Disclaimer

Mirror Sub-Assembly End-Effector Design

By:

Becky Butlin

MS Plan II – Technical Report for Professor Michael Hill
University of California, Davis
12/29/06

Table of Contents

Table of Figures

Introduction

The Optic Assembly Building (OAB) is a facility where large optical mirror units are assembled and installed into Line Replaceable Units (LRUs) for deployment into the National Ignition Facility (NIF) laser system. The New Optics Insertion Device (NOID), shown in Figure 1, is a powered jib crane specially designed to handle large optical assemblies. The NOID arm has three degrees of freedom. It can rotate about the vertical boom, travel up and down the boom, and extend away from and retract in towards the boom. The NOID is used to assist in the assembly of five types of Laser Mirror (LM) LRUs. These five LMs have been creatively named, LM4, LM5, LM6, LM7, and LM8. The LM4 and LM5 LRUs each contain four Mirror Sub-Assemblies (MSAs). The LM6, LM7, and LM8 LRUs each contain 2 MSAs. The MSAs are assembled apart from the LRU and are then installed in the LRU at the LM4-8 workstations shown in Figure 1.

An MSA NOID End-Effector is required to interface with the MSAs and install them into the LRUs. The End-Effector must attach to the Robo-hand on the end of the NOID arm. At the time the MSA NOID End-Effector was being designed the NOID, the LM4-5 workstation, and the LM6-8 workstation were already installed in the OAB. The LRUs and the MSAs designs were also complete. The MSA NOID End-Effector design had to work with the assembly equipment and LRU designs that were already in place.
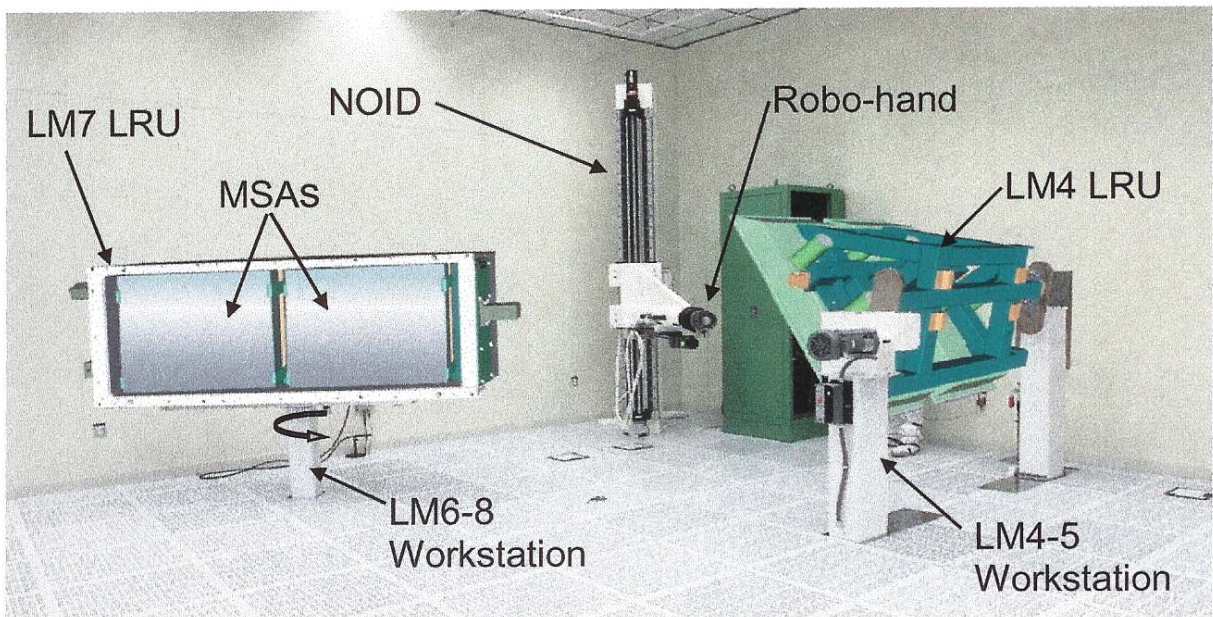

Figure 1 LM4-8 Workstation in the OAB

Design Requirements

The most general, but most important design requirement is that the MSA NOID End-Effecter must be able to engage the MSAs and place them in the LM4-8 LRUs in the correct orientation. The MSA design for each LM4-8 LRU is very similar but varies in size. The widths of the MSAs range from 440mm to 685mm. The MSA NOID End-Effector must be able to handle all sizes. All the MSAs have four mounting lugs on the sides as shown in Figure 2. These mounting lugs were added to the MSA design with the intent that they would be used to handle the MSAs in the OAB.

When the MSA is installed in the LRU, there is very little clearance between the MSA and the LRU frame, which creates a space constraint for the End-Effector design. In addition to this space constraint there is also restraint on the length of the End-Effector. The length of the MSA NOID End-Effector has to be less than 495mm or the MSA will interfere with the LRU during installation.

In order to install the MSA into the LRU the MSA NOID End-Effector has to have several degrees of freedom. The End-Effector must be able to manipulate the MSA so that the bolt holes on the back of the MSA can be aligned with the clearance holes in the LRU. The MSA must also be oriented so that it is parallel to the mating surface on the LRU. In order for this to be accomplished the End-Effector has to have a minimum of ±5° twist rotation, ±5° pitch rotation, and ±20° yaw rotation.

MSA Mounting Lugs
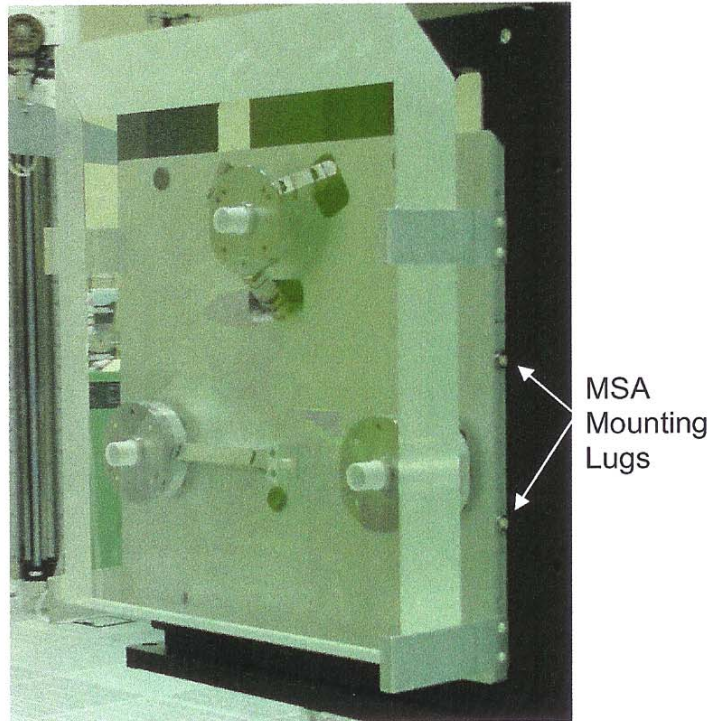
Figure 2 LM4 MSA

The MSA NOID End-Effector must maneuver the MSAs safely and comply with LLNL design standards for lifting devices. To maneuver the MSAs safely, the degrees of freedom of the End-Effector must be controlled to some degree so that the End-Effector is not free to rotate. The End-Effector must be able to hold and maneuver the heaviest MSA which weighs 270 lbs. To comply with the LLNL design standards the static safety factor on each component must be equal to or greater than 3 on yield or 4 on ultimate. Seismic load calculations require a minimum safety factor of 1 on yield or 1.25 on ultimate. There is also a maximum allowable bending moment on the NOID Robo-hand of 6000 in-lbs.

The MSA NOID End-Effector will be used in the OAB, which is a class 100 cleanroom. All exposed materials of the End-Effector must be cleanroom compatible and approved by the NIF cleanliness steering committee. The End-Effector should also be designed in such a way that particle generation is minimized when the End-Effector is used.

The last design requirement is that the final cost of the MSA NOID End-Effector should be minimal. There was no set value, just a requirement to keep the cost down.

Design Interface Requirements

As previously discussed, the MSA NOID End-Effector has to attach to the Robo-hand on the NOID arm and interface with each LM4-8 LRU assembly. The End-Effector must install the MSAs into the appropriate LRUs without interfering with the LRU frame. The last piece of equipment the End-Effector must interface with is the MSA Handling Fixture shown in Figure 3. The MSA Handling Fixture is what is used to rotate the MSA from a horizontal to a vertical orientation. The End-Effector must accept the MSA from the MSA Handling Fixture while it is in the vertical orientation.
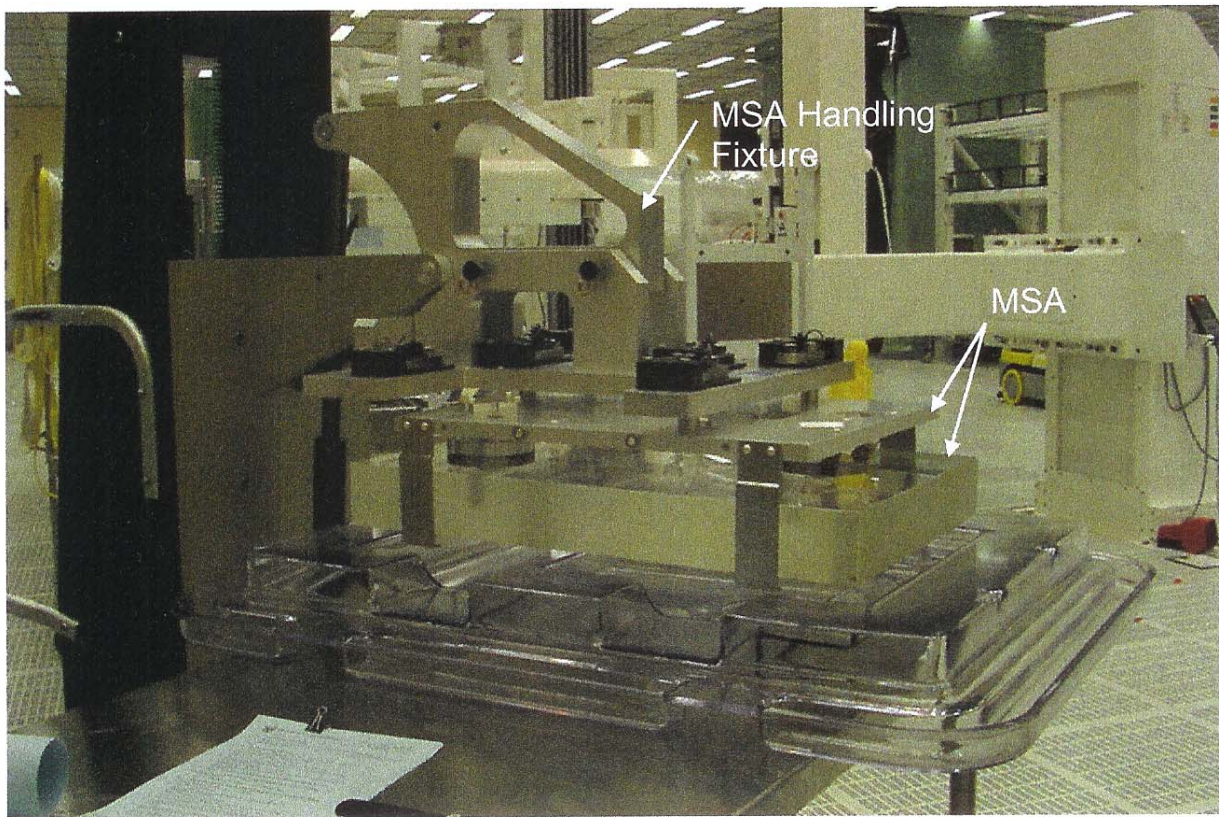
Figure 3 MSA Handling Fixture lifting the MSA

The MSA NOID End-Effector Design

Design Overview

The purpose of the MSA NOID End-Effector is to remove an assembled Mirror Sub-Assembly (MSA) from the MSA Handling Fixture and place it into the LRU. It was a challenge to obtain all the required degrees of freedom in such a compact system. For each degree of freedom, I came up with several conceptual designs. With each conceptual design, I made several sketches

and created a decision matrix to determine which of the design concepts best met the design requirements. I also shared some of the design concepts with the assembly technicians that would be using the End-Effector to gather their thoughts. Once a design concept was chosen, I modeled the design in Pro/DESKTOP and created detailed drawings for each component. The drawings can be found in Appendix A. Great care was taken to make sure the dimensions and tolerances for each part were correct so that the parts would fit together as designed. Once the design was complete a Final Design Review was held. I presented the design to a large group of NIF engineers, mechanical technicians, OAB operations personnel, cleanliness and materials personnel, and safety personnel. The design review was a success and I was given permission to procure the MSA NOID End-Effector.
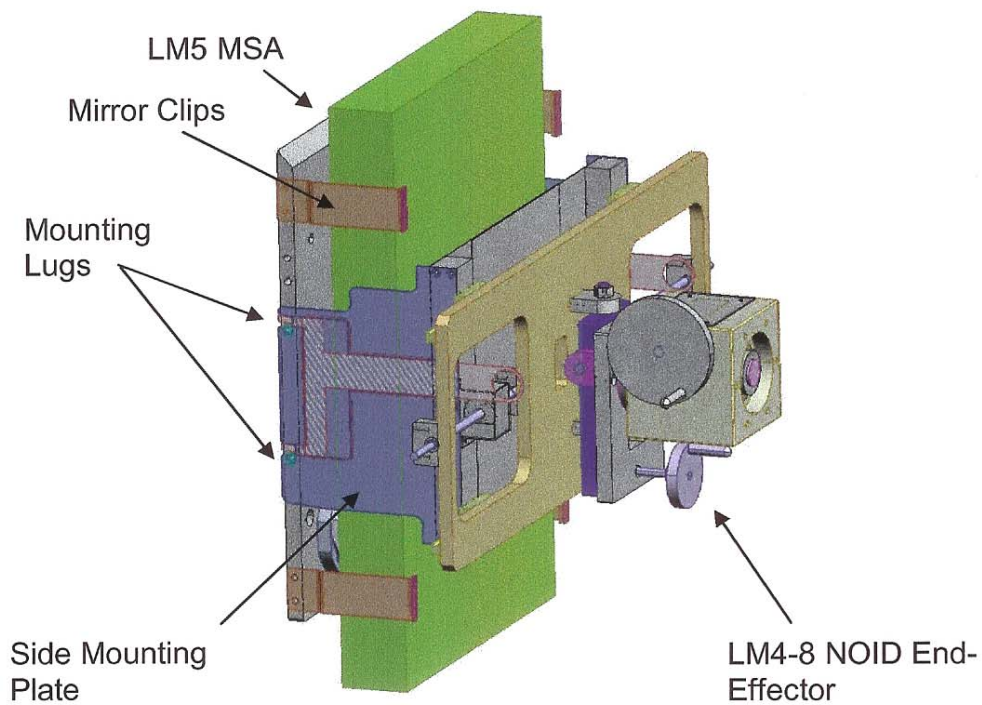
Figure 4 MSA NOID End-Effector holding an LM5 MSA

The final End-Effector design is shown in Figure 4. The MSA NOID End-Effector is a mechanical device, which approaches the MSA from the front and holds the MSA using the four mounting lugs on the MSA. A hook and latch mechanism is incorporated into the Side Mounting Plates so that a positive capture of the MSA is assured. The NOID can then lift the MSA from the MSA Handling Fixture and place it into the LRU.

The MSA NOID End-Effector design consists of three Sub-Assemblies as shown in Figure 5. The main function of the Rail Sub-Assembly is to interface with each of the MSAs. The distance between the Side Mounting Plates is adjustable to accommodate the varying widths of the MSAs. The main function of the Tilt Sub-Assembly is to provide Y-axis (yaw) and Z-axis (pitch) rotation. Rotation about the Y-axis is only allowed when a friction nut has been loosened. Rotation about the Z-axis is adjusted manually by turning a hand wheel. The main

function of the Twist Sub-Assembly is to provide rotation about the X-axis (twist). The rotation about the X-axis is also manually adjusted by turning a hand wheel. In this case, turning the hand wheel causes a shaft to rotate, which in turn causes the first two Sub-Assemblies to rotate about the X-axis. The following three sub-sections give more details about the each Sub-Assembly and discuss the conception of the designs.
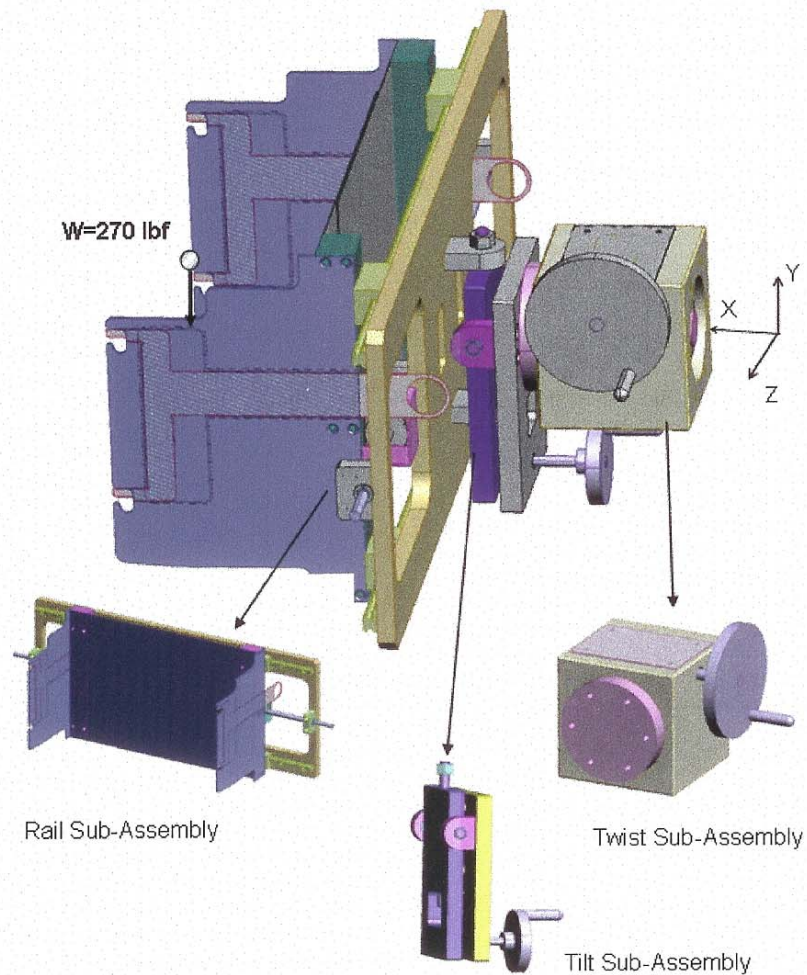
Figure 5 Three Sub-Assemblies of the MSA NOID End-Effector

Rail Sub-Assembly Design

As mentioned earlier, the main function of the Rail Sub-Assembly is to interface with all the MSAs. It has to safely hold the MSAs and protect the optics from particles the End-Effector may generate. The main design constraint of the Rail Sub-Assembly was the size because there is very little clearance between the MSAs and the LRU frames.

There were two main conceptual designs considered for the Rail Sub-Assembly, which will be discussed later in this section. The final design is shown in Figure 6. The main components of the Rail Sub-Assembly are the side mounting plates, the protecting shields, the rail system and ACME rod, and the rail plate.
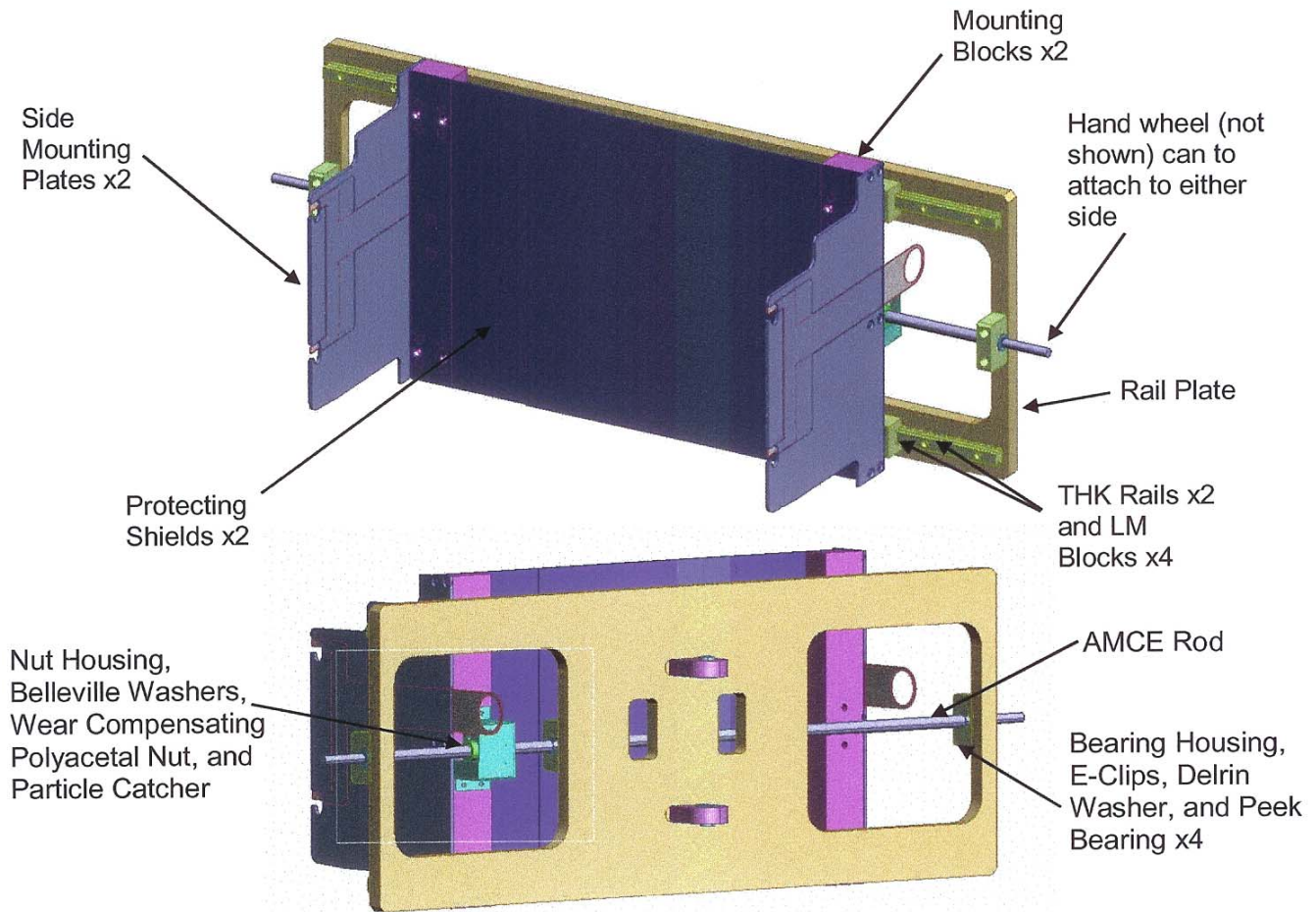
Figure 6 Components of the Rail Sub-Assembly
(Additional components shown in figure 7)

Vacuum grippers are often used to handle optical assemblies, but this was not an option for the End-Effector design because of the 4 Mirror Clips on the MSA (shown in Figure 4). The gasket of the vacuum gripper is only allowed to touch the optic outside of the clear aperture (where the NIF laser beam will not hit). In the case of the LM4-8 optics, there is only 14 mm around the edge of the optic that is outside of the clear aperture. The Mirror Clips on the MSA would interfere with a vacuum gripper design. Also, since there are several different MSA optic sizes, there would need to be different vacuum plates for each optic size. It was clear that using the 4 mounting lugs on the sides of the MSAs was the best choice. The mounting lugs for each MSA are in the same location. The design constraints did not leave too many options for the design of the side mounting plates. The plate geometry was completely driven by the tight clearances

between the MSAs and the LRU frames. Stainless Steel was chosen as the plate material primarily because of its strength and durability over aluminum. Stainless steel also particulates less than aluminum. A latching mechanism was incorporated into the side mounting plates so that a positive capture of the MSA is assured. The side mounting plates consist of three pieces sandwiched together. The two outer pieces are welded together and the middle piece is free to slide in an out. This middle piece is pulled back while engaging the MSA and then pushed forward to lock the MSA in the plate hooks while the End-Effector is handling the MSA.

Because the MSAs vary in width, the distance between the plates needs to be adjustable. There were two conceptual designs considered. The first design concept was to have two parallel rods that the side mounting plates could slide freely on. The side mounting plates would attach to a block that would have a bearing sleeve in them. The rods would be placed through the bearing sleeves and the side plates could simply be pushed or pulled along the rods into the correct position. In order to keep an MSA from moving once it was engaged, clamps would be placed on the rails. The second design considered was a rail and lead screw design. With this design, the side mounting plates could slide in and out on a set of linear rails. A lead screw would be used to drive one of the side plates and the other one would be able to slide freely. A decision matrix was used to choose the best design. This matrix can be found in Table 1.

Table 1 Decision matrix for the mechanism to adjust the width of the side mounting plates
(Each criterion was given a value between 1 and 5, with 1 being the lowest and 5 being the highest. The design with the highest total was deemed the best overall design.)

Design Criterion	% Weight of Each Criterion	Criterion Value	
		Rod and Clamp design	Rail and Lead Screw design
Cost	10%	5	3
Safety / Risk to the optic and personnel	40%	2	4
Ease to operate and control the motion	30%	2	4
Particle generation / need for lubrication	10%	3	3
Wear over time	10%	3	4
Total (Sum of % weight x the criterion value)		**2.5**	**3.8**

As shown in the decision matrix, the rail and lead screw concept was the better design choice. I chose to use THK stainless steel linear rails and mounts. I was able to work with the vendor to have them use Krytox LVP grease, which is a cleanroom approved lubricant. In choosing the lead screw I needed to find a screw and nut combination that would require little to no lubrication, would last a long time, and would be made out of the acceptable cleanroom materials. This was much harder task than I had expected. I wanted to find something off the shelf because it would most likely be less expensive than designing and fabricating something from scratch. I finally found a Precision Modified-ACME Lead screw and nut combination from McMaster-Carr. The screw is made of 303 stainless steel and the nut is a polyacetal, self lubricating, wear-compensating nut. The material of the nut had not been approved for use in the OAB. I worked with the NIF materials group and received approval for the use of this material for my application. To use the nut, they required that I put a particle catcher around the nut. I also added protective shields near the face of the optic to help mitigate contamination on the mirror surface.

The rail plate of the Rail Sub-Assembly was designed to hold the THK rails and the lead screw as well as interface with the Tilt Sub-Assembly. The width of the plate was chosen based on the widest MSA. The height of the plate was chosen based on the appropriate spacing of the THK rails. As the distance between the rails increases, the load on each rail decreases, but the overall weight of the plate increases. The plate height selected resulted in safety factors well above the required value and an acceptable weight for the Rail Sub-Assembly. Aluminum was chosen as the material for the rail plate to help keep the weight of the End-Effector down. Large cutouts were also placed in the plate to reduce the weight.

There are 4 aluminum blocks that attach to the rail plate to hold the lead screw. These are called the bearing housings and are shown in Figures 6 and 7. The thread of the lead screw was turned down at the location of the four blocks. Peek bushings were placed in each block to lower the friction between the lead screw and the block and to reduce the particle generation. Peek was chosen over other bearing materials because of its low coefficient of friction and durability. E-clips were placed at the two end blocks to prevent the lead screw from translating. There is one other block that holds the lead screw nut. This aluminum block is called the nut housing and is shown in Figure 6. It is important that a majority of the MSA weight is transferred to the THK rails and not onto the lead screw. To transfer the load correctly, the nut needs to be perfectly aligned with the lead screw both when the End-Effector is loaded and unloaded. To minimize the load transferred to the lead screw, Belleville washers were placed between the nut housing and the bar it is mounted to.

The last pieces of the Rail Sub-Assembly are the two hinges shown in Figure 7. These are used to interface with the Tilt Sub-Assembly and assist with the ±20° yaw rotation. This will be discussed further in the next section.

Bearing Housing and Peek bushing (slotted to help with assembly)

Hinges and Peek bearings (interfaces with the Rail Sub-Assembly)

Slots so that the hinges on the tilt sub- assembly did not interfere with the rail plate
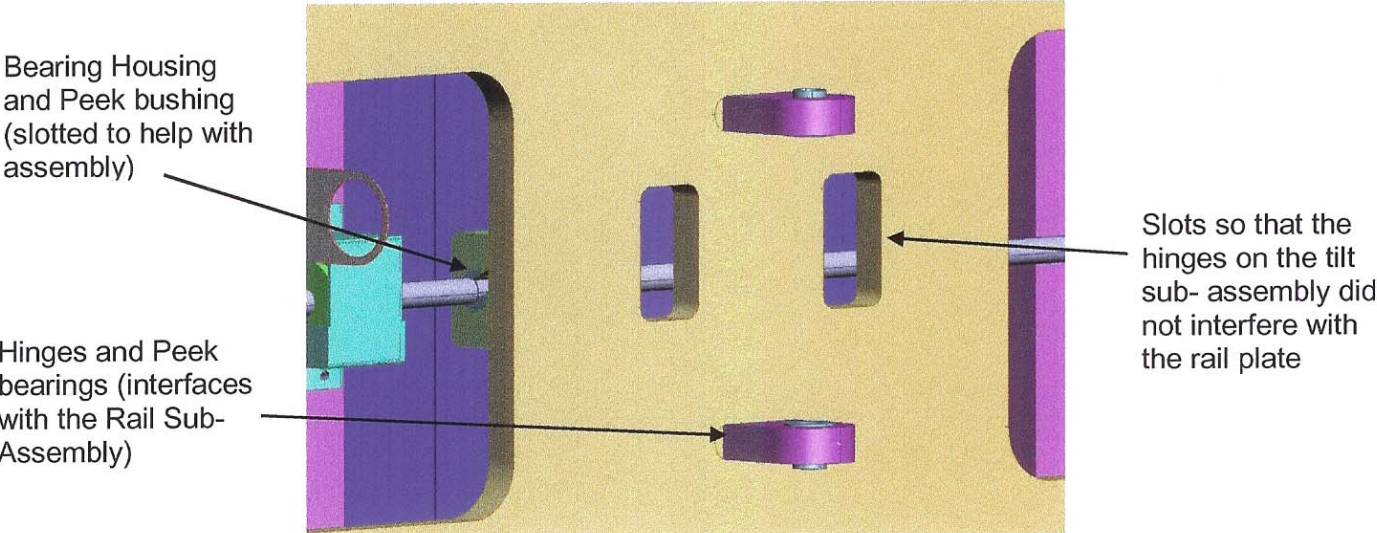
Figure 7 Additional components of the Rail Sub-Assembly

Tilt Sub-Assembly Design

The main function of the Tilt Sub-Assembly is to safely rotate the MSA in pitch and yaw. Like the Rail Sub-Assembly, the main constraint of the Tilt Sub-Assembly design was the size. Since the total length of the End-Effector is limited to 495 mm, the total length of the Tilt Sub-Assembly had to be minimal. The other constraint that played a large role in this part of the design was the requirement to be able to safely control the angular motion. Lastly, the weight was a consideration because of the maximum moment allowed on the NOID Robo-hand.

There were several conceptual designs considered for the Tilt Sub-Assembly, which will be discussed later in this section. The final design is shown in Figures 8 and 9. The main components of the Tilt Sub-Assembly are the two parallel aluminum plates, the rotating pins and hinges, the pitch adjustment system, and the yaw locking system.

Figure 8 Components of the Tilt Sub-Assembly

Figure 9 Pitch adjustment system of the Tilt Sub-Assembly

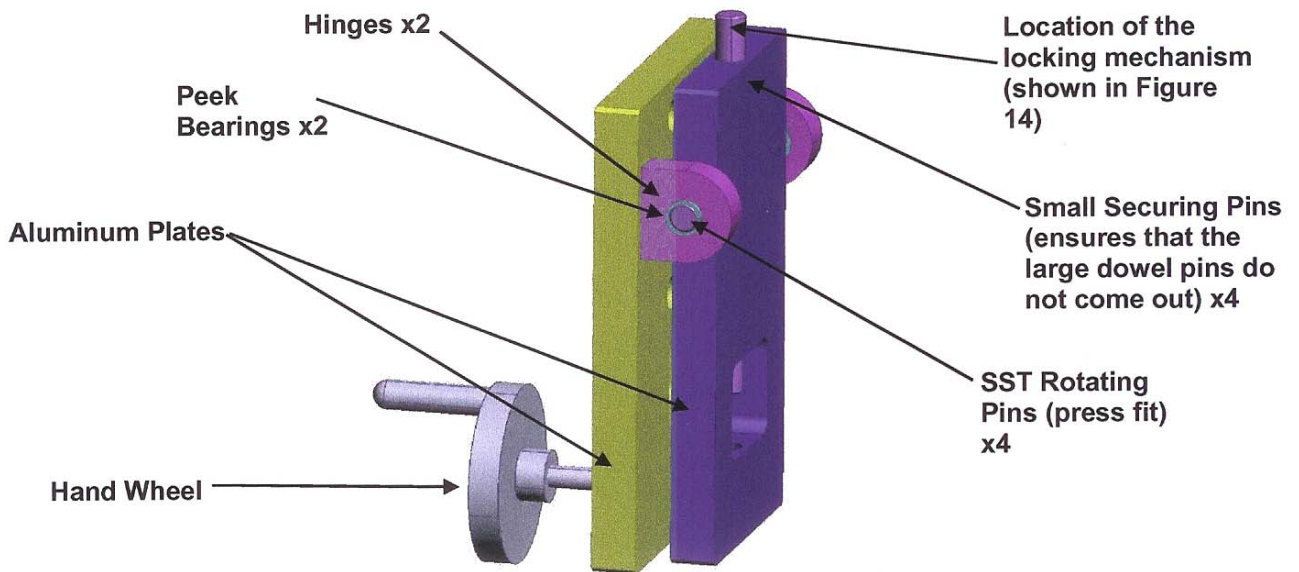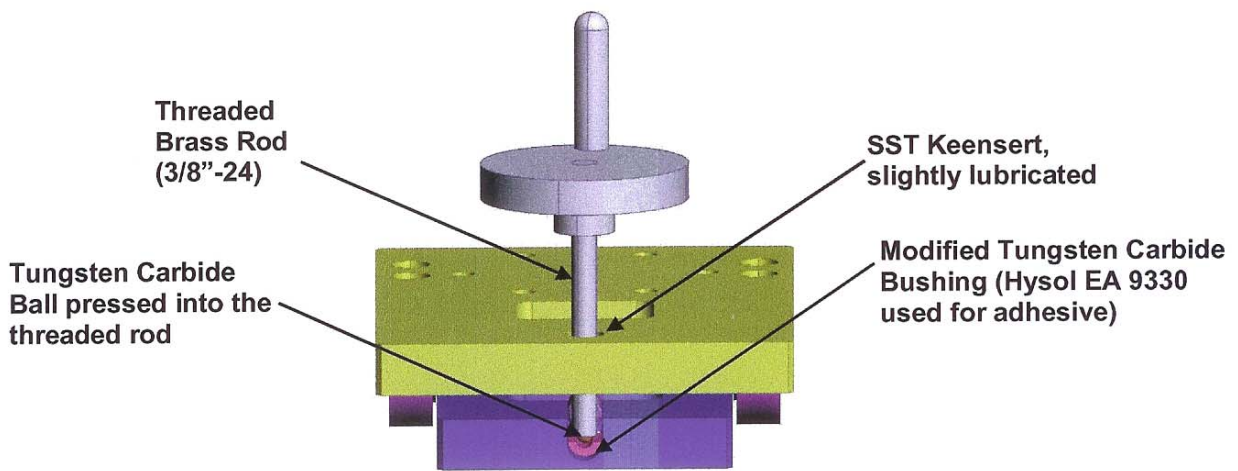When an MSA is being installed in an LRU, the MSA must be parallel to the mating surface on the LRU. In order for this to happen, the yaw orientation of the MSA relative to the NOID arm must continually be changing as the NOID arm is extended. For this reason, the yaw rotation of the End-Effector must be easy to control and adjust. The pitch rotation, on the other hand, only needs to be adjusted once or twice during the installation of an MSA.

To keep the design costs down, the first idea for this portion of the design was to see if there was anything commercially available that would meet the requirements. Unfortunately, I did not find anything that would meet the space, material, and load requirements of this system. The next idea was to use a combination of a commercially available joint with a custom design. There were two joint options: I could use a ball and socket joint, which would allow rotation in all directions or I could use a universal joint that would only allow pitch and yaw rotation. In both cases I would need to design mechanisms that would safely control the rotation. A sketch of this design idea is shown in Figure 10. The other idea was to design a mechanism that would control the pitch rotation while allowing the End-Effector to freely rotate in yaw about a vertical pin. I came up with several mechanisms that would adequately adjust and control the pitch motion. Two of the mechanisms considered are shown in Figure 10. A decision matrix was used to choose the best conceptual design, as shown in Table 2.
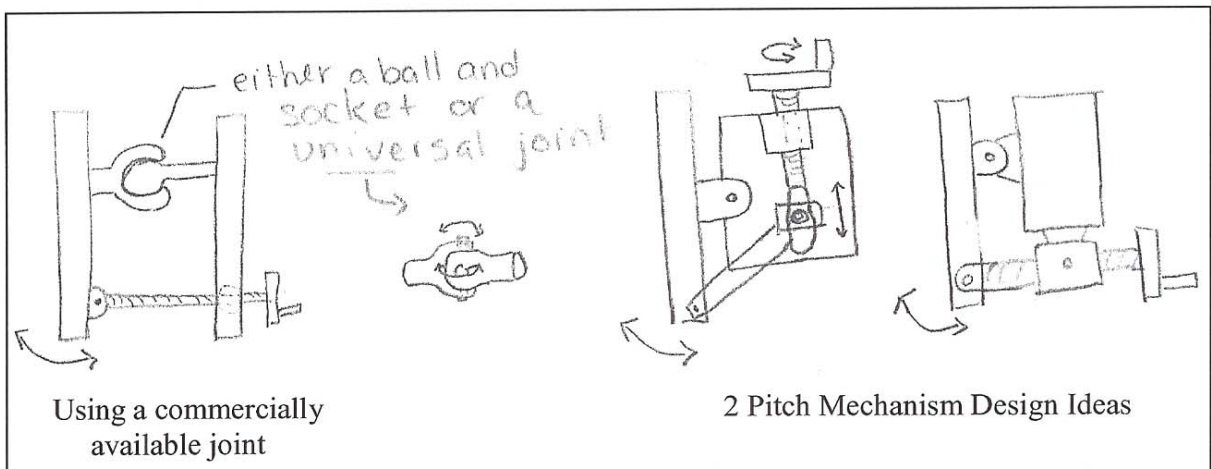

Using a commercially available joint

2 Pitch Mechanism Design Ideas

Figure 10 Tilt Sub-Assembly design idea sketches

Table 2 Decision matrix for the mechanism to adjust the pitch and yaw
(Each criterion was given a value between 1 and 5, with 1 being the lowest and 5 being the highest. The design with the highest total was deemed the best overall design.)

Design Criterion	% Weight of Each Criterion	Criterion Value	
		Design using a commercially available joint	Pitch Mechanism Design
Total design and manufacturing cost	15%	4	2
Safety / Good motion control	35%	3	4
Particle generation / need for lubrication	15%	3	2
Wear over time	10%	3	3
Light weight desired	10%	4	2
Short length desired	15%	4	1
Total (Sum of % weight x the criterion value)		**3.4**	**2.8**

From the decision matrix it was clear that the joint design was best, however, after looking at the commercially available joints I decided that it would be better to design a custom universal joint that would better fit my application. The pitch and yaw plate assembly previously shown in Figure 8 is the universal joint that I designed. This design is significantly more compact than any of the designs I could come up with using a commercially available joint. The two vertically oriented pins interface with the hinges and flanged peak bearings on the Rail Sub-Assembly. These pins allow for the yaw rotation. I decided to leave the yaw rotation free so that the MSA could be easily adjusted by hand while the MSA is being installed in the LRU. I added a locking system to the top pin so that the yaw rotation could be held if desired. The locking system was added later and is therefore not shown in the model. It can be seen in Figure 14 on page 18. The top vertical pin was made longer and was threaded on the top end. The two threaded rings of the locking system were designed to act like two locking nuts. The friction of the bottom ring against the top hinge is what prevents the Rail Sub-Assembly from rotating in Yaw when the locking system is engaged.

The two horizontal pins allow for the pitch rotation of the End-Effector. The last part of the Tilt Sub-Assembly to be designed was the mechanism to adjust and control the pitch rotation. There were two options that were considered. The first idea was to use a threaded rod through one plate that would contact the second plate near the bottom. With this design gravity would push the second plate against the threaded rod. As the rod is threaded in or out, the plate would tilt up or down. The second idea also used a threaded rod through one plate, but this time the rod would be attached to the second plate. For this second design to work the threaded connection in the first plate would have to be able to rotate and the connection with the second plate would have to mimic a ball and socket joint. The second design is safer than the first because it would positively connect the threaded rod to both plates, but it is significantly more complicated. I ended up selecting the first design, because the added safety of the second design was not worth the extra design complications. The final pitch adjustment system is shown in Figure 9. The hand wheel is attached to a fine pitch threaded rod which is threaded through a stainless steel keensert in the aluminum plate. Since it is common for similar materials to gall together in a cleanroom, brass was selected for the threaded rod material. Cleanroom approved lubrication

was also placed on the keensert to lower the friction and allow for easier turning of the handle. A lot of thought was put into the end of the threaded rod that would slide up and down the second plate as the threaded rod is turned. Particle generation, contact stresses, and friction were all factors in this part of the design. A Tungsten Carbide ball from McMaster-Carr was inserted into the end of the threaded rod. A modified Tungsten Carbide bushing from Reid Tool was chosen as the contact surface for the ball. The bushing was cut in half and glued to the second plate, providing a curved surface for the ball to slide against. A curve surface was selected over a flat surface to reduce the contact stresses. Tungsten Carbide was chosen because of its strength, low friction, and wear resistant properties. Other Tungsten Carbide designs were considered, but they would have required new parts to be designed and manufactured. The ball and modified Tungsten Carbide bushing design was chosen because the parts were commercially available and inexpensive.

Twist Sub-Assembly Design

The main function of the Twist Sub-Assembly is to provide a controlled twist rotation of the MSA. The other function of this sub-assembly is to attach the whole end-effector to the NOID Robo-hand. The main design considerations of the Twist Sub-Assembly design were the size, the load capacity, and safety. Since the Twist Sub-Assembly is furthest from the MSA, there is a large moment load it must withstand.

There were several conceptual designs considered for the Twist Sub-Assembly, which will be discussed later in this section. The final design is shown in Figure 11. The main components of the Twist Sub-Assembly are the aluminum shaft housing, the shaft, two roller bearings, a clamp collar, and the twist mechanism.
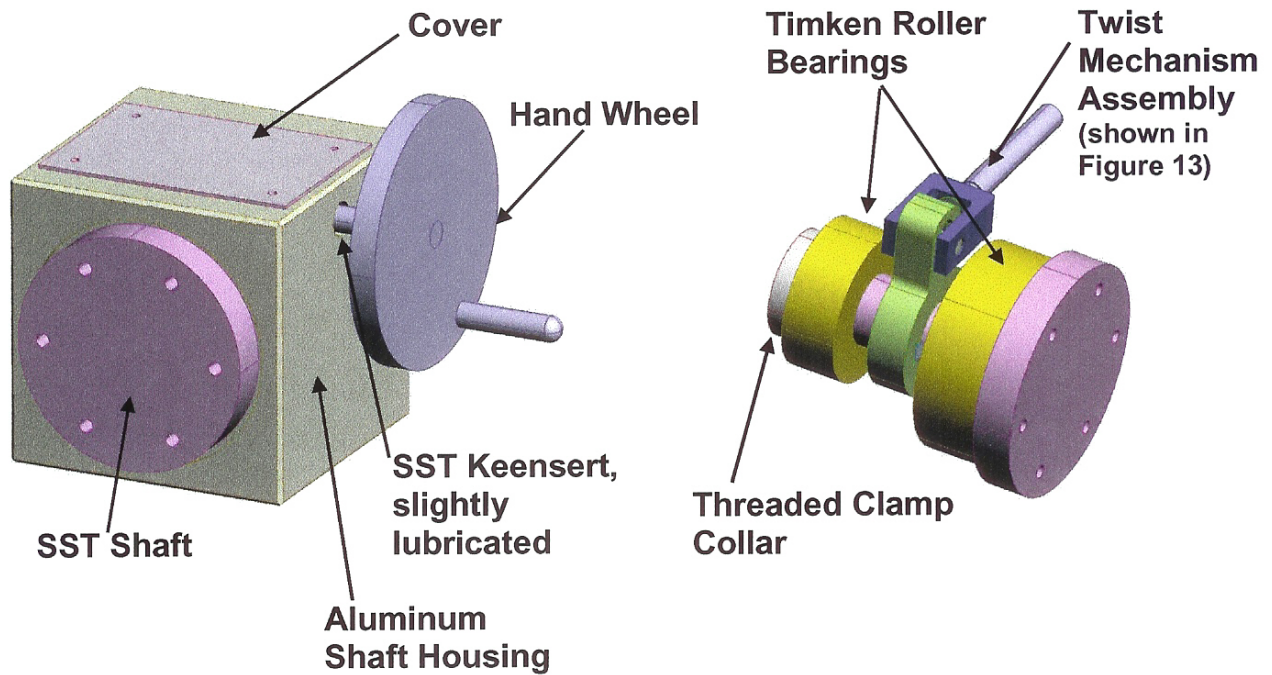

Figure 11 Twist Sub-Assembly Components

There were four conceptual designs considered for the Twist Sub-Assembly. Sketches of the four concepts are shown in Figure 12. The first concept is the most simple. It simply uses bolts, washers, and a plate with arced slots. The bolts would always be tight until a twist adjustment is required, at which time the bolts would be slightly loosened and the End-Effector (with an MSA) would be rotated by hand. This design concept is simple, but it has some very apparent safety concerns. If the MSA was not centered on the Rail Sub-Assembly it is possible that the End-Effector would twist on its own if the bolts were not tight enough. Also, when the bolts are loose it is risky to have personnel control the twist by hand. The second design concept considered is also very simple. It uses a large threaded shaft that would turn inside a shaft housing as shown in Figure 12. There would be a locking nut to hold the shaft from rotating when twist rotation was not desired. When twist rotation is required, the nut would be loosened and then the End-Effector (with an MSA) would be rotated to the desired position and the nut would be tightened again. This design is simple, but the process could be cumbersome if several iterations were required to get the exact twist orientation required. The design also has safety issues when the locking nut is loosened. The third concept uses a worm gear assembly to rotate a shaft that is connected to the rest of the End-Effector, also shown in Figure 12. This design provides a safer and more controlled twist adjustment. The challenges with this design were to find gears that are cleanroom compatible, provide the fine adjustment required, and have minimal backlash. The last design concept consists of a shaft, a shaft housing, and a mechanism that will cause the shaft to rotate. At this point in the design, there were a few design ideas for the mechanism that would control the twist rotation, but nothing had been finalized. Once again the pros and cons of each concept were considered and a design was chosen. The pros and cons are summarized in the decision matrix found in Table 3.
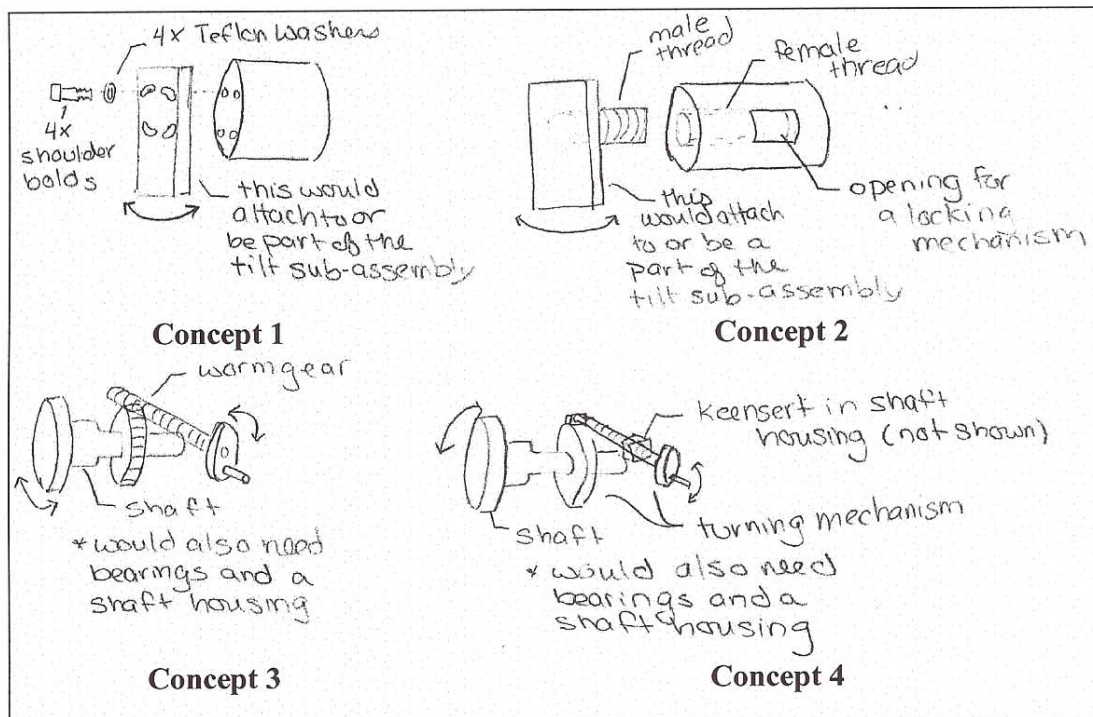

Figure 12 Twist Sub-Assembly design idea sketches

Table 3 Decision matrix for the twist design concept
(Each criterion was given a value between 1 and 5, with 1 being the lowest and 5 being the highest. The design with the highest total was deemed the best overall design.)

Design Criterion	% Weight of Each Criterion	Criterion Value			
		Bolt in slotted holes design	Threaded shaft design	Worm gear design	Twist mechanism design
Total design and manufacturing cost	10%	5	4	3	2
Safety / Risk to the optic and personnel	30%	1	1	5	4
Smooth and easy operation under a large moment load	20%	2	2	5	5
Fine adjustment	15%	1	2	3	5
Cleanliness: Particle generation / need for lubrication	5%	4	3	2	3
Wear over time/ backlash	10%	3	5	3	4
Short length desired	10%	5	3	3	3
Total (Sum of % weight x the criterion value)		**2.35**	**2.35**	**3.95**	**4.00**

As shown in Table 3, the worm gear and twist mechanism designs were far superior. Even though the twist mechanism design required more design effort, it was the superior design concept because of the need for fine adjustment and minimal backlash. After that concept was chosen, the twist sub-assembly had to be designed. First, I focused on the shaft and the shaft housing. The shaft design was mostly driven by the moment load it would carry and the size of the bearings that would support the shaft. The detailed drawing of the shaft can be found in Appendix A. The face of the shaft is 4.5 inches in diameter and interfaces directly with the Tilt Sub-Assembly. This way, when the shaft is rotated the tilt and rail sub-assemblies will also rotate. The shaft diameter steps down twice, once to fit the inner diameter of the larger bearing and again to fit the inner diameter of the smaller bearing. A radius was added at each step down to reduce the stress concentrations in the shaft. The end of the shaft was threaded so that a threaded clamp collar could be placed at the end of the shaft to hold the bearings in place. The two bearings in the Twist Sub-Assembly are tapered roller bearings from Timken. These were lubricated with cleanroom approved lubrication. I chose tapered roller bearings because they can take both thrust and radial loads. The threaded clamp collar was from King Bearing. The bearings and the clamp collar are concealed in a shaft housing. The main functions of the shaft housing are to support the shaft bearings and to connect the whole End-Effector to the NOID Robo-hand. The detailed drawing of the shaft housing can be found in Appendix A. The length of the shaft housing was chosen to give the whole End-Effector the optimal length for installing the MSAs into the LRUs. The internal cut outs were sized to properly hold the bearings and to allow for easy assembly of the Twist Sub-Assembly. The last component needed for this Sub-Assembly was the mechanism that would be used to turn the shaft.

After a lot of brainstorming and several sketches I came up with the twist mechanism shown in Figure 13. The twist mechanism consists of a threaded brass rod that is threaded through a slightly lubricated keensert in the shaft housing. As the threaded rod is turned, it pushes/pulls a fork which then pushes/pulls the force ring, causing the shaft to rotate.
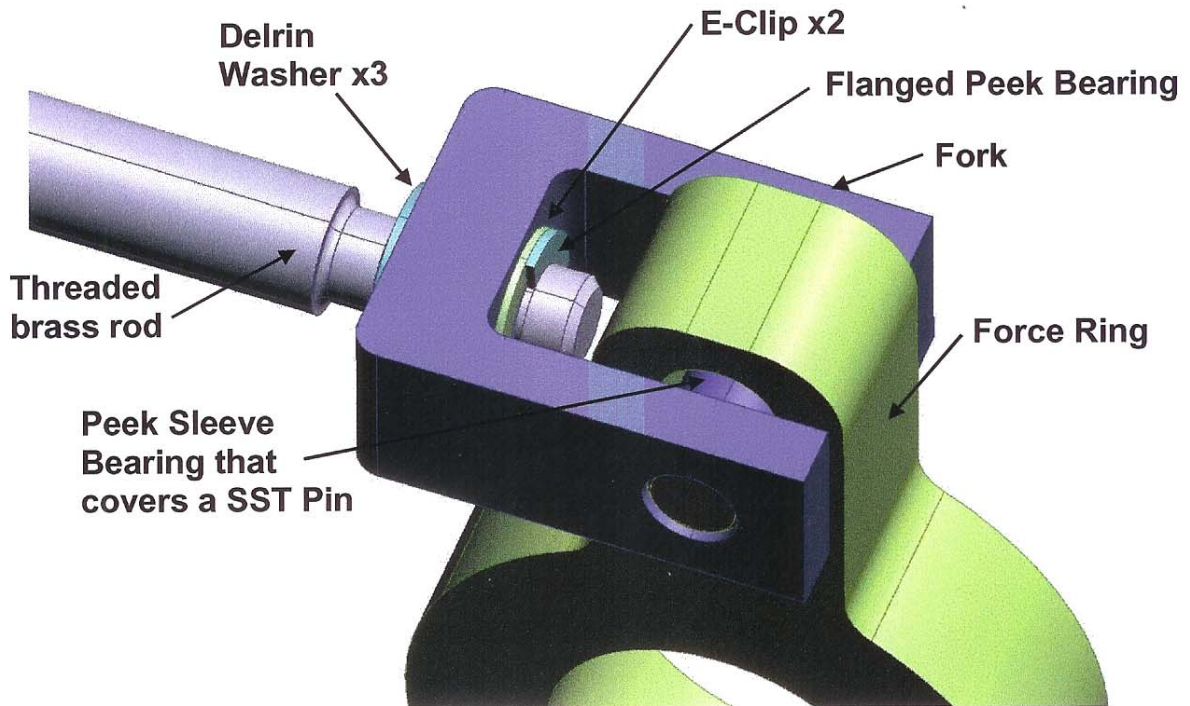
Figure 13 Twist mechanism of the Twist Sub-Assembly

The threaded rod is made from brass and has a ½-20 fine pitch thread. Brass was chosen because it is unlikely to gall with the stainless steel keensert. The fine pitch thread was chosen to allow for fine adjustments of the twist rotation. The threaded rod is turned down where it connects to the aluminum fork as shown in Figure 13. There is a flanged peak bearing sleeve that goes through the fork and allows the rod to turn in the fork. Peek was chosen for the bearing sleeve because of its low friction and wear resistant properties. The rod is held in place with two E-clips. A Delrin washer was placed between the E-clip and the fork on the non-flanged side of the peak bearing. On the other end of the force ring there is a stainless steel dowel pin that is pressed into the fork and goes through the vertical slot in the force ring. This dowel pin has a peek bearing sleeve around it that allows for a low friction interaction between the dowel pin and the force ring as the pin moves up and down in the slot. The peek sleeve was selected so that inner diameter just fit over the dowel pin. The slot in the force ring was sized so that there was very little slop between it and the peek sleeve. This was carefully done to minimize the backlash in the twist mechanism assembly. There are also two Delrin washers on the dowel pin axel that were placed on each side of the force ring. These washers were placed there so that the aluminum force ring did not side against the aluminum fork and create particles. The last piece of the twist mechanism is the stainless steel key that locks the force ring in position against the shaft.

The Design as a Whole

The last components to be selected were the hand wheels. The Rail Sub-Assembly, Tilt Sub-Assembly, and Twist Sub-Assembly each need one hand wheel. The hand wheel sizes were selected based on the clearance between the other components and on the estimated amount of force required to turn the wheel when the End-Effector is holding an MSA. All hand wheels were purchased from Carr Lane and are made of stainless steel. The hand wheel for the Rail Sub-Assembly is attached in such a way that it can easily be removed and moved to either side of the lead screw as required.

With the hand wheels in place and all three Sub-Assemblies mated together, the MSA NOID End-Effector design is complete and all design requirements have been met. The completed End-Effector is shown installed on the NOID in Figure 14.
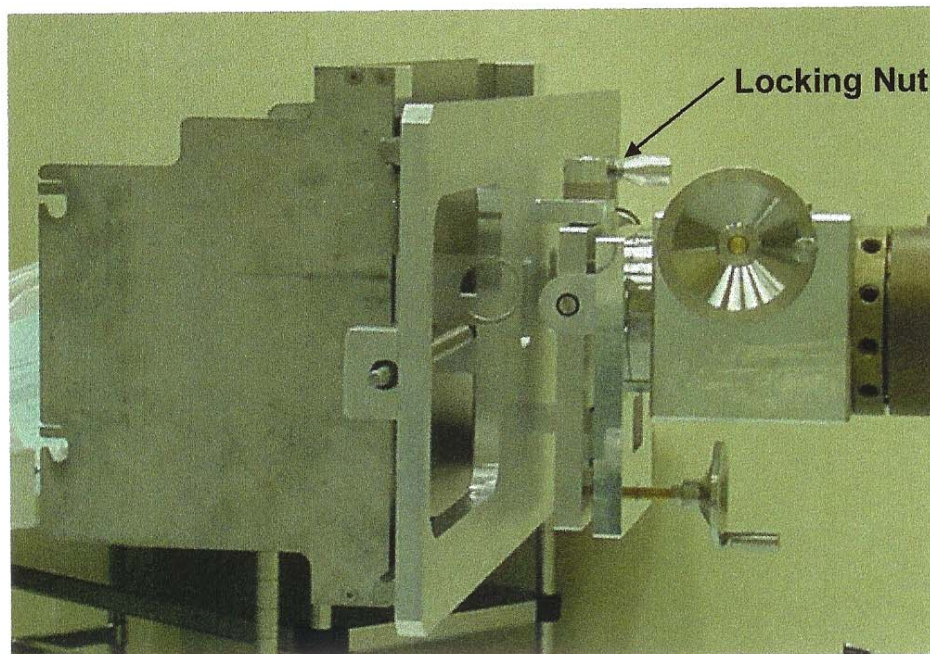

Figure 14 MSA NOID End-Effector installed on the NOID arm

As stated earlier, I modeled the MSA NOID End-Effector design in Pro/DESKTOP. I used the End-Effector assembly model to perform an interface study to make sure that in all cases the MSA NOID End-Effector could install the MSAs without interfering with the LRU frames. The clearance is tight in some cases, but the study showed that the appropriate MSAs can be installed in each LRU type. Figure 15 shows some of the tight clearances that were found in the study. As can be seen in the figure, the Rail Sub-Assembly hand wheel would have to be moved to a specific side of the lead screw in order for the MSA to be installed into an LM4 LRU.
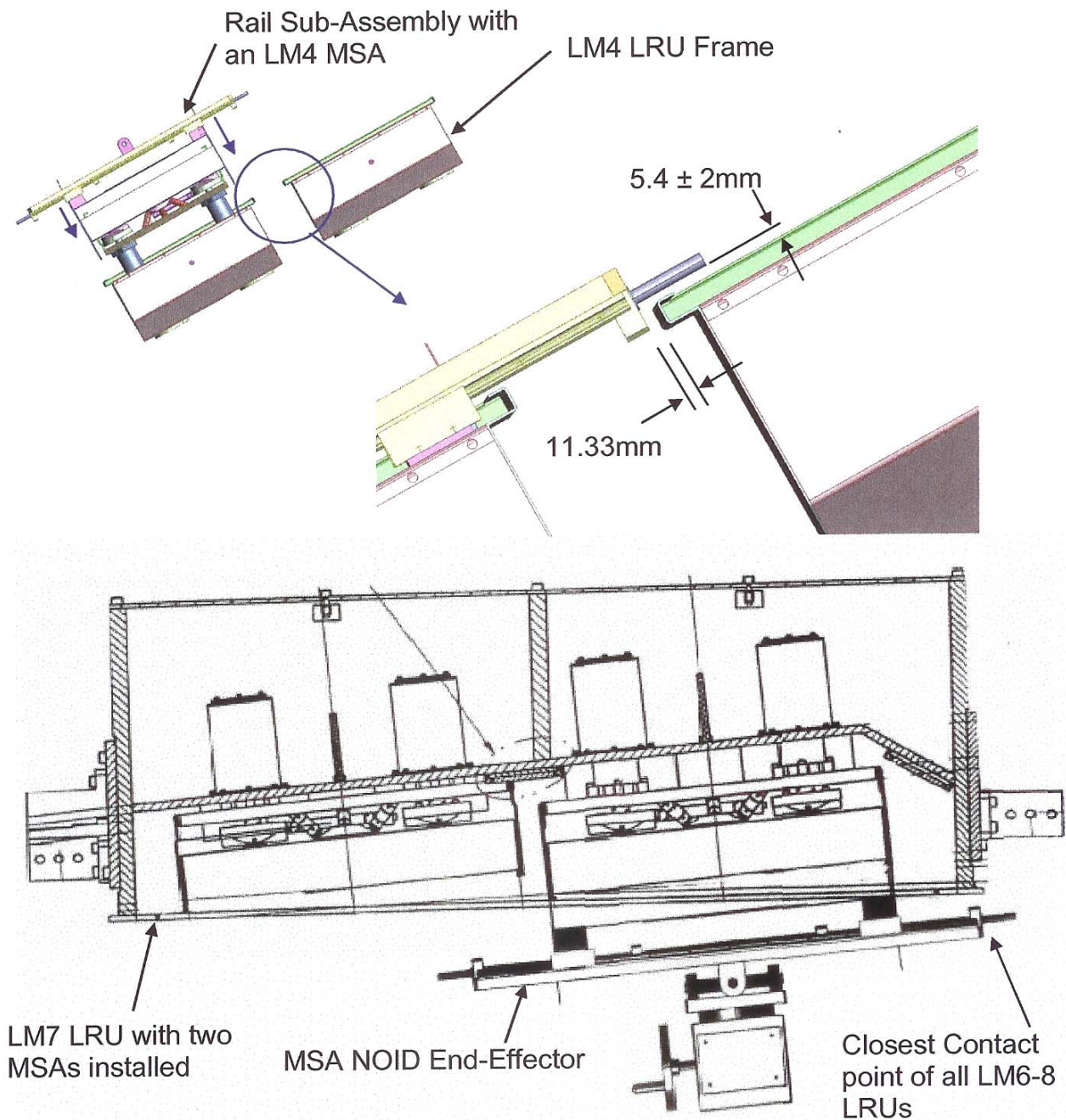
Figure 15 MSA NOID End-Effector interface with an LM4 (upper) and an LM7 (lower)

Design Calculations

Before the any equipment designed at Lawrence Livermore National Laboratory can be used, a safety note must be written and approved. The safety note for the MSA NOID End-Effector addresses the structural integrity of the End-Effector. The safety note calculations can be found in Appendix B. The End-Effector was designed per guidelines of the Lawrence Livermore National Laboratory Mechanical Engineering Design Safety Standards. In most cases the Design

Safety Standards require a minimum safety factor of 3 on yield or 4 on ultimate for static loads and a minimum safety factor of 1 on yield or 1.25 on ultimate for seismic loads.

The largest MSA weight of 270 lbs was used for the calculations. When the MSA is cantilevered off of the Side Mounting Plates, an over turning moment is created throughout the assembly. Overall, the machined components were considered very robust and the critical load path was assumed to be through the fasteners connecting the components and through the shaft of the Twist Sub-Assembly. It was also conservatively assumed that the MSA was rotated 20° in yaw and was 4 inches off center along the THK rails on the Rail Sub-Assembly. This placed an additional shear load on most of the End-Effector components.

Both static and seismic calculations were performed on the End-Effector. For the seismic calculations it was assumed that the LM4-8 NOID End-Effector will be subject to 1.3g vertical and 1.3g horizontal loads during a seismic event. It is unrealistic to have 100% of the seismic load applied to all three directions (X, Y, and Z) at the same time. For this reason, NIF seismic guidance was applied to the calculations. That is, vertical seismic load (Y) =100%, horizontal seismic loads (X and Z) = 40%. Due to the free-swinging nature of the NOID arm around the center column, horizontal accelerations in "Z" were considered nil.

As can be seen in Table B1 of Appendix B, most of the MSA NOID End-Effector components met the Design Safety Standards. The few that did not meet the Design Safety Standards were accepted based on the improbability of occurrence or acceptable consequences of the component yielding. The worst case static moment load on the Robo-hand was calculated to be 5319 in-lb which is just under the manufactures rated load of 6000 in-lb.

Procure, Assemble, and Test

The drawings found in Appendix A were sent to several machine shops for bidding. Lukas Machine, in Seattle Washington had the lowest bid at $4409.00 and was awarded the job. When the cost of the machined components is totaled with the cost of the "off the shelf" items, the total manufacturing cost of the MSA NOID End-Effector was just under $6,000, which met the low cost requirement of the design.

Once all the components were received, a dirty fit up was performed to make sure that all the pieces fit together as designed. At the time of the dirty fit up, a new yaw locking system was designed and fabricated. The parts were then disassembled and precision cleaned. The final assembly was performed in the OAB and the assembly was attached to the NOID as shown previously in Figure 14.

The final step before the MSA NOID End-Effector could be used for production was a load and function test. The load test was performed with 405 lbs hung from the End-Effector at the location of the MSA's center of gravity, as shown in Figure 16. The 405 lbs represented 150% of the weight of the heaviest MSA. The test was performed with the End-Effector at the extreme twist and yaw angles and with the MSA weight off center by 4 inches on the Rail Sub-Assembly. The load was held for 10 minutes and then the End-Effector was inspected for yielding. During the inspection it was discovered that the E-clips that were used in the Twist Sub-Assembly were not strong enough to carry the load at the extreme angles. The E-clips were replaced with a set

screw design and the assembly was reload tested. There were no signs of yielding during the second load test.

The function test of the End-Effector followed the load test. The function test was first performed using aluminum blanks for optics and installing the blank MSAs into an LM4 LRU frame. The second function test was performed using practice optics and installing the practice MSAs into an LM5 LRU frame. A picture of this function test is shown in Figure 17. Due to hardware restraints at the time a fit test was not performed with an LM6, LM7, or LM8 LRU. Since the designs are very similar, there is not expected to be any problems installing the MSAs into these LRUs.
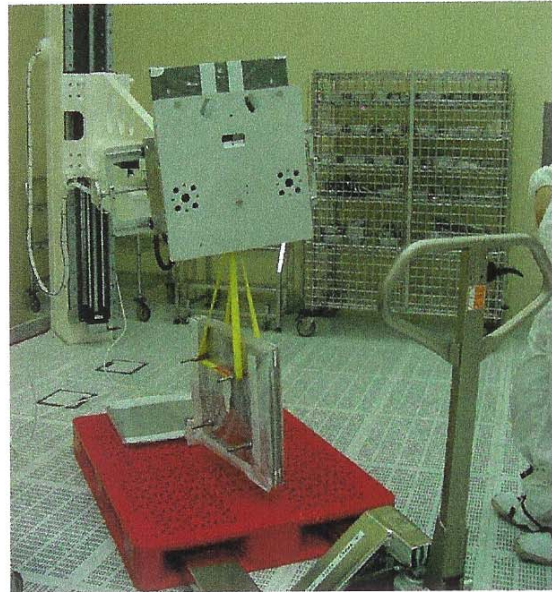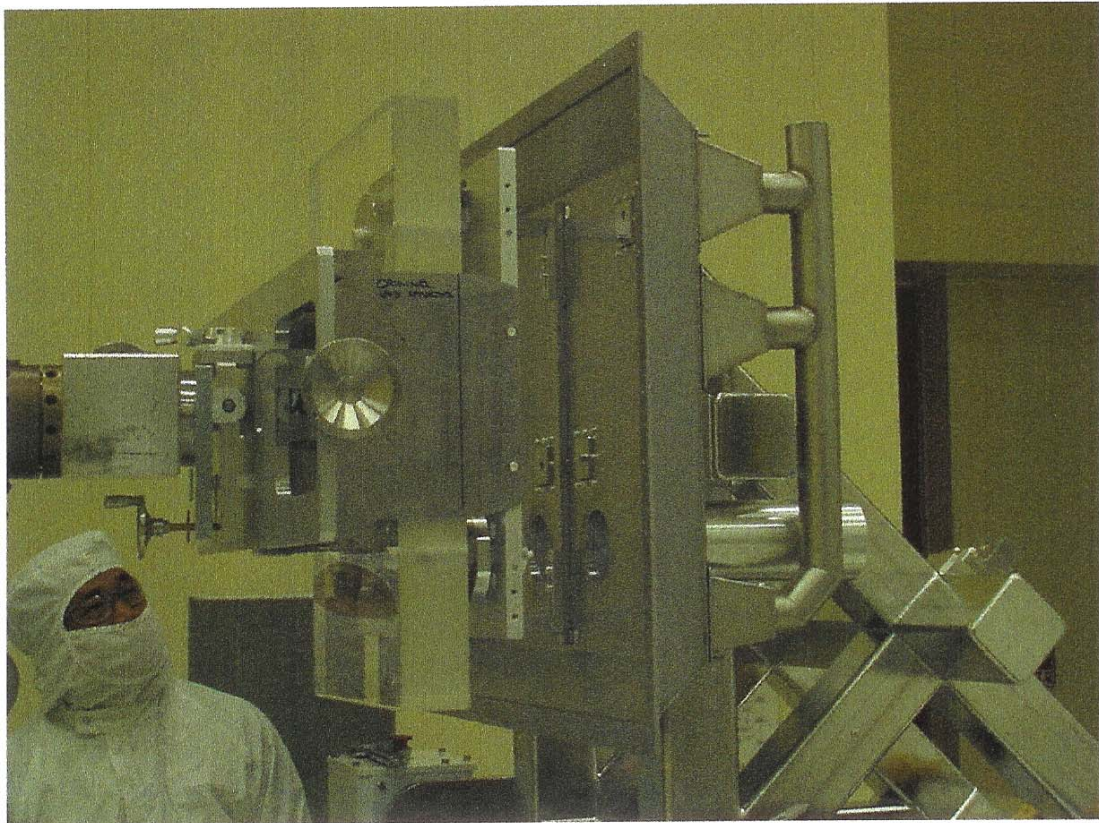

Figure 16 Load test of the MSA NOID End-Effector

Figure 17 Fit and function test of the MSA NOID End-Effector with an LM5 Practice optic

Conclusion

The final testing of the MSA NOID End-Effector was completed in June, 2004. Soon after the testing was complete, the LM4-8 LRU production was put on hold for a couple of years while other parts of the NIF laser were built. It was only in October, 2006 that the End-Effector was used to build its first production LM5 LRU. It is now December, 2006 and the End-Effector is currently being used to build a production LM4 LRU. So far the End-Effector has worked well and has been successful in installing 8 MSAs. LM6, LM7, and LM8 LRUs are scheduled to be assembled in January, 2007. I have no doubt that the MSA NOID End-Effector will effectively install the MSAs into these LRUs.
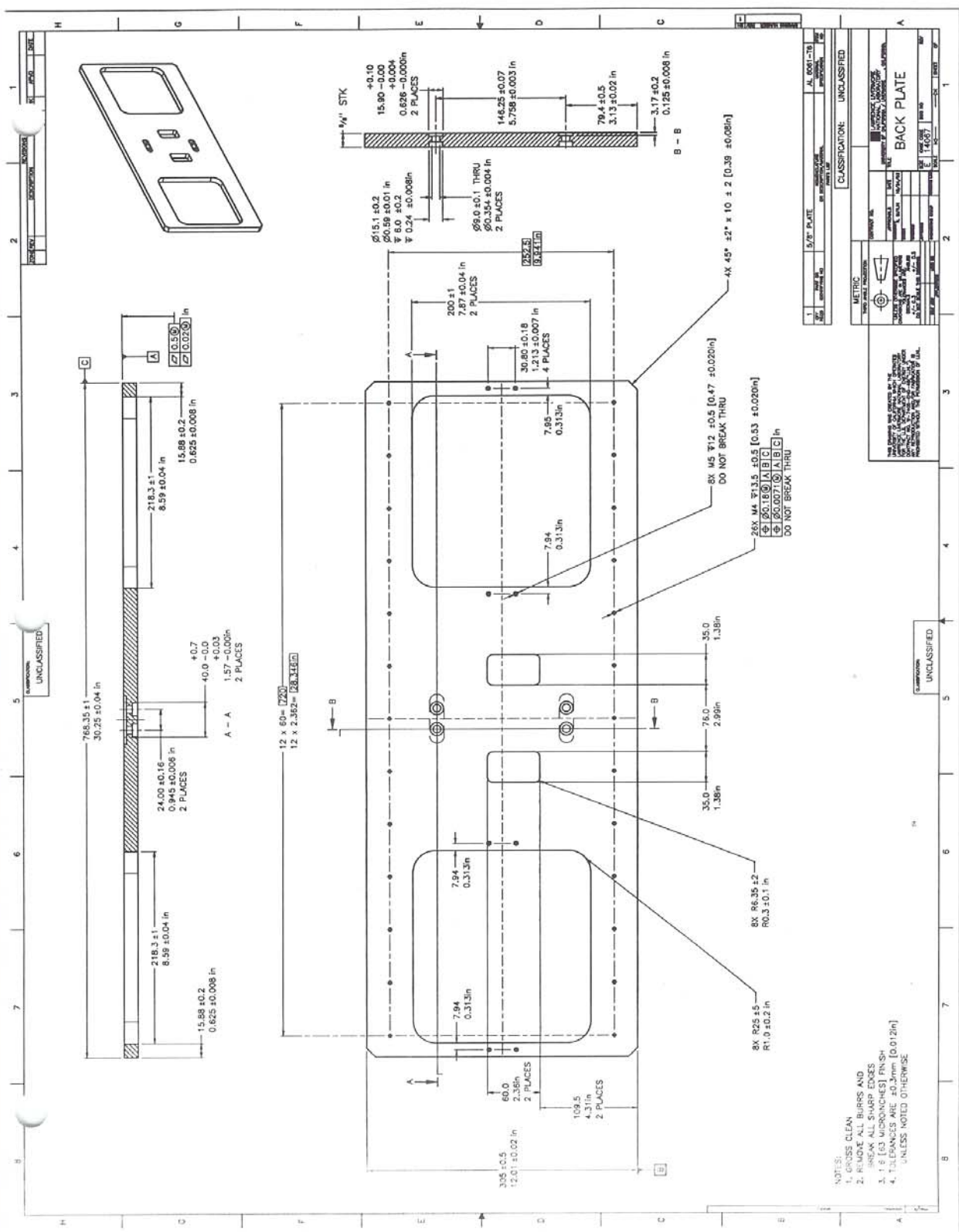
Appendices

Appendix A: MSA NOID End-Effector component drawings.

Appendix B: MSA NOID End-Effector Safety Note Calculations
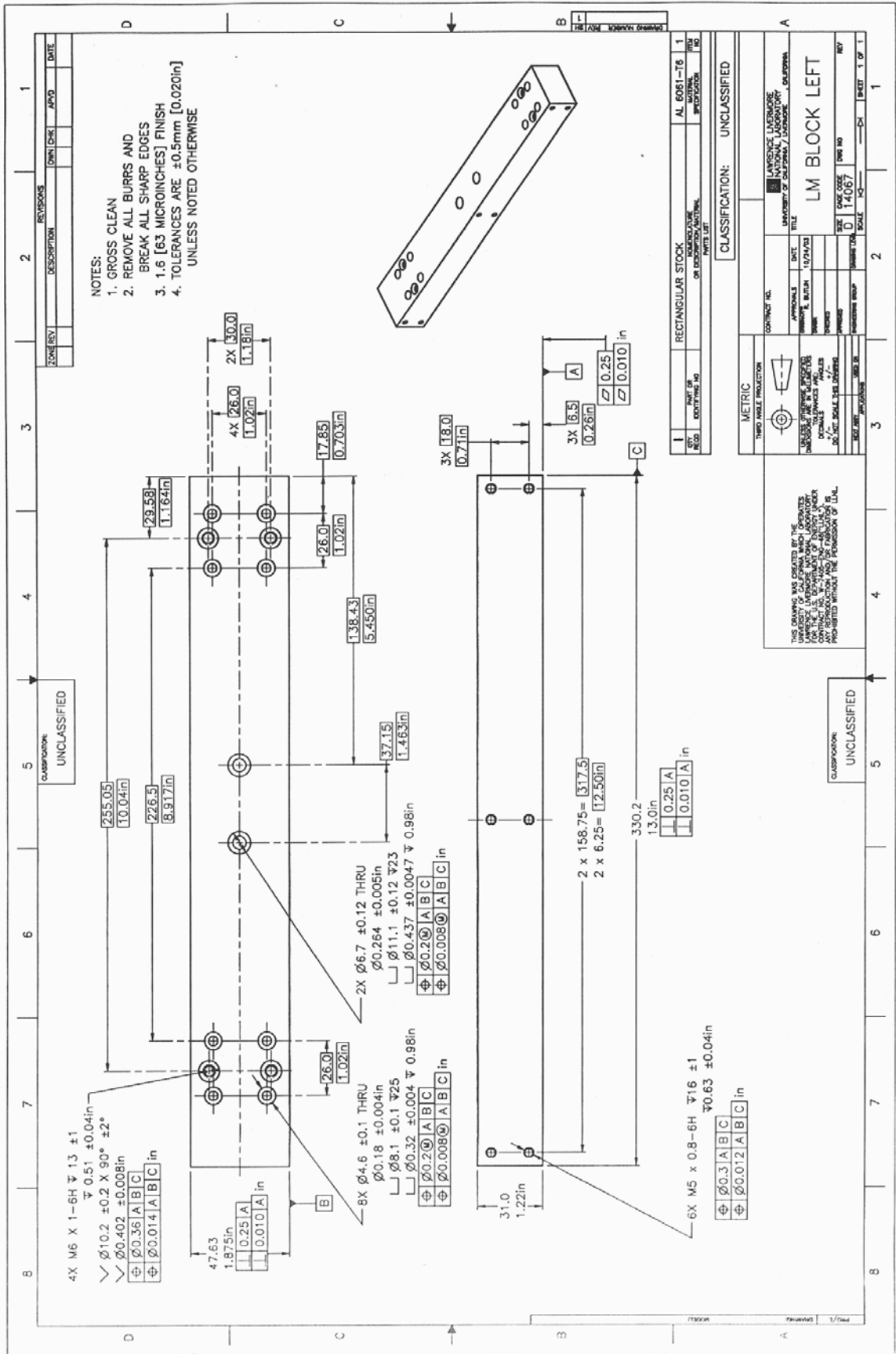
Appendix A-Reference Drawings

26 attached pages

BACK PLATE

AL 6061-T6
5/8" PLATE
METRIC

CLASSIFICATION: UNCLASSIFIED

LAWRENCE LIVERMORE NATIONAL LABORATORY

NOTES:
1. CROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.3mm [0.012in] UNLESS NOTED OTHERWISE

B – B

15.90 +0.10 / -0.00
0.626 +0.004 / -0.000in
2 PLACES

146.25 ±0.07
5.758 ±0.003 in

79.4 ±0.5
3.13 ±0.02 in

3.17 ±0.2
0.125 ±0.008 in

Ø15.1 ±0.2
Ø0.59 ±0.01 in
⌵ 6.0 ±0.2
0.24 ±0.008in

Ø9.0 ±0.1 THRU
Ø0.354 ±0.004 in
2 PLACES

4X 45° ±2° × 10 ± 2 [0.39 ±0.08in]

252.5
9.941in

200 ±1
7.87 ±0.04 in
2 PLACES

30.80 ±0.18
1.213 ±0.007 in
4 PLACES

7.95
0.313in

8X M5 ⌵12 ±0.5 [0.47 ±0.020in]
DO NOT BREAK THRU

26X M4 ⌵13.5 ±0.5 [0.53 ±0.020in]
⌖ Ø0.18 Ⓜ |A|B|C|
⌖ Ø0.0071 Ⓜ |A|B|C| in
DO NOT BREAK THRU

7.94
0.313in

35.0
1.38in

76.0
2.99in

35.0
1.38in

8X R6.35 ±2
R0.3 ±0.1 in

8X R25 ±5
R1.0 ±0.2 in

7.94
0.313in

60.0
2.36in
2 PLACES

109.5
4.31in
2 PLACES

305 ±0.5
12.01 ±0.02 in

768.35 ±1
30.25 ±0.04 in

218.3 ±1
8.59 ±0.04 in

15.88 ±0.2
0.625 ±0.008 in

24.00 ±0.16
0.945 ±0.006 in
2 PLACES

40.0 +0.7 / -0.0
1.57 +0.03 / -0.00in
2 PLACES

A – A

218.3 ±1
8.59 ±0.04 in

15.88 ±0.2
0.625 ±0.008 in

12 × 60= 720
12 × 2.362= 28.344in

⌖ 0.5 Ⓜ in
⌖ 0.020 Ⓜ in

"W" STK

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in] UNLESS NOTED OTHERWISE

REVISIONS
ZONE | REV | DESCRIPTION | DWN | CHK | APPD | DATE

RECTANGULAR STOCK.
NOMENCLATURE OR DESCRIPTION/MATERIAL
PARTS LIST

AL 6061-T6
MATERIAL SPECIFICATION

CLASSIFICATION: UNCLASSIFIED

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA, LIVERMORE, CALIFORNIA
TITLE
LM BLOCK LEFT

APPROVALS
DRAWN E. BUTLAN 10/24/13

SIZE D | CAGE CODE 14067 | DWG NO
SCALE | SHEET 1 OF 1

METRIC
THIRD ANGLE PROJECTION

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES ARE
ANGLES +/-
DO NOT SCALE THIS DRAWING

2X 30.0 [1.18in]
4X 26.0 [1.02in]
17.85 [0.703in]
29.58 [1.164in]
26.0 [1.02in]
138.43 [5.450in]
37.15 [1.463in]
255.05 [10.04in]
226.5 [8.917in]

4X M6 X 1-6H ▼ 13 ±1
▼ 0.51 ±0.04in
⌵ Ø10.2 ±0.2 X 90° ±2°
⌵ Ø0.402 ±0.008in
⊕ Ø0.36 | A | B | C
⊕ Ø0.014 | A | B | C | in

8X Ø4.6 ±0.1 THRU
Ø0.18 ±0.004in
⌴ Ø8.1 ±0.1 ▼25
⌴ Ø0.32 ±0.004 ▼ 0.98in
⊕ Ø0.2 Ⓜ | A | B | C
⊕ Ø0.008 Ⓜ | A | B | C | in

2X Ø6.7 ±0.12 THRU
Ø0.264 ±0.005in
⌴ Ø11.1 ±0.12 ▼23
⌴ Ø0.437 ±0.0047 ▼ 0.98in
⊕ Ø0.2 Ⓜ | A | B | C
⊕ Ø0.008 Ⓜ | A | B | C | in

47.63 [1.875in]
⏥ 0.25 | A
⏥ 0.010 | A | in
B

3X 18.0 [0.71in]
3X 6.5 [0.26in]
A
⏥ 0.25
⏥ 0.010 | in
C

2 x 158.75 = [317.5] [12.50in]
2 x 6.25 = [12.50in]
330.2 [13.0in]
⏥ 0.25 | A
⏥ 0.010 | A | in

31.0 [1.22in]

6X M5 x 0.8-6H ▼16 ±1
▼0.63 ±0.04in
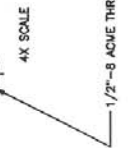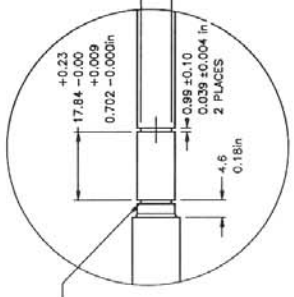⊕ Ø0.3 | A | B | C
⊕ Ø0.012 | A | B | C | in

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in] UNLESS NOTED OTHERWISE

RECTANGULAR STOCK

AL 6061-T6

CLASSIFICATION: UNCLASSIFIED

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA, LIVERMORE, CALIFORNIA

LM BLOCK RIGHT

DWG NO. 14067

METRIC
THIRD ANGLE PROJECTION

2X Ø6.7 ±0.1 THRU
Ø0.264 ±0.005 in
⌴ Ø11.1 ±0.12 ▽23
Ø0.437 ±0.0047 ▽ 0.98in
⊕ Ø0.2Ⓜ A B C
⊕ Ø0.008Ⓜ A B C in

6X M5 X 0.8-6H ▽16 ±1
▽0.63 ±0.04in
⊕ Ø0.3 A B C
⊕ Ø0.012 A B C in

4X M6-x-1-6H ▽ 13 ±1
▽ 0.51 ±0.04in   Ø10.2
⊻ Ø10.2 ±0.2 X 90° ±2°
⊻ Ø0.402 ±0.008in
⊕ Ø0.36 A B C
⊕ Ø0.014 A B C in

8X Ø4.6 ±0.1 THRU
Ø0.18 ±0.004in
⌴ Ø8.1 ±0.1 ▽27
Ø0.32 ±0.004 ▽ 0.98in
⊕ Ø0.2Ⓜ A B C
⊕ Ø0.008Ⓜ A B C in

⌖ 0.25
⌖ 0.010 in

3X 6.5
0.26in

3X 18.0
0.71in

330.2 ⌖ 0.25 A
13.0in   ⌖ 0.010 A in

2 x 158.75= 317.5
2 x 6.25= 12.50in

33.0
1.30in

29.58
1.164in

2X 30.0
1.18in

4X 26.0
1.02in

26.0
1.02in

17.85
0.703in

158.43
5.450in

255.05
10.04in

226.5
8.917in

37.15
1.463in

26.0
1.02in

Ø10.2

47.63
1.375in

⌖ 0.25Ⓜ A
⌖ 0.010Ⓜ A in

CLASSIFICATION: UNCLASSIFIED

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in] UNLESS NOTED OTHERWISE

$4X \vee \emptyset 12.0 \pm 0.2 \times 90° \pm 2°$
$\vee \emptyset 6.47 \pm 0.01$ in
⊕ $\emptyset 0.05$ Ⓜ |A|B|C|
⊕ $\emptyset 0.0024$ Ⓜ |A|B|C| in

45.53
1.79in

330.2
13.0in

255.05
10.04in

2X 7.25
0.31in

2X .30
1.18in

361.0
15.0in

0.5 B in
0.02 B

METRIC
THIRD ANGLE PROJECTION

304 SST
0.0351 IN THK (20 GA) SHEET

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA • LIVERMORE, CALIFORNIA

COVER SHEET LEFT

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in] UNLESS NOTED OTHERWISE

COVER SHEET RIGHT

304 SST
0.0351 IN THK (20 GA) SHEET

METRIC
THIRD ANGLE PROJECTION

4X ⌀12.0 ±0.2 X 90° ±2°
⌀6.0 ±0.2 in
⌀0.06 Ⓜ |A|B|C|
⌀0.0024 Ⓜ |A|B|C| in

330.2
13.0in

255.05
10.04in

29.52
1.16in

381.0
15.0in

2X 7.75
0.31in

2X [30]
1.18in

0.5 |B|
0.02 |B| in

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in] UNLESS NOTED OTHERWISE

874.4 ±1.0
34.42 ±0.04 in

74.2
2.92in

283.0
11.14in

23.9
0.94in

23.9
0.94in

283.0
11.14in

74.2
2.92in

6X R1.0 MAX
R0.04in

SEE VIEW A

4X SCALE

Ø9.525 +0.100 −0.000
Ø0.375 +0.004 −0.000in
4 PLACES

1/2"−8 ACME THREAD

Ø7.696 +0.025 −0.076
Ø0.303 +0.001 −0.003in
2 PLACES

17.84 +0.23 −0.00
0.702 +0.009 −0.000in

0.99 ±0.10
0.039 ±0.004 in
2 PLACES

4.6
0.18in

4X R0.2 MAX
R0.008in

VIEW A
SCALE 3X

303 SST

1/2" ACME THREAD

THREADED ROD

SIZE D   CAGE CODE   DWG NO   REV
              14067
SCALE   SHEET 1 OF 1

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA, LIVERMORE

METRIC
THIRD ANGLE PROJECTION

REVISIONS
ZONE | REV | DESCRIPTION | DATE | APPD

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.3mm [0.012in]
   UNLESS NOTED OTHERWISE

35.3
1.39in

[A]

3X M4 ⍋16 ±1 [0.63 ±0.04in]
3X 120° on Ø28.58 [1.125in]
⊕ | Ø0.3 Ⓜ | A | B |
⊕ | Ø0.012 Ⓜ | A | B | in

Ø20.0 THRU
Ø0.79in
⊕ | Ø0.30 Ⓜ | A | B |
⊕ | Ø0.012 Ⓜ | A | B |

[B]

50.8
2.00in

2X 45° ± 1° X 3
0.12in

19.07
0.751in

2X M6 ⍋19 ±1 [0.748 ±0.04in]

1" STK

18.58 ±0.12
0.731 ±0.005 in

12.70
0.500in

18.57 ±0.12
0.731 ±0.005 in

UNCLASSIFIED

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in]
   UNLESS NOTED OTHERWISE

31.0 +0.0 −0.5
1.22 +0.00 −0.02in

15.86 +0.00 −0.04
0.624 +0.000 −0.002in

$\boxed{A}$

Ø11.906 +0.000 −0.013
Ø0.4687 +0.0000 −0.0005in
⊕ | Ø0.2 | A | B
⊕ | Ø0.008 | A | B | in

2X 45° ±2° x 3 ±1 [0.12 ±0.04in]

7.00 ±0.05
0.276 ±0.002 in

2X Ø5.5 ±0.1 THRU
Ø0.217 ±0.004 in
⌴ 10.0 ±.1 ▽ 19.1 ± 1
⌴ 0.39 ±.004 ▽ 0.75 ±0.04in
⊕ | Ø0.2 Ⓜ | A | B
⊕ | Ø0.008 Ⓜ | A | B in

30.80
1.213in

50.8
2.00in

$\boxed{B}$

CLASSIFICATION: UNCLASSIFIED

METRIC
THIRD ANGLE PROJECTION

MOUNTED BEARING
(BEARING HOUSING)

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA / LIVERMORE

CLASSIFICATION: UNCLASSIFIED

AL 6061-T6
MATERIAL SPECIFICATION

5/8" PLATE
NOMENCLATURE OR DESCRIPTION/MATERIAL
PARTS LIST

2 QTY REQ'D

ITEM NO 1

SIZE D | CAGE CODE | DWG NO 14067 | REV
SCALE K:1 | SHEET 1 OF 1

REVISIONS
ZONE | REV | DESCRIPTION | DATE | DWN | CHK | APVD

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH

4.76 ±0.05
0.188 ±0.002 in

9.525 +0.120 −0.000
0.375 +0.005 −0.000in

CLASSIFICATION: UNCLASSIFIED

REVISIONS
ZONE | REV | DESCRIPTION | DWN/CHK | APVD | DATE

QTY REQD	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION/MATERIAL	MATERIAL SPECIFICATION	ITEM NO
2	5527K19	3/8" PEEK BUSHING	McMASTERS	1

PARTS LIST

METRIC
THIRD ANGLE PROJECTION

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES ON
DECIMALS ANGLES
±
DO NOT SCALE THIS DRAWING

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA / LIVERMORE, CALIFORNIA

TITLE
PEEK BUSHING

SIZE D | CAGE CODE | DWG NO 14067 | REV
SCALE | SHEET 1 OF 1

CONTRACT NO.
APPROVALS | DATE
DRAWN R. BUTLIN 11/12/03
CHECKED
APPROVED
ENGINEERING GROUP

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in] UNLESS NOTED OTHERWISE

25.4
1.00in

⁵/₈″ STK

[A]

| 2 | PART OR IDENTIFYING NO | 5/8" PLATE | NOMENCLATURE OR DESCRIPTION/MATERIAL | AL 6061–T6 | 1 |
| QTY REQD | | | PARTS LIST | MATERIAL SPECIFICATION | ITEM NO |

CLASSIFICATION: UNCLASSIFIED

METRIC

THIRD ANGLE PROJECTION

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA · LIVERMORE, CALIFORNIA

TITLE
MOUNTED BRG B
(BEARING HOUSING)

| SIZE D | CAGE CODE | DWG NO 14-067 | REV |
| SCALE | | SHEET 1 OF 1 | |

APPROVALS
DRAWN R. BUTLIN DATE 11/20/03

THIS DRAWING WAS CREATED BY THE UNIVERSITY OF CALIFORNIA WHICH OPERATES LAWRENCE LIVERMORE NATIONAL LABORATORY UNDER CONTRACT NO. W-7405-ENG-48 WITH U.S. DEPARTMENT OF ENERGY. ANY REPRODUCTION AND OR FABRICATION IS PROHIBITED WITHOUT THE PERMISSION OF LLNL.

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS
DECIMALS   ANGLES
±/–   ±/–
DO NOT SCALE THIS DRAWING

+0.07
4.76 –0.00
+0.003
0.187 –0.000in

+0.07
4.76 –0.00
+0.003
0.188 –0.000in

30.0°

7.00 ±0.05
0.276 ±0.002 in

2X 45° ±2° x 3 ±1 [0.12 ±0.04in]

+0.000
Ø11.906 –0.013
+0.0000
Ø0.4687 –0.0005in

⊕ Ø0.2 | A | B
⊕ Ø0.200 | A | B in

[B]

30.80
1.213in

50.8
2.00in

2X Ø5.5 ±0.1 THRU
Ø0.217 ±0.004 in
⌴ 10.0 ±.1 ▽ 12.7 ±1
0.39 ±.004 ▽ 0.50 ±0.04in
⊕ Ø0.2 Ⓜ | A | B
⊕ Ø0.008 Ⓜ | A | B in

Note: This was the only drawing created by one of the NIF designers in Pro E. I was the engineer to check and approve the drawing (back when my last name was Patton)
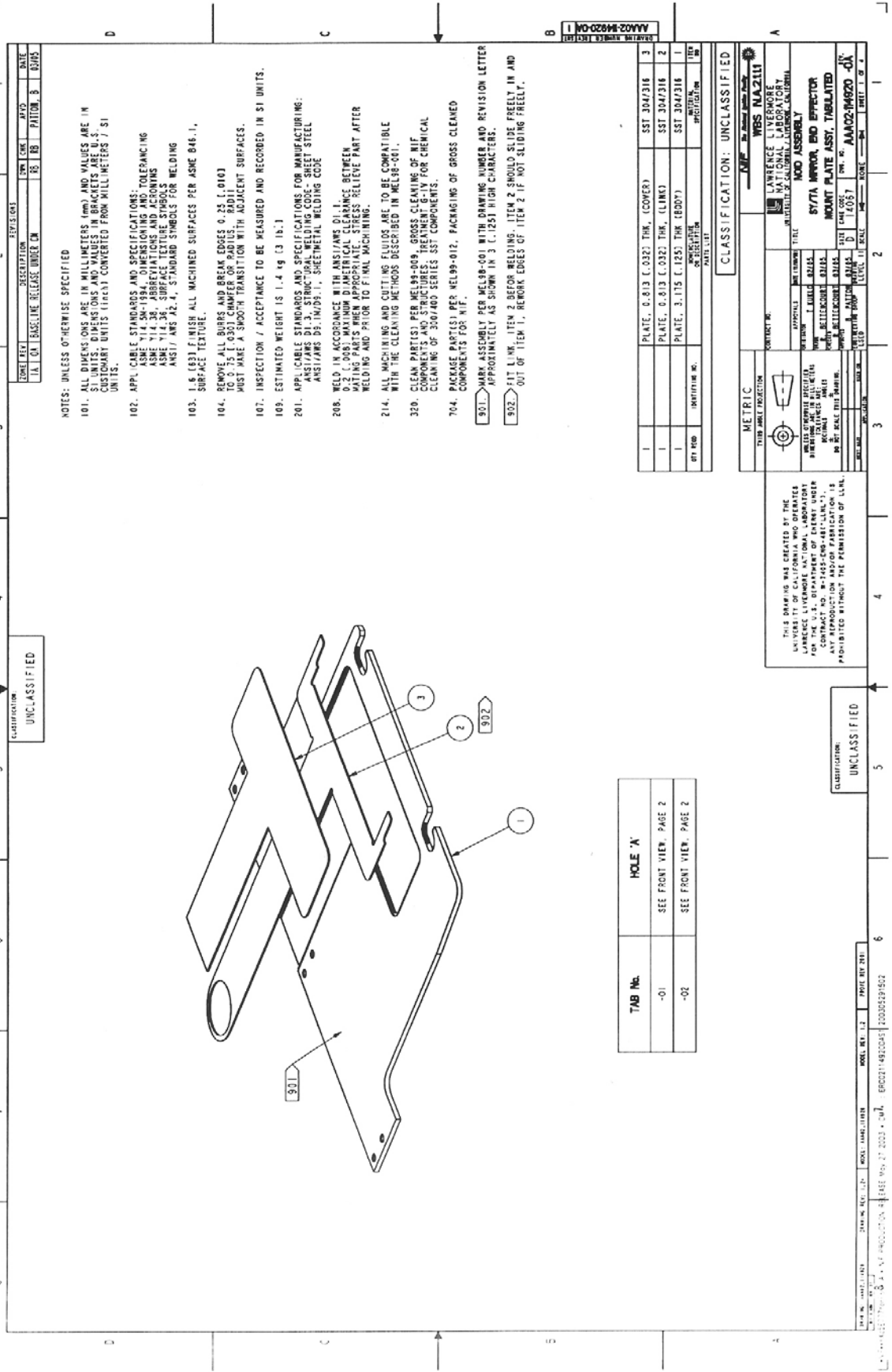
NOTES: UNLESS OTHERWISE SPECIFIED

101. ALL DIMENSIONS ARE IN MILLIMETERS (mm) AND VALUES ARE IN SI UNITS. DIMENSIONS AND VALUES IN BRACKETS ARE U.S. CUSTOMARY UNITS (inch) CONVERTED FROM MILLIMETERS / SI UNITS.

102. APPLICABLE STANDARDS AND SPECIFICATIONS:
ASME Y14.5M-1994, DIMENSIONING AND TOLERANCING
ASME Y14.38, ABBREVIATIONS AND ACRONYMS
ASME Y14.36, SURFACE TEXTURE SYMBOLS
ANSI/AWS A2.4, STANDARD SYMBOLS FOR WELDING

103. 1.6 [63] FINISH ALL MACHINED SURFACES PER ASME B46.1, SURFACE TEXTURE.

104. REMOVE ALL BURRS AND BREAK EDGES 0.25 [.010] TO 0.75 [.030] CHAMFER OR RADIUS. RADII MUST MAKE A SMOOTH TRANSITION WITH ADJACENT SURFACES.

107. INSPECTION / ACCEPTANCE TO BE MEASURED AND RECORDED IN SI UNITS.

109. ESTIMATED WEIGHT IS 1.4 kg [3 lb.]

201. APPLICABLE STANDARDS AND SPECIFICATIONS FOR MANUFACTURING:
ANSI/AWS D1.3, STRUCTURAL WELDING CODE- SHEET STEEL
ANSI/AWS D9.1M/D9.1, SHEETMETAL WELDING CODE

208. WELD IN ACCORDANCE WITH ANSI/AWS D1.1.
0.2 [.008] MAXIMUM DIAMETRICAL CLEARANCE BETWEEN MATING PARTS WHEN APPROPRIATE. STRESS RELIEVE PART AFTER WELDING AND PRIOR TO FINAL MACHINING.

214. ALL MACHINING AND CUTTING FLUIDS ARE TO BE COMPATIBLE WITH THE CLEANING METHODS DESCRIBED IN MEL98-001.

320. CLEAN PART(S) PER MEL99-009, GROSS CLEANING OF NIF COMPONENTS AND STRUCTURES TREATMENT G-IV FOR CHEMICAL CLEANING OF 300/400 SERIES SST COMPONENTS.

704. PACKAGE PART(S) PER MEL99-012, PACKAGING OF GROSS CLEANED COMPONENTS FOR NIF.

901. MARK ASSEMBLY PER MEL98-001 WITH DRAWING NUMBER AND REVISION LETTER APPROXIMATELY AS SHOWN IN 3 [.125] HIGH CHARACTERS.

902. FIT LINK, ITEM 2 BEFOR WELDING. ITEM 2 SHOULD SLIDE FREELY IN AND OUT OF ITEM 1. REMORK EDGES OF ITEM 2 IF NOT SLIDING FREELY.

TAB No.	HOLE 'A'
-01	SEE FRONT VIEW, PAGE 2
-02	SEE FRONT VIEW, PAGE 2

	DESCRIPTION		
	PLATE, 0.813 [.032] THK, (COVER)	SST 304/316	3
	PLATE, 0.813 [.032] THK, (LINK)	SST 304/316	2
	PLATE, 3.175 [.125] THK (BODY)	SST 304/316	1

CLASSIFICATION: UNCLASSIFIED

WBS NA2111

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA, LIVERMORE, CALIFORNIA

MOD ASSEMBLY
SY/TA MIRROR, END EFFECTOR
MOUNT PLATE ASSY, TABULATED

AAA02-IM920 -0X

METRIC
THIRD ANGLE PROJECTION

CLASSIFICATION: UNCLASSIFIED

ZONE	REV	DESCRIPTION	DWN	CHK	APVD	DATE
1A	OA	BASELINE RELEASE UNDER CM	IB	BB	PATTON	03/05

SECTION C-C

0.15 (.005)

1.87±0.2 CUT
(.046)±.0078]

12.6±0.2 CUT
(.49)±.007]

4X R 0.4 MAX

2±0.12
(.079)±.004]

3.175
(.125001)

DETAIL B
2X ITEM 1 SHOWN

2X R1.1±0.3
(.061)±.011]

16.7±0.2
(.657)±.007]

3±0.2
(.118)±.007]

4.5±0.2
(.17)±.007]

2.5±0.3 X 45°±3°

R6 +0.22
-0.05
[.236 +.008
-.001]

⌖ ⌀0.2 (.007) Ⓜ A B Ⓜ C

⌖ ⌀0.2 (.007) Ⓜ A B Ⓜ C

266.78±1
(10.503±.039]

⌖ ⌀0.2 (.007) Ⓜ A B Ⓜ C

152.4
[6.000]

Right view / TAB-02

(259)
(10.196853]

129.5±0.5
(5.098±.019]
TO THEOR SHARP CORNER

2X R10±0.5
(.39±.019]
TAB-02 ONLY

167.18±1
(6.582±.039]

TAB-02 SHOWN

FRONT VIEW
ITEM 1, TAB-01 SHOWN

R10 MAX
(.3937)
TAB-01 ONLY

2X R6±0.5
(.236±.019]
TAB-01 ONLY

30±0.5
(1.18)±.019]
CUT OUT
TAB-01 ONLY

SEE DETAIL B

129.5±0.5
(5.098±.019]
TO THEOR SHARP CORNER

R10±0.5
(.39±.019]
TAB-01 ONLY

55±0.2
(2.205±.007]

9±0.5
(.354±.019]
CUT OUT
TAB-01 ONLY

22.5±0.5
(.120±.019]

2X (209.71)
(8.256]

4X R±0.12
(.238±.004]

2X R10±0.5
(.394±.019]

8 [.108]

3X 8 [.108]

135.7±2
(5.34±.007]

167.18±1
(6.582±.039]

3X 6.35 [.250]

2X 48.6±1
(1.913±.039]

Φ 2±0.12
(.079±.004]

43.4±0.2
(1.709±.007]

10.2±0.2
(.402±.007]

82.3±0.2
(3.240±.007]

HOLE 'A'

6X ⌀5.5 +0.18
(.217±.007] THRU

⌀10.2±0.2 X 90°±0.5°
(.402±.007] X 90°±0.5°
ON FAR FACE FOR TAB-01 SHOWN
ON NEAR FACE FOR TAB-02

⌖ ⌀0.2 (.007) Ⓜ A B Ⓜ C

2X 317.5
(12.500]

330.1±1
(13.00±.039]

LINK DETAIL

UNCLASSIFIED

DETAIL A
SCALE 4.000

SEE DETAIL A

4X R1.6 [.063] MAX
CORNER RELIEF

R2.5±0.5 [.098±.019]

7.5±0.5 [.295±.019]

15°±5°

3±0.2 [.118±.007]

18±0.5 [.70±±.019]

1±0.2 [.43±.007]

163.4±0.2 [6.433±.007]

141.4±0.2 [5.561±.007]

30.8±0.2 [1.213±.007]

55.8±0.2 [2.197±.007]

R4±0.5 [.236±.019]

281.1±0.2 [11.061±.007]

60.3±0.2 [2.371±.007]

R20.9±0.2 [.82±.007]

Ø3±0.3 [1.378±.011]

0.15 [.005] | A | B | C

0.15 [.005] | A | B | C

0.15 [.005]

0.813 [.032]

B

C

WBS NA.2111
AAA02-M920 -0A

UNCLASSIFIED

(330.2)
[13.00000]

(152.4)
[6.00000]

(5.4)
[.21260]

A

A

(302)
[11.88976]

(59.6)
[2.34646]

3.18 STK
[.125]

LASER WELD

0.81±0.12
[.032±.004]

2±0.12
[.079±.004]

(1.187)
[.04683]

// 0.15 [.0051] A

ITEM 1 ONLY

0.2 MAX GAP
[.008]

0.76
[.030]

⊥ [.030] M

SECTION A-A

3
2
1

(1.175)
[.04626]

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in]
   UNLESS NOTED OTHERWISE

5/8" STK

34.00 ±0.05
1.339 ±0.002 in

⌀15.080 −0.013 / +0.000
⊕ ⌀0.5 A B
⌀0.5937 −0.0005in / +0.0000
⊕ ⌀0.02 A B

40.0
1.57in

B

12.0
0.472in

12.0
0.472in

2X M8 ▼ 1B ±1 [0.71 ±0.04in]
⊕ ⌀0.24 Ⓜ A B
⊕ ⌀0.009 Ⓜ A B in

A

REVISIONS
ZONE REV DESCRIPTION DWN CHK APVD DATE

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in]
   UNLESS NOTED OTHERWISE

⅝" STK

$\varnothing$15.080 $^{+0.000}_{-0.013}$
⊕ $\varnothing$0.5 A B
$\varnothing$0.5937 $^{+0.0000}_{-0.0005}$in
⊕ $\varnothing$0.02 A B

19.05 ±0.05
0.750 ±0.002 in

40.0
1.57in

2X M8 ⊽14 ±0.5 [0.55 ±0.02in]
⊕ $\varnothing$0.24 Ⓜ A B
⊕ $\varnothing$0.009 Ⓜ A B in

12.0
0.472in

12.0
0.472in

A

B

AL 6061-T6

5/8" PLATE

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA, LIVERMORE, CALIFORNIA

TITLE
HINGE SHORT

SIZE D   CAGE CODE 14067   DWG NO
SCALE   SHEET 1 OF 1

METRIC
THIRD ANGLE PROJECTION

REVISIONS
ZONE REV   DESCRIPTION   DWN CHK APVD   DATE

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.3mm [0.012in] UNLESS NOTED OTHERWISE

34.00 ±0.16
1.339 ±0.006 in

24.00 ±0.16
0.945 ±0.006 in

56.3
2.22in

38.1
1.5in

30.8
1.21in

127.00 ±0.10
5.000 ±0.004 in

12.70 ±0.10
0.500 ±0.004 in

76.20 ±0.10
3.000 ±0.004 in

4X Ø9.0 THRU
Ø14.5 ⯆ 9.0
Ø0.35 THRU
Ø0.57in ⯆ 0.35in

4X R6.35 MAX
R0.25in

3/8–24" INTERNAL THREAD (KEENSERT) THREADED FROM THIS SIDE ONLY

6X Ø9.0 THRU ⨅ Ø14.5 ⯆ 8.0
Ø0.35in THRU ⨅ Ø0.57in ⯆ .31in

4X 45° ±2° X 2 [0.079in]

3/4"

54.61
2.150in

76.20
3.00in

50.8
2.00in

50.8
2.00in

152.4
6.00in

Ø88.90 ±0.16
Ø3.500 ±0.006 in

6X 60.0°

202.6 ±1.0
7.98 ±0.04 in

PITCH PLATE

AL 6061–T6
MATERIAL SPECIFICATION

3/4" PLATE

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA / LIVERMORE, CALIFORNIA

DWG NO D 14067

SIZE D   CAGE CODE
SCALE   SHEET 1 OF 1

METRIC
THIRD ANGLE PROJECTION

APPROVALS   DATE
DRAWN   R. BUTLER   10/27/03

REVISIONS
ZONE REV   DESCRIPTION   DWN CHK   APVD   DATE

DRAWING NUMBER   REV SH

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.3mm [0.012in]
   UNLESS NOTED OTHERWISE

107.7
4.24in

12.70
0.50in

A – A

3/4" PLATE

3/4" STK

207.1 ±1.0
8.15 ±0.04 in

9.53 ±0.05
0.375 ±0.002 in
2 PLACES

50.50 ±0.10
1.988 ±0.004 in
2 PLACES

4X Ø12.685 ±0.005 ⌴19.05
Ø0.4994 ±0.0002 ⌴0.75in

4X R6.35 MAX
R0.25in

4X 45° ±2° X 2 [0.079in]

9.53 ±0.05
0.375 ±0.002 in
3 PLACES

53.87 ±0.05
2.121 ±0.002 in
3 PLACES

28.5
1.12in

28.5
1.12in

28.5
1.12in

+0.0
127.0 –0.2
+0.00
5.00 –0.01in

42.0
1.65in

A

A

REVISIONS

ZONE | REV | DESCRIPTION | DWN/CHK | APVD | DATE

AL 6061-T6
MATERIAL SPECIFICATION

NOMENCLATURE
OR DESCRIPTION/MATERIAL
PARTS LIST

PART OR
IDENTIFYING NO

QTY
REQD

METRIC

THIRD ANGLE PROJECTION

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES ARE:
DECIMALS   ANGLES
±/– 0.3   +/– 0.5
±/– 0.5
DO NOT SCALE THIS DRAWING

CONTRACT NO.

APPROVALS   DATE
DRAWN   R. BUTLIN   10/27/03
CHECKED
APPROVED
ENGINEERING GROUP

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA · LIVERMORE, CALIFORNIA

TILT PLATE

SIZE CAGE CODE   DWG NO
D   14067

SCALE   SHEET 1 OF 1   REV

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.02in]
   UNLESS NOTED OTHERWISE

⌀5.540 ±0.008 ▽ 2.00 ±0.05
⌀0.2181 ±0.0003 in ▽ 0.079 ±0.002in

3/8"–24 FULLY THREADED ROD

102 ±2
4.0 ±0.1 in

–2.0–

REVISIONS
ZONE | REV | DESCRIPTION | DWN | CHK | APVD | DATE

3/8" FINE THREADED ROD
BRASS

CLASSIFICATION: UNCLASSIFIED

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA , LIVERMORE

PITCH THREAD

SIZE D | CAGE CODE | DWG NO 14067 | REV
SCALE | SHEET 1 OF 1

METRIC
THIRD ANGLE PROJECTION

DRAWN R. BUTLAR DATE 10/26/03

APPROVALS

PART OR
IDENTIFYING NO | NOMENCLATURE
OR DESCRIPTION

QTY
REQD

ITEM
NO 1

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. ROUGHEN THE OUTER
   SURFACE OF THE LARGE
   DIA

$\varnothing 1/4''$ STK

$\varnothing 1/2''$ STK

1.0 ±0.2
0.04 ±0.01 in

$3/4''$ STK

REVISIONS
ZONE | REV | DESCRIPTION | DWN | CHK | APVD | DATE

TUNGSTEN CARBIDE BUSHING
REID TOOL

NOMENCLATURE OR DESCRIPTION
MATERIAL SPECIFICATION
PARTS LIST

METRIC
THIRD ANGLE PROJECTION

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA / LIVERMORE

TITLE

TC BUSHING

SIZE CAGE CODE DWG NO
D   14067

SCALE   SHEET 1 OF 1

ADAPTER BLOCK

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. CHAMFER ALL 6 EDGES OF THE BLOCK 45° ±3° X 2mm [0.012in]
5. TOLERANCES ARE ±0.3mm [0.3mm [0.012in] UNLESS NOTED OTHERWISE

MATERIAL SPECIFICATION: AL 6061-T6

METRIC
THIRD ANGLE PROJECTION

1/2-20 INTERNAL THREAD (KEENSERT) ▼ THRU TO NEXT FACE

DWG NO 14-067
SIZE D
SHEET 1 OF 1

REVISIONS

ZONE	REV	DESCRIPTION	DMN	CHK	APPD	DATE

UNCLASSIFIED

Ø114.3 ±1.0
⊕ | Ø1.0 Ⓜ | A | B Ⓜ
⊕ | Ø4.500 ±0.039 in | A | B Ⓜ
⊕ | Ø0.0 Ⓜ | A | B Ⓜ

Ø41.275 ±0.013
▯ | Ø0.13 Ⓜ | A
⊕ | Ø1.6250 ±0.0005 in | A | B Ⓜ
⊕ | Ø0.005 Ⓜ | A | B Ⓜ

6X M8 X 1.25–6H THRU
ON Ø 88.9 [3.50in]
⊕ | Ø0.6 Ⓜ | A | B Ⓜ
⊕ | Ø0.02 Ⓜ | A | B Ⓜ

6X 60.0°

STANDARD THREAD RELIEF

Ø34.849 ±0.013
⊕ | Ø0.13 Ⓜ | A | B Ⓜ
⊕ | Ø1.3720 ±0.0005 in | A | B Ⓜ
⊕ | Ø0.005 Ⓜ | A | B Ⓜ
| B |

146.8 ±0.5
5.78 ±0.02 in

128.8 ±0.5
5.07 ±0.02 in

R3.2
R0.125in

19.05
0.750in

R2.0
R0.08in

53.8
2.12in

19.0
0.75in

15.6 ±1.0
0.61 ±0.04 in

1.25–12UNF
⊕ | Ø1.2 Ⓜ | A | B Ⓜ
⊕ | Ø0.05 Ⓜ | A | B Ⓜ in

▱ | 0.013 Ⓜ
▱ | 0.0005 Ⓜ in

∥ | 0.13 Ⓜ | A
∥ | 0.005 Ⓜ | A in

| A |

4.762 ±0.020
4.763 ±0.020

11.48 ±0.20
0.452 ±0.008 in

A – A

A   A

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. KEY SLOT MUST BE PLACED IN POSITION SHOWN W.R.T THE THREADED HOLES
5. TOLERANCES ARE ±0.3mm [0.012in] UNLESS NOTED OTHERWISE

CLASSIFICATION: UNCLASSIFIED

NOMENCLATURE OR DESCRIPTION/NATIONAL PARTS LIST

304 SST — MATERIAL SPECIFICATION

4.5" DIA STK

| QTY REQD | PART OR IDENTIFYING NO | | |

METRIC
THIRD ANGLE PROJECTION

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA , LIVERMORE , CALIFORNIA

TITLE: SHAFT

APPROVALS — DATE
DRAWN K. BUTLER 11/13/03

SIZE D   CAGE CODE   DWG NO 14067   REV
SCALE   SHEET 1 OF 1

UNCLASSIFIED

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.02in]
   UNLESS NOTED OTHERWISE

A – A

19.05 +0.00 +0.20
0.750 +0.000 +0.008in

68.0 +0.7 −0.0

4.820 ±0.025
0.1898 ±0.0010 in

4.820 ±0.025
0.1898 ±0.0010 in

Ø35.0 ±0.1
Ø1.378 ±0.004 in

21.7 ±0.3
0.854 ±0.012 in

2X R0.15 MAX
R0.006in

49.0 ±0.5
1.93 ±0.02 in

12.0 +0.1 −0.0
0.472 +0.004 −0.000in

25.4 ±0.3
1.00 ±0.01 in

Ø63.5 ±1.0
Ø2.50 ±0.04 in

57.0 +0.1 −0.5
2.24 +0.00 −0.02in

AL 6061-T6
MATERIAL SPECIFICATION

3/4" STK

CLASSIFICATION: UNCLASSIFIED

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA / LIVERMORE, CALIFORNIA

FORCE RING

CAGE CODE DWG NO
14067

SIZE D   REV
SCALE 1 OF 1

CONTRACT NO.
APPROVALS   DATE
DRAWN R. BUTLER   11/13/03

METRIC
THIRD ANGLE PROJECTION

THIS DRAWING WAS CREATED BY THE
UNIVERSITY OF CALIFORNIA WHICH OPERATES
LAWRENCE LIVERMORE NATIONAL LABORATORY
FOR THE U.S. DEPARTMENT OF ENERGY UNDER
CONTRACT NO. W-7405-ENG-48. THE U.S.
GOVERNMENT RETAINS AND THE PUBLISHER, LLNL'S
ANY REPRODUCTION AND/OR THE FOR LLNL'S
PROHIBITED WITHOUT THE PERMISSION OF LLNL.

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.3mm [0.012in]
   UNLESS NOTED OTHERWISE

16.85
0.663in

Ø11.906 +0.000 −0.013
Ø0.4687 +0.0000 −0.0005in

9.53
0.375in

11.000 ±0.025
0.4331 ±0.0010 in

2X R2.5 MAX
R0.10in

21.00 ±0.10
0.827 ±0.004 in

2X R3.0 ±1.0
R0.12 ±0.004 in

46.0 ±1.0
1.81 ±0.04 in

6.35
0.25in

6.35
0.25in

9.53 ±0.10
0.375 ±0.004 in

34.5 +0.5 −0.0
1.36 +0.02 −0.00in

Ø9.525 +0.000 −0.010
Ø0.3750 +0.0000 −0.0004in

STK

3/4" STK

AL 6061–T6
MATERIAL SPECIFICATION

3/4" STK

METRIC
THIRD ANGLE PROJECTION

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN MILLIMETERS
TOLERANCES ARE:
DO NOT SCALE THIS DRAWING

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA, LIVERMORE, CALIFORNIA

FORK

CAGE CODE: 14067
SIZE D
DWG NO
SCALE
SHEET 1 OF 1

DRAWN R. BUTLIN   DATE 11/13/03

APPROVALS   DATE

REVISIONS
ZONE | REV | DESCRIPTION | DWN | CHK | APPD | DATE

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH

A – A

102 ±3
4.00 ±0.12 in

25.4 ±1.0
1.00 ±0.04 in
+0.23
12.98 −0.00
+0.009
0.511 −0.000in

4X R0.2 MAX
R0.008in

+0.10
0.99 −0.00
+0.004
0.039 −0.000in
2 PLACES

+0.025
Ø7.696 −0.076
+0.001
Ø0.303 −0.003in
2 PLACES

45° ±2° X 1 [0.04in]

+0.0
5.0 +0.5
+0.00
0.20 +0.02in

R1.0 ±0.5
R0.04 ±0.02 in

1/2"–13 THREAD
+0.100
Ø9.525 −0.000
+0.004
Ø0.375 −0.000in

A — A

CLASSIFICATION: UNCLASSIFIED

UNCLASSIFIED

	QTY REQD	PART OR IDENTIFYING NO	NOMENCLATURE OR DESCRIPTION PARTS LIST	MATERIAL SPECIFICATION
1	1	1/2–13 THREADED ROD		Brass

METRIC
THIRD ANGLE PROJECTION

UNLESS OTHERWISE SPECIFIED DIMENSIONS ARE IN MILLIMETERS TOLERANCES ARE: ANGLES ±
DECIMALS ±
DO NOT SCALE THIS DRAWING

LAWRENCE LIVERMORE NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA / LIVERMORE, CALIFORNIA

TITLE
TWIST THREAD

SIZE	CAGE CODE	DWG NO	REV
D	14067		

SCALE 1:2 — CH SHEET 1 OF 1

APPROVALS
DRNGR/DL BUTUH   DATE 11/12/03

REVISIONS

ZONE	REV	DESCRIPTION	DWN	CHK	APPVD	DATE

CLASSIFICATION: UNCLASSIFIED

UNCLASSIFIED

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±1.0mm [0.04in]
   UNLESS NOTED OTHERWISE

88.9
3.5in

4X ⌀4.0 ±0.4 THRU
⌀0.16 ±0.02 in

127.0
5.0in

101.6 ±0.3
101.60 ±0.30 mm

63.5 ±0.3
2.50 ±0.01 in

12.7
0.50in

12.7
0.50in

REVISIONS

ZONE	REV	DESCRIPTION	DWN	CHK	APVD	DATE

AL 6061-T6

1/16" STK SHEET

METRIC
THIRD ANGLE PROJECTION

THIS DRAWING WAS CREATED BY THE
UNIVERSITY OF CALIFORNIA WHICH OPERATES
LAWRENCE LIVERMORE NATIONAL LABORATORY
FOR THE U.S. DEPARTMENT OF ENERGY UNDER
CONTRACT NO. W-7405-ENG-48. ANY
REPRODUCTION AND/OR FABRICATION IS
PROHIBITED WITHOUT THE PERMISSION OF LLNL.

LAWRENCE LIVERMORE
NATIONAL LABORATORY
UNIVERSITY OF CALIFORNIA / LIVERMORE, CALIFORNIA

CLASSIFICATION: UNCLASSIFIED

TITLE

COVER

SIZE CASE CODE   DWG NO
D   14.067

SCALE   SHEET 1 OF 1

DATE 11/13/03

Appendix B-Supporting Calculations

55 attached pages

Weight Definitions:

W_1 = 276 lb (Weight of largest MSA + Side Mounting Plates)

W_2 = 6 lb (Weight of 2 LM Mounts)

W_3 = 26.5 lb (W_2 + Weight of THK Rails, LM Blocks, Threaded Rod, and Rail Plate)

W_4 = 3 lb (Weight of Tilt Plate and Hinges)

W_5 = 6.7 lb (W_4 + Weight of Pitch Plate and Pitch Crank)

W_6 = 17.9 lb (Weight of Shaft + Adapter Block)

Table B1 Safety Factors calculated for the MSA NOID End-Effector

Component Description	Failure Mode	S.F. Static	S.F. Dynamic
Rail Sub-Assembly			
Side Mounting Plates	Bending stress	133.3	48.1
	Von Mises Stress in the tangs	N/A	1.6
Screws securing Side Mounting Plates to the LM Mounts (M5)	Von Mises Bolt Stress	8.3[1]	2.9[1]
Screws securing the LM Blocks to the LM Mounts (M4)	Von Mises Bolt Stress	7.9[1]	3.1[1]
THK Rails	Manufacturer's rated load	16.7	N/A
Screws securing the THK Rails to the Back Plate (M4)	Von Mises Bolt Stress	7.3[1]	2.8[1]
	Al thread stripping	18.4[1]	N/A
Screws securing the Hinges to the Back Plate (M8)	Von Mises Bolt Stress	5.8[1]	2.1[1]
	Al thread stripping	27.7[1]	10.0[1]
Tilt Sub-Assembly			
Tilt Plate Pins	Von Mises Stress	4.0[1]	1.6[1]
Screws securing the Hinges to the Pitch Plate (M8)	Von Mises Bolt Stress	4.2[1]	1.6[1]
Threaded Rod through the Pitch Plate	Brass thread stripping	13.5[1]	5.0[1]
	Max force to turn handle	13.1 lbf	N/A
Tungsten Carbide ball	Contact stress	1.1[2]	N/A
Twist Sub-Assembly			
Screws securing the Pitch Plate to the Shaft (M8)	Von Mises Bolt Stress	4.4[1]	1.6[1]
	SST thread stripping	5.4	3.3
Timken Bearing #526/522	Manufacturer's rated load	6.5	N/A
Timken Bearing #23620/23690	Manufacturer's rated load	4.5	N/A
Shaft	Von Mises Stress (pt A)	3.7	1.6
	Von Mises Stress (pt B)	4.4	N/A[3]
	Von Mises Stress (pt C)	4.3[1]	1.1
Shaft Key	Shear	4.8[1]	1.9[1]
Force Ring Pin	Shear	4.6[1]	1.8[1]
Force Ring	Shear at the slot	2.9[4]	1.1
	Von Mises at the base	4.4	1.7
Fork	Axial Pull Stress	7.2	2.8
Threaded Rod through the Adapter Block	Axial Pull Stress	3.8[1,4]	1.5[1]
	Brass thread stripping	12.4[1]	4.8[1]
	Max force to turn handle	7.6 lbf	N/A
E-Clip	Shear	3.4[1,4]	1.3[1]
Screws securing the Adapter Block to the Robohand (M8)	Von Mises Bolt Stress	4.5[1]	2[1]
	Al thread stripping	9.4	3.6

1. Ultimate Strength used, in most cases the Ultimate Strength is all the manufacturer provided

2. Not a critical part and yielding would be acceptable

3. Calculation was not needed, see notes within the calculations

4. Values are slightly lower than the Safety Factors recommended in the MEDSS, but the stress on these components are primarily a result of a twist moment caused by the unlikely event of the MSA being 3" off center on the rails

Table B2 MSA NOID End-Effector purchased items

Part	Sub-Assembly	Quantity	Manufacturer	Description
Twist Thread	Twist	1	McMasters	Brass, 1/2-20 Thread
Keensert	Twist	1	LLNL Shop	SST, 1/2-20 Inner Thread
Delrin Washer	Twist/Rail	4	McMasters	3/8" ID, 0.625 OD, 0.031" THK
E-Clips	Twist/Rail	4	McMasters	3/8" Stainless Steel
Peek Flanged Bearing	Twist	1	McMasters	3/8" ID, 15/32 OD, 1/2" Lg
Peek Flanged Bearing	Rail	4	McMasters	3/8" ID, 15/32 OD, 3/4" Lg
Pin	Twist	1	Fabroy	3/8" DIA Oversized, 1.25" Lg, A2 or A4 equivalent
Peek Sleeve Bearing	Twist	1	McMasters	3/8" ID, 15/32 OD, 3/4" Lg
Open Cage Roller Bearing	Twist	1	Timken	4" OD
Open Cage Roller Bearing	Twist	1	Timken	2.875" OD
Clamp Collar	Twist	1	King Bearing	1 1/4-12 SST
Key stock	Twist	1	McMasters	3/8"x3/8" 303 (8-18) oversized
Pitch Thread	Tilt	1	McMasters	Brass, 3/8" -24
Keensert	Tilt	1	LLNL Shop	SST, 3/8" -24
Tungsten Carbide Ball	Tilt	1	McMasters	7/32" DIA Ball
Tungsten Carbide Bushing	Tilt	1	Reid Tool	1/4" ID, 1/2" OD, 3/4" Lg
Pins	Tilt	4	Fabroy	1/2" DIA Oversived, 1.5" Lg, A2 or A4 equivalent
Peek Flanged Bearing	Tilt/Rail	4	McMasters	1/2" ID, 19/32" OD, 3/4" Lg
Rail Thread	Rail	1	McMasters	1/2" Precision Modified-ACME Lead Screw, SST
Flanged Nut	Rail	1	McMasters	1/2" ACME Nut, wear compensating, Polyacetal
Belville Washers	Rail	8	McMasters	1/4" screw size, SST, Serrated
Rail with two LM Blocks	Rail	2	THK	Rails and two LM Mounts, 760mm Lg
Pin	Rail	4	McMasters	M2 DIA, 10mm 18-8 SST
All Fasteners	All	99	Fabroy	All Fasteners
Standard Dowel Pin	Tilt	1	McMasters	3/16 DIA, 3/4" Type 416 SST
Set Screws	Twist	8	Danforth	10-32 x 3/16", ½ dogend, Heat-treated alloy steel

Safety Calculations.

Heaviest MSA = <u>270</u> lb (FABS)

Load Path:

Part	QTY	Weight (lb)	pg # of calc	NOTES
✓ MSA	1	270	N/A	MESN02-116
Mounting plates	2	3	2	SST 4130
LM Mount	2	3	7	AL 6061
✓ LM Block	4	0.5	7	THK
✓ Rail	2	2	10	THK
✓ Rail Block	1	0.5	12	AL
✓ Threaded rod	1	0.5	12	SST
✓ Rail Plate	1	14	13	AL 6061-T6
✓ tilt Plate		2.8	18	AL 6061-T6
✓ pitch Plate	1	2.7	21	AL 6061-T6
✓ Pitch Crank	1	1.0	26	SST
✓ Shaft	1	7.9	29	SST 304
✓ Force ring/Fork	1	.2	41	AL
✓ Adapter block		10	49	AL
Robotend	1	N/A	53	max moment allowed = 6000 in.lb

NOTE: the critical path is through the bolts. To be conservative the Sut value for A2-50 bolts will be used, even though most will be A2-70.

Bending stress in the side mounting plates

$\leftarrow 6.34^{in} \rightarrow$

$13 in = h$

$W = 270 lb$

$\frac{W}{2}$
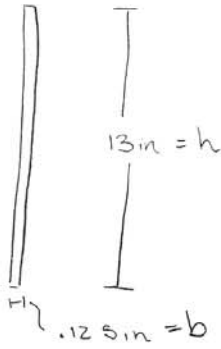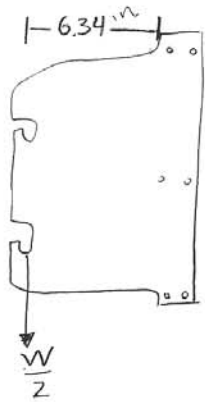
$.125 in = b$

$$I = \frac{bh^3}{12} = \frac{.125(13)^3}{12} = 22.885 \, in^4$$

$$C = \frac{d}{2} = 6.5 in$$

$$M_b = \frac{W}{2}(6.34) = 855.9 \, in\cdot lb$$

$$\sigma_b = \frac{Mc}{I} = 243.1 \, lb/in^2 \quad (psi)$$

S_y of 304 SST $= 30 \, ksi$

$$FS = \frac{S_y}{\sigma_b} = \frac{30,000 \, psi}{243.1} = 123.4 \quad > 3 \quad \therefore \quad OK$$

Seismic:

Apply 100-40-40 rule. Due to the fact that the NOID arm is free swinging in Z, assume Z accelerations are negligable.

$\frac{SD_X}{2}$

Y_{mom}

$SDX_{tensile}$

$\frac{SD_Y}{2}$

$\longrightarrow$
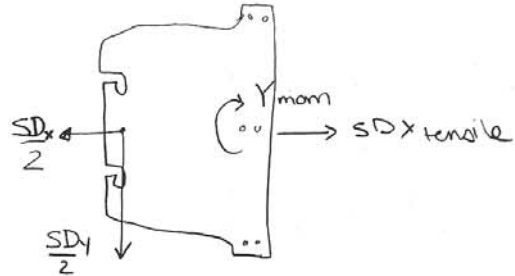
$SD_x = W \cdot C_P \cdot I_P \cdot A_{cc} \cdot N_P \ (40\%)$

$SD_y = W \cdot C_P \cdot I_P \cdot A_{cc} \cdot N_P \ (100\%)$

$A_{cc} = 1.3g$ (seismic acceleration)

$C_P = 1$ (force factor)

$I_P = 1$ (impatance factor)

$N_P = 1.2$ (correction factor PC1/PC2)

$SD_x = 270 \, lb \times 1 \times 1 \times 1.3 \times 1.2 (.4) = 168.5 \, lb$

$SD_y = 270 \, lb \times 1 \times 1 \times 1.3 \times 1.2 (1) = 421.2 \, lb$

$A = b \, h = 1.625 \, in^2$ (plate area)

Stress from SD_x:

$SD_x \text{ tensile} = \dfrac{SD_x / 2}{A} = 51.8 \, psi$

Stress from SD_y:

$Y_{mom} = \dfrac{SD_y}{2} \cdot (6.34") = 1.33 \times 10^3 \, in \, lb$

$\sigma_{Y_b} = \dfrac{Y_{mom} \cdot c}{I} = \dfrac{1.33 \times 10^3 \, (6.5)}{22.885} = 379.2 \, psi$

Combine static & seismic stresses:

$\sigma_{total} = \sigma_{Y_b} + SD_x \text{ tensile} + \sigma_b = 674.2 \, psi$

$FS = \dfrac{S_y}{\sigma_{total}} = \dfrac{30,000 \, psi}{674 \, psi} = \boxed{44.5}$

> 1.0 $\therefore$ ok

Look at the tangs bending + shearing due to SDx
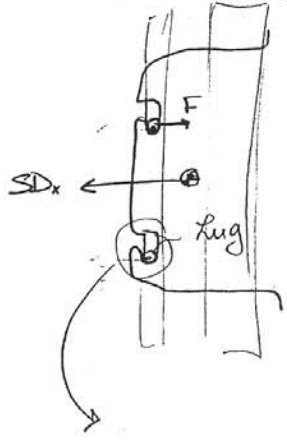
$$F = \frac{SD_x}{4} \leftarrow \text{four lugs total}$$

$SD_x = 168.5$ lb (from pg 3)

$$F = \frac{168.5}{4} = 42.13 \text{ lb}$$

Worst-case, F would be at the top of the tang.

THK = 3.175 mm

5.0mm (make a conservative assumption that the width is 5.0mm all the way to the base)
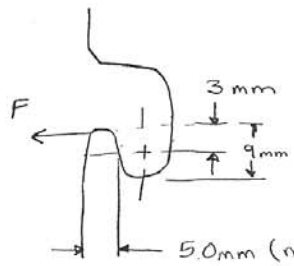
$$C = \frac{5mm}{2} = 2.5mm = 0.098''$$

$$A = 5mm \times 3.175mm = 15.875mm^2$$
$$= 0.025 in^2$$

$$I = \frac{bh^3}{12} = \frac{3.175(5)^3}{12}$$
$$= 33.07 mm^4$$
$$= 7.95 E-5 in^4$$

$$M = F \times \frac{9mm}{25.4 \, mm/in} = 42.13 \times \frac{9}{25.4} = 14.93 \text{ in lb}$$

$$\sigma = \frac{Mc}{I} = \frac{14.93 \text{ in lb} \times 0.098 in}{7.95 E-5 in^4} = \underline{18.41 \text{ ksi}}$$

$$\tau = \frac{3}{2} \frac{F}{A} = \frac{3}{2} \frac{(42.13 lb)}{.025 in^2} = \underline{2.53 \text{ ksi}}$$
↑ shear of a rectangle

$$\sigma_{vms} = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{18.41^2 + (2.53^2)3} = \underline{18.92 \text{ ksi}}$$

$S_y = 30$ ksi

$$F.S = \frac{30.0}{18.92} = \boxed{1.59} \geq 1 \quad \therefore \text{ ok}$$

Side Plates (shear in screws)
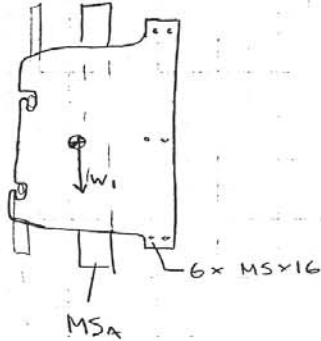
$W_1 =$ MSA weight $+ 2\times$ side plate weight

$W_1 = 270 + 6 = 276\,lb$

A moment is created which places a shear stress on the bolts.

6× M5×16

MSA
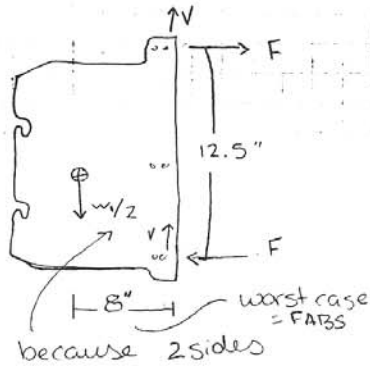
$F = \dfrac{W/2 \times 8''}{12.5''} = 88.3$

For both $F + V$ assume that only two screws carry the load due to misalignment.

$V = \dfrac{W/2}{2} = 67.5\,lb$

12.5"

8" — worst case = FABS because 2 sides

Total force on screw $= \sqrt{88.3^2 + 67.5^2} = 111.2\,lb$

$A_s = 14.2\,mm^2 = 0.022\,in^2$

$\sigma = \dfrac{111.2}{.022\,in^2} = \underline{5.05\,ksi}$

because the bolt is in shear : $S_{ut} = \dfrac{72.5\,ksi}{\sqrt{3}} = 41.9\,ksi$

Note: 72.5 ksi, S_{ut} came from Fabroy catalog for A2-50 SST bolts.

$FS = \dfrac{41.9\,ksi}{5.05\,ksi} = \boxed{8.3}$ on ultimate $\quad \underline{ok \geq 4}$

Bolt stress under seismic Load:
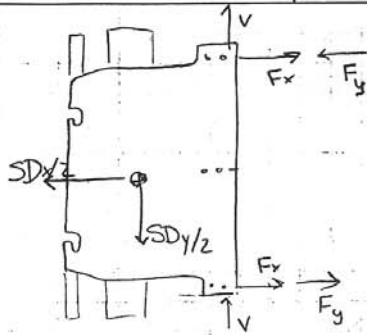
$Acc = 11.3g$ For 3 axis from NOID amplification

Using 100-40-40 Rule:

$C_p = 1$ $N_p = 1.2$ } same for all future calculations

$I_p = 1$

$SD_x = W \cdot C_p \cdot I_p \cdot Acc \cdot N_p \cdot (40\%) = 276(1)(1)(1.3)(1.2)(.4) = \underline{172.2}$

$SD_y = W \cdot C_p \cdot I_p \cdot Acc \cdot N_p \cdot (100\%) = 276(1)(1)(1.3)(1.2)(1) = \underline{430.56}$

Note: Do not neet to look at SD_z because the NOID arm is free in that direction. The friction nut of the tilt sub assembly prevents rotation of the rail subassembly.

Load from $SD_y \Rightarrow$

$F_y = \dfrac{SD_{y/2} \times 8}{12.5} = 137.8 \, lb$ } In addition to static load

$V = \dfrac{SD_y/2}{2} = 107.6 \, lb$

Load from $SD_x \Rightarrow$

assume load is taken by only 2 screws

$\dfrac{SD_x/2}{2} = 43.1 \, lb$

worste case screw:

$\sqrt{(137.8 + 88.3 + 43.1)^2 + (107.6 + 67.5)^2} = 321.1 \, lb$

$\sigma = \dfrac{321.1 \, lb}{0.022 \, in^2} = 14.60 \, ksi$

$F.S = \dfrac{41.9 \, ksi}{14.6 \, ksi} = \boxed{2.9} \quad > 1.25 \therefore ok$

Torque for side plate bolts:

$$T = K \times D_n \times F_p$$

$K = 0.3$ (high friction in cleanroom)

$D_n = 0.171$ in

$F_p = 65\% \ (72.5\,\text{ksi} \times (0.022\,\text{in}^2)) = 1037$ lb
(there is no axial force in this case)

$T = 0.3\,(.171)(1037) = \boxed{53.2 \text{ lb in}}$

AL stripping:

$A_s = \frac{\pi}{2}\left(\frac{\overset{D_n}{5}}{25.4}\right)\left(\frac{\overset{L_e}{12}}{25.4}\right)$

$= 0.146\,\text{in}^2$

need $FS \geq 3$

$S_y = 40\,\text{ksi}$

$\tau = \frac{F_p}{A_s}$, $FS \geq \frac{S_y}{\sqrt{3}\,\tau}$

$3 \triangleq \frac{40}{\sqrt{3}\,(F_p/.146)} \rightsquigarrow$

$F_p \leq 1.12$ kips

$\therefore F_p = 1037 = ok$

LM Mount + LM Block

— LM Block 4x (THK product: SHS 15V)

— LM Mount 2x (Aluminum)
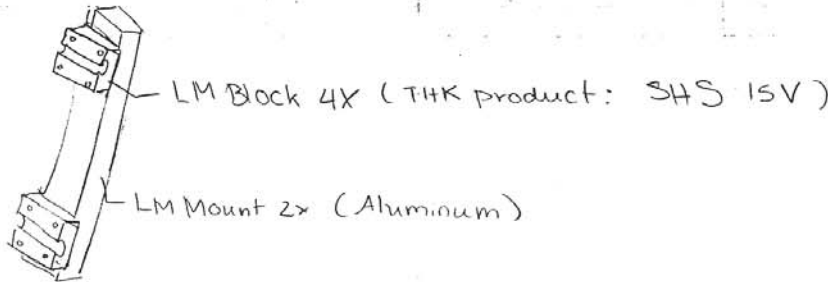
Each block has 4 M4 screws threaded into it 6mm deep.
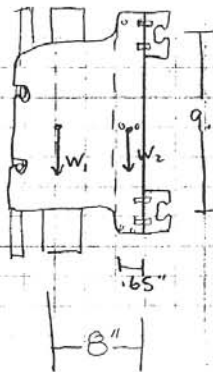
$W_1 = \underline{276}$ lb (from pg 5)

$W_2 = 2 \times$ LM Mount $= 6$ lb
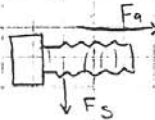
Moment created: $M = 276(8.0) + 6(.65) = 2211.9$ in lb

Force on screws $= \frac{M}{9.9''} = \frac{2211.9}{9.9} = \underline{223.4\text{ lb}}$

The tension induced from the moment is shared by the top 8 screws (4 on 2 LM blocks). One screw would hold

$$\frac{223.4}{8} = \underline{27.9 \text{ lbs}} \quad \text{axial}$$

Assuming only 4 out of the 16 screws hold the share (load) due to misalignment:

$$F_s = \frac{W_1 + W_2}{4} = \frac{276 + 6}{4} = \underline{70.5 \text{ lb}}$$

As for M4 = $8.78 \text{ mm}^2 = 0.0136 \text{ in}^2$

Bolt stress = $\frac{1}{A_s}\sqrt{F_a^2 + 3F_s^2} = \frac{1}{0.0136}\sqrt{27.9^2 + 3 \times 70.5^2}$

$$\sigma = 9.21 \text{ ksi}$$

$$FS = \frac{72.5 \text{ ksi}}{9.21 \text{ ksi}} = \boxed{7.9} \quad \begin{array}{l}\text{based on Ultimate}\\ \geq 4 \therefore \text{ok}\end{array}$$

Seismic Load:
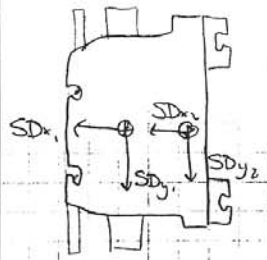
$$SD_{x1} = W_1 (C_p)(I_p)(Acc)(N_p)(40\%)$$

$$SD_{y1} = W_1 (C_p)(I_p)(Acc)(N_p)(100\%)$$

$$SD_{x2} = W_2 (C_p)(I_p)(Acc)(N_p)(40\%)$$

$$SD_{y2} = W_2 (C_p)(I_p)(Acc)(N_p)(100\%)$$

$SD_{x1} = 172.2$ $SD_{x2} = 6(1)(1)(1.3)(1.2)(.4) = 3.75 \text{ lb}$

$SD_{y1} = 430.56$ $SD_{y2} = 6(1)(1)(1.3)(1.2)(1.0) = 9.36 \text{ lb}$

$$\text{Pull from } SD_y = \frac{(SD_{y1} + W_1)(8.0") + (SD_{y2} + W_2)(.65")}{9.9" \times 8_s} = 71.5 \text{ lb}$$

eight screws sharing the load

Pull from $SD_x = \dfrac{SD_{x_1} + SD_{x_2}}{16} = \dfrac{172.2 + 3.75}{16} = \underline{11.0 \text{ lb}}$

16 ↗ 16 screws
Sharing the load

Total seismic $F_a = 71.5 + 11.0 = \underline{82.5 \text{ lb}}$

$F_{S_s} = \dfrac{SD_{y_1} + W_1 + SD_{y_2} + W_2}{4} = \underline{179.1 \text{ lb}}$

4 ↖ assume only 4 screws
share the shear load

$\sigma_s = \dfrac{1}{0.0136} \sqrt{(82.5)^2 + 3(179.1)^2} = 23.6 \text{ ksi}$

↑
seismic

$FS = \dfrac{72.5 \text{ ksi}}{23.6 \text{ ksi}} = \boxed{3.1} \qquad \geq 1.25 \quad \therefore \text{ ok}$

Bolt Torque:

$T = K_F \cdot D_n \cdot F_P$

$F_P \geq 1.6(F_a) = 1.6(27.9) = 44.6 \text{ lb}$

$F_P \leq 0.65(S_y)(A_t) = 72.5 E3 (0.65)(0.0136) = 640.9 \text{ lb}$

↳ 65%

Choose $F_P = 350 \text{ lb}$

$K = 0.3$ (cleanroom, high friction)

$T = 0.3 \left(\dfrac{4}{25.4}\right)(350) = \boxed{16.5 \text{ in lb}}$

NOTE: Thread pull out is included in the manufacturers
rated load for the LM Blocks.

LM Block + Rail Rating:

THK model # SHS-15R (see attached manufacturer data sheets)

- This model was selected because it is designed to have "4-way equal-load-rating"

Manufacturer Rating:

Static, C_0 = 24.2 kN = 5,440 lb

Dynamic, C = 9.8 kN = 2,200 lb

The 4 LM Blocks will resist the same moment + shear load that was used on the screws.
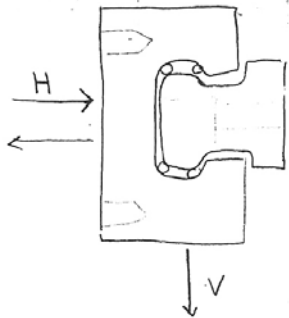
$$V = \frac{W_1 + W_2}{4} = 70.5 \text{ lb}$$

$$H = \frac{M}{9.9"} \times \frac{1}{2} = \frac{2211.9}{2(9.9)} = 111.7 \text{ lb}$$

⌐ 2 LM blocks resisting tension

Total load per LM block = $\sqrt{70.5 + 111.7^2}$ = 132 lb

This is far from the 2,200 lb Dynamic rated load

Rail Screws

Each of the two rails have a single row of M4 screws placed 2.36" apart.
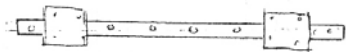
The same moment load applies.

(assumes the LM block's weight would contribute an insignificant amount to the moment at this point)

Assume that 2 screws take the tension load for each LM Block in tension. (4 total)

$$F_a = \frac{M}{4(9.9)^{''}} = \underline{55.9 \text{ lb}}$$

Assume only 4 screws take the shear load due to misalignment

$$F_s = \frac{W_1 + W_2 + Z}{4} \overset{\text{weight of LM blocks}}{} = 71.0$$

$$\sigma = \frac{1}{0.0136} \sqrt{55.9^2 + 3 \cdot 71.0^2} = \underline{9.87} \text{ ksi}$$

$$FS = \frac{72.5}{9.87} = \boxed{7.3} \geq 4 \therefore \text{okay}$$

use same torque as M4 screws in the LM Block. T = 16.5 in lb (see pg 9)

Seismic:

- use same loads as seismic case for LM Block screws:

$SD_{x_1} = 172.2$	$SD_{x_2} = 3.75$ lb	$W_1 = 276$ lb
$SD_{y_1} = 430.52$	$SD_{y_2} = 9.36$ lb	$W_2 = 6$ lb

$$\text{Pull from } SD_y = \frac{(SD_{y_1} + W_1)(8.0) + (SD_{y_2} + W_2)(0.65)^{''}}{9.9^{''} \times 4 \curvearrowleft 4\text{screws}} = 143 \text{ lb}$$

$$\text{Pull from } SD_x = \frac{SD_{x_1} + SD_{x_2}}{8 \curvearrowleft 8\text{screws}} = 22.0 \text{ lb}$$

$$Fa_s = 143 + 22.0 = 165 \text{ lb}$$

$$F_s = \frac{W_1 + SD_{y_1} + W_2 + SD_{y_2}}{4 \curvearrowleft 4\text{screws}} = 179.1 \text{ lb}$$

$$\sigma_s = \frac{1}{0.0136} \sqrt{165^2 + 3(179.1)^2} = 25.8 \text{ ksi}$$

$$FS = \frac{72.5}{25.8} = \boxed{2.8} \geq 1.25 \therefore \text{ok}$$

Look at the thread in the aluminum stripping:

Seismic: $F_a = 165\,lb$

Hole M4 ↧ 13.5 mm

an M4×20 bolt give an engagement length of 12 mm

$$A_t = \frac{\pi}{2} \times D_n \times L_e = \frac{\pi}{2}(.157") \left(\frac{12\,mm}{25.4\,in}\right) = 0.117 \; in^2$$

$$\tau = \frac{F_a}{A_t} = \frac{165\,lb}{.117\,in^2} = 1.41 \; ksi$$

$$FS = \frac{45}{\sqrt{3}(1.41)} = \boxed{18.4} \qquad \ge 1.25 \; \therefore \; ok$$

↑ cause in shear

Do not need to look at static case!

Rail Block + Threaded Rod

The rail block is designed so that there will be minimal force exerted on it. The weight + moment from the cantilevered MSA will be taken up mostly by the rail + LM block assemblies because the rail block has slightly compressed belville washers to take up any misalignment

There are no calculations needed for these pieces.

NOTE: There is a shield around the rail block + threaded rod to catch any particulate that may be created.

belville washers
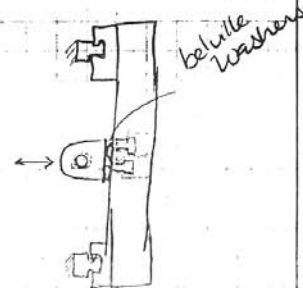
Rail Plate Hinges
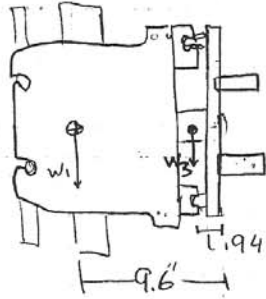
$W_1 = 276$

$W_3 = 2 \times \text{Rail} + \text{LM Block} \times 4 + \text{Threaded Rod} + \text{Rail Plate} + W_2$

$= \underline{26.5 \, lb}$

4x M10 Nut

4x M18 Bolt (2/hinge)

MC

5.76"

hinge x2
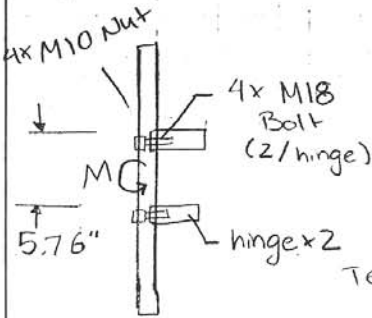
$M = 9.6(W_1) + 1(W_3)$

$= 2676.1 \, in \, lb$

Tension load on top two bolts:

$F_a = \dfrac{2676.1}{2(5.76")} = \underline{232.3 \, lb}$

↳ two bolts

$\text{shear from weight} = \dfrac{W_1 + W_3}{2} = \underline{151.3 \, lb}$

↳ assume only two bolts carry the load due to miss alignment
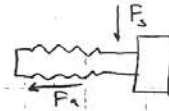
There is a possibility that the mirror subassembly is not centered on the rails. In the LRU assembly procedure, it does say to make sure it is centered, but there is that possibility it is not, which will create additional shear stress due to twist.
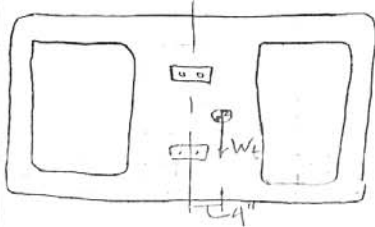
only W_1 and the 7 lbs from the 2 LM mounts and the 4 LM Blocks will add to the twist moment

$W_t = 276 + 7 = 283 \, lb$

$SD_{4L} = 283 (C_p)(I_p)(A_{cc})(N_p) = 441.5 \, lb$

→

Shear from twist (assume two bolts due to misalignment)
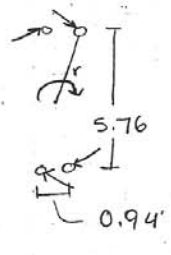
$$M_t = 283 \times 4'' = 1132 \ in \cdot lb.$$

$$r = \sqrt{\left(\frac{5.76}{2}\right)^2 + \left(\frac{.94}{2}\right)^2} = 2.92 \ in$$

$$\tan^{-1}\left(\frac{5.76''}{.94'}\right) = 80.7°$$

Shear twist Force $= \dfrac{M_t}{2r} = \dfrac{1132}{2(2.92)} = 193.84 \ lb$

F_s = worst case vector build up
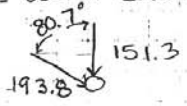
$$F_s = \sqrt{(151.3 + 193.8 \cos 80.7)^2 + (193.8 \sin 80.7)^2}$$

$$= 264.47 \ lb$$

There is also a small moment on the bolt where the hinge meets the yaw plate

$W_1 + W_2 = 302.5$
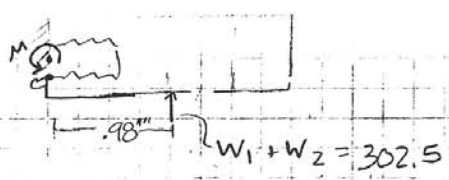
$$M = 0.98 (302.5) = 296.5 \ in \cdot lb$$

$$\sigma_{Bend} = \frac{Mc}{I}$$

The bottom is free so the cross section for $c + I$ will be taken to be from the bottom of the screws to the top of the hinge

$h = .45''$

$$C = \frac{\frac{.625}{2} + .134''}{2} = .22''$$

$I = \frac{1}{12} bh^3$ (because beam is actually the hinge + we are just finding force at c.)
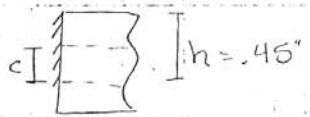
$$I = \frac{1}{12}(1.57'')(0.45)^3$$

$$= .0119 \ in^4$$

$\rightarrow$

$$\sigma = \frac{Mc}{I} + \frac{Fa}{As} = \frac{296.5 \, lb \, (.22)}{0.0119 \, in^4} + \frac{232.3}{0.0567}$$

$$\sigma = 9.58 \, ksi \qquad\qquad As = 0.0567 \, in^2$$
$$\text{for M8 bolt}$$

$$\tau = \frac{Fs}{As} = \frac{264.47}{.0567} = 4.66 \, ksi$$

$$\sigma_{vms} = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{9.58^2 + 3(4.66)^2} = \underline{12.53 \, ksi}$$

$$FS = \frac{72.5}{12.53} = \boxed{5.8} \qquad \geq 4 \quad \therefore ok$$

Note: this calculation was conservative because it did not take into account that the hinge is sealed in a slot of the yaw plate. It is designed to have a small interference fit which will take much of the shear load off the bolts.
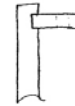
Thread Stripping:
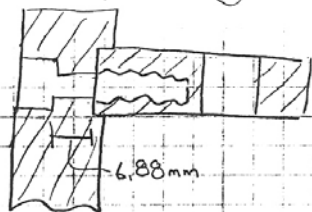
M8 × 20

$$\frac{20 - 6.88 \, mm}{25.4} = .5'' \text{ length of engagement}$$

6.88mm

$$A_t = \frac{\pi}{2} D_n \, Le = \frac{\pi}{2} \left(\frac{8}{25.4}\right)(.5) = .247$$

$$\tau = \frac{Fa}{A_t} = \frac{232.3}{.247} = 0.94 \, ksi$$

$$FS = \frac{45 \, {}^{AL}}{\sqrt{3}(0.94)} = \boxed{27.7} \qquad \geq 4 \quad \therefore ok$$
$$\uparrow \text{ cause in shear}$$

Bolt Torque.

$T = K \, D_n \, F_p$

$F_p \geq 1.6 \, F_a = 1.6 \, (282.3) = 371.5 \; lb$

$F_p \leq 0.65 \, (A_s)(S_{ut}) = 0.65 \, (.0567)(72.5 \, E3) = 2672 \; lb$

choose $F_p = 1600 \; lb$

$T = 0.3 \, (8/25.4)(1600) = \boxed{151 \quad in \; lb}$

Seismic Rail Plate Bolts:

$SD_x = W(C_p)(I_p)(A_{cc})(N_p)(40\%)$

$SD_y = W(C_p)(I_p)(A_{cc})(N_p)(100\%)$

$A_{cc} = 1.3g$ all axis

$C_p = I_p = 1$

$N_p = 1.2$

$SD_{x_1} = 172.2 \; lb \qquad SD_{x_3} = 16.5 \; lb$

$SD_{y_1} = 430.6 \; lb \qquad SD_{y_3} = 41.3 \; lb \qquad SD_{y_t} \,(Pg \,12) = 441.5 \; lb$

Pull from $SD_y = \dfrac{(SD_{y_1} + W_1)(9.6'') + (SD_{y_2} + W_3)(1'')}{2(5.76'')}$

$= 594.73 \; lb$

Pull from $SD_x = \dfrac{SD_{x_1} + SD_{x_3}}{4} = 47.2$

$F_a = 594.7 + 47.2 = 641.9 \; lb$

Shear from weight $= \dfrac{SD_{y_1} + W_1 + SD_{y_3} + W_3}{2} = 387.2 \; lb$

Shear from twist $= \dfrac{(W_t + SD_{y_t})4''}{(2.92)2} = \dfrac{(283 + 441.5)4''}{5.84} = 496.2 \; lb$

$F_s = \sqrt{(387.2 + 496.2 \cos 80.7)^2 + (496.2 \sin 80.7)^2} = 676.9 \; lb$

Bending moment :

(next page)
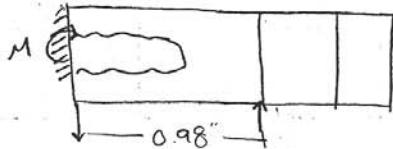
$$M = 0.98'(302.6 + SD_{y_1} + SD_{y_2}) = 900.0 \text{ in lb}$$

Same C + I from before

$$\sigma = \frac{Mc}{I} + \frac{F_a}{A_s} = \frac{900.0(.22)}{.0119} + \frac{641.9}{.0567}$$

$$= 27.96 \text{ ksi}$$

$$\tau = \frac{F_s}{A_s} = \frac{676.9}{.0567} = 11.94 \text{ ksi}$$

$$\sigma_{vms} = \sqrt{27.96^2 + 3(11.94)^2} = 34.78 \text{ ksi}$$

$$FS = \frac{72.5}{34.78} = \boxed{2.1} \quad \geq 1.25 \quad \text{ok}$$

NOTE: There is a conteracting moment on the pin that was not taken into account when calculating the bending moment on the hinge. If this was factored in the moment on the screws would have decreased making the safety factor higher.
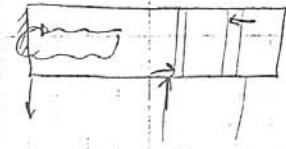
Tilt Plate Pins
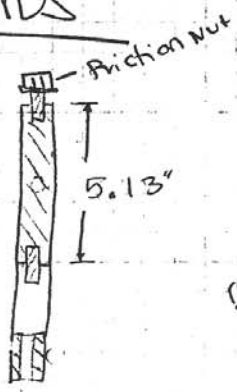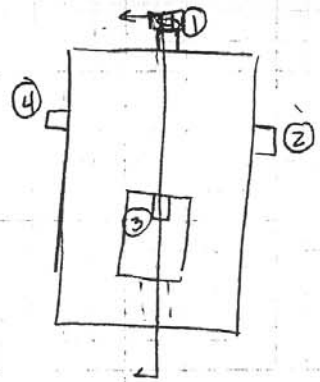
Pins ① + ③ are used with the yaw plate
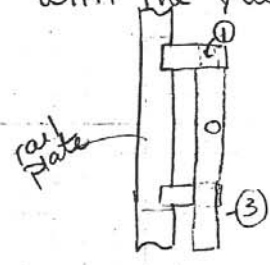
Pins ② + ④ are used with the pitch plate

5.13"

Friction Nut

rail plate

5.4"
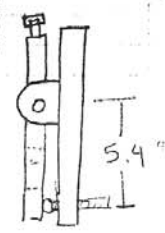
° all pins are A2-70 with ½" DIA $S_{ut} = 101.5 ksi$

and are pressed into the tilt plate

Pins ① + ③ are worst case because Pin ① has a significant bending stress that the others don't.

.98"

F_M

5.13"

F_M

1.92"

10.58

W_1 W_3
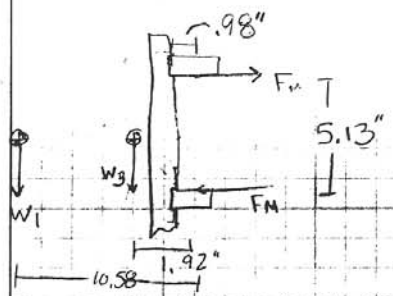
$W_1 = 276$ lb   } from pg 13
$W_3 = 26.5$ lb

$M = 276 (10.58") + 26.5 (1.92")$
$= 2970.9$ in lb

$F_{M'} = \dfrac{2970.9}{5.13"} = 579.1$ lb

There is also shear from the twist moment from the small chance that in mirror is off center.

$M_t = 1132$ in lb (from pg 13)

$F_t = \dfrac{M_t}{5.13} = 220.66$ lb

F_t M_t F_t

TOP of PIN:

F_t

F_M
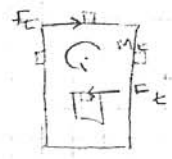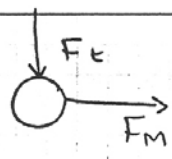
total Shear force, $F_s = \sqrt{F_m^2 + F_t^2}$

$= \sqrt{579.1^2 + 220.66^2}$

$= \underline{619.7 \text{ lb}}$

round section

$\tau = \dfrac{4}{3}\dfrac{F_s}{A}$

$A = \pi r^2 = \pi\left(\tfrac{1}{4}"\right)^2 = 0.196 \text{ in}^2$

$\tau = \dfrac{4/3(619.7)}{.196} = \underline{4.21 \text{ ksi}}$

There is no axial force, but there is a bending moment on the top pin.
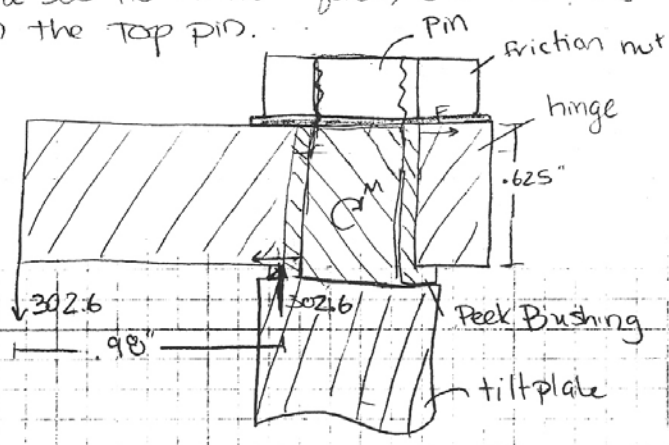
$M = 302.6(.98) = 296.5 \text{ in lb}$

$\sigma = \dfrac{MC}{I} \qquad C = 1/4"$

$I = \dfrac{\pi}{4}(.25)^4 = .00307 \text{ in}^4$

$\sigma = \dfrac{296.5(.25)}{.00307} = 24.15 \text{ ksi}$

$$\sigma_{vms} = \sqrt{\sigma^2 + 3\tau^2}$$

$$= \sqrt{24.15^2 + 3(4.21)^2} = 25.23 \text{ ksi}$$

S_{ut} for A2-70 pin = 101.5 ksi

$$FS = \frac{101.5}{25.23} = \boxed{4.0} \quad \geq 4 \quad \text{so OK}$$

Note: this was a conservative calculation because some of the bending moment in the top pin would be taken up by the friction on the bottom pin.

Seismic

There has been no added weight so the seismic values from the rail plate bolts can be used.

$SD_{x1} = 172.2 \text{ lb}$ $SD_{y1} = 430.6 \text{ lb}$ $SD_{yt} = 441.5 \text{ lb}$

$SD_{x3} = 16.5 \text{ lb}$ $SD_{y3} = 41.3 \text{ lb}$

shear from SD_x:

$$F_x = \frac{SD_{x1} + SD_{x3}}{2} = 94.35 \text{ lb}$$
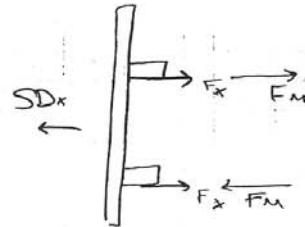
shear from Moment:

$$F_M = \frac{(SD_{y1} + W_1)(10.58") + (SD_{y3} + W_3)(1.92")}{5.13_n}$$

$$= 1482.6 \text{ lb}$$

shear from twist:

$$F_t = \frac{4(W_t + SD_{yt})}{5.13} = 568.7 \text{ lb}$$

Total Shear:
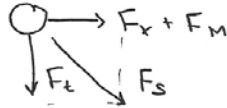
$$F_s = \sqrt{(F_M + F_s)^2 + (F_t)^2}$$

$$= \sqrt{(1482.6 + 94.35)^2 + (568.7)^2}$$

$$F_s = 1676.4 \text{ lb}$$

$$\tau = \frac{4}{3} \frac{F_s}{A} = \frac{(4/3)1676.4}{.196 \text{ in}^2} = \underline{11.4} \text{ ksi}$$

bending moment:

$$M = (W_1 + W_3 + SD_{y_1} + SD_{y_3})(.98") = 759.01 \text{ in lb}$$

$$\sigma = \frac{Mc}{I} = \frac{759.01(.25)}{.00307} = 61.81 \text{ ksi}$$

$$\sigma_{vms} = \sqrt{(61.81)^2 + 3(11.4)^2} = 64.89 \text{ ksi}$$

$$FS = \frac{101.5}{64.89} = \boxed{1.6} \qquad \geq 1.25 \quad \therefore OK$$

Note: This really includes two rare events. 1. an earthquake + 2. the mirror being off center by 4".

Pitch Plate:

* The Rail Plate can rotate ±20° relative to the rest of the NOID End-Effector + NOID arm. The worst case stresses occur when the Rail Plate is rotated the full 20° because it adds to the twist moment That places a shear stress on the rest of the assembly's fasteners. For the remainder of the calculations (except where noted), calculations will assume the 20° rotated position.

TOP VIEW

tilt Plate

.75"

Pitch Plate

10°

could be 4" off center ↓

Hinge × 2

Hinge Short × 2

W₂

W₁

2.1"

10.9"
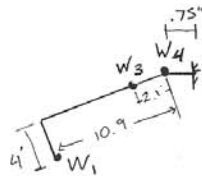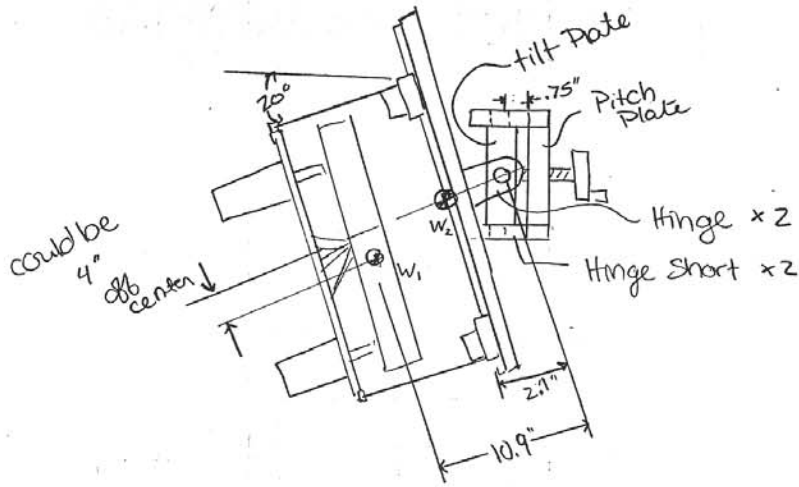
$W_1 = 276 \ (MSA)$

$W_3 = 26.5 \ lb$

$W_4 = 3.0 \ lb$

(tilt plate and four hinges)

.75"

W₃ W₄

1.2.1'

10.9"

4'

W₁

20°

.75

.72

1.97

3.73"

3.76"

9.62"
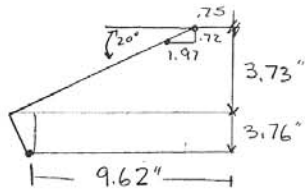
Looking at Pitch Plate bolts

$$M_t = (W_1)(3.73 + 3.76) + W_3(.72)$$

$$= 2086.3 \ in \cdot lb$$

$$M_b = W_1(9.62") + W_3(1.97 + .75) + W_4(.75)$$

$$= 2730.7 \ in \cdot lb$$

Pitch Plate

M_t

M_b

4 × M8

5.4"

threaded rod

tungsten carbide ball rests against curved tungsten carbide
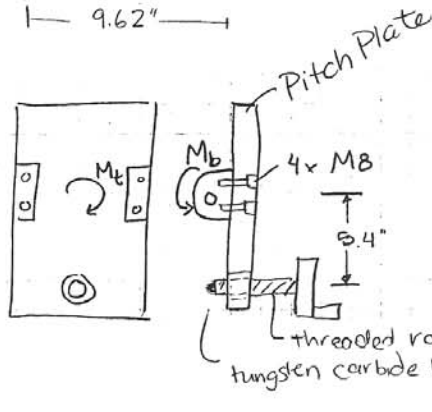
M_t is taken only by the 4 screws

M_b is shared between the screws + the threaded rod
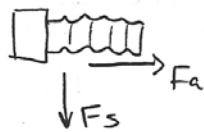
$$F_a = \frac{M_b}{4(5.4)} = \frac{2730.7}{4(5.4')} = 126.4 \text{ lb}$$

↳ four screws

$$F_{s_{weight}} = \frac{\sum W}{2} = \frac{276 + 26.5 + 3}{2} = \frac{305.5}{2} = 152.8 \text{ lb}$$

↳ assumes 2 screws share load due to misalignment
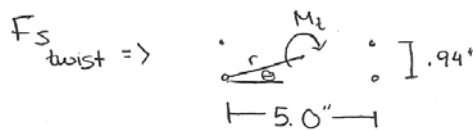
$F_{s_{twist}} \Rightarrow$

$$\theta = \tan^{-1}\left(\frac{.94}{5}\right) = 10.6°$$

$$r = \sqrt{\left(\frac{.94}{2}\right)^2 + \left(\frac{5.0}{2}\right)^2} = 2.54'$$

$$F_{s_{twist}} = \frac{M_t}{2(2.54)} = \frac{2086.3}{2(2.54)} = 410.08 \text{ lb}$$

↳ assumes 2 screws share load

Total Shear:
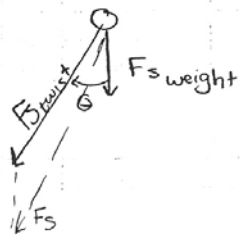
$$F_s = \sqrt{\left(F_{s_{weight}} + F_{s_{twist}} \cos\theta\right)^2 + \left(F_{s_{twist}} \sin\theta\right)^2}$$

$$= \sqrt{(152.8 + 403.1)^2 + (75.4)^2}$$

$$= 560.97 \text{ lb}$$

$$\sigma = \frac{1}{A_t}\sqrt{F_a^2 + 3F_s^2} = \frac{1}{0.0567}\sqrt{(126.4)^2 + (560.97)^2 \times 3}$$

$$= 17.28 \text{ ksi}$$

$$FS = \frac{72.5}{17.28} = \boxed{4.2} \geq 4 \therefore Ok$$

Note: Many conservative assumptions were made in this calculation:
1. Assembly at max (20°) angle
2. MSA off center on rails 4"
3. Only 2 out of the 4 screws take the shear load for weight + twist
4. A2-50 Sut was use instead of the A2-70 Sut value..

- Thread striping: similar to other hinges pg 15 which had a very high FS.
- Bolt Torque: use same torque as M8 bolts in the rail plate hinges. (pg 16)

Seismic

$$SD_x = W(c_P)(I_P)(A_{cc})(N_P)(40\%)$$
$$SD_y = W(c_P)(I_P)(A_{cc})(N_P)(100\%)$$

$N_P = 1.2$
$A_{cc} = 1.3$ all axis
$I_P = C_P = 1$

$SD_{1x} = 172.2$ $SD_{1y} = 430.6$

$SD_{3x} = 16.5$ $SD_{3y} = 41.3$ } lb

$SD_{4x} = 1.9$ $SD_{4y} = 4.7$

$$M_{t_s} = (SD_{1y} + W_1)(3.73 \, r \, 3.76) + (SD_{y3} + W_3)(.72)$$
$$= 5341.3 \text{ in lb}$$

$$M_{b_s} = (SD_{1y} + W_1)(9.62") + (SD_{y3} + W_3)(1.97+.75) + (SD_{y4} + W_4)(.75)$$
$$= 6987.7 \text{ inlb}$$

Because the rail assembly is free to rotate about "Y" it is assumed that SD_x puts an even axial pull across the four bolts.

$$F_a = F_{a_b} + F_{a_{SD_x}}$$

$$F_{a_{SD_x}} = \frac{\Sigma SD_x}{4} = \frac{172.2 + 16.5 + 1.9}{4} = 47.65 \text{ lb}$$

$$F_{a_b} = \frac{M_b}{5.4''(4)} = \frac{6987.7}{4(5.4)} = 323.5 \text{ lb}$$

$$F_a = 47.65 + 323.5 = 371.15 \text{ lb}$$

$F_s \equiv$ resultant of $F_{s_{weight}} + F_{s_{twist}}$

$$F_{s_{weight}} = \left(\frac{\Sigma W + \Sigma SD_y}{2}\right) = 391.05 \text{ lb}$$

$$F_{s_{twist}} = \frac{M_t}{2r} = \frac{5341.3}{2(2.54)} = 1051.44 \text{ lb}$$

$$F_s = \sqrt{\left(F_{s_{weight}} + F_{s_{twist}} \cos\theta\right)^2 + \left(F_{s_{twist}} \sin\theta\right)^2}, \quad \theta = 10.6°$$

$$F_s = \sqrt{(391.05 + 1033.5)^2 + (193.4)^2}$$

$$F_s = 1437.6 \text{ lb}$$

$$\sigma = \frac{1}{A_t}\sqrt{F_a^2 + 3F_s^2} = \frac{1}{0.0567}\sqrt{(371.15)^2 + 3(1437.6)^2}$$

$$\sigma = 44.40 \text{ ksi}$$

$$FS = \frac{72.5}{44.40} = \boxed{1.63} \geq 1.25 \quad \therefore ok$$

Note: same stack up & conservative assumptions apply here (see pg 21)

Threaded Rod Through Pitch Plate
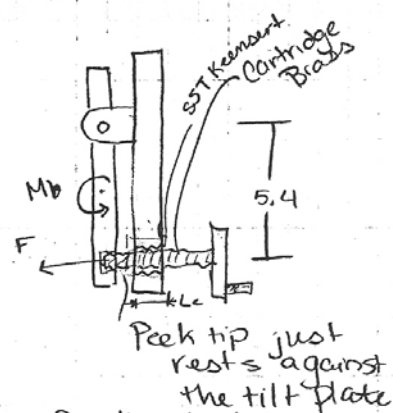
Worst case for the threaded rod is when there is no rotation in yaw

Peek tip just rests against the tilt plate
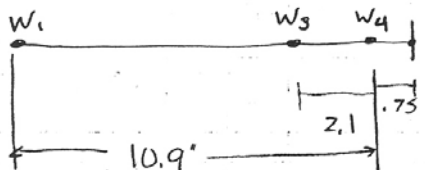
Looking for thread shear:

$$M_b = W_1 (10.9 + .75) + W_3 (2.1 + .75) + W_4 (.75)$$

$$= 3293.2 \text{ in lb}$$

$$F = \frac{M_b}{5.4} = \frac{3293.2}{5.4} = \underline{609.85} \text{ lb}$$

Brass is the weaker material with $S_{ut} = 55,000$ psi

(range of $55 - 75$ ksi given by manufacturer)

$$L_e = \tfrac{1}{2}" \qquad D_n = \tfrac{3}{8}" \qquad TPI - 24 \text{ (Fine)}$$

$$A_s = \frac{\pi}{2}(D_m)(L_e) \quad , \quad D_m = D_n - \frac{1}{TPI}$$

$$A_s = \frac{\pi}{2}\left(\tfrac{3}{8} - \tfrac{1}{24}\right)\left(\tfrac{1}{2}\right) = 0.26 \text{ in}^2$$

$$\tau = \frac{F}{A_s} = \frac{609.85}{.26} = 2.35 \text{ ksi}$$

$$FS = \frac{S_{ut}}{\sqrt{3}\,\tau} = \frac{55.0 \text{ ksi}}{\sqrt{3}(2.35)\text{ksi}} = \boxed{13.5} \qquad \geq 4 \; \therefore \text{ ok}$$

↑ because it's in shear

Seismic:

all seismic values from Pg 24 still apply!

$$M_{b_{seismic}} = (SD_{y_1} + W_1)(10.9 + .75) + (SD_{y_3} + W_3)(2.1 + .75) + (SD_{y_4} + W_4)(.75)$$

$$= 8430.9 \text{ in lb}$$

$$F_{SD_y} = \frac{8430.9}{5.4°} = 1561.3 \text{ lb}$$

$$F_{SD_x} = \frac{SD_{x_1} + SD_{x_3} + SD_{x_4}}{2} = 95.3 \text{ lb}$$

2 assume other half is taken by pins.

Total $F = 1561.3 + 95.3 = 1656.6 \text{ lb}$

$$\tau = \frac{F}{A_s} = \frac{1656.6}{.26} = 6.37 \text{ ksi}$$

$$FS = \frac{S_{ut_{brass}}}{\sqrt{3}\ \tau} = \frac{55.0}{\sqrt{3}(6.37)} = \boxed{5.0} \qquad \geq 1.25 \therefore ok$$

→ Contact Stress: (from Roark's Formulas for Stress + Strain)

Ø 7/32
Ø 1/4
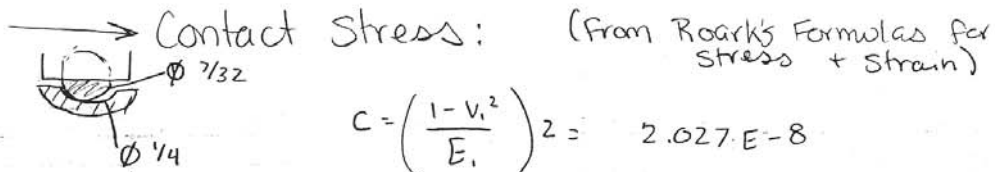
$$C = \left(\frac{1 - V_1^2}{E_1}\right) 2 = 2.027 E-8$$

$$k = \frac{D_1 \cdot D_2}{D_1 - D_2} = \frac{1/4 \cdot 7/32}{1/4 - 7/32} = 1.75$$

Tungsten Carbide

$V:\quad .26$

$E\quad 9.2 E 7 \text{ psi}$

$d_1 = 1/4 \qquad d_2 = 7/32$

$S_{compression} = 790 \text{ ksi}$

$$a = .721 \sqrt[3]{P \cdot K \cdot C} = .721 \sqrt[3]{(609.85)(1.75)(2.027 E-8)}$$

$$= 0.02 \text{ in}^2$$

$$\sigma = \frac{1.5 P}{\pi a^2} = 7.215 E5 \text{ psi}$$

$$SF = \frac{7.90 E 5 \text{ psi}}{7.215 \text{ psi}} = \boxed{1.1}$$

- This is not a critical part and yielding will only decrease the stress!

Force to turn the crank:

tungsten carbide ball
SST Keensert
3/8"-24 Lubricated Brass
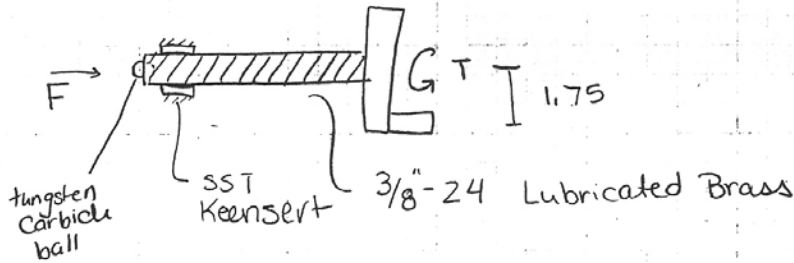
$T = K(D_n)(F)$

Where $F = 609.85$ lb
    $D_n = 3/8"$
    $K = .10$ (Estimated value from the K values in the ME Design Standards. It's low because lubricated brass + SST have a low coefficient of friction)

$T = .10 (3/8")(609.85 \text{ lb}) = 22.87$ in lb

$F_{handle} = \dfrac{T}{1.75"} = \boxed{13.1 \text{ lb}}$

This is doable considering this is the worst case scenario. Most mirror assemblies weigh 2/3 the weight used in the calculation and the friction is probably less that what was estimated. If there is a problem when it is assembled a larger handle can be use to decrease the force required to turn the handle.

Seismic does not have to be calculated because no one will have to turn the handle at such a time.

Shaft

6 x M8 (Ag PLD SST)

3.04

Ø 3.5"

shaft
roller bearings
threaded collar
adapter block

20°

1.4"
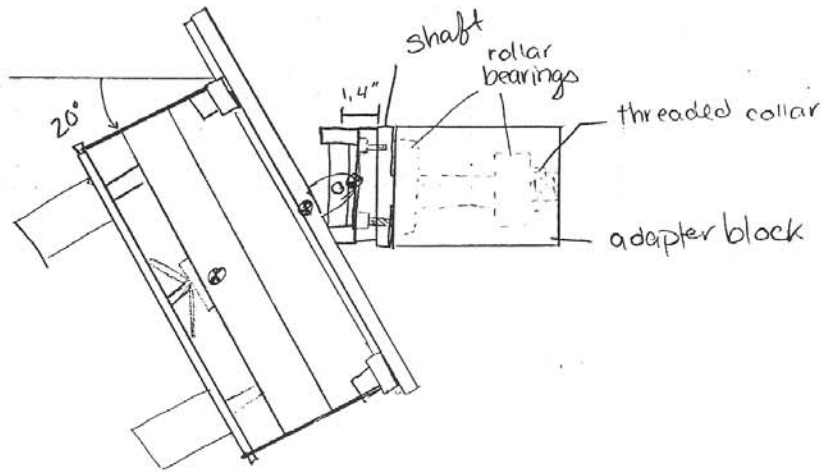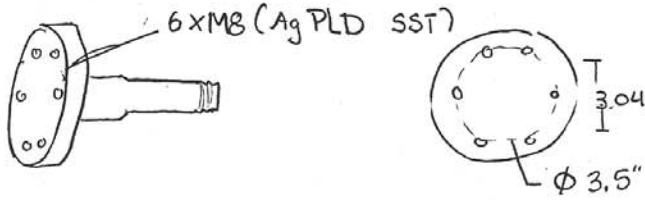
$W_1 = 276$ (MSA)

$W_3 = 26.5$ lb

$W_5 = 6.7$ lb
(w₄ + pitchplate, + crank)

1.4'
W_5
10.9'
W_3
12.1'
.8"
$\frac{4}{1}$ • W_1
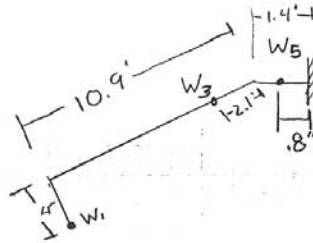
$M_t = 2086.3$ (from pg 22)

$M_b = W_1 (10.27") + W_3 (1.97 + 1.4) + W_5 (.8')$

$= 2929.2$ in lb

20°
1.4
.72
1.97
7.49'
10.27'
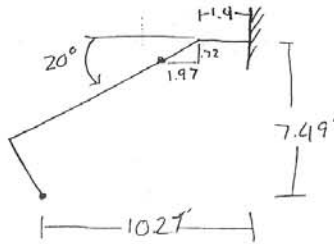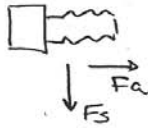
$$F_a = \frac{M_b}{2(3.04)} = \frac{2929.2}{2(3.04)} = 356.1 \ lb$$

$$F_{s\ weight} = \frac{\Sigma W}{3} = \frac{276 + 26.5 + 6.7}{3} = 103.07\ lb$$

assumes 3 of the 6 screws share the load
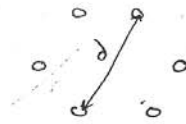
F_s twist ⇒

$$r = \frac{d}{2} = \frac{3.5}{2} = 1.75"$$

$$F_{s\ twist} = \frac{M_t}{3(1.75)} = \frac{2086.3}{3(1.75)} = 397.4 \ lb$$

worst case:
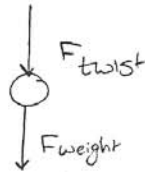
$$F_s = F_{weight} + F_{twist}$$
$$= 103.07 + 397.4$$
$$= 500.46\ lb$$

$A_t = 0.0567$ for M8

$$\sigma = \frac{1}{A_t}\sqrt{F_a^2 + 3F_s^2} = \frac{1}{0.0567}\sqrt{(356.1)^2 + 3(500.46)^2}$$

$$= 16.53\ ksi$$

$$FS = \frac{72.5}{16.53} = \boxed{4.4} \geq 4 \therefore ok$$

Pull out:

Fa worst case is when the rail plate has 0° rotation
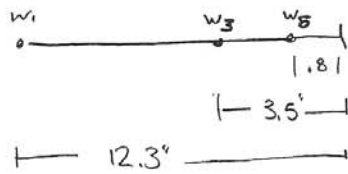
$$M_b = W_1 (12.3) + W_3 (3.5) + W_5 (.8) = 4722.9 \text{ in lb}$$

$$F_a = \frac{M_b}{2(304")} = 776.8 \text{ lb/screw}$$

screws: M8 × 25 into 304 SST ($S_y = 35 \text{ksi}$)

$L_e = .5"$
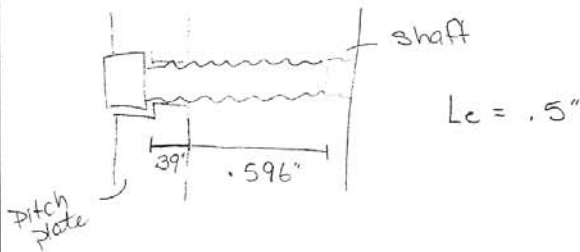
$$A_s = \frac{\pi}{2} (D_n - TPI^{-1}) L_e$$

$$= \frac{\pi}{2} (.315 - .05)(.5) = .208 \text{ in}^2$$

$$\tau = \frac{F_a}{A_s} = \frac{776.8}{.208} = 3.73 \text{ ksi}$$

$$FS = \frac{35 \text{ ksi}}{\sqrt{3} (3.73)} = \boxed{5.4} \geq 3 \therefore \text{ ok}$$

τ cause in shear

Torque:

$$F_p \geq 1.6 (F_a) = 1.6 (776.8) = 1242.9 \text{ lb}$$

$$F_p \leq .65 (A_t)(S_{ut}) = .65 (.0567 \text{ in}^2)(72.5 E3 \frac{\text{lb}}{\text{in}^2}) = 2672.0$$

choose $F_p = 2000$ lb

$T = K \cdot D_n \cdot F_p$

$= .25 \times .315 \times 2000 = 157.5 \rightsquigarrow \boxed{160 \text{ in lb}}$

↑ cleanroom + AgPLD

Seismic

$SD_y = W (C_p)(IP)(Acc)(N_p)$

$SD_x = 40\% (SD_y)$

$SD_{1x} = 172.2$	$SD_{1y} = 430.6$
$SD_{3x} = 16.5$	$SD_{3y} = 41.3$
$SD_{5x} = 4.2$	$SD_{5y} = 10.6$

} lb

$M_{ts} = 5341.3$ (same as previous calc on pg 24)

$M_{b_s} = (SD_{1y} + W_1)(10.27) + (SD_{3y} + W_3)(1.97 + 1.4) + (SD_{5y} + W_5)(.8')$

$= 7499.0$ in lb

$F_a = F_{a_b} + F_{a_{SD_r}}$

$F_{a_{SD_r}} = \dfrac{\Sigma SD_r}{6} = 32.15$ lb   6 screws

$F_{a_b} = \dfrac{M_b}{2(3.04)} = \dfrac{7499}{2(3.04)} = 1233.4$ lb

$F_a = 32.15 + 1233.4 = 1265.5$ lb

$F_s = F_{s_{twist}} + F_{s_{weight}}$

$F_{s_{weight}} = \dfrac{\Sigma W + \Sigma SD_y}{3} = 263.9$ lb

↖ assumes 3 out of the 6 screws share the load

$$FS_{twist} = \frac{M_{ts}}{3\ r} = \frac{5341.3}{3\ (1.75)} = 1017.4$$

$$F_S = 263.9 + 1617.4 = 1281.3$$

$$\sigma = \frac{1}{0.056} \sqrt{1265.5^2 + 3(1281.3)^2} = 45.6\ ksi$$

$$FS = \frac{72.5}{45.6} = \boxed{1.6} \geq 1.25 \ \therefore \ ok$$

Stresses in the Shaft and Bearings
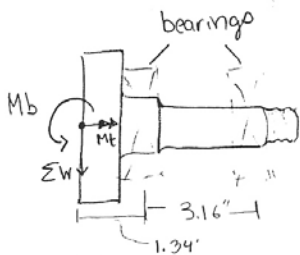
$$M_b = 2929.2$$

$$M_t = 2086.3$$

$$\Sigma W = 276 + 26.5 + 6.7 = 309.2\ lb$$

from pg 29
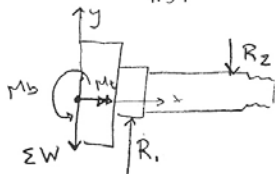
$$\Sigma F_y = 0 \qquad \Sigma W + R_2 = R_1 \qquad \textcircled{1}$$

$$\Sigma M_z = 0 \ \text{at origin}$$

$$M_b + 1.34(R_1) = R_2(1.34 + 3.16)$$

$$2929.2 + 1.34 R_1 = 4.5 R_2 \qquad \textcircled{2}$$

substitute eqn ① into eqn ② :

$$2929.2 + 1.34(309.2 + R_2) = R_2(4.5)$$

$$3343.5 = 3.16\ R_2$$

$$R_2 = 1058.1\ lb$$

$R_i = 1367.3$ lb

Bearing Rating (from Timken.com)
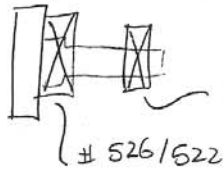
	#526/522	#23620/23690
Dynamic Radial load	8,900	4,800

Note: there is very little thrust on these bearings so there is no need to look at it. The rated thrust load for the small bearing is 3040 lb

$$FS = \frac{8900}{1367.3} = \boxed{6.51} + \frac{4800 \, lb}{1058 \, lb} = \boxed{4.54}$$
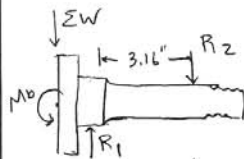
Shaft:

$M_b = 2929.2$ in lb

$\Sigma W = 309.2$ lb

$R_1 = 1367.3$ lb

$R_2 = 1058.1$ lb
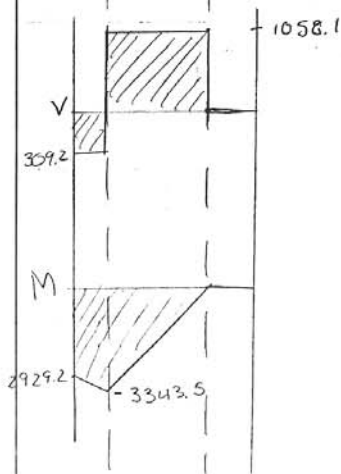
In addition to shear and the bending moment theres is also a twist, M_t the shaft is carrying.
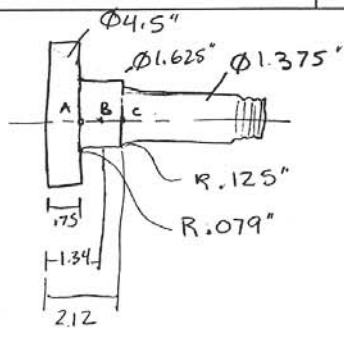
A, B, + C are the critical stress areas of the shaft. The stress will be calculated at all three.

At Point A:

$M_A = 2929.2 \text{ in lb} + 309.2(.75) = 3161.1 \text{ in lb}$

$V_A = 309.2 \text{ lb}$

$M_t = 2086.3 \text{ in lb}$

For DIA = 1.625" $\rightsquigarrow$

$I = \frac{\pi}{4} r^2 = 0.342 \text{ in}^4$ $A = \pi \frac{d^2}{4} = 2.07 \text{ in}^2$

$C = \frac{1}{2} d = 0.813 \text{ in}$

$J = \frac{1}{2} \pi c^4 = .684 \text{ in}^4$

$\sigma_o = \frac{M \cdot C}{I} = \frac{3161.1 (.813)}{.342} = 8.44 \text{ KSi}$

The decrease in Diameter causes a sress concentration.

$\frac{D}{d} = \frac{4.5}{1.625} = 2.77$ from "Machinery's Handbook
                                                    25ª Edition" pg 260

$\frac{r}{d} = \frac{.079}{1.625} = .049$ $K: 2.2$

$\sigma = 8.44 (2.2) = \underline{18.57} \text{ KSi}$

shear $\rightarrow$
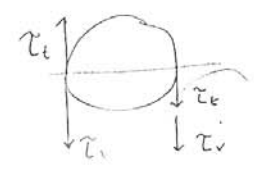
$$\tau_v = \frac{V_a}{A} = \frac{309.2}{2.07} = 149.4 \ psi$$

$$\tau_{t_0} = \frac{M_t C}{J} = \frac{2086.3 \ in \ lb \ (.813 \ in)}{.684 \ in^4} = 2.47 \ ksi$$

From pg 199 of the "Machinery's Handbook 25ᵗʰ edition":

$$K = 1.8$$

$$\tau_t = 2.47 \ (1.8) = 4.45 \ ksi$$

$$\tau = \tau_v + \tau_t = \underline{4.60} \ ksi$$

$$\sigma_{vms} = \sqrt{\sigma^2 + 3\tau^2} = \sqrt{18.57^2 + 3(4.6)^2} = 20.2 \ ksi$$

$$Sut \ SST304 = 85 \ ksi$$

$$SF = \frac{85}{20.2} = \boxed{4.2} \geq 4 \ \therefore OK$$

NOTE: if the assumption is made that there are no stress concentrations because SST304 is ductile + the SF is calculated based on $S_y = 35ksi$ $\boxed{SF = 3.7}$ * still good

At Point B:

$$M_B = 3343.5 \ inlb$$

$$V_B = 1367.3 \ lb$$

$$M_t = 2086.3$$

$$DIA = 1.625''$$

Same I, A, C, + J as Point A.

(there are no concentration factors at this point)

$$\sigma = \frac{M_B C}{I} = \frac{3343.5 \ (.813)}{.342} = 7.95 \ ksi$$

$$\tau = \frac{V_B}{A} + \frac{M_t C}{J} = \frac{1367.3}{2.07} + \frac{2086.3 \ (.813)}{.684} = .66 \ ksi$$

$$\sigma_{vms} = \sqrt{7.95^2 + 3(.66)^2} = 8.03 \ ksi$$

	ultimate	yield

$$SF = \frac{85 \text{ ksi}}{8.03 \text{ ksi}} = \boxed{10.6} \qquad \frac{35}{8.03} = \boxed{4.36} \geq 3 \therefore \text{ ok}$$

At Point C :

$M_c = 3343.5 - 1058.1 (2.12" - 1.34") = 2518.2 \text{ in lb}$

$V_c = 1058.1 \text{ lb}$

$M_t = 2086.3 \text{ in lb}$

$DIA = 1.375" \rightsquigarrow$

$A = \frac{\pi}{4} (1.375)^2 = 1.48 \text{ in}^2$

$I = \frac{\pi}{4} (.688)^4 = 0.175 \text{ in}^4$

$C = 0.688 \text{ in}$

$J = 0.35 \text{ in}^4$

$\sigma_o = \frac{M_c C}{I} = \frac{2518.2 (.688)}{.175} = 9.90 \text{ ksi}$

$\frac{D}{d} = \frac{1.625}{1.375} = 1.18 \qquad \frac{r}{d} = \frac{0.125}{1.375} = .091$

$K = 1.65$

$\sigma = 1.65 (9.9) = \underline{16.34 \text{ ksi}}$

$\tau_v = \frac{V_c}{A} = \frac{1058.1}{1.48} = 0.715 \text{ ksi}$

$\tau_{t_o} = \frac{M_t C}{J} = \frac{2086.3 (.688)}{0.35} = 4.10 \text{ ksi}$

$K = 1.35$

$\tau_t = 1.35 (4.10) = 5.54 \text{ ksi}$

$$\gamma = \tau_v + \tau_t = .715 + 5.54 = \underline{6.25} \text{ ksi}$$

$$\sigma_{vms} = \sqrt{16.34^2 + 3(6.25)^2} = 19.6 \text{ ksi}$$

$$SF = \frac{85}{19.6} = \boxed{4.3} \geq 4 \quad \therefore OK$$

NOTE: If the assumption is made that there are no stress concentrations because SST304 is ductile + the S.F. is calculated based on $S_y = 35$ ksi

$$\boxed{FS = 2.9}$$

Seismic   (pt. A)

$M_{b_s} = 7499$ in lb

$M_{t_s} = 5341.3$ in lb $\left.\begin{array}{c} \\ \\ \\ \end{array}\right\}$ from pg 32

Weight $= \Sigma W + \Sigma SD_y = 791.7$ lb

any SD_x will be taken by the bearings, not carried in the shaft.
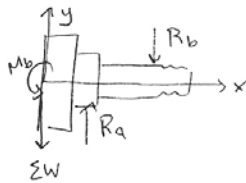
$\Sigma F_y = 0 \qquad 791.7 + R_b = R_a$ ①

ΣM_z at origin $= 0$

$$M_{b_s} + R_a(1.34) = R_b(1.34 + 3.16)$$

$$7499 + 1.34 R_a = 4.5 R_b \quad ②$$

Substitute eqn ① into ②

$$7499 + 1.34(791.7 + R_b) = 4.5 R_b$$

$$R_b = 2708.8 \text{ lb}$$

$$R_a = 3500.5 \text{ lb}$$

Shear + Moment Diagram on next page
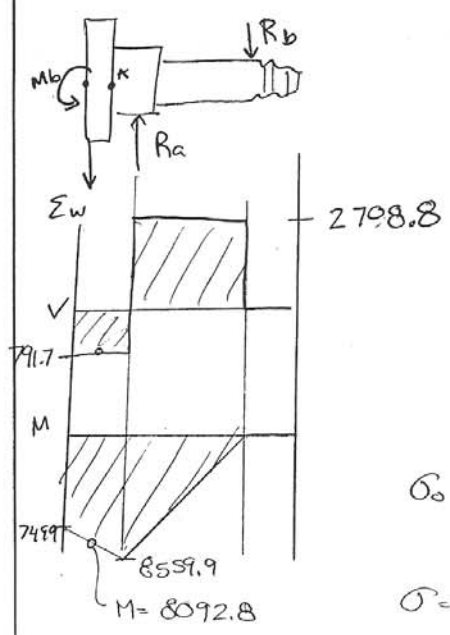
DIA = 1.625"
(I, A, C, +J from pg34)

At pt A:

$M = 8092.8 \text{ in lb}$

$V = 791.7 \text{ lb}$

$M_{ts} = 5341.3 \text{ in lb}$

$$\sigma_o = \frac{Mc}{I} = \frac{8092.8 \times .813}{.342} = 19.24 \text{ ksi}$$

$$\sigma = K\sigma_o = 2.2(19.24) = \underline{42.32 \text{ ksi}}$$

$$\tau_v = \frac{V}{A} = \frac{791.7}{2.07} = 382.5 \text{ psi}$$

$$\tilde{\tau}_{t_o} = \frac{M_{ts} \, c}{J} = \frac{5341.3 \times .813}{.684} = 6.35 \text{ ksi}$$

$$\tau_t = K\tilde{\tau}_{t_o} = 1.8(6.35) = 11.43 \text{ ksi}$$

$$\tau = \tilde{\tau}_v + \tau_t = .3825 + 11.43 = \underline{11.81 \text{ ksi}}$$

$$\sigma_{vms} = \sqrt{42.32^2 + 3(11.81)^2} = 47.00 \text{ ksi}$$

$$SF = \frac{85}{470} = \boxed{1.8} \qquad \geq 1.25 : \text{ ok}$$

NOTE: If the assumption is made that there are no stress concentrations because SST 304 is a ductile material & the factor of safety is calculated based on yield ($s_y = 35 \text{ ksi}$) $\rightsquigarrow$ $\boxed{SF = 1.56}$ $\geq 1 \; \therefore \; \text{ok}$

Seismic (pt. B) → Not needed because pts. A+C
had higher stresses in the static case.

Pt. C (Seismic):

$M_c = 8559.9 - 2708.8(21.2" - 1.34") = 6447$

$V_c = 2708.8$

$\sigma_o = \dfrac{M_c C}{I} = \dfrac{6447\,(.688)}{.175} = 25.35$

$\sigma = 1.65\,(25.35) = \underline{41.8\ \text{ksi}}$

$\tau_v = \dfrac{V_c}{A_c} = \dfrac{2708.8}{1.48} = 1.83\ \text{ksi}$

$\tau_{t_o} = \dfrac{5341.3 \times (.688)}{0.35} = 10.5\ \text{ksi}$

$\tau_t = 1.35\,(10.5) = 14.17\ \text{ksi}$

$\tau = \tau_v + \tau_t = \underline{14.18}$

$\sigma_{vms} = \sqrt{41.8^2 + 3(14.18)^2} = 49.03$

$F_s = \dfrac{85}{49.03} = \boxed{1.73} \geq 1.25 \ \therefore\ ok$

Assuming no concentration factor $\overset{\text{because it's ductile}}{\text{the stress is:}}$

$\sigma_{vms} = \sqrt{25.35^2 + (12.33)^2 \times 3} = 33.15\ \text{ksi}$

$FS = \dfrac{35}{33.15} = \boxed{1.1} \geq 1.0 \quad \text{based on yield}$

Key :
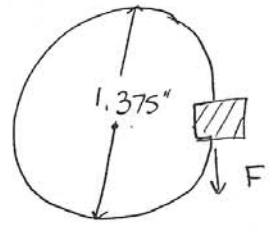
The key will take a shear load from the twist moment.

$$M_t = 2086.3 \text{ in lb}$$

$$F = \frac{M_t}{(1.375/2)} = 3034.6 \text{ lb}$$

  T .3/8" ⊥
├── .75" ──┤
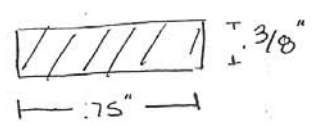
$$A = 0.281 \text{ in}^2$$

$$\tau = \frac{F}{A} = \frac{3034.6}{.281} = 10.79 \text{ ksi}$$

$$FS = \frac{90 \text{ksi} \quad (303SST)}{\sqrt{3}\,(10.79)}$$
$$\tau_{\text{in shear}}$$

$$\boxed{FS = 4.8} \quad \geq 4 \therefore ok$$

Seismic

$$M_{ts} = 5341.3 \quad {}^{\text{Pg 24}}$$

$$F = \frac{M_{ts}}{(1.375/2)} = 7769.2 \text{ lb}$$

$$\tau = \frac{7769.2}{.281} = 27.65 \text{ ksi}$$

$$FS = \frac{90 \text{ksi}}{\sqrt{3}\,(27.65)} = \boxed{1.9} \quad \geq 1.25 \therefore ok$$

Force Ring

The force ring and its components take only the twist moment
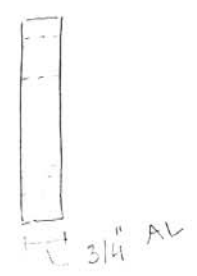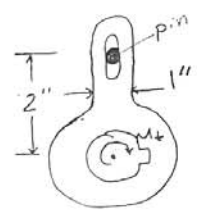
3/4" AL
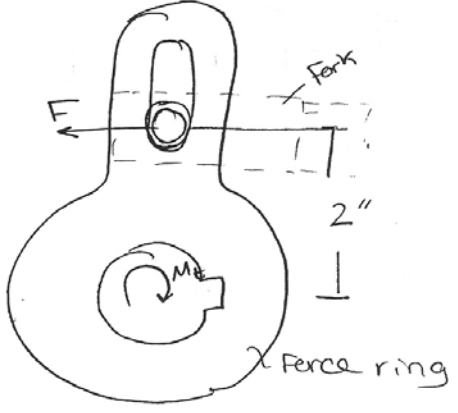
Looking for shear in the pin:

$$F = \frac{Mt}{2''}$$

$M_t = 2086.3 \text{ in lb}$

$F = 1043.2 \text{ lb}$

Pin: DIA = $3/8$"   A2-70
                      $S_{ut} = 101.5$

$$A = \frac{\pi d^2}{4} \sim .11 \text{ in}^2$$

$$\tau = \frac{4}{3}\frac{F}{A} = \frac{4(1043.2)}{3(.110)} = 12.64 \text{ ksi}$$

$$FS = \frac{101.5 \text{ ksi}}{\sqrt{3}(12.64)} = \boxed{4.6} \geq 4 \therefore ok$$
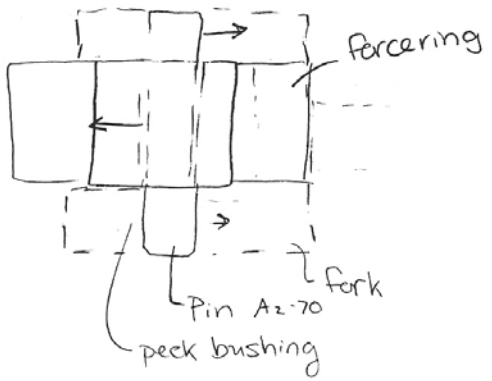
Seismic:

$M_{ts} = 5341.3$

$$F = \frac{5341.3}{2''} = 2670.65 \text{ lb}$$

$$\tau = \frac{4(2670.65)}{3(0.110 \text{ in})} = 32.37 \text{ ksi}$$

$$FS = \frac{101.5}{\sqrt{3}(3237)} = \boxed{1.81} \geq 4 \therefore d$$

Stresses in the force ring:
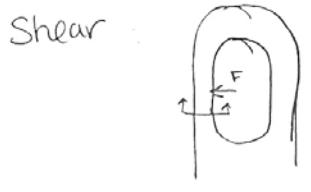
Shear

cross section

$F = 1043.2 \text{ lb}$

.75

.264'
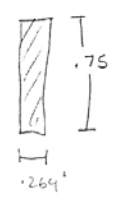
$$\tau = \frac{3}{2}\frac{F}{A} = \frac{3}{2}\frac{1043.2}{(.264 \times .75)} = 7.9 \text{ ksi}$$

$\hookleftarrow$ rectangular section

$$F.S. = \frac{40}{\sqrt{3}(7.9)} \quad \nwarrow \quad S_y \text{ AL 6061-T6}$$

$\downarrow$ cause in shear

$$F.S. = \boxed{2.92}$$

• based on yield ∴ should be greater or equal to 3

$\downarrow$ considered ok because it is a rare event that the mirror is off center 4" which is causing a majority of the twist moment.

Stress at the base

$$\tau_A = \frac{3}{2}\frac{F}{A} = \frac{3}{2}\frac{1043.2}{.75 \times 1} = 2.09 \text{ ksi}$$
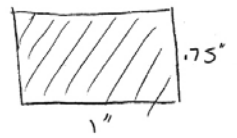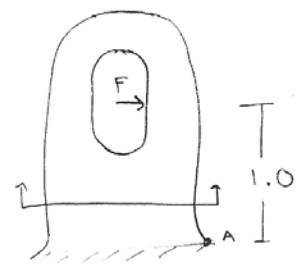
rectangular cross section

$$\sigma_A = \frac{Mc}{I} = \frac{(1.0'' \times 1043.2)(.5'')}{\left(\frac{.75(1)^3}{12}\right)}$$

$\downarrow \frac{bh^3}{12}$

$$\sigma_A = \frac{521.6}{.0625} = 8.35 \text{ ksi}$$

$$\sigma_{vms} = \sqrt{8.35^2 + 3(2.09)^2}$$

$$= 9.10 \text{ ksi}$$

$$F.S = \frac{40}{9.1} = \boxed{4.4} \geq 3 \therefore \text{ok}$$

based on yield

Seismic

Shear at top:

$F = 2670.65$ (pg 42, seismic)

$$\tau = \frac{3}{2}\left(\frac{2670.65}{.264 \times .75}\right) = 20.23$$

$$F.S. = \frac{40}{\sqrt{3}\,(20.23)} = \boxed{1.14} \quad \geq 1 \quad \therefore \text{ ok}$$

based on yield

Stress at the base:

Same seismic F: $F = 2670.65$ lb

$$\tau = \frac{3}{2}\frac{(2670.65)}{.75" \times 1'} = 5.34 \text{ ksi}$$

$$\sigma_\lambda = \frac{Mc}{I} \qquad M = F \times 1" = F$$

$$I = \frac{bh^3}{12}$$

$$= \frac{2670.65\,(0.5)}{\left(\frac{.75 \times 1^3}{12}\right)}$$

$$= 21.37 \text{ ksi}$$
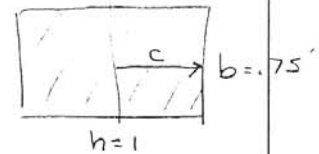
$$\sigma_{vms} = \sqrt{21.37^2 + 5.34^2 \times 3} = 23.29 \text{ ksi}$$

$$FS = \frac{40}{23.29} = \boxed{1.72} \quad \geq 1 \quad \therefore \text{ ok}$$

based on yield

Fork

section

$\frac{3}{8}"$ $.75"$ $\frac{1}{4}$ from Pg 42 $\left(\frac{Mt}{2}"\right)$

$.25"$
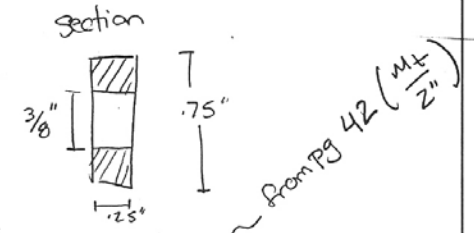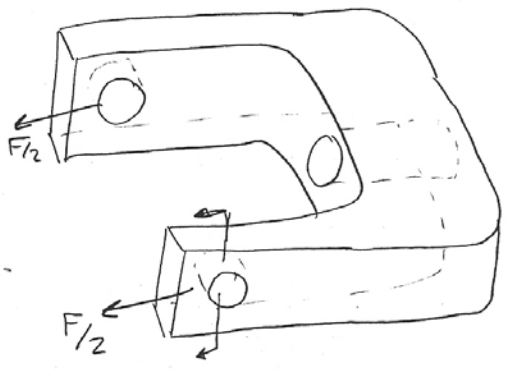
$F/2 = \dfrac{1043.2\ lb}{2} = 521.6\ lb$

$A = .25\ (.75 - \frac{3}{8}) = .0938\ in^2$

$\sigma = \dfrac{F/2}{A} = \dfrac{521.6}{.0938}$

$\sigma = 5.56\ ksi$

$FS = \dfrac{40\ ksi}{5.56} = \boxed{7.19}$ $> 3\ \therefore\ ok$

based on yield

Seismic

from page 42

$\dfrac{F}{2} = \dfrac{2670.65}{2} = 1335.3\ lb$

$\sigma = \dfrac{1335.3}{.0938} = 14.24\ ksi$

$FS = \dfrac{40}{14.24} = \boxed{2.81}$ $\geq 1.0\ \therefore\ ok$

Twist Threaded Rod

changed see Pg 54

petri washer
Peek bushing

$\frac{1}{2}$ thread (Cartrige Brass)
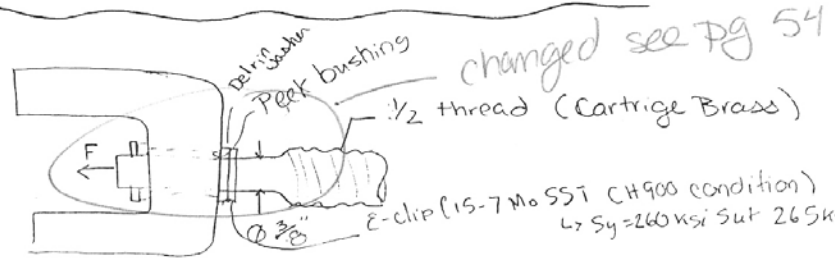
E-clip (15-7 Mo SST CH900 condition)
↳ Sy = 260 ksi Sut 265 ksi — from manufacturer

$\phi \frac{3}{8}"$

$$A = \frac{\pi d^2}{4} = .0721$$

$d = .303''$ for E-clip

$\phi = 3/8''$
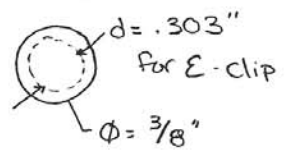
$F = 1043.2$ lb

$$\sigma = \frac{F}{A} = \frac{1043.2}{.0721} = 14.47$$

$$FS = \frac{55}{14.47} = \boxed{3.8}$$ based on ultimate

Sut Brass

Seismic:

$F_S = 2670.65$

$$\sigma = \frac{2670.65}{.0721} = 37.04 \text{ ksi}$$

$$F.S = \frac{55}{37.04} = \boxed{1.48}$$

E-clip:

$\phi = 3/8$

Area: $\pi d (THK) \times \frac{1}{2}$ only half of DIA in shear

THK = 0.039" $= 0.023 \text{ in}^2$
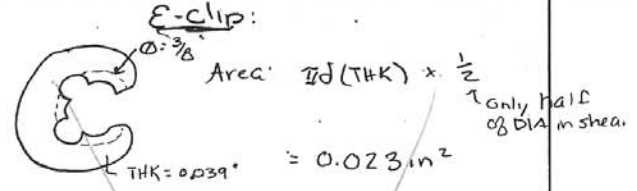
$$\sigma = \frac{F}{A} = \frac{1043.2}{.023} = 45.36 \text{ ksi}$$

$$FS = \frac{265}{\sqrt{3}(45.36)} = \boxed{3.4}$$ based on ultimate

↑ shear

↳ this is considered OK because it is a rare case for the mirror to be off center 4" and the end effector at the full 20° yaw angle.

E-clip

$$\sigma = \frac{2670.65}{(.023)} = 116.1 \text{ ksi}$$

$$F.S. = \frac{265}{\sqrt{3}(116.1)} = \boxed{1.32} \geq 1.25 \therefore OK$$

Thread Stripping:

1/2" - 20   Brass

$F = 1043.2$ lb

$Le = .61''$

$$A_s = \frac{\pi}{2}\left(D_n - \frac{1}{TPI}\right) Le$$

$$= \frac{\pi}{2}\left(\frac{1}{2} - \frac{1}{13}\right)(.61)$$
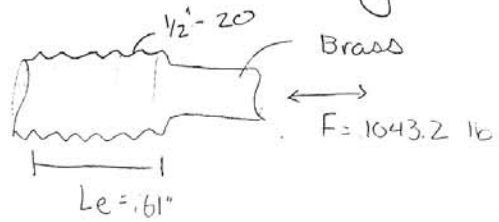
$$= 0.405 \text{ in}^2$$

$$\hat{t} = \frac{F}{A_s} = \frac{1043.2}{.405} = 2.57 \text{ ksi}$$

$$F.S. = \frac{55 \text{ ksi}}{\sqrt{3}(2.57)} = \boxed{12.4} \quad \geq 4 \therefore \text{ ok}$$

Seismic:

$$F = 2670.65$$

$$\hat{t} = \frac{F}{A_s} = 6.59 \text{ ksi}$$

$$F.S. \frac{55}{\sqrt{3}(6.59)} = \boxed{4.82} \quad \geq 1.25 \therefore \text{ ok}$$

Force to turn Crank:

NOTE: The force (F=1043.2 lb) used in the previous calculations was from the twist moment created when the assembly was at its extreem yaw angle + the mirror was off center 4 inches. These are rare cases, but the previous calculations needed to incorporate these assumptions just in case. Under normal circumstances the mirror will be off centered by ≈1" (only because it is roughly centered by sight without meausuring). Also at the time when the "twist" of the assembly would need to be adjusted, the yaw angle would be no more than 8 deg. for the heaviest mirror sub-assembly (FABS). The calculation to find the force needed to turn the crank will be performed under the "normal circumstance" position!
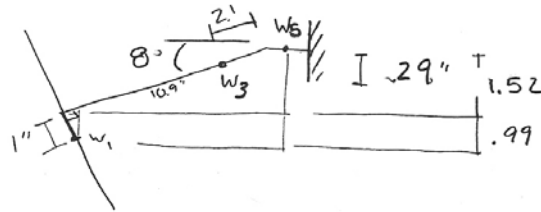
Max angle for FABs:
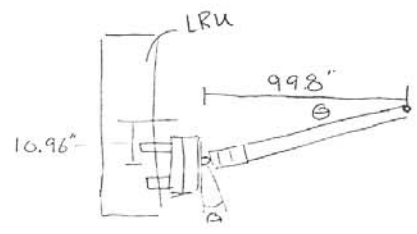
$$\theta = \tan^{-1}\left(\frac{10.96}{99.8}\right)$$

$$\theta = 6.3°$$

to be conservative use $\theta = 8°$

Find new twist moment:

$W_1 = 276$ lb

$W_3 = 26.5$ lb
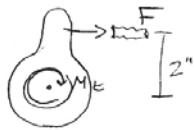
$W_5 = 6.7$ lb

} from page 29

$M_t = W_1 (1.52" + .99") + W_3 (.29")$
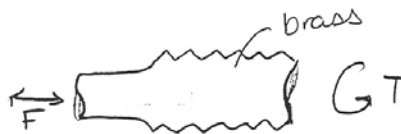
$= 276 (2.51) + 26.5(.29) = 700.4$ in lb

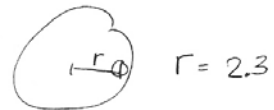$F = \dfrac{M_t}{2"} = 350.2$ lb

↑ force on threads

$T = K (D_n)(F)$

$K = .1$ (see pg 27)

$D_n = 0.5$ in

brass

GT

$T = .1 (.5)(350.2) = 17.5$ in lb

Force to turn wheel $= \dfrac{T}{r_{wheel}}$ $r = 2.3$
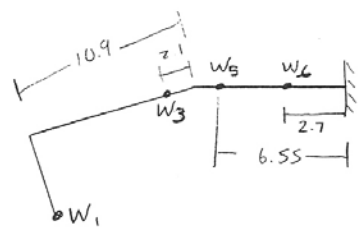
$= \dfrac{17.5}{2.3} = \boxed{7.6 \text{ lb}}$

Once again this is for the heavies mirror assembly.
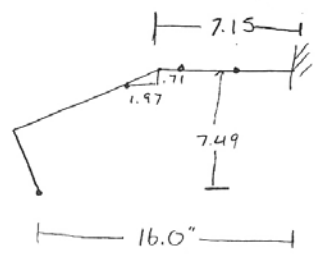Most cases will be about 2/3 the force.

Adapter Block
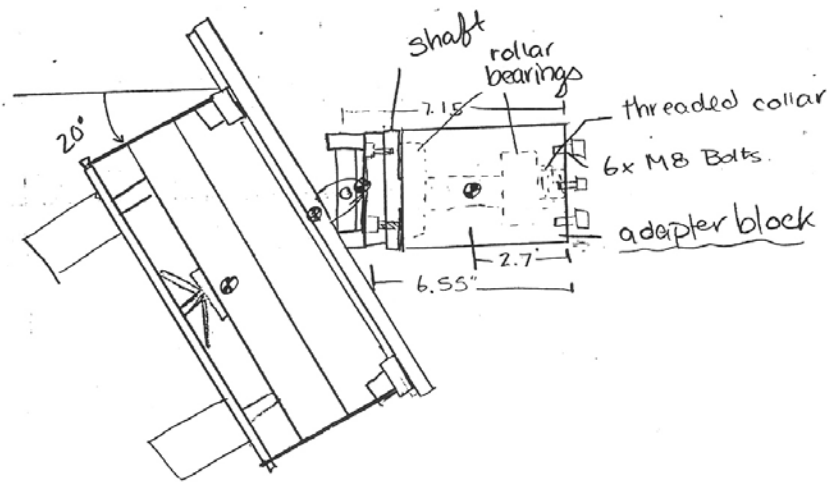
$W_1 = 276 \text{ lb}$

$W_3 = 26.5 \text{ lb}$

$W_5 = 6.7 \text{ lb}$

$W_6 = 17.9 \text{ lb}$
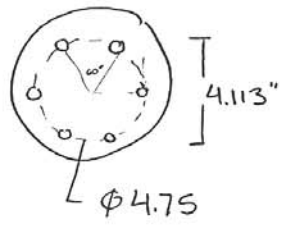(shaft + Adapter block)

$M_t = 2086.3$ (pg 21)

$M_b = W_1(16.0) + W_3(1.97 + 7.15)$
$\quad + W_5(6.55) + W_6(2.7)$

$\quad = 4749.9 \text{ in lb}$

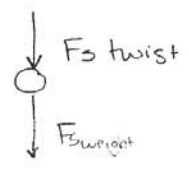$$F_a = \frac{M_b}{2(4.113)} = \frac{4749.9}{2(4.113)} = 577.4 \text{ lb}$$

$$FS_{weight} = \frac{\Sigma W}{3_\kappa} = 109.03 \text{ lb}$$

assumes only 3 out of the 6 screws take the load.

$$FS \text{ twist} \Rightarrow \frac{M_t}{3(\phi/2)} = \frac{2086.3}{3(2.375'')} = 292.8 \text{ lb}$$

↳ assumes only 3 out of the 6 screws take the load

Fs ⇒

Fs twist

F_sweight

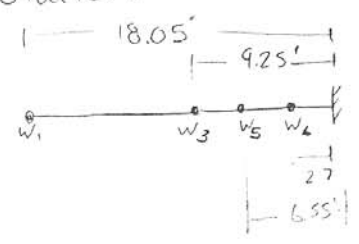$$\Rightarrow F_s = 109.03 + 292.8 = 401.84 \text{ lb}$$

$$\sigma = \frac{1}{A_t}\sqrt{F_a^2 + 3 F_s^2}, \quad A_t = 0.0567 \text{ in}^2 \quad \text{for M8}$$

$$\sigma = \frac{1}{0.0567}\sqrt{(577.4)^2 + 3(401.84)^2} = 15.95 \text{ ksi}$$

$$FS = \frac{72.5}{15.95} = \boxed{4.5} \qquad \geq 4 \therefore \text{ ok}$$

Pull out:

Fa, worst case is when the rail plate has 0° rotation

$$M_b = W_1(18.05) + W_3(9.25) + W_5(6.55) + W_6(2.7) = \Rightarrow$$

$$M_b = 5319.1 \text{ in lb}$$

$$F_a = \frac{M_b}{2(4.113)} = 646.6 \text{ lb } / \text{ M8 bolt}$$

$$A_s = \frac{\pi}{2}\left(D_n - \frac{1}{TPI}\right) L_e$$

$$= \frac{\pi}{2}(.315 - .08)(.63") = 0.26 \text{ in}^2$$

$$\tau = \frac{646.6}{.26} = 2465.9 \text{ psi} = 2.466 \text{ ksi}$$

$$FS = \frac{40 \swarrow^{S_y \text{ AL}}}{\sqrt{3}(2.466)} = \boxed{9.36} \geq 3 \therefore \text{ ok}$$

Torque:

$$T = K \cdot D_n \times F_p$$

$$F_p \geq 1.6 F_a = 1034.6 \text{ lb}$$

$$F_p \leq 0.65 (A_s)(S_{ut}) = .65(.0567)(72.5E3)$$
$$= 2672.0$$

Choose $F_p = 1850 \text{ lb}$

$$K = 0.3 \quad (\text{SST} \to \text{AL} ; \text{Dry})$$

$$T = 0.3(8/25.4)(1850) = 174.8 \rightsquigarrow \boxed{175 \text{ in lb}}$$

Seismic

$SP_{x_1} = 172.2$

$SD_{x_3} = 16.5$

$SP_{x_5} = 4.2$

$SD_{x_6} = 11.2$

$SD_{y_1} = 430.6$

$SD_{y_3} = 41.3$

$SD_{y_5} = 10.5$

$\left.\begin{array}{c} \\ \\ \end{array}\right\}$ $\frac{\text{lb}}{(\text{from page 32})}$

$SD_{y_6} = 279$

$M_{t_s} = 5341.3\,lb$ (W_4 does not add to twist, so it's the same as on pg 40)

$M_{b_s} = (W_1 + SD_{y_1})(16) + (W_3 + SD_{y_3})(1.97 + 7.15) + (W_5 + SD_{y_5})(6.55)$
$\quad\quad + (W_6 + SD_{y_6})(2.7)$

$\quad\quad = 12{,}160\ in \cdot lb$

$F_a = F_{a_{SD_x}} + F_{a_{M_b}}$

$\quad F_{a_{SD_x}} = \dfrac{\sum SD_x}{6} = 34.02\ lb$

$\quad F_{a_{M_b}} = \dfrac{M_b}{3(4.113")} = 985.49\ lb$

$\quad\quad F_a = \underline{1019.5\ lb}$

$F_s = F_{s_{twist}} + F_{s\ weight}$

$\quad F_{s_{twist}} = \dfrac{M_{ts}}{3(\phi/2)} = \dfrac{5341.3}{3(2.375)} = 749.66\ lb$

$\quad F_{s\ weight} = \dfrac{\sum SD_y + \sum W}{3} = 279.13\ lb$

$\quad\quad F_s = \underline{1028.8\ lb}$

$\sigma = \dfrac{1}{0.0567}\sqrt{(1019.5)^2 + 3(1028.8)^2} = 36.21\ ksi$

$FS = \dfrac{72.5}{36.2} = \boxed{2.0} \geq 1.25 \therefore ok$

Pull out:
Find M_b with zero yaw rotation.

$$M_b = (W_1 + SD_{y1})(18.05) + (W_3 + SD_{y3})(9.25) + (W_5 + SD_{y5})(6.55)$$
$$+ (W_6 + SP_{y6})(2.7)$$

$$= 13,617.6$$

$$F_c = \frac{M_b}{2(4.113)} = 1655.4$$

$$\tau = \frac{1655.4}{.26} = 6.37 \text{ ksi}$$

$$FS = \frac{40}{\sqrt{3}(6.37)} = \boxed{3.63} \geq 1 \therefore ok$$

Robohand

Manufacturers rated load   Applied Load

$$M_{b_{max}} = 6000 \text{ in lb}$$ 5319.1 in lb (Pg 50)
                                            ↑ non rotated
                                               position

$$M_{t_{max}} = 9000 \text{ in lb}$$ 2086.3 in lb

- in both cases the applied load is
  under the maximum.