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SYSTEM DEVELOPMENT FOR AUTOMATED PAVEMENT CRACK SEALING

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System Development for Automated Pavement Crack Sealing

BY

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ABSTRACT

In an effort to make highway crack sealing more efficient, safe, and cost effective, the Advanced Highway Maintenance and Construction Technology (AHMCT) Center at University of California, Davis constructed the Automated Crack Sealing Machine (ACSM) in 1993. While a success, this technology demonstration prototype was not practical for use in the highway maintenance system. The ACSM was comprised of numerous systems that would be more effective in the highway maintenance system if they were redesigned for individual use. Two of the less complicated individual machines are the Longitudinal Crack Sealing Machine (LCSM) and the Operator Controlled Crack Sealing Machine (OCCSM). Together these two machines will be able to seal all types of cracks encountered on roadways.

This thesis presents the design and development of various systems that are essential to the operation of the LCSM and OCCSM. A pneumatically powered linkage is designed to position the sealant applicator on the LCSM. For the OCCSM, a retractable camera boom and a camera housing are designed for use with the image processing system and a sealant hose management system is developed that allows the hose to reach all regions of the workspace. Also, an electrically heated sealant hose is developed and efforts are made to increase sealant melting capabilities for all crack sealing operations. This thesis discusses each of the above topics in detail and provides tests results where available and recommendations for future development.

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CHAPTER 1: INTRODUCTION

Government agencies spend a total of $18 billion annually on road maintenance alone. Nearly $200 million of the government's annual expenditure is spent on crack sealing efforts (Highway Statistics, 1992). Approximately two thirds of this cost is labor, one quarter equipment and maintenance, and the remainder attributed to materials (Velinsky, 1993a). Highway maintenance work is not only expensive, but also very dangerous. Workers are exposed to the hazards of errant drivers as well as the dangers of working with heavy machinery. Equipment that efficiently automates road maintenance significantly lowers overall labor costs and, perhaps more significantly, reduces workers exposure time on the roadway. An automated system also has the advantage of keeping workers protected in the maintenance vehicle and reducing the number of workers required to support the operation. To these ends, the Advanced Highway Maintenance and Construction Technology (AHMCT) Center at University of California, Davis constructed the Automatic Crack Sealing Machine (ACSM) in 1993.

The ACSM, a technology demonstration prototype, would sense, prepare and seal all the cracks in a highway lane in one pass (Velinsky, 1993b). Although this machine was a success, there were several factors that made it impractical for use in the highway maintenance system. The main factor was its production cost, which was estimated to be nearly 1500% of the entire annual crack sealing budget for an average county. While a detailed cost benefit analysis showed that the machine would return its initial investment in 2 years, highway maintenance organizations can not easily procure the funds necessary to purchase the machine (Velinsky, 1993a). Maintenance of the numerous systems on the ACSM was another concern. Because of the problems associated with the complex

ACSM, recommendations were made to develop simpler machines that would focus on individual crack sealing tasks. The purpose of this thesis is to document the development of systems essential to two simpler crack sealing machines.

1.1 Longitudinal Pavement Cracks

The first machine, the Longitudinal Crack Sealing Machine (LCSM), is dedicated to sealing straight cracks that run parallel to the roadway. Typically, these cracks lie between the roadway shoulder and lane, serving as a thermal expansion joint between the two surfaces. The expansion joint is important because of the difference in expansion rates of the lane material, usually made of Portland Cement Concrete (PCC), and the shoulder material, usually Asphalt Concrete (AC). The linearity of longitudinal cracks readily leads to automation of the task. While sealing straight cracks is relatively simple it comprises, at most, 25% of crack sealing operations (Velinsky, 1993a). The remaining 75% of cracks that require sealing meander randomly through the pavement and require a more complex sealing machine.

1.2 Random Pavement Cracks

The Operator Controlled Crack Sealing Machine (OCCSM) has been designed to seal the random cracks. These cracks are typically narrow and often lie in the lanes where the pavement bears the weight of vehicles. The primary cause of this type of cracking is the cyclic loads of vehicles. Thermal cycles cause another large portion of random cracks. These cracks can run across the entire roadway. The OCCSM's 3.7 x 3.7 m (12 x 12 ft) workspace extends far enough to seal cracks across a complete lane width. Figures 1.1 and 1.2 show examples of the two types of cracks.

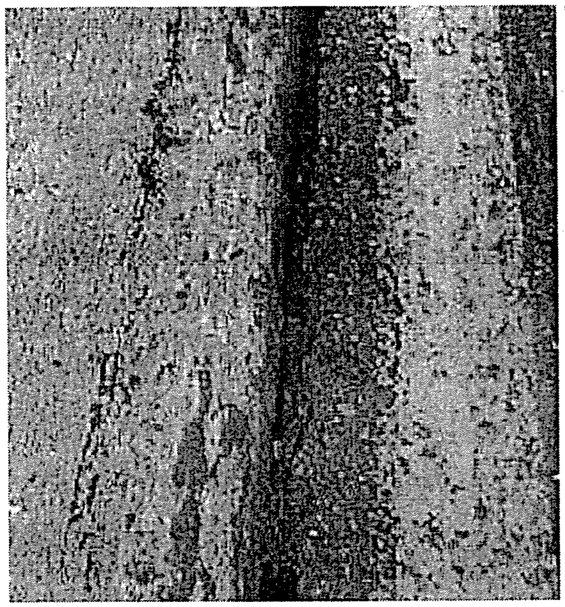

Figure 1.1 *A Longitudinal Crack*

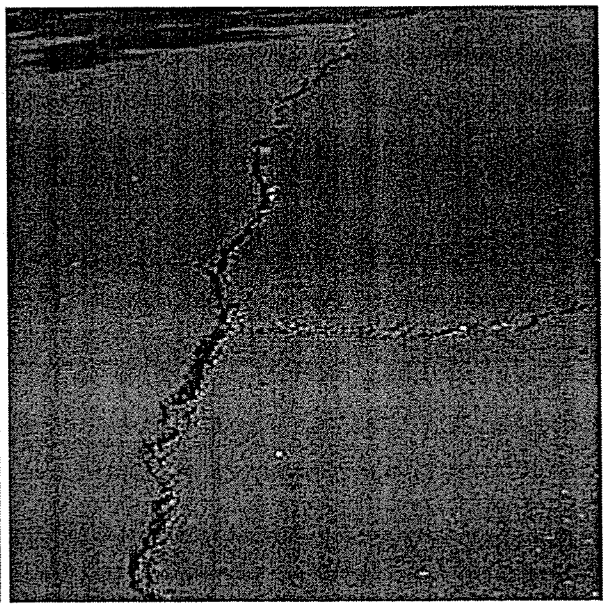

Figure 1.2 *A Random Crack*

1.3 Overview of the Longitudinal Crack Sealing Machine

As previously mentioned the LCSM has been designed to seal longitudinal cracks in the roadway, most often the construction joint between the PCC road surface and the AC shoulder material. The main system on the LCSM designed for this specific task is a linkage assembly attached to the front of the vehicle that moves the sealant applicator from its stowed position to the working position. When stowed, the entire assembly is located within the width of the vehicle, as specified by the design requirements. In the working position the sealant applicator is about 0.61m (2 ft) to the side of the truck and contacts the pavement in view of the operator in the truck cab. An important feature of the linkage is that it can be arranged to seal cracks on either side of the vehicle. A hose is used to supply the sealant to the sealant applicator from a sealant melter, which melts the sealant material and raises it to its working temperature of approximately 193°C (380°F).

The following paragraph describes the operating procedure of the LCSM. While still in the equipment yard, the linkage on the front of the truck is setup to work on a specific side of the truck. Once at the work site, the operator controls the linkage to move the sealant applicator to its working position and then positions the truck so the sealant applicator is over the crack. Next, the sealant is turned on, begins to flow out the sealant applicator, and the operation starts to move. The driver controls the flow of sealant while driving along the crack, steering carefully to keep the sealant applicator over the crack. When the job is finished the operator stops the sealant flow and, using the linkage, positions the sealant applicator back in the stowed position.

This thesis discusses the design of the LCSM sealant applicator positioning linkage in Chapter 4. Design requirements are reviewed and explanation is made of how the overall configuration satisfies these requirements. Then individual components, including the motion actuators, are analyzed in more detail.

1.4 Overview of the Operator Controlled Crack Sealing Machine

The OCCSM is comprised of several systems which must work together in order to seal random cracks. The primary system on the OCCSM is the robotic arm (see Baker, 1998), which operates in the workspace behind the truck. The sealant applicator, different from the applicator used on the LCSM, attaches to the end of the arm and follows the crack while applying sealant. Above the workspace a camera, attached to the end of a boom, supplies pictures via computer to the operator in the truck cab. This boom, called the camera boom, mounts above the truck bed and retracts when not in use. The operator views the picture supplied by the camera system on a touch screen and traces out a crack in the workspace. This path is sent to a motion controller which commands the actuators that move the robotic arm. The sealant melter in the truck bed supplies liquid sealant, through a hose, out to the end of the arm. A hose retraction system keeps tension in the hose while it extends and retracts with the arm. All of the systems comprising the OCCSM reside on one vehicle, including the sealant melter and delivery system.

The basic operating procedure of the OCCSM is as follows. The operator positions the vehicle just in front of the crack to be sealed. The overhead camera takes a picture of the road surface in the workspace and the picture is displayed in the cab for the operator. The operator traces a crack on the touch screen and the robotic arm follows the crack while dispensing sealant.

This thesis documents the development of some of the systems present in the OCCSM. The camera boom and its retraction mechanism are discussed in Chapter 5. Vibration characteristics of the camera boom are analyzed, and braces, that were determined necessary from the vibration analysis, were designed. The hand operated crank mechanism, which extends and retracts the camera boom by means of a rack and pinion design, is examined. The design of the camera housing that attaches to the end of the boom and provides adjustments for camera alignment is also discussed. Chapter 6 focuses on the hose retraction system and several design options that were considered for powering the mechanism. The chapter also explores how the system was designed to fit in the small area beneath the truck bed and between the frame rails.

1.5 Sealant Melting and Supply

The LCSM and OCCSM have a system in common that is equally important to each machine. Neither machine would accomplish any crack sealing without their sealant melting and supply system. The sealant melter itself can be purchased commercially from one of several manufacturers. The hose that delivers the sealant from the melter to the sealant applicator is typically custom-made for each machine. There are problems with the current sealant melter and supply system for both the LCSM and OCCSM.

Sealant melters currently available do not have the ability to keep up with the amount of sealant demanded by the LCSM. The OCCSM is projected to have this problem as well. Sealant is supplied in solid blocks, approximately 33 x 25 x 18 cm (13 x 10 x 7 in) in size, and workers constantly add the material to the melter during the process of sealing cracks. Even with the constant addition of sealant, work crews must stop sealing approximately every hour and wait at least a half-hour for the melter to catch up. This problem and a solution, aimed at melting sealant blocks more quickly, are discussed in the next chapter.

Supplying the sealant through a hose to the sealant applicator presents its own unique problems. Once a hose is used, it remains at least partially full of sealant that will solidify between uses. Therefore, the sealant left in the hose must be melted before the operation can begin again. Because the OCCSM design requires the hose to reside permanently inside the robotic arm, a hose heating system had to be developed for the machine. A heated sealant hose also provides advantages for the LCSM, which currently, and inconveniently, keeps the hose hot using heat from the melter. Chapter 3 of this thesis discusses the development and advantages of a self contained heated sealant hose.

Together, the LCSM and OCCSM automatically seal all types of cracks in the roadway. They both require the operator to “tell” the system where the cracks are, but neither requires a worker on the roadway to apply the sealant. With both of these systems every crewmember enjoys the safety of being in the vehicle. Both systems have the ability to increase the rate at which cracks are sealed and to improve worker safety. The purpose of the work documented in this thesis is to assist in the automation of pavement crack sealing through the design and development of the LCSM and the OCCSM.

CHAPTER 2: THE SEALANT MELTER

Every crack sealing operation requires the sealant to be liquid when applied to the crack. The most common method to obtain liquid sealant uses a sealant melter to melt solid blocks of material. Melters are commercially available either on a trailer, as used with the LCSM, or free standing, which the OCCSM uses. The block melting process begins with the melter's diesel burner, the heat source for the operation. The burner supplies heat to the sealant via heat transfer oil that is pumped through a network of coils dispersed throughout the sealant vat. Heat transfer oil is used because heating the sealant directly from the burner would overheat and coke the material. While this system works, it has the major drawback that it cannot keep up with the amount of sealant demanded by many crack sealing operations. The sealant blocks added to the melter are not melted fast enough to replace the liquid sealant used by the operation. Even with large 400 gallon melters, current manual operations must stop approximately every 1 ½ hours to wait for the melter to recover (Technology Development Center, 1993). Faster, automated operations will run out of sealant even more quickly.

The simplest way to solve this problem without redesigning the melter was to reduce the size of the sealant blocks. Because conduction and convection increase linearly with surface area, an increase in area would cause a proportional decrease in melting time. The main objective of the design discussed in this chapter is to increase the amount of surface area per volume of sealant.

2.1 Surface Area Increase from Slicing Sealant Blocks

The increase in surface area depends on the number of cuts made through the block as well as their orientation. A block of width w, height h and depth d has an initial surface area given by

$$A_i = 2wh + 2wd + 2hd. \tag{2.1}$$

If the block is cut n times parallel to the w-h plane, the new total surface area is

$$A_f = A_i + 2nwh. \tag{2.2}$$

The percent increase in area is

$$A_{inc} = 2nwh/A_i * 100\%. \tag{2.3}$$

A typical sealant block measures 33 x 25 x 18 cm (13 x 10 x 7 in), although they are available in several different sizes. Making three slices through the block would increase the surface area by a maximum of 130%. From equation 2.3, it is clear that cutting parallel to the largest side of the block provides the greatest increase in surface area for a given number of cuts. However, because sealant blocks are rolled down a ramp from the back of the truck into the melter, shown in Figure 2.1, it is not possible to limit their orientation to the ideal case. Furthermore, the number of cuts is restricted by the amount of power available to heat the cutting blades and to push the blocks through the blades.

2.2 Sealant Block Shredder

The system used to slice blocks, called the sealant block shredder, consists of two main components. The first of these components, the blades, slice the blocks into smaller pieces. The second component, which pushes sealant blocks through the blades, is a hydraulic press mechanism that had already been designed and fabricated. The press mechanism and blades are enclosed by sheet metal and the assembly mounts to the top of the melter as shown in Figure 2.1. The blades are positioned about level with the top of the melter so the sliced blocks fall directly into the sealant vat. The next several sections discuss the blade design and aspects of controlling the shredder press.

2.2.1 Shredder Blade Design

Blades needed to be designed to slice the blocks into smaller pieces. The tough, rubber-like quality of the sealant material dictated that the blades had to be heated in order to slice through the blocks. The method used to heat the blades had to supply enough heat to slice through a block in no more than a minute, as well as heat the blades back up to operating temperature between block cutting cycles. Tests were performed with several blade designs, varying both the blade geometry and the location of the heating elements, to determine the best configuration.

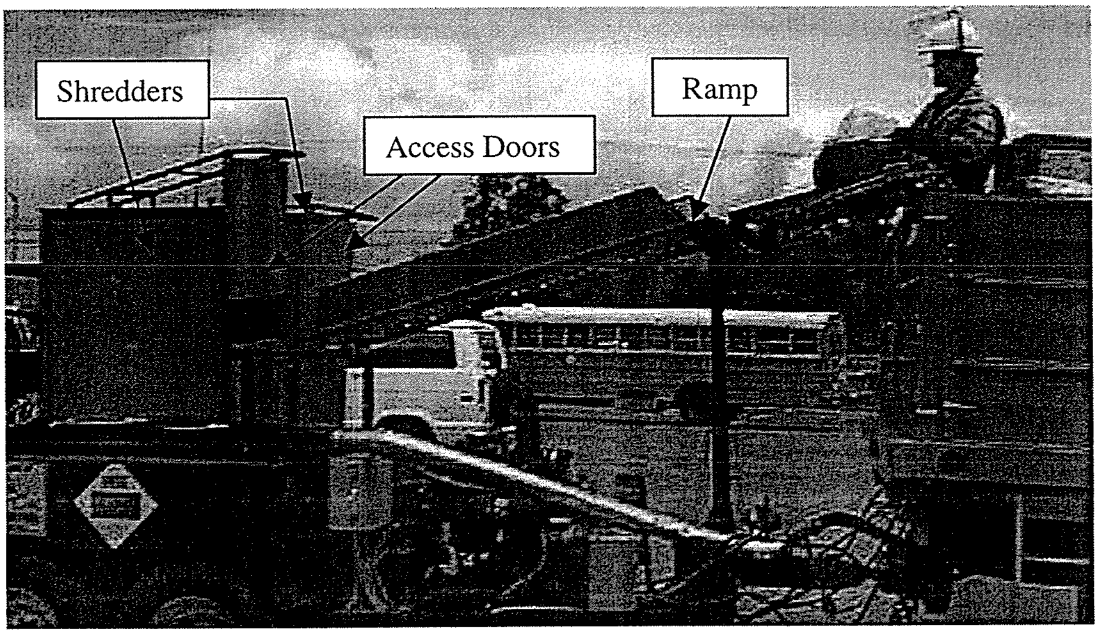

Figure 2.1 *Sealant Supply System*

2.2.1.1 Blade Heating System

Two heat sources were considered for heating the blades, hot oil and electric heating elements. Hot oil could be supplied by tapping into the melter's heat transfer oil system and routing hot oil through tubes built into the blades. This method was not chosen because of the difficulty involved in routing additional oil lines and the inconvenience of heating up the oil system to test each new blade design. Even though using electric power required generators, it was chosen due to being the most practical for prototype purposes. Two generators, providing 8 kW of electricity, were used to power a total of eight blades, four in each shredder. Thermocouples, placed in the blades and connected to a temperature controller, regulated power to the heating elements. Although electric power was the immediate choice, once a proven blade design was found, heating could be converted to hot oil and the need for generators eliminated.

2.2.1.2 Blade Structure Design

Experiments were done with different blade designs to determine the effectiveness of each. First, a design with the heating element used as the tip of the blade and supported by an aluminum bar was tried; see the cross section shown in Figure 2.2. With this design, a cut as wide as the heating element had to be made in the block for each slice. Because so much material was being melted by the heating element, blocks were sliced slowly even though the heating element was in direct contact with the sealant. Heating elements that were significantly smaller and delivered the same amount of power were not available to make this design practical.

Another design had the heating element sandwiched between two pieces of sheet metal; a cross-section is shown in Figure 2.3. This design provided a relatively sharp tip

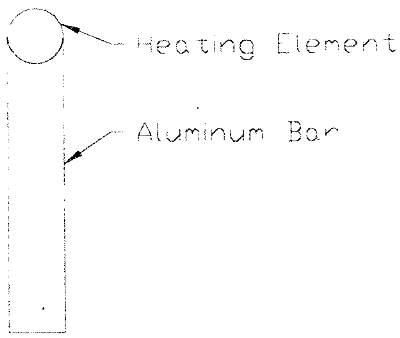

Figure 2.2 *First Blade Design*

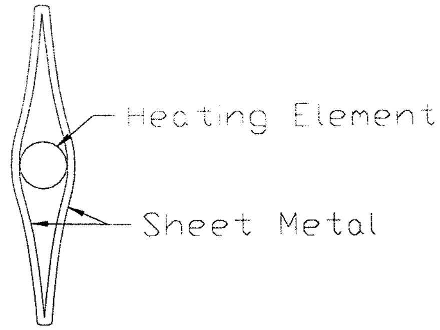

Figure 2.3 *Second Blade Design*

on the blade, reducing the amount of sealant that needed to be melted to make a cut. While this blade was effective at cutting sealant blocks, it would not be strong enough to withstand the forces applied by the sealant press. The final design combined aspects of these first two blade designs.

The best blade design, whose cross-section is shown in Figure 2.4, was found to be one with a relatively sharp leading edge made from sheet metal. The heating element was put as close to the edge as possible. To provide strength, these components were mounted to a 5.08 x .635 cm (2 x .25 in) steel bar. A heat conductive epoxy was placed

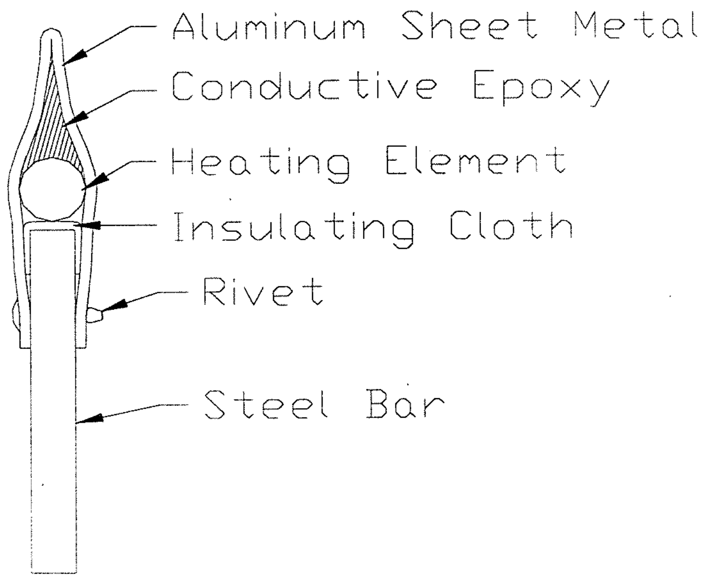

Figure 2.4 *Final Blade Design*

between the blade tip and the heating element, and an insulating cloth was placed between the heating element and the steel bar to direct most of the heat to the blade tip. Thermocouples, used to control the electric heating elements, were placed in the tip of the blade and next to the heating element to ensure that the temperature did not exceed the sealant's maximum recommended temperature of 204°C (400°F). Figure 2.5 shows a set of four completed blades in the melter. The tube coils that route the heat transfer oil through the sealant vat can be seen below the blades.

2.2.2 Controlling the Shredder Press

The hydraulic cylinders power each press and are controlled by the operator using switches located on the end of the ramp that extends from the melter to the back of the truck; see Figure 2.6. To add a block to the melter, the operator pushes the switch, raising the press and access door, and then rolls blocks down the ramp onto the blades. Pulling the switch shuts the access door and causes the press to begin pushing the blocks through

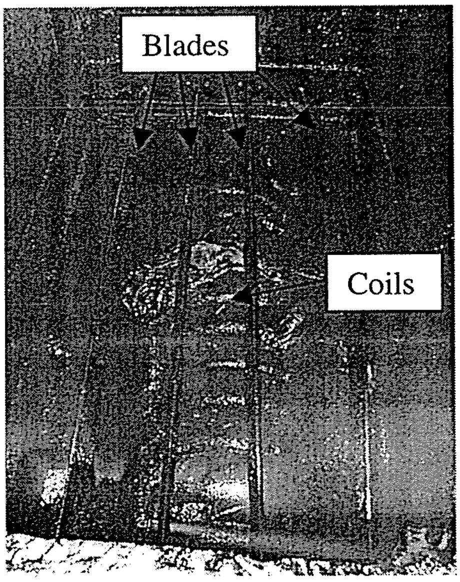

Figure 2.5 *Four Blades in the Shredder*

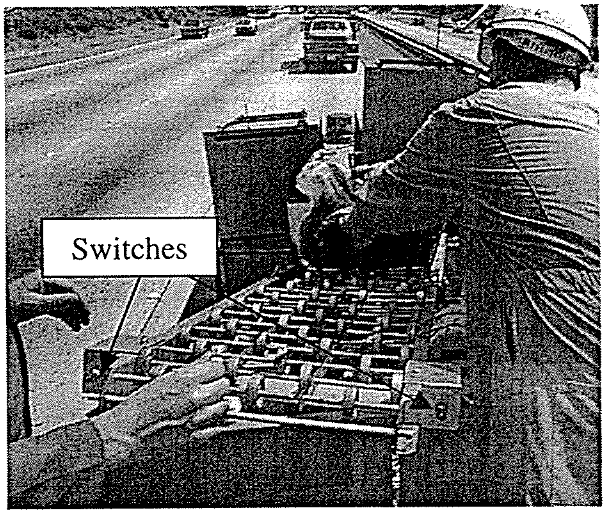

Figure 2.6 *Control Switches on Ramp*

the blades. The switches work to control each press through a programmable logic controller (PLC), allowing effective control of each press and providing essential safety features.

2.2.2.1 Position Switches

Magnetic bolts strategically placed at three locations on the mechanism trigger a magnetic position switch that is connected to the PLC. The signal from the position switch allows the PLC to monitor the press mechanism's position relative to three important locations. Two of the locations are simply end of travel positions, one at the bottom signaling that the press has completed slicing a set of blocks, and one at the top signaling that the system is ready to accept more blocks. Incorporating a safety feature into the system, the third location marks the press mechanism's position when the access door should just barely be closed. As the press lowers, if anything prevents the door from closing when the mechanism reaches this position, the press stops until the obstruction is cleared and the operator issues another command for the press to move. In fact, anytime the press mechanism is between the position where the door closes and its bottom end of travel, the press will stop if the door is not completely shut. A separate magnetic switch tells the PLC whether the access door is open or closed.

2.2.2.2 The PLC Program

The PLC tracks a total of 8 inputs for the block cutting system. Each shredder requires four: a position magnetic switch, the door status magnetic switch, and two for the push/pull switch located at the end of the ramp. Four outputs are used, two for each shredder, to control the hydraulic cylinders. Other input and output channels on the PLC are used for the LCSM mechanism discussed in Chapter 4. A program, created in the Relay Ladder Logic (RLL) programming language, monitors all the inputs and controls the outputs accordingly. Key features of this program include position monitoring using the magnetic switches described above, the safety check ensuring that the access door is closed, and one touch shredder motion control.

Because of the limited number of input channels, a single position switch must be used to monitor the mechanism's location relative to the three important positions described in the previous section. The program employs a counter that counts up or down, depending on whether the mechanism is moving up or down, whenever the position switch is triggered. The counter value is zero when the mechanism is at the bottom of its travel, then becomes one just before it reaches the position where the access door begins to open, and finally becomes two when the upper end of travel is reached. On the way down the counter value drops to one as soon as the mechanism moves below the upper end of travel position, then drops to zero at the middle position where the door closes. The counter value divides the mechanism's position into two main regions, the first when the counter's value is zero and the second for a counter value of one.

These two regions help provide a safety feature by defining that the access door should be closed when the counter value is zero. Checking the door status and the counter value helps prevent foreign objects from getting caught in the press. For example, in order to shred more blocks at a time, a worker may attempt to make room for more blocks by adjusting the sealant blocks already in the shredder, possibly using a pole or their own hand. If the device the worker is using prevents the access door from closing, the program stops the mechanism when the counter reaches zero, hopefully preventing any injury or damage.

The PLC program allows the shredder's motion to be controlled by a single push or pull on its switch. To start the press the operator either pushes the switch to raise the mechanism or pulls it to lower the mechanism. Once in motion, the press will continue until an end of travel position is reached, an access door fault occurs, or the operator stops the press. The operator can stop the press by either pushing or pulling the switch, providing a quick way to stop the mechanism if a problem arises. The one touch control of the operation permits the operator to spend more time on duties other than controlling the shredder. For example, while one shredder slices blocks, the operator can open packages of sealant and load the other shredder.

2.3 Testing and Results

When the modified melter was tested in a crack sealing operation, the shredders worked well, however, an increase in operation speed was not realized due to deficiencies in the melter itself. The test was conducted on a longitudinal crack and began with the melter full of sealant at the operating temperature of 193°C (380°F). Using the shredders, blocks were added to the melter once sealant began to be drawn from the melter. Each shredder performed well, slicing a set of two or three blocks in about a minute. The operation ran smoothly for about 45 minutes, at which time the sealant's temperature in the melter had dropped below the suitable range for application. At this point the operation stopped to wait for the melter to heat up.

Even though the sealant in the melter was below application temperature, the majority of it was liquid. This is notably different from using a melter without block slicing capabilities, where the operation stops because the sealant blocks have not melted and little liquid sealant remains in the melter. These observations show the qualitative result that slicing the blocks causes them to melt faster, but at the cost of lowering the temperature of all the sealant in the melter, much like dropping ice cubes in a glass of hot water. At this point it became clear that the melter could not deliver enough heat to the sealant.

Studying the operation of the melter itself revealed two key facts leading to the deficient element of the system. First, the heat transfer oil was always near its maximum temperature; indicating heat transfer from the diesel burner to the oil is sufficient. Second, the diesel burner ran only about half of the time in order to avoid over-heating the oil, proving that the diesel burner can supply ample heat. Since the oil remained at maximum temperature and the sealant was relatively cool, it may be concluded that the real problem lies in transferring heat from the oil to the sealant. That is, the coils containing the heat transfer oil do not provide enough surface area to adequately transfer heat to the sealant.

2.4 Summary

This chapter has discussed the design of a system used to slice sealant blocks in an effort to expedite their melting for roadway crack sealing operations. A set of tests were designed and performed to assess the benefits of this system and the overall abilities of a sealant melter. From the tests, it can be concluded that slicing blocks certainly expedites the melting of individual blocks, but the actual limiting factor in the melting process is the melter's ability to transfer heat from the heat transfer oil to the sealant. It is clear that significant increases in sealant melting can only be accomplished through improving this heat transfer subsystem, but the block slicing system developed here will be an essential feature in maximizing melting ability.

CHAPTER 3: DEVELOPMENT OF A HEATED SEALANT HOSE

Complications arise in the sealant delivery systems of crack sealing operations resulting from the characteristics of the sealant material and the nature of the operation. After every use, sealant remains in each part of the delivery system; the pump, sealant applicator, pipes, and hoses. Because of the impracticality of cleaning sealant from all the parts, the sealant remaining in the system is left to solidify after the operation finishes. Before the machine can be used again, the sealant delivery system must be heated to liquefy the sealant in the system. Among the parts requiring heating is the sealant hose that carries sealant from the melter to the sealant applicator. Typically, as with the LCSM, sealant hoses have an inner diameter of about 3.1 cm (1.2 in) and are made of synthetic rubber with a steel wire braid reinforcement. However, to allow for a tighter bend radius and greater overall flexibility, the OCCSM's sealant hose is made of convoluted Teflon with a stainless steel wire braid cover. The goal of the efforts discussed in this chapter focuses on the development of a heated sealant hose that will solve problems associated with current heated hose designs.

3.1 Current Hose Heating Methods

The current method used to heat a sealant hose works but is inconvenient, unreliable and time consuming. This method prescribes placing the hose in a metal enclosure on the melter, called the hot box, which also encloses a portion of the exhaust pipes from the melter's diesel burner; see Figure 3.1. While in the hot box, the hose is heated from the burner's exhaust as the burner melts sealant in the melter. Once the sealant in the hose liquefies, the hose can be removed from the hot box and is kept free

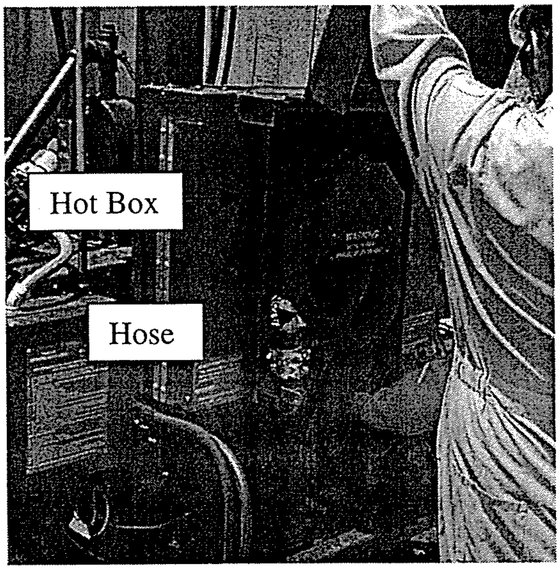

Figure 3.1 *Sealant Hose in the Hot Box*

flowing by maintaining a constant flow of hot sealant through it. While this method appears acceptable at first, a few problems appear once the method is used.

The first problem results from the standard procedure of heating up the melter before driving to the work site. This is done to minimize disruption of traffic, as well as to avoid the dangers of traffic while waiting over an hour for the melter to heat up. Then, while en route to the work site, the problem is caused by the sealant in the hose cooling and solidifying, due to relatively cool air from highway travel flowing around the hot box. When this occurs, time will be spent waiting at the work site for the hose to heat back up in the hot box. A second inconvenience of the current hose heating method is having to coil the hose and place it in the hot box any time the operation pauses for more than a couple of minutes, for example, while waiting for the melter to melt more sealant blocks. While heated hoses are currently available, they are unreliable and, because their heating element wraps around the hose, normal flexing will eventually cause their heater to fail.

Because of the current problems associated with heating the sealant hose, a more reliable method would be a significant improvement for crack sealing operations. However, the main motivation for a better hose heating system comes from the requirements of the OCCSM. The sealant hose on the OCCSM must connect the melter in the back of the truck with the sealant applicator at the end of the robotic arm. To connect these two points, the hose is routed between the frame rails under the truck bed, then into and through the center of the arm to reach the sealant applicator on the end. Under the truck bed, the hose goes through a mechanism, discussed in Chapter 6, which keeps tension on the hose and adjusts its length as the robotic arm extends and retracts. Because of the complicated routing of the hose, it is impractical to remove the hose every time it needs to be heated in the hot box, making a heated hose a necessity for the OCCSM.

3.2 Heat Source for the Hose

The heat source used in the sealant hose design had to meet several requirements based on the hose path and connections. First, it is imperative that the heater be flexible enough to bend with the hose, precluding the use of solid cartridge type electrical heaters. Second, in order to provide more effective heating, a goal of the design was to incorporate the heater inside the hose. To achieve this goal, the heater must fit in the hose while minimally affecting the sealant flow. Also, because connections inside the hose are unacceptable, the heater must be available as a continuous piece for the required length, and the overall design must allow for the heater connections to be made outside of the hose. Furthermore, the heater must be durable and withstand the sealant environment.

Meeting these requirements will result in a heated hose that is more rugged and reliable than those currently available.

Two options were available as a heat source, hot oil from the melter and electricity provided by a generator, both with the ability to heat the entire length of hose while in place. An electric heating element was chosen for its advantages and because initially, the hose connections prevented the use of oil. An electric element is smaller than a suitable oil line and provides uniform heating along the length of the hose, unlike the oil option. Also, the use of oil heating runs the risk of contaminating the sealant if a hole develops in the oil line, a problem that does not present itself with electric heating. The heating element chosen for the hose design meets the stated requirements and provides 39 W/m (12 W/ft) at 120 Volts. The next step in development was to test the heater alone to develop an understanding of its operational temperature range.

3.3 Preliminary Heater Tests

Before any testing was done to characterize the heater, a method of accurately measuring and recording temperature was needed. A few experiments were performed to determine the type of thermocouple and the method of attaching it to the heater that produced the most reliable results. The best combination was found to be a thin, surface mount thermocouple, tightly fastened to the heater using electrical tape. To obtain the temperature reading the thermocouple was connected to either a hand held digital thermometer or LabVIEW data acquisition hardware.

For the first tests, designed to explore the heater's response, the heater was powered at its rated voltage of 120 V. To provide some insulation, the heater was wrapped in an insulating cloth jacket, similar to the jacket that would be used in the

OCCSM to protect the hose and wires that run out to the sealant applicator alongside the hose. The first test with this setup showed that the heater had a slow response and barely reached 149°C (300°F), the sealant's melting temperature. To see if a better response could be attained, the decision was made to increase the voltage supplied to the heater. The graph in Figure 3.2 shows the response differences for varying voltage levels. Raising the voltage to 150 V provided a significant increase in both response time and steady state temperature. The next 30 V increase, to 180 V, produced a similar but less dramatic increase in response time and would have resulted in a higher steady state temperature if the heater's maximum temperature of 260°C (500°F) had not been reached. The sudden decline in the 180 V temperature curve after 570 seconds, resulted from the heater failing due to overheating. When the heater's maximum temperature was

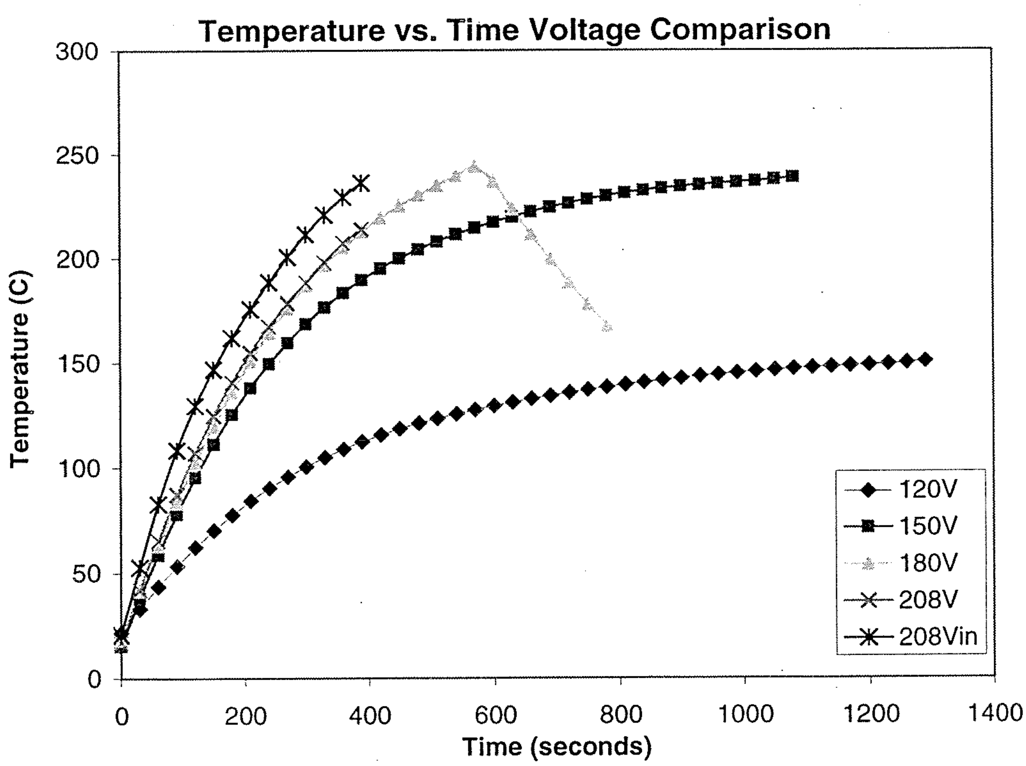

Figure 3.2 Response of Heater at Different Voltage levels

exceeded, the insulation on the power wires inside the heater melted, causing a short circuit. While the graph does not show that the maximum temperature was ever reached, an exploration into the thermocouple location and other tests reveal that this was indeed the case.

The thermocouple used in obtaining the temperature data was located on the outer sheath of the heater. Logically, the temperature would be greater inside the heater, directly next to the resistive heating wires. In preparation for the next test, a section of the outer sheath of the heater was temporarily removed so a thermocouple could be placed on the inner sheath directly beside the resistive heating wires. This provided measurements of the temperature inside the heater. For comparison, a second thermocouple was positioned outside the outer sheath, where the data for all the previous tests had been taken, see Figure 3.3.

For the test using the two thermocouples, the heater voltage was set at 208 V. The curves in Figure 3.2 labeled 208Vin and 208V represent the data taken by the thermocouple located in the heater and the thermocouple located outside the heater, respectively. During this test, the temperature of each thermocouple was monitored

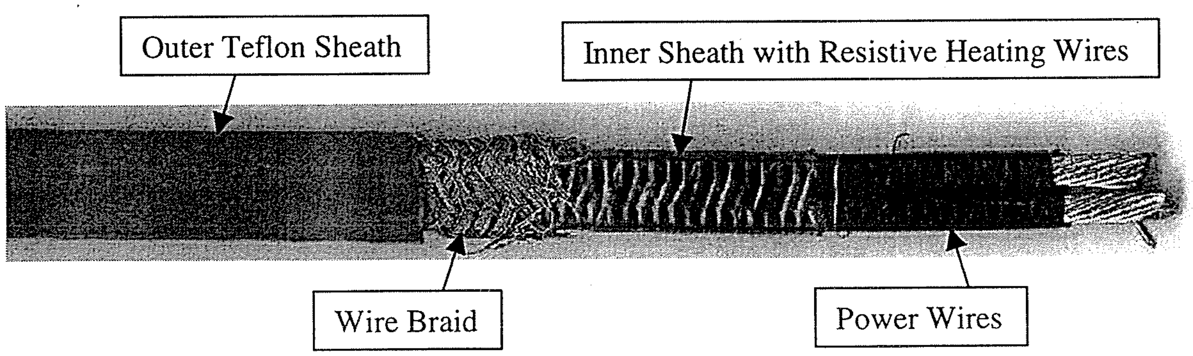

Figure 3.3 *View Showing the Heater's Layers*

closely and power to the heater was turned off when the thermocouple inside the heater reached 232°C (450°F). As the graph shows, the temperature difference between the two thermocouples used in this test, grew to nearly 28°C (50°F) before the power was shut off, which explains why the heater burned out in the previous test. However, little difference exists between the response of the thermocouple outside the heater in this test, and the response from the same thermocouple in the 180 V test, indicating that this last voltage increase has little significance.

To see if powering the heater above its rated voltage had any effect over time, the heater was run for a period of 8 hours at 208 V, using a temperature controller to ensure that the temperature in the heater never exceeded 232°C (450°F). A 180-degree, 1.9 cm (0.75 in) radius bend was put in the heater to further increase stress and help assess the heater's reliability. After this test, the heater was inspected and no sign of overheating or any other damage was found. From the results of the tests performed, it was concluded that the heater could safely be powered above its rated voltage provided that the temperature inside the heater was closely monitored. Also, a power level of 180 V allowed for adequate heater response, and further voltage increases would not be advantageous.

Even though heat transfer differences exist between the conditions tested here and a hose full of sealant, the consequences of a heater failing while in use made it necessary to test the heater as thoroughly as possible before adding sealant. For example, if a heater in a hose full of sealant fails, the option of putting a new heater in that hose is not worth considering. This is primarily due to the complication involved in removing the failed heater and putting another one inside the hose, while keeping the sealant liquid at over

149°C (300°F). If a heater did burn out, the hose could possibly be used with the hot box, but a new one, at the cost of $1000, would need to be purchased for use with a new heater.

3.4 Temperature Control Issues

With the issue of powering the heater solved, the next step in development was to put a heater in an empty hose and proceed with testing. While the majority of the heater resides inside the hose, in order to make the necessary electrical connections, the ends fit through a hole drilled in the side of the hose end fitting; see Figure 3.4. To provide more heat to the sealant, the heater runs the length of the hose twice, from one end to the other and back. With this design, less time is required to heat up the hose, and both ends of the heater are at the same end of the hose, protruding through a single hole in the hose fitting. Figure 3.4 shows the arrangement with both ends of the heating element side by side at their exit from the hose end fitting.

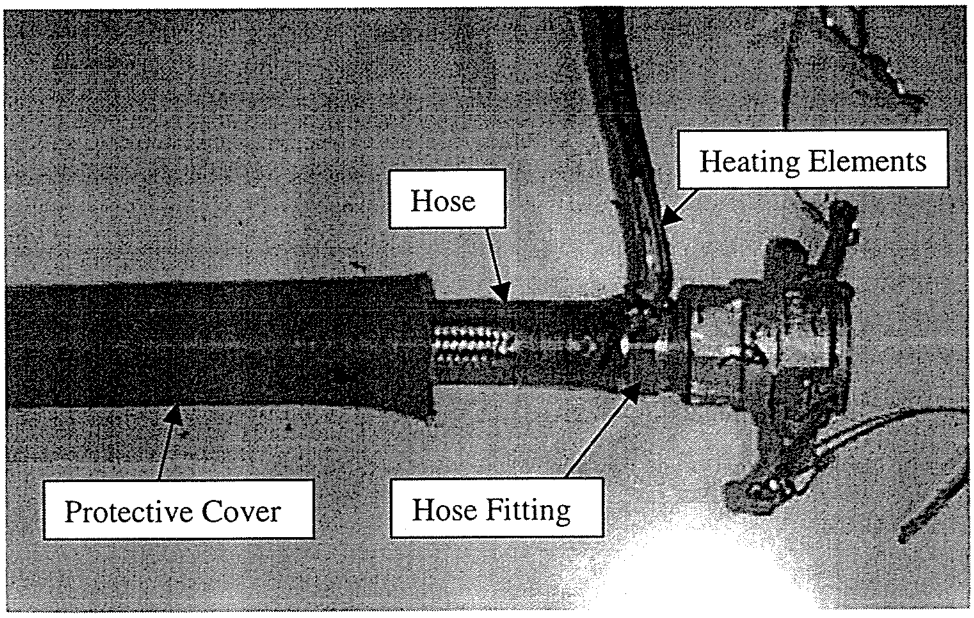

Figure 3.4 *Hose End With Heating Element*

Next, it was then necessary to determine the best location for the thermocouple used to control the heater's power. If the thermocouple could be in the section of heater that is outside of the hose, it would take less effort to reseal the heater's outer sheath after inserting the thermocouple. If the thermocouple needed to reside in the hose, a method of resealing the heater would have to be developed. To determine the importance of the thermocouple's location, a test was performed using two thermocouples, one located in the section of heater outside of the hose and the other in the section inside the hose. The heater was powered until the thermocouple inside the hose read 154°C (310°F), at which point the thermocouple outside the hose read only 110°C (230°F), indicating the importance of placing the thermocouple inside the hose. When the hose is filled with sealant, a more extreme temperature difference is expected between these two locations, since the sealant is a poor heat conductor. While locating the thermocouple inside the hose requires resealing the outer sheath of the heater, it does provide for more accurate monitoring of the sealant's temperature in the hose. Setting the temperature controller below 204°C (400°F) guarantees that the sealant in the hose will never exceed its maximum temperature.

After a few methods were considered to reseal the heater sheath, the most feasible one proved to be using heat shrink Teflon tubing. Because the outer sheath of the heater is made of Teflon as well, the temperature limits of the heater and its compatibility with sealant are unchanged. Since the tubing must reach about 315°C (600°F) before it shrinks, the thermocouple reading was monitored to ensure that the heater was not damaged while a heat gun was used to shrink the tubing.

The uniformity of temperature along the length of the heater was another issue of concern. Considering the fact that the heater runs the length of the hose twice, a hot spot would be created if the two sections were in contact anywhere along their length. This area of contact would likely be significantly hotter than the section that contains the thermocouple, and could cause the heater to overheat. To alleviate this potential problem, the heater section with the thermocouple is bound tightly to the adjacent section returning from the far end of the hose. This construction ensures that the thermocouple will control the hottest section of the heater, further reducing the chances of damaging the heater. This design held up well when tested with a full length heater in an empty hose, indicating that satisfactory control of the heater had been achieved.

3.5 Heated Hose Tests and Results

With the necessary precautions taken to prevent the heater from burning out, the hose was filled with sealant in preparation for tests that would determine the time necessary for the heater to melt the sealant in the hose. For each of the three tests, 175 V was supplied to the heater. The temperature controller was set at 149°C (300°F), approximately the melting temperature of sealant, for the first test. The temperature was increased to 177°C (350°F) for the second test, and to 190°C (375°F), just 14°C (25°F) below the sealant's maximum recommended temperature, for the final test. Table 3.1 summarizes the time necessary to liquefy the sealant to a state in which it oozed out of the hose. Also shown, is the time when the temperature controller began cycling, or equivalently, how long until the area of heater with the thermocouple reached the specified control temperature.

Control Temperature	Time until Liquid (min)	Time to Cycle (min)
149°C (300°F)	35	8
177°C (350°F)	25	10
191°C (375°F)	20	11

Table 3.1 *Heater Test Results*

These tests demonstrate that a sealant hose can be heated very effectively using an electric heater. While operating at 230°C (375°F), well below the heater's maximum temperature of 260°C (500°F), the hose was ready for use in a mere 20 minutes. Furthermore, at each of the tested control temperatures, the time taken to liquefy the sealant lies within an acceptable range, especially when considering that the melter takes at least an hour to heat up. None of the tests showed any problems with the heater burning out due to overheating or being powered above its rated voltage. Along with heating the hose in a reasonable amount of time, the design satisfies all the other requirements imposed by the OCCSM. This design will also alleviate the sealant hose problems associated with other crack sealing operations.

3.6 Recommendations

Further testing of this hose design would help make the system more efficient and reliable. An optimum trade off could be found between voltage level and control temperature, based on the allowable time for the hose to heat up. Also, if the hose does not need to heat at a rate faster than the melter, then it may be possible to use a single length of heater through the hose. However, if it is necessary to have two lengths of heater through the hose, the use of two separate heaters should be considered. In the event that one heater becomes damaged, the remaining heater could aid in replacing the damaged heater. Another option worth considering is to use a custom built heater designed for use at a higher voltage to provide more power and containing an internal thermocouple. All of these options would be simple modifications of the successful design developed here.

CHAPTER 4: THE LONGITUDINAL CRACK SEALING MACHINE

The Longitudinal Crack Sealing Machine (LCSM) is designed solely for use on straight cracks running along the roadway. No special guiding system is needed because of the crack's linearity. The sealant applicator can be positioned by driving the vehicle parallel to the crack. As shown in Figure 4.1, the LCSM consists of a truck and a sealant melter on a trailer. The sealant material is fed from the back of the truck into the melter, where it is melted, then through a hose to the sealant applicator on the front of the truck. In the previous version of the LCSM, the sealant applicator attached to the side of the truck near the front wheel. While this system worked well, it had one inherent disadvantage, cracks could only be sealed on one side of the truck. In order to seal a crack between the right traffic lane and the shoulder, the whole sealing operation would have to work against the flow of traffic, which is not the ideal way for work crews to operate. This brings up the design requirements for the new LCSM.

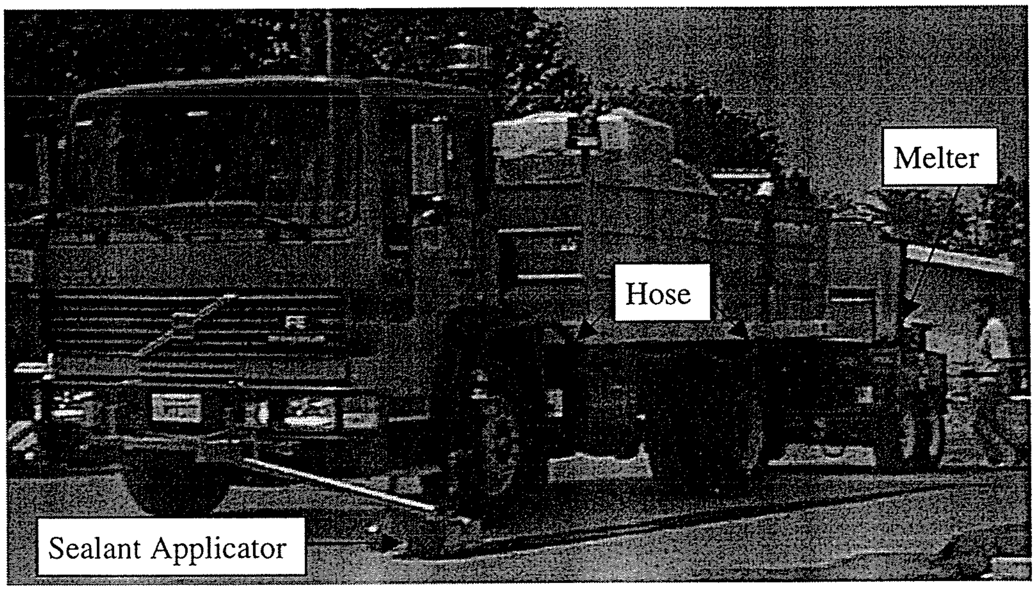

Figure 4.1 *The Longitudinal Crack Sealing Machine*

4.1 Design Requirements

The main objective for the new LCSM system is for it to operate on both sides of the vehicle. A dual steer truck was obtained for use in the project so the operator could drive and see the sealant applicator when sealing a crack on either side of the truck. Because crack sealing operations typically work on only one side of the road during the day, the LCSM is designed to be set up in the yard at the beginning of the day to work on a given side of the truck. For minimum set up and to provide the operator a good view of the sealant applicator, the new mechanism attaches to the front of the truck. When sealing, the operating position is about 0.61 m (2 ft) to the side of the truck and approximately in line with the front bumper.

Two motions are necessary to move the sealant applicator to the specified working position. The first moves the sealant applicator out to the side of the truck. This is necessary because while driving from the yard to the work site, all components are required to fit within the width of the truck. The second motion lowers the sealant applicator to the road surface. Whatever mechanism is designed to perform these two motions cannot obstruct the lights on the front of the truck or the ability to tilt the cab in order to access the engine compartment.

4.2 Mechanical Configuration

Two options were considered to perform the side to side motion. The first was a rack and pinion arrangement. While this would easily provide the necessary motion, there were concerns that the mechanism would collect dirt and require constant maintenance. For this reason the second configuration, a 4-bar linkage, was chosen. As shown in Figure 4.2, the base link of the 4-bar attaches to two pins in the front bumper of

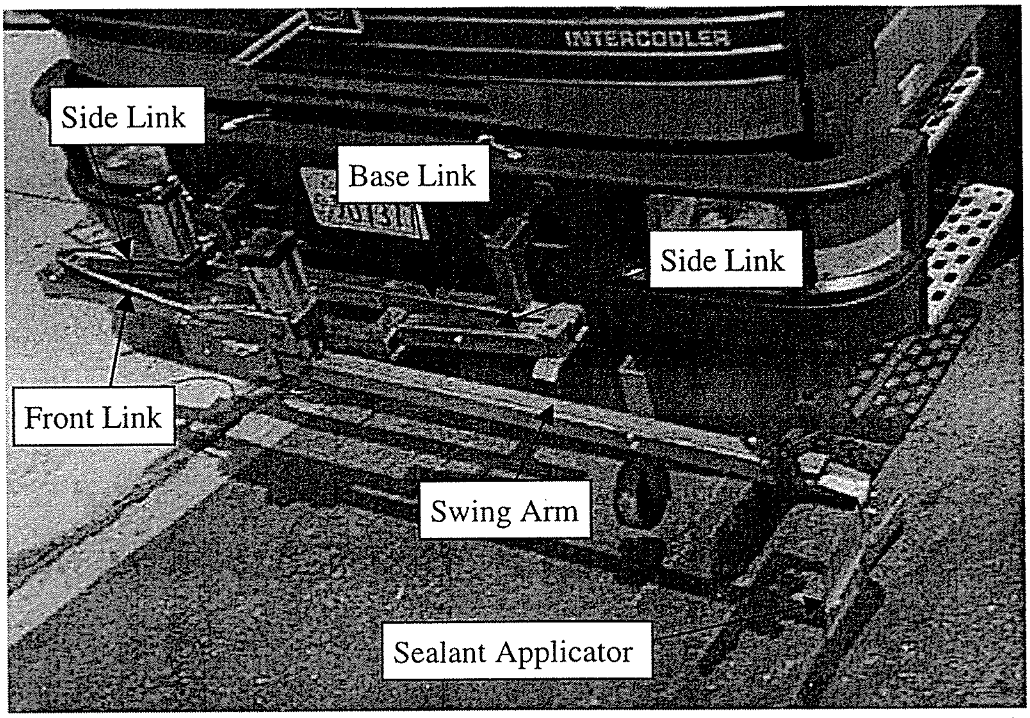

Figure 4.2 *4-Bar Mechanism in Stowed Configuration*

the truck. Another bar, the swing arm, attaches to the midpoint of the front link and has the sealant applicator attached to its other end. The ability of the swing arm to rotate 180 degrees allows the sealant applicator to be positioned on either side of the truck. This orientation is set up in the yard before leaving for the work site. Figure 4.2 shows the mechanism set up for use on the left side of the truck and in the stowed configuration. Simply by moving the 4-bar mechanism, the sealant applicator is moved to the side of the truck and is ready to be lowered to the road surface; see Figure 4.3. The swing arm provides the motion to lower the sealant applicator to the pavement, as shown in Figure 4.4.

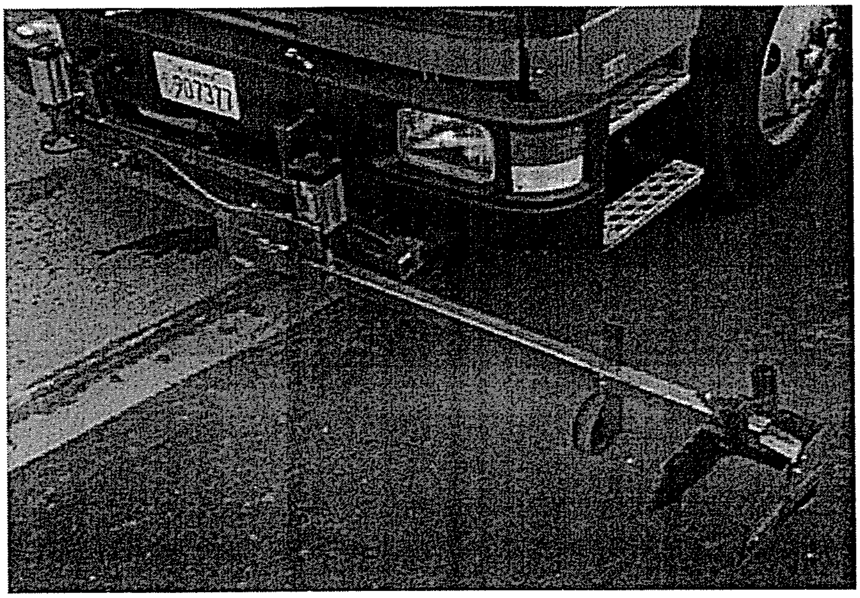

Figure 4.3 *Mechanism in Extended Configuration*

Figure 4.4 *Mechanism in Use on the Left Side of the Truck*

4.2.1 Motion Actuators

The two options available to power the motions described above were pneumatic and electric. Pneumatic actuators were chosen because they do not require position switches and can constantly hold the mechanism in a given configuration. A cylinder was the most reasonable choice to raise and lower the swing arm. The stroke of the cylinder was determined from the distance it needed to lower the sealant applicator to the road. While this distance is 38.1 cm (15 in), in order to accommodate dips in the road, an extra 15 cm (6 in) of travel was provided. From the geometry of the swing arm, the required cylinder stroke was found to be slightly less than 10 cm (4 in). A 8.26 cm (3.25 in) bore was chosen based on the minimum of 621 kPa (90 psi) available, geometry of the swing arm, and the weight of the previous version of the LCSM sealant applicator. This ended up being oversized because the new design is significantly lighter than the earlier design.

For the motion of the 4-bar linkage, both cylinders and rotary actuators were considered. The chosen actuator needs the ability to hold the linkage in the stowed and working positions to avoid using more components for this task. This is where the cylinder had its main disadvantage. When in either of these two positions, a cylinder would provide very little moment to hold the linkage in place due to the way it would be mounted. For both positions, the cylinder force would act almost in line with the link it was holding in place, resulting in a very short moment arm, as can be seen in Figure 4.5. The rotary actuator was chosen because it provides a constant torque throughout its motion and was simpler to mount. No method was readily available to calculate the torque required to rotate the 4-bar linkage, so the actuator size was based on rough estimates. The selected rotary actuator provided 33.9 N-m (300 in-lb) at 689 kPa (100

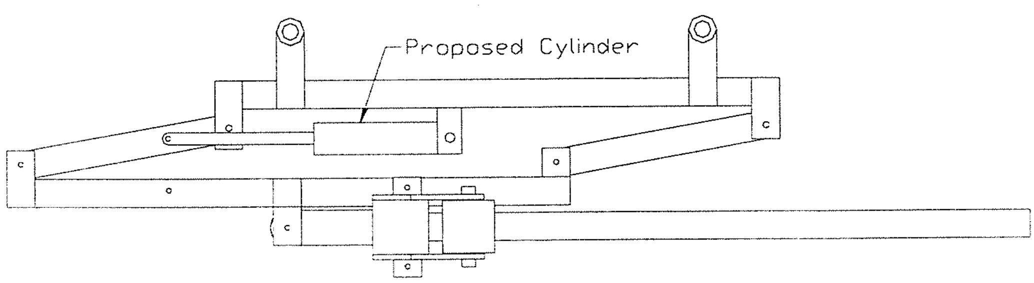

Figure 4.5 *Design with Proposed Pneumatic Cylinder*

psi). The system was designed to be used with only one rotary actuator, but a second could easily be added if necessary.

4.2.2 Materials and Detailed Design

Material selection was based on cost, simplicity of design, and ease of assembly. Square steel tubing was chosen for each of the bars because it best satisfied the above criteria. All of the plates and tabs for the rest of the assembly structure were constructed with hot rolled steel plate or bar. For corrosion resistance, an electroless nickel plating was applied to each of the steel parts. The remaining pins and connectors were either stainless steel or another non-corrosive material.

The length of each of the links is determined from the truck's width and the requirement that the sealant applicator needs to extend 61 cm (24 in) beyond the side of the truck. The side links need to be approximately 30.5 cm (12 in) long to provide the necessary motion to the side of the truck. However, these links were made slightly longer, 36.8 cm (14.5 in), because two factors reduce the mechanisms extension capability. First, to avoid going through a singular configuration, the 4-bar mechanism does not quite rotate a full 180 degrees. Second, because of the rotational motion of the swing arm, as the sealant applicator is lowered to the ground, it also moves back towards the truck. When the system is in the stowed position, the swing arm should extend from the midpoint of the front link to the side of the bumper. This distance is 145 cm (57 in).

For simplicity, each link was fabricated from the same size tubing. The stress analysis below shows that a 5.1 x 5.1 x .318 cm wall (2 x 2 x .125 in wall) tube is a reasonable size. For the load case shown in Figure 4.6, the maximum bending stress occurs at point A. For the tube size chosen and the estimated 445 N (100 lb) load, this stress is 15.51 MPa (2250 psi). Note that even if all of the weight were to rest on only one of the tubes, the stress would still be well below the material's yield stress, which is 250 MPa (36 ksi).

The bending stress in a tube is also critical at the location where the cylinder attaches to the swing arm. To avoid making any extra parts, this connection was created by removing material from the swing arm allowing the ball joint on the end of the cylinder to fit inside the tube. The compact design provided the swing arm, and hence the sealant applicator, more clearance from the roadway. However, these advantages came at the cost of weakening the swing arm at the position subject to the greatest bending stress. For the load case shown in Figure 4.7, the maximum bending stress is 42.2 MPa (6150 psi). This provides a factor of safety of 5.9, and again, the chosen tube size is acceptable.

The pin joints are other areas in the design in which the stresses may be critical. The main concern here is shear stress in the 1.27 cm (0.5 in) diameter pins. Figure 4.8 shows the loads on the pin joint between the base link and the side link. The two horizontal forces shown are reactions to the moment imposed by the 445 N (100 lb) load.

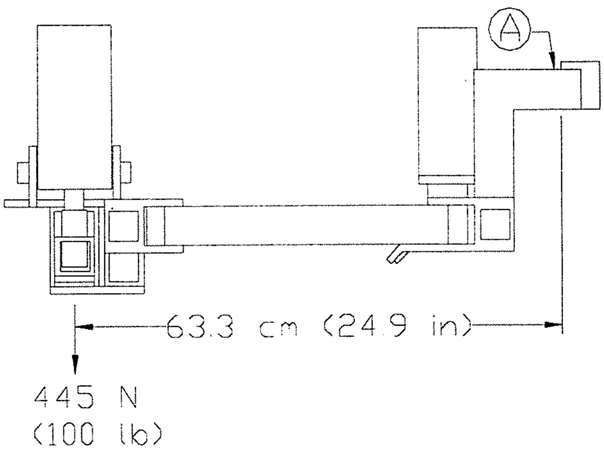

Figure 4.6 *Side View of Linkage Assembly*

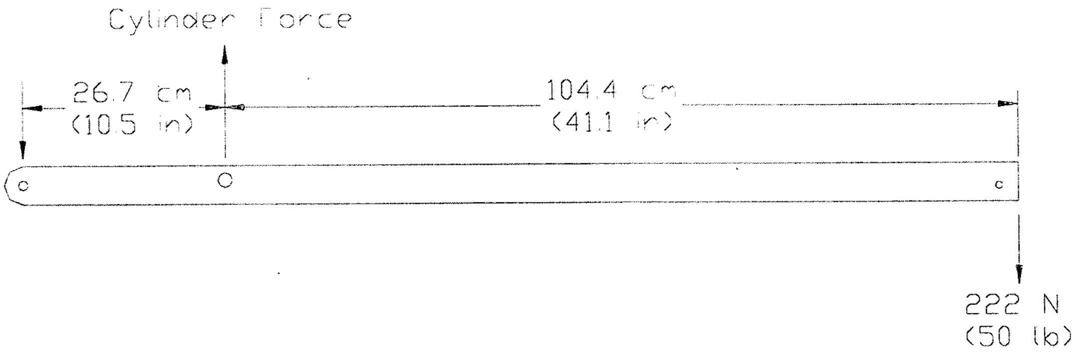

Figure 4.7 *Load Case on the Swing Arm*

Assuming that the load is evenly distributed between the two side links, the average shear stress in the pin is 17.3 MPa (2510 psi). Note that the stress level would be acceptable even if the entire load was supported by a single side link. Also, during normal operation the linkage is only in the position shown in Figure 4.8 when it is midway through the deploying or retracting motions. Most of the time the linkage is in the more compact deployed or retracted positions, which puts even less stress on the pin joints.

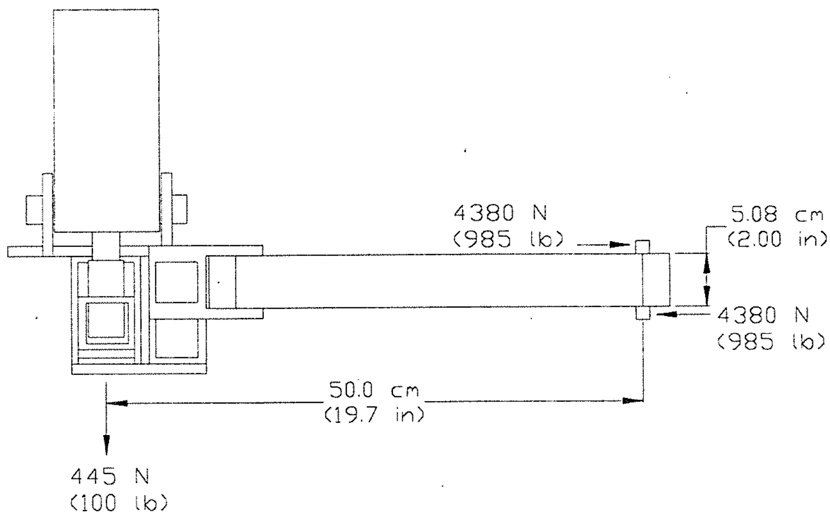

Figure 4.8 *Load on Pin Joint*

The pin joint connecting the swing arm to the front link was also an area of concern, primarily because the swing arm is relatively long. To reduce shear stress in this pin, the joint height was increased to 10.16 cm (4 in.). With the arrangement and the estimated load of 222 N (50 lb), the average shear stress is 23.7 MPa (3440 psi). This static analysis does not consider the effect of dynamic loads encountered while driving the vehicle down the road. However, the tab that constrains the swing arm's 180 degree rotation also rests on the front link. This tab provides support at all times except when the swing arm is being repositioned for use on a different side of the truck. Therefore, any dynamic load is distributed between the pin joint and the tab.

4.3 Testing and Recommendations

Testing the LCSM proved that the machine was successful, but some modifications are necessary. The problem encountered during testing resulted from too much weight resting on the sealant applicator. The drag caused by this weight resulted in an unstable, jerky motion of the mechanism when sealing a crack. For immediate solution to this problem, a wheel was added to take weight off the sealant applicator. However, a better solution needs to be identified. One possibility is to use the cylinder, which raises and lowers the swing arm, to support some of the weight. This could be achieved by just reducing the pressure, instead of bleeding off all of the pressure, in the cylinder when lowering the swing arm. The amount of pressure left in the cylinder would be adjusted to provide adequate support. A second option would be to design a new mechanism that would lower the sealant applicator but would not put extra weight on the sealant applicator.

Testing also revealed that an adjustment needs to be incorporated into the mechanism to adjust the distance to the side of the truck that the sealant applicator extends when in the working position. This is necessary because not all roadway lanes are the same width and the sealing operations must be able to work in narrow lanes without having to close adjacent lanes. Simply making the length of the swing arm adjustable can solve this problem.

One other aspect of this mechanism that can be improved is the weight of each of the components. Because simplicity of design and assembly was a primary goal, the current design is heavier than necessary. Removing material from several areas as well as changing the primary material to aluminum could effectively reduce weight.

A longitudinal crack sealing machine has been developed which has the ability to increase the speed and safety of current crack sealing operations. The design meets all the requirements, is simple and the machine produces a high quality seal as shown in Figure 4.9. With the modifications suggested above, the Longitudinal Crack Sealing Machine can be a useful machine for all longitudinal crack sealing operations.

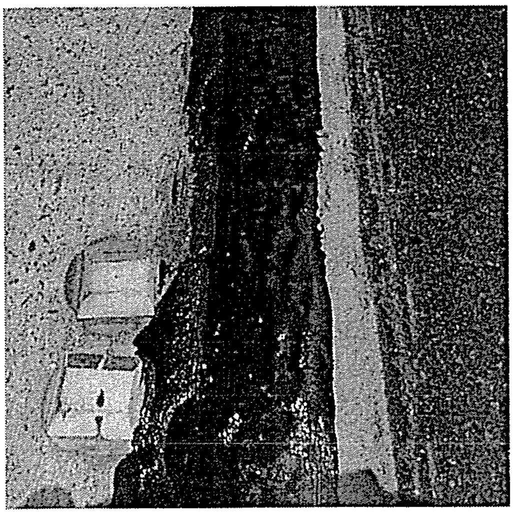

Figure 4.9 *Finished Longitudinal Crack Seal*

CHAPTER 5: THE OCCSM CAMERA BOOM

As mentioned in the introductory chapter, the OCCSM uses a camera centered over the robotic arm's workspace to obtain an image of the roadway. The camera is mounted on the end of a boom that extends behind the truck and supplies pictures to the operator in the truck's cab. During operation, the camera boom positions the camera over the center of the workspace, but retracts once work is completed. A housing encloses the camera at the end of the boom and provides adjustments to align the camera with the workspace coordinates. This chapter focuses on the design of the camera boom, its retraction mechanism, and the camera housing. Vibration analysis of the camera boom and braces designed to minimize camera movement are also discussed.

5.1 Camera Boom Design Requirements

The main purpose of the camera boom is to position the camera over the center of the workspace. Added to this purpose are several constraints and requirements that guide the boom's design. The frame members supporting the canopy over the truck bed also serve as mounts for the brackets that hold the camera boom. While the boom rests in the brackets below the canopy frame, it must not interfere with overhead clearance for workers on the truck bed. Furthermore, to allow the truck to enter garages or shops, the top of the canopy is restricted to a maximum of 3.6 m (12 ft) off the ground, leaving 2.3 m (7.5 ft) between the floor of the truck bed and the bottom of the canopy frame members. Mounting the boom just below the canopy places the camera high enough above the roadway to capture an image of the entire 366 x 366 cm (144 x 144 in) workspace.

The boom extension and retraction mechanism must be capable of moving the boom from its stowed position under the canopy, to the position that locates the camera in the center of the workspace, a distance of 2.3 m (7.5 ft). Because the boom moves between the two positions relatively infrequently, only at the start and completion of a job, a manually operated mechanism is suitable for the task. However, a single worker should be able to operate the mechanism while remaining in the truck bed.

A method of securing the camera boom in both the extended and retracted positions is necessary. Because the camera must line up accurately with the robotic arm's coordinates, the boom's position securing system must eliminate all movement as well as provide repeatability for the extended configuration. Camera motion due to vibration must also be minimized to obtain as clear an image as possible.

5.2 Camera Boom Supports

The camera boom receives support from the two rear canopy frame members that are 140 cm (54 in) apart. A bracket made from three pieces of steel angle and one steel bar welded into a box shape, hangs from each canopy frame member; see Figure 5.1. Delrin plastic fastened to the inner surfaces of the box acts as a bearing material between the boom and the bracket. A slot in one side of the plastic allows the chain, a component of the extension mechanism described in Section 5.3, to pass through the brackets as the boom extends and retracts.

To constrain the boom in the extended and retracted positions, bolts thread through the brackets and into the boom. To maximize the effectiveness of the bolts, the one on the rear support pulls the boom down against the bottom of the support, while the

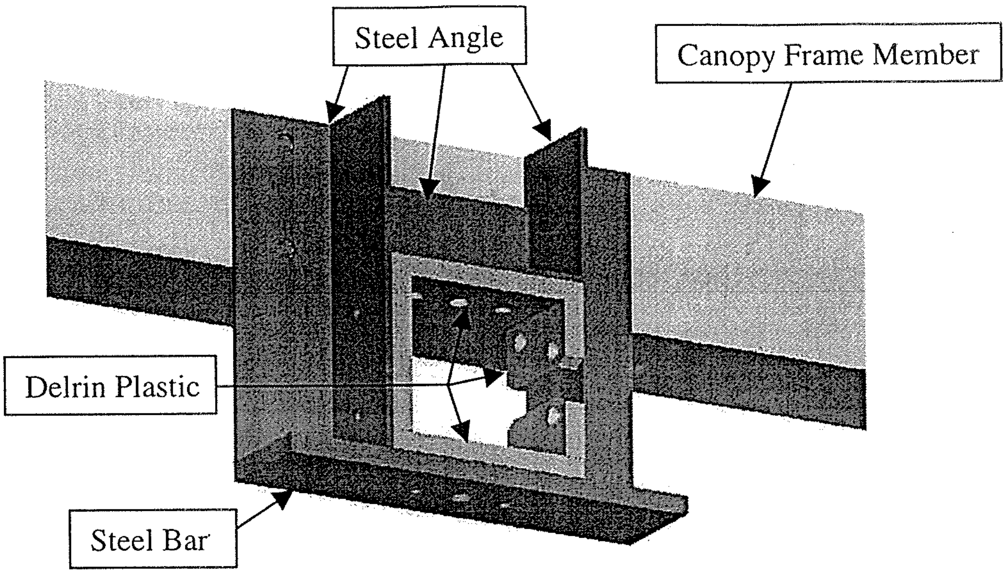

Figure 5.1 *Model of Camera Boom Bracket*

front one pushes the boom against the top of its support. Applying the bolt forces in this manner augments the loads naturally acting on the supports due to the boom's weight when in the extended position. To make the bolts easy to install and remove by hand, handles were welded onto the bolts, forming a "T" out of the assembly.

5.3 Extension Mechanism

The extension and retraction mechanism must extend the boom out to the center of the workspace, 3.2 m (10.4 ft) behind the truck bed. Two options were considered to produce this motion: a pulley system and a rack and pinion mechanism. The rack and pinion mechanism was chosen because it required fewer parts and was simpler to integrate with the boom supports. The rack portion of the mechanism attaches to the side of the boom, while the pinion remains fixed and mounts to one of the boom brackets. A chain stretched alongside the boom is used as the rack and this approach keeps the design lightweight while providing ample strength. The pinion gear, simply a chain sprocket, attaches to a shaft with a handle that will be turned by the operator to move the boom.

The sizes of the mechanism's components were determined by a compromise between the force the operator needs to apply to the handle and the number of revolutions made by the pinion. Equation 5.1 relates the pinion's pitch diameter, d, to the number of pinion revolutions, t, necessary to extend the boom a distance l.

$$d = \frac{l}{\pi \cdot t} \tag{5.1}$$

The boom needs to extend about 230 cm (90 in) to reach its working position, and should do so with a reasonable number of revolutions of the pinion gear. To achieve the required extension with 10 revolutions of the pinion gear requires a pitch diameter of just under 7.4 cm (2.9 in). A suitable chain sprocket was found with a pitch diameter of 6.510 cm (2.563 in), requiring 11.2 revolutions to fully extend the boom.

Next, an estimation of the force required to extend the boom was used to determine the length of handle needed to turn the shaft. Equation 5.2 describes the effect of the handle length, h, and the pinion's pitch diameter on the force required by the operator, P. The friction force, F, opposing the boom's motion, is approximated as shown in equation 5.3, using a coefficient of friction, μ, and the reaction forces at the boom's supports, F_1 and F_2.

$$P = \frac{F \cdot d}{2 \cdot h} \tag{5.2}$$

$$F = \mu \cdot (F_1 + F_2) \tag{5.3}$$

The reaction forces depend on the weight of the boom and its position relative to the supports. A simple analysis reveals that the reaction forces will be largest when the boom is fully extended. Using the aluminum beam, whose selection is detailed in Section 5.4, the sum of the two reaction forces totals about 490 N (110 lb). With a relatively large estimated friction coefficient of 0.35, the maximum friction force opposing the beam's motion is calculated from Equation 5.3 to be nearly 180 N (40 lb). Applying Equation 5.2 with a handle length of 20 cm (8 in) and the sprocket mentioned above, results in a force of less than 31 N (7 lb) being required by the operator. This force lies well within the range that the operator should be able to exert on the overhead crank mechanism. The above analysis shows that the chosen chain sprocket size and handle length are acceptable in the design.

The parts of the crank assembly, shown in Figure 5.2, are fastened together using traditional methods. The sprocket slides on the shaft and is held in place by two set-screws. The shaft rotates in, and receives support from, two bronze bushings located on either side of the sprocket. A collar keeps the shaft from sliding out of the bushings by transferring the weight of the assembly to the flanged, lower bushing. The shaft extends far enough below the boom so that the handle welded to the end of the shaft clears the boom bracket.. A handgrip screws into the end of the handle and may be folded up when not in use to provide more overhead clearance.

5.4 Selection of Boom Size and Material

The size of the boom and its material were key issues considered to minimize camera movement while keeping the design relatively lightweight. At full extension, the boom is essentially cantilevered 318 cm (125 in) off the back of the truck, with nearly an additional 180 cm (70 in) of the boom remaining between the boom support brackets. A square cross-section was chosen for the boom, as opposed to a circular cross-section, because it does not have to be constrained from rotating to keep the camera in alignment.

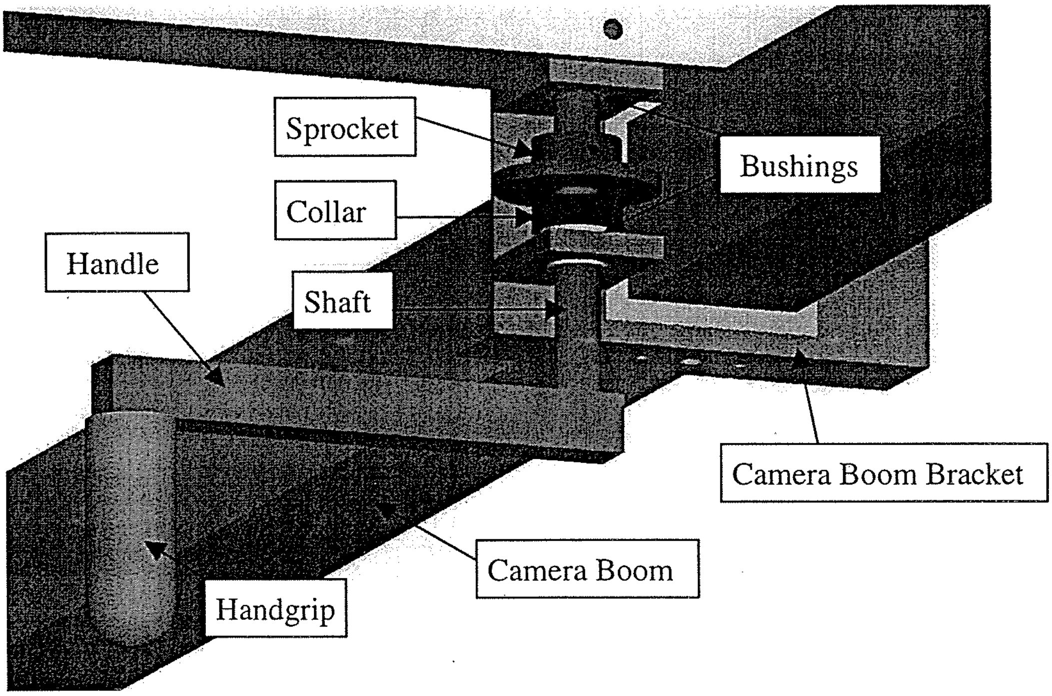

Figure 5.2 *Model of Camera Boom Crank Assembly*

To begin the analysis of different materials and wall thicknesses, a boom with a 7.6 cm (3 in) square cross-section was chosen as a starting point. Throughout the design this size was deemed the most reasonable. A larger boom would begin to cut into the height clearance in the truck bed. Using a smaller boom would only further exaggerate problems encountered with vibration. Three different materials were compared: steel, aluminum, and a fiberglass composite. Each material was analyzed with varying wall thicknesses, ranging from 0.16 cm (0.063 in) to 0.953 cm (0.375 in). Table 5.1 lists the specific weight and Modulus of Elasticity for each of the materials. Four characteristics were examined to compare different materials and wall thicknesses: weight, static end deflection, vibration natural frequency, and maximum forced vibration amplitude.

5.4.1 Weight Analysis

The total weight of each prospective boom was obtained by multiplying the boom's volume by the specific weights listed in Table 5.1. To find the boom's volume, the cross-sectional area for each wall thickness was found, then the area was multiplied by the boom's total length. The graph in Figure 5.3 shows the results of this analysis, and provides an easy comparison for each combination of material and wall thickness.

Figure 5.3 shows that the use of steel becomes undesirable as wall thickness increases. The weight of both aluminum and fiberglass composite increase slowly with increased wall thickness when compared with steel. Being the lightest, the fiberglass boom is the preferred choice based on weight analysis. However, aluminum comes in a close second and should remain a possibility pending the results of the other analyses.

Material	**Modulus of Elasticity**	**Specific Weight**
Steel	200 GPa $29*10^6$ psi	0.077 N/cm^3 0.284 lb_f/in^3
Aluminum	69 GPa $10*10^6$ psi	0.027 N/cm^3 0.098 lb_f/in^3
Fiberglass Composite	21.7 GPa $3.15*10^6$ psi	0.014 N/cm^3 0.053 lb_f/in^3

Note: All data was adapted from Hibbeler (1994); except for the Specific Weight of the Fiberglass obtained from the supplier's catalog and the Fiberglass Modulus of Elasticity, which was determined by experiment.

Table 5.1 *Properties of Possible Boom Materials*

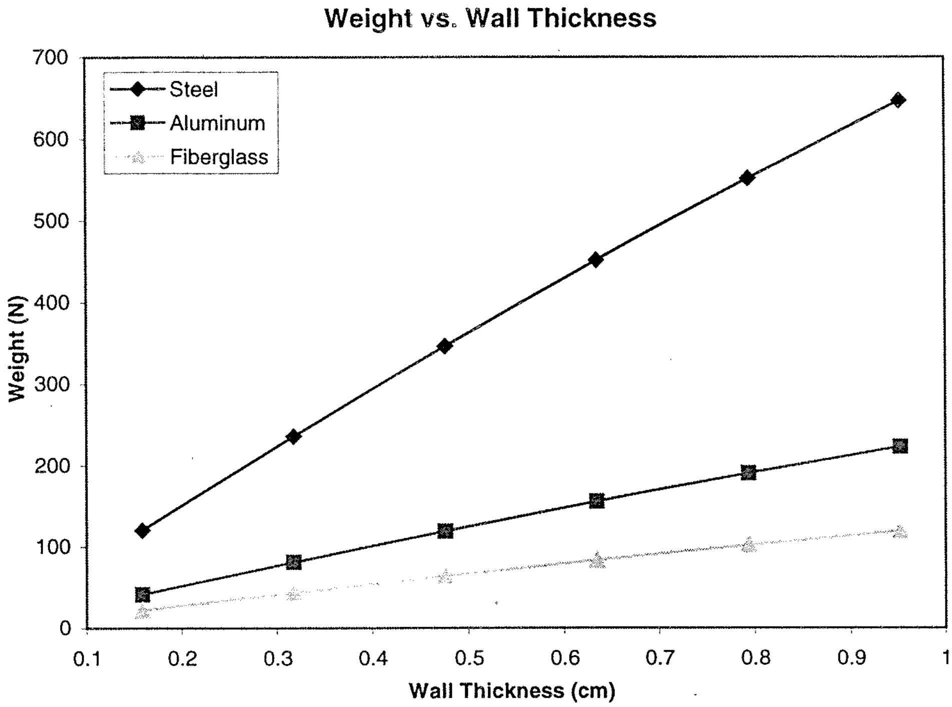

Figure 5.3 *Results of Boom Weight Analysis*

5.4.2 Static End Deflection Analysis

The static deflection of the boom was determined using Mechanics of Materials methods, specifically Equation 5.4. Figure 5.4 shows the free body diagram of the boom and the two coordinates required in formulating the deflection equations. Note that two deflection equations result from this analysis: one for the cantilevered section and the other for the section between the two supports. The force, F_c, on the end of the boom, includes the estimated weight of the camera and housing, about 22 N (5 lb), while the distributed load, *w*, is comprised of the boom's distributed weight and the distributed weight of the chain. Applying Equation 5.4, the moment expression, *M(x)*, was integrated twice to obtain the beam's deflection equation, *v(x)*.

$$\mathrm{E} \cdot \mathrm{I} \cdot \frac{\partial^2 v}{\partial x^2} = M(x) \tag{5.4}$$

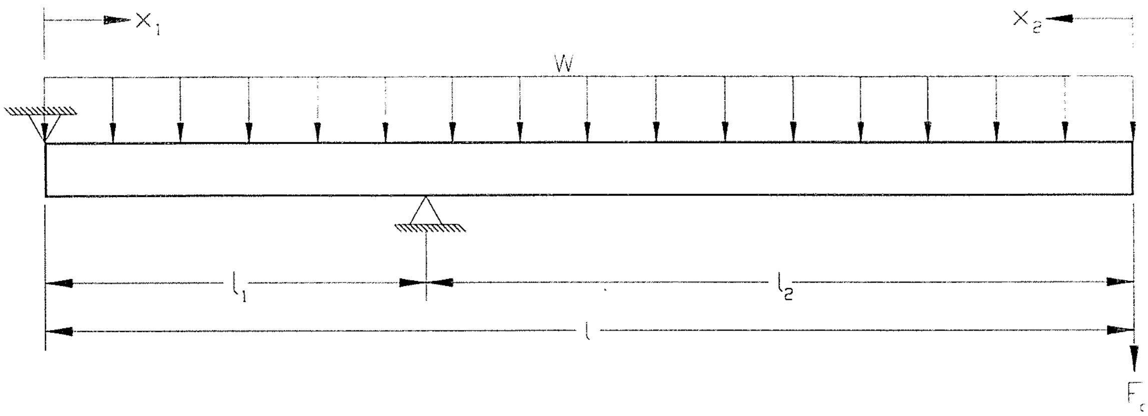

Figure 5.4 *Free Body Diagram of Camera Boom*

Boundary conditions were used to solve for the constants of integration, giving final expressions for the boom's deflected shape. From these expressions, a simplified equation for the deflection at the end of the beam can be written as

$$v_{end} = \frac{-1}{\mathrm{E} \cdot \mathrm{I}} \cdot \left(\frac{w}{24} \cdot l_2^3 \cdot l + \frac{w}{8} \cdot l_2^2 \cdot l^2 - \frac{w}{24} \cdot l_2 \cdot l^3 + \frac{F_c \cdot l_2^2 \cdot l}{3} \right). \qquad (5.5)$$

The end deflections for the various beam materials and cross-sections may be compared by examining Figure 5.5.

Figure 5.5 shows the inadequacy of the fiberglass boom for supporting weight with minimal deflection. Although the deflection of the fiberglass boom improves with increased wall thickness, it remains nearly double the values obtained using either steel or aluminum. The deflection of both steel and aluminum booms are relatively close, especially with larger wall thicknesses, showing that these materials are preferred based on the deflection analysis.

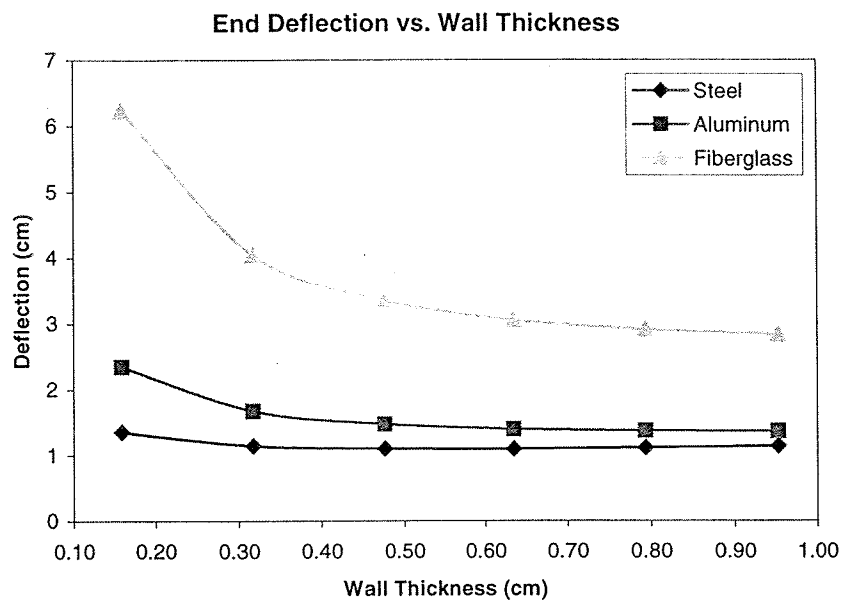

Figure 5.5 *Results of Camera Boom End Deflection Analysis*

5.4.3 Natural Frequency Analysis

To obtain an estimate of the boom's vibration natural frequency, Rayleigh's Method was employed. Using Rayleigh's Method, the familiar equation for natural frequency, shown in Equation 5.6, is used, but an equivalent mass, m_{eq}, must be found to approximate the continuous system as a point mass.

$$\omega_n = \sqrt{\frac{k}{m_{eq}}} \tag{5.6}$$

The equivalent mass is found by developing an expression for the maximum kinetic energy, T_{max}, in the system. The basic formulation of the solution is shown here, while the calculations for the analysis were performed in Matlab, using the program in Appendix A.

An essential assumption in finding the kinetic energy of the system is that the dynamic mode shape may be closely approximated by the static deflection shape, which was found in Section 5.4.2. Finding the maximum deflection, v_{max}, in each section of the boom allows the two equations for the static shape to be written in the form

$$v = \frac{v}{v_{\max}} \cdot v_{\max}. \tag{5.7}$$

Now, differentiating with respect to time gives an expression for the velocity profile, $\dot{v}$, as

$$\dot{v} = \frac{v}{v_{\max}} \cdot \dot{v}_{\max}. \tag{5.8}$$

Using this result and applying Equation 5.9, a form of the kinetic energy equation where V represents velocity and w_m represents mass per unit length, to each section of the boom leads to Equation 5.10 for the boom's total kinetic energy.

$$T = \frac{1}{2} \cdot \int_0^l \left(w_m \cdot V^2\right) dx \tag{5.9}$$

$$T_{\max} = \frac{w_m \cdot \dot{v}_{1,\max}^2}{2 \cdot v_{1,\max}^2} \int_0^{l_1} v_1^2 \, dx_1 + \frac{w_m \cdot \dot{v}_{2,\max}^2}{2 \cdot v_{2,\max}^2} \int_0^{l_2} v_2^2 \, dx_2 \tag{5.10}$$

In Equation 5.10, the numeric subscripts differentiate between the two sections of the boom and w_m represents the mass per unit length.

At this point, the only two unknowns on the right side of the equation are the two maximum velocities, $\dot{v}_{1,\max}$ and $\dot{v}_{2,\max}$. Before the equivalent mass can be found, one of these unknowns must be eliminated. Making the assumption that the ratio of the maximum velocities is equal to the ratio of maximum deflections, r, allows one of the unknowns to be eliminated, and we have

$$r = \frac{v_{1,\max}}{v_{2,\max}} = \frac{\dot{v}_{1,\max}}{\dot{v}_{2,\max}} \tag{5.11}$$

and

$$\dot{\upsilon}_{1,\max} = r \cdot \dot{\upsilon}_{2,\max}. \tag{5.12}$$

Using Equation 5.12, and letting I_1 and I_2 denote the numeric values of the integrals in Equation 5.10, the maximum kinetic energy of the boom becomes

$$T_{\max} = \frac{1}{2} \cdot \left(\frac{w_m \cdot I_1 \cdot r^2}{\upsilon_{1,\max}^2} + \frac{w_m \cdot I_2}{\upsilon_{2,\max}^2} \right) \cdot \dot{\upsilon}_{2,\max}^2 = \frac{1}{2} \cdot m_{eq} \cdot \dot{\upsilon}_{2,\max}^2. \tag{5.13}$$

Equation 5.13 shows how the expression for the boom's equivalent mass is easily found by comparison to the standard expression for kinetic energy, shown on the far right side of the equation. Adding the mass of the camera and housing, C_m, the equivalent mass of the entire system can be represented as

$$m_{eq} = \frac{w_m \cdot I_1 \cdot r^2}{\upsilon_{1,\max}^2} + \frac{w_m \cdot I_2}{\upsilon_{2,\max}^2} + C_m. \tag{5.14}$$

The final piece of information needed to calculate the vibration natural frequency is the boom's stiffness, k. Because stiffness is simply force divided by distance, finding the deflection resulting from a force applied on the unsupported end of the beam gives the necessary information. Using Equation 5.5 and letting w=0, gives the following expression for the boom's stiffness.

$$k = \frac{F}{\upsilon_{2,\max}} = \frac{3 \cdot \mathrm{E} \cdot \mathrm{I}}{l \cdot l_2^2} \tag{5.15}$$

Finally, Equation 5.6 may be employed to calculate estimates of the beam's vibration natural frequency. The natural frequency of the boom was calculated for the three materials and varying wall thicknesses. The graph in Figure 5.6 shows the results of the natural frequency analysis for comparison.

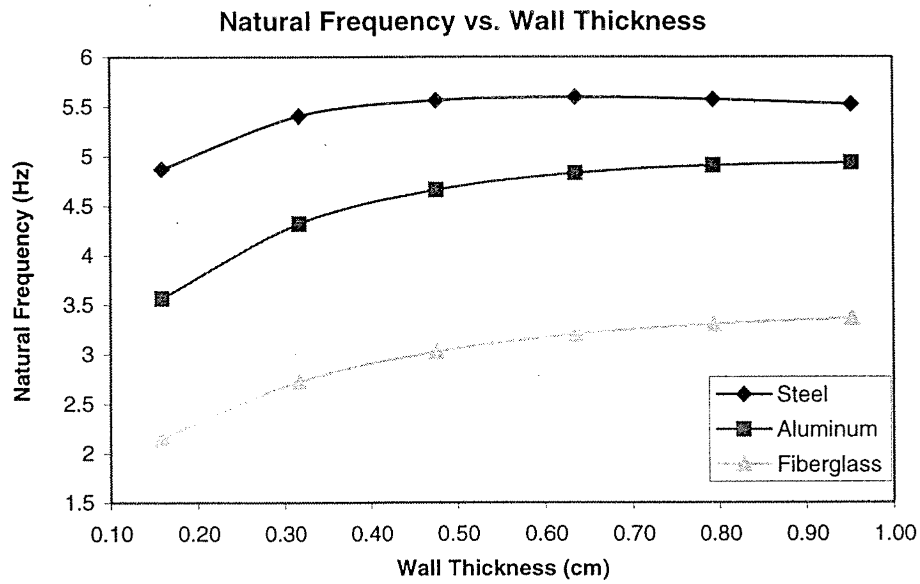

Figure 5.6 *Results of Camera Boom Natural Frequency Analysis*

The validity of the solution was checked against the natural frequencies of two purely cantilevered beams using formulas also derived by Rayleigh's method (see Rao, 1995). The first beam was as long as the portion of the camera boom that extends off the back of the truck, while the second had a length equal to the entire camera boom. As expected, the natural frequency of these two cantilevered beams bracket the result from the detailed analysis described above, indicating the validity of the solution.

The boom can not be picked to avoid having its natural frequency near the frequency of the input, because of the undefinable nature of the vibration input. Therefore, a boom with a higher natural frequency is desired because, in general, a higher frequency corresponds to smaller amplitude vibrations. As with the deflection analysis results, the fiberglass boom lacks the natural frequency characteristics desired. A steel camera boom is clearly preferred, having a natural frequency about 1 Hz higher than aluminum and 3 Hz higher than the fiberglass boom. However, this advantage will have to be weighed carefully against the disadvantage in a steel boom's weight.

5.4.4 Forced Vibration Analysis

The final analysis used to determine the best camera boom material and size was the study of a forced, damped vibration response. Because the actual vibration input to the system is very complex, the analysis uses a sinusoidal force input applied on the free end of the boom. For each combination of boom material and wall thickness, the applied force had a magnitude of 4.4 N (1 lb). However, the forcing frequency, ω_f, is different for each combination since it was chosen to produce the maximum amplitude response.

The choice of a damping value was based on experiment and a couple of assumptions. Observations made of the vibrations of a beam equivalent in size to the camera boom revealed that the vibration amplitude decreased approximately ninety percent in a time span of ten seconds. From this observation, the logarithmic decrement, δ, can be obtained from

$$\delta = \frac{1}{n} \cdot \ln\left(\frac{x_1}{x_{n+1}}\right) \tag{5.16}$$

where n represents the number of cycles observed and, x_1 and x_{n+1} are the amplitudes before and after n cycles, respectively. The number of cycles, n, occurring during the observation time, t, are calculated as

$$n = \frac{t \cdot \omega_n}{2 \cdot \pi} \cdot \sqrt{1-\xi^2}. \tag{5.17}$$

Assuming a relatively small damping ratio, ξ, allows the radical in Equation 5.17 to be approximated as unity. The damping constant, c, is then expressed as

$$c = \frac{m_{eq} \cdot \omega_n \cdot \delta}{\pi}. \tag{5.18}$$

The sources of damping in the camera boom system may be split into two categories: internal damping in the boom itself and damping resulting from the boom mounts and the environment. Assuming that different boom materials have similar damping properties, allows the use of one damping value for all combinations of materials and wall thicknesses. Specifically, the damping value obtained using the data and procedure described in the previous paragraph will be used throughout the forced vibration analysis of the boom. While the damping value remains constant, the damping ratio differs from one combination of boom material and wall thickness to the next. The damping ratio, ξ, is expressed as

$$\xi = \frac{c}{2 \cdot \sqrt{k \cdot m_{eq}}} \tag{5.19}$$

and will be used to calculate the forced amplitude response and the forcing frequency.

The final parameter necessary to calculate the forced amplitude response is the forcing frequency, ω_f. As mentioned earlier, the forcing frequency is chosen to produce the maximum response and differs for each combination of boom material and wall thickness. The forcing frequency that will produce the maximum amplitude response in terms of the undamped natural frequency and the damping ratio is expressed as

$$\omega_f = \omega_n \cdot \sqrt{1 - 2 \cdot \xi^2}. \tag{5.20}$$

The amplitude response, X, is expressed as

$$X = \frac{F_0}{k \cdot \sqrt{\left[1 - \left(\frac{\omega_f}{\omega_n}\right)^2\right]^2 + \left[2 \cdot \xi \cdot \frac{\omega_f}{\omega_n}\right]^2}} \tag{5.21}$$

and using the forcing frequencies found from Equation 5.20 will ensure that this is the maximum vibration amplitude.

The results of the analysis appear in Figure 5.7 and clearly show that the fiberglass boom has a vibration amplitude nearly twice that of a steel boom. The amplitudes of the steel and aluminum booms are much closer, with the aluminum boom amplitude an average of about 20% greater than steel. Similar to the natural frequency analysis, the order of preference places steel first, aluminum second, and fiberglass last. Again, steel only shows minor advantages over aluminum, emphasizing the weight consideration.

5.4.5 Review of Analysis Results

Reviewing the results of the four characteristics analyzed, a fiberglass boom seems a poor choice since it exhibits significant disadvantages in three of the four analyses. However, the choice between steel and aluminum is not readily apparent. While steel appears better in all characteristics except for weight, the values for

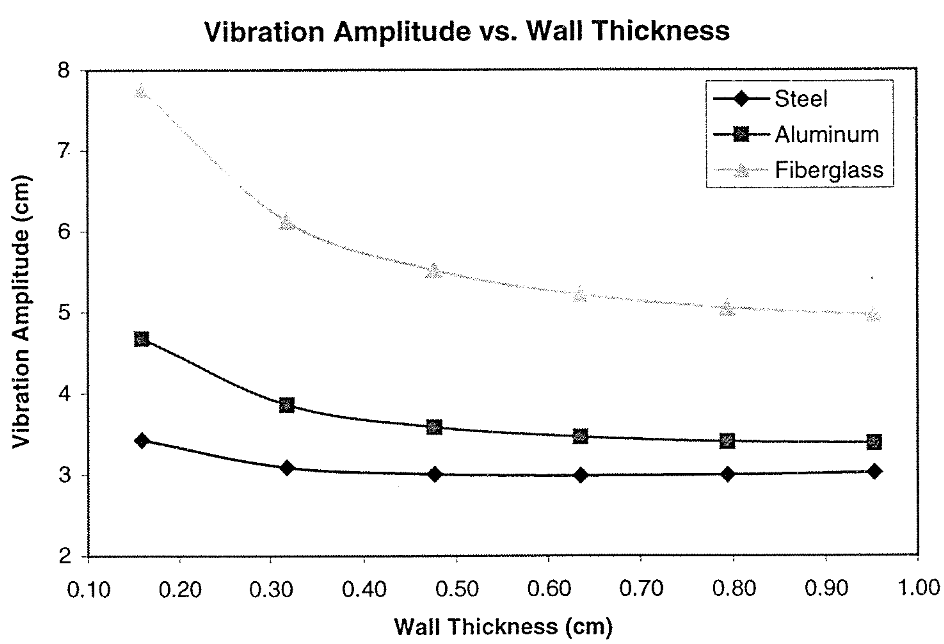

Figure 5.7 *Results of Maximum Vibration Amplitude Analysis*

aluminum are usually within 20% of those for steel. Further comparisons reveal the similar characteristics of the two specific possibilities for boom material and size: steel with a 0.16 cm (0.063 in) wall thickness and aluminum with a 0.478 cm (0.188 in) wall thickness. Both of these combinations possess very similar qualities in the four characteristics analyzed, and either is a suitable choice from that standpoint. However, in other areas an aluminum boom has two advantages over a steel one. First, since the OCCSM will be subjected to weather, an aluminum boom has the property that it does not require extra weather protection. Second, fastening parts to an aluminum boom will be easier since the thicker wall allows the direct use of threaded fasteners. For these reasons the aluminum boom with a 0.478 cm (0.188 in) wall thickness was selected.

The forced vibration analysis discussed in Section 5.4.4 suggests that the camera boom will likely experience too much vibration. In the analysis, the excitation force was applied in a manner to produce the largest response possible, on the very end of the boom and at the resonant frequency. Therefore, the results, which predict vibration amplitudes large enough to significantly reduce the quality of images taken by the camera, were not unexpected. However, even though the forcing function was applied to produce the worst case response, the magnitude of the force was only 4.4 N (1 lb). While the forcing function does not represent the real input, the boom will likely experience vibrations of this magnitude or greater at some time. Simple tests performed with the aluminum tube, acquired for use as the camera boom, supported the notion that vibrations would be above acceptable levels. A method of reducing the camera boom vibrations is needed to ensure the quality of images the camera supplies to the operator.

5.5 Camera Boom Braces

In addition to reducing camera movement to a reasonable level, the system used to reduce camera boom vibrations had to meet two other important requirements. First, the system must not require an additional worker besides the one who operates the camera boom crank mechanism. Second, any required assembly or adjustments should be accessible to the worker while remaining on the truck bed.

Two options were considered to provide the extra support to the camera boom. One option employed a cable attached to the end of the boom near the camera. The other end of the cable would attach to a removable post above the truck bed canopy, as shown in Figure 5.8. While this arrangement would reduce vibrations in the vertical direction, it would have minimal effect on horizontal vibrations. Also, the support post above the canopy would have to be removable or retractable in order to meet the overall truck height requirement. The extra complication involved in lowering the support post was not desirable, especially in the prototype machine, and can be avoided by using a simpler method to support the camera boom.

The second option that was considered to support the camera boom was chosen because it reduces vibration in all directions while remaining mechanically simple. The fundamental design involves two braces that attach to the cantilevered section of the camera boom and then extend to the two corner truck bed posts on the rear of the truck. The braces will be made from round steel tube with a 3.81 cm (1.5 in) diameter and a 0.318 cm (0.125 in) wall thickness to provide reasonable strength against buckling. Before further specifications could be made, a plan for how the braces would deploy and attach needed to be developed.

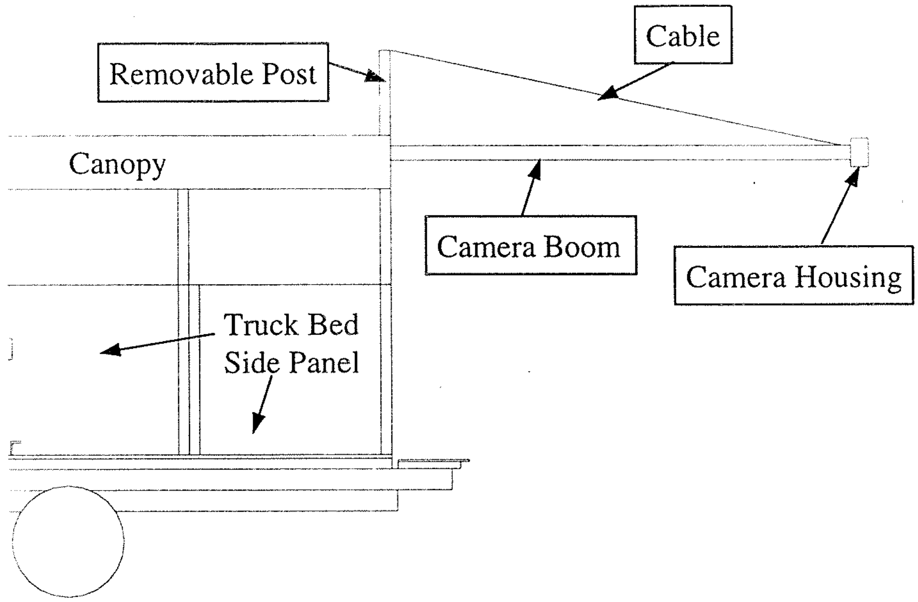

Figure 5.8 *Proposed Camera Boom Cable Support*

5.5.1 Camera Boom Brace Arrangement

For convenience, the braces are stowed next to the camera boom, one brace on each side. To reach this stowage position, the braces disconnect from the truck bed posts and then swing up parallel to the boom, all while remaining attached to the boom. The connection between the boom and the braces uses a square bracket, similar to the two main boom support brackets that hang below the canopy. The braces connect to posts located far enough to the sides of the square bracket so that they clear the rest of the camera boom assembly, see Figure 5.9. This connection forms a rotational joint designed to allow the braces to swing from the stowed position, parallel to the boom, to the connection on the truck bed posts. The square bracket is held in place lengthwise along the boom by a removable pin. For stowage, the pin is removed, and the bracket slides along the boom towards the camera housing, allowing the boom to be fully retracted. On deployment, when the is boom partially extended, the pin will be replaced, then the boom

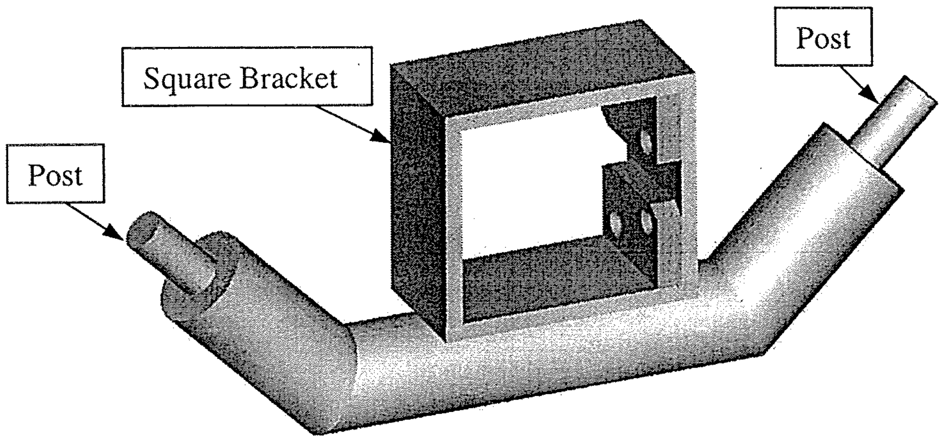

Figure 5.9 *Square Bracket with Brace Attachment Posts*

can be extended the rest of the way, and the braces can rotate down to connect with the truck bed posts.

Ideally, the braces would connect to the truck bed posts near the truck bed floor to create the largest angle possible between the braces and the boom. However, the rear panel of the truck bed prevents the braces from swinging to the bottom of the posts. Either the braces would have to be shortened to clear the panel and then be lengthened again to reach the post, or the boom would have to over-extend temporarily to allow the braces to clear the panel. To avoid either of these complications, the braces are designed to connect to the truck bed posts just above the rear panel, even though the arrangement reduces the angle between the braces and the camera boom. Either of the more complicated options may be employed if the current design does not provide enough rigidity.

5.5.2 Brace Bracket Location

The location on the camera boom where the square bracket attaches must also be considered. If the bracket attaches at the end near the camera, the angle between the braces and the boom will be relatively small, decreasing the effectiveness of the braces.

As the bracket location moves towards the truck, the noted angle increases. However, this is accompanied by the disadvantage of leaving a greater length of the boom cantilevered beyond the bracket.

Examining the vibration characteristics of the boom also provides insight to the problem. With the bracket near the camera, the primary mode shape will be similar to a fixed-pinned beam, with the pinned end being the end with the brace bracket. As the bracket location is moved towards the truck, the mode shape will change and the natural frequency will increase until it reaches a maximum. Moving the bracket further inward will cause the natural frequency to decrease as the cantilevered mode shape becomes dominant.

Both of the above discussions indicate that an ideal location exists along the boom's length for the attachment of the square bracket. To determine the location where the bracket should be fixed, a vibration analysis was used. While a static deflection analysis was considered, the dynamic loads involved required the consideration of a vibration analysis. The following vibration analysis examines the lowest natural frequency of the boom for varying bracket locations. Again, since the vibration input is not defined, the analysis will focus on maximizing the natural frequency.

The analysis begins with the diagram shown in Figure 5.10, which considers only the section of the boom that extends beyond the back of the truck. The fixed support, on the right of the diagram, represents the end of the boom supported by the canopy brackets, while the pinned support, on the boom's midspan, represents the square brace bracket. Rather than using Rayleigh's Method, as with the previous vibration analysis, this analysis will use vibration methods for continuous systems. Again, the support at

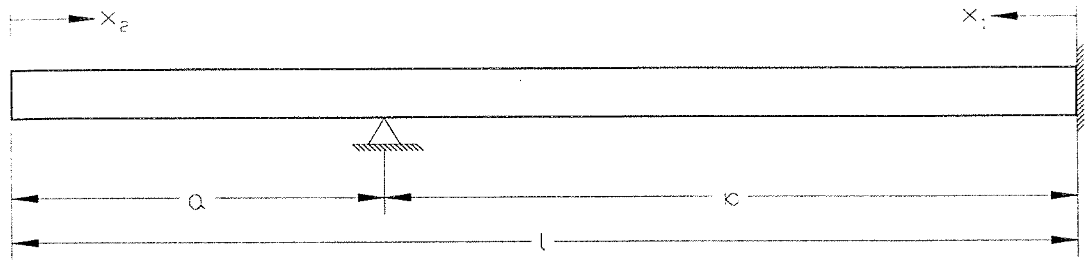

Figure 5.10 *Diagram of Camera Boom for Brace Location Analysis*

midspan makes it necessary to split the boom into two sections and use two coordinates, x_1 and x_2, in the analysis. The weight of the camera and its housing is neglected to simplify the analysis. This simplification will not alter the trend of the results and should not change numerical results significantly.

The vibration mode shapes are described by Equations 5.22 and 5.23, where: C_1 through C_8 are constants to be determined, β is a constant containing boom properties and the unknown natural frequency, and W_1 and W_2 are the lateral displacements of the two sections of the boom.

$$W_1 = C_1 \cdot \cos(\beta \cdot x_1) + C_2 \cdot \sin(\beta \cdot x_1) + C_3 \cdot \cosh(\beta \cdot x_1) + C_4 \cdot \sinh(\beta \cdot x_1) \quad (5.22)$$

$$W_2 = C_5 \cdot \cos(\beta \cdot x_2) + C_6 \cdot \sin(\beta \cdot x_2) + C_7 \cdot \cosh(\beta \cdot x_2) + C_8 \cdot \sinh(\beta \cdot x_2) \quad (5.23)$$

Boundary conditions are used to solve for the constants, C_1 through C_8. Applying the two boundary conditions on the fixed end, zero slope and zero deflection, yields

$$C_3 = -C_1 \quad (5.24)$$

and

$$C_4 = -C_2. \quad (5.25)$$

For the free end of the beam, the shear force and moment are zero, giving

$$C_7 = C_5 \quad (5.26)$$

and

$$C_8 = C_6. \quad (5.27)$$

The four remaining boundary conditions describe the pinned support: zero deflection, which gives one boundary condition for each section of the boom; the two section's slopes are equal; and the moments of the two sections are equal. These last four boundary conditions together with Equations 5.24-5.27 yield

$$0 = C_1 \cdot [\cos(\beta \cdot b) - \cosh(\beta \cdot b)] + C_2 \cdot [\sin(\beta \cdot b) - \sinh(\beta \cdot b)] \tag{5.28}$$

$$0 = C_5 \cdot [\cos(\beta \cdot a) + \cosh(\beta \cdot a)] + C_6 \cdot [\sin(\beta \cdot a) + \sinh(\beta \cdot a)] \tag{5.29}$$

$$C_1 \cdot [\sin(\beta \cdot b) + \sinh(\beta \cdot b)] - C_2 \cdot [\cos(\beta \cdot b) - \cosh(\beta \cdot b)]$$
$$= C_5 \cdot [\sinh(\beta \cdot a) - \sin(\beta \cdot a)] + C_6 \cdot [\cos(\beta \cdot a) + \cosh(\beta \cdot a)] \tag{5.30}$$

$$C_1 \cdot [\cos(\beta \cdot b) + \cosh(\beta \cdot b)] + C_2 \cdot [\sin(\beta \cdot b) + \sinh(\beta \cdot b)]$$
$$= C_5 \cdot [\cos(\beta \cdot a) - \cosh(\beta \cdot a)] + C_6 \cdot [\sin(\beta \cdot a) - \sinh(\beta \cdot a)]. \tag{5.31}$$

A Matlab program, shown in Appendix B, was used to solve for the natural frequency using Equations 5.28-5.31. To find the natural frequency, the trigonometric coefficients of the four remaining constants, C_1, C_2, C_5, and C_6, are put in 4 x 4 matrix. The values of β that make the determinant of this matrix equal to zero will return the natural frequencies of the system. Using the smallest value of β satisfying the above condition, the lowest natural frequency is expressed as

$$\omega = \beta^2 \cdot \sqrt{\frac{E \cdot I}{\rho \cdot A}} \tag{5.32}$$

where A denotes the cross-sectional area, I denotes the area moment of inertia, ρ denotes the density of the boom material, and E is the material's Modulus of Elasticity. Figure 5.11 shows the variation in the lowest natural frequency with the distance between the free end of the boom and the pinned support.

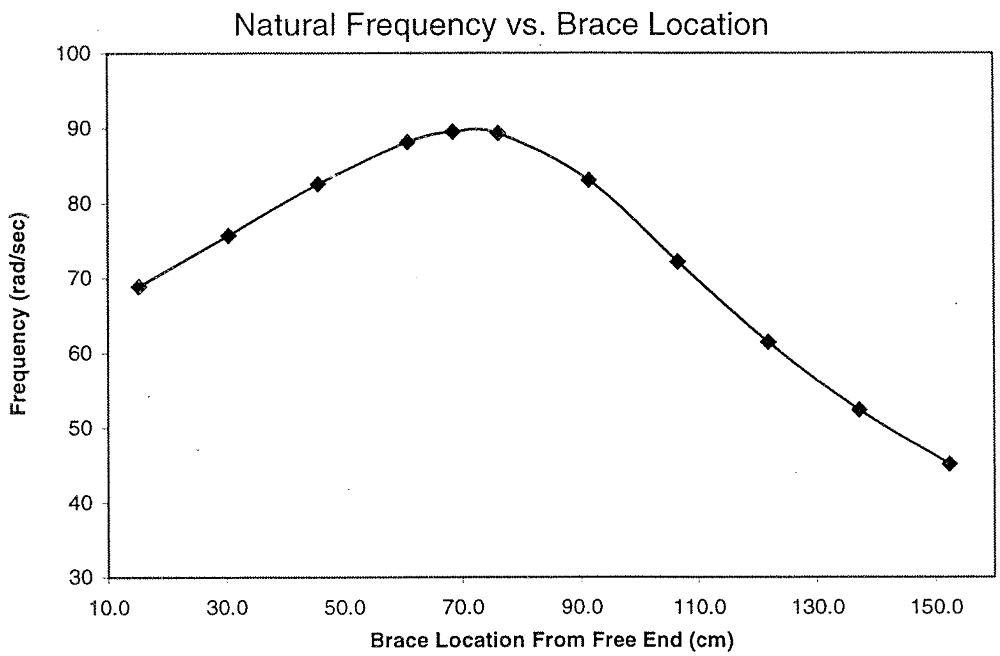

Figure 5.11 *Natural Frequency Results for Brace Analysis*

The graph shows that, based on the vibration analysis, the ideal location for the brace bracket is about 75 cm (30 in) from the free end. However, moving the bracket closer to the truck will increase the angle between the braces and the camera boom, effectively making the braces stiffer. For this reason, the brace bracket will be located about 50 cm (20 in) closer to the truck.

5.6 The Camera Housing

A camera housing is needed to rigidly constrain the camera at the end of the boom and meet several other requirements. Besides providing a place to mount the camera, the design needs to completely enclose the camera to protect it from the weather. While the OCCSM will not be used in adverse weather, it will likely be stored outside when not in use. Also, because the camera needs to align with the coordinate system of the OCCSM's main robotic arm, camera adjustments are key features necessary in the camera housing. The camera boom sets the camera's physical location, but the camera housing must provide for the rotational orientation of the camera. Extra consideration was taken to ensure that the rotation adjustments would withstand the rough and possibly jarring environment. Additionally, since the weight of the camera and housing has a significant impact on camera boom vibration, the design must be lightweight.

A few basic elements will be present in a camera housing designed to meet the mentioned specifications. First, since the housing will completely enclose the camera, the bottom surface of the housing needs to incorporate a clear, colorless lens, through which the camera can view the road surface. An abrasion-resistant Polycarbonate plastic was chosen for the lens material. Second, including the lens surface, the housing effectively needs three horizontal surfaces. The other two surfaces being the top of the housing and the mounting surface for the camera. Finally, the third important element in the camera housing design is an exit for the cable that connects the camera to the image processing computer.

Two designs were developed for the camera housing. The first uses a custom fabricated sheet metal box to cover an internal aluminum framework. While this design contains all the essential features, it is not considered as robust or as rigid as the second. The main component of the second design is an aluminum tube with an inner diameter large enough for the camera to fit inside. An aluminum cap covers the top of the tube, while the polycarbonate lens fits into the bottom of the tube, making an enclosed chamber for the camera. The camera body mounts to an aluminum ring, and the camera lens fits through a hole in the center of the ring. Once inside the main tube, the aluminum ring holding the camera rests on a ledge machined into the inner wall of the tube. Figure 5.12 shows a cross-section of the assembly.

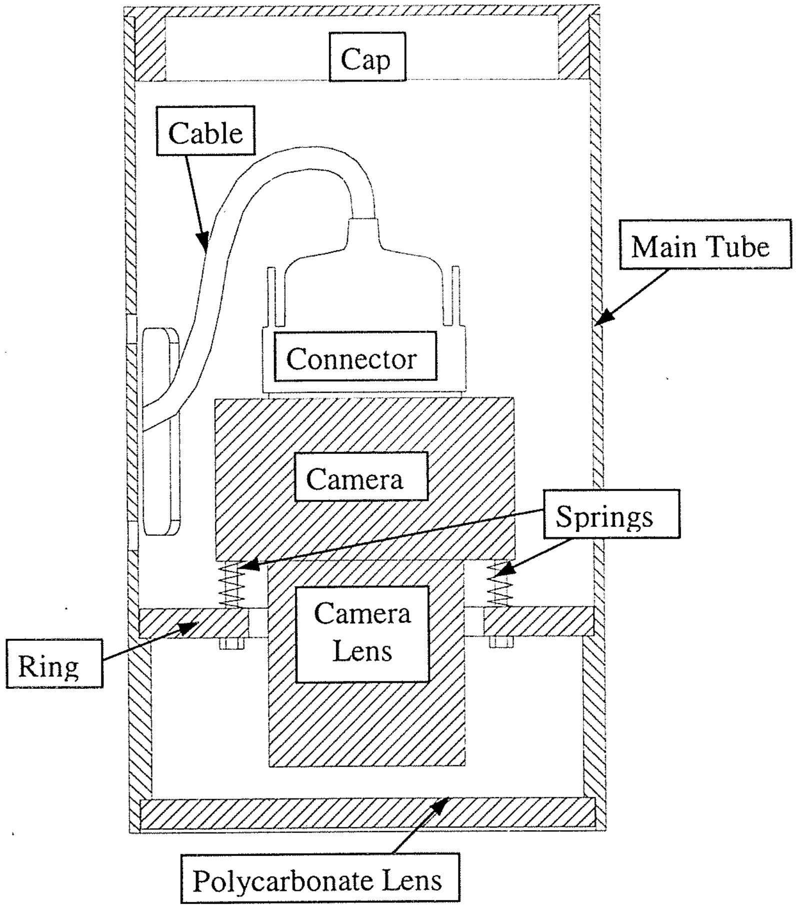

Figure 5.12 *Cross-Section of Camera Housing*

The rotational adjustments for aligning the camera are provided by the connections between the tube, the ring, and the camera. To hold the camera on the ring, bolts protrude through holes on the ring and thread into the four corners of the camera body. Springs around each bolt keep the camera raised off the surface of the ring. Tightening or loosening the appropriate bolts will adjust the camera's tilt angle. The rotational orientation about the axis of the camera lens is set by rotating the ring inside the tube. Once the correct orientation is achieved, screws thread into the ring through slots in the wall of the tube. The slots are long enough to allow 20 degrees of adjustment about the axis.

To allow for a flush connection between the camera housing and the camera boom, a circular cut is machined in the end of the boom. The camera housing bolts to a 2.5 cm (1 in) wide aluminum bar fastened in the center of the camera boom's cross-section. The cable connecting the camera to the computer extends up the center of the camera boom from the computer and fits through a cut in the side of the camera housing, visible in Figure 5.12. The cut is made large enough to accommodate the 16-pin connector that plugs into the back of the camera.

5.7 Summary and Recommendations

This chapter has discussed the design of a mechanism used in obtaining an overhead image of the workspace of the OCCSM's robotic arm. First, the crank mechanism that extends and retracts the camera boom was described. Pending testing, this mechanism is expected to be reliable and easy to use. Next, the selection of the camera boom material was aided by several analyses, including vibration considerations. While the best option available was chosen, once the system has been assembled and used, a judgment can be made regarding whether enough room exists to use a boom with a larger cross-section for the next generation machine. A larger boom may preclude using the braces that were described in the section following the selection of the camera boom size. Finally, the camera housing design was discussed. The camera housing is expected to work well, and should require few, if any, modifications for future versions of the OCCSM.

CHAPTER 6: THE SEALANT HOSE RETRACTION SYSTEM

Supplying sealant to the end of the OCCSM's robotic arm required two new systems to be developed. One of the new systems, the heated sealant hose connecting the melter in the truck bed to the sealant applicator on the end of the arm, has already been discussed in Chapter 3. The other system that needed to be developed is the hose retraction system that adjusts the effective length of the hose to compensate for the extension and retraction of the arm. During crack sealing, the arm on the OCCSM may extend as far as 4.03 m (13.3 ft) (Baker, 1998) to reach the far corners of the workspace. The hose retraction system must store the excess hose, allow the hose to be drawn out as the arm extends, and then bring the hose back into storage as the arm retracts. This chapter discusses the development of the system that manages the hose, allowing it to move with the arm. The system requirements and overall layout of the system are reviewed, options for powering the mechanism are analyzed, and details of the final design are discussed.

6.1 Design Requirements

As mentioned, the main requirement of the hose retraction system is to adjust the hose to match the length of the arm. Naturally, the system should achieve this requirement in the simplest manner, using parts that are reliable, commercially available or simple to make, and require minimal maintenance. Because the arm can supply power to pull the hose out as it extends, the hose retraction system does not need to do more than guide the hose during extension. However, the system must be able to pull the hose back into storage, since the hose can not be pushed back in by the arm. To avoid extra complication involved in determining when the hose should pulled back or left to extend,

6.3 Retraction Power System

Using the pulley and track design, the power system used to retract the hose will attach to the carriage that runs on the track. The system will constantly apply a force acting to pull the carriage and sheave towards the front of the truck bed. Because the hose wraps around the sheave, the force applied to the carriage must be double the force required to overcome friction forces opposing hose movement. All of the options considered to apply a force on the sheave employed a cable connecting the carriage to the power system. Using the cable, the power system will remain stationary allowing it to protrude below the bed frame rails into any space available between the truck's frame rails.

Several options were considered to supply power to keep the cable connected to the carriage under tension; all were powered by either electric, hydraulic, or spring devices. The majority of them employed a reel to wind the cable on, and they only differed in the method of applying torque to the reel. Both the torque required by the power device and the number of turns necessary to fully extend and retract the hose were varied by considering different size cable reels. A gear reduction mechanism, using a series of chain sprockets and shafts, was also examined to adjust torque levels required by the power system.

6.3.1 Electric and Hydraulic Motor Options

While using an electric motor to provide torque is a possibility, it would require extra systems to prevent the motor from overheating when stalled. For example, power to the motor would need to be regulated based on whether the robotic arm was in motion or not. In order to avoid the problems associated with an electric motor, the use of a

The method that was chosen to store the hose consists of a pulley and track system running along the length of the truck bed. The hose wraps around the sheave once, so the sheave needs to move half the distance that the arm extends. In order to allow the sheave to move, it is mounted on a carriage that rides in tracks parallel to the frame rails, as shown in Figure 6.1. With this design, the hose coming from the melter will enter the area between the frame rails just beyond the point where the sheave stops when the arm is completely extended. Also, the system will be offset as shown in Figure 6.1 to locate the hose going to the arm as close as possible to the centerline of the truck. The 10 cm (4 in) EMT conduit, which routes the hose over the truck's differential, will also direct the hose the remaining distance to the centerline of the truck so the hose can enter the robotic arm.

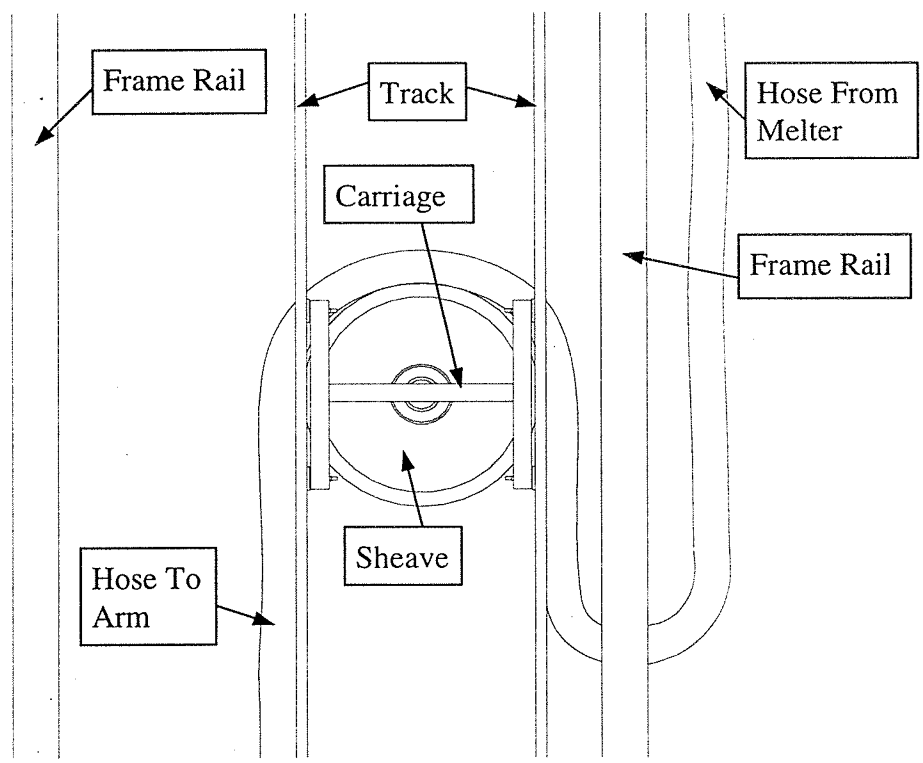

Figure 6.1 *Hose Retraction System Layout*

hydraulic motor was considered. While the hydraulic motor could supply constant torque, it also had a few disadvantages. First, it required routing hydraulic lines from the truck's PTO unit to the location under the truck bed. Also, to provide a constant torque, a constant flow of hydraulic fluid through the motor was required, making the system unnecessarily inefficient.

6.3.2 Hydraulic Cylinder Options

The use of hydraulic cylinders also required the installation hydraulic lines, but did not have the inefficiency associated with hydraulic motors. The first of two cylinder powered designs utilized the gear reduction mechanism described above. In addition to the cable connecting the carriage to the output of the gear mechanism, another cable and reel assembly connects the input of the gear mechanism to the cylinder rod. The configuration of the gear reduction allowed the cylinder to have a reasonable stroke length, but required it to produce higher forces. The main disadvantages of this system are the necessity for gearing, the additional parts needed to mount the gear mechanism, and the second reel and cable assembly.

The second design that utilized a hydraulic cylinder was based on a block and tackle. The cable connected to the carriage would be wound through the pulleys in the block and tackle, and the cylinder would simply provide a constant force to expand the mechanism. This design was mechanically simpler and easier to mount beneath the truck bed than the gear reduction design. However, this design still required a hydraulic supply, and a suitable block and tackle would have to be found or fabricated. Avoiding the hydraulics in either of these two designs would be possible by using gas springs.

While this would eliminate the need for hydraulics, a simpler method is available where all the necessary parts, including the cable, are available as a single unit.

6.3.3 Spring Powered Tool Balancer

The use of a spring mechanism proved to be the simplest way to solve the hose retraction problem. A spring mechanism requires no electrical or hydraulic connections since it stores up energy as the arm extends. After examining several options of constant force and constant torque springs, a tool balancer, which uses a constant torque spring, was found that met all requirements.

A tool balancer is typically used to support a hand tool hanging above a manufacturing workstation, making the given operation more efficient and physically less stressful for the operator. The main body of the tool balancer encloses the spring and a reel for the cable. The spring is typically preloaded so it provides a nearly constant tension on the cable throughout its extension. The only significant disadvantages of the tool balancer design are the weight and cost of the tool balancer, and the spring's fatigue life. However, the weight and cost factors are likely very similar to the other options considered, which required more parts and involved more complication. The only data offered by the tool balancer manufacturer about the spring's fatigue life was that in typical operation they last for a few years. The spring's life is not a major concern because the OCCSM is expected to undergo limited use and the spring in the tool balancer can be replaced if failure occurs. The tool balancer used in this mechanism will provide about 330 N (75 lb) of tension on the cable with an extension length of 200 cm (78 in).

6.4 Detailed Design

With the conceptual design completed, the focus was shifted to the design of the individual parts. The track and rollers were the first items specified in detail since they work together and interact with a majority of the other parts. Several different configurations and track styles were considered. Commercially available track and roller combinations in general had high precision and tolerance, and were therefore also high in price. The best alternative was determined to be one using a standard 3.8 x 1.9 x 0.32 cm (1.5 x 0.75 x 0.125 in) C-channel for a track and having custom steel rollers made.

The rollers, which ride down the middle of the C-channel, were designed with a few features that were essential for their task. First, since the legs of the C-channel are slightly tapered, the roller surface in contact with the channel legs was machined with a matching taper, ensuring a maximum amount of contact area. Second, a lip was left on the roller that will come in contact with the end of the channel legs to support any side loads. The middle of the roller was bored out to accommodate a bearing and a shoulder bolt that mounts the roller to the carriage.

The carriage is made from three pieces of 2.5 cm (1 in) square steel tubing welded together to form an "H", as shown in Figure 6.2. The four rollers bolt to the ends of the two side tubes, and the shaft constraining the sheave is welded to the middle of crossbar that forms the center of the "H". The size of the carriage places the wheels at the corners of a 30 cm (12 in) square, making the reaction forces on the rollers less than 156 N (35 lb) under the expected operating conditions. With the expected loads, the maximum bending stress in the carriage is 5.2 MPa (750 psi), providing a factor of safety against yield of 48.

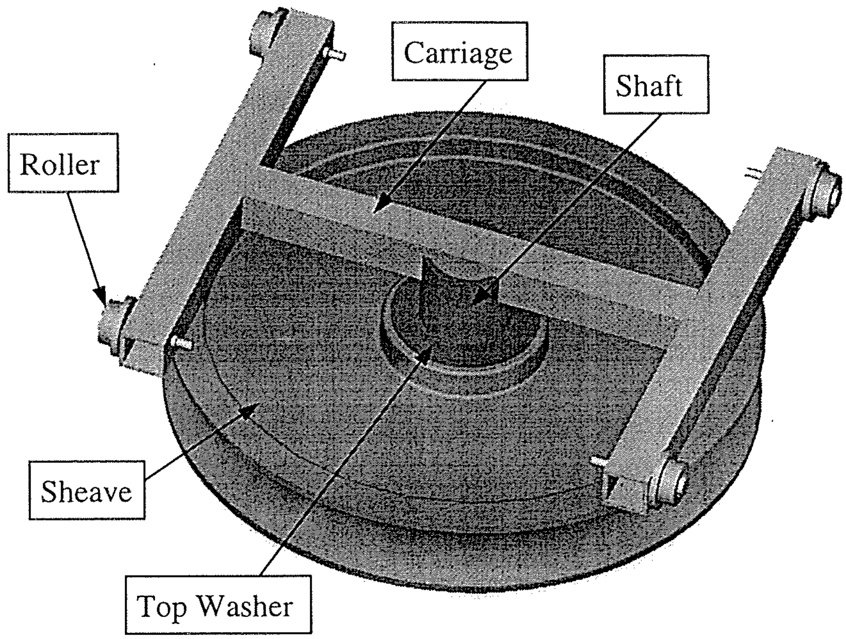

Figure 6.2 *Model of Carriage and Sheave*

Finding something to function as a sheave required searching through several catalogs and contacting suppliers. Most of the difficulty resulted from the requirement for a relatively large diameter with a thin profile. Even though the hose was chosen because of its small bend radius, the minimum radius was still 18 cm (7 in), requiring a 36 cm (14 in) diameter sheave. The best solution found was a 41 cm (16 in) diameter textite wheel that was modified to fit the application. The main modifications include making the wheel thinner, cutting a round slot around the diameter of the wheel for the hose, and removing unnecessary material to reduce weight.

With the sheave design complete, its mounting shaft could be designed. As mentioned, the shaft attaches to the middle of the carriage and is made from a steel tube. The shaft's outer diameter is cut to match the 6.17 cm (2.43 in) inner diameter of the wheel. No bearing is used because the toughness of the wheel material and the relatively large wearing area provides a design life adequate for the prototype machine. To constrain the wheel in the axial direction, two custom flat washers are used, which contact flat surfaces near the wheel's hub. The top washer can be seen in Figure 6.2, and both are visible in the cross-section view shown in Figure 6.3. The lower washer bolts to the end of the shaft once the parts are assembled.

Brackets are used to connect all the components to the truck. These brackets are spaced along the length of the track and mounted to the crossbars that run across the width of the truck on top of the bed frame rails. The brackets are fabricated from 3.81 cm (1.5 in) steel angle, making it easy to connect the track to the crossbars, which run perpendicular to each other. Because the inside of the C-channel must be left clear for the rollers, the brackets are welded to the tracks, but are bolted to the crossbars so that the assembly is still removable.

While the design of the primary components of the hose retraction system was complete, a method of supporting the hose under the truck bed was still needed. There are two lengths of hose requiring support. The first length of hose stretches from where it enters the area under the truck bed to where it wraps around the sheave. The second stretch is between the sheave and the conduit that routes the hose around the truck's differential. Obviously, both of these lengths change as the hose extends and retracts with the robotic arm. To support the hose throughout its range of motion, a sheet metal tray hangs just far enough below the sheave for clearance, see Figure 6.3. The groove around the diameter of the sheave should slightly move the hose up and off the tray as the hose moves around the sheave. The same brackets used to mount the track to the truck bed are also used to hold the tray. The brackets' geometry accommodates the tray, which is about 15 cm (6 in) wider than the distance between the tracks

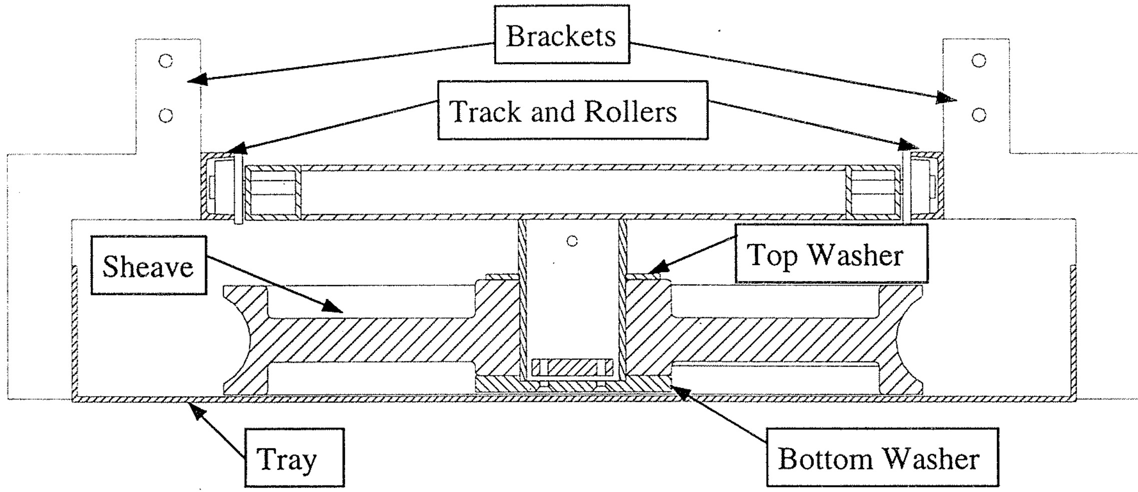

Figure 6.3 *Cross-Section of Hose Retraction Assembly*

6.5 Summary and Recommendations

This chapter has presented the requirements and design of the hose retraction system on the OCCSM. The hose retraction system is needed to adjust the effective length of the sealant hose as the robotic arm extends and retracts. An important and challenging aspect of the design was placing the system in the limited space between the truck bed frame rails. Section 6.2 discussed the conceptual design and the reasons for using a pulley and track system. The several options for powering the mechanism were presented with the advantages and disadvantages of each discussed in Section 6.3. The power option using a tool balancer was chosen mainly because it did not need an additional supply of power since it stores up energy as the arm extends. Other factors influencing the decision were that the tool balancer required only parts used for mounting it to the truck bed and its availability commercially as a complete unit. Section 6.4 discussed the detailed design and important considerations associated with the individual parts.

Many of the system components have already been made or purchased, but they have not been assembled because of delays in receiving the vehicle. Once the truck arrives, the system will be assembled and the hose installed. However, the tool balancer will not be purchased until a measurement can be made of force required to move the hose, allowing the most suitable balancer to be chosen.

Once completed, the system is expected to operate smoothly. However, a few areas should be checked periodically for excessive wear or improper operation. One area of concern involves the hose cover's ability to withstand its interaction with the sheave while the hose is under up to 156 N (35 lb) of tension. Another issue is the effect, if any, that the hose tension will have on the performance of the robotic arm. Most of the concerns result from the tension in the hose coming from friction forces opposing motion. If a problem does arise, adding rollers to specific portions of the hose path could reduce friction in the system. The hose retraction system designed here meets the requirements specified at the beginning of the chapter and should require few modifications for future versions of the OCCSM.

CHAPTER 7: SUMMARY

This thesis focuses on worked aimed at making pavement crack sealing more efficient, safe, and cost effective by automating the application of sealant. Systems were designed that are essential to two different machines dedicated to working on two different types of cracks. The LCSM seals cracks that run parallel to the roadway, typically between the shoulder and the adjacent lane. The linearity of this type of crack readily leads to automating the application of sealant. The second machine, the OCCSM, is designed to seal all other cracks that meander randomly through the pavement, requiring a more complex automation system.

Both machines require a system that melts sealant and then delivers the liquid sealant to the sealant applicator. Chapters 2 and 3 discussed designs that improved the melting and delivery systems, respectively. Current melters cannot even keep up with manual sealing operations, and will delay the work of automated operations even more. In an effort to melt the solid sealant blocks more quickly, a sealant block shredder was developed. While this system performed its task, the melting time was not significantly reduced because of the melter's inability to transfer heat to the sealant quickly. Significant decreases in the time required to melt sealant can only be accomplished through an improved melter and the sealant block shredder developed here.

The hose that delivers sealant from the melter to the sealant applicator must be heated before each use to liquefy sealant left in the hose from the previous use. A heated hose was developed in Chapter 3 to replace current hose heating methods that are, in general, unreliable and not usable with the OCCSM. The design incorporated the electric heater inside the hose where it can most effectively melt the sealant. A thermocouple placed inside the heater is used to control the temperature to prevent overheating either the heater or the sealant. The heater does not reduce the flexibility of the hose, and during testing, it liquefied sealant in the hose in as little as 20 minutes. Further testing of the hose could be performed to determine an optimum voltage level and maximum temperature for the heater. Testing could also include changing the number of times the heater runs the length of the hose and whether one continuous length of heater is used or several for multiple passes. However, as designed, the hose will work on the OCCSM and may be used with other crack sealing operations.

Chapter 4 discussed the design of a linkage for the LCSM that allows the application of sealant on either side of the truck. The linkage mounts to the front bumper of the dual steer truck and will be configured to work on a specific side before leaving for the work site. Once at the work site, the driver controls pneumatic actuators to position the sealant applicator on the pavement beside the truck. The linkage is a 4-bar mechanism with an additional arm, called the swing arm, attached to the front link. Moving the 4-bar linkage from its stowed position extends the swing arm and positions the sealant applicator beside the truck for use.

Testing of this mechanism led to a few recommended modifications. First, the weight of the swing arm must be taken off of the sealant applicator because the drag resulting from the excessive weight prevents the applicator from sliding smoothly across the pavement. Second, an adjustment needs to be incorporated into the mechanism, allowing the machine to easily accommodate different lane widths. Additionally, the weight of the mechanism could be reduced by using a material other than steel and removing unnecessary material from individual parts.

Chapters 5 and 6 discussed the design of systems for the OCCSM. The operation of the OCCSM requires an overhead image of the workspace behind the truck. The operator will use the image to trace a crack and create a path for the robotic arm holding the sealant applicator to follow. For this purpose, a camera boom was developed that holds a digital camera over the center of the workspace. When not in use, the camera boom is retracted by means of a hand operated crank mechanism that is also used to extend the boom. The camera boom and crank mechanism mount to the underside of the canopy that covers the OCCSM's truck bed. Vibration characteristics of the camera boom were used to select a material and an appropriate wall thickness for the square tubular boom. To further minimize camera movement, braces were added between the boom and the rearmost truck bed posts. While the boom positions the camera directly over the center of the workspace, the camera's mount must provide rotational adjustments to align the camera with the workspace coordinates. An aluminum camera housing was developed to perform this task and to protect the camera from the weather. The main option available to improve the system developed here would be an increase in the cross sectional size of the boom. However, before a judgment on this option can be made, the overhead clearance in the truck bed must be evaluated.

Because the OCCSM's robotic arm may extend up to 4.03 m (13.3 ft), the sealant hose must also be able to extend this distance. The sealant hose retraction system discussed in Chapter 6 was designed to adjust the hose length to match the effective length of the arm. The hose runs through a sheave mounted to a carriage that rolls along a track, allowing the hose to extend and retract with the arm. A spring powered tool balancer stores energy as the arm pulls the hose out, then as the arm retracts, the tool balancer pulls the hose back in. A tool balancer was chosen because it does not require any electric or hydraulic power and was a single item that performed all the necessary functions. Testing of the hose retraction system will be performed once the truck arrives and the system can be assembled on the truck.

The work detailed in this thesis focused on developing essential systems for two crack sealing machines, the LCSM and OCCSM. Together, the LCSM and the OCCSM can seal all the cracks on a roadway. While each still requires an operator to locate the crack, both of the machines will increase the rate and quality of crack sealing through automation. An additional benefit resulting from the machines is worker safety. Neither machine requires any worker to be outside the vehicle where they are directly exposed to danger from traffic flow. The systems developed in this thesis will allow the realization of these advantages as well as pave the way for the future of highway maintenance.

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APPENDIX A: MATLAB Program for Camera Boom Size Selection

```
***vib.m***
%This MATLAB file calculates the weight, maximum deflection, natural
%frequency and forced vibration amplitude of the camera boom. Discussion
%of the analysis maybe found in Chapter 5 of this thesis.

clear all
close all

F=5;                        %end load (lb)
l=182;                      %total length (in)
l2=125;                              %cantilevered length (in)
b=3;                        %beam width, OD (in)
h=3;                        %beam height, OD (in)
t=.125;                     %beam wall thickness (in)
E=29*10^6;                  %modulus of elasticity (lb/in^2)
dns=.0088145;               %beam material density (slug/in^3)
  %note: g/cm^3 * 1.12287e-3 = slug/in^3
xA=b*h-(b-2*t)*(h-2*t);              %cross section area (in^2)
Wt=dns*xA*l*32.2            %total beam weight (lbf)
wl=dns*xA + 6/(32.2*l);              %mass per length (slug/in)(includes chain)
w=wl*32.2;                  %distributed load (lb/in)
I=(b*h^3-(b-2*t)*(h-2*t)^3)/12;      %moment of inertia (in^4)
l1 = l-l2;

%%%COEFFICIENTS OF THE DEFLECTION EQUATIONS%%%
%The a terms are the coefficients of x1 and x2 in the deflection equations.
%The first 'subscript' refers to deflection equation 1 or 2, and the second
%'subscript' corresponds to the power of x the coefficient multiplies.
a14 = -w/24;
a13 = -((-w*l^2 + 2*w*l*l2 + 2*F*l2)/(l-l2))/12;
a12 = 0;
a11 = -(w*l^3)/24 + (w*l2*l^2)/8 - (w*l*l2^2)/24 - (w*l2^3)/24 + F*(l*l2-l2^2)/6;
a10 = 0;
a24 = -w/24;
a23 = -F/6;
a22 = 0;
a21 = -(w*l^3)/24 + (w*l2*l^2)/8 + (w*l*l2^2)/24 + (w*l2^3)/24 + F*(2*l*l2+l2^2)/6;
a20 = (w*l2*l^3)/24 - w*(l^2)*(l2^2)/8 - (w*l*l2^3)/24 - (F*l*l2^2)/3;


%%%MAXIMUM DEFLECTIONS - and the ratio r%%%
v2max = a20/(E*I)           %maximum deflection for the cantilevered section

slope1eq = [a14*4 a13*3 0 a11];      %coefficients of the slope eq for section 1
```

```
x1maxs = roots(slope1eq);     %one of these roots will be the value of x1 at v1max

for i = 1:3                   %this for loop determines the correct root
if 0 < x1maxs(i) < l1
 x1max = x1maxs(i);
end
end

v1max = (a14*x1max^4 + a13*x1max^3 + a11*x1max)/(E*I);     %max defl of section 1 OK

r = abs(v1max/v2max);               %ratio of the maximum deflections of the two sections

%%%squaring v1 and v2 and evaluating the integrals I1 and I2 in the Tmax expression%%%

a1 = [a14 a13 a12 a11 a10]*(1/(E*I));          %vector of poly coeff of v1
a2 = [a24 a23 a22 a21 a20]*(1/(E*I));          %vector of poly coeff of v2

c1 = conv(a1,a1);                   %vector of poly coeff of v1^2
c2 = conv(a2,a2);                   %vector of poly coeff of v2^2

I1 = (c1(1)*l1^9)/9 + (c1(2)*l1^8)/8 + (c1(3)*l1^7)/7 + (c1(4)*l1^6)/6 + (c1(5)*l1^5)/5 +
(c1(6)*l1^4)/4 + (c1(7)*l1^3)/3 + (c1(8)*l1^2)/2 + (c1(9)*l1);
I2 = (c2(1)*l2^9)/9 + (c2(2)*l2^8)/8 + (c2(3)*l2^7)/7 + (c2(4)*l2^6)/6 + (c2(5)*l2^5)/5 +
(c2(6)*l2^4)/4 + (c2(7)*l2^3)/3 + (c2(8)*l2^2)/2 + (c2(9)*l2);

%%%finally, the equivalent mass!!!%%%
meqb = (wl*I1*r^2)/v1max^2 + (wl*I2)/v2max^2;  %equliv mass of beam (slug)
mF = F/32.2;                                   %mass of end weight (slug)
meq = meqb + mF;                               %total equivalent mass (slug)

%%%the beam stiffness%%%
k = 3*E*I/(l*l2^2);                 %the beam stiffness (lb/in)

%%%the natural frequency%%%
Wn = sqrt(k*12/meq);                %(rad/s)
Hz = Wn/(2*pi)                                 %(hz)

%%%Forced Vibration Amplitude%%%
c=.291;                             %damping value (lb*s/ft)
z=c/(2*sqrt(12*k*meq));             %damping ratio
Fo=1;                         %magnitude of harmonic force (lb)
Wf=Wn*sqrt(1-2*z^2);                %forcing freq for max amp (rad/s)

den1=(1-(Wf/Wn)^2)^2;
den2=(2*z*Wf/Wn)^2;
amp=Fo/(k*sqrt(den1 + den2))        %maximum vibration amplitude (in)
```

APPENDIX B: MATLAB Program for Camera Boom Brace Location

This appendix contains two MATLAB programs. The first program is the main program, brace.m, shown on this page. The second program, deter.m, shown on the following page, is a function that is called by the main program.

brace.m

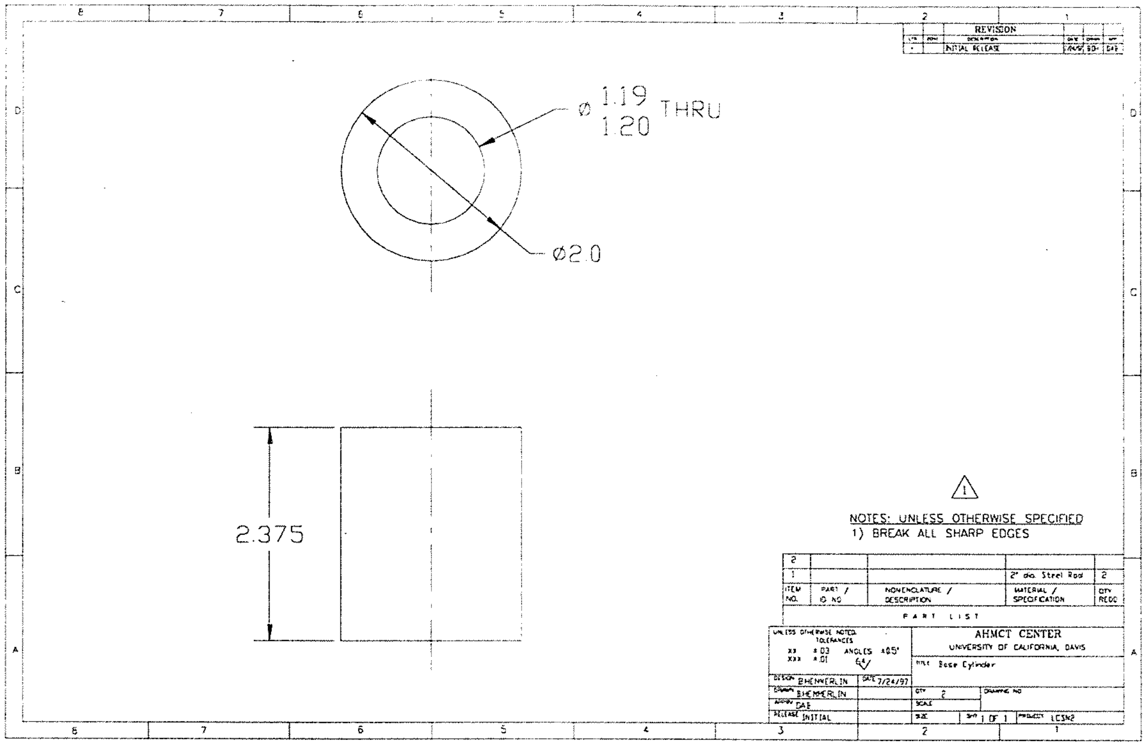
```
clear all
clear global
close all
global dens area I E a b
height=3;
width=3;
thick=.0625;
area=width*height-(width-2*thick)*(height-2*thick);
I=(width*height^3-(width-2*thick)*(height-2*thick)^3)/12;

l=130;                  %length of beam (in)

%These are for the Fiberglass Composite beam
%E=3.15*10^6;                  %modulus of elasticity (psi)
%dens=.0525/32.2;    %density (slugs/in^3)
%These are for a steel beam
%E=29*10^6;              %modulus of elasticity (psi)
%dens=.0088145;                %density (slugs/in^3)
%These are for an aluminum beam
E=10*10^6;               %modulus of elasticity (psi)
dens=.0030430;                 %density (slugs/in^3)
aa=[6 12 18 24 27 30 36 42 48 54 60];
lwrs=ones(size(aa,2));
uprs=100*ones(size(aa,2));

for ainc=1:size(aa,2)
  a=aa(1,ainc);
  b=l-a;
  lwr=lwrs(1,ainc);
  upr=uprs(1,ainc);

  ww(ainc)=fzero('deter',50);
end

figure(1)
plot(aa,ww,'+')
```

deter.m

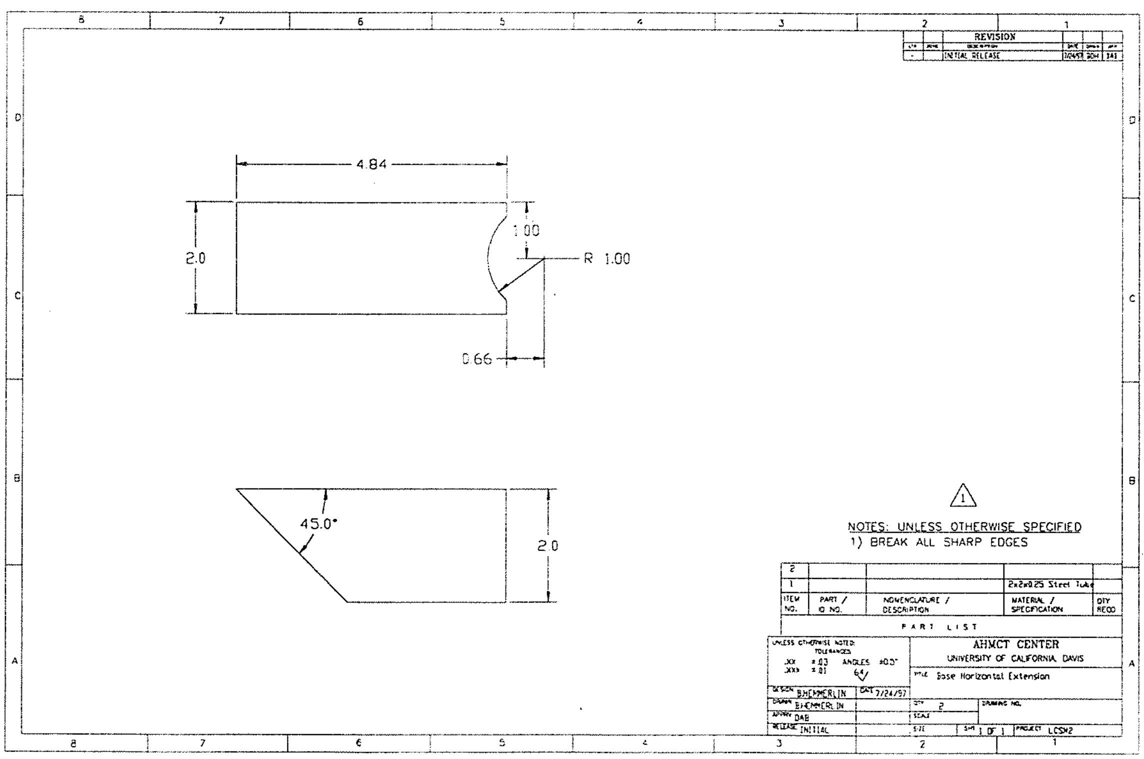
```
function [detr] = deter(w)

global dens area E I a b

beta=(((dens*area)/(E*I))^.25)*sqrt(w);
sa=sin(beta*a);
sha=sinh(beta*a);
sb=sin(beta*b);
shb=sinh(beta*b);
ca=cos(beta*a);
cha=cosh(beta*a);
cb=cos(beta*b);
chb=cosh(beta*b);

matrx=[[(cb-chb) (sb-shb) 0 0]
    [0 0 (ca+cha) (sa+sha)]
    [(-sb-shb) (cb-chb) (sha-sa) (ca+cha)]
    [(-cb-chb) (-sb-shb) (ca-cha) (sa-sha)]];

detr=det(matrx);
```

APPENDIX C: Part Drawings for the LCSM Linkage

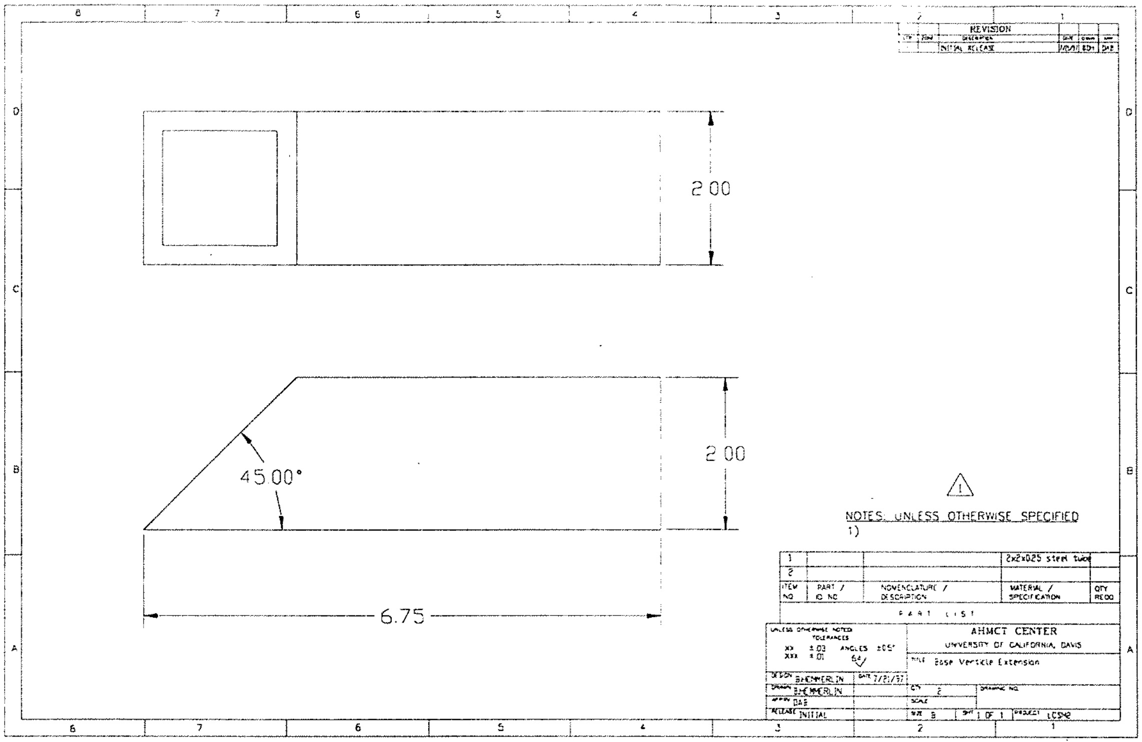
TABLE OF CONTENTS

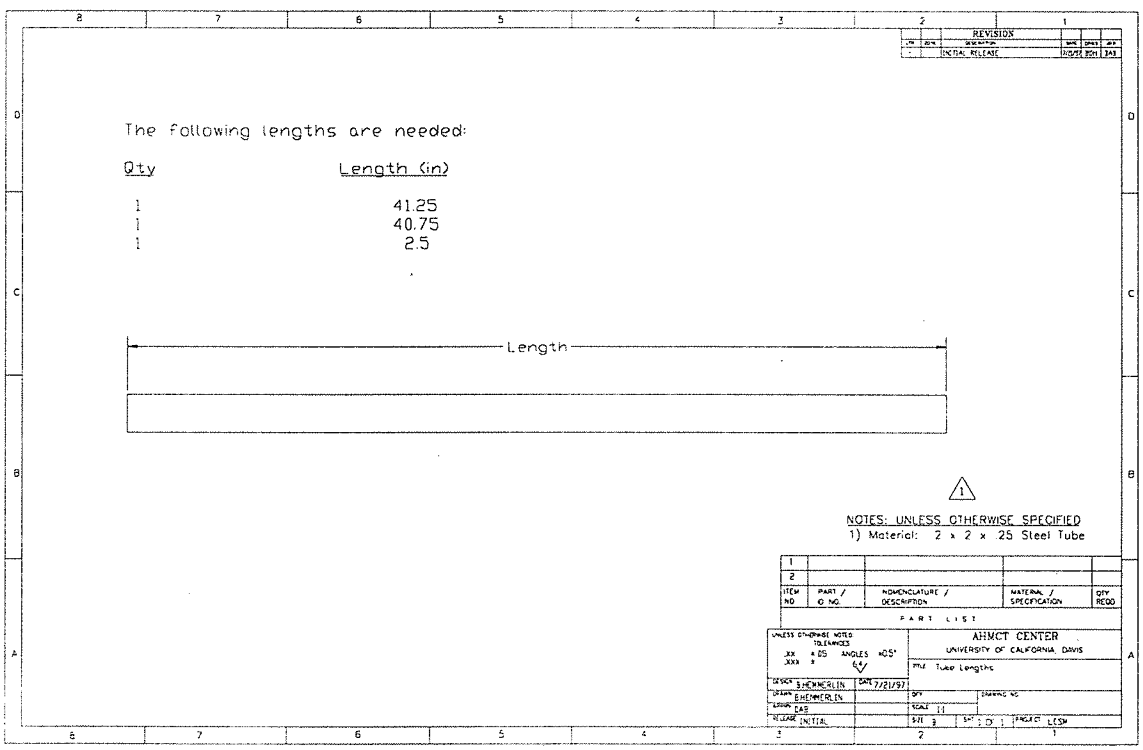


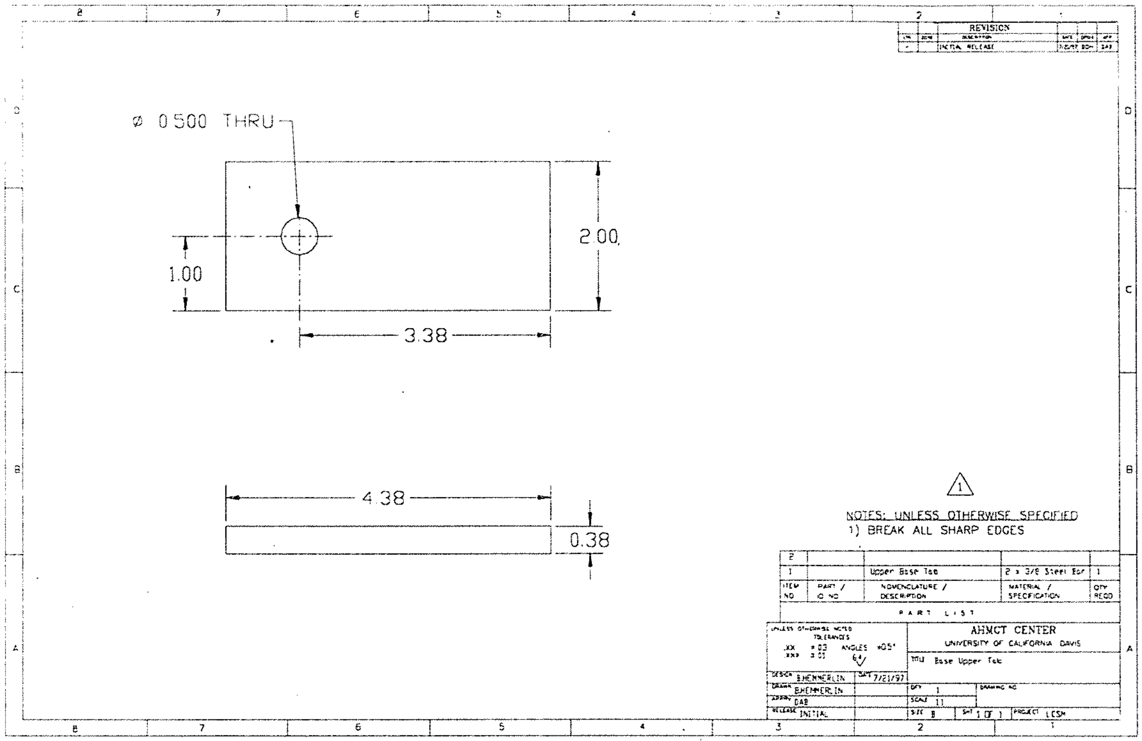
REVISION
INITIAL RELEASE
Ø 1.19 1.20 THRU
Ø2.0
2.375
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2" dia. Steel Rod
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Base Cylinder
B.HEMMERLIN
7/24/97
INITIAL
LCSN2



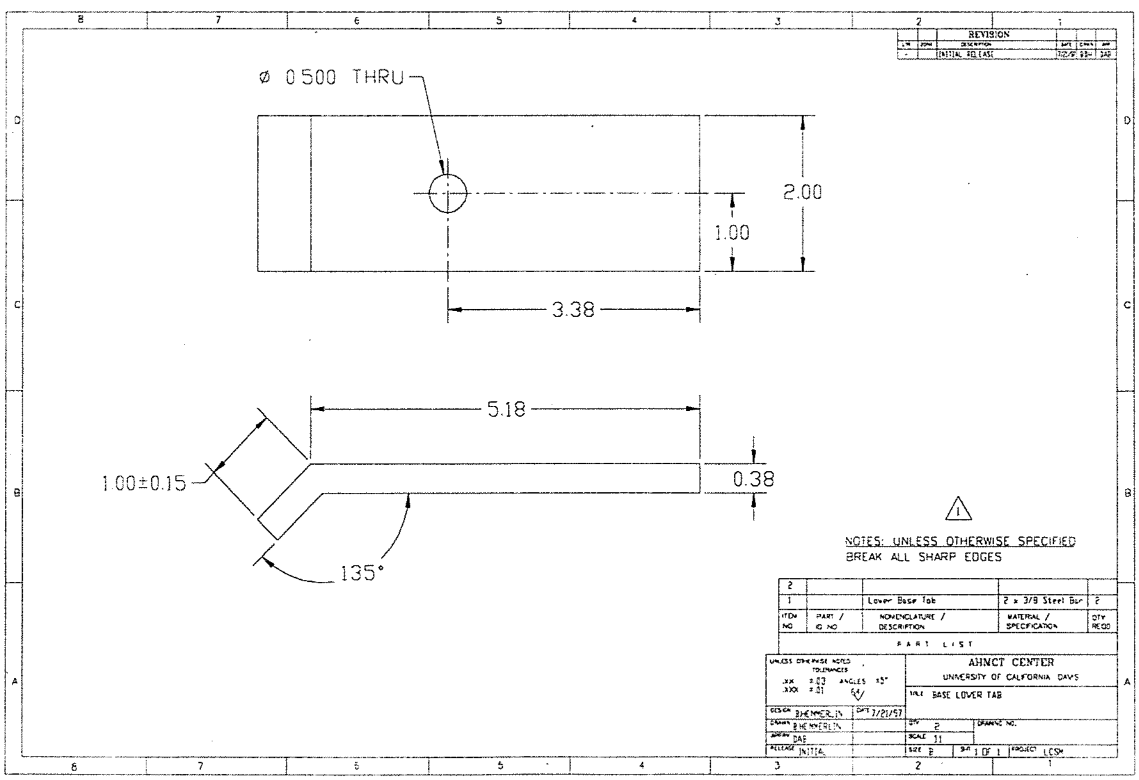
REVISION
INITIAL RELEASE
4.84
2.0
1.00
R 1.00
0.66
45.0°
2.0
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Base Horizontal Extension
B.HEMMERLIN
7/24/97
INITIAL
LCSN2


REVISION
INITIAL RELEASE
2.00
2.00
45.00°
6.75
NOTES: UNLESS OTHERWISE SPECIFIED
1)
2x2x025 steel tube
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Base Vertical Extension
BHEMMERLIN
7/21/97
DAB
INITIAL

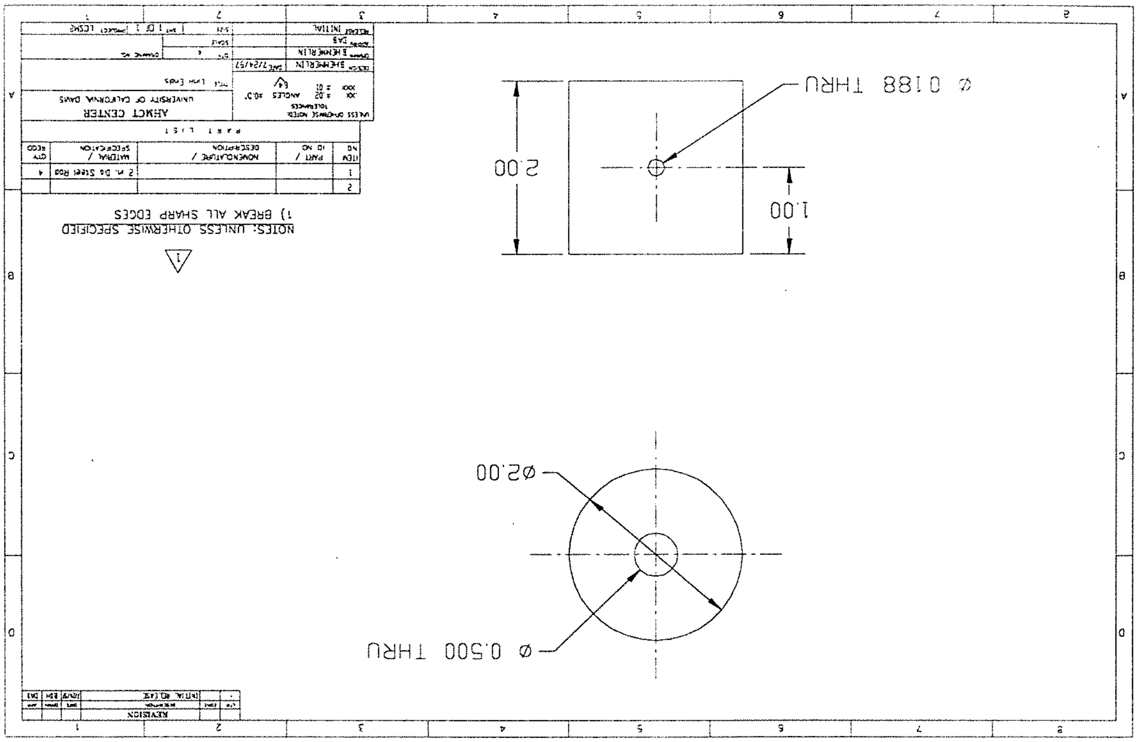

REVISION
INITIAL RELEASE
The following lengths are needed:
Qty
Length (in)
1 41.25
1 40.75
1 2.5
Length
NOTES: UNLESS OTHERWISE SPECIFIED
1) Material: 2 x 2 x .25 Steel Tube
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Tube Lengths
BHEMMERLIN
7/21/97
DAB
INITIAL

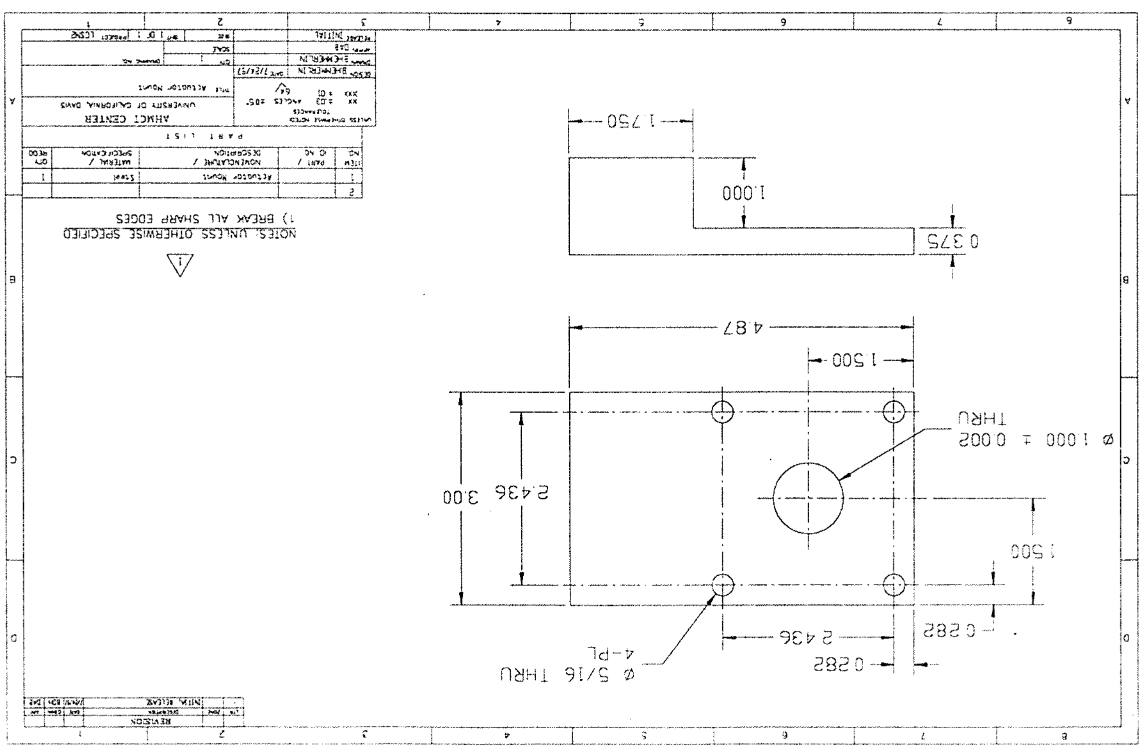


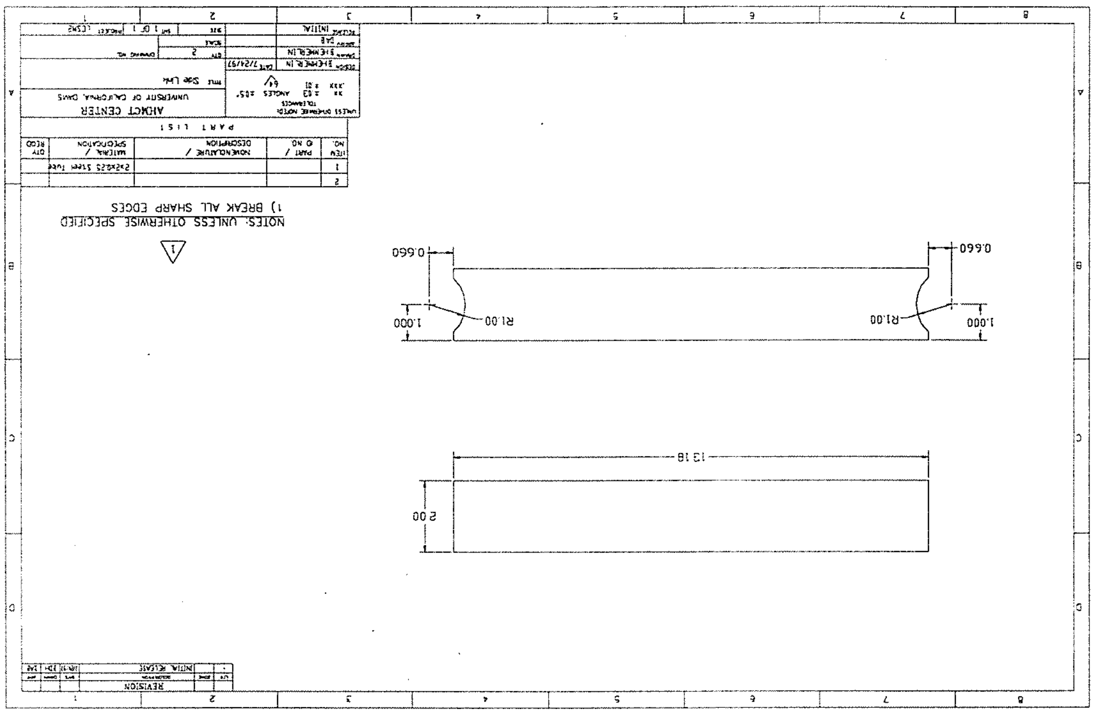
Ø 0.500 THRU
2.00
1.00
3.38
4.38
0.38
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
Upper Base Tab
AHMCT CENTER
UNIVERSITY OF CALIFORNIA DAVIS
Base Upper Tab
BHEMMERLIN
7/21/97

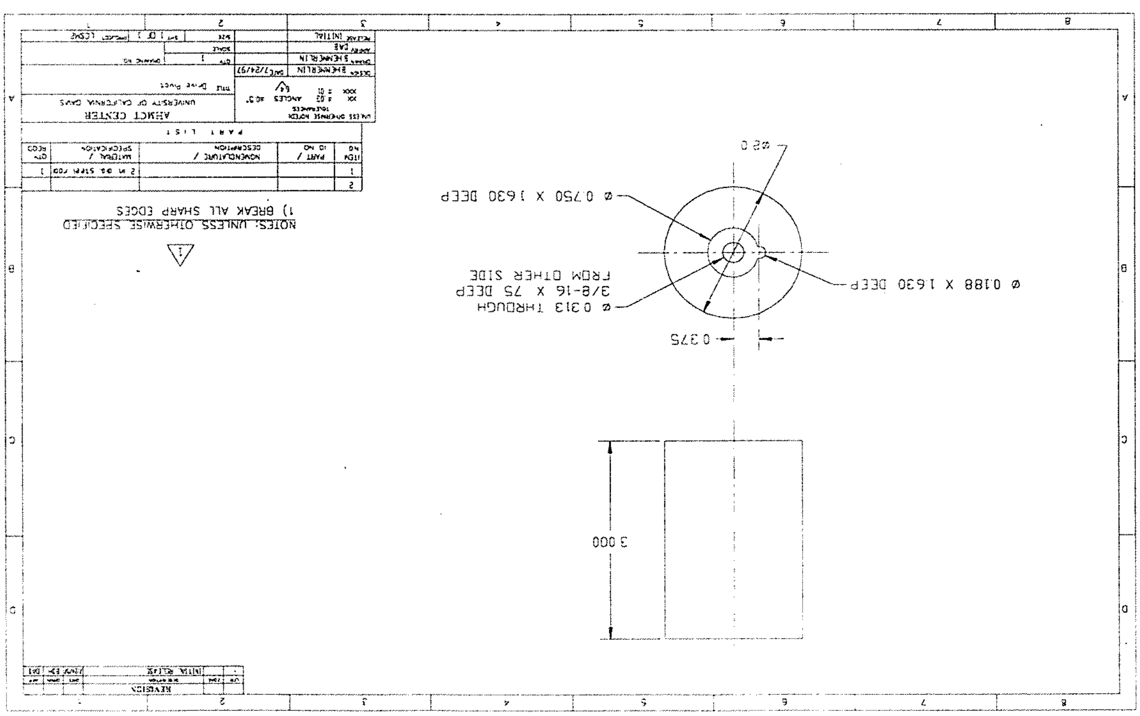


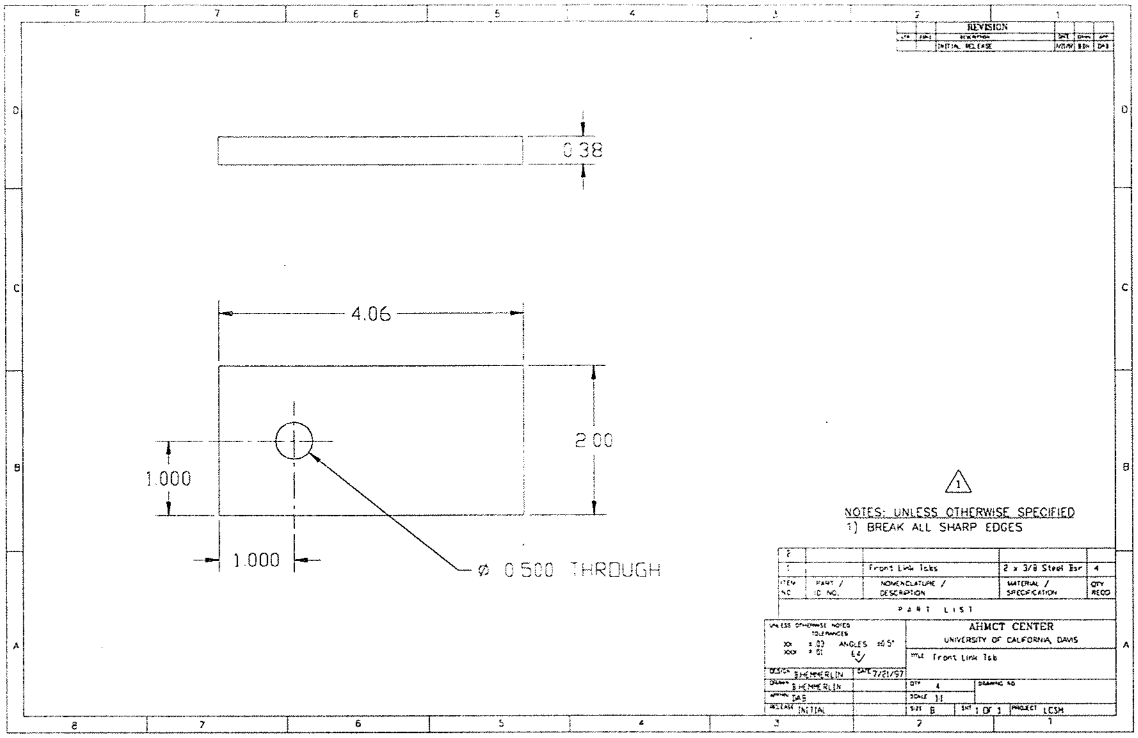
Ø 0.500 THRU
2.00
1.00
3.38
5.18
1.00±0.15
0.38
135°
NOTES: UNLESS OTHERWISE SPECIFIED
BREAK ALL SHARP EDGES
Lower Base Tab
AHMCT CENTER
UNIVERSITY OF CALIFORNIA DAVIS
BASE LOWER TAB
BHEMMERLIN
7/21/97

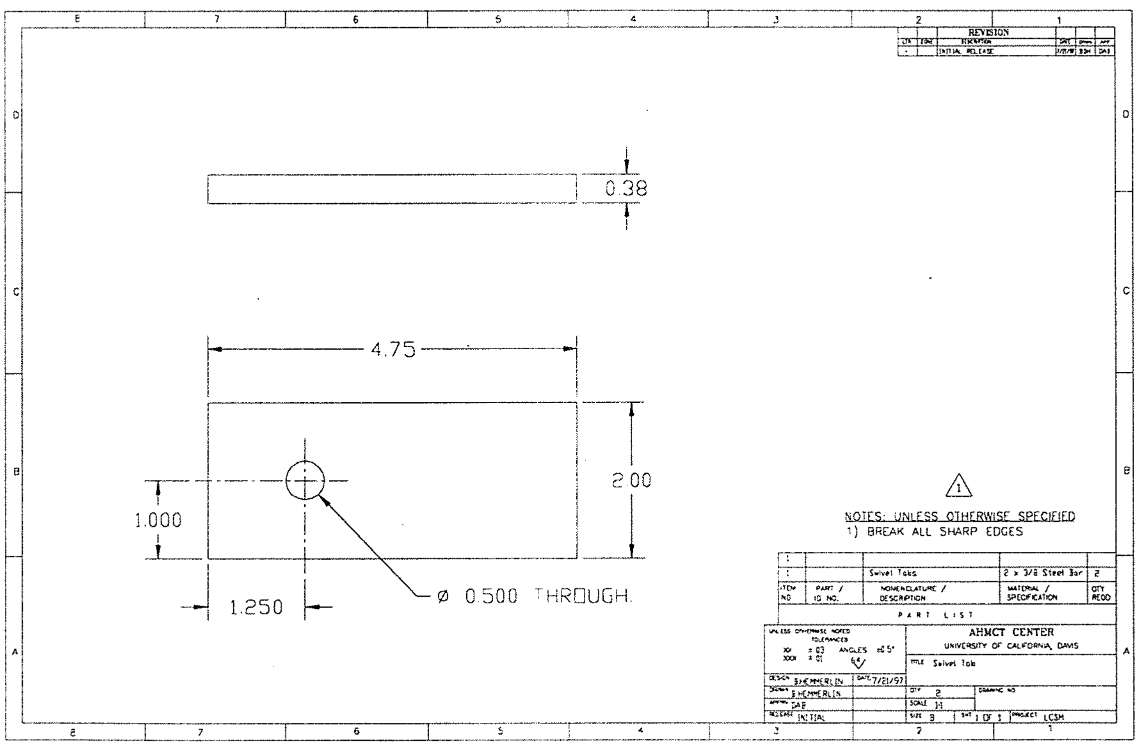

Ø 0.188 THRU
2.00
1.00
Ø2.00
Ø 0.500 THRU
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
REVISION

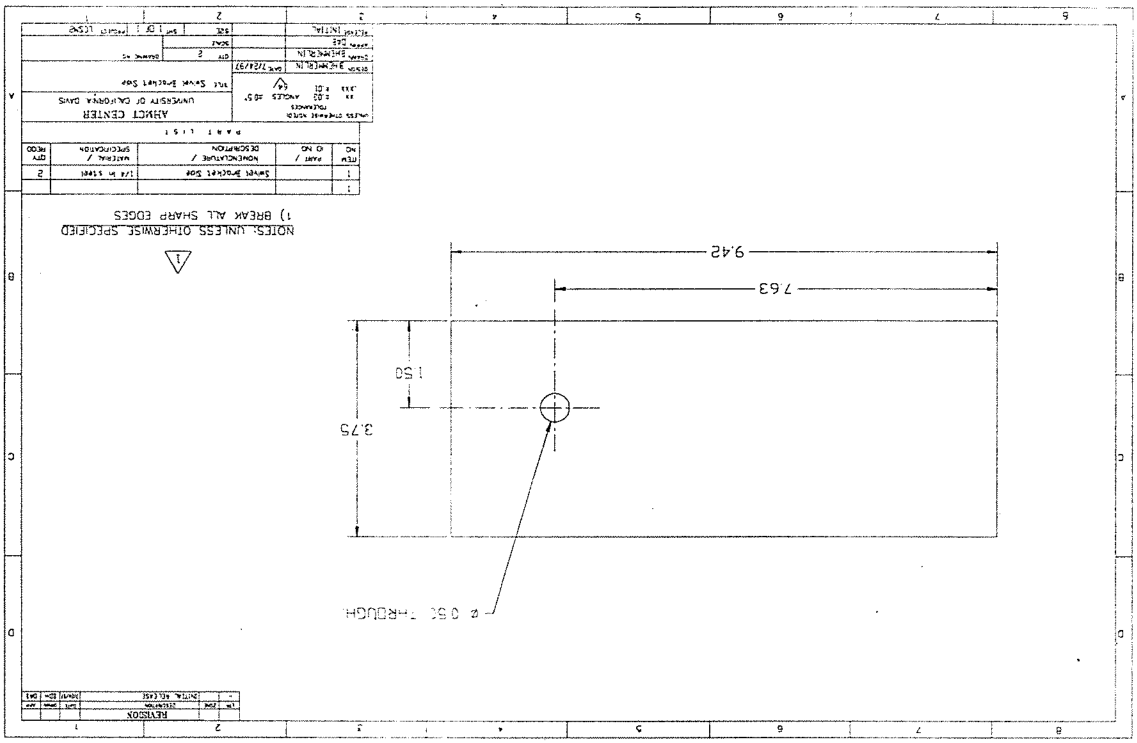

1.750
1.000
0.375
4.87
1.500
Ø 1.000 ± 0.002
THRU
2.436
3.00
0.282
Ø 5/16 THRU
4-PL
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Actuator Mount
Steel
REVISION

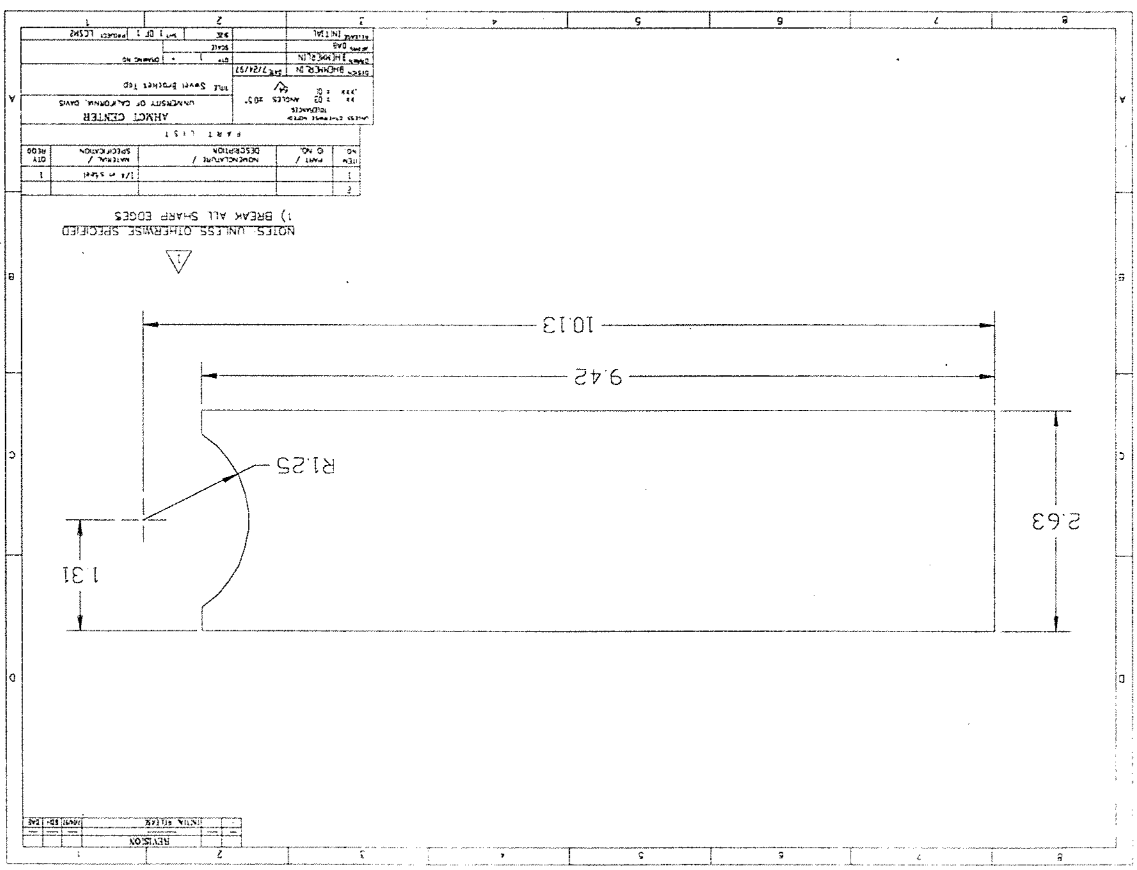

AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
PART LIST
2x2x.25 Steel Tube
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
0.660
0.660
R1.00
R1.00
1.000
1.000
13.18
2.00
REVISION

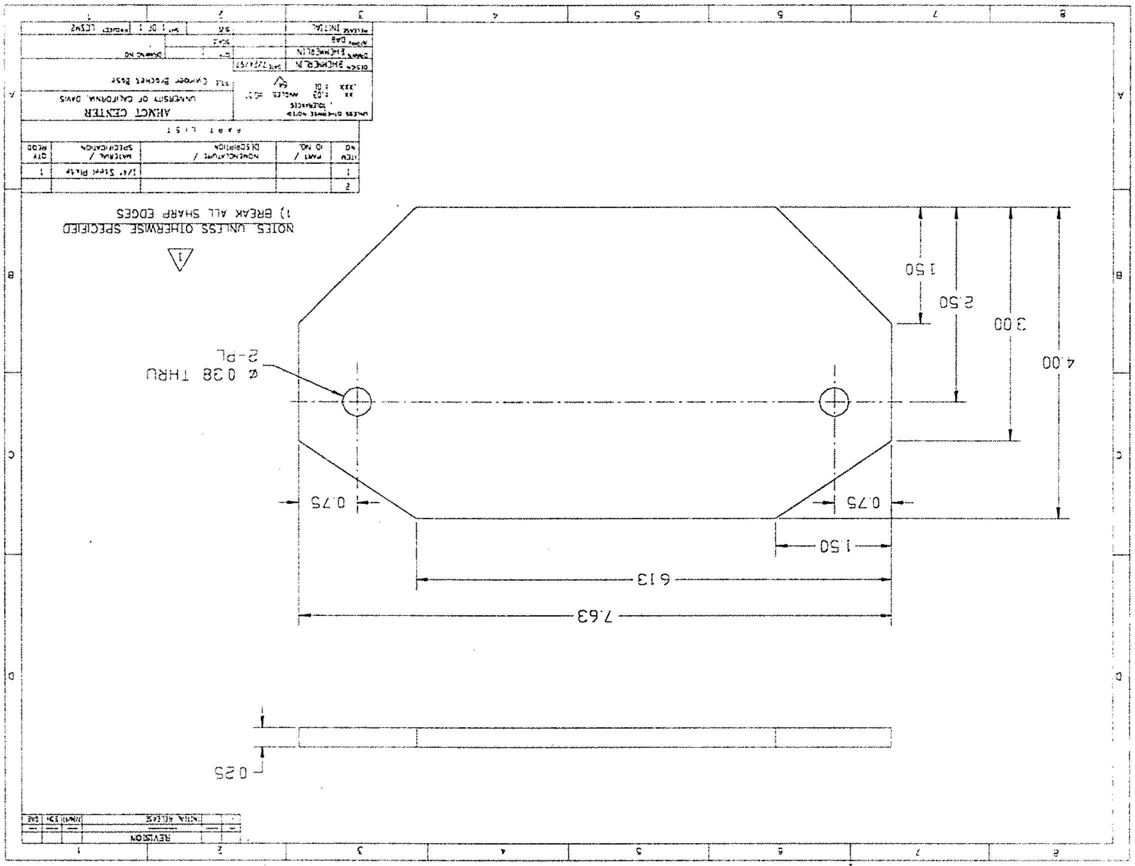

AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Drive Plugs
PART LIST
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
⌀ 0.750 X 1.630 DEEP
⌀ 0.188 X 1.630 DEEP
⌀ 0.313 THROUGH
3/8-16 X .75 DEEP
FROM OTHER SIDE
⌀2.0
0.375
3.000
REVISION

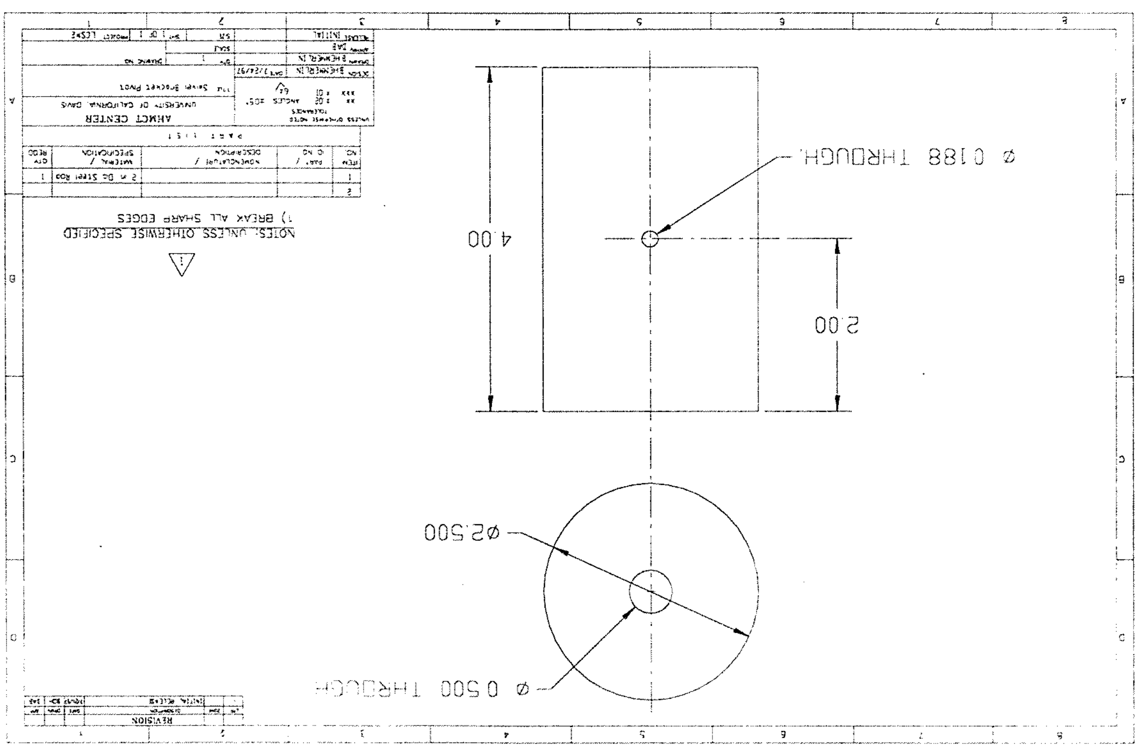

0.38
4.06
2.00
1.000
1.000
Ø 0.500 THROUGH
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
Front Link Tabs
2 x 3/8 Steel Bar
4
ITEM NO
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Front Link Tab
7/21/97
DAS
INITIAL
1 OF 1
LCSM

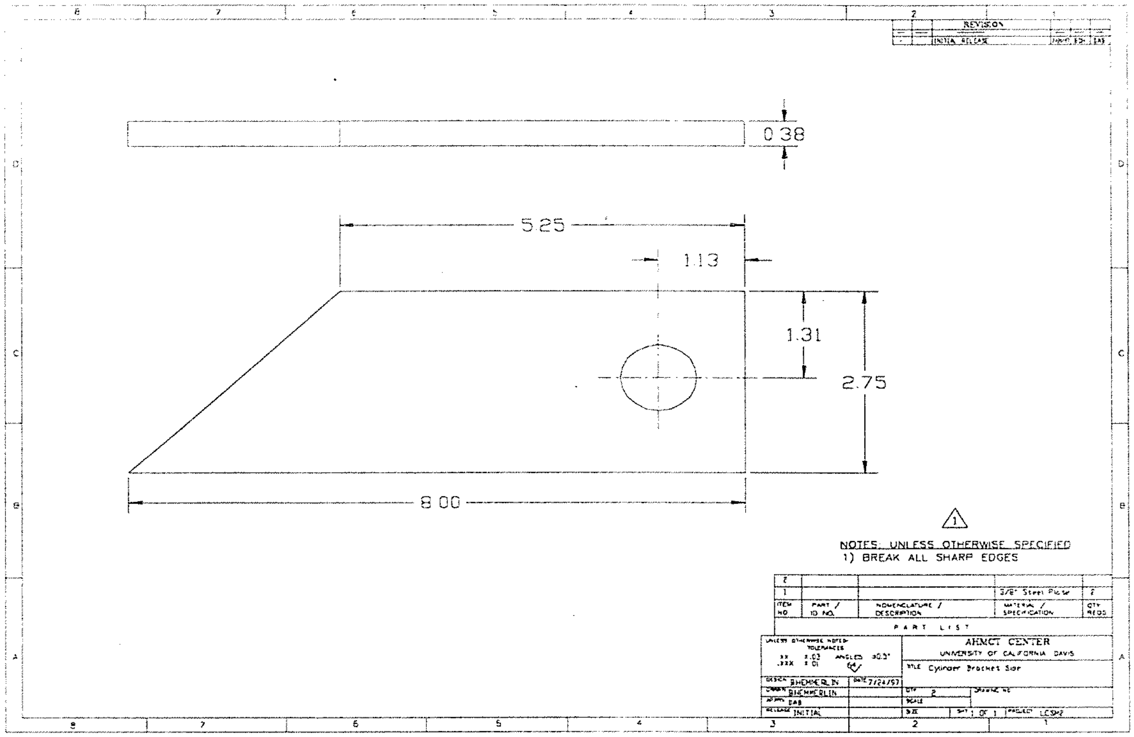

0.38
4.75
2.00
1.000
1.250
Ø 0.500 THROUGH.
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
Swivel Tabs
2 x 3/8 Steel Bar
2
ITEM NO
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Swivel Tab
7/21/97
INITIAL
1 OF 1
LCSM

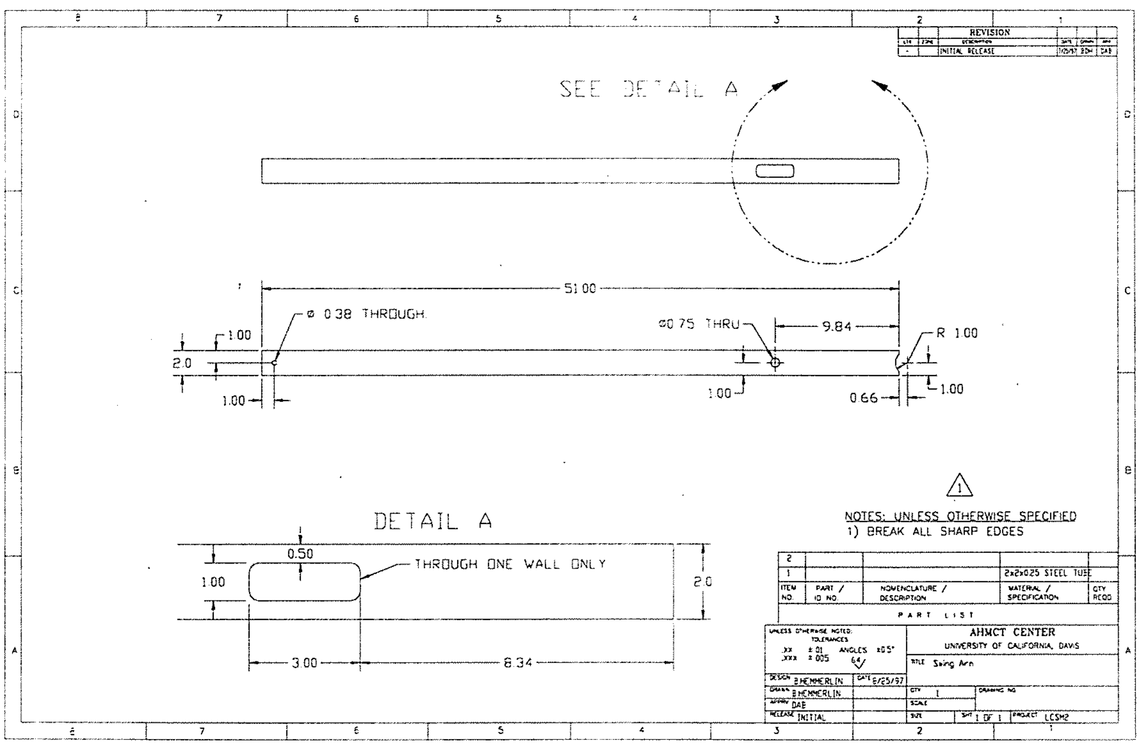

NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
AHMCT CENTER
UNIVERSITY OF CALIFORNIA DAVIS
Swivel Bracket Side
9.42
7.63
1.50
3.75
⌀ 0.50 THROUGH
REVISION


NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
AHMCT CENTER
UNIVERSITY OF CALIFORNIA DAVIS
Swivel Bracket Top
10.13
9.42
R1.25
2.63
1.31
REVISION


NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
1/4" Steel Plate
⌀ 0.38 THRU
2-PL
1.50
2.50
3.00
4.00
0.75
0.75
1.50
6.13
7.63
0.25
REVISION

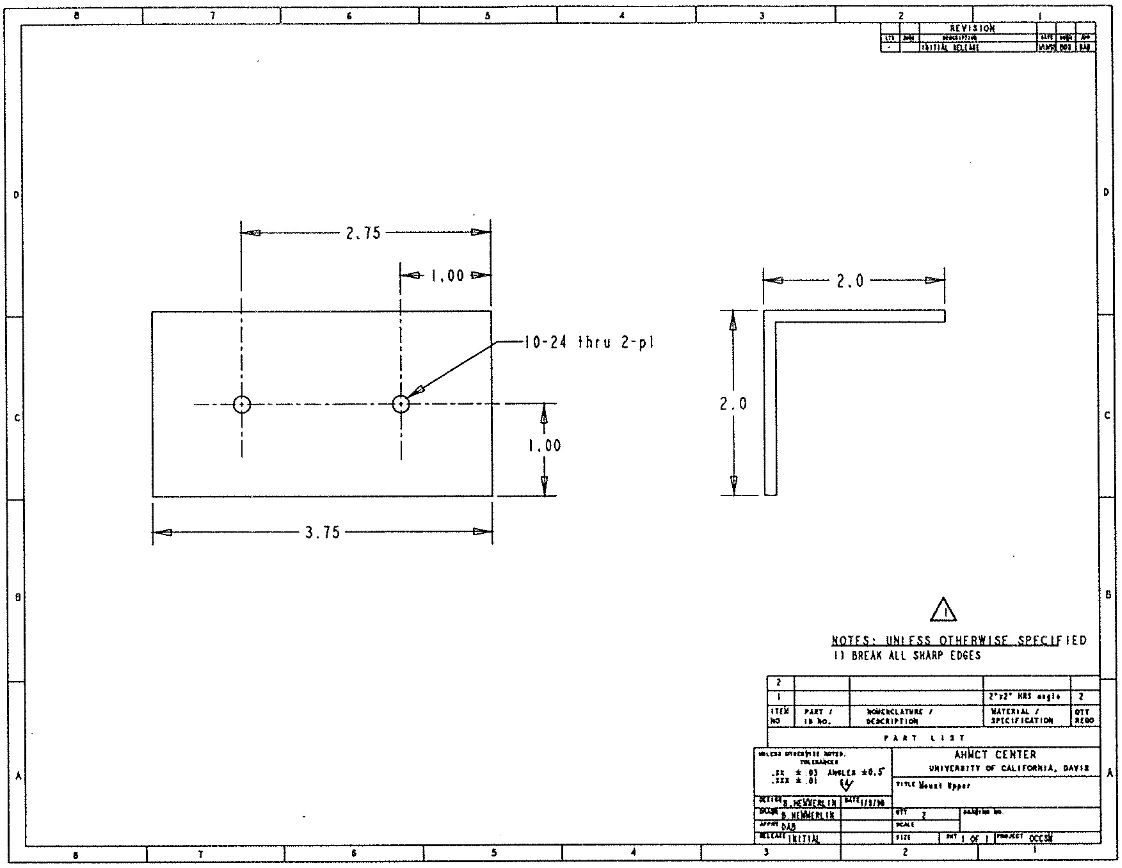

NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
⌀ 0.188 THROUGH
4.00
2.00
⌀2.500
⌀ 0.500 THROUGH
REVISION

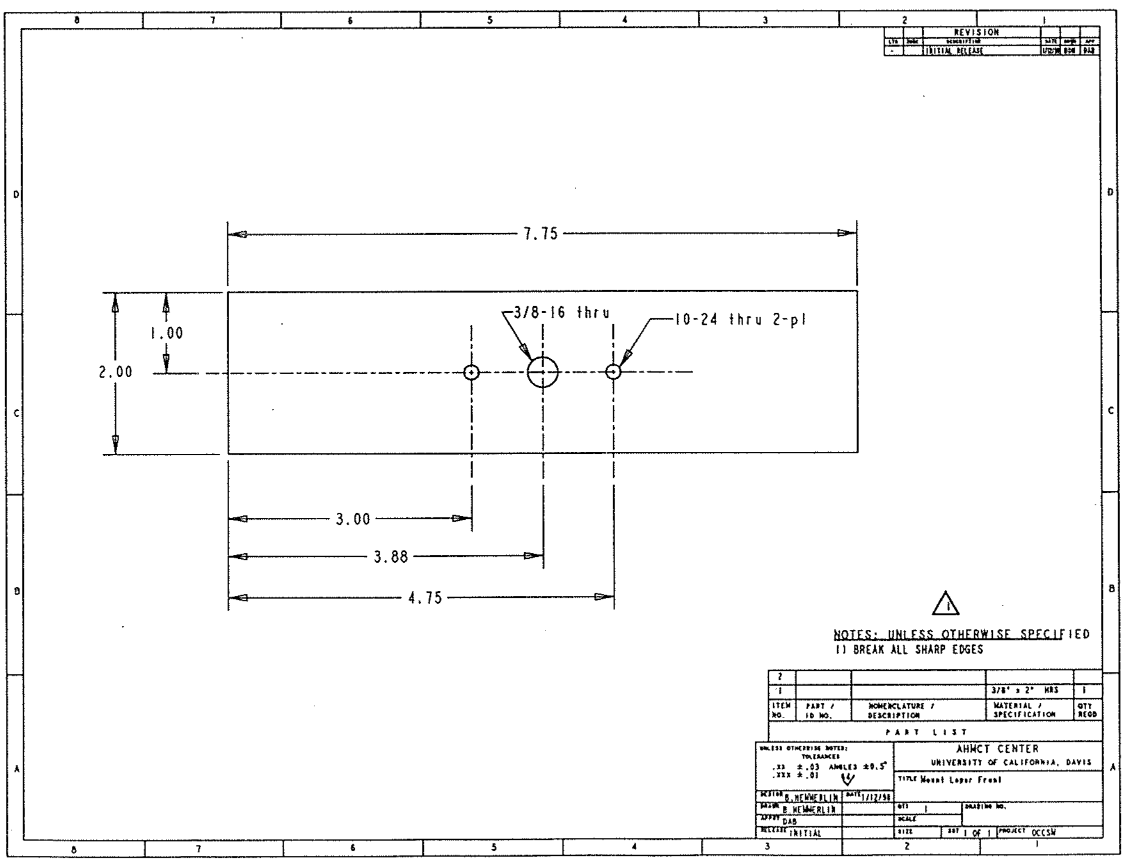

0.38
5.25
1.13
1.31
2.75
8.00
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA DAVIS
B.HEMMERLIN
7/24/97
INITIAL
LCSM2

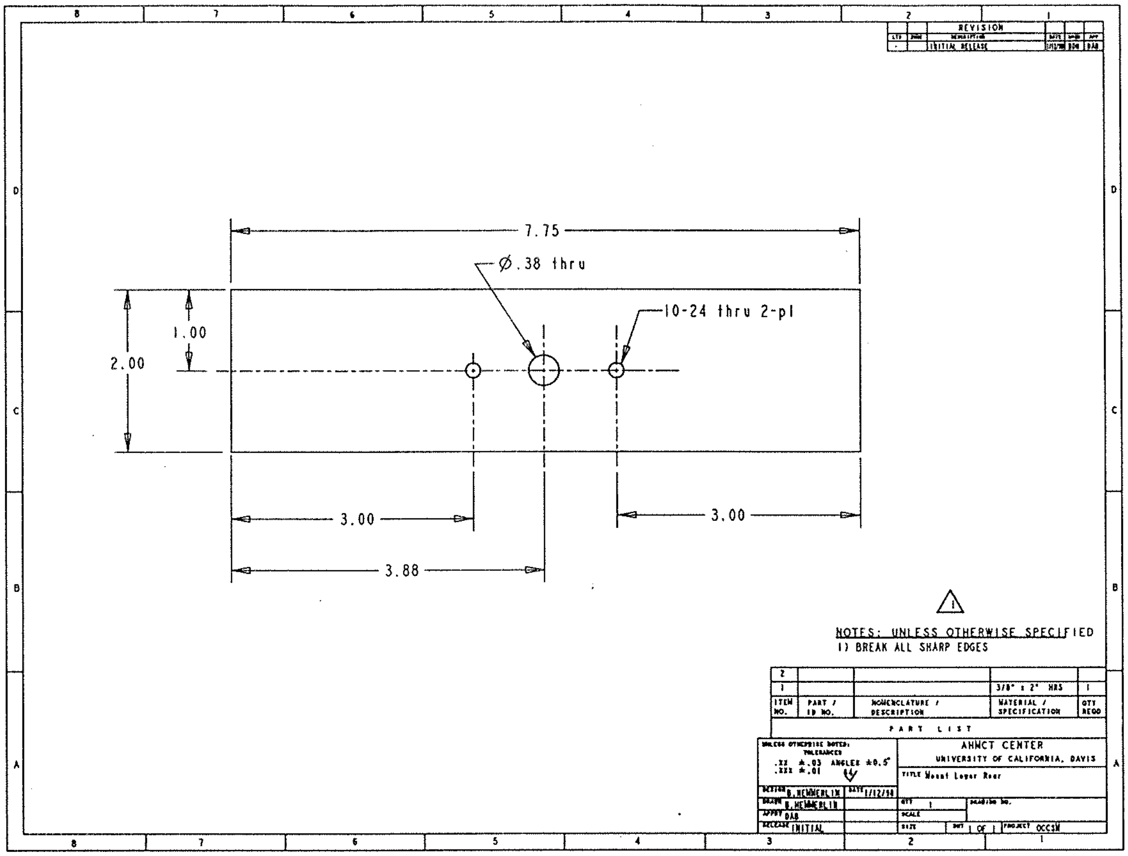

REVISION
INITIAL RELEASE
SEE DETAIL A
51.00
⌀ 0.38 THROUGH
⌀0.75 THRU
9.84
R 1.00
1.00
2.0
0.66
DETAIL A
0.50
THROUGH ONE WALL ONLY
3.00
6.34
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2x2x025 STEEL TUBE
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Swing Arm
B.HEMMERLIN
6/25/97
DAB
INITIAL
LCSM2

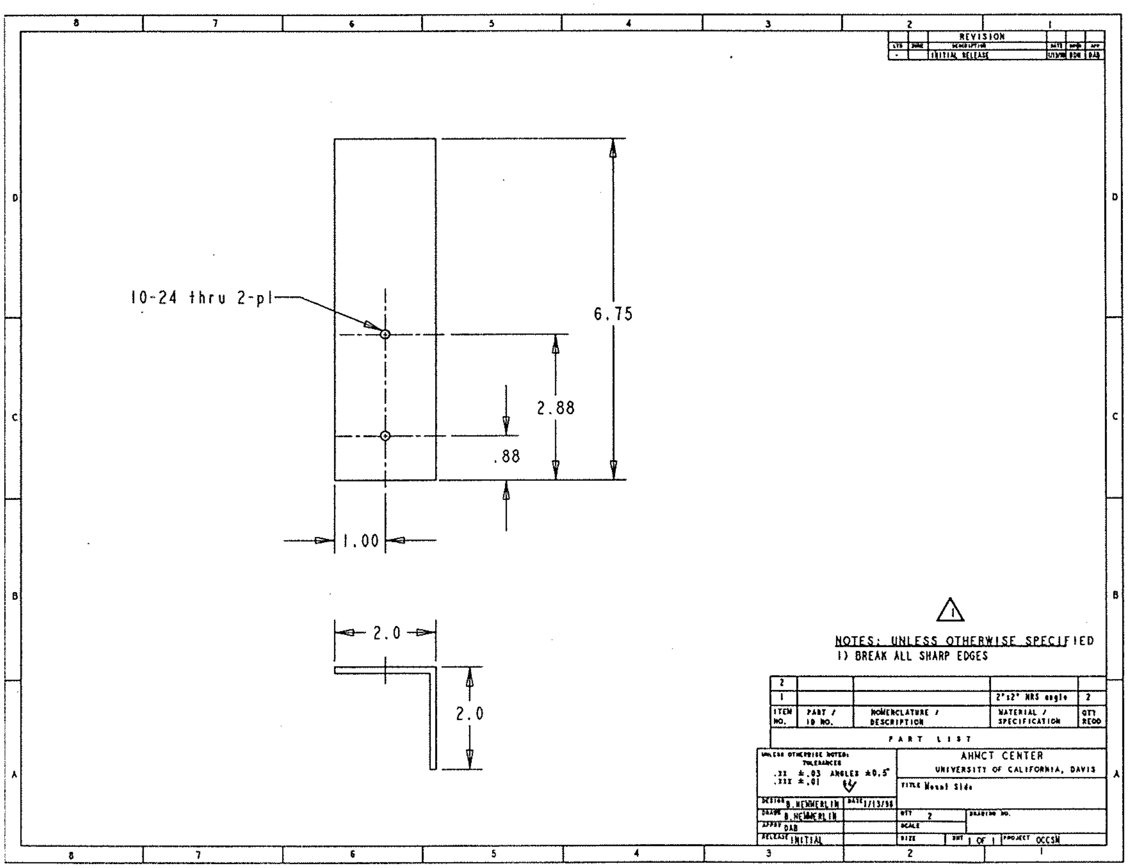
APPENDIX D: Part Drawings for the Camera Boom Extension Mechanism

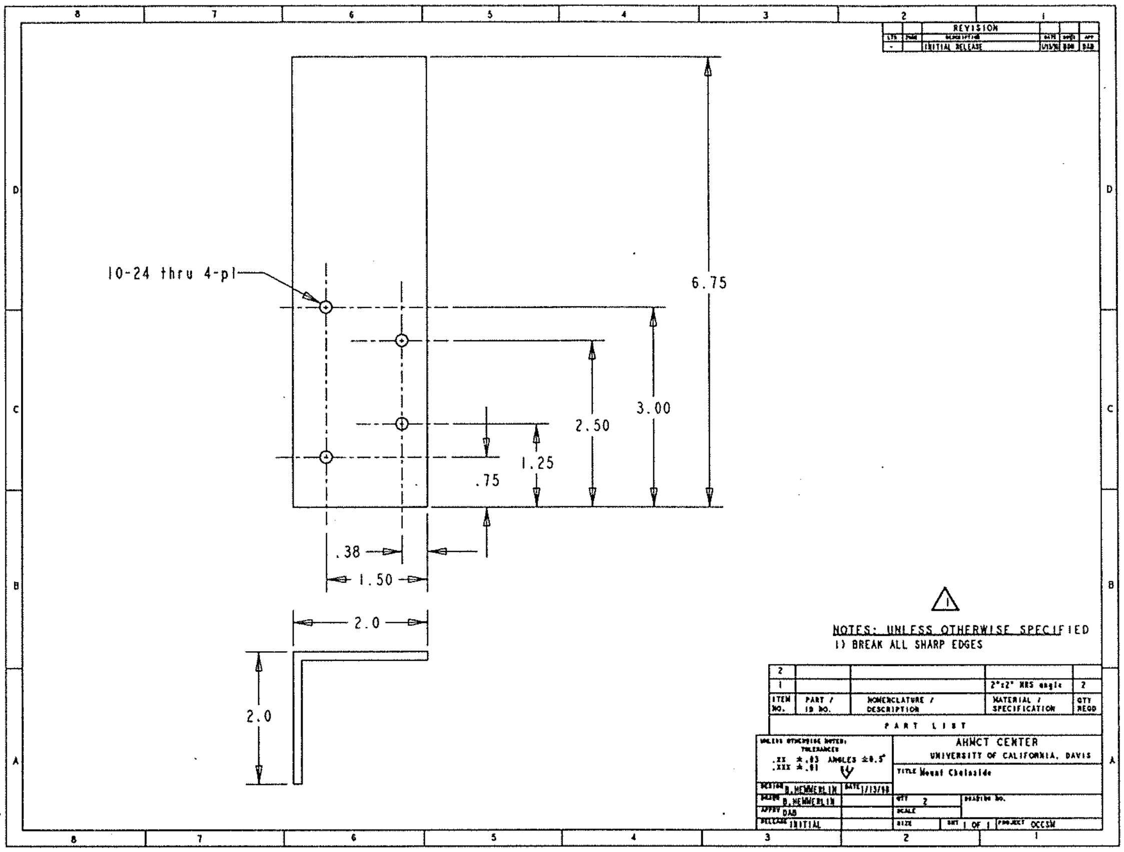
TABLE OF CONTENTS

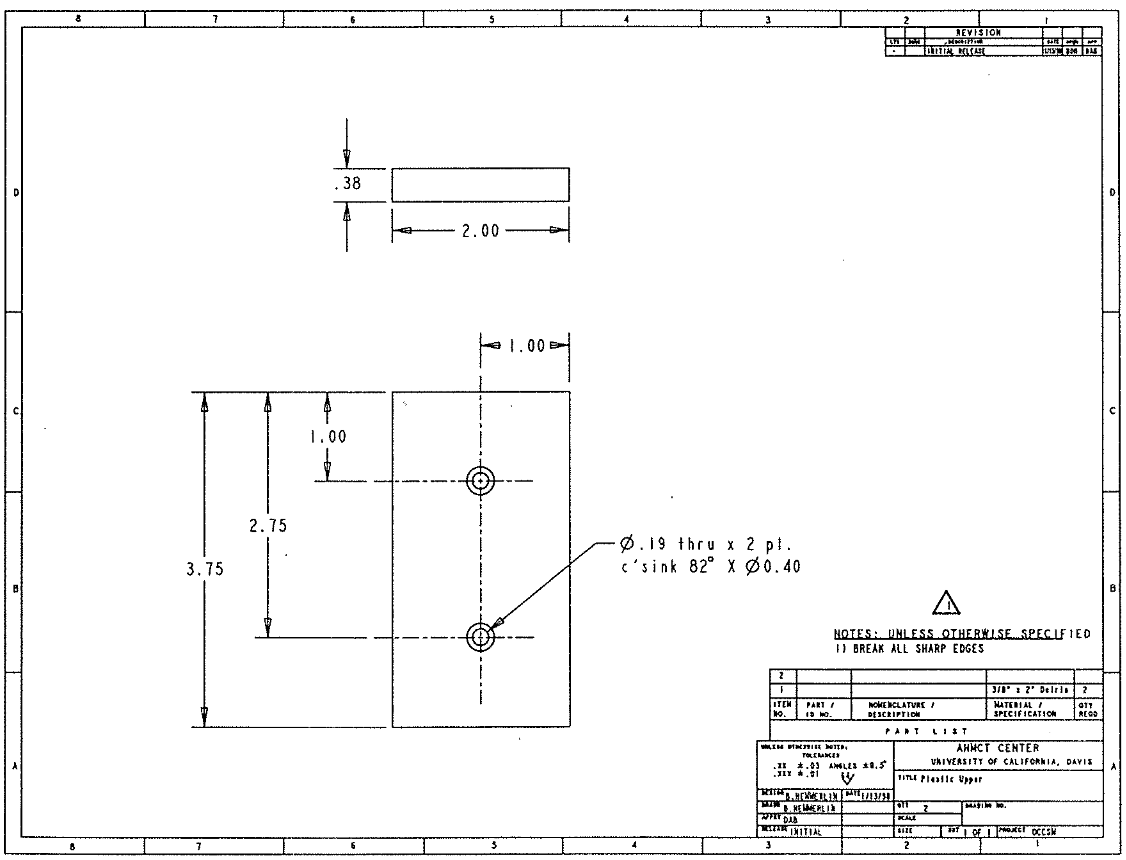

REVISION
INITIAL RELEASE
2.75
1.00
10-24 thru 2-pl
1.00
3.75
2.0
2.0
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2"x2" HRS angle
2
ITEM NO
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ± .03 ANGLES ±0.5°
.XXX ± .01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
TITLE Mount Upper
DESIGN B. NEMMERLIN
DRAWN B. NEMMERLIN
APPRV DAB
RELEASE INITIAL
QTY 2
DRAWING NO.
SCALE
SIZE
SHT 1 OF 1
PROJECT OCCSM

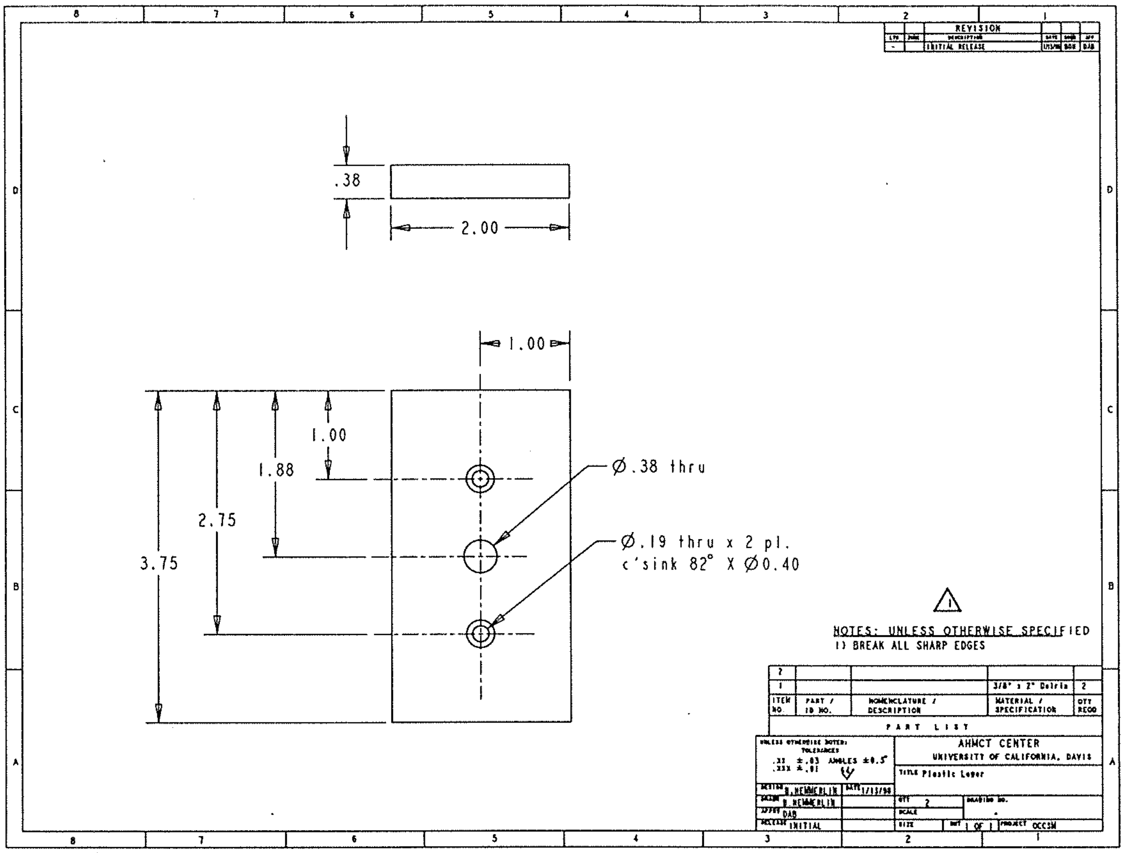

REVISION
INITIAL RELEASE
7.75
1.00
2.00
3/8-16 thru
10-24 thru 2-pl
3.00
3.88
4.75
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
3/8" x 2" HRS
1
ITEM NO.
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ± .03 ANGLES ±0.5°
.XXX ± .01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
TITLE Mount Laser Front
DESIGN B. NEMMERLIN
DATE 1/12/98
DRAWN B. NEMMERLIN
APPRV DAB
RELEASE INITIAL
QTY 1
DRAWING NO.
SCALE
SIZE
SHT 1 OF 1
PROJECT OCCSM

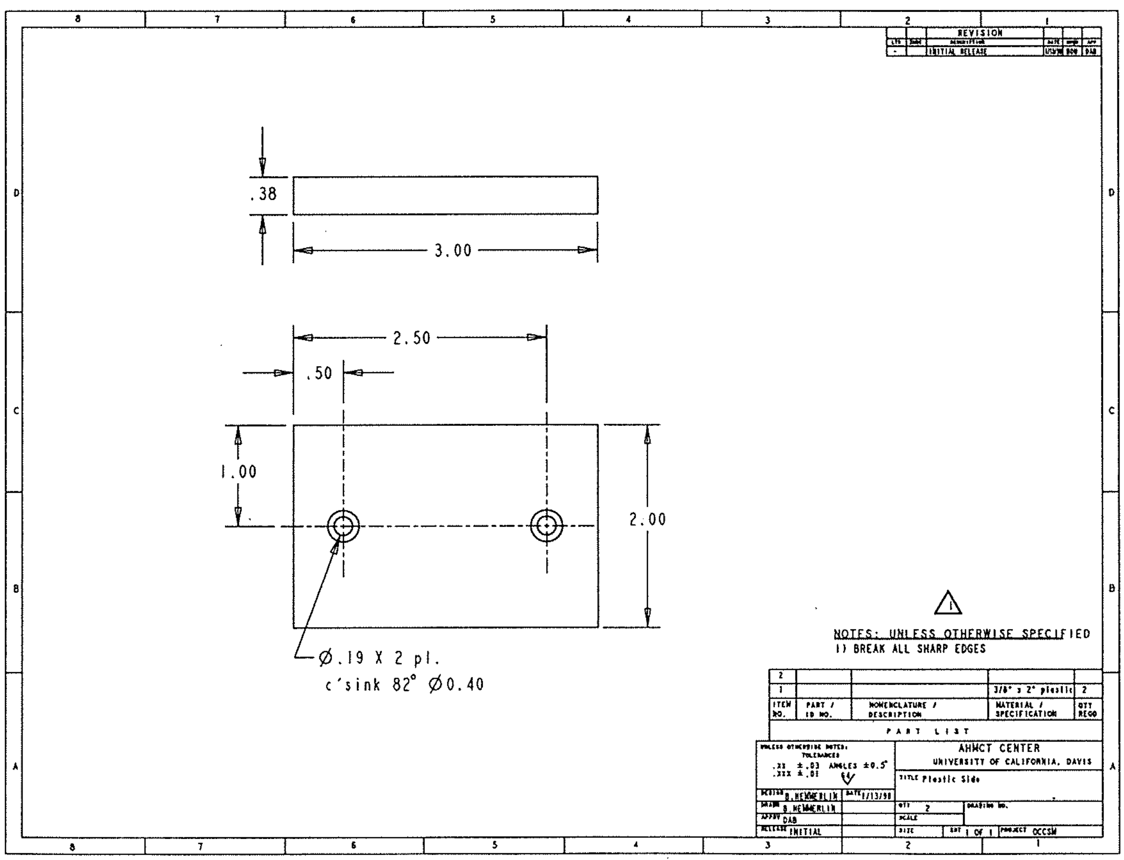

7.75
Ø.38 thru
10-24 thru 2-pl
1.00
2.00
3.00
3.00
3.88
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
3/8" x 2" HRS
ITEM NO.
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
.XX ±.03 ANGLES ±0.5°
.XXX ±.01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
TITLE Mount Lower Rear
DESIGN B. HEMMERLIN DATE 1/12/98
DRAWN B. HEMMERLIN
APPVD DAB
RELEASE INITIAL
QTY 1
SHT 1 OF 1
PROJECT OCCSM
REVISION
INITIAL RELEASE

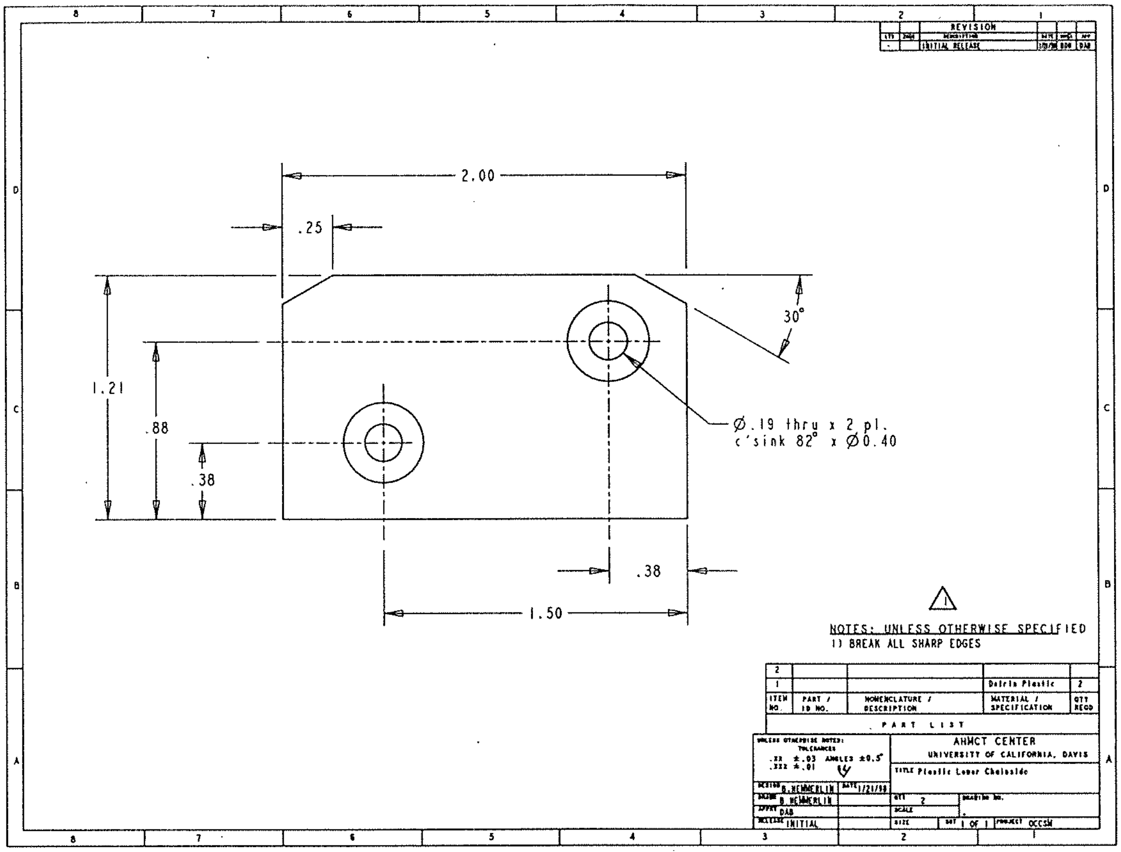

10-24 thru 2-pl
6.75
2.88
.88
1.00
2.0
2.0
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2"x2" HRS angle
ITEM NO.
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
.XX ±.03 ANGLES ±0.5°
.XXX ±.01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
TITLE Mount Side
DESIGN B. HEMMERLIN DATE 1/13/98
DRAWN B. HEMMERLIN
APPVD DAB
RELEASE INITIAL
QTY 2
SHT 1 OF 1
PROJECT OCCSM
REVISION
INITIAL RELEASE

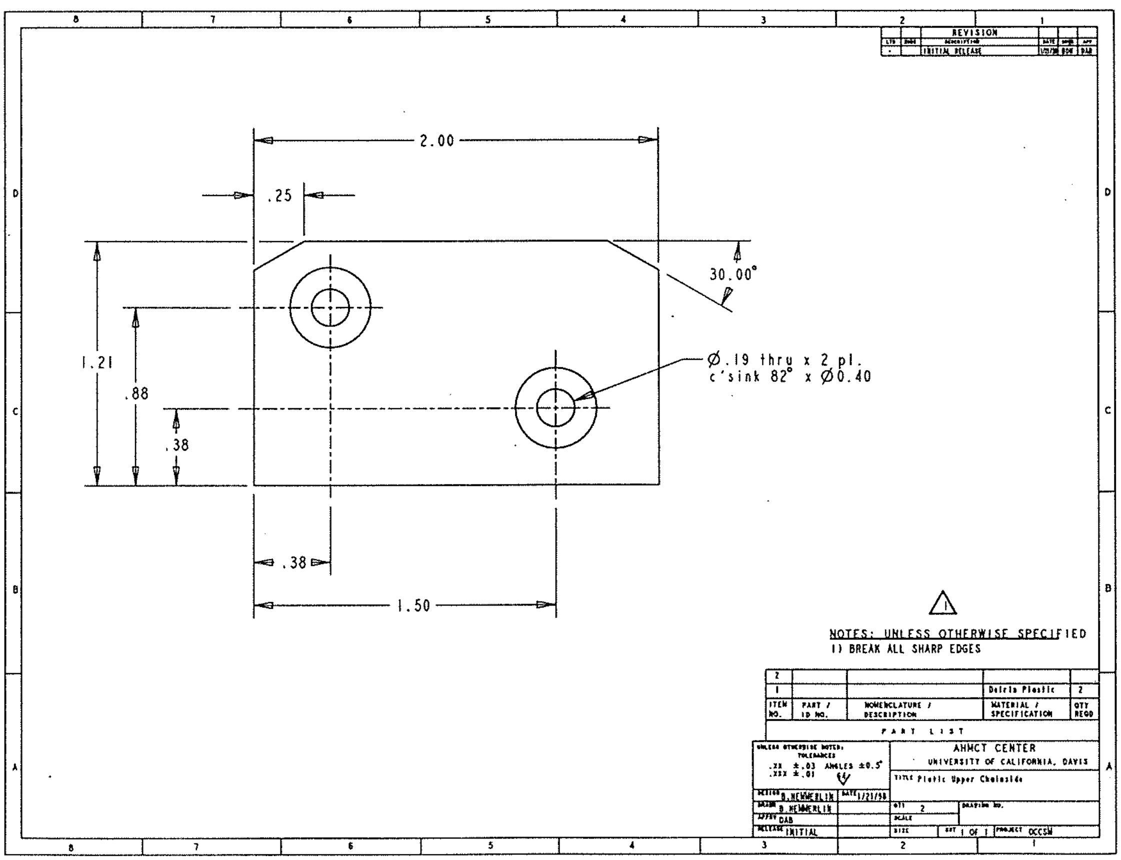

10-24 thru 4-pl
6.75
3.00
2.50
1.25
.75
.38
1.50
2.0
2.0
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2"x2" HRS angle
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
B. HEMMERLIN
DAB
INITIAL
OCCSM

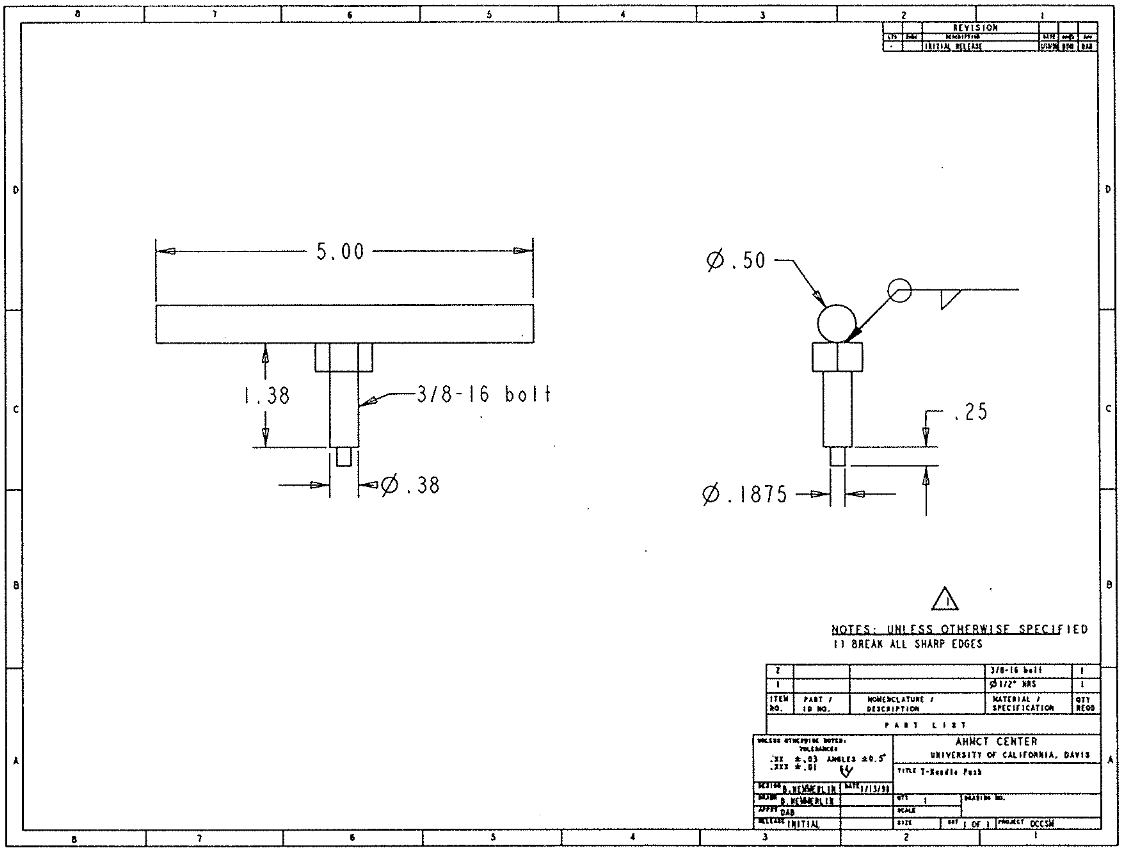

.38
2.00
1.00
1.00
2.75
3.75
Ø.19 thru x 2 pl.
c'sink 82° X Ø0.40
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Plastic Upper
B. HEMMERLIN
DAB
INITIAL
OCCSM


REVISION
INITIAL RELEASE
.38
2.00
1.00
1.00
1.88
2.75
3.75
Ø.38 thru
Ø.19 thru x 2 pl.
c'sink 82° X Ø0.40
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
3/8" x 2" Delrin
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Plastic Lever
B.HEMMERLIN
DAB
INITIAL
OCCSM

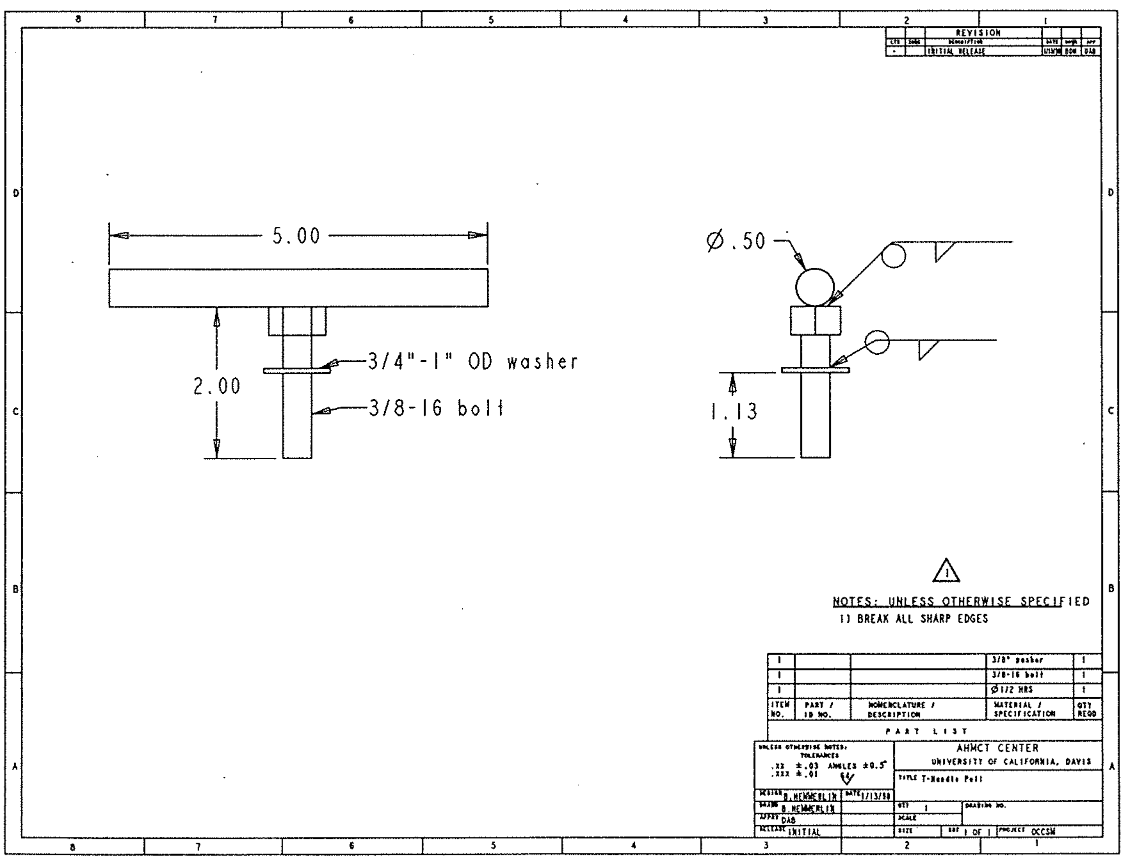

REVISION
INITIAL RELEASE
.38
3.00
2.50
.50
1.00
2.00
Ø.19 X 2 pl.
c'sink 82° Ø0.40
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
3/8" x 2" plastic
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Plastic Side
B.HEMMERLIN
DAB
INITIAL
OCCSM


2.00
.25
30°
1.21
.88
.38
Ø.19 thru x 2 pl.
c'sink 82° x Ø0.40
.38
1.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
REVISION
INITIAL RELEASE
Delrin Plastic
ITEM NO.
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
TOLERANCES
.XX ±.03 ANGLES ±0.5°
.XXX ±.01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
B. HEMMERLIN
1/21/98
DAB
INITIAL
1 OF 1
OCCSM

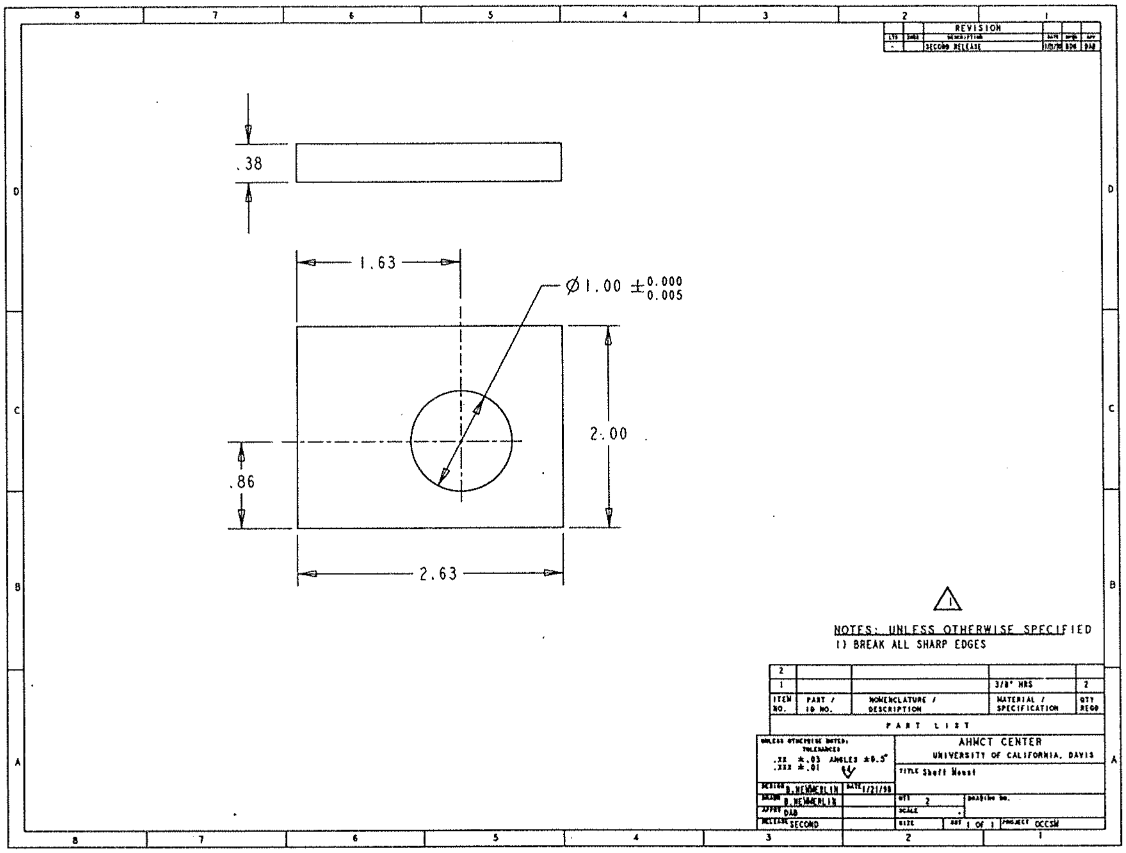

2.00
.25
30.00°
1.21
.88
.38
Ø.19 thru x 2 pl.
c'sink 82° x Ø0.40
.38
1.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
REVISION
INITIAL RELEASE
Delrin Plastic
ITEM NO.
PART / ID NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
TOLERANCES
.XX ±.03 ANGLES ±0.5°
.XXX ±.01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
B. HEMMERLIN
DAB
INITIAL
1 OF 1
OCCSM



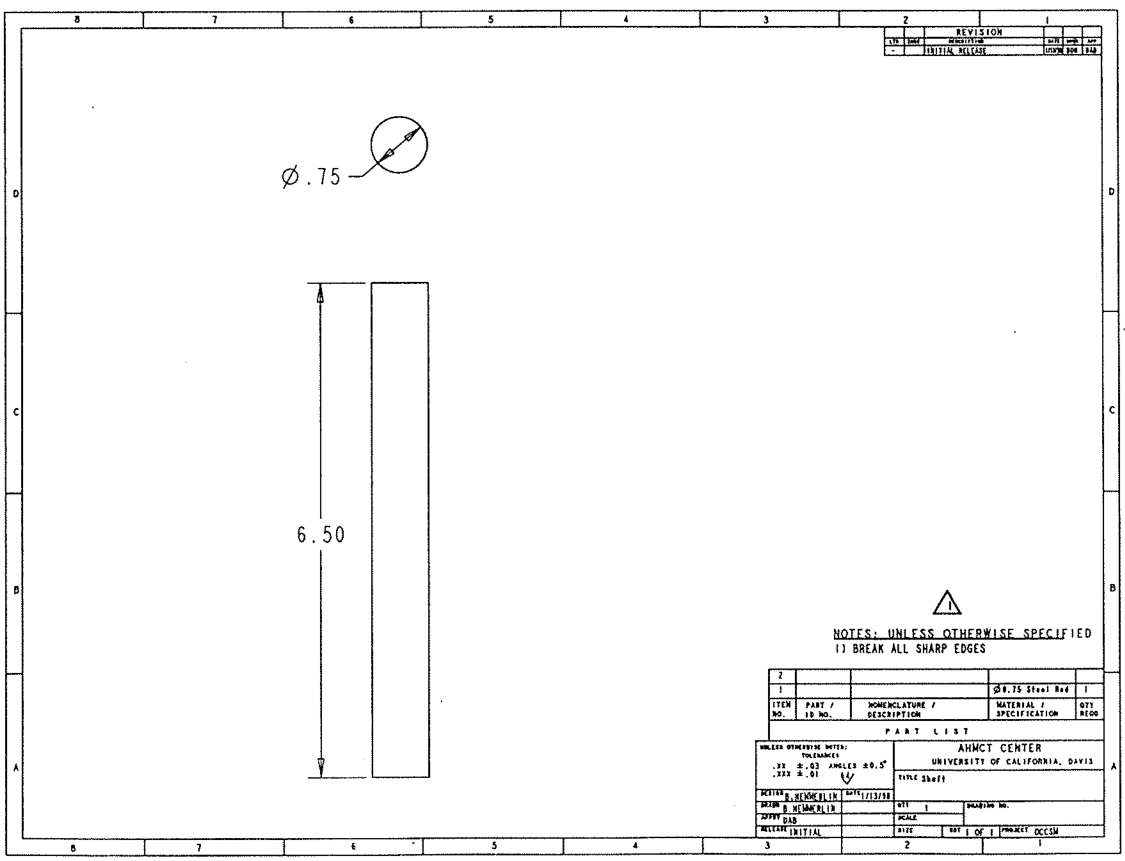
REVISION
INITIAL RELEASE
5.00
1.38
3/8-16 bolt
Ø.38
Ø.50
.25
Ø.1875
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2 3/8-16 bolt 1
1 Ø1/2" HRS 1
ITEM NO. PART / ID NO. NOMENCLATURE / DESCRIPTION MATERIAL / SPECIFICATION QTY REQD
PART LIST
TOLERANCES
.XX ±.03 ANGLES ±0.5°
.XXX ±.01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
B. HEMMERLIN
DAB
INITIAL
1 OF 1
DCCSM



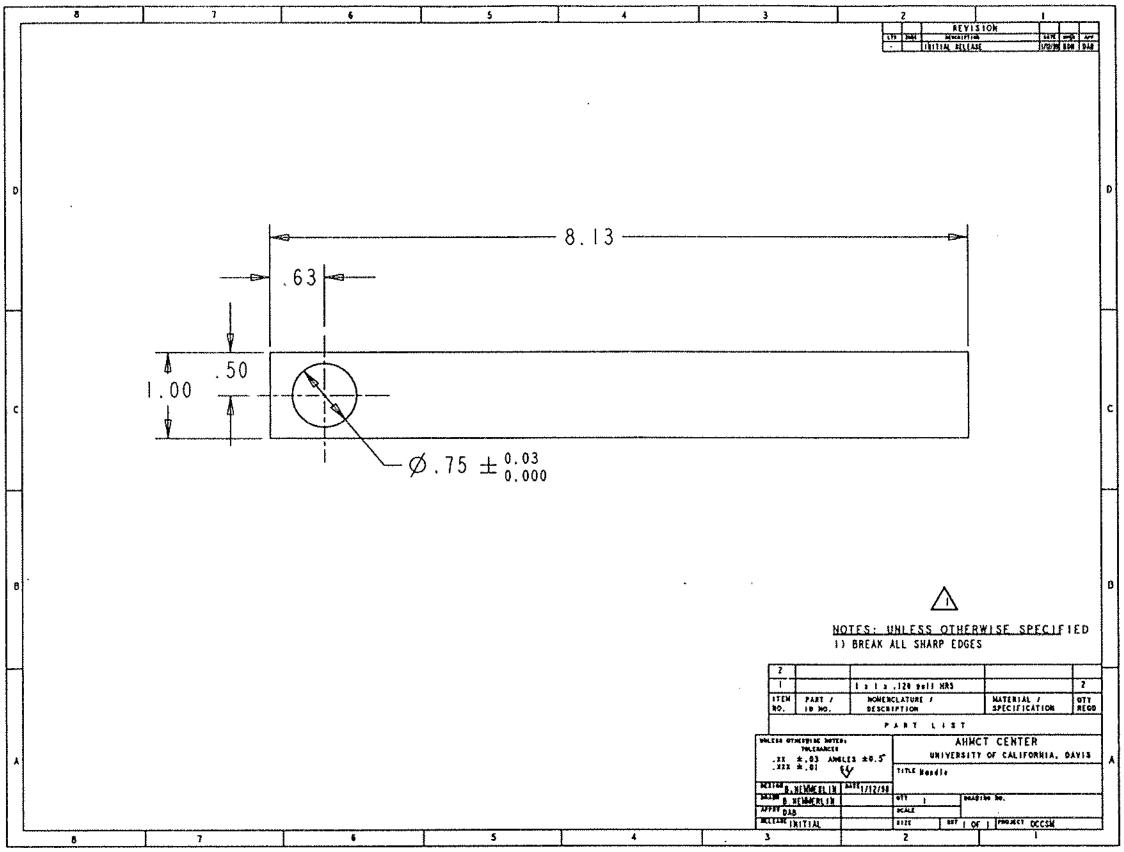
REVISION
INITIAL RELEASE
5.00
2.00
3/4"-1" OD washer
3/8-16 bolt
Ø.50
1.13
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 3/8" washer 1
1 3/8-16 bolt 1
1 Ø1/2 HRS 1
ITEM NO. PART / ID NO. NOMENCLATURE / DESCRIPTION MATERIAL / SPECIFICATION QTY REQD
PART LIST
TOLERANCES
.XX ±.03 ANGLES ±0.5°
.XXX ±.01
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
B. HEMMERLIN
DAB
INITIAL
1 OF 1
DCCSM

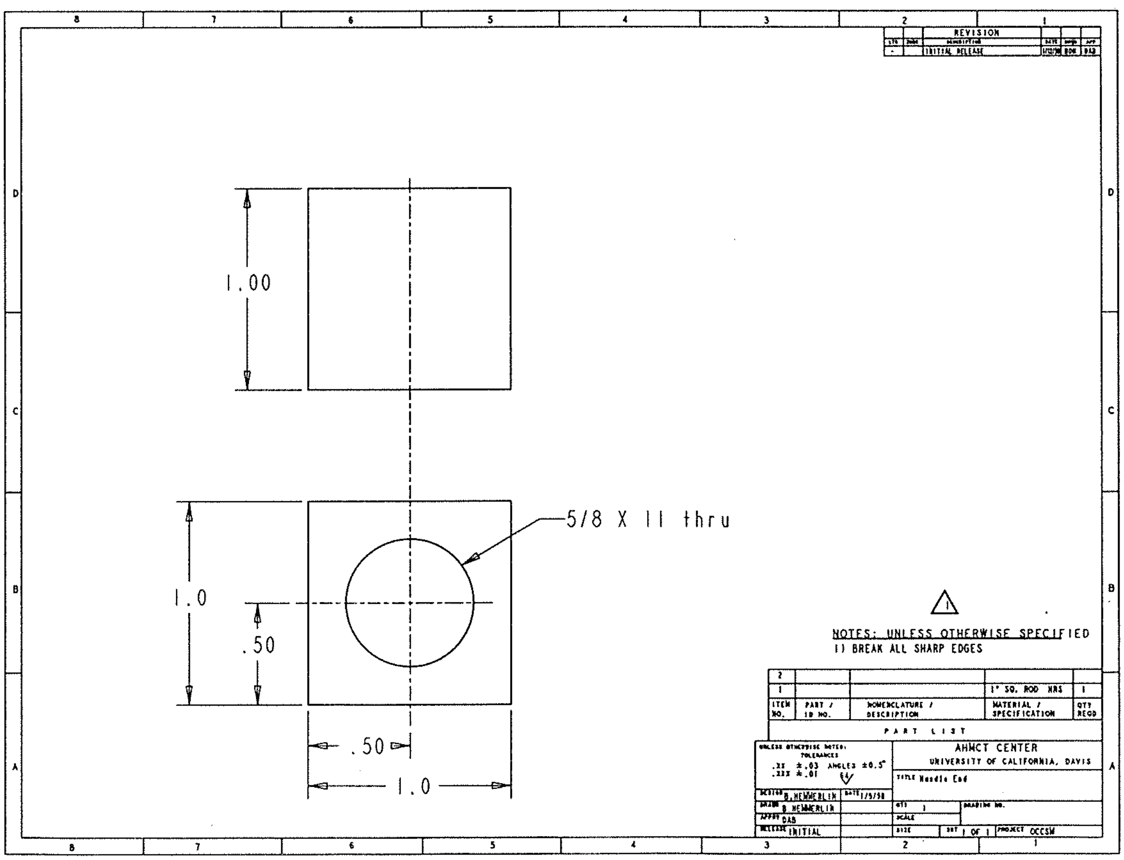


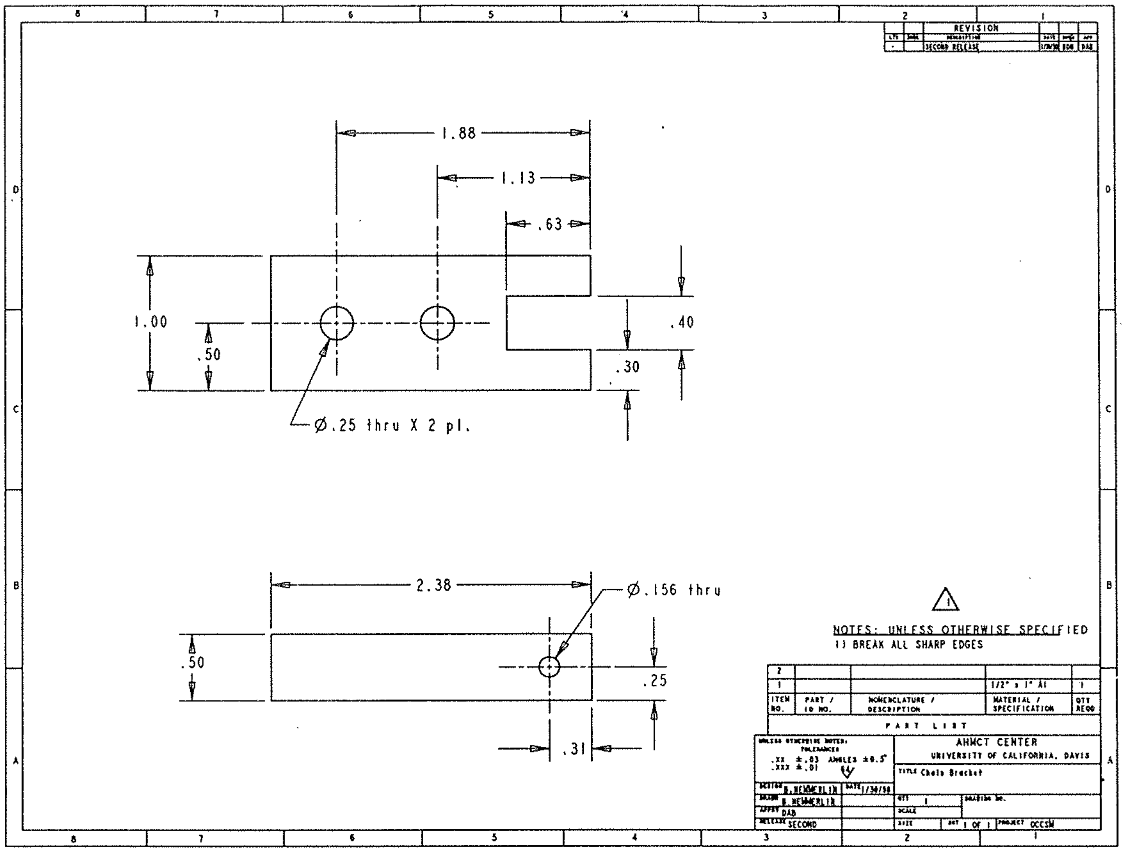
REVISION
SECOND RELEASE
.38
1.63
Ø1.00 ±0.000 0.005
2.00
.86
2.63
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
3/8" HRS
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Shaft Mount

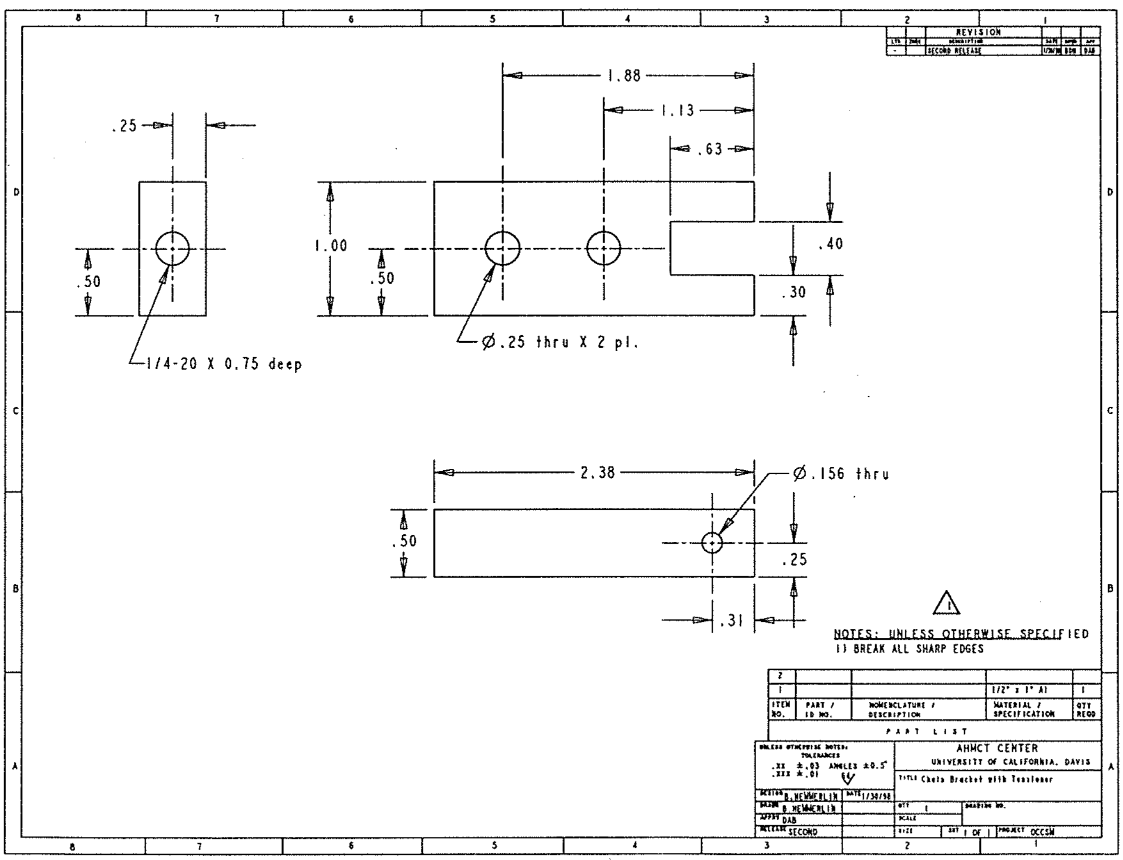


REVISION
INITIAL RELEASE
Ø.75
6.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
Ø0.75 Steel Rod
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
Shaft


8.13
.63
.50
1.00
Ø.75 ± 0.03 / 0.000
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
TITLE Handle
B. NEMMERLIN
1/12/98
DAB
INITIAL
1 OF 1
OCCSM


1.00
5/8 X 11 thru
1.0
.50
.50
1.0
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1" SQ. ROD HRS
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS
TITLE Handle End
B. NEMMERLIN
DAB
INITIAL
1 OF 1
OCCSM


REVISION
SECOND RELEASE
1.88
1.13
.63
1.00
.50
.40
.30
Ø.25 thru X 2 pl.
2.38
Ø.156 thru
.50
.25
.31
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS


REVISION
SECOND RELEASE
.25
.50
1/4-20 X 0.75 deep
1.88
1.13
.63
1.00
.50
.40
.30
Ø.25 thru X 2 pl.
2.38
Ø.156 thru
.50
.25
.31
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
PART LIST
AHMCT CENTER
UNIVERSITY OF CALIFORNIA, DAVIS

APPENDIX E: Part Drawings for the Camera Boom Braces

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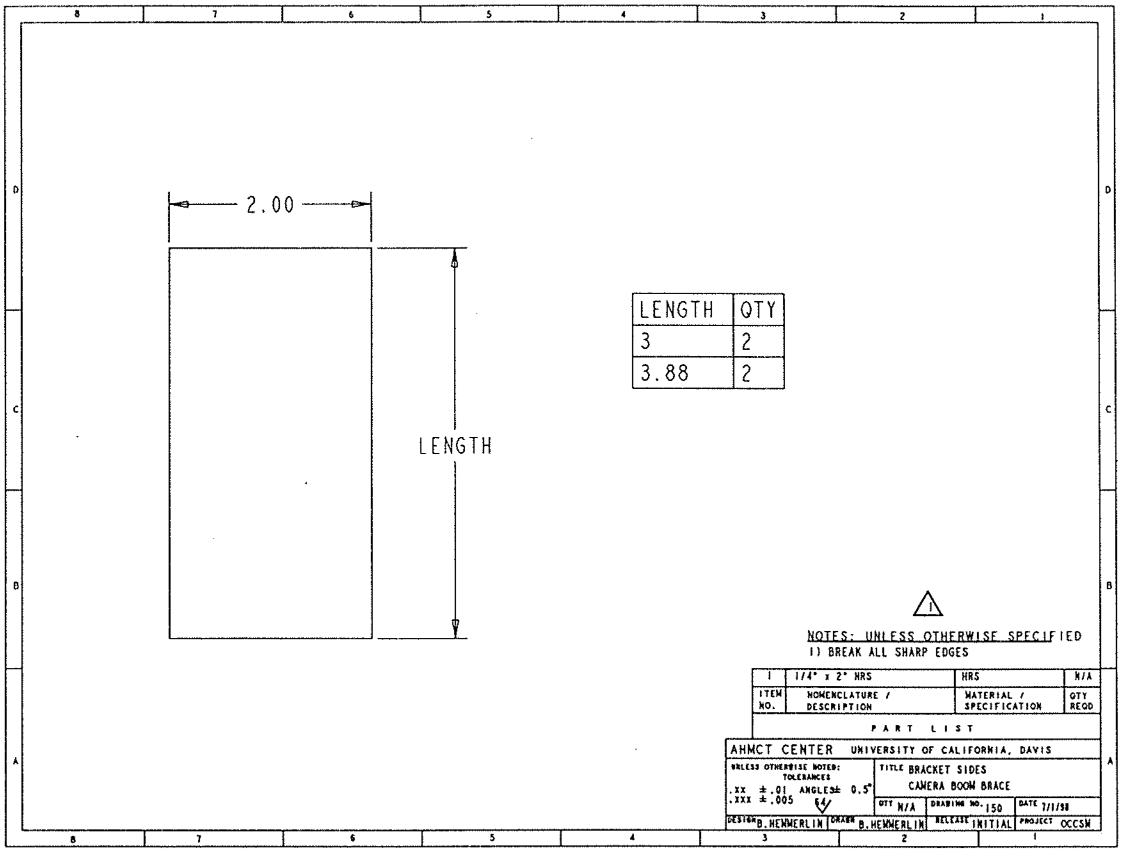
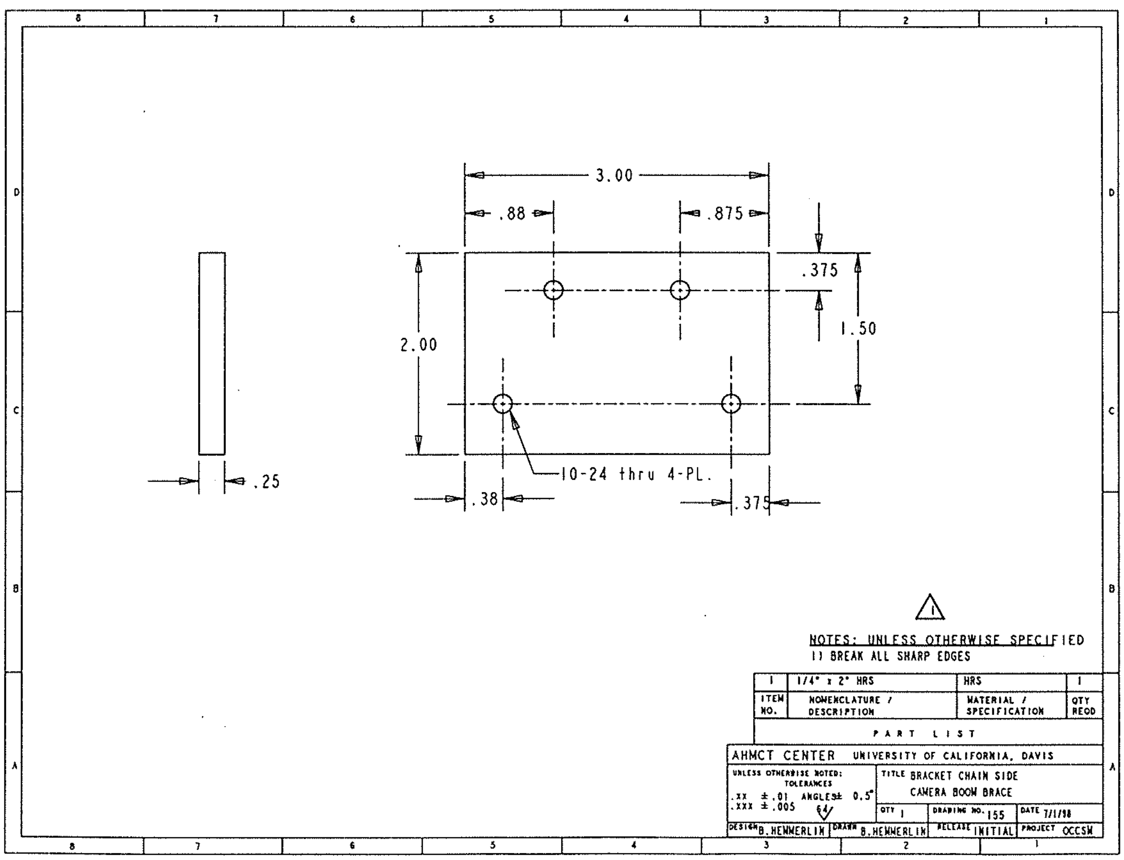
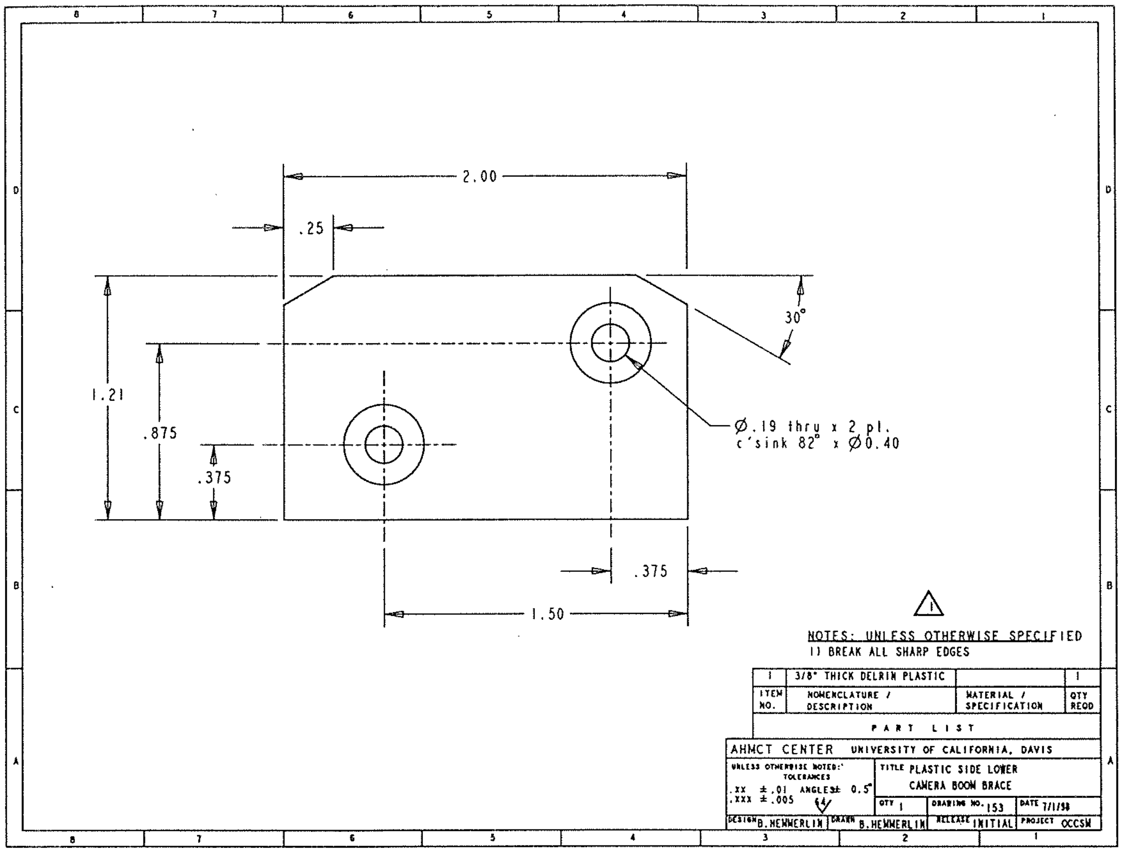
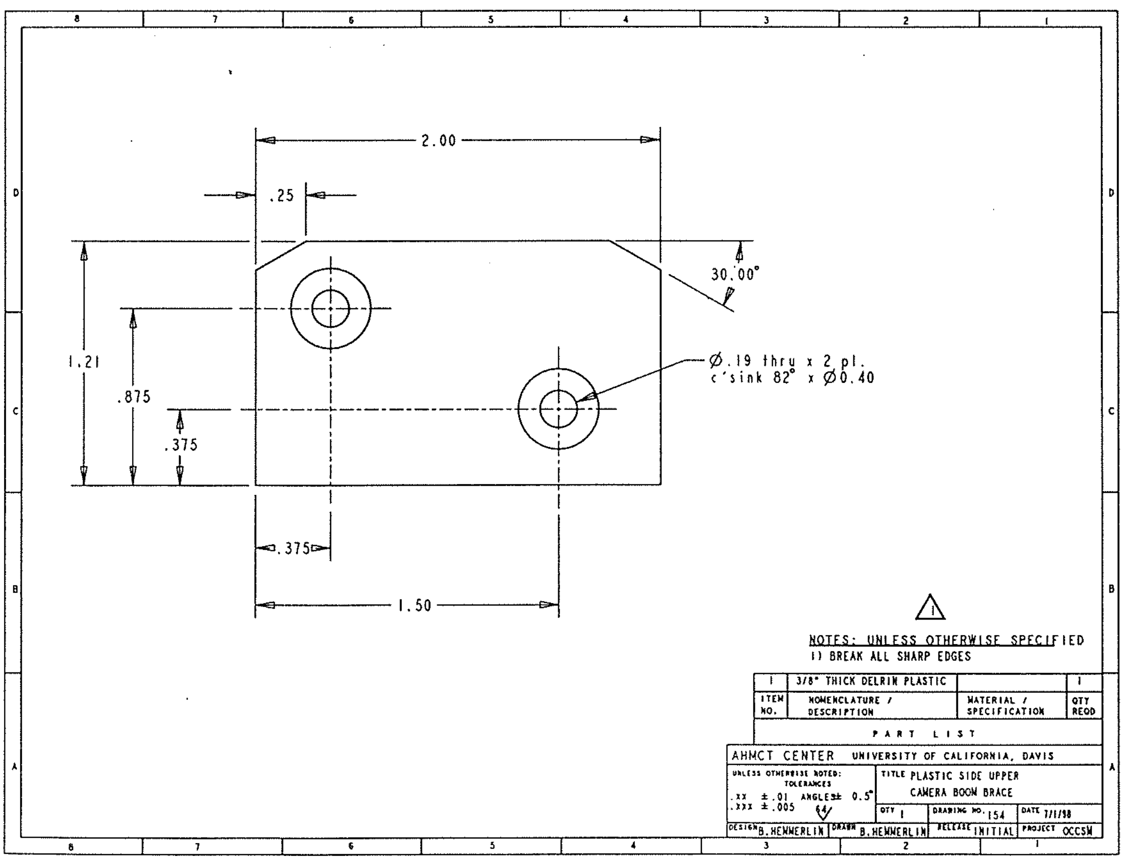

2.00
LENGTH
LENGTH | QTY
3 | 2
3.88 | 2
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1/4" x 2" HRS | HRS | N/A
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
TITLE BRACKET SIDES
CAMERA BOOM BRACE
QTY N/A
DRAWING NO. 150
DATE 7/1/98
DESIGN B.HENNERLIN
DRAWN B.HENNERLIN
RELEASE INITIAL
PROJECT OCCSM

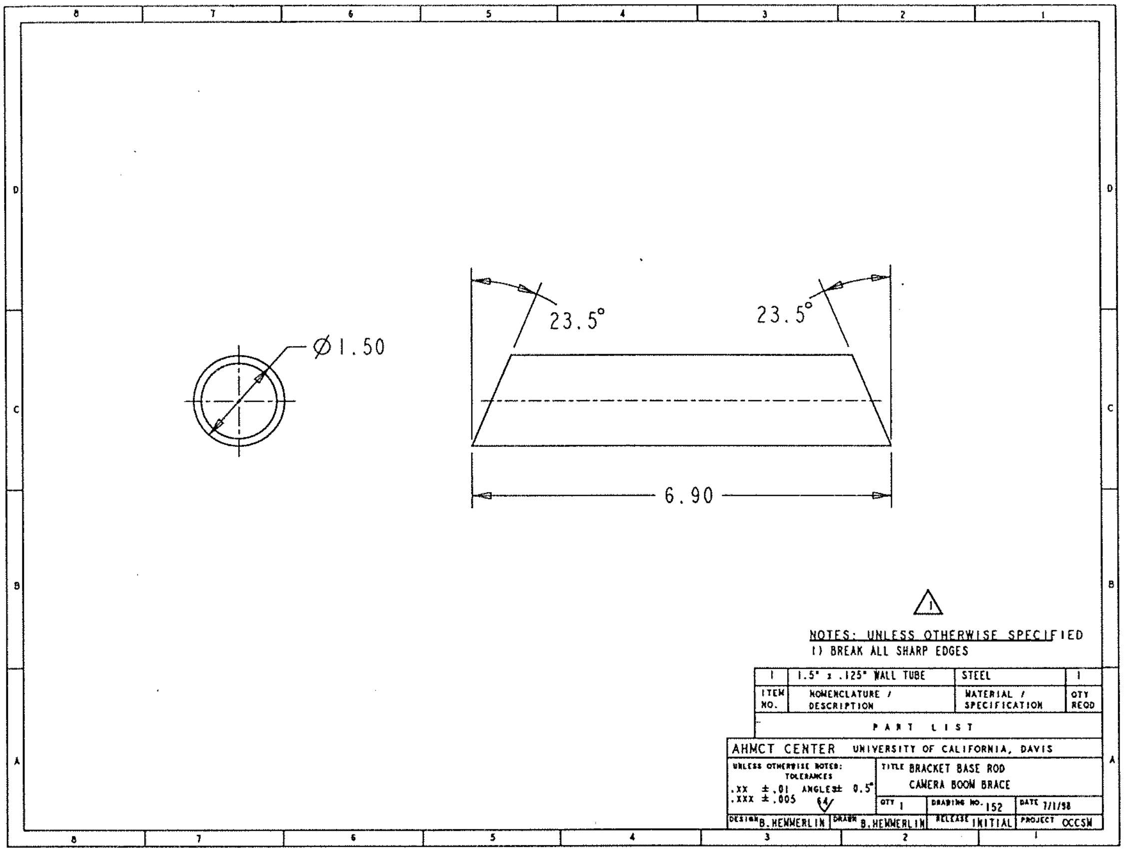

3.00
.88
.875
.375
1.50
2.00
.25
10-24 thru 4-PL.
.38
.375
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1/4" x 2" HRS | HRS | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
TITLE BRACKET CHAIN SIDE
CAMERA BOOM BRACE
QTY 1
DRAWING NO. 155
DATE 7/1/98
DESIGN B.HENNERLIN
DRAWN B.HENNERLIN
RELEASE INITIAL
PROJECT OCCSM



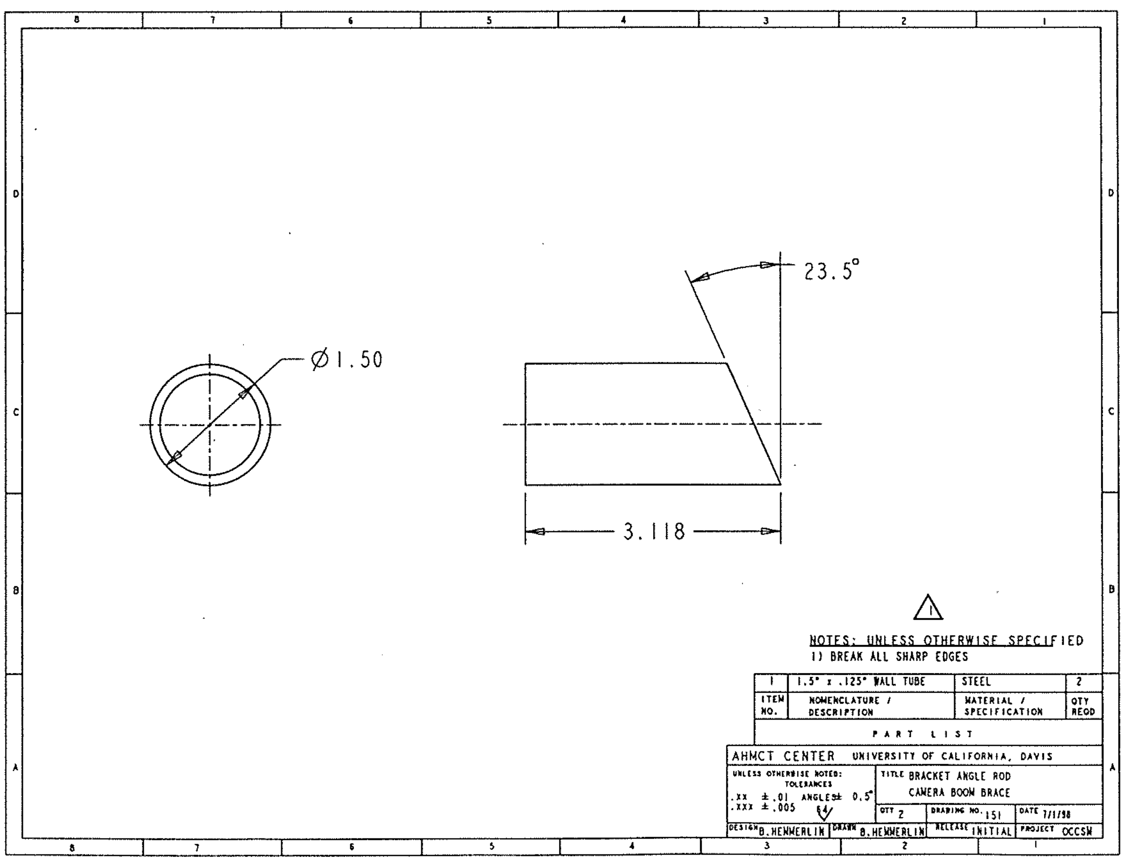
2.00
.25
30°
1.21
.875
.375
Ø.19 thru x 2 pl.
c'sink 82° x Ø0.40
.375
1.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 3/8" THICK DELRIN PLASTIC 1
ITEM NO. NOMENCLATURE / DESCRIPTION MATERIAL / SPECIFICATION QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE PLASTIC SIDE LOWER CAMERA BOOM BRACE
QTY 1 DRAWING NO. 153 DATE 7/1/98
DESIGN B. HEMMERLIN DRAWN B. HEMMERLIN RELEASE INITIAL PROJECT OCCSM



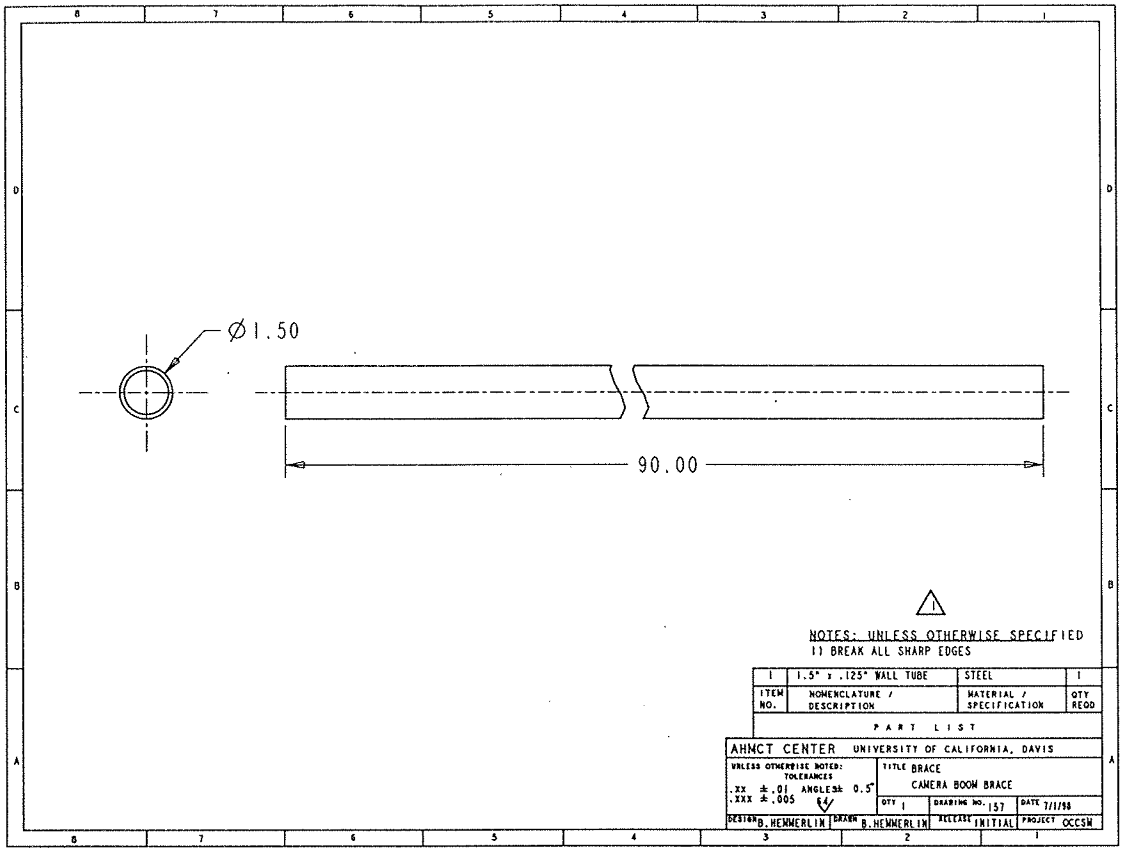
2.00
.25
30.00°
1.21
.875
.375
Ø.19 thru x 2 pl.
c'sink 82° x Ø0.40
.375
1.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 3/8" THICK DELRIN PLASTIC 1
ITEM NO. NOMENCLATURE / DESCRIPTION MATERIAL / SPECIFICATION QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE PLASTIC SIDE UPPER CAMERA BOOM BRACE
QTY 1 DRAWING NO. 154 DATE 7/1/98
DESIGN B. HEMMERLIN DRAWN B. HEMMERLIN RELEASE INITIAL PROJECT OCCSM



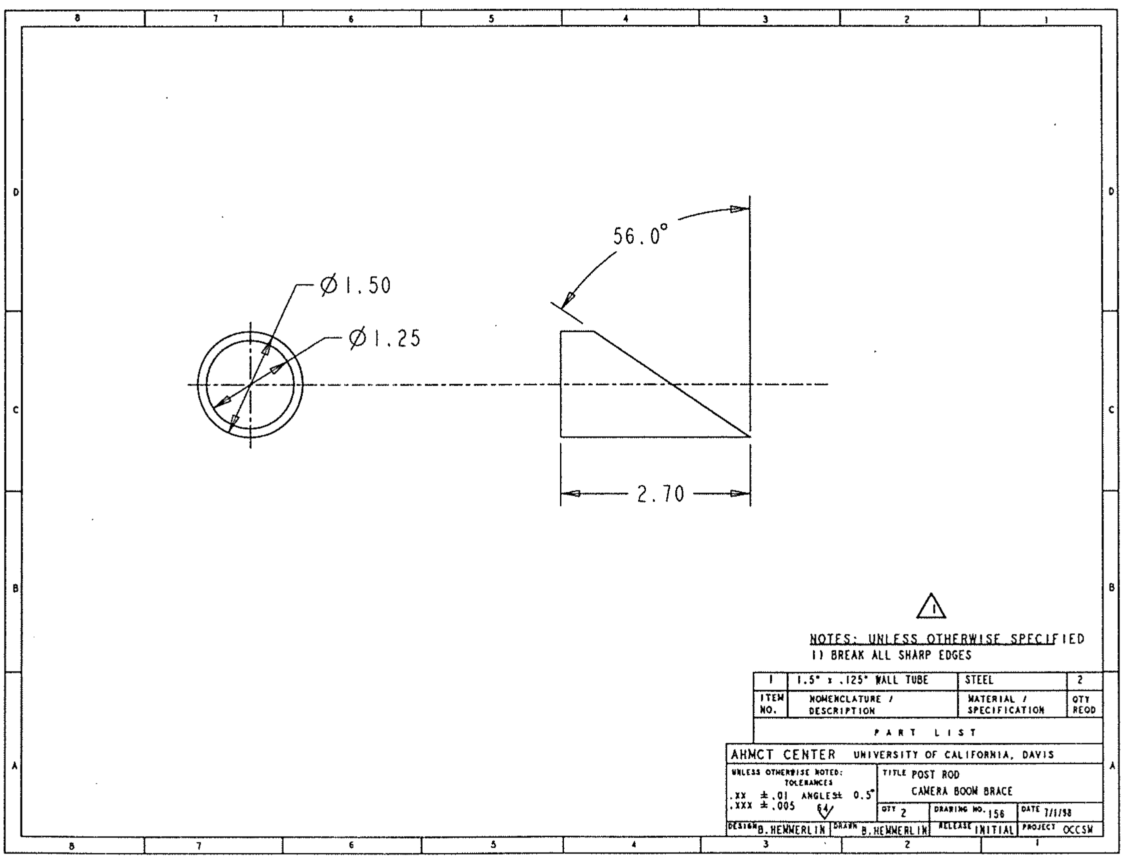
Ø1.50
23.5°
23.5°
6.90
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1.5" x .125" WALL TUBE | STEEL | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE BRACKET BASE ROD CAMERA BOOM BRACE
QTY 1 DRAWING NO. 152 DATE 7/1/98
DESIGN B. HEMMERLIN DRAWN B. HEMMERLIN RELEASE INITIAL PROJECT OCCSM



Ø1.50
23.5°
3.118
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1.5" x .125" WALL TUBE | STEEL | 2
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE BRACKET ANGLE ROD CAMERA BOOM BRACE
QTY 2 DRAWING NO. 151 DATE 7/1/98
DESIGN B. HEMMERLIN DRAWN B. HEMMERLIN RELEASE INITIAL PROJECT OCCSM



Ø1.50
90.00
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1.5" x .125" WALL TUBE | STEEL | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE BRACE
CAMERA BOOM BRACE
QTY 1
DRAWING NO. 157
DATE 7/1/98
DESIGN B. HEMMERLIN
DRAWN B. HEMMERLIN
RELEASE INITIAL
PROJECT OCCSW



56.0°
Ø1.50
Ø1.25
2.70
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1.5" x .125" WALL TUBE | STEEL | 2
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE POST ROD
CAMERA BOOM BRACE
QTY 2
DRAWING NO. 156
DATE 7/1/98
DESIGN B. HEMMERLIN
DRAWN B. HEMMERLIN
RELEASE INITIAL
PROJECT OCCSW

APPENDIX F: Part Drawings for the Camera Housing

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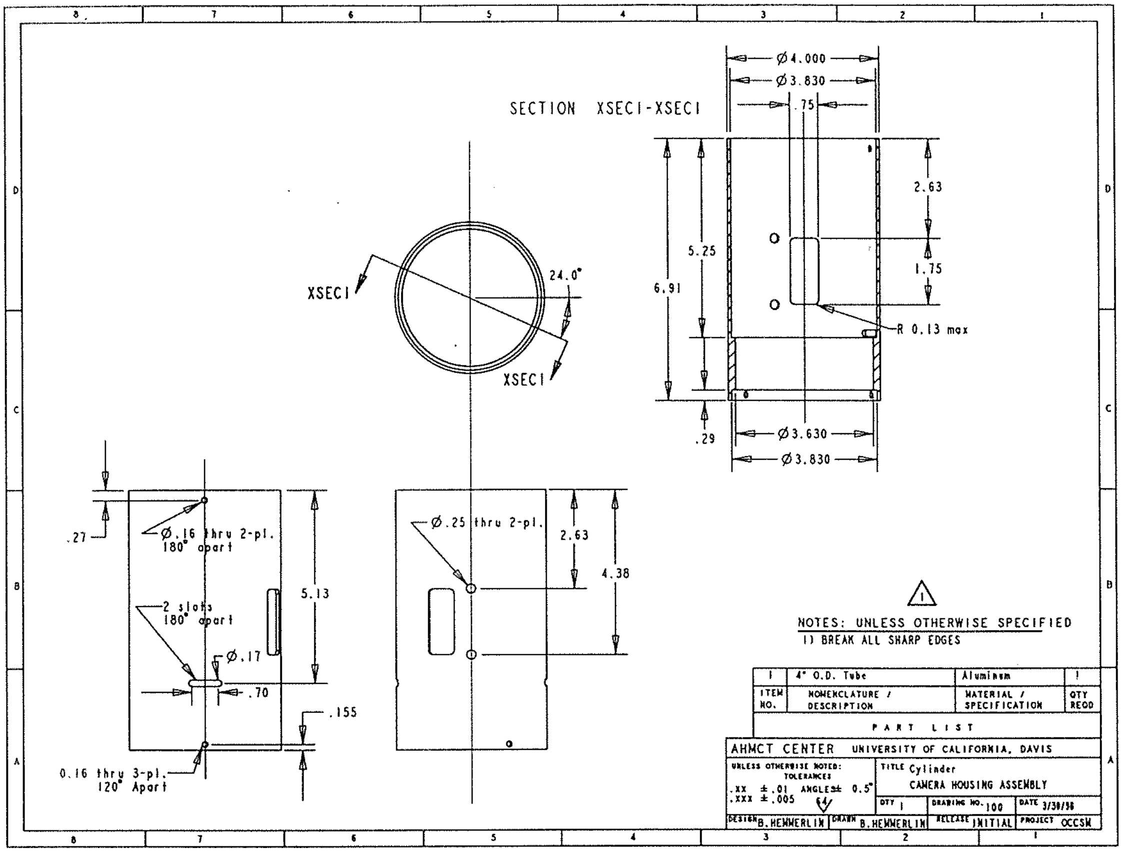
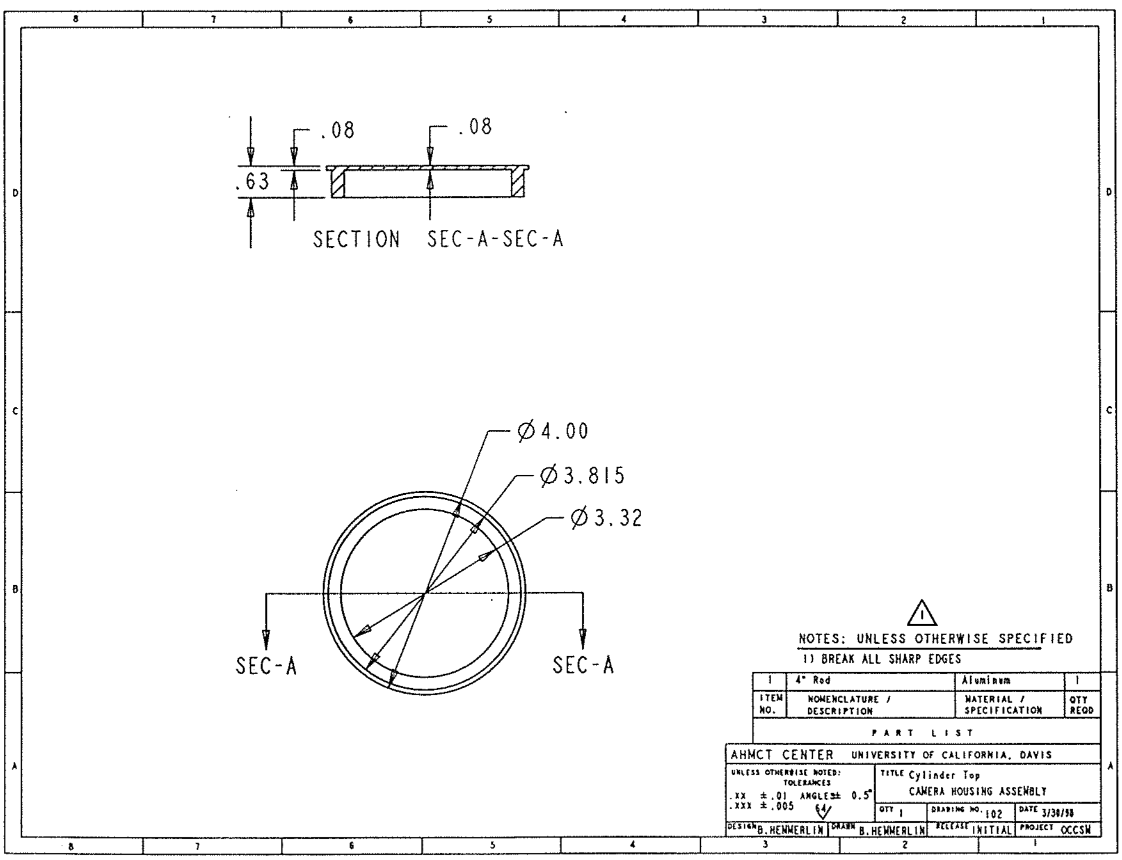
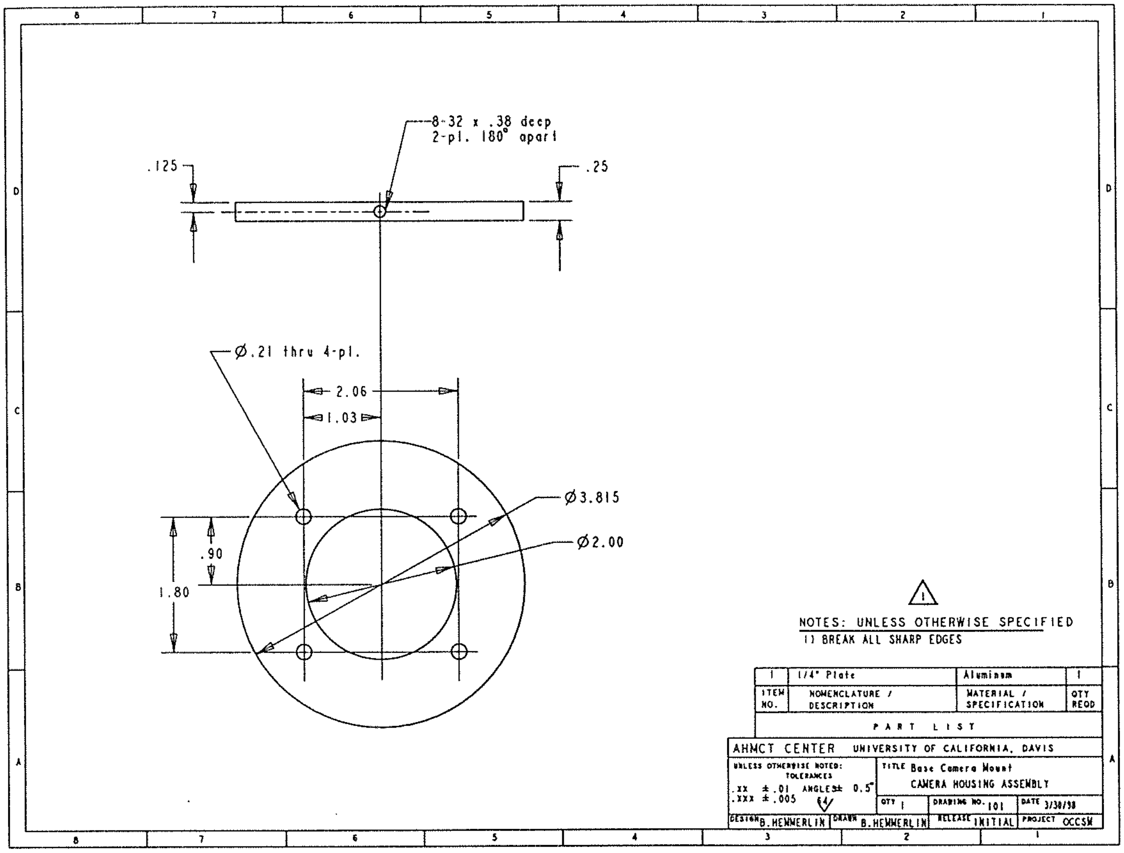
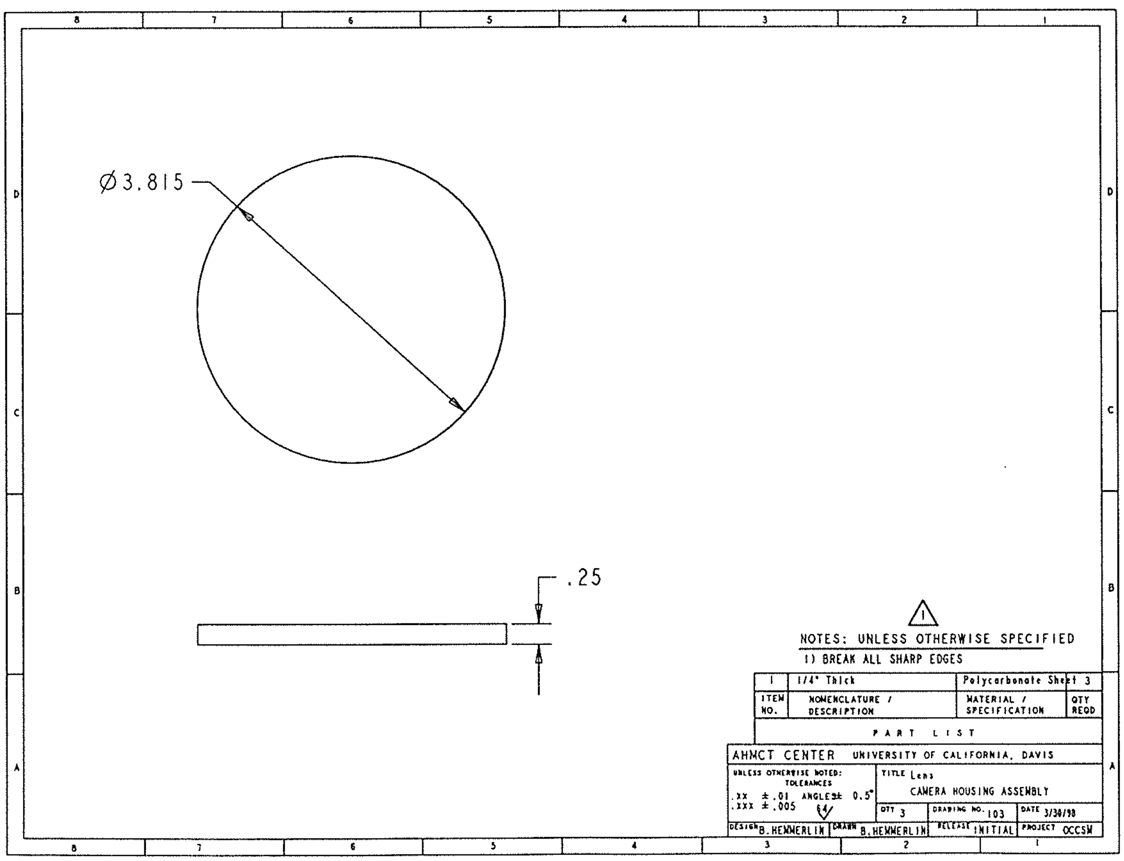

SECTION XSEC1-XSEC1
Ø4.000
Ø3.830
.75
2.63
1.75
5.25
6.91
R 0.13 max
.29
Ø3.630
Ø3.830
XSEC1
24.0°
XSEC1
.27
Ø.16 thru 2-pl. 180° apart
2 slots 180° apart
Ø.17
.70
5.13
.155
0.16 thru 3-pl. 120° Apart
Ø.25 thru 2-pl.
2.63
4.38
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 4" O.D. Tube | Aluminum | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLE± 0.5° .XXX ±.005
TITLE Cylinder CAMERA HOUSING ASSEMBLY
QTY 1 DRAWING NO. 100 DATE 3/30/98
DESIGN B. HEMMERLIN DRAWN B. HEMMERLIN RELEASE INITIAL PROJECT OCCSW

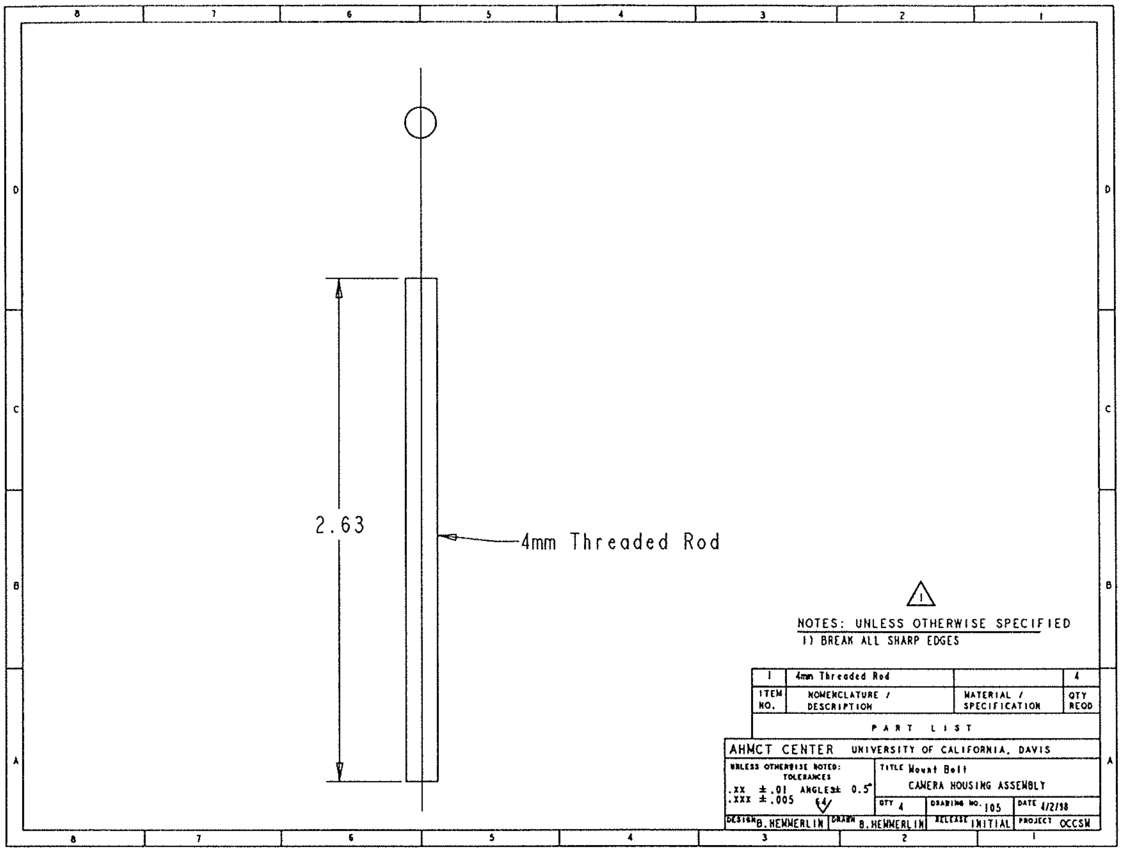

.08
.08
.63
SECTION SEC-A-SEC-A
Ø4.00
Ø3.815
Ø3.32
SEC-A
SEC-A
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 4" Rod | Aluminum | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLE± 0.5° .XXX ±.005
TITLE Cylinder Top CAMERA HOUSING ASSEMBLY
QTY 1 DRAWING NO. 102 DATE 3/30/98
DESIGN B. HEMMERLIN DRAWN B. HEMMERLIN RELEASE INITIAL PROJECT OCCSW

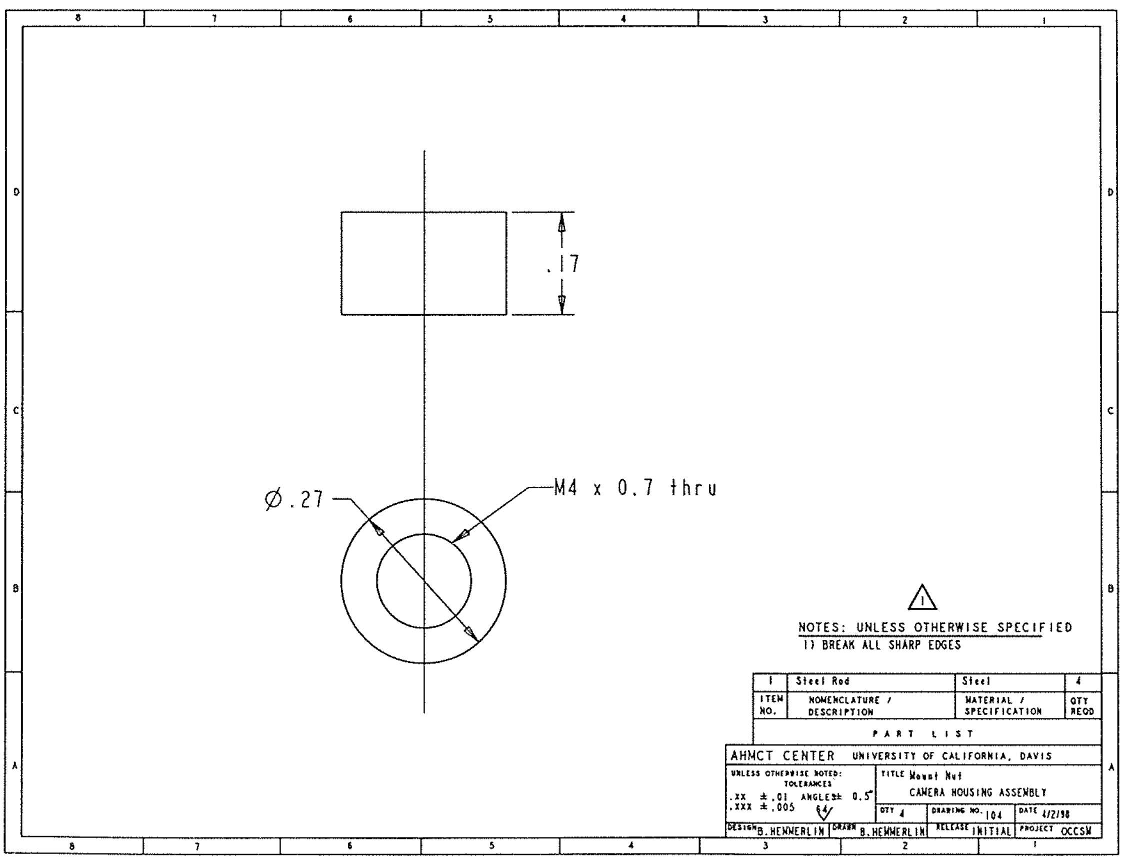


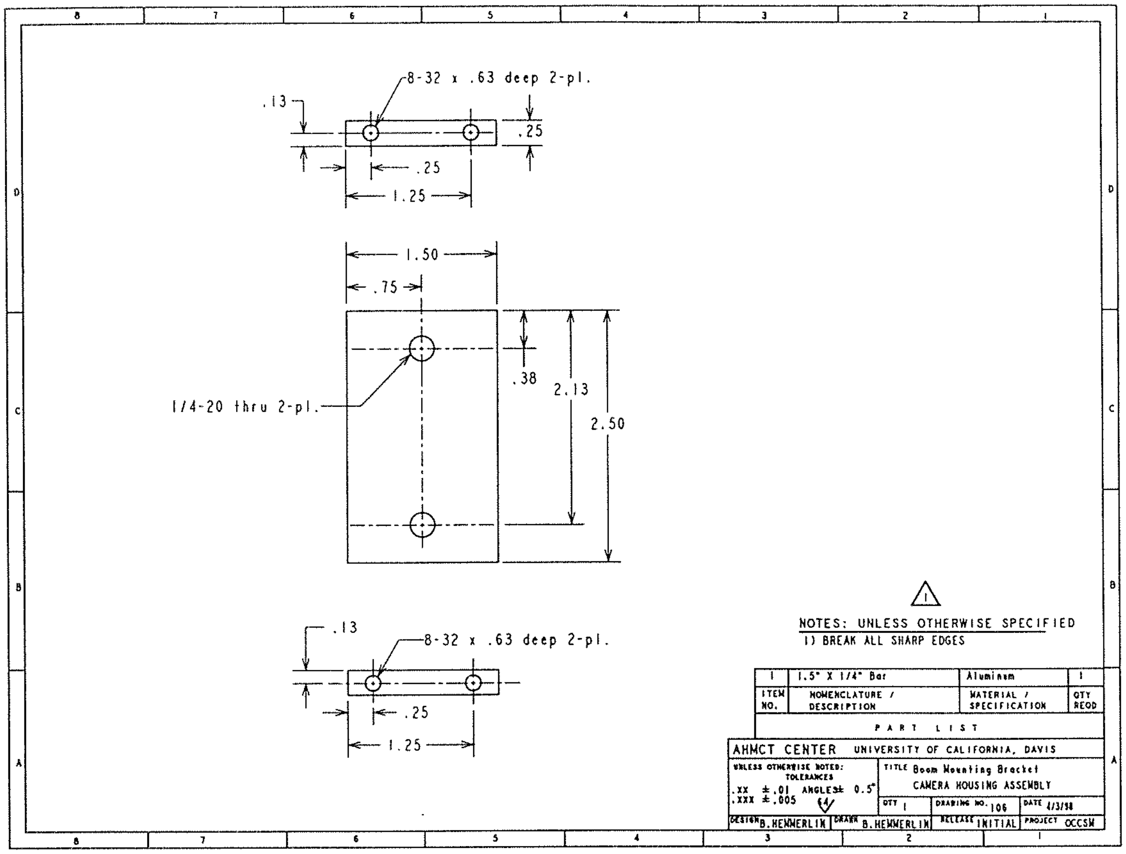
8-32 x .38 deep
2-pl. 180° apart
.125
.25
∅.21 thru 4-pl.
2.06
1.03
∅3.815
∅2.00
.90
1.80
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1/4" Plate | Aluminum | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE Base Camera Mount CAMERA HOUSING ASSEMBLY
QTY 1 DRAWING NO. 101 DATE 3/30/98
DESIGN B.HEMMERLIN DRAWN B.HEMMERLIN RELEASE INITIAL PROJECT OCCSM



∅3.815
.25
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1/4" Thick | Polycarbonate Sheet | 3
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE Lens CAMERA HOUSING ASSEMBLY
QTY 3 DRAWING NO. 103 DATE 3/30/98
DESIGN B.HEMMERLIN DRAWN B.HEMMERLIN RELEASE INITIAL PROJECT OCCSM


2.63
4mm Threaded Rod
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 4mm Threaded Rod | | 4
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
64
TITLE Mount Bolt
CAMERA HOUSING ASSEMBLY
QTY 4
DRAWING NO. 105
DATE 4/2/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM


.17
Ø.27
M4 x 0.7 thru
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | Steel Rod | Steel | 4
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
64
TITLE Mount Nut
CAMERA HOUSING ASSEMBLY
QTY 4
DRAWING NO. 104
DATE 4/2/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM


8-32 x .63 deep 2-pl.
.13
.25
.25
1.25
1.50
.75
.38
2.13
2.50
1/4-20 thru 2-pl.
.13
8-32 x .63 deep 2-pl.
.25
1.25
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1.5" X 1/4" Bar | Aluminum | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
TITLE Boom Mounting Bracket
CAMERA HOUSING ASSEMBLY
QTY 1
DRAWING NO. 106
DATE 4/3/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM

APPENDIX G: Part Drawings for the Hose Retraction System

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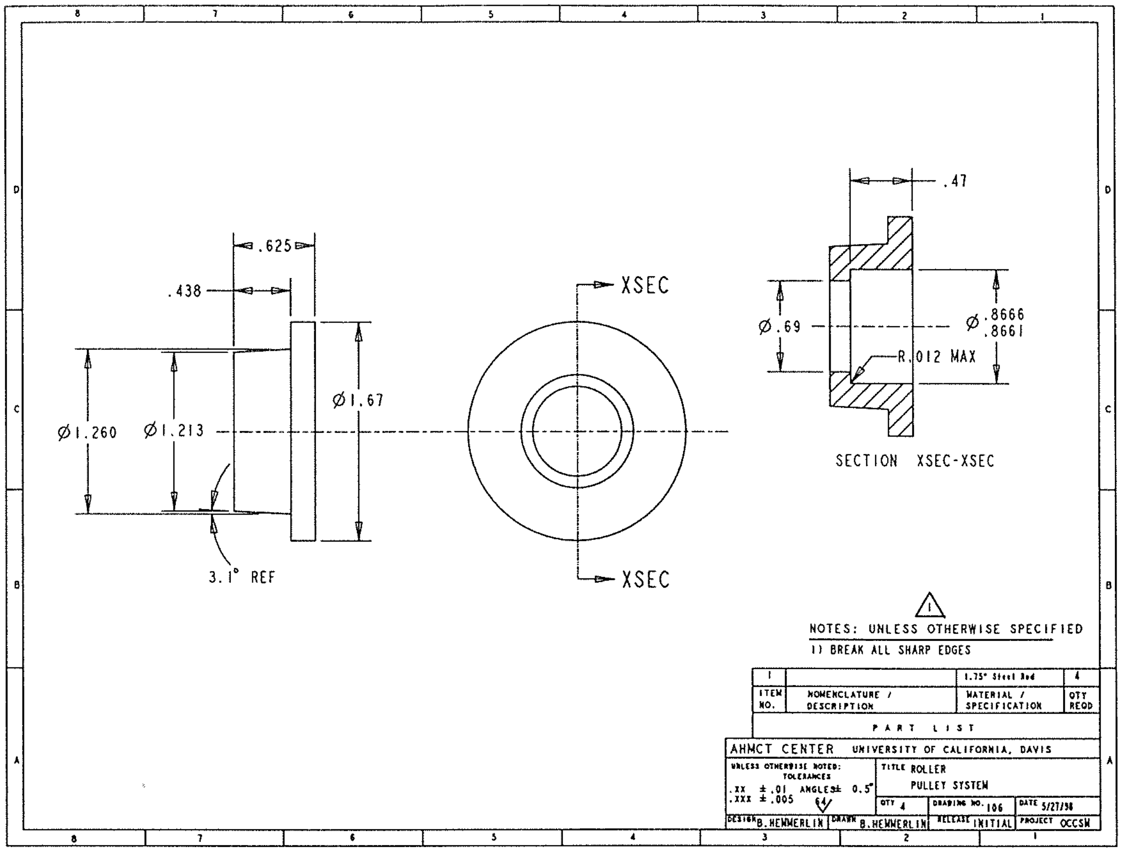
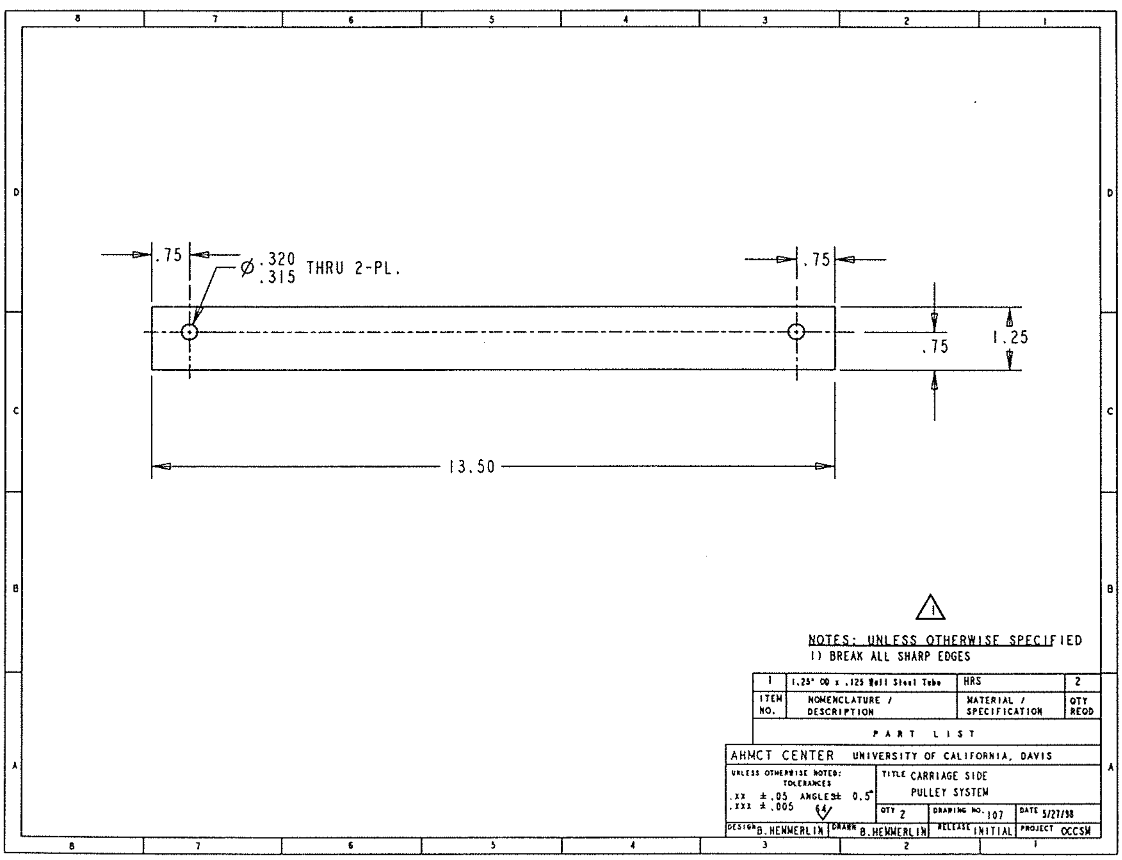
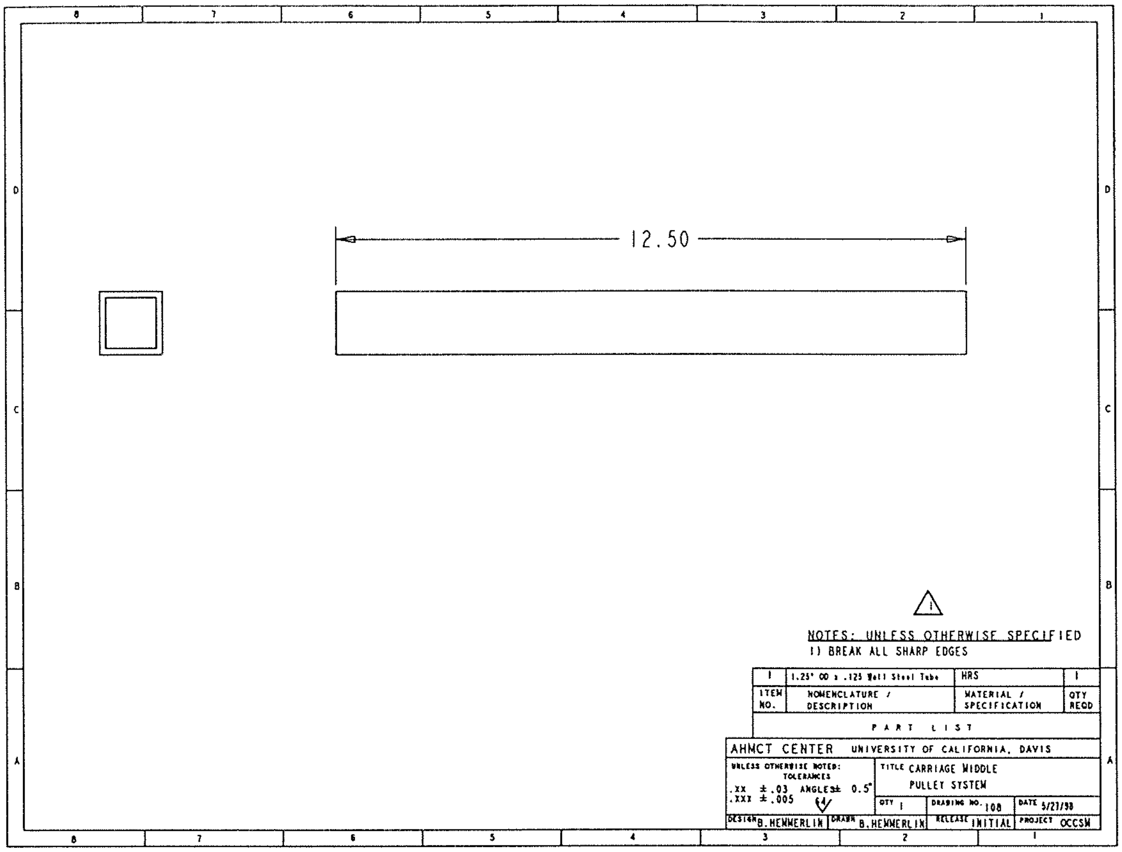
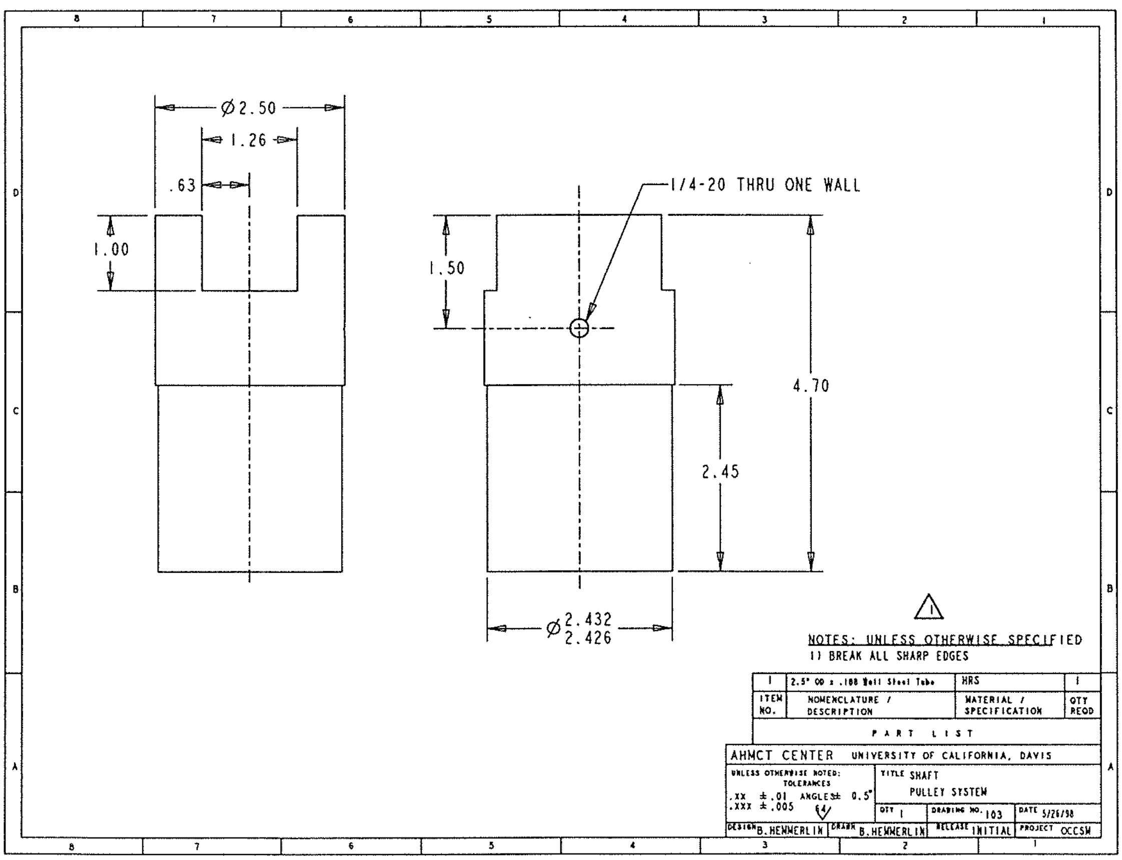
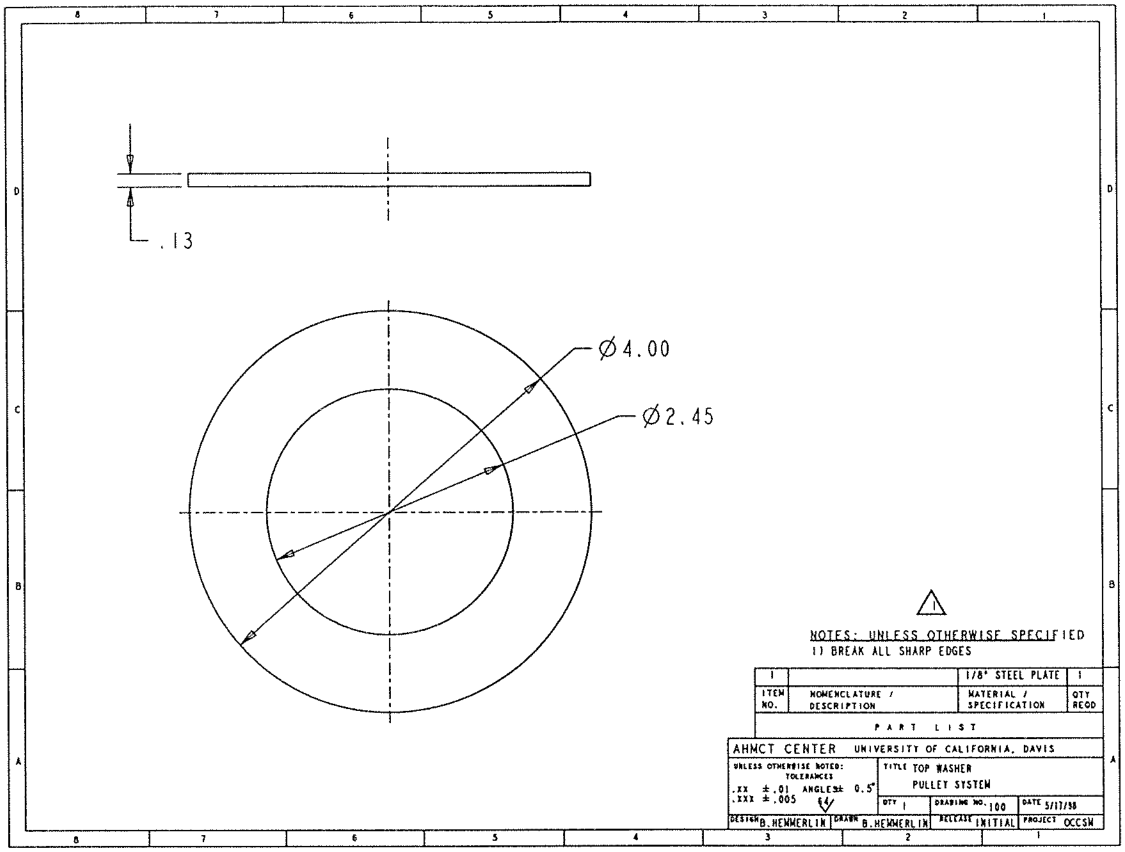
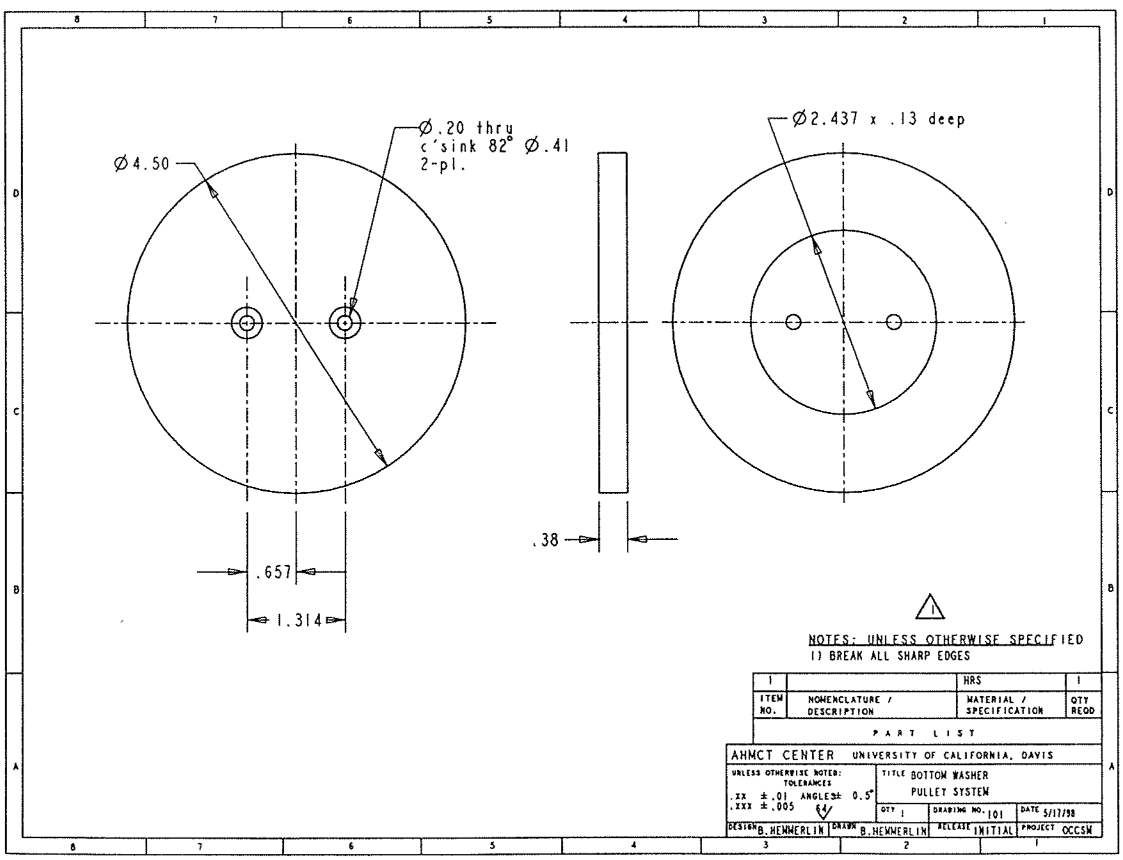

.625
.438
Ø1.260
Ø1.213
Ø1.67
3.1° REF
XSEC
XSEC
.47
Ø.69
Ø.8666 .8661
R.012 MAX
SECTION XSEC-XSEC
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 1.75" Steel Rod 4
ITEM NO. NOMENCLATURE / DESCRIPTION MATERIAL / SPECIFICATION QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.01 ANGLES± 0.5° .XXX ±.005
TITLE ROLLER PULLEY SYSTEM
QTY 4 DRAWING NO. 106 DATE 5/27/98
DESIGN B.HEMMERLIN DRAWN B.HEMMERLIN RELEASE INITIAL PROJECT OCCSM

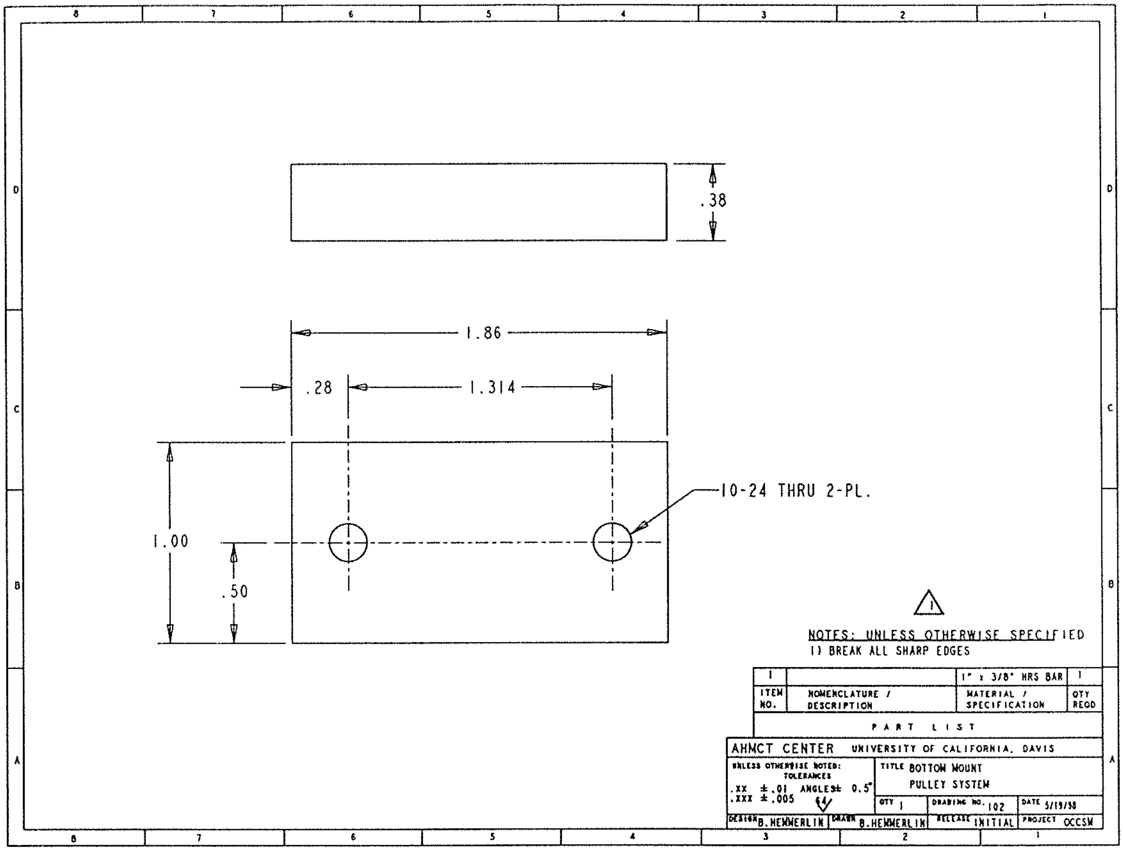

.75
Ø .320 .315 THRU 2-PL.
.75
.75
1.25
13.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 1.25" OD x .125 Wall Steel Tube HRS 2
ITEM NO. NOMENCLATURE / DESCRIPTION MATERIAL / SPECIFICATION QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES .XX ±.05 ANGLES± 0.5° .XXX ±.005
TITLE CARRIAGE SIDE PULLEY SYSTEM
QTY 2 DRAWING NO. 107 DATE 5/27/98
DESIGN B.HEMMERLIN DRAWN B.HEMMERLIN RELEASE INITIAL PROJECT OCCSM

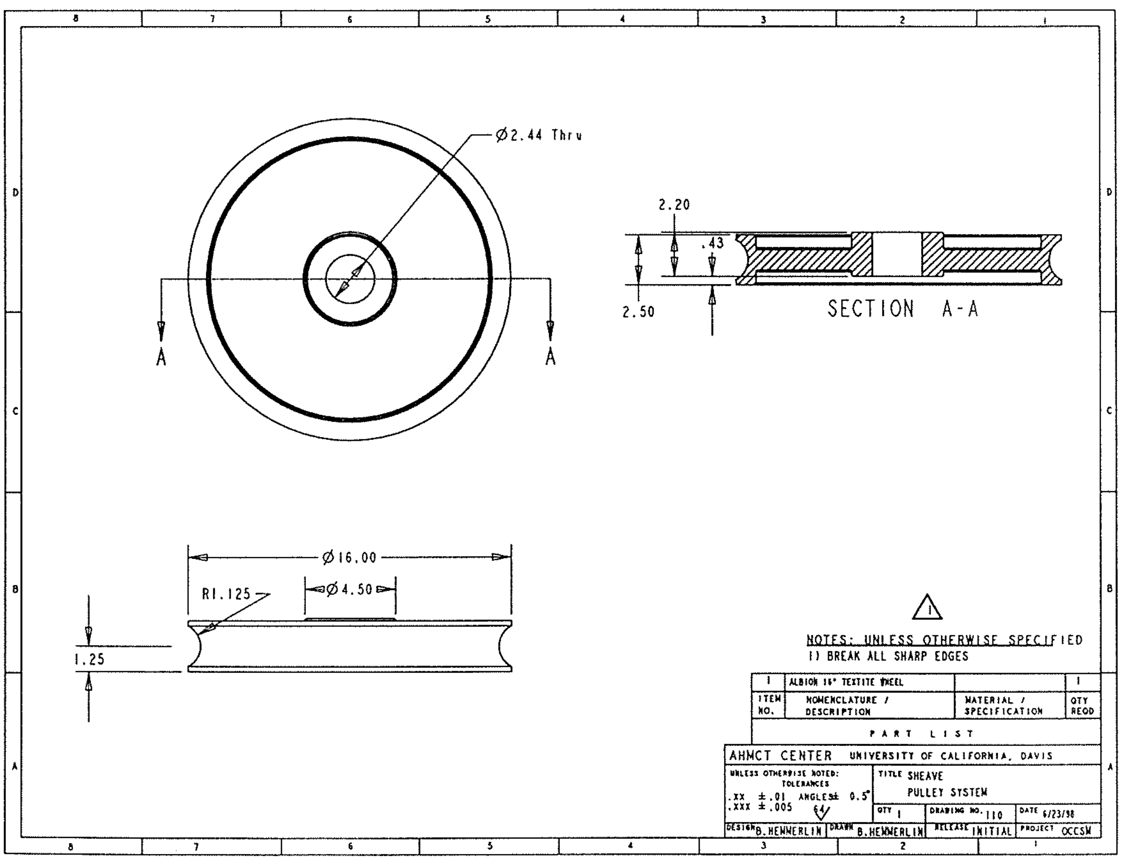

12.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 1.25" OD x .125 Wall Steel Tube | HRS | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.03 ANGLES± 0.5°
.XXX ±.005
TITLE CARRIAGE MIDDLE
PULLEY SYSTEM
QTY 1
DRAWING NO. 108
DATE 5/27/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM

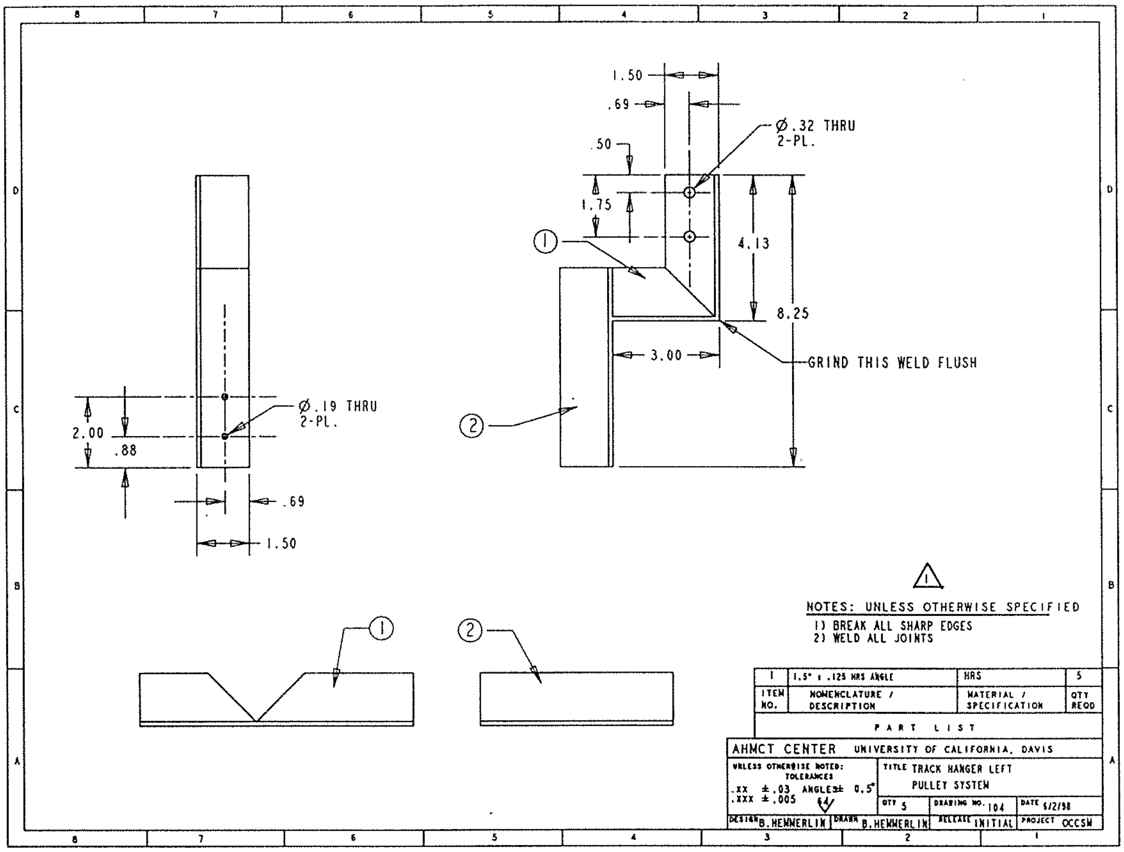

Ø2.50
1.26
.63
1.00
1/4-20 THRU ONE WALL
1.50
4.70
2.45
Ø2.432
2.426
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | 2.5" OD x .188 Wall Steel Tube | HRS | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
TITLE SHAFT
PULLEY SYSTEM
QTY 1
DRAWING NO. 103
DATE 5/26/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM

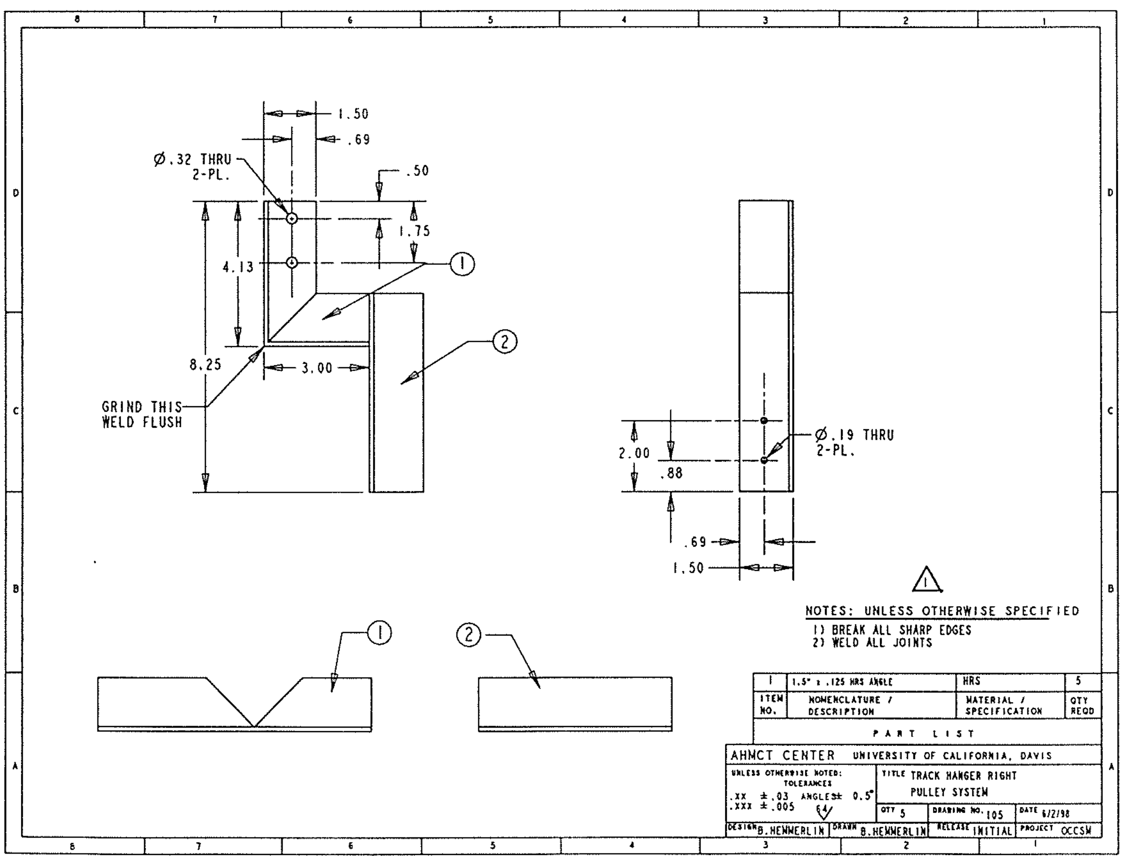

.13
Ø4.00
Ø2.45
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | | 1/8" STEEL PLATE | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
TITLE TOP WASHER
PULLEY SYSTEM
QTY 1
DRAWING NO. 100
DATE 5/17/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM

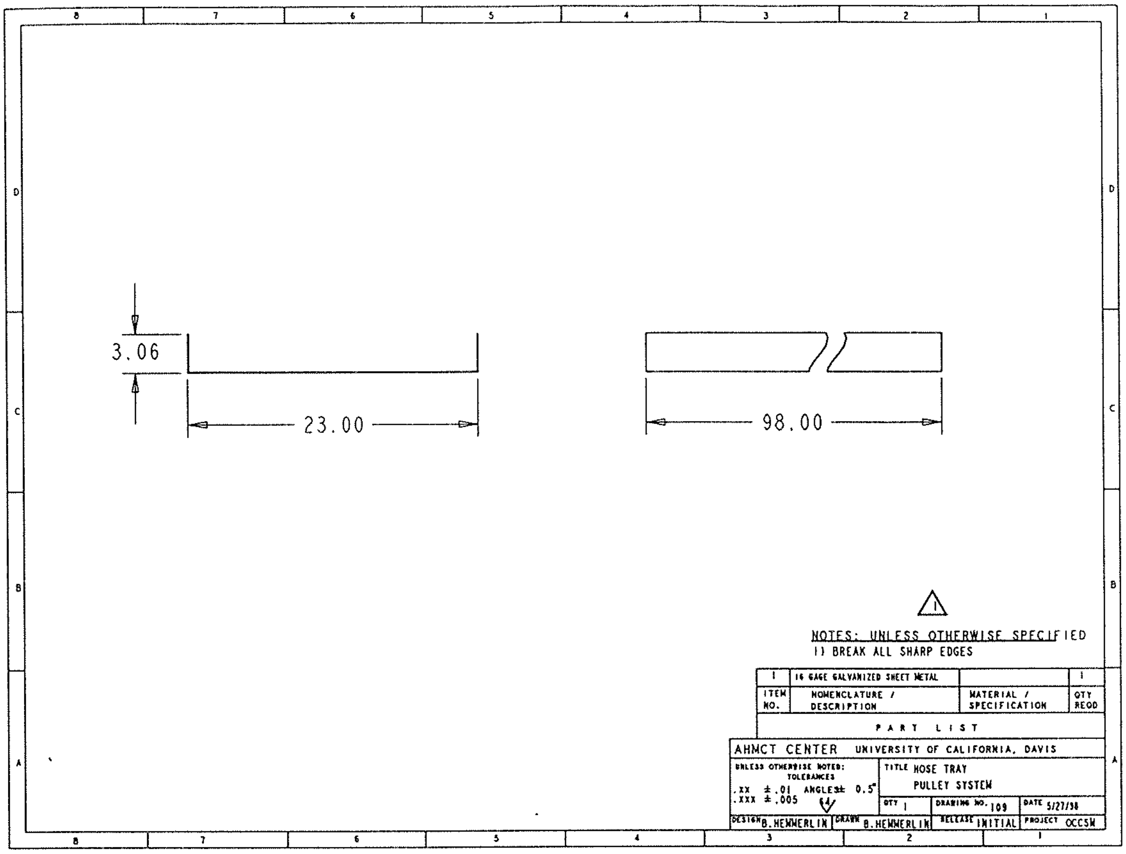

Ø4.50
Ø.20 thru
c'sink 82° Ø.41
2-pl.
Ø2.437 x .13 deep
.657
1.314
.38
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1 | | HRS | 1
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
TITLE BOTTOM WASHER
PULLEY SYSTEM
QTY 1
DRAWING NO. 101
DATE 5/17/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM


.38
1.86
.28
1.314
10-24 THRU 2-PL.
1.00
.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1" x 3/8" HRS BAR
ITEM NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
TITLE BOTTOM MOUNT PULLEY SYSTEM
DRAWING NO. 102
DESIGN B. HEMMERLIN
DRAWN B. HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM


Ø2.44 Thru
2.20
.43
2.50
SECTION A-A
A
A
Ø16.00
R1.125
Ø4.50
1.25
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
ALBION 16" TEXTITE WHEEL
ITEM NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
TITLE SHEAVE PULLEY SYSTEM
DRAWING NO. 110
DESIGN B. HEMMERLIN
DRAWN B. HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM


1.50
.69
.50
Ø.32 THRU
2-PL.
1.75
4.13
8.25
3.00
GRIND THIS WELD FLUSH
Ø.19 THRU
2-PL.
2.00
.88
.69
1.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2) WELD ALL JOINTS
1 | 1.5" x .125 HRS ANGLE | HRS | 5
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.03 ANGLES± 0.5°
.XXX ±.005
TITLE TRACK HANGER LEFT PULLEY SYSTEM
QTY 5
DRAWING NO. 104
DATE 6/2/98
DESIGN B. HEMMERLIN
DRAWN B. HEMMERLIN
RELEASE INITIAL
PROJECT OCCSW


1.50
.69
Ø.32 THRU
2-PL.
.50
1.75
4.13
8.25
3.00
GRIND THIS WELD FLUSH
Ø.19 THRU
2-PL.
2.00
.88
.69
1.50
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
2) WELD ALL JOINTS
1 | 1.5" x .125 HRS ANGLE | HRS | 5
ITEM NO. | NOMENCLATURE / DESCRIPTION | MATERIAL / SPECIFICATION | QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.03 ANGLES± 0.5°
.XXX ±.005
TITLE TRACK HANGER RIGHT PULLEY SYSTEM
QTY 5
DRAWING NO. 105
DATE 6/2/98
DESIGN B. HEMMERLIN
DRAWN B. HEMMERLIN
RELEASE INITIAL
PROJECT OCCSW


3.06
23.00
98.00
NOTES: UNLESS OTHERWISE SPECIFIED
1) BREAK ALL SHARP EDGES
1
16 GAGE GALVANIZED SHEET METAL
1
ITEM NO.
NOMENCLATURE / DESCRIPTION
MATERIAL / SPECIFICATION
QTY REQD
PART LIST
AHMCT CENTER UNIVERSITY OF CALIFORNIA, DAVIS
UNLESS OTHERWISE NOTED: TOLERANCES
.XX ±.01 ANGLES± 0.5°
.XXX ±.005
TITLE HOSE TRAY PULLEY SYSTEM
QTY 1
DRAWING NO. 109
DATE 5/27/98
DESIGN B.HEMMERLIN
DRAWN B.HEMMERLIN
RELEASE INITIAL
PROJECT OCCSM