Cold Roll Extrusion Machine Redesign
Final Design Report

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Statement of Disclaimer

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Executive Summary

This report is a comprehensive review of a redesign of subsystems for a Cold Roll Extrusion Machine, a forging operation, for PCC Rollmet. Three subsystems were addressed in this project: A new Loading system, Feed system, and Lubrication system. The Loading system, and support system during developmental design stages, addressed the timely loading process of pipes onto the forging machine. After two quarters of design iteration, the group decided on a lever arm, hydraulic actuated, loading design. In the final quarter of the project, a quarter scale model of this lever arm was constructed to prove the concept of the design. Computer models of the loading system were generated in Abaqus and Adams which allowed preliminary performance analysis. According to the results from these models, if a full scale model were to be implemented at Rollmet, it would perform as intended. The Feed system refers to the system that powers the axial motion during the forging operation. It dealt with the speed at which pipe could be returned to the starting position after one pass of deformation. The final design investigates the feasibility of a cable driven feed system and the findings show it to be a bulky and dangerous design. The Lubrication system addressed problem of the ease and speed at which lubricant is currently applied. A drip system was conceptualized to reduce increase efficiency for the single operator. The current system employs a folgers cup which the operator manually fills with lubricant and pours on the operating surfaces. The conclusion of this project is that the Lever Arm Loading device needs full scale testing, but would greatly increase Rollmet's manufacturing efficiency if implemented.

Introduction

This project is to begin a new design of a Cold-Roll Extrusion machine. PCC Rollmet currently possesses two cold-roll extrusion machines, uniquely designed and operated solely at their location in Irvine. These machines take short thick-walled cylinders and turn them into long thin-walled pipe. The machines vary a little in their designs, and due to this variance can produce different thicknesses and lengths of pipe. These machines were both built in the 60's and 70's. The focus of the new machine will be producing PCC Rollmet's high volume pipes of a minimum length of 25 feet to a maximum length of 35 feet. The expectation is that it will be able to produce the pipes quickly and easily, unlike the current designs. This expectation will be met by upgrading the loading system, the feed system and the lubrication system. To design this machine we will be working with Dr. Julian Roberts and Dr. Jack Hyzak at PCC Rollmet.

Background

Cold-Rolling is a metallurgy process that compresses metal and strains it to provide a thinner and longer final product. Cold work occurs at room temperature. Straining the material increases its strength and it provides good dimensional tolerances and a high quality surface finish. The rolling process is performed by deforming the part between two rollers, which is most often manifested in a process like the one shown in Figure 1 [1] and Figure 2 [2] below.

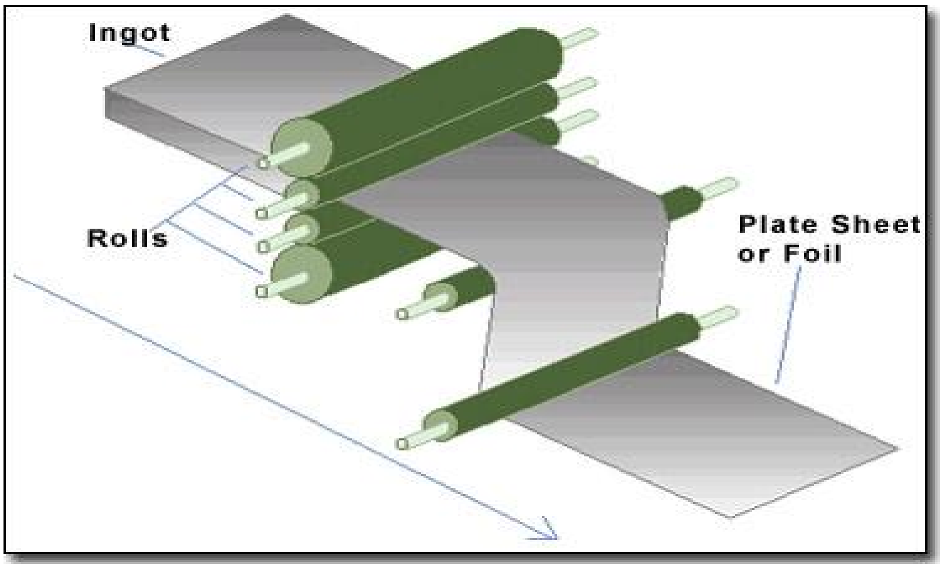

Figure 1: Cold Rolling

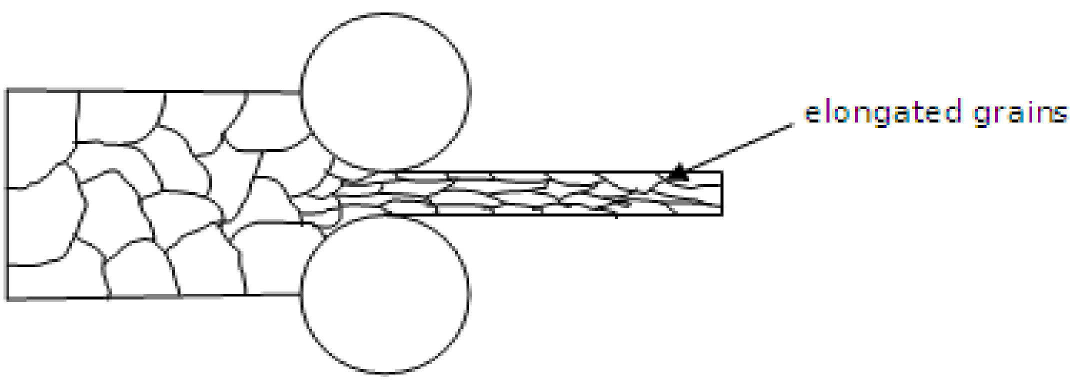

Figure 2: Grain Deformation

Extrusion is another metallurgy process used in industry to create different shapes of solid material and pipes. This forging process can be performed hot or cold. Extrusion is usually performed with hydraulic presses that push metal through a die to produce a cylindrical rod or pipe. Figure 3 [3] shows how extrusion is performed. The extrusion process is most often performed hot; however there are cold extrusion

processes as well such as back extrusion. The concept behind back extrusion is to compress the working metal in a container with a single open face. Using a push rod to compress, the working metal extrudes out and around the rod as shown in Figure 4 [4]. The comparison of benefits and costs of cold and hot working is further explained below.

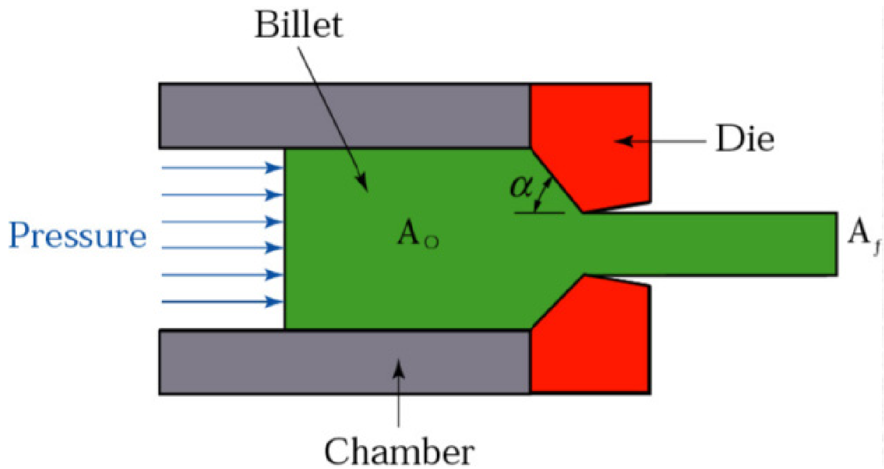

Figure 3: Extrusion

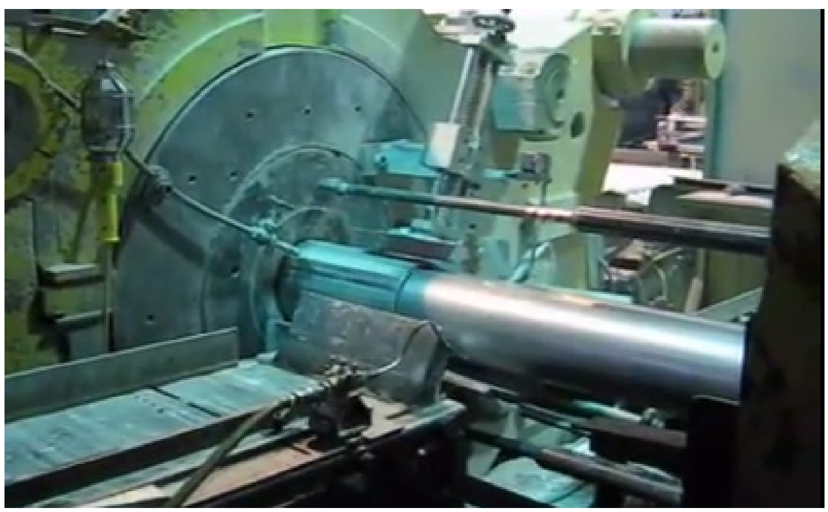

Figure 4: Back Extrusion

PCC Rollmet has produced a machine and a process that combines both rolling and extruding, hence named roll extrusion. This process is unique to PCC Rollmet and is not used anywhere else in the world. Using roll and extrusion in conjunction allows this machine to handle pipes of any diameter between 6 and 26 inches OD and thicknesses of 0.04 - 1.5 inches. Unlike back extrusion seen in Figure 4 [4], cold roll extrusion can form piping to any size within these parameters without the need of extra tooling. Along with this freedom to control the ID and OD, the machine can produce the desired piping to tolerances as small as ±0.005 inches on the OD. The combination of

cold roll and extrusion is advantageous in the manufacturing of piping also because of its ability to handle material of any origin. The machine requires this base material to be a cylinder because it must slide on a steel mandrel to be rolled. There are many different processes that will achieve a cylinder for this base material, and the cold roll machine will accept any of them. The machine can even handle pipes that have defects because the cold roll process massages out the flaws and produces a smooth surface finish. In these ways the cold roll extrusion machine is superior to other processes available today. The reason that it is uniquely used by PCC Rollmet is that it requires a very large capital investment to produce the machine and the knowledge of how to operate it well is kept at PCC Rollmet.

<u>Material Benefits of Cold Operations as Compared to Hot:</u>

The process of strengthening a material by plastic deformation is known as strain-hardening. When metal is plastically deformed its strength increases (at a cost of ductility) as a result of dislocation motion and dislocation generation within the crystal structure of the metal. Strain-hardening is, for the most part, a positive outcome from cold working a non-brittle material.

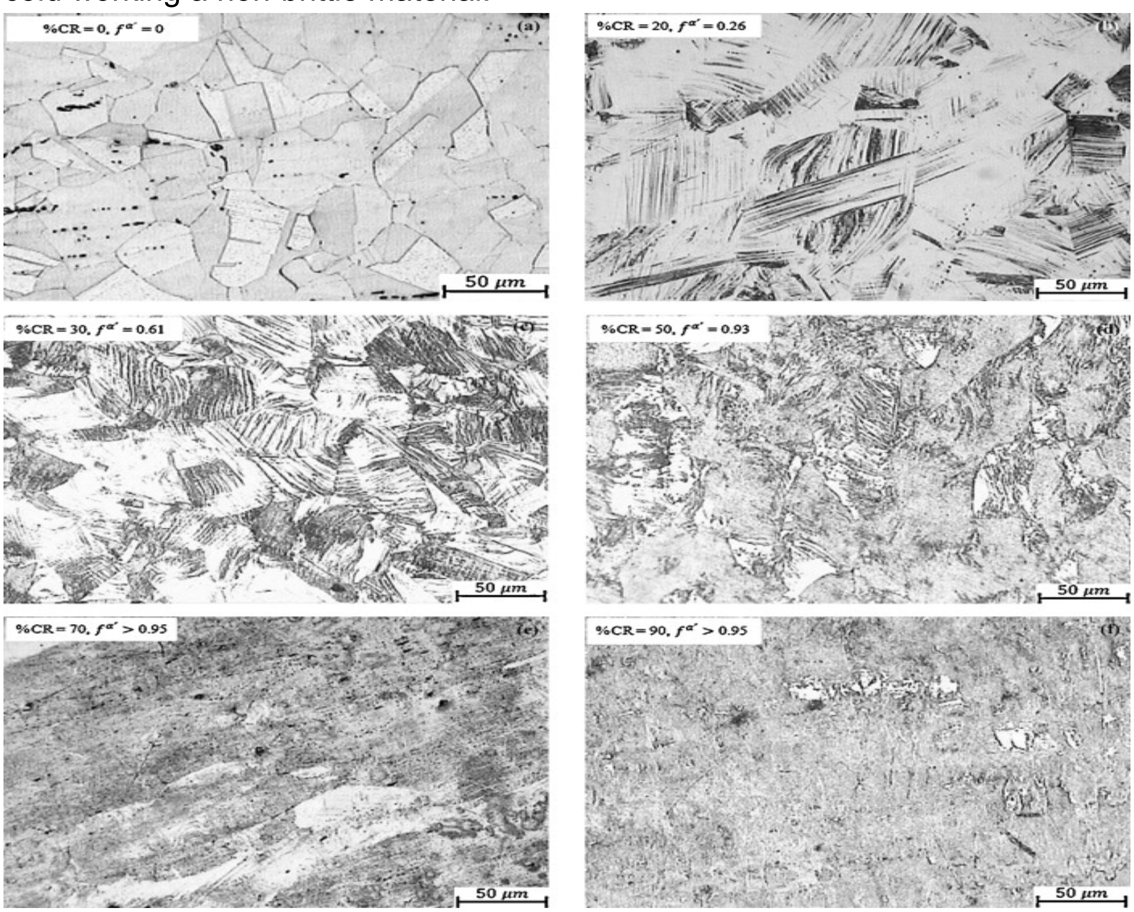

Figure 5: Effects on grain size with different amounts of strain

In Figure 5 [5] it is easy to see the effect of additional strain on AISI 304L stainless steel at 0 ºC. From top left to bottom right the pictures show a cold roll reduction of 0%, 20%, 30%, 50%, 70%, and 90%. It is apparent at a cold roll reduction of 90% that the grains of the steel have become extremely small, almost impossible to see even with the optical micrograph used to take these photos. This leaves the steel with increased yield strength and increased fracture strength.

The plug rolling process (below in Figure 6 [6]) is another competing process for cold roll extrusion in the construction of long thin 40ft pipe. It begins in a similar way to the cold roll extrusion process starting with a large hollow bloom. The difference in plug rolling is the hollow bloom begins at a temperature of roughly 1280 ºC. The hollow is then bitten by two large rollers and guided onto a mandrel with a plug on the end that has an OD with the desired ID of the finished pipe. The hot pipe is then pushed over the plug and onto the mandrel. At this point the plug is pushed off and two smaller rollers are lowered to roll the finished product off the mandrel to be sent to cooling tanks.

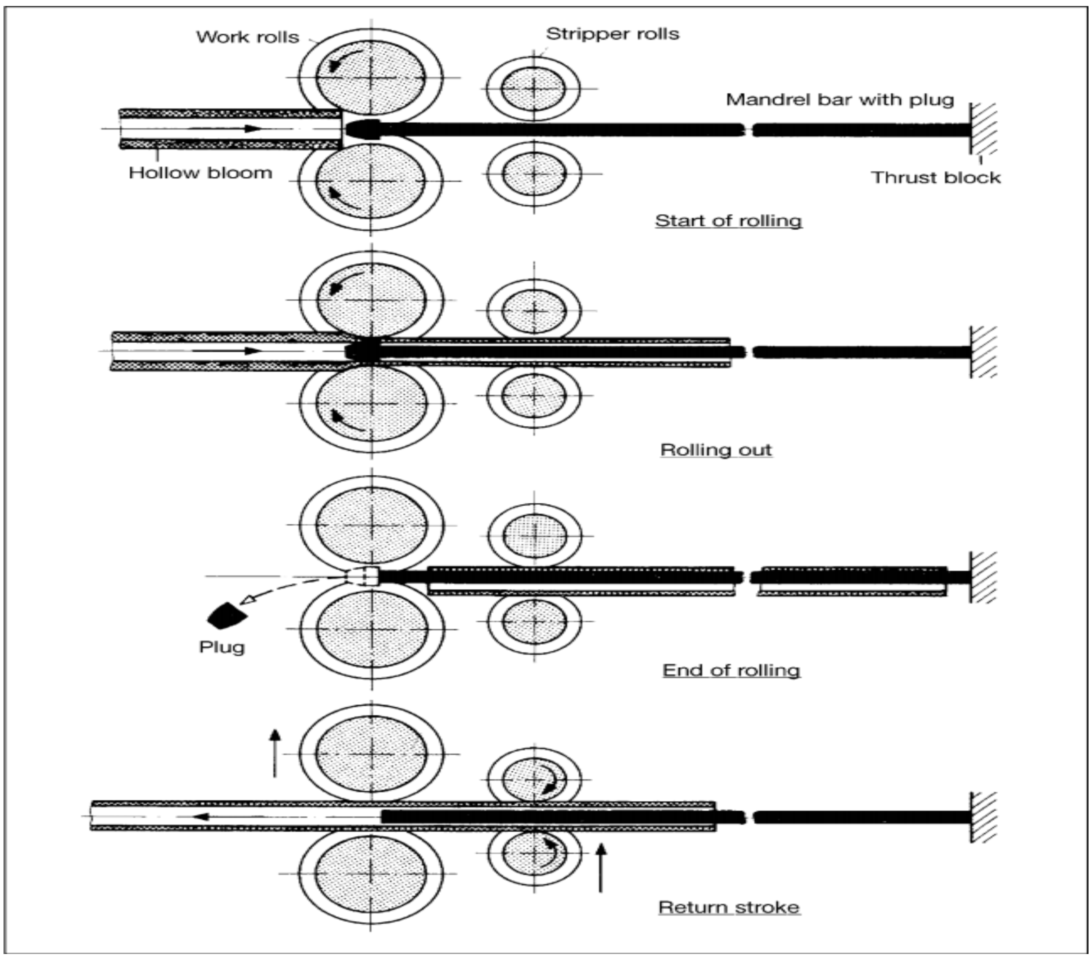

Figure 6: Infographic on plug rolling process

Using hot hollow bloom vs. a cold hollow bloom has its pros and cons. A hot bloom is easier to work, therefore requiring less force to roll. This translates to less overall power consumed by the machine per part, which can save a production facility money spent on electrical power in the long run. The negatives to using a hot hollow bloom, however, seem to outweigh its cheaper production cost. For one, once the pipe is rolled it must be cooled, this takes time which in a production facility is equivalent to costing money. Compare this to a cold process where the pipe is ready to ship as soon as it's removed from the machine, therefore saving a good chunk of time per part. A hot rolled pipe also has a lower yield strength because of its larger grain size. Figure 7 [7] clearly shows the difference in grain structure between hot and cold rolled steel.

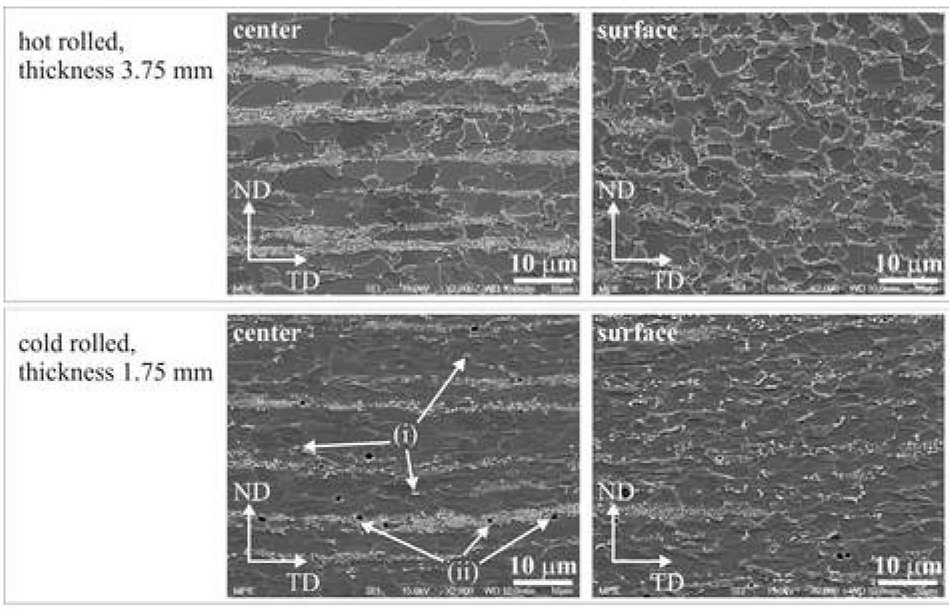

Figure 7: Difference in grain size between hot and cold rolled galvanized steel

The benefit of using cold extrusion is it allows PCC Rollmet to meet specifications required by customers in fields of high stress application. One of Rollmet's customers is the oil industry, which has very high standards for piping which is introduced below:

Oil Specification, API: ISO 13680 [8]:
Specification for Corrosion Resistant Alloy Seamless Tubes for Use as Casing, Tubing and Coupling Stock (includes Errata 1 dated August 2011)
Specifies the technical delivery conditions for corrosion-resistant alloy seamless tubulars for casing, tubing, and coupling stock for two product specification levels.
This edition of API Spec 5CRA is the identical national adoption of
ISO 13680:2010. Pages: 87

Other markets of PCC Rollmet's cold roll extruded piping include: Aerospace (missiles), nuclear power plants and submarines. Standard ASTM specifications for Nickel, Nickel Alloys Seamless Pipe and Tubes have been included in Appendix C.

Although cold roll extrusion is a process unique to PCC Rollmet, there are some competing manufacturing processes that also produce seamless piping. Plug rolling was one mentioned above. Another such process received a U.S. Patent, no. EP2111932 B1 (below in Figure 8 [9]), in June of 2012. This process uses a tubing process called 'piercing' which is very similar to plug rolling except that the mandrel is pointed and creates the cylinder hole without a plug and must be done hot. The patent includes pipe straightening techniques shown in sketch c) and d) of Figure 8 [9] below.

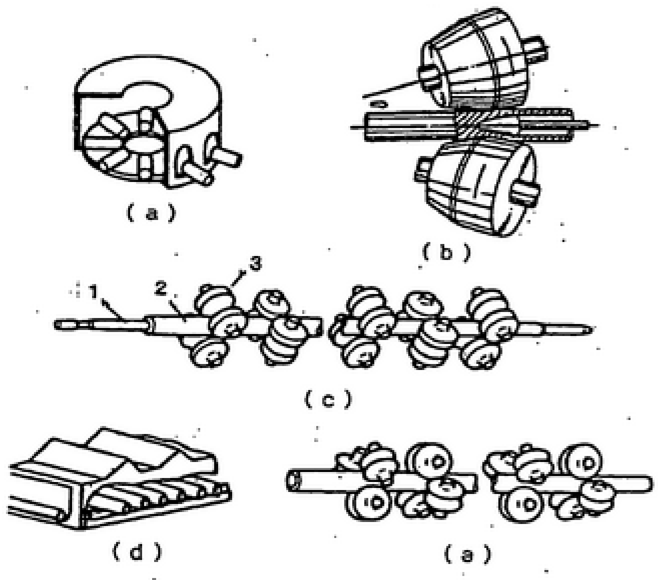

Figure 8: Patent Sketches EP2111932 B1

Cold roll extrusion is unique in its performance but there are many competing manufacturing processes that produce high quality piping efficiently. Cold roll extrusion has an advantage in quality on a lot of these competing processes, but in order to keep the corporation ahead, it needs to streamline its manufacturing process to be more time effective. The manufacturing process in Figure 8 shows the high volume capabilities that of continuous rolling and heat treating that lead to the ability to meet more purchasing orders.

Design Requirements

The need of PCC Rollmet is a new External Roll Extrude (ERE) machine to produce high volume parts quickly. The goal of our project is to design major subsystems that will allow this machine to work faster than the existing models. The layout and overall design of the current machine is effective, and has been working for a long time. Our goal in the design of a new machine, therefore, is not to completely redesign the machine, but rather improve the systems that inhibit its speed.

The need for a new cold roll extrusion machine at PCC Rollmet stems from the lengthy and difficult load process. Because the current machines were designed and built in the late 60's, the flaws and shortcomings are now more evident. The current machine lacks efficiency and cannot run at high speeds without the feed transmission wearing out quickly. It is manually loaded by a worker and a 10-ton crane, a process which consumes a lot of time. The lubrication is applied manually, and there is no way to recycle and reuse the majority of it. The feed for the extrusion is done by a worm gear which is extremely slow and limits the potential of the machine. These machines are unique and are not used anywhere else in the world and PCC would like a new, more efficient, machine to increase production and efficiency.

We will be working in conjunction with Dr. Roberts and Dr. Hyzak for the redesign. The subsystems are broken up into the loading mechanism, lubrication, die box, feed system and control system/user interface. We will focus on redesigning the loading mechanism with thought given to how the feed and lubrication systems work in conjunction. These will be designed around the existing die box (fixed design). The design will be presented to PCC Rollmet as a set of concepts and detailed drawings for the lubrication, feed, and loading systems. In addition, for the loading system we will also do one of two plans for development of the loading system depending on the need of the project after the Critical Design Review (CDR) on April 18th. Our first plan is to do a more in-depth redesign of the loader including detailed drawings and computer generated model. This will also be complemented by a purchasing plan for parts and cost estimates of the loading mechanism. The second plan is to build a prototype scale model of the loading mechanism approved in the CDR. We will pick one of these plans depending on what Dr. Roberts and our team decide will be most beneficial for the project.

In Table 1 (below), we have stated the various requirements necessary in the design of a new machine. This table is a more simplified version of the QFD. QFD stands for Quality Function Deployment. The QFD analyzes the project requirements by taking a lot of different things into account. In the QFD we took the requirements given by PCC Rollmet and wrote how we will attempt to meet them. We then checked to ensure that our attempts to meet the requirements actually do meet them and are not considered extraneous by PCC Rollmet. We do this by giving the design specifications

a positive, negative, or neutral symbol, comparing to PCC's current, for each of PCC's requirements. From there we compared the requirements of the new machine to the specifications of the current machines at PCC Rollmet. Unfortunately, we we're unable to receive the specifications of the two current machines from PCC. The reason we are unable to get these requirements is because they are buried in engineering drawings and are currently inaccessible to us. The results will help us identify and focus upon the most critical engineering specifications. As well as identifying the most critical features we also identify features that are not very important. From our QFD we determined that every engineering specification we included is relevant to the project and must be addressed. The full QFD is included in Appendix B.

Table 1: Engineering Requirements

Spec #	Parameter Description	Requirement	Tolerance	Risk	Compliance
1	Working Stroke (Feed)	35ft	Minimum	L	A,S
2	Feed Rate (Feed)	36ipm-360ipm	Min-Max	M	A
3	Feed Power (Feed)	150hp	Min	L	S
4	Die Box Swing Angle (Die Box)	20°	Min	M	A,S
5	Rotational Speed (Feed)	0-120rpm	Min-Max	M	A,S
6	Rotational Power (Feed)	400hp	Min	L	S
7	Die Slide Travel (Die Box)	24 in	Min	L	S
8	Die Feed Load (Die Box)	500 tons	Min	M	A,S
9	Die Locked Load (Die Box)	1000 tons	Min	M	A,S
10	Machine centerline (Loading)	60 in	Min	L	A

To start this project we will analyze the existing machines and learn how they work. Then we will go into detail with the specific subsystems that we plan to redesign. From there we plan to brainstorm various ways to upgrade the subsystems. Afterwards

we will take the top ideas we came up with and detailed plans will be created. The detailed plans will then be presented to PCC Rollmet for analysis and evaluation. After hearing the feedback, we will perform redesigns as necessary. Then a cost-benefit analysis will be done to determine whether upgrading is worth it and how long the upgrade will take to pay itself off. The final design and analysis will then be presented to PCC Rollmet.

Safety on our project is a huge concern because our project is in a heavy manufacturing environment. Not only are we dealing with 500 tons of compressive force inside of our die box. But we have multiple cables under very high tension, large hydraulic actuators, and rotating machinery. We must also consider the electrical hazard of powering the multiple 150+ horsepower motors required to make this machine run. As if this machine was already not hazardous enough it will also generate constant high decibel noise that will require the use of ear protection to be around for even very brief inspections. These concerns will be passed on to PCC Rollmet as we will not be building a full-scale model of this machine.

Concept Generation

Prior to brainstorming we have broken up the Cold Roll Extrusion machine into separate subsystems which we plan to redesign. These subsystems are the loading subsystem, the feed subsystem and the lubrication subsystem. To design these subsystems we had a series of brainstorms using various different tactics. Some of the different styles of brainstorming we used were brainwriting, individual brainstorms and group discussions. We quickly discovered that we were most successful with individual brainstorms followed by a group discussion. This led to back-to-back brainstorming which produced very good results. Also we came up with ideas during non-brainstorming sessions and used these as well.

Loading System

The loading mechanism is what moves the pipe from the ground onto the machine itself. This mechanism is responsible for lining up the pipe with the mandrel as well as securing the pipe in place as the pipe is pushed fully onto the mandrel. The current loading system is a 10-ton crane operated by a manufacturing laborer. The laborer secures the pipe to the crane, then he lubricates both the mandrel and the hanging pipe. Having lubricated all of the key components the laborer carefully uses the crane to slide the pipe onto the mandrel. He keeps the crane fastened to the pipe and uses the cranes controls to line up the mandrel with the tail stop so he can pin the two together. This process is effective and is a generally safe, easy and reliable loading process. The one shortcoming is that it is not very fast. The loading process in total takes about 30-45 minutes. Here is where the process can be improved. Figure 9, Figure 10 and Figure 11 below highlight some of the extensive brainstorming that we did to come up with a faster loading system. Figure 9 has 5 different sketches we came up with for the loading system. These ideas are from the early brainstorming phase and would not become refined until later.

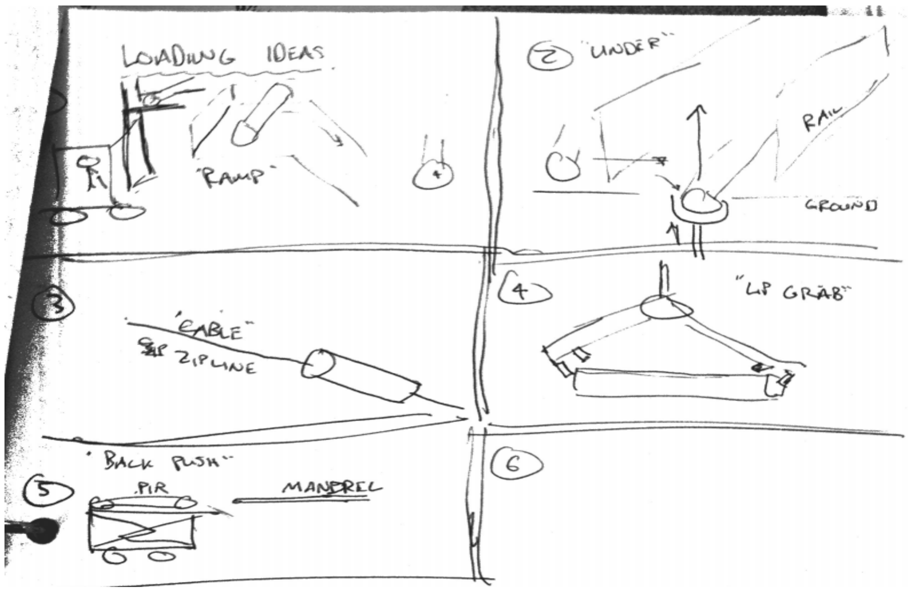

Figure 9: Logbook Sketches of Loading Mechanisms Brainstorming

Figure 10 and Figure 11 (below) come from a bit later in the brainstorming phase and have been refined. These ideas still need further refinement as the technical aspects of the concepts have been ignored thus far.

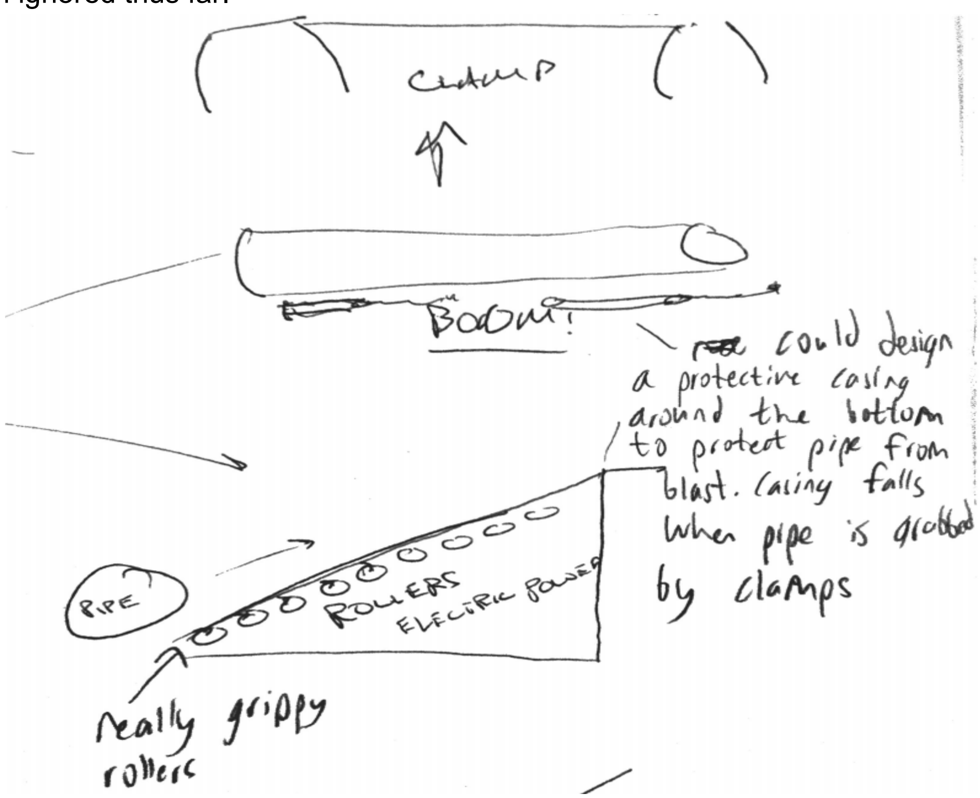

Figure 10: Loading Sketches

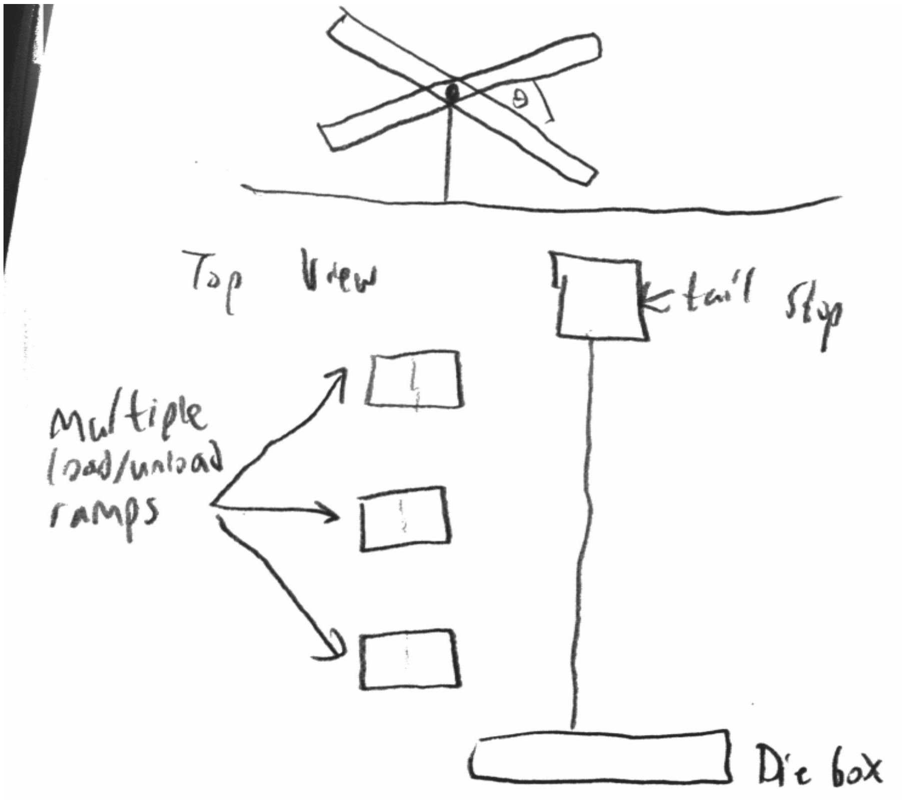

Figure 11: Loading Sketches

After refining the different concepts we decided to approach the technical aspect of loading the pipe onto the mandrel. For example we started to consider whether the pipe would be supported by the loading system or a separate set of supports would be needed. How precise would the loading system be as we have to get it centered within a .125” tolerance. Designs like the ramp and conveyor belt would require another support system to catch and center the piping. Meanwhile the lever arm would not. The question then is, would it be better to use a loading system with separate supports or have the supports incorporated into the loading system. The new system must also be faster than the crane is. It can also incorporate more than just the crane does as it could have a designated loading area where it is easy for the loading system to pick up the pipe and load it onto the mandrel.

We had a variety of creative ideas that were very outside of the box in their design. For example the zip line loader show below in Figure 12 was different from the all the others designs but still seemed to accomplish all the functions, even centering and loading. It however lacks a couple very important components. First, the mandrel’s rear supporting cart (support cart) would have to be moved out of the way. Second, the unloading capability is limited. This is the start of the technical analysis done upon each concept to ensure it is feasible.

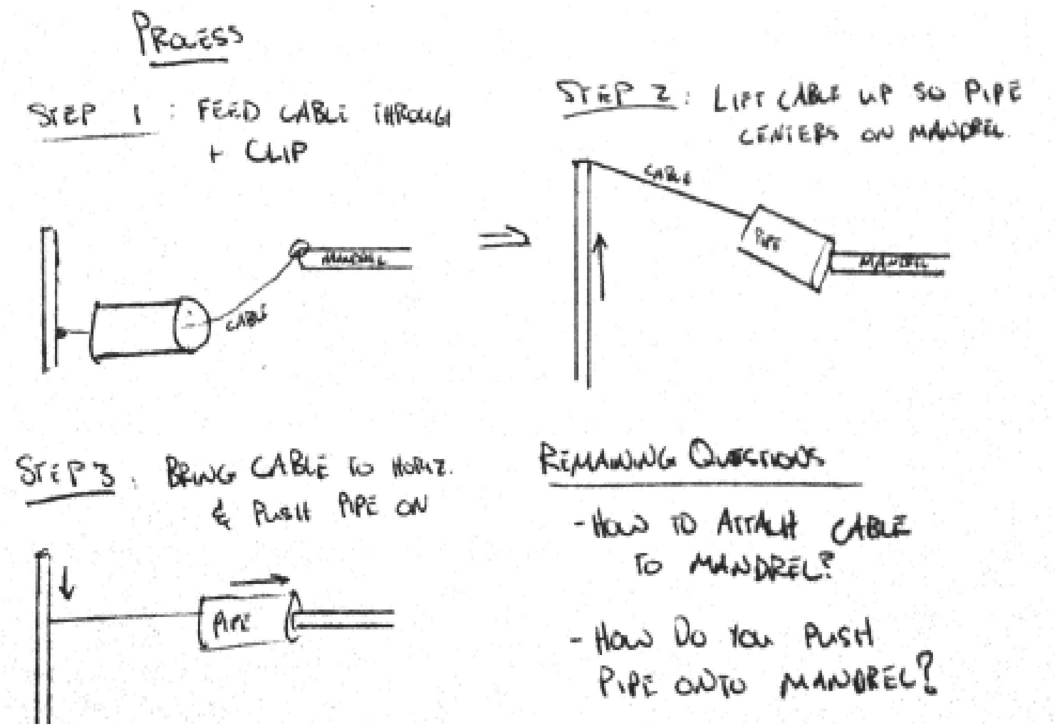

Figure 12: Zip line Function Sketches

In order to analyze these capabilities and flaws we created a Pugh matrix (below in Table 2) for our top loading ideas. On the first column we write down features that are necessary for the loading system, such as durability, cost, loading time, etc. We compared each of different ideas to the current system (crane) with respect to each feature. If the system is better it gets a “+”, “-“ for being worse and an “S” for being the same. However each feature is not weighted, thus we are only given a rough idea of what ideas might work well.

Table 2: First Loading Pugh Matrix with Crane as Datum

Concepts: Needs	Crane	Ramp	Conveyor Belt	Lever Arm	Hydraulic Press	Rear Loader	Zipline	Teter Totter
Loader Size		-	-	-	-	-	S	-
# of Operators	D	S	S	S	S	S	S	S
Ability To Handle Size		S	S	S	S	S	S	S
Reliability	A	S	S	S	S	S	S	S
Repair		+	S	S	-	S	+	S
Durability	T	+	S	S	+	S	S	S
Operation Cost		+	+	S	-	+	+	S
Capitol Cost	U	+	S	-	-	S	+	-
Unloading Capability		-	S	S	S	S	-	S
Loading Time	M	+	+	+	+	+	+	+
Unloading Time		-	+	+	+	+	-	+
Need for Support		-	-	S	-	S	S	-
Sum +	0	5	3	2	3	3	4	2
Sum -	0	4	2	2	5	1	2	3
Total	0	1	1	0	-2	2	2	-1

After the first Pugh matrix we generated another one, except we now used the top performing idea from the original as the datum. This new Pugh matrix is displayed below in Table 3. After doing this we saw that a new idea took the top score, and that was interesting as the original Pugh matrix did not show that idea winning. The Pugh matrix was a valuable tool to begin evaluating our ideas; however more evaluation is still needed.

Table 3: Second Loading Pugh Matrix with Rear Loader as Datum

Concepts								
Needs	Rear Loader	Crane	Ramp	Conveyor Belt	Lever Arm	Hydraulic Press	Zipline	Teeter Totter
Loader Size		+	S	-	S	+	+	S
# of Operators	D	S	S	S	S	S	S	S
Ability To Handle Size		S	S	S	S	S	-	S
Reliability	A	S	+	S	S	S	S	S
Repair		S	+	S	S	S	+	S
Durability	T	S	+	S	S	S	S	-
Operation Cost		S	+	S	S	S	+	S
Capitol Cost	U	S	+	S	S	S	+	S
Unloading Capability		+	-	S	S	S	-	S
Loading Time	M	-	S	-	S	S	S	S
Unloading Time		-	S	S	S	S	-	S
Need for Support		S	-	-	-	S	S	-
Sum +	0	2	5	0	0	1	4	0
Sum -	0	2	2	3	1	0	3	2
Total	0	0	3	-3	-1	1	1	-2

With these results in mind, we proceeded to create a decision matrix shown in Table 4. Here we weighted each of the design criteria with a percentage of importance in comparison to each other. We decided that the most important criteria are the reliability of mechanism to perform its operation (21%) and the speed it can do this (24%). We proceeded to rank each of our design ideas on a scale of 1-10, for each of the performance columns. 10's were given to the design that performed the best in each category and the remaining designs were ranked below that depending on how far inferior they were decided the score gap. Each column was multiplied by its corresponding weight and totals were added in the last column. The designs with the highest scores were decided to be the best. Consider the matrix below to see our results.

Table 4: Decision Matrix for Loading

Design Criteria	Loading Speed	Reliability	Size	Ease of Operation	Durability	Handling Size	Operation Cost	Capital Cost	Repair	Overall Score
Wieght	0.24	0.21	0.1	0.1	0.15	0.05	0.05	0.05	0.05	1
Crane	5	10	10	6	8	10	6	9	7	7.7
Ramp	9	8	5	8	9	8	10	10	10	8.39
Conveyer Belt	8.5	9	4	9	7	8	7	7	7	7.73
Lever Arm	10	10	7	10	9	7	8	6	7	8.95
Access Port	9.5	10	8	10	8	7	9	7	7	8.88
Rear Loader	7.5	9	6	8	7	6	7	6	7	7.44
Zipline	8	8	6	7	7	6	8	7	8	7.4
Teter Totter	8.5	8	5	9	8	8	8	6	8	7.82
Underneath Ramp	9.5	9	8	10	10	7	10	8	9	9.17

This matrix aided our discussions about which loading system is most appropriate to implement with a new cold roll extrusion machine at PCC Rollmet. We are not yet experts in this field, and so our judgments on rankings and weightings is limited, so we used this matrix only as a tool in our decision making process. Our deepening analysis of the feasibility of these loading components prompted many discussions that moved us to a better understanding of the need at Rollmet.

From the decision matrix we determined that our top design was the underneath ramp (Figure 13). This was followed by the lever arm, the access port, the normal ramp, and the conveyor belt. However, as stated earlier we started to understand which designs were feasible after the matrix was completed and we recognized that all of the winning designs focused on side loading. For this reason we have decided to keep the rear loader in contention for the best design because it does not load from the side. We also decided to get rid of normal ramp design as the underneath ramp incorporates everything the normal ramp does. Thus we end up with the underneath ramp, the lever arm, the access port, the conveyor belt and the rear loader as viable loading systems.

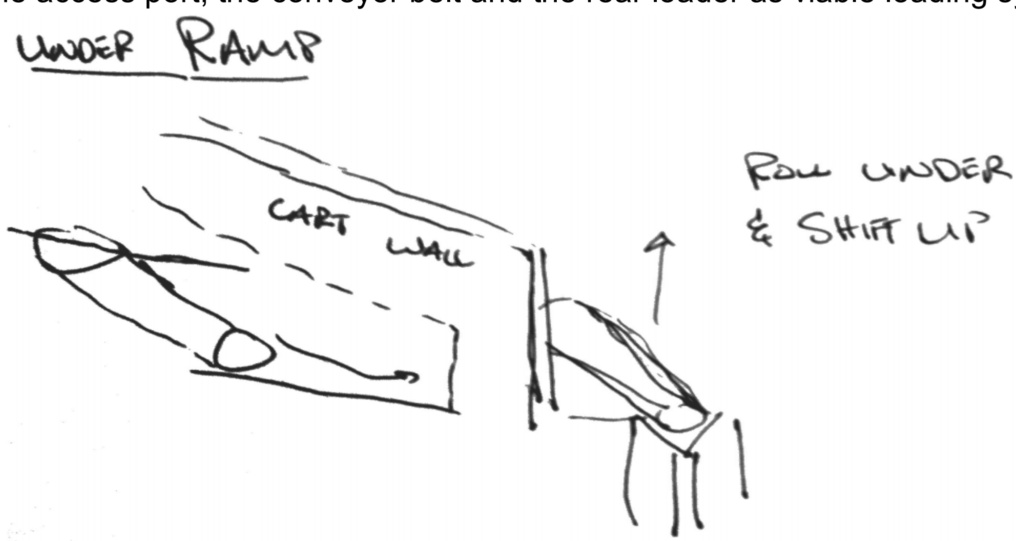

Figure 13: Underneath Ramp Concept Sketch

The underneath ramp requires a hole in the side of the machine. The ramp will roll the pipe through this hole and onto the support system. The ramp will load relatively quickly and not take up very much space. It is also very durable and repair will not be difficult upon it. The main negative with a ramp is the worry that the pipe will roll out of control, the speed of it is hard to control as it is dependent upon the slope of the ramp. It is a simple mechanical solution to the problem with no electronics needed with it.

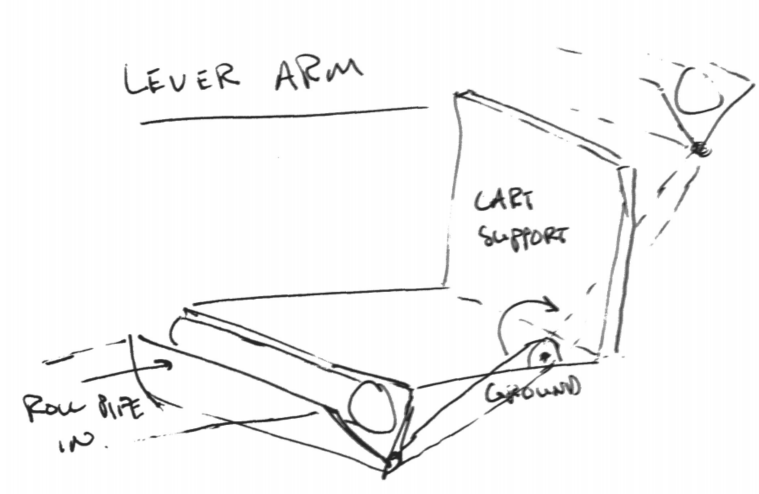

Figure 14: Lever Arm Concept Sketch

The lever arm (Figure 14) will be attached to the side of the machine and will have multiple hands which will support the pipe. This design does not require supports as the arms double as the support. A forklift will place the pipe into the lever arm, and then it will be actuated into place so the pipe can go over the mandrel. The lever arm will be bulky and require a controls system to ensure that it moves the same way every time.

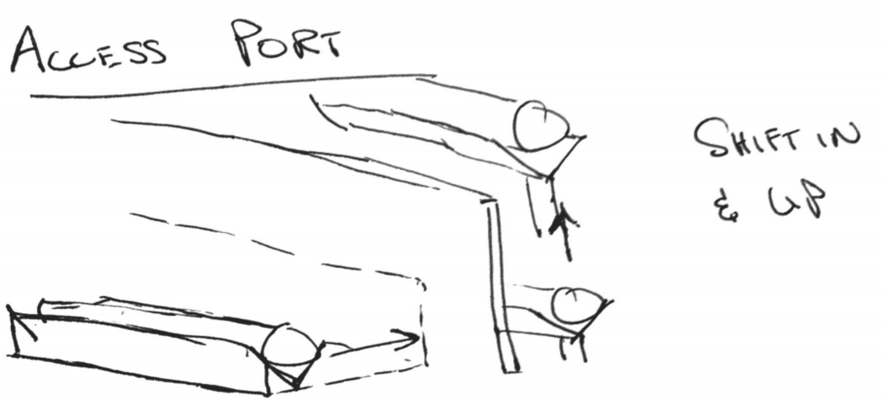

Figure 15: Access Port Concept Sketch

The access port (Figure 15) is a drawer that will open up from the machine, and the pipe will be loaded into this. The port will then close and move the pipe over the supports. This design will be very bulky with a motor that will actuate the port open and closed. It does allow for very easy loading and will not take up much space as the port will be built into the machine.

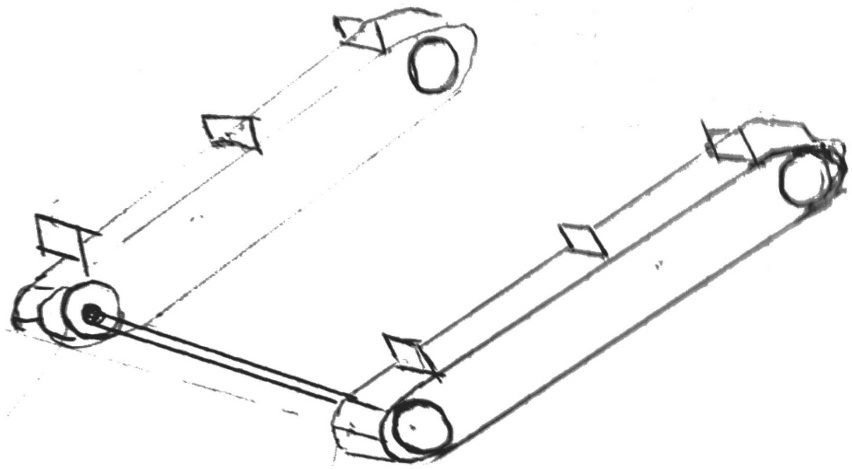

Figure 16: Conveyor Belt Concept Sketch

The conveyor belt (Figure 16) is similar to the ramp in the fact that it will load onto supports. The conveyor belt will need to be rated for the weight of the pipe and requires a motor to power it. The conveyor belt will provide a controlled motion of the pipe unlike the ramp, this makes it a safer and more reliable option. It also allows for placement of multiple pipes onto the belt to minimize loading time.

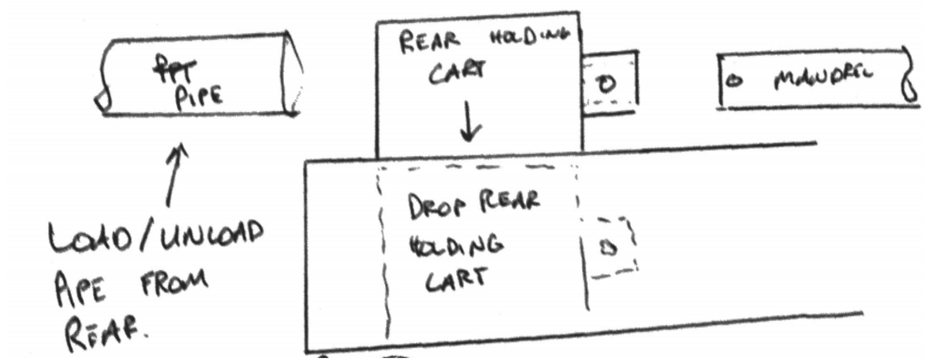

Figure 17: Rear Loader Concept Sketch

The rear loader (Figure 17) is the only loading device that does utilize the side of the machine to load the pipe onto the mandrel. Instead it will be placed above the tailstock and have a gripper for the pipe. The tailstock will then move it forward so that the pipe will slide over the mandrel. The tailstock will then move back and raise up so it can connect onto the mandrel. After this the extrusion process can occur. This concept is the only one that can load from the back. It minimizes loading time and also makes unloading easier.

Feed System

The feed system is what moves the mandrel back and forth along the machine. The pipe that will be extruded fits over the mandrel and is then pressed in between the

dies at the die box. The feed system then forces the pipe through the die box and the process of extrusion commences. Currently the machine at PCC Rollmet is fed by a large power screw. The benefits of this being a huge intrinsic gear ratio, roughly 100:1, which translates to a lot of available power for extrusion. The drawback of this design is naturally a very slow feed rate. We want to design a new feed system that has a faster feed rate to decrease overall cycle time. Figure 18 below showcases some of the original ideas we came up with at the start of brainstorming.

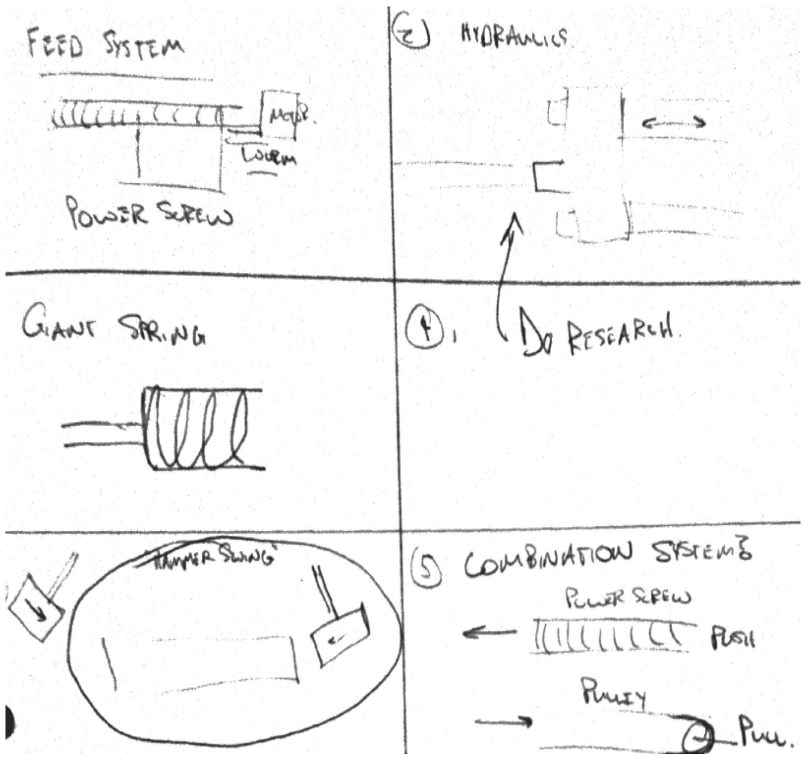

Figure 18: Logbook Sketches of Feed System Brainstorming

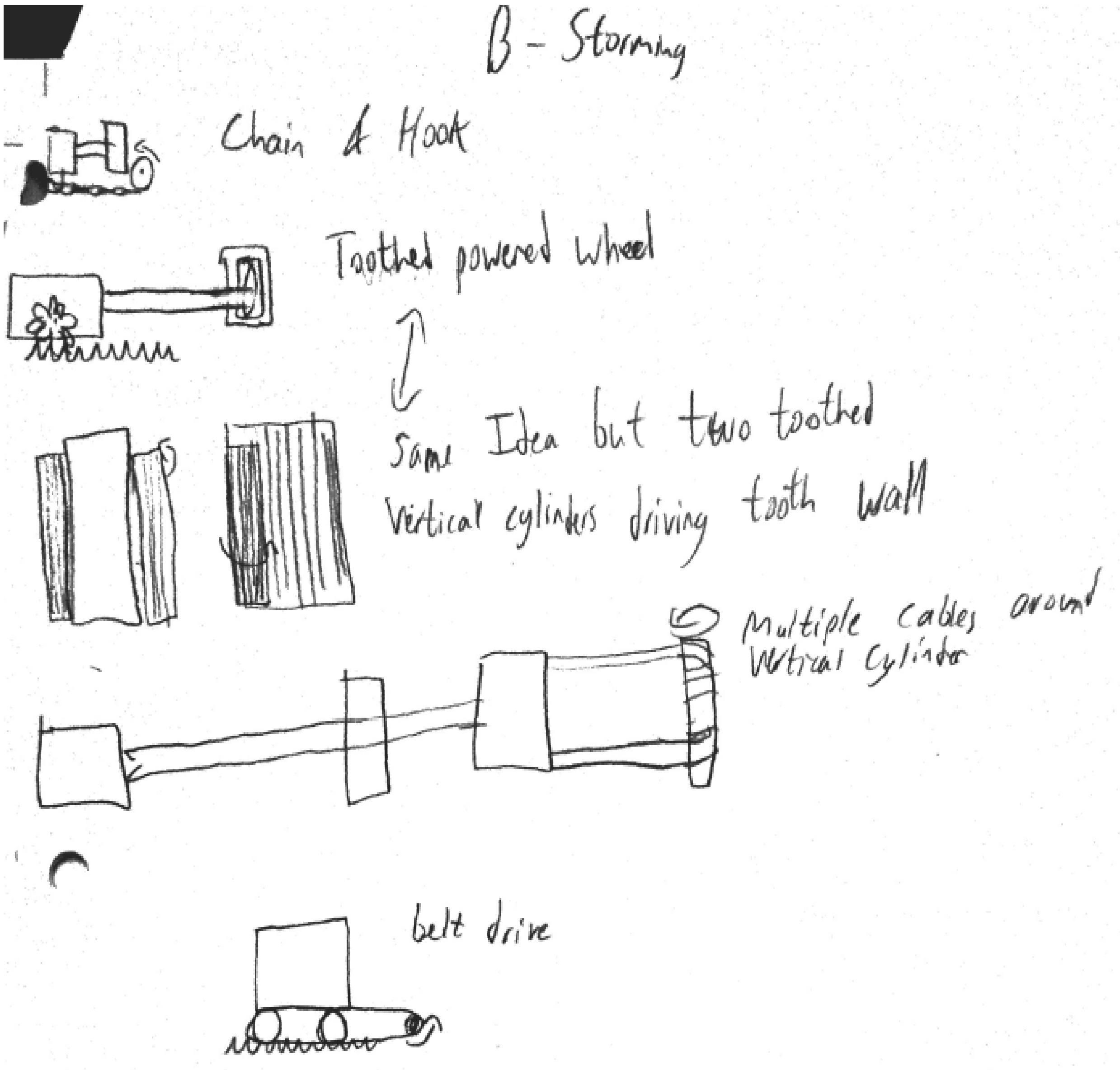

Figure 19: Logbook Sketches of Feed System Brainstorming

Figure 19 depicts some of the early sketches we came up with during brainstorming. These ideas all came out as very rough and slowly more analysis was done upon them. Soon after brainstorming we realized that these ideas must be able to move the feed system in both directions up to 40 feet which is the working stroke of the machine. It must also provide enough power to move the feed box and provide a large enough load to push the pipe and mandrel through the die box.

Table 5: First Pugh Matrix of Feed System with Power Screw as Datum

Concepts								
Needs	Feed Screw	Chain Drive	Toothed Power Drive	Hydraulics	Cable	Belt	Combination Feed/Cable	Linear Actuator
Speed of Return	D	+	+	+	+	+	+	+
Gear Box Needed	A	-	-	S	-	-	-	S
Reliability	T	S	S	S	S	S	S	S
Durability	U	-	-	-	-	-	-	-
Repair	M	+	+	+	+	+	-	+
Size	T	S	S	S	S	S	-	S
Capitol Cost		S	S	S	S	S	-	S
Sum +	0	2	2	2	2	2	1	2
Sum -	0	2	2	1	2	2	5	1
Total	0	0	0	1	0	0	-4	1

After finishing the brainstorms and coming up with multiple feasible ideas, we proceeded ahead with our first Pugh matrix, shown in Table 5. This matrix used the power screw as the datum. After completing the decision matrix we discovered that the hydraulics and linear actuator system rated the highest.

Table 6: Second Pugh Matrix of Feed System with Hydraulics as Datum

Concepts								
Needs	Hydraulics	Chain Drive	Toothed Power Drive	Cable	Belt	Combination Feed/Cable	Linear Actuator	Feed Screw
Speed of Return	D	+	+	+	+	+	S	-
Gear Box Needed	A	-	-	-	-	-	S	S
Reliability	T	S	S	S	-	S	S	S
Durability	U	+	+	+	+	+	S	+
Repair	M	+	+	+	S	-	S	-
Size	T	S	S	S	S	S	S	S
Capitol Cost		S	S	S	S	-	S	S
Sum +	0	3	3	3	2	2	0	1
Sum -	0	1	1	1	3	3	0	2
Total	0	2	2	2	-1	-1	0	-1

For the second Pugh matrix (Table 6) we choose hydraulics as the datum and this time cable drive, chain drive and tooth powered drive scored the highest.

Table 7: Decision Matrix for Feed System

Design Criteria	Speed of Return	Power	Repair	Duribility	Reliability	Size	Cost	Overall Score
Weight	0.2	0.18	0.16	0.14	0.12	0.1	0.1	1
Power Screw	5	10	7	10	10	8	8	8.12
Hydraulics	9	10	6	9	9	6	7	8.2
Chain Drive	8	7	8	7	9	8	10	8
Cable	10	6	10	8	9	8	6	8.28
Belt	8	5	8	6	9	8	10	7.5

Finally a decision matrix (Table 7) was done for designs that we deemed feasible after the Pugh matrix. The highest weights were put upon the speed of return of the feed system, the available power and the ease of repair. Size and cost were the least important aspects of selecting the feed system. In the end the Cable design scored the highest, followed by the hydraulics and then the power screw which is what PCC Rollmet currently uses.

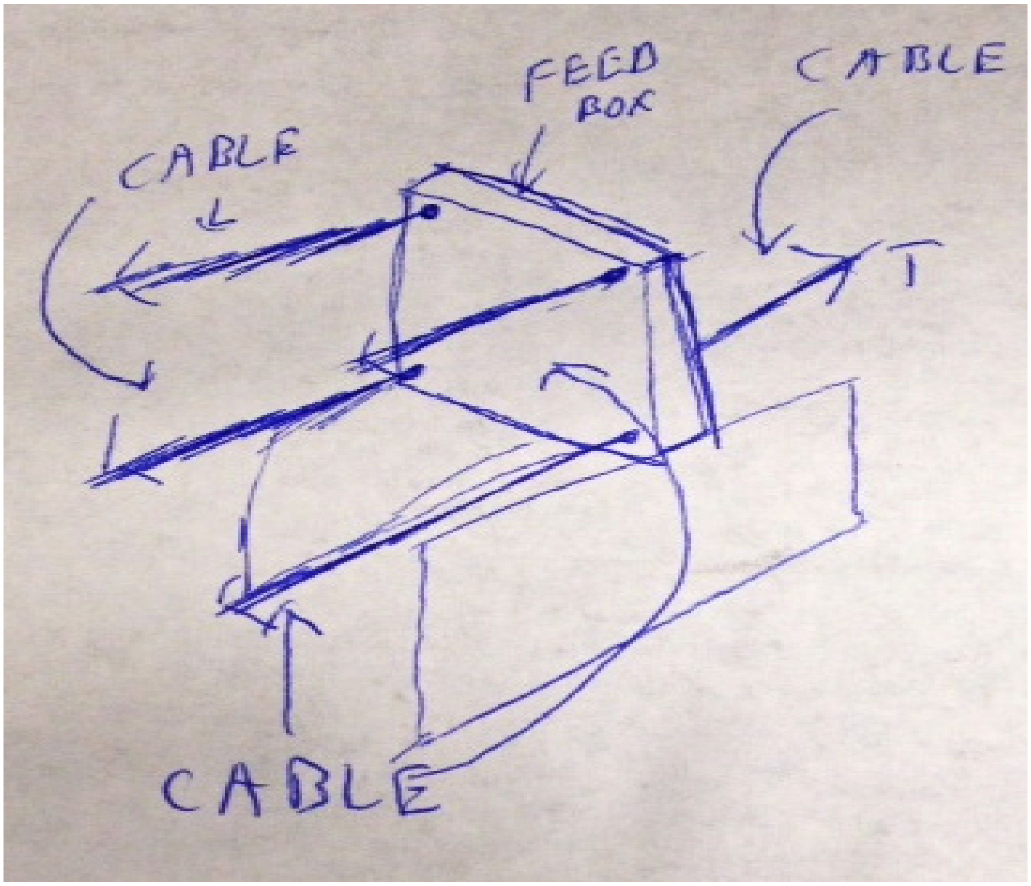

Figure 20: Cable Feed System

The cable design functions like a sideways elevator. Four spools of cables will be placed within the die box and connected to the feed block. They will be connected at equal distances away from the center of mass to get rid of the bending moment as shown in Figure 20. The spools will then have a motor attached that rotates them and pulls the feed box towards the die box. For the return mechanism here, there will be a cable attached to the rear part of the feed box and it will pull the feed box away from the die box. This design offers very high return speeds as well as the ability to be concealed efficiently within the die box so the spools don't take up much space. Also if more power is needed, more cables can be added; however, it is a very complex system and will require a controls system to ensure that each cable is pulling with an equal amount of force.

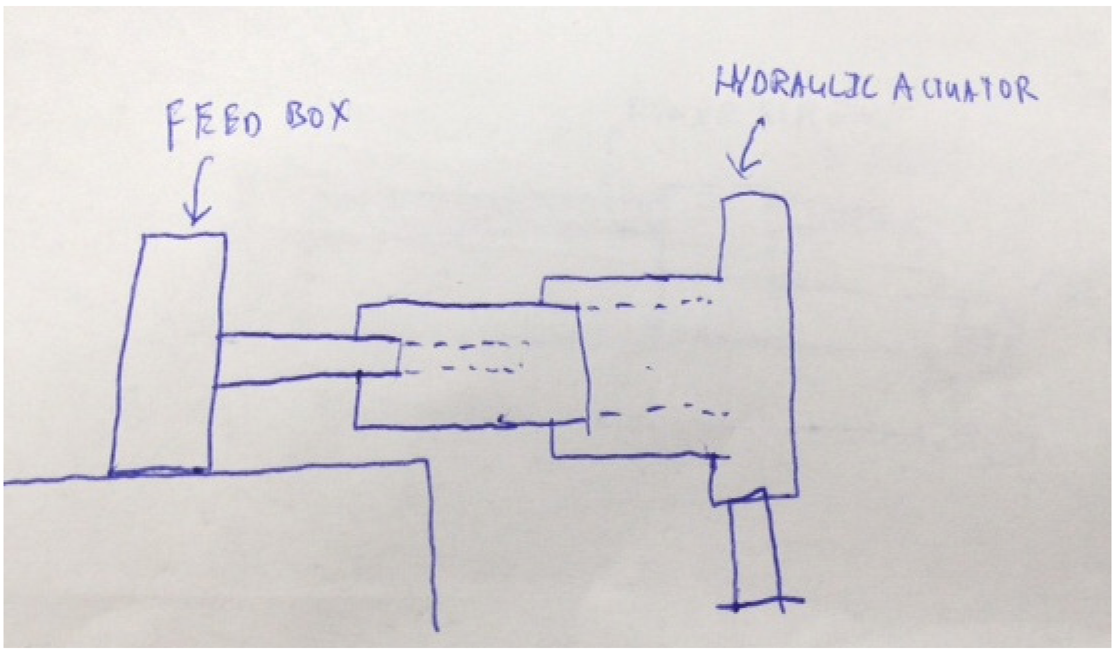

Figure 21: Hydraulic Feed System

The hydraulics design shown in Figure 21 is extremely complex as it involves a multi-stage hydraulic system. These sorts of systems require some sort of mechanical synchronization system, otherwise there would be large shock forces to the system every time a stage fully extended. The hydraulic system must be able to have a working stroke of 40 feet and would take a large amount floor space. The idea has enough potential to be considered for use at PCC Rollmet, but seems to have more drawbacks than benefits.

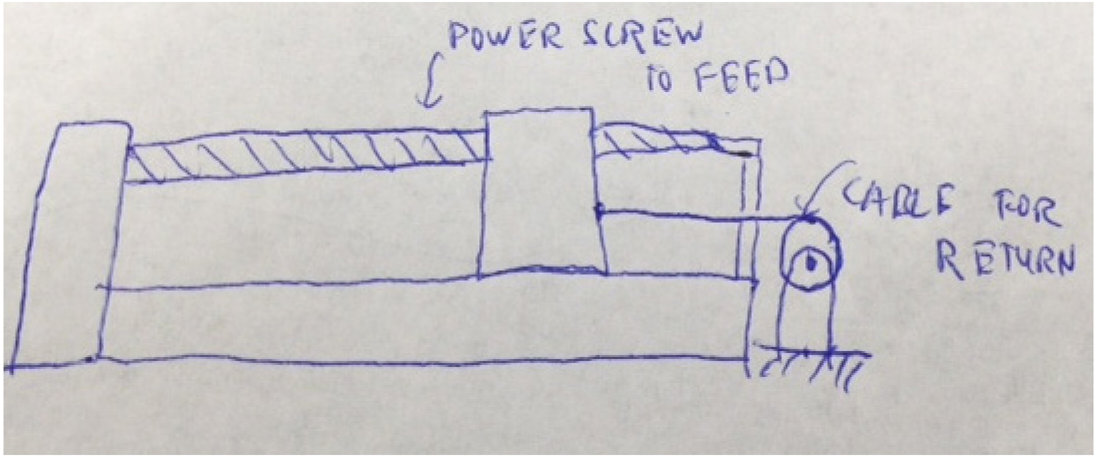

Figure 22: Power Screw and Cable Return Feed System

The power screw design is very effective at moving the feed box toward the die box and extruding the pipe. The drawback of this design is that it is slow, which is appropriate for the rolling portion of the process, but inconvenient on the return when no work is being done. To combat this we have looked into upgrading this design to make returning the feed box to its original location much faster. To do this we have considered the possibility of disengaging the feed box from the power screw and adding a cable return system (Figure 22). The cable system would solely be used to move the feed box away from the die box and would lead to a decrease in cycle time. At this point we do not have enough understanding of the operation of the current drive system to propose ideas of how to disengage the drive system to allow for cable return. If this could be achieved, though, the idea might provide a cost effective upgrade to the current system and benefit even the current systems.

Support System

The support system is not actually an entire subsystem. Supports are needed in order to aid many of our designs for the loading system. Currently there is no need for supports, because the crane fills this roll by holding the pipe while it is centered on the mandrel and then pushed forward to the die box where it is secured. Many of our loading designs will instead load the pipe onto the supports which will position and hold the pipe in such a way that the mandrel can easily slide over the pipe. We are venturing to say that the time saved by switching away from the crane is worth the addition of a support system.

The need for supports was realized after all the decision matrices for a loading system were completed. Without supports, some of the loading systems are not feasible. We proceeded to brainstorm to generate different types of supports and discuss the feasibility of our designs based on this new shortcoming. Three main ideas came up and these are shown below in Figure 23.

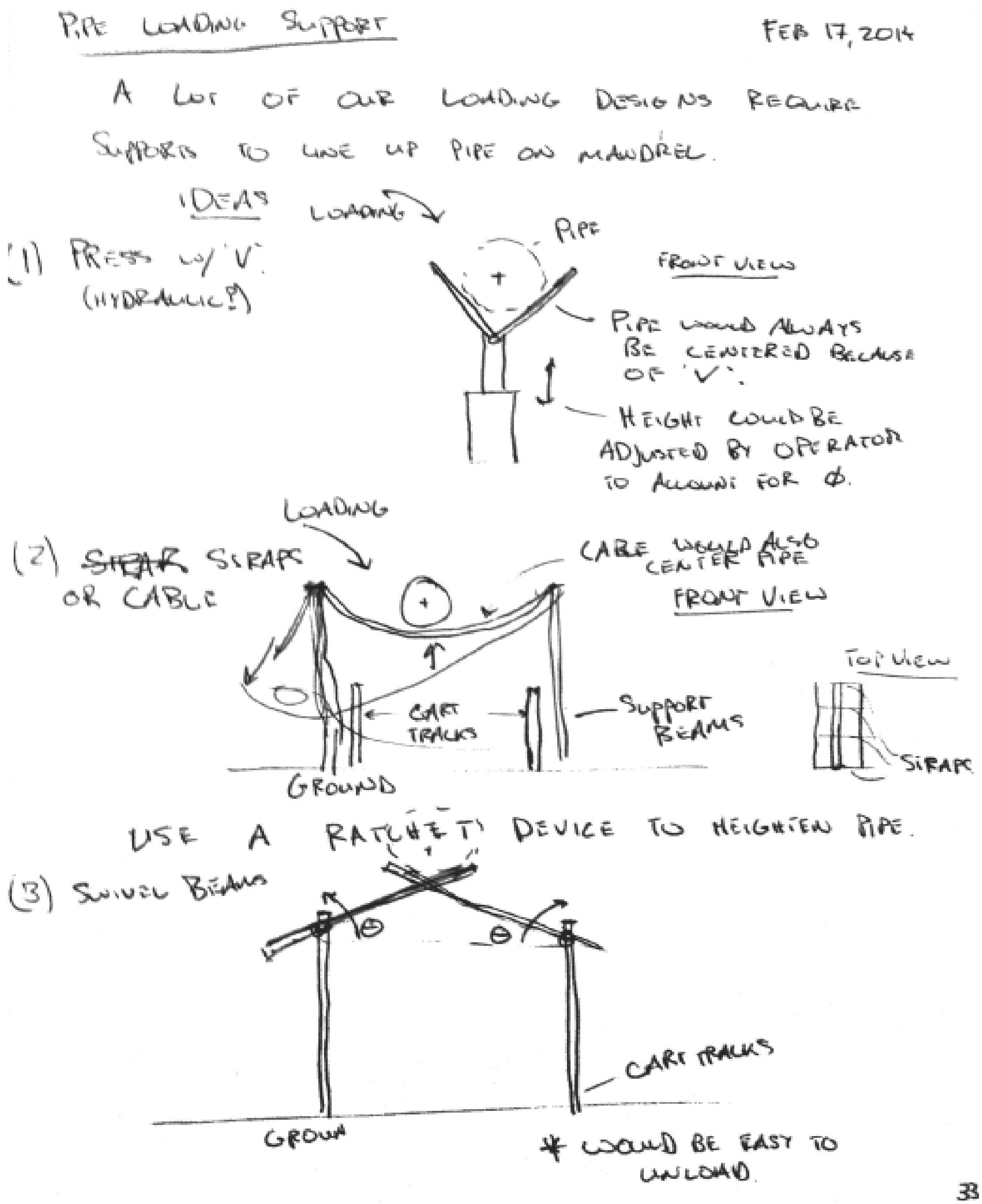

Figure 23: Logbook Sketches of Support Mechanisms Brainstorming

After the brainstorm a discussion occurred over the different types of supports we came up with. The discussion focused mostly on the feasibility and simplicity of the supports as there was not enough time to do a more in-depth analysis prior to the preliminary design report. The V support, (1) in Figure 23, was based on the idea that a cylindrical pipe would always be centered and therefore no horizontal positioning is needed for any sized pipe. When the diameters of the pipes vary though, the centering

height will vary and this would require a mechanism to allow vertical positioning to be manipulated. Hydraulic presses would allow the support to be raised and lowered with a pipe on it and would allow for ease of unloading but would also demand a higher capital cost and maintenance.

Another thought was to have a mechanical locking device that would allow the operator to select a height before the pipe was introduced. In an attempt to combat these difficulties of vertical positioning, the strap,(2) in Figure 23, and swiveling beam, (3) in Figure 23, ideas were generated. These ideas allow pipes to be more easily positioned vertically while still maintaining a horizontal center. These designs have concerns analytically however as there are large bending moments created so it was determined that the “V” single support would be the easiest to design, therefore it was chosen to be used in conjunction with the conveyor belt loading system.

We encountered yet another problem with the support system before we decided that it was appropriate to attempt to implement in our design. We had not considered the function, which the crane performs rather easily, of pushing the pipe up the mandrel and into its locked position on the die box. This problem we can only justify in conjunction with a feed system that has a superior return speed than the current system. With a quick return system we would be able to push the pipe up to the feed system as the crane does. This process is shown in Figure 24 below.

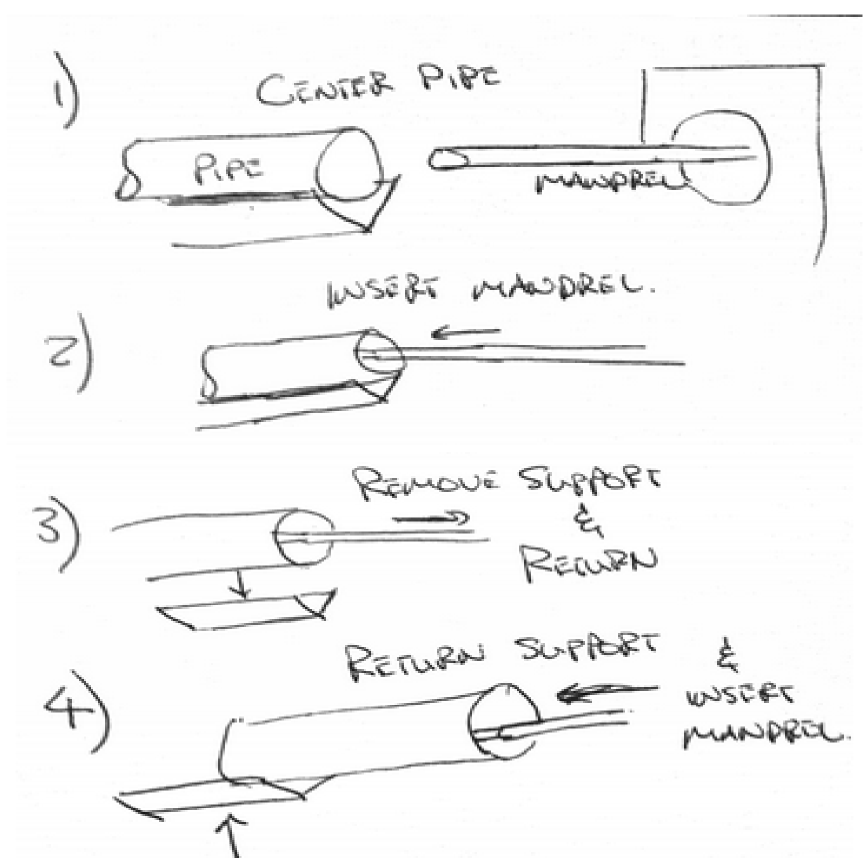

Figure 24: Logbook Sketches of Pipe Moving

Lubrication System

The lubrication system consists of three distinct portions. The three areas that must be lubricated are the outside of the mandrel, the inside of the pipe and the outside of the pipe. All three of these areas must be lubricated otherwise the operation will not run smoothly. The current method of lubrication involves the operator filling a Folgers cup with Mobilgear 600 XP 220 oil and pouring it on the mandrel, inside and outside of the pipe. To lubricate the inside of the pipe, the operator pours the oil into the pipe while it is still on the crane so that he can tilt it up and allow the oil to stream down the whole length of the pipe. Our initial brainstorming, shown below in Figure 25, helped us identify the need for a system that can lubricate all the surfaces (mandrel, inside of pipe, outside of pipe) as well as an operator with a Folgers cup.

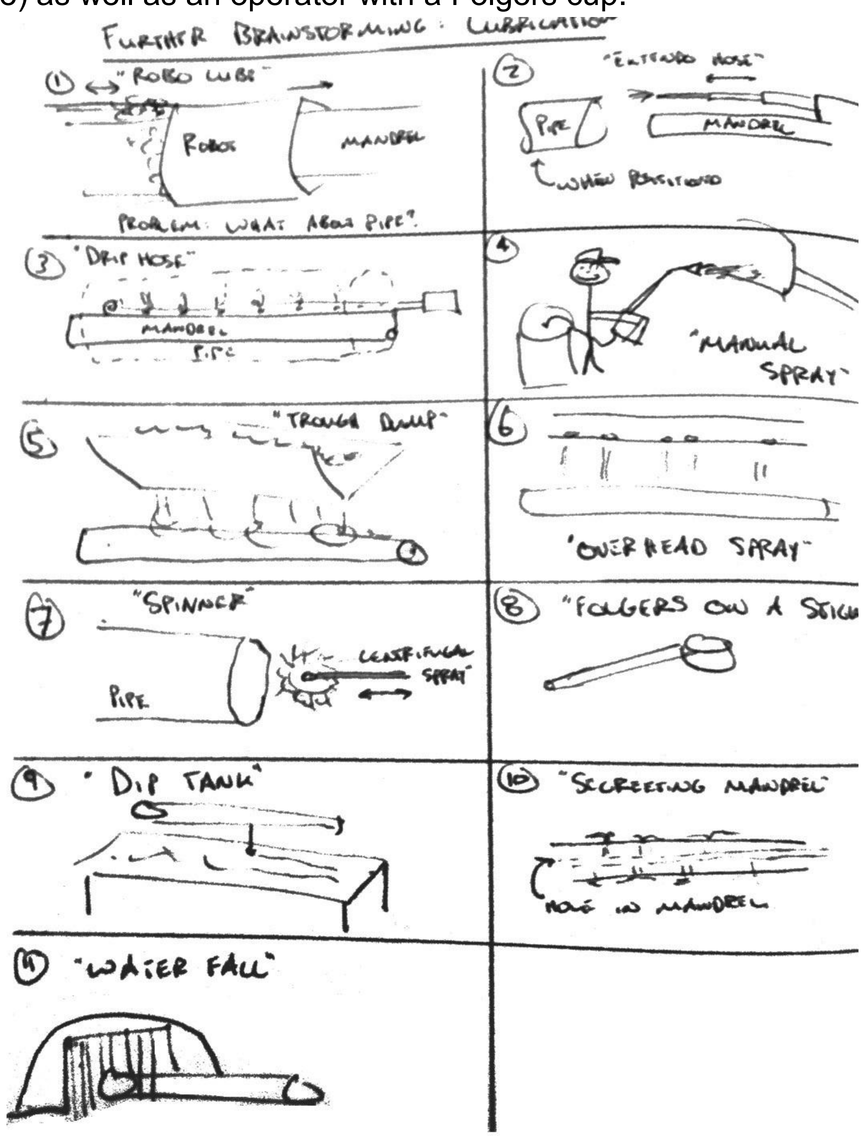

Figure 25: Logbook Sketches of Lubrication Brainstorming

We came up with a lot of creative ideas for quick and effective lubrication, but many of our ideas addressed only one or two of the needed surfaces. For example. Idea number 3, the drip hose, addresses lubricating the mandrel and inside of the pipe using a roll out spray hose that could fit inside the pipe while it is on the mandrel. Some of our ideas also became exceedingly technical in hopes of very efficient lubrication. In order to begin to separate out the truly feasible ideas, we generated a set of Pugh matrices which are shown below in Table 8 and Table 9.

Table 8: First Lubrication Pugh Matrix with Folgers Cup as Datum

Concepts										
Needs	Robot	Overhead Spray/Drip	Folgers on a Stick	Manual Spray	Spinner	Waterfall	Dip Tank	Mandrel Secretes Oil	Plastic/Low Friction Bushings	Lube with Loading
Ease of Operation	+	+	+	+	+	+	-	+	+	+
Speed of Process	+	+	+	+	+	+	-	+	+	+
Efficiency	+	+	S	+	+	+	+	+	+	+
Reliability	+	+	S	+	+	+	+	+	+	+
Repair	-	-	S	-	-	-	+	-	-	-
Capitol Cost	-	-	S	S	-	-	-	-	-	-
Size Effects	S	-	S	S	-	-	-	+	-	S
Mandrel	S	S	S	S	-	S	-	S	S	-
Outside Pipe	S	S	S	S	-	S	S	-	S	S
Inside Pipe	-	-	S	S	S	-	S	S	S	S
Durability	-	-	S	-	-	-	-	-	S	-
Sum +	4	4	2	4	4	4	3	5	4	4
Sum -	4	5	0	2	6	5	5	4	3	4
Total	0	-1	2	2	-2	-1	-2	1	1	0

Our first Pugh matrix (above in Table 8) compared all our ideas from brainstorming to the current method of lubrication, using a Folgers cup. From this Pugh matrix the top design was the overhead spray along with the Folgers on a stick.

Table 9: Second Lubrication Pugh Matrix with Folgers on a stick as the Datum

Concepts										
Needs	Folgers on a Stick	Robot	Overhead Spray/Drip	Manual Spray	Spinner	Waterfall	Dip Tank	Mandrel Secretes Oil	Plastic/Low Friction Bushings	Lube with Loading
Ease of Operation	D	+	+	+	+	+	-	+	+	+
Speed of Process	+	+	+	+	+	+	-	+	+	+
Efficiency	A	+	+	S	+	S	+	+	+	+
Repair	+	-	-	-	-	-	-	-	-	-
Capitol Cost	T	-	-	S	-	-	-	-	-	-
Size Effects	-	-	-	S	-	-	-	-	-	-
Mandrel	U	S	S	S	-	S	-	S	S	-
Outside Pipe	S	S	S	S	-	S	S	-	-	S
Inside Pipe	M	-	-	S	S	-	S	S	S	S
Durability		-	-	-	-	-	+	-	S	-
Reliability		+	+	+	+	+	+	+	+	+
Sum +	0	4	4	3		3	3	4	4	4
Sum -	0	5	5	2	4	5	6	5	4	5
Total	0	-1	-1	1	6	-2	-3	-1	0	-1

A second Pugh matrix was then done with the Folgers on a stick as the datum (above in Table 9). This second Pugh matrix came out with the manual spray as the top design. From here we moved on to doing a decision matrix to apply weights to our design criteria and sort the designs out accordingly.

Table 10: Lubrication Decision Matrix

Lubrication Subsystem								Location of Lub				
Design Criteria	Ease of Operation	Speed of Process	Efficiency	Reliability	Repair	Capitol Cost	Size Effects	Mandrel	Outside Pipe	Inside Pipe	Durability	Overall Score
Weight	0.11	0.12	0.07	0.07	0.07	0.05	0.05	0.12	0.12	0.12	0.1	1
Robot	6	7	10	10	6	5	8	5	5	0	6	5.17
Overhead Spray	9	10	7	10	7.5	7.5	7	5	5	0	8	5.83
Folgers on a Stick	8	8	7	9	10	10	10	5	5	5	10	5.86
Spinner	6	7	8	10	6	6	8	0	0	5	6	3.88
Water Fall	10	10	10	10	8	8	9	5	5	0	8	6.31
Low Friction Bushing	7	5	7	10	10	9	6	5	0	5	10	4.61

The decision matrix (above in Table 10) had the heaviest weight on the speed of process along with the ability to lubricant the different surfaces. The least important aspects were the capital cost and size effects. Size effects are how much space the subsystem would take up. From this matrix the top three lubrication designs are: Waterfall, Overhead spray and Folgers on a stick. However, one thing that is not shown is that we also want to implement part of the lubrication system into the loading system. The feasibility of this is evaluated when our final design is produced.

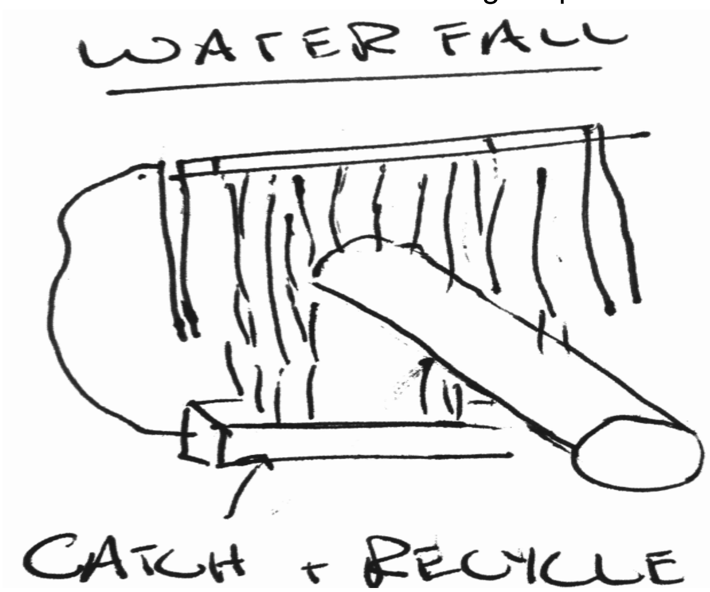

Figure 26: Waterfall Concept Sketch

The top idea was the waterfall (Figure 26). The waterfall is a lubricant drip system. It is positioned in front of the dye box and can successfully lubricate the outside of the pipe and mandrel. The waterfall will drip on top of the mandrel as the mandrel passes under it and then onto the pipe as the pipe passes through it. This is highly advantageous as the motion of the mandrel and pipe do not have to be stopped as lubing occurs during the operation. Another advantage is that the waterfall will have a large tub underneath it to collect the excess lubricant from the operation. This high recycling rate makes the waterfall an efficient and reliable option as there will always be enough lubricant on the pipe and mandrel. It also takes up very little space and is not very complex as the machine operator will just press a button to start and stop the

waterfall.

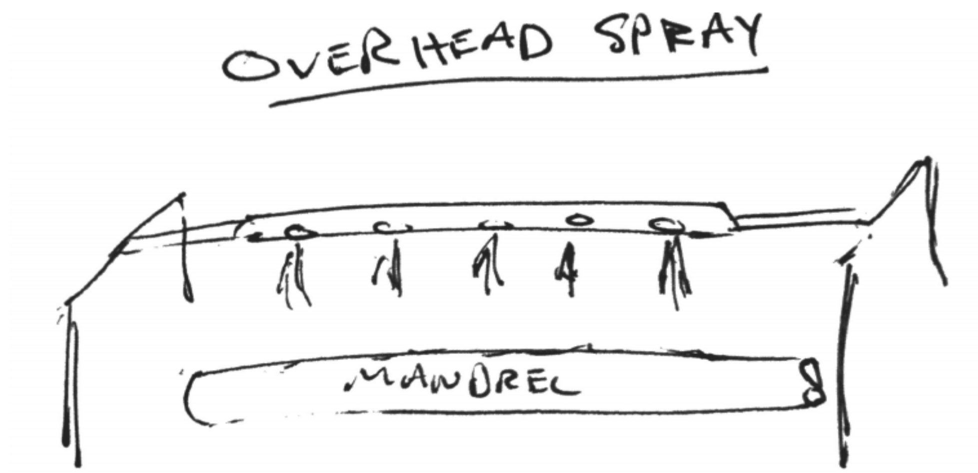

Figure 27: Overhead Spray Concept Model

One alternative idea is the overhead spray system (Figure 27). The lubricant used by PCC Rollmet is actually too viscous to spray. Therefore this system will actually drip the lubricant onto the mandrel and the outside of the pipe. This is essentially the waterfall; however it is positioned in a different direction. Instead of having to move the pipe through the overhead spray, this system will activate once over the mandrel and once over the pipe and then it will turn off. This will lead to a single quick dispensing of the lubricant as opposed to the waterfall which will dispense the lubricant for a long time. The overhead spray system will require less recycling of the lubricant, however it is much bigger.

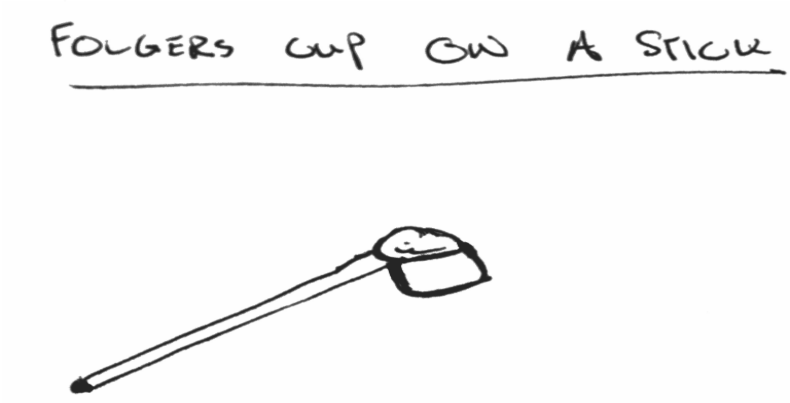

Figure 28: Folgers Cup on a Stick Concept

The final alternative idea is essentially an upgrade over the current lubrication system PCC Rollmet uses. Currently a Folgers coffee cup is used to pour lubricant on the inside of the pipe, outside of the pipe and the mandrel. This requires the operator to stand up on the machine and lean over the pipe which is not the safest act. We believe

adding a handle to the Folgers cup (Figure 28) will increase the speed at which lubricant can be poured and it'll make this process safer by allowing the operator to stand farther away from the pipe while applying the lubricant.

Design Development

The different subsystems were combined to form the final overall design of the cold-roll extrusion machine. Not all of these subsystems were the highest ranked after the decision matrix; however, after much discussion the loading system chosen was the conveyor belt onto a set of supports. The feed system was the cable system and the lubrication system was a waterfall device along with side-spraying lubricant. The side-spraying lubricant is integrating the lubrication system into the loading system. The pipes will be waiting to get loaded onto the mandrel, and during this time, two side spraying arms will come out and spray the inside of the pipe. We performed a feasibility analysis done upon it. The feasibility analysis included a solid model to ensure everything would fit where it was supposed to fit. The CAD model is very basic but it shows that the subsystems can be combined without any interference from the machine or from each other.

Loading System

The loading system that we decided upon after the first quarter of work was a conveyor belt that will be rated to hold the weight of the pipes that will sit on it. It will have a metal stand that a forklift will be able to load the pipe into. This can be seen in Figure 29. The conveyor belt is then angled upwards to carry the pipe to the supports. It will drop the pipe off at the supports and while that pipe begins to get extruded. The next pipe will be loaded onto the conveyor belt and it'll wait until the pipe finishes being extruded. This will reduce cycle time and also create a time in which lubing can be done without changing the overall run time of the system. The tricky part of lubrication is lubing the inside of the pipe. While the pipe is sitting on the conveyor belt and waiting for the pipe in front of it to finish extruding, the operator will take this time to turn on side sprayers mounted on the conveyor belt. The side sprayers will spray the inside of the pipe with lubricant. However, we still need to lubricant the outside of the pipe and the outside of the mandrel.

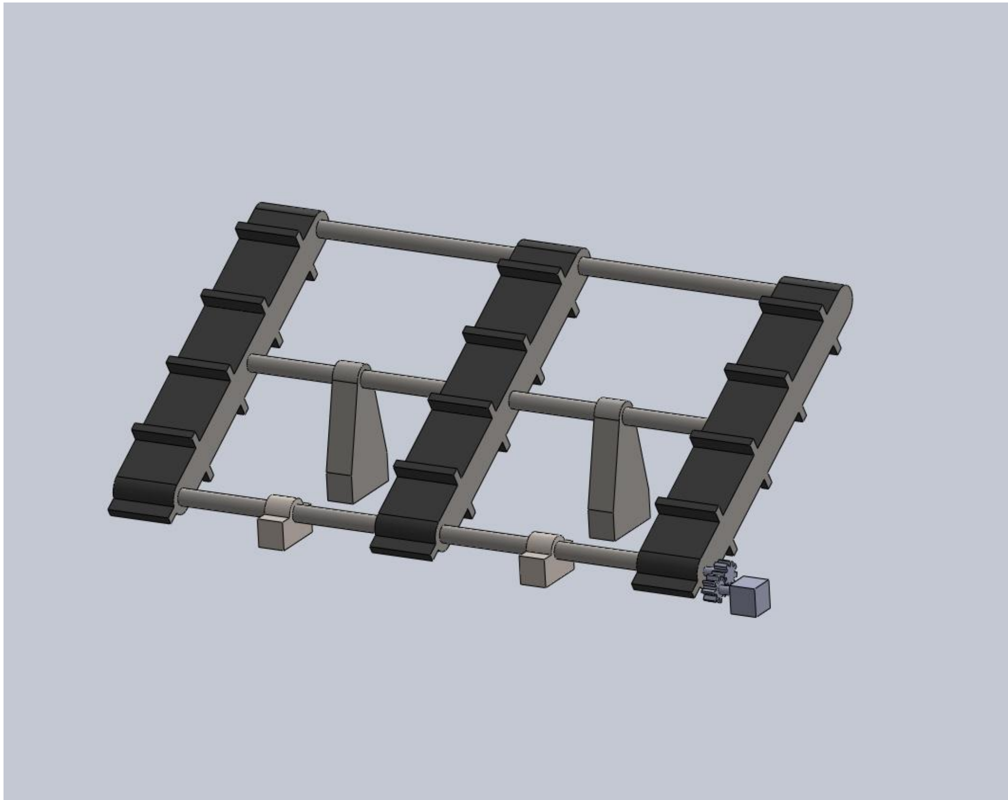

Figure 29: Conveyor Belt CAD Model

We presented the final design (from above) to Dr. Roberts and discussed each concept design for the different subsystems. After discussing each concept Dr. Roberts provided feedback which included his opinion on the feasibility of each design. The first thing we discussed was the loading system. Dr. Roberts was not a big fan of the conveyor belt loading system. After discussing this it was agreed that a lever arm would work better for the purpose of loading. This is because the lever arm would act as its own support, not have the pipe drop down from the conveyor belt and it is easier to load with. We agreed with Dr. Roberts and then got the okay to do an in-depth design of the lever arm. Dr. Roberts also specified that the pipe must only be loaded by the lever arm and that the sky crane will do the unloading for this new machine while the lever arm gets ready to load the next pipe. After this we discussed the feed system and this is where it got interesting. The loading system that we decided upon with Dr. Roberts was changed to a lever arm and we now proceeded into another iteration of design for a lever arm. The main problem remains for our design: we need to get a 2000 lb pipe over the dead zone where the rails for the support cart run. In addition, we are no longer aided by supports to help hold the load of the pipe, but now the lever arm must become the support. In our early brainstorming stages we had 3 basic ideas for how to load a pipe with a lever arm which are shown below in Figure 30. The first concept uses two moving links. A) The base and bottom gripping jaw for grabbing the pipe can slide and in B) both

gripping jaws can move. In the second concept, the pipe could be pushed up vertically to the correct height and then moved horizontally into position. The third and final concept was to have a single degree of freedom mechanism push the pipe on a centerline until it reached the loading position.

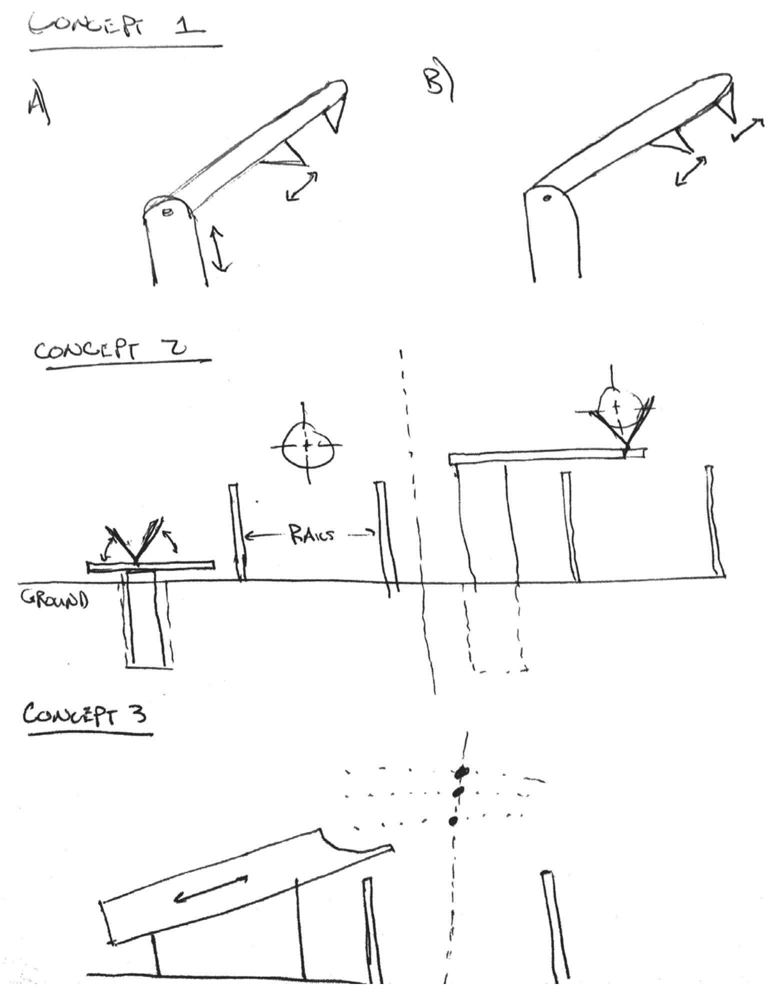

Figure 30: Initial Lever Arm Concepts

We preferred Concept #3 because of the simplicity of a single degree of freedom. The single degree of freedom is feasible if the mechanism that grips the pipe is self-centering. If such a pair of grippers can be designed, then the pipe could be pushed along a single axis until its center became concentric with the mandrel's center. We checked the feasibility of geometry with the current machine's set up and found the angles it to appear reasonable. Note the geometry below in Figure 31, the center line below minimum as it touches the support cart rail.

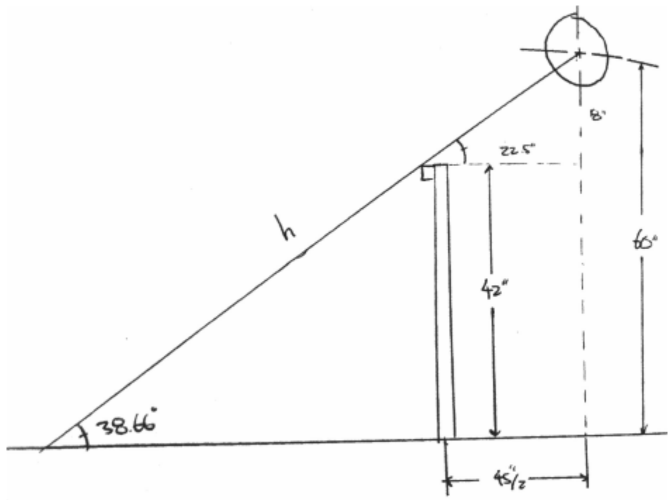

Figure 31: Concept #3 Geometry

With the confidence that this idea might work, and operate more simply than conventional robotic arms with multiple joints, we brainstormed a number of ideas that followed the idea of a single degree of freedom. Our first idea was simply to mount a piston at an angle and push the pipe up a ramp, show below in Figure 32. Our gripper initially took the form of our 'V-support' concept with a strap or other restraining mechanism holding it in the notch. The largest concern with this idea was the bending that the piston might endure. We had to make sure that extending piston arm was stiff enough that it would not deflect out of the centerline and thus out of concentricity with the mandrel. Over the duration of our hydraulic piston research, it became more and more evident that there were no applications in which a piston supported a bending moment. Upon further investigation we found that this was because of the seals that retain the hydraulic in the contact between the two cylinders. When talking to a representative from Parker Hydraulics, we confirmed what we suspected: although some high grade seals exist, there are no applications where a hydraulic piston is subjected to extreme bending moments.

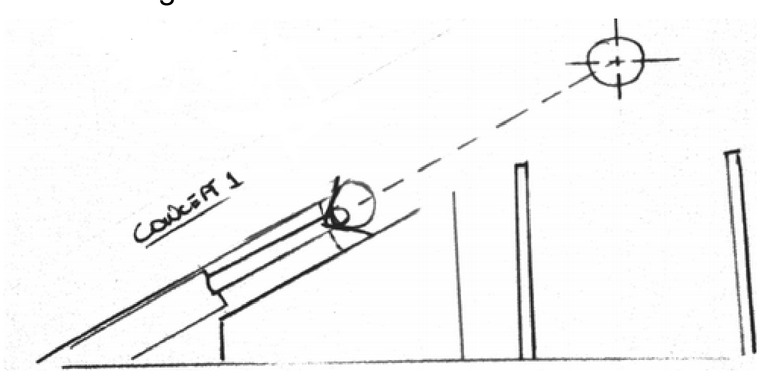

Figure 32: Piston Ramp Push

With this basic understanding of hydraulic piston operation learned, we continued to

brainstorm centerline ideas. In the design show in Figure 33, the hydraulic piston would pull the gripper and pipe up a rail. This idea could work but was stifled by the extreme length needed. Although it is not shown in the diagram, the length of the arm behind the gripper would have to be much longer in order to get the pipe over the rail and up the height of the mandrel.

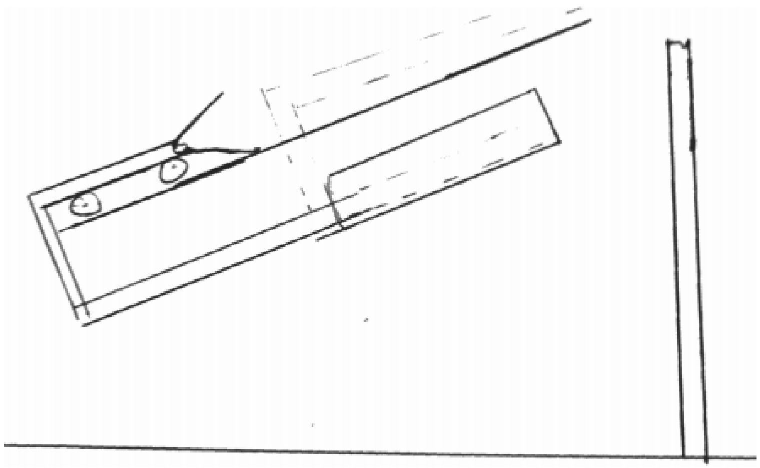

Figure 33: Piston Pull below Gripper

We also had some ideas that attempted to take advantage of geometry to provide a mechanical advantage. The concept show below in Figure 34 shows three different points from loading the pipe onto the arm (1), to full extension and loading the mandrel (3). The concept is essentially the same as the one shown above but it creates a linkage that does not require the hydraulic piston to be as long while still moving the pipe the same distance. The same problem applies here as it did above though; the gripper arm will have to be very long to reach the mandrel.

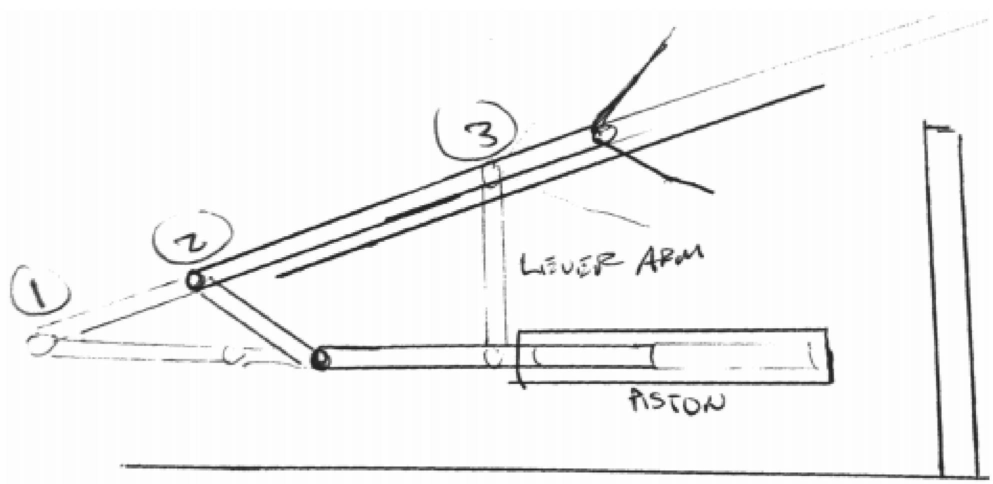

Figure 34: Mechanical Advantage via Geometry

There are a couple key design flaws with all of our previous designs. The first one is the length of the lever arm that holds the gripper. The length in and of itself is not the issue, but the amount of floor space that it requires begins to become an issue. Another problem is the loading of the lever arm. Since the lever arm is facing towards the mandrel, a forklift would have lift a pipe over and down into jaws. Figure 35 shows that a forklift could fit in between two lever arm installments in order to mount a pipe. The rails show would allow the pipe to roll down and onto the V-supports when they are fully retracted. In office hours with Dr. Niku (Robotics Professor at Cal Poly), we discussed the importance of having only two powered lever arms. If we implement more than two lever arms, we introduce the possibility of bending a pipe if any one of the lever arms malfunctions and operates differently than the other two. If we only operate two powered lever arms and one malfunctions, the pipe cannot bend, only the angle will become skewed. There is the possibility of including a third non-powered lever arm that acts only as a guide for the pipe and cannot cause bending in the case of a malfunction. This third non powered lever arm would be helpful for loading longer pipes.

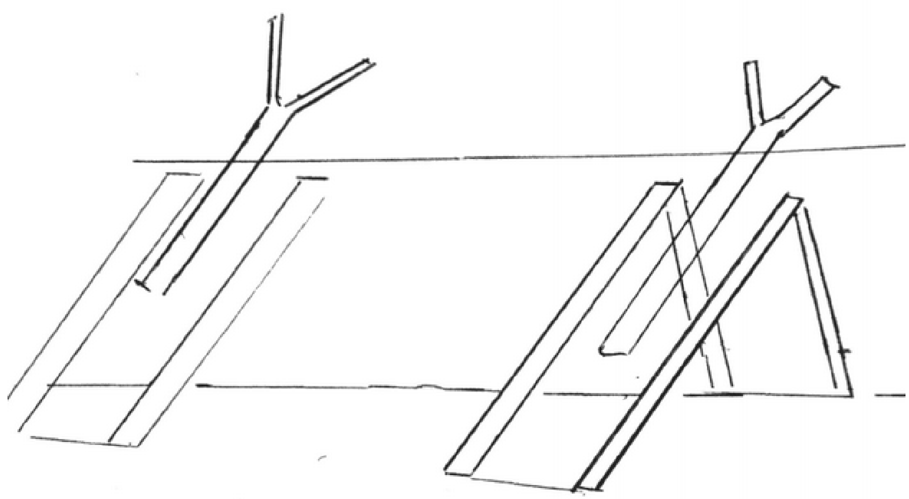

Figure 35: Slider Supports

The next concept that we generated was sparked when we followed our advisor's suggestion to examine how dump truck pistons work. This design, show below in Figure 36, addresses the problems that surfaced in our previous ideas. The mechanism begins at state (1) folded upon itself and as the piston applies pressure it begins to unfold through state (2).The lever arm itself also acts as an extending piston and when it completes its rotation to the centerline, a solenoid will unlock, releasing the lever arm to linear motion. The extendible portion of the lever arm can then push out along the centerline until it reaches concentricity with the mandrel at state (3). This concept maintained the singe degree of freedom centerline push, and the need for only one hydraulic piston. These components are retained while introducing a compact design that faces towards the shop to load pipe, the two major flaws from the previous designs.

Although we were happy with the basic concept of this design, there were still a number of flaws that needed to be addressed.

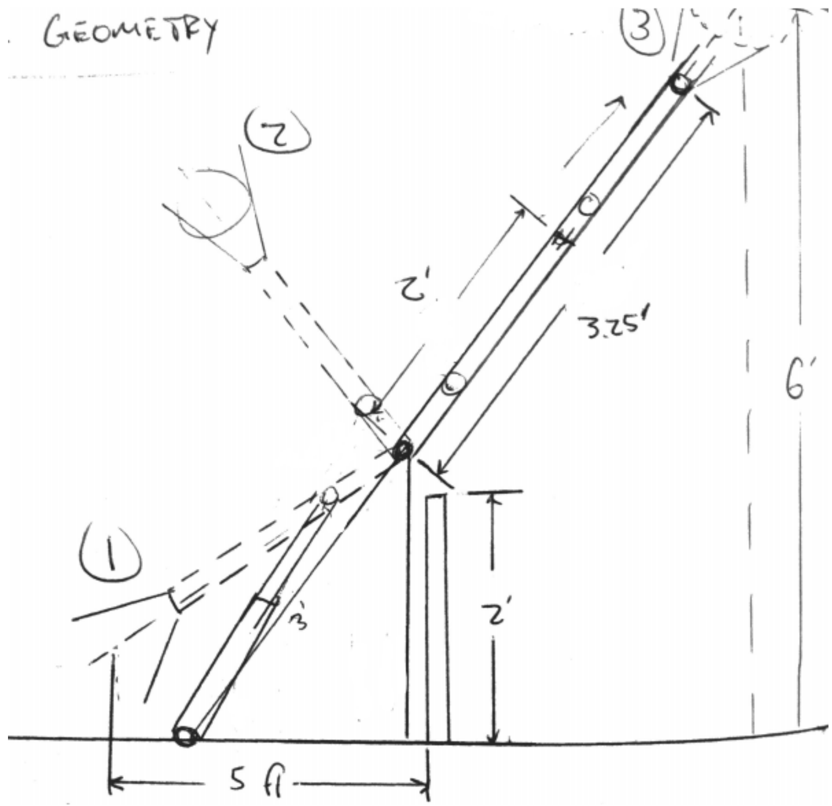

Figure 36: Dump Truck Mechanism

The first flaw that we encountered was discovered when we performed loading analysis for each angle of operation.

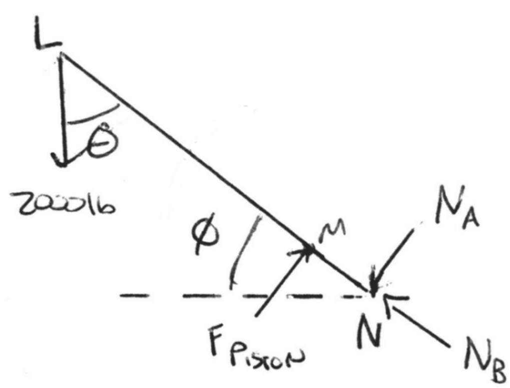

Figure 37: FBD Load Analysis

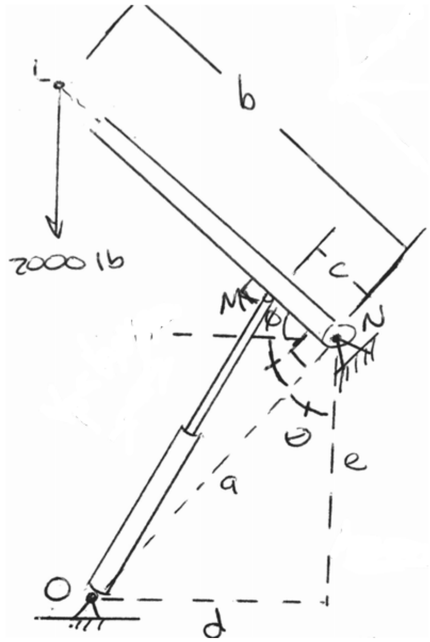

Figure 38: Load Analysis Angle Calculations

Figure 38 depicts the variables we input into a spreadsheet. We used this spreadsheet to optimize the geometry for our loading arm design.

Sample Calculations

$$\sum F_{x'} = 0$$

$$2000\cos(\theta) - N_B = 0$$

$$N_B = 2000\cos(\theta)$$

$$\sum F_{y'} = 0$$

$$-2000\sin(\theta) + F_p - N_A = 0$$

$$\sum M_N = 0$$

$$-2000\sin(\theta) * b = F_{piston} * c$$

Table 11: Load Analysis Angle Variables

Variable	Value	Unit
Weight	1500	lb
a	2.31482464	ft
b	3	ft
c	0.5	ft
d	1.16550982	ft
e	2	ft
g	0.5	ft
h	0.16666667	ft
i	4	ft
j	5	ft
alpha	36.8698976	ft
beta	56.8698976	ft
k	0.41871584	ft
l	1	ft
m	0.27327101	ft
p	1.82194519	ft

With these calculations and variable estimates given in Table 11, we determined the locations of max load. In the graph shown below we observe a large load when phi (the angle between the lever arm and the piston) is equal to 60 degrees. The next jump occurs when phi approaches 180 degrees. We realized that this is occurring because when the arm reaches 180 degrees, it has become in line with the piston and therefore there is no longer a moment between the two joints, only reaction forces on the joints.

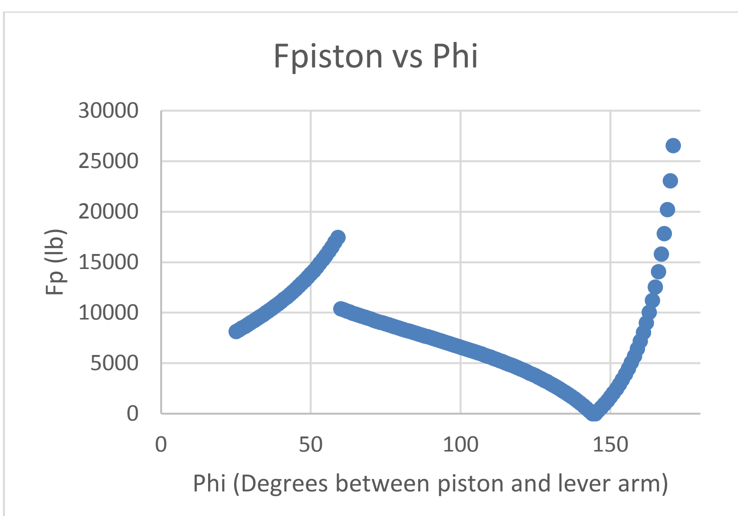

Figure 39: Max Loading of Lever Arm

In order to combat this problem of having the piston and the lever arm in line with each other at 180 degrees, we proposed two ideas. The first was simply to keep the angle of operation of the lever arm less than 180 degrees as shown below in Figure 40. The second was to install a small stub that sticks up from the lever arm that provides the piston with a small moment no matter what angle the lever arm is at.

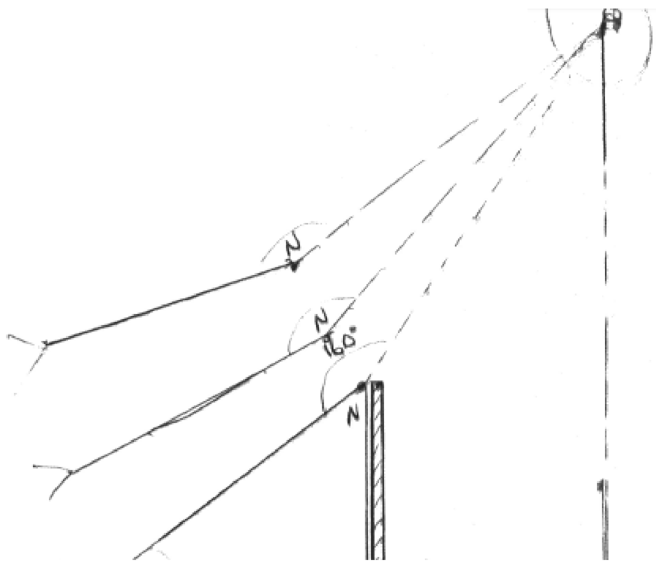

Figure 40: Φ<160

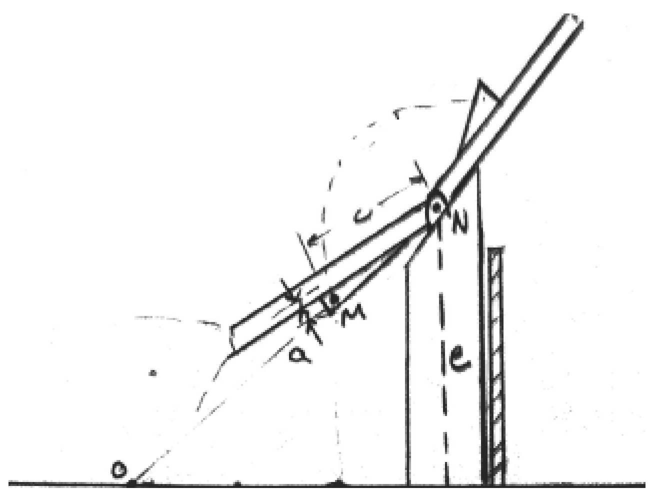

Figure 41: Stub on Lever Arm

With these modifications we set out to determine the final geometry of the design. We created an excel spreadsheet that correlated the distance the extended and contracted lengths of the piston to find out what would be feasible and what was not. Refer to Appendix E to see this relationship. A piston cannot extend passed twice its original length, and comfortably it should stay within 1.75 times its compressed length.

Feed System

Basic stress analysis was done along with the CAD model. The stress analysis focused upon immediate failure and ignored fatigue life failure. This is because fatigue life is easier to lengthen via designing for fatigue. The failure analysis was conducted upon the cables to ensure they would not break while pulling the feed box. It was also done upon the support structure that the pipe will get placed upon. The stress analysis was done using Excel. The sample calculation for this spreadsheet is visible in Appendix E.

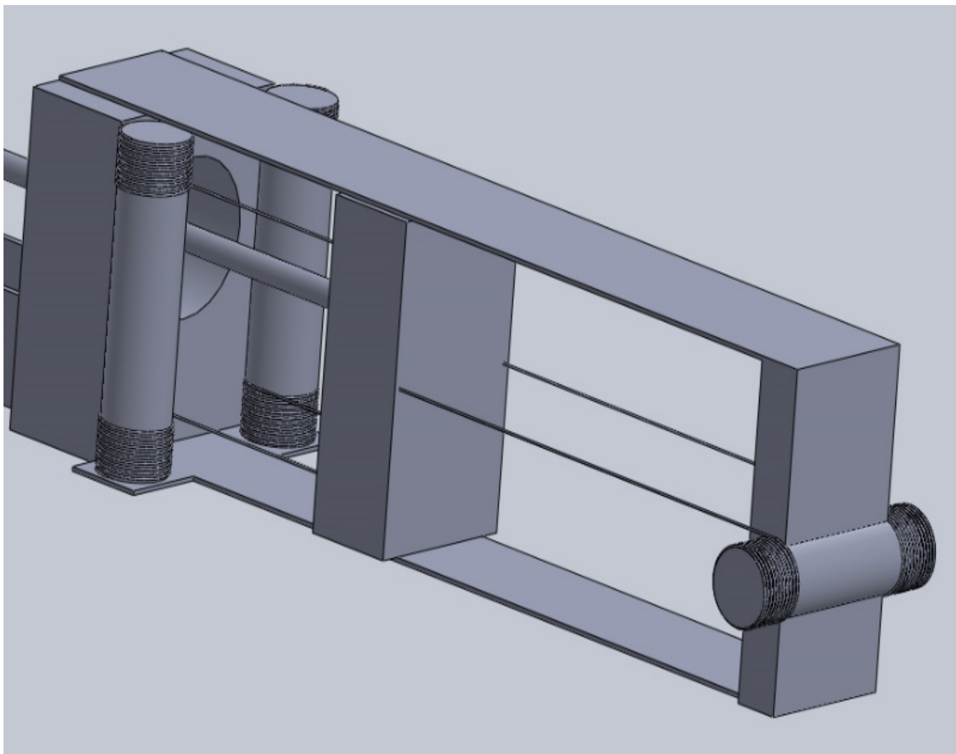

Figure 42: Cable Feed System CAD Model

We presented our top design of the cable system in a Preliminary Design Report and Dr. Roberts was at first skeptical of it being feasible and he suggested a rack and pinion system. We then proceeded to discuss the cable system and mentioned that it would in essence be a horizontal elevator and thus would be able to take the high loads caused by the extrusion process. After this discussion Dr. Roberts gave the okay to continue with the design of the cable system. We first discussed the lubrication system.

Support System

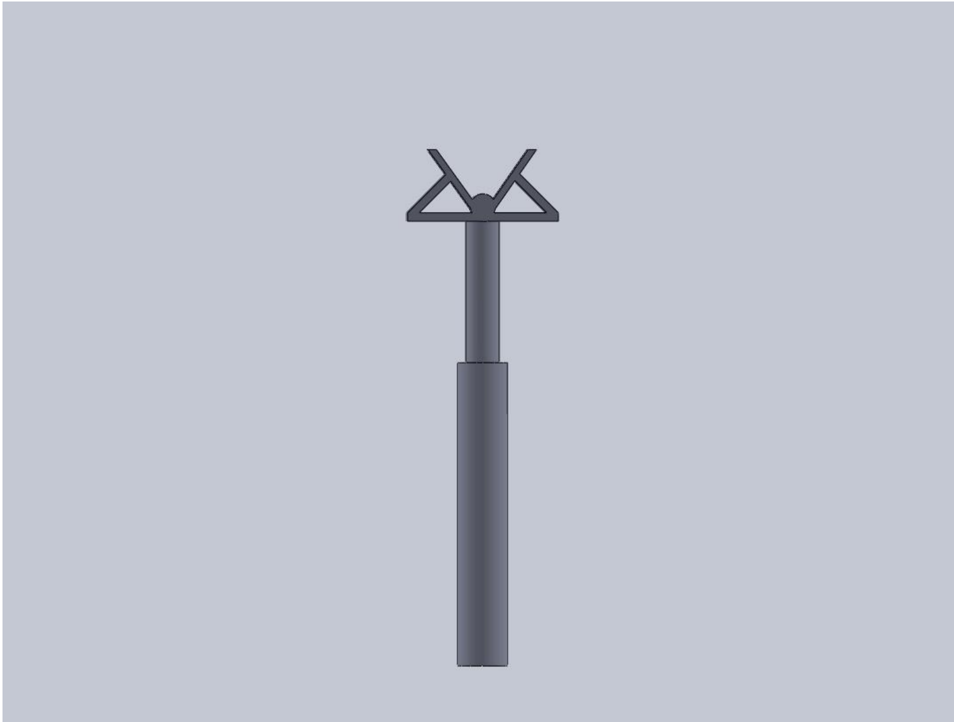

Figure 43: Support CAD Model

At first the support structure (Figure 43) started off with a simple holder in the shape of a “V”. It was then realized that this would have a bending moment and higher stresses than necessary, so another two linkages were added on to form a sort of truss. The weight of the pipe was determined by taking the density of steel and multiplying by the length and thickness of the pipe. We can use symmetry to do the force analysis on the support. The method of joints (Figure 44) was then used (assuming a 45 degree angle) for the load.

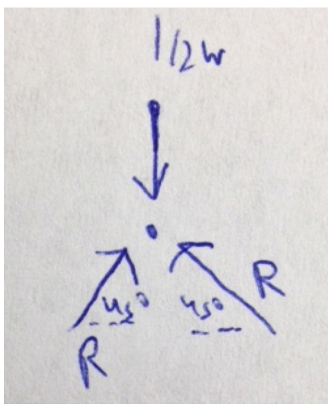

Figure 44: Method of Joints at Pipe Contact Point

$$R = \frac{w}{4} = 946\ lb \quad \text{(Eqn. 5)}$$

Then the load passed onto the major support which is shown in Figure 43.

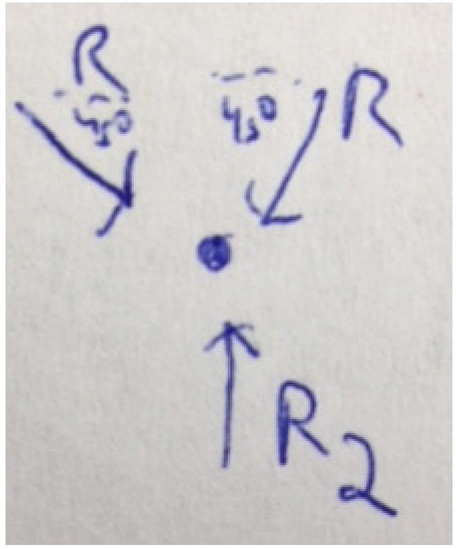

Figure 45: Method of Joints at Main Strut

$$R_2 = 2 * R = \frac{w}{2} = 1893\ lb \quad \text{(Eqn. 6)}$$

We have only normal stress in this system so we used the following equation to determine the stresses from the weight of the pipe.

$$\sigma = \frac{R_2}{A} = 37\ Psi \quad \text{(Eqn. 7)}$$

$$\sigma = \frac{R}{A} = 75\ Psi \quad \text{(Eqn. 8)}$$

From here we compare to the allowable limit and we can see that it is within the allowable load.

We also plan to add padding of some sort to decrease the odds of the pipe scraping against the support. The supports will be able to actuate up and down to allow the mandrel to fit perfectly into the pipe. They will also be placed on a track so they can move forwards and backwards to push the pipe along the mandrel.

Lubrication System

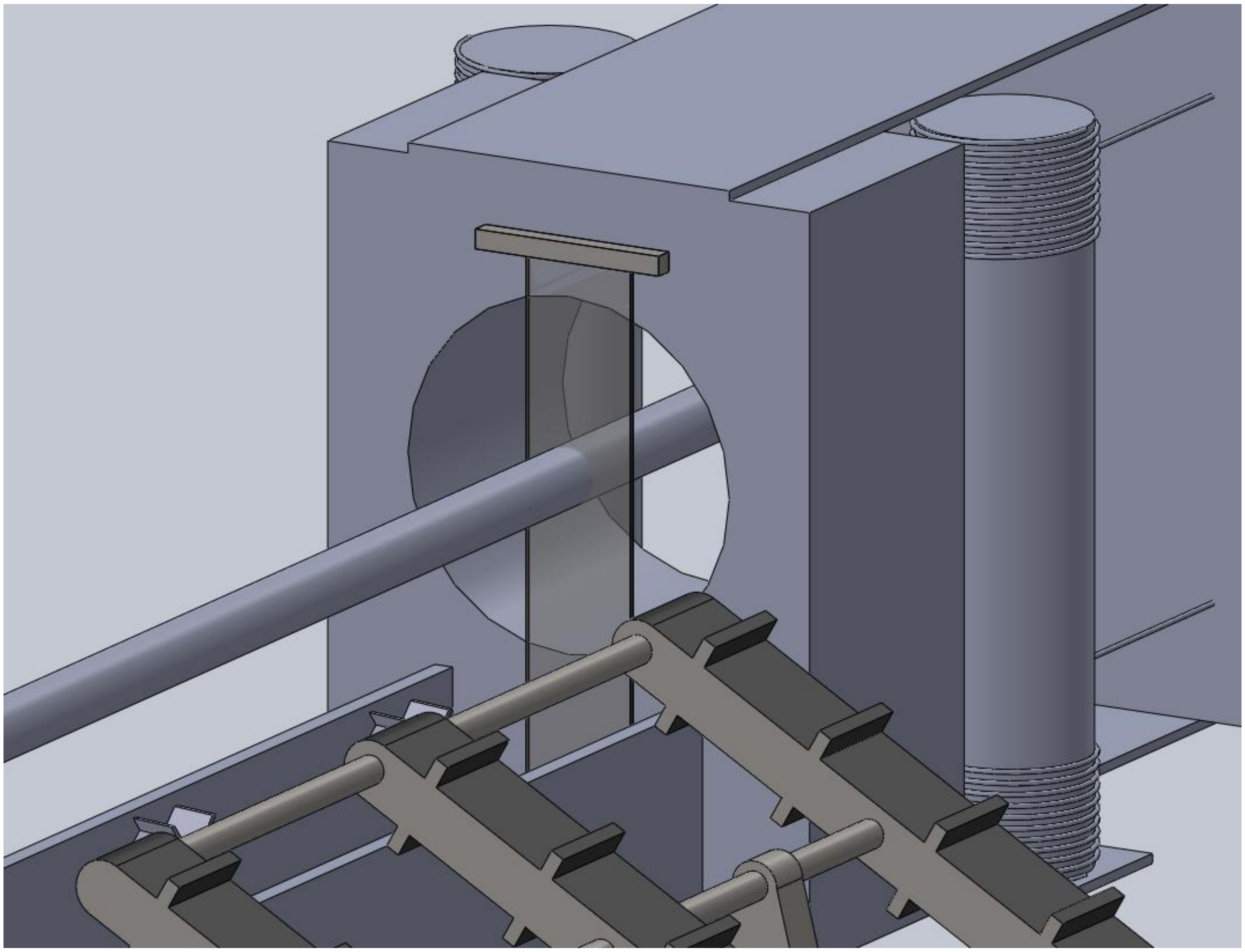

Figure 46: Waterfall System over the Mandrel

Part of the lubrication is done by the loading system. However, it is still vital to lubricant both the outside of the mandrel and the outside of the pipe. To deal with this we have decided to add a waterfall that drips lubricant down onto the mandrel and the pipe. The waterfall is located just before the die box as seen in Figure 46.

The full system is shown below in Figure 47. The Conveyor belt will have up to three pipes sitting on it at one time. During this waiting time the inside of the pipe will be lubricated. The mandrel will then be moved back by the feed system so the conveyor belt can load the pipe onto the supports that are not shown in the below image. The supports will raise the pipe up and position it so that it is concentric with the mandrel. The cable system will then push the mandrel into the pipe. While the mandrel is going into the pipe, the waterfall system will turn on and lubricate the outside of the mandrel. When the pipe is fully over the mandrel, the cable system will change direction and bring the pipe through the die box whilst the waterfall lubricates the outside of the pipe.

When the pipe and mandrel are pulled all the way through the die box will pinch the pipe and the feed system will reverse direction again and push the pipe through the die box and the extrusion process will commence. This will be repeated for as many passes that are required. The pipe will then be unloaded from the supports onto another conveyor belt that will take it to a storage area.

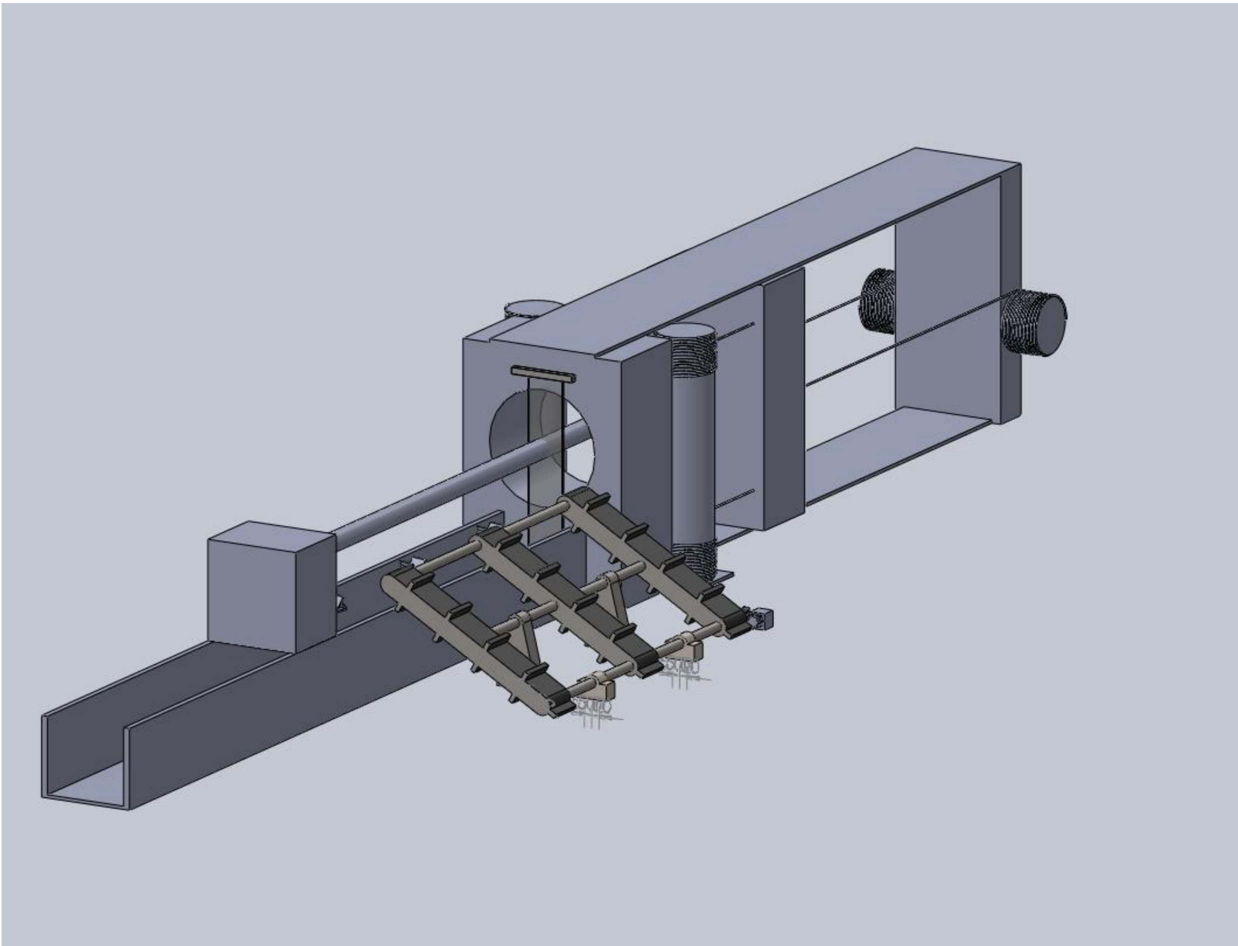

Figure 47: Cold Roll Extrusion Machine

Dr. Roberts agreed with our top design of the Waterfall and the Side Sprayers during our Preliminary Design Report presentation. He then gave us the okay to design these components for the lubrication system.

Final Design

After the PDR with our sponsor and more redesign, we came up with our final designs for each subsystem. We will be using the cable concept for the feed system, lever arm for the loading system and a waterfall with Folgers cup combination for lubrication system. The overall system model is visible below in Figure 48.

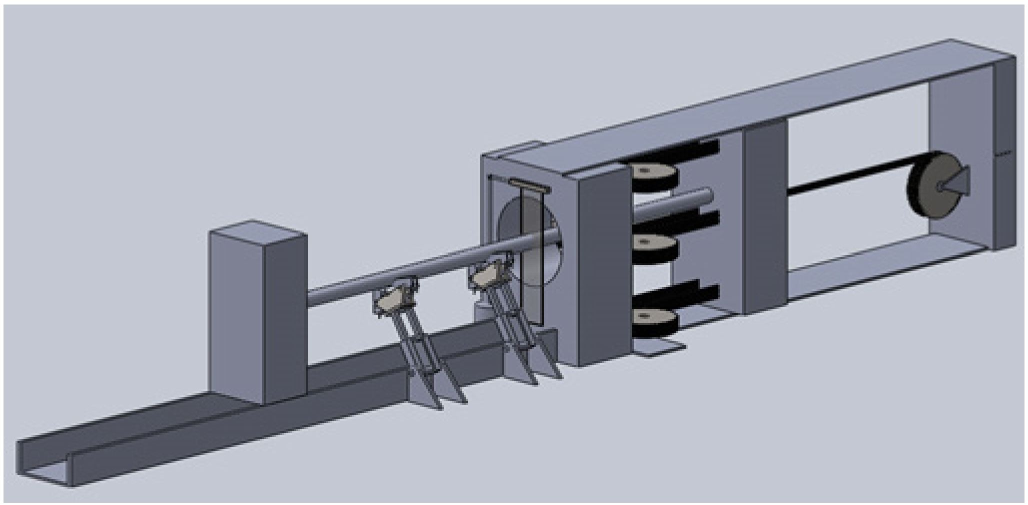

Figure 48. Final Design

Loading System

The final design of the Loading system is a combination of two lever arms working in parallel to lift a pipe into concentricity with the mandrel. We decided to use the dump truck inspired concept that rotates into position and then pushes the pipe along a centerline to concentricity. The Hydraulic Piston is sized at 4 inch diameter housing and 4 feet in length. We contacted Parker Hannifin and verified that we could purchase a piston of these dimensions that operates on a 3,000 psi and push at a force of 37698 lb. The piston will be mounted at pin joints on both ends, making it a two force member that does not experience bending moments. At one end it mounts to the protruding stub joint which will be welded onto the telescoping portion of the lever arm. At the other end, the piston will be fastened to a swivel joint that will be mounted 1.5 feet below the shop floor. This may or may not be an issue at the location in Irvine but is necessary for the geometry of a single piston design. If it is absolutely necessary not to cut into the shop floor, we could change the lever arm design to operate with two pistons. The joint below the shop floor will also be welded to a steel plate and anchored to the ground. In our calculations of bolt shearing stress, we found that we would only need two 3/4 A490, type 1 bolts in order to safely fix the loading system to the ground. A final concept sketch with loading positions is shown below in Figure 49. Most of our components in the loading system will be manufactured with 1030 Steel Quenched and Treated at 400 °F.

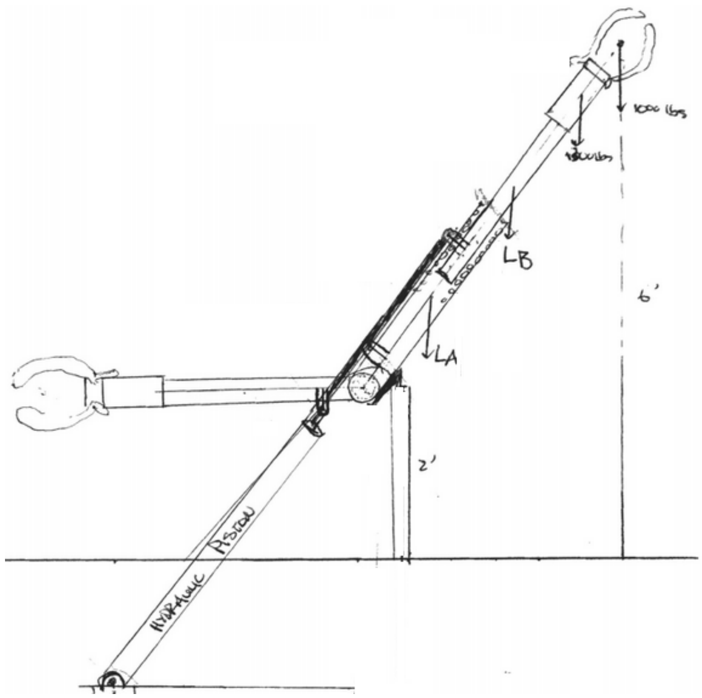

Figure 49: Final Concept Sketch

The final design operates in multiple stages. The first stage is when the hydraulic piston is fully retracted and the lever arm faces the shop. In this stage the pipe is loaded into the gripper using a forklift. The pipe gripper clamps are engaged to hold the pipe in place. The main hydraulic pump is then engaged and rotates the lever arm around a shaft. This shaft is mounted to the main supports which are 3 inches thick and bolted to the concrete shop floor.

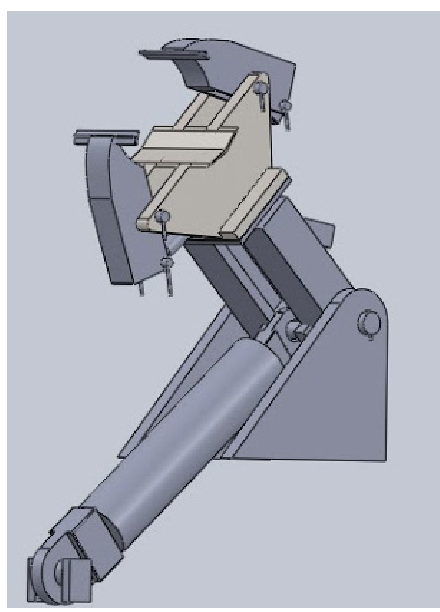

Figure 50: Loading Arm Stage 1

This rotation brings the lever arm to stage 2 where the rotational motion has finished. In the second stage the lever arm has an angled protrusion that rests on a permanent mount between the rails. We determined that the mount should be at least 1.79 inches square to hold the moment from the arm, but decided to make it 4 inches square to have a safety factor greater than two.

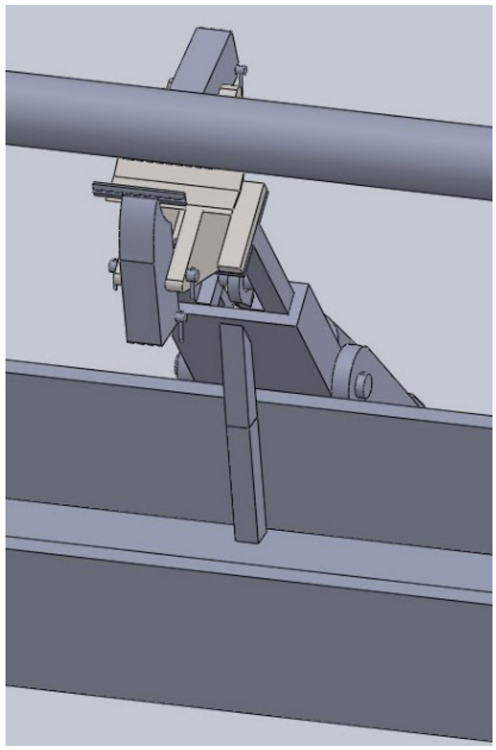

Figure 51: Moment Support Arm Engaged at Stage 2

Once the arm is in position in stage two, the motion changes from rotational to linear and the piston pushes out the telescoping arm with the gripper. The stationary portion of the arm is 30 inches long while the telescoping segment is 28 inches long. The amount of linear motion is dependent on the diameter of the pipe. The larger the pipe the farther the arm has to extend. This extension must be done very accurately as there is a .125 inch clearance between the pipe and mandrel.

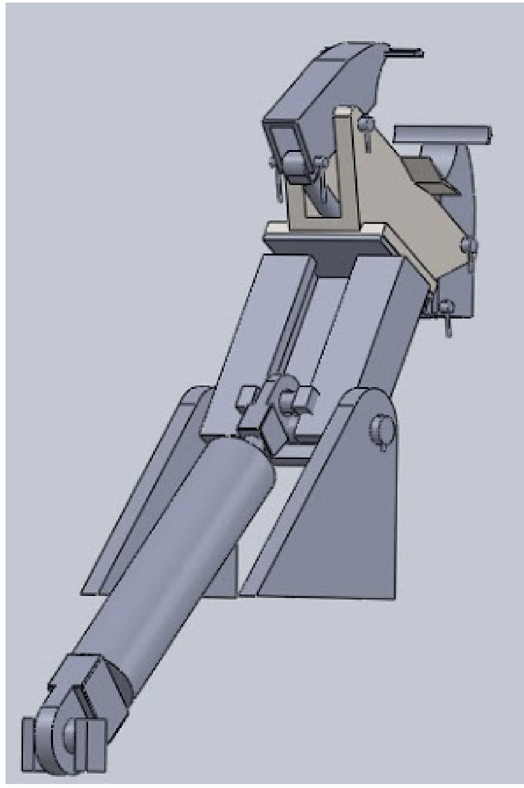

Figure 52: Lever Arm Stage 2

The lever arm has the possibility of failing due to fatigue. This would cause it to fall and damage the rails as well as possibly dropping the pipe and injuring someone. To mitigate the odds of this happening, maintenance will be carried out once a month to ensure all the components of the lever arm are in working order.

A detailed drawing and an exploded view of the lever arm are visible in Appendix B. The majority of the components in the lever arm will be manufactured at PCC. The only component that would be purchased, as opposed to manufacture in house, is the main hydraulic piston and the two hydraulic pistons in the gripper. This would be purchased from Parker and the specifications for the piston are available in Appendix D.

Gripping Hand

Our chosen loading system requires that the pipe is pushed along a centerline until the pipe becomes concentric with the mandrel. In order for this to work, we must make sure that the center of the pipe stays on this centerline as shown in Figure 53 below. In order to maintain that centerline, we designed a self-centering gripping system that leaves the only concern of the operator to push the pipe far enough along the centerline so that it is concentric with the mandrel as shown by the red dots in Figure 53.

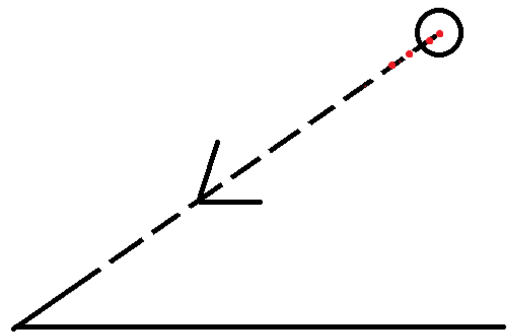

Figure 53: Centerline Theory

The gripper is designed to use two small hydraulic pistons that are connected to the same pump as the main hydraulic piston in the lever arm. These two pistons will be exposed to the same pressures and have symmetrical geometry so that they will push at the same rate and angle to center the pipe on the gripper, and consequently on the center line. In Figure 54 below, the gripper demonstrates that it can hold both the minimum and maximum piping diameters (6-20 inches). The gripper maintains three points of contact on any sized pipes. Two points of contact are provided by the rotating clamp arms. The third contact of support is a 1 inch V notch that helps to center any sized pipe as well as provide a consistent resting place.

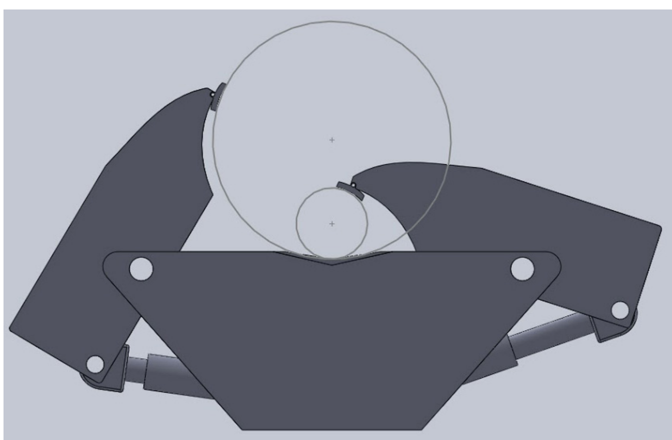

Figure 54: Gripper at Larges and Smallest Pipe Diameter

Extending Arm

The extending arm is connected to the piston as shown below in Figure 55.

It does not have a locking mechanism as the weight of the gripper and pipe will keep it unextended as the lever arm rotates. When the lever arm aligns with the center axis the extender will activate and the arm will begin to telescope out. The telescoping system is operated with a brass bushing liner in the housing component of the lever arm. The brass bushing allows steel to slide, even under extreme pressure with small amount of wear on the arm. The brass is used because it has a different crystalline structure than steel and so the materials don't form a bond in compression. If we left the inside liner as a steel and so have steel on steel contact, the material would begin to bond to itself under the high loads and experience large amounts of wearing. We also considered using roller bearings in the telescoping arm, but the bearings would be much too large and bulky.

Figure 55. Extending Arm

Feed System

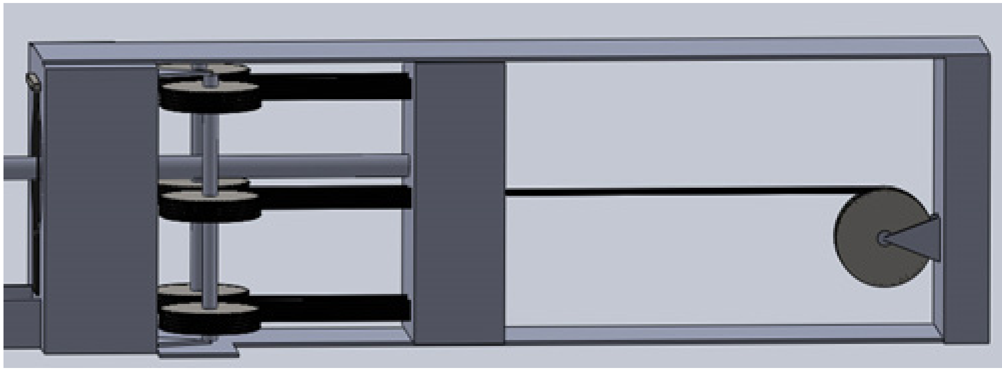

Figure 56. Feed System

The feed system utilizes 30 cables to pull the feed box towards the die box and commence the extrusion process. There are 6 sheaves that hold these 30 cables at 5 cables per sheave. Each system of 3 sheaves has a shaft running through it which is hooked up to an individual motor. The motor turns the shaft which rotates the sheaves and pulls the pipe. The rear of the feed box has 5 cables attached to it as well. These cables are used solely for returning the feed box to its pre-extrusion position. We analyzed the loads experienced during the extrusion process and sized the cable system to be able to withstand them. The analysis is shown in Appendix E. The force analysis (Figure 57) was done at the instant in time prior to the pipe beginning to rotate and extrude.

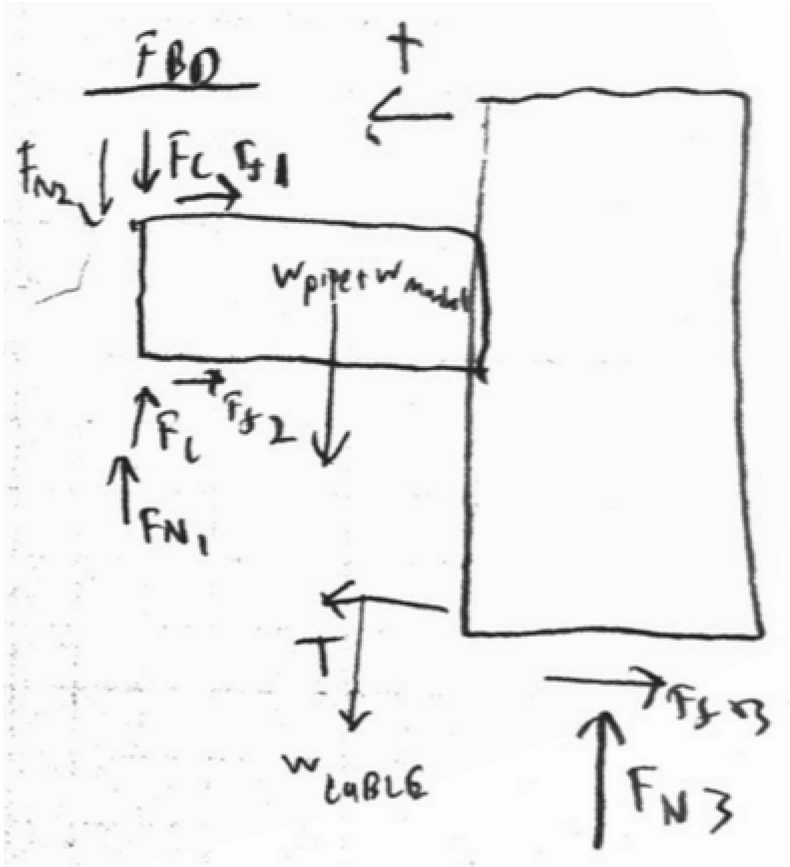

Figure 57. Free Body Diagram Feed System

At this moment in time the cables will have the highest loads so fatigue analysis is done with this in mind. This also leads to having a small built in safety factor in the analysis. After determining the loads in the cables we had to determine the fatigue life of the cable system. To do this we utilized Shigley's Mechanical Engineering Design Book [10]. Chapter 17-6 is a section on design with wire ropes. The rope must be wrapped around a sheave and the diameter of the sheave relative to the diameter of the rope has a huge effect on fatigue life. We set up the required equations in Microsoft Excel and iterated through with different sized ropes and different sized sheaves. In the end we opted to purchase wire rope from Woods Logging and Industrial Supply. The wire rope is 2 inches thick. We will use 30 of these ropes with seven separate 52 inch diameter sheaves where each sheave holds 5 ropes. These sheaves could be custom-made by eSheaves due to their size requirements. This ends up giving us a safety factor of 6.61 for a minimum fatigue life for 547,500 runs of the system which correlates to a 30 year life. This safety factor also exceeds OSHA's standards of a safety factor of 5.

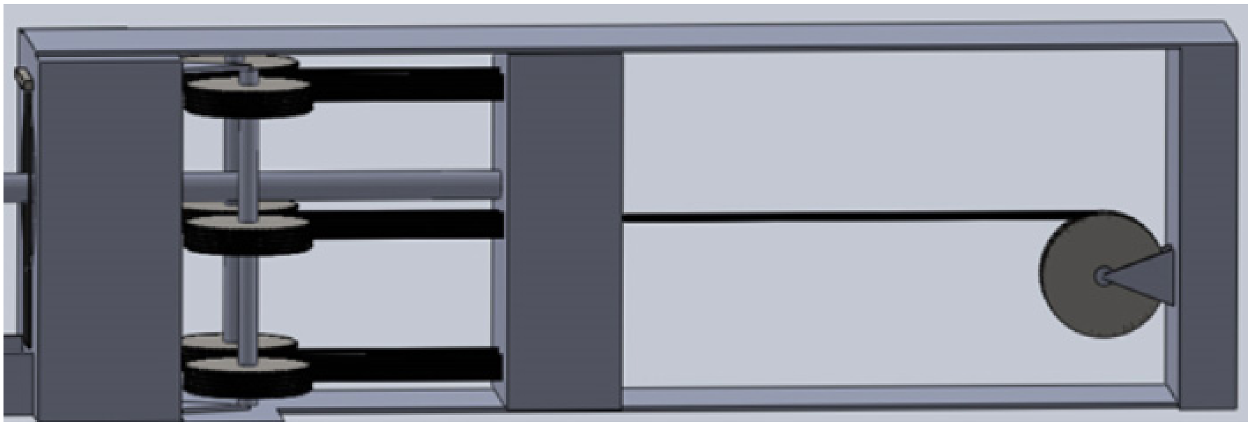

Figure 58. Feed System

We will also be using cables to pull the pipe and mandrel back before the machine can start another pass. The forces on the cables that must take this load are much lower as they do not experience the forces due to extrusion. To return the feed box to its pre-extrusion location we will be using a single sheave with 5 ropes. This will have the same properties as the sheave used during the extrusion process. However, due to much lower loads the factor of safety is 67.5 for 30 years (547,500 runs). To alleviate safety concerns about the cables snapping we will add a barrier between the cable system and the operator. This analysis shows that the cable feed system is a viable option, but further analysis needs to be done to size the brackets connecting the sheaves to the die box, the cables to the feed box and a load analysis on the feed box must be done.

Our goal here was to prove that it is feasible to use cables instead a worm screw to provide the necessary force for the extrusion process. We have shown that it is possible. The feasibility calculations are far enough along that the rest of the analysis consists of sizing a couple components, which we know will work. The biggest question with this design was whether the cables would even be able to withstand the loads produced by the extrusion process. Our calculations show that the cables would have a minimum life of thirty years, which means that this system is completely feasible.

The remaining analysis of this system is outside of the scope of this project; however it is listed below. A motor must be sized for the feed system, the critical speed and shaft size must be determined for the shaft passing through the sheaves, a more definite rail system must be created for the die box to move on and stress analysis should be done on the feed box.

Below in Table 11 we have the cost estimate for the feed system. Originally we had planned on purchasing 1.25 inch thick Python Ultra Wire Rope. However, during a call with the sales engineer working at the retailer for wire, the engineer suggested purchasing a Korean import rope that was half the cost. We ran the numbers on this new rope and discovered it was viable, so we decided to purchase it instead. This decision saved around $22,000 in our cost estimate. The rope is purchased from Wood's Logging and Industrial Supply. Meanwhile, the sheave to hold the rope could be custom ordered from Sheaves. The cost of the rest of the

system that was not analyzed will cause this to increase but these two pieces of equipment are the main expenditures for the feed system.

Table 12: Cost Estimate Feed System

Cost Estimate	
Rope	$21,542.50
Sheave	$70,000
Total	$91,542.50

The feed system must be maintained to ensure that it functions properly and does not break. If the cables in the feed system broke they could cause a large amount of damage. The cables also have the potential to injure or even kill the operator. To mitigate the chances of serious injury, we will be installing a barrier between the feed system and the operator. This will ensure that any failures of the feed system will stay contained within the barrier and not injure anyone in the vicinity of the machine. Maintenance will be carried out by the head of maintenance and safety checks of the system will be done once a month.

Lubrication System

The lubrication system has been changed to use the Folgers cup instead of side sprayers. The waterfall will remain to lubricate the outside of the mandrel and pipe. We have opted to switch for the Folgers cup because it is a lot cheaper than the side sprayers and makes more sense with our new loading system. Our lubricant is extremely viscous and only works in specialty pumps. The Bayer drum pump is made specifically for high viscosity fluids and it can deliver up to 11 gpm of lubricant and utilizes a 4hp air motor which will give us 98 feet of head. This is more than enough to move the lubricant from the drum to the waterfall.

The piping system will be manufactured by PCC and will be placed relatively close to the machine. The lubricant drum will be moved next to the machine and hooked up to the pump. From here the pipe will run up over the top of the mandrel. When it reaches the top of the mandrel the pipe will have holes drilled in it. These holes will be equally spaced and a check valve will be placed in each hole. These valves will be opened and closed electronically. The valves will open as the mandrel passes under it and it will then close. They will open again as the pipe passes underneath it and will do

these processes as necessary. A lubricant catch will be placed underneath the waterfall and will collect excess flow.

The head of maintenance will check the pipe flow every morning prior to the start of the work day and will perform maintenance as necessary.

Management Plan

We have created a Gantt chart (Appendix A) to keep track of our progress throughout senior project. The Gantt chart is a step by step timeline of what we will be working on every week until the project is completed. Foja will be in charge of maintaining the chart and ensuring the completion dates of various parts of the project are met. The chart will be updated as needed to account for any major changes.

One major complication of the chart is that we have yet to decide what will be happening during the final quarter of our senior project. There are currently two routes that can be taken during that time. We can either continue onto building a small scale working model of the loading system we've decided is the best. The other option is to reiterate the design process and improve our system and implement new ideas that we have come up with after the initial design. These two routes were determined after discussion with Dr. Rossman and Dr. Roberts of PCC Rollmet. We will wait until the end of the second design review to determine which route to choose.

Our group meets every Monday, Tuesday and Thursday with additional meetings as necessary. This time is used exclusively to work on our project. Weekly status reports are written every Thursday to ensure we have done enough work during the week. If we are behind then we use the weekends to catch up. When a disagreement comes up during project work; it is discussed and then voted upon. This method allows for in-depth discussions of the different ideas and gives a chance for everyone to have input and build off of each other's statements. Tasks are divided via our skill sets. Bryan and David are in charge of creating the SolidWorks models, whereas Foja is in charge of the analysis. Brainstorming is done in group sessions and design decisions are made as a group. This allows us to work on what our skills are suited for and still take advantage of being a team by working together on important decisions.

Product Realization (Manufacturing)

Streamlining the manufacturing processes at PCC Rollment was decided to be most effectively achieved in the design of a Loading Device. As discussed above, the loading process takes up to 45 minutes so there is a lot of room for improvement. For this reason, and because the loading mechanism that was designed is nontraditional, a functional prototype of the loading arm was constructed. This functional prototype was built to demonstrate that the motion and mechanics of the model. In order to do this, it was agreed that a ¼ scale model would an appropriate size. In order to manufacture a quarter scale model the necessary materials were compiled in a list show below in

Table 13. To secure the model and demonstrate the different levels of shop floor, the model was constructed on a wooden base. The base was constructed of plywood and was mounted on a frame constructed with 2x4 wooden beams. The hydraulic piston that was ordered was double acting, and was ordered from McMaster Carr.

Table 13: Materials list for Quarter Scale Model

Item	*Vendor*	*Part #*	*Quantity*	*Cost/per part*
1/4" 1018 Steel Sheet 12"x24"	*McMaster-Carr*	*6544K76*	*2*	*$90.61*
1" D 1556 Steel Shaft 2' Long	*McMaster-Carr*	*6061K55*	*1*	*$31.05*
1018 Steel Bar 1"x1" 24" Long	*McMaster-Carr*	*9143K21*	*1*	*$26.01*
1/4" 1018 Steel Sheet 6"x6"	*McMaster-Carr*	*6544K22*	*1*	*$19.81*
1/16" 360 Brass Sheet 4"x24"	*McMaster-Carr*	*8951K07*	*1*	*$42.27*
1/4 - 20 x 3 Steel Hex Cap Bolt (25pc)	*McMaster-Carr*	*91257A554*	*1*	*$9.79*
2" Bore with 12" Stroke Length Large Footprint Cylinders Hydraulic Piston	*McMaster-Carr*	*62205K743*	*1*	*$474.02*
J-B Weld Epoxy	*McMaster-Carr*	*7605A12*	*1*	*$17.50*
Total Cost				*$801.67*

The entire frame of the lever arm, including the supports, Lower and Upper Arms, and Gripper, were constructed out of quarter inch 1018 steel plating. From the plating, the appropriate pieces were cut out and then welded together to form the correct structures. Figure 59 below displays the plan we developed to cut out all of our pieces on the sheet metal that we ordered.

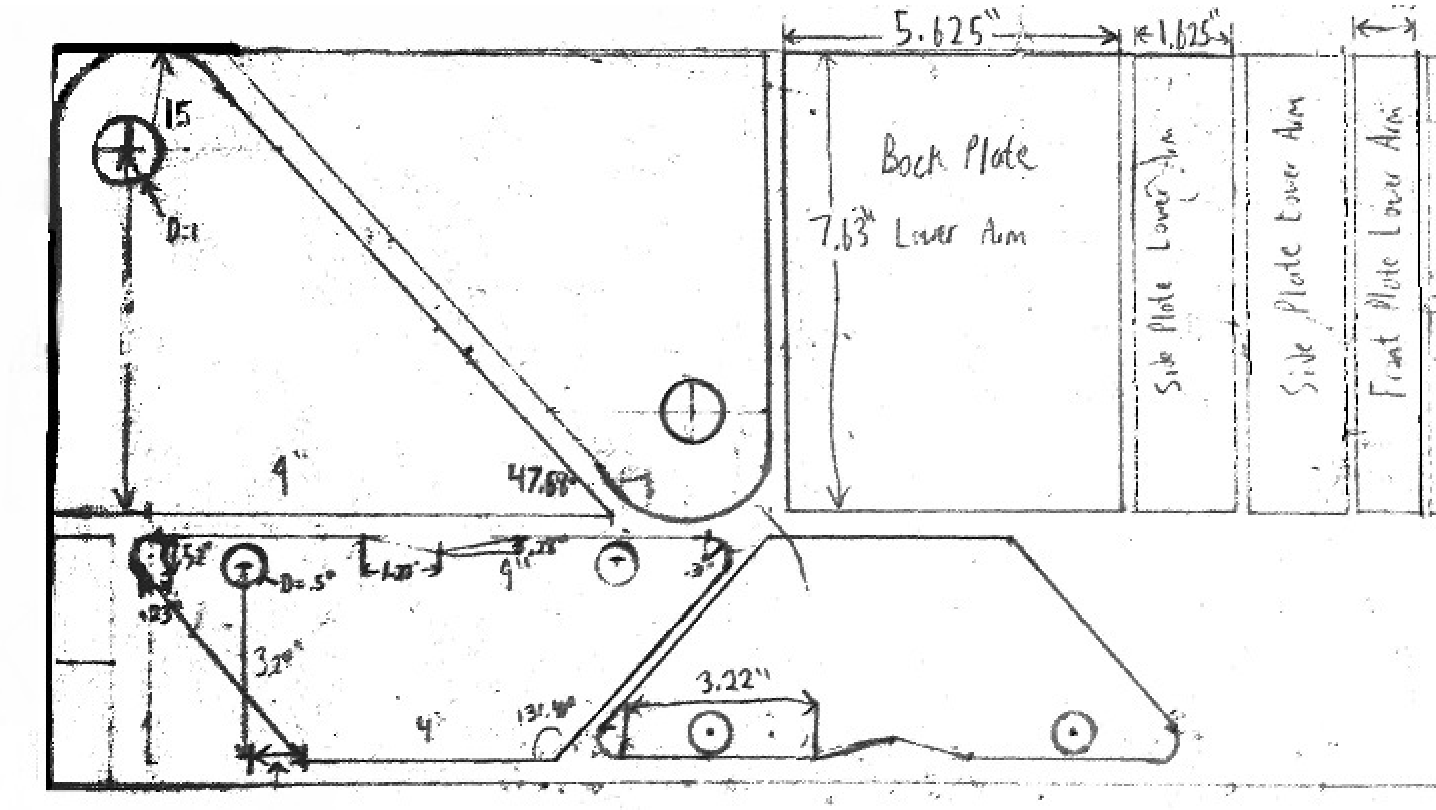

Figure 59: Cut-up plan for quarter inch 1018 steel

The beginning of the manufacturing process was to cut out these pieces of sheet metal. Different cutting processes were implemented in this process, and some were found to be more effective than others. The first process used was a plasma cutter which is shown below in Figure 60. The plasma cutter was rigidly attached to a vision system which traced a line drawn on a sheet of paper. Unfortunately, the vision system was not operational this quarter so it could not trace the curved lines that some of our designs needed, but it could still operate in a linear motion. The plasma cutter was used to cut out the large triangular support pieces for the upper, main base of the lever arm, as well as the corresponding bolt flanges. The plasma cutter needed two passes to fully partition the sheet metal. After the cut, the edges were rough, divoted and covered with slag in some areas. A steel bench grinder, shown in Figure 61, was used to polish off these rough edges, and to achieve the curves shown in the design. Since the rough edges could be cleaned up, the pieces for the Lower Lever Arm were also cut out on the plasma cutter and polished with the bench grinder.

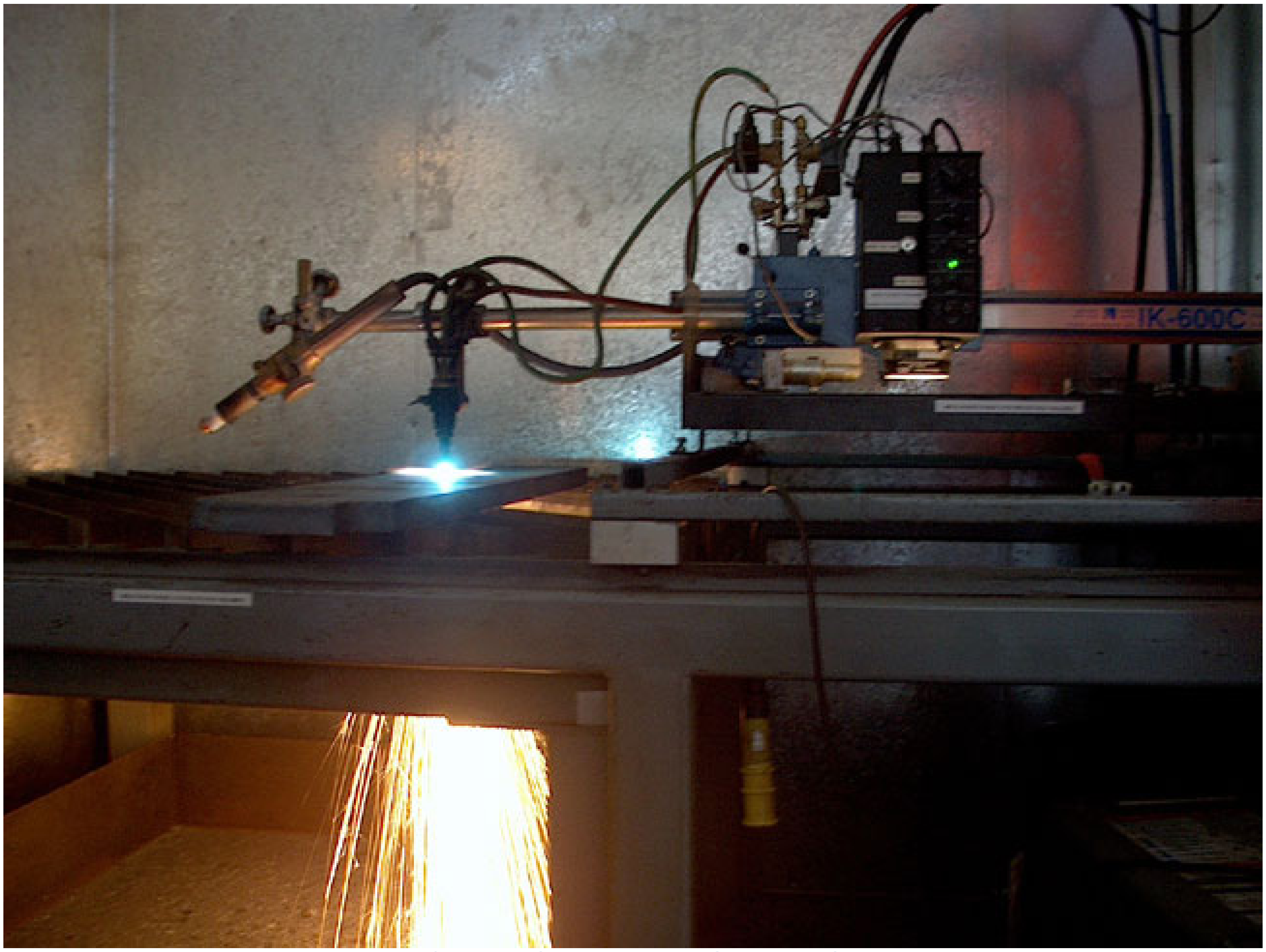

Figure 60. Example of Plasma Cutter Tracingy System in Cal Polys Hanger

Figure 61. Bench grinder similar to the model in the Cal Poly Hanger

Although the bench grinder allowed the edges to be smoothed out, it was very difficult to produce straight edges and perpendicular angles, not to mention meet the correct dimensions. Because the edges were not completely straight, the cut outs did not sit flush upon each other and therefore the welds could not be properly performed. In order to create straight and perpendicular edges, the Cal Poly Shop Technicians (Shop Techs) recommended using a mill. Using the mills in the Cal Poly Hanger required the use of the machining speed equation $N = 4V/D$. V is the cutting speed for different materials (steel $V = 50$), and D is the diameter of the cutting tool. This equation is used a guide, but in practice we often cut the steel a little slower to avoid overheating and dulling the endmill. Coolant and cutting fluid was also used to aid the cutting process and maintain adequate temperature levels. The mill allowed us to create parallel and perpendicular edges when using a reference edge, as shown below in Figure 62. Typically one of the edges of a piece of cut sheet metal was “as received” from the manufacturer. These edges were straight and could clamped on an edge in the vice, allowing everything machined to be parallel or perpendicular to it. Milling was a slow process, especially when machining material as tough as steel, but it produced the straight edges that were required. In order to reduce milling time we also switched our remaining cutting operations to use the bandsaw. The bandsaw made cleaner cuts than the plasma cutter, and if done carefully, there was less clean up work required by the mill. An example of the bandsaw cutting operation is shown below in Figure 63.

Figure 62. Milling Piece for Lower Arm at Cal Poly Hanger


Figure 63. Bryan using the Bandsaw to cut sheet metal

The mills were also used to accurately tap the holes in the Triangle Base supports and the Lower Lever Arm, all of which was to hold the 1” shaft. Mills are equipped with measuring tools that can be calibrated to measure the precise location of holes on a part. This was more accurate than tracing the location of the hole by hand and using a drill press. An edge finder tool was used to calibrate the mill and find the same location for each drilling operation. The Cal Poly Hanger had nominal drill sets with standard drill bits up to ½ inch in diameter. Using a Mill chuck to hold the drill bits, we started small and worked up to the ½ inch diameter. It was more difficult from there,

but the Hanger did have some larger drill bits that worked up to the 1 inch Inner diameter. A few times, the drill bit bent off center, and didn't plunge straight through the metal as intended. In order to compensate for that, we used an endmill suitable for plunging to realign the hole.

After drilling these holes we realized that there should be some sort of bearing supporting the shaft through the Triangle Base supports. A little research uncovered that there are a couple different options for mounting a bearing in ¼ inch thick metal. A roller bearing did not seem appropriate, because it would require a precise press fit. Pillow block bearings and single piece sleeve bearings, however, fit the application well. Pillow block bearings, shown below in Figure 64, operate by being bolted on to the plate. Brass Sleeve bearings shown below in Figure 65 operate on a press fit, but the flanged end allows the bearing to be easily aligned. Since the brass sleeve bearings are cheap and simple, they were chosen to be implemented in our design, press fit into the holes cut for the Triangle Base supports.

Figure 64. Picture of Block Bearing from McMaster Carr Online Catalog

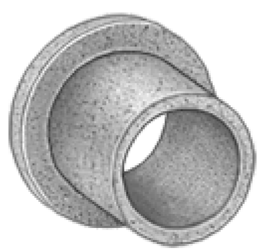

Figure 65. Picture of Brass Sleeve Bearing from McMaster Carr Online Catalog

In order for the bearings to fit, the 1 inch ID hole in the Triangle Base had to be expanded. To gradually step up the cut to the appropriate press fit, a fly cutter tool was used. The fly cutter, as shown in Figure 66 below, allowed us to shave off the inner diameter of the hole in the Triangle Base until it was large enough to have a .003 inch interference with the outer diameter of the brass sleeve bearings. The interference created a press fit when the brass sleeve bearing was pressed into the hole in the Triangle Base. After the brass sleeves were press fit into the Triangle Bases, it was discovered that the shaft did not fit as easily or rotate freely. The press fit had deformed the brass bearings enough to slightly choke the shaft. In order to combat this, the brass bearings were filed in order to increase the inner diameter. This allowed the shaft so again spin freely. To complete the shaft assembly with the Lower Lever Arm and

Triangle Supports, the shaft needed something to keep it from moving in the axial direction and potentially slipping out of the bearings. This was most easily solved by ordering shaft collars which tighten down on the OD of the shaft without requiring any extra machining.

Figure 66. Adjustable Fly Cutter Tools for mount in a mill

Cutting the square bar and circular shaft was done on the horizontal bandsaw and the chopsaw. The 1018 steel square bar cut easily on the horizontal bandsaw, but hardly made a notch on the stainless steel circular shaft. We learned that stainless steel is strain hardening and therefore is difficult to machine. We were able, however, to cut it on the abrasive chop saw. Cutting the square bar at an angle, so that it could attach to the Lower Lever Arm, turned out to be a challenge. The horizontal band saw did allow for angled cuts, but not as extreme as was needed.

Welding the pieces together was the next step. Many of the welds were to add flanges to the supports, which allowed the model to be bolted down to the wooden base. The most critical welds were those made in the construction of the Lower Lever Arm. These welds were done by one of the shop techs so that the because the tolerances are very fine and the angles of the welds are difficult to keep perpendicular. This weld was done with a TIG welder. Our group did some of the other welds, with MIG on the bolt flanges and TIG on the stud support and connections. The welds on the upper arm were also high tolerance welds as two bars needed to be welded in parallel a shaft needed to fit through tabs on the other end.

When the mounting brackets for the piston, their shaft diameters were for ¾ inch shafts, not 1 inch shafts. We attempted to compensate for this by lathing some of the 1 inch stainless steel shaft down to ¾ inch, but quickly found that machining stainless steel was extremely difficult. We decided, due to the time and cost of parts, to order ¾ inch shaft and remanufacture the supporting tabs to accommodate the piston mounts. See Figure 67 for an example.

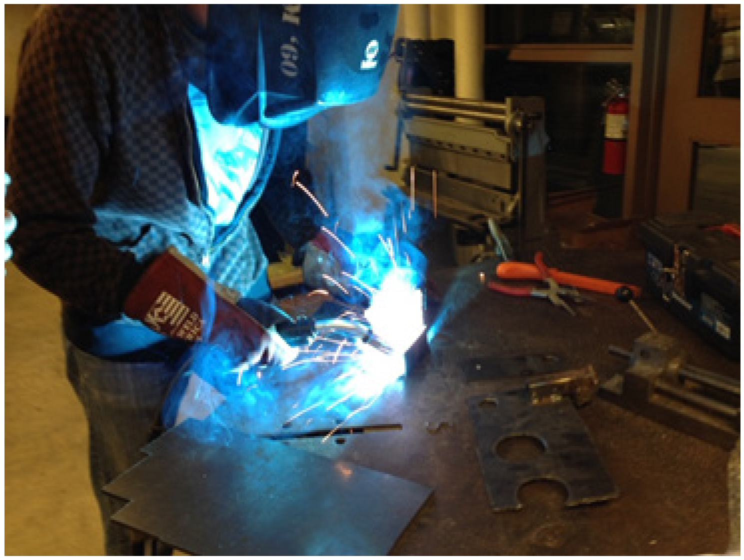

Figure 67. David MIG welding a mounted bracket flange

Figure 68. Example of MIG weld on lower piston support

Figure 69. Picture showing TIG welds on Lower Arm and Shaft Assembly

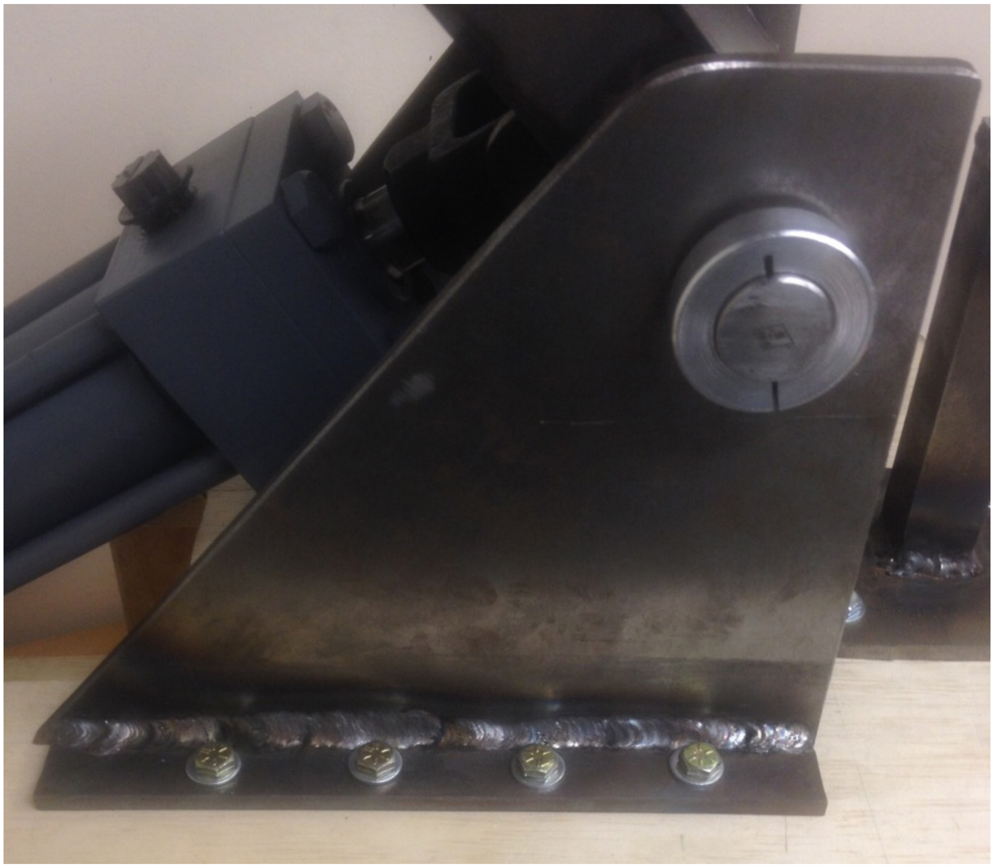
Figure 70. Triangle Base Support with Shaft Collar

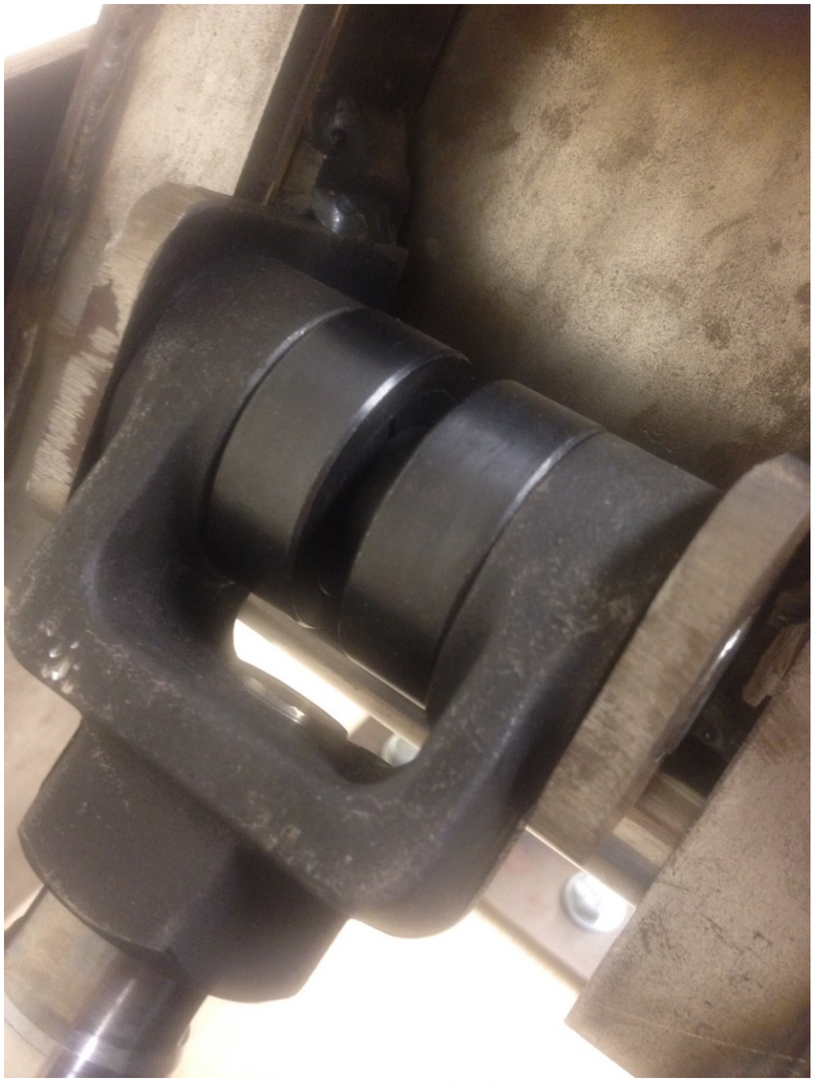

Figure 71. Upper Arm Shaft Assembly

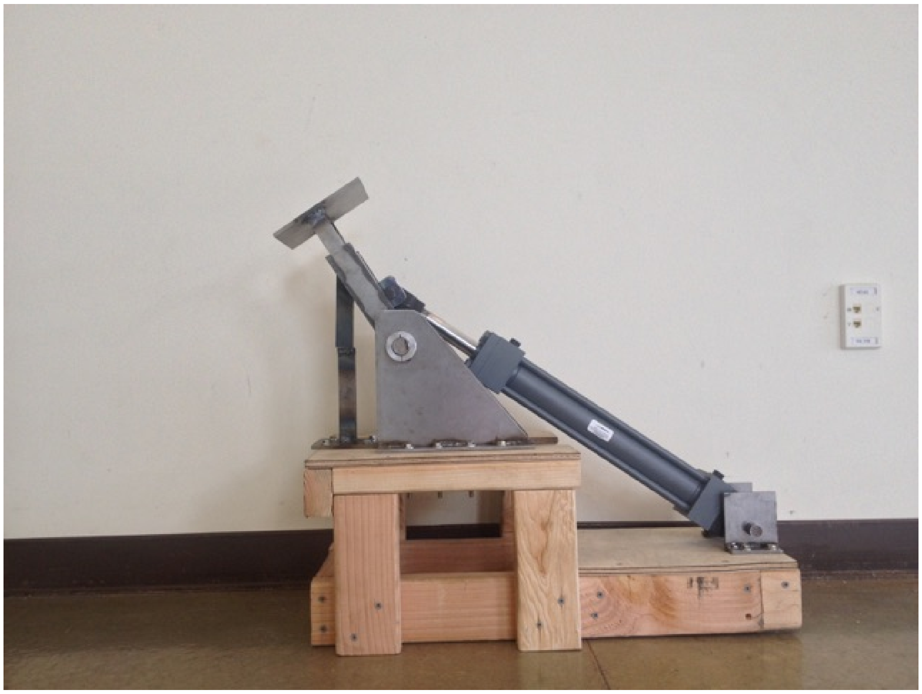

Figure 72. Final Model

Design Verification (Testing)

The testing for a proof-of-concept model is much different than that of a prototype. For our proof-of-concept tests want to show that the design idea is feasible. A prototype meanwhile is used to test stresses, deflection, controls, fatigue and anything else that comes to mind.

Proof-of-Concept Test

To test our model we had originally planned to use a pump that the Cal Poly Mechanical Engineering department had. However, the pump was unable to run at our design pressure. Testing above our design pressure was deemed unsafe and this test was canceled. To make up for this we conducted a manual test of the model.

Manual Test

1. Open the ports on each side of the hydraulic cylinder
2. Rotate the lever arm so that it comes to a rest on the dead stop, whilst observing the cylinder.
3. Actuate the upper arm by sliding it out of the lower arm, whilst observing the cylinder.
4. Slide the upper arm back into the lower arm
5. Rotate the lever arm back to its original position of off the dead stop

Our manual test proved that it was possible to move our model through the desired positions. This proved our model worked and would warrant moving on to building a Prototype in the future.

Future Tests for a Prototype

1. Running the lever arm until failure to determine its fatigue life.
2. Setting up a controls system and optimizing response time and error.
3. Positioning accuracy for the different sized pipe.
4. Ability to withstand the force applied the the mandrel as it is put into the pipe.

Concluding Remarks and Recommendations

Over this yearlong project we have learned quite a bit. We have delved into three major portions of mechanical engineering. Design, analysis and manufacturing. Multiple design iterations were done during the first phase of this project. We learned that the longer we spend designing, the better our project becomes, however this slows down the overall process of getting a product out into the market. The second phase consisted of thorough analysis to ensure nothing would fail. This analysis required us to ask for help from multiple professors to handle things we haven't seen before in our classes. Finally we manufactured a proof of concept model. The model was made of steel and this proved to be very time consuming to machine. Along with this we began to grasp a better idea of how to design with manufacturing in mind to ensure that the parts could actually be produced.

We redesigned three separate subsystems of the Cold Roll Extrusion Machine. The feed system was redesigned to be a cable pull system. After analysis was done, this was determined to be feasible, but unsafe and much too costly. This was a cool discovery, because we proved a design to be a bad option so more time wouldn't be spent looking at it. Meanwhile, the lever arm was proved to be feasible and cost effective. Thus we made a concept model of it to prove that it works.

After examining the FEA of the lever arm it was discovered that our loading arm was experiencing more deflection than initially expected. The largest area of deflection was underneath the two bars of the upper arm. To combat this the design could be improved by splitting the single support member into two support members, one under each of the two bars of the upper arm. After some buckling analysis it was discovered that our support members were far too thick yielding safety factors at an upwards of 100. By making the support members smaller we can decreased the total material used and therefore the overall cost of production.

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[6] http://www.smrw.de/files/steel_tube_and_pipe.pdf

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[10] Budynas, Richard G, J K. Nisbett, and Joseph E. Shigley. *Shigley's Mechanical Engineering Design*. New York: McGraw-Hill, 2011. Print.

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Cal Poly Shop Techs

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Loren – Assembly
Carlos - Milling

APPENDIX A

Task Name	Duration	Start	Finish
Write Introduction Letter to Sponsor	4 days	Thu 1/9/14	Tue 1/14/14
Write Team Contract	3 days	Tue 1/14/14	Thu 1/16/14
Sponsor Visit	2 days	Sun 1/19/14	Mon 1/20/14
Write Project Proposal	9 days	Tue 1/21/14	Fri 1/31/14
Complete QFD	4 days	Mon 1/27/14	Thu 1/30/14
Complete Team Evaluation	4 days	Mon 2/3/14	Thu 2/6/14
Brainstorm Concept Models	16 days	Mon 2/3/14	Mon 2/24/14
Build & Present Concept Models	6 days	Tue 2/11/14	Tue 2/18/14
Concept Design Review with Sponsor	1 day	Tue 2/25/14	Tue 2/25/14
Work on Decision Matrices	3 days	Tue 2/25/14	Thu 2/27/14
Work on Concept Design Report	6 days	Tue 2/25/14	Tue 3/4/14
Work on Preliminary Design presentation	1 day?	Tue 2/25/14	Tue 2/25/14
Present Prelminary Design Presentations to Class	3 days	Tue 3/4/14	Thu 3/6/14
Edit Presentation from in-class Critiqes	2 days	Thu 3/6/14	Fri 3/7/14
Schedule Preliminary Design Review	1 day	Fri 3/7/14	Fri 3/7/14
Present to Sponsor	1 day	Thu 3/13/14	Thu 3/13/14
Update Top Design To Reflect Sponsor Critiques	16 days	Fri 3/14/14	Fri 4/4/14

Figure 73: Winter Quarter Task List

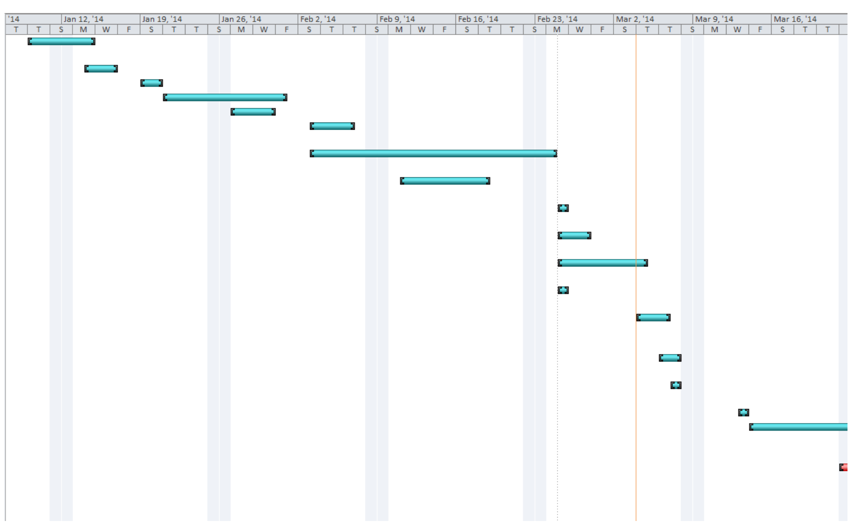


Figure 74: Winter Quarter Task List Schedule View

Scheduling Overview

Task Name	Duration	Start	Finish
Stress Analysis and CAD of Feed System	7 days	Tue 4/15/14	Wed 4/23/14
Schedule Concept Design Review	1 day	Thu 4/24/14	Thu 4/24/14
Final Design Report	7 days	Thu 4/24/14	Fri 5/2/14
Redesign	22 days	Mon 5/5/14	Tue 6/3/14
Complete Critical Design Review presentation with Sponsor	1 day	Fri 5/9/14	Fri 5/9/14
Order Parts	9 days	Tue 5/13/14	Fri 5/23/14
Manufacturing Plan	7 days	Mon 5/26/14	Tue 6/3/14
End of Quarter Report	2 days	Thu 6/5/14	Fri 6/6/14

Figure 75: Spring Quarter Task List

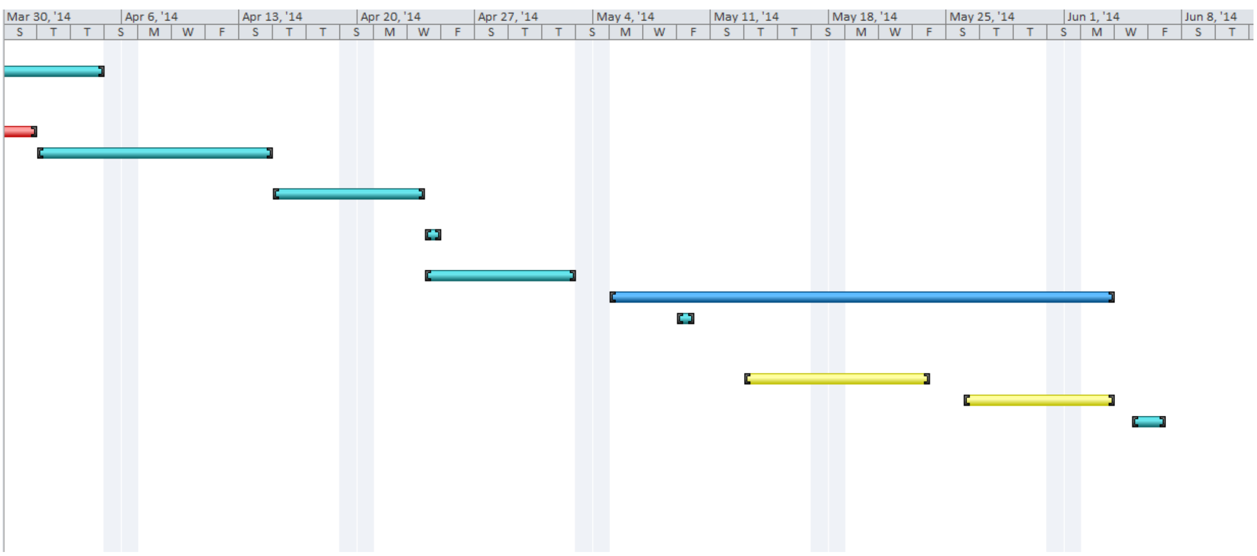


Figure 76: Spring Quarter Task List Schedule View

Due to the nature of our project we are unsure about what we will be doing during spring and fall quarter. Figure 73 and Figure 74 display the schedule for winter quarter; next Figure 75 , Figure 76, Figure 77 and Figure 78 show the two routes we have the option of taking. In Figure 77 and Figure 78 the first route we can take is doing an iteration of design and is depicted by the blue bar. Meanwhile the second route is building and testing the loading system to prove feasibility. We will determine which route to take after the critical design review in conjunction with our sponsor. This will then let us determine which route we will take during fall quarter. As we do not currently know what we are doing we made a schedule to account for both possibilities.

Task Name	Duration	Start	Finish
Build Prototype	16 days	Mon 9/22/14	Mon 10/13/14
Redesign	24 days	Mon 9/22/14	Thu 10/23/14
Test Prototype	20 days	Tue 10/14/14	Sun 11/9/14
CAD & Analysis of Redesign	12 days	Fri 10/24/14	Sun 11/9/14
Senior Design Expo	9 days	Mon 11/10/14	Thu 11/20/14
Final Project Report	10 days	Fri 11/21/14	Thu 12/4/14

Figure 77: Fall Quarter Task List

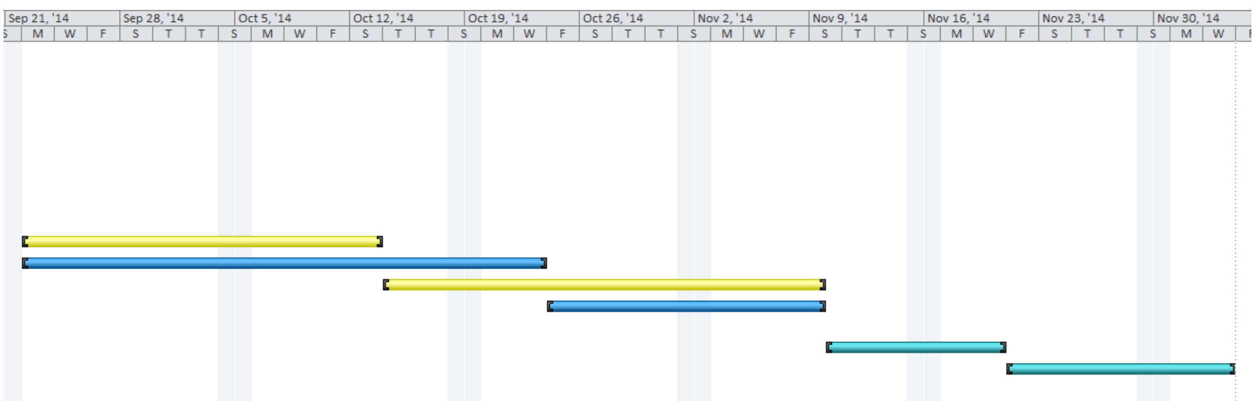


Figure 78: Fall Quarter

QFD

Proje: Cold Roll Extrusion Team
Revi: 1
Date: Tuesday, January 28, 201

Correlations	
Positive	+
Negative	−
No Correlation	

Relationships	
Strong	●
Moderate	○
Weak	▽

Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼

+ +
+
− − −
+ +

	Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
	Direction of Improvement	▲	▼	◇													
	HOW: Engineering Specifications	Working	Motor Fee	Die Box S	Motor for	Die Slide	Die Feed	Die Locke	Machine	Automati	Manual L	Loading M					

WHO: Customers — NOW: Current Product Assesment - Customer Requirements

Row	Weight Chart	Relative Weight[1]	PCC E	PCC S	Opera		Maxi	WHAT: Customer Requirements (explicit & implicit)	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	Curre	Curre	Row
1		7%	9	5	1		9	Machine Length	●	○	▽	▽	▽	▽	▽	▽	○	○	●								1
2		9%	9	9	1		9	Part Length up to 45 feet	●	○	▽	▽	○	○	○	▽	○	●	●								2
3		9%	9	9	1		9	Part Diameter: 6"-20" OD	▽	○	●	○	●	●	●	○	○	○	●								3
4		6%	9	1	5		9	Rotational Speed: 0 - 120 RPM (Variab	▽	▽	▽	●	○	○	○	▽	●	●	▽								4
5		6%	9	1	5		9	Rapid Feed Rate	▽	●	▽	○	○	○	○	▽	●	●	▽								5
6		14%	9	9	9		9	OSHA Compliant	○	○	○	○	○	○	○	○	●	●	●								6
7		14%	9	9	9		9	Decreased Load Time	○	▽	▽	▽	▽	▽	▽	○	●	●	●								7
8		11%	9	9	5		9	High Quality Production	▽	●	●	○	○	○	●	▽	○	○	▽								8
9		8%	9	1	7		9	Part must be 60" above ground	▽	▽	○	▽	▽	▽	▽	●	▽	●	○								9
10		8%	9	1	8		9	Part Lubrication	○	○	▽	▽	▽	○	●	○	●	●	▽								10
11		8%	9	1	8		9	Equipment Lubrication	▽	○	○	○	○	○	●	▽	●	▽	▽								11
12		0%																									12
13		0%																									13
14		0%																									14
15		0%																									15
16		0%																									16

0 1 2 3 4 5

X values column cannot be of type string

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
HOW MUCH: Target	35 ft	150 HP	20 Degrees	400 HP	24"	500 Tons	1000 Tons	60 Inches above Ground								
Max Relationship	9	9	9	9	9	9	9	9	9	9	9					
Technical Importance Rating	295.9938	351.3816	321.5100													
Relative Weight	31%	36%	33%													
Weight Chart																
Curre Produ — Our Product	9	5	4	2												

PCC Rollmet Machine (Big)	2	3	5	1												
PCC Rollmet Machine (small)	5	1	4	5												

Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16

Template Revision: 0.9 Date: 4/23/2010
Christopher Battles

Design Criteria	Size	Ease of Operation	Handling *size	Reliability	Repair	Durability	Operation Cost	Capital Cost	Loading Speed	Overall Score
Wieght	0.1	0.1	0.05	0.21	0.05	0.15	0.05	0.05	0.24	1
Crane	10	6	10	10	7	8	6	9	5	7.7
Ramp	5	8	8	8	10	9	10	10	9	8.39
Conveyer Belt	4	9	8	9	7	7	7	7	8.5	7.73
Lever Arm	7	10	7	10	7	9	8	6	10	8.95
Access Port	8	10	7	10	7	8	9	7	9.5	8.88
Rear Loader	6	8	6	9	7	7	7	6	7.5	7.44
Zipline	6	7	6	8	8	7	8	7	8	7.4
Teter Totter	5	9	8	8	8	8	8	6	8.5	7.82
Underneath Ramp	8	10	7	9	9	10	10	8	9.5	9.17

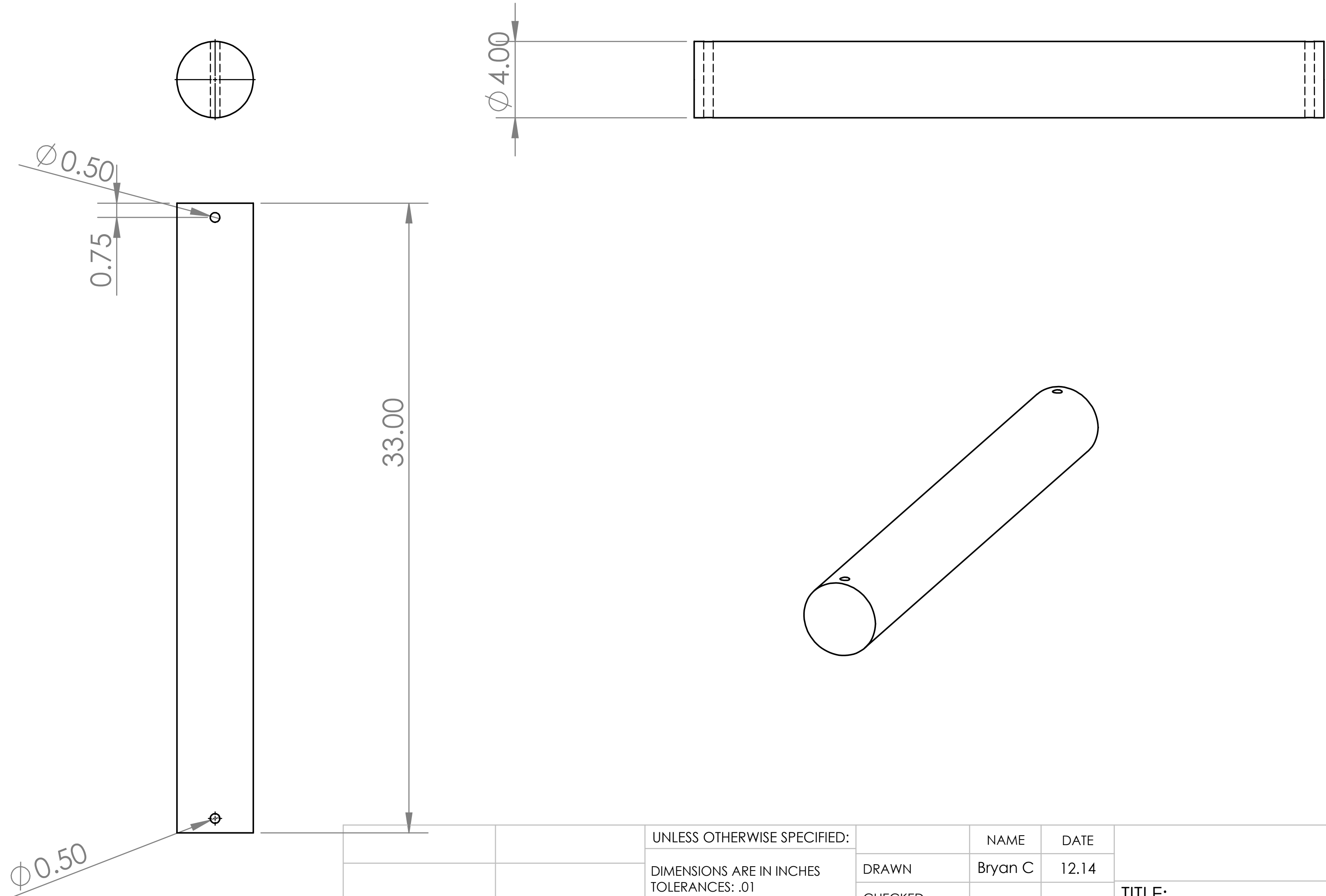
Top 3
Underneath Ramp
Lever Arm
Access Port

Wildcard
Rear Loader

Design Criteria	Ease of Operation	Speed of Process	Efficiency	Reliability	Repair	Capitol Cost	Size Effects	Location of Lub			Durability	Overall Score
								Mandrel	Outside Pipe	Inside Pipe		
Weight	0.11	0.12	0.07	0.07	0.07	0.05	0.05	0.12	0.12	0.12	0.1	1
Robot	6	7	10	10	6	5	8	5	5	0	6	5.17
Overhead Spray	9	10	7	10	7.5	7.5	7	5	5	0	8	5.83
Folgers on a Stick	8	8	7	9	10	10	10	5	5	5	10	5.86
Spinner	6	7	8	10	6	6	8	0	0	5	6	3.88
Water Fall	10	10	10	10	8	8	9	5	5	0	8	6.31
Low Friction Bushing	7	5	7	10	10	9	6	5	0	5	10	4.61
Lube with loading												0

Design Criteria	Speed of Return	Power	Reliability	Duribility	Repair	Size	Cost	Overall Score
Weight	0.2	0.18	0.12	0.14	0.16	0.1	0.1	1
Power Screw	5	10	10	10	7	8	8	8.12
Hydraulics	9	10	9	9	6	6	7	8.2
Chain Drive	8	7	9	7	8	8	10	8
Cable	10	6	9	8	10	8	6	8.28
Belt	8	5	9	6	8	8	10	7.5
Combinations								0
								0

APPENDIX B



UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: .01
FRACTIONAL ±.005
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C	12.14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

NEXT ASSY	USED ON
APPLICATION	

PROPRIETARY AND CONFIDENTIAL

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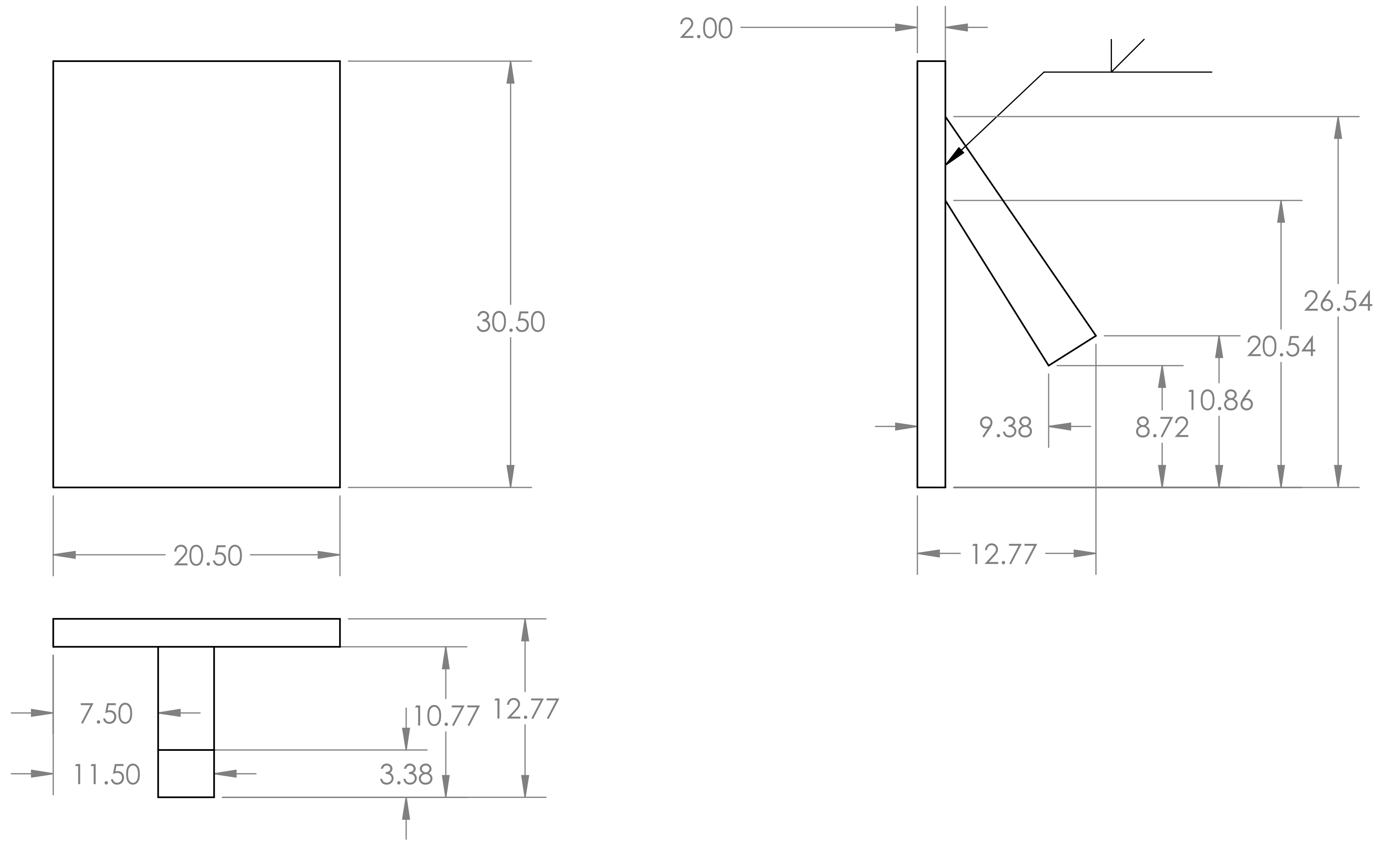
TITLE:

ArmBasePin

SIZE A | DWG. NO. | REV

SCALE: 1:8 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 Steel	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

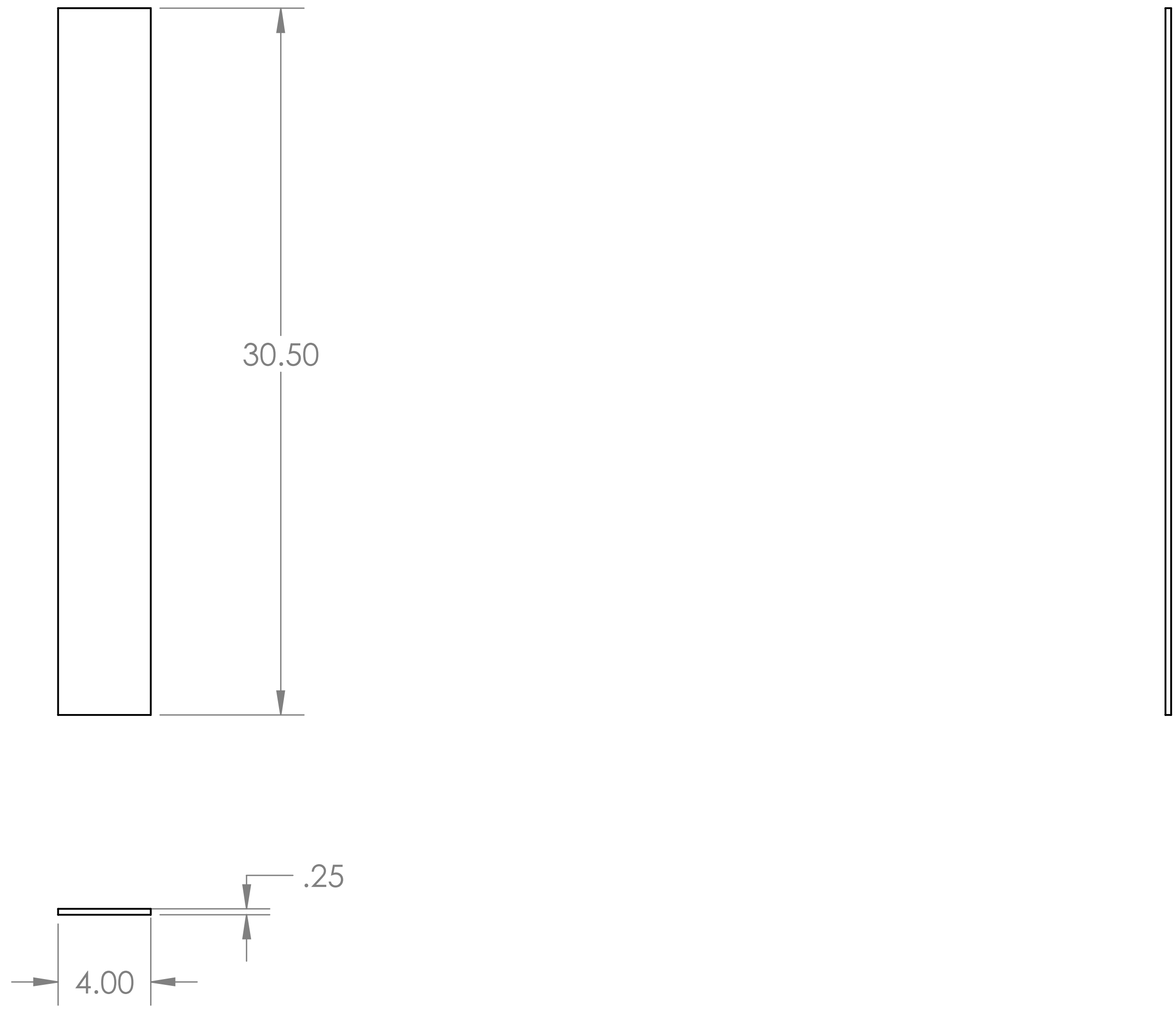
TITLE:

Bottom Arm Back Plate

SIZE	DWG. NO.	REV
A		

SCALE: 1:12 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



			NAME	DATE
		DRAWN	Bryan C	12.4.14
		CHECKED		
		ENG APPR.		
		MFG APPR.		
		Q.A.		
NEXT ASSY	USED ON	COMMENTS:		
APPLICATION				

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:.01
FRACTIONAL ±.005
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
Brass

FINISH

DO NOT SCALE DRAWING

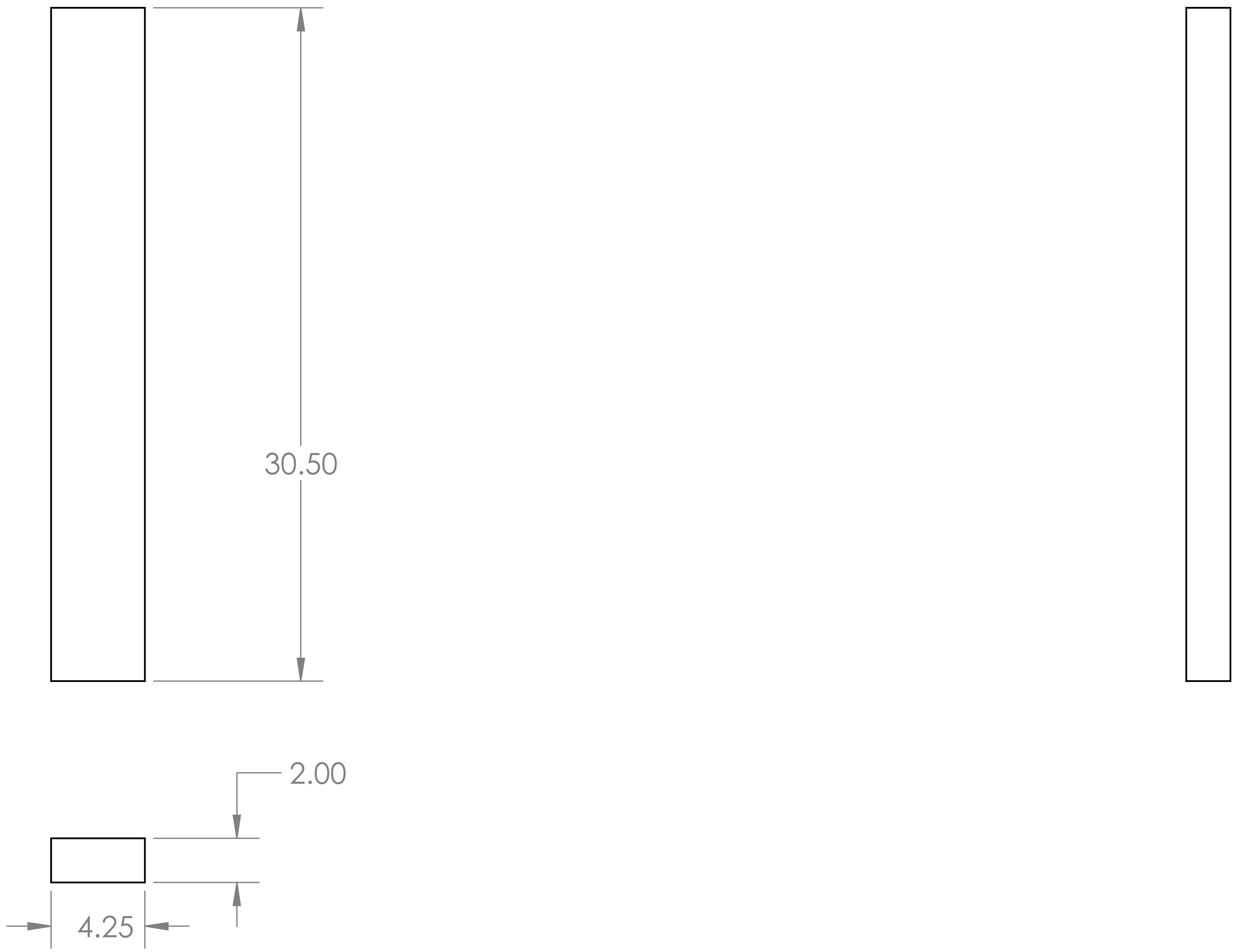
TITLE:

Bottom Arm Front & Back Brass Plate

SIZE **A** DWG. NO. REV

SCALE: 1:8 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



PROPRIETARY AND CONFIDENTIAL

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: .01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1018 Steel

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C	12/4/14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

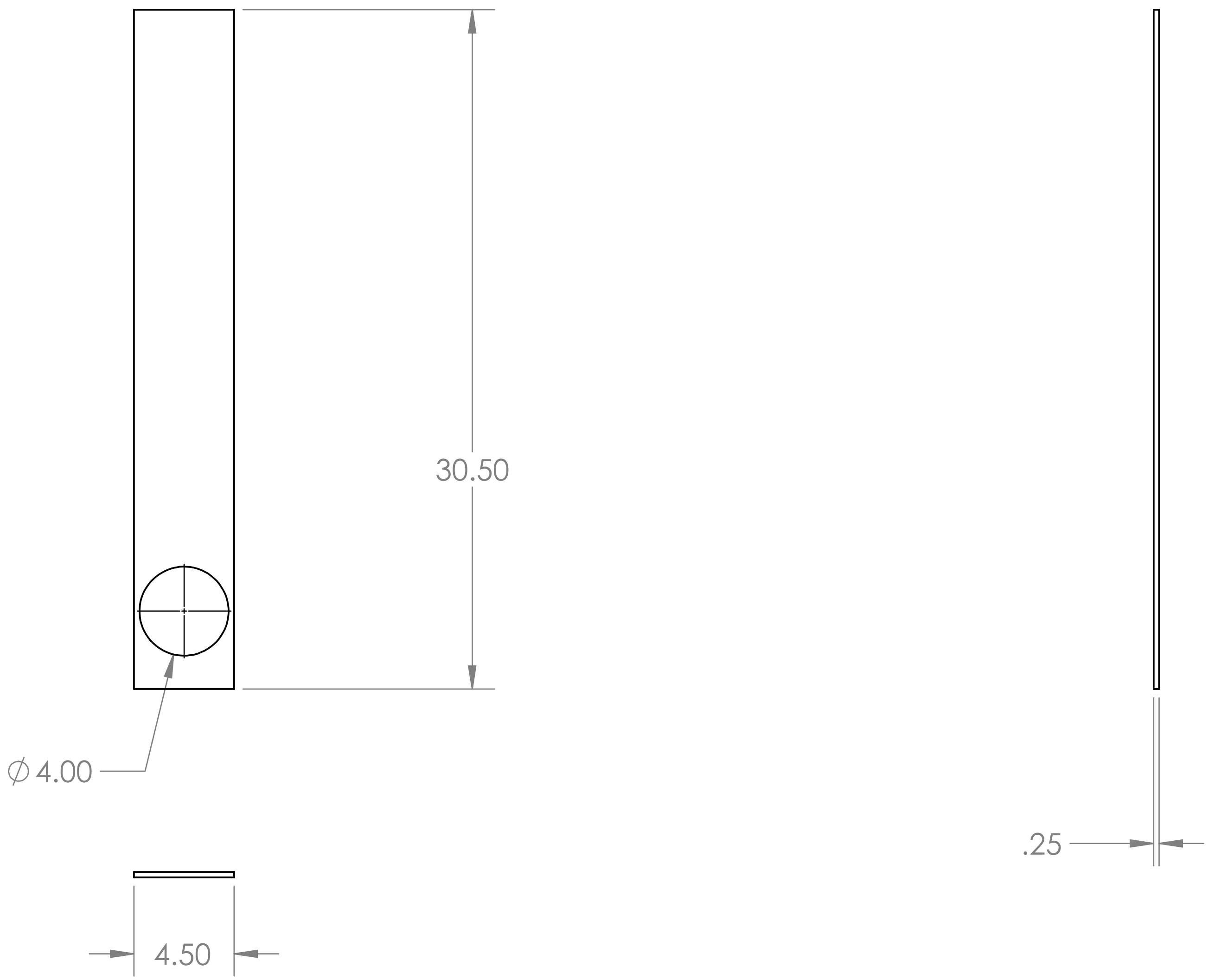
TITLE:

Bottom Arm Front Plate

SIZE	DWG. NO.	REV
A		

SCALE: 1:8 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:.01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
Brass

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C	12/4/14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

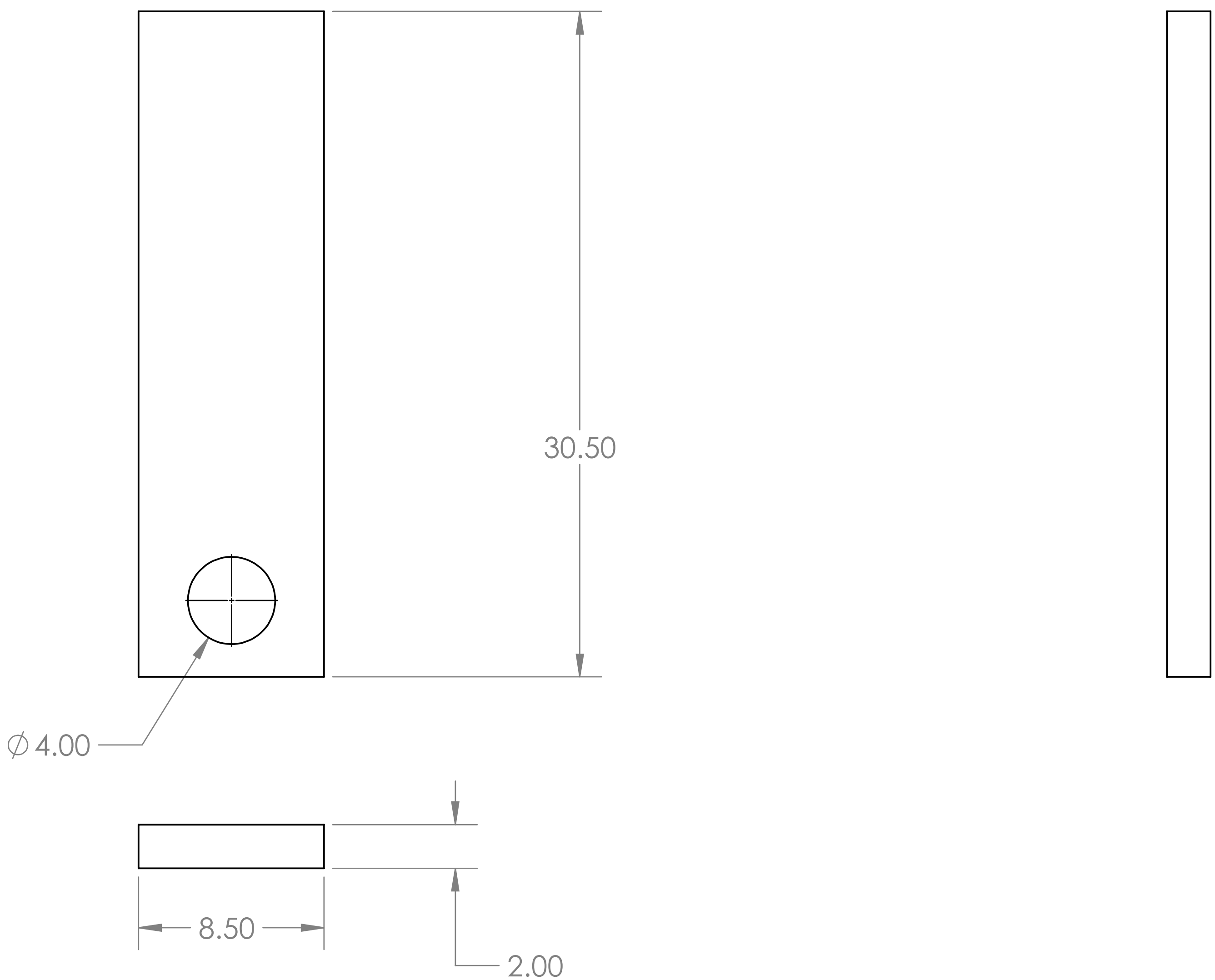
TITLE:

Bottom Arm Side Brass Plate

SIZE	DWG. NO.	REV
A		

SCALE: 1:8 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: .01
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1018 Steel

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C	12.4.14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

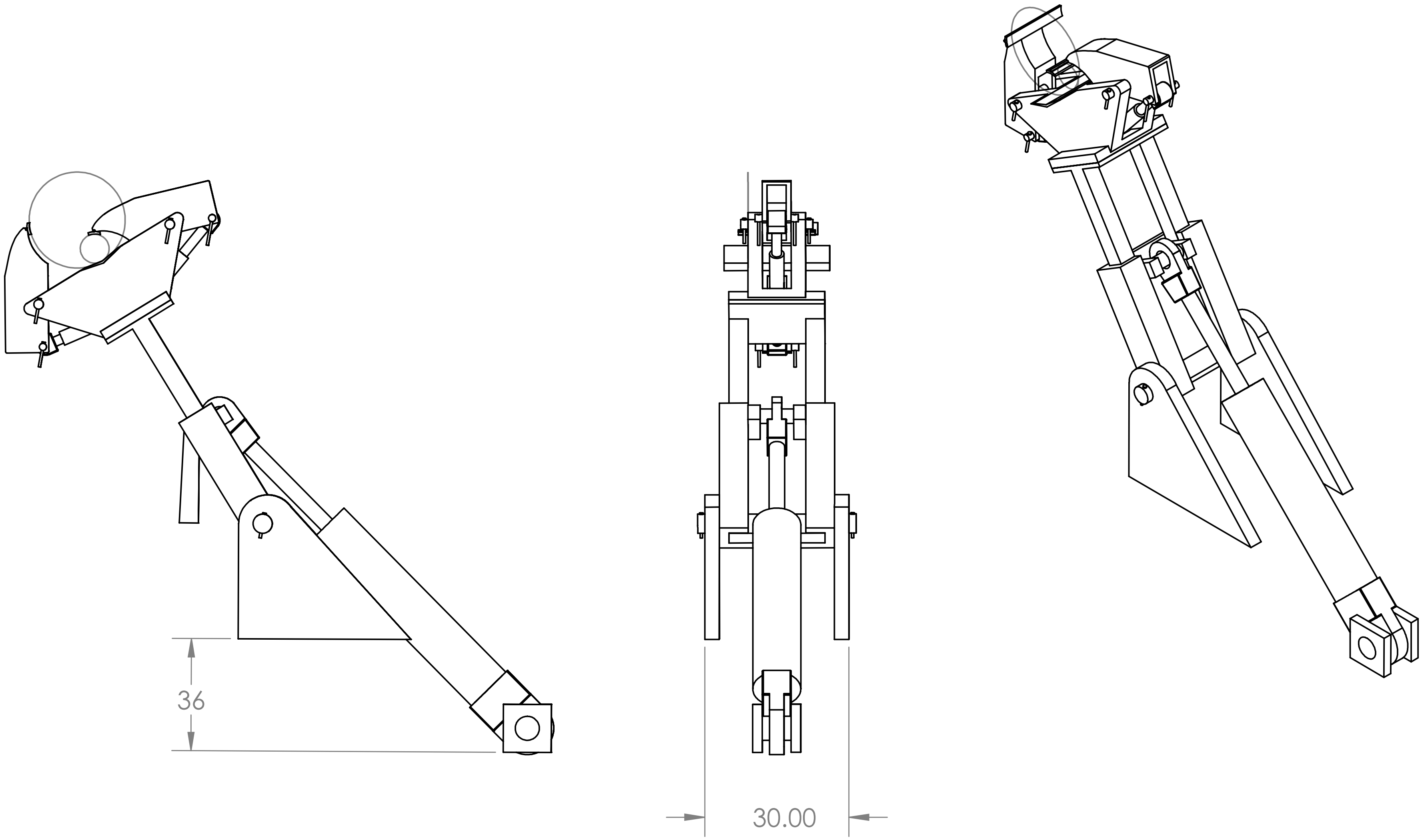
TITLE:

Bottom Arm Side Plate

SIZE A | DWG. NO. | REV

SCALE: 1:8 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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			CHECKED		
			ENG APPR.	David V	12/5/14
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

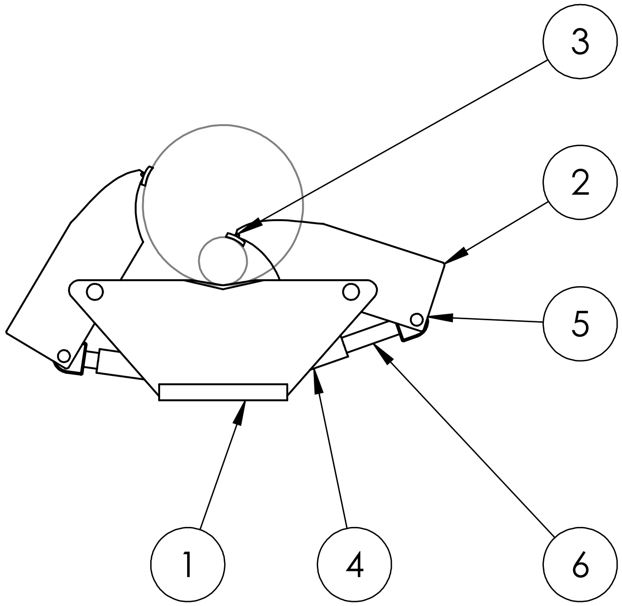
TITLE:

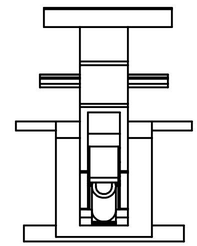
Final Assembly

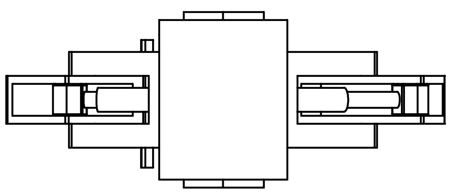
SIZE	DWG. NO.	REV
A		

SCALE:1/32 WEIGHT: SHEET 1 OF 1

5 4 3 2 1







ITEM NO.	PART NUMBER	QTY.
1	GripperBasePlate	1
2	GripperFinger	2
3	GrippingContact	2
4	BaseTubeHydraulic Gripper	2
5	CapHydraulicGripp er	2
6	UpperTubeHydrauli cGripper	2

NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C	12/14
CHECKED		
ENG APPR.	David V	12/14
MFG APPR.		
Q.A.		

COMMENTS:

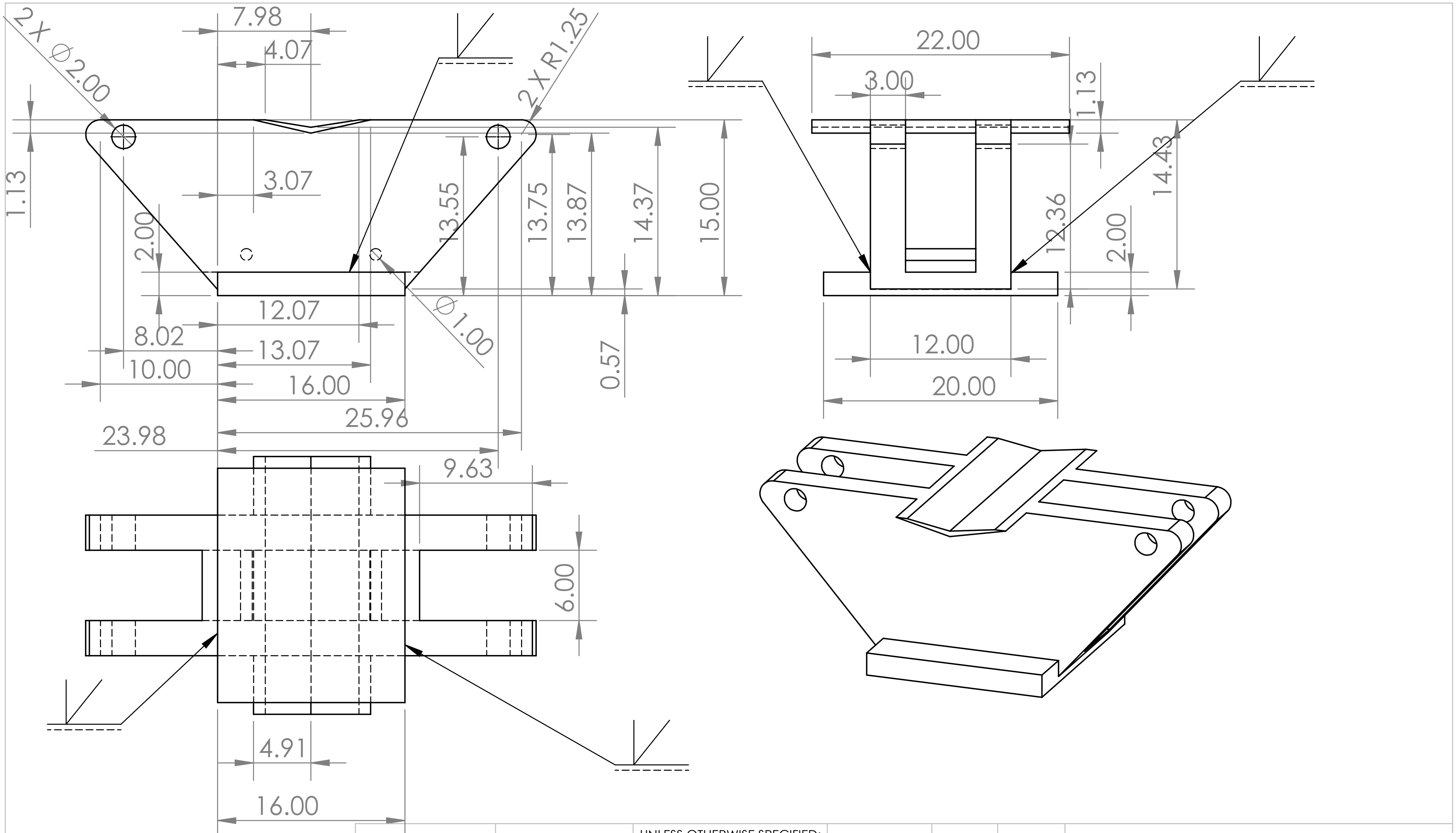
TITLE:

Gripper Assembly

SIZE **A** DWG. NO. REV

SCALE: 1:24 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES: 0.01
FRACTIONAL ±0.005
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C	12/14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

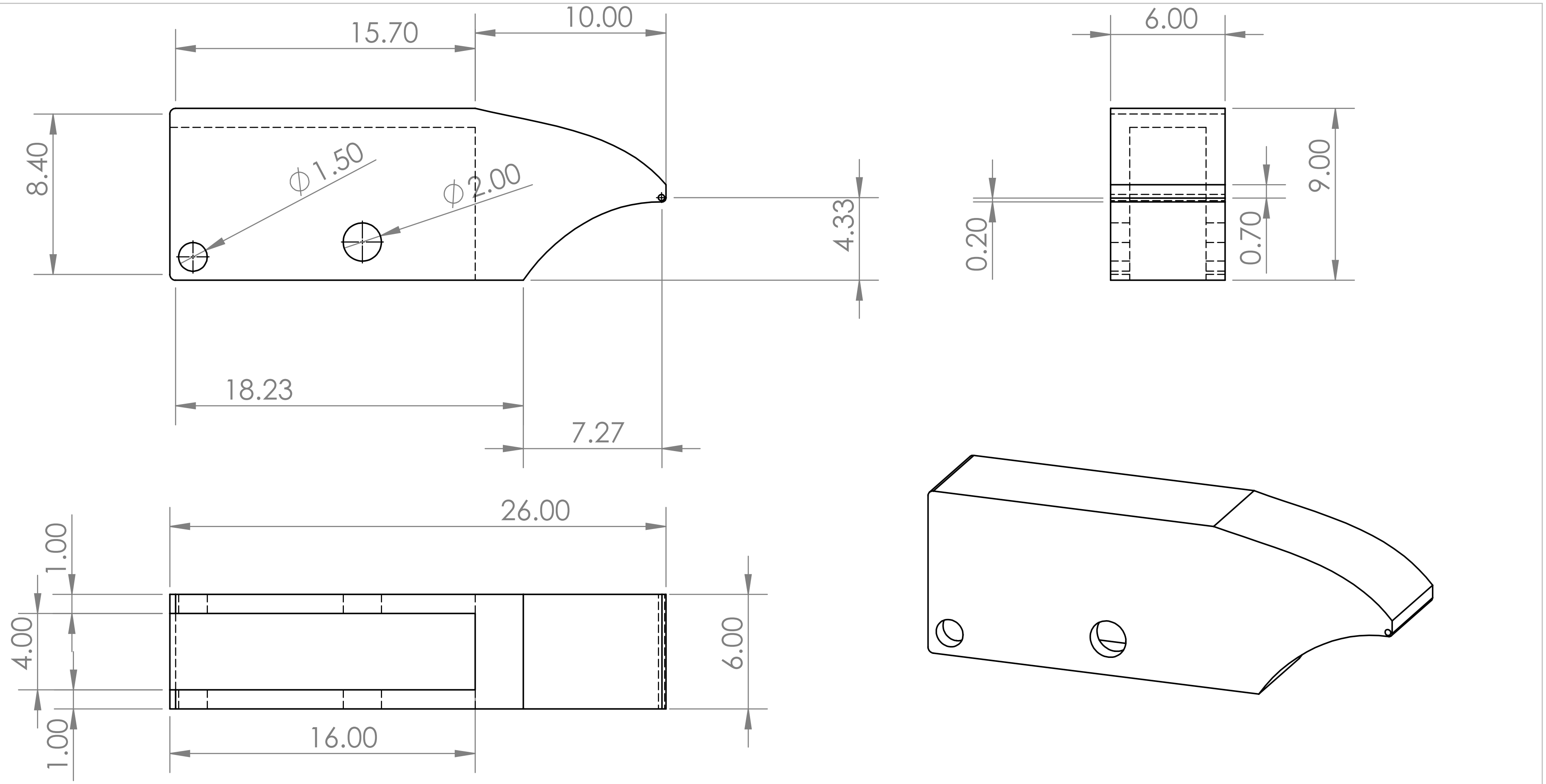
TITLE:

GripperBasePlate

SIZE **A** DWG. NO. REV

SCALE: 1:12 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



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			CHECKED		
			ENG APPR.		
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

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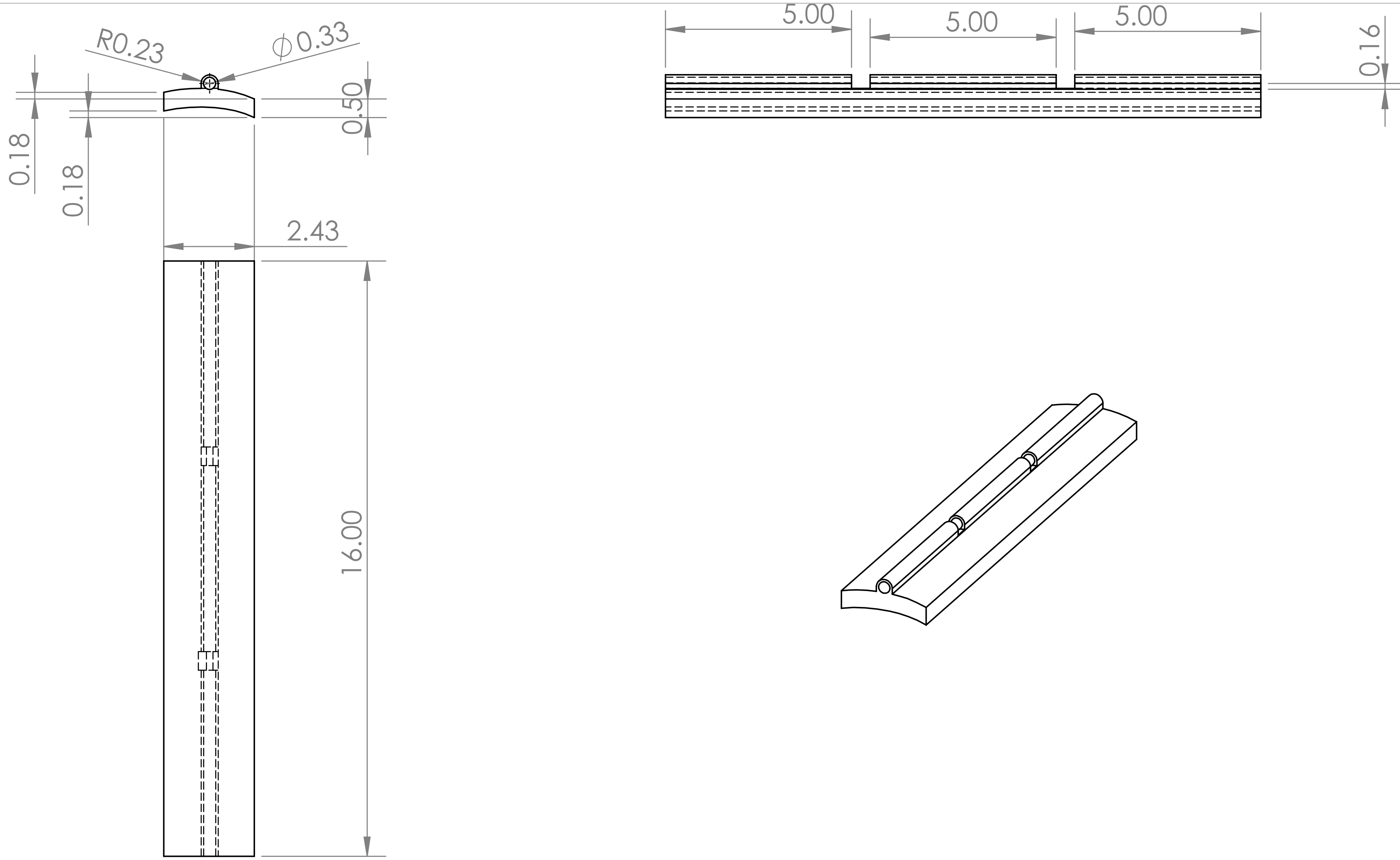
TITLE:

GripperFinger

SIZE	DWG. NO.	REV
A		

SCALE: 1:8 WEIGHT: SHEET 1 OF 1

5 4 3 2 1



		UNLESS OTHERWISE SPECIFIED:
		DIMENSIONS ARE IN INCHES TOLERANCES: .01 FRACTIONAL ±.005 ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± THREE PLACE DECIMAL ±
		INTERPRET GEOMETRIC TOLERANCING PER:
		MATERIAL 1018 Steel
NEXT ASSY	USED ON	FINISH
APPLICATION		DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C	12/14
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		

COMMENTS:

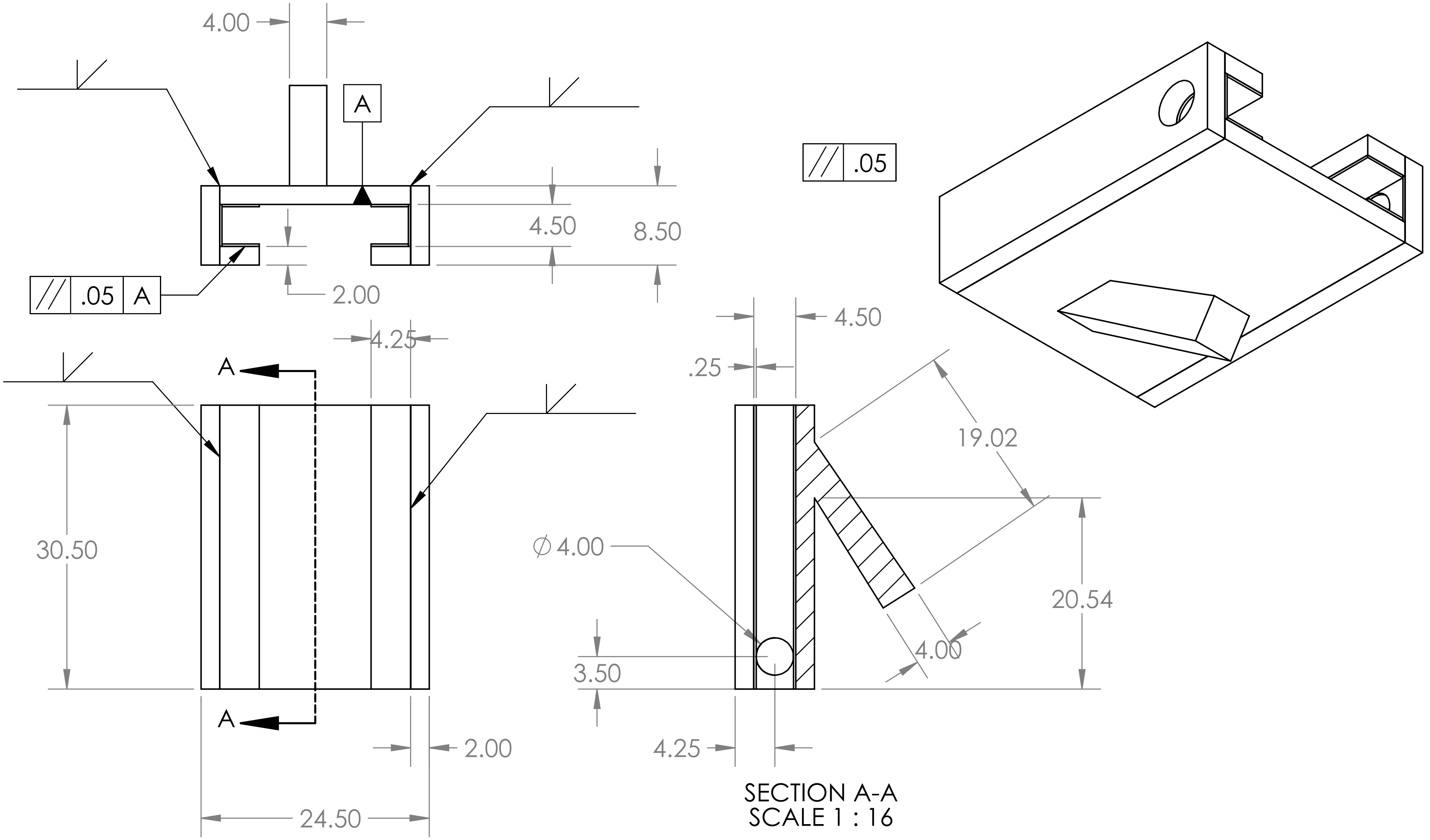
TITLE:

GrippingContact

SIZE	DWG. NO.	REV
A		

SCALE: 1:4 | WEIGHT: | SHEET 1 OF 1

5 4 3 2 1



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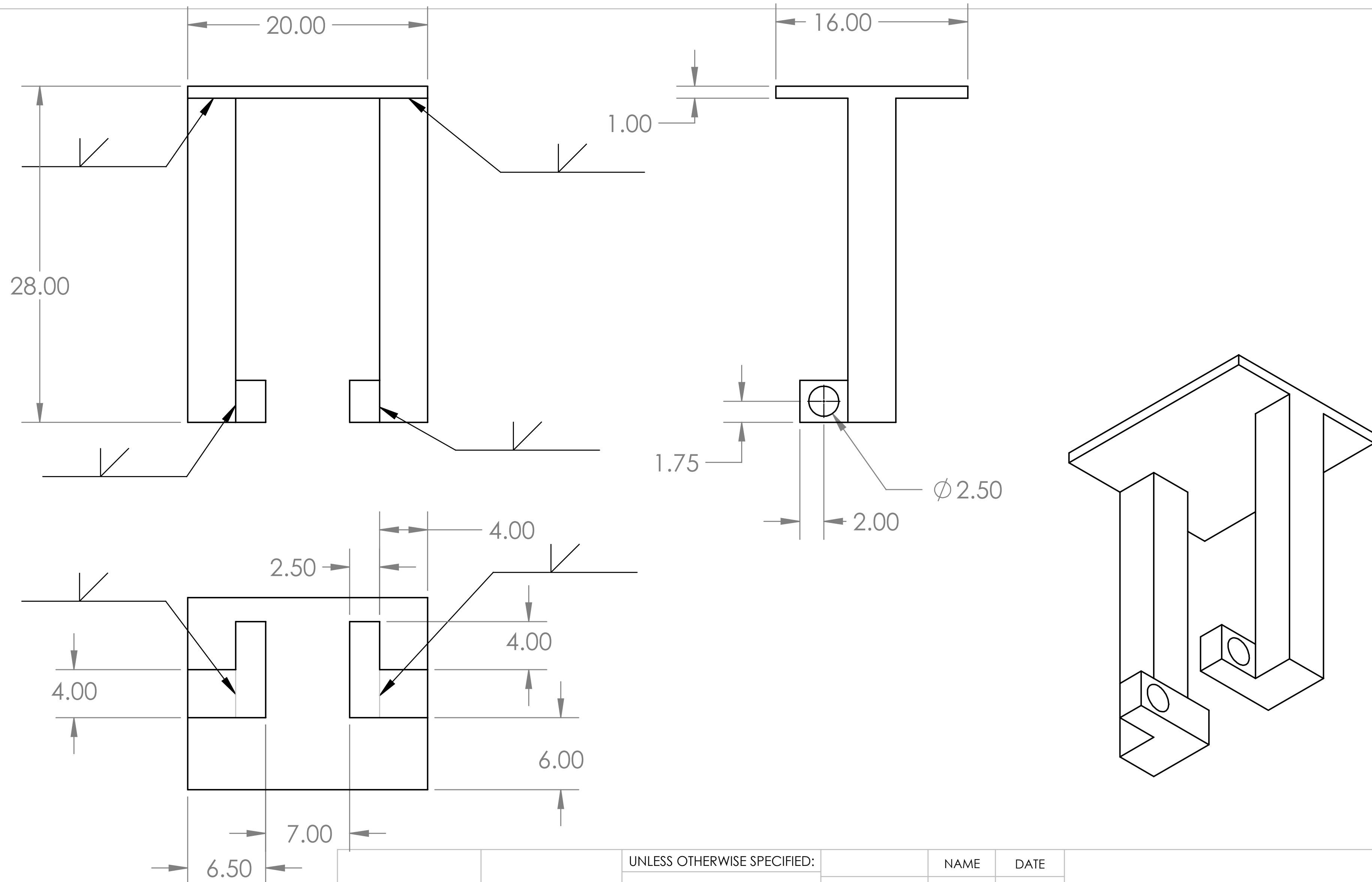
			CHECKED		
			ENG APPR.	David V.	12/5
			MFG APPR.		
		INTERPRET GEOMETRIC TOLERANCING PER:	Q.A.		
		MATERIAL 1018 Steel	COMMENTS:		
NEXT ASSY	USED ON	FINISH			
APPLICATION		DO NOT SCALE DRAWING			

TITLE:

Lower Arm Asm

SIZE	DWG. NO.	REV
A		
SCALE: 1:1		SHEET 1 OF 1

5 4 3 2 1



Lever Arm	
NEXT ASSY	USED ON
APPLICATION	

UNLESS OTHERWISE SPECIFIED:

DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±
THREE PLACE DECIMAL ±

INTERPRET GEOMETRIC TOLERANCING PER:

MATERIAL
1018 Steel

FINISH

DO NOT SCALE DRAWING

	NAME	DATE
DRAWN	Bryan C.	12/4/14
CHECKED		
ENG APPR.	David V.	12/5/14
MFG APPR.		
Q.A.		

COMMENTS:

TITLE:

Upper Lever Arm

SIZE	DWG. NO.	REV
A		

SCALE: 1:12 WEIGHT: SHEET 1 OF 1

5 4 3 2 1

APPENDIX C

Vendors	Contact Info	Component	Pricing
Parker Hannifin	1-800-272-7537	Hydraulic Piston	~$700.00
Woods Logging and Industrial Supply	1-360-577-8030	Wire Rope	$12.31 per foot
eSheaves	1-860-449-1128	Sheaves	~$10000.00
Baytec Containers	1-888-460-3786	Drum Pump	$932.80

APPENDIX D

Rope Size: 1 3/8" Rope
Outside Dimensions: 52" OD
Bearing Type: Bronze Sleeve Bearing
Shaft Size: 10 Shaft Size
Hub Width: 3.625" Hub Width

Item # 455174044

Check price and availability

Check with us for delivery. May not be a stock part

Description:
1 3/8 Rope, 52" OD Bronze Sleeve Bearing 10 shaft size 3.625" Hub Width Forgefab

Features:

- 51 Approx wt lbs.,3.25 Rim Width 1.5 Web Thickness 14" Hub O.D Forgefab

Technical Specs:

- 6 to 8 weeks estimated delivery, call for current availability

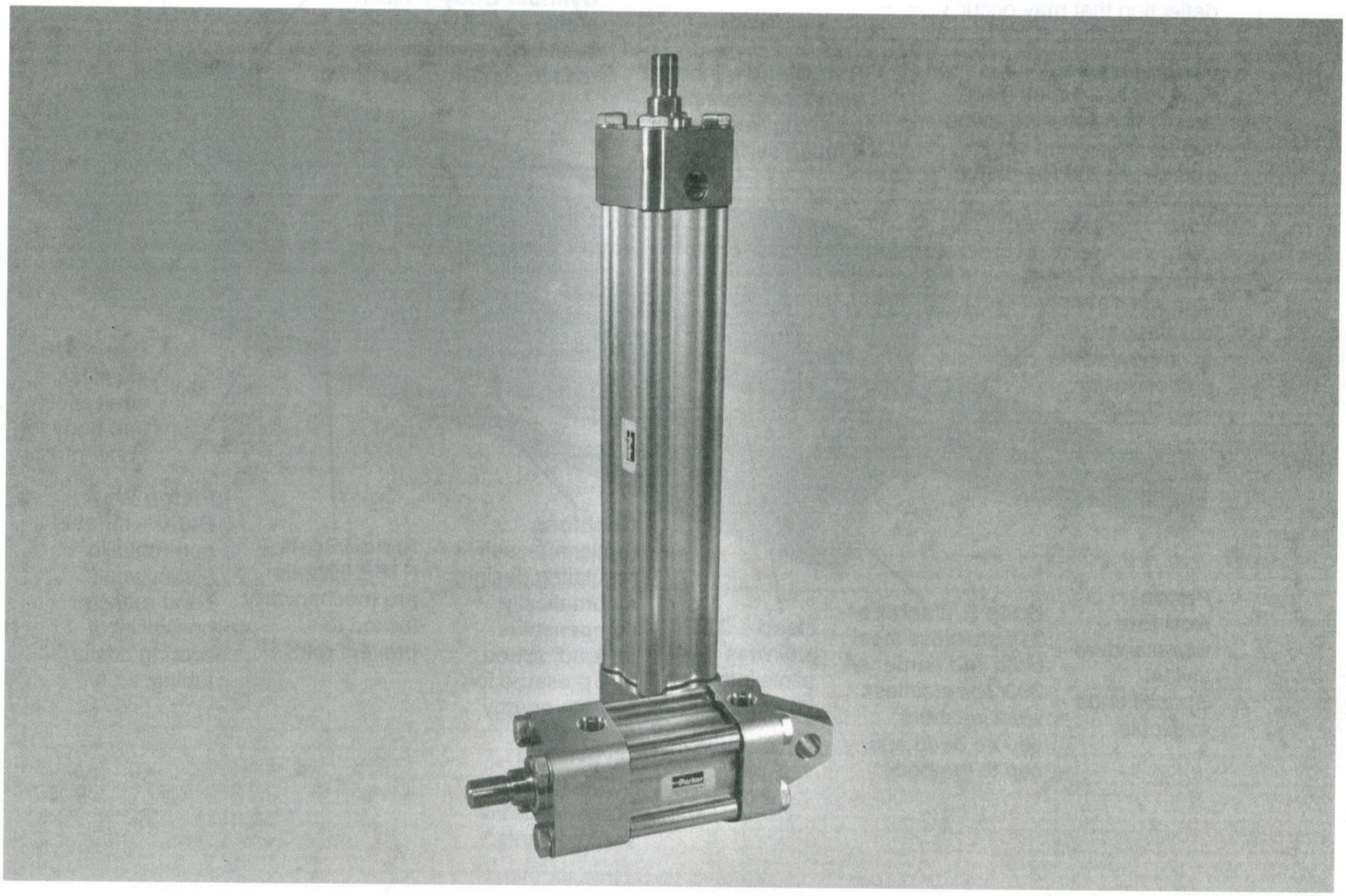
Reference:

- 6 to 8 weeks estimated delivery, call for current availability

Series SH/SHG

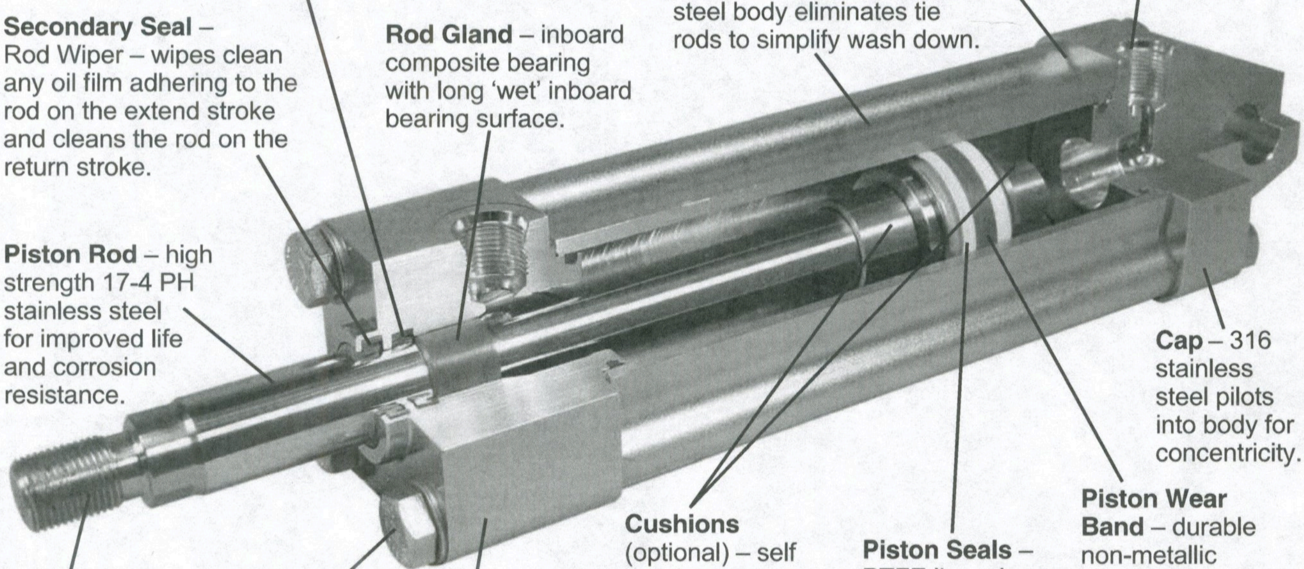
Stainless Steel Hydraulic Cylinders

Catalog HY08-1138-4/NA
January, 2011



Series SH – Your best value in Heavy Duty Stainless Steel Hydraulic Cylinders

- 316 stainless steel heavy duty hydraulic cylinder construction offers superior corrosion resistance for greater durability in harsh environments
- Integral stainless steel gland-in-head design for longer life than traditional retained style gland materials
- Self-regulating cushions do not require adjustment which reduces set-up cost*



* When cushions are specified on 1½", 2", 2½" bores with oversize rod, a fixed cushion without adjusting screw is supplied on the head end. Fixed cushion rate of deceleration is constant and does not vary with load, speed, and pressure like self-regulating style cushions.

** Hi-Load piston is supplied as standard in 1½", 2" and 2½" bores with oversize rod.

Series SHG – Your best value in heavy duty hydraulic cylinders for food processing applications

In addition to features of Series SH cylinders, Series SHG includes exterior surfaces that are electropolished, food grade wiperseal material, Stat-O-Seal™ washers under bolt heads for maximum protection against contamination, and assembly with H-1 rated lubricant.

Series SHG Feature	Benefit	Series SHG Value
Rounded Corners on Head and Cap and "Clean" Profile Body without Tie Rods	Material does not collect around cylinder tie rods or build up on head and cap surfaces. Extruded construction simplifies repairability compared to tie rod style.	Wash down time is reduced with 'clean' profile style body and rounded head and cap corners. Equipment down time for repair is reduced.
Standard PTFE Piston Seals with Non-Metallic Piston Wear Band	PTFE piston seals have greater fluid and temperature resistance than elastomeric seals. Non-metallic wear band for greater protection against scoring costly tubing.	Longer piston seal and tube life than typical cylinders used in food processing applications.
Self-Regulating Cushions	Automatic compensation for load, speed, and pressure ensures optimal cushion performance. Saves manual adjustment labor. No needle or check valve cavities to collect contamination.	Time required for machine set-up, operational adjustments and cleaning is reduced.
Electropolished Cylinder Exterior	Electropolishing significantly improves corrosion resistance and improves surface condition to resist bacteria.	Reduced corrosion increases cylinder life and replacement interval. Added bacterial resistance promotes better food handling practices.
Integral Gland with Food Grade Wiperseal	Integral stainless steel gland with inboard bearing is more corrosion resistant than traditional retained style gland materials. Food grade wiperseal accepts food contact.	Gland service interval is increased which reduces maintenance cost.
USDA H-1 Rated Fluid Used for Cylinder Assembly Lubrication and Testing	Eliminates possibility of hydraulic system contamination by unknown lubricants from the cylinder.	Reduced setup time by eliminating need to purge cylinder.

Table of Contents

Page No.

In line with our policy of continuing product improvement, specifications and information contained in this catalog are subject to change.

⚠ Warning

FAILURE OR IMPROPER SELECTION OR IMPROPER USE OF THE PRODUCTS AND/OR SYSTEMS DESCRIBED HEREIN OR RELATED ITEMS CAN CAUSE DEATH, PERSONAL INJURY AND PROPERTY DAMAGE.

This document and other information from Parker Hannifin Corporation, its subsidiaries and authorized distributors provide product and/or system options for further investigation by users having technical expertise. It is important that you analyze all aspects of your application, including consequences of any failure and review the information concerning the product or system in the current product catalog. Due to the variety of operating conditions and applications for these products or systems, the user, through its own analysis and testing, is solely responsible for making the final selection of the products and systems and assuring that all performance, safety and warning requirements of the application are met.

The product described herein, including without limitation, product features, specifications, designs, availability and pricing, are subject to change by Parker Hannifin Corporation and its subsidiaries at any time without notice.

Offer of Sale

The items described in this document are hereby offered for sale by Parker Hannifin Corporation, its subsidiaries or its authorized distributors. This offer and its acceptance are governed by provisions stated on a separate page of the document entitled 'Offer of Sale'.

Model Ordering Code for SH and SHG

2"	C	K	BE	SHG	L*	T	V	S	1	4	A	C	12.000
Bore Size	Cushion Head	Double Rod Cylinder	Mounting Style	Series	Piston	Ports	Common Modifications	Special Modifications	Piston Rod Number	Piston Rod Thread Style	Piston Rod Thread Type	Cushion Cap	Stroke

(The K, V and S boxes are printed empty in the example code.)

Bore Size – Specify bore size in inches: 1½, 2, 2½, 3¼, 4

Cushion Head – Use "C" only if head end cushion is required.

Double Rod Cylinder – Use "K" only if double rod cylinder is required.

Mounting Style – Specify:
- T = No tie rods extended
- TB = Tie rods ext. head
- TC = Tie rods ext. cap
- TD = Tie rods extended both ends
- JJ = Head rectangular
- HH = Cap rectangular
- BE = Cap eye
- F = Side tapped
- D = Head trunnion

Series –
- SHG = Food Grade Stainless Steel Cylinder
- SH = Standard Stainless Steel Cylinder

Piston – L = Lipseal Piston

* 'K' Hi-Load piston is supplied as standard for 1½", 2" & 2½" bores with oversize rods and is not available in any other bore and rod combination.

Ports – Specify:
- T = SAE Ports
- U = NPTF Ports
- R = BSPP Ports

Common Modifications – V = Fluorocarbon

Special Modifications – Use "S" for Special Modification other than rod end, and specify modification.

Piston Rod Number – For Single Rod Cylinders, select one rod number only. Refer to rod number listing on pages 6 thru 15.
See Cylinder Stroke Chart on page 19 for minimum piston rod diameter.

Piston Rod Thread Style – Specify:
- 4 = Small Male
- 8 = Intermediate Male
- 9 = Female
- 55 = Flange Coupler
- 3 = Special (see note below)

To order special thread specify "3" and give the desired dimensions for KK, A, and W (WD for D mount or WF for JJ mount) or furnish a dimensioned sketch.

Piston Rod Thread Type – Specify:
- A = UNF
- M = Metric

Cushion Cap – Use "C" only if cap end cushion is required.

Stroke – Specify Stroke Length Required in inches.

Double Rod Cylinders
For double rod cylinders, specify rod number and rod end symbols for both piston rods. A typical double rod model number would be:
2" KJJ-SHGLT14A/14AX12"

Standard Specifications

- 6 Standard mounting styles
- Bore sizes – 1½" to 4"
- Strokes – up to 72"
- Piston Rod Diameters – ⅝" to 2½"
- Working pressure up to 3000 psi
- Single and double rod construction available
- Temperature range – -10°F (-23°C) to +250°F (+121°C) (depending on seal class)

Seal Classes	Typical Fluids	Temperature Range
1 – Standard Nitrile, Polyurethane & PTFE	**ISO Grade 32 NSF / USDA H-1 Oils** (Acceptable for use in Food Industry)– Approved products include Chevron FM32, Petro-Canada Purity FG AW32, Mobil DTE FM32, ConocoPhillips 200 **Hydraulic Oil, MIL-H-5606 Oil**	-10°F (-23°C) to +165°F (+74°C)
5 – Optional (At extra cost) Fluorocarbon Seals	High Temperature	-10°F (-23°C) to +250°F (+121°C) Class 5 seals may be operated up to +400°F (+204°C) with reduced service life

Note: Class 5 seals are not suitable for use with Skydrol fluid, but can be used with hydraulic oil if desired.
Rod seal, wiperseal, and body o-rings are fluorocarbon; piston seals are spring loaded PTFE.
Contact the factory before specifying Series SH or SHG for use with phosphate ester fluid.

Theoretical Push and Pull Forces

The cylinder output forces are derived from this formula:

$$F = P \times A$$

Where F = Force in pounds.
P = Pressure at the cylinder in pounds per square inch.
A = Effective area of cylinder piston in square inches.

To determine the bore size for the application, follow the steps below.

1. Select the Operating Pressure column closest to that desired.
2. In the same column, identify the force required to move the load (always rounding up). If the piston rod is in compression use the 'Push' row and if the piston rod is in tension use the 'Pull' row.
3. In the row to the left is the bore required. To select the correct rod diameter for the stroke required use the Piston Rod-Stroke Selection Chart on page 19.

If the cylinder envelope dimensions are too large for the application, increase the operating pressure to the maximum pressure in the table below, if possible, and repeat steps 1 - 3.

Pressure Ratings

Series SH and SHG hydraulic cylinders are recommended for pressures to **3000 psi** for heavy-duty hydraulic service with hydraulic oil.

Maximum Pressure Ratings

Bore Ø	Rod Ø	Heavy Duty Service (psi)
1 1/2	5/8, 1	3000
2	1, 1 3/8	3000
2 1/2	1, 1 3/4	3000
3 1/4	1 3/8, 2	3000
4	1 3/4, 2 1/2	3000

Push and Pull Force in Pounds

Bore Ø	Rod Ø	Operating Direction	Piston Area (inches²)	Operating Pressure in psi					
				100	250	500	1000	2000	3000
1 1/2	5/8	Push	1.767	177	442	884	1767	3534	5301
		Pull	1.460	146	365	730	1460	2920	4380
	1	Push	1.767	177	442	884	1767	3534	5301
		Pull	0.982	98	246	491	982	1964	2946
2	1	Push	3.142	314	786	1571	3142	6284	9426
		Pull	2.357	236	589	1179	2357	4714	7071
	1 3/8	Push	3.142	314	786	1571	3142	6284	9426
		Pull	1.652	165	413	826	1652	3304	4956
2 1/2	1	Push	4.909	491	1227	2455	4909	9818	14727
		Pull	4.124	412	1031	2062	4124	8248	12372
	1 3/4	Push	4.909	491	1227	2455	4909	9818	14727
		Pull	2.499	250	625	1250	2499	4998	7497
3 1/4	1 3/8	Push	8.296	830	2074	4148	8296	16592	24888
		Pull	6.806	681	1702	3403	6806	13612	20418
	2	Push	8.296	830	2074	4148	8296	16592	24888
		Pull	5.154	515	1289	2577	5154	10308	15462
4	1 3/4	Push	12.566	1257	3142	6283	12566	25132	37698
		Pull	10.156	1016	2539	5078	10156	20312	30468
	2 1/2	Push	12.566	1257	3142	6283	12566	25132	37698
		Pull	7.656	766	1914	3828	7656	15312	22968

Cylinder Weights

To determine the weight of a Series SH or SHG cylinder, first select the basic zero stroke weight for the mounting required, and then calculate the weight of the cylinder stroke and add the results to the basic weight. For extra rod extension, use piston rod weights per inch in Table C.

Table A – Single Rod End SH & SHG Cylinder Weights in Pounds

Bore Ø	Rod Ø	Single Rod Cylinders Basic Weight - Zero Stroke					Add Per Inch of Stroke
		T, TB, TC, TD, F	JJ	D	BE	HH	
1 1/2	5/8	7.3	9.1	7.8	7.7	8.4	0.7
	1	8.2	10.0	8.6	8.6	9.3	0.8
2	1	11.8	14.8	13.1	12.9	13.4	1.1
	1 3/8	13.3	16.3	14.5	14.4	14.8	1.3
2 1/2	1	17.1	20.7	18.3	18.6	19.0	1.6
	1 3/4	20.7	24.3	21.9	22.2	22.6	2.0
3 1/4	1 3/8	33.5	40.2	36.0	36.3	37.1	2.4
	2	37.6	44.4	40.2	40.5	41.2	2.9
4	1 3/4	45.8	53.8	48.2	51.8	49.9	3.2
	2 1/2	53.4	61.4	55.8	59.4	57.5	3.9

Table B – Double Rod End SH & SHG Cylinder Weights in Pounds

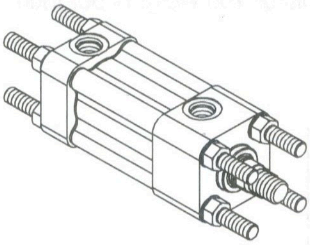
Bore Ø	Rod Ø	Double Rod Cylinders Basic Weight - Zero Stroke			Add Per Inch of Stroke
		KT, KTB, KTD, KF	KJJ	KD	
1 1/2	5/8	8.7	10.5	9.2	0.8
	1	10.5	12.3	11.0	1.1
2	1	14.7	17.7	16.0	1.3
	1 3/8	17.6	20.6	18.8	1.7
2 1/2	1	20.8	24.4	22.1	1.8
	1 3/4	28.0	31.6	29.2	2.7
3 1/4	1 3/8	40.8	47.6	43.4	2.8
	2	49.2	55.9	51.7	3.8
4	1 3/4	56.9	65.0	59.3	3.9
	2 1/2	72.1	80.2	74.5	5.3

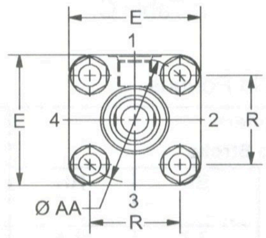
Table C – Piston Rod Weights in Pounds

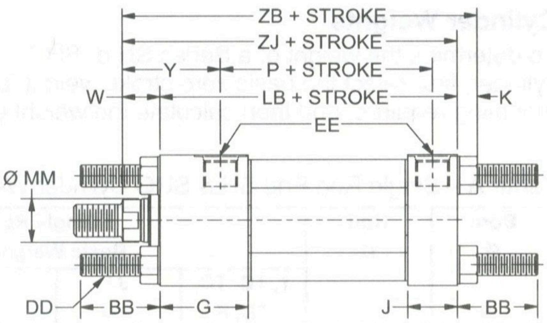
Rod Ø	Piston Rod Weight Per Inch
5/8	0.09
1	0.22
1 3/8	0.42
1 3/4	0.68
2	0.89
2 1/2	1.40

TD Mount – Single Rod End*

1½" to 4" Bore Size







T, TB, TC, TD Mount Single Rod End – Envelope and Mounting Dimensions

Bore Ø	AA	BB	DD	E	EE NPTF	EE SAE	G	J	K Max	R	Add Stroke LB	Add Stroke P
1 1/2	2.3	1 3/8	3/8-24	2 1/2	1/2	8	1 3/4	1 1/8	7/16	1.63	5	3 1/4
2	2.9	1 13/16	1/2-20	3	1/2	8	2	1 1/8	1/2	2.05	5 1/4	3 5/16
2 1/2	3.6	1 13/16	1/2-20	3 1/2	1/2	8	2	1 1/8	1/2	2.55	5 3/8	3 7/16
3 1/4	4.6	2 5/16	5/8-18	4 1/2	3/4	10	2 3/8	1 3/8	5/8	3.25	6 1/4	3 15/16
4	5.4	2 5/16	5/8-18	5	3/4	12	2 1/2	1 3/8	5/8	3.82	6 5/8	4 1/4

T, TB, TC, TD Mount Single Rod End – Rod Dimensions

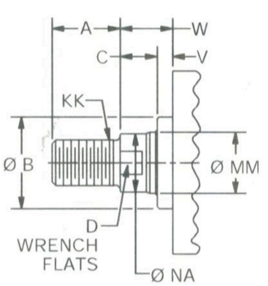
Bore Ø	Rod No.	MM Rod Ø	A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	W	WH
			Rod Extensions and Pilot Dimensions											
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	5/8	3/4
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1	1 1/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1	1 1/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 1/4	1 3/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	7/8	15/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 1/4	1 5/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1	15/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	1 3/8	1 11/16

Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke ZB	Add Stroke ZJ
1 1/2	1	5/8	1/2-20	7/16-20	1 13/16	6 1/16	5 5/8
	2	1	7/8-14	3/4-16	2 3/16	6 7/16	6
2	1	1	7/8-14	3/4-16	2 1/8	6 1/2	6
	2	1 3/8	1 1/4-12	1-14	2 3/8	6 3/4	6 1/4
2 1/2	1	1	7/8-14	3/4-16	2 1/8	6 5/8	6 1/8
	2	1 3/4	1 1/2-12	1 1/4-12	2 5/8	7 1/8	6 5/8
3 1/4	1	1 3/8	1 1/4-12	1-14	2 1/2	7 3/4	7 1/8
	2	2	1 3/4-12	1 1/2-12	2 7/8	8 1/8	7 1/2
4	1	1 3/4	1 1/2-12	1 1/4-12	2 11/16	8 1/4	7 5/8
	2	2 1/2	2 1/4-12	1 7/8-12	3 1/16	8 5/8	8

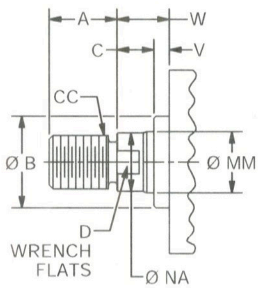
*Style T – no tie rods extended, Style TB – tie rods extended head end, and Style TC – tie rods extended cap end can be dimensioned from Style TD shown.

Rod End Dimensions

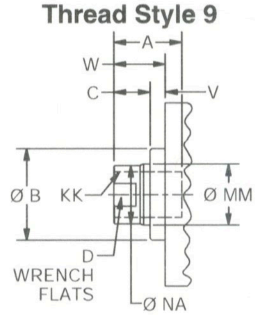
Thread Style 4



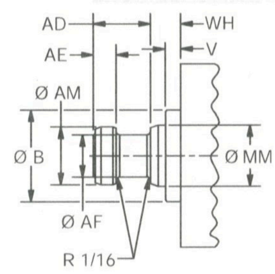
Thread Style 8



Thread Style 9



Style 55



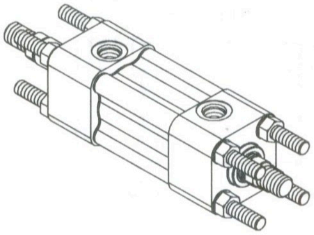
"Special" Thread Style 3

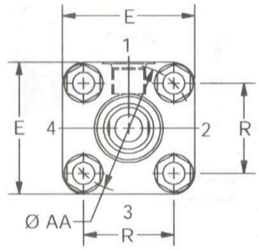
Special thread, extension, rod eye, blank, etc. are also available.

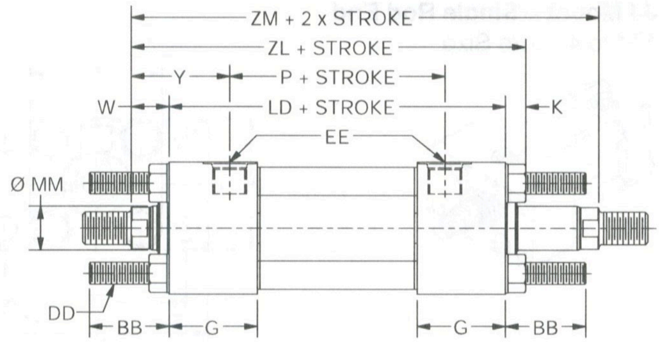
To order, specify "Style 3" and give desired dimensions for KK, A, & W. If otherwise special furnish dimensional sketch.

TD Mount – Double Rod End*

1½" to 4" Bore Size







T, TB, TD Mount Double Rod End – Envelope and Mounting Dimensions

Bore Ø	AA	BB	DD	E	EE NPTF	EE SAE	G	K Max	R	Add Stroke LD	Add Stroke P
1 1/2	2.3	1 3/8	3/8-24	2 1/2	1/2	8	1 3/4	7/16	1.63	5 5/8	3 1/4
2	2.9	1 13/16	1/2-20	3	1/2	8	2	1/2	2.05	6 1/8	3 3/8
2 1/2	3.6	1 13/16	1/2-20	3 1/2	1/2	8	2	1/2	2.55	6 1/4	3 1/2
3 1/4	4.6	2 5/16	5/8-18	4 1/2	3/4	10	2 3/8	5/8	3.25	7 1/4	4
4	5.4	2 5/16	5/8-18	5	3/4	12	2 1/2	5/8	3.82	7 3/4	4 3/8

T, TB, TD Mount Double Rod End – Rod Dimensions

Bore Ø	Rod No.	MM Rod Ø	Rod Extensions and Pilot Dimensions A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	W	WH
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	5/8	3/4
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1	1 1/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1	1 1/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 1/4	1 3/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	7/8	15/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 1/4	1 5/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1	15/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	1 3/8	1 11/16

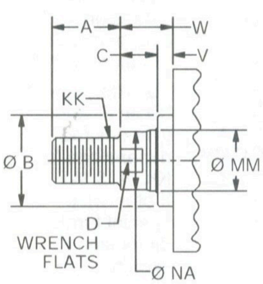
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke ZL	Add 2X Stroke ZM
1 1/2	1	5/8	1/2-20	7/16-20	1 13/16	6 11/16	6 7/8
	2	1	7/8-14	3/4-16	2 3/16	7 1/16	7 5/8
2	1	1	7/8-14	3/4-16	2 1/8	7 3/8	7 5/8
	2	1 3/8	1 1/4-12	1-14	2 3/8	7 5/8	8 1/8
2 1/2	1	1	7/8-14	3/4-16	2 1/8	7 1/2	7 3/4
	2	1 3/4	1 1/2-12	1 1/4-12	2 5/8	8	8 3/4

Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke ZL	Add 2X Stroke ZM
3 1/4	1	1 3/8	1 1/4-12	1-14	2 1/2	8 3/4	9
	2	2	1 3/4-12	1 1/2-12	2 7/8	9 1/8	9 3/4
4	1	1 3/4	1 1/2-12	1 1/4-12	2 11/16	9 3/8	9 3/4
	2	2 1/2	2 1/4-12	1 7/8-12	3 1/16	9 3/4	10 1/2

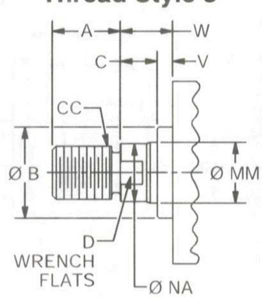
*Style T – no tie rods extended and Style TB – tie rods extended one end can be dimensioned from Style TD shown.

Rod End Dimensions

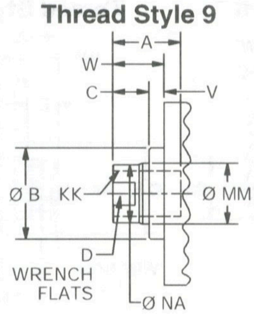
Thread Style 4



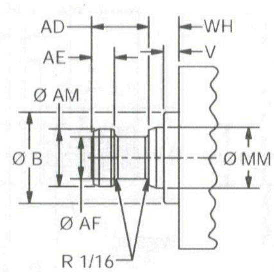
Thread Style 8



Thread Style 9



Style 55



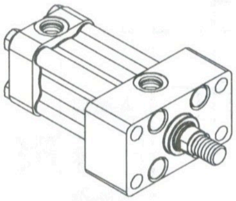
"Special" Thread Style 3

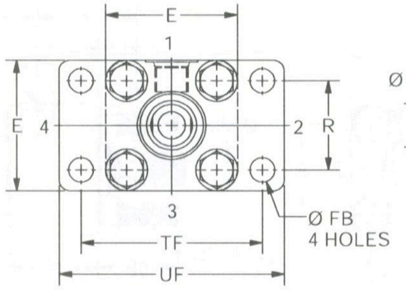
Special thread, extension, rod eye, blank, etc. are also available.

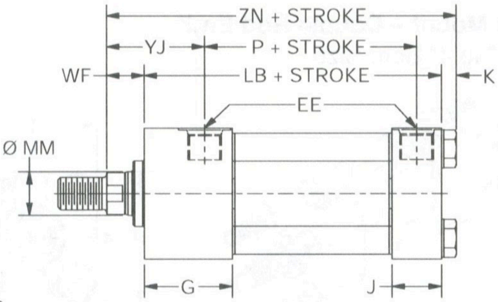
To order, specify "Style 3" and give desired dimensions for KK, A, & W. If otherwise special furnish dimensional sketch.

JJ Mount – Single Rod End

1½" to 4" Bore Size







JJ Mount Single Rod End – Envelope and Mounting Dimensions

Bore Ø	E	EE NPTF	EE SAE	FB	G	J	K Max	R	TF	UF	Add Stroke LB	Add Stroke P
1 1/2	2 1/2	1/2	8	7/16	1 3/4	1 1/8	7/16	1.63	3 7/16	4 1/4	5	3 1/4
2	3	1/2	8	9/16	2	1 1/8	1/2	2.05	4 1/8	5 1/8	5 1/4	3 5/16
2 1/2	3 1/2	1/2	8	9/16	2	1 1/8	1/2	2.55	4 5/8	5 5/8	5 3/8	3 7/16
3 1/4	4 1/2	3/4	10	11/16	2 3/8	1 3/8	5/8	3.25	5 7/8	7 1/8	6 1/4	3 15/16
4	5	3/4	12	11/16	2 1/2	1 3/8	5/8	3.82	6 3/8	7 5/8	6 5/8	4 1/4

JJ Mount Single Rod End – Rod Dimensions

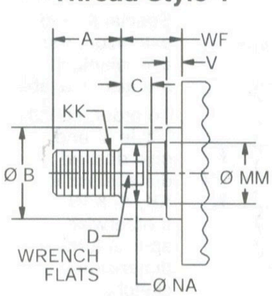
Bore Ø	Rod No.	MM Rod Ø	Rod Extensions and Pilot Dimensions A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	WF	WK
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	1	1 1/8
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1 3/8	1 7/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 3/8	1 7/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1 5/8	1 11/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 3/8	1 7/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 7/8	1 13/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	1 5/8	1 11/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	2	2 1/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1 7/8	1 13/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	2 1/4	2 9/16

Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	YJ	Add Stroke ZN
1 1/2	1	5/8	1/2-20	7/16-20	2 3/16	6 7/16
	2	1	7/8-14	3/4-16	2 9/16	6 13/16
2	1	1	7/8-14	3/4-16	2 3/4	7 1/8
	2	1 3/8	1 1/4-12	1-14	3	7 3/8
2 1/2	1	1	7/8-14	3/4-16	2 3/4	7 1/4
	2	1 3/4	1 1/2-12	1 1/4-12	3 1/4	7 3/4

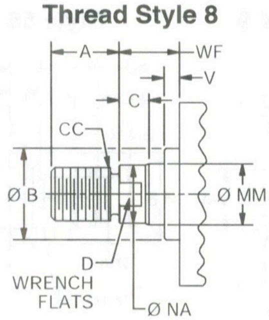
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	YJ	Add Stroke ZN
3 1/4	1	1 3/8	1 1/4-12	1-14	3 1/4	8 1/2
	2	2	1 3/4-12	1 1/2-12	3 5/8	8 7/8
4	1	1 3/4	1 1/2-12	1 1/4-12	3 9/16	9 1/8
	2	2 1/2	2 1/4-12	1 7/8-12	3 15/16	9 1/2

Rod End Dimensions

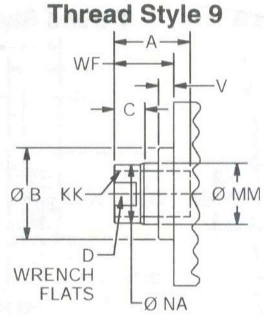
Thread Style 4



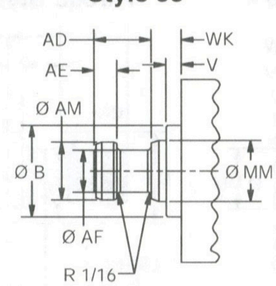
Thread Style 8



Thread Style 9



Style 55



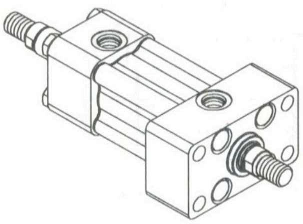
"Special" Thread Style 3

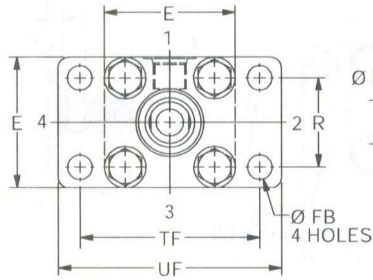
Special thread, extension, rod eye, blank, etc. are also available.

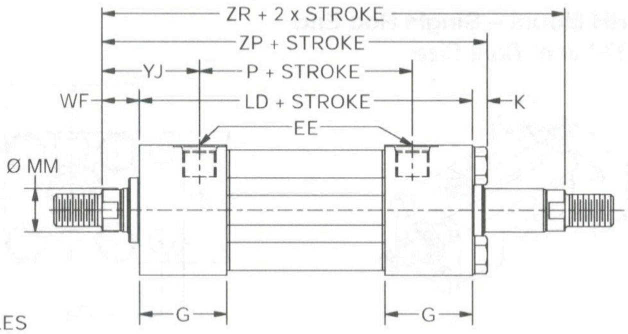
To order, specify "Style 3" and give desired dimensions for KK, A, & WF. If otherwise special furnish dimensional sketch.

JJ Mount – Double Rod End

1½" to 4" Bore Size







JJ Mount Double Rod End – Envelope and Mounting Dimensions

Bore Ø	E	EE NPTF	EE SAE	FB	G	K Max	R	TF	UF	Add Stroke LD	Add Stroke P
1 1/2	2 1/2	1/2	8	7/16	1 3/4	7/16	1.63	3 7/16	4 1/4	5 5/8	3 1/4
2	3	1/2	8	9/16	2	1/2	2.05	4 1/8	5 1/8	6 1/8	3 3/8
2 1/2	3 1/2	1/2	8	9/16	2	1/2	2.55	4 5/8	5 5/8	6 1/4	3 1/2
3 1/4	4 1/2	3/4	10	11/16	2 3/8	5/8	3.25	5 7/8	7 1/8	7 1/4	4
4	5	3/4	12	11/16	2 1/2	5/8	3.82	6 3/8	7 5/8	7 3/4	4 3/8

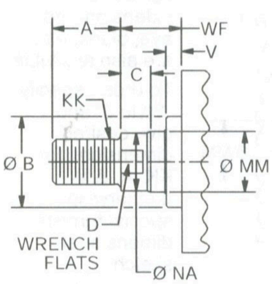
JJ Mount Double Rod End – Rod Dimensions

Bore Ø	Rod No.	MM Rod Ø	Rod Extensions and Pilot Dimensions A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	WF	WK
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	1	1 1/8
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1 3/8	1 7/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 3/8	1 7/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1 5/8	1 11/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 3/8	1 7/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 7/8	1 13/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	1 5/8	1 11/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	2	2 1/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1 7/8	1 13/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	2 1/4	2 9/16

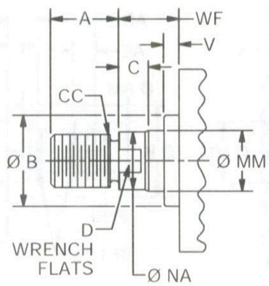
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	YJ	Add Stroke ZP	Add 2X Stroke ZR
1 1/2	1	5/8	1/2-20	7/16-20	2 3/16	7 1/16	7 1/4
	2	1	7/8-14	3/4-16	2 9/16	7 7/16	8
2	1	1	7/8-14	3/4-16	2 3/4	8	8 1/4
	2	1 3/8	1 1/4-12	1-14	3	8 1/4	8 3/4
2 1/2	1	1	7/8-14	3/4-16	2 3/4	8 1/8	8 3/8
	2	1 3/4	1 1/2-12	1 1/4-12	3 1/4	8 5/8	9 3/8
3 1/4	1	1 3/8	1 1/4-12	1-14	3 1/4	9 1/2	9 3/4
	2	2	1 3/4-12	1 1/2-12	3 5/8	9 7/8	10 1/2
4	1	1 3/4	1 1/2-12	1 1/4-12	3 9/16	10 1/4	10 5/8
	2	2 1/2	2 1/4-12	1 7/8-12	3 15/16	10 5/8	11 3/8

Rod End Dimensions

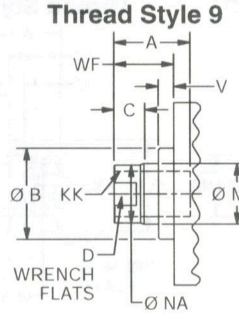
Thread Style 4



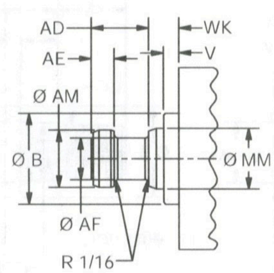
Thread Style 8



Thread Style 9



Style 55



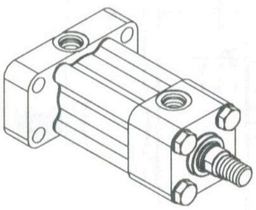
"Special" Thread Style 3

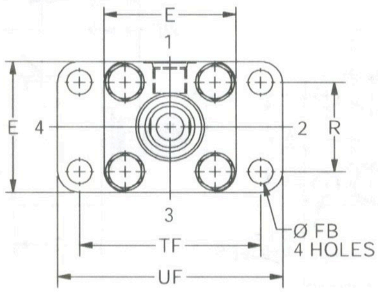
Special thread, extension, rod eye, blank, etc. are also available.

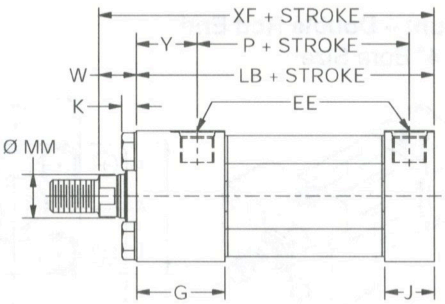
To order, specify "Style 3" and give desired dimensions for KK, A, & WF. If otherwise special furnish dimensional sketch.

HH Mount – Single Rod End

1½" to 4" Bore Size







HH Mount Single Rod End – Envelope and Mounting Dimensions

Bore	E	EE NPTF	EE SAE	FB	G	J	Max K	R	TF	UF	Add Stroke LB	Add Stroke P
1 1/2	2 1/2	1/2	8	7/16	1 3/4	1 1/8	7/16	1.63	3 7/16	4 1/4	5	3 1/4
2	3	1/2	8	9/16	2	1 1/8	1/2	2.05	4 1/8	5 1/8	5 1/4	3 5/16
2 1/2	3 1/2	1/2	8	9/16	2	1 1/8	1/2	2.55	4 5/8	5 5/8	5 3/8	3 7/16
3 1/4	4 1/2	3/4	10	11/16	2 3/8	1 3/8	5/8	3.25	5 7/8	7 1/8	6 1/4	3 15/16
4	5	3/4	12	11/16	2 1/2	1 3/8	5/8	3.82	6 3/8	7 5/8	6 5/8	4 1/4

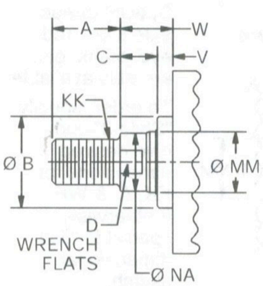
HH Mount Single Rod End – Rod Dimensions

Bore Ø	Rod No.	MM Rod Ø	Rod Extensions and Pilot Dimensions A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	W	WH
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	5/8	3/4
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1	1 1/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1	1 1/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 1/4	1 3/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	7/8	15/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 1/4	1 5/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1	15/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	1 3/8	1 11/16

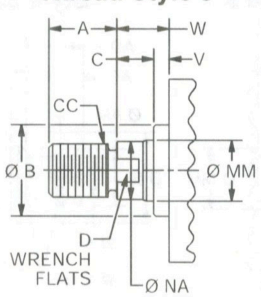
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke XF
1 1/2	1	5/8	1/2-20	7/16-20	1 13/16	5 5/8
	2	1	7/8-14	3/4-16	2 3/16	6
2	1	1	7/8-14	3/4-16	2 1/8	6
	2	1 3/8	1 1/4-12	1-14	2 3/8	6 1/4
2 1/2	1	1	7/8-14	3/4-16	2 1/8	6 1/8
	2	1 3/4	1 1/2-12	1 1/4-12	2 5/8	6 5/8
3 1/4	1	1 3/8	1 1/4-12	1-14	2 1/2	7 1/8
	2	2	1 3/4-12	1 1/2-12	2 7/8	7 1/2
4	1	1 3/4	1 1/2-12	1 1/4-12	2 11/16	7 5/8
	2	2 1/2	2 1/4-12	1 7/8-12	3 1/16	8

Rod End Dimensions

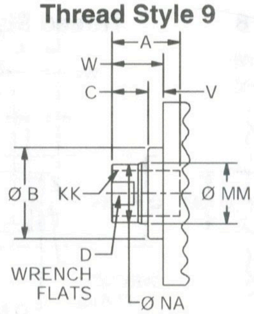
Thread Style 4



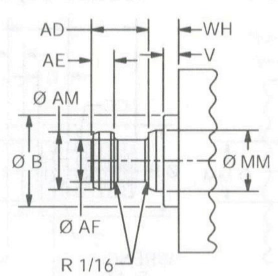
Thread Style 8



Thread Style 9



Style 55



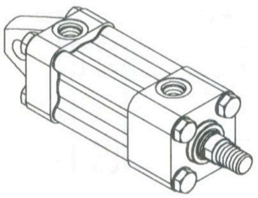
"Special" Thread Style 3

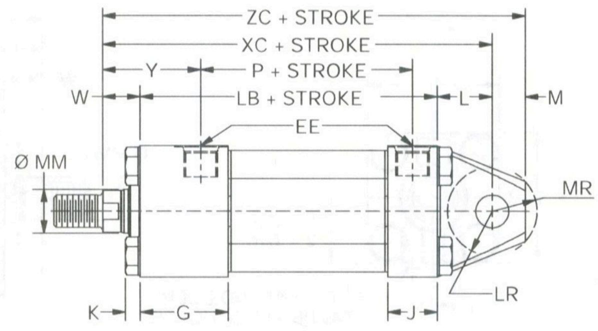
Special thread, extension, rod eye, blank, etc. are also available.

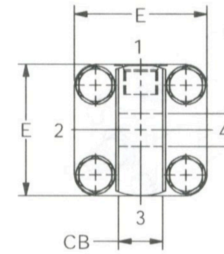
To order, specify "Style 3" and give desired dimensions for KK, A, & W. If otherwise special furnish dimensional sketch.

BE Mount – Single Rod End

1½" to 4" Bore Size







BE Mount Single Rod End – Envelope and Mounting Dimensions

Bore Ø	CB	CD +.002 -.000	E	EE NPTF	EE SAE	G	J	K Max	L	LR	M	MR	Add Stroke LB	Add Stroke P
1 1/2	3/4	0.502	2 1/2	1/2	8	1 3/4	1 1/8	7/16	3/4	9/16	1/2	11/16	5	3 1/4
2	1	0.752	3	1/2	8	2	1 1/8	1/2	1 1/4	1	3/4	1 1/16	5 1/4	3 5/16
2 1/2	1 1/4	0.752	3 1/2	1/2	8	2	1 1/8	1/2	1 1/4	15/16	3/4	1 1/16	5 3/8	3 7/16
3 1/4	1 1/2	1.002	4 1/2	3/4	10	2 3/8	1 3/8	5/8	1 1/2	1 1/4	1	1 3/8	6 1/4	3 15/16
4	2	1.376	5	3/4	12	2 1/2	1 3/8	5/8	2 1/8	1 3/4	1 3/8	1 3/4	6 5/8	4 1/4

BE Mount Single Rod End – Rod Dimensions

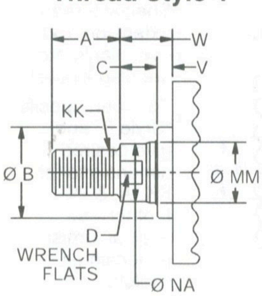
Bore Ø	Rod No.	MM Rod Ø	Rod Extensions and Pilot Dimensions A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	W	WH
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	5/8	3/4
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1	1 1/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1	1 1/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 1/4	1 3/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	7/8	15/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 1/4	1 5/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1	15/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	1 3/8	1 11/16

Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke XC	Add Stroke ZC
1 1/2	1	5/8	1/2-20	7/16-20	1 13/16	6 3/8	6 7/8
	2	1	7/8-14	3/4-16	2 3/16	6 3/4	7 1/4
2	1	1	7/8-14	3/4-16	2 1/8	7 1/4	8
	2	1 3/8	1 1/4-12	1-14	2 3/8	7 1/2	8 1/4
2 1/2	1	1	7/8-14	3/4-16	2 1/8	7 3/8	8 1/8
	2	1 3/4	1 1/2-12	1 1/4-12	2 5/8	7 7/8	8 5/8

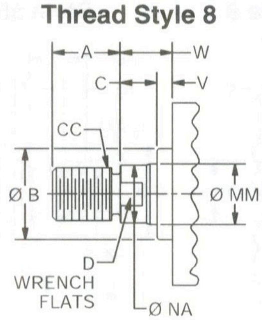
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke XC	Add Stroke ZC
3 1/4	1	1 3/8	1 1/4-12	1-14	2 1/2	8 5/8	9 5/8
	2	2	1 3/4-12	1 1/2-12	2 7/8	9	10
4	1	1 3/4	1 1/2-12	1 1/4-12	2 11/16	9 3/4	11 1/8
	2	2 1/2	2 1/4-12	1 7/8-12	3 1/16	10 1/8	11 1/2

Rod End Dimensions

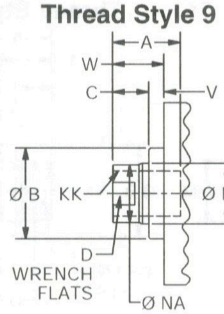
Thread Style 4



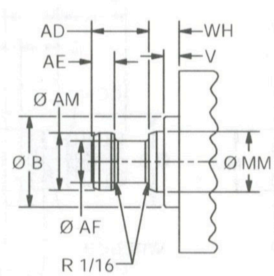
Thread Style 8



Thread Style 9



Style 55



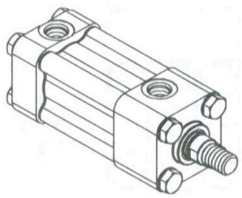
"Special" Thread Style 3

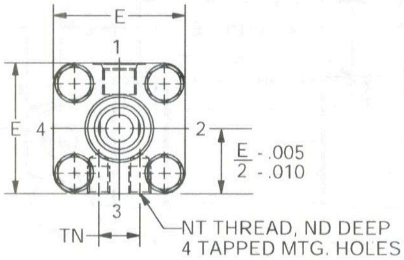
Special thread, extension, rod eye, blank, etc. are also available.

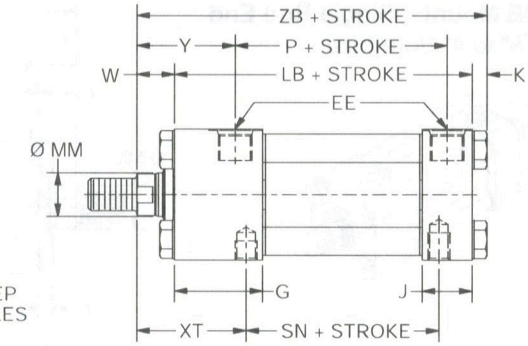
To order, specify "Style 3" and give desired dimensions for KK, A, & W. If otherwise special furnish dimensional sketch.

F Mount – Single Rod End

1½" to 4" Bore Size







F Mount Single Rod End – Envelope and Mounting Dimensions

Bore Ø	E	EE NPTF	EE SAE	G	J	K Max	NT	TN	Add Stroke LB	Add Stroke P	Add Stroke SN
1 1/2	2 1/2	1/2	8	1 3/4	1 1/8	7/16	3/8-16	3/4	5	3 1/4	2 7/8
2	3	1/2	8	2	1 1/8	1/2	1/2-13	15/16	5 1/4	3 5/16	2 7/8
2 1/2	3 1/2	1/2	8	2	1 1/8	1/2	5/8-11	1 5/16	5 3/8	3 7/16	3
3 1/4	4 1/2	3/4	10	2 3/8	1 3/8	5/8	3/4-10	1 1/2	6 1/4	3 15/16	3 1/2
4	5	3/4	12	2 1/2	1 3/8	5/8	1-8	2 1/16	6 5/8	4 1/4	4

F Mount Single Rod End – Rod Dimensions

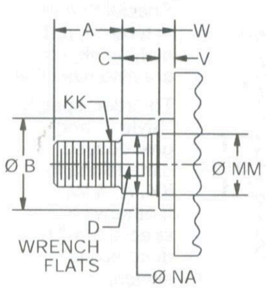
Bore Ø	Rod No.	MM Rod Ø	A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	W	WH
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	5/8	3/4
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1	1 1/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1	1 1/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 1/4	1 3/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	7/8	15/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 1/4	1 5/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1	15/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	1 3/8	1 11/16

Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	ND	XT	Y	Add Stroke ZB
1 1/2	1	5/8	1/2-20	7/16-20	3/8	2	1 13/16	6 1/16
	2	1	7/8-14	3/4-16	3/8	2 3/8	2 3/16	6 7/16
2	1	1	7/8-14	3/4-16	7/16	2 3/8	2 1/8	6 1/2
	2	1 3/8	1 1/4-12	1-14	7/16	2 5/8	2 3/8	6 3/4
2 1/2	1	1	7/8-14	3/4-16	1/2	2 3/8	2 1/8	6 5/8
	2	1 3/4	1 1/2-12	1 1/4-12	1/2	2 7/8	2 5/8	7 1/8

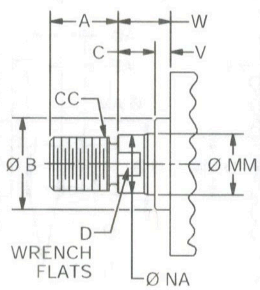
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	ND	XT	Y	Add Stroke ZB
3 1/4	1	1 3/8	1 1/4-12	1-14	11/16	2 3/4	2 1/2	7 3/4
	2	2	1 3/4-12	1 1/2-12	11/16	3 1/8	2 7/8	8 1/8
4	1	1 3/4	1 1/2-12	1 1/4-12	11/16	2 7/8	2 11/16	8 1/4
	2	2 1/2	2 1/4-12	1 7/8-12	11/16	3 3/4	3 1/16	8 5/8

Rod End Dimensions

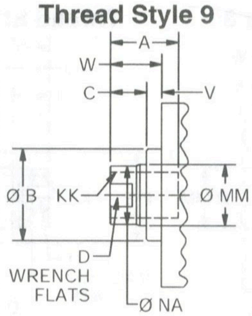
Thread Style 4



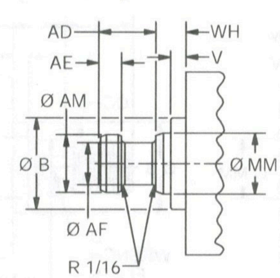
Thread Style 8



Thread Style 9



Style 55



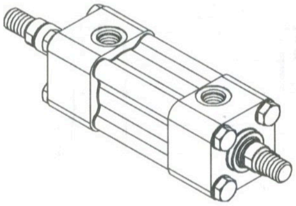
"Special" Thread Style 3

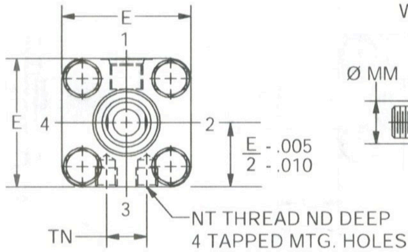
Special thread, extension, rod eye, blank, etc. are also available.

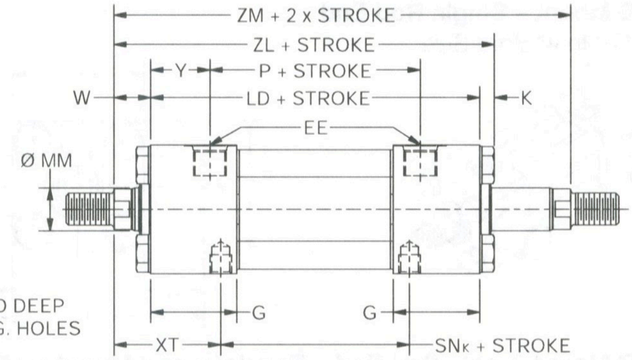
To order, specify "Style 3" and give desired dimensions for KK, A, & W. If otherwise special furnish dimensional sketch.

F Mount – Double Rod End

1½" to 4" Bore Size







F Mount Double Rod End – Envelope and Mounting Dimensions

Bore Ø	E	EE NPTF	EE SAE	G	K Max	NT	TN	Add Stroke LD	Add Stroke P	Add Stroke SNK
1 1/2	2 1/2	1/2	8	1 3/4	7/16	3/8-16	3/4	5 5/8	3 1/4	2 7/8
2	3	1/2	8	2	1/2	1/2-13	15/16	6 1/8	3 3/8	2 7/8
2 1/2	3 1/2	1/2	8	2	1/2	5/8-11	1 5/16	6 1/4	3 1/2	3
3 1/4	4 1/2	3/4	10	2 3/8	5/8	3/4-10	1 1/2	7 1/4	4	3 1/2
4	5	3/4	12	2 1/2	5/8	1-8	2 1/16	7 3/4	4 3/8	4

F Mount Double Rod End – Rod Dimensions

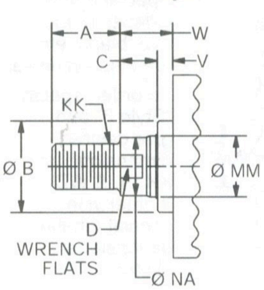
Bore Ø	Rod No.	MM Rod Ø	A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	W	WH	ND	XT
			Rod Extensions and Pilot Dimensions													
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	5/8	3/4	3/8	2
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1	1 1/16	3/8	2 3/8
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16	7/16	2 3/8
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1	1 1/16	7/16	2 5/8
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	3/4	13/16	1/2	2 3/8
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 1/4	1 3/16	1/2	2 7/8
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	7/8	15/16	11/16	2 3/4
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 1/4	1 5/16	11/16	3 1/8
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1	15/16	11/16	2 7/8
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	1 3/8	1 11/16	11/16	3 3/4

Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke ZL	Add 2X Stroke ZM
1 1/2	1	5/8	1/2-20	7/16-20	1 13/16	6 11/16	6 7/8
	2	1	7/8-14	3/4-16	2 3/16	7 1/16	7 5/8
2	1	1	7/8-14	3/4-16	2 1/8	7 3/8	7 5/8
	2	1 3/8	1 1/4-12	1-14	2 3/8	7 5/8	8 1/8
2 1/2	1	1	7/8-14	3/4-16	2 1/8	7 1/2	7 3/4
	2	1 3/4	1 1/2-12	1 1/4-12	2 5/8	8	8 3/4

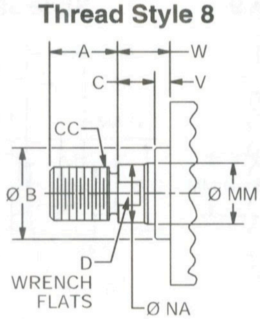
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	Y	Add Stroke ZL	Add 2X Stroke ZM
3 1/4	1	1 3/8	1 1/4-12	1-14	2 1/2	8 3/4	9
	2	2	1 3/4-12	1 1/2-12	2 7/8	9 1/8	9 3/4
4	1	1 3/4	1 1/2-12	1 1/4-12	2 11/16	9 3/8	9 3/4
	2	2 1/2	2 1/4-12	1 7/8-12	3 1/16	9 3/4	10 1/2

Rod End Dimensions

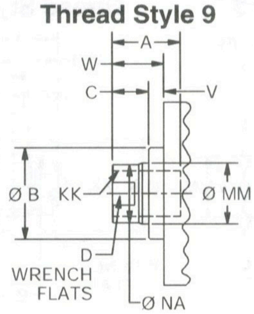
Thread Style 4



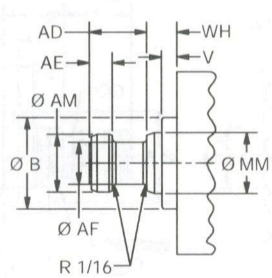
Thread Style 8



Thread Style 9



Style 55



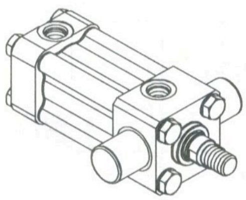
"Special" Thread Style 3

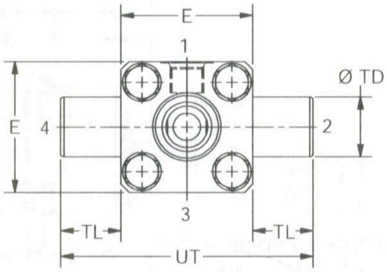
Special thread, extension, rod eye, blank, etc. are also available.

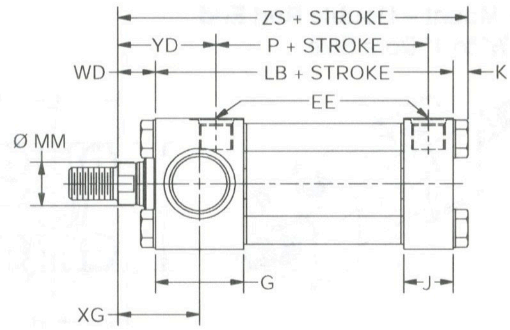
To order, specify "Style 3" and give desired dimensions for KK, A, & W. If otherwise special furnish dimensional sketch.

D Mount – Single Rod End

1½" to 4" Bore Size







D Mount Single Rod End – Envelope and Mounting Dimensions

Bore Ø	E	EE NPTF	EE SAE	G	J	K Max	TD +.000 -.001	TL	UT	Add Stroke LB	Add Stroke P
1 1/2	2 1/2	1/2	8	1 3/4	1 1/8	7/16	1.000	1	4 1/2	5	3 1/4
2	3	1/2	8	2	1 1/8	1/2	1.375	1 3/8	5 3/4	5 1/4	3 5/16
2 1/2	3 1/2	1/2	8	2	1 1/8	1/2	1.375	1 3/8	6 1/4	5 3/8	3 7/16
3 1/4	4 1/2	3/4	10	2 3/8	1 3/8	5/8	1.750	1 3/4	8	6 1/4	3 15/16
4	5	3/4	12	2 1/2	1 3/8	5/8	1.750	1 3/4	8 1/2	6 5/8	4 1/4

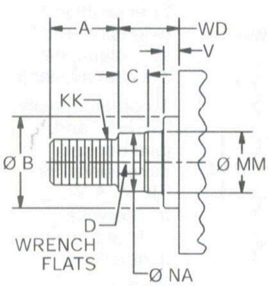
D Mount Single Rod End – Rod Dimensions

Bore Ø	Rod No.	MM Rod Ø	A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	WD	WJ
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	1	1 1/8
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1 3/8	1 7/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 1/4	1 5/16
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1 1/2	1 9/16
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 1/4	1 5/16
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 3/4	1 11/16
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	1 7/16	1 1/2
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 13/16	1 7/8
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1 5/8	1 9/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	2	2 5/16

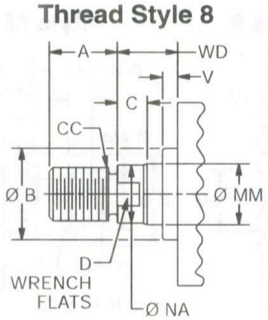
Bore Ø	Rod No.	MM Rod Ø	Thread Style 8 CC	Thread Style 4 & 9 KK	XG	YD	Add Stroke ZS
1 1/2	1	5/8	1/2-20	7/16-20	1 7/8	2 3/16	6 7/16
	2	1	7/8-14	3/4-16	2 1/4	2 9/16	6 13/16
2	1	1	7/8-14	3/4-16	2 1/4	2 5/8	7
	2	1 3/8	1 1/4-12	1-14	2 1/2	2 7/8	7 1/4
2 1/2	1	1	7/8-14	3/4-16	2 1/4	2 5/8	7 1/8
	2	1 3/4	1 1/2-12	1 1/4-12	2 3/4	3 1/8	7 5/8
3 1/4	1	1 3/8	1 1/4-12	1-14	2 5/8	3 1/16	8 5/16
	2	2	1 3/4-12	1 1/2-12	3	3 7/16	8 11/16
4	1	1 3/4	1 1/2-12	1 1/4-12	2 7/8	3 5/16	8 7/8
	2	2 1/2	2 1/4-12	1 7/8-12	3 1/4	3 11/16	9 1/4

Rod End Dimensions

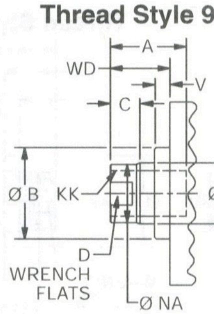
Thread Style 4



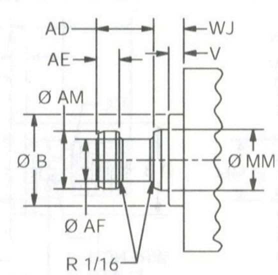
Thread Style 8



Thread Style 9



Style 55



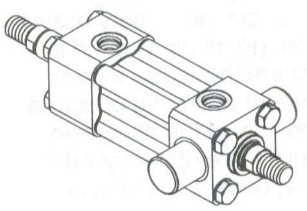
"Special" Thread Style 3

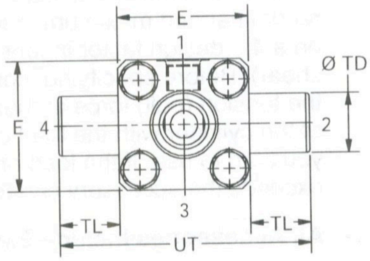
Special thread, extension, rod eye, blank, etc. are also available.

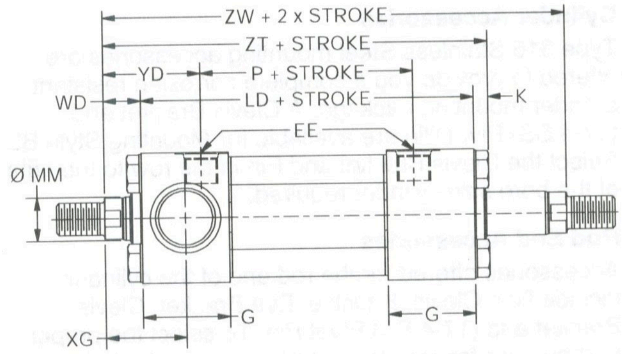
To order, specify "Style 3" and give desired dimensions for KK, A, & WD. If otherwise special furnish dimensional sketch.

D Mount – Double Rod End

1½" to 4" Bore Size







D Mount Double Rod End – Envelope and Mounting Dimensions

Bore Ø	E	EE		G	Max K	TD +.000 -.001	TL	UT	Add Stroke	
		NPTF	SAE						LD	P
1 1/2	2 1/2	1/2	8	1 3/4	7/16	1.000	1	4 1/2	5 5/8	3 1/4
2	3	1/2	8	2	1/2	1.375	1 3/8	5 3/4	6 1/8	3 3/8
2 1/2	3 1/2	1/2	8	2	1/2	1.375	1 3/8	6 1/4	6 1/4	3 1/2
3 1/4	4 1/2	3/4	10	2 3/8	5/8	1.750	1 3/4	8	7 1/4	4
4	5	3/4	12	2 1/2	5/8	1.750	1 3/4	8 1/2	7 3/4	4 3/8

D Mount Double Rod End – Rod Dimensions

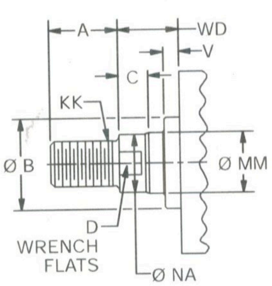
Bore Ø	Rod No.	MM Rod Ø	Rod Extensions and Pilot Dimensions												XG	YD
			A	AD	AE	AF	AM	B +.000 -.002	C	D	NA	V	WD	WJ		
1 1/2	1	5/8	3/4	5/8	1/4	3/8	0.57	1.124	3/8	1/2	9/16	1/4	1	1 1/8	1 7/8	2 3/16
	2	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/2	1 3/8	1 7/16	2 1/4	2 9/16
2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 1/4	1 5/16	2 1/4	2 5/8
	2	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	3/8	1 1/2	1 9/16	2 1/2	2 7/8
2 1/2	1	1	1 1/8	15/16	3/8	11/16	0.95	1.499	1/2	7/8	15/16	1/4	1 1/4	1 5/16	2 1/4	2 5/8
	2	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/2	1 3/4	1 11/16	2 3/4	3 1/8
3 1/4	1	1 3/8	1 5/8	1 1/16	3/8	7/8	1.32	1.999	5/8	1 1/8	1 5/16	1/4	1 7/16	1 1/2	2 5/8	3 1/16
	2	2	2 1/4	1 11/16	5/8	1 3/8	1.95	2.624	7/8	1 11/16	1 15/16	3/8	1 13/16	1 7/8	3	3 7/16
4	1	1 3/4	2	1 5/16	1/2	1 1/8	1.70	2.374	3/4	1 1/2	1 11/16	1/4	1 5/8	1 9/16	2 7/8	3 5/16
	2	2 1/2	3	1 15/16	3/4	1 3/4	2.45	3.124	1	2 1/16	2 3/8	3/8	2	2 5/16	3 1/4	3 11/16

Bore Ø	Rod No.	MM Rod Ø	Thread		Add Stroke	Add 2X Stroke
			Style 8 CC	Style 4 & 9 KK	ZT	ZW
1 1/2	1	5/8	1/2-20	7/16-20	7 1/16	7 1/4
	2	1	7/8-14	3/4-16	7 7/16	8
2	1	1	7/8-14	3/4-16	7 7/8	8 1/8
	2	1 3/8	1 1/4-12	1-14	8 1/8	8 5/8
2 1/2	1	1	7/8-14	3/4-16	8	8 1/4
	2	1 3/4	1 1/2-12	1 1/4-12	8 1/2	9 1/4

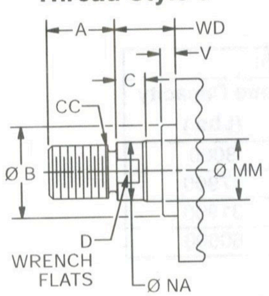
Bore Ø	Rod No.	MM Rod Ø	Thread		Add Stroke	Add 2X Stroke
			Style 8 CC	Style 4 & 9 KK	ZT	ZW
3 1/4	1	1 3/8	1 1/4-12	1-14	9 5/16	9 9/16
	2	2	1 3/4-12	1 1/2-12	9 11/16	10 5/16
4	1	1 3/4	1 1/2-12	1 1/4-12	10	10 3/8
	2	2 1/2	2 1/4-12	1 7/8-12	10 3/8	11 1/8

Rod End Dimensions

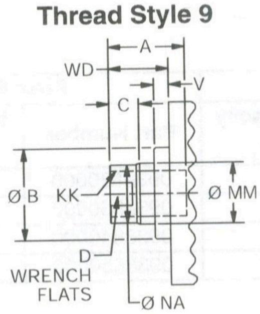
Thread Style 4



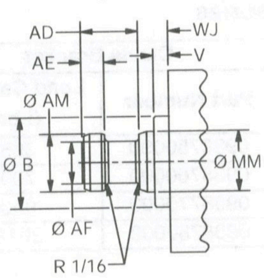
Thread Style 8



Thread Style 9



Style 55



"Special" Thread Style 3

Special thread, extension, rod eye, blank, etc. are also available.

To order, specify "Style 3" and give desired dimensions for KK, A, & WD. If otherwise special furnish dimensional sketch.

Parker Hannifin Corporation
Cylinder Division
Des Plaines, Illinois USA

Cylinder Accessories

Type 316 Stainless Steel mounting accessories are offered to provide you a complete corrosion resistant cylinder mounting package. A Clevis Bracket and (17-4 SS) Pivot Pin are available for Mounting Style BE. Select the Clevis Bracket and Pin in the row to the right of the bore size cylinder required.

Rod End Accessories

Accessories offered for the rod end of the cylinder include Rod Clevis, Knuckle, Eye Bracket, Clevis Bracket and (17-4 SS) Pivot Pin. To select the proper part number for any desired accessory, refer to the table below or on the opposite page and look in the row to the right of the rod thread in the first column. For economical accessory selection, it is recommended that rod end style 4 be specified on your cylinder order.

Accessory Load Capacity

The various accessories have been load rated for your convenience. The load capacity in lbs. is the recommended maximum load for that accessory based on a 4:1 design factor in tension. (Pivot pin is rated in shear). Before specifying, compare the actual load or the tension (pull) force at maximum operating pressure of the cylinder with the load capacity of the accessory you plan to use. If the load or pull force of the cylinder exceeds the accessory capacity, consult the factory.

All Accessories Include Electropolishing

Rod End Accessories

Thread Size	Rod Clevis		Eye Bracket		Pivot Pin	
	Part Number	Load Capacity (Lbs.)	Part Number	Load Capacity (Lbs.)	Part Number	Load Capacity (Lbs.)
7/16-20	0938480000	2125	0938680000	2050	0938820000	8000
1/2-20	0938490000	2450	0938680000	2050	0938820000	8000
3/4-16	0938500000	5600	0938690000	5800	0938830000	17900
7/8-14	0938510000	9400	0938700000	12200	0938840000	31900
1-14	0938520000	9750	0938700000	12200	0938840000	31900
1 1/4-12	0938530000	22300	0938710000	12720	0938850000	60500
1 1/2-12	0938540000	30400	0938720000	32900	0938860000	98000
1 3/4-12	0938550000	43700	0938730000	46600	0938870000	127700
1 7/8-12	0938560000	43700	0938730000	46600	0938870000	127700
2 1/4-12	0938570000	65400	0938740000	62800	0938880000	199600

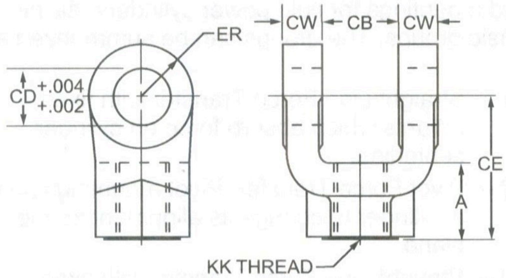
Rod End Accessories

Thread Size	Knuckle		Clevis Bracket		Pivot Pin	
	Part Number	Load Capacity (Lbs.)	Part Number	Load Capacity (Lbs.)	Part Number	Load Capacity (Lbs.)
7/16-20	0938580000	2700	0938750000	3650	0938820000	8000
1/2-20	0938590000	3100	0938750000	3650	0938820000	8000
3/4-16	0938600000	7200	0938760000	7000	0938830000	17900
7/8-14	0938610000	7800	0938770000	9600	0938840000	31900
1-14	0938620000	13000	0938770000	9600	0938840000	31900
1 1/4-12	0938630000	20000	0938780000	20120	0938850000	60500
1 1/2-12	0938640000	30000	0938790000	20300	0938860000	98000
1 3/4-12	0938650000	35500	0938800000	19700	0938870000	127700
1 7/8-12	0938660000	50000	0938800000	19700	0938870000	127700
2 1/4-12	0938670000	65000	0938810000	20900	0938880000	199600

Cylinder Accessories

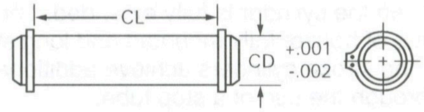
Bore Ø	Clevis Bracket		Pivot Pin	
	Part Number	Load Capacity (Lbs.)	Part Number	Load Capacity (Lbs.)
1 1/2	0938750000	3650	0938820000	8000
2, 2 1/2	0938760000	7000	0938830000	17900
3 1/4	0938770000	9600	0938840000	31900
4	0938780000	20120	0938850000	60500

Rod Clevis Dimensions



Part Number	A	CB	CD	CE	CW	ER	KK
0938480000	3/4	3/4	1/2	1 1/2	1/2	1/2	7/16-20
0938490000	3/4	3/4	1/2	1 1/2	1/2	1/2	1/2-20
0938500000	1 1/8	1 1/4	3/4	2 3/8	5/8	3/4	3/4-16
0938510000	1 5/8	1 1/2	1	3 1/8	3/4	1	7/8-14
0938520000	1 5/8	1 1/2	1	3 1/8	3/4	1	1-14
0938530000	1 7/8	2	1 3/8	4 1/8	1	1 3/8	1 1/4-12
0938540000	2 1/4	2 1/2	1 3/4	4 1/2	1 1/4	1 3/4	1 1/2-12
0938550000	3	2 1/2	2	5 1/2	1 1/4	2	1 3/4-12
0938560000	3	2 1/2	2	5 1/2	1 1/4	2	1 7/8-12
0938570000	3 1/2	3	2 1/2	6 1/2	1 1/2	2 1/2	2 1/4-12

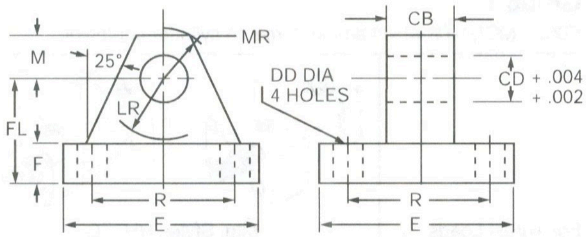
Pivot Pin Dimensions



Part Number	CD	CL
0938820000	1/2	1 7/8
0938830000	3/4	2 5/8
0938840000	1	3 1/8
0938850000	1 3/8	4 1/8
0938860000	1 3/4	5 3/16
0938870000	2	5 3/16
0938880000	2 1/2	6 3/16

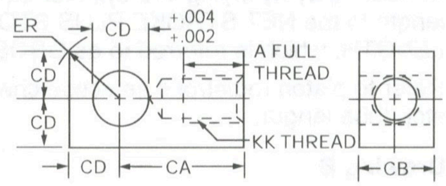
1. Pivot Pins are furnished with (2) retainer rings.
2. Pivot Pins must be ordered as a separate item if to be used with Rod Clevises or Clevis Brackets.

Eye Bracket Dimensions



Part Number	CB	CD	DD	E	F	FL	LR	M	MR	R
0968680000	3/4	1/2	13/32	2 1/2	3/8	1 1/8	3/4	1/2	9/16	1.63
0938690000	1 1/4	3/4	17/32	3 1/2	5/8	1 7/8	1 1/4	3/4	7/8	2.55
0938700000	1 1/2	1	21/32	4 1/2	7/8	2 3/8	1 1/2	1	1 1/4	3.25
0938710000	2	1 3/8	21/32	5	7/8	3	2 1/8	1 3/8	1 5/8	3.82
0938720000	2 1/2	1 3/4	29/32	6 1/2	1 1/8	3 3/8	2 1/4	1 3/4	2 1/8	4.95
0938730000	2 1/2	2	1 1/16	7 1/2	1 1/2	4	2 1/2	2	2 7/16	5.73
0938740000	3	2 1/2	1 3/16	8 1/2	1 3/4	4 3/4	3	2 1/2	3	6.58

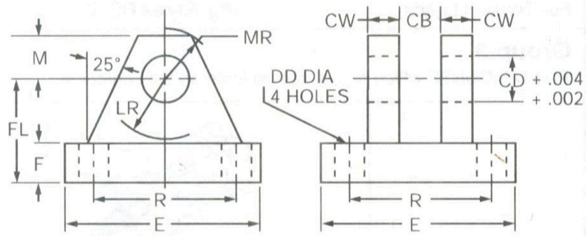
Knuckle Dimensions



Part Number	A	CA	CB
0938580000	3/4	1 1/2	3/4
0938590000	3/4	1 1/2	3/4
0938600000	1 1/8	2 1/16	1 1/4
0938610000	1 1/8	2 3/8	1 1/2
0938620000	1 5/8	2 13/16	1 1/2
0938630000	2	3 7/16	2
0938640000	2 1/4	4	2 1/2
0938650000	2 1/4	4 3/8	2 1/2
0938660000	3	5	2 1/2
0938670000	3 1/2	5 13/16	3

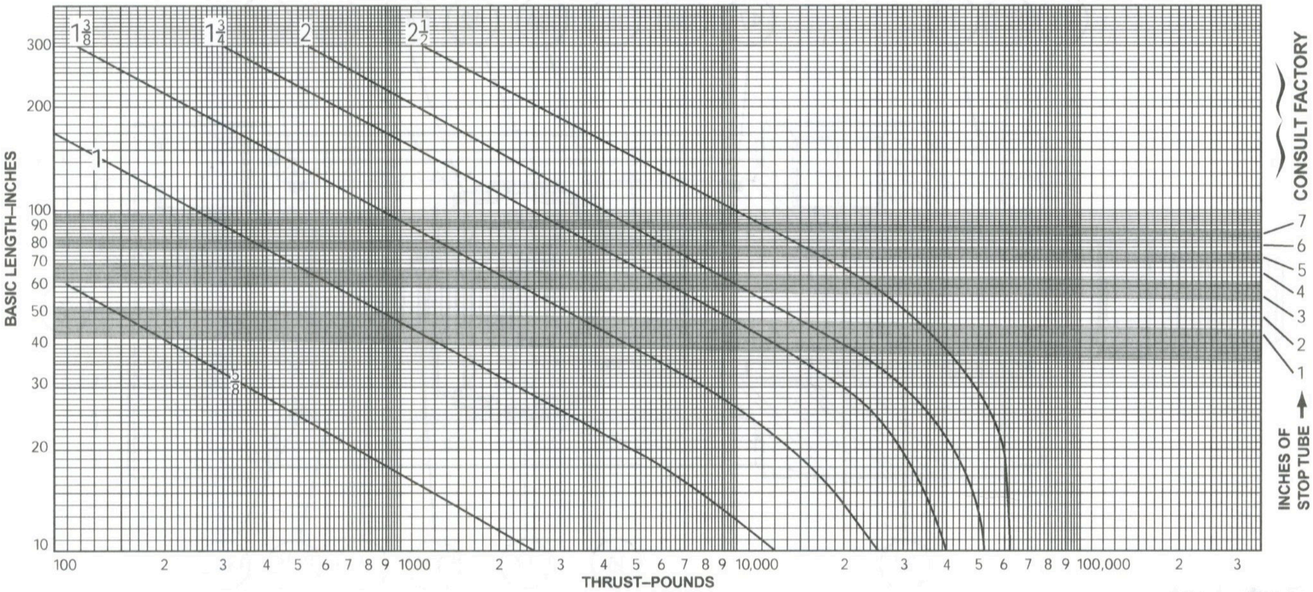
Part Number	CD	ER	KK
0938580000	1/2	23/32	7/16-20
0938590000	1/2	23/32	1/2-20
0938600000	3/4	1 1/16	3/4-16
0938610000	1	1 7/16	7/8-14
0938620000	1	1 7/16	1-14
0938630000	1 3/8	1 31/32	1 1/4-12
0938640000	1 3/4	2 1/2	1 1/2-12
0938650000	2	2 27/32	1 3/4-12
0938660000	2	2 27/32	1 7/8-12
0938670000	2 1/2	3 9/16	2 1/4-12

Clevis Bracket Dimensions



Part Number	CB	CD	CW	DD	E	F	FL	LR	M	MR	R
0938750000	3/4	1/2	1/2	13/32	3 1/2	1/2	1 1/2	3/4	1/2	5/8	2.55
0938760000	1 1/4	3/4	5/8	17/32	5	5/8	1 7/8	1 3/16	3/4	29/32	3.82
0938770000	1 1/2	1	3/4	21/32	6 1/2	3/4	2 1/4	1 1/2	1	1 1/4	4.95
0938780000	2	1 3/8	1	21/32	7 1/2	7/8	3	2	1 3/8	1 21/32	5.73
0938790000	2 1/2	1 3/4	1 1/4	29/32	9 1/2	7/8	3 5/8	2 3/4	1 3/4	2 7/32	7.50
0938800000	2 1/2	2	1 1/2	1 1/16	12 3/4	1	4 1/4	3 3/16	2 1/4	2 25/32	9.40
0938810000	3	2 1/2	1 1/2	1 3/16	12 3/4	1	4 1/2	3 1/2	2 1/2	3 1/8	9.40

Piston Rod — Stroke Selection Chart



How to Use the Chart

The selection of a piston rod for thrust (push) conditions requires the following steps:

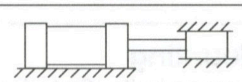
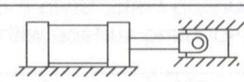
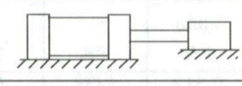
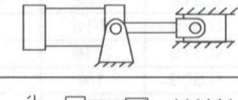
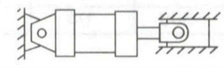
1. Determine the type of cylinder mounting style and rod end connection to be used. Then consult the chart below and find the "stroke factor" that corresponds to the conditions used.
2. Using this stroke factor, determine the "basic length" from the equation:

$$\text{Basic Length} = \text{Actual Stroke} \times \text{Stroke Factor}$$

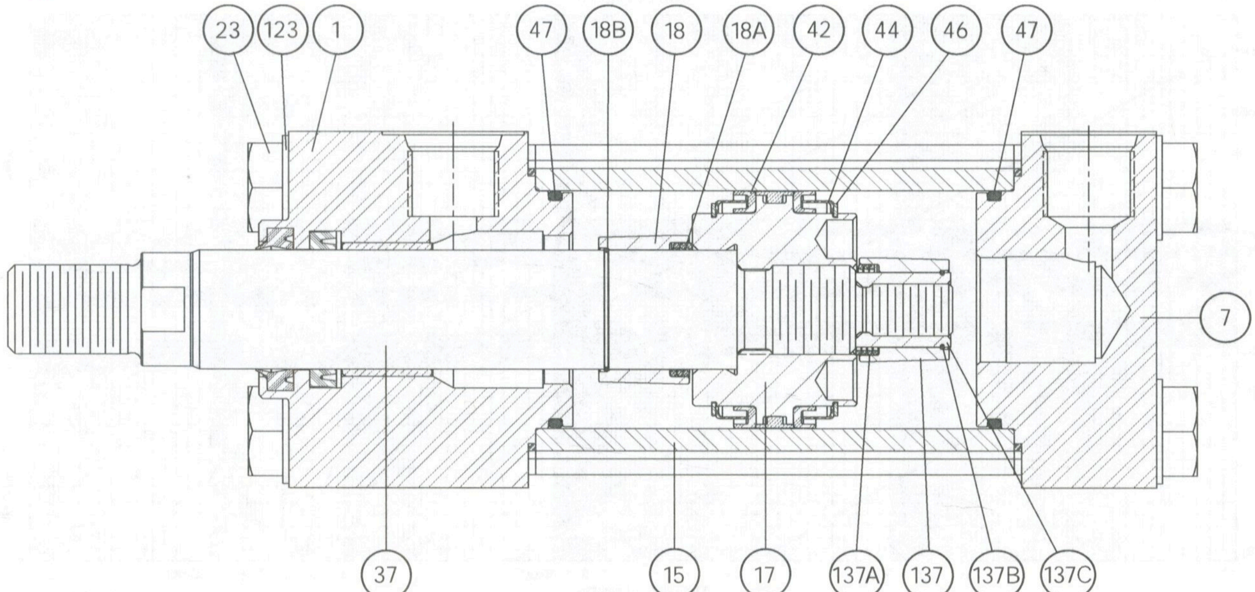
   The graph is prepared for standard rod extensions beyond the face of the head. For rod extensions greater than standard, add the increase to the stroke in arriving at the "basic length."
3. Find the load imposed for the thrust application by multiplying the full bore area of the cylinder by the system pressure.
4. Enter the graph along the values of "basic length" and "thrust" as found above and note the point of intersection:
   A) The correct piston rod size is read from the diagonally curved line labeled "Rod Diameter" next *above* the point of intersection.
   B) The required length of stop tube is read from the right of the graph by following the shaded band in which the point of intersection lies.
   C) If required length of stop tube is in the region labeled "consult factory," submit the following information for an individual analysis:
   1) Cylinder mounting style.
   2) Rod end connection and method of guiding load.
   3) Bore, required stroke, length of rod extension (Dim. "A" and "W") if greater than standard, and series of cylinder used.
   4) Mounting position of cylinder. (Note: If at an angle or vertical, specify direction of piston rod.)
   5) Operating pressure of cylinder if limited to less than standard pressure for cylinder selected.

Warning ⚠

Piston rods are not normally designed to absorb bending moments or loads which are perpendicular to the axis of piston rod motion. These additional loads can cause the piston rod end to fail. If these types of additional loads are expected to be imposed on the piston rods, their magnitude should be made known to our Engineering Department so they may be properly addressed. Additionally, cylinder users should always make sure that the piston rod is securely attached to the machine member.

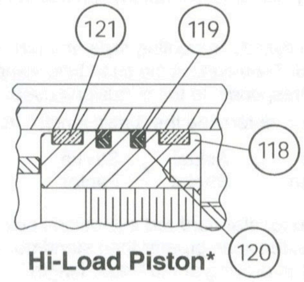
Recommended Mounting Styles for Maximum Stroke and Thrust Loads	Rod End Connection	Case		Stroke Factor
Groups 1 or 3 Long stroke cylinders for thrust loads should be firmly fixed at one end and aligned to take the principal force. Additional mounting should be specified at the opposite end, which should be used for alignment and support. An intermediate support may also be desirable for long stroke cylinders mounted horizontally. Machine mounting pads can be adjustable for support mountings to achieve proper alignment.	Fixed and Rigidly Guided	I		.50
	Pivoted and Rigidly Guided	II		.70
	Supported but not Rigidly Guided	III		2.00
Group 2 Style D — Trunnion on Head	Pivoted and Rigidly Guided	IV		1.00
Style BE — Eye on Cap	Pivoted and Rigidly Guided	V		2.00

Parts List



Parts List

Symbol	Description
1	Head
7	Cap
15	Cylinder Body
17	Piston, lipseal type
18	Cushion sleeve, head end cushion
18A	Cushion check spring, head end cushion
18B	Cushion retaining wire, head end cushion
23	Bolt, head and cap to body
37	Piston rod, single rod type
42	Lipseal, piston
44	Anti-roll ring, piston lipseal
46	Retaining ring, piston lipseal
47	O-ring, cylinder body to head and cap seal
118	Piston, Hi-Load type*
119	Outer ring
120	Inner ring
121	Wear ring
123	Washer
137	Cushion sleeve, cap end cushion
137A	Cushion check spring, cap end cushion
137B	Cushion retaining wire, cap end cushion
137C	Cushion support, cap end cushion



Hi-Load Piston*

Piston and Rod Assemblies
Factory assembled piston and rod assemblies (that include seals for the piston type specified) are recommended.

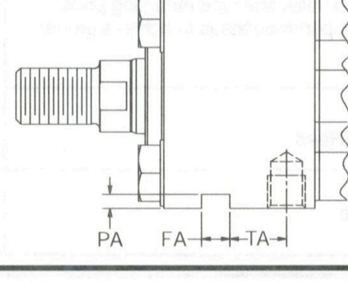
*Hi-Load Piston design available only in 1$^1/_2$", 2" and 2$^1/_2$" bores with oversize rod.

Thrust Key Mounting

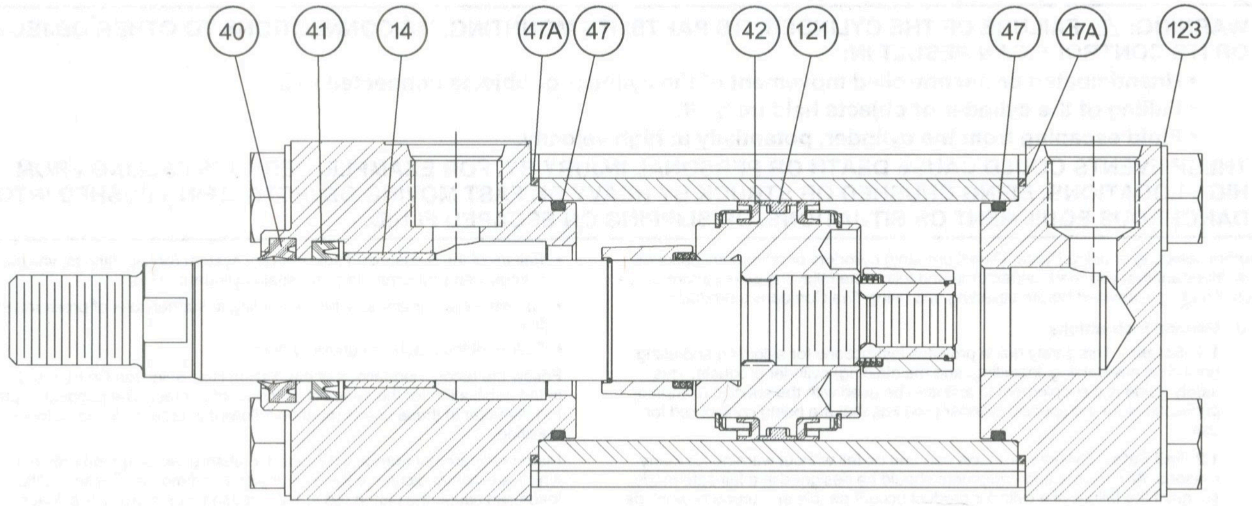
In addition to mounting bolts, Style F cylinders should be keyed to the mounting surface with a thrust key.

An optional groove can be supplied in the head for installing a thrust key.

Bore	+.001 -.000 FA	PA	TA
1 1/2	0.312	5/32	5/8
2	0.375	3/16	3/4
2 1/2	0.375	3/16	3/4
3 1/4	0.500	1/4	7/8
4	0.500	1/4	7/8



Parts Identification

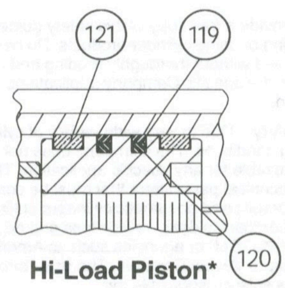


Parts List

Symbol	Description
14	Rod bearing
40	Rod wiper
41	Rod Seal
42	Lipseal, piston
47	O-ring, cylinder body to head and cap seal

Symbol	Description
47A	Gasket, cylinder body to head and cap seal
119	Outer Ring
120	Inner Ring
121	Wear ring
123	Washer

*Hi-Load Piston design available only in 1½", 2" and 2½" bores with oversize rod.



Hi-Load Piston*

Seal Kits

See Model Code and Standard Specifications page for compatibility.

Piston Seal Kits

Bore Ø	Rod Ø	Class 1		Class 5	
		Piston Lipseal Kits (contains: 2 Each Sym. # 42, 47 & 47A)	Hi-Load Piston Seal Kits (contains: 2 Each Sym. # 119, 120, 121, 47 & 47A)	Piston Lipseal Kits (contains: 2 Each Sym. # 42, 47 & 47A)	Hi-Load Piston Seal Kits (contains: 2 Each Sym. # 119, 120, 121, 47 & 47A)
1 1/2	5/8	PK15SHG001	N/A	PK15SHG005	N/A
	1	N/A	PK15SHGK01	N/A	PK15SHGK05
2	1	PK20SHG001	N/A	PK20SHG005	N/A
	1 3/8	N/A	PK20SHGK01	N/A	PK20SHGK05
2 1/2	1	PK25SHG001	N/A	PK25SHG005	N/A
	1 3/4	N/A	PK25SHGK01	N/A	PK25SHGK05
3 1/4	All	PK32SHG001	N/A	PK32SHG005	N/A
4	All	PK40SHG001	N/A	PK40SHG005	N/A

Note: Lipseal piston design is not available in 1½", 2", and 2½" bores with oversize rod. Use Hi-Load piston seal kit to service these bore and rod combinations.

Rod Bearing and Seal Kits

Bore Ø	Rod Ø	Class 1	Class 5
		Rod Bearing & Seal Kits (contains: 1 Each Sym. # 14, 40, 41, 47 & 47A)	Rod Bearing & Seal Kits (contains: 1 Each Sym. # 14, 40, 41, 47 & 47A)
1 1/2	5/8	RGSHG15061	RGSHG15065
	1	RGSHG15101	RGSHG15105
2	1	RGSHG20101	RGSHG20105
	1 3/8	RGSHG20131	RGSHG20135
2 1/2	1	RGSHG25101	RGSHG25105
	1 3/4	RGSHG25171	RGSHG25175
3 1/4	1 3/8	RGSHG32131	RGSHG32135
	2	RGSHG32201	RGSHG32205
4	1 3/4	RGSHG40171	RGSHG40175
	2 1/2	RGSHG40251	RGSHG40255

Stat-O-Seal™ Washer Kit for Series SHG†

Bore Ø	Stat-O-Seal Washer Kit (contains: 8 Each Sym. # 123)	Head & Cap to Body Bolt Torque†† (ft. lbs.)
1 1/2	WK15SHG001	18 - 19
2, 2 1/2	WK25SHG001	46 - 49
3 1/4	WK40SHG001	120 - 124
4	WK40SHG001	131 - 135

†Stat-O-Seal washers must be replaced when reassembling a Series SHG cylinder.
††Anti-seize lubricant required on bolt thread.
H-1 rated anti-seize lubricant must be used for Series SHG.

Safety Guide for Selecting and Using Hydraulic, Pneumatic Cylinders and Their Accessories

WARNING: ⚠ FAILURE OF THE CYLINDER, ITS PARTS, ITS MOUNTING, ITS CONNECTIONS TO OTHER OBJECTS, OR ITS CONTROLS CAN RESULT IN:

- **Unanticipated or uncontrolled movement of the cylinder or objects connected to it.**
- **Falling of the cylinder or objects held up by it.**
- **Fluid escaping from the cylinder, potentially at high velocity.**

THESE EVENTS COULD CAUSE DEATH OR PERSONAL INJURY BY, FOR EXAMPLE, PERSONS FALLING FROM HIGH LOCATIONS, BEING CRUSHED OR STRUCK BY HEAVY OR FAST MOVING OBJECTS, BEING PUSHED INTO DANGEROUS EQUIPMENT OR SITUATIONS, OR SLIPPING ON ESCAPED FLUID.

Before selecting or using Parker (The Company) cylinders or related accessories, it is important that you read, understand and follow the following safety information. Training is advised before selecting and using The Company's products.

1.0 General Instructions

1.1 Scope – This safety guide provides instructions for selecting and using (including assembling, installing, and maintaining) cylinder products. This safety guide is a supplement to and is to be used with the specific Company publications for the specific cylinder products that are being considered for use.

1.2 Fail Safe – Cylinder products can and do fail without warning for many reasons. All systems and equipment should be designed in a fail-safe mode so that if the failure of a cylinder product occurs people and property won't be endangered.

1.3 Distribution – Provide a free copy of this safety guide to each person responsible for selecting or using cylinder products. Do not select or use The Company's cylinders without thoroughly reading and understanding this safety guide as well as the specific Company publications for the products considered or selected.

1.4 User Responsibility – Due to very wide variety of cylinder applications and cylinder operating conditions, The Company does not warrant that any particular cylinder is suitable for any specific application. This safety guide does not analyze all technical parameters that must be considered in selecting a product. The hydraulic and pneumatic cylinders outlined in this catalog are designed to The Company's design guidelines and do not necessarily meet the design guideline of other agencies such as American Bureau of Shipping, ASME Pressure Vessel Code etc. The user, through its own analysis and testing, is solely responsible for:

- Making the final selection of the cylinders and related accessories.
- Determining if the cylinders are required to meet specific design requirements as required by the Agency(s) or industry standards covering the design of the user's equipment.
- Assuring that the user's requirements are met, OSHA requirements are met, and safety guidelines from the applicable agencies such as but not limited to ANSI are followed and that the use presents no health or safety hazards.
- Providing all appropriate health and safety warnings on the equipment on which the cylinders are used.

1.5 Additional Questions – Call the appropriate Company technical service department if you have any questions or require any additional information. See the Company publication for the product being considered or used, or call 1-847-298-2400, or go to www.parker.com, for telephone numbers of the appropriate technical service department.

2.0 Cylinder and Accessories Selection

2.1 Seals – Part of the process of selecting a cylinder is the selection of seal compounds. Before making this selection, consult the "seal information page(s)" of the publication for the series of cylinders of interest.

The application of cylinders may allow fluids such as cutting fluids, wash down fluids etc. to come in contact with the external area of the cylinder. These fluids may attack the piston rod wiper and or the primary seal and must be taken into account when selecting and specifying seal compounds.

Dynamic seals will wear. The rate of wear will depend on many operating factors. Wear can be rapid if a cylinder is mis-aligned or if the cylinder has been improperly serviced. The user must take seal wear into consideration in the application of cylinders.

2.2 Piston Rods – Possible consequences of piston rod failure or separation of the piston rod from the piston include, but are not limited to are:

- Piston rod and or attached load thrown off at high speed.
- High velocity fluid discharge.
- Piston rod extending when pressure is applied in the piston retract mode.

Piston rods or machine members attached to the piston rod may move suddenly and without warning as a consequence of other conditions occurring to the machine such as, but not limited to:

- Unexpected detachment of the machine member from the piston rod.
- Failure of the pressurized fluid delivery system (hoses, fittings, valves, pumps, compressors) which maintain cylinder position.
- Catastrophic cylinder seal failure leading to sudden loss of pressurized fluid.
- Failure of the machine control system.

Follow the recommendations of the "Piston Rod Selection Chart and Data" in the publication for the series of cylinders of interest. The suggested piston rod diameter in these charts must be followed in order to avoid piston rod buckling.

Piston rods are not normally designed to absorb bending moments or loads which are perpendicular to the axis of piston rod motion. These additional loads can cause the piston rod to fail. If these types of additional loads are expected to be imposed on the piston rod, their magnitude should be made known to our engineering department.

The cylinder user should always make sure that the piston rod is securely attached to the machine member.

On occasion cylinders are ordered with double rods (a piston rod extended from both ends of the cylinder). In some cases a stop is threaded on to one of the piston rods and used as an external stroke adjuster. On occasions spacers are attached to the machine member connected to the piston rod and also used as a stroke adjuster. In both cases the stops will create a pinch point and the user should consider appropriate use of guards. If these external stops are not perpendicular to the mating contact surface, or if debris is trapped between the contact surfaces, a bending moment will be placed on the piston rod, which can lead to piston rod failure. An external stop will also negate the effect of cushioning and will subject the piston rod to impact loading. Those two (2) conditions can cause piston rod failure. Internal stroke adjusters are available with and without cushions. The use of external stroke adjusters should be reviewed with our engineering department.

The piston rod to piston and the stud to piston rod threaded connections are secured with an anaerobic adhesive. The strength of the adhesive decreases with increasing temperature. Cylinders which can be exposed to temperatures above +250°F (+121°C) are to be ordered with a non studded piston rod and a pinned piston to rod joint.

2.3 Cushions – Cushions should be considered for cylinder applications when the piston velocity is expected to be over 4 inches/second.

Cylinder cushions are normally designed to absorb the energy of a linear applied load. A rotating mass has considerably more energy than the same mass moving in a linear mode. Cushioning for a rotating mass application should be review by our engineering department.

2.4 Cylinder Mountings – Some cylinder mounting configurations may have certain limitations such as but not limited to minimum stroke for side or foot mounting cylinders or pressure de-ratings for certain mounts. Carefully review the catalog for these types of restrictions.

Always mount cylinders using the largest possible high tensile alloy steel socket head cap screws that can fit in the cylinder mounting holes and torque them to the manufacturer's recommendations for their size.

2.5 Port Fittings – Hydraulic cylinders applied with meter out or deceleration circuits are subject to intensified pressure at piston rod end.

The rod end pressure is approximately equal to:

$$\frac{\text{operating pressure x effective cap end area}}{\text{effective rod end piston area}}$$

Contact your connector supplier for the pressure rating of individual connectors.

3.0 Cylinder and Accessories Installation and Mounting

3.1 Installation

3.1.1 – Cleanliness is an important consideration, and cylinders are shipped with the ports plugged to protect them from contaminants entering the ports. These plugs should not be removed until the piping is to be installed. Before making the connection to the cylinder ports, piping should be thoroughly cleaned to remove all chips or burrs which might have resulted from threading or flaring operations.

3.1.2 – Cylinders operating in an environment where air drying materials are present such as fast-drying chemicals, paint, or weld splatter, or other hazardous conditions such as excessive heat, should have shields installed to prevent damage to the piston rod and piston rod seals.

3.1.3 – Proper alignment of the cylinder piston rod and its mating component on the machine should be checked in both the extended and retracted positions. Improper alignment will result in excessive rod gland and/or cylinder bore wear. On fixed mounting cylinders attaching the piston rod while the rod is retracted will help in achieving proper alignment.

3.1.4 – Sometimes it may be necessary to rotate the piston rod in order to thread the piston rod into the machine member. This operation must always be done with zero pressure being applied to either side of the piston. Failure to follow this procedure may result in loosening the piston to rod-threaded connection. In some rare cases the turning of the piston rod may rotate a threaded piston rod gland and loosen it from the cylinder head. Confirm that this condition is not occurring. If it does, re-tighten the piston rod gland firmly against the cylinder head.

For double rod cylinders it is also important that when attaching or detaching the piston rod from the machine member that the torque be applied to the piston rod end of the cylinder that is directly attaching to the machine member with the opposite end unrestrained. If the design of the machine is such that only the rod end of the cylinder opposite to where the rod attaches to the machine member can be rotated, consult the factory for further instructions.

3.2 Mounting Recommendations

3.2.1 – Always mount cylinders using the largest possible high tensile alloy steel socket head screws that can fit in the cylinder mounting holes and torque them to the manufacturer's recommendations for their size.

3.2.2 – Side-Mounted Cylinders – In addition to the mounting bolts, cylinders of this type should be equipped with thrust keys or dowel pins located so as to resist the major load.

3.2.3 – Tie Rod Mounting – Cylinders with tie rod mountings are recommended for applications where mounting space is limited. The standard tie rod extension is shown as BB in dimension tables. Longer or shorter extensions can be supplied. Nuts used for this mounting style should be torqued to the same value as the tie rods for that bore size.

3.2.4 – Flange Mount Cylinders – The controlled diameter of the rod gland extension on head end flange mount cylinders can be used as a pilot to locate the cylinders in relation to the machine. After alignment has been obtained, the flanges may be drilled for pins or dowels to prevent shifting.

3.2.5 – Trunnion Mountings – Cylinders require lubricated bearing blocks with minimum bearing clearances. Bearing blocks should be carefully aligned and rigidly mounted so the trunnions will not be subjected to bending moments. The rod end should also be pivoted with the pivot pin in line and parallel to axis of the trunnion pins.

3.2.6 – Clevis Mountings – Cylinders should be pivoted at both ends with centerline of pins parallel to each other. After cylinder is mounted, be sure to check to assure that the cylinder is free to swing through its working arc without interference from other machine parts.

4.0 Cylinder and Accessories Maintenance, Troubleshooting and Replacement

4.1 Storage – At times cylinders are delivered before a customer is ready to install them and must be stored for a period of time. When storage is required the following procedures are recommended.

4.1.1 – Store the cylinders in an indoor area which has a dry, clean and noncorrosive atmosphere. Take care to protect the cylinder from both internal corrosion and external damage.

4.1.2 – Whenever possible cylinders should be stored in a vertical position (piston rod up). This will minimize corrosion due to possible condensation which could occur inside the cylinder. This will also minimize seal damage.

4.1.3 – Port protector plugs should be left in the cylinder until the time of installation.

4.1.4 – If a cylinder is stored full of hydraulic fluid, expansion of the fluid due to temperature changes must be considered. Installing a check valve with free flow out of the cylinder is one method.

4.1.5 – When cylinders are mounted on equipment that is stored outside for extended periods, exposed unpainted surfaces, e.g. piston rod, must be coated with a rust-inhibiting compound to prevent corrosion.

4.2 Cylinder Trouble Shooting

4.2.1 – **External Leakage**

4.2.1.1 – Rod seal leakage can generally be traced to worn or damaged seals. Examine the piston rod for dents, gouges or score marks, and replace piston rod if surface is rough.

Rod seal leakage could also be traced to gland wear. If clearance is excessive, replace rod bushing and seal. Rod seal leakage can also be traced to seal deterioration. If seals are soft or gummy or brittle, check compatibility of seal material with lubricant used if air cylinder, or operating fluid if hydraulic cylinder. Replace with seal material, which is compatible with these fluids. If the seals are hard or have lost elasticity, it is usually due to exposure to temperatures in excess of 165°F. (+74°C). Shield the cylinder from the heat source to limit temperature to 350°F. (+177°C.) and replace with fluorocarbon seals.

4.2.1.2 – Cylinder body seal leak can generally be traced to loose tie rods. Torque the tie rods to manufacturer's recommendation for that bore size.

Excessive pressure can also result in cylinder body seal leak. Determine maximum pressure to rated limits. Replace seals and retorque tie rods as in paragraph above. Excessive pressure can also result in cylinder body seal leak. Determine if the pressure rating of the cylinder has been exceeded. If so, bring the operating pressure down to the rating of the cylinder and have the tie rods replaced.

Pinched or extruded cylinder body seal will also result in a leak. Replace cylinder body seal and retorque as in paragraph above.

Cylinder body seal leakage due to loss of radial squeeze which shows up in the form of flat spots or due to wear on the O.D. or I.D. – Either of these are symptoms of normal wear due to high cycle rate or length of service. Replace seals as per paragraph above.

4.2.2 – **Internal Leakage**

4.2.2.1 – Piston seal leak (by-pass) 1 to 3 cubic inches per minute leakage is considered normal for piston ring construction. Virtually no static leak with lipseal type seals on piston should be expected. Piston seal wear is a usual cause of piston seal leakage. Replace seals as required.

4.2.2.2 – With lipseal type piston seals excessive back pressure due to over-adjustment of speed control valves could be a direct cause of rapid seal wear. Contamination in a hydraulic system can result in a scored cylinder bore, resulting in rapid seal wear. In either case, replace piston seals as required.

4.2.2.3 – What appears to be piston seal leak, evidenced by the fact that the cylinder drifts, is not always traceable to the piston. To make sure, it is suggested that one side of the cylinder piston be pressurized and the fluid line at the opposite port be disconnected. Observe leakage. If none is evident, seek the cause of cylinder drift in other component parts in the circuit.

4.2.3 – **Cylinder Fails to Move the Load**

4.2.3.1 – Pneumatic or hydraulic pressure is too low. Check the pressure at the cylinder to make sure it is to circuit requirements.

4.2.3.2 – Piston Seal Leak – Operate the valve to cycle the cylinder and observe fluid flow at valve exhaust ports at end of cylinder stroke. Replace piston seals if flow is excessive.

4.2.3.3 – Cylinder is undersized for the load – Replace cylinder with one of a larger bore size.

4.3 Erratic or Chatter Operation

4.3.1 – Excessive friction at rod gland or piston bearing due to load misalignment – Correct cylinder-to-load alignment.

4.3.2 – Cylinder sized too close to load requirements – Reduce load or install larger cylinder.

4.3.3 – Erratic operation could be traced to the difference between static and kinetic friction. Install speed control valves to provide a back pressure to control the stroke.

4.4 Cylinder Modifications, Repairs, or Failed Component – Cylinders as shipped from the factory are not to be disassembled and or modified. If cylinders require modifications, these modifications must be done at company locations or by The Company's certified facilities. The Cylinder Division Engineering Department must be notified in the event of a mechanical fracture or permanent deformation of any cylinder component (excluding seals). This includes a broken piston rod, tie rod, mounting accessory or any other cylinder component. The notification should include all operation and application details. This information will be used to provide an engineered repair that will prevent recurrence of the failure.

It is allowed to disassemble cylinders for the purpose of replacing seals or seal assemblies. However, this work must be done by strictly following all the instructions provided with the seal kits.

Offer of Sale

The items described in this document and other documents or descriptions provided by Parker Hannifin Corporation, its subsidiaries and Divisions ("Company") and its authorized distributors, are hereby offered for sale at prices to be established by the Company, its subsidiaries and its authorized distributors. This offer and its acceptance by any customer ("Buyer") shall be governed by all of the following Terms and Conditions. Buyer's order for any such item, when communicated to the Company, its subsidiary or an authorized distributor ("Seller") verbally or in writing, shall constitute acceptance of this offer.

1. Terms and Conditions of Sale: All descriptions, quotations, proposals, offers, acknowledgments, acceptances and sales of Seller's products are subject to and shall be governed exclusively by the terms and conditions stated herein. Buyer's acceptance of any offer to sell is limited to these terms and conditions. Any terms or conditions in addition to, or inconsistent with those stated herein, proposed by Buyer in any acceptance of an offer by Seller, are hereby objected to. No such additional, different or inconsistent terms and conditions shall become part of the contract between Buyer and Seller unless expressly accepted in writing by Seller. Seller's acceptance of any offer to purchase by Buyer is expressly conditional upon Buyer's assent to all the terms and conditions stated herein, including any terms in addition to, or inconsistent with those contained in Buyer's offer. Acceptance of Seller's products shall in all events constitute such assent.

2. Payment: Payment shall be made by Buyer net 30 days from the date of delivery of the items purchased hereunder. Amounts not timely paid shall bear interest at the maximum rate permitted by law for each month or portion thereof that the Buyer is late in making payment. Any claims by Buyer for omissions or shortages in a shipment shall be waived unless Seller receives notice thereof within 30 days after Buyer's receipt of the shipment.

3. Delivery: Unless otherwise provided on the face hereof, delivery shall be made F.O.B. Seller's plant. Regardless of the method of delivery, however, risk of loss shall pass to Buyer upon Seller's delivery to a carrier. Any delivery dates shown are approximate only and Seller shall have no liability for any delays in delivery.

4. Warranty: Seller warrants that the items sold hereunder shall be free from defects in material or workmanship for a period of 18 months from date of shipment from the Company. **THIS WARRANTY COMPRISES THE SOLE AND ENTIRE WARRANTY PERTAINING TO ITEMS PROVIDED HEREUNDER. SELLER MAKES NO OTHER WARRANTY, GUARANTEE, OR REPRESENTATION OF ANY KIND WHATSOEVER. ALL OTHER WARRANTIES, INCLUDING BUT NOT LIMITED TO, MERCHANTABILITY AND FITNESS FOR PURPOSE, WHETHER EXPRESS, IMPLIED, OR ARISING BY OPERATION OF LAW, TRADE USAGE, OR COURSE OF DEALING ARE HEREBY DISCLAIMED.**

NOTWITHSTANDING THE FOREGOING, THERE ARE NO WARRANTIES WHATSOEVER ON ITEMS BUILT OR ACQUIRED WHOLLY OR PARTIALLY, TO BUYER'S DESIGN OR SPECIFICATIONS.

5. Limitation of Remedy: SELLER'S LIABILITY ARISING FROM OR IN ANY WAY CONNECTED WITH THE ITEMS SOLD OR THIS CONTRACT SHALL BE LIMITED EXCLUSIVELY TO REPAIR OR REPLACEMENT OF THE ITEMS SOLD OR REFUND OF THE PURCHASE PRICE PAID BY BUYER, AT SELLER'S SOLE OPTION. IN NO EVENT SHALL SELLER BE LIABLE FOR ANY INCIDENTAL, CONSEQUENTIAL OR SPECIAL DAMAGES OF ANY KIND OR NATURE WHATSOEVER, INCLUDING BUT NOT LIMITED TO LOST PROFITS ARISING FROM OR IN ANY WAY CONNECTED WITH THIS AGREEMENT OR ITEMS SOLD HEREUNDER, WHETHER ALLEGED TO ARISE FROM BREACH OF CONTRACT, EXPRESS OR IMPLIED WARRANTY, OR IN TORT, INCLUDING WITHOUT LIMITATION, NEGLIGENCE, FAILURE TO WARN OR STRICT LIABILITY.

6. Changes, Reschedules and Cancellations: Buyer may request to modify the designs or specifications for the items sold hereunder as well as the quantities and delivery dates thereof, or may request to cancel all or part of this order, however, no such requested modification or cancellation shall become part of the contract between Buyer and Seller unless accepted by Seller in a written amendment to this Agreement. Acceptance of any such requested modification or cancellation shall be at Seller's discretion, and shall be upon such terms and conditions as Seller may require.

7. Special Tooling: A tooling charge may be imposed for any special tooling, including without limitations, dies, fixtures, molds and patterns, acquired to manufacture items sold pursuant to this contract. Such special tooling shall be and remain Seller's property notwithstanding payment of any charges by Buyer. In no event will Buyer acquire any interest in apparatus belonging to Seller which is utilized in the manufacture of the items sold hereunder, even if such apparatus has been specially converted or adapted for such manufacture and notwithstanding any charges paid by Buyer. Unless otherwise agreed, Seller shall have the right to alter, discard or otherwise dispose of any special tooling or other property in its sole discretion at any time.

8. Buyer's Property: Any designs, tools, patterns, materials, drawings, confidential information or equipment furnished by Buyer, or any other items which become Buyer's property, may be considered obsolete and may be destroyed by Seller after two (2) consecutive years have elapsed without Buyer placing an order for the items which are manufactured using such property. Seller shall not be responsible for any loss or damage to such property while it is in Seller's possession or control.

9. Taxes: Unless otherwise indicated on the face hereof, all prices and charges are exclusive of excise, sales, use, property, occupational or like taxes which may be imposed by any taxing authority upon the manufacture, sale or delivery of the items sold hereunder. If any such taxes must be paid by Seller or if Seller is liable for the collection of such tax, the amount thereof shall be in addition to the amounts for the items sold. Buyer agrees to pay all such taxes or to reimburse Seller therefore upon receipt of its invoice. If Buyer claims exemption from any sales, use or other tax imposed by any taxing authority, Buyer shall save Seller harmless from and against any such tax, together with any interest or penalties thereon which may be assessed if the items are held to be taxable.

10. Indemnity For Infringement of Intellectual Property Rights: Seller shall have no liability for infringement of any patents, trademarks, copyrights, trade dress, trade secrets or similar rights except as provided in this Part 10. Seller will defend and indemnify Buyer against allegations of infringement of U.S. patents, U.S. trademarks, copyrights, trade dress and trade secrets (hereinafter "Intellectual Property Rights"). Seller will defend at its expense and will pay the cost of any settlement or damages awarded in an action brought against Buyer based on an allegation that an item sold pursuant to this contract infringes the Intellectual Property Rights of a third party. Seller's obligation to defend and indemnify Buyer is contingent on Buyer notifying Seller within ten (10) days after Buyer becomes aware of such allegations of infringement, and Seller having sole control over the defense of any allegations or actions including all negotiations for settlement or compromise. If an item sold hereunder is subject to a claim that it infringes the Intellectual Property Rights of a third party, Seller may, at its sole expense and option, procure for Buyer the right to continue using said item, replace or modify said item so as to make it noninfringing, or offer to accept return of said item and return the purchase price less a reasonable allowance for depreciation. Notwithstanding the foregoing, Seller shall have no liability for claims of infringement based on information provided by Buyer, or directed to items delivered hereunder for which the designs are specified in whole or part by Buyer, or infringements resulting from the modification, combination or use in a system of any item sold hereunder. The foregoing provisions of this Part 10 shall constitute Seller's sole and exclusive liability and Buyer's sole and exclusive remedy for infringement of Intellectual Property Rights.

If a claim is based on information provided by Buyer or if the design for an item delivered hereunder is specified in whole or in part by Buyer, Buyer shall defend and indemnify Seller for all costs, expenses or judgements resulting from any claim that such item infringes any patent, trademark, copyright, trade dress, trade secret or any similar right.

11. Force Majeure: Seller does not assume the risk of and shall not be liable for delay or failure to perform any of Seller's obligations by reason of circumstances beyond the reasonable control of Seller (hereinafter "Events of Force Majeure"). Events of Force Majeure shall include without limitation, accidents, acts of God, strikes or labor disputes, acts, laws, rules or regulations of any government or government agency, fires, floods, delays or failures in delivery of carriers or suppliers, shortages of materials and any other cause beyond Seller's control.

12. Entire Agreement/Governing Law: The terms and conditions set forth herein, together with any amendments, modifications and any different terms or conditions expressly accepted by Seller in writing, shall constitute the entire Agreement concerning the items sold, and there are no oral or other representations or agreements which pertain thereto. This Agreement shall be governed in all respects by the law of the State of Ohio. No actions arising out of sale of the items sold hereunder or this Agreement may be brought by either party more than two (2) years after the cause of action accrues.

APPENDIX E

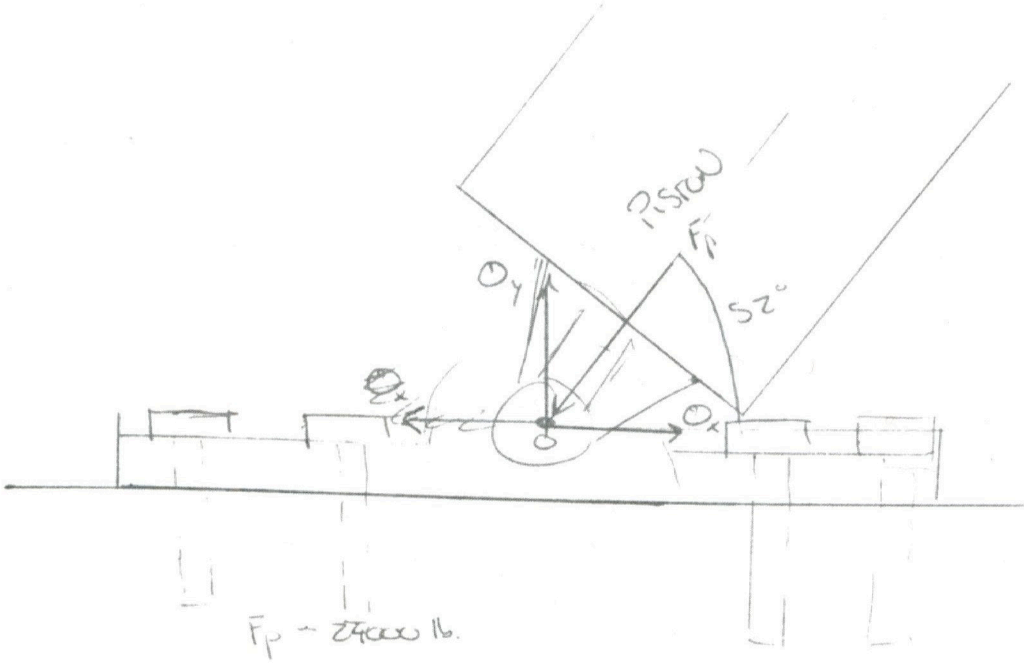
Weight =	1500	lb
a=	2.314825	ft
b =	3	ft
c =	0.5	Ft

l =

2.5		
P1		
Min	Max	
2.704318	5.656854	2.091786
2.770146	5.692319	2.054881
2.838457	5.728001	2.017998
2.90911	5.763896	1.981327
2.98197	5.8	1.945023
3.056915	5.836309	1.909215
3.13383	5.872819	1.874007
3.212612	5.909526	1.839477
3.293166	5.946427	1.805687
3.375403	5.983519	1.772683
3.459245	6.020797	1.740495
3.544621	6.058259	1.709142
3.631465	6.0959	1.678634
3.719719	6.133718	1.648973
3.809331	6.17171	1.620156
3.900254	6.209871	1.592171
3.992443	6.2482	1.565007
4.08586	6.286692	1.538646
4.18047	6.325346	1.513071
4.27624	6.364157	1.48826
4.638841	5.874851	1.266448

d	e	i	j	alpha	beta	k	m
3.564473	2	4	5	36.8699	56.8699	0.418716	0.273271
3.805064	2.1	3.9	4.920366	37.56859	57.56859	0.422017	0.268145
4.060736	2.2	3.8	4.841487	38.29016	58.29016	0.42536	0.262809
4.333223	2.3	3.7	4.763402	39.03551	59.03551	0.428743	0.257253
4.624535	2.4	3.6	4.68615	39.80557	59.80557	0.432162	0.251468
4.93702	2.5	3.5	4.609772	40.60129	60.60129	0.435612	0.245442
5.273427	2.6	3.4	4.534314	41.42367	61.42367	0.43909	0.239165
5.637009	2.7	3.3	4.459821	42.27369	62.27369	0.44259	0.232624
6.031632	2.8	3.2	4.386342	43.15239	63.15239	0.446105	0.22581
6.461932	2.9	3.1	4.313931	44.06081	64.06081	0.449629	0.218708
6.933521	3	3	4.242641	45	65	0.453154	0.211309
7.453248	3.1	2.9	4.172529	45.97102	65.97102	0.45667	0.203599
8.029567	3.2	2.8	4.103657	46.97493	66.97493	0.460167	0.195567
8.673038	3.3	2.7	4.036087	48.01279	68.01279	0.463634	0.1872
9.397018	3.4	2.6	3.969887	49.08562	69.08562	0.467057	0.178486
10.21866	3.5	2.5	3.905125	50.19443	70.19443	0.470424	0.169415
11.16036	3.6	2.4	3.841875	51.34019	71.34019	0.473717	0.159974
12.25193	3.7	2.3	3.780212	52.52382	72.52382	0.476921	0.150155
13.53396	3.8	2.2	3.720215	53.74616	73.74616	0.480016	0.139947
15.06311	3.9	2.1	3.661967	55.00798	75.00798	0.482981	0.129342
-10.4899	4	2	3.605551	90	110	0.469846	-0.17101

Bolt Shearing Calculations

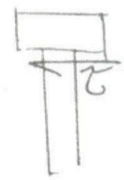


$F_p \sim 24000$ lb.

$\Sigma F_x = 0$

$F_p \cos(52^\circ) = O_x$

$O_x = 14775$ lb



$\tau = \frac{F}{A}$

n_d = design factor = 3

N = # of bolts

$$\tau = \frac{O_x}{N \pi d^2/4} = 0.577 \frac{S_p}{n_d}$$

$$N = \frac{O_x}{\frac{\pi d^2}{4}\left(0.577 \frac{S_p}{n_d}\right)}$$

$$= \frac{14.775 \text{ kips}}{\frac{\pi (0.75 \text{ in})^2}{4}(0.577)\left(\frac{120 \text{ kpsi}}{3}\right)}$$

$N = 1.45$ bolts

YAY!

PIPE WEIGHT CALCULATION

GIVEN: $\rho_{STEEL} = .29\ lb/in^3$ PIPE OD = 20 inch

$L_{STEEL} = 96$ inch PIPE ID = 18.5 inch

ANALYSIS:

$$VOLUME\ PIPE = L_{STEEL} \cdot \pi (OD - ID)^2$$

$$VOLUME\ PIPE = 96\ inch \cdot \pi \left(\left(\frac{20}{2}\right)^2 - \left(\frac{18.5}{2}\right)^2\right) = 8164\ inch^3$$

$$PIPE\ WEIGHT = \rho_{STEEL} \cdot VOLUME\ PIPE$$

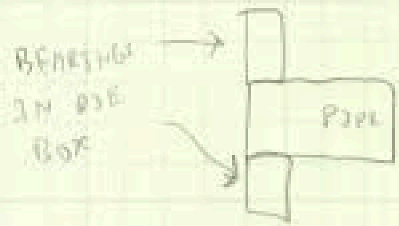
$$PIPE\ WEIGHT = .29 \frac{lb}{in^3} \cdot 8164\ inch^3$$

PIPE WEIGHT = 2367 lb

THIS CALCULATION HAS BEEN VERIFIED FOR AN EXCEL SPREADSHEET. THIS SAME CALCULATION HAS BEEN DONE FOR THE MANDREL

DIE SYSTEM FRICTION FORCE

GIVEN:



ANALYSIS:

$\mu_s = .16$ STEEL ON STEEL LUBRICATED

$F_{N_1} = 1{,}000{,}000\ lb$

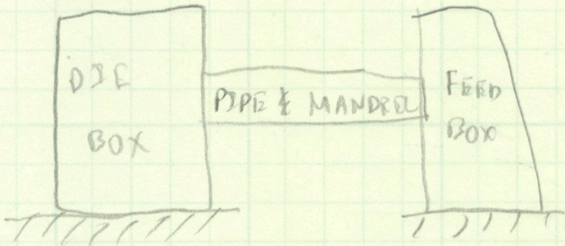
$F_{f_1} = F_{N_1} \mu_s = 1{,}000{,}000 \cdot .16 = 160{,}000\ lb$

$F_{N_2} = 1{,}000{,}000\ lb$

$F_{f_2} = F_{N_2} \mu_s = 1{,}000{,}000 \cdot .16 = 160{,}000\ lb$

FEED SYSTEM CABLE TENSION CALC

GIVEN: SYSTEM DRAWING



$F_{N1} = F_{N2} = 1{,}000{,}000\ lb = F_c$

$\mu_s = .16$ [STEEL ON STEEL LUBRICATED]

$W_{pipe} = 2367\ lb$

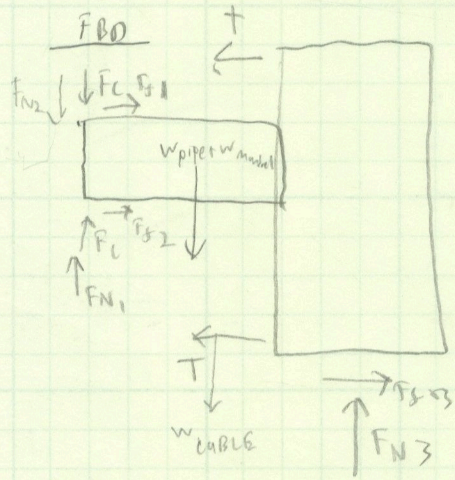
$W_{mandrel} = 29500\ lb$

$W_{cable} = 844.8\ lb$

n(# OF CABLES) = 30

$F_{f1} = 160{,}000 = F_{f2}$

ANALYSIS: FBD



$\Sigma F_y = 0$

$F_{N3} + F_{N1} - F_{N2} + F_c - F_c - W_{pipe} - W_{mandrel} - W_{cable} = 0$

$F_{N3} = 1{,}000{,}000 - 1{,}000{,}000 + 1{,}000{,}000 + 2367 + 29500 + 844.8 - 1{,}000{,}000$

$F_{N3} = 32{,}712\ lb$

$F_{f3} = \mu F_{N3} = .16 \cdot 32712 = 5{,}2339\ lb$

$\Sigma F_x = 0$

$F_{f3} + F_{f1} + F_{f2} - T = 0$

$T = 52339 + 160{,}000 + 160{,}000$

$T = 372339\ lb$

FOR 30 ROPES: $T_{per\ rope} = 12{,}411\ lb$

DETERMINE FATIGUE FACTOR OF SAFETY FOR CABLES

GIVEN: TENSION FORCE = 325,098 lb
# OF ROPES = 30
E = 12 MPSI , ROPE DIAMETER = 2 inches = d
D = SHEAVE DIAMETER = 52 inches
S_u = 251600 PSI
p/S_u = .0055 (FROM FIG 17-21 SHIGLEYS

$\frac{D}{d} = 26$

ANALYSIS:

$$F_f = \frac{(p/S_u) \cdot S_u \cdot D \cdot d}{2} = \frac{.0055 \cdot 251600 \cdot 26 d^2}{2} = 21120 d^2$$

$$F_b = \frac{E \cdot d_w \cdot A_m}{D} = \frac{12 \cdot 10^6 \cdot .067 d \cdot .4 d^2}{26} \cdot d = 2473 d^3$$

$$\frac{F_f - F_b}{\frac{F_t}{m}} = n_f = \frac{21120 d^2 - 2473 d^3}{\frac{325{,}098}{30}} = \boxed{6.61}$$

SAFETY FACTOR IS OKAY. OSHA WANTS >5

D/d RATIO & DIAMETERS WERE ITERATED TO FIND THE MOST FAVORABLE RESULT.

THESE CALCS ARE IN EXCEL.

RETURN CABLE FATIGUE LIFE

GIVEN: $F_T = 1063$ lb , $F_f = 19520$ lb, $F_B = 19790$ lb

$m = 5$

THE RETURN SYSTEM ONLY HAS TO DEAL WITH FRICTION OF THE FEED BOX & THIS MAKES IT HAVE A VERY LONG LIFE.

ANALYSIS:

$$FOS = \frac{F_f - F_B}{F_T} = \frac{91520 - 19790 \text{ lb}}{\frac{1063 \text{ lb}}{5}}$$

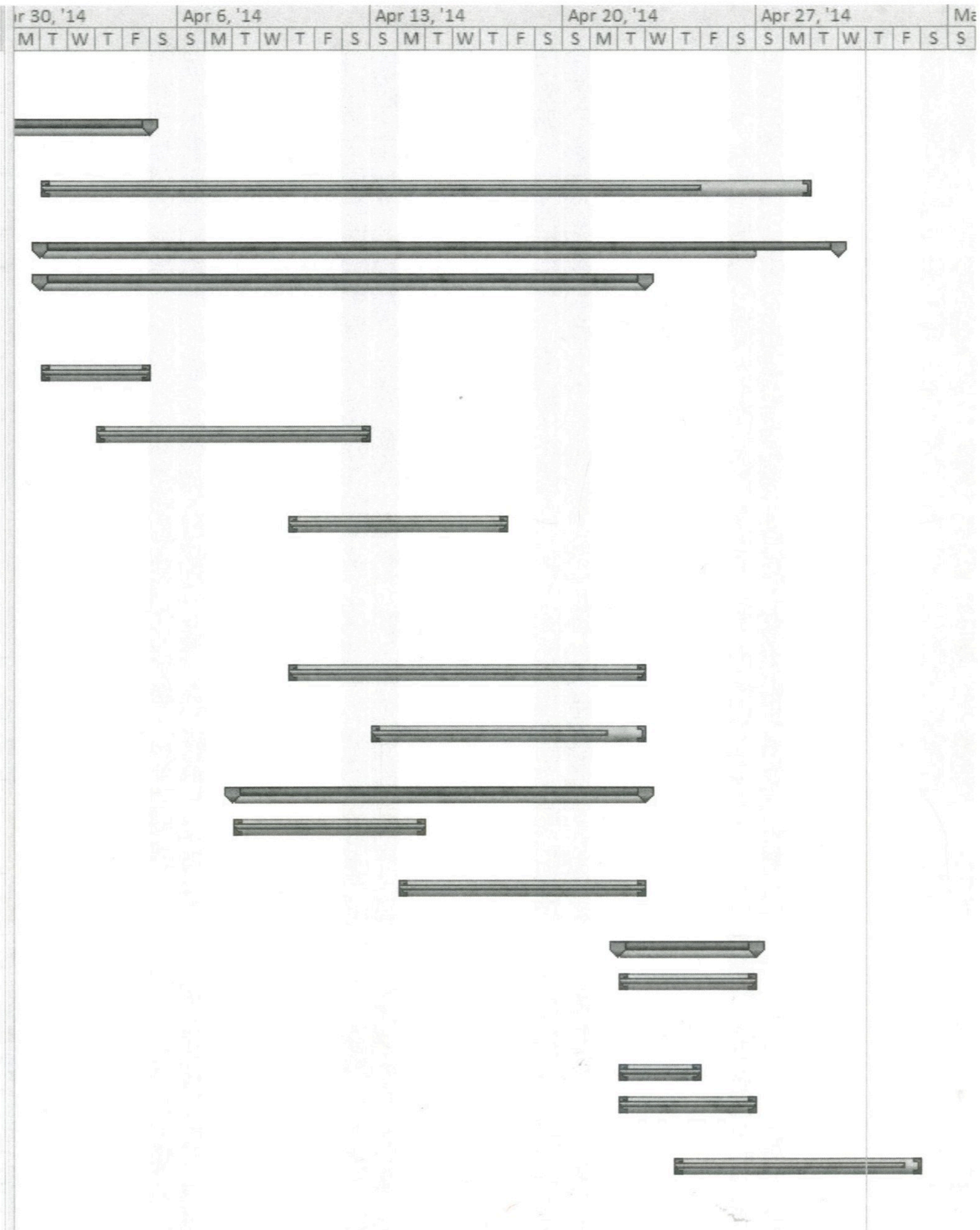
FOS = 67.5 ✓

FOR A TOTAL OF 30 YEARS @ 50 RUNS PER DAY

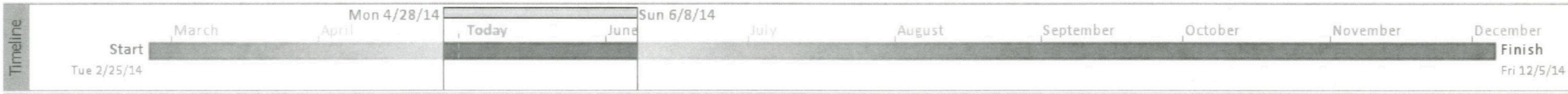
APPENDIX F

Appendix F: Spring Quarter Gantt Chart

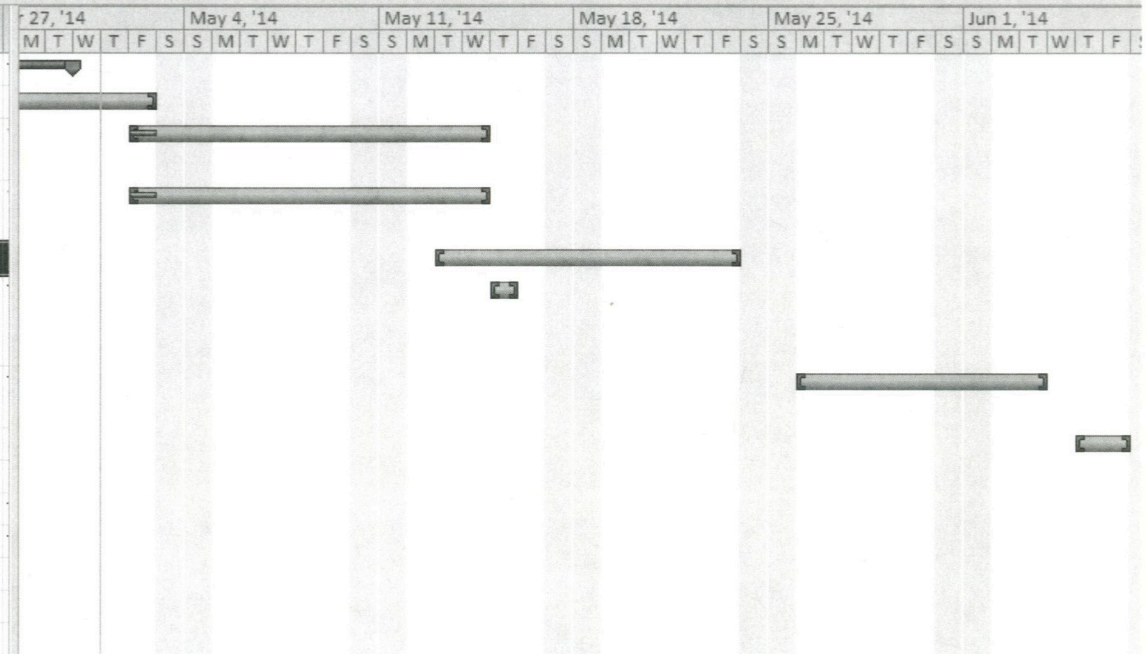
	ℹ	Task Mode	Task Name	Duration	Start	Finish
1	✓		Concept Design Review with Sponsor	1 day	Tue 2/25/14	Tue 2/25/14
2	✓		+ **Concept Design (ME 428)**	**29 days**	**Tue 2/25/14**	**Fri 4/4/14**
11			Schedule Concept Design Review	20 days	Tue 4/1/14	Mon 4/28/14
15			− **CDR**	**21 days**	**Tue 4/1/14**	**Tue 4/29/14**
23			− **Stress Analysis and CAD of Loading System**	**16 days**	**Tue 4/1/14**	**Tue 4/22/14**
28	✓		Pick Lever Arm Design	4 days	Tue 4/1/14	Fri 4/4/14
24	✓		Calculate Axial Stress on Arms during loading	8 days	Thu 4/3/14	Sat 4/12/14
25	✓		Determine Design Specifics (foundation, bearing sizes, bolt sizes)	6 days	Thu 4/10/14	Thu 4/17/14
26	✓		Perform Analysis on Design Specifics	9 days	Thu 4/10/14	Tue 4/22/14
27			Solid Works Drawing	8 days	Sun 4/13/14	Tue 4/22/14
20	✓		− **Cable Design**	**11 days**	**Tue 4/8/14**	**Tue 4/22/14**
21	✓		Sizing Analysis for Cable and Spindle	5 days	Tue 4/8/14	Mon 4/14/14
22	✓		Solid Works Drawing	7 days	Mon 4/14/14	Tue 4/22/14
16	✓		− **Lubrication Design**	**5 days**	**Tue 4/22/14**	**Sat 4/26/14**
17	✓		Analysis (Pump Sizing, Filtration, Viscosity)	5 days	Tue 4/22/14	Sat 4/26/14
18	✓		Design Layout	3 days	Tue 4/22/14	Thu 4/24/14
19	✓		Solid Works Drawing	5 days	Tue 4/22/14	Sat 4/26/14
29			Final Design Report	7 days	Thu 4/24/14	Fri 5/2/14
14			Complete Critical Design Review presentation	1 day	Thu 5/15/14	Thu 5/15/14



Appendix F: ME 430 Fall Quarter Gantt Chart



	Task Mode	Task Name	Duration	Start
17		+ **CDR**	**21 days**	**Tue 4/1/14**
33		Final Design Report	7 days	Thu 4/24/14
15		Develop Prototype Concept	9 days	Fri 5/2/14
16		Develop Redesign Concept	9 days	Fri 5/2/14
36		Order Parts/Redesign	9 days	Tue 5/13/14
14		Complete Critical Design Review presentation with Sponsor	1 day	Thu 5/15/14
37		Manufacturing Plan/Redesign	7 days	Mon 5/26/14
35		End of Quarter Report	2 days	Thu 6/5/14
38		Summer Break No Work	71 days	Sun 6/15/14
12		− **Senior Design Expo**	**9 days**	**Mon 11/10/14**
13		Concept Design Repor	6 days	Tue 2/25/14
34		Final Project Report	10 days	Mon 11/24/14



APPENDIX G

Assembling the loading arm is a very simple process, but order of assembly is of utmost importance. First, press fit the composite bushings into the triangular base plates. To create a stable base, measure out the distance between the two triangular base plates as defined in the detailed drawings. Drill holes into the concrete where the anchors will be placed at coordinates indicated by the drawings. Once the triangular plates have been anchored into place the hole for the main hydraulic piston needs to be cut. After this has been cut the base bracket of the main piston with shaft, pins, and hydraulic cylinder attached can be anchored into the ground at a distance defined by the drawings. With that taken care of, the next order of business is to anchor the support member inside of the main machines sidewalls. This can be done in an orientation shown by the design drawings. Once the support is anchored the lower lever arm can be aligned with the holes in the triangular base plates and the main shaft can be slid through. Pin the shaft to avoid any movement in the axial direction during use of the machine. Now that the lower arm is secured, attach the upper arm/gripper assembly to the free end of the main hydraulic cylinder. Next slide the upper arm/gripper assembly into the slot of the lower arm. At this point the hydraulic hosing can be attached to the main cylinder as well as the two smaller cylinders inside the gripper. Make sure the gripper cylinders are connected in series, ones exit to the others inlet. At this point the loading arm is ready for use.

When using the arm be cautious to not overextend the upper arm, doing this can cause the upper arm to disconnect from the lower arm causing the load of the pipe and upper arm to be applied as a point force to the extending arm of the main hydraulic cylinder causing structural failure of the cylinder and exposing the surrounding area to pressurized hydraulic fluid. As a rule never stand within a 5 foot radius of the system while it's pressurized, the highly pressurized hydraulic fluid can easily penetrate skin and cause sever damage and infection, resulting in surgery. Another injury that could occur by getting to close to the loading arm during operation is getting caught between two moving pieces of metal.