UNIVERSITY OF OKLAHOMA

GRADUATE COLLEGE

THE DESIGN OF AN EXPERIMENTAL TESTING FIXTURE TO TEST SEALS OF VARIOUS GEOMETRIES EXPOSED TO EXTREME OPERATIONAL CONDITIONS WHILE OPERATING IN DYNAMIC RECTILINEAR MOTION

A THESIS

SUBMITTED TO THE GRADUATE FACULTY

in partial fulfillment of the requirements for the

Degree of

MASTER OF SCIENCE

By

Jon Keegan

Norman, Oklahoma

2023

THE DESIGN OF AN EXPERIMENTAL TESTING FIXTURE TO TEST SEALS OF VARIOUS GEOMETRIES EXPOSED TO EXTREME OPERATIONAL CONDITIONS WHILE OPERATING IN DYNAMIC RECTILINEAR MOTION

A THESIS APPROVED FOR THE
SCHOOL OF AEROSPACE AND MECHANICAL ENGINEERING

BY THE COMMITTEE CONSISTING OF

Dr. Zahed Siddique, Chair

Dr. Yingtao Liu

Dr. Shivakumar Raman

ACKNOWLEDGMENTS

I would like to start by thanking Dr. Zahed Siddique, my research advisor. We have known each other for many years, and I have always appreciated his patience and knowledge. Dr. Siddique, your support helped me to achieve all my academic goals and I will always be thankful for that.

To my fellow mates, Brandon and Richard, you guys gave me such trouble both good and bad. We had some fun times and some very stressful times. Nonetheless, we always were able to function as a cohesive bunch, mostly 85% efficient. Thank the two of you for all the talks, beverages, cigars, and conversations.

Dad and Sister, you guys were annoying. Always asking when I was going to complete this thesis, and never letting me forget it. Thank you for guilting me into finally bringing this chapter in life to an end. In all seriousness, thanks, the anxiety you two pressed onto me not only kept me awake at night, but also forced me to look at the amount of work I have done while in grad school and made me realize that this is just a paper.

A great big thank you to some of the AME staff, Bethany, Billy, Greg, and Melissa. The bunch of you really made my undergraduate and graduate years manageable. The amount of work all of you have done cannot be overlooked and I sincerely appreciate everything.

Lastly, Olga Umanskaya, we worked closely all through grad school. I was always happy to help you with your project alongside Adam. We had some very interesting adventures on North Campus. You defended your thesis and a few days after you made me promise to complete mine. Well, promise made. I value every conversation we had and have had since then. Thank you, Olga.

TABLE OF CONTENTS

LIST OF FIGURES

ABSTRACT

Radial reciprocating seals are available in a multitude of materials and physical configurations. The most used seal configuration is a circular cross section polymer ring shaped seal, O-ring seal. While affordable and small in cost, seals are essential to the functionality of many mechanical direct to consumer purchased products and industrial implements. Seal design and material selection is essential to the performance and functionality of the seal. With the multitude of different seal configurations, quantifying seal performance characteristics remains difficult. Efforts taken by the manufacturer to publish technical data of seals often only tailors to a very specific application. Being that a seals performance relies highly on parameters such as pressure, compression, lubrication, and application specifics such as remaining static or sealing dynamically, some means of classifying sealing behavior needs development.

In this thesis an outline of the eight-phase morphology of design process will be employed to develop a means of identifying inefficiencies and achievements of varying seal architectures used for seals in components of dynamic rectilinear motion. With this design process an experimental testing fixture will be designed and commissioned. This fixture serves as a device to answer research questions revolving around the testing of seals experiencing the exposure to assorted operational environments. Performance data of interest will be recorded in real time and then analyzed to give a statistical identifier of seal efficiencies or inefficiencies.

CHAPTER 1: INTRODUCTION

1.1 PROBLEM STATEMENT

In industry it is often desired to know physical performance characteristics of a designed product before deployment of that product into a system. A good measure of performance is the closeness of correlation between simulated analytical and experimental data. Often gathering the simulated data is difficult and costly, therefore, if the decision is made to proceed with real-world testing, a high refinement to the overall holistic view of the design process needs to be considered.

In very specialized situations an equally as specialized testing procedure and testing fixture needs to be thoroughly thought out. In order for these things to be attained, some detail needs to be brought forward about the overall end objective and goal. The following question was brought forth by the industry sponsor and principal investor:

Can an experimental testing fixture be designed and developed to quantify the operating performance characteristics of reciprocating rod seals while being exposed to varying harsh operational conditions? Can this experimental testing fixture be safely commissioned to expose seals to extreme high pressures while assuring operational safety?

System was required by sponsor to meet the following key operational parameters:

- High-pressure hydraulic fluid boosting capabilities 0-30ksi with resolution of +/- 100psi.
- System is required to operate at critical temperatures ranging from min -40 degF to max 400 degF.
- Force required to move reciprocating rod must be measured and attain a resolution of +/- 10lbf.

- Reciprocating rod speed must be accurately controlled. Minimum linear speed set at 1 cycle/hr and maximum linear speed set at 1 cycle/min.

1.2 RESEARCH OBJECTIVES

In order to address the research questions brought forth in section 1.1 this thesis will focus on the design and building of an experimental testing fixture to test the sealing performance characteristics of dynamic radial reciprocating seals. Scope of this paper is to describe in detail the decision making process undertaken in the conceptualization, design, and development process involved in the determination of actions taken to achieve the primary investors reach goals. A design of an experimental testing fixture should be outlined and agreed upon by all parties involved. Proofing this proposed design will narrow the vision of directions moving forward and will ultimately determine key components to accomplish research goals. Completing component selections, the preliminary design process will begin followed by the critical design review and ultimately concluded with the build and commissioning of the electromechanical machine seal testing fixture.

(Pressure Variation)

- Iteration 1
  - Cycle time: 1 Reciprocation/Min
  - Pressure: 15000 psi
  - Temperature: Ambient
- Iteration 2
  - Cycle time: 1 Reciprocation/Min
  - Pressure: 500 psi
  - Temperature: Ambient

(Cycle Time Iterations)

- Iteration 3
  - Cycle time: .1 Reciprocation/Min
  - Pressure: 15000 psi
  - Temperature: Ambient
- Iteration 4
  - Cycle time: 0.1 Reciprocation/Min
  - Pressure: 500 psi
  - Temperature: Ambient

(At Temperature)

- Iteration 5
  - Cycle time: 1 Reciprocation/Min
  - Pressure: 15000 psi
  - Temperature: 150DegC
- Iteration 6
  - Cycle time: 1 Reciprocation/Min
  - Pressure: 500 psi
  - Temperature: 150DegC
- Iteration 7
  - Cycle time: 0.1 Reciprocation/Min
  - Pressure: 15000 psi
  - Temperature: 150DegC
- Iteration 8
  - Cycle time: 0.1 Reciprocation/Min
  - Pressure: 500 psi
  - Temperature: 150DegC

(Dwell Time Iterations

- Iteration 9
  - Cycle time: 1 Reciprocation/Min
  - Pressure: 15000 psi
  - Temperature: Ambient
  - Dwell Time: 5 minutes @ 1000Cycles
- Iteration 10
  - Cycle time: 1 Reciprocation/Min
  - Pressure: 500 psi
  - Temperature: Ambient
  - Dwell Time: 5 minutes @ 1000Cycles
- Iteration 11
  - Cycle time: 0.1 RPM Reciprocation/Min
  - Pressure: 15000 psi
  - Temperature: Ambient
  - Dwell Time: 5 minutes @ 1000Cycles
- Iteration 12
  - Cycle time: 0.1 RPM Reciprocation/Min
  - Pressure: 500 psi
  - Temperature: Ambient
  - Dwell Time: 5 minutes @ 1000Cycles

1.3 OVERVIEW OF APPROACH

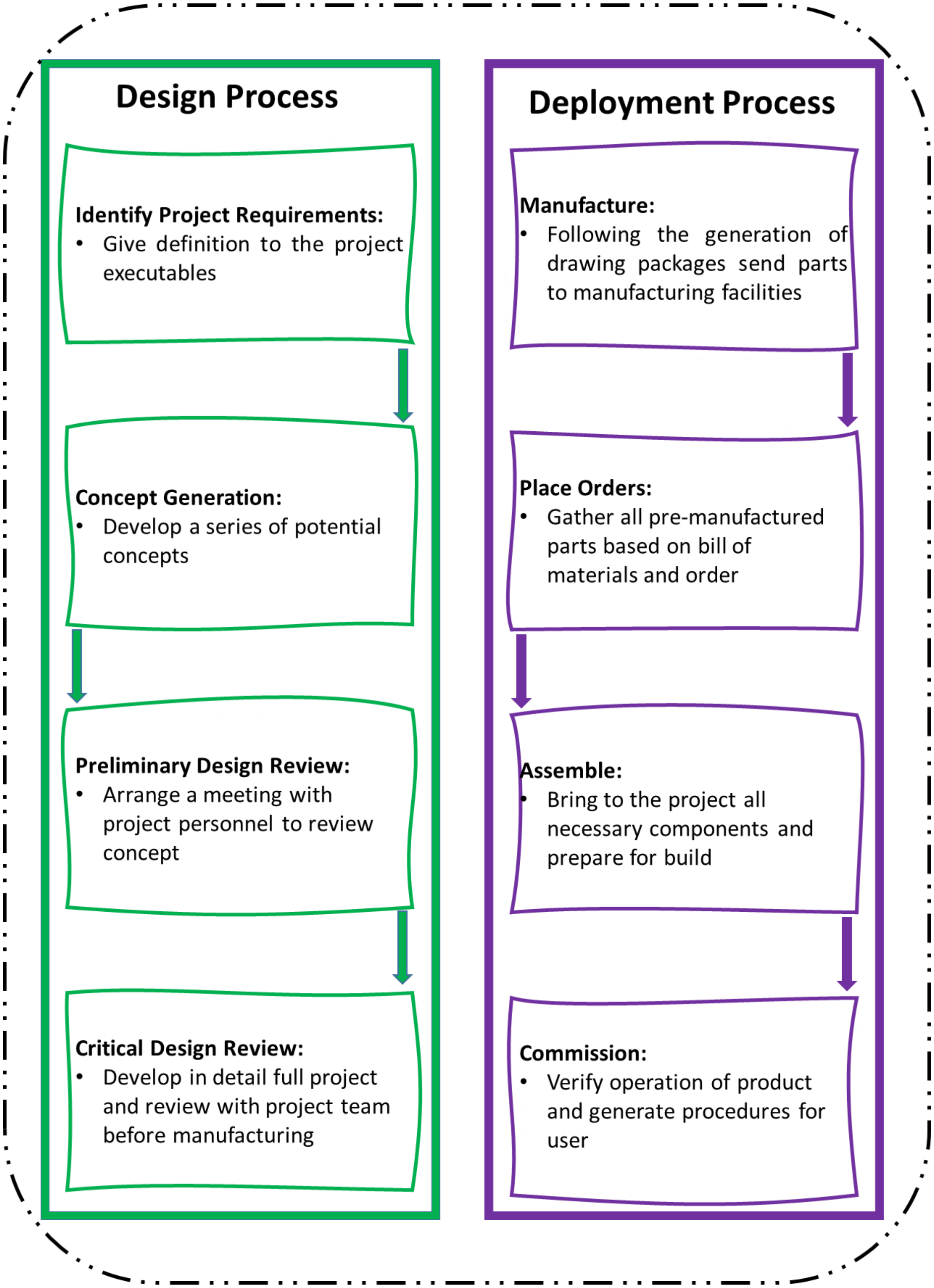


Figure 1: Graphical representation showing thesis overview of approach.

The overview of approach map in Figure 1 breaks the research project into two aspects, design, and deployment. These processes are represented as serial and executed stepwise. Of the two processes the most special attention should be given to the critical design review as this juncture breaks the barrier between analytical and physical.

Presented with a description of the project and understanding the project objectives, iterate the requirements among principal investor and project team. The concreting of mutually understood executables keeps parties involved synchronized. Using the project definition, a series of rough concepts were drafted and presented in a preliminary design review with sponsors.

Design reviews aide in keeping the project on track and act to help create feedback opportunities do drive project to its milestone. The most important review would be the critical design review process. At this time the project is well defined and a concept has been decided upon. This process can easily derail a project, therefore, significant time should be allotted for open communication.

Beginning the deployment process a couple of steps can be executed in parallel. The sending of engineered parts to manufacturing facilities and gathering of off the shelf products where applicable. This process generally takes time due to lead time on items. In this interim time attack paths moving forward were generated to keep project flow.

After performing quality control on manufactured parts and after receiving of pre-made items the assembly process begins. Attention to detail in assembly will directly influence the commissioning process. Product commissioning concludes the process map. Commissioning is verified when all functions and subsystems are functional. Procedural steps are created then the product is taken to final inspection before deployment.

1.4 OUTLINE OF THESIS

Discussions in this section will illustrate a holistic view of thesis chapters. Problem statement and research objectives will be reviewed and have been introduced in preceding sections.

Chapter1 describes the problem and research objectives. In this chapter a problem statement is defined and an overview of approach given. The design and deployment process are graphically represented for clarity then are defined in more detail.

Chapter 2 outlines the eight-tier morphology of design process. This process consists of two large groupings broken down into more detailed action items. The large bounding categories are the conceptual phase of design, and the second category relates to design embodiment. Within the conceptual phase, problem definition, information gathering, concept generation, and concept evaluation are shown. The embodiment design phase is the most time consuming and detailed. In this phase component selections are made, and product architecture is described before detailing and commissioning the product.

Chapter 3 shows in detail the last part of the design embodiment, the detail design process. Concept generation is finalized, and functionality analyzed. Finite element analysis is utilized to promote data driven decision making about the viability of designed components. Pre-manufactures items are chosen to accomplish research objectives and reasoning behind choices is given in this chapter. Final deployment and commissioning shows project completion.

Chapter 4 presents ways in which the product mains safe operation. Engineered components are conferred with behavior of operation and purpose. Summary of flow loop regime and importance of high-pressure fluid flow interrupters. Caution is given to operational procedures and best user practices presented.

Chapter 5 concludes the thesis. Research obstacles are presented in this chapter. Most issues arose with miscommunications between manufacturing facilities, parts arrived not within specifications listed on prints and needed rework. Project future recommendations are given such as addition of a pulsation dampening device to reduce hydraulic fluid hammer. This dampener also serves to increase the overall fluid volume held pressurized in the system. Addition of a linear positioning sensor affixed to the ball screw jack neck could allow real time position readout of intensifier piston allowing a more accurate way of monitoring fluid loss.

1.5 Introduction To Reciprocating Seals

Seals fall into three principal categories. Seals which do not move and remain stationary, seals that capture rotating equipment, and seals that are used in linear motion. Reciprocating seals are a part of the last category where the seal functions in directional linear movement, Figure 2.

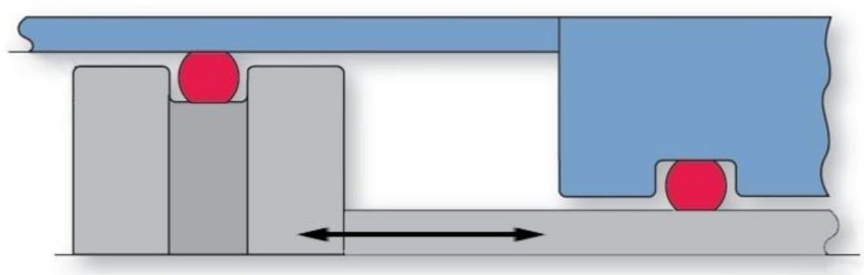


Figure 2: Reciprocating rod/shaft seal[1].

Mechanics of interest are isolated to the reciprocating/shuffling motion of a piston or rod within a stationary bore. There exist two common methods of sealing fluids within dynamic reciprocating applications, piston seals and rod seals. Rod seals lie within a machined seal gland located on the inside of a bore. These types of seals remain stationary while a rod travels linearly across the seals inner diameter, right side of Figure 2. Alternatively, piston seals move in unison with the rod. Piston seals are constrained in a machined gland located on the peripheral of a rod. These seals utilize their outer diameter to seal against a stationary bore, left side of Figure 2.

One of the primary failure methods for circular cross section seals results from improper tolerancing of the piston outer diameter and the bore internal diameter. Lose fitment allows for an increase in diametral clearances between these parts. This tolerance stack up allows seals under pressure to deform and pass into the gap, Figure 3

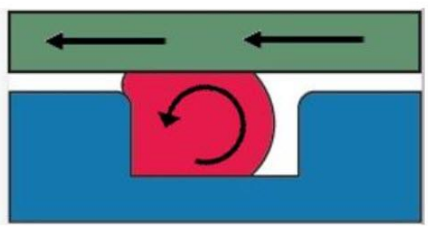


Figure 3: Seal extrusion gap with rolling[1].

Resulting from this action the seal begins to get pinched on both the compression and tension stroke. Ultimately the seal will either roll or tear failing potentially causing machine damage. Suggestion is to maintain an extrusion gap less than five thousandths of an inch or less for all pressure ranges above five hundred psi. Sealing is difficult at low pressures as deformation of seal is minimal. With proper seal gland design deformation can be controlled improving sealing capability. There are a few fundamentals to dynamic sealing and are as follows:

Friction

To lower friction:

- Change lubrication

- Alter gland to result in less percent squeeze
- Change speed of reciprocation
- Reduce seal to rod surface area contact
- Reduce surface finish contacting the moving wall

Wear

Degradation of seals are a product of the following:

- Chosen fluid
- Surface finish of gland
- Surface finish of moving wall
- Seal compression

Sealability

To improve sealability:

- Increase pressure
- Increase seal to moving wall surface area
- Decrease surface finish of gland
- Decrease surface finish of moving wall

Extrusion

To decrease possibility of seal extruding:

- Change material stiffness
- Decrease extrusion gap
- Decrease pressure

The O-ring seal gland is designed and dimensioned so that, when a seal is installed a percentage of deformable compression is present. This percentage of compression typically resides in the %5-%20 range for dynamic applications and can be as high as %30 in static applications. Gland grooves are suggested to be designed with 5deg sloping walls and all corners broke with a small radius. Generally, a surface finish greater than 32 micron inch is undesirable and will cause sealing failure.

CHAPTER 2: DESIGN PROCESS

2.1 INTRODUCTION

Specialized design projects such as this one as well as many others spanning a multitude of different engineering sectors require the understanding of an execution process and how to keep oneself or team accountable through the development of a product. There exist a few different adopted organizational schemes to outline the design process. Schemes include the Universal Design Theory proposed by Grabowski, Rodenacker’s function-based design methodology, and Roth’s categorization of elements based on classification of function [2, 3]. The scope of this research will be taking a slightly more generalized direction as the project design process needs to adhere to a full scope holistic design overview. The direction of this project will use the universally accepted eight-step design process proposed in the 1960’s by Morris Asimow. This design process is outlined in Figure 4. This chapter overviews the conceptual, embodiment and detail design process surrounding the design of the experimental setup to test seals exposed to extreme operational conditions while operating in rectilinear motion.

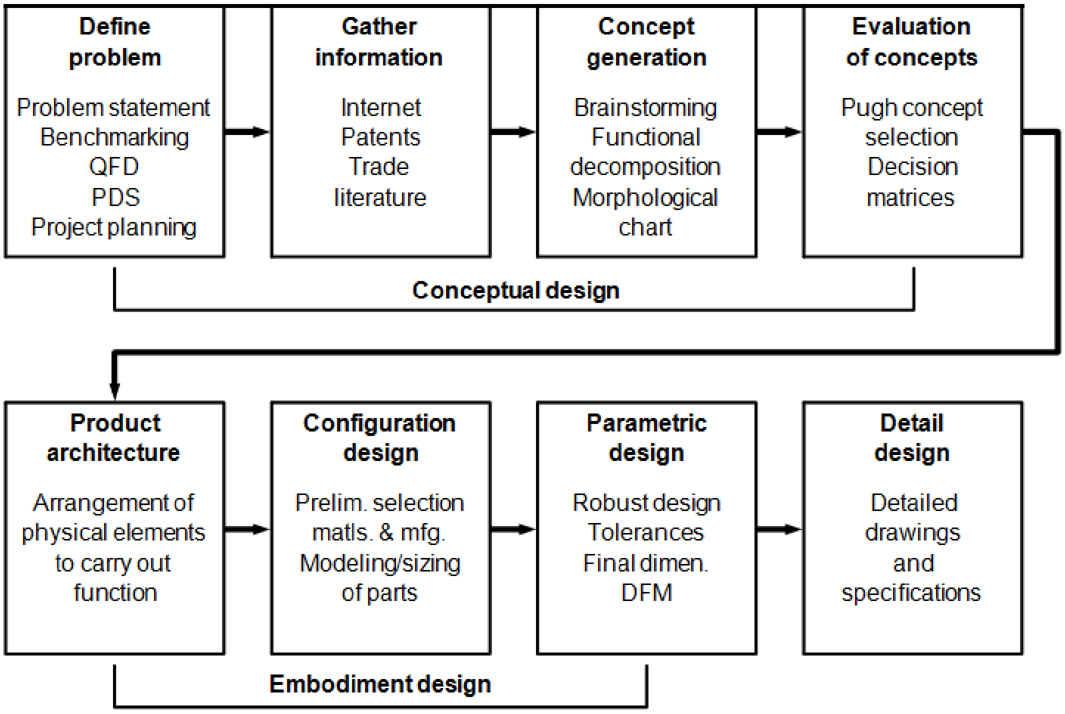


Figure 4: Morris Asimow eight phase morphology of design[3].

2.2 CONCEPTUAL DESIGN

The conceptual design phase consists of assessing what precisely the principal investor requires, what are the desired outcomes and what approaches can be taken to accomplish the desired outcome for the project. Conceptual phase should consist of risk assessment, tentative timelines, and benchmarks[4]. Within this phase there should be a generation of a few possible solutions to the select problem, it is imperative here to open a means of open-door communication with the sponsor to generate ideas and understand concessions by all who are involved. The conceptual phase remains the most fluid of all phases and consequentially requires the most creativity and flexibility. In this section the design process will be broken down into action items associated with the eight-step design morphology process.

Conceptual Action Item 1: Identify Project Requirements

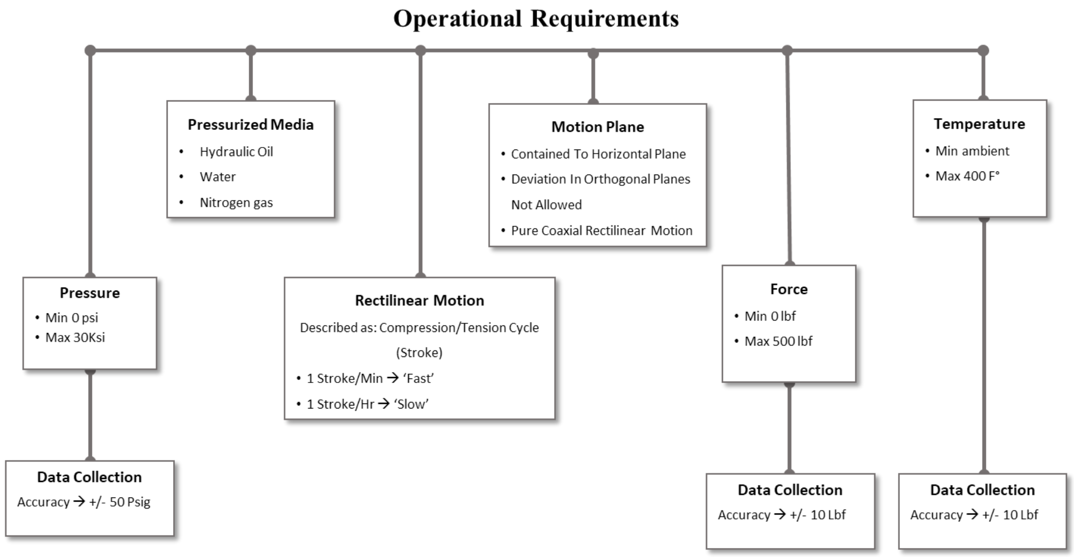


Figure 5: Key deliverables required by project principal investor.

Essential to the beginning of any project, there must be a clearly understood bounding box to understand what is needed and what can be accomplished within the timeframe set forth. The earliest stages of the project initiation involve a series of discussions with the principal investor, where within each iterative meeting the project scope becomes more concrete. Resulting from these discussions it is not uncommon for needs and desires to change, Figure 5 gives a visual representation of the resulting requirements decided upon giving this project a clear prospective while moving forward. These product design specifications illustrated above give direction for the remainder of the conceptualization phase whilst eventually transitioning to the definition stage and finally the detail stage where the product is brought to competition and commissioned.

Conceptual Action Item 2: Isolating Main Components

Building upon the knowledge obtained from action item one, the project now has an understood outline. The parameters and outputs of interest are now undergoing inspection and the direction moves now to develop a plan which outlines preferred methods of attaining these required critical aspects. Development of a project block diagram is essential to the conceptualization phase as it gives an idea as how key components need to interface with each other and how those components might occupy some fit/form factor allowing for a foundational understanding of the project scope.

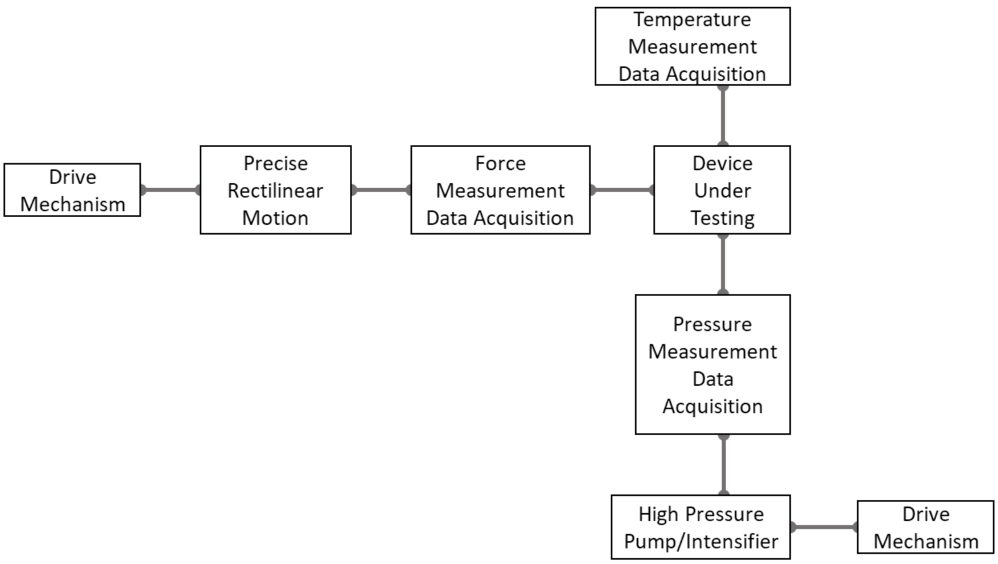


Figure 6: Projected component flow.

During the development of the project block diagram some questions of interest arose. Those questions circled around ideas of joining two separate subsystems or subassemblies into the overall design and build. In order to accomplish some form of precise rectilinear motion to test the spring energized bore seals, and in order to create a large pressure gradient for the seals to operate under, the thought of developing two distinctly different subsystems arose. To accomplish the required linear motion for the reciprocating rod a reciprocating rod bed assembly could be created, and to accomplish high pressure intensification, a separate subsystem. Each subsystem would have the ability to operate independently which allows for some future proofing and adaptability for future changes to the device under testing, i.e. different seal geometries and vessel configurations.

Conceptual Action Item 3: Proposed Component Selection

Readily available premanufactured items are the preferred means of accomplishing key operational requirements as depicted in Figure 5, however, with the specificity of these requirements an almost entirely organic restructuring of design and execution might need to be accepted.

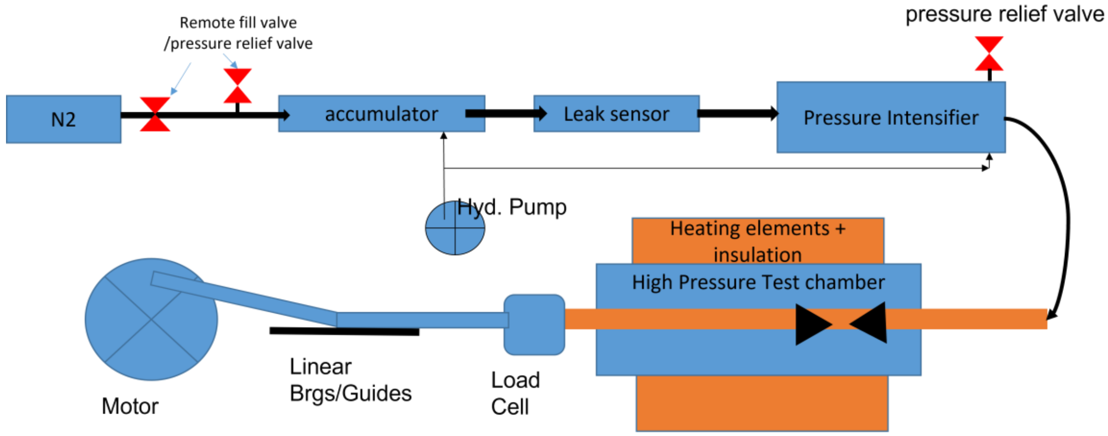


Figure 7: Conceptual design on using mostly off shelf items.

Keeping clear of an entirely foundation up custom component build a couple of design iterations were developed. Collaborating with the principal investor group led to a design containing a series of off-shelf items. Figure 7 gives one example of utilizing some common industrial components and draws a rough idea of potential steps forward. This version would use a compressed nitrogen tank to apply pressure to a hydraulic accumulator, the accumulator then pushes highly pressurized hydraulic fluid upstream through a leak rate sensor, pressure intensifier, and ending into the device under testing. Linear motion would be accomplished using a cam and arm attached to a linear guide rail carriage. The carriage mounts to a load cell and the pair push and pull the rod through the pressurized seal housing.

Off the shelf mech. Items	Electronic components	Machined parts
High pressure hydraulic pump, fluid reservoir, autoclave valves/fittings/tubing, accumulator, pressure gage (1), relief valve, rails, linear bearings, pressure intensifier, gear reducer, cart, …	Larger HP motor & motor controller, NI DAQ with >10 input lines & >4 output for pressure/temp & for heater/motor/pump control lines, load cell, pressure transducer (2), heater bands and controller, leak sensor, solenoid valves (3),…	Machined drive components, base plate, clamps,…

Figure 8: Item availability low pressure cam arm drive concept.

This build contains mostly readily available components, Figure 8, but also lacks some of the primary investors required metrics. This build would only be able to produce pressures maxing at 10ksi, and speed selection does not have the control desired.

Reassessing the projects deliverables, the primary investor was not prepared to sacrifice the important metrics such as speed control and high-pressure utility. Discussions about directions forward were taken and as a result the idea of two separate subsystems became desirable. One subsystem to control the pressurized media and a separate system to handle the rectilinear motion of the reciprocating rod as it completes its compression/tension cycles.

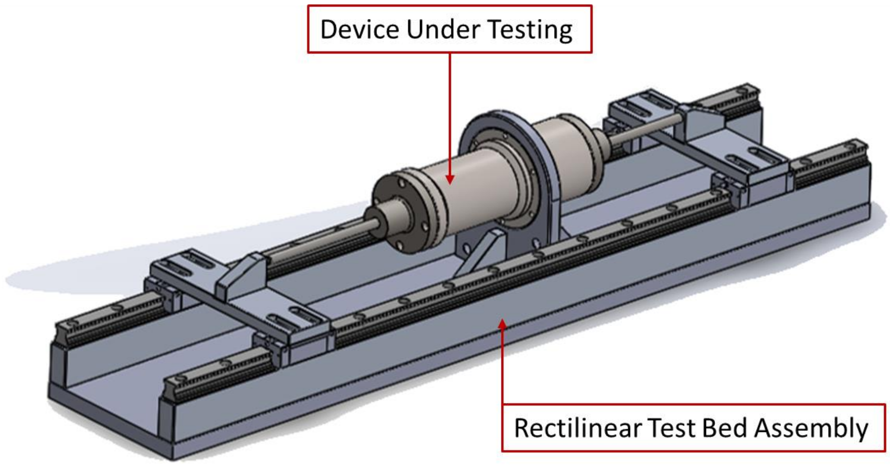


Figure 9: Concept proof of a completely constrained rectilinear bed and device under testing.

Following a few derivations of potential bed assemblies, a mutually accepted solution was found. A rough concept proof of the test bed can be seen in Figure 9. The benefits to this solution include a fully constrained rod while it travels through the stroke cycle. The rod here would remain perfectly parallel to the test bed as well as perfectly parallel to the plane orthogonal to the test bed. Two variants to the drive selection were suggested. Variant one would utilize a cam arm connector driven by an ac induction motor controlled by a variable frequency drive. Variant two uses a servo or stepper motor to drive a linear ball screw.

From these selections we needed to look further into the critical requirements of the project and understand more about what is physically more feasible to implement.

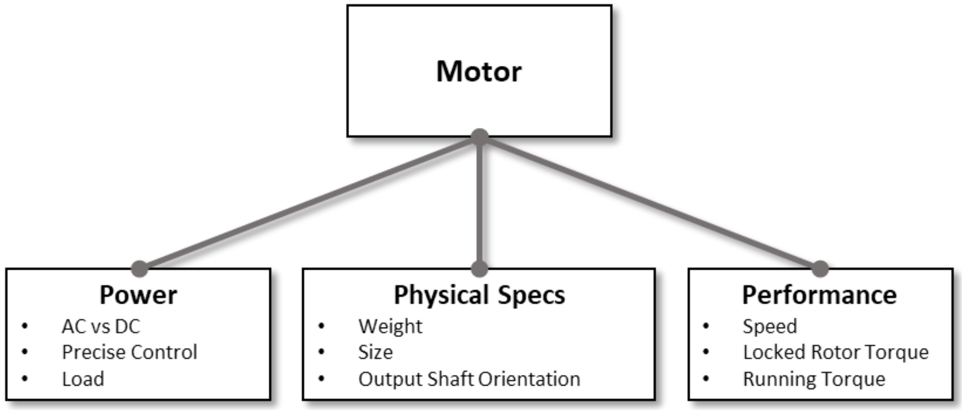


Figure 10: Drive motor variant selection.

The specifics as to which variant is best suited for our process will be outlined in more depth in a later section of this chapter. For transparency we will introduce Figure 10 here as the fluidity of the conceptualization process makes the designer take note of what avenues are open to them to accomplish a specific task.

The resulting solution to the pressure intensification problem developed as discussions were made with high pressure pump manufacturers. Typically, high pressure hydraulic systems operate in some sort of a closed-circuit application and the pumps are constantly running in a loop. The purpose of pressurization in this system is to energize seals which allows force measurements to be taken. Transient effects produced by pumps engaging and disengaging is undesirable under our testing criteria.

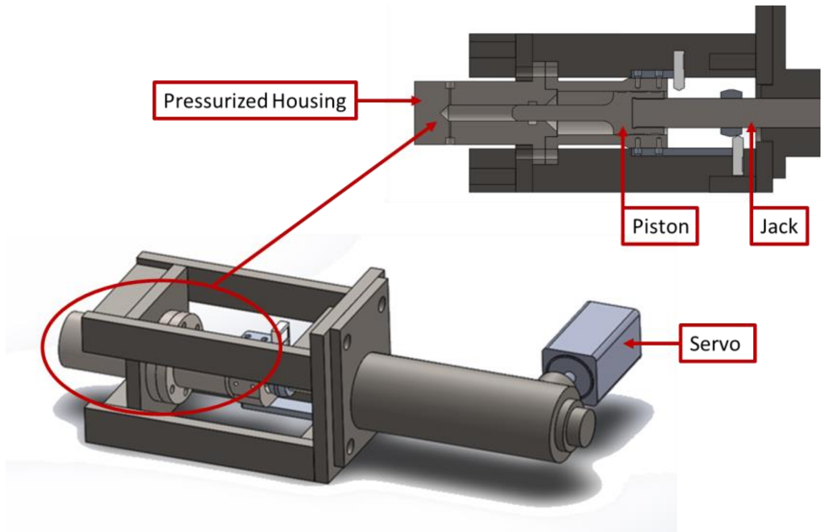


Figure 11: Concept poof pressure intensifier.

The proposed solution to the pressure intensification problem can be seen in Figure 11. The concept proof acts as a single stage reciprocating pump. The idea is to simply use a piston to compress fluid in a blind bore chamber. This fluid, which is being compressed by the piston, driven by a strong actuator such as a jack or linear actuator, then travels upstream and reaches the device under testing to energize and lubricate the seals.

Before moving onto the embodiment design phase, a critical design review was performed. In this review action items such as scheduling, feasibility, and commitment to critical values were discussed. The design team and the primary investors found common ground on the project being economically worthwhile and executable within the given budget and timeframe. Product architecture, where the overall system is broken into subsystems will be developed in unison with milestones during the definition phase of the morphology design process.

2.3. EMBODIMENT DESIGN

The embodiment design phase is sometimes referred to as the preliminary design phase or the system level design phase. In this phase the architecture of the design project is broken into more understandable sub-assemblies and preliminary component selections. Critical design items such as product architecture and design configurations are a result of this embodiment design phase[5]. In this phase selections will be made and proposed best design will be placed into the detail design phase. Any changes in this phase will detrimentally influence all remaining phases of the project.

Embodiment Design Action Item 1: Architecture

Continuing from the conceptual design process, the destination and how to get there is now better known. The concept is well outlined, and knowledge of key requirements is understood. The subsystem and component level architecture needs to be outlined before moving to the detail design phase. Moving from a macro to a more microscopic view of the system we are dissolving certain assemblies into sub-assemblies as seen in Figure 12, Figure 13, and Figure 14.

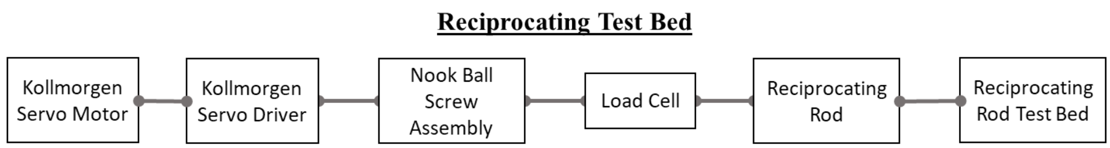


Figure 12: Subcomponent architecture reciprocating rod test bed.

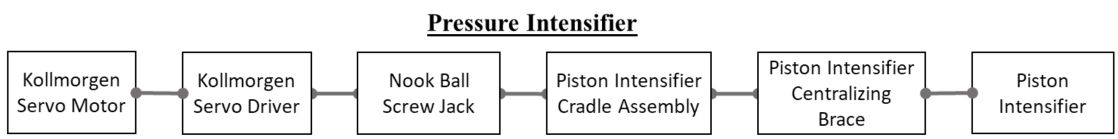


Figure 13: Subcomponent architecture Pressure intensifier.

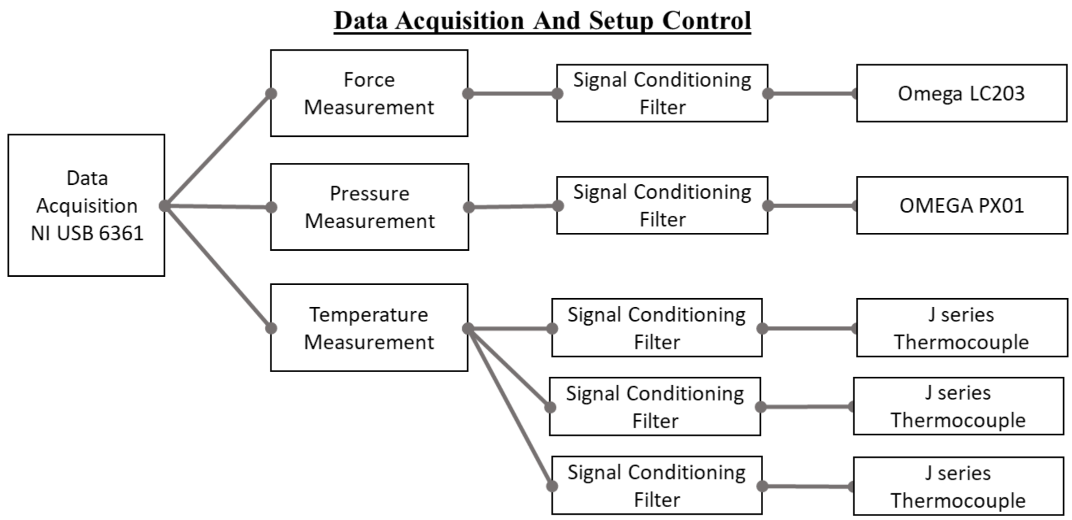


Figure 14: Subcomponent architecture measurement and automation.

Dissecting the preliminary designs produced from the conceptual design process leads to a more robust understanding of sub system architecture. To produce the reciprocating rod rectilinear motion, a servo controlled linear ball screw assembly will be implemented and will be powered by a Kollmorgen servo motor. The reciprocating rod will need to have a load cell on the driven end of the rod. The pressure intensifier will not be able to be purchased off the shelf therefore a custom build is necessary. The intensifier will take the form of a single ended dead bore piston. The linear motion of the piston will come from a ball screw jack powered by a Kollmorgen servo motor, terminating in a specialty designed pressure vessel. Data Acquisition and motion control will be accomplished by a national instrument’s multifunction data acquisition device. Recorded signals of interest require signal conditioning filters and channels of consideration are force, pressure, and temperature as specified in Figure 5.

Embodiment Design Action Item 2: Risk Assessment And Milestones

Assessing the unknowns and potential risks of a design are critical before moving onto the project execution. These inherent risks can significantly degrade the progress of a project and need to be known beforehand. Looking back at Figure 12, Figure 13, and Figure 14 of the architecture break down we can categorize component selections within subsystems and analyze their inherent risk factor. When assessing risk factors, time will be the sole metric. Main risk categories are listed in figure X an the evaluation criteria is assessed on a scale of one to five, one being the lowest risk and five being the highest assessed risk.

Inherent Risks and Risk Assessment

Component	Availability/Lead Time/Comittment	Risk
Kollmorgen (Servos, Drives, Cabling)	(2-4 Weeks)	High==4
Nook (Ball Screw Jack, Linear Ball Screw Assembly)	(3-5 Business Days)	Low==2
Machined Components (Device under testing, Intensifier, etc.)	(4-6 Weeks)	High==4
Water-Jet Components (Cradle Components)	(2-3 Business Days)	Low==2
Weldments (Frame, Cradle Tension Bars, Electrical Enclosure)	(1 Week)	Medium==3
Powder Coated Components (Cradle Assembly)	(2 Weeks)	Medium==3
Heat Treat (Intensifier Pressure Housing & Piston)	(2 Business Days)	Low==2
Instrumentation (Transducers, Signal Conditioners)	(3 Business Days)	Low==2
Integration (Mechanical Assembly And Testing)	(8-10 Weeks)	High==5
Fabricated Equipment (Nuts, Bolts, Wire, Valves, Piping)	(Next Day Delivery)	Low==1
National Instruments (Data Acquisition, Power Modules, Relays)	(7-10 Business Days)	Medium==3
Electrical Enclosure (Organization And Wire Routing)	(4-6 Weeks)	High==4
Coding (Coding For Stepper Motors, Servo Motors, And LabVIEW)	(6+ Weeks)	High==5

Figure 15: Project deliverables and connected risk values.

After assessing project risks the time commitment is better understood and it is devised that the instances of highest risk would be time allocated to coding and automation/integration of the system. Uncovering these risks early into the project allows clear milestones to present themselves. Critical areas are given additional support as needed, and easily executable tasks are organized to grant efficient project throughput.

2.4 CHAPTER SUMMARY

In review, chapter three gave a summary of design phases essential to proper technical execution of the ideation and commissioning of a specialized and unique testing apparatus which characterizes the performance characteristics of radial seals operating in exaggerated wear conditions while operating in rectilinear motion. Through the conceptual and embodiment design phases, project needs were outlined and means of fulfilling those needs were presented and decided upon. Milestones and benchmarks are apparent in any large project and associated risk factors are essential to evaluate in an effort to keep project timelines intact. Architectural breakdown of the project gives understanding to key component selections and drives the decision making process of the detailed design portion of the eight tier morphology of design process. Within the detailed design definition type, a less holistic look at the project deliverables and details will be discussed. Specific component selection and discussions on why certain components will take place. Reviewing specific risk factors and evaluating potential unknowns allows resource designations and gives a clear travel forward. Chapter four will be an overview in more specific detail of the detailed design process.

CHAPTER 3: DESIGN AND FABRICATION PROCESS

3.1 INTRODUCTION

Specialized designs are often necessary to accomplish specialized needs, this chapter will explore the design and fabrication process connected to creating a testing fixture which accomplishes the needs described in preceding chapters. The commissioned assembly will be introduced and subassemblies will be defined in this chapter. Giving some depiction and acting as a prologue to the remainder of the design and fabrication process chapter a introduction to mechanical, electrical, and other associated functional components of the testing equipment can be seen in Figure 16, and Figure 17.

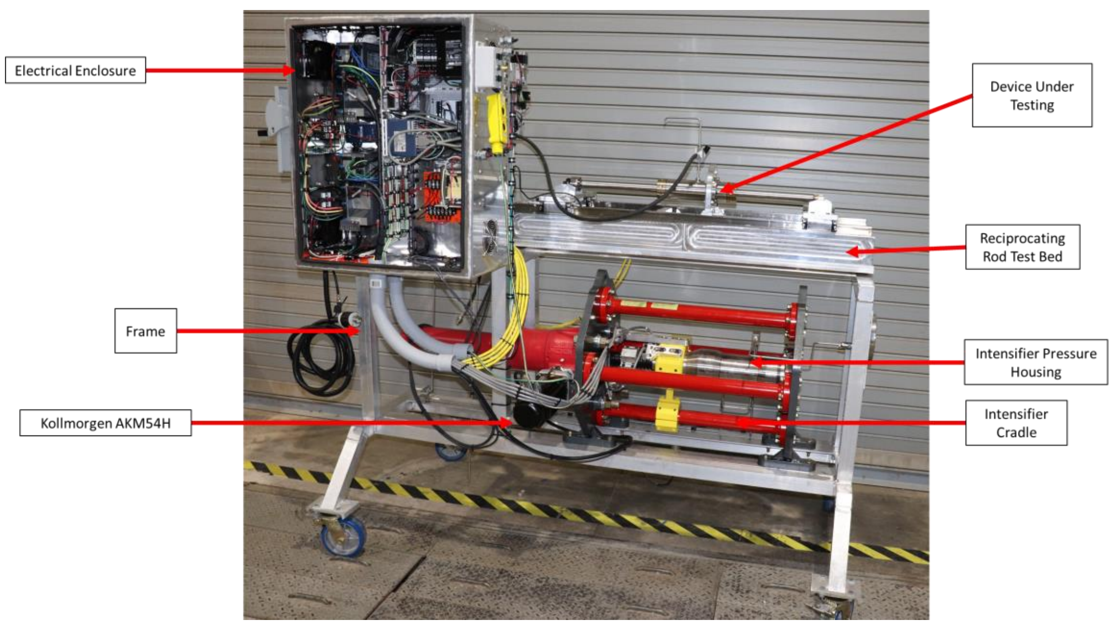


Figure 16: Rear view rectilinear seal testing fixture.

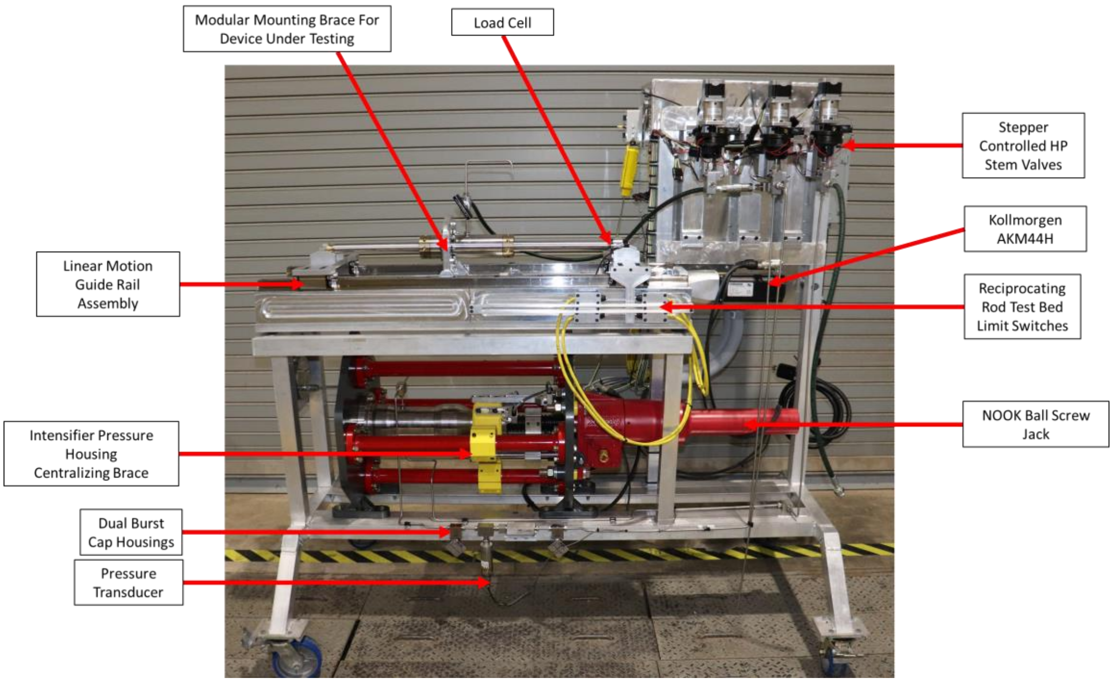


Figure 17: Front view rectilinear seal testing fixture.

The testing fixture consists of many components but is decidedly dissected into a handful of distinct subsystems. The main driven component remains the high-pressure intensifier and cradle assembly. Secondary driven component gives motion to a reciprocating rod which travels bidirectionally through a device under testing who holds either rod seals or piston seals. Internal geometries of the testing device are meant to be variable depending on the desired type of seal under testing. Large removable reciprocating rod test bed allows for modularity of system while keeping core functionality given by the intensifier, automation, piping and instrumentation layout.

3.2 PRESSURE INTENSIFIER SUBASSEMBLY

The pressure intensifier is a vital system component and is required to satisfy one of the primary objectives of this systems operation. Since no reasonable premade options presented themselves during the embodiment of design process outlined in chapter two, a engineered assembly became the foremost option. A rendering of the as-build subassembly can be seen in Figure 18.

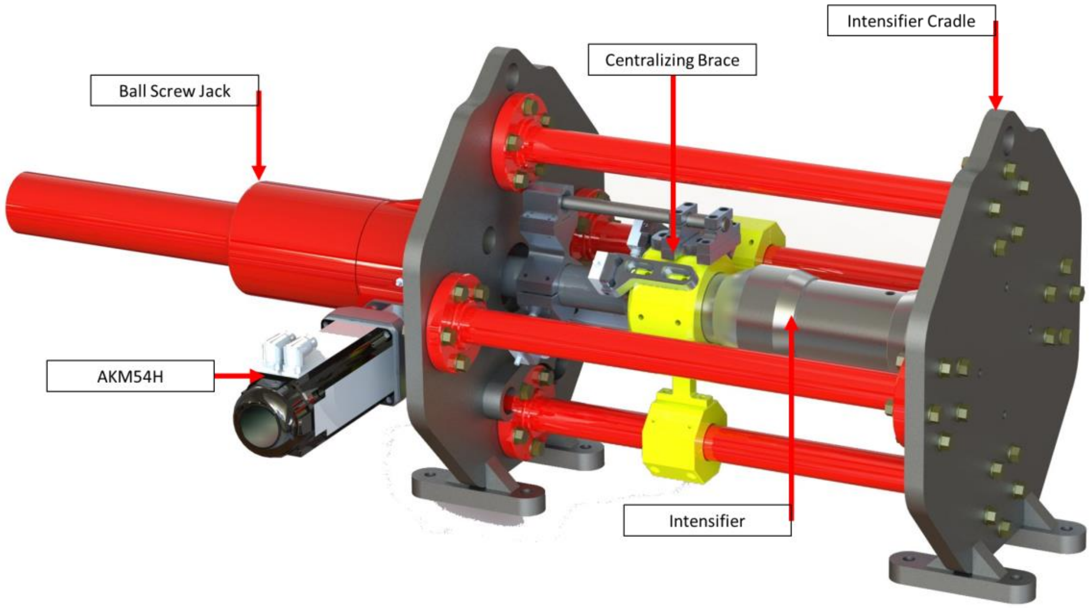


Figure 18: Intensifier cradle, pressure housing, centralizing brace assembly.

The intensifier subassembly consists of four key components. Primarily the pressure intensifier pressure housing which acts as the fluid supply to the downstream system. The fluid is pressured by using a blind bored casing and piston assembly. The power comes in the form of a ball screw jack driven by a servomotor. All these components are affied to the cradle/frame and are aligned by the intensifier centralizing brace. Piping and fluid routing is not shown here for image cleanliness. Piping and instrumentation will be outlined in a subsequent portion of this chapter.

3.2.1 Intensifier Pressure Vessel Housing Assembly

The pressure vessel housing assembly consists of seven discrete machined components. A cross sectional view of the assembled sub assembly can be found in Figure 19. Starting from the left the intensifier subassembly contains a mounting plate which allows either vertical or horizontal operating orientations depending on desired fit, form, and function. Intensifier bore plug and piston plug are machined from 4140 alloy steel which is a dissimilar material then that of the intensifier end cap, piston guide and primary pressure housing. The choice of material selection was determined by the need for threaded parts to resist thread galling and thread wear.

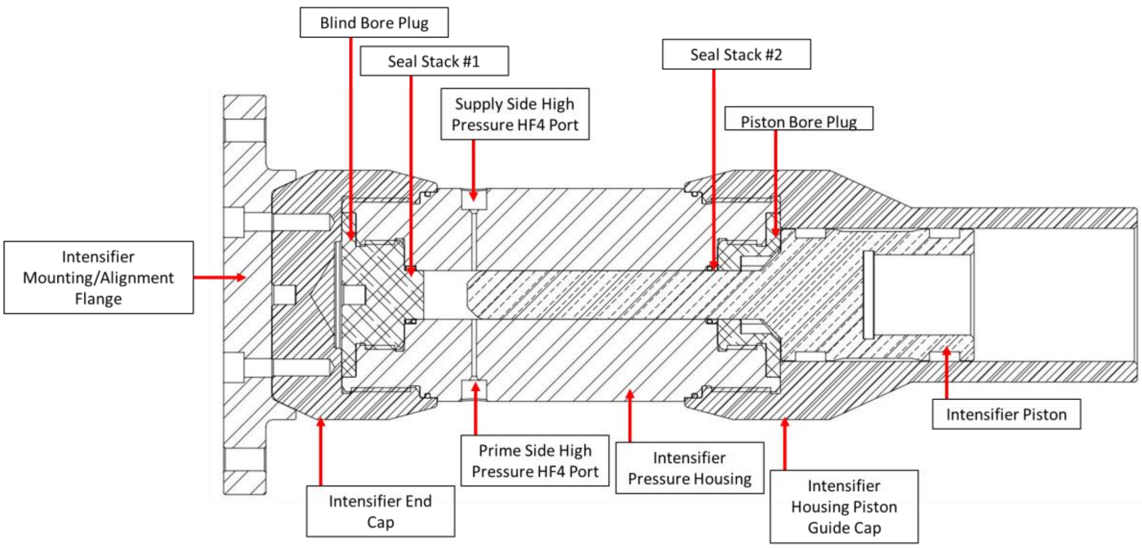


Figure 19: Cross section view of intensifier subassembly.

The main housing and piston were made from 17-4 ph balanced stainless steel and then heat treated to the H900 condition. For ease of assembly and disassembly all threaded machined parts were tolerance to a six-pitch stub acme. A through bore was adopted for ease of manufacturability and to allow for the use of a reaming tool to accommodate the low surface finish requirements. Drawings for reference can be found in the appendix. The means of creating high pressure is accomplished by exhibiting a force on the driven end of the intensifier piston. A demonstration of intensifier actuation and pressure boosting can be found in Figure 20.

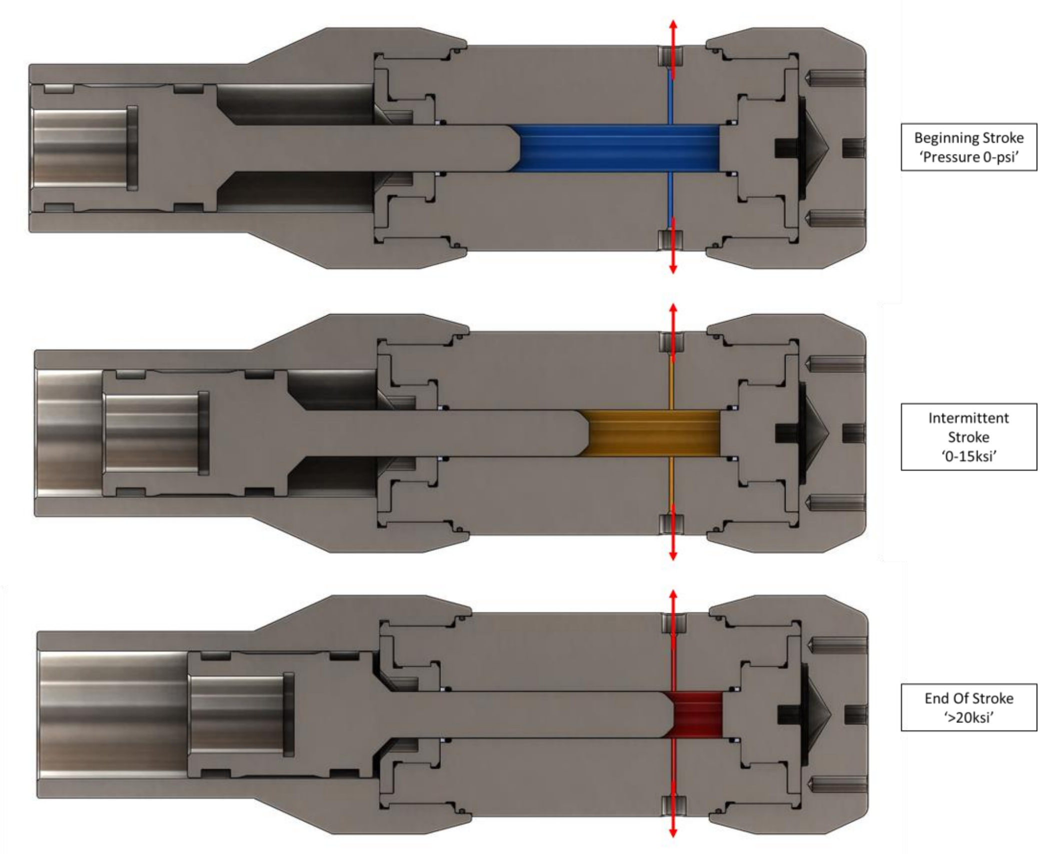


Figure 20: Intensifier actuation to boost pressure.

In order to achieve pressures above 15ksi the driving force required by the piston actuator needs to be known. Knowing the area of the piston and setting 30ksi as the desired max operating force we can determine piston force requirements.

$$F = \pi * .5^2 inch * 30000 lb/in\text{^}2$$

$$F = 23561 lbf$$

$$F = 104755 N$$

After determining force needed to attain downstream pressure and setting a design parameter of piston diameter equal to one inch, our bounding box for intensifier analysis can begin. Items of interest that present themselves and come forth with the accepted intensifier design direction are housing integrity, piston column buckling, sealing capability, and potential issues with possible thread shearing. For intensifier housing integrity and piston buckling we shall simulate exaggerated worst case scenario measures using finite element analysis software.

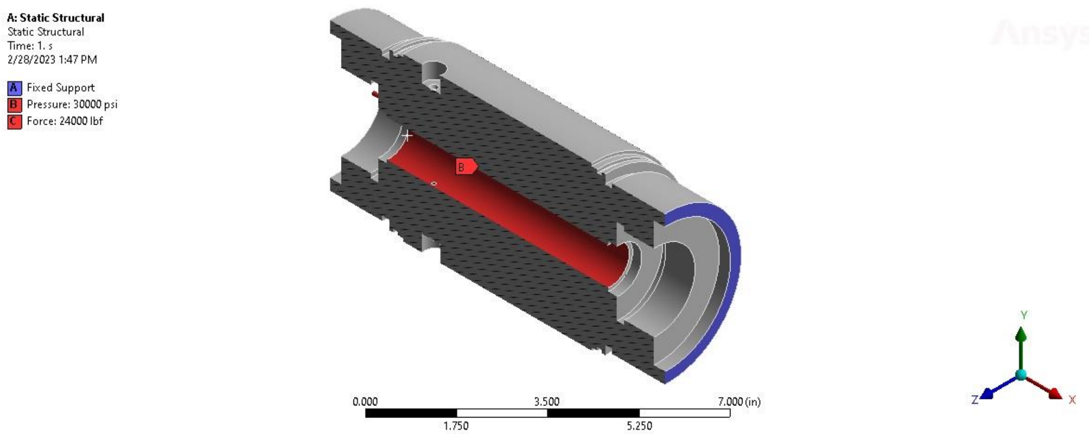


Figure 21: FEA setup of intensifier housing.

Simplifying model geometry allowed for a straightforward structural setup. Setting bounding conditions and simulating max operational pressure loads let to a more rigid understanding of the proposed design. Material setup used the mechanical material properties as seen in **Error! Reference source not found.**.

Figure 22: 17-4 PH balanced H900 condition material properties[6].

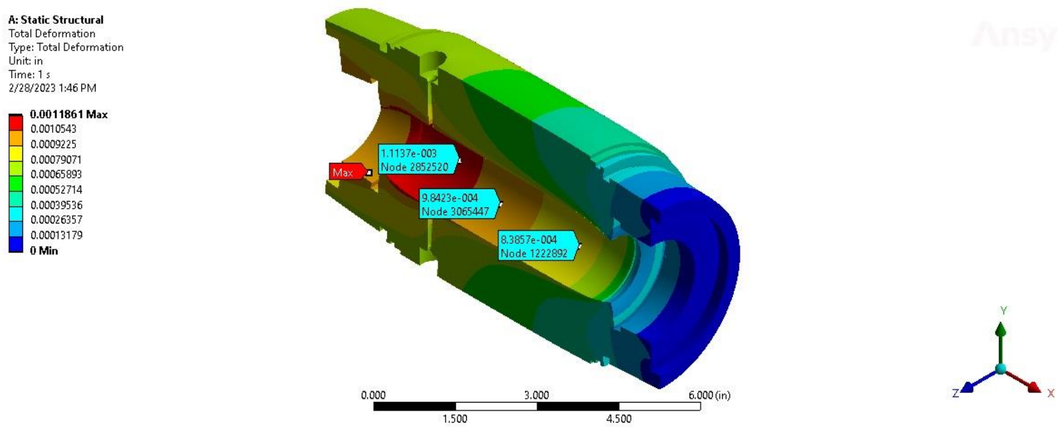


Figure 23: Deformation intensifier pressure housing.

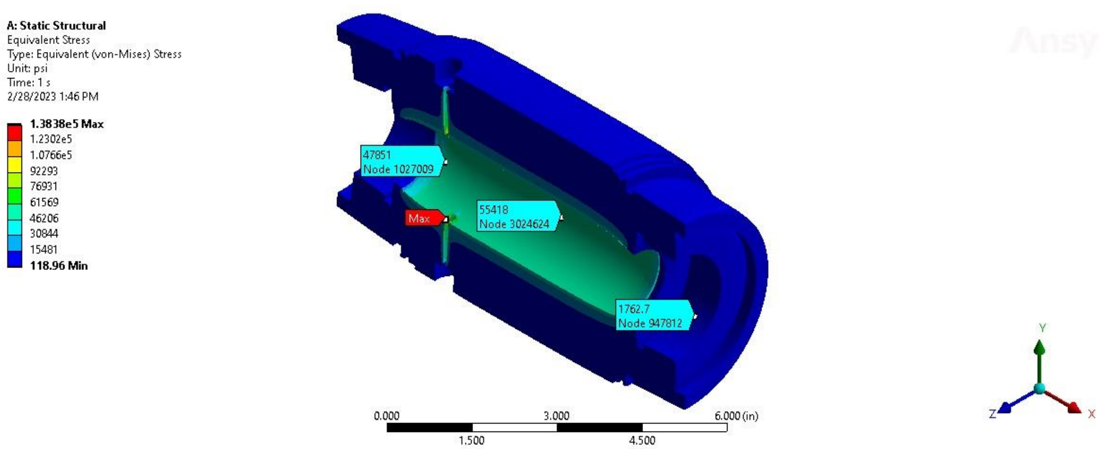


Figure 24: Equivalent von-mises stress.

Equivalent maximum deformation can be seen in Figure 23 and the location lies on the end of the bore opposing the fixed end boundary constraint. The deformation scale is small and can be considered negligible due to the knowing that additional parts of the overall assembly will add rigidity to the structure. The equivalent stress peaks around roughly 138ksi and is located at areas of mesh sharpness because of constraining geometries. Taking a more detailed look with probes we can average the stress in the operational zones to be less than 60ksi giving a safety factor of three to four times over expected operating pressures.

Mentioned before we had adopted a through chamber bore rather than a blind bore for the pressure chamber. This was a engineering choice because of the long and small diameter fluid chamber, a simple deep drilling process followed by a final reaming process allowed machine work to be significantly less expensive. This through chamber bore brought forth ideas of capping the ends of the chamber so fluid can remain retained. The capping of ends to the chamber needed two critical items. Item one was a type of seal and item two the caps needed to be removable to have the intensifier be serviceable.

Making the fabrication and assembly process easier, high strength stub acme threads were chosen for the pressure vessel plugs as well as the end caps. Outside diameter threaded parts are commonly more prone to failure than that of the mating inside diameter counterparts[7]. Calculating the thread axial shear rating of the male threaded part using the following equations.

[7] $$\text{Shear\_Rating} = \frac{\text{LOE} * \text{ShearAreaPerInch} * \text{ShearFactor} * \text{MinimumYield}}{\text{SafetyFactor}} \text{ (lbf)}$$

$$\text{ShearAreaPerInch} = \pi * \text{MaxMinorBoxDia} * \left[\frac{1}{2} + \left(\frac{1}{P} * \tan(14.5^\circ)\right) * (\text{MinPitchDiaPin} - \text{MaxMinorBoxDia})\right] \text{(in)}$$

$$ShearAreaPerInch = \pi * 2.0917 * [\frac{1}{2} + \left(\frac{1}{6} * \tan(14.5°)\right) * (2.1536 - 2.0917)]$$

$$ShearAreaPerInch = 3.30317$$

$$Shear_{Rating} = \frac{1.1 * 3.30317 * 2 * 50000}{4}$$

$$Shear_{Rating} = 90837.1\ lbf$$

The shear rating of ~90837 lbf with a safety factor of four considerably covers the needed 23550 lbf required by the booster to raise internal fluid pressures to 30ksi. As an added measure of security the pressure intensifier assembly was designed in such a way that the thread on end caps double as a reinforcement clasp for the bore plugs. An example can be found in Figure 25.

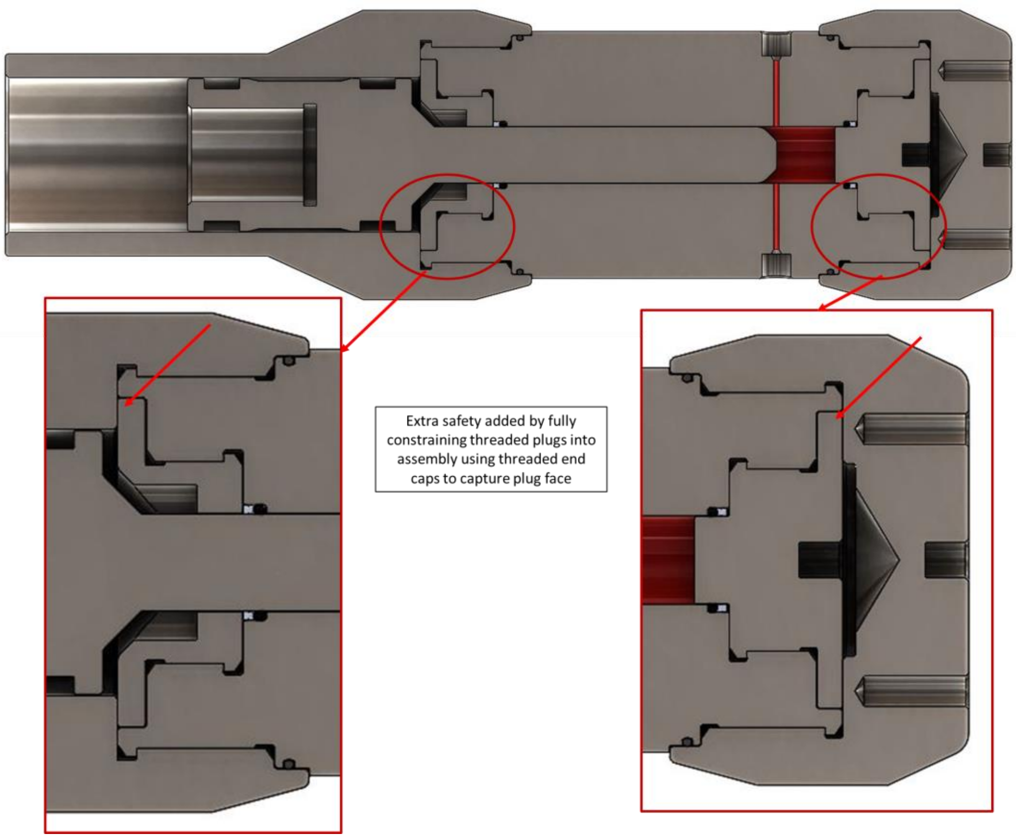


Figure 25: Captive end caps and bore plugs.

Lastly, to contain the fluid media inside the intensifier, a company was contracted to make a set of specialty v-seals as well as a custom guide ring for the piston. These specialty seals are designed to seal around the piston on the power end and on the opposing end act as a bore plug. Seal design was based on a seal stack the company had produced in the past for a specialty pump company, these seals are unique to our design however because of their ability to contain pressures more than 40ksi. The guide ring for the piston is made from a high wear resistant polymer, for assembly purposes there was a specific feature that needed to be added, a sipe which allows for the expansion of the guide ring letting the inner diameter of the ring to slip past ends of the piston.

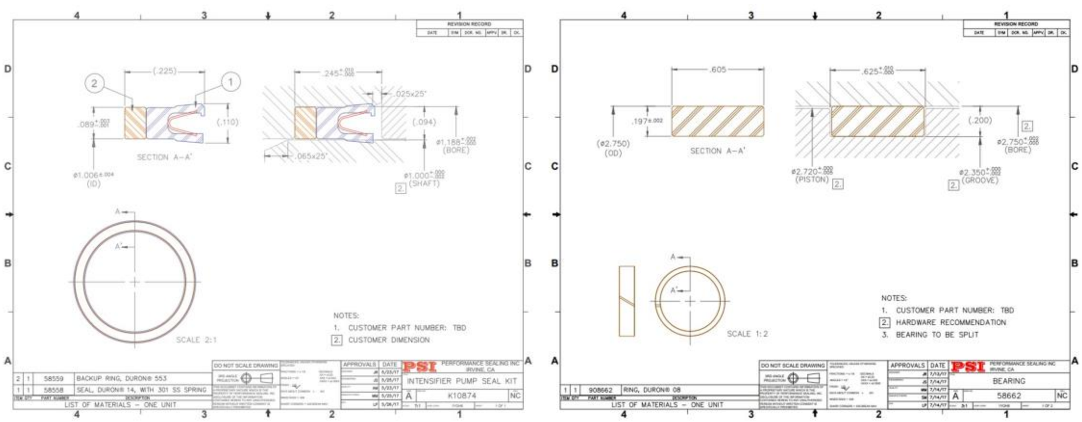


Figure 26: Custom seal and guide bushing design[8].

The last item to outline in the design portion of the intensifier assembly is the piston. Concerns early in the development of this intensifier assembly arose around the piston. There was concern that the small one-inch diameter piston might undergo some small deflection as it is acted upon by high fluid pressures. To be sure of the design additional FEA was used.

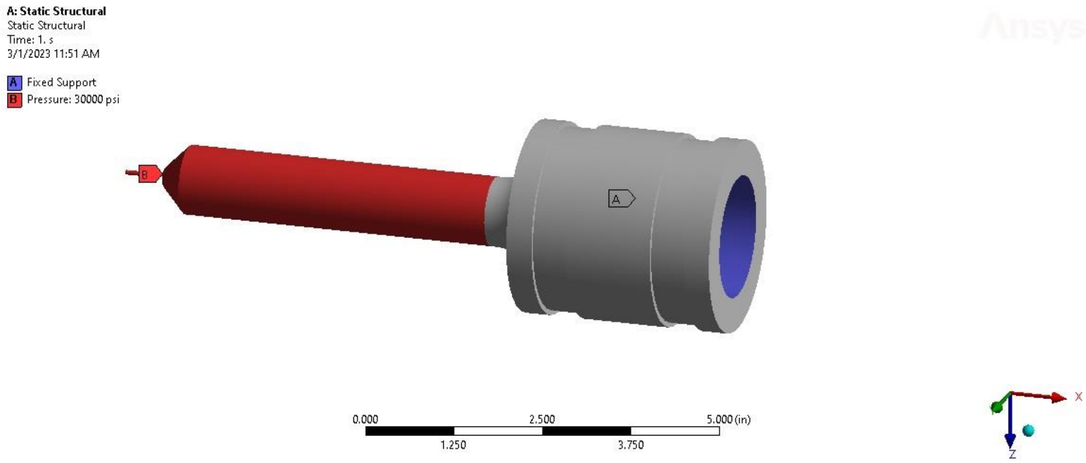


Figure 27: Pressure loading setup intensifier piston.

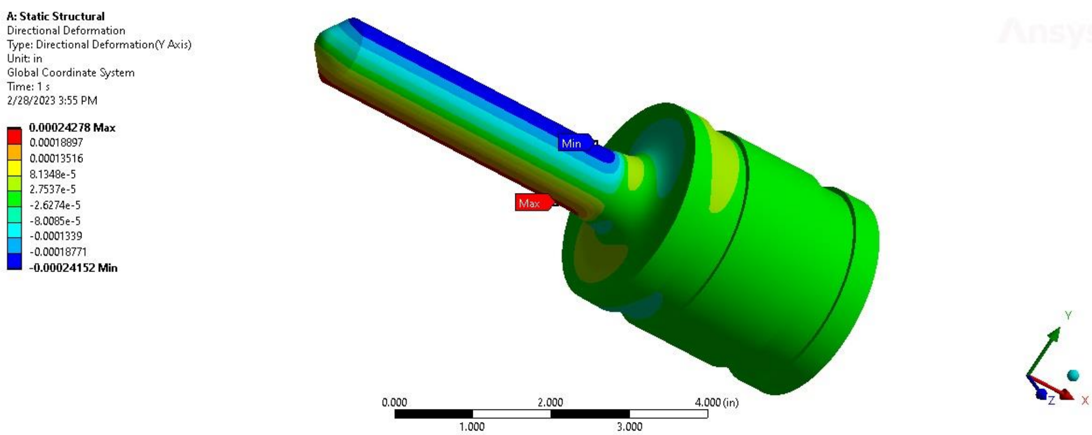


Figure 28: Deformation along Y axis.

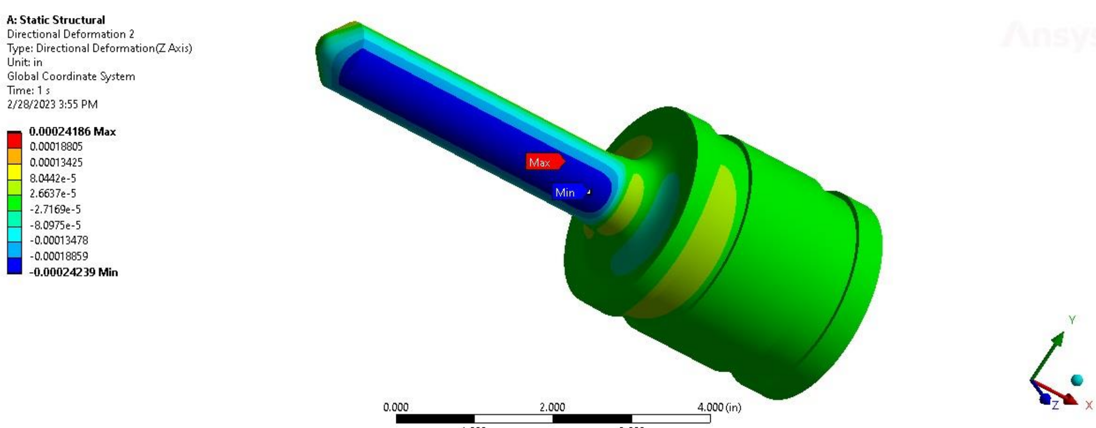


Figure 29: Deformation along Z axis.

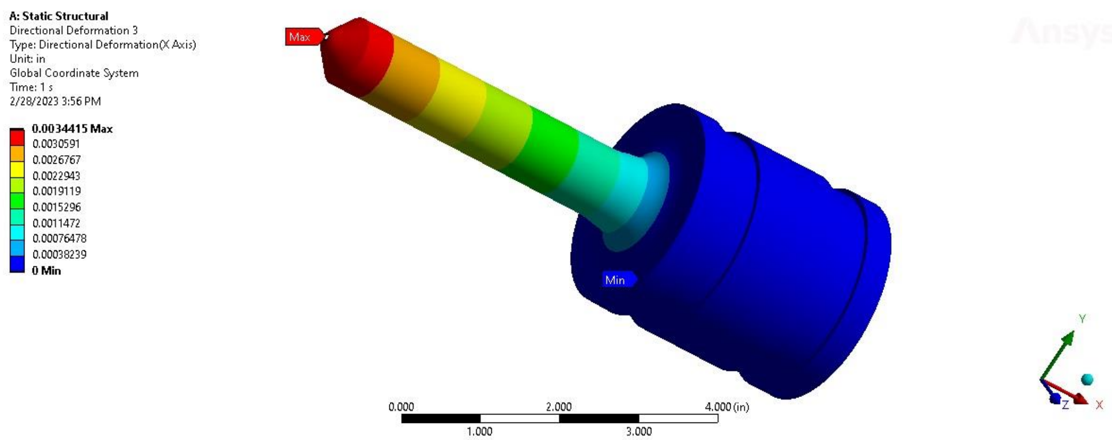


Figure 30: Deformation along X axis.

According to simulation the resulting total deformation along the y and z axis are equal and opposite. The reasoning for this aligns with the expected symmetrical distortion as the planes which these axis's intercept are orthogonal and the loading of the piston is consistent around the cylinder circumference. Largest expected deformation lies in the x axis direction and this deformation would result in some small length change in the piston.

3.2.2. Intensifier Cradle

The next step in the design process concerns the answering of the question, how will we contain the forces exerted by the ball screw jack acting onto the intensifier? Several different orientations for the intensifier were explored and in the preliminary design review with the sponsor it was suggested to have the intensifier lay horizontally so when it comes time to prime the system, trapped air in the fluid would be easier to bleed. From here a cradle concept was created and can be overviewed in Figure 31.

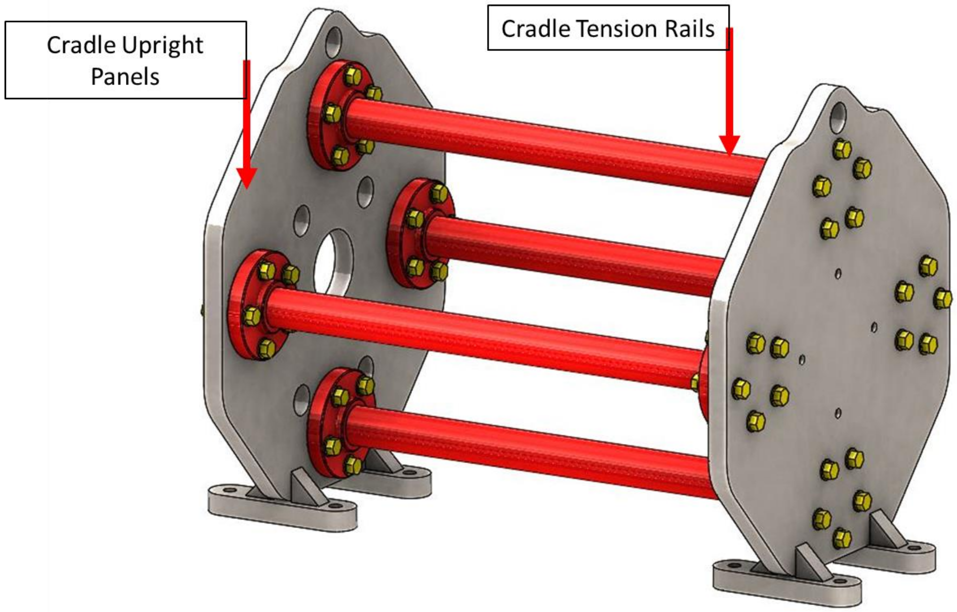


Figure 31: Intensifier cradle.

Main components of the intensifier cradle are the tension rails and the upright panels. The tension rails need to be capable of producing a reactive force greater than the force being produced by the ball screw jack and intensifier. The chosen material for the uprights in this assembly became A572-50 and the material for the tension rails A513-Type 5 which is a electric-resistance-welded drawn over mandrel structural tube[9]. Tension rails are weldments, and the flanges are mated with a socket style weld. Weld beads of a quarter inch form a weld path on both the exterior of the tube as well as interior of the tubes. This style of weld joint is commonly accepted and practiced when quasi full weld bead penetration is desired but unachievable[10].

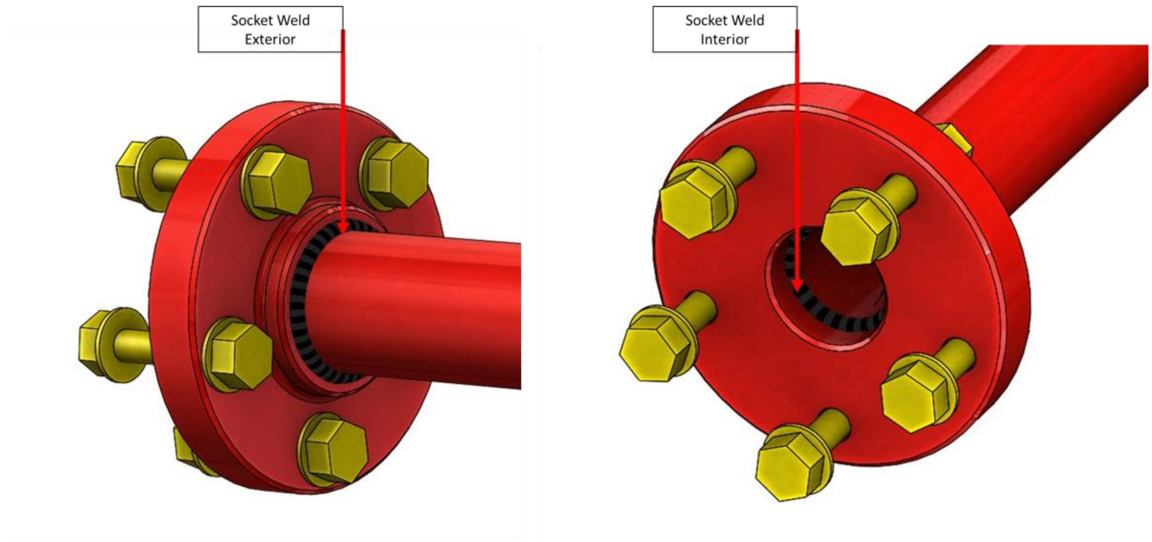


Figure 32: Tension rail socket weld example.

Knowing material selection and design footprint it became necessary to do some preliminary analysis on the cradle. Reactive forces were applied to the cradle to mimic actual operation, fixed supports were placed at areas where the ball screw jack bolts to the frame, and pretension was applied to all bolts creating a as-build clamping force for more realistic simulation results. Bolt preload was set to 9550 lbf [11-13] and a force of 50k lbf applied to the intensifier mounting flange.

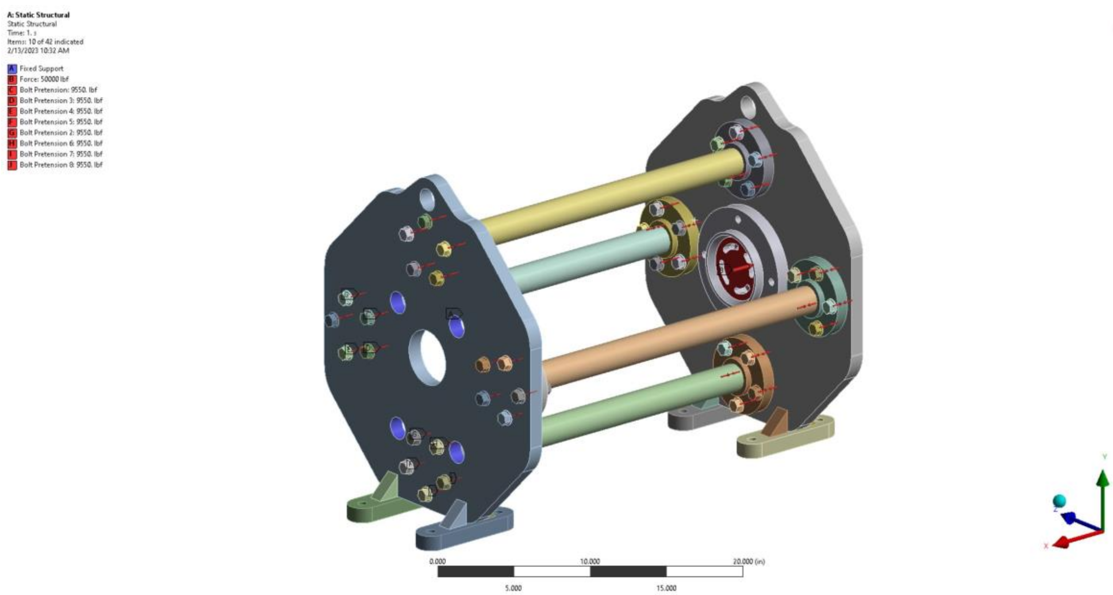


Figure 33: Setup cradle reactive forces.

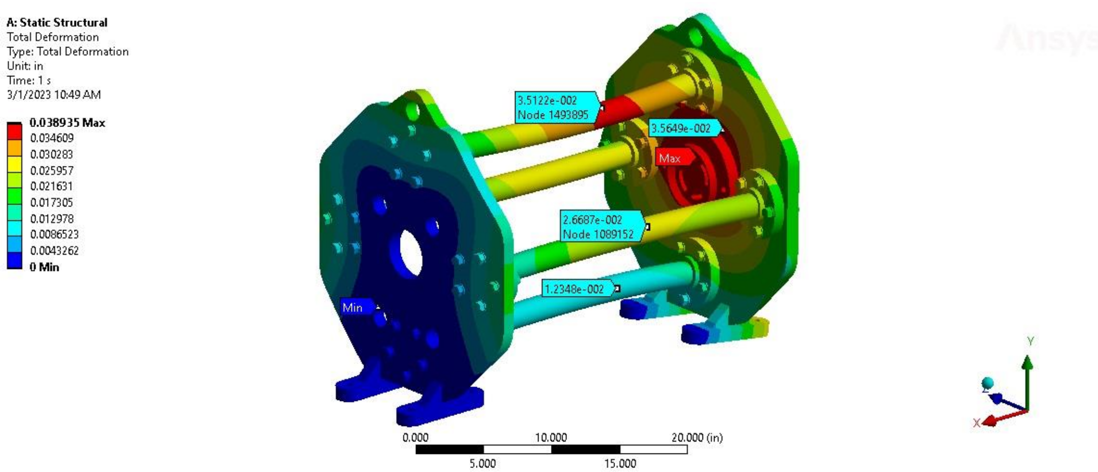


Figure 34: Cradle total expected deformation.

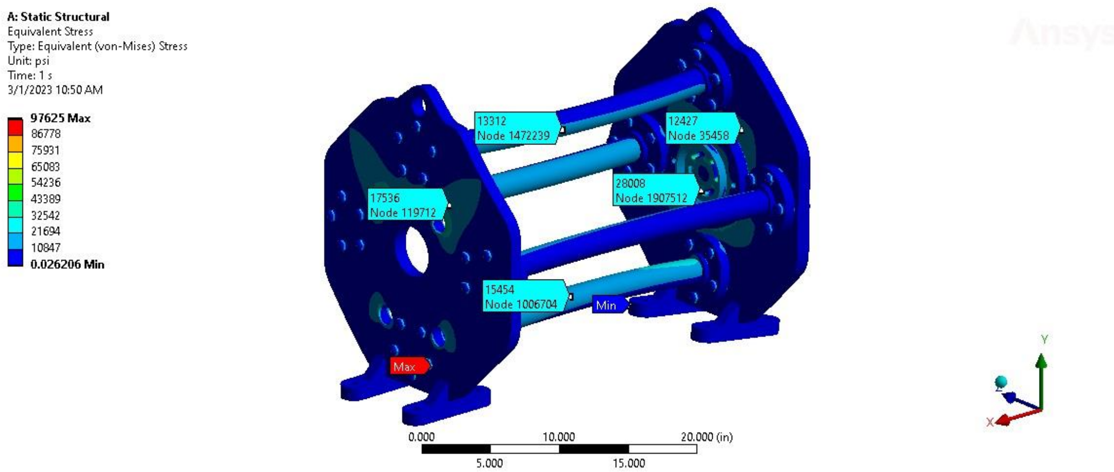


Figure 35: Cradle equivalent von-mises stress.

Reviewing simulation results an expected total deformation of thirty thousandths of an inch is present along the x axis and located at the intensifier mounting flange. The resulting equivalent von-mises stress is approximately 97ksi and is located along the axis of one of the fasteners.

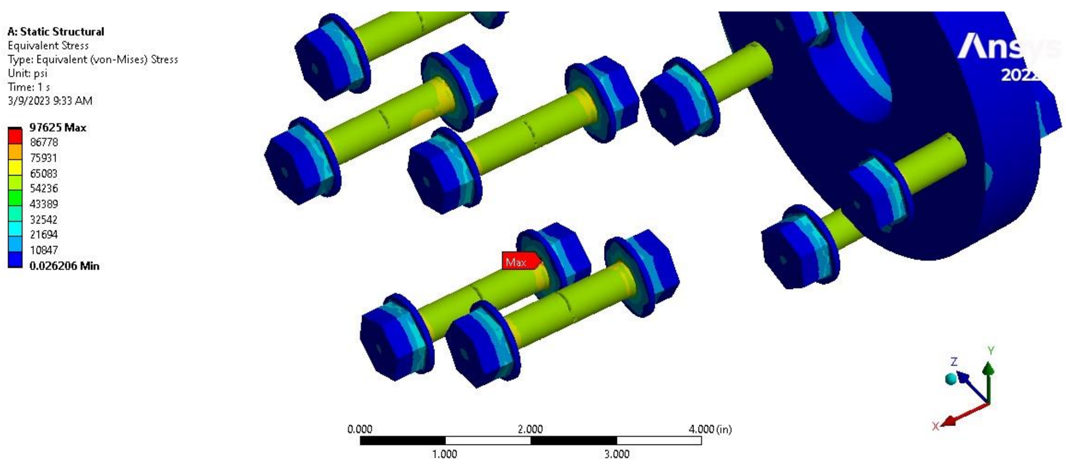


Figure 36: Cradle bolt equivalent von-mises stress.

Given that the fasteners are grade 9 bolts, with a 190ksi tensile yield strength rating[11], and recalling that the applied force in the simulation is double that of expected operational force, the design performance is acceptable. Further it is again in need of mentioning that areas of critical values are influenced highly by FEA meshing and instances of sharpness and non-compliance in corners and edges.

3.2.3. Intensifier Centralizing Brace

Critical to keep binding and misalignment to a minimum, and to give some real estate for mounting of overtravel limit switches the following centralizing alignment brace was developed.

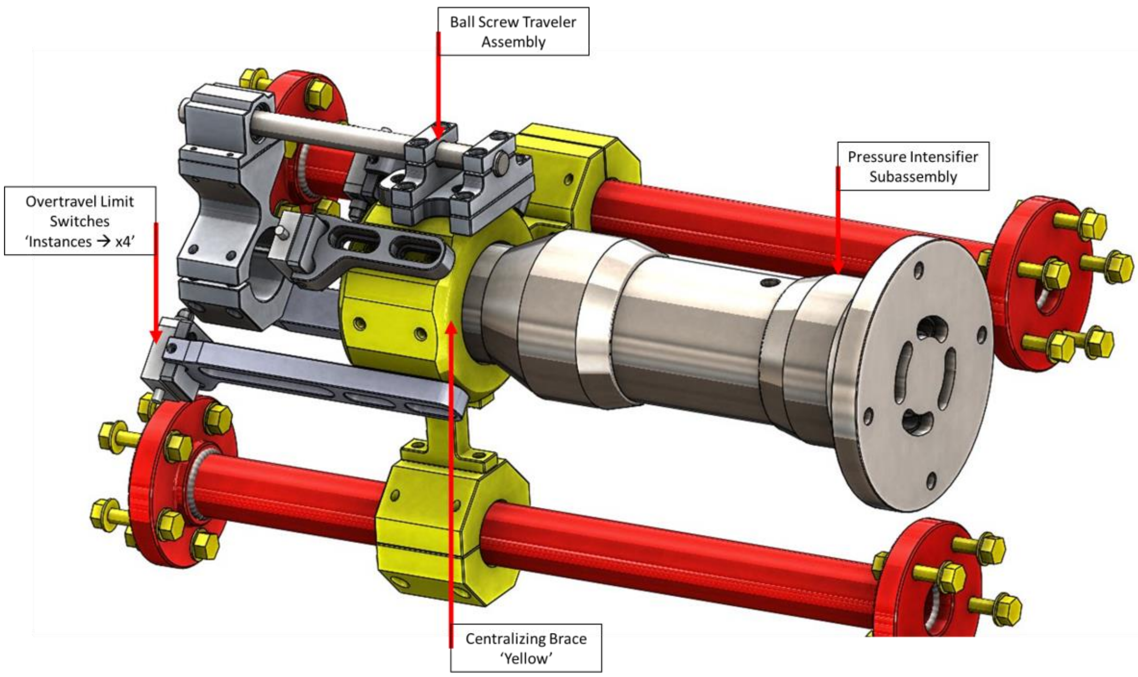


Figure 37: Intensifier centralizing brace.

A more detailed view of the centralizing brace present within the pressure intensifier subassembly can be seen in Figure 37. Most fasteners have been excluded to give clarity. The larger function of this manufactured unit is to allow correct coaxial alignment of pressure intensifier to linear motion of ball screw jack travel. Additional discrete parts act as peripheral bolt on accessories, such as overtravel limit switches which will be explained in more detail in a later chapter, and the ball screw traveler assembly. The ball screw traveler assembly acts as both a triggering unit for the limit switches and clamps to the output shaft of the ball screw jack. The fixturing of the clamping portion of the ball screw traveler keeps the linear extending and retraction motion of the ball screw jack concise, and without some backlash in movement is prevalent. Considering this portion of the intensifier cradle subassembly as an alignment device and as a accessory mounting platform, easily machinable low carbon steel was selected, 1018. Considerable time under simulation was not of concern as this assembly when correctly aligned would not expect significant side loading effects. The bracing unit was mostly designed as a precautionary implement and thus serves as such.

3.2.4 Servomotor, Servo Drive, Shear Coupling, Ball Screw Jack

Driving the pressure intensifier there exists a series of daisy chained components. Fundamentally the driving force behind the intensifier was chosen to be a Nook Industries ball screw jack. This jack operates in simplicity as a conventional propping mechanism, however, here the function has been modified to press upon a piston isolated within a pressure housing. Stated before, the linear pushing of the jack acts upon a single rod which in turn has the capability to build pressure inside of its respected bore and housing assembly. The first option for this piston driving force was narrowed to a more specific scope because of controllability. Original options included a hydraulic bottle style jack, and an electric linear actuator. It was evident that in order to more accurately control the fluid pressurization process there existed a need for very robust controls and mechanic, therefore the following components were chosen, Figure 38.

Servomotor	Shear Coupling	Ball Screw Jack	Drive Controller
			
AKM44H-ACCNC-00	Manufactured Part	0631-0500 SRT RA/HK/HN/24.00/SBN1011 3/S	AKD-T00607-ICAN

Figure 38: Driving components behind pressure intensifier.

Seen from left to right, Figure 27, the drive assembly for the pressure intensifier consists of a Kollmorgen servomotor, adapter coupling which bridges the output shaft of the servomotor to the input shaft of the ball screw jack, a Nook SBN1011 ball screw jack, and the control function a servo drive controller. Recalling the required force needed to act upon the intensifier, 23550lbf, to create a fluid pressure of 30ksi, the SBN1011 became the component of choice.

MODEL	Gear ratio	Capacity (tons)	Lifting Screw Dia (in)	Screw Lead (in)	Root Dia (in)	Turns of Worm for 1" travel	Max input Torque (in.-lb.)	Max Input (hp)	Max Worm Speed at Rated Load (rpm)	Max Load at 1,750 rpm (lb)	Torque to Raise 1 lb. (in-lb)	Tare Drag Torque (in-lb)	Backdrive Holding Torque (ft-lb)
20-BSJ	8:1	20	2¼	.50	1.850	16	626	7½	755	17,204	.0157	40	27
	24:1	20	2¼	.50	1.850	48	314	2½	501	11,397	.0079	40	7

Figure 39: Nook ball screw jack model 20-BSJ[14].

Following the selection of a Nook Industries ball screw jack it was understood torque requirements at the input shaft required to move a desired load. The 24:1 gearing ratio single thread start ball screw jack allows for one pound of lift given a .0079in-lb input. Placing this into the intensifier application we can iterate the following input requirements to accomplish 23550lbf of compressive force.

$$Torque_{Input} = 23550 * .0079$$

$$Torque_{Input} = \sim 186\ in - lb$$

Knowing this design requirement, we can properly size a drive motor. Two separate readily available automation friendly options were available. Option one involves a coil wound AC induction style motor and the second option involves a servomotor selection.

AC Induction Motor vs. Servo Motor

Servo Motor	AC Induction Motor
• Encoder included	• Motor brake
• High resolution rotation	• Encoder not included
• Programmable drives	• No active feedback
• High locked rotor torque	• Current monitoring sole feedback source
	• Variable frequency drive
	• Low locked rotor torque

Figure 40: Electric motor comparison.

Given that the servomotor gives exceptional performance at stall or locked rotor torque, and precise rotary controls as well as feedback, a Kollmorgen servo and drive were chosen to drive the intensifier. Generated operational torque curve can be found in Figure 41.

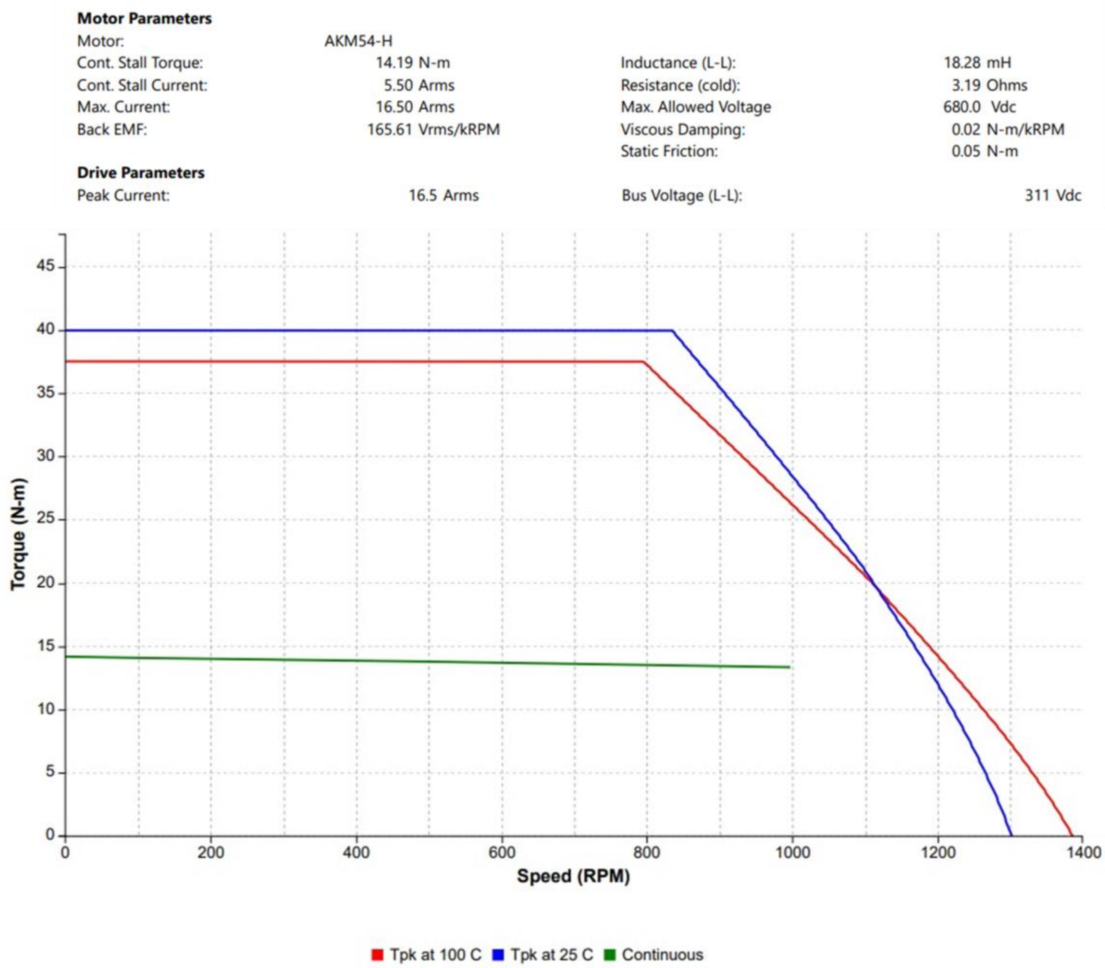


Figure 41: Kollmorgen AKM54H speed vs. torque curve[15].

Recalling that ~186 in-lb is required to place the appropriate force on the intensifier, the above graph states a locker rotor torque of 40 N-m which equates to 354 in-lb. This torque output of the chosen AKM54H is more than sufficient.

3.3 RECIPROCATING ROD AND BED SUBASSEMBLY

The reciprocating rod test bed is the main platform for the testing of seals. This platform gives motion to a rod held within a pressurized housing by some geometry of seals. The rod is affixed at the power end to a load cell and the opposing end of the rod is constrained. The motion is driven by a Kollmorgen servomotor attached to a Nook industries linear ball screw. Linear guide rails spanning the length of the bed allow for precise linear motion of the rod. The device under testing is held and mounted to the bed through a machined upright bracket, Figure 42.

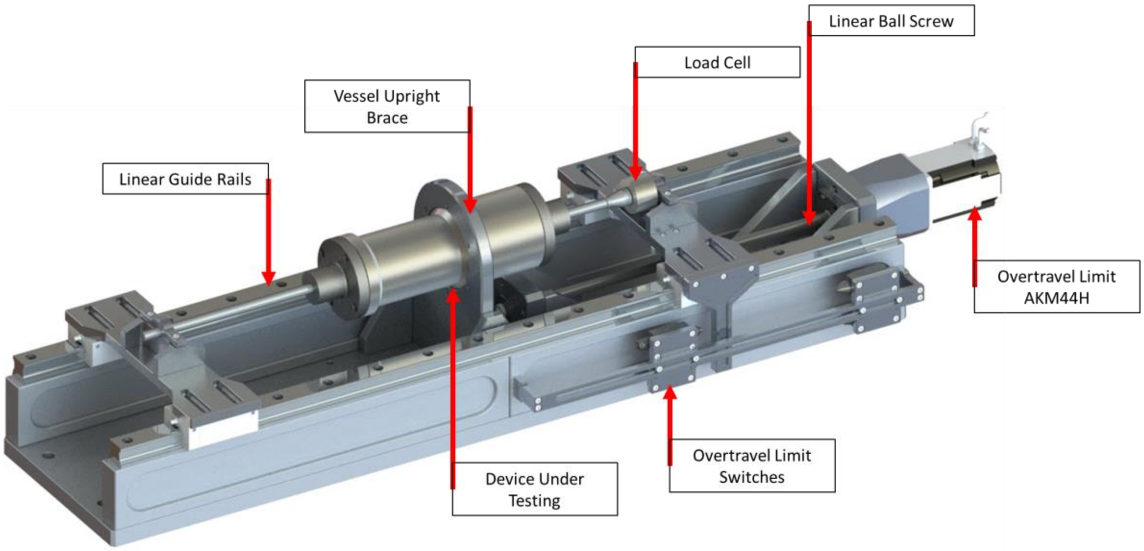


Figure 42: Reciprocating rod test bed introduction.

Special considerations were made to the design of the bed components. Manufactured components on the test bed are all modular and can be dissected into more subassemblies. This allows for easy replacement or alteration of such subassemblies to receive different future devices under testing.

Decisions on material choice for all manufactured parts on the test bed for the exception of the pressurized device under testing became 6061-t6 aluminum. The driving decisions here were focused on cost and ease of machinability. Further, expected reactional forces exerted on the rod by the seals was projected to be less than 500lbf. Setting this reactive force as a design constraint and after getting approval from the sponsor about overall design concessions, base FEA simulations were used to better understand potential means of failure. Two different simulations were run and will be introduced hereafter. Firstly, the model will be constrained, and forces applied in a way to mimic the reciprocating rod test bed undergoing a compression stroke. Following this the parameters will be changed to view associated expected worst-case scenario while rod test bed is subjected to a tension stroke.

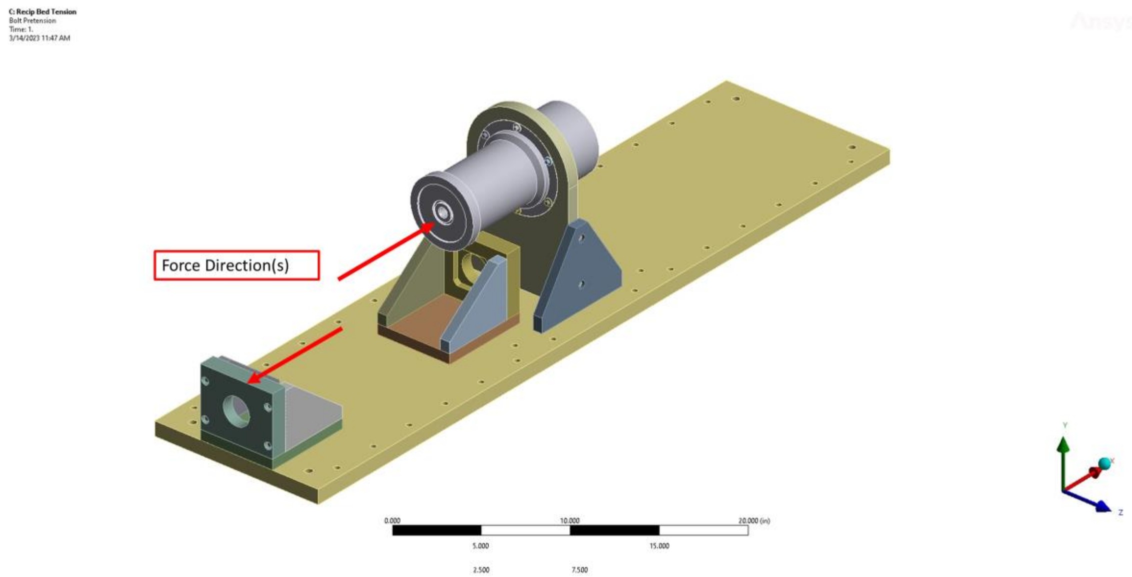


Figure 43: Reciprocating g rod test bed reactive forces set to simulate compression stroke.

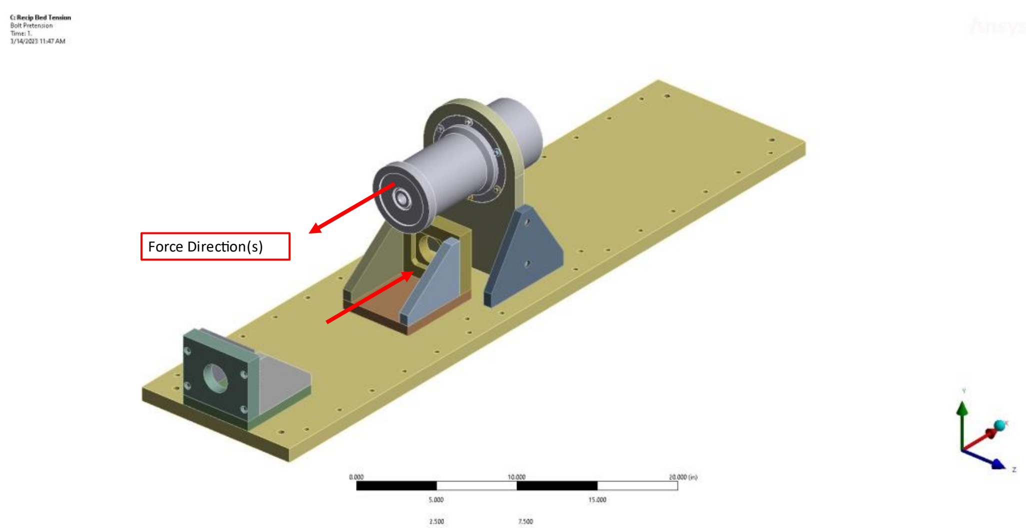


Figure 44: Reciprocating rod test bed reactive forces set to simulate tension stroke.

The reactive forces set in Figure 43, and Figure 44 are assuming an opposite and equal reaction between the upright mounts for the nook linear ball screw. This mirrors worst case force loading due to the instance of not allowing upright mounts to be bridge by the ball screw. There is in turn no force sharing between these mounts as there would be in actuality. For ease of simulation some components have been omitted and assumptions made.

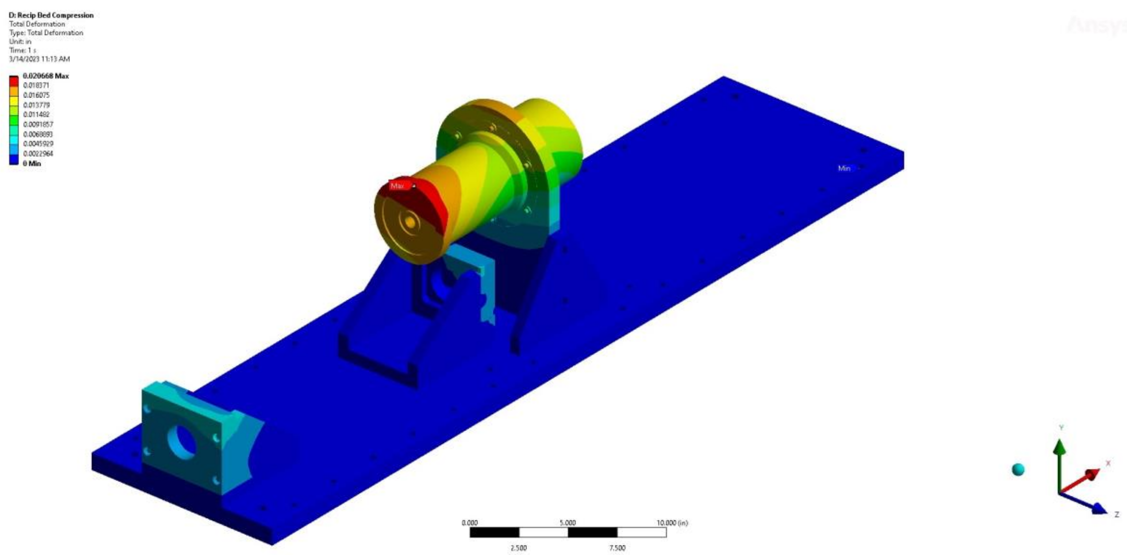


Figure 45: Total deformation of rod test bed and components under compression stroke.

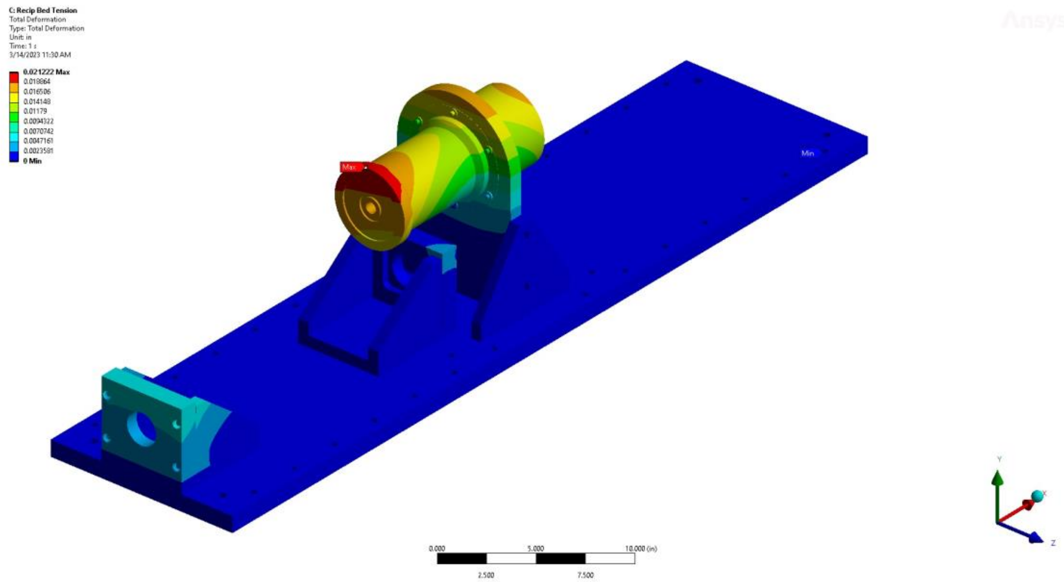


Figure 46: Total deformation of rod test bed and components wile under tension stroke.

From the figures representing the reciprocating rod test bed under both compression and tension stroke we see a maximum value reoccurring at the same localized point. Under the tension and compression stroke the unsymmetrical form of the device under testing acts as a lever arm and raises slightly. The largest value of deformation in compression stroke is along the expected y-axis, the y-axis, and has a value of 0.02inch. Likewise, the value of total deformation along the y-axis in tension has a negligible value of 0.21inch. Demising here from the deformation plots it seems some rigidity for future designs might need to be assessed for the vessel upright mount, this mount experiences the most deformation due to its height, bracing, and possible material choice.

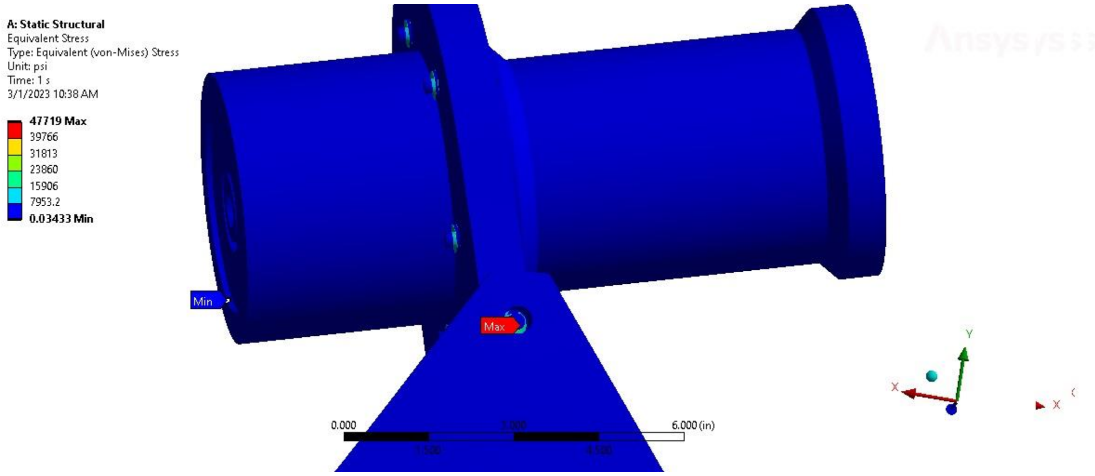


Figure 47: Equivalent von-mises stress at fastener location on upright vessel mount.

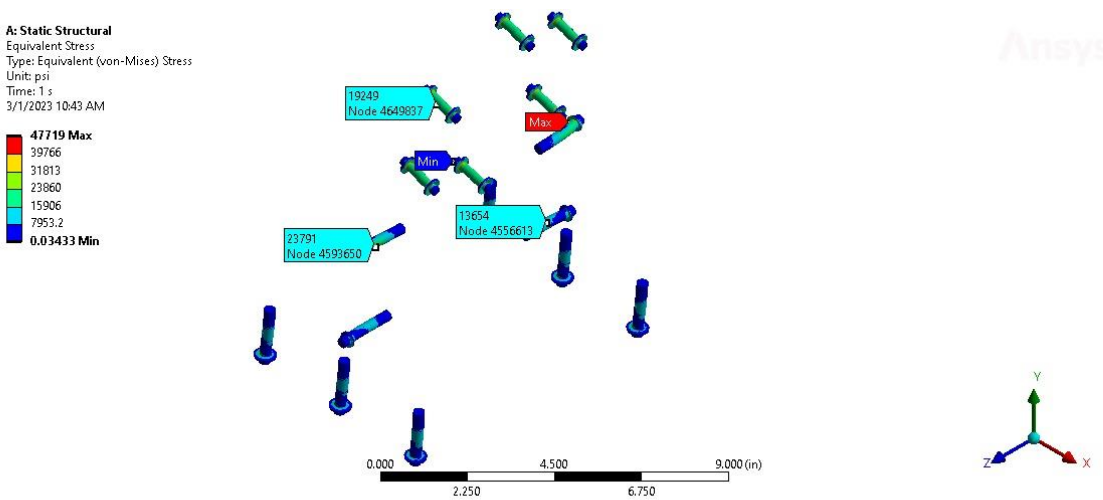


Figure 48: Removing components systematically von-mises stress found along fastener on upright vessel mount.

Results for total equivalent von-mises stress appears at bolt locations on the upright vessel mount bracket while undergoing compression stroke. These bolt locations are acting in shear and were expected to be components experiencing the highest evaluated stresses. Omitting some components to clean the visualization of the model it is found that the area of excessive assessed stress to be underneath the simulated bolt head, again where mesh convergence is sharp. Taking the average von-mises stress over the body of the bolts we see an average value of 23ksi, well within the design parameters of shear loading of bolts.

Since resulting principal stress and overall equivalent von-mises stress plots of the reciprocating test bed in its entirety were showing no large component stress concentrations the are not represented here. Largest areas of interest before beginning analysis were overall deformation of the upright vessel mount and stresses at bolt locations. Specifically stresses expected around the female threaded holes in manufactured parts. Concluding with the added rigidity of suppressed components and that also of a rigid mount to the setup frame the design was acceptable and the research objectives continued.

3.3.2 Servomotor, Servo Drive, Shear Coupling, Linear Ball Screw

Powering the rectilinear motion of the rod traveling through its compression stroke and then returning in its tension, a combination of items have been chosen. Keeping with common components, both a Kollmorgen AKM servomotor and Kollmorgen AKD drive were selected. The advancement and retracting of the rod assembly through the device under testing decidedly was achieved by using a linear ball screw driven by the AKD, AKM. And shear coupling combination. Discussions concerning the shear coupling will be present in proceeding chapters. A brief summary of the reciprocating rod test bed drive components can be seen in Figure 49.

Servomotor	Shear Coupling	Ball Screw Linear Motion Assembly	Drive Controller
			
AKM54H-ACC2C-00	Manufactured Part	0631-0500 SRT RA/HK/HN/24.00/SBN10113 /S	AKD-T00607-ICAN

Figure 49: Component drive selection summary for reciprocating rod test bed.

Very specific linear control of the reciprocating rod was keystone to fulfilling the primary investor's needs. Recalling that such needs were specific velocity control, acceleration, deceleration control, and torque for loosening seals as they move into their dynamic state from their static state. The combination of the AKD, AKM, smart feedback device configurations allow the servomotor to be controlled in either a velocity, torque, or position mode. The position mode was the deciding factor for these component selections. All AKM servomotors contain a 20 bit signal resolution, and 16,777,213 counts for every one revolution of the output shaft[16].

Before deciding upon a specific servomotor build it was firstly decided to find a readily available linear ball screw assembly for direct bolt on application to the reciprocating rod test bed. Due to Nook industries great customer service and dense product catalog it was decided to look into premade double flange ball screw assemblies. Fitting within our space constraints the Nook 0631-0500 SRT double start ball nut and screw assembly fit nicely[17].

	BALL NUT									SCREW			
	Ball Nut Number	Helix	Max. Adj. Preload (lb)	Dynamic Load (lb)	Static Load (lb)	Balls per Circuit/Nut	Nominal Ball Dia. (in)	Nut Wt. (lb)	Torque to Raise 1 lb (in-lb)	Ball Circle Dia. (in)	Lead (in)	Root Dia. (in)	Screw Wt. (lb/ft)
0631-0500 SRT **double start** 0.631 Ball Circle Dia. (in) 0.500 Lead (in)	SBN10113	RH	0	960	5,565	37/74	0.125	0.27	0.088	0.631	0.500	0.500	0.82

Figure 50: Nook industries 0631-500 SRT ball screw specifications[17].

Using the information from Figure 50, knowing the ball screw diameter and pitch configuration, a linear position precision can be shown through the following

$$\frac{16{,}777{,}213\ counts}{1\ rev} * \frac{1\ rev}{0.5in} = \frac{33{,}554{,}426\ counts}{in}$$

$$And\ multiplying\ by, \qquad 1/100$$

$$Linear\ Percision = 33554.4\ counts = \ .001inch\ of\ linear\ travel$$

$$Linear\ Percision = \frac{1}{500th} revolution\ of\ servomotor = \ .001inch\ of\ linear\ travel$$

Knowing two key design parameters now of maximum force expected to reciprocate rod through the device under testing and also the linear ball screw assemblies required torque to drive a load as given from the Nook ball screw assembly catalog Figure 51, we can now determine selection of Kollmorgen servomotor using the driving torque equation as our governing metric.

$$T_d = \frac{F \times P_h}{2\pi \times \eta_1} \times 10^{-3}$$

- T_d = Driving torque (N·m)
- F = Axial load (N)
- P_h = Lead (mm)
- η_1 = Normal efficiency (90%)

Figure 51: Torque formula required by a ball screw assembly to move a load[17].

$$T_d = \frac{2224.11N * 12.7mm}{2 * \pi * .9} * 10^{-3}$$

$$T_d = 4999.7\frac{N}{mm} = 5.0\frac{N}{m}$$

CALCULATE REQUIRED TORQUE TO MOVE A GIVEN LOAD

INPUTS:

B.C.D.	0.631	in	Ball Circle Diameter
Lead	0.5	in	Ball screw lead
Threaded Length	18	in	Usable threaded length of ball screw
Dynamic Load Capacity	1000	lbf	Rated dynamic load capacity listed for ball screw
Preload Amount	0%		Preload applied (0%, 2%, or 5%)
Screw Class	5		Lead Accuracy (T3, T5, T7, and T10)
Operating Load	500	lbf	Applied Load to the ball screw assembly

OUTPUTS:

Operating Load Torque	5.0	Nm	Torque required to move the applied load
Applied Preload	0	N	Amount of load due to preload
Preaload Drag Torque Max.	**0.000**	**Nm**	Max torque fluctuation due to Preload torque variance
Preaload Drag Torque Min.	0.000	Nm	Min torque fluctuation due to Preload torque variance

TOTAL TORQUE:

Torque Variance Max.	**5.0**	**Nm**
Torque Variance Min.	**5.0**	**Nm**

Added torque from wiper seals can be as much as 2x that
of the calculated torque from preload.
If the system is not fashioned with a preloaded ball screw,
but does utilize ball nut wipers, up to 0.4Nm needs to be added
to the calculations.

Figure 52: Nook industries calculation of driving torque for the linear ball screw assembly[18].

Verifting hand calculation of torque required to drive the desired load the online Nook industries calculator was used for self verification, Figure 52. Knowing all needed drive parameters for servomotor the AKM44H[19] was chosen. This servomotor os one case size smaller than the servomotor used for the pressure intensifier assembly but programming and operation remains identical. Torque justification can be assimilated from the Kollmorgen AKM44H performance curves seen in Figure 53.

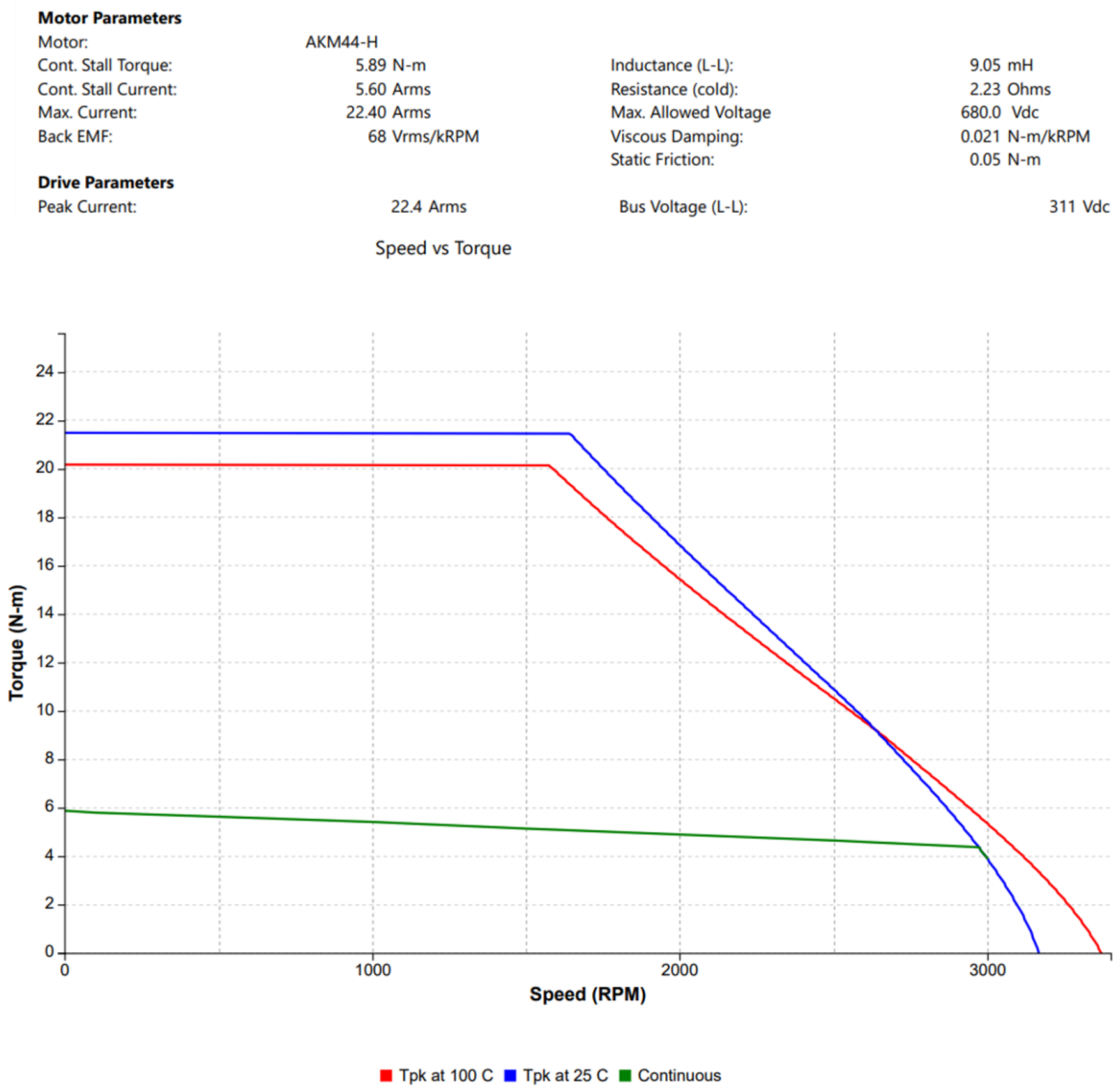


Figure 53: Kollmorgen AKM44H speed vs. torque curve[20].

3.4 FRAME

As a crucial aspect to the portability and modular design required by this research, the principal investors requested a rigid and easily portable frame to be designed. The frame needed a few key features including encasing the entire operations of the system within the frames footprint. Adaptability to allow future devices under testing to be easily affixed and prepared for testing. Ergonomics for the technician servicing components and preparing experiments, and lots of visibility to components. Visibility helping the technician assess any potential or occurring issues while device is operational.

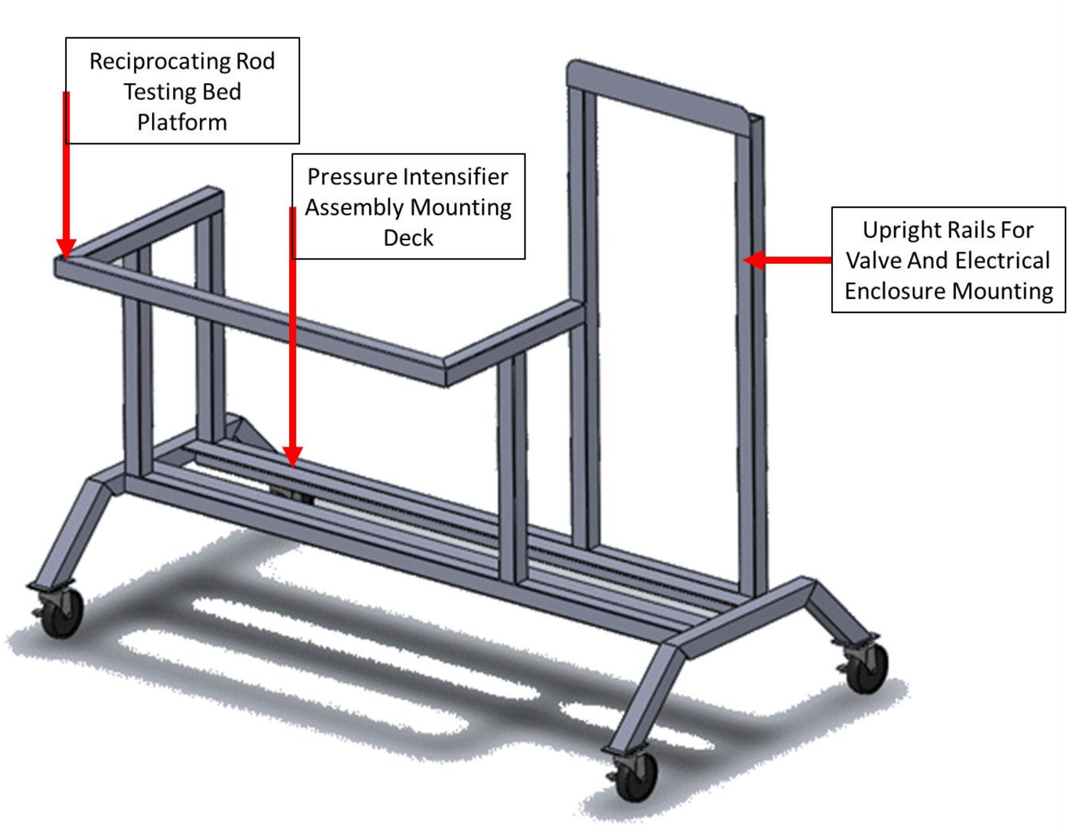


Figure 54: Introduction to frame and related design needs.

Seen in the figure above a quick introduction to the proposed design will be explained. The structure is made from two-inch square aluminum tubing, quarter inch wall thickness, and all joints welded for strength. Sitting forty inches from the ground, a platform large enough for the reciprocating rod test bed. The height of this platform allows the technician to access all components on the testing bed while standing.

Located at a lower center of gravity lies a deck for the mounting of the pressure intensifier assembly. The large opening or ‘C’ shape made into the reciprocating rod testing bed testing platform allows for the pressure intensifier assembly to be lifted onto the frame with a overhead crane before the mounting of the reciprocating rod testbed assembly. Lastly, welded rigid upright mounting rails are available for electrical enclosure mounting and automated valve mounting.

Main concerns with an airframe assembly such as this are high stress concentrations at joints, and potential deformation and sagging of the middle of the testing frame. Using the mass properties calculations derived from the 3D modeling software, the values in Figure 55 will be used as the force loading vectors in frame analysis.

Mass Properties Of Pressure Intensifier Assembly

Mass	631 pounds
Volume	2428 cubic inches
Center Of Gravity	X = 29.9 inches
	Y = -1.5 inches
	Z = -5.3 inches

Mass Properties Of Reciprocating Rod Test Bed Assembly

Mass	230 pounds
Volume	1791 cubic inches
Center Of Gravity	X = 15 inches
	Y = 21 inches
	Z = 40 inches

Figure 55: Mass properties for pressure intensifier and reciprocating rod test bed assemblies.

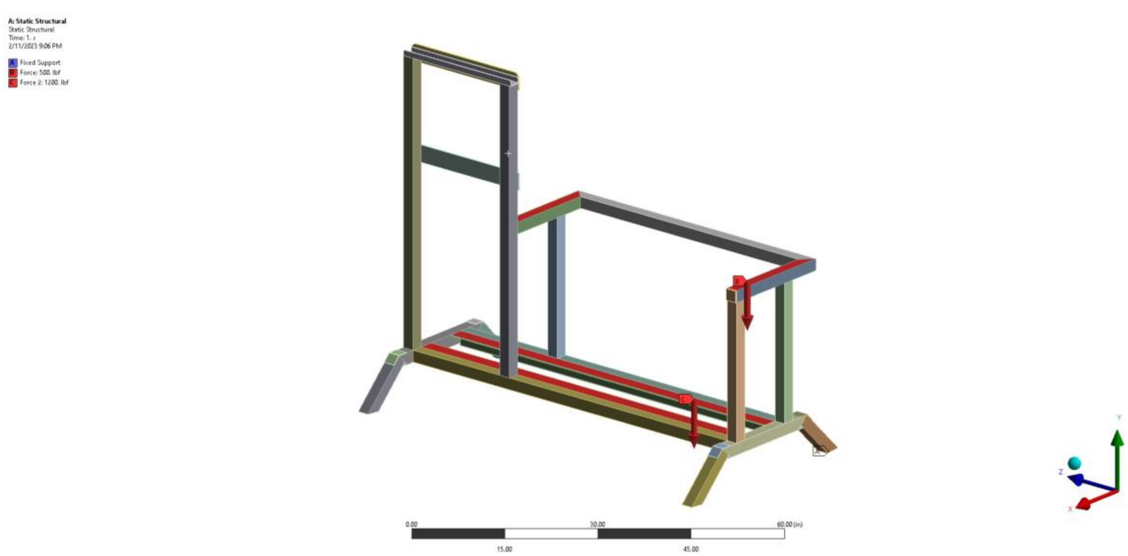


Figure 56: FEA setup for analysis of frame.

Analysis of frame assembly begins by considering all mating parts to be perfectly bonded contacts to immolate welded part end terminations. Fixed supports are applied to the bottom surface of two of the four frame legs, legs located at the bottom right-hand corner of Figure 56. Legs on the opposing end of the frame are allowed to translate in the zx-plane. Using weights calculated from mass properties for the pressure intensifier and reciprocating bed assemblies in Figure 55, associated distributed forces are applied at locations where the reciprocating rod test bed and pressure intensifier mount.

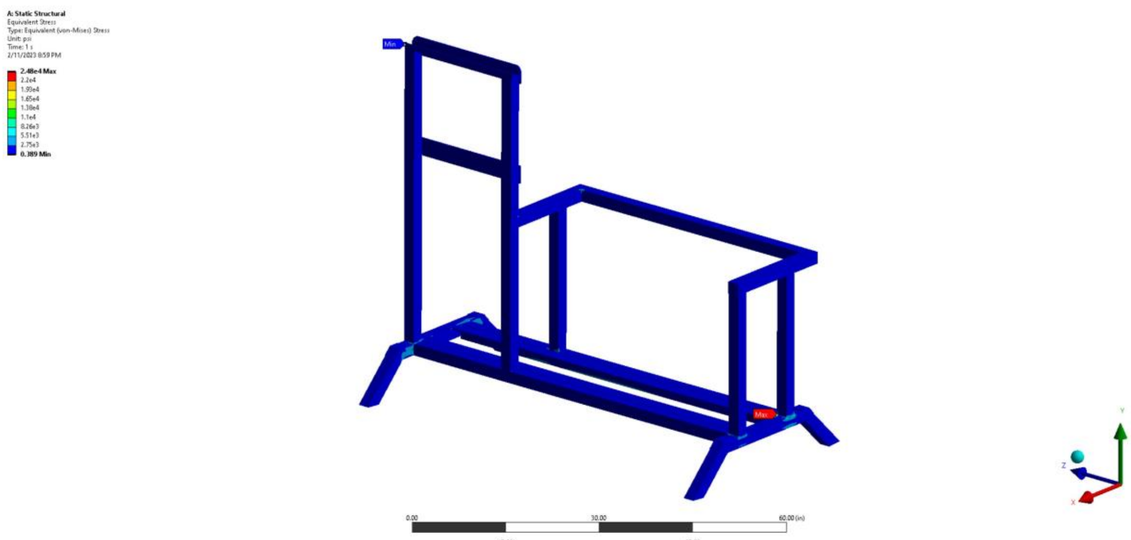


Figure 57: Equivalent von-mises stress results of frame.

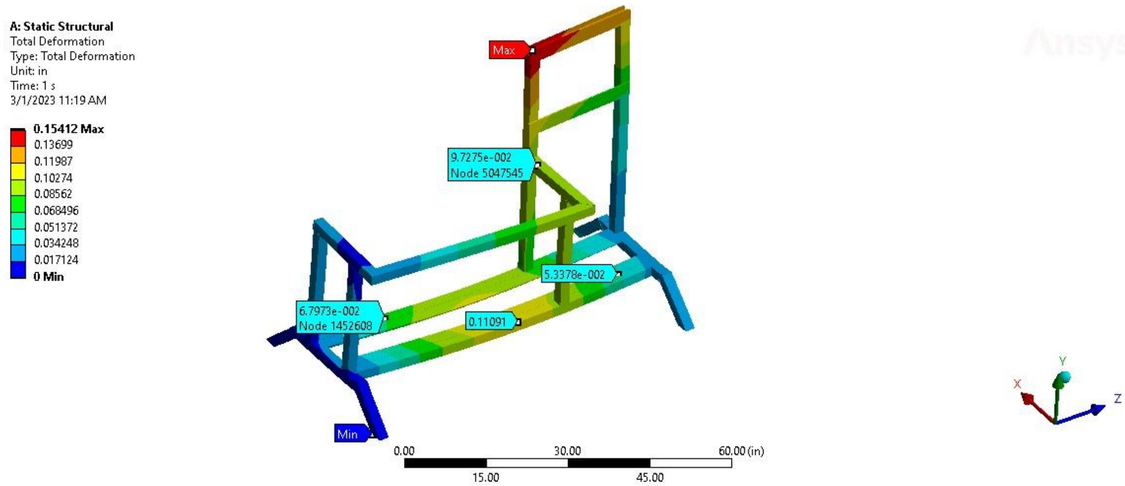


Figure 58: Total deformation result of frame.

Maximum equivalent von-mises stresses as seen in Figure 57, are again located at isolated sharp corners of meshed model. These instances of extreme stress concentrations are artificial. Taking a wholistic view of the results realistic areas of reasonable equivalent von-mises stress are seen around the frame legs, which is where assumed high stress concentrations were suggested to be located before analysis began. These equivalent von-mises stresses were averaged and values of approximate 5ksi-8ksi were found, significantly less than material yield values published in Figure 59for the structural square aluminum tube used to create the experimental test setup frame.

6063-T52 ALUMINUM SQUARE TUBE

MECHANICAL INFORMATION

	Imperial	Metric
Density	0.1 lb/in3	2.7 g/cc
Ultimate Tensile Strength	27,000psi	186 MPa
Yield Tensile Strength	21,000psi	145 MPa

Figure 59: Mechanical properties of square aluminum tube used for frame analysis.

Viewing the total equivalent deformation of the structure, an exaggerated bow along the midplane is seen in Figure 58. This deformation of 0.015inch could be considered negligible as applied forces from analysis setup were applied not as point vectors but of overall distributed loads. True loading moves forces further from frame center plane and more towards edges where bracing is more plateful.

3.5 PIPING SYSTEM

Industrial applications requiring the transfer of flow media are comprised of a variety of flow and fluid transfer components. In this chapter we will overview the function of this often-overlooked subsystem and review approach taken to both pressurize fluid media and exhaust media after experiments come to competition. Critical design requirements for this research will be addressed by illustrating how intensified fluid pressure flows from the pressure intensifier through the system while being monitored by pressure transducer. Priming of system will also be discussed as will depressurization recommendations.

3.5.1 Piping and Instrumentation Diagram (P&ID)

The process flow of the accepted piping and instrumentation for this experimental setup has undergone a great deal of simplification processes. The resulting routing below in the process and instrumentation diagram has been developed to allow ease of understanding and operation.

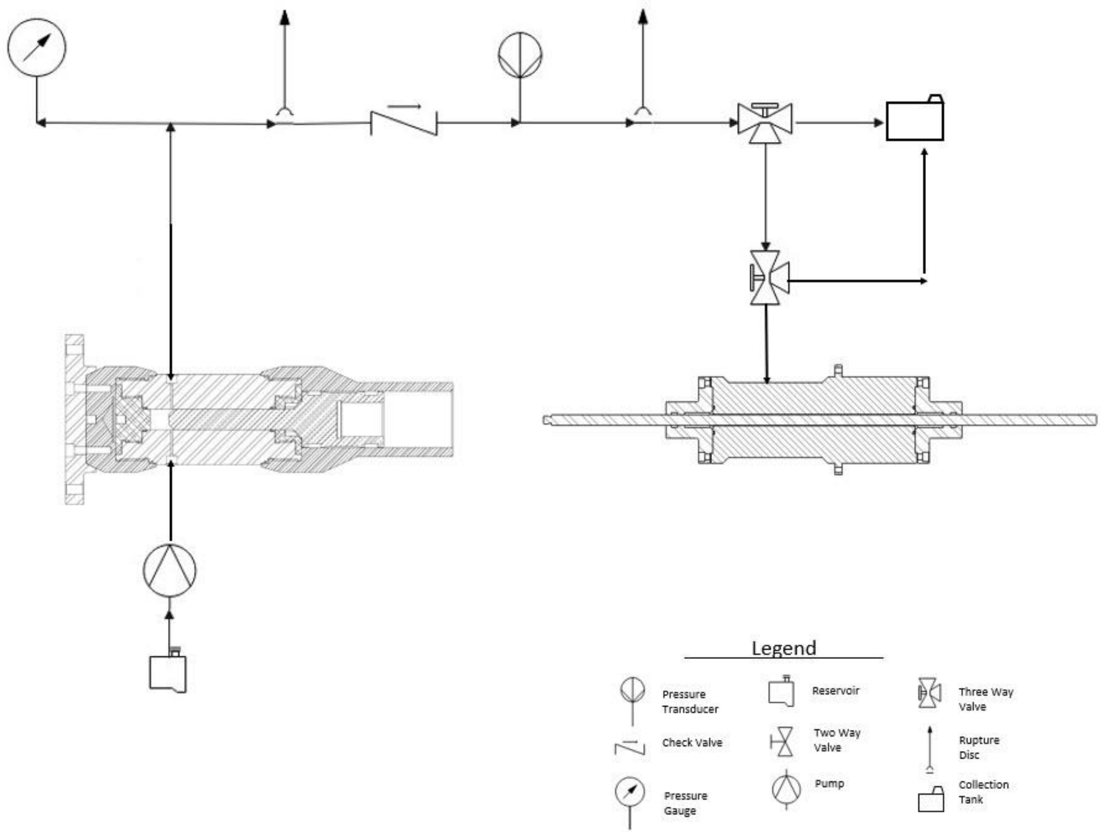


Figure 60: Piping and instrumentation diagram of system.

The two large components in the designed system are the pressure intensifier and the device under testing. Recall that the pressure intensifier is required by this research to accomplish the large pressure differentials which act upon the seals held within the device under testing.

As seen in Figure 60, following the direction of fluid flow, the fluid circuit consists of thirteen components. A fluid reservoir feeds a pump which passes fluid upstream to the pressure intensifier flooding the intensifier’s cavity with fluid. The primed intensifier then passes fluid into two legs. These two legs flow fluid to an analog pressure gauge and pressure relief valve. After fluid passes through the pressure relief valve, the fluid flows through a pressure transducer and then through another pressure relief valve. The exiting fluid branches into a three-way valve which is normally open to flow fluid down into a second three-way valve. This three-way valve always remains closed for the exception of it being opened during the depressurization process. At this junction the fluid can either flow into the device under testing or while the valve is fully open fluid flows back into a collection tank.

Priming the system begins by opening the single direction needle valve located between the pressure intensifier and the pressure gauge/relief valve leg. The third valve is a three-way valve with legs entering the valve from left to right and top to bottom being normally open. During normal operation this needle valve remains closed. Needle valve number three is also a three-way valve with ports where fluid enters from top to bottom remaining normally open, this valve needs to be opened to allow flow into the collection tank. Following the valve sequencing a pump pushes fluid through the circuit which eventually primes the device under testing and bleeds any trapped fluid containing bubbles into the collection tank. The priming process is cycled until no bubbles are present in the collection tank. When no air is determined to be in the system the three-way valve closest to the device under testing is closed and the system is primed for operation.

3.5.2 Valves, Tubing, Fittings

High pressures required by this project led to the selection of uniquely designed and engineered fluid flow componentry. The best suited company which had all suitable components for this build was identified as High Pressure Equipment Company. This company supplies a wide range of equipment all ranging from pressure ratings sub 5ksi building above 100ksi.

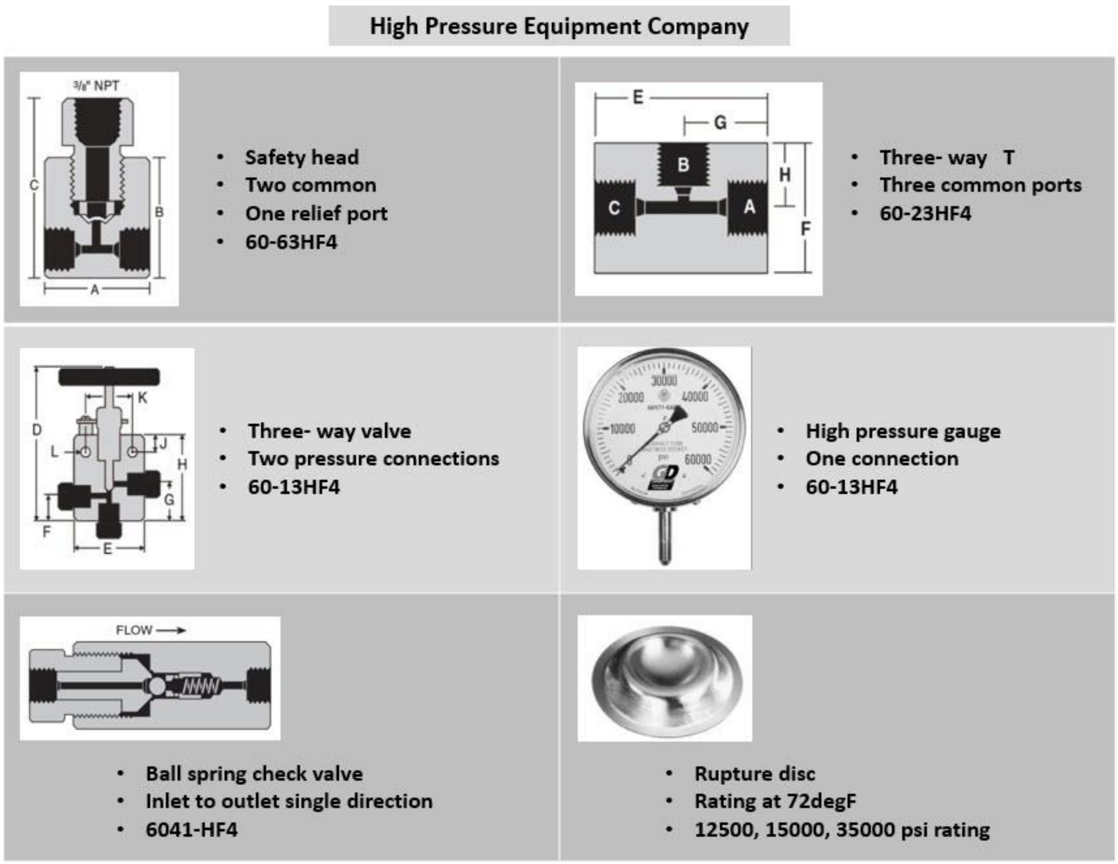


Figure 61: High Pressure Equipment Company component selection summary[21].

Considering the unknowns of developing a novel system such as the figure above outlines chosen key fluid flow components utilized in this system. Here all components are sized to accept a cone and threaded quarter inch outside diameter high pressure rigid stainless-steel tubing. The tubing and all fitting components are rated for 60ksi. For commissioning purposes, a series of different rupture discs were purchased. These discs are solid diaphragm style pressure relief style valve. When acted upon by a pressure rating above that which the disc is rated for the disc bursts and pressure it immediately released. All threaded ports on these high pressure couplings are designed to accept an HF4 autoclave fitting which is a standard 7/16-20 threaded port with a 60deg coned sealing surface at the bottom of the port[21]. These fittings allow the extreme high pressures to be diverted to this small surface area rather than stressing the threaded connection.

3.6 INSTRUMENTATION AND CONTROL

Designing and building a testing apparatus capable of applying high pressure to seals located within a device under testing subjected to rectilinear motion was only one critical function of this research. The variables of force, pressure, and device under testing temperature measurements were essential to the research of characterizing performance characteristics of tested dynamic reciprocating rod/bore seals. To accomplish this a series of interwound components were chosen and the reasoning behind those choices will be outlined in this chapter section. Giving an overview briefly the system input and output functions are functioning in an active feedback loop between a National Instruments data acquisition device and the programming software behind the overall function tree required by the project. Input devices are both sensors and recorded metrics of interest such as force, pressure, and temperature. Outputs are triggers and they cause some component to choose a state of operation.

3.6.1 Load Cell, Hight Pressure Transducer, Thermocouple

To measure the inputs of interest and to actively record the desired input signals for further analysis and post-processing a narrowed selection of components presented themselves. Two components caused some issues, the load cell and pressure transducer. The load cell is needed to send signals while in both tension and compression stroke, where most specialize in a single direction. Pressure transducers operating above 15ksi are typically long lead time items and special builds and only a few were available but output signal choices were of poor accuracy.

A bidirectional high accuracy dual stud pancake style load cell from OMEGA engineering. The LCO3 series load cell full scale output accuracy, linearity and repeatability all below +/-0.15% made this selection the ideal choice for our tension and compression measurements[22]. The output signal of this load cell is measured as mV/V citation for full scale output. Which simply means if you power the load cell strain gauge using 10Vdc and the load cell is rated to 100lbf, at 100lbf exerted on the load cell the output signal gives 10mv/V[23].

Another OMEGA engineering product was found to solve the pressure data acquisition requirement. A very high accuracy pressure transducer capable of recording pressures up to 30ksi, the PX01. This transducer is a simple strain gauge transducer that outputs 4-20ma scaled and mapped to the transducers pressure reading range. At zero psi the transducer reads 4mA and at 30ksi the transducer reads 20mA. Largest deciding factor in choosing this transducer is because of the very high accuracy output, accuracy, repeatability, and linearity below +/-0.05% of full-scale output[24].

The temperature measurement portion of the device under testing was easily accomplished by deciding between type K, and type J thermocouples. J type thermocouples are commonly used in industrial furnaces measuring ranges up to 14000F, having a common range of -346-1400F if thermocouple grade extension wire is used. Type K thermocouples exhibit a higher potential of heat measurement, -454-2300F if thermocouple extension wire is used. With respect to reliability and expense both are equivalent where Type J typically have a shorter lifespan[25]. Type K thermocouples will be the temperature measuring instruments for measurements taken both on the peripheral and also inside the device under testing.

3.6.2 NI USB 6353, LabVIEW

The choices of the process measurement transducers in chapter 4.61. drove the selection of the data acquisition unit. To accomplish complete automation there needed to be a data acquisition unit capable of sending and receiving many digital inputs and outputs in conjunction with analog inputs and outputs. The control device needed to also be able to sample data at high frequencies and have expandability on number of inputs for future research needs.

The choices of transducers brought issue to accomplishing the complex needs of utilizing one device for all input and output needs as required by this research. A solution to condition and altering the transducers signals became the obvious choice. A data acquisition device from National Instruments x-series multifunction flatform was chosen as a good alternative choice to fulfil all needs of the required inputs and outputs multiplicities of function. Knowing that signals could be transformed into a readable mode by the data acquisition device we could then narrow the selection to a specific series within the x-series family.

National Instruments multifunction digital input, digital output device USB-Series 6353 gives 32 analog inputs with a sampling rate of 1.25MS/s, 4 analog outputs, and 48 programmable digital input or output lines[26]. This device is a suitable unit for the projects data logging and open-loop, closed-loop automation control.

Referring to the overall choice and selection of this data acquisition device, not only does it function as a sole source unit for all input and output functions. Readable input for this data acquisition device ranges on the analog input pins +/-10V. Through clever signal conditioning filters, we can take our pressure transducer with an output of 4-

20mA, our thermocouple with a 10mV/C, and the load cell with a mV output depending on excitation voltage and attain a +/-10Vdc signal output, Figure 62.

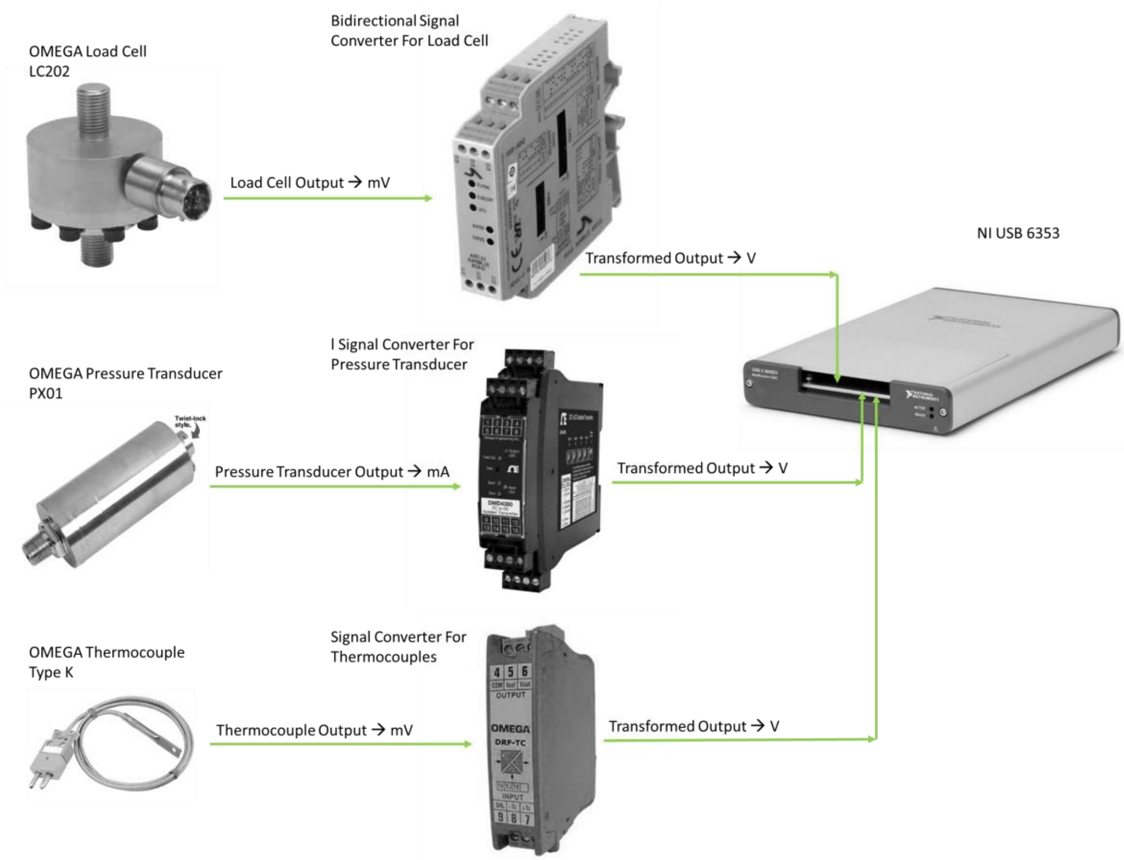


Figure 62: Model view of transducers to signal condition filters to transform signals.

The active feedback loop of the systems automation as described in the introduction to this chapter. This section defines the hardware and software portion of automation. Hardware has been reviewed in detail, the focus now leans towards the software, LabVIEW. LabVIEW is a graphical programming language that focuses on automation in both industry and academic settings. Decision making, automation, reading, and writing data can all be accomplished with understanding the fundamentals of case structures, and software loop execution behaviors. An overview of the front graphical user interface will be shown in, Figure 63.

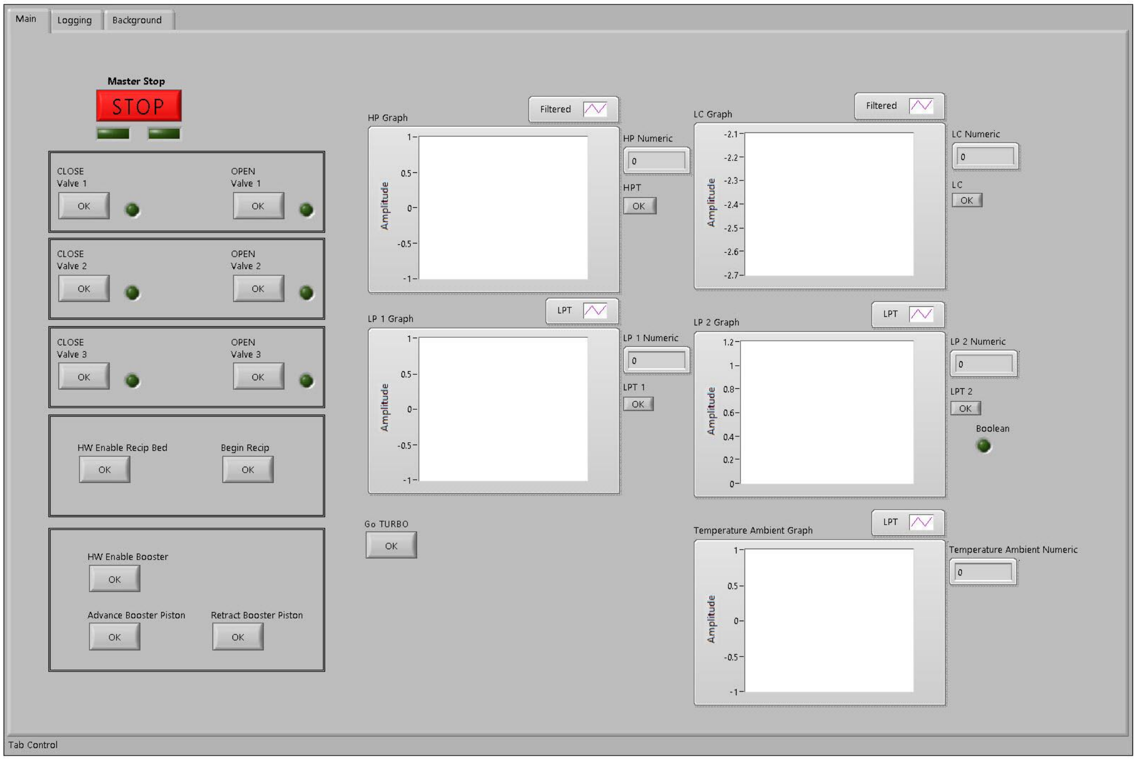


Figure 63: Graphical user interface LabVIEW programming controls.

The interactive graphical interface gives the operator a real time view of the system's status and allows the user to operate the system remotely depending on need. The interface shows current system pressure from pressure transducers, force felt by load cell, and temperature from thermocouple inputs. As an added addition to the software side, there is a background toggle switch that gives the ability to use low pressure transducers. This bonus is a future proofing modification in case of low-pressure testing or monitoring of low-pressure leak chamber.

Top left-hand corner of the interface controls the automated valves, opening and closing of the valves is accomplished here. When valves reach their respective open and closed limits the user is notified by a green light showing movement competition. The operator has the ability to disengage system operations at any time, hardware enable can be depressed to cancel the operation of either the intensifier or reciprocating motion of rod. Pressure intensifier can be manually retracted and advanced using associated piston booster buttons. Programmed into the non-volatile memory of the servo controller controlling the reciprocating bed, there are a series of prewritten programs which can be called and then automatically executed by latching the begin reciprocation button.

At any time, the operator can press the master stop button to fully shut down system. During the master stop shutdown, a series of steps are executed in sequence. The program looks firstly at load cell forces and pressure values, the reciprocating motion will be killed first then the program will prompt the operator if depressurization is required. If depressurization is required the program opens the automatic valve, venting fluid to the containment container. If no loss in pressure is required, the program dwells allowing remaining data to pass through the queue and be written to file before shutting down fully.

Data acquisition process and writing process is accomplished using a multitier producer consumer formatted data queuing structure. A producer consumer structure is dual loop structure where the consumer structure runs continuously as fast as it is allowed. The consumer structure acquires the data at very high frequencies and then passed that data into a queue[27]. The queue then feeds the consumer loop which filters data and writes it to the user interface to be used in graphs and indicators. Before filtering the data in the consumer loop the data is written to a file to be analyzed and postprocessed statistically later.

3.6.3 Kollmorgen AKD Servomotor Drives

Kollmorgen servomotors and servomotor drives are the main movement performers and control both the linear movement of the rod on the reciprocating rod test bed and drive the ball screw jack on the pressure intensifier assembly. Kollmorgen servomotor and servomotor controller selection on the intensifier assembly drove the decision to replicate similar choices for the driving of the linear ball screw assembly on the reciprocating rod test bed. Keeping within the Kollmorgen family allows for easy transfer of knowledge and adds some symmetry to the project. Additionally, from a commissioning of project standpoint, component pinouts and wire routing diagrams are known making for a more seamless and succinct process flow.

Reciprocating Rod Test Bed Combination		**Pressure Intensifier Combination**	
			
AKM54H-ACC2C-00	AKD-T00607-ICAN	AKM44H-ACCNC-00	AKD-T00607-ICAN

Figure 64: Kollmorgen component selection review for experimental testing setup.

The Kollmorgen AKD servomotor controller units are one of the forefront leaders in motion control and are commonly chosen over competitors because of the ability to upgrade controllers to The BASIC package. The BASIC package gives precise motion control using Kollmorgen's proprietary BASIC motion control programming language[28]. The Kollmorgen BASIC software allows writing and compiling of advanced movement and motion control and is an additional feature provided on the AKD-T00607-ICAN series controllers. The identical the AKD controllers are sized to function off both an input voltage of 208-220Vac and 440Vac three phase. The two different input voltages allow an output DC bus voltage of either 320Vdc or 440Vdc respectively. The different DC bus voltages then supply the servomotors and allow for different performance characteristics such as locked rotor torque, running torque, speed, and current draw. This added measure is to improve adaptability of the system and to prepare for future improvements or modifications to the device under testing.

Looking specifically at the servomotor controller operating the linear motion of the reciprocating rod, a tailored motion control program can be made based on the needs of a specific experimental matrix of design of experiments. A program is created within the Kollmorgen code compiler and uploaded to the non-volatile memory of the controller. Parameters of interest such as linear speed, acceleration, deacceleration, and linear tracking can be altered for the reciprocating rod, Figure 4.

AKD BASIC Parameters

```
'--------------DeviceParams-----------------------
Params
        DRV.OPMODE=2'positionoperationmode
        DRV.CMDSOURCE=5'commandsource=AKDBASICTG
        UNIT.PROTARY=4'16bitpositionunits,65536counts/rev
        UNIT.VROTARY=0'velocityunits=rpm
        UNIT.ACCROTARY=0'accelerationunits=rpm/sec
'--------------Define(dim)GlobalVariables--------
AliasONE_REV=65536
DimLOOP,COUNT,RunSpeed,ReverseSpeedasInteger
'--------------MainProgram-------------------------
Main
        'Settingupmoveparameter
        Move.Acc=10000
        Move.Dec=10000
        Move.RunSpeed=100
        DRV.EN
        drv.swenable=1
        RunSpeed=100
        ReverseSpeed=50
        While(1)
                'DI-#1=HandheldToggleEnable
                'DI-#2=HandheldToggleOUTButton
                'DI-#3=HandheldToggleINButto
                'DI-#4=LabviewTriggerStartRecipMotionRelay1
                'DI-#6=LabviewTriggerCWMoveRelay2(NotUsed
                'DI-#21=LimitSwitchSoft@Side1=ClosestToServo
                'DI-#22=LimitSwitchHard@Side1=ClosestToServo
                'DI-#23=LimitSwitchSoft@Side2=FurthestFromServo
                'DI-#24=LimitSwitchHard@Side2=FurthestFromServ
                'MoveshaftINTODUT"ManualINButton"
        If(DIN1.STATE=1)And(DIN4.STATE=0)And(DIN2.STATE=0)And(DIN3.STATE=1)And(DIN21.STATE=0)And(DIN22.STATE=0)And(DIN23.STATE=0)And(DIN24
.STATE=0)Then
                        CallS2_RotateIN
                'MoveshaftOUTofDUT"ManualOUTButton"
        Elseif(DIN1.STATE=1)And(DIN4.STATE=0)And(DIN2.STATE=1)And(DIN3.STATE=0)And(DIN21.STATE=0)And(DIN22.STATE=0)And(DIN23.STATE=0)And(DI
N24.STATE=0)Then
                        CallS3_RotateOUT
```

Figure 65: AKD BASIC operational parameter setup example.

Notice in Figure 65, the program has outlined operating parameters and has defined digital inputs. These inputs function as the instigator to some desired action by the feedback system. The digital inputs defined and stored in the AKD servomotor driver for the reciprocating bed receive inputs from a handheld manual override controller, inputs from LabVIEW, and limit switches. The limit switch input, and handheld toggle inputs will be overviewed in the 'Safety' chapter. LabVIEW communicates with the controller by sending signals to digital input four and six. Digital input four calls a subprogram pre-programmed in the AKD to start the automatic reciprocating motion of the rod, full program can be seen in the appendix. Digital input six in this program is not used and can be adapted for future use if needed. Originally digital input six was meant to be another subprogram call to start automatic reciprocation of the rod but differing from the subprogram called using digital input for because the start direction would be reversed. Simply, digital input four would start motion in compression stroke and alternatively digital input five would start the motion in tension.

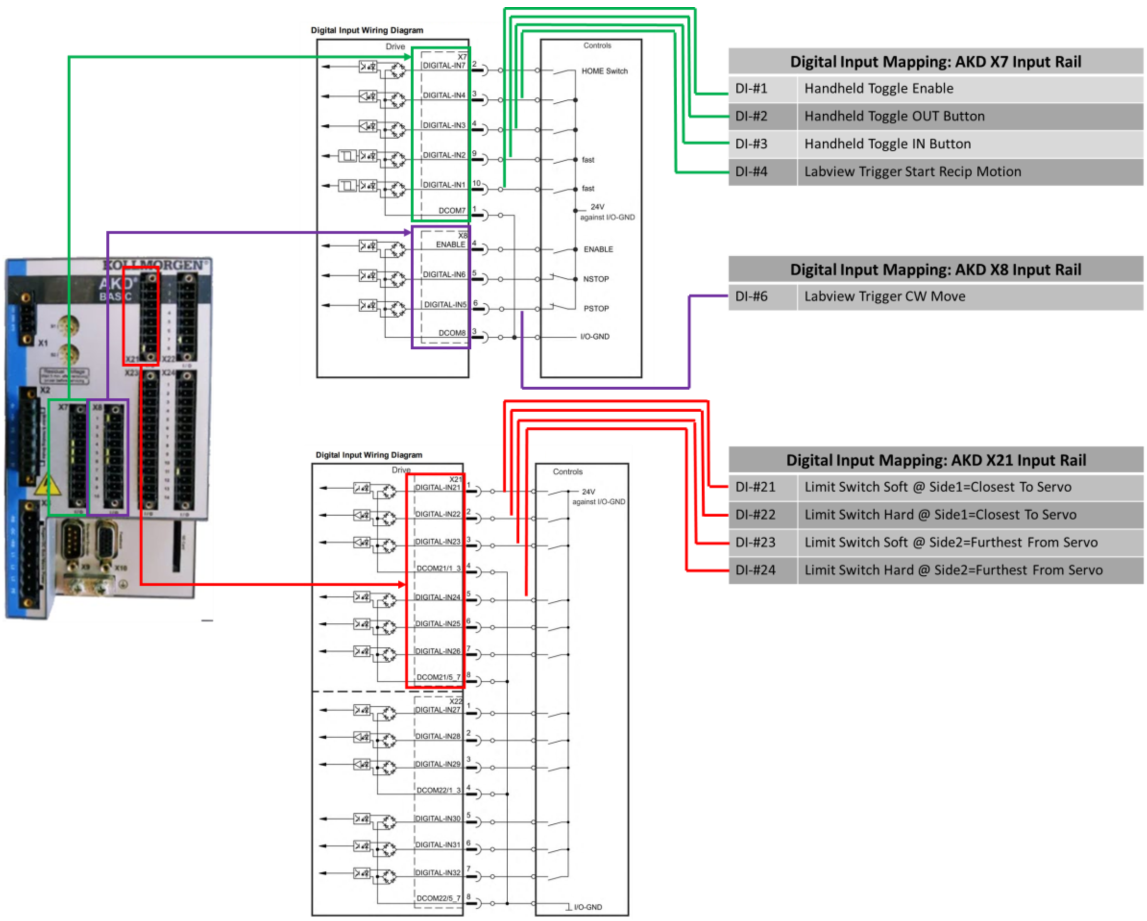


Figure 66: Digital input mapping to Kollmorgen AKD X7, X8, X21 input rails for AKD controlling the AKM44H.

The Kollmorgen the AKD-T00607-ICAN servomotor controller has four input rails. The X7 rail contains input pins for digital inputs 1-4, and 7. The X8 rail has input pins 5, 6, and has a special hardware enable pin at location 4, input rail X21 maps pins 21-26, and the X22 rail maps pins 27-32, Figure 66.

Considering that both the reciprocating rod test bed and the pressure intensifier assembly share the same servomotor controller the pinout remains the same. For completeness, Figure 67 shows the digital input mapping for the AKD-T00607-ICAN controlling the pressure intensifier assembly.

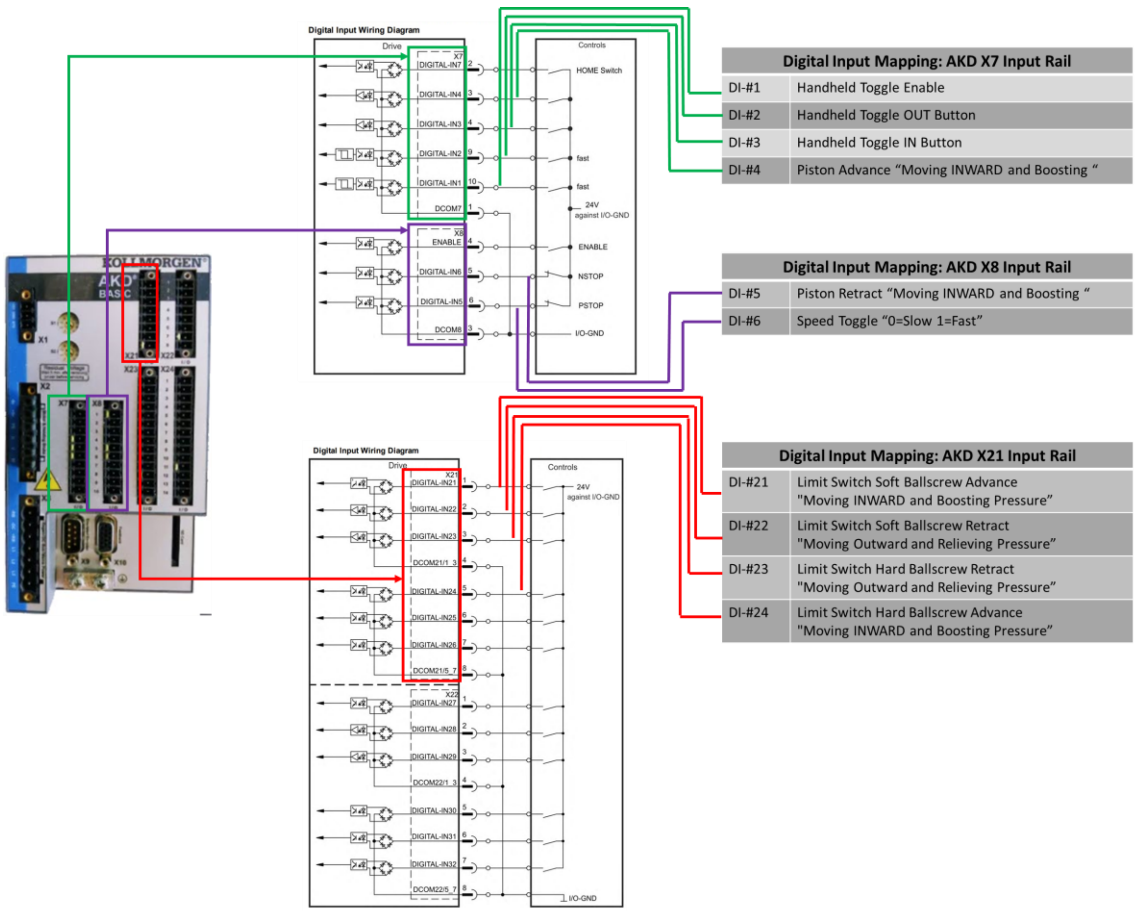


Figure 67:Digital input mapping to Kollmorgen AKD X7, X8, X21 input rails for AKD controlling the AKM54H.

Difference between digital input mapping between the two different servomotor controllers is minute and differ only because of desired functional needs of each subsystem. Duplicate mapping is seen in both the X7, and X21 input rail. The handheld toggle and limit switch mapping do not vary. Differences are concerned with digital inputs 4-6 where the definition of these inputs relate to the speed of the servomotor and direction of rotation.

3.6.4 Kollmorgen P70530 Stepper motor Drives

The Kollmorgen P70530 stepper motor drives are a series of micro -stepping servo drivers in Kollmorgen's P7000 series stepper motor controller products. This stepper motor powers any traditional four wire stepper motor. Power outputs are any voltages between 48-75Vdc and can supply up to 10amps of continuous current[29]. The drive can receive any 5Vdc TTL logic signal and is capable of sourcing up to 5Vdc at terminal J4-25 with a current load sourcing of 50mA. The J4 input and output command port is a high-density D subminiature female serial connector and is capable of receiving and sending various commands on pins J4-1 → J4-26.

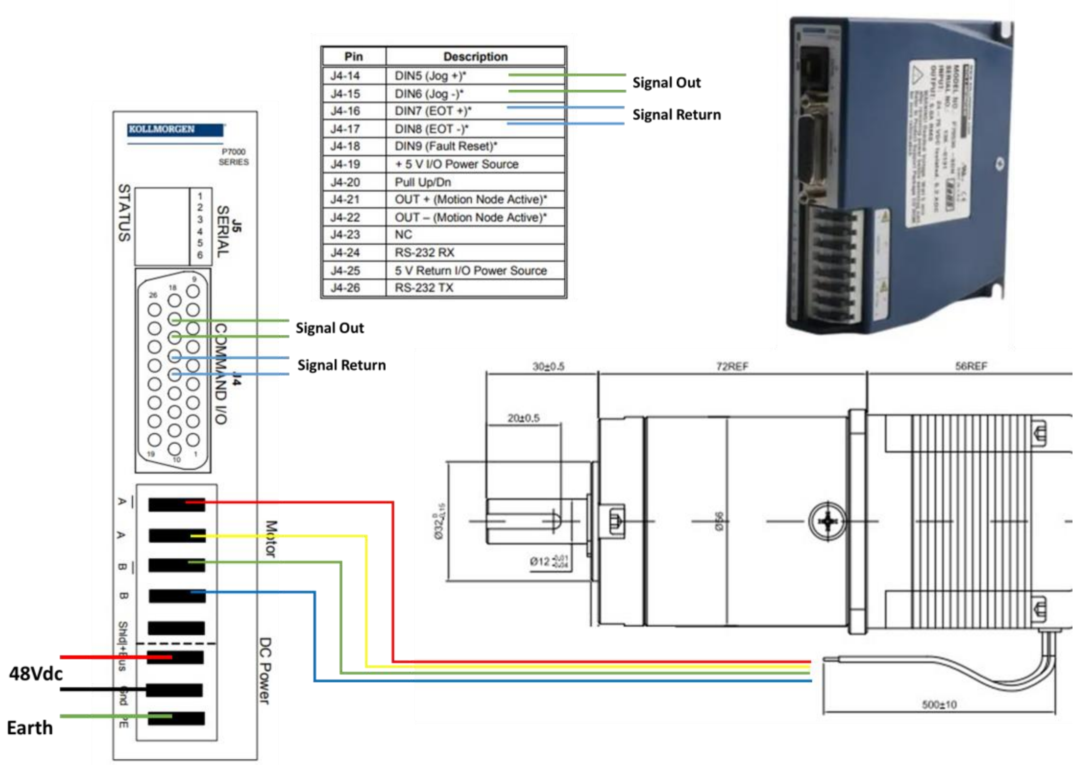


Figure 68: Kollmorgen P70530 DC micro-stepping stepper motor controller pinout.

The system utilizes three of the Kollmorgen P70530 micro-stepping stepper motor controllers. Controllers power and control three identical Nema-23 gearhead stepper motors. These stepper motors automatically actuate needle valves used in the fluid process flow and will be discussed more in a later chapter. The controller's operations were setup in the P7000 setup software and then downloaded to the controller's memory. The first operation of interest for the function used in this setup was to jog the motors forward and in reverse (ccw/cw shaft rotation). Secondly the end of travel signal return signal. The NI-6353 data acquisition device sends a digital signal to the controller specifying either a forward jog or a revere jog. Movement completion and shaft rotation is ceased when the end of travel limit switches are tripped. Tripping of end of travel limit switches send a 5Vdc TTL logic signal back to the data acquisition device signaling to terminate action and notify the operator via a indicator on the graphical user interface.

3.7 ELECTRICAL SYSTEM

Division of power to the system is done through wire runs and routing within the electrical enclosure. Primary power required to turn on sensors, data acquisition device, external pawer supplies, servomotor controllers, stepper motor controllers and automatic valves is controlled by a standard power switch which excites all components running off 120Vac power. High voltage three-phase 208-220Vac power enters the electrical enclosure via a disconnect throw switch mounted to the enclosure. The large, powered components, heating circuit, and servomotors require the large 208-220Vac three-phase. The servo motor controllers receive the three-phase power and convert that power to direct current voltage with a series of rectifiers within the controllers. The heating circuit connection takes two of the three phase legs and runs that power to a step-up transformer. The step-up transformers ratio is 1:2 at 15kVA and therefore gives a output voltage of approximately 440Vac. Transformer was sized and connections routed to accept high wattage heating rods or bands.

To keep inrush currents manageable, inline fuses are located along each of the three. Three phase power legs inside the disconnect box and are each rated at 30amp. Smaller 120Vac power current is regulated at the switching box routed to the outlet plug inside the electrical enclosure, in series, and before power is routed to the 120Vac power distribution bus bar plate.

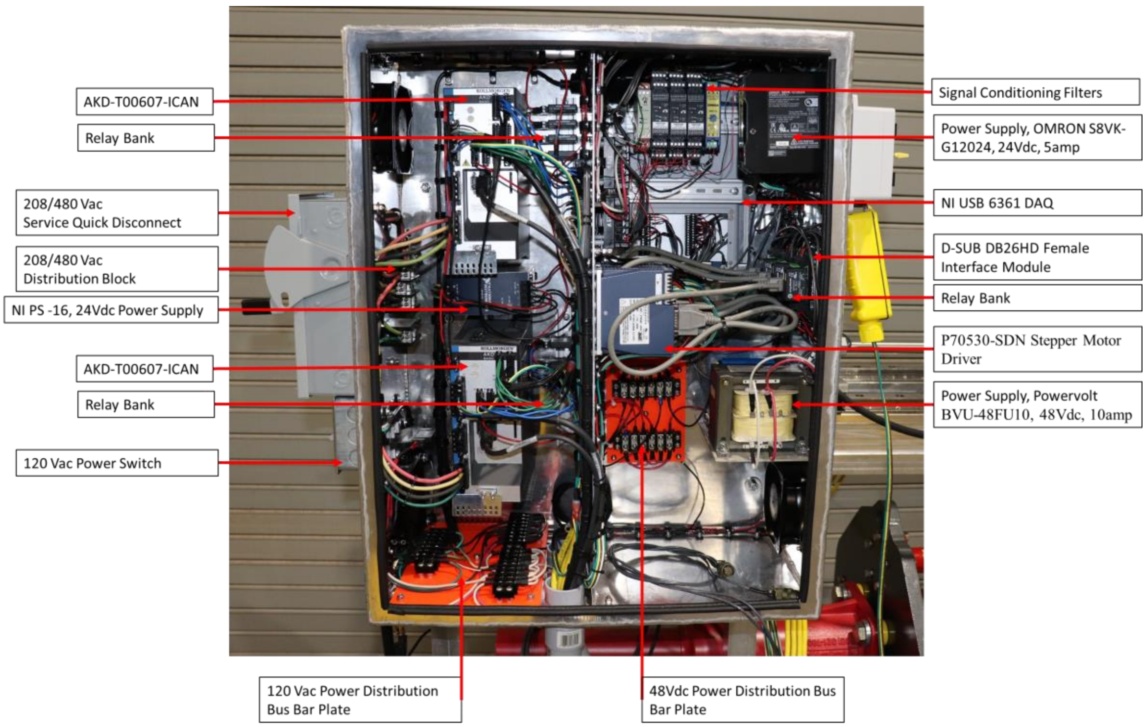


Figure 69: Electrical enclosure AC powered components.

The above figure illustrates the large components drawing and supplying power inside the electrical enclosure. Starting with the mapping of the 120Vac power. The 120Vac enters the left-hand side of the electrical enclosure then is routed down to the 120Vac power distribution bus bar plate. This bus bar plate branches power to a few Vac-Vdc power converters. Powervolt 120Vac-48Vdc power supply sends power to the stepper motor controllers and enclosure fans, NI 120Vac-24Vdc power supply powers the Kollmorgen AKD servomotor controllers. The NI 6353 receives power from this bus

120Vac bus bar. The 120Vac-24Vdc OMORON supply powers the signal conditioner filters for the transducers. The NI PSS-10 120Vac-10Vac supplies the load cell and the NI PSS-12 powers the high-pressure transducer.

Large power, 208/480 Vac entering the left-hand side of the enclosure takes a much simpler route. The leads enter inside the enclosure and are routed to a distribution block located in close proximity to the AKD servomotor controllers to keep wire lengths and voltage drop minimal. After entering the distribution block the lines split and branch directly to the servomotor high voltage input terminals.

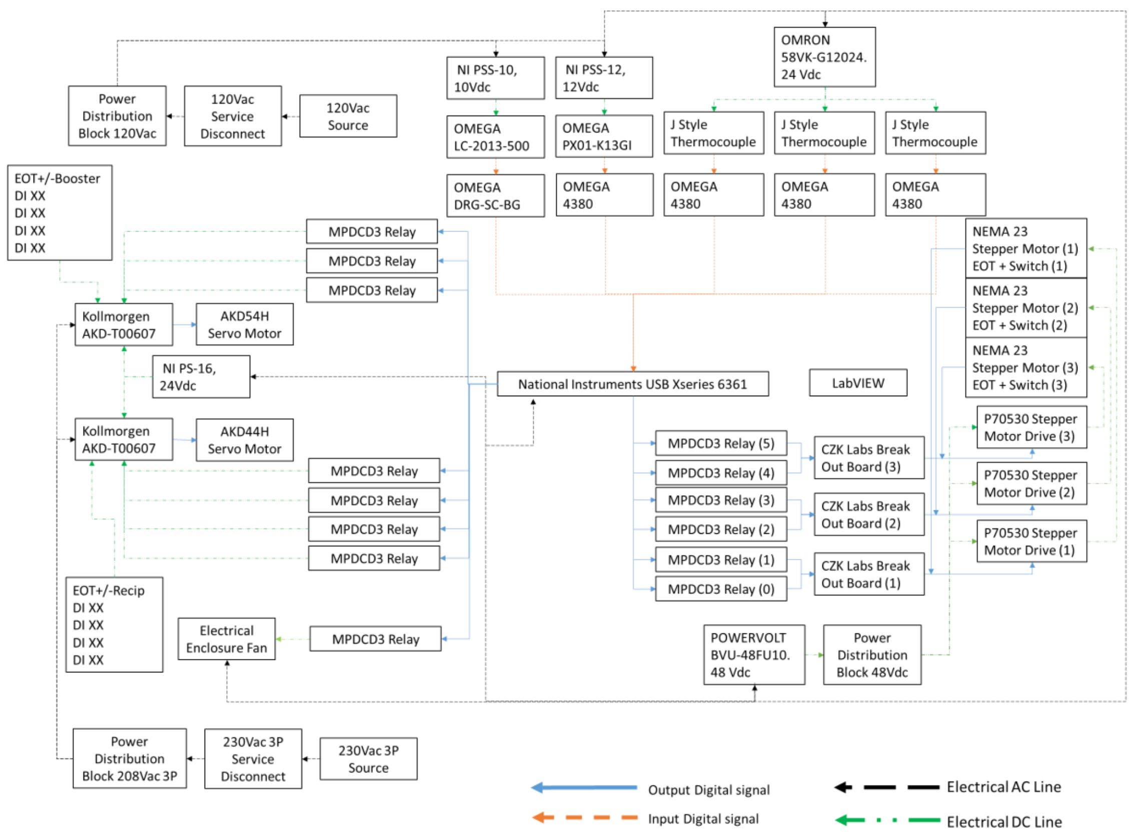


Figure 70: Diagram of power and signal routing inside electrical enclosure.

Tracing of power distribution and signal distribution can be seen in Figure 70. Shown above marks an important note to how the AKD servomotor controllers receive signal feedback at inputs. The controllers run on 24Vdc machine logic[16]. In order to make the signal sent by the data acquisition device more powerful, the data acquisition device is simply triggering a relay bank connected to machine logic of 24Vdc. This is how the data acquisition device communicates with the servomotor controllers.

3.8 CHAPTER SUMMARY

The detail design process was presented in great length in chapter four. This chapter outlined the data driven decision making process of the selection of key components needed to accomplish research objectives and gave insight to the manufacturing of select components. Original ideas were presented and reasoning behind choices were discussed. Chapter four brought to competition the morphology of design process outlined in chapter three by completing the design process, outlining subassemblies, and component functions.

The subassemblies presented in this chapter included the pressure intensifier assembly, the reciprocating rod test bed, frame, piping and instrumentation, and electrical subsystem. Within the pressure intensifier assembly analysis was conducted on the intensifier cradle, intensifier pressure housing and intensifier piston. Operation of the pressure intensifier was presented, and choice and sizing of driving components were given. Development of the reciprocating rod test bed took place in this chapter with corresponding structural analysis and functional operation explanations. The experimental setup frame was presented and associated expected stress and deformation analysis shown. Critical supporting subsystems including, piping and instrumentation and the electrical system were defined. This chapter gave a micro scale delineation of the overall holistic view presented in chapter three and concludes the chronological structure of the morphology of design process.

CHAPTER 4: SAFETY

4.1 INTRODUCTION TO SAFE OPERATIONS

Operational safety is essential to keeping the health and well-being of any operator or technician using industrial equipment. Keeping in mind that this experimental testing setup has the potential to develop extremely high hydraulic pressures and the subassemblies built and designed start and stop automatically, safety measures need developed. Three large areas of interest specific to this experimental testing apparatus can be split into a couple categories. Category one concerns itself with motion control and how to implement some best practices safety measures. Category two can be classified as fluid flow control and how safe remote operation is achieved.

4.2 Linear Motion Travel Limit Switches

Common to many automated pieces of equipment the utilization of motion safety features are implemented to minimize potential machine and operator damage and harm. In the instance of linear transversal motion deployed in the experimental seal testing fixture represented here, there are two specific motion limits of interest. The pressure intensifier unit increases pressure by the driving of a piston into a blind bore cylinder and the reciprocating rod test bed constrains linear movement of a device under testing.

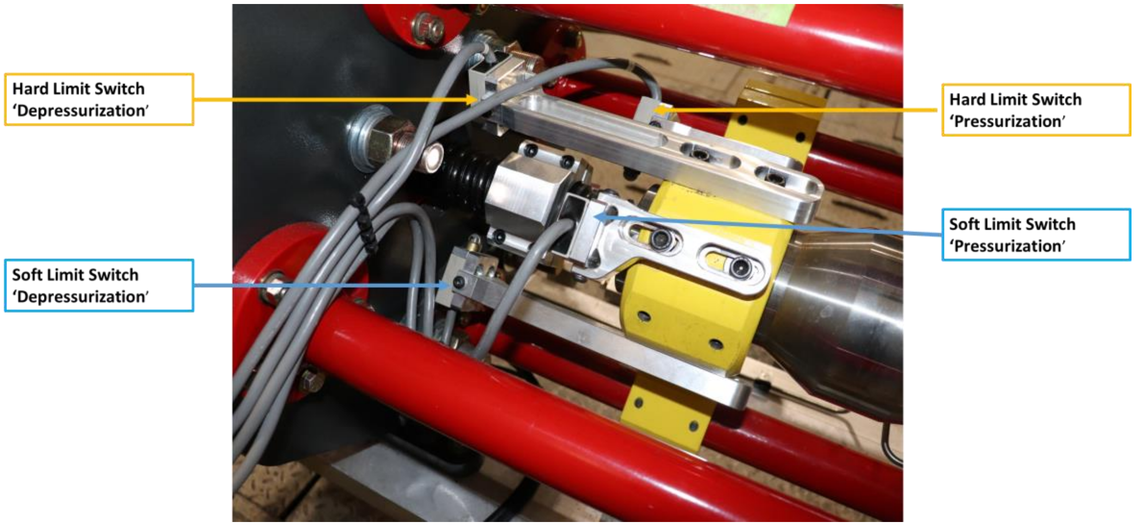


Figure 71: Motion limit switches pressure intensifier.

Operational routines for the pressure intensifier are defined in Kollmorgen's AKD BASIC code and control the maximum working conditions of the pressure intensifier. The functioning window for the movement of the piston inside of the intensifier assembly is determined by four different end of travel limit switches. Quantity two switches are in place for the haptic feedback of full extension of the pressurization stroke and quantity two limit switches give input to the completion of the depressurization stroke. For both the pressurization and depressurization movements, a soft limit switch is implemented.

The functionality behind the soft limit switch is solely to send a signal to the servomotor driver communicating to the driver that no more advancement or detraction of the piston can be completed. After this signal is received the controller transmits a signal to the data acquisition device prompting the user that the intensifier has reached its functional limits. In the case that the first limit switches fail mechanically there are a secondary pair of limit switches. This secondary pair of switches are a worst-case scenario dead man trigger, Figure 71. In the case that there exists some delay in the feedback loop between the servomotor controller and the data acquisition device or in the instance of mechanical failure of the soft limit switch, a hard limit switch is present. The hard limit switch acts as a power disruptor. After this switch is tripped the power to the servo is disconnected and all movement ceased.

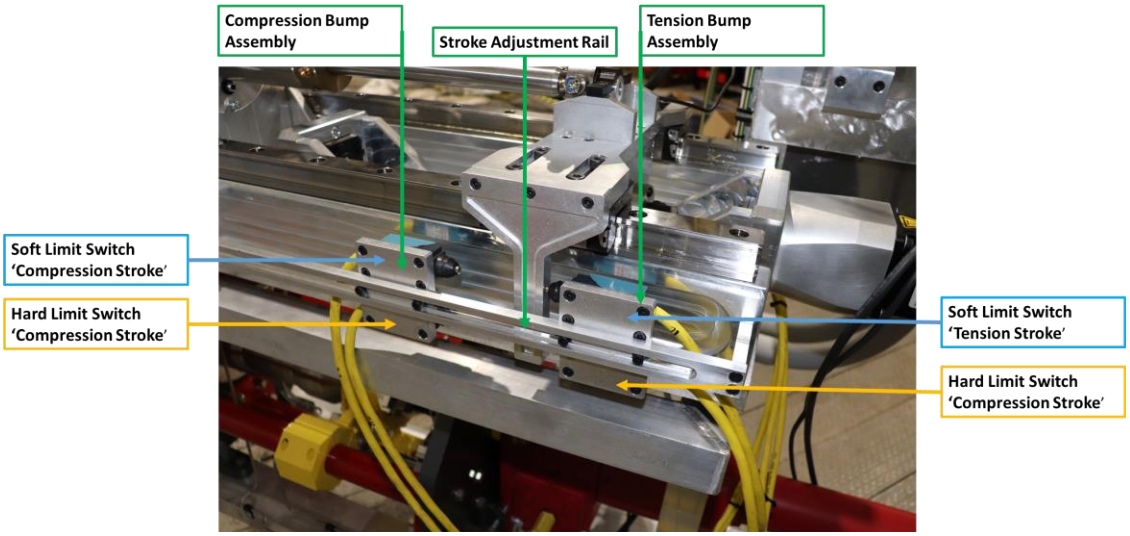


Figure 72: Motion limit switches reciprocating rod test bed.

Analogous to the methodology behind the safe operating parameters of the pressure intensifier, the reciprocating rod test bed employs similar motion limits. The overall length of compression and tension stroke are determined by the placement of limit switches on a stroke adjustment rail. This rail allows for a variable stroke length of zero to seventeen inches. Adjustment and measuring is simple, two set screws per bump assembly can be loosened and then the assembly slides along the adjustment rail. Depending on the stroke length addressed by the design of experiments, a simple measurement can be taken between the upper limit switches which ultimately will control the length of linear travel.

Located on the bump assemblies towards the top are the soft limit switches, Figure 72. These limit switches control the stroke length of the reciprocating rod. The servomotor controller governing the rectilinear motion of the reciprocating rod is programmed to execute a series of functions. Predominantly the simplest of uses are to govern the reversal of motion during actuation. The controller receives a signal when soft limit switches are depressed, and after this signal is received the direction of the reciprocation rod is reversed. This reversal of motion in both the tension and compression stroke create a left to right shuttling motion.

Repeating a parallel procedure like that discussed from the pressure intensifier the reciprocating rod test bed contains another duplicate pair of travel limit switches, the hard travel limit switches. Unlike the hard travel limit switches used and programmed in the pressure intensifier these switches do not immediately terminate power to the servomotor controller. The hard travel limit switches used in the reciprocating rod test bed actuate the rod in a reversal motion before power is disconnected. Simply, if the rod is traveling in either tension or compression, and the soft limit switch fails, the hard travel switch is triggered. Following the triggering of this hard travel limit switch motion is reversed and the rod travels in an opposing direction for three inches before the servomotor controller flags a hardware disable signal terminating movement.

4.3 Shear Couplings

Common to a wide arrangement of rotary power transmissions there exists a need to adapt the output shaft of a driving device to the input shaft of a corresponding driven device. Traditionally a simple solution in the form of a coupling device is used. This device can come in many different forms and size factors depending on the specific needs of the power transmission required. Some instances require a specialized component to protect the power unit or the unit under power. Major concerns with rotational equipment is the possibility of binding or fundamentally an instance of mechanical friction between power and powered units. Helping to minimize overall irreversible failure of the power train, a specialized ‘shear coupling’ was developed in the early nineteen fifties. Over the decades the shear-based coupling has been refined and developed to solve not only torsion issues, but also issues of vibration dampening.

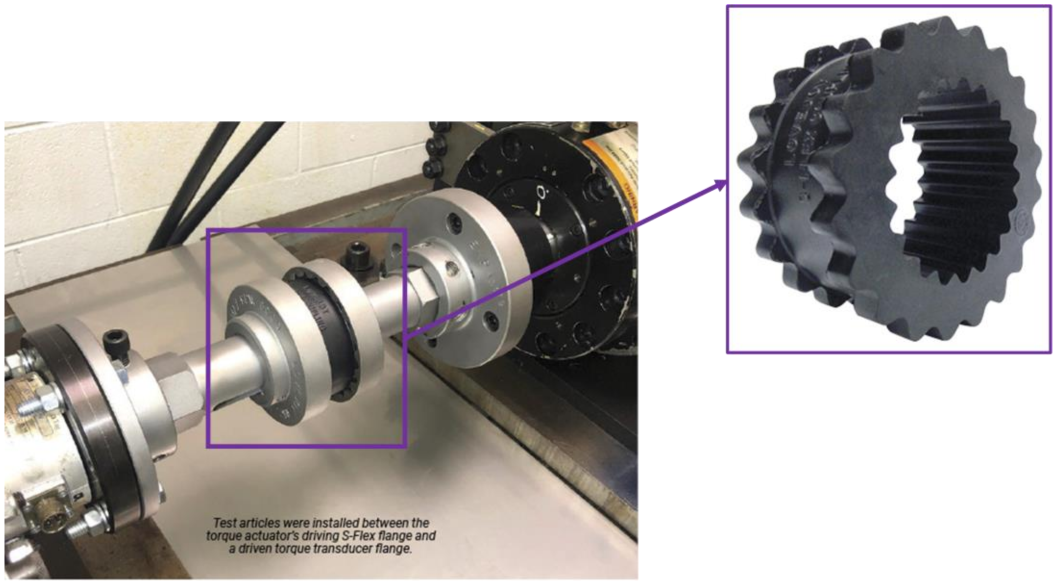
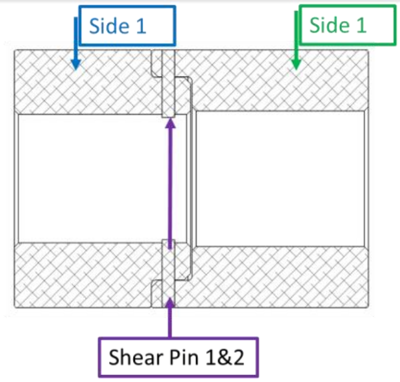


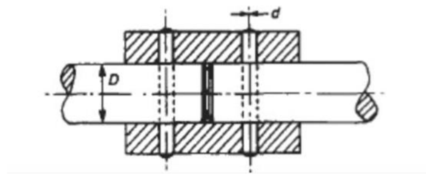
Figure 73: LoveJoy elastomer shear coupling[30].

Initially designed specifically for pumping applications, the LoveJoy shear coupling utilizes a elastomeric buffering insert between power shafts. Rigid couplings are sized to accept the respective input and output shafts and joined by a disposable interim sleeve component. The LoveJoy company has pioneered the area of shear couplings, and due to the new flanges insert design have both accomplished mechanical rigidity and vibrational resistances caused by unbalanced inertial fluctuations.

Unfortunately, due to space constraints and shaft sizing these LoveJoy couplings could not be utilized on either the reciprocating rod test bed or the pressure intensifier. Consequentially a more tailored design was needed. The decision to create a pin-shear coupling was determined to be the best route.

Shear Coupling 'A' AKM44H → Linear Ball Screw	Shear Coupling 'B' AKM54H → Ball Screw Jack	Coupling Assembly Cross Section
		

Two different coupling assemblies were developed for the experimental testing fixture. Coupling assembly 'A' joins the AKM44H to the linear ball screw used to move the rod through its rectilinear motion. Coupling assembly 'B' connects the AKM54H to the ball screw jack of the pressure intensifier assembly. Both couplings are two-piece assemblies. The assemblies consist of two through bore cylindrical parts. Bores are tolerance to match a slip fit of the corresponding input shaft and are then broached for each respective keyway. Side one of the coupling assembly's telescopes inside of a counter bored face of side two. Located on the counterbore lip exists two through holes orientated one hundred and eighty degrees from each other. These holes align the two halves of the coupling assemblies and join them with a pressed fit shear pin.



$$Torque = \pi \frac{\tau}{4} d^2 \mathrm{D}$$

D = Shaft Diameter
d = Pin Diameter
T = Torque Capacity
τ = Allowable Shear Stress

Figure 74: Pin sleeve shaft allowable torque equation[31].

The shear pin size for each respective coupling assembly is determined by using the pinned sleeve shaft coupling torque capacity equation from the mechanical engineer's handbook, Figure 74. Material selection for the pin was chosen to be PEEK because of its ability to be readily purchased as a direct to consumer off-the-shelf good pre-made in various dowel pin sizes. PEEK is a high strength engineered plastic with good dimensional stability at varying temperatures and has a very low moisture uptake. For this material we will reference published sheer strengths and take the upper limit for calculations to remain conservative[32].

Known Values For AKM44H andAKM54H Servomotors			
	Shaft Diameter	Torque Capacity	Allowable Shear Stress
AKM54H	1.495	186in-lb	10ksi
AKM44H	1.495	44in-lb	10ksi

Figure 75: Known values for shear pin sleeve shaft calculations.

Referencing from the previous chapter, Figure 75 the largest expected output torque of the servomotor driving the pressure intensifier was 186in-lb and the expected driving force of the servomotor powering the reciprocating rod test bed was 44in-lb. Using the known values of maximum expected, the value of allowable shear, and the shaft diameters we can rearrange the equation seen in Figure 74 and solve for the shear pin diameter. Note that the shaft diameters were designed to be identical to be manufactured as a batch reducing cost and machine time. The only differences between coupling mated parts are the bore diameters respectively.

$$\sqrt{\frac{4T}{\pi\tau D}} = d$$

$Shear\ pin\ diameter\ AKM54H..$ $$\sqrt{\frac{4 * 186in - lb}{\pi * 10ksi * 1.495in}} = d$$

$$\sqrt{\frac{744in - lb}{46943psi}} = d$$

$$.1256in = d$$

$Shear\ pin\ diameter\ AKM44H..$ $$\sqrt{\frac{4 * 44in - lb}{\pi * 10ksi * 1.495}} = d$$

$$\sqrt{\frac{176in - lb}{46943psi}} = d$$

$$.061 = d$$

From the populated shear pin diameters calculated above, diameter selection for the pins used in each coupling assembly can be selected. In the case of the coupling assembly used on the AKM54H, a shear pin diameter of 0.125inch was chosen and for coupling assembly associated with the AKM54H, a pin diameter of 0.0625inch was selected. The function of these pins are a single use item. In the event of binding, pinching, or buckling of powered mechanical components, these pins break allowing the subassembly to be flexible. After pins are sheared the two component coupling subassemblies can rotate freely and individually and driven components will become stationary until coupling assembly is repaired.

4.4 Automated Needle Valves

Fluid control in industrial systems can be controlled in a variety of different ways and multiple different control options are available. One of the most crucial centers of a system which routes fluid is the selection of valving. Common styles of valves such as globe, gate, needle, and butterfly are often used. Selection of the valve style is specific to the application and flow regime. When dealing with extreme pressures such as those generated by this experimental setup only one true valving option exists. The solid body precision milled autoclave needle valves from High Pressure Equipment Company are some of the highest quality and best performing valves available.

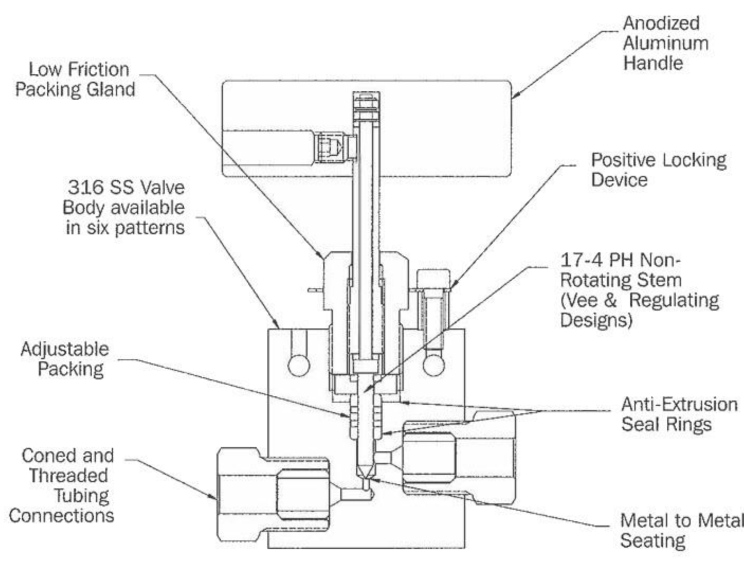


Figure 76: Cross sectional view of high-pressure needle valve.

The needle valves from High Pressure Equipment Company are constructed to create a true seal by disrupting fluid flow using a rotating stem which makes a metal-to-metal mechanical seal in the base of the valve housing, Figure 76. All needle valves from this company follow a similar design and adopt autoclave threaded ports for tubing. One of the most unique design features is the adjustable packing around the valve stem. This packing helps to centralize the stem inside the valve body and controls the amount of torque required to twist the handle for the opening and closing of fluid flow.

Applications for these needle valves are typically in a setting where the valves are manually actuated. Manual operation of these valves does not adhere to the objectives outlined by this research project, therefore, some solutions needs applying. The largest issue with automation of these valves is that a very slow opening is expected to be needed to slowly relieve pressure in the system. If pressure is relieved too quickly there is a risk of seal blowout and can possibly cause damage to the containment reservoir.

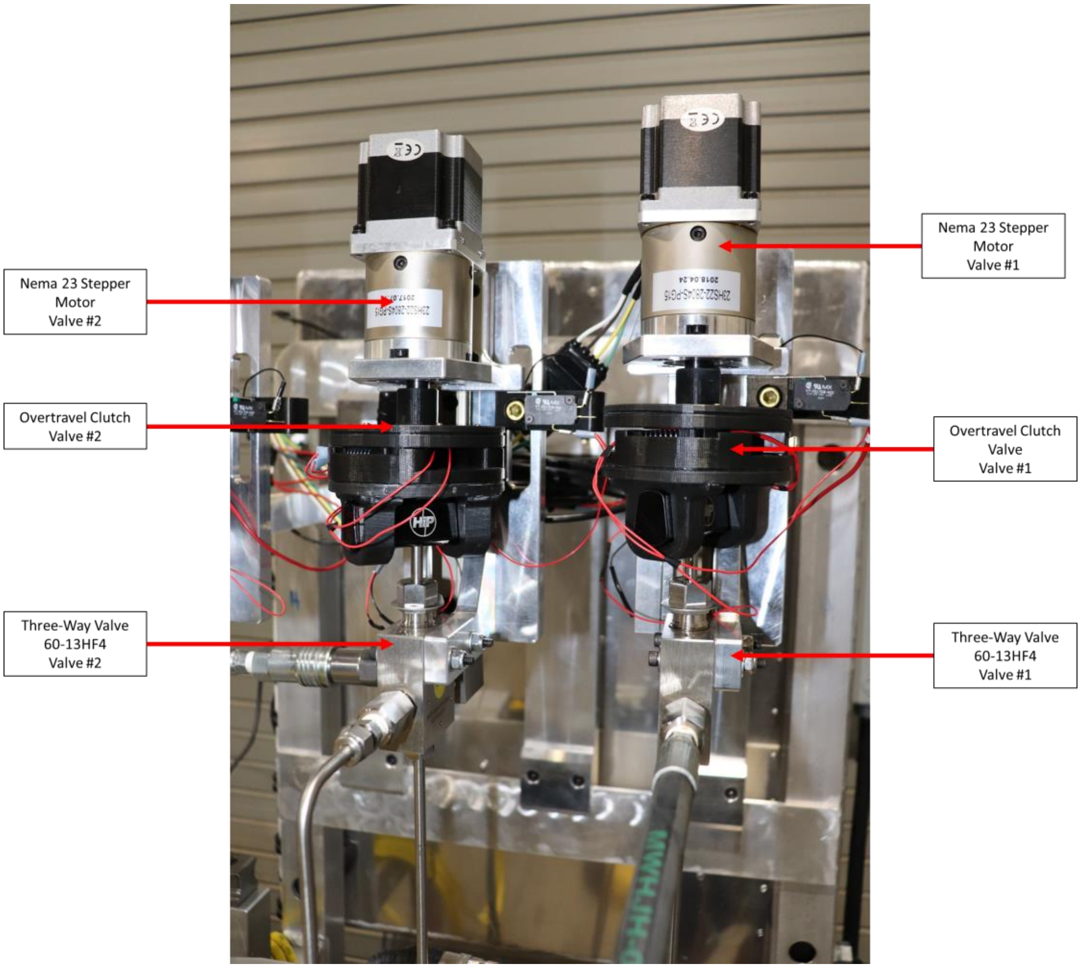


Figure 77: Automated stepper driven valves with overtravel clutch.

Finding no feasible readily available automated valves on the market the above solution developed. The solution uses needle valves from High Pressure Equipment Company, NEMA 23 stepper motors, and uniquely designed overtravel clutches. Process of reasoning behind this automated valve is that the stepper motors have the ability to very slowly open the valves while keeping valve stem movement smooth without coaxial loading.

The stepper motors are controlled by a Kollmorgen micro-stepping, stepper motor controller and is capable of speeds as low as one-hundred arc minutes. Between the stepper motor and the valve stem handle there is in place a type of rotary clutch. This clutch operated by acting as a torque limiter in the clockwise rotation direction. While the valve is being closed the spring clutch internals remain inactive. When the valve becomes properly seated at the end of its rotational movement, the clutch springs engage and apply a torque of 20in-lbs to the valve stem. After the valve is torqued shut the clutch engages a limit switch which completes a loop back to the stepper motor controller triggering the controller to terminate movement. Following the termination of movement, the controller sends a signal to the data acquisition device illuminating a indicator on the graphical user interface panel. This automated valve assembly design allows safe remote operation of the system and reduces the chance of seal blow-out due to rapid depressurization.

CHAPTER 5: CONCLUDING REMARKS

5.1 SUMMARY OF THESIS

This thesis takes a detailed view into the design and commissioning of a experimental testing fixture to test rectilinear seals at high pressures while operating through variable operational conditions. Parameters of interest from the principal investor was force required to reciprocate rod, pressures seals can operate under, and temperatures seals can be exposed to.

The research began with the defining of exact operational needs of the system. After understanding the desired output of the system, the conceptualization process where more detail to ideas to solve the research problem were generated. Regular meetings with the principal investors drove the decision-making process toward a specialized design. The design utilizes a custom designed hydraulic fluid pressure intensifier to create downstream pressures in excess of 20ksi. This intensifier is the prime mover in the system and acted as the primary early focus of the research.

The seals used in this testing setup are to be analyzed while the system is moving a rod through the device under testing. In order to move this rod in a reciprocating motion a test bed needed to be developed. As a result, a reciprocating rod test bed consisting of a powered ball screw to initiate linear rod movement came to be the best foreseeable option. This test bed contains linear guide rails attached to rod carriages keeping the rod perfectly linear through compression and tension strokes. Attached to this rod in line with the drive assembly exist a bi-directional load cell capable of measuring rod driving force with a precision less than one pound force.

Sensory equipment on the testing fixture sends signals to a data acquisition device which actively records real time data. The data acquisition unit also functions as the main brain to the system. Active feedback loops and triggers allow an operator to utilize a graphical user interface to control remotely the operation of the testing fixture while remaining at a safe distance.

Additionally, secondary safety measures were taken to maximize operator wellbeing. Measures included redundancy of inline rupture disc valves. These valves relieve pressure in the case of a pressure spike or if the intensifier overshoots the target pressure. Existing on both the pressure intensifier assembly and the reciprocating rod test bed are motion overtravel limit switches. Soft limit travel switches cancel movement and notify operator. Hard travel limit switches kill power to movement and physical resetting is required. Lastly, stepper motor actuated needle valves were implemented to create a hands-free slow depressurization process.

5.2 LIMITATIONS AND CHALLENGES

Through the design and build process very few real challenges presented themselves. The largest of problems existed due to improper quality control inspections of machined parts. Specific to the device under testing, cylindricity of the piston rod assembly was found to out of tolerance, Figure 78.

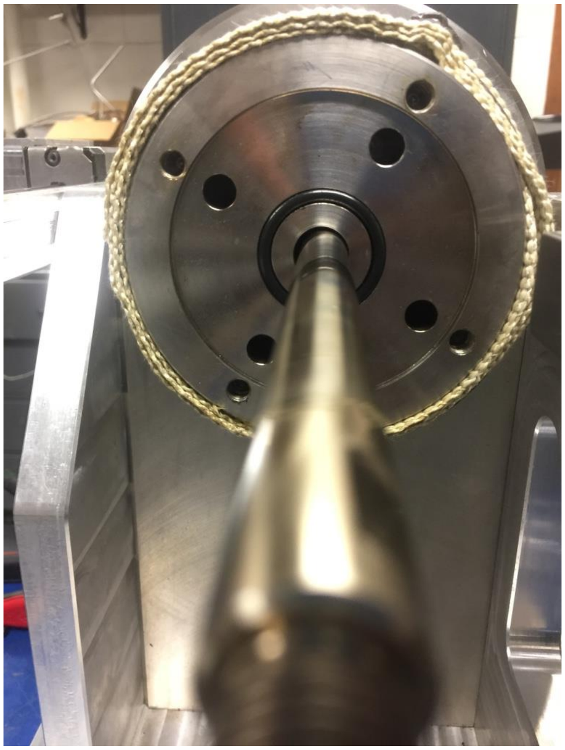


Figure 78: Quality control issue with reciprocating rod.

The front view of the deice under testing shows the reciprocating rod inside the device under testing. The uneven axial spacing around the rod in relation to the bore led the investigation causing the underlining issue. After inspection it was found that the through bore of the pressurized housing was indeed straight and symmetrical. The root cause of the alignment issue was a bend in the reciprocating rod assembly. Thoughts on this issue are that the raw material used for the turned parts held some residual stress and upon material removal those stresses deformed the connected components in the rod assembly.

5.3 FUTURE RECCOMENDATIONS

When retrospectively looking at the design process behind the testing fixture presented in this thesis, there are a few changes I would consider. Overall functionality and future adaptability can be accomplished by substituting the NI USB X-Series 6353 for a more user-friendly National Instruments C-Serie DAQ and chassis. This addition to the experimental testing fixture would free real estate in the electrical enclosure, smooth signals of transducers more efficiently, give the ability to grow the system organically, and reduce the programming difficulty.

Additionally, I believe the experimental setup can be functionally improved by introducing some method of fluid volume addition. This additional fluid volume would help to reduce transient effects felt by the seals inside the device under testing, Figure 79.

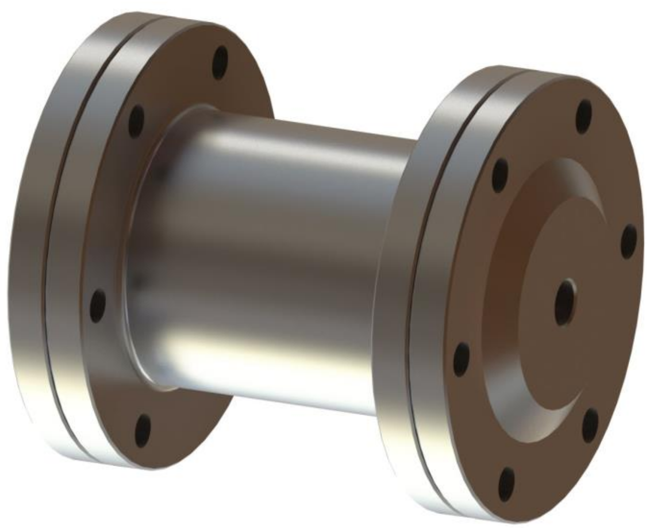


Figure 79: Pulsation dampener concept.

This proposed concept would act as an inline hydraulic fluid pressurized reservoir. The idea here is that with increased volume the effects of hydraulic hammering of the testing fluid would be lessened with additional volume utilized as a buffer. The concept is conceptual but has been used in industry on systems producing pressure differentials less than expected here. Figure 80 shows a basic outline of the routing process for clarification.

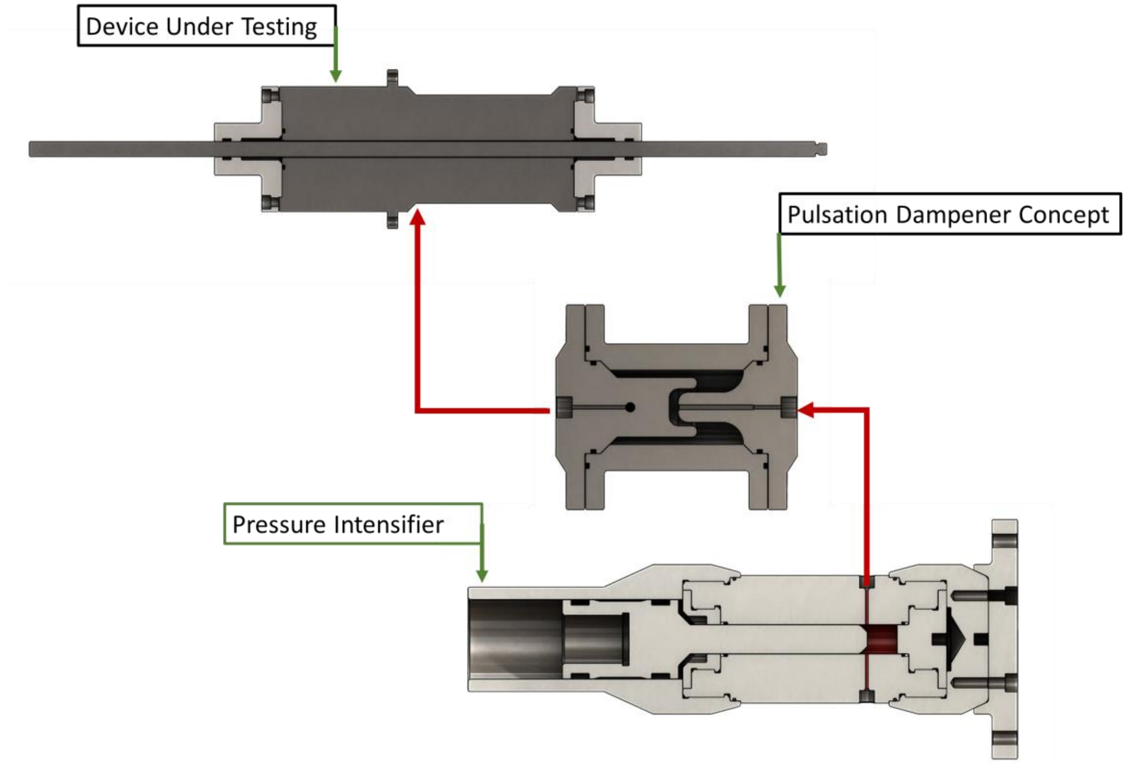


Figure 80: Generalized flow process of dampener concept sub assembly.

From the above figure, the pressure intensifier boosts hydraulic fluid pressure and passes that pressure upstream to the dampening device. The dampening device has engineered baffle systems to decrease transient effects of fluid pulsations resulting from the reversal of reciprocating rod motion. In steady state, while the reciprocating rod is in motion variations to pressure remain negligible, only at the end of travel movements does the system experience spikes in pressure. When the piston rod transitions form the compression stroke to the tension stroke and vice versa a moment of interim fluid turbulence becomes present. This turbulence then raises pressures felt by the seals inside the device under testing. Not only does this variation in fluid pressure affect the characterization of seals within the device under testing but also this pressure spike effects the mechanical response of the testing apparatus.

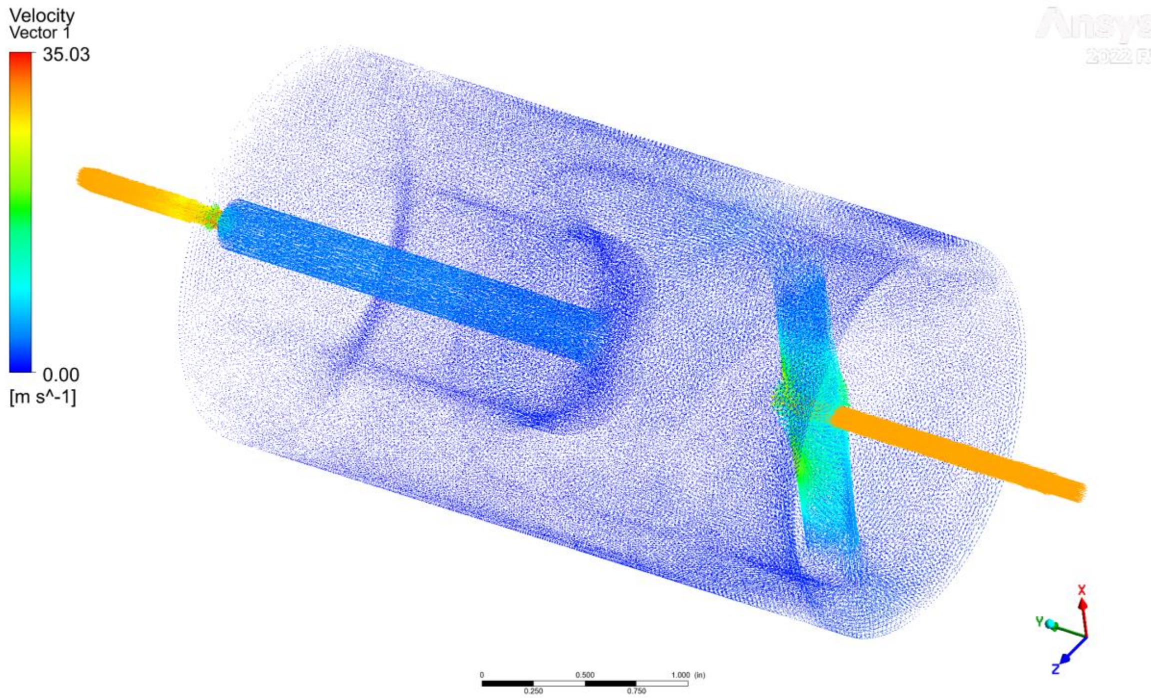


Figure 81: Computational fluid dynamics simulation result of transient behavior pertaining to the conceptual fluid dampener.

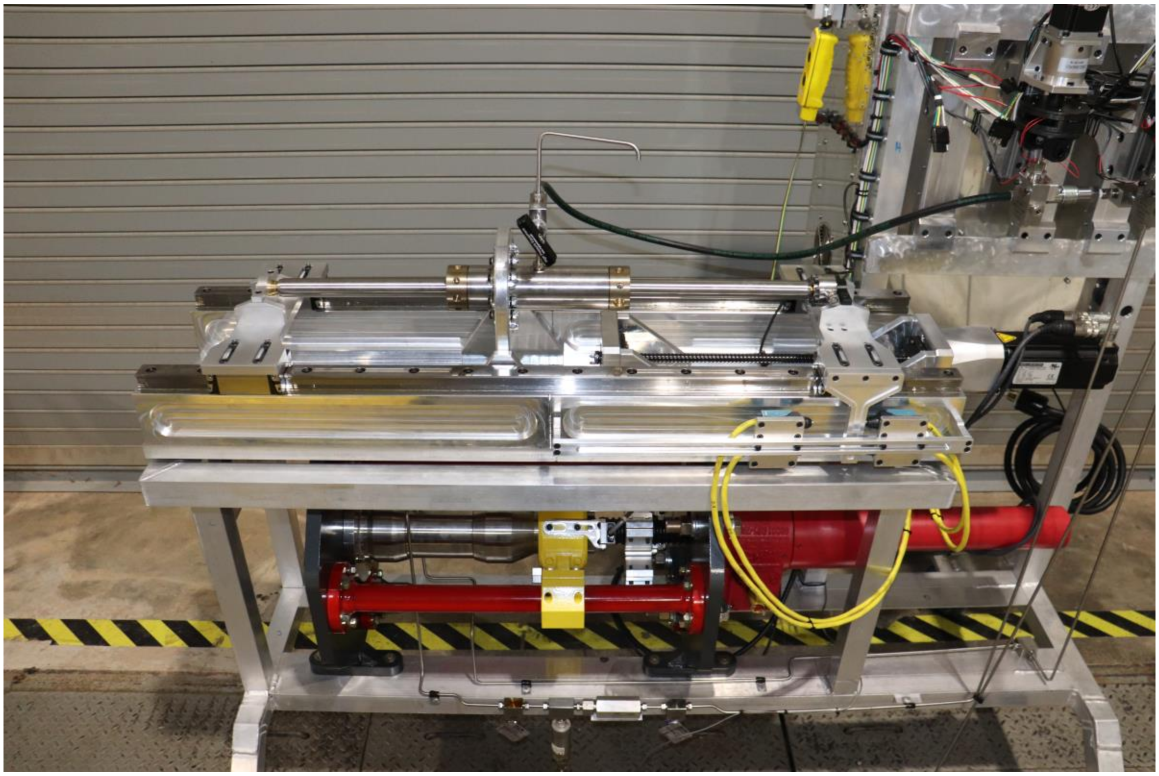
Utilizing ANSYS CFX, a computational fluid dynamics analysis software, preliminary views of the dampening subassembly concept shows expected flow regime changes between at the inlet and outlet. Cyclical loading of the fluid entering the dampening device is applied to the inlet boundary condition and then expected at the outlet. Mass flow is conserved; however, the altering of inlet and outlet port sizing can provide a fluid velocity difference, in turn, increasing or decreasing transient pressure spikes resulting from the dynamic motion.

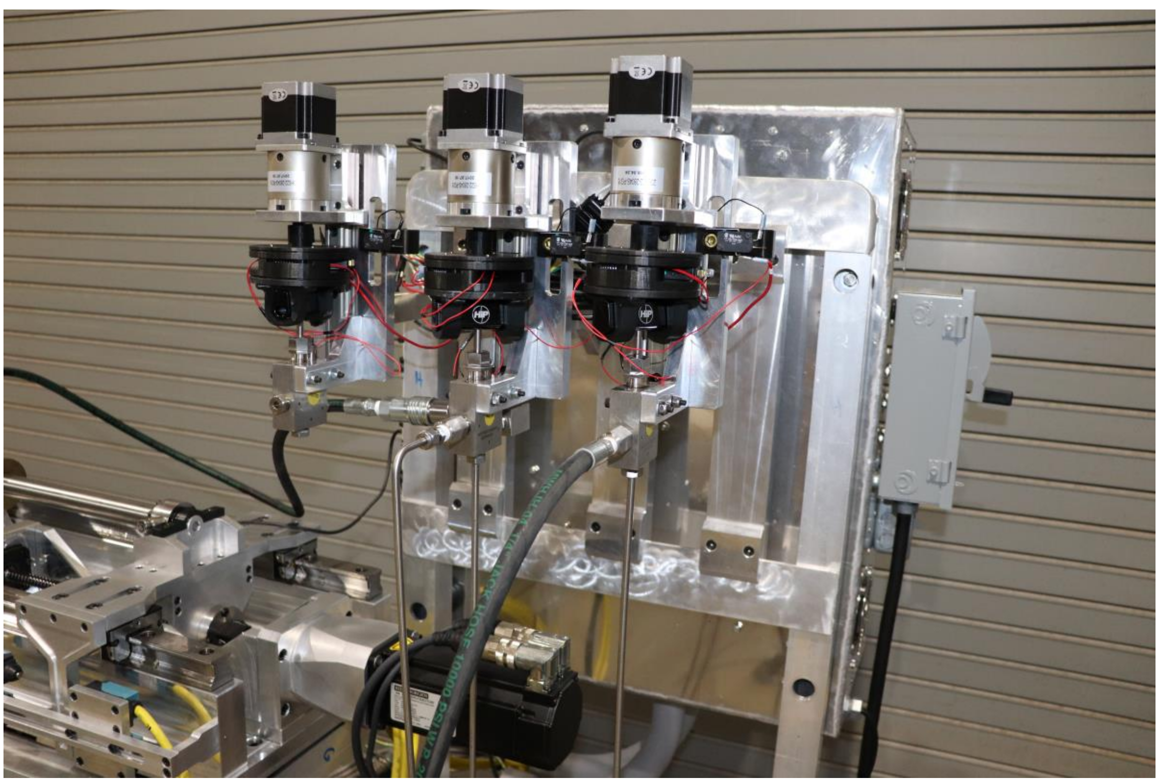
REFERENCES

1. Rubber, A. *Dynamics of reciprocating seals*. 2023; Available from: applerubber.com.
2. Ullman, D.G., *The mechanical design process*. 3rd ed. McGraw-Hill series in mechanical engineering. 2003, Boston, Mass: McGraw-Hill. xv, 415 pages : illustrations.
3. Grabowski, H., S. Rude, and G. Grein, *Universal design theory*. 1998, Aachen: Shaker Verlag.
4. Heisler, S.I., *The Wiley project engineer's desk reference : project engineering, operations, and management*. 1994, New York: Wiley. xv, 504 p.
5. *Design Process*. 2023; Available from: https://engineeringproductdesign.com/article-categories/product-design-process/.
6. KVAstainless. *17-4 Mechanical Properties*. 2014; Available from: http://www.kvastainless.com/17-4.html.
7. Wihaga, *Thread Size Optimization (Acme/Stub Acme Threads)*. 2021.
8. Incorporated, P.S., *VSeal and Bearing Design*. 2017: Irvine, CA.
9. Corporation, I.S.a.T. *Drawn over mandrel tubing selection*. 2018; Available from: https://www.industrialtube.com/drawn-over-mandrel/.
10. TPMCSTEEL. *Differences between but welds and socket welds*. Available from: https://tpmcsteel.com/quality/socket-weld-butt-weld/.
11. Spaenaur, *Bolt Clamping Force vs. Tightning Torque For Unlubricated Steel Bolts*. 2017.
12. McMaster-Carr. *Extreme-Strength Grade 9 Steel Hex Head Screw. 7/16"-14, 2" length*. 2017 [cited 2020 February 1]; Available from: https://www.mcmaster.com/90201A363/.
13. Carvill, J., *Mechanical engineer's data handbook*. 2012, Place of publication not identified: Butterworth-Heinemann.
14. Nook, *Nook Jack Catalog*. 2017. p. 53.
15. Kollmorgen, *Kollmorgen AKM54H Speed vs. Torque 220Vac*. 2018.
16. Kollmorgen Corporation, *AKD User Guide*. Kollmorgen Corporation.
17. Industries, N., *Linear Ball Screw Assemblies* 2017. p. 89.
18. Industries, N. *Ball Screw (PMBS) Torque* 2017; Available from: https://www.nookindustries.com/resources/engineering-calculators/.
19. Kollmorgen Corporation, *AKM Servo Motor*. 2017, Kollmorgen Corporation.
20. Kollmorgen, *Kollmorgen AKM44H Speed vs. Torque at 220Vac*. 2017.
21. Company, H.P.E., *High Pressure Full Catalog*. 2020. p. 46, 48, 50, 51, 52, 55.
22. Engineering, O., *High-Accuracy Minerature Universal Load Cell*, in 2022. p. 1.
23. Liptak, B.G., *Instrument engineers' handbook*. 4th ed. 2006, Boca Raton, Fla: CRC/Taylor & Francis.
24. Engineering, O., *Very-High Accuracy Current Output Transducer* 2022. p. 1, 2.

25. Guyer, E.C. and D.L. Brownell, *Handbook of applied thermal design*. 2nd . ed. 1999, Philadelphia, Pa. ; London: Taylor & Francis. 1 v. (various pagings) : ill..
26. Instruments, N., *DAX X Series User Manual*. 2019. p. 110,111,221.
27. Jennings, R. and F.D.L. Cueva, *Labview graphical programming : practical applications in instrumentation and control, fifth edition*. Fifth edition. ed. 2019, New York, NY: McGraw-Hill Education. xxii, 618 pages : illustrations, charts.
28. Kollmorgen, *AKD BASIC Programming Guide*. 2013. p. Chapter 3.6.
29. Kollmorgen, *Kollmorgen P70530 (DC) High Performance Micro-Stepping Drive* 2012. p. 7,8,15.
30. Journal, F.P. *The Reinvention Of The Shear Type Coupling*. 2019; Available from: https://fluidpowerjournal.com/reinvention-shear-type-coupling/.
31. Society of Allied Weight, E., *Weight engineers handbook*. Rev., 1986. ed. 1986, Los Angeles Society of Allied Weight Engineers. 1 v. (various pagings) : ill.
32. *Overview of materials for Polyetheretherketone, Unreinforced*. 2023; Available from: https://www.matweb.com/search/datasheet_print.aspx?matguid=2164cacabcde4391a596640d553b2ebe.

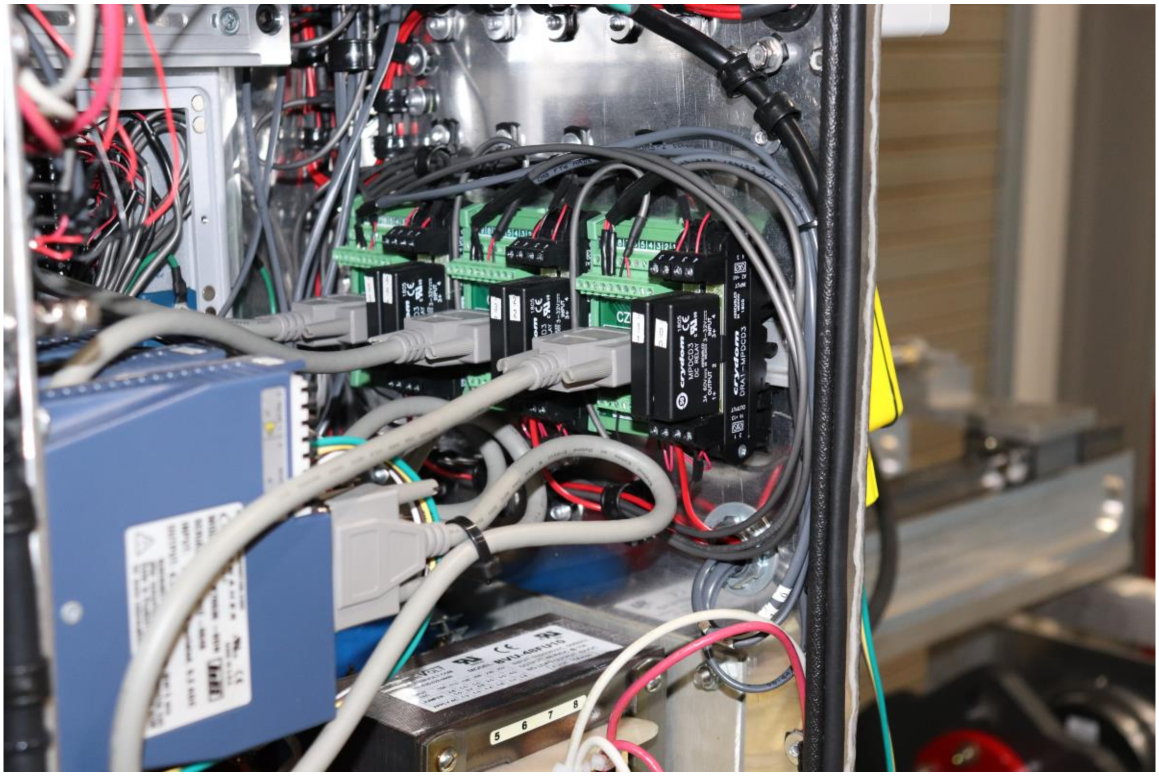
APPENDIX

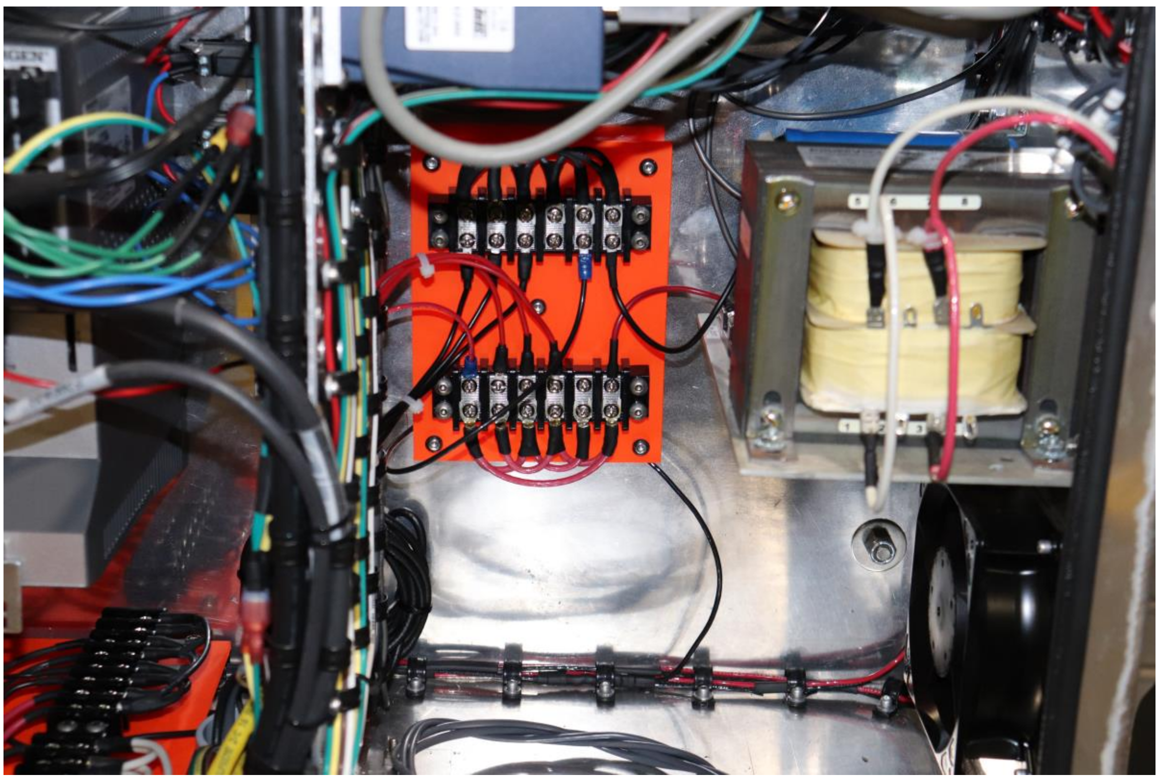


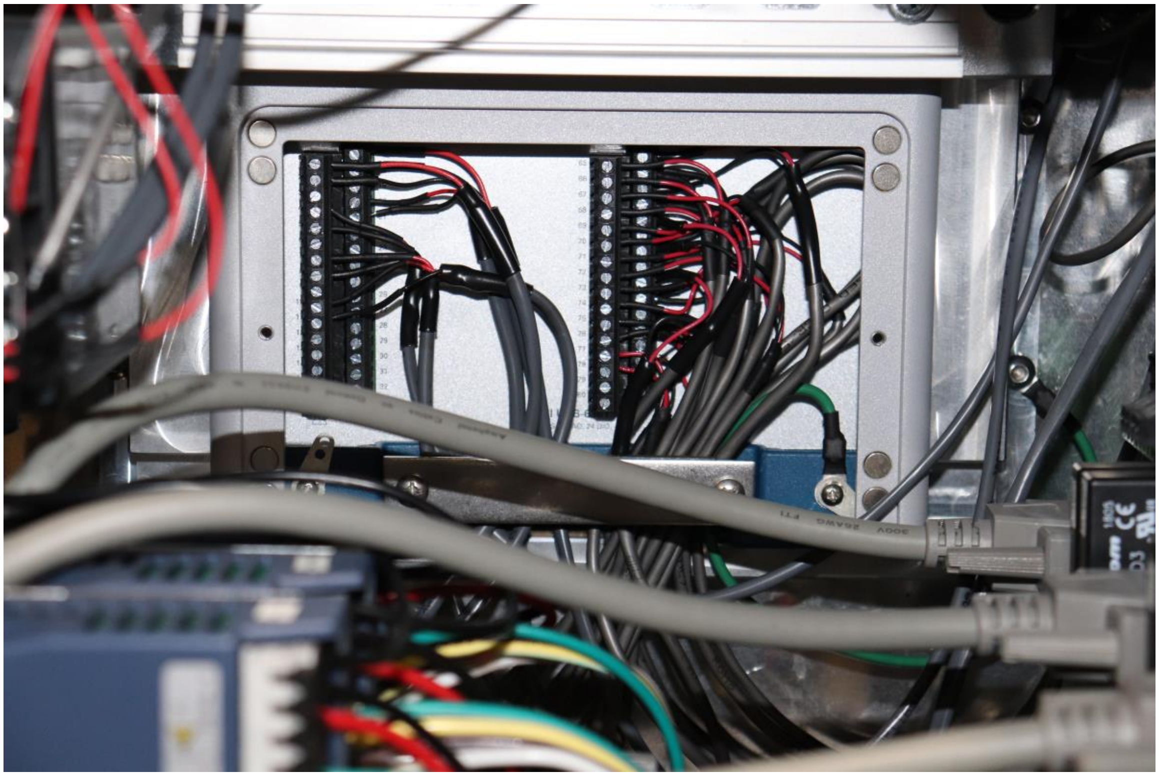


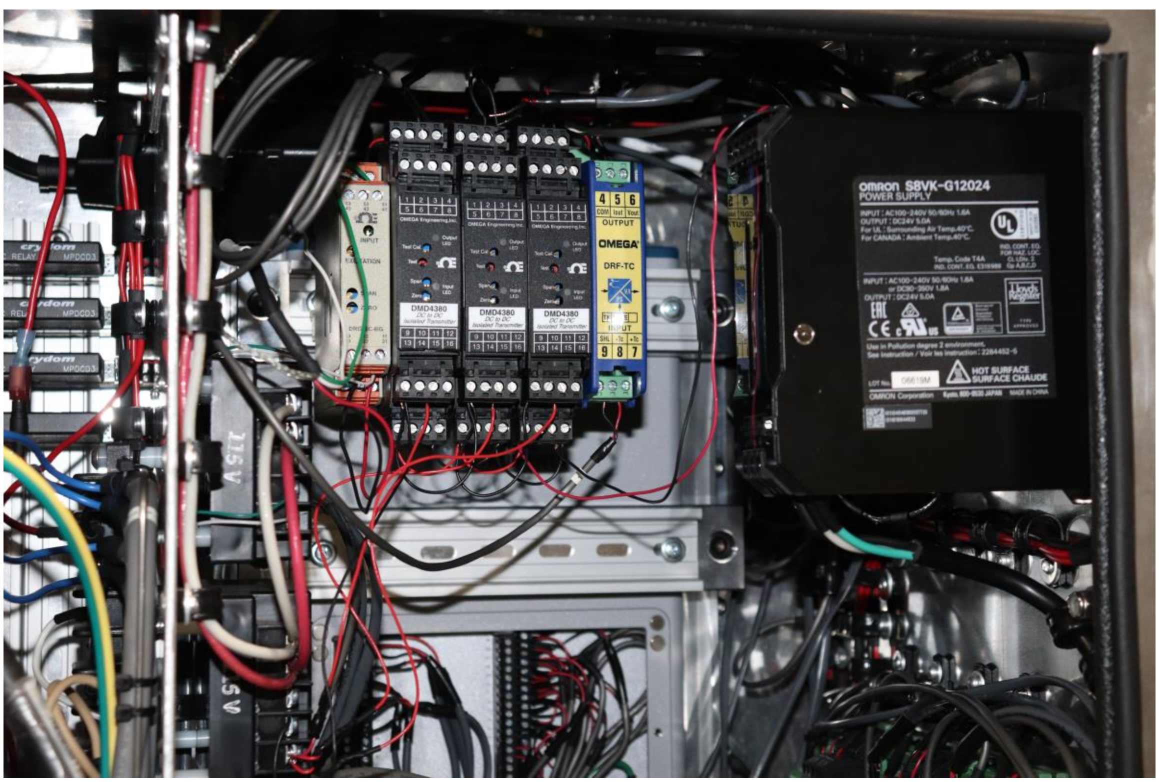

DANGER
PELIGRO

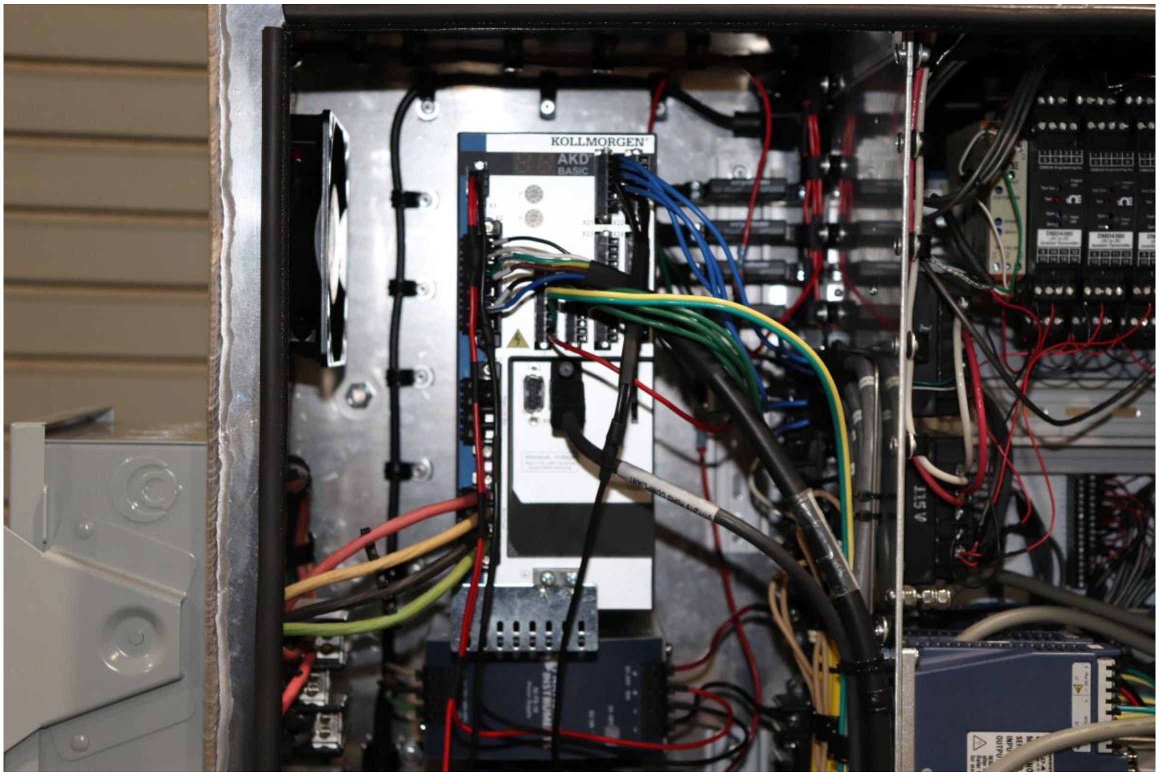

KOLLMORGEN
AKD
BASIC
KOLLMORGEN
AKD
BASIC
OMRON S8VK-G12024
POWER SUPPLY
OMEGA

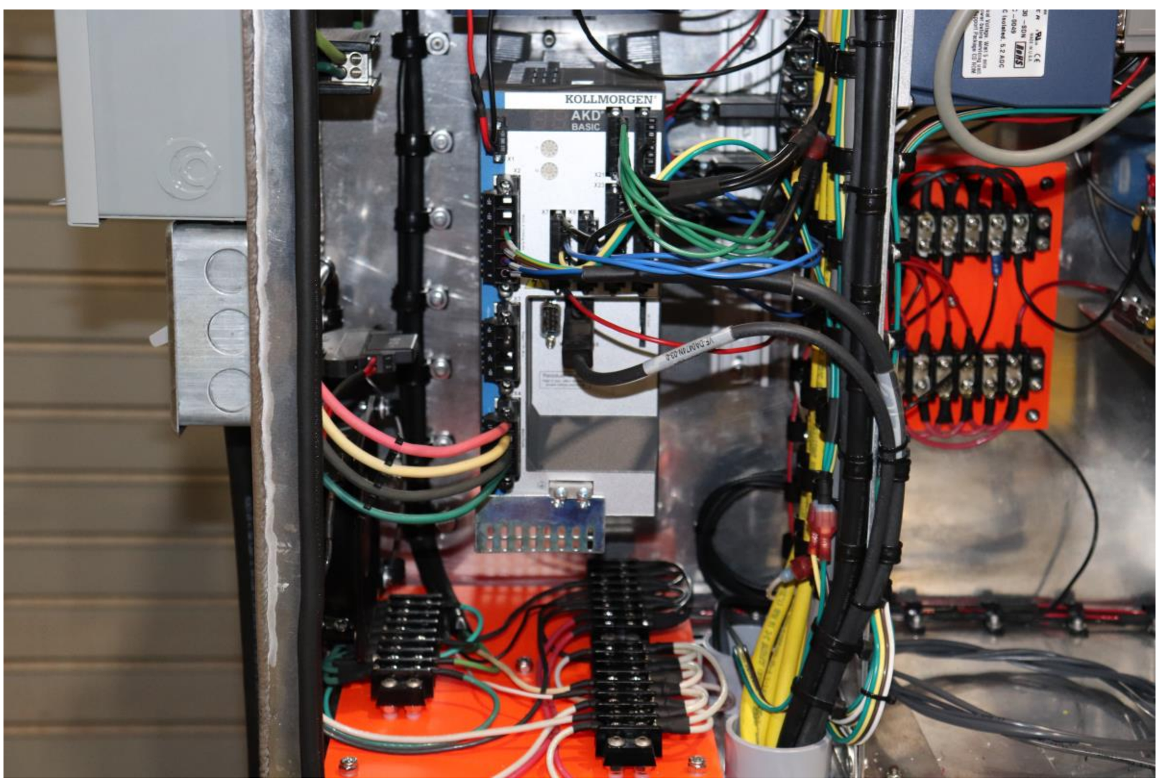


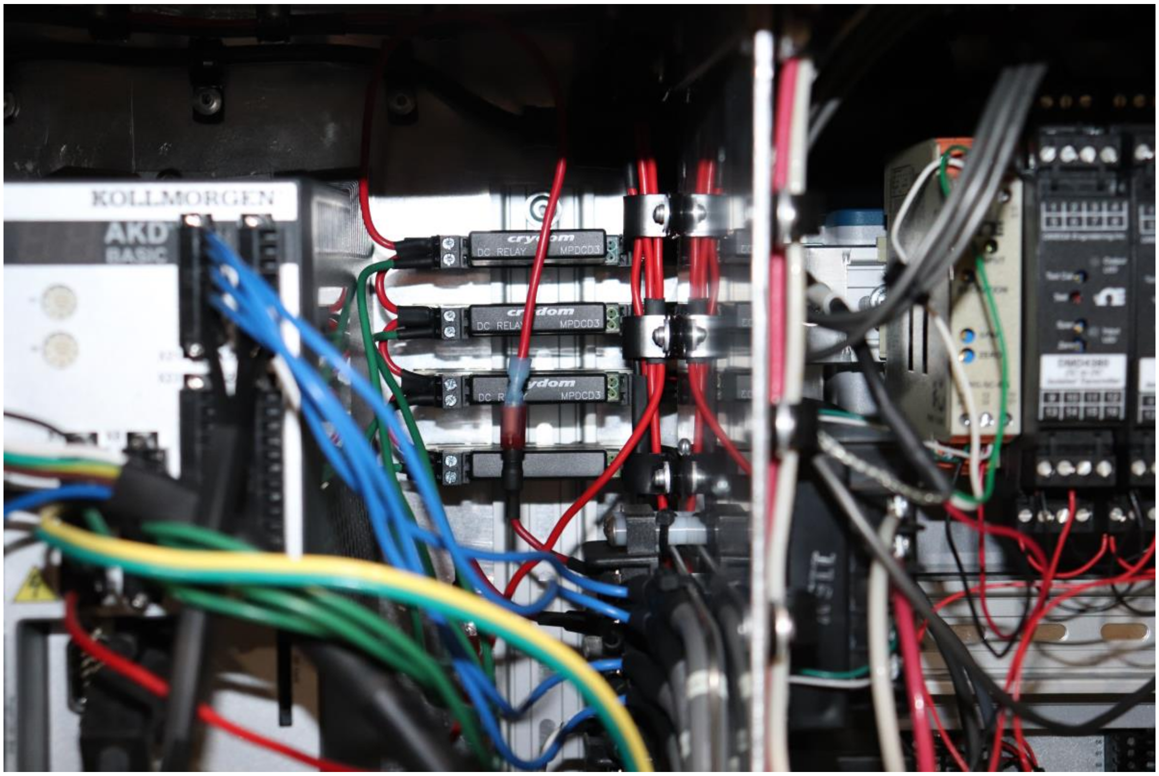


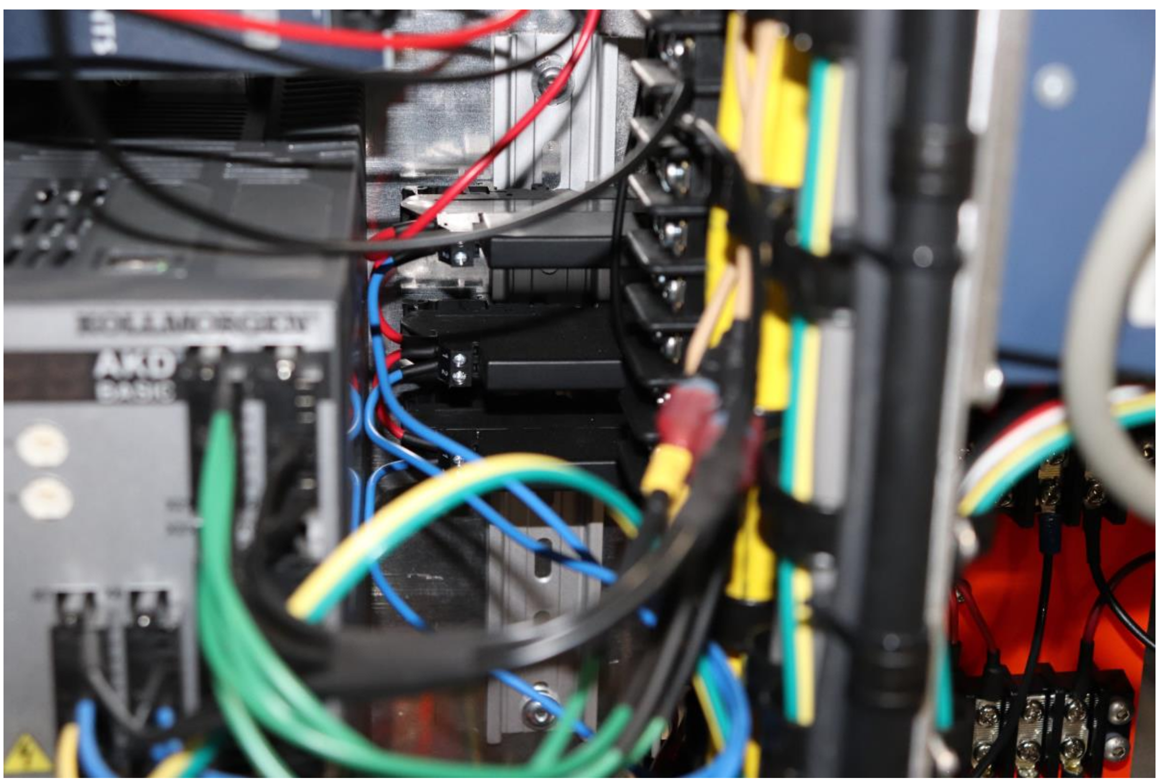


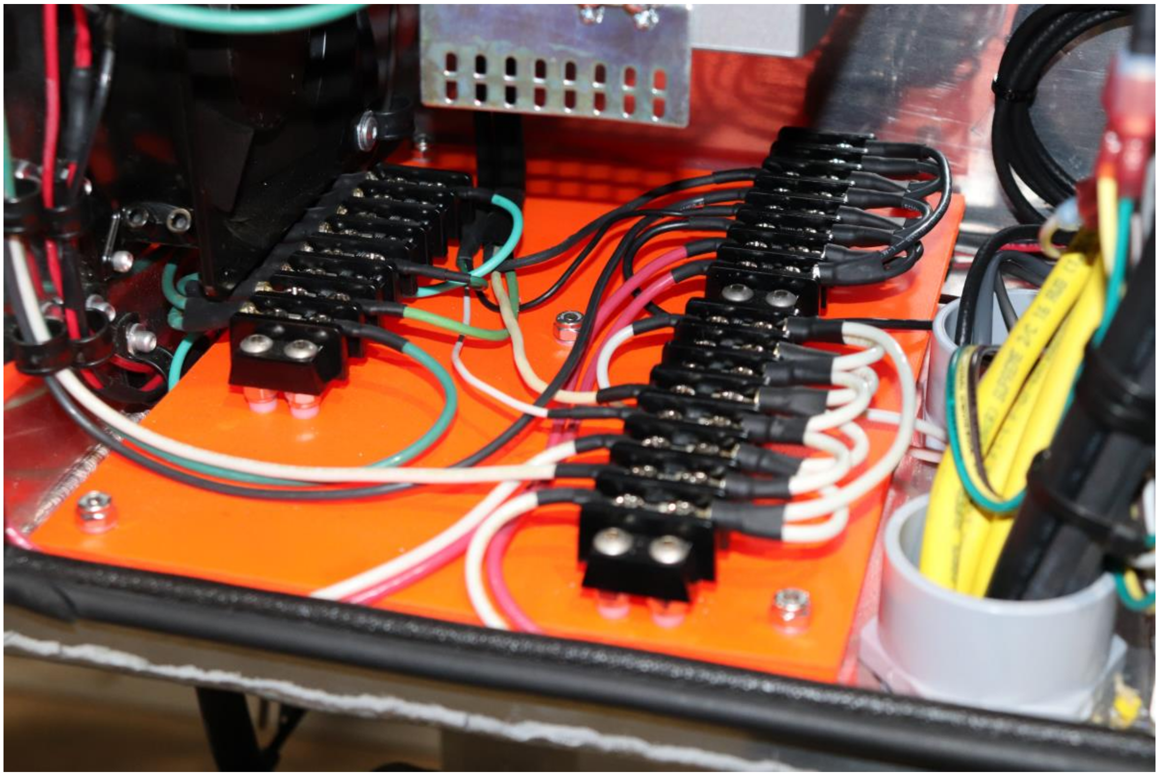

OMRON S8VK-G12024
POWER SUPPLY
DMD4380
DMD4380
DMD4380
OMEGA
DRF-TC
OUTPUT
INPUT
115 V
HOT SURFACE
SURFACE CHAUDE

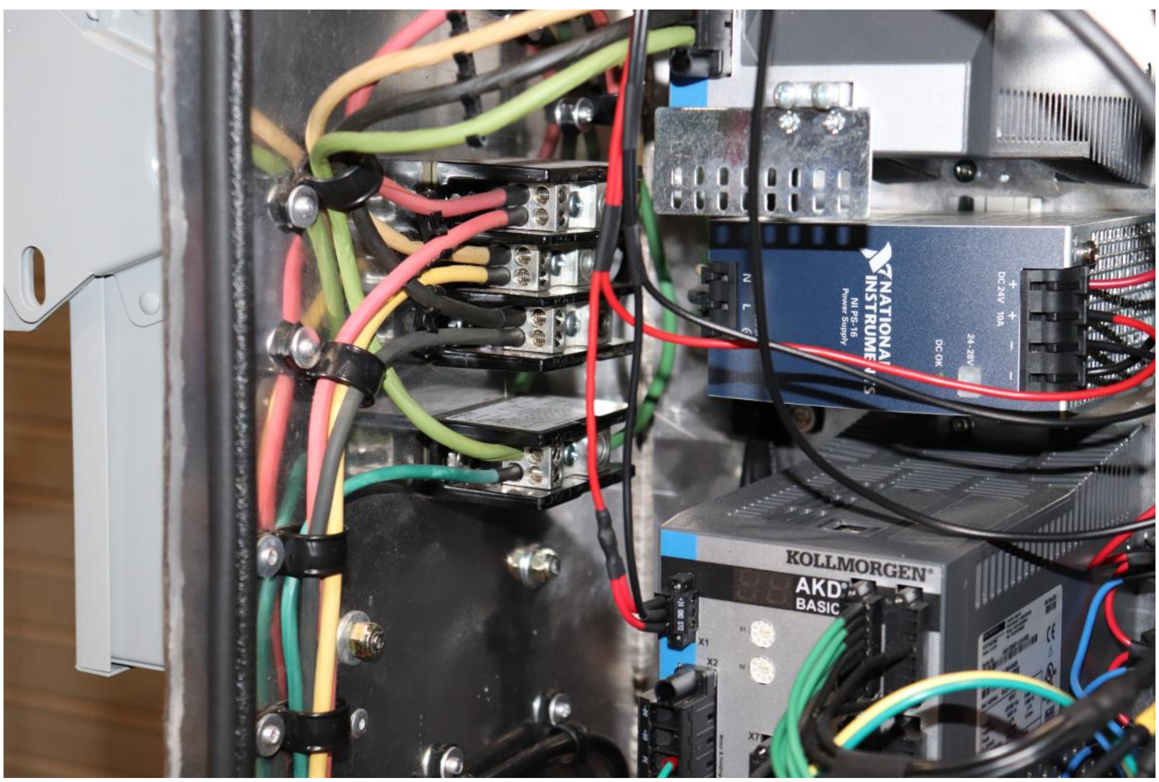

KOLLMORGEN
AKD
BASIC

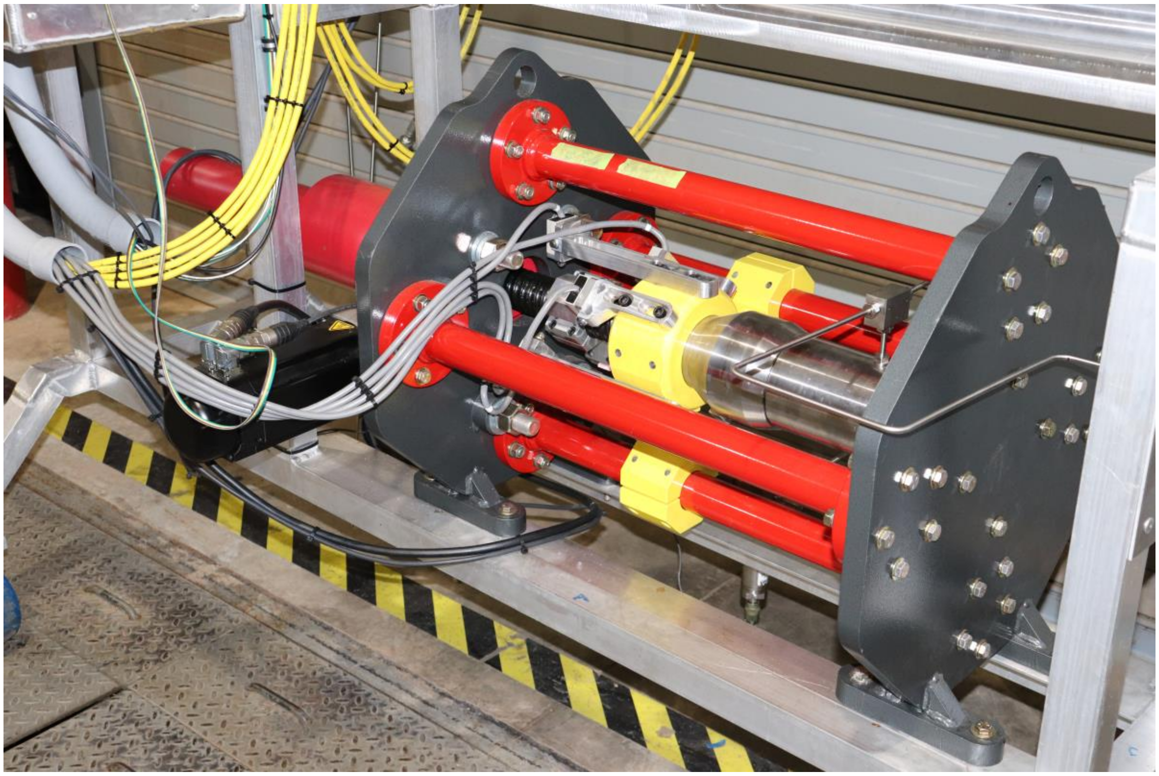

KOLLMORGEN
AKD
BASIC

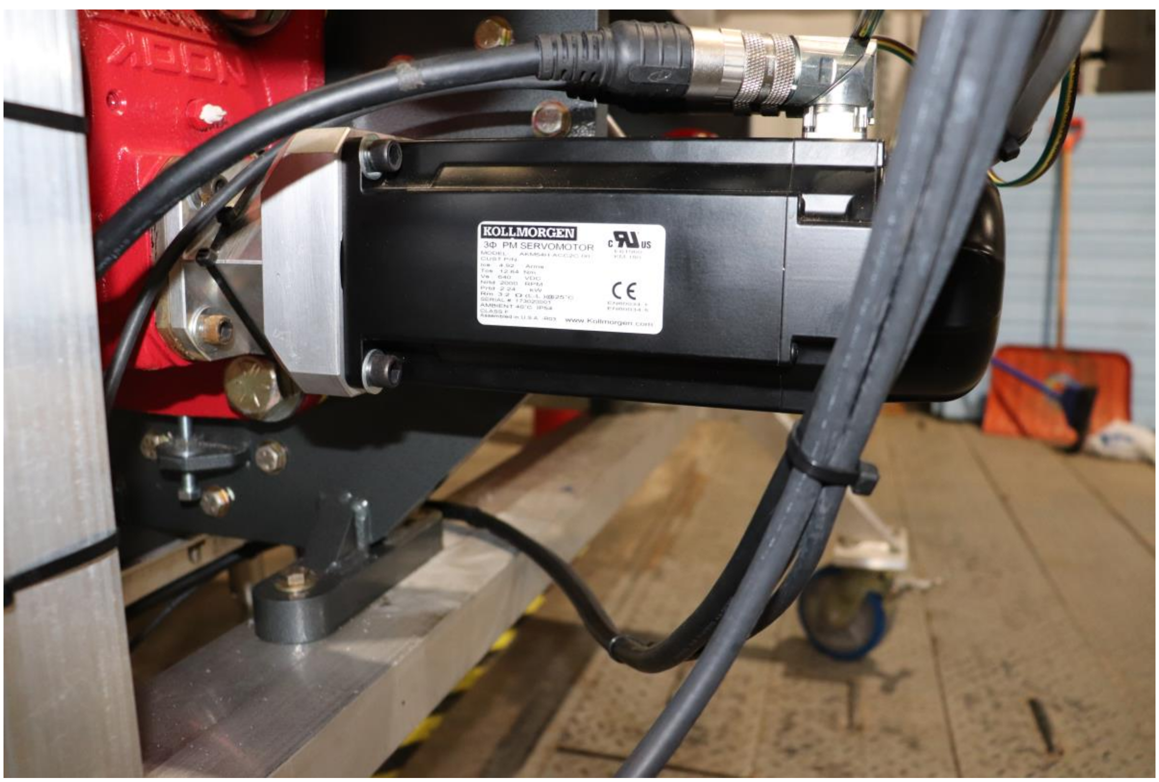

KOLLMORGEN
AKD
BASIC
crydom
DC RELAY
MPDCD3

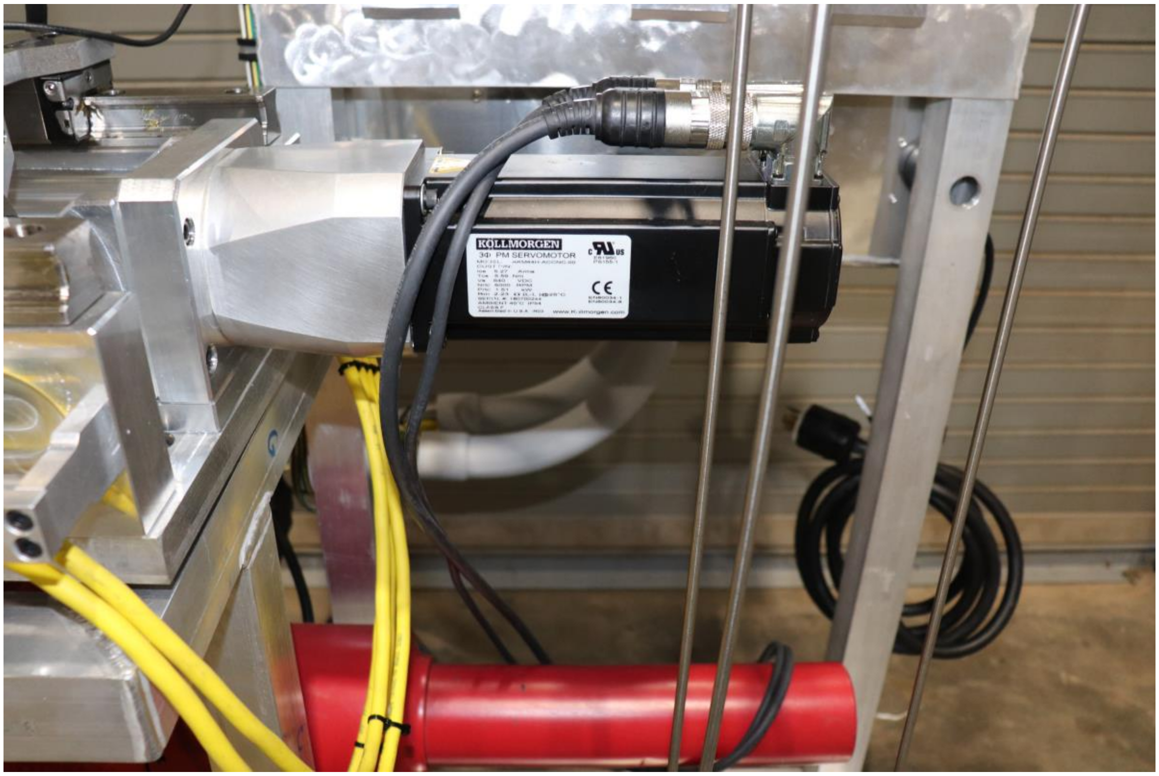

KOLLMORGEN
AKD
BASIC

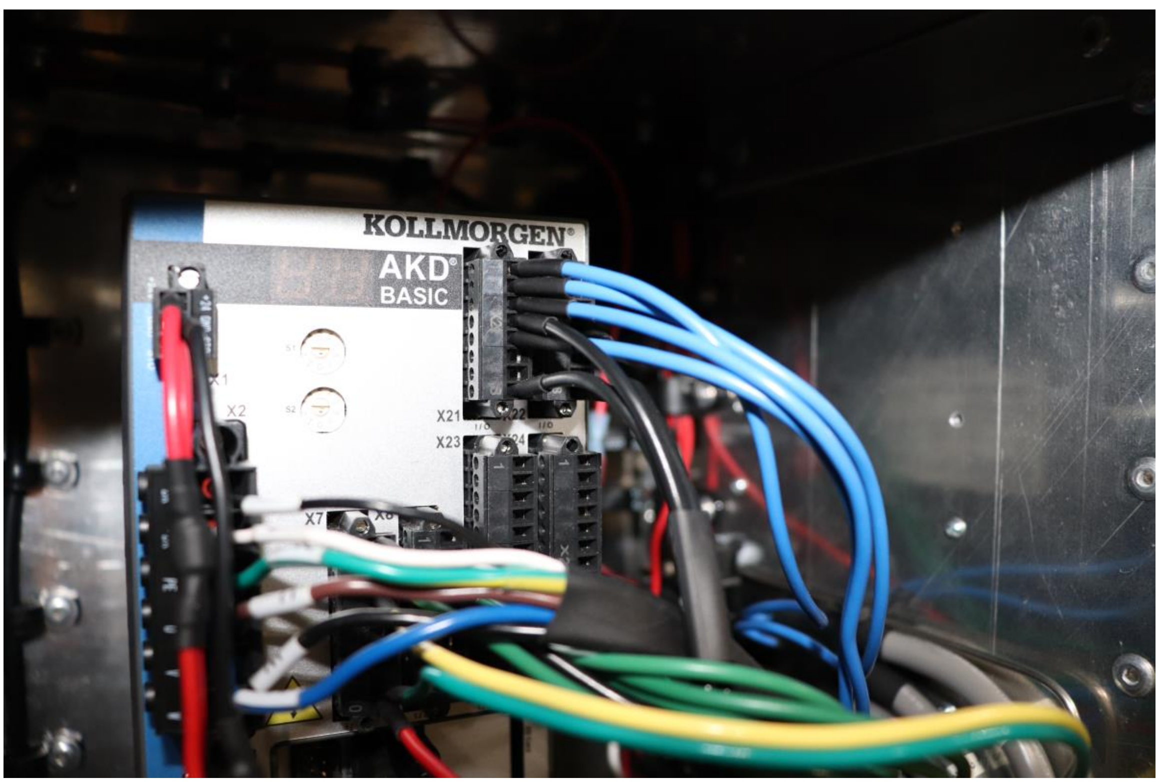


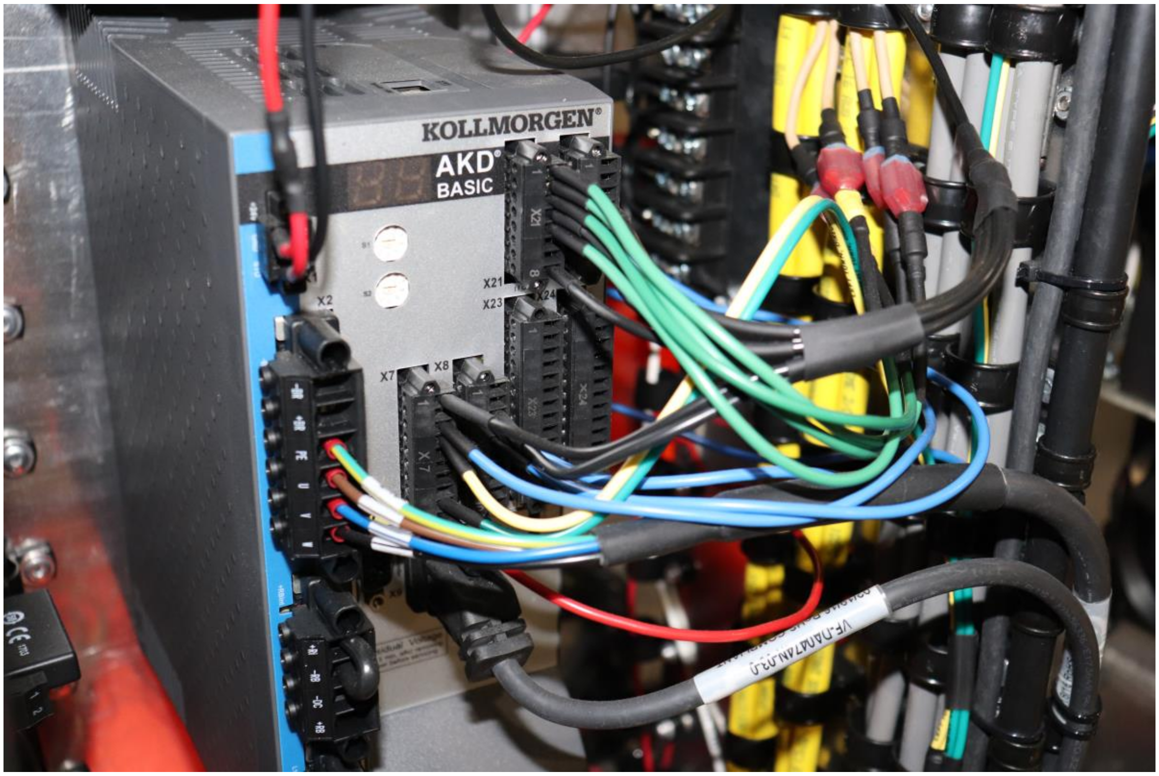

NI PS-16
Power Supply
KOLLMORGEN
AKD
BASIC

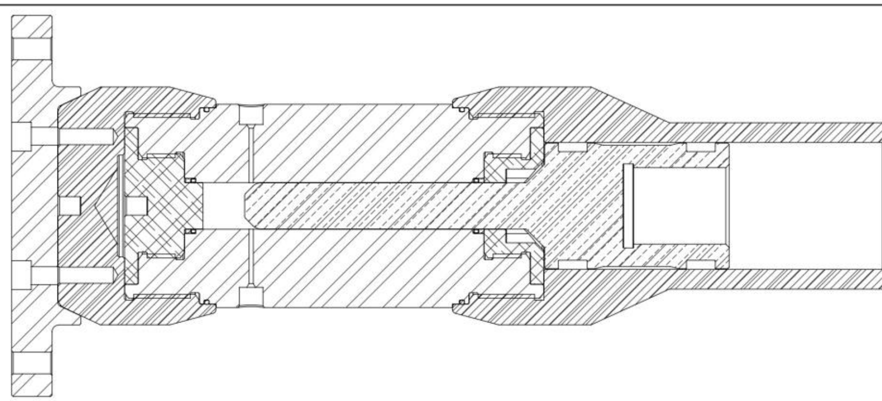



KOLLMORGEN
3Φ PM SERVOMOTOR
www.Kollmorgen.com


KOLLMORGEN
3Φ PM SERVOMOTOR


KOLLMORGEN
AKD
BASIC
X1
X2
X21
X22
X23
X24
X7





ITEM NO.	DESCRIPTION	PART NUMBER	QTY.
1	Housing_High_Pressure	10203-30K-SLBHP-RECIP	1
2	2.25_ACME_Piston_Bore_Plug	10204-30K-SLBHP-RECIP	1
3	4.00_ACME_Housing_OD_End_Cap_Piston_Guide	10202-30K-SLBHP-RECIP	1
4	4.00_ACME_Housing_OD_End_Cap	10201-30K-SLBHP-RECIP	1
5	2.25_ACME_Blind_Bore_Plug	10205-30K-SLBHP-RECIP	1
6	PSI Rod Seal	N/A	2
7	155_Oring	N/A	2
8	Piston	10206-30K-SLBHP-RECIP	1
9	Bolt_On_Flange_HP_Housing	10207-30K-SLBHP-RECIP	1

NOTES:

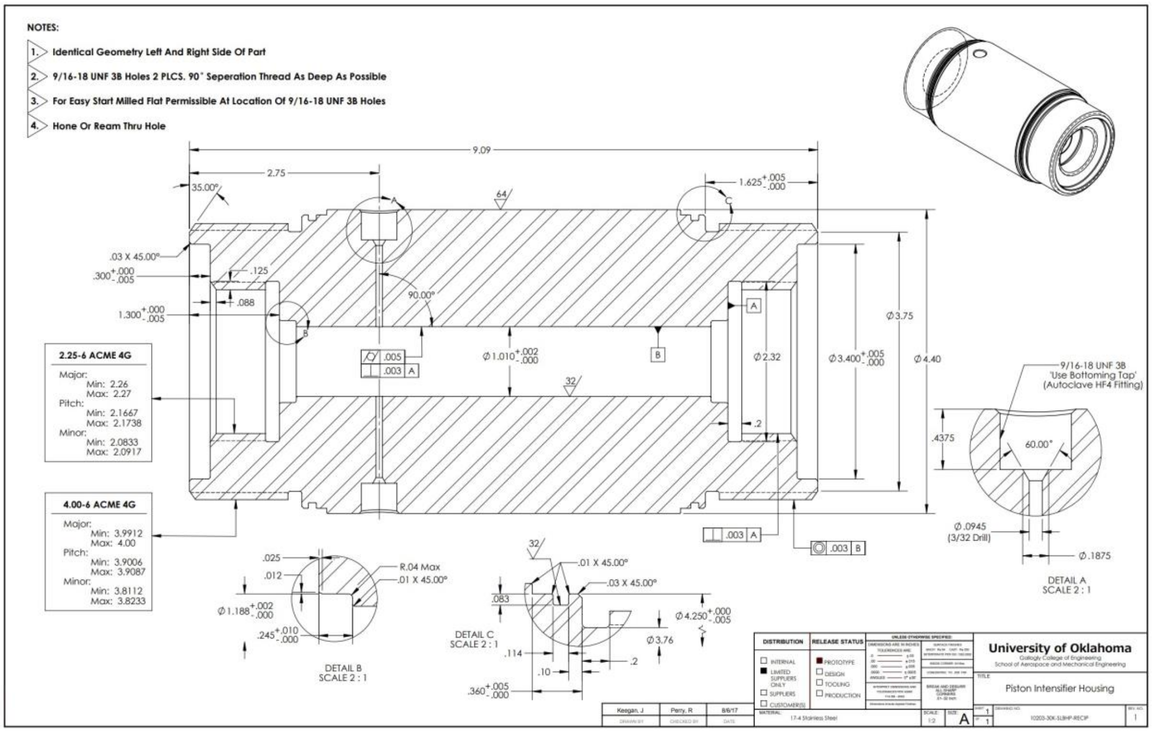
1. Assembly Excludes Fasteners
2. AssemblyExcludes High Pressure Fittings
3. All Threaded Parts Require AntiSieze

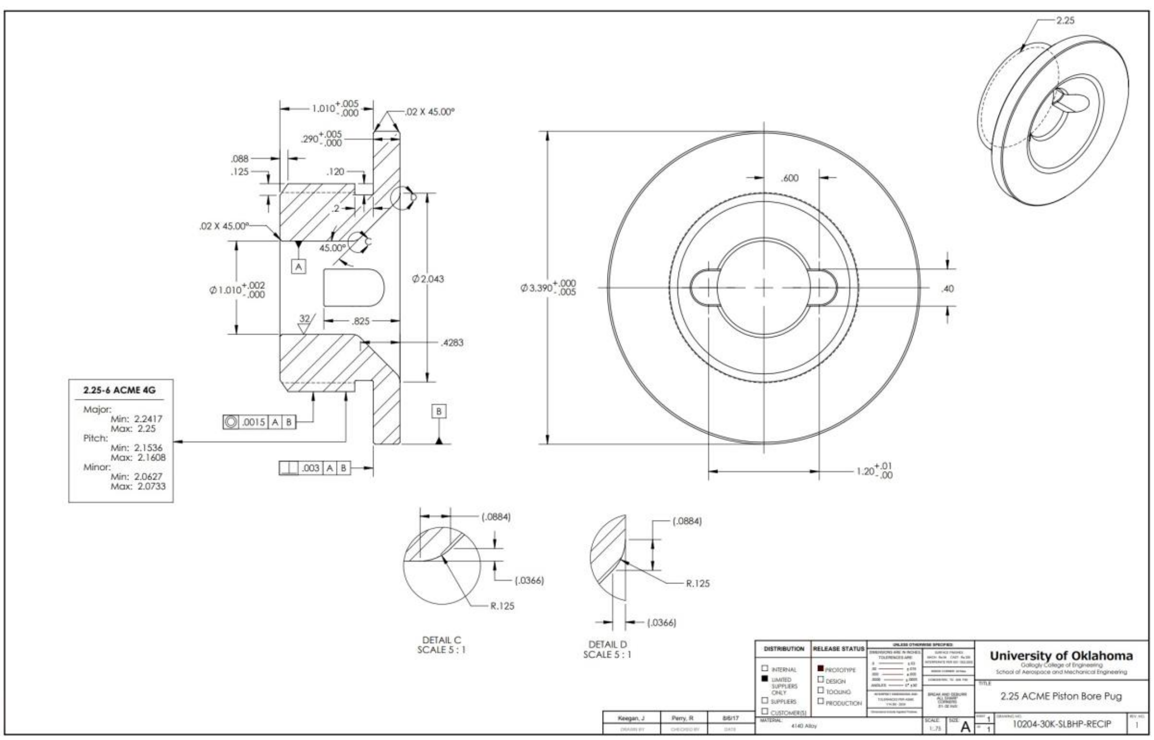
DISTRIBUTION	RELEASE STATUS
☐ INTERNAL	■ PROTOTYPE
■ LIMITED SUPPLIERS ONLY	☐ DESIGN
☐ SUPPLIERS	☐ TOOLING
☐ CUSTOMER(S)	☐ PRODUCTION

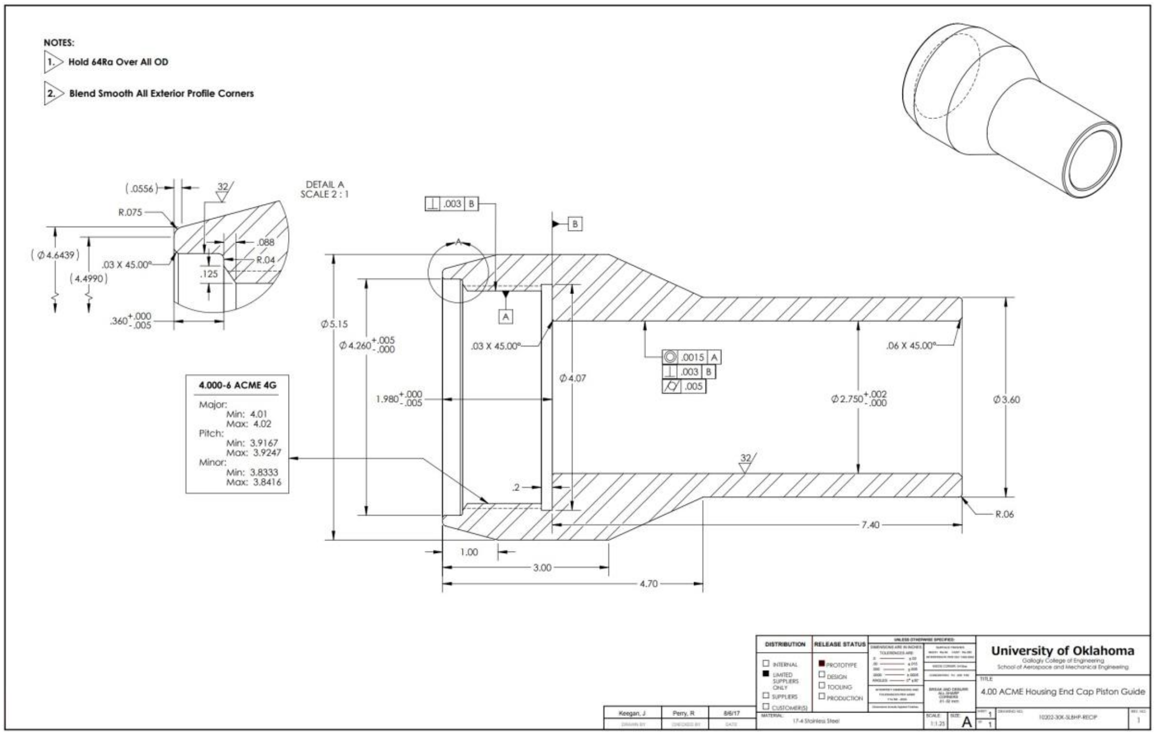
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering

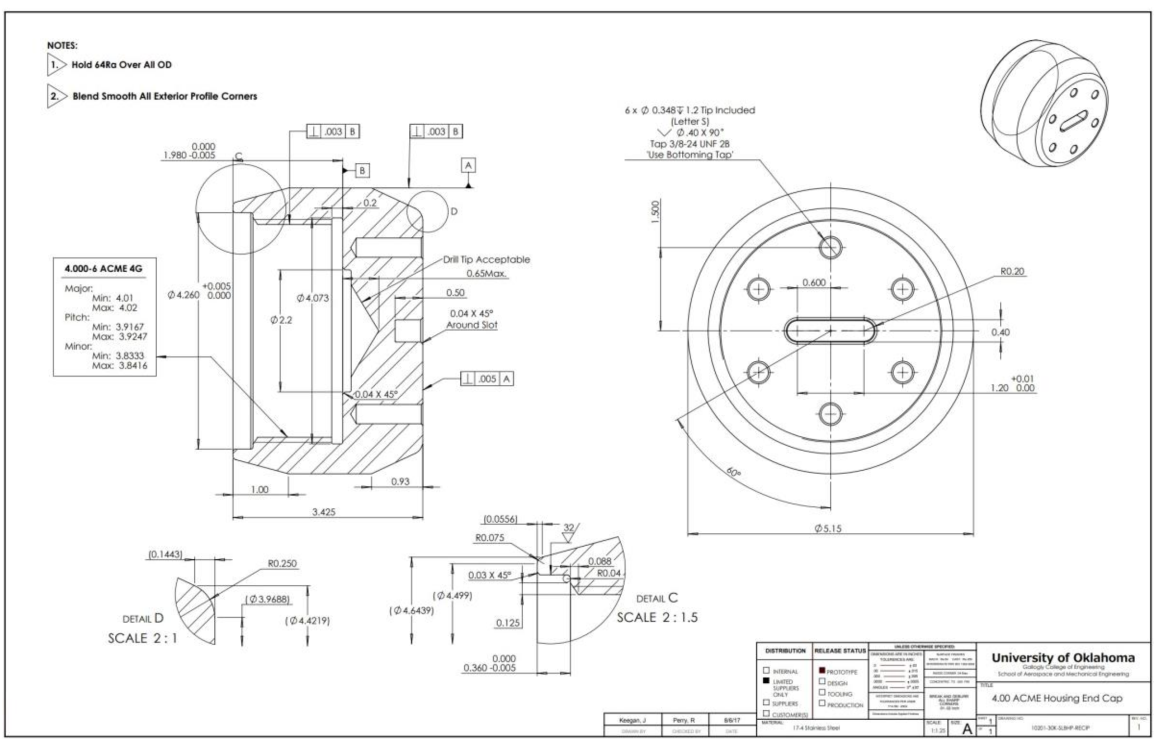
TITLE: Piston Intensifier Housing Assembly

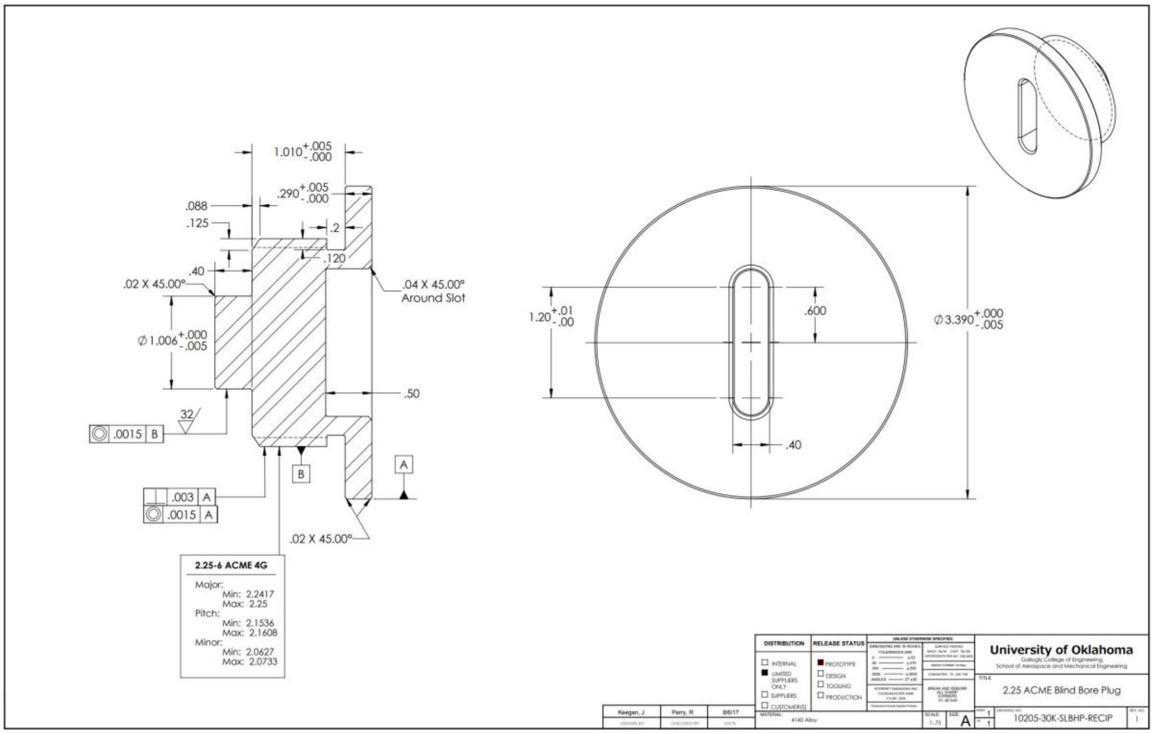
DRAWN BY	CHECKED BY	DATE	MATERIAL	SCALE	SIZE	SHEET	DRAWING NO.	REV. NO.
Keegan, J		7/25/17	17-4 SS And 4140 Alloy		A	1 of 1	10200-30K-SLBHP-RECIP	0

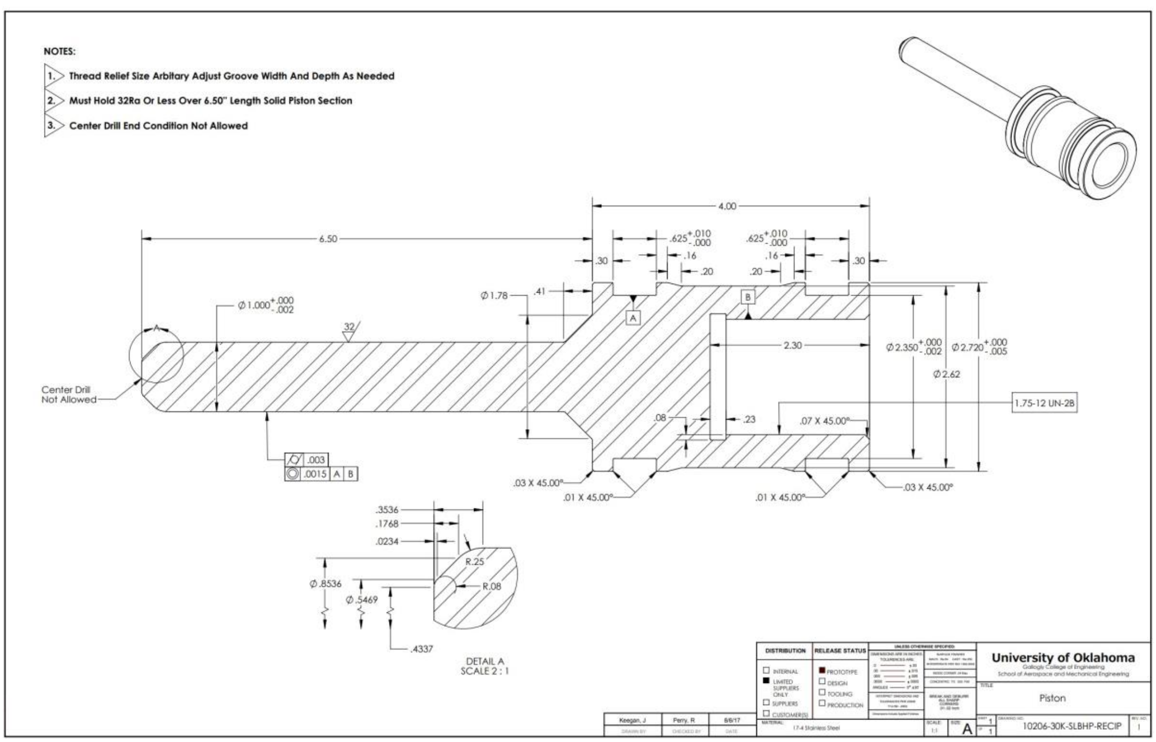

NOTES:
1. Identical Geometry Left And Right Side Of Part
2. 9/16-18 UNF 3B Holes 2 PLCS. 90° Seperation Thread As Deep As Possible
3. For Easy Start Milled Flat Permissible At Location Of 9/16-18 UNF 3B Holes
4. Hone Or Ream Thru Hole
2.25-6 ACME 4G
4.00-6 ACME 4G
DETAIL A
SCALE 2 : 1
DETAIL B
SCALE 2 : 1
DETAIL C
SCALE 2 : 1
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Piston Intensifier Housing
10203-30K-SLBHP-RECIP

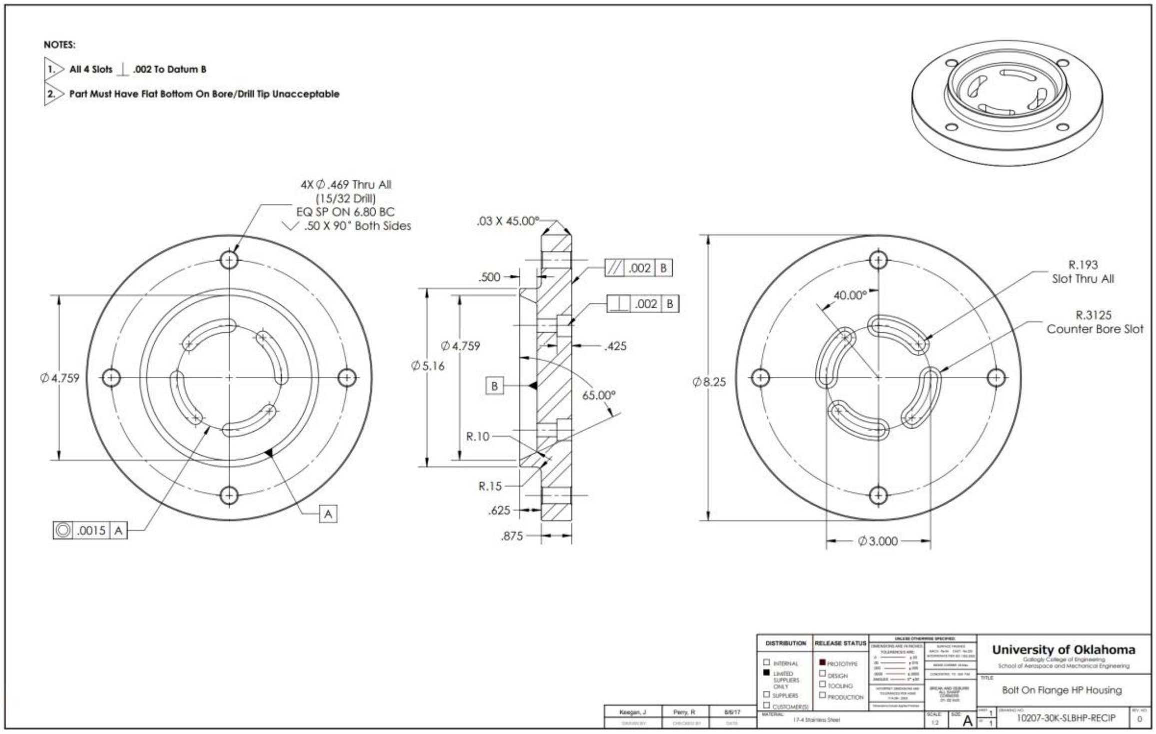

2.25-6 ACME 4G
DETAIL C
SCALE 5 : 1
DETAIL D
SCALE 5 : 1
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
2.25 ACME Piston Bore Pug
10204-30K-SLBHP-RECIP


NOTES:
1. Hold 64Ra Over All OD
2. Blend Smooth All Exterior Profile Corners
DETAIL A
SCALE 2 : 1
4.000-6 ACME 4G
Major:
Min: 4.01
Max: 4.02
Pitch:
Min: 3.9167
Max: 3.9247
Minor:
Min: 3.8333
Max: 3.8416
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
4.00 ACME Housing End Cap Piston Guide
Keegan, J
Perry, R
8/6/17
17-4 Stainless Steel

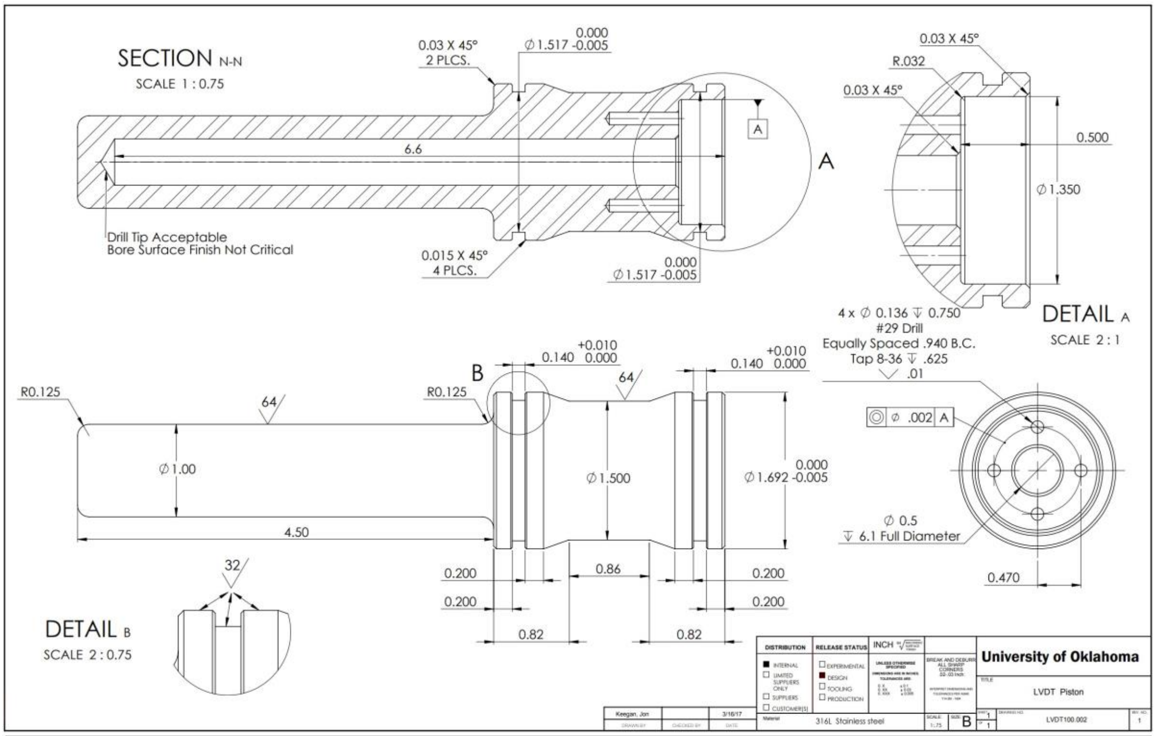
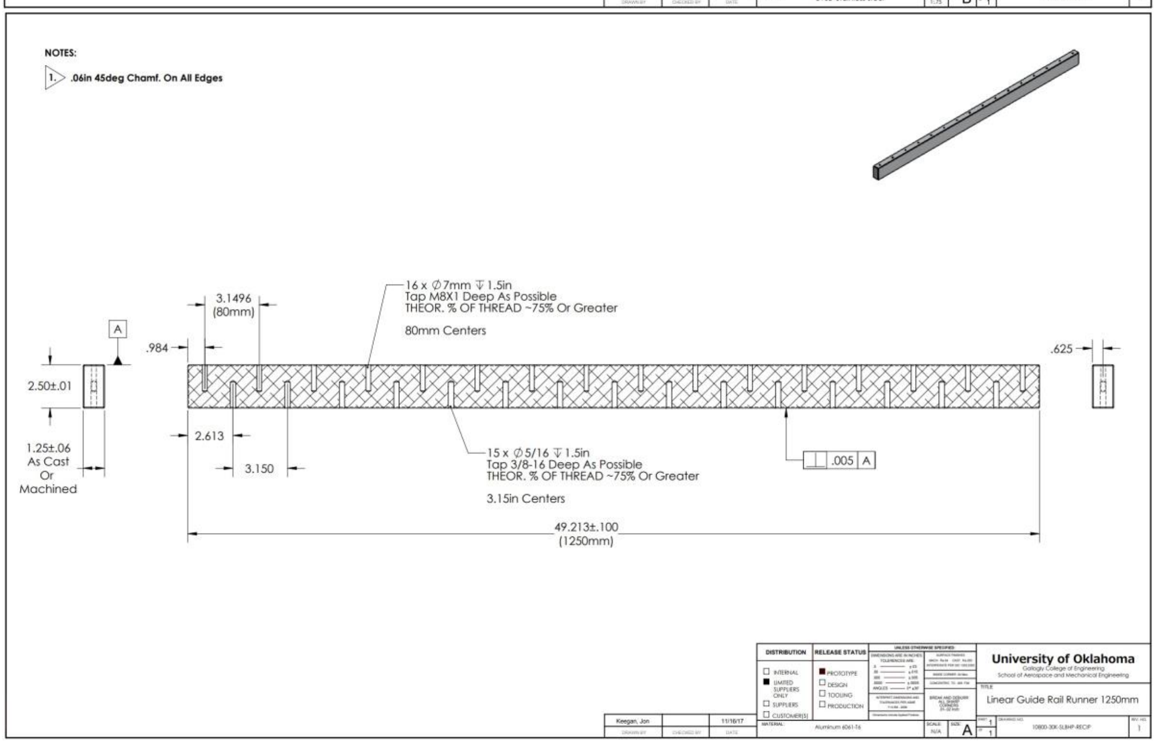

NOTES:
1. Hold 64Ra Over All OD
2. Blend Smooth All Exterior Profile Corners
4.000-6 ACME 4G
Major:
Min: 4.01
Max: 4.02
Pitch:
Min: 3.9167
Max: 3.9247
Minor:
Min: 3.8333
Max: 3.8416
Drill Tip Acceptable
0.65Max.
0.04 X 45°
Around Slot
6 x Ø 0.348 ↧ 1.2 Tip Included
(Letter S)
Ø .40 X 90°
Tap 3/8-24 UNF 2B
'Use Bottoming Tap'
DETAIL D
SCALE 2 : 1
DETAIL C
SCALE 2 : 1.5
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
4.00 ACME Housing End Cap
Keegan, J
Perry, R
8/6/17
17-4 Stainless Steel

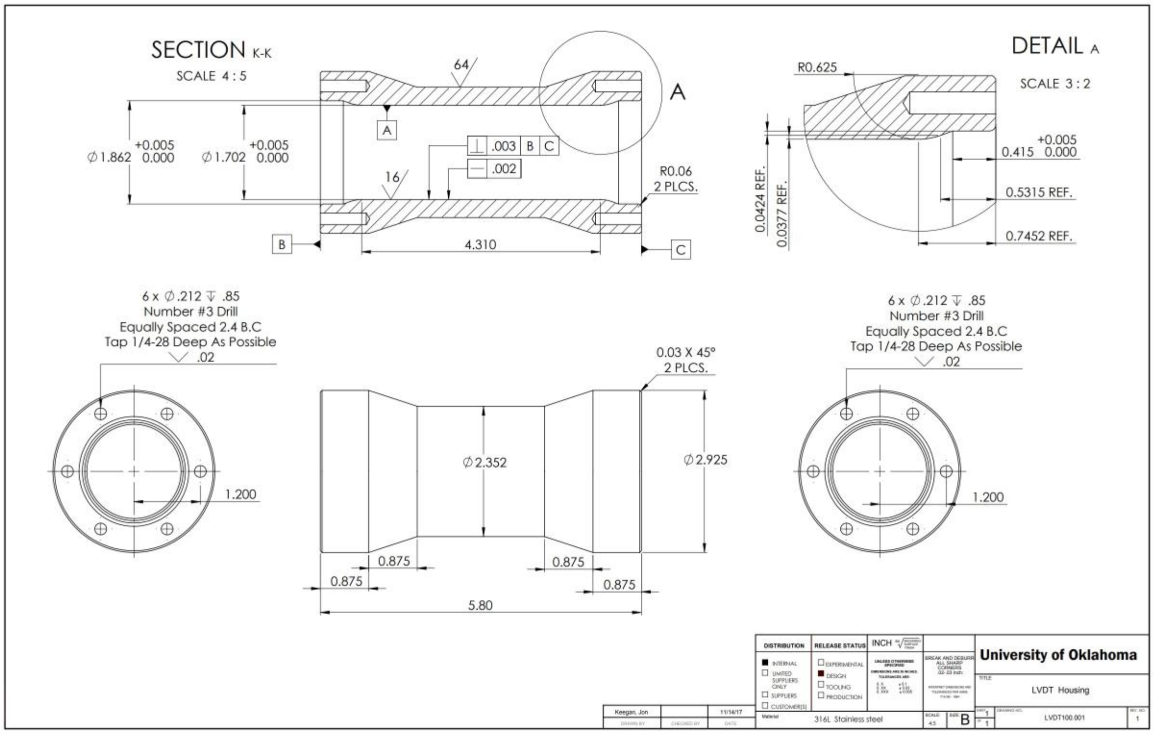

2.25-6 ACME 4G
Major:
Min: 2.2417
Max: 2.25
Pitch:
Min: 2.1536
Max: 2.1608
Minor:
Min: 2.0627
Max: 2.0733
.04 X 45.00° Around Slot
.02 X 45.00°
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
2.25 ACME Blind Bore Plug
10205-30K-SLBHP-RECIP
4140 Alloy
Keegan, J
Perry, R
8/6/17

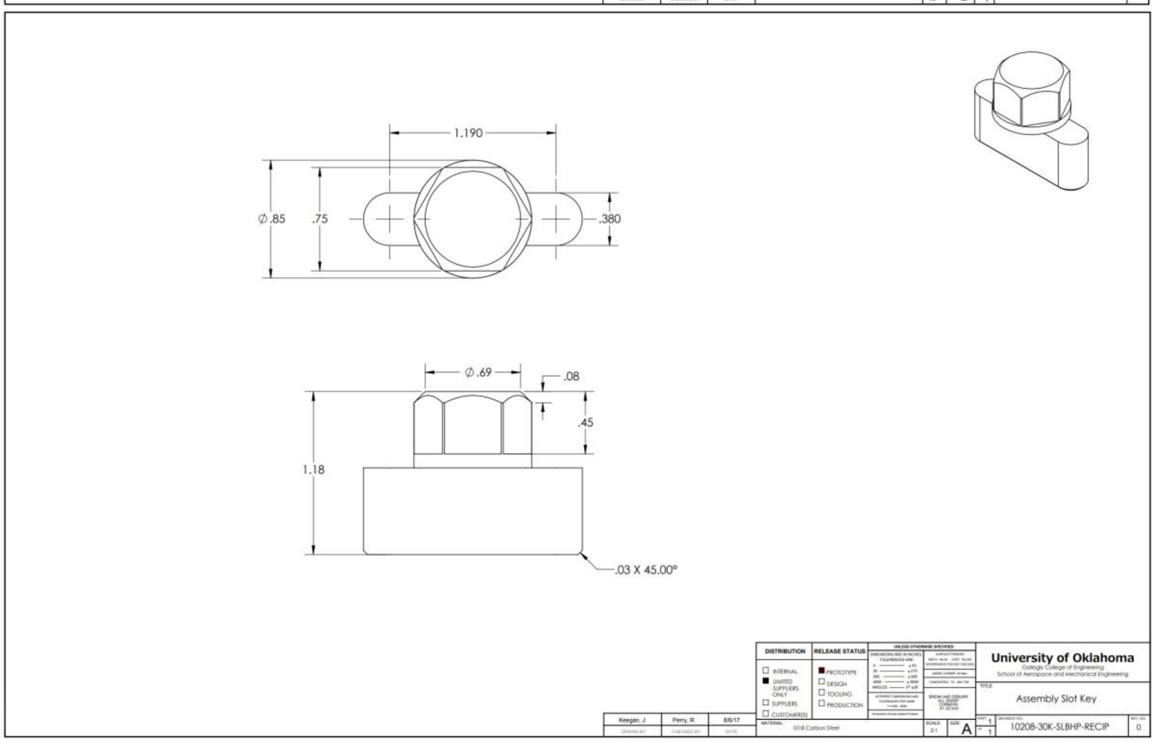

NOTES:
1. Thread Relief Size Arbitary Adjust Groove Width And Depth As Needed
2. Must Hold 32Ra Or Less Over 6.50" Length Solid Piston Section
3. Center Drill End Condition Not Allowed
Center Drill Not Allowed
1.75-12 UN-2B
DETAIL A
SCALE 2 : 1
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Piston
10206-30K-SLBHP-RECIP
17-4 Stainless Steel
Keegan, J
Perry, R
8/6/17

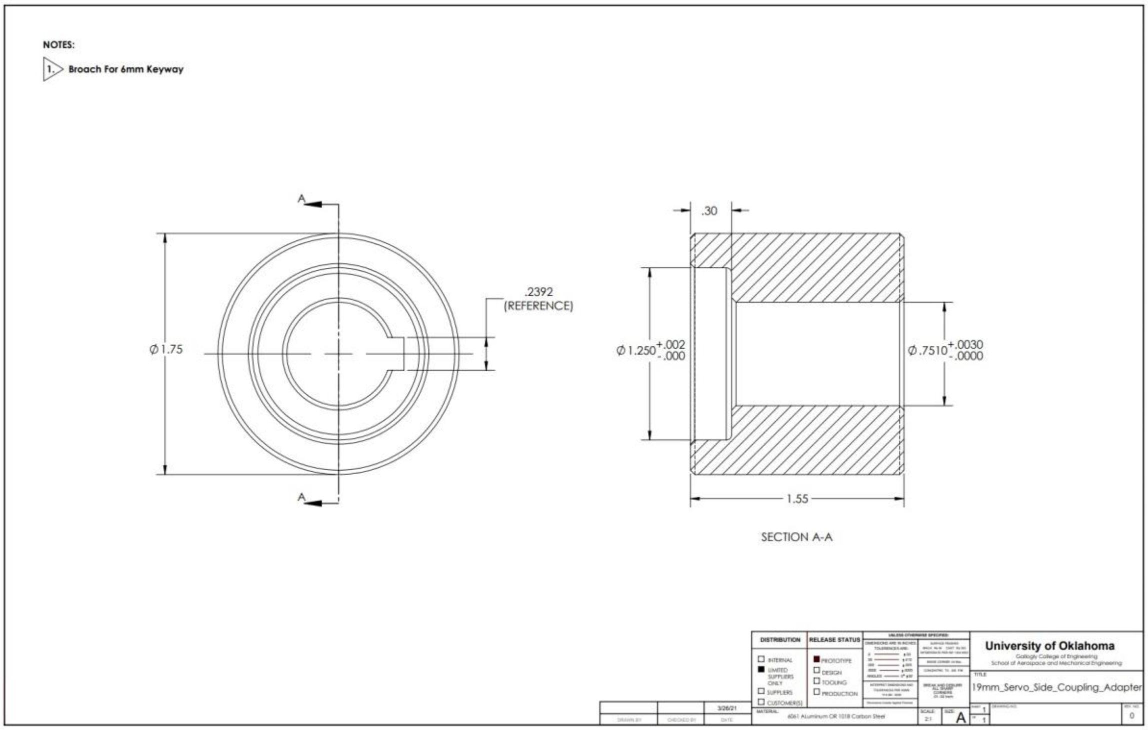


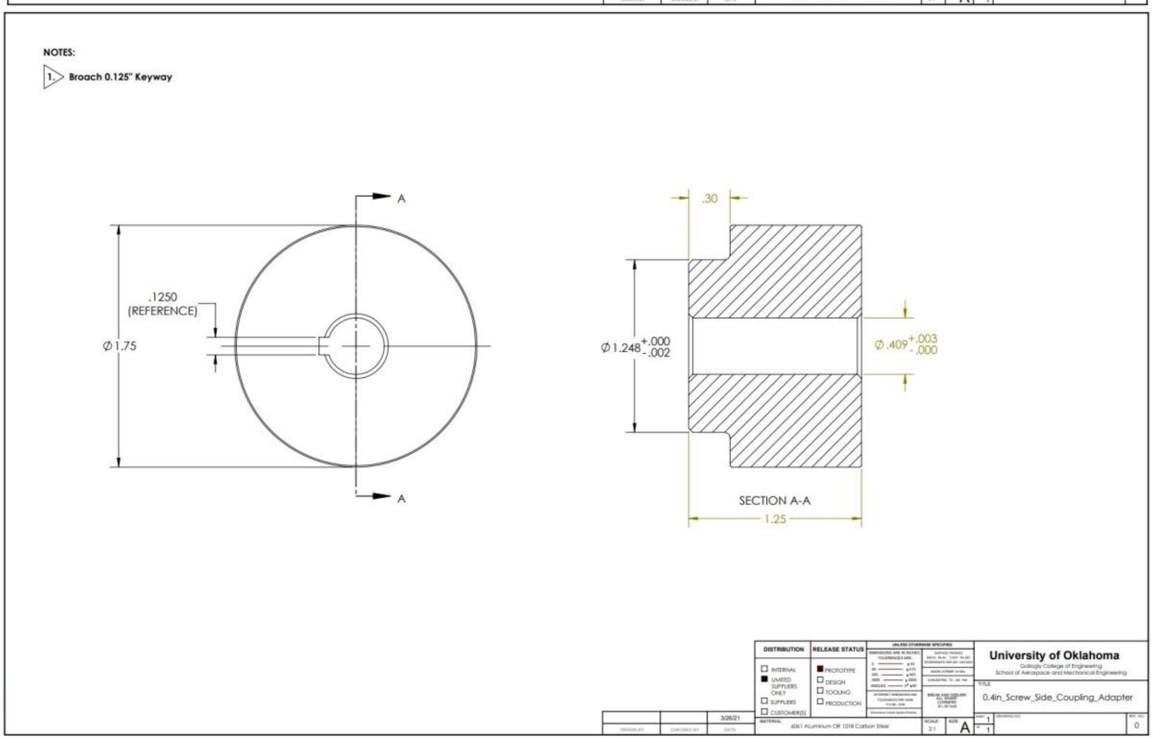
NOTES:
1. All 4 Slots ⊥ .002 To Datum B
2. Part Must Have Flat Bottom On Bore/Drill Tip Unacceptable
4X ∅ .469 Thru All
(15/32 Drill)
EQ SP ON 6.80 BC
.50 X 90° Both Sides
∅4.759
.0015 A
A
.03 X 45.00°
.500
.002 B
.002 B
∅4.759
∅5.16
.425
B
65.00°
R.10
R.15
.625
.875
40.00°
R.193
Slot Thru All
R.3125
Counter Bore Slot
∅8.25
∅3.000
University of Oklahoma
Bolt On Flange HP Housing
17-4 Stainless Steel
10207-30K-SLBHP-RECIP
Keegan, J
Perry, R
8/6/17

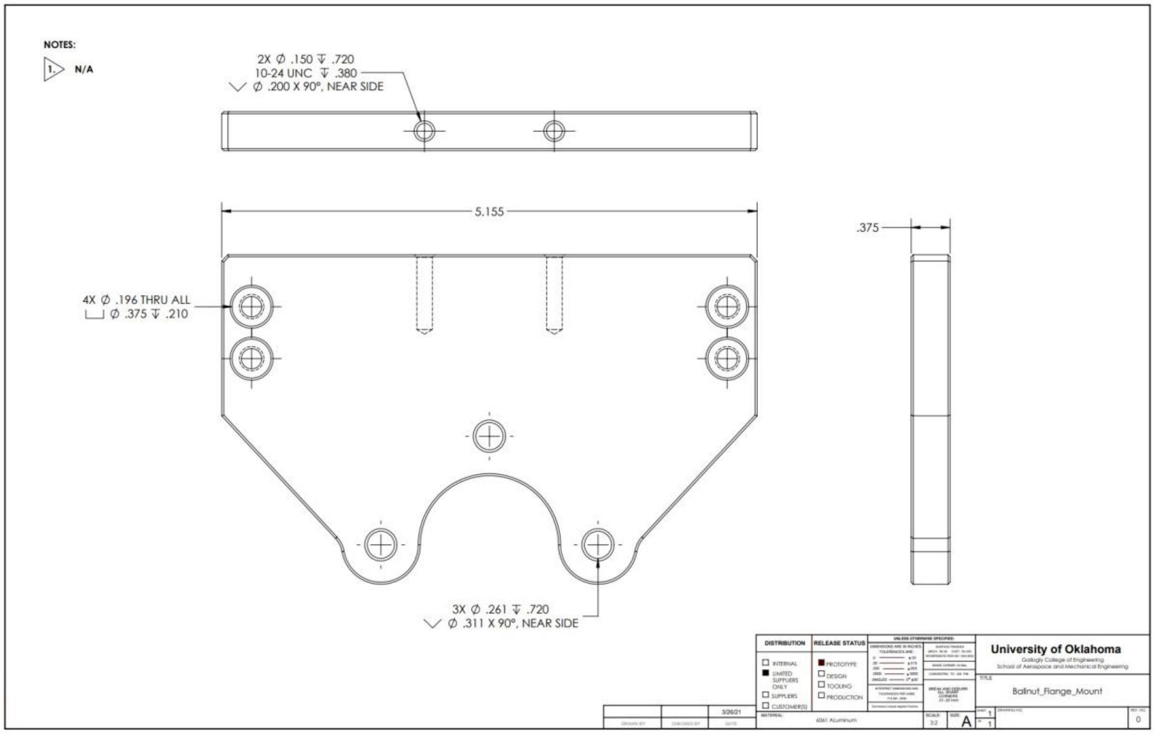

SECTION N-N
SCALE 1 : 0.75
Drill Tip Acceptable
Bore Surface Finish Not Critical
DETAIL A
SCALE 2 : 1
4 x ∅ 0.136 ↧ 0.750
#29 Drill
Equally Spaced .940 B.C.
Tap 8-36 ↧ .625
∅ 0.5
↧ 6.1 Full Diameter
DETAIL B
SCALE 2 : 0.75
University of Oklahoma
LVDT Piston
316L Stainless steel
LVDT100.002
Keegan, Jon
3/16/17

NOTES:
1. .06in 45deg Chamf. On All Edges
16 x ∅7mm ↧ 1.5in
Tap M8X1 Deep As Possible
THEOR. % OF THREAD ~75% Or Greater
80mm Centers
15 x ∅5/16 ↧ 1.5in
Tap 3/8-16 Deep As Possible
THEOR. % OF THREAD ~75% Or Greater
3.15in Centers
49.213±.100
(1250mm)
1.25±.06
As Cast
Or
Machined
University of Oklahoma
Linear Guide Rail Runner 1250mm
Keegan, Jon
11/16/17

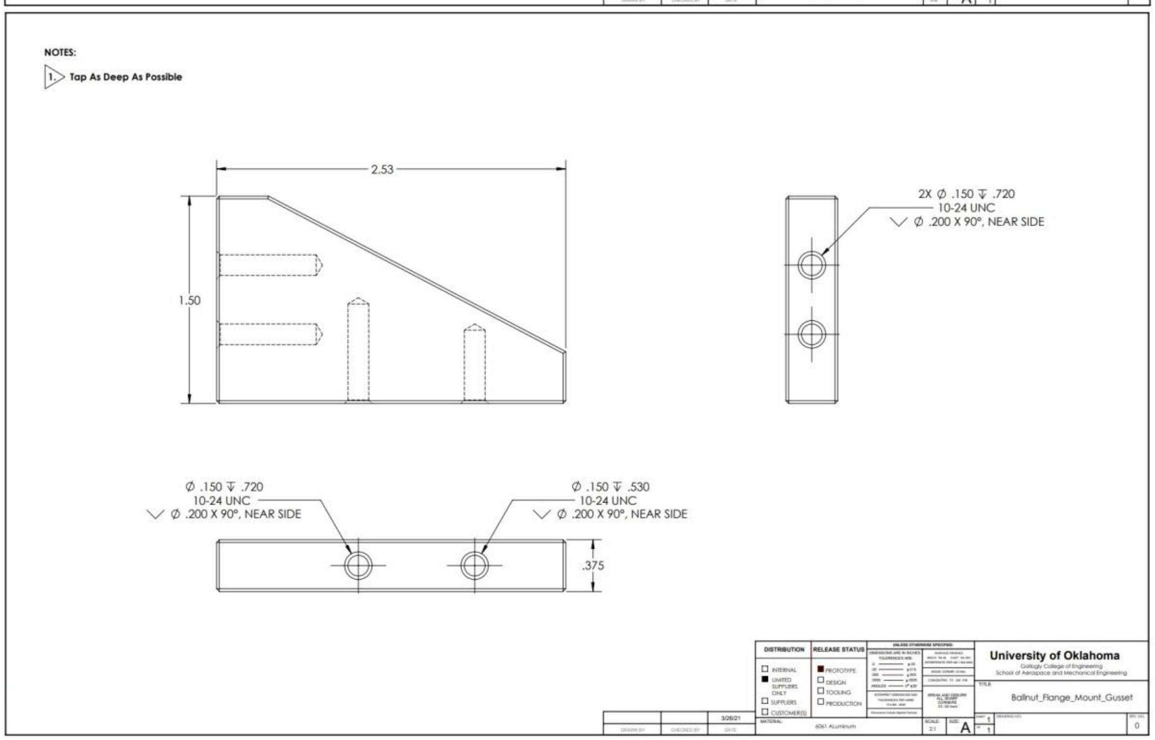

SECTION K-K
SCALE 4 : 5
DETAIL A
SCALE 3 : 2
A
64
16
R0.625
R0.06
2 PLCS.
+0.005
0.000
Ø 1.862
Ø 1.702
.003 B C
.002
4.310
0.415
0.5315 REF.
0.7452 REF.
0.0424 REF.
0.0377 REF.
B
C
6 x Ø .212 ⌵ .85
Number #3 Drill
Equally Spaced 2.4 B.C
Tap 1/4-28 Deep As Possible
.02
0.03 X 45°
2 PLCS.
Ø 2.352
Ø 2.925
1.200
0.875
5.80
University of Oklahoma
LVDT Housing
316L Stainless steel
Keegan, Jon
11/14/17
LVDT100.001

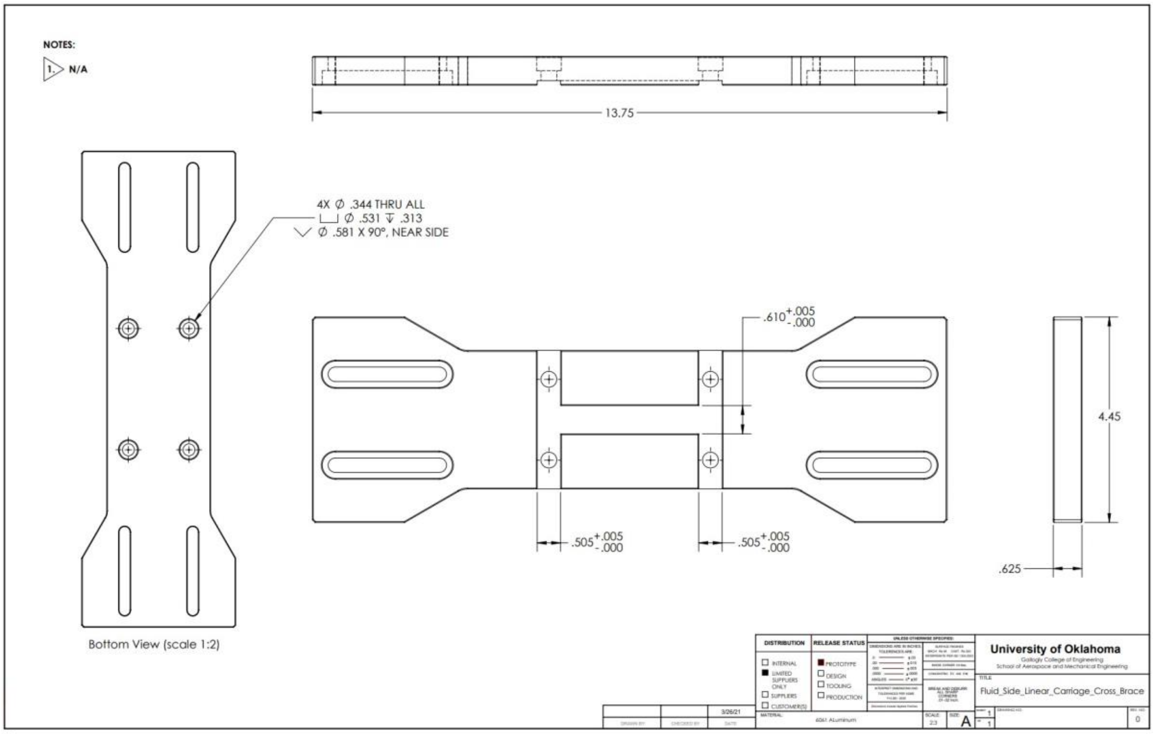
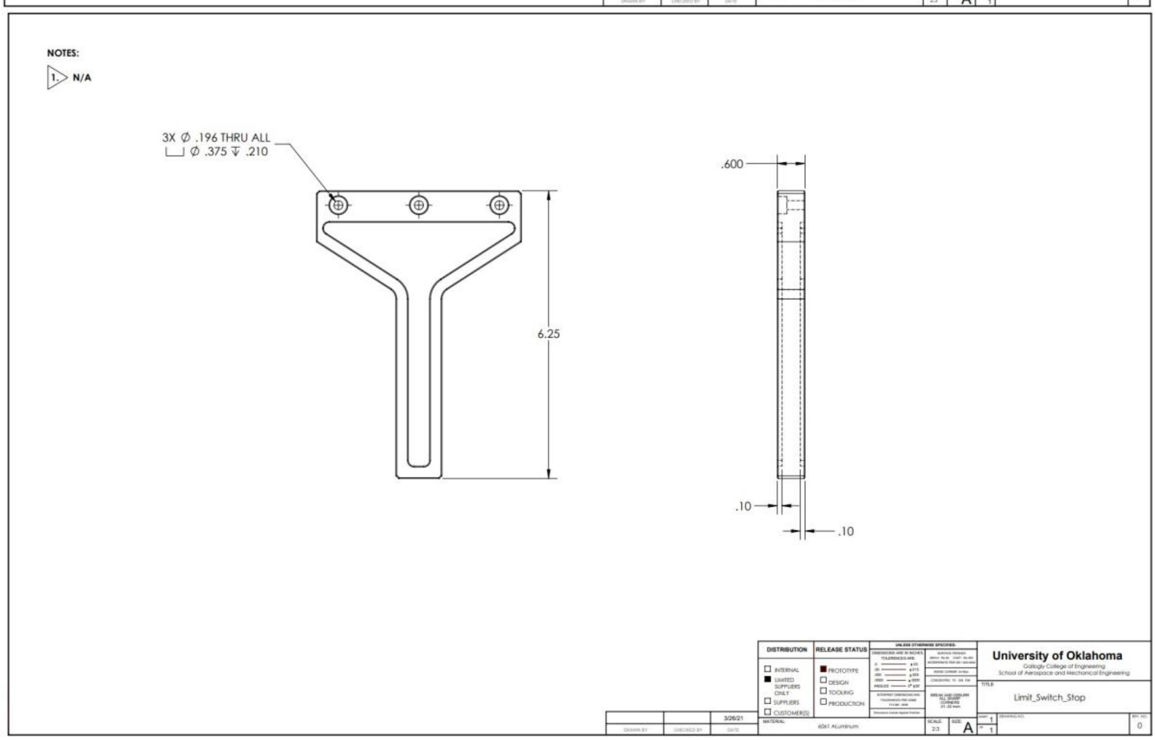

1.190
Ø .85
.75
.380
Ø .69
.08
.45
1.18
.03 X 45.00°
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Assembly Slot Key
1018 Carbon Steel
Keegan, J
Perry, R
8/6/17
10208-30K-SLBHP-RECIP

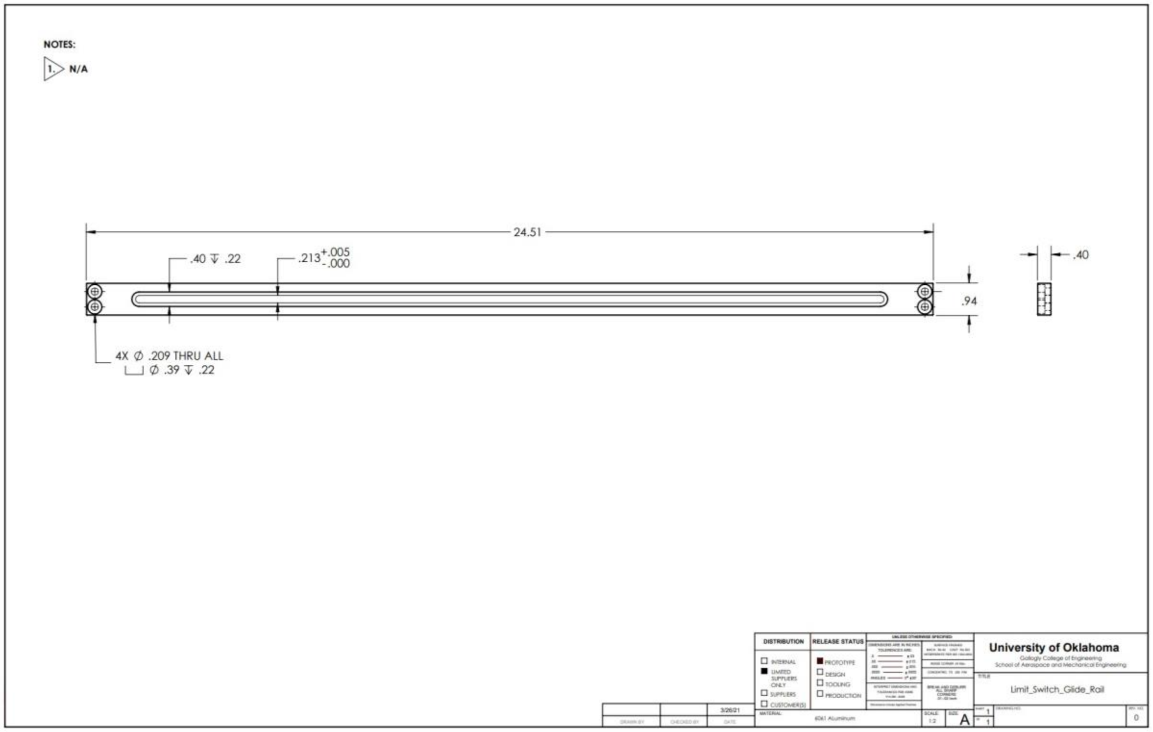

NOTES:
1. Broach For 6mm Keyway
.2392
(REFERENCE)
Ø1.75
.30
Ø1.250 +.002 -.000
Ø.7510 +.0030 -.0000
1.55
SECTION A-A
University of Oklahoma
19mm_Servo_Side_Coupling_Adapter
6061 Aluminum OR 1018 Carbon Steel
3/26/21

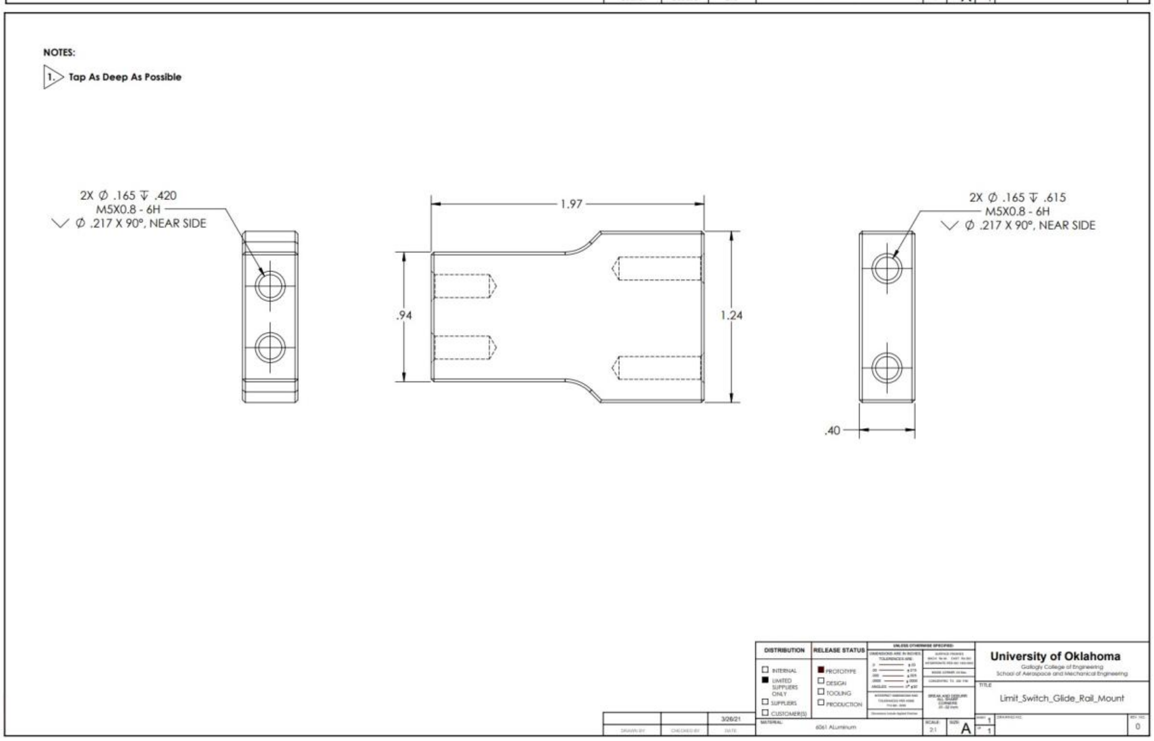

NOTES:
1. Broach 0.125" Keyway
.1250
(REFERENCE)
Ø1.75
.30
Ø1.248 +.000 -.002
Ø.409 +.003 -.000
SECTION A-A
1.25
University of Oklahoma
0.4in_Screw_Side_Coupling_Adapter
6061 Aluminum OR 1018 Carbon Steel
3/26/21

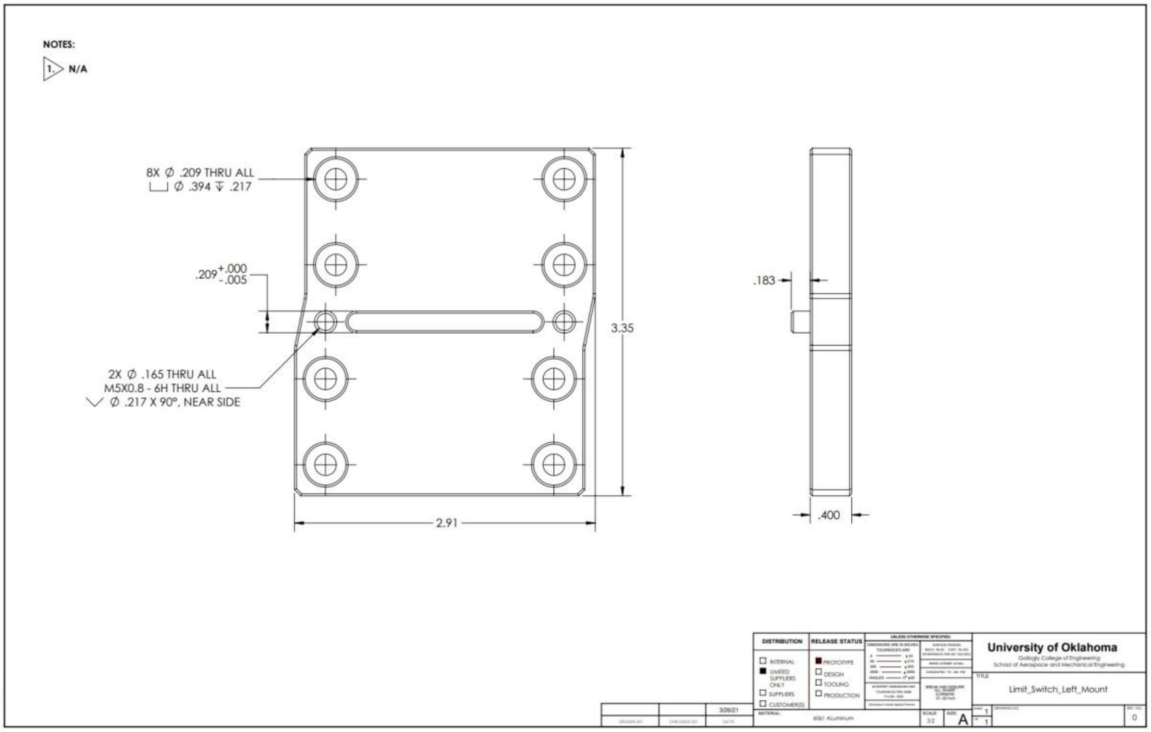
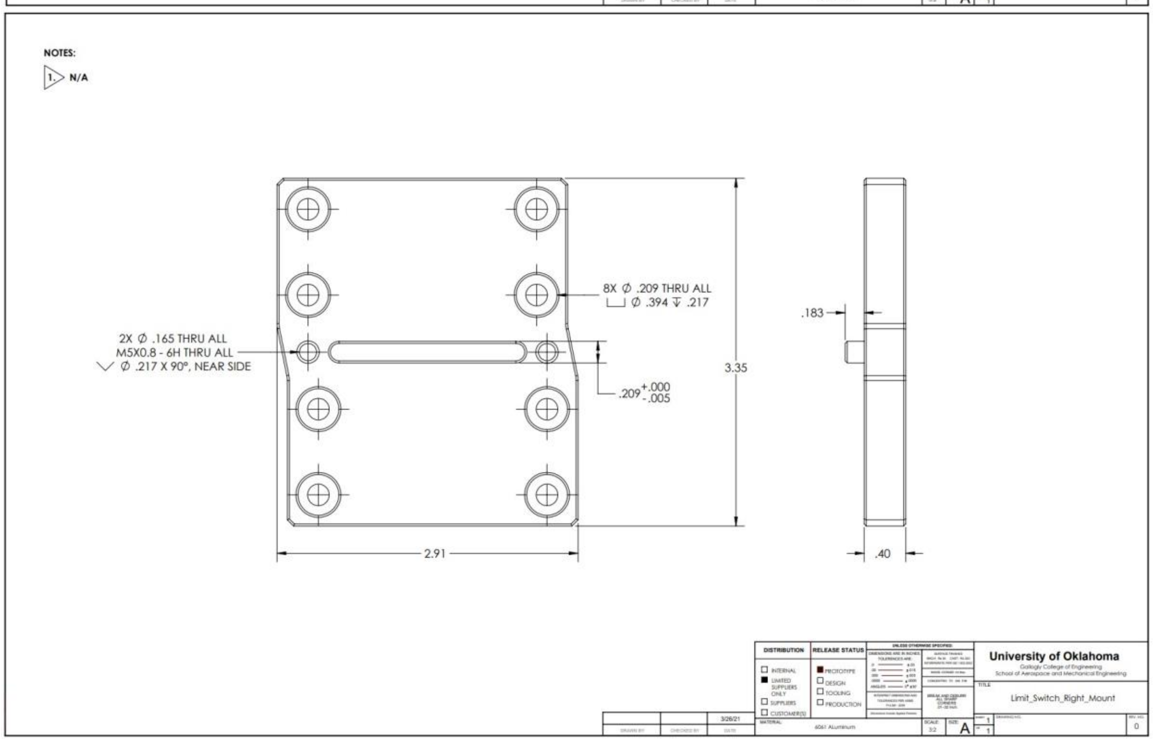

NOTES:
1. N/A
2X Ø .150 ↧ .720
10-24 UNC ↧ .380
⌵ Ø .200 X 90°, NEAR SIDE
5.155
.375
4X Ø .196 THRU ALL
⌴ Ø .375 ↧ .210
3X Ø .261 ↧ .720
⌵ Ø .311 X 90°, NEAR SIDE
University of Oklahoma
Ballnut_Flange_Mount
3/26/21
6061 Aluminum
A

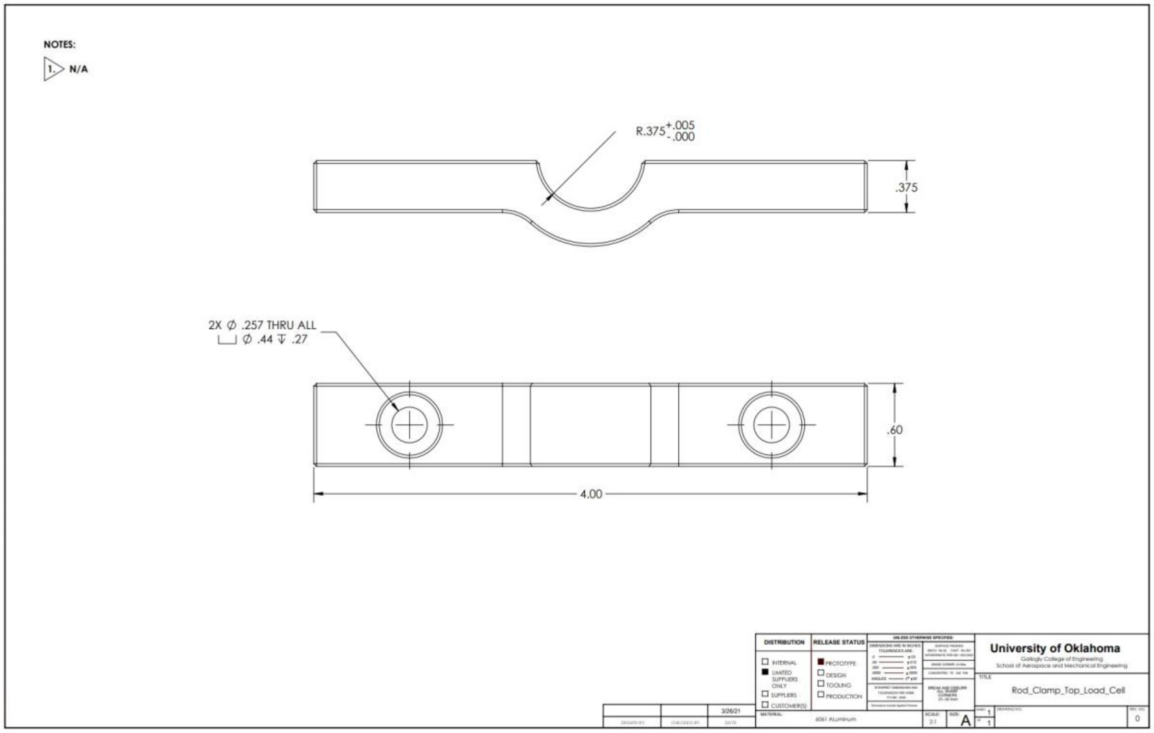

NOTES:
1. Tap As Deep As Possible
2.53
1.50
2X Ø .150 ↧ .720
10-24 UNC
⌵ Ø .200 X 90°, NEAR SIDE
Ø .150 ↧ .720
10-24 UNC
⌵ Ø .200 X 90°, NEAR SIDE
Ø .150 ↧ .530
10-24 UNC
⌵ Ø .200 X 90°, NEAR SIDE
.375
University of Oklahoma
Ballnut_Flange_Mount_Gusset
3/26/21
6061 Aluminum
A

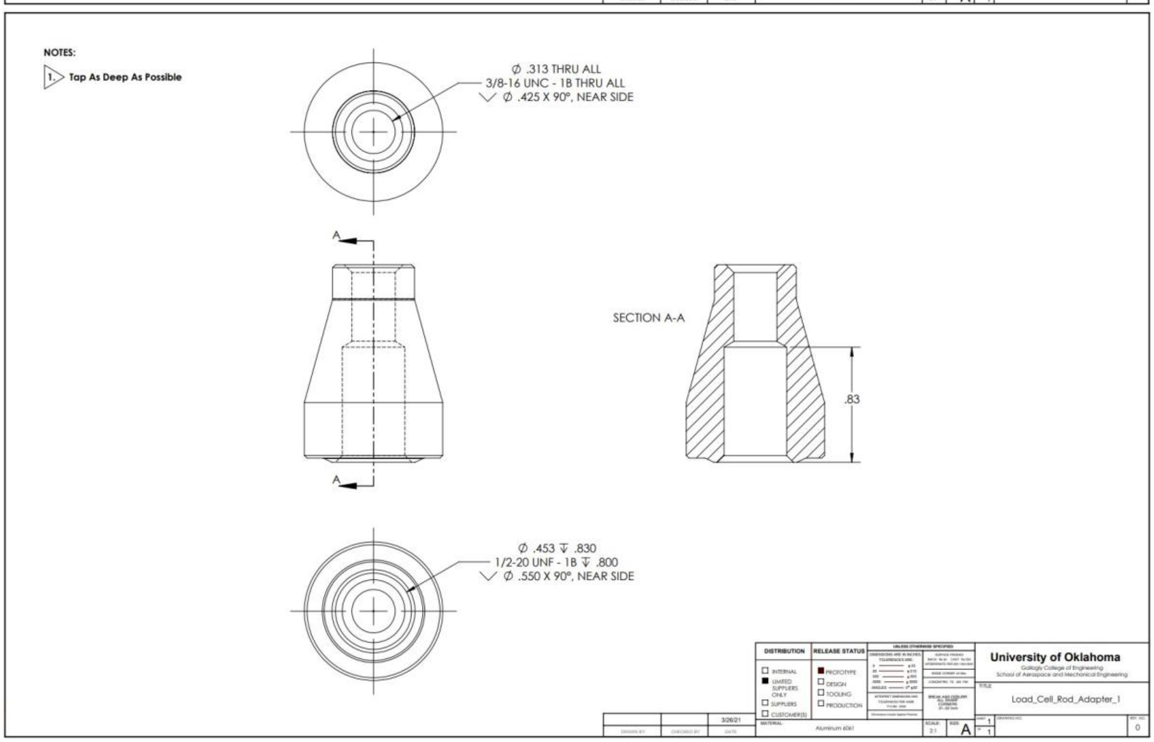

NOTES:
1. N/A
13.75
4X Ø .344 THRU ALL
⌴ Ø .531 ↧ .313
⌵ Ø .581 X 90°, NEAR SIDE
.610 +.005 -.000
4.45
.505 +.005 -.000
.505 +.005 -.000
.625
Bottom View (scale 1:2)
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
TITLE
Fluid_Side_Linear_Carriage_Cross_Brace
3/26/21
MATERIAL
6061 Aluminum
A
0

NOTES:
1. N/A
3X Ø .196 THRU ALL
⌴ Ø .375 ↧ .210
.600
6.25
.10
.10
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
TITLE
Limit_Switch_Stop
3/26/21
MATERIAL
6061 Aluminum
A
0

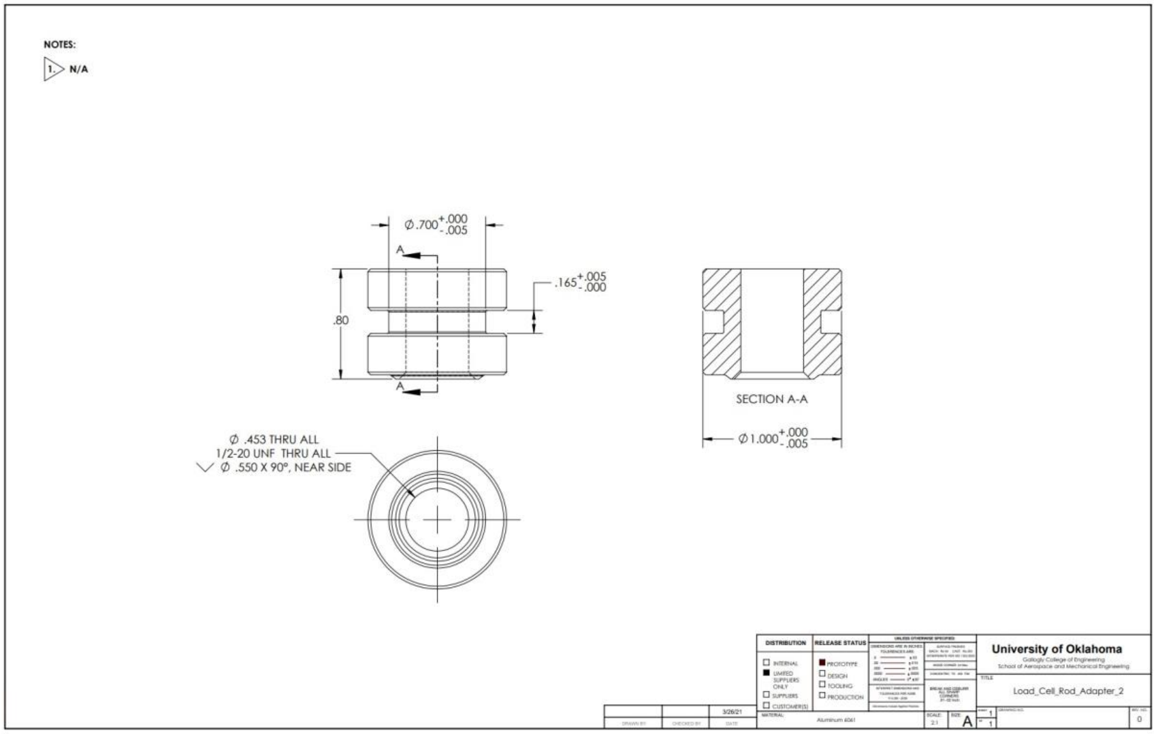

NOTES:
1. N/A
24.51
.40 ↧ .22
.213 +.005 -.000
.40
.94
4X Ø .209 THRU ALL
⌴ Ø .39 ↧ .22
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Limit_Switch_Glide_Rail
3/26/21
MATERIAL:
6061 Aluminum
1:2
A
0


NOTES:
1. Tap As Deep As Possible
2X Ø .165 ↧ .420
M5X0.8 - 6H
⌵ Ø .217 X 90°, NEAR SIDE
1.97
.94
1.24
2X Ø .165 ↧ .615
M5X0.8 - 6H
⌵ Ø .217 X 90°, NEAR SIDE
.40
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Limit_Switch_Glide_Rail_Mount
3/26/21
MATERIAL:
6061 Aluminum
2:1
A
0

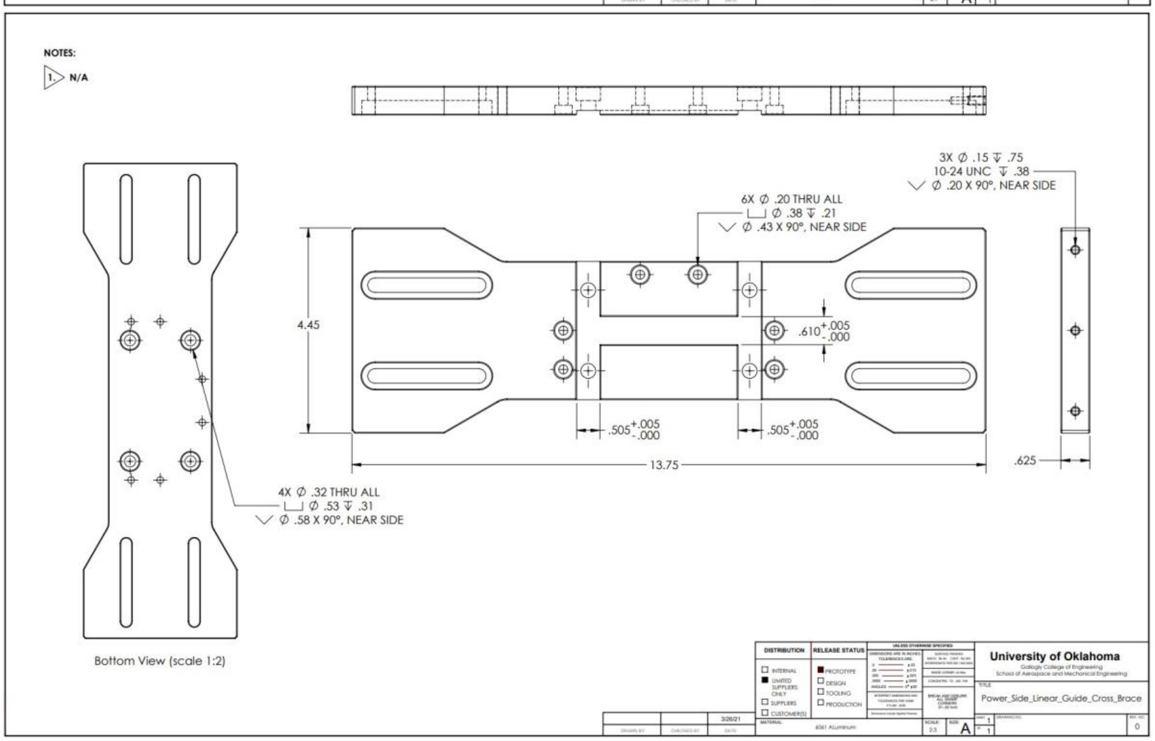

NOTES:
1. N/A
8X Ø .209 THRU ALL
⌴ Ø .394 ↧ .217
.209 +.000 -.005
2X Ø .165 THRU ALL
M5X0.8 - 6H THRU ALL
⌵ Ø .217 X 90°, NEAR SIDE
3.35
2.91
.183
.400
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
TITLE
Limit_Switch_Left_Mount
3/26/21
MATERIAL
6061 Aluminum
SCALE
3:2
SIZE
A
0

NOTES:
1. N/A
8X Ø .209 THRU ALL
⌴ Ø .394 ↧ .217
2X Ø .165 THRU ALL
M5X0.8 - 6H THRU ALL
⌵ Ø .217 X 90°, NEAR SIDE
.209 +.000 -.005
3.35
2.91
.183
.40
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
TITLE
Limit_Switch_Right_Mount
3/26/21
MATERIAL
6061 Aluminum
SCALE
3:2
SIZE
A
0


NOTES:
1. N/A
R.375 +.005 -.000
.375
2X ∅ .257 THRU ALL
∅ .44 ↧ .27
.60
4.00
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
TITLE
Rod_Clamp_Top_Load_Cell
3/26/21
MATERIAL
6061 Aluminum
SCALE 2:1
SIZE A
0

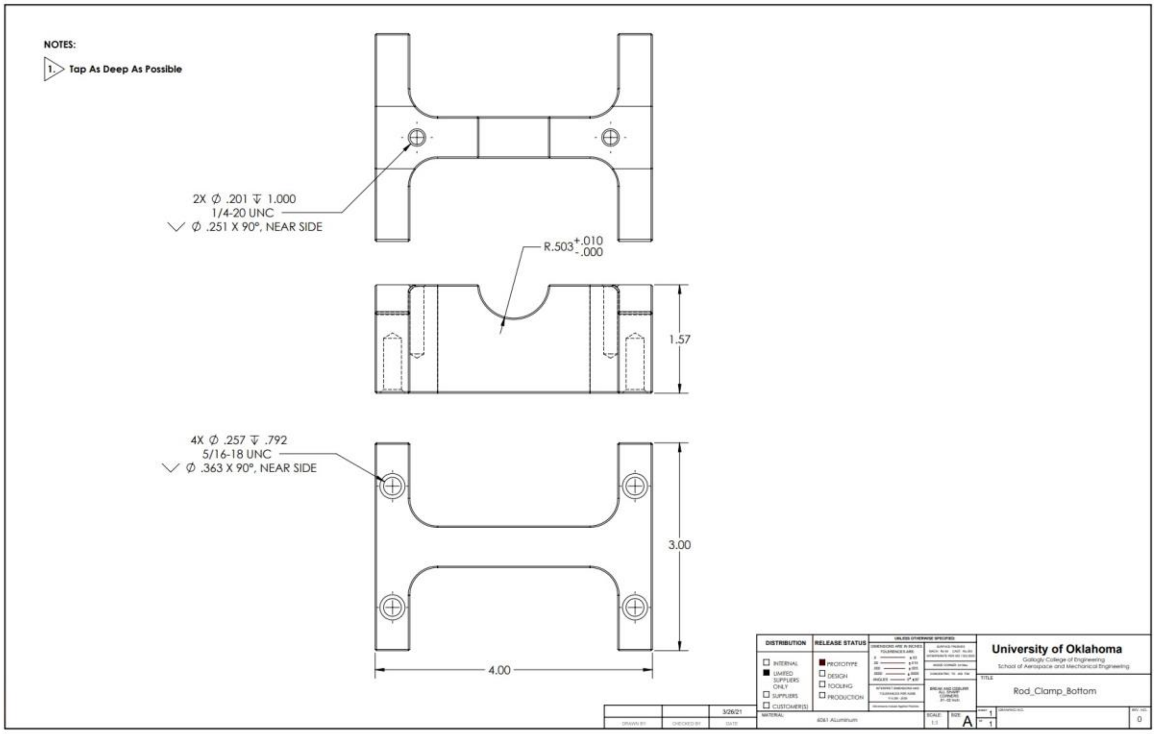
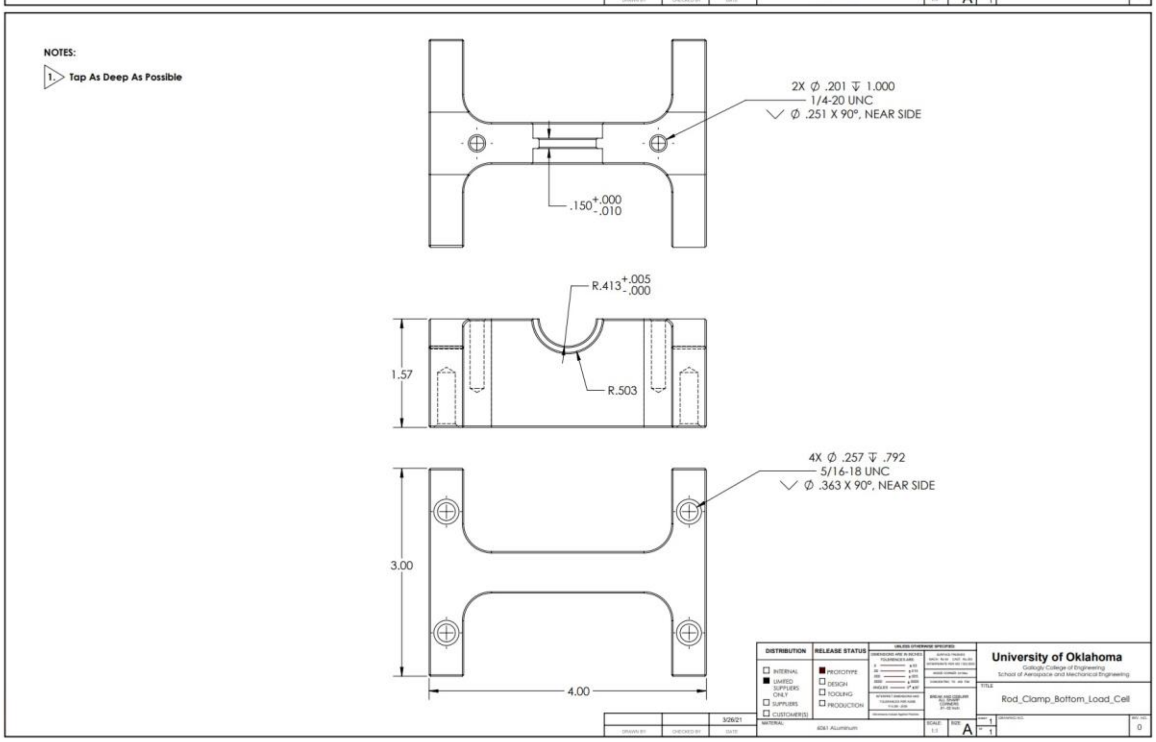

NOTES:
1. Tap As Deep As Possible
∅ .313 THRU ALL
3/8-16 UNC - 1B THRU ALL
∅ .425 X 90°, NEAR SIDE
A
A
SECTION A-A
.83
∅ .453 ↧ .830
1/2-20 UNF - 1B ↧ .800
∅ .550 X 90°, NEAR SIDE
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
TITLE
Load_Cell_Rod_Adapter_1
3/26/21
MATERIAL
Aluminum 6061
SCALE 2:1
SIZE A
0

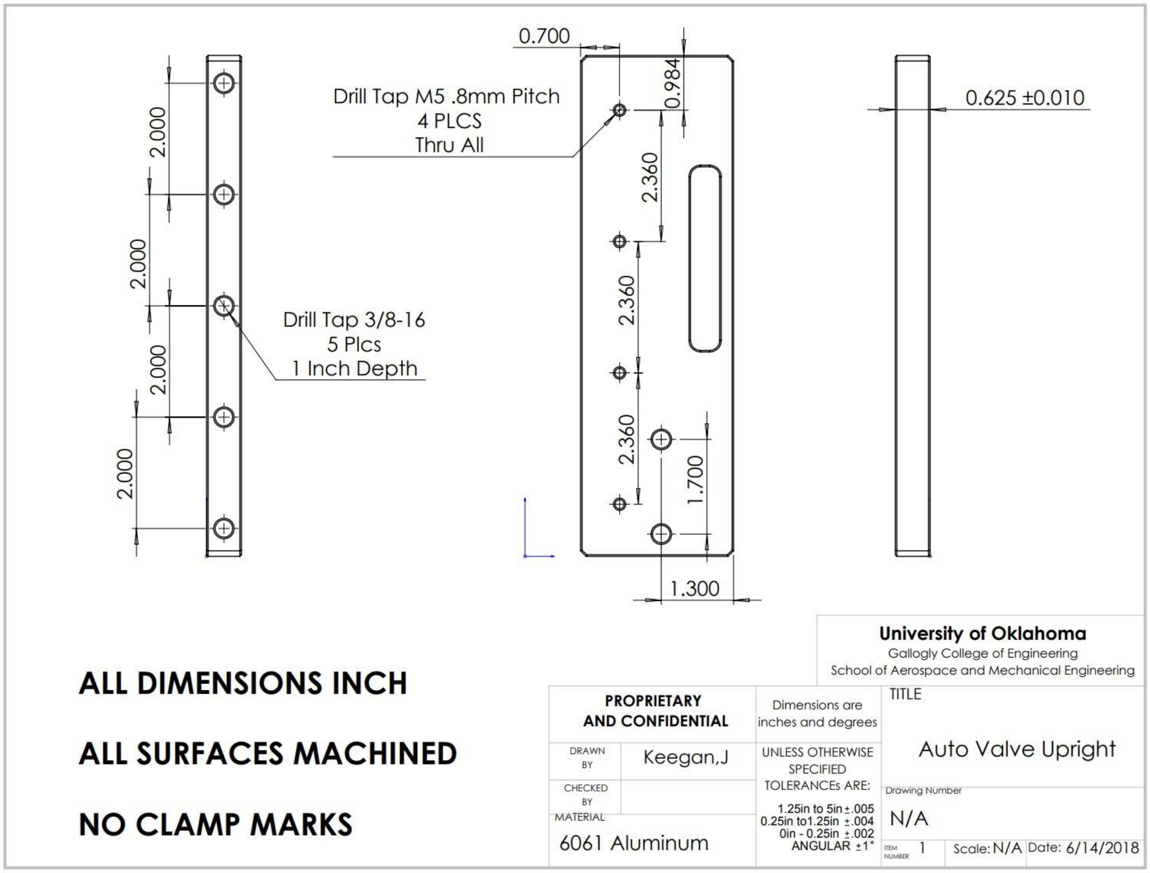
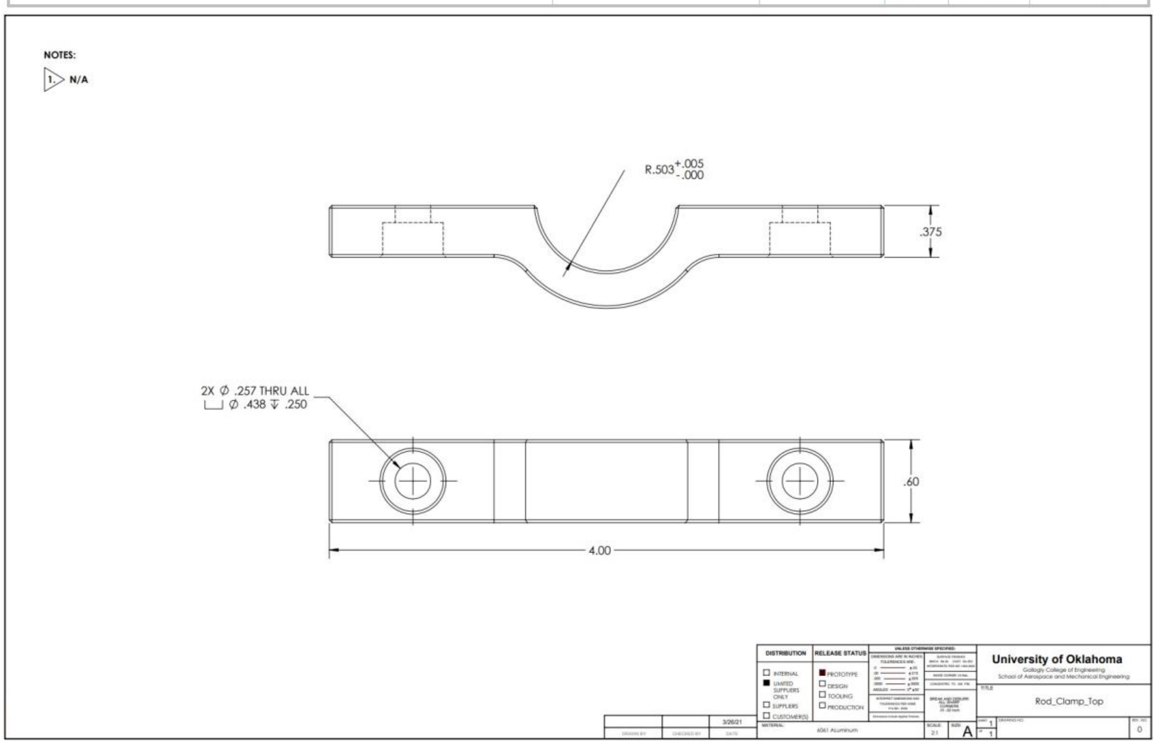


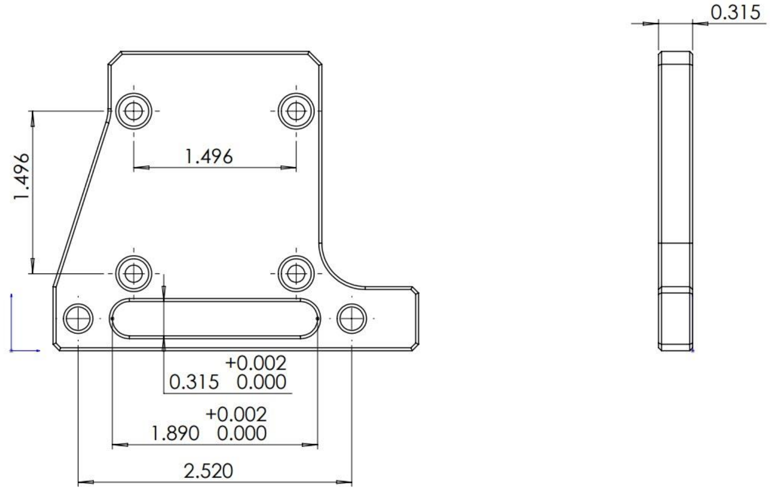
NOTES:
1. N/A
Ø .700 +.000 -.005
.165 +.005 -.000
.80
SECTION A-A
Ø 1.000 +.000 -.005
Ø .453 THRU ALL
1/2-20 UNF THRU ALL
Ø .550 X 90°, NEAR SIDE
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Load_Cell_Rod_Adapter_2
Aluminum 6061
3/26/21



NOTES:
1. N/A
3X Ø .15 ↧ .75
10-24 UNC ↧ .38
Ø .20 X 90°, NEAR SIDE
6X Ø .20 THRU ALL
⌴ Ø .38 ↧ .21
Ø .43 X 90°, NEAR SIDE
4.45
.610 +.005 -.000
.505 +.005 -.000
.505 +.005 -.000
13.75
.625
4X Ø .32 THRU ALL
⌴ Ø .53 ↧ .31
Ø .58 X 90°, NEAR SIDE
Bottom View (scale 1:2)
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Power_Side_Linear_Guide_Cross_Brace
6061 Aluminum
3/26/21


NOTES:
1. Tap As Deep As Possible
2X Ø .201 ↧ 1.000
1/4-20 UNC
⌵ Ø .251 X 90°, NEAR SIDE
R.503 +.010 -.000
1.57
4X Ø .257 ↧ .792
5/16-18 UNC
⌵ Ø .363 X 90°, NEAR SIDE
3.00
4.00
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Rod_Clamp_Bottom
6061 Aluminum
3/26/21
A

NOTES:
1. Tap As Deep As Possible
2X Ø .201 ↧ 1.000
1/4-20 UNC
⌵ Ø .251 X 90°, NEAR SIDE
.150 +.000 -.010
R.413 +.005 -.000
R.503
1.57
4X Ø .257 ↧ .792
5/16-18 UNC
⌵ Ø .363 X 90°, NEAR SIDE
3.00
4.00
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Rod_Clamp_Bottom_Load_Cell
6061 Aluminum
3/26/21
A


0.700
Drill Tap M5 .8mm Pitch
4 PLCS
Thru All
0.984
0.625 ±0.010
2.000
2.000
2.000
2.000
2.360
2.360
2.360
1.700
1.300
Drill Tap 3/8-16
5 Plcs
1 Inch Depth
ALL DIMENSIONS INCH
ALL SURFACES MACHINED
NO CLAMP MARKS
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
PROPRIETARY AND CONFIDENTIAL
Dimensions are inches and degrees
TITLE
Auto Valve Upright
DRAWN BY
Keegan,J
UNLESS OTHERWISE SPECIFIED TOLERANCEs ARE:
CHECKED BY
1.25in to 5in ±.005
0.25in to1.25in ±.004
0in - 0.25in ±.002
ANGULAR ±1°
MATERIAL
6061 Aluminum
Drawing Number
N/A
ITEM NUMBER 1
Scale: N/A
Date: 6/14/2018

NOTES:
1. N/A
R.503 +.005 -.000
.375
2X Ø .257 THRU ALL
⌴ Ø .438 ↧ .250
.60
4.00
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Rod_Clamp_Top
3/26/21
MATERIAL
6061 Aluminum
A
0



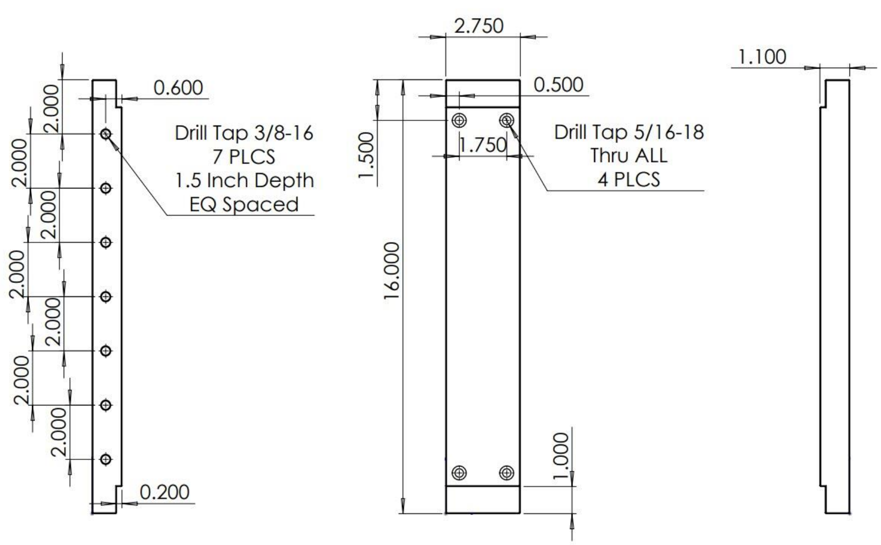
ALL DIMENSIONS INCH

ALL SURFACES MACHINED

NO CLAMP MARKS

University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering

PROPRIETARY AND CONFIDENTIAL	Dimensions are inches and degrees	TITLE Carriage Stepper Base
DRAWN BY: Keegan,J	UNLESS OTHERWISE SPECIFIED TOLERANCEs ARE:	Drawing Number N/A
CHECKED BY:	1.25in to 5in ±.005 0.25in to1.25in ±.004 0in - 0.25in ±.002 ANGULAR ±1°	ITEM NUMBER 1
MATERIAL 6061 Aluminum		Scale: N/A Date: 6/14/2018



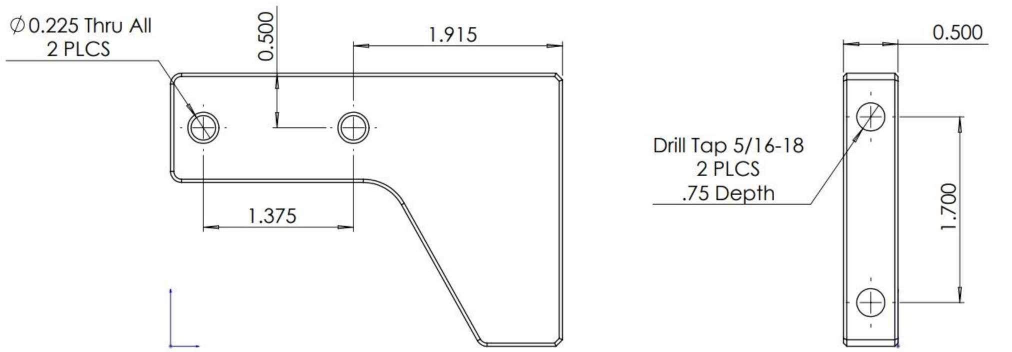
ALL DIMENSIONS INCH

Machine All Surfaces

No Clamp Marks

University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering

PROPRIETARY AND CONFIDENTIAL	Dimensions are inches and degrees	TITLE Frame Mount Upright
DRAWN BY: Keegan,J	UNLESS OTHERWISE SPECIFIED TOLERANCEs ARE:	Drawing Number N/A
CHECKED BY:	1.25in to 5in ±.005 0.25in to1.25in ±.004 0in - 0.25in ±.002 ANGULAR ±1°	ITEM NUMBER 1
MATERIAL 6061 Aluminum		Scale: N/A Date: 6/18/2018



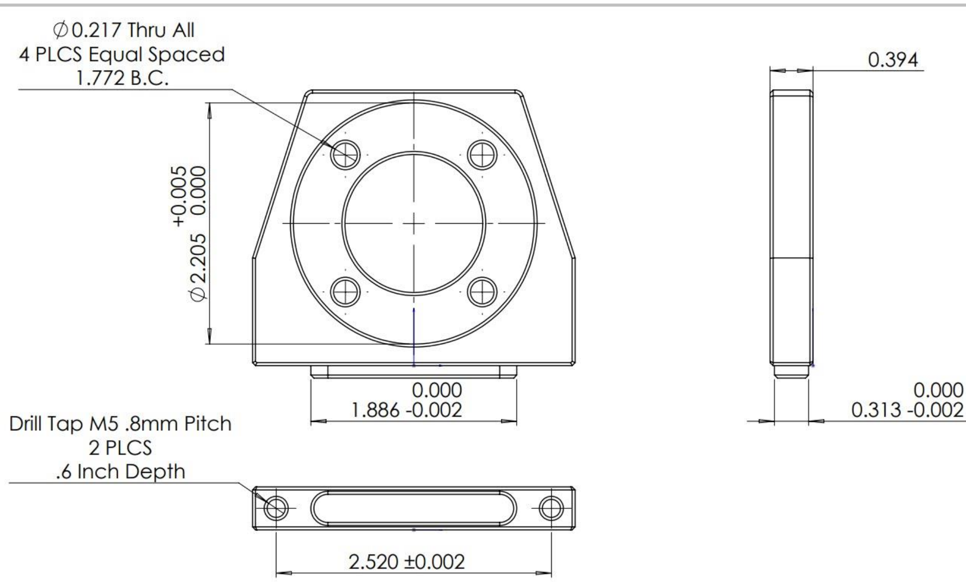
ALL DIMENSIONS INCH

ALL SURFACES MACHINED

NO CLAMP MARKS

University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering

PROPRIETARY AND CONFIDENTIAL		Dimensions are inches and degrees	TITLE
DRAWN BY	Keegan,J	UNLESS OTHERWISE SPECIFIED TOLERANCEs ARE:	Valve Mounting Bracket
CHECKED BY		1.25in to 5in ±.005 0.25in to1.25in ±.004 0in - 0.25in ±.002 ANGULAR ±1°	Drawing Number N/A
MATERIAL 303 Stainless Steel			ITEM NUMBER 1 — Scale: N/A — Date: 6/14/2018



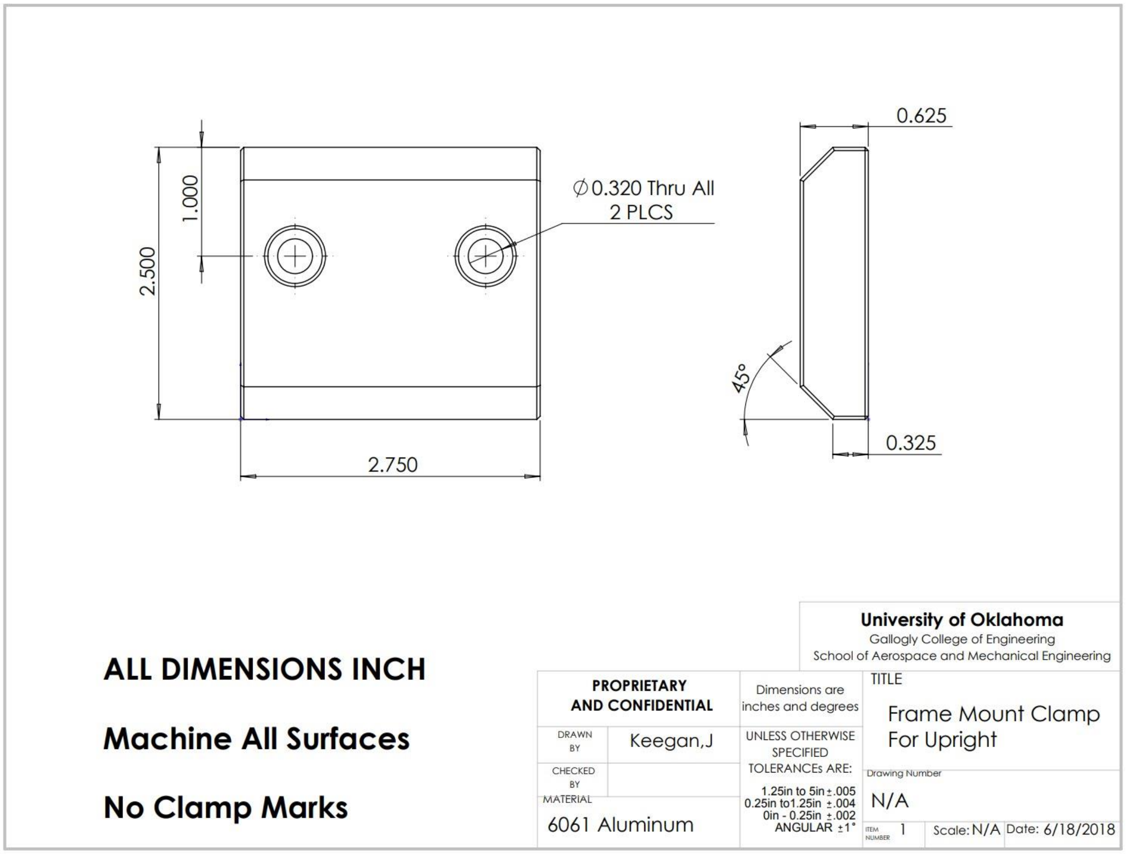
ALL DIMENSIONS INCH

ALL SURFACES MACHINED

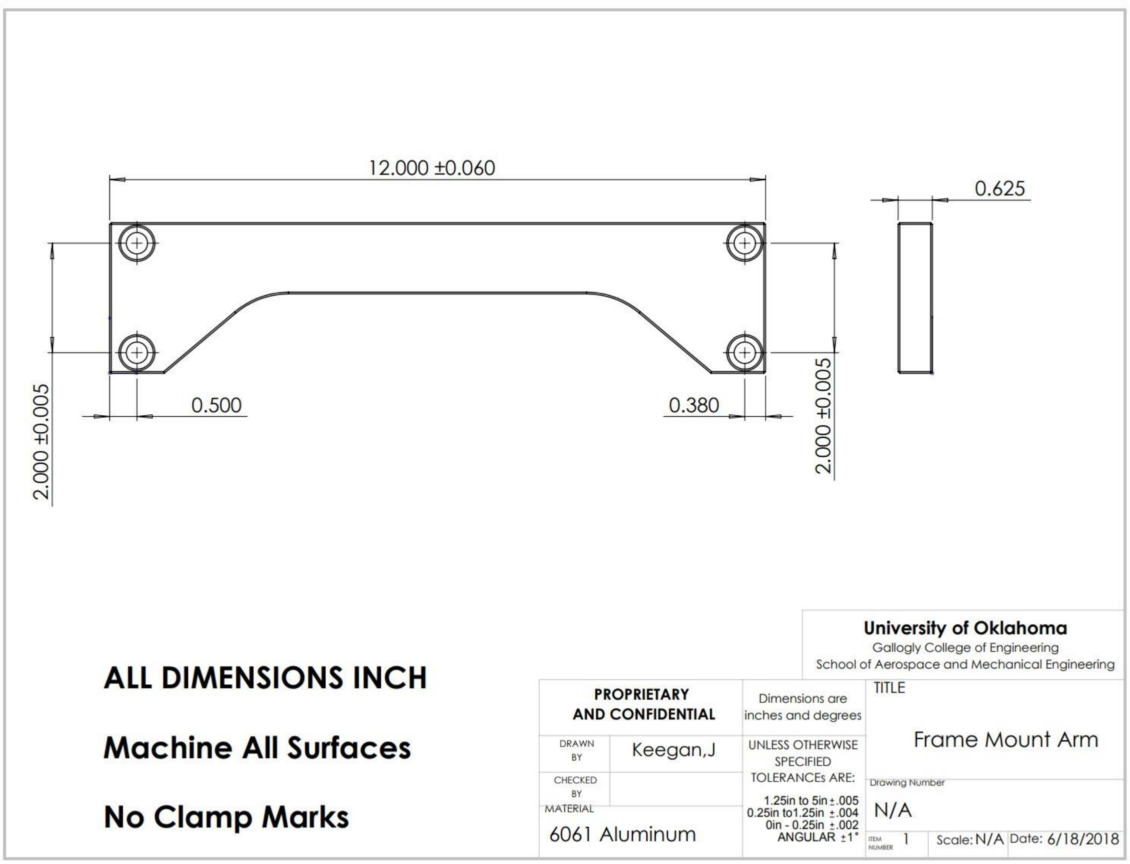
NO CLAMP MARKS

University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering

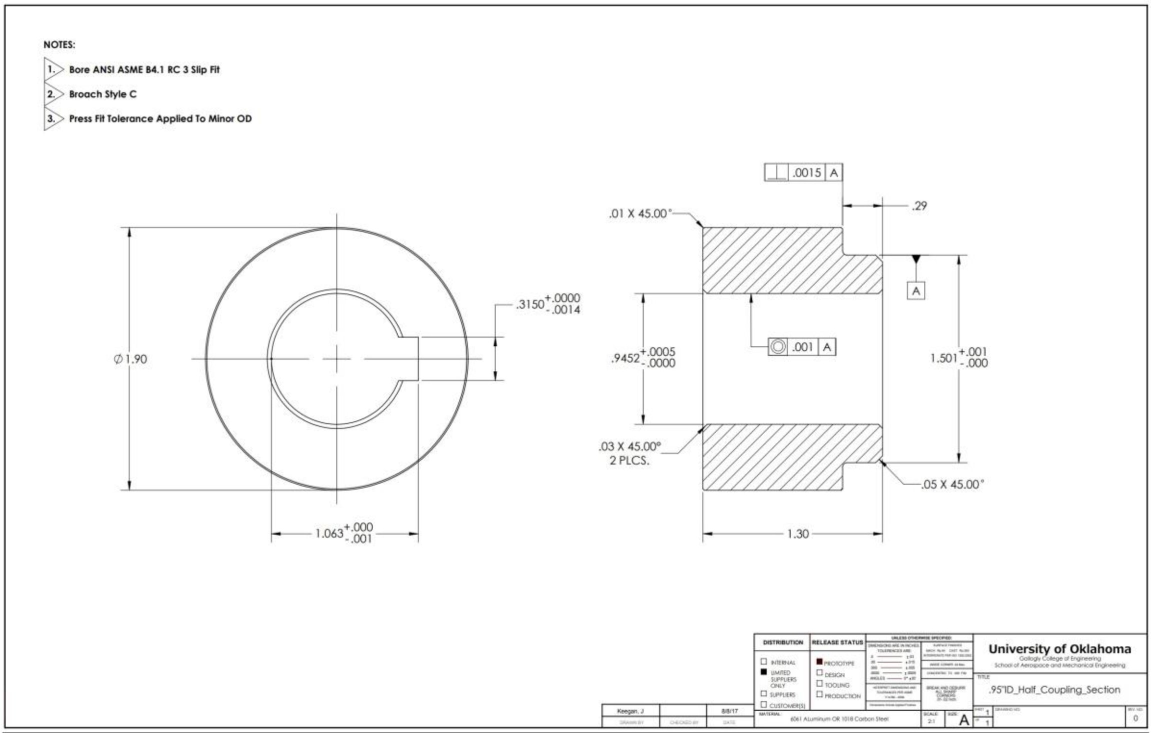
PROPRIETARY AND CONFIDENTIAL		Dimensions are inches and degrees	TITLE
DRAWN BY	Keegan,J	UNLESS OTHERWISE SPECIFIED TOLERANCEs ARE:	Carriage Stepper Upright
CHECKED BY		1.25in to 5in ±.005 0.25in to1.25in ±.004 0in - 0.25in ±.002 ANGULAR ±1°	Drawing Number N/A
MATERIAL 6061 Aluminum			ITEM NUMBER 1 — Scale: N/A — Date: 6/14/2018

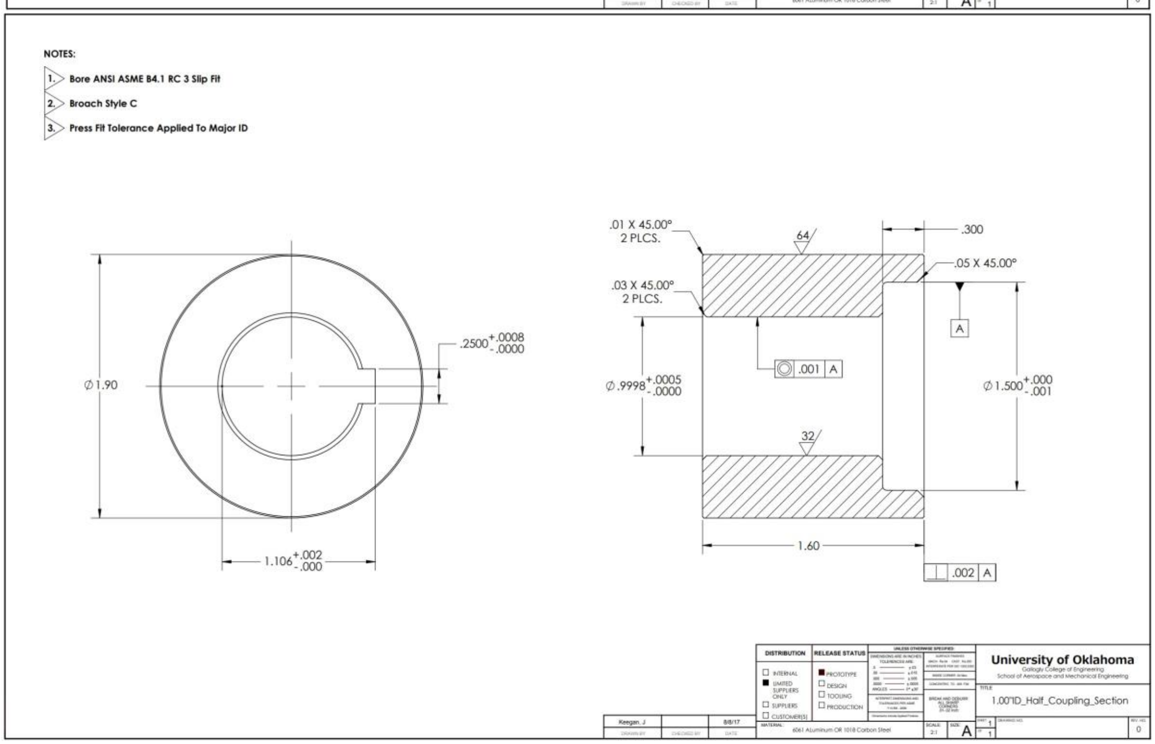


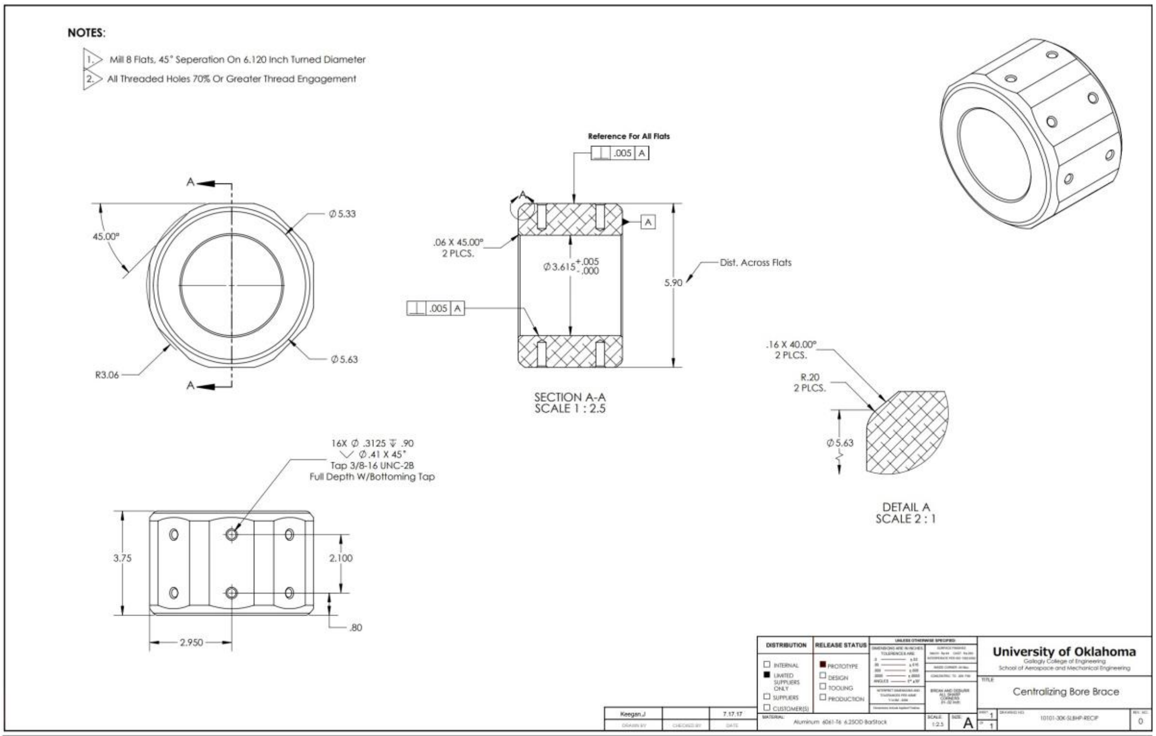
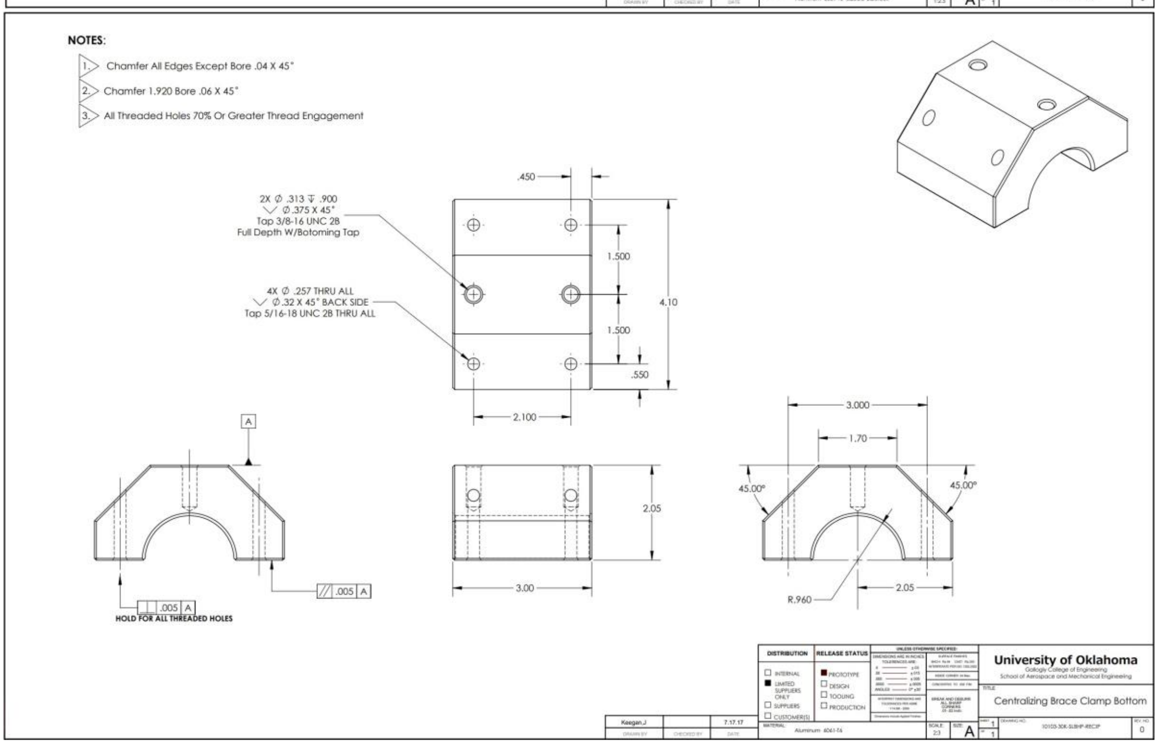
0.625
⌀0.320 Thru All
2 PLCS
1.000
2.500
45°
0.325
2.750
ALL DIMENSIONS INCH
Machine All Surfaces
No Clamp Marks
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
PROPRIETARY AND CONFIDENTIAL
Dimensions are inches and degrees
TITLE
Frame Mount Clamp For Upright
DRAWN BY
Keegan,J
UNLESS OTHERWISE SPECIFIED TOLERANCES ARE:
CHECKED BY
Drawing Number
MATERIAL
1.25in to 5in ±.005
0.25in to1.25in ±.004
0in - 0.25in ±.002
ANGULAR ±1°
N/A
6061 Aluminum
ITEM NUMBER 1
Scale: N/A
Date: 6/18/2018

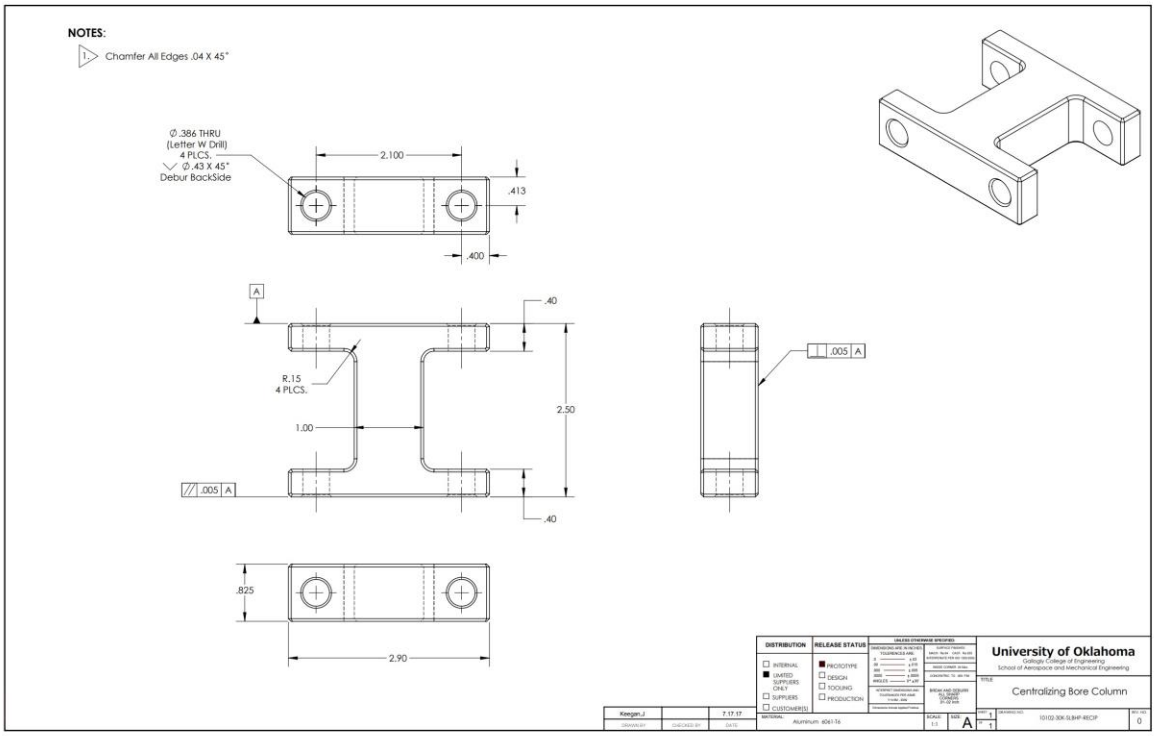


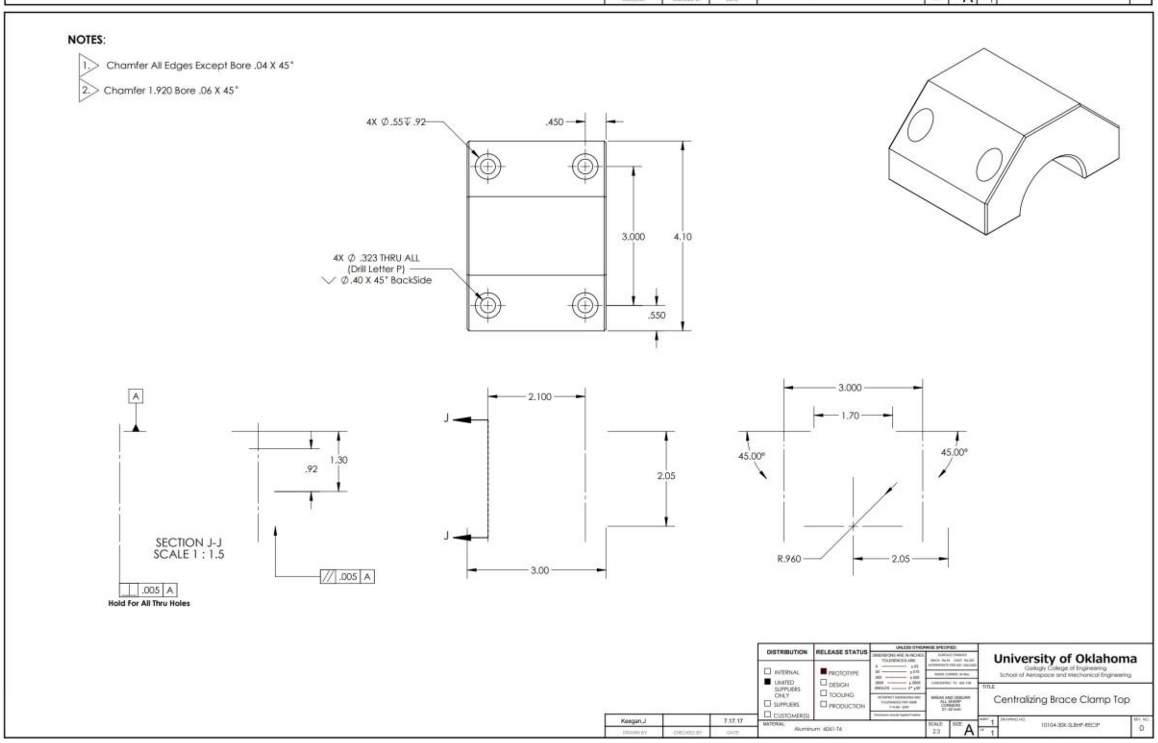
12.000 ±0.060
0.625
0.500
0.380
2.000 ±0.005
2.000 ±0.005
ALL DIMENSIONS INCH
Machine All Surfaces
No Clamp Marks
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
PROPRIETARY AND CONFIDENTIAL
Dimensions are inches and degrees
TITLE
Frame Mount Arm
DRAWN BY
Keegan,J
UNLESS OTHERWISE SPECIFIED TOLERANCES ARE:
CHECKED BY
Drawing Number
MATERIAL
1.25in to 5in ±.005
0.25in to1.25in ±.004
0in - 0.25in ±.002
ANGULAR ±1°
N/A
6061 Aluminum
ITEM NUMBER 1
Scale: N/A
Date: 6/18/2018


NOTES:
1. Bore ANSI ASME B4.1 RC 3 Slip Fit
2. Broach Style C
3. Press Fit Tolerance Applied To Minor OD
.0015 A
.29
.01 X 45.00°
A
.3150 +.0000 -.0014
.9452 +.0005 -.0000
.001 A
1.501 +.001 -.000
Ø1.90
.03 X 45.00° 2 PLCS.
.05 X 45.00°
1.063 +.000 -.001
1.30
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
.95"ID_Half_Coupling_Section
Keegan, J
8/8/17
6061 Aluminum OR 1018 Carbon Steel
2:1
A
1
0

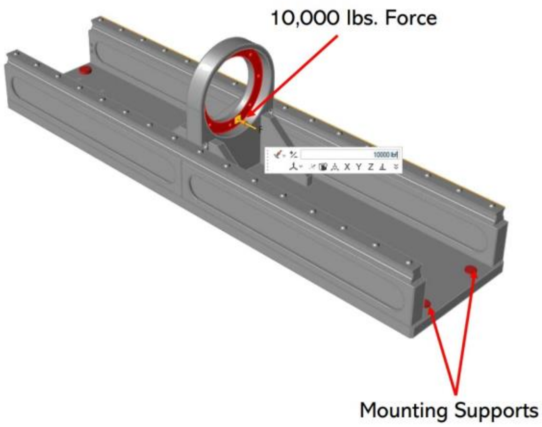

NOTES:
1. Bore ANSI ASME B4.1 RC 3 Slip Fit
2. Broach Style C
3. Press Fit Tolerance Applied To Major ID
.01 X 45.00° 2 PLCS.
64
.300
.05 X 45.00°
.03 X 45.00° 2 PLCS.
A
.2500 +.0008 -.0000
.001 A
Ø1.90
Ø.9998 +.0005 -.0000
Ø1.500 +.000 -.001
32
1.60
.002 A
1.106 +.002 -.000
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
1.00"ID_Half_Coupling_Section
Keegan, J
8/8/17
6061 Aluminum OR 1018 Carbon Steel
2:1
A
1
0

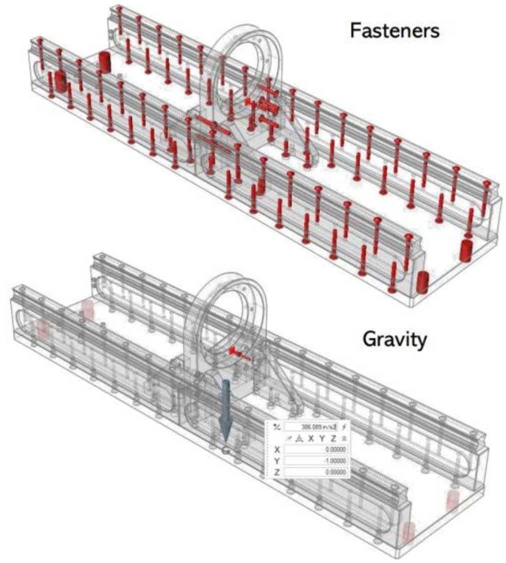

NOTES:
1. Mill 8 Flats, 45° Seperation On 6.120 Inch Turned Diameter
2. All Threaded Holes 70% Or Greater Thread Engagement
Reference For All Flats
.005 A
45.00°
Ø5.33
Ø5.63
R3.06
.06 X 45.00°
2 PLCS.
Ø3.615 +.005 -.000
5.90
Dist. Across Flats
SECTION A-A
SCALE 1 : 2.5
.16 X 40.00°
2 PLCS.
R.20
2 PLCS.
Ø5.63
DETAIL A
SCALE 2 : 1
16X Ø .3125 ↧ .90
⌵ Ø .41 X 45°
Tap 3/8-16 UNC-2B
Full Depth W/Bottoming Tap
3.75
2.100
.80
2.950
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Centralizing Bore Brace
Keegan,J
7.17.17
Aluminum 6061-T6 6.25OD BarStock
1:2.5
A
0

NOTES:
1. Chamfer All Edges Except Bore .04 X 45°
2. Chamfer 1.920 Bore .06 X 45°
3. All Threaded Holes 70% Or Greater Thread Engagement
2X Ø .313 ↧ .900
⌵ Ø .375 X 45°
Tap 3/8-16 UNC 2B
Full Depth W/Botoming Tap
4X Ø .257 THRU ALL
⌵ Ø .32 X 45° BACK SIDE
Tap 5/16-18 UNC 2B THRU ALL
.450
1.500
1.500
4.10
.550
2.100
3.000
1.70
45.00°
45.00°
A
2.05
3.00
R.960
2.05
.005 A
HOLD FOR ALL THREADED HOLES
// .005 A
DISTRIBUTION
RELEASE STATUS
INTERNAL
LIMITED SUPPLIERS ONLY
SUPPLIERS
CUSTOMER(S)
PROTOTYPE
DESIGN
TOOLING
PRODUCTION
University of Oklahoma
Gallogly College of Engineering
School of Aerospace and Mechanical Engineering
Centralizing Brace Clamp Bottom
Keegan,J
7.17.17
Aluminum 6061-T6
2:3
A
0


NOTES:
1. Chamfer All Edges .04 X 45°
Ø .386 THRU (Letter W Drill) 4 PLCS. Ø .43 X 45° Debur BackSide
R.15 4 PLCS.
University of Oklahoma
Centralizing Bore Column
Aluminum 6061-T6
Keegan.J 7.17.17

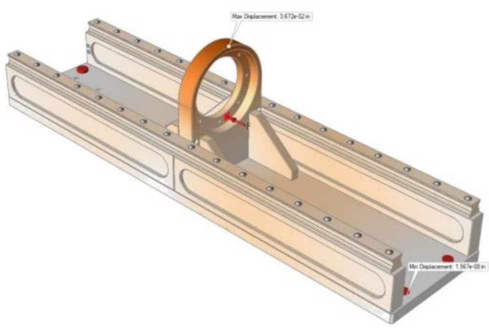

NOTES:
1. Chamfer All Edges Except Bore .04 X 45°
2. Chamfer 1.920 Bore .06 X 45°
4X Ø .323 THRU ALL (Drill Letter P) Ø .40 X 45° BackSide
SECTION J-J SCALE 1 : 1.5
Hold For All Thru Holes
University of Oklahoma
Centralizing Brace Clamp Top
Aluminum 6061-T6
Keegan.J 7.17.17

Setup

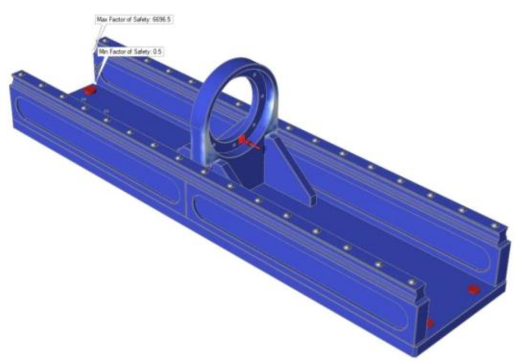




Forward Results

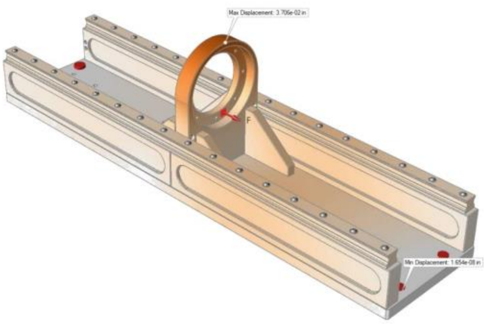


MAX Deflection: 3.672e-02in

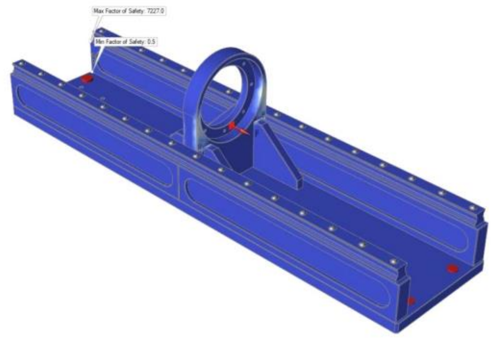


MIN Safety Factor: 0.5
Negligible result in small region near edge
Nothing below 2.0

Backward Results



MAX Deflection: 3.706e-02in



MIN Safety Factor: 0.5
Negligible result in small region near edge
Nothing below 2.0

Component Cost Overview Intensifier Assy.

Lead Time:	Pressure Intensifier:	Quantity:	Approximate Cost: Each:	Volume:
5 Weeks	Ball Screw Jack	1	3800.00	3800
In Stock	AKD-Analog Servo Driver 6AMP	1	1300.00	1300
In Stock	AKM 54H Servo	1	1750.00	1750
In Stock	Limit Switch	2	38.00	76
2 Weeks	Pressure Vessel HP Housing 316 SS	1	1200.00	1200
2 Weeks	Pressure Vessel Piston Assy. 17-4 SS	1	380.00	380
2 Weeks	Cradle/Mounting Assy.	1	800.00	800
4 Weeks	Intensifier Seal Kit	5	210.00	1050
In Stock	Pressure Relief Solenoid	2	350.00	700
		Total Cost: (Pressure Intensifier)		$11,056.00

Schlumberger

The UNIVERSITY of OKLAHOMA
School of Aerospace and Mechanical Engineering

Component Cost Overview Reciprocating Assy.

Lead Time:	Transducer/Electrical:	Quantity:	Approximate Cost: Each:	Volume:
1 Week	Transducer Power Supply	2	210.00	420.00
8 Weeks	0-30,000PSIG Pressure Transducer	1	1025.00	1025.00
In Stock	0-200LB Load Cell	1	635.00	635.00
In Stock	Thermocouple	3	33.00	99.00
In Stock	VDC Relay	5	33.00	165.00
In Stock	AKD-Basic Servo Driver 6AMP	1	1255.00	1255.00
In Stock	AKM 44H Servo	1	1260.00	1260.00
In Stock	Limit Switch	2	38.00	76.00
		Total Cost: (Transducer/Electrical)		$4,935.00

Lead Time:	Mechanical:	Quantity:	Approximate Cost: Each:	Volume:
In Stock	Linear Guide Rail	2	421.00	842.00
In Stock	Linear Guide Carriage	4	145.00	580.00
In Stock	40mm x 10mm Ball Screw	1	480.00	480.00
In Stock	40mm x 10mm Ball Nut	1	206.00	206.00
	Material Aproximate		800.00	
		Total Cost: (Mechanical)		$2,908.00

Schlumberger

The UNIVERSITY of OKLAHOMA
School of Aerospace and Mechanical Engineering

Component Cost Overview Data Acquisition

Lead Time:	Data Acquisition (Daq):		Approximate Cost:	
		Quantity:	Each:	Volume:
In Stock	"USB 6331"	1	1976.00	1976.00
In Stock	iDRN-TC Thermocouple C_Filter	3	280.00	840.00
In Stock	iDRN-ST Strain Guage C_Filter	1	305.00	305.00
In Stock	iDRX-PR Process C_Filter	2	290.00	580.00
In Stock	iRDN-PSPower Supply C_Filter	1	158.00	158.00
		Total Cost: (Daq)		$3,859.00

Schlumberger

The UNIVERSITY of OKLAHOMA
School of Aerospace and Mechanical Engineering

Seal it right. Right from Apple.℠
APPLE RUBBER PRODUCTS INC.

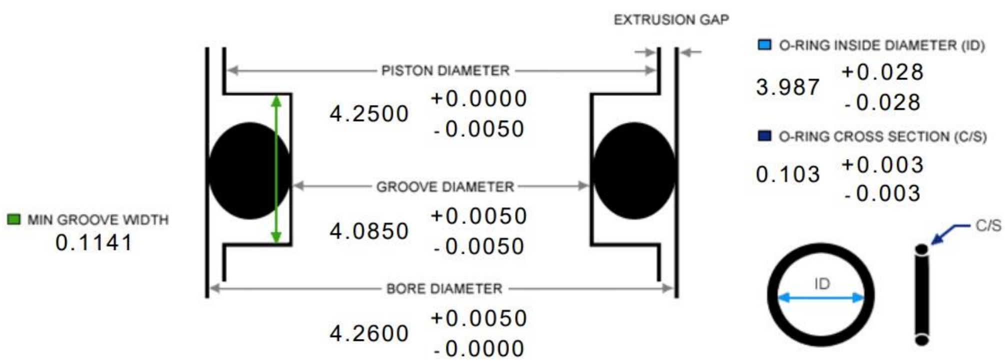
O-Ring Gland Calculation

Static Concentric Piston Seal

Ref: ARP67417

Input Values

	Nominal	+ Tolerance	- Tolerance
Bore Diameter	4.2600	0.0050	0.0000
Piston Diameter	4.2500	0.0000	0.0050
Groove Diameter	4.0850	0.0050	0.0050
O-Ring ID	3.987	0.028	0.028
O-Ring CS	0.103	0.003	0.003
Volume Swell	90%		



Results*

* Value out of recommended range.

	Concentric Condition		
	Minimum	Nominal	Maximum
% Compression	7.5*	15.05	19.81
Extrusion Gap	0.005	0.005	0.01
% Stretch	1.62	2.46	3.31
Squeeze	0.0075	0.0155	0.021

*Note: The advice given is believed to be accurate and reliable, however, it is the responsibility of the user to determine its safety and suitability. The proposed design should be thoroughly tested for each specific application.

Apple Rubber Products
1-800-828-7745
716-684-6560
applerubber.com

DRILL SIZE DECIMAL EQUIVALENT & TAP DRILL CHART

DECIMAL EQUIVALENT CHART

Drill Size	MM	Decimal Equivalent	Drill Size	MM	Decimal Equivalent	Drill Size	MM	Decimal Equivalent	Drill Size	MM	Decimal Equivalent	Drill Size	MM	Decimal Equivalent
–	0.10	.0039	53	1.51	.0595	–	4.00	.1575	J	7.04	.2770	–	14.00	.5512
–	0.20	.0079	1/16	1.59	.0625	21	4.04	.1590	K	7.14	.2810	9/16	14.29	.5625
–	0.25	.0098	52	1.61	.0635	20	4.09	.1610	9/32	7.14	.2812	37/64	14.68	.5781
–	0.30	.0118	51	1.70	.0670	19	4.22	.1660	L	7.37	.2900	–	15.00	.5906
80	0.34	.0135	50	1.78	.0700	18	4.31	.1695	M	7.49	.2950	19/32	15.08	.5938
79	0.37	.0145	49	1.85	.0730	11/64	4.37	.1719	19/64	7.54	.2969	39/64	15.48	.6094
1/64	0.40	.0156	48	1.93	.0760	17	4.39	.1730	N	7.67	.3020	5/8	15.88	.6250
78	0.41	.0160	5/64	1.98	.0781	16	4.50	.1770	5/16	7.94	.3125	–	16.00	.6299
77	0.46	.0180	47	1.99	.0785	15	4.57	.1800	–	8.00	.3150	41/64	16.27	.6406
–	0.50	.0197	–	2.00	.0787	14	4.62	.1820	O	8.03	.3160	21/32	16.67	.6562
76	0.51	.0200	46	2.06	.0810	13	4.70	.1850	P	8.20	.3230	–	17.00	.6693
75	0.53	.0210	45	2.08	.0820	3/16	4.76	.1875	21/64	8.33	.3281	43/64	17.07	.6719
74	0.57	.0225	44	2.18	.0860	12	4.80	.1890	Q	8.43	.3320	11/16	17.46	.6875
–	0.60	.0236	43	2.26	.0890	11	4.85	.1910	R	8.61	.3390	45/64	17.86	.7031
73	0.61	.0240	42	2.37	.0935	10	4.91	.1935	11/32	8.73	.3438	–	18.00	.7087
72	0.64	.0250	3/32	2.38	.0938	9	4.98	.1960	S	8.84	.3480	23/32	18.26	.7188
71	0.66	.0260	41	2.44	.0960	–	5.00	.1968	–	9.00	.3543	47/64	18.65	.7344
–	0.70	.0276	40	2.50	.0980	8	5.05	.1990	T	9.09	.3580	–	19.00	.7480
70	0.71	.0280	39	2.53	.0995	7	5.11	.2010	23/64	9.13	.3594	3/4	19.05	.7500
69	0.74	.0292	38	2.58	.1015	13/64	5.16	.2031	U	9.35	.3680	49/64	19.45	.7656
–	0.75	.0295	37	2.64	.1040	6	5.18	.2040	3/8	9.53	.3750	25/32	19.84	.7812
68	0.79	.0310	36	2.71	.1065	5	5.22	.2055	V	9.56	.3770	–	20.00	.7874
1/32	0.79	.0313	7/64	2.78	.1094	4	5.31	.2090	W	9.80	.3860	51/64	20.24	.7969
–	0.80	.0315	35	2.79	.1100	3	5.41	.2130	25/64	9.92	.3906	13/16	20.64	.8125
67	0.81	.0320	34	2.82	.1110	7/32	5.56	.2188	–	10.00	.3937	–	21.00	.8268
66	0.84	.0330	33	2.87	.1130	2	5.61	.2210	X	10.08	.3970	53/64	21.03	.8281
65	0.89	.0350	32	2.95	.1160	1	5.79	.2280	Y	10.26	.4040	27/32	21.43	.8438
–	0.90	.0354	–	3.00	.1181	A	5.94	.2340	13/32	10.32	.4062	55/64	21.84	.8594
64	0.91	.0360	31	3.05	.1200	15/64	5.95	.2344	Z	10.49	.4130	–	22.00	.8661
63	0.94	.0370	1/8	3.18	.1250	–	6.00	.2362	27/64	10.72	.4219	7/8	22.23	.8750
62	0.97	.0380	30	3.26	.1285	B	6.05	.2380	–	11.00	.4331	57/64	22.62	.8906
61	0.99	.0390	29	3.45	.1360	C	6.15	.2420	7/16	11.11	.4375	–	23.00	.9055
–	1.00	.0394	28	3.57	.1405	D	6.25	.2460	29/64	11.51	.4531	29/32	23.02	.9062
60	1.02	.0400	9/64	3.57	.1406	1/4	6.35	.2500	15/32	11.91	.4688	59/64	23.42	.9219
59	1.04	.0410	27	3.66	.1440	E	6.35	.2500	–	12.00	.4724	15/16	23.81	.9375
58	1.07	.0420	26	3.73	.1470	F	6.53	.2570	31/64	12.30	.4844	–	24.00	.9449
57	1.09	.0430	25	3.80	.1495	G	6.63	.2610	1/2	12.70	.5000	61/64	24.21	.9531
56	1.18	.0465	24	3.86	.1520	17/64	6.75	.2656	-	13.00	.5118	31/32	24.61	.9688
3/64	1.19	.0469	23	3.91	.1540	H	6.76	.2660	33/64	13.10	.5156	–	25.00	.9842
55	1.32	.0520	5/32	3.97	.1562	I	6.91	.2720	17/32	13.49	.5312	63/64	25.00	.9844
54	1.40	.0550	22	3.99	.1570	–	7.00	.2756	35/64	13.89	.5469	1"	25.40	1.0000

TAP DRILL CHART

Tap Size	Drill Size	Probable % Thread	Tap Size	Drill Size	Probable % Thread	Tap Size	Drill Size	Probable % Thread
0 – 80	3/64	71 – 81	10 – 32	21	68 – 76	5/8 – 18	37/64	58 – 65
M1.6 x .35	1.25 mm	67 – 77	M5 x .8	4.2 mm	69 – 77	M16 x 2	35/64	76 – 81
1 – 64	53	59 – 67	12 – 24	17	66 – 72	3/4 – 10	21/32	68 – 72
M2 x .4	1/16	72 – 79	12 – 28	15	70 – 78	3/4 – 16	11/16	71 – 77
1 – 72	53	67 – 75	M6 x 1	10	76 – 84	M20 x 2.5	11/16	74 – 78
2 – 56	51	62 – 69	1/4 – 20	7	70 – 75	7/8 – 9	49/64	72 – 76
2 – 64	50	70 – 79	1/4 – 28	3	72 – 80	7/8 – 14	13/16	62 – 67
M2.5 x .45	2.05 mm	69 – 77	5/16 – 18	F	72 – 77	M24 x 3	53/64	72 – 76
3 – 48	5/64	70 – 77	5/16 – 24	I	67 – 75	1 – 8	7/8	73 – 77
3 – 56	46	69 – 78	M8 x 1.25	6.7 mm	74 – 80	1 – 12	59/64	67 – 72
4 – 40	44	65 – 71	3/8 – 16	5/16	72 – 77	1 – 14	15/16	61 – 67
4 – 48	42	61 – 68	3/8 – 24	Q	71 – 79	1-1/8 – 7	63/64	72 – 76
M3 x .5	40	70 – 79	M10 x 1.5	8.4 mm	76 – 82	1/18 – 12	1-3/64	66 – 72
5 – 40	39	65 – 72	7/16 – 14	U	70 – 75	M30 x 3.5	1-3/64	75
5 – 44	38	63 – 71	7/16 – 20	25/64	65 – 72	1-1/4 – 7	1-7/64	76
M3.5 x .6	33	72 – 81	M12 x 1.75	13/32	69 – 74	1-1/4 – 12	1-11/64	72
6 – 32	36	71 – 78	1/2 – 13	27/64	73 – 78	1-3/8 – 6	1-7/32	72
6 – 40	33	69 – 77	1/2 – 20	29/64	65 – 72	1-3/8 – 12	1-19/64	72
M4 x .7	3.25 mm	74 – 82	M14 x 2	15/32	76 – 81	M36 x 4	1-1/4	82
8 – 32	29	62 – 69	9/16 – 12	31/64	68 – 72	1-1/2 – 6	1-11/32	72
8 – 36	29	70 – 78	9/16 – 18	33/64	58 – 65	1-1/2 – 12	1-27/64	72
10 – 24	25	69 – 75	5/8 – 11	17/32	75 – 79			

Metric Tap & Clearance Drill Sizes		Tap Drill				Clearance Drill			
		75% Thread for Aluminum, Brass, & Plastics		50% Thread for Steel, Stainless, & Iron		Close Fit		Standard Fit	
Screw Size (mm)	Thread Pitch (mm)	Drill Size (mm)	Closest American Drill	Drill Size (mm)	Closest American Drill	Drill Size (mm)	Closest American Drill	Drill Size (mm)	Closest American Drill
M1.5	0.35	1.15	56	1.25	55	1.60	1/16	1.65	52
M1.6	0.35	1.25	55	1.35	54	1.70	51	1.75	50
M 1.8	0.35	1.45	53	1.55	1/16	1.90	49	2.00	5/64
M 2	0.45	1.55	1/16	1.70	51	2.10	45	2.20	44
	0.40	1.60	52	1.75	50				
M 2.2	0.45	1.75	50	1.90	48	2.30	3/32	2.40	41
M 2.5	0.45	2.05	46	2.20	44	2.65	37	2.75	7/64
M 3	0.60	2.40	41	2.60	37	3.15	1/8	3.30	30
	0.50	2.50	39	2.70	36				
M 3.5	0.60	2.90	32	3.10	31	3.70	27	3.85	24
M 4	0.75	3.25	30	3.50	28	4.20	19	4.40	17
	0.70	3.30	30	3.50	28				
M 4.5	0.75	3.75	25	4.00	22	4.75	13	5.00	9
M 5	1.00	4.00	21	4.40	11/64	5.25	5	5.50	7/32
	0.90	4.10	20	4.40	17				
	0.80	4.20	19	4.50	16				
M 5.5	0.90	4.60	14	4.90	10	5.80	1	6.10	B
M 6	1.00	5.00	8	5.40	4	6.30	E	6.60	G
	0.75	5.25	4	5.50	7/32				
M 7	1.00	6.00	B	6.40	E	7.40	L	7.70	N
	0.75	6.25	D	6.50	F				
M 8	1.25	6.80	H	7.20	J	8.40	Q	8.80	S
	1.00	7.00	J	7.40	L				
M 9	1.25	7.80	N	8.20	P	9.50	3/8	9.90	25/64
	1.00	8.00	O	8.40	21/64				
M 10	1.50	8.50	R	9.00	T	10.50	Z	11.00	7/16
	1.25	8.80	11/32	9.20	23/64				
	1.00	9.00	T	9.40	U				
M 11	1.50	9.50	3/8	10.00	X	11.60	29/64	12.10	15/32
M 12	1.75	10.30	13/32	10.90	27/64	12.60	1/2	13.20	33/64
	1.50	10.50	Z	11.00	7/16				
	1.25	10.80	27/64	11.20	7/16				
M 14	2.00	12.10	15/32	12.70	1/2	14.75	37/64	15.50	39/64
	1.50	12.50	1/2	13.00	33/64				
	1.25	12.80	1/2	13.20	33/64				
M 15	1.50	13.50	17/32	14.00	35/64	15.75	5/8	16.50	21/32
M 16	2.00	14.00	35/64	14.75	37/64	16.75	21/32	17.50	11/16
	1.50	14.50	37/64	15.00	19/32				
M 17	1.50	15.50	39/64	16.00	5/8	18.00	45/64	18.50	47/64
M 18	2.50	15.50	39/64	16.50	41/64	19.00	3/4	20.00	25/32
	2.00	16.00	5/8	16.75	21/32				
	1.50	16.50	21/32	17.00	43/64				
M 19	2.50	16.50	21/32	17.50	11/16	20.00	25/32	21.00	53/64
M 20	2.50	17.50	11/16	18.50	23/32	21.00	53/64	22.00	55/64
	2.00	18.00	45/64	18.50	47/64				
	1.50	18.50	47/64	19.00	3/4				

LittleMachineShop.com

the premier source of parts and accessories for mini lathes and mini mills
LittleMachineShop.com • (800) 981-9663

Tap & Clearance Drill Sizes

Screw Size	Major Diameter	Threads Per Inch	Minor Diameter	Tap Drill: 75% Thread for Aluminum, Brass, & Plastics — Drill Size	Dec. Eq.	Tap Drill: 50% Thread for Steel, Stainless, & Iron — Drill Size	Dec. Eq.	Clearance Drill: Close Fit — Drill Size	Dec. Eq.	Clearance Drill: Free Fit — Drill Size	Dec. Eq.
0	.0600	80	.0447	3/64	.0469	55	.0520	52	.0635	50	.0700
1	.0730	64	.0538	53	.0595	1/16	.0625	48	.0760	46	.0810
		72	.0560	53	.0595	52	.0635				
2	.0860	56	.0641	50	.0700	49	.0730	43	.0890	41	.0960
		64	.0668	50	.0700	48	.0760				
3	.0990	48	.0734	47	.0785	44	.0860	37	.1040	35	.1100
		56	.0771	45	.0820	43	.0890				
4	.1120	40	.0813	43	.0890	41	.0960	32	.1160	30	.1285
		48	.0864	42	.0935	40	.0980				
5	.125	40	.0943	38	.1015	7/64	.1094	30	.1285	29	.1360
		44	.0971	37	.1040	35	.1100				
6	.138	32	.0997	36	.1065	32	.1160	27	.1440	25	.1495
		40	.1073	33	.1130	31	.1200				
8	.1640	32	.1257	29	.1360	27	.1440	18	.1695	16	.1770
		36	.1299	29	.1360	26	.1470				
10	.1900	24	.1389	25	.1495	20	.1610	9	.1960	7	.2010
		32	.1517	21	.1590	18	.1695				
12	.2160	24	.1649	16	.1770	12	.1890	2	.2210	1	.2280
		28	.1722	14	.1820	10	.1935				
		32	.1777	13	.1850	9	.1960				
1/4	.2500	20	.1887	7	.2010	7/32	.2188	F	.2570	H	.2660
		28	.2062	3	.2130	1	.2280				
		32	.2117	7/32	.2188	1	.2280				
5/16	.3125	18	.2443	F	.2570	J	.2770	P	.3230	Q	.3320
		24	.2614	I	.2720	9/32	.2812				
		32	.2742	9/32	.2812	L	.2900				
3/8	.3750	16	.2983	5/16	.3125	Q	.3320	W	.3860	X	.3970
		24	.3239	Q	.3320	S	.3480				
		32	.3367	11/32	.3438	T	.3580				
7/16	.4375	14	.3499	U	.3680	25/64	.3906	29/64	.4531	15/32	.4687
		20	.3762	25/64	.3906	13/32	.4062				
		28	.3937	Y	.4040	Z	.4130				
1/2	.5000	13	.4056	27/64	.4219	29/64	.4531	33/64	.5156	17/32	.5312
		20	.4387	29/64	.4531	15/32	.4688				
		28	.4562	15/32	.4688	15/32	.4688				
9/16	.5625	12	.4603	31/64	.4844	33/64	.5156	37/64	.5781	19/32	.5938
		18	.4943	33/64	.5156	17/32	.5312				
		24	.5114	33/64	.5156	17/32	.5312				
5/8	.6250	11	.5135	17/32	.5312	9/16	.5625	41/64	.6406	21/32	.6562
		18	.5568	37/64	.5781	19/32	.5938				
		24	.5739	37/64	.5781	19/32	.5938				
11/16	.6875	24	.6364	41/64	.6406	21/32	.6562	45/64	.7031	23/32	.7188
3/4	.7500	10	.6273	21/32	.6562	11/16	.6875	49/64	.7656	25/32	.7812
		16	.6733	11/16	.6875	45/64	.7031				
		20	.6887	45/64	.7031	23/32	.7188				
13/16	.8125	20	.7512	49/64	.7656	25/32	.7812	53/64	.8281	27/32	.8438
7/8	.8750	9	.7387	49/64	.7656	51/64	.7969	57/64	.8906	29/32	.9062
		14	.7874	13/16	.8125	53/64	.8281				
		20	.8137	53/64	.8281	27/32	.8438				
15/16	.9375	20	.8762	57/64	.8906	29/32	.9062	61/64	.9531	31/32	.9688
1	1.000	8	.8466	7/8	.8750	59/64	.9219	1-1/64	1.0156	1-1/32	1.0313
		12	.8978	15/16	.9375	61/64	.9531				
		20	.9387	61/64	.9531	31/32	.9688				

D

Technical Data

Suggested Tightening Torque1 Values To Produce Corresponding Bolt Clamping Loads

Size	Bolt Dia. D (in.)	Tensile Stress Area A (sq. in.)	SAE Grade 2 Bolts: Tensile Strength (min psi)	SAE Grade 2: Proof Load (psi)	SAE Grade 2: Clamp[2] Load P (lb.)	SAE Grade 2: Tightening Torque Dry K=0.20	SAE Grade 2: Tightening Torque Lub. K=0.15	SAE Grade 5 Bolts: Tensile Strength (min psi)	SAE Grade 5: Proof Load (psi)	SAE Grade 5: Clamp[2] Load P (lb.)	SAE Grade 5: Tightening Torque Dry K=0.20	SAE Grade 5: Tightening Torque Lub. K=0.15	SAE Grade 7[3]: Clamp[2] Load P (lb.)	SAE Grade 7: Tightening Torque Dry K=0.20	SAE Grade 7: Tightening Torque Lub. K=0.15	SAE Grade 8[4]: Clamp[2] Load P (lb.)	SAE Grade 8: Tightening Torque Dry K=0.20	SAE Grade 8: Tightening Torque Lub. K=0.15
						lb. in.	lb. in.				lb. in.	lb. in.		lb. in.	lb. in.		lb. in.	lb. in.
4-40	0.1120	0.00604	74,000	55,000	240	5	4	120,000	85,000	380	8	6	480	11	8	540	12	9
4-48	0.1120	0.00661			280	6	5			420	9	7	520	12	9	600	13	10
6-32	0.1380	0.00909			380	10	8			580	16	12	720	20	15	820	23	17
6-40	0.1380	0.01015			420	12	9			640	18	13	800	22	17	920	25	19
8-32	0.1640	0.01400			580	19	14			900	30	22	1100	36	27	1260	41	31
8-36	0.1640	0.01474			600	20	15			940	31	23	1160	38	29	1320	43	32
10-24	0.1900	0.01750			720	27	21			1120	43	32	1380	52	39	1580	60	45
10-32	0.1900	0.02000			820	31	23			1285	49	36	1580	60	45	1800	68	51
1/4-20	0.2500	0.0318			1320	66	49			2020	96	75	2500	120	96	2860	144	108
1/4-28	0.2500	0.0364			1500	76	56			2320	120	86	2860	144	108	3280	168	120
						lb. ft.	lb. ft.				lb. ft.	lb. ft.		lb. ft.	lb. ft.		lb. ft.	lb. ft.
5/16-18	0.3125	0.0524			2160	11	8			3340	17	13	4120	21	16	4720	25	18
5/16-24	0.3125	0.0580			2400	12	9			3700	19	14	4560	24	18	5220	25	20
3/8-16	0.3750	0.0775			3200	20	15			4940	30	23	6100	40	30	7000	45	35
3/8-24	0.3750	0.0878			3620	23	17			5600	35	25	6900	45	45	7900	50	35
7/16-14	0.4375	0.1063			4380	30	24			6800	50	35	8400	60	45	9550	70	55
7/16-20	0.4375	0.1187			4900	35	25			7550	55	40	9350	70	50	10700	80	60
1/2-13	0.5000	0.1419			5840	50	35			9050	75	55	11200	95	70	12750	110	80
1/2-20	0.5000	0.1599			6600	55	40			10700	90	65	12600	100	80	14400	120	90
9/16-12	0.5625	0.1820			7500	70	55			11600	110	80	14350	135	100	16400	150	110
9/16-18	0.5625	0.2030			8400	80	60			12950	120	90	16000	150	110	18250	170	130
5/8-11	0.6250	0.2260			9300	100	75			14400	150	110	17800	190	140	20350	220	170
5/8-18	0.6250	0.2560			10600	110	85			16300	170	130	20150	210	160	23000	240	180
3/4-10	0.7500	0.3340			13800	175	130			21300	260	200	26300	320	240	30100	380	280
3/4-16	0.7500	0.3730			15400	195	145			23800	300	220	29400	360	280	33600	420	320
7/8-9	0.8750	0.4620	60,000	33,000	11400	165	125			29400	430	320	36400	520	400	41600	600	460
7/8-14	0.8750	0.5090			12600	185	140			32400	470	350	40100	580	440	45800	660	500
1-8	1.0000	0.6060			15000	250	190			38600	640	480	47700	800	600	54500	900	680
1-12	1.0000	0.6630			16400	270	200			42200	700	530	52200	860	660	59700	1000	740
1-1/8-7	1.1250	0.7630			18900	350	270	105,000	74,000	42300	800	600	60100	1120	840	68700	1280	960
1-1/8-12	1.1250	0.8560			21200	400	300			47500	880	660	67400	1260	940	77000	1440	1080
1-1/4-7	1.2500	0.9690			24000	500	380			53800	1120	840	76300	1580	1100	87200	1820	1360
1-1/4-12	1.2500	1.0730			26600	550	420			59600	1240	920	84500	1760	1320	96600	2000	1500
1-3/8-6	1.3750	1.1550			28600	660	490			64100	1460	1100	91000	2080	1560	104000	2380	1780
1-3/8-12	1.3750	1.3150			32500	740	560			73000	1680	1260	104000	2380	1780	118400	2720	2040
1-1/2-6	1.5000	1.4050			34800	870	650			78000	1940	1460	111000	2780	2080	126500	3160	2360
1-1/2-12	1.5000	1.5800			39100	980	730			87700	2200	1640	124005	3100	2320	142200	3560	2660

Notes:

1. Tightening torque values are calculated from the formula $T = KDP$, where T = tightening torque, lb-in.; K = torque-friction coefficient; D = nominal bolt diameter, in.; and P = bolt clamping load developed by tightening. lb.

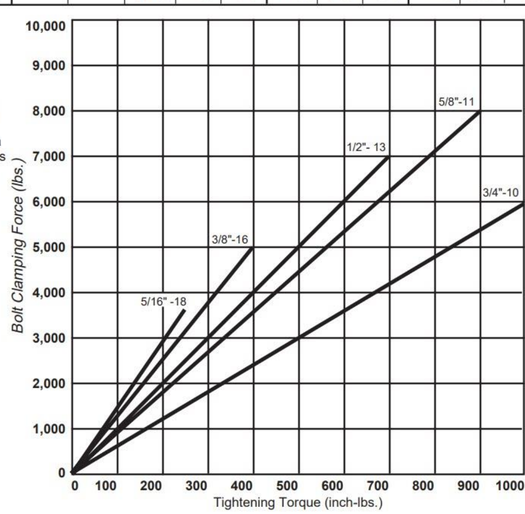
2. Clamp load is also known as preload or initial load in tension on bolt. Clamp load (lb.) is calculated by arbitrarily assuming usable bolt strength is 75% of bolt proof load (psi) times tensile stress area (sq. in.) of threaded section of each bolt size. Higher or lower values of clamp load can be used depending on the application requirements and the judgement of the designer.

3. Tensile strength (min psi) of all Grade 7 bolts is 133,000. Proof load is 105,000 psi.

4. Tensile strength (min psi) of all Grade 8 bolts is 150,000 psi. Proof load is 120,000 psi.

Ref.: Fastening Reference. Machine Design. Nov. 1977.

Bolt Clamping Force vs. Tightening Torque for Unlubricated Steel Bolts.



CATALOG 13

SPAENAUR

AKM44H Servomotor Controller Code (Reciprocating Rod Test Bed)

```
REM Uploaded from AKM44H_JK on 5/18/2021 9:25:24 PM
REM #line 0 "C:\ SLB - High Pressure -
General\Kollmorgen\Manual_Operation_v2_Keegan_4.14.21.bas"
'-------------- Device Params -----------------------
Params
	DRV.OPMODE = 2'position operation mode
	DRV.CMDSOURCE = 5'command source = AKD BASIC TG
	UNIT.PROTARY = 4'16 bit position units, 65536 counts/rev
	UNIT.VROTARY = 0'velocity units = rpm
	UNIT.ACCROTARY = 0'acceleration units = rpm/sec
End Params
'-------------- Define (dim) Global Variables --------
Alias ONE_REV = 65536
Dim LOOP, COUNT, RunSpeed, ReverseSpeed as Integer
'-------------- Main Program -------------------------
Main
	' Setting up move parameter
	Move.Acc = 10000
	Move.Dec = 10000
	Move.RunSpeed = 100
	DRV.EN
	drv.swenable = 1
	RunSpeed = 100
	ReverseSpeed = 50
	While (1)
		'DI-#1 = Handheld Toggle Enable
		'DI-#2 = Handheld Toggle OUT Button
		'DI-#3 = Handheld Toggle IN Button
		'DI-#4 = Labview Trigger Start Recip Motion Relay1
		'DI-#6 = Labview Trigger CW Move Relay2 (Not Used)
		'DI-#21 = Limit Switch Soft @ Side1=Closest To Servo
		'DI-#22 = Limit Switch Hard @ Side1=Closest To Servo
		'DI-#23 = Limit Switch Soft @ Side2=Furthest From Servo
		'DI-#24 = Limit Switch Hard @ Side2=Furthest From Servo
		'Move shaft INTO DUT "Manual IN Button"
		If (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE = 0) And
(DIN3.STATE = 1) And (DIN21.STATE = 0) And (DIN22.STATE = 0) And (DIN23.STATE =
0) And (DIN24.STATE = 0) Then
			Call S2_RotateIN

		'Move shaft OUT of DUT "Manual OUT Button"
		Elseif (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE = 1) And
(DIN3.STATE = 0) And (DIN21.STATE = 0) And (DIN22.STATE = 0) And (DIN23.STATE =
0) And (DIN24.STATE = 0) Then
			Call S3_RotateOUT
		'Shaft IN overtravel correction "Soft Limit Switch Trigger"
```

```
		Elseif (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE = 0) And
(DIN3.STATE = 1) And (DIN21.STATE = 0) And (DIN22.STATE = 0) And (DIN23.STATE =
1) And (DIN24.STATE = 0) Then
			Call S4_INcorrection
		'Shaft OUT overtravel correction "Soft Limit Switch Trigger"
		Elseif (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE = 1) And
(DIN3.STATE = 0) And (DIN21.STATE = 1) And (DIN22.STATE = 0) And (DIN23.STATE =
0) And (DIN24.STATE = 0) Then
			Call S5_OUTcorrection
		'Reciprocating Program
		Elseif (DIN1.STATE = 0) And (DIN4.STATE = 1) And (DIN22.STATE = 0)
And (DIN24.STATE = 0) Then
			Call S6_Reciprocating
		'Shaft IN overtravel HARDSTOP "Hard Limit Switch Trigger"
		Elseif (DIN22.STATE = 0) And (DIN24.STATE = 1) Then
			Call S9_INHardStop
		'Shaft OUT overtravel HARDSTOP "Hard Limit Switch Trigger"
		Elseif (DIN22.STATE = 1) And (DIN24.STATE = 0) Then
			Call S10_OUTWHardStop
			'Stop Motor
		Else
			Call S1_Stopped
		End If
	Wend
End Main
'-------------- Subroutines and Functions ------------
Sub S1_Stopped
	'Stop Servo
	CS.DEC = 1000
	DRV.STOP
End Sub
Sub S2_RotateIN
	'CCW Rotation
	MOVE.DIR = 0
	MOVE.RUNSPEED = RunSpeed
	MOVE.GOVEL
End Sub
Sub S3_RotateOUT
	'CW Rotation
	MOVE.DIR = 1
	MOVE.RUNSPEED = RunSpeed
	MOVE.GOVEL
End Sub
Sub S4_INcorrection
	'Soft Limit Switch Trigger
	'CCW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 1
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (3)
```

```
End Sub
Sub S5_OUTcorrection
	'Soft Limit Switch Trigger
	'CW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 0
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (3)
End Sub
Sub S6_Reciprocating
	While (DIN4.STATE = 1) And (DIN1.STATE = 0) And (DIN22.STATE = 0) And
(DIN24.STATE = 0)
		MOVE.RUNSPEED = RunSpeed
		MOVE.GOVEL
		If (DIN21.STATE = 1) And (DIN23.STATE = 0) AND (DIN4.STATE = 1) Then
			Call S8_ReciprocateIN
		ElseIf (DIN21.STATE = 0) And (DIN23.STATE = 1) AND (DIN4.STATE = 1)
Then
			Call S7_ReciprocateOUT
		End If
	Wend
End Sub
Sub S7_ReciprocateOUT
	While (DIN4.STATE = 1) And (DIN1.STATE = 0) And (DIN21.STATE = 0) And
(DIN22.STATE = 0) And (DIN24.STATE = 0)

		MOVE.DIR = 1
		MOVE.RUNSPEED = RunSpeed
		MOVE.GOVEL
	Wend
End Sub
Sub S8_ReciprocateIN
	While (DIN4.STATE = 1) And (DIN1.STATE = 0) And (DIN23.STATE = 0) And
(DIN22.STATE = 0) And (DIN24.STATE = 0)

		MOVE.DIR = 0
		MOVE.RUNSPEED = RunSpeed
		MOVE.GOVEL
	Wend
End Sub
Sub S9_INHardStop
	'CCW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 1
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (3)
	DRV.DIS
End Sub
Sub S10_OUTWHardStop
```

```
	'CCW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 0
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (3)
	DRV.DIS

End Sub
```

AKM54H Servomotor Controller Code (Pressure Intensifier)

```
REM Uploaded from AKM54H_JK on 5/18/2021 9:25:24 PM
REM #line 0 "C:\SLB - High Pressure -
General\Kollmorgen\Manual_Operation_v2_Keegan_4.14.21.bas"
'-------------- Device Params -----------------------
Params
	DRV.OPMODE = 2'position operation mode
	DRV.CMDSOURCE = 5'command source = AKD BASIC TG
	UNIT.PROTARY = 4'16 bit position units, 65536 counts/rev
	UNIT.VROTARY = 0'velocity units = rpm
	UNIT.ACCROTARY = 0'acceleration units = rpm/sec
End Params
'-------------- Define (dim) Global Variables --------
Alias ONE_REV = 65536
Dim RunSpeed, ReverseSpeed as Integer
'-------------- Main Program -------------------------
Main
	' Setting up move parameter
	Move.Acc = 10000
	Move.Dec = 10000
	Move.RunSpeed = 100
	DRV.EN
	drv.swenable = 1
	RunSpeed = 150
	ReverseSpeed = 200
	While (1)
		'DI-#1 = Handheld Toggle Enable
		'DI-#2 = Handheld Toggle OUT Button
		'DI-#3 = Handheld Toggle IN Button
		'DI-#4 = Labview Trigger Relay1
		'DI-#5 = Labview Trigger Relay2
		'DI-#6 = Labview Trigger Relay4
		'DI-#21 = Limit Switch Soft Ballscrew Advance "Moving INWARD and
Boosting Pressure"
		'DI-#22 = Limit Switch Soft Ballscrew Retract "Moving Outward and Relieving
Pressure"
		'DI-#23 = Limit Switch Hard Ballscrew Retract "Moving Outward and Relieving
Pressure"
```

```
                'DI-#24 = Limit Switch Hard Ballscrew Advance "Moving INWARD and
Boosting Pressure"
'------------------------------------------------------------------------Manually Triggered------------------
------------------------------------------------------------------------------'
                        'Move And Advance Ballscrew "Manual IN Button"
                If              (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN5.STATE
= 0) And (DIN2.STATE = 0) And (DIN3.STATE = 1) And (DIN21.STATE = 0) And
(DIN22.STATE = 0) And (DIN23.STATE = 0) And (DIN24.STATE = 0) Then
                        Call S2_RotateIN
                        'Move And Retract Ballscrew "Manual OUT Button"
                Elseif  (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN5.STATE = 0)
And (DIN2.STATE = 1) And (DIN3.STATE = 0) And (DIN21.STATE = 0) And (DIN22.STATE
= 0) And (DIN23.STATE = 0) And (DIN24.STATE = 0) Then
                        Call S3_RotateOUT
                        'Ballscrew Advance Correction "Soft Limit Switch Trigger"
                Elseif  (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN5.STATE = 0)
And (DIN2.STATE = 0) And (DIN3.STATE = 1) And (DIN21.STATE = 1) And (DIN22.STATE
= 0) And (DIN23.STATE = 0) And (DIN24.STATE = 0) Then
                        Call S4_INcorrection
                        'Ballscrew Retract Correction "Soft Limit Switch Trigger"
                Elseif  (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN5.STATE = 0)
And (DIN2.STATE = 1) And (DIN3.STATE = 0) And (DIN21.STATE = 0) And (DIN22.STATE
= 0) And (DIN23.STATE = 1) And (DIN24.STATE = 0) Then
                        Call S5_OUTcorrection
'------------------------------------------------------------------------LabView Triggered------------------
------------------------------------------------------------------------------'
                        'Move And Advance Ballscrew "DI-#4 LabView Signal"
                'If             (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE
= 1) And (DIN3.STATE = 0) And (DIN21.STATE = 0) And (DIN22.STATE = 0) And
(DIN23.STATE = 0) And (DIN24.STATE = 0) Then
                '       Call S2_CWROTATION
                        'Move And Retract Ballscrew "DI-#5 LabView Signal"
                'Elseif (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE = 0)
And (DIN3.STATE = 1) And (DIN21.STATE = 0) And (DIN22.STATE = 0) And
(DIN23.STATE = 0) And (DIN24.STATE = 0) Then
                '       Call S3_CCWROTATION
                        'Ballscrew Advance Correction "Soft Limit Switch Trigger" ---------------
--------PENDING ACTION----------------------------------
                'Elseif (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE = 0)
And (DIN3.STATE = 1) And (DIN21.STATE = 1) And (DIN22.STATE = 0) And
(DIN23.STATE = 0) And (DIN24.STATE = 0) Then
                '       Call S4_CWCORRECTION
                        'Ballscrew Retract Correction "Soft Limit Switch Trigger" -----------------
------PENDING ACTION----------------------------------
                'Elseif (DIN1.STATE = 1) And (DIN4.STATE = 0) And (DIN2.STATE = 1)
And (DIN3.STATE = 0) And (DIN21.STATE = 0) And (DIN22.STATE = 0) And
(DIN23.STATE = 1) And (DIN24.STATE = 0) Then
                '       Call S5_CCWCORRECTION
'------------------------------------------------------------------------Stoping Conditions-------------------
-----------------------------------------------------------------------------'
                        'Ballscrew Advance Hard Stop "Hard Limit Switch Trigger"
```

```
		Elseif (DIN21.STATE = 0) And (DIN24.STATE = 1) Then
			Call S9_INHardStop
			'Ballscrew Retract Hard Stop "Hard Limit Switch Trigger"
		Elseif (DIN22.STATE = 1) And (DIN23.STATE = 0) Then
			Call S10_OUTWHardStop
			'Stop Motor
		Else
			Call S1_Stopped
		End If
	Wend
End Main
'-------------- Subroutines and Functions ------------
'Move Direction Set=1 Advances Ballscrew
'Move Direction Set=0 Retracts Ballscrew
Sub S1_Stopped
	'Stop Servo
	CS.DEC = 1000
	DRV.STOP
End Sub
Sub S2_RotateIN
	'Advances Ballscrew
	'CW Rotation
	MOVE.DIR = 1
	MOVE.RUNSPEED = RunSpeed
	MOVE.GOVEL
End Sub
Sub S3_RotateOUT
	'Brings Ballscrew back
	'CCW Rotation
	MOVE.DIR = 0
	MOVE.RUNSPEED = RunSpeed
	MOVE.GOVEL
End Sub
Sub S4_INcorrection
	'Soft Limit Switch Trigger
	'CCW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 0
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (3)
End Sub
Sub S5_OUTcorrection
	'Soft Limit Switch Trigger
	'CW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 1
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (3)
End Sub
```

```
Sub S9_INHardStop
	'Disables Software after hard limit swtich press
	'CCW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 0
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (5)
	DRV.DIS
End Sub
Sub S10_OUTWHardStop

	'Disables Software after hard limit swtich press
	'CW Rotation for 3 Seconds
	Move.Acc = 10000
	MOVE.DIR = 1
	MOVE.RUNSPEED = ReverseSpeed
	MOVE.GOVEL
	Pause (5)
	DRV.DIS
End Sub
```