Design of a 600HP Diesel Power Gearbox Test Stand

Final Design Report

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Executive Summary

This report details the design of a gearbox test stand dynamometer for Elmer's Manufacturing, a Manitoba-based producer of industrial agricultural equipment. The purpose of this project is to improve Elmer's testing procedure for its Haulmaster grain cart gearboxes. The gearbox test stand will replace the tractor and agricultural equipment containing the test gearbox which are part of the current testing equipment.

The design features a 600 HP Deutz diesel engine as its power source. The power from the diesel engine is transferred through a reduction gearbox, which lowers the rate of rotation to a regular PTO speed of roughly 1000 RPM, into the PTO shaft leading to the test gearbox. A clutch located between the reduction gearbox and the diesel engine allows unloaded idling of the engine for cooldown.

A hydraulic pump dynamometer system applies an independent, variable load to each test gearbox output. The hydraulic pump dynamometer consists of two hydraulic pumps operating at up to 5000 psi at 206 gpm, electronic load control valves, oil filters, oil cooler, and a 680-gallon oil reservoir. Power is transferred from the test gearbox to the hydraulic pumps via a PTO shaft and an increaser gearbox. The increaser gearbox raises the shaft speed to 1500 rpm which is the operational speed of the hydraulic pumps. The angles of both the hydraulic pump dynamometer and the test gearbox mount are adjustable to accommodate operational angles of the Haulmaster gearbox, ranging between 30 and 60 degrees.

An overview of all the systems, their components and their function are provided throughout the report. Where applicable, the justification for the component design is provided in the form of preliminary FEA studies. Over the course of the project the design team has provided all the project deliverables, including CAD models of the final design, preliminary engineering drawings for parts fabricated in-house, overall assembly drawings, a detailed bill of materials, a list of purchased parts for the final design, and a cost estimate for the final design. The proposed final design achieves all of the initially proposed metrics with the exception of a target cost of $200,000 CAD as the final design costs approximately $270,000 CAD.

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Glossary

Gearbox Test Stand	For the purposes of this report, the entire designed system including the power unit and the gearbox test unit, is called the "Gearbox Test Stand".
Gearbox Test Unit	Within the gearbox test stand, the gearbox test unit contains the components required for holding Elmer's gearboxes and applying the load(s) to the gearbox outputs.
Inventor	Autodesk Inventor is the CAD software used to complete the design of the project.
Mini-Skid	The skid that contains the test gearbox mounting
Modules	Within the overall gearbox test stand design, the major design components are referred to as modules in the report.
Programmable Logic Controller	A programmable logic controller, referenced in the report as PLC, is a digital controller used in industrial applications to automate processes.
Power Takeoff	A power takeoff, referenced in the report as PTO, is used on agricultural equipment to transfer power from one source to another. Usually, a PTO shaft is used to connect the output of a tractor to the input of the implement.
Power Unit	Within the gearbox test stand, the power unit contains the components to create the power and provide the PTO output through a gearbox.

1 Introduction

This report documents the design of a gearbox test stand for Elmer’s Manufacturing. Elmer’s Manufacturing is a Manitoba based company that designs, develops, and manufactures a variety of agricultural equipment. A large portion of the agricultural equipment produced by Elmer’s Manufacturing is designed to be powered directly from a tractor powertrain through a power takeoff (PTO) shaft. The tractor’s PTO output shaft is connected to the agricultural equipment’s gearbox, which transmits the power to onboard equipment and accessories. Evaluation of gearbox designs is the driving force behind the gearbox test stand project.

The purpose of the gearbox test stand project is to provide a versatile test platform for the gearboxes and simplify the current test procedure. The first chapter of this report will outline the background of the project, including the current testing procedure at Elmer’s and the ways in which it can be improved by the implementation of this project. The first chapter also provides a detailed project purpose statement.

The second chapter of this report is a summary of the concept development and analysis process which the team used throughout the project. This chapter provides the concept generation and selection methodology, a list of the concepts which were considered for those components which are independent of other system components, and the summary of the concept of the final design.

The third chapter of the report is focused on the design of the gearbox test stand. The design and its function are detailed, as is the general design methodology. This chapter also provides an overview of the design process for each assembly in the final design and its individual components.

The final chapter contains a summary of the entire report along with an initial cost estimate, a list of achieved metrics, and recommendations for further development of the design presented by the project team. Project deliverables, including the assembly drawings, preliminary engineering drawings, bill of materials (BOM), and list of purchased parts, are omitted in the body of the report and can be found in Appendix A and Appendix B.

1.1 Project Background

The gearbox is a vital component of the agricultural equipment manufactured by Elmer’s, and prototypes are thoroughly tested under various loading conditions. Elmer’s current methodology for testing their prototype gearboxes requires a fully functioning tractor to act as the power source. To apply load to the gearboxes, the agricultural equipment is operated as it would be when in service. For example, Elmer’s Haulmaster grain cart is filled up with grain, connected to the drivetrain of a farm tractor and cycled for the duration of the test. Other equipment, such as the Wolverine ditcher, is used in a nearby field.

The tractors which are used to test the gearbox are borrowed from a nearby farm, which shares a common owner with Elmer’s Manufacturing. The farm also provides a land area for testing when required. Although Elmer’s farm and Elmer’s Manufacturing share the same owner, they are still separate companies that are pursuing different areas of the agriculture business.

The current testing methodology is particularly inefficient for two of Elmer’s products: the Haulmaster grain cart shown in Figure 1, and the Wolverine ditcher shown in Figure 2. These products are the focus of the gearbox test stand project. There is a total of three gearbox designs that must be accommodated by the project, one used in the Haulmaster and two used in the Wolverine. Haulmaster’s gearbox is shown in Figure 3. There are three standard 21 tooth splines used in the gearbox, two output splines which run the cross auger and the unload auger, and one input spline. The two Wolverine ditcher gearboxes are arranged similarly to the Haulmaster gearbox but have a different output spline orientation which accommodates the specific toolset of the ditcher.

Figure 1. Elmer's Haulmaster grain cart during operation. Adapted from [1].

Figure 2. Elmer's Wolverine ditcher during operation. Adapted from [1].

Figure 3. Elmer's Haulmaster grain cart gearbox.

1.2 Problem Statement

Elmer's Manufacturing has four main issues with the current gearbox testing methodology. Firstly, Elmer's testing procedures are taxing on the tractor which is providing power to the agricultural equipment. Damage that occurs during testing and the shortened service intervals associated with increased wear both result in extra costs and downtime of the farm's tractors.

Secondly, the selection of test tractors is limited to the equipment which the farm has purchased for agricultural operations. There is a risk that the farm will not always own the most advanced tractors and that Elmer's ability to test the compatibility of their equipment with the top-of-the-line tractors will be hindered.

Thirdly, since the tractors used for testing are borrowed from a farm, the testing must be scheduled for such a time when they can be spared. The scheduling requirement adds unnecessary constraints on the development timeline. Since the development of new equipment lasts several seasons, unforeseen delays in the development process or unpredictable weather which impacts the farming operation can cause scheduling conflicts.

Lastly, the need for a full-size tractor and associated agricultural equipment for testing greatly limits the potential testing locations and vastly increases the test time. When testing the Haulmaster grain cart gearbox, the cart must first be filled with grain. The grain must be sourced, and a container needs to be prepared for the Haulmaster to unload the grain. The testing location for the Haulmaster must accommodate the tractor, the grain cart, and the container, all of which combine into an area of several hundred square feet.

1.3 Project Scope

The scope of this project includes the design of a gearbox test stand which can adequately simulate the operating conditions of Elmer's gearboxes. More specifically, the scope includes the specification and, if applicable, preliminary design of the following:

1) Power unit with a power output comparable to that of the latest tractors.
2) Gearbox which reduces the power source output to PTO speeds
3) Gearbox mounting system which can accommodate the Haulmaster gearbox at a range of angles between 30 and 60 degrees
4) System which applies variable resistive loading independently to each gearbox output

Preliminary designs of components include fully dimensioned models of the parts and initial verification of the parts using either finite element analysis (FEA) or a combination of hand calculations and FEA. Certain components are excluded from the verification requirement including any skids used to house all of the assemblies and gears in any gearboxes designed over the course of the project.

The scope also includes the creation of a computer-aided design (CAD) model of the final design. Assembly drawings which include exploded views and a detailed list of materials are required for all assemblies in the final design. Preliminary drawings were prepared for components designed by the project team. Preliminary drawings do not adhere to any engineering standard and only contain intolerance dimensions. Lastly, a list of purchased parts and an overall cost estimate for the final design are required.

The scope does not include an exhaustive validation process for any components proposed by the design team. Fatigue, fracture and buckling analyses, as well as detailed FEA studies, were considered out of scope for this project due to time constraints. Lubrication analysis for the gearbox was also omitted from the project scope. While the aforementioned analyses are essential to perform prior to the implementation of the design, the project timeline was not sufficient to accommodate them.

There were two major scope changes that were made over the course of the project in order to avoid delays in the final deadline for the deliverables. The first scope change included eliminating the accommodation for the Wolverine ditcher gearbox. The client requested that the design team focussed efforts on the Haulmaster gearbox alone. The second scope change was to eliminate the FEA verification of the skids which all the components in the final design are mounted on. The initial scope of the project called for FEA verification to be performed on the skids as well as all the other components designed by the team. However, due to the number of skids and the time constraints of the design process, there were no resources available to complete this task. Both of the aforementioned changes were discussed with the client over the course of several meetings and were approved before implementation.

1.4 Technical Specifications

To ensure that the final design meets the client’s requirements, the design team developed target specifications for the project in close cooperation with the client. These specifications are used to evaluate the final design. The specifications consist of a comprehensive list of client needs, a list of constraints and limitations, and lastly a list of metrics.

1.4.1 Needs

After the initial meeting with the client, the design team developed a list of needs for the project based on the client's feedback. The needs of the design are grouped into five categories as shown below.

1. Physical Properties. This category contains all the needs which relate to the geometry and mass of the test stand assembly.
2. Power Unit Capabilities. This category contains all of the needs which relate to the output and operating requirements of the test stand's power source.
3. Gearbox Test Unit Capabilities. This category contains all of the needs which relate to the input and output specifications of the gearbox test unit, including the number of inputs and outputs and their physical orientation and adjustability.
4. Operation. This category contains all of the needs which relate to operation, usability, serviceability/maintenance, and safety of the gearbox test stand assembly.
5. Manufacturability. This category contains all of the needs which relate to the manufacturing processes and raw materials specified for the gearbox test stand assembly components.

Within these five categories, the importance of each need was determined by assigning a priority number system. The priority number system is a scale from one to five, where five denotes the highest level of importance and one represents the lowest importance level. The list of needs, along with their importance rankings, was submitted to and approved by Elmer's. TABLE I gives an overview of the project needs.

TABLE I: CLIENT NEEDS

Category	Need Number	Need	Priority Number
1. Physical Properties	1.1	The system has a reasonable weight	5
	1.2	The system is easily moveable	4
	1.3	The system has two separate units (gearbox test unit and power unit)	4
	1.4	The system is stable	3
	1.5	The system is of reasonable size	2
2. Power Unit Capabilities	2.1	The system has a power source with a high-power output	5
	2.2	The power unit has no external dependencies	5
3. Gearbox Test Unit Capabilities	3.1	The inputs to the test gearbox are at standard PTO speeds	5
	3.2	The loads applied to the test gearboxes can be varied	5
	3.3	Test gearboxes are mounted at an angle similar to their actual operation angle	5
	3.4	The gearbox test unit can accommodate different gearbox types	4
	3.5	The materials used are readily available	3

Category	Need Number	Need	Priority Number
4. Operation	4.1	The system is safe to operate	5
	4.2	The system is operable in a variety of conditions	4
	4.3	The system has a reasonable continuous run time	3
	4.4	The power unit and gearbox test unit are capable of being operated independently of each other.	3
	4.5	The system has reasonable longevity	3
	4.6	The system is simple to operate	2
	4.7	The units are easily serviceable	2
5. Manufacturability	5.1	The system has a reasonable cost	3
	5.2	The materials used are readily available	3

1.4.2 Constraints and Limitations

The constraints and limitations placed on the project were identified during client and team meetings and were organized into two main categories: design constraints and team constraints. Design constraints are the constraints placed upon the design of the gearbox test stand, such as maximum weight and size. The team constraints are factors that restrict how the design team operates, such as project scheduling and software usage. The design constraints and limitations are shown in TABLE II, while the team constraints and limitations are shown in TABLE III.

TABLE II: DESIGN CONSTRAINTS AND LIMITATIONS

#	Design Constraint/Limitation	Description and Implications
D.1	The design is limited to power sources capable of operating with no external dependencies	The testing unit must be operable in different locations at the client's facility, i.e. the unit will not have a permanent "home". Therefore, the source of power for the test unit must not require any external dependencies.
D.2	The power unit and gearbox test unit must be independently operable	For testing purposes, the gearbox testing system of the design must be able to be used independently of the power unit. Therefore, the two units must be designed to accommodate the independent operation.
D.3	Input to the gearbox test unit is limited to a standard PTO speed	In the agricultural industry, standard PTO outputs of tractors are set to either 540 rpm or 1000 rpm, with typical high horsepower implements using 1000 rpm. Therefore, the input to the gearbox test unit must be set to 1000 rpm.
D.4	Loads applied to gearbox must be variable	Current tractors available on the market have PTO power outputs approaching 600 HP. To accurately test gearboxes in a variety of scenarios, the applied loads must be controllable and variable between 0-600 HP.
D.5	The design must be movable using the equipment available by the client	The client has requested that the design is to be movable using equipment that is readily available at their location. Currently, a forklift with a maximum lift capacity of 9000 lbs is available. Therefore, the total weight of components in the design must not exceed 9000 lbs.

#	Design Constraint/Limitation	Description and Implications
D.6	Power unit and test unit must be contained on a skid(s)	Coinciding with the design being movable using the client's available equipment, the client has also requested that the design is contained on a skid(s) with forklift pockets. Therefore, the design must be mounted on a skid(s) capable of containing all required components for operation.
D.7	Physical dimensions of units are limited to not exceed those of a grain cart	Elmer's facility was designed to allow for the transportation of grain carts within facilities. Therefore, to maintain the ability to move the design with a forklift, the overall dimensions must not exceed those of a grain cart. The dimensions of the allowable size are not yet determined.
D.8	The project budget must be reasonable	The client did not provide a set dollar value for the project budget, however, an internal maximum allowable budget of $200000 was set by the design team and approved by Elmer's. If necessary, the value of "reasonable" can be changed upon further research.
D.9	The design must be able to be operated in a variety of environmental conditions	As the designed system will be operated outdoors in Manitoba, the design must be able to operate in temperatures as low as -15°C while maintaining an IP68 or IP69K rating where possible.

#	Design Constraint/Limitation	Description and Implications
D.10	The design must meet safety requirements	As the design will be operated by multiple employees at Elmer's, the design must be safe to use. Provincial safety requirements must be met on all parts of the design to avoid legal repercussions. Therefore, standard safety equipment such as kill switches, guarding, and fire suppression must be included in the design.
D.11	Emission regulations must be met for the power source	Any emissions from the design must meet the legal requirements. Therefore, a minimum emission standard rating of Tier 2 must be met according to the Government of Canada.

TABLE III: TEAM CONSTRAINTS AND LIMITATIONS

#	Team Constraint/Limitation	Description and Implications
T.1	CAD models and drawings must use Autodesk Inventor	Elmer's Manufacturing currently uses Autodesk Inventor as their CAD software. Therefore, to allow for a smooth transition during the design phase of the project the design team must use Autodesk Inventor (2018 or newer version).
T.2	Student budget is limited to the allowable amount provided by the Mech 4860 course	The design team has $400 allotted to help cover project-related expenses such as printing reports and travel. The design team costs must remain under $400.

#	Team Constraint/Limitation	Description and Implications
T.3	Project timeline is limited to the Fall 2019 term at the University of Manitoba	Throughout the Mech 4860 course during the 2019 Fall term, certain deadlines for reports and presentations must be met with the design project concluding on December 6, 2019. These deadlines, viewable in Section 7, must be satisfied by the design team to meet the course requirements for graduating as well as the client's needs.
T.4	Proprietary information must be kept confidential	To avoid patent disputes and leaking of trade secrets, the design team will not share any confidential information as outlined by the signed and dated Confidentiality Agreement.

1.4.3 Metrics

After the needs and constraints were defined, metrics were assigned to each of the needs to provide quantitative specifications that were targeted throughout the design process and used to evaluate the final design. A priority number scale ranging from one to five was used to indicate the importance of the metric, with one being the lowest importance and five being the highest importance. The importance of each metric was determined by the design team based on its associated needs. Target and allowable values denote the ideal and minimum acceptable values respectively for each metric. The metrics, along with their associated information, are outlined in TABLE IV.

TABLE IV: DESIGN METRICS

Metric Number	Associated Needs	Priority Number	Metric	Units	Target	Allowable
1	2.1	5	Maximum power output	HP	600	600-700
2	1.1	4	Movable with forklift	Binary	Yes	Yes
3	1.2	2	Overall dimensions	inch	H x W x L 151x167x447	H x W x L 216x167x500
4	1.3	5	Maximum weight of each unit	lbs	<8000	<9000
5	1.5, 4.2	5	Units can be operated independently	Binary	Yes	Yes
6	3.1	5	Maximum PTO speed	rpm	1000	1000
7	3.2	5	Range of power absorbed	HP	0-800	0-600
8	5.1	3	Total cost	$CAD	<$100k	<$200k
9	5.2	3	Lead time	Business Days	<30	<60
10	5.2	3	Cost to acquire raw materials	% of total budget	<10%	<20%
11	4.1	4	Lowest starting temperature	°C	-15°C	-10°C
12	4.1	4	Resistance to elements	IP Rating	IP69K	IP68

Metric Number	Associated Needs	Priority Number	Metric	Units	Target	Allowable
13	3.3	5	Range of operating angle	Degrees	0-60°	45°
14	3.4	4	Number of gearbox types	N.A.	3+	1
15	2.2	5	Number of external connections	N.A.	0	0
16	4.3	2	Components are accessible	Subjective	N.A.	N.A.
17	4.4	3	Corrosion resistance	Years	20	10
18	4.4	3	Lifetime	Hours	5000	2500
19	4.4	3	Total number of starts	Cycles	45000	15000
20	1.4	3	Amplitude of vibrations	mm	1	5
21	1.4	3	Maximum center of mass height	m	<1	<1.5
22	4.5	2	Setup time	Minutes	10	60
23	4.5	2	Simple user interfaces	Subjective	N.A.	N.A.
24	4.6	3	Noise Levels	dB	<85	<100
25	4.6	3	Emissions Level	Tier Rating	Tier 4	Tier 2
26	4.6	5	Safety guards	Binary	Yes	No
27	4.6	5	Kill switch	Binary	Yes	Yes

Metric Number	Associated Needs	Priority Number	Metric	Units	Target	Allowable
28	4.6	5	Fire suppression	Binary	Yes	No
29	4.6	5	Resistance to overloading components	Binary	Yes	Yes
30	4.7	3	Running time	Hours	10	8

TABLE IV was continually updated throughout the design process and several changes were made since the table's first iteration. Since the concept definition phase of the project was completed, the metrics were updated to include the exact target and allowable dimensions for metric 3 listed in TABLE IV. During the early stages of the concept definition phase of the project, it was revealed that Elmer's is not considering using the gearbox test stand in any areas which are partially or completely enclosed, making noise and emissions less of a priority. As a result of this new information, metrics 24 and 25 were both lowered from priority number 5 to priority number 3. Lastly, after a number of meetings with Elmer's both the design team and the client decided that a run-time of 10 hours was desirable. As a result, the target and allowable metrics were changed from 8 and 5 were changed to 10 and 8, respectively.

1.5 Overall Expectations

The gearbox test stand project will significantly improve the testing methodology which Elmer's currently uses for their Haulmaster gearboxes. The complete assembly will decrease the repair and maintenance costs of the tractors owned by Elmer's farm, decrease the time spent on testing, eliminate scheduling conflicts associated with borrowing tractors for testing and decrease the land area required for testing. The deliverables of the project are as follows:

1) CAD models of the final design.
2) Preliminary engineering drawings for parts fabricated in-house.

3) Overall assembly drawings including exploded views, locations of features and a detailed bill of materials (BOM) of the final design.
4) A list of purchased parts for the final design.
5) A cost estimate for the final design.

Only the renders of the CAD models and cost estimate of the final design are provided in the body of this report. The preliminary engineering drawings and the assembly drawings of the final design are contained in Appendix A of this document while the BOM and the list of purchased parts are contained in Appendix B.

2 Concept Development and Selection Summary

Rather than consider the entire gearbox test stand during the concept generation process, the project was split into multiple major sections, or modules. The modules were treated as individual design problems with unique requirements to avoid a large number of concept variations. After analysis and selection, the chosen concepts were combined to create the conceptual assembly of the gearbox test stand. In addition to simplifying the design process, this approach encourages the design of the gearbox test stand in terms of separable units, corresponding to client need 4.4 in TABLE I.

The gearbox test stand was split into five modules: power source, transmission system, torque limiter, load application system, and gearbox mounting system. These modules were chosen as they are the major components of the final design and will dictate how the other minor components will be designed. A visual representation of the gearbox test stand assembly and each of the modules is shown in Figure 5.

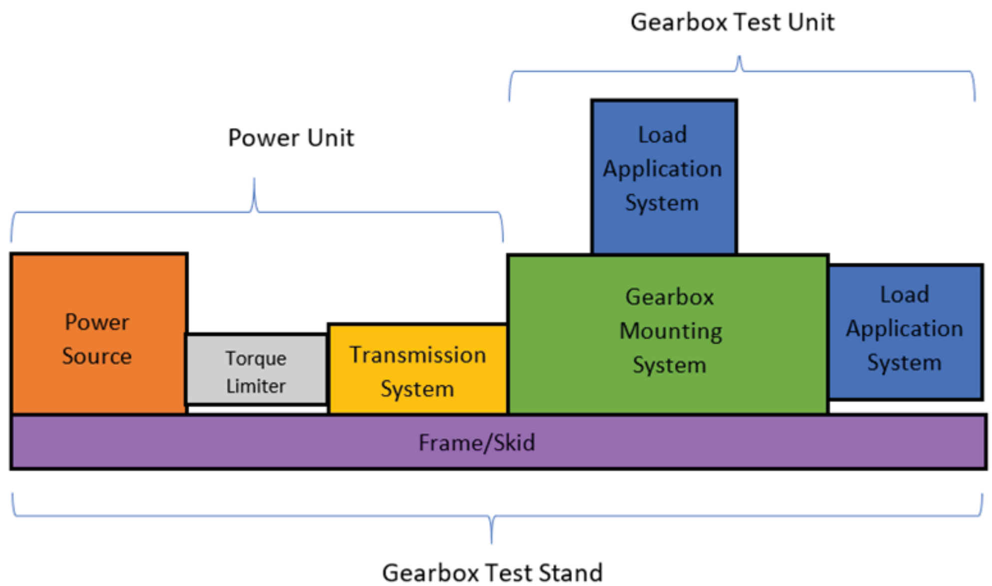


Figure 5. Updated visual representation of the basic components of the final design.

A basic design methodology was defined and followed for each of the five modules to ensure a smooth design process for this project. This section of the report documents the methodology for the concept generation and selection for each of the modules and provides an overview of the selection process for the best concepts.

2.1 Methodology

The project team used several different methods to generate concepts for the gearbox test stand modules. Before developing any concepts, the team categorized the modules based upon whether the solution is designed or sourced. A list of the modules with their assigned category and explanation is shown in TABLE V.

TABLE V: CATEGORIZED MODULES

Module	Category	Explanation
Power Source	Source	As the power source is a complex system, designing a unique solution for this project is impractical. Therefore, the power source will be sourced based on the project requirements.
Transmission System	Source/Design	Due to the varying complexity of concepts, the transmission system could either be sourced or designed.
Torque Limiter	Source	As torque limiters are readily available, this module will be sourced. It is impractical to attempt to design this system when a commercially available equivalent is more effective in regard to cost and design time.
Load Application	Source/Design	Similar to the transmission system, the load application complexity can greatly vary depending on the concept. Therefore, this module was left open to sourcing or designing the component until a concept had been selected.
Gearbox Mounting	Design	Due to the unique design problem, a sourced gearbox mounting component is not likely available. Therefore, this module will be a solution designed by the team.

With module categories fully defined, concepts were brainstormed for each module, first individually and then as a group. For the source modules, concepts were developed using available products. For the design modules, concepts were developed using an external search of patents in addition to brainstorming. The patent search was performed both to generate additional concepts and to determine potential patent infringements.

Once a sufficient list of concepts was generated, a weighted decision matrix (WDM) was used to compare the concepts. WDM criteria for each module were created during project group meetings to suit the specific requirements of each module. The weights of the criteria were then determined individually and averaged for each criterion. Any large discrepancies between group member's values were discussed by the project team. If the discrepancies were due to a misunderstanding of the criterion, the errors were fixed. However, if the discrepancies were due to a difference in opinion, they were left untouched.

The WDM results, along with the weights for the WDM criteria, were discussed with the client to ensure that they aligned with the client's vision for the project. If the client deemed necessary, this process could be repeated using other concept generation methods, such as SCAMPER or TRIZ 40. To select the final concept for each module, the team considered the results of the WDM along with the client input to ensure the design solution is unbiased and will achieve the project requirements. The WDM for each design module and weights of the criteria are contained in Appendix C of this document

2.2 Power Source

The power source is the heart of the power unit, which provides the power required to test Elmer's gearboxes. The selected power source must be capable of providing 600 HP to the system, where the power can be routed to different outputs via the load application method. Several engine types were considered for the power source module, including the diesel engine, gasoline engine, electric motor, gas turbine engine, steam turbine, and a diesel-electric hybrid engine. The team also considered the possibility of purchasing a standalone engine for the gasoline and diesel concepts instead of a complete unit and designing the cooling, fuel and emissions systems. The concepts were weighed against one another in a WDM using the following set of criteria listed in TABLE VI.

TABLE VI: POWER SOURCE DEFINED CRITERIA

Criterion	Explanation
Weight	The projected weight of the power unit which incorporates the concept.
Cost	The projected overall cost of the concept.
Size	The physical dimensions of the concept.
Torque	The torque output of the concept corresponding to the required power output.
External Dependencies	The relative number of permanent or semi-permanent connections required for the operation of the concept.
Low Starting Temperature	The lowest temperature at which the concept will operate.
Vibration	Vibrations caused by the operation of the concept.
Running Time	The longest continuous running time that the concept is capable of.
Noise Level	The noise output of the concept.
Implementation	The ease with which the concept can be implemented into the final design.
Emissions	The difficulty with which the concept will meet the applicable emission standards.

The complete diesel power unit performed best in the weighing scheme due to its ease of implementation, lack of external dependencies and excellent torque capability. Additionally, since the tractors which provide power to Elmer's equipment in the field are powered by diesel engine, a diesel power unit is the best representation of real-world use. The results of the concept analysis were presented to Elmer's and the selection of a diesel power unit for the final design was approved.

2.3 Transmission System

In order to replicate realistic testing conditions, the power from the power unit to the test gearbox must be transmitted at a standard PTO shaft output of 1000 rpm. Since the diesel engine selected for the power unit provides maximum power at a shaft speed of 2000 rpm, the gearbox test stand will require a reduction gearbox with a ratio of 2:1 between the test gearbox and the power source. Three different types of gearboxes were considered for this purpose: planetary gearbox, compound gearbox, and single-stage gearbox. Each gearbox concept had both helical and spur gear variations which were evaluated. Other transmission concepts, including a belt-drive, chain drive, and a torque converter were also considered. The concepts were weighed against one another in a WDM using the criteria outlined in TABLE VII.

TABLE VII: TRANSMISSION CONCEPT SELECTION CRITERIA

Criterion	Explanation
Weight	The projected weight of the concept.
Cost	The projected overall cost of the concept.
Size	The physical dimensions of the concept.
Power Transmission Capability	The ease with which the concept can be designed to accommodate the required power output.

Criterion	Explanation
Simplicity	The relative simplicity of the concept in regard to its operating principle and the amount of design work required for implementation.
Life	The relative length of the projected service life of the concept.
Vibration	The vibration caused by the operation of the concept.
Efficiency	The efficiency of power transmission provided by the concept.
Noise	The noise output of the concept.

The single-stage gearbox concept achieved the highest rating out of all the concepts due to its relative simplicity, high power transmission capability, low weight, and long expected service life. The spur gears were selected for the concept to simplify the design process for the gearbox. An additional benefit to choosing a single-stage gearbox with spur gears is that it is the same concept as Elmer's uses for their test gearboxes. This creates potential for the reduction gearbox design to share components with Elmer's gearboxes, further simplifying the design process. The team discussed the results of the concept analysis for the reduction gearbox with Elmer's, who approved the single-stage spur gearbox concept for the final design.

2.4 Torque Limiter

The torque limiter is a device that is placed between a power source and the power input, which in the context of this project are the power unit and the gearbox input. The torque limiter will interrupt the connection between the power source and the power input if a certain specified torque value is exceeded. Similar to circuit breakers in an electrical circuit, torque limiters offer protection for the weaker components in the system in the event that the design load is exceeded. A total of four torque limiter concepts were considered for the final design, including the shear pin torque limiter, ball detent torque limiter, slip clutch torque limiter and the air engaged torque limiter. The concepts were weighed against one another according to the criteria outlined in TABLE VIII.

TABLE VIII: TORQUE LIMITER CONCEPT SELECTION CRITERIA

Criterion	Explanation
Weight	The projected weight of the concept.
Cost	The projected overall cost of the concept.
Size	The physical dimensions of the concept.
Torque Capability	The relative availability of variants of the concept which are able to transmit the torque output of the power source
Reusability	The capability of the concept to withstand multiple overloading scenarios.
External Dependencies	The relative number of permanent or semi-permanent connections required for the operation of the concept.
Availability	The commercial availability of torque limiters using the specific concept.
Resistance to Elements	The ability of the torque limiter to remain reliable when exposed to outdoor conditions.
Response Time	The duration of time between when the system begins to experience a load which exceeds the design load, and when the torque limiter disengages the power source from the protected component.

The slip clutch was the highest-rated concept due to its wide availability, low cost and the relative availability of designs with high torque ratings in relation to other concepts. An additional advantage of the slip clutch concept, Elmer’s Manufacturing already has an established relationship with a vendor that offers PTO shafts with integrated slip clutches. The team presented the results of the concept selection to the client for feedback. Elmer’s Manufacturing approved of the outcome of the concept selection.

2.5 Load Application

The load application module provides resistance to the rotation of the test gearbox output shafts. The load application system must be capable of absorbing all the power and torque supplied to it by the diesel engine in the power unit through the various gearboxes installed in between the two modules. The load application system must also be variable by the client and provide loads of 0-600 HP to accommodate need 3.4 in TABLE I. The team considered eight different options for the load application system, including an eddy-current dynamometer, water brake dynamometer, hydraulic pump dynamometer, air brake dynamometer, hysteresis dynamometer, magnetic powder dynamometer, electric generator dynamometer, and friction brake dynamometer. All these concepts were weighed against one another in a WDM using the criteria outlined in TABLE IX.

TABLE IX: LOAD APPLICATION SYSTEM DEFINED CRITERIA

Criterion	Explanation
Weight	The projected weight of the concept.
Cost	The projected overall cost of the concept.
Size	The physical dimensions of the concept.
Power Absorption Capability	The relative availability of variants of the concept which are able to absorb the power output of the power source

Criterion	Explanation
External Dependencies	The relative number of permanent or semi-permanent connections required for the operation of the concept.
Simplicity	The relative simplicity of the concept in regard to its operating principle and the amount of design work required for implementation.
Life	The relative length of the projected service life of the concept.

The water brake dynamometer achieved the highest rating according to the team's weighting criteria due to its low cost, weight, and availability of units that have high power absorption capabilities. The hydraulic pump dynamometer followed closely, therefore the results of the WDM were forwarded to the client for feedback. The feedback which the team received from Elmer's Manufacturing strongly recommended to select the hydraulic pump concept for the final design. The basis for the client's recommendation was the large reservoir size and difficulty in sourcing pumps capable of the flowrate required by a water brake which fulfills the project requirements. The hydraulic pump dynamometer requires far less flow rate, and as a result a far smaller reservoir than a water brake. The hydraulic pump also uses oil rather than water as the working fluid, which allows the load application module to be stored in subfreezing temperatures without the need for heating. For these reasons and in accordance with client recommendations, the team selected the hydraulic pump dynamometer.

2.6 Gearbox Mounting

The gearbox mounting module consists of a custom mount that restrains the Haulmaster gearbox during testing while also allowing the angle of the gearbox's unload auger output to be adjusted between 30 and 60 degrees to the horizontal. The adjustability of the output angle mimics the adjustability of a grain cart's unloading auger. A variety of concepts for the gearbox mount were developed by the team in the initial concept generation stage. These concepts were evaluated in accordance with the criteria outlined in TABLE X.

TABLE X: GEARBOX MOUNTING DEFINED CRITERIA

Criterion	Explanation
Weight	The projected weight of the concept.
Cost	The projected overall cost of the concept.
Size	The physical dimensions of the concept.
Adjustability	The relative number of positions which the concept can accommodate.
Setup Time	The relative time required to adjust the test gearbox and begin testing while using the concept.
Manufacturability	The relative availability of the components and or manufacturing processes required to implement the concept.
Life	The relative length of the projected service life of the concept.
Stability	The relative projected deflection of the test gearbox during operation while using the concept

The three concepts which achieved the highest rating in the WDM were the plate mounting concept, the adjustable arm mounting concept and the pivoting U-bracket mounting concept. These concepts generally offered good manufacturability, short setup times, acceptable adjustability and provided a rigid mounting structure for the gearbox. Prior to selection of the final concept, the U-bracket, and adjustable arm concepts were modified to include a skid base which allowed for quick swaps between the Haulmaster and Wolverine ditcher gearboxes.

The top three concepts were compared against each other and their skid-based variants in a WDM using the same criteria as outlined earlier in this subsection. The skid-based variant of the adjustable arms mount achieved the highest rating out of the group due to a good combination of service life adjustability and short setup time.

2.7 Summary of Concept Development

The final gearbox test stand concept consisted of the final choices for each module as presented in this chapter. The description of the final gearbox test stand concept broken down by modules is as follows:

- A diesel power unit as the power source.
- A single-stage spur gearbox as the transmission system.
- A slip clutch as the torque limiter.
- A hydraulic pump dynamometer as the load application system.
- Adjustable arms on a mini-skid to mount the gearboxes.
- Miscellaneous components such as the skids and fuel tank.

These components were all selected individually without reference to one another to ensure that the most appropriate concept within each module was selected. A cost estimate was performed during the closing of the concept definition stage of the project, to ensure that the project adhered to the internal budget set in D.8 in TABLE II. Prior to designing any components, the initial cost estimate of the entire concept was $157,000 CAD.

3 Final Design

With the concepts for each major design module selected, the team developed the concepts into the final design. The following sections go into the details of the design, first explaining a general overview of the system, the general design methodology used for the individual components, followed by the detailed explanation of the design process for each component.

3.1 Overview

The components were assembled to create the final gearbox test stand design using a rigorous design methodology which explored various concepts. The final design which integrates all components is shown in Figure 4.

[redacted]

Figure 4. Labelled render of Final Gearbox Test Stand Design.

As shown in the render, there are two major systems within the final design; the power unit and the gearbox test unit. The power unit contains the necessary components to provide power to the client's gearbox input, while the gearbox test unit contains the components required to hold the client's gearbox and apply a load to the gearbox. This section will briefly cover the components within this design, beginning with the power unit. A render of the power unit identifying the individual components, is shown in Figure 5.

[redacted]

Figure 5. Labelled render of final power unit assembly.

The power unit is centered around a Deutz TCD 13.5 engine, capable of producing 600 HP and 2000 ft-lb at 2000 rpm. One of the deciding factors for selecting this engine was that there were clutches readily available for this application, through the assistance of Pritchard Group. The clutch selected for this application is an IB 314P from TwinDisc, which allows power from the engine to be decoupled from the rest of the system. This clutch was necessary for the design as the engine must be de-coupled when started or idling during a cool-down period.

Accompanying the diesel engine is a diesel fuel tank. The tank capacity was calculated by using the specific fuel consumption with an assumed maximum runtime of 10 hours to be 1183 L. Rather than designing a custom tank to this capacity, a 1300 L tank was chosen from Westeel, as their product is more effective than a custom-designed tank.

Gearboxes are typically driven by a PTO, where the speed of the engine needed to be reduced to 1000 rpm. This speed reduction uses a specially designed gearbox with a 2:1 reduction and capable of transmitting the required horsepower. The reduction gearbox was designed to feature components of Elmer’s Haulmaster gearbox, with some minor modifications. The Haulmaster gearbox was chosen in this application because the parts used to manufacture this gearbox are already available to the client. Furthermore, it was known that the highest amount of torque throughout the system will occur within the test gearbox, meaning that the test gearbox will fail before the reduction gearbox. By using the same components, the reduction gearbox would not fail under regular testing conditions. A custom adapter was designed to connect the output of the clutch to the reduction gearbox.

The clutch was specified to attach to the engine’s flywheel housing and mount plates were designed to support the reduction gearbox. The skid was extended using standard C-channel sizes which were then attached to the existing engine skid using designed plates.

With the transmitted power reduced to the correct input speed, a PTO shaft featuring a slip-clutch is used to send the power to the test gearbox, which is housed within the gearbox test unit, shown in Figure 6. The slip-clutch acts as a safety mechanism that will limit the maximum amount of torque through the system. This ensures that the powertrain is not able to be overloaded, which would cause significant damage.

[redacted]

Figure 6. Labeled render of the final gearbox test unit design.

As previously discussed, the Haulmaster gearbox was required to be tested at a range of possible operating angles. The desired range of operating angles is possible through the design of the test gearbox mounting system which features two mounts; the pivot mount and the adjustable mount. The pivot mount was designed through the use of preliminary FEA to be rigid and support the large amounts of torque being transmitted through the system, while the adjustable mount was designed analytically to provide the range of necessary testing angles. The mounts were designed with the main goal of reducing set-up time and the required effort of the operators.

The mounting system was designed to be contained on a separate mini-skid, allowing the operator to remove the mini-skid from the entire assembly, giving more space to attach the test gearbox to the system. The pivot mount was designed so that the Haulmaster gearbox could be attached while lying flat on the mini-skid, reducing the need for additional supports. The adjustable mount includes a dedicated lifting point so that the test gearbox could be lifted once attached to the pivot mount. The adjustable arm is attached by a pin to the mounting plate, greatly reducing set-up time. Finally, the adjustable arm allows 6" of extension from the smallest to largest length, limiting the amount of operator effort required.

With the gearbox at the appropriate testing angle and securely mounted, the load is then applied to a Kawasaki K3VG-280DT hydraulic pump. The pump was selected because it is capable of producing a maximum of 5000 psi at up to 206 gpm, which is equivalent to a variable load up to 600 HP. Since the pumps operate at a rated speed of 1500 rpm, an increaser gearbox was designed to achieve the required speed. The increaser gearbox features a planetary gearbox to keep the system compact while handling the required 6300 ft-lb. The gearbox was designed to have a 3:1 increase, where the power is inputted to the sun and transmitted out of the carrier with the ring gear held in place. An adapter was designed to connect the output of the gearbox to the pump. The strength of the gearbox casing was verified using FEA to withstand the resultant forces and mount to the pump brackets.

To accommodate the range of desired or gearbox operating angles, two separate pump mounting systems were designed. The adjustable pump mount supports the weight and resultant loads of the pump and planetary gearbox at a range of discrete angles. The system features two main members, the vertical support and diagonal support, which includes holes in specific locations to give a range of possible testing angles that are secured with bolts. Additional supports were designed to support the structure while keeping the overall size of the design compact. To ensure that the gearbox could be tested at any possible angle between 30-60 degrees, a PTO shaft was selected to transmit the power from the test gearbox to the increaser gearbox. This PTO shaft allows for slight misalignment, eliminating the effects of the discrete operating angles of the pump, increasing the simplicity of the design.

As the operating angle of the test gearbox rotates, the output shaft location changes. Therefore, the stationary pump mount was designed so that the pump was centered between the possible output shaft locations. Using the same pump brackets as the adjustable pump mount, the stationary pump mount features a simple structure made of sheet metal and was verified for strength through preliminary FEA. The structure is to be rigidly welded and bolted in place on the skid to maintain its location relative to the other components of the system.

With the physical power transformed into fluid flow through the pumps, the actual load application was applied through the use of a restricting valve. Due to the high flow rates, each pump was designed to split the output into two cartridge valves from Sun Hydraulics, operating in parallel. These valves were selected as they are able to handle flow rates of 200 gpm each, and can be controlled electronically, allowing for a completely variable load. These valves and the necessary manifold blocks are to be mounted onto a bracket which is attached to the gearbox test unit skid, allowing for them to be easily accessible.

The hydraulic oil system design includes filters, removing any possible contaminants for the hydraulic system. Once passed through the filters, the oil flows through a cooler since the load applied across the valve results in large amounts of heat being introduced to the oil. The cooler is rated for 707 HP at a flow rate of 206 gpm, ensuring that the oil will maintain a safe and stable operating temperature during testing. The cooler accepts the two inputs from the adjustable and stationary pump and returns the oil from a single output to the reservoir. The reservoir was calculated to hold 680 gallons of oil, ensuring that there is sufficient oil within the system. Additionally, the reservoir includes a baffle plate to remove entrained air within the hydraulic oil, therefore improving the hydraulic dynamometer’s performance. Other components, such as an input diffuser and sight-glass were incorporated into the reservoir design to ensure proper conditions for the oil within the hydraulic dynamometer were met.

Finally, the entire hydraulic oils system design includes the design of the hoses which were evaluated for the required pressure and flow rate at each point throughout the system. Additionally, locations for sensors such as pressure sensors and temperature sensors were included to allow for the client to equip the system with a data acquisition system as desired.

With the final design summarized, the following section describes the design methodology that was applied for each of the individual components.

3.2 General Design Methodology

A general design methodology was followed to ensure a thorough and consistent process throughout the design of each of the individual components. Since there were certain aspects of the design that were sourced rather than designed, there were slightly different procedures for each case.

The first step in the design process was performing all of the necessary calculations for sourced components. The possible alternatives were then identified based on the results of the analysis. A selection was then made on which alternative best suits the design, however, there were limited alternatives for the majority of components that required sourcing.

For components that were designed, the first step in the procedure was determining the level of complexity required for the design. The complexity of the component and its loading conditions determined whether the design was created with an analytical approach or solely using preliminary FEA. For simpler components, the analytical approach was performed to ensure that the component is designed using proven equations. These components were then verified using preliminary FEA. For complex components and loading conditions, performing the design process with analytical equations was not feasible. Therefore, the design process was iterative, beginning with an initial concept and then using preliminary FEA to improve the component until the design met the specific criteria.

A general procedure for the preliminary FEA was created to ensure design consistency across the individual components was kept. The FEA was performed through Autodesk Inventor's stress analysis tool. The mesh settings were initially set to one-tenth of the part's bounding box length. The minimum element size was set to 0.2, as a fraction of the average size. These mesh settings are relatively coarse but allowed for most designs to be evaluated within a reasonable time frame. For components with a disproportionally large length, these mesh settings were reduced as much as possible to improve the quality of the results. The grading factor and maximum turn angle were set at 1.5 and 60 degrees, respectively. All studies were allowed a maximum of 5 mesh refinements with a stop criterion of 10%. Additionally, the h refinement threshold was set at 0.75.

These settings allowed for the assemblies and components to be analyzed within a reasonable timeframe but often resulted in a lack of convergence of study results. However, the scope of the FEA throughout this design process was to provide general insight. Properly analyzing each individual component through FEA analysis was impossible in the time frame provided to the design team. The study results for all of the models were displayed with contour shading of the von Mises stress and actual displacement. Additionally, each stress plot is accompanied by the convergence plot.

The material choice for all of the designed components was restricted to a list of commonly available and client recommended materials shown in TABLE XI. This ensured the consistency of materials for all designs, requiring a smaller number of vendors for sourcing. Usage of similar materials also ensured that the materials are compatible for welding.

TABLE XI: LIST OF COMMON MATERIALS THROUGHOUT FINAL DESIGN

Material	Yield Strength [ksi]
1566 Carbon Steel	75.0 [2]
ASTM A36	36.3 [3]
ASTM A193	105 [4]
ASTM A513	72.0 [5]
ASTM A500C	50 [6]
ANSI 65-45-12 Ductile Iron	65 [7]
AISI 4140H (UNS H41400)	152 [8]

3.3 Design Details

As previously mentioned, the design process of the gearbox test stand was split into two main categories being the design of the power unit, and the design of the gearbox test unit. The detailed design of the components within each of these two systems are shown in the following subsections.

3.3.1 Power Unit

As shown in Figure 7, the power unit for the gearbox test stand project includes three major components, being the diesel power source itself, a reduction gearbox to achieve a PTO speed of 1000 rpm from the engine rpm, and a fuel system for the diesel power source.

Figure 7. Render of final power unit design, including engine skid and fuel tank.

3.3.1.1 Diesel Power Unit

The concept development conducted by the design team lead to the selection of a diesel power unit as the power source. These diesel power units include a diesel engine, a cooling system, and the appropriate emissions systems to meet the legal requirements set by the Canadian government, with all options being factory-installed. To begin, a scope table was created to ensure the diesel power unit meets the needs of the client while not exceeding the requirements of the project. The scope table outlines the tasks performed by the team and the tasks that were not included in the project. The scope table is shown in TABLE XII.

TABLE XII: SCOPE OF DIESEL POWER UNIT DESIGN

In Scope	Out of Scope
Sourcing of diesel power unit with an emissions system and controls from the engine manufacturer	Sourcing fasteners to secure clutch to flywheel housing
Sourcing clutch for flywheel housing on the diesel power unit	Specifying the required engine and clutch lubrication methods

Cummins and Pritchard Dieseltech were contacted for their options regarding diesel power units, including the available options, power ratings, and pricing. Both companies provided information on diesel power units that are acceptable for the gearbox test stand. The Cummins unit is based on a QSX15 engine, while the Pritchard Dieseltech option is a Deutz TCD 13.5. The relevant specifications of these diesel power units are shown in TABLE XIII.

TABLE XIII: SPECIFICATIONS OF CUMMINS QSX15 AND DEUTZ TCD 13.5

Specification	Cummins QSX15 Power Unit [9]	Deutz TCD 13.5 Power Unit [10]
Displacement [L]	14.9	13.5
Power [HP]	600	600
Torque [ft-lb]	2050	2046
System Voltage [V]	24	24
Block Heater Option	No	Yes

Specification	Cummins QSX15 Power Unit [9]	Deutz TCD 13.5 Power Unit [10]
Clutch Available	No*	No**
Skid Available	Yes	Yes
Price [$CAD]	***	85,000

**It is up to the end-user to find a clutch for the QSX15 power unit.*

***Clutches are not readily available from Deutz, however, the Pritchard Group offered to help source a clutch for the Deutz power unit.*

****Cummins has requested that all pricing discussions are left between Elmer's and Cummins. An NDA was signed by the team before receiving any pricing from Cummins.*

The information in TABLE XIII was presented to the client during a meeting, where the decision was made to use the Deutz. As both engines have the same output power and similar torque numbers, the main deciding factor came to price. Elmer's chose to use the Deutz as the option saves over $25,000 when compared to the Cummins.

The Deutz engine was also chosen over the Cummins engine as the Deutz dealer, Pritchard Dieseltech, offered to help with the selection of a clutch. With Pritchard Dieseltech's assistance, the IB 314P clutch from TwinDisc was selected for this application. The clutch selected is a triple plate clutch capable of handling 2430 ft-lb, which is greater than the output of the Deutz output of 2046 ft-lb. A model of the diesel power unit and clutch assembly are shown in Figure 8.

Figure 8. Render of Deutz TCD 13.5 with IB 314P clutch.

It is noted that the model provided for the engine is not the final model of the engine and does not show the emissions system. The actual engine does include the necessary emissions components, which was included in the given price. With the engine and clutch specified, the next designed component in the power unit was the reduction gearbox.

3.3.1.2 Clutch to Reduction Gearbox Adapter

An adapter was designed to transmit the power from the clutch to the reduction gearbox, since the clutch output is a keyed shaft and the reduction gearbox input is a splined shaft. A basic shaft was designed for this purpose. One end of the shaft is hollow with a keyway which fits over the clutch's keyed output shaft, shown in Figure 9.

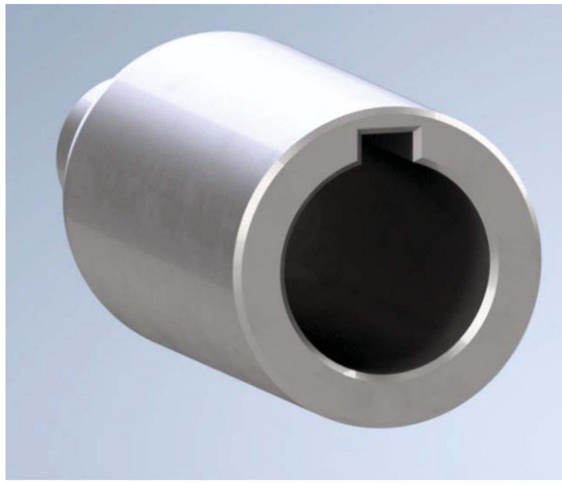


Figure 9. Render of clutch adapter featuring the clutch end.

The opposite end of the adapter has an internal spline that matches the reduction gearbox input shaft spline, shown in Figure 10.

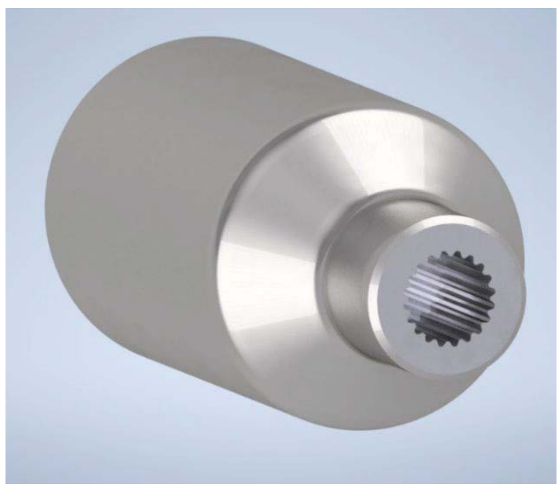


Figure 10. Render of clutch adapter featuring the gearbox end.

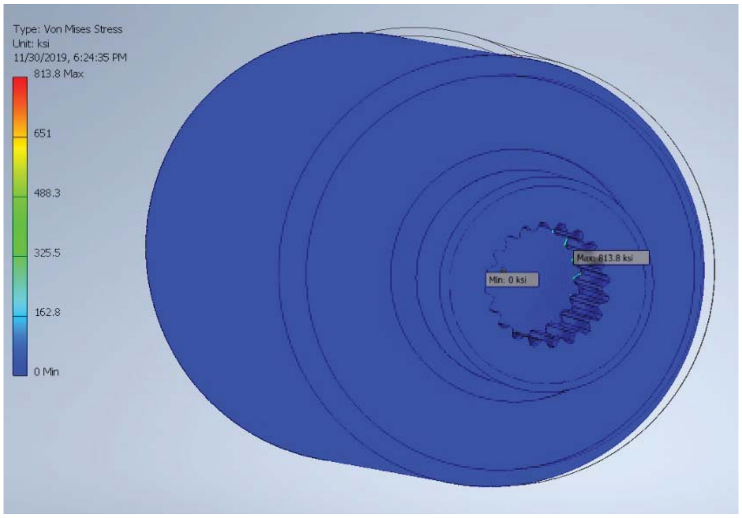
The middle area of the shaft between these two hollow sections is solid and the shaft tapers to reduce its overall weight. A side view of the adapter is shown in Figure 11.

Figure 11. Render of clutch adapter displaying a side view.

The shaft material selected was ANSI 65-45-12 ductile iron, whose material properties are shown in TABLE XI. Preliminary finite element analysis was performed to verify that the adapter can transmit the torque without deforming. The spline faces that will resist the twisting motion of the shaft were fixed and a distributed load was applied to the face of the keyway that will resist the twisting motion of the clutch output shaft. The magnitude of the distributed load on the keyway was calculated using the following equation:

$$F = \frac{T}{(D/2)}$$

where F is the force on the keyway, T is the torque being transmitted and d is the diameter of the keyed shaft. This calculation resulted in a force of 9543.6 lbf being applied to the keyway. The results of the finite element analysis are shown in Figure 12.



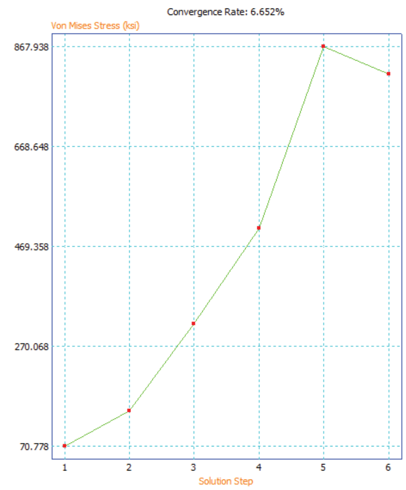


Figure 12. Results of preliminary FEA for clutch adapter.

As can be seen from the preliminary FEA results, the adapter has a high factor of safety in all areas except the sharp edges where the internal splines meet the solid section of the shaft. This is likely due to the fixed constraint being applied to the spline faces. In reality where some flexibility of the spline faces is allowed this stress concentration may not be an issue. This potential issue is recommended to be further investigated by Elmer's before manufacturing the part and implementing the design.

3.3.1.3 Reduction Gearbox

A reduction gearbox between the diesel engine and the test gearbox was required in order to have the correct input torque and speed to the test gearbox to replicate the gearbox being driven from a PTO shaft. This section covers the design of this reduction gearbox, with the scope of the design defined in TABLE XIV.

TABLE XIV: SCOPE OF REDUCTION GEARBOX DESIGN

In Scope	Out of Scope
Initial Design of Gearbox Components	Fastener Selection
Preliminary FEA of Case Ends	Fatigue Analysis
	Fracture Analysis
	Lubrication

The reduction gearbox consists of a single spur reduction with a 2:1 ratio. The design features splined input and output shafts and case ends which can be directly bolted to the mounting plates on the power unit skid. An overview of the gearbox is shown in Figure 13 and Figure 14.

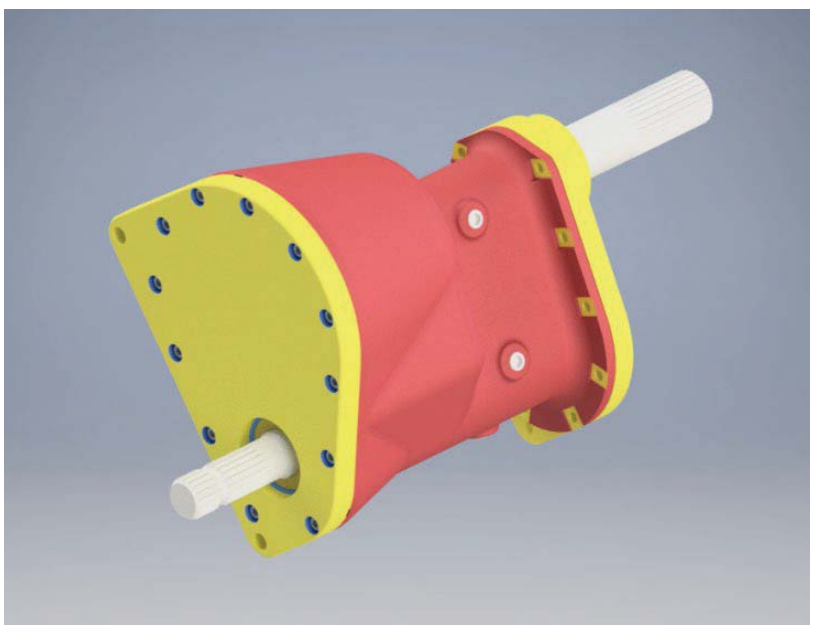


Figure 13. Render of reduction gearbox design.

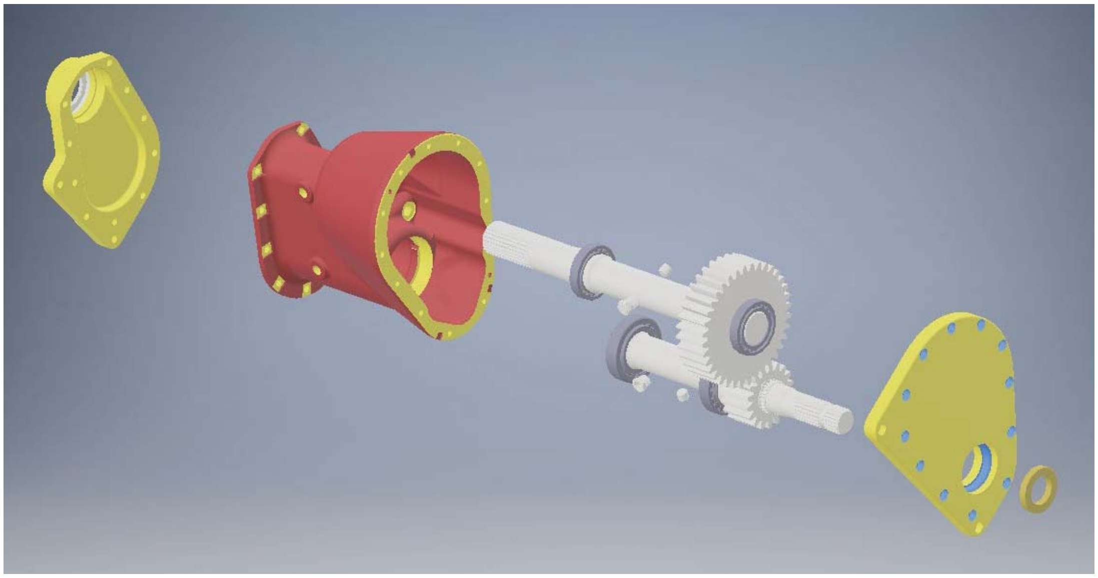


Figure 14. Exploded view of reducer gearbox design.

Since Elmer’s Haulmaster grain cart gearbox has a 2:1 reduction ratio, components of this gearbox were reused, and some were modified, in order to create all the parts of the reduction gearbox. This was done to reduce costs and the amount of design work, a decision that was approved by Elmer’s. The main components of the Haulmaster gearbox that were reused without modification are the main casing section shown in red, the pinion gear, the fill and drain plugs and breather, as well as the shaft bearings and seals. The input and output shafts, output gear and case ends were created by modifying components of the Haulmaster gearbox.

3.3.1.3.1 Gears

The pinion gear of the Haulmaster gearbox was reused as the pinion gear of the reduction gearbox. The Haulmaster’s output shaft spur gear was reused as the output gear of the reduction gearbox with some modifications from the original design. The dog teeth grooves that were used to engage and disengage the gear in the Haulmaster gearbox were eliminated in order to simplify and strengthen the design. The modified gear is shown in Figure 15.

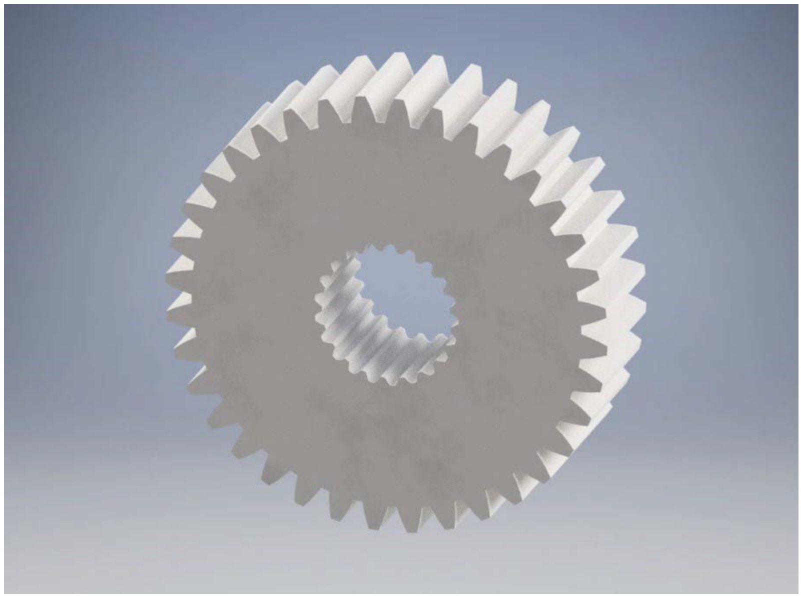


Figure 15. Render of modified Haulmaster gear for reducer gearbox.

Internal splines were created on the output gear, allowing the gear to be connected by spline to the output shaft. This spline is the same spline used to transmit the power from the output shaft gear to the output shaft in the Haulmaster gearbox, however this spline is now on the output gear itself instead of on the engagement mechanism’s dog gear. Analysis of the strength of this internal spline was not necessary, as it is longer than the one used on the engagement dog gear and only has to transmit half the torque it transmits in the Haulmaster gearbox.

Full calculation of the bending and contact stresses experienced by the gears was not necessary, as the AGMA stress equations can be used along with the known loading and gear speed values to determine whether the gears will fail before the test gearbox fails. Since the reduction gearbox gears are at least as strong as the Haulmaster gearbox gears, a simple comparison of the transmitted load and number of loading cycles experienced by each gearbox is adequate. The reduction gearbox is only subjected to half the torque that the Haulmaster gearbox would experience, meaning that the transmitted load for the reduction gearbox gears will also be half that of the Haulmaster gearbox gears. According to the AGMA stress equations, this reduces the bending and contact stresses by 50% and 30% respectively.

While the torque is lower, the gears are rotating at double the speed compared to the gears in the Haulmaster gearbox and therefore are exposed to double the loading cycles, which affects safety factors against bending stress and pitting resistance. Specifically, this affects the values of the stress cycle factors used in the factor of safety calculations for bending and contact stresses. However, doubling the number of cycles has a minimal effect on the stress cycle factors, which would not result in enough change to offset the reduction in bending and contact stresses to a point where the factor of safety would decrease [11]. Therefore, it can be concluded that the reduction gearbox gears should not fail before the test gearbox.

3.3.1.3.2 Shafts

The speed reducer gearbox was designed to have one input and one output shaft which run parallel to each other. Modified versions of the Haulmaster gearbox's parallel input and output shafts were used as the input and output shafts of the reduction gearbox. The input shaft was shortened on the output end to fit within the casing.

The output shaft required more extensive modifications from the original Haulmaster part. A render of the reduction gearbox output shaft design is shown in Figure 16.

Figure 16. Render of modified Haulmaster output shaft for reducer gearbox.

The modified shaft was designed by shortening the shaft to fit in the reduction gearbox casing. The spline was designed to be longer, and its location was moved so that the output gear could be directly splined onto the shaft. This spline is the same as the one used in the Haulmaster gearbox on the engagement dog gear to transmit the torque from the output gear through the engagement gear to the output shaft. Since the same shaft material, shaft diameter and spline type as in the Haulmaster gearbox was used, and the length of the spline that is transmitting the torque is longer, with lower torque requirements, this spline will be more than adequate for the needs of the reduction gearbox. The retaining ring used in the reducer gearbox to prevent axial movement of the output gear was selected to be the same as the one used in the Haulmaster gearbox, as are the retaining ring groove size and location relative to the gear splines. The same shoulder used on the output shaft in the Haulmaster gearbox to support the output end of the output shaft was also selected on the reduction gearbox output shaft to prevent axial movement of the shaft. The input side of the reduction gearbox output shaft was designed to butt up against the case to fully constrain axial movement of the shaft. Since the shear force and bending moment on the reducer gearbox output shaft will be less at the locations of the various shaft features than in the Haulmaster gearbox, there should be no issues with failure due to the incorporation of these features and further analysis is not necessary.

3.3.1.3.3 Casing

The casing of the reduction gearbox consists of three pieces, one main housing section shown in red and two end caps shown in yellow in Figure 13. The main housing section is reused from the Haulmaster gearbox, which is cast in five separate pieces that are secured together with fasteners. The main housing section is one of these five pieces. Custom endplates were designed to cover the exposed ends of the main housing section and facilitate mounting of the reduction gearbox.

The case endplates were designed with the same inner geometry and shaft support structure as the original Haulmaster gearbox end caps in order to support the shafts using the same methods used in the Haulmaster gearbox. The case endplates were also designed to be machined from the same cast 65-45-12 ductile iron material used to cast the main housing section of the case. This material was chosen to keep the case material consistent, avoiding issues such as galvanic corrosion and different rates of thermal expansion between case sections.

Preliminary finite element analysis was performed on the case ends to verify they would stand up to the forces exerted on them. To determine the forces the case will experience when in service, the output shaft bearing forces were first determined by calculating the separation force that results from the gears meshing, then using a sum of moments to determine the forces on each bearing, and consequently the forces exerted on each bearing support built that is built into the case end caps. To simulate the loading conditions on the case end plates, the inner surfaces of the reduction mounting holes were fixed and the bearings support surfaces were split and the appropriate forces were applied to the upper bearing support surface sections of each case endplate in an outwards vertical direction. The general convergence settings outlined in Section 3.2 - General Design Methodology were used. The results of the finite element analyses can be seen in Figure 17, Figure 19 and Figure 18.

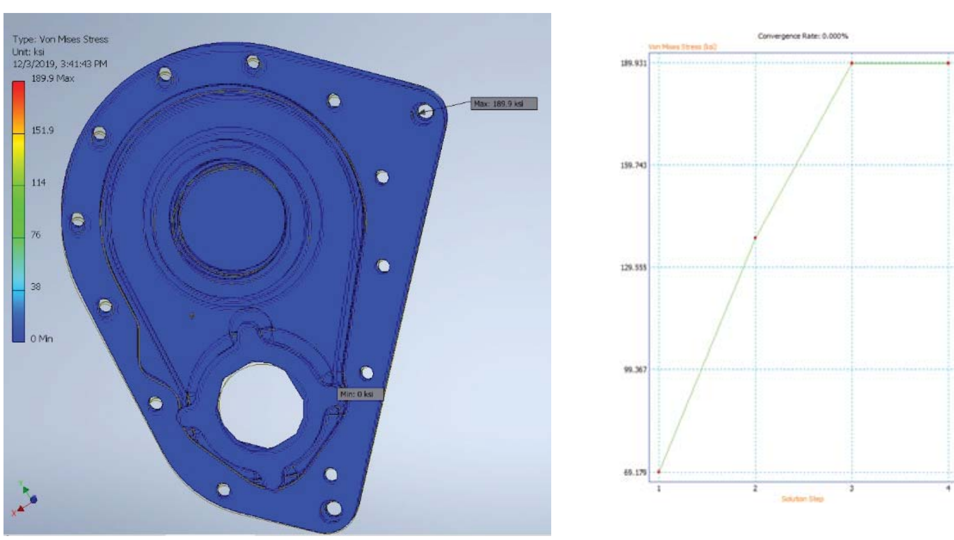


Figure 17. Results of preliminary FEA for the reducer gearbox input endplate.

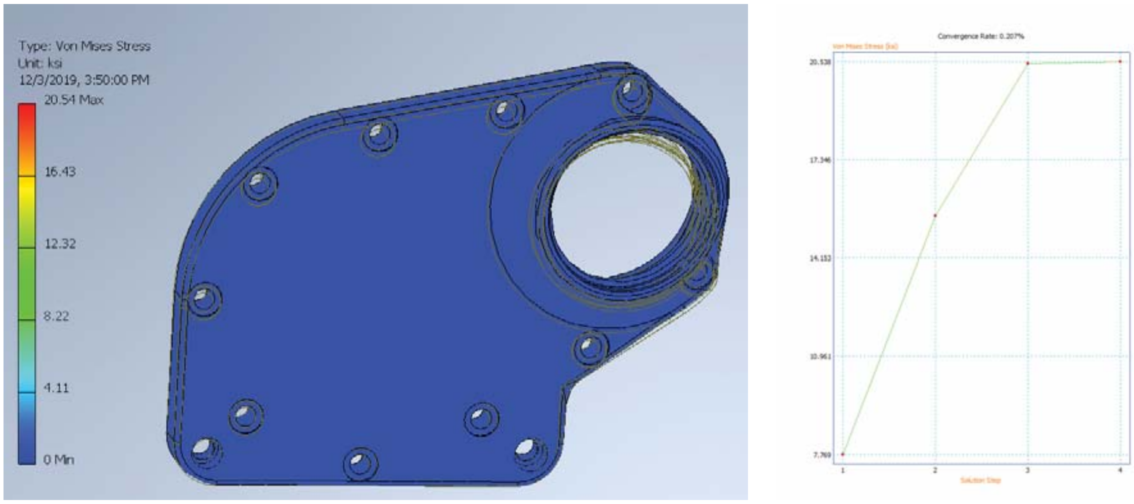


Figure 18. Results of preliminary FEA for reducer gearbox output endplate.

The finite element analysis showed that the case ends have a high factor of safety. The only areas that had high-stress values were the sharp edges of the gearbox mounting holes. This stress concentration likely appeared due to the loading conditions applied to the study, and therefore would not be expected to actually occur. Since the surrounding mounting area shows minimal stress, this concentration is likely not of concern. An example of this stress concentration can be seen in Figure 19.

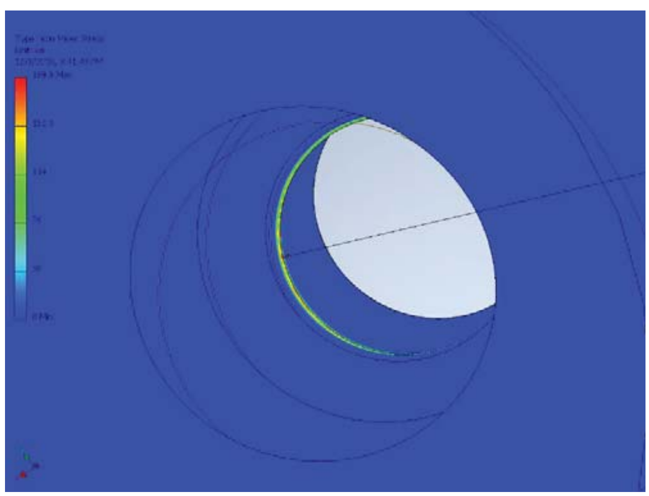


Figure 19. Stress concentration on mounting hole chamfer of reducer gearbox input endplate.

As with the other FEA results, these are preliminary results and the actual stress may be different, therefore further analysis is recommended.

The shape of the case endplates, and the location of the mounting holes were designed so that the reduction gearbox is mounted at a 45-degree angle with respect to the ground. This angle was chosen in order to minimize lubrication issues. The Haulmaster gearbox can be mounted at a range of angles from 30 to 60 degrees when installed in the grain cart and used in the field. Thus, by mounting the reduction gearbox at a 45-degree angle the reduction gearbox should have adequate lubrication if the same lubrication method is used as the Haulmaster gearbox. However, it is recommended that Elmer's further investigate how to adequately lubricate the reduction gearbox.

3.3.1.3.4 Bearings and Seals

The same bearings and shaft seals used in the Haulmaster gearbox are intended to be reused in their same positions in the reducer gearbox. Due to the reuse of the shafts from the Haulmaster gearbox, as well as the matching the end cap geometry in the areas surrounding and supporting the shafts to that of the Haulmaster gearbox, the same bearings and shaft seals used in the Haulmaster gearbox will fit into the reducer gearbox. Since the output shaft is now supported closer to the gear on the output end of the shaft, the force on the output shaft bearing on the input side of the gearbox will be less than in the Haulmaster gearbox. Conversely, the output shaft bearing on the output end of the gearbox will now take a higher percentage of the shaft separation force than it did in the Haulmaster gearbox. However, this will still be far less than the force it would be subjected to in the Haulmaster gearbox. In the case of the input shaft, the forces on the bearings will be the same, as the Haulmaster gearbox as the location of components did not change. Thus the bearings from the Haulmaster gearbox should be able to withstand the forces exerted on them with a high factor of safety.

It is expected that the existing bearings and shaft seals will be able to deal with the increased rotational speeds that the reducer gearbox shafts will experience compared to the Haulmaster gearbox shafts. The specific bearing and seal models used were not known but bearings and shaft seals of this size can typically deal with shaft speeds of multiple thousands of rpm which is more than sufficient for this application where the highest shaft speed will be 2000 rpm on the input shaft [12] [13]. However, it is recommended that Elmer's verify that the Haulmaster gearbox's bearing and shaft seals are rated to for these speeds. Should these bearings and seals not meet the necessary shaft speed requirements, other bearings and seals that meet these requirements can easily be selected and made to fit the reduction gearbox with no or minor modifications to case endplates.

3.3.1.4 Fuel System

To allow for continuous operation of the gearbox test stand, a sourced fuel tank was implemented in the project, as shown in Figure 20.

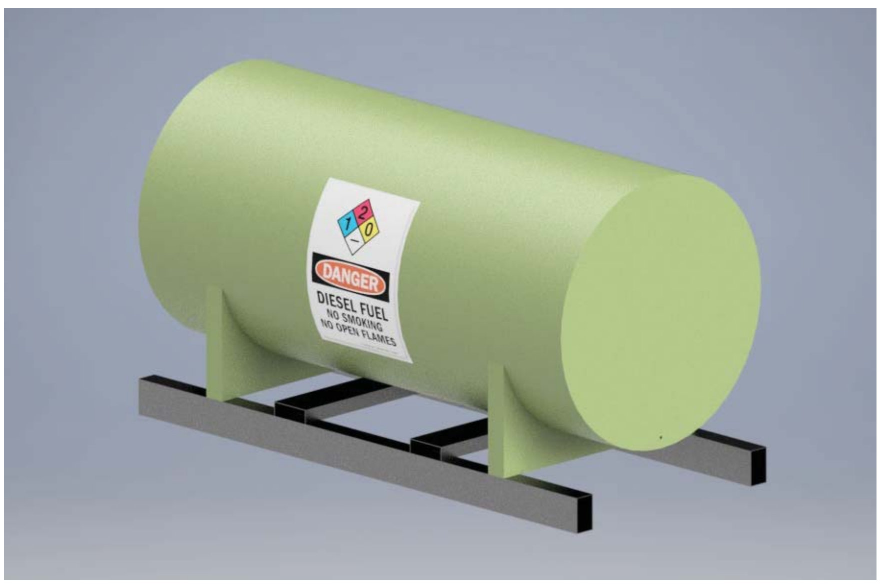


Figure 20. Render of Final Fuel Tank Design.

The scope of its implementation is defined in the scope table shown in TABLE XV.

TABLE XV: SCOPE OF POWER UNIT FUEL SYSTEM

In Scope	Out of Scope
Determining fuel usage under full 600 HP load	Specifying fuel line sizes between the fuel tank and engine
Sourcing of a fuel tank from a local, Manitoban vendor	

To connect the fuel tank to the Deutz diesel power unit, a standard 1.5” diesel fuel line, and NPT fittings were sourced from McMaster. 1.5” NPT fittings were selected as the outlet of the fuel tank uses this size as specified by Westeel. Elmer’s is responsible for creating an adapter between the 1.5” NPT female end of the fuel tank and the connection to the Deutz power unit. Pritchard Dieseltech should be contacted for further information regarding the specific fuel line sizes used by Deutz on the TCD 13.5 diesel engine, as the information was not available to the team.

The fuel consumption data from Deutz was used to estimate the required fuel tank size. The power curve of the diesel engine was consulted to determine the approximate rpm range during operation. This power curve is shown in Figure 21.

Figure 21. Power curve of Deutz TCD 13.5 engine (ISO 9242) [10].

From Figure 21, the blue curve representing the engine tuned to 600 HP, or 450 kW, is the curve being examined. It is evident that to produce 600 HP, the engine must operate from approximately 1700 rpm to its maximum speed at 2100 rpm. Using these operating speeds, the fuel consumption of the engine was estimated using the specific fuel consumption plot in Figure 22.

Figure 22. Specific fuel consumption of Deutz TCD 13.5 engine (ISO 9249) [10].

Using an operating range of 1700 rpm to 2100 rpm, the specific fuel consumption values range from 200 g/kWh to approximately 220 g/kWh. While it is unlikely that the diesel engine will be operating at 100% load for the entirety of its operation, the extreme case of 220 g/kWh was used to calculate the required fuel tank size. From metric 30, the target runtime was set to 10 hours. The following calculations assume a diesel fuel density of 0.832 kg/L [14].

$$Fuel\ Required = \frac{0.220\ kg}{kWh} * \frac{0.7457\ kWh}{HP} * 600\ HP * 10\ hr * \frac{1\ L}{0.832\ kg} = 1183\ L$$

From the calculation, at 100% load, the diesel engine will consume approximately 1183 L of fuel over a period of 10 hours. This value was then used to perform a search for commercially available fuel tanks. Designing a fuel tank was not considered as the guidelines behind storing fuel are highly regulated by the law.

The actual tank capacity was selected to be 1300L from Westeel. When looking at manufacturers for the fuel tank, the team looked at two Manitoban companies for fuel storage solutions. Both Meridian Manufacturing and Westeel offer fuel tanks on skids, with many options for tank sizes and fuel outlet locations on the tank. To remain portable, the main consideration in selecting a fuel tank was to select the smallest tank that was closest to the required capacity of 1183 L. Meridian offers tanks starting at 2300 L, while Westeel offers tanks as small as 1300 L. Therefore, the decision was made to move forward with the selection of a fuel tank from Westeel as the excess capacity from a Meridian tank is unneeded.

Manitoba law governs that any aboveground fuel storage system must have a secondary containment apparatus in the form of a dike or a double-wall fuel tank [15]. A double-wall tank was selected to maintain portability and to avoid the requirement of building a dike around the fuel tank each time the tank moves. The gravity-fed tank was selected over a standard tank which requires a fuel transfer pump to move the fuel through an outlet at the top of the tank. The gravity flow tank's outlet is located at the bottom of the front of the tank, allowing for fuel lines to easily be created.

3.3.1.5 Power Unit Skid

To ensure that the power unit skid design met the expectations of the client, the scope was first defined and is shown in TABLE XVI.

TABLE XVI: SCOPE OF POWER UNIT SKID DESIGN

In Scope	Out of Scope
The initial design of skid	FEA of skid
Center of mass calculation	Detailed analysis of components

As shown in Figure 8, the Deutz engine comes with a skid from the manufacturer. The team decided to modify the existing skid rather than design a new one and waste the existing skid. A render displaying the side view of the entire power unit focusing on the modifications, is shown in Figure 23.

Figure 23. Render of Side View of the Power Unit.

The existing skid channels were not made from a standard channel size. For the extension, the standard size was chosen from Metals Depot to match the height of the existing channels. The lengths of the channels were extended a small amount further than the reduction gearbox, but to the nearest inch as the dimension is not critical, giving a length of 48”. A bracket was designed with four holes to mount these channel extensions to the existing skid. The team did not design the extensions on the assumption that welding is possible since the material of the existing skid is unknown. The plates were designed using 1/4” A36 sheet metal with 9/16” holes. Holes are required to be drilled into the existing skid, as well as into the channel extensions, shown in Figure 24.

Figure 24. Render of Skid Channel Extensions and Attaching Brackets.

No mounts were designed for the clutch since the clutch is mounted directly to the engine. Mounting holes were designed into the endplates of the reduction gearbox, so the appropriate mounting plates were designed by extending from the flat face of the skid channel extensions to these mounting holes. For these components, 1” A36 steel was used to ensure stability and that the mount is able to support the weight and reaction forces of the reduction gearbox. Additionally, these mount plates will act as cross members, supporting the skid transversely, while also allowing the reduction gearbox to be easily removed if necessary. These mount plates were designed to be welded to the channel extensions to complete the skid extension.

Figure 25. Render of Reduction Gearbox Mount Plates.

The consequence of modifying the existing skid is that the fork pockets of the existing skid were located to be under the center of mass of the engine system alone. Therefore, the new center of mass was calculated to determine whether the fork pockets required modification. To calculate the new center of mass, the approximate center of mass for each of the major components were measured.

$$\bar{x} = \frac{4000(42) + 600(84.5) + 200(116)}{4000 + 600 + 200} = 50.4"$$

The new center of mass is 50.4” from the front of the engine skid, compared to the previous 42”. As this 50.4” was above one of the existing fork pockets, a modification was required. A blocking plate was designed out of A36 sheet metal to fit over the existing fork pocket. This plate was not required structurally but instead works as a preventative measure to stop any operators from using the pocket and potentially causing injury. Additionally, a new fork pocket was designed, however, the location of the pocket was restricted by the attaching bracket and gussets on the existing skid. Due to these constraints, the fork pocket was placed between the gusset and the attaching brackets, allowing the center of mass of the entire assembly to be between the fork pockets once again. However, the exact positioning of the fork pocket is not critical and could be subject to change by the client.

FEA of the skid was not performed as the analysis was too complex based on the initial project outline and was therefore deemed out of scope. Therefore, the team moved forward into designing the next major component of the test stand, the gearbox test unit.

3.3.2 Gearbox Test Unit

As shown in Figure 26, within the gearbox test stand, the gearbox test unit houses the mounting system to hold the test gearbox, the load application system in the form of a hydraulic pump dynamometer, and the gearboxes required between the test gearbox output and the hydraulic pump input.

Figure 26. Render of the gearbox test unit.

3.3.2.1 Hydraulic Pump Dynamometer System

From the concept selection process, a hydraulic dynamometer system was chosen to apply a load to Elmer's gearboxes. A hydraulic pump dynamometer is desirable as it is able to operate without any external connections to power grids or waterbodies. To ensure that this module meets the needs of the client and does not exceed the requirements of the project, a scope table was developed. An overview of the sections included and excluded from the scope of this project are listed in TABLE XVII.

TABLE XVII: SCOPE OF HYDRAULIC PUMP DYNAMOMETER DESIGN

In Scope	Out of Scope
Sourcing of pump, load control valve, filter, and oil cooler	Routing case-drain lines from hydraulic pumps
Reservoir design	Specifying hydraulic circuit for cooler fan hydraulic motor

In Scope	Out of Scope
Preliminary FEA for the designed components.	
Specifying hydraulic hose requirements and pressure drops between major components	

To begin the design of the hydraulic dynamometer, the basic circuit components were determined, as shown in Figure 27.

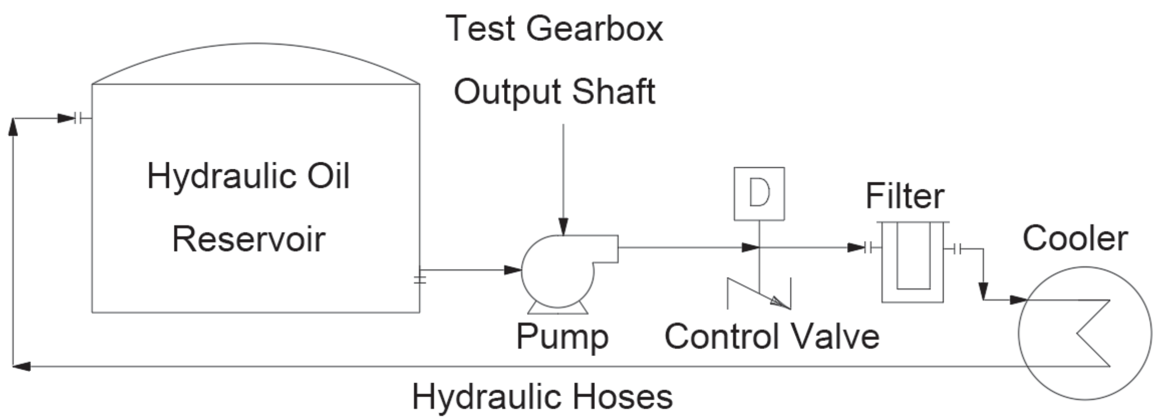


Figure 27. Simple schematic of the hydraulic circuit. Adapted from [16].

The hydraulic dynamometer consists of seven major components as shown below.

1) Hydraulic oil. The working fluid of the hydraulic dynamometer is hydraulic oil, and the type and specifications of the oil used determine the components of the hydraulic circuit to create the dynamometer.
2) Hydraulic pumps. The input shaft of the hydraulic pump is driven by the output shaft of the test gearbox. The hydraulic pump then pumps oil through the hydraulic circuit which creates a resistance load to the test gearbox output. Based on the number of outputs of the test gearbox, up to two hydraulic pumps are required in the dynamometer system. Note: A speed increaser gearbox is required between the test

gearbox output and the pump input to allow the hydraulic pump to operate at its rated speed.

3) Load control valves. The resistive loads applied to the client's gearboxes are provided by restricting the outlets of the hydraulic pumps, through both pressure and flow control. Restricting the pump outlet increases the pressure of the system, requiring more horsepower to overcome the restriction.
4) Filters. Contaminants within the hydraulic circuit affect the performance of the hydraulic pump and can cause damage to the system. Therefore, filters are installed in the system to remove the contaminations in the oil flow.
5) Oil cooler. The act of pumping oil and forcing the oil through a restriction generates heat. The cooler removes the generated heat from the hydraulic oil to keep the system within acceptable operating temperatures.
6) Oil reservoir. The oil reservoir contains an essential amount of hydraulic oil for the dynamometer system. Along with storing the oil for the hydraulic system the reservoir allows for any entrained air within the oil to be eliminated, enhancing the system performance. The reservoir also permits for minor heat dissipation from the oil through the reservoir walls, and for any solid contaminants to settle out of the oil.
7) Hydraulic hoses. Hoses are used to contain the flow of oil between components of the hydraulic circuit and must be rated for the appropriate pressures and flows in each section of the hydraulic circuit.

It is noted that the circuit depicted in Figure 27 only shows the components used to test one output of the gearbox. Duplicate components, such as the pump and control valves, are required to test two outputs of a gearbox. The design and specification of the hydraulic dynamometer components and the reasoning behind duplicating components are explained in the following subsections.

3.3.2.1.1 Hydraulic Oil

Hydraulic oil is the heart of the dynamometer. As the working fluid, the oil must be able to operate in a variety of conditions to allow for the testing of Elmer's gearboxes. When selecting an oil for the dynamometer, the team contacted Elmer's to determine which oil is used in the equipment they manufacture. Currently, Elmer's uses AGRIPRO Universal Tractor Hydraulic Fluid, UTHF, purchased from Oil Mart in Winnipeg. Oil Mart was contacted by the team for pricing per 1040 L tote of oil, where a value of $2797.60 per tote was provided [17]. Properties of the UTHF oil are shown in TABLE XVIII.

TABLE XVIII: PROPERTIES OF HYDRAULIC OIL USED BY ELMER'S [17].

Specification	Value
Product Number	07-120700
SDS Number	S059.1
Viscosity @ 100°C, cSt	9.18
Viscosity @ 40°C, cSt	56.93
Viscosity Index	142
API Gravity at 60°F	30.0
Pour Point, °C	-48
Brookfield @ -35°C, cP	43,650
Color ASTM	L3.0

From TABLE XVIII, the main specification that is used for pump selection is the viscosity of the oil, which ranges from 9.18 cSt to 56.93 cSt. The viscosity of the fluid mainly affects the selection of the hydraulic pump, which is covered in Section 3.3.2.1.2. However, the viscosity range of the hydraulic oil is stereotypical of most mineral oils, therefore most hydraulic pumps are able to be selected for the hydraulic circuit.

3.3.2.1.2 Hydraulic Pump Selection

Within the gearbox test unit, two hydraulic pumps are required to test the Haulmaster gearbox. However, as both outputs of the Haulmaster gearbox must be able to be tested with loads ranging from 0-600 HP, the same model of pump is used for both gearbox outputs. Two variations of the Kawasaki K3VG-280DT hydraulic pump are used for the gearbox test stand project, one version with an auxiliary gear pump and one without. Both pumps are equipped with electronic power and pressure control. Renders of the two hydraulic pumps are shown in Figure 28 and Figure 29 [18].

Figure 28. Render of K3VG-280DT pump with auxiliary pump.

Figure 29. Render of K3VG-280DT pump.

The operating parameters for the pump were determined using the following equation:

$$Power\ [HP] = \frac{\Delta p[psi] * Q[gpm]}{1714}$$

Within this equation, Δp is the pressure difference across the pump, and Q is the flow rate through the pump. As both the pressure difference and flow can be controlled based on which type of pump control is selected, the system pressure was set to a constant value to simulate the maximum operating conditions. By assuming the extreme operating scenario of one pump resisting all 600 HP from the diesel engine through only one output of the gearbox, the team calculated the maximum required flow through one loop of the hydraulic circuit. This rearranged the power formula to provide the required flow rate which is shown in the following formula:

$$Q[gpm] = \frac{Power\ [HP] * 1714}{\Delta p[psi]}$$

The system pressure was initially set to a standard hydraulic system operating pressure of 3000 psi. This led to a maximum system flow rate of 342.8 gpm, which is a relatively high value when compared to hydraulic circuits in industry. A brief search of pump manufacturers including Parker, Danfoss, and others was conducted, where insufficient pump flows were found at a pressure of 3000 psi. The client was then contacted to discuss options, where it was suggested that the design team look into 5000 psi systems. The client also suggested looking into Eaton and Kawasaki for large displacement hydraulic pumps. When using a 5000 psi system, the maximum required flow rate is reduced to 205.68 gpm.

After exploring all options through both Eaton and Kawasaki, one commercially available pump from Kawasaki met the requirements for the project, being a 5000+ psi system capable of flows over 200 gpm. The K3VG-280DT series of pumps from Kawasaki uses two 280 cc/rev swashplate axial piston pumps mounted in tandem for a total of 560 cc/rev, with multiple flows and pressure control options [19]. The general specifications of the pump are summarized in TABLE XIX.

TABLE XIX: K3VG-280DT GENERAL SPECIFICATIONS [19]

Specification	Value
Rated Speed [rpm]	1500
Rated Pressure [psi]	5075
Displacement [cc/rev]	560
Max. Flow @ Rated Speed [gpm]	206.054
Rated Power [HP]	684
Viscosity Range [cSt]	10-200
Weight [lb]	660

As the K3VG-280DT pump from Kawasaki was the only pump found to meet the requirements of the hydraulic circuit, it was selected for the project.

The selected pump is configurable in multiple aspects including flow control, power control, flow direction, mounting brackets, power control, and auxiliary pump options. Elmer’s indicated they would be implementing the control system through a programmable logic controller (PLC), therefore electronic pump controls are crucial to the operation of the gearbox test stand. The Kawasaki pump has the option of electronic control for both power and displacement of the hydraulic pump, allowing for complete control over the pump’s operation.

Through the selection of both power and displacement control within the pump, the hydraulic circuit for each dynamometer can apply 0-684 HP load, varying infinitely between 0 and 684 HP due to the ability to change the angle of the swashplates within the pump. By running the pumps with no angle on the swashplates, the pump is effectively operating at 0 cc/rev. Zero displacement translates to zero horsepower produced. By increasing the angle of the swashplate, the effective displacement of the pump is increased, therefore, creating power within the hydraulic system. Increasing the swashplate angle to the maximum setting achieves the desired output of 600+ HP.

In addition, the combination of electronic power and displacement control allows for the flow rates of each pump to be specified based on the load. As the combined outputs of the tested gearbox cannot exceed 600 HP, which is limited by the diesel engine, the total flow rate of the system can be said to have a maximum value of 206 gpm. Regardless of which pump is providing the oil flow, the system cannot achieve a flow rate of over 206 gpm. This flow rate is used for the selection of the remaining components in the hydraulic circuit.

The rotation of the pump from Kawasaki is configurable to be either clockwise or counter-clockwise when looking at the input shaft end of the pump. Based on the rotations of the diesel engine and all other gearboxes in the system, both selected pumps require a clockwise rotation. When considering the option of auxiliary gear pumps, only one of the selected

Kawasaki pumps is required to be equipped with an auxiliary gear pump. This auxiliary pump is used to power the hydraulic fan motor on the oil cooler, creating a hydraulic circuit independent of the main load application circuit.

As with any hydraulic system, inefficiencies are introduced with a pumping operation. The selected pump for the hydraulic dynamometer is no different, with efficiencies ranging from 0% efficiency near low displacement settings, to 91% as the peak efficiency. At any operating speed, the power that is not transferred through the system is turned to heat within the pump. Calculating the efficiency and heat produced in the system does not provide any beneficial data, as the efficiency calculated is for one scenario only. Changing the pump displacement or pressure level then requires a new calculation. However, the efficiency at any given time within the hydraulic pump is not important as power lost is still being absorbed by the hydraulic circuit and a load is still being applied to the test gearbox.

Rather than transmitting the power through the system in the form of pressurized fluid flow, the lost load is converted to heat within the pump. In the case of a hydraulic dynamometer, a lower efficiency pump is desired as a lower efficiency pump lowers the oil flow rate through the hydraulic circuit, reducing the size of the required components in the hydraulic circuit. It is noted that with a hydraulic system operating with low efficiencies, an oil cooler must be implemented to prevent the overheating of the hydraulic oil.

3.3.2.1.3 Load Control Valve Selection

To induce a load on the test gearbox, a hydraulic control valve is placed after the pump. A render of the final control valve assembly is shown in Figure 30.

Figure 30. Render of the final control valve assembly [20] [21] [22].

The control valve assembly operates by restricting the flow of oil through two electro proportional relief valves mounted in parallel, therefore, creating a pressure drop over the valves, resulting in a load being applied to the hydraulic pump. If the pump is under a higher load, the tested gearbox is also under a higher load. The complete control valve assembly relies on the input oil flow from the pump to be split into two flows using a tee-body manifold, where each flow is then routed through the aforementioned relief valves. The oil flows through the two control valves are then combined back into one flow in another tee-body manifold, where the flow can then move to the filters.

The control valve is electronically controlled to allow Elmer's to implement their PLC for dynamic loading through pump control and valve control. An electro proportional pressure relief valve was selected as the flow in the hydraulic pump dynamometer is only in one direction. Using a pressure relief valve ensures that the pressure upstream of the hydraulic valve is maintained at the desired pressure level. The ability to set the relief pressure of the valve allows Elmer's to set the load on the test gearbox through the pump.

The type of electro proportional pressure relief valve was determined by the team and through consulting Elmer's. The client recommended using cartridge valves, which are then mounted in a manifold. Common hydraulic component vendors such as Parker Hannifin, Eaton, Hydac, and Hydraforce were consulted to find a solution. However, most options from the aforementioned suppliers were limited by flow rates of under 100 gpm, with limited valves allowing for 160 gpm. Running two relief valves in parallel allows for flow rates surpassing 206 gpm.

Elmer's recommended that the team look at using components from Sun Hydraulics as they had previous experience with the vendor. Together, the team and client selected components from Sun Hydraulics where acceptable valves were found by mounting the valves in parallel. The two highest-capacity cartridge pressure relief valves manufactured by Sun Hydraulics are rated for 5000 psi systems with flow ratings of 100 gpm and 200 gpm. Therefore, to achieve a flow rate of 206 gpm, mounting two valves in parallel was the only option. The team selected two 200 gpm valves to be mounted in parallel, model number RPKC8WV [22]. Two 200 gpm valves are used to simplify Elmer's implementation of their PLC; rather than use a 200 gpm valve and a different valve with a lower flow rating (example of 200 gpm valve and a 25 gpm valve to achieve 225 gpm capacity), The selected relief valve requires an additional pilot control valve which Sun Hydraulics gives one option for, being model number RBAPXWV [22].

As the relief valves are running in parallel, the flow from the pump must be split into two parallel flows, one flow into each manifold and cartridge valve. A tee-body junction was selected with the size determined by the outlet size of the pump. The K3VG-280DT pump uses a 2" outlet with standard SAE 4-bolt flanged connections. Therefore, a tee-body junction with a 2" SAE 4-bolt inlet was selected, being the WAI/S from Sun Hydraulics. This junction splits the flow into two 1.5" outlets with SAE 4-bolt flanged connections. These 1.5" connections then determined the manifolds used to house the cartridge valves.

Manifolds for the cartridge valve were selected using Sun Hydraulics' manifold configurator. Simple 90° manifolds with gage ports were selected, as they were the only manifolds available without internal passthrough which allows the oil to flow through the manifold, bypassing the relief valve. These manifolds use 1.5" SAE 4-bolt flanges for the inlet and outlet [21].

An additional tee-body junction was implemented to bring the two flows through the valve back into one flow. To do this, the WAI/S tee-body junction was used [20]. To connect all of the components to one another, the client recommended that the team used fittings and adapters from Sealum Industries. From Sealum Industries, 1.5” SAE Code 62 O-ring unions were selected to connect components together [23]. SAE Code 62 components were selected over Code 61 as the pressure ratings for the Code 61 components is only 3000 psi, whereas Code 62 components are rated for 6000 psi. Along with the selection of the 1.5” O-ring unions, Code 62 split-flanges were selected to attach the unions to the tee-body junctions and valve manifolds. A simple sketch of the load control system is shown in Figure 31.

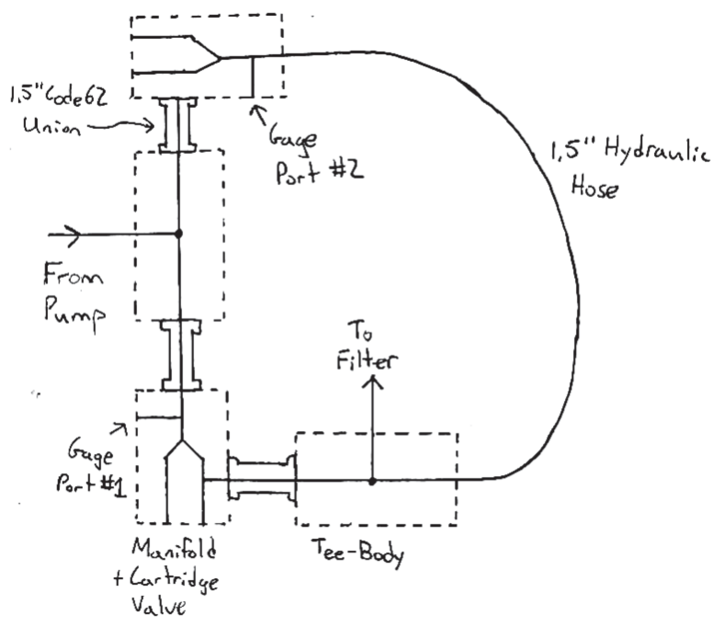


Figure 31. Simple sketch of the load control valve layout for the hydraulic circuit.

Within Figure 31, two gage ports are visible in each valve manifold. By placing the manifolds in opposing directions, the side port of the manifold is the inlet of one valve and the outlet of another. Similarly, the bottom port on one block is the inlet while the other is the outlet. By positioning the manifolds in this way, the gage ports in the two manifolds are placed allowing for pressure readings both before and after the valves. As the valves are run in parallel, Gage Port #1 is sufficient to read the pressure before the valves. Likewise, Gage Port #2 reads the pressure after the valves. Placing gage ports before and after the valves allows the pressure drop over the valves to be read, which is critical information for Elmer's when implementing a PLC for the valve control.

It is noted that most of the components are connected using the 1.5" Code 62 union from Sealum Industries. However, due to the different mounting of the cartridge valve manifolds, and the union being available at only one length, it is not possible to use unions and elbows from Sealum Industries to complete the control valve circuit. Therefore, a hydraulic hose is required to connect one of the manifolds to the tee-body joining the two oil flows into one. This hydraulic hose is also shown in the aforementioned Figure 31.

Two of the circuits shown in Figure 31 are required for the variable control of the load applied to each test gearbox output. Using one circuit allows for the application of a variable load to the test gearbox, however, the load would be applied equally to each output. By separating the controls for the oil flows from each pump on the test gearbox, the load to each output of the test gearbox is then variable. Therefore, two of the control valve circuits are required for the correct operation of the gearbox test stand. A summary of the components required to complete one of the control valve circuits is shown in TABLE XX.

TABLE XX: HYDRAULIC LOAD CONTROL COMPONENTS REQUIRED PER CONTROL VALVE CIRCUIT AND THE COMPONENT SPECIFICATIONS

Component	Vendor	Part Number	Quantity
Electro Proportional Relief Valve [24]	Sun Hydraulics	RPKC8WV	2
Pilot Control [25]	Sun Hydraulics	RBAPXWV	2
Tee-Body Junction [26]	Sun Hydraulics	WAI/S	2
Valve Manifold [27]	Sun Hydraulics	LB6/S	2

Component	Vendor	Part Number	Quantity
1.5" SAE Code 62 O-Ring Union [28]	Sealum Industries	1959-24-24	3
1.5" SAE 4-Bolt Split-Flange Kit [29]	Sealum Industries	1825-24	8
1.5" Hydraulic Hose w/Straight SAE Flange Ends (3ft long)	Vendor of Choice	TBD	1

After determining the components of each control valve circuit, the components of the circuit were modeled in Inventor, where the mount plate was designed and is shown in Figure 32.

Figure 32. Mounting plate to hold cartridge valves and manifolds.

The plate was designed by placing the components on a flat 3/8" sheet metal mounting plate for a general alignment. Holes were added to the sheet metal mounting plate to allow for bolts to secure the components to the mounting plate. Additionally, the mount plate was designed with a flange with clearance holes for 5/8” bolts to allow for the mounting plate to be secured to a skid.

3.3.2.1.4 Filter Selection

A W041 inline cartridge filter from Donaldson Filters, as shown in Figure 33, is selected as the oil filters for the dynamometer system [30]. The main purpose of having filters in the hydraulic pump dynamometer system is to trap the contaminations and keep the system clean.

Figure 33. W041 inline cartridge filter from Donaldson [30].

In this project, the selection of the filter contained three major considerations: the position of the filters in the hydraulic circuit, the operating pressure of the filter, and the operating flow rate passing through the filter.

The filter was designed to be placed between the control valve and the cooler, with one filter per hydraulic pump circuit. Initially, the team planned to install a filter at the suction side of each hydraulic pump to keep any contaminants from reaching the pump. However, after discussing options with Elmer’s, it was suggested that any filters should be installed on the return line after the hydraulic pump and before the coolers, rather than on the suction line before the hydraulic pumps. As the filter traps contaminations, the pressure drop through the filter increases, which reduces the suction pressure of the hydraulic pump and affects the performance of the pump. Similarly, the filter should be installed before the cooler. If the filter is clogged and it is installed after the cooler, the oil pressure in the cooler builds and could exceed the burst pressure of the cooler.

The maximum operating pressure of the selected filters should always be higher than the maximum pressure at the location where the filters are installed in the system to keep the system safe. The pressure at the filter is the same as the sum of the resistance in the hydraulic hose from the filter to the cooler, the resistance within the cooler, and the resistance in the hose from the cooler to the hydraulic oil reservoir. After the calculation in Section 3.3.2.1.7, it was determined that the maximum operating pressure of the filter is 13.45 psi.

Different types of filters have their own flow rate capacity, therefore, the maximum flow rate through the filter should be considered as well. As mentioned in Section 3.3.2.1.2, the maximum flow rate of the hydraulic pump is 206 gpm, requiring the flow rate capacity of the filter to be more than 206 gpm. Considering the pressure rating and the flow rate capacity, the W041 inline cartridge filter from Donaldson was chosen. The specifications of the filter are shown in TABLE XXI.

TABLE XXI: HYDRAULIC OIL FILTER SPECIFICATIONS

Component	Vendor	Part Number	Maximum Pressure (psi)	Maximum Flow Rate (gpm)	Port Size	Quantity
Inline Cartridge Filters	Donaldson	W041	500	300	2" SAE 4-Bolt Flange Code 61	2

3.3.2.1.5 Oil Cooler Selection

The act of forcing oil through a restriction creates heat and usually requires the use of an oil cooler to prevent the overheating of the hydraulic oil. In a normal hydraulic circuit, most of the input power is used by actuators, like cylinders and motors. In the case of the hydraulic dynamometer, all of the power produced by the pump will eventually be converted to heat as no work is being done by the hydraulic circuit. As the hydraulic system will be used as a load application for loads up to 600 HP, the hydraulic circuit must be able to dissipate heat and a cooler must be implemented. The cooler used in the hydraulic dynamometer system is shown rendered in Figure 34.

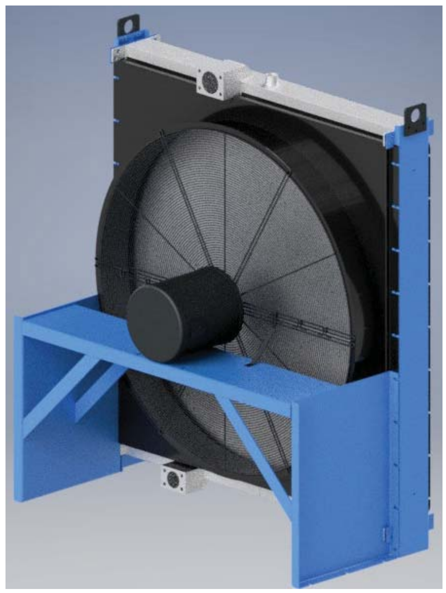


Figure 34. Render of oil cooler used in the hydraulic dynamometer system [31].

The cooler selected for the hydraulic dynamometer is a forced-air cooler from Thermal Transfer Products. The cooler is rated for flows of up to 500 gpm and with a maximum heat dissipation capacity of 707 HP [31].

The cooler was selected by determining the required heat dissipation capability of the cooler. By assuming perfect power transfer through the gearbox test stand, from the engine through the pumps, a cooling system can be designed for a “worst-case-scenario”. Under these conditions, it is assumed that the maximum heat transfer requirement for the circuit is to dissipate 600 HP, which represents 100% of the input power being converted to heat. This allows the team to assume a flow rate of approximately 206 gpm and a required heat dissipation of 600 HP.

By conducting an online search for industrial oil coolers, preferably forced-air coolers to maintain simplicity, the team found that Thermal Transfer Products manufactures high-flow capacity and high-power dissipation forced-air coolers. A cooler for the project was selected using the scenario of requiring 600 HP of heat dissipation and the performance curve shown in Figure 35.

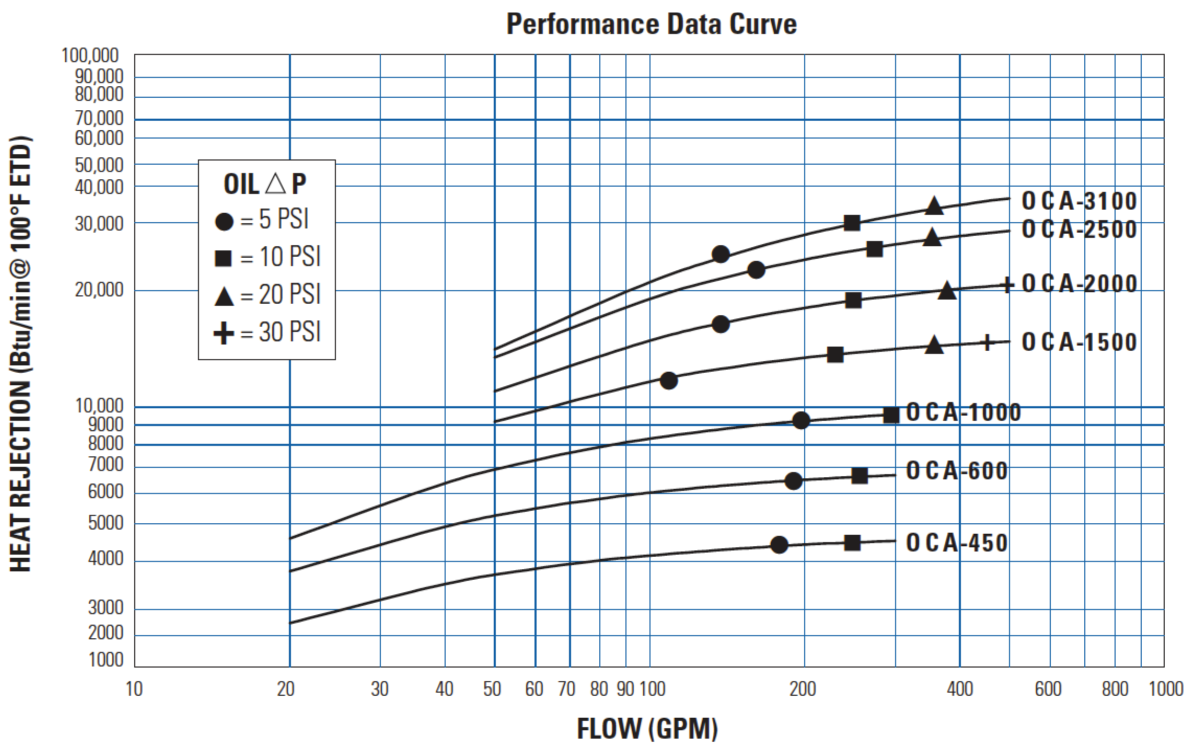


Figure 35. Performance curves for OCA-series forced-air heat exchanger [31].

From this graph, a cooler was selected by calculating the heat dissipation requirements from the applied load to BTU/min, as shown next [31]:

$$\frac{BTU}{min} = HP * 42.4 = 600\ HP * 42.4 = 25{,}440\ BTU/min$$

From Figure 35, the OCA-2500 meets the requirements of dissipating 25,440 BTU/min at 206 gpm, as does the OCA-3100. Elmer’s was presented the information on the coolers where they selected the largest cooler available, the OCA-3100. The OCA-3100 was selected as the cooler for the gearbox test stand since Elmer’s did not want to risk overheating the hydraulic oil.

To maintain the requirement for no external dependencies, a hydraulic motor was selected over an electric motor to operate the fan. Using a hydraulic motor is an ideal choice as the hydraulic pumps selected in Section 3.3.2.1.2 have the option for an auxiliary pump to be mounted in tandem. The requirements for the hydraulic motor are a flow rate of 12.8 gpm, as stated by the manufacturer [31]. The auxiliary pump for the K3VG-280DT pump has a displacement of 32.5 cc/rev. At the standard operating speed of 1500 rpm, the flow capacity of the auxiliary gear pump translates to 12.89 gpm, meeting the requirements of the hydraulic fan motor for the cooler. Therefore, a hydraulic circuit dedicated to the operation of the cooling fan motor must be implemented on one of the hydraulic pumps used in the dynamometer.

Due to the scope constraints and limited information on the specifications of the hydraulic motor on the cooler, the exact specifics of this hydraulic loop are not defined. It is recommended that the client request more information from the oil cooler vendor to determine the requirements of the oil cooler fan motor hydraulic circuit. For the purposes of this report, the fan motor circuit is shown as a closed-loop hydraulic circuit. The team recognizes that ideally the return line from the hydraulic motor should be integrated into the hydraulic reservoir with a suction line going from the reservoir to the auxiliary pump, however, without the specifics about the fan motor, an accurate design of the auxiliary hydraulic circuit could not be completed.

3.3.2.1.6 Oil Reservoir Design

The oil reservoir was designed to contain enough hydraulic oil for the hydraulic pump dynamometer operation. Although two hydraulic pump dynamometers are required for testing the Haulmaster gearbox, one shared hydraulic oil tank was designed to satisfy the requirements of the hydraulic circuit. Figure 36 shows the CAD model of the finalized hydraulic oil tank.

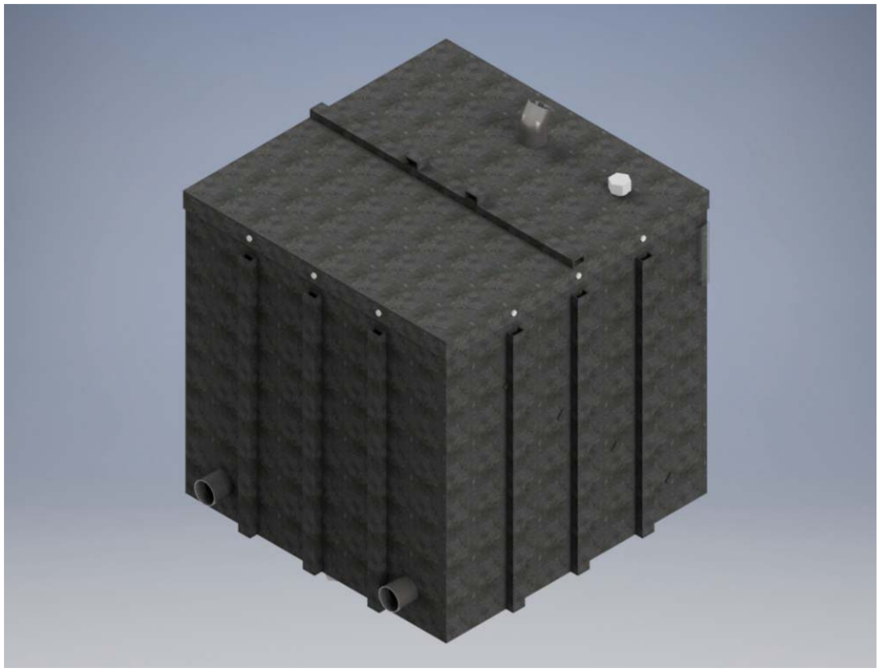


Figure 36. Render of the final hydraulic oil tank CAD model.

As shown in Figure 37, a typical hydraulic oil reservoir should contain the following sub-components:

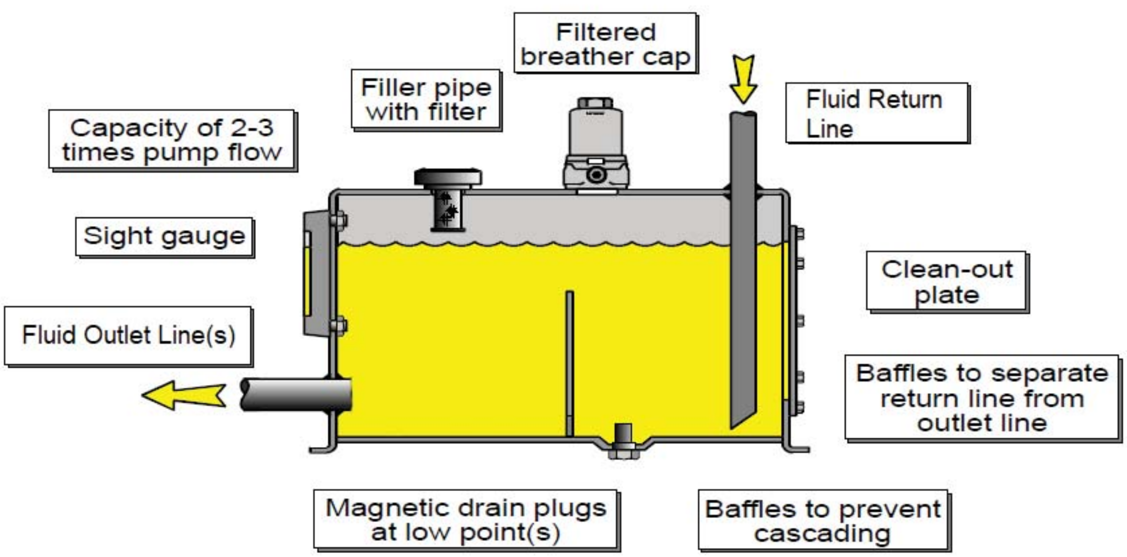


Figure 37. Essential components for a hydraulic reservoir. Adapted from [32].

From Figure 37, the essential components of a hydraulic reservoir can be summarized as:

1) Tank body which has the volume capacity of 2-3 times the pump flow.
2) A fluid return line which leads the hydraulic oil to return to the reservoir.
3) A fluid outlet line where the hydraulic oil exits the reservoir.
4) A sight gauge, which indicates the oil level in the reservoir.
5) A filler pipe with a filter, which allows the operators to fill the oil to the reservoir.
6) A clean-out plate, which allows the operators to access the inner part of the reservoir for repair and cleaning.
7) A baffle plate which separates the fluid return line from the fluid outlet line.
8) A magnetic drain plug at a low point, which collects the metal contaminations in the reservoir and allows the operators to drain the oil from the reservoir.

The following subsections cover the design and sourcing of the aforementioned hydraulic reservoir components.

3.3.2.1.6.1 Hydraulic Oil Reservoir Body

As shown in Figure 38, the main body of the hydraulic oil reservoir is a cubic tank which is made of A36 sheet steel. The inner volume of the tank is 4.5’x4.5’x4.5’ and the thickness of the tank wall is 0.25”. Three 54” long 2.5”x1.5”x0.25” rectangular tubes on the top plate and the side plates strengthen the structure of the tank body. The following subsections detail the design and optimization process of the tank body.

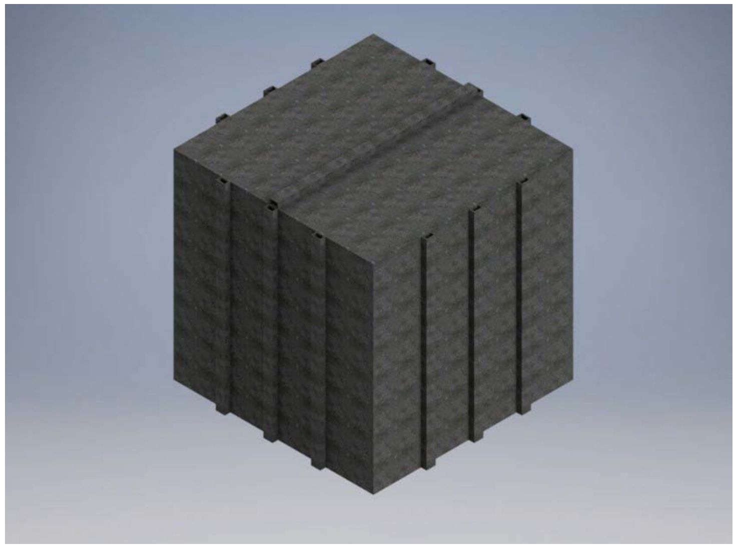


Figure 38. Render of the designed hydraulic oil tank body CAD model.

The size of the hydraulic oil tank body is related to the total flow rate through the system [33]. As mentioned in Section 3.3.2.1.4, the maximum flow rate in the hydraulic pump dynamometer system is 206 gpm. The calculated reservoir volume range for the operation of one hydraulic dynamometer is approximately 2 to 3 times the flow rate, which is 412 gallons to 618 gallons. As the system will primarily convert work into heat, the higher capacity oil reservoir of 618 gallons was selected. The hydraulic oil reservoir should also contain additional space, equals to at least 10% of its fluid capacity, allowing for thermal expansion of the fluid as well as the gravity drain-back during shutdown. As a result, the oil reservoir size can be calculated as shown next.

$$V_{tank} = Q_{flow} \cdot 3 \cdot 1.1 = 680\ gallons = 2574\ L$$

Since an oil cooler is installed in the system, the tank does not need to have the maximum surface area to cool the oil down. A cubic shape tank design is sufficient and was approved by Elmer’s. The length, width, and height of the tank (ignoring the thickness of the wall) are calculated using the following equation:

$$L_{tank}, W_{tank}, H_{tank} = \sqrt[3]{2574L} = 137.05cm \approx 4.50\ ft$$

The thickness of the tank wall can be distinguished by the mass analysis and the stress analysis. As mentioned in Section 1.4.3, the total mass of the unit skid cannot exceed 9000 lbs, which is about 4082.3 kg. Therefore, the total mass of the tank itself cannot weigh over 4082.3 kg. If the tank wall is t meters thick, the mass of the tank can be calculated as:

$$M_{Tank} = \rho_{A36\ Metal} \cdot V = 7850 \frac{kg}{m^3} \cdot [(1.37m + 2t)^3 - (1.37m)^3]$$

For this maximum condition, the thickness was determined to be 0.04337 m. The plot of the M_{Tank} can be generated by using Google online plotter, as shown in Figure 39. As shown in the plot, M_{Tank} increases as t increases. Therefore, the wall thickness t should be less than 0.04337 m (1.71”) to ensure the designed tank does not exceed 9000 lbs.

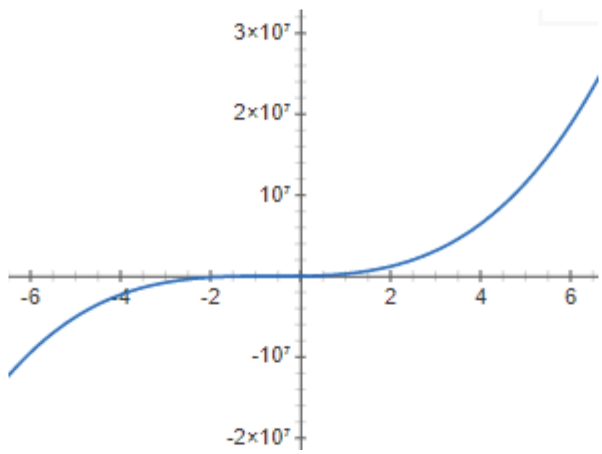


Figure 39. Function plot of the total mass of the tank.

Additionally, the wall should be thick enough to withstand a horizontal pressure from the depth of the liquid and a vertical pressure from the weight of the tank.

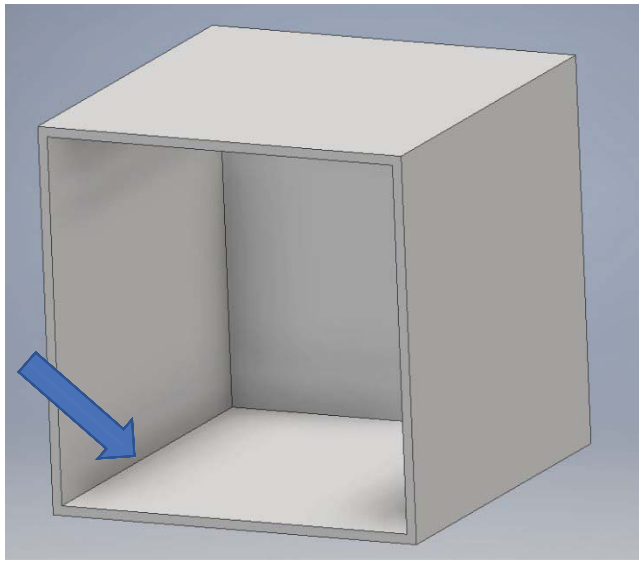


Figure 40. Open view of a 4.5 ft size cubic tank with the maximum stress location noted.

As shown in Figure 40, the maximum liquid stress can be estimated as:

$$\sigma_{liquid} = \rho_{liquid} \cdot g \cdot H_{tank} = 876.2\frac{kg}{m^3} \cdot \frac{9.81m^2}{s} \cdot 1.37m = 11{,}775.86pa = 1.71psi$$

Wall thickness is assumed to be uniform with a thickness of t meters. The total weight of the upper structure, which equals the total mass minus the mass of the bottom plate, can be estimated as:

$$W_{upper} = \rho_{A36\ Metal} \cdot V \cdot g$$
$$= 7850\frac{kg}{m^3} \cdot [(1.37m + 2t)^2 * (1.37m + t) - (1.37m)^3] \cdot \frac{9.81m^2}{s}$$

The stress due to the upper structure's weight can be estimated as

$$\sigma_{weight} = \frac{W_{upper}}{A} = \frac{W_{upper}}{(1.37 + 2t)^2 - 1.37^2}$$
$$= \frac{7850 \cdot [(1.37 + 2t)^2 * (1.37 + t) - (1.37)^3] \cdot 9.81}{(1.37 + 2t)^2 - 1.37^2}$$

A function plot of σ_{weight} is generated by using MATLAB, and is shown in Figure 41.

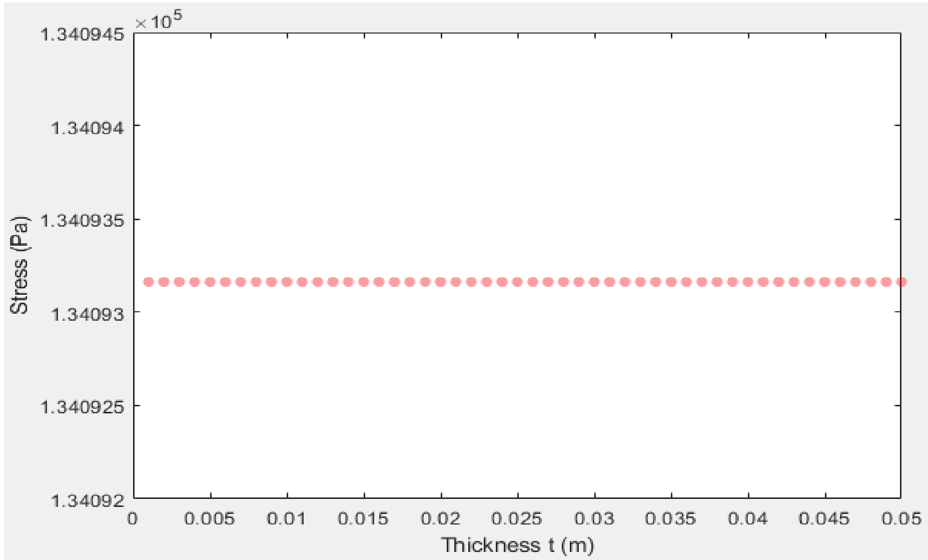


Figure 41. Function plot of σ_{weight} by using MATLAB.

The plot indicates that the σ_{weight} is about 134000 Pa, when $0 < \mathrm{t} \leq 0.04337\ \mathrm{m}$. When t=0.04337 m (1.71”):

$$\sigma_{weight} = 1.34 \cdot 105\ pa = 19.44\ psi$$

Therefore, the maximum total force applied on the oil tank wall, when $0 < \mathrm{t} \leq 0.04337\ \mathrm{m}\ (1.71")$, is calculated as shown next:

$$\sigma_{max} = \sqrt{(\sigma_{liquid})^2 + (\sigma_{weight})^2} = 19.52\ psi$$

The calculated maximum stress is significantly below the yield strength (36,000 psi) and the shear modules (11500 ksi) of the A36 steel. Therefore, the wall thickness of the tank must be less than or equal to 1.71”.

After a number of discussions with the client, the 0.25” thick A36 sheet metal was chosen for building the hydraulic oil tank. A preliminary FEA was performed to determine the maximum expected stress and maximum expected displacement in the 0.25” thick tank when it is completely filled with fluid. Since Inventor FEA does not have an option to apply a variable load across the face of a model, the tank is divided into 10 equal layers. A pressure load of 1.71 psi is applied to the bottom layer of the tank to simulate the maximum fluid capacity scenario. The loading condition is shown in Figure 42. The pressure applied to successively higher layers decreases in increments of 0.171 psi. The bottom face of the tank is fixed during the study.

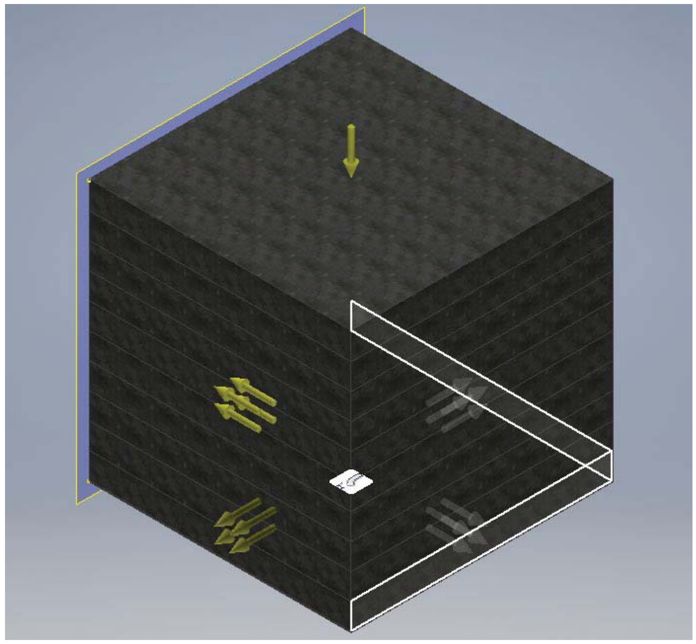


Figure 42. The loading condition for the bottom layer.

As shown in Figure 43, under the applied load, the value of maximum stress in the tank is 15.19 ksi, which is significantly below the yield stress of the A36 sheet metal (36 ksi). As shown in Figure 43, under the applied load, the converged number of the maximum displacement in the tank is 0.2782” (7.06 mm), which is not acceptable to the client. Therefore, the deformation of the tank should be decreased by using proper methods.

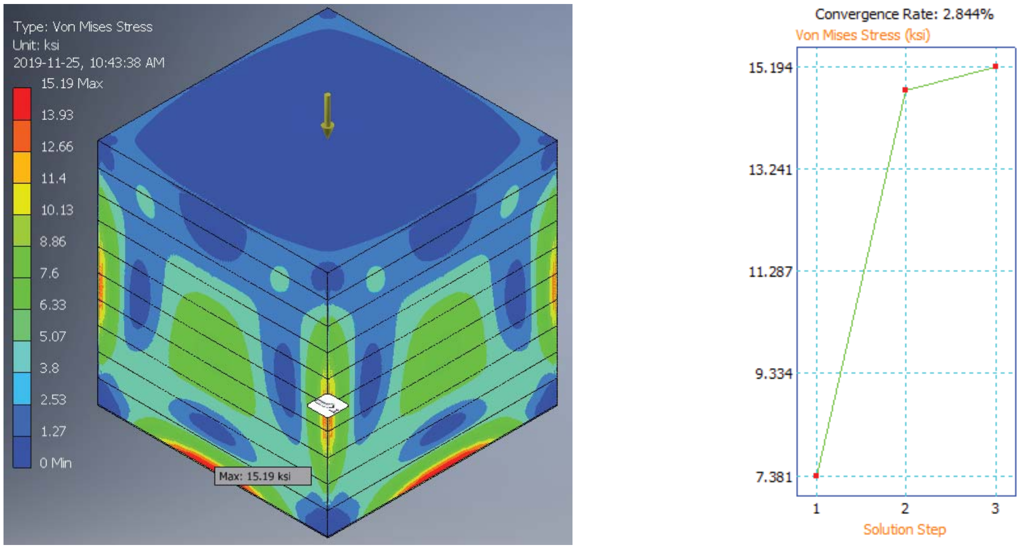


Figure 43. Results of preliminary FEA for the 0.25" thick tank under gravity and fluid pressure.

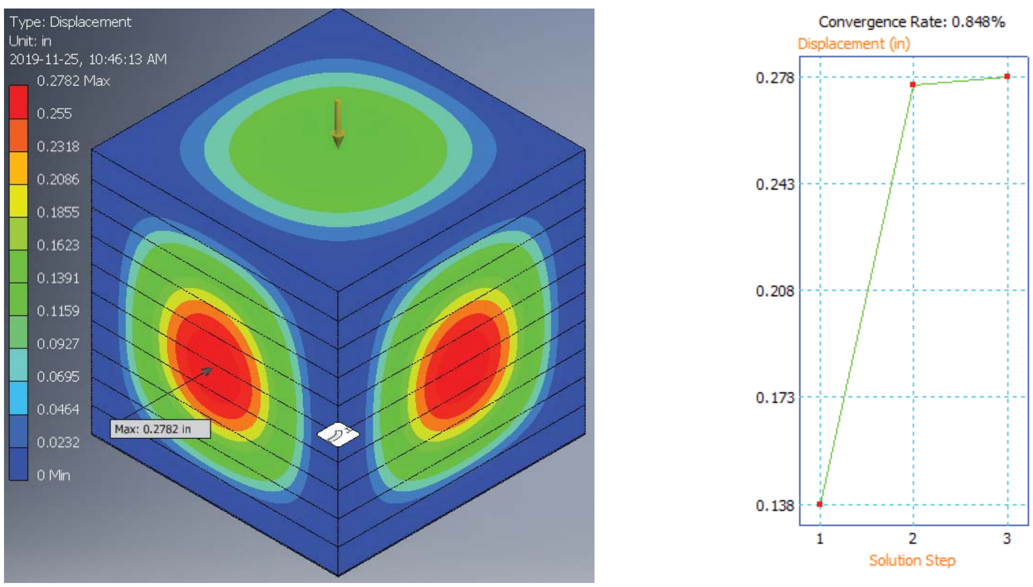


Figure 44. Displacement distribution of the 0.25" thick tank under gravity and fluid pressure.

As indicated in Figure 44, the deformations mainly occur at the centers of the side plates and the top plate. The deformation can be decreased by increasing the bending strength of these plates. One of the simplest ways is through welding steel tubes on these plates.

As shown in Figure 45 and Figure 46, the maximum stress and the deformation of the tank decreases, after installing the long steel tubes on the top plate and the side plates. With these tubes, the maximum stress decreases from 15.19 ksi to 5.647 ksi and the maximum deformation decreases from 0.2782” to 0.01374”. The magnitude of deformation, which is 0.01374” (0.35 mm), is accepted by the client.

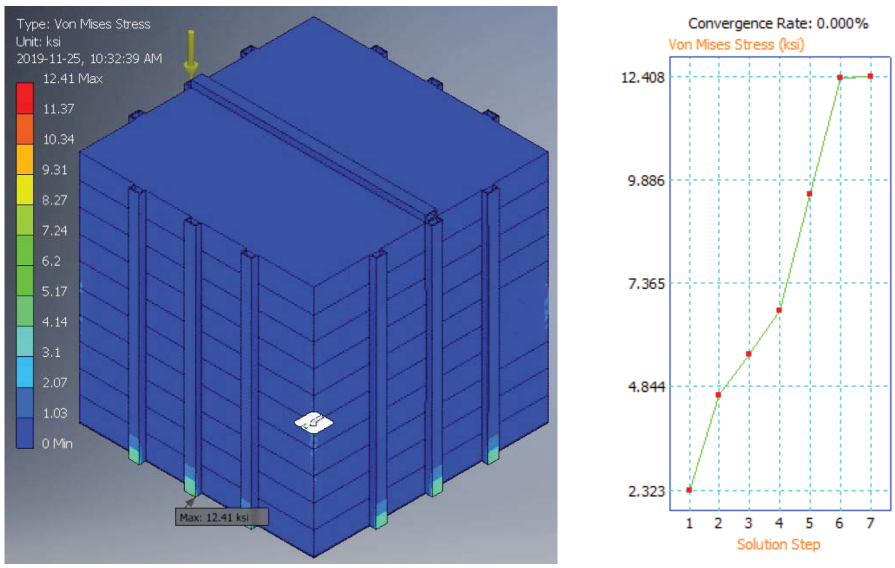


Figure 45. Stress distribution of the modified 0.25" thick tank under gravity and fluid pressure

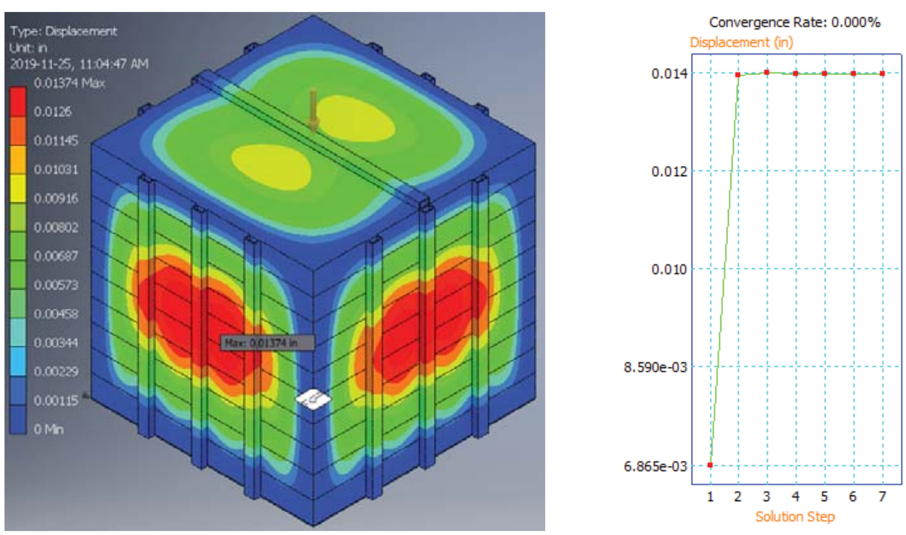


Figure 46. Displacement distribution of the modified 0.25" thick tank under gravity and fluid pressure.

The redesigned tank body is strong enough to stand the gravity force and the oil pressure during the gearbox test. A summary of the components required to complete a full assembly of the hydraulic oil tank assembly is shown in TABLE XXII.

TABLE XXII: COMPONENTS REQUIRED PER HYDRAULIC OIL TANK ASSEMBLY AND THE COMPONENT SPECIFICATIONS

Component	Vendor	Part Number	Size	Quantity
A36 Sheet Metal 0.25” Thick [34]	Metals Depot	P114	60”x120”	3
A500 Rectangle Steel Tube 2.5”x1.5”x0.25” [35]	Metals Depot	T121211214	10’	6

3.3.2.1.6.2 Fluid Return Line Assembly

The fluid return line assembly leads the process fluid going back to the fluid reservoir from the cooler as shown in Figure 47. The fluid return line consists of three components, which are a 45-degree elbow pipe, an extension tube, and a diffuser.

Figure 47. Cut view of the hydraulic tank to show the fluid return line assembly.

A 45-degree elbow pipe is installed on the top of the oil tank which has a bore size of 4” and matches the bore size of the cooler outlet. With the elbow, the hydraulic hose that connects the oil tank the cooler has a lower chance to exceed the hydraulic hose’s minimum bend radius. Figure 48 shows the model of the 45-degree elbow pipe. The bottom of the elbow is welded on the top of the hydraulic oil tank. The thread on the top of the pipe is used to connect the hoses between the oil tank and the cooler.

Figure 48. Render of a 45-degree elbow pipe model created by the team.

Once the hydraulic oil flows into the oil tank through the elbow pipe, the oil enters a 20” long round steel tube, which has a bore size of 4” and a 4” to 2” adapter on the end. The function of this long tube is to ensure the hydraulic oil returns to the reservoir at a certain liquid depth, preventing foaming within the tank.

Figure 49. Render of the extension tube model created designed by the team.

An oil flow diffuser is also installed on the bottom of the long extension tube. As shown in Figure 50, heat and foam are generated through turbulence when the hydraulic oil flows back to the oil tank through the return line if there is no flow diffuser. However, if a flow diffuser is installed on the return line, the diffuser helps to slow down fluid returning to the reservoir, reducing foaming and pump cavitation from flow disturbances at the inlet, and providing fluid mixing without agitation [33].

[redacted]

Figure 50. The return flow without a diffuser and with a diffuser [33].

A summary of the components required to complete a full assembly of the fluid return line is shown in TABLE XXIII.

TABLE XXIII: COMPONENTS REQUIRED PER FLUID RETURN LINE ASSEMBLY

Component	Vendor	Part Number	Quantity
4” size 45-degree elbow [36]	Acklands Grainger	WWG1LTC4	1
2” threaded tube [37]	Acklands Grainger	GGM10E691	1
Flow Diffuser [38]	Parker	5563	1

3.3.2.1.6.3 Fluid Outlet Lines

As shown in Figure 51, the hydraulic oil tank body has two fluid outlet lines. The bore size of the fluid outlet line in the hydraulic oil tank is 4” which is the same as the bore size of the hydraulic pump inlet. The fluid outlet lines are two 4” threaded pipes welded on the front face on the tank. The threads are used to connect the hydraulic oils.

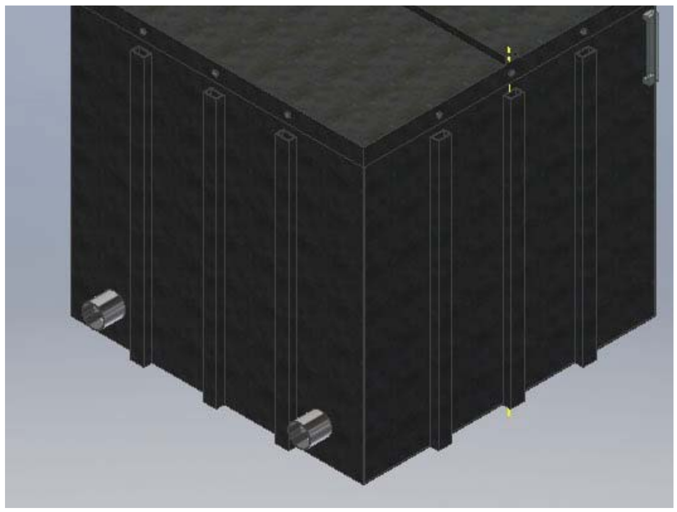


Figure 51. Render of the tank focusing on the two fluid outlet lines.

The specifications of the threaded tube are shown in TABLE XXIV.

TABLE XXIV: SPECIFICATIONS OF FLUID OUTLET THREADED LINES

Component	Vendor	Part Number	Quantity
4” Threaded Tube [39]	Acklands Grainger	GGM1LRZ7	2

3.3.2.1.6.4 Sight Gauge

A 10” long fluid level gauge is selected as the sight gauge of the hydraulic oil tank. The level gauge has the marks of the maximum oil level as well as the minimum oil level, which are used to help the gearbox test stand operators with monitoring the hydraulic oil level in the hydraulic reservoir tank. The sight gauge is positioned on the wall of the hydraulic tank such that the middle of the sight gauge is at the oil's resting level within the hydraulic reservoir.

Figure 52. Technical drawings of the 10" fluid level gauge [38].

The specifications of the fluid level gauge are shown in TABLE XXV.

TABLE XXV: SPECIFICATIONS OF THE SELECTED SIGHT GAUGE

Component	Vendor	Part Number	Length	Quantity
Fluid Level Gauge [38]	Parker	FL69321	10"	1

3.3.2.1.6.5 Breather and Filler

A pressurized metal filler breather, as shown in Figure 53, is installed on the top of the hydraulic oil tank to allow Elmer's to refill the oil tank with filtered oil. Additionally, the attached filtered breather allows the exchange of air between the oil tank and the atmosphere when the fluid level changes due to temperature, pressure, and system operation [40].

Figure 53. Pressurized metal filler breather from Parker [38].

The specifications of the filler breather are shown in TABLE XXVI.

TABLE XXVI: SPECIFICATIONS OF THE SELECTED FILLER BREATHER

Component	Vendor	Part Number	Pressurization Option	Quantity
Metal Filler Breather [38]	Parker	PAB.1730.10.5	5 psi	1

3.3.2.1.6.6 Clean-out plate

As shown in Figure 54, the top plate of the hydraulic oil tank was designed as a clean-out plate to allow access to the inner part of the tank for cleaning and maintenance. Two short tubes are welded on the top bar of the tank lid which can be used to lift the top lid with the forklift. The two holes on the top of the clean-out plate are cut for installing the fluid return line and the filler breather. Additionally, three ¾" holes are drilled on each side of the clean-out plate and the tank body sidewalls. Steel nuts are also welded on the inner side of the sidewalls, which are shown in Appendix A. These holes and weld nuts help the clean-out plate with being bolted with the hydraulic oil tank body. The material of the clean-out plate is the same as the tank body, which is A36 sheet metal.

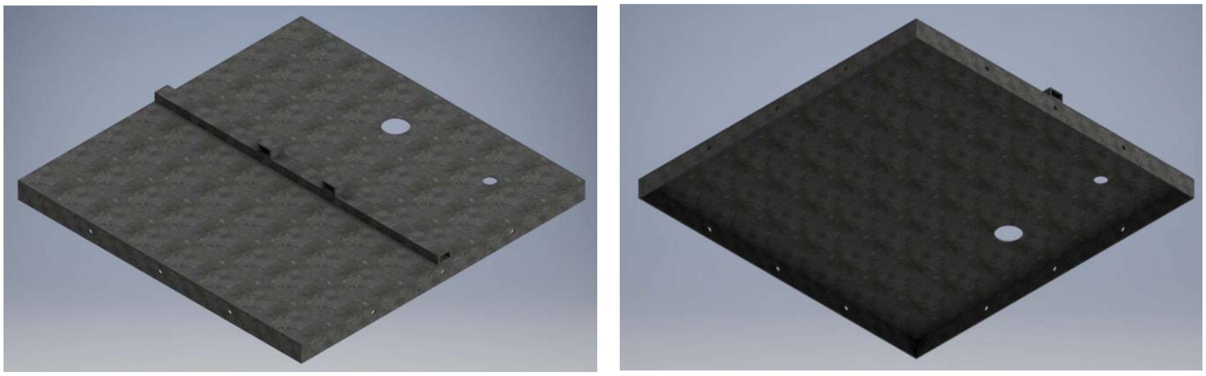


Figure 54. Render of the clean-out plate from both top and bottom view.

Preliminary FEA was performed for testing if the two welded short tubes are strong enough to hold the whole lid during lifting. As shown in Figure 55, a gravity force is applied to the lid and the top of the two short tubes are fixed during the stress analysis. The converged maximum stress is 6.106 ksi which is approximately 1/6 of the yield stress of A36 sheet metal (36 ksi). Therefore, the two short tubes are designed to be strong enough to hold the clean-out plate during lifting.

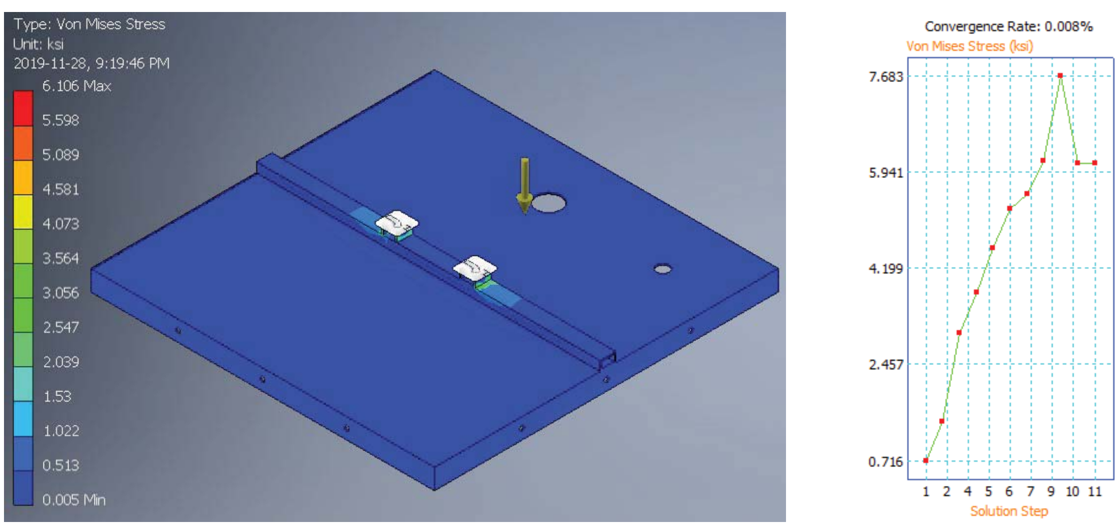


Figure 55. Results of preliminary FEA for the tank top plate under gravity force when lifted.

3.3.2.1.6.7 Baffle Plate

As shown in Figure 57, a baffle plate is installed inside the hydraulic oil tank. The baffle plate separates the fluid return line from the fluid outlet line, permits foreign substances to settle at the bottom of the reservoir and removes entrained air from the hydraulic oil [41]. Additionally, a baffle plate prevents localized turbulence in the reservoir and promotes heat dissipation from the reservoir walls.

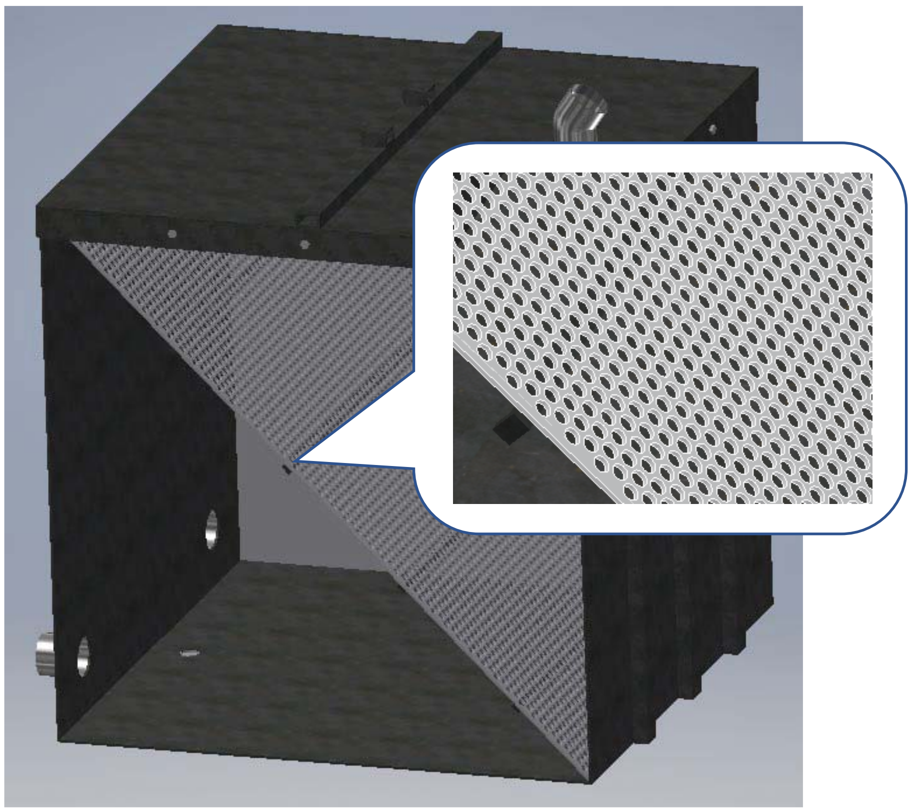


Figure 56. Render of auxiliary view of the hydraulic oil tank CAD model focusing on the baffle plate.

Apart from the baffle plate, five 54” long braces are welded on the sidewalls of the tank to hold the baffle plate, as shown in Figure 57. A cut-to-size 2’x5’x0.5” A36 sheet metal can be purchased and laser cut to the proper size of the braces.

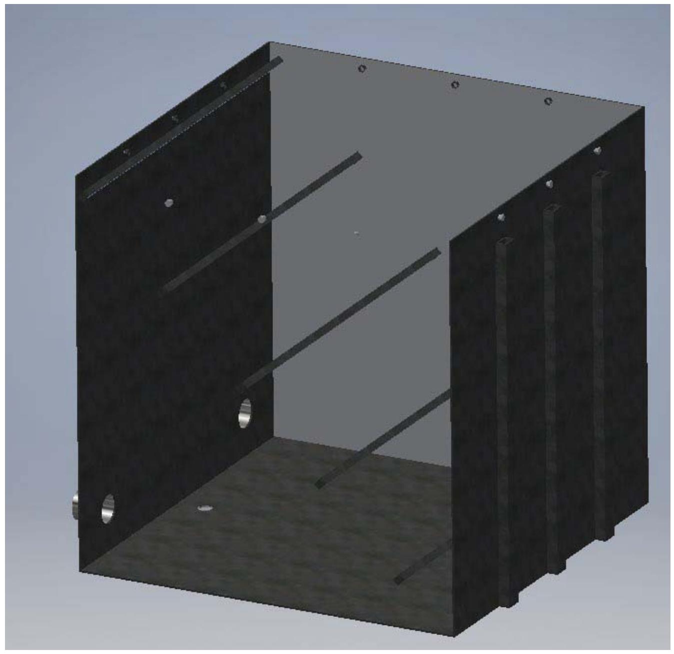


Figure 57. Render of the oil tank assembly focusing on the braces designed to hold the baffle plate.

A nut is welded at both ends of the braces, as shown in Figure 58.

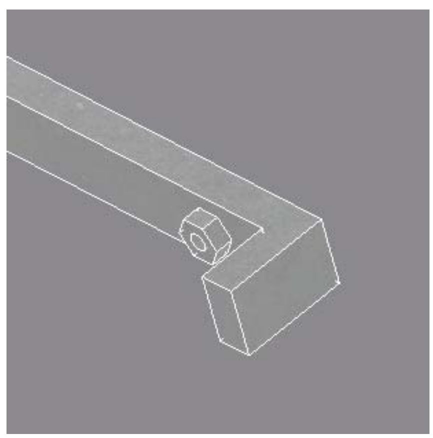


Figure 58. CAD model to show the welded nut on the brace.

The baffle plate, which is 76.3” long and 54” wide steel perforated sheet, is bolted on the top of the braces with the help of the welded nuts. In this case, the baffle plate can be easily removed for cleaning the tank. However, the team found that the maximum size of the steel perforated sheet available for purchase is 48”x120”, which is not big enough to be the create the 76.3”x54” baffle plate. Therefore, two cut-to-size 38.15”x54” sheets need to be purchased and welded together to be the baffle plate. A summary of the components required to complete a full assembly of the baffle plate is shown in TABLE XXVII.

TABLE XXVII: COMPONENTS REQUIRED PER BAFFLE PLATE ASSEMBLY AND COMPONENT SPECIFICATIONS

Component	Vendor	Part Number	Size	Quantity
Perforated Steel Sheet [42]	Metals Depot	PS31614	38.15”x54”x0.1875”	2
A36 Sheet Metal 0.5” Thick [34]	Metals Depot	P112	24”x54”	1
Nuts	TBD	TBD	1”	10

3.3.2.1.6.8 Drain plug

As shown in Figure 59, a 2.5” diameter magnetic drain plug is installed on the bottom of the tank body to ensure the hydraulic oil tank can be regularly maintained. With the drain plug, operators can easily drain dirty hydraulic oil from the reservoir and remove any contaminations within the reservoir. The magnetic drain plug can also collect the metal contaminations to keep the oil in the tank clean. Additionally, a 2.5” threaded tube is welded to the bottom plate of the tank so the drain plug can be properly installed to the tank.

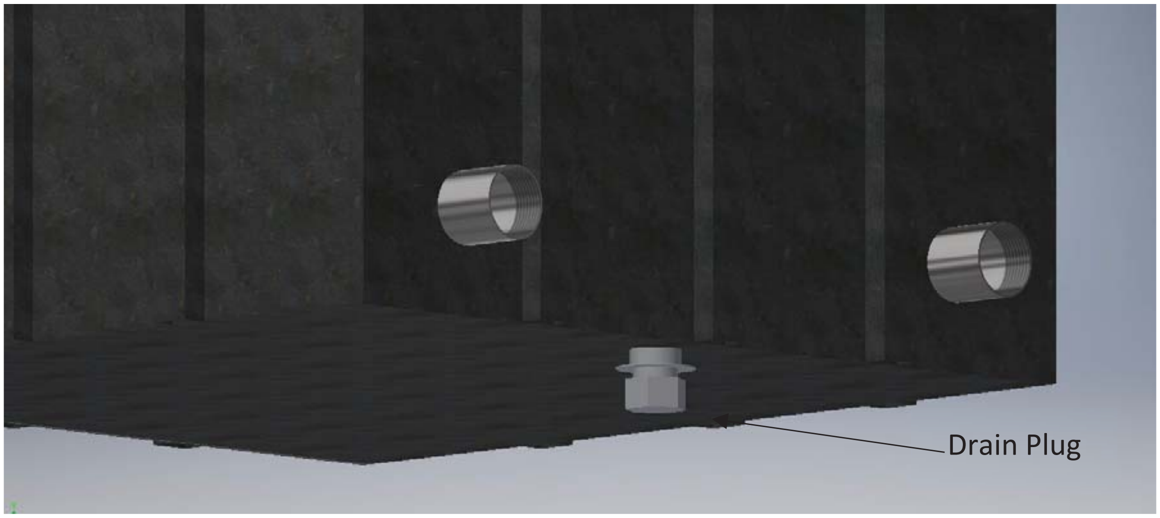


Figure 59. Render of the oil tank CAD model to show the drain plug.

The specifications of the drain plug are shown in TABLE XXVIII.

TABLE XXVIII: SPECIFICATIONS OF THE SELECTED DRAIN PLUG

Component	Vendor	Part Number	Quantity
Magnetic Drain Plug [43]	Lenz	32STP-MAG	1
2.5” threaded tube [44]	Acklands Grainger	GGM2TV90	1

3.3.2.1.7 Hydraulic Hoses and Line Losses

Three sizes of hydraulic hose are required for the gearbox test stand, being 4”, 2”, and 1.5” diameter hoses. The 4” hoses within the gearbox test stand are used as the suction and return lines for the K3VG-280DT hydraulic pumps. As the 4” lines are suction and return lines, the pressure ratings of the 4” lines are relatively low. The pressure requirements are calculated further in this section of the report. The outlet of the pump is a 2” in diameter, set by Kawasaki. Therefore, a 2” hydraulic line rated for 6000 psi is used from the pump to the valve manifolds.

Within the control valve assembly, SAE Code 62 unions are used and are defined in Section 3.3.2.1.3, as well as a 1.5” diameter hydraulic line to connect one of the manifolds to a tee-body block. The unions before the valve manifolds must be rated for SAE Code 62, which translates to a pressure rating of 6000 psi. However, after the valve manifolds, the pressure in the hydraulic system is much lower as the only pressure in the system is caused by the resistance in circuit elements from the valve manifold to the reservoir.

Typically, the resistive pressures through the remaining components, being the filter, cooler, and hydraulic lines in the hydraulic circuit are much lower than the output pressure of the pump. While hoses rated at 6000 psi may not be required for the remaining hydraulic hoses, calculations were performed to determine the resistive pressures in the hydraulic lines. The equations used to determine the resistive pressures in the hydraulic lines, measures as pressure losses, are given in the following equations [32].

$$\Delta p = \frac{\lambda * L * SG * Q^2}{74.3 * D^5}$$

$$\lambda = \begin{cases} \dfrac{64}{Re}, Re < 2000 \Rightarrow Laminar \\ \dfrac{0.3164}{Re^{0.25}}, Re > 2000 \Rightarrow Turbulent \end{cases}$$

$$Re = \frac{3164Q}{vD}$$

Within these equations, L is the length of the hydraulic line, SG is the specific gravity of the oil, Q is the flow rate of the oil through the hose, D is the diameter of the hose, v is the viscosity of the oil, and Re is the Reynolds number.

As the lengths of the hydraulic hoses are determined in Section 3.3.2.5.2, the aforementioned equations were used to determine the pressure drops as a function of length for each diameter of hydraulic hose. This was achieved by dividing the pressure drop by the length, giving $\frac{\Delta p}{L}$. Within these calculations, the oil viscosity of the oil was taken as 56.93 cSt rather than 9.18 cSt as the higher viscosity value provides a worst-case pressure drop. The specific gravity of the oil was also taken to be 0.86 [32]. The calculations for the 4” hoses are as follows:

$$Re = \frac{3164(206)}{(56.93)(4)} = 2862 > 2000 \Rightarrow Turbulent$$

$$\lambda = \frac{0.3164}{2862^{0.25}} = 0.0433$$

$$\frac{\Delta p}{L} = \frac{0.0433 * 0.86 * 206^2}{74.3 * 4^5} = 0.02075\, psi/ft$$

The calculations for the 4” hose use a flow rate of 206 gpm, as this is the maximum flow that is experienced by either the suction or return lines in the hydraulic circuit. As a worst-case scenario, the pressure drop over any length of 4” hose is $0.02075\, psi/ft$.

The calculations for the 2” hydraulic hoses are as follows:

$$Re = \frac{3164(206)}{(56.93)(2)} = 5724 > 2000 \Rightarrow Turbulent$$

$$\lambda = \frac{0.3164}{5724^{0.25}} = 0.0363$$

$$\frac{\Delta p}{L} = \frac{0.0363 * 0.86 * 206^2}{74.3 * 2^5} = 0.558\, psi/ft$$

The calculations for the 2” diameter also use a maximum flow rate of 206 gpm to represent the worst-case scenario. This scenario can be visualized by assuming that the full 600 HP load is applied to only one output of the test gearbox, producing the full 206 gpm flow through one hydraulic hose.

The pressure drop through the 1.5” hydraulic lines are given by the following:

$$Re = \frac{3164(103)}{(56.93)(1.5)} = 3816 > 2000 \Rightarrow Turbulent$$

$$\lambda = \frac{0.3164}{3816^{0.25}} = 0.0403$$

$$\frac{\Delta p}{L} = \frac{0.0403 * 0.86 * 103^2}{74.3 * 1.5^5} = 0.651\, psi/ft$$

Within the calculations for the 1.5" hydraulic hoses, a flow rate of 103 gpm was used as this is the maximum flow rate. Referring back to Figure 31, which shows the control valve layout, the 206 gpm flow of oil into the control is split evenly into two flows, with only one of which flows through the 1.5" hydraulic hose.

After determining the required sizes of hydraulic hoses, hose models from Parker Hydraulics were sourced. The four types of hydraulic hose are summarized in TABLE XXIX.

TABLE XXIX: HYDRAULIC HOSE PROPERTIES [45]

Parker Hose Series	Hose Inner Diameter (in)	Pressure Rating (psi)	Bend Radius (in)	Calculated Pressure Drop (psi/ft)
7219	4	75	13.4	0.02075
797TC	2	6000	15	0.558
787TC	2	5000	12.5	0.558
787TC	1.5	5000	12.5	0.651

A simple sketch of the required hoses in the system is shown in Figure 60.

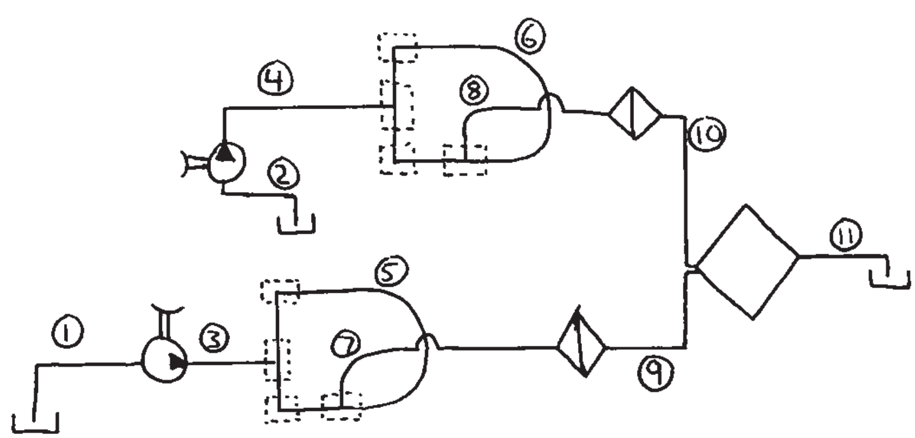


Figure 60. Numbered hydraulic hoses required for the hydraulic dynamometer.

The hoses labeled in Figure 60 are summarized in TABLE XXX.

TABLE XXX: SPECIFICATIONS OF REQUIRED HYDRAULIC HOSES

#	ID Size (in)	Pressure Rating (psi)	Length* (ft)	Start Fitting	End Fitting
1	4	75	22	4” NPT (Male)	4” SAE 4-Bolt Flange (Straight)
2	4	75	15	4” NPT (Male)	4” SAE 4-Bolt Flange (Straight)
3	2	6000	8	2” SAE 4-Bolt Flange (45°)	2” SAE 4-Bolt Flange (Straight)
4	2	6000	3	2” SAE 4-Bolt Flange (Straight)	2” SAE 4-Bolt Flange (90°)
5	1.5	5000	3	1.5” SAE 4-Bolt Flange (Straight)	1.5” SAE 4-Bolt Flange (Straight)
6	1.5	5000	3	1.5” SAE 4-Bolt Flange (Straight)	1.5” SAE 4-Bolt Flange (Straight)
7	2	5000	6	2” SAE 4-Bolt Flange (Straight)	2” SAE 4-Bolt Flange (Straight)
8	2	5000	6	2” SAE 4-Bolt Flange (Straight)	2” SAE 4-Bolt Flange (Straight)
9	2	5000	7	2” SAE 4-Bolt Flange (Straight)	2” SAE 4-Bolt Flange (Straight)
10	2	5000	7	2” SAE 4-Bolt Flange (Straight)	2” SAE 4-Bolt Flange (Straight)
11	4	75	5	4” SAE 4-Bolt Flange (Straight)	4” NPT (Male)

**Note that the lengths of the hydraulic lines were determined after designing the skids for the system, covered in Sections 3.3.2.5 and 3.3.2.5.2, and their relative positions. The client is free to change the lengths of the hydraulic hoses as they see fit.*

Within TABLE XXX, only lines #3 and #4 are required to be rated for 6000 psi, as they are the only two hydraulic lines that experience pressures over 5000 psi. All other lines within the system are theoretically only required to be rated for low pressures of under 1000 psi. However, any hydraulic lines from a reputable manufacturer with an internal diameter of 1.5" or 2" are rated at 5000 psi. Therefore, to remain cost-efficient, the team recommends that the 6000 psi lines are used only where needed and the remaining lines are set to be 5000 psi lines. Combining the pressure drops listed in TABLE XXIX and the lengths from TABLE XXX allows for the line losses of the hydraulic circuit to be calculated, which are shown in TABLE XXXI.

TABLE XXXI: PRESSURE DROPS IN HYDRAULIC HOSES

#	Length (ft)	Pressure Drop per Foot (psi/ft)	Calculated Pressure Drop (psi)
1	22	0.02075	0.46
2	15	0.02075	0.31
3	8	0.558	4.46
4	3	0.558	1.67
5	3	0.651	1.95
6	3	0.651	1.95
7	6	0.558	3.35
8	6	0.558	3.35
9	7	0.558	3.91
10	7	0.558	3.91
11	5	0.02075	0.10

These values are then used to ensure that the burst pressure of components throughout the hydraulic circuit is not exceeded. Important components for which pressures are required are the filters and the cooler in the system. As the pressure of the oil reservoir is atmospheric, 0 psi, all component pressures can be calculated by working backward from the reservoir. Regarding the cooler, the only component affecting the pressure through the cooler is hose #11, the oil return line between the cooler and the reservoir. The pressure in this hose is 0.1 psi, significantly lower than the burst pressure of 250 psi. Therefore, the selected cooler was acceptable to be used in the hydraulic circuit.

The other component requiring a check for burst pressure calculations are the hydraulic oil filters, located before hose #9 and hose #10, as shown previously in Figure 60. The pressure at the filter is equal to the sum of the pressure drop over line #9 or line #10, 3.35 psi, the pressure drop through the oil cooler, given in Figure 35 as approximately 10 psi through the cooler at a flow rate of 206 gpm, and the pressure drop through the return line, 0.1 psi. Therefore, the pressure the filters must withstand a maximum operating pressure of at least 13.45 psi. As the selected filters are rated for up to 500 psi, they are acceptable to use in the hydraulic circuit.

3.3.2.2 Increaser Gearbox

A planetary speed increaser gearbox between the test gearbox output and the hydraulic pump input is designed to drive the hydraulic pumps at the correct speed. An overview of the preliminary speed increaser gearbox design is shown in Figure 61 and Figure 62.

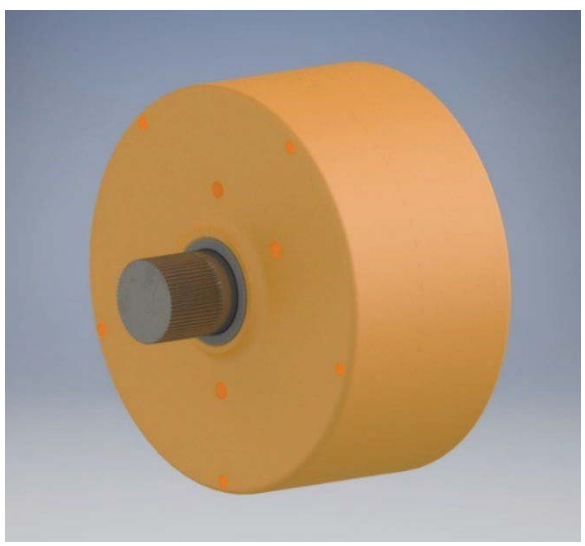


Figure 61. Render of increaser planetary gearbox design.

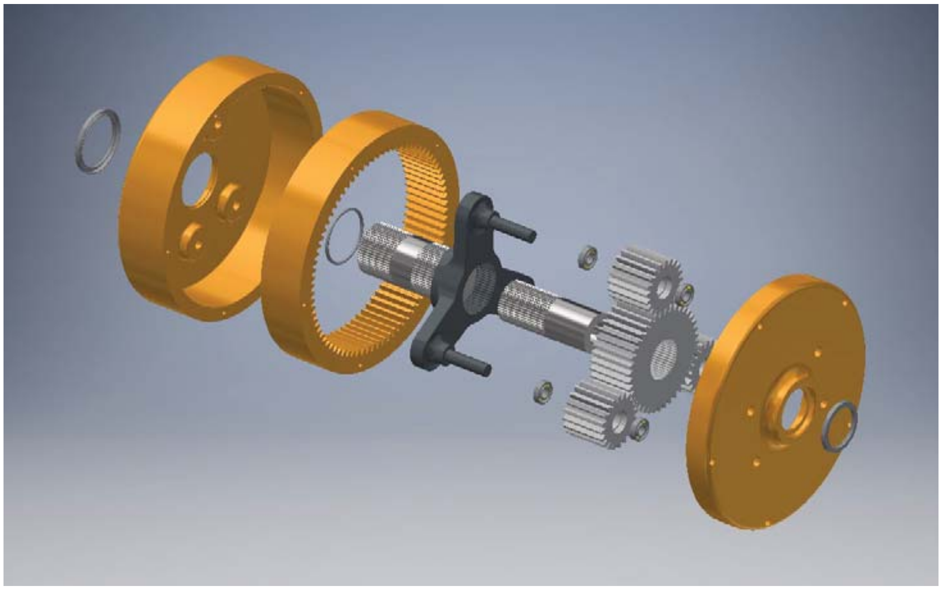


Figure 62. Exploded view of increaser planetary gearbox design.

To ensure that the increaser gearbox design met the expectations of the client, the scope was first defined and is shown in TABLE XXXII.

TABLE XXXII: SCOPE OF INCREASER GEARBOX DESIGN

In Scope	Out of Scope
Development of Gearbox Layout	Fatigue Analysis
Basic Gear Analysis	Determination of Appropriate Backlash for Gears
Basic Carrier Design and Analysis	Lubrication of Gearbox
Basic Shaft Design	Analysis of Forces on Case
	Retaining Ring Selection
	Design of Adapter for Input Shaft

The selected gear ratio for the gearbox is 0.333:1. The required input speed of the selected hydraulic pump is at a range between 1200 rpm and 1800 rpm, while the speed of the output shaft of the test gearbox is approximately 500 rpm. This design results in an input speed to the hydraulic pumps of 1500 rpm. Two speed increaser gearboxes are required for the system depending on the number of the outputs of the test gearbox.

A planetary gearbox was used instead of a single-stage spur gearbox. A single-stage spur gearbox of this type that could meet the necessary requirements would be too large and heavy to suit the rest of the gearbox test stand design. A compound spur gearbox was also considered, but had the same problems. This change in concept was discussed with and approved by Elmer’s.

The design team first looked to source a planetary gearbox as this was the best option to ensure the reliability of the unit and reduce the scope of the project. Upon conducting research, a planetary gearbox that met all the necessary torque, speed and life requirements could not be found so the design team created a preliminary planetary gearbox design for the speed increaser gearbox.

The gearbox uses the carrier as the input, while the ring gear is fixed and forms part of the gearbox casing, and the sun gear is the output. Spur gears are used for all gears for ease of manufacturing and optimal efficiency and so axial forces do exist in the gearbox. Three planets are used as was recommended by Elmer’s. The planet, sun, and ring gears have tooth counts of 20, 40 and 80 teeth respectively providing the required reduction. End caps were designed to be fixed to the ring gear using threaded fasteners and contain threaded extrusions for mounting. Input and output shafts are splined onto the carrier and sun gear. The gearbox has an overall diameter of 24” and an approximate width of 14.5”. Each component will be discussed in further detail in the following subsections.

It should be noted that the speed increaser gearbox was designed to handle 6302 lb-ft, the torque that is transmitted through the system when the engine is producing at 2000 rpm. This was done to keep the gearbox as light and compact as possible and load should not be applied to the test gearbox until these operating conditions are reached as the diesel engine is capable of producing more torque at lower engine speeds.

Note that the calculations made when designing the speed increaser gearbox were performed in Excel and the files with these calculations were submitted to Elmer's.

3.3.2.2.1 Gears

As previously mentioned, the gears have tooth counts of 20, 40 and 80 teeth for the planet, sun and ring gears respectively giving an overall ratio of 0.333 to 1. The gears have a face width of 4.5", a pressure angle of 20 degrees and a diametral pitch of 4 teeth per inch. ANSI 14NiCr18 alloy steel was chosen as the gear material of each gear due to its high allowable bending and contact stress numbers, which allowed for the gearbox to reasonably compact considering the torque it must transmit.

The sun gear is splined onto the output shaft and is prevented from moving axially by retaining rings. Each planet gear features machined holes in each end as shown in Figure 63, into which bearings will be shrink fit and pressed in. This assembly will be shrink fit onto the carrier and constrained axially by the carrier shoulder on one end and a rotating ring on the other. The ring gear was designed to be held stationary by the case. The machined holes in the planet gears are shown in Figure 63.

Figure 63. Render of increaser planet gear focusing on machined bearing holes.

Information regarding the methodology used to design the Haulmaster grain cart gears was unavailable. Therefore, in order to ensure the reliability of the speed increaser gearbox, the AMGA stress equations were used for the gear design. The AGMA stress equations were chosen as the use of these equations results in a conservatively designed gearset for a given set of operating conditions. Due to this fact as well as the conservative selection of factors used in the AGMA equations, the design team felt comfortable using a safety factor of 1.5 for the gear design. Hand calculations using the AGMA equations were performed, then the accuracy of the results was checked using the gear design accelerator in Autodesk Inventor. The gear design accelerator in Autodesk Inventor does not allow the user to specify values for all the AGMA equation factors so it was only used as a rough check of the hand calculations, which were trusted in the design of the gears.

The effect of the three planet gears sharing the load was approximated by dividing the value of the transmitted load for each gear by three resulting in bending and contact stress values being 0.33 and 0.577 times the values for a typical two-gear gear mesh. While the full calculations and derivations of AGMA stress equation factors will not be discussed here, the following describes some of the key factors selected.

A gear quality number of 5 was selected as typical gear quality numbers range from 3 to 7 for most commercially available gears. The design team felt picking the average value within this range was reasonable. This number affects the dynamic factor, used in the calculation of both the bending and contact stresses [11]. An overload factor of 1.5 was selected as this is the typical value used when the driving machine is a multi-cylinder internal combustion engine with a medium impact load [46]. It was assumed there will be no detrimental surface finish effect and thus surface condition factor of 1 was used. Likewise, it was assumed that there is no detrimental size effect with the gears used so a size factor of 1 was used. When calculating the load distribution factor, C_{mc} was set equal to 1 as the teeth are not crowned. The values used to calculate C_{ma} were those used for commercial enclosed units; C_{pm} was set to 1.1 since the gears are not straddle mounted and C_e was set to 1. When calculating the hardness ratio factor it was assumed that the pinion and gear will have the same hardness. When calculating the stress cycle factors, there was a range of values that could be used due to the number of load cycles, so the lowest values in the range were used for these factors to account for the worst-case scenario. The desired reliability of 99% was selected giving a reliability factor of 1. It was assumed that the temperature of the gears will not exceed 120 degrees Celsius when in service so a temperature factor of 1 was used. The rim thickness of each gear was kept at a minimum of 1.2 times the tooth height in order for a rim thickness factor of 1 to be used at all times [11]. It is recommended that Elmer's closely review these assumptions and selections to verify the accuracy of the results and make adjustments where necessary.

The final design ended up with a minimum factor of safety of 1.477 (approximately 1.5) against failure due to pitting for the planet gears. The failure mode of each gear was determined by comparing the bending stress safety factor to the contact stress safety factor squared as dictated by the AGMA stress equations. From this method, it was concluded that when the gears fail, the planet, sun and ring gears will fail due to bending stress, contact stress and bending stress respectively.

It should be noted that the gears were modeled using the design accelerator in Autodesk Inventor. As a result, the gears do not have any backlash modeled. However, determination of the correct backlash necessary for smooth operation and proper lubrication is considered out of the scope of this project and Elmer's should perform analysis to determine the correct backlash.

3.3.2.2.2 Shafts

The speed increaser gearbox was designed with two shafts. One shaft is the input shaft, shown in Figure 64, which is splined on both ends and transmits torque to the carrier.

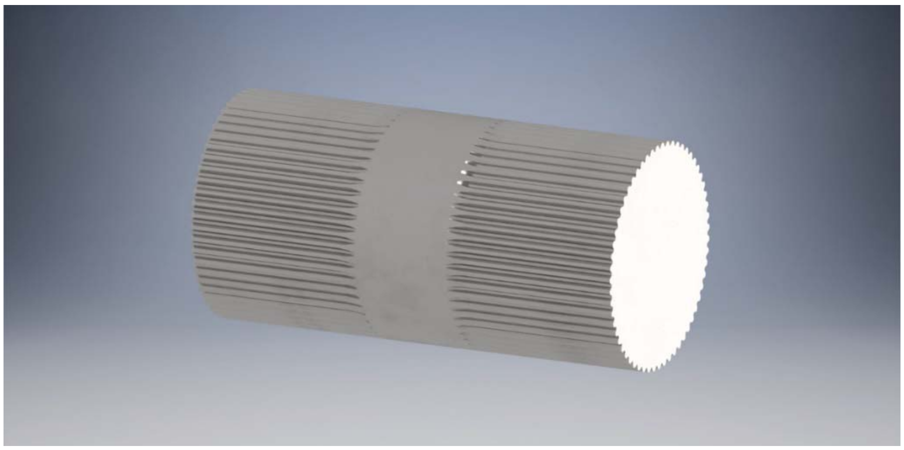


Figure 64: Render of increaser gearbox input shaft

The other shaft is the output shaft which is splined onto the sun gear on one end and whose other end is hollow with a keyseat cut into it as can be seen in Figure 65.

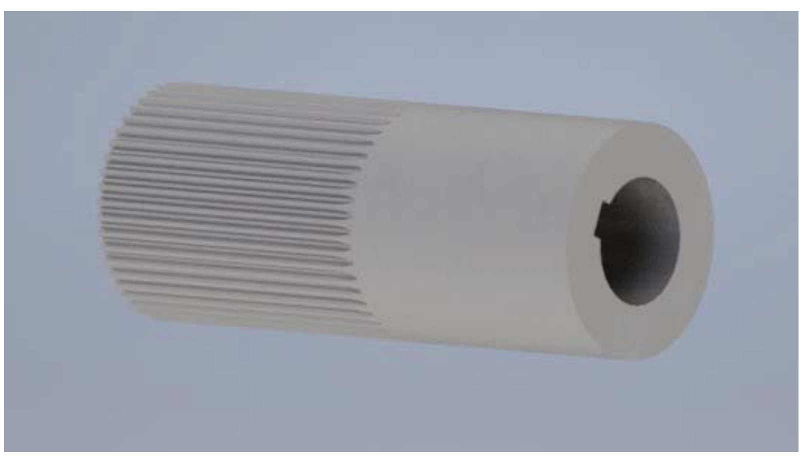


Figure 65: Render of speed increaser gearbox output shaft showing keyway

This hollow end was designed to fit over the hydraulic pump's keyed input shaft, allowing the speed increaser gearbox to directly drive the pump. Finite element analysis was not performed on the keyway, therefore it is recommended that the keyway is examined further by the client to ensure that the design will not fail.

The main design consideration for the shafts was the torque transmitted through the gearbox. There should be no axial forces on any components and since it is a planetary gearbox there will be no radial shaft forces on the input or output shafts because the shaft separation forces created by the planets cancel each other out. To determine shaft an appropriate shaft and spline size, the involute spline design accelerator in Autodesk Inventor was used. This software uses the ANSI strength calculation method for ANSI spline types and analyzes the overall shaft hub spline connection. The design accelerator outputs the required minimum shaft diameter and spline length once the spline type, shaft and hub material properties, hub size, desired factor of safety, spline calculation factors and operating conditions were specified. A factor of safety of 2 was used when selecting the splines type and length to use for the various applications. The various spline calculation factors were determined using the Machinery's Handbook (30th Edition) [47].

The first spline calculation factor that was determined was the spline application factor. A value of 2.4 was selected to be on the safe side, as the loading the spline will see would most likely be considered light shock but intermittent shock might also occur. Next, the fatigue life factor was calculated, which is based on the number of starts and stops (torque cycles) that the spline will be subjected to. Elmer's specified that each test gearbox would undergo 15000 stops and starts for a complete test. Since Elmer's indicated that they may use this testing system for up to three gearboxes, a torque cycle number of 45000 was used in order to determine the fatigue life factor. Since this falls between the 10^4 and 10^6 torque cycle values for which fatigue life factors are listed, the values for 10^6 torque cycles were considered. Since the increaser gearbox will not be operated in reverse the unidirectional value was selected. This resulted in a fatigue life factor of 0.5 being selected. Finally, since fixed splines are being used, the load distribution factor is equal to 1 [47]. Case hardened steel with allowable compressive and shear stresses of 5000 and 50000 ksi respectively was selected as the material used for the shafts as this was one of the materials automatically built into the design accelerator software with high allowable stress values.

Using these parameters along with the design accelerator calculations, input and output shaft diameters of 5” and 4” respectively were selected. ANSI B92.1 – 37.5 degree, fillet root, side fit 12/24 class 5, 12 teeth per inch 59 tooth splines used on both ends of the input shaft. An ANSI B92.1 – 37.5 degree, fillet root, side fit 12/24 class 5, 12 teeth per inch 47 tooth spline is used on the gear end of the output shaft. The splines on the input shaft have lengths of 3.5”, with the spline on the carrier side having an engagement length of 3”. The spline on the output shaft has a length of 5” with an engagement length of 4.5”. These shaft specifications exceed the requirements to transmit the required torque.

Retaining rings were not selected and grooves were not designed into the shafts as any retaining ring should be adequate to retain the gearbox components in their correct position the shafts since there should not be any axial force on any of the internal gearbox components. The retaining rings only need to prevent movement of the components due to vibration, etc. The client is recommended to select any retaining rings they prefer to use and then analyze the shafts to ensure that adding grooves will not cause any problems. However and overhang of 0.5” was accounted for and modeled where each shaft splines into another gearbox component to allow space for a retaining ring and groove to be added.

3.3.2.2.3 Carrier

The planet gear carrier of the increaser gearbox was designed with shoulders to help with the installation and retention of the planet gears. The carrier was designed with a splined center for transmission of torque from the input shaft. The carrier material is AISI 4140H (UNS H41400) due to its high strength and machinability, and the carrier design is shown in Figure 66 [7].

Figure 66. Render of increaser gearbox carrier.

The spline was selected using the spline design accelerator, following the process outlined in the previous section to verify that the carrier's spline is adequate to transmit the torque. The forces that will be exerted on each of the planet gear shafts were calculated by taking the input torque to the gearbox, then dividing by the distance from the carrier centre to the centre of one of the planet gear shafts on the carrier and dividing this value by three. This resulted in a force of 3361 lbf acting on each planet shaft of the carrier. Preliminary finite element analysis was performed on the carrier to verify that these forces would not cause failure of the carrier. In this finite element study, the spline faces that would oppose the rotational motion of the carrier were held fixed and the 3361 lbf force was applied as a bearing load to each planet gear shaft at the appropriate angle. The results of this finite element analysis are shown in Figure 67.

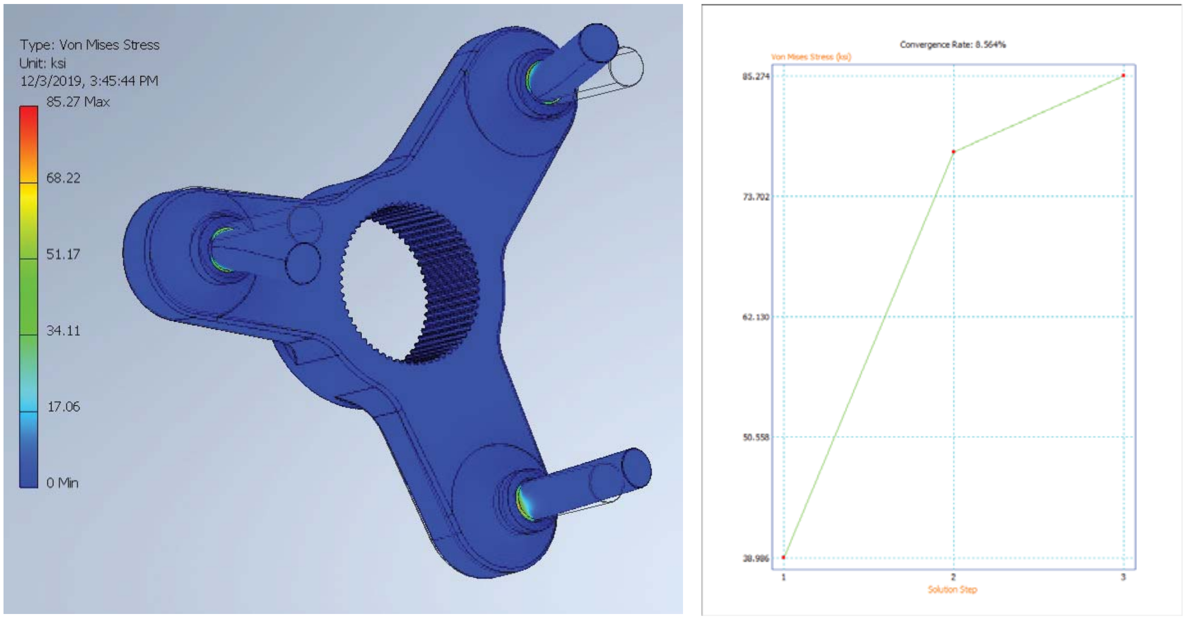


Figure 67. Results of preliminary FEA for increaser gearbox carrier.

The preliminary FEA suggests that the carrier has a high factor of safety over most areas of the design. However, the stress concentrations at the base of the planet gear shafts limit the factor of safety to approximately 2. Therefore, further analysis by the client is recommended to verify the design.

3.3.2.2.4 Bearings and Seals

This gearbox was designed to feature bearings for the planet gears, as well as the input and output shafts, and rotary shaft seals were selected for the input and output shafts of the increaser gearbox. When selecting bearings, the four parameters considered were maximum dynamic load, maximum allowable rotation speed, size and cost. A safety factor of approximately two was used when selecting bearings.

The bearings were chosen as part number NTN XLS51GC3 from Motion Canada and 6656K16 from McMaster Carr for the output and input shafts respectively [48] [49]. The cheapest bearings that meet the size and speed requirements were selected since there are no radial forces on the input and output shafts. Only one bearing was needed for each shaft, as they are supported by the gears on the other end.

The bearing selected for the planet gears is the SKF RLS8. The forces that act on each arm of the carrier are equal and opposite to the forces that will be exerted on the planet bearings, so this value was used to select the planet bearings. Each planet gear uses a pair of bearings in order to meet the required safety factor of two and provide better support for the gear.

3.3.2.2.5 Casing

The speed increaser gearbox casing consists of the rings gear and two case ends that are secured to the ring gear with threaded fasteners. The case ends were designed so that the increaser gearbox can be directly bolted to the hydraulic pump mounting bracket. Each case end features threaded extrusions for this purpose as can be seen in Figure 68.

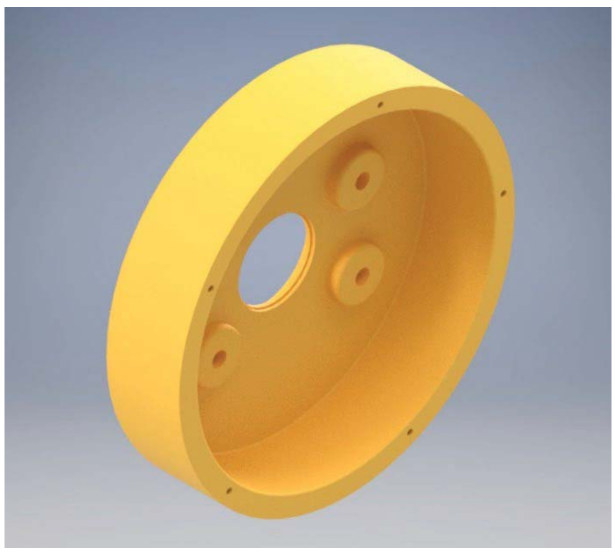


Figure 68. Render of increaser gearbox case displaying mounting locations.

The casing was designed out of 65-45-12 ductile iron. The mounting points were designed to keep the system compact by enabling the gearbox to directly drive the hydraulic pump without any need for intermediate connections. A basic calculation to calculate the tangential force on the ring gear at the location of the fasteners was performed in order to determine the number of fasteners required to hold the sections of the gearbox casing together. By converting the torque in the ring gear to a tangential force, then dividing this force by the number of fasteners, it was determined that five ½-13 UNF grade 5 fasteners would be adequate to secure the case sections together with a factor of safety of 2.5. The casing was designed to be 1.381” thick, however the required thickness was not determined through FEA. It is recommended that Elmer’s perform further analysis to ensure the case will withstand the stress from the applied torque using the designed mounting methods.

3.3.2.3 Test Gearbox Mounting

As one of the five major design modules, the test gearbox mounting system was one of the areas of this project which required significant design. To ensure that this module meets the needs of the client and does not exceed the requirements of the project, a scope table was developed. An overview of the sections included and excluded from the scope of this project are listed in TABLE XXXIII.

TABLE XXXIII: SCOPE OF TEST GEARBOX MOUNTING DESIGN

In Scope	Out of Scope
CAD models of Haulmaster mounting components	Detailed analysis using FEA
Analytical calculations of simple components	Vibration analysis of assembly
Preliminary FEA of major components	Wolverine ditcher gearbox mounting system
Material selection of all components	
Initial skid design	

With the scope limitations of this module set, the team then moved into designing the test gearbox mounting system. A render of the final gearbox mounting system is shown in Figure 69 and Figure 70.

Figure 69. Render of test gearbox mounting system focusing on pivot mount.

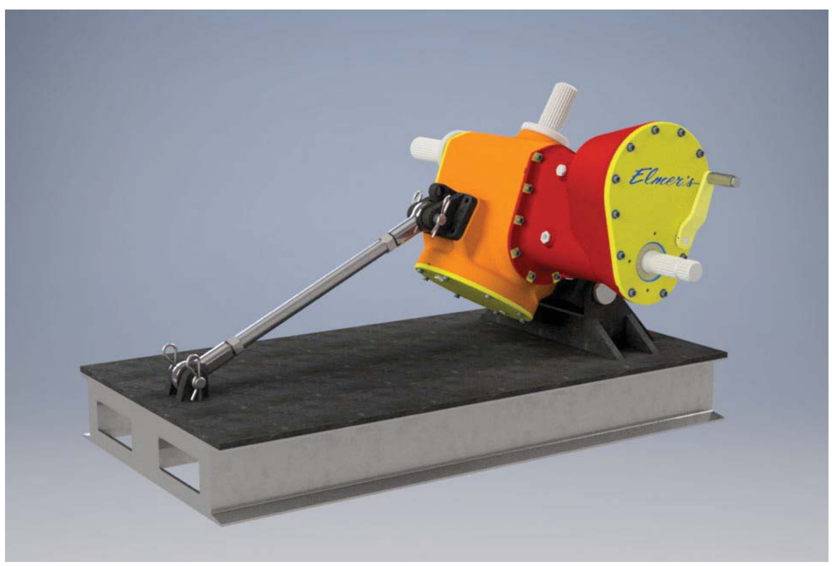


Figure 70. Render of test gearbox mounting system focusing on adjustable mount

The team designed the mounts by first calculating the forces that will be acting on the mounting system during testing.

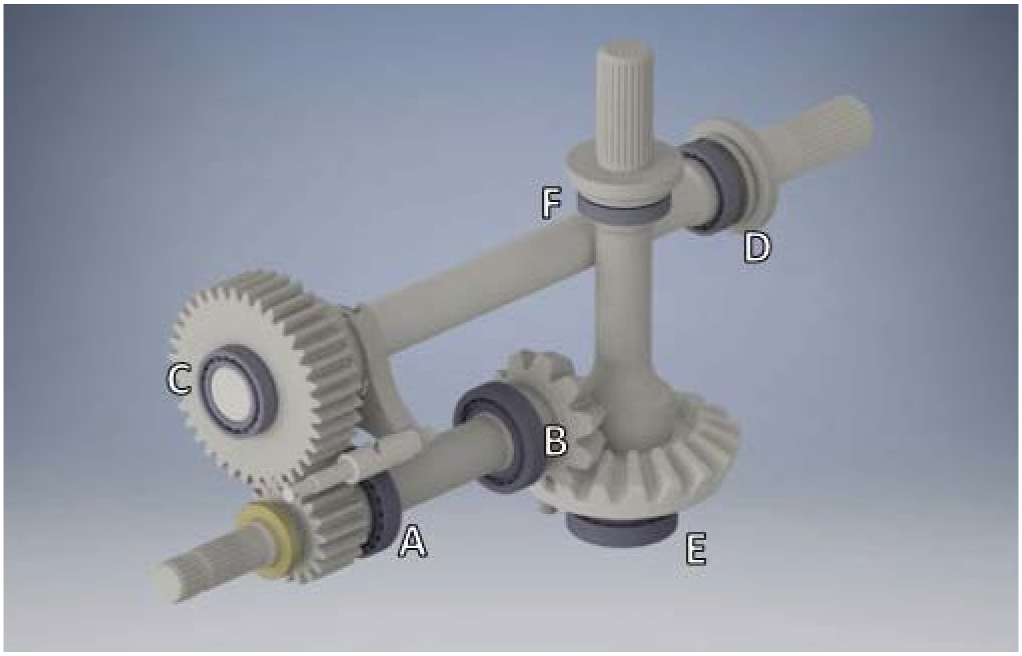


Figure 71. Isometric view of the internal components of the Haulmaster gearbox.

Each of the Haulmaster shafts was analyzed individually, under the assumption that 2000 ft-lb of torque, was applied to the input at 1000 rpm. The forces upon the bearings were calculated using the radial component of the force in the spur and bevel gears, as well as the axial force which only appeared in the bevel gears. These forces were applied at approximately the center of contact between the gears. A moment balance was performed about one of the bearings for each shaft, allowing for the reaction forces to be solved. A summary of these bearing forces, converted to the universal coordinate system, is shown in TABLE XXXIV.

TABLE XXXIV: FORCES ACTING ON HAULMASTER BEARINGS UNDER 2000 FT-LB TORQUE LOAD

Bearing	Force in X [lbf]	Force in Y [lbf]	Force in Z [lbf]
A	3457	-7845	0
B	-812	7315	-10087
C	-2510	6029	0
D	-135	324	0
E	0	10087	-7308
F	0	0	1484

The forces on the case were calculated by summing the forces and moments about the mounting locations. Since the Haulmaster will be tested at different operating angles, the forces for different angles were calculated and the maximum forces expected at the mounting points were recorded in TABLE XXXV.

TABLE XXXV: REACTION FORCES ACTING ON THE TEST GEARBOX MOUNT POINTS

Force on Pivot [lbf]			Moment on Pivot [lb ft]			Force on Adj. Rod [lbf]
x	y	z	x	y	z	Axial
22046.2	-5959.0	15911.0	3483.8	-7713.9	0	-24479.7

3.3.2.3.1 Pivot Mount

As shown in Figure 72, the pivot mount was designed to provide most of the support for the test gearbox. A render of the final pivot mount design is shown in Figure 72.

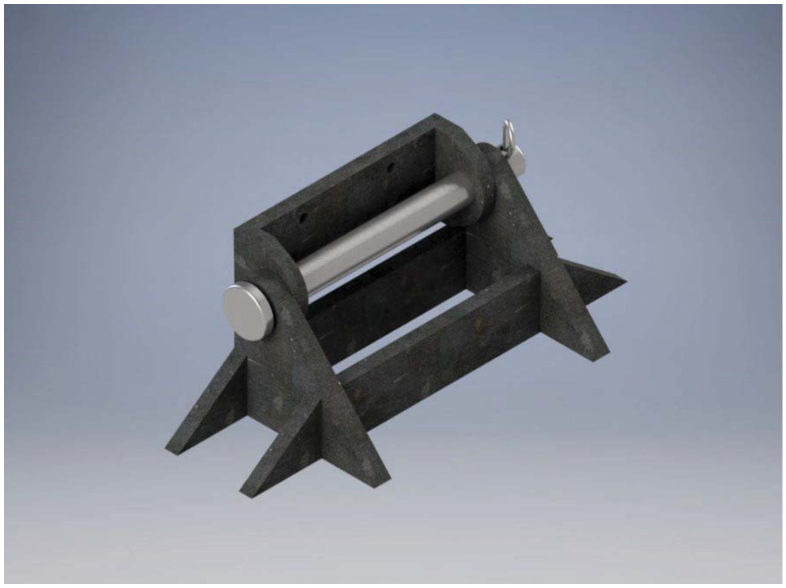


Figure 72. Render of final pivot mount design.

The components of the final pivot mount design were designed iteratively through preliminary FEA. The base of the mount was designed to be 8.0” wide and 21.5” long to provide a large area for the mount to distribute forces. The plates securing the pin were designed to be 11.5” apart, as this distance creates a large reaction moment to counteract the large forces and moments from the Haulmaster during testing. The height of the pin was measured to be 5.625” as this allows for the Haulmaster gearbox to sit flat on the ground and line up with the mounting holes.

The team recognized that it is beneficial for the gearbox to be initially attached while lying flat on the skid. The mounting plate was designed to adapt the existing mounting holes on the gearbox onto the pivoting mount. Additionally, the plate components of the pivot mount were designed out of 3/4” and 1” thick A36 sheet metal to ensure that the mount will support the load. The plates were designed to be welded together because the large thickness prevented the plates from being bent into shape. Slots were designed into each of the supporting plates ensuring the pieces could be easily assembled and welded into place.

The pin was designed to be 1 1/2” diameter and made of 1566 Carbon Steel. The pin was designed to be this size through preliminary FEA since this component will be transmitting all of the load from the attaching mount plate into the base plates. The total length of the pin was designed to be 15.5” and features a steel plate welded to the end to provide the axial constraint for the pin joint. A through-hole and appropriately sized cotter pin were selected as the constraint for the opposite side of the pin as no load is applied. The results of the FEA for the final pivot mount and the convergence plot for the analysis, is shown in Figure 73.

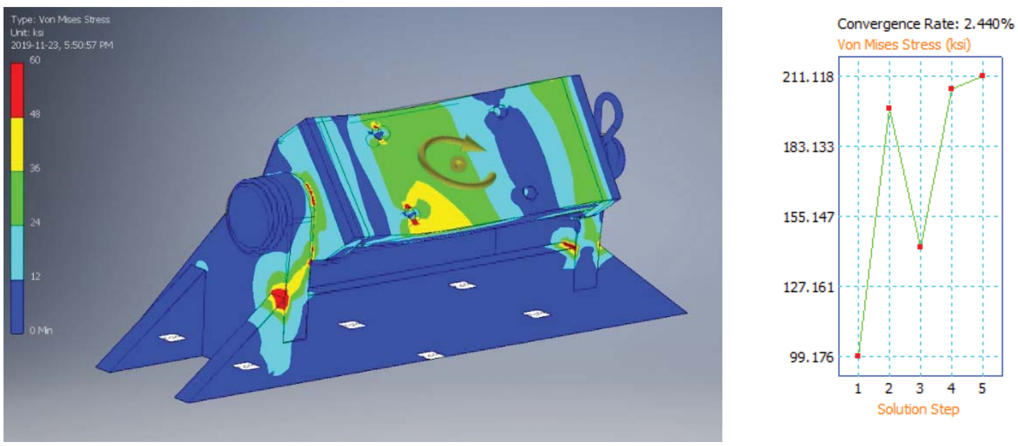


Figure 73. Preliminary finite element analysis results for final pivot mount design.

As shown by the convergence plot the study did not converge, meaning that further analysis would be recommended. There were many stress concentrations due to the solver assuming that two parts are bonded, whereas in the actual system they would not be. This setting was used to reduce runtimes and simplify the analysis. It is recommended that the client perform further finite element analysis to affirm the design.

This final concept was significantly modified from the original concept selected during the concept generation phase of the report. A physical model was created, and the team realized by changing one side to a pivot, there would be a greater range in operating angle for the same change in length.

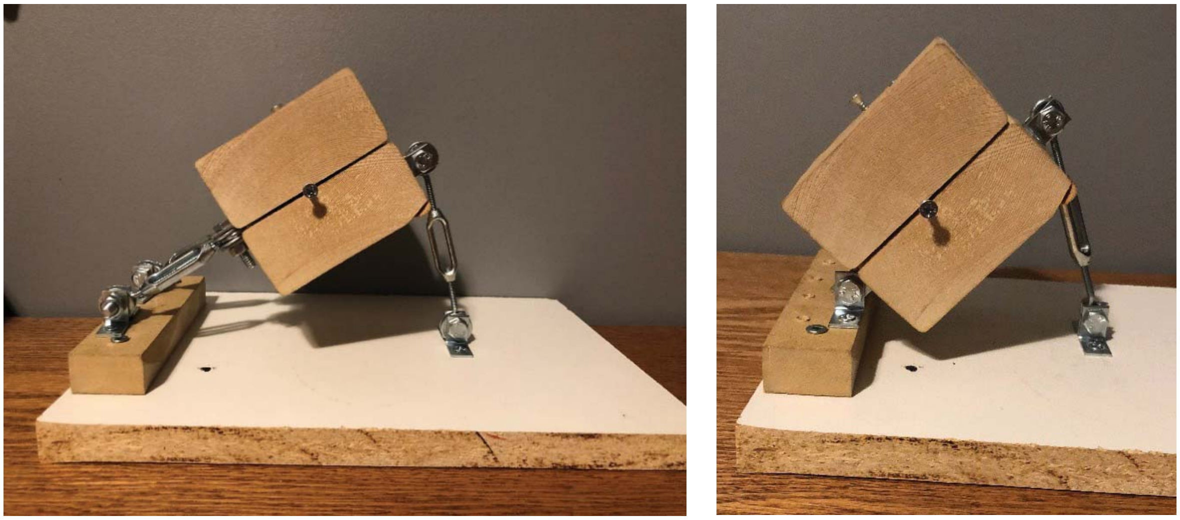


Figure 74. View of the initial (left) and modified (right) physical model.

3.3.2.3.2 Adjustable Rod Mount

The adjustable rod mount was calculated using analytical formulas, then verified using FEA due to the simpler loading scenario. The final adjustable rod mount design is shown in Figure 75.

Figure 75. Render of final adjustable mount design

The threaded rod sections for this mount was selected to be 1”-12 ASTM A193 steel. This thread size was selected by looking at the different sizes and types of material available from a recommended vendor and determining the most effective option. TABLE XXXVI shows the tensile area and calculated stress for different thread sizes.

TABLE XXXVI: TENSILE AREA AND CALCULATED STRESS OF SELECT THREAD SIZES [50]

Thread Size	Tensile Area (in^2)		Calculated Stress (PSI)	
	UNF	UNC	UNF	UNC
3/4	0.373	0.334	65629.32	73292.62
7/8	0.509	0.462	48093.78	52986.44
1	0.663	0.606	36922.68	40395.6
1 1/4	1.073	0.969	22814.29	25262.88
1 1/2	1.581	1.405	15483.7	17423.3

The length of the threaded rod was calculated to be 10” by determining the maximum and minimum lengths needed to achieve the desired operating angles. The maximum length of the arm was calculated to be 34.851” and the minimum length, 27.936”. Therefore, the arm would only require 6.915” of extension to rotate the gearbox between 30 and 60 degrees. These lengths were calculated using the known position of the pivot mount and the dimensions of the Haulmaster. The additional length of the threaded rod is to ensure that a significant section remains threaded to distribute the load and to attach the rod ends. For the proper extension of the arm, the upper arm was designed to have the opposite thread direction.

1”-12 hex nuts were selected out of the same material to allow for the rod to thread into the middle member [51]. These hex nuts were designed to be welded to the middle tube, allowing for the center section to be rotated from the nuts, which would extend both rods equally. As with the threaded rod, the upper nut was designed to have the opposite thread direction for proper operation. The middle tube was selected as a 1.315” diameter tube with a 12-gauge thickness made of ASTM A513. The dimensions were calculated by finding the allowable stress and the known force through the member.

$$d_{in} = d_{out} - 2t,\ \ \sigma_{all.} = \frac{F}{A} = \frac{4F}{\pi({d_{out}}^2 - {d_{in}}^2)}$$

The equation was used with an assumed inner diameter, which allowed for a thickness to be found. The final result was then rounded up to the nearest standard size available by a recommended vendor. The length of the middle tube was determined to be 16” to achieve the maximum and minimum lengths, without the threaded rods contacting. The middle section was designed from steel tube and hex nuts to minimize cost, as designing a custom machined part with the required thread would not be cost-effective.

The rod end was selected from a vendor using the chosen thread size of the threaded rod and the required load. The rod ends were designed to receive a 1” pin, therefore a 1” pin used for this application. The material of the pin was selected to be 1566 carbon steel to attain this diameter, while still also achieving the desired safety factor. Appropriately sized hairpin cotter pins were used to constrain the joint due to their simplicity and the lack of axial loads applied to the pin.

Figure 76. Render of adjustable rod mount focusing on the mounting plate.

The mount plate tabs receiving the pins were designed to be 1” thick and made of A36 steel. Using the desired safety factor of 2, the thickness was calculated to be 0.75” using the bearing stress. However, during the preliminary FEA study performed on these components, the thickness was increased to 1” to provide additional support and alleviate stress concentration factors. Additionally, the tabs were designed to extend across the entire face of the mounting plate, giving additional support in the direction that the rod will be stressed. The distance between the tabs was determined by the width of the rod end. These tabs were then modified to attach to a flat surface and were used as the bottom mounts for the threaded rod assembly. A lifting tab was designed by modifying the larger tab and positioning it at the top of the plate. This lifting tab would allow for the Haulmaster to have a rigid lifting point, reducing the amount of setup time.

For the mount tabs that connect to the skid, the same mount plate tab was used. The only modification was that the overall length of the tab was reduced to shrink the overall footprint of the design.

The mounting plate was designed to have a thickness of 3/4” and made from A36 steel. Holes were added to the plate to allow it to attach to the respective mounting holes on the

Haulmaster gearbox. Additionally, slots were added to the plate so that during manufacturing, the tabs could be welded in the correct position with ease. Preliminary FEA was performed on this plate by fixing the bolt hole locations and applying the load through the previously designed mount plate tabs. The results of this analysis are shown in Figure 77.

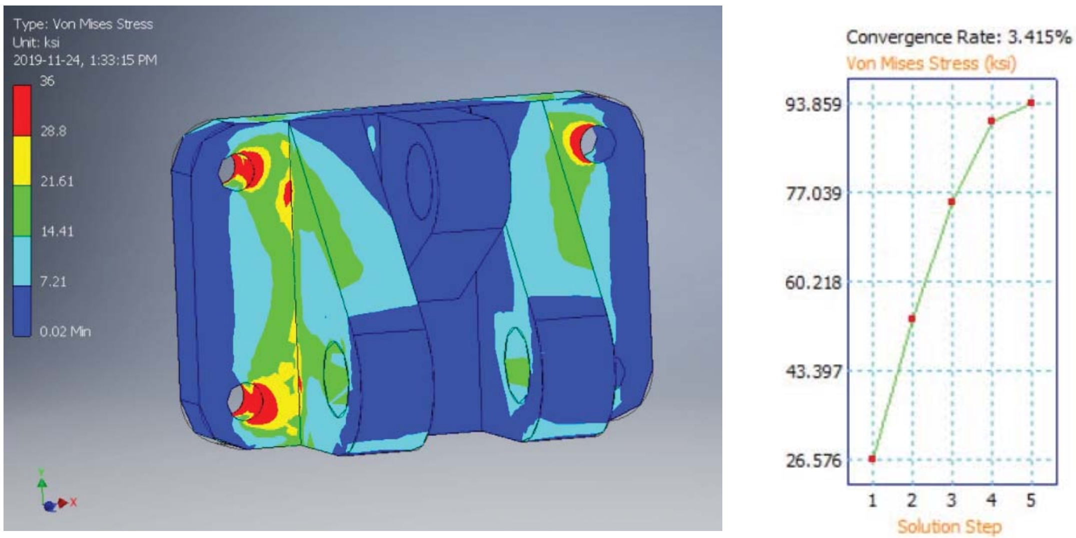


Figure 77. Preliminary finite element analysis results for final adjustable mount plate design.

The stress concentrations shown at the base of the mount plate tab is due to the lack of weldments. It is recommended that the client further analyzes the mounting plate, specifically around the bolt holes.

3.3.2.3.3 Mini-Skid

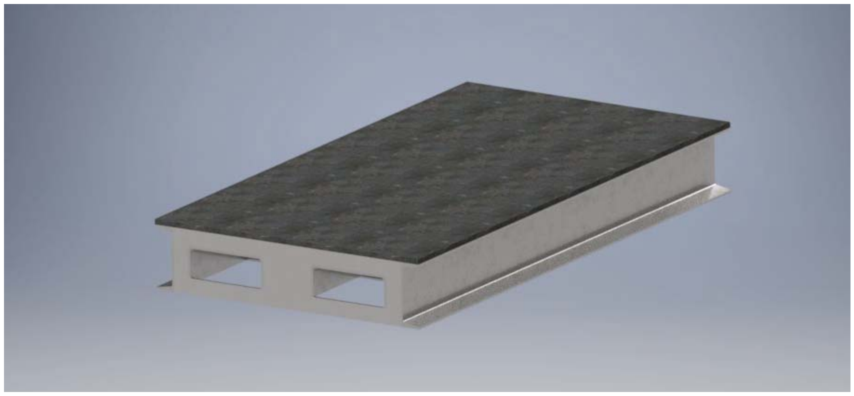


Figure 78. Render of final mini-skid design.

The skid was designed based upon readily available materials from Russel Metals, as they were a recommended vendor for Elmer's. For the base of the skid, two 52" long ANSI C 6X8.2 channel sections were chosen and designed to fit the required dimensions for the mounting components. The length of the channel was determined from the length of the test gearbox mounting components. Four cross supports were designed to be 26" long and designed from the same cross-section of channel used for the sides of the skid. These cross supports feature holes cut through them to house fork pockets which were designed from sheet metal that is bent to shape. These fork pockets extend through the entire skid, allowing the system to be lifted from either side. The skid plate was designed to fit exactly on top of the C-channel base, allowing for a large surface that could have the mounting components welded to. This plate would then be welded to the C-channel base.

Due to scope restrictions, the skid was designed without the use of FEA. It is recommended that the client performs further finite element analysis on the skid, as the complex design would require a more rigorous approach than the team was asked to perform. Therefore, this skid outlines the basic requirements for the design but requires further analysis before being implemented.

Our design was based upon the assumption that the mounting components would be welded to the skid. For testing multiple gearboxes, the skid could be reproduced, and the appropriate mounting components would be welded. The skid was designed to have a gap around the mounting components for the case that the client decided that they would prefer a single skid with removable mounting systems. This gap would allow for the mounting components to have room to be bolted down to the skid. A render displaying the test gearbox mounting system, focusing on the gap is shown in Figure 79.

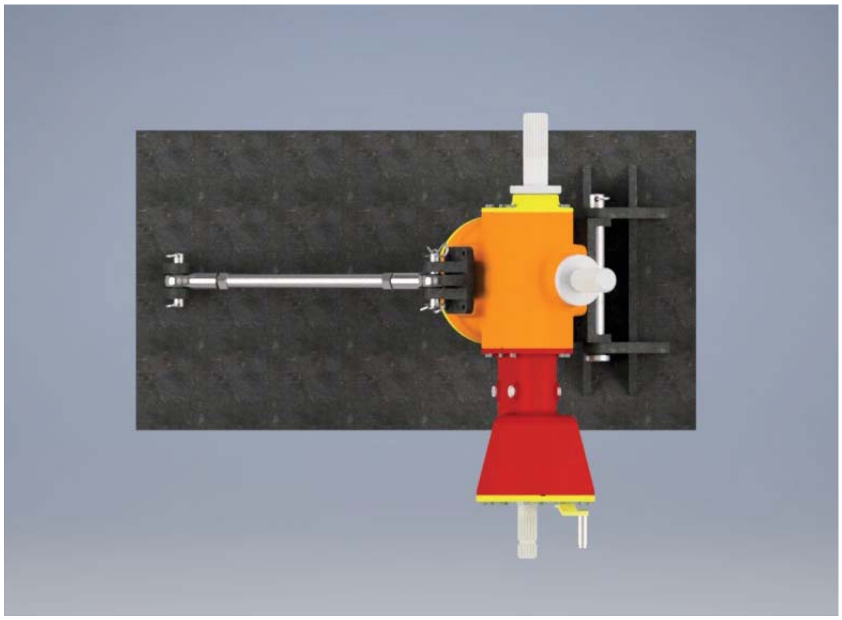


Figure 79. Render of the test gearbox mounting system focusing on mounting components locations on the mini-skid.

3.3.2.4 Hydraulic Pump Mounting

The hydraulic pumps selected require dedicated mounts to ensure that they remain in the correct position while loaded. The scope of the hydraulic pump mounting design is outlined in TABLE XXXVII.

TABLE XXXVII: SCOPE OF HYDRAULIC PUMP MOUNTING DESIGN

In Scope	Out of Scope
Initial FEA study for all assembly components	Detailed validation of designs using FEA studies
Simple force and moment analysis of the major components	Exhaustive manual analysis of all the likely failure modes of the components, including fatigue, fracture and buckling analyses
CAD models of the assembly	CAD models of fasteners

The hydraulic pump mounting assemblies are responsible for orienting and restraining the hydraulic pumps which apply load to the test gearboxes. Two hydraulic pump mount assemblies were designed, one accommodates the pump which loads the unload auger (vertical) output of the Haulmaster gearbox and one accommodates the cross auger (horizontal) output of the Haulmaster gearbox. TABLE XXXVII shows the scope of the design process for these assemblies The two assemblies are shown side by side in Figure 80. The assembly on the right side of Figure 80 is the stationary pump mounting assembly, which houses the hydraulic pump applying load to the cross auger output of the test gearbox. The assembly on the left side of Figure 80 is the adjustable pump mounting assembly. Both assemblies and their components will be discussed in the following subsections.

Figure 80. Render of pump mounting assemblies on a skid.

3.3.2.4.1 Adjustable Pump Mounting

The adjustable pump mounting assembly was designed to cradle the hydraulic pump between two accessory brackets of the hydraulic pump. This assembly was designed to tilt the hydraulic pump along with the test gearbox as the angle of the unloading auger output is adjusted. The tilting mechanism consists of a pivot joint located at the middle set of feet in Figure 80. The tilting action is restrained by a pin joint between the two diagonal members which cradle the pump between them and the two vertical members in the assembly.

Unlike the test gearbox adjustment mechanism, the angle of the adjustable pump mount was designed to be varied in discrete intervals, with each interval corresponding to a 5-degree tilt in the test gearbox. This limitation of the adjustment system means that the unloading auger output of the test gearbox and the input of the increaser gearbox are misaligned for most adjustment settings. However, this misalignment does not cause any issues, as all PTO shafts allow for some offset between the input and output splines due to the use of U-joints at the connection points. This compromise was chosen to simplify the required effort of the operator to adjust the system. The entire assembly is constructed from 5”x3”x0.375” rectangular tube stock made from A500C steel with 50 ksi yield strength.

The following subsections detail the design of each component of the main assembly. Initial FEA studies of each component were performed to ensure that the designs are able to withstand the loads applied to them during operation. The pin joints in the assembly will be referred to as joint A and joint b, as shown in Figure 81.

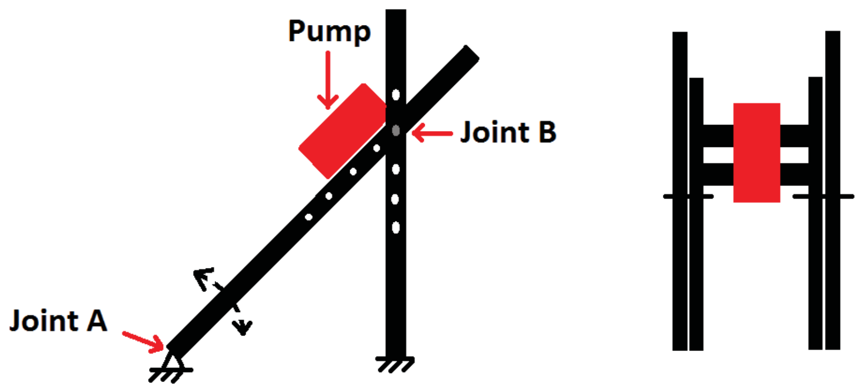


Figure 81. Functional schematic of the adjustable pump mount.

3.3.2.4.1.1 Diagonal Member

Figure 82. Render of diagonal members.

The diagonal member of the adjustable pump mount assembly shown in Figure 82 is a 134” long section of A500C steel rectangular tube. The length was designed to house the hydraulic pump and the increaser gearbox while allowing for a two-foot-long PTO shaft to connect the increaser gearbox to the test gearbox. The dimensions of the tube are 5x3x0.375” and are identical to all the other components in the assembly. The holes along the side of the tube are designed to fit a 0.75” diameter pin, which joins the diagonal member to the vertical member. The holes are spaced such that each location of joint B corresponds to 5 degrees of rotation in the test gearbox.

Initial calculations using sums of moments and forces showed that the diagonal member experiences the highest loads out of the pump mounting assembly and determines the dimension of the rectangular tube material which is chosen. Other than stress considerations, the dimensions of the rectangular tube are limited to those that are easily available to the client. The potential rectangular tube dimensions were chosen from a list of stocked material provided by Elmer’s manufacturing. The material of all the rectangular tube stock on the list is A500C steel with 50 ksi yield strength. A safety factor of 2 is used throughout the design, making the maximum allowable stress 25 ksi. However, before the numerical value of the stress on the diagonal member was calculated, a pin dimension for both joints B and A was chosen.

Since the bearing area depends on the unknown thickness of the rectangular tube, an arbitrary value of 0.75” for the diameter of the pin. The area and second moment of inertia were then tabulated for the entire list of rectangular tube stock provided by Elmer’s and the results were used to calculate bending, bearing and shear stresses. The resultant stresses for each rectangular tube cross-section were compared to the maximum allowable stress of 25ksi. The square tube with the smallest cross-sectional area which exceeded this value was then modeled in Autodesk Inventor and subjected to an FEA static study. When a model failed the static study, the next available rectangular tube cross-section was chosen for the next iteration of the model.

Inventor FEA static studies on the diagonal member were completed using a maximum loading scenario. This scenario occurs when the test gearbox is inclined to 60 degrees, resulting in the lowest possible angle of the diagonal member and the shortest possible distance from joint A to joint B. H-adaptive meshing was used to maximize the accuracy of the static study. Unfortunately, due to the limitations of the computing power available to the design team, a mesh small enough to fit two elements into the thickness of the rectangular tube was not able to be made for the diagonal member, introducing the potential for shear locking. However, the FEA showed a good convergence trend and closely matched the results of the preliminary calculations. von Mises stress distribution, along with a convergence graph of the static study of the model under load is shown in Figure 83.

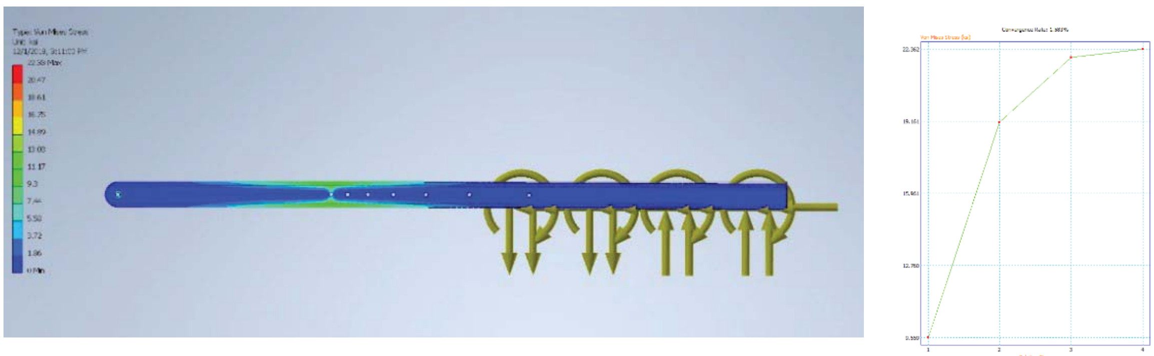


Figure 83. Results of preliminary FEA for the diagonal member.

The rectangular tube with the smallest cross-section which completed the FEA static study without exceeding the allowable stress (excluding stress concentrations at infinitely sharp edges) had the following dimensions: 5x3x0.375”. Using this thickness of the tube, the bearing stress calculated on the pin was found to be well within the allowable stress limit. The pin diameter was increased to 1” in an attempt to reduce the cross-sectional area but increasing the diameter of the pin did not lower the stress in the next smallest available cross-section below the allowable stress. The diameter of the pin was therefore left at 0.75”. Due to concerns about the deflection of the diagonal and vertical members resulting in axil forces being applied to the pin, a bolt will be used as the pin in joint B.

Despite the potential of shear locking, the results of the FEA appear as expected in regard to the stress distribution. There appear to be no significant stress concentrations around the pinholes. The highest stress areas are the concentrations are located at the edges of the bearing pin constraint in joint B, which are reasonable given the sharp edges along the contour of the hole and the forces acting on the joint. The next highest stress location is located at the top and bottom faces of the rectangular tube around joint B, which matches the expected location of the highest bending moment according to preliminary calculations. The stress in all areas is far less than predicted by hand calculations, with a maximum of 22.36 ksi at the bearing stress concentration in joint B. The stress due to bending above joint B approaches 11 ksi. Both of these stress values are below the allowable stress of 25 ksi for the material. The convergence graph shows that the results are well on their way to converging with 1.583% difference between the last two results. Overall, the FEA results match expectations in their distribution and magnitude. The client is advised to perform further FEA with a smaller mesh to confirm these initial findings.

3.3.2.4.1.2 Diagonal Member Pivot Mount

Figure 84. Renders of diagonal member mount and components

The diagonal member pivot mount is designed to resist the forces applied to it by the diagonal member. The design consists of two tabs with holes which hold the joint pin and a base to which they are welded. Figure 84 shows the complete assembly on the left and the individual components on the right. The tabs are designed to be welded to the base. The welding process is aided by the rectangular cutouts in the pivot mount’s base. The cutout in the mount base fits bottom flanges on the mount tabs, allowing easy placement for welding. The base of the mount is dimensioned to position the bolt holes above the C-channels running through the gearbox test unit skid. Both the tabs and the base were designed to be laser-cut from a single 0.375” thick sheet of A36 steel.

The pin for the mount has a diameter of 0.75”, the same as the pin used in joint B in the adjustable pump mount assembly. The pin is restrained from axial motion using a set of retaining rings, which are able to resist the relatively minor axial forces shown to be present in the joint by the FEA of the diagonal member. The two tabs through which the pin is slotted are 0.375” thick, exactly the same as the rectangular tubes used throughout the assembly. Therefore, the bearing area in joint A is identical to the bearing are in joint B. Evaluation of the diagonal member pivot mount assembly was performed using Inventor FEA. The evaluation was performed for two worst-case scenarios. The first scenario was maximum load application during normal operation. The loads for this scenario were derived from the reaction forces found by completing an FEA of the diagonal mounts. Half of the load of the entire adjustable pump mount assembly was applied to the pivot mount in the second case to simulate the completely vertical position of the pump cradle, which could be approached during the adjustment process. The results for the first scenario and its convergence plot are shown in Figure 85.

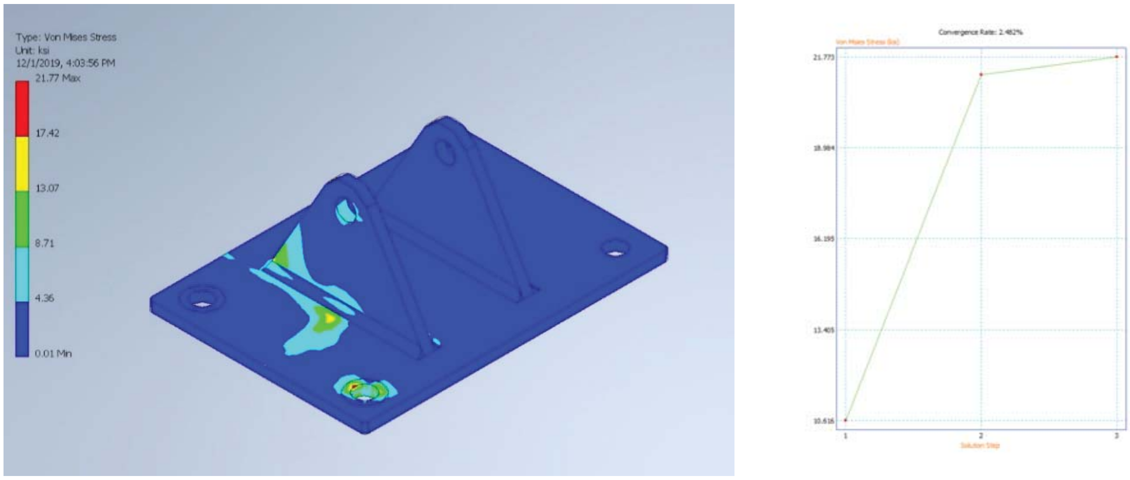


Figure 85. Results of preliminary FEA for the first loading scenario of the diagonal member mount.

The top result in Figure 85 shows the stress distribution in the first scenario. Stresses in this scenario reached as high as 18.13 ksi in stress concentrations around the bolt holes in the base of the assembly. While this value is still 0.02 ksi below the allowable stress for A36 steel, employing washers around the base of the bolt could reduce it significantly be expanding the bearing area of the part. The bolts experienced a maximum of 3.47 ksi tensile stress and insignificant shear forces in this scenario. The part performed well in this scenario, never exceeding the allowable stress for the material. The FEA results for the next scenario are shown in Figure 86.

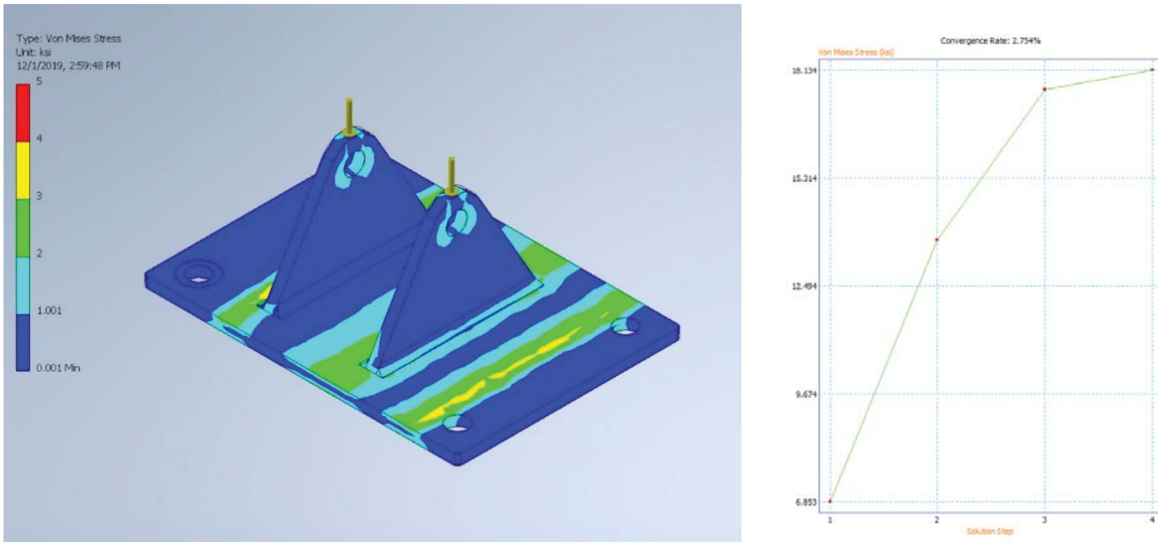


Figure 86. Results of preliminary FEA for the second loading scenario of the diagonal member mount.

FEA results for the second scenario showed a different stress distribution, and a much lower maximum stress value than the first scenario. The maximum observed in the part was less than 4ksi, excluding stress concentrations due to infinitely sharp geometry at the corners of the weld guide holes. Reaction forces observed in the bolt holes resulted in a maximum of 0.38ksi tensile stress on a 0.75” bolt, which is similar to the value found in the first scenario. Overall the FEA results show that the pivot mount design is easily able to sustain a worst-case scenario loading condition. The shape of both of the convergence graphs found in Figure 85 and Figure 86 suggests that the maximum value of von Mises stress is reaching convergence.

3.3.2.4.1.3 Vertical Member

Figure 87. Render of vertical members

The vertical members are shown in Figure 87. These sections of the rectangular tubes are connected to the diagonal members by assembly joint B, thereby restraining the motion of the diagonal members. The dimensions of the cross-section of the vertical mounts were determined by the outcome of FEA and hand calculations completed for the diagonal members as it shares the same cross-section. The hole pattern for the vertical members was designed in tandem with the one for the diagonal members, allowing the holes to align in 5-degree intervals of the test gearbox orientation. Due to excessive moments placed on the vertical member at the most inclined position of the pump a support member was added to carry some of the horizontal load. The dimensions of the support member are designed around the gearbox test unit skid, providing ample clearance for the test gearbox input PTO shaft.

There were no hand calculations performed to determine stress in the vertical member after the addition of the support member. The two members were expected to experience less stress than the diagonal member once joined because the single biggest source of stress in the vertical member was calculated to be the bending moment at its base, which is eliminated by the presence of the support member. A model of the vertical members was analyzed using an FEA study in Autodesk Inventor to confirm this expectation. The results of the study, along with the convergence graph are shown in Figure 88.

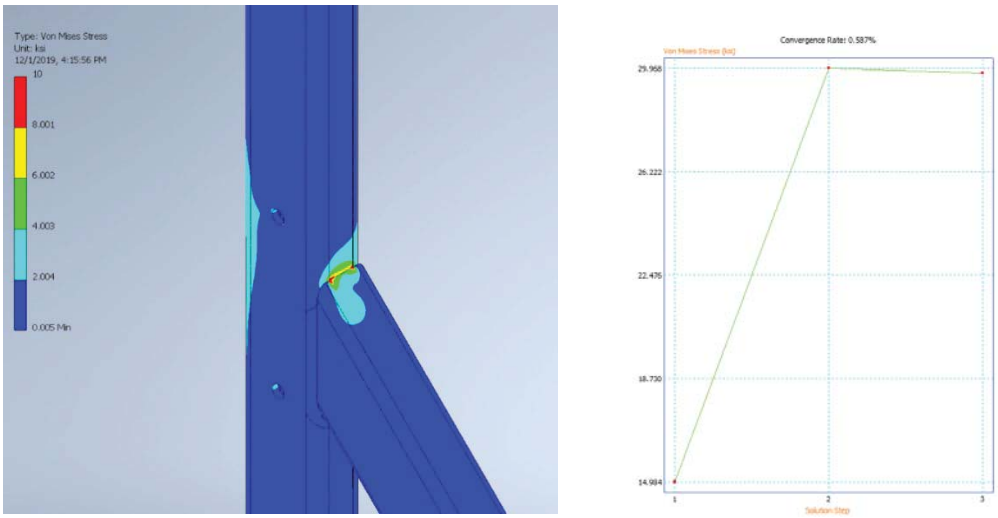


Figure 88. Results of preliminary FEA for the vertical member.

Figure 88 shows the vertical member subjected to the worst-case scenario loading condition according to the preliminary hand calculations. This scenario occurs when the test gearbox is tilted to 30 degrees, resulting in the minimum angle between the vertical and diagonal members, where the maximum amount of horizontal force is transferred. The area of the largest von Mises stress is the stress concentration in the corner of the bond between the support member and the vertical member. The von Mises stress in this area reached above 30ksi. This maximum stress value was interpreted by the design team as an artifact of the infinitely sharp corner of the bond geometry and was ignored.

The general area immediately around the connection, however, was expected to carry a significant amount of stress relative to the rest of the model, as it was intuitively the location of the highest bending moment in the vertical member. The stress in the immediate vicinity of this connection was measured using the probe tool and was found to have a range of 5-10ksi, which is well below the allowable stress of 25 ksi. The next largest area of stress is located on the opposite side of the connection between the vertical and support members. The stress in this area was found to be about 3 ksi, well below the allowable stress. Overall the stress distribution in the vertical member adhered to expectations. The stress gradient in the model, along with the trend of the convergence plot leads the team to trust the FEA analysis on the model. However, as with all the other FEA analyses found in this report, the client is advised to perform further FEA with a smaller mesh to confirm these initial findings.

3.3.2.4.1.4 Vertical Member Base

Figure 89. Renders of the vertical member base and the base of its support.

Figure 89 shows the vertical member base on the left and support member base on the right. These bases were designed to be welded to the rectangular tubes which they support. The perimeter dimensions of the bases for both the vertical member and the vertical member support were designed to position the fastener holes above the C-channel beams in the gearbox test unit skid. Space was provided on both bases for fastener heads to clear the rectangular tube, although the base of the support member had to be shrunk in order to fit the skid thereby limiting available fastener space. Socket head bolts are recommended for the support member base to ensure that no interference takes place between the fasteners and the welded joint. Both of these bases, as well as many other parts in this assembly, can be laser cut from a single 0.375” thick sheet of A36 steel.

The loading condition for both mounts is relatively complex and was not evaluated using hand calculations. Inventor FEA was used to verify that no yielding occurred at the fastener connections and the area in the vicinity of the weld connection for worst-case scenario loading. The worst-case scenario for the vertical base assembly occurs when the test gearbox is inclined to 30 degrees above horizontal, which requires the tallest setting for joint B and results in the largest horizontal force on the member. Figure 90 shows the vertical member base and support member base under worst-case loading, along with convergence graphs of their studies, top, and bottom respectively.

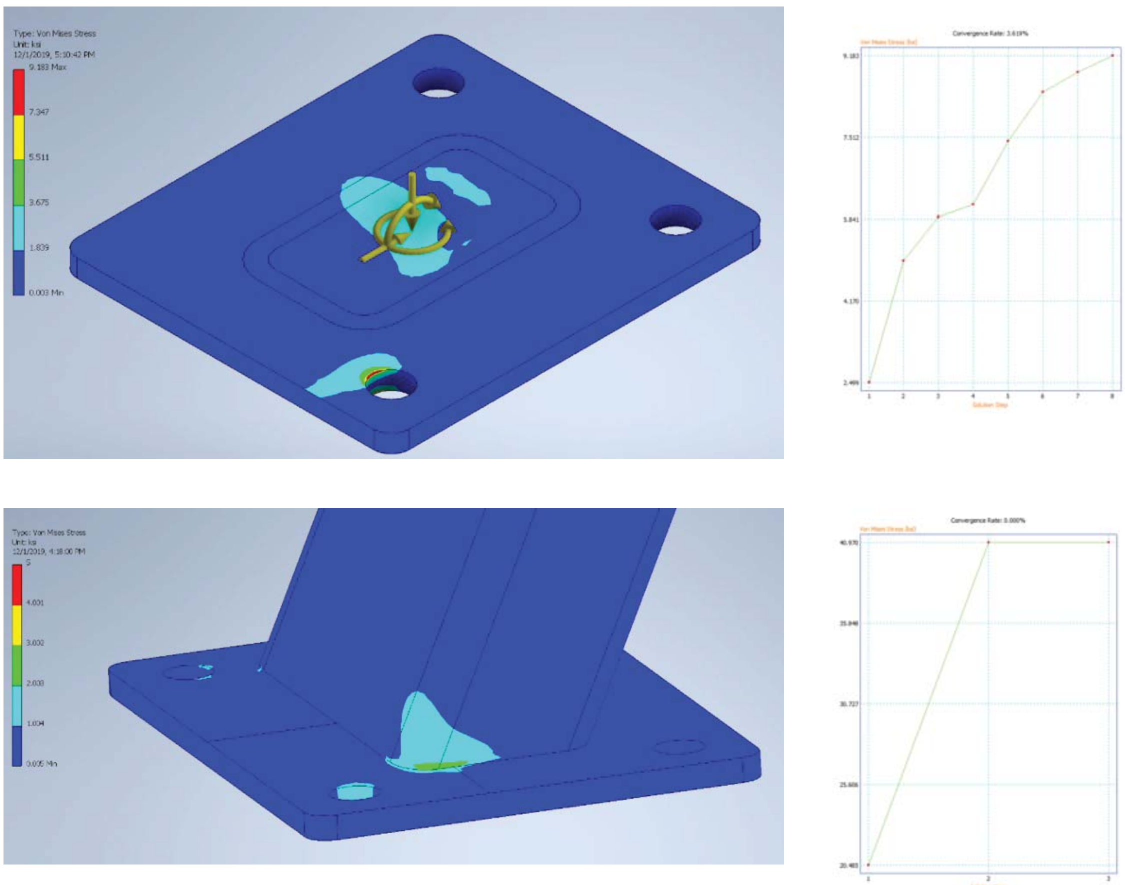


Figure 90. Results of preliminary FEA for support member base and vertical member base.

The bearing stresses about the fasteners for both of the bases were less than 2 ksi, which is well below the yield point of 18 ksi. Reaction forces on the fasteners result in a maximum of 0.48ksi average shear stress and 0.26 ksi tensile stress on the 0.75” diameter fasteners. These values of stress are well below the yield stress of any steel bolt. Overall the results of the FEA show that the base designs can easily resist any loading conditions applied to it during normal operation. The shapes of the convergence graphs suggest that the results of the FEA are in the process of converging at a maximum von Mises stress value nearly identical to that mentioned in this subsection. Additional FEA work is recommended to verify the results of this initial study.

3.3.2.4.1.5 Cross-member

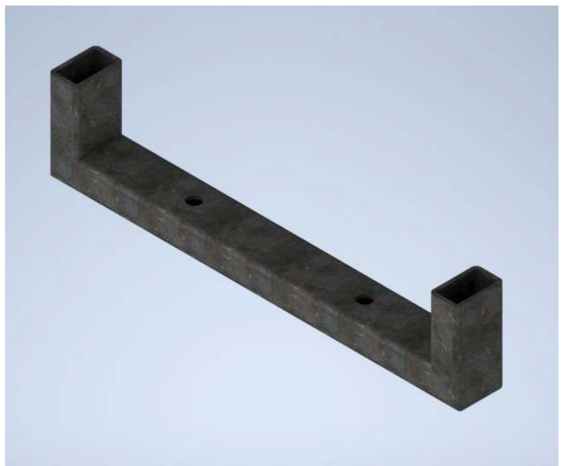


Figure 91. Render of the cross member.

The cross member is shown in Figure 91 and is the mounting point for the pump accessory brackets. Four of these members are present in the design, all using the same 5x3x0.375” A500C steel rectangular tube as the other members in the assembly. This member can either be made by welding vertical pieces of rectangular tube to the horizontal tube or cutting slices out of a single rectangular tube section and bending it into shape prior to welding. The purpose of the bends in the cross member is to position the hydraulic pump inline with the diagonal member. Offsetting the pump mounting location prevents interference between the cross member and joint B.

The cross member was evaluated using an Inventor FEA study for the worst-case loading scenario. The worst-case scenario occurs when the gearbox is tilted to an angle of 60 degrees, resulting in the lowest setting for joint B. The results of the FEA and the associated convergence graph are shown in Figure 92.

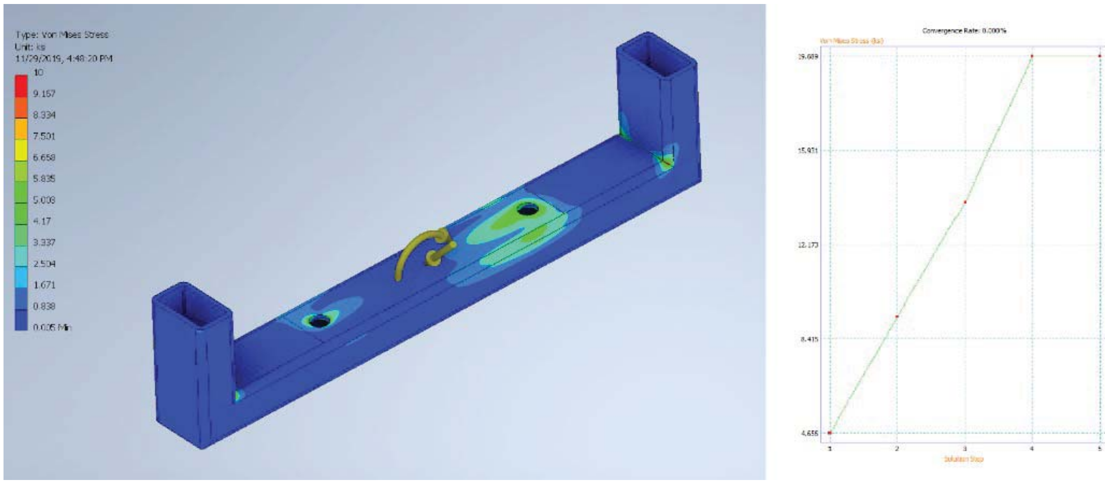


Figure 92. Results of preliminary FEA for the cross member.

The stress distribution shown in Figure 92 is much more complex than revealed by a simple free body diagram. However, the areas of the largest stress concentrations are located around the connection between the vertical and horizontal tubes in the model. While stress concentrations are expected in this area, the maximum stress of 19.69 ksi is likely due to an infinitely sharp corner in the joint. Even so, the maximum stress shown by Inventor is still significantly below 25 ksi allowable stress for the rectangular beam material.

There are also stress concentrations around the holes for the accessory pump bracket fasteners. The stresses in this location do not exceed 7 ksi, which is still well below the allowable yield stress. The convergence graph shown in Figure 92 shows that the study is trending towards convergence. Note that the reaction forces from the bracket fasteners were not applied accurately to the fastener holes for the pump, but rather directly to the face of the part. This concession was made due to time constraints and would have likely resulted in higher values of stress around the fastener holes. While the team does not expect this simplification to make a significant difference to the maximum stress value experienced by the part, further FEA studies should be performed to verify this assumption.

3.3.2.4.2 Stationary Pump Mounting

The stationary pump mount features the pump brackets that were available with the selected pump, with a designed structure to position the pump in the correct position. A render of the final stationary pump mount design is shown in Figure 93.

Figure 93. Render of stationary pump mounting system.

The horizontal plate that the pump brackets attach to was designed to have hole locations matching the pump brackets. The thickness was then determined using preliminary FEA, resulting in a final thickness of 0.5” and was made of A36 steel. The plate was designed so that the speed increaser gearbox was mounted flush with the side of the plate, as the gearbox was larger than the mounting brackets and otherwise interfered. The plates acting as the feet for the design were created to have a height of 14.2”. This height located the pumps at the center of the possible locations of the output shaft.

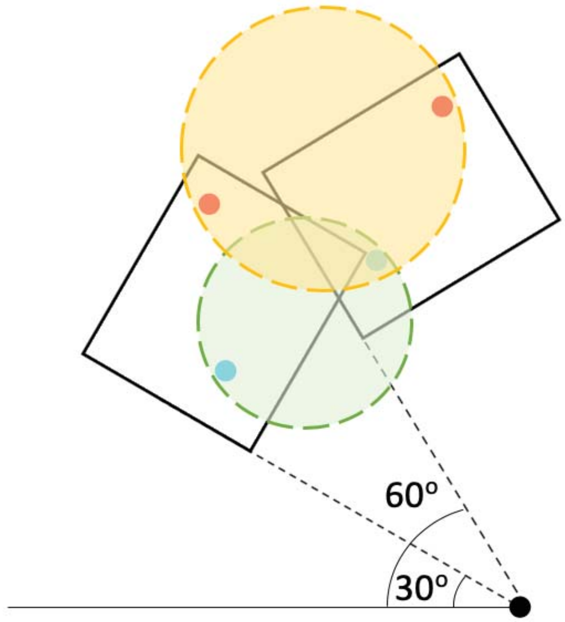
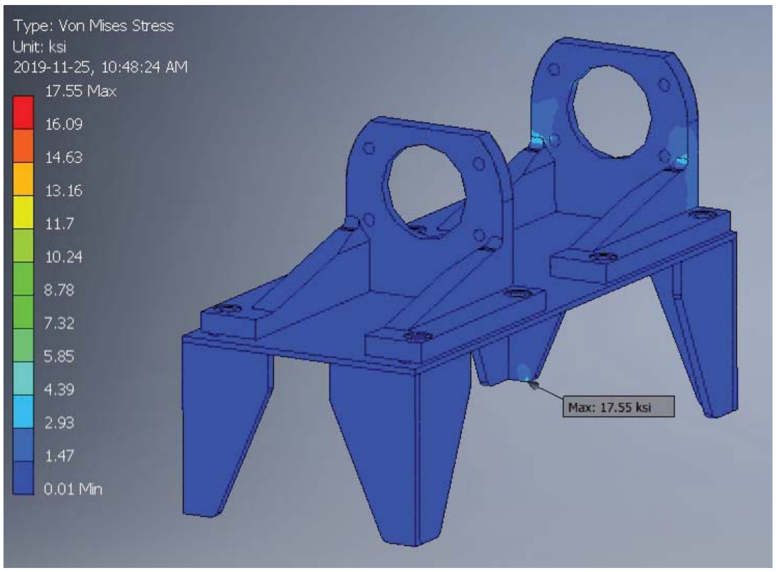


Figure 94. Diagram of input and output shaft movement for varying operating angles.

As the gearbox rotates, the positions of the input and output shafts will change. The yellow circle represents the area that the PTO shaft must be able to connect to the gearbox output. Therefore, by designing the input shaft of the pump to be at the center of the possible test gearbox output positions, the misalignment of the PTO was minimized. The footplates were designed to widen towards the top, giving additional strength. Also, the feet were designed to be made of two plates welded together at each corner of the horizontal plate to increase the transverse stability. As with the other sheet metal parts, these components were designed out of A36 steel and these specific components were designed with a thickness of 0.75”, determined through finite element analysis. These components were then designed to be welded together, and the pump brackets bolted onto the plate creating the final assembly.

Additionally, the distance from the side of the skid to the side of the pump mount was determined to be 14.6”. This distance will locate the pump in the horizontal direction to minimize the PTO misalignment.

This pump assembly was analyzed using some initial FEA by placing the expected torque applied to the pump and gearbox, along with the weights, onto the mounting brackets. The results of the FEA study and a convergence plot are shown in Figure 95.



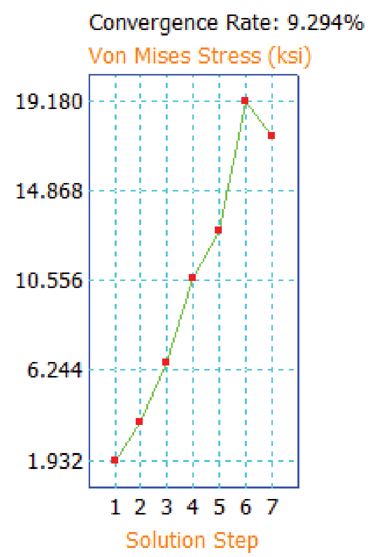


Figure 95. Preliminary finite element analysis results for stationary pump mounting design.

As with the other assemblies analyzed using FEA, this study was preliminary, and the results did not completely converge. Therefore, the results of this study are not to be taken as completely accurate, but rather as a preliminary view on how the team expects the system to behave. Additionally, some of the stress concentrations shown in the results are due to the lack of fillets that actually are present in the actual design. These fillets are incorporated through welding the mounting system to the overall gearbox test unit skid.

3.3.2.5 Skids

Two skids are used to contain the components of the gearbox test unit, namely the gearbox test unit skid and the hydraulic systems skid. The gearbox test unit skid is designed to hold the client's gearbox at the desired angle during testing, the hydraulic pumps with their respective mounts, and the control valve assemblies. The hydraulic systems skid contains the components required to keep the hydraulic oil within its operating parameters, specifically the reservoir, filters, and cooler. A render of the gearbox test unit skid and the hydraulic system skid is shown in Figure 96.

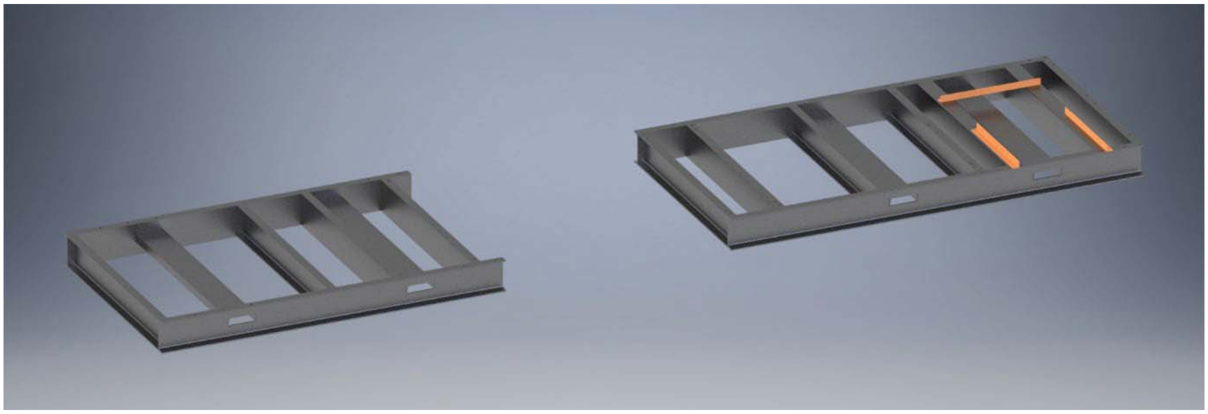


Figure 96. Render of the gearbox test unit and hydraulic system skids.

TABLE XXXVIII outlines the scope of the skid designs.

TABLE XXXVIII: SCOPE OF GEARBOX TEST UNIT SKID DESIGN

In Scope	Out of Scope
Initial layout of components and assemblies on skids	Detailed analysis using FEA
Structural channel selection for all skid members	Sourcing of PTO shafts between components
	Detailed placement of fork pockets

The first step in designing the skid for the gearbox test unit was to determine how many skids were required based on the component weights. As metric 4 requires that the total weight of each unit remain under 9000 lbs, the combined weights of the components determined how many skids are required. An estimated total weight breakdown of the components is shown in TABLE XXXIX.

TABLE XXXIX: APPROXIMATE WEIGHTS OF COMPONENTS WITHIN THE GEARBOX TEST STAND

Component	Weight (lbs)	Quantity	Total Weight (lbs)
Hydraulic Oil	7.2 (lb/gal)	600 gal	4320
K3VG280DT Pump	660	2	1320
Control Valve Assembly	192	2	384
Oil Filter	48.5	2	97
OCA-3100 Oil Cooler	1806	1	1806
Oil Reservoir	2460	1	2460
Elmer's Test Gearbox	450	1	450
Speed Increaser	790	2	1580
Stationary Pump Mount	675	1	675
Adjustable Pump Mount	1635	1	1635
Mini-Skid	509	1	509
		Total	**15,236**

From TABLE XXXIX, it is evident that designing one skid to hold all of the components and remain under 9000 lbs is impossible as the combined weight of only the major components is over 15,000 lbs. Therefore, two skids are required for the gearbox test unit.

3.3.2.5.1 Gearbox Test Unit Skid

The gearbox test unit skid, shown in Figure 97, was designed based upon the components that it was designed to contain. Cross members were designed in specific locations to ensure that the valves, pump mounting assemblies, as well as the gearbox mounting system, are positioned correctly.

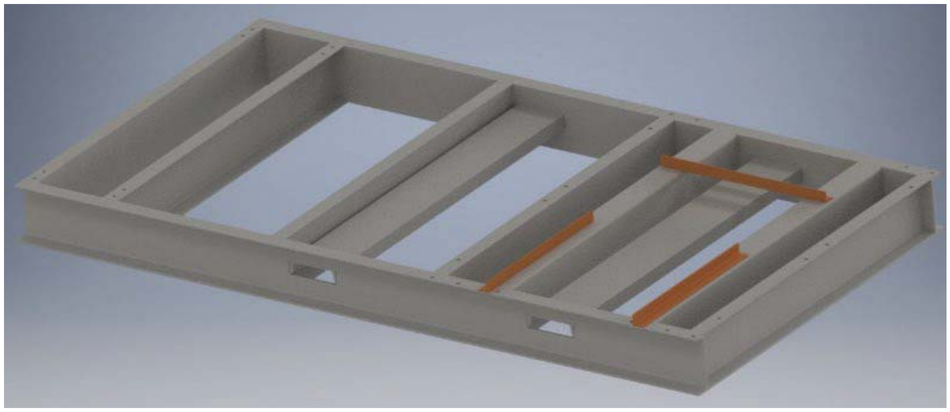


Figure 97. Render of gearbox test unit skid design.

As with the previous skid designs, the procedure for designing this skid began with determining the maximum required dimensions based on the dimensions of the components on the skid. The mini-skid was positioned such that the input of the test gearbox is pointed to an open end of the skid. The adjustable pump mount assembly, including the pump with an auxiliary gear pump and a speed increaser gearbox, was placed around the mini-skid in a position allowing for the adjustment of the angle of the hydraulic pump and for a PTO shaft to be connected to the test gearbox input. These two assemblies set the width of the skid to 70.0”. Next, the stationary pump mount, pump, and speed increaser gearbox were aligned with the center position of the maximum and minimum testing angles, as previously shown in Figure 94. A distance of 30.7” was set between the rear output of the test gearbox and the input to the speed increaser gearbox on the stationary pump mount to allow for standard length PTO shaft to be implemented by Elmer’s. Lastly, both control valve assemblies were placed on either side of the stationary pump mount, accounting for the minimum bend radii of the hydraulic lines connecting the pumps to the control valve assemblies.

The main rails and cross members of the skid were chosen to be constructed from C10x20 channel, of which the design took inspiration from both the diesel engine skid from Deutz and the mini-skid. Six c-channel cross members are used to allow for the mounting of the stationary and adjustable pump mounts, and the control valve assembly. Two rectangular tubes, 10x6x3/8 are also used to allow for the mini-skid to be set upon the main skid in a secure fashion. Two 2”x2”x1/4” angle irons are placed in parallel, 28” apart, on the top of the rectangular tubes with a third angle iron spanning the gap between the two parallel angle irons. These angle irons create an aligning cradle, allowing for precise placement of the mini-skid. This cradle is shown in Figure 98.

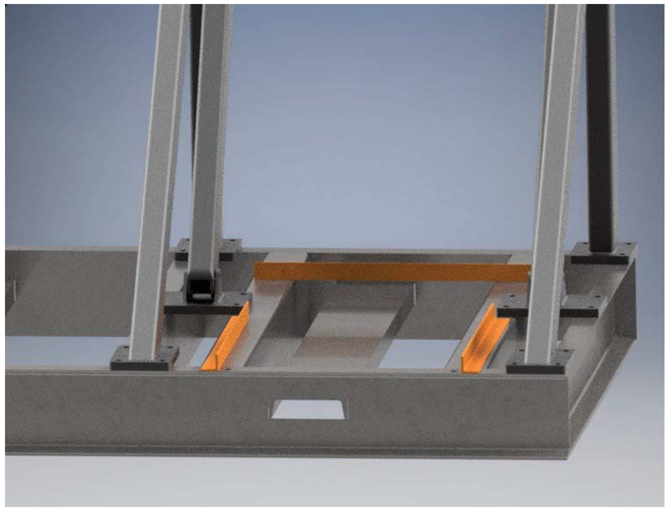


Figure 98. Render of mini-skid cradle highlighted by orange aligning angle-irons.

To allow for a forklift to move the gearbox test unit skid two C9x13.4 c-channels are placed facing downwards spanning the two main rails of the skid, creating fork pockets for a forklift. The pockets are placed with a center-to-center distance of 48” apart. Ideally, the fork pockets should be centered under the center of mass, however, the cross members of the skid dictated the placement of the fork pockets. The rectangular tubes used to support the mini-skid interfered with the placement of the fork pockets. Therefore, the pockets were moved between the two rectangular tubes and one of the c-channel cross members.

Using Inventor’s mass calculator to determine the weight of the main skid members, the total weight of the gearbox test unit including the mounted assemblies was then calculated and is shown in TABLE XL.

TABLE XL: GEARBOX TEST UNIT SKID COMPONENT WEIGHTS

Component	Weight (lbs)	Quantity	Total Weight (lbs)
K3VG280DT Pump	660	2	1320
Control Valve Assembly	192	2	384
Elmer's Test Gearbox	450	1	450
Speed Increaser	790	2	1580
Stationary Pump Mount	675	1	675
Adjustable Pump Mount	1635	1	1635
Mini-Skid	509	1	509
Skid Main Rails	230	2	460
Skid C-Channel Cross Members	115	6	690
Skid Rect. Tube Cross Members	130	2	260
Aligning Angle Irons	8	3	24
		Total	**7987**

As the total weight of the skid of 7987 lbs is under the target weight of 9000 lbs, the skid containing the components required to load the test gearbox is able to be moved by a forklift.

3.3.2.5.2 Hydraulic System Skid

The hydraulic system skid contains the critical components to ensure the oil in the hydraulic dynamometer is properly maintained at operating conditions. Components on the hydraulic system skid include the oil filters, cooler, and reservoir. The hydraulic system skid is shown in Figure 99.

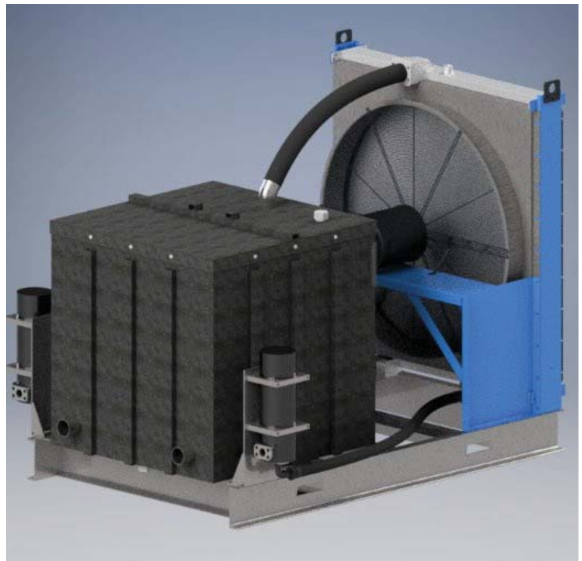


Figure 99. Render of the hydraulic system skid.

Following a similar procedure as used to design the gearbox test unit skid, the size of the skid was determined by the maximum dimensions of the components on the skid. The skid width was set to 68” to match the oil cooler as the cooler is the widest component to be mounted to the skid. The length of the skid was then set to 110” to allow for the placement of the oil reservoir, the cooler, and to allow for adequate room between the cooler and reservoir. The space between the cooler and reservoir is required to allow for hydraulic hoses to reach the inlet to the cooler without exceeding the minimum bend radii of the hydraulic hoses.

On the skid, the cooler was oriented such that the airflow through the cooler is blown away from the reservoir. This was done to avoid heat transfer to the oil through convection in the hydraulic reservoir. In the orientation shown in Figure 99, the hot air discharged from the oil cooler is blown away from the entire system. The reservoir was mounted to the skid with both suction lines on the reservoir facing the same direction as the inlets to the cooler. Doing so ensured that all hydraulic hoses connecting the components of the hydraulic system skid and the gearbox test unit skid are routed in the same direction. Lastly, one filter and its mounting bracket are placed on each side of the oil reservoir, allowing for in-line filtering of the hydraulic oil between the control valve assembly and the oil cooler.

The hydraulic system skid frame is constructed solely of C10x20 c-channel. Four cross members are used between the two skid rails to create mounting points for the reservoir, cooler, and filters. Like the gearbox test unit skid, two C9x13.4 c-channels are placed facing downwards spanning the two main rails of the skid, creating fork pockets for a forklift. As none of the cross members in the hydraulic system skid are critical to the mounting of components, the cross members were able to be adjusted to allow room for fork pockets to be placed exactly under the skid's center of gravity. The assembled frame of the hydraulic system skid is shown in Figure 100.

Figure 100. Render of the hydraulic system skid frame members.

Using Inventor's mass calculator to determine the weight of the main skid members, the total weight of the hydraulic system skid is summarized in 12TABLE XLI.

12TABLE XLI: HYDRAULIC SYSTEM SKID COMPONENT WEIGHTS

Component	Weight (lbs)	Quantity	Total Weight (lbs)
Hydraulic Oil	7.2 (lb/gal)	600 gal	4320
Oil Filter	48.5	2	97
OCA-3100 Oil Cooler	1806	1	1806
Oil Reservoir	2460	1	1819
Skid Main Rails	192	2	384
Skid C-Channel Cross Members	117	4	468
		Total	**8894**

As the total weight of the skid is under the target weight of 9000 lbs, the skid containing the components required to maintain the hydraulic oil during operation is able to be moved by a forklift.

3.3.3 Connection of Power Unit and Gearbox Test Unit

To connect the power unit to the gearbox test stand, the design team chose a PTO shaft with a slip clutch, as determined during the concept selection process. After presenting this idea and getting approval from Elmer's on using a PTO shaft with a slip clutch, the client suggested that their current PTO shaft supplier was used, as they have an option for the application within the gearbox test stand. The client also stated that they are able to handle sourcing the PTO shaft and for the design team to leave the sourcing and specifying the shaft out of the scope. Elmer's requested that the design team ensured that the PTO output spline of the power unit was a standard spline, which was achieved in Section 3.3.1.3, and Elmer's would be able to make adapters at a later date to fit the sourced PTO shaft.

4 Summary

With the design completed, the team developed an initial cost estimate, reflected on the design process to determine the achieved metrics, recommendations for the client, and a final conclusion.

4.1 Initial Cost Estimate

An initial cost analysis was performed using prices of major purchased components within the design and using Elmer's internal cost estimation technique for bulk steel components, costing $2 per pound of material. The initial cost estimate is shown in TABLE XLII.

TABLE XLII: INITIAL COST ESTIMATE.

Component	Cost ($CAD)
TCD 13.5 Diesel Engine [10]	85,000
K3VG280DT Pump [52]	55,000
K3VG280DT Pump with Auxiliary Pump [52]	60,000
OCA-3100 Oil Cooler [31]	18,000
Hydraulic Oil [17]	5595
Fasteners	1000
Hydraulic Hoses, Fittings, and Connections	5000
Speed Reducer Gearbox	3000
Speed Increaser Gearboxes	20,000
Approx. 8500 lbs Steel Components	17,000
Total	**269,595**

From TABLE XLII, the estimated cost of the gearbox test stand is approximately $270,000 CAD. In the initial cost estimate, exact prices for fasteners, hydraulic lines, and speed increaser gearboxes were unavailable. Approximate values for these components were created based on the design team's previous experience. It is noted that the design will likely cost more than the listed $270,000 due to the procurement of all purchased parts and raw material, as well as the cost of labour to fabricate and assemble the gearbox test stand.

4.2 Achieved Metrics

To ensure that the final design met the established metrics in TABLE IV, the achieved specifications of the final design were compared to the target and allowable metrics. This comparison is shown in TABLE XLIII.

TABLE XLIII: FINAL DESIGN ACHIEVED METRICS

Metric Number	Associated Needs	Metric	Units	Target	Allowable	Achieved
1	2.1	Maximum power output	HP	600	600-700	600
2	1.1	Movable with forklift	Binary	Yes	Yes	Yes
3	1.2	Overall dimensions	inch	H x W x L 151x167x 447	H x W x L 216x167x500	H x W x L 124x80x132
4	1.3	Maximum weight of each unit	lbs	<8000	<9000	<9000
5	1.5, 4.2	Units can be operated independently	Binary	Yes	Yes	Yes
6	3.1	Maximum PTO speed	rpm	1000	1000	1000

Metric Number	Associated Needs	Metric	Units	Target	Allowable	Achieved
7	3.2	Range of power absorbed	HP	0-800	0-600	0-684
8	5.1	Total cost	$CAD	<$100k	<$200k	>$270k
9	5.2	Lead time	Business Days	<30	<60	N.A.
10	5.2	Cost to acquire raw materials	% of total budget	<10%	<20%	N.A.
11	4.1	Lowest starting temperature	°C	-15°C	-10°C	-10°C
12	4.1	Resistance to elements	IP Rating	IP69K	IP68	N.A.
13	3.3	Range of operating angle	Degrees	0-60°	45°	30°-60°
14	3.4	Number of gearbox types	N.A.	3+	1	1
15	2.2	Number of external connections	N.A.	0	0	0
16	4.3	Components are accessible	Subjective	N.A.	N.A.	N.A.
17	4.4	Corrosion resistance	Years	20	10	N.A.
18	4.4	Lifetime	Hours	5000	2500	5000
19	4.4	Total number of starts	Cycles	45000	15000	45000
20	1.4	Amplitude of vibrations	mm	1	5	N.A.

Metric Number	Associated Needs	Metric	Units	Target	Allowable	Achieved
21	1.4	Maximum center of mass height	m	<1	<1.5	0.9
22	4.5	Setup time	Minutes	10	60	N.A.
23	4.5	Simple user interfaces	Subjective	N.A.	N.A.	N.A.
24	4.6	Noise Levels	dB	<85	<100	N.A.
25	4.6	Emissions Level	Tier Rating	Tier 4	Tier 2	Tier 4
26	4.6	Safety guards	Binary	Yes	No	Yes
27	4.6	Kill switch	Binary	Yes	Yes	Yes
28	4.6	Fire suppression	Binary	Yes	No	No
29	4.6	Resistance to overloading components	Binary	Yes	Yes	Yes
30	4.7	Running time	Hours	8	5	10

From TABLE XLIII, the final design of the gearbox test stand meets all calculable allowable metrics with the exception of metric 8: total cost. The design team worked hard to ensure the metrics were considered throughout the project, continuously referring back to the metric table. However, by ensuring that the certain allowable metrics were hit, the estimated cost of the design went up.

The design is capable of applying a 600 HP load to the client's gearbox through the use of a diesel power unit and a hydraulic dynamometer. These two components also ensure that the number of external connections of the entire gearbox test stand is maintained at zero. The design is also movable by a forklift as the physical size of the largest module is less than the target size listed in metric 3 and the weight of each module is less than 9000 lbs. The clutch on the power unit and the PTO connections between the client's test gearbox and the hydraulic dynamometer ensure that the design's units are independently operable.

To achieve a 1000 rpm output of the power unit, a speed reducer gearbox was designed. Likewise, to operate the hydraulic pumps, custom speed increaser gearboxes were designed. All gearboxes of the system were designed for 5000 hours of operation, corresponding to 45,000 start-stop cycles.

Due to scope changes, the gearbox test stand accommodates only the Haulmaster gearbox, however, the target angle adjustment range of 0-60° is maintained, allowing for Elmer's to implement a mounting system for the Wolverine gearbox in the future.

4.3 Recommendations

The gearbox test stand design project focused on providing a preliminary design that satisfied the client's needs. The design team made a number of concessions due to the limited time allowed for the project. This section detai

ls six recommendations from the design team for future work on this project. Note that the contents of this subsection do not represent an exhaustive list of changes that are required for the validation of the design.

The first recommendation relates to the accommodation of the Wolverine ditcher gearbox. The Wolverine was removed from the scope project during the design phase of the project, as elaborated upon in Section 1.3 of this report. Should the client desire to test the Wolverine gearbox in the future, the team recommends that only a separate mini skid that accommodates both the adjustable pump mount and gearbox test unit skid is designed. All other design elements of the gearbox test stand are compatible with the Wolverine gearbox. The design of the mini skid will be straightforward as, unlike the Haulmaster gearbox, the angle of the Wolverine gearbox output does not need to be varied.

The second recommendation is that the skid designs outlined in this report are thoroughly evaluated using FEA prior to the implementation of the final design. Due to time restrictions, the thorough verification of the skid design was removed from the scope. As such, no theoretical calculations, or FEA studies were completed to verify the skid designs.

The third recommendation is that the design work detailed in this report is further investigated by fatigue and fracture analyses, as well as more thorough FEA studies for each of the project components prior to the implementation of the design. Fatigue and fracture analyses were omitted from the project due to time constraints placed upon the design process. However, fatigue is likely to have a significant impact on the lifetime of various assemblies in the gearbox test stand due to the cyclic nature of the test procedure. The severity of the impact should be analyzed to assure that no changes to the design are necessary. Furthermore, the FEA studies presented in this report are only meant as preliminary analysis rather than complete and thorough verification of the designs. Additional studies should be performed which more accurately replicate the loads and constraints applied to each component.

The fourth recommendation is to further develop and analyze the reduction and increaser gearboxes. It is recommended that Elmer’s develops lubrication systems for both gearbox designs. Both the reducer and increaser gearbox designs contain rotating components that require lubrication. However, analyses of the lubrication requirements were not performed for the gearboxes due to time constraints. Determination of the type and method of lubrication for these components is essential to ensure a reasonable lifetime. Additionally, Elmer’s should perform analysis of the forces on the increaser gearbox casing and carrier to verify the design and make improvements where necessary. Elmer’s should also select retaining rings for the increaser gearbox and check that the addition of retaining ring grooves to the input and output shafts will not cause failure of the shafts. For the speed reducer gearbox, it is recommended that Elmer’s verify that the bearings and the shaft seals from the Haulmaster gearbox can deal with the shaft speeds of the reduction gearbox, and make changes if necessary.

The fifth recommendation is that proper tolerancing for all the designed components is performed prior to manufacturing. Only preliminary drawings were developed for all the designs presented in this report in accordance with the client request. These drawings do not adhere to any engineering drawing standards and may require further dimensioning and tolerancing for manufacturing.

The last recommendation is that a safety procedure and a list of personal protective equipment (PPE) is developed for the design prior to operation. The safe operating procedure is paramount with large scale, high power output machinery like the gearbox test stand. Improper procedures and lack of PPE can lead to injuries and death in certain scenarios. The risk of these scenarios should be minimized as much as possible prior to the operation of the final design.

4.4 Conclusion

The gearbox test stand project was devised as an improvement of Elmer’s manufacturing current testing methodology for their industrial agricultural equipment gearboxes. The gearbox test stand reduces land area required for testing, eliminates the scheduling conflicts involved in borrowing a tractor from Elmer’s farming operations, removes the maintenance costs resulting from using tractors for testing, and reduces the testing time. The design proposed in this report consists of a power unit skid, gearbox test unit skid, the hydraulic system skid, fuel tank skid, and gearbox mini skid. The complete system is shown in Figure 101.

Figure 101. Render of the designed gearbox test stand.

The final design was developed using a consistent and rigorous design methodology to ensure that the result met the client's needs. Throughout this process, the design was verified through preliminary FEA, while keeping the client involved to ensure all stakeholders were being considered.

Due to time constraints, there are certain deviations from the original scope. These include the omissions of designing the Wolverine ditcher gearbox skid and the design of the skid structure. Excluding these items, the team has provided all of the deliverables, including CAD models of the final design, preliminary engineering drawings overall assembly drawings and a detailed bill of materials (BOM) of the final design a list of purchased parts for the final design, and a cost estimate for the final design. All of the metrics developed for this project were achieved with the exception of the internal maximum budget of $200,000 CAD. The final price for the entire assembly as proposed in this report is $270,000. Having provided an initial design for the gearbox test stand, the design team looks forward to the verification and validation stages of the project.

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Appendix A

Assembly Drawings and Preliminary Engineering Drawings

Table of Contents

1 Introduction

Appendix A contains all of the assembly drawings of the main components of the designed gearbox test stand for Elmer’s Manufacturing along with the preliminary Engineering drawings of the parts under each assembly.

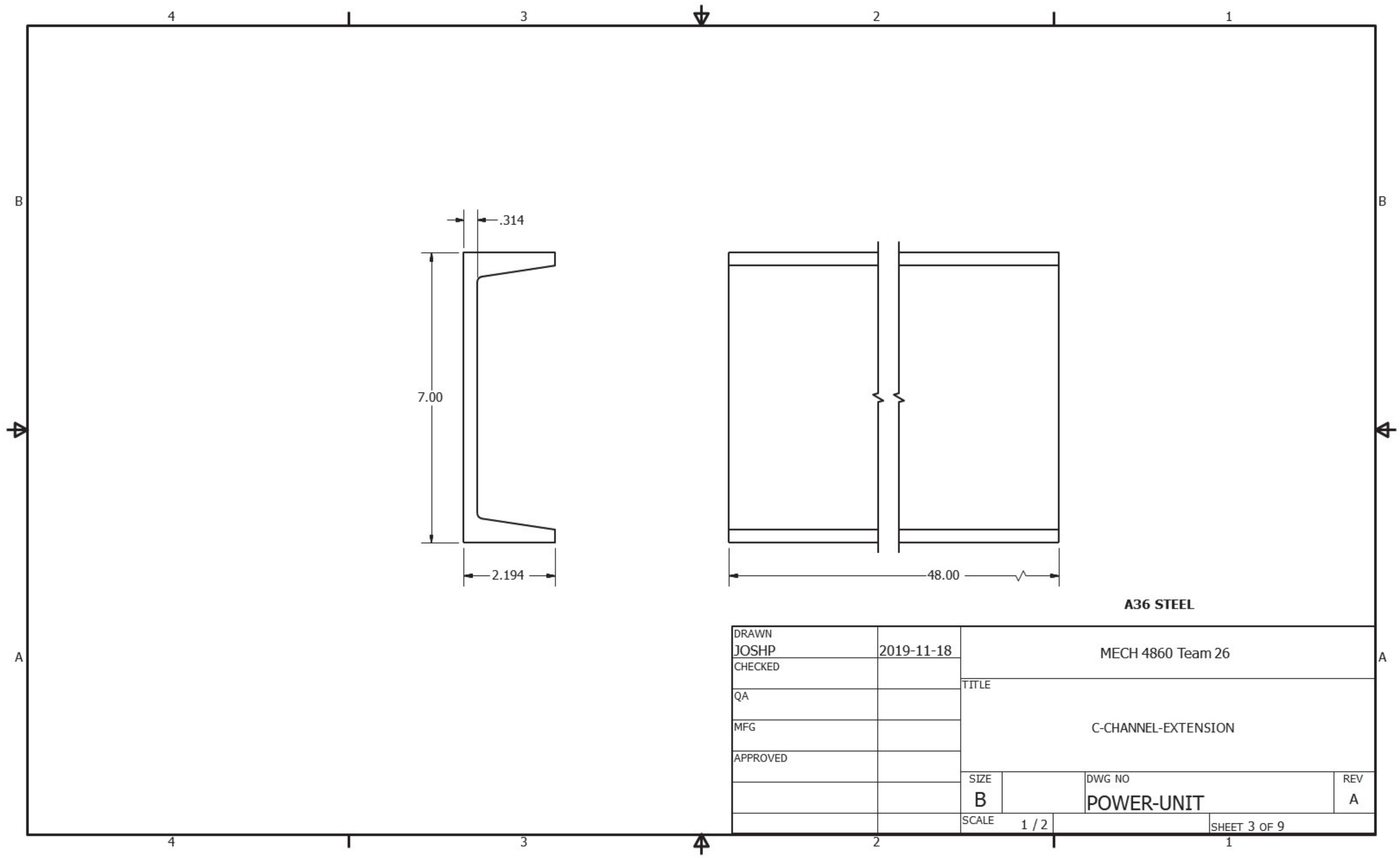
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	TCD 13.5	
2	1	IB314P103	
3	1	MOUNT-PLATE	
4	2	C-CHANNEL-EXTENSION	
5	2	CONNECTING-PLATE	
6	1	POWER-UNIT-FORK-CHANNEL	
7	2	POWER-UNIT-FORK-PLATE	
8	1	REDUCER-GEARBOX	
9	1	MOUNT-PLATE-B	
10	1	CLUTCH-ADAPTER	

DRAWN JOSHP	2019-11-18	MECH 4860 Team 26
CHECKED		TITLE
QA		EXPLODED VIEW OF POWER UNIT
MFG		
APPROVED		SIZE B — DWG NO POWER-UNIT — REV A
		SCALE 1 / 20 — SHEET 1 OF 9

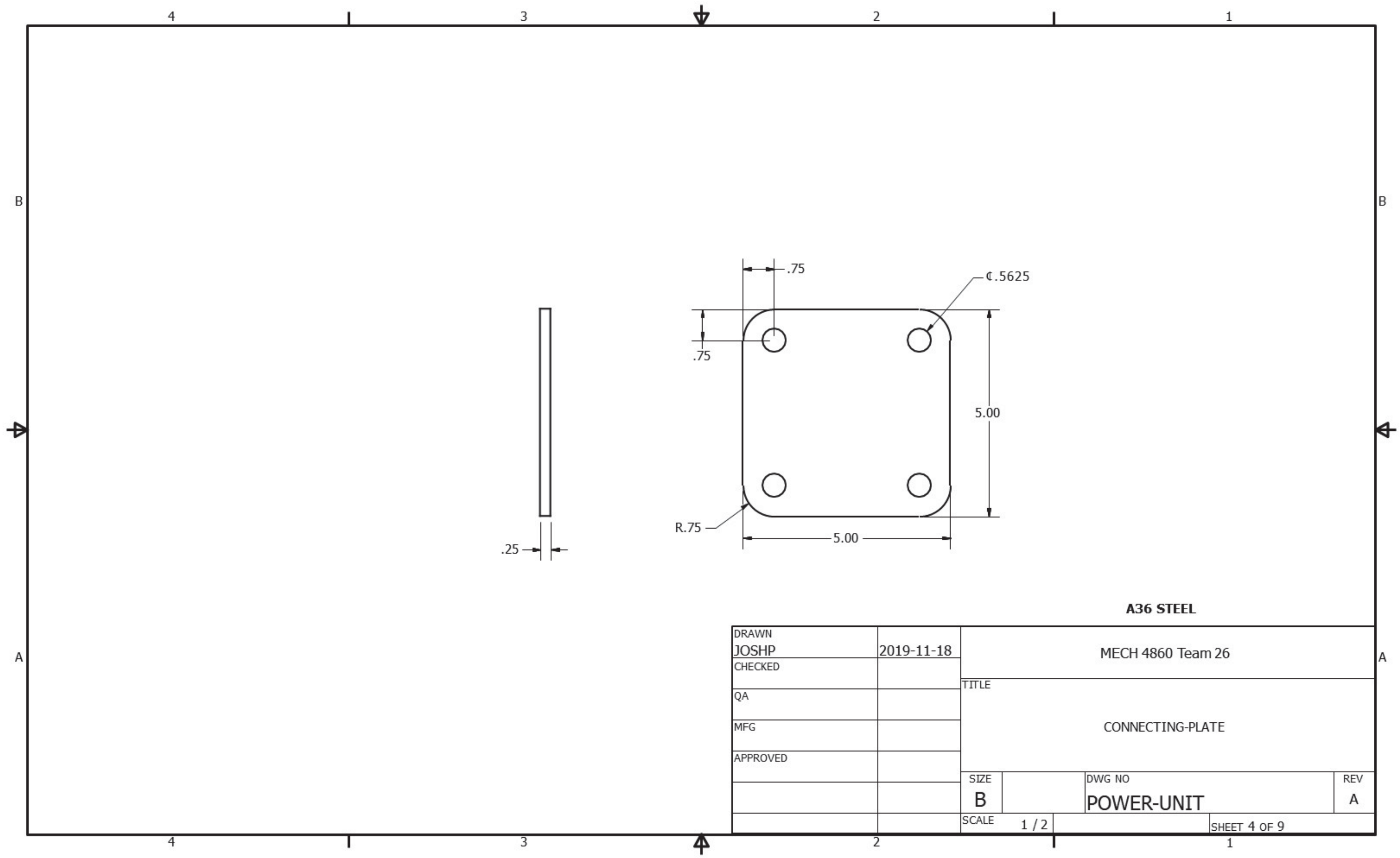
[redacted]

SAME FOR BOTH SIDES

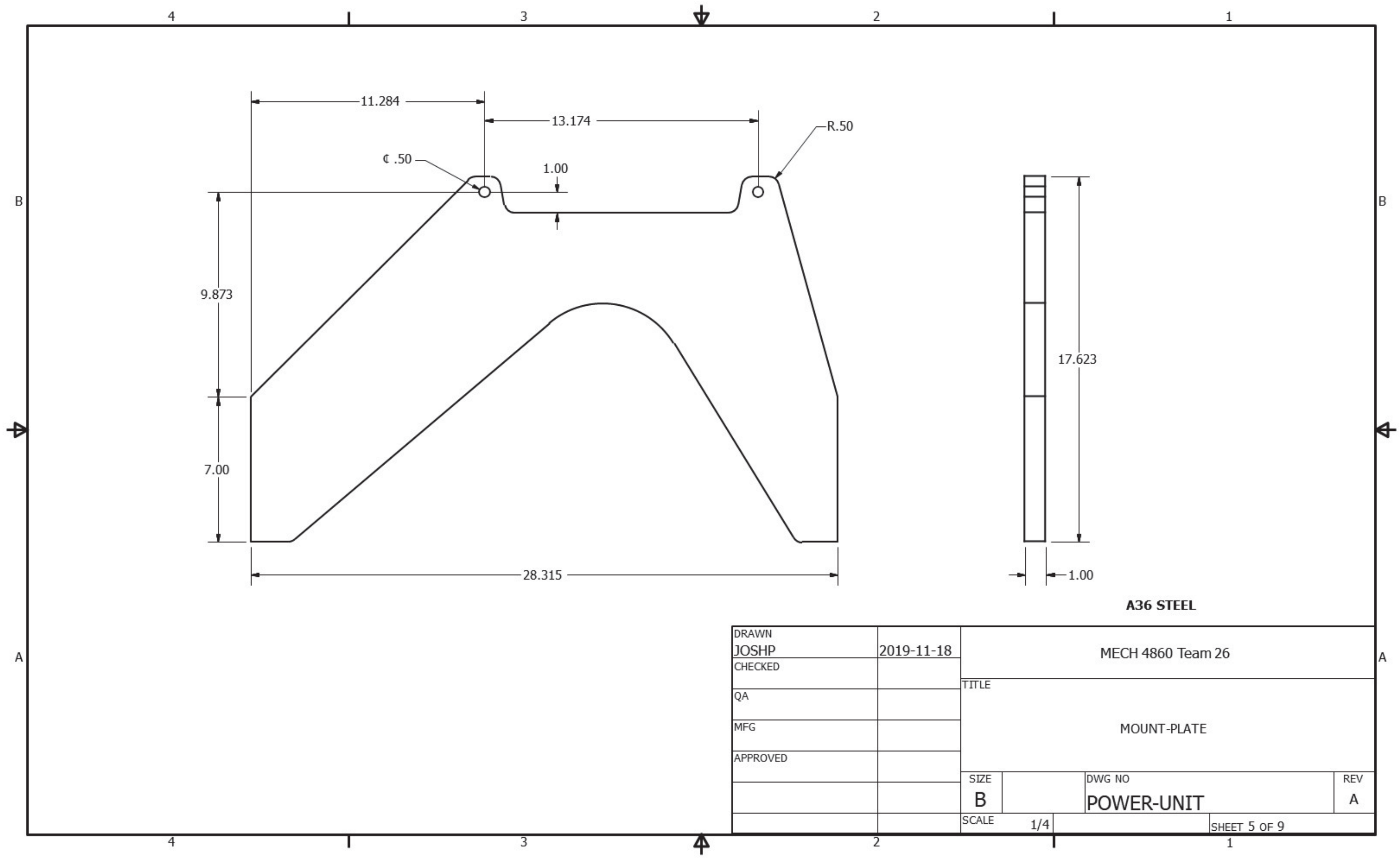
DRAWN JOSHP	2019-11-18	MECH 4860 Team 26
CHECKED		TITLE WELDING REDUCTION GEARBOX MOUNTS, FORK POCKETS, AND BLOCKING PLATES
QA		
MFG		
APPROVED		SIZE B / DWG NO POWER-UNIT / REV A
		SCALE 1/20 / SHEET 2 OF 9

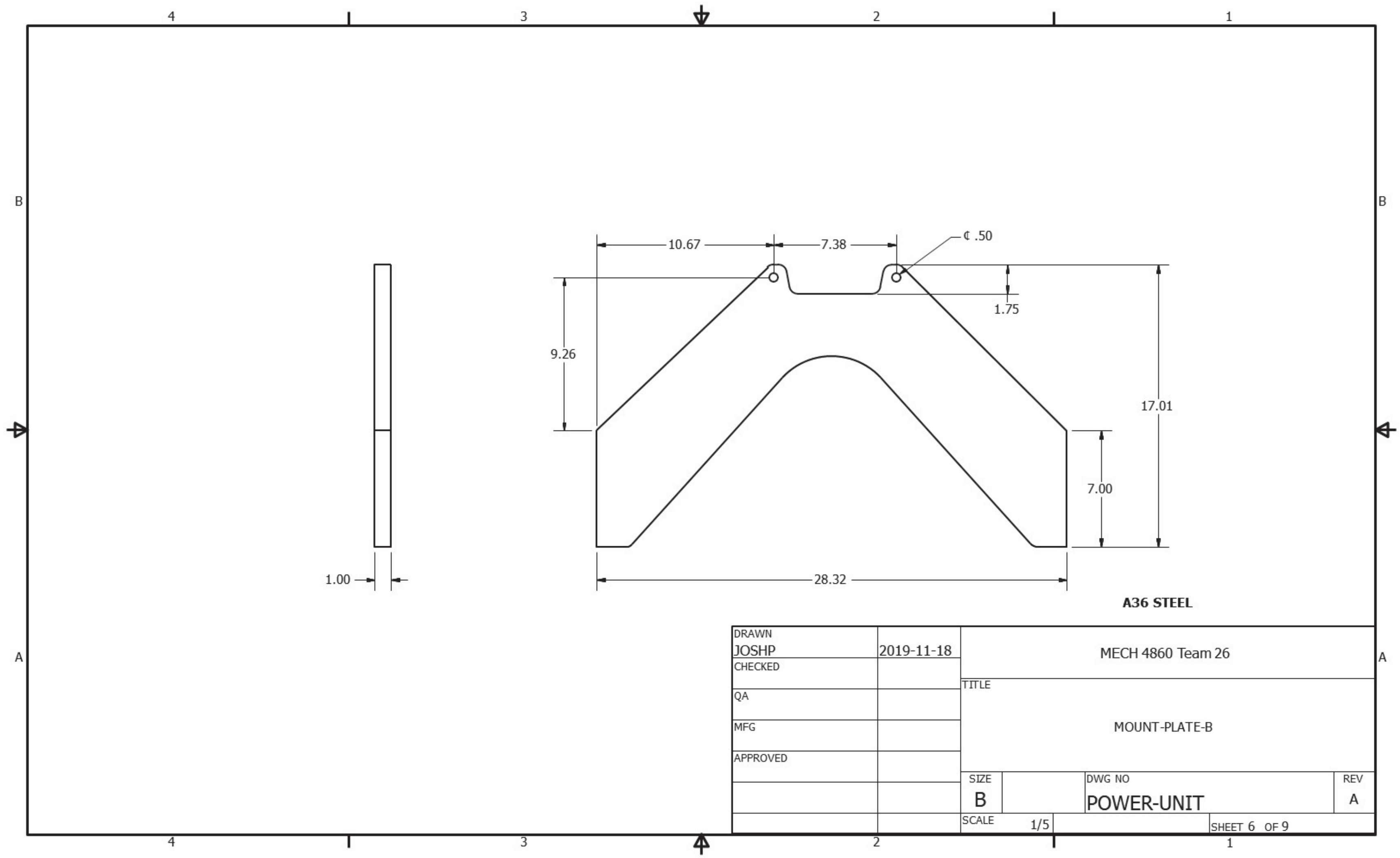


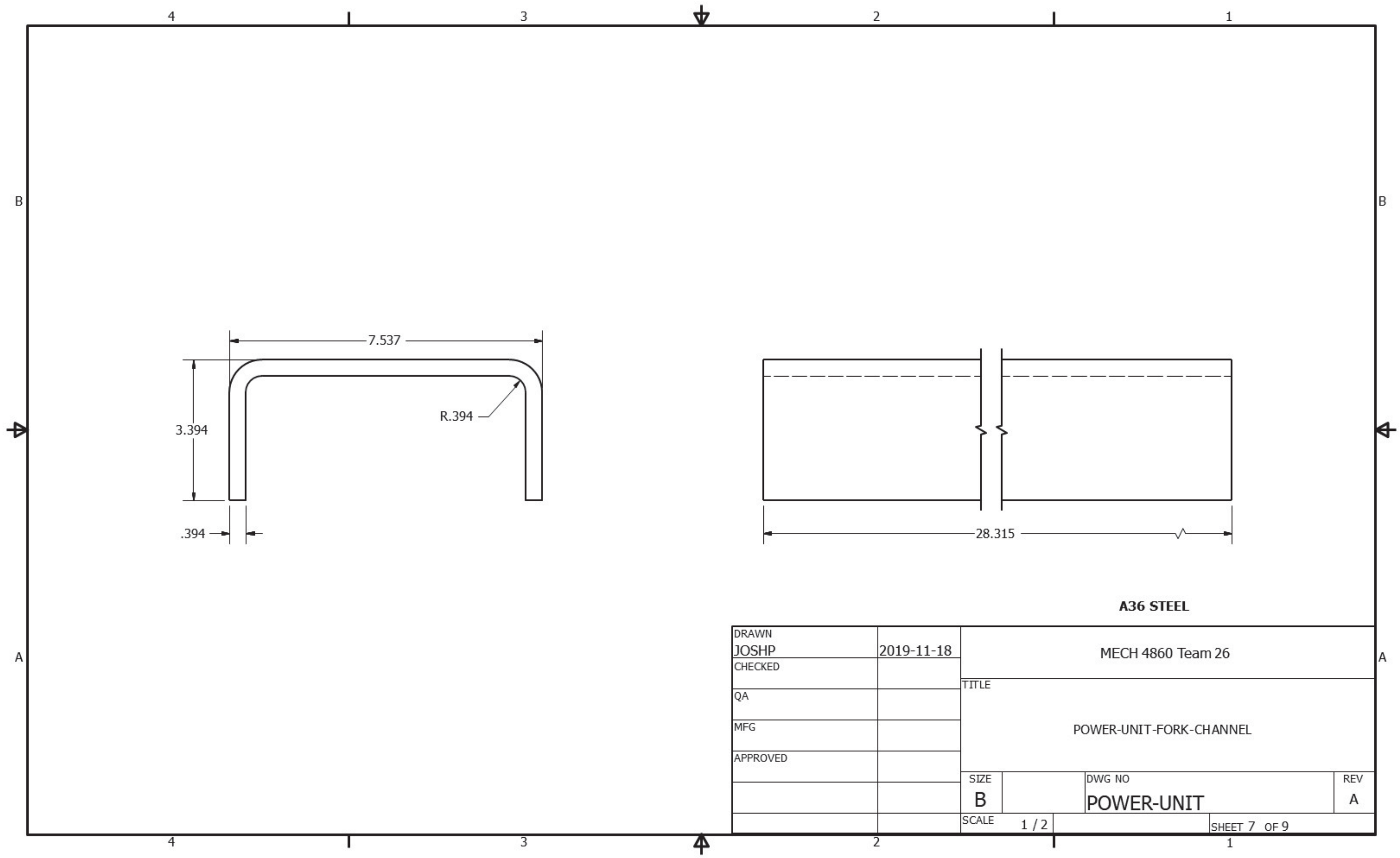
.314
7.00
2.194
48.00
A36 STEEL
DRAWN
JOSHP
2019-11-18
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
C-CHANNEL-EXTENSION
SIZE
B
DWG NO
POWER-UNIT
REV
A
SCALE 1 / 2
SHEET 3 OF 9

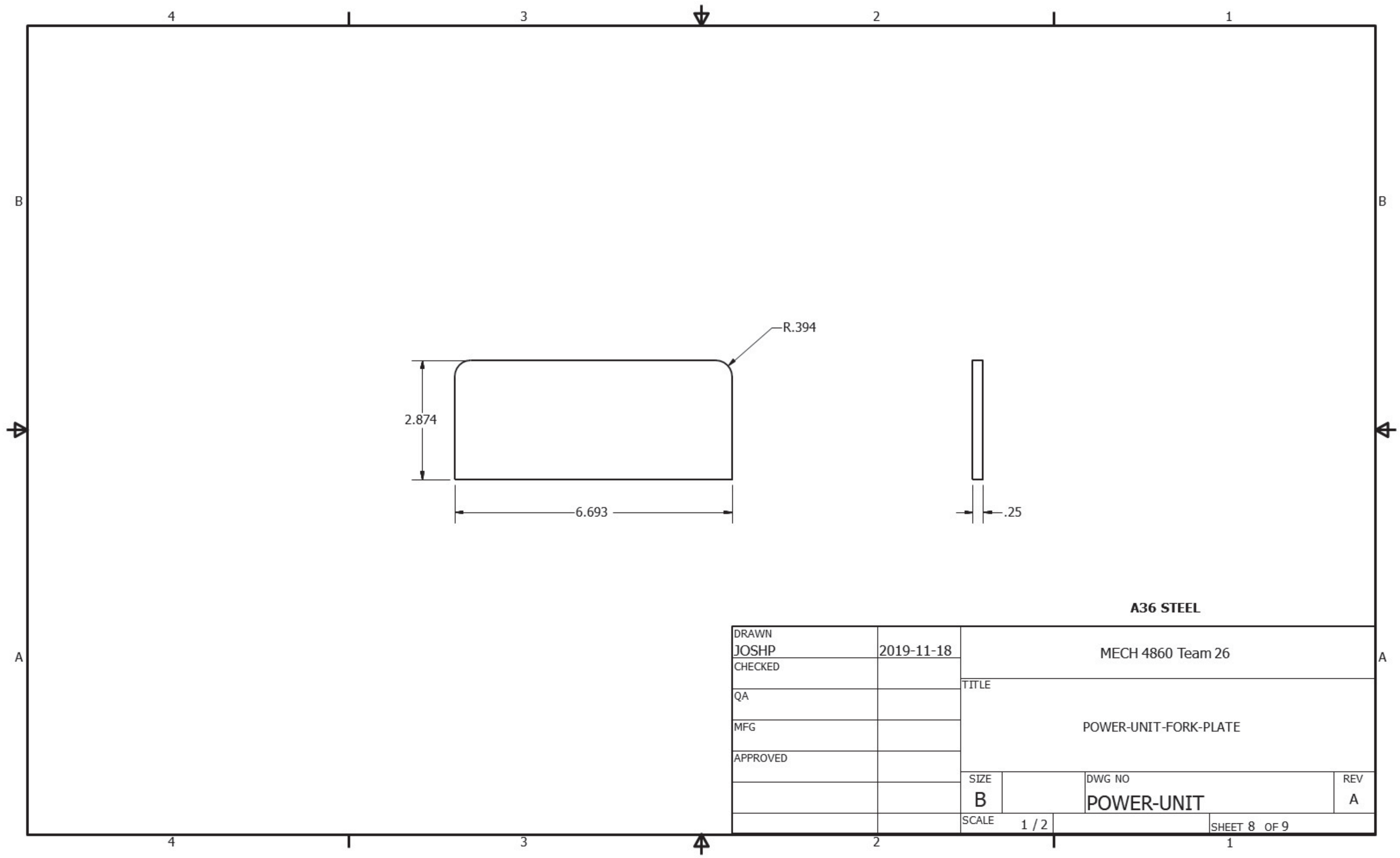


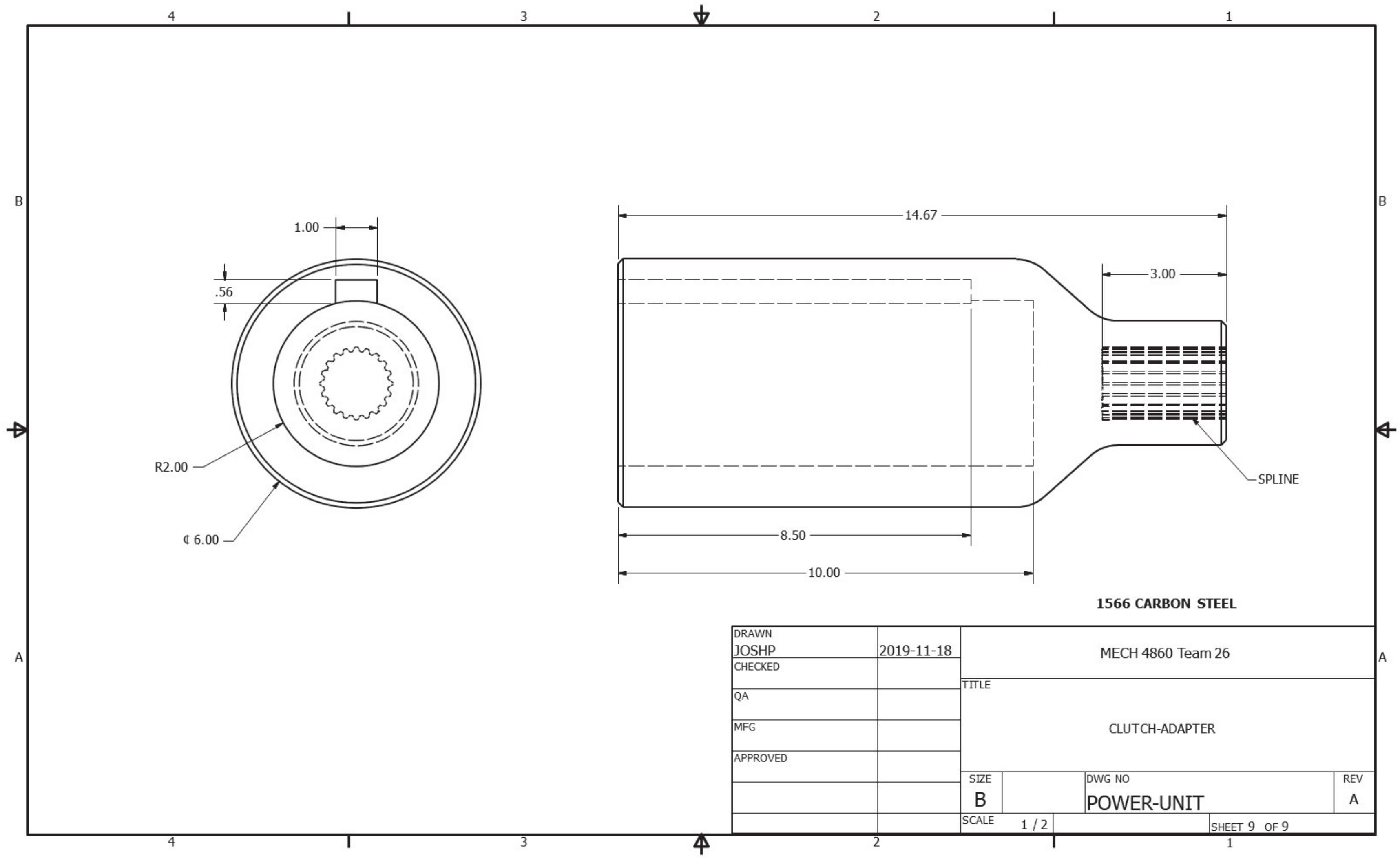
4
3
2
1
B
A
.75
⌀.5625
.75
5.00
R.75
5.00
.25
A36 STEEL
DRAWN
JOSHP
2019-11-18
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
CONNECTING-PLATE
SIZE
B
DWG NO
POWER-UNIT
REV
A
SCALE
1 / 2
SHEET 4 OF 9

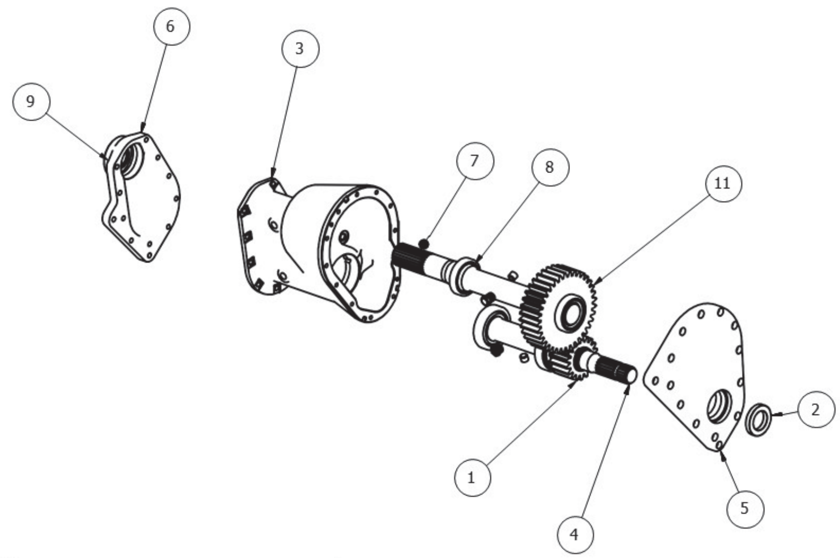

11.284
13.174
R.50
¢ .50
1.00
9.873
7.00
28.315
17.623
1.00
A36 STEEL
DRAWN
JOSHP
2019-11-18
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
MOUNT-PLATE
SIZE
B
DWG NO
POWER-UNIT
REV
A
SCALE
1/4
SHEET 5 OF 9


10.67
7.38
⌀ .50
1.75
9.26
17.01
7.00
28.32
1.00
A36 STEEL
DRAWN
JOSHP
2019-11-18
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
MOUNT-PLATE-B
SIZE
B
DWG NO
POWER-UNIT
REV
A
SCALE
1/5
SHEET 6 OF 9


7.537
3.394
R.394
.394
28.315
A36 STEEL
DRAWN
JOSHP
2019-11-18
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
POWER-UNIT-FORK-CHANNEL
SIZE
B
DWG NO
POWER-UNIT
REV
A
SCALE 1 / 2
SHEET 7 OF 9

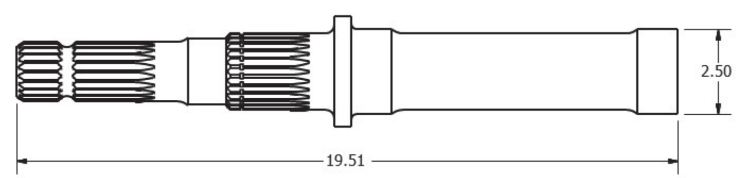

4
3
2
1
B
A
R.394
2.874
6.693
.25
A36 STEEL
DRAWN
JOSHP
2019-11-18
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
POWER-UNIT-FORK-PLATE
SIZE
B
DWG NO
POWER-UNIT
REV
A
SCALE
1 / 2
SHEET 8 OF 9


4
3
2
1
B
A
1.00
.56
R2.00
¢ 6.00
14.67
3.00
SPLINE
8.50
10.00
1566 CARBON STEEL
DRAWN
JOSHP
2019-11-18
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
CLUTCH-ADAPTER
SIZE
B
DWG NO
POWER-UNIT
REV
A
SCALE
1 / 2
SHEET 9 OF 9



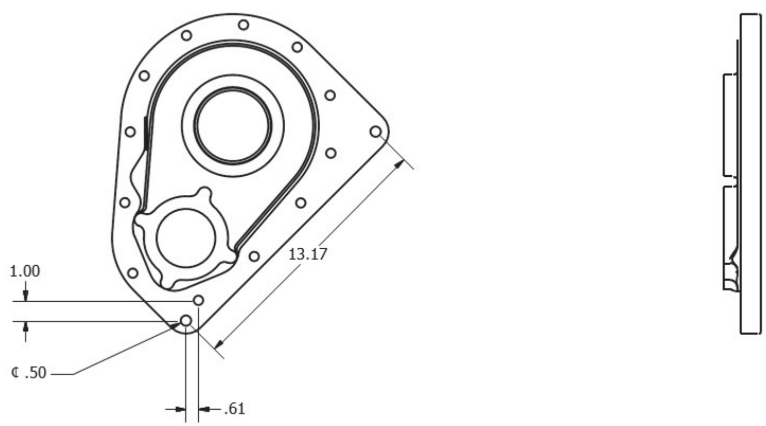
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	13432700	INPUT SPUR GEAR
2	1	13555000	ORIGINAL HAULMASTER BEARINGS AND SEALS
3	1	13331100	CASING
4	1	INPUT-SHAFT	
5	1	INPUT-END-PLATE	
6	1	OUTPUT-END-PLATE	
7	1	OUTPUT-SHAFT	
8	1	13337600	RELOCATED OUTPUT BEARING
9	1	13336600	OUTPUT SEAL
10	1	13336900	RETAINING RING
11	1	OUTPUT-SPUR-GEAR	OUTPUT SPUR GEAR

DRAWN JOSHP	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE
QA		EXPLODED VIEW OF REDUCER GEARBOX
MFG		
APPROVED		SIZE B · DWG NO REDUCER-GEARBOX · REV A
		SCALE 1 / 10 · SHEET 1 OF 6



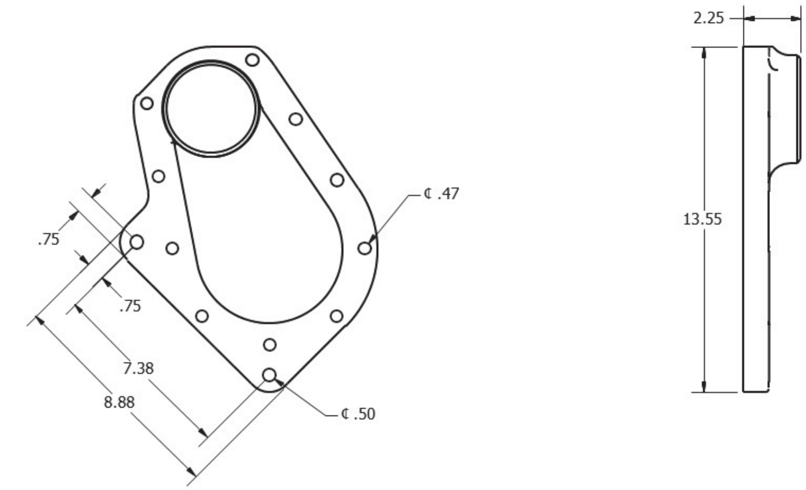
THIS SHAFT WAS DESIGNED BY SHORTENING THE HAULMASTER INPUT SHAFT TO REMOVE THE BEVEL GEARS, ALL OTHER FEATURES ARE THE SAME AS THE ORIGINAL INPUT SHAFT.

DRAWN JOSHP	2019-11-29	MECH 4860 Team 26			
CHECKED		TITLE INPUT-SHAFT			
QA					
MFG					
APPROVED					
		SIZE B		DWG NO REDUCER-GEARBOX	REV A
		SCALE 1 : 3		SHEET 2 OF 6	

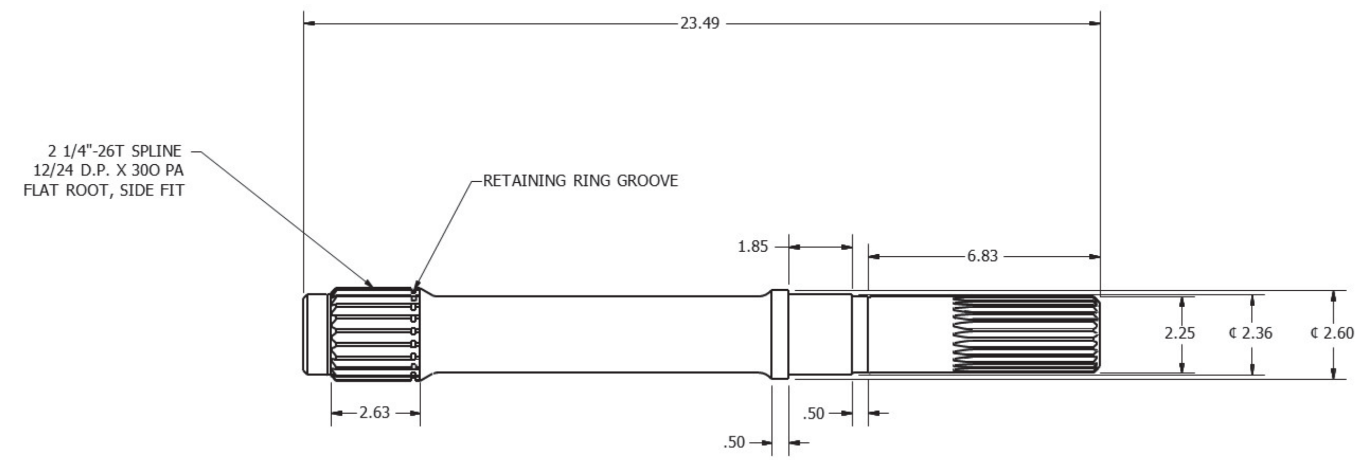


THIS END PLATE WAS DESIGNED USING THE HAULMASTER INPUT PLATE,
THE ONLY MODIFICATION IS EXTENDED THE PLATE TO ADD MOUNT HOLES.

DRAWN JOSHP	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE INPUT-END-PLATE
QA		
MFG		
APPROVED		SIZE B · DWG NO REDUCER-GEARBOX · REV A
		SCALE 1 / 5 · SHEET 3 OF 6

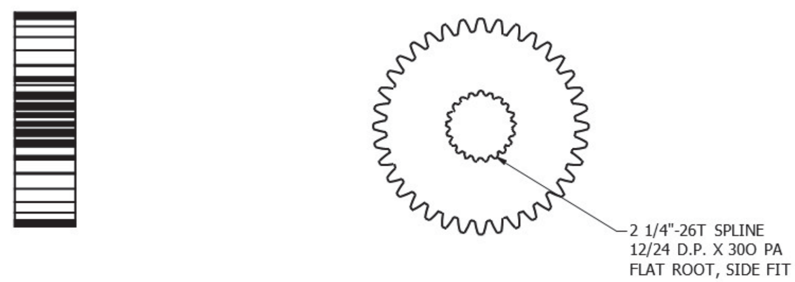


DRAWN JOSHP	2019-11-29	MECH 4860 Team 26		
CHECKED		TITLE		
QA		OUTPUT-END-PLATE		
MFG				
APPROVED		SIZE B	DWG NO REDUCER-GEARBOX	REV A
		SCALE 1 / 4		SHEET 4 OF 6



THIS SHAFT WAS DESIGNED BY SHORTENING THE HAULMASTER OUTPUT SHAFT TO REMOVE THE EXTRA LENGTH AND A SPLINE WAS ADDED TO FIX THE OUTPUT SPUR GEAR.

DRAWN JOSHP	2019-11-29	MECH 4860 Team 26		
CHECKED		TITLE OUTPUT-SHAFT		
QA				
MFG				
APPROVED		SIZE B	DWG NO REDUCER-GEARBOX	REV A
		SCALE 1 / 3	SHEET 5 OF 6	



THIS GEAR WAS DESIGNED USING THE HAULMASTER OUTPUT GEAR,THE MODIFICATION WAS TO REMOVE THE DOGTOOTH COUPLER AND BE FIXED BY AN INTERNAL SPLINE.

DRAWN JOSHP	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE OUTPUT-SPUR-GEAR
QA		
MFG		
APPROVED		SIZE B · DWG NO REDUCER-GEARBOX · REV A
		SCALE 1 / 4 · SHEET 6 OF 6

[redacted]

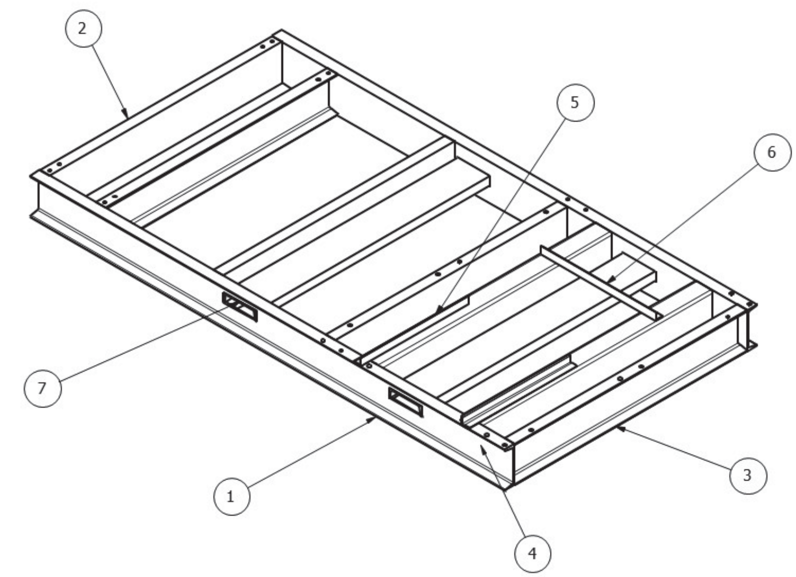
PARTS LIST

ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	GTU-SKID	
2	1	GEARBOX-MOUNTING-SYSTEM	
3	2	CONTROL-VALVE-ASSEMBLY	
4	8	2in Code 61 Split Flange	
5	4	4in Code 61 Split Flange	
6	2	INCREASER-GEARBOX	
7	1	STATIONARY-PUMP-MOUNT	
8	1	ADJ-PUMP-MOUNT	
9	1	K3VG280DT	
10	1	K3VG280DT with Auxiliary Pump	

DRAWN JOSHP	2019-11-30	MECH 4860 Team 26	
CHECKED		TITLE EXPLODED VIEW OF GEARBOX TEST UNIT ASSEMBLY	
QA			
MFG			
APPROVED			
		SIZE B	DWG NO GTU-ASSEMBLY	REV A
		SCALE 0.03 : 1	SHEET 1 OF 2

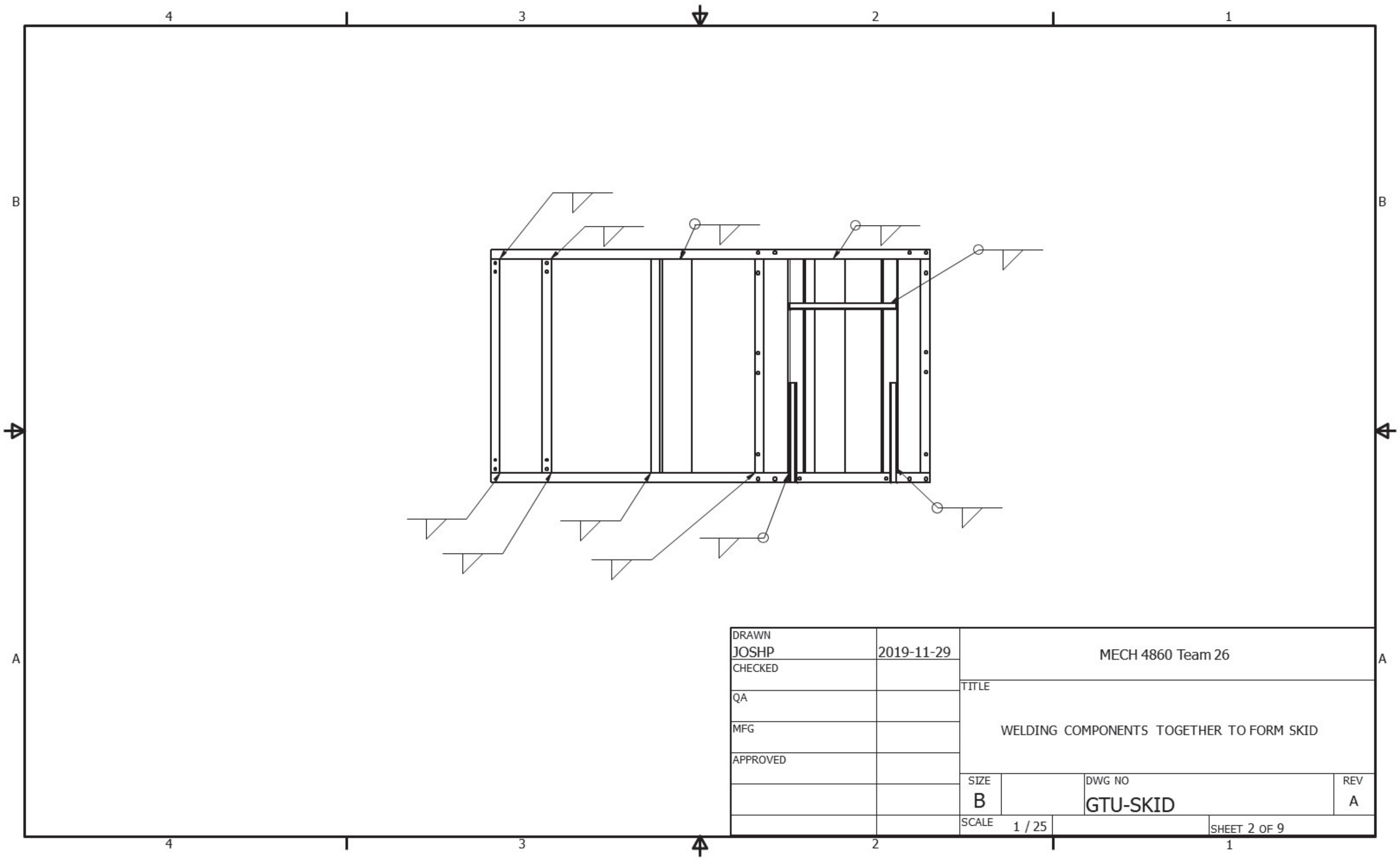
[redacted]

DRAWN		MECH 4860 Team 26		
JOSHP	2019-11-30			
CHECKED		TITLE		
QA		WELDING STATIONARY PUMP MOUNT TO SKID		
MFG				
APPROVED		SIZE B	DWG NO GTU-ASSEMBLY	REV A
		SCALE 1 / 25		SHEET 2 OF 2

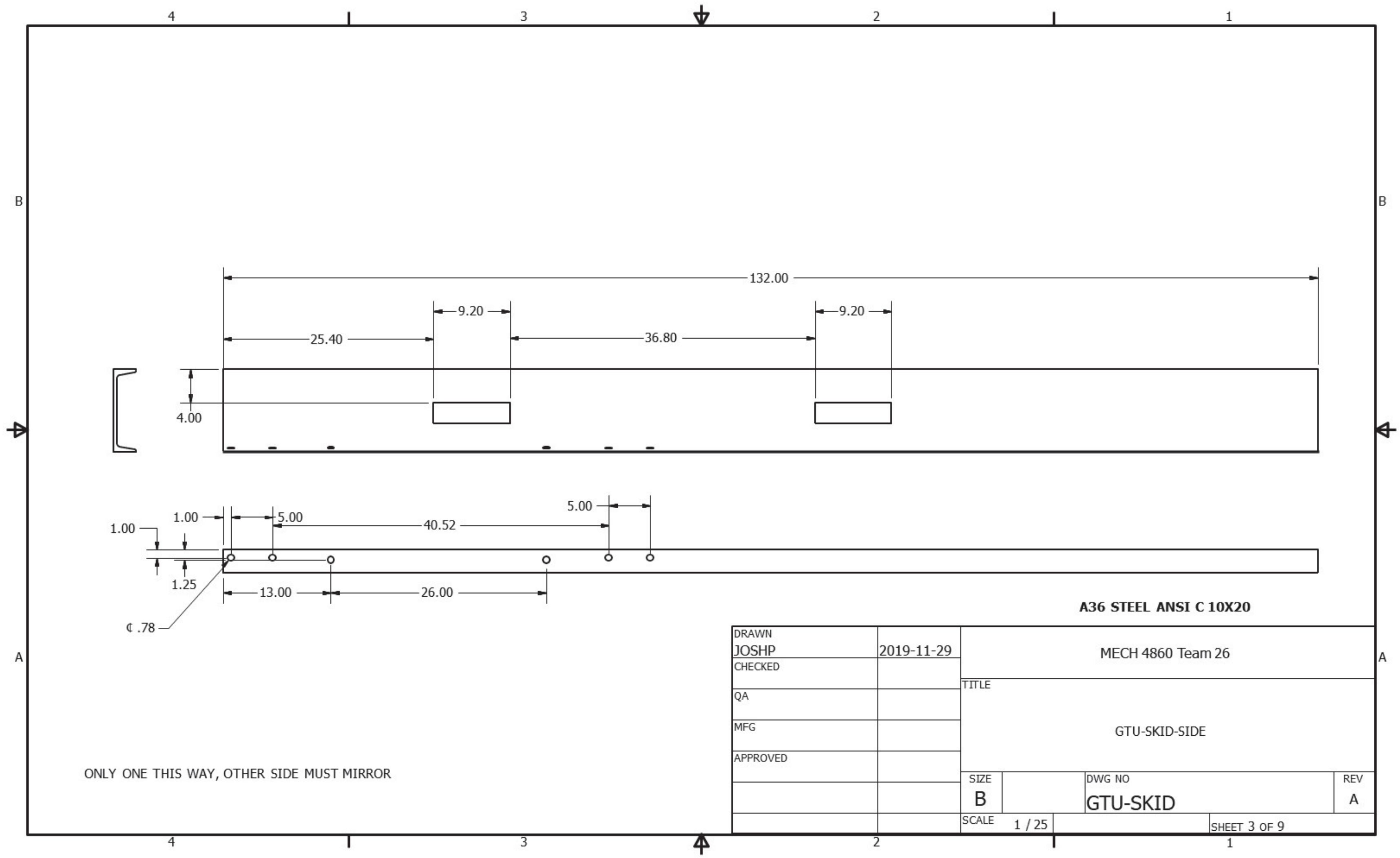


PARTS LIST			
I TEM	QTY	PART NUMBER	DESCRIPTION
1	2	GTU-SKID-SIDE	
2	3	GTU-BACK-BRACE	
3	2	GTU-FRONT-BRACE	
4	2	GTU-BOX-BRACE	
5	2	GTU-ANGLE-SIDE	
6	1	GTU-ANGLE-BACK	
7	2	GTU-FORK-CHANNELS	

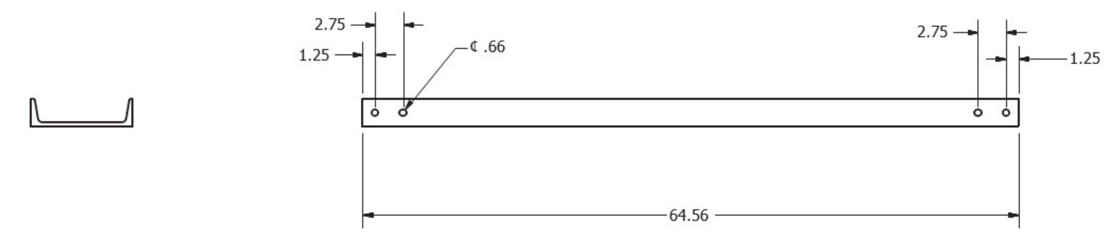
DRAWN JOSHP	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE
QA		EXPLODED VIEW OF GTU SKID
MFG		
APPROVED		
SIZE B	DWG NO GTU-SKID	REV A
SCALE 1 / 20		SHEET 1 OF 9



DRAWN JOSHP	2019-11-29	MECH 4860 Team 26		
CHECKED		TITLE WELDING COMPONENTS TOGETHER TO FORM SKID		
QA				
MFG				
APPROVED		SIZE B	DWG NO GTU-SKID	REV A
		SCALE 1 / 25	SHEET 2 OF 9	

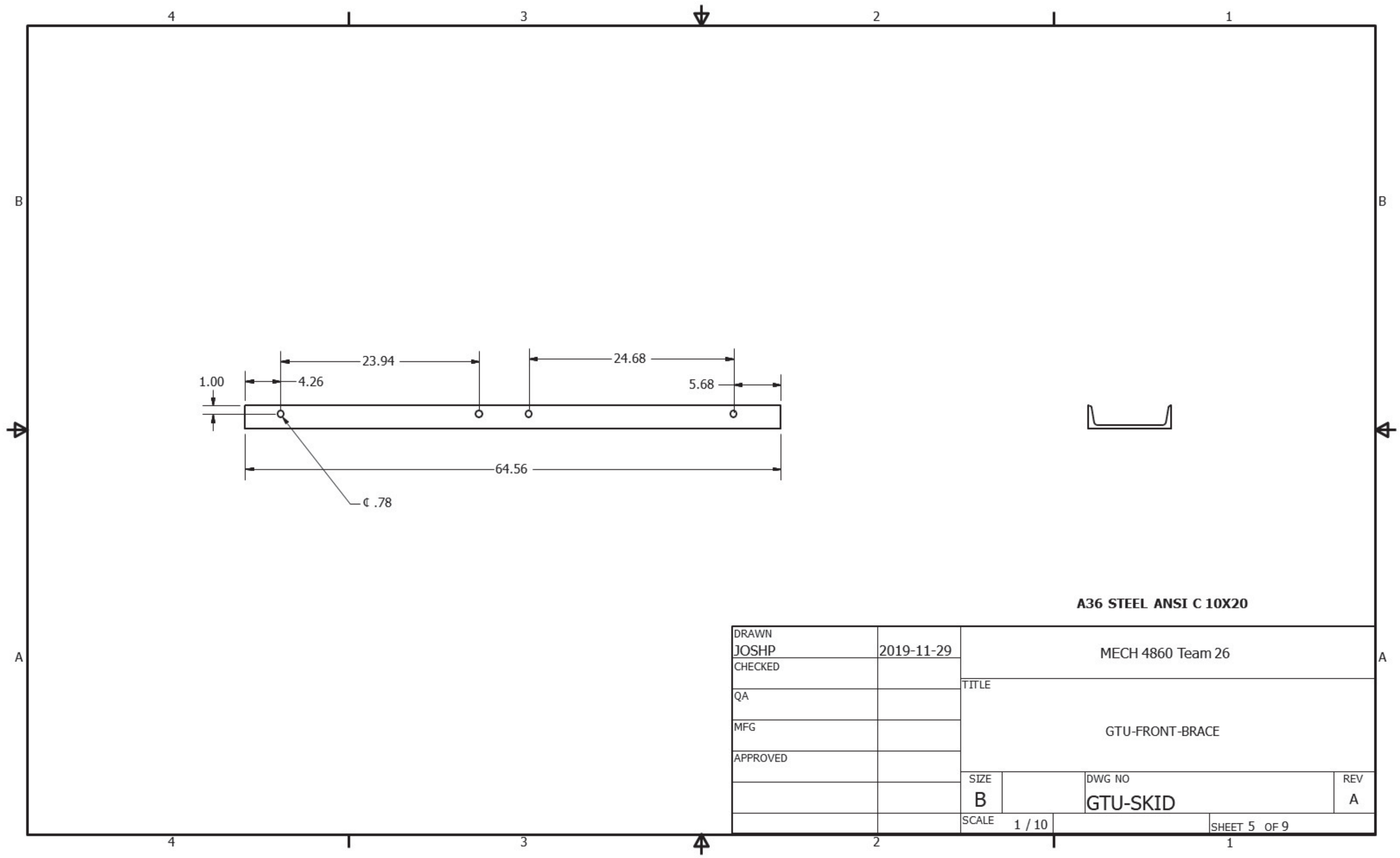


132.00
25.40
9.20
36.80
9.20
4.00
1.00
1.00
5.00
40.52
5.00
1.25
13.00
26.00
¢ .78
A36 STEEL ANSI C 10X20
ONLY ONE THIS WAY, OTHER SIDE MUST MIRROR
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
GTU-SKID-SIDE
SIZE
B
DWG NO
GTU-SKID
REV
A
SCALE
1 / 25
SHEET 3 OF 9

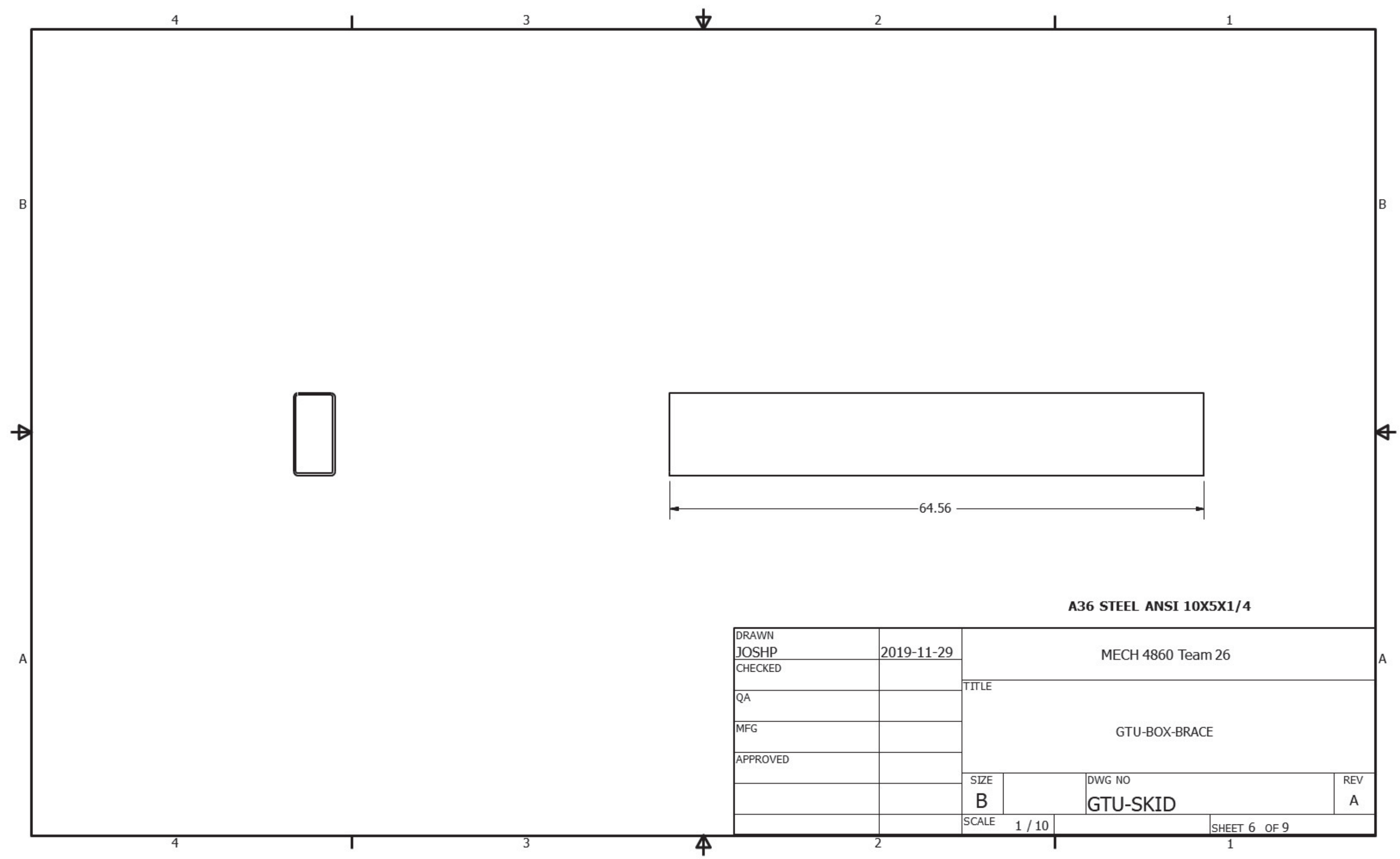


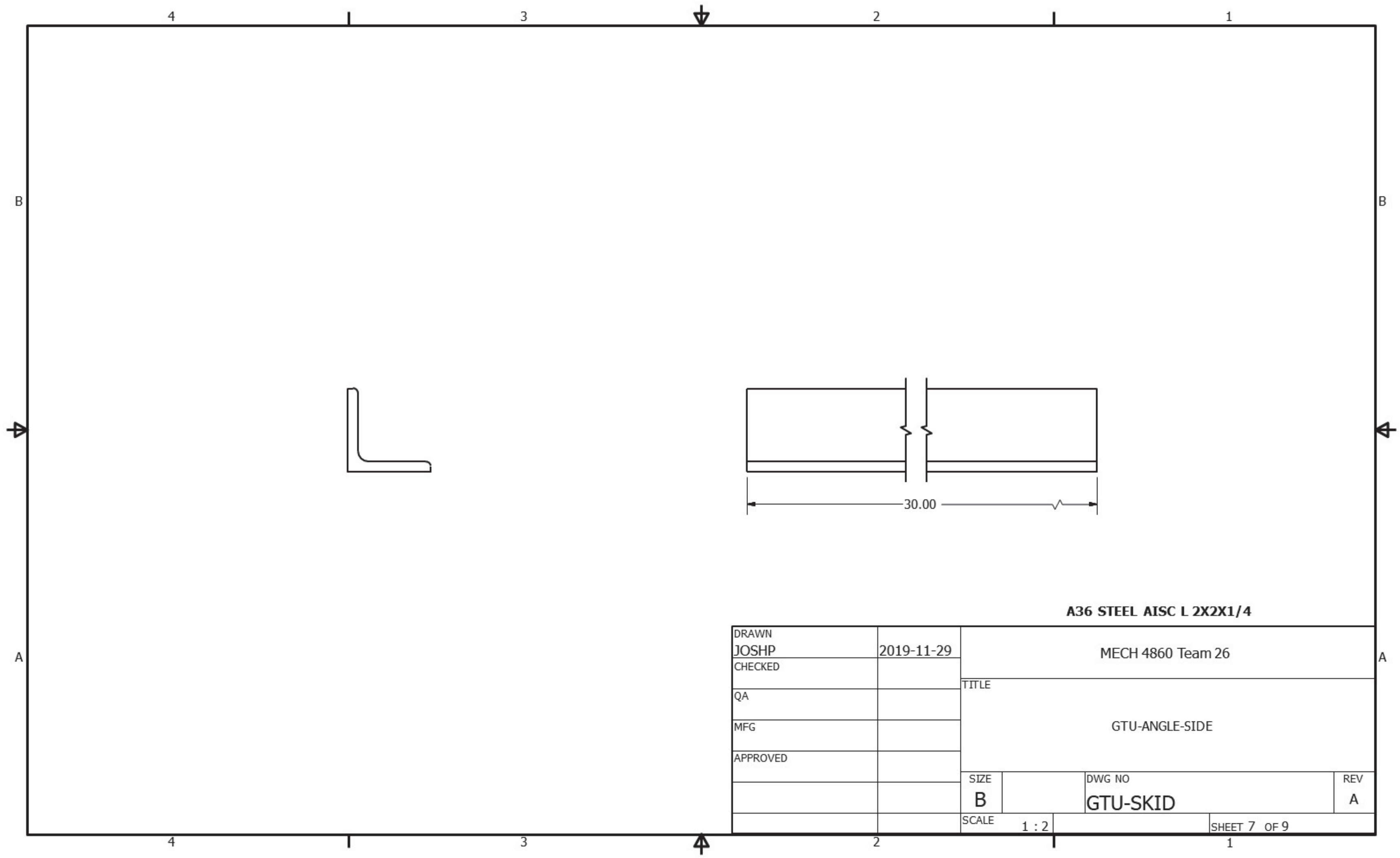
A36 STEEL ANSI C 10X20

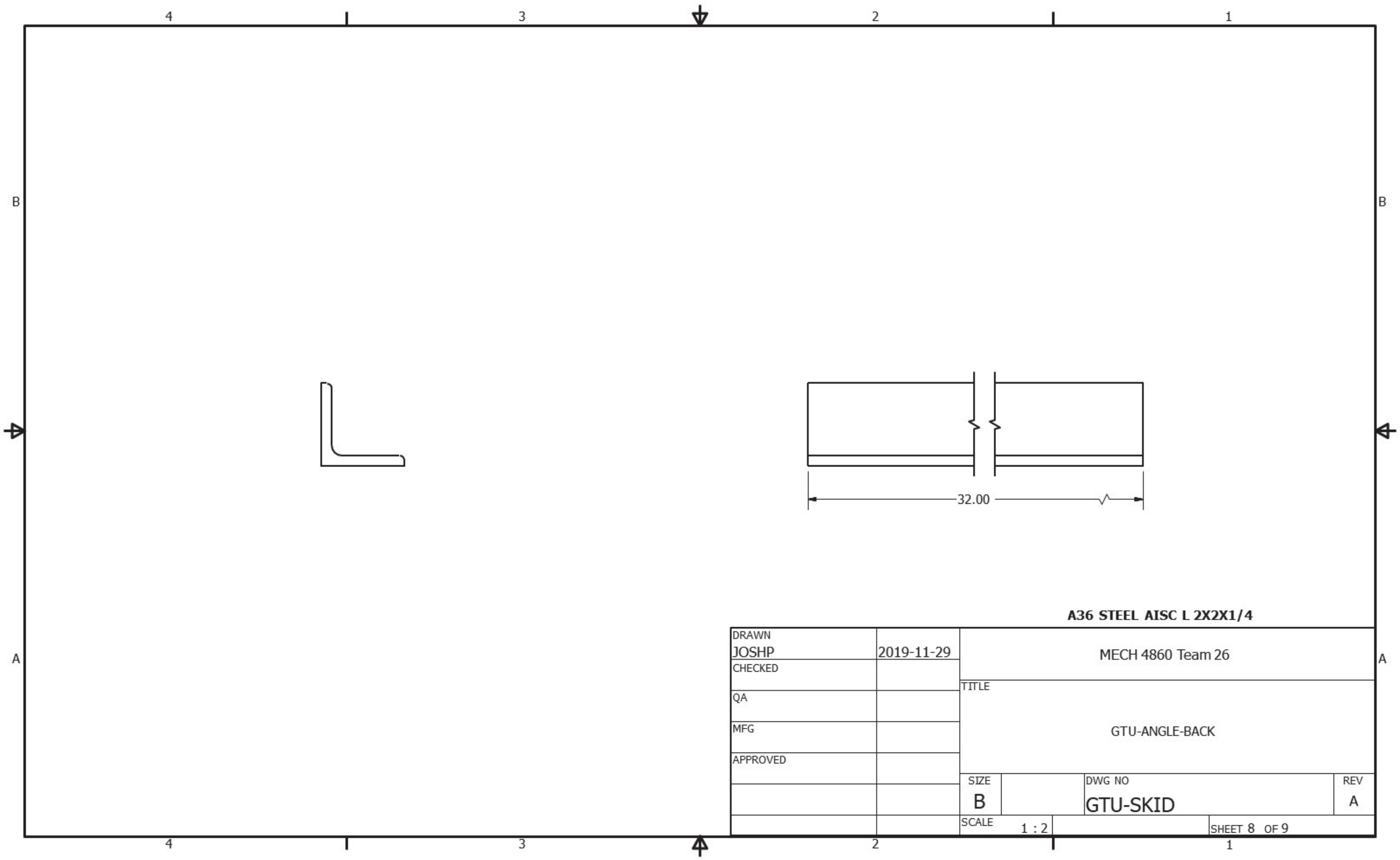
DRAWN JOSHP	2019-11-29	MECH 4860 Team 26		
CHECKED		TITLE GTU-BACK-BRACE		
QA				
MFG				
APPROVED		SIZE B	DWG NO GTU-SKID	REV A
		SCALE 1 / 10	SHEET 4 OF 9	

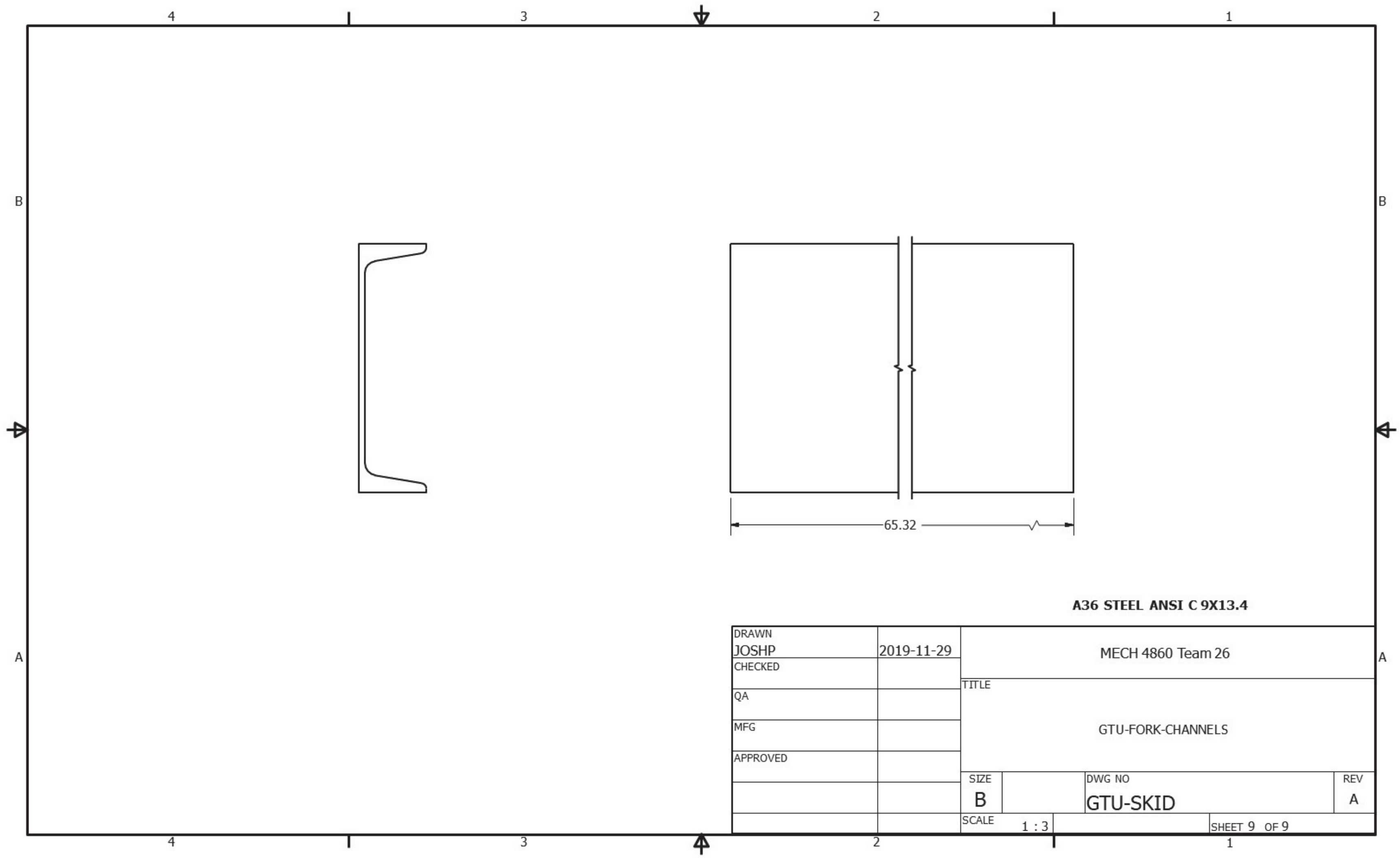


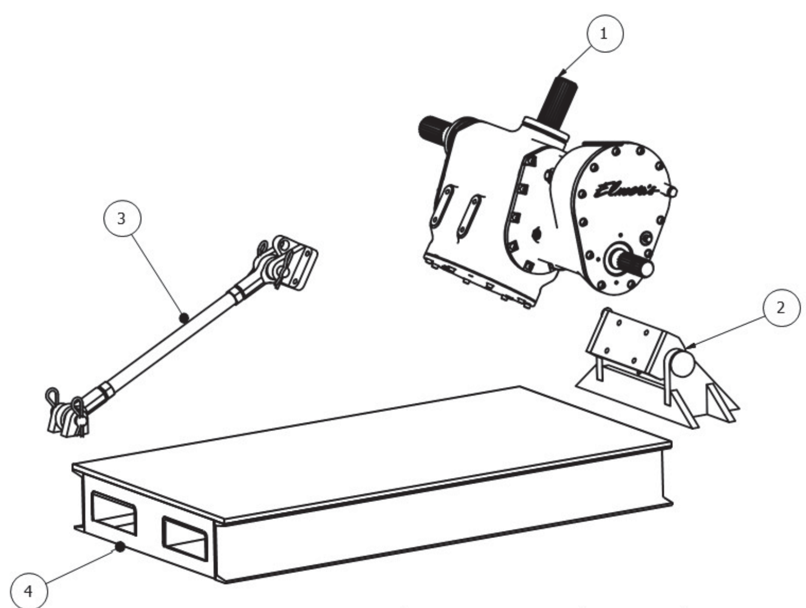
DRAWN JOSHP	2019-11-29	MECH 4860 Team 26			
CHECKED		TITLE GTU-FRONT-BRACE			
QA					
MFG					
APPROVED		SIZE B	DWG NO GTU-SKID		REV A
		SCALE 1 / 10		SHEET 5 OF 9	


64.56
A36 STEEL ANSI 10X5X1/4
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
GTU-BOX-BRACE
SIZE
B
DWG NO
GTU-SKID
REV
A
SCALE 1 / 10
SHEET 6 OF 9


30.00
A36 STEEL AISC L 2X2X1/4
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
GTU-ANGLE-SIDE
SIZE
B
DWG NO
GTU-SKID
REV
A
SCALE
1 : 2
SHEET 7 OF 9

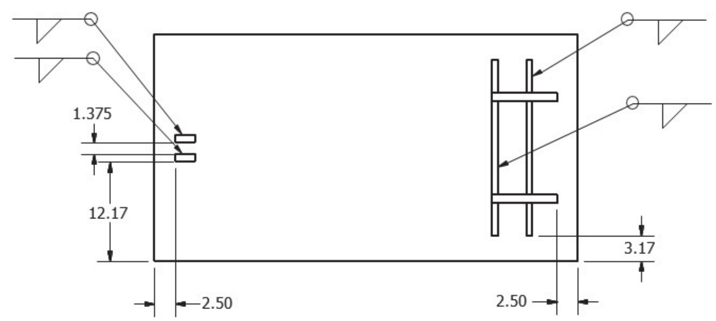

4
3
2
1
B
A
32.00
A36 STEEL AISC L 2X2X1/4
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
GTU-ANGLE-BACK
SIZE
B
DWG NO
GTU-SKID
REV
A
SCALE
1 : 2
SHEET 8 OF 9


4
3
2
1
B
A
65.32
A36 STEEL ANSI C 9X13.4
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
GTU-FORK-CHANNELS
SIZE
B
DWG NO
GTU-SKID
REV
A
SCALE 1 : 3
SHEET 9 OF 9

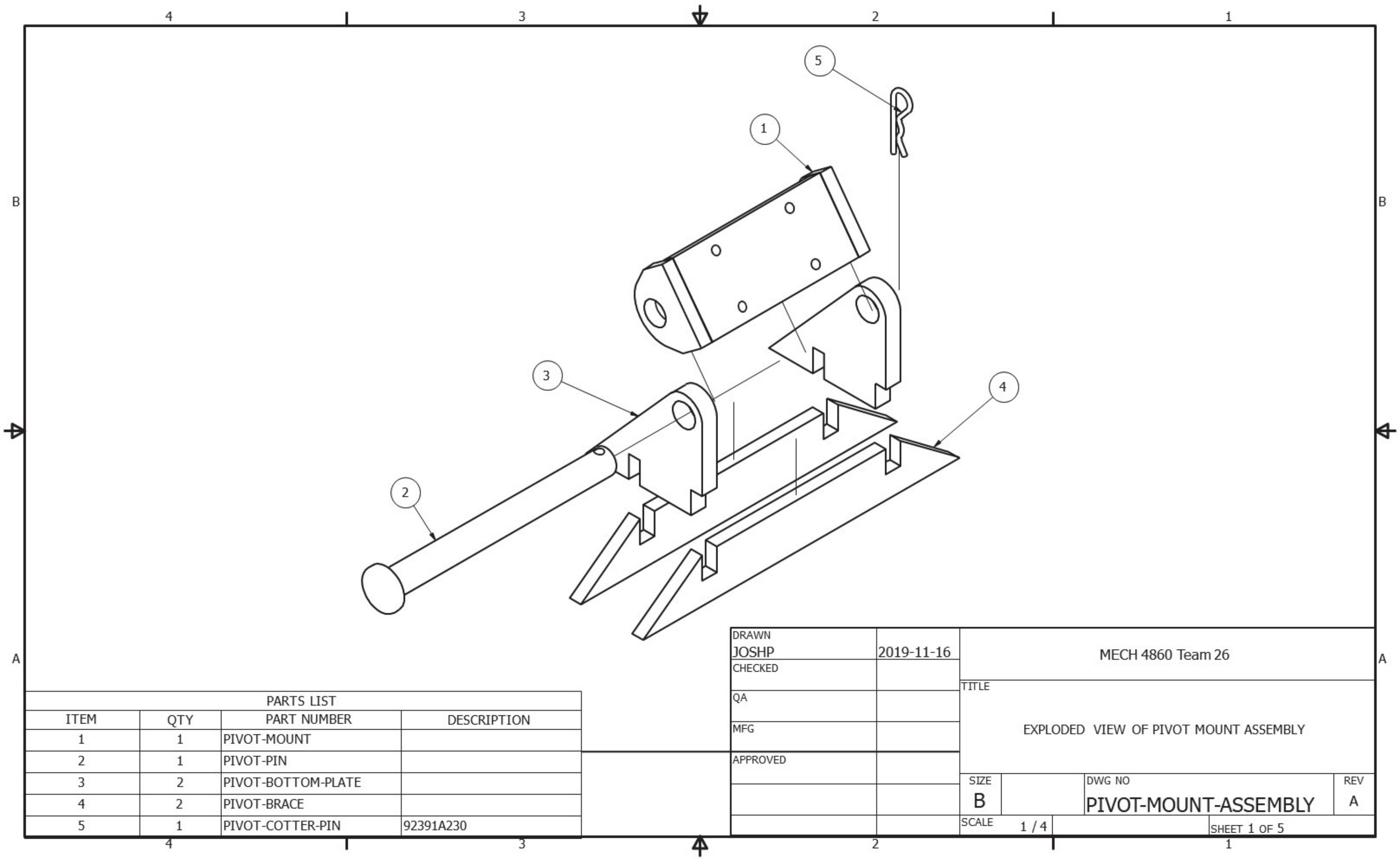


PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	13432800	HAULMASTER GEARBOX
2	1	PIVOT-MOUNT-ASSEMBLY	
3	1	ADJ-MOUNT-ASSEMBLY	
4	1	MINI-SKID	

DRAWN JOSHP	2019-11-17	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF GEARBOX MOUNTING SYSTEM
QA		
MFG		
APPROVED		SIZE B · DWG NO GEARBOX-MOUNTING-SYSTEM · REV A
		SCALE 1 / 10 · SHEET 1 OF 2



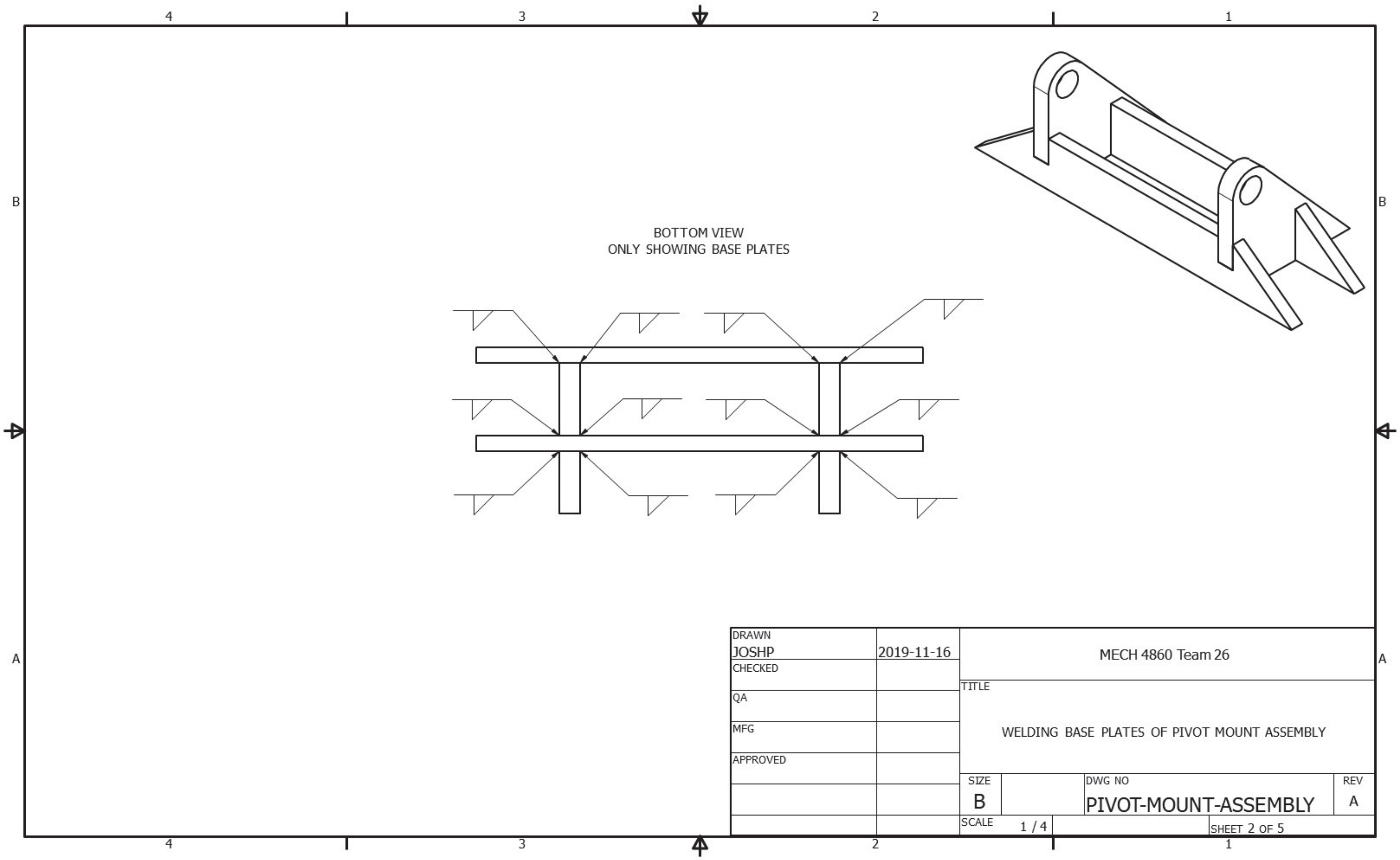
DRAWN JOSHP	2019-11-17	MECH 4860 Team 26		
CHECKED		TITLE WELDING MOUNT PLATES TO MINI-SKID		
QA				
MFG				
APPROVED		SIZE B	DWG NO GEARBOX-MOUNTING-SYSTEM	REV A
		SCALE 0.08 : 1	SHEET 2 OF 2	

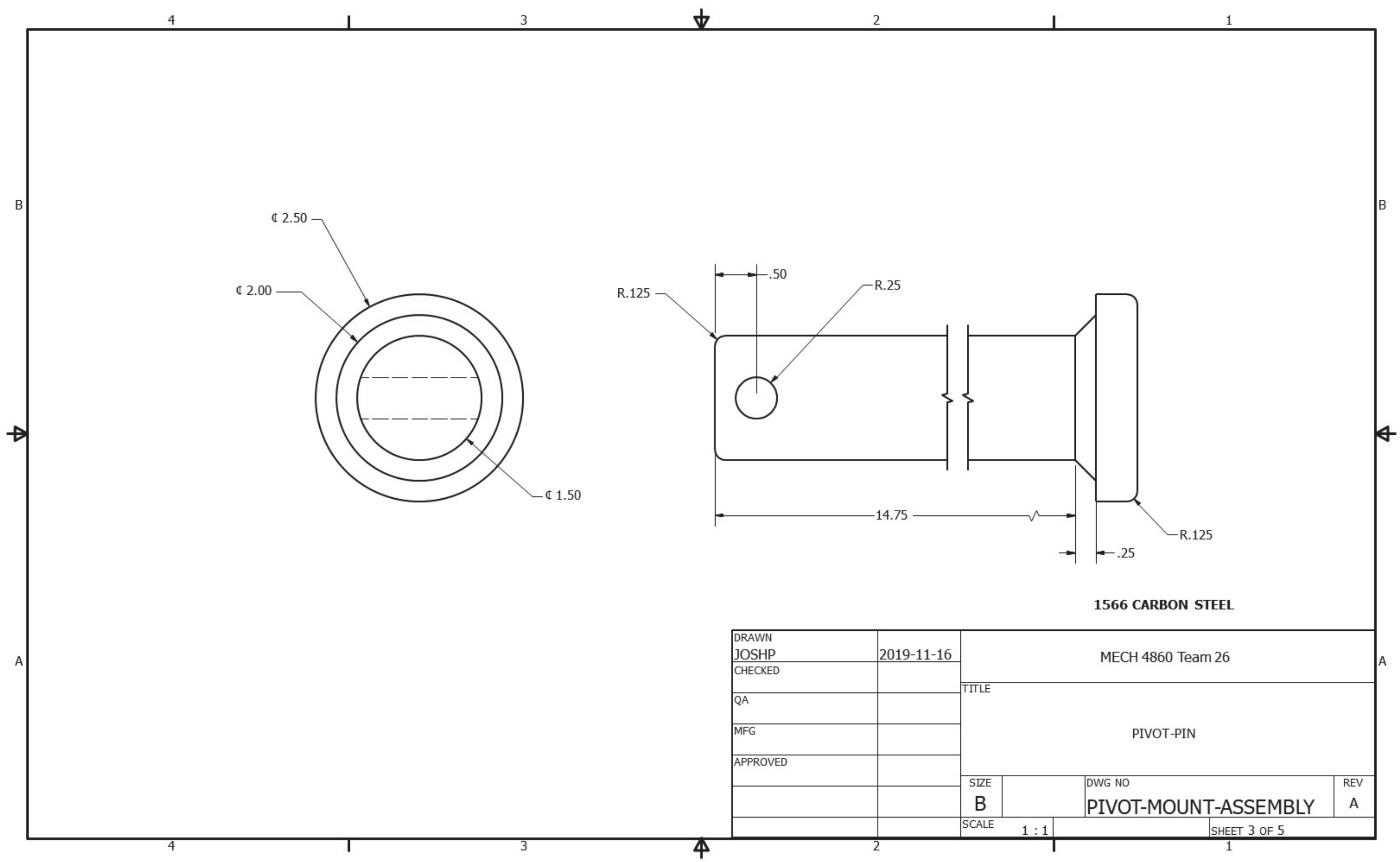


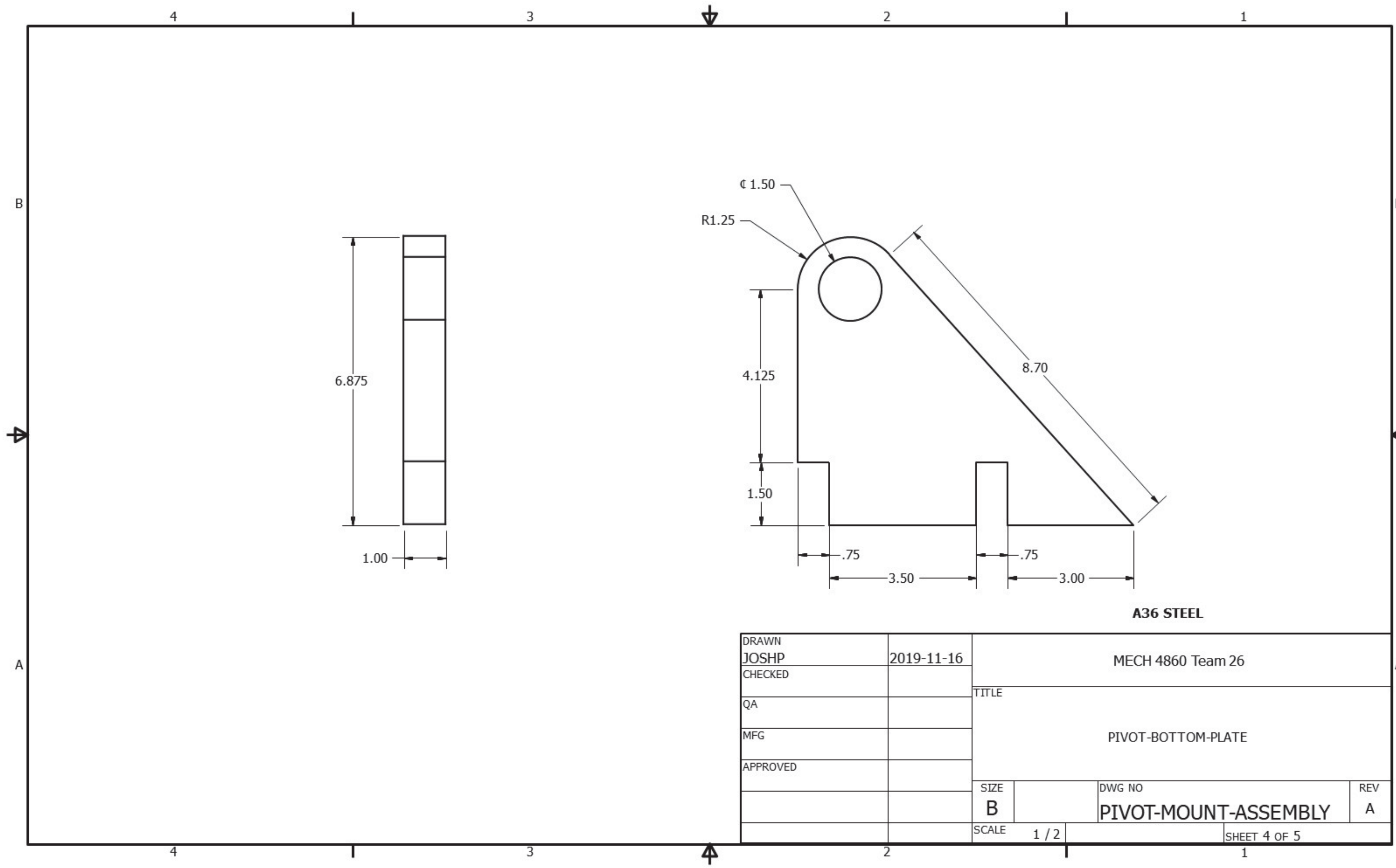
PARTS LIST

ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	PIVOT-MOUNT	
2	1	PIVOT-PIN	
3	2	PIVOT-BOTTOM-PLATE	
4	2	PIVOT-BRACE	
5	1	PIVOT-COTTER-PIN	92391A230

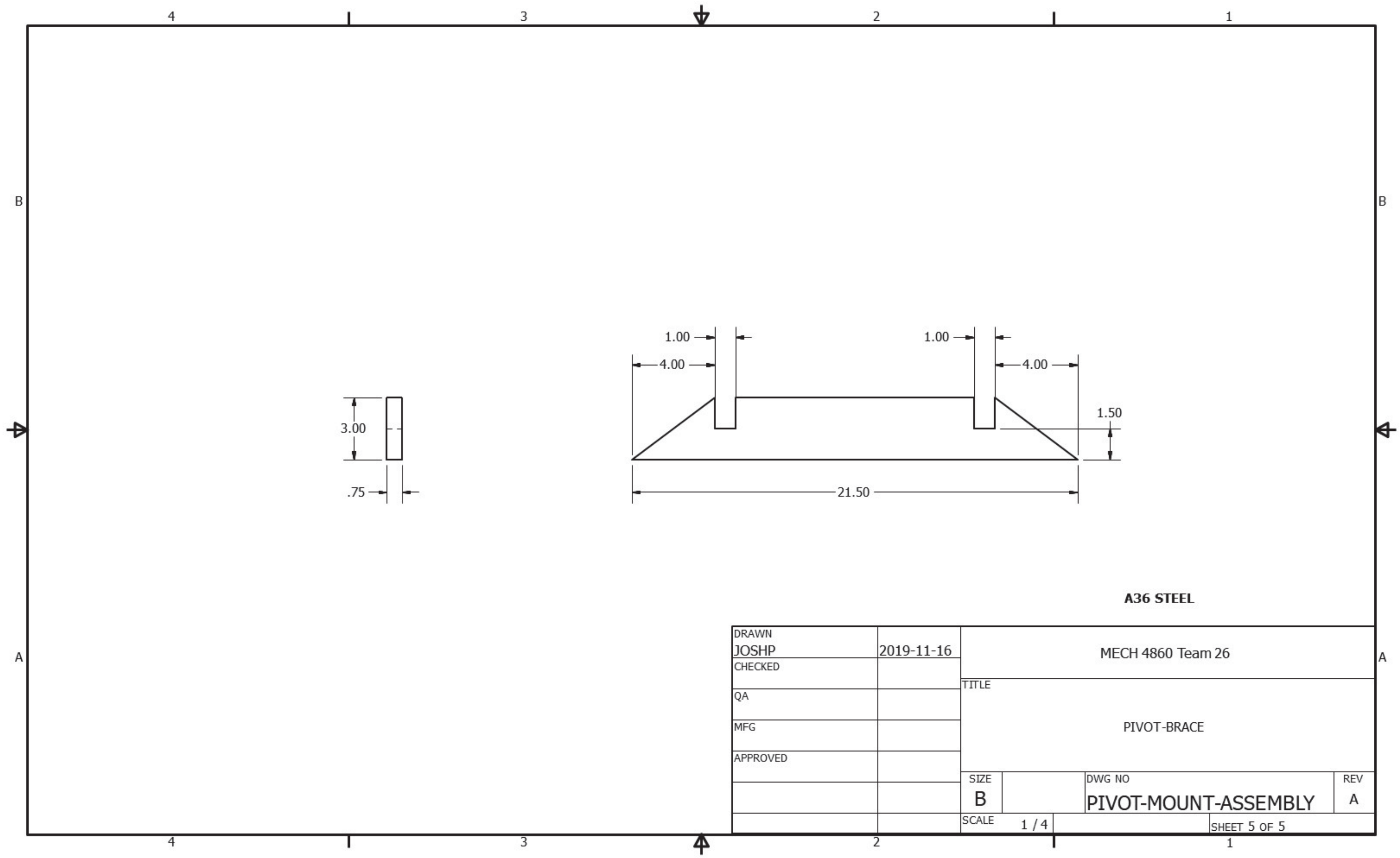
DRAWN JOSHP	2019-11-16	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF PIVOT MOUNT ASSEMBLY
QA		
MFG		
APPROVED		SIZE B; DWG NO PIVOT-MOUNT-ASSEMBLY; REV A
		SCALE 1 / 4; SHEET 1 OF 5

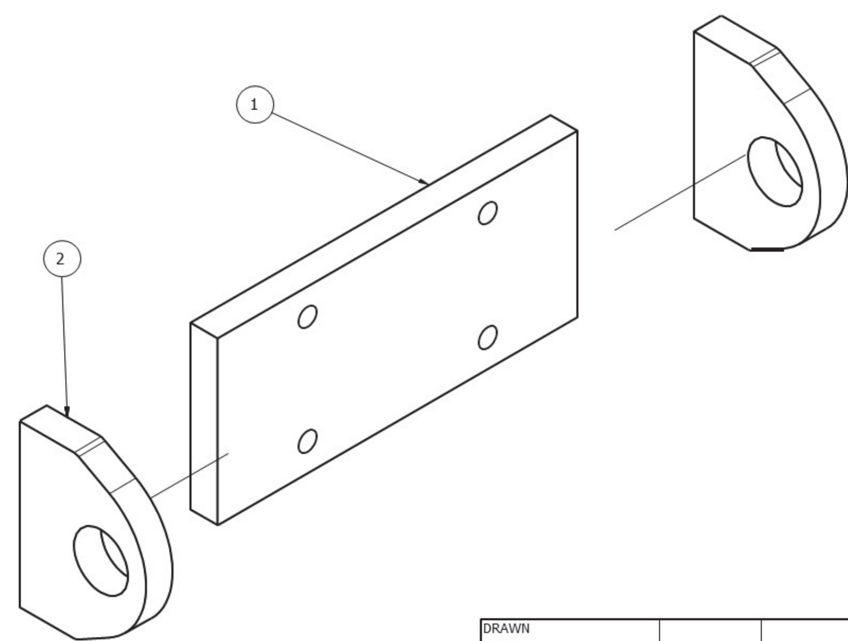

4
3
2
1
B
A
BOTTOM VIEW
ONLY SHOWING BASE PLATES
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
WELDING BASE PLATES OF PIVOT MOUNT ASSEMBLY
SIZE
B
DWG NO
PIVOT-MOUNT-ASSEMBLY
REV
A
SCALE
1 / 4
SHEET 2 OF 5


4
3
2
1
B
A
¢ 2.50
¢ 2.00
¢ 1.50
R.125
.50
R.25
14.75
R.125
.25
1566 CARBON STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
PIVOT-PIN
SIZE
B
DWG NO
PIVOT-MOUNT-ASSEMBLY
REV
A
SCALE
1 : 1
SHEET 3 OF 5



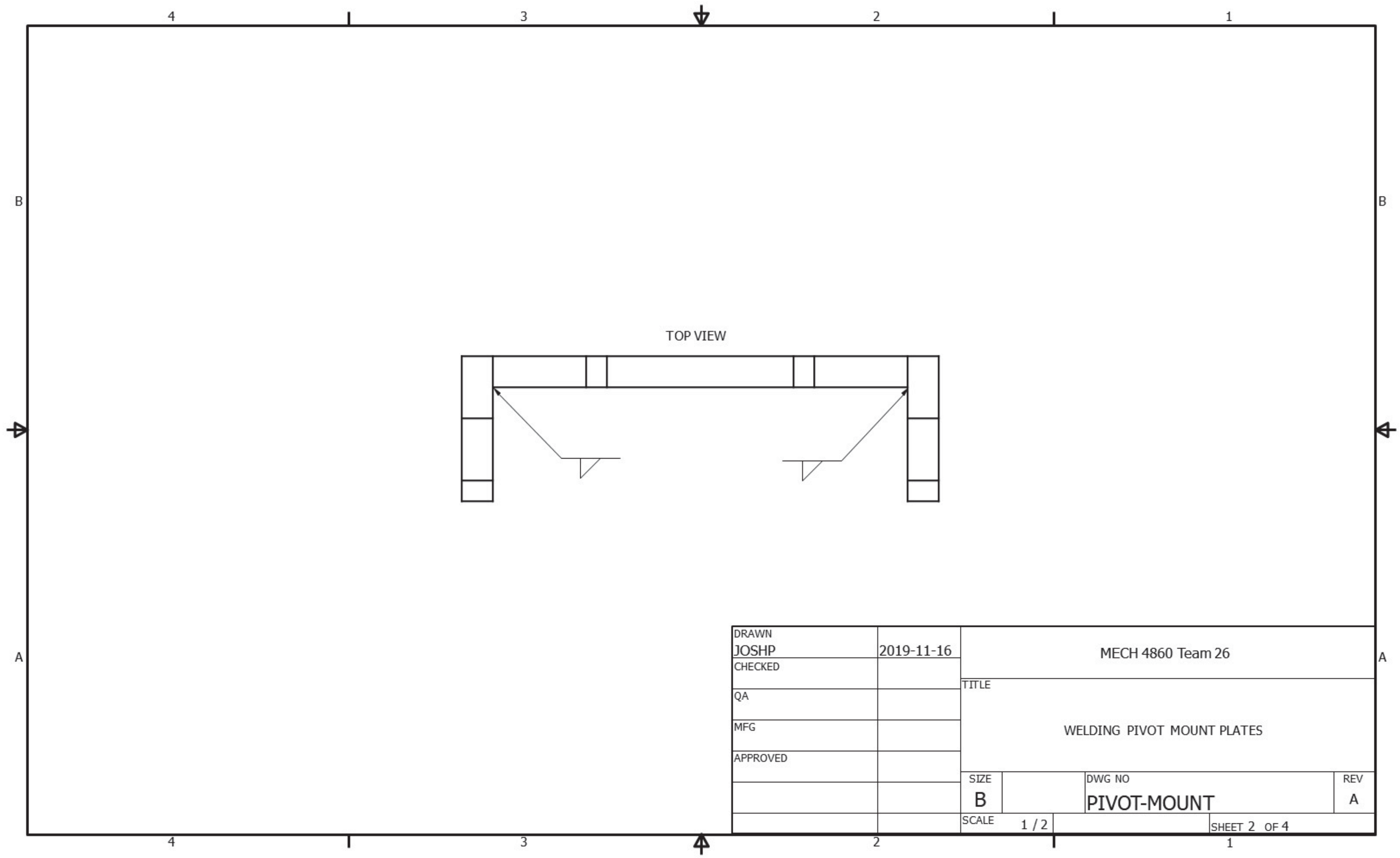
4
3
2
1
B
A
6.875
1.00
¢ 1.50
R1.25
4.125
1.50
8.70
.75
3.50
.75
3.00
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
PIVOT-BOTTOM-PLATE
SIZE
B
DWG NO
PIVOT-MOUNT-ASSEMBLY
REV
A
SCALE
1 / 2
SHEET 4 OF 5

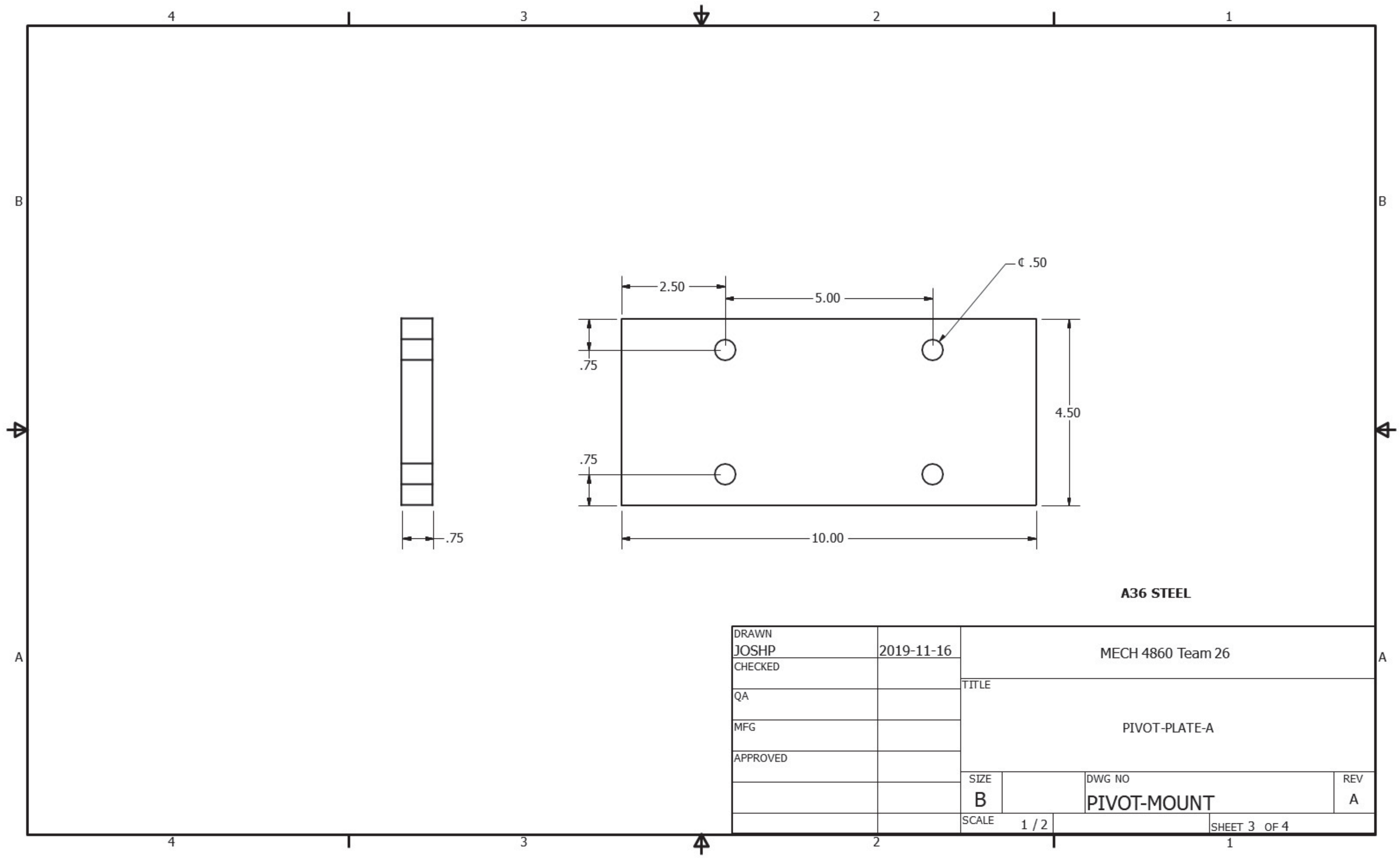

4
3
2
1
B
A
1.00
4.00
1.00
4.00
1.50
3.00
.75
21.50
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
PIVOT-BRACE
SIZE
B
DWG NO
PIVOT-MOUNT-ASSEMBLY
REV
A
SCALE
1 / 4
SHEET 5 OF 5

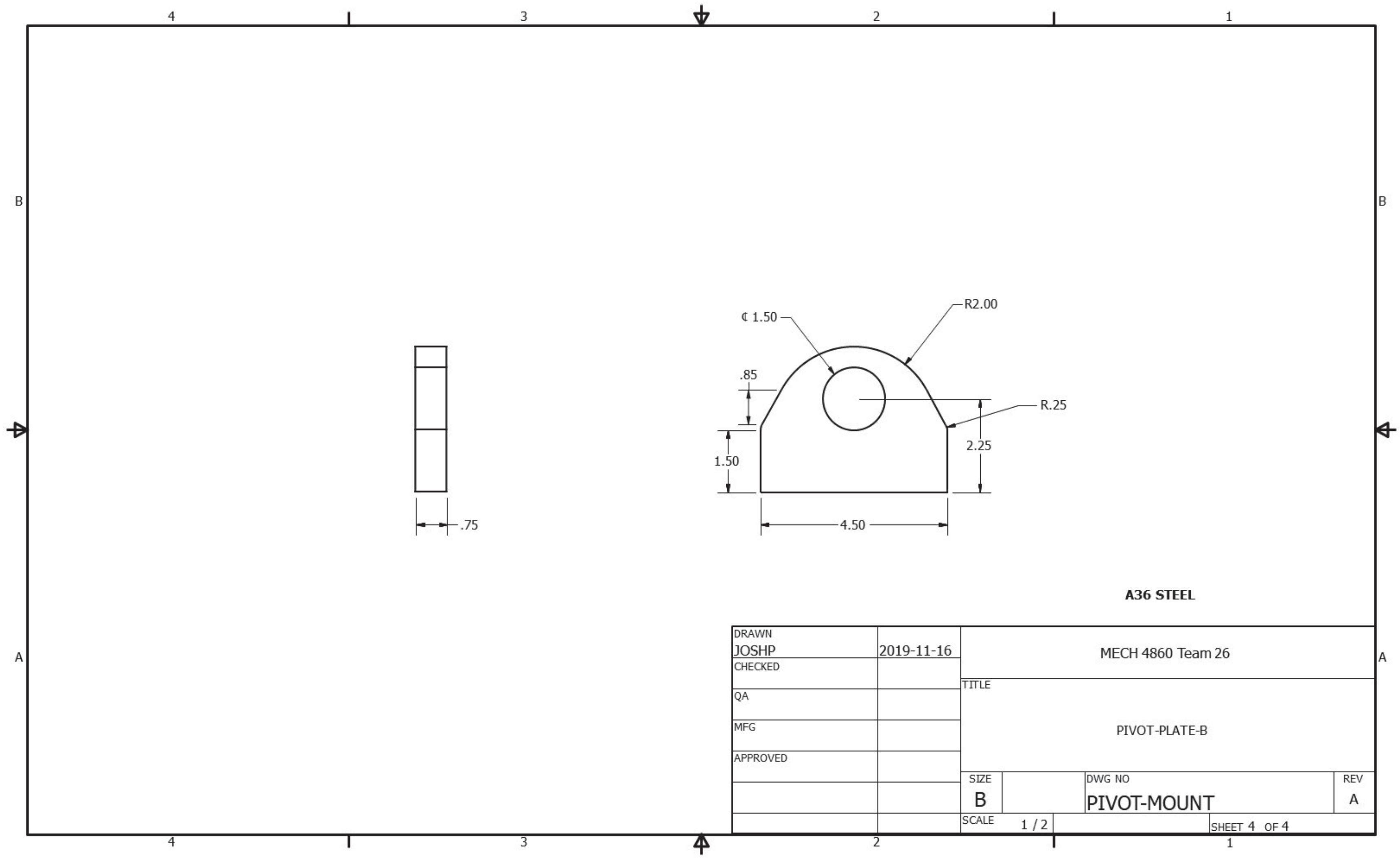


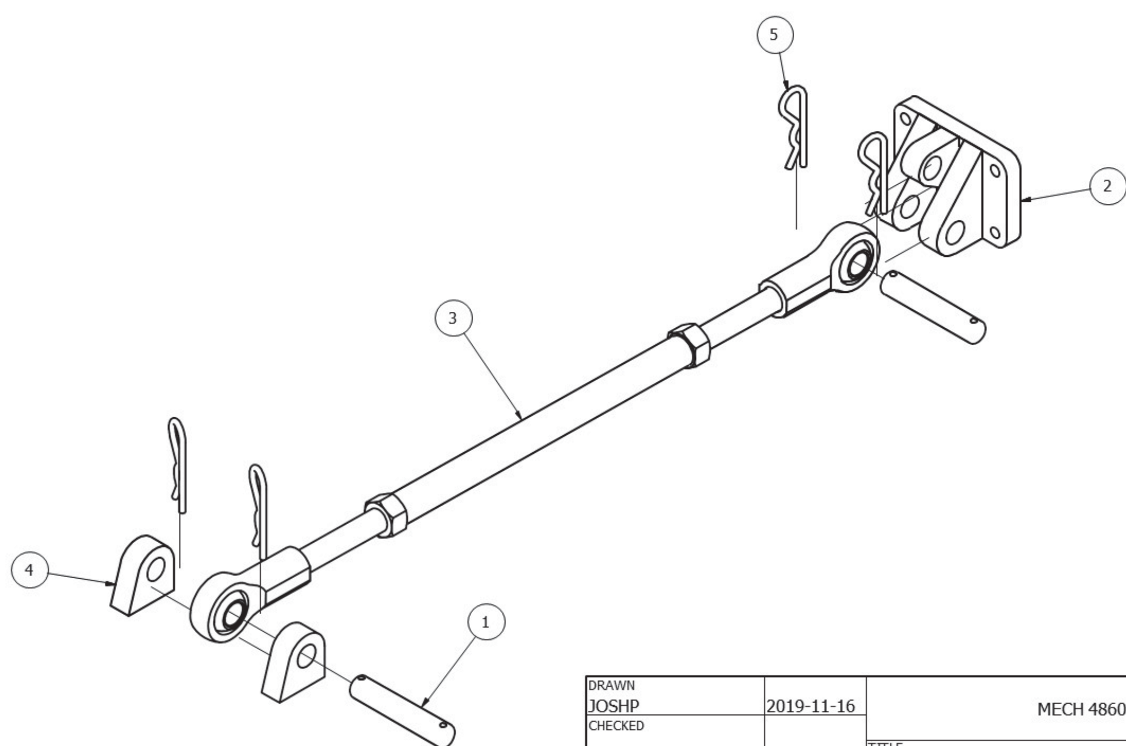
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	PIVOT-PLATE-A	
2	2	PIVOT-PLATE-B	

DRAWN JOSHP	2019-11-16	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF PIVOT MOUNT PLATE
QA		
MFG		
APPROVED		
		SIZE B / DWG NO PIVOT-MOUNT / REV A
		SCALE 1 / 2 / SHEET 1 OF 4


4
3
2
1
B
A
TOP VIEW
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
WELDING PIVOT MOUNT PLATES
SIZE
B
DWG NO
PIVOT-MOUNT
REV
A
SCALE
1 / 2
SHEET 2 OF 4


2.50
5.00
¢ .50
.75
.75
4.50
10.00
.75
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
PIVOT-PLATE-A
SIZE
B
DWG NO
PIVOT-MOUNT
REV
A
SCALE
1 / 2
SHEET 3 OF 4

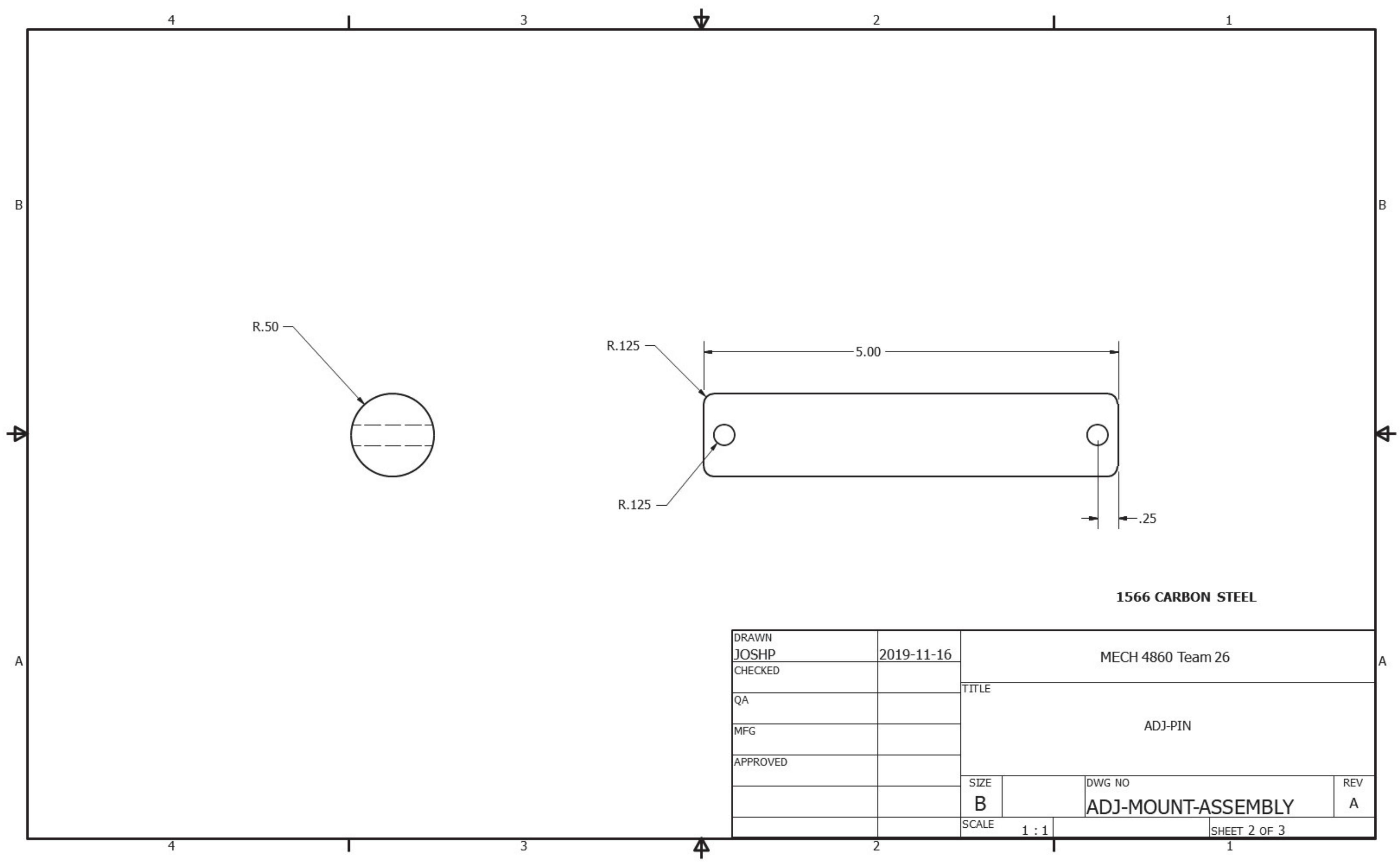

⊄ 1.50
R2.00
.85
R.25
1.50
2.25
4.50
.75
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
PIVOT-PLATE-B
SIZE
B
DWG NO
PIVOT-MOUNT
REV
A
SCALE
1 / 2
SHEET 4 OF 4

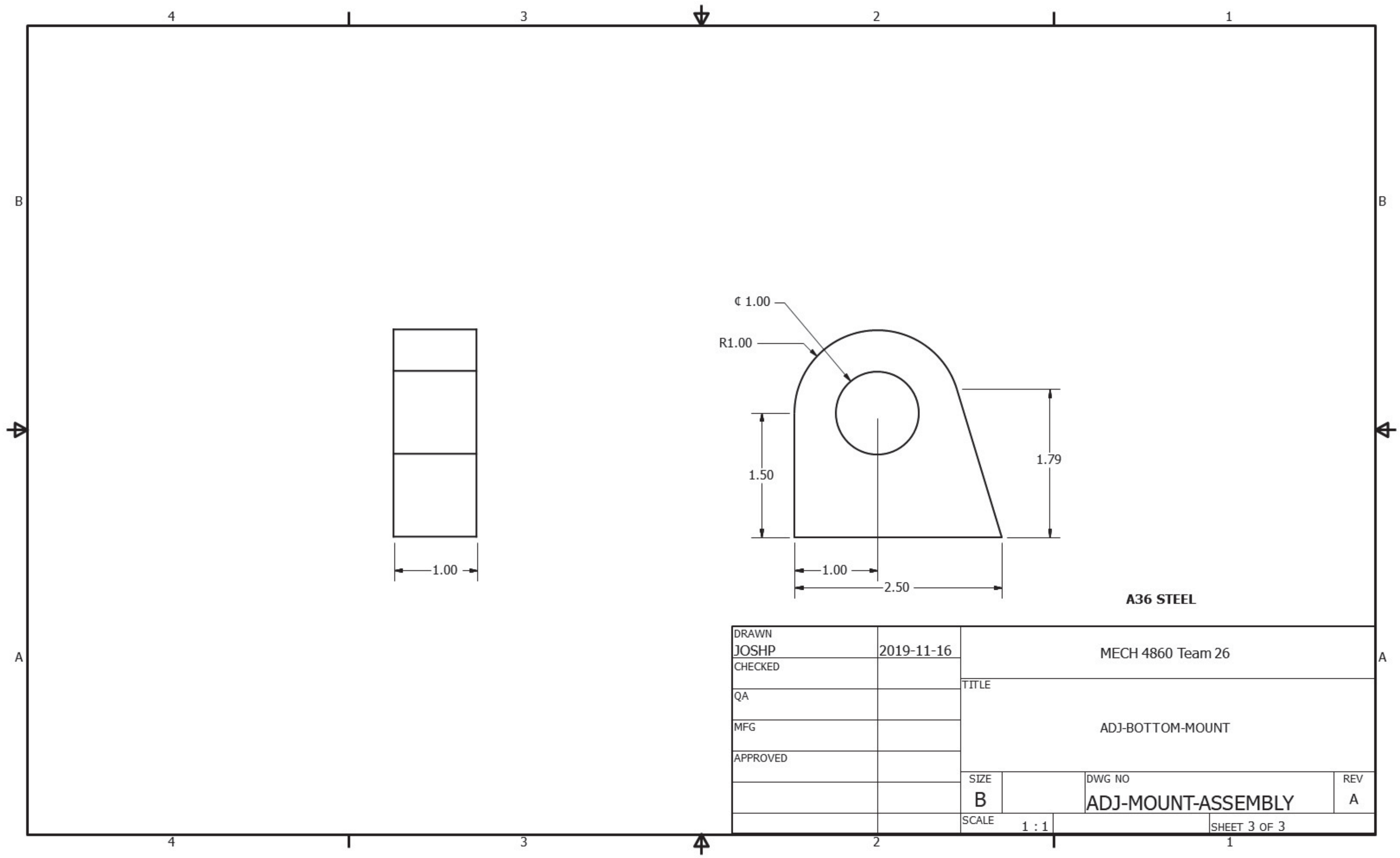


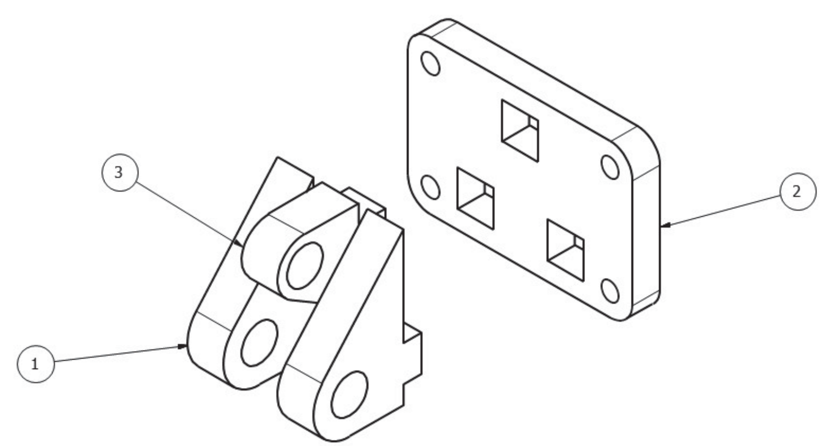
PARTS LIST

ITEM	QTY	PART NUMBER	DESCRIPTION
1	2	ADJ-PIN	
2	1	ADJ-MOUNT	
3	1	ADJ-ROD	
4	2	ADJ-BOTTOM-MOUNT	
5	4	ADJ-COTTER-PIN	92391A210

DRAWN JOSHP	2019-11-16	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF ADJUSTABLE MOUNT ASSEMBLY
QA		
MFG		
APPROVED		SIZE B; DWG NO ADJ-MOUNT-ASSEMBLY; REV A
		SCALE 1 / 4; SHEET 1 OF 3

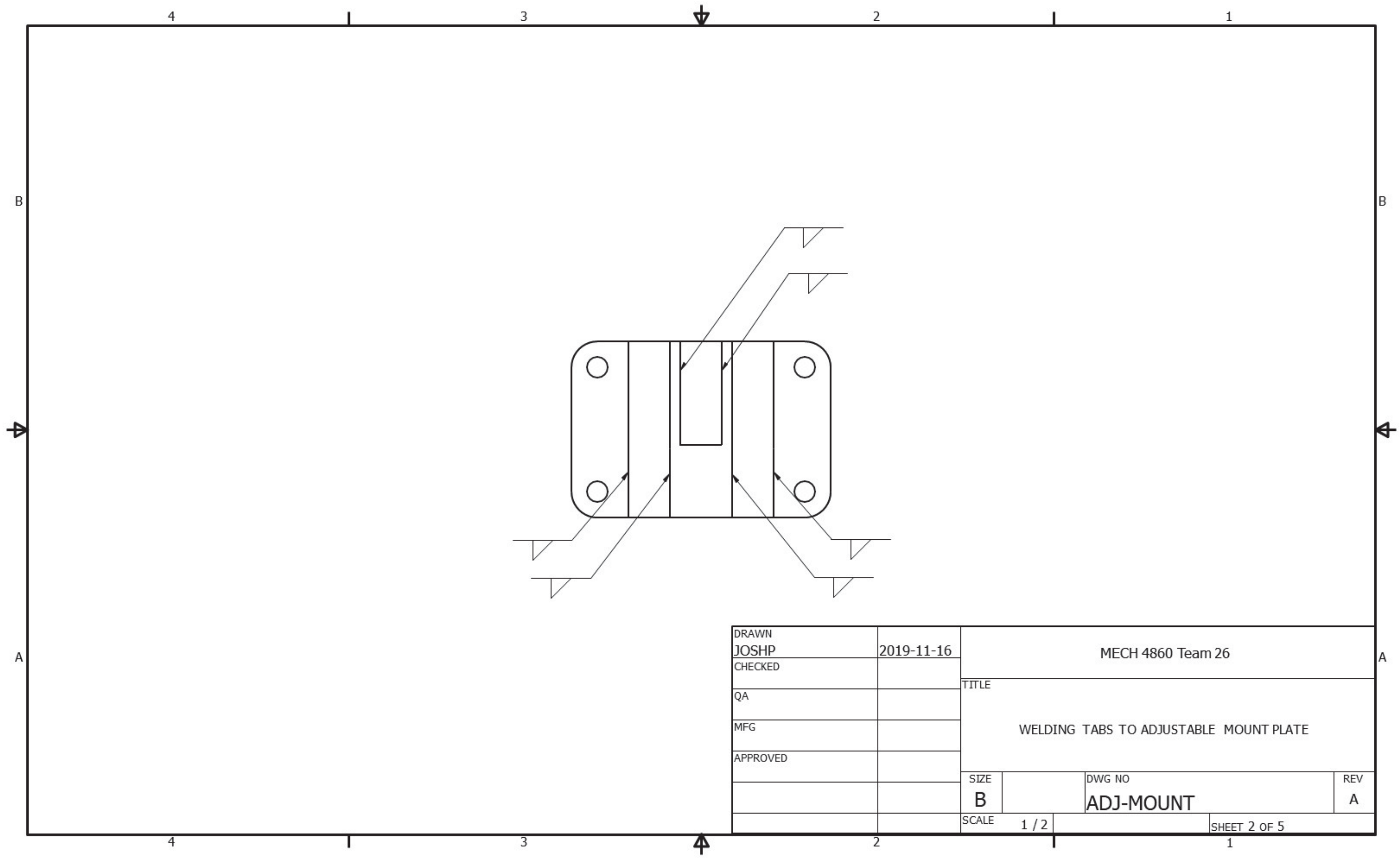

4
3
2
1
B
A
R.50
R.125
R.125
5.00
.25
1566 CARBON STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ADJ-PIN
SIZE
B
DWG NO
ADJ-MOUNT-ASSEMBLY
REV
A
SCALE
1 : 1
SHEET 2 OF 3


Ø 1.00
R1.00
1.50
1.79
1.00
2.50
1.00
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ADJ-BOTTOM-MOUNT
SIZE
B
DWG NO
ADJ-MOUNT-ASSEMBLY
REV
A
SCALE
1 : 1
SHEET 3 OF 3

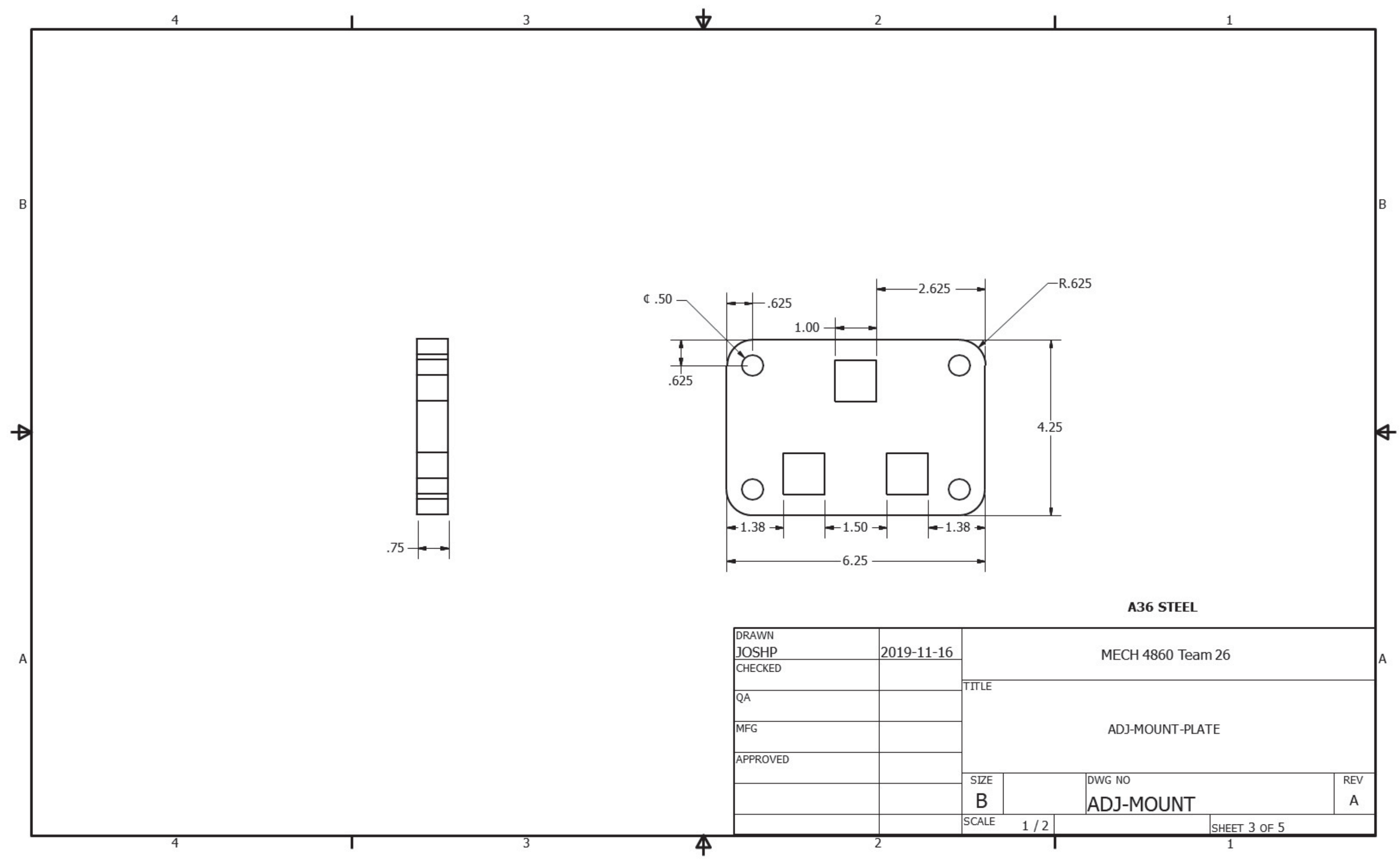


PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	2	MOUNT-BRKT-ADJ	
2	1	MOUNT-PLATE-ADJ	
3	1	ADJ-MOUNT-LIFT-BRKT	

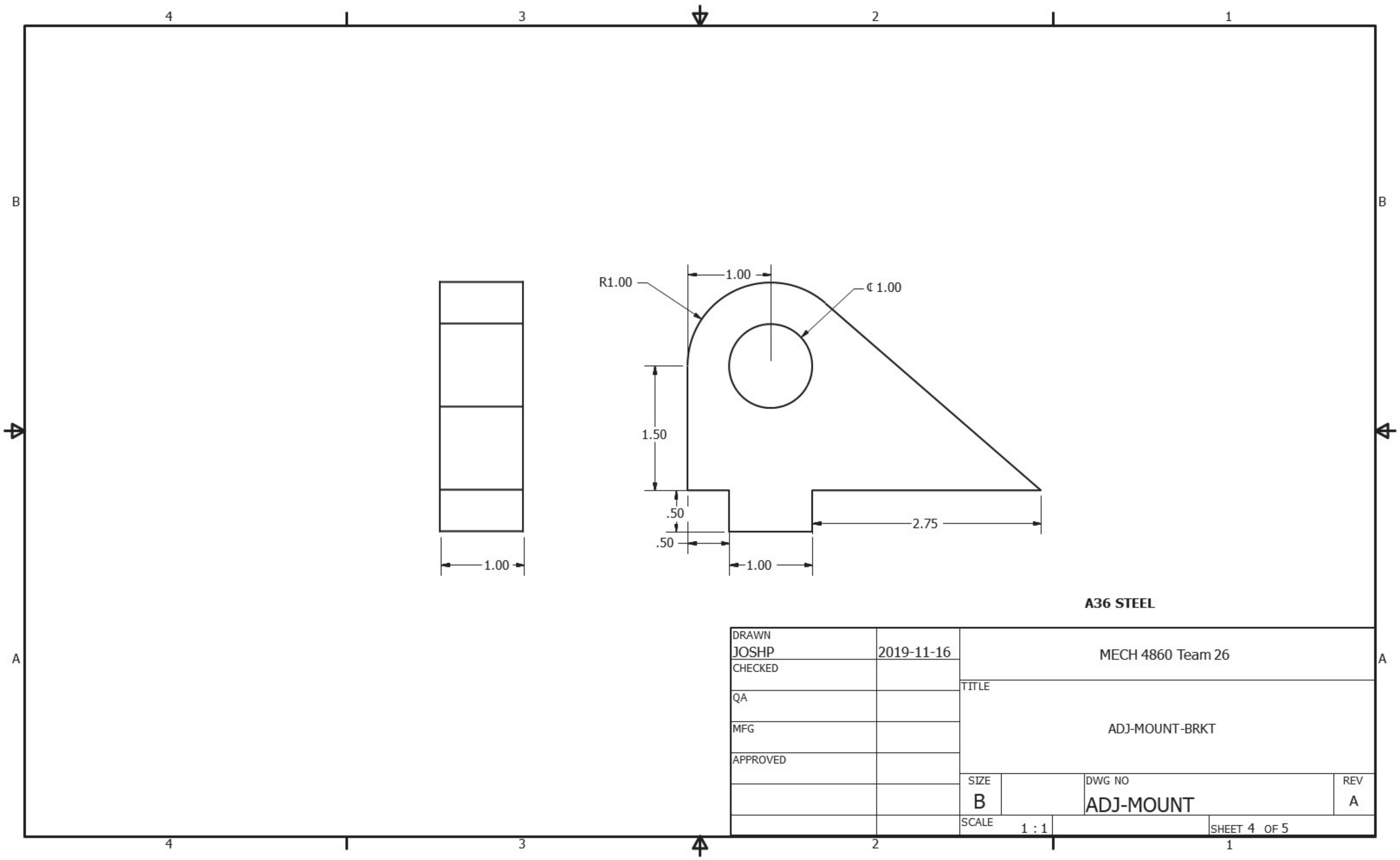
DRAWN JOSHP	2019-11-16	MECH 4860 Team 26
CHECKED		TITLE
QA		EXPLODED VIEW OF ADJUSTABLE MOUNT PLATE
MFG		
APPROVED		SIZE B · DWG NO ADJ-MOUNT · REV A
		SCALE 1 / 2 · SHEET 1 OF 5

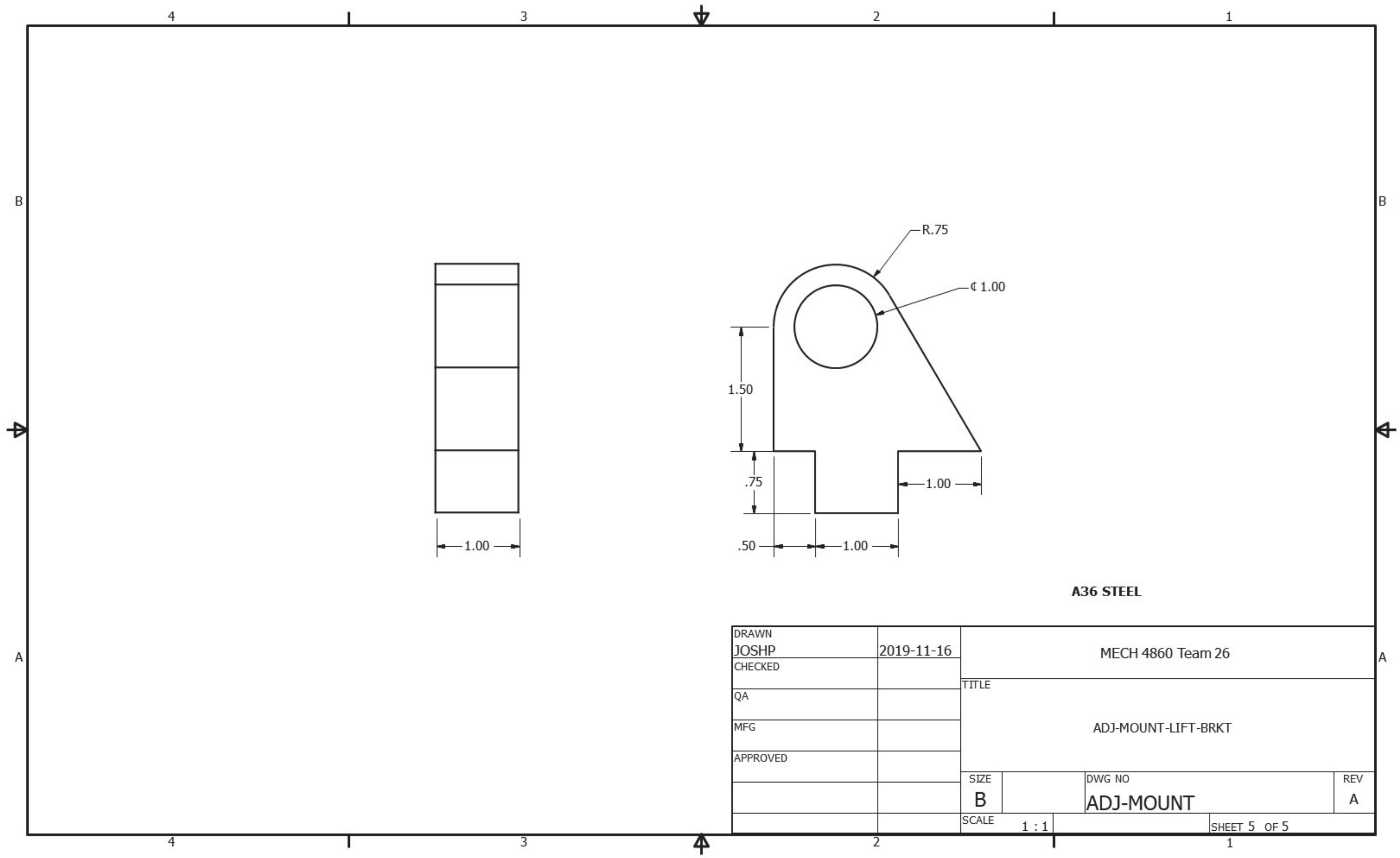


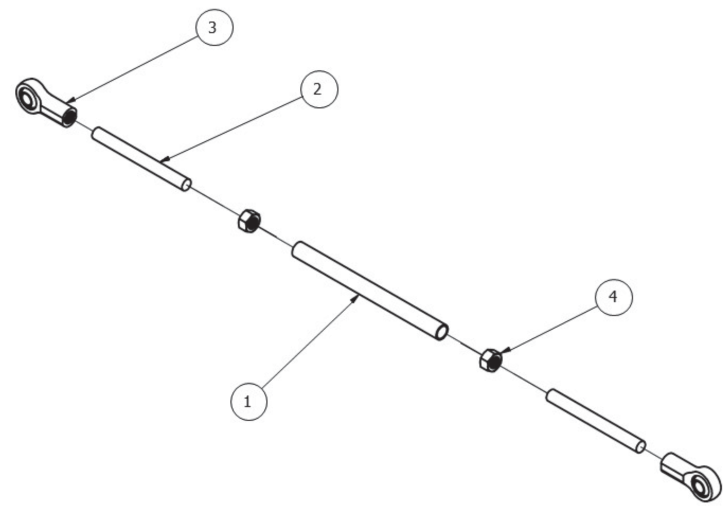
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
WELDING TABS TO ADJUSTABLE MOUNT PLATE
SIZE
B
DWG NO
ADJ-MOUNT
REV
A
SCALE
1 / 2
SHEET 2 OF 5



DRAWN JOSHP	2019-11-16	MECH 4860 Team 26	
CHECKED		TITLE ADJ-MOUNT-PLATE	
QA			
MFG			
APPROVED			
SIZE B	DWG NO ADJ-MOUNT		REV A
SCALE 1 / 2		SHEET 3 OF 5	

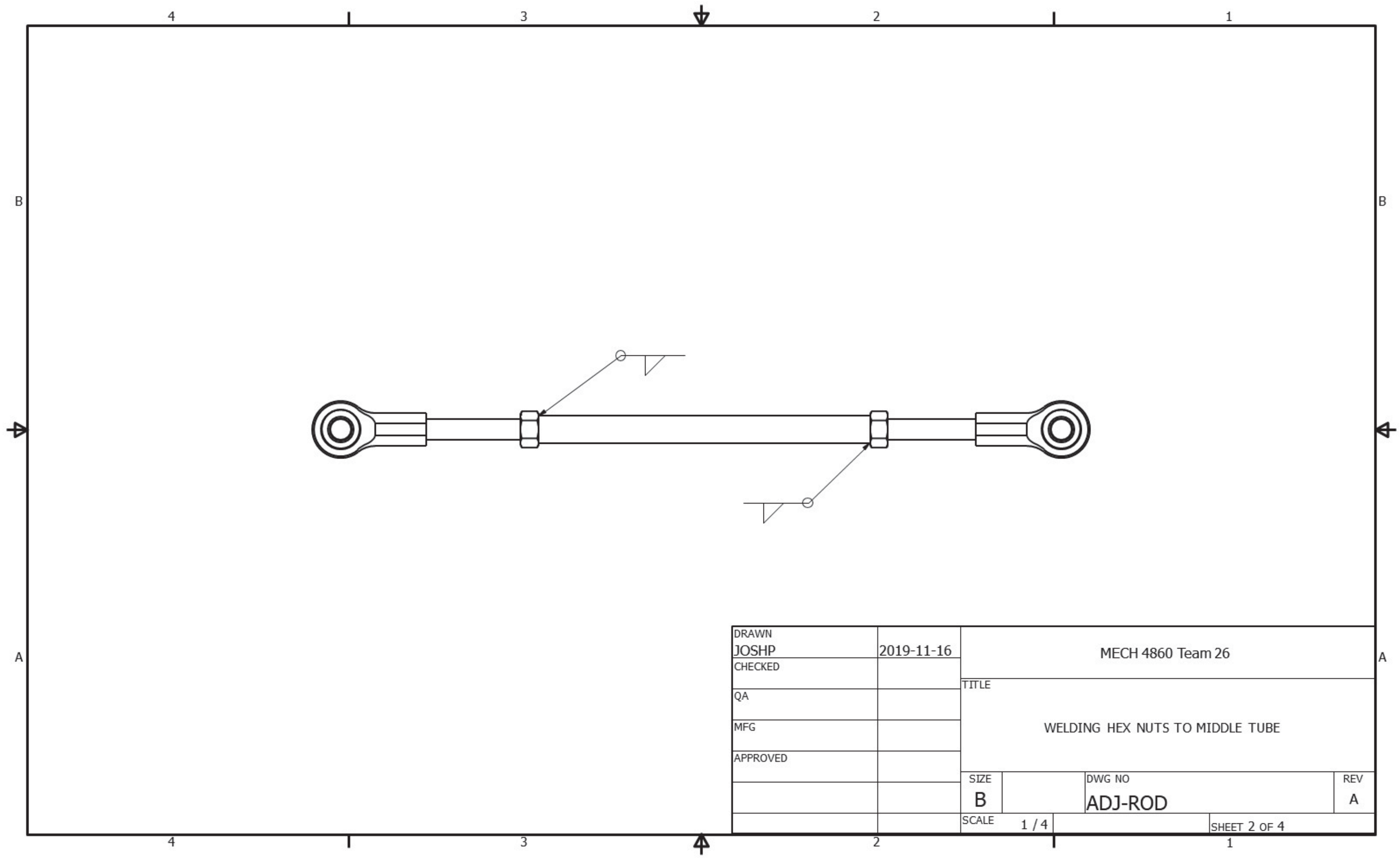

R1.00
1.00
⌀ 1.00
1.50
.50
.50
1.00
2.75
1.00
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ADJ-MOUNT-BRKT
SIZE
B
DWG NO
ADJ-MOUNT
REV
A
SCALE
1 : 1
SHEET 4 OF 5

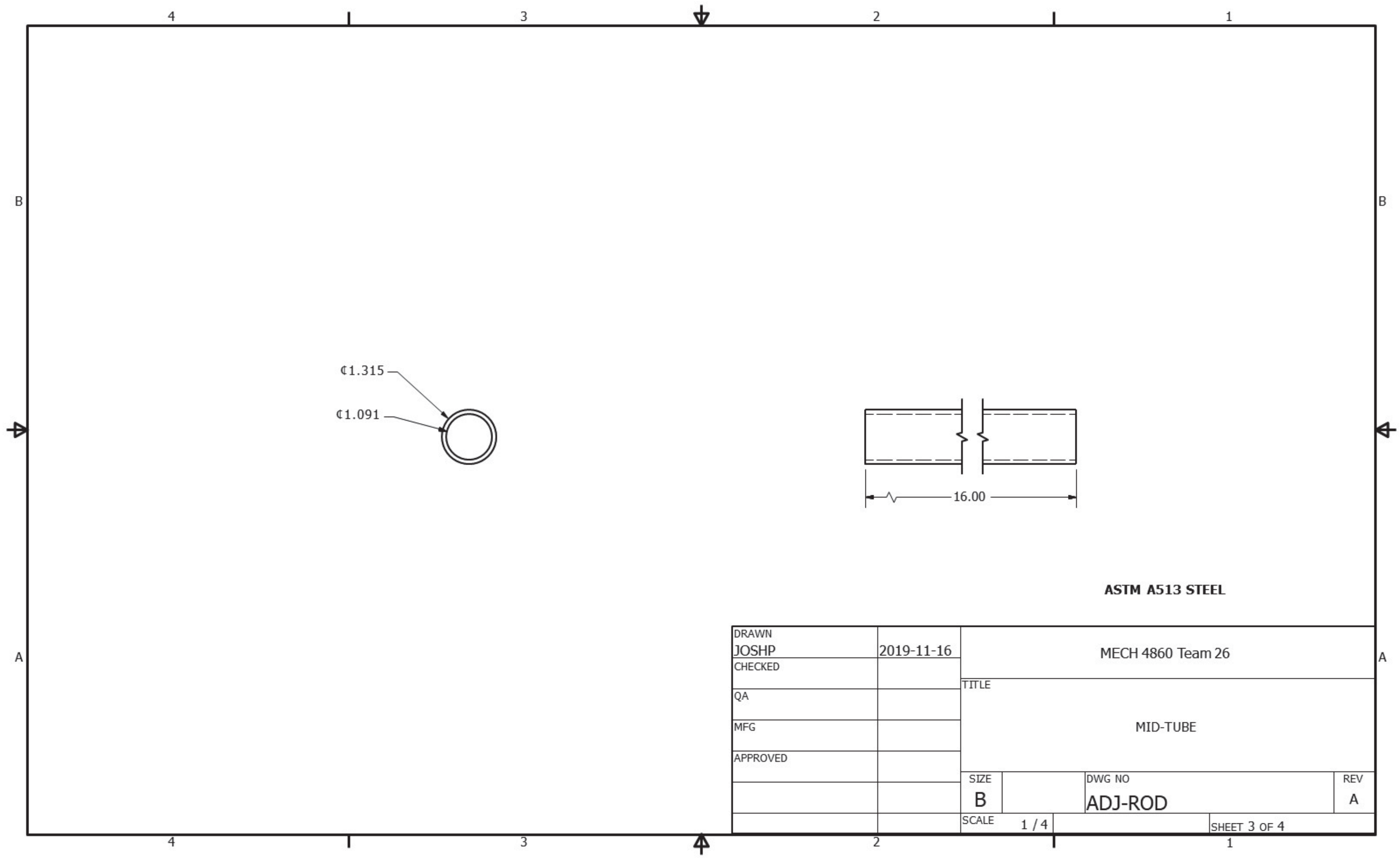

4
3
2
1
B
A
R.75
¢ 1.00
1.50
.75
1.00
.50
1.00
1.00
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ADJ-MOUNT-LIFT-BRKT
SIZE
B
DWG NO
ADJ-MOUNT
REV
A
SCALE
1 : 1
SHEET 5 OF 5



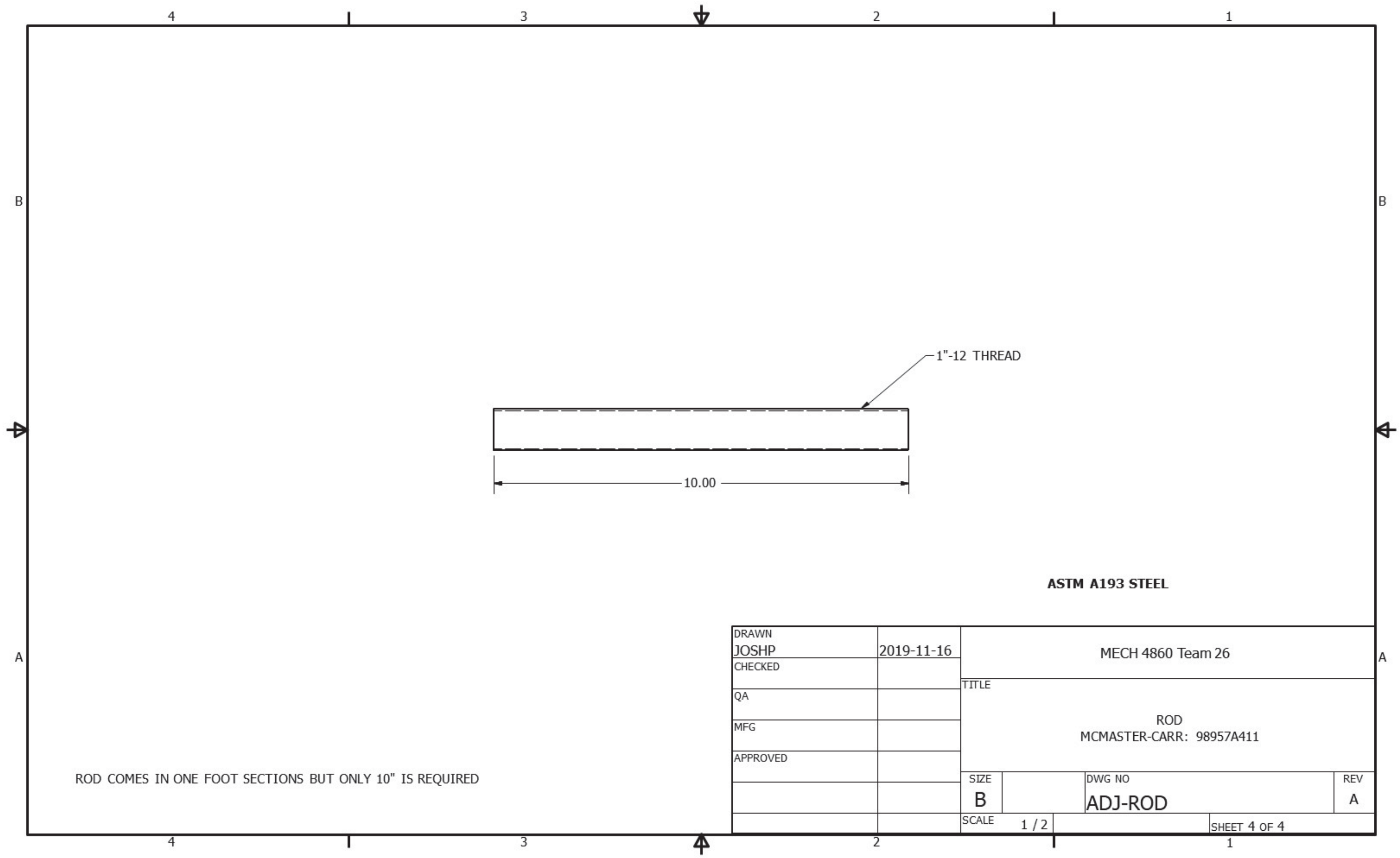
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	MID-TUBE	
2	2	ROD	98957A411
3	2	ROD-END	60645K72
4	2	HEX-NUT	95462A550

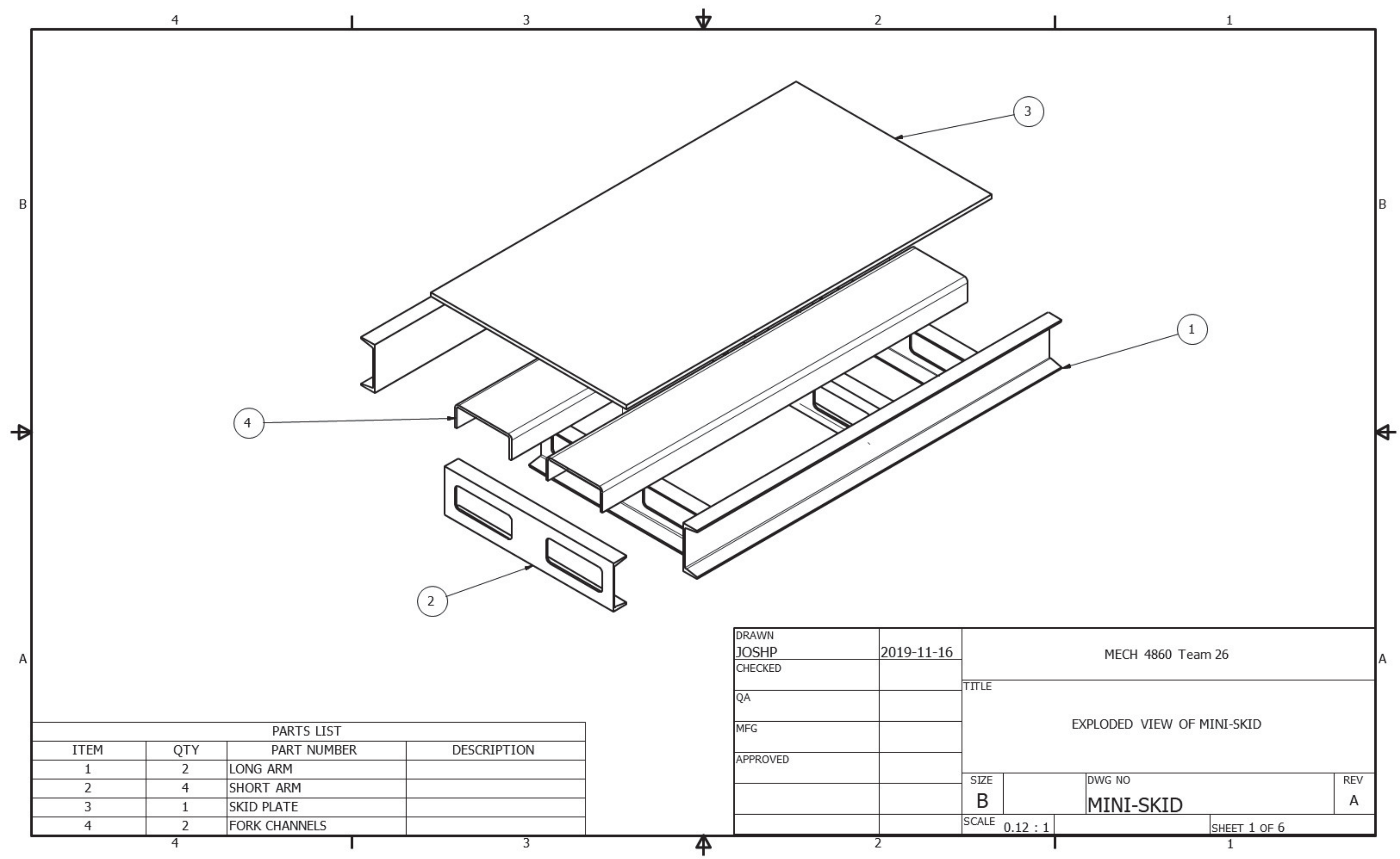
DRAWN JOSHP	2019-11-16	MECH 4860 Team 26
CHECKED		TITLE
QA		EXPLODED VIEW OF ADJUSTABLE ROD ASSEMBLY
MFG		
APPROVED		SIZE B — DWG NO ADJ-ROD — REV A
		SCALE 1 / 8 — SHEET 1 OF 4


4
3
2
1
B
A
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
WELDING HEX NUTS TO MIDDLE TUBE
SIZE
B
DWG NO
ADJ-ROD
REV
A
SCALE
1 / 4
SHEET 2 OF 4



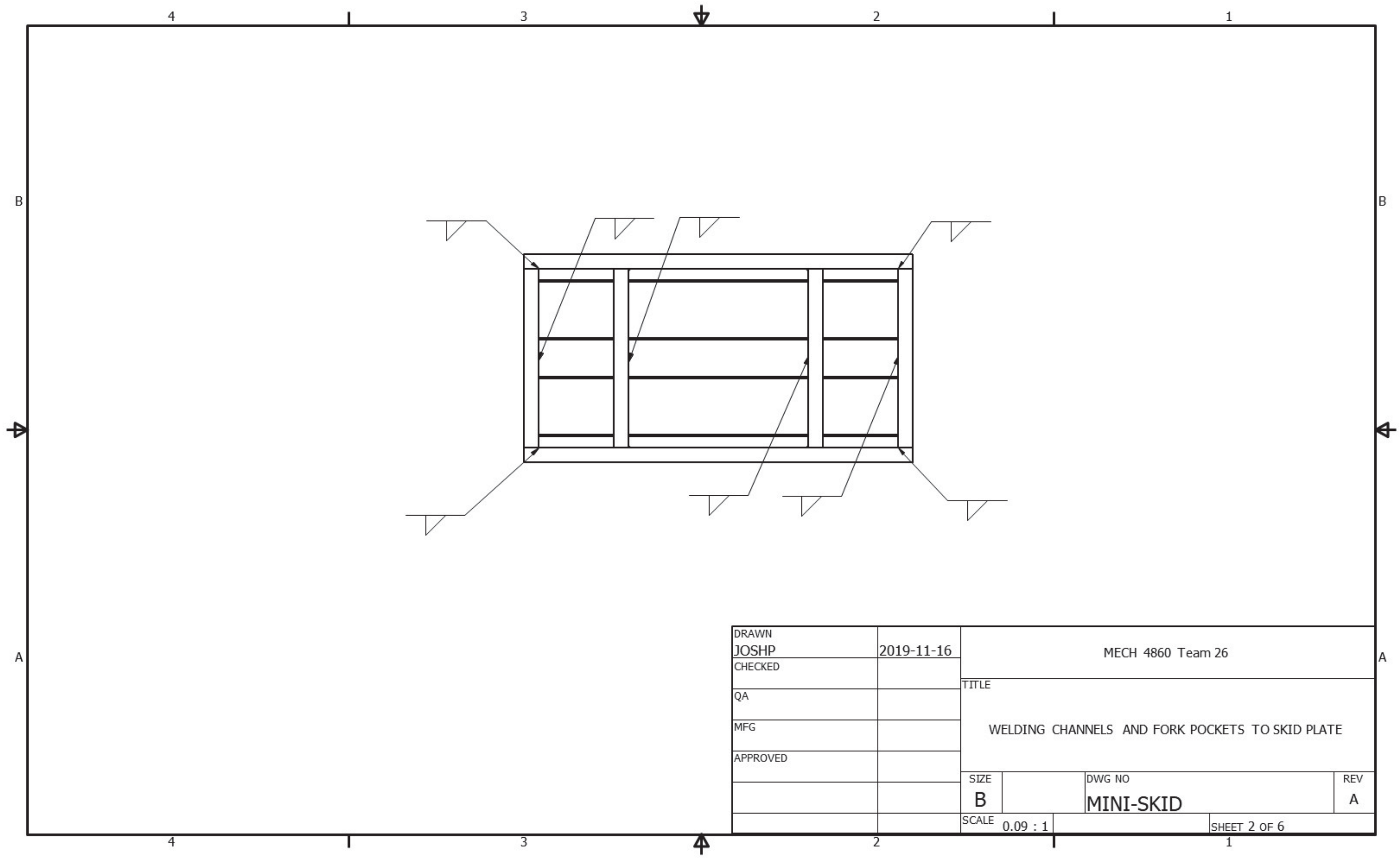
4
3
2
1
B
A
¢1.315
¢1.091
16.00
ASTM A513 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
MID-TUBE
SIZE
B
DWG NO
ADJ-ROD
REV
A
SCALE
1 / 4
SHEET 3 OF 4

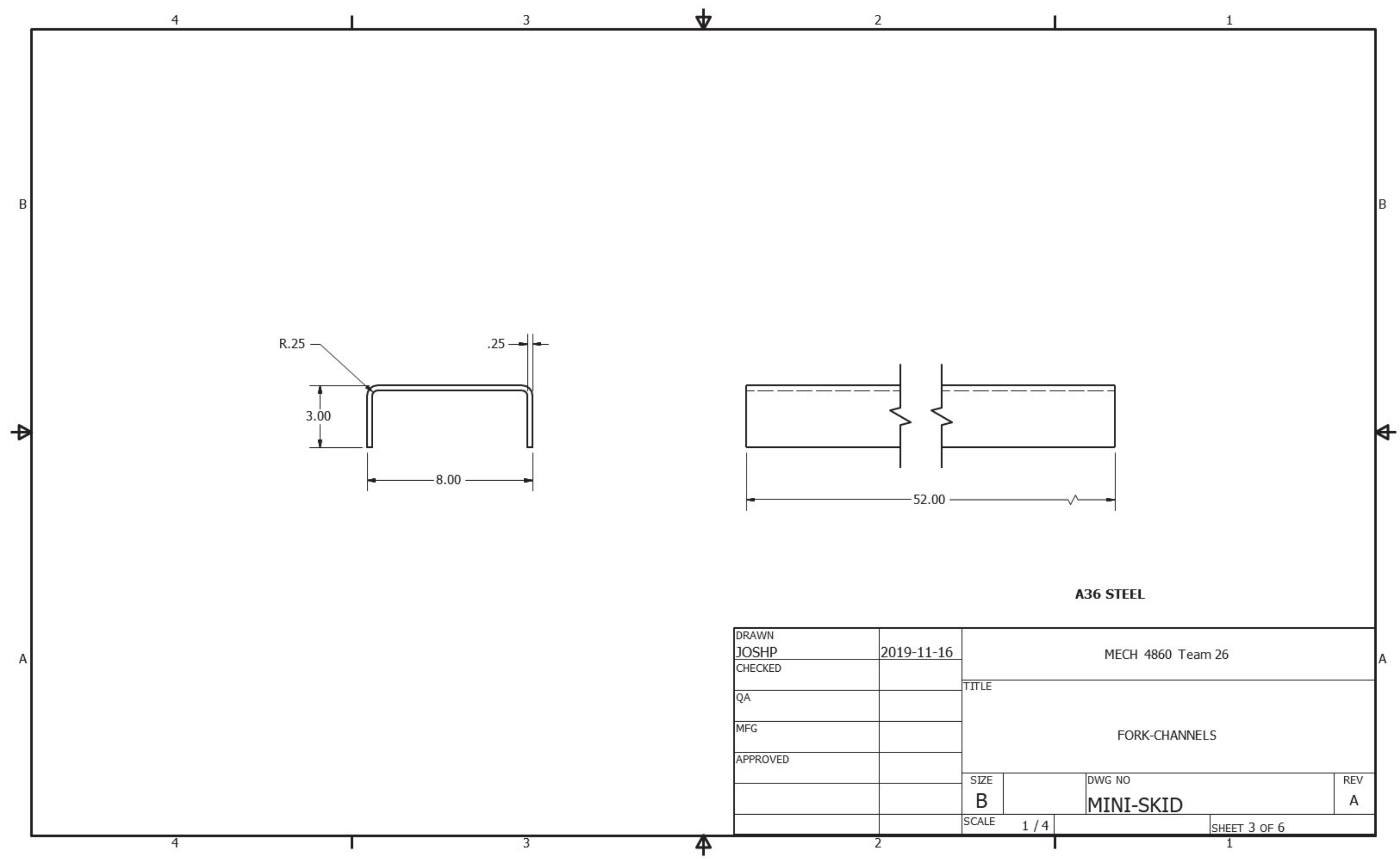

4
3
2
1
B
A
1"-12 THREAD
10.00
ASTM A193 STEEL
ROD COMES IN ONE FOOT SECTIONS BUT ONLY 10" IS REQUIRED
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ROD
MCMASTER-CARR: 98957A411
SIZE
B
DWG NO
ADJ-ROD
REV
A
SCALE
1 / 2
SHEET 4 OF 4



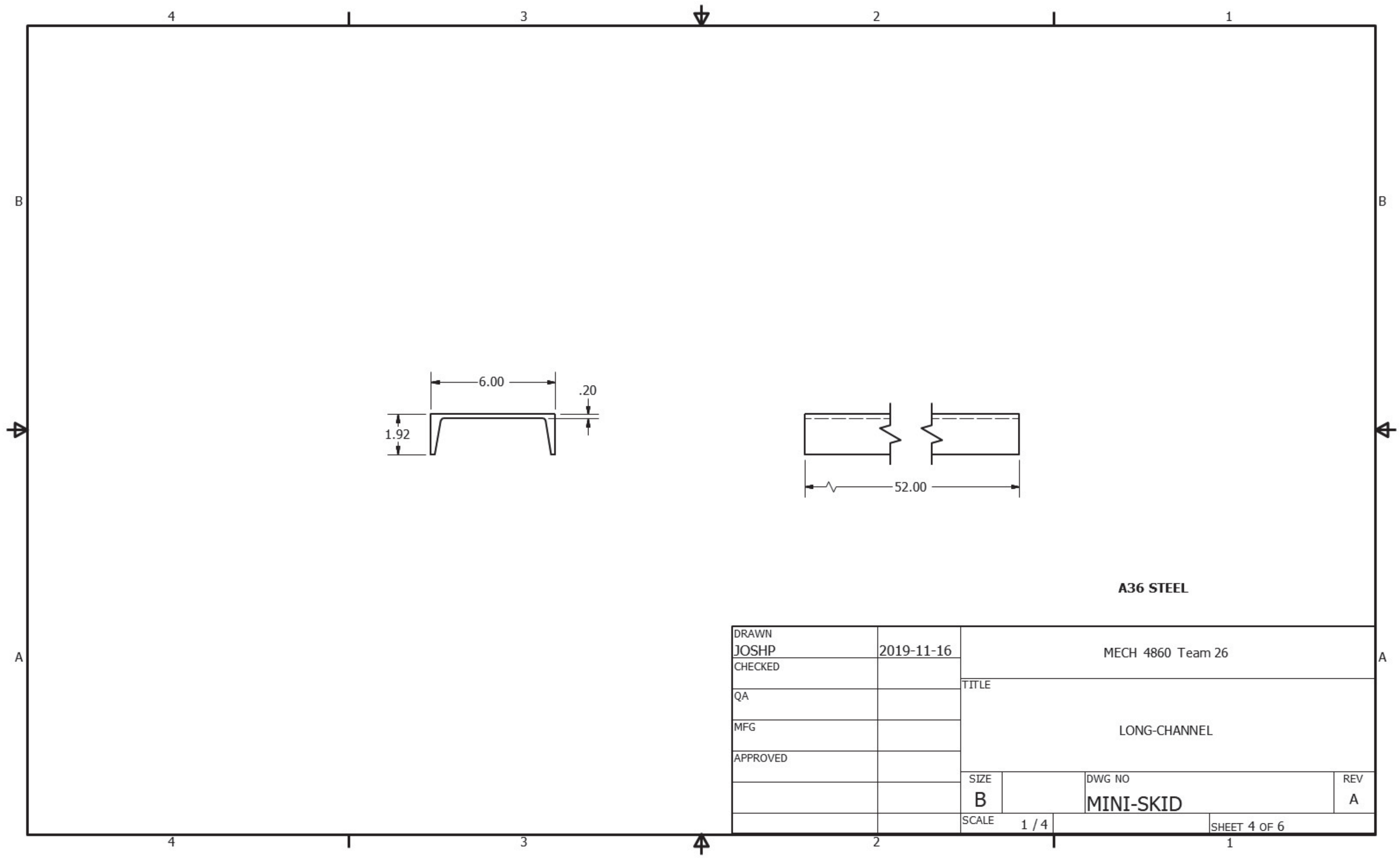
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	2	LONG ARM	
2	4	SHORT ARM	
3	1	SKID PLATE	
4	2	FORK CHANNELS	

DRAWN JOSHP	2019-11-16	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF MINI-SKID
QA		
MFG		
APPROVED		SIZE B — DWG NO MINI-SKID — REV A
		SCALE 0.12 : 1 — SHEET 1 OF 6

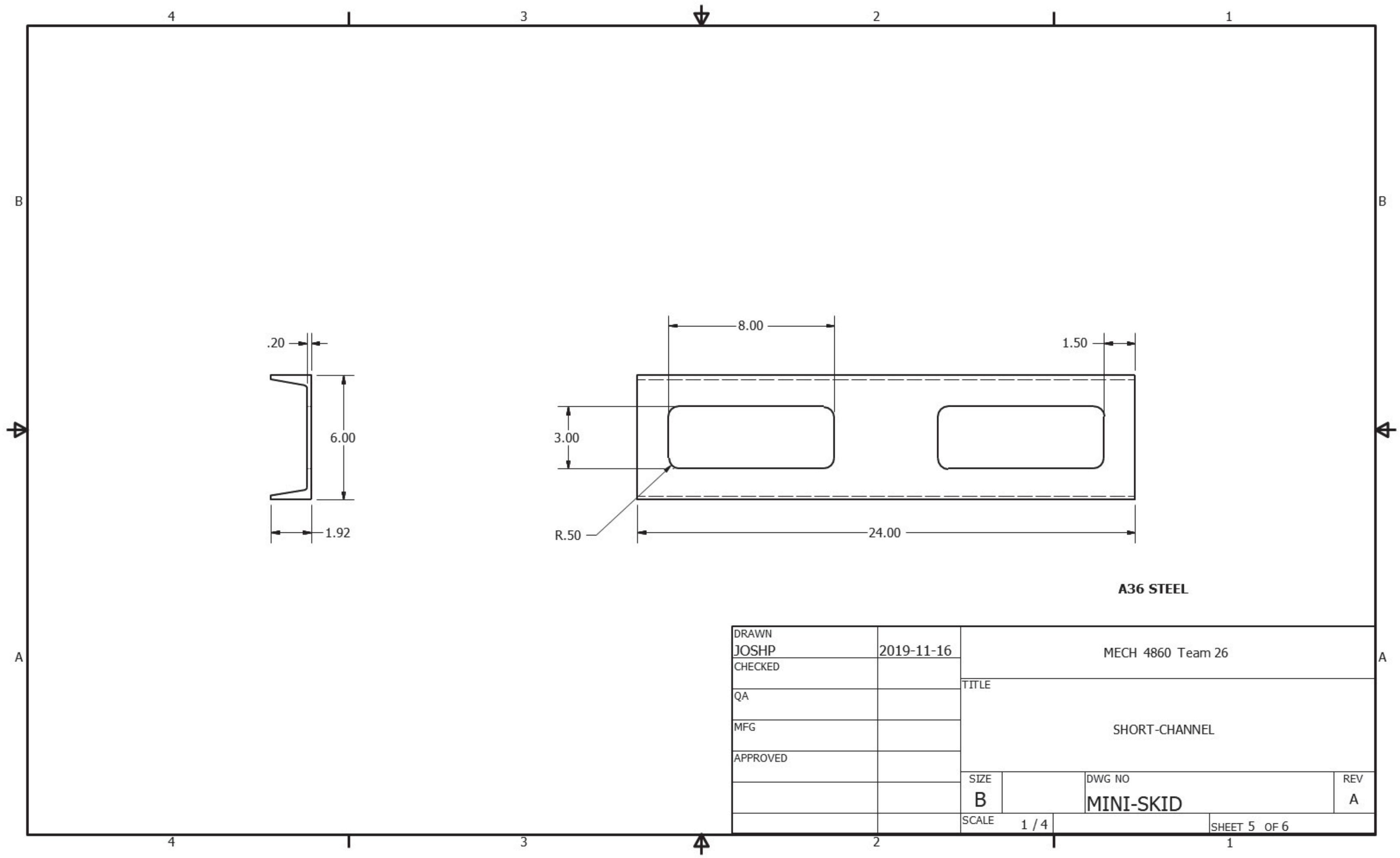

4
3
2
1
B
A
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
WELDING CHANNELS AND FORK POCKETS TO SKID PLATE
SIZE
B
DWG NO
MINI-SKID
REV
A
SCALE 0.09 : 1
SHEET 2 OF 6

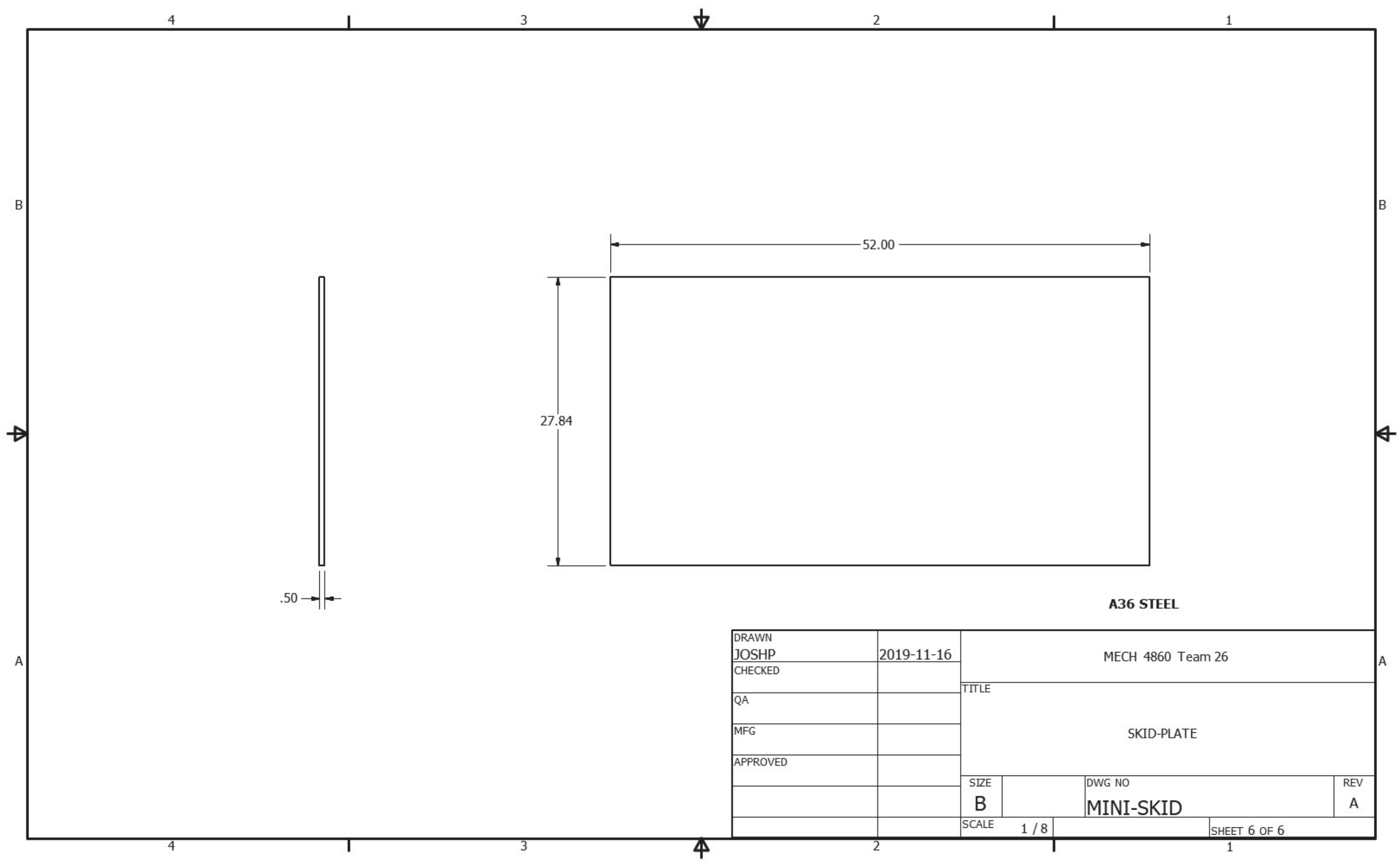


DRAWN JOSHP	2019-11-16	MECH 4860 Team 26	
CHECKED		TITLE FORK-CHANNELS	
QA			
MFG			
APPROVED			
		SIZE B	DWG NO MINI-SKID	REV A
		SCALE 1 / 4	SHEET 3 OF 6



6.00
.20
1.92
52.00
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
LONG-CHANNEL
SIZE
B
DWG NO
MINI-SKID
REV
A
SCALE
1 / 4
SHEET 4 OF 6

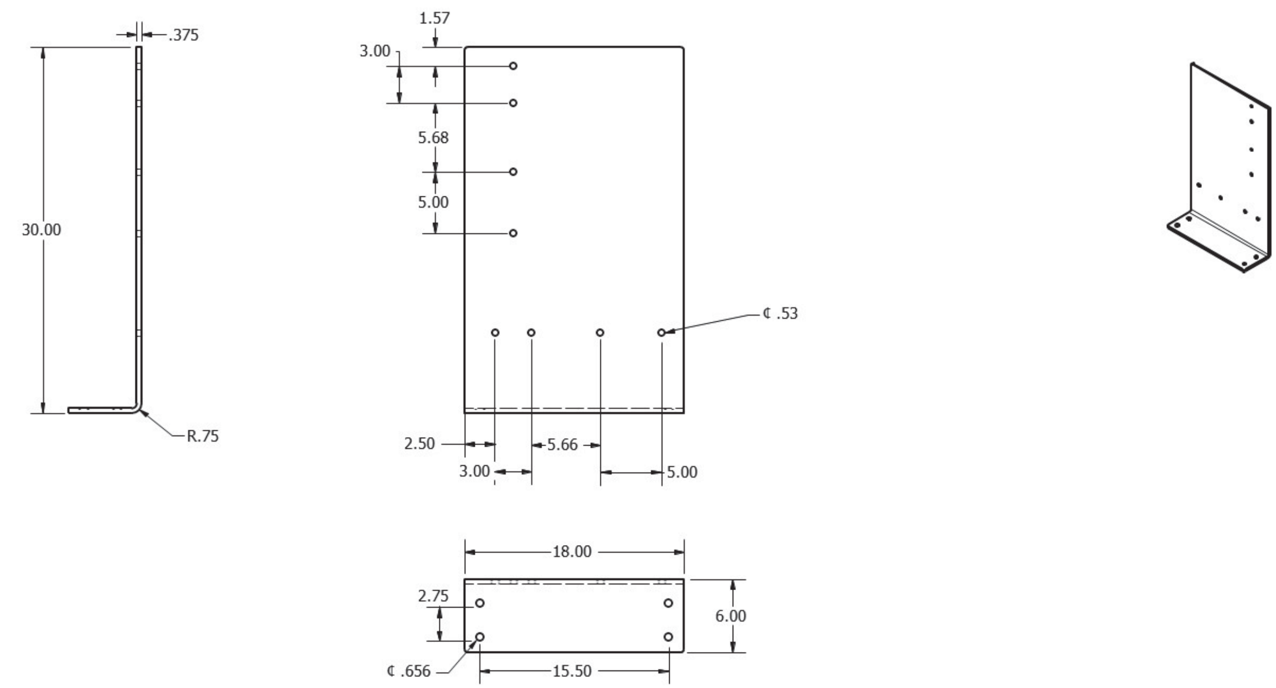

.20
6.00
1.92
8.00
1.50
3.00
R.50
24.00
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
SHORT-CHANNEL
SIZE
B
DWG NO
MINI-SKID
REV
A
SCALE 1 / 4
SHEET 5 OF 6


52.00
27.84
.50
A36 STEEL
DRAWN
JOSHP
2019-11-16
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
SKID-PLATE
SIZE
B
DWG NO
MINI-SKID
REV
A
SCALE 1 / 8
SHEET 6 OF 6

[redacted]

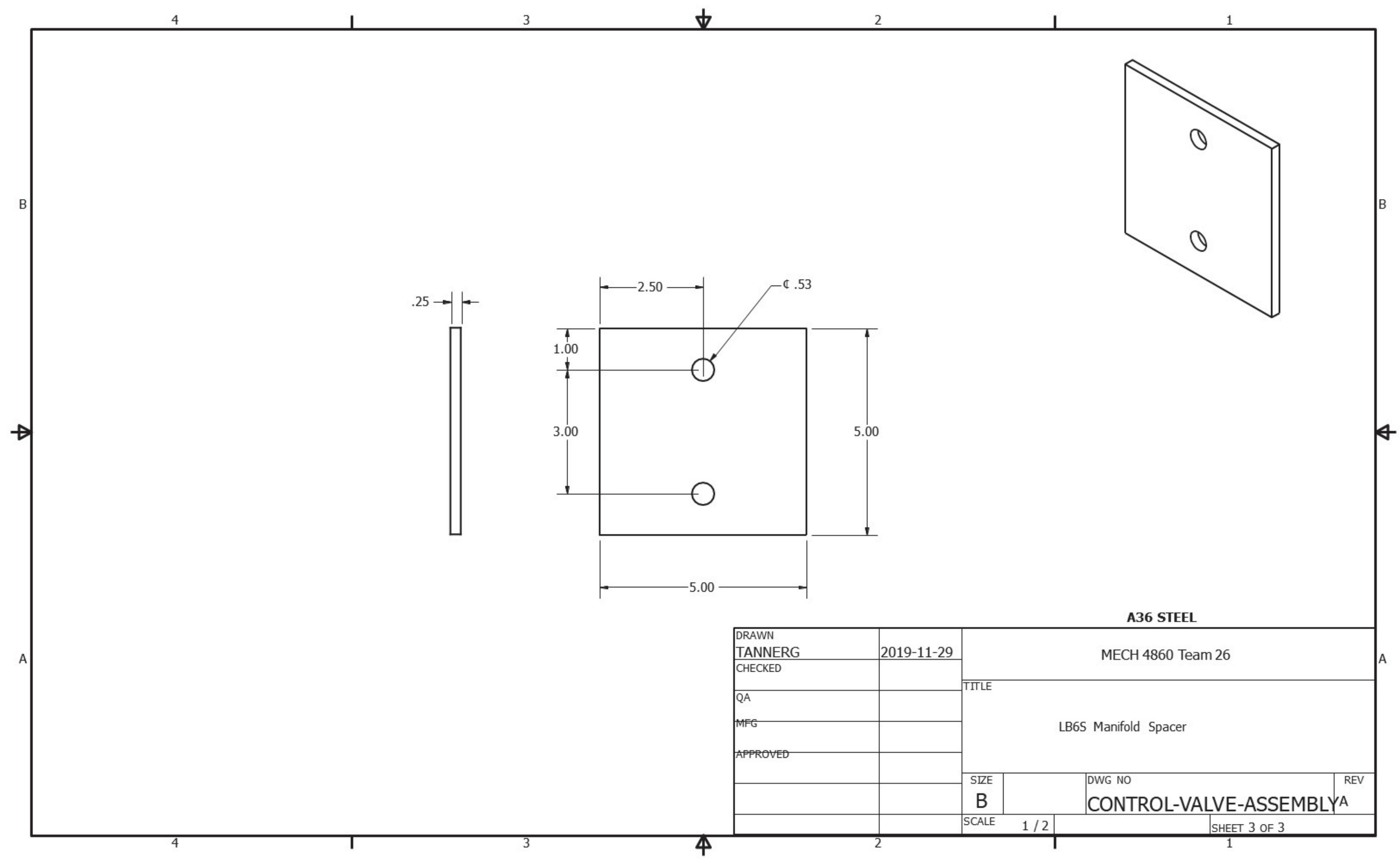
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	Manifold Mount Plate	
2	2	WAIS T-Block	
3	2	LB6S Manifold	
4	3	Sealum Code 62 Union 4.69in 1959-24-24	
5	2	LB6S Manifold Spacer	
6	16	1.5in Code 62 Split Flange	
7	2	RPKC8WV Control Valve	
8	1	Run04	

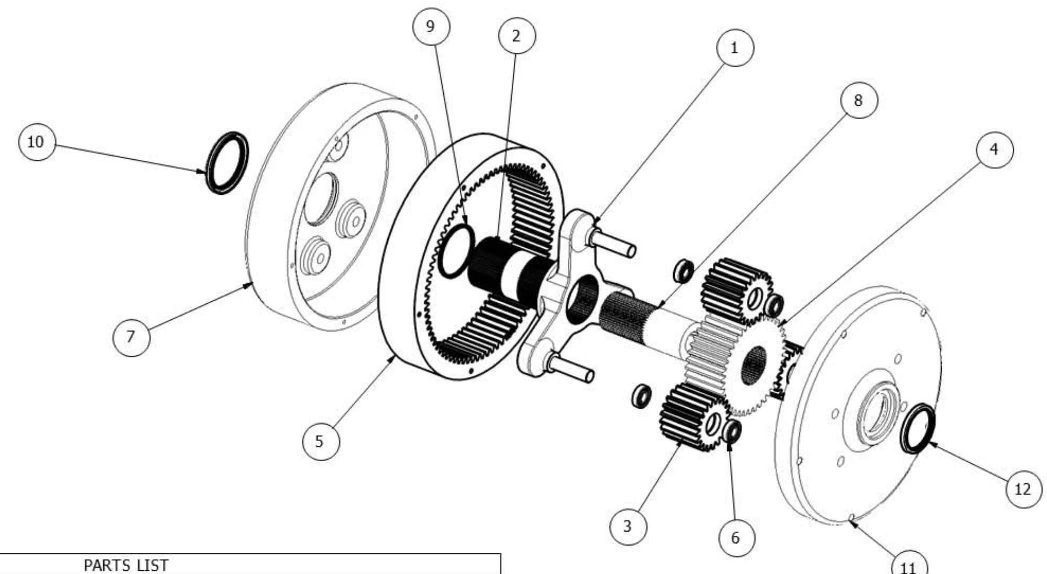
DRAWN TANNERG	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF CONTROL VALVE ASSEMBLY
QA		
MFG		
APPROVED		SIZE B; DWG NO CONTROL-VALVE-ASSEMBLY; REV A
		SCALE 0.12 : 1; SHEET 1 OF 3



A36 STEEL

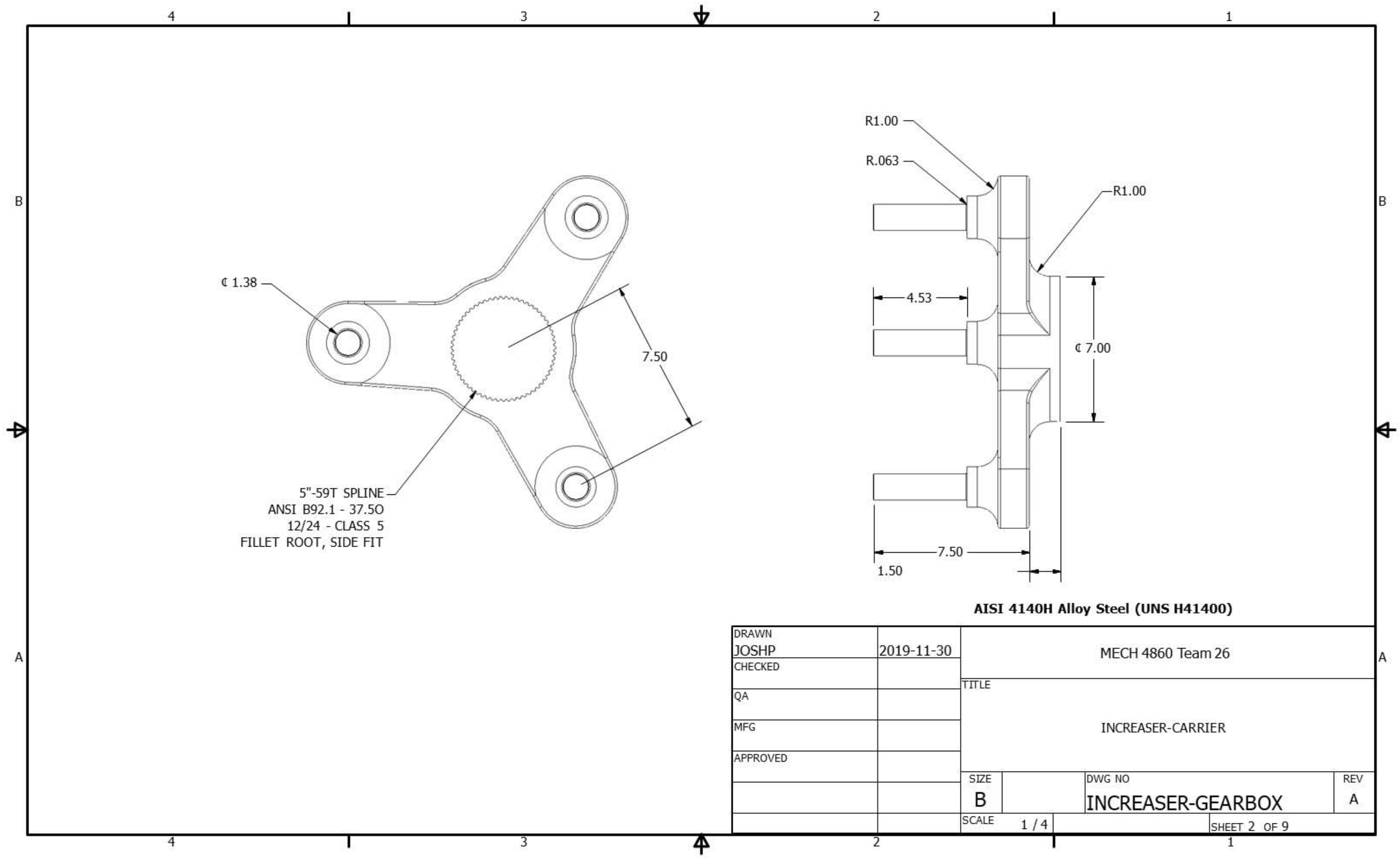
DRAWN TANNERG	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE CONTROL VALVE MOUNTING PLATE
QA		
MFG		
APPROVED		SIZE B / DWG NO CONTROL-VALVE-ASSEMBLY / REV A
		SCALE 0.06 : 1 / SHEET 2 OF 3

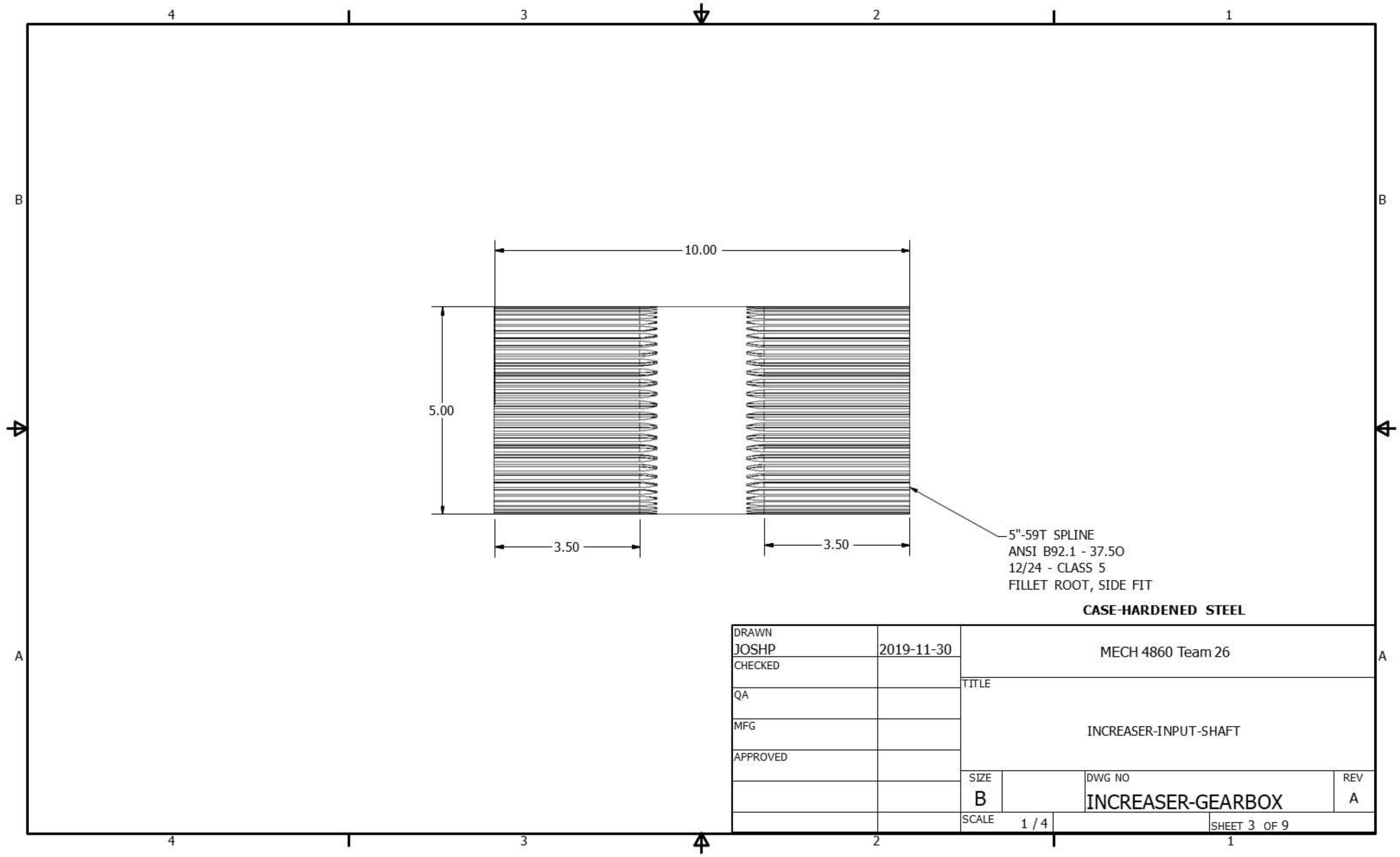

.25
2.50
⌀ .53
1.00
3.00
5.00
5.00
A36 STEEL
DRAWN
TANNERG
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
LB6S Manifold Spacer
SIZE
B
DWG NO
CONTROL-VALVE-ASSEMBLY
REV
A
SCALE
1 / 2
SHEET 3 OF 3

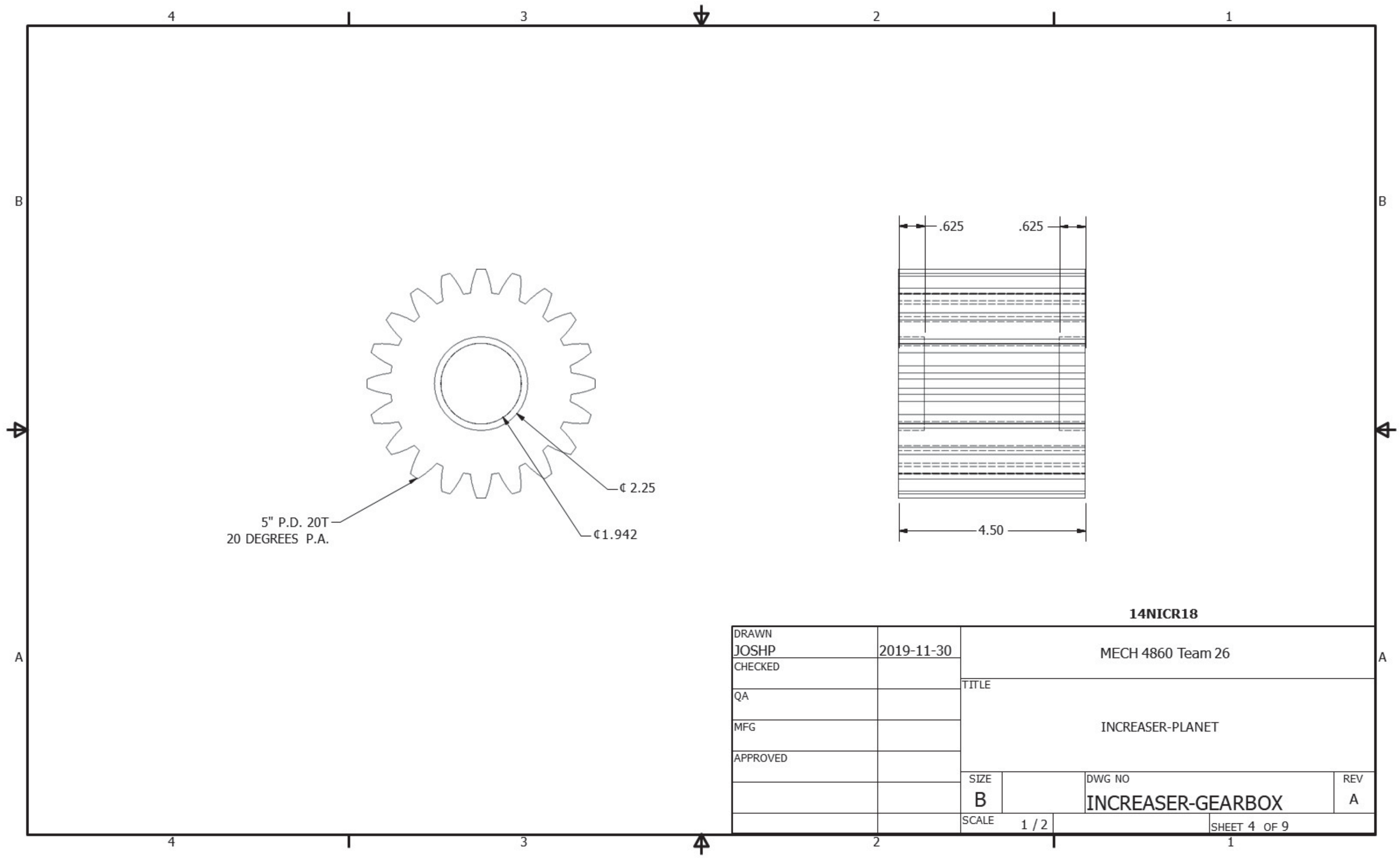


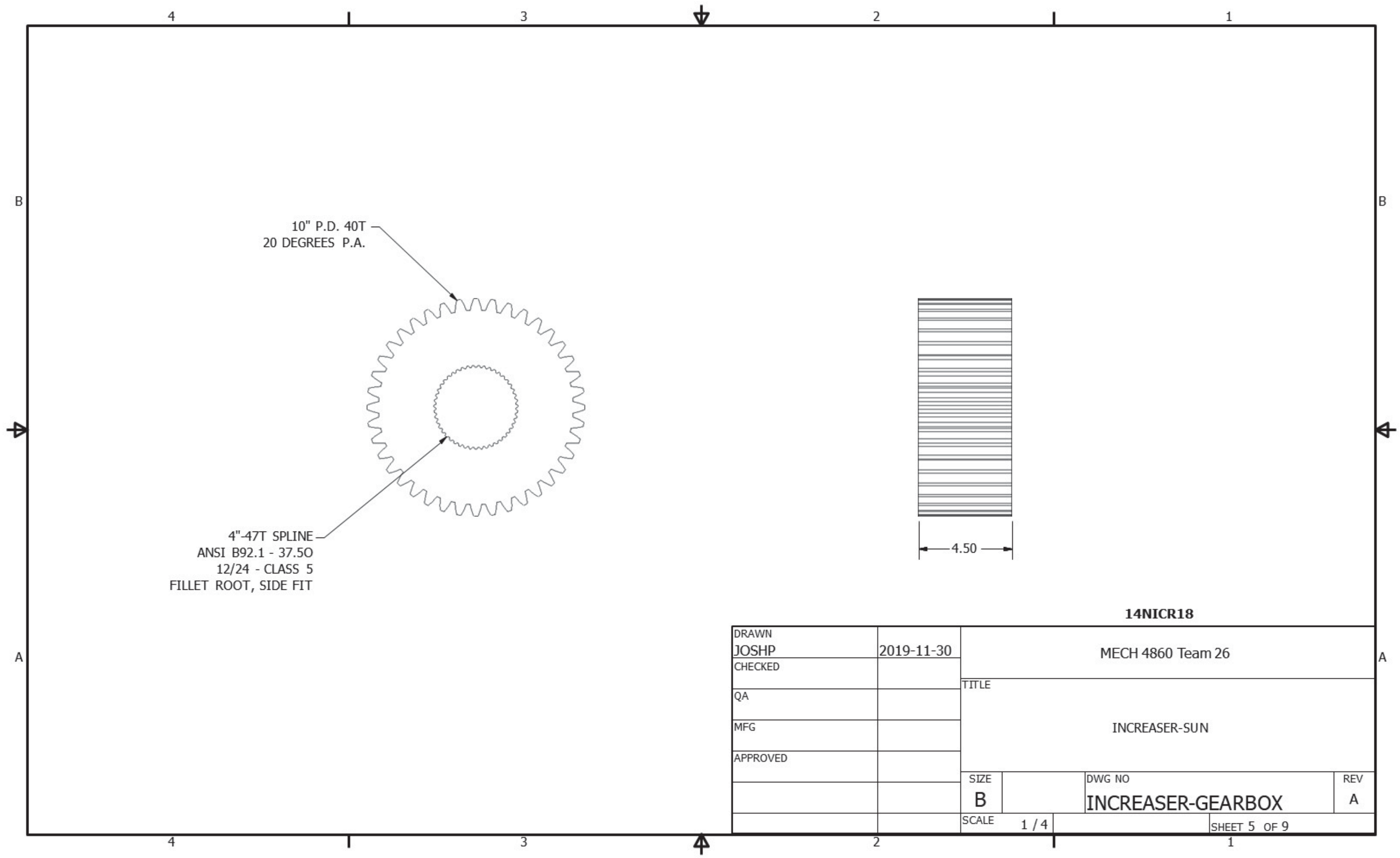
PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	INCREASER-CARRIER	
2	1	INCREASER-INPUT-SHAFT	
3	3	INCREASER-PLANET	
4	1	INCREASER-SUN	
5	1	INCREASER-RING	
6	6	INCREASER-PLANET-BEARING	SKFRLS8
7	1	INCREASER-INPUT-END-PLATE	
8	1	INCREASER-OUTPUT-SHAFT	
9	1	INCREASER-INPUT-BEARING	6656K160
10	1	INCREASER-INPUT-SEAL	SKF49985
11	1	INCREASER-OUTPUT-END-PLATE	
12	1	INCREASER-OUTPUT-SEAL	SKF39934

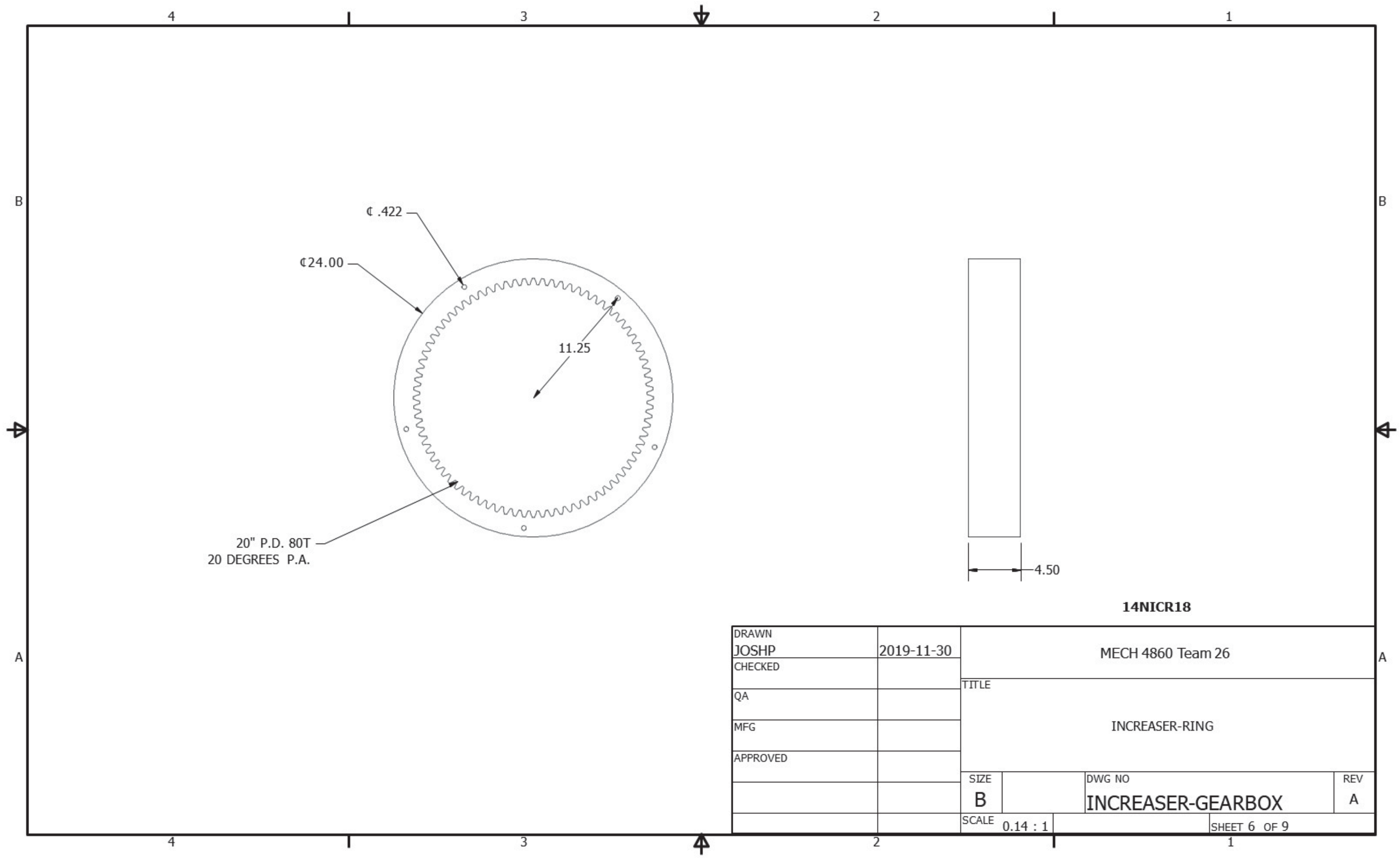
DRAWN JOSHP	2019-11-30	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF INCREASER GEARBOX
QA		
MFG		
APPROVED		SIZE B · DWG NO INCREASER-GEARBOX · REV A
		SCALE 1 / 10 · SHEET 1 OF 9

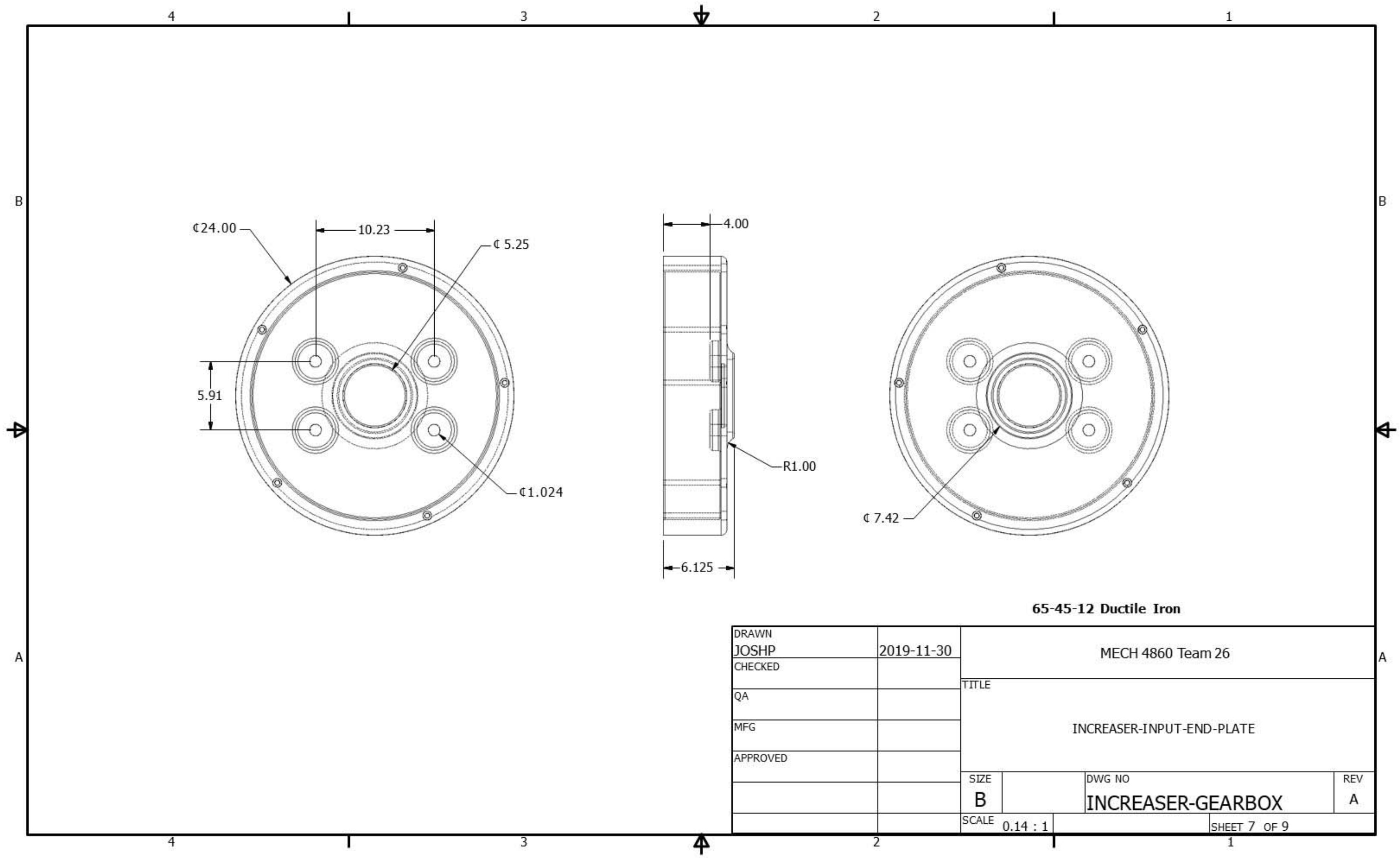

4
3
2
1
B
A
⌀ 1.38
7.50
5"-59T SPLINE
ANSI B92.1 - 37.5O
12/24 - CLASS 5
FILLET ROOT, SIDE FIT
R1.00
R.063
R1.00
4.53
⌀ 7.00
7.50
1.50
AISI 4140H Alloy Steel (UNS H41400)
DRAWN
JOSHP
2019-11-30
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
INCREASER-CARRIER
SIZE
B
DWG NO
INCREASER-GEARBOX
REV
A
SCALE
1 / 4
SHEET 2 OF 9

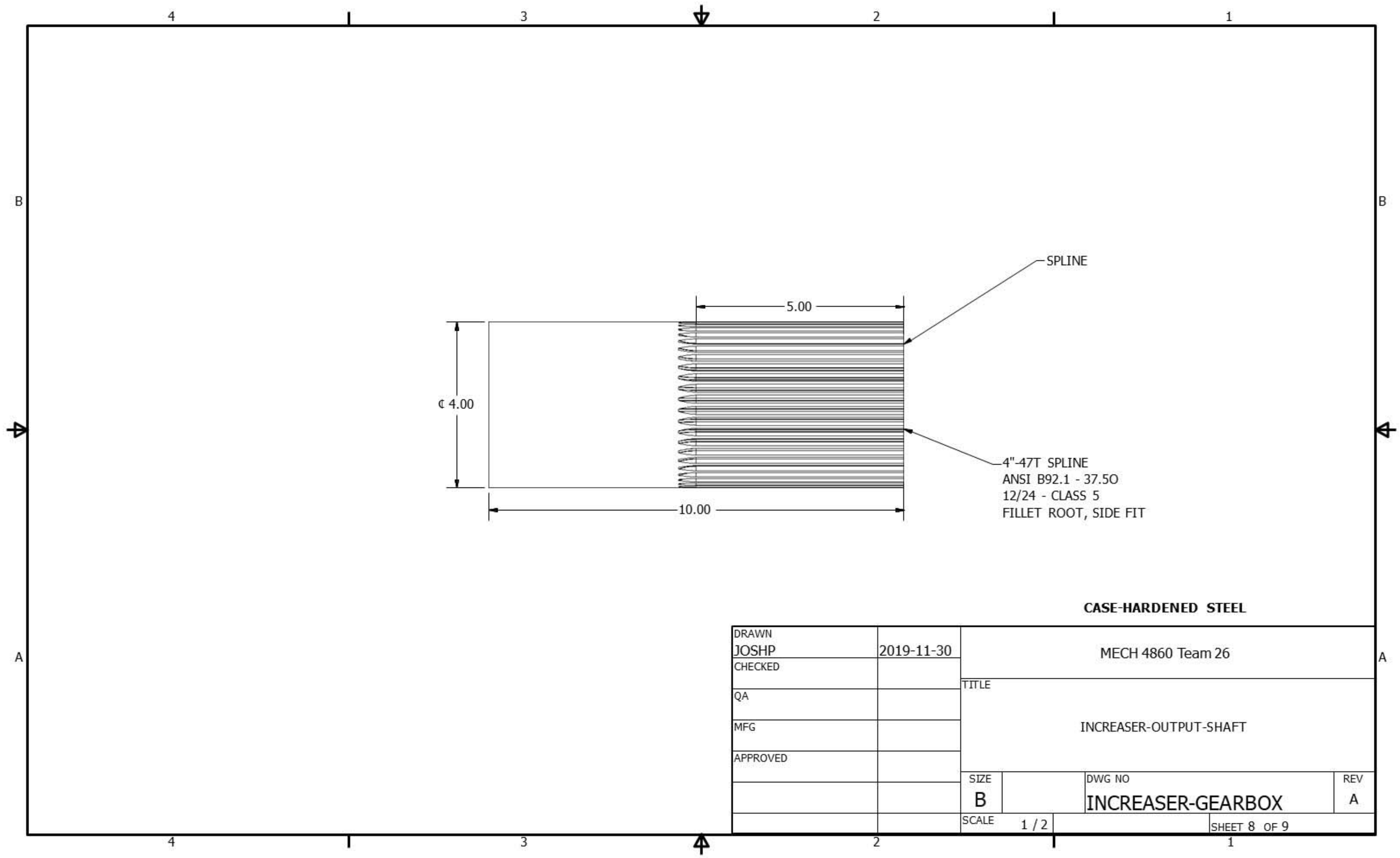

10.00
5.00
3.50
3.50
5"-59T SPLINE
ANSI B92.1 - 37.5O
12/24 - CLASS 5
FILLET ROOT, SIDE FIT
CASE-HARDENED STEEL
DRAWN
JOSHP
2019-11-30
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
INCREASER-INPUT-SHAFT
SIZE
B
DWG NO
INCREASER-GEARBOX
REV
A
SCALE
1 / 4
SHEET 3 OF 9

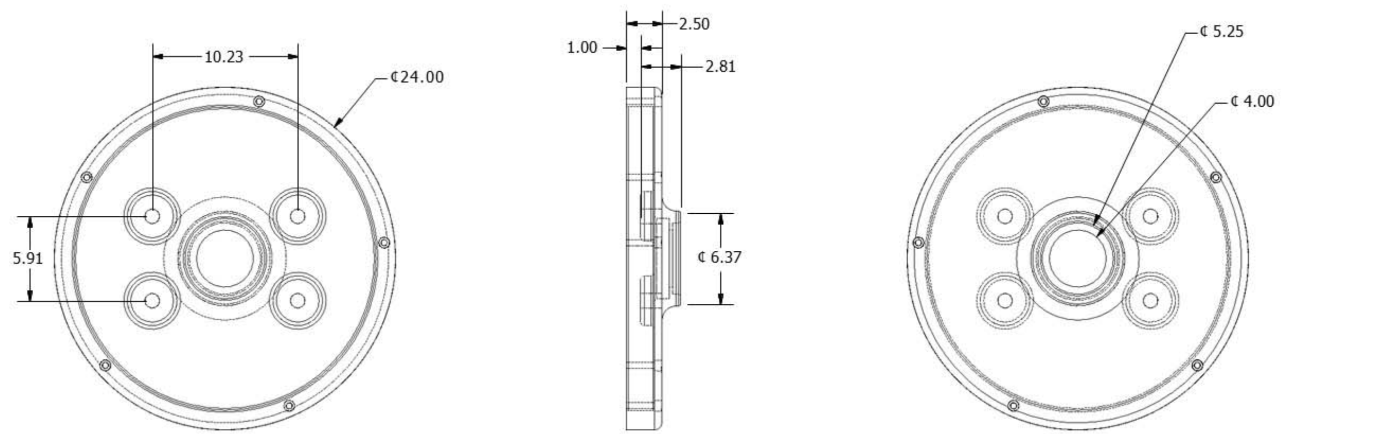

4
3
2
1
B
A
5" P.D. 20T
20 DEGREES P.A.
¢ 2.25
¢1.942
.625
.625
4.50
14NICR18
DRAWN
JOSHP
2019-11-30
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
INCREASER-PLANET
SIZE
B
DWG NO
INCREASER-GEARBOX
REV
A
SCALE
1 / 2
SHEET 4 OF 9


4
3
2
1
B
A
10" P.D. 40T
20 DEGREES P.A.
4"-47T SPLINE
ANSI B92.1 - 37.5O
12/24 - CLASS 5
FILLET ROOT, SIDE FIT
4.50
14NICR18
DRAWN
JOSHP
2019-11-30
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
INCREASER-SUN
SIZE
B
DWG NO
INCREASER-GEARBOX
REV
A
SCALE 1 / 4
SHEET 5 OF 9


¢ .422
¢24.00
11.25
20" P.D. 80T
20 DEGREES P.A.
4.50
14NICR18
DRAWN
JOSHP
2019-11-30
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
INCREASER-RING
SIZE
B
DWG NO
INCREASER-GEARBOX
REV
A
SCALE 0.14 : 1
SHEET 6 OF 9

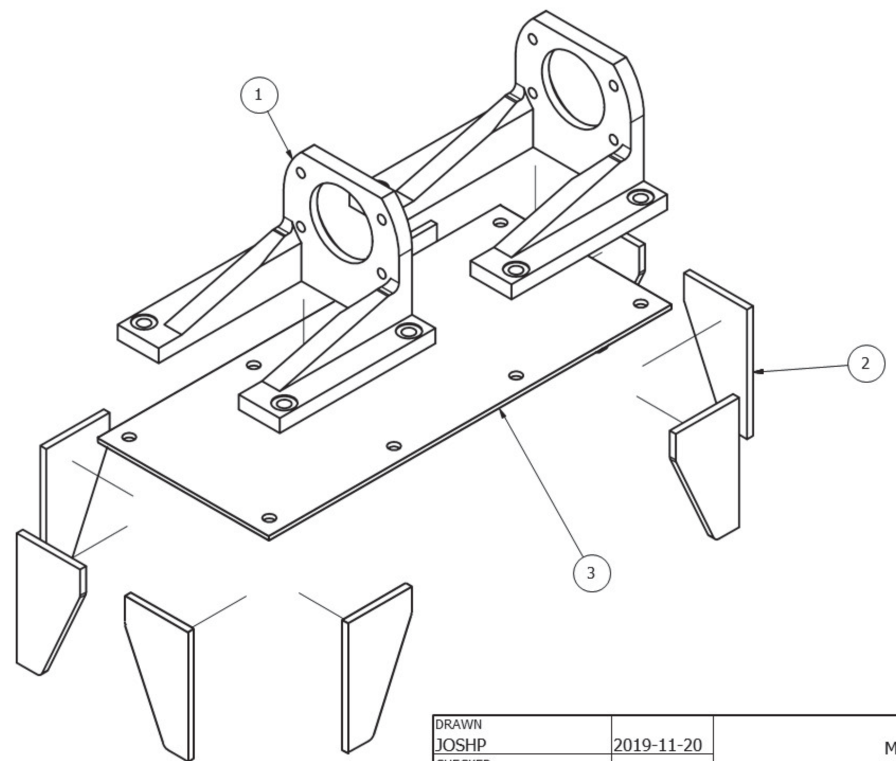

4
3
2
1
B
A
¢24.00
10.23
¢ 5.25
5.91
¢1.024
4.00
R1.00
6.125
¢ 7.42
65-45-12 Ductile Iron
DRAWN
JOSHP
2019-11-30
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
INCREASER-INPUT-END-PLATE
SIZE
B
DWG NO
INCREASER-GEARBOX
REV
A
SCALE 0.14 : 1
SHEET 7 OF 9


SPLINE
5.00
¢ 4.00
10.00
4"-47T SPLINE
ANSI B92.1 - 37.5O
12/24 - CLASS 5
FILLET ROOT, SIDE FIT
CASE-HARDENED STEEL
DRAWN
JOSHP
2019-11-30
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
INCREASER-OUTPUT-SHAFT
SIZE
B
DWG NO
INCREASER-GEARBOX
REV
A
SCALE 1 / 2
SHEET 8 OF 9



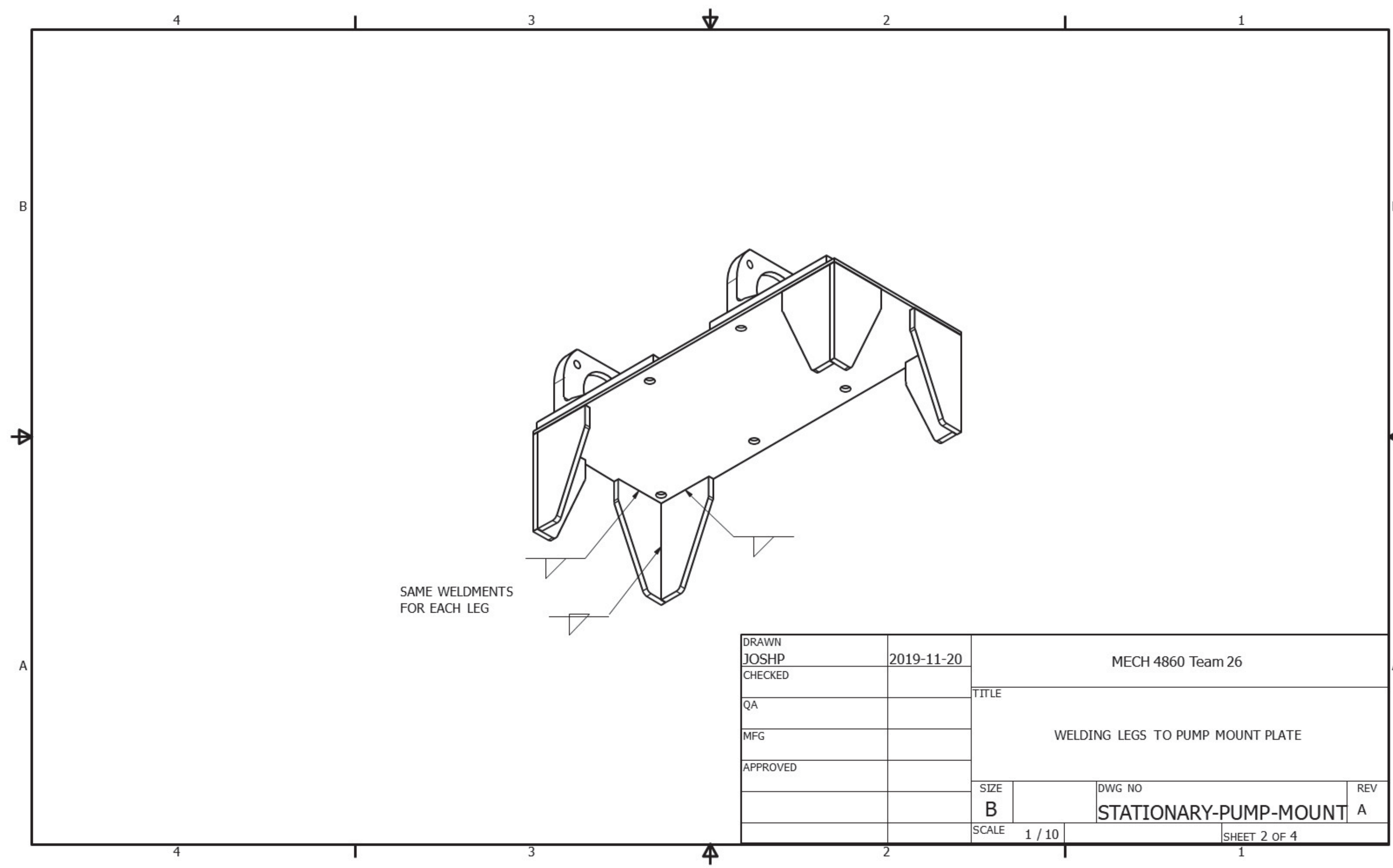
65-45-12 Ductile Iron

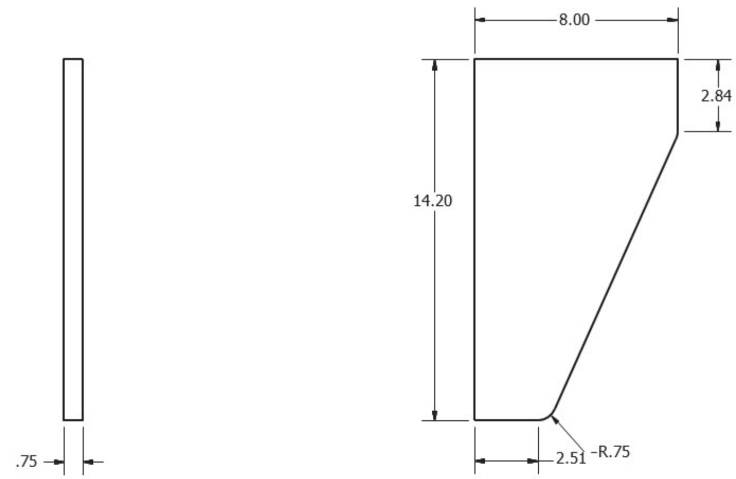
DRAWN JOSHP	2019-11-30	MECH 4860 Team 26		
CHECKED		TITLE INCREASER-OUTPUT-END-PLATE		
QA				
MFG				
APPROVED		SIZE B	DWG NO INCREASER-GEARBOX	REV A
		SCALE 0.14 : 1	SHEET 9 OF 9	



PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	2	MOUNTING BRACKET	
2	8	PUMP-MOUNT-LEG	
3	1	PUMP-MOUNT-PLATE	

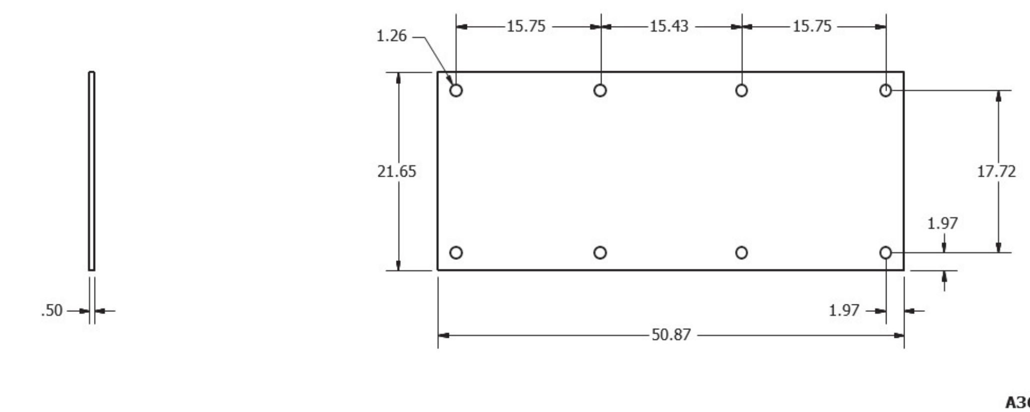
DRAWN JOSHP	2019-11-20	MECH 4860 Team 26
CHECKED		TITLE
QA		EXPLODED VIEW OF STATIONARY PUMP MOUNT
MFG		
APPROVED		SIZE B — DWG NO STATIONARY-PUMP-MOUNT — REV A
		SCALE 0.11 : 1 — SHEET 1 OF 4


SAME WELDMENTS
FOR EACH LEG
DRAWN
JOSHP
2019-11-20
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
WELDING LEGS TO PUMP MOUNT PLATE
SIZE
B
DWG NO
STATIONARY-PUMP-MOUNT
REV
A
SCALE 1 / 10
SHEET 2 OF 4

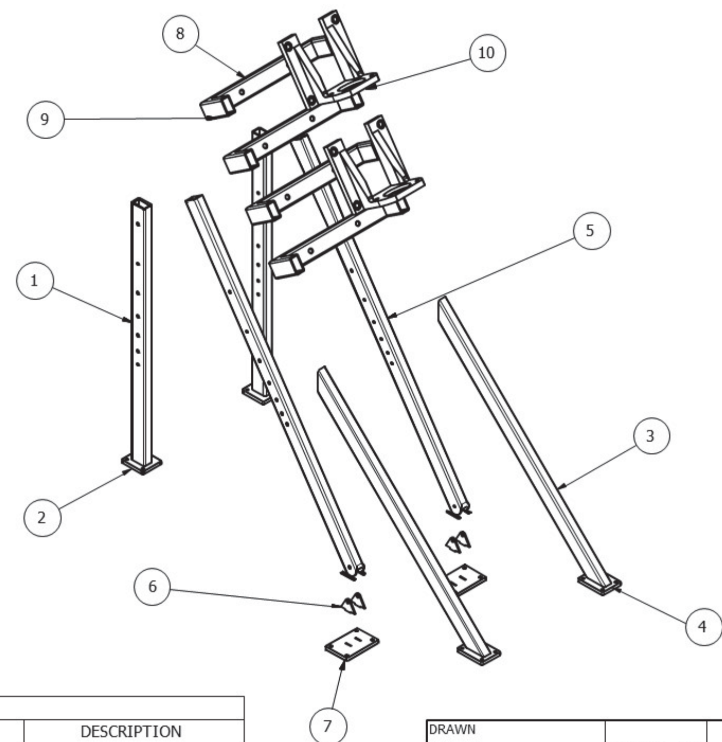


A36 STEEL

DRAWN JOSHP	2019-11-20	MECH 4860 Team 26			
CHECKED		TITLE PUMP-MOUNT-LEG			
QA					
MFG					
APPROVED		SIZE B		DWG NO STATIONARY-PUMP-MOUNT	REV A
		SCALE 1 / 4		SHEET 3 OF 4	



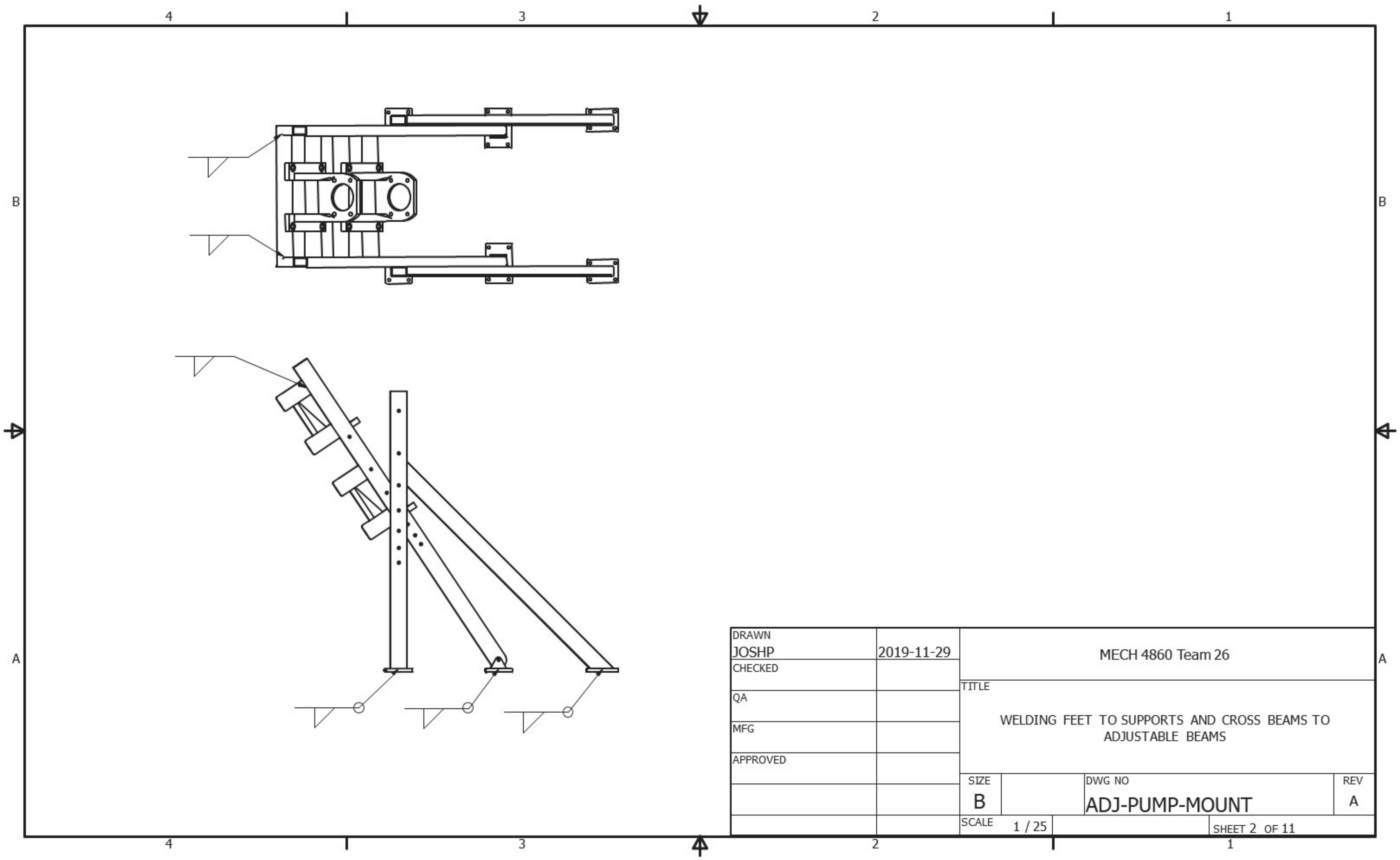
DRAWN JOSHP	2019-11-20	MECH 4860 Team 26			
CHECKED		TITLE PUMP-MOUNT-PLATE			
QA					
MFG					
APPROVED		SIZE B		DWG NO STATIONARY-PUMP-MOUNT	REV A
		SCALE 0.09 : 1		SHEET 4 OF 4	

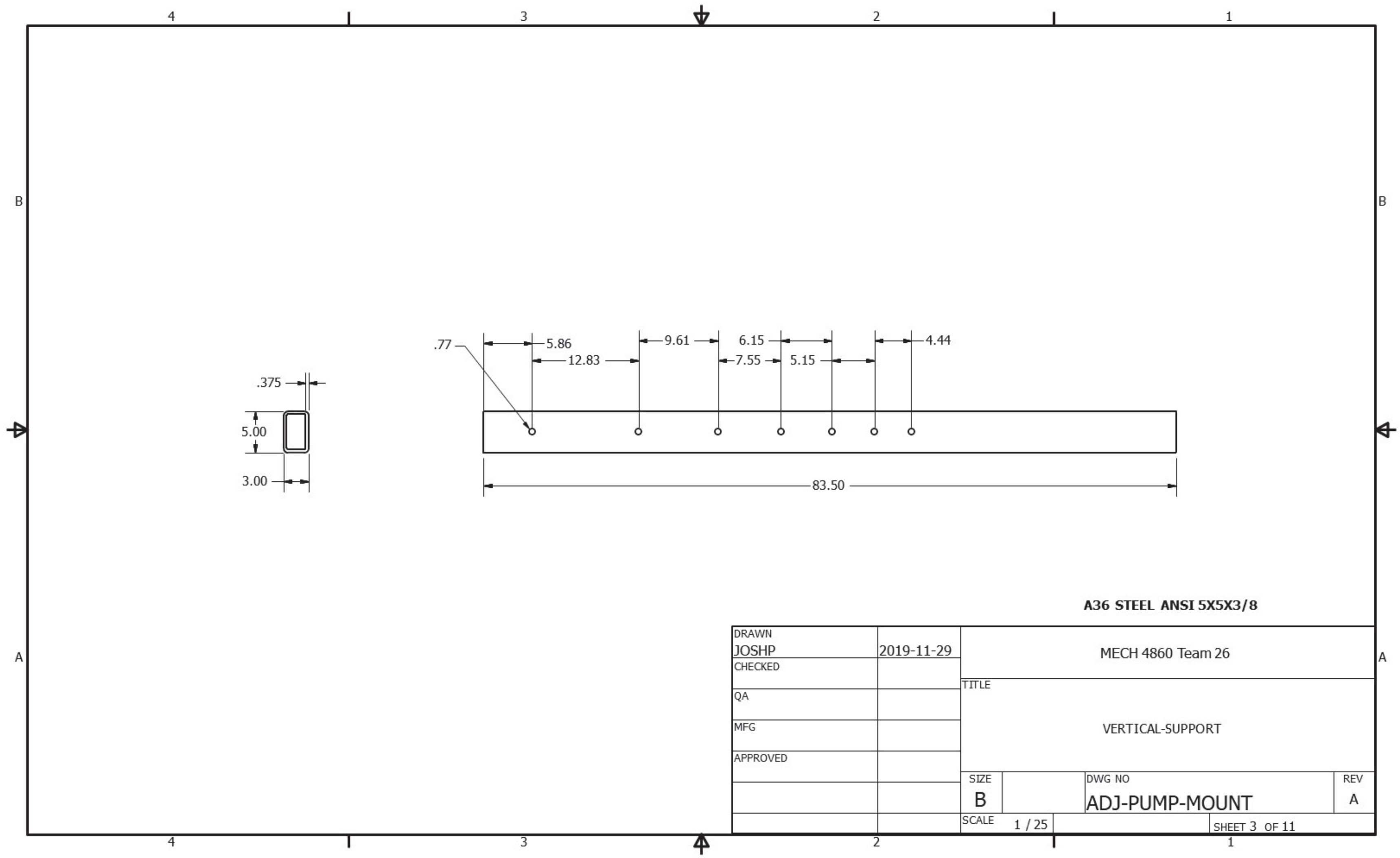


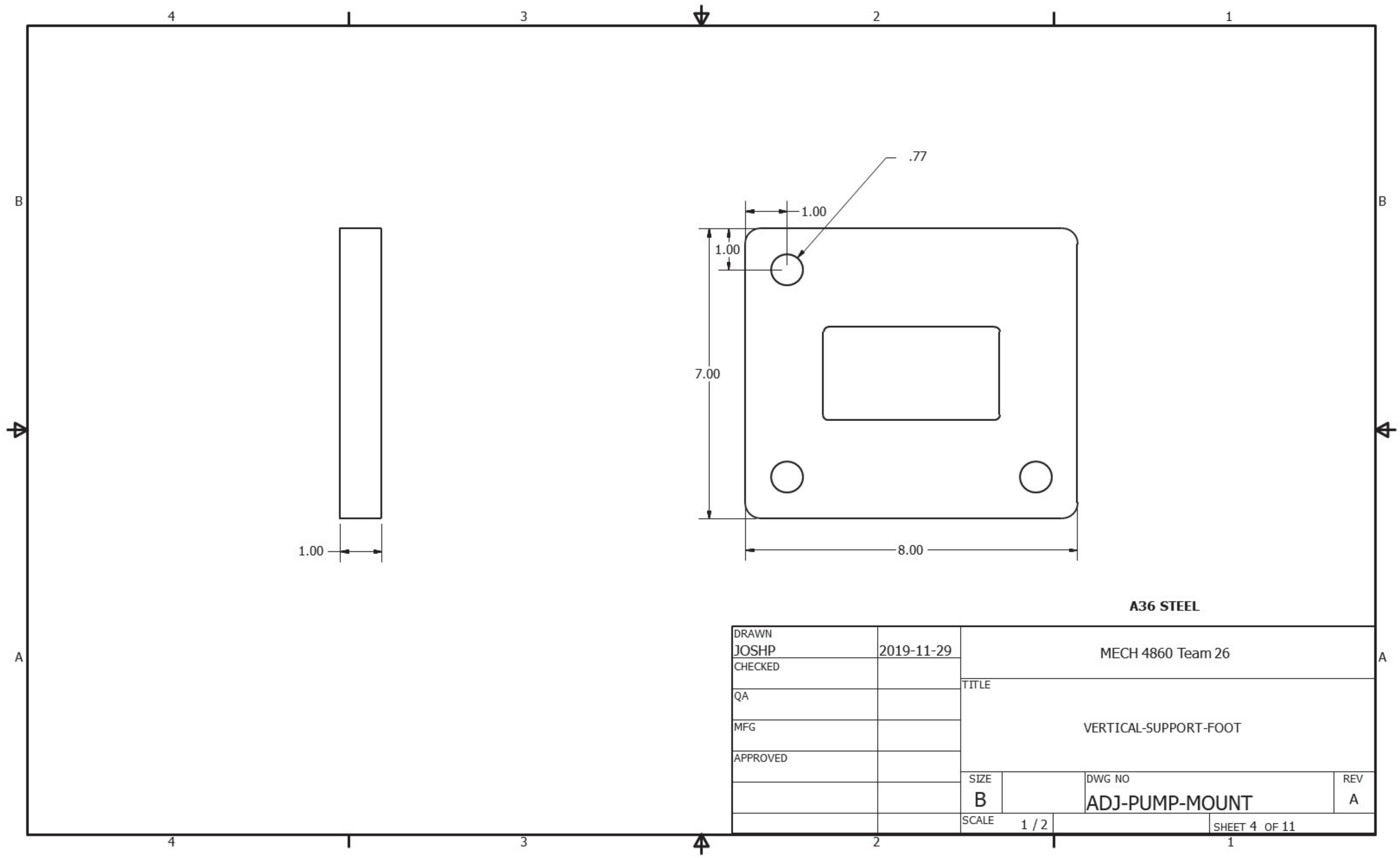
PARTS LIST

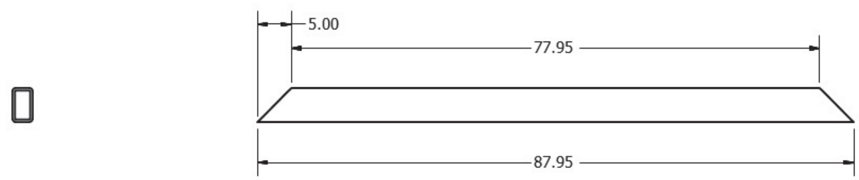
ITEM	QTY	PART NUMBER	DESCRIPTION
1	2	VERTICAL-SUPPORT	
2	2	VERTICAL-SUPPORT-FOOT	
3	2	DIAGONAL-SUPPORT	
4	2	DIAGONAL-SUPPORT-FOOT	
5	2	ADJ-BEAM	
6	4	ADJ-BRKT	
7	2	ADJ-FOOT	
8	4	CROSS-BEAM-HORZ	
9	8	CROSS-BEAM-VERT	
10	2	PUMP-BRACKET	

DRAWN JOSHP	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF ADJUSTABLE PUMP MOUNT ASSEMBLY
QA		
MFG		
APPROVED		SIZE B · DWG NO ADJ-PUMP-MOUNT · REV A
		SCALE 1 / 27 · SHEET 1 OF 11


4
3
2
1
B
A
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
WELDING FEET TO SUPPORTS AND CROSS BEAMS TO ADJUSTABLE BEAMS
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE 1 / 25
SHEET 2 OF 11

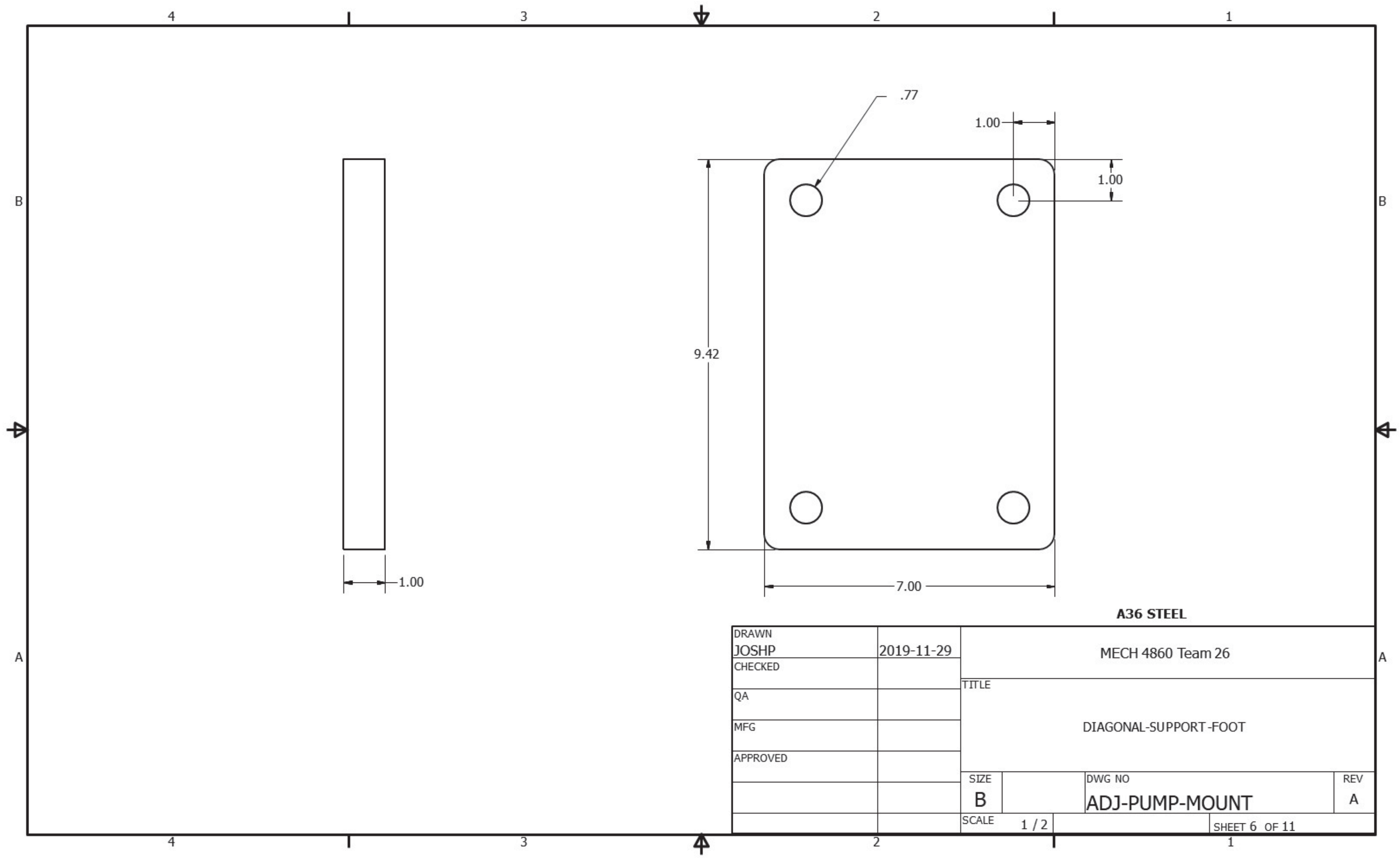

.375
5.00
3.00
.77
5.86
12.83
9.61
6.15
7.55
5.15
4.44
83.50
A36 STEEL ANSI 5X5X3/8
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
VERTICAL-SUPPORT
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE 1 / 25
SHEET 3 OF 11

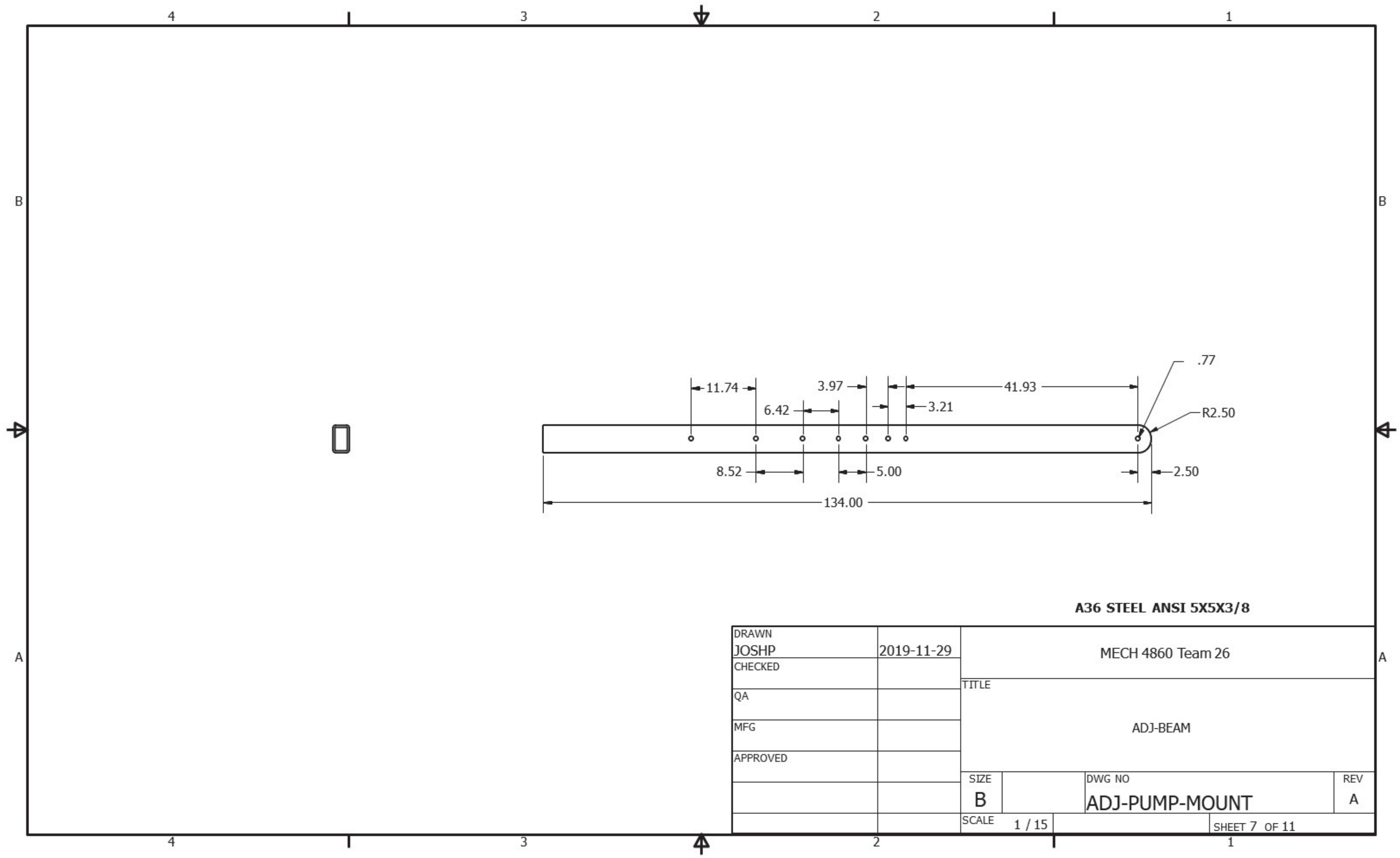

.77
1.00
1.00
7.00
8.00
1.00
A36 STEEL
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
VERTICAL-SUPPORT-FOOT
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE
1 / 2
SHEET 4 OF 11

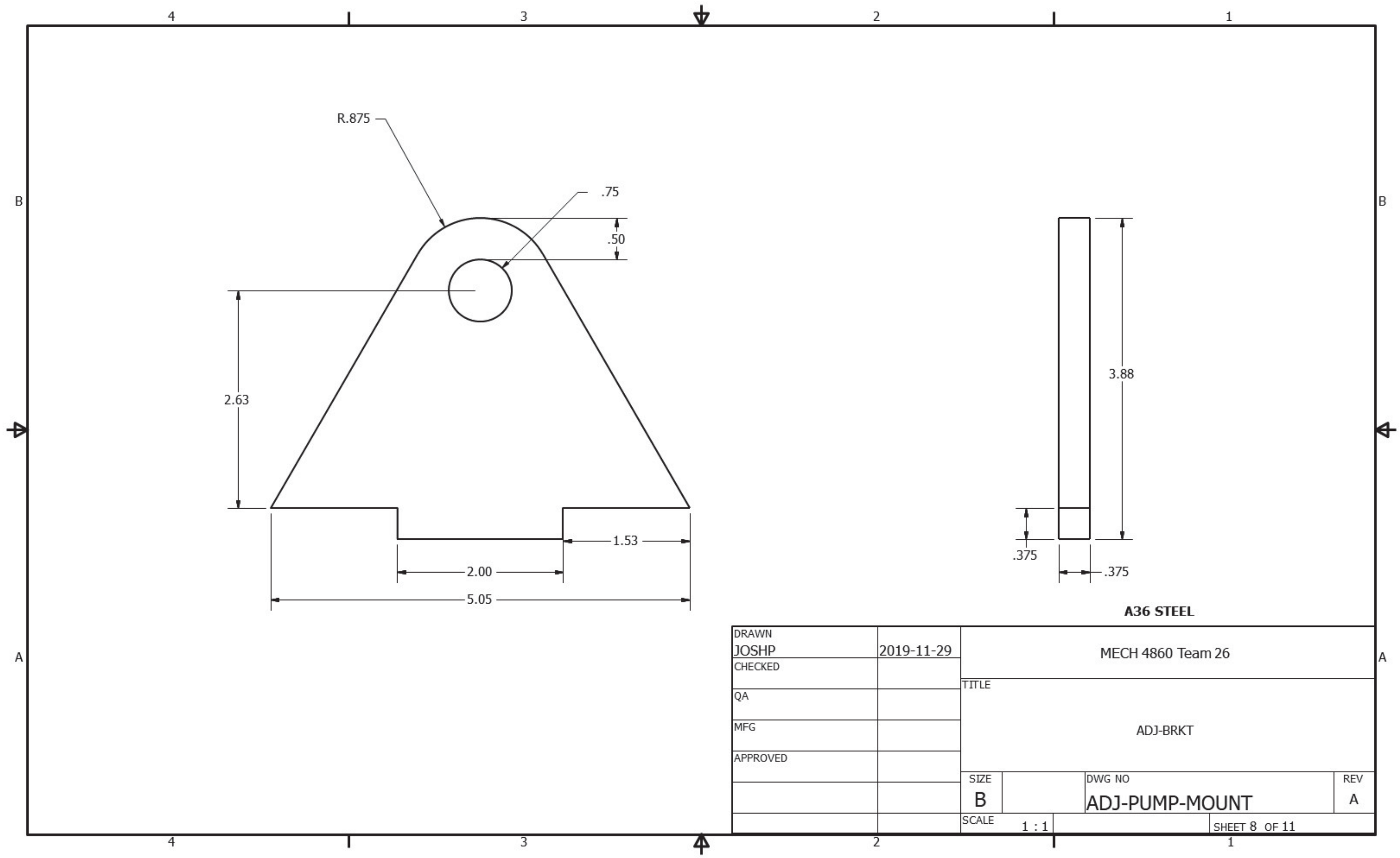


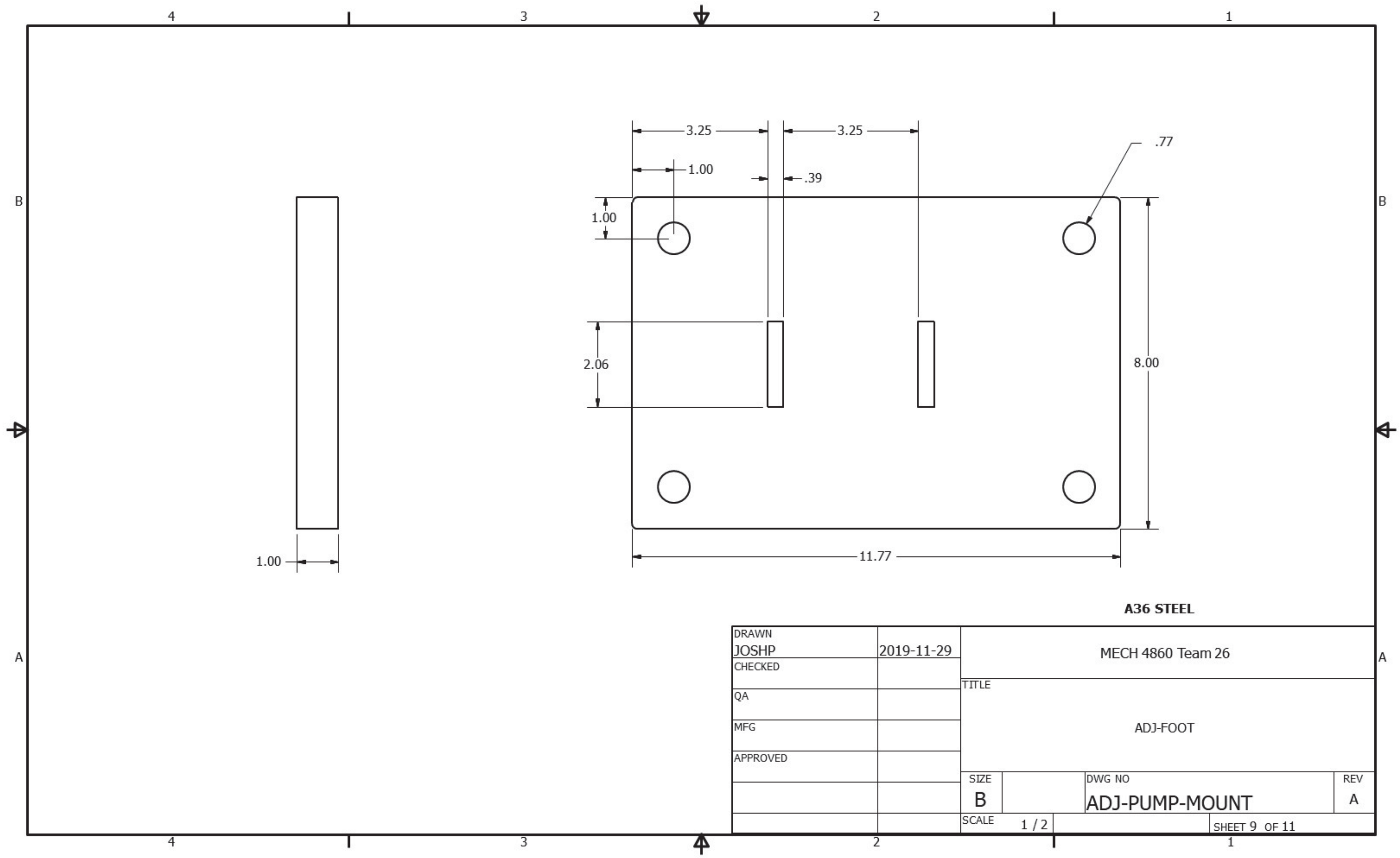
A36 STEEL ANSI 5X5X3/8

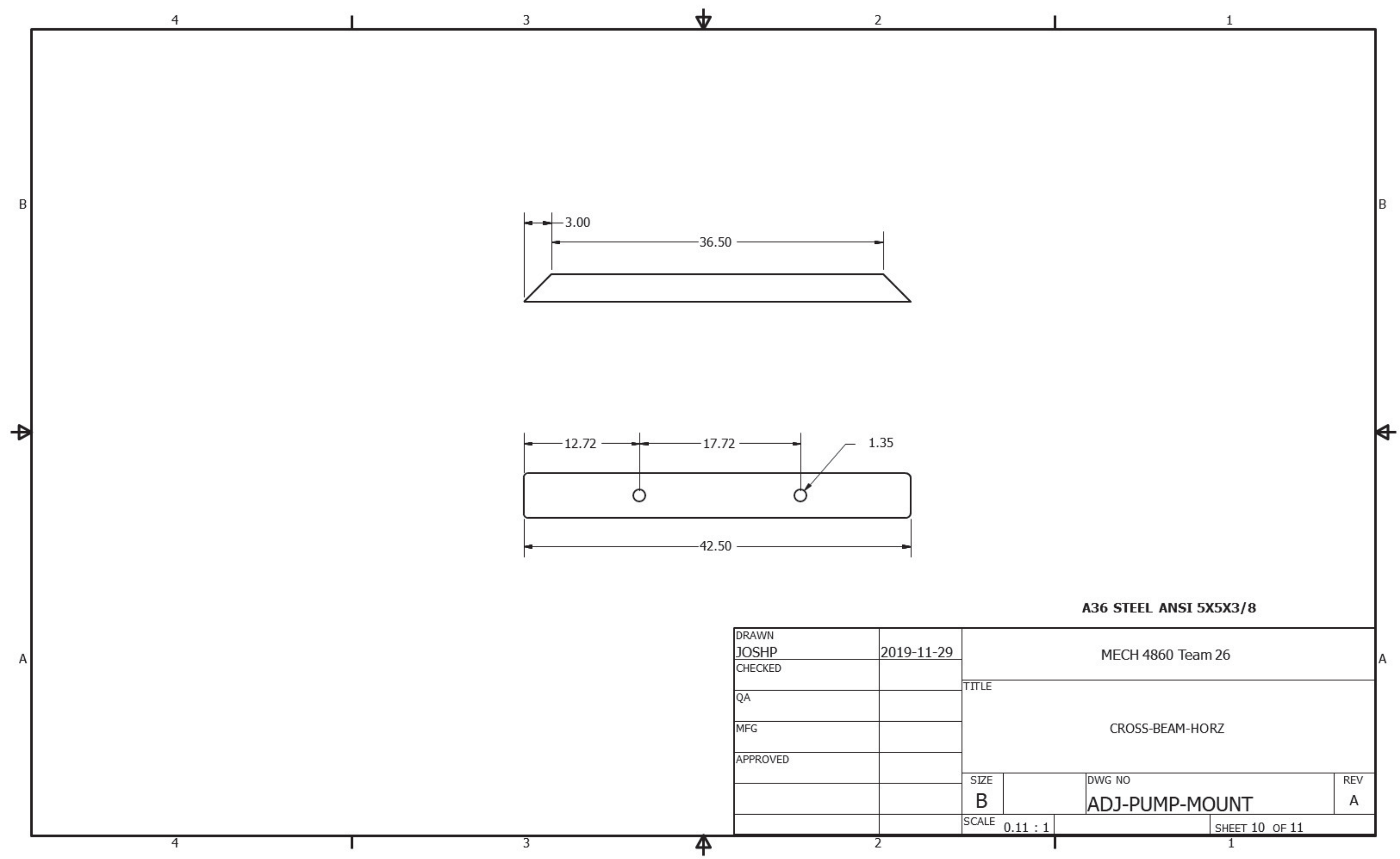
DRAWN JOSHP	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE DIAGONAL-SUPPORT
QA		
MFG		
APPROVED		SIZE B / DWG NO ADJ-PUMP-MOUNT / REV A
		SCALE 1 / 15 / SHEET 5 OF 11

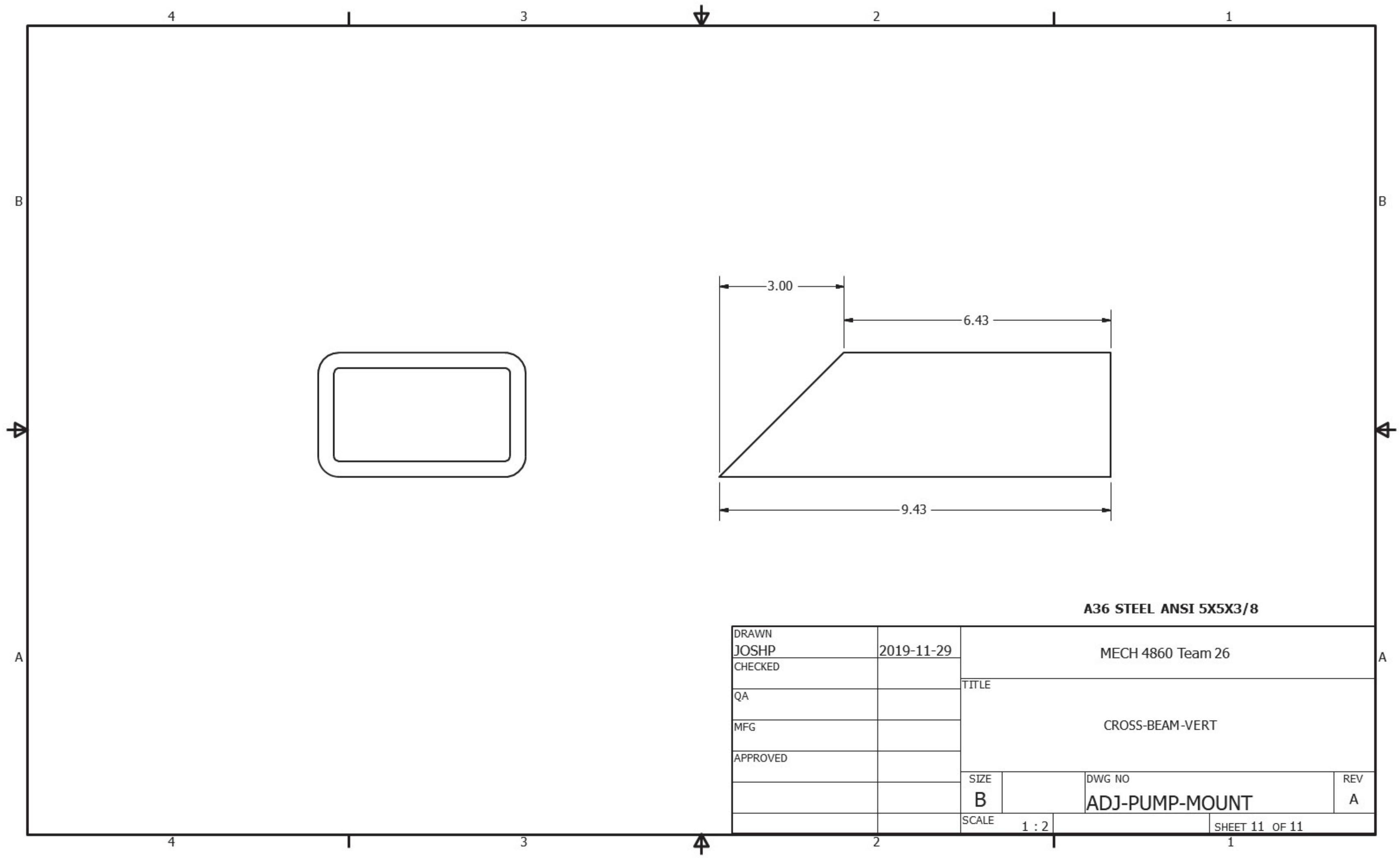

.77
1.00
1.00
9.42
7.00
1.00
A36 STEEL
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
DIAGONAL-SUPPORT-FOOT
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE
1 / 2
SHEET 6 OF 11

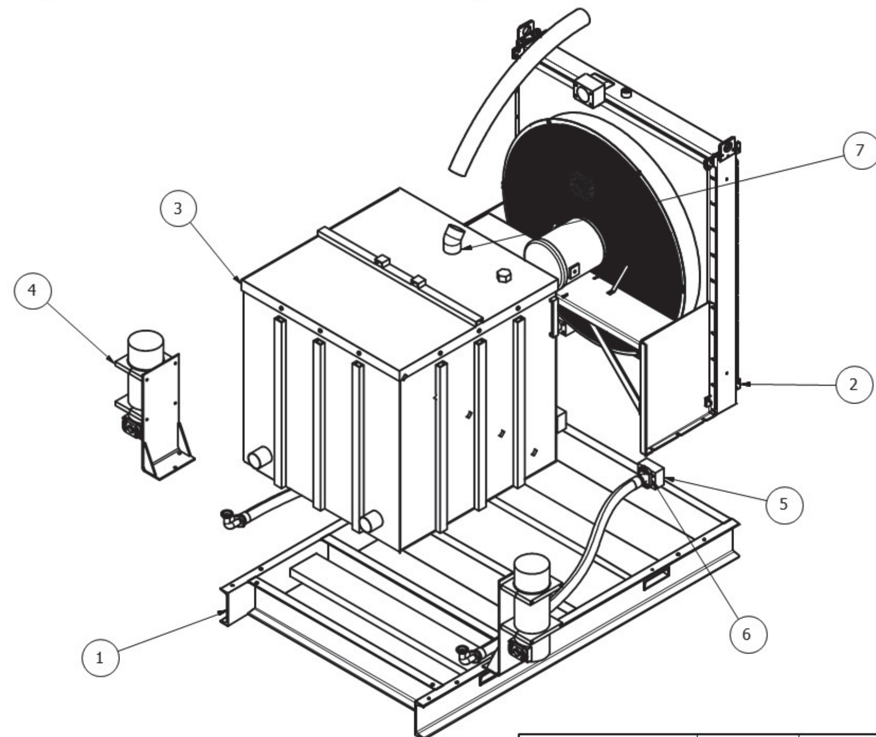

11.74
6.42
3.97
3.21
41.93
.77
R2.50
8.52
5.00
2.50
134.00
A36 STEEL ANSI 5X5X3/8
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ADJ-BEAM
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE 1 / 15
SHEET 7 OF 11


4
3
2
1
B
A
R.875
.75
.50
2.63
1.53
2.00
5.05
3.88
.375
.375
A36 STEEL
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ADJ-BRKT
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE
1 : 1
SHEET 8 OF 11


4
3
2
1
B
A
3.25
3.25
1.00
.39
.77
1.00
2.06
8.00
1.00
11.77
A36 STEEL
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
ADJ-FOOT
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE
1 / 2
SHEET 9 OF 11


3.00
36.50
12.72
17.72
1.35
42.50
A36 STEEL ANSI 5X5X3/8
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
CROSS-BEAM-HORZ
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE 0.11 : 1
SHEET 10 OF 11

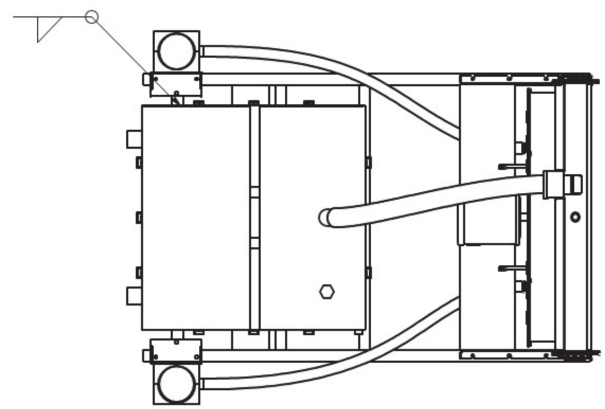

4
3
2
1
B
A
3.00
6.43
9.43
A36 STEEL ANSI 5X5X3/8
DRAWN
JOSHP
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
CROSS-BEAM-VERT
SIZE
B
DWG NO
ADJ-PUMP-MOUNT
REV
A
SCALE
1 : 2
SHEET 11 OF 11



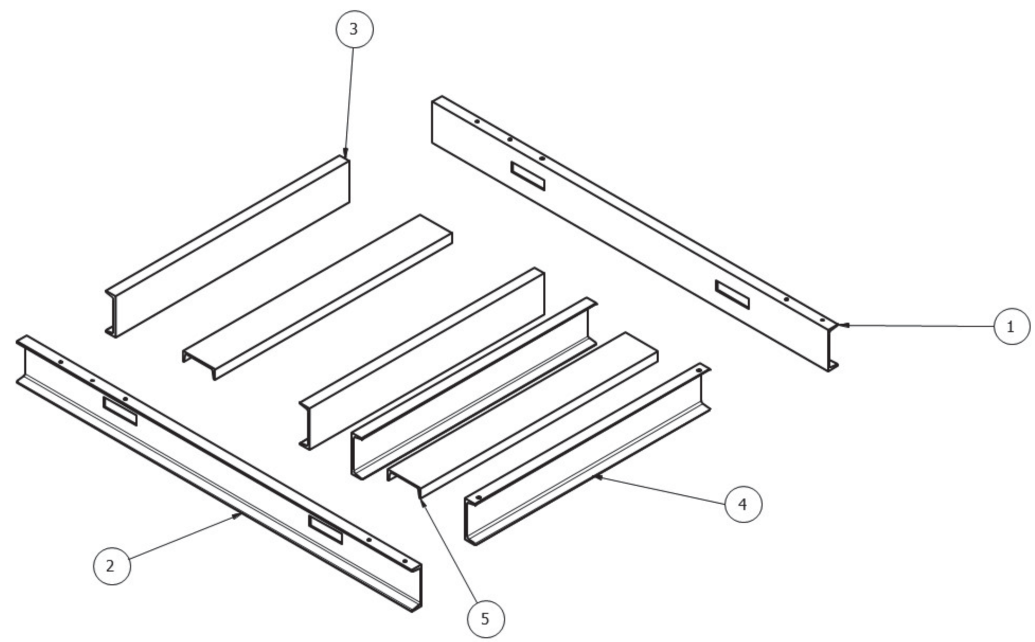
I TEM	QTY	PART NUMBER	DESCRIPTION
1	1	OIL-SKID	
2	1	Cooler	
3	1	OIL-TANK	
4	2	FILTER-ASSEMBLY	
5	2	3in SAE to 2in Adapter	
6	4	2in Code 61 Split Flange	
7	2	4in Code 61 Split Flange	

PARTS LIST

DRAWN JOSHP	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE
QA		EXPLODED VIEW OF OIL ASSEMBLY
MFG		
APPROVED		
		SIZE B — DWG NO OIL-ASSEMBLY — REV A
		SCALE 1 / 25 — SHEET 1 OF 2

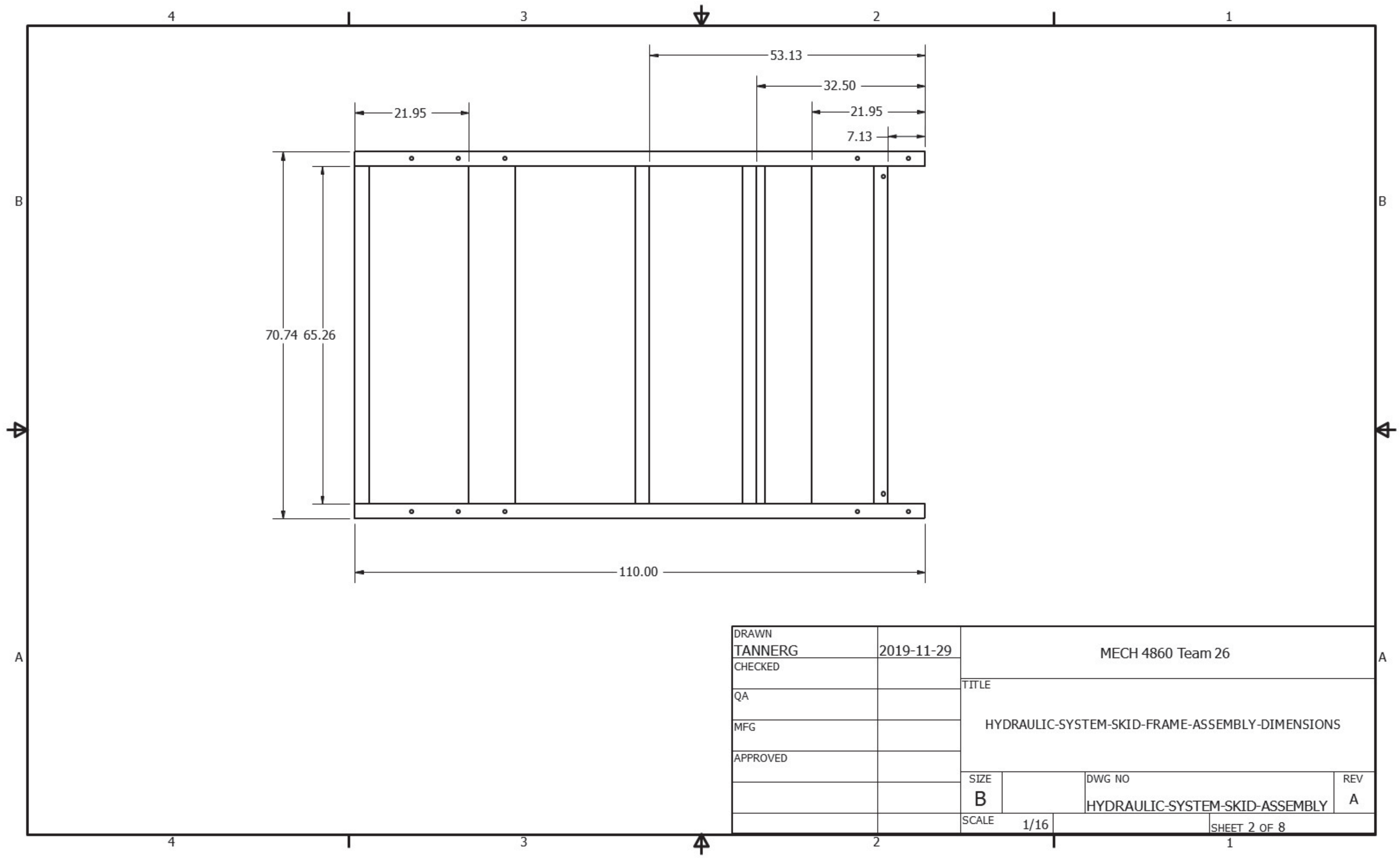


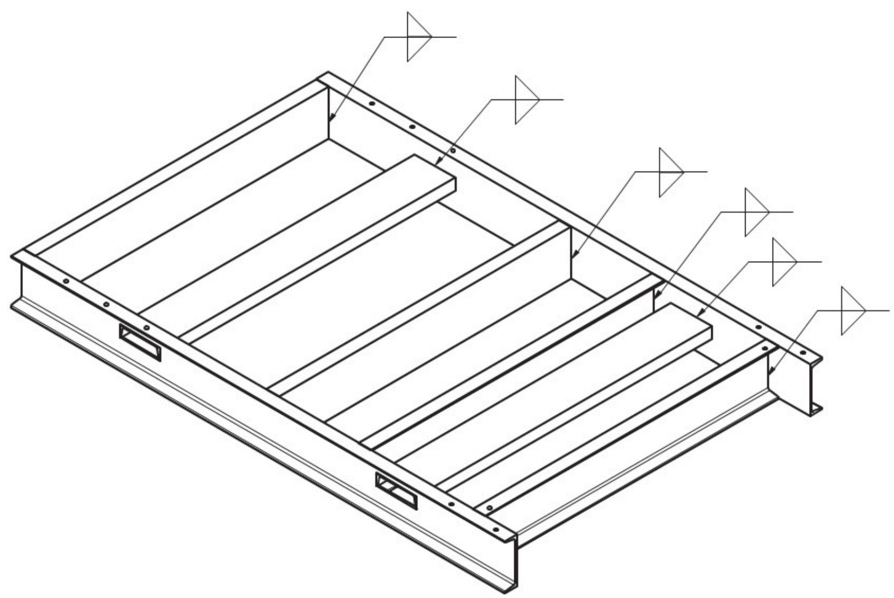
DRAWN JOSHP	2019-11-29	MECH 4860 Team 26			
CHECKED		TITLE WELDING OIL TANK TO SKID			
QA					
MFG					
APPROVED		SIZE B	DWG NO OIL-ASSEMBLY		REV A
		SCALE 1 / 25		SHEET 2 OF 2	



PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	LEFT-MAIN-RAIL	
2	1	RIGHT-MAIN-RAIL	
3	3	CROSS-MEMBER	
4	1	FRONT-CROSS-MEMBER	
5	2	FORK-CHANNELS	

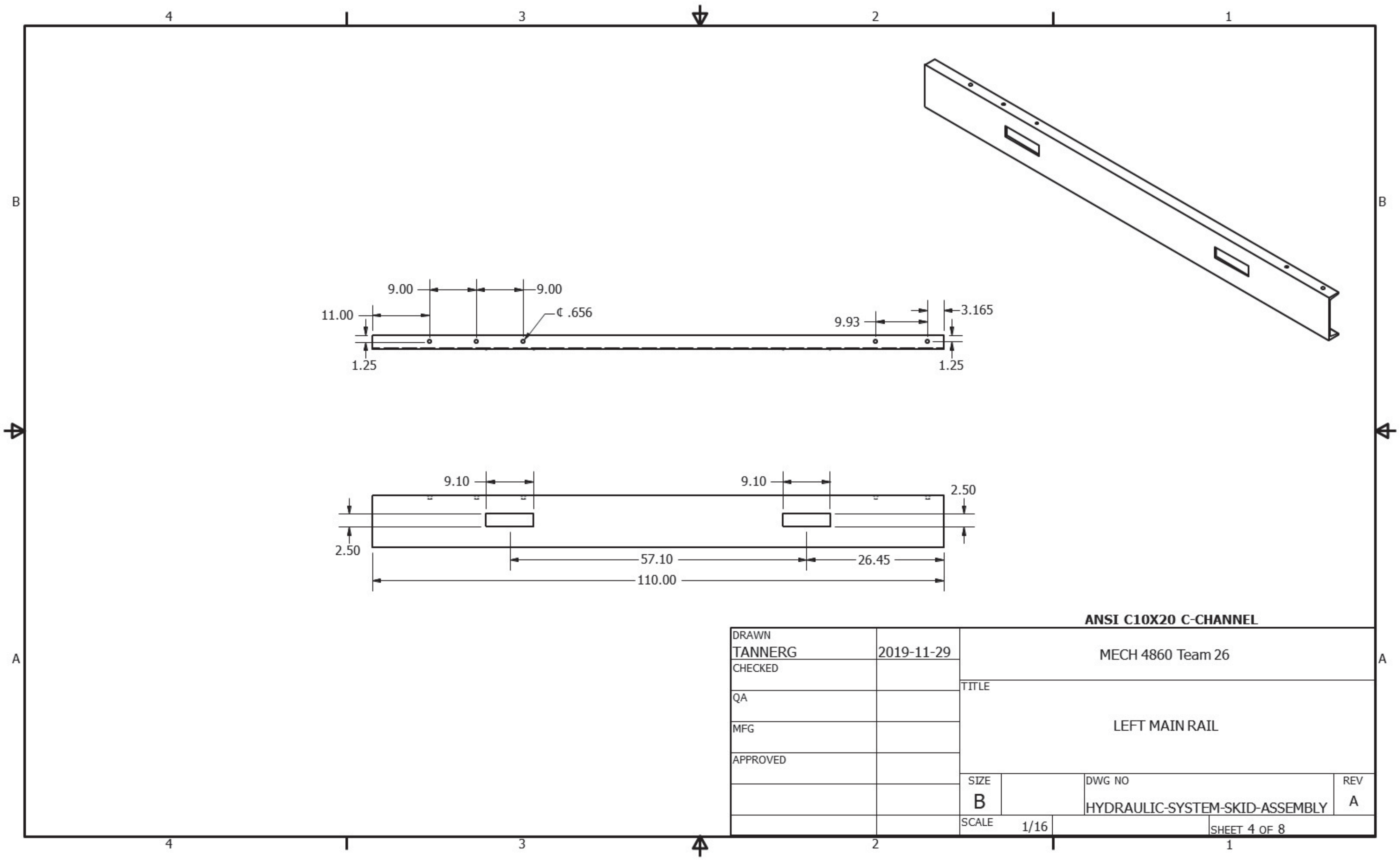
DRAWN TANNERG	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF HYDRAULIC SYSTEM SKID
QA		
MFG		
APPROVED		SIZE B; DWG NO HYDRAULIC-SYSTEM-SKID-ASSEMBLY; REV A
		SCALE 1 / 20; SHEET 1 OF 8

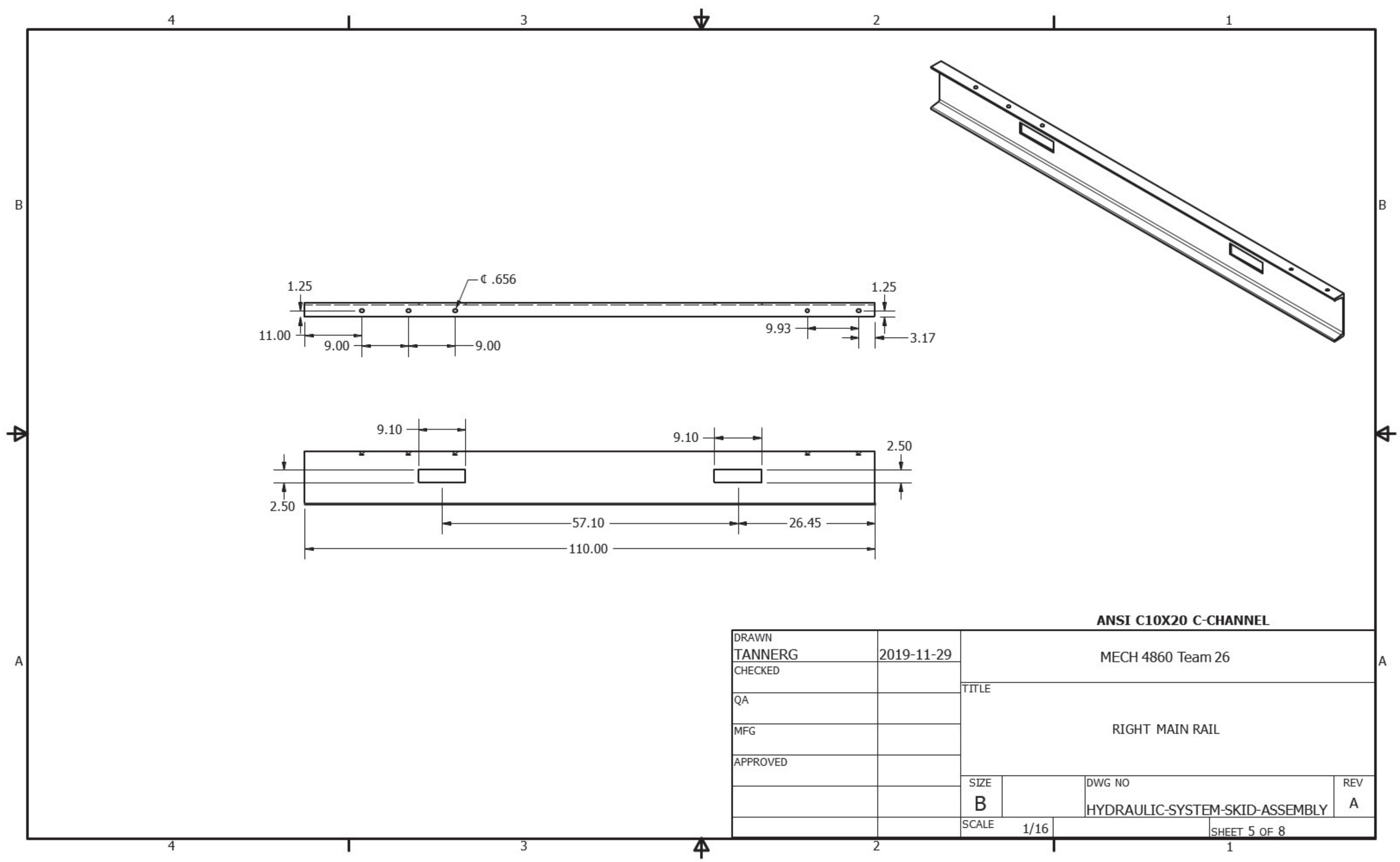

53.13
32.50
21.95
7.13
21.95
70.74
65.26
110.00
DRAWN
TANNERG
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
HYDRAULIC-SYSTEM-SKID-FRAME-ASSEMBLY-DIMENSIONS
SIZE
B
DWG NO
HYDRAULIC-SYSTEM-SKID-ASSEMBLY
REV
A
SCALE
1/16
SHEET 2 OF 8



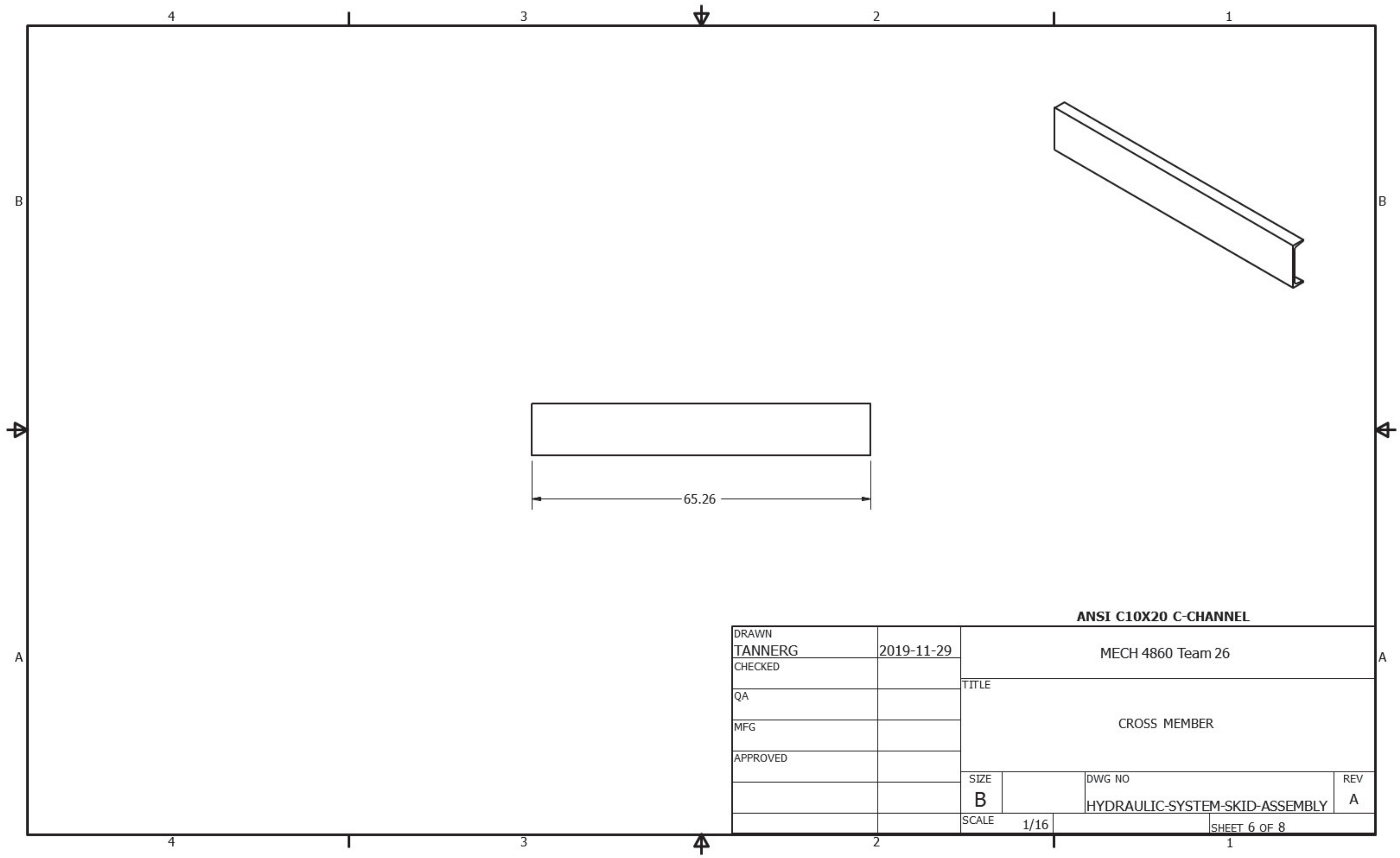
NOTE: REPEAT WELDS SHOWN ON LEFT MAIN RAIL ON THE RIGHT MAIN RAIL

DRAWN TANNERG	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE HYDRAULIC SYSTEM SKID WELDMENTS
QA		
MFG		
APPROVED		SIZE B
		DWG NO HYDRAULIC-SYSTEM-SKID-ASSEMBLY
		REV A
		SCALE 1/16
		SHEET 3 OF 8


9.00
9.00
11.00
¢ .656
9.93
3.165
1.25
1.25
9.10
9.10
2.50
2.50
57.10
26.45
110.00
ANSI C10X20 C-CHANNEL
DRAWN
TANNERG
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
LEFT MAIN RAIL
SIZE
B
DWG NO
HYDRAULIC-SYSTEM-SKID-ASSEMBLY
REV
A
SCALE
1/16
SHEET 4 OF 8



4
3
2
1
B
A
1.25
⊄ .656
1.25
11.00
9.00
9.00
9.93
3.17
9.10
9.10
2.50
2.50
57.10
26.45
110.00
ANSI C10X20 C-CHANNEL
DRAWN
TANNERG
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
RIGHT MAIN RAIL
SIZE
B
DWG NO
HYDRAULIC-SYSTEM-SKID-ASSEMBLY
REV
A
SCALE
1/16
SHEET 5 OF 8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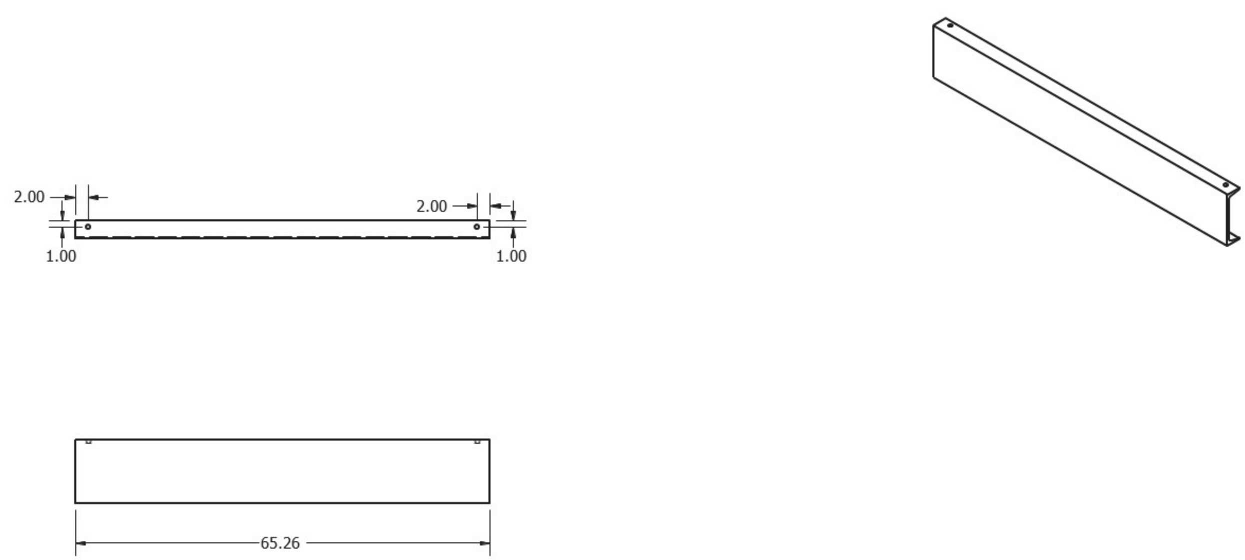
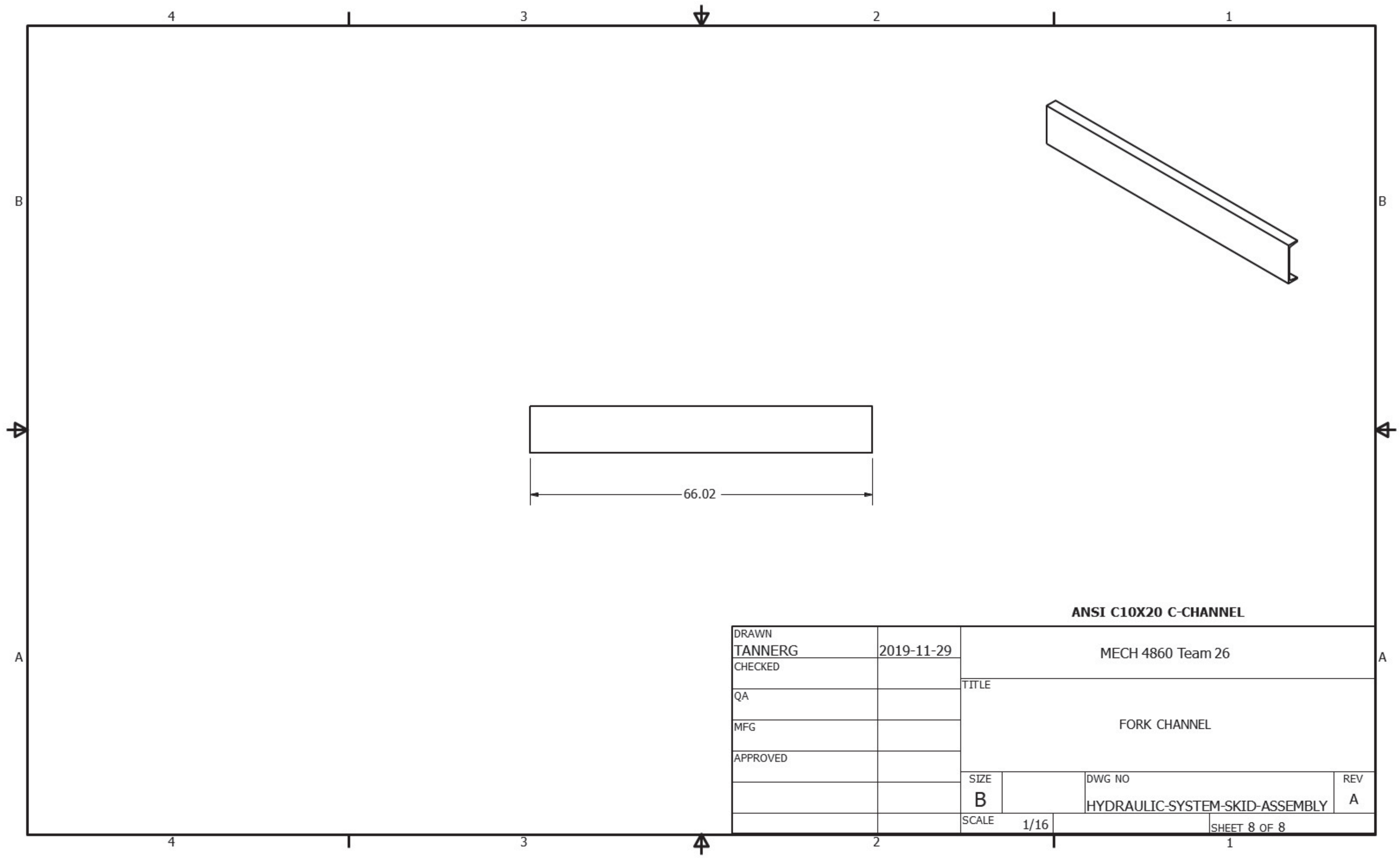
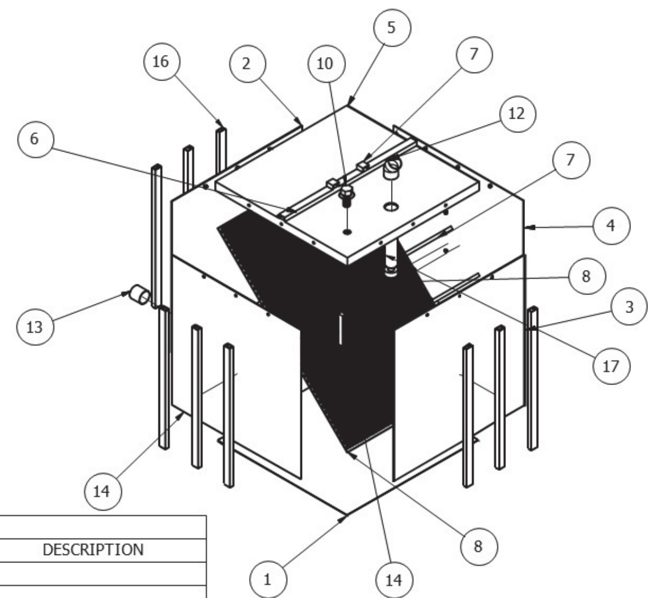
DRAWN
TANNERG
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
CROSS MEMBER
SIZE
B
DWG NO
HYDRAULIC-SYSTEM-SKID-ASSEMBLY
REV
A
SCALE
1/16
SHEET 6 OF 8



ANSI C10X20 C-CHANNEL

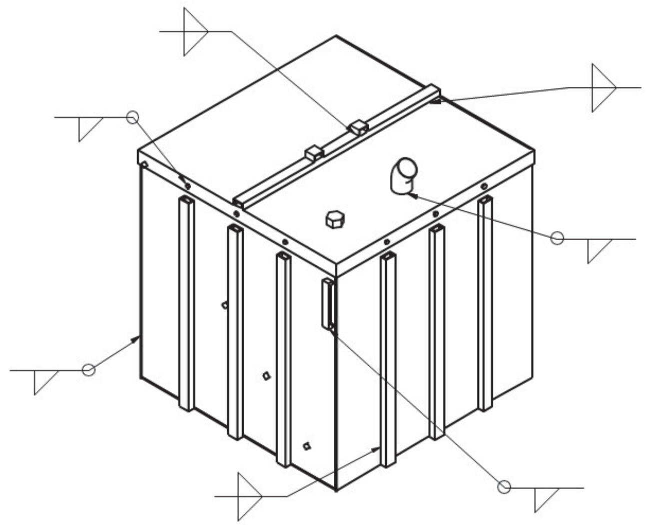
DRAWN TANNERG	2019-11-29	MECH 4860 Team 26			
CHECKED		TITLE FRONT CROSS MEMBER			
QA					
MFG					
APPROVED		SIZE B		DWG NO HYDRAULIC-SYSTEM-SKID-ASSEMBLY	REV A
		SCALE 1/16		SHEET 7 OF 8	

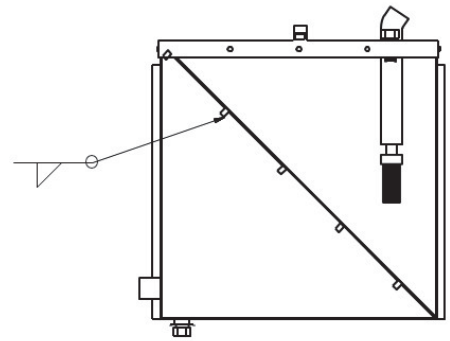

4
3
2
1
B
A
66.02
ANSI C10X20 C-CHANNEL
DRAWN
TANNERG
2019-11-29
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
FORK CHANNEL
SIZE
B
DWG NO
HYDRAULIC-SYSTEM-SKID-ASSEMBLY
REV
A
SCALE
1/16
SHEET 8 OF 8



PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	BOTTOM-PLATE	
2	1	FRONT-PLATE	
3	1	BACK-PLATE	
4	2	SIDE-PLATE	
5	1	TOP-PLATE	
6	1	LIFTING-BRACE	
7	2	LIFTING-HOLES	
8	2	BAFFLE-PLATE	
9	5	BAFFLE-BRACE	
10	1	Metal Filler Breather	
11	1	Drain Cap	
12	1	Return Elbow	
13	2	Outlet Fitting	
14	1	Sightglass	
15	1	Diffuser	
16	12	TANK-BRACE	
17	1	RETURN-EXTENSION	
18	1	Threaded Drain	

DRAWN JOSHP	2019-11-28	MECH 4860 Team 26	
CHECKED		TITLE EXPLODED VIEW OF OIL TANK ASSEMBLY	
QA			
MFG			
APPROVED			
		SIZE B	DWG NO OIL-TANK	REV A
		SCALE 1 / 30	SHEET 1 OF 12

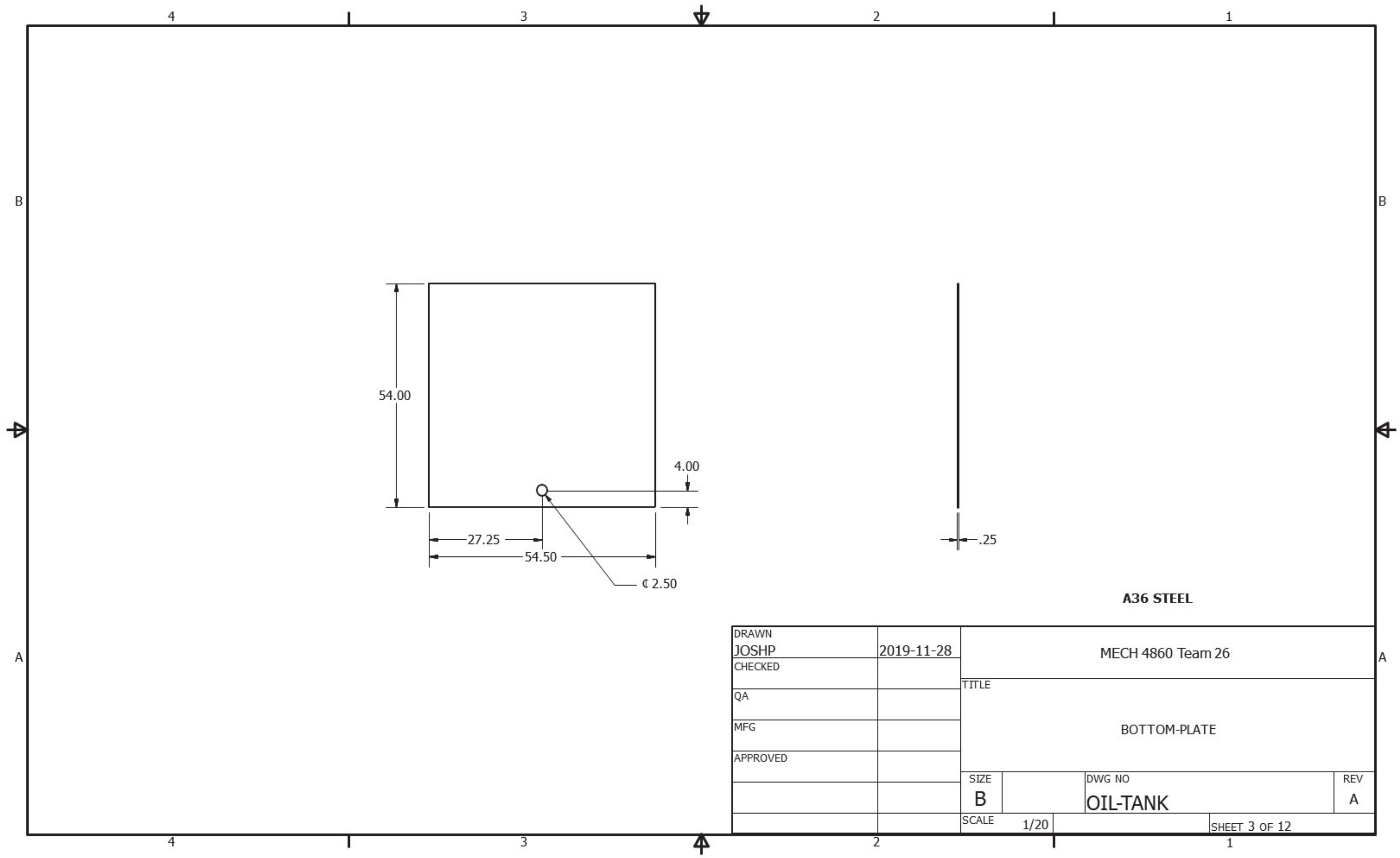


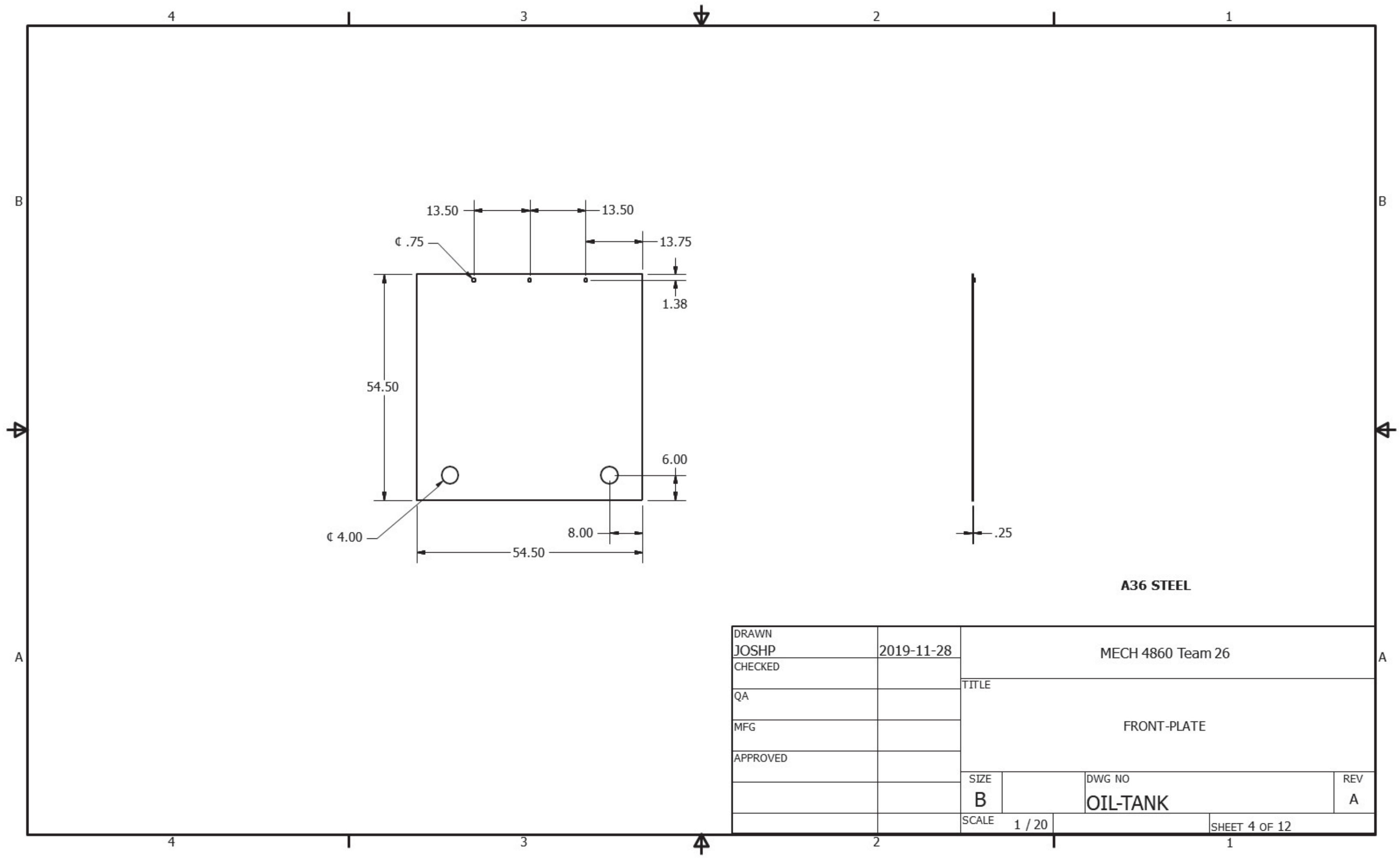


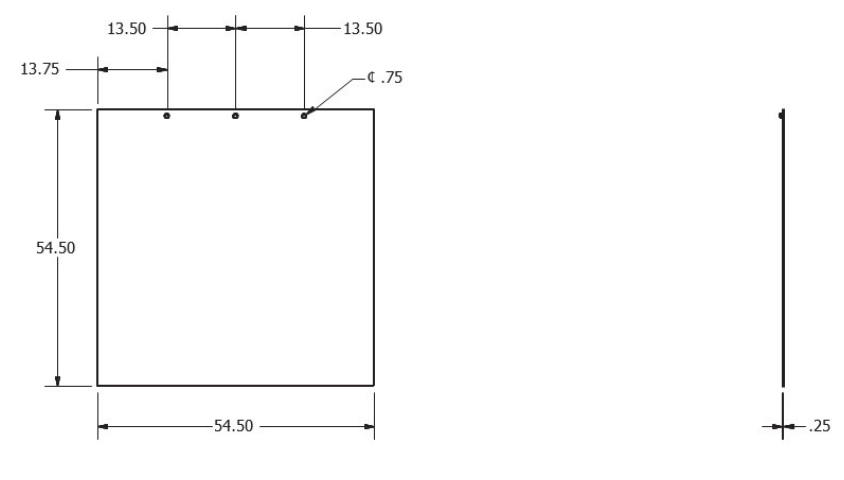
SIDE VIEW WITH PANEL AND BACING HIDDEN

TOP PLATE IS DESIGNED TO BE ATTACHED USING NUTS, ALL OTHER PLATES ARE DESIGNED TO BE WELDED TOGETHER

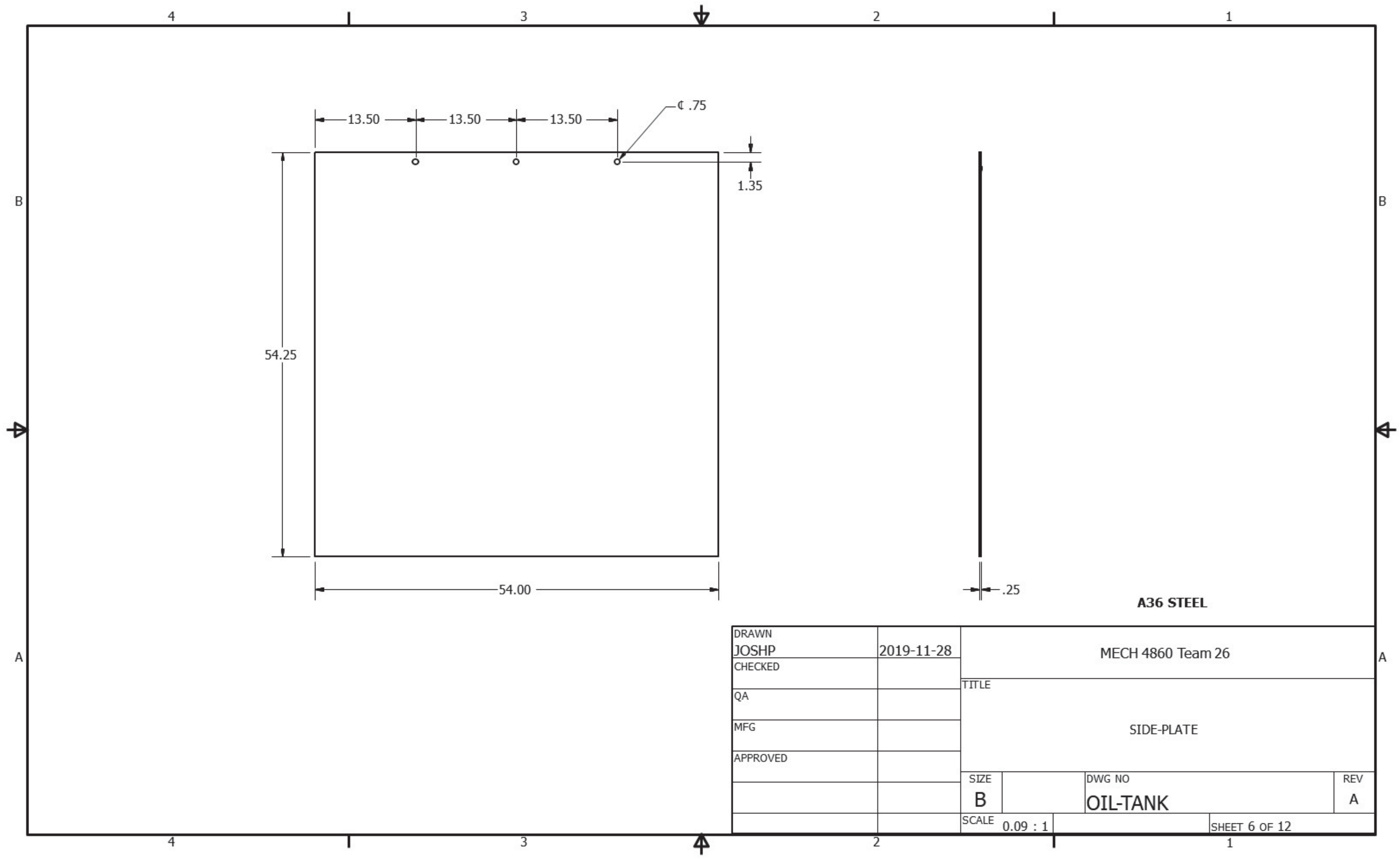
DRAWN JOSHP	2019-11-28	MECH 4860 Team 26	
CHECKED		TITLE WELDING PLATES, BRACES, FITTINGS, AND SIGHTGLASS	
QA			
MFG			
APPROVED			
		SIZE B	DWG NO OIL-TANK	REV A
		SCALE 1/20	SHEET 2 OF 12

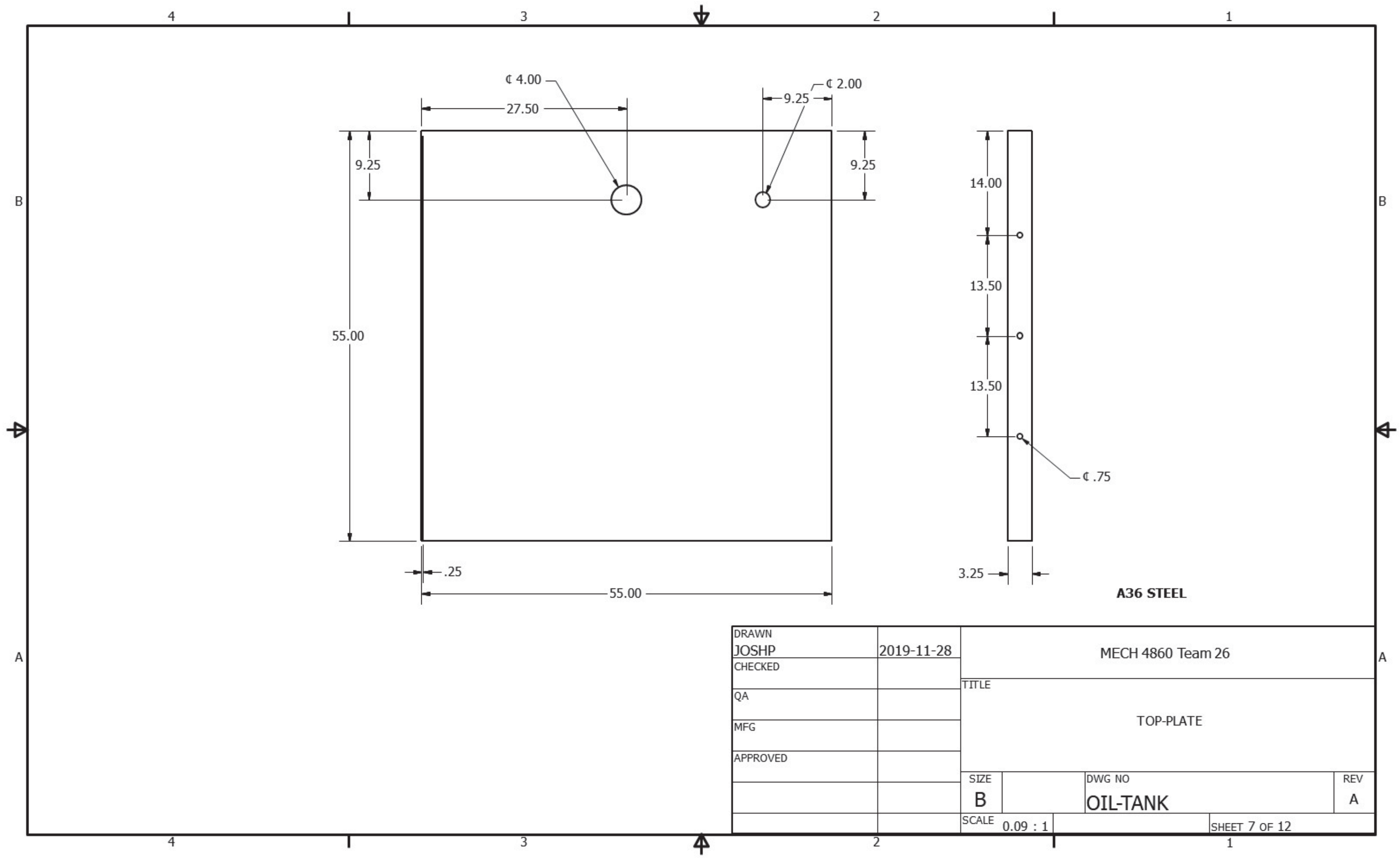

4
3
2
1
B
A
54.00
4.00
27.25
54.50
¢ 2.50
.25
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
BOTTOM-PLATE
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE
1/20
SHEET 3 OF 12


4
3
2
1
B
A
13.50
13.50
¢ .75
13.75
1.38
54.50
6.00
¢ 4.00
8.00
54.50
.25
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
FRONT-PLATE
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE 1 / 20
SHEET 4 OF 12

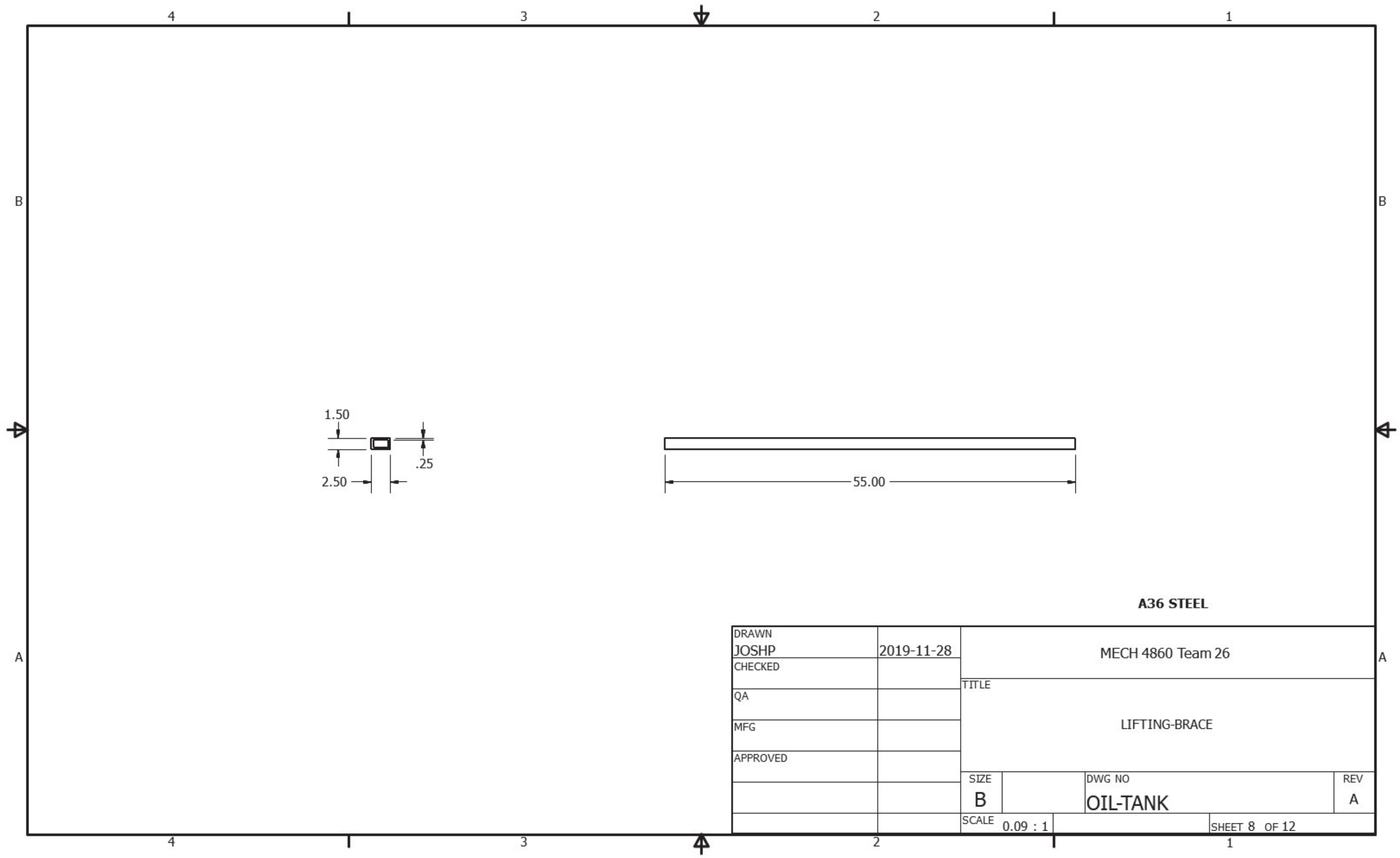


DRAWN JOSHP	2019-11-28	MECH 4860 Team 26	
CHECKED		TITLE BACK-PLATE	
QA			
MFG			
APPROVED		SIZE B	DWG NO OIL-TANK	REV A
		SCALE 1 / 20	SHEET 5 OF 12

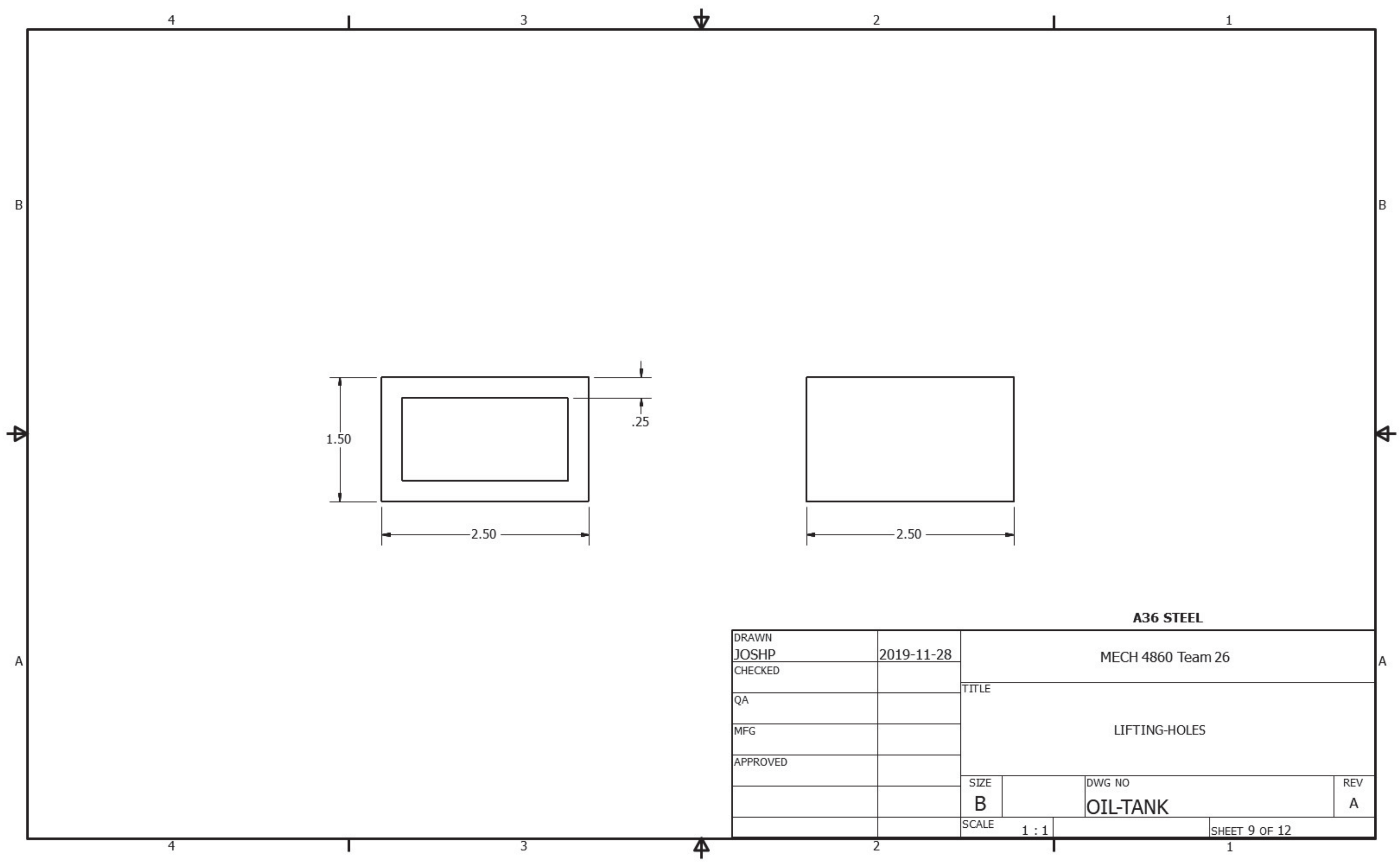

13.50
13.50
13.50
⊄ .75
1.35
54.25
54.00
.25
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
SIDE-PLATE
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE 0.09 : 1
SHEET 6 OF 12



¢ 4.00
27.50
¢ 2.00
9.25
9.25
9.25
55.00
.25
55.00
14.00
13.50
13.50
¢ .75
3.25
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
TOP-PLATE
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE 0.09 : 1
SHEET 7 OF 12



1.50
.25
2.50
55.00
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
LIFTING-BRACE
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE 0.09 : 1
SHEET 8 OF 12


4
3
2
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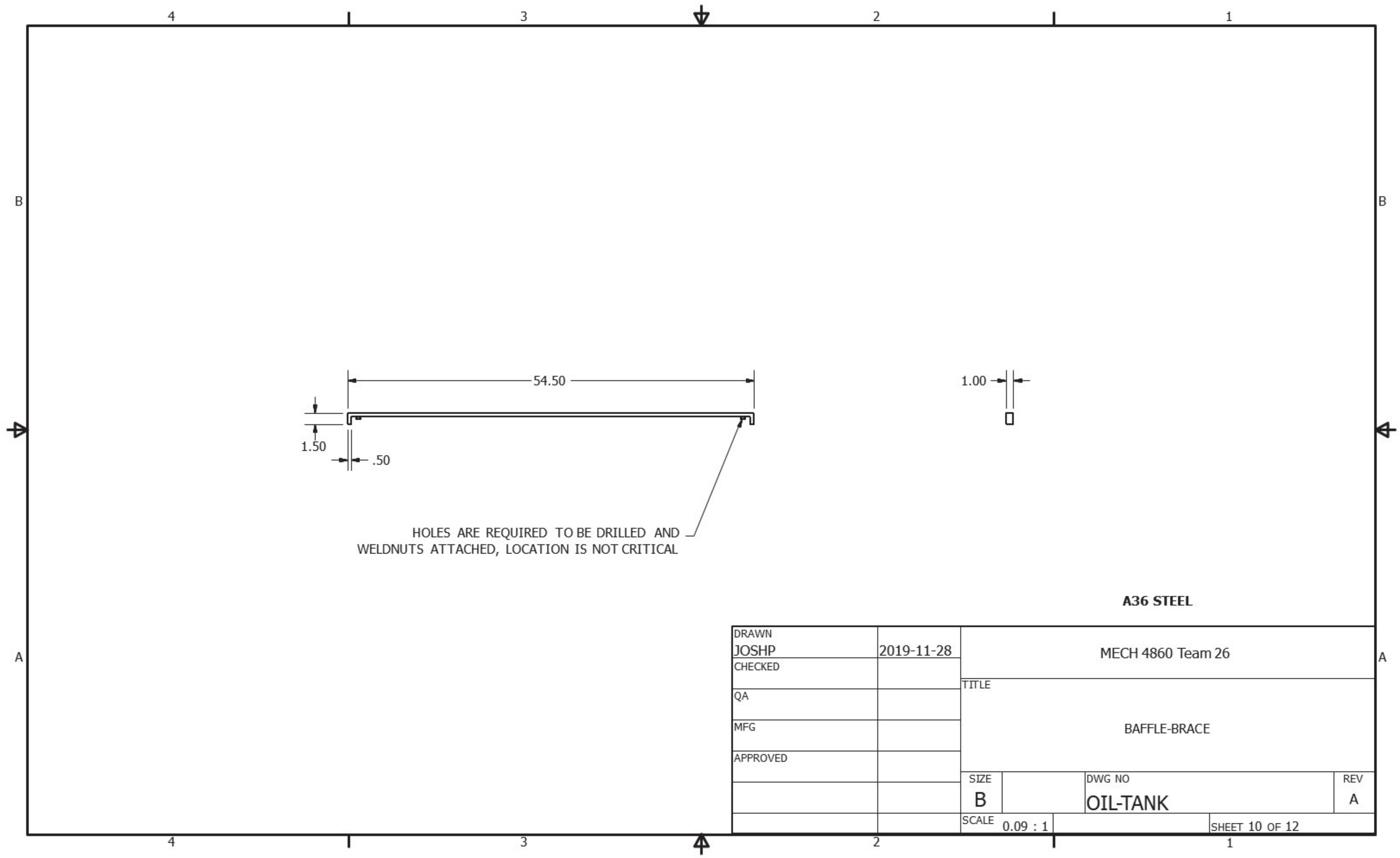
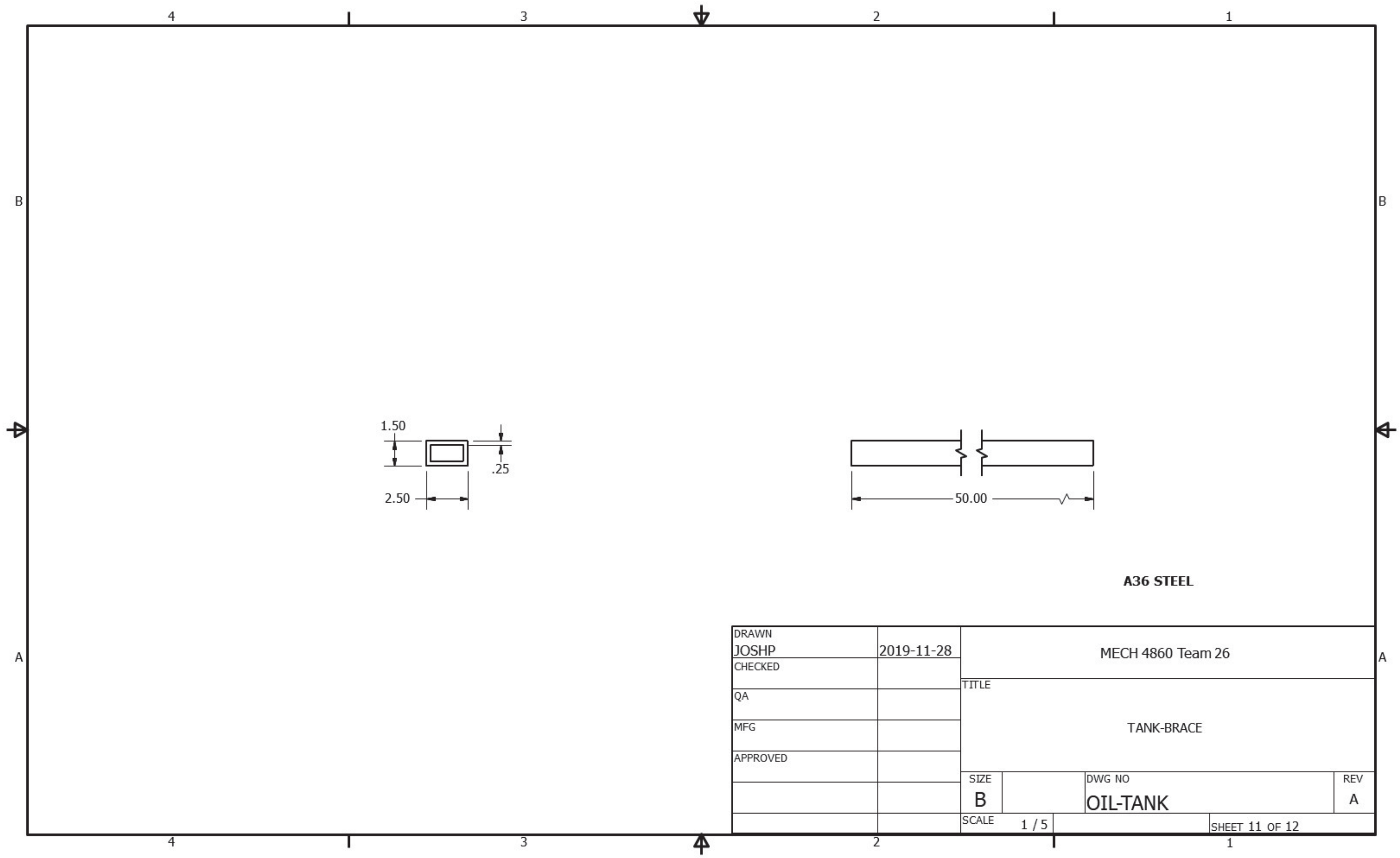
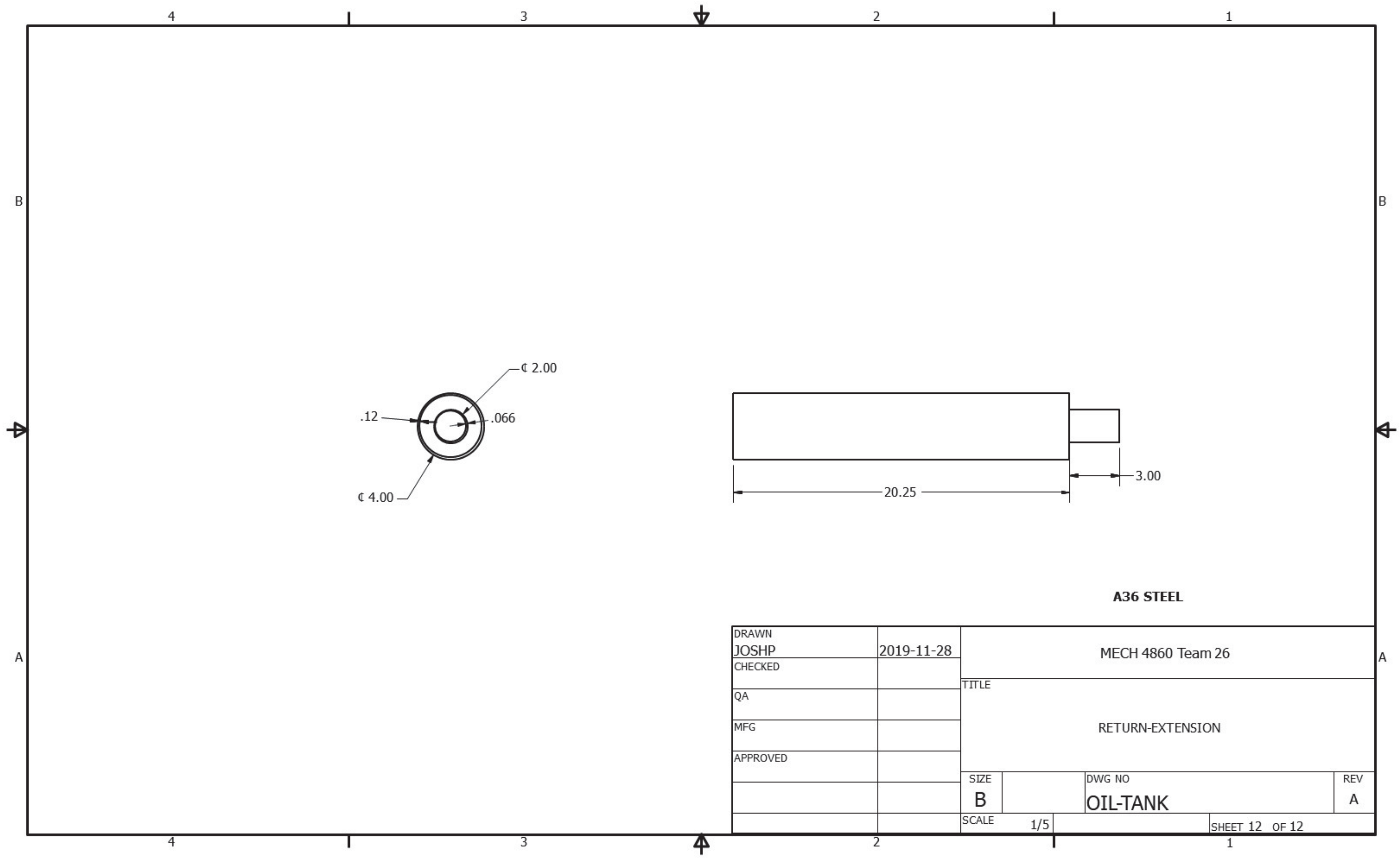
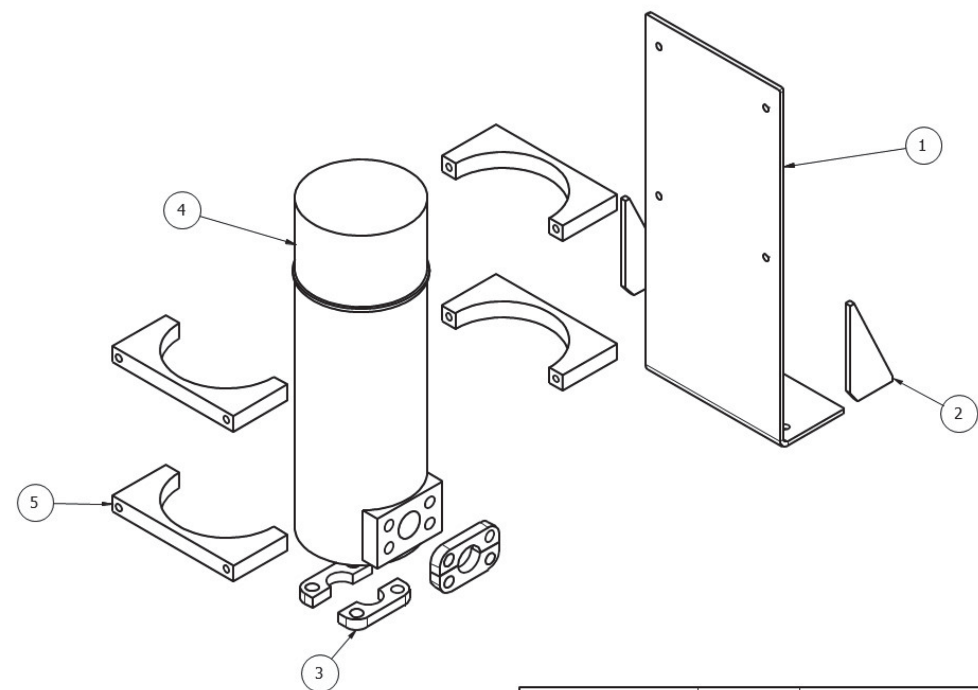
B
A
1.50
.25
2.50
2.50
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
LIFTING-HOLES
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE
1 : 1
SHEET 9 OF 12


54.50
1.50
.50
1.00
HOLES ARE REQUIRED TO BE DRILLED AND
WELDNUTS ATTACHED, LOCATION IS NOT CRITICAL
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
BAFFLE-BRACE
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE 0.09 : 1
SHEET 10 OF 12


1.50
.25
2.50
50.00
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
TANK-BRACE
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE
1 / 5
SHEET 11 OF 12

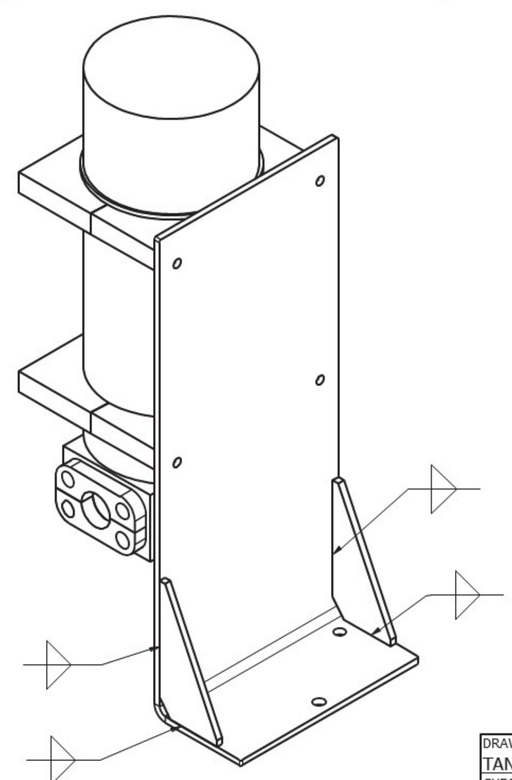


4
3
2
1
B
A
¢ 2.00
.12
.066
¢ 4.00
20.25
3.00
A36 STEEL
DRAWN
JOSHP
2019-11-28
CHECKED
QA
MFG
APPROVED
MECH 4860 Team 26
TITLE
RETURN-EXTENSION
SIZE
B
DWG NO
OIL-TANK
REV
A
SCALE
1/5
SHEET 12 OF 12



DRAWN TANNERG	2019-11-29	MECH 4860 Team 26
CHECKED		TITLE EXPLODED VIEW OF FILTER ASSEMBLY
QA		
MFG		
APPROVED		
SIZE B	DWG NO FITLER-ASSEMBLY	REV A
SCALE 0.15 : 1		SHEET 1 OF 5

PARTS LIST			
ITEM	QTY	PART NUMBER	DESCRIPTION
1	1	Filter Bracket	
2	2	Filter Bracket Brace	
3	4	2in Code 61 Split Flange	
4	1	Filter	
5	4	Filter Clamp	



DRAWN TANNERG	2019-11-29	MECH 4860 Team 26			
CHECKED		TITLE WELDING FILTER BRACKET BRACE TO FILTER BRACKET			
QA					
MFG					
APPROVED					
		SIZE B	DWG NO FITLER-ASSEMBLY		REV A
		SCALE 0.2 : 1		SHEET 2 OF 5	

Appendix B

Bill of Materials and List of Purchased Parts

Table of Contents

1 Introduction

Appendix B contains the bill of materials as well as the list of purchased parts. The bill of materials is broken into three main sections, which are the power unit and fuel tank, the gearbox test unit skid, and the hydraulic systems skid.

2 Bill of Materials

2.1 Power Unit

Part Number	Procurement	Quantity	Description
TCD 13.5	Purchased	1	TCD 13.5
MOUNT-PLATE-B	Fabricated	1	
CLUTCH-ADAPTER	Fabricated	1	
IB314P	Purchased	1	IB314P
MOUNT-PLATE	Fabricated	1	
C-CHANNEL-EXTENSION	Fabricated	2	
CONNECTING-PLATE	Fabricated	2	
POWER-UNIT-FORK-CHANNEL	Fabricated	1	
POWER-UNIT-FORK-PLATE	Fabricated	2	
13432700	Purchased	1	INPUT SPUR GEAR
ANSI B18.3 - 3_8-16 UNC - 2(31)	Purchased	10	
ANSI B18.2.2 - 3_8 - 16(2)	Purchased	10	
11826100	Purchased	2	
18710	Purchased	5	
Cap Screw, Socket Head	Purchased	12	
11826200	Purchased	2	
13337600	Purchased	1	
13433300	Purchased	1	
13432900	Purchased	1	
13336500	Purchased	1	
13343700	Purchased	1	
13434400	Purchased	1	
20022400	Purchased	1	
13336900	Purchased	1	RETAINING RING
OUTPUT-SPUR-GEAR	Fabricated	1	
13331100	Purchased	1	CASING
INPUT-SHAFT	Fabricated	1	
INPUT-END-PLATE	Fabricated	1	
OUTPUT-END-PLATE	Fabricated	1	
OUTPUT-SHAFT	Fabricated	1	
13337600	Purchased	1	RELOCATED OUTPUT BEARING
13336600	Purchased	1	OUTPUT SEAL
Fuel Tank	Purchased	1	GFDW1300

2.2 Gearbox Test Unit Skid

Part Number	Procurement	Quantity	Description
Hose #4	Purchased	1	
Parker SAE Flanged Connection - 45° Elbow 1 1/2 x 2	Purchased	1	
Hose #3	Purchased	1	
GTU-FORK-CHANNEL	Fabricated	2	
GTU-SKID-SIDE	Fabricated	2	
INCREASER-CARRIER	Fabricated	1	
INCREASER-INPUT-SEAL	Purchased	2	SKF49985
INCREASER-OUTPUT-END-PLATE	Fabricated	2	
INCREASER-OUTPUT-SEAL	Purchased	2	SKF39934
INCREASER-INPUT-SHAFT	Fabricated	2	
INCREASER-PLANET	Fabricated	6	
INCREASER-SUN	Fabricated	2	
INCREASER-RING	Fabricated	2	
INCREASER-PLANET-BEARING	Purchased	12	SKFRLS8
INCREASER-INPUT-END-PLATE	Fabricated	2	
INCREASER-OUTPUT-SHAFT	Fabricated	2	
INCREASER-INPUT-BEARING	Purchased	2	6656K160
GTU-BACK-BRACE	Fabricated	3	
GTU-FRONT-BRACE	Fabricated	2	
GTU-BOX-BRACE	Fabricated	2	
K3VG280DT with Auxiliary Gear Pump	Purchased	1	K3VG-280DT-1AFLS-7EH2-0-1
DIAGONAL-SUPPORT	Fabricated	2	
DIAGONAL-SUPPORT-FOOT	Fabricated	2	
CROSS-BEAM-HORZ	Fabricated	4	
MOUNTING BRACKET	Purchased	4	Contact Vendor
CROSS-BEAM-VERT	Fabricated	8	
VERTICAL-SUPPORT	Fabricated	2	
ADJ-BEAM	Fabricated	2	
VERTICAL-SUPPORT-FOOT	Fabricated	2	
ADJ-BRKT	Fabricated	4	
ADJ-FOOT	Fabricated	2	
PUMP-MOUNT-LEG	Fabricated	8	
PUMP-MOUNT-PLATE	Fabricated	1	

Part Number	Procurement	Quantity	Description
K3VG280DT Single	Purchased	1	K3VG-280DT-1AFLS-7EH2-0-0
GTU-ANGLE-SIDE	Fabricated	2	
GTU-ANGLE-BACK	Fabricated	1	
PIVOT-PIN	Purchased	1	
PIVOT-BOTTOM-PLATE	Fabricated	2	
PIVOT-BRACE	Fabricated	2	
ADJ-BOTTOM-MOUNT	Fabricated	2	
LONG-CHANNEL	Fabricated	2	
SHORT-CHANNEL	Fabricated	4	
SKID-PLATE	Fabricated	1	
FORK-CHANNELS	Fabricated	2	
PIVOT-COTTER-PIN	Purchased	1	92391A230
PIVOT-PLATE-A	Fabricated	1	
PIVOT-PLATE-B	Fabricated	2	
ADJ-MOUNT-BRKT	Fabricated	3	
ADJ-MOUNT-PLATE	Fabricated	1	
ROD-END	Purchased	2	60645K72
MID-TUBE	Purchased	1	
ROD	Purchased	2	98957A411
ADJ-PIN	Fabricated	2	
ADJ-COTTER-PIN	Purchased	4	92391A210
Manifold Mount Plate	Fabricated	2	
WAIS T-Block	Purchased	4	WAI/S
LB6S Manifold	Purchased	4	LB6/S
Sealum Code 62 Union 4.69in 1959-24-24	Purchased	6	1959-24-24
LB6S Manifold Spacer	Fabricated	4	
1.5in Code 62 Split Flange Kit	Purchased	16	1825-24
RPKC8WV Control Valve	Purchased	4	RPKC8WV
RBAPXWV	Purchased	4	RBAPXWV
Parker SAE Flanged Connection 1 1/2 x 1 1/2	Purchased	4	
Hose #8	Purchased	1	
Parker SAE Flanged Connection - 90° Elbow 1 1/2 x 2	Purchased	2	
Hose #2	Purchased	1	
Hose #1	Purchased	1	

Part Number	Procurement	Quantity	Description
Hose #6	Purchased	1	
Hose #5	Purchased	1	

2.3 Hydraulic System Skid

Part Number	Procurement	Quantity	Description
Hose #9	Purchased	1	
Hose #11	Purchased	1	
4in Code 61 Split Flange Kit	Purchased	1	1585-64
LEFT-MAIN-RAIL	Fabricated	1	
COOLER	Purchased	1	OCA-3100-1-30-9-G-SFG
TOP-PLATE	Fabricated	1	
LIFTING-BRACE	Fabricated	1	
LIFTING-HOLES	Fabricated	2	
DRAIN-CAP	Purchased	1	32STP-MAG
RETURN-EXTENSION	Fabricated	1	
Threaded Drain	Purchased	1	GGM2TV90
Outlet Fittings	Purchased	2	GGM1LRZ7
Return Elbow	Purchased	1	WWG1LTC4
Sightglass	Purchased	1	FL69321
Diffuser	Purchased	1	5563
TANK-BRACE	Fabricated	12	
BOTTOM-PLATE	Fabricated	1	
FRONT-PLATE	Fabricated	1	
BACK-PLATE	Fabricated	1	
SIDE-PLATE	Fabricated	2	
BAFFLE-BRACE	Fabricated	5	
BAFFLE-PLATE	Purchased	2	PS31614
Metal Filler Breather	Fabricated	1	
Filter Bracket	Fabricated	2	
Filter Bracket Brace	Fabricated	4	
2in Code 61 Split Flange Kit	Purchased	10	1585-32
Filter	Purchased	2	W041
Filter Clamp	Fabricated	8	
3in SAE to 2in Adapter	Fabricated	2	

Part Number	Procurement	Quantity	Description
Parker SAE Flanged Connection - 90° Elbow 2 x 2	Purchased	2	
Parker SAE Flanged Connection 2 x 2	Purchased	7	
Hose #10	Purchased	1	
HYDRAULIC OIL 1040L TOTE	Purchased	2	UHTFT
RIGHT-MAIN-RAIL	Fabricated	1	
CROSS-MEMBER	Fabricated	3	
FRONT-CROSS-MEMBER	Fabricated	1	
FORK-CHANNELS	Fabricated	2	
HYDRAULIC OIL 1040L TOTE	Purchased	2	UHTFT

3 List of Purchased Parts

Part Number	Quantity	Description	Price ($CAD)	Vendor
Hose #9	1		Contact Vendor	Vendor of Choice
Hose #11	1		Contact Vendor	Vendor of Choice
4in Code 61 Split Flange Kit	1	1585-64	Contact Vendor	Sealum
COOLER	1	OCA-3100-1-30-9-G-SFG	$ 18,000.00	Southwest Thermal Technology
DRAIN-CAP	1	32STP-MAG	Contact Vendor	Lenz
Threaded Drain	1	GGM2TV90	Contact Vendor	Acklands Grainger
Outlet Fittings	2	GGM1LRZ7	Contact Vendor	Acklands Grainger
Return Elbow	1	WWG1LTC4	Contact Vendor	Acklands Grainger
Sightglass	1	FL69321	Contact Vendor	Parker
Diffuser	1	5563	Contact Vendor	Parker
BAFFLE-PLATE	2	PS31614	Contact Vendor	Metals Depot
2in Code 61 Split Flange Kit	10	1585-32	Contact Vendor	Sealum
Filter	2	W041	Contact Vendor	Donaldson
Parker SAE Flanged Connection - 90° Elbow 2 x 2	2		Contact Vendor	Parker
Parker SAE Flanged Connection 2 x 2	7		Contact Vendor	Parker

Part Number	Quantity	Description	Price ($CAD)	Vendor
Hose #10	1		Contact Vendor	Parker
Hose #10	1		Contact Vendor	Parker
Hose #4	1		Contact Vendor	Parker
Parker SAE Flanged Connection - 45° Elbow 1 1/2 x 2	1		Contact Vendor	Vendor of Choice
Hose #3	1		Contact Vendor	Parker
INCREASER-INPUT-SEAL	2	SKF49985	Contact Vendor	SKF
INCREASER-OUTPUT-SEAL	2	SKF39934	Contact Vendor	SKF
INCREASER-PLANET-BEARING	12	SKFRLS8	Contact Vendor	SKF
INCREASER-INPUT-BEARING	2	6656K160	Contact Vendor	McMaster Carr
K3VG280DT with Aux. Gear Pump	1	K3VG-280DT-1AFLS-7EH2-0-1	$ 60,000.00	Kawasaki
MOUNTING BRACKET	4	Contact Vendor	Contact Vendor	Kawasaki
K3VG280DT Single	1	K3VG-280DT-1AFLS-7EH2-0-0	$ 55,000.00	Kawasaki
PIVOT-PIN	1		Contact Vendor	McMaster Carr
PIVOT-COTTER-PIN	1	92391A230	$ 48.76	McMaster Carr
ROD-END	2	60645K72	$ 107.28	McMaster Carr
MID-TUBE	1		Contact Vendor	Russel Metals
ROD	2	98957A411	$ 14.97	McMaster Carr
ADJ-COTTER-PIN	4	92391A210	Contact Vendor	McMaster Carr
WAIS T-Block	4	WAI/S	Contact Vendor	Sun Hydraulics
LB6S Manifold	4	LB6/S	Contact Vendor	Sun Hydraulics
Sealum Code 62 Union 4.69in 1959-24-24	6	1959-24-24	Contact Vendor	Sealum
1.5in Code 62 Split Flange Kit	16	1825-24	Contact Vendor	Sealum
RPKC8WV Control Valve	4	RPKC8WV	Contact Vendor	Sun Hydraulics
RBAPXWV	4	RBAPXWV	Contact Vendor	Sun Hydraulics
Parker SAE Flanged Connection 1 1/2 x 1 1/2	4		Contact Vendor	Parker

Part Number	Quantity	Description	Price ($CAD)	Vendor
Hose #8	1		Contact Vendor	Parker
Hose #8	1		Contact Vendor	Parker
Parker SAE Flanged Connection - 90° Elbow 1 1/2 x 2	2		Contact Vendor	Parker
Hose #2	1		Contact Vendor	Parker
Hose #1	1		Contact Vendor	Parker
Hose #6	1		Contact Vendor	Parker
Hose #5	1		Contact Vendor	Parker
TCD 13.5	1	TCD 13.5	$ 85,000.00	Pritchard Dieseltech
IB314P	1	IB314P	Contact Vendor	Twindisc
13432700	1	INPUT SPUR GEAR		Elmer's
ANSI B18.3 - 3_8-16 UNC - 2(31)	10			Elmer's
ANSI B18.2.2 - 3_8 - 16(2)	10			Elmer's
11826100	2			Elmer's
18710	5			Elmer's
Cap Screw, Socket Head	12			Elmer's
11826200	2			Elmer's
13337600	1			Elmer's
13433300	1			Elmer's
13432900	1			Elmer's
13336500	1			Elmer's
13343700	1			Elmer's
13434400	1			Elmer's
20022400	1			Elmer's
13336900	1	RETAINING RING		Elmer's
13331100	1	CASING		Elmer's
13337600	1	RELOCATED OUTPUT BEARING		Elmer's
13336600	1	OUTPUT SEAL		Elmer's
Fuel Tank	1	GFDW1300	Contact Vendor	WESTEEL
HYDRAULIC OIL 1040L TOTE	2	UHTFT	$ 5,595.20	OilMart

Appendix C

WEIGHTED DECISION MATRIX (WDM)

Table of Contents

List of Tables

1 Introduction

The design team created on weighed decision matrix (WDM) for each main module of the designed gearbox test stand. Each WDM is used to compare the generated concepts for each module and select the best concept for the model. The following sections show the WDM created for each module.

2 WDM for Power Source

Table I shows the comparing criteria for selecting the power source as well as gives a brief explanation of the criteria. Each team member weighed the criteria individually and the results were then averaged to obtain the final weighting that was used in the WDM.

TABLE I: POWER SOURCE DEFINED CRITERIA

Criterion	Criterion Weight	Explanation
Weight	0.14	The weight of the power unit is a criterion considered so that the design remains portable at Elmer's facility. The power unit must remain under 9,000 lbs. Therefore, power sources that are over 9,000 lbs will not meet the customer's requirements of remaining portable. As the forklift cannot lift over 9000 lbs, this criterion was given a relatively high weight.
Cost	0.04	The cost of the power source will affect rankings. While the team does not have a specific customer-stated budget for the design, a power source, which cost multi-million dollars, should not be selected. However, for most concepts the price differences between concepts were not substantial, therefore cost received a lower weight.

Criterion	Criterion Weight	Explanation
Size	0.04	The size of the power source will affect the final design to remain portable the system must fit on a skid that is movable by Elmer's forklift. Most power sources at the 600 HP level are of similar size, thus this criterion is weighted with lower scores than others.
Torque	0.17	The torque output of the power source is crucial to testing the gearboxes in the test stand and thus must be accounted for in the criteria weightings. As the torque output of the power source will ultimately be testing the gearboxes, it received a high weighting.
External Dependencies	0.16	Limiting the number of external dependencies of the power source will help keep the design portable and non-reliant on operating locations. The client has emphasized that the design must be portable. Therefore the external dependencies have a high weighting.
Low Starting Temperature	0.08	Accounting for the lowest allowable starting temperature for the concept power sources is important as the power source will be operated within Manitoba, where operating temperatures may be below freezing. Low starting temperatures of the power source will allow for the gearboxes to be tested in a variety of scenarios, and hence the weighting of the criterion is moderate.

Criterion	Criterion Weight	Explanation
Vibration	0.04	The vibration of the power unit is important as it affects the overall life of the test stand components and the easy and safe operation of the entire test stand. As a result of this, the weight for the criterion is low.
Running Time	0.08	As the system will be used to simulate gearboxes in use, the running time of the power source should be considered to allow for adequate time to perform gearbox tests. Therefore, the running time criterion was given a moderate weight.
Noise Level	0.05	The noise level of the power source is important as quieter operating power sources generally are safer for operators, while simultaneously providing greater operator comfort. The client stated that personal protective equipment is available to the operators, therefore the criterion was given a low weight.
Implementation	0.13	The ease of implementation of the power source will determine how much work goes into both designing the system from the team's perspective. The ease of implementation will also determine how much time and effort will be required from Elmer's Manufacturing to implement the design. Therefore, the weight of the criterion was relatively high.

Criterion	Criterion Weight	Explanation
Emissions	0.06	Emissions of the power source should be accounted for, as environmental standards are beginning to change to protect the environment (such as carbon emission, etc.). As laws must be followed, the emissions weighting was considered moderate.

The initial concepts were then ranked in the WDM based on a scale from one to five for each criterion, with one being the worst rating and five being the best rating, is shown in Table II.

TABLE II: POWER SOURCE WDM

Criteria	Weight	Diesel Engine	Diesel Power Unit	Gas Engine	Gas Power Unit	Gas Turbine	Electric Motor	Steam Turbine	Diesel Electric
Weight	0.14	5	4	5	4	5	2	4	2
Cost	0.04	4	3	4	3	1	2	2	1
Size	0.04	3	4	3	4	4	1	2	1
Torque	0.17	5	5	3	3	2	5	4	5
Ext. Dependencies	0.16	5	5	5	5	4	2	1	5
Low Starting Temp.	0.08	3	3	4	4	2	5	2	3
Vibration	0.04	4	4	4	4	5	5	5	4
Running Time	0.08	4	4	3	3	5	5	5	4
Noise Level	0.05	3	3	2	2	1	4	2	2
Implementation	0.13	3	5	3	5	2	2	1	2
Emissions	0.06	3	3	3	3	2	5	3	3
Total (out of 5)		**3.92**	**4.04**	**3.53**	**3.65**	**2.96**	**3.08**	**2.57**	**3.12**

**The concepts that were further considered are highlighted in green.*

3 WDM for Transmission System

Table III shows the comparing criteria for selecting the transmission system as well as gives a brief explanation of the criteria. Each team member weighed the criteria individually and the results were then averaged to obtain the final weighting that was used in the WDM.

TABLE III: TRANSMISSION CONCEPT SELECTION CRITERIA

Criterion	Criterion Weight	Explanation
Weight	0.16	In order to keep the system portable via forklift, the weight of the power unit, which will include the transmission system, must be under 9,000 lbs. Thus, the weight of the transmission system needs to be considered when selecting a concept. Weight was a high importance need so the weight of the transmission is a highly weighted criterion.
Cost	0.08	The transmission system must be reasonably priced as the maximum allowable budget of the project is $200,000. Due to the client not specifying a maximum allowable budget, with this value instead it being determined by the design team and the cost of the transmission system being unlikely to make up a substantial portion of the cost of the gearbox test stand, this criterion has a relatively low weight.

Criterion	Criterion Weight	Explanation
Size	0.14	The gearbox must not take up too much physical space as the entire gearbox test stand must be smaller than a grain cart. This criterion relates to needs and metrics involving the overall dimensions, portability, and stability of the test stand, which had moderately high importance. Therefore, size was given moderately high weight.
Power Transmission Capability	0.23	The transmission system must be able to reliably transmit 600 HP from the power source to the test gearbox. It should be considered if a concept requires extensive design work to be able to achieve this. This criterion was given the highest priority because the main purpose of the transmission is to transfer power and if the transmission system is not capable of transmitting the required power it cannot be used.
Simplicity	0.06	The gearbox should be as simple as possible for reliability as well as ease of design, manufacturing, service, and setup and disconnection of the gearbox test stand. A design could still meet all the needs and metrics of the project without being simple, so this criterion was given a relatively low weight.
Life	0.14	The gearbox needs to last for many hours of stop-start testing and continuous running. The test stand may be used for new gearbox designs for a long time into the future. Thus, life was given a relatively high weight.

Criterion	Criterion Weight	Explanation
Life	0.14	The gearbox needs to last for many hours of stop-start testing and continuous running. The test stand may be used for new gearbox designs for a long time into the future. Thus, life was given a relatively high weight.
Vibration	0.09	Vibration is undesirable as it will cause accelerated wear of components and could result in fatigue failure. In addition, excessive vibration will reduce the stability of the system which is a potential safety hazard. Thus, the vibration was given a moderate weight.
Efficiency	0.05	Elmer's specified that they would like to be able to apply as close to 600 HP as possible to the gearboxes being tested. Thus, the method of power transmission should be as efficient as possible to minimize power losses from the power source to the test gearboxes. However, the variance in efficiency between different transmission types is unlikely to be significant. Therefore, efficiency was given a relatively low weight.
Noise	0.05	Although the gearbox test stand is to be operated outdoors, a lower noise transmission system is desirable as excessive noise will be an annoyance to those working nearby and operators of the test stand will need to wear hearing protection if the system is too loud. However, the client mentioned that the gearbox test stand would be only be used outdoors, lessening the importance of this criterion. Thus, the noise was given relatively low weight.

The initial concepts were then ranked in the WDM based on a scale from one to five for each criterion, with one being the worst rating and five being the best rating, is shown in Table IV.

TABLE IV: TRANSMISSION CONCEPT WDM

Criteria	Weight	Planetary Gearbox with Spur Gears	Planetary Gearbox with Helical Gears	Compound Gearbox with Spur Gears	Compound Gearbox with Helical Gears	Single Stage Gearbox with Spur Gears	Single Stage Gearbox with Helical Gears	Torque Converter	Belt	Chain
Weight	0.16	2	2	3	3	5	5	3	3	3
Cost	0.10	3	1	4	1	5	2	1	1	2
Size	0.14	5	5	4	4	4	4	4	1	1
Power Transmission Capability	0.23	4	5	4	5	4	5	5	2	3
Simplicity	0.06	2	1	3	2	5	3	1	4	4
Life	0.14	4	5	4	5	4	5	5	1	2
Vibration	0.08	1	2	1	2	2	3	5	5	4
Efficiency	0.05	4	3	4	3	5	3	1	1	5
Noise	0.04	1	2	1	2	2	3	5	5	4
Total (out of 5)		**3.24**	**3.43**	**3.42**	**3.51**	**4.12**	**4.11**	**3.72**	**2.21**	**2.76**

**The concepts that were further considered are highlighted in green.*

4 WDM for Torque Limiter

Table V shows the comparing criteria for selecting the transmission system as well as gives a brief explanation of the criteria. Each team member weighed the criteria individually and the results were then averaged to obtain the final weighting that was used in the WDM.

TABLE V: TORQUE LIMITER CONCEPT SELECTION CRITERIA

Criterion	Criterion Weight	Explanation
Weight	0.08	The weight of the torque limiter assembly. The weight of every component in the gearbox test stand is considered to ensure that the design adheres to the client's requirements. However, the weight of the torque limiter relative to the entire assembly is minuscule, and this criterion carries an appropriately low weight.
Cost	0.02	The cost of the torque limiter unit as well as implementation. The cost of every component in the gearbox test stand is considered to provide the most efficient solution to the client, and there is a substantial range in cost between the considered concepts. However, even the most expensive choice is relatively inexpensive compared with the power unit, and the weight of this criterion is low as a result.
Size	0.03	The physical size of the torque limiter. A smaller size is preferred as larger units could lead to difficulties with implementation in the gearbox test stand design, but even the largest choice will be tiny in comparison to other assemblies in the gearbox test stand. As a result, the weighting for this criterion is low.

Criterion	Criterion Weight	Explanation
Torque Capability	0.21	This criterion scores the concepts based on the obtainability and ease of implementation of a commercial design that meets the project load requirements. A concept that is easy to obtain and implement is highly favored due to the associated savings in cost and time. As a result, this criterion has the highest weight.
Reusability	0.12	The capability of the concept to withstand multiple overloading scenarios. A lack of reusability results in downtime and continuous expense which may exceed the upfront cost advantage of the design. As downtime is undesirable, this criterion has a relatively high weighting.
External Dependencies	0.15	The number of systems or devices that are not contained within the gearbox test stand that are required for a specific concept to function and also the ease of implementation of those systems. External dependencies are extremely undesirable as they inhibit the portability of the gearbox test stand assembly, and this criterion was weighted highly as a result.

Criterion	Criterion Weight	Explanation
Availability	0.07	The commercial availability of torque limiters using the specific concept. The wide availability of the torque limiter assembly corresponds to a wide availability in vendors and replacement parts. Flexibility in vendors and availability of replacement parts are desirable qualities but not essential, resulting in a moderate weighting of this criterion.
Resistance to Elements	0.15	The ability of the torque limiter to remain reliable when exposed to outdoor conditions. The gearbox test stand assembly is designed to be used outdoors and it is essential for the torque limiter to continue to function in outdoor conditions. As a result, the weighting for this criterion is relatively high.
Response Time	0.17	The duration of time between when the system begins to experience a load which exceeds the design load, and when the torque limiter disengages the power source from the protected component. Lower response times are desired, as they decrease the likelihood of damage to the protected component. The weighting of this criterion is relatively high, as damage to the protected component is undesirable.

The initial concepts were then ranked in the WDM based on a scale from one to five for each criterion, with one being the worst rating and five being the best rating, is shown in Table VI.

TABLE VI: TORQUE LIMITER WDM

Criteria	Weight	Shear Pin	Ball Detent (Ratcheting) Torque Limiter	Slip Clutch	Air-Engaged Torque Limiter
Weight	0.08	5	4	4	4
Cost	0.02	5	2	4	1
Size	0.03	5	3	4	3
Torque Capability	0.21	2	4	5	2
Reusability	0.12	1	5	4	5
External Dependencies	0.15	5	5	5	2
Availability	0.07	5	3	5	3
Resistance to Elements	0.15	5	4	5	4
Response Time	0.17	5	4	4	4
Total (out of 5)		**3.88**	**4.13**	**4.58**	**3.24**

**The concepts that were further considered are highlighted in green.*

5 WDM for Load Application System

Table VII shows the comparing criteria for selecting the transmission system as well as gives a brief explanation of the criteria. Each team member weighed the criteria individually and the results were then averaged to obtain the final weighting that was used in the WDM.

TABLE VII: LOAD APPLICATION SYSTEM DEFINED CRITERIA

Criterion	Criterion Weight	Explanation
Weight	0.16	The load application systems must have a reasonable weight to keep the total weight of the gearbox test unit less than 9000 lbs to ensure the gearbox test unit can be lifted by Elmer’s forklift. Therefore, this criterion has a relatively high weight.
Cost	0.09	Since Elmer’s does not have a specific requirement for the project budget, the cost of the load application systems is not relatively important. However, the load application systems must be reasonably priced. Hence, the weighting of the criterion is moderate.
Size	0.10	The load application systems must not take up too much physical space as the entire gearbox test stand must be smaller than a grain cart. Hence, the weighting of the criterion is moderate.
Power Absorption Capability	0.28	The main purpose of having the load application systems is to reliably apply an adjustable load, from 0 to 600 HP, on test gearbox. Thus, the power that the system is capable of absorbing is the most important criterion.

Criterion	Criterion Weight	Explanation
External Dependencies	0.25	Based on the client’s needs, the load application systems must need minimum external dependencies to keep the test stand portable and not rely on the test locations. Therefore, the number of external dependencies is a vital criterion to select the load application system.
Simplicity	0.02	The load application systems must be as simple as possible for reliability as well as ease of design, manufacturing, and service. Since purchasing on-shelf dynamometers is an acceptable option for Elmer’s, the simplicity of the design and manufacturing processes is not a significant criterion.
Life	0.10	The load application systems need to have a reasonable lifetime without any replacement. Hence, the weighting of the criterion is moderate.

The initial concepts were then ranked in the WDM based on a scale from one to five for each criterion, with one being the worst rating and five being the best rating, is shown in Table VIII.

TABLE VIII: LOAD APPLICATION SYSTEM WDM

Criteria	Weight	Eddy Current (Liquid-cooled)	Eddy Current (Air-cooled)	Water Brake	Hydraulic Pump	Air Brake	Hysteresis	Magnetic Powder	Electric Generator	Friction Brake
Weight	0.16	2	2	5	2	5	2	2	1	2
Cost	0.09	2	3	5	2	4	3	2	1	2
Size	0.10	4	5	3	4	1	4	4	4	4
Power Absorption Capability	0.28	5	4	5	5	1	2	3	5	3
External Dependencies	0.25	2	3	2	4	5	3	3	3	4
Simplicity	0.02	2	3	4	5	5	3	3	3	3
Life	0.10	5	2	5	3	5	2	4	4	2
Total (out of 5)		**3.35**	**3.22**	**4.03**	**3.70**	**3.39**	**2.56**	**2.96**	**3.27**	**3.00**

**The concepts that were further considered are highlighted in green.*

6 WDM for Gearbox Mounting System

Table IX shows the comparing criteria for selecting the transmission system as well as gives a brief explanation of the criteria. Each team member weighed the criteria individually and the results were then averaged to obtain the final weighting that was used in the WDM.

TABLE IX: GEARBOX MOUNTING DEFINED CRITERIA

Criterion	Criterion Weight	Explanation
Weight	0.11	The weight will play a role in the concept that is selected since the overall design is restricted by weight. Minimizing the weight in the mounting system is beneficial to the overall design. Therefore, the weighting of this criterion was given a high value.
Cost	0.06	The cost will also affect the concepts as the project budget should not be larger than necessary, and minimizing cost is important. However, most concepts are expected to be similar in price, so the cost criterion was given a relatively low value.
Size	0.04	The overall size of the mounting system contributes towards the success of the concept as the design should be easily manageable and size should be limited. However, since the design will be accommodating Elmer's gearboxes, the concepts should be of similar size and therefore the weighting is low.
Adjustability	0.23	Since there are multiple gearboxes being tested at different operating angles, having adjustability will lead towards a more successful design. The design must adjust to accommodate the different gearboxes. Therefore, adjustability was given a high weighting.

Criterion	Criterion Weight	Explanation
Setup Time	0.11	For the ease of operators, setup time should be considered for each of the concepts. Reducing setup time will contribute towards operators approving of the design. Therefore, the setup time was given a moderate weight.
Manufacturability	0.13	As this system will be unique, it is important that manufacturing is considered for the concepts to ensure the designs can be produced. As the design should be manufacturable at the client's facility, manufacturability was given a moderate weighting.
Life	0.11	This system will be under load for extended amounts of time during test cycles, so it is important that the mounting system has a large lifetime. As such, the life of the design was given a moderate weighting.
Stability	0.21	The mounting system must also be able to hold the test gearbox in place during the extended test cycles. Therefore, stability is an important parameter and is weighted high.

The initial concepts were then ranked in the WDM based on a scale from one to five for each criterion, with one being the worst rating and five being the best rating, is shown in Table X.

TABLE X: GEARBOX MOUNTING WDM

Criteria	Weight	Plates	Adjustable Arms	Pivoting U-Bracket	Adjustable Arms Skid	Pivoting U-Bracket Skid
Weight	0.11	4	4	3	3	2
Cost	0.06	5	4	3	3	2
Size	0.04	4	3	3	2	3
Adjustability	0.23	1	5	5	5	4
Setup Time	0.11	4	4	5	4	5
Manufacturability	0.13	5	4	3	4	3
Life	0.11	5	4	5	5	5
Stability	0.21	5	4	4	5	5
Total (out of 5)		**3.83**	**4.19**	**4.11**	**4.30**	**3.91**

**The concepts that were further considered are highlighted in green.*