Gangplank System Redesign

Final Design Report

MECH 4860 – Engineering Design

Submitted by:	Team 23	
	Thomas Jaeger	____________________
	Jeffrey Lach	____________________
	Cassandra Purtill	____________________
	Steven Vanderwees	____________________
Project Advisor:	Dr. Malcom Xing	
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Steven Vanderwees

Dr. Paul Labossiere
75 Chancellors Circle
E2-327F EITC
Faculty of Mechanical Engineering
University of Manitoba
Winnipeg, Manitoba
R3T 5V6

Dear Dr. Labossiere,

On behalf of MECH 4860 Engineering Design's Team 23 we are pleased to present our Final Deign Report for the Gangplank Redesign Project initialized by Triple E Recreational Vehicles for the Fall 2016 term.

The purpose of this report is to develop a preliminary design for a dynamic gangplank system used for working on the rooves of luxury travel vans. The current design consists of a solid platform which is lifted in and out of place, by two employees, which spans the gap between two side scaffolds. Two main issues with the current design are strain on employees from overhead manual lifting and the resource requirements required to install and remove the design during workstation changeovers.

The new design is permanently attached to one side of the scaffolding and extends automatically to bridge the gap to the other scaffold. Due to height constraints, the new design is split into two portions which are connected with a hinge such that they can fold out to form a bridge in the deployed position, or fold in to reduce the overall height in the storage position. To power the new design, a DC electric winch and DC electric linear actuator are used in conjunction to lift the first portion of the folding gangplank and to fold the second portion near the first, respectively. Major design components of the new deign include the under deck supports, power system mounts, supporting end plates, center hinge, and safety devices.

The proposed design provides a walking surface that is approximately 600 mm wide and 2700 mm long in its deployed state. All weight bearing members satisfy the Manitoba Heath and Safety regulations requirement of being able to support 4 times the rated load of 600 lbs. In its storage state the new design occupies a space of 2100 mm by 650 mm by 460 mm and can retract completely out of the way allowing the recreational vehicles to be moved from station to station. The estimated final cost of the new design is broken down into three components: power system, hardware, and raw materials. The power system and hardware are expected to cost $235.62 and $52.30, respectfully. The exact material costs for the new design are difficult to calculate as the materials used are a combination of what the client already stocks and materials that will need to be special ordered. Due to the low quantity of units being produced it is impossible to give an accurate cost of the raw materials therefore a total cost of the gangplank system is not given.

Concluding the project, Team 23 have developed a functional, safe, and successful design that is easily implementable for all workstations, throughout the client's manufacturing facility. In presenting this report we would like to acknowledge the assistance given by our client, Triple Recreational Vehicles, and our project advisor, Dr. Malcolm Xing, in providing input and guidance during this project. If you have any further questions about the report please do not hesitate to contact me by phone at or email at

Sincerely,

Steven Vanderwees
Team Leader

EXECUTIVE SUMMARY

In this report, details of the gangplank redesign project initiated by the client, Triple E Recreational Vehicles, are presented. These details include background information on the client, details of the issue they tasked the design team to solve, and the solution which the team created.

Triple E RV, located in Winkler Manitoba, is a leading manufacturer of luxury Class B motorhomes which produced on an assembly line. These motorhomes are often called RVs. After assembling the bodies of the RVs, they are moved from one workstation to the next where various components are added to the top of the vehicle. Using raised scaffolds which flank the sides of the RV, multiple workers install items such as trim, sealants, air conditioners, vents, decals, and many other components on the roof. Some of the installed components are located near the front and rear of the vehicle which require a gangplank to bridge between the side scaffolds creating 360° access to the RVs roof. The current gangplank consists of a platform supported by solid 6 inch aluminum channels which are manually lifted overhead into and out of place by two workers. The lifting process is of concern as there is a chance the workers could injure themselves when lifting the 66 lbs gangplank over their heads. When installed, the gangplank blocks the lane which the RVs use to move from workstation to workstation; therefore, the gangplanks must be removed each time the assembly line advances. The objective of this project was to develop a new gangplank system which could be deployed safely by one person to reduce labor resource requirements and to prevent injury from overhead lifting. The new design needed to complete these objectives while maintaining the existing gangplank's functionality.

After investigating various solutions, the V-Fold design was determined to be the optimal configuration. This design has two connected bridge portions that fold out of the way during workstation moves. The first and second bridge portion are 1639 mm and 1061 mm long, respectively. The width of the bridge is 609.6 mm, 24 inches, wide. The design is attached to one side of the scaffolding and mounted 460 mm away from the edge. The bridge portions use two 4x1.5x0.25 inch C-channels as the main under deck supports frame designed to a minimum safety factor of 4 as per Manitoba Workplace Heath and Safety Standards. The folded configuration has a height of 2096 mm and is fully contained within

the side scaffolding. The center hinge and actuator mounting brackets are made of CSA G40.21 50W steel. All other manufactured components made of 6061-T6 aluminum.

The design uses two powered devices to deploy and retract the gangplank. First, a 2000 lb ATV winch mounted to the scaffold is hooked to the floor of the first portion of bridge and the bridge begins being raised. After a one second delay, to avoid interference, a 600 lb 12” stroke DC linear actuator is used to retract the second portion of the bridge. Deployment is estimated to take 28 seconds. Using the new design, no heavy lifting is required for operation.

The attached safety railing consists of four parts. Two vertical posts used to support the railing, one horizontal tube acting as an intermediate rail, and a bent tube with two rounded ends creating the end of the railing and the top rail. The intermediate and top rail locations were governed by Manitoba Workplace Heath and Safety regulations and are positioned 1000 mm and 500 mm from the top of the gangplank, respectively.

The power systems and hardware are estimated to cost $235.62 and $52.30, respectfully. The exact material costs for the new design are difficult to calculate as the materials used are a combination of what the client already stocks and materials that will need to be special ordered. Due to the low quantity of units being produced it is impossible to give an accurate cost of the raw materials therefore a total cost of the gangplank system is not given.

Enclosed in this report are calculation based analysis of components and assemblies, engineering drawings, a bill of materials, and a 3D CAD model of the finalized design.

After the conclusion of this project, the team recommends that the client have a certified professional engineer review the calculations and drawing presented in this report. In addition, the team recommends that a prototype be developed for life cycle testing and validation of the calculation based analysis.

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1 INTRODUCTION

1.1 BACKGROUND

Initiated by Triple E Recreational Vehicles, the client, the team's goal for this project was to design a replacement for the current gangplank system in use at Triple E RV.

1.1.1 CLIENT INFORMATION

Triple E RV is a luxury recreational vehicle manufacturer located in Winkler, Manitoba [1]. The company was founded in 1965. Initially, Triple E RV manufactured small tow-behind trailers and eventually transitioned to being a leading manufacturer of Class B luxury motorhomes as known as RVs. Figure 1 shows an example of an RV produced by Triple E.

Figure 1: Triple E Serenity recreational vehicle [2].

1.1.2 CURRENT DESIGN

In Triple E RV's manufacturing facility, each RV is assembled by hand on several assembly lines. Each of the individual assembly lines follow the same general assembly procedure described below.

To begin the process of building an RV the floor is laid on top of a cut away chassis consisting of a fully fabricated cab, where the driver sits, and a bare steel frame which sits on the rear suspension. On the bare frame, plywood covered in vinyl is laid to make the interior floor. Preconstructed cabinets are then screwed down to create the interior layout of the RV. After the cabinets are in place the walls, roof, and rear cap are installed enclosing the interior and creating the exterior of the RV. Once the roof is on, raised scaffolds which flank the sides provide easy access to the roof such that multiple workers can install items such as trim, sealants, air conditioners, vents, decals, and many other components on the roof. Some of the installed components are located near the front and rear of the vehicle which require a gangplank to bridge between the side scaffolds creating 360° access to the RVs roof. Figure 2 shows the flanking scaffolds in blue and the gangplank in grey.

Figure 2: Current gangplank system [3].

As seen in Figure 2 the gangplank provides access to the top rear of the RV. An additional gangplank is used at the front of the RV to access the area above the cab while spanning over the hood of the vehicle.

The current gangplank consists of a platform supported by solid 6 inch aluminum channels which are manually lifted overhead into and out of place by two workers. When installed, the gangplanks block the lane which the RVs use to move from workstation to workstation; therefore, the gangplanks must be removed each time the assembly line advances.

To secure the gangplank to the flanking scaffolds two clams are used to prevent the scaffolds from pulling apart which would cause the gangplank to fall. If workers are on the gangplank when this happens injury would certainly occur in addition to the damage caused on the RV. Figure 3 shows the clamps used to attach to the scaffold. The bridge clamps onto both scaffolds, tying them together and stabilizing the entire structure.

Figure 3: Bridge clamp [4].

Presently, the operating procedure for installation and tear down of the gangplanks are as follows:

1. The RV is pushed into place between the scaffolds.
2. The front and rear gangplanks are manually lifted, above head height, from the floor and placed on the scaffold in front of and behind the RV. Two employees are required to lift each gangplank.
3. The gangplanks are clamped to the scaffolds on each side.
4. The guardrails are placed into the receiving mounts on the gangplanks.
5. After the work is done on the RV, the guard rail is removed.
6. The gangplanks are unclamped from the scaffolds.

7. The gangplanks are lifted off the scaffold, again above shoulder height, and placed on the floor. Two employees are required for this step.
8. The RV is pushed out of the workstation.

The gangplank and scaffold system is used in multiple locations around the manufacturing facility on the different assembly lines. The two main types of scaffolds are non-movable, which are fixed to the ceiling, and movable, which are rolled on the ground. The fixed scaffolds are spaced further apart then the movable and are located closer to the ceiling creating more geometrical restrictions.

1.1.3 PROBLEMS WITH CURRENT DESIGN

The current design has the following problems:

- The gangplank is heavy enough to require two workers to manipulate. This requires resources which could be used elsewhere while the RVs are being moved from workstation to workstation.
- The gangplank must be lifted overhead height increasing risk of injury and strain on employees due to lifting.
- The gangplank is stored on the floor while RVs are being moved to the next workstation, introducing a tripping hazard.
- The gangplanks take approximately 40 seconds per plank to set up and ensure it is secure. This slows down cycle time.

1.2 PROJECT OBJECTIVES

The objective of this project was to design a system to replace the current gangplanks in use at Triple E RV. The reasons for the requested new design were to:

1. Improve employee safety by reducing or eliminating the manual lifting requirements of the current design.
2. Reduce the labour resource requirements.

A successful design must therefore satisfy both these requirements.

The scope of the design created for this project was confined to the gangplank portion spanning the front and back of the RVs including the power systems. The design does not include the specifics of any control circuity needed to coordinate multiple power sources. Additionally, the project does not include the redesign of the side flanking scaffolds.

To reduce the amount of repeated work during the project, the design was restricted to the fixed scaffolds as they are further apart and closer to the ceiling creating more restrictions. The final design is easily adaptable to bridge the smaller gap between the movable scaffolds and will not interfere with the higher ceilings.

The deliverables for this project include calculation based analysis, SolidWorks CAD models of all components and assemblies, a bill of materials, and engineering drawings of the final design.

1.3 NEEDS AND SPECIFICATIONS

The team created a list of target specifications based on the needs presented by the client. The target specifications were used as parameters to evaluate each design concept's performance.

1.3.1 PRODUCT NEEDS

The following is a list to summarize the client's needs in order of priority.

1. Allows workers to access top of cab.
2. Supports the weight of workers and equipment.
3. Allows RVs to move from workstation to workstation when necessary.
4. Includes safety measures to prevent workers from falling.
5. Is safe to deploy.
6. Is easy to deploy.
7. Is quick to deploy.
8. Protects RVs from damage.
9. Meets station size restraints.
10. Does not cause scaffold to tip over.
11. Secures scaffolds and prevent them from moving.
12. Is adjustable for different workstations.
13. Implements safety measures to prevent equipment from falling.
14. Is durable and resistance to daily usage.
15. Allows roof cap to be transferred from floor to top of cab.
16. Is lightweight.
17. Meets budget requirements.
18. Is manufacturable at client facility.
19. Is aesthetically pleasing.
20. Can deploy quietly.

1.3.2 TARGET SPECIFICATIONS

To identify the target specifications for the final design, a house of quality analysis was used. The house of quality relates the needs, listed in Section 1.3.1, of the project to the engineering functional requirements of the final design. The functional requirements are the measurable or otherwise assessable characteristics of the design, such as weight and aesthetics.

The needs are related to the functional requirements by proportion; if a certain need is affected by a functional requirement then the relation between them is given a value proportional to the strength of the relationship. For example, when the new design is in its storage position, the width of the opening created between the scaffolds needs to be enough for the RV to pass from one workstation to another. The relationship described above between the functional requirement and the need is classified as strong and therefore a value of nine is assigned. A moderate relationship is given a value of three, a weak relationship is assigned as one, and no relationship as zero.

The needs are then rated for importance on a scale of one to five, which is then converted to a relative weight out of 100. The importance of each functional requirement is then calculated by summing the products of the relative weights of each need and the strength of the relationships to the functional requirement.

Using a house of quality analysis, target specifications for all the functional requirements were set using information from the client and the intuition of the design team. TABLE I lists the functional requirements identified for the design, the target specifications for each functional requirement, and the relative importance of meeting each target as determined in the house of quality.

TABLE I: FUNCTIONAL REQUIREMENTS AND TARGET SPECIFICATIONS

Functional Requirement	Target Specification	Relative Importance
Weight	<35 kg	5.7
Aesthetically Pleasing	Subj. 5/5	0.7
Rated Load	250 kg	8.2
Deployment Time	< 30 sec	7.4
Operation Force	90 N	5.7
Centre of Gravity Remains Within Scaffold	Yes	6.0
Height of Fall Prevention Rail	Top Rail: 0.9 to 1.06 m Intermediate Rail: 0.45 to 0.53 m	6.8
Length	2.48 to 2.71 m Dependent on Workstation	9.2
Width	> 0.5 m	3.9
Toe Board Height	> 0.125 m	3.2
Unit Lifetime	15 years	2.0
Midpoint Deflection	< 0.01 m	2.7
Percent Manufactured in House	100%	2.4
Cost of Parts	$200	1.3
Number of People for Operation	1 person	9.3
Clearance Gap with RV	< 0.3 m	7.4
Station Length Adjustment	±0.07 m	4.3
Height of Platform from Ground	1.715 to 1.87 m	9.3
Undeployed Gap Width	2.1 m	3.8
Operation Noise level	80 dB	0.7

1.4 CONSTRAINTS AND LIMITATIONS

Based on consultation with Triple E RV, our team determined that the two main limitations to consider for the new gangplank system were:

1. Physical size restriction.
2. Health and safety regulations/

Introduced by existing systems, several size constraints were placed on the project. The existing side scaffolds and the width of the RV, shown in Figure 4, create length restrictions due to the fixed distance between the two side scaffolds shown in blue. In addition, the fixed working height of the scaffolds creates a height restriction between itself and the manufacturing facility's roof.

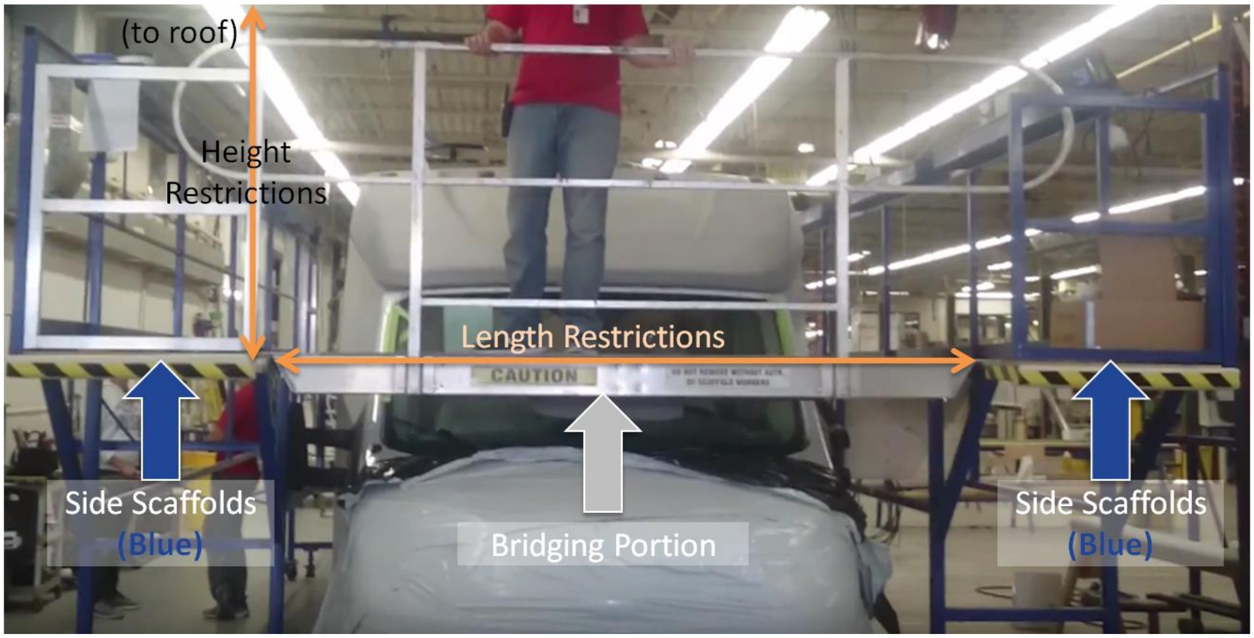


Figure 4: Scaffold components and restrictions [5].

If the team were to ignore this restriction, the new design would have been too short to bridge the gap between the scaffolds; too long, exceeding maximum useable length; or too high, causing interference with the roof. Any of the above situations would have deemed the project to be a functional failure.

The second major restriction for this project was the safety of the workers using the product. Many of the restrictions are mandated by health and safety codes outlined in

Manitoba's Workplace Safety and Health Regulation Act [6]. Sections which are most relevant to our project include the following:

- 4.5 Slipping and Tripping Hazards
- 14.3-14.5 Guardrail Systems
- 28 Scaffolds and Other Elevated Work Platforms

These regulations restrict parameters such as maximum rated load that can be supported, inclusion of fall prevention devices, minimum or maximum dimensions of fall prevention devices, and minimum strength requirements for certain elements of the design.

If these regulations are not followed during the design process two major incidents could occur. The first and less impactful incident would be failure to pass safety inspections before use, leading to the scrapping or redesign of the product. The second and more impactful incident would be the injury or death of an employee while using our product due to insufficient design. The cost of injury, loss of limb, or death of an employee is unacceptable in every situation.

2 CONCEPT GENERATION AND SELECTION

Various concepts were created to satisfy the client's needs which were presented earlier in this report. These concepts drew inspiration from various sources. After creation, the concepts were put through a systematic procedure determine the best theoretical solution.

2.1 CONCEPT GENERATION METHODOLOGY

To create concepts various internal and external methods were used. These methods included research into existing designs, patent searches, consultation with lead users, consultation with technical experts, and brainstorming. These methods provided a wide variety of mechanism and power source inspirations.

When search for existing designs much inspiration was taken from large vehicle bridges which move to clear a lane for ships to pass under or through. Various concepts created from this inspiration include a drawbridge, lifting bridge, v-fold bridge, and many more. Additional design inspiration was taken from devices that were invented for different applications such as scissor lifts, drawer slides, and telescopic cranes.

To ensure the new gangplank design did not conflict with any existing patentable designs a search was completed. The team found that while most patents pertaining to our design space were filed more than 20 years ago, there were some that could infringe on newer patents. Steps were taken to avoid or change the patentable designs such that they would not infringe if used at Triple E RV.

To ensure that the new design considered the preferences of the employees who would be using the final design the team interviewed several lead users. While not yielding any new concepts, by consulting with the lead users the team found that a concept which attaches to the scaffolds would be preferable to one that attaches to the ceiling. Consultation with the operations manager and the R&D manager yielded a preferred choice of four power sources: manual, electric, pneumatic, and hydraulic.

Consulting with a technical expert, Ron Vanderwees P.Eng, who designed the gangplank system currently in use at Triple E RV, the team found additional sources of inspiration for the gangplank mechanism, an example of which are the ramps used to load and unload flat deck trailers. Mr. Vanderwees also gave guidance on how to integrate the power systems into the mechanism designs.

Finally, the concepts gained through the other generation methods were combined and modified to create more concepts. Different components from separate concepts were combined to create additional ideas. In all thirteen mechanisms and four power sources were generated.

2.2 CONCEPT SELECTION METHODOLGY

To select the best of the generated concepts three stages of concept selection were completed: mechanism screening, power system screening, and integrated concept scoring.

To narrow the 13 mechanisms into a more manageable number, a systematic screening method was used. To judge the concepts several criteria were generated from the list of client needs presented in Section 1.3.1. The criteria related to the mechanisms only with no regard to the power system. Each of the 13 concepts were then compared to a baseline to see if they were better, worse, or the same with regards to each criterion. The score of each concept was then summed and the top five mechanisms were moved to the next concept selection stage.

Using the top five mechanisms, the four proposed power sources were then applied to create 20 concepts. To prevent the results from the first screening from being overshadowed the top two power system from each of the five mechanisms were carried forward to the third and final stage of concept selection. Though a similar method was employed to screen the power systems different criteria were used. These criteria related to the power system with little regard to the mechanism.

The final stage of concept selection was the integrated concept soring which is a more rigorous method then concept screening. First a list of criteria was selected which related to both the mechanism for the gangplank and the power system used to move it. These criterial were then weighed to find their relative importance to one another. Each integrated concept was then scored on a scale of one to ten with respect to each criterion. Multiplying the concept's score by each criteria's weight and summing the result, total score for each concept was found. The highest scoring concept was then deemed the best theoretical solution to solve the client's problem.

2.3 CONCEPT SELECTION RESULTS

After screening and scoring the concepts, the V-fold bridge mechanism, shown in Figure 5, was chosen as the best theoretical to solution to Triple E RV's problems. Additionally, the theoretical optimal power source was selected a linear pneumatic system consisting of multiple cylinders.

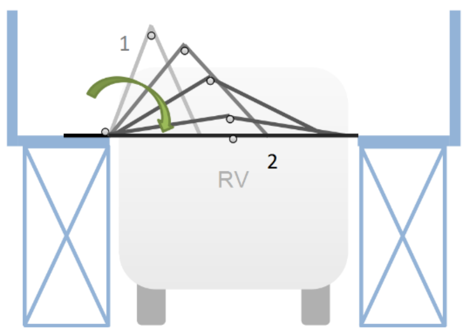


Figure 5: V-fold concept.

While this solution was deemed the best using the concept screening and scoring process, after further investigation various factors prevented the use of pneumatics to power the V-fold design. These factors include the need to mount one cylinder inside the scaffolding requiring major modification, the quantity of force required in the allowable space, and the coordination of multiple pneumatic cylinders using expensive programable logic controllers.

3 DESIGN DETAILS

This section describes the final design and the steps taken in designing the various gangplank system components. Design of the gangplank involved computational dynamic analysis in addition to the design of major parts and specification of material

3.1 DESIGN INTRODUCTION

Shown in Figure 6 and Figure 7 is the final design of the new gangplank system in its deployed and storage state, respectively.

Figure 6: Final gangplank design in deployed position.

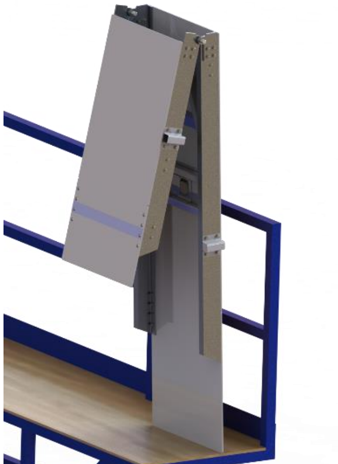


Figure 7: Final gangplank design in storage position.

The design consists of 4 inch aluminum channels supporting the upper walking surface. Approximately 60% of the way across the 2700 mm gap, the gangplank is hinged using four 1/2 inch steel plates held to the gangplank sections using bolts. These plates are locked together by 3/4 inch steel pins which allow the two gangplank sections to rotate independent of each other. This hinge is referred to as the center hinge. Powering the folding action of the gangplank one DC linear actuator provides 600lbs of force transferred to the gangplank sections via bolted hybrid aluminum and steel mount points and supports. The gangplank is lifted from its deployed position using a ATV winch and is attached to one side of the scaffold using the pivot mount hinge.

To ensure the workers who use the gangplank do not fall a safety rail is provided per Manitoba Health and Safety regulations. The safety rail is held onto the gangplank by two bolted railing receivers attached to the main support channels

3.2 KINEMATICS AND DYNAMICS

The team used kinematic and dynamic analysis to determine the necessary load requirements in the pivot mount hinge, the center hinge, the linear actuator, and winch. Position analysis was used to ensure the physical feasibility of the configuration such that the gangplank sections do not fold into one another and the linear actuator is not over extended. Velocity and acceleration analysis were then used to determine variables needed in the subsequent force analysis and to ensure the maximum speed stays below a safe level. Finally, a force analysis was used to determine the load requirements of the aforementioned components.

3.2.1 ANALYSIS SETUP

The team started with an initial configuration for the position analysis, shown in Figure 8.

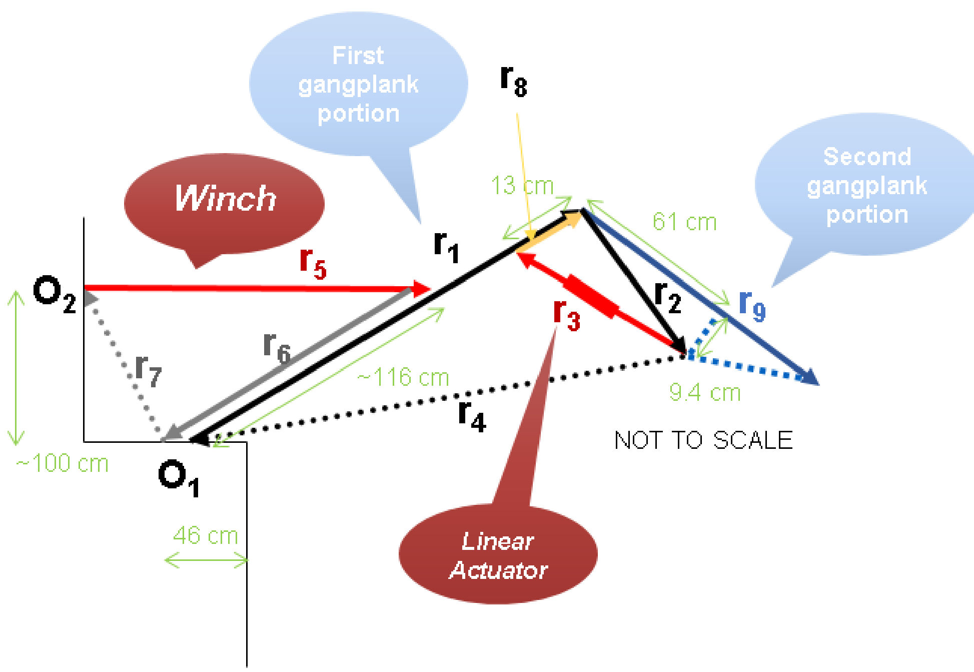


Figure 8: Initial dynamic analysis configuration.

The gangplank is represented by 2D vectors R1 through R9. R1 represents the first portion of the gangplank, and R9 represents the second portion. O1 and O2 are the mounting points of the pivot mount hinge and the winch, respectively. Detailed descriptions of each vector are given in TABLE II along with their dimensions. Note that some vectors represent physical components of the gangplank while others are added to complete vector loops for the position, velocity, acceleration, and force analysis. All kinematic and dynamic calculations were done using Maple math software.

TABLE II: VECTOR DESCRIPTIONS AND DIMENSIONS

Vector	Description	Length (m)
R1	First portion of bridge including end plate	1.740
R2	Vector from middle hinge to pick up point on second portion of gangplank	0.615
R3	Linear Actuator	0.748 to 0.481
R4	Vector from pick up point on second portion of gangplank to the pivot mount hinge	N/R
R5	Winch cable	1.668 to 0.280
R6	Winch cable pick up point on first portion of bridge	1.160
R7	Vector from pivot mount hinge location to winch mount location	1.026
R8	Portion of R1 connected to R2 and R3	0.134
R9	Second portion of gangplank	1.160

3.2.2 POSITION ANALYSIS

Positional analysis was used to first ensure that the configuration was physically feasible and met our target clearance requirements. Snapshots of the extension animation are shown in Figure 9. This analysis was critical for determining the pick-up points of the linear actuator, represented by vector R3, to ensure the gangplank could fully retract and clear the space such that the RV can be pushed to the next workstation.

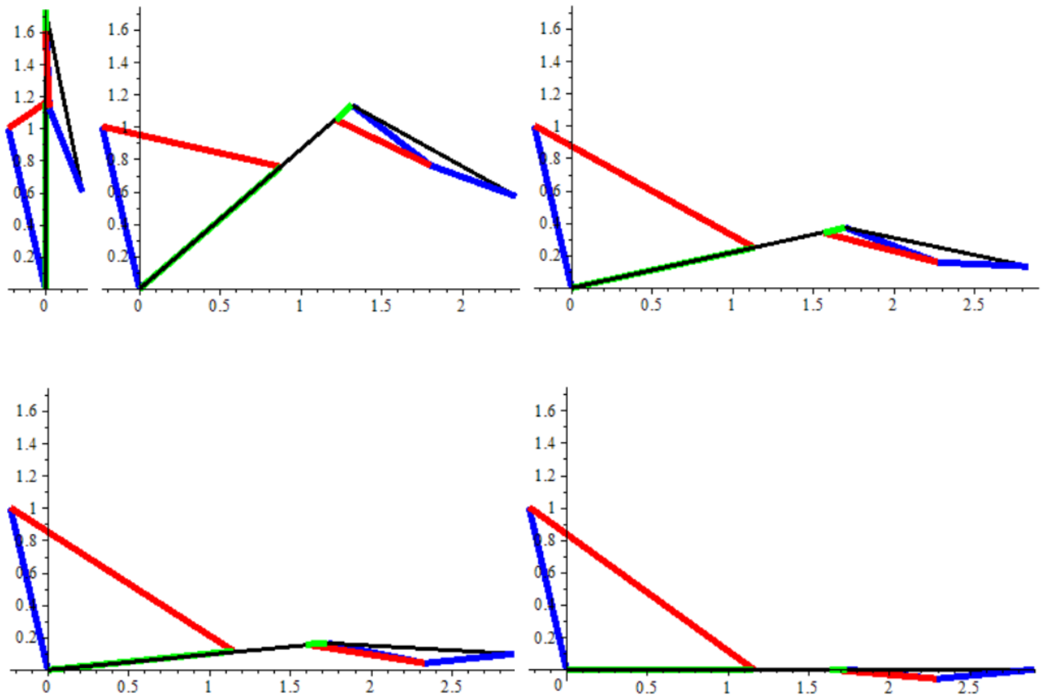


Figure 9: Snapshots of position animation.

The red vectors in Figure 9 represent the winch and linear actuator. The remaining vectors in the position animation are colored to aid with differentiation between one another and are not otherwise relevant.

It was determined that a one second delay would be required so that the second portion of the gangplank does not interfere with the opposite side scaffold during extension and retraction. The final snapshot of Figure 9 shows the linear actuator being fully extended before touching down on the other scaffold when being deployed. Similarly, during gangplank retraction, the linear actuator needs to be delayed until the winch lifts the first portion of the gangplank slightly above the horizontal to prevent interference.

While the electrical circuit portion of the design is outside the scope of this project, a circuit with a simple control switch to activate both the winch motor and linear actuator, with a one second delay, would be necessary given the results of the positional analysis. The delay for both the extension and retraction of the bridge is shown in Figure 10.

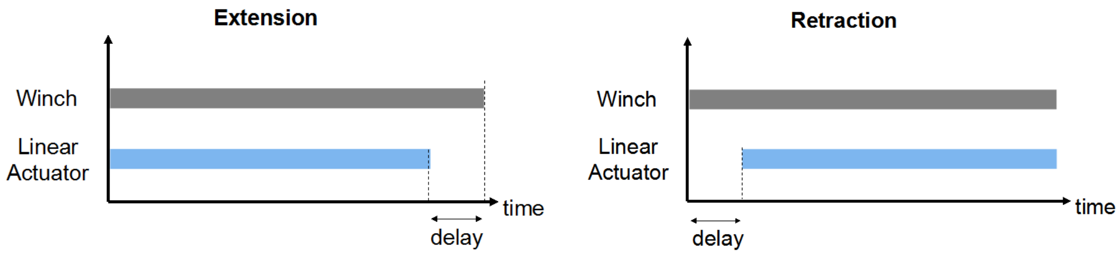


Figure 10: Extension and retraction delay requirements.

3.2.3 VELOCITY PROFILES

A constant velocity profile produces straight line displacement and zero acceleration throughout the transition. The problem with this type of profile arises at the end points, which theoretically result in an instantaneous change in velocity, and thus infinite acceleration [7]. Infinite acceleration would give erroneous results in the force analysis and can be avoided by use of cycloidal curves. The characteristics of a cycloidal curve are shown in Figure 11.

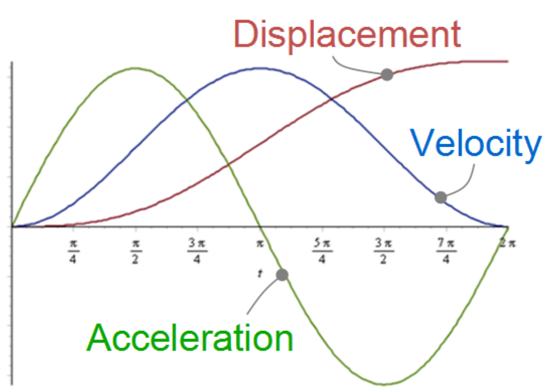


Figure 11: Characteristics of a cycloidal curve.

The input of both the winch and linear actuator use a cycloidal curve over a period of 25 seconds. The time period was chosen based on the speed of a 10” stroke actuator extending its full range at a speed of 0.4”/sec.

3.2.4 FORCE ANALYSIS

The free body diagram for the two portions of the gangplank are shown in Figure 12 and Figure 13. Both figures include the reaction, gravitational, and inertial forces. Note that all directions are assumed and may be evaluated as negative values which implies a reversed direction.

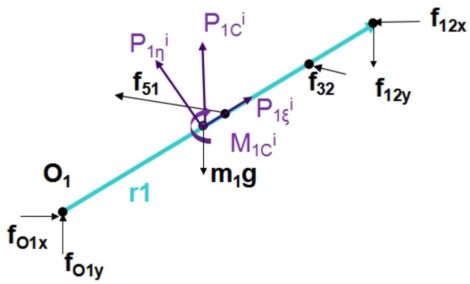


Figure 12: Free body diagram of the first gangplank portion.

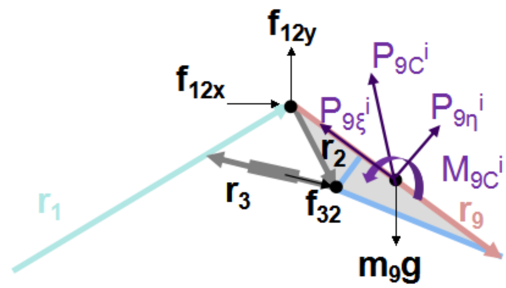


Figure 13: Free body diagram of the second gangplank portion.

The reaction, gravitational, and inertial force components used later for the analysis and selection of components are summarized in TABLE III. The naming convention for force f_{AB} describes a force at the joint between vector R_A and R_B.

TABLE III: FORCE AND MOMENT COMPONENT VARIBLE NAMES

Variable	Description
$\boldsymbol{f_{12}}$	Centre Hinge Force
$\boldsymbol{f_{32}}$	Linear Actuator Force
$\boldsymbol{f_{O1}}$	Pivot Mount Hinge Force
$\boldsymbol{f_{51}}$	Winch Force
$\boldsymbol{P_{51}}$	Winch Power
$\boldsymbol{m_1 g}$	Gravitational Force, Link 1
$\boldsymbol{m_9 g}$	Gravitational Force, Link 9
$\boldsymbol{P^i_{1C}}$	Inertial Force, Link 1
$\boldsymbol{M^i_{1C}}$	Inertial Moment, Link 1
$\boldsymbol{P^i_{9C}}$	Inertial Force, Link 9
$\boldsymbol{M^i_{9C}}$	Inertial Moment, Link 9

Using D'Alembert's Principle, the inertial forces, F_{inert}, and moments, M_{inert}, were determined from the following equations [8],

$$\boldsymbol{F}_{inert} = -m_j \ddot{\boldsymbol{r}}_{cj}$$

$$\boldsymbol{M}_{inert} = -I_j \ddot{\theta}_j$$

where m_j is the mass of the link, $\ddot{r}_{cj}$ is the second derivative of the position vector to the mass center of link j with respect to time, I_j is he moment of inertia of the link, and $\ddot{\theta}_j$ is the second derivative of the angular coordinate of the link j with respect to time. The resulting inertial force and moment for Vector R1 and R9 are shown in Figure 14 and Figure 15, respectively.

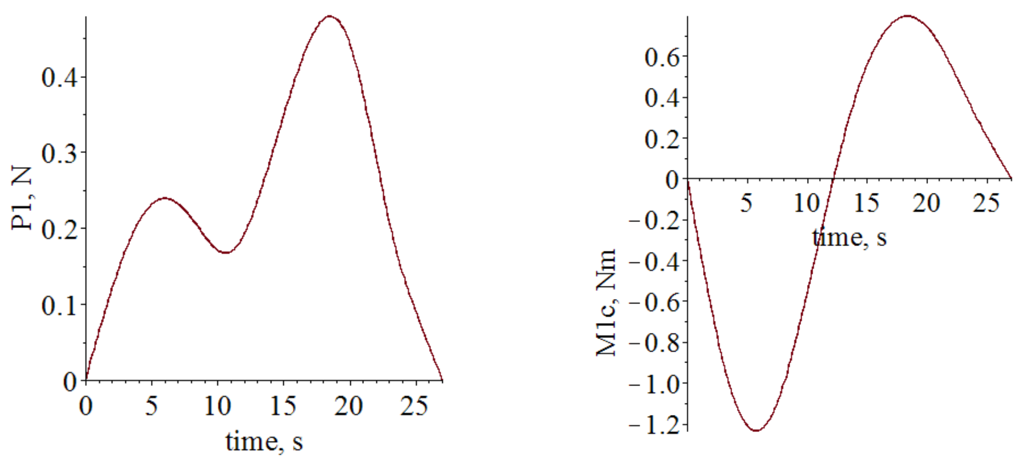


Figure 14: Inertial force P_1 and inertial moment M_{1c} on vector R1.

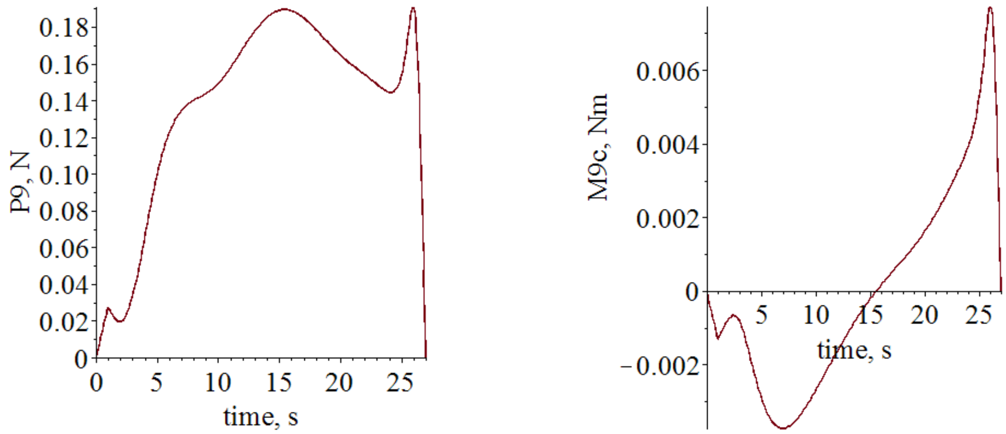


Figure 15: Inertial force P_9 and inertial moment M_{9c} on vector R9.

These figures show that the inertial contributions are minimal due to the large time period and low speed of the mechanism. However, the inertial effects were kept in the analysis in case the speed of the mechanism were to increase.

These inertial forces and moments were then used in the following modified static equilibrium equations.

$$\sum F_x + F_{x,inert} = 0$$

$$\sum F_y + F_{y,inert} = 0$$

$$\sum M + M_{inert} = 0$$

Beginning with the free body diagram of the second gangplank portion, the force at the center hinge was broken into the x-component, f_{12x}, and y-component, f_{12y}. The force in the linear actuator is variable f_{32}. Using D'Alembert's Principle to represent the inertial forces of the system, the three unknown variables were determined by solving the static equilibrium equations. The results are shown in Figure 16 and Figure 17.

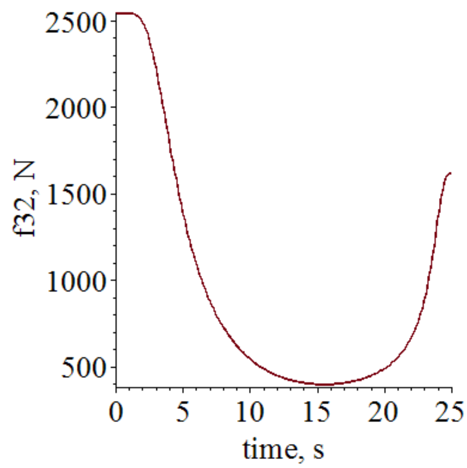


Figure 16: Force in linear actuator.

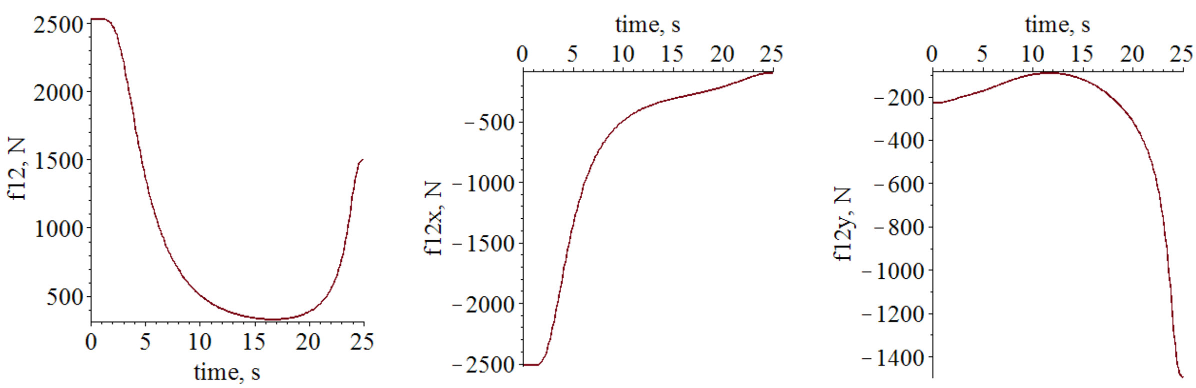


Figure 17: Force at center hinge, f_{12}, and the x and y components.

Given the forces in the center hinge and the linear actuator, the same modified static equilibrium equations were then applied to the first bridge portion. The results of these equations solve for the unknown force in the winch cable, f_{51}, and the force in the pivot mount hinge, f_{O1}. Like the force at the center hinge, the force at the pivot mount hinge was calculated as separate x and y component, f_{O1x} and f_{O1y}, respectively. The forces in the pivot mount at O1 are shown in Figure 18.

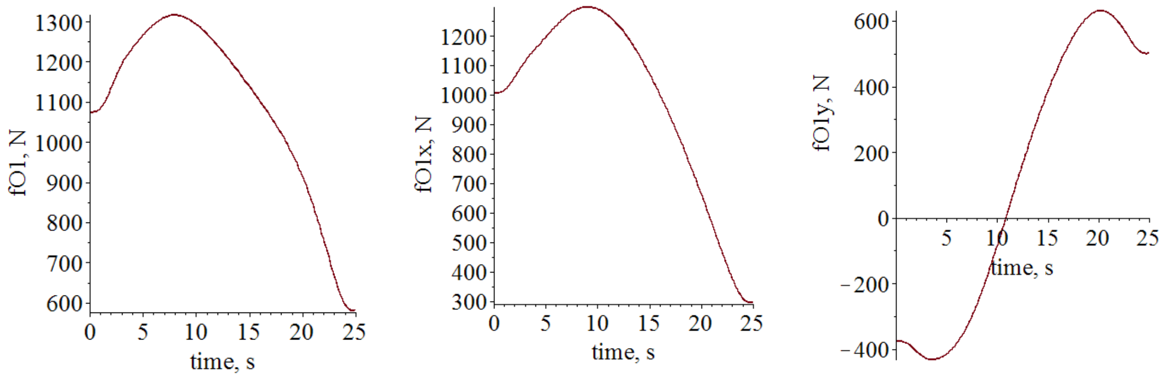


Figure 18: Force in pivot hinge at O1 and the x and y components.

Since power is simply force multiplied by velocity, the force in the winch cable and the rate at which the length of cable decreases was used to determine the power required for the winch motor. The force and power results for the winch cable are shown in and Figure 19.

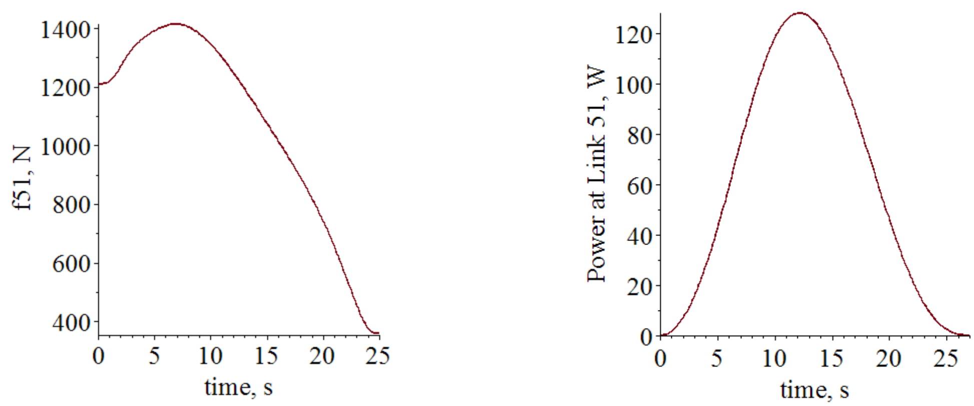


Figure 19: Force in winch cable and power required by winch motor.

The peak force values for each of the critical design components are summarized in TABLE IV.

TABLE IV: SUMMARY OF MAXIMUM FORCE RESULTS

Variable	**Description**	**Maximum (SI)**	**Maximum (lbs)**
$\boldsymbol{f_{12}}$	Center Hinge Force	2528 N	568
$\boldsymbol{f_{32}}$	Linear Actuator Force	2541 N	571
$\boldsymbol{f_{O1}}$	Pivot Mount Hinge Force	1316 N	296
$\boldsymbol{f_{51}}$	Winch Force	1414 N	318
$\boldsymbol{P_{51}}$	Winch Power	140 W	--

The linear actuator force, the winch force, and winch power will be used in Section 3.3 to determine a suitable linear actuator and winch mechanism for purchase. The pivot mount and center hinge forces will be used in Sections 3.6 and 3.7 in the hinge design details. Lastly, the linear actuator force was used to design mount points for the linear actuator in Section 3.8.

3.3 POWER SYSTEMS

Using the linear actuator force, the winch force, and winch power found from the dynamic analysis the two powered elements were specified for purchase: the linear actuator and winch.

With a maximum force requirement of 2541 N, 571 lb, the team's recommended choice of linear actuator is the PA-03 12V DC Linear Actuator from ActuatorZone [9] whose technical specifications are shown in TABLE V. This vendor also has bulk order discounts starting at 10-unit batch orders to further reduce costs if the new design is implemented through out the client's facility.

TABLE V: PA-03 DC LINEAR ACTUATOR FROM ACUATORZONE [9]

Specification	Value
Vendor	ActuatorZone.com
SKU	PA-03
Cost	$133.99 CAD
Stroke	12"
Force	600 lbs
Speed	0.39"/sec
Current: Full Load	7.6 A
Input Voltage	12 VDC

The rated speed on this actuator is 0.39"/sec. which gives a total period of 27 seconds for full retraction for the stroke range determined and listed previously in TABLE II. Given that a one second delay is required, found from the position analysis in Section 3.2.2, the total time for retraction or extension will be 28 seconds. This time period is similar to the initially assumed period of 25 seconds and thus would have similarly negligible inertial forces, leaving the rest of the determined forces near identical.

Each linear actuator requires a power supply and control system. There are many options available, but based on the number of actuators required, the team recommends the PS-10 37.5A power supply available from ActuatorZone for $98.99 CAD which can power up to four actuators at once [10]. The control system for the linear actuators can either be a control box or simple rocker switch. Momentary rocker switches are available from ActuatorZone for $18.00 CAD [11]. The client may however have a more cost effective local supplier they can purchase a polarity reversing DPDT momentary switch.

From TABLE IV, the winch must be able to withstand 1414 N, 318 lb, and must supply 140 W of power for this configuration. A heavy-duty Keeper 2000 lbs. ATV winch, available from Princess Auto, satisfies the loading requirements [12]. This winch is available for $58.88 CAD. Furthermore, the load capacity of the selected winch is more then double what is required; therefore, a single winch will be able to power two gangplanks in a single station if a pully system is implemented. For now, a single winch will be specified to power one gangplank.

A summary of purchased parts required for the power systems and their costs are listed in TABLE VI.

TABLE VI: COST OF POWER SYSTEM COMPONENTS

Item	SKU	Vendor	Cost ($CAD)	Cost per gangplank
DC Linear Actuator [9]	PA-03	ActuatorZone	$133.99	$133.99
Power Supply [10]	PS-10	ActuatorZone	$98.99	$24.75
Rocker Switch Momentary [11]	RC-03	ActuatorZone	$18.00	$18.00
Keeper 2000 lb 12V ATV Winch [12]	8605941	Princess Auto	$58.88	$58.88
Total				$235.62

3.4 UNDER DECK SUPPORT STRUCTURE

Structural sections run beneath the decking of the gangplank to support the weight of the applied load. To meet target specifications for weight, dimensional constrains, and deflection the gangplank supports must be light, strong, and stiff. To achieve these specifications two main factors dictated the design: the material and cross-sectional geometry. To reduce the cost of the overall design, the team decided to restrict material choice to what the client currently has in inventory or could be ordered from their current supplier, Russel Metals.

To estimate the deflection of the bridge, the existence of the center hinge was ignored. This omission created a simply supported beam with a pinned connection on one side and a roller connection on the other. This simplification was justified as the configuration of the center hinge allowed the structural sections to bear against each other, allowing for transfer of moment and shear across the connection maintaining a deformed shape like that of the aforementioned continuous beam. Assuming a load of three 200lb people is distributed over the length of the bridge, the following equation can be used to describe the gangplank's elastic curve where ω is the distributed load, x and L are the distance from the end of the gangplank and length of the gangplank respectively, E is the material stiffness, and I is the geometric stiffness.

$$y = \frac{\omega}{24EI}(x^4 - 2Lx^3 + L^3)$$

As seen in the above equation, both the material and geometric stiffness play a roll in reducing the deflection. Using the same simplification assumption, the stress in the gangplank due to bending can be estimated using the following equation.

$$\sigma = -\frac{My}{I}$$

Breaking down the above equation, the stress due to bending in the bridge is only dependent on the geometry of the cross-section, I, and not the material stiffness.

3.4.1 MATERIAL AND CROSS-SECTION SELECTION

Restricting the choice of materials to what Russel Metals could supply to the client, two materials were considered: aluminum and steel. To keep the weight of the bridge to a reasonable value the decision was made to construct the support structure for the bridge from aluminium, specifically 6061-T6. This aluminum alloy is comparable in strength to mild steels while being a fraction of the weight [13]. In addition, the corrosion resistance of aluminum removes the need to paint the structure to prevent oxidation build up.

Three cross-sections were considered for the under-deck support structure: tubing, structural c-channel, and custom bent sheet aluminum cross-sections. All three options are either already available from the client's material inventory or can be ordered from Russel Metals. The choice was made to avoid bent aluminum supports to reduce issues with cracking and residual stresses from tight bend radii. The weight per unit geometric stiffness was then calculated for the remaining two cross-sections. The results showed that to maintain a similar stiffness, the tubing would be slightly heavier then the structural c-channels. To limit the weight of the design the c-channels were selected.

The deflection and bending stress was estimated for all 6061-T6 structural c-channels available from Russel Metals. Note that all calculations assumed that two c-channels would be used to support the gangplank. Results from the calculation showed that the limiting factor for the gangplank was deflection and not stress. Selecting a 4" tall channel with a flange width of 1.5" inches and a web thickness of 0.25" yielded a maximum estimated stress and deflection of 17.4 MPa and 3.78 mm, respectively, at the center position of the gank plank. Shown in Figure 20 is the selected cross section.

Figure 20: 6061-T6 aluminum c-channel cross-section

3.4.2 FATIGUE STUDY

To ensure that the selected supports will not fail over time, a Goodman type analysis was completed to check for failure due to fatigue. Additionally, a linear damage model was assumed to account for the variable amplitude of load being supported by the structure.

3.4.2.1 FATIGUE LOADING

The three cases being accounted for in this analysis were a load of three people distributed over the length of the gangplank, two people each a third the distance from the two ends of the gangplank, and one person standing at the midpoint. For each case people were assumed to weigh 200 lbs each.

To account for unforeseen loading a worse case assumption was made. This assumption stated that when a person crosses the gangplank, each step will load and unload the under-deck support channels. Using an average step length of 762 mm, 30 inches, it would take approximately 3.5 steps to cross the 2700 mm long gangplank [14]. Multiplying the step count by the estimated number of occurrences for each loading scenario per day, a cycle count was calculated. For this analysis, it was assumed that one person would walk over the bridge every 1 minute, two people every 3 minutes, and three people every 5 minutes through out the course of an 8-hour workday. Using these numbers the cycle counts were calculated and are presented in TABLE VII.

TABLE VII: FATIGUE LIFE CYCLE COUNTS PER 8 HOUR DAY

	Case 1	**Case 2**	**Case 3**
Number of People	3	2	1
Load (lbs)	600	400	200
Cycle Count per Day	42	560	1680

3.4.2.2 GOODMAN ANALYSIS

For most fatigue tests, data is collected using a material specimen subjected to fully reversed bending with zero mean stress. For the team's design the mean stress is not zero therefore a different method from the standard is needed to predict the life of the bridge. Using a modified Goodman analysis both the effects of the oscillating and mean stress can be accounted for using the following equation where S_a is the stress amplitude, S_m is the mean stress, S_{Nf} is the equivalent fully reversed fatigue strength, S_y is the yield strength of the material, MF is the overall fatigue modifying factor, and SF is the safety factor.

$$\frac{S_a}{S_{Nf}}(MF) + \frac{S_m}{S_y}(SF) = 1$$

Solving the left-hand side of the above equation the sum of the two terms must be less then 1 to be deemed a safe design. By taking the right-hand side of the equation as exactly one, the stresses at the threshold of safety were calculated.

Rearranging the previous equation to solve for the fully reversed fatigue strength the following equation was created.

$$S_{Nf} = \frac{MF * S_a}{1 - \frac{S_m}{S_y}(SF)}$$

To calculate the various stresses in the above equation, the maximum stress, when loaded with three people, and minimum stress, when not loaded, were calculated using the bending stress equation presented earlier. This calculation was then repeated for each of the three loading cases. The stress was assumed to oscillate between the minimum and maximum stresses calculated previously. To find the mean stress the maximum and minimum stresses were averaged. The stress amplitude was then calculated by subtracting the mean from the maximum stress.

The safety factor was taken as 4 as per Manitoba heath and safety regulation Section 28 for scaffolding and elevated platforms . The remaining overall modifying factor was calculated using the following equation where the C's represent individual modifying factors presented in TABLE VIII.

$$MF = C_{st}C_R C_S C_T C_C \quad [15]$$

TABLE VIII: FATIGUE MODIFIYNG FACTORS

Factor Symbol	Factor Name	Situation	Value
C_{st}	Type of Stress	Bending	1.0
C_R	Reliability	99.9%	0.75
C_S	Size	4" C-Channel	0.81
C_T	Temperature	Room Temperature	1.0
C_C	Corrosion	Non-Corrosive Environment	1.0

Multiplying the individual modifying factors together the overall modifying factor was calculated as 0.51. Combining all the stresses and factors, the equivalent fully reversed fatigue strength was calculated. The results of the calculation are shown in TABLE IX.

TABLE IX: EQUIVILENT FULLY REVERSED BENDING STRESS

Load Case	1 Person	2 People	3 People
Equivalent Fully Reversed Fatigue Strength	2.94 MPa	4.33 MPa	4.60 MPa

To combine the loading scenarios a linear damage model was adopted. The model states that each element of the loading, the three load cases, takes up some of the life of the gangplank. The effects of the damage and can be accounted for using equation using the following equation which predicts the expected life time of the gangplank.

$$N_f = \frac{1}{\sum_{i=1}^{3} \frac{n_i}{N_i}}$$

In the above equation, the n_i's represent the cycle counts, in shown in TABLE VII, and the N_i's are the number of cycles until fatigue failure at the calculated fully reversed fatigue strength calculated for each load case. Since the calculated stresses were so low the number of cycles for fatigue failure were very large and extended past the end of the fatigue curve for 6061-T6 aluminum shown in Figure 21.

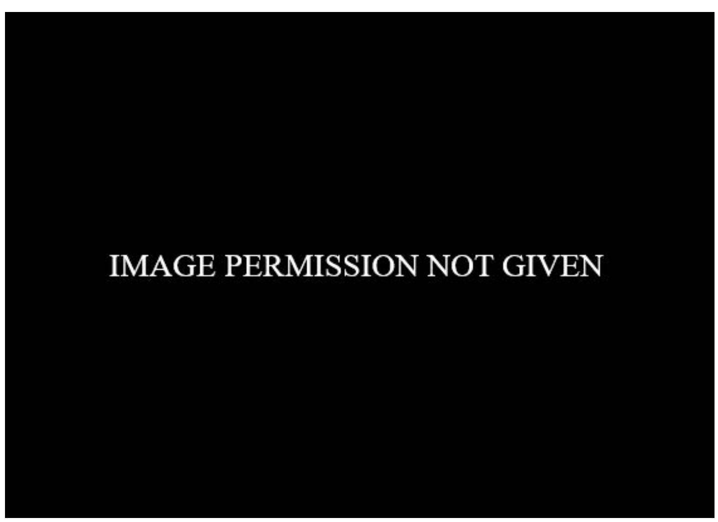


Figure 21: 6061-T6 aluminum fatigue curve [16].

Extrapolating past the end of the curve using a power line of best fit yields the following equations which fit the fully reversed bending line of Figure 21 with an R^2 value of 0.9997.

$$S_{Nf} = 86.818 {N_i}^{-0.106} \qquad \text{Where } S_{Nf} \text{ is in ksi.}$$

OR

$$S_{Nf} = 598.59 {N_i}^{-0.106} \qquad \text{Where } S_{Nf} \text{ is in MPa.}$$

Solving for N_i, the results from the above equations were then plugged back into expected life equation. The calculation yielded an expected life of 2.3×10^{17} days. This result is wildly inaccurate due to the assumption used to extrapolate off the end of Figure 21. Replacing the extrapolated fatigue failure cycle count with the maximum cycle count on Figure 21, 10^8 cycles, the expected life drops to approximately 44000 days which is still well above the 15-year planned life expectancy of the gangplank.

3.5 SAFETY RAILING AND RECIEVER

Manitoba heath and safety regulations require that for all open sides of an elevated work platform or scaffold a guard rail must be present to prevent workers from falling and injuring themselves. This guard rail must cover the length of the platform, be between 0.90 and 1.06 meters tall, have an intermediary rail 0.45 to 0.53 meters from the walking surface, and withstand a minimum load of 900N in any direction. The safety rail consists of two main sections: the railing and the railing receiver. To keep weight down it was decided that aluminum would be an ideal choice for both elements due to its high strength to weight ratio.

3.5.1 SAFETY RAILING

The proposed safety railing design in shown in Figure 22.

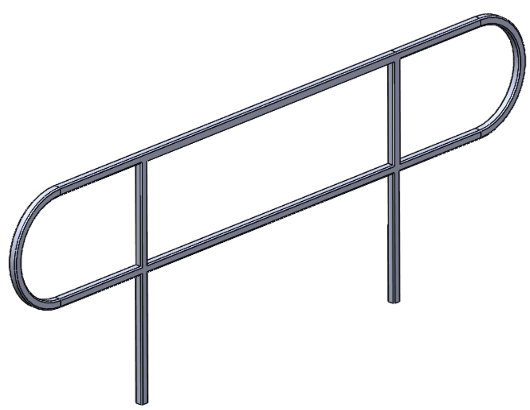


Figure 22: Safety railing.

The safety railing consists of three major parts: the two vertical supports, the looped railing, and intermediate rails. Russel Metals, the client's metal supplier, offers two alloys of aluminum tubing, 6063-T5 and 6061-T6. Both alloys are weldable and therefore are suable for manufacturing the safety rail. As previously mentioned, to be considered safe the railing must withstand a static load of 900N. This load can either be applied to the top or intermediate railing shown in Figure 22. To account for the worst-case scenario the load was applied on the top rail to create the largest moment possible. Additionally, the load was assumed to be applied near the ends of the top rail such that only one of the vertical supports resisted the bending caused by the load.

The analysis of the railing was completed assuming the bottom is fixed such that there is no rotation at the base and reacts like a cantilever beam. The fixturing is supplied by the railing receivers discussed later A free body diagram of the loading is shown in Figure 23.

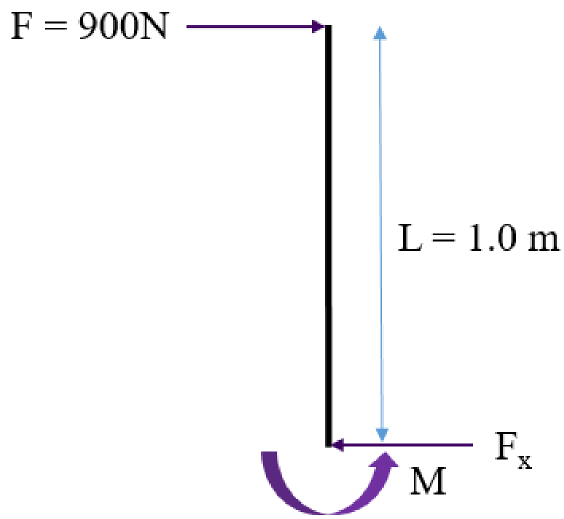


Figure 23: Railing free body diagram.

Setting the top rail 1.0 m from the top of the walking surface, the maximum internal moment in the vertical support, located at the base, can be described using the following equation where M is the internal moment, L is the length, and F is the applied load.

$$M = L * F \quad [17]$$

Using the internal moment, the maximum stress caused by bending in the railing was then calculated using the following equation where σ is the bending stress, c is the maximum distance from the tube's neutral axis to the outside edge, and I is the geometric stiffness of the cross-section.

$$\sigma = \frac{Mc}{I} \quad [17]$$

For a hollow square tube the geometric stiffness was estimated using the following equation where b is the outer dimension of the tube and t is the wall thickness.

$$I = \frac{b^4 - (b-2t)^4}{12}$$

After calculating the stresses in the vertical rail support, the minimum cross section that could support the 900N load while not yielding was selected. The smallest and lightest option was a 1.5”x1.5”x0.120” 6061-T6 aluminum tube. Under the worst case loading the vertical support experienced 194 MPa of bending stress which is lower then 6061-T6’s yield strength of 276 MPa.

3.5.2 SAFETY RAIL RECIEVER

To secure the safety rail to the bridge, two bent aluminum receivers are fastened to the support channels using 4 bolts. The receivers are 4” deep and capped at the bottom to prevent the railing from falling through. The proposed design is shown in Figure 24.

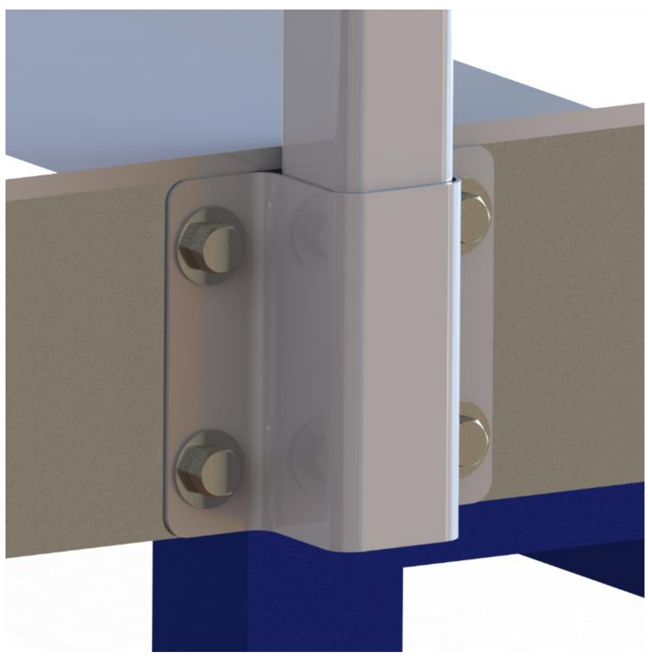


Figure 24: Design of safety railing receiver.

Using the same worse case assumption as the safety rail, the 900 N force pushes on the top of the railing creating an internal moment, M, and a horizontal force, F_x, which is resisted by the receiver. A free body diagram of the loading is shown in Figure 25.

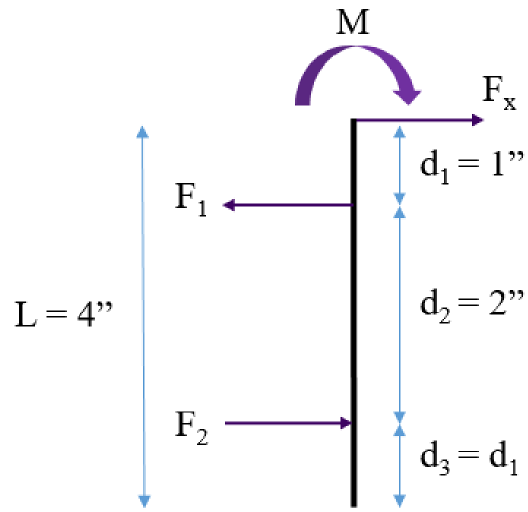


Figure 25: Safety railing receiver free body diagram.

Using the previous calculated force and moment, the clamping force for the upper bolts, F_1, was calculated as 15343 N. Assuming two bolts share the load, the clamping force is divided between them and becomes 7672 N or 1724 lbs each. To maintain a safety factor of two the clamping force was doubled to 3448 lbs for each bolt. For a grade 8 bolt the maximum clamping force is taken as 0.75 times a proof load of 120 ksi. The minimum tensile area was then calculated as 0.0383 in^2. Up sizing to the next largest nominal diameter the bolts became 5/16”-18.

To calculate the torque required to produce the required clamping force the following equation is used where T is the bolt torque, K is a lubrication factor, D is the nominal diameter, and P is the required clamping force.

$$T = KDP \quad [15]$$

To produce enough clamping force the bolts will need to be torqued to a minimum of 215 in-lbs. Additional clamping force will have no negative effects on the design therefore the 5/16”-18 bolts should be torqued to their specified maximum, 295 in-lbs [18].

To ensure the receiver does not fail plastically, the effect of the axial and bending stress were investigated. The receiver was assumed to be rigidly attached to the support channels creating the loading scenario show in in the left diagram of Figure 26. Note that the two diagrams shown in Figure 26 depict equivalent loading scenarios.

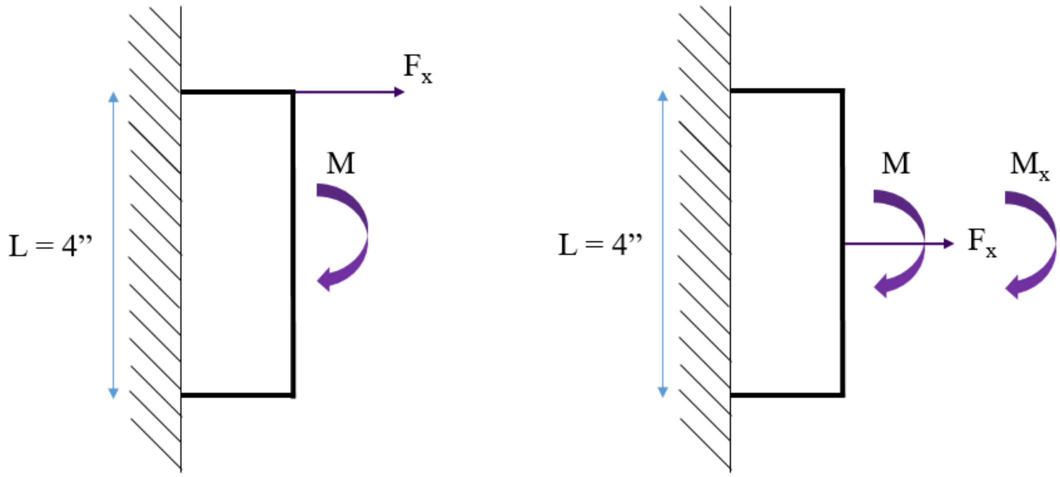


Figure 26: Eccentric loading of railing receiver.

Replacing the eccentric axial force with a force and moment pair the stress in the section can be calculated by combining the effect of each component using the law of superposition. Maintaining a safety factor of two the minimum 6061-T6 sheet thickness was calculated as 0.051”.

Washers were added to prevent tear out of the bolted connections. Using the previously calculated clamping force for each bolt, 7626 N, and a 1” outer diameter washer, the minimum thickness of the channel web and railing receiver, t, were calculated using the following where SF is the safety factor, F is the force, σ_y is the yield strength, and D is the outer diameter of the washer.

$$t = \frac{SF * F}{\sigma_y * \pi * D}$$

Using the above equation, the minimum thickness of sheet aluminum to prevent tear out was calculated as 0.0366”. This minimum is lower then the railing receiver thickness calculated from the loading scenario shown in Figure 26, therefore the thickness will be taken as 0.051”.

3.6 CENTER HINGE

The gangplank was deigned to support the weight of three average adults or approximately 270 kg. The target weight for the structure is 30 kg; therefore, the total weight the gangplank must hold is 300 kg. As mentioned earlier in the report the gangplank is hinged 60% of the way across the span to accommodate the height restrictions imposed by low ceilings. This hinge must support the weight of the structure and the rated load. By locating the hinge on the bottom centerline of the beam and applying a downward force, the cross-sections of the two halves bear against each other allowing for the transfer of moment and shear across the hinge. Considering the gangplank as a beam in bending between two supports the maximum load on the center hinge was calculated as the force necessary to transfer the moment and shear load through this point. Figure 27 illustrate the loading scenario for the gangplank in the deployed position.

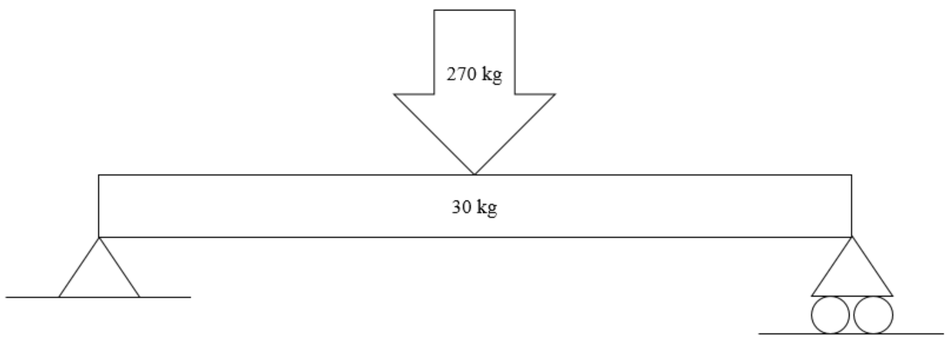


Figure 27: Diagram of support and loading configuration of the gangplank.

As the loading is symmetrical the weight must be equally supported by both ends; therefore, the reaction at each end of the gangplank was considered as half of the rated load. The hinge itself is not located at the center of the gangplank, rather it is located at a distance from the pivot end that is 60% of the total span. Distributing the total load between the two gangplank portions, 60% of the load is supported by the first 60% percent of the gangplank while 40% of the load is taken by the remainder of the gangplank. To determine the load on the hinge a free body diagram of the 40% gangplank portion was created and is shown in Figure 28.

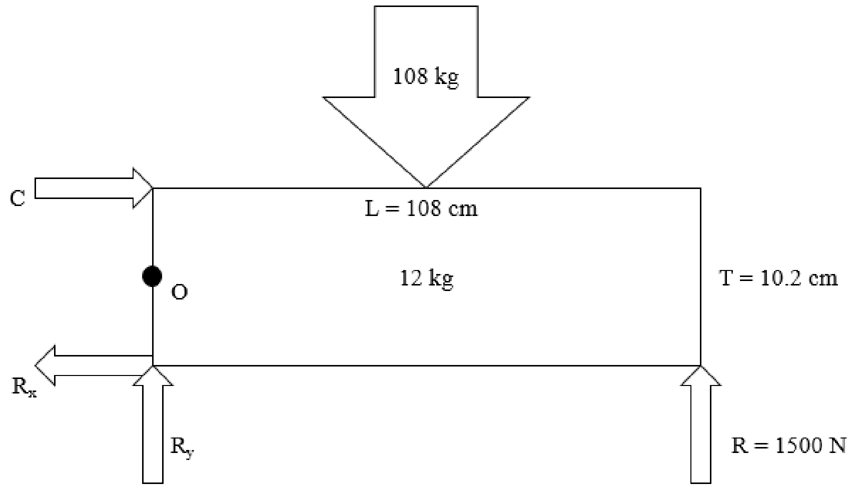


Figure 28: Free body diagram of second gangplank portion.

Static analysis was used to determine the unknowns in Figure 28 which are listed in TABLE X.

TABLE X: RESULTS GANGPLANK SECTION STATIC ANALYSIS

Variable	Description	Value
R_x	Horizontal load on hinge	9646 N
R_y	Vertical load on hinge	302.5 N
R	Combined load on hinge	9651 N
C	Load on bearing surface	9646 N

3.6.1 HINGE PIN SPECIFICATIONS

When considering the pin for the center hinge, the team determined that the loading scenario could be taken to resemble a cantilever beam with a point load on the end equal to that of the combined load on the hinge. Figure 29 illustrates the free body diagram of the hinge pin under the described loading were B is the previously calculated reaction force on the hinge pin

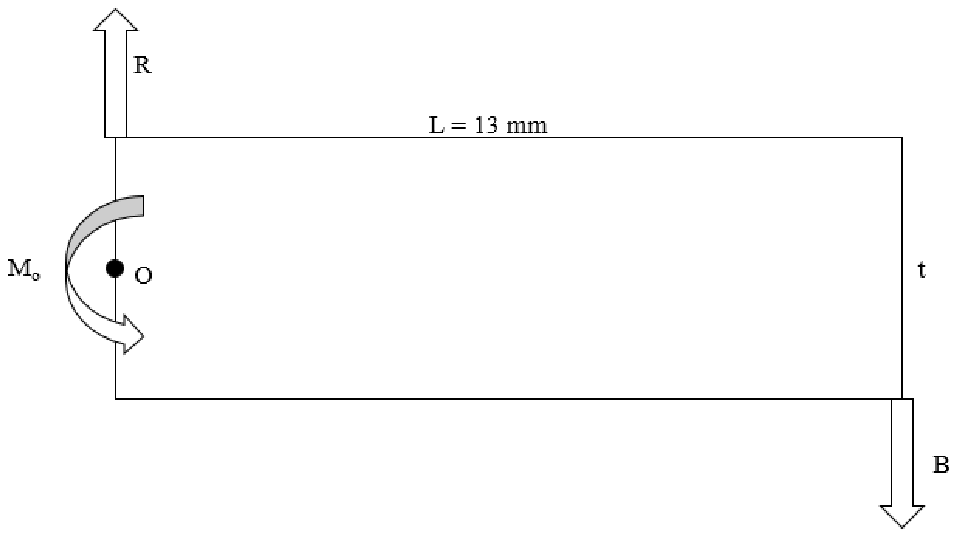


Figure 29: Free body diagram of hinge pin.

Static load analysis of Figure 29 was used to determine the unknown variables which are listed in TABLE XI.

TABLE XI: RESULTS OF HINGE PIN STATIC ANALYSIS

Variable	Description	Value
R	Reaction due to combined load	$R = 9651\ N$
M_p	Moment due to combined load	$M_P = 70.62\ Nm$

To ensure the pin will not fail the specified diameter must be large enough to withstand the combined stress consisting of shear from the load and bending from the moment. To combine the two components to a single scaler value, the von Mises yield criterion was used. Once calculated, this scaler value was then compared to one quarter of the selected material's yield strength as per Manitoba Workplace Health and Safety regulations that specify all platforms used to support people must be able to withstand a minimum of four times the rated load. Comparing the calculated von Mises and allowable stresses, a minimum pin diameter was then calculated which satisfied the required safety factor of four. Additionally, the material for the pin was selected to minimize its diameter while maintaining the required safety factor.

TABLE XII lists the formulae used to calculate the maximum stress in the hinge pin in addition to determining the required diameter and material properties.

TABLE XII: FORMULAE FOR HINGE PIN STRESS CALCULATION [17]

Variable	Equation
Shear Stress due to Reaction	$\sigma_S = \frac{R}{A}$
Tensile Stress due to Bending	$\sigma_B = \frac{Rlc}{I}$
Area of a Circular Cross-section	$A = \frac{\pi d^2}{4}$
Maximum Distance from Neutral Axis	$c = \frac{d}{2}$
Moment of Inertia about Neutral Axis for a Circular Cross-section	$I = \frac{\pi d^4}{64}$
Combined von Mises Stress for one Shear and One Axial Stress	$\sigma_v = \sqrt{\sigma_B{}^2 + 3\sigma_S{}^2}$
Maximum Allowable Stress	$\sigma_{max} = \frac{\sigma_{yield}}{4}$

TABLE XIII lists the input variables required to calculate the combined stress on the hinge pin.

TABLE XIII: INPUT VALUES FOR HINGE PIN STRESS CALCULATIONS

Input	Value
Reaction force	$R = 9651\ N$
Length	$l = 13\ mm$

Using the equations from TABLE XII and inputs from TABLE XIII the minimum diameter and required material were found. TABLE XIV lists the material and size specifications for the hinge pin.

TABLE XIV: HINGE PIN SPECIFICATIONS [19].

Variable	Specification
Material	AISI 1215 (σ_{yield} = 60200 psi)
Size	¾″ Dia. Round Cold Finished Bar

3.6.2 HINGE PLATE SPECIFICATIONS

The previously calculated reaction loads and moment on the hinge pin are transferred to the C-channels by bolted plates which contain a ¾ inch hole. This hole allows two plates, one bolted to each channel, to be connected by the pin to form the center hinge on the gangplank. Figure 30 shows the final hinge plate design.

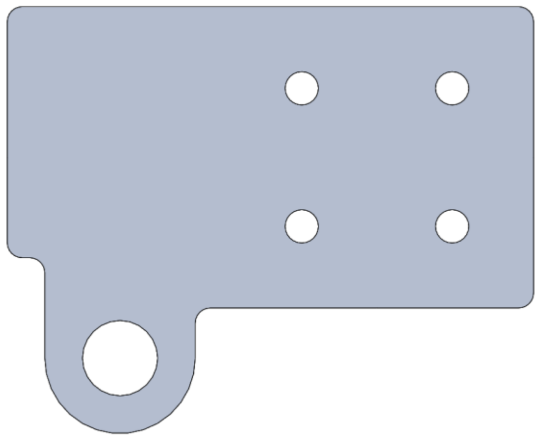


Figure 30: Hinge plate.

These plates were considered to be cantilever beams with a tangential load and a torsion moment applied to them. The load on the pin is transferred to the plate as the tangential load while the bending moment on the pin is transmitted as torsion. The hole in the plate must have enough thickness to resist failure of the hinge plate due to bearing stress from the pin. This bearing stress was found to not be the limiting factor in specifying the plate thickness.

The three stress modes on the hinge plate are: straight shear from the tangential load, shear from the torsion moment from pin, and bending stress from the tangential load. The highest stress from this loading scenario occurs at the mid point of the plate at its base near the neutral axis. This stress is the combined stress from the shear and torsion. The maximum stress from the bending occurs at the edge of the member but is lower than the combined stress of the other two load modes. Figure 31 illustrates the free body diagram of the hinge plate.

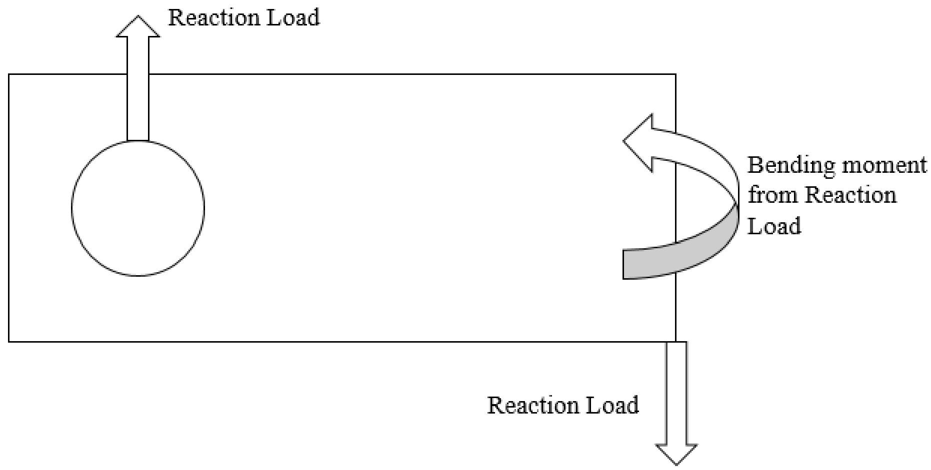


Figure 31: Hinge plate shear and moment reaction due to reaction force from hinge pin.

Figure 32 illustrates the action of torsion on the hinge plate.

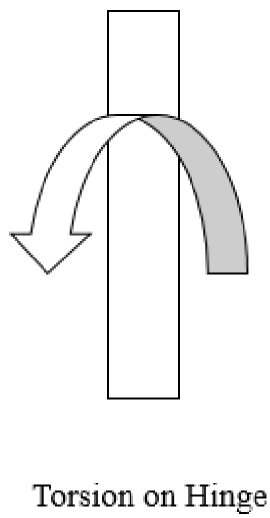


Figure 32: Torsion on hinge plate due to reaction moment on hinge pin.

Torsion on a rectangular cross section does not produce a linear stress profile. According to research the maximum stress in a rectangular cross section under torsion occurs at the centre of the long face of the cross section [20]. The material furthest from the centre carries the lowest strain; therefore, the shear along the short face of the cross section is zero. The formula for calculating maximum shear on a rectangular cross section in torsion contains a modifying factor α. This modifying factor is calculated as a function of the ratio w/h. TABLE XV lists the values of α at specific ratios. Figure 33 illustrates the approximate stress contours within a rectangular cross section in torsion.

TABLE XV: TORSION MODIFYING FACTORS [20]

$\frac{w}{t}$	$\alpha = f(\frac{w}{t})$
1	0.208
1.5	0.231
1.75	0.239
2	0.246
2.5	0.258
3	0.267
4	0.282
6	0.299
8	0.307
10	0.313
∞	0.333

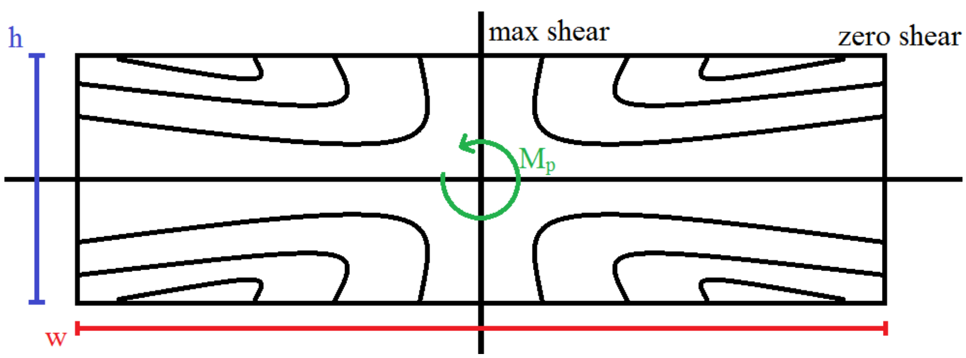


Figure 33: Shear stress contours of a rectangular cross section in torsion [20].

Knowing location of greatest stress in the hinge plate, each components of stress was calculated using the equations shown in TABLE XVI and the input values from TABLE XVII.

TABLE XVI: FORMULAE FOR HINGE PLATE STRESS CALCULATIONS [20]

Description	Formula
Shear Stress due to Reaction Force	$\sigma_s = \frac{R}{A}$
Stress due to Bending	$\sigma_B = \frac{Rlw}{2I}$
Shear Stress due to Torsion	$\sigma_T = \frac{M_p}{\alpha w h^2}$

TABLE XVII: INPUT VALUES FOR HINGE PLATE STRESS CALCULATIONS

Input	Value
Reaction Force	$R = 9651\ N$
Torsion Moment	$M_P = 70.62\ Nm$
Width	$w = 40\ mm$
Safety Factor	SF =4

Comparing the stress in the hinge plate to allowable stress dictated by the material choice and the safety factor, a minimum plate thickness was calculated. Specifications for the hinge plate are shown in TABLE XVIII.

TABLE XVIII: HINGE PLATE MATERIAL SPECIFICATIONS [21].

Variable	Specification
Material	CSA G40.21 44W
Size	½″ Plate

3.6.2.1 HINGE PLATE FASTENERS

To transfer the load on the hinge plate to the C-channels, a bolted connection is used. By tightening the bolts the steel hinge plate is pressed into the aluminum channel creating friction which resists the shear and moment from hinge pin imparting a force on the hinge plate. Figure 34 illustrates the loading scenario present for the interaction between the hinge plate and C-channel interface.

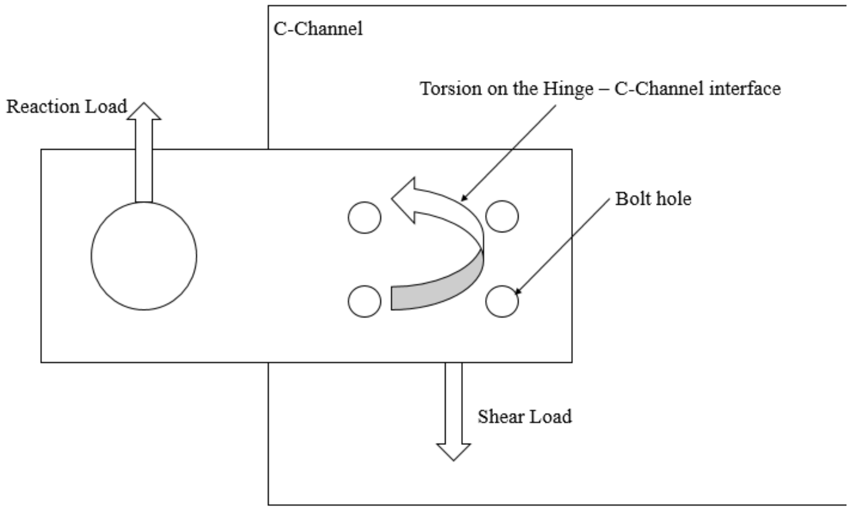


Figure 34: Hinge plate to C-channel interface configuration.

TABLE XIX lists the input values from static analysis used for calculating the load on the hinge plate to C-channel interface.

TABLE XIX: INPUT VALUES FOR HINGE PLATE FASTENER CALCULATIONS

Input	Value
Reaction Load	$R = 9651$ N
Moment from Pin Reaction Load	$M_o = 451$ Nm
Coefficient of static friction between steel and aluminum	$\mu = 0.61$
Number of bolts	$N = 4$

TABLE XX lists the formulae used for calculating the load on the interface between the hinge plate and the C-channel in addition to the load and minimum requirements for the fasteners.

TABLE XX: FORMULAE FOR HINGE PLATE FASTENER CALCULATIONS [15]

Description	Formula
Shear Load	$V = R$
Combined Friction Load	$F_f = V + r_b M_o$
Clamping Load	$F_n = \frac{F_f}{\mu}$
Clamping Load per Bolt	$F_{bolt} = \frac{F_n}{N}$
Minimum Bolt Diameter	$d_{min} = \sqrt{\frac{4\, F_{bolt}}{\pi\, \frac{\sigma_{yeild}}{FS}}}$
Minimum Torque on Bolt	$T = KDF_n$
Proof Strength of SAE Grade 8 Bolts	$\sigma_{yeild} = 130000\, psi$
Safety Factor	$FS = 4$

Using the equations presented in TABLE XX, the minimum bolt diameters were calculated. Results from the calculations are presented in TABLE XXI where the diameter and torque specifications of the bolts are listed.

TABLE XXI: HINGE PLATE FASTENER SPECICFCATIONS [18]

Description	Specification
Bolt size	5/16″-18 SAE Grade 8
Torque	295 in-lbs

3.7 PIVOT PLATE SPECIFICATIONS

As per its name, when the gangplank is being extended or retracted the first portion of bridge pivots around the end of the plate on a hinge. According the dynamic analysis preformed in Section 3.2, the pivot plate experiences its highest load when the gangplank begins to lift from its deployed position. This dynamic load was calculated as 1400 N which is transmitted by an axial force through the length of the plate. Likening the pivot plate to an eccentrically loaded column, the maximum stress governed by buckling and yielding were found. Using the lower of the two maximum stress the required minimum thickness of the plate was calculated. Shown in TABLE XXII and TABLE XXIII are the formulae and inputs used to calculate the minimum thickness of the plate.

TABLE XXII: FORMULAE FOR PIVOT PLATE STRESS CALCULATIONS [17]

Variable	Equation
Secant Stress	$\sigma_{sec} = \frac{P}{A}\left(1 + \frac{ec}{r^2}\sec\left(\frac{l}{2}\sqrt{\frac{P}{EI}}\right)\right)$
Axial Stress	$\sigma_{axial} = \frac{P}{A}$
Maximum Allowable Stress	$\sigma_{max} = \frac{\sigma_{yield}}{4}$
Area of a rectangular cross section	$A = wt$
Moment of Inertia about Neutral Axis for a rectangular cross section	$I = \frac{wt^3}{12}$
Maximum distance from Neutral Axis	$c = \frac{t}{2}$
Eccentricity of load from Neutral Axis	$e = \frac{t}{2}$
Radius of Gyration	$r = \frac{t}{\sqrt{3}}$

TABLE XXIII: INPUT VALUES FOR PIVOT PLATE STRESS CALCULATIONS [15].

Input		Value
Load		$P = 1400\ N$
Width of plate		$w = 61\ cm$
Length of plate from pivot to C-channel		$l = 46\ cm$
6061-T6 aluminum	Yield strength	$\sigma_{yield} = 275\ MPa$
	Modulus of Elasticity	$E = 70\ GPa$

After calculation, the team found that eccentric buckling was the limiting factor in the design and not plastic failure. TABLE XXIV lists the material type and dimensions of the pivot plate found from the calculations.

TABLE XXIV: SPECIFICATIONS OF PIVOT PLATES.

Description	Specifications
6061-T6 Aluminum Plate	0.250″ x 28.000″ x 24.000″

3.7.1 PIVOT PLATE FASTENERS

The maximum load on the fasteners holding the pivot plate to the C-channel was calculated from the combined moment of the vertical reaction force from the hinge, R, and the rated load on the first 60% portion of the gang plank, W. Figure 35 illustrates the static free body diagram of the first portion of the gangplank with fasteners located at points A, B, C, and D.

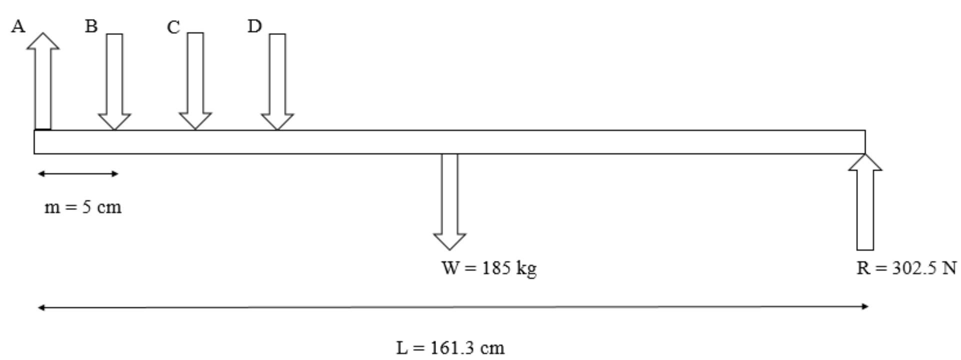


Figure 35: Free body diagram of primary portion of the gangplank.

After calculating each axial force component, fastener D, was found be under the highest tensile force at 2068 N.

The maximum shear load that this bolted connection will experience occurs when the gangplank begins to lift when being retracted. The shear load is equal to the dynamic force on the pivot plate calculated earlier, 1400 N, and will be countered by the friction caused by the bolt's clamping force. Using the inputs listed in TABLE XXV and the equations in TABLE XXVI, the minimum bolt diameters were calculated.

TABLE XXV: INPUTS FOR PIVOT PLATE FASTENER CALCULATIONS

Input	Value
Shear Load	$P = 1400\ N$
Coefficient of static friction between two aluminum faces	$\mu = 1.1$
Number of bolts	$n = 8$
Yield Strength of SAE Grade 8 bolts	$\sigma_{yield} = 130000\ psi$
Safety Factor	$FS = 4$
Maximum Axial Force from Static Analysis	$d = 2068\ N$

TABLE XXVI: FORMULAE FOR PIVOT PLATE FASTENER CALCULATIONS [15]

Description	Formula
Total Clamping load	$F_n = \frac{P}{\mu}$
Clamping load per bolt	$F_C = \frac{F_n}{n}$
Total Required load per bolt	$F_B = F_C + D/2$
Minimum Diameter	$d_{min} = \sqrt{\frac{4\ F_C}{\pi \frac{\sigma_{yield}}{FS}}}$
Minimum Torque	$T = KDF_B$

From the calculations, 8 5/16"-18 bolts would be sufficient to support the gangplank both when loaded statically, creating axial force, and dynamically, creating shear. Each of the bolts must be torqued to a minimum of 149 in-lbs. Adding additional clamping force is beneficial to resisting the shear force on the interface therefore the bolts will be torqued to their recommended maximum specification shown in TABLE XXVII.

TABLE XXVII: SPECIFICATIONS OF PIVOT PLATE FASTENERS [15] [18]

Description	Specification
Bolt size	5/16″-18 SAE Grade 8
Torque	295 in-lbs

Note that the same size and number of bolts are used on the other side of the gangplank to secure another plate to the end. This plate rests on the opposites scaffold when the gangplank is deployed. The same process was used to confirm if the same fastener configuration would work for both the pivot and secondary plate.

3.8 LINEAR ACTUATOR MOUNTS

As mentioned previously, the gangplank is split into two portions to prevent interference with the roof when folding: the larger primary portion and the smaller secondary portion. To facilitate the folding of the secondary gangplank portion, a DC linear actuator is connected at each end to the two gangplank portions via its mount brackets which are attached to cross channels. To transfer the generated force, the cross channels are then bolted to the under deck support structure. Figure 36 shows the linear actuator and its mount assembly.

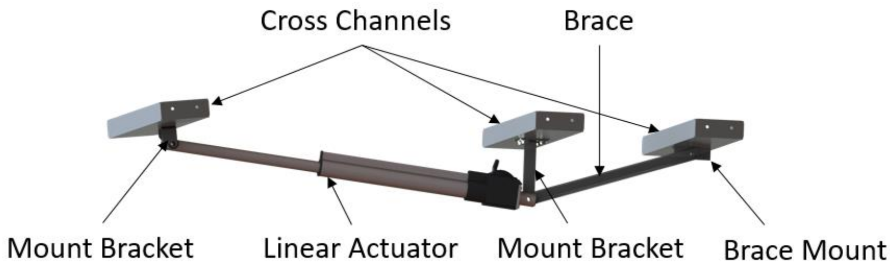


Figure 36: Linear actuator and mounting assembly.

The assembly consists of the linear actuator, two mount brackets, one brace mount, the brace, and three cross channels.

To determine the optimal mount points for the linear actuator the team performed a position analysis as outlined in Section 3.2.2. The dimensions of the mounting points are highlighted in Figure 37 where the points are measured from the bottom of the c-channels. The geometry presented in Figure 37 requires a 12 inch stroke linear actuator which has a length of 48 cm when fully retracted and 78 cm when fully extended. The mount point on the secondary gangplank is moved 9.4 cm down such that the axis of the linear actuator is below the hinge when the gangplank is fully extended to prevent the design from attempting to fold the wrong way causing damage.

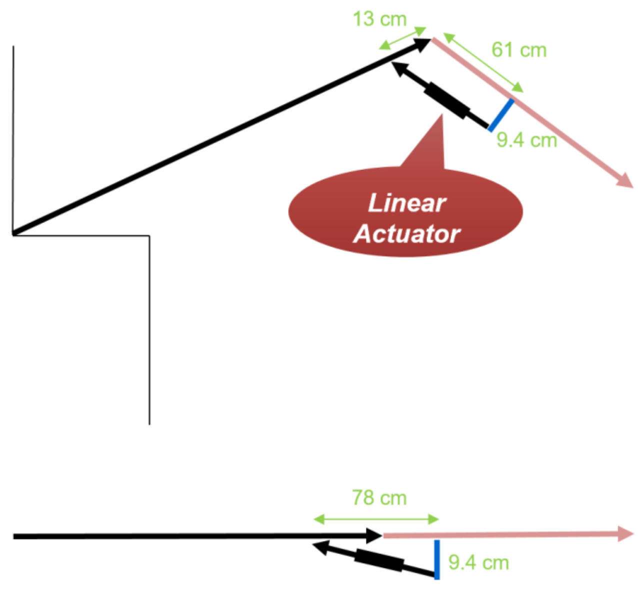


Figure 37: Dimensions of linear actuator mounting points.

ActuatorZone offers mounting brackets for their PA-03 linear actuator. The mounting bracket BRK-02 is made out of iron steel and designed to withstand over 1000lbs of force which is suitable for our application on the first portion of the gangplank. Due to the vertical drop of the second mount point no commercial mounting bracket offered from ActuatorZone satisfied the team's need, therefore a custom bracket was designed. This bracket is made from 1″x1″x0.125″ angle iron and 1/8″ sheet steel which is bolted to a cross channel. To brace the mount a second piece of angle iron is run back to another mount which is also bolted to a cross channel. Figure 38 shows the design of the mounting bracket and bracing for the second bridge mounting point.

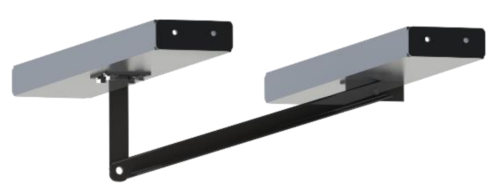


Figure 38: Linear actuator custom designed mounting bracket and brace.

This structure must carry the load of the secondary portion of the gangplank. To analyze the stress in the structure a static analysis was conducted with the force from the linear actuator acting perpendicular to the mount point creating the highest moment and bending stress in the vertical angle iron. Figure 39. The direction of the actuator force, B, was taken as if the actuator were expanding such that the brace will be subject to compression.

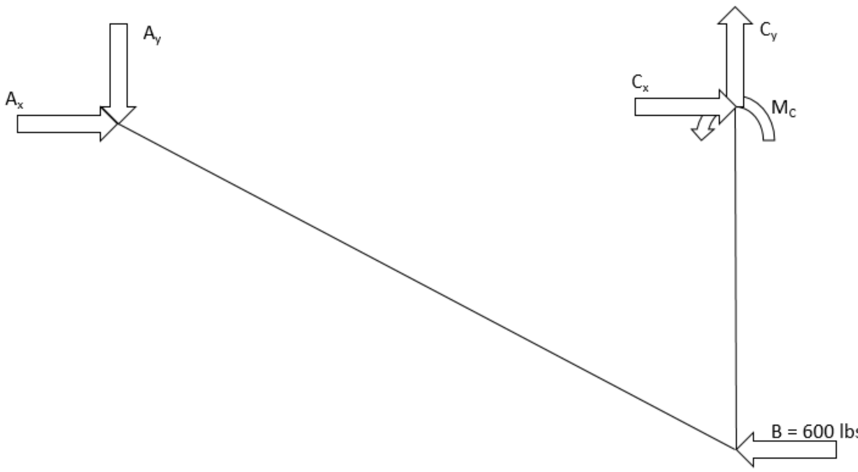


Figure 39: Custom actuator mount and bracing free body diagram.

Due to the fixed support at the top of the vertical section, C, this system is statically indeterminate. Assuming small deflection in the vertical member B-C the system was analyzed. Setting the horizontal deflection in the diagonal brace A-B equal to the deflection in the vertical section the proportion of force being supported by each member was calculated. The formulae for the deflection analysis of the support structure are defined in TABLE XXVIII.

TABLE XXVIII: FORMULAE FOR DEFLECTION ANALYSIS [17]

Description	Formula
Deflection in Vertical Section	$\delta_x = \frac{\overrightarrow{C_x}\overline{BC}}{3EI}$
Horizontal Deflection in Diagonal Brace	$\delta_x = \frac{\vec{A}\,\overline{AB}}{AE}\cos\alpha$
Where: **A = Area** **I = Moment of inertia about neutral axis** **E = Modulus of Elasticity of Mild Steel** **α = Angle of diagonal brace to horizontal**	

Using the equations from TABLE XXVIII and the free body diagram in Figure 39 the forces acing on each member were calculated and are tabulated in TABLE XXIX.

TABLE XXIX: MOUNT POINT ANALYSIS RESULTS

Variable	Value
A_x	2580 N
A_y	827 N
C_x	89 N
Cy	827 N
M_C	12 Nm

3.8.1 WELD ANALYSIS PROCEDURE

The 1″x1″x0.125″ angle iron of the vertical section is welded to a plate which is in turn bolted to the cross member. Analysis of the welded joint utilized the weld line method which uses the following procedure [15]:

1. Identify all loads on the joint.
2. Calculate the magnitude and direction of the force on the weld due to each load.
3. Identify the point of maximum load.
4. Calculate the magnitude and direction of the combined force on the weld at the point of the maximum load.
5. Use TABLE XXX, which defines the maximum allowable force on a weld, to specify a weld material having a maximum allowable force per weld greater than the maximum force on the weld.

TABLE XXX: ALLOWABLE SHEAR STRESS AND FORCE ON WELDS [15]

Base Metal	Electrode/ Filler Alloy	Allowable Shear Stress [psi]	Allowable Force per Inch of Leg [lb/in]
A36 Steel	E60	13600	9600
A36 Steel	E70	15800	11200
6061 Aluminum	4043	5000	3800
6061 Aluminum	5356	7000	5200

6. The minimum required weld leg is calculated using:

$$Minimum\ Required\ Leg = \frac{Maximum\ Force\ on\ Weld}{Allowable\ Force\ on\ weld}$$

7. Using TABLE XXXI specify the appropriate weld leg which greater than the minimum calculated in the previous step.

TABLE XXXI: MINIMUM WELD SIZE FOR THICK PLATES [15]

Plate Thickness [in]	Minimum Leg Size for Fillet Weld [in]
≤ .500	.188
> .500 - .750	.250
> .750 – 1.500	.313
> 1.750 – 2.250	.375
> 2.250 – 6.000	.500
> 6.000	.625

The vertical section of the mount is loaded as a cantilever beam with the force aligned along one of the legs of the angle iron. The weld will experience the highest force at the end of the leg in line with the load application. Figure 40 shows the loading scenario and the location of highest force on the angle iron.

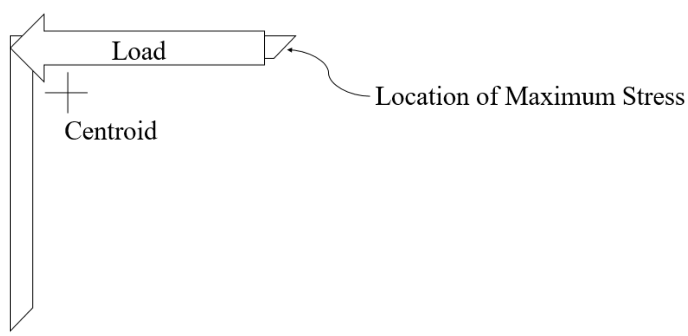


Figure 40: Cross section of angle iron and location of maximum force.

The calculated minimum requirements of the weld, the specified electrode and weld leg size are defined in TABLE XXXII.

TABLE XXXII: CUSTOM ACTUATOR MOUNT WELD SPECIFICATION [15]

Description	Calculated Minimum	Allowable Minimum	Specification
Force per inch of weld leg	419.9 lb/in	9600 lb/in	A36 Steel; E60 Electrode
Leg Size	.109″	.188″	.188″

To analyse the bolts used to hold the actuator mount to the cross channel, the following free body diagram was used. Note that for the worst case loading the actuator is assumed to be extending causing the vertical angle iron to pulled away from the plate. Note that F_1 and F_2 each represent the axial force of two bolts.

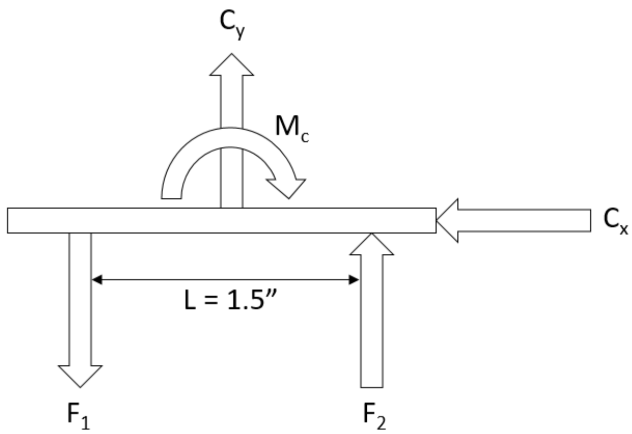


Figure 41: Free body diagram for mount point bolt analysis.

Using the forces and moment from TABLE XXIX, the axial force in the bolts were calculated with F_1 being greater of the two. Dividing the required clamping force by the number of bolts represented by F_1, the minimum bolt diameter was found using the same methods outlined in the previous sections of this report taking into account both the axial and shear forces in the connection

A similar method was used to evaluate the required bolts for the brace mount. Assuming that the actuator is retracting, subjecting the brace to tensile loads, the following free body was used to calculate the minimum bolt diameters.

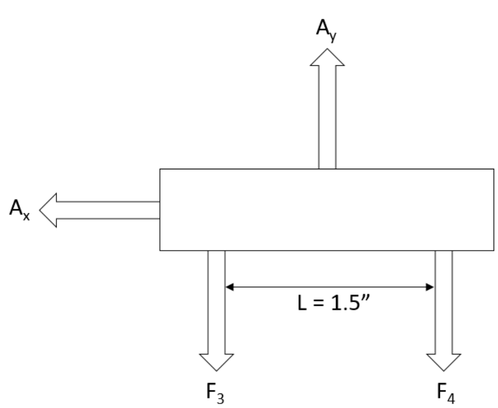


Figure 42: Free body diagram for brace mount bolt calculations.

Again, using the forces from TABLE XXIX the required clamping force to resist the upward pull of the brace and the shear load in the connection was valuated. From these results the minimum bolt diameter and torque were calculated. Shown in Table xxx are the calculated bolt and torque specs for both the actuator mount and the brace mount.

TABLE XXXIII: MOUNT POINT AND BRACE MOUNT FATENER SPECIFICATIONS

Description	Specification
Bolt size	1/4″-20 Grade 8
Torque	143 in-lbs

3.8.2 CROSS MEMBER SUPPORTS

The three mounts for the linear actuator are fixed to three separate cross members used to transfer the force and movement to the second portion of the gangplank. These cross members must be able to carry the load applied from the linear actuator as well as the torsion created by the vertical distance from the mount points to the cross member. The cross members are made of the same aluminum C-channel extrusion as the gangplank under deck supports.

To ensure that the cross members would not yield the effects of each load and the torque, shown in Figure 43, were investigated.

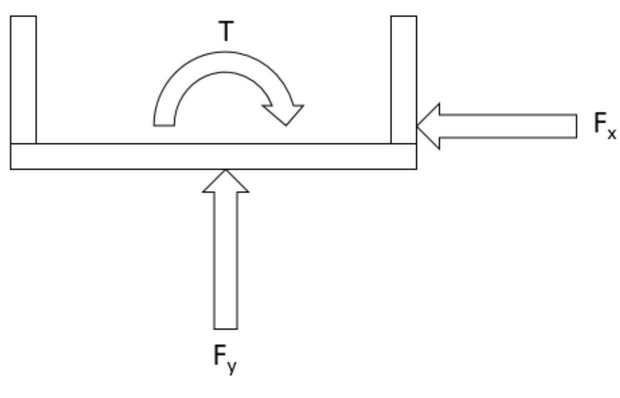


Figure 43: Cross member combined loading.

Each of the loads subjected on the cross members create bending stress and shear. The torsion, created by the horizontal force applied on the mount point and the vertical offset distance from the cross member, induces shear stress in the flanges and web of the C-channel proportional to their respective polar moment of inertia. The stresses from these loads were then combined using the von Mises yield criterion and compared to the yield strength of the aluminum channel. Shown in TABLE XXXIV: FORMULAE FOR CROSS MEMEBR ANALYSIS [17] are the formulae used to calculate the combined stress.

TABLE XXXIV: FORMULAE FOR CROSS MEMEBR ANALYSIS [17]

Description	Formula
Horizontal Bending Moment	$M_h = \dfrac{F_x\, L}{8}$
Horizontal Bending Stress	$\sigma_h = \dfrac{M_h\, c_h}{I_v}$
Vertical Bending Moment	$M_v = \dfrac{F_y\, L}{8}$
Vertical Bending Stress	$\sigma_h = \dfrac{M_v\, c_v}{I_h}$
Torsional Shear Stress, Flange	$\tau_f = \dfrac{(T/2)\,(J_f/J_{tot})}{\alpha_f\, w\, {t_f}^2}$
Torsional Shear Stress, Web	$\tau_f = \dfrac{(T/2)\,(J_w/J_{tot})}{\alpha_w\,(h-2t_f)\,{t_w}^2}$
Where: **L = Length of Cross Member** **c_h and c_v = Maximum Horizontal and Vertical Distance from Neutral Axis** **I_h and I_v = Moment of Inertia About Horizontal and Vertical Axis** **J_f, J_w, and J_{tot} = Polar Moment of Inertia of Channel Flange, Channel Web, and Total** **α_f and α_w = Channel Flange and Web Rectangular Cross Section Polar Moment of Inertia Modifying Factors** **w = Channel Width** **h = Channel Height** **t_f and t_w = Flange and Web Thicknesses**	

Using the results from the above equations the required weld geometry and fasteners were calculated.

To bolt the cross members to the support channels 1/4" aluminum plates are welded on either end of the cross members. The weld between the plates and the cross member were analyzed using the same procedure as the weld used to attach the vertical mount point structure to a steel plate. The minimum requirements and specifications of the weld are defined in TABLE XXXV.

TABLE XXXV: CROSS MEMBER END PLATE WELD SPECFICATIONS [15]

Description	Calculated Minimum	Allowable Minimum	Specification
Force per inch of weld leg	209.8 lb/in	3800 lb/in	6061 Aluminum; 4043 Filler
Leg Size	.053″	.188″	.188″

The plates of the cross members are bolted to the gangplank support channels to transfer the moment of the cylinder to the gangplank. The minimum diameter and torque of the fasteners are specified in the same method as in previous sections and are shown in TABLE XXXVI.

TABLE XXXVI: CROSS MEMBER TO SUPPORT CHANNEL FASTENER SPECIFICATIONS [18]

Description	Specification
Bolt size	1/4″-20 Grade 8
Torque	143 in-lbs

3.9 HARDWARE COSTS

Throughout the various designed components many bolted connections are used. To provide the client with a more accurate cost each purchased nut, bolt, washer, pins, and various accessories must be tabulated. To reduce the number of bolt sizes required all bolts were sized up to one of two standard sizes for this project, 1/4”-20 or 5/16”-18.

Shown in TABLE XXXVII are the final hardware counts and estimated costs for the project. This list includes all items not manufactured by the client or that come with the purchase of other components such as the pins included with the DC linear actuator.

TABLE XXXVII: HARDWARE SUMMARY

Item	Length	Quantity	Cost per Unit	Total Cost	Supplier
Bolts					
1/4”-20 Hex Grade 8	1.125”	12	$0.30	$3.60	McMaster-Carr
1/4”-20 Hex Grade 8	0.750”	8	$0.12	$0.96	McMaster-Carr
5/16”-18 Hex Grade 8	1.250”	16	$0.20	$3.20	McMaster-Carr
5/16”-18 Hex Grade 8	0.875”	8	$0.20	$1.60	McMaster-Carr
5/16”-18 Carriage Grade 8	1.000”	16	$0.80	$12.80	Bolt Depot
Nuts					
1/4”-20 Grade 8	-	20	$0.0322	$0.64	McMaster-Carr
5/16”-18 Grade 8	-	40	$0.0491	$1.96	McMaster-Carr
Washers					
1/4”	-	38	$0.0825	$3.14	McMaster-Carr
5/16”	-	40	$0.118	$4.47	McMaster-Carr
3/4" Nylon	-	4	$0.23	$0.92	Fastenal
Pins					
1/4" Clevis Pin	9/16”	1	$5.26	$5.26	McMaster-Carr
Cotter Pin	-	4	$0.0923	$0.37	McMaster-Carr
Miscellaneous					
Tie Down Ring	-	1	$4.88	$4.88	McMaster-Carr
Bracket (BRK-02)	-	1	$8.50	$8.50	ActuatorZone
Total				$52.30	

3.10 OPERATIONAL STEPS

To operate the new v-fold gangplank design the following steps are taken.

Starting with the gangplank in its deployed state:

1. After completing work on the RV one employee removes the safety railing and sets it aside.
2. ATV winch is then attached to its pick-up point on the first portion of the gangplank.
3. Employee activates the actuator and winch using a single rocker switch causing the gangplank to be retracted.
4. The RV is moved to the new workstation.
5. Employee activates the actuator and winch again using the same rocker switch causing the gangplank to return to its original position.
6. ATV winch is disconnected from the pick-up point.
7. Safety railing is replaced and work resumes.

4 DESIGN SUMMARY

Using the analysis presented in Section 3.0, the final gangplank design was created. The deigned structure folds using a custom designed hinge located approximately 60% the way through the 2700 mm span between the scaffolds. Figure 44 and Figure 45. Shoe the final design

Figure 44: Gangplank in deployed state.

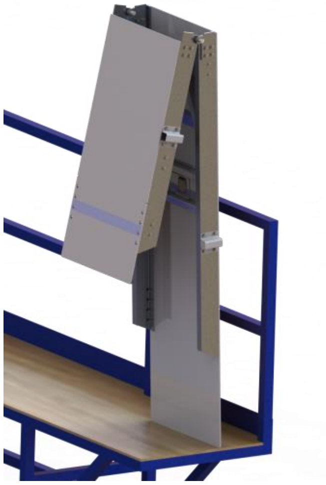


Figure 45: Gangplank in retracted state.

The V-Fold design attaches to one side of the scaffolding and is mounted 460 mm away from the edge. The first and second bridge portions are 1639 and 1061 mm long, respectively. Both portions are supported by an under-deck support structure composed of two 4”x1.5”x0.25” web C-channels designed with a minimum safety factor of 4. The width of the bridge is 609.6 mm and the folded configuration has a height of 2096 mm.

The power system is comprised of a 2000 lb. ATV winch attached to the scaffold and the first portion of the gangplank. A 600 lb. 12” stroke DC linear actuator is used to extend the second portion of the gangplank during deployment with a one second delay between the two.

To retract the gangplank, first the safety rail is removed. Next the winch hook is attached to the pick-up point on the first portion of the bridge after which all workers dismount from scaffold. A switch is pressed and the gangplank is folded into its storage position. Deployment is the done by reversing these steps.

This V-fold Gangplank design was able to meet the critical target specifications listed previously in TABLE I. The final design specifications are listed in TABLE XXXVIII. Note that some target specifications are labeled future work. These elements will require future investigation by the client to confirm the proposed design meets the target specification or is an area where the client can improve upon the deign during subsequent design updates.

TABLE XXXVIII: FINAL DESIGN TECHNICAL SPECIFICATION COMPAIRISON

Functional Requirement	Target Specification	V-Fold Design
Weight	<35 kg	~45 kg Without Railing
Aesthetically Pleasing	Subj. 5/5	Subj. 5/5
Rated Load	250 kg	270 kg
Deployment Time	< 30 sec	28 sec
Operation Force	90 N	Rocker Switch ~ 3.1 N [22]
Centre of Gravity Remains Within Scaffold	Yes	Future Work
Height of Fall Prevention Rail	Top Rail: 0.9 to 1.06 m Intermediate Rail: 0.45 to 0.53 m	1.0 m 0.5 m
Length	2.48 to 2.71 m Dependent on Workstation	2.71 m – Design can be scaled
Width	≥ 0.5 m	0.609 m (24")
Toe Board Height	≥ 0.125 m	0.125 m
Unit Lifetime	15 years	Future Work
Midpoint Deflection	< 0.01 m	0.0038 m
Percent Manufactured in House	100%	100%
Cost of Parts	$200	$235.62
Number of People for Operation	1 person	1 person
Clearance Gap with RV	< 0.3 m	Future work
Station Length Adjustment	±0.07 m	0
Height of Platform from Ground	1.715 to 1.87 m	Matches Scaffold
Undeployed Gap Width	2.1 m	2.1 m
Operation Noise level	80 dB	<45 dB (Actuator)

5 DESIGN FAILURE MODES EFFECT ANALYSIS

Design Failure Modes Effects Analysis, DFMEA, was used to identify vulnerable components of the design and opportunities for improvement. This method of analysis considers components of a design and identifies potential failure modes of each component. Each failure mode is rated on a scale of 1-10 in the following categories: Frequency of Occurrence (F), Severity of Impact (S), and Probability of Detection (D). Failure modes with a low frequency of occurrence are rated 1 and a high frequency rated 10. Failure modes with a low severity of impact are rated 1 and a high severity rated 10. Failure modes with a high probability of detection prior to failure are rated 1 and a low probability rated 10. The product of the Frequency of Occurrence rating, the Severity of Impact rating, and the Probability of Detection rating is defined as the Risk Priority Number (RPN) thus:

$$RPN = F \times S \times D$$

The RPN has a range of 1-1000. For this analysis, all failure modes with an RPN $\geq$ 100 will be addressed. This is an analysis of a design and not a continuous process; therefore, the probability of detection of failure is the probability of detection before failure occurs rather than the probability of detection of defective products. A maintenance and inspection schedule takes the place of continuous quality monitoring. Shown in TABLE XXXIX is the completed DFMEA.

TABLE XXXIX: DFMEA ANALYSIS MATRIX

Function	Potential Failure Mode	Impact of Failure	Severity of Impact (1-10)	Causes of Failure	Frequency of Occurrence (1-10)	Detection Mode	Probability of Detection (10-1)	Risk Priority Number
Support Weight of workers and equipment	Center Hinge -beam fastener failure	Bridge collapse	10	Fasteners yielding or loosening	3	Inspection of fastener integrity and joint integrity	6	180
	Center Hinge plate failure	Bridge collapse	10	Crack in plate, wear of pin hole, sudden impact over rated load	2	Inspection of hinge for cracks or wear	4	80
	Bridge beam failure	Bridge collapse	10	Crack in beam, foreign damage, tampering	2	Inspection of beams	4	80
	Pivot plate failure	Bridge collapse	10	Buckling of plate during retraction	1	Observation during retraction	10	100
	Pivot Plate - Bridge beam fastener failure	Bridge collapse	10	Fasteners yielding or loosening	3	Inspection of fastener integrity and joint integrity	6	180
	Bridge-scaffold interface failure	Bridge collapse	10	Cracking, wear, or fatigue of pivot hinge	1	Inspection of pivot hinge	5	50
	Hinge pin failure	Bridge collapse	10	Wear or crack in hinge pin	2	Inspection of hinge pin	5	100
Raise and lower gangplank	Loss of power to power system	Inability to move gangplank automatically	5	Power interruption	1	Observation during retraction	10	50
	Linkage or cable failure	Bridge collapse	10	Wear or damage to linkage or cable	1	Inspection of cables and linkage	4	40
	Bridge-link interface failure	Bridge collapse	10	Wear, or over loading of hook	1	Inspection of linkage integrity	4	40
	Power system infrastructure failure (pulleys etc.)	Inability to move gangplank automatically	5	Wear, damage or improper loading of pulleys	3	Inspection of pulleys	4	60
Fold center hinge	Hinge actuator failure to contract	Inability to fully retract gangplank	3	Internal failure of actuator	1	Observation during retraction	10	30
	Hinge actuator failure to extend	Inability to deploy gangplank	8	Internal failure of actuator	1	Observation during deployment	10	80
	Hinge actuator fracture	Collapse of secondary portion	9	Wear or damage to actuator	1	Inspection of Actuator	4	36
	Actuator mount point failure	Collapse of secondary portion	9	Wear, damage, or overloading of secondary portion of gangplank	2	Inspection of mount point	4	72
Prevent workers and equipment from falling	Safety Rail failure	Workers fall off gangplank or refuse to work on gangplank	8	Damage to safety rail	1	Inspection of Safety Rail	10	80
	Toe board failure	Tools roll off gangplank	5	Damage to toe board	1	Inspection of toe board	10	50

5.1 CORRECTIVE ACTIONS

Based on the RPN ≥ 100 there are four high risk failure modes. TABLE XL lists the failure modes with their respective RPN values and corrective actions.

TABLE XL: FAILURE MODES AND CORECTIVE ACTIONS

Failure Mode	RPN	Action Taken
Fasteners on the hinge loosening or yielding	180	Specify SAE Grade 8 fasteners and add locking device such as Loctite or lock nuts.
Buckling of the pivot plate	100	Specify thickness of plate well in excess of required thickness to withstand buckling load.
Pivot plate fasteners loosening or yielding	180	Specify SAE Grade 8 fasteners and add locking device such as Loctite or lock nuts.
Failure of the hinge pin due to wear or cracking	100	Specify diameter of pin to a Safety Factor of 4.

5.2 RECOMENDATIONS

In addition to the specified actions taken in the design it is recommended that the system be fully inspected on a monthly basis for any wear, cracks, or warping. Special attention should be paid to the hinge pin, fasteners, and pivot plate.

6 DRAWINGS AND ASSEMBILES

To provide instructions on how the V-Fold design would be manufactured and assembled, preliminary engineering drawings have been created to facilitate the process. Shown in TABLE XLI is the bill of materials which includes the main assembly, the sub-assemblies, and individual components. Purchased parts are included in this bill of materials but drawings are not attached. Instead, the purchased SKU number will be given for reference. Drawings for the manufactured parts and assemblies presented following TABLE XLI.

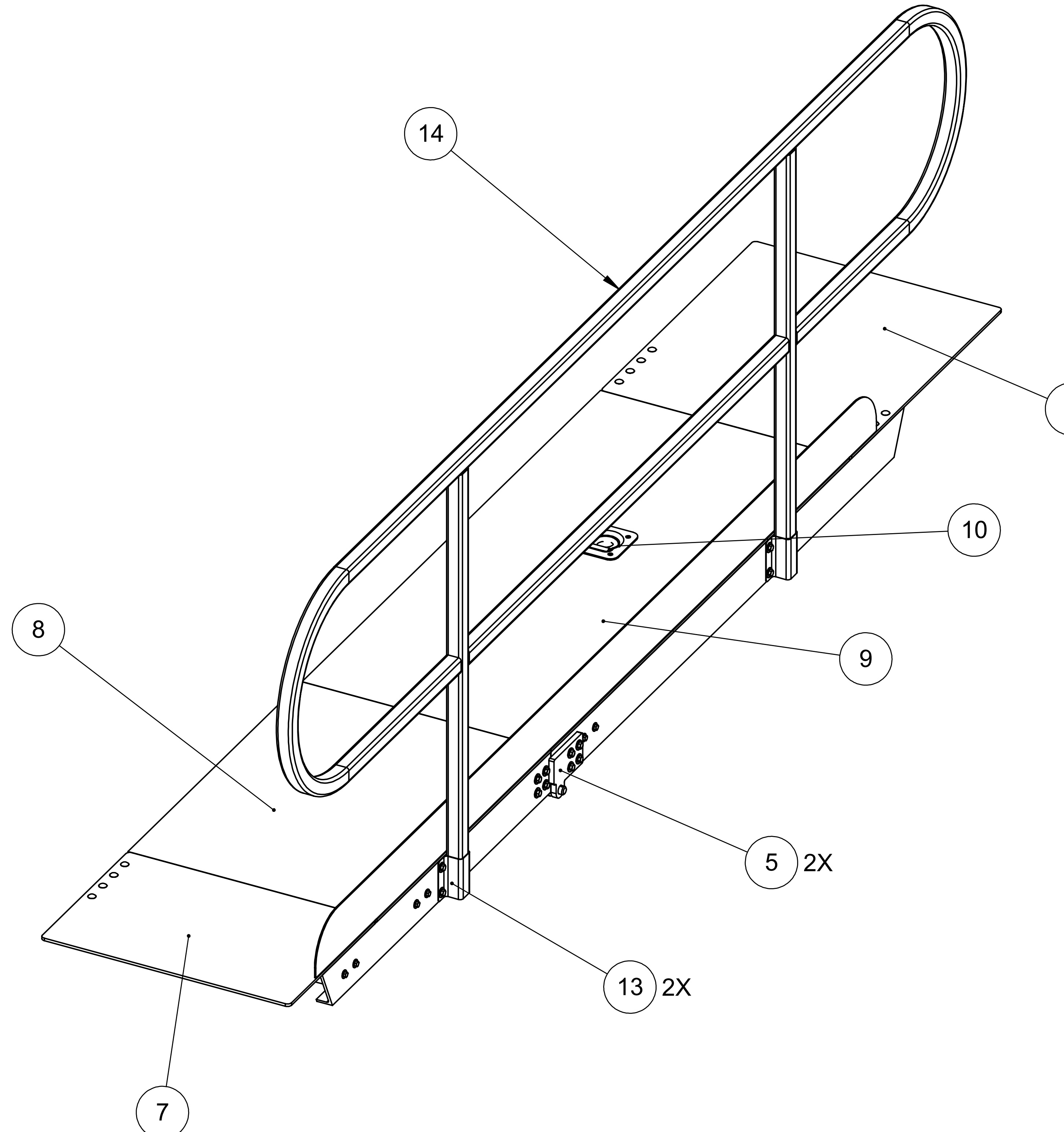
TABLE XLI: BILL OF MATERIALS

LVL	Part #	Description	Parent #	Stock Material	Qty
0	**A01**	**Main Assembly**	**N/A**	**N/A**	1
1	**SR1-0**	**Safety Rail Assembly**	**A01**	**N/A**	**1**
2	SR1-1	Railing Curved	SR1-0	38 X 38 X .120 TUBE 6061-T6 ALUMINUM	1
2	SR1-2	Railing Inter	SR1-0		1
2	SR1-3	Railing Vertical	SR1-0		2
2	SR1-4	Toe board	SR1-0	0.125” SHEET 6061-T6 ALUM	1
2	**SR2-0**	**Railing Receiver Assembly**	**SR1-0**	**N/A**	**2**
3	SR2-1	Railing Receiver	SR2-0	16GA 6061-T6 ALUMINUM	1
3	SR2-2	Bottom Receiver Plate	SR2-0		1
3	N/A	5/16 Flat Washer	SR2-0	Purchased	4
3	N/A	Grade 8 Hex Bolt 5/16-18X.875	SR2-0	Purchased	4
3	N/A	Grade 8 Hex Nut 5/16-18	SR2-0	Purchased	4
1	**CH1-0**	**Center Hinge Assembly**	**A01**	**N/A**	**2**
2	CH1-1	Plate, Hinge	CH1-0	CSA G40.21 50W	2
2	H1	Washer 3/4” Screw, 0.125” THK	CH1-0	NYLON	2
2	CH1-2	Pin & cotter	CH1-0	AISI 1215 STEEL ROD 0.75 dia	1
1	**n/a**	**Deck Support (no drawing)**			1

LVL	Part #	Description	Parent #	Stock Material	Qty
2	DS1-1	Plate, Mount	A01	0.250 SHEET 6061-T6 ALUMINUM	1
2	DS1-2	Plate, Support	A01		1
2	DS1-3	Channel, Long, left	A01	4 X 1.5 X .25 WEB C-CHANNEL 6061-T6 ALUMINUM	1
2	DS1-4	Channel, Long, right	A01		1
2	DS1-5	Channel, Short, left	A01		1
2	DS1-6	Channel, Short, right	A01		1
2	DS1-7	Wood, Large	A01	PLYWOOD	1
2	DS1-8	Wood, Small	A01	PLYWOOD	1
1	**LA1-0**	**Mount Assembly, Motor Side**	**A01**	**N/A**	**1**
2	LA1-1-0	Mount Pickup, Motor Side	LA1-0	N/A	1
3	LA1-1-1	Cross Channel Brace, Motor Side	LA1-1-0	4 X 1.5 X .25 WEB C-CHANNEL	1
3	LA1-1-2-0	Vertical Mount Assy	LA1-1-0	N/A	1
4	LA1-1-2-1	Vertical Mount	LA1-1-2-0	1” angle iron ASTM A36 STEEL	1
4	LA1-1-2-2	Base Plate	LA1-1-2-0	11GA CSA G40.21 50W	1
3	EP-1	End Plate	LA1-1-0	.25 IN. 6061-T6 Aluminum	2
3	N/A	¼ in. Flat Washer	LA1-1-0	Purchased	8
3	N/A	Grade 8 Hex Bolt 1/4-20X1	LA1-1-0	Purchased	4
3	N/A	Grade 8 Hex Nut 1/4-20	LA1-1-0	Purchased	4
2	LA1-2-0	Mount, Reinforcement	LA1-0	N/A	1
3	LA1-2-1	Cross Channel Brace Reinforcement	LA1-2-0	4 X 1.5 X .25 WEB C-CHANNEL	1
3	LA1-2-2	Brace	LA1-2-0	1” angle iron ASTM A36 STEEL	1
3	EP-1	End Plate	LA1-2-0	.25 IN. 6061-T6 Aluminum	2

LVL	Part #	Description	Parent #	Stock Material	Qty
3	N/A	¼ in. Flat Washer	LA1-2-0	Purchased	4
3	N/A	Grade 8 Hex Bolt 1/4-20X1	LA1-2-0	Purchased	2
3	N/A	Grade 8 Hex Nut 1/4-20	LA1-2-0	Purchased	2
2	LA1-3	Angle Brace	LA1-0	1" angle iron ASTM A36 STEEL	1
2	N/A	½ in. Clevis Pin	LA1-0	Purchased	1
1	**LA2-0**	**Mount Pivot Side**	**A01**	**N/A**	**1**
2	LA2-1	Cross Channel Brace Pivot Side	LA2-0	4 X 1.5 X .25 WEB C-CHANNEL	1
2	BRK-02	Mount bracket - ActuatorZone	LA2-0	Purchased	1
2	EP-1	End Plate	LA2-0	.25 IN. 6061-T6 Aluminum	2
3	N/A	¼ in. Flat Washer	LA2-0	Purchased	2
3	N/A	Grade 8 Hex Bolt 1/4-20X1	LA2-0	Purchased	2
3	N/A	Grade 8 Hex Nut 1/4-20	LA2-0	Purchased	2
1	**PA-03**	**DC Linear Actuator 12" Stroke 600 lb**	**A01**	**Purchased**	**1**
1	**PS-10**	**Power Supply**	**LA1-0**	**Purchased**	**1**
	N/A	**Winch Mount (no drawing)**	A01		**1**
1	8304T130	Recessed Tie-Down Ring	A01	Purchased	1
1	WM1-1	Bracket	A01	.125" THK SHEET ALUMINUM 6061-T6	1
1	WM1-2	Brace	A01	38 X 38 X .120 TUBE 6061-T6 ALUMINUM	2
	8605941	2000 lb ATV Winch Motor	ND	Purchased	1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	DS1-3	CHANNEL, LONG, LEFT	1
2	DS1-4	CHANNEL, LONG, RIGHT	1
3	DS1-5	CHANNEL, SHORT, LEFT	1
4	DS1-6	CHANNEL, SHORT, RIGHT	1
5	CH1-0	ASSY - CENTRE HINGE	2
6	DS1-1	PLATE, MOUNT	1
7	DS1-2	PLATE, SUPPORT	1
8	DS1-8	WOOD, SMALL	1
9	DS1-7	WOOD, LARGE	1
10	8304T130	Recessed Tie-Down Ring, McMaster-Carr	1
11	WM1-1	TIE-DOWN BRACKET	1
12	WM1-2	BRACE	2
13	SR2-0	ASSY - RAILING RECEIVER	2
14	SR1-0	ASSY - SAFETY RAIL	1
15	RHBOLT 0.3125-18x1x1-N	GRADE 8 ROUND HEAD BOLT 5/16-18X1	16
16	HNUT 0.3125-18-D-N	GRADE 8 HEX NUT 5/16-18	32
17	LA1-0	ASSY - MOUNT, MOTOR SIDE	1
18	LA2-0	ASSY - MOUNT, PIVOT SIDE	1
19	HBOLT 0.3125-18x1.25x0.875-N	GRADE 8 HEX BOLT 1/4-20X1.25	16
20	PA-03	DC Linear Actuator 12" Stroke, 600 lbs	1
21	Preferred Wide FW 0.3125	WIDE FLAT WASHER 5/16"	32
22	Preferred Wide FW 0.25	WIDE FLAT WASHER 1/4"	24
23	HNUT 0.2500-20-D-N	HEX NUT 1/4-20	12
24	HBOLT 0.2500-20x1.125x0.75-N	HEX BOLT 1/4-20X1.125	12



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BOLT TORQUE
FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE

1/4-20"	143 IN-LBS
5/16"	195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

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MATERIAL

FINISH NO PLATING REQUIRED

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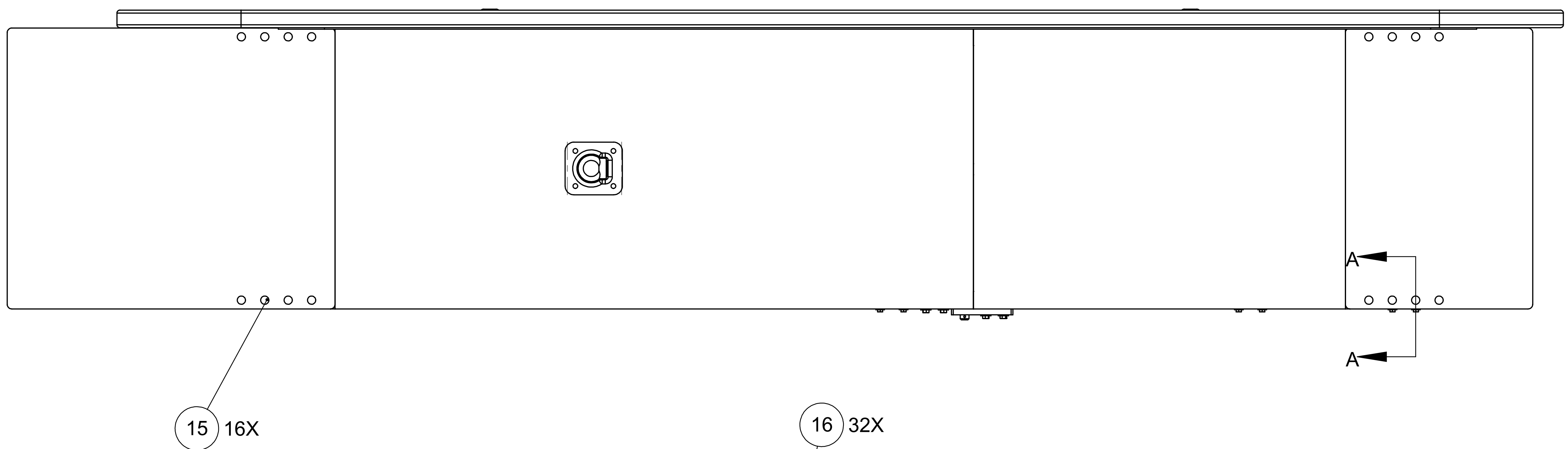

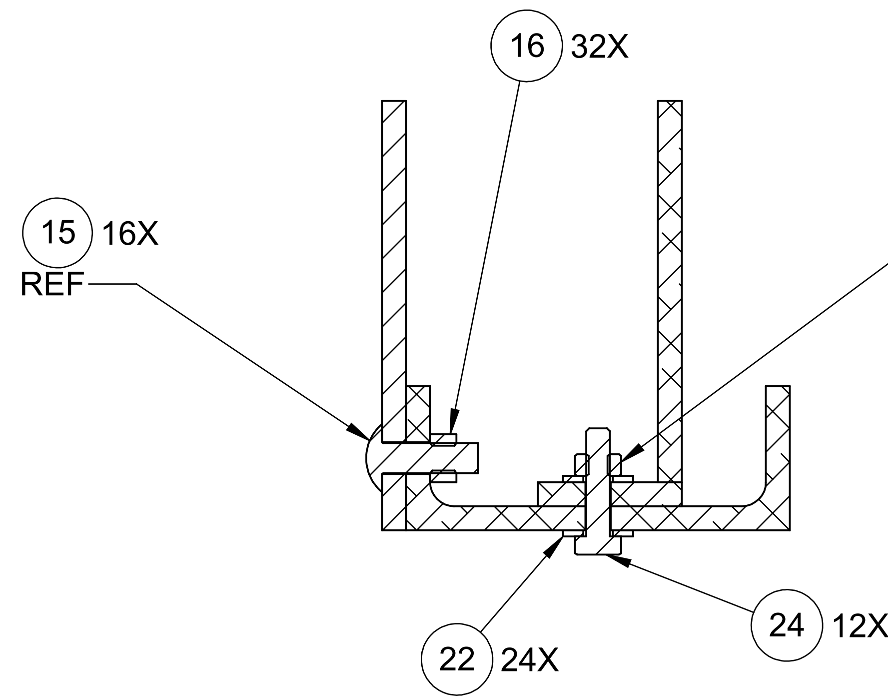

SECTION A-A
SCALE 1 : 2

BOLT TORQUE
FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE

1/4-20"	143 IN-LBS
5/16"	195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

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MATERIAL

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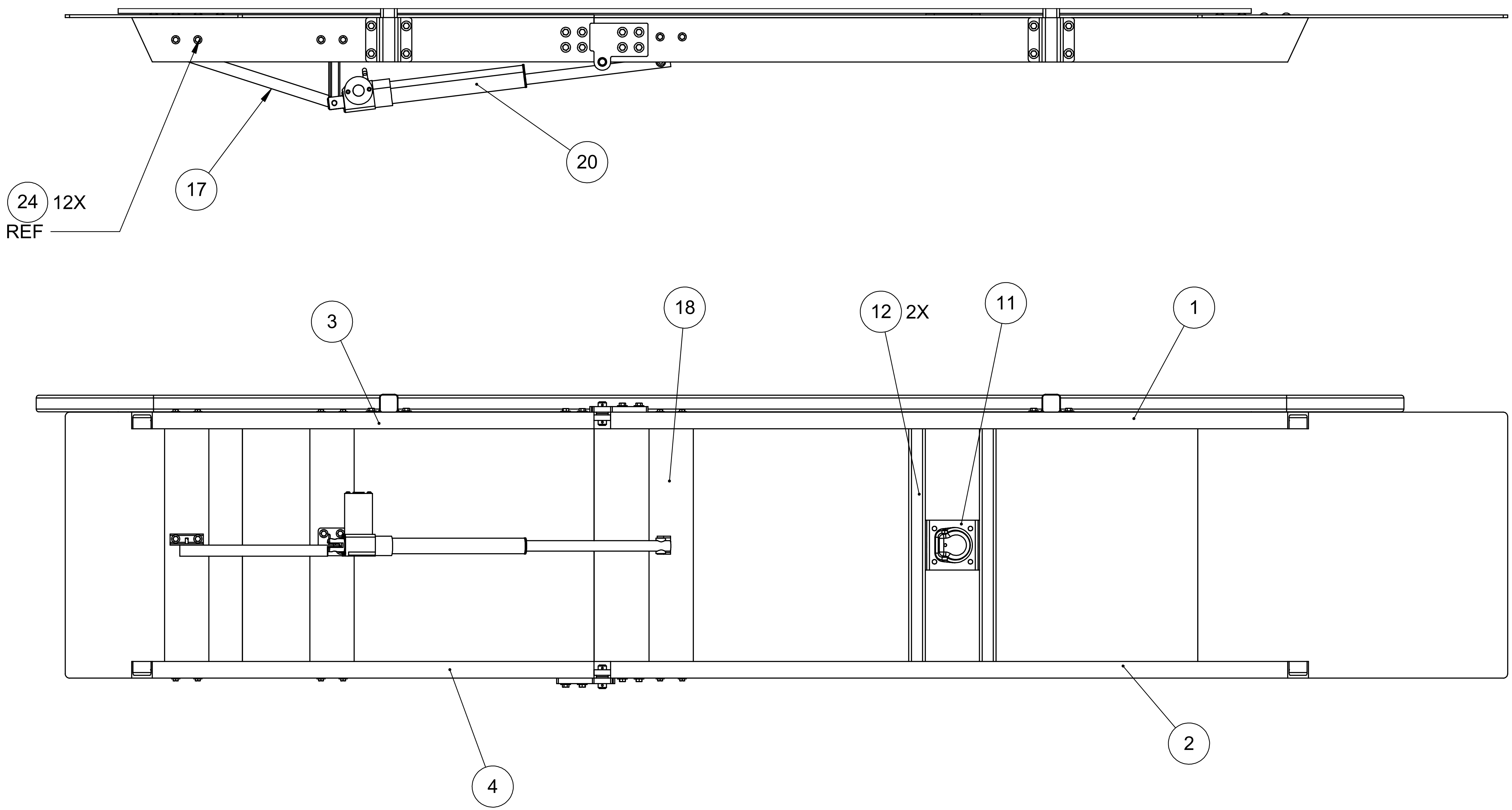
REV 0 A01 DWG. NO.



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			DWN / CKD	
0	16/12/06	PRELIMINARY DWG	DWN / CKD	C.P.

BOLT TORQUE
FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE

1/4-20"	143 IN-LBS
5/16"	195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

MATERIAL

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

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TITLE:

SIZE **B**

DWG. NO. A01

REV 0

SCALE: 1:10 WEIGHT: N/A SHEET 3 OF 3

DWG. NO. A01 REV 0

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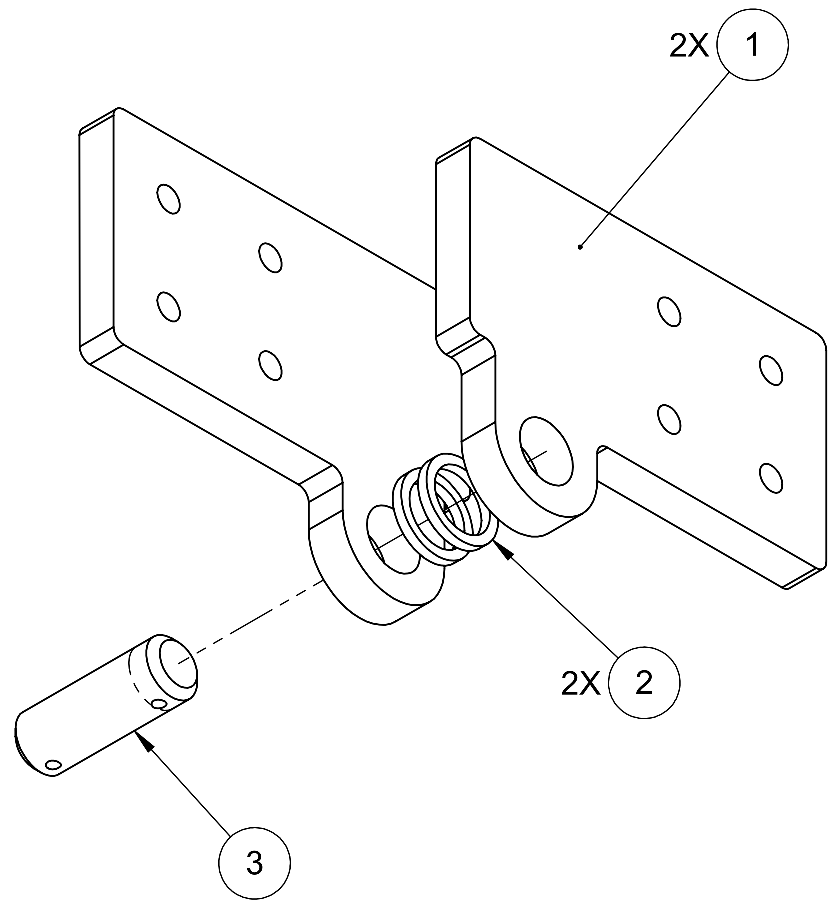
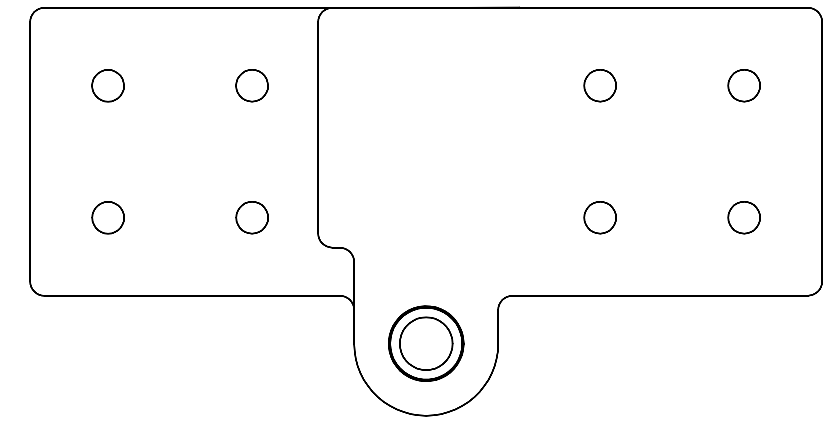
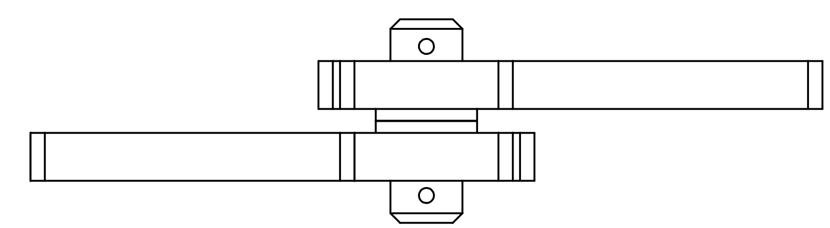
1	CH1-1	PLATE, HINGE	2
2	H1	NYLON WASHER, 3/4" Screw Size, 0.812" ID, 1.057" OD, 0.125" THK	2
3	CH1-2	PIN	1







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TITLE: ASSY - CENTER HINGE

MATERIAL: N/A

FINISH: NO PLATING REQUIRED

SIZE: B

DWG. NO. CH1-0

REV 0

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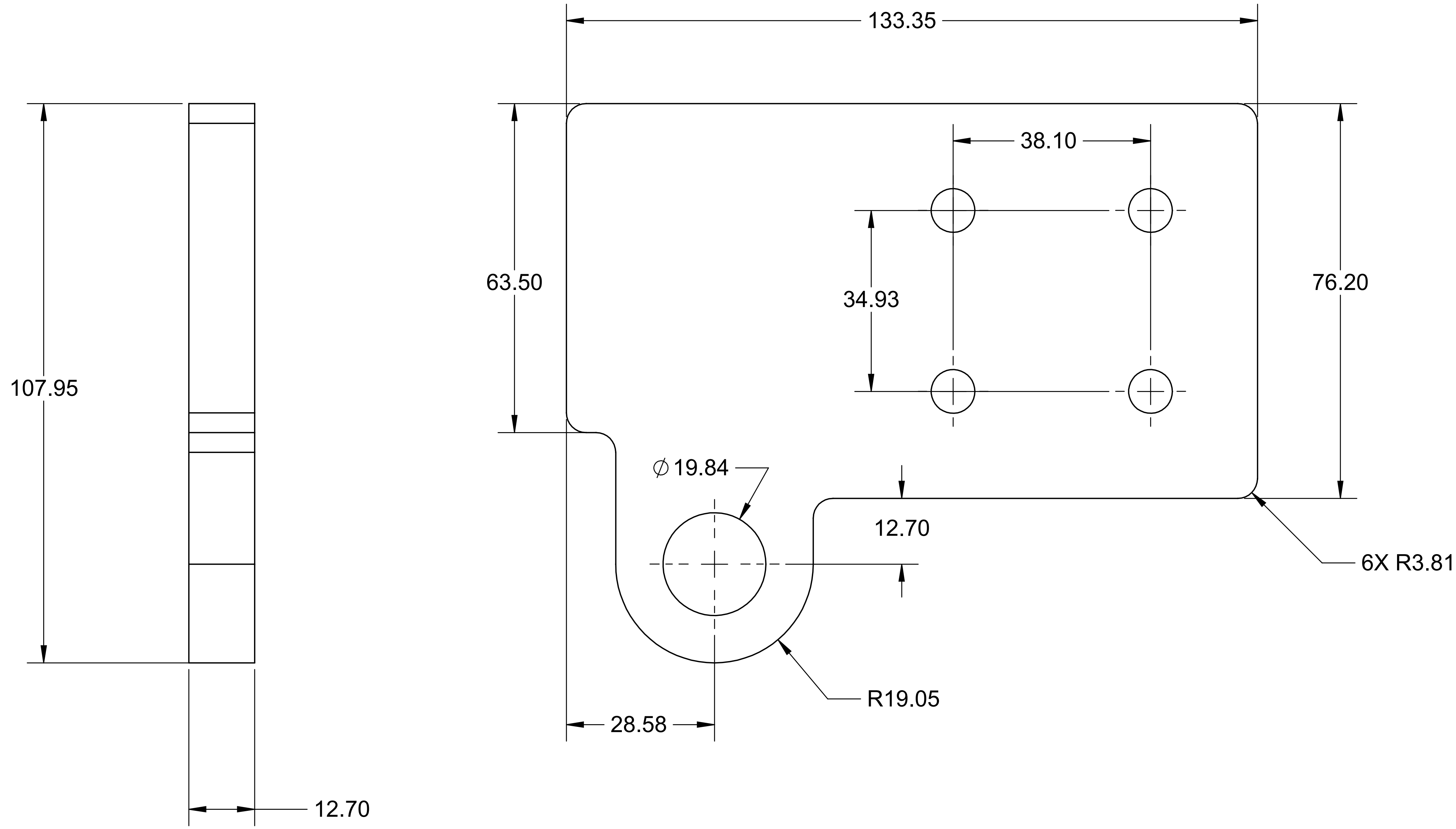
DWG. NO. CH1-0 REV 0



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UNLESS OTHERWISE SPECIFIED ANGULAR ± 2° LINEAR 0.XX ± 0.5 0.XXX ± 0.15	MATERIAL CSA G40.21 50W	TITLE:	PLATE, HINGE
	FINISH NO PLATING REQUIRED	SIZE **B**	DWG. NO. CH1-1 — REV 0
	DO NOT SCALE DRAWING	SCALE: 1:1	WEIGHT: N/A — SHEET 1 OF 1

REV	DATE Y/M/D	DESCRIPTION	DWN / CKD	
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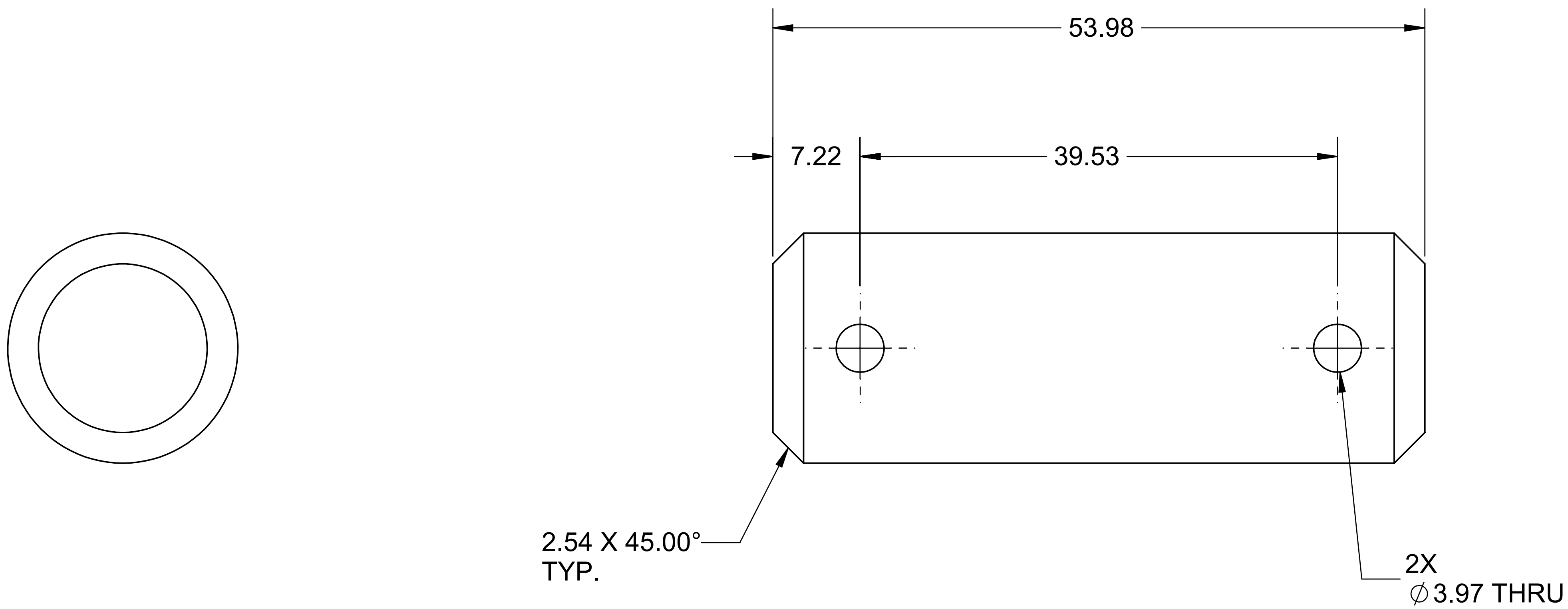
REV 0

DWG. NO. CH1-1



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DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
AISI 1215 STEEL
0.75 IN. ROD

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

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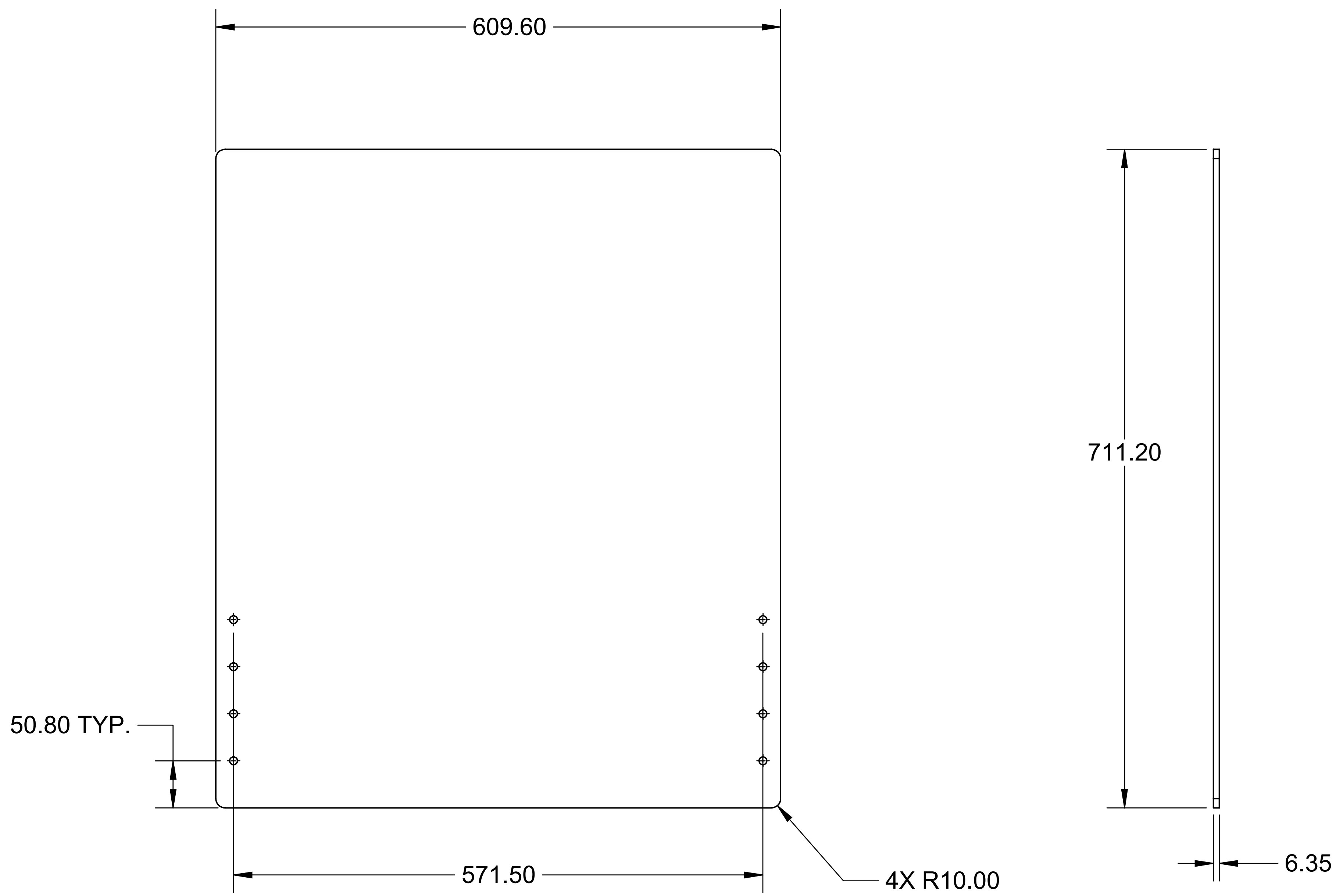
TITLE:

SIZE B | DWG. NO. CH1-2 | REV 0

SCALE: 2:1 | WEIGHT: N/A | SHEET 1 OF 1

REV 0
DWG. NO. CH1-2



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			DWN	
			CKD	
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			CKD	
			DWN	
			CKD	
0	16/12/02	PRELIMINARY DWG	DWN	C.P.
			CKD	
REV	DATE Y/M/D	DESCRIPTION		

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
0.250 IN. SHEET 6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

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TITLE:
PLATE, MOUNT

SIZE	DWG. NO.	REV
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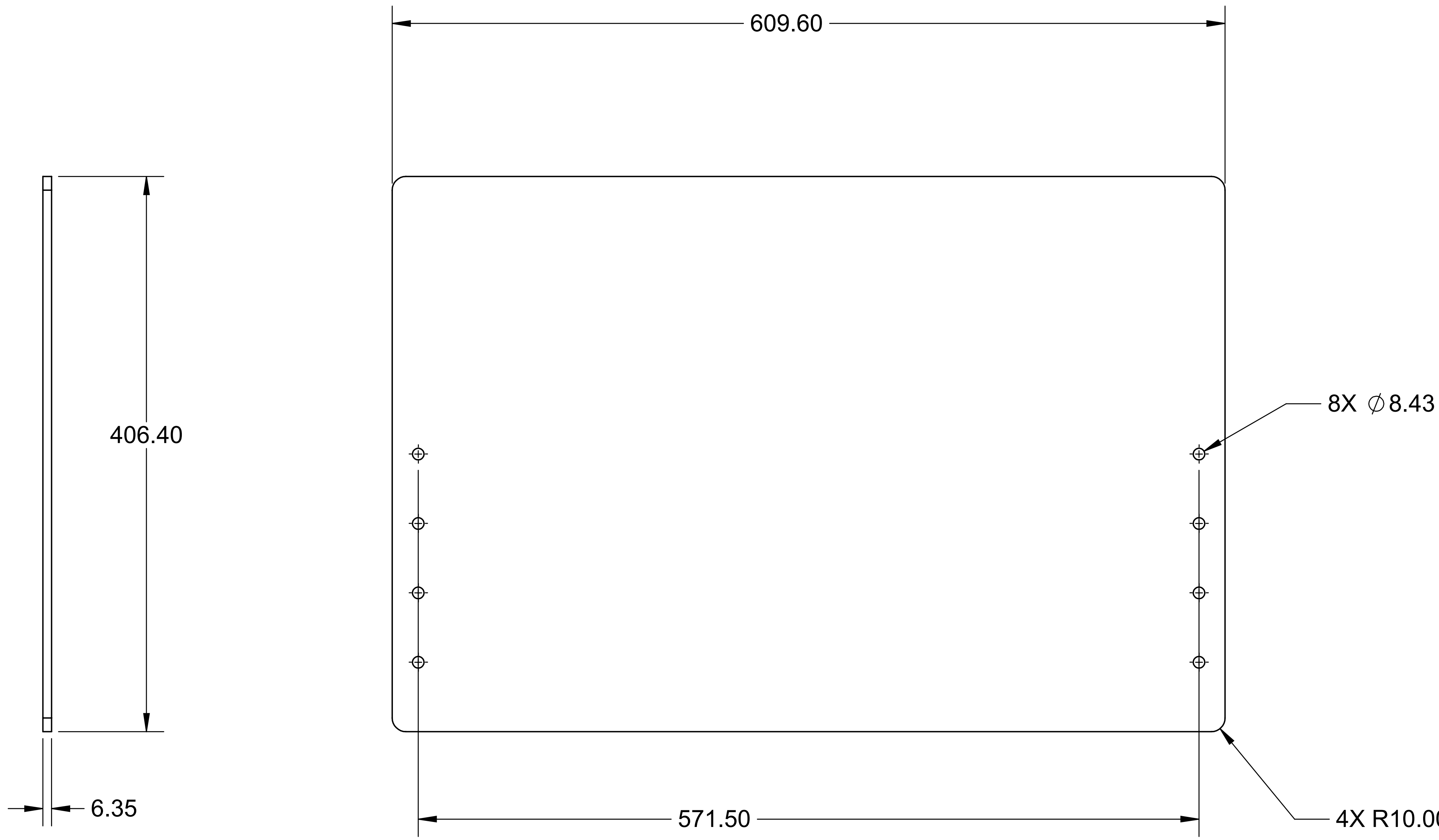
SCALE: 1:6 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. DS1-1 REV 0



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
0.250 IN. SHEET 6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

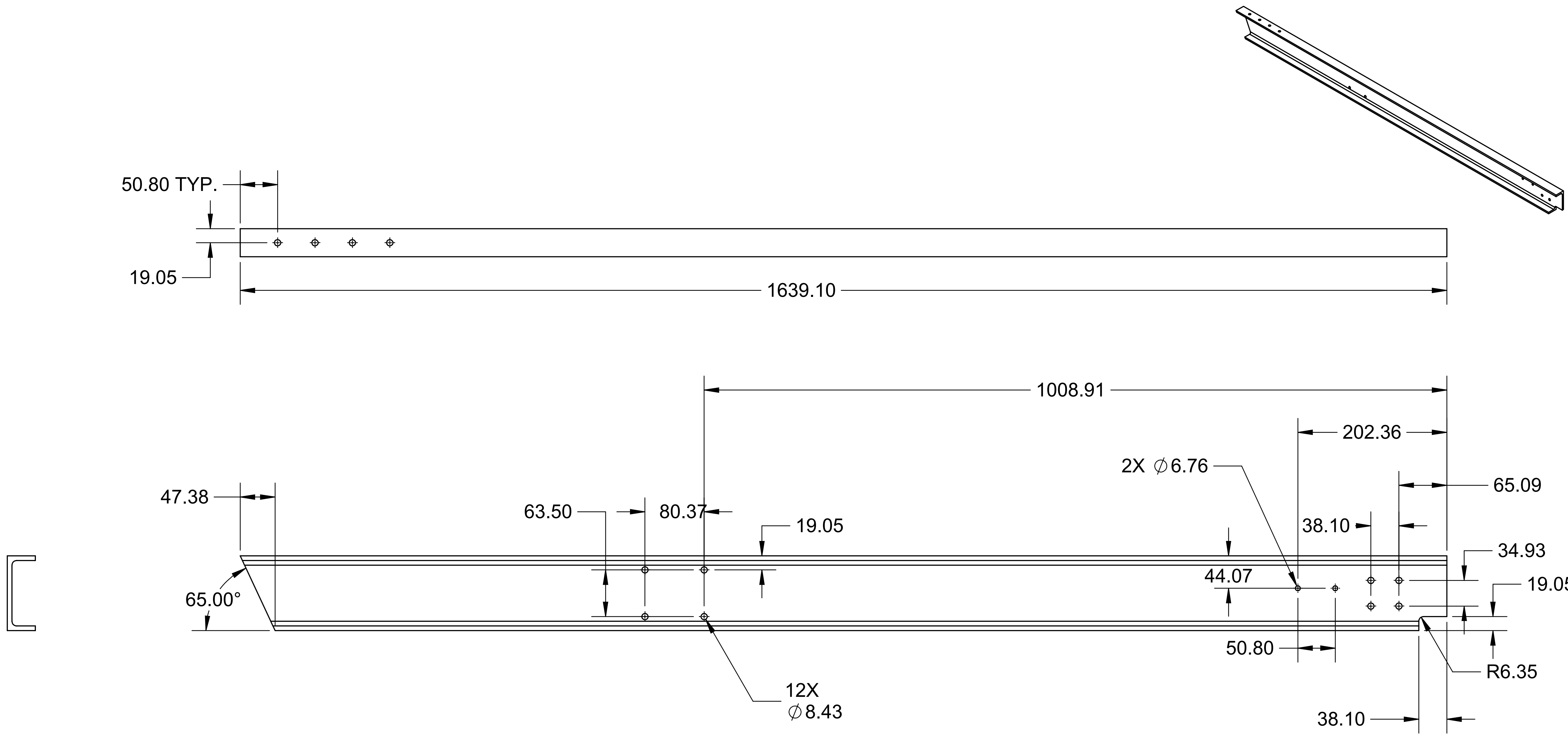
UNIVERSITY OF MANITOBA
MECH 4860 - 2016 TEAM 23

TITLE:
PLATE, SUPPORT

SIZE **B** DWG. NO. DS1-2 REV 0

SCALE: 1:4 WEIGHT: N/A SHEET 1 OF 1

DWG. NO. DS1-2 REV 0



DS1-3 REV 0

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

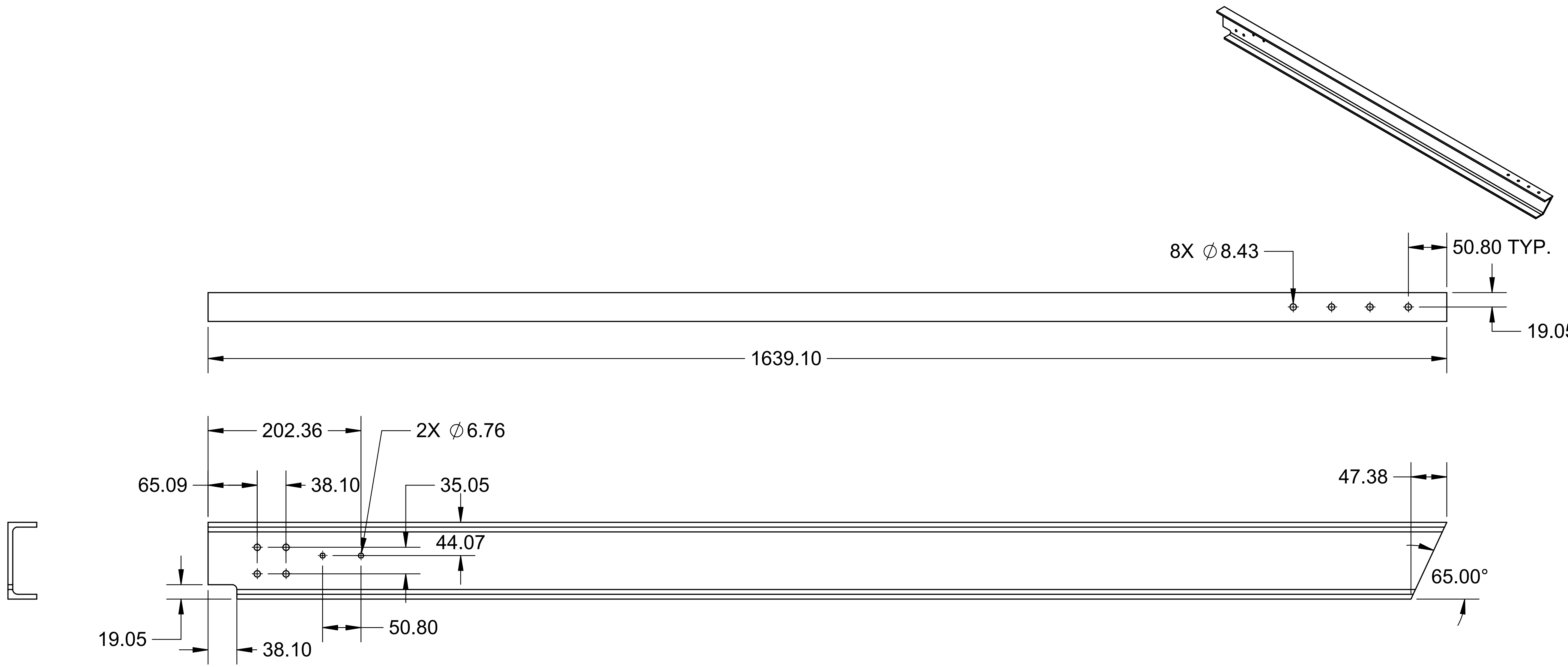
UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
CHANNEL, LONG, LEFT
4 X 1.5 X .25 WEB C-CHANNEL

SIZE B | DWG. NO. DS1-3 | REV 0

SCALE: 1:6 | WEIGHT: N/A | SHEET 1 OF 1



REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

EDUCATIONAL AND PROPRIETARY
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MATERIAL: 6061-T6 ALUMINUM

FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

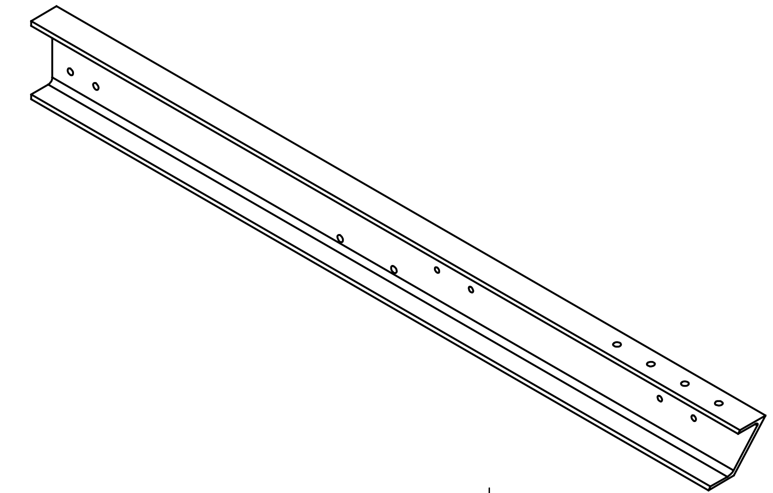
MECH 4860 - 2016 TEAM 23

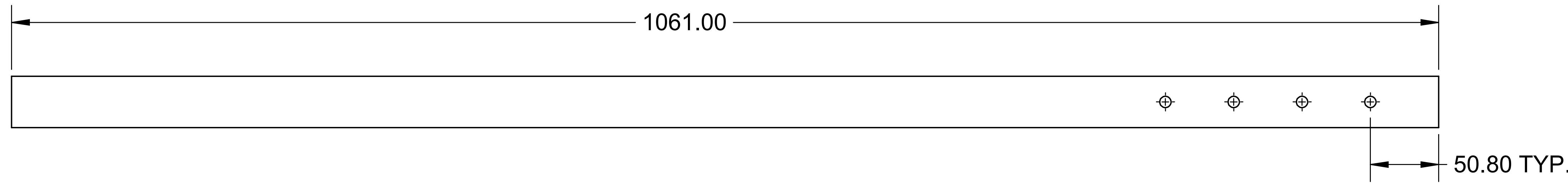
TITLE:
CHANNEL, LONG, RIGHT
4 X 1.5 X .25 WEB C-CHANNEL

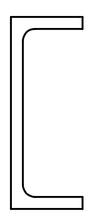
SIZE: B | DWG. NO. DS1-4 | REV 0

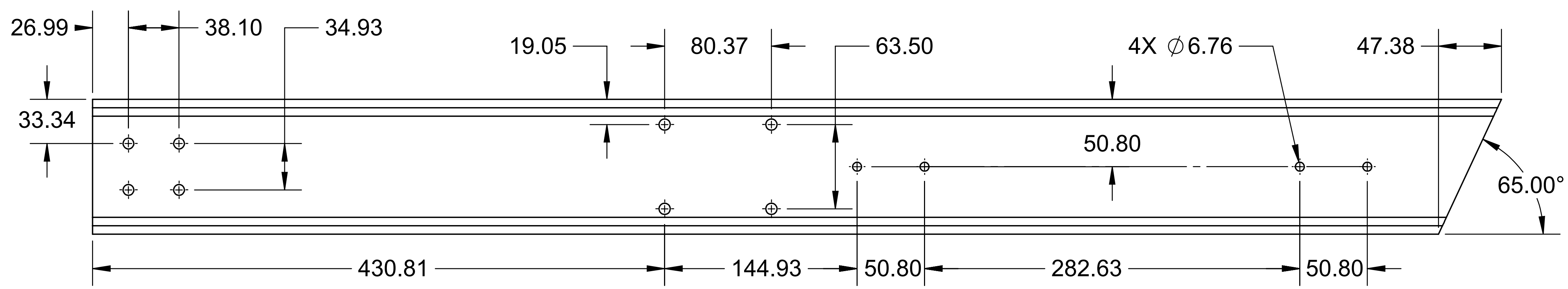
SCALE: 1:6 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. DS1-4 REV 0









REV	DATE Y/M/D	DESCRIPTION		
			DWN NAME #	
			CKD NAME #	
			DWN NAME #	
			CKD NAME #	
			DWN NAME #	
			CKD NAME #	
			DWN NAME #	
			CKD NAME #	
0	16/12/02	PRELIMINARY DWG	DWN NAME #	C.P.
			CKD NAME #	

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
CHANNEL, SHORT, LEFT
4 X 1.5 X .25 WEB C-CHANNEL

SIZE **B**

DWG. NO. DS1-5

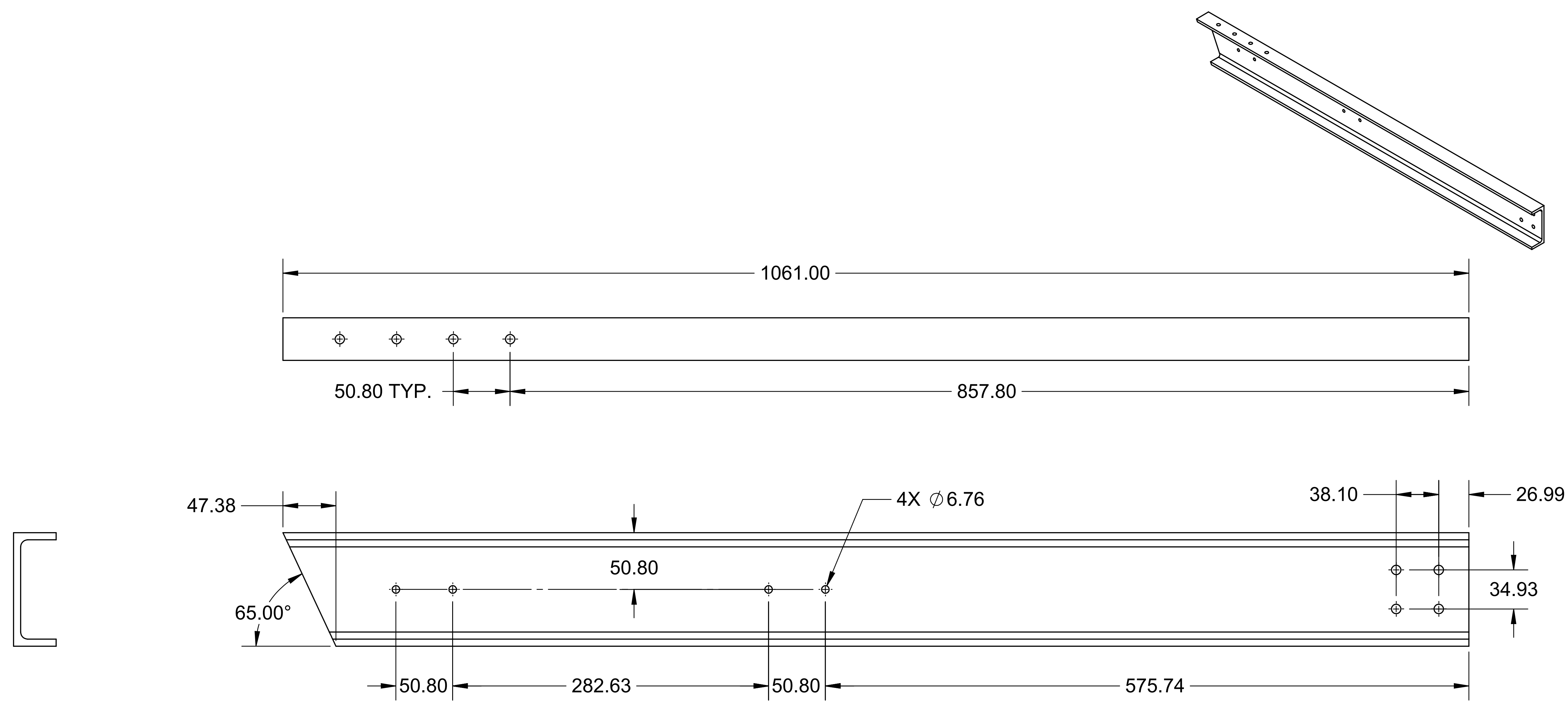
REV 0

SCALE: 1:4

WEIGHT: N/A

SHEET 1 OF 1

DWG. NO. DS1-5 REV 0



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL: 6061-T6 ALUMINUM

FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

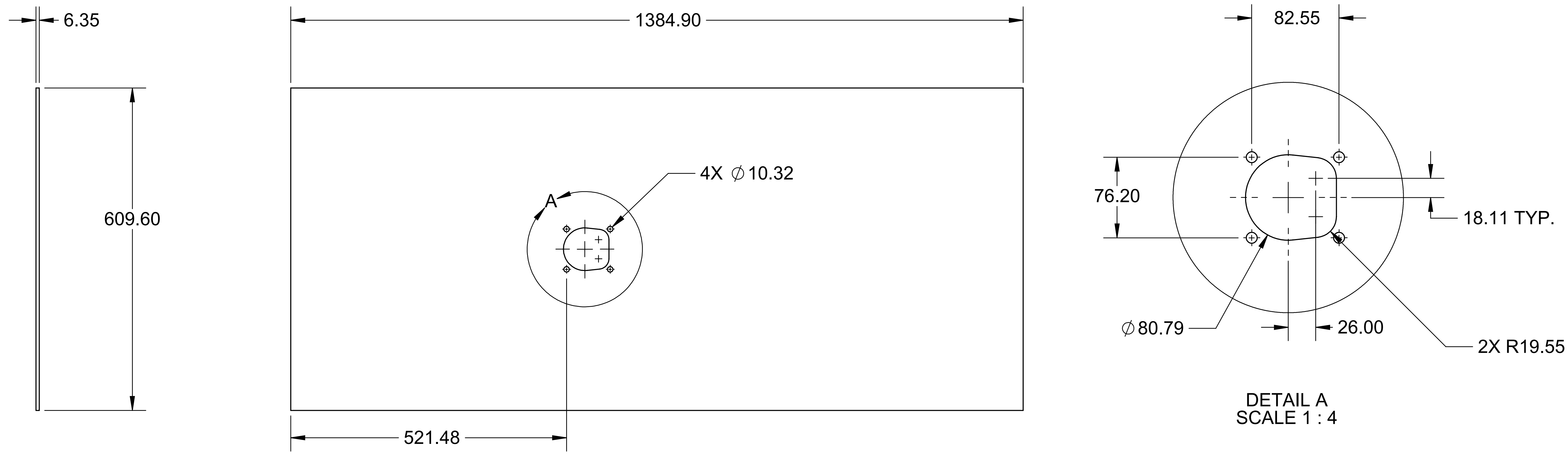
MECH 4860 - 2016 TEAM 23

TITLE:
CHANNEL, SHORT, RIGHT
4 X 1.5 X .25 WEB C-CHANNEL

SIZE: B | DWG. NO.: DS1-6 | REV: 0

SCALE: 1:4 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. DS1-6 REV 0



REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

MATERIAL: PLYWOOD

FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

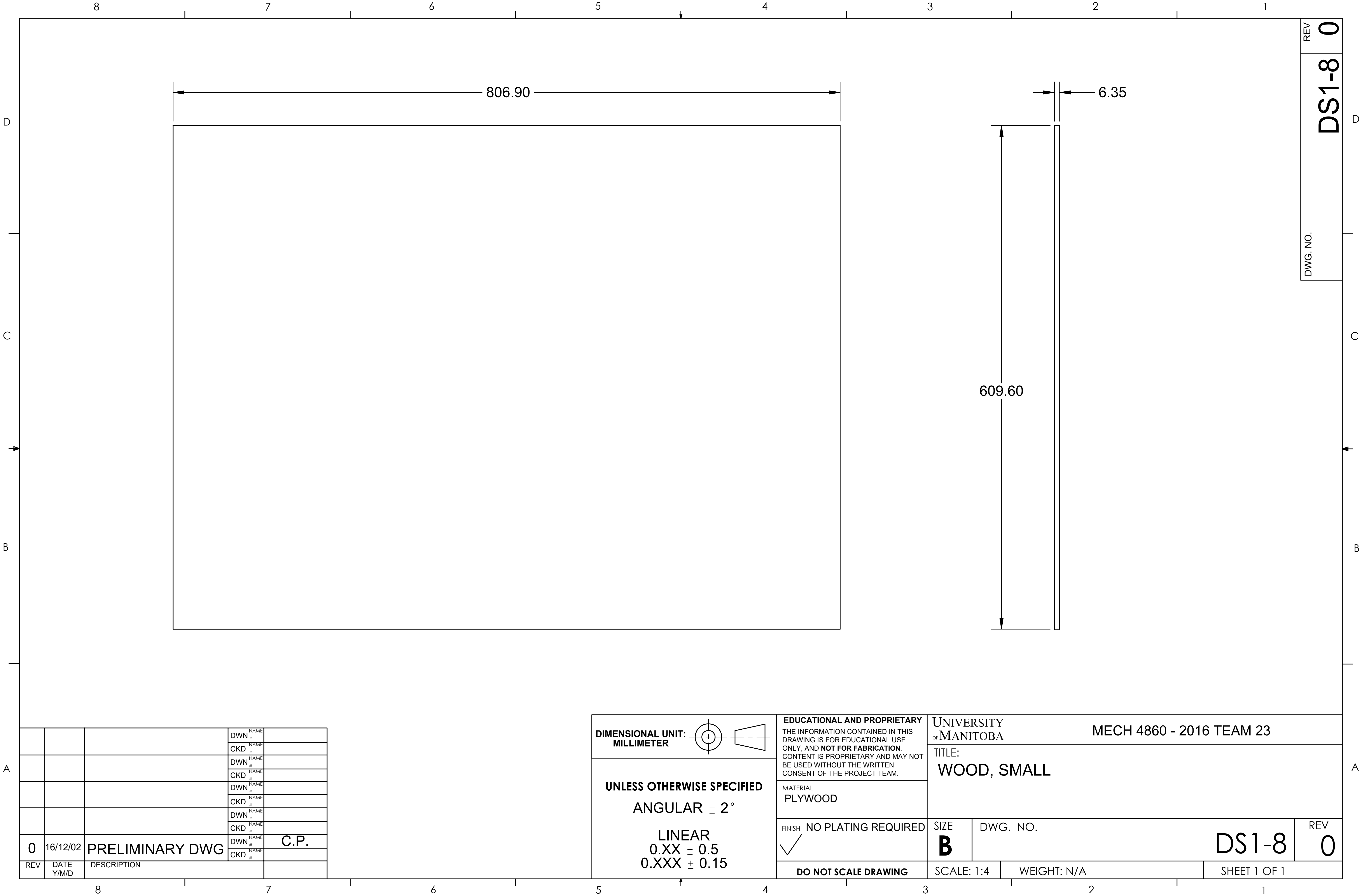
UNIVERSITY OF MANITOBA — MECH 4860 - 2016 TEAM 23

TITLE: WOOD, LARGE

SIZE	DWG. NO.	REV
B	DS1-7	0

SCALE: 1:8 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. DS1-7 REV 0


806.90
6.35
609.60
REV 0
DWG. NO. DS1-8
DIMENSIONAL UNIT: MILLIMETER
UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15
EDUCATIONAL AND PROPRIETARY
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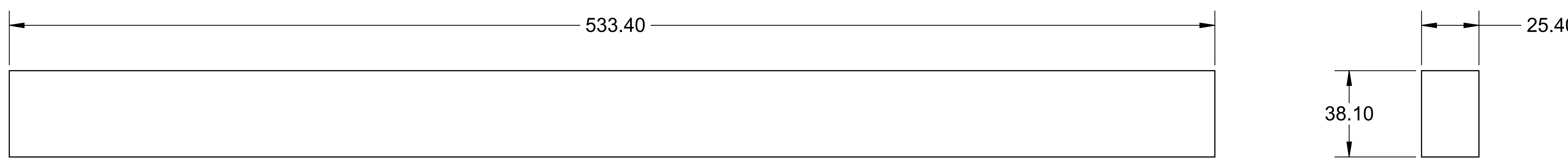
MATERIAL PLYWOOD
FINISH NO PLATING REQUIRED
DO NOT SCALE DRAWING
UNIVERSITY OF MANITOBA
MECH 4860 - 2016 TEAM 23
TITLE:
WOOD, SMALL
SIZE B
DWG. NO. DS1-8
REV 0
SCALE: 1:4
WEIGHT: N/A
SHEET 1 OF 1
0 16/12/02 PRELIMINARY DWG DWN C.P.
REV DATE Y/M/D DESCRIPTION

DS1-9 REV 0



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL: SPRUCE

FINISH: ✓

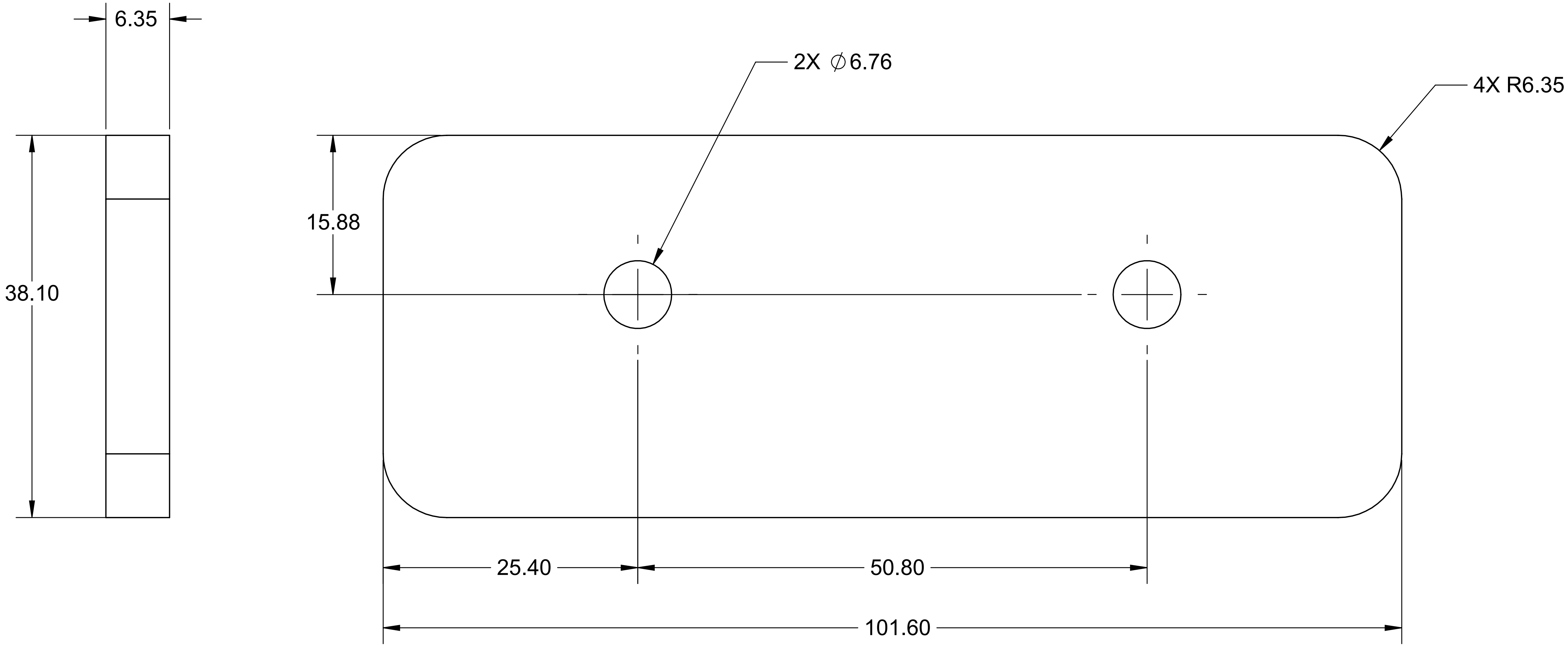
DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 - 2016 TEAM 23

TITLE: WOOD STIFFENERS

SIZE: B | DWG. NO.: DS1-9 | REV: 0

SCALE: 1:2 | WEIGHT: N/A | SHEET 1 OF 1



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
.25 IN. 6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

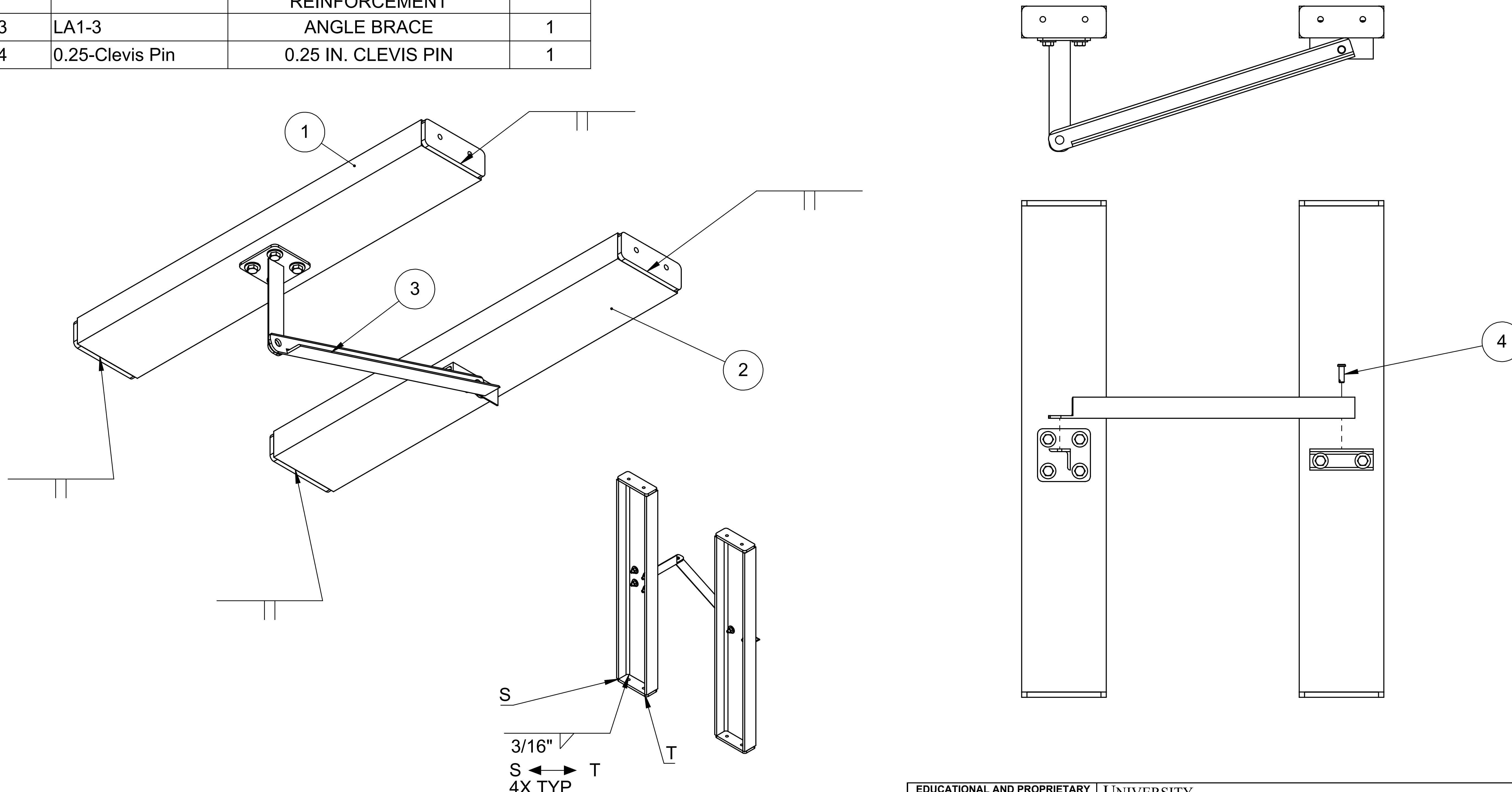
TITLE:
END PLATE

SIZE B | DWG. NO. EP-1 | REV 0

SCALE: 2:1 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. EP-1 REV 0

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	LA1-1-0	ASSY - MOUNT PICK UP, MOTOR SIDE	1
2	LA1-2-0	ASSY - MOUNT, REINFORCEMENT	1
3	LA1-3	ANGLE BRACE	1
4	0.25-Clevis Pin	0.25 IN. CLEVIS PIN	1



REV 0

DWG. NO. LA1-0

BOLT TORQUE

FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE

1/4-20"	143 IN-LBS
5/16"	195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

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MATERIAL

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

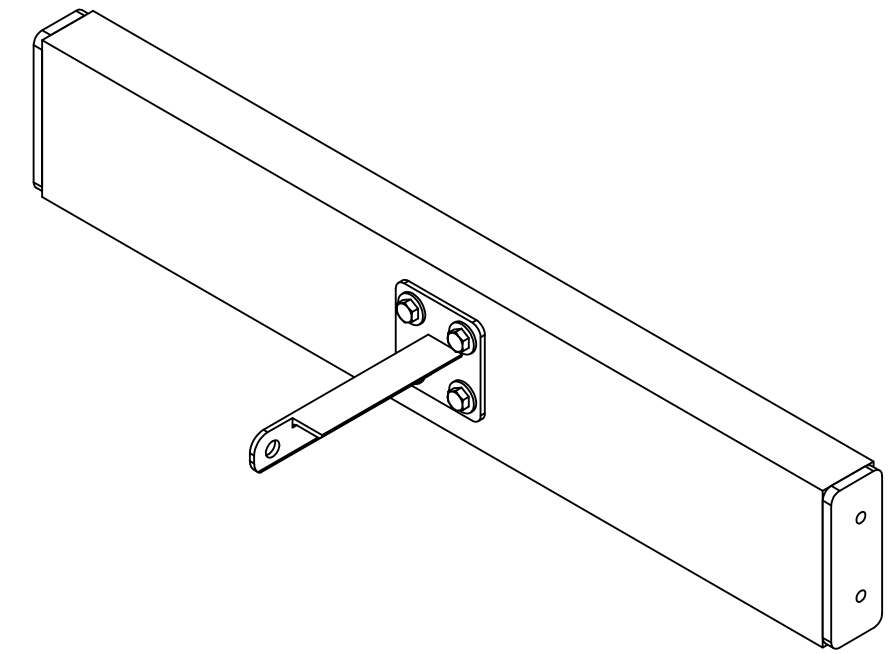
MECH 4860 - 2016 TEAM 23

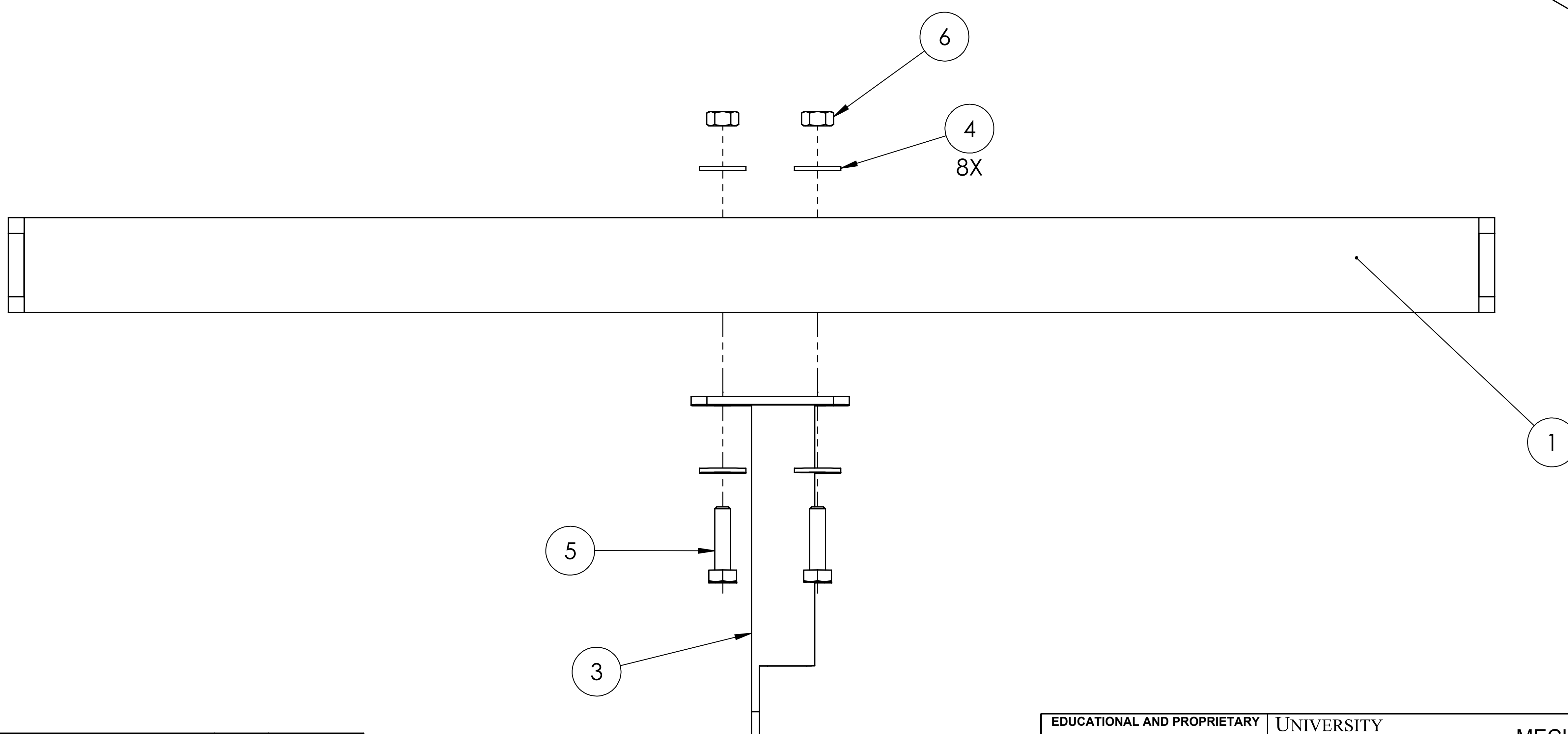
TITLE: ASSY - MOUNT, MOTOR SIDE

SIZE **B** | DWG. NO. LA1-0 | REV 0

SCALE: 1:5 | WEIGHT: N/A | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	LA1-1-1	CROSS CHANNEL BRACE, MOTOR SIDE	1
2	EP-1	END PLATE	2
3	LA1-1-2-0	ASSY - MOUNT, VERTICAL	1
4	Preferred Wide FW 0.25	0.25 IN. WASHER	8
5	HBOLT 0.2500-20x1x0.75-N	GRADE 8 HEX BOLT 1/4-20X1	4
6	HNUT 0.2500-20-D-N	GRADE 8 HEX NUT 1/4-20	4





REV 0 | DWG. NO. LA1-1-0

REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

BOLT TORQUE
FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE

1/4-20"	143 IN-LBS
5/16"	195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

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MATERIAL

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

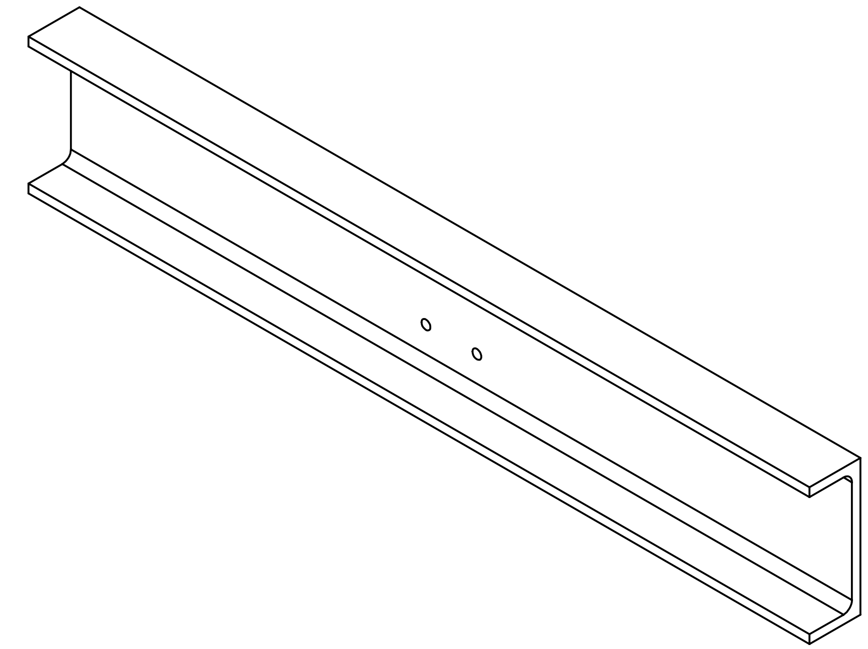
UNIVERSITY OF MANITOBA

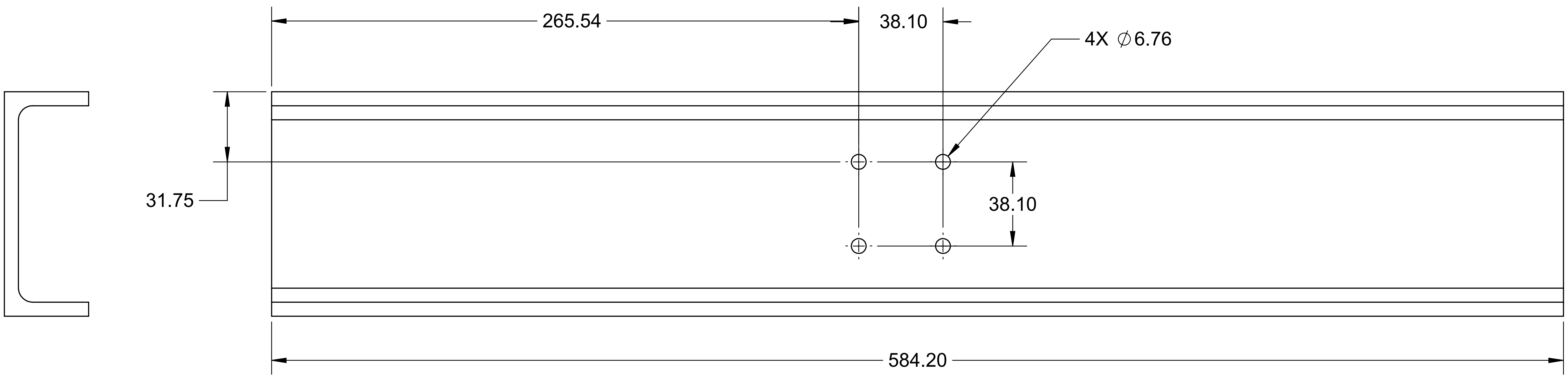
MECH 4860 - 2016 TEAM 23

TITLE: ASSY - MOUNT PICKUP, MOTOR SIDE

SIZE **B** | DWG. NO. LA1-1-0 | REV 0

SCALE: 1:2 | WEIGHT: N/A | SHEET 1 OF 1





REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

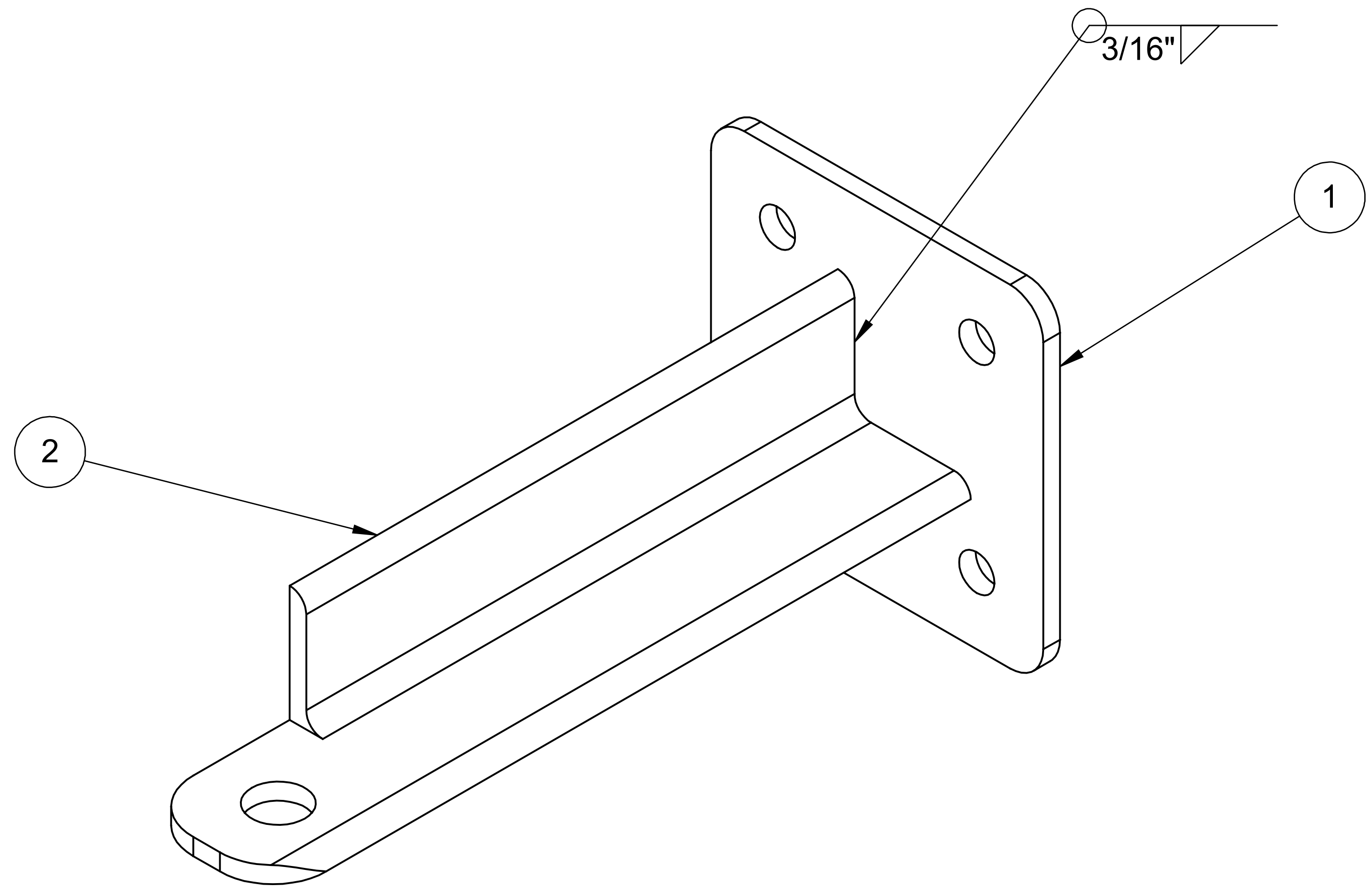
TITLE:
CROSS CHANNEL BRACE, MOTOR SIDE
4 X 1.5 X .25 WEB C-CHANNEL

SIZE B | DWG. NO. LA1-1-1 | REV 0

SCALE: 1:2 | WEIGHT: N/A | SHEET 1 OF 1

REV 0 | LA1-1-1 | DWG. NO.

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	LA1-1-2-2	MOUNT, BASE PLATE	1
2	LA1-1-2-1	MOUNT, VERTICAL BRACE	1



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE: ASSY - VERTICAL MOUNT

MATERIAL

FINISH NO PLATING REQUIRED

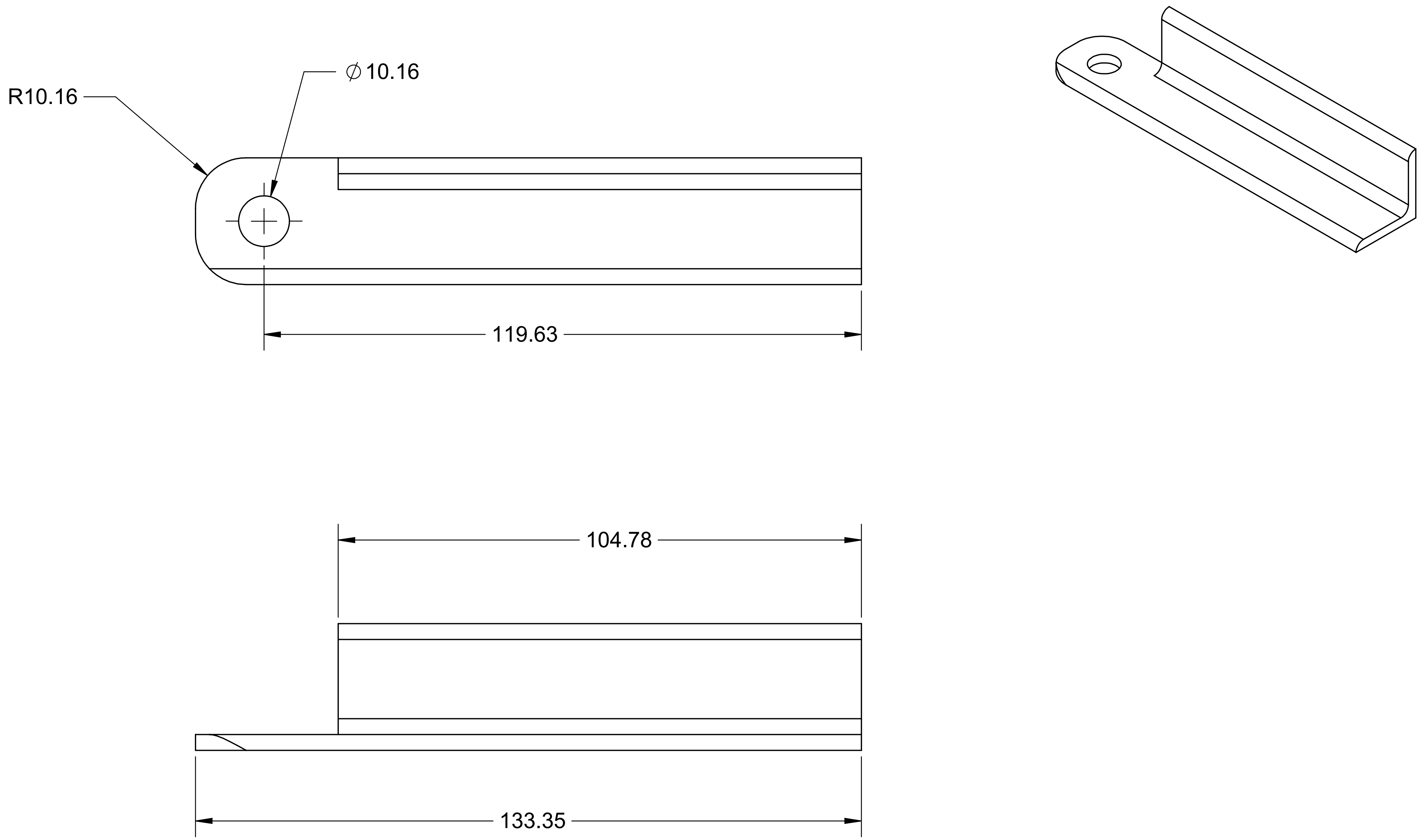
SIZE B

DWG. NO. LA1-1-2-0

REV 0

DO NOT SCALE DRAWING

SCALE: 1:1 | WEIGHT: N/A | SHEET 1 OF 1



REV 0

DWG. NO. LA1-1-2-1

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
ASTM A36 STEEL

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
VERTICAL MOUNT
1 IN. ANGLE IRON

SIZE **B**

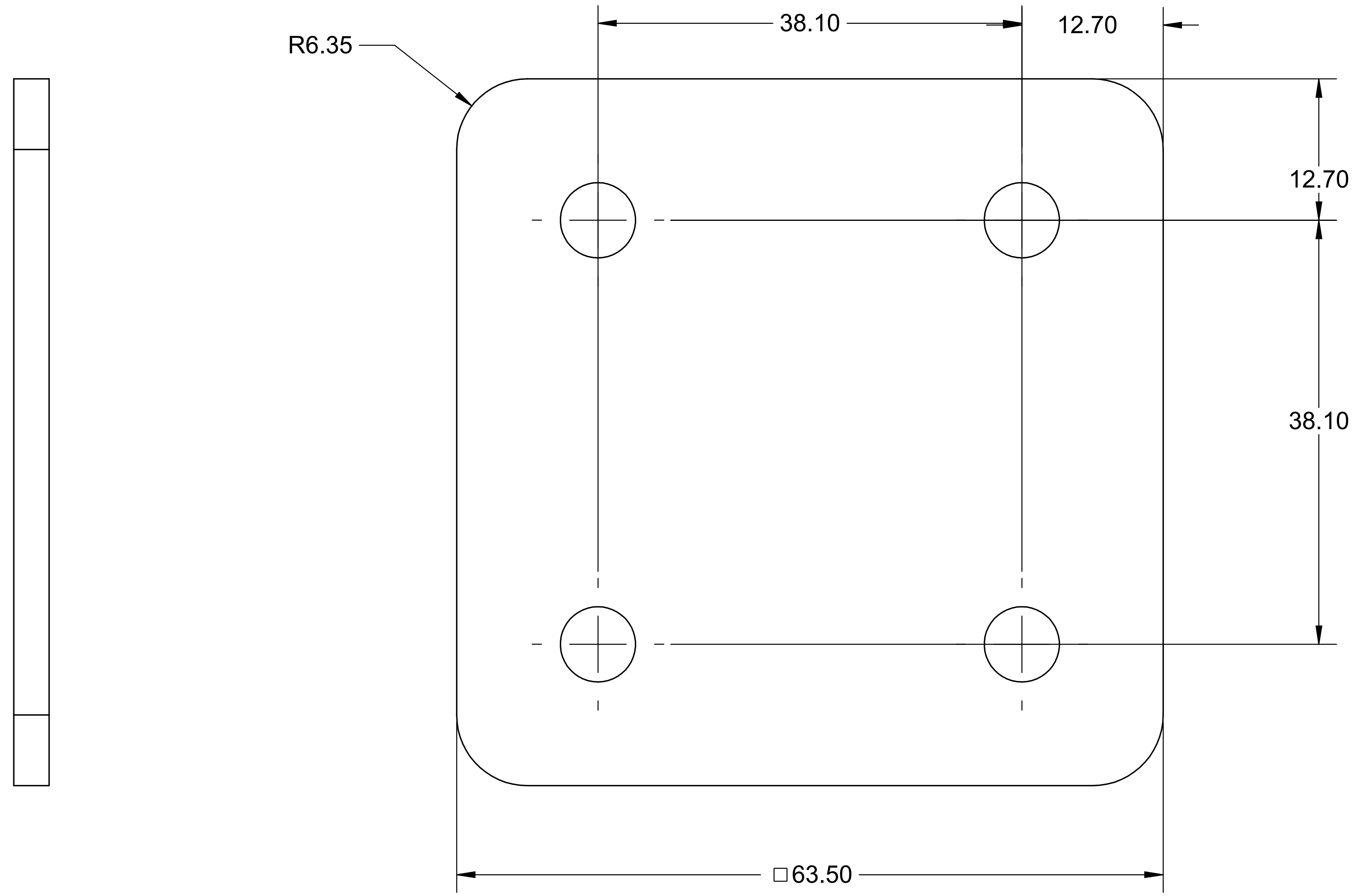
DWG. NO. LA1-1-2-1

REV 0

SCALE: 1:1

WEIGHT: N/A

SHEET 1 OF 1



LA1-1-2-2 | REV 0 | DWG. NO.

REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

EDUCATIONAL AND PROPRIETARY
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MATERIAL: 11GA CSA G40.21 50W

FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

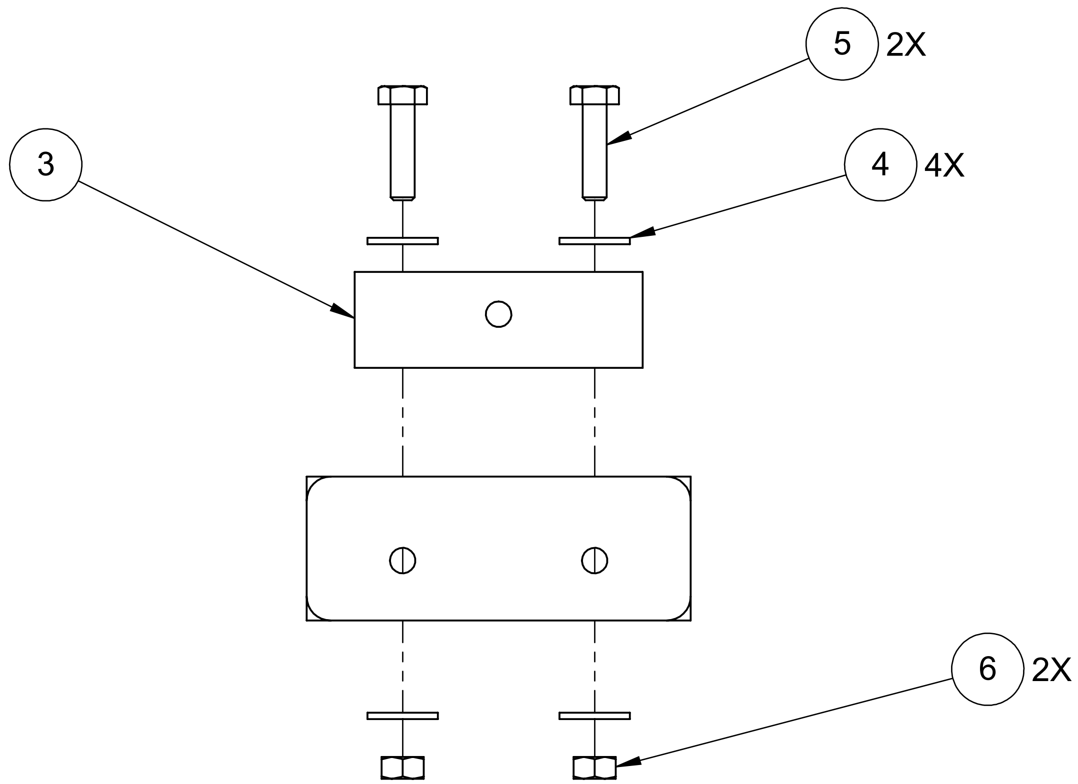
MECH 4860 - 2016 TEAM 23

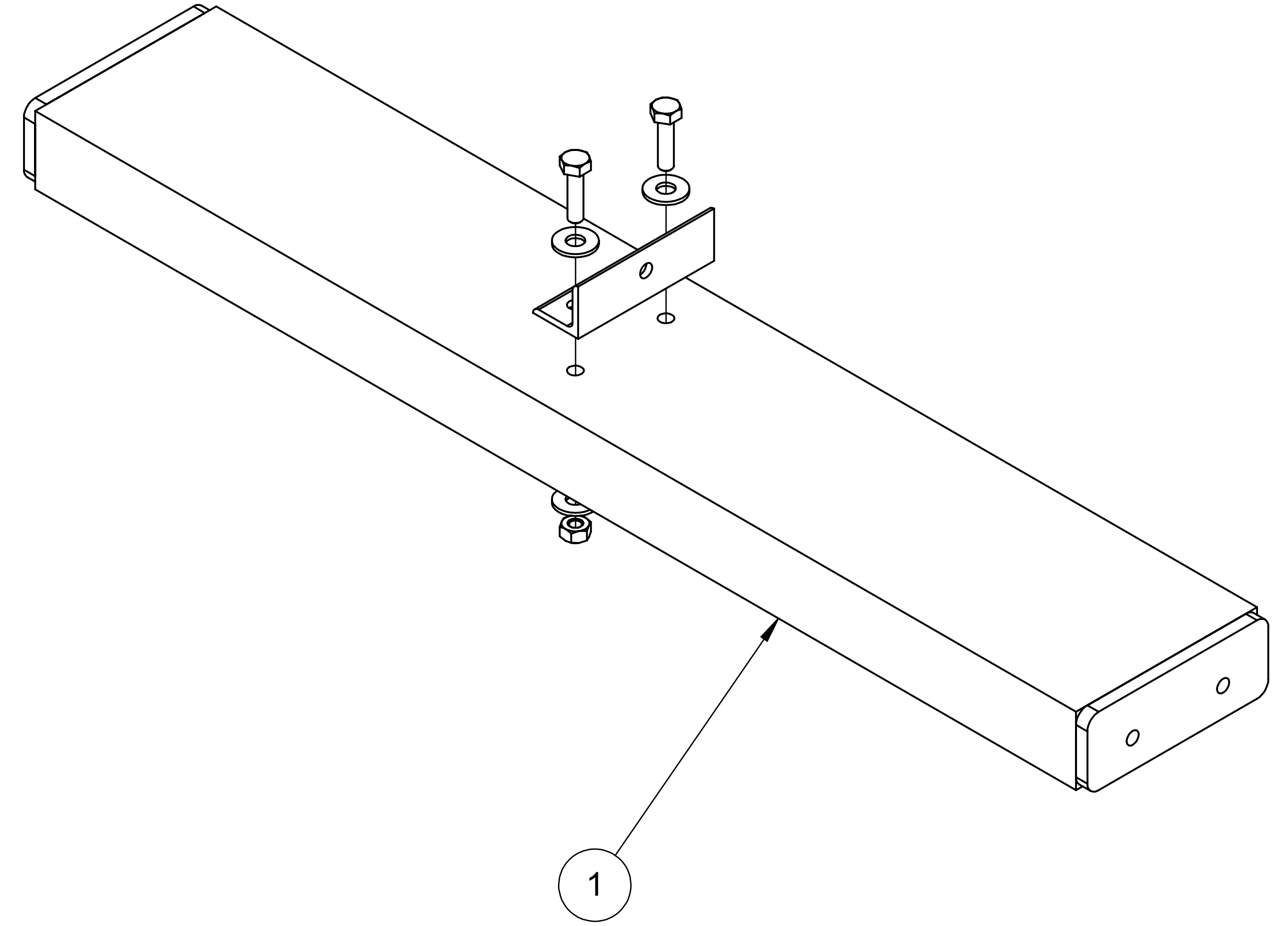
TITLE: BASE PLATE

SIZE: B | DWG. NO. LA1-1-2-2 | REV 0

SCALE: 2:1 | WEIGHT: N/A | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	LA1-2-1	CROSS CHANNEL BRACE, REINFORCEMENT	1
2	EP-1	END PLATE	2
3	LA1-2-2	BRACE	1
4	Preferred Wide FW 0.25	1/4" WIDE WASHER	4
5	HBOLT 0.2500-20x1x0.75-N	GRADE 8 HEX BOLT 1/4-20X1	2
6	HNUT 0.2500-20-D-N	GRADE 8 HEX NUT 1/4-20	2





REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN	C.P.
			CKD	

BOLT TORQUE
FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE
1/4-20" 143 IN-LBS
5/16" 195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

EDUCATIONAL AND PROPRIETARY
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MATERIAL

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

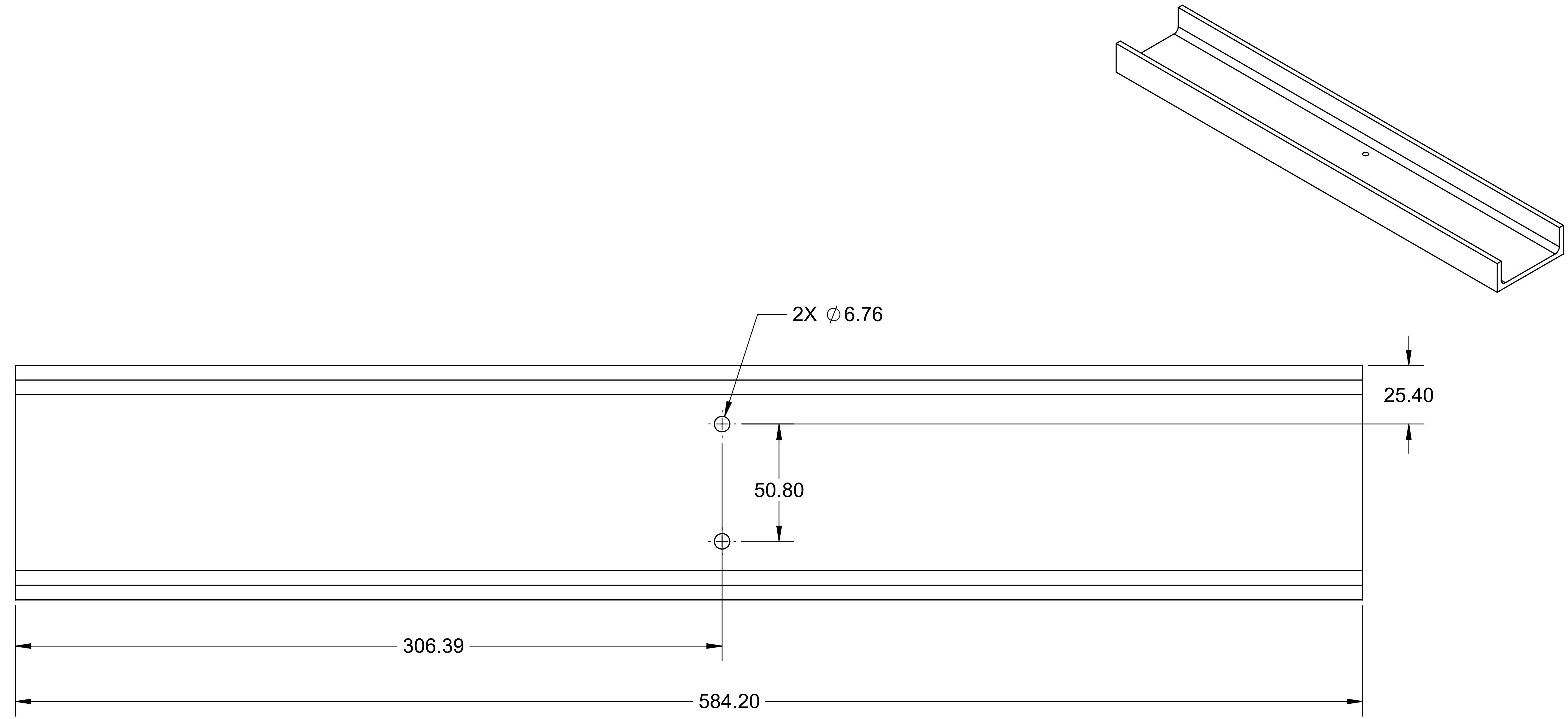
MECH 4860 - 2016 TEAM 23

TITLE: ASSY - MOUNT, REINFORCEMENT

SIZE B | DWG. NO. LA1-2-0 | REV 0

SCALE: 1:3 | WEIGHT: N/A | SHEET 1 OF 1

REV 0 DWG. NO. LA1-2-0



REV 0
DWG. NO. LA1-2-1

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

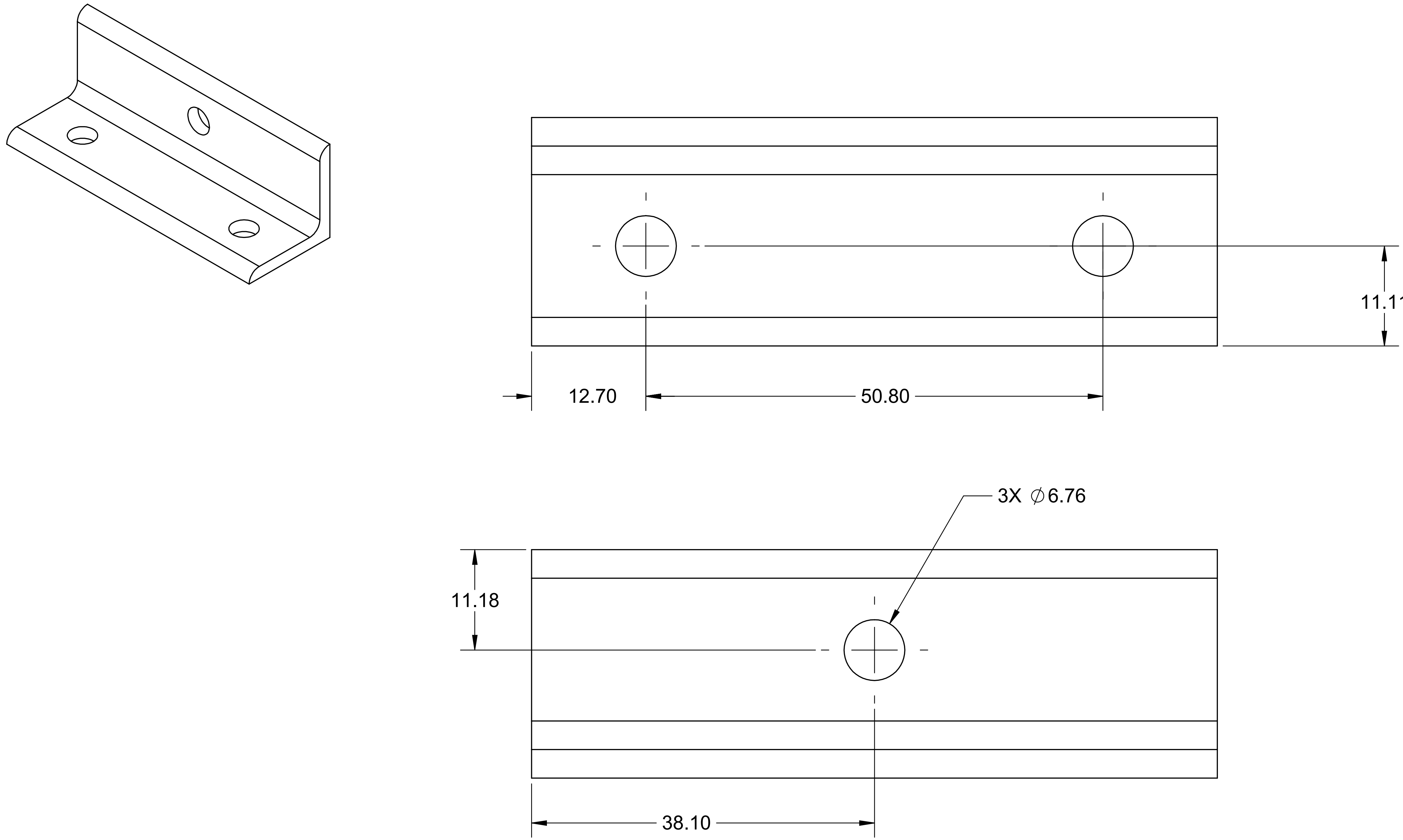
UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
CROSS CHANNEL BRACE, REINFORCEMENT
4 X 1.5 X .25 WEB C-CHANNEL

SIZE	DWG. NO.	REV
B	LA1-2-1	0

SCALE: 1:2 | WEIGHT: N/A | SHEET 1 OF 1



REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

MATERIAL: ASTM A36 STEEL

FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

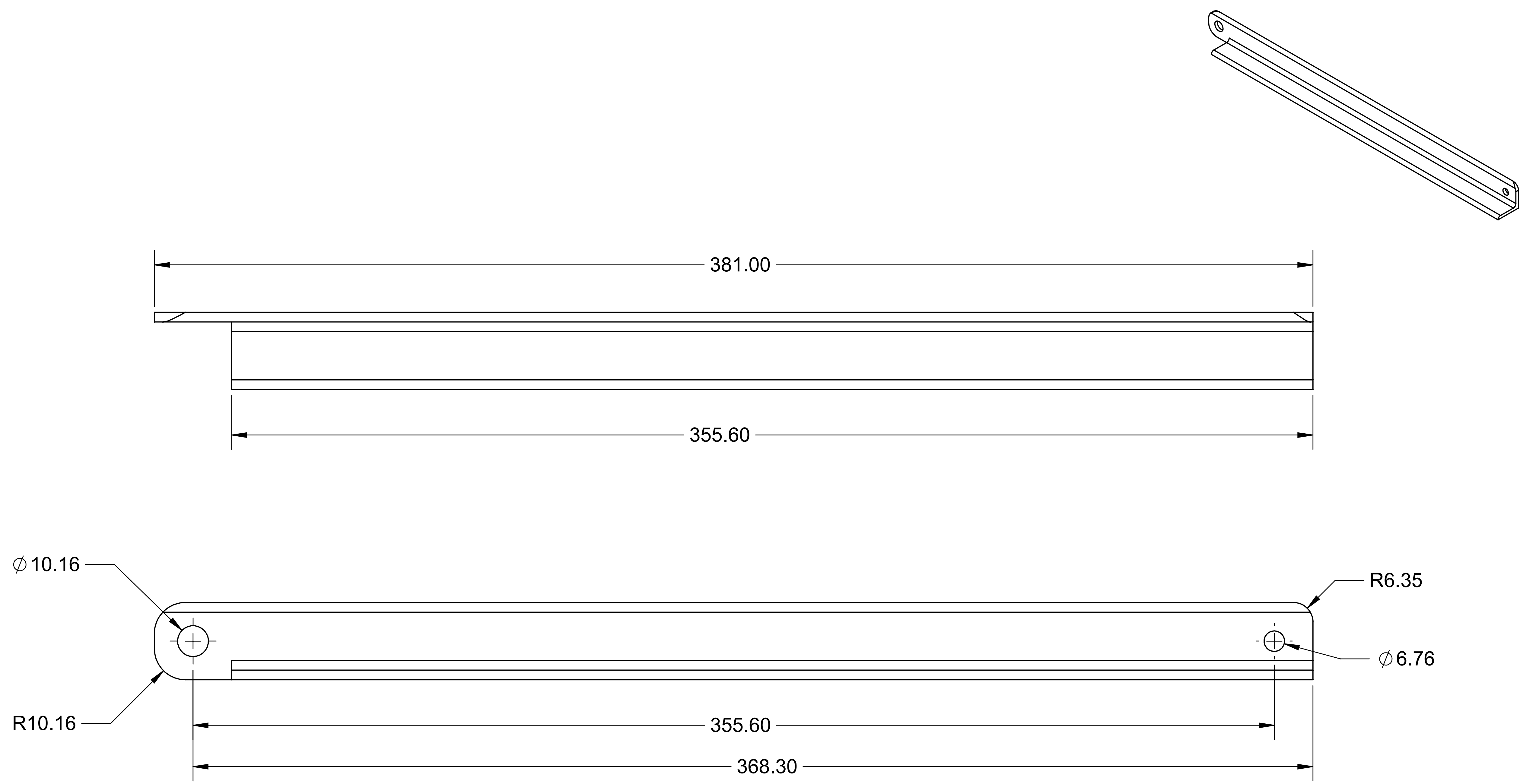
MECH 4860 - 2016 TEAM 23

TITLE:
BRACE
1 IN. ANGLE IRON

SIZE: B | DWG. NO.: LA1-2-2 | REV: 0

SCALE: 2:1 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. LA1-2-2 REV 0



REV 0
DWG. NO. LA1-3

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL: ASTM A36 STEEL

FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

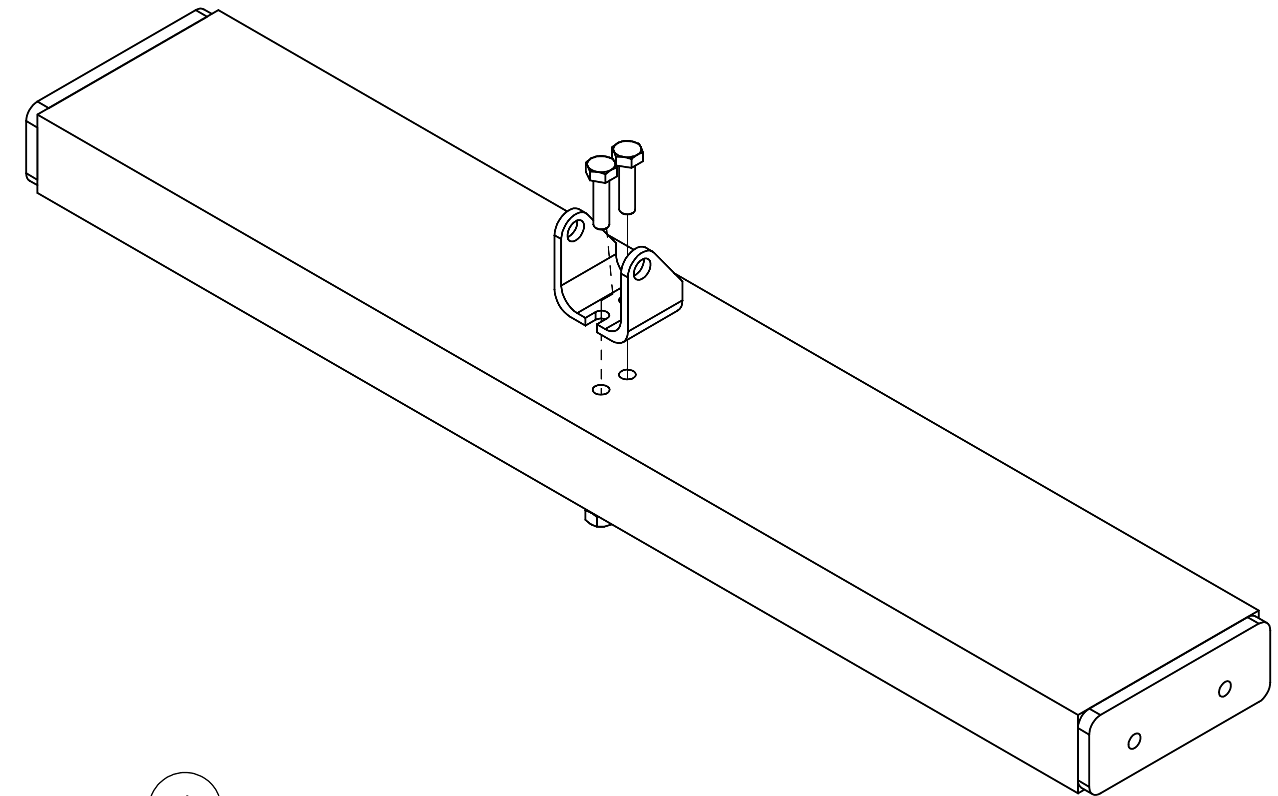
UNIVERSITY OF MANITOBA — MECH 4860 - 2016 TEAM 23

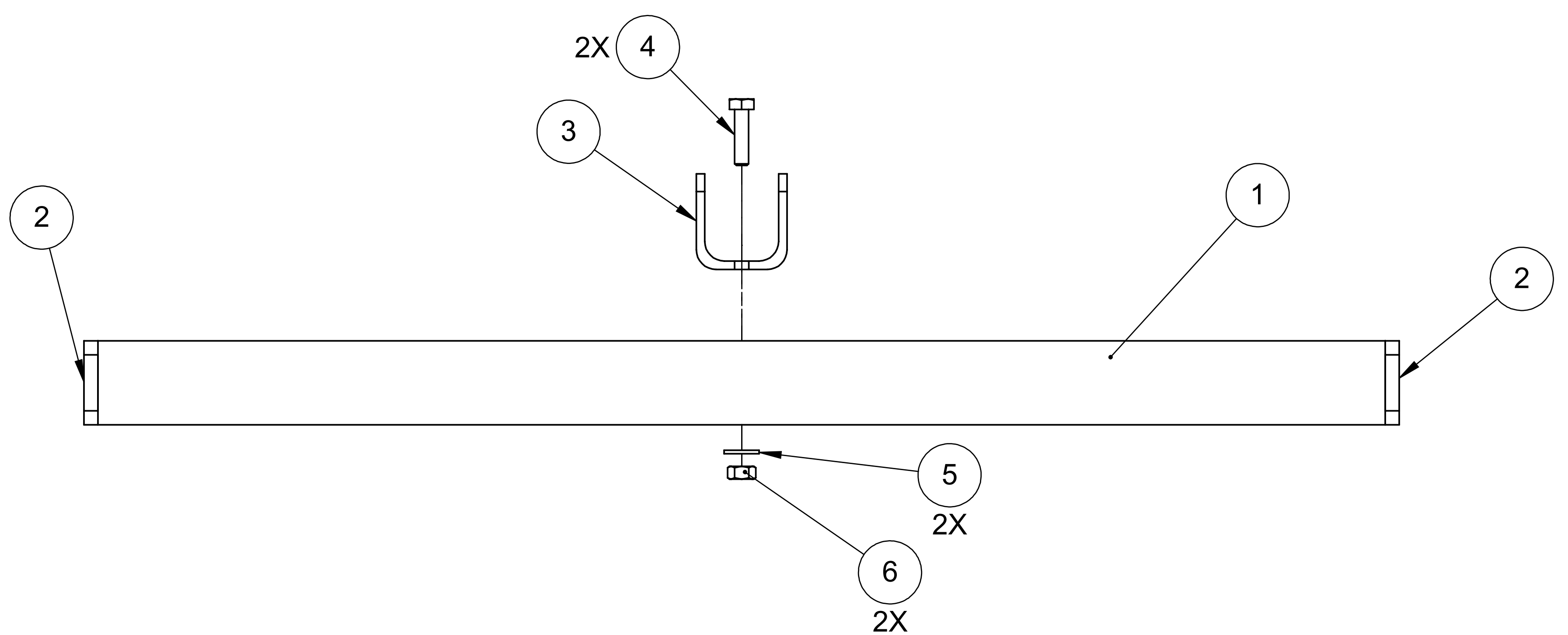
TITLE: ANGLE BRACE
1 IN. ANGLE IRON

SIZE	DWG. NO.	REV
B	LA1-3	0

SCALE: 2:3 | WEIGHT: N/A | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	LA2-1	CROSS CHANNEL BRACE, PIVOT SIDE	1
2	EP-1	END PLATE	2
3	BRK-02	MOUNT BRACKET (ACTUATORZONE)	1
4	HBOLT 0.2500-20x1x0.75-N	GRADE 8 HEX BOLT 1/4-20X1	2
5	Preferred Narrow FW 0.25	1/4 NARROW WASHER	2
6	HNUT 0.2500-20-D-N	GRADE 8 HEX NUT 1/4-20	2





REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

BOLT TORQUE
FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE

1/4-20"	143 IN-LBS
5/16"	195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

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MATERIAL

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

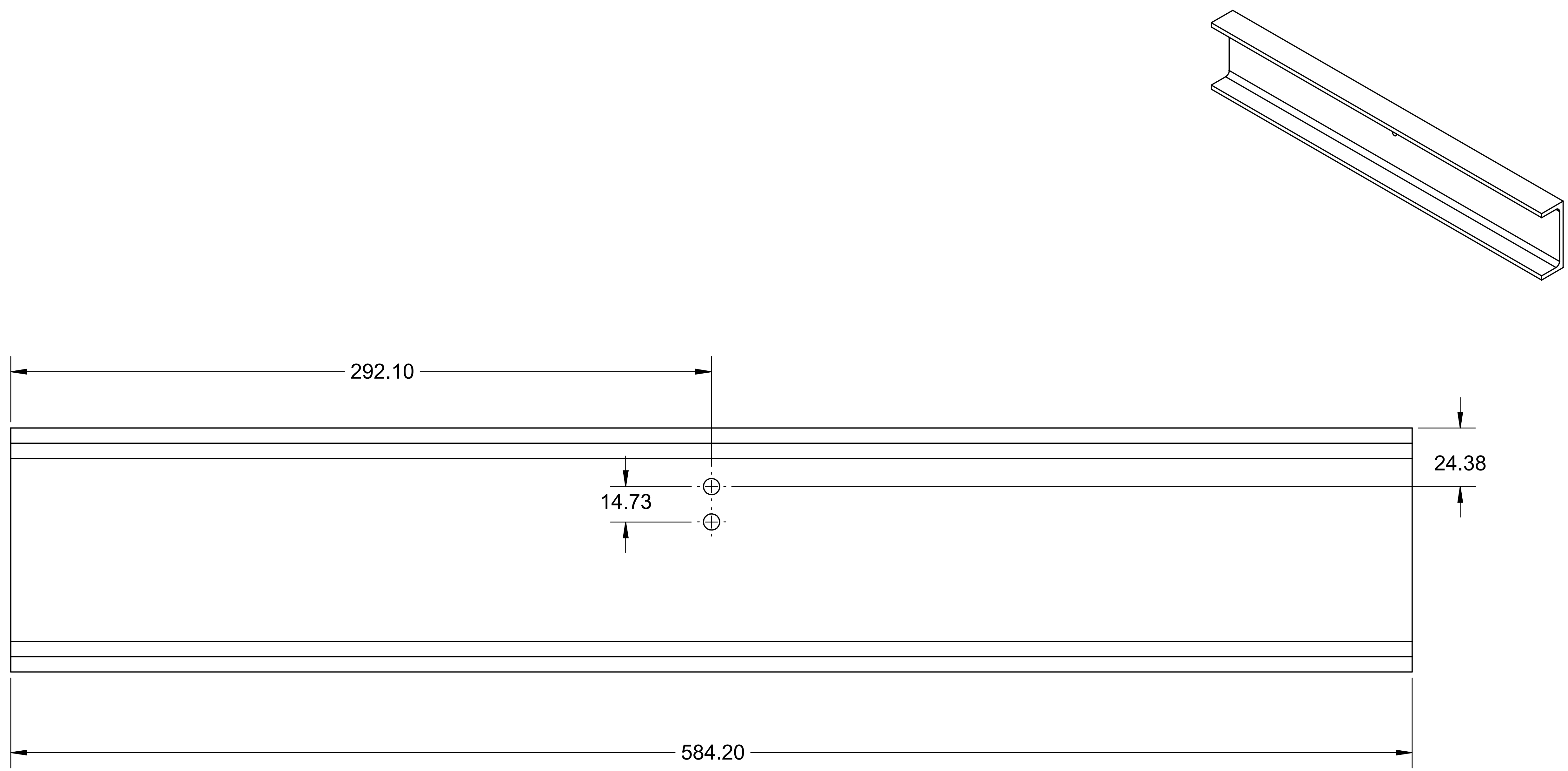
MECH 4860 - 2016 TEAM 23

TITLE:
ASSY - MOUNT, PIVOT SIDE

SIZE	DWG. NO.	REV
B	LA2-0	0

SCALE: 1:3 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. LA2-0 REV 0



REV 0

DWG. NO. LA2-1

0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.	
REV	DATE Y/M/D	DESCRIPTION			

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

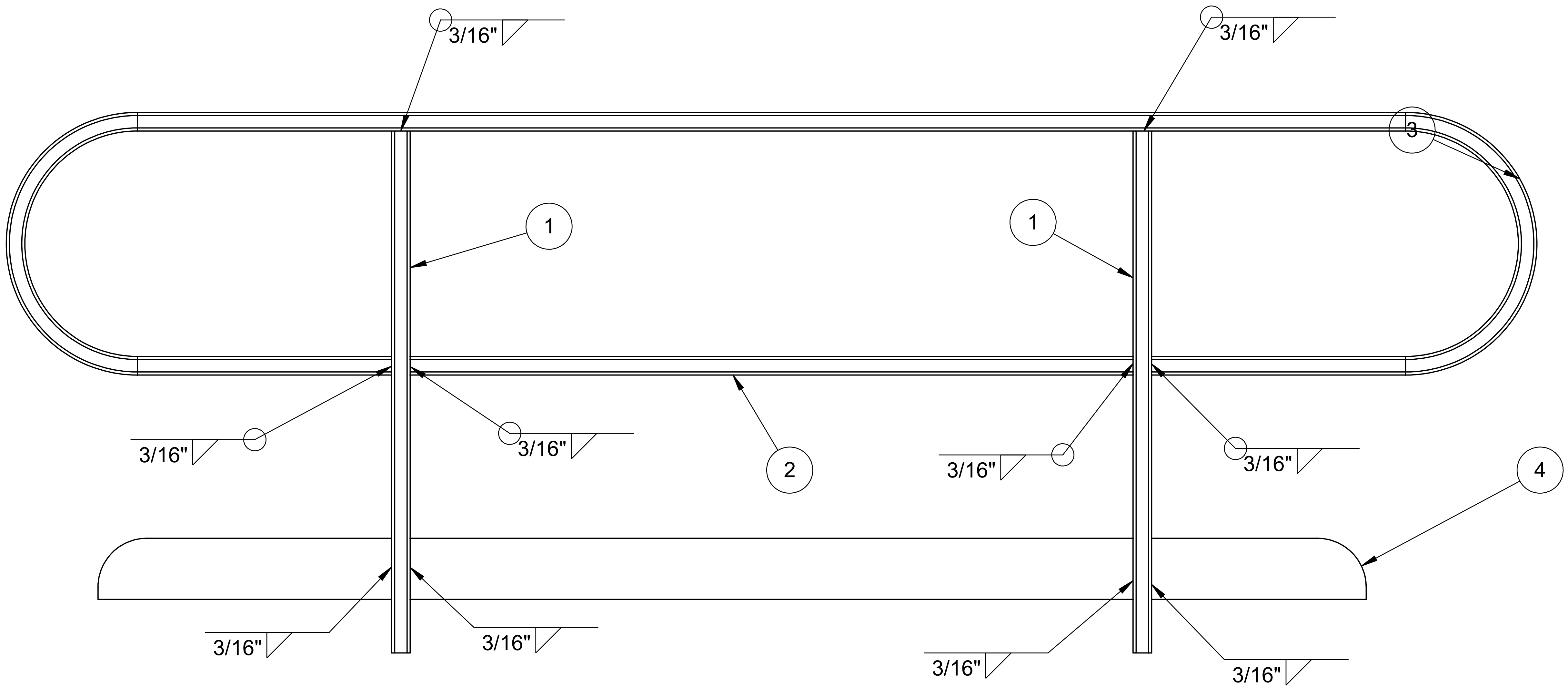
MECH 4860 - 2016 TEAM 23

TITLE:
CROSS CHANNEL BRACE, PIVOT SIDE
4 X 1.5 X .25 WEB C-CHANNEL

SIZE	DWG. NO.	REV
B	LA2-1	0

SCALE: 1:2 | WEIGHT: N/A | SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	SR1-3	RAILING, VERTICAL	2
2	SR1-2	RAILING, INTER	1
3	SR1-1	RAILING, CURVED	1
4	SR1-4	TOE BOARD	1



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

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MECH 4860 - 2016 TEAM 23

TITLE: ASSY - SAFETY RAIL

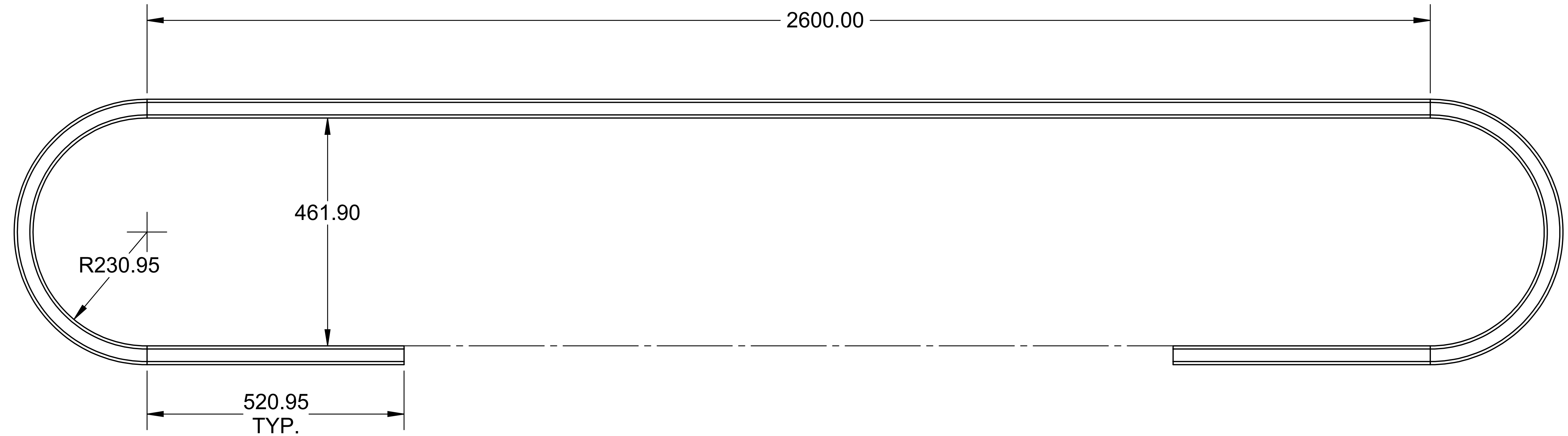
MATERIAL: 6061-T6 ALUMINUM

FINISH: NO PLATING REQUIRED

SIZE: B | DWG. NO.: SR1-0 | REV: 0

DO NOT SCALE DRAWING | SCALE: 1:10 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. SR1-0 REV 0



REV 0

DWG. NO. SR1-1

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
38 X 38 X .12 TUBE
6061 T-6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
RAILING, CURVED

SIZE **B**

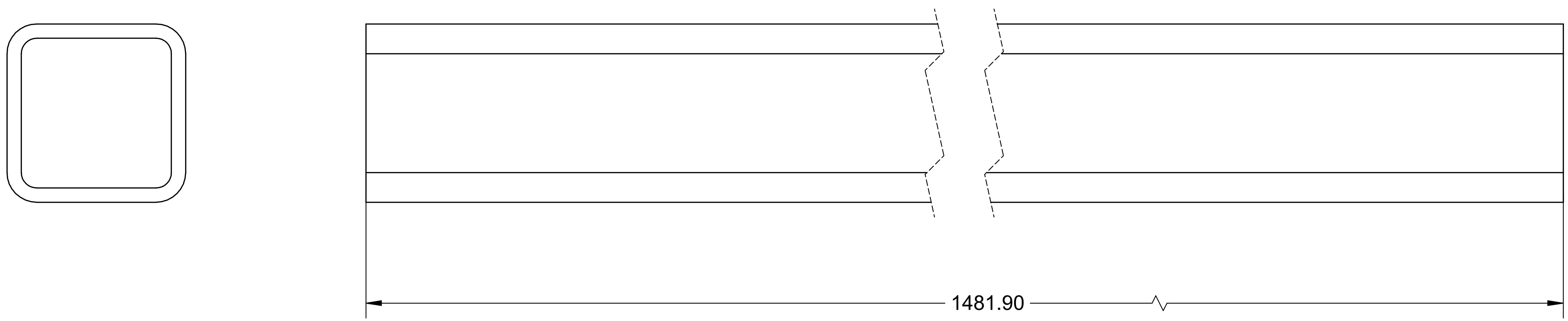
DWG. NO. SR1-1

REV 0

SCALE: 1:10

WEIGHT: N/A

SHEET 1 OF 1



			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
0	16/12/02	PRELIMINARY DWG	DWN	C.P.
			CKD	
REV	DATE Y/M/D	DESCRIPTION		

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

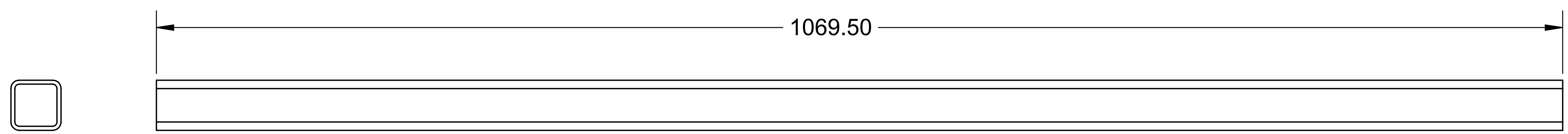
MECH 4860 - 2016 TEAM 23

TITLE:
RAILING, INTER
38 X 38 X .120 TUBE

SIZE	DWG. NO.	REV
B	SR1-2	0

SCALE: 1:1 | WEIGHT: N/A | SHEET 1 OF 1

REV 0 | DWG. NO. SR1-2



REV 0

DWG. NO. SR1-3

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
RAILING, VERTICAL
38 X 38 X .120 TUBE

SIZE B

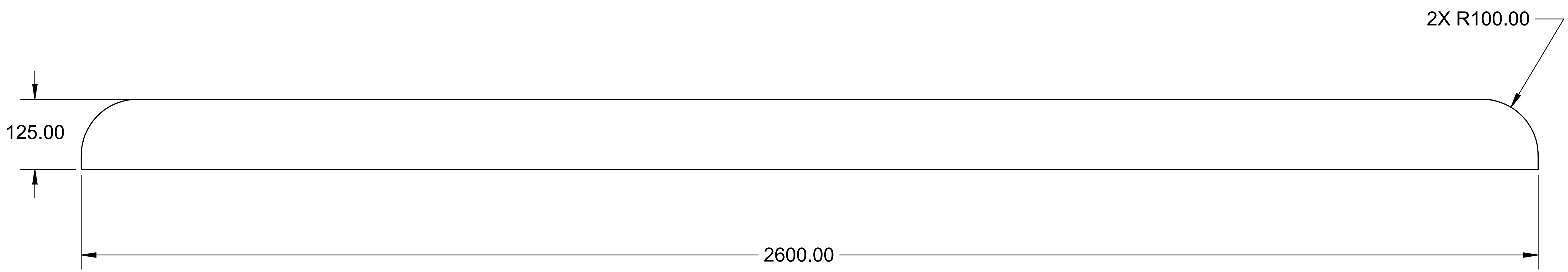
DWG. NO. SR1-3

REV 0

SCALE: 1:4

WEIGHT: N/A

SHEET 1 OF 1



			DWN		
			CKD		
			DWN		
			CKD		
			DWN		
			CKD		
			DWN		
			CKD		
0	16/12/06	PRELIMINARY DWG	DWN		C.P.
			CKD		
REV	DATE Y/M/D	DESCRIPTION			

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
0.125" SHEET 6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

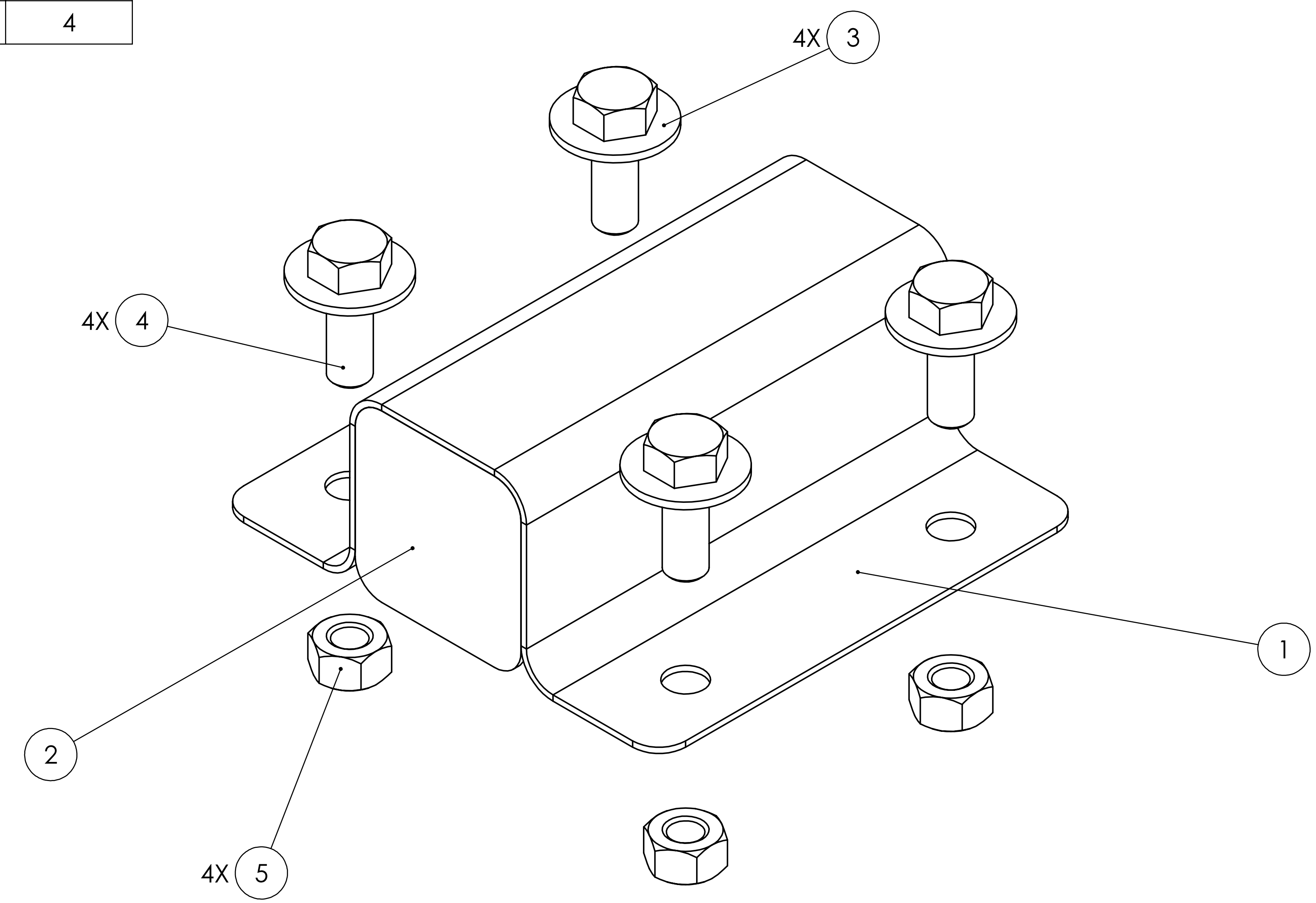
TITLE:

SIZE **B** | DWG. NO. SR1-4 | REV 0

SCALE: 1:8 | WEIGHT: N/A | SHEET 1 OF 1

REV 0 | DWG. NO. SR1-4

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	SR2-1	RAILING RECEIVER	1
2	SR2-2	BOTTOM RECEIVER PLATE	1
3	Regular FW 0.3125	5/16 FLAT WASHER	4
4	HBOLT 0.3125-18x0.875x0.875-N	GRADE 8 HEX BOLT 5/16-18X.875	4
5	HNUT 0.3125-18-D-N	GRADE 8 HEX NUT 5/16-18	4



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

BOLT TORQUE
FOR ALL BOLTS TIGHTEN TO SPECIFIED TORQUE

1/4-20"	143 IN-LBS
5/16"	195 IN-LBS

DIMENSIONAL UNIT: MILLIMETER

MATERIAL N/A

FINISH N/A

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

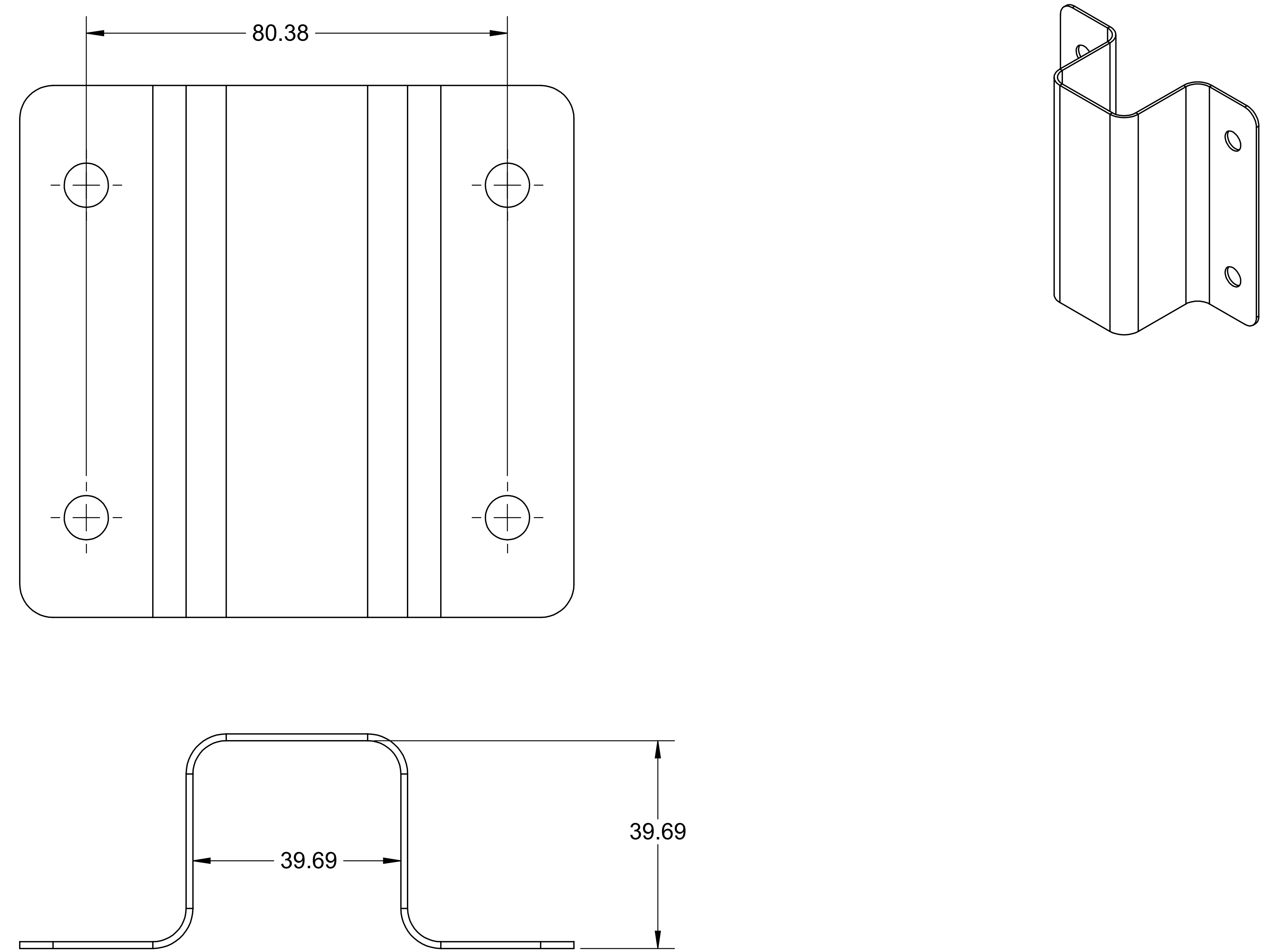
MECH 4860 - 2016 TEAM 23

TITLE:
RAILING RECEIVER ASSY

SIZE **B** | DWG. NO. SR2-0 | REV 0

SCALE: 1:1 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. SR2-0 REV 0



REV 0
SR2-1
DWG. NO.

REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL: 16GA 6061-T6 ALUMINUM

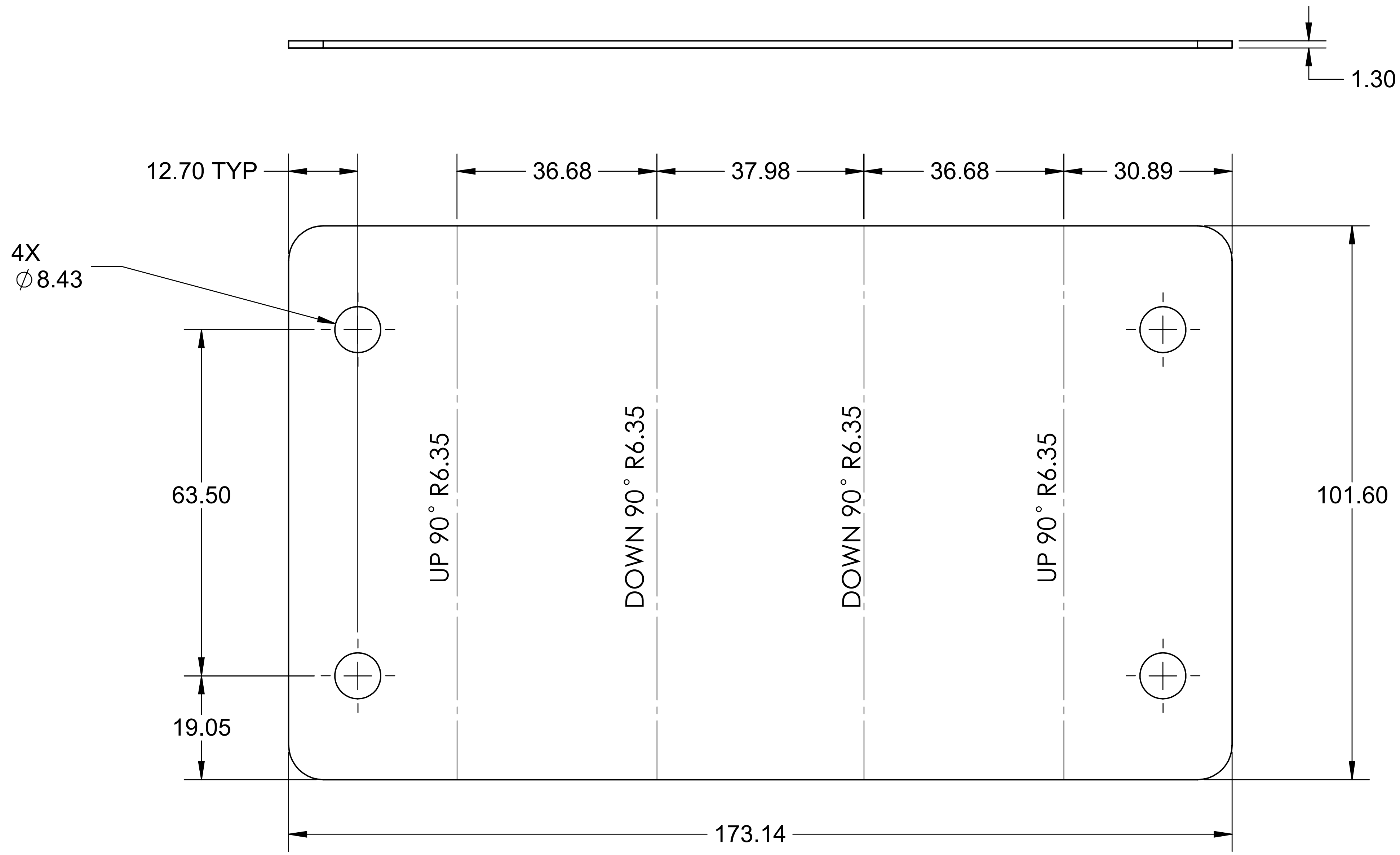
FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA
MECH 4860 - 2016 TEAM 23

TITLE:
RAILING, RECEIVER
FOLDED

SIZE: B
DWG. NO.: SR2-1
REV: 0
SCALE: 1:1
WEIGHT: N/A
SHEET 1 OF 2



			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.
REV	DATE Y/M/D	DESCRIPTION		

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
16GA 6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

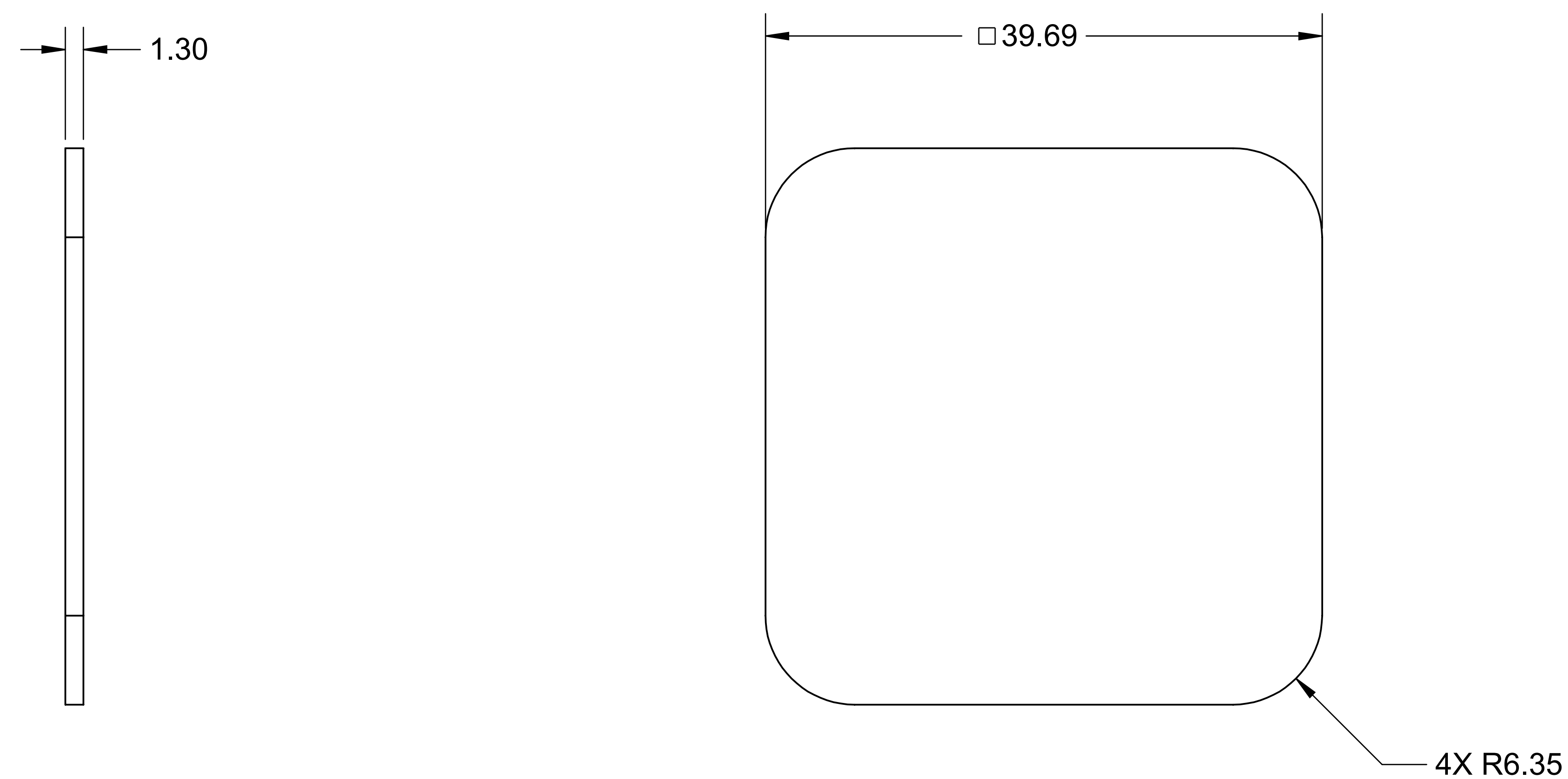
TITLE:

FLAT

SIZE	DWG. NO.	REV
B	SR2-1	0
SCALE: 1:1	WEIGHT: N/A	SHEET 2 OF 2

REV 0

DWG. NO. SR2-1



REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL: 16GA 6061-T6 ALUMINUM

FINISH: NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

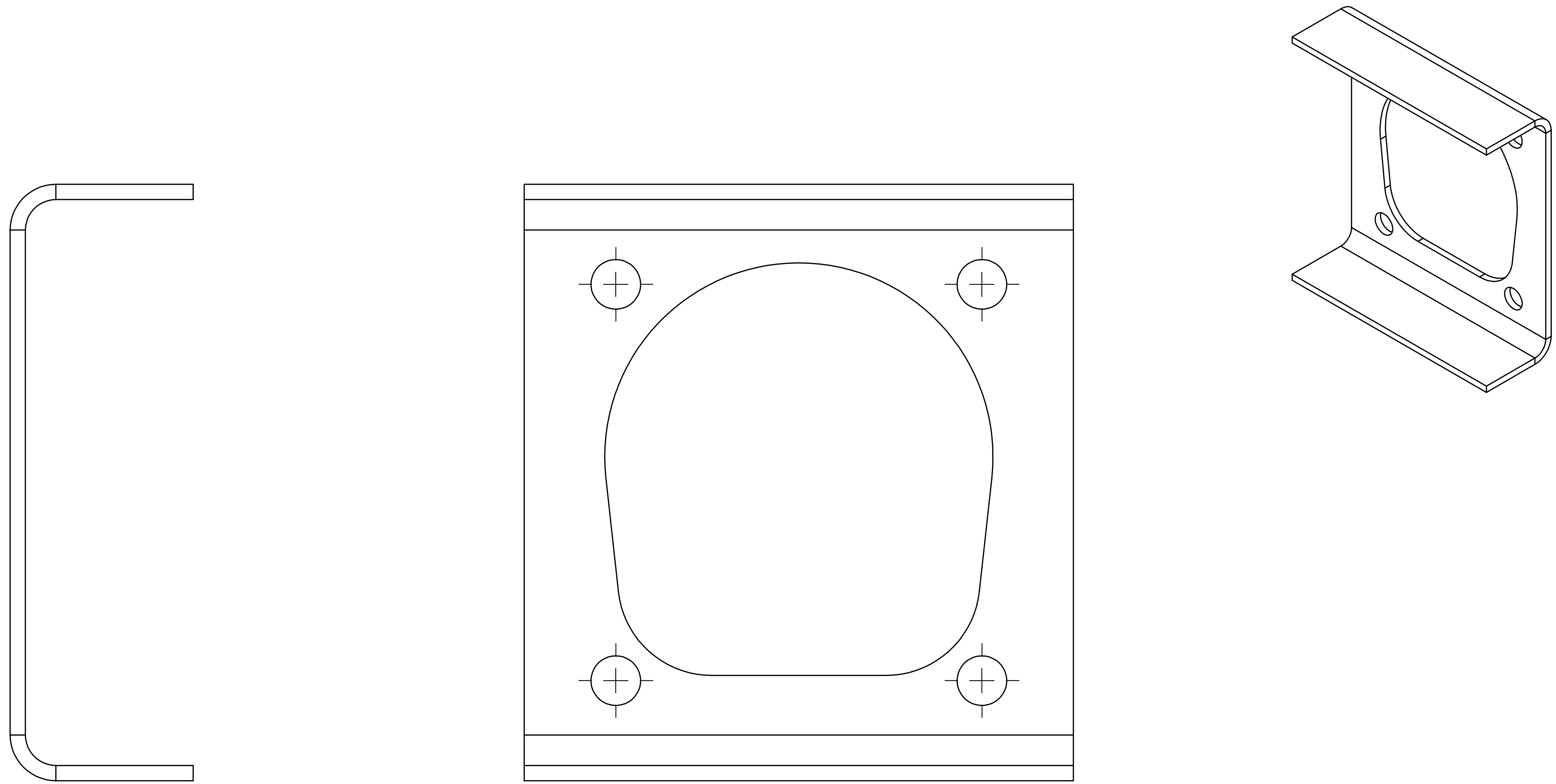
MECH 4860 - 2016 TEAM 23

TITLE: BOTTOM RECEIVER PLATE

SIZE	DWG. NO.	REV
B	SR2-2	0

SCALE: 2:1 | WEIGHT: N/A | SHEET 1 OF 1

REV 0
DWG. NO. SR2-2



DWG. NO. WM1-1 REV 0

REV	DATE Y/M/D	DESCRIPTION		
			DWN NAME #	
			CKD NAME #	
			DWN NAME #	
			CKD NAME #	
			DWN NAME #	
			CKD NAME #	
			DWN NAME #	
			CKD NAME #	
0	16/12/02	PRELIMINARY DWG	DWN NAME #	C.P.
			CKD NAME #	

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

MATERIAL
.125 IN. SHEET
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

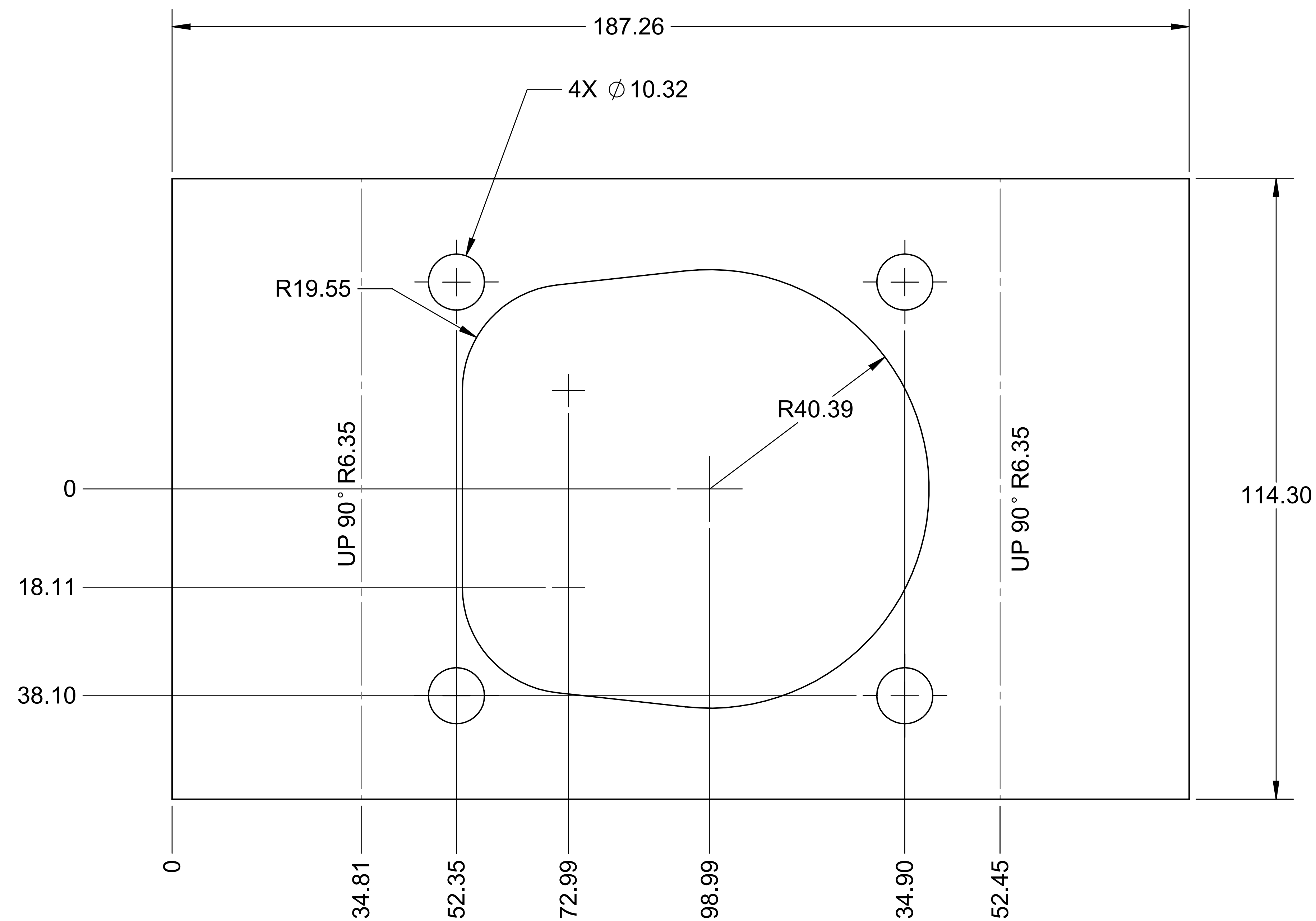
UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
BRACKET

SIZE	DWG. NO.	REV
B	WM1-1	0

SCALE: 1:1 | WEIGHT: N/A | SHEET 1 OF 2



REV	DATE Y/M/D	DESCRIPTION		
0	16/12/02	PRELIMINARY DWG	DWN NAME # / CKD NAME #	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED

ANGULAR ± 2°

LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
.125 IN. SHEET
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

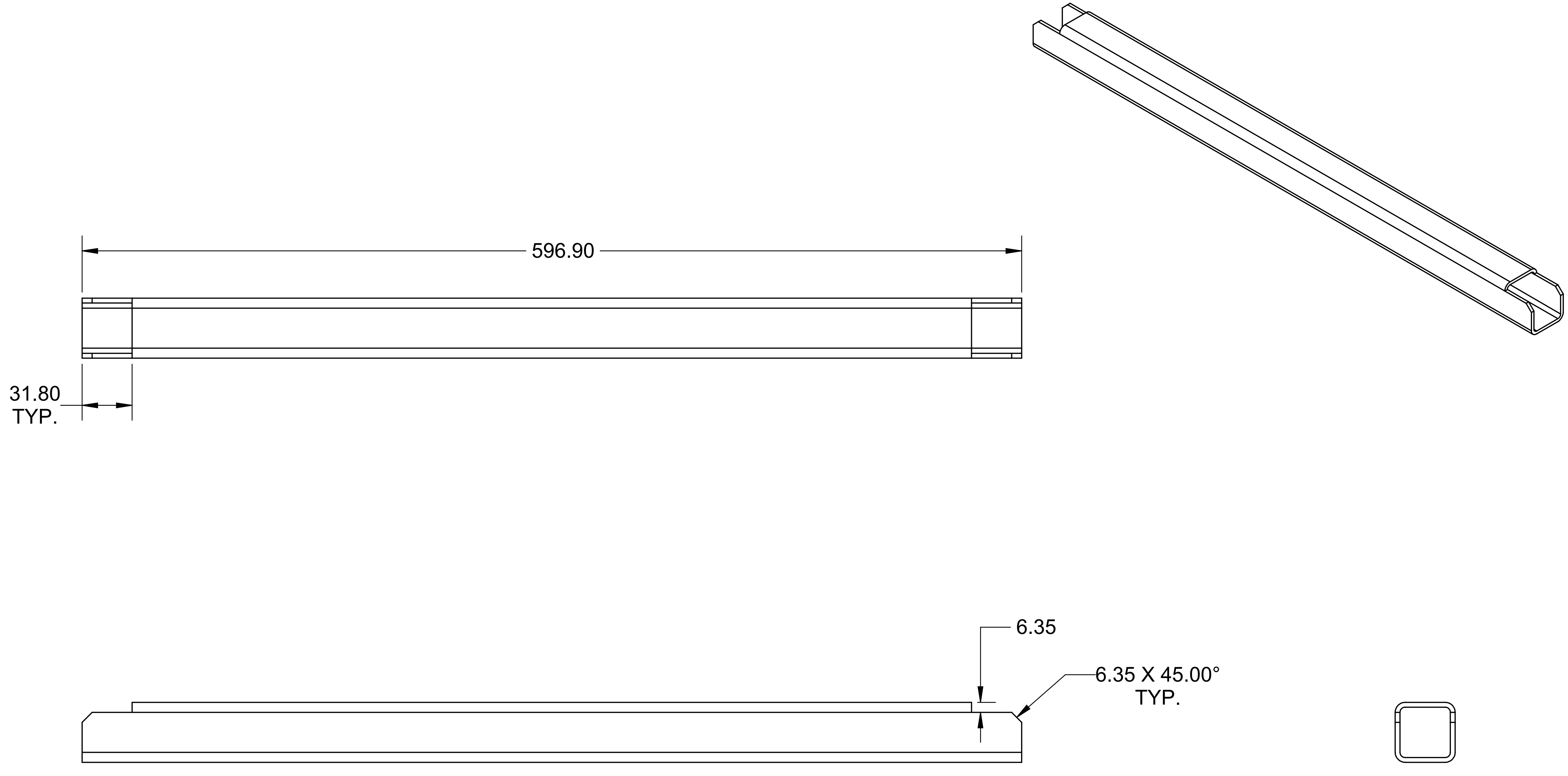
MECH 4860 - 2016 TEAM 23

TITLE:

SIZE	DWG. NO.	REV
B	WM1-1	0

SCALE: 1:1 | WEIGHT: N/A | SHEET 2 OF 2

REV 0 | DWG. NO. WM1-1



REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
0	16/12/02	PRELIMINARY DWG	DWN / CKD	C.P.

DIMENSIONAL UNIT: MILLIMETER

UNLESS OTHERWISE SPECIFIED
ANGULAR ± 2°
LINEAR
0.XX ± 0.5
0.XXX ± 0.15

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MATERIAL
6061-T6 ALUMINUM

FINISH NO PLATING REQUIRED

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 - 2016 TEAM 23

TITLE:
BRACE
38 X 38 X .120 TUBE

SIZE B | DWG. NO. WM1-2 | REV 0

SCALE: 1:3 | WEIGHT: N/A | SHEET 1 OF 1

DWG. NO. WM1-2 REV 0

7 CONCLUSIONS AND RECOMMENDATIONS

The two major objectives of this project were to develop a new gangplank system that can be deployed safely by one employee reducing risk of injury and labor resource requirements when moving RVs from workstation to workstation. The proposed V-Fold gangplank design eliminates the need for any heavy overhead lifting by automating the process of moving the gangplank out of the way. The proposed design is operated safely by one worker holding a rocker switch which powers a linear DC actuator and winch. Deployment time is estimated at 28 seconds, reduced from 40 seconds using the old manual system. The total cost per gang plank is broken down into three categories. First the power systems consisting of the linear actuator, winch, switches, and power supplies is estimated to cost $235.62 per gangplank. The second is the cost of hardware estimated to be approximately $52.30. Finally, the third cost is the raw materials required. The exact material costs for the new design are difficult to calculate as the materials used are a combination of what the client already stocks and materials that will need to be special ordered. Due to the low quantity of units being produced it is impossible to give an accurate cost of the raw materials therefore a total cost of the gangplank system is not given.

In the future, the team recommends that the client should build a prototype for life cycle testing and validation of the calculation based analysis. Additionally, the team recommends that simple inserts that match the profile of the RV's cab be created to fill in the gaps between the scaffold, gangplank, and RV itself. For the electronic components, the team recommends the client investigate methods of delaying and precisely controlling the two electric power system elements to minimize forces on the center hinge, linear actuator, and linear actuator mount points. Finally, the team recommends investigating the use of one winch per two gangplanks to minimize cost as the winch is over specified and can handle more than twice the required load.

Overall, the newly proposed design is a safe, functional, and successful solution to the client's design problem than can be easily adapted for use at all necessary workstations around the client's manufacturing facility.

8 REFERENCES

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