University of Manitoba
Faculty of Engineering

Mechanical and Manufacturing Department

MECH 4860 - Engineering Design

Aircraft Skin Handling Device for Magellan Aerospace

Final Design Report

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To Dr. Labossiere,

This is a proposal submission for a material handling device project for Magellan Aerospace entitled "Aircraft Skin Handling Device for Magellan Aerospace." Our team worked hard to research and complete the project to the best of our knowledge.

The report content provides in depth understanding of the purpose of the project and other key introductory information. It is followed by concept generation, methodology, screening and scoring analysis. The report further analyzes individual functions for the respective design to meet all objectives set by the client and the company. The report concludes with recommendations and how each objective was achieved followed by appendix sections.

Our team would like to thank Kathryn Atamanchuk and Aidan Topping for their technical and communicational support throughout the project. Their insight on ways to improve on the content and layout of the project was informative, precise, and appreciated. Our team would also like to thank yourself, Dr. Labossiere, for supporting during times of uncertainty about the project and also for information about each milestone throughout the project.

Thank you,

Regards,

Team 28
Innovision Design Team

Executive Summary

Magellan Aerospace required a material handling device for composite aircraft skins to reduce the amount of damage that occurred during manual handling and operation by the operators. The damage that occurs on the composite skin delays the production of the horizontal tail assembly and creates large overhead cost for the company.

The material handling device, as required by the clients and operators, had to transport the composite skin in the near vertical position and allow for rotation to the horizontal position for various work processes. The device also needed to support a change in the skin as the skin becomes trimmed in one of the work processes. In addition, the skin needs to be height adjustable for ergonomic purposes. Further requirements from the client were safety, economical to manufacture, sturdy, rugged and easy to maintain. The objective of this report was to design the most efficient material handling device that will meet all of the clients' requirements. This objective was achieved by establishing customer needs, developing various design concepts and optimizing the highest scored concept.

The device consists of a base frame that sits on four caster wheels. The frame has two vertical members. Attached to the members is a vertical adjustment mechanism and a rotational adjustment mechanism. An upper frame is connected to the rotational mechanism and has welded clamps. The clamps are used to mount the aircraft skin to the device.

The base frame is made from aluminum 6061-T6 tubing with an overall dimension of 60x108x50 inches. A truss design minimized the effect of bending in the members. Under maximum loading, the frame has a minimum yield stress factor of safety of 2. The height adjustment mechanism consists of a square tubing sleeve that sits outside of the vertical member

of the frame, free to slide up and down. Pin holes are placed through the frame to lock the sleeve at a height range of 34 inches to 39 inches from the ground using a ½ inch pin. An extension spring is connected to the sleeve and the frame to support the skin during vertical adjustments. The rotational mechanism consists of two bearings housed between two machined aluminum blocks. Pin holes are aligned with the housing and the upper frame to lock the skin at 180 degrees and 80 degrees from the horizontal. The upper frame assembly is made from AISI 1020 steel to support higher bending stresses that incur during work processes. The upper frames allows for ease of change between two skin types as the inner member slides along the outer member. The skin is mounted to the device using five custom made clamps. The clamps are located relative to the skin such that the skin will deflect a maximum of 0.1 inches when subjected to 25 lbf working forces. The maximum stress that the skin will experience due to an operator falling forward and pushing down on the skin has a stress factor of safety of 4.9.

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1. Introduction

Magellan Aerospace requires a device that can transport an aircraft skin throughout their facility. The material handling device needs to not only transport the skin but also allow for work to be performed on the skin. Section 1.1 will convey in depth information about the company, the project and the current process. Section 1.2 will describe the objective of needing a material handling device. Section 1.3 will analyze customer needs and section 1.4, 1.5 and 1.6 will provide in depth information about constraints, assumption, and technical specifications respectively. Section 2 discusses how concepts were generated, procedure used in analysing concepts and a brief conclusion of the results. Section 3 is the full design detail of material handling designs and its different parts. Section 4 will use a Design Failure Mode and Effect Analysis (D-FMEA) table to analyze potential failure, effects of those failure, possible solution, and prevention and detection techniques. Lastly is the conclusion and further recommendations to the company before developing prototypes and using this product.

1.1 Background Information

Magellan Aerospace is a global aerospace company that manufactures products through systematic and complex assemblies for different aerospace, defence and space agencies [1]. At the Winnipeg facility, Magellan Aerospace manufactures horizontal tail assemblies for the F-35 Lightning II. These horizontal tail assemblies are used for the conventional take-off and landing (CTOL) variant of the F-35A Lightning II Joint Strike Fighter Jet [2]. The horizontal tail assembly includes many parts, one of which is a horizontal composite skin. The material of the horizontal skin is a carbon fibre material with bismaleimide (BMI) [3] and undergoes various processes at the Winnipeg facility [4].

One process that occurs is the trimming of the composite skin to perform more processes on the smaller composite skin. The difference in size and shape between the trimmed and untrimmed skin is show in Figure 1 below. The overall height and length of the untrimmed skin is 84.64 inches and 83.9 inches, respectively, and the overall height and length of the trimmed skin is 81.45 inches and 66.55 inches, respectively. The thickness of the skin varies as it ranges from 3/16 inches thick to 1/2 inch thick.

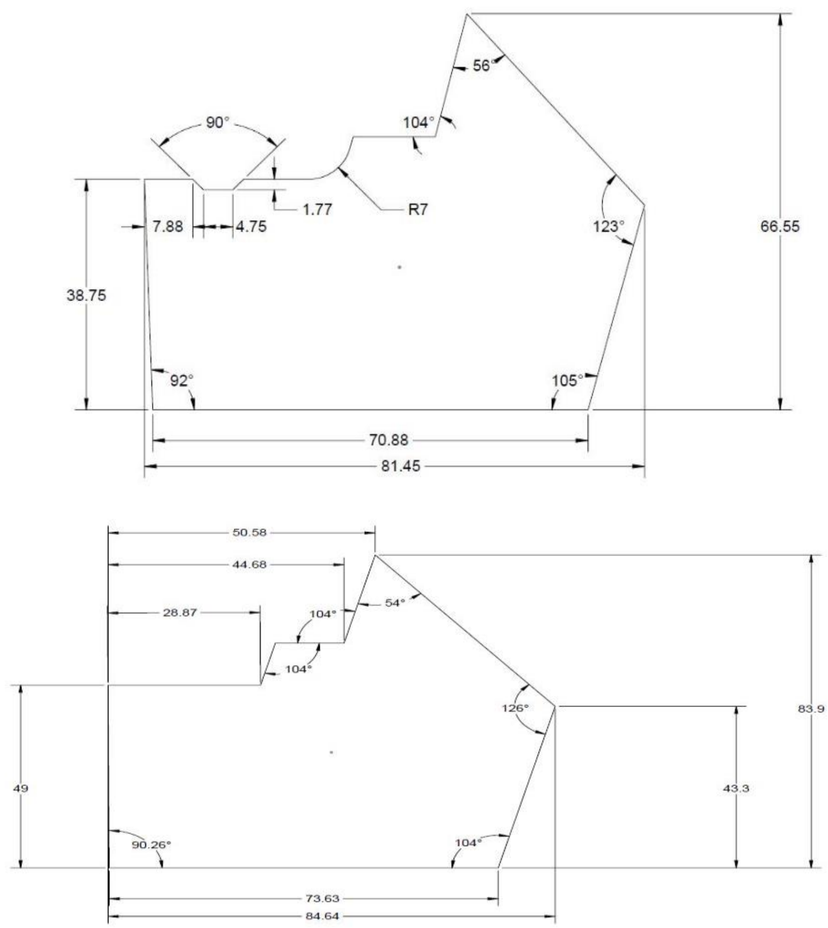

Figure 1: Sketches and dimensions of the untrimmed and trimmed horizontal composite skin respectively.

Currently, two operators handle the composite skin by manually loading and unloading onto an A-frame cart. Once transported to a specific location for a specific process, it is again unloaded from the A-frame cart, by two operators, and placed onto a worktable. The multiple manual handling of the skin results in damage of the skin from either being dropped or bumped against a table or wall. Damage to the skin slows down the completion of the horizontal tail assemblies used for the F-35 variant and results in unnecessary rework and replacement cost to the company.

1.2 Project Objectives

Our client has asked for a device that can mount and support both trimmed and untrimmed skin for transportation and work processing. The device also needs to be able to rotate the skin from the horizontal to the near vertical position to allow work to be performed. The aircraft skin handling device will reduce the amount of operator contact with the skin, thereby minimizing damage to the composite skin.

The deliverables for this project are:

- Various design concepts
- Mechanical and CAD models of the final device
- Complete bill of materials
- Stress analysis
- Stability analysis and
- D-FMEA analysis

Using a systematic process from exploring the main objectives, creating a design space with target specifications, developing concepts, mathematical and computational testing of the final product, our team will design a safe to use, rugged, easy to maintain, economical to manufacture and ergonomic material handling device.

1.3 Customer Needs Analysis

Our client has requested that the handling device adjust to both the trimmed and untrimmed forms of the skin. The handling device should transport the skin in the near vertical position and allow work to be performed in the horizontal and near vertical position only. Operators suggested a near vertical position of 80 degrees from the horizontal and the ability to adjust the vertical height in this position. The device needs to be comfortable for the operators to work efficiently and hence ergonomic standards set by Canadian federal government will be used to achieve an ergonomic design. Lastly, our client has further requested that the device is safe, rugged, easy to maintain, economical to manufacture, and aesthetically pleasing.

The rank and weight of each need was determined by comparing each need individually. The need that was the more important of the two gained a "hit" and the ranking was based on the total number of hits. The following three questions were considered when comparing the needs: What is the consequence if the need was not met? How much emphasis has the client put on the need? Which function is performed more than another?

From the three questions, we realized that there is a high negative consequence if there is damage to property and injuries to personnel. Thus, a high precedence was given to safety. It was realized that functionality was the next second to safety in which the client has emphasized the need to be able to transport in the vertical position and allow for work to be performed at different positions. The weight of each need is determined by the following equation:

$$Weight = \frac{\#\ of\ hits}{sum\ of\ all\ hits\ from\ all\ needs} \qquad \text{Eq. 1}$$

The list of needs, associative ranks and weights are shown in TABLE , below.

TABLE I: LIST OF CUSTOMER NEEDS IN PROPER RANKING AND WEIGHTING

Rank	Needs	Weight
1	Device is safe to use and prevents injuries to users.	6.90
2	Device prevents damage to the skin during transportation and all work functions performed by operators.	6.65
3	Devices does not cause excessive strain to the skin while stationary, lifting or rotating.	6.40
4	Device does not fall over with/without skin.	6.16
5	Device supports weight of the skin and maximum external loading of 30-pound force.	5.91
6	Device adjusts to dimensional variation of skin; trimmed and untrimmed skin.	5.67
7	Device has controlled movements	5.42
8	Device should rotate.	5.17
9	Device keeps skin stable when in working position.	4.93
10	Device is able to work at ergonomic position for range of workers (height, length, and width).	4.68
11	Device keeps skin stable and firm during transportation.	4.43
12	Device experiences minimal deflections when subjected to external loads.	4.19
13	Device is stationary when in working position.	3.94
14	Device is portable with or without a skin.	3.69
15	Device maximizes all working surface while in working position.	3.45
16	Device is easy and efficient to load and operate.	3.20
17	Device has a long lifespan.	2.96
18	Devices’ material is based on environmental condition.	2.71
19	Movement requires minimal force and motions are smooth.	2.46
20	Device is designed such that professional engineering processes can be used to manufacture it.	2.22
21	Device is comfortable for operators to work around and have ample room at any one location	1.97
22	Device is a simple design.	1.72
23	Device is easy to maintain.	1.48
24	Device minimizes footprint.	1.23
25	Device can manoeuvre through tight corners (and pivot like a chalkboard).	0.99
26	Device is pleasing to the eyes.	0.74
27	Device does not have unpleasant noise while performing any function.	0.49
28	Device has reasonable cost to manufacture.	0.25
29	Device design minimizes environmental impact.	0.00

1.4 Constraints Analysis

Magellan Aerospace does not allow our team to disclose work processes techniques that Magellan uses to manufacture the F-35A horizontal composite skin. In addition, information for specifications of test data of the composite skin is not allowed to be disclosed and only information publicly available sources is used in the analysis. Lastly, our team is not allowed to disclose environmental conditions the skin and material handling device will be exposed to. These constraints restrict the design space that is used to generate concept designs. Any designs that exceeds the design space will be eliminated. The complete list of constraints is provided in TABLE below.

TABLE II: DESIGN CONSTRAINTS FOR THE MATERIAL-HANDLING DEVICE

#	**Constraints**
1	Design is to be accomplished by December 6th 2017.
2	Four team members are working on the design of the device.
3	Maximum lifting force for operators is 50 pounds, set by union rules.
4	Device produces a deflection of 0mm on the skin with a maximum of 1mm.
5	Device material choice is biocompatible with operators and prevents health issues such as skin irritation, inhalation problems and etc.
6	The extent of testing will only be performed through computational analysis. Further prototyping and physical testing will not be performed in order to understand possible failures.
7	Due to security restriction of proprietary information of the part, our team is unable to:
a	Discuss any processes the skin goes through
b	Discuss any environments that the skin may be exposed to
c	Acquire and have access to any specifications or test data of the skin (e.g. density, tensile strength etc.)

1.5 Assumption Analysis

A list of assumptions illustrated how our team approached the design of the handling device based on the constraints. As our team is not allowed to discuss work processes that occur at Magellan and not allowed to acquire actual data of the composite skin, assumptions will be made to oversee all possible worst-case scenarios for work processes and a product that is commonly used for aircraft skins for actual data of composite skin. TABLE , below, lists an in-depth description of each assumption that our team assumed to design the handling device.

TABLE III: DESIGN ASSUMPTIONS ACQUIRED AFTER ANALYZING CONSTRAINTS AND THE APPROACH TO DESIGN THE HANDLNIG DEVICE

#	Assumptions
1	Material handling device will be designed for worst case scenarios and still be feasible.
2	Weight of the skin will use a carbon fibre with the highest density and the skin is the volume of the enclosure
3	Tensile strength of the skin will be used with the weakest material properties of carbon fiber in order to incorporate and gage strain of skin that is less than 2 inches enclosure.
4	Operators will require 100% access to surface of the skin as we are not allowed to know the type of work that is performed on the skin.

1.6 Technical Specifications Analysis

Each of the needs from section 1.3 were provided measurements or values in order to set ideal and marginal targets. Thirty-eight metrics were established and shown in TABLE , below. The current process specifications for the A-frame cart is also provided in the TABLE . Target specifications were assigned to overcome values of the A-frame and create a better handling device.

TABLE IV: TARGET SPECIFICATIONS AND CURRENT PROCESS SPECIFICATIONS

Metric	Need #	Units	Ideal Value	Marginal Value	A-Frame
Number of crush points	1	#	0	0	0
Laceration points	1	#	0	0	0
Local Deformation at mounting points	2	Binary	None	None	None
Skin stress	3	Pascal	YSs	YSs	0

Metric	Need #	Units	Ideal Value	Marginal Value	A-Frame
			(FS=5)	(FS=3)	
Device is upright at all times	4	Binary	Yes	Yes	Yes
Maximum stress on device	5, 20	Pascal	YSd (FS=5)	YSd (FS=3)	0
Both skin types are compatible with device	6	Binary	Yes	Yes	Yes
Time to complete full rotation	7, 16	Seconds	3	5	N/A
Maximum force required to prevent movement in the undesirable direction	7	Newton	225	250	Unknown
Time to complete height adjustments	7	Seconds	15	20	N/A
Ease of applying stopping (slowing down) force	7	Subjective	Easy	Easy	Moderate
Range of rotation	8	Degree	90	90	N/A
Range of motion of all co-ordinate frame for working position (tolerance)	9	x,y,z (mm)	0	1	N/A
Allowable change in angle at position	9	Degree	0	1	N/A
Design allows for agronomical mounting for user	1, 10	Binary	Yes	Yes	No
Height adjustment	10	m	0.7-1.1	0.8-1.0	N/A
Range of motion of all co-ordinate frame for transfer position (tolerance)	11	x, y, z, pheta, phi	0	1	N/A
Device protects skin on all side upon collision	12	Binary	Yes	Yes	Yes
Maximum deflection under external load	12	mm	0	1	N/A
Maximum force required to move device when in working position	13	Newton	500	300	N/A
Device is portable	14	Binary	Yes	Yes	Yes
Maximize working surface area	15	ft^2	9.5	9	N/A
Time to setup and load skin onto device	16, 22	Seconds	45	60	45
Time required before device fails	17, 20	Years	40	10	40
Amount of visible corrosion on device over lifespan	18	Binary	None	Moderate	None
Force required to adjust height position	19	Binary	Minimal	Moderate	N/A
Force required to make rotational adjustment	19	Binary	Minimal	Moderate	N/A
Force required to start moving device with/without skin	19	Binary	Minimal	Moderate	Moderate
Workable surface that the worker can access	21	ft^2	10	9	N/A
Hours to replace/fix/maintain any one skin of device	22, 23	Hours	0.5	1	N/A
Specialized maintenance tools	23	List	1	3	N/A
Maximum dimensions of device	24	m x m x m	1.5 x 2 x 3	2 x 2 x 3	1.75 x 1.14 x 2.49
Device rotates about center of vertical axis	25	Degree	360	360	360
Sound during usage of device, include all motion	26	dB	20	60	20
Manufacturing cost of one unit (standardization)	20, 22, 27	US$	2,000	10,000	$2,000
Manufacturing process has minimal environmental impact	28	Binary	Yes	Yes	N/A
Materials recyclable	28	Binary	Yes	Yes	yes
Device is aesthetically pleasing	29	Binary	Pass	Pass	Fail

The ideal target specifications in the table above were set to improve the specifications of the current process. There are no stresses or strains caused by the A-frame cart on the skin, which results in a non-applicable value for most of the metrics.

As the design space is fully defined based on the objectives, needs, constraints, assumptions and technical specifications, concepts were generated and analyzed, in section 2, to come one step closer in reaching the goal of designing a material handling device that is easy to use, rugged and ergonomic, and safe to use.

2. Concept Generation and Analysis

The handling device requires several functions such as transportation, rotation, height adjustment and the ability to support untrimmed and trimmed skin. Section 2.1 describes the steps before concept generation began. All concepts are in Appendix A, for reference and section 2.2 describes the result of all the concepts and which final concept was chosen for detail design.

2.1 Concept Generation Methodology

The desired functions of the overall design were separated and listed in a morph chart. The chart provides either a method, mechanism or assembly of mechanisms that could be used to satisfy the desired function. Each of the components listed were acquired from patent searches or components of products used by competitors. The major sections of the morph chart are transportation, frame, rotational mechanism and vertical mechanism. After a full morph was constructed, it was later refined to remove concepts that were not practical and a modified morph chart was created which is shown in Appendix A. From the modified morph chart, concepts were generated which could be found in Appendix A.

2.2 Concept Analysis Result from Screening and Scoring

Screening was performed on all preliminary designs using twelve criteria extracted from the top needs. The material handling device needs to improve the current handling and work process of the composite skin through the use of the A-frame cart and hence that A-frame cart was used as a reference to compare all the preliminary designs. The A-frame cart was given a zero grading for each objective and each design was given a plus if it exceeded the A-frame cart, a negative if it failed or a zero if remained the same against the A-frame cart. The top five concepts that passed the screening phase are truss frame with steering wheel control, Triangular Frame with Sliding Bars, External Truss Pull to Turn, Pulley Lift Design and Circular Mounting Gear Frame on Vertical Bars. The full analysis of the screening phase is described in Appendix A.

The five concepts were then scored in order to determine a winner. There was no reference design that was scored against; rather every design was compared with each other for each need. A rating value of five was given if a design performed well for a given objective and a number one rating if it performed terribly. Using weighted scoring of the need, a total score was given to all five concepts. The number one rank from the scoring analysis was the truss frame with steering wheel control design. A full description of the scoring table and methodology can be found in Appendix A.

The truss frame with steering wheel control was further optimized in order to increase the scoring of the design. Each function that needed to be modified was compared to another potential design and was scored. The full analysis of each individual function is in Appendix A. The result is a modified truss frame design with spring mechanism for vertical translation and a frame with clamps to expose both sides of the skin for work processes. Section 3 will describe in further detail the overall design and each function of the design.

3. Detailed Design Analysis

The final design comprises of a triangular base frame with rotational arms that can rotate the composite skin. The rotational arms uses clamps to attach to the device and uses a spring mechanism to adjust to various height. Section 3.1 and 3.2 will analyze bottom truss frame and rotational mechanism respectively. Section 3.3 and 3.5 will discuss size adjustability and vertical adjustability respectively. Section 3.4 will discuss the skin to frame interface using clamping system. Section 3.8 will summarize the whole device by comparing the technical specification that this device achieves compared to the ideal values as shown in TABLE , above. Section 3.9 will review the complete bill of material. Below in Figure 2 is a render of the final design.

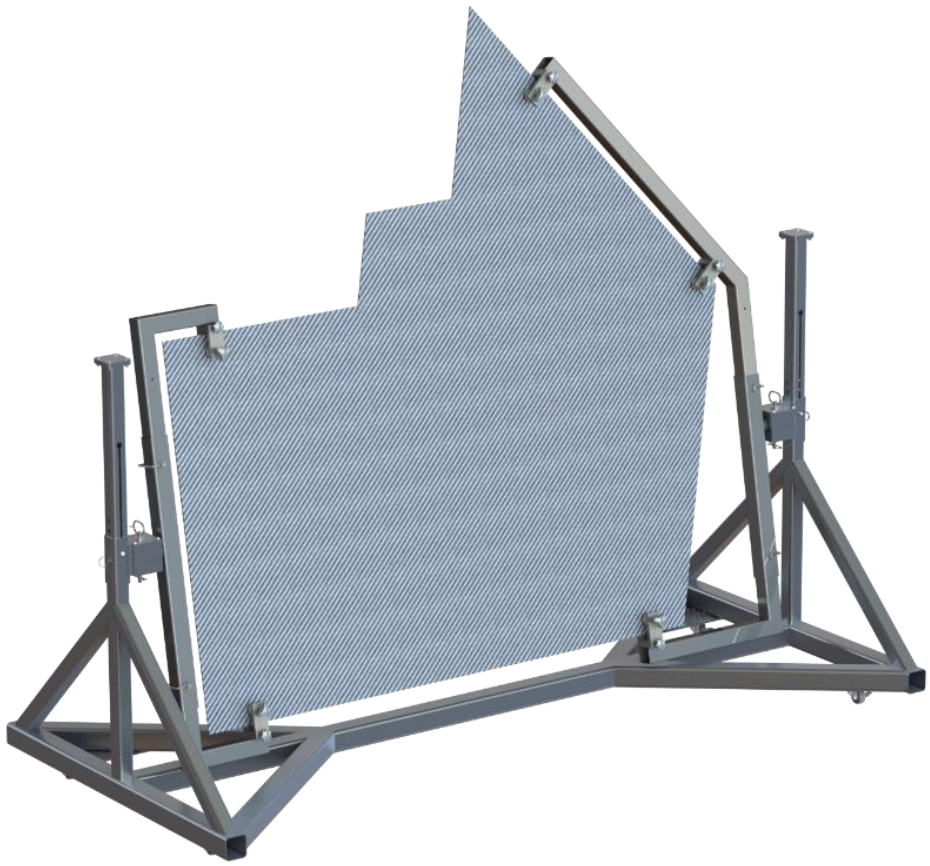

Figure 2: Finalized Design Concept for Ergonomic movement of Aircraft Skins

3.1 Base Frame Assembly

The starting point for the final design was setting the design space constraints. These constraints would lead to the minimum spatial geometry that the bottom frame would need to achieve. The largest contributing factors to these constraints are

- size of the skin
- height of the skin in the working position and
- width of the device for stability purposes.

With the skin having two configurations, untrimmed (largest size) and trimmed (smaller size) the bottom frame was designed to fit the untrimmed size with adjustment being made in the upper part of the frame.

The skin working height position was to be a minimum of 34 inches and a maximum of 39 inches. This allows the skin to be moved to an ergonomic height for the employee working with the skin.

The last constraint was the width of the device as to resist the motion of the device tipping over in the event of a force being applied horizontally on the skin. Below in Figure 3 is a rendering of the bottom frame.

Figure 3: Render of the bottom frame portion of the device

This frame is made of 6061-T6 extruded aluminum tubing and has two main cross sections to the frame. The lower portion highlighted in Figure 4 below has a square 3 inch cross section with a 0.125 inches wall thickness. The vertical members highlighted in Figure 5 below have a square 2.25-inch cross section with a 0.125 inches wall thickness. The engineering drawings for the frame structure is found in Appendix F with tolerances added respectively.

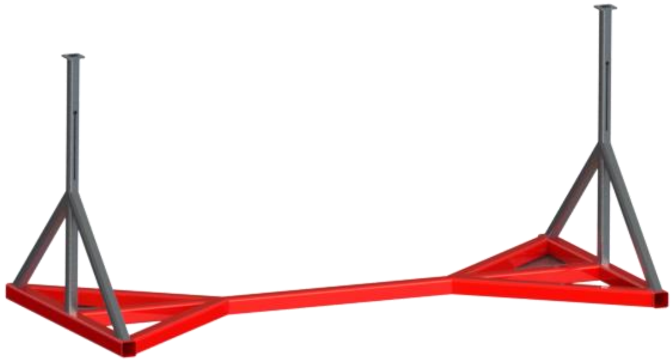

Figure 4: Bottom frame showing highlight to horizontal members

Figure 5: Bottom frame showing highlight to vertical members

Two different cross sections were chosen to withstand the loading scenarios, while going with minimum sized members. To validate the frame, preliminary Finite Element Analysis was performed for each of the loading scenarios our team developed. The results of these simulations can be found in Appendix C.

Since the device is portable, the wheels used to transport were mounted on the bottom plane of the frame. A 0.25-inch-thick aluminum plate was placed in each corner to give a mounting location for the caster. This plate is to be welded directly to the frame to ensure the plate will not shift under any foreseeable loading scenario. As seen below in Figure 6 the caster is located within the bottom frame to avoid a tripping hazard when the workers are loading the skin.

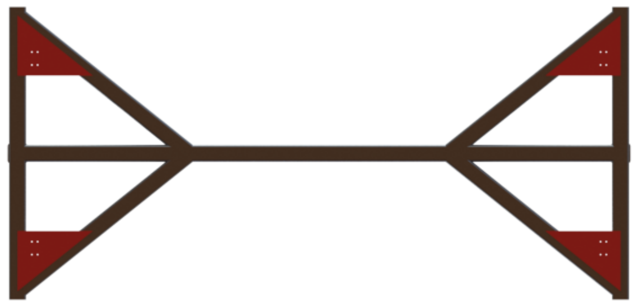

Figure 6: Highlight of caster plates

3.2 Rotational Mechanism

The rotational assembly includes many components that work in unison to allow the skin to rotate about a center axis. The rotational assembly is welded to the sleeve, which is mounted to the vertical members on the bottom frame. The sleeve will be discussed in Section 3.5. Figure 7 below, depicts each component and a render of the rotational assembly of the final design.

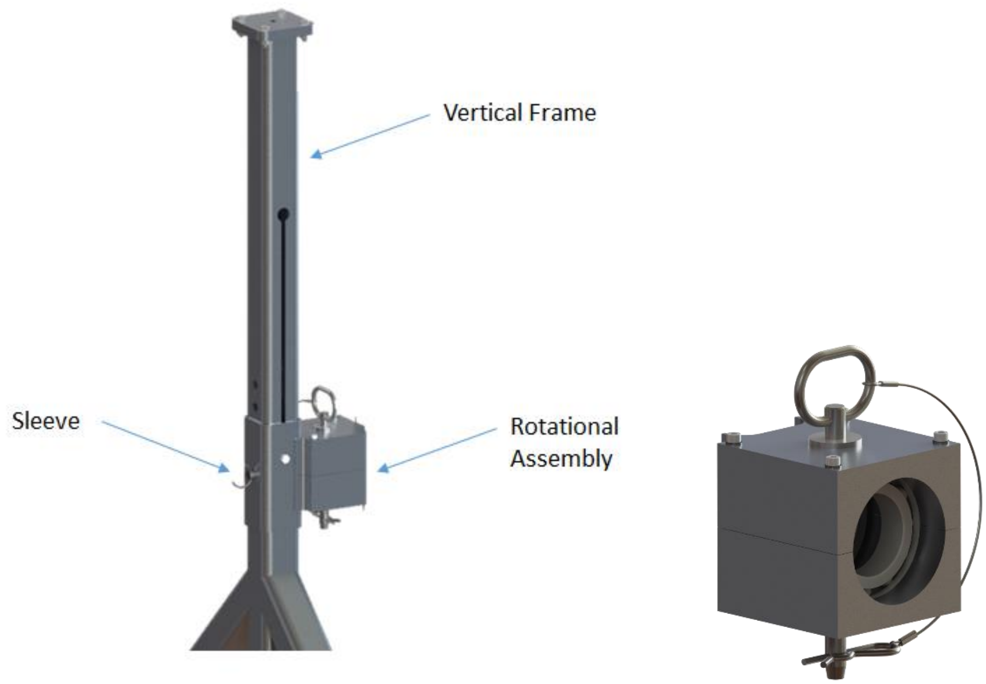

Figure 7: Rotational assembly of final design

The primary part of the rotational assembly is the aluminum housing that encompasses the bearings and locating hole for the rotating upper frame. The housing is split in two halves with the hole threading and locating holes being the minor differences between the two. The housing is designed to be manufactured with a CNC machine, as there are tolerances for the bearing surfaces and pin that require precision. The tolerance and the full drawings are listed and shown in Appendix I. This part can be seen below in Figure 8.

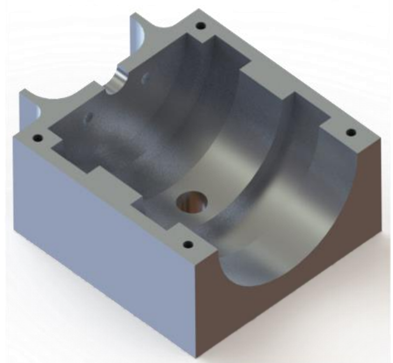

Figure 8: Isometric view of the lower half rotational block

There are some notable features which allow for ease of manufacturing and assembly. One of these features is the two M5 holes located on the back side of the block. The holes are designed to line up with adjacent holes on the sleeve component. Using two M5 bolts, the holes can be aligned for welding the bottom block to the slider sleeve. The top half aluminum block does not have these holes as it is a removable component for assembly purposes. Figure 9, below, is the pictorially description of the above description of the alignment between the rotational block and the slider component.

Figure 9: Isometric view of slider sleeve and lower rotational block

Other than the difference in the bottom block having locating holes, the holes used to fasten the two halves together are machined differently. The top block has straight through clearance holes while the bottom is threaded for an M5x0.8 bolt. This feature allows the M5 bolt to be partially threaded and for the worker assembling the component to not have to tighten the bolt the whole length of the two blocks. Figure 10, below, shows the exploded view of the M5 bolt with the top and bottom rotational block. Full drawings of the whole component are shown in APPENDIX F.

Figure 10: Top and bottom block of the rotational assembly with M5 bolt alignment

The two large cut-outs seen in Figure 8 above are used to position the deep groove bearings. These bearings were chosen to fit the outer diameter of the upper frame tube and be able to withstand the vertical loading they are exposed to. Two bearings helps to distribute the load and keep the round tube of the upper frame aligned horizontally to the floor. An open style bearing was used because dust contamination is not an issue with the application of this device and allows for a cheaper part. After the bottom housing is welded to the slider sleeve, the bearings can be placed inside, and the top housing attached. By using four M5 bolts the two housings can be tightened together.

In order to lock the rotation of the skin and thus the upper frame, a single 0.5-inch locking pin is used on each side of the device. This pin is made of cold rolled steel with a zinc coating for corrosion resistance. By having two pins placed on both sides of the device, it holds the skin in place no matter the applied load location. The dimension of the pin was based on a worst-case scenario, where a 125lbs load was applied on the skin at the furthest location for the rotating axis. With the material properties of cold rolled steel and a factor of safety of two, the pin will not shear under the load scenario. The full analysis of choosing the locking pin can be found in APPENDIX C.

3.3 Upper Frame Assembly

The upper frame of the device is made of AISI 1020 steel to give the members a higher tensile than that of 6061-T6 aluminum. With worst case loading scenarios, these members can see large bending stresses which the lower strength aluminum cannot handle. Adjustment to the different skin sizes is made with coinciding square members. By having the square cross sections, the members are constrained to move in a single direction.

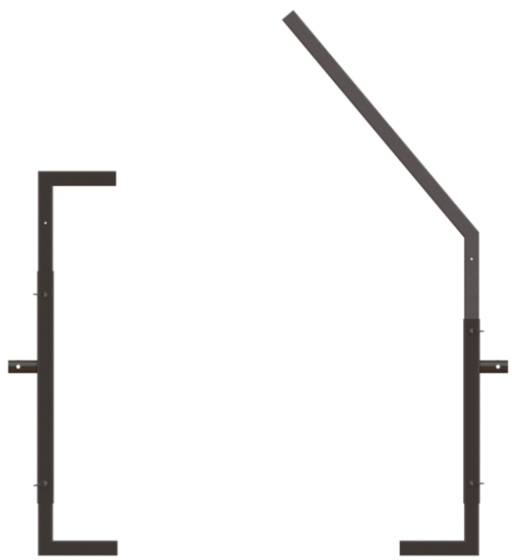

Figure 11: Upper frame members used to hold the skin in place

The larger cross section member has an outer diameter of 2.25 inches and a wall thickness of 0.125 inches. With the outer diameter of the smaller tube being less than the inner diameter of the larger tube they can slide within one another. Figure 12 below shows the two tubes coincident with one another and APPENDIX F is the drawings of the upper frame members.

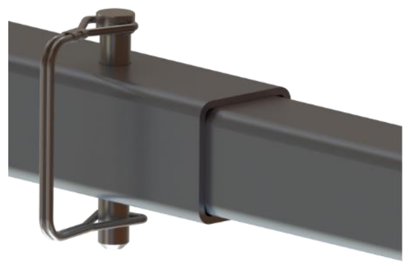

Figure 12: Detail illustration of the upper frame members assembly

As discussed before in the rotational assembly, there is a circular member that slides into the rotational block perpendicularly. This circular tube has an outer diameter of 1.967 inches to tightly fit in the inner race of the bearings. To achieve this diameter, a 2 inches diameter steel tube with a 0.25 inches wall thickness was chosen. The tube must then be lathed down to the correct diameter and the laser cut profile to follow. By doing this procedure, a proper fitting for the interaction between the tube and bearings is achieved.

In order to simplify the position of the adjustable arms, there are six locating holes. The bottom arms, highlighted in Figure 13 below, do not have multiple hole locations but rather one. By having only one-hole location the worker does not have to worry about adjusting one side prior to loading. Also, the length of the arms is designed to keep the centre of mass closest to the axis of rotation in both the trimmed and untrimmed configurations. This helps to balance the skin when rotating to a new position. To secure the bottom arms at their specified location, laser cut holes are cut in the tubes and then fastened with a pin for improved install and assembly. The locking pin used to secure the bottoms arms is a 0.25-inch steel pin with retainer so that the pin will not fall out when rotating. This diameter was chosen because of its ability to withstand the load of the skin in the vertical position and any expected load it may be exposed to.

Figure 13: Fixed bottom members of the upper frame assembly

The upper arms seen in Figure 14 below, support the skin in the same way the lower arms do. There are two differences however and that is the number of position holes and the geometry. There are two locating holes on the tube that allow adjustment between the two skins. With only two holes, the worker can easily identify which hole the pin should be placed in to get the arm at the correct length for the skin. With one side of the skin being angled, an angled support arm was designed to best match the profile and support the skin as close as possible.

Figure 14: Adjustable top members of the upper frame assembly

3.4 Skin to Frame Interface

Two key components need to be optimized for the interaction between the composite skin and the clamps. The first is the clamps and the section is the location that the clamps will engage onto the skin.

3.4.1 Clamping Mechanism

A final render of the clamping mechanism is shown in Figure 15 below.

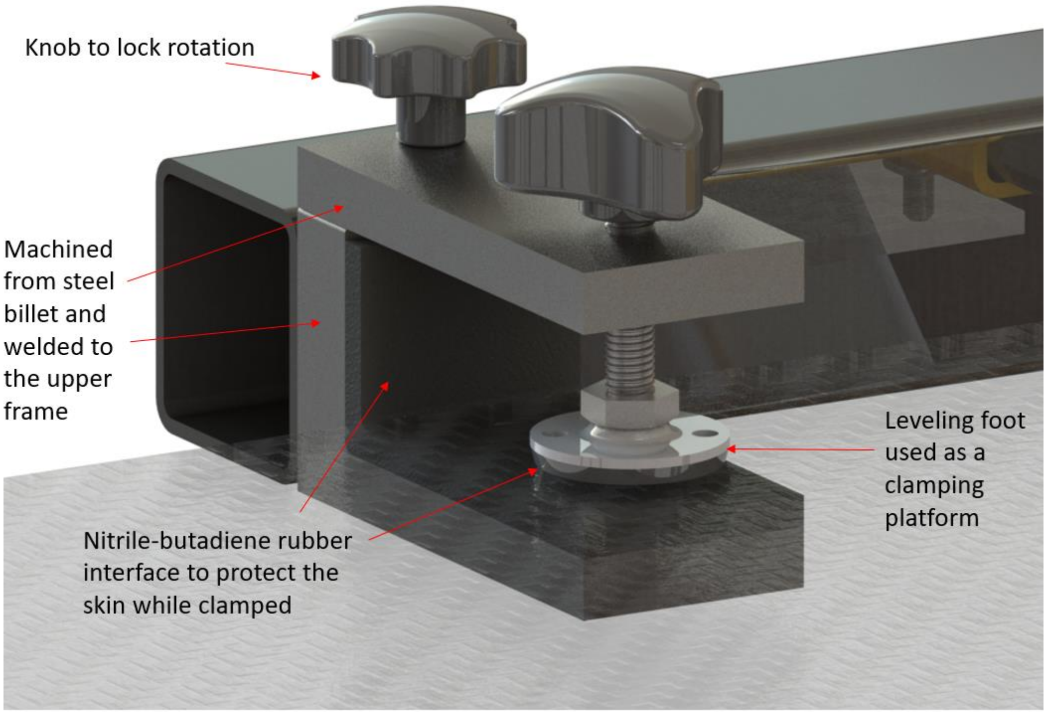

Figure 15: Final render of clamping mechanism (Front view)

The figure above shows the five components of the clamp. Our team could not find any off the shelf clamps available on the market that are designed similar to the selected concept. As a result, there are parts that require machining and assembly along with existing products on the market. The threaded platform and the knobs are both components that will be purchased and assembled.

Table leveling feet are to be used the threaded platform that can be tightened to clamp down on the skin. The selected part is sold by Elesa+Ganter and contains an M10 bolt rated for 2700 lbs [5]. The foot has a 40mm diameter platform that can rotate up to 6° to allow for the clamp to contact slight angularities that may occur on the surface of the skin. There is a 3.5 mm thick nitrile-butadiene rubber (NBR) layer at the bottom of the platform. The rubber has a shore

A hardness of 70 and is in place to protect the skin during clamping as well as prevent the surface from gliding.

A ¼ inch thick layer of adhesive backed NBR is applied to the bottom machined part of the clamp for contact with the bottom of the skin. The machined parts are from stock cold rolled steel billets so that they can be welded to the steel upper frame. The top and bottom of the steel parts are 0.5 inches thick to ensure the maximum stress remains under below the material yield strength with a factor of safety of 3.8. The knobs can be purchased off the shelf and assembled to the remaining components. The knob at the back of the clamp contains an m8 threaded bolt. When the bolt is loosened, the top of the clamp can rotate 360° for 100% working exposure on the surface of the skin. The figure below shows the back side of the clamps while the top is rotated out of the way.

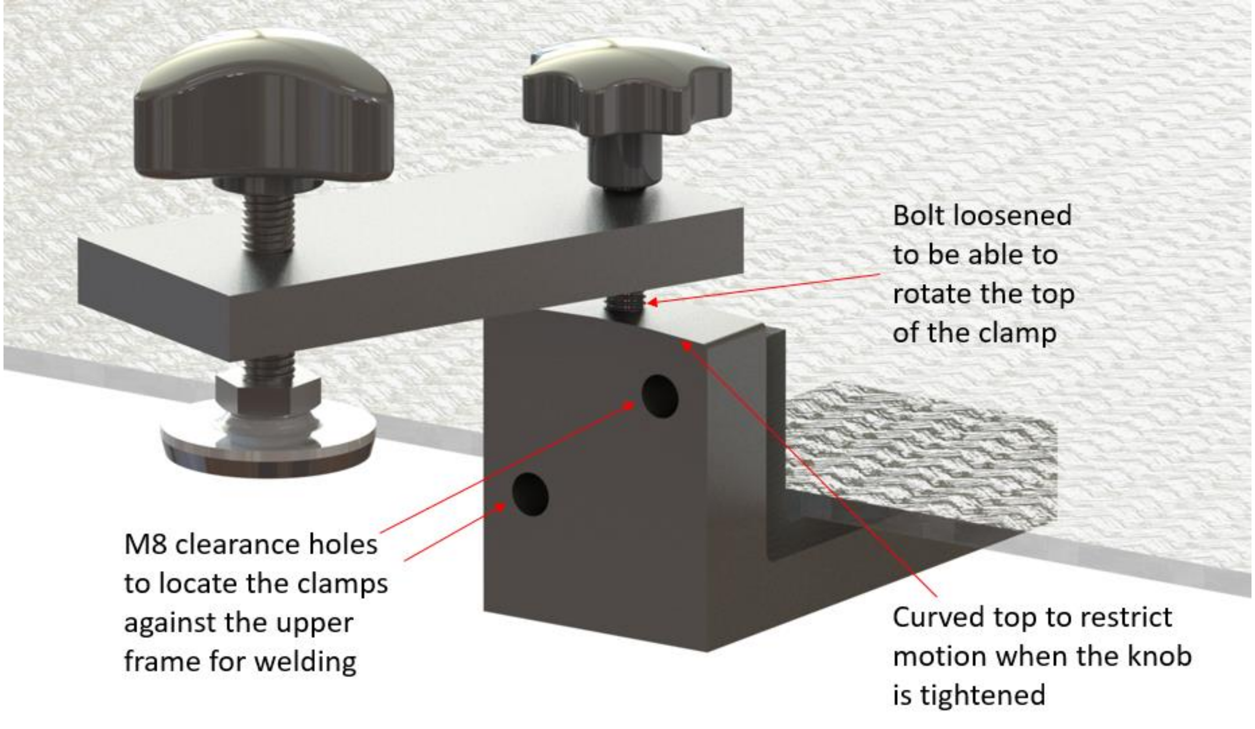

Figure 16: Final render of clamping mechanism (back view while the top of the clamp is rotated)

The slight curvature in the figure above applied on the back of the clamp is to restrict all rotational degrees of freedom when the knob is tightened. There are two m8 clearance holes at the back of the clamp that are positioned to align with the same clearance holes on the upper frame members. The holes are in place so that two M8 pins can be inserted to locate the clamp against the frame during welding. The total weight of the clamping assembly is 3.8 lbs. A full design analysis is provided in APPENDIX C, which provides details on part selection as well as stress/strain calculations on the machined parts.

3.4.2 Location of Mounting Points

A total of five clamps are required the skin is adequately support the skin such that the skin does not experience excessive stresses and deflections. The locations were designed such that the upper frame arms were minimized while ensuring the following:

- The skin does not deflect more than 0.1inche while subjected to working loads
- The skin does not deflect more than 0.5inches if someone were to fall forward and push down on the skin
- The stresses in the skin remain under the open-hole compressive strength with a factor of safety of at least 4.

The final location of the mounting points is shown in Figure 17 below.

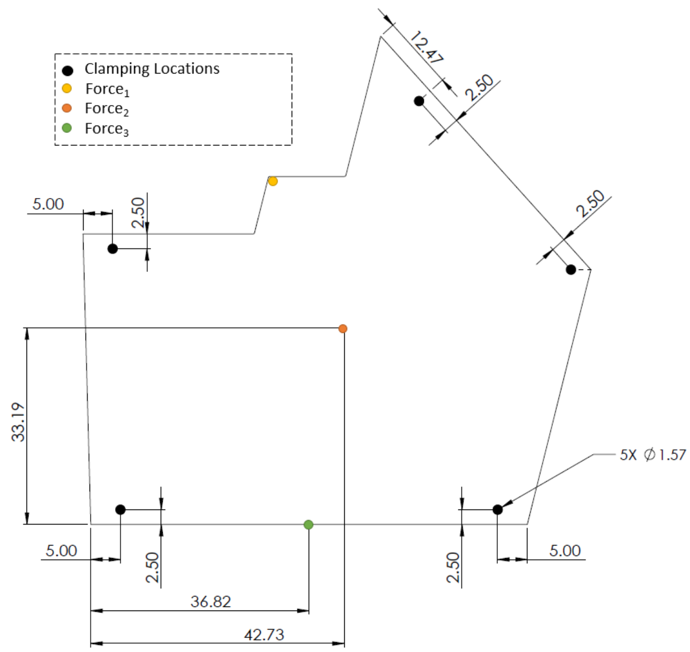

Figure 17: Location of clamp mounting points and forces used in the FEA

The black circles in the figure above indicates where the clamping platform will sit on the skin when the camps are engaged. The figure above also indicates three locations where forces were applied in the FEA analysis to determine the maximum deflections and stresses. The analysis is provided in APPENDIX C and the results are provided in the table below. The final column indicates the amount of flex that the operator would experience on the skin while working on it (deflection due to gravity is negated).

TABLE V: SUMMARY OF STRESSES AND DEFLECTIONS ON THE SKIN

Force location	**Minimum stress factor of safety**	**Maximum skin deflection if an operator were to fall forward on the skin (inches)**	**Maximum skin deflection while subjected to working loads**
Force 1	4.9	0.46	0.086
Force 2	19.8	0.23	0.038
Force 3	10.1	0.42	0.078

The table above indicates that the highest stresses and deflections occurs at the location of force 1 and meets the stress / deflection requirements.

3.5 Vertical Adjustment

Extension springs were used for the vertical translation motion in order to adjust the height for various operators. From the mathematical analysis, found in Appendix C, the springs chosen were an off-the-shelf unit with specific dimensions from Acxess Springs manufacturer. The best-suited springs have the following dimensions [6]

TABLE VI: BEST-SUITED SPRINGS WITH RESPECTIVE DIMENSIONS

Parameter	Unites	Values
Outside Diameter	inch	1.750
Insider Diameter	inch	1.396
Wire Diameter	inch	0.177
Mean Diameter	Inch	1.573
Original Length inside Hook	inch	14
Overall Length	Inch	14.354
Initial Tension	lbf	16
Total Number of Coils	-	66.266
Spring Constant	Lbf/in	5.470
Spring Index	-	8.887
Wahl factor	-	1.164
Maximum Suggested Load	lbf	89
Maximum Deflection	inches	13

The springs chosen are manufactured using hard drawn carbon steel wire, ASTM A-227. Hard drawn steel is a high carbon spring steel which is most commonly used, least expensive, readily available, easily worked, and has the following material characteristics [7]

TABLE VII: MATERIAL CHARACTERISTICS OF HIGH CARBON SPRING STEEL [8]

Parameter	Units	Values
Density	lb/in3	0.282
Modulus of Elasticity (E)	psi	30E6
Ultimate Tensile Strength	psi	2.56E5
Modulus of Torsion (G)	psi	11.5E6
Shear Modulus	psi	11.6E6
Poisson's Ratio (υ)	-	0.29

The spring uses a machine hook and has a zinc finish plated onto the springs to provide a galvanized layer of protected against corrosion as provided by the manufacturer. The machine hook uses 75% of the last coil and is bent to form a hook. It is one of the two most common hook types next to cross over hook but it is stronger because the radius of bend is not as pronounced [9]. Lastly, a zinc finish can protect the hard-drawn carbon steel wire during any high heat processes as it has a higher melting point [10].

Two extension springs will be mounted onto the top plate through an eye bolt on both sides of the device, as shown in Figure 18, below.

Figure 18: Mounting of the extension spring to the eye bolt

The bottom hook of the spring is mounted on a bolt which is attached through the sleeve. The bolt and the sleeve are illustrated in Figure 19, below, and Figure 18 shows the bottom hook end of the spring attached to the bolt.

Figure 19: Bottom end of spring attached to bolt

As the operator pulls on the outside of the sleeve, the spring will stretch to a specific height location. A locking pin, located one inch below the bolt (also part of the sleeve) is used to hold the spring mechanism in place. Once the pin is inserted to hold the spring mechanism, the load applied onto the skin affects the locking pin and not the extension springs. The size of the locking pin is dependent on the load applied on the skin and the material of the pin. The maximum force experienced by the pin is 325-pound force (factor of safety of 2 included) from the weight of the skin and the load applied at any one time. Using 316 Stainless Steel ring grip quick release locking pin from McMaster-Carr with an ultimate tensile strength of 75,000 psi, the diameter needed to support the 325-pound force load is 0.08 inches. A 0.5-inch diameter locking pin will be used because of safety reasons. It is also important to note that the 325-pound force will be distributed to pins on both sides of the device. Figure 20 below is the overall render on one side of the spring including the pin, bolt and sleeve.

Figure 20: Internal renders of the spring mechanism

The first stretched position occurs when the composite skin is loaded. At that position, the springs are 39 inches off the ground including the castors and 21.21 inches from the top plate. Hence the total height of the device is 60.21 inches. The springs can be further stretched five inches to be 34 inches off the ground, including casters or 26.21 inches from the top. The total suggested deflection is 13 inches and with an overall original length of 14.354 inches, the maximum suggested extension is 27.354 inches. Holes are placed one inch apart from the first stretched position to the last for any height variation in between. These measurements were expressed by operators of different heights and the ease of being able to perform work at those heights. Furthermore, the low spring rate helps to ease the movement of the vertical translation as it requires only 5 pounds of force to move the mechanism one inch. In order to eliminate the probability of twisting/torsion of the composite skin during vertical translation, each hole drilled during manufacturing of the device has tight tolerances and will match closely. Co-ordination between two operators is required on both sides in order to synchronize height adjustment and place the locking pin.

The force experienced by the spring on the first stretched position when the skin is loaded is 55.44-pound force. After the skin is stretched another 5 inches, the load experienced by the extension springs is at maximum of 82.79-pound force. The maximum suggested load that the extension spring can experience is 89-pound force. The eye bolt shown in Figure 20 is able to experience a vertical load up to 1,300 lbf. The full force analysis can be found in Appendix C. Stress analysis was further conducted to ensure the spring remains in the elastic region and does not plastically deform. The stresses were analyzed in the body of the coil and in the hook region of the extension spring, as shown in Figure 21, below.

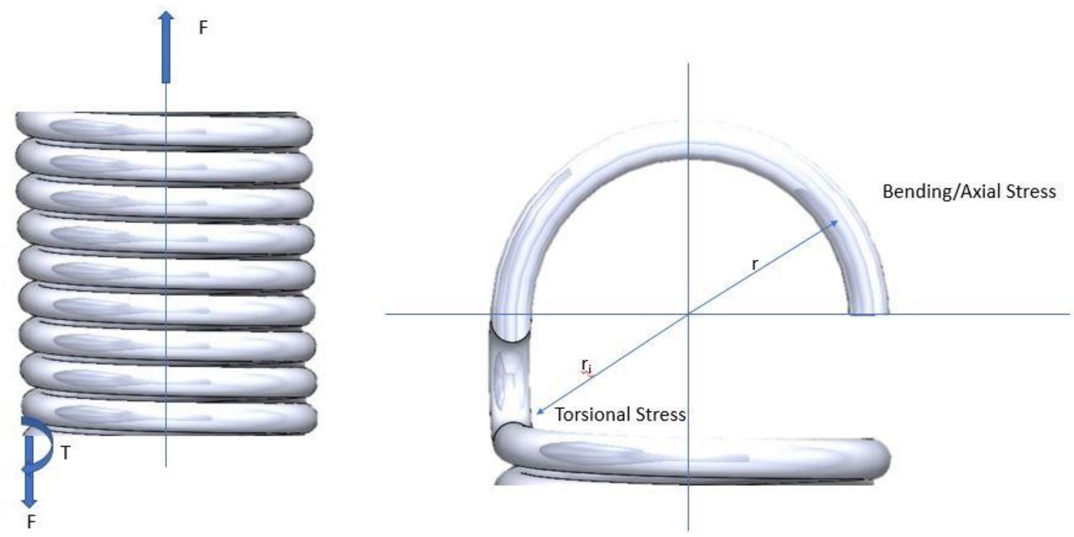

Figure 21: Stresses induced in the body and hook end of the extension spring during extension

Forces experienced in the middle of the coil induces direct stress and torsional stresses and at the maximum stretched position the super-position of the two stresses induces stress of 69,628.02 psi. The shear modulus for hard drawn spring is 11,600,000 psi which is above the calculated shear stress ensuing the coil to remain in the elastic region. The hook experiences bending in the middle of the hook and torsional stresses in the inside of the hook, as shown in **Error! Reference source not found.**.

The bending stresses at the maximum extended length is 133,917.10 psi and the torsional stress is 66,315.28 psi. The ultimate tensile stress and modulus of torsion for hard drawn carbon steel wire is 256,000 psi and 11,500,000 psi and hence the hook end of the extension spring remains in the elastic region.

Furthermore, even though no forces applied on the skin are experienced by the spring, the spring is stretched for a specific amount of time and the stresses incurred in the body affects the bolt. Using the maximum force and stress experienced by the spring at the maximum extended position, the minimum bolt size that can be used on each size of the device is 0.039 inches. Hence using bolt from McMaster-Carr the bolt size used will be 0.25 inches. Full analysis of all stress experienced on the spring, bolt and locking pin can be found in Appendix F.

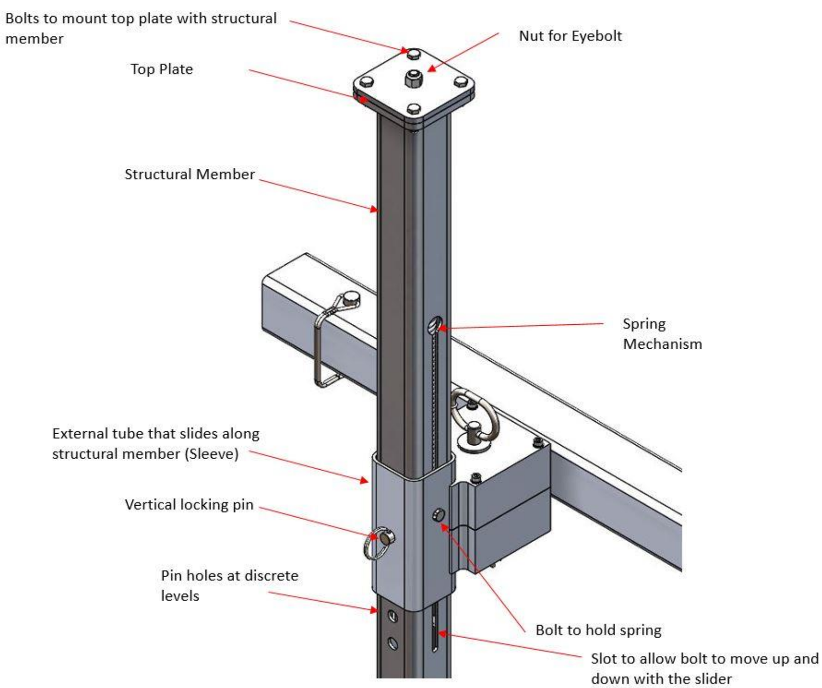

Figure 22: Height Adjustment Mechanism

3.6 Centre of Mass

Stability for any object is the measure of how likely they can topple over when pushed or moved. The width of the base of an object and height of the centre of mass are two factors that can affect the stability of an object [11].

The center of gravity for any object can be calculated using the sum of its moments divided by the overall weight of the object. [12].

$$C.G = \frac{W_1 d_1 + W_2 d_2 + W_1 d_3 + \cdots}{W} \qquad \text{Eq. 2}$$

Where W_1 is the weight at position 1, W is the total Weight of the device and d_1 is the distance between origin and position 1. Evaluating the centre of mass when designing any device is therefore a key step in identifying how stable that object can be. Devices with low centre of mass and a wide base ends up reducing the chances for tipping over.

3.6.1 Stability of the device

In order to evaluate the tipping force for the device when it is in the horizontal or vertical position, the centre of gravity and the mass of the whole system must be known. The centre of gravity when in the vertical position is 30.93 inches from the ground and mass for the whole assembly with the untrimmed skin 246 lbs was evaluated using SolidWorks. Figure 23 and Figure 24 below shows the location of the centre of mass.

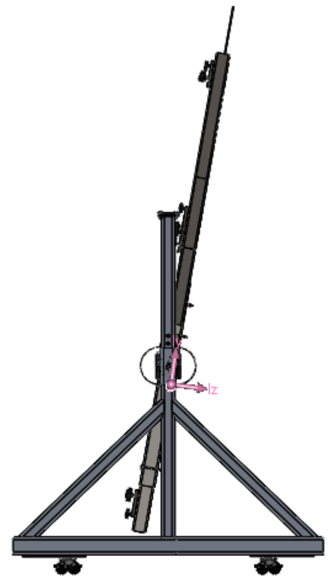

Figure 23: Side view of the Centre of Gravity for Device

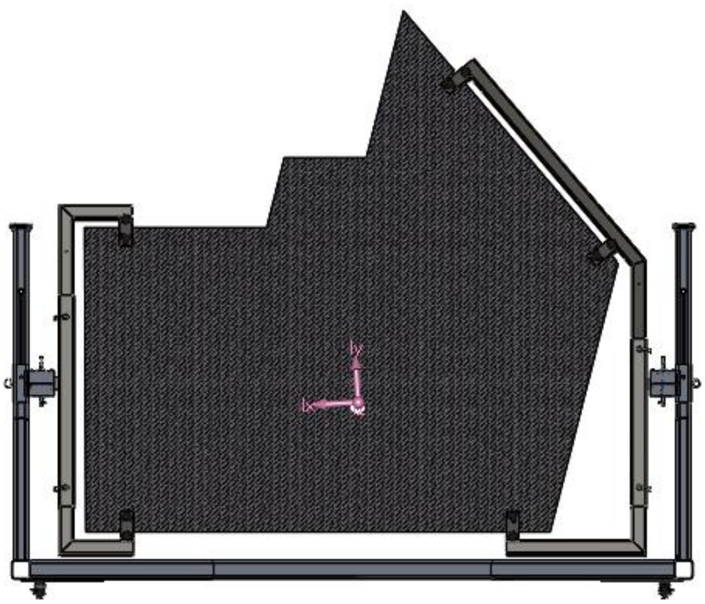

Figure 24: The Front View Device and Skin with Centre of Gravity

3.6.2 Vertical Stability

The tipping force for the device when in vertical position is calculated using the following equations.

$$\left(\overrightarrow{F_{net}}\right)_y = \sum F_y = 0 \qquad \text{Eq. 3}$$

$$\left(\overrightarrow{F_{net}}\right)_x = \sum F_x = 0 \qquad \text{Eq. 4}$$

$$\left(\overrightarrow{M_{net}}\right)_{C.G} = \sum M_{CG} = 0 \qquad \text{Eq. 5}$$

Where F_y are the forces in the Y direction, F_x is the forces in the x direction and M_{CG} is the Moment about the centre of gravity

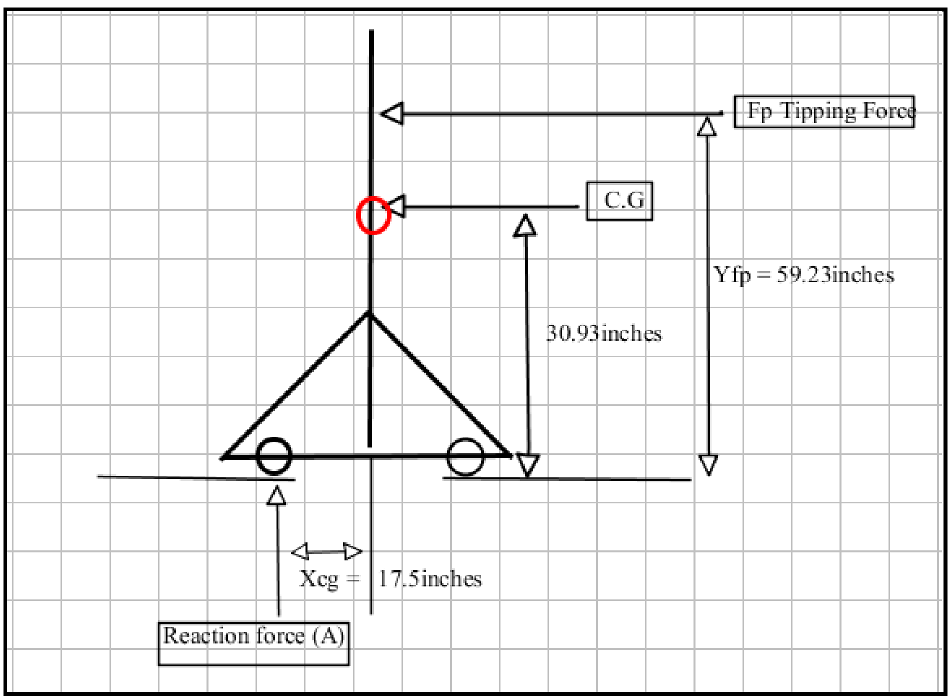

Figure 25: FBD showing the tipping force and the Centre of gravity with their locations

Figure 25 above is a FBD of the device in a vertical position. In diagram, the centre of gravity and the applied tipping force are located 17.5inches and 59.23 inches away from the wheels. Using Eq. 5 and taking the moment about position A the tipping force of the device can be calculated.

$$\sum M_A = 0$$

$$\sum M_A = W_{CG}X_{CG} - F_P Yfp = 0 \qquad \text{Eq. 6}$$

Inserting known values into Eq. 6, the tipping force when the structure is in a vertical position can be calculated as thus:

$$Fp = \frac{246*17.5}{59.33} = 72.56\text{lbs}$$

3.6.3 Horizontal Stability

The device when subjected to a horizontal force as shown in Figure 26 and **Error! Reference source not found.** is more stable when compared to a body subjected to a vertical force.

Case I: Force applied horizontally to the edge of the horizontal platform.

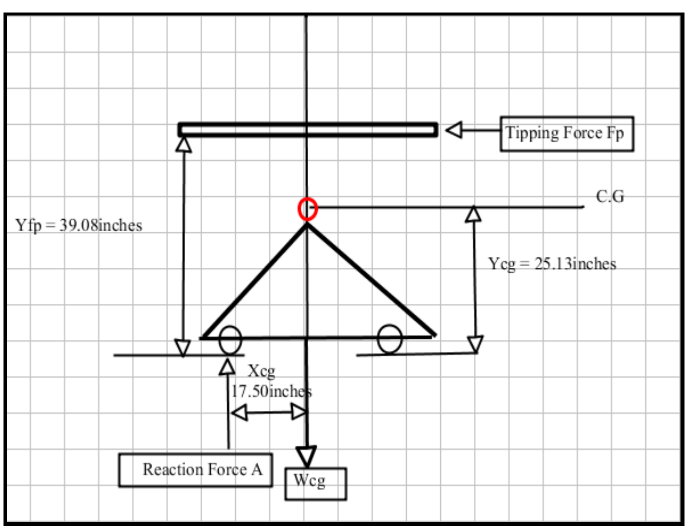

Figure 26: FBD of the device with tipping force applied in the horizontal position

Applying Eq. 6 and using the FBD in Figure 26 (X_{fp} now Y_{fp} = 39.08 inches) the applied tipping force (F_p) was resolved to 110.16 lbs

$$Fp = \frac{246*17.5}{39.08} = 110.16 \text{ lbs}$$

Case II: Force applied veritcally to the horizontal platform

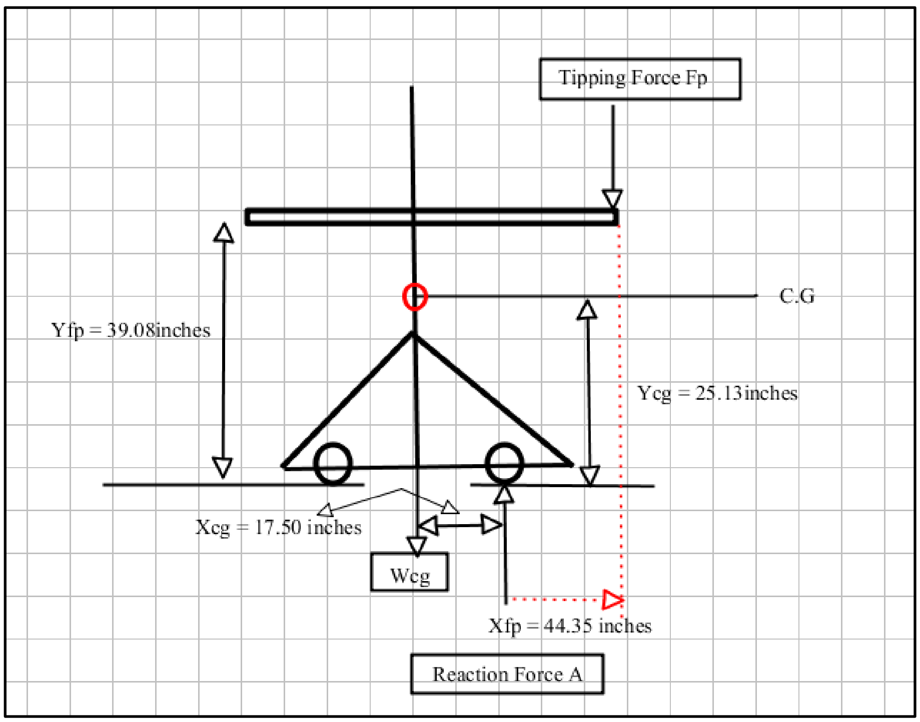

Figure 27: FBD diagram showing the device with an applied force on top the horizontal platform

If the same equation is applied again to the FBD in Figure 27, the tipping force applied at the top of the horizontal platform (Xf_p = 44.35 inches) is 97.67lbs. This scenario is the worst case scenario where the force is applied on the very tip of the skin. If the force is applied to an upper arm at the furthest location from the caster, it resulted in a force of 128 lbs to start tipping the device. When compared to the force applied in the horizontal direction, case B would be more stable. However, this case is the most critical situation because the width cannot be moved too inwards.

3.7 D-FMEA

The D-FMEA table was required in order to provide the client information on the different means of failure for each function. The table identifies potential risks of each function, its possible effect on the composite skin or operators using the device, prevention and detection techniques. By incorporating a D-FMEA table with the design, risks are mitigated and identified which in turn increases and improves the safety of the skin and operators using the device and performance of the device.

TABLE VIII, below, lists each function of the device, and their specific dimensions in the requirements section. Potential failure modes are described next which has a range from minor to devastating failures. As the failures are listed, potential effects are described and the severity of the effect and classification is identified. The table ranking of the severity and the table of the classification is found in Appendix G. Potential causes of the failures is described next along with current prevention techniques. The level of occurrence was given a value through a ranking table also found in Appendix G. The level of occurrence takes into consideration prevention techniques in place to reduce the amount of time that failure would occur. Lastly current control detection was listed along with a detection ranking and the risk priority number (RPN) is the multiplication of the severity, occurrence and detection ranking. Some failures modes had a low severity with occurrence occurring very likely which also increases the RPN number. The D-FMEA table provides client possible failure that can occur on the device and a full disclosure of ways to prevent, detect the failure mode to mitigate the risk and increase safety and performance.

TABLE VIII: D-FMEA TABLE

Item / Function	Requirements	Potential Failure Mode	Potential Effects of Failure	S E V	Potential Causes / Mechanisms of Failure	Current Design Controls Prevention	O C C	Current Design Controls Detection	D E T	R P N
Transportation	Size, load per castor, source,	Castors does not roll	Skin assemblies does not finish	4	Brakes are applied/Coefficient of friction is too high	Prior castor technology with specific specification of material to reduce static friction coefficient	1	Design Reviews	1	4
		Castors weak and cannot support load	Device detach from castor and damages skin and operator	10	Capacity per load on one castor is too low/Material too weak	Castor with adequate load capacity will be used	2	. Design Review and Test to failure	1	20
		Brake cannot be applied	Skin moves during work operation damaging skin and operator	8	Castors does not have brake characteristics/Brakes are broken/	Castors with durable and working brakes characteristics will be used	1	Design Review and Verification Test Method	3	24
		Castors does not pivot device	Device will not transport skin to work station and work will not be performed	5	Castors does not have swivel characteristics	Castor with swivel characteristic will be used	1	Design Review	1	5
		Hard to control during transportation when skin loaded	Damage against wall or obstacle damaging skin.	9	Castors load capacity too low/ Kinetic coefficient of friction of too high/Castors not suited for floor to material interface	Specific castors will be used to prevent high kinetic coefficient friction, distribute load more evenly and suited for floor type	3	Verification Test Method	3	81
		Skin damages when the device contacts wall / table during transportation.	Damage to the skin causing rework and delays.	10	The device does not adequately enclose the outer boundary of the skin and objects in the facility can be orientated to contact the skin directly during transportation. The external boundaries of the frame are subjected to failure resulting in a deflection that contacts the skin.	Adequate design features must be placed into the device to prevent damage of skin against wall/tables	7	Design review	3	210
Vertical Adjustment	Size of spring and other measurement, source	Spring does not hook onto top plate	No vertical adjustment mechanism	6	Wire thickness to thick/ top plate loop to small	Top plate loop will be specifically designed and manufactured so that spring hook end can fit	2	Design Review	1	12
		Spring does not extend	Operator cannot perform work comfortably	7	Spring constant too large/Spring initial tension is too high	Low spring constant and adequate suggested deflection of spring will be used for ease of extension	1	. Design Review and Verification Test Method	1	7
		Spring breaks upon extending with loaded skin	Skin falls onto ground and is damaged. Potential injury to operators	10	Spring material to soft/ Spring suggested load not high enough/ Spring hook can't handle load	Adequate spring geometry will be used to compensate failure mode	2	Test to Failure	7	140
		Spring plastically deforms over period of time	Spring will not recoil back upon release of locking pin.	6	Spring doesn't remain in elastic region/ Spring suggest load is not high enough	Adequate spring material characteristics will be used	7	Test to failure	7	294
		Spring corrodes due to environmental exposure	e.	8	Spring material has no protection/ Spring material not sufficient for type of environment/	Adequate finishing of spring and spring material will be used to prevent against corrosion	7	Degradation testing	8	448
		Hard to push on spring	Skin will not be set for ergonomic height for operators and potential injury to operator upon applying pressure onto spring and damage to skin also.	8	Spring constant is too high	Adequate and low spring constant will be used	1	Design Review	2	16

Item / Function	Requirements	Potential Failure Mode	Potential Effects of Failure	S E V	Potential Causes / Mechanisms of Failure	Current Design Controls Prevention	O C C	Current Design Controls Detection	D E T	R P N
Vertical Locking Pin	Size of pin, and other measurements, source	Locking pin shears due to load applied during work processes	Skin is recoiled upwards causing injury to operator and potential damage to skin	10	Pin diameter to low to withstand load	Adequate pin size will be used by performing mathematical and FEA	6	Verification test Method	2	120
		Locking pin cannot fit in hole/Pin doesn't stay in hole	No locking mechanism and skin cannot maintain any position upon setting at specific height	4	Pin is too small/Pin doesn't have a ball point at the end	Adequate pin size and proper design technique will be used		Design Review	2	8
		Locking gets lost or misplaced	No locking mechanism and skin cannot maintain any position upon setting at specific height	3	Pin not secured to device	Pin will be secured to device	1	Design Review	1	3
Size Adjustment	Size of sliding bar, hole measurement, number of hole	Sliding bar cannot to adjust	Skin cannot be fixed onto the device and device fails	5	Sliding bar has high friction against outside bar	Adequate design of bar using mathematical and FEA	2	Design Review	2	20
		Bars can't support load of skins	Skin falls onto ground causing damage and possible injury to operators	10	Bars are not thick enough to support load/Wrong choice of material for bar.	Adequate design of bar using mathematical and FEA	6	Test to Failure	7	420
		Locations of the adjustments slightly off	1) The back of the clamp will either not rest flush with the side of the skin resulting in the potential for the skin to slide vertically on the clamp, the sliding can damage the surface of the skin. 2) The back of the clamp contacts the skin before the pin can be inserted into the adjustment mechanism and the location cannot be locked in place.	1) 9 2) 2	Incorrect measurement taken of the skin.	Measurements taken multiple times to ensure that the location are correct. A verification with the technical drawing of the skin shall be performed.	1) 3 2) 6	Ensure Protoypes are constructed before actual fabrication of device. Design Review	1) 8 2) 8	1) 216 2) 96
Clamps/Padding	Size of clamps	Clamps scratch skin upon sliding for work process	Surface of skin becomes damaged and needs to be fixed causing delays for completion of assembly	9	Clamps are not padded properly/ Wrong choice of padding material/ Padding is not thick enough	Proven clamp characteristic will be used	4	Design Review	1	36
		Clamps does not hold skin and slides off	Skin falls onto ground and is damaged, possible injury to operators	9	Clamps cannot tighten/Interface between material and skin doesn't interact as per required	Adequate design consideration between two interfaces will be performed as well as prior technology will be used	4	Design Review	1	36
		Clamps deforms skin/Clamps leaves indentation on skin upon relieving clamp	Skin is damaged	10	Clamps are too tightly screwed /Padding is not thick enough	Design of clamps will use prior technology, training will be performed for operators.	4	Design Review and Verification Test Method	2	80
		Applied load breaks clamps	Skin falls onto ground and is damaged, possible injury to operators	10	Clamp material, geometry is not strong enough to withstand load applied	Adequate design of clamps using FEA will be performed. Ensure that off the shelf components are purchased from reputable supplier.	6	Test to Failure. Inspect off the shelf components upon arrival.	7	420
		Clamps not able to slide and expose skin for work	delay in completing work processes as operators have to re-adjust skin	5	Interface between clamp padding and skin is to rough and has high coefficient of static friction/ Clamps are not loosened/	Prior clamp technology will be reviewed and training will be provided for operators.	3	Design Review and Verification Test Method	1	15

Item / Function	Requirements	Potential Failure Mode	Potential Effects of Failure	S E V	Potential Causes / Mechanisms of Failure	Current Design Controls Prevention	O C C	Current Design Controls Detection	D E T	R P N
		Applied load to the skin causes the skin to yield/fracture	Skin is severely damaged and needs to be fixed, and possibly scrapped, causing delays for completion of assembly	10	Supporting material underneath the skin is positioned such that there are areas that are subjected high stresses and deflections when loaded.	Perform analytical calculations and FEA stress on the skin when mounted to the device. This is to determine the maximum stress and deflection when subjected to maximum normal forces.	2	Design review and verification test method.	9	180
Rotational Adjustment (Including Pin)	Size of bearings, box, locking pin	Top of skin fall backwards once pin is loaded	Damage to skin and possible injuries to operators	10	Tolerance in hole is not tight enough/Adjustment pin is to thin	Adequate pin design will be performed using mathematics and FEA	5	Verification Test Method	3	150
		Operator cannot support skin for rotational adjustment	Skin is not adjusted to perform specific work processes	8	No place for operator to hold while rotating/ Skin and device too heavy to adjust using one operator/ Center of gravity out of balance (too high) and hard to control by operator/Skin is too heavy and wants to fall backwards	Adequate device design and prior handling techniques will be used	4	Design Review	2	64
		Locking pin shears due to applied load on skin	Skin topples over damaging skin and possible injury to operator	10	Locking pin diameter to thick enough	Adequate pin design using mathematical and FEA	6	Test to Failure	7	420
		Locking pin gets lost	Cannot hold skin, device fails as skin can topple over and damage the skin	4	Locking pin not secured to device	Pin will be secured against device	1	Design Review	1	4
		Locking pin cannot fit and is loose inside hole	Cannot hold skin, device fails as skin can topple over and damage the skin	4	Locking pin diameter not right for hole. No secure ball at end of pin	Pin adequate designed through mathematics and FEA	1	. Design Review	1	4
		Bearing fails	The shaft cannot easily rotate and the bearing will need to be replaced	2	Bearing selection does not meet required loading capacity.	Purchase bearings of high quality from reputable supplier.	1	Part inspection upon arrival	f.	f.
Frame of Device	Overall Size of Device	Device cannot support skin and buckles or breaks	Skin falls to ground and is damaged, possible injuries to operators	10	Dimension of device cannot support load of skin and applied load	Device will be adequately designed using mathematics and FEA	6	Test to Failure	7	420
		Device a tripping hazard for operator	Injuries operators and possible damage to skin	10	Device sticks out and is in the way of the operators' movement	Padding will be placed on device and device will be adequately design to identify trip hazards.	2	Design Review Design Review and Verification Test Method	3	60
		Device causes ergonomic problems to operator during work	Operator experience health issues, device fails and possible damage to skin	8	Device does not follow ergonomic standard/ Device is not designed for different operators' size and height	Device will follow ergonomic standards	2		1	16
		The frame tips over when subjected to external loads.	Damage to the device and to the skin, potential damage to the operator.	10	The the base of the device is not wide enough and the center of gravity is too high to prevent the device from tipping over	Analytical calculations to determine the tipping force to ensure that the device will not tip over when subjected to maximum external loads.	1	(TEST EXTERNAL LOAD TO PROVE IT WON'T TIP OVER – WHAT TO CALL THAT?)	1	10

3.8 Technical Specification Review Analysis

Each designed component in the material handling device achieves all ideal values and surpasses the current process, the A-frame, as prescribed by the target specification table in Section 1.6. Thorough force and stress analysis allowed achieved values surpass or meet the ideal values. TABLE IX below, compares achieved value for the material handling device to the ideal and A-frame values.

TABLE IX: FINAL TECHNICAL SPECIFICATION REVIEW TABLE

Metric	Units	Ideal Values	A-Frame	Achieved Values
Number of crush points	#	0	0	0
Laceration points	#	0	0	0
Local Deformation at mounting points	Binary	None	None	None
Maximum skin stress	lbf	YSs(FS=4)	0	FS = 4.9
Skin deflection while subjected to working loads	inches	0.1	0	0.087
Maximum possible skin deflection	inches	0.5	0	0.46
Device is upright at all times	Binary	Yes	Yes	Yes
Maximum stress on device	psi	YSd(FS=5)	0	FS=2
Both skin types are compatible with device	Binary	Yes	Yes	Yes
Time to complete full rotation	Seconds	3	N/A	3
Maximum force required to tip the device over	lbs	50.582	Unknown	97
Time to complete height adjustments	Seconds	15	N/A	15
Ease of applying stopping (slowing down) force	Subjective	Easy	Moderate	Easy
Range of rotation	Degree	90	N/A	180
Range of motion of all co-ordinate frame for working position (tolerance)	x,y,z (mm)	0	N/A	0
Allowable change in angle at position	Degree	0	N/A	0
Design allows for ergonomic mounting for user	Binary	Yes	No	Yes
Height adjustment	Inches	4 to 8	N/A	5
Range of motion of all co-ordinate frame for transfer position (tolerance)	x, y, z, pheta, phi	0	N/A	0
Device protect skin on all side upon collision	Binary	Yes	Yes	Yes
Maximum deflection under external load	Millimeters	20	N/A	18
Device is portable	Binary	Yes	Yes	yes
Maximize working surface area	ft^2	9.5	N/A	100%
Time to setup and load skin onto device	Seconds	45	45	45

Metric	Units	Ideal Values	A-Frame	Achieved Values
Time require before device fails	Years	40	40	40
Amount of visible corrosion on device over lifespan	Binary	None	None	None
Force required to adjust height position,	Binary	Minimal	N/A	Minimal
Force required to make rotational adjustment	Binary	Minimal	N/A	Minimal
Force required to start moving device with/without skin	Binary	Minimal	Moderate	Minimal
Workable surface that the worker can access	ft^2	10	N/A	100%
Hours to replace/fix/maintain any one skin of device	Hours	0.5	N/A	0.5
Specialized maintenance tools	List	1	N/A	1
Maximum Dimensions of Device	in x in x in	60x79 x118.11	68.89x44.88 x98.03	60.21x51 x108.00
Device rotates about center of vertical axis	Degree	360	360	360
Sound during usage of device, include all motion	dB	20	20	20
Manufacturing Cost of One Unit (Standardization)	US$	2,000	2,000	1400
Manufacturing process has minimal environmental impact	Binary	Yes	N/A	Yes
Materials recyclable	Binary	Yes	Yes	Yes
Device looks nice and is pleasing to the eyes	Binary	Pass	Fail	Pass

3.9 Complete Bill of Material

The table below lists the complete bill of material required to manufacture the material handling device. The table is broken into the different components including frame design, clamp design and rotational block design. Within each component, each material has part name, description according to the name in the source, source of where to acquire the part and the source part numbers, quantity and the source of each material. The part number is referred to the drawings in Appendix F. Materials that are machined are used from a block material and machined to the size desired as per the drawings in Appendix F.

Part Name	Description	Quantity	Part Number	Source	Source Part Number
Frame Design					
Mighty-Lite Caster with 3" x 1-3/4" Plate	Swivel with Brake and 2" Diameter 90A Durometer Rubber Wheel	4	4-190	McMaster Carr	2406T65
Caster Plate - Aluminum Pate 6061T6	Aluminum 6061T6 Plate 0.250in	4	4-110	Metal Supermarket	-
Base Frame - 2.25X2.25 Aluminum Tube Sqaure	Aluminum 6061T6 Plate Square Tube	Total Length	4-110	Metal Supermarket	-
Base Frame 3X3 Aluminum Tube Sqaure 6061T6	Aluminum 6061T6 Sqaure Tube	Total Length	4-110	Metal Supermarket	-
18-8 Stainless Steel Hex Head Screw	1/4"-20 Thread Size, 1" Long	8	4-160	McMaster Carr	92240A542
18-8 Stainless Steel Hex Head Screw	1/2"-13 Thread Size, 3" Long, Fully Threaded	8	4-200	McMaster Carr	92240A724
High-Strength Steel Nylon-Insert Locknut	Grade 8, Zinc Yellow-Chromate Plated, 1/4"-20 Thread Size	8	4-170	McMaster Carr	97135A210
Top Plate - Aluminum Plate 6061T6	Aluminum 6061T6 0.250in	2	4-120	Metal Supermarket	-
Vertical Extension Spring	PE177-1750-66.266-HD-14.000-MH-Z-IN	2	4-180	Acxess Spring	-
18-8 Stainless Steel Ring-Grip Quick-Release Pin	1/2" Diameter, 3" Usable Length	2		McMaster Carr	98404A510
Steel Eyebolt with Shoulder - for Lifting	3/8"-16 Thread Size, 5/8" Thread Length	2	4-150	McMaster Carr	3014T956
High-Strength Steel Nylon-Insert Locknut	Grade 8, 3/8"-16 Thread Size	2	4-140	McMaster Carr	90630A121
Upper Frame Assembly					
Steel Cold Rolled Tube for Upper Frame Left	1020 Steel Cold Tube 2.5x2.5x0.125	1	3-110	Metal Supermarket	-
Steel Cold Rolled Tube for Upper Frame Right	1020 Steel Cold Tube 2.5x2.5x0.125	1	3-140	Metal Supermarket	-
Steel Cold Rolled Tube Round	1020 Steel Cold Round Tube 2x0.25	1	3-110, 3-140	Metal Supermarket	-
Steel Cold Rolled Tube for Upper Frame Top Left	1020 Steel Cold Tube 2.25x2.25x0.125	1	3-120	Metal Supermarket	-
Steel Cold Rolled Tube for Upper Frame Bottom Left	1020 Steel Cold Tube 2.25x2.25x0.125	1	3-130	Metal Supermarket	-
Steel Cold Rolled Tube for Upper Frame Top Right	1020 Steel Cold Tube 2.25x2.25x0.125	1	3-150	Metal Supermarket	-
Steel Cold Rolled Tube for Upper Frame Bottom Right	1020 Steel Cold Tube 2.25x2.25x0.125	1	3-160	Metal Supermarket	-
Clamp Design					
Steel Cold Rolled C1018 for Clamp Bottom Part	Cold Rolled C1018 Sqaure Bar 2.5 x 5	5	4-110	Metal Supermarket	CSQ1018/212
Steel Cold Rolled C1018 for Clamp Top Part	Cold Rolled C1018 Flat Bar 0.5 x 5 x 2.5	5	4-120	Metal Supermarket	CF1018/125
Rubber Buna-N 1/4in Thick 2X36in	EJM4070-1/4XTAPE	5	4-130	Acklands Grainger	-
Metric Comfort-Grip Plastic Multi-Arm Knob with Threaded Studs	M10 X 1.5mm Threaded hole, 45mm Wide Head	5	4-150	McMaster Carr	2776K65
Plastic Multi-Lobe Knob	M8 x 1.25mm Threaded 40mm Long Stud	5	4-160	McMaster Carr	2776k68
Sleeve and Rotational Block Design					
Ball Bearing	Trade No. 6210, for 50 mm Shaft Diameter, 90 mm OD	4	2-160	McMaster Carr	5972K129
Rotational Block - Aluminum Sqaure Bar 6061 T6	Aluminum 6061T6 Swayre Bar 4.000	4	2-120, 2-130	Metal Supermarket	ASQ6061/312
Sleeve - Aluminum Tube Sqaure 6061T6	Aluminum 6061T6 Sqaure Tube 2.5*2.5*.120	2	2-110	Metal Supermarket	-
316 Stainless Steel Washer	5.3 mm ID, 10 mm OD	8	2-180	McMaster Carr	90965A160
Black-Oxide Alloy Steel Socket Head Screw	M5 x 0.8 mm Thread, 75 mm Long	8	2-170	McMaster Carr	91290A274
18-8 Stainless Steel Hex Head Screw	1/4"-20 Thread Size, 3" Long, Partially Threaded	8	2-140	McMaster Carr	92198A554
Medium-Strength Steel Nylon-Insert Locknut	Grade 5, Zinc-Plated, 1/4"-20 Thread Size	8	2-150	McMaster Carr	95615A120
Zinc-Plated Steel Ring-Grip Quick-Release Pin	1/2" Diameter, 2-13/16" Usable Length	2	2-200	McMaster Carr	98320A508
Clevis Pin with Hairpin Cotter Pin and Lanyard	316 Stainless Steel, 1/2" Diameter, 5" Usable Length	2	2-190	McMaster Carr	98412A434

4. Recommendations

To prove or further analyze the design of the device, a prototype device should be made to confirm manufacturability and durability. The numerical analysis performed on the skin indicated that the stresses and deflections were within allowable limits, however, experimental tests should be conducted to validate the results. Incorrect assumptions could leave the skin vulnerable to excessive stresses and damage. Below in

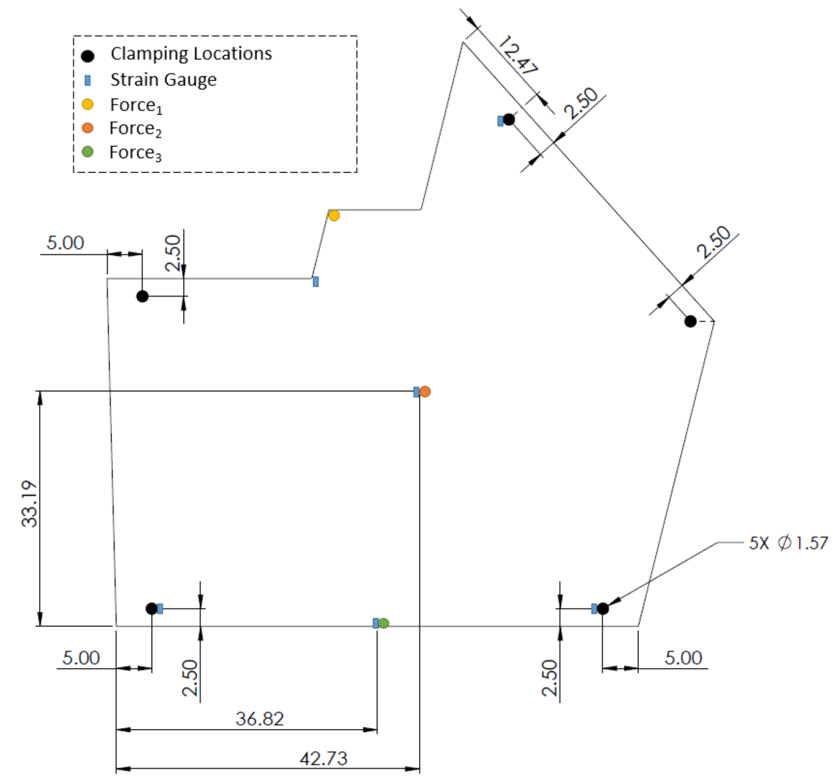

Figure 28 is the layout for strain gauges to be placed on the skin and the proper locations for the clamps and applied loads. These strain gauges would be used as the confirmation tool for the numerical results. Additional strain gauges may be required depending on the actual contour of the skin.

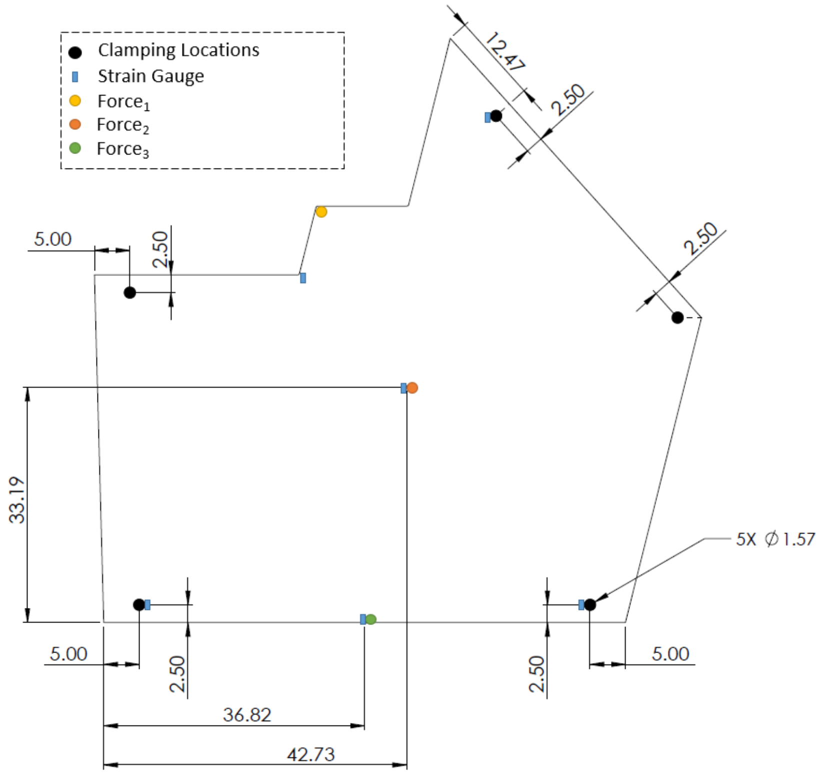

Figure 28: Strain gauges on skin for experimental confirmation

5. Conclusion

Magellan Aerospace required a material handling device, in the Winnipeg Facility, that can transport a composite aircraft skin in the near vertical position and rotate the skin to allow work perform in the horizontal position. The need of the device was to reduce the amount of manual handling by the operators, which in turn reduces the amount of damage that occurs on the composite skin due to manual handling prone accidents. Creating and ranking the objective list helped to organize mandatory requirements that the device had to perform. Constraints and assumptions were defined to further delineate the project that led to the creation of the technical specification analysis. Each objective was given a defined method and metric to achieve each objective.

The use of morph table helped our team find different methods to perform different functions of the material handling device. Concepts were generated by connecting each possible method to one another and twelve concepts were created. Defining strengths and weaknesses of each design, screening and scoring analysis helped to filter weaker designs and promote a stronger definite design. Further optimization was conducted to achieve higher scoring and a more definite design to meet all objectives.

Each function underwent detail design analysis where force and stress analysis were evaluated to choose materials and parts that would meet all requirements for a safe, durable, out-performing material handling device. After which, technical specification was further reviewed to determine whether the design created met all metrics and objectives. Other than the cost, each metric was achieved for a safe, rugged, ergonomic, durable material handling device. The client regarded cost as a non-importance as safety of the composite and towards the operators overly out-weighed the cost. Defining a D-FMEA table and evaluating stability of the device horizontally and vertically, design failure analysis was conducted. The D-FMEA provided the customer different potential failure modes that can occur by the device, the consequence of the failure, the cause and prevention and detection methods of the failure. The D-FMEA table would provide the customer means of failure at unexpected times. The stability analysis included the overall size of the device, the different center of masses between both skin types and an overall force required to turn over the device. The stability analysis showed that the device could not be easily tipped over through its wide base and sturdy material selection

It is highly recommended that further stress/strain analysis be performed through a prototype model within the company to further improve on the computational stress analysis. Overall, our team created not only a functional and durable material handling device but also a product that completes all work processes easily, provide ergonomic features, provide safety for the composite skin and for operators through its rugged and sturdy design. This device through intense rigorous design work from finding different methods for all functions to providing detail design analysis for each function meets all the requirements that was demanded and required by the client and Magellan Aerospace.

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APPENDIX A: Concept Generation and Selection

The method and results from the concept generation and selection is provided in this appendix.

1. Morph Chart

TABLE A- I, below, is the modified version of the morph chart that was used to generate concepts. The modified morph chart broke the design of the material handling device into its separate function and further subsection of each function was created in order to fully define each function. The possible solutions for each subsection are solutions that made the most sense for the design of the material handling device and by combining concepts from each subsection; a possible design was created and generated.

TABLE A- I: MODIFIED MORPH CHART DEPICTING DIFFERENT FUNCTION, ITS SUBSECTIONS AND POSSIBLE METHODS OF PERFORMING THOSE SUBSECTIONS

Modified Morph Chart for Handling Device										
Means ------------>> Principal Functions (Below)	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Transportation										
Ground contact	4 Large Rubber Wheels Filled with Air	Shopping Cart Wheels (2 Casters in the Front, 2 Fixed Wheels in the Rear)	3 Casters Triangle	Solid Casters with Locking Brakes	4 Duel Wheel Caster with Brakes	4 Top Plate Caster Wheel with Brake	4 Swivel Caster with Brakes	4 Ball Caster Wheel		
Slow down when in motion	Arm bars for the user to hold onto the whole time and is able to pull back to slow the device down	Bike hand breaks								
Parking	Wheel Bearing Friction	Wheels lift up so that frame sits on the ground	Foot break							
Frame										
Material	Steel Tubing Truss with Welded Joints	Aluminum Tubing with Welding [13]	Plastic	Titanium Tubing with Welded Joints [14]	Stainless Steel Tubing					
Top Part	Open	2 side railings with one bar through the middle to support skin	Flat plate with mesh dipped in poly with supports on all side [15]	Side bar with extendable length and adjustable width	Grill Type Framing	Two vertical bars with a hollow circle frame on each bar	Flat thin vertical rectangular stand			

Modified Morph Chart for Handling Device										
Means ------------>> **Principal Functions (Below)**	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Bottom	One Triangular-frame	Two Triangular-frame connected with a bar running down the middle	Two Triangular-frame connected with bars through the border of the two frame	Two Y-frame connected with a bar running down the middle	One Y-frame placed in the middle of the circular base	Two X-type vertical frame, which sits on a rectangular bar. The rectangular bar has a horizontal bar attachment in the middle	A flat wide thin rectangular base	A handheld cart frame structure	Four legged system with inner and outer bar.	Four crossed adjustable truss system mounted to a control tool box base
Mounting	Suction Cups [16]	One long clamp holding base of skin	Two long rectangular clamps holding the sides of the skin	Plate with supports on side and bottom with pin to prevent sliding	Two small diameter flat surface C-clamps on each side of skin [17]	Two small rectangular clamps on each side of skin	C-clamps and hooks attached to bars [17]	A pulley that attaches to bars with hooks at the bottom	Hooks attached to tube	A flat plate with adjustable hooks on all four sides
Rotational Mechanism										
Motor assisted Drive Mechanism	Gears [18]	Belt	Chain	Hydraulics						
Manual drive mechanism	Hook in a hole to act as an axis to rotate about	A wheel with a shaft attached to top mounting [19]	Gear system which is attached to top mounting which is rotated through manual force [20]	A sphere which has a track that allows for not only vertical translation but also rotational adjustment [21]	Pulley system that helps with rotation [22]	Top mounting attaches to a rotating arm which pivots about its fixed point [23]				
Prevent back movement	Friction fit	Design similar to ratchet where if you rotate it one way it turns but if you attempt to rotate it in the other direction it locks								
Locking mechanism	Ladder rungs [24]	Pin [25]	Knob loosens tooth design which will then rotate and then you can retighten, locking the teeth							
Vertical Mechanism										
Height Adjustment	Lift and insert into next rung similar to increasing height of ladder	Crank to slide up similar to drywall lift	A hydraulic system similar to a office chair system [26]	A-rack and pinion mechanism	Elevator Style Lift	A smaller bar inside a larger bar that is adjustable by pins	Manual Hydraulic System [26]	Rack and Pinion combined with Pin system		

Modified Morph Chart for Handling Device										
Means ------------>> Principal Functions (Below)	**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**
Locking mechanism	Ladder rungs	Pin [25]	Press to release mechanism which will then only allow for vertical translations (new)	Press fit	Locking slot	Locking hook				

2. Concept generation

From the modified morph chart, each possible solution from each function was combined together to create competent and possible concepts. A total of twelve concepts that satisfy the customers needs were generated. Each concept is provided and described below.

Concept 1: Locking Spline

The first concept consists of a triangular frame, a shaft, a rotational locking mechanism and clamps to hold the skin in place. A sketch of the concept is shown in the figure below:

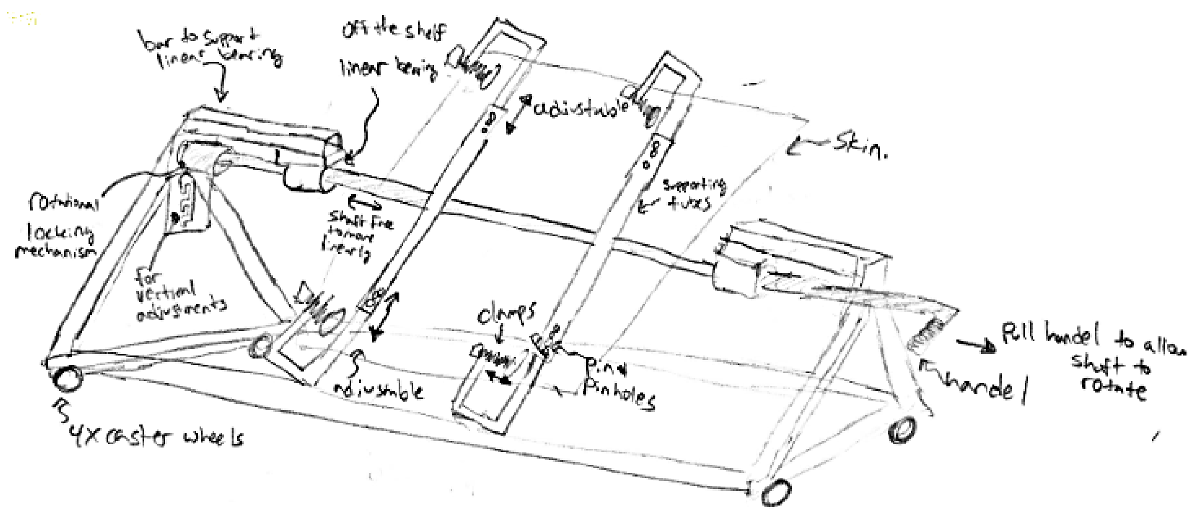


Figure A- 1: Locking Spline – Pull to Rotate

Figure A- 1, above, shows a shaft that is supported by linear bearings.

The shaft connects to a locking mechanism on one end and a handle on the other. Two structural square tubes are welded to the shaft to connect to clamps. Inner tubes are placed inside the larger square tubes with the ability to slide linearly to adjust the length. Pins are used to lock the tubes after adjustments have been made. The clamps consist of a bolt, a platform at the bottom of the bolt and a handle at the top of the bolt so that the user can lower the platform placing a compression force between the skin and the tube. Foam padding is placed on the tube and bolt as a soft material in contact with the skin. Four caster wheels are placed at the bottom of the frame to allow the device to move around the facility and rotate about the vertical axis for an ease of placement at each station.

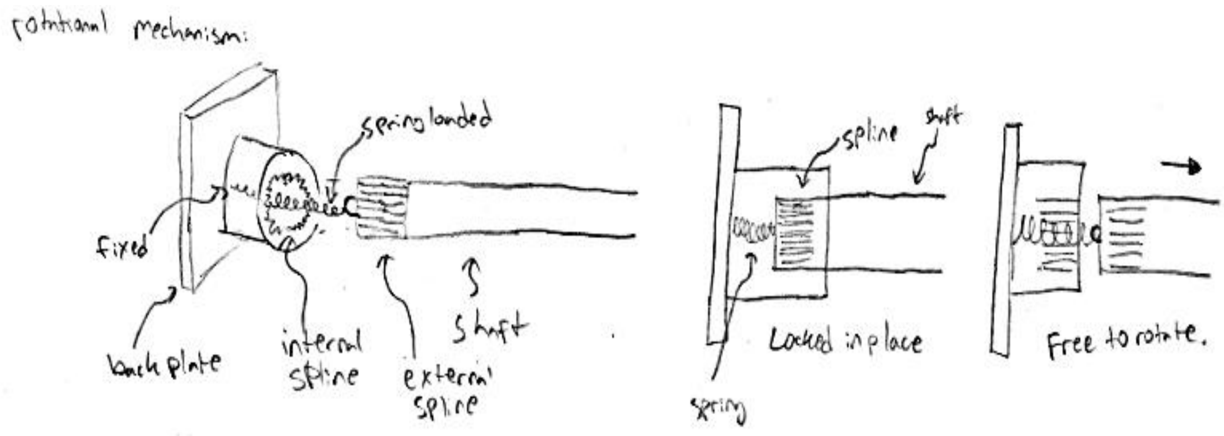


Figure A-2: Locking mechanism for "Locking spline pull to rotate"

Details of the locking mechanism are provided in Figure A-2 above shows a larger diameter tube with an internal spline welded to a back plate. The end of the shaft has an external spline. The shaft would be unable to rotate when it is inside of the larger diameter tube as the spline walls would be in contact. The shaft would be attached to a spring which would force it in the locked position. If the handle is pulled the shaft will move out of the tube and will be free to rotate. The bar to support the liner bearings and the rotational locking mechanism are attached to a plate that has pegs sticking out the end. The pegs rest in a back plate as seen in Figure A-3: Height adjustment for "Locking spline pull to rotate"

, below:

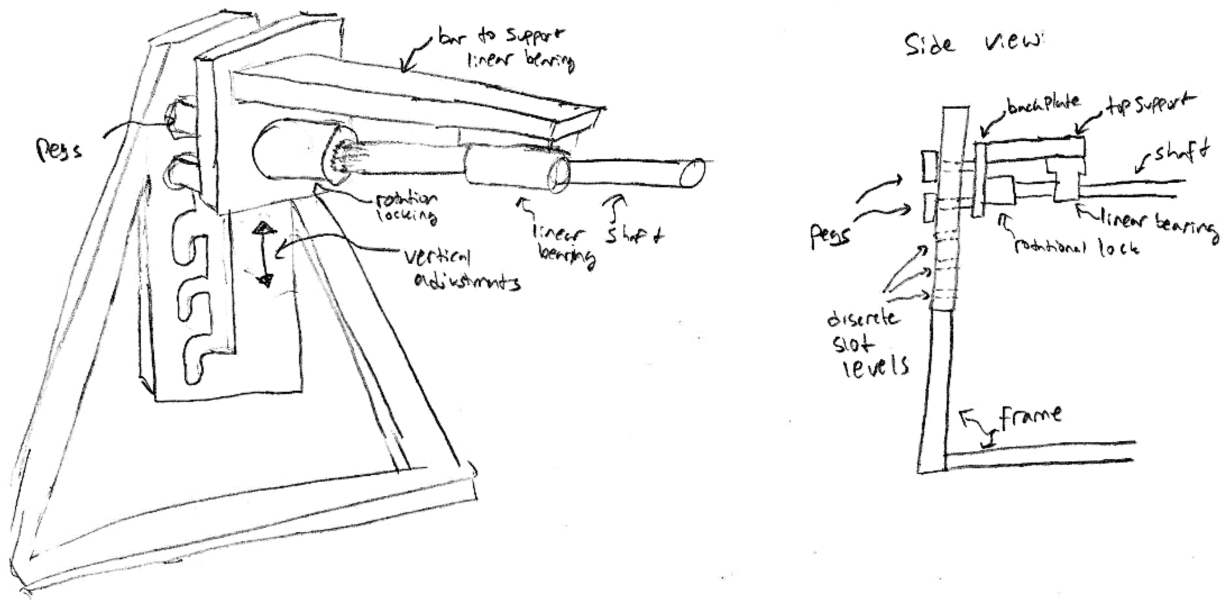


Figure A-3: Height adjustment for "Locking spline pull to rotate"

When the skin is not attached to the frame, the shaft can be picked up, shifted back, then raised or lowered to a different peg location.

The strengths and weaknesses to this concept are as follows:

Strengths

- The device is stable as the center of gravity is close to the center of the frame when the skin is either horizontal or vertical.
- The device is stiff as the frame is a truss design where all members are either in tension or compression (with minimal bending).
- The skin is supported in the center so there is minimal stress on the skin when mounted to the device.
- The skin can lock in many increments between 0° and 90° which will assist with loading the skin onto the device.

- There is minimal rotational tolerance when the locking mechanism is engaged.
- There is minimal area covered by the clamps.
- It is easy to load and unload the skin as it can rest on the tubes that are welded to the shaft.
- The external frame acts as the first point of contact in the event of a collision into a wall or desk, thus protecting the skin.

Weaknesses

- Vertical adjustments can only be made when the skin is not mounted onto the device and the adjustment is difficult to perform.
- The frame has a large footprint and the bottom structural members can act as tripping hazards.
- The back side of the skin is covered by the shaft and the supporting tubes, thus restricting workable area.
- Components of the device are difficult and complex to manufacture and assemble.
- The shaft will become free to rotate if the operator pulls on the handle to stop the device while it is in motion.

Concept 2: Worm gear crank with an offset rotational axis

The worm gear design consists of an off the shelf worm gear crank to adjust the height and an offset axis that will allow for the skin to rotate and be stable under gravity. A sketch of the concept is shown in Figure A- 4 below.

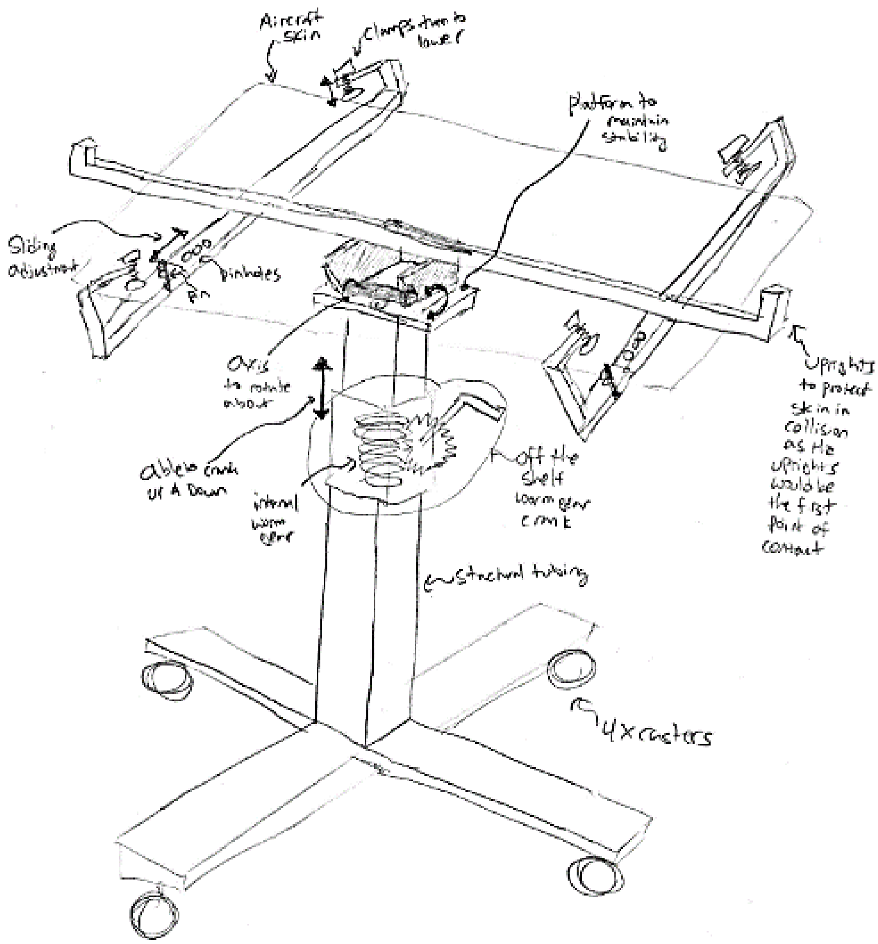


Figure A- 4: Worm gear crank with an offset rotational axis

This design uses the same clamping mechanism. Additional uprights are placed at both ends of the skin to act as the first point of contact in the event of a collision. Vertical translations are completed using a worm gear crank. Four castors are placed on the bottom of the device. Square tubing is welded on the bottom and top of the crank to act as the center structural column. The top tube has a platform welded on the top with a rod welded above the plate to act as a rotational axis. There is a bent steel plate that is connected to the axis and is not connected to any locking mechanisms when the skin is horizontal. The plate is designed such that when it is vertical, the flat edge of the plate is in contact with the platform, as shown in Figure A-5, below.

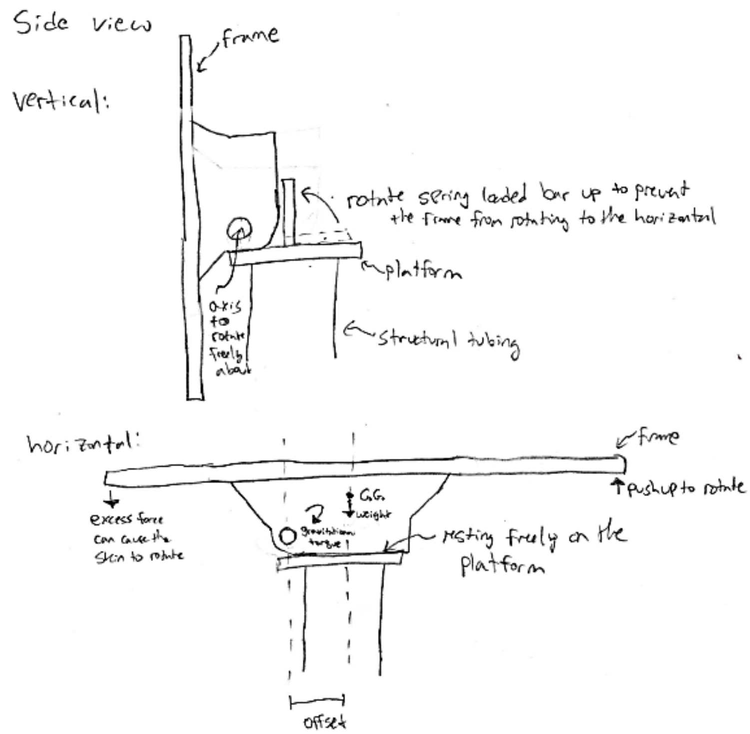


Figure A-5: Offset axis close up

When the frame is vertical, a spring-loaded plate is moved upwards to prevent the frame from rotating back down. When the plate is horizontal there is another edge that is in contact with the platform. Due to the size of the skin, the bent plate or platform cannot be reached by the operator when the skin is in the horizontal position, thus a locking mechanism cannot be used. The skin remains horizontal by the opposing rotational inertia of the weight of the skin and frame.

The strengths and weaknesses to this concept are as follows:

Strengths

- The vertical translations are smooth as the worm gear bears the weight of the skin instead of the operator.

- The height adjustment can be virtually being placed in any position within the maximum and minimum height.
- The device has a small footprint as the structure is in the center of the skin so there is almost nothing in the way when walking around the skin.
- The skin is easy to rotate as there are minimal locking mechanisms in place.

Weaknesses

- There is no rotational locking mechanism in reach of the operator when the skin is horizontal. As a result, if the operator pushes down on the skin then the torque could overcome the inertia and cause the skin to rotate.
- The center of gravity is offset from the center of the frame which reduces the stability of the device when the skin is in the vertical position.
- The device is not as stiff as the truss design as there is a center beam that can bend and buckle.
- Vertical adjustments can only be made when the skin is in the vertical position because the operator would not be able to reach the crank when the skin is horizontal.
- There is minimal material on either side of the skin that would protect the skin while in transport. The skin can hit a desk that is below the height of the uprights.
- There is uncertainty if the off the shelf worm gear can easily be welded and attached to the structural tubing.

Concept 3: Singular Triangular Frame with a Long Clamp Attachment

The "singular triangular frame with a long clamp", shown in Figure A- 6, is a simple handling device design as there is only one frame and one long clamp that holds the base of the composite skin. The concept is shown in Figure A- 6 below.

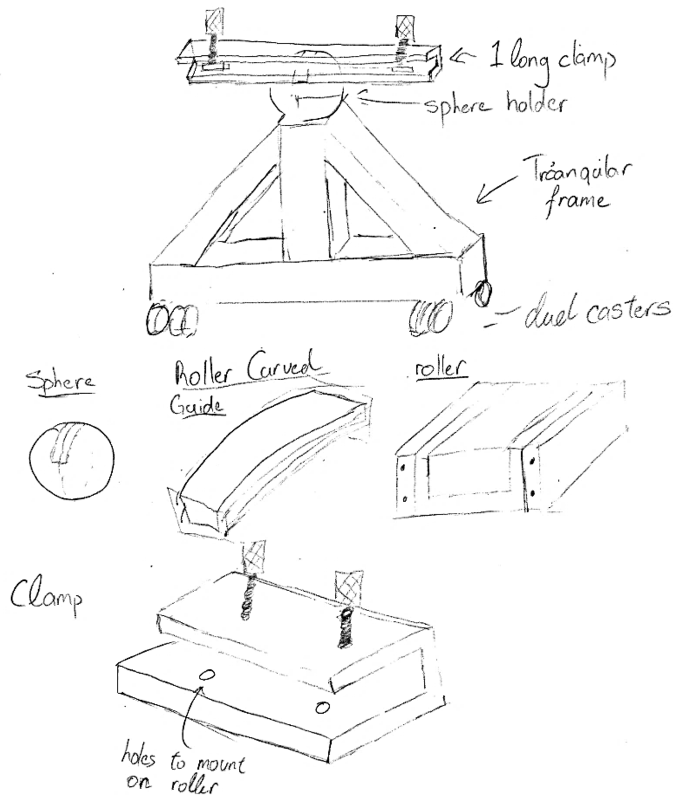


Figure A- 6: Singular triangle frame with long clamp attachment

The clamp sits on a roller and a roller curved guide which is mounted on a sphere. The sphere is welded onto the triangular frame. The roller curved guide is a curved version of the roller linear guide. The roller curved guide sits a quarter of the way down on the sphere and helps with vertical translation for enhanced ergonomics.

The strengths and weaknesses to this concept are as follows:

Strengths

- Vertical translation will be smooth and will be locked through a stopper at both ends of the curved roller guide.
- The clamps are able to hold both composite skin types

Weaknesses

- There is no rotational mechanism with this design
- The roller linear guide will have to be specially machined as there is no off-the-shelf roller curved guide available resulting in a high cost to manufacture.
- The skin can deform as it only supported in the center on both sides.
- The device is not very stable.
- The device does not protect either ends of the skin in the event of a collision
- The triangular frame has a large footprint.

Concept 4: Single Y-arm Using Gears and Hydraulics Mechanism

The “single Y-arm using gears and hydraulics mechanism” can adjust vertically, rotate and adjust to the two types of composite skin. The design is shown in Figure A-7 below.

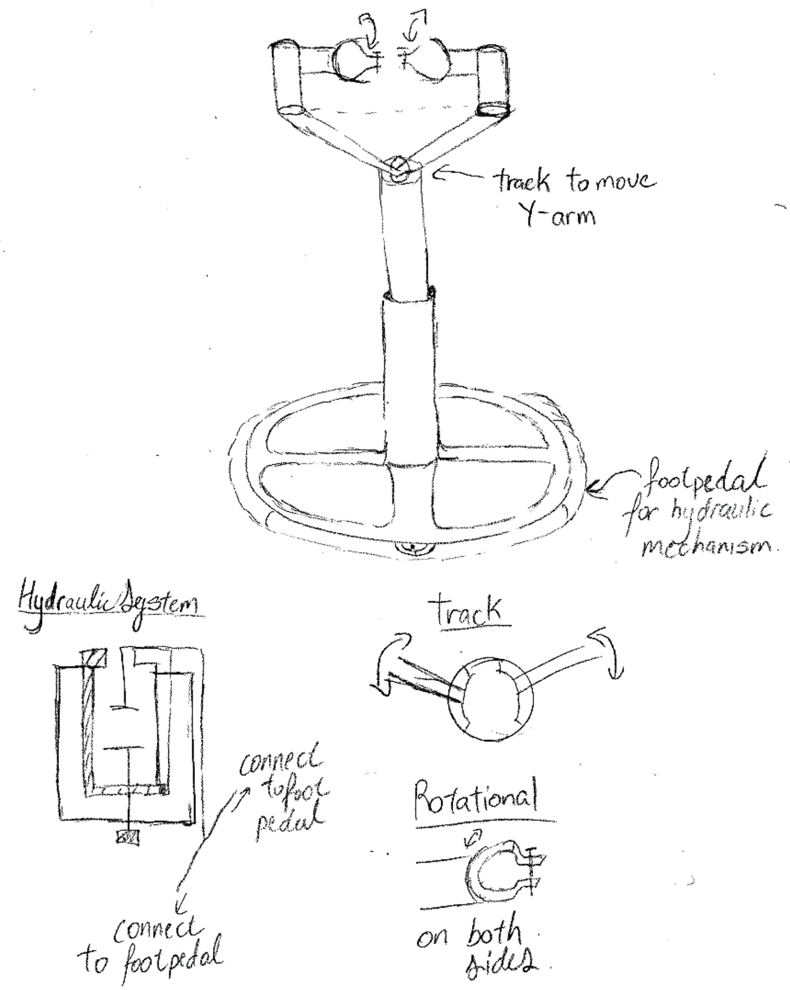


Figure A-7: Single Y-arm using gear and hydraulic mechanism

The device has a single vertical support in the middle that sits on a circular base. The vertical support houses a manual hydraulic system much like an office chair and is attached to a foot pedal at the bottom to activate the vertical translation. The Y-arms are able to extend back and forth from the vertical through a motorized system that will help to support both types of composite skin, trimmed and untrimmed. Through the use of gear system, the C-clamps are adjustable about its x-axis which allows for work to be performed on the front, near vertical position and back side of the composite skin. The design uses four single casters below the circular support with brake system.

The strengths and weaknesses to this concept are as follows:

Strengths

- The device performs all desired functions.
- This device is able to support both skin types.
- Applied load on the skin during work processes would not damage or deflect the skin as the clamps are structurally strong and well manufactured.
- The design is aesthetically pleasing.

Weaknesses

- The device is complex and will require complex manufacturing processes and result in a higher cost.
- The circular base will have a large footprint to prevent the skin from tipping over.
- There are a number of parts that need to be assembled which will reduce the lifespan of the device as there are more parts that could lead to failure.

Concept 5: Circular Mounting Gear Frame on Vertical Bars

The "circular mounting gear frame on vertical bars" design sits on a singular bar with two circular frames attached to height adjustable bars, as shown Figure A-8 below.

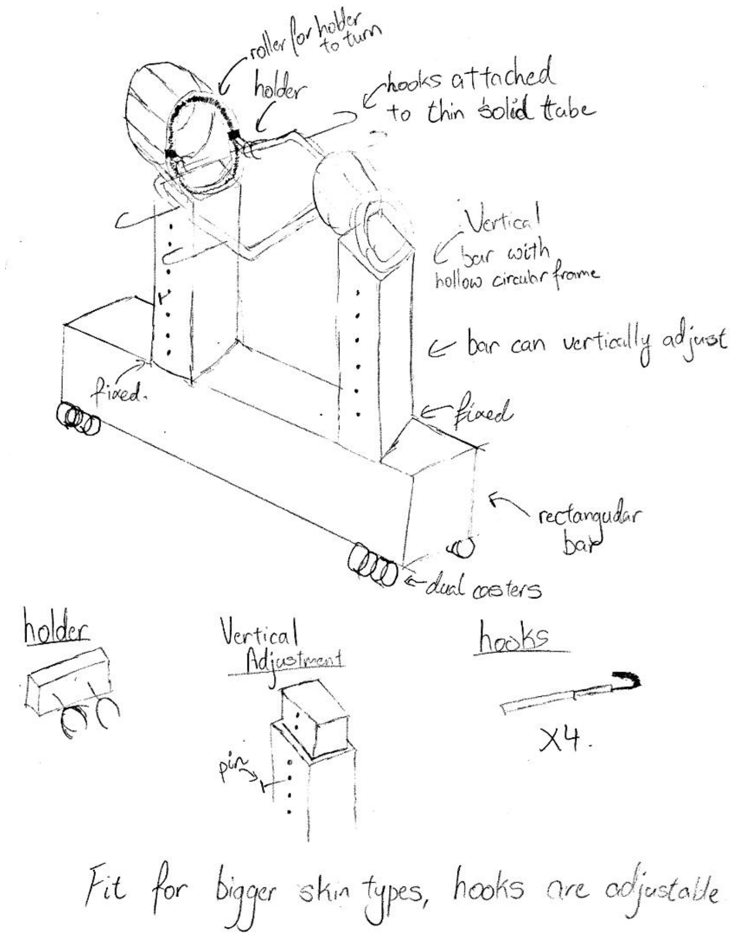


Figure A-8: Circular mounting gear frame on vertical bars

The bars adjust vertically as the inner bar lifts up and locked in position with a pin. The circular hollow frame houses a track in it which can rotate in order for the skin to be worked on the back and front side. The track attaches to the holder and the holder is fixed onto the tubing. The tubing is manufactured for the bigger skin shape and the hooks on the tubing are adjustable in order to hold both skin types in place when in near vertical position. In order to minimize the footprint, the bottom frame variation can be two bars with an attached bar in the middle and the circular mounting sits on the two bars. Also for extra stability, dual casters with locking foot breaks are used.

The strengths and weaknesses to this concept are as follows:

Strengths

- The design is able to perform all required functions.
- The device protects the skin from damage.

Weaknesses

- Two operators are to perform the height adjustment which reduces operational efficiency.
- The track in the hollow circle is difficult to machine resulting in a high manufacturing cost.
- The base will have a large footprint to prevent the skin from tipping over.

Concept 6: Triangular frame with sliding bars

The “triangular frame with sliding bars” consists of a truss design with clamps as shown in Figure A- 9, below.

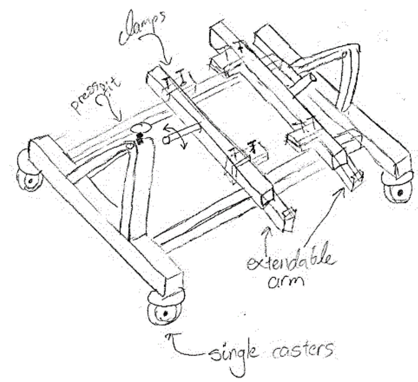


Figure A- 9: Triangle frame with sliding bars

The top of the triangular mount is attached to a rod which is fixed to a horizontal bar with two clamps. The rod is able to slide in and out in order to adjust to skin variation and is also able to rotate to any desired position in order to work on the composite skin. The lock on the bar is a spring pin lock mechanism. The horizontal bar has two clamps that can adjust along the bar and has an extendable bar for skin variation. The clamps are locked with a press fit screw and the extendable bar is locked with a pin system.

The strengths and weaknesses to this concept are as follows:

Strengths

- There truss design results in minimal material required to maintain the required stability and stiffness properties.
- The design simple and does not use many parts. As a result, it is aesthetically pleasing.
- The cost to manufacturing this type of design is relatively low as there are no complex systems.

- The device fully protects the skin during transportation.

Weaknesses

- The skin is susceptible to deflect in the center as there are no bars supporting it.

Concept 7: Flat Plate with Hydraulic System

The “flat plate with hydraulic system”, shown in Figure A-10**Error! Reference source not found.** below, is hydraulic system used to guide and hold the angled railings to any position desired.

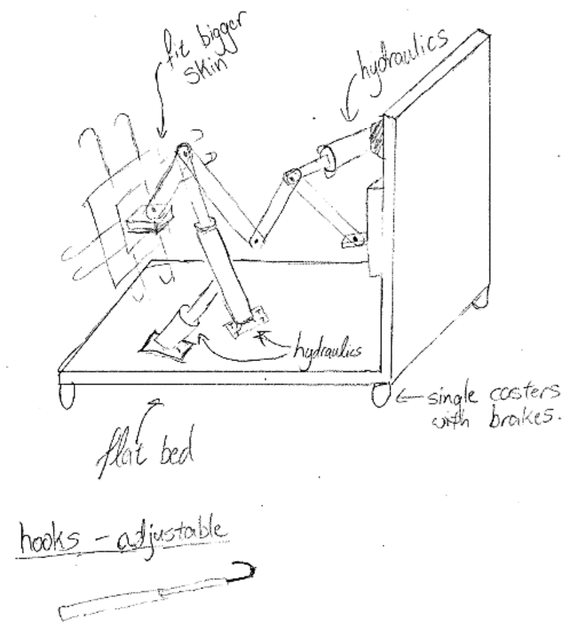


Figure A-10: Flat plate with hydraulic system

The end of the railing is attached to a plate with hooks. The skin is mounted inside the hooks and the hooks are adjustable in order to support the skin in the near vertical position. The hydraulic system must be fully capable to hold not only the railings but also both skin type and the hydraulic system must not fail if any external loads are applied to it and the skin is moved up or down. As movement of the skin is adjusted for enhanced ergonomics purposes, the side to side movements are uncontrolled and failure is possible on the railings and hydraulic frame.

The strengths and weaknesses to this concept are as follows:

Strengths

- The design has controlled movements as the hydraulics will only move to the specific location as per operators guide and will remain at that position.
- The design is aesthetically pleasing
- The device has a small footprint

Weakness

- The design is very complex design with many uncertainties in the design and a large margin of error on designing the hydraulic system.
- There are many moving mechanisms that can fail, especially in the hydraulic frames which could lead to damage on the skin.
- The operator has a low visibility during transportation.
- The skin is only partially protected as it could still be bumped against walls.
- The design is costly to implement.
- The device is not able to move in the horizontal position without increasing the complexity of the design.

Concept 8: Suction Step Design

The "suction step design" is made up of a cantilever bar, tube supports for skin basket, gear system, rubber clamps, and a hand device to help rotate basket. Figure A-11, below shows a sketch of the design.

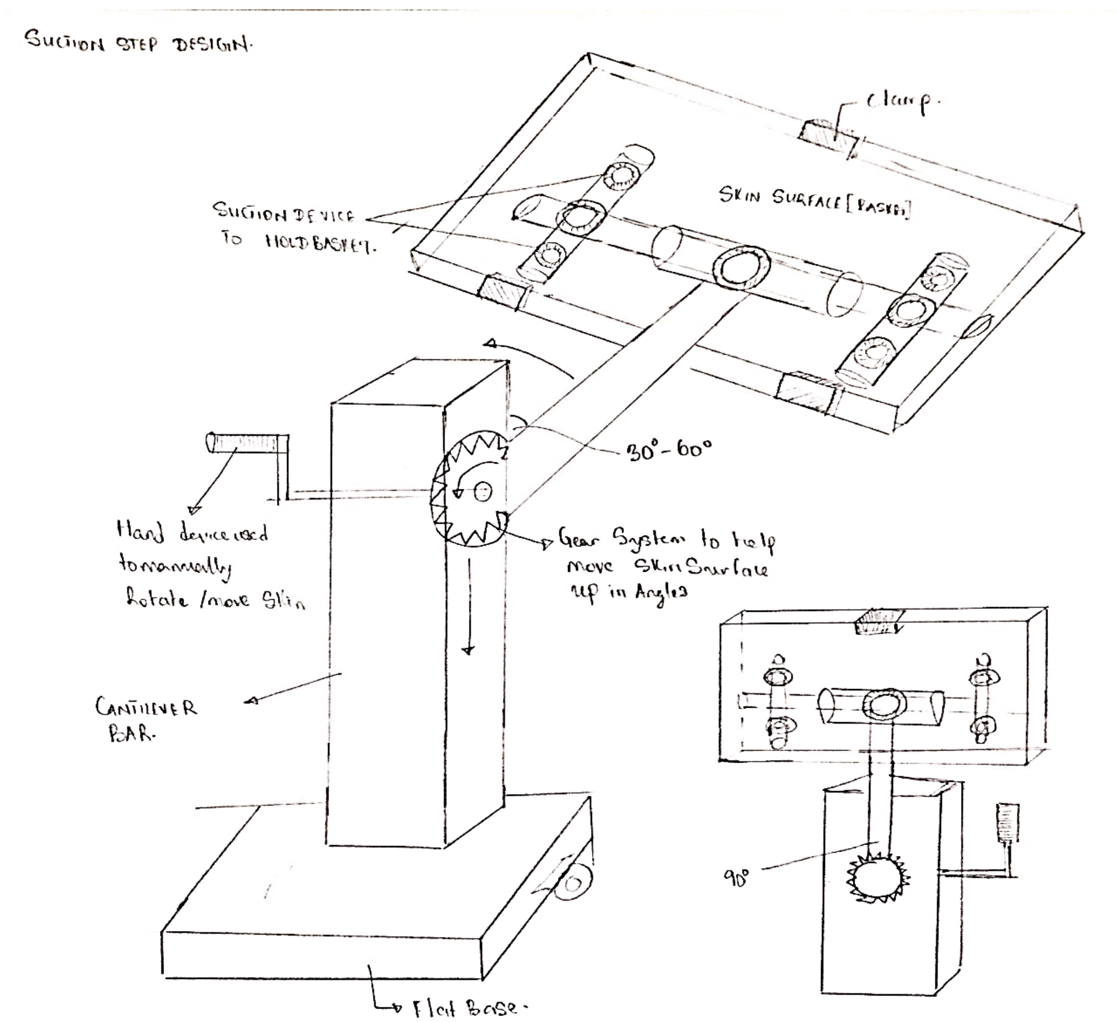


Figure A-11: Sketch of the Suction Step Design

The suction step design is also made up of a cantilever beam with a gear system embedded in the solid beam. Two gears (one for the handle and the other for the cantilever beam) are connected together to control the angular movement (angles of 30°, 60° and 90°) of the arm carrying the skin basket. The handle with the smaller gear turns clockwise or counter clockwise to move the angular bar to various angles. The device there are seven suction cups that hold two cylindrical tubes connected to the cantilever beam. One suction cup is placed in the middle while three are placed on either side of the cylindrical tubes (See Figure A-11 above). This prevents the skin from falling off the basket. The edges of the basket frame have rubber clamps that are used to hold the skin from falling off. Each of the clamps would have a foamy inner surface to prevent any scratch or damage to the skin. The clamps are screwed onto the skin basket.

The strengths and weaknesses to this concept are as follows:

Strengths

- The skin is easy to rotate with this particular design concept since it has gear and connecting bar that helps the operator rotate the device
- The angular bar in this design, would make loading the skin on the device easy. It prevents excessive strain on the operators back when lifting the skin to various heights.

Weaknesses

- The height adjustment mechanism for this device is poorly designed. If device is in the near vertical position then the operators would find it difficult to work on the top ends of the device.
- The extended arm holding the basket makes the device unstable. While in a working position, an operator could easily lean on the arm and this can cause the device to fall over.
- The extended arm can cause injuries to operators passing by the working station.

Concept 9: Four Leg Pinned System

The "four legged" design concept consist of a rectangular skin basket, four cantilever beams, a hydraulic control button, and inner tubes connected to a pin roller to make the basket move to different angles. The figure below is a sketch of the four-legged system.

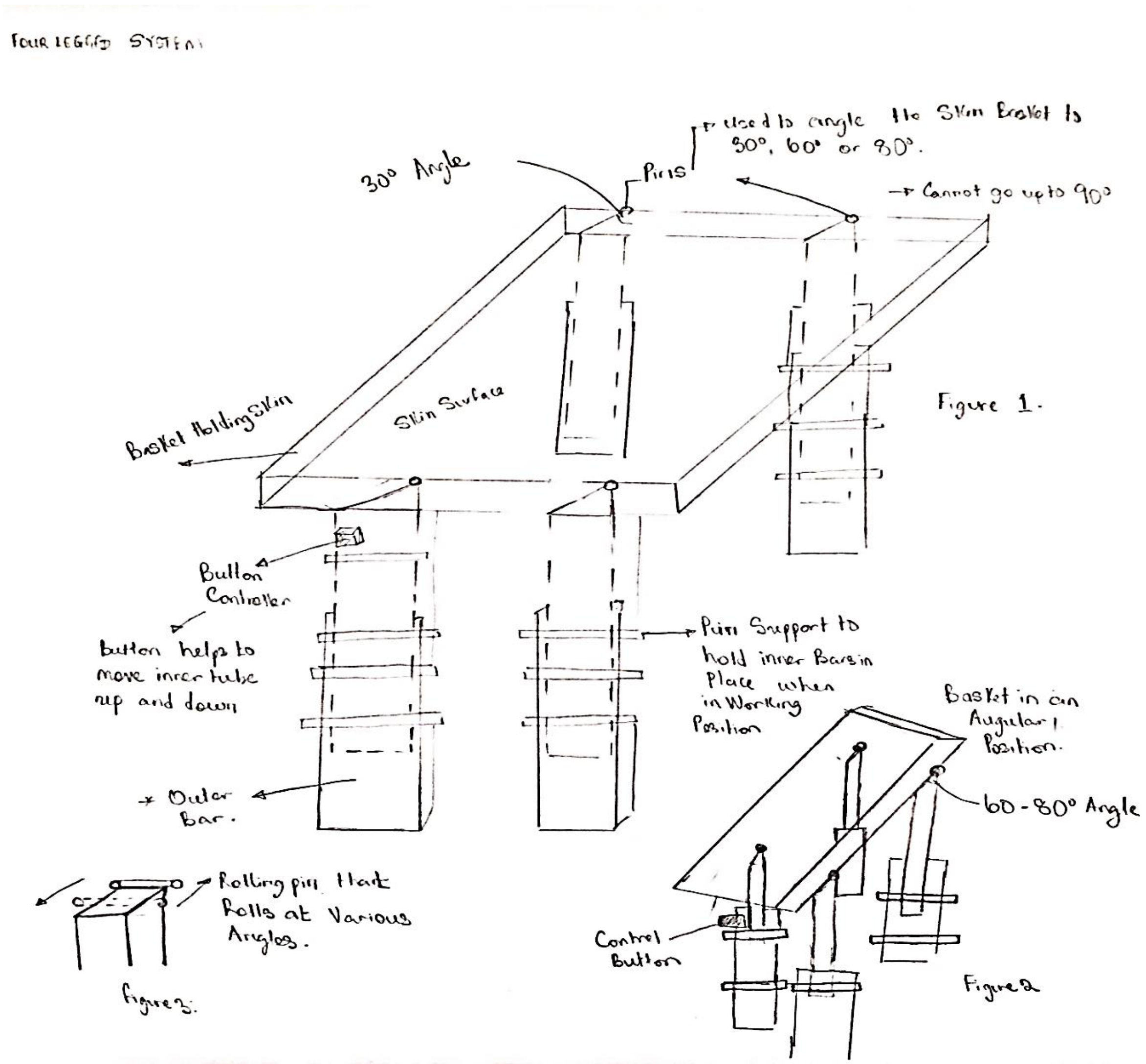


Figure A-12: Four leg pinned system

The four-pinned system is designed to be a rectangular table that has a flat working surface. The flat rectangular surface can tilt from a 30° angle to a 60° angle. A rolling pin sits underneath the rectangular basket. Each leg has an inner tube with angular tips connected to the rolling pins. These angular tips are designed in such a way that they can attain 30°, 60° or 180° angles. The device can be operated easily with a button on the side of front legs. The button controls the inner tubes with angular tips. The back legs can move higher than the front legs such that the working surface is in an angle. The four-pinned system also has a pin lock mechanism that prevents the device from falling down when in working position. Each leg have holes in between the inner and outer tubes. The pins are inserted into these holes to keep both tubes in place thereby making the device stable and safe.

The strengths and weaknesses to this concept are as follows:

Strengths

- Height adjustment are easy with a built-in button to control the vertical movements.
- The device is stable and upright in its working position.

Weaknesses

- The pin tip in this design concept makes it impossible for the device to rotate to a 90° position.
- This design concept would cover a large footprint with components in the way of operators working around the device.

Concept 10: Angular Truss Design

The "angular truss design" is made up of several trusses, a control box, clamps and rotating pin connections. A detailed sketch of the design is seen in the figure below.

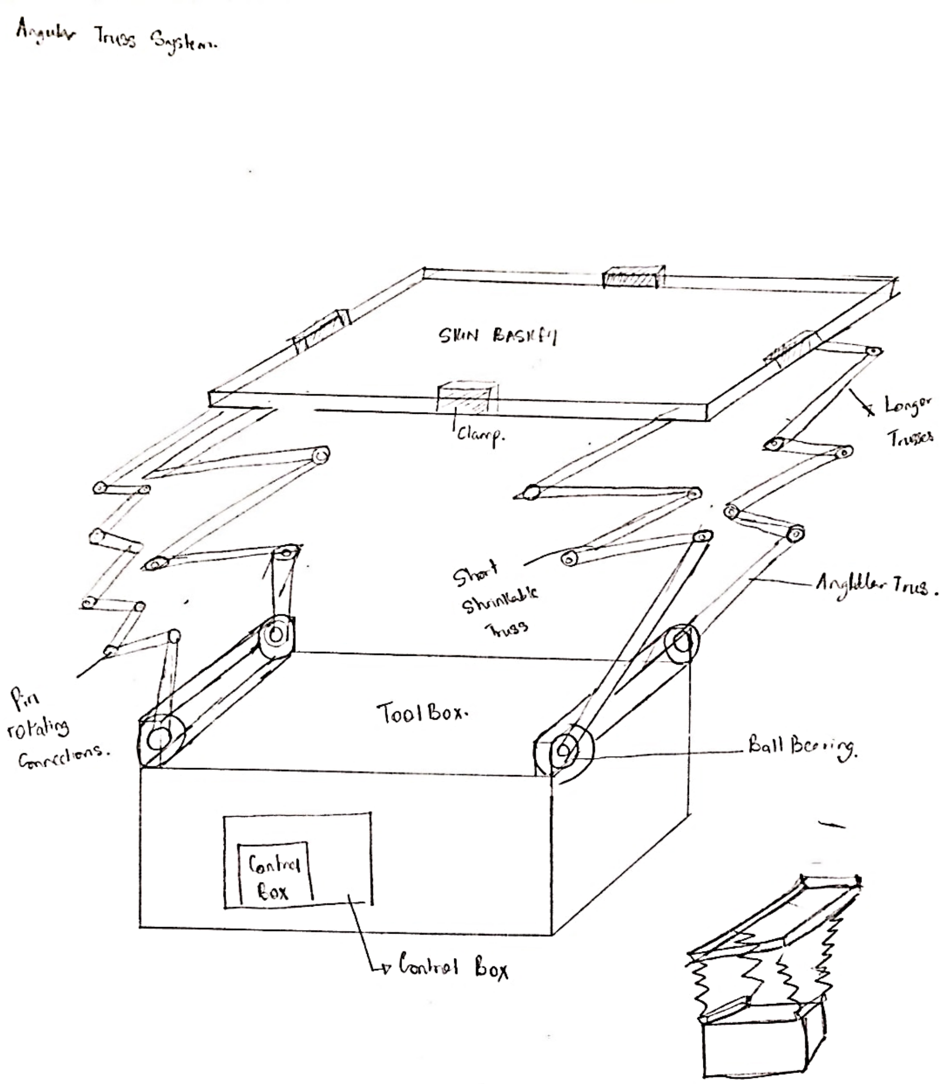


Figure A-13: Angular truss Design Concept

The angular truss spring system is a box with foldable spring trusses. The elbows of the each trusses are joined using connecting pins as shown in Figure A-13, above. The movement of the trusses is controlled using a control box. The control box is equipped with the ability to move the rear trusses to a higher length than the front trusses. This makes it possible for the workers to work on the device in an angular position. At the ends of the spring trusses close to the control box, bearings are attached to help the trusses rotate easily.

The strengths and weaknesses to this concept are as follows

Strengths

- The spring truss design is portable and provides storage space for working tools.
- The device is designed to move easily to various heights just by pressing a button. The extended spring trusses extends to various heights and is ergonomically friendly.

Weaknesses

- It is difficult to implement the design to rotate the skin 90°. The larger the angle results in an increased probability of the device to tip over.
- The spring system has the potential to fail, resulting in damage to the device and potential injury to the operator.
- The maintenance cost would be high if the motor fails. Also, it would be timely to conduct the repairs which would slow down the production of skins.

Concept 11: Pulley Lift Design

The pulley lift design concept is made up of two pulleys, a hydraulic pedal system, a skin basket, a rotating handling device and a belt or chain. The detailed sketch of the design is shown in Figure A-14 below.

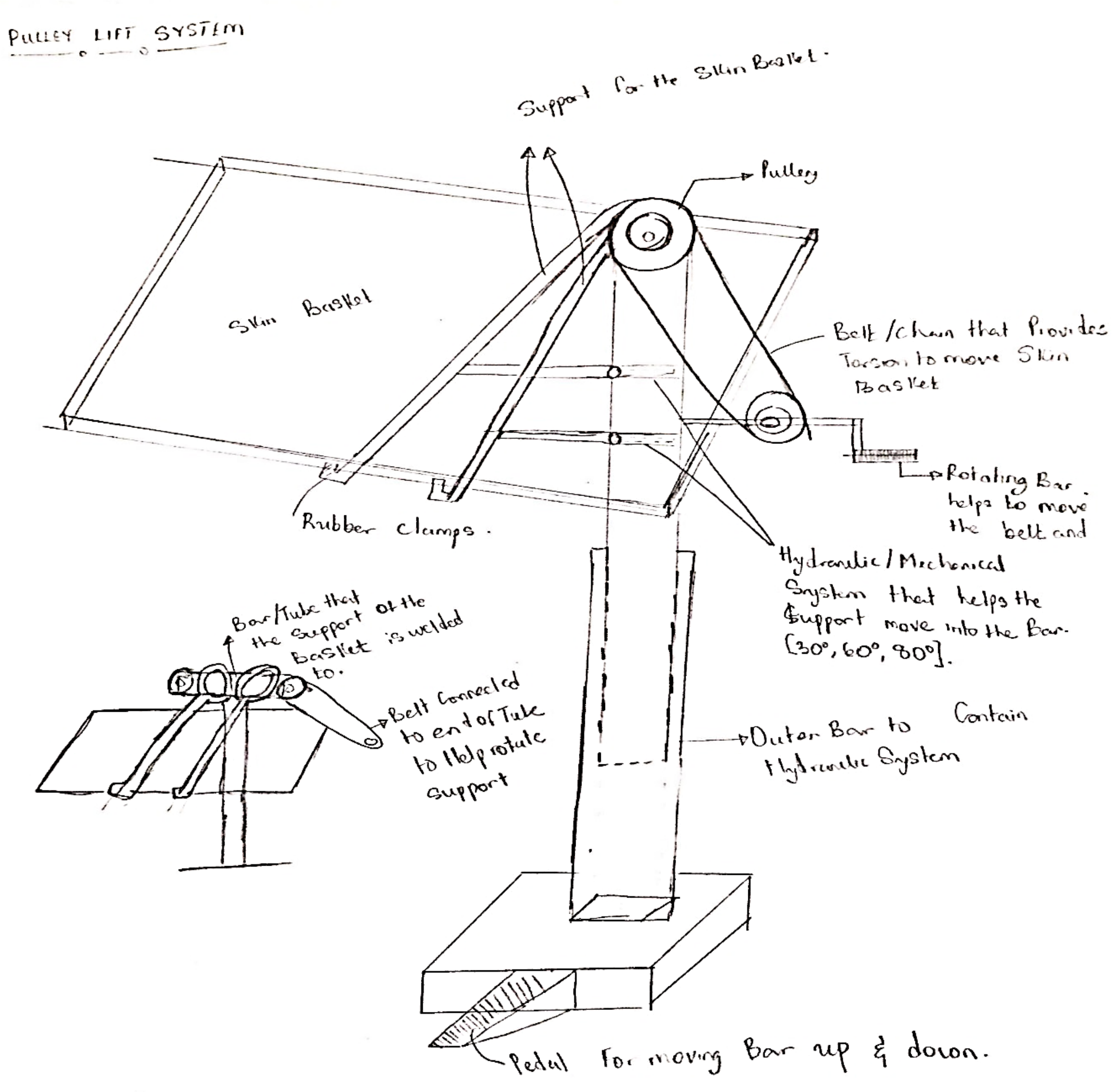


Figure A-14: Pulley Lift Design Concept

The pulley lift system is controlled using two pulleys and a hydraulic lift system. Both pulleys use the tension of a belt or chain drive to control the movement of the skin bar support. The both bars are welded to the main cylindrical bar connected to the cantilever beam. The cantilever beam can be adjusted easily with a hydraulic pedal system. The bigger pulley is connected to one end of the cylindrical bar and then connected with a belt to the smaller pulley. The smaller pulley is connected with bearings to the handle of the device. The handle moves clockwise or counter clockwise to move the support bars to different angles. With the supporting bars there are connecting tubes that move outwards or inwards using a hydraulic mechanism. When the belt is in tension the hydraulic system extends outward and if it is in a relaxed mode the hydraulic system shrinks inward. The skin basket is supported on supporting bars with rubber clamps that holds the skin in place in the basket.

The strengths and weaknesses to this concept are as follows:

Strengths

- The device does not apply excessive strain on the skin when it is stationary or in a rotating motion.
- The belt and hydraulic pedal system makes the pulley system safe and easy to use.
- The operator can easily perform the height adjustments.

Weaknesses

- The pulley design concept is made up of two separate systems; maintaining or manufacturing both systems would be complicated.
- There are multiple components that are likely to fail from fatigue or wearing of jointed parts. If a part were to fail during operation then the skin could be damaged or an operator can become injured.

Concept 12: Truss frame with steering wheel control

The “Truss frame with steering wheel control” consists of a truss like frame with a basket frame to which the skin can be placed into. To rotate the basket and thus the skin, a steering wheel mounted to the upper basket assembly is used. Supporting this rotation is accomplished with a machined two-piece aluminum mount and a simple pin to lock the desired angle. A sketch of the "Truss frame with steering wheel control" is found below in Figure A- 15.

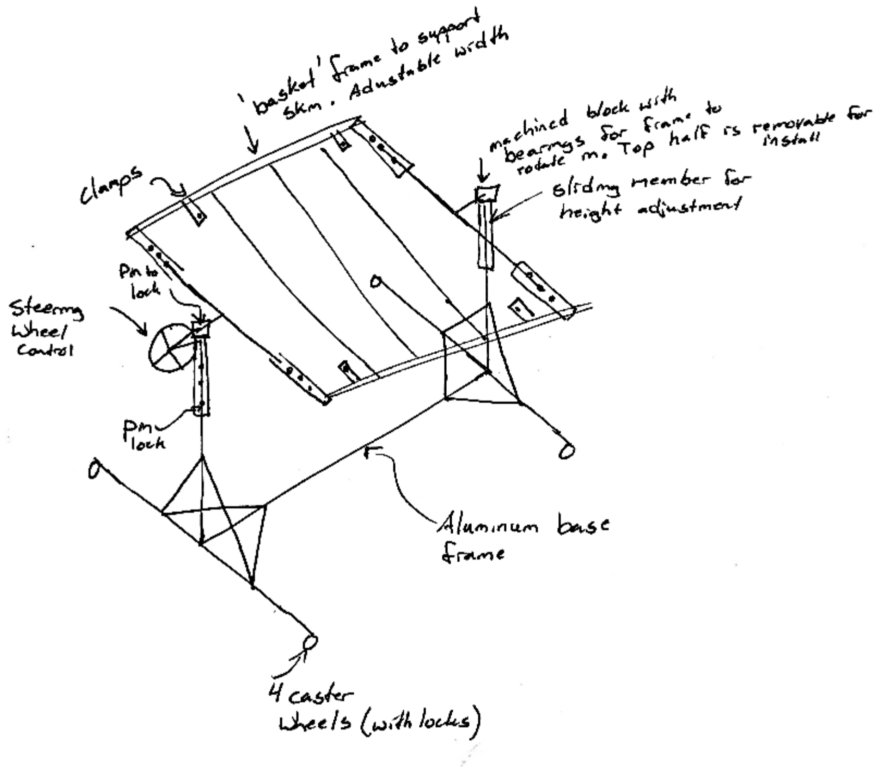


Figure A- 15: Truss frame with steering wheel control

One of the many needs that this design meets is the height adjustability for the worker. This operation is done by removing a pin and sliding the vertical member attached to the steering wheel up and down. The bottom frame that supports the upper basket portion of the design, is constructed from welded together aluminum square tubes. By using this material, the base frame will have greater structural rigidity and not deflect under the expected loads. Moving this device around is done with the use of four caster wheels in the location seen in Figure A- 15. These caster wheels will also have the ability to lock when at the desired working location. The locking will help the device from moving around when work is performed on the skin.

The strengths and weaknesses to this concept are as follows:

Strengths

- The device protects skin.
- The skin will not deflect due to support bars on the non-working surface.
- The basket is adjustable to different size skins.
- The steering wheel helps control rotation.
- The height is adjustable for workers.
- The single middle bar in the base frame allows for close access to skin.
- The locking pins are a fast, easy way to lock in needed position.
- The clamps are simple and maximize working surface.

Weaknesses

- The operator bears the weight of the skin when adjusting the height which can result in a back injury.
- Height adjustments must be performed by two workers.
- The steering wheel may block access to skin working space near it.
- The pins and pin holes may lose tolerance over time from wear and cause unwanted rotation.

2.1. Summary of Concept Designs

All sixteen designs presented in the above sections had features describing their strengths and weaknesses. By using a systemized evaluation process, the most optimal concept was chosen. The screening process consists of evaluating all sixteen designs against the major objectives. The screening process allowed the team to converge to designs that are closer to meeting all of the needs, while staying within the design space. The scoring process further evaluated the top five designs concepts that passed through the screening process and was compared against the list of needs. The following sections describe the scoring and screening process, as well as a sensitivity analysis on the top scoring concept from the scoring analysis.

2.2.Concept Screening

The screening table, TABLE A-II below, is the screening of all twelve generated concepts against the A-frame. The A-frame is taken as a reference concept with grading of zero as it is the current concept. By placing a plus sign or a minus sign, the concept is better or worse than the A-frame cart. Each concept is graded against the major function and need for a quick filtration of the concepts. The results of the pluses, minuses and zeroes of the screening process were summed separately, and a net total was evaluated. The summations of all the designs were ranked. The top five designs that passed the screening process were the truss frame with steering wheel control, triangle frame with sliding bars, external truss pull to turn design, pulley lift design and circular mounting gear frame on vertical bars.

TABLE A-II: SCREENING GRADING OF ALL TWELVE GENERATED CONCEPTS

	Concepts												
Selection Criteria	Single Y-Arm using Gear and Hydraulic Mechanism	Circular Mounting Gear Frame on Vertical Bars	Single Triangular Frame with Long Clamp	Triangular Frame with Sliding Bars	Flat Plate with Hydraulic System	External Truss Pull to Turn	Truss Frame with Steering Wheel Control	Internal Center - drywall lift	Suction Step Design	Four leg Pinned System	Angle Truss Design	Pulley Lift Design	Ref - A-frame
Safe for Operators to Use	-	+	-	+	+	+	+	-	-	+	-	0	0
Skin is not changed in anyway shape or form	-	+	-	+	-	+	+	0	-	+	+	+	0
Ease of Use - (load, function on device)	-	-	-	+	0	+	+	+	+	-	+	+	0
Stability	-	-	-	0	-	0	-	-	-	0	-	-	0
Rotational Ease	-	+	-	+	+	+	+	+	+	-	-	+	0
Height Adjustment	+	+	-	0	+	-	+	+	-	+	+	+	0
Portability	+	+	+	+	+	+	+	+	+	+	+	+	0
Device helps expose skin for work	-	0	-	-	0	0	0	0	0	0	0	0	0
Ease of Manufacturability	-	-	-	-	-	-	-	-	-	-	-	-	0
Complexity in Design	-	-	-	-	-	-	-	-	-	-	-	-	0
Able to attach to variant skin sizes	0	0	0	0	0	0	0	0	0	0	0	0	0
Aesthetics/Professional	+	+	+	+	+	+	+	+	+	+	+	+	0
Pluses	3	6	2	6	5	6	7	5	4	5	5	6	
Same	1	2	1	3	3	3	2	3	2	3	2	3	
Minuses	8	4	9	3	4	3	3	4	6	4	5	3	
Net	-5	2	-7	3	1	3	4	1	-2	1	0	3	
Rank		5		2		3	1					4	
Continue				yes		yes	yes					yes	

2.3.Concept Scoring

The scoring matrix, TABLE A-III, is the rating of each of the top five screened concepts. Each need was listed along with its respective weighting. The weight of each need was the needs' individual hit from the introduction section over the total number of hits hence giving a percentage value. Assigning a value from one to five for each need, a weighted score was calculated for each of the five concepts for the respective need. A number five rating was given if a design performed well for a given objective and a number one rating if it performed terribly. A total score was given to all five concepts and the truss frame with steering wheel control design was the highest scoring concept.

TABLE A-III: SCORING MATRIX OF TOP 5 SCREENING CONCEPTS

Needs	**Weight %**	**Concepts**									
		Truss frame with steering wheel control		**Triangular Frame with Sliding Bars**		**External Truss Pull to Turn**		**Pulley Lift Design**		**Circular Mounting Gear Frame on Vertical Bars**	
		Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Device is safe to use and prevents injuries to users.	6.90	5	0.34	5	0.34	5	0.34	5	0.34	5	0.34
Device prevents damage to the skin during transportation and all work functions performed by operators.	6.65	5	0.33	5	0.33	5	0.33	5	0.33	5	0.33
Devices does not cause excessive strain to the skin while stationary, lifting or rotating.	6.40	5	0.32	3	0.19	5	0.32	5	0.32	5	0.32
Device does not fall over with/without skin.	6.16	5	0.31	4	0.25	5	0.31	3	0.18	2	0.12
Device supports weight of the skin and maximum external loading of 30 pound force.	5.91	5	0.30	4	0.24	4	0.24	3	0.18	3	0.18
Device adjusts to dimensional variation of skin; trimmed and untrimmed skin.	5.67	5	0.28	2	0.11	5	0.28	5	0.28	2	0.11
All movements are controlled.	5.42	5	0.27	5	0.27	4	0.22	2	0.11	3	0.16
Able to rotate.	5.17	5	0.26	5	0.26	4	0.21	2	0.10	0	0.00
Skin is stable when in working position.	4.93	4	0.20	4	0.20	5	0.25	2	0.10	5	0.25
Device is able to work at ergonomic position for range of workers (height, length, and width).	4.68	3	0.14	0	0.00	3	0.14	5	0.23	5	0.23
The skins stays stable and firm during transportation.	4.43	4	0.18	5	0.22	5	0.22	2	0.09	5	0.22
Device needs to experience minimal deflections when subjected to external loads.	4.19	5	0.21	5	0.21	5	0.21	5	0.21	5	0.21
Device needs to be stationary when in working position.	3.94	5	0.20	5	0.20	5	0.20	5	0.20	5	0.20
Device is portable with or without a skin.	3.69	5	0.18	3	0.11	4	0.15	2	0.07	4	0.15
Device needs to maximize all working surface while in working position.	3.45	4	0.14	4	0.14	3	0.10	2	0.07	4	0.14
Device is easy and efficient to load and operate.	3.20	4	0.13	4	0.13	5	0.16	3	0.10	4	0.13
Device needs to have a long lifespan.	2.96	5	0.15	5	0.15	5	0.15	5	0.15	5	0.15
Device choice of material needs to be based on environmental condition.	2.71	5	0.14	5	0.14	4	0.11	3	0.08	4	0.11
Movement requires minimal force and motions are smooth.	2.46	5	0.12	5	0.12	5	0.12	5	0.12	5	0.12
Device is designed such that professional engineering processes can be used to manufacture it.	2.22	4	0.09	4	0.09	3	0.07	5	0.11	5	0.11
Operators are comfortable to work around the device and have ample room at any one location.	1.97	5	0.10	5	0.10	5	0.10	4	0.08	5	0.10
Device needs be a simple design.	1.72	4	0.07	5	0.09	3	0.05	2	0.03	5	0.09
Device is easy to maintain.	1.48	5	0.07	5	0.07	3	0.04	2	0.03	3	0.04
Device needs to minimize footprint.	1.23	5	0.06	5	0.06	3	0.04	2	0.02	3	0.04

Needs	Weight %	Concepts									
		Truss frame with steering wheel control		**Triangular Frame with Sliding Bars**		**External Truss Pull to Turn**		**Pulley Lift Design**		**Circular Mounting Gear Frame on Vertical Bars**	
		Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score	Rating	Weighted Score
Device can maneuver through tight corners (and rotate like a chalkboard).	0.99	5	0.05	5	0.05	4	0.04	3	0.03	3	0.03
Device does not have unpleasant noise while performing any function.	0.74	5	0.04	5	0.04	5	0.04	5	0.04	5	0.04
Device has reasonable cost to manufacture.	0.49	5	0.02	5	0.02	5	0.02	5	0.02	5	0.02
Device design minimizes environmental impact.	0.25	4	0.01	5	0.01	4	0.01	3	0.01	3	0.01
Device is pleasing to the eyes.	0.00	5	0.00	5	0.00	5	0.00	5	0.00	5	0.00
Net Total		4.70		4.14		4.46		3.65		3.95	
Ranking		1		3		2		5		4	

2.4. Optimization of Truss Frame with Steering Wheel Control

The concept that passed the screening and scoring is the truss frame with steering wheel control concept. The sections of this appendix discuss the optimization of this finalized design in order to increase the scores that this concept did poorly on. Rotational mechanism and upper frame scored the lowest for the truss with steering wheel concept. APPENDIX A described which concept was used to integrate into the winning design in order to create a functional rotational mechanism and how the scoring changed. APPENDIX C described changes to the upper frame. Changes to the upper frame led to changes to the vertical adjustment and skin to frame interface which will be discussed in APPENDIX C.

2.4.1. Rotational Mechanism Optimization

The two-piece housing design from the truss frame with steering wheel control is the best for assembling the device, however, the locking mechanism was further optimized. The two cases to analyze the locking mechanism were a pin locking mechanism, shown in Figure A-16 and a screw locking mechanism shown in Figure A- 17.

TABLE A-IV, below, shows the criteria used for scoring the two design options and which mechanism ranked highest. The criteria and weights are based off scoring table in Appendix A, using the needs that are relevant to the rotation mechanism function.

Figure A-16: Rotational mechanism with pin

Figure A- 17: Rotational mechanism with screw

TABLE A-IV: SCORING OR ROTATIONAL MECHANISM

Rotational Component					
Need	Weight	Pin		Screw	
		Rate	Weighted score	Rate	Weighted score
Device is safe to use and prevents injuries to users.	6.9	5	0.35	5	0.35
Device supports weight of the skin and maximum external loading of 30-pound force	5.91	5	0.30	5	0.30
Skin is stable when in working position	4.93	5	0.25	4	0.20
Device needs to have a long lifespan	2.96	5	0.15	5	0.15
Device needs be a simple design	1.72	5	0.09	3	0.05
Device is easy and efficient to load and operate	3.2	5	0.16	3	0.10
Device has reasonable cost to manufacture	0.49	5	0.02	4	0.02
	Total	1.31		1.15	

The pin design used as a locking mechanism scored the highest and was used in the final concept. To further expand on the final concept for the rotational mechanism, two bearings were used to help with smooth rotational motion. These bearings are both deep groove ball bearings and located by grooves cut into the two-piece block. Securing the bearings and two halves together was performed using four M5 bolts, threaded into the block itself.

2.4.2. Upper Frame Optimization

The “Truss with Steering Wheel Control” also had low scoring points with the “Maximize working surface of skin” need. The design did not incorporate any possible way for the back side of the composite skin to be worked on as per operator’s demand. The top scorer for this particular need was the “Triangle frame with sliding bar” design as it allowed for both sides of the skin to be accessed; maximizing the working surface. Integrating the upper frame design of “Triangle frame with sliding bar concept” to the “Truss with Steering Wheel Control” improved the design and allowed it to best meet the need.

Further scoring was performed on the selection of member type for the upper frame. Figure A-18: Round tubing for upper frame and Figure A-4, below, show the two options of round tubing and square tubing for the upper frame and were scored against each other in order to select the best option. The results of the scoring are seen below in Figure A-1 and based on the highest scoring value which option was chosen.

Figure A-18: Round tubing for upper frame

Figure A-19: Square tubing for upper frame

TABLE A-V: UPPER FRAME SCORING

Upper Frame Component		
Selection Criteria	**Round tubing**	**Square tubing**
Safe for operators to use	5	5
Cost	5	5
Complexity	5	4
Ease of sliding	5	5
Single direction of motion	0	5
Total	20	24

TABLE A-V above, shows the square tubing was the better choice between the two options. The round tubing failed for the "Single direction of motion" criteria because as translation along the inner tube slides through the outer tube, it will rotate. However, with two square tubes over each other they will slide along one another but will not rotate due to their edges. Due to the rotation of the inner circular tube, the skin would experience additional load as the clamping mechanism also rotates.

The upper frame ties into the rotational mechanism with holes in the round tube seen below in Figure A-20. These holes are located such that the skin can be orientated in the horizontal position with access to one side or rotated 180^{o} and locked again for access to the other side of the skin. There is also a set of holes that lock the skin in the 80^{o} from the horizontal during transportation of the skin.

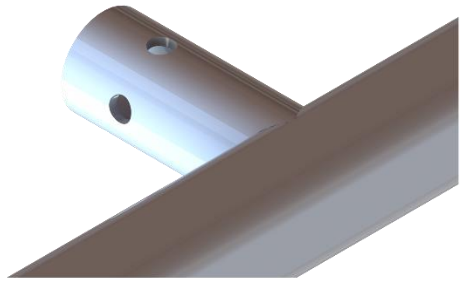


Figure A-20: Close-up of holes that tie into pin of rotational mechanism

2.4.3. Vertical Adjustment

Vertical adjustments were difficult to perform in the original truss frame with steering wheel control concept because the upper frame has a huge force downward during height translation. Multiple operators are required and they must bear the full load of the skin and upper frame when the pin is removed from the vertical adjustment mechanism. The systems that use hydraulics and worm gears to adjust vertically performed the best for vertical adjustments, however, those designs used a center column instead of two columns spaced on either side of the skin. If a crank/piston were placed on both sides of the skin then the adjustment would need to synchronize on both sides. Two operators could perform the adjustment at the same time but there is the risk that one side becomes slightly higher than the other. This would place additional strain on the device and could make for an uneven working surface. Control systems can be put in place to force the pistons to move in union, however, there are no readily available systems off-the-shelf and there would be added complexity to design such a system.

A variation to the current system is to use the mechanism shown in Figure A- 21, below.

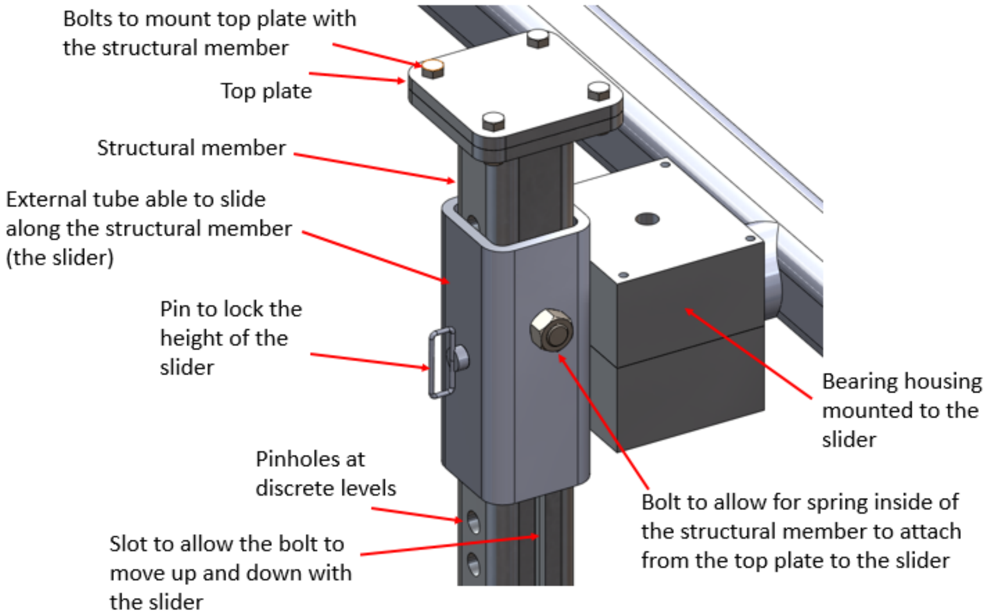


Figure A- 21: Close-up render of the height adjustment mechanism

The mechanism above uses pins to lock the adjustment at discrete levels so that the final height on both sides would be equal after the adjustment was completed. Pin holes are cut into the side of the structural member to allow for a pin to be inserted through a hole in the slider to lock the vertical position. The slider has a bolt that runs through the structural member to allow for a spring to hook onto the slider. The other end of the spring hooks onto the top plate. The spring is used to bear a portion of the skin and clamping frame load once the pin is removed. This allows for the operators to apply minimal force to move the skin up and down. The reduced force will prevent muscular injuries and will help when the pin needs to be inserted back into the hole. A slot is cut into the structural member so that the bolt can move up and down freely. The bearing housing is mounted to the side of the slider so it will move up and down as the slider moves. Internal images of the slider assembly are shown in Figure A- 22, below.

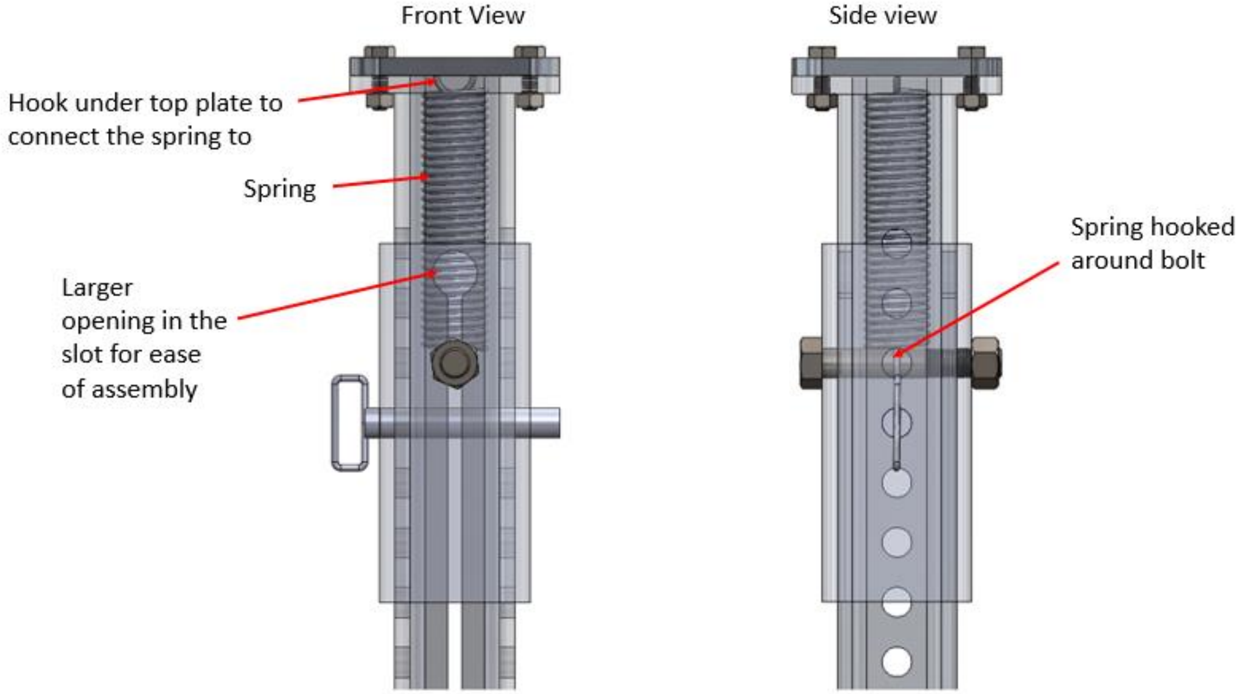


Figure A- 22: Internal renders of the spring mechanism

The larger opening in the slot shown in the figure above is in place to assist with the assembly. The bottom hook in the spring and the bolt hole in the slider would be aligned at the opening in the slot so that the bolt can inserted.

A variation for the pin mechanism is to combine the pin and the bolt into one part that complete both designed functions at once. This is completed by placing the pin holes along the same side as the slot and modifying the bolt; a render of this can be seen in Figure A- 23, below.

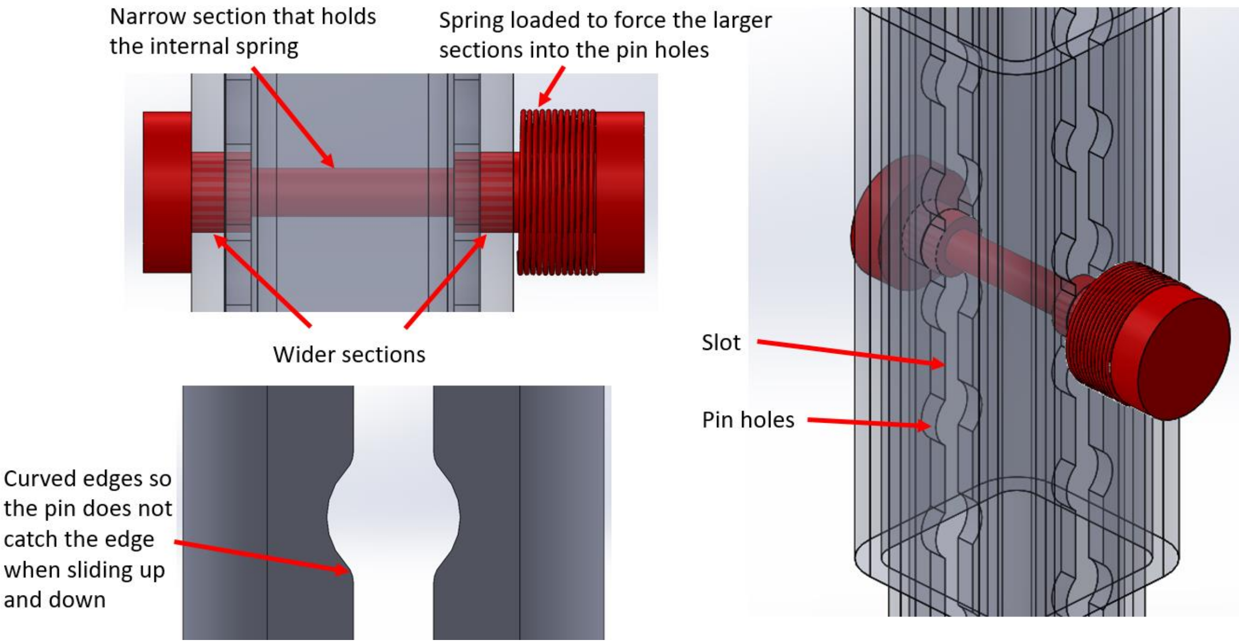


Figure A- 23: Height adjustment mechanism with the pin and bolt combined

The pin has wider sections that are slightly smaller than the size of the pin holes. The narrow section of the pin is slightly smaller than the diameter of the slot. Thus, when the pin is shifted, the wider sections will no longer be in contact with the edges of the pin hole and the pin is free to move up and down. There is spring added to the assembly to force the pin to remain in the locked position. The sharp edges between the pin hole and the slot would be curved to ensure that the pin doesn't get caught when sliding up and down. One advantage to the variation is that less parts are required in the assembly. Also, the pin can never become lost which is more efficient to use in operation. The downside is that the pin will be more difficult and costly to manufacture as it cannot be purchased off the shelf. Also, the spring is susceptible to wear reducing the overall lifespan of the device. Both concepts were scored against each other to determine the concept to move forward with in the final design.

TABLE A-VI: VERTICAL ADJUSTMENT SCORING

Vertical alignment locking mechanism					
Need	**Weight**	**Pin and bolt combined**		**Pin and bolt used separately**	
		Rate	**Weighted score**	**Rate**	**Weighted score**
Device is safe to use and prevents injuries to users.	6.9	5	0.35	5	0.35
Device supports weight of the skin and maximum external loading of 30 pound force	5.91	5	0.30	5	0.30
Device is easy and efficient to load and operate	3.2	5	0.16	4	0.13
Device needs to have a long lifespan	2.96	3	0.09	5	0.15
Device needs be a simple design	1.72	2	0.03	5	0.09
Device has reasonable cost to manufacture	0.49	2	0.01	5	0.02
Total		0.93		1.03	

The design that used the pin and bolt separately scored higher and will be used as the mechanism moving forward into the final design.

2.4.4. Skin to Frame Interface

Now that the structural bars behind the skin have been removed, the mounting system needs to be redesigned. The interface needs to hold the skin in place and allow for the skin to be easily loaded and unloaded. One way to achieve both is to have a modified c-clamp style mechanism that is shown in the Figure A-24 below.

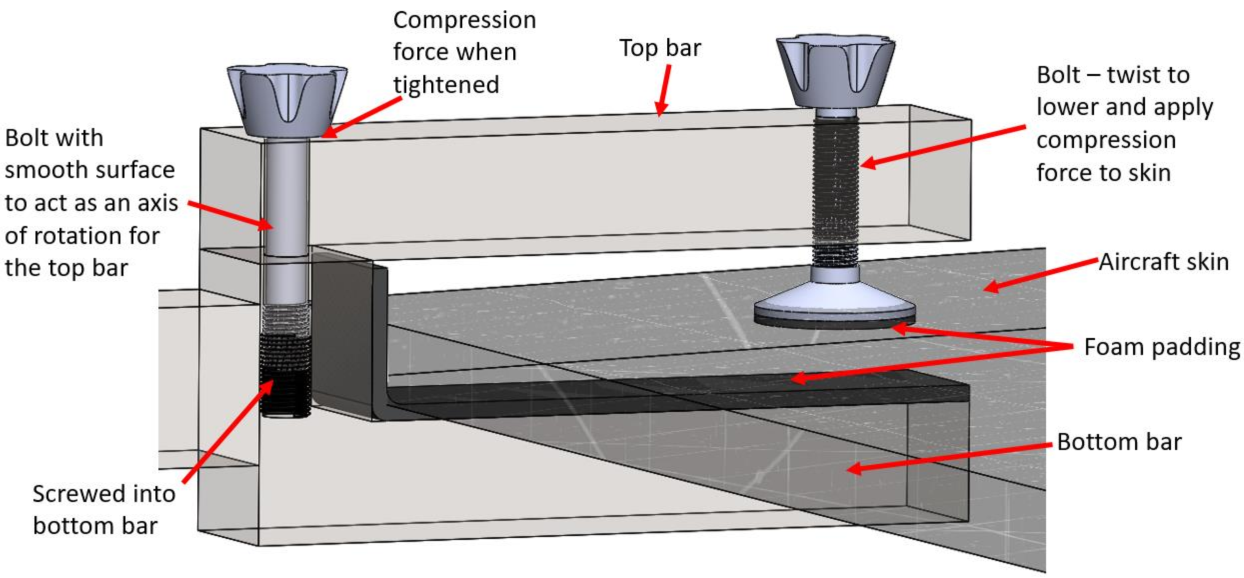


Figure A-24: C-clamp style mount

The device allows for the skin to rest on foam padding while a bolt can be turned to lower another platform that applies a compressive force to the skin. The design also allows for the bottom edge skin to rest against the back of the clamp which will take some of the stress when the skin is in the vertical position. The top and bottom of the clamp are independent from each other. The top bar is free to rotate about the second bolt with the smooth surface in the figure above. The same bolt can be tightened which will provide a clamping force between the top bar and bottom bar which will prevent the top bar from rotating. The rotation function allows for the top bar to be moved out of the way for an ease of loading the skin on the bottom bar. It also allows for the top surface to be fully exposed when the operator is performing work on it. Figure A-25, below, shows the top bar performing the movement.

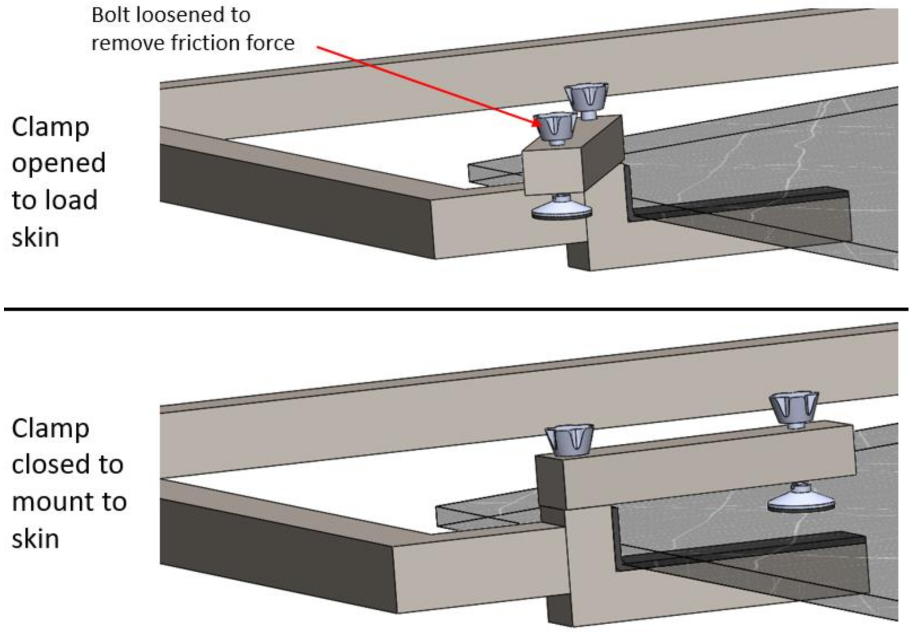


Figure A-25: Top bar rotation function on the clamp

The bottom bar covers a section of the back of the skin. The clamp would need to be loosened and the skin shifted over in order to access the covered area to perform work on it.

One variation for the clamp was to allow the clamp to rotate about an axis normal to the skin. The top right of skin is at an angle relative to the rest of the skin. The ability to rotate allows for the back end of the clamp to be in contact with the skin at that side. The function also allows for the entire clamp to move out of the way to expose the top and bottom surface of the skin. This clamp variation can be seen in Figure A-26, below.

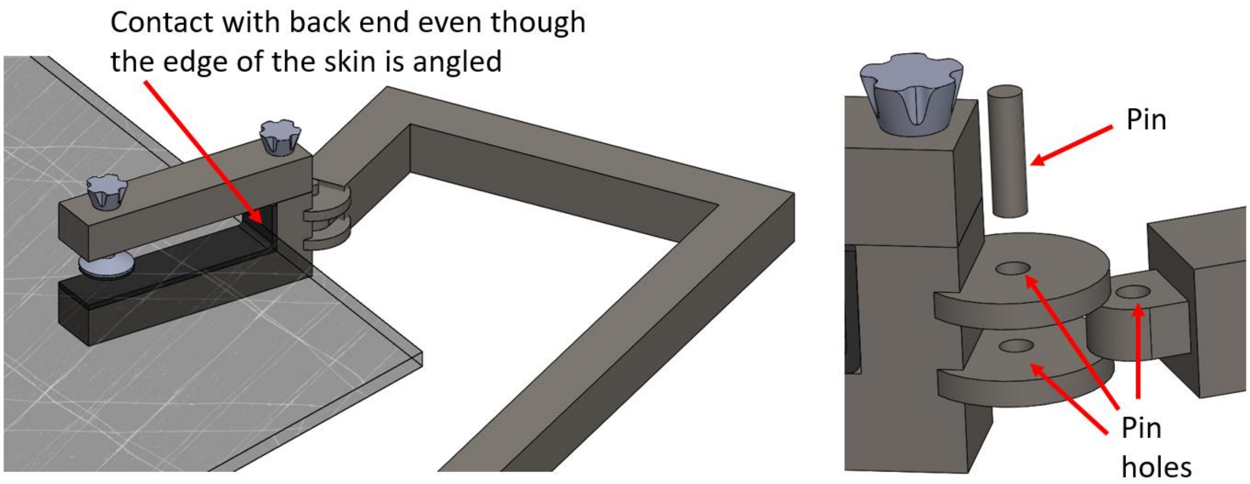


Figure A-26: Rotating clamp design mechanism

The clamp is able to rotate by having circular disks extend out of both the clamp and the frame. A pin is placed in the center of both disks which will act as an axis for the clamp to rotate about. The concepts were scored against each other to determine the concept to move forward with.

TABLE A-VII: SKIN TO FRAME INTERFACE SCORING

Clamping mechanism							
Need	Weight	Parallel clamp		Angled clamp		Rotational clamp	
		Rate	Weighted score	Rate	Weighted score	Rate	Weighted score
Device is safe to use and prevents injuries to users.	6.9	5	0.35	5	0.35	5	0.35
Device supports weight of the skin and maximum external loading of 30-pound force	5.91	5	0.30	4	0.24	5	0.30
Device is easy and efficient to load and operate	3.2	5	0.16	5	0.16	5	0.16
Device needs to have a long lifespan	2.96	5	0.15	4	0.12	3	0.09
Device needs be a simple design	1.72	5	0.09	4	0.07	4	0.07
Device has reasonable cost to manufacture	0.49	5	0.02	5	0.02	4	0.02
Total		1		0.95		0.99	

The parallel clamp scored the highest and was the concept used moving forward.

3.1Summary of Optimized Truss Design

The adjusted truss design was scored against the original concept to determine if the adjustments resulted in a higher score.

TABLE A-VIII: SCORING TABLE OF ADJUSTED TRUSS DESIGN TO ORIGINAL CONCEPT

Needs	**Weight %**	**Concepts**			
		Truss frame with steering wheel control		**Truss frame optimized**	
		Rating	Weighted Score	Rating	Weighted Score
Device is safe to use and prevents injuries to users.	6.9	5	0.34	5	0.34
Device prevents damage to the skin during transportation and all work functions performed by operators.	6.65	5	0.33	5	0.33
Devices does not cause excessive strain to the skin while stationary, lifting or rotating.	6.4	5	0.32	4	0.26
Device does not fall over with/without skin.	6.16	5	0.31	5	0.31
Device supports weight of the skin and maximum external loading of 30 pound force.	5.91	5	0.3	5	0.3
Device adjusts to dimensional variation of skin; trimmed and untrimmed skin.	5.67	5	0.28	5	0.28
All movements are controlled.	5.42	5	0.27	5	0.27
Able to rotate.	5.17	5	0.26	5	0.26
Skin is stable when in working position.	4.93	4	0.2	5	0.25
Device is able to work at ergonomic position for range of workers (height, length, and width).	4.68	3	0.14	5	0.23
The skins stays stable and firm during transportation.	4.43	4	0.18	5	0.22
Device needs to experience minimal deflections when subjected to external loads.	4.19	5	0.21	5	0.21
Device needs to be stationary when in working position.	3.94	5	0.2	5	0.2
Device is portable with or without a skin.	3.69	5	0.18	5	0.18
Device needs to maximize all working surface while in working position.	3.45	4	0.14	5	0.17
Device is easy and efficient to load and operate.	3.2	4	0.13	4	0.13
Device needs to have a long lifespan.	2.96	5	0.15	5	0.15
Device choice of material needs to be based on environmental condition.	2.71	5	0.14	5	0.14

Needs	Weight %	Concepts			
		Truss frame with steering wheel control		Truss frame optimized	
		Rating	Weighted Score	Rating	Weighted Score
Movement requires minimal force and motions are smooth.	2.46	5	0.12	5	0.12
Device is designed such that professional engineering processes can be used to manufacture it.	2.22	4	0.09	5	0.11
Operators are comfortable to work around the device and have ample room at any one location.	1.97	5	0.1	5	0.1
Device needs to be a simple design.	1.72	4	0.07	4	0.07
Device is easy to maintain.	1.48	5	0.07	4	0.06
Device needs to minimize footprint.	1.23	5	0.06	5	0.06
Device can maneuver through tight corners (and rotate like a chalkboard).	0.99	5	0.05	5	0.05
Device does not have unpleasant noise while performing any function.	0.74	5	0.04	5	0.04
Device has reasonable cost to manufacture.	0.49	5	0.02	5	0.02
Device design minimizes environmental impact.	0.25	4	0.01	5	0.01
Device is pleasing to the eyes.	0	5	0	5	0
Net Total		4.7		4.87	
Ranking		2		1	

The table indicates that the adjustments resulted in a higher score. Thus, the adjusted truss design is the final concept to be optimized.

APPENDIX B –Material Properties used in the analysis

1. Aircraft skin material properties

The client is unable to provide exact material specifications or available test data on the skin due to proprietary restrictions. As a result, a material will need to be selected and modeled. The table below indicates the uncertainties with the skin properties and the assumptions used in order to conduct an analysis.

TABLE B-I: MATERIAL UNCERTAINTY

Uncertainty based on the limit on the information allowed to disclose.	**Assumption used in the analysis**
Material type	It is publically available that the aircraft skin on the F-35 is comprised mostly of a carbon fiber with BMI provided from Cyntec [27]. As a result material properties of the skin are taken from the BMI datasheets provided by Cyntec [28].
BMI composition throughout the skin	Skin is assumed to have a 100% BMI composition
The contour of the skin	Skin is assumed to have a constant thickness and is planar. The thickness used is 11/32 inches which was the approximate average of the least thick and greatest thickness sections of the skin.
They layup of the carbon fiber	Strength and modulus properties are provided for a layup of [+45,0,-45,90]2s. As a result, the material is treated as isotropic and linear elastic with a design strength of 45 ksi and a modulus of elasticity of 9.1 Msi. The selected design strength is the open hole compressive strength of the material with the [+45,0,-45,90]2s layup.
The temperatures that the skin will be exposed to in any of the processes	Material properties are taken at room temperature (24°C).
The poisson’s ratio for the material is unavailable on the datasheet	A poisson’s ratio of 0.36 is selected based on the value used for BMI in table II of the American Society of Composites-28th Technical Conference [29].
Material density	The density used for gravitational analysis on the skin was set to 0.205 lb/ft^2 in order for the weight of the model to equal 75 lbs.

Due to the uncertainties about skin specifications, a high factor of safety of 4 is used in the analysis to ensure that the skin does not experience excessive stresses when subjected to external loads. Any major deviations from the assumptions used need to be considered prior to implementing the design.

2. AISI 1020 cold rolled steel properties

The material properties for AISI 1020 used in the analysis of the report is provided in the table below [30]:

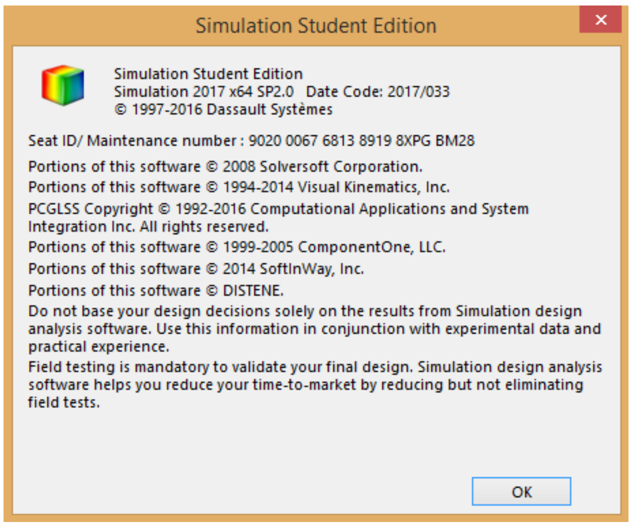
TABLE B- II:MATERIAL PROPERTIES FOR AISI 1020 STEEL

Properties	Imperial
Tensile strength	60900 psi
Yield strength	50800 psi
Modulus of elasticity	29700 ksi
Shear modulus (typical for steel)	11600 ksi
Poisson’s ratio	0.29

APPENDIX C – Detail Design Mathematical Analysis

This section provides the mathematical analysis and methodology used to design and optimize each component of the device. This section also includes the analytical and computational calculations used to determine the final specifications on each component. The loading force used on each section is 125 lbf which is used to represent the force of an operator falling forward and pushing down on the skin or the device. This value was determined by placing a scale on a table and recording the maximum dial reading caused by a team member falling forward and pushing down on the scale. The weight of the skin used in the analysis is 75 lbs which is the value that the client has indicated.

All FEA analysis was performed using Solidworks 2017 student edition and the specification information about the software used can be seen below.


Simulation Student Edition

Simulation Student Edition
Simulation 2017 x64 SP2.0 Date Code: 2017/033
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Seat ID/ Maintenance number : 9020 0067 6813 8919 8XPG BM28

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Do not base your design decisions solely on the results from Simulation design analysis software. Use this information in conjunction with experimental data and practical experience.
Field testing is mandatory to validate your final design. Simulation design analysis software helps you reduce your time-to-market by reducing but not eliminating field tests.

OK

Figure C-1: SolidWorks simulation information

1. Base Frame

Load applied at an offset to show bending

Offset loading was used to simulate the load going through the rotational block on the bearing surface and transferred to the bottom frame. The loading scenario was a vertical load applied on the skin in the working position. The load case used, considered the 75lb weight of the skin and an applied load of 125lb with a factor of safety of 2. Below in Figure C- 2 is the stress analysis using Solid works FEA set to display von Mises stresses. The material applied was 6061 – T6 aluminum and has a yield strength of 275Mpa. Also analyzed was the deflection due to the load case, which resulted in a max deflection of 1.796mm towards the middle of the frame.

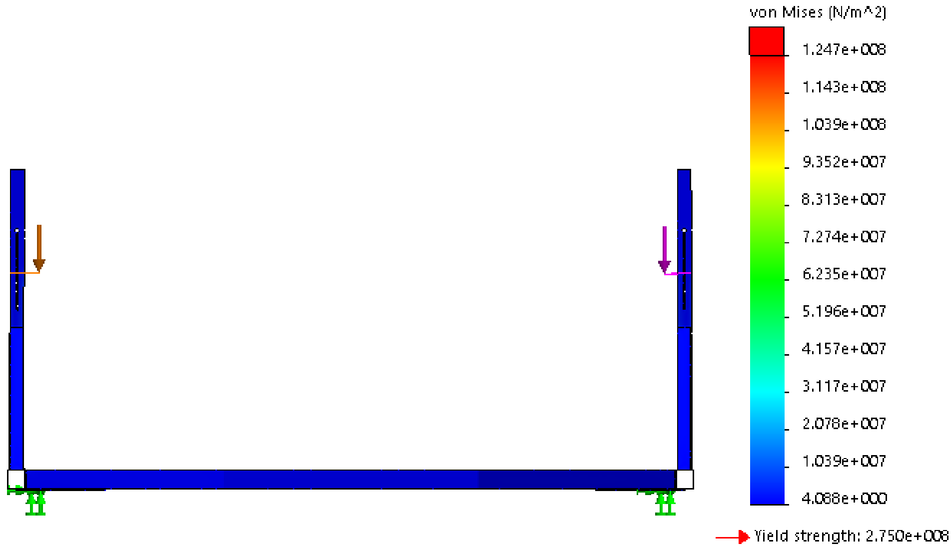


Figure C- 2: Stress Analysis using Solidworks FEA to display of the Von Misses stresses on the Base frame

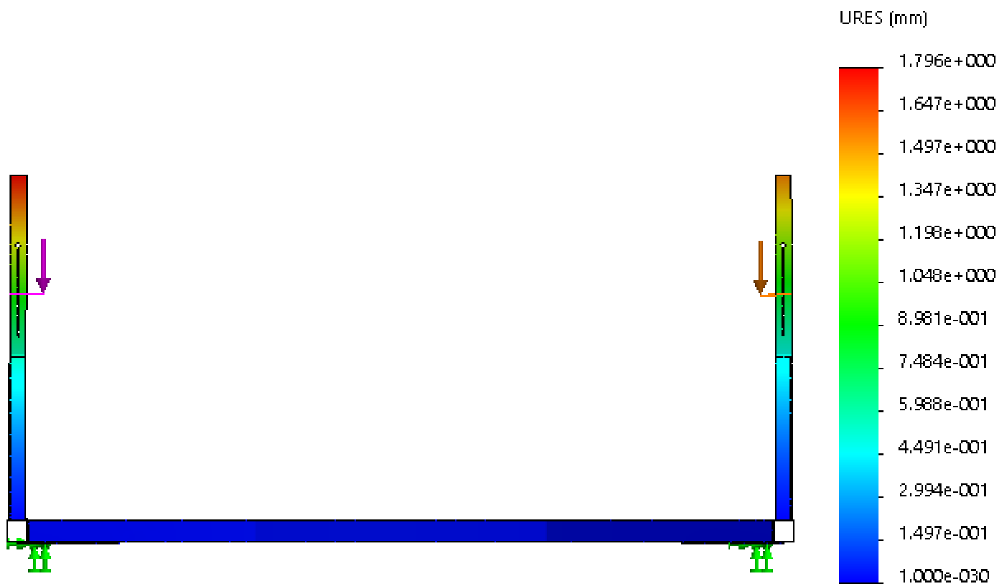


Figure C- 3: Stress analysis of the Base frame showing its maximum deflection

The analysis of offset load was done using an h-adaptive method for the mesh control. Running through 5 iterations the mesh converged with a 98% accuracy value. Below in TABLE C-I is the mesh details including the number of nodes, elements, and element size used.

TABLE C-I: TABLE SHOWING THE MESH DETAILS USED IN ANALYZING THE BASE FRAME

Mesh Details

Study name	offset top loading (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	1.41699 in
Tolerance	0.0708494 in
Mesh quality	High
Total nodes	186386
Total elements	99819
Maximum Aspect Ratio	66.662
Percentage of elements with Aspect Ratio < 3	24.9
Percentage of elements with Aspect Ratio > 10	37.3
% of distorted elements (Jacobian)	0

Height adjustment pin

To ensure that the material beneath the vertical adjustment pin did not fail under a max vertical load scenario a Solidworks FEA was performed. The loading scenario was a vertical load applied on the skin in the working position. The load case used, considered the 75lb weight of the skin and an applied load of 125lb with a factor of safety of 2. As it is shown in Figure C- 4 below, the aluminum 6061-T6 tubing is well within the yield strength.

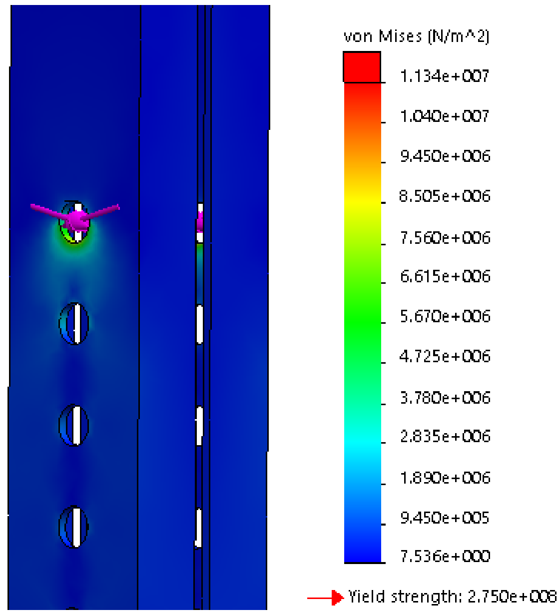


Figure C- 4: FEA performed to check if the Vertical Adjustment Pin would fail under vertical load

The analysis of the height pin location was done using an h-adaptive method for the mesh control. Running through 5 iterations the mesh converged with a 98% accuracy value. Below in TABLE F- II is the mesh details including the number of nodes, elements, and element size used.

TABLE C-II: TABLE SHOWING THE MESH DETAILS USED IN ANALYZING THE VERTICAL ADJUSTMENT PIN

Mesh Details

Study name	Static 1 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	1.18926 in
Tolerance	0.0594629 in
Mesh quality	High
Total nodes	350237
Total elements	199184
Maximum Aspect Ratio	77.198
Percentage of elements with Aspect Ratio < 3	27.3
Percentage of elements with Aspect Ratio > 10	22.5
% of distorted elements (Jacobian)	0

Corner impact force on bottom frame

To ensure that the device holding the skin would withstand a collision in the corner of a wall, a finite element analysis was used for a given scenario. The scenario u the mass of the device and skin, velocity of impact into wall and time of impact. The weight used was 250lb and included the untrimmed skin weight, as well the weight of the device. A velocity of 1m/s was chosen based on the cruising speed of a worker while moving this device. An impact time of 0.1s was chosen based on similar times used when measuring impacts of solid objects. Using these values, the equation for the force used was,

$$F = \frac{m\Delta v}{\Delta t} \qquad \text{Eq. 7}$$

Where m is the weight, Δv is the change in velocity, and Δt is the impulse time. This resulted in a force of 1311.2N and with a factor of safety of 1.5 the force was 1696.8N. The factored value was used in the Solidworks FEA. The stress results can be seen below in Figure C- 5.

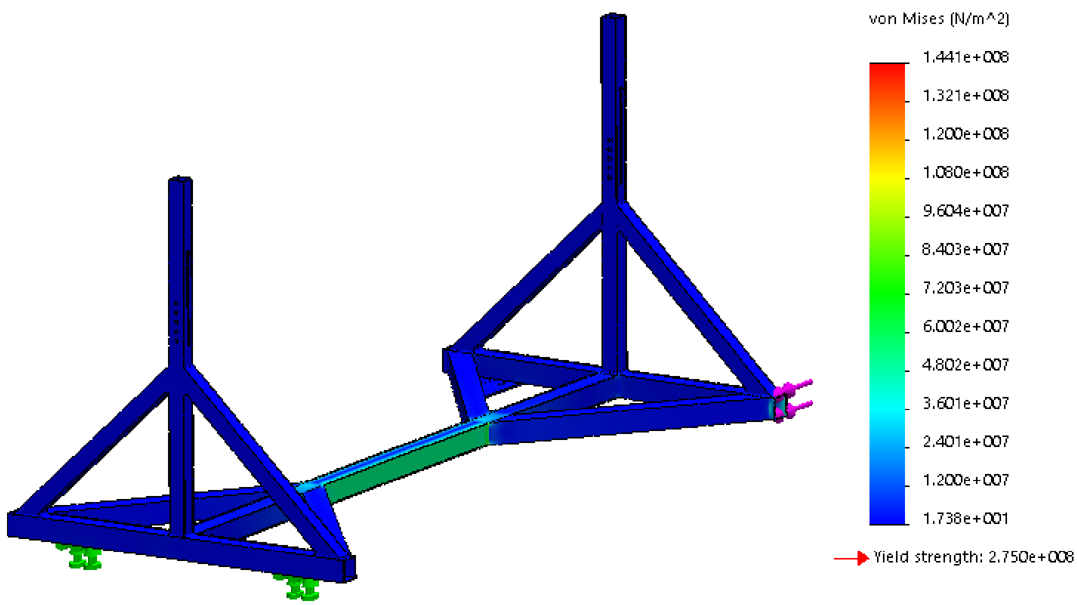


Figure C- 5: Stress Analysis to display the Von Misses stresses on the device holding the skin

Shown above in Figure C- 5 the stresses with the structure are below the yield strength of the 6061-T6 aluminum. The location of the majority stress, from an off centred load would be located in the single beam in the middle of the structure and has been accurately shown here.

The analysis of the corner wall collision used an h-adaptive method for the mesh control. Running through 5 iterations the mesh converged with a 98% accuracy value. Below in TABLE C-III is the mesh details including the number of nodes, elements, and element size used.

TABLE C-III: MESH PROPERTIES OF DEVICE HOLDING THE SKIN

Mesh Details

Study name	bottom corner load (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	1.23986 in
Tolerance	0.0619932 in
Mesh quality	High
Total nodes	300514
Total elements	173707
Maximum Aspect Ratio	151.1
Percentage of elements with Aspect Ratio < 3	25.6
Percentage of elements with Aspect Ratio > 10	25.7
% of distorted elements (Jacobian)	0

2. Rotational Block

Rotational block shear pin load

The main vertical pin of the rotational block assembly resists the rotational motion of the skin when unwarranted. When loads are applied at the furthest location from this pin, this results in the largest stresses the pin will see. To ensure that the pin will not fail under a given loading scenario, analytical calculations were performed. The furthest location from a pin would be when a force is applied at the pointed tip of the skin. A 2D sketch depicting this scenario can be seen in Figure C-6 below.

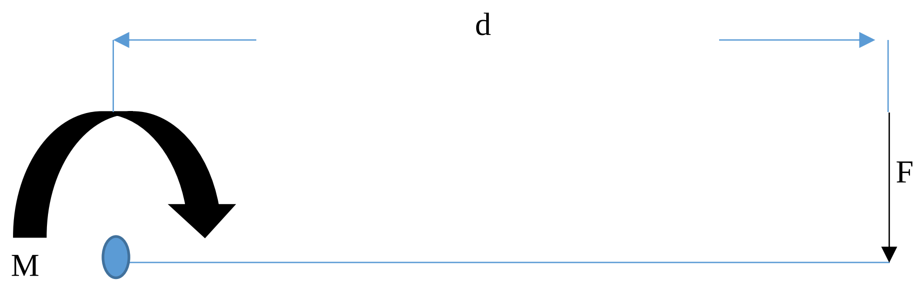


Figure C-6: 2D sketch showing the furthest location of the force applied at the tip of the skin

Where M is the moment, d is the distance and F is the applied force.

The equation created by these three variables is,

$$M = F * d \qquad \text{Eq. 8}$$

The moment was then converted into a force on the pin and this is depicted in the sketch Figure C- 7 below,

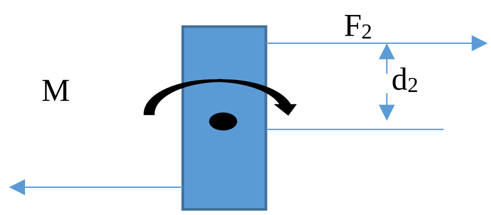


Figure C- 7: Sketch showing how the moment on the pin was converted into the force on the pin

Where F_2 is the force resulting from the distance d_2 and the moment M. Since there are two sides of the pin taking load from the moment the value of F_2 was cut in half.

This lead to determining the minimum diameter needed to support the loading scenario. The equation used can be seen below.

$$D = \sqrt{\frac{4F_2}{\tau\pi}} \qquad \text{Eq. 9}$$

Where D is the minimum diameter of the bolt and τ is the shear strength of the bolt material.

A bolt made from 316 Stainless steel, having a shear strength of 348 Mpa. This shear strength was determined by taking the ultimate tensile strength and multiplying by 60%. It was found during research that this is an assumption when the shear strength value is unknown.

With an applied load value 125lb at a distance d, of 1.146m and a factor of safety of 2, the resulting force, F was 1110N. This resulted in the moment M being 1571.76Nm. From this moment and a distance d_2 of 0.0254m, the resulting force F_2 was 61880.3N. Taking half of this force and applying it to the diameter equation, the minimum diameter required was 10.64 mm.

As manufactures don't have stock 10.64 mm pins, a 12.7mm pin was chosen instead.

Bottom rotational block under side pin loading

After analysis on the shear pin, the same scenario was used to analyze the bottom rotational block component. This component is machined from a block of 6061-T6 aluminum due to its need for welding to the sleeve. Applying the force 30940.15N or half of F_2 found above, on one side of the pin hole, the stresses were analyzed. As this component is more complicated than what analytical calculations can provide, Solidworks FEA was used as the analyzing tool. Below in Figure C-8 is the stresses that occur in the block under a static load of 30940.15N.

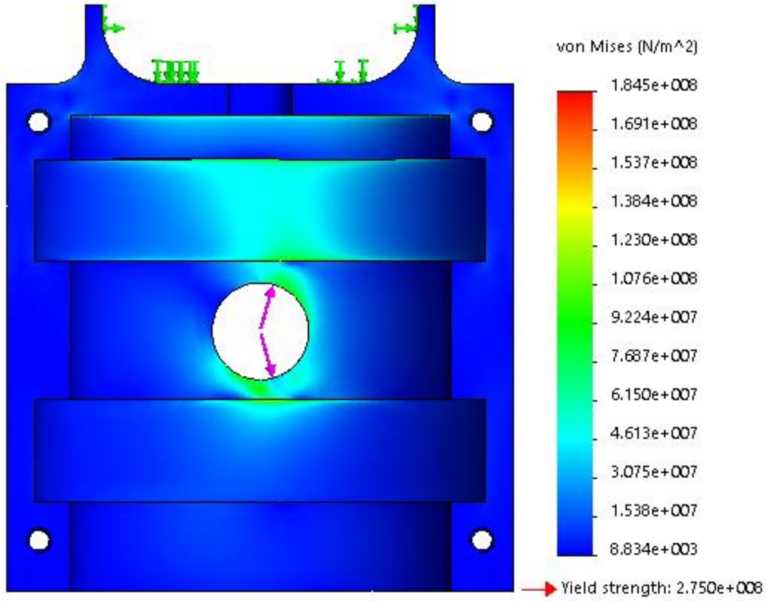


Figure C-8: Figure showing the Stresses that occur in the block when under a Static load

As seen by the chart on the right side of Figure C-8 above, the block is under the yield strength of the 6061-T6 aluminum.

The analysis of the bottom rotational block subjected to shear pin loading used an h-adaptive method for the mesh control. Running through 5 iterations the mesh converged with a 98% accuracy value. Below in TABLE C-IV is the mesh details including the number of nodes, elements, and element size used.

TABLE C-IV: MESH PROPERTIES USED TO ANALYZE THE BOTTOM ROTATIONAL BLOCK

Mesh Details

Study name	Static 2 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	0.162516 in
Tolerance	0.00812579 in
Mesh quality	High
Total nodes	175460
Total elements	116117
Maximum Aspect Ratio	14.497
Percentage of elements with Aspect Ratio < 3	99.3
Percentage of elements with Aspect Ratio > 10	0.0258

3. Upper Frame

Pin used to hold adjustable arms

The pins used to hold all four sliding arms are of the same material, size and length. This was determined by the scenario where the skin was in the vertical position and all the weight applied to one pin. A 2D sketch below can be used to depict the loading scenario on one half of a pin.

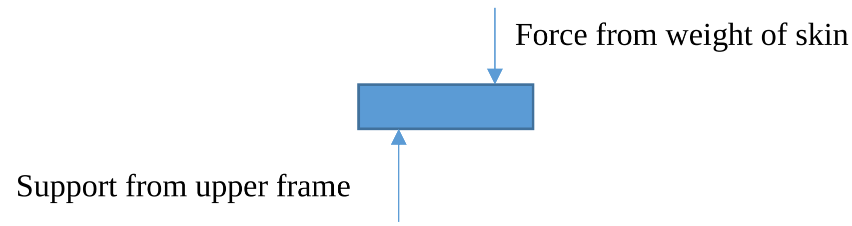


Figure C- 9: sketch showing the forces on the pin used to hold the adjustable arms

The force due to the weight of the skin was taken as 1400N with a factor of safety of 4. This factor was chosen because if the pins were to fail there would be severe damage done to the skin. Using the following equation, the minimum diameter of the bolt was determined.

$$D = \sqrt{\frac{4F}{\tau\pi}} \qquad \text{Eq. 10}$$

Where F is the force, and τ is the shear strength of the pin material.

The material chosen for the pins was 304 Stainless Steel as it was the material choice of the pin manufacturing company. An ultimate tensile strength of 505 MPa was used and translated into a shear strength value of 303 MPa. From the material values and the applied load of 700N, a minimum diameter of 2.43mm. As the pin manufacture does not make this size, a 6.35mm or 0.25-inch pin was chosen.

Adjustable arm under load

With the angled adjustable arm being the longest member and thus exposed to the largest bending stress when a load is applied at the end, this member was analyzed using FEA. The material used for the adjustable arms was AISI 1020 Steel due to its weld-ability and higher than aluminum strength. Applying a load of 500N at the end of the member resulted in a max von

Mises stress of 247.8 MPa and a max deflection of 8.114 mm. The results for the stress analysis can be seen in Figure C- 10: Stress Analysis of the Adjustable Arm.

below, and the deflection analysis in Figure C- 11: Deflection Analysis of the Adjustable Arm

below.

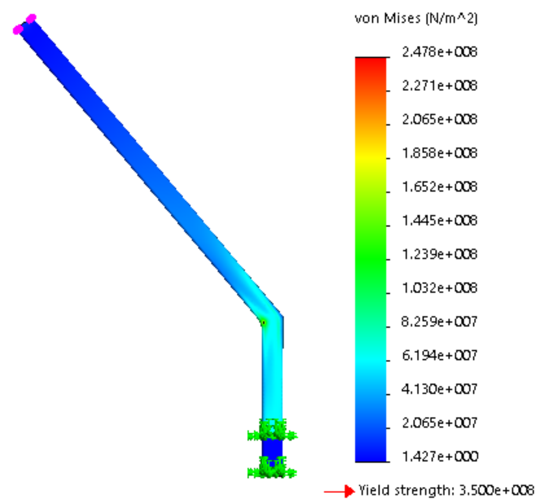


Figure C- 10: Stress Analysis of the Adjustable Arm.

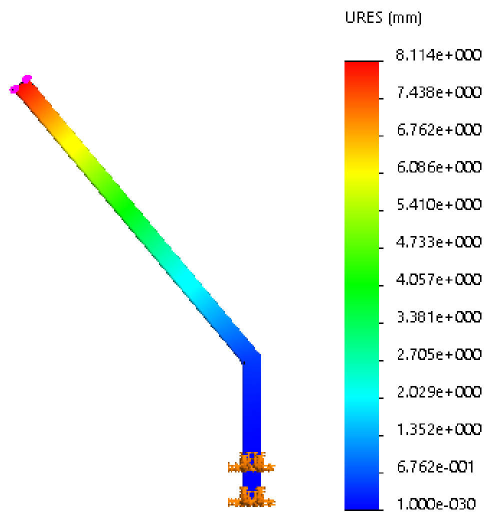


Figure C- 11: Deflection Analysis of the Adjustable Arm

The analysis of the bottom rotational block subjected to shear pin loading used an h-adaptive method for the mesh control. Running through 5 iterations the mesh converged with a 98% accuracy value. Below in TABLE C-V is the mesh details including the number of nodes, elements, and element size used.

TABLE C-V: MESH PROPERTIES FOR THE ADJUSTABLE ARM ANALYSIS

Mesh Details

Study name	Static 1 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	0.41472 in
Tolerance	0.020736 in
Mesh quality	High
Total nodes	114759
Total elements	63562
Maximum Aspect Ratio	9.7687
Percentage of elements with Aspect Ratio < 3	44.9
Percentage of elements with Aspect Ratio > 10	0
% of distorted elements (Jacobian)	0

4. Clamping mechanism

The clamps were designed to satisfy the following requirements:

1. The skin will not be able to slide out of the clamps
2. All components of the clamp will not fail during any possible loading conditions
3. The size is minimized so that the clamp is not in the way when the operator is working

The nitrile-butadiene rubber applied on the leveling foot and on the bottom of the clamp is in place to ensure the skin does not glide when subjected to horizontal forces. The rubber material has a dynamic coefficient of friction of 1.02 and a static coefficient of friction of 1.1 [31]. Given the material properties of the rubber, a normal force can be calculated to ensure that the part does not slide out of the clamps. It is desirable that the skin does not slide out of the clamps when subjected to 125 lbf normal to the clamping vector. The minimum normal force required is calculated using the following equation:

$$fn\_\min = \frac{125}{\mu_s * 2} = \frac{125}{2 * 1.1} = 56.8\ lbf \qquad \text{Eq. 11}$$

Where μ_s is the static coefficient of friction between the rubber and the skin. It is assumed that the friction coefficient of rubber is identical to the coefficient between rubber and the skin BMI material. A factor of two was applied in the denominator of Eq. 11**Error! Reference source not found.** as it is assumed that both rubber interfaces contacting the top and bottom of the skin have the same static friction coefficient. The resulting minimal normal force required is 56.8 lbf.

It is ideal for the operator to be able to torque the threaded platform by hand (without the use of additional tools) in order to achieve the required normal force. While the rubber is compressed against the skin, there will be a frictional force resisting the rotation. The torque that would need to be applied to the handle to overcome the frictional force is shown in the following equation [32]:

$$T = \frac{2}{3} * \mu_k * fn * r \qquad \text{Eq. 12}$$

Where T is the applied torque, μ_k is the kinematic friction coefficient, *fn* is the normal force and *r* is the radius of the leveling foot. The larger the radius results in a larger torque that would need to be applied in order to achieve the required clamping force. There is test data available for the torque capacity of the human hand using various handle shapes and sizes [33]. The results indicate that men can output 3.6 Nm to 6.8 Nm on using a triangular handle with a side length ranging from 25.4mm to 63.5mm. The result for females using the same range of handle sizes was 2.1 Nm to 3.7 Nm. The naming convention for the handles used in the dataset is as follows:

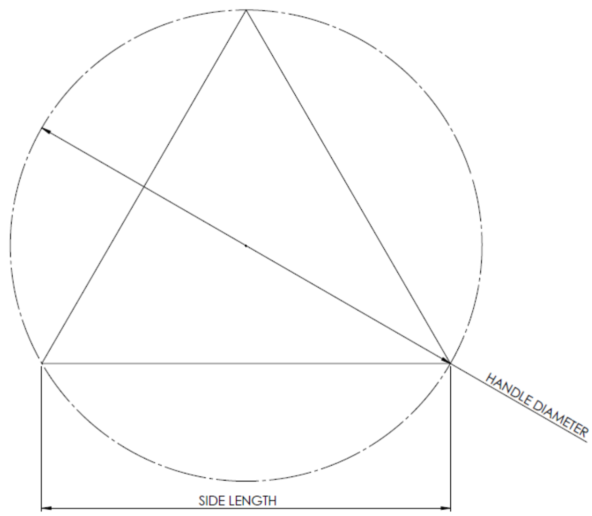


Figure C- 12: Handle dimension convention [33].

The required torque needs to be within the capacity of all employees, thus less than 3.7 Nm. Using Eq. 12, a torque of 3.7 Nm requires the clamping platform to be smaller than 43.1 mm. It is desirable to maximize the size of the platform to reduce the compression stress exerted on the skin. However, a larger diameter reduces the working surface that is exposed. Given the importance to protect the skin, the diameter was maximized, thus a diameter of 40 mm was selected. Using Eq. 12, a diameter of 40 mm requires a torque of 3.44 Nm to achieve the required normal force. Using the available test data, the average female can output a moment of 3.5 Nm using a handle with a side length of 44.5mm (51.4 mm diameter). Elesa+Ganter also sells knobs for the leveling feet, however, these knobs are sold in packs of 25 and the price for all the handles would total $395 CAD [5]. Turning knobs from McMaster are sold individually, reducing the total cost to $56 [34] & [35]. The largest available triangular handles provided my McMaster are 55mm diameter, which are selected, as they are larger than the 51.4 mm diameter requirement.

For ease of assembly, the clamps will be welded to the steel frame. As a result, the machined components will need to be also made of steel. Some of the dimensions on the parts were selected to satisfy geometric requirements as indicated in the figure below.

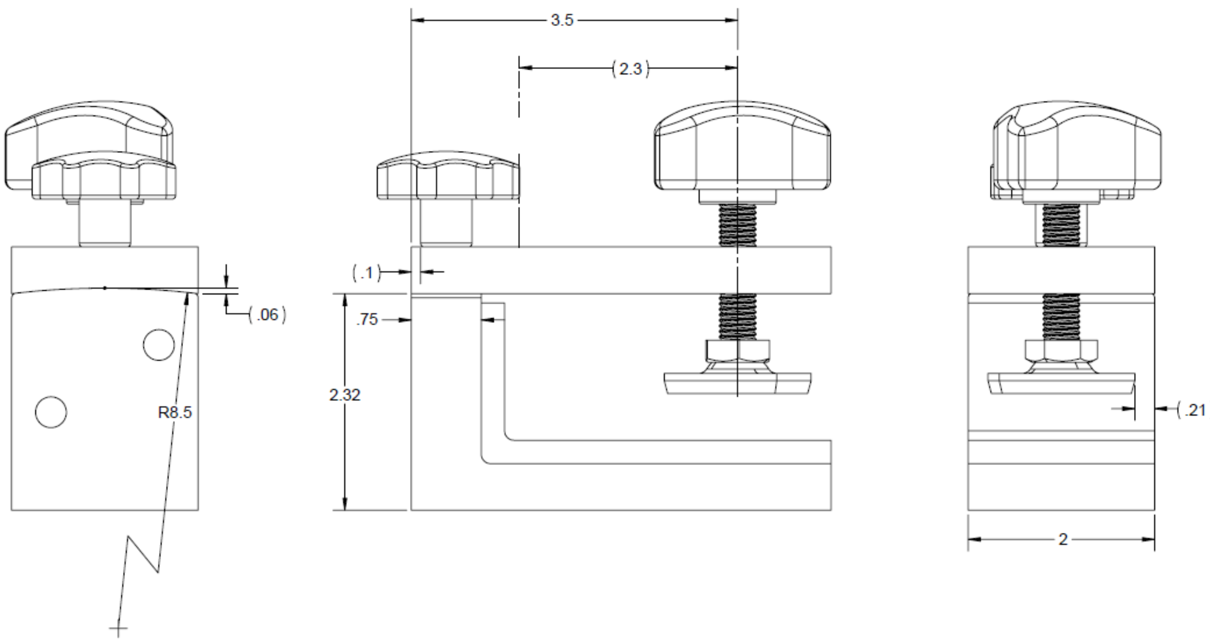


Figure C-13: Clamp selected dimensions to satisfy geometric requirements

The figure above indicates that a 2 inch clamp width ensures that the clamping platform is within the envelope of the machined part (within 0.21 inches). The clamping bolt is located at a distance of 3.5 inches from the edge of the clamp to provide a 2.3 inch gap from the other knob. The gap ensures that the operators hand will not contact the other knob.

The 0.75 inches thick vertical member was used to ensure that the locking knob does not hang over the edge (with a 0.1 inch clearance). The 2.32 inch height for the bottom clamp was selected to allow for a one inch vertical translation for the clamping platform. The height also ensures that the handle on the leveling foot does not contact the top bar if the clamp is lowered to the 3/16 inches section of the skin. An 8.5 inch radius of curvature was applied at the interface between the two machined parts so that the difference between the high point and low point (0.06inch) matched the 1.5mm thread pitch. As a result the top bar can be free to rotate with a single turn of the knob.

A stress analysis was performed to determine the thickness of the top and bottom of the clamp. The thickness of the top and bottom members need to be selected to ensure that the material does not yield or have excessive deflections when loaded. The maximum loading that would be experienced by the clamps is shown in the following image.

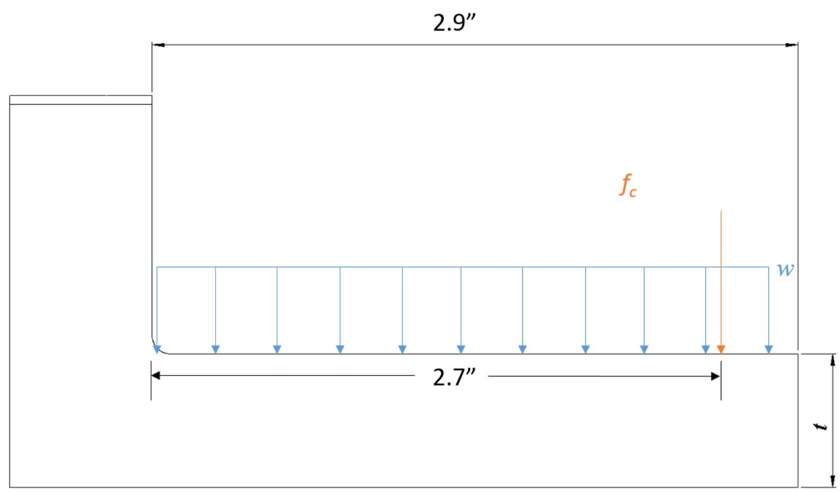


Figure C- 14: Figure showing the maximum load experienced by the clamps

Where *w* is a distributed load applied to the bottom of the clamp. The load is a combination of the weight of the skin and a load applied by the operator. The weight of the skin is approximately 75 lbs as indicated by the client. It is assumed that a clamp in a corner will experience ¼ the weight of the skin. A value of 125 lbs was used to represent an operator applying a vertical load on the skin near the clamp. It is assumed that the applied load is equally distributed along the length of the clamp. As a result, w is set to 125 + 75/4 = 162.5 lbf. f_c is the normal clamping force that is applied from the clamping platform. The actual load would be distributed across the clamp as the stress moves through the skin, however it is treated as a point load in the analysis to be conservative. The maximum value of f_c analyzed uses the maximum torque capacity (6.8 Nm) for humans on a handle with the same dimensions as used in the design [33] . Inserting a moment of 6.8 Nm into Eq. 12 results in a clamping force of 500N (112.4 lbf).

The bottom length of the bar would act as a cantilever beam with maximum stress and deflection. The maximum stress and deflection can be calculated by evaluating the cantilever beam equations for a point load as well as a distributed load [36]. The material properties used in calculation for AISI 1020 Carbon Steel can be found in Appendix H. The resulting maximum stress and deflections can be calculated using the following equations:

$$d_{max} = \frac{f_c * h^3}{3 * E * I} + \frac{w * h^4}{8 * E * I} = \frac{112.4 * 2.7^3 * 12}{3 * 29.7E^6 * 2 * t^3} + \frac{162.5 * 2.9^4 * 12}{8 * 29.7E^6 * 2 * t^3} \qquad \text{Eq. 13}$$

$$\sigma_{max} = \frac{M_{max} * \frac{t}{2}}{I} = \frac{M_{max} * 6}{2 * t^2} \qquad \text{Eq. 14}$$

Where the maximum moment is calculated using the following equation:

$$M_{max} = f_c * 2.7 + w * \frac{2.9}{2} = 539.1\ lb \bullet in \qquad \text{Eq. 15}$$

It is desirable to have ridged clamps; thus the thickness is designed to ensure that the deflections do not exceed 0.05in. Also, the maximum stress in the clamps is targeted to have a yield stress factor of safety of 3.

Solving Eq. 13 for a deflection of 0.05 results in a thickness of 0.21 inches.

Solving Eq. 14 for a stress of 50800*3 results in a thickness of 0.18 inches

As a result, the thickness must be set to 0.21inches in order to satisfy the deflection requirements. There will be a stress concentration at the corner fillet that is not accounted for in the calculations. As a result, the thickness to satisfy the stress requirement is multiplied by a factor of 3 to account for the concentration. The resulting thickness is 0.18*3= 0.54 inches. The result is rounded to a ½ inch thick bar as it is a standard size. The actual stress concentration at the fillet is further analyzed using an FEA software. The clamping model using the dimensions provided in the drawing # 4-110 was analyzed using FEA software. The resulting stress and deflection plots are shown in the figures below. The back surface that contacts the upper frame was set to fixed geometry. A distributed load of 162.5 lbf was applied across the top surface as shown as the purple arrows in the figure below. A distributed load of 112.4 lbf was applied within the sketched 40 mm diameter circle to represent the force transmitted from the clamping platform.

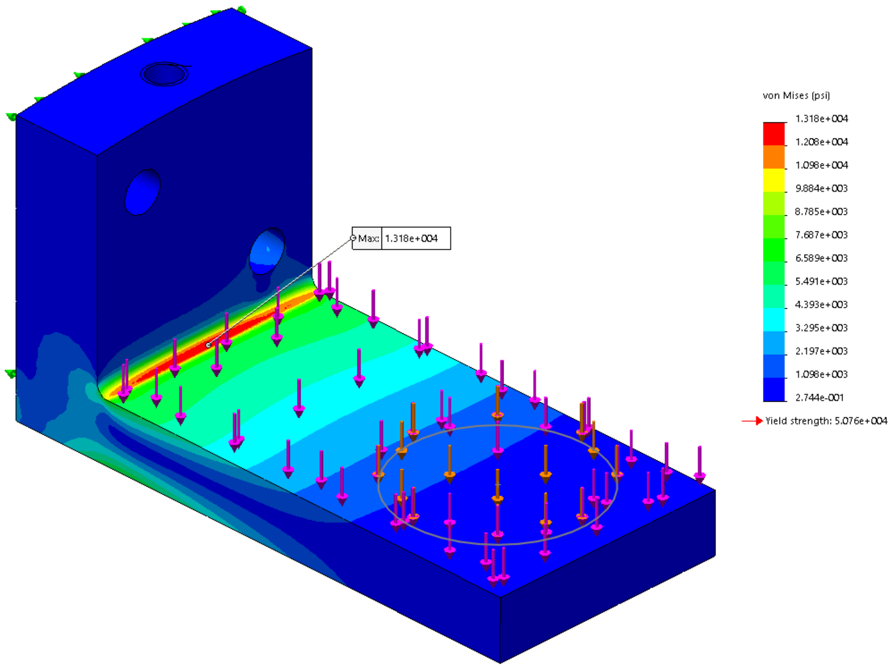


Figure C-15: Stress FEA plot on the clamp

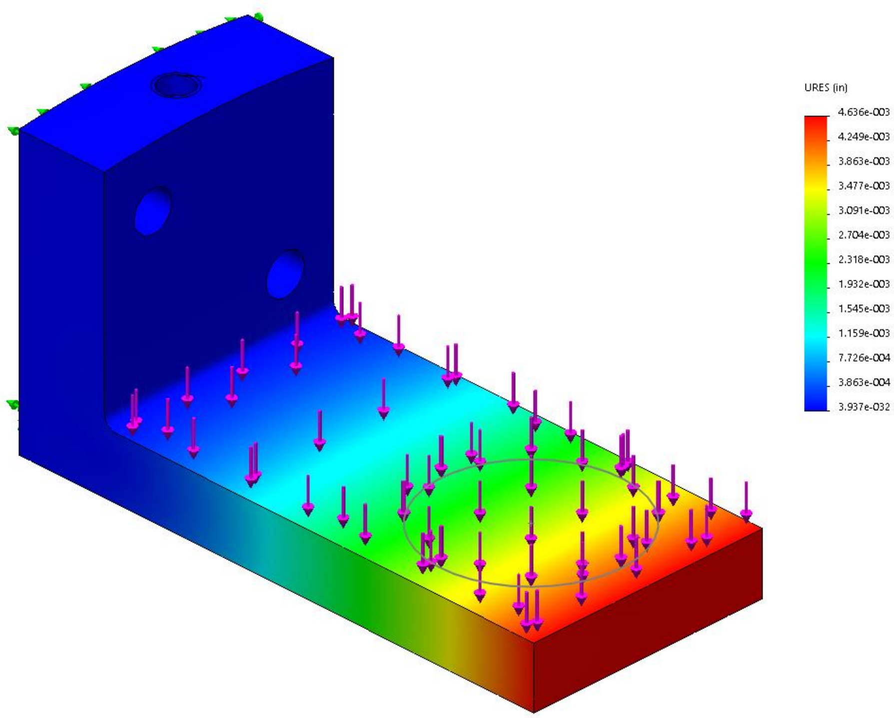


Figure C-16: Deflection FEA plot on the clamp

The result converged to within an error of 0.94% as shown in the convergence plot below.

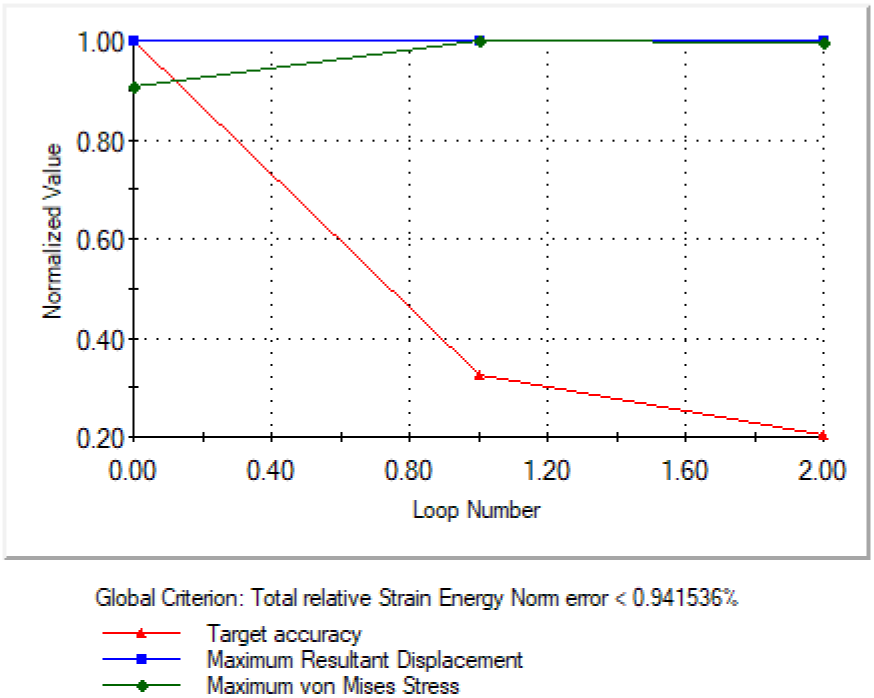


Figure C-17: Convergence plot for FEA on clamps

Figure C-17 above indicates that the highest stress concentration occurs at the 0.1 inch fillet. The resulting stress is 13.2 ksi, which is 3.8 times smaller than the material yield stress. As a result the design has a factor of safety of 3.8. Figure F-15 indicates that the maximum deflection of the clamp is 0.0046 inches which is more than 10 times lower than the 0.05 inch requirement. As a result, a thickness of 0.5 inches satisfies both stress and deflection requirements. The same thickness is used for the top bar as it has similar length and will experience similar loading forces.

It is also critical to verify that the compressive force applied on the skin does not cause the skin to yield/fracture. Material properties used to analyze the skin are provided in

APPENDIX C. Using the same 112.4 lbf maximum clamping force, the stress exerted on the skin is calculated with the following equation:

$$Compressive\ stress = \frac{F}{A} = \frac{112.4 * 4}{\pi * 1.57^2} = 58.1\ psi$$

As a result the stress has a design factor of safety of 77.5. The maximum deformation is calculated with the following equation:

$$Deformation = \frac{\mathrm{\sigma}}{\mathrm{E}} * t = \frac{58.9}{9100000} * \frac{11}{32} = \ 2.22\ E - 06\ inches$$

The resulting deformation is negligible as it would not be noticeable to the human eye.

5. Locations of clamp mounting points

It is optimal to have the clamps located in each corner so long as the stresses and deflection are within allowable limits. This will ensure that the upper frame is out of the way of the operators as much as possible. Given the dimensions of clamps, the clamping platforms can be placed anywhere within a 2.5 inch offset inwards from the edge of the skin as shown in the figure below.

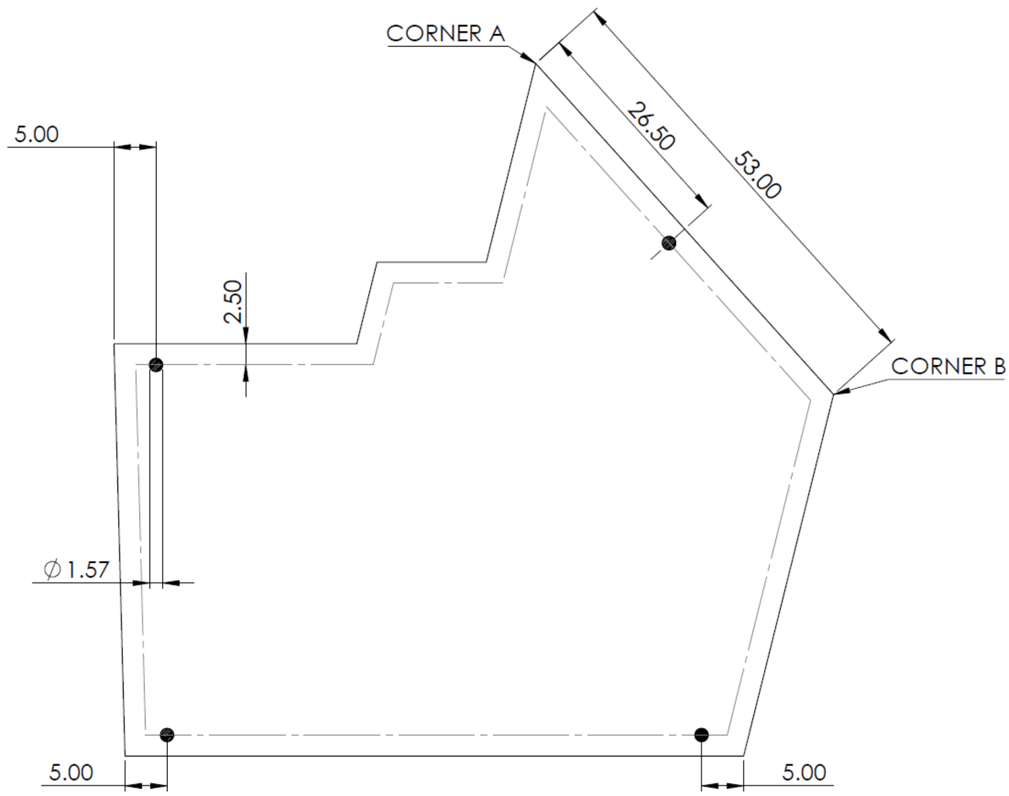


Figure C- 18: Schematic of trimmed Skin showing the location of mounting points

The black circles are ideal locations of the clamping platforms. It is optimal to have the clamp on the top right of the skin to be located at corner B, however it is preliminary placed at an equal distance between corner A and corner B to prevent major deflections when there is a load applied at corner A. The remaining clamps are placed 5 inches inwards from the closest corner to allow for extra space that the skin can be placed when the operators load the skin onto the device. An FEA was performed on the skin where the four clamping locations shown in Figure C- 18: Schematic of trimmed Skin showing the location of mounting points

were fixed. The fixture was completed assuming the fixed surface on the bottom of the skin was a projection of the 1.57 inch diameter fixture on the top of the skin. The fixture used in the model is shown in the figure below.

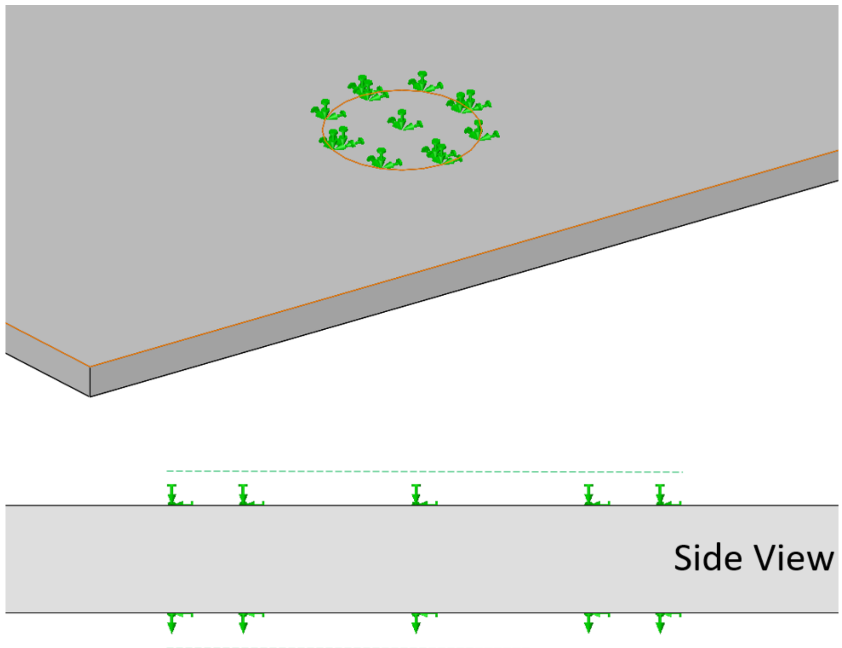


Figure C- 19: FEA analysis fixture for skin deflections

The analysis on the skin considered loadings on the untrimmed skin as the trimmed skin is smaller in size, thus will experience smaller stresses and deflections. The loading force is 125 lbs normal to the skin. There is also a 75 lbf gravitational force due to the weight of the skin. An h-adaptive mesh setting was used in Solidworks in order to refine the mesh to within a 98% accuracy. TABLE C-VI, Figure C- 20 and Figure C- 21 below shows the mesh details, convergence plot and deflection plot respectively for the FEA analysis when a load is applied at corner ‘A’.

TABLE C-VI: MESH DETAILS ‘ FOUR CLAMPS WITH A LOAD AT CORNER A’

Mesh Details	
Study name	Static 2 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	0.75 in
Tolerance	0.0375 in
Mesh quality	High
Total nodes	88365
Total elements	49150
Maximum Aspect Ratio	7.3137
Percentage of elements with Aspect Ratio < 3	93.8
Percentage of elements with Aspect Ratio > 10	0
% of distorted elements (Jacobian)	0

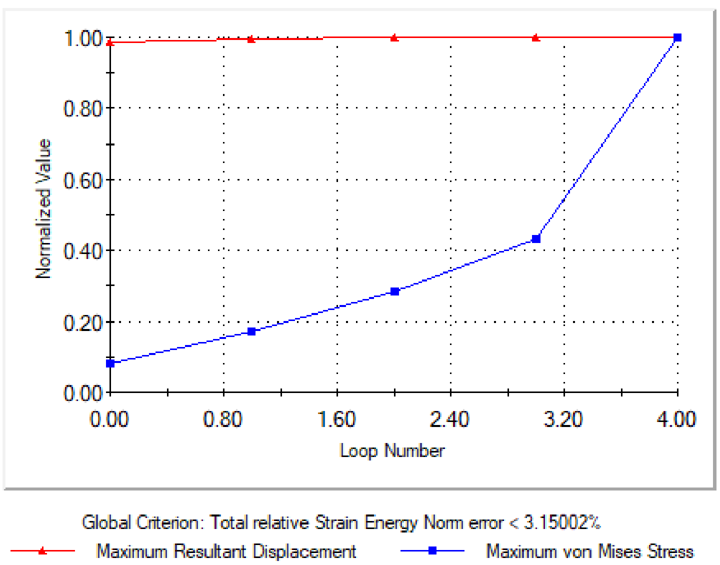


Figure C- 20: Convergence plot ‘Four clamps with a load at corner A’

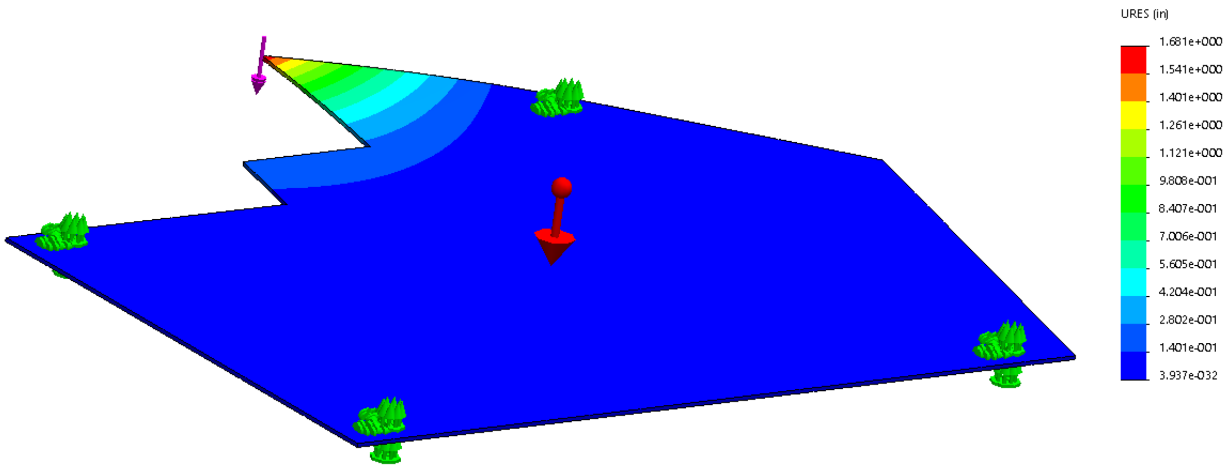


Figure C- 21: FEA deflection plot ‘Four clamps with a load at corner A’

Figure C- 21 indicates that the deflections converge to within 98 percent accuracy, however, the stresses trend to infinity as the mesh size decreases. Figure C- 21 can be used to determine the maximum deflection as the maximum deflections converge within 98% accuracy. The purple arrow in Figure C- 21 represents the force applied by the operator and the red arrow represents the distributed load due to the force of gravity on the skin. The resulting deflection at corner ‘A’ is 1.681 inches. The same method was applied to a vertical load at corner ‘B’. TABLE C-VII, Figure C-22 and Figure C-23 show the mesh details, convergence plot and the FEA deflection plot for a load at corner ‘B’.

TABLE C-VII: MESH DETAILS ‘ FOUR CLAMPS WITH A LOAD AT CORNER B’

Mesh Details

Study name	Static 2 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	0.75 in
Tolerance	0.0375 in
Mesh quality	High
Total nodes	115770
Total elements	65329
Maximum Aspect Ratio	7.3137
Percentage of elements with Aspect Ratio < 3	93.8
Percentage of elements with Aspect Ratio > 10	0
% of distorted elements (Jacobian)	0

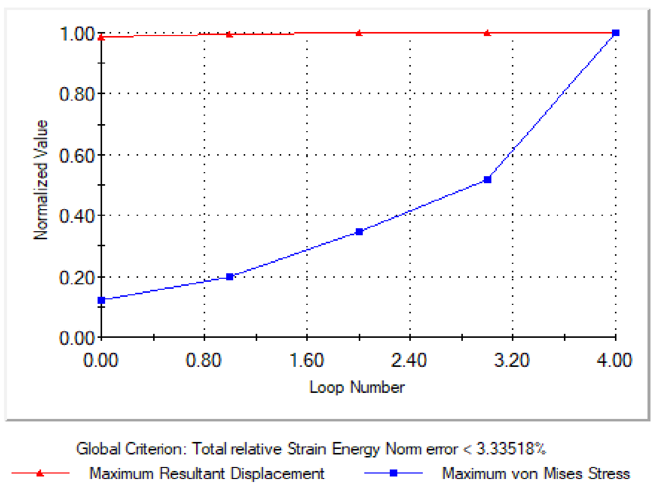


Figure C-22: Convergence plot ‘Four clamps with a load at corner B’

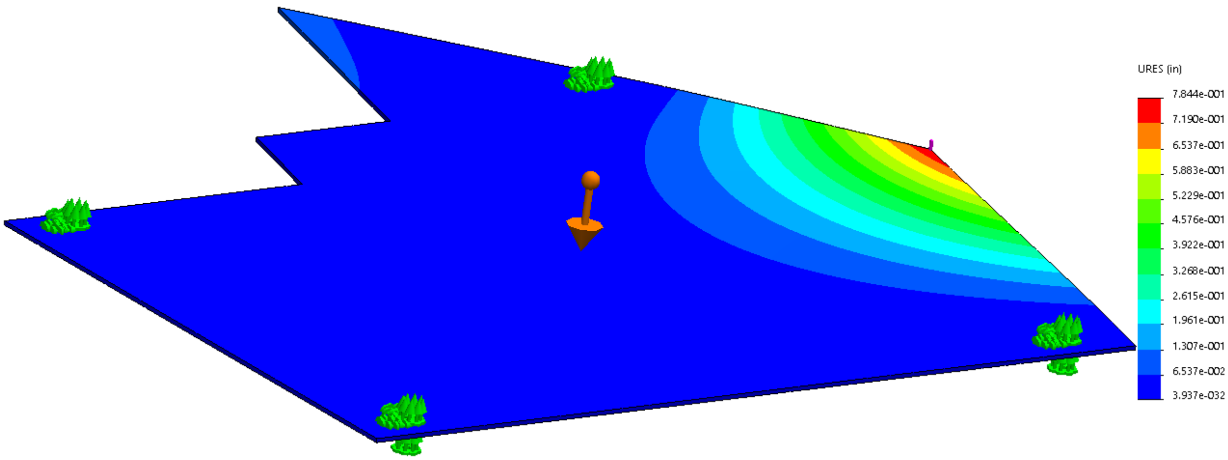


Figure C-23: FEA deflection plot ‘Four clamps with a load at corner B’

Figure C-23 above indicates that the maximum deflection converged but the maximum stress did not. The resulting maximum deflection as indicated in Figure C-23 is 7.84 inches. A summary for the deflections generated while using four clamps is provided in the table below.

TABLE C-VIII: SUMMARY OF RESULTING DEFLECTIONS ON THE SKIN WITH FOUR CLAMPS

Force location	Maximum deflection (inches)
Corner A	1.68
Corner B	7.84

The table above indicates that a force applied at either corner A or corner B will cause the skin to deflect more than 0.5 inches. As a result, the deflections cannot be reduced to within the allowable limits regardless where the clamp between corner A and corner B is placed. Thus two clamps must be used in the top corner of the skin to support both corner ‘A’ and corner B.

Two clamps will need to be used, one placed in each of the corners. For the clamp in the top corner, the location cannot be placed directly in the corner because then the clamp would not mate with the untrimmed surface when the adjustment arm is lowered. An FEA analysis was performed on the skin while supported by five clamps. The analysis was completed using three different applied load locations in order to determine the maximum possible deflection and stress. The resulting locations of the clamps and the locations of the forces in the FEA analysis are provided in Figure C- 24 below.

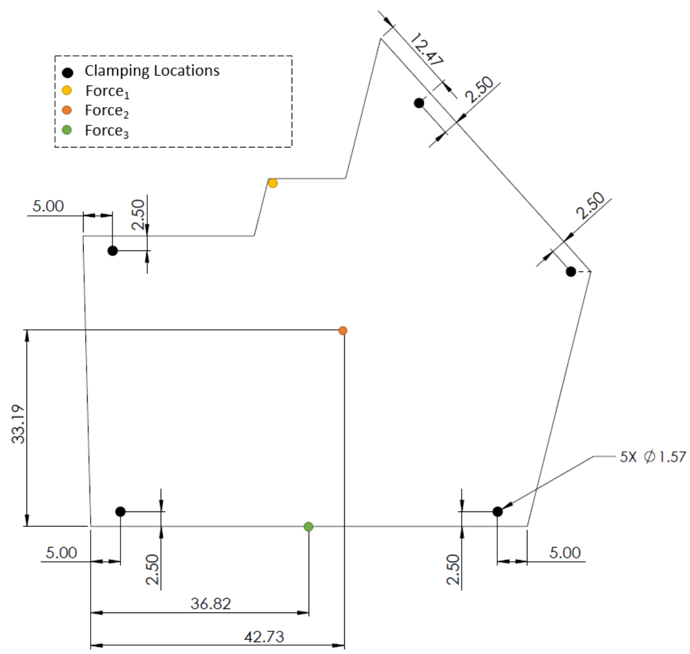


Figure C- 24: Locations of clamps and forces used in the FEA analysis

Figure C- 24 indicates that the highest placed clamp is at an offset of 12.47 inches from the nearest corner. The offset is in place to ensure that the clamp will make contact with the skin when the skin is trimmed. Force 1 in the figure above was selected because there are likely high stress concentrations at the two nearest corners of the skin. Force 3 is at an equal distance between the bottom two clamps and was selected for the analysis because it is the longest distance between two clamps on the edge of the skin. Force 2 is placed at the average X and Y locations relative to the remaining clamps. Thus the relative distance from the bottom edge of the skin for force 2 is calculated as follows:

$$X_{force2} = \frac{\sum_{n=1}^{5} X_i}{5} = 36.82\ inches$$

Where X_i is the location of each clamp relative to the bottom left edge of the skin. The same was applied for the Y dimension. A 125 lbf applied load with a 75 lbf skin weight is used for each loading scenario.

The following three sections indicate the FEA results using five clamps with the three separate loading scenarios. The applied force on the top surface of the skin is distributed about a 4 inch diameter circle in the model centered at the locations indicated in Figure C- 24. A fillet of 0.3 inches was applied to all eight sharp corners to avoid stresses diverging to infinity at the corners.

5.1. AIRCRAFT SKIN FEA 'FIVE CLAMPS SUBJECTED TO FORCE 1'

The mesh details, convergence plot, FEA deflection plot and FEA stress plot are shown in TABLE C-IX, Figure C-25, Figure C-26 and Figure C-27 respectively.

TABLE C-IX: MESH DETAILS 'FIVE CLAMPS SUBJECTED TO FORCE 1'

Mesh Details

Study name	Static 2 (-Default-)
Mesh type	Solid Mesh
Mesher Used	Standard mesh
Automatic Transition	Off
Include Mesh Auto Loops	Off
Jacobian points	4 points
Element size	0.75 in
Tolerance	0.0375 in
Mesh quality	High
Total nodes	148080
Total elements	84004
Maximum Aspect Ratio	10.73
Percentage of elements with Aspect Ratio < 3	93.7
Percentage of elements with Aspect Ratio > 10	0.00392
% of distorted elements (Jacobian)	0

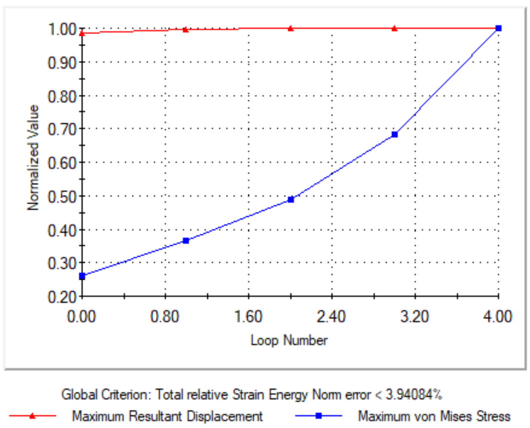


Figure C-25: Convergence plot 'Five clamps subjected to force 1'

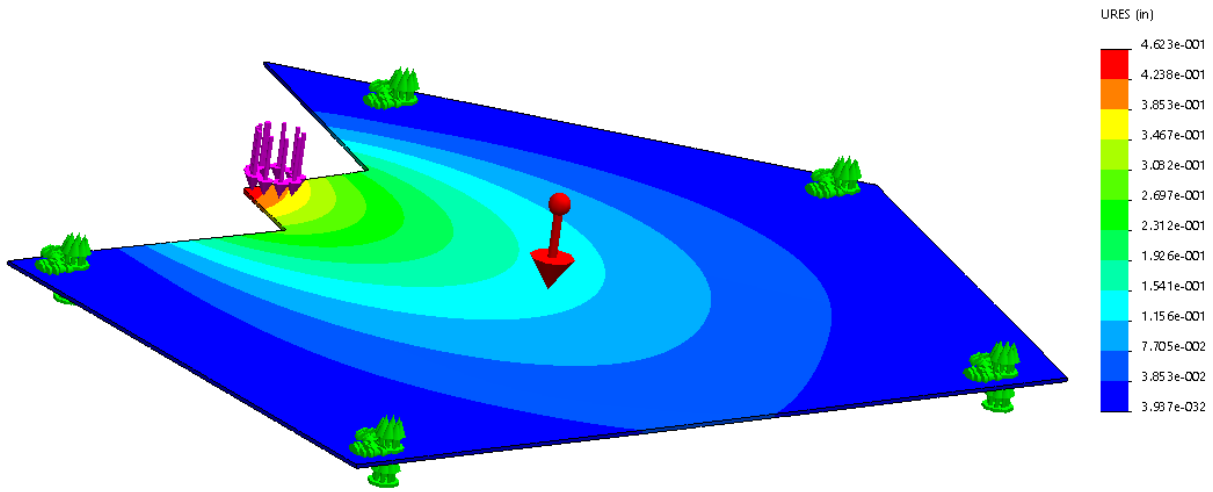


Figure C-26: FEA deflection plot ‘Five clamps subjected to force 1’

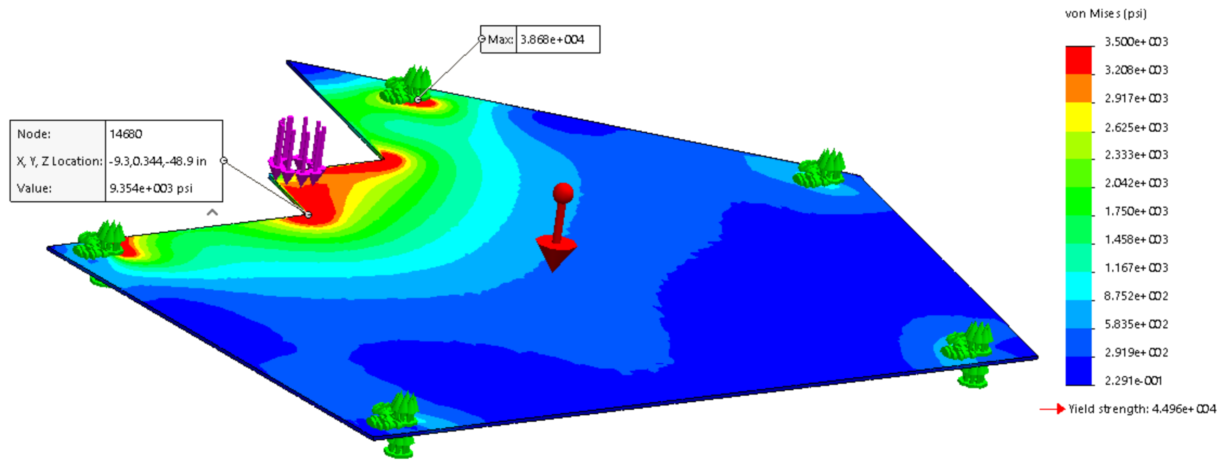


Figure C-27: FEA stress plot ‘Five clamps subjected to force 1’

Figure C-25 indicates that the maximum deflection converges within 98% accuracy; however the maximum stress does not. Figure C-26 indicates that the maximum deflection experienced is 0.46 inches. Figure C-27 indicates that the maximum stress trends to infinity at the location of the clamps, which is due the mesh created in the model. The scale on the plot limits the max stress fringe to 3.5 ksi for visual purposes as the diverging stress skews the plot. It is likely that the maximum stress will occur at the nearby fillet (Node 14680) as there is a change in geometry. The h-adaptive mesh will not prove convergence at any location other than at the maximum stress, thus an additional model needs to be used to prove convergence at Node 14680. To prove convergence, the model was constrained such that the edges of the sheet had a fixed geometry instead of the clamping locations. The new geometry removed the diverging stress result at the clamping location. The convergence plot and FEA stress plot with the new geometry are shown respectively in

Figure C- 28: Convergence plot 'adjusted fixed geometry subjected to force 1'

and Figure C- 29 below.

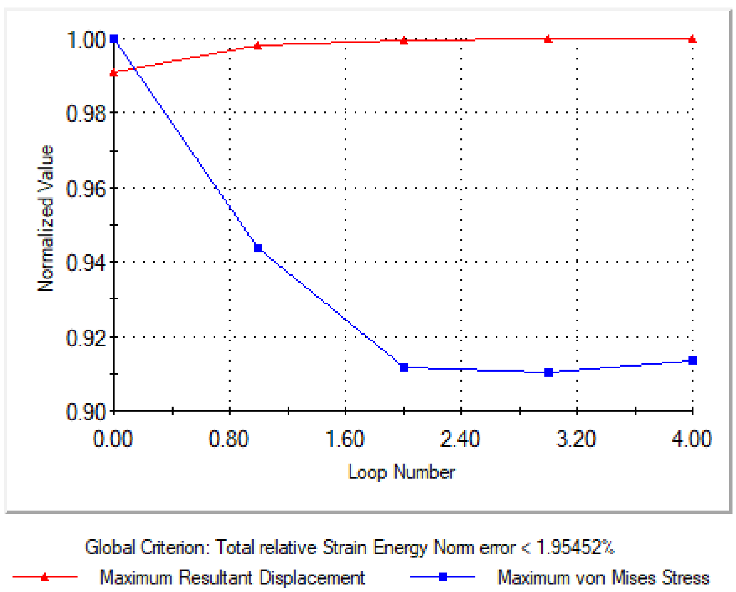


Figure C- 28: Convergence plot 'adjusted fixed geometry subjected to force 1'

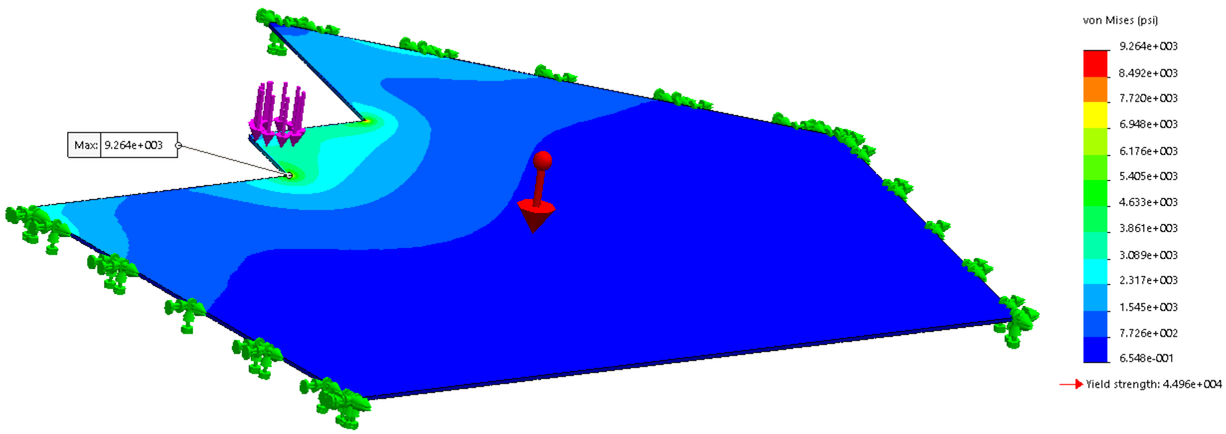


Figure C- 29: Converging stress FEA plot 'adjusted fixed geometry subjected to force 1'

Figure C- 28 indicates that the stresses now converge within 98% accuracy. The resulting maximum displacement (located at the same fillet) is 9.26 ksi which agrees with the 9.35 ksi result from Figure C- 29 to within 1%. As a result, the adjusted fixed geometry did not significantly effect the maximum stress result. An FEA analysis was performed with the same model as in but with the skin gravitational force removed. The resulting maximum deflection is 0.43 inches. The result indicates that the skin will have a total deflection of 0.46 inches, however, the operator will only feel a maximum deflection of 0.43 inches when applying a load to the skin. When the operator performs work on the skin, the maximum force they would apply is only 25 lbs which is 20% that of the force that would occur if an operator were to fall forward and push down on the skin. The resulting maximum deflection that the operator would feel is 0.2*0.43 = 0.086 inches.

Given the strength of the skin is 45 ksi (as determined in APPENDIX B), the resulting stress factor of safety is 45/9.35 = 4.9 ksi.

5.2. AIRCRAFT SKIN FEA 'FIVE CLAMPS SUBJECTED TO FORCE 2'

The same method of analysis was used for force 2. The resulting FEA displacement plot and FEA stress plot and FEA stress plot with the adjusted fixed geometry is shown in Figure C-30, Figure C- 31 and Figure C- 32.

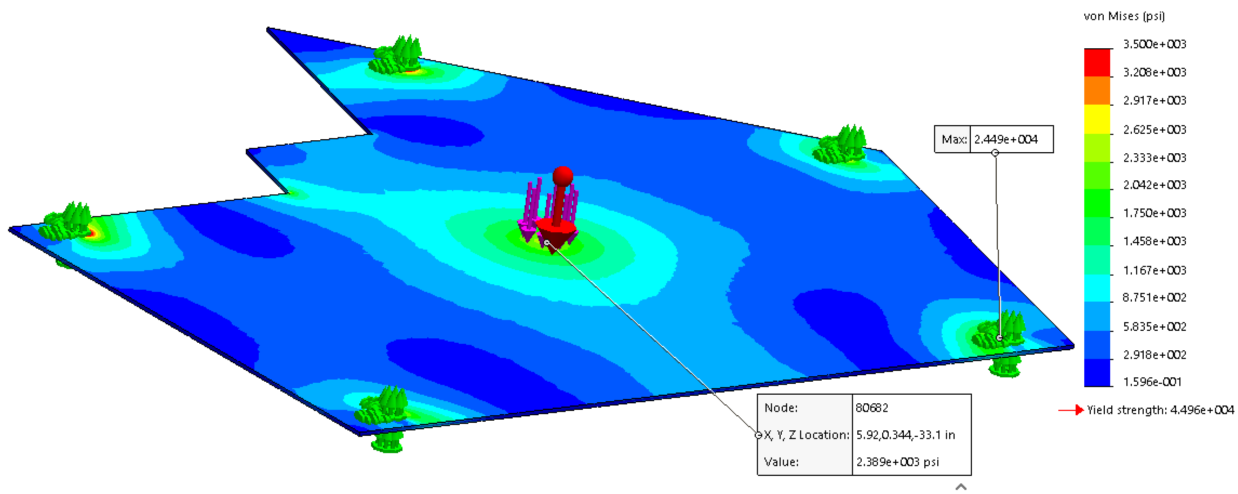


Figure C-30: FEA stress analysis 'five clamps subjected to force 2'

The maximum stress fringe was set to 3.5 psi to be able to visually show the locations of stress concentrations.

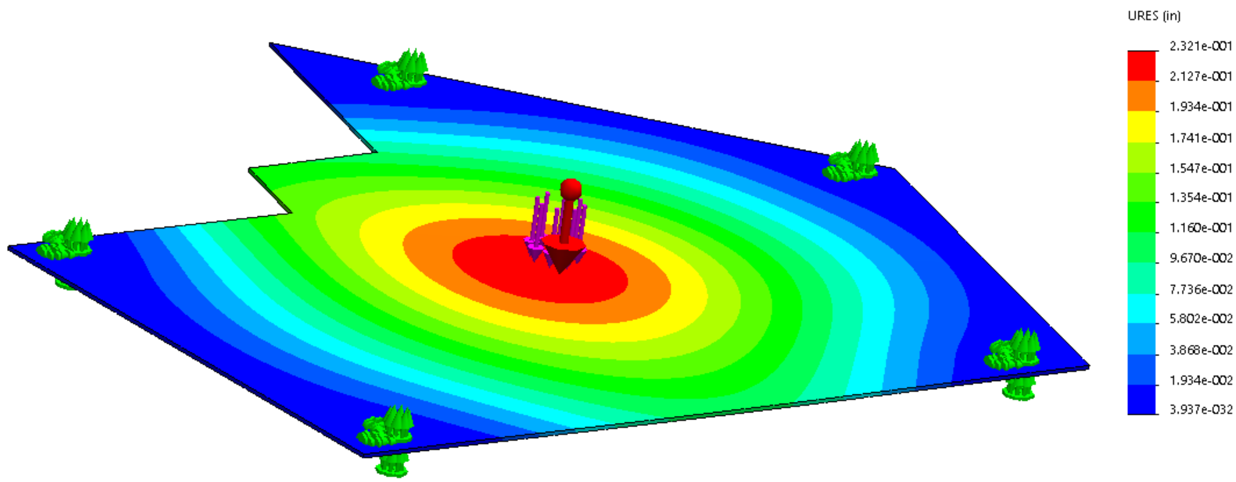


Figure C- 31: FEA deflection plot ‘five clamps subjected to force 2’

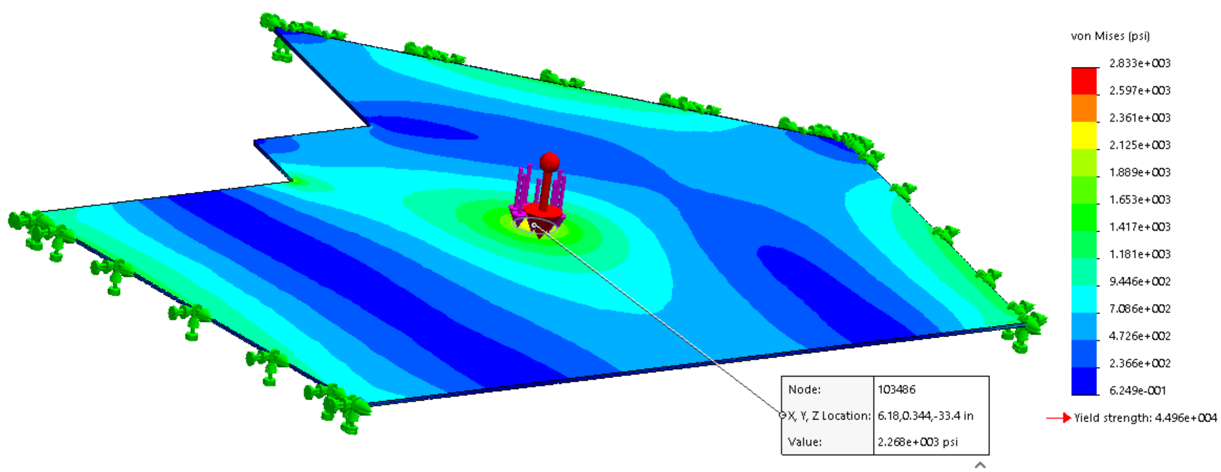


Figure C- 32: FEA stress plots ‘adjusted fixed geometry subjected to force 2’

The maximum deflection without gravity was 0.19 inches. The FEA results indicate that the maximum possible deflection is 0.23 inches. The deflection caused by working loads is 0.19*0.2 = 0.038 inches. The maximum stress is 2.39 ksi, which has a factor of safety of 19.8 when compared to the strength of the skin.

5.3. AIRCRAFT SKIN FEA 'FIVE CLAMPS SUBJECTED TO FORCE 3'

The same method of analysis was used for force 3. The resulting FEA displacement plot and FEA stress plot and FEA stress plot with the adjusted fixed geometry is shown in Figure C-33, Figure C-34 and Figure C-35 below.

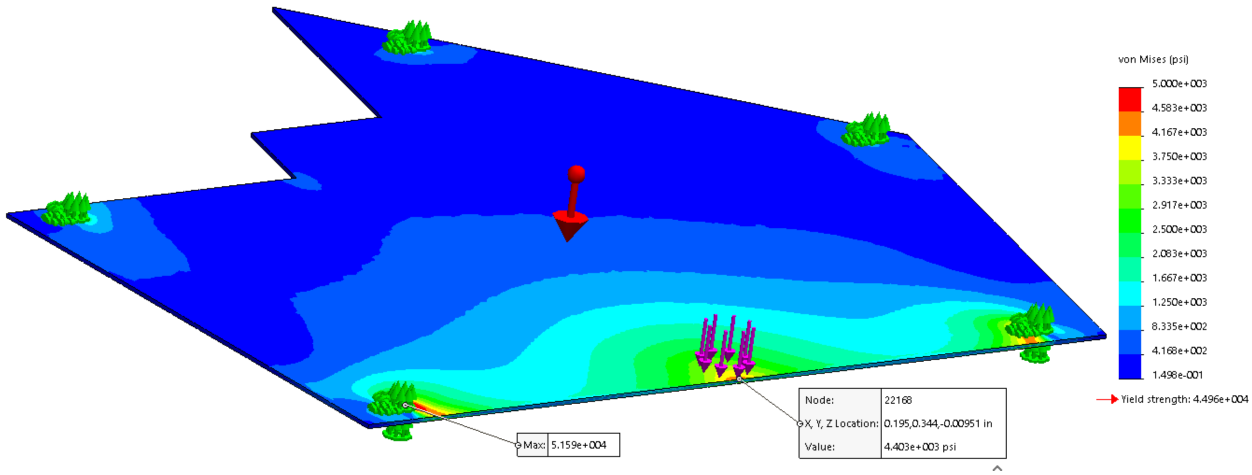


Figure C-33: FEA stress analysis 'five clamps subjected to force 3'

The maximum stress fringe was set to 5 psi to be able to visually show the locations of stress concentrations.

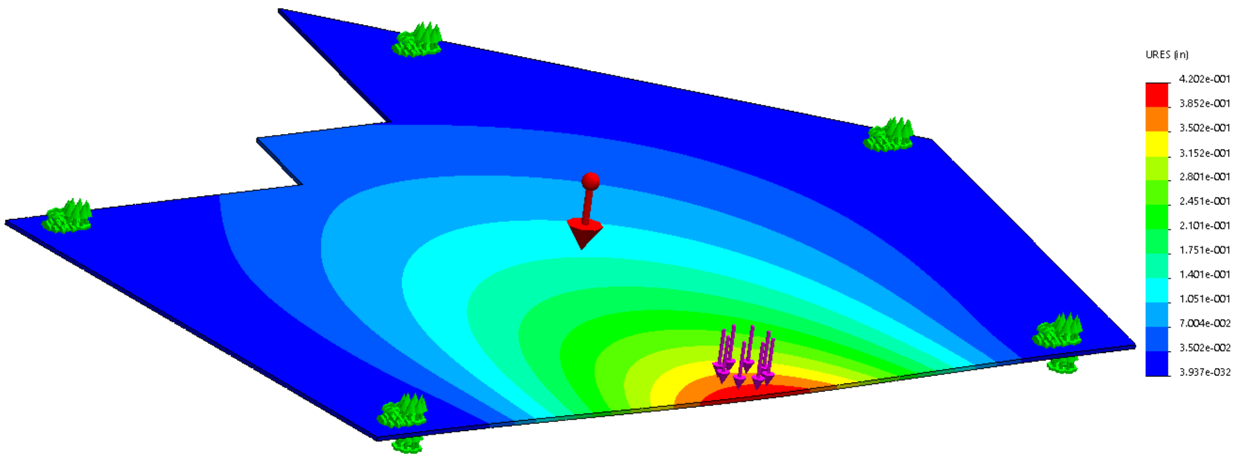


Figure C-34: FEA deflection plot ‘five clamps subjected to force 3’

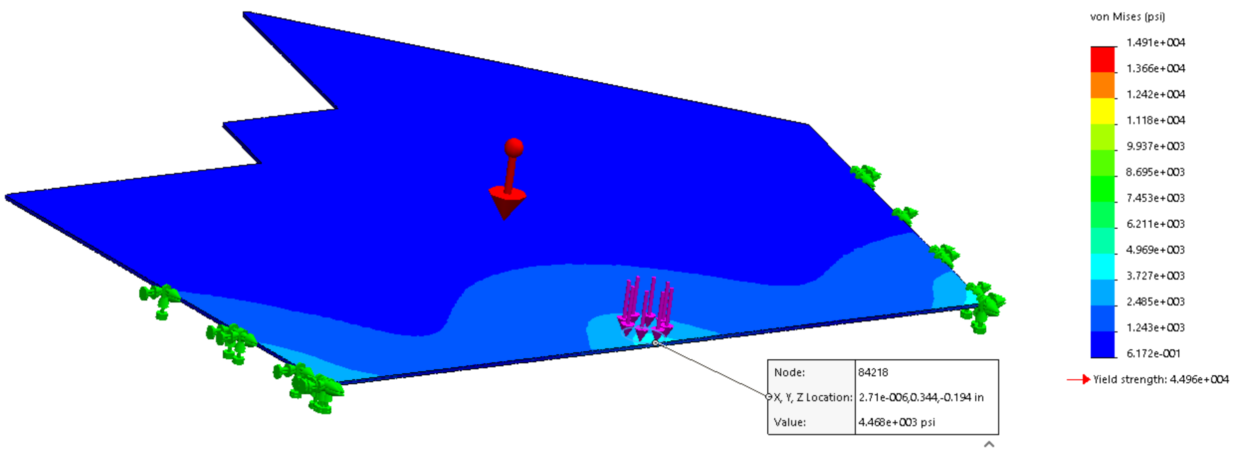


Figure C-35: FEA stress plots ‘adjusted fixed geometry subjected to force 3’

The maximum deflection without gravity was 0.39 inches. The FEA results indicate that the maximum possible deflection is 0.42 inches. The deflection caused by working loads is 0.39*0.2 = 0.078 inches. The maximum stress is 4.42 ksi, which has a factor of safety of 10.1 when compared to the strength of the skin.

6. Vertical Adjustment

The two springs had to support 75 pounds composite skin at all height position. Since the springs are in parallel, each spring had to support 37.5 pounds. Operators had expressed a height of 39 inches from the ground for the taller operator and 34 inches from the ground for the smaller operator; hence, the spring had to be able to stretch 5 inches from the ground. In addition, cost is not an issue as expressed by the client and the amount of force to move the spring and adjust the height had to be as low as possible. The spring is attached from the top and hence the operators need only to pull down the skin. This requires an extension spring where it is tightly coiled when un-disturbed and is only stretched when downward forces are applied to it. In doing so, if any slippage or incident occurred during height adjustment, the spring would want to move back up protecting the skin from hitting the ground or the handling device. Using spring manufacturer, Acxess Springs, and inputting all the requirement listed above from the extension springs, an extension spring that can withstand 100 pounds force is required. However, through further research into the Acxess Spring catalog, an extension spring that has a maximum suggested load of 89 pounds was discovered. The spring had the following properties

TABLE C-X: TABLE SHOWING THE SPRING PROPERTIES

Outside diameter (OD)	1.75 inches
Inside Diameter(ID)	1.396 inches
Wire diameter (d)	0.177 inches
Number of coils (Na)	66.266
Original Length (inside hook)	14 inches
Initial Tension	16 lbs.

The mean diameter, spring index and spring constant was calculated using the following equations

$$D = OD - d$$
$$D = 1.75 - 0.177$$
$$D = 1.573\ inches \qquad \text{Eq. 16}$$

$$c = \frac{D}{d}$$
$$c = \frac{1.573}{0.177}$$
$$c = 8.89 \qquad \text{Eq. 17}$$

$$k = \frac{Gd^4}{8 * D^3 * Na}$$
$$k = 5.47\ lnf/in \qquad \text{Eq. 18}$$

The Wahl factor is calculated using the following formula.

$$K_W = \frac{4c - 1}{4c - 4} + \left(\frac{0.615}{c}\right)$$
$$K_W = 1.16 \qquad \text{Eq. 19}$$

As expressed by the operators for the difference in height for different operators. The first point of 39 inches from the ground was achieved when the skin is loaded and the skin is deflected downwards due to weight of the spring. Using the weight of 37.5 inches (half of the weight of the skin), the first deflection point from top of the top plate is

$$FP = \left(\frac{37.5}{k}\right) + OL$$
$$FP = \left(\frac{37.5}{5.47}\right) + 14.354$$
$$FP = 21.21\ inches \qquad \text{Eq. 20}$$

Hence the overall height of the material handling device is

$$OH = 39 + 21.21$$
$$OH = 60.21\ inches \qquad \text{Eq. 21}$$

The last deflection point is

$$SP = 5 + FP$$
$$SP = 26.21\ inches \qquad \text{Eq. 22}$$

The maximum suggested deflection as stated by Acxess Spring is 13 inches hence the total suggested stretched length of the extension spring is

$$Suggested\ Deflection = 14.354 + 13$$
$$Suggested\ Deflection = 27.35\ inches \qquad \text{Eq. 23}$$

Therefore, the last deflection point is lower than the total suggested stretched length. The forces that the spring will temporarily be stressed to at the first and last position is 55.44 lbf and 82.79 lbf and the suggested maximum load is 89 lbf. The loads are calculated using the following equation

$$F_{firspoint} = k(FP - OL) + IT$$
$$F_{firstpoint} = 55.44 \text{ lbf}$$
$$F_{secondpoint} = k(SP - OL) + IT \qquad \text{Eq. 24}$$
$$F_{secondpoint} = 82.79\ lbf$$

Even though the forces are well below the suggested maximum load that can be applied onto the spring, it is important to calculate stresses and compare with the material of the extension spring. Stress due to forces occur in the coil of the spring and in the hook end of the extension spring. Force applied in the middle of the coil induces direct stress and torsion which is super imposed to the following equation

$$\tau = \frac{Tr}{J} + \frac{F}{A}$$
$$\tau = K_W\left(\frac{8FD}{\pi d^3}\right)$$
$$\tau = 1.16\,\frac{(8 * 89 * 1.573)}{\pi * 0.177^3}$$
$$\tau = 74{,}895.98\ psi$$

Eq. 25

Shear stress of 74,895.98 psi is the calculated for the lowest stretched position of the extension spring as it incurs the most forces. The shear modulus of hard drawn of carbon steel wire is 11,600,000 psi and hence the shear stresses are well below shear limits and will not fail [37]. There are two types stresses that occur on the hook of the spring; bending and torsional stress. Bending stress occur in the middle of the hook while torsional stress occurs in the inside of the hook. The torsional stresses are given by the following equation

$$\tau_{torsion} = K_{W_2}\left(\frac{8FD}{\pi d^3}\right)$$
$$K_{w_2} = \left(\frac{4C_2 - 1}{4C_2 - 4}\right)$$
$$C_2 = \frac{2r_i}{d} = \frac{ID}{d} = 7.89$$
$$K_{W_2} = 1.11$$
$$\tau_{torsion} = 71{,}332.60\ psi$$

Eq. 26

And bending stresses are given by

$$\sigma_{bend} = K_b\left(\frac{16FD}{\pi d^3}\right) + \left(\frac{4F}{\pi d^2}\right)$$
$$K_b = \left(\frac{4C_1^2 - C_1 - 1}{4C_1(C_1 - 1)}\right)$$
$$C_1 = \frac{D}{d} = c = 8.89 \qquad \text{Eq. 27}$$
$$K_b = 1.09$$
$$\sigma_{bend} = 144{,}049.09\ psi$$

Maximum bending and torsional stresses also occurs when the spring is fully extended and experienced 144,049.09 psi and 71,332.60 psi respectively of stress. The ultimate tensile strength and the modulus of torsion for hard drawn spring steel is 256,000 psi and 11,500,000 psi, respectively which are well above the calculated stresses and will not fail.

APPENDIX D – D-FMEA Tables

The tables below represent the different levels of severity, occurrence and ranking used for each individual failure mode in the D-FMEA Table in Section 3. Each table ranks from low to high and has specific description for each ranking. The severity table describes the level of consequence that is associated with the failure mode. The occurrence table indicates how often the failure mode might occur even after prevention mechanism are in place and the detection table indicates how easily the failure mode is detected before and after the design is made. As any further changes occur on the material handling design after this project, these tables can be used as a reference guide for the new set of failure modes for the changes made to the design.

TABLE D-I: D-FMEA SEVERITY RANKING

Severity Ranking		
Rank	Evaluation	Direct Risk to Personnel
1	No Effect	The personnel will not notice any failure effect as there are none
2 to 4	Minor	Annoyance or squeak or rattle; visual defects which does not affect function
5 to 6	Limited Effect	Degradation or loss of a secondary function of the item studied
7 to 8	High Significant Effect	Degradation or loss of a primary function of the item studied
9 to 10	Critical to Unacceptable	Regulatory/Safety Implications

TABLE D-II: D-FMEA OCCURENCE RANKING TABLE

Occurrence Ranking			
Ranking Grid		Level of Occurrence	Definition
Ranking	Criteria		
1	Highly Unlikely	Remote	Prevented Causes due to using known design standard
2	Low	Failure are few and far amongst them	Identical or similar design with no history of failure
3		Few failures	Isolated Failures
4	Moderate	Infrequent failures	

Occurrence Ranking			
Ranking Grid		Level of Occurrence	Definition
Ranking	Criteria		
5		Failure occur occasionally	Occasional failures have been experienced in the field or in development / verification testing
6		Failure seen intermittently	
7	High	High	New design with no history (based on current technology)
8		Repeated failures	
9		Failures occur regularly	
10	Very High	Failures are certain & persistent	New design with no experience with technology

TABLE D- III: D-FMEA DETECTION RANKING TABLE

Detection Ranking		
Ranking	Evaluation	Design Description
1	Almost Certain	Failure Prevented through design solution, standard and material standard
2	Very High	Use of CAD analysis highly correlated to real world stress profile
3	High	Test to failure with measurement of output tracking degradation (before DV)
4	Moderately High	Test to failure (Design Verification DV)
5	Moderate	Bogey Test, test to pass to 1 life and suspend the test (DV)
6	Low	Test to failure with measurement of output tracking degradation (after DV)
7	Very Low	Test to failure (product validation PV)
8	Remote	Bogey Test, test to pass to 1 life and suspend the test (PV)
9	Very Remote	Use of CAD analysis but not yet correlated to real world stress profile
10	Almost impossible	Cannot evaluate, no test available, current test does not excite the cause/failure mode

APPENDIX E – Bill of Material with Cost

TABLE E- I: BILL OF MATERIALS WITH COST

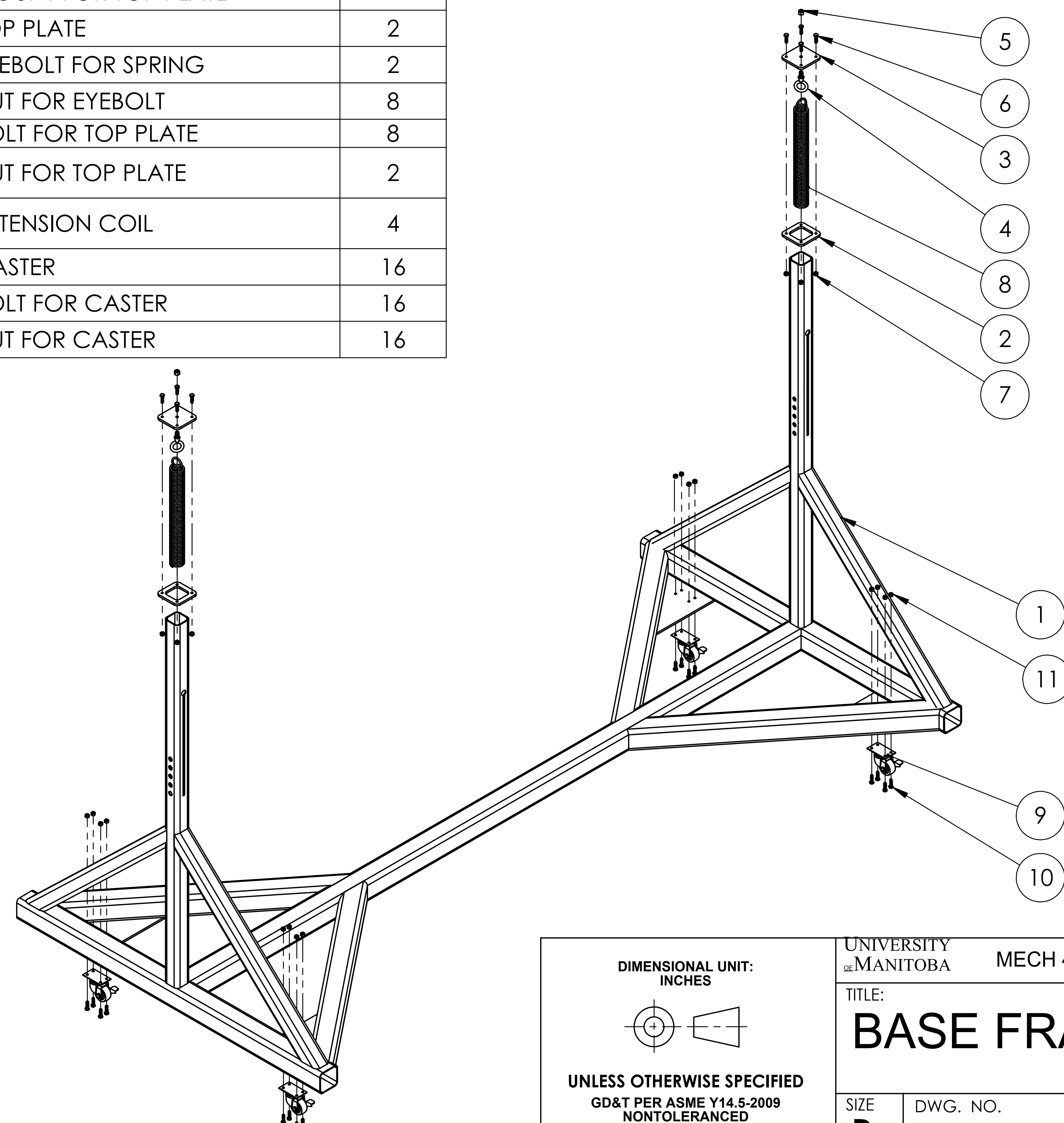
Part Name	Quantity	Source	Estimated Cost Each (USD)
Frame Design			
Mighty-Lite Caster with 3" x 1-3/4" Plate	4	McMaster Carr	$ 10.26
Caster Plate - Aluminum Pate 6061T6	4	Metal Supermarket	$ 34.56
Base Frame - 2.25X2.25 Aluminum Tube Sqaure	Total Length	Metal Supermarket	$ 171.37
Base Frame 3X3 Aluminum Tube Sqaure 6061T6	Total Length	Metal Supermarket	$ 278.98
316 Stainless Steel Washer	8	McMaster Carr	4.62 per pack of 100
Black-Oxide Alloy Steel Socket Head Screw	8	McMaster Carr	3.88 per pack of 25
18-8 Stainless Steel Hex Head Screw	8	McMaster Carr	$6.48 per pack of 25
18-8 Stainless Steel Hex Head Screw	8	McMaster Carr	$3.41 per pack of 10
18-8 Stainless Steel Hex Head Screw	8	McMaster Carr	$5.29 per pack of 50
18-8 Stainless Steel Hex Head Screw	8	McMaster Carr	$4.22 per pack of 100
Medium-Strength Steel Nylon-Insert Locknut	8	McMaster Carr	$3.43 per pack of 25
High-Strength Steel Nylon-Insert Locknut	8	McMaster Carr	$3.43 per pack of 25
High-Strength Steel Nylon-Insert Locknut	8	McMaster Carr	$4.31 per pack of 10
Zinc-Plated Steel Ring-Grip Quick-Release Pin	2	McMaster Carr	$ 2.47
Zinc-Plated Steel Ring-Grip Quick-Release Pin	2	McMaster Carr	$ 2.63
Clevis Pin with Hairpin Cotter Pin and Lanyard	2	McMaster Carr	$ 18.03
Zinc-Plated Steel Locking Pin with Wire Retainer	2	McMaster Carr	$ 6.61
Top Plate - Aluminum Plate 6061T6	2	Metal Supermarket	$ 13.48
Vertical Extension Spring	2	Acxess Spring	$ 51.88
18-8 Stainless Steel Ring-Grip Quick-Release Pin	2	McMaster Carr	$ 6.56
Steel Eyebolt with Shoulder - for Lifting	2	McMaster Carr	$ 3.15
High-Strength Steel Nylon-Insert Locknut	2	McMaster Carr	$3.08 per pack of 20

Part Name	Quantity	Source	Estimated Cost Each (USD)
Frame Design			
Upper Frame Assembly			
Steel Cold Rolled Tube for Upper Frame Left	1	Metal Supermarket	$ 35.71
Steel Cold Rolled Tube for Upper Frame Right	1	Metal Supermarket	$ 31.46
Circle	1	Metal Supermarket	$ 21.52
Steel Cold Rolled Tube for Upper Frame Top Left	1	Metal Supermarket	$ 30.05
Steel Cold Rolled Tube for Upper Frame Bottom Left	1	Metal Supermarket	$ 30.05
Steel Cold Rolled Tube for Upper Frame Top Right	1	Metal Supermarket	$ 38.65
Steel Cold Rolled Tube for Upper Frame Bottom Right	1	Metal Supermarket	$ 35.00
Clamp Design			
Steel Cold Rolled C1018 for Clamp Bottom Part	5	Metal Supermarket	$ 40.65
Steel Cold Rolled C1018 for Clamp Top Part	5	Metal Supermarket	$ 10.91
Rubber Buna-N 1/4in Thick 2X36in	5	Acklands Grainger	$ 14.36
Metric Comfort-Grip Plastic Multi-Arm Knob with Threaded Studs	5	McMaster Carr	$ 2.60
Plastic Multi-Lobe Knob	5	McMaster Carr	$ 7.97
Sleeve and Rotational Block Design			
Ball Bearing	4	McMaster Carr	$ 23.82
Rotational Block - Aluminum Sqaure Bar 6061 T6	4	Metal Supermarket	$ 39.42
Sleeve - Aluminum Tube Sqaure 6061T6	2	Metal Supermarket	$ 13.00

With a conversion rate of 1.27 USD to CAD and the incoming taxes of 13% the total cost would be $1400 without manufacturing costs.

APPENDIX F – Drawings

ITEM NO.	PART NUMBER	DESCRIPTION	Default/ QTY.
1	4-110	BASE FRAME	1
3	4-120	MOUNT FOR TOP PLATE	2
3	4-130	TOP PLATE	2
4	4-140	EYEBOLT FOR SPRING	2
5	4-150	NUT FOR EYEBOLT	8
6	4-160	BOLT FOR TOP PLATE	8
7	4-170	NUT FOR TOP PLATE	2
8	4-180	EXTENSION COIL	4
9	4-190	CASTER	16
10	4-200	BOLT FOR CASTER	16
2	4-210	NUT FOR CASTER	16



REV	DATE Y/M/D	DESCRIPTION		
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			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
1	17/11/25	RELEASED	DWN / CKD	M. STEPHANSON

DIMENSIONAL UNIT: INCHES

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED
DIMENSIONS ARE BASIC

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

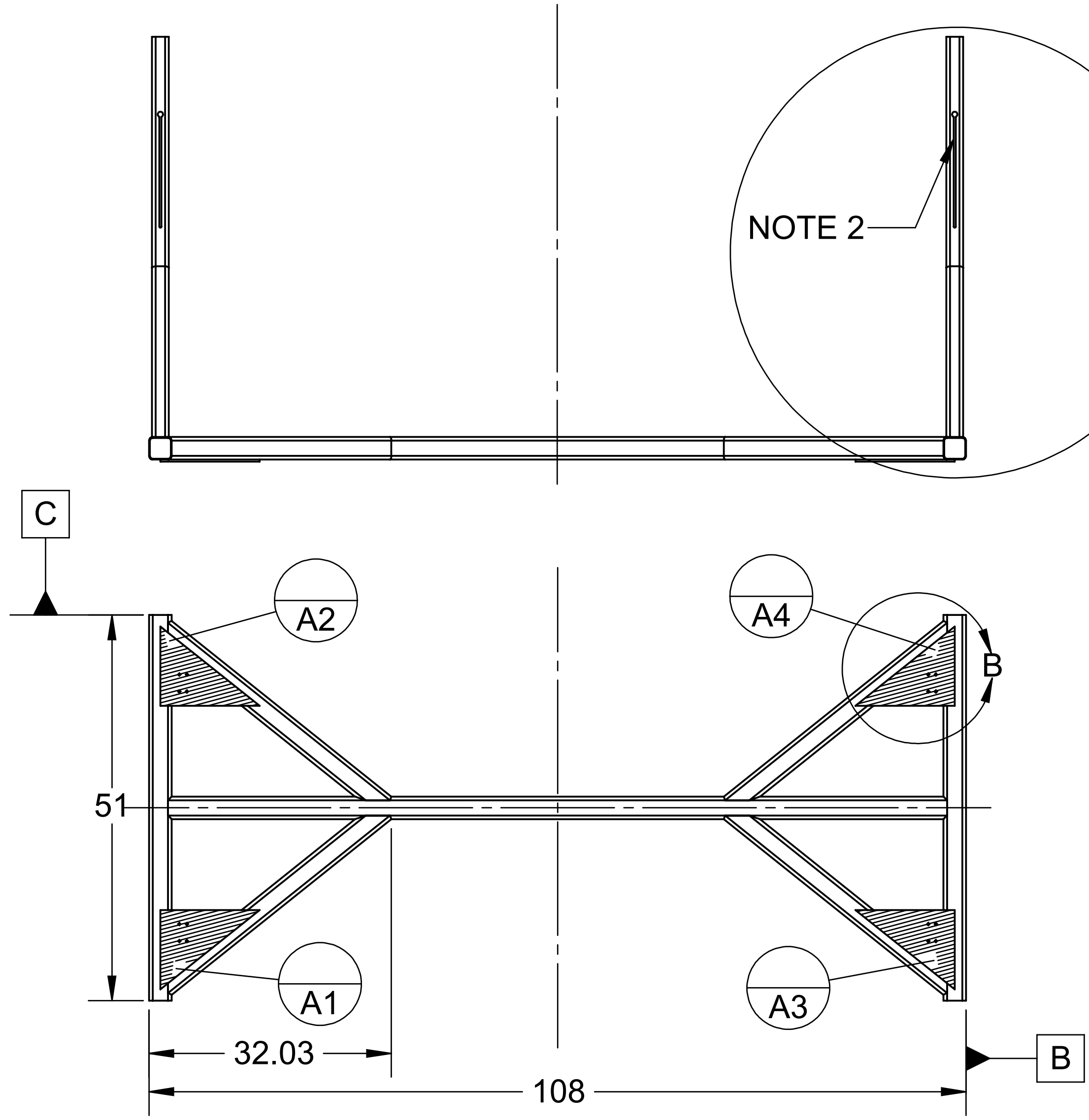
TITLE:

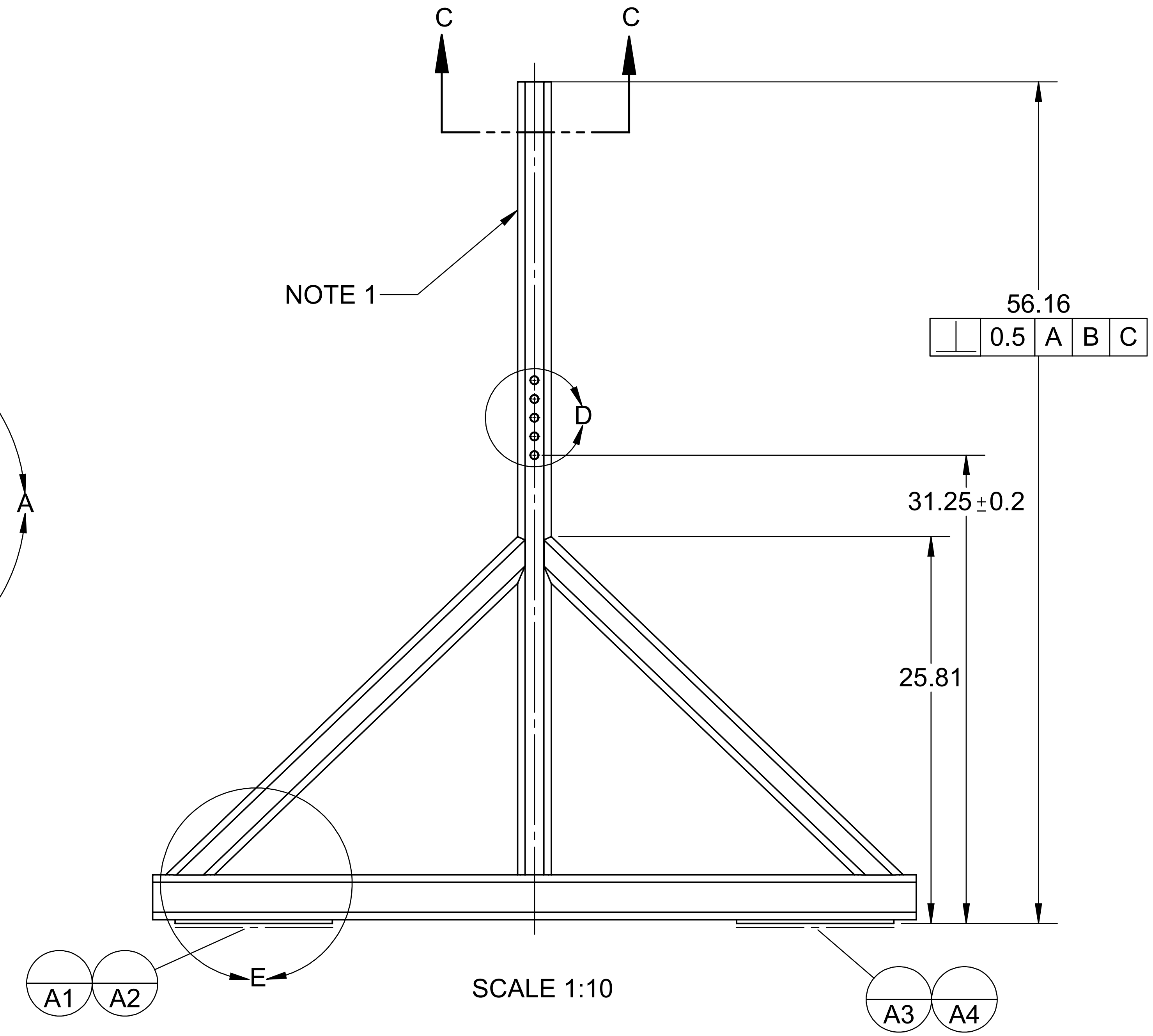
BASE FRAME

SIZE: B | DWG. NO. 1-100 | REV 1

SCALE: 1:50 | WEIGHT: 75.9 LBS | SHEET 1 OF 1

DWG. NO. 1-100 REV 1





NOTE 1: ROUND SHARP EDGES

NOTE 2: SLOT CUT THROUGH BOTH TUBE WALLS

REV	DATE Y/M/D	DESCRIPTION		
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			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE ⌓ 0.2 | A | B | C

THREADED FEATURES EXTERNAL THREADS AT ⌀MAJOR INTERNAL THREADS AT ⌀MINOR

CHAMFERS 0.3 ± 0.1 X 45° ± 2°

FILLETS 0.5 ± 0.02

MATERIAL
ALUMINUM 6061-76

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 ENGINEERING DESIGN

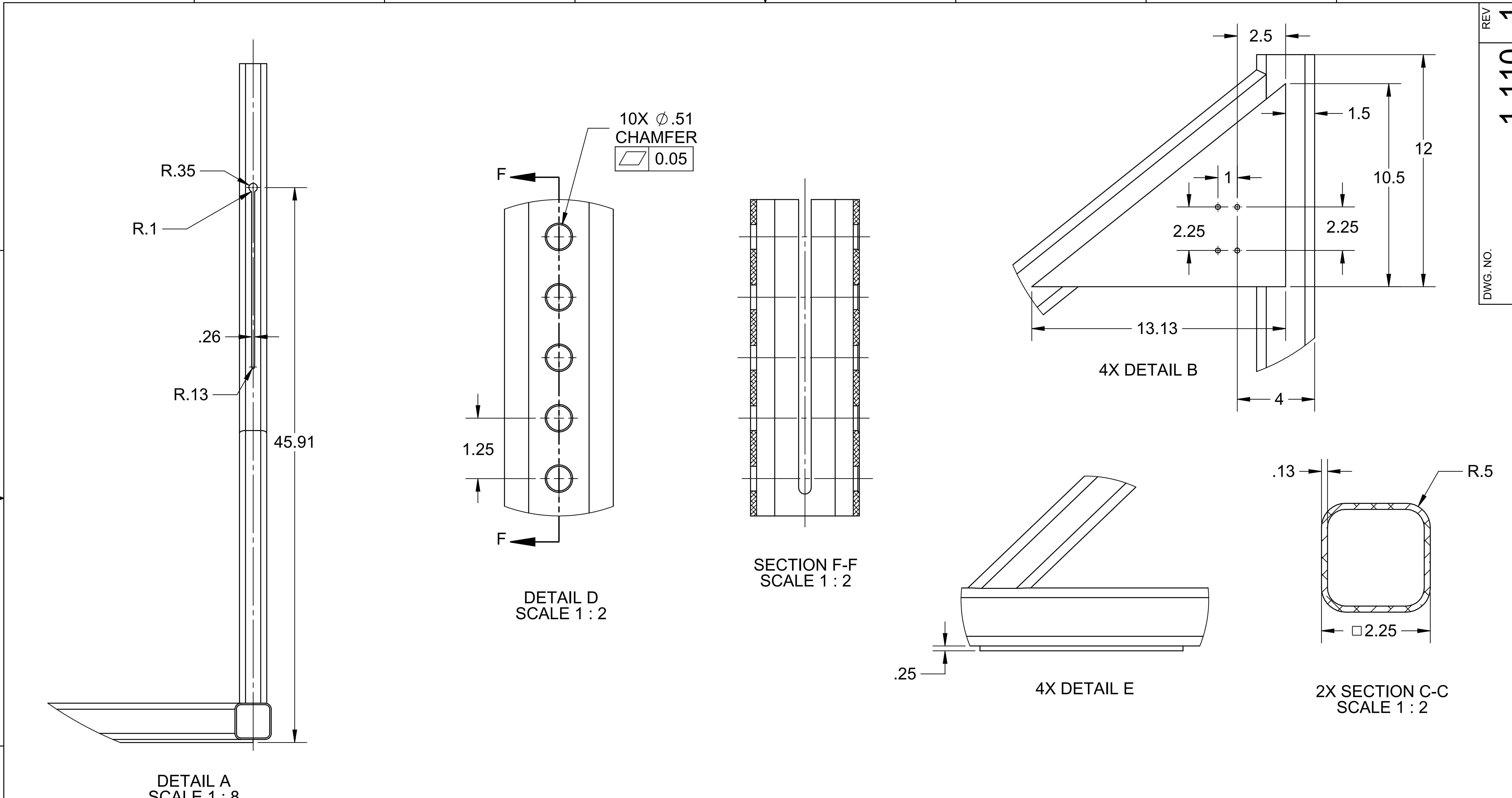
TITLE:

BASE FRAME

SIZE	DWG. NO.	REV
B	1-110	1

SCALE: 1:25 | WEIGHT: 73.23 LBS | SHEET 1 OF 2

REV 1 | DWG. NO. 1-110



REV	DATE Y/M/D	DESCRIPTION		
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			CKD	
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			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE | 0.2 | A | B | C |

CHAMFERS 0.3 ± 0.1 X 45° ± 2°

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR INTERNAL THREADS AT ∅MINOR

FILLETS 0.5 ± 0.02

MATERIAL
ALUMINUM 6061-T6

DO NOT SCALE DRAWING

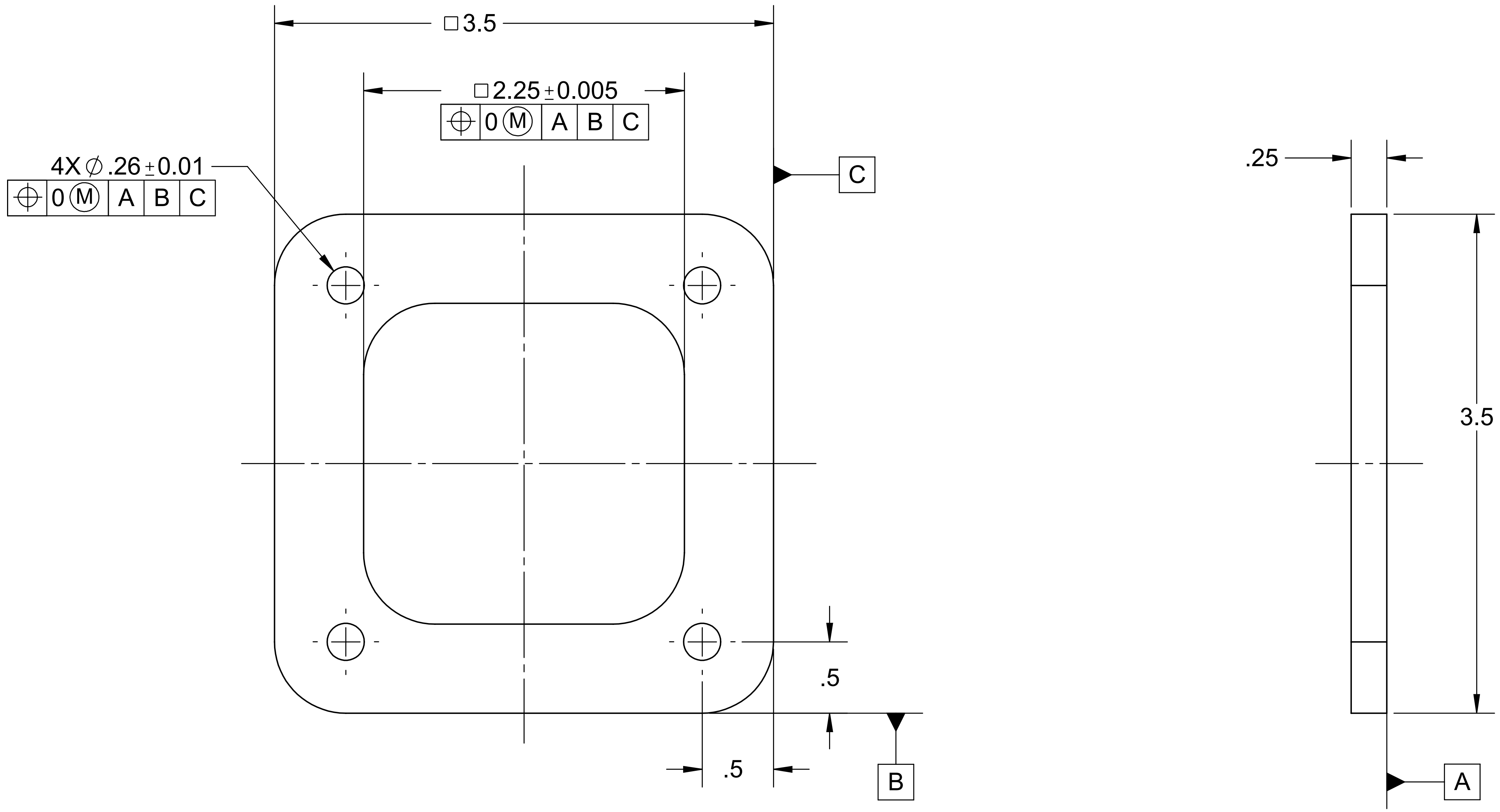
UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:

BASE FRAME

SIZE	DWG. NO.	REV
B	1-110	1

SCALE: 1:5 | WEIGHT: 73.23 LBS | SHEET 2 OF 2



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
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			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE ⌓ 0.02 | A | B | C

CHAMFERS N/A

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR, INTERNAL THREADS AT ∅MINOR

FILLETS 0.5 ± 0.02

MATERIAL
ALUMINUM 6061-T6

DO NOT SCALE DRAWING

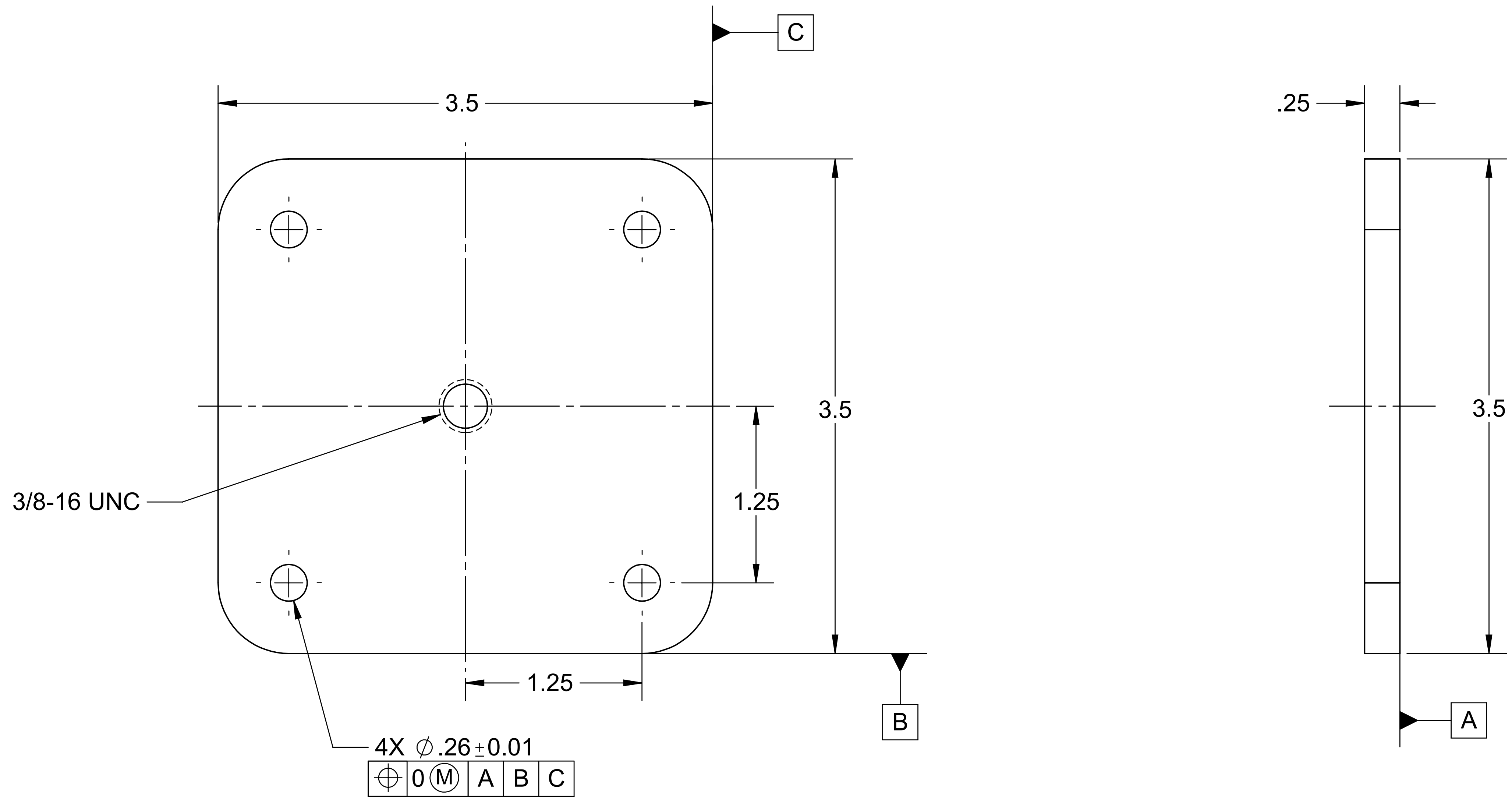
UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
MOUNT FOR TOP PLATE

SIZE: **B** | DWG. NO. 1-120 | REV 1

SCALE: 1:1 | WEIGHT: 0.17 LBS | SHEET 1 OF 1

DWG. NO. 1-120 REV 1



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
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			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE ⌓ 0.02 | A | B | C

CHAMFERS N/A

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR, INTERNAL THREADS AT ∅MINOR

FILLETS 0.5 ± 0.02

MATERIAL
ALUMINUM 6061-T6

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

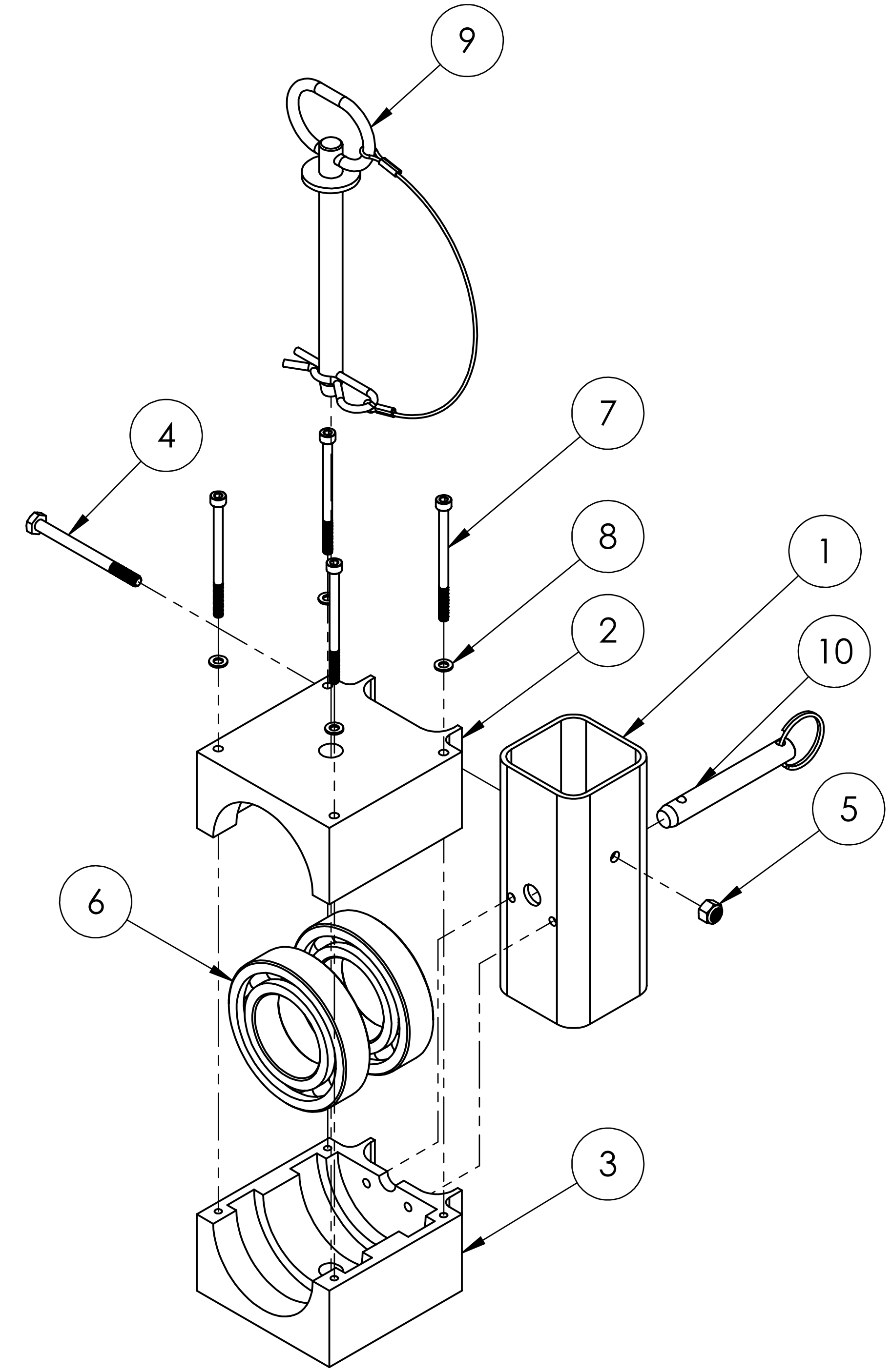
TITLE:
TOP PLATE

SIZE: B | DWG. NO. 1-130 | REV 1

SCALE: 1:1 | WEIGHT: 0.29 LBS | SHEET 1 OF 1

DWG. NO. 1-130 REV 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	2-110	SLEEVE	1
2	2-120	ROTATIONAL BLOCK TOP	1
3	2-130	ROTATIONAL BLOCK BOTTOM	1
4	2-140	SLEEVE BOLT TO HOLD SPRING	1
5	2-150	NUT FOR SLEEVE BOLT	1
6	2-160	ROTATIONAL BEARINGS	2
7	2-170	SCREW FOR ROTATIONAL BLOCK	4
8	2-180	WASHER FOR ROTATIONAL BLOCK	4
9	2-190	PIN TO LOCK ROTATIONAL ADJUSTMENT	1
10	2-200	PIN TO LOCK HEIGHT ADJUSTMENT	1



REV	DATE Y/M/D	DESCRIPTION		
1	17/11/25	RELEASED	DWN	H. SANDHU

DIMENSIONAL UNIT: INCHES

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED
DIMENSIONS ARE BASIC

DO NOT SCALE DRAWING

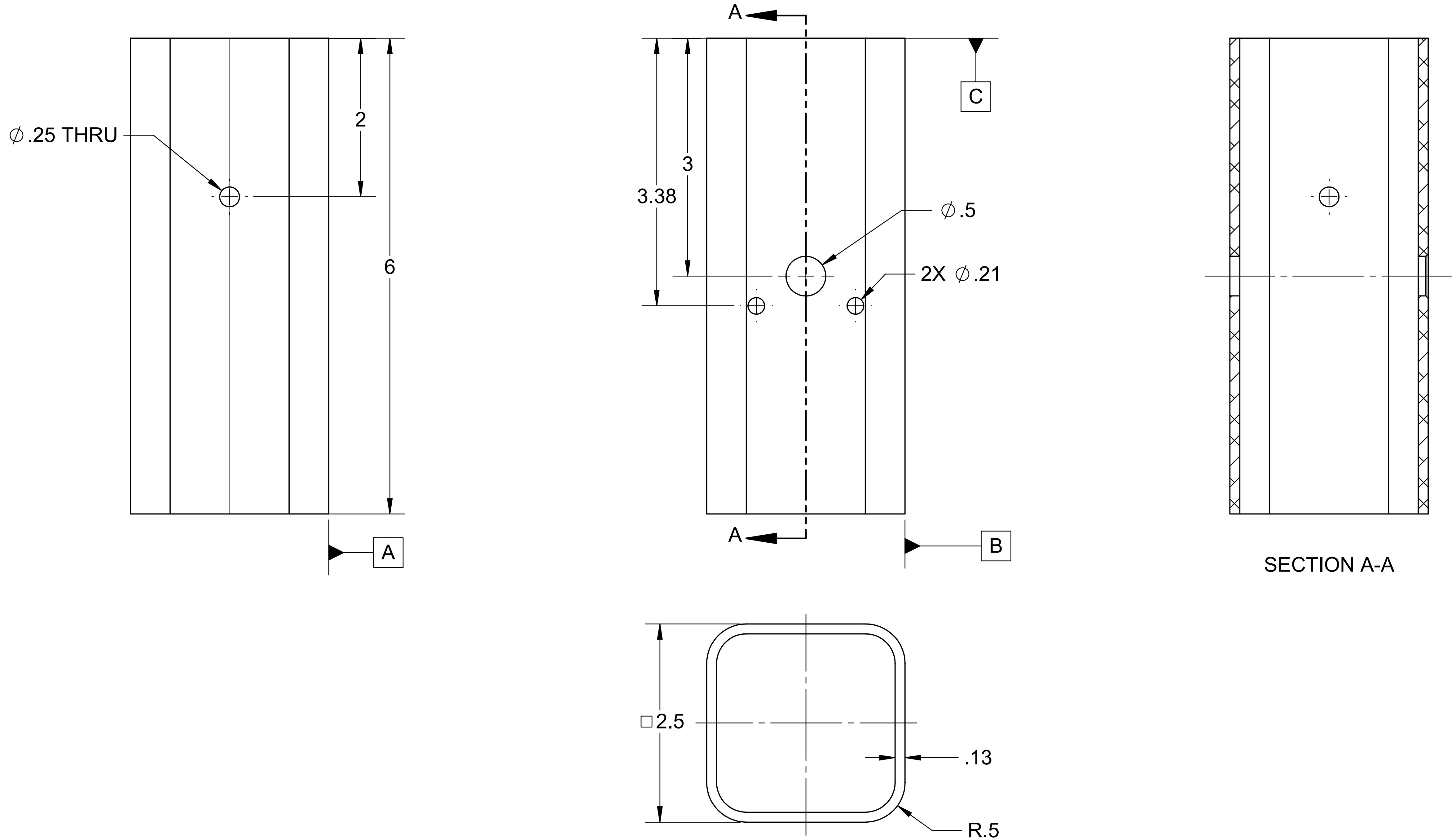
UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:

BASE FRAME

SIZE: B — DWG. NO. 2-100 — REV 1

SCALE: 1:10 — WEIGHT: 5.03 LBS — SHEET 1 OF 1



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE

⌓	0.02	A	B	C

CHAMFERS 0.03 ± 0.01 X 45° ± 2°

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR INTERNAL THREADS AT ∅MINOR

FILLETS 0.05 ± 0.02

MATERIAL
ALUMINUM 6061-T6

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA

MECH 4860 ENGINEERING DESIGN

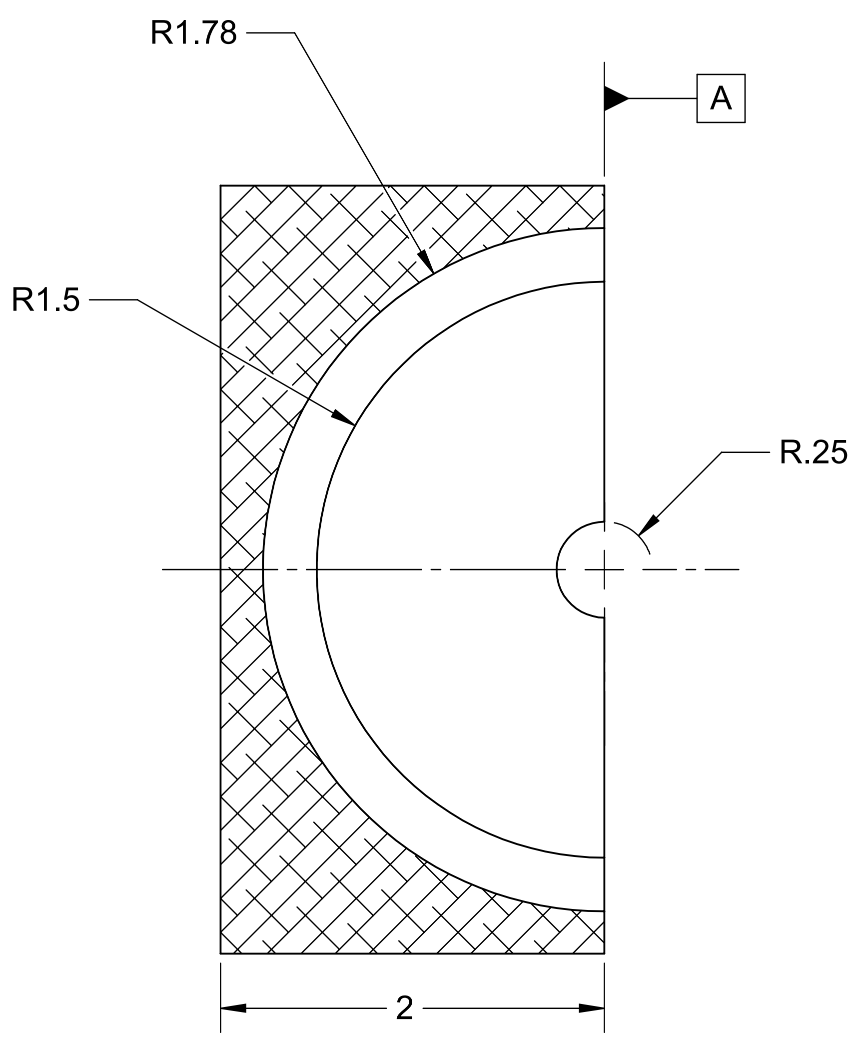
TITLE:

SLEEVE

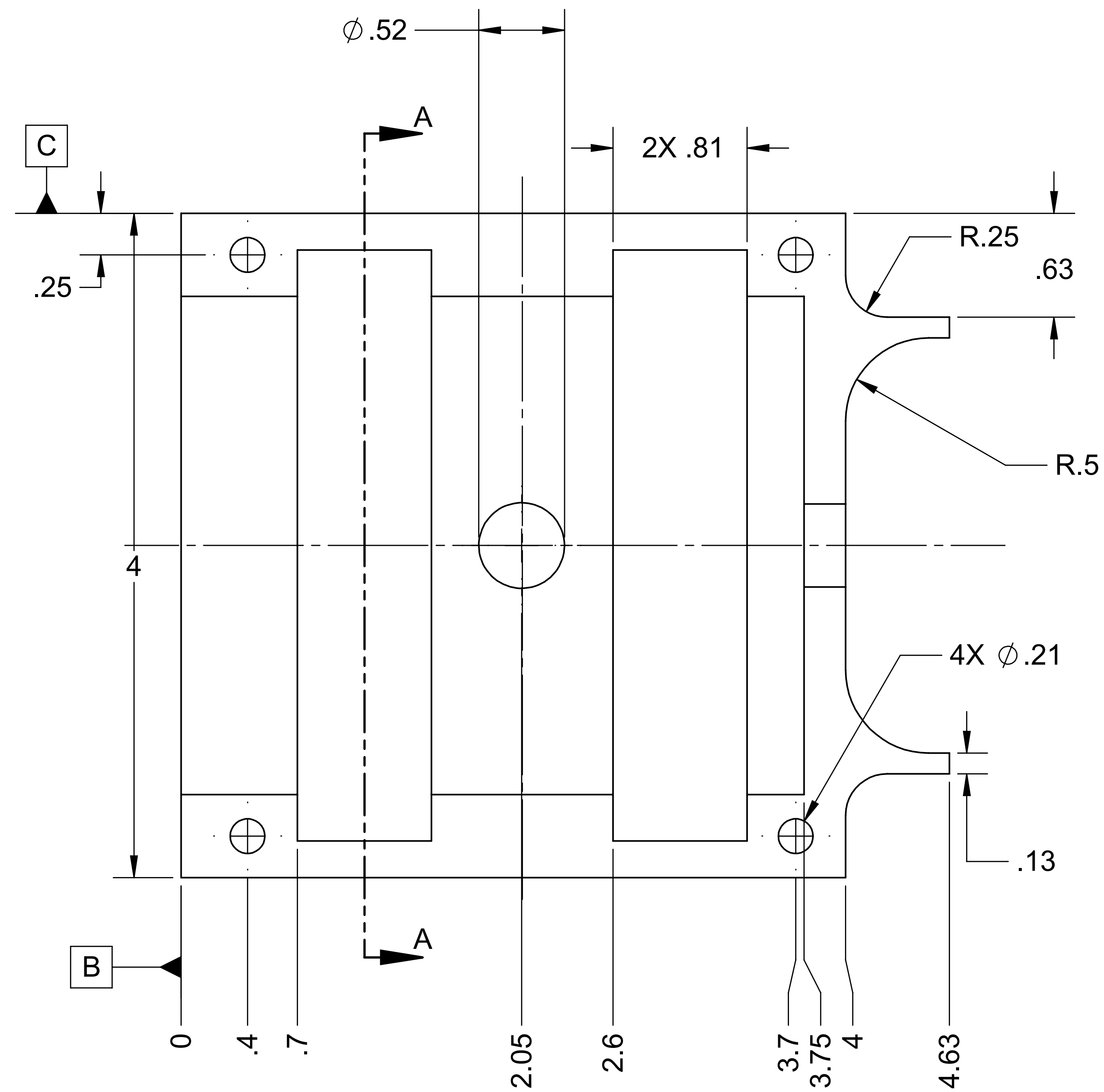
SIZE	DWG. NO.	REV
B	2-110	1

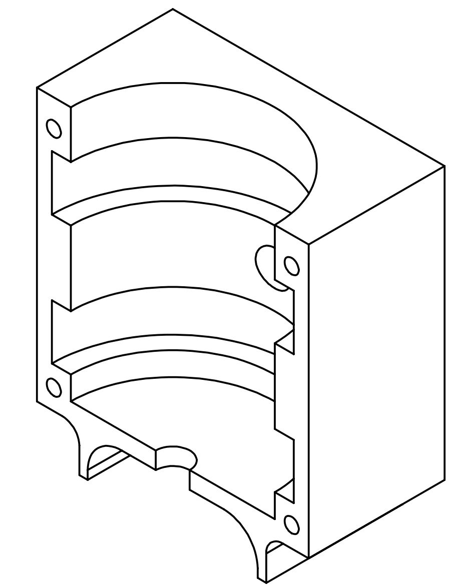
SCALE: 2:3 | WEIGHT: 0.63 LBS | SHEET 1 OF 1

DWG. NO. 2-110 REV 1



SECTION A-A





SCALE 1 : 2

NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE
⌓ 0.02 | A | B | C

CHAMFERS N/A

THREADED FEATURES EXTERNAL THREADS AT ⌀MAJOR, INTERNAL THREADS AT ⌀MINOR

FILLETS 0.05 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

UNIVERSITY of MANITOBA

MECH 4860 ENGINEERING DESIGN

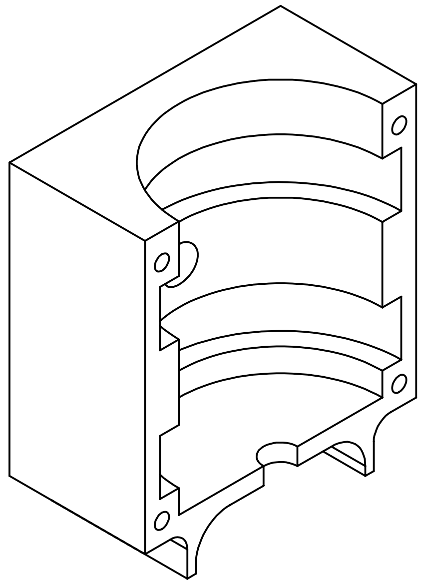
TITLE:

ROTATIONAL BLOCK TOP

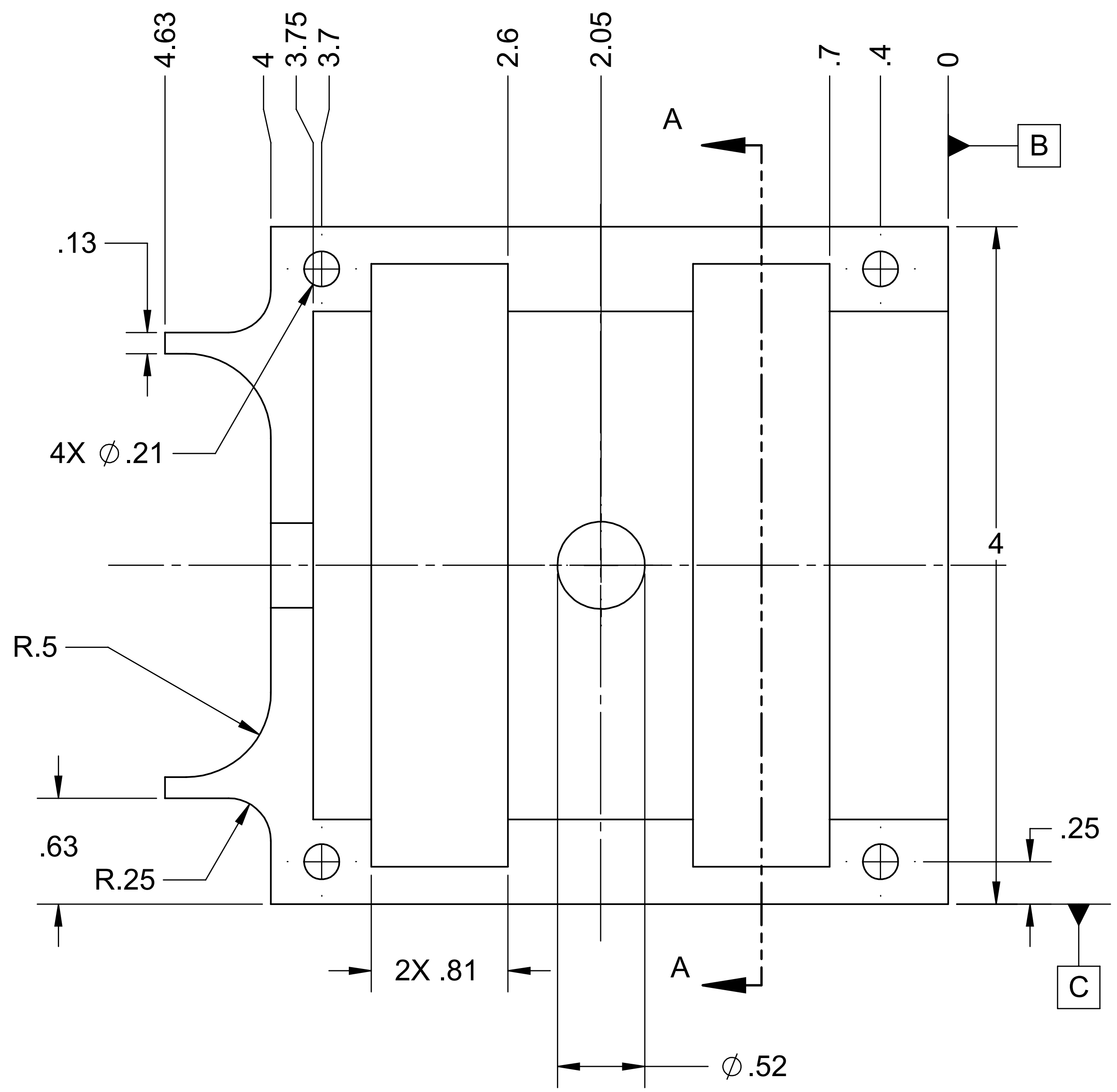
SIZE **B** | DWG. NO. 2-120 | REV 1

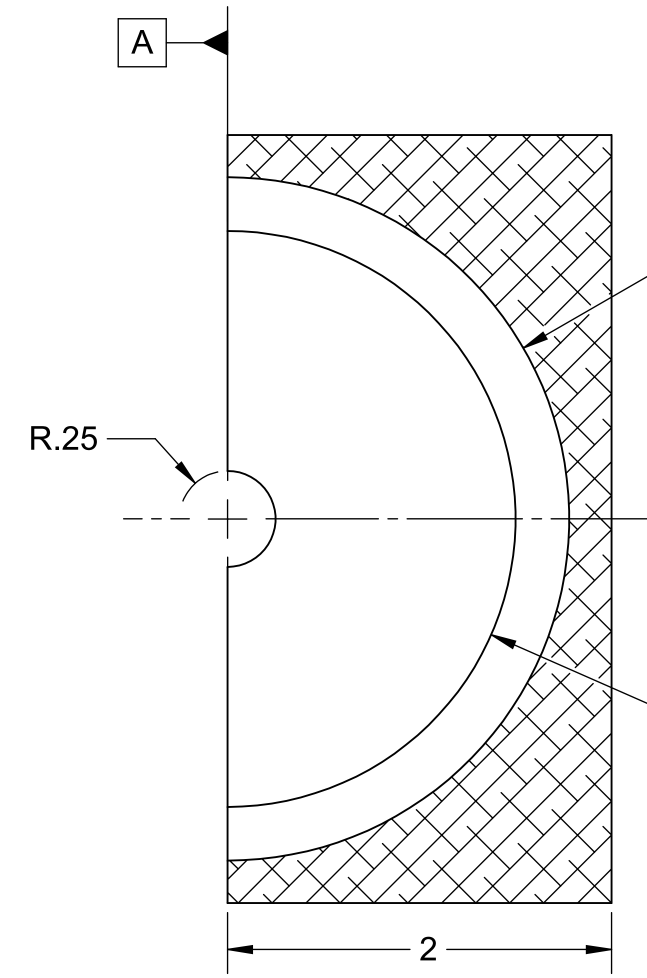
SCALE: 1:1 | WEIGHT: 1.62 LBS | SHEET 1 OF 1

DWG. NO. 2-120 REV 1



SCALE 1 : 2





SECTION A-A

NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE

| 0.02 | A | B | C |

CHAMFERS N/A

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR INTERNAL THREADS AT ∅MINOR

FILLETS 0.05 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:

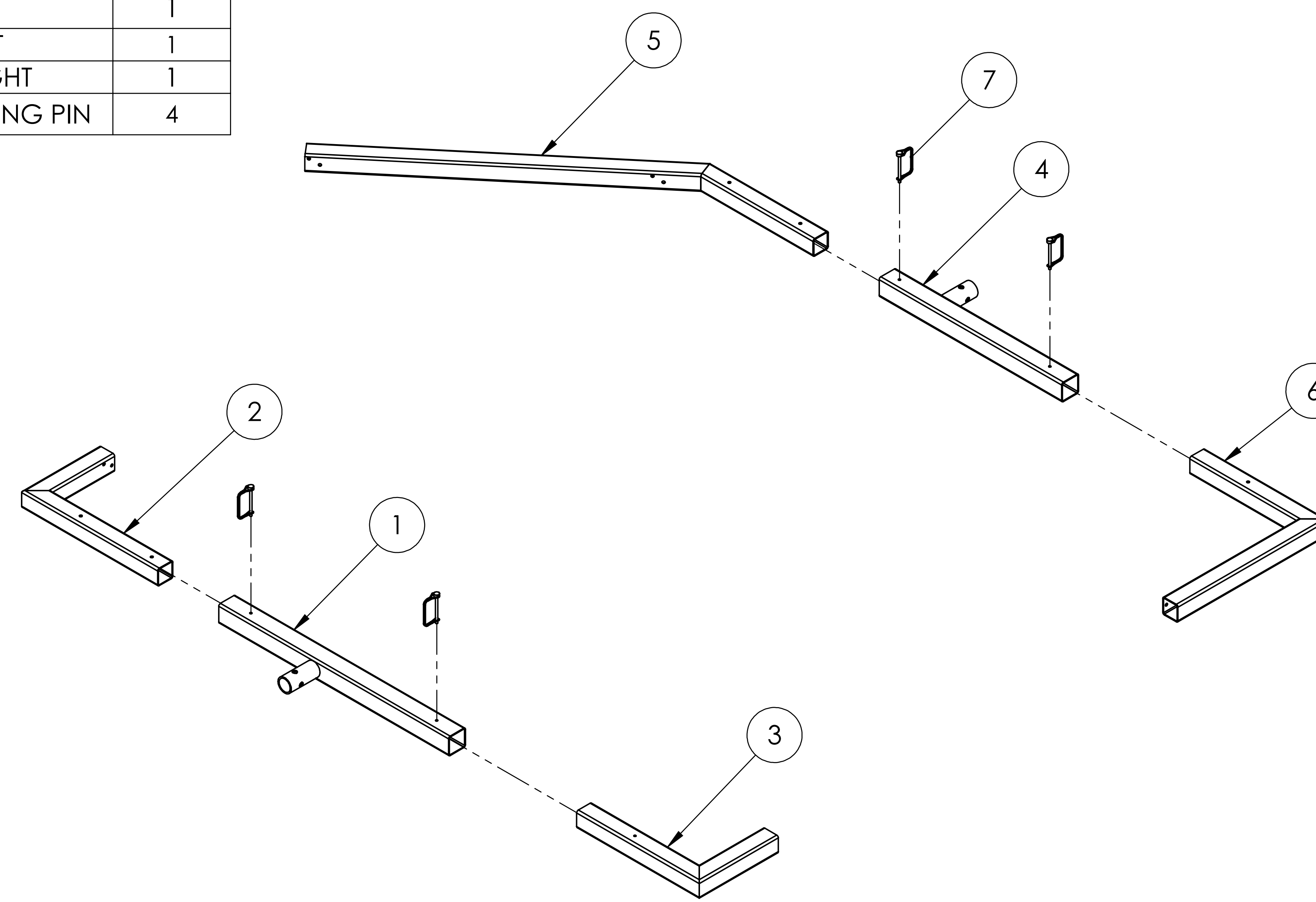
ROTATIONAL BLOCK BOTTOM

SIZE	DWG. NO.	REV
B	2-130	1

SCALE: 1:1 | WEIGHT: 1.62 LBS | SHEET 1 OF 1

DWG. NO. 2-130 REV 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	3-110	UPPER FRAME LEFT	1
2	3-120	UPPER FRAME TOP LEFT	1
3	3-130	UPPER FRAME BOTTOM LEFT	1
4	3-140	UPPER FRAME RIGHT	1
5	3-150	UPPER FRAME TOP RIGHT	1
6	3-160	UPPER FRAME BOTTOM RIGHT	1
7	3-170	UPPER FRAME TOP RIGHT LOCKING PIN	4



REV	DATE Y/M/D	DESCRIPTION		
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	M. STEPHANSON
			CKD	

DIMENSIONAL UNIT: INCHES

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED
DIMENSIONS ARE BASIC

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

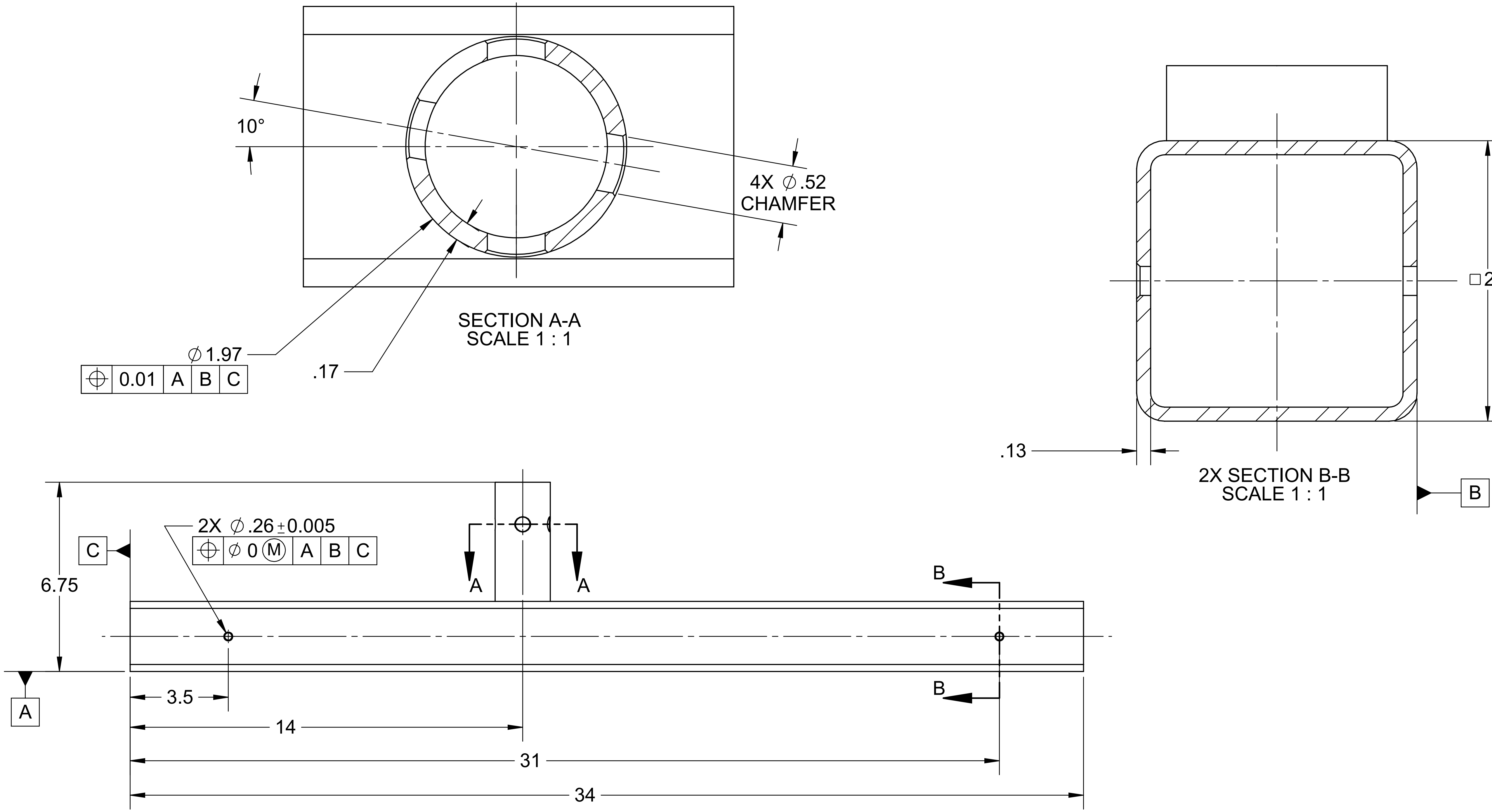
TITLE:

UPPER FRAME

SIZE: B | DWG. NO. 3-100 | REV 1

SCALE: 1:50 | WEIGHT: 66.88 LBS | SHEET 1 OF 1

DWG. NO. 3-100 REV 1



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
			DWN / CKD	
1	17/11/25	RELEASED	DWN	M. STEPHANSON
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE ⌓ 0.02 | A | B | C

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR INTERNAL THREADS AT ∅MINOR

CHAMFERS 0.03 ± 0.01 X 45° ± 2°

FILLETS 0.05 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

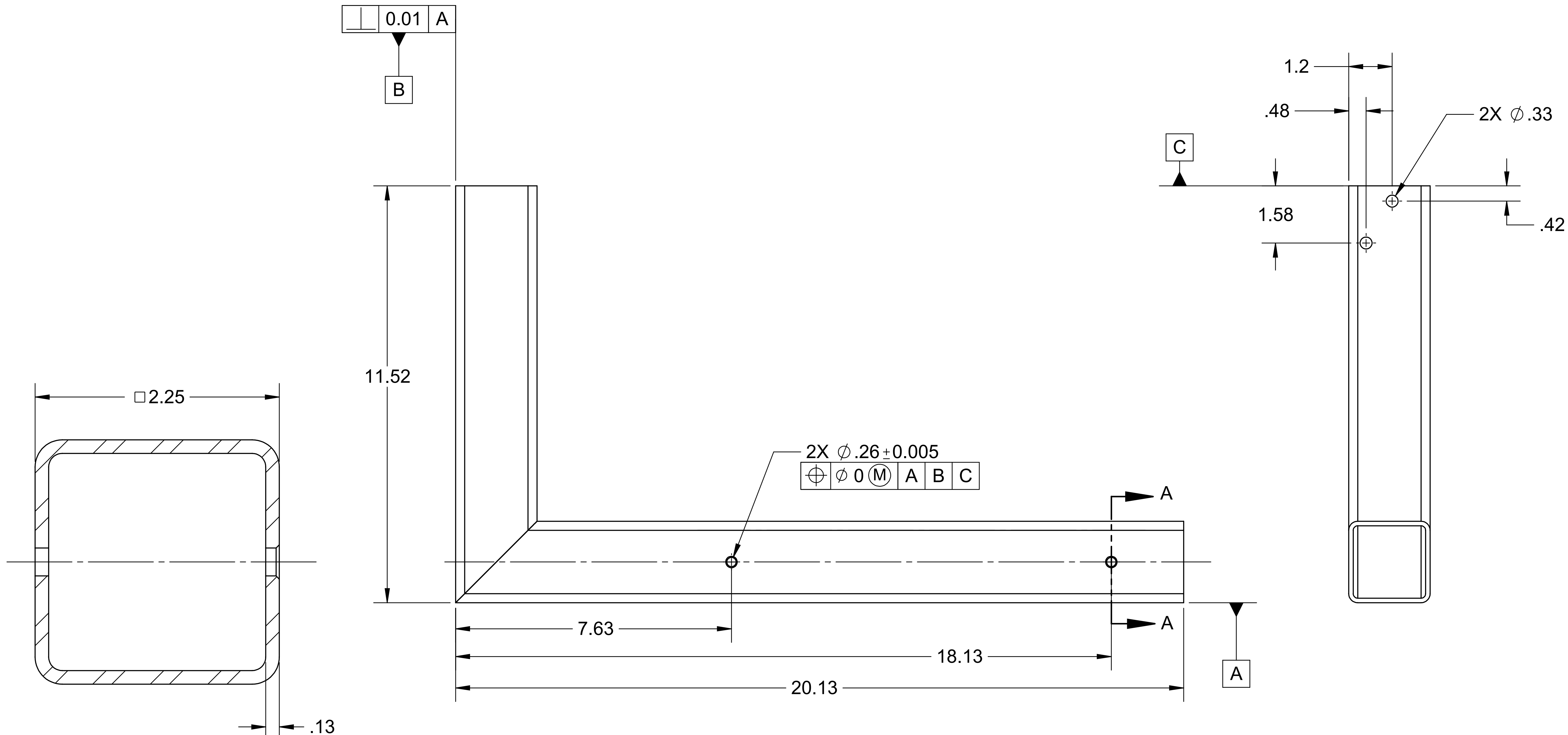
UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
UPPER FRAME LEFT

SIZE	DWG. NO.	REV
B	3-110	1

SCALE: 1:4 | WEIGHT: 12.1 LBS | SHEET 1 OF 1

DWG. NO. 3-110 REV 1



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE 0.02 | A | B | C

CHAMFERS 0.03 ± 0.01 X 45° ± 2°

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR, INTERNAL THREADS AT ∅MINOR

FILLETS 0.25 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

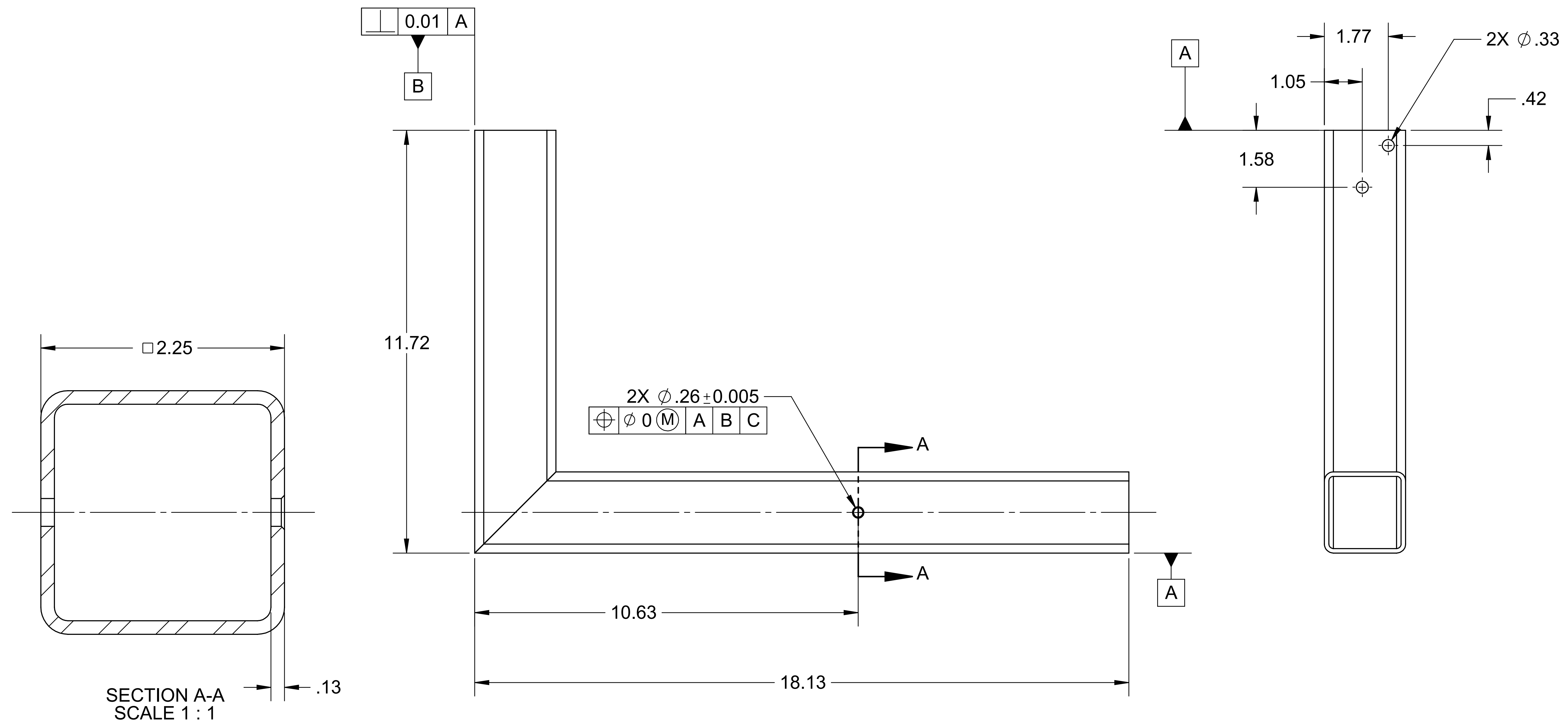
UNIVERSITY of MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
UPPER FRAME TOP LEFT

SIZE: B | DWG. NO. 3-120 | REV 1

SCALE: 1:3 | WEIGHT: 8.45 LBS | SHEET 1 OF 1

DWG. NO. 3-120 REV 1



NOTE 1: ROUND SHARP EDGES

			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	H. SANDHU
			CKD	
REV	DATE Y/M/D	DESCRIPTION		

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE 0.02 | A | B | C

CHAMFERS 0.03 ± 0.01 X 45° ± 2°

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR INTERNAL THREADS AT ∅MINOR

FILLETS 0.25 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

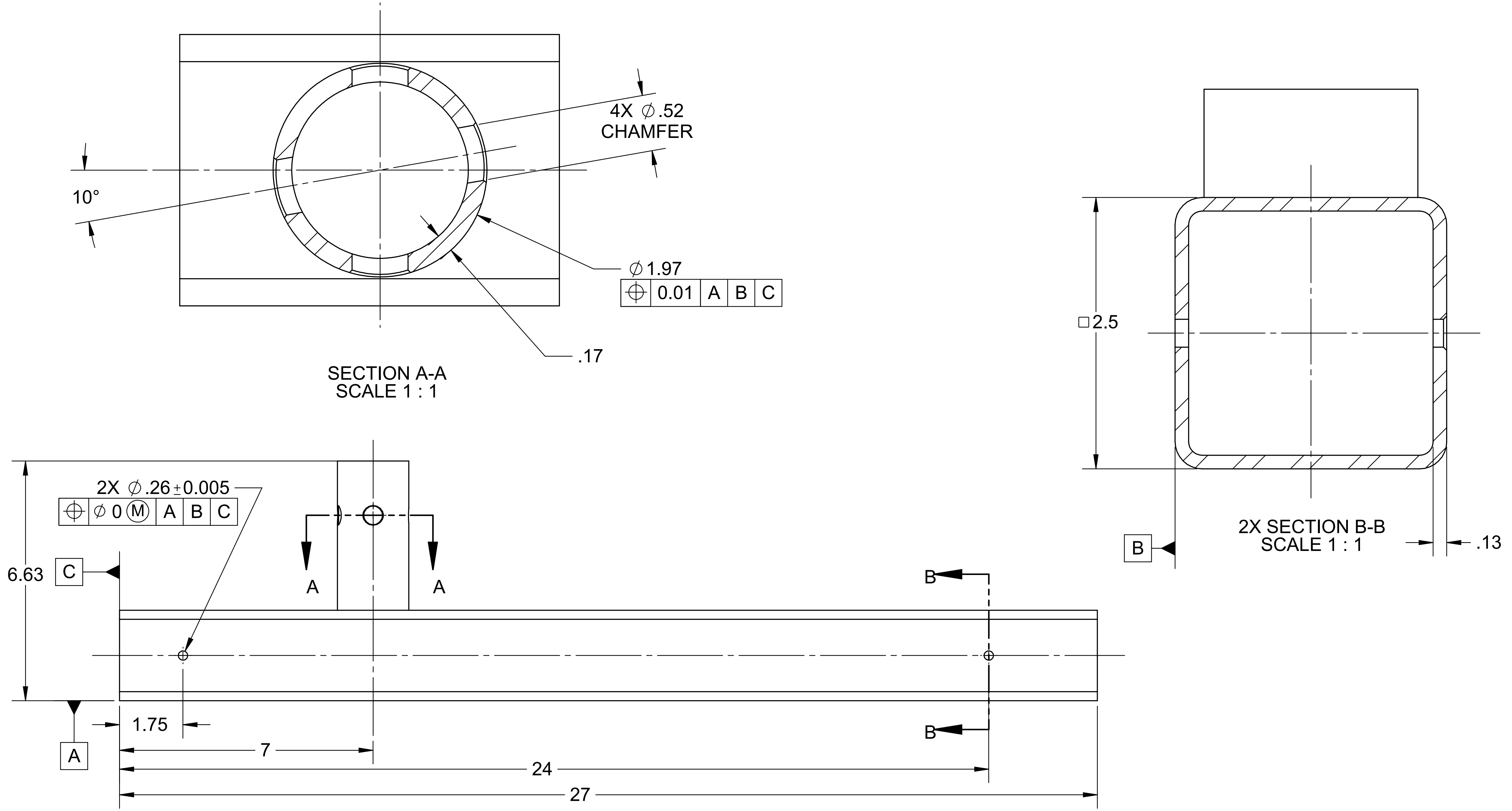
UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
UPPER FRAME BOTTOM LEFT

SIZE	DWG. NO.	REV
B	3-130	1

SCALE: 1:3 | WEIGHT: 7.94 LBS | SHEET 1 OF 1

REV 1 — DWG. NO. 3-130



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE
⌓ 0.02 | A | B | C

CHAMFERS 0.03 ± 0.01 X 45° ± 2°

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR INTERNAL THREADS AT ∅MINOR

FILLETS 0.25 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

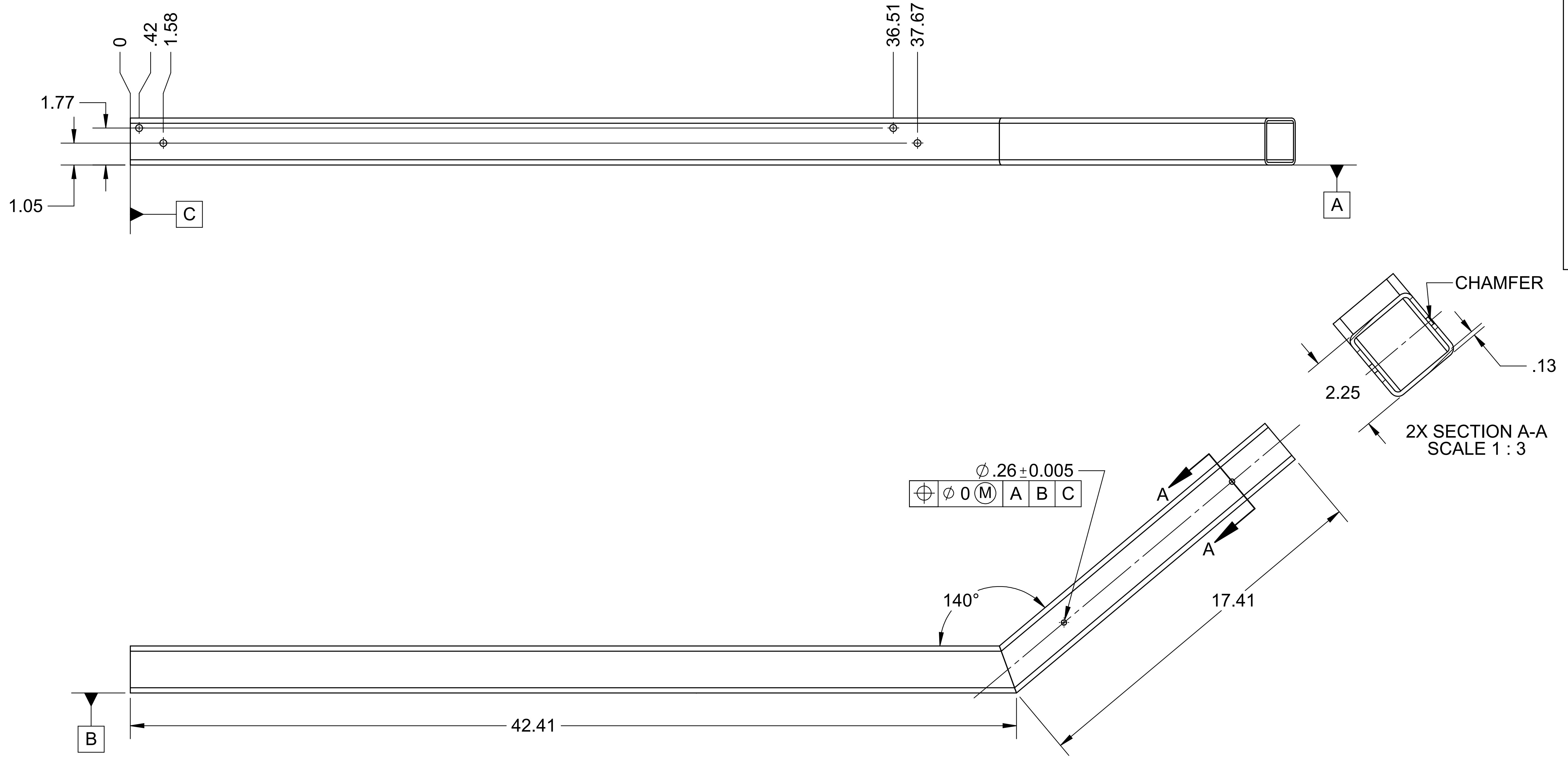
DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA
MECH 4860 ENGINEERING DESIGN

TITLE:
UPPER FRAME RIGHT

SIZE	DWG. NO.	REV
B	3-140	1
SCALE: 1:3	WEIGHT: 9.81 LBS	SHEET 1 OF 1

REV 1
DWG. NO. 3-140



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE ⌓ 0.02 | A | B | C

CHAMFERS 0.03 ± 0.01 X 45° ± 2°

THREADED FEATURES EXTERNAL THREADS AT ⌀MAJOR, INTERNAL THREADS AT ⌀MINOR

FILLETS 0.25 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

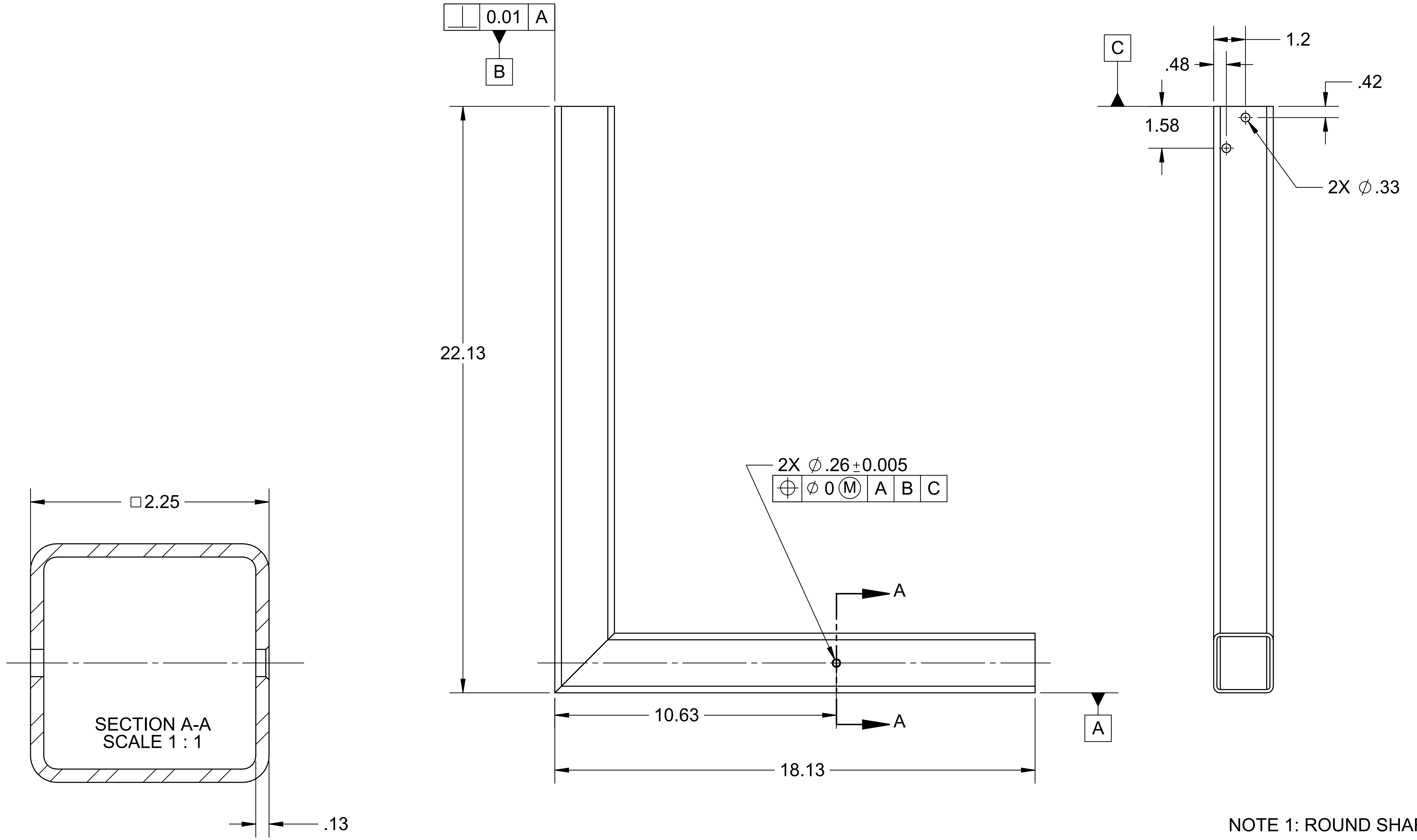
UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
UPPER FRAME TOP RIGHT

SIZE **B** | DWG. NO. 3-150 | REV 1

SCALE: 1:5 | WEIGHT: 17.13 LBS | SHEET 1 OF 1

DWG. NO. 3-150 REV 1



NOTE 1: ROUND SHARP EDGES

			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	
REV	DATE Y/M/D	DESCRIPTION		

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE 0.02 | A | B | C

CHAMFERS 0.03 ± 0.01 X 45° ± 2°

THREADED FEATURES EXTERNAL THREADS AT ∅MAJOR INTERNAL THREADS AT ∅MINOR

FILLETS 0.25 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

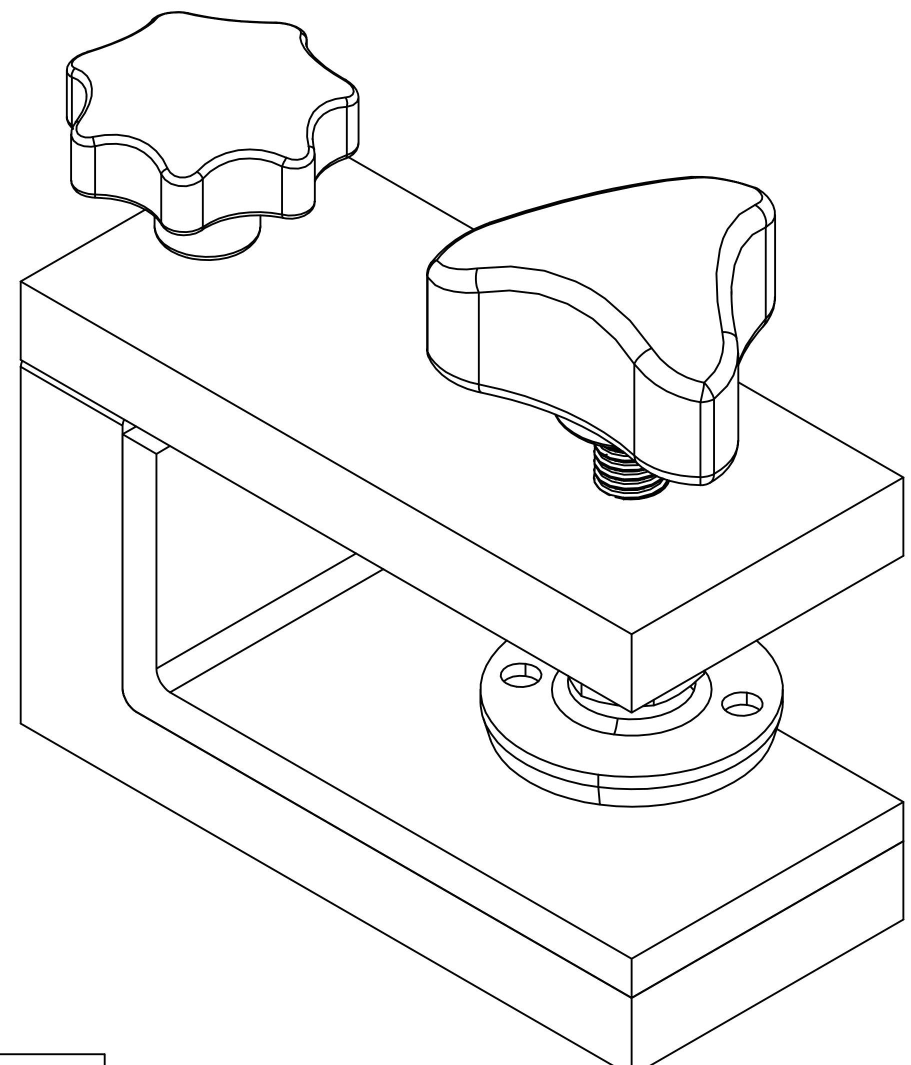
TITLE:
UPPER FRAME BOTTOM RIGHT

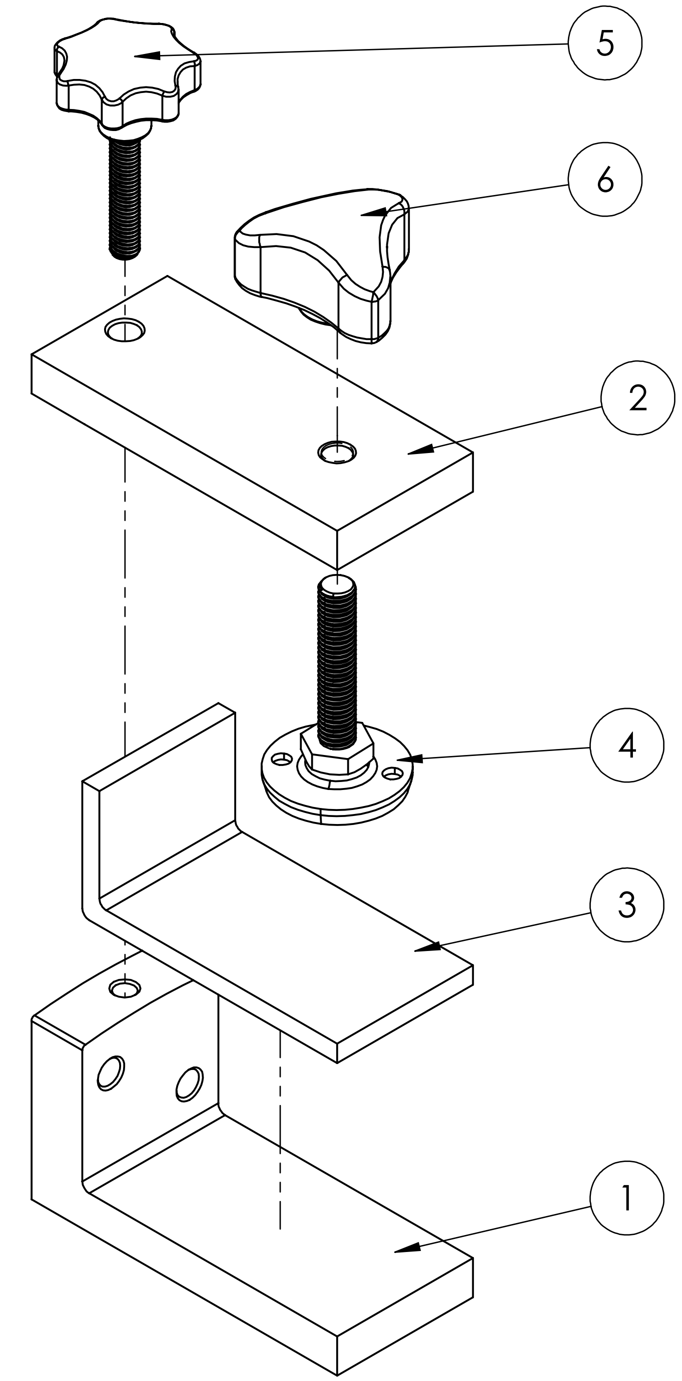
SIZE **B** | DWG. NO. 3-160 | REV 1

SCALE: 1:4 | WEIGHT: 10.94 LBS | SHEET 1 OF 1

DWG. NO. 3-160 REV 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
2	4-110	CLAMP BOTTOM	1
1	4-120	CLAMP TOP	1
3	4-130	FOAM PADDING	1
4	4-140	CLAMPING PLATFORM	1
5	4-150	ROTATIONAL LOCKING HANDLE	1
6	4-160	CLAMPING HANDLE	1





SCALE 1:2

REV	DATE Y/M/D	DESCRIPTION		
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

DIMENSIONAL UNIT: INCHES

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DO NOT SCALE DRAWING

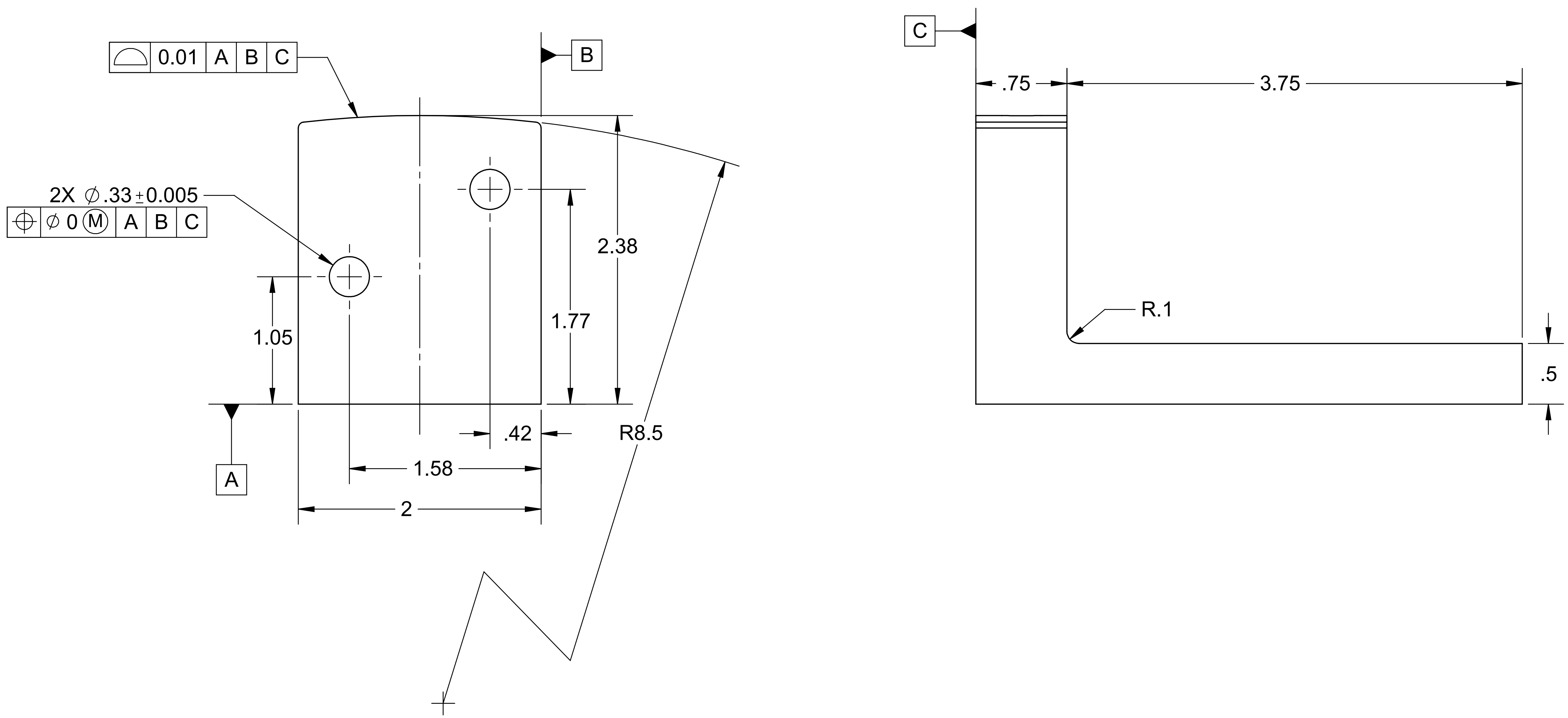
UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
CLAMP ASSEMBLY

SIZE	DWG. NO.	REV
B	4-100	1

SCALE: 1:1 | WEIGHT: 3.78 LBS | SHEET 1 OF 1

DWG. NO. 4-100 REV 1



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE 0.02 A B C

CHAMFERS N/A

THREADED FEATURES EXTERNAL THREADS AT ⌀MAJOR INTERNAL THREADS AT ⌀MINOR

FILLETS 0.05 ± 0.02

MATERIAL
AISI 1020 COLD ROLLED STEEL

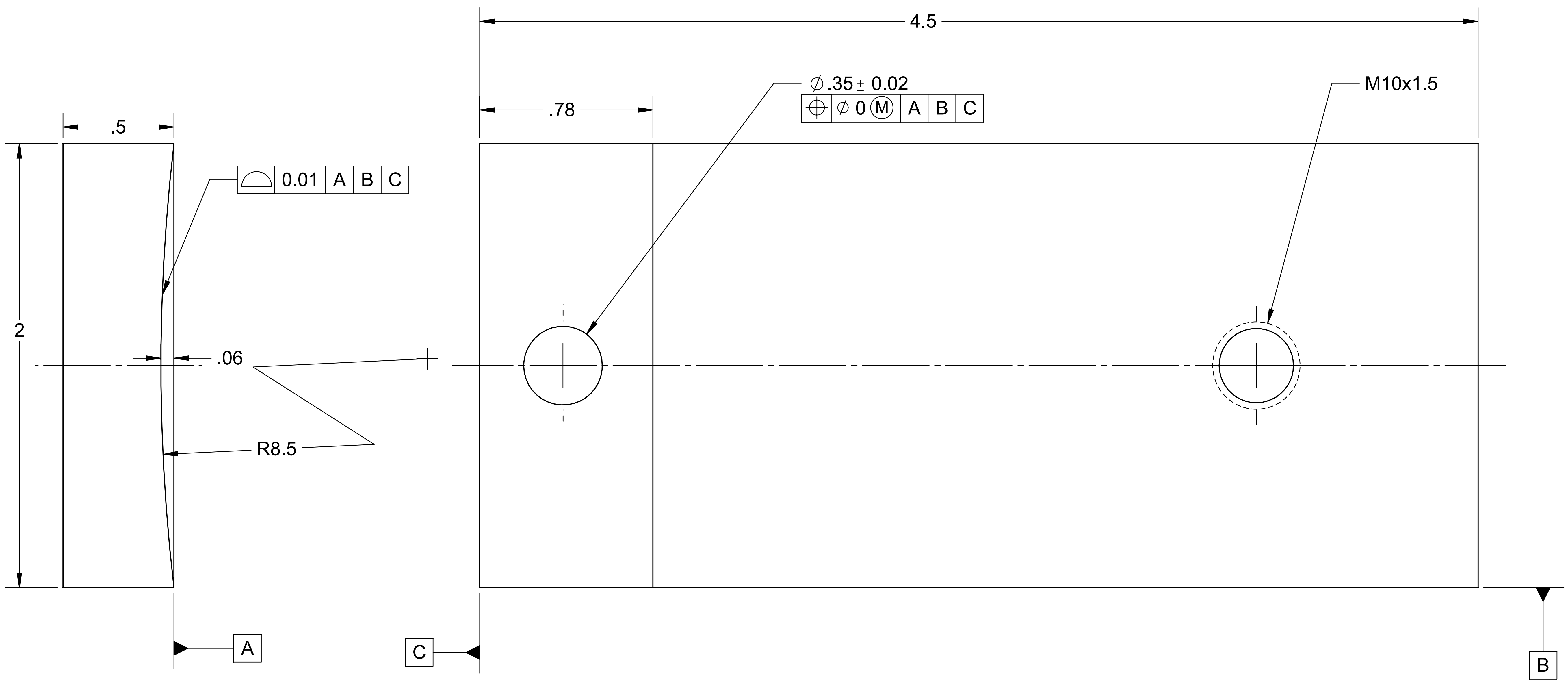
DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
CLAMP BOTTOM

SIZE	DWG. NO.	REV
B	4-110	1
SCALE: 1:2	WEIGHT: 2.02 LBS	SHEET 1 OF 1

DWG. NO. 4-110 REV 1



NOTE 1: ROUND SHARP EDGES

REV	DATE Y/M/D	DESCRIPTION		
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
			DWN	
			CKD	
1	17/11/25	RELEASED	DWN	J. FURLAN
			CKD	

UNLESS OTHERWISE SPECIFIED
GD&T PER ASME Y14.5-2009
NONTOLERANCED DIMENSIONS ARE BASIC

DIMENSIONAL UNIT: INCHES

N/A = NOT APPLICABLE

GENERAL TOLERANCE 0.02 | A | B | C

CHAMFERS 0.3 ± 0.1 X 45° ± 2°

FILLETS N/A

THREADED FEATURES EXTERNAL THREADS AT ⌀MAJOR, INTERNAL THREADS AT ⌀MINOR

MATERIAL
AISI 1020 COLD ROLLED STEEL

DO NOT SCALE DRAWING

UNIVERSITY OF MANITOBA — MECH 4860 ENGINEERING DESIGN

TITLE:
CLAMP TOP

SIZE	DWG. NO.	REV
B	4-120	1
SCALE: 2:1	WEIGHT: 1.15 LBS	SHEET 1 OF 1