Final Design Report
Volume I

RYERSON SHEET TRANSFER SYSTEM

MECH 4860
Team 19
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Sponsor:

[redacted]

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Executive Summary

The purpose of the project was to design a sheet metal transfer system for Ryerson Canada Inc. The current sheet metal handling process presents a safety concerns such as repetitive strain injuries, back pain and minor lacerations to employee's hand and forearm. These safety concerns arise due to the current sheet metal transfer process being fully manual, and extremely difficult for one employee to complete. To solve the problem, the Team identified the constraints and target specifications for the project. A systematic approach was taken to determine the most viable concept to pursue. Finally the team took the most viable concept and developed it into a detailed design that Ryerson could submit for proper engineering approval.

Within the detailed design the Team generated a package that consists of preliminary technical drawings, a bill of materials, material costs estimates, and a list of recommendations that pertain to the continued development of the transfer system's design. Unfortunately the Team was required to reduce the scope it set out during the project definition phase due to resource and time constraints. Items omitted from the machine's design include the electrical and pneumatic system routing, PLC selections, and a drive system. With the reduction in scope the Team focused its efforts on the core components of the system in the form of the lifting and sliding mechanism.

The machine designed by Team 19 is intended to be integrated onto an existing cantilever racking systems. The transfer system utilizes a vacuum lifter to move sheet material from the stock skid to the customer order pallet. Vertical actuation of the vacuum lifter is accomplished with a 28-inch stroke pneumatic cylinder that attaches to the lifting mechanism. The lifting mechanism moves between the loading and unloading areas along a pair of modified cantilever. This sliding mechanism is retractable to minimize the machine's footprint. The machine designed is expected to lift sheets of material up to 250 lbs that is approximately 4 feet wide and 10 feet long which represents roughly 65% of the sales orders at Ryerson's Winnipeg location. The machine is divided up into three distinct components that, as a whole, will reduce the likelihood of injury occurrence to employees while simultaneous preventing addition accidently material defects from being created.

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NOMENCLATURE

Ryerson	Ryerson Canada Inc.
Team	Team 19
Client	Ryerson Canada Inc.
CAD	Computer Aided Design
BOM	Bill of Materials
PLC	Programmable Logic Controller
QFD	Quality Function Deployment
WBS	Work Breakdown Structure
CSA	Canadian Standards Association
FOS	Factor of Safety
FMEA	Failure Mode and Effects Analysis
FBD	Free Body Diagram
OEM	Original Equipment Manufacturer
FEA	Finite Element Analysis

SECTION 1 PROJECT DEFINITION

This report has been completed in accordance with the expectations of the Client, Ryerson Canada Inc. and the requirements of MECH 4860: Engineering Design in mind. The report is structured around a systematic approach to engineering which will provide the Client with the ground work to design, build, test, and validate an automated sheet transfer system to be used in metal distribution facilities across the world.

1.1 Project Background

Team 19's Engineering Design Project was sponsored by Ryerson Canada Inc. Ryerson is one of the world's largest metal processers and metal distributors in the industry [1]. Ryerson has an extensive portfolio of products and has the capacity for state-of-the-art fabrication and processing techniques. Ryerson has asked Team 19 to improve the procedure for filling customer orders. Currently, material is obtained from the facility's storage racks with a side loader which is then moved to a staging table. Once the side loader reaches the staging area, it is raised to the staging table's height where two employees either pick up or slide the required number of sheets onto the staging table for the order. Figure 1 shows the method of filling sheet orders, that has not seen major change or improvement for nearly 100 years aside from improved personal protection equipment [2].

[redacted]

Figure 1: Current sheet metal transfer process [2]

The current method is antiquated, inefficient, and presents a high risk to staff health and safety. Injuries often include repetitive strain injuries and lacerations to arms, hands, and fingers. This procedure also typically requires two operators, and becomes an issue when one employee is unable to make their shift. There is an even greater risk of injury when a warehouse employee is absent from their shift and the sheets must be moved by only one employee. To minimize the danger to the lone employee, an employee from another area of the warehouse is required to help with the sheet lifting. Additionally, the sheets can be scratched when one sheet is dragged over the next. This creates an inventory of what Ryerson classifies as secondary material, which is damaged inventory material that must be sold below prime market value.

1.2 Objectives

The objective of the project is to improve Ryerson's sheet metal transfer process by providing a detailed design of a metal sheet transfer system to assist in the transfer process. Team 19's objective is to design a machine to improve the efficiency of the transfer process by reducing the number of operators required to one. The reduction of required operators will be accomplished by automating the sheet transfer process while maintaining the current capacity of the process and simultaneously minimizing health and safety accidents.

The team must provide the following deliverables to Ryerson [2]:

1. Preliminary load and stress calculations
2. Original 3D CAD models made in Solidworks
3. Sub-assembly and detail part drawings
4. Detailed bill of materials

The preliminary load and stress analysis calculations will be used to determine the viability of the design. The 3D CAD models will be used to evaluate the design's footprint in the facility. The Team will deliver all listed deliverables and all detailed design documents to the customers by December 7th, 2016 [3].

1.3 Scope of Work

The Team will provide a detailed design for a mechanical system to move sheet material from a stock pile located on a side loader and transfer it to a packaging pallet. The detailed design will include the mechanical design of the sheet metal transfer machine with all required support structures and transfer mechanisms [2]. All detailed design parts and assemblies will be accompanied by preliminary stress and load calculations in imperial units. The team will also provide the Client with preliminary engineering drawings for the design parts and assemblies.

The following items will be excluded from the scope: routing of hydraulic, electrical and pneumatic systems, selection of PLC controller, motors and motor controllers, the integration of an automated skid strapping device, integration of built in scales and sheet counters, and the modification of current cantilever beam racking systems. This scope was selected based on an expected workload for only three team members.

1.4 Stakeholders

After consulting with the Client, the Team identified three groups of stakeholders as detailed in TABLE I.

TABLE I: LIST OF STAKEHOLDERS

Group	Stakeholder	Name
Project Sponsor	Ryerson Canada Inc.	Scott Eardley
	Future Project Lead	Dominic Casar
	Ryerson Canada Operators	Various
University of Manitoba	Course Co-Ordinator	Paul Labossiere
	Faculty Advisor	Vern Campbell
	University of Manitoba	Faculty of Engineering
Team 19	Team 19 Members	David Dy Andrew Kerr Yong Lee

By utilizing the CSA Z432-04: Safeguarding of Machinery standard, Ryerson Canada employees can be further broken down in accordance with Section 5.3.2.1: Information for Risk Assessment [4]. The Team determined the Ryerson Canada Operators will include:

1. Machine operators and helpers;
2. Maintenance individuals;
3. Technicians;
4. Installation and removal personnel;
5. Trainees;
6. Passers-by, as defined in CSA Z432-04;
7. Safety personnel; and
8. Safety consultants.

The Team include the above stakeholders due to their function relative to the design, development, implementation, and operation of the sheet metal transfer system.

1.5 Needs And Target Specifications

To develop targets, the Team first identified the core customer needs. These needs were then ranked to provide a qualitative measure of their relative importance. The Team then developed specific metrics to capture these needs in a quantitative manner using a Quality Function Deployment matrix. The results of the QFD, in combination with recommendations of the Client, were used to develop the target specifications.

1.5.1 Customer Needs

The Team identified a list of customer needs based on three site visits, as well as an information package provided to by the Client. During the three site visits, multiple interviews were conducted with the Team's main Ryerson contacts, Scott Eardley and Dominic Casar. The Team also interviewed employees from the warehouse who may interact with the machine on a daily basis. These site visits included two tours of the warehouse to document the current procedures used for moving sheet material.

The Team ranked the needs from 1 to 5 based on the perceived importance. Five was the rank of highest importance. Afterwards, Ryerson Canada also ranked the needs independently. The two rankings provided a comparison of viewpoints, as well as a quick evaluation of the team's understanding of the problem. This list is detailed in TABLE II.

TABLE II: CATEGORIZED AND RANKED LIST OF CUSTOMER NEEDS

Category	Customer Need	Team Rank	Client Rank
Safety	The machine can be operated safely.	5	5
	The machine is safe to those in its surroundings.	5	5
Morale	The machine reduces physical exertion on workers.	3	5
	The machine is easy to operate.	3	4
Quality	The machine moves sheet metal from one pallet to another.	5	5
	The machine maintains the surface finish of the materials.	4	4
	The machine's operations are accurate and repeatable.	4	4
	The machine facilitates individual sheet separation.	4	5
	The machine facilitates paper insert removal.	2	4
Efficiency	The machine can be operated by one operator.	4	5
	The machine operates quickly.	4	3
Robustness	The machine can handle a range of sheet materials and thicknesses.	4	3
	The machine can handle a range of order sizes.	4	2
	The machine is suited for a range of workspaces.	3	2
	The machine's design facilitates future desired features.	3	3
	The machine can handle different sheet sizes.	2	5
	The machine can handle different loading methods.	2	3
	The machine operates normally after repeated collisions from loader traffic.	4	4
Cost	The machine efficiently uses the available space.	4	3
	The machine requires minimal maintenance.	3	3
	The machine is affordable to purchase for a small sheet metal supplier.	2	2
	The machine is affordable to operate.	2	3
	The machine is easy to install.	2	4
	The machine is easy to service.	3	3

These needs were organized into six different categories shown on the left-hand side of the table. The categories include safety, worker morale, quality, efficiency, robustness, and cost. By grouping the customer needs, we ensured that these six defining areas of the problem were adequately represented in the list of needs.

The customer ranking was used as the true weighting for analysis and to help the Team determine is most valuable to the customer. The highest weighted needs are highlighted in yellow. The highest priority needs for the machine can be summarized as the machine needs to be safe, it must be operable by one worker, and it must be able to move a wide variety of sheet sizes.

1.5.2 Constraints and Limitations

Team 19 identified four constraints related to cost, size, and schedule. The first identified constraint was cost. Cost was considered a constraint for the team due to The Client providing a topline budget between $15 000 and $25 000 [5]. The second and third constraints are in regards to the size the designed equipment. The nominal footprint size must be a consistent 140" wide, and its dimension when stored cannot exceed 40" when retracted. The final constraint was not set by the Client but by the course, as the final report must be submitted by December 7th, 2016 [3]. A description of the four constraints identified and how Team 19 plans to adhere to each constraint can be found in TABLE III.

TABLE III: PROJECT CONSTRAINTS AND LIMITATIONS

Constraint	Value	Description	Plan to Adhere
Cost	$25 000 [5]	The basic machine elements for transferring sheet metal from a side loader to a pallet should not exceed $25 000.	The team will keep a detailed budget of all purchased components and raw materials then raise concerns when they arise.
Footprint (In-Use)	140” Wide [5]	The machine must not exceed 140” in width.	Each designer will have a clear understanding of this constraint prior to any design work commences.
Footprint (Stored)	80” Retracted Depth [5]	When the machine is in its stored and retracted position, it must not exceed 80” in depth.	Each designer will have a clear understanding of this constraint prior to any design work commences.
Schedule	December 8, 2016 [3]	All design documents and reports will be provided to the Client and the University no later than December 8, 2016.	Weekly meetings with all team members, the advisor, and client will occur to ensure progress is being made and the team is on schedule.

As the Team completed the problem definition phase of the project the identified constraints were in constant consideration to ensure the Client was satisfied with the final product.

1.5.3 Target Specifications

The insight provided by the QFD analysis and input from the Client were combined to develop the target specifications. TABLE IV defines each metric in detail and categorizes the metrics in a similar structure to the customer needs. The priority column is based on the values calculated in the QFD.

TABLE IV: TARGET SPECIFICATIONS

Category	Metric	Description of Metric	Priority	Target	Unit	Justification for Target
Safety	Worker Contact Time Per Sheet	The time each worker is in direct contact with each piece of sheet material moved.	-100	< 1	seconds	We expect potential minor manual adjustments. Currently the contact time for one sheet is around 5s to 30s depending on the job and number of workers involved.
	New Hazards to Workplace	Number of new hazards introduced to the workspace with the new equipment or procedure.	-88	1	each	One new hazard includes the hazard zone created by main automated sheet metal mover components. No other new hazards such as excessive noise, increased likelyhood of crushing or lacerations, etc.
	Encroachment into Traffic Area	Maximum distance into an area of the shop that is currently a high traffic area.	-53	0.8	meters	Based on a maximum retracted depth constraint of 80" assuming the machine is partially located within their current stock material shelving system.
Robustness	Pre-Shift Check Time	The amount of time required to perform the pre-shift maintenance check.	-41	5	minutes per shift	Typical for equivalent machinery at Ryerson Canada warehouse.
	Initial Sheet Position Variance	The maximum deviation from the ideal location of the sheet.	8	13	mm	Assumed this metric was strongly correlated to final sheet position.
	Loading Methods	The number of different methods the machine can be presented with material.	-5	1	each method	The desired function is to take the sheet from a pallet held by a side loader. Not a high priority.
	Adaptability for Future Features	The number of future features that were accomodated after this first iteration.	11	1	each feature	Will likely account for future addition of more advanced automation controls. Other features will be considered, but they are not high priority.
	Different Order Sizes	The ideal number of sheets it is to move for one order.	14	1 to 10	sheets per order	Machine's primary function will be to replace the manual material moving.
	Handled Materials	The number of different materials it is capable of moving.	15	4	types	This is all the sheet material types stocked by the Ryerson Canada warehouse in Winnipeg. Ie. Carbon steel, stainless, aluminum, nickel.
	Handled Sheet Sizes	The range of sheet sizes that the machine will be able to move.	26	$T \leq 0.5$ $36 \leq W \leq 72$ $96 \leq L \leq 144$	inches	T = thickness, W = width, L = length These are the maximum/minimum dimensions of a sheet that could be moved by this machine.
Quality	Final Sheet Position Variance	The maximum deviation from the ideal placement location of the sheet.	-42	25	mm	Estimated reasonable deviation to correct manually prior to banding the material.
	Maximum sheet droop	Maximum amount of deflection in a sheet caused by it's own weight while it is lifted.	-18	50	mm	Based on the recommended maximum amount of deflection from Anver, the vacuum material handling specialists.
Morale	Unsuccessful Sheet Moves	Any instance where the normal movement of a sheet is interrupted or completed incorrectly.	-86	0.0001 - 0.02	%	Assumed 1 in 5000 sheets has minor problem, 1 in 1 million sheets has major problem is reasonable for preliminary design.
	Operating Steps for a Single Sheet	The number of sub tasks required by the operator to move a sheet.	-50	3	each	Current manual process can be broken into 3-5 steps if two workers are involved.

Category	Metric	Description of Metric	Priority	Target	Unit	Justification for Target
Morale	Job setup time	The length of time required to pick up the first sheet of material once the skids are in place.	-33	1.5	minutes	Reasonable amount of time compared with the current methods of moving sheet metal for orders.
Efficiency	Number of Operators	The minimum number of workers required to move sheet material with the machine.	-71	1	workers	Reducing the number of workers required to move a sheet was highly desirable for the Client. This was reinforced by the QFD analysis.
	Task Time Per Sheet	The average time required to move one sheet of material.	-26	5	seconds	Comparable to average speed at which two workers could safely move one smaller sheet.
Cost	Annual Costs	Annual cost to maintain and run the machine excluding manhours.	-73	$ 2,000	CAD$	power (15kW, 8hrs a day) + maintenance ($500)
	Installation Cost	Cost of installation including labour.	-38	$ 2,000	CAD$	Assumed one day with 2 workers, plus opportunity cost and an electrician.
	Steps for Installation	Major sub-tasks required to install.	-32	10	steps	Estimated major sub-tasks required for installation for initial concept from Ryerson.
	Total Footprint	Total area occpied by the machine.	-14	78	square feet	Estimate based on the maximum footprint of the machine when not in use set by the constraints.
	Steps for Servicing Access	Major sub-tasks required to service the internal components.	-18	2	steps	Assumed access requires lockout/tagout and the removal of a panel for most service applications.
	Capital Cost	Total initial cost to purchase the machine.	38	$15,000	CAD$	Estimate provided by the Client and seems reasonable.

Safety metrics had the highest priority, and therefore had the most aggressive targets. The most useful metrics included worker contact time and new hazards to workplace because they are both representative of safety and easy to measure. The target specifications for handled material types and sizes were focused only on materials that are regularly sold at the Ryerson Canada Winnipeg location. The Team understood that attempting to accommodate a larger range of material sizes could have detrimental effects on other metrics of the machine performance. The unsuccessful sheet moves metric was ranked relatively highly because it has some correlation to safety and potential damage to sheets. There was also strong justification for reducing both the number of workers required to operate the machine and the annual cost.

SECTION 2 CONCEPTUAL DESIGN

With the identification of the customer's needs and relevant target specifications for the project, the Team initiated the concept generation. We began by researching existing solutions for the problem through patents, as well as products already available on the market. The Team also brainstormed new ways of innovating existing technology. For the initial concept research and generation, the machine was decomposed into its core functions. These key functions were referred to as the critical components. These components were:

1. Control Mechanisms – how the machine will be controlled by an operator
2. Grab Methods – how the machine will grab individual sheets off of the unload/supply pallet
3. Lift Methods – how the machine will lift individual sheets off of the unload/supply pallet
4. Machine Movement – how the machine will move to transfer the sheet to the packaging pallet
5. Sheet Movement – how individual sheets will move from the unload/supply pallet to the packaging pallet
6. Support Structure – how the machine will be supported
7. Sheet Separation – how the machine will prevent the sheets from sticking together as they are lifted
8. Drive System – how the machine will move the sheets
9. Skid Alignment – how skids will be aligned to ensure the process is repeatable

10. Sheet Stick Detection – how the machine will determine if more than one sheet is being moved
11. Skid Holding Method – how the skids will be held in place to ensure the process is accurate and repeatable

The Team devised a list of concepts for each of these machine components from the research and brainstorming. An extensive screening and scoring process was applied to these concepts. Additionally, we iteratively combined and expanded on the concepts throughout the process.

Eventually, the component concepts were developed into a small number of full machine concepts. At this time the Team explored the feasibility of the most promising designs in close detail. This feasibility study consisted of a close evaluation of the concepts' limitations, preliminary stress calculations, a cost estimate, and a prediction of the expected performance specifications.

2.1 External Research

Before researching components for the identified functions, the team performed an external search for documents that related to designs featuring transfer systems to accommodate moving sheet metal from one skid to another. The sources of external search included competitor's products, patents, technical journals and standards.

The Team determined from the external research that there are numerous competitors in the market that currently utilize a mechanical vacuum head system that operates by creating a vacuum to grab and lift the material with suction cups. However, the details of the design such as the support structure, separation mechanism and sheet stick detection method varied from one system to another. TABLE V shows the list of major sheet metal handling machine manufactures and distributors.

TABLE V: COMMONLY USED SHEET HANDLING DEVICES

Company	Features	Illustrations
AP&T	• Fully automated station [6] • Grab: vacuum head • Support structure: frame on floor • Controls: terminal • Drive method: motorized • Machine movement: roll on rail • Sheet separators • Max sheet size: 98.4”x177.1” • Min sheet size: 5.9”x11.8” • Thickness: 0.02” to 0.30” • Weight capacity: 110 lbs	[redacted] [6]
Vacuum Lifter Automation	• Fully automated station [7] • Grab: vacuum head • Support structure: frame on floor/ceiling • Controls: manual or terminal • Drive method: motorized • Machine movement: conveyors (motor/belt) • Sheet separators	[redacted] [7]
Ergonomic Partners	• Grab: vacuum head [8] • Support structure: frame on floor • Machine movement: swing arm • Drive method: piston • Controls: manual • Weight capacity: 4400 lbs	[redacted] [8]

The Team discovered through the completed research that fully automated sheet metal transfer stations (from AP&T and Vacuum Lifter Automation) are used in facilities where frequent loading and unloading of materials need to occur. These stations are fixed in one location either on floor or ceiling and contain loading and unloading zones [6], [7]. In most automated systems, when a sheet separation mechanism is implemented, it is typically located in the loading zone. Once the material is placed in the loading zone, the automated lifter enters the loading zone, picks up the material with the vacuum head, and moves it to the unloading zone to complete the transfer process. The fully automated transfer system is the ideal existing solution for the problem at hand. However, the downside with it is the massive footprint of the machinery, as well as the cost constraint.

Another set of pre-existing designs the Team investigated was manually driven sheet metal transfer systems. As shown in TABLE V, a manual system incorporates a handle or switch which is attached to the vacuum lifter for controlling the machine movements [8]. Once the material is ready to be lifted at its initial location, an operator brings the vacuum lifter to the location to manually adjust the height of the lifter, which is followed by the operator bringing the lifter to the desired height and guiding it to the unloading zone. After the material is brought to the designated location, the operator drops the material using the controls. To aid the manual handling, these types of designs usually contain additional drive mechanisms such as hydraulic or pneumatic pistons or motorized hoist systems. The advantage of the design is in its simplicity which means that it will achieve the objective of the project while keeping the cost low. The disadvantage of the design is the manual handling aspect of the design. Since the operator is always in close proximity of the vacuum lifters, there is always a risk of the operator becoming injured due to lacerations and repetitive strain injuries.

Upon the completion of investigating existing products, the Team explored related patents registered in both Canada and United States of America. The patent search was to research design solutions and to determine what aspects of the potential designs is patentable. During the research, the Team discovered that the sheet metal transfer system was commonly used in the industry. In numerous patent documents, the system was incorporated into existing machinery designs to satisfy different functional requirements. The Team identified from the research three patents which solely dealt with the sheet metal transfer system. TABLE VI shows the details of these patents including the operating mechanisms.

TABLE VI: SUMMARY OF PATENTS DESCRIBING THE OPERATING MECHANISMS

Canadian Patent	Patent Title	Transfer Mechanisms
CA 1180360	A Sheet Transfer Apparatus	1. Material is picked up by vacuum lifts [9]. 2. Vacuum lift is shifted to move the material to the conveyer. 3. The conveyer transports the material from one location to another. 4. Two longitudinally spaced bars which are controlled by a push button provides pressure onto the material at the end of the conveyer to drop the material from the conveyer to unloading zone.

American Patents	Patent Title	Transfer Mechanisms
CA 1254917	Sheet Transfer Device and Method	1. Vacuum heads grab the material [10]. 2. Base unit provides pivoting action of a beam to separate the material from one another and to lift the material above the unloading skid. 3. Once suction is terminated from the vacuum heads, the material is dropped onto the skid to finish the transfer.
US 6746063B1	Device for separating, lifting and moving aluminium sheets or other non-ferromagnetic material	1. Magnetic field separates the top sheet from the rest of the material [11]. 2. Robot arm carries the vacuum lift to the loading zone. 3. Robot arm moves the vacuum lift downwards to grab the material and upwards to lift it. 4. Robot arm carries material to the destination location.

The Team decided not to investigate the patents any further due to the designs not possessing the core functionality that the project requires, not meeting the design constraints, or simply the patent being too general to accurately interpret. Patent CA 1180360 did not contain enough details to achieve the separation of sheets and the integration of the listed mechanisms into a single system. The Team did not pursue the exploration of CA 1254917 because the design did not specify how the pivoting action will lift the material to a certain height without eliminating the risk of the material falling down. Also, the use of pivoting mechanism would require a large footprint which would not meet the design's constraint. Finally, patent US 6746063B1 was not applicable for the project. The design was solely aimed for aluminum and non-ferrous material which is not usable on ferrous materials such as steel.

2.1.1 Machine Component Research

While researching the overall system designs, the need for looking into the specific components arose since the current designs did not encompass all of the potential alternatives. The components for the general functions such as the support structure, machine movement method and grabbing and lifting mechanisms were researched briefly as the Team was unfamiliar with them. An in-depth investigation to achieve sheet separation and sheet stick detection was required. In addition, an assessment of the vacuum heads was required to understand the differences between varying products and model lines.

2.1.1.1 Sheet Separation Mechanism

The Client, Ryerson, handles different sizes of sheet metals varying from 3' x 8' to 6' x 12', where 4' x 8' and 8' x 10' are the vast majority of the orders [5]. The problems with larger sheet metals is that sheets occasionally stick together during the material handling process [12]. This leads to sheet not in contact with the vacuum to not be grabbed properly, thus creating a hazard to operators as well as potentially damaging other stock material below. To prevent these causes of concern, a sheet separation mechanism requires incorporation into the sheet transfer system to ensure that the new system does not create new problems. TABLE VII summarizes the list of methods that are typically used in the industry.

TABLE VII: COMMONLY USED SHEET SEPARATING MECHANISMS [12]

Sheet Separation Mechanism	Description
Manual	Operator manually separates sheets.
Magnetism	Closely placed magnets create magnetic fields to separate the sheets from each other. Not applicable for non-ferrous materials.
Strategic peeling	Interface inserted between the material to peel off top sheet.
Air knife	Device shoots compressed air toward the sheets to life the top sheet to achieve separation.
Vibration	Device applies vibration on the stack of sheets through a short span of time to separate them.
Mix of Magnetism and Air Knife	Magnetic field separates ferrous sheets while air knife separates non-ferrous sheets [13].

From all of the options presented in TABLE VII, the air knife and magnetism are the two separation methods that are frequently utilized [14]. However, due to magnetism only being viable on ferrous materials, an air knife is can additionally be utilized handle non-ferrous material [13]. By combining the two concepts, magnetism to automatically separate ferrous, and utilizing an air knife with non-ferrous materials, a viable solution is present. The method reduces operation cost as the major part of the separation process is processed spontaneously. Methods such as vibration and strategic peeling were not assessed due the mechanisms being likely to damage the sheets while achieving the same functionality as the other options.

2.1.1.2 Sheet Sticking Detection

Through researching the separation system, the Team discovered that risks of the materials sticking together always exist even with the implementation of automated sheet separation system [14]. To prevent and eliminate the issue, industries utilize sticking detection systems. The results of the research are presented in TABLE VIII.

TABLE VIII: COMMONLY USED DOUBLE SHEET DETECTION MECHANISM [14]

Sticking Detection Mechanism	**Description**
Visual inspection	Operator manually checks if only the top sheet is lifted.
Laser	Laser sensor is located at a specific height depending on material thickness. If two sheets are lifted, the laser will hit the sheets instead of the other end of the transfer station to detect it.
Ultrasonic	Two ultrasonic sensors are located across from each other on the opposing sides of the sheet metals [15]. As the wave signals pass through the sheets, the amplitude decreases. The results are picked up by the sensors to distinguish if the wave signals are passing through one or more number of sheets.
Weight	The weight of the material lifted is compared with the database.
Inductance	Sensors utilize magnetic fields for ferrous materials and eddy currents for non-ferrous materials [16]. When multiple sheets are lifted, the magnetic field or eddy currents will be weakened in-between the sheets which are picked up by the sensors.

The most commonly used mechanism is inductance since both ferrous and non-ferrous sheets can be handled, the process is quick and the device itself is small [14]. The ultrasonic method has the same advantages; however, it is only suitable for thin gauges [15].

2.1.1.3 Vacuum Lifter

Preliminary research of the components revealed that there were varieties of vacuum heads in the market with different loading capacities and material handling capabilities, which needed to be thoroughly researched. Upon investigation, we identified that the classification of the systems and the corresponding specifications were based on the power source. The specifications of the vacuum heads are presented in TABLE IX.

TABLE IX: TYPES OF VACUUM LIFTER UNITS [17]

Systems	**Weight Capacity**	**Applications**	**Movements**
Mechanical Self Powered Units	18 000 lb	Only for sheet and plate	Horizontal
Electrical Powered Units	60 000 lb	Most materials; better for non-porous and semi-porous materials	Horizontal, vertical, tilting, rotation
Battery Powered Units	5 000 lb	Mostly for glass	Not specified
Air Powered Units	5 000 lb	Most materials; better for non-porous and semi-porous materials	Horizontal, vertical, tilting, rotation

All of the identified systems excluding the battery powered units are viable for the project specific application. They all satisfy the maximum weight capacity of 2000lbs. while being suitable for sheet metals.

2.1.2 Applicable Standards

Research into manufacturing standards revealed that there are four major standards that the design is required to comply with [18]. The following is a list of the four CSA standards the Team identified as relevant for the environment the product can be implemented in:

1. Z432 Safeguarding of Machinery
2. Z142 Code for Power Press Operation: Health, Safety & Safeguarding Requirements
3. Z434 Industrial Robots and Robot Systems – General Safety Requirements
4. Z460 Control of Hazardous Energy – Lockout and Other Methods

From initial review of the standards, the Team realized that CSA Z142 and CSA Z460 were not directly applicable to the project. CSA Z142 was entirely focused on power press, which was too specific to utilize for the project. On the contrary, the topics covered in CSA Z460 were too general, which meant the contents of the document were already covered in other CSA documents. Consequently, only CSA Z432 and CSA Z434 were analyzed in detail to see the design requirements. The overview of the relevant standards is shown in TABLE X. The descriptions show the categories of the standards that will be utilized to ensure the designs comply with regulations.

TABLE X: SUMMARY OF APPLICABLE STANDARDS FROM CSA Z432-04 AND CSA Z434-03

	Standards	
	CSA Z432-04 [4]	**CSA Z434-03** [19]
Descriptions	Risk assessment and reduction	Design requirements for robots
	Hazardous environment	Mechanical requirements
	Safety Device designs	Electrical requirements
	Emergency stop device	Information label
	Ergonomics	Control system

The Team assed the concepts and final designs accordingly to the standards such that the machinery components meet the design requirements stated in CSA Z434-03 while all of the components are safely guarded via following CSA Z432-04.

2.1.3 Summary of Research

Through investigating pre-existing designs, patents and specific component designs, the Team explored alternative methods of achieving the overall project objective. Although the Team did not utilize the researched system designs completely, both results from competitor product research and patents demonstrated potential design solutions for the decomposed functionalities. The results of the research are shown in TABLE XI.

TABLE XI: SUMMARY OF RESEARCHED COMPONENTS

Function	Discovered Components
Support Structure	Frame on floor, frame on ceiling
Machine Movement Mechanism	Roll on rail, conveyer belt, swing arm
Drive Method	Motor, manual
Controls	Terminal, handle, switch
Grabbing Mechanism	Vacuum head
Sheet Separation Mechanism	Magnetic field generator, pivot action
Lifting Mechanism	Pivot action, piston
Sticking Detection Mechanism	Not Applicable
Skid Alignment and Holding	Fully automated unloading zone

By completing external research, the Team identified the most commonly used components in the industry to achieve the functionality of transferring sheet metals from one skid to another. The discovered components listed in TABLE XI were utilized and expanded through brainstorming to develop refined and more specific design concepts.

2.2 Preliminary Concept Generation

In conjunction with external research, the Team performed internal brainstorming to develop design concepts. TABLE XII shows the list of concepts generated through internal brainstorming and expansion of researched concepts for each system.

TABLE XII: CONCEPTS GENERATED FOR EACH SYSTEM/COMPONENT

System/Component	Concepts
Control Mechanism	1. By hand
	2. Machine mounted hand controlled dashboard
	3. Machine mounted foot controlled dashboard
	4. Machined mounted at point of use
	5. Handle (manual)
	6. Chain controls (manual)
	7. Wired foot control removed
	8. Wired hand control removed
	9. Wireless remote controls
	10. Computer controlled
	11. Sensor controlled
	12. Cellphone app controlled
Grab Methods	1. By hand
	2. Vacuum head
	3. Magnets
	4. Clamps
	5. Hooks
	6. Adhesive
	7. Attachable handles
	8. Sling
	9. Straps
Lift Methods	1. By hand
	2. Wedge and roller
	3. Pneumatic piston
	4. Hydraulic piston
	5. Winch
	6. Hoist
	7. Linear actuator
	8. Air bag

System/Component	**Concepts**
Lift Methods	9. Air cushion
	10. Robotic arm
	11. Counter weighted arm
	12. Motor assisted arm
	13. Hydraulic assisted arm
	14. Pneumatic assisted arm
Machine Movement	1. Slide on rail
	2. Swing arm
	3. Roll on ground
	4. Swing on vertical chain
	5. Sliding on horizontal cable
	6. Automated guided vehicles
	7. Robot arm
	8. Manual lift
Sheet Movement	1. Manual lift and slide
	2. Push
	3. Pull
	4. Launch
	5. Slide over other sheets
	6. Gravity feed
	7. Rollers
	8. Air cushion
	9. Conveyor
	10. Pivot
	11. Vibration
Support Structure	1. Cantilever racking
	2. Ceiling mounted frame
	3. Ceiling mounted cables/chains
	4. Wall mounted to building
	5. Side loader mounted
	6. Frame on floor
	7. Packaging table mounted
	8. Skid mounted
	9. Sheet mounted
	10. Exoskeleton
Sheet Separation	1. Air knife
	2. Wedge
	3. Vibration
	4. Strategic lifting (corner)
	5. Strategic lifting (wave)
	6. Skew stack
	7. Magnetic
	8. By hand
	9. Impact

System/Component	Concepts
Drive System	1. By hand
	2. Magnets
	3. Hydraulic piston
	4. Pneumatic piston
	5. Geared motor
	6. Motor driven chain
	7. Motor driven belt
	8. Motor driven lead screw
	9. Winch
	10. Pulley
	11. Wind-up spring
Skid Alignment	1. Visual inspection
	2. Mechanical stops, non-adjustable
	3. Mechanical stops, flip up/down

With the concepts listed in TABLE XII, the SCAMPER brainstorming technique, as defined in TABLE XIII, was used.

TABLE XIII: PRINCIPLES OF SCAMPER TECHNIQUE [20]

Variable	Expanded Name	Questions to Ponder
S	Substitute	Can you substitute components?
C	Combine	Can you mix, combine, or integrate components?
A	Adapt	Can you alter, change the function, or use it differently?
M	Modify	Can you change the component scale, shape, or other attributes?
P	Put to other use	Can you use the component differently?
E	Eliminate	Can you remove the component to simplify or reduce core functionality?
R	Reverse	Can you change the component's orientation?

The SCAMPER brainstorming technique was utilized to reduce the number of ideas brought into the concept selection phase. The reduction in the number of concepts brought into the concept selection phase was accomplished by investigating each concept and identifying its strength and weaknesses. The Team simultaneously applied the thought process provided in SCAMPER to further improve concepts and reduce the number of concepts scored.

2.3 Concept Selection

To select the best candidate concept, the Team chose to conduct both concept screening and concept scoring. This process of screening and scoring is an iterative process due to new information becoming available and issues in concepts arising. The team took a systematic approach to concept selection to ensure due diligence was completed and all decisions were justifiable. The Team began the concept selection process with concept screening to compare various concepts to a datum value to determine what designs were better or worse than what is currently utilised at the Ryerson, Winnipeg facility. Once values were assigned to the screening matrix, the number of viable concepts were reduced prior to them being scored in a weighted scoring matrix. The reason the Team chose to reduce the number of concepts scored was to ensure the design space did not become too crowded and prevent repetitive or weak concepts from consuming the Teams attention during the concept selection phase. A feasibility analysis was performed on the best candidate concept to ensure it was viable and able to be pursued by the team.

2.3.1 Concept Screening

For each of the screening matrices, a datum was selected based on the existing sheet metal transfer process of solutions found during the external search completed in the concept generation phase. Every customer need for the datum was assigned a “0” value and each concept was assigned either a “-“ which indicated it performs worse relative to the datum, or a “+” to indicate it performs better than the datum. With the customer need screened, a net score was determined by summing up each concepts score by assigning a -3 to each “-“, a 0 to each “0”, and a 3 to each “+”. With the net score determined, the Team discussed the results to ensure the results made sense and fell in line with the customer’s expectations. Shown in TABLE XIV is the screening matrix for the machine movement system as an example.

TABLE XIV: PUGH CHART FOR MACHINE MOVEMENT

	Concept Screening							
	Datum	**Design 1**	**Design 2**	**Design 3**	**Design 4**	**Design 5**	**Design 6**	**Design 7**
Description	*Slide On eail*	*Swing arm*	*Roll On Ground*	*Swing On Vertical Chain*	*Sliding On Horizonal Cable*	*AGV*	*Robot Arm*	*Manual Lift*
Safe to operate	0	0	-	-	0	-	-	-
Safe to the surroundings	0	-	-	-	0	-	-	0
Transfers metal from one skid to another	0	0	0	-	0	0	0	-
Operated by one operator	0	0	0	0	0	0	0	-
Maintains the material surface finish	0	0	0	-	0	0	0	-
Operations are accurate and repeatable	0	0	-	-	-	+	+	-
Efficient use of space	0	0	-	+	0	-	-	+
Handles a range of sheet materials and thicknesses	0	0	0	0	0	0	0	-
Handle a range of order sizes	0	0	0	0	0	0	0	0
Operates quickly	0	0	-	-	0	+	+	-
Suited for a range of workspaces	0	+	+	+	-	-	-	+
Facilitates future desired features	0	0	0	-	0	0	0	-
Ensures correct shipping quantity	0	0	0	0	0	+	+	0
Reduces physical exertion on workers	0	0	0	0	0	0	0	-
Easy to operate	0	0	0	0	-	-	-	+
Requires minimal maintenance	0	0	0	+	0	-	-	+
Affordable to purchase	0	0	0	+	0	-	-	+
Affordable to operate	0	0	0	0	0	-	-	0
Handles different sizes	0	0	0	0	0	0	0	-
Handles different loading methods	0	0	+	0	0	+	+	-
Durable	0	0	0	0	-	-	-	-
Facilitates individual sheet separation	0	0	0	0	0	0	0	0
Facilitates paper insert removal	0	0	0	0	0	0	0	0
Easy to install	0	0	+	+	-	-	-	+
Easy to service	0	0	0	0	0	-	-	+
+	0	1	3	5	0	4	4	7
0	25	23	17	13	20	10	10	6
-	0	1	5	7	5	11	11	12
Net Score:	0	0	-6	-6	-15	-21	-21	-15

These matrices showed which concepts were potentially stronger than other concepts. The strength of a concept was relative to each of the other concepts explored. The stronger concept was identified with a net score that was higher than other concepts screened. A range of between three and fifteen concepts for each component or system were identified for the next phase, concept scoring. There was a range of concepts taken to scoring, due to the Team feeling some concepts were repetitive or could be consolidated.

By performing the screening process on components, the concepts identified in TABLE XV were determined as the best candidates for scoring. Due to the systematic approach taken by the Team, we felt that the best solution is captured within the reduced component list.

TABLE XV: RESULT OF SCREENING OF COMPONENTS

Component	Concepts
Base Mechanics	Boom Hoist
	Automated Frame with Terminal
	Automated Rack with Remote
	Motorized Hoist
	Manual Frame
	Autmated Rack with Terminal
Vacuum	Electric
	Mechanical
	Battery DC Powreed
	Air Powered
Sheet Separation	Peeling
	Air Knife
	Air Knife and Magnets
Sheet Sticking Detection	Visual Inspection
	Weight
	Inductance
Drive System	Geared Motor
	Motor and Chain
	Motor and Belt
	Direct Motor
	Motor with Lead Screw
Skid Alignment	Mechanical, Not Adjustable
	Mechanical, Inserts
	Skid Pockets
	Markings with Guides
	Markings

2.3.2 Concept Scoring

With the screening process completed, the Team analysed the results and determined various concepts could be grouped together for example hydraulic pistons and pneumatic pistons could be scored together as a single concept titled pistons. The Team also determined that four components, grab, lift, machine movement, and support structure, would make up the core design of the machine. All other components were considered independent of the core mechanics of the machine. With this in mind, two paths were taken the first was to generate the base line mechanics of the system and score them. While the second task was to score the independent parts. For the core mechanics of the system, the team utilized the screening matrices for grab, lift, machine movement, and support structure to determine the most viable concepts. These promising component concepts were combined to develop permutations of the core mechanical design of the machine.

TABLE XVI: CONCEPTS TO PERMUTATE AND COMBINE

Component	Grab Sheet	Lift Sheet	Machine Movement	Support Structure
Concepts	Vacuum head Hooks/clamp Attachable handles	Hoist/winch Wedge and rollers Piston	Slide on rail Swing on arm Roll on ground Swing on vertical chain	Frame on floor Cantilever racking

Using TABLE XVI, a total of 72 possible concepts were generated. The Team discussed all possible combinations and eliminated concepts that contained contradictions, duplications, or did not make sense. This process reduced the options to 32 concepts. These concepts were then distributed between the three team members. Each member sketched a unique interpretation for each of their assigned concepts. The intent was to devise a large number of unique uses for the different component concepts.

Upon the completion of the initial system designs, the Team conducted a quick screening of the 32 conceptual designs to determine which concepts were more competitive than the others. The major factors the Team looked into were how suitable they were for the project and the viability of the designs based on the information gathered from external research. Since the Team generated a total of 32 concepts, even when the overall design of the concept was not realistic, the Team could salvage creative ways of achieving the desired functionalities. At the end of the process, the Team obtained a total of five viable combinations which were identified as the following:

1. Hoist on swing arm system installed at floor
2. Piston on sliding on rail system installed at cantilever racking
3. Piston on sliding on rail system installed at floor
4. Hoist on sliding on rail system installed at floor

Based on the selected concepts, the Team added machine drive and control methods that could be implemented mechanically into the concepts with ease. The resulting conceptual designs are listed below.

1. Boom hoist design
2. Motorized rack design with
    a. Remote control system or,
    b. Machine mounted terminal
3. Motorized frame design
4. Motorized hoist design
5. Manual frame design

The manual hoist system illustrated in Figure 2 is an expansion of the original hoist on swing arm design listed above. A vacuum head is attached to a hoist system by a hook which controls the height of the unit. Once the vacuum head picks up the sheet metal, the operator manually moves the vacuum head towards the unload skid. This design was selected by the Team because it is commonly used throughout the industry; in addition, the design is capable of achieving the same desired features with a minimal cost and little complexity. However, the downside of the concept is the low viability for future automation process.

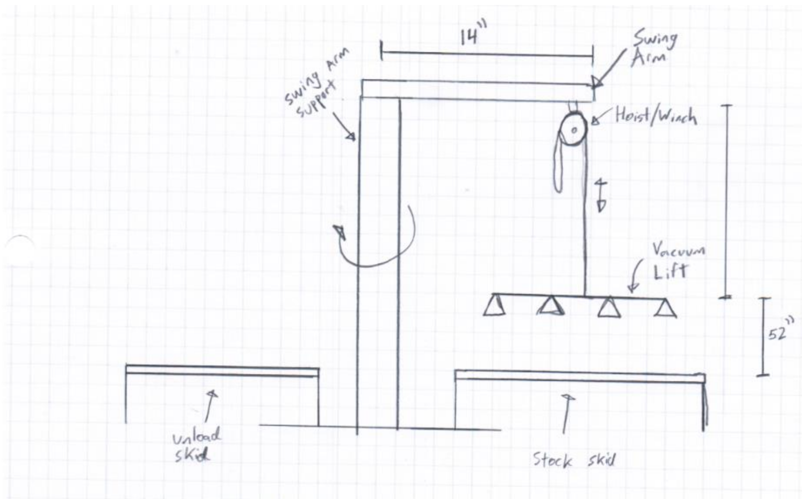


Figure 2: Sketch of the manual hoist design

The motorized rack design has two similar systems with the only difference between them being the control methods. Both designs utilized a vacuum head system carried by a piston which is mounted on a sliding rail system. The advantage of utilizing a cantilever racking system is the small footprint of the entire system when implemented. The basic principles of the design are depicted in Figure 3.

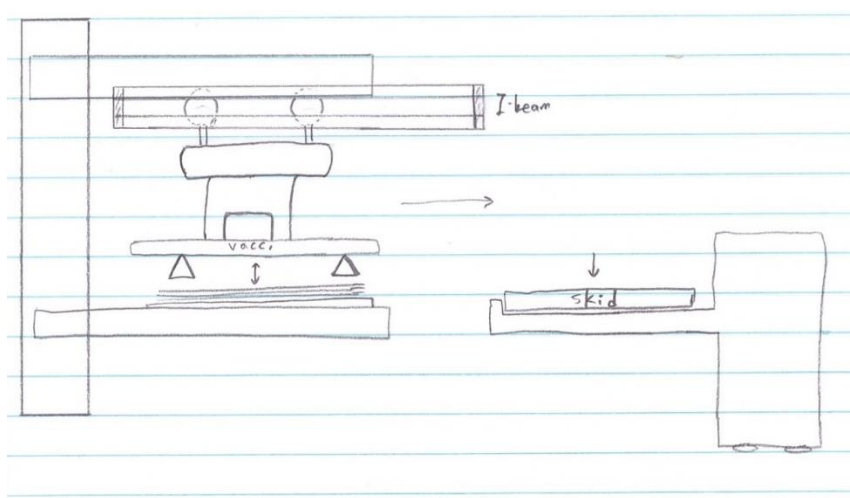


Figure 3: Sketch of the motorized rack design

The motorized frame design shown in Figure 4 is a similar design to the motorized rack design with the only difference being the support structure being installed on floor.

The issue with these support structures are the footprint which could go over the assigned design constraint.

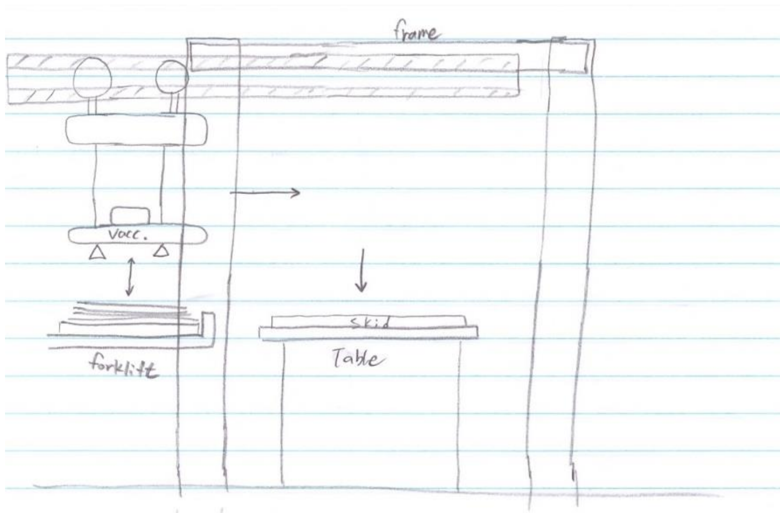


Figure 4: Sketch of the motorized frame design

Figure 5 illustrates the motorized hoist design. The concept is an alteration to the second concept – the motorized rack design. Instead of utilizing a piston system to lift and drop materials, the system uses a motorized hoist to achieve the same function. As with other cantilever systems, the design occupies a small footprint which is an important factor for selection.

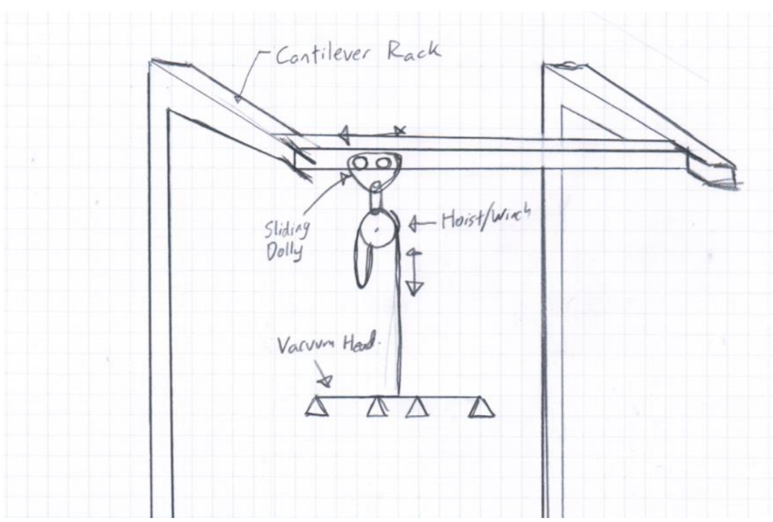


Figure 5: Sketch of the motorized hoist design

The manual frame design depicted in Figure 6 utilizes a handle instead of a motorized drive system shown in Figure 4 to to simply the design. It is less complicated than the motorized frame design; also, due to the nature of the design, it is suitable for adapting future automation projects.

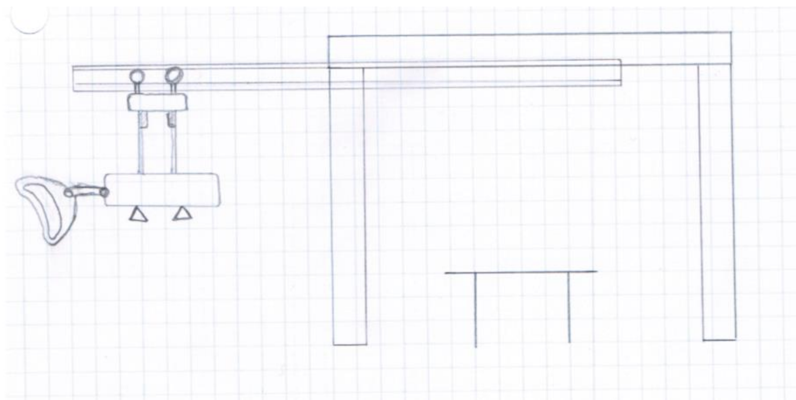


Figure 6: Sketch of the manual frame design

With all screening completed and initial system designs generated, the team moved into scoring the system designs. Each of the components and systems were scored using a weighted decision matrix similar to the QFD. The decision matrix was populate using the customer needs. The weights for each were assigned based on the values provided by the Client. The weighted decision matrix for the selection of the base machine mechanics can be seen in TABLE XVII.

TABLE XVII: SCORING OF BASE MACHINE SYSTEM

Description	Weight	C6. Motorized, Rack w/ Terminal		C2. Motorized, Frame w/ Terminal		C3. Motorized, Rack w/ Remote		C1. Manual, Boom Hoist	
		Rating	Score	Rating	Score	Rating	Score	Rating	Score
Low inertia	5	O	5	O	5	O	5	OOO	45
Pinch Points	5	OO	15	OO	15	OO	15	O	5
Low potential for overhead risks	5	OO	15	OO	15		0	O	5
Low potential of risk to people in the area	5	OO	15	OO	15	OO	15	O	5
Operated by one operator	5	OO	15	OO	15	OO	15	O	5
Handles a range of sizes	5	O	5	O	5	O	5		0
Reduces physical exertion on workers	5	OOO	45	OOO	45	OOO	45	OO	15
Maintains the material surface finish	4	OOO	36	OOO	36	OOO	36	OO	12
Operations are accurate and repeatable	4	OOO	36	OOO	36	OOO	36	O	4
Easy to operate	4	OO	12	OO	12	OO	12	OO	12
Reliablity	4	O	4	O	4	O	4	OO	12
Resistance to bumps and collisions	4	OO	12	OO	12	OO	12	O	4
Facilitates individual sheet separation	4	O	4	O	4	OO	12	OO	12
Easy to install	4	O	4		0	OO	12	OOO	36
Efficient use of space	3	OO	9		0	OO	9	OO	9
Handles a range of thicnkesses	3	OO	9	OO	9	OO	9	O	3
Affordable to operate	3	OOO	27	OOO	27	OO	9	OO	9
Can integrate autostrapping into the system	3	O	3	OO	9	O	3	O	3
Facilitates paper insert removal	3	O	3	O	3	OO	9	OO	9
Operates quickly while transfering sheets	3	OOO	27	OOO	27	OOO	27	OO	9
Machine setup time	3	O	3	O	3	O	3	OOO	27
Can integrate automation into the system	3	OOO	27	OOO	27	OOO	27	O	3
Ease of maintenance	3	O	3	O	3	O	3	OOO	27
Handle a range of order sizes	2		0		0		0	O	2
Ensures correct shipping quantity	2	O	2	O	2	OO	6	OO	6
Versitale for various workspaces	2	O	2	OO	6	O	2	OOO	18
Low cost to implement	2	O	2	O	2	O	2	OOO	18
Total Score:		**385**		**382**		**338**		**320**	
		0%		**-1%**		**-12%**		**-17%**	

For each weighted decision matrix, a symbol was placed into the raw score that corresponded to a number that introduced an exponential scale to help exaggerate the total scores determined for each concept. A legend for the symbols used in each matrix can be found in TABLE XVIII.

TABLE XVIII: SCORING VALUES USED FOR THE SCORING MATRIX

Symbol	Value
	0
O	1
OO	3
OOO	9

If the weighted decision matrix yielded a result which was unexpected a 3-color gradient was applied to each row and column independently to highlight the high and low values for each customer need across each concept and how each customer need affected the overall concept. With the color gradient in place the Team conducted a sensitivity analysis on the matrix.

Thorough assessment of the concepts showed that both motorized rack and motorized frame systems with terminal control method were the best solution to achieve base machine movement function. With the best system designs, the Team started analyzing the additional components including the types of vacuum heads, sheet separation mechanisms, sheet sticking detection method, detailed motor drive system and skid alignment method. The summary of the additional components to be scored is shown in TABLE XIX.

TABLE XIX: DETAILED CONCEPTS SCORED

Component/ System	Concept
System Mechanics	Boom Hoist
	Automated Frame with Terminal
	Automated Rack with Remote
	Automated Rack with Terminal
	Manual Frame
	Motorized Hoist
Vacuum Head	Electric Powered
	Mechanical Self Powered
	Battery DC Powered
	Air Powered
Sheet Separation	Peeling
	Air Knife
	Air Knife and Magnet
Sheet Stick Detection	Visual Inspection
	Weight
	Inductance
Drive System	Geared Motor
	Motor Driven Chain
	Motor Driven Belt
	Direct Motor
	Motor Driven Lead Screw
Skid Alignment	Mechanical Stops, Non-adjustable
	Mechanical Stops, Inserts
	Skid Pocket
	Markings and Mechanical Stops
	Visual Markings

After completing the scoring process on each component and it passing the sensitivity analysis, the Team was able to determine the most promising concepts. The best candidate was determined to be an automated frame on floor that has a vacuum lift system attached to sliding rail system articulated using a motor driven belt or chain, that utilizes an air knife to separate the sheets when the weight detected by the machine indicates that more than one sheet is present on the lift. The machine would then transfer the sheets from the unload skid to the packaging skid that was aligned using visual inspection.

2.3.3 Final Concept Selection

The remaining concepts of the selection process were expanded upon prior to deciding the best concept to pursue as the final design. At this stage, the remaining concepts were sketched to scale to capture rough sizes for the machines and to check for interference issues. We also determined the possible locations in the warehouse that these machines could operate. The three locations shown in Figure 7 were deemed as acceptable locations for the machine during site tours earlier in the project.

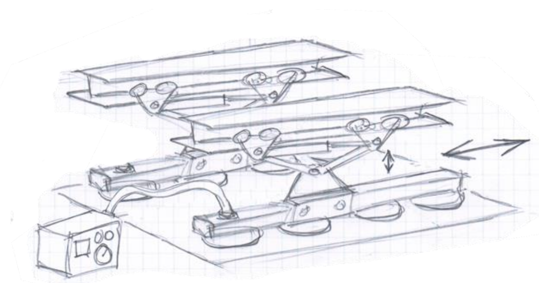
[redacted]

Figure 7: Layout of warehouse and the three considered machine locations [2]

A machine at location A would be underneath a pallet storage cantilever racking system. The machine would be either integrated into the racking system or supported by a dedicated structure. Location B is similar to A in that it is within a cantilever racking storage area. This racking system currently houses secondary material. Machines in either of these locations would consume storage space, but they would not significantly protrude into the loader traffic lanes. Location C would be near the current customer order packing area in the center of the shop floor.

The Team also began a thorough feasibility analysis of the final concepts as the concept selection process neared its completion. The full details of the feasibility study are provided in Section 2.4 of this report. Through this study, we found additional limitations of the vacuum lifter that had a profound impact on the performance of the machine. Specifically, the vacuum lifters are sensitive to large variations in sheet sizes and thickness. It became apparent that the machine would not easily lift every sheet size in our target specification without dramatically increasing cost and complexity. The Team decided to expand on the best options from the previous scoring matrix, shown in TABLE XVII, to explore the effect of compromising different performance specifications such as cost, task time, and the range of handled sheet sizes.

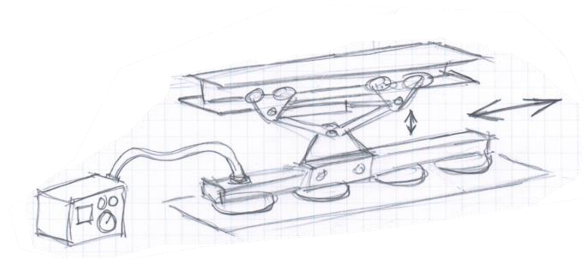
TABLE XX: FOUR FINAL CONCEPTS AND DESCRIPTIONS

<u>Concept CI</u> The figure shows CI without the full support structure. CI was the evolution of concepts C2 and C6 from the previous scoring matrix. This concept compromises complexity and cost to maintain the range of handled sheet sizes. The machine would operate using two separate vacuum lifters on two separate sliding rail systems. They would operate in tandem to lift larger sheets or separately to move smaller sheets. This arrangement reduces the manual adjustments required for the vacuum lifter to carry different sizes of sheets. This concept would be situated in location B of the warehouse.	

<table>
<tr><td><u>Concept CII</u>

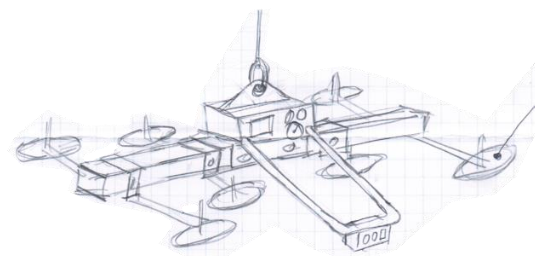
The figure shows CII without the full support structure.

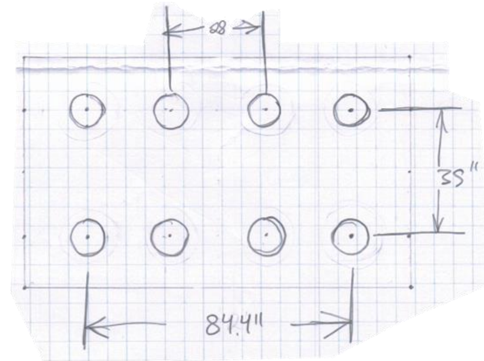
This concept would be the implementation of one of the vacuum lifters from CI. This concept restricts the size of movable sheets to only three or four feet wide. This machine would keep costs and complexity low.

CII would also be situated in location B.</td><td></td></tr>
<tr><td><u>Concept CIII</u>

The figure omits the hoist and swing-arm support structure of CIII.

Concept CIII was an expansion of the manual boom hoist, concept C1, from the previous scoring matrix. This concept requires frequent manual adjustments and manual positioning of the vacuum lifter which is attached to a hoist. This concept sacrifices task time and worker contact time to dramatically reduce cost while maintaining the ability to lift a large range of sheets.

This concept could be placed in any of the three locations of the warehouse; however, it’s small footprint would be most useful in locations 1 and 3.</td><td></td></tr>
</table>

<u>**Concept CIV**</u> The figure shows only the specific pad arrangement for concept CIV. Concept CIV was also an evolution of concepts C2 and C6. This concept was a single vacuum lifter on a motorized rail system. The concept attempted to balance between cost, task time and the range of sheet sizes it could handle. The vacuum lifter was set to an optimized suction pad arrangement that required no adjustment to lift any 4 or 5 foot-wide sheet. These sheets comprise more than 80% of Ryerson's sold material [5]. This concept was designed to be situated in location A.	

These four concepts were scored in the decision matrix shown below. The Team applied the new information about the machine components that was found during the research of the feasibility study.

TABLE XXI: SCORING MATRIX FOR THE FINAL CONCEPT SELECTION

		CIV. Single Rail, Restricted		CII. Half Dual-Rail, Restricted		CIII. Adjustable Vacuum Hoist		CI. Dual-Rail	
Description	Weight	Rating	Score	Rating	Score	Rating	Score	Rating	Score
Low inertia	5	O	5	O	5	OOO	45		0
Pinch Points	5	OO	15	OO	15		0	O	5
Low potential for overhead risks	5	OO	15	OO	15		0	O	5
Low potential of risk to people in the area	5	OO	15	OO	15		0	OO	15
Operated by one operator	5	OO	15	OO	15	O	5	OO	15
Handles a range of sizes	5	OO	15		0	OOO	45	OO	15
Reduces physical exertion on workers	5	OOO	45	OOO	45	OO	15	OOO	45
Maintains the material surface finish	4	OO	12	OO	12	O	4	OO	12
Operations are accurate and repeatable	4	OOO	36	OOO	36	O	4	OO	12
Easy to operate	4	O	4	OO	12	OO	12	O	4
Reliablity	4	OOO	36	OOO	36	OO	12	OO	12
Resistance to bumps and collisions	4	O	4	OO	12	OO	12	O	4
Facilitates individual sheet separation	4	OO	12	OO	12	OO	12	OO	12
Easy to install	4	OO	12	OO	12	OOO	36		0
Efficient use of space	3	OO	9	OO	9	OOO	27	O	3
Handles a range of thicnkesses	3	O	3		0	OOO	27	OO	9
Affordable to operate	3	OO	9	OO	9	OOO	27	O	3
Can integrate autostrapping into the system	3	O	3	O	3	OO	9	O	3
Facilitates paper insert removal	3	O	3	O	3	OO	9	O	3
Operates quickly while transfering sheets	3	OO	9	OO	9		0	O	3
Machine setup time	3	OOO	27	OOO	27		0	OO	9
Can integrate automation into the system	3	OO	9	OO	9		0	OO	9
Ease of maintenance	3	OO	9	OO	9	OO	9		0
Handle a range of order sizes	2	O	2	O	2	OO	6	O	2
Ensures correct shipping quantity	2	O	2	O	2	OO	6	O	2
Versitale for various workspaces	2	OO	6	OO	6	OOO	18	O	2
Low cost to implement	2	OO	6	O	2	OOO	18		0
Total Score:		**383**		**377**		**358**		**249**	
		2%		**0%**		**-5%**		**-34%**	

We found that the two outstanding concepts were CIV and CII. These concepts both used a single vacuum lifter that was actuated using linear actuators. The machine would be controlled by an operator via a terminal. The team agreed that since the concept CIV handled a larger range of sheet sizes it was likely the best concept to develop as a final design. Before this decision was solidified, the concepts were brought to the Client to verify that the Team's opinion aligned with that of Ryerson's. The Client confirmed with the Team that concept CIV was the most promising design; however, the Client emphasized the footprint constraint of the machine. To accommodate this constraint, this concept was integrated into the existing cantilever racking system in location A, shown in Figure 8.

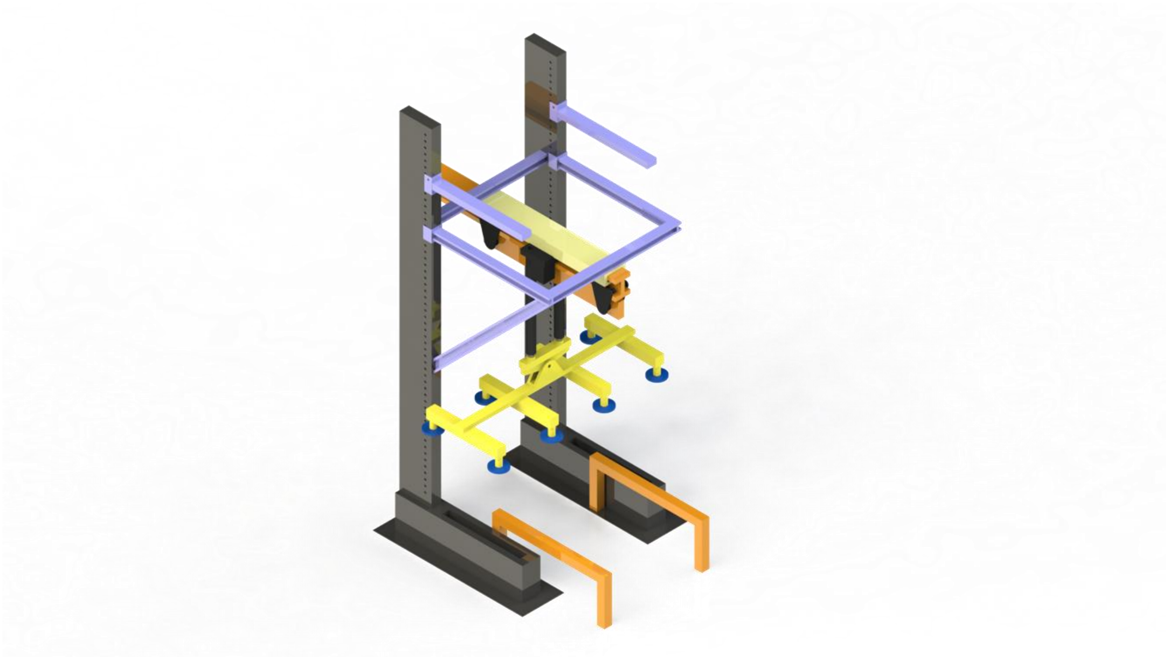


Figure 8: Representation of the final concept

Additionally, the rail that the vacuum head travelled along was designed to be telescoping in order to reduce the footprint of the machine when not in use. This was suggested by the Client [2]. The mechanism is shown in Figure 9.

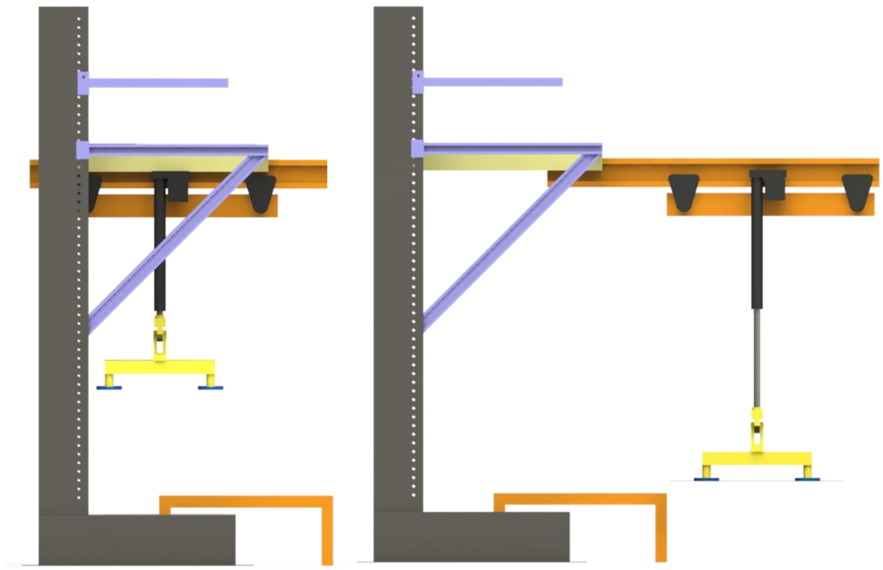


Figure 9: Side view of final concept shown in the retracted and extended positions

It was agreed by the Team and the Client that this concept would be carried forward into the detailed design phase of the project. For the remainder of this report, this concept will be referred to as the final concept.

2.4 Feasibility

As the Team neared the final concept selection process as described in Section 2.3.3, we began to quantify the feasibility of the most promising machine concepts. We researched available products to integrate into our design concepts. The major components were the vacuum lifter, the sheet separator mechanisms, the rail trolleys, the actuators and the available types of steel beams. The specifications of these components revealed some important limitations final design which will be discussed in Section 2.3.3. This information was applied to estimate new performance specifications. Additionally, the actual products found from this research were later used to develop a mock-up of the final concept. The Team also performed preliminary stress calculations on the support structure to ensure the required size of the structural steel members were reasonable. Finally, the Team developed a cost estimate for the final concept and re-evaluated the schedule to ensure the project was still feasible in respect to both of these constraints.

2.4.1 Preliminary Stress Analysis

To perform a feasibility analysis on the new concept, five failure modes were investigated at a high level with numerous assumptions made. The failure modes investigated can be found in TABLE XXII.

TABLE XXII: SUMMARY OF FAILURE MODES AND ASSOCIATED ASSUMPTIONS

Failure Mode	Assumptions
Forces in Members	• The max force due to gravity experienced in the structure is approximately 2200 lbs. • The structure was treated as a simple truss. • All distributed loads were translated into equivalent point loads.
Deflection	• All fillets and stress concentrations were ignored.
Yielding	• The cantilever beam supporting the trolley would be the most likely component to plastically fail.
Buckling	• The only members in the structure that will experience Mode 1 buckling are the two support 45^{o} support members.
Bearing Stress	• The hole pattern utilized is similar to that found on the current racking system at Ryerson.

With the assumptions found in TABLE XXII, and the data found in TABLE XXIII to determine a baseline hanging weight a truss analysis could be completed on the structure.

TABLE XXIII: WEIGHT OF COMPONENTS

Variable	Imperial Value	Metric Value
Maximum Sheet Weight	1600 lbs	711.11 kg
Vacuum Lifter Weight	280 lbs	124.44 kg
Cylinder Weight	100 lbs	44.44 kg
Total Weight	1980 lbs	~880 kg

Using the total weight listed in TABLE XXIII, the force due to gravity can be determined:

$$F_g = (mass)(gravity) \qquad \text{Eq. 2.4.1.1}$$

$$\mathrm{F_g} = (880\ \mathrm{kg})\left(9.81\ {}^{\mathrm{m}}/_{s^2}\right)$$

$$F_g = 8632.80\ N$$

With the force due to gravity determined, the forces in the sliding cantilever I- beam could be determined. The reaction forces are illustrated in Figure 10 with associated calculations following.

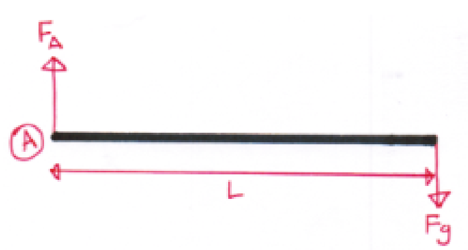


Figure 10: Force balance diagram for the cantilever I beam

$$\sum F_y = 0 \qquad \text{Eq. 2.4.1.2}$$

$$0 = F_g + F_A \qquad \text{Eq. 2.4.1.3}$$

$$0 = 8632.80\,N + F_A$$

$$F_A = 8632.80\,N$$

With the forces determined in the sliding I beam cantilever the forces could be translated into the C – Channel illustrated in Figure 11 with the solution to the translated forces found proceeding the depiction.

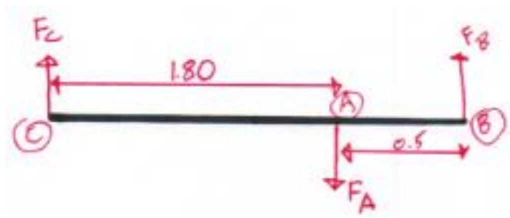


Figure 11: Force balance diagram for the C-channel beam

$$\sum F_y = 0 \qquad \text{Eq. 2.4.1.4}$$

$$0 = F_C + F_B - F_A \qquad \text{Eq. 2.4.1.5}$$

$$\sum M_C = 0 \qquad \text{Eq. 2.4.1.6}$$

$$0 = F_B(0.50\,m) - F_A(1.80\,m) \qquad \text{Eq. 2.4.1.7}$$

$$0 = F_B(0.50\,m) - (8632.80\,N)(1.80\,m)$$

$$F_B = 31\,078\,N$$

$$F_C = F_A - F_B \qquad \text{Eq. 2.4.1.8}$$

$$F_C = -22\,445\,N$$

From the C Channel, the forces were translated and determined in both the front and rear horizontal support beam as illustrated in Figure 12 and Figure 13, respectively.

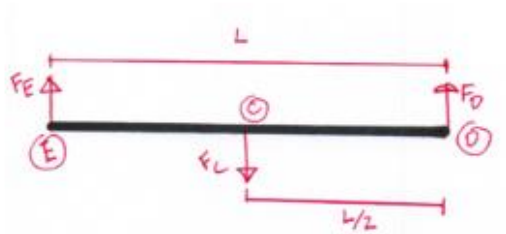


Figure 12: Force balance diagram for front horizontal support beam

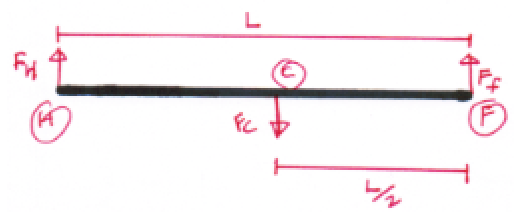


Figure 13: Force balance diagram for the rear horizontal support beam

Due to symmetry, the front and rear horizontal beams, the following equations can be utilized to determine the force translation.

$$F_E = F_D = 2F_B \quad \text{Eq. 2.4.1.9}$$

$$F_E = F_D = 15\,539\ N$$

$$F_F = F_H = 2F_C \quad \text{Eq. 2.4.1.10}$$

$$F_F = F_H = 11\,222.50\ N$$

Finally, the last part of the support structure could be analysed using Figure 14 and conducting a force balance on Node E.

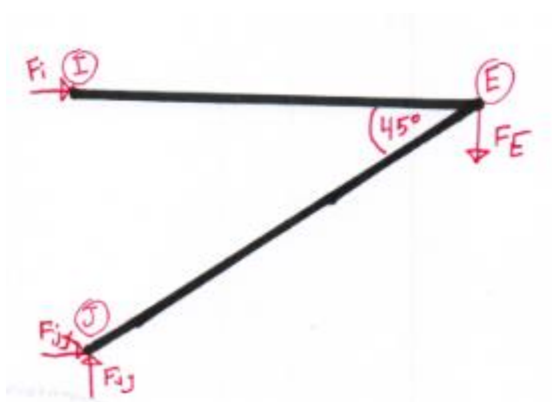


Figure 14: Force balance diagram for the frame support structure

$$F_{IY} = F_{JY} = \frac{15\ 539}{Sin(45)} = 21{,}975.50\text{ N}$$

$$F_{IX} = F_{JX} = \frac{15\ 539}{Tan(45)} = 21{,}975.50\text{ N}$$

After performing the truss analysis, the forces to be experienced in the members appeared reasonable and this failure mode was considered to not be of concern for the Team.

The next failure mode explored was the deflection of the sliding I-Beam at its tip experiencing a load generated by the weight of the vacuum lifter and all associated components. For this calculation, a beam type assumption as made that the beam parameters equated to W250 x 22.3. The weight of the beam was taken into account with the point load using superposition. The formula for max deflection was found to be:

$$Max\ Deflection = \frac{-PL^3}{3EI} - \frac{wL^4}{8EI} \qquad \text{Eq. 2.4.1.11}$$

$$= \frac{-(8632.80N)(2.92\ m)^3}{3(200e9)(2.87e-5)} - \frac{(218.77\ N/m)(2.92\ m)^4}{8(200e9)(2.87e-5)}$$

$$Max\ Deflection = 0.00431\ m$$

The deflection of the beam found in Equation 2.4.1.11 was determined to be acceptable for the assigned parameters.

Following the calculation for the deflection of the beam it was checked for plastic failure. Once again, superposition was utilized to combine the effects of the weight of the beam with the point load applied as a result of the weight of the vacuum lifter and its associated components. To determine the actual stress experienced in the beam, the maximum moment was determined for both load cases then substituted into the maximum stress Equation 2.4.1.13. Finally, both equations were summed up to determine the actual stress located in the beam. The calculations for the distributed load are as followed:

$$M = \frac{-wL^2}{2} \qquad \text{Eq. 2.4.1.12}$$

$$M = \frac{-(218.76\,N)(2.92)^2}{2}$$

$$M = -932.61\,Nm$$

$$\sigma = \frac{(M)(y_{max})}{I} \qquad \text{Eq. 2.4.1.13}$$

$$\sigma = \frac{(932.61)(0.127)}{2.87e-5}$$

$$\sigma = 4.126\,MPa$$

The calculations for the point load are as followed:

$$M = Fd \qquad \text{Eq. 2.4.1.14}$$

$$M = (8632.80)(2.92)$$

$$M = 25207.77\,Nm$$

$$\sigma = \frac{(25207.77)(0.127)}{2.87e-5}$$

$$\sigma = 111\,546\,604.60\,Pa$$

The final actual stress determined using superposition was found to be 115 MPa. The factor of safety was determined using Equation 2.4.1.15.

$$Factor\ of\ Safety = \frac{\sigma_{yield}}{\sigma_{actual}} \qquad \text{Eq. 2.4.1.15}$$

$$Factor\ of\ Safety = \frac{250\,MPa}{115\,MPa} = 2.2$$

The factor of safety was deemed acceptable, which helps prove the feasibility of the concept. The next failure mode investigated was buckling in the 45^{o} support members. Two approaches were taken to check for feasibility due to this failure mode. The first was what the critical buckling load would be when using a W250 x 223 beam. The critical buckling load was found using Equation 2.4.1.16.

$$P_{Cr} = \frac{\pi^2(E)(I)}{(y_{max})^2} \qquad \text{Eq. 2.4.1.16}$$

$$P_{Cr} = \frac{\pi^2(200e9)(2.87e-5)}{(1.015)^2}$$

$$P_{Cr} = 54\ kN$$

The second approach taken was to utilize a factor of safety of 8, work backwards through EQN 16 to determine a second moment of inertia and compare that value to common I beam cross sections to check for feasibility. By utilizing EQN 2.4.1.16, the second moment of inertia was found to be 1.15e-8 m^4.

The final failure mode investigated for feasibility was utilizing bearing stress to determine if a reasonable thickness was possible to support a factor of safety of 8. Equation 2.4.1.17 was utilized to perform this feasibility analysis.

$$\frac{Yield\ Strength}{Factory\ of\ Safety} = \frac{Load}{td} \qquad \text{Eq. 2.4.1.17}$$

$$\frac{250e6}{8} = \frac{15539}{t(0.02)}$$

Equation 2.4.1.17 suggested the minimum thickness required to ensure a factor of safety of 8 would be 0.025m which the Team determined to be reasonable. By combining all five stress calculations, the Team determined that this concept was feasible.

SECTION 3 DETAILS OF THE DESIGN

The preliminary feasibility analysis of the machine demonstrated that the structural beams built from ASTM A36 carbon steel would be able to withstand the load applied on the beams by the lifter. From the preliminary analysis, the Team initiated detailed analysis of the sheet metal lifting machine to determine the specifications of the design components. To begin, we decomposed the machine into two main assemblies – lifter and support structure. The detailed analysis encompassed both static and dynamic loading cases to design the machine for the worst loading scenario.

Based on the worst-case, iterative engineering failure analysis, which included von Mises yielding, fatigue, deflection and buckling, took place until the design was optimized in terms of footprint and weight while meeting the design constraints. During the iterative process, the feasibility of the design was continuously checked with a stability analysis to ensure that the machine would not lose balance due to the cantilever actions of the components.

Prior to entering the detailed design phase, a failure mode and effects analysis was conducted to account for the potential failure modes in the finalized design. Through the evaluation of the FMEA, engineering failure modes were mitigated or eliminated through design. However, failure modes due to material wear and defects were handled by providing proactive recommendations to the Client.

The following section of the report outlines all details of the final design as determined by the Team.

3.1 Preliminary Loading Calculations

The Team undertook a systematic engineering approach by dividing the machine into three main sections: the support structure, table and, lifting mechanism. Starting at the lifting mechanism, the Team identified three unique loading scenarios which represented the different ways an operator would handle the lifter. For instance, an operator pulling the lifter towards his own body, pushing the lifter to the sides or trying to move the lifter past the maximum distance would create different moments and forces on the lifter.

As the Team progressed through the details of the design, it was determined that an iterative approach was required in all calculations to ensure that the final design was capable of meeting all safety requirements. An iterative approach was required due to all of the design components being interconnected with each other which caused the analysis of the components to become interconnected and dependent on one other.

In order to design for the worst-case scenario, the Team explored three loading cases on the lifter design. By evaluating each loading scenario, the Team identified the worst-case scenario which was used throughout the analysis to calculate the reaction forces.

3.1.1 Preliminary Design Approximations

The Team began by establishing assumptions for the machines general composition and operation. These initial assumptions were based heavily on the initial concept of the machine, but were adjusted during engineering analysis as needed to comply with the structural and functional design requirements. The list of global design assumptions is tabulated in TABLE XXIV.

TABLE XXIV: PRELIMINARY ASSUMPTIONS OF THE MACHINE DESIGN

Assumption	Justification
The general function and construction of the machine will loosely comply with the initial concept.	This is standard practice. Changes would be made as potential deficiencies or opportunities were identified in the concept.
The structural components will be created using AISI 1020 steel sheet or standard ASTM A36 structural steel elements where appropriate.	These assumptions were applied to minimize the material and manufacturing costs and streamline the analysis of the structure.
Precision machining will only be used where absolutely necessary.	
The structure will be welded where appropriate and where disassembly is not necessary.	This was to minimize the number of components and fasteners.

The four assumptions were necessary to begin the analysis and refinement of the design. For example, the Team needed to assume the initial structural steel elements throughout the design to develop the equations for calculating the loads and stresses acting on the machine structure.

In addition to the listed assumptions, the Team performed preliminary calculations to approximate the overall dimensions and weight of the machine. The calculations were performed by using yielding as the design criterion and were focused on reducing the footprint of the machine. To begin the analysis, the weight of the lifter assembly was approximated by taking the mass of the sheet, lifter frame and lifting structure to consideration. The weight used for the analysis is listed in TABLE XXV.

TABLE XXV: ESTIMATED WEIGHTS OF THE LIFTER COMPONENTS AND THE ASSEMBLY

Weight Estimate (lb)		Accuracy
Sheet (max)	250	Datasheet [21]
Lifter frame & cups	185	Datasheet [22]
Lifting structure	150	Estimation
Weight of other sliding structures	285.6	Estimation
TOTAL	**870.6**	

For the analysis, the weight of the lifter was assumed to be distributed evenly among the members. With the assumption, the reaction forces, shear force and maximum moment along the sliding I-beam were calculated from the weight. The details of the load analysis are shown in Section 3.1.4.3.

From the maximum moment, the Team found the maximum bending stress induced on the I-beam. Since the Team only accounted for static loading in the initial calculation, the resulting bending stress was used to optimize the length of the I-beam. The length of the I-beam was determined based on design constraints and other components. These factors include:

1. When fully retracted, the design should not exceed 80 inches in length from the column.
    a. It is preferred to keep the fully retracted length as short as possible to reduce the footprint of the machine.
2. When fully retracted, the back of the sliding I-beam should not exceed 12 inches in length from the column.
3. The height of the I-beam needs to be adequate (minimum of 8 inches) to contain two roller bearings (each with diameter of 2.94 inches) in-between them.
    a. The associated c-channel bracket should be taller than the I-beam to compensate for the flange thickness (minimum of 9 inches).

Based on the identified constraints, the Team set up a Microsoft Excel Solver function using the conditions shown in TABLE XXVI to perform iterative calculations in order to find the optimal length and size of the I-beam.

TABLE XXVI: SUMMARY OF CONDITIONS USED FOR BEAM OPTIMIZATION

Components	Conditions	Reasons
Length of I-beam	≤ 80 inch	Design footprint constraint
Distance between bearings	≥ 17.81 inch	Minimum length required for containing the bearings, dampers and dampers' full stroke length
Weight of I-beam	≥ 18.4 lbs/feet	Minimum height of the I-beam to contain the required components within the C-channel bracket
Factor of Safety	≥ 8	Factor of safety typically used for overhead cranes

Through optimization, the Team identified the most optimal I-beam configuration to be C8 X 18.4. Since the beam met the minimum height requirement the Team initially set, the C-channel bracket was determined to be C9 X 13.4. The length of the C-channel was calculated by finding the distance from the column to the estimated forklift location. The optimization result can be found in TABLE XVII.

TABLE XXVII: OPTIMIZED I-BEAM AND C-CHANNEL BRACKET SIZES

	Beam Designation	Beam Length	Distance between bearings
I-beam	C8 X 18.4	4.76 feet	1.485 feet
C-channel bracket	C9 X 13.4	6.55 feet	N/A

Since the results did not account for dynamic loading scenarios, the Team conducted a detailed load analysis encompassing the worst-case loading scenario after the initial dimension calculations were performed. The detailed analysis is presented in Section 3.1.4.

The designed C-channel bracket was to be mounted onto the columns with the help of structural tie rods. Since the Team pursued reducing the overall footprint of the design, we analyzed the optimal length of the tie rod based on the adjustment angle and force applied on it. By assuming that a single tie rod was holding the entire weight of the lifter assembly, the Team applied the load induced by the lifter to the tie rod. Then, the angle of the tie rod was adjusted iteratively until a minimum angle was determined while ensuring a factor of safety of eight was met. TABLE XXVIII shows the conditions used for the iterations.

TABLE XXVIII: CONDITIONS USED FOR THE OPTIMAL TIE ROD ANGLE

Components	Conditions	Reasons
Load induced by lifter	870 lb	Weight of the lifter assembly
Angle	≥ 10 degrees	Estimation
Factor of Safety	≥ 8	Factor of safety typically used for overhead cranes

From the analysis, the Team found that with a minimum angle of 20 degrees from the horizontal, the tie rod would be able to withstand the forces applied on them.

Upon finding the required configuration for the structural beams, the Team solved for the required height of the lifter assembly. We estimated the maximum and minimum heights of the vacuum suction cups from the material and components to determine the actuator stroke length. To find the minimum height, the Team analyzed the maximum distance the lifter needs to travel to pick up the sheet metals from the loading area. The analysis was based on the heights of the platform and the air knife. The analysis result is shown in TABLE XXIX.

TABLE XXIX: ESTIMATED MINIMUM VACUUM HEAD HEIGHT

Unloading Area				
Measurement	**Value**	**Unit**	**Accuracy**	**Calculations**
Platform height	20	in	Calc.	Slightly above the column base
Table top thickness (max)	2	in	Est.	
Air knife (overall)	12	in	Datasheet [23]	
Air knife nozzles	10	in	Datasheet [23]	

Bottom of air knife	8	in	Calc.	Distance from floor to the bottom of air knife
Lowest sheet	10	in	Calc.	Distance from floor to the lowest point in the air knife nozzle
Extra range	2	in	Est.	
Min vac head height	8	in	Calc.	Minimum vacuum head height including buffer extra range

After the minimum vacuum head height was calculated, the Team analyzed the maximum vacuum head height. The analysis was conducted from the calculated platform height and four estimated height values as shown in TABLE XXX.

TABLE XXX: ESTIMATED MAXIMUM VACUUM HEAD HEIGHT

Unloading Area				
Measurement	**Value**	**Unit**	**Accuracy**	**Calculations**
Platform height	20	in	Calc.	Slightly above the column base
Skid thickness	6.00	in	Est.	
Max stack height (load)	4	in	Est.	
Min lifting gap	4	in	Est.	
Extra range	2	in	Est.	
max vac head height	36	in	Calc.	Sum of the measurements

From the maximum and minimum height calculations, the Team could identify the required actuator stroke length. The summary of the analysis including the resulting actuator stroke length is shown in TABLE XXXI.

TABLE XXXI: SUMMARY OF HEIGHT CALCULATIONS

Lifting Requirements			
Measurement	**Value**	**Unit**	**Accuracy**
Max vac head height	36	in	Calc.
Min vac head height	8	in	Calc.
Actuator stroke	28	in	Calc.

As shown in TABLE XXXI, an actuator stroke length of 28 inch was required for the lifter assembly to reach the maximum and minimum vacuum head heights.

3.1.2 Loading Scenarios

Early in the design phase, it was important to understand the types of loads that the machine would experience. The Team developed a list of forces and moments that may be applied to the machine during normal operation. Most of these loads were assumed to be directly applied to the vacuum lifter. These loads are listed in TABLE XXXII. The coordinate system was established throughout this report so that the Z-axis was vertically downward, the X-axis was directed along the sliding mechanism of the machine, and the Y-axis was directed laterally (left and right). Note that these loads do not represent the first approximations that we made, rather the most recent values that were used for the final calculations.

TABLE XXXII: PREDICTED EXTERNAL LOADS ON MACHINE

		Forces (lb)			Moments (ft-lb)		
ID	**Direction:**	**Z**	**Y**	**X**	**Mz**	**My**	**Mx**
1	Unbalance, Sheet Static, B	250				125	
2	Unbalance, Sheet Static, A	250					125
3	Weight, Vacuum Lifter	185					
4	Weight, Bearing Rails	92.2					
5	Weight, Cylinder	40					
6	Weight, Lifter Beam	162					
7	Load, Lifting	707					
8	Unbalance, sheet lifting, B	707				353.5	
9	Unbalance, sheet lifting, A	707					353.5
10	Worker push/twist B		184		1028.6		
11	Worker push/twist A			184	1028.6		
12	Contacting shock absorbers			300			

The unbalanced sheet loads were calculated based off a 250 lb sheet held aloft by the vacuum lifter that is positioned 6 inches out of alignment from center. The weights represent conservative approximations based on the initial concept. The lifting load was based off the maximum lifting force of the cylinder at the operating pressure. The worker push load was an estimate of a strong worker pushing the lifter at a moment arm of roughly five and a half feet from the center axis of the lifter. Finally, the 300lb load from the shock absorbers was a target deceleration value for stopping the machine travelling at maximum velocity.

These loading scenarios were combined to create three loading scenarios that the Team believed would represent the worst case scenarios. The loading scenarios were split into two versions, A and B, that represented primarily X-direction loading or primarily Y-direction loading. These loading scenarios are listed in TABLE XXXIII.

TABLE XXXIII: DESCRIPTIONS OF THE LOADING SCEANRIOS

Loading Scenario	Loads Involved	Description
Case 1A	2*, 3, 4, 5, 6, 12*	This scenario represents the vacuum lifter colliding with the dampers while holding a 250lb unbalanced sheet.
Case 1B	1*, 3, 4, 5, 6, 12*	
Case 2A	9**	This scenario represents lifting an unbalanced sheet.
Case 2B	8**	
Case 3A	1**, 3, 4, 5, 6, 11**	This scenario represents a large external force (184lb) applied on the vacuum lifter to induce a torque on the lifting mechanism.
Case 3B	2**, 3, 4, 5, 6, 10**	

*load applied when vacuum lifter is raised
**load applied when vacuum lifter is extended

The loading scenarios listed in TABLE XXXIII were used to develop the equations to describe the forces and stresses throughout the machine and will be referenced throughout the remainder of the report.

3.1.3 Loading on the Lifting Mechanism

The lifting mechanism was defined as the assembly of the machine that slides with the I beam and lifts the sheet metal from the stock skid. The purpose of the lifting mechanism is to position the vacuum head to the height of the stock material on the supply pallet and slides it along the machine's beams to the off-load pallet found on the platform.

The mechanism connected to the standard ANVER vacuum lifter frame is a custom steel clevis. The clevis is supported vertically by a single 28-inch stroke, 3-inch bore, pneumatic cylinder which can lift a 250 lb sheet of material. The five remaining degrees of freedom for the vacuum lifter are constrained by four linear bearing carriages that travel along two rails mounted on 3-inch square tube. These bearings guide the vertical movement of the lifter. One pair of bearing carriages is designed to allow "float" in the Y-direction to account for bearing rail misalignment. The bearing carriages and cylinder are supported along a composite spreader beam created from two C6x13 steel channels and a number of sheet steel stiffener plates.

The lifting mechanism was separated into components to analysis the loading effects on the complex geometry of the lifter. Each of these components could be analysed separately as a beam or column. The lifter is composed of the vacuum lifter frame, the bearing rail supports, the bearing carriages, the pneumatic cylinder, the cylinder supports, and the spreader beam. Figure 15 shows where these components are located.

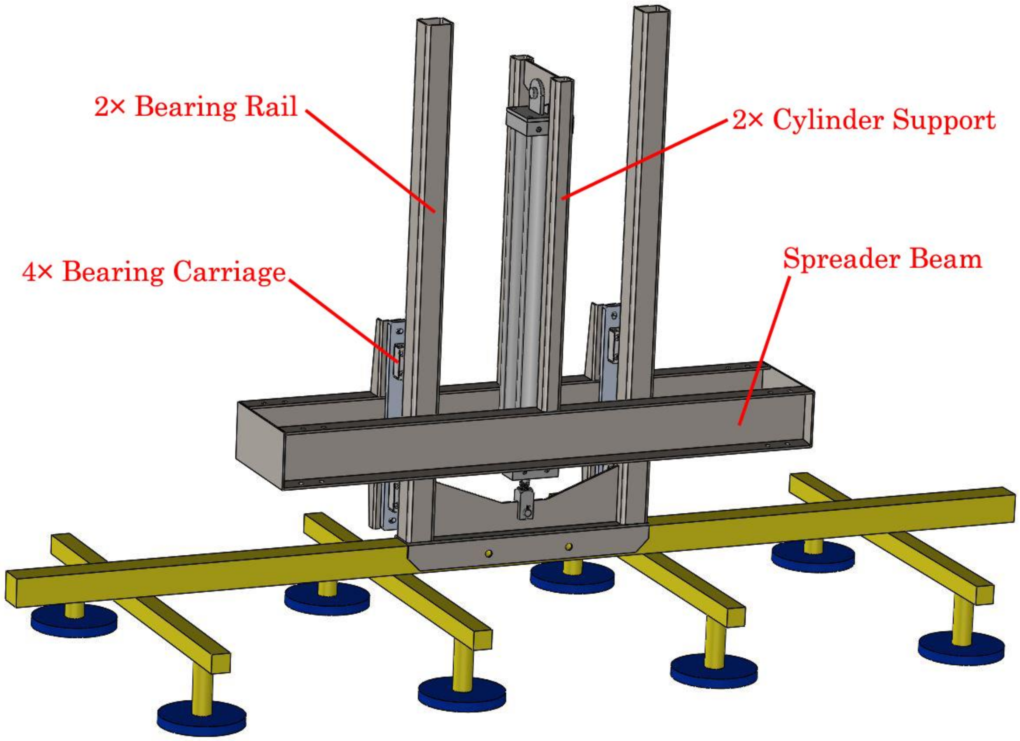


Figure 15: The lifting mechanism in the raised position with the components identified

These components were selected for thorough analysis as they were expected to carry the largest loads, especially in bending which typically induces the highest stresses. The three loading scenarios for the machine were translated to the equivalent loading on each component. The worst loading scenarios for each component was used to determine the maximum stresses in the machine.

The Team began by analysing the loads acting on the bearing carriages and bearing rail supports as these were closely correlated. Loads on the bearing rails and bearing carriages are caused by the forces (F_x, F_y, and F_z) and moments (M_x, M_y, and M_z) acting on the vacuum lifter frame. This interaction is shown by the free body diagram (FBD) in Figure 16 where the each "R" denotes a reaction load on the bearing carriage and corresponding rail support. The pneumatic cylinder is also shown carrying the vertical force component. The four bearing carriages are labeled as "A" through "D." Also, the section length of the bearing rail, $h_{bearing}$, is variable depending on if the lifter is extended or raised.

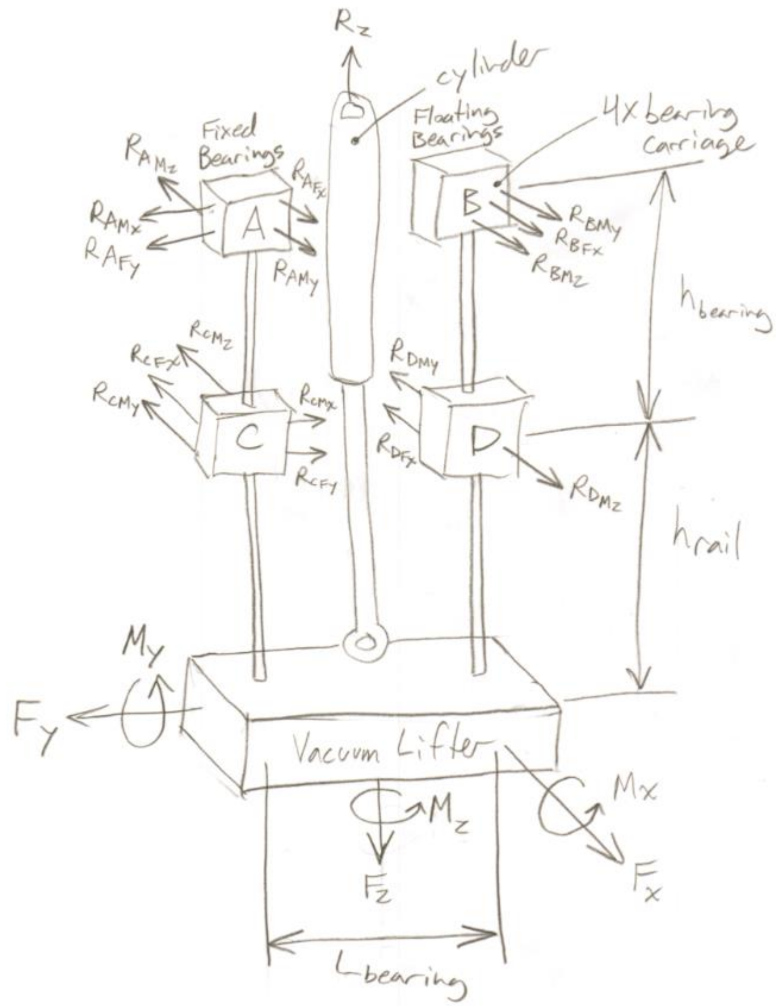


Figure 16: A three-dimensional free body diagram showing the forces and moments acting on the vacuum lifter and the corresponding reaction forces at the four bearing carriages A to D

There are no reaction moments shown on individual bearing carriages since it was assumed that the bearings were spaced far enough apart that these moments would act largely as force couples between bearings. In addition, one set of bearings is designed to "float" horizontally, so it does not allow for any reaction forces along the Y axis. This floating bearing set was recommended by Igus and allows for parallelism error between the two bearing rails [24].

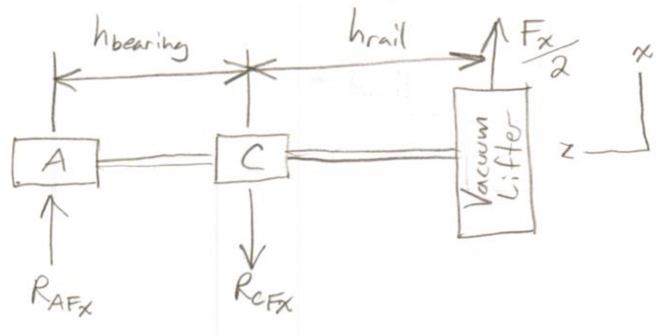
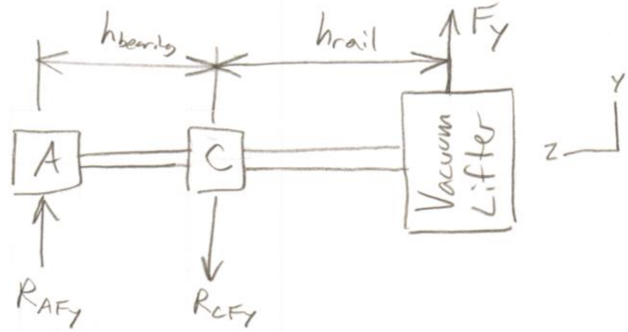
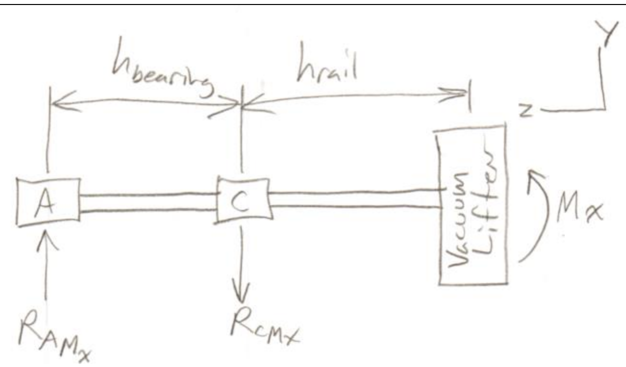
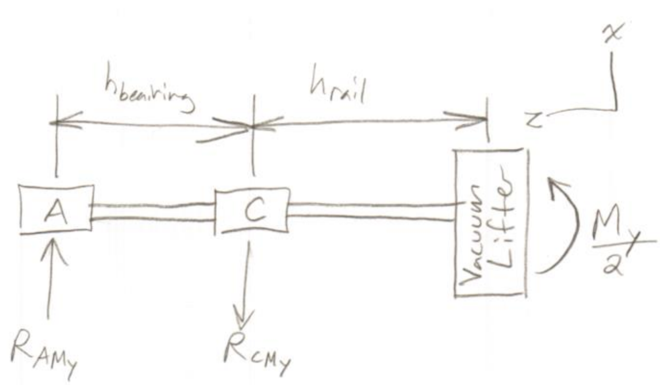
The three-dimensional FBD was converted into a set on two-dimensional FBDs for each force and moment as shown in TABLE XXXIV.

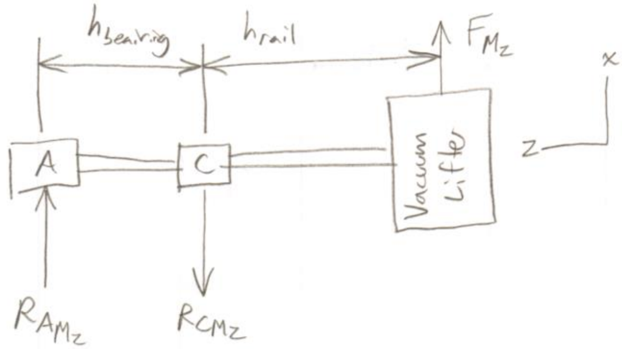
TABLE XXXIV: BASIC EQUATIONS FOR REACTION FORCES AT THE BEARING CARRIAGES

Name	Applied Load	Carriage A Reaction	Carriage C Reaction
Bearing FBD 1	Force along X	$R_{AF_x} = \frac{F_x \cdot h_{rail}}{2 \cdot h_{bearing}}$	$R_{CF_x} = \frac{1}{2} F_x + R_{AF_x}$
Bearing FBD 2	Force along Y	$R_{AF_y} = \frac{F_y \cdot h_{rail}}{h_{bearing}}$	$R_{CF_y} = F_y + R_{AF_y}$
Bearing FBD 3	Force along Z	$R_{AM_x} = \frac{M_x}{h_{bearing}}$	$R_{CM_x} = R_{AM_x}$
Bearing FBD 4	Force along Y	$R_{AM_y} = \frac{M_y}{2 \cdot h_{bearing}}$	$R_{CM_y} = R_{AM_y}$
Bearing FBD 5	Force along Z	$R_{AM_z} = \frac{F_{M_z} \cdot h_{rail}}{h_{bearing}}$	$R_{CM_z} = F_{M_z} + R_{AM_z}$

The diagrams show only the loads acting on the fixed bearing rail, but the derived equations are applicable for both rails. These diagrams were then combined to represent the different loading scenarios to determine the stresses in the components. These FBDs will be referred to as the bearing FBDs one through five throughout this analysis section.

TABLE XXXV: FREE BODY DIAGRAMS OF THE BEARING CARRIAGES AND RAILS FOR EACH LOAD TYPE

Name	Free Body Diagram
Bearing FBD 1	
Bearing FBD 2	
Bearing FBD 3	
Bearing FBD 4	

Name	Free Body Diagram
Bearing FBD 5	

The equation for the reaction force at bearing carriage A was found in each case by performing a moment balance about bearing carriage C. The moment balance for the bearing FBD 1 is shown below.

$$\sum M_c = 0 = \tfrac{1}{2} F_x \cdot h_{rail} - R_{AF_x} \cdot h_{bearing} \qquad \text{Eq. 3.1.3.1}$$

Notice that the applied force is divided by two because this force is actually supported by both bearing rails. The second bearing rail with carriages B and D is not shown. The moment balance for the second bearing FBD, shown in Eq. 3.1.3.2, does not have the factor of a half because the other bearing rail is floating in the direction Y-axis. This means this bearing rail does not support any load in the Y direction.

$$\sum M_c = 0 = F_y \cdot h_{rail} - R_{AF_y} \cdot h_{bear} \qquad \text{Eq. 3.1.3.2}$$

The bearing A reaction forces for the bearing FBD 3 and 4 are shown as equations 3.1.3.3 and 3.1.3.4 respectively. The moment M_y is shared between the two bearing rails so there is once again a factor of one half.

$$\sum M_c = 0 = M_x - R_{AM_x} \cdot h_{bearing}$$ Eq. 3.1.3.3

$$\sum M_c = 0 = \frac{M_y}{2} - R_{AM_y} \cdot h_{bearing}$$ Eq. 3.1.3.4

For the moment about the Z-axis in bearing FBD 5, it was assumed that the moment would be transferred to the two bearing rail supports as the force couple F_{Mz}. This force couple equation is shown below where $L_{bearing}$ is the size of the gap between the two bearing rail supports.

$$F_{M_z} = \frac{M_z}{L_{bearing}}$$ Eq. 3.1.3.5

The moment balance about carriage C for this FBD is as follows.

$$\sum M_c = 0 = \frac{M_y}{2} - R_{AM_y} \cdot h_{bearing}$$ Eq. 3.1.3.6

The five moment balances in equations 3.1.3.1, 3.1.3.2, 3.1.3.3, 3.1.3.4 and 3.1.3.6 were rearranged to solve for the reaction force at carriage A. The reaction force at A was used to perform a force balance for each FBD and develop the equation for the reaction force at carriage C. These final basic equations are all summarized in TABLE XXXVI.

TABLE XXXVI: BASIC EQUATIONS FOR REACTION FORCES AT THE BEARING CARRIAGES

Name	Applied Load	Carriage A Reaction	Carriage C Reaction
Bearing FBD 1	Force along X	$R_{AF_x} = \frac{F_x \cdot h_{rail}}{2 \cdot h_{bearing}}$	$R_{CF_x} = \frac{1}{2} F_x + R_{AF_x}$
Bearing FBD 2	Force along Y	$R_{AF_y} = \frac{F_y \cdot h_{rail}}{h_{bearing}}$	$R_{CF_y} = F_y + R_{AF_y}$
Bearing FBD 3	Moment about X	$R_{AM_x} = \frac{M_x}{h_{bearing}}$	$R_{CM_x} = R_{AM_x}$
Bearing FBD 4	Moment about Y	$R_{AM_y} = \frac{M_y}{2 \cdot h_{bearing}}$	$R_{CM_y} = R_{AM_y}$
Bearing FBD 5	Moment about Z	$R_{AM_z} = \frac{F_{M_z} \cdot h_{rail}}{h_{bearing}}$	$R_{CM_z} = F_{M_z} + R_{AM_z}$

Once these basic equations were established they were combined to derive the equations for the complex loading scenarios. The worst-case scenarios for bearing carriages were predicted as cases 1B, 3A, and 3B. Each of these scenarios is shown TABLE XXXVII. To reiterate, case 1B represents the lifter moving an unbalanced sheet and colliding with the shock absorbers at end of the sliding I-beam mechanism at full speed. Case 3A and 3B represent a 184lb external force applied at the end of a 10-foot long, unbalanced sheet while it is being held by the machine at maximum vertical extension. This 184lb force is comparable to the maximum strength of a worker [25]. Cases 3A and 3B are distinct in the direction of the external load.

The table also contains the variables that represent each scenario and the full equation for the reaction forces on the carriage according to these variables. The values for the variables correspond to those derived in Section 3.1.2 and summarized in TABLE XXXII. The reaction force equations are written with reference to the basic equations shown previously in TABLE XXXVI.

TABLE XXXVII: REACTION FORCES AT BEARING CARRIAGE ACCORDING TO THE COMPLEX LOADING SCENARIOS

Case	Variable	Description	Value	Carriage A Reaction Forces
1B	M_y	Unbalanced sheet	125 ft-lb	$R_{Ax} = R_{AF_x} + R_{AM_y}$ $R_{Ay} = 0$
	F_x	Collision with shock absorber	300 lb	
	h_{rail}	Raised position	14.64 in	
	$h_{bearing}$	Bearing spacing	14 in	
3A	F_x	External force	184 lb	$R_{Ax} = R_{AF_x} + R_{AM_z} + R_{AM_y}$ $R_{Ay} = 0$
	M_z	Torsion from external force	1029 ft-lb	
	M_y	Unbalanced sheet	125 ft-lb	
	h_{rail}	Lowered position	42.64 in	
	$h_{bearing}$	Bearing spacing	14 in	
	$L_{bearing}$	Bearing spacing	24 in	
3B	M_z	Torsion from external force	1029 ft-lb	$R_{Ax} = R_{AM_z}$ $R_{Ay} = R_{AF_y} + R_{AM_x}$
	h_{rail}	Lowered position	42.64 in	
	$h_{bearing}$	Bearing spacing	14 in	
	$L_{bearing}$	Bearing spacing	24 in	
	F_y	External force	184 lb	
	M_x	Unbalanced sheet	125 ft-lb	

The total X and Y reaction forces for bearing carriage A were found using these values and formulae. The results are compiled in TABLE XXXVIII.

TABLE XXXVIII: TOTAL REACTION FORCES AT BEARING CARRIAGE A FOR THREE CRITICAL LOADING SCENARIOS

Case	R_{Ax}	R_{Ay}
1B	210.4 lb	0
3A	1900.2 lb	0
3B	1566.4 lb	667.6 lb

The equations for the forces acting on carriage C were found using the exact same method. These equations are summarized in TABLE XXXIX.

TABLE XXXIX: REACTION FORCES AT BEARING CARRIAGE C ACCORDING TO THE COMPLEX LOADING SCENARIOS

<table>
<tr><th>Case</th><th>Variable</th><th>Description</th><th>Value</th><th>Carriage C Reaction Forces</th></tr>
<tr><td rowspan="4">1B</td><td>M_y</td><td>Unbalanced sheet</td><td>125 ft-lb</td><td rowspan="4">$R_{cx} = R_{cF_x} + R_{cM_y}$
$R_{cy} = 0$</td></tr>
<tr><td>F_x</td><td>Collision with shock absorber</td><td>300 lb</td></tr>
<tr><td>h_{rail}</td><td>Raised position</td><td>14.64 in</td></tr>
<tr><td>$h_{bearing}$</td><td>Bearing spacing</td><td>14 in</td></tr>
<tr><td rowspan="6">3A</td><td>F_x</td><td>External force</td><td>184 lb</td><td rowspan="6">$R_{cx} = R_{cF_x} + R_{cM_z} + R_{cM_y}$
$R_{cy} = 0$</td></tr>
<tr><td>M_z</td><td>Torsion from external force</td><td>1029 ft-lb</td></tr>
<tr><td>M_y</td><td>Unbalanced sheet</td><td>125 ft-lb</td></tr>
<tr><td>h_{rail}</td><td>Lowered position</td><td>42.64 in</td></tr>
<tr><td>$h_{bearing}$</td><td>Bearing spacing</td><td>14 in</td></tr>
<tr><td>$L_{bearing}$</td><td>Bearing spacing</td><td>24 in</td></tr>
<tr><td rowspan="4">3B</td><td>M_z</td><td>Torsion from external force</td><td>1029 ft-lb</td><td rowspan="4">$R_{cx} = R_{cM_z}$
$R_{cy} = R_{cF_y} + R_{cM_x}$</td></tr>
<tr><td>h_{rail}</td><td>Lowered position</td><td>42.64 in</td></tr>
<tr><td>$h_{bearing}$</td><td>Bearing spacing</td><td>14 in</td></tr>
<tr><td>$L_{bearing}$</td><td>Bearing spacing</td><td>24 in</td></tr>
</table>

Again, these formulae reference the equations previously derived and summarized in TABLE XXXVI. The total force components on carriage C are summarized in TABLE XL.

TABLE XL: TOTAL REACTION FORCES AT BEARING CARRIAGE C FOR THREE CRITICAL LOADING SCENARIOS

Case	R_{cx}	R_{cy}
1B	360.4 lb	0
3A	2414.5 lb	0
3B	2080.7 lb	851.6 lb

Next, the forces on the bearing carriages A and C were used to determine the loads on the bearing rail supports. Only the worst loading case was examined for the bending loads in the bearing rail supports. Case 3A was selected as the worst case based on the results of the bearing carriage loads which showed large bending loads in the XZ plane. Figure 17 shows the FBD of the rail.

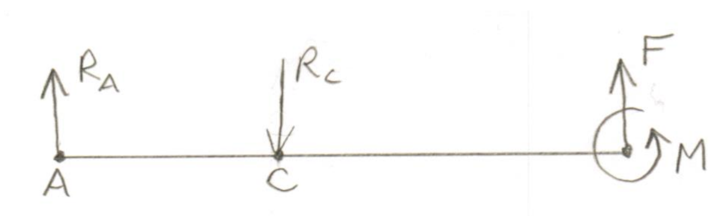


Figure 17: Free body diagram for the bearing rail support

The reaction forces were equal to the forces for case 3A in TABLE XXXVIII and TABLE XL. These forces are 1900.2 lb and 2414.5 lb for R_A and R_C respectively. The exact values for moment M and force F in Figure 17 can be solved for using force balance and moment balance equations, but they were not necessary to solve for to determine the maximum loads and stresses in the rail. The maximum loads were found by creating shear force and moment diagrams shown in Figure 18 and Figure 19.

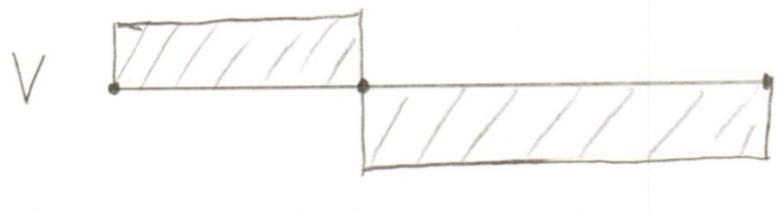


Figure 18: Shear force diagram for the rail support

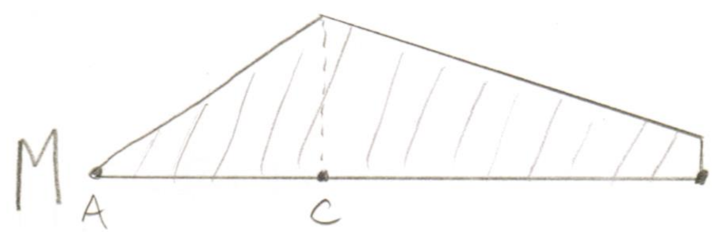


Figure 19: Moment diagram for the rail support

From these two diagrams, it is apparent that the largest shear force and bending moment are both at the location of bearing carriage C. The maximum shear force was 2414.5 lb and the maximum bending moment was 2216.9 ft-lb. The third area of interest was the spreader beam. Again, the highest load scenario was case 3A. The free body diagram for the scenario is shown in Figure 20. Notice F_x is applied remotely at a distance "h" where the lifting head would be located.

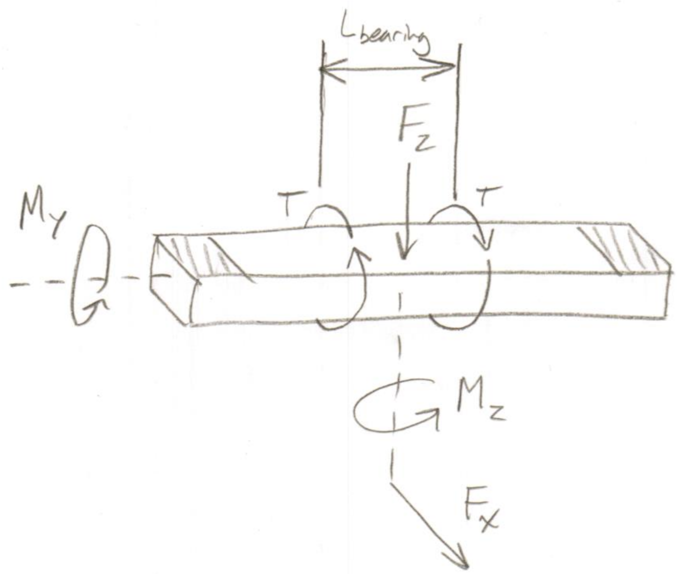


Figure 20: Free body diagram of spreader beam

These loads follow those laid out originally for case 3A. They are shown again below. Note that the distance, h, from F_x to the spreader beam is equal to the sum of h_{rail} and half of $h_{bearing}$. The torque, T, shown around the center of the beam is a consequence of the moment M_z as it acts on and bends the bearing rails. This torque was calculated by finding the reaction force couple acting on the bearing rails due to the moment. The force couple in the bearing rails was then used to calculate the reaction torque on the spreader, T. The reaction forces on the bearing rails were given by the forces on the bearing carriages. The force couple can be calculated from the bearing separation, $h_{bearing}$, and the x-component force on bearing carriage A, Fx, which was given in TABLE XLI.

TABLE XLI: LOADS ACTING ON THE SPREADER BEAM

Case	Variable	Description	Value
3A	F_x	External force	184 lb
	M_z	Torsion from external force	1029 ft-lb
	M_y	Unbalanced sheet	125 ft-lb
	h_{rail}	Lowered position	42.64 in
	$h_{bearing}$	Bearing spacing	14 in
	$L_{bearing}$	Bearing spacing	24 in

This concluded the load calculations for the lifter assembly. These forces and moments were used later to perform stress analysis and determine the size requirements of the structural components of the machine.

3.1.4 Loading on Support Structure and Columns

The structure, in which the lifter discussed in Section 3.1.3 is to be mounted, is composed of multiple structural beams. In order to adapt the lifter design and to enhance the versatility of the design in terms of being adaptable to different facilities, the Team has changed the final conceptual design shown in Section 2.3.3 to the double beam design. As the name suggests, the modified design utilizes two sliding beams that are mounted onto the columns instead of a single beam sliding through the channel to transfer sheets.

Each C-channel bracket, which the sliding beams roll through, is consisted of two individual C-channels. The C-channels are positioned vertically and in a way that the tapered parts of the beams are facing towards each other. Through this, the c-channel extrusions act as the rail that the sliding I beam and tapered roller bearings move along.

The designed brackets are mounted onto the base columns at each end of the C-channel bracket. Near the columns, the bracket is directly mounted onto the columns with the use of C-shaped support members. The free-end of the bracket is held in place with an adjustable structural tie-rod. The length of the tie-rod is adjusted accordingly to the length of the bracket to allow the user to level the bracket adequately.

For the purpose of mounting the spreader to the structure, the spreader beams shown in Figure 15 are used in the design. They are positioned vertically so that the top flange of the beam is fixed onto the sliding I-beam itself and the bottom flange is fixed onto the lifter. The channels are set at a distance from each other which equals the depth of the lifter design. The placement of the beams also limits lateral movements which would have been caused otherwise due to the double sliding beam design. The overview of the support structures including the lifter assembly is shown in Figure 21.

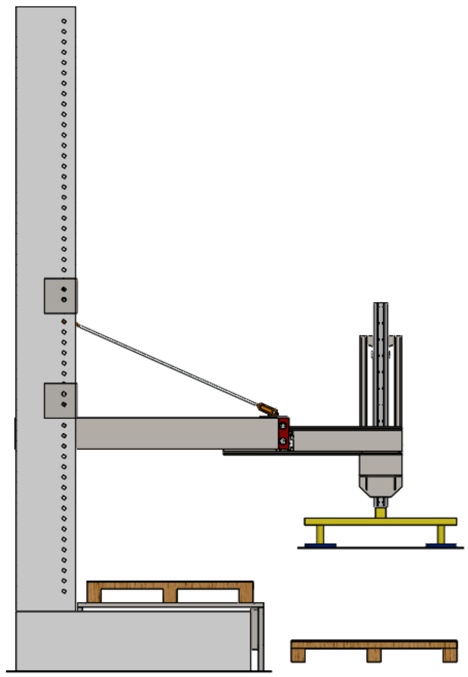


Figure 21: Render of the support structure to interface the lifter assembly with the cantilever beam racking

3.1.4.1 Spreader Reaction Force Calculation

Through the analyse of the lifter assembly, the Team calculated the loads and moments on the spreader beams. Out of the three different loading cases, the Team used case 3A for the design since the structure experienced the largest reaction forces from case 3A as discussed in Section 3.1.2. The loads and moments for case 3A and the corresponding reaction forces and moments at the connection points can be found in Figure 22.

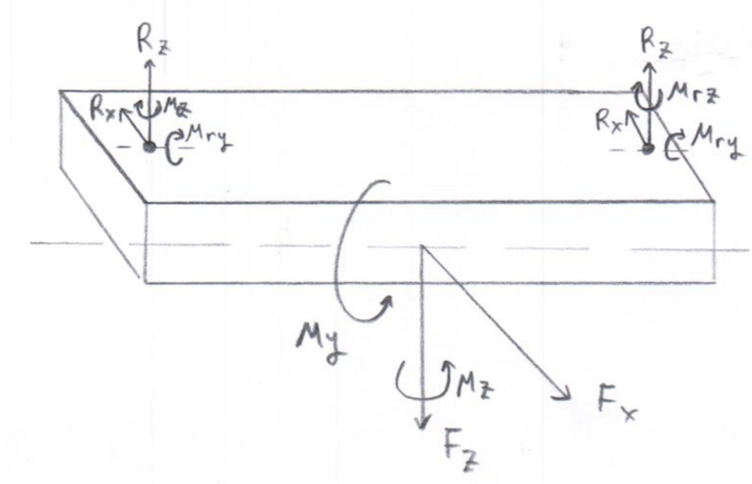


Figure 22: Reaction forces and moments at the fixed points of the spreader beam

The spreader was fixed to the sliding I-beam at a total of four points. Since the spreader was symmetric with respect to both the x-axis and y-axis, we treated the reaction forces and moments at each side of the spreader to be identical to perform the analysis. Based on the assumption, force balance equation stated below were used to solve for the reaction force along the x-axis, R_x.

$$\Sigma F_x = 0 = F_x - 2R_x$$

$$R_x = \frac{F_x}{2} \qquad \text{Eq. 3.1.4.1}$$

The moments induced by the operator about the z-axis and y-axis were analyzed using the same principle with the following moment balance equations.

$$\Sigma M_z = 0 = M_z - 2M_{rz}$$

$$M_{rz} = \frac{M_z}{2} \qquad \text{Eq. 3.1.4.2}$$

$$\Sigma M_y = 0 = M_y - 2M_{ry}$$

$$M_{ry} = \frac{M_y}{2} \qquad \text{Eq. 3.1.4.3}$$

The reaction force in the Z-axis, R_z, was affected by the vertical force, F_z, and the weight of the spreader itself, W_s, as shown in Figure 23.

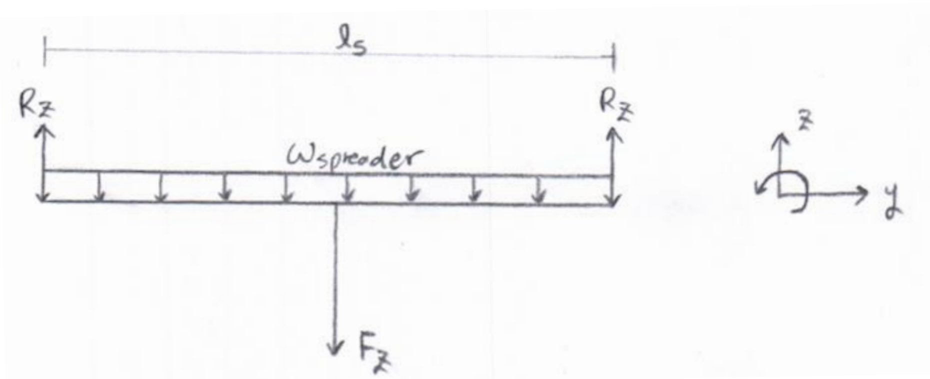


Figure 23: Load Diagram of the Spreader beam in the YZ plane

With both factors, the reaction force in the Z-axis was analyzed with equation 3.1.4.4.

$$\Sigma F_z = 0 = 2R_z - F_z - W_s \cdot l_s$$

$$R_z = \frac{F_z + W_s l_s}{2} \qquad \text{Eq. 3.1.4.4}$$

The calculated reaction forces and moment at the fixture points can be found in TABLE XLII.

TABLE XLII: REACTION FORCES AND MOMENTS AT THE SPREADER BEAM FIXTURE POINTS

	Force [lb]		Moment [lb*ft]	
Values	R_x	R_z	M_{ry}	M_{rz}
	150	315.33	501.15	559

After the reaction forces and moments were found, the Team transferred them to the next section of the design – sliding I-beam.

3.1.4.2 Sliding I-beam Analysis

The I-beam was analyzed in the XZ plane to assess the effect of the reaction force in Z-axis, R_z, and the moment about the Y-axis, M_{ry}. The weight of the I-beam W_I, reaction force R_z and moment M_{ry} were supported by two roller bearings. The roller bearings had been represented as R_1 and R_2 throughout the analysis. The load diagram illustrating the positions of forces and moments can be found in Figure 24.

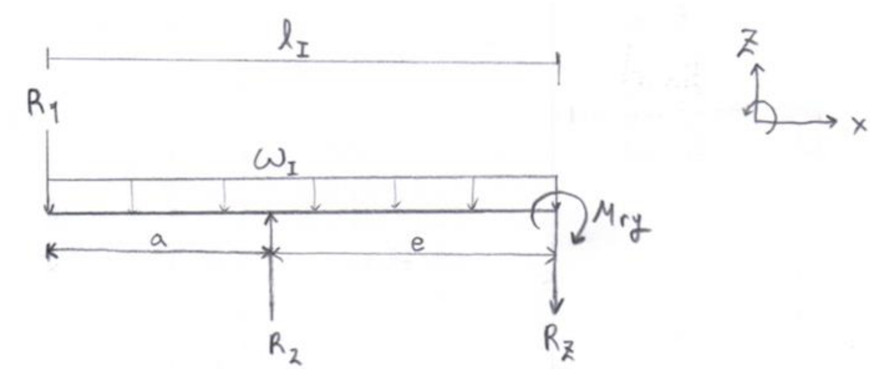


Figure 24: Load Diagram of the I-beam in the XZ plane

From the load diagram, the reaction forces at the first bearing, R_1, and second bearing, R_2, were calculated by using both force balance equation 3.1.4.5 and moment balance equation 3.1.4.6.

$$\sum F_z = 0 = -R_1 + R_2 - R_z - \omega_I l_I \qquad \text{Eq. 3.1.4.5}$$

$$\sum M_1 = 0 = R_2 a - R_z l - M_{ry} - \omega_I l_I \left(\frac{l_I}{2}\right) \qquad \text{Eq. 3.1.4.6}$$

$$R_2 = \frac{R_z l + M_{ry} + \omega_I l_I^2 (0.5)}{a}$$

$$R_1 = R_2 - R_z - \omega_I l_I$$

With the reaction forces, we integrated the load equation once to obtain the shear equation, v(x), along the I-beam. The derived equations are presented in Equation 3.1.4.7 and 3.1.4.8 and the resulting shear diagram is shown in Figure 25.

$$V(x) = V_0 - \int_0^x w(x)\,dx = -R_1 - W_I x \qquad (0 \leq x \leq a)$$

$$V(x) = V_2 - \int_a^x w(x)\,dx = R_2 - R_1 - W_I a - (W_I x - W_I a)$$

$$V(x) = R_2 - R_1 - W_I x \qquad (a \leq x \leq l_I)$$

$$V(x) = \begin{cases} -R_1 - W_I x, & 0 \leq x \leq a \qquad \text{Eq. 3.1.4.7} \\ R_2 - R_1 - W_I x, & a \leq x \leq l_I \qquad \text{Eq. 3.1.4.8} \end{cases}$$

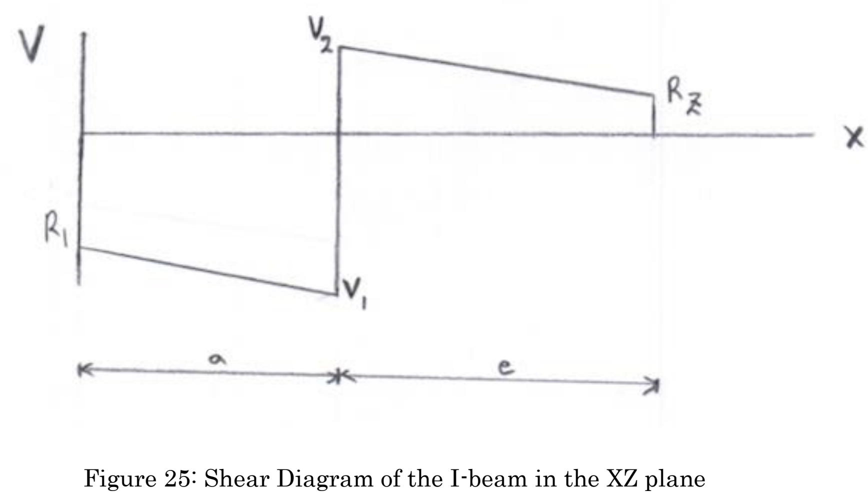


Figure 25: Shear Diagram of the I-beam in the XZ plane

From the shear diagram, we found that the maximum shear force, V_{max}, would be observed at the second bearing location (distance a). By using Equation 3.1.4.7, we calculated the maximum shear force.

$$V_{max} = V_1 = -R_1 - \omega_I a$$ Eq. 3.1.4.9

Then, the Team integrated the shear equation to find the bending moment equation, M(x), along the I-beam. The derived equations along the two sections of the beams are shown in Equation 3.1.4.10 and 3.1.4.11.

$$M(x) = \int_0^x V(x)\,dx = -R_1 x - \frac{\omega_I}{2}x^2 \qquad (0 \le x \le a)$$

$$M(x) = M_a + M_{ry} + \int_a^x V(x)\,dx \qquad (a \le x \le l_I)$$

$$= -R_1 a - \frac{\omega_I}{2}a^2 + M_{ry} + \left[R_2 x - R_1 x - \frac{\omega_I}{2}x^2\right]\Big|_a^x$$

$$= R_2 (x-a) - R_1 x - \frac{\omega_I}{2}x^2 + M_{ry}$$

$$M(x) = \begin{cases} -R_1 x - \frac{\omega_I}{2}x^2, & 0 \le x \le a \\ R_2 (x-a) - R_1 x - \frac{\omega_I}{2}x^2 + M_{ry}, & a \le x \le l_I \end{cases}$$

Eq. 3.1.4.10

Eq. 3.1.4.11

Figure 26 illustrates the bending moment diagram representing the derived equations along the beam.

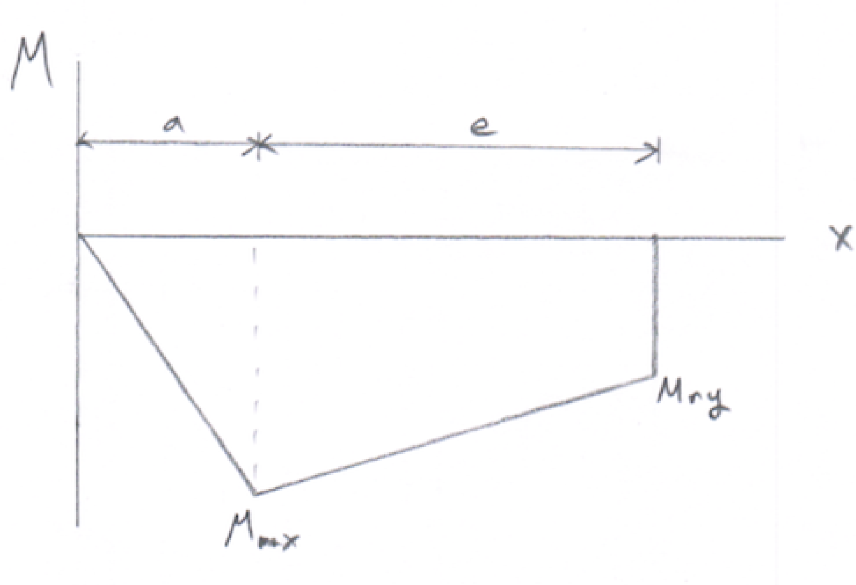


Figure 26: Bending Moment Diagram of the I-beam in the XZ plane

As shown in the bending moment diagram, the maximum moment along the I-beam was observed at the second bearing location. Using Equation 3.1.4.10, the Team found the maximum moment.

$$M_{max} = M(x=a) = R_1 \cdot a + \omega_I \cdot \frac{a^2}{2} \qquad \text{Eq. 3.1.4.12}$$

Once the analysis of the forces and moments in the XZ plane were completed, the I-beam was analyzed in the XY plane to study the effect of the moment M_Z. To perform the analysis, the Team assumed that the bearings would resist the lateral movement when the moment was applied. The visualization of the moment and the reaction forces on the bearings can be found in Figure 27.

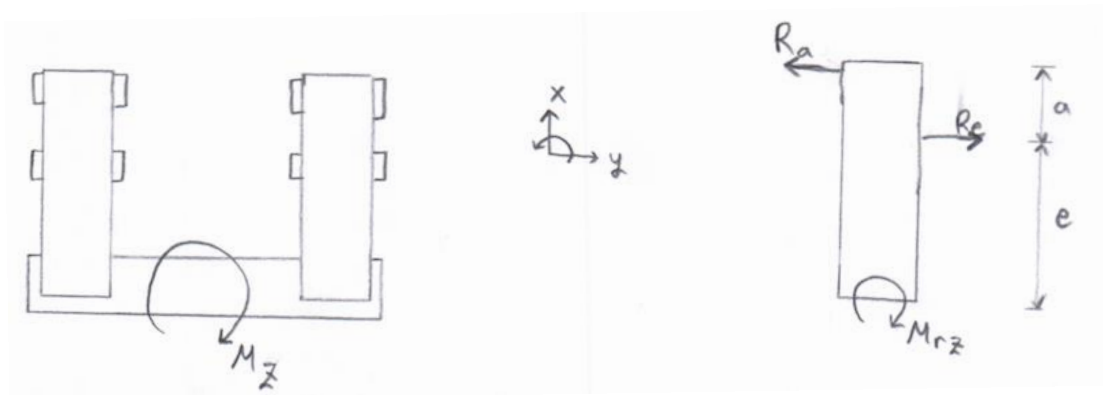


Figure 27: Visualization of moment M_z and the reaction forces

From the load diagram shown in Figure 27, the reaction forces in the y-direction at both bearings, R_a and R_e, were determined by using Equation 3.1.4.13 and 3.1.4.14.

$$\Sigma M_a = 0 = +R_e a - M_z \qquad \text{Eq. 3.1.4.13}$$

$$R_e = +\frac{M_z}{a}$$

$$\Sigma F_y = 0 = -R_a + R_e \qquad \text{Eq. 3.1.4.14}$$

$$R_a = +\frac{M_z}{a}$$

The reaction forces were used to find the distribution of the shear force along the beam. Figure 28 shows the results of the analysis.

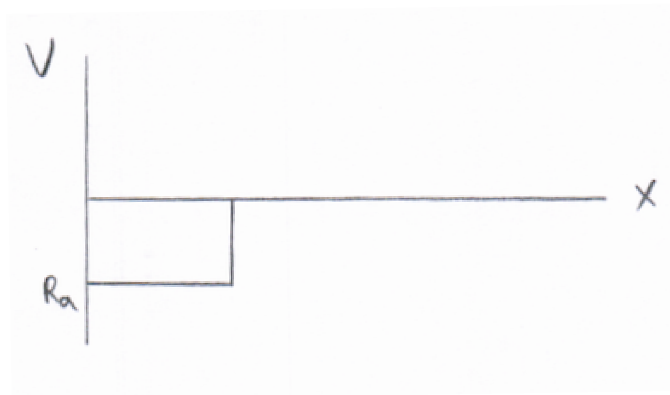


Figure 28: Shear diagram of the I-beam in the xy plane

Since the same shear force was observed in-between the bearings (distance a), the maximum shear force was simply the reaction force R_a. Furthermore, the integration of the shear equation resulted in a linear increase in moment throughout distance "a" as shown in Figure 29.

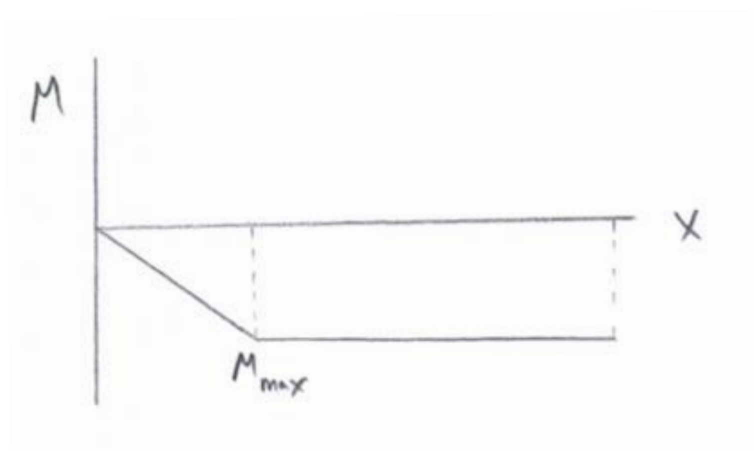


Figure 29: Bending moment diagram of the I-beam in the xy plane

A linear relationship between length and moment resulted in the maximum moment being applied at the second bearing.

$$M_{max} = R_a \cdot a \qquad \text{Eq. 3.1.4.15}$$

With the maximum shear forces and moments found for both XZ and XY planes, the Team completed the analysis for the sliding I-beam. The results of the reaction force calculations can be found in TABLE XLIII and the maximum shear force and moment are shown in TABLE XLIV.

TABLE XLIII: SELECTED I-BEAM AND RESULTING REACTION FORCES

	Reaction Force [lb]				Length [ft]	
Beam	R1	R2	Ra	Re	a	L_I
S8 x 18.4	1085.69	1488.60	376.43	376.43	1.485	4.31

TABLE XLIV: MAXIMUM SHEAR FORCE AND MOMENT CALCULATED FOR THE I-BEAM

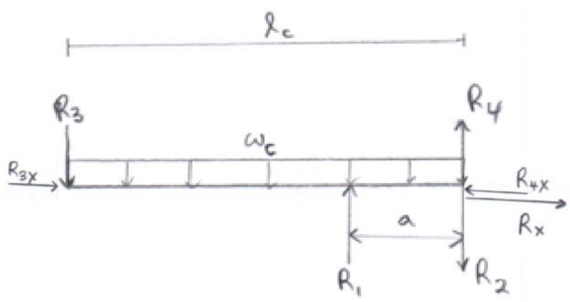
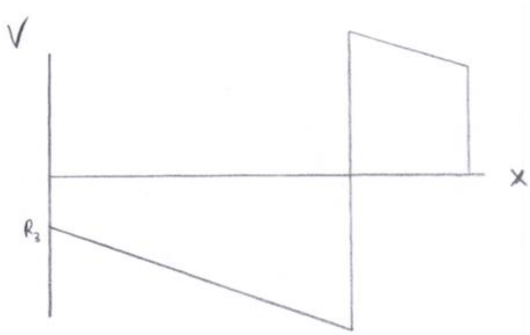
	XZ plane		XY plane	
Beam	$V_{max,xz}$ [lb]	$M_{max,y}$ [lb*ft]	$V_{max,xy}$ [lb]	$M_{max,z}$ [lb*ft]
S8 x 18.4	1113.01	3743.78	376.43	559

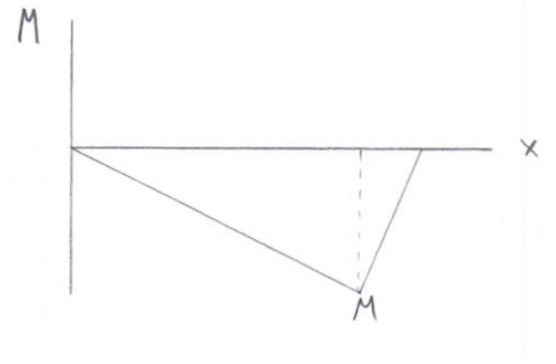
The reaction forces at the two bearings were transferred to the C-channel bracket as the bearings were the points of contact between the I-beam and C-channel bracket.

3.1.4.3 C-channel Analysis

The reaction forces on the roller bearings along the z-axis and y-axis were transferred to the C-channel to analyze the beams. Using the same design methodologies described in the I-beam section, the Team developed the load, shear and bending moment diagrams to determine the reaction forces on the C-channel and the resulting bending and shear stresses. The diagrams and corresponding analysis results in the XZ plane can be found in TABLE XLV.

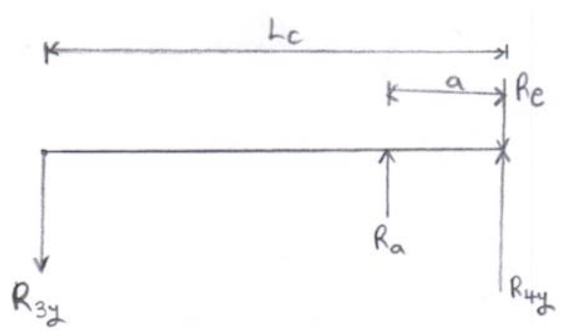
TABLE XLV: FORCE AND STRESS ANALYSIS IN THE XZ PLANE

Diagrams	Equations
	$\Sigma M_R = 0 = R_3 l_c + w_c l_c\left(\frac{l_c}{2}\right) - R_1 a$ $R_3 = \frac{R_1 a - w_c l_c^2/2}{l_c}$ $\Sigma F_z = 0 = -R_3 + R_4 - w_c l_c + R_1 - R_2$ $R_4 = R_3 + w_c l_c + R_2 - R_1$ $R_{4x} = \frac{R_4}{\tan\theta}$ $\Sigma F_x = 0 = R_{3x} + R_x - R_{4x}$ $R_{3x} = R_{4x} - R_x$
	$V(x) = -R_3 - w_c x$ $V_{max} = -R_3 - w_c(L_c - a)$

Diagrams	Equations
	$M(x) = \int_0^x V(x)\,dx$ $M(x) = -R_3\,x - W_c\,\frac{x^2}{2}$ $M_{max} = -R_3\,(L_c - a) - W_c\,\frac{(L_c - a)^2}{2}$

After analyzing the C-channel bracket in the XZ plane, the Team assessed the effects of the reaction forces R_a and R_e in the XY plane. The analysis process is shown in TABLE XLVI.

TABLE XLVI: FORCE AND STRESS ANALYSIS IN THE XY PLANE

Diagrams	Equations
	$\Sigma M_R = 0 = R_{3y}\cdot L_c - R_a\cdot a$ $R_{3y} = \frac{R_a\cdot a}{L_c}$ $\Sigma F_y = 0 = R_{4y} - R_{3y} + R_a - R_e$ $R_{4y} = R_{3y} - R_a + R_e$

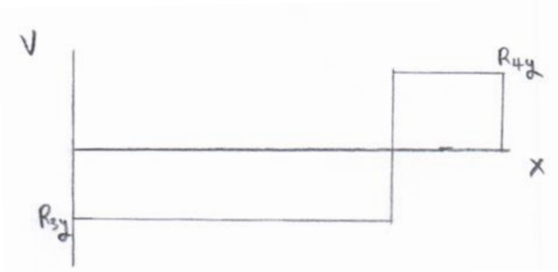
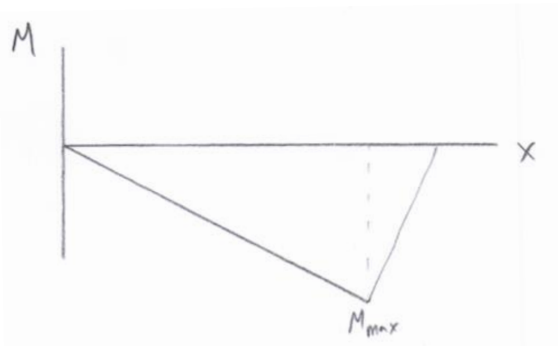
Diagrams	Equations
	$V_{max} = R_{3y}$
	$M_{max} = R_{3y}(L_c - a)$

TABLE XLVII shows the resulting forces on the C-channel bracket using the identified length and TABLE XLVIII summarizes the maximum shear force and moment calculation results.

TABLE XLVII: SELECTED C-CHANNEL AND RESULTING REACTION FORCES

	Length [ft]	Reaction Force at Fixed End [lb]			Reaction Force at Free End [lb]		
Beam	**L_c**	**R3**	**R3x**	**R3y**	**R4**	**R4x**	**R4y**
C9 x 13.4	6.55	158.37	1874.42	85.34	736.73	2049.07	85.34

TABLE XLVIII: MAXIMUM SHEAR FORCE AND MOMENT CALCULATED FOR C-CHANNEL

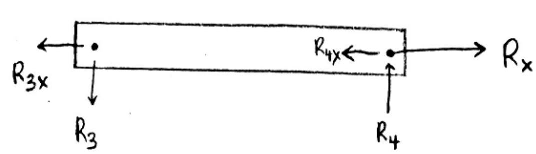
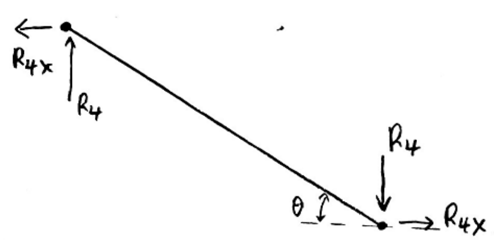
	XZ plane		XY plane	
Beam	**$V_{max,xz}$ [lb]**	**$M_{max,y}$ [lb*ft]**	**$V_{max,xy}$ [lb]**	**$M_{max,z}$ [lb*ft]**
C9 x 13.4	1108.318	1145.93	85.34	432.26

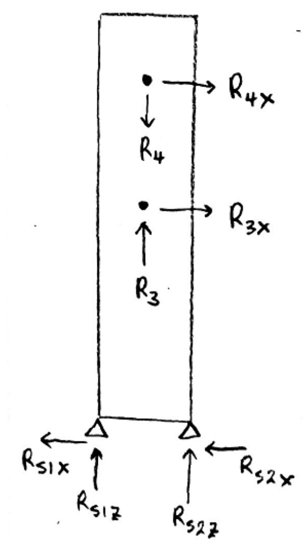
The 3D reaction forces at points 3 and 4 were transferred from the C-channel bracket to the columns to analyze the effect of the forces on the column.

3.1.4.4 Column Analysis

The column in the facility needs to be analyzed to ensure that the column can withstand the additional weight of the design components. To assess the column, the reaction forces shown in TABLE XLVII are transferred to the columns through two support members. As described in Section 3.1.4, an adjustable tie rod holds the free-end of the C-channel bracket in place while the other end is fixed directly onto the columns with pin supports. TABLE XLIX shows how the forces from the C-channel bracket are transmitted to the columns. The tabulated governing equations demonstrate how the reaction forces at the support are calculated, where l_{co} is the width of the column, h_3 is the height of the R_3 pin support location and h_4 is the height of the R_4 pin support location.

TABLE XLIX: TRANSMITTAL OF REACTION FORCES FROM THE BRACKET TO COLUMN SUPPORTS

Parts	Drawings	Governing Equations
C-channel bracket		N/A
Tie rod		N/A

Parts	Drawings	Governing Equations
Column		$\sum M_{s1} = R_{s2} l_{co} + R_3 \frac{l_{co}}{2} - R_4 \frac{l_{co}}{2} + R_{3x} h_3 - R_{4x} h_4 - \omega_{co} \frac{l_{co}}{2}$

The angle of the tie rod, θ, was desired to be as small as possible to decrease the overall footprint. From using the smallest allowable angle determined in Section 3.1.1, the Team identified the reaction forces at the base of the columns. The result of the analysis and the final reaction forces of the structure are shown in TABLE L.

TABLE L: THE COLUMN DIMENSIONS AND REACTION FORCES AT 20 DEGREES

Column dimension [in]			Reaction Force [lb]			
Width	Depth	Thickness	R_{s1x}	R_{s1z}	R_{s2x}	R_{s2z}
5.13	17.25	0.25	75	-2191.90	75	3936.15

With the column analysis, the support structure analysis was completed. Since the Team knew the parameters and dimensions of the design components, the Team performed in-depth stability analysis to test the feasibility of the design.

3.1.5 Stability Analysis

Once the basic design parameters were determined through the analysis with a maximum sheet size of 5 ft., the Team conducted a stability analysis of the design with the worst-case loading scenario. The purpose was to evaluate whether the newly designed structure would lose balance in the worst setting. The analysis led to the machine being unstable in use, which created a safety concern for those in the vicinity of the machine.

The stability analysis was performed by applying a single governing equation on the overall structure. Figure 30 shows the schematics of the structure where F_o and F_{end} represent the two essential support forces.

$$\sum M = Fd \qquad \text{Eq. 3.1.4.16}$$

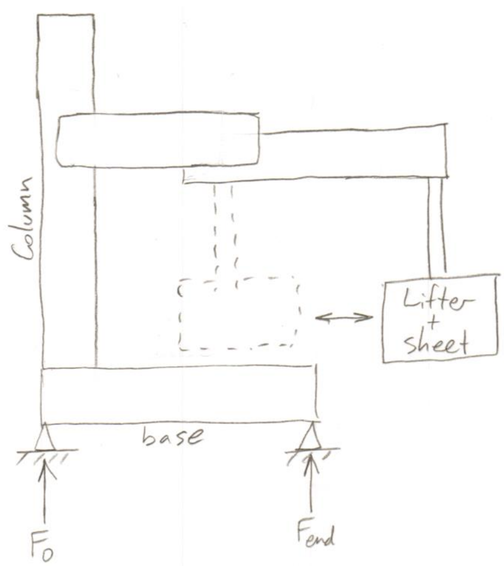


Figure 30: Stability analysis of the structure

All of the structures including the lifter and sheet assembly, structural beams and columns were converted into a point load positioned at the centre of each application point. TABLE LI shows the overall composition of the structure with the associated weights and distances from the origin, F_o, tabulated as well. In the table, "max lift force" represents the maximum force applied while picking up a sheet by the vacuum lifter and "stopping force" represents the force required by dampers to stop the sliding vacuum lifter. For "stopping force," "out" means it is the scenario where the machine is moving away from the column and "in" represents the machine when it moves towards the column.

TABLE LI: STABILITY ANALYSIS SETUP

	QTY	UNIT WEIGHT (LB)	WEIGHT (LB)	CENTER (IN)
Column	2	1165.8	2331.6	0
C-channel bracket	2	198.8	397.5	44.5
Sliding I-beam	2	82.8	165.6	74
Rollers and other parts	1	120.0	120.0	44 to 96 (retracted or extended)
Weight of vacuum lifter	1	572.3	572.3	
Weight of remaining lifter assembly	1	262.5	262.5	

With the values in TABLE LII, we took the sum of the moments about each supporting location – once at F_o and another at F_{end} – to calculate the support loads. Additionally, the Team considered three extreme loading scenarios that could induce instability.

TABLE LII: THREE LOADING CASES FOR STABILITY ANALYSIS

Scenarios	**Load**	**Loading Case Description**
Max lift force	1414 lb downward vertical load applied 96in away from column.	Maximum lifting force from cylinder at maximum sliding beam extension with a factor of safety of 2 for the lifting force of 707 lb.
Stopping force out	900 lb force directed away from the columns and applied at the height of the raised vacuum lifter.	Maximum deceleration force from dampers as the lifter slides away from the column with a factor of safety of 3 for the deceleration force (not initial velocity).
Stopping force in	900 lb force directed towards the columns and applied at the height of the raised vacuum lifter.	Maximum deceleration force from dampers as the lifter slides towards the column with a factor of safety of 3 for the deceleration force (not initial velocity).

The basic principle of the stability analysis was that neither one of the two support loads, F_o and F_{end}, had to be zero. The structure would become more unstable as one of the two support forces approached zero. At its best, the two supports would be perfectly balanced in all of the cases resulting in the structure being stable. The results from the force/moment balance calculations for the 3 scenarios are shown in TABLE LIII.

TABLE LIII: RESULT OF STABILITY ANALYSIS USING 4 FT. MAXIMUM SHEET SIZE

Scenarios	Total Moment	F_0	F_{end}
Max lift force	16,867 ft-lb	1715 lb	2977 lb
Stopping force out	13,958 ft-lb	1386 lb	2463 lb
Stopping force in	1454 ft-lb	3593 lb	257 lb

The first two scenarios are shown to have large moments, but they are reasonable according to typical racking system use. For example, four 2000 lb stacks of 5ft wide sheet placed on that racks would equate to roughly 24,000 ft-lbs. However, the third scenario has a relatively low force component on the end of the beam base. The team recommends using a double ended column to improve stability in this direction.

We initially attempted to design the machine to lift a 5ft wide, 600 lb sheet as set out in the target specifications. However, as the design phase progress and the stability analysis was developed, we found that the factor of safety was impermissibly low for the level of uncertainty in the calculations. The instability led to the lifting assembly to be reworked which required the support structure to also be reworked. This process of design-analysis-adjust was completed numerous times until the Team determined an acceptable balance between functionality, cost, and safety was reached.

3.2 Component Selection

This section of the report discusses the non-designed parts of the machine. Both the lifter and support structure provide the framework for the machine to achieve its goal of transferring sheets. However, for the machine to be able to function within the given framework, the design requires specific components. The non-designed parts include the roller the I-beam uses to slide in the C-channel bracket, damper which slows down the velocity of the sliding beam at both ends, ANVER vacuum lifter products and the air knife fanners to achieve sheet separation.

3.2.1 Roller

Each sliding I beam contains four identical tapered rollers. Each tapered roller is to be installed with a press-fit mate between the roller and roller install bracket. The rollers are then secured using a locknut as indicated by the manufacturer's specifications. A tapered roller was selected for the final design due to the self-centering properties of tapered cams when used in pairs on a tapered surface. For the machine, the CIR-3.00-R tapered roller was selected from PCI Manufacturing. The roller was selected by analyzing the loads the stud would see product specifications as listed in TABLE LIV.

TABLE LIV: SPECIFICATIONS FOR CIR-3.00E-R SUPPLIED BY PCI MANUFACTURING [26]

Part Number	Basic Radial Dynamic Load Rating (lbs)	Basic Radial Static Load Rating (lbs)	Thrust Dynamic Load Rating (lbs)	Thrusts Static Load Rating (lbs)	Stud Capacity (lbs)	Retaining Ring Capacity (lbs)
CIR-3.00E-R	3624	2136	1563	534	3020	900

To determine the viability of the rollers a shear and moment diagram was generated to assist in the creation of equations to determine the loads experienced by each roller. The sliding I beam was treated as a beam with a force applied downwards due to the weight of the lifter and two reaction forces found at each location of the roller as discussed in Section 3.1.4.2.

The maximum reaction forces for the rollers were found to be 376.4 lbs. This maximum reaction force was then compared to the thrust loads and was found to be smaller so the CIR-3.00E-R was determined to be sufficient for the application. The 3D CAD model of the bearing is shown in Figure 31.

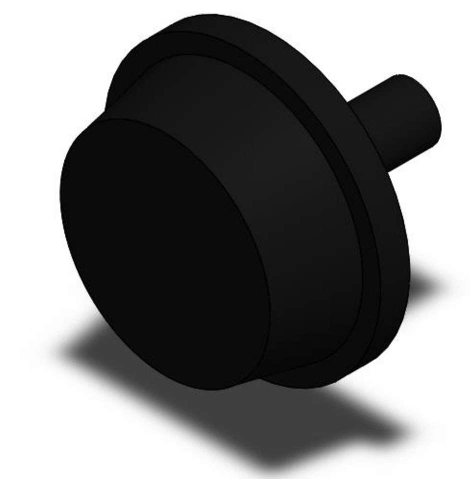


Figure 31: Solidworks model of the CIR-3.00E-R tapered roller manufactured by PCI

To hold the upper tapered roller bearings in place the Team designed a roller support bracket that is to be machined out of plain carbon steel. It was determined that the roller support will be bolted to the I beam and not permanently welded to the structure to allow for easy assembly and maintenance of the part. The bracket can be seen in Figure 32.

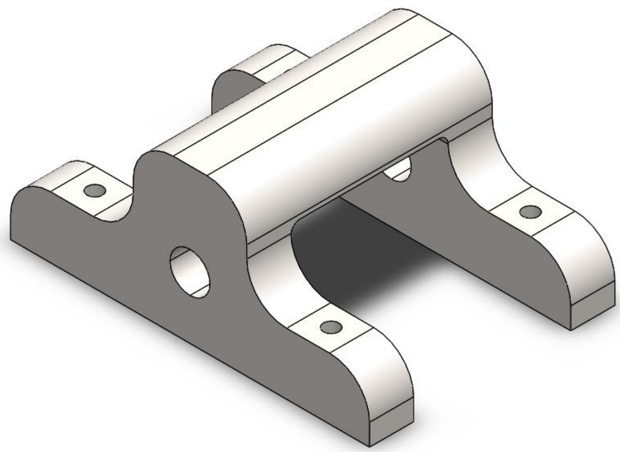


Figure 32: Render of the upper tapered roller bearing support bracket

The final design also calls for two more tapered roller bearings to be installed along the web of the I beam to provide vertical support to the sliding I beam mechanism. This bracket is to be machined out of plain carbon steel and will also be fastened to the I beam and not welded for servicing and manufacturing reasons. The lower roller mount can be seen in Figure 33.

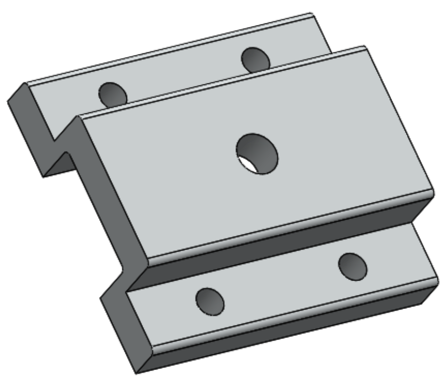


Figure 33: Render of the lower tapered roller bearing support bracket

3.2.2 Damper

The Team determined that the design requires four dampers to slow down the sliding I beam assembly and held mitigate one of the failure modes identified. The damper was determined by first determining the kinetic energy the damper would be required to absorbed using Equation 3.2.2.1.

$$E_K = \frac{W}{772} \times V^2$$ Eq. 3.2.2.1

With the kinetic energy of the damper determined a stroke length of 3 inches was selected by the Team. The stroke length and kinetic energy was then used to calculate the work energy of the damper by using Equation 3.2.2.2.

$$E_W = F_D \times S$$ Eq. 3.2.2.2

The kinetic energy and work energy were then summed up to determine the total energy requirements for the damper using Equation 3.2.2.3.

$$E_T = E_K + E_W$$ Eq. 3.2.2.3

Finally, the shock force value for the damper could be determined using Equation 3.2.2.4.

$$F_P = \frac{E_T}{S \times 0.85}$$ Eq. 3.2.2.4

With the assumption made that the weight of sliding assembly would be 850 lbs, the stroke length being 3 inches, and finally the velocity of the sliding assembly would not be exceeding 2 ft/s. Using the above equations, the values found at each step are outlined in TABLE LV.

TABLE LV: SUMMARY OF VALUES FOR DAMPER CALCUATIONS

<u>Variable</u>	<u>Value</u>
Kinetic Energy	32.2 in-lbs
Work Energy	150 in-lbs
Total Energy	784.2 in-lbs
Shock Force	307.5 lbs

The Team selected a non-adjustable mid-bore hydraulic damper, part number, PMXT 1575, from ITT Enidine.

[redacted]

Figure 34: OEM picture of the PMXTT 1575 damper from ITT Enidine [27]

3.2.3 Vacuum lifter and Generator

Two ANVER products were used in the design to lift the sheet metals and to provide the required vacuum to lift the sheet metals. The first product was the lifting frame. ANVER supplied varieties of lifting frames with options between in-line and cross-arm suction cup configurations. For the purpose of stabilizing the sheet metal once it was picked up by the machine, the Team decided to use the cross-arm configuration. Additionally, in order to provide with the option of increasing the capacity of the machine in the future, the longest eight cup adjustable cross-arm lifting frame was selected. With these design factors in mind, to reduce the overall weight of the lifter assembly, the Team selected the lifting frame model L80M8-110-4/44 as it was capable of withstanding the lifter weight while being low weight.

The second product was the vacuum generator. For the purpose of reducing the weight of the lifter assembly in general, the Team used VPF-57R-AC for the design. The particular component was selected because the part was capable of handling up to 2200 lb of material while being low weight and consuming less power than the alternative. TABLE LVI shows the specifications of the selected components.

TABLE LVI: SPECIFICATIONS OF THE SELECTED ANVER PRODUCTS [22], [28]

	Component #	Rated Load Capacity	Unit Weight	Incoming Power Requirement AC
Lifting Frame	L80M8-110-4/44	800 lb	190 lb	N/A
Vacuum generator	VPF-57R-AC	2200 lb	80 lb	115V/1PH/60Hz – 6 Amps

3.2.4 Air Knife Fanner

To separate both ferrous and non-ferrous materials and prevent sheet sticking the Team chose to incorporate an air knife, magnetic fanner combo which is designed and sold by Industrial Magnets Incorporation, part number AKF12. The reason the AKF12 was selected was due to the height of the unit and it being usable for all the varying sheet sizes. The air knife fanner would be installed underneath the platform in line with the platform legs to avoid the risk of the forklift pressing onto the air knife fanner. The specifications of the selected air knife fanner are shown in TABLE LVII.

TABLE LVII: TECHNICAL SPECIFICATION OF AIR KNIFE FANNER AKF12 [23]

Part number	Overall Height	# of Air knifes	Weight	Operating Pressure
AKF12	12 inch	5	35 lb.	10 – 100 psi

3.2.5 Pneumatic Cylinder

A large pneumatic cylinder was used to provide the lifting force of the machine. Pneumatic was chosen to provide linear actuation because it was more inexpensive than hydraulics and electric. The size of cylinder was selected so that it would not damage the structure in the event that an overweight sheet was lifted or the sheets stuck together. At a line pressure of 100 psi, the cylinder can exert 707 lbs of force which was considered during the stress analysis of the machine structure. The machine also required a relatively long, 28 in stroke. This was to accommodate the change in elevation from the air knives to the top of the loading table, as well as to account for up to 5” of sheet sag when lifting.

The cylinder was sourced from McMaster-Carr. The major specifications for the air cylinder are listed in TABLE LVIII.

TABLE LVIII: SPECIFICATIONS FOR THE PNEUMATIC CYLINDER [29]

Part Number	Bore	Stroke	Force at 50 psi	Force at 100 psi	Air Cushioned?	Actuation
6491K673	3 inch	28 inch	354 lb	707 lb	Adjustable	Double Acting

3.2.6 Linear Bearings

Two sets of linear bearings are used to guide the vertical motion of the lifting mechanism. Each set consists of two bearing carriages and a vertically positioned bearing rail. One of the bearing sets allows for one millimeter of lateral shift, or “float”, to account for misalignment between the two bearing rails. These particular bearings were selected based on their high load bearing capacity and low cost. These bearings are supplied by a local company, Igus. TABLE LIX summarizes the selected bearing products and their capacities.

TABLE LIX: SPECIFICATIONS FOR THE IGUS COMPONENTS [30]

Description	Part Number	QTY	Capacity (According to machine coordinate system)
Heavy duty linear bearing rail, 1.3m	TS-01-30	2	N/A
Heavy duty linear bearing carriage	TW-02-30	2	Fx=3140 lb, Fy=1570 lb, Mx=74 ft-lb , My= 74 ft-lb, Mz=148 ft-lb
Heavy duty linear bearing carriage, floating	TW-02-30LLz	2	Fx=3140 lb, My= 74 ft-lb, Mz=148 ft-lb

3.3 Preliminary Engineering Failure Analysis

From the worst-case scenario, the stress was calculated analytically using hand calculations that were verified and optimized using spreadsheets. For components of high critical stress or components that hand calculations would be difficult to complete on, the Team opted to use finite element analysis to determine the stress, deflection, and factor of safety values for the components.

The structural performance of the machine was heavily dependent on the materials that were selected. ASTM A36 steel and AISI 1020 steel were the primary materials used for the machine. Aluminum alloy 6061-T6 was used very sparingly where appropriate. The properties of these materials can be found in TABLE LX.

TABLE LX: PRIMARY MATERIAL PROPERTIES

Material	Density (lb/in^3)	Modulus of Elasticity (ksi)	Yield Strength (ksi)
ASTM A36 [31]	0.284	29,000	36.0
AISI 1020 [32]	0.284	29,000	51.0
AA 6061-T6 [33]	0.098	10,100	35.0

These material properties were used throughout the entire engineering analysis of the machine. Additionally, the target factor of safety was 8. We intentionally set a high target safety factor knowing that the maximum stresses in the most complex components would be unknown until the finite element analysis was performed.

3.3.1 Theory

Material under forces and moments experience stress which is the amount of force applied on the cross-sectional area of the material. When the force is acting perpendicular to the area, the stress is designated as normal stress with tensile and compressive stresses implying that the material is in tension and compression. On the other hand, when the force is applied on a surface parallel to the direction of the force, it is called as shear stress.

Both normal and shear stresses induce failure modes in the affected materials in a variety of ways. The types of engineering failure modes that could occur for the design and the details of each failure mode can be found in TABLE LXI.

TABLE LXI: VARIOUS ENGINEERING FAILURE MODES

Failure Mode	Descriptions
Yielding	Plastic deformation of material. Once the stress level on the material surpasses the yield strength of a material, the deformed shape of it does not return to the original state, resulting in loss of material property.
Fatigue	Material under continuous repeated motion can fail and fracture despite the stress level being below the yield strength. As the number of repeated cycle increases, the stress level required for the material to fail by fatigue is decreased.
Deflection	Vertical or horizontal displacement of the material which is induced by a significant load applied at a free end of the material. The further away from the fixed point the free end is, the more susceptible the material is to deflection.
Buckling	Sudden failure of the material due to strong compressive force. The failure occurs in a way such that positions in-between the nodes bend in the direction perpendicular to the compressive force.

Since some failure modes only affected components that were under specific conditions, the Team did not perform analysis of all of the failure modes for each component. Rather, the Team identified which failure modes would be applicable to which parts and analyzed the parts selectively. Out of the four failure modes discussed in TABLE LXI, yielding and fatigue applied on all of the main components as the entire machine was under loads and moments. However, deflection was only applicable to the fully extended I-beam as the other parts of the design were fixed. Buckling only affected components under compression which were only the columns in the design.

3.3.2 Von Mises Yield Criterion

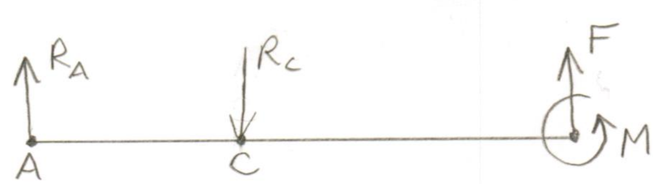
The Von Mises yield criterion was used to evaluate the likelihood of yielding failure. The equation is shown below.

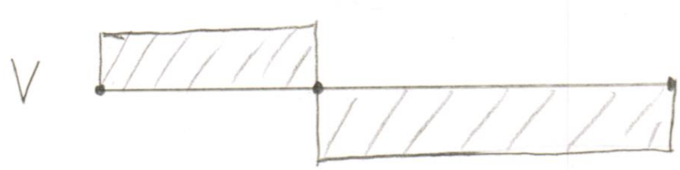
$$\sigma_{VM} = \sqrt{\frac{(P_1 - P_2)^2 + (P_2 - P_3)^2 + (P_3 - P_1)^2}{2}} \qquad \text{Eq. 3.3.2.1}$$

The maximum stresses were found using the loads derived in Section 3.1. The process for finding the maximum stress in the steel members started with the shear force and bending moment diagrams. The maximum shear stress was found from the shear force, while the maximum normal stress was found from the bending moments. The principal stresses (P_1, P_2, and P_3) were determined from the coordinate-based normal and shear stresses using either the stress tensor technique or Mohr's Circle. Equations and sample calculations will be provided throughout the analysis section.

3.3.2.1 Lifter Assembly

The two areas of interest for the lifter assembly were the bearing rail supports and the spreader beam. The shear force (V) and bending moment (M) diagrams for the bearing rails are below. The diagrams were produced from the free body diagrams and the calculated loads for loading case 3A from Section 3.1.1 as it was the worst-case scenario.





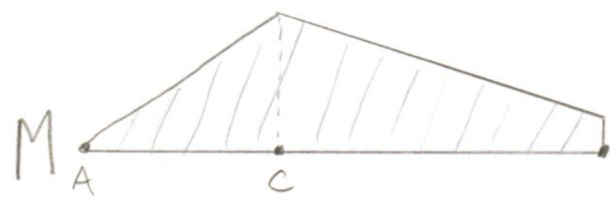


Figure 35: Shear force (V) and bending moment (M) diagrams for the bearing rails

These diagrams were applicable to loading of the rails in the Y-direction and the X-direction. The diagrams indicated that the maximum shear force and bending moments were both located on bearing carriage C. The shear stress at point C was calculated from the reaction force R_c and the cross-sectional area of the rail. The maximum normal stress was found using the below equation where "M" was the moment at point C, "I" was the second moment of area of the rail, and "c" was half the width of the rail (equivalent to the maximum distance from the neutral axis).

$$\sigma_{max} = \frac{M \cdot c}{I} \qquad \text{Eq. 3.3.2.2}$$

The second moment of area (I) and the distance to the neutral axis (c) were calculated based on 3 inch square tube according to the final design. From this approach, the maximum shear stress and maximum bending stress with the bearing rail for case 3A were found. The results are shown in TABLE LXII.

TABLE LXII: STATE OF STRESS IN BEARING RAIL FOR CASE 3A

Shear Stress σ_{xy}	**Bending (Normal) Stress** σ_z
0.88 ksi	11.42 ksi

A similar approach was used to determine the maximum stresses in the spreader beam. Again, the Team only considered scenario 3A as it was expected to induce the highest stress. The loading on the spreader beam was significantly more complex, however they equated to a relatively simple state of stress since the shear stress in the YZ plane was negligible. The results are summarized in TABLE LXIII.

TABLE LXIII: STATE OF STRESS IN THE SPREADER BEAM FOR CASE 3A

Shear Stress σ_{zx}	**Shear Stress** σ_{zx}	**Bending (Normal) Stress** σ_{y}
3.24 ksi	Negligible.	6.92 ksi

As indicated by the subscripts, these states of stress were dependant on the XYZ-coordinate system we established at the beginning of the design process. In reality, the highest stress values are independent of our established coordinate system. The highest stress values are represented by the principal stresses (P_1, P_2, P_3). The Team converted the stress values into the three principal stresses for both the spreader beam and bearing rail by using Mohr's circle. The figure below shows this graphical method for the bearing rail state of stress.

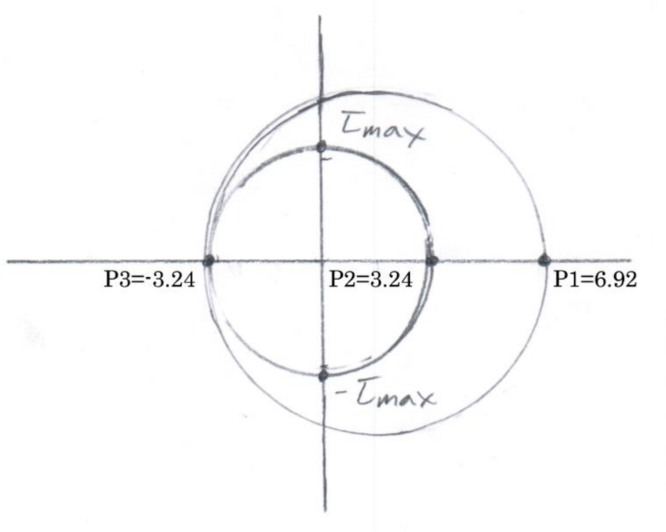


Figure 36: Mohr's circle showing the graphical method of determining the principal stresses

The principal stresses in the spreader beam and bearing rails for loading case 3A are tabulated in TABLE LXIV.

TABLE LXIV: THE PRINCIPAL STRESSES WITHIN THE LIFTER ASSEMBLY STRUCTURE

	3rd Principal Stress	**2nd Principal Stress**	**1st Principal Stress**
Spreader Beam	-3.24 ksi	3.24 ksi	6.92 ksi
Bearing Rail	-0.88 ksi	0.88 ksi	11.42 ksi

These principal stresses were used to calculate the Von Mises stresses based on equation 3.3.2.1. The Von Mises stresses for the spreader and bearing rail were 8.91 ksi and 11.52 ksi respectively. The yield stress for these components was 36.0 ksi as they were made from A36 structural steel. Therefore, the factor of safety for the spreader beam was 4.04 and the factor of safety for the bearing rail was 3.13. These were lower than the target safety factor because the machine needed to remain relatively lightweight. The team discovered that increasing the strength, and therefore weight, of these components would have a significant detrimental effect on the machine stability.

3.3.2.2 Support Structure

From the forces and moments calculated in Section 3.1.4, the Team calculated the resulting normal and shear stresses to begin the analysis. For the I-beam, since the shear would be carried mostly by the web, we took the cross-sectional area of the web to analyze the resulting shear stress, τ_{ZX}, from the maximum shear force found. The cross-sectional dimension of the I-beam used for calculating the web area is shown in Figure 37.

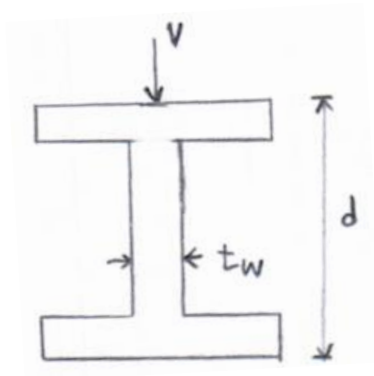


Figure 37: Cross-sectional view of the I-beam

$$A_{web} = t_w \cdot d$$

$$\tau_{zx} = \frac{V_{max}}{A_{web}} \qquad \text{Eq. 3.3.2.3}$$

By using the maximum moment, the bending stress, σ_z, was calculated, where S_{yy} was the section modulus of the beam along the y-axis.

$$\sigma_{bending} = \sigma_z = \frac{M_{max}}{S_{yy}} \qquad \text{Eq. 3.3.2.4}$$

For the xy plane of the I-beam, the maximum shear stress, τ_{YX}, and the bending stress, σ_y, was calculated by using the equations below, where S_{zz} was the section modulus of the beam in the z-axis.

$$\tau_{yx} = \frac{V_{max}}{A_{web}} \qquad \text{Eq. 3.3.2.5}$$

$$\sigma_{bending} = \sigma_y = \frac{M_{max}}{S_{zz}} \qquad \text{Eq. 3.3.2.6}$$

Although the reaction force in the X-axis, F_x, was omitted in the load diagrams as it did not have direct impact on the reaction forces in the XY and XZ planes, the force F_x was still acting on the beams. By taking the force into account, the normal stress in the x-axis could be solved with the following equation, where A_{beam} was the cross-sectional area of the I-beam.

$$\sigma_x = \frac{F_x}{A_{beam}} \qquad \text{Eq. 3.3.2.7}$$

The variables of the analysis are listed in TABLE LXV and the calculation results are shown in TABLE LXVI.

TABLE LXV: VARIABLES USED FOR THE I-BEAM STRESS ANALYSIS

	XZ plane			XY plane	
Beam	**$V_{max,xz}$ [lb]**	**$M_{max,y}$ [lb*ft]**	**F_x [lb]**	**$V_{max,xy}$ [lb]**	**$M_{max,z}$ [lb*ft]**
S8 x 18.4	1113.01	3743.78	150	376.43	559
	A_{web} [in²]	**S_{yy} [in³]**	**A_{beam} [in²]**	**A_{web} [in²]**	**S_{zz} [in³]**
	2.168	13.6	5.41	2.168	1.84

TABLE LXVI: RESULTS OF THE I-BEAM STRESS ANALYSIS

	τ_{ZX} [psi]	σ_z [psi]	σ_x [psi]	τ_{YX} [psi]	σ_y [psi]
Results	513.38	3303.34	27.72	173.63	3645.65

For the C-channel bracket, with the maximum shear forces and moments, the corresponding shear and bending stresses were found from the equations stated below. The first two equations accounted for the XZ plane and the last two accounted for the XY plane.

$$\tau_{zx} = \frac{V_{max}}{A_{web}} = \frac{V(x = L_c - a)}{A_{web}} \qquad \text{Eq. 3.3.2.8}$$

$$\sigma_{bending} = \sigma_z = \frac{M(x = L_c - a)}{S_{yy}}$$ Eq. 3.3.2.9

$$\tau_{yx} = \frac{V_{max}}{A_{web}} = \frac{R_{3y}}{A_{web}}$$ Eq. 3.3.2.10

$$\sigma_{bending} = \sigma_y = \frac{M_{max}}{S_{zz}}$$ Eq. 3.3.2.11

Similar to the I-beam stress analysis, the variables are listed in TABLE LXVII and the analysis results are shown in TABLE LXVIII.

TABLE LXVII: VARIABLES USED FOR THE C-CHANNEL STRESS ANALYSIS

	XZ plane			XY plane	
Beam	$V_{max,xz}$ [lb]	$M_{max,y}$ [lb*ft]	F_x [lb]	$V_{max,xy}$ [lb]	$M_{max,z}$ [lb*ft]
S9 x 13.4	1108.318	1145.93	150	85.34	432.26
	A_{web} [in^2]	S_{yy} [in^3]	A_{beam} [in^2]	A_{web} [in^2]	S_{zz} [in^3]
	2.097	10.6	7.88	2.097	0.954

TABLE LXVIII: STRESS ANALYSIS RESULTS FOR THE C-CHANNEL

	τ_{ZX} [psi]	σ_z [psi]	σ_x [psi]	τ_{YX} [psi]	σ_y [psi]
Results	528.53	1297.28	19.04	40.70	5437.29

For the columns, the Team determined that the loads on the columns could be summarized by the reaction forces at the two pin supports, R_3 and R_4. To see the effect of two reaction forces acting together on the columns, the normal and shear stresses were calculated taking both forces into account.

$$\sigma_z = \frac{R_3}{A} + \frac{R_4}{A}$$ Eq. 3.3.2.12

$$\tau_{xz} = \frac{R_{3y}}{A} + \frac{R_{4y}}{A}$$ Eq. 3.3.2.13

$$\tau_{yz} = \frac{R_{3x}}{A} + \frac{R_{4x}}{A}$$ Eq. 3.3.2.14

TABLE LXIX shows the summary of variables used for the analysis and TABLE LXX tabulates the stress analysis results.

TABLE LXIX: VARIABLES USED FOR THE COLUMN ANALYSIS

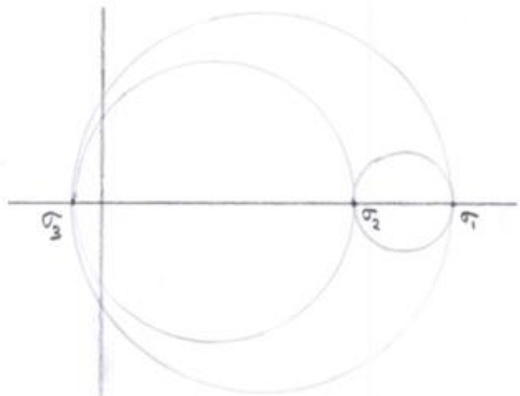
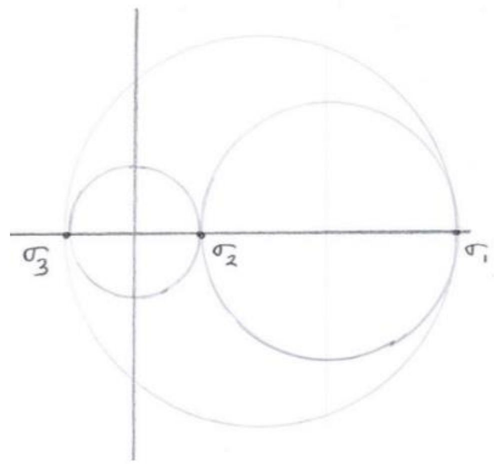
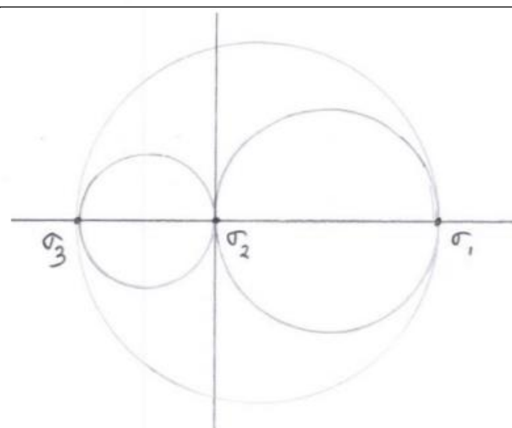
Column Area [in^2]	Reaction Force at Fixed End [lb]			Reaction Force at Free End [lb]		
A_{column}	R3	R3x	R3y	R4	R4x	R4y
21.38	158.37	1874.42	85.34	736.73	2049.07	85.34

TABLE LXX: STRESS ANALYSIS RESULTS FOR THE COLUMNS

	τ_{ZX} [psi]	σ_z [psi]	τ_{ZY} [psi]
Results	182.36	41.87	7.98

With all of the bending and shear stresses on the I-beam found, the Team analyzed the principal stresses by using Mohr circles. The Mohr circles for the beams were different from each other. TABLE LXXI summarizes the schematics of the Mohr circles and the associated principal stresses.

TABLE LXXI: SUMMARY OF MOHR CIRCLES FOR SUPPORT STRUCTURES AND PRINCIPAL STRESSES

Component	Mohr Circles	Principal Stresses
I-beam		P_1 = 3626.24 lb P_2 = 3354.19 lb P_3 = -78.57 lb
C-channel Bracket		P_1 = 5418.56 lb P_2 = 1468.47 lb P_3 = -190.22 lb
Column		P_1 = 204.49 lb P_2 = 0 lb P_3 = -162.62 lb

From the principal stresses, Equation 3.3.2.1 was used to assess if the structures would yield under case 3A. The analysis result of the main components can be found in TABLE LXXII.

TABLE LXXII. VON MISES YIELDING ANALYSIS OF THE STRUCTURE

	I-beam	C-channel	Column
Von Mises Stress [psi]	3576.56	4990.64	318.62
Factor of Safety	10.1	7.21	113.0

As shown in the Table, the factor of safety for all of the structural members were sufficiently high to conclude that the design would not experience yielding.

3.3.3 Fatigue

The principal stresses for the components represented the minimum and maximum stresses with P_1 showing the maximum and P_3 showing the minimum. The differences in the stress levels were calculated by transforming them into the mean stress and the stress amplitude using Equation 3.3.3.1 and 3.3.3.2.

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} \quad \text{Eq. 3.3.3.1}$$

$$\sigma_{amp} = \frac{\sigma_{max} - \sigma_{min}}{2} \quad \text{Eq. 3.3.3.2}$$

With the equations, both mean stress and stress amplitude were analyzed. The results of the calculations are shown in TABLE LXXIII.

TABLE LXXIII: SUMMARY OF MEAN STRESSES AND STRESS AMPLITUDES FOR DESIGN COMPONENTS

	Spreader	Rail	I-beam	C-channel	Column
Mean Stress [lb]	1844.02965	5270.025	1138.94	2357.73	20.94
Stress Amplitude [lb]	5079.09424	6148.014	1431.81	2631.82	183.56

The mean stresses and stress amplitudes were assessed by using the Goodman analysis to determine if the design would experience fatigue. Goodman analysis was performed by taking the endurance limit of the material in use, which was ASTM A36 for the core design components, and applying the following factors to estimate the actual endurance limit of the material.

1. Endurance strength, S_n
   - The endurance strength of the material, which was the stress level that would not cause fatigue failure, could only be determined through external research. The Team found that the endurance strength of ASTM A36 was 160 MPa or 23206.08 psi [34].
2. Material factor, C_m
   - Since ASTM A36 was a carbon steel, the Team determined the material to be a steel casting, resulting in a material factor of 0.8 [35]
3. Stress factor, C_{st}
   - Stress factor of 1 was used for the design as the design components were mostly in bending [35].
4. Reliability factor, C_r
   - A reliability of 99.90% was assumed to be the design factor so the reliability factor of 0.75 was used [35].
5. Size factor, C_s
   - Size factor was derived from equivalent diameter. Since the parts were designed based on the worst-case scenario, the Team used the largest dimensions to analyze the equivalent diameter by using Equation 3.3.3.3 [35].

$$D_{eq.} = 0.808\sqrt{width \cdot height} \qquad \text{Eq. 3.3.3.3}$$

$$D_{eq.} = 0.808\sqrt{2.43'' \times 9''} = 3.78 \text{ in.}$$

- With the analysis, the size factor for the worst-case scenario was determined to be 0.778.

With all of the factors determined, the estimated actual endurance limit, S_n', was analyzed with the following equation [35].

$$S_n' = S_n \times C_m \times C_{st} \times C_r \times C_s \qquad \text{Eq. 3.3.3.4}$$

$$S_n' = 10841.71 \text{ psi}$$

To conduct Goodman analysis, the estimated actual endurance limit was plotted on the y-axis of a graph while the ultimate stress of the material, 58 ksi, was plotted on the x-axis to form a linear line called the Goodman line. Then, each design component shown in TABLE LXXIII was plotted on the graph with the mean stress and stress amplitude values serving as coordinates. The resulting graph is shown in Figure 38.

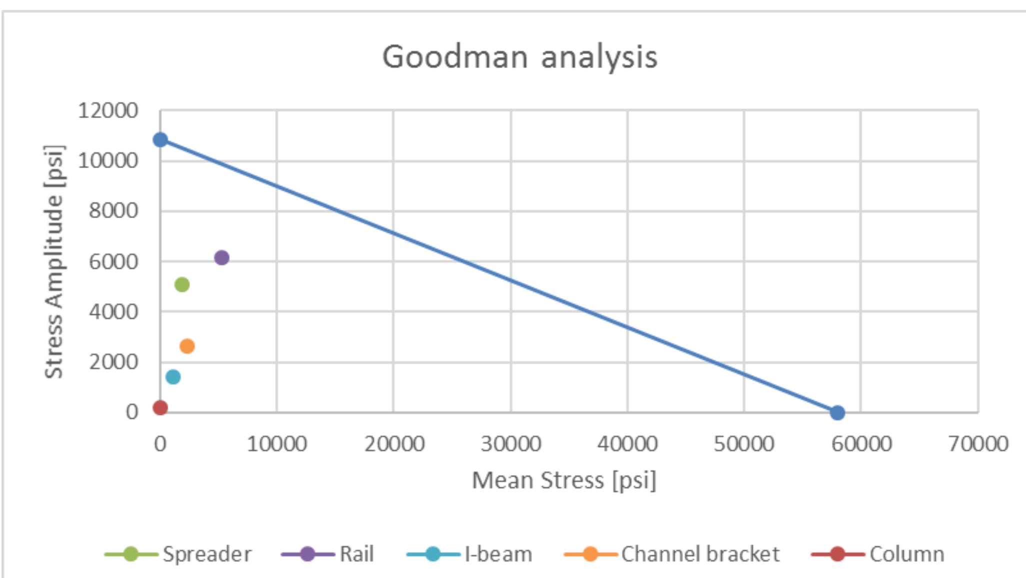


Figure 38: Fatigue analysis using the Goodman line

From the preliminary analysis, the Team found out that the components were less likely to experience fatigue failure as they were within the Goodman line. However, the analysis would not guarantee that the design would not fail by fatigue; consequently, the machine would need to be periodically checked. As shown in the Figure, the lifter assembly including the rail and spreader was the closest to the Goodman line so it would be strongly recommended to inspect the rail and spreader in a regular basis.

3.3.4 Deflection

Since a large portion of the I-beam would extend out from the c-channel with no supports, the deflection at the end of the I-beam needed to be analyzed to ensure that the beam would stay level. To assess deflection, the Team integrated the moment equations in the ZX plane once to obtain the angle of deflection, v', and once again to develop an equation for deflection, v.

For the first section of the beam (in-between the bearings), the deflection equation was developed as the following, where "E" was the elastic modulus and "I" was the second moment of area.

$$EIv'' = -R_1 x - \frac{W_I \cdot x^2}{2} \qquad \text{Eq. 3.3.4.1}$$

$$EIv' = -\frac{R_1 x^2}{2} - \frac{W_I \cdot x^3}{6} + C_1 \qquad \text{Eq. 3.3.4.2}$$

$$EIv = -\frac{R_1 x^3}{6} - \frac{W_I \cdot x^4}{24} + C_1 x + C_2 \qquad \text{Eq. 3.3.4.3}$$

The constants C_1 and C_2 were found by applying the boundary conditions. The conditions were that there would be no deflections at the bearing positions. By applying these conditions in Equation 3.3.4.3, we found both constants.

$$v(x=0) = 0 = 0 - 0 + 0 + C_2$$

$$C_2 = 0$$

$$v(x=a) = 0 = -\frac{R_1 a^3}{6} - \frac{W_I a^4}{24} + C_1 a$$

$$C_1 = \frac{R_1 a^2}{6} + \frac{W_I a^3}{24} \qquad \text{Eq. 3.3.4.4}$$

For the second section of the beam (from the second bearing to free-end), the deflection equation was developed as the following.

$$EI v_2'' = R_2 x - R_1 x - \frac{W_I x^2}{2} - R_2 a \qquad \text{Eq. 3.3.4.5}$$

$$EI v_2' = \frac{R_2 x^2}{2} - \frac{R_1 x^2}{2} - \frac{W_I x^3}{6} - R_2 a x + C_3 \qquad \text{Eq. 3.3.4.6}$$

$$EI v_2 = \frac{R_2 x^3}{6} - \frac{R_1 x^3}{6} - \frac{W_I x^4}{24} - \frac{R_2 a x^2}{2} + C_3 x + C_4 \qquad \text{Eq. 3.3.4.7}$$

To solve for the constants C_3 and C_4, two boundary conditions were used. The first condition was that the angle of deflection, v', would be the same for the two sections at x=a. By applying the condition to Equation 3.3.4.6, the Team found the first constant C_3.

$$v_1'(x=a) = v_2'(x=a)$$

$$-\frac{R_1 a^2}{2} - \frac{W_I a^3}{6} + \frac{R_1 a^2}{6} + \frac{W_I a^3}{24} = \frac{R_2 a^2}{2} - \frac{R_1 a^2}{2} - \frac{W_I a^3}{6} - R_2 a^2 + C_3$$

$$C_3 = \frac{R_1 a^2}{6} + \frac{W_I \cdot a^3}{24} + \frac{R_2 a^2}{2} \qquad \text{Eq. 3.3.4.8}$$

The second condition was that there would be no deflection at the second bearing position. Through applying this condition to Equation 3.3.4.7, the constant C_4 was found.

$$v_2(x=a) = 0 = \frac{R_2 a^3}{6} - \frac{R_1 a^3}{6} - \frac{W_I a^4}{24} - \frac{R_2 a^3}{2} + C_3 a + C_4$$

$$C_4 = -\frac{R_2 a^3}{6} + \frac{R_1 a^3}{6} + \frac{W_I a^4}{24} + \frac{R_2 a^3}{2} - \frac{R_1 a^3}{6} - \frac{W_I a^4}{24} - \frac{R_2 a^3}{2}$$ Eq. 3.3.4.9

With all of the constants identified, the maximum deflection at the free-end of the beam could be analyzed Equation 3.3.4.8 and 3.3.4.9 into Equation 3.3.4.10.

$$v_{max} = v_2(x = l_I)$$

$$v_2(x=l_I) = \left[\frac{R_2 l_I^3}{6} - \frac{R_1 l_I^3}{6} - \frac{W_I l_I^4}{24} - \frac{R_2 a l_I^2}{2} + C_3 l_I + C_4\right]\left(\frac{1}{EI}\right)$$ Eq. 3.3.4.10

The result of the deflection analysis is shown in TABLE LXXIV.

TABLE LXXIV: SLIDING I-BEAM DEFLECTION ANALYSIS

Beam	a [lb]	l_I [lb]	W_I [lb/ft]	R1 [lb]	R2 [lb]	I [in^4]	v [in]
S8x18.4	1.485	4.76	18.4	1085.69	1488.60	57.5	5.57e-6

As shown in the Table, the resulting deflection on the I-beam from the lifter assembly was negligible with a value of 5.57×10^{-6} inches.

3.3.5 Buckling

In the design, columns were the components which were under compression forces exerted by the reaction force R_z. Although the force was not enough to cause yielding, the critical buckling stress of the columns was studied to ensure that the column would not undergo buckling at any instances.

The critical buckling stress of a member under compression could be calculated with the following equation where r_{gy} is the radius of gyration.

$$\sigma_{crit.} = \frac{\pi^2 \cdot E}{(192 \cdot r_{gy})^2} \qquad \text{Eq. 3.3.5.1}$$

$$r_{gy} = \frac{I}{A} \qquad \text{Eq. 3.3.5.2}$$

$$I = \frac{1}{12}(d_{co} \times \omega_{co}{}^3) - \frac{1}{12}(d_{co} - 2t_{co}) \times (\omega_{co} - 2t_{co})^3 \qquad \text{Eq. 3.3.5.3}$$

The resulting critical stress was 33 ksi which included a safety factor of eight in it. Since the critical buckling stress was almost identical to the yield stress, the Team decided that the column, which would not yield according to the von Mises yielding criterion, would not buckle as well.

3.3.6 Finite Element Analysis

With the rollers identified, two support brackets were required to mount the roller to the sliding I beam assembly. The upper bracket mount is seen in Figure 32, and is expected to see the highest forces under worst loading conditions. Due to its complex geometry, the Team completed a finite element analysis on the part to determine its safety factor. To ensure the Team had confidence in the results of the finite element analysis the Team generated convergence plots. The purpose of a convergence plot is to determine how close your solution is to the exact answer. The denser the model mesh the closer it is to an exact. As the mesh becomes denser, the elements more accurately illustrate the true geometry of the geometric model. With every increase in mesh density, either the stress at a node or displacement can be plotted on a graph. This is done repeatedly until the stress values stabilize or a predetermined rate of change between the measured values or rate of convergence is reached.

With the two sensors, the Team defined in Solidworks both plots converged which indicates the results are approaching an exact answer. The results of the FEA can be seen in Figure 39 with the associated convergence plot for sensor 1 and sensor 2 found in Figure 40 and Figure 41 respectively.

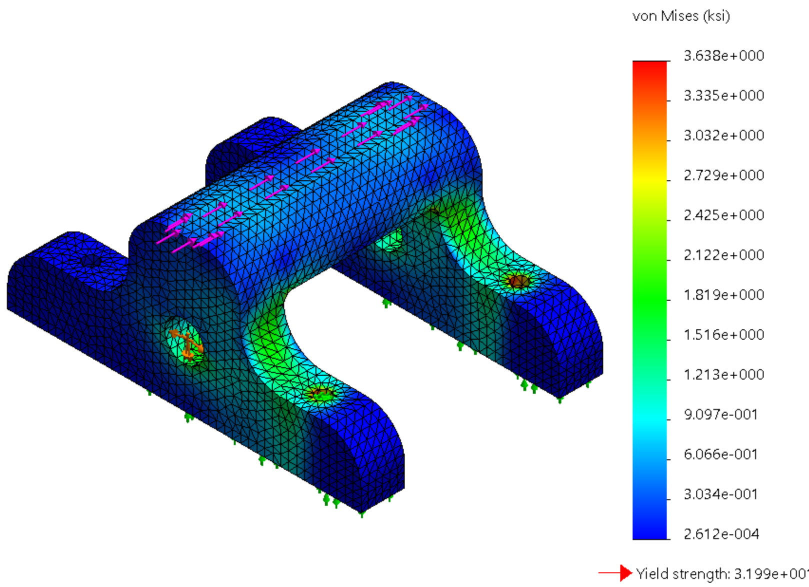


Figure 39: FEA results for the upper roller mount

For the above worst case FEA study it was assumed that two bolts failed so the part was belt attached to the I beam with only two fasteners. To achieve this fixture scenario a fixed fixture was set on the interior cylindrical faces and a sliding fixture was applied to the underside of the part to simulate it sliding along the top of the I beam. For worst case loading scenario, a split face was created in each of the mounting locations of the roller. The purpose of the split was to apply a total of 1845 lbf was applied upwards to simulate the roller stud interacting with the bracket. Next to simulate the thrust load a 376 lbf was applied inwards on the bearing surfaces cylindrical face to simulate the thrust loading the part would experience.

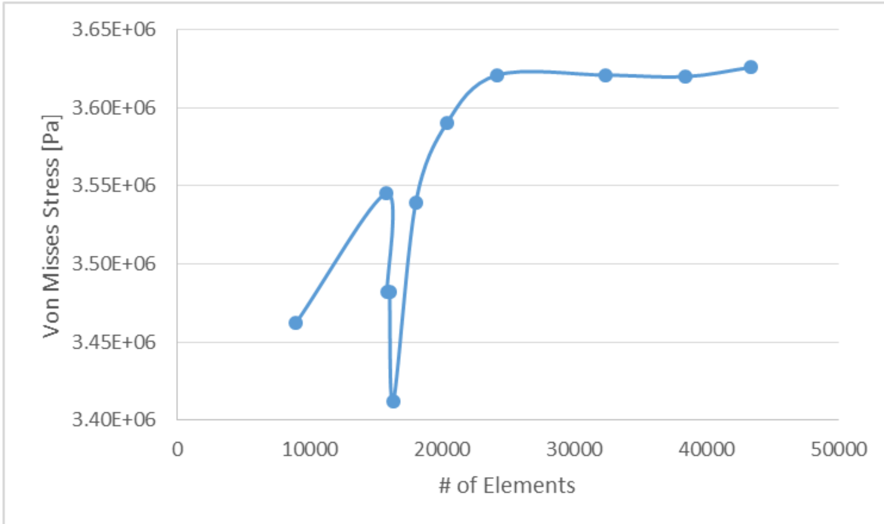


Figure 40: Convergence plot for upper roller mount, Team defined sensor #1

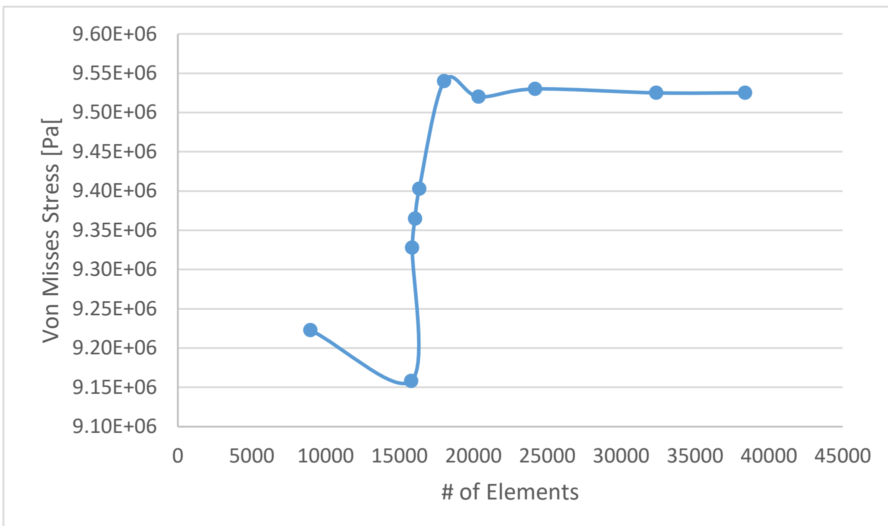


Figure 41: Convergence plot for upper roller mount, Team defined sensor #2

After running the simulation, the Team is confident that the factor of safety on the bracket is 8.2 with a maximum Von Mises Stress value of 3.926 ksi. The deflection of the part was found to be 3.7145e-4, and found to be negligible. The Team is confident in these results due to Figure 40 and Figure 41 indicated convergence for the finite element analysis study.

Due to the complex nature of the second taper roller bearing mount, the Team determined that FEA was required to be completed on the part as hand calculations would be too complicated to completed. For the FEA study two different fixtures were applied to the part. The first was a sliding roller fixture along the face of the part that would be interacting with the I-beam, and the second being a fix fixture along the bottom two cylindrical faces that the bolts would interact with. This situation represents the worst case loading scenario where two of the four bolts have failed and the lifter is picking up the largest sheet sizes. The load scenario with fixtures can be seen in Figure 42.

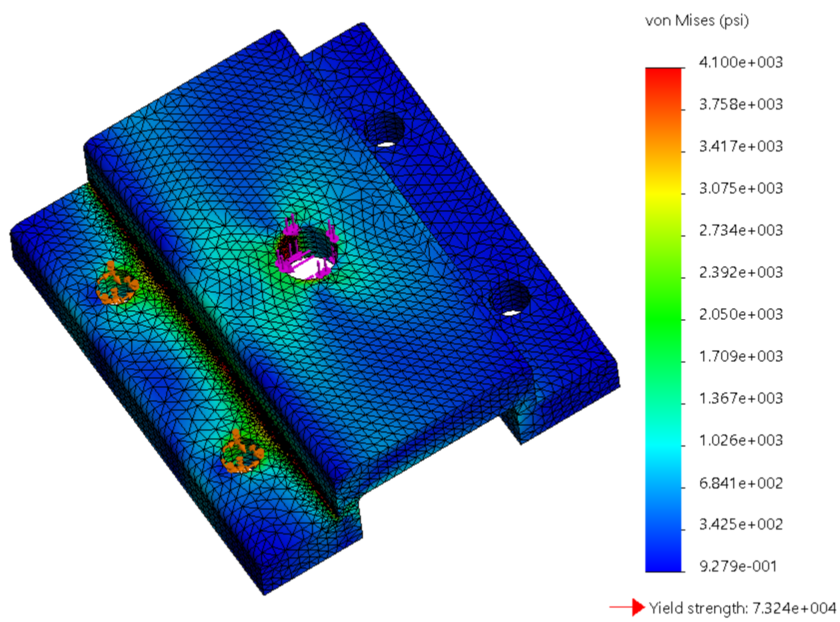


Figure 42: Worst case scenario FEA study on the lower roller mount

The same process as the upper taper bearing mount was taken for the lower bearing mount. Through convergence the Team determined that the maximum Von Mises Stress would be 1.218e4 PSI, maximum deflection of 6.42e-4 inches which yields a factor of safety of 6.

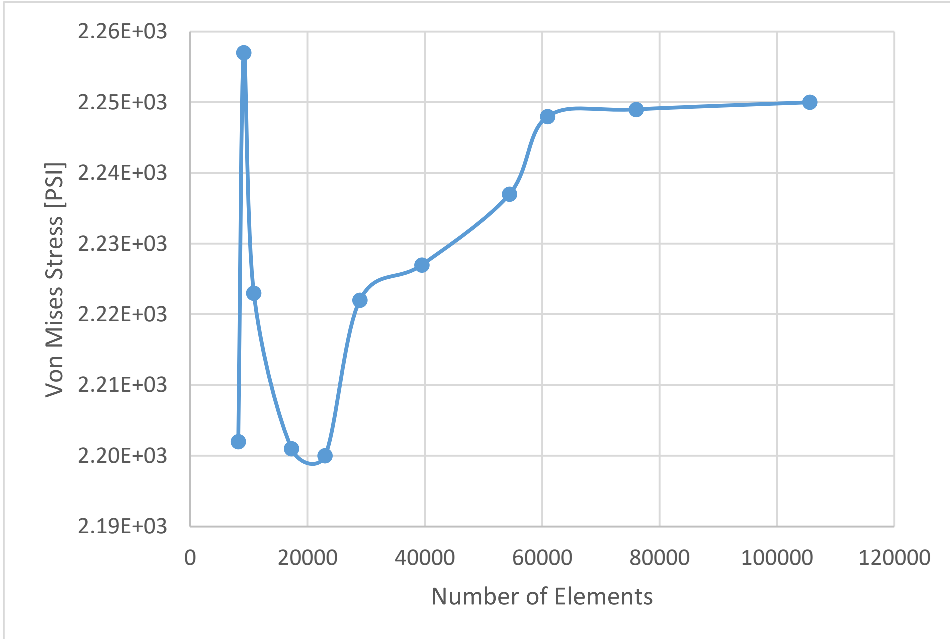


Figure 43: Convergence plot for lower mount, Team defined sensor #1

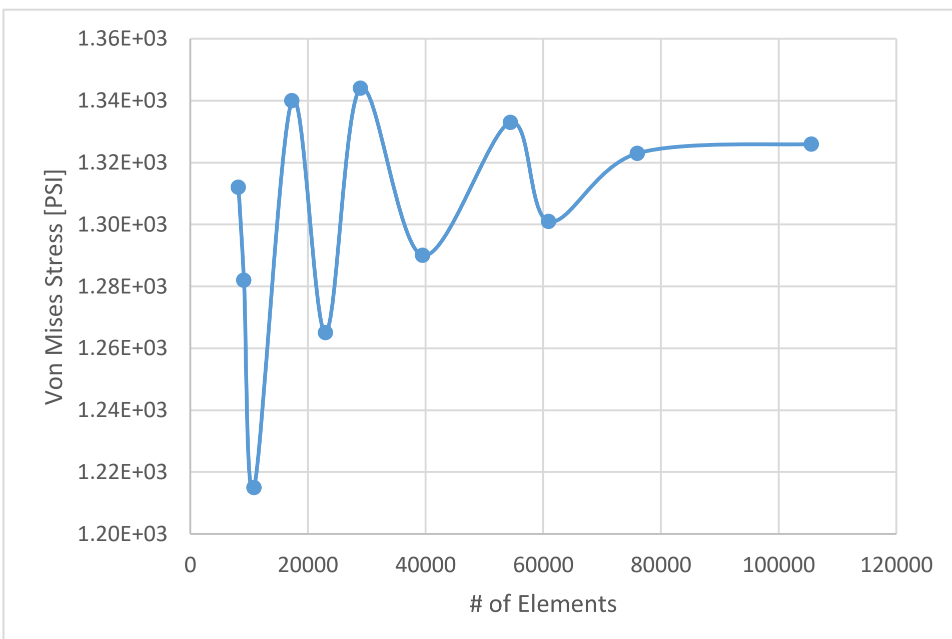


Figure 44: Convergence plot for lower mount, Team defined sensor #2

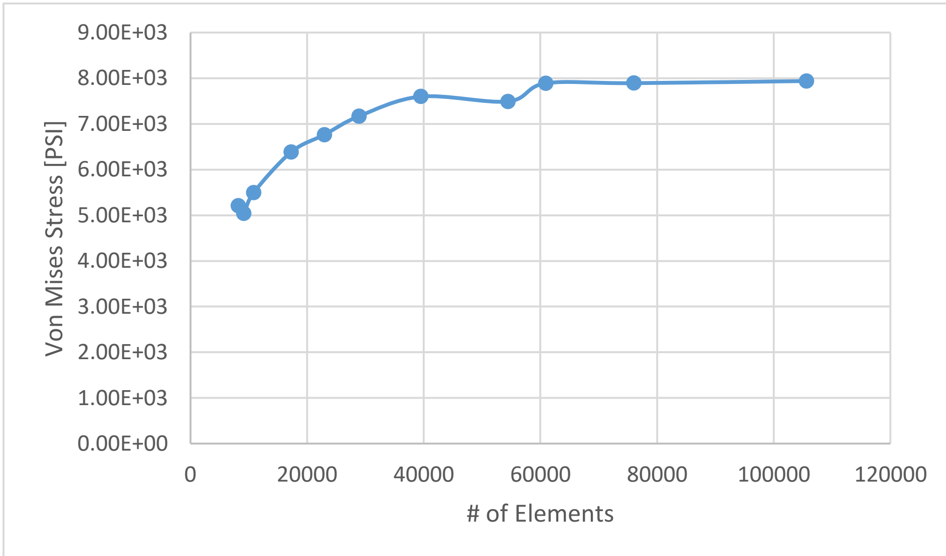


Figure 45: Convergence plot for lower mount, Team defined sensor #3

The Team is confident in the results due to plotting for convergence for three different sensors and all sensors converging to an exact value which can be seen in Figure 43, Figure 44 and, Figure 45.

For the lifter, a worst case loading scenario was applied to see how the assembly would perform while experiencing load case 3A. A summary of load case 3A can be found in TABLE LXXV.

TABLE LXXV: SUMMARY OF LOADS FOR LOADING CASE 3A

<table>
<tr><th>Case</th><th>Load</th><th>Description</th><th>Value</th><th>Location</th></tr>
<tr><td rowspan="5">3A</td><td>F_x</td><td>External force</td><td>184 lb</td><td rowspan="3">Remote: at lifter frame</td></tr>
<tr><td>M_z</td><td>Torsion from external force</td><td>1029 ft-lb</td></tr>
<tr><td>M_y</td><td>Unbalanced sheet</td><td>125 ft-lb</td></tr>
<tr><td>T</td><td>Torque from M_z</td><td>2216.9 ft-lb</td><td>Bearing carriage locations</td></tr>
<tr><td>F_z</td><td>Weight on cylinder</td><td>550 lb</td><td>Cylinder mounting bracket</td></tr>
</table>

To simulate the external force and moment created by an employee pushing the lifter, a remote load was applied to the assembly. This remote load created a moment around the cylinder's center axis, which caused a force couple at the bearing carriage location, as seen in Figure 46. This loading scenario also considered the moment of an unbalanced sheet which contributed 125 ft-lb.

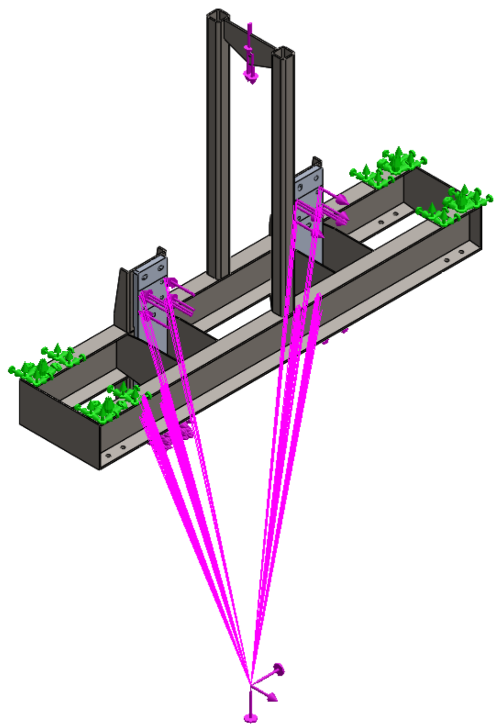


Figure 46: Loading scenario 3A applied on the lifter assembly

The study illustrated points of high stress between the frame and stiffener plates, as well as between PS3M36730 and PSM13440. The points of interest with the highest stresses can be seen in Figure 47.

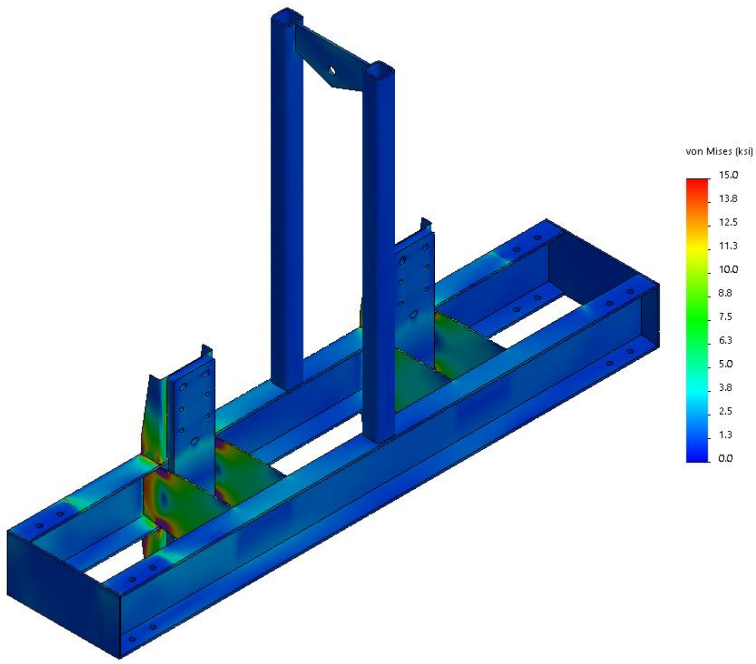


Figure 47: Isometric view of the lifter assembly

These high stress points are found at locations where welding would occur, which illustrates the importance of a quality weld being completed at the high stress locations to prevent failure at the joints. These locations can be seen in Figure 48.

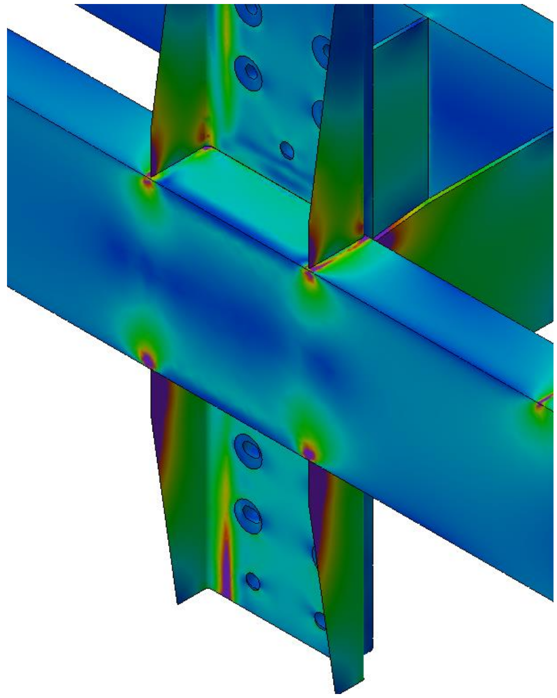


Figure 48: Illustration of highest stress locations in the lifter assembly

After completing numerous FEA studies, the highest Von Mises stress in the c-channel or part PS2M13440 was found to be 17.4 ksi for a factor of safety of 2.07. The highest Von Mises stress in the bracket, part PS3M36730, was found to be 29.4 ksi for a factor of safety of 1.73. The stiffener plate was found to have a maximum Von Mises stress of 31.9 ksi for a safety factor of 1.60, while the aluminum bearing plates had a Von Mises stress of 3.9 ksi for a safety factor of 8.98.

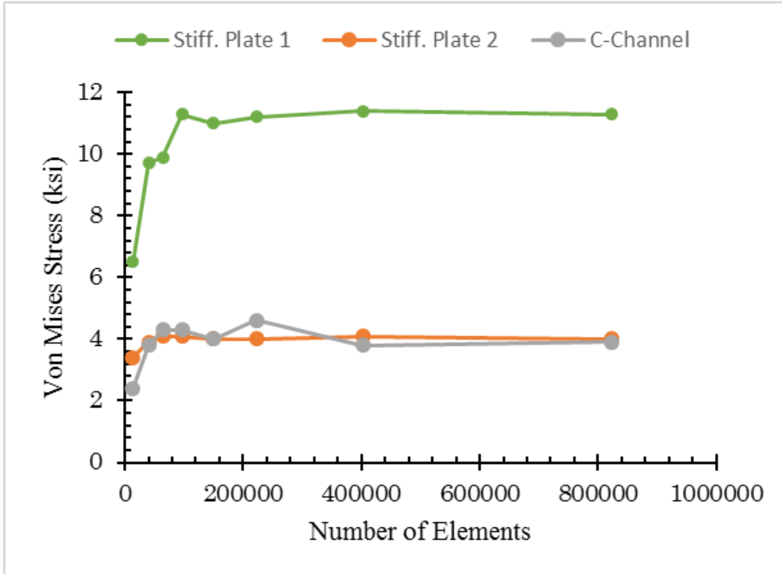


Figure 49: Convergence plot for lifter assembly

The Team validated the results obtained through FEA by proving convergence. As seen in Figure 49, the two stiffener plates and c-channel that were probed all converge to an exact value, so the Team is confident in the results obtained.

3.4 Failure Mode and Effects Analysis

To identify potential failure modes of the sheet metal lifting machine and to develop action plans for mitigating the failure modes, the Team utilized failure mode and effect analysis in the design process. For the finalized design to be proactive in terms of accounting for the future failure modes, the Team decomposed the machine into smaller components and identified possible failure modes associated with each component.

The failure modes were identified from both internal brainstorming and external research. Hazards induced from engineering failure analysis such as yielding, buckling and fatigue were deduced internally. For the failure modes which were caused by mechanical failure and wear, the Team utilized standard CSA-Z432-04 to look for hazards associated with machinery. The Team also researched and found multiple failure modes in existing vacuum lifting products from Health and Safety Executive (HSE) and Fenlock-Hansen Ltd. [36], [37]. In the process, the potential causes of the failure modes were determined by using CSA-Z432-04.

The identified failure modes were evaluated by three factors: severity, occurrence and detectability. Severity represented how severe the impact of the failure modes was on the component, occurrence was measured based on how frequently the failure modes would occur and detectability accounted for how probable it was to detect them. The rating scale systems used for the three factors are shown in TABLE LXXVI, TABLE LXXVII and TABLE LXXVIII.

TABLE LXXVI: RATING SCALE AND CRITERIA FOR SEVERITY [38]

Severity		Rating
Minor	Failure has no effect on performance.	1
Low	Failure causes only slight performance degradation.	2,3
Moderate	Failure causes noticeable performance degradation.	4,5,6
High	Failure causes serious performance degradation	7,8
Very High	Failure causes serious personal safety hazards	9,10

TABLE LXXVII: RATING SCALE AND CRITERIA FOR OCCURRENCE [38]

Occurrence		Failure Rates	Rating
Remote	Failure is unlikely.	<1 in 20,000	1
Very Low	Very few failures.	1 in 20,000	2
Low	Few failures.	1 in 4,000	3
Moderate	Moderate failure probability.	1 in 1,000	4
		1 in 400	5
		1 in 80	6
High	High probability of failure.	1 in 40	7
		1 in 20	8
Very High	Failure is almost inevitable.	1 in 8	9
		1 in 2	10

TABLE LXXVIII: RATING SCALE AND CRITERIA FOR DETECTABILITY [38]

Detectability		Rating
Very High	Design controls will almost certainly detect failure.	1,2
High	Design controls have a high chance of detecting failure.	3,4
Moderate	Design controls may detect failure.	5,6
Low	Design controls have a low chance of detecting failure.	7,8
Very Low	Design controls probably will not be able to detect failure.	9
Non-Detection	Design controls will not detect failure.	10

After the failure modes were analyzed with the three factors, the associated risk priority numbers (RPN), which were the product of severity, occurrence and detectability, were calculated and assigned to the failure modes. A higher RPN showed that the particular failure mode posed higher risks to the operators and their surroundings and required immediate or frequent actions.

TABLE LXXIX shows the summary of the potential failure modes the Team identified and analyzed for the design. The high detectability ratings for many failure modes are due to the control method. With the finalized design, which utilizes manual control of machine, the only available method of design control is visual inspection, which in turn has caused the detectability of many failure modes to score high. As the design becomes more automated in the future, the Team expects the detectability rating to decrease accordingly.

TABLE LXXIX. FMEA FOR THE SHEET METAL LIFTING STATION DESIGN [36], [37], [38]

#	Component	Potential Failure Mode	Potential Failure Effect	SEV	Potential Causes	OCC	Design Controls	DET	RPN	Action Recommendations
1	Support structure	Structure losing stability	Entire transfer station falls over	10	Loss of stability	2	Visual inspection	1	20	Reduce Capacity to reduce the moment causing instability
2		Structure yields and/or buckles	One or more structural beams fails, damaging the whole system	10	Station used for material with higher than recommended mass / material wear	5	Visual inspection	1	50	Apply warning labels to machine and provide adequate training to staff to ensure the capacity of the machine is understood
3		Structure fractures/experiences fatigue	One or more structural beams fails, damaging the whole system	10	Inadequate mechanical strength for repetitive motion / material wear	4	Visual inspection	4	160	Design to ensure failure due to fatigue is unlikely
4		Sheets located above transfer station falls down	Material damage / operator injury	10	Vibration induced by kinetic energy (movement of vacuum lifter)	1	Visual inspection	1	10	Utilize vibration dampers when possible
5		I-beams experience downwards deflection (do not stay level)	Entire transfer station loses stability	10	Material wear	1	Visual inspection	5	50	Determine maintenance schedule and inspection schedules
6		Mechanism holding the rack and I-beam together fails	I-beam with vacuum lifter is separated from the rack, damaging material and/or injuring operator	10	I-beam and vacuum lifter assembly extending to the limit causes tipping over of the assembly	7	Visual inspection	7	490	Design stops to ensure this condition is not reached
7		Brake Pins Shear/ fail	Shear and fatigue	10	Dynamic loading and them being hammered into place during installation	1	Visual Inspection	8	80	Determine maintenance schedule and inspection schedules
8		Lifter moved while in its fully extended position	vacuum head assembly experienced bending	3	Improper operation	3	Visual Inspection	1	9	Determine maintenance schedule and inspection schedules
9		Sideloader crashing into the transfer station	Operator injury / structural damage	8	Relative location of the transfer station in the workplace	8	Visual inspection	3	192	Apply appropriate guarding around the machine to protect it from impact
10	Vacuum lifter	Stuck with suction closed	Fails to pick up sheet metals	8	Mechanical failure	3	Visual inspection	1	24	Utilize ANVER safety system
11		Stuck with suction opened	Fails to release sheet metals	8	Mechanical failure	3	Visual inspection	1	24	Utilize ANVER safety system
12		Fails to provide vacuum	Releases load while transferring material, damaging material and/or injuring operator	9	Mechanical damage to vacuum seal / power failure / suction cup material wear	4	Visual inspection	3	108	Utilize ANVER safety system
13		Component attaching the vacuum lifter to linear actuator fails	Vacuum lifter is released, damaging material and/or injuring operator	10	Inadequate mechanical strength	7	Visual inspection	7	490	Design for appropriate FOS
14	Linear actuator	Fails to move vacuum lifter up/down	Fails to transfer sheet metals	8	Mechanical failure	3	Visual inspection	1	24	Determine maintenance schedule and inspection schedules
15		Fails to hold vacuum lifter in position	Vacuum lifter damage / material damage	8	Mechanical failure	3	Visual inspection	1	24	Determine maintenance schedule and inspection schedules
16	Air knife	Pressurized gas ejection towards the operator	Operator injury	7	Relative location of the air knife to the operator	5	Visual inspection	8	280	Define operator keepout zone for loading scenarios
17		Air knife fails to provide compressed air	Fails to separate both non-ferrous and ferrous sheet metals	4	Mechanical failure	3	Visual inspection	1	12	Reprime airlines and consult maintenance staff
18		Air knife pushes the sheets off the skid	Material damage / operator injury	4	Inconsistent or inaccurate control of compressed air	1	Visual inspection	6	24	Ensure the sideload is able to with stand the impact of the sheet
19		Magnet loses power over years to separate sheets	Fails to transfer ferrous sheet metals	5	Material wear	3	Visual inspection	1	15	Consult manufacturer and maintenance
20	Manual control	Operator fails to align skid to accurate position	Transportation of vacuum lifter to inaccurate loading and unloading locations	2	Human error / procedure problem	9	Visual inspection	1	18	Mitigate with proper training for operators and ensure markings are clearly visible to operators
21		Sliding mechanism fails to lock into position	The entire sliding mechanism is free to move	7	braking system fails	2	Visual inspection	1	14	Ensure redundancy
22	Vacuum Lifter	Supply lines is cut	Accident / collision	7	loss of suction to vacuum lifter	2	Visual Inspection	1	14	Guard all supply lines

The failure modes with the highest risks were item number 6 and 13. Both were associated with the failure of mechanism holding two separate components together. Since the design was composed of components which extended out from the columns, the failure of the connection points meant the disassembly of the core parts. The disassembly would pose very high risk of personal injury and lead to complete stoppage of the process. Thus, frequent and periodic inspection of the connection points was recommended.

The failure mode with the second highest risk was item number 16 which involved the core mechanism of the air knife separation system. Since the air knife would be pointing towards the side loader at all times to separate the sheet metals, the operator would always be at risk of being exposed to highly compressed air. Thus, setting up keep out zone around the air knife device would be recommended for the safety of the operator.

3.5 Design Summary

The final design is consisted of two main assemblies: lifter and support structure assemblies. The lifter, which is capable of picking up the sheet metals, is made of ANVER 8-cup lifting frame L80M8-110-4/44 with suction pads and IGUS heavy duty linear bearing rails TS-01-30 and carriages TW-02-30. While the ANVER components are used to pick up the sheet metals through vacuum suction cups, IGUS components in conjunction with pneumatic linear actuator allow the lifter to achieve the required vertical movements. Figure 50 shows the vertical displacement of the lifter assembly with the sheet metal attached to the suction cups.

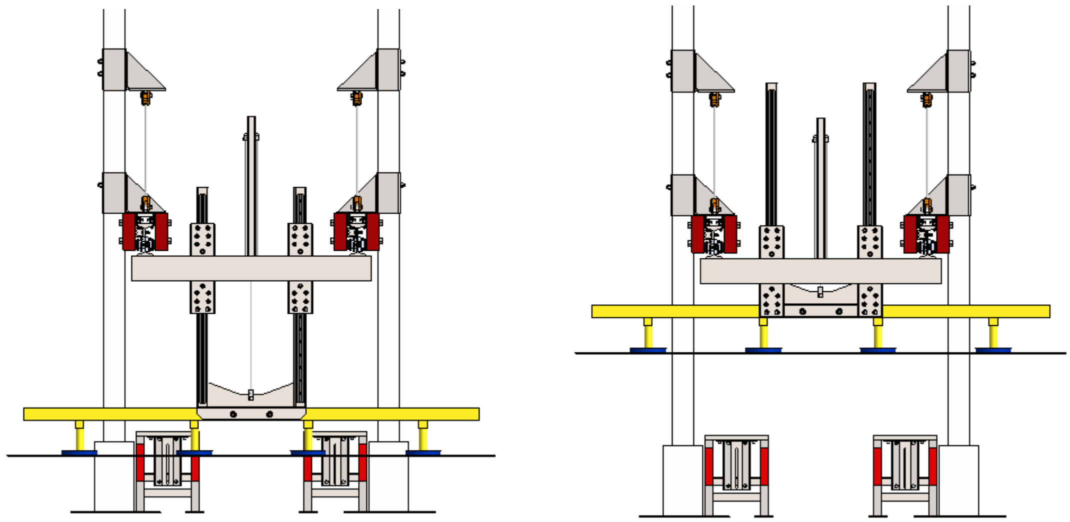


Figure 50. Frontal view of the extended and retracted lifter assembly

For vacuum generation, instead of using a generator which is mounted onto the frame itself, the Team has selected a remote electric powered vacuum generator VPF-57R-AC from ANVER. The main reasoning behind the choice is for the reduction of weight on the lifter assembly. Additionally, both the mounted and remote units have the same load capacity which has allowed the remote unit to be competitive.

The final piece of the lifter assembly is the spreader in which the lifter parts excluding the remote vacuum generator are mounted onto. The spreader is made of two standard C-channels which are fixed to the support structures at each end. The lifter is placed at the center of the spreader to evenly distribute the weight of the lifter assembly among the two sliding I-beams. Figure 51 shows the orientation of the spreader and the fixture points where the sliding I-beams and spreader join.

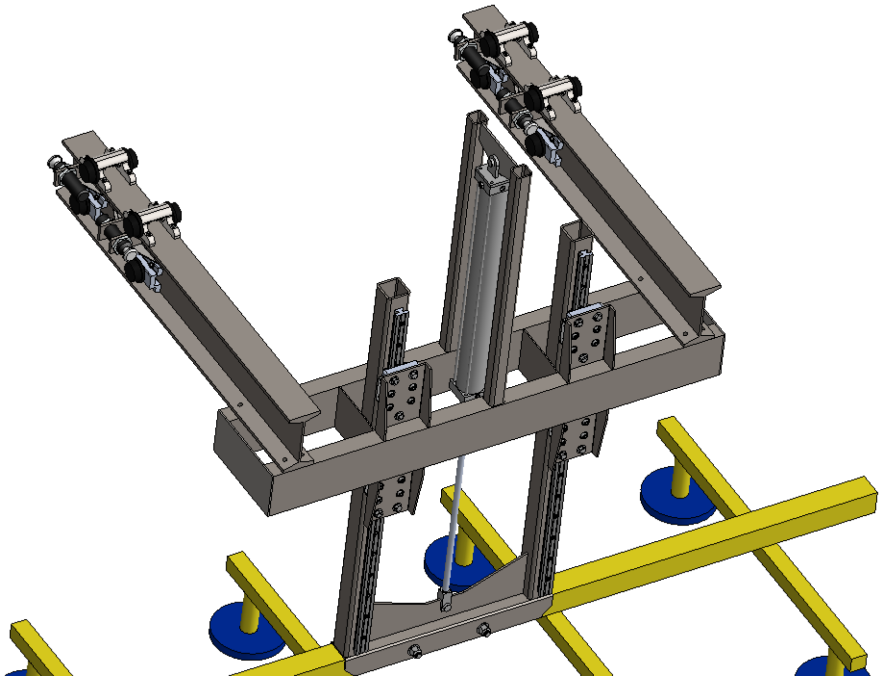


Figure 51. Spreader configuration

The two main structural beams used in the design are the C-channel brackets and sliding I-beams, which are made with standard ASTM A36 material. The sliding I-beams, which the spreader is mounted to, slide within the C-channel brackets via the use of roller bearings CIR-3.00E-R. With the rollers, the I-beams can extend out from the fixed C-channel bracket towards the forklift. Once the I-beam is reaching the maximum allowable extended length, dampers installed on the I-beam, PMXT 1575, hit the pre-installed pins on the C-channel bracket to reduce the velocity of the moving I-beam. Figure 52 illustrates the cross-sectional view of the sliding I-beam and C-channel assembly to demonstrate the locations of rollers and dampers on the I-beam.

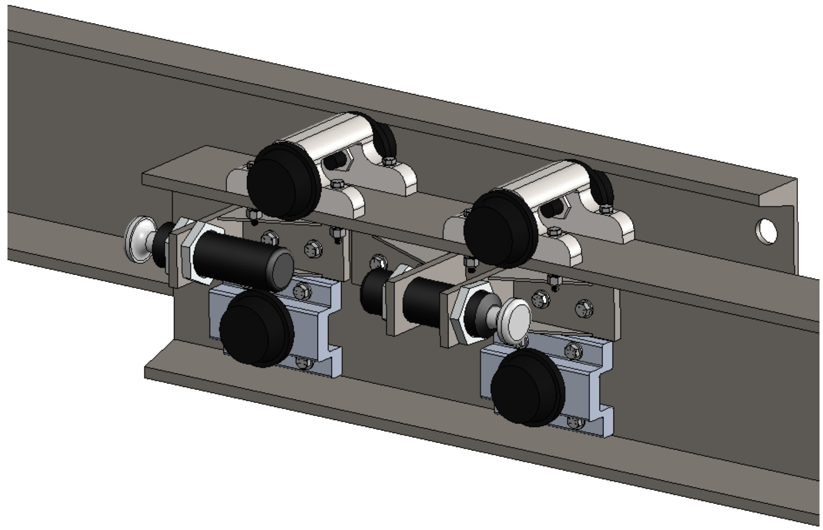


Figure 52. Side view of I-beam and C-channel bracket assembly

Once the sheet metal is picked up vertically with the lifter and moved horizontally towards the unloading zone by the rollers, the sheet is placed on a platform. To reduce the speed of the I-beam sliding back into the C-channel bracket, another set of linear dampers and stopping pins are installed at the end of the beams. The platform, which the sheet metal is delivered to, contains two features: visual marking and air knife fanner. Visual marking is for the operator to locate where to unload the metal. The air knife fanner AKF12 which is installed underneath the platform serves as a mechanism to separate incoming sheet metals to prevent sticking. Figure 53 shows how the air knife fanner would be utilized to achieve sheet metal separation.

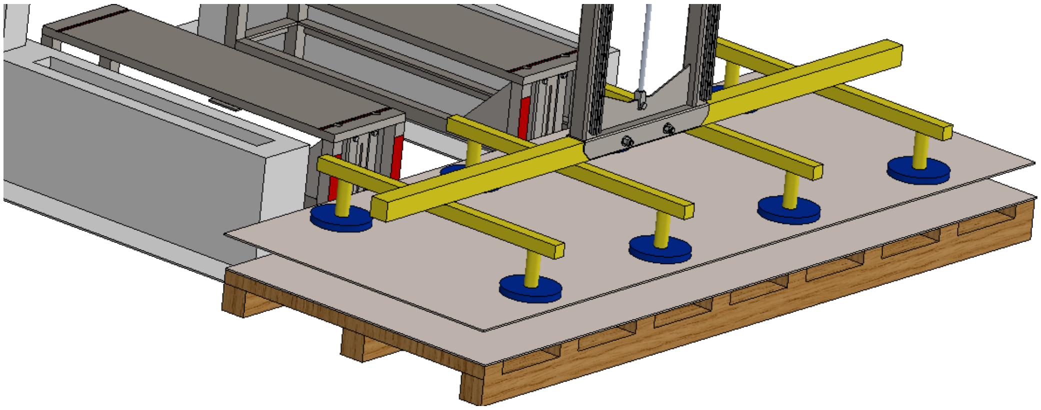


Figure 53. Close up view of incoming sheet metals being separated by air knife fanner

The complete assembly of the design encompassing the lifter, sliding I-beam, C-channel bracket and platform can be found in Figure 54.

Figure 54. Final design of the sheet metal lifting machine

3.5.1 Machine Performance

Upon the completion of the design, the finalized design was evaluated to assess the performance of the machine. By using the design metrics, the Team had set up at the project definition phase, the performance level and estimated specification of the design were analyzed in terms of seven criteria: safety, robustness, quality, morale, efficiency and cost. TABLE LXXX summarizes the target specification and how the final design performs with respect to them.

TABLE LXXX: TARGET AND ESTIMATED FINAL DESIGN SPECIFICATIONS

Category	Metric	Target	Design Estimate	Unit	Reason for Deviation
Safety	Worker Contact Time Per Sheet	< 1	2	seconds	It is expected that the contact time per sheet may increase if the sheets require more manual alignment after the sheet is placed on the loading skid.
	New Hazards to Workplace	1	2	each	1) Slide-on-roller overhead equipment 2) High velocity air (air knives)
	Encroachment into Traffic Area	2.1	1.6/2.6	m	Retracted/extended. The retracted dimensions meet the basic functionality.
Robustness	Pre-Shift Check Time	5	5	minutes per shift	No deviation.
	Initial Sheet Position Variance	13	50	mm	The alignment of the unloading skid will depend strongly on the operator of the side loader.

Category	Metric	Target	Design Estimate	Unit	Reason for Deviation
	Loading Methods	1	1	each method	No deviation.
	Adaptability for Future Features	1	1	each feature	No deviation.
	Different Order Sizes	1 to 10	1 to 10	sheets per order	No deviation.
	Handled Materials	4	4	types	No deviation.
	Handled Sheet Sizes	$T \leq 0.5$ $36 \leq W \leq 72$ $96 \leq L \leq 144$	$T \leq 0.5$ $36 \leq W \leq 48$ $96 \leq L \leq 120$	inches	The range of sheet sizes was reduced according to the capabilities of the vacuum lifter and air knives. This restricted range covers more than 65% of Ryerson's orders.
Quality	Final Sheet Position Variance	25	50	mm	The alignment of the final sheet placement will depend heavily on the alignment of the sheet when it is first lifted.
	Maximum sheet droop	50	125	mm	This value was changed to match the maximum allowable deflection according to Anver.
Morale	Unsuccessful Sheet Moves	0.0001 - 0.02	0.0001 - 0.02	%	No deviation.

Category	Metric	Target	Design Estimate	Unit	Reason for Deviation
	Operating Steps for a Single Sheet	3	6	each	Increase in steps compared to the current manual method of moving sheets because of the extra attention required for aligning the equipment safely.
	Job setup time	1.5	1.5	minutes	No deviation.
Efficiency	Number of Operators	1	1	workers	No deviation.
	Task Time Per Sheet	5	12	seconds	The task time increased based on the specifications of the linear actuators that were considered for the cost estimate.
Cost	Annual Costs	$ 2,000	$ 2,000	CDN$	No deviation.
	Installation Cost	$ 2,000	$ 2,000	CDN$	No deviation.
	Steps for Installation	10	10	steps	No deviation.
	Total Footprint	78	68/124	square feet	Retracted position/operating position. Dimensions of machine for basic functionality.

Category	Metric	Target	Design Estimate	Unit	Reason for Deviation
	Steps for Servicing Access	2	2	steps	No deviation.
	Capital Cost	$ 25,000	$ 21,017	CDN$	Based on final design cost estimate.

The most noticeable deviation from the target specification is the reduction in handled sheet sizes. The reduction in maximum sheet size is due to the structure losing balance in the worst loading scenario. The loss in balance would induce the entire structure to tip over resulting in a catastrophic failure. To prevent the failure mode, the Team has performed stability analysis iteratively by reducing the maximum sheet size. Through the analysis, the Team concluded that a sheet size of 48 by 122 inches is the maximum allowable size to avoid the failure mode.

Another notable compromise was the task time per sheet. The Team identified that high speed sheet moving operations require a significant amount of automation. This added complexity would increase cost an exorbitant amount. To reduce the overall cost of the system, we determined to use a linear actuator instead of a motorized hoist. By using the linear actuator, we reduced the estimated task time per sheet from the initial projection time of 15 seconds to 12 seconds. We predicted that the designed lifter would require approximately 4 seconds to lift a sheet metal to height, 4 seconds to move horizontally towards the unloading skid with an average speed of 2 feet per second, and additional 4 seconds to drop the sheet to the skid.

3.5.2 Preliminary Engineering Drawings

3.5.2.1 Part Naming Conventions

The Team determined that a modified S-Code was required to name all parts and assemblies for the structure. A modified S-Code is important as it defines primary features and properties of each part and assembly. The Team generated a nine digit code to identify each part. If Ryerson chooses to expand the structure into other facilities, the same s-code can be utilized to name the parts and track machine components.

TABLE LXXXI. MODIFIED S-CODE FOR THE RYERSON SHEET METAL TRANSFER STATION

Digit Location	Property	Value	Descption
1	Part/ Assembly	P	Part
		A	Assembly
2	Assembly Location	T	Table
		L	Lifter
		S	Support Structure
		R	Sliding Rail Support Structure
3	Mating Relationship	1	Primary
		2	Secondary
		3	Tertiary
4	Part Type	P	Purchased Part
		M	Manufactured
		A	Assembly
5	Material	0	Assmbly
		1	Purchased Part
		2	A36 Steel
		3	1020 Steel
6	Made from	0	Assemble
		1	I Beam
		2	Angle Iron
		3	U Channel
		4	Bar Stock
		5	Round Stock
		6	Sheet
		7	Plate
		8	Square Tube
7	Primary Operation	0	Assemble
		1	Water Jet
		2	Lathe
		3	Mill
		4	Drill
		5	Hand Modification
		6	Weld
		7	Laser Cut
8	Largest Length	1	Smaller than 1 IN
		2	1 < L < 4
		3	4 < L < 12
		4	L > 12
9	Side Location	0	No Side
		1	Right Side
		2	Left Side

The explanation of how the s-code is applied is explained below for the sheet metal bracket seen in Figure 55.

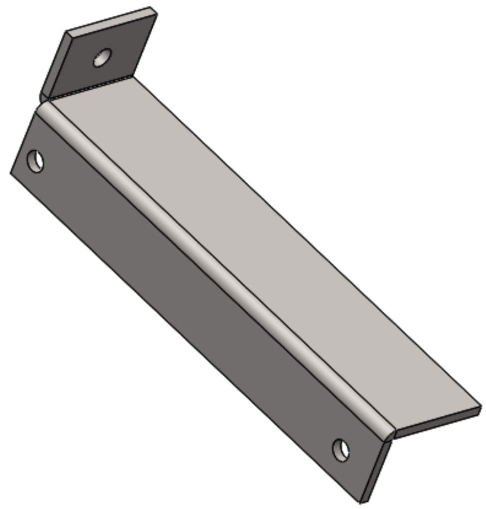
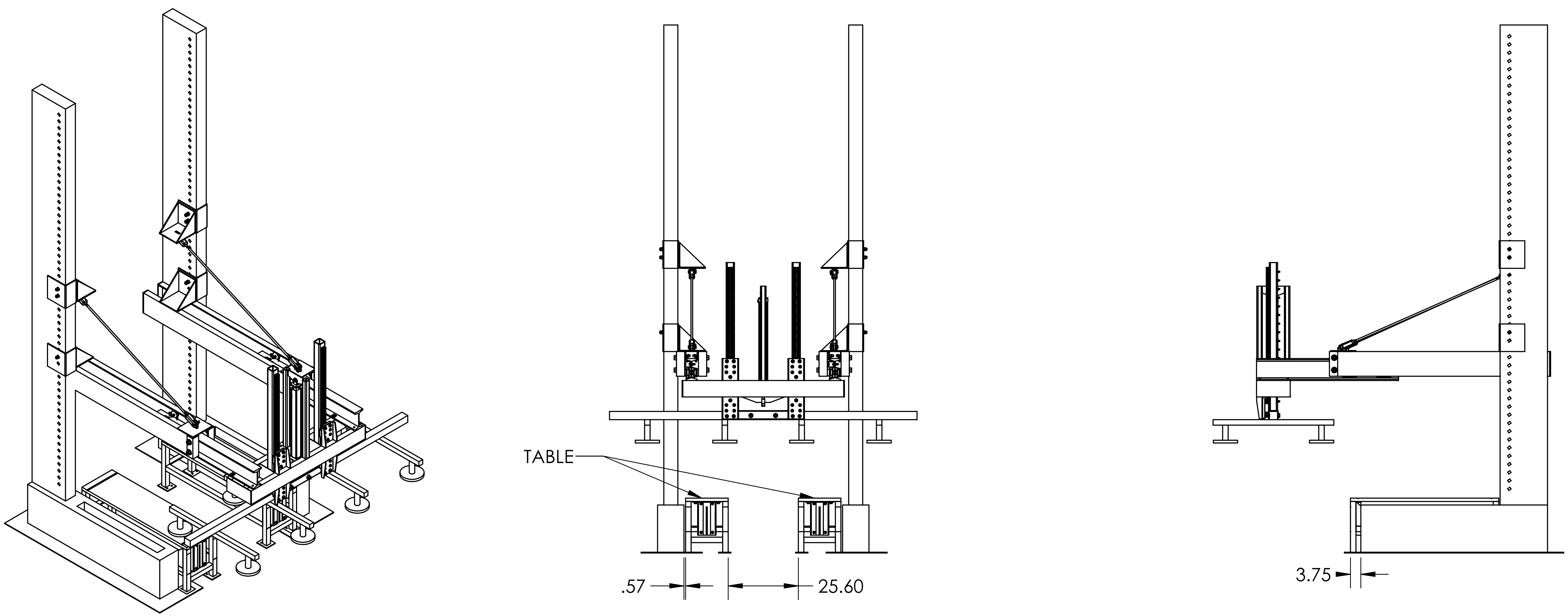


Figure 55. A depiction of detailed part PT2M36731

The taper bearing mount is a detailed part so it would be assigned a character value of “P”. It is integrated the table thus the second character is a “T”. In the table assembly, it acts as the intermediate connection part between the air knife and table which makes it possess a secondary mating relationship which yields a third character value of “2”. The fourth character value represents the part type, since it is a part to be manufactured by Ryerson and not purchased it will poses a value of “M”. Character value five and six represents the material and make from, as the part made from AISI 1020 sheet metal, values of “3” and “6” will be used respectively. The Team predicted the primary manufacturing method as laser cutting, so character value 7 will have a value of “7”. As the part is longer between 4 and 12 inches a character value of “3” is assigned. For the final character digit it is to be used on the right side of the assembly thus a value of “1” is assigned. The resulting s-code vale is PT2M36731.

3.5.2.2 Preliminary Drawings

The following pages are representing the technical drawings required to manufacture and assemble the designed sheet metal transfer station. It is important to note that these drawings are preliminary and all calculations presented in this report should be verified and validated by a qualified registered engineer to ensure the machine is safe and will not pose a risk. With this in mind the following drawings are preliminary and the final design should not be manufactured from them until signed off by a qualified engineer.



SOLIDWORKS Educational Product. For Instructional Use Only

REV	DATE	DESCRIPTION	
A	16/12/06	DRAFT	DWN: DDY CKD:AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

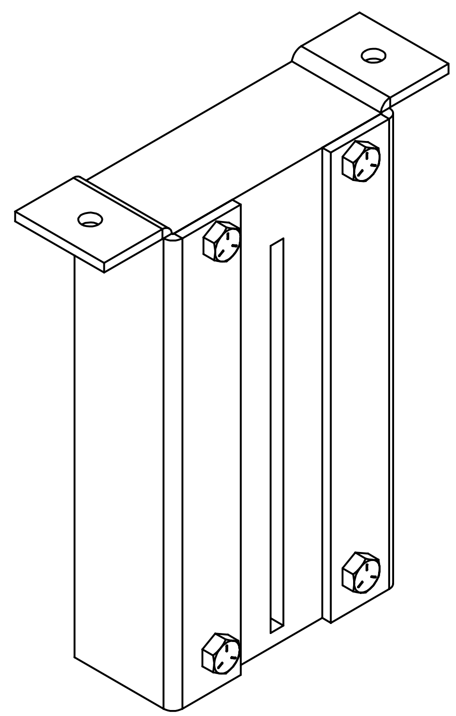
DIMENSIONAL
UNIT: INCHES

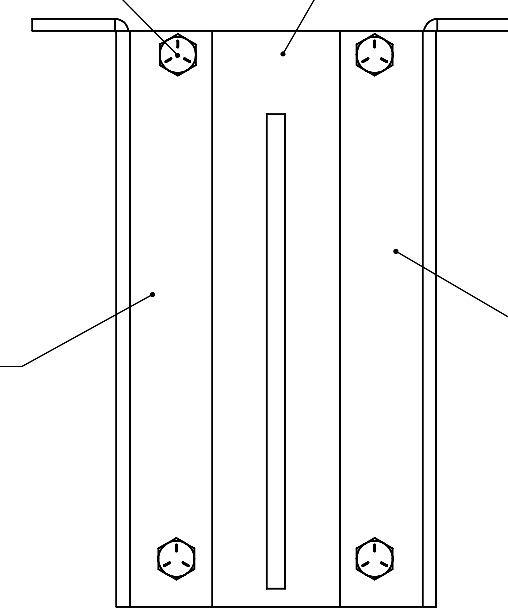
DO NOT SCALE DRAWING

DESCRIPTION:
TABLE

SIZE B DWG. NO. TABLE REV A

SCALE: 1:40 WEIGHT: N/A SHEET 1 OF 1





REV	DATE	DESCRIPTION	
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			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

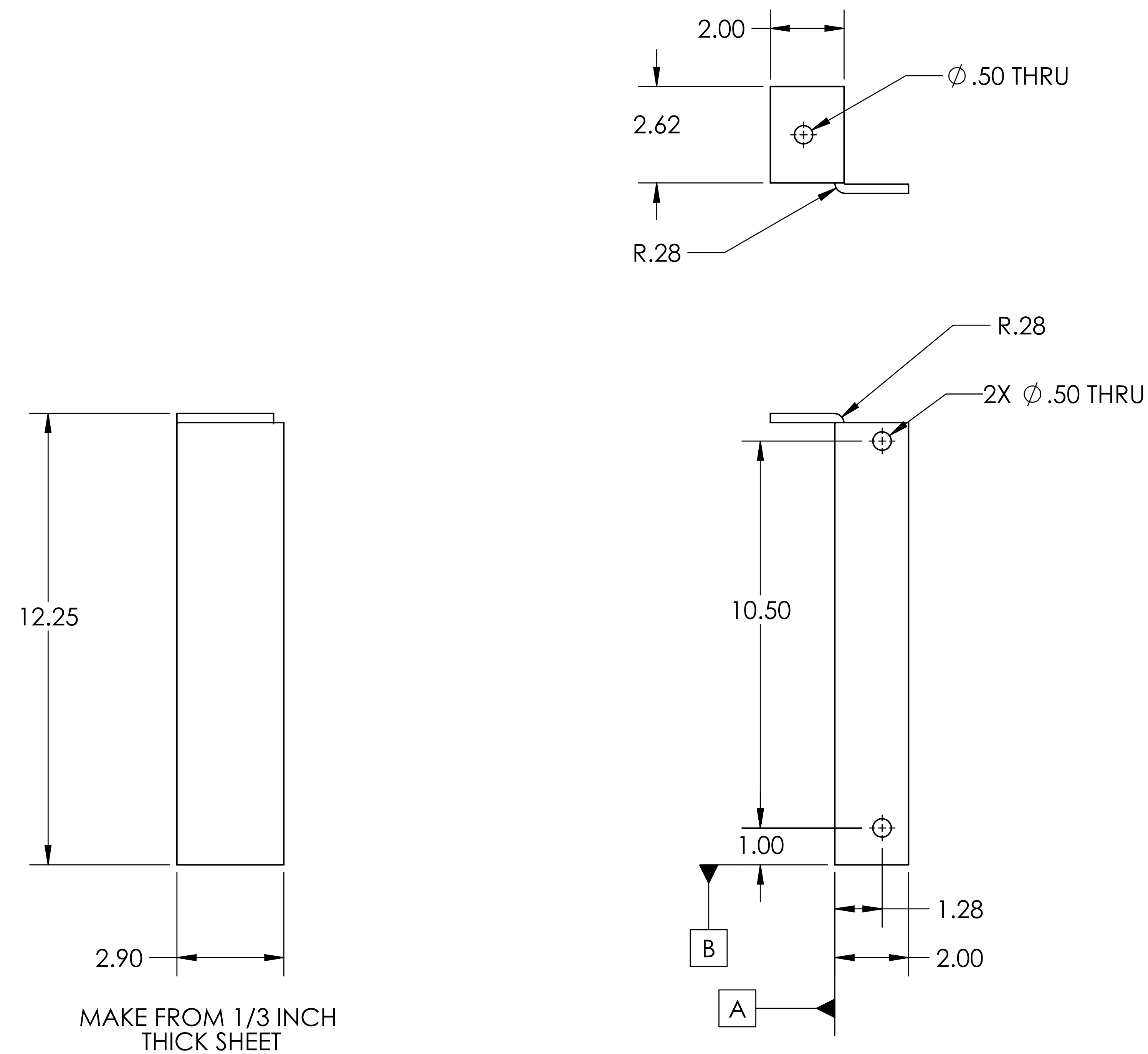
DESCRIPTION:
AIR KNIFE ASSEMBLY

SIZE B

DWG. NO. AIR KNIFE ASSEMBLY

REV A

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



REV	DATE	DESCRIPTION	
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			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL AISI 1020

FINISH NO PLATING 3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL UNIT: INCHES

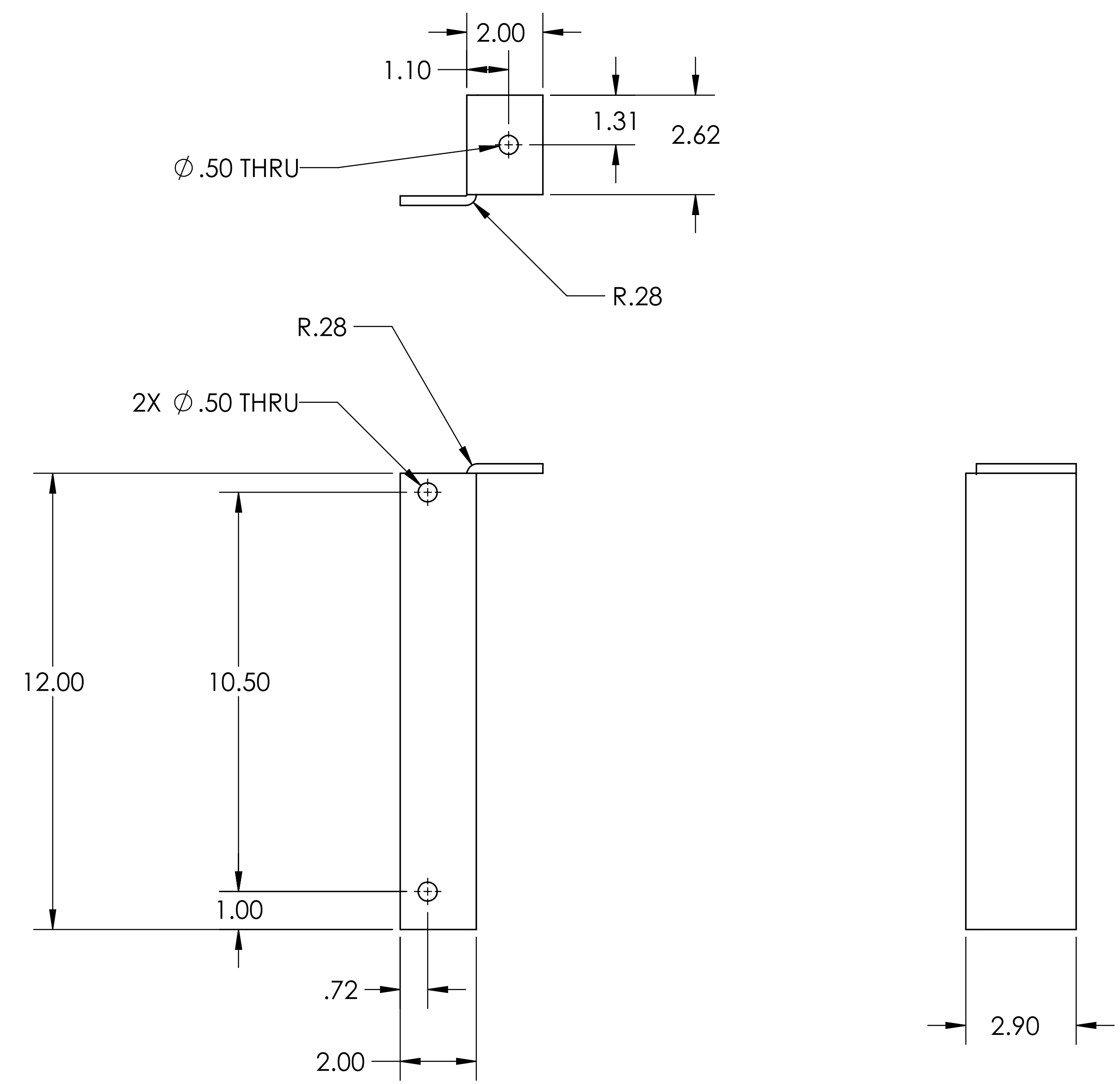
DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. PT2M36732 | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



MAKE FROM
0.25 INCH THICK SHEET

REV	DATE	DESCRIPTION	
A	16/12/06	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL UNIT: INCHES

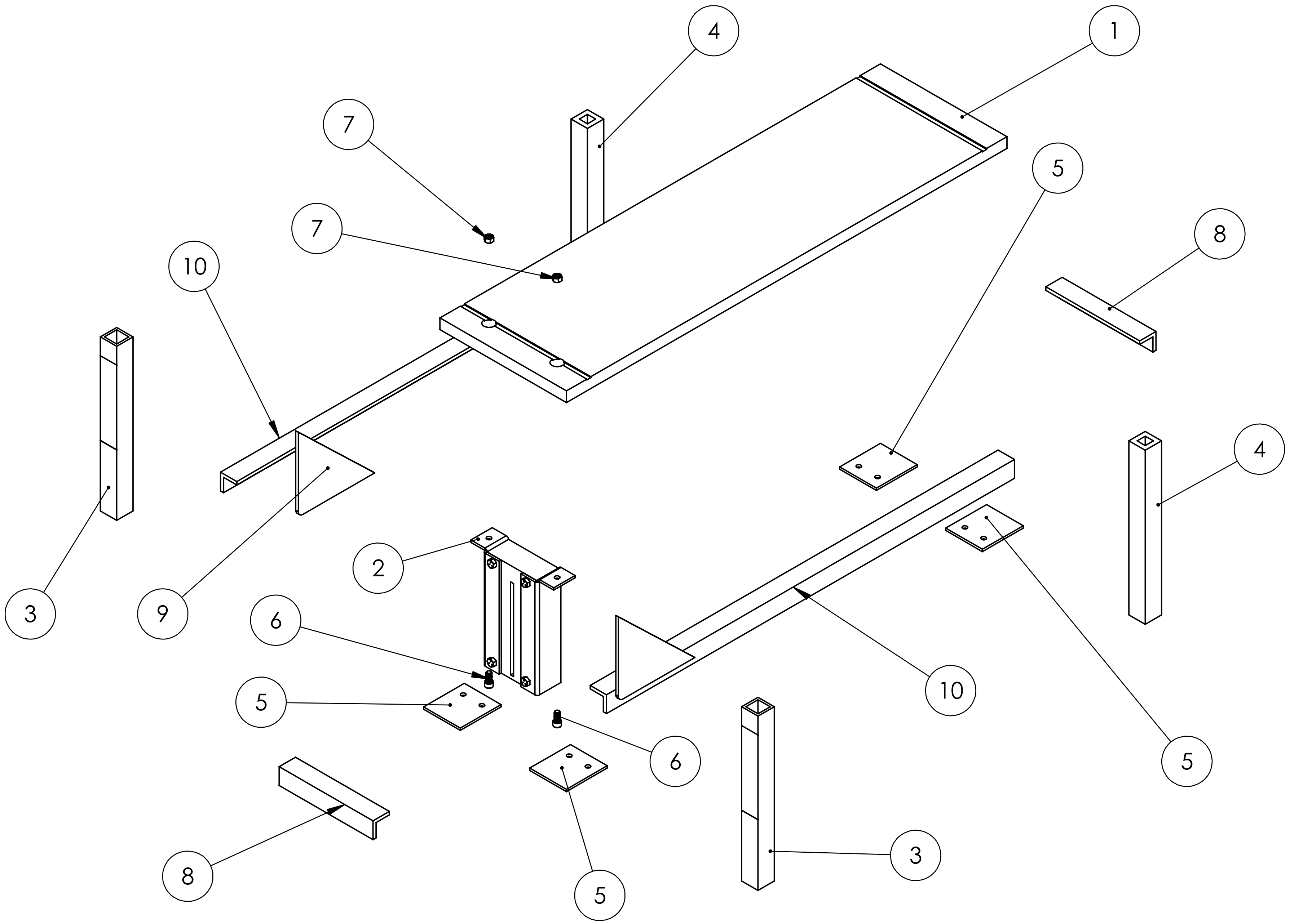
DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. PT2M36731 | REV A

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

ITEM NO.	PART NUMBER	QTY.
1	PT1M37140	1
2	Air knife assembly	1
	AKF12	1
	PT2M36732	1
	PT2M36731	1
	92865A714	4
3	PT2M34630 - Marked	2
4	PT2M34630	2
5	Table bottom	4
6	91251A712	2
7	94895A823	2
8	Table Bracket Rear	2
9	PS1M36730	2
10	Table Bracket	2



REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL UNIT: INCHES

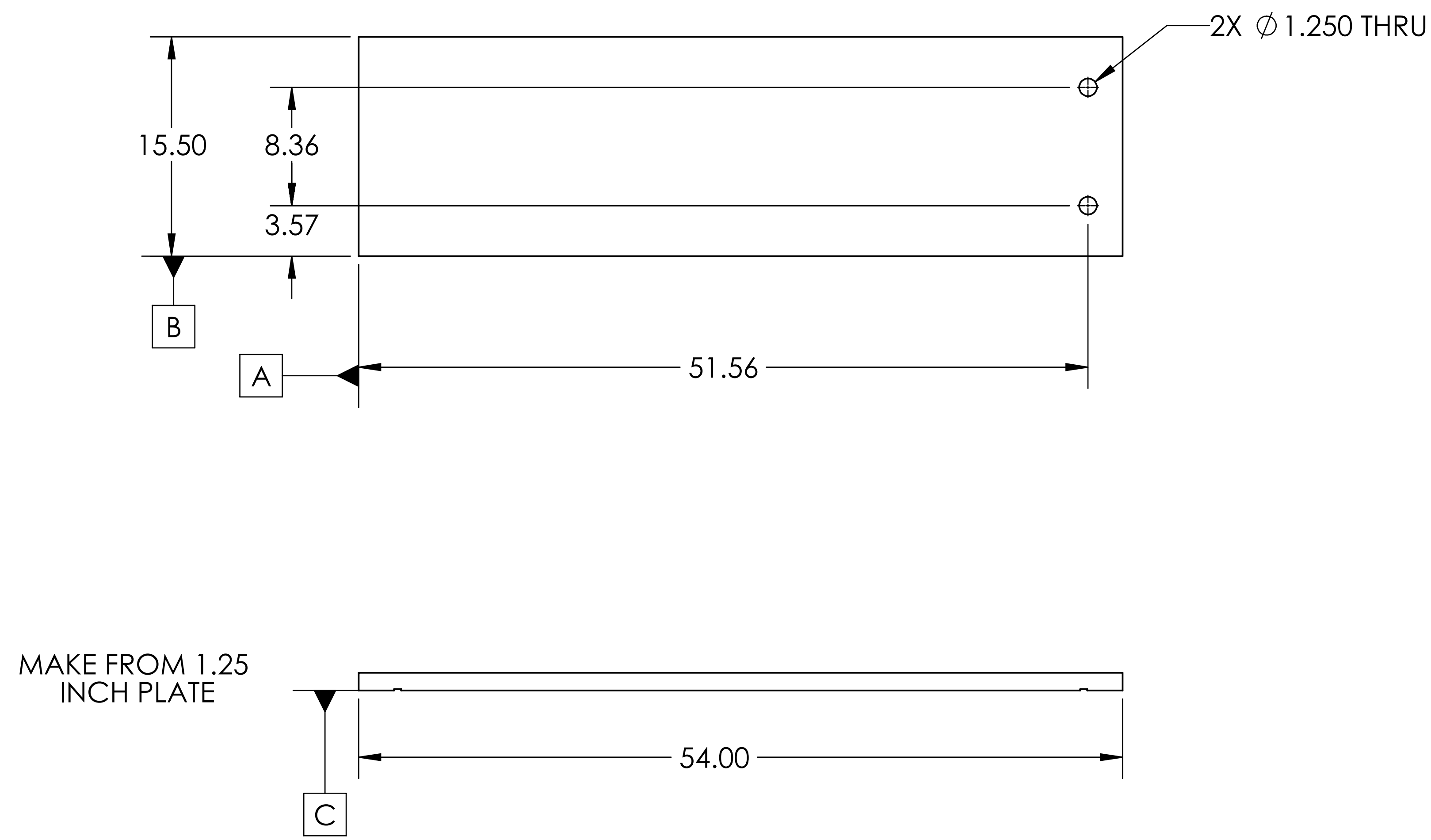
DO NOT SCALE DRAWING

DESCRIPTION:
TABLE

SIZE B | DWG. NO. AT1M00640 | REV A

SCALE: 1:32 | WEIGHT: N/A | SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN:
CKD:
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A26

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

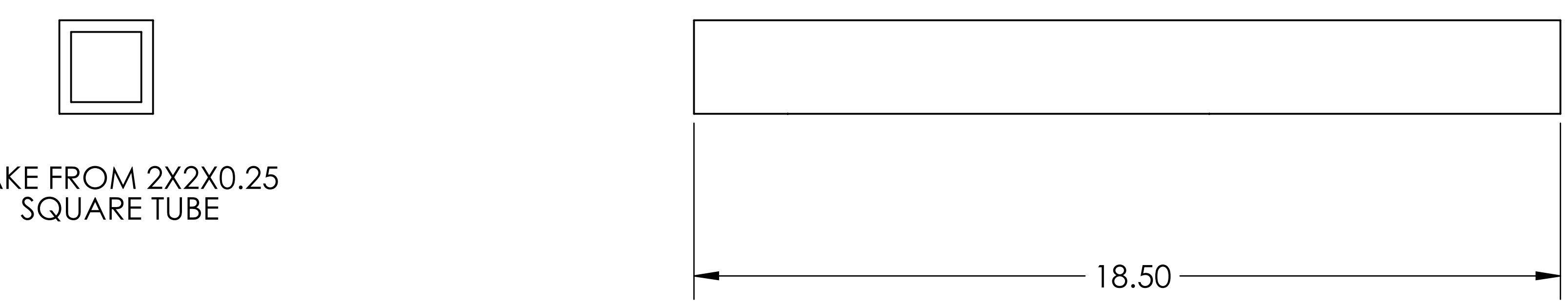
DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B DWG. NO. PT1M37140 REV A

SCALE: 1:12 WEIGHT: N/A SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only



MAKE FROM 2X2X0.25
SQUARE TUBE

REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

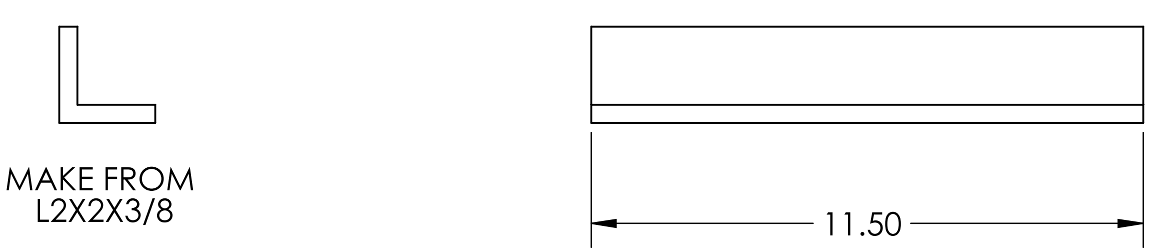
DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B DWG. NO. PT2M34630 REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

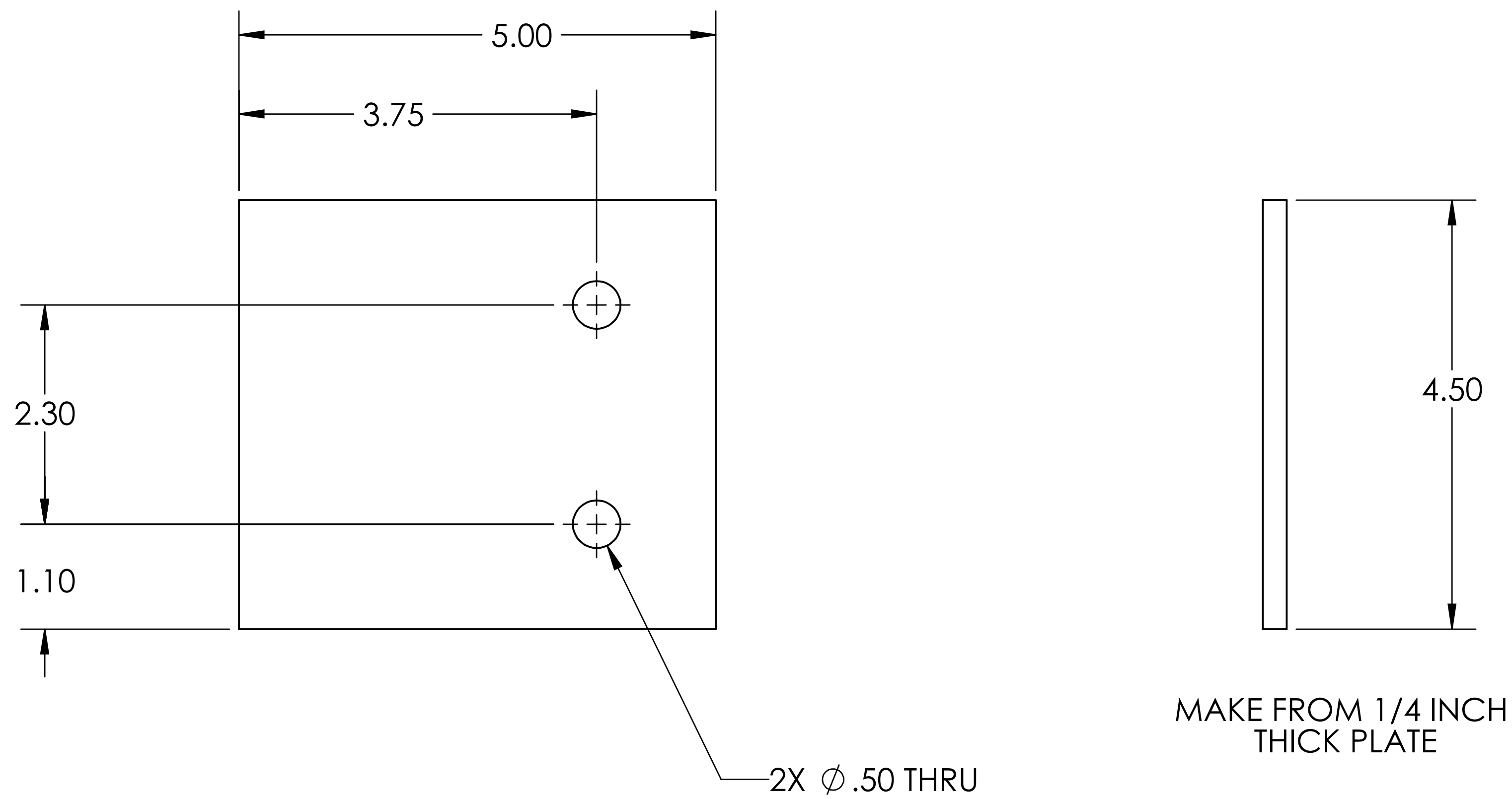
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
TABLE BRACKET SHORT

SIZE B
DWG. NO.
TABLE BRACKET SHORT
REV A

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



REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36

FINISH NO PLATING

3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

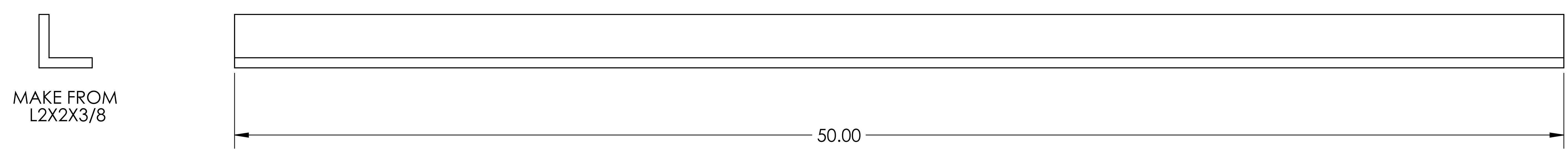
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
TABLE BOTTOM

SIZE B | DWG. NO. TABLE BOTTOM | REV A

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



REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL UNIT: INCHES

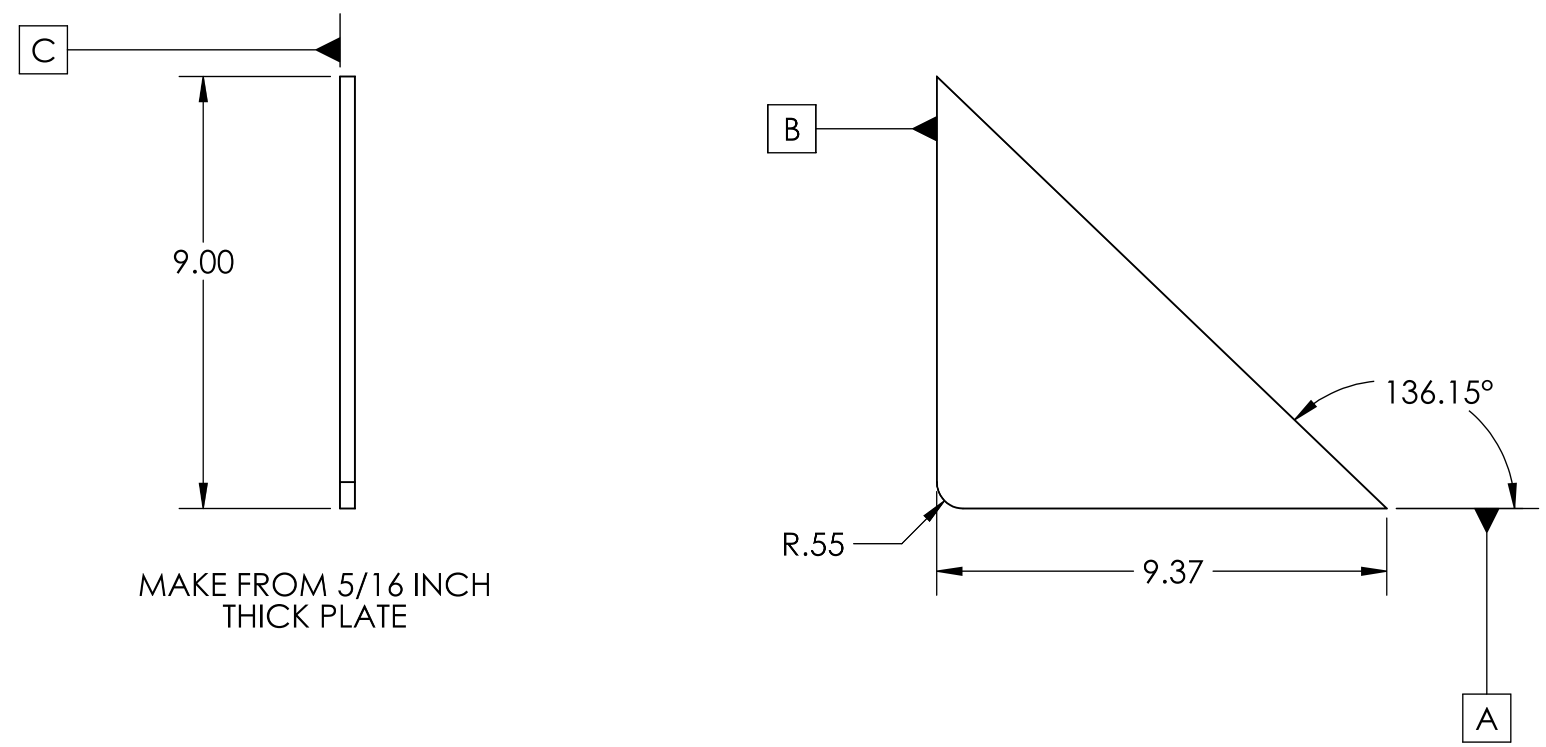
DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. TABLE BRACKET | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION
A	16/12/03	DRAFT

DWN: DDY
CKD: AKERR

DWN:
CKD:

DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

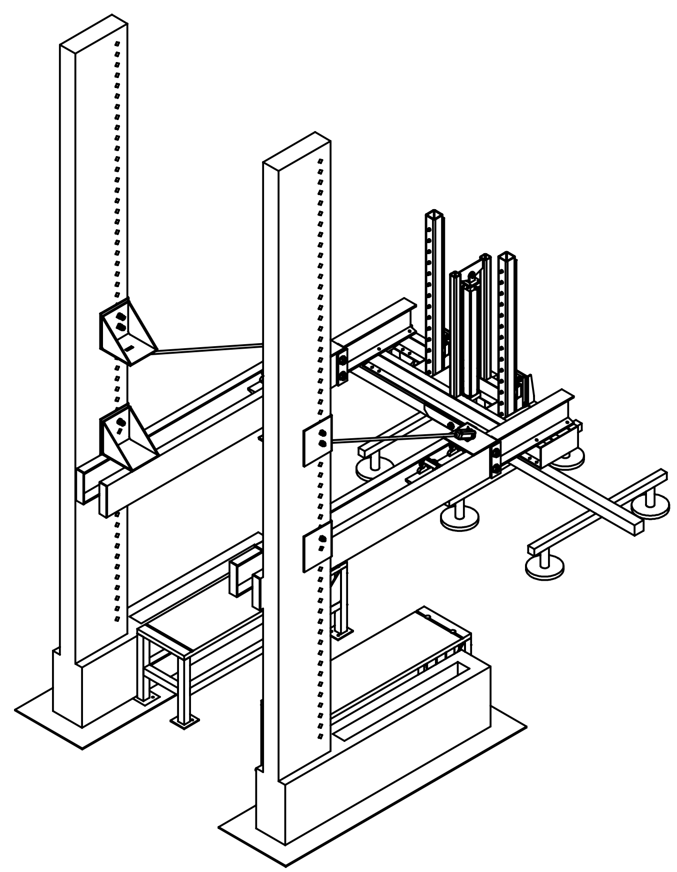
DIMENSIONAL
UNIT: INCHES

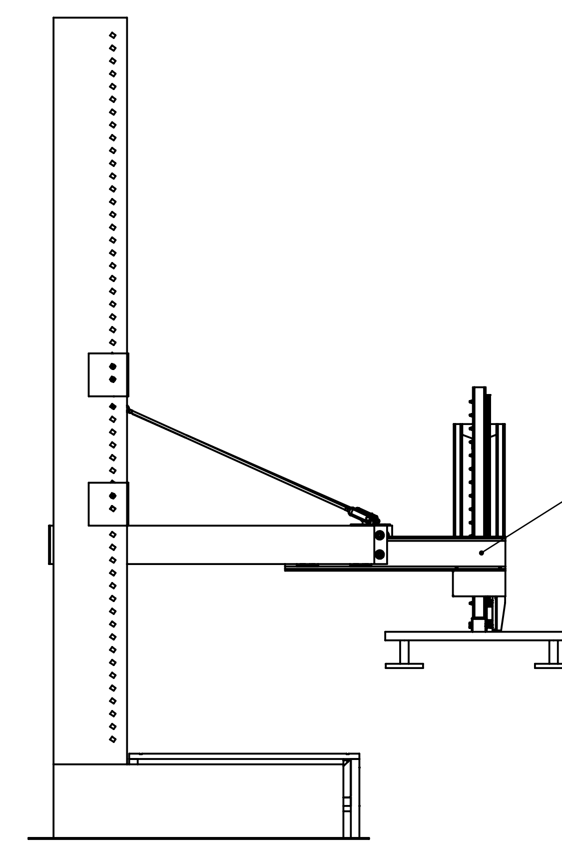
DO NOT SCALE DRAWING

DESCRIPTION:
GUSSET

SIZE B | DWG. NO. PS1M36730 | REV A

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





REV	DATE	DESCRIPTION
A	16/12/07	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/07
CHECKED	AKERR	16/12/07
ENG APPR.		

DIMENSIONAL UNIT: INCHES

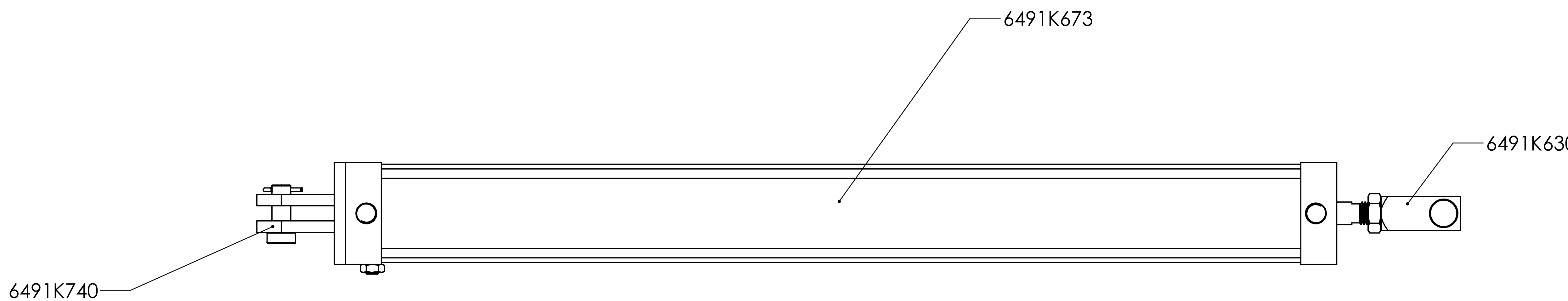
DO NOT SCALE DRAWING

DESCRIPTION:

LIFTER ASSEMBLY

SIZE **B** DWG. NO. LIFTER REV A

SCALE: 1:64 WEIGHT: N/A SHEET 1 OF 1



REV	DATE	DESCRIPTION	
A	16/12/06	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ADD MATERIAL

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

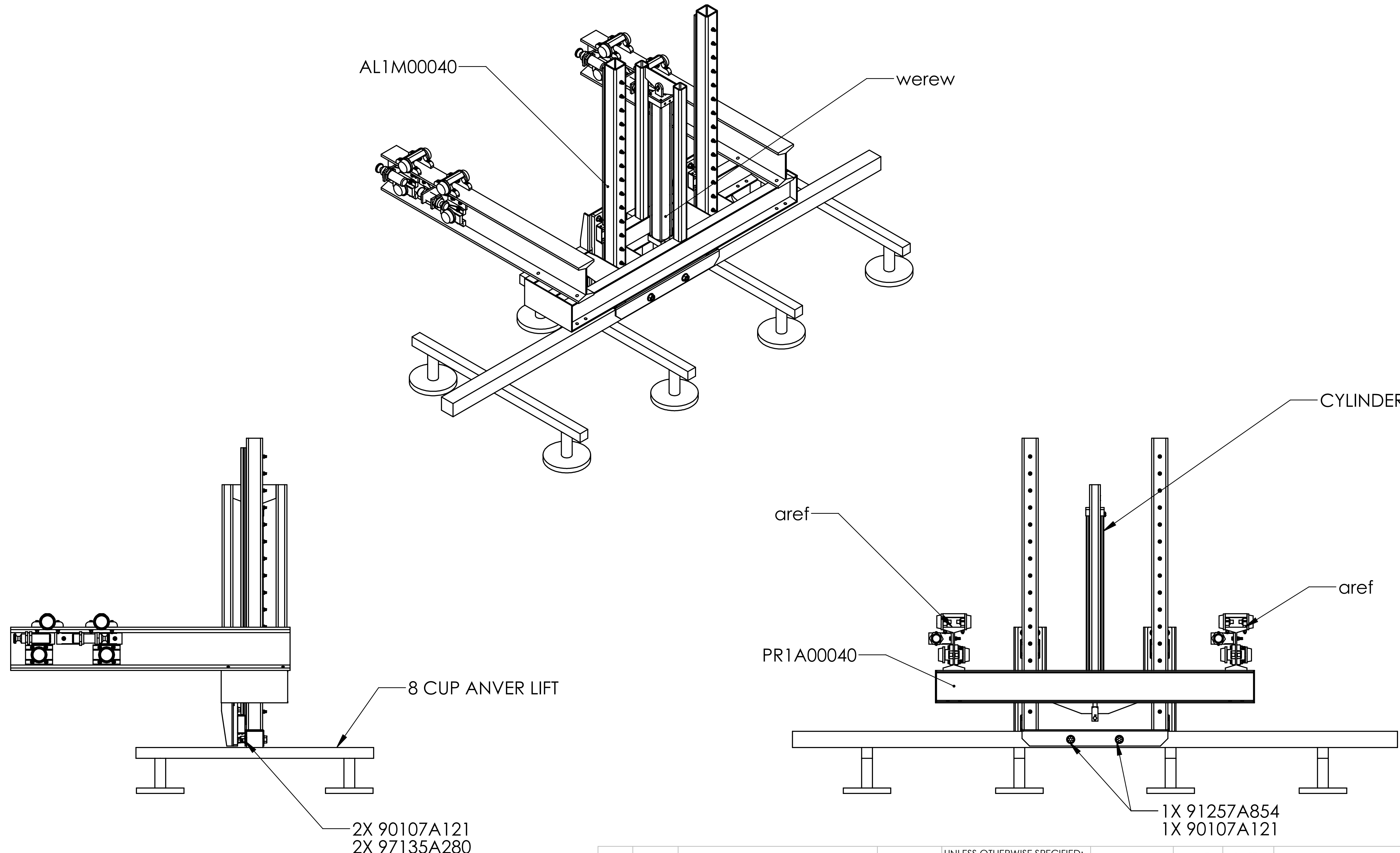
DO NOT SCALE DRAWING

DESCRIPTION:
CYLINDER ASSEMBLY

SIZE B | DWG. NO. CYLINDER ASSEMBLY | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR
DWN:
CKD:
DWN:
CKD:

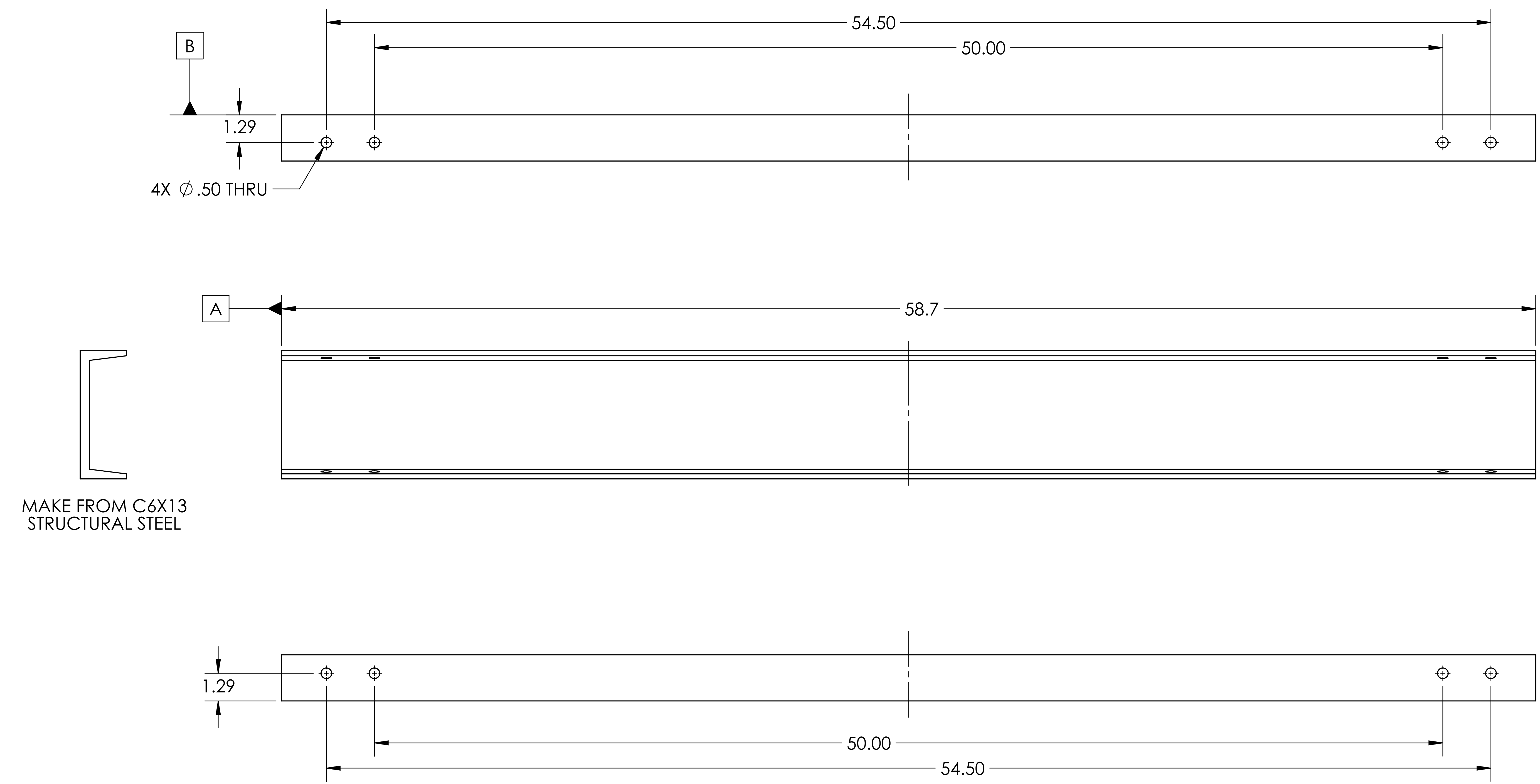
UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009
MATERIAL N/A
FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL UNIT: INCHES
DO NOT SCALE DRAWING

DESCRIPTION:
LIFTER FRAME

SIZE B
DWG. NO. LIFTER
REV A
SCALE: 1:32 WEIGHT: N/A SHEET 1 OF 1



REV	DATE	DESCRIPTION	
A	16/12/04	DRAFT	DWN: AKERR CKD: DDY
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36 STEEL

FINISH 3.2 AS MACHINED

	NAME	DATE
DRAWN	AKERR	16/12/04
CHECKED	DDY	16/12/04
ENG APPR.		

DIMENSIONAL UNIT: INCHES

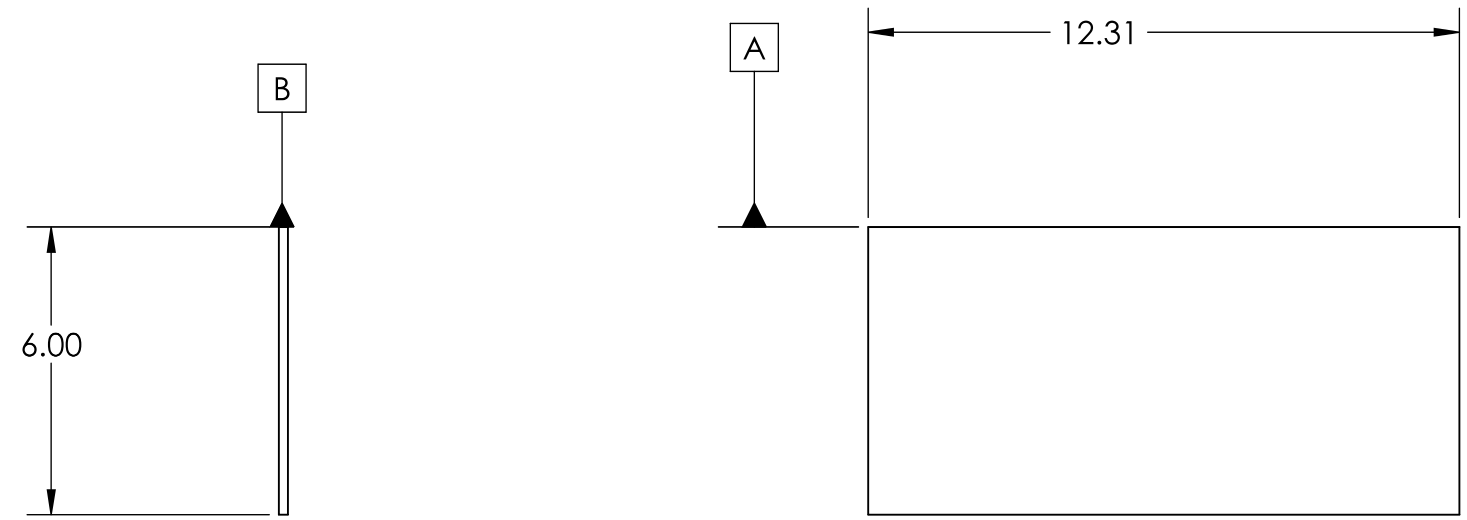
DO NOT SCALE DRAWING

DESCRIPTION:

C-CHANNEL

SIZE	DWG. NO.	REV
B	PS2M13440	A
SCALE: 1:5	WEIGHT: N/A	SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only



MAKE FROM
0.19 INCH THICK SHEET

REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

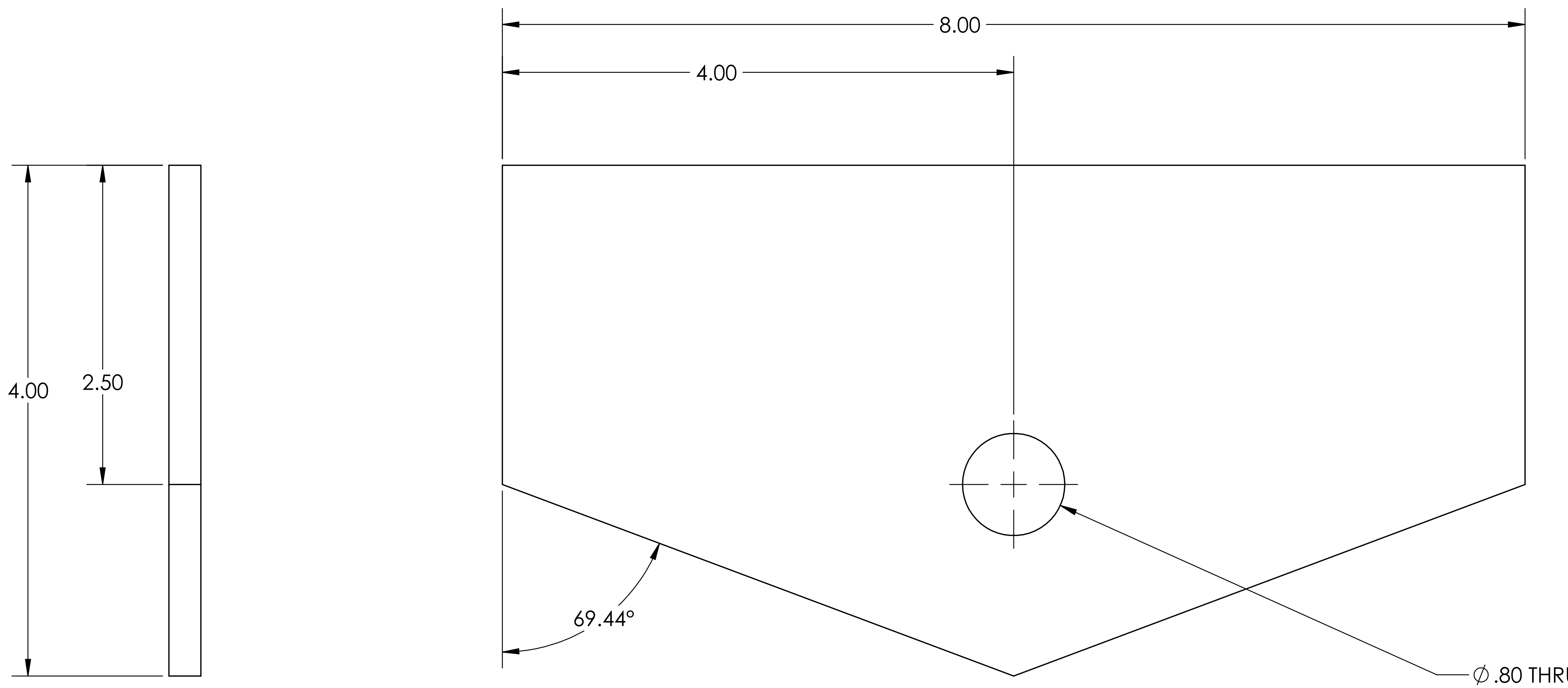
DO NOT SCALE DRAWING

DESCRIPTION:
PLATE

SIZE **B** DWG. NO. P2M37740 REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



MAKE FROM
0.25 INCH THICK PLATE

REV	DATE	DESCRIPTION	
A	16/12/06	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH
3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

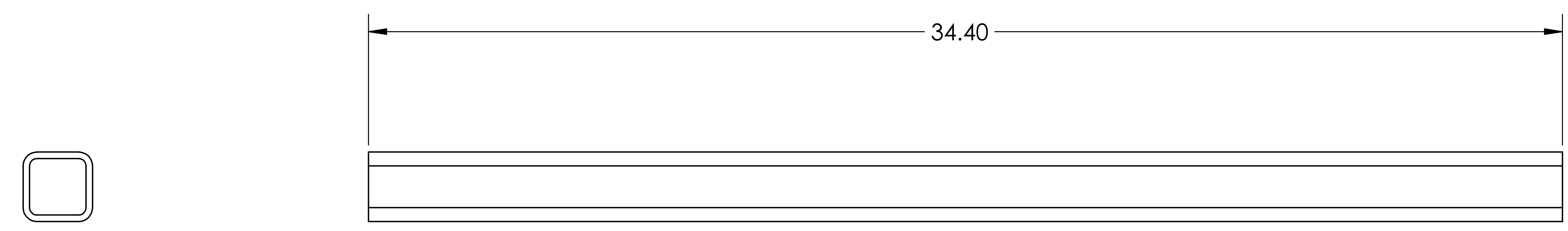
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
CLEVIS PLATE

SIZE	DWG. NO.	REV
B	PS3M27140	A

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



REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

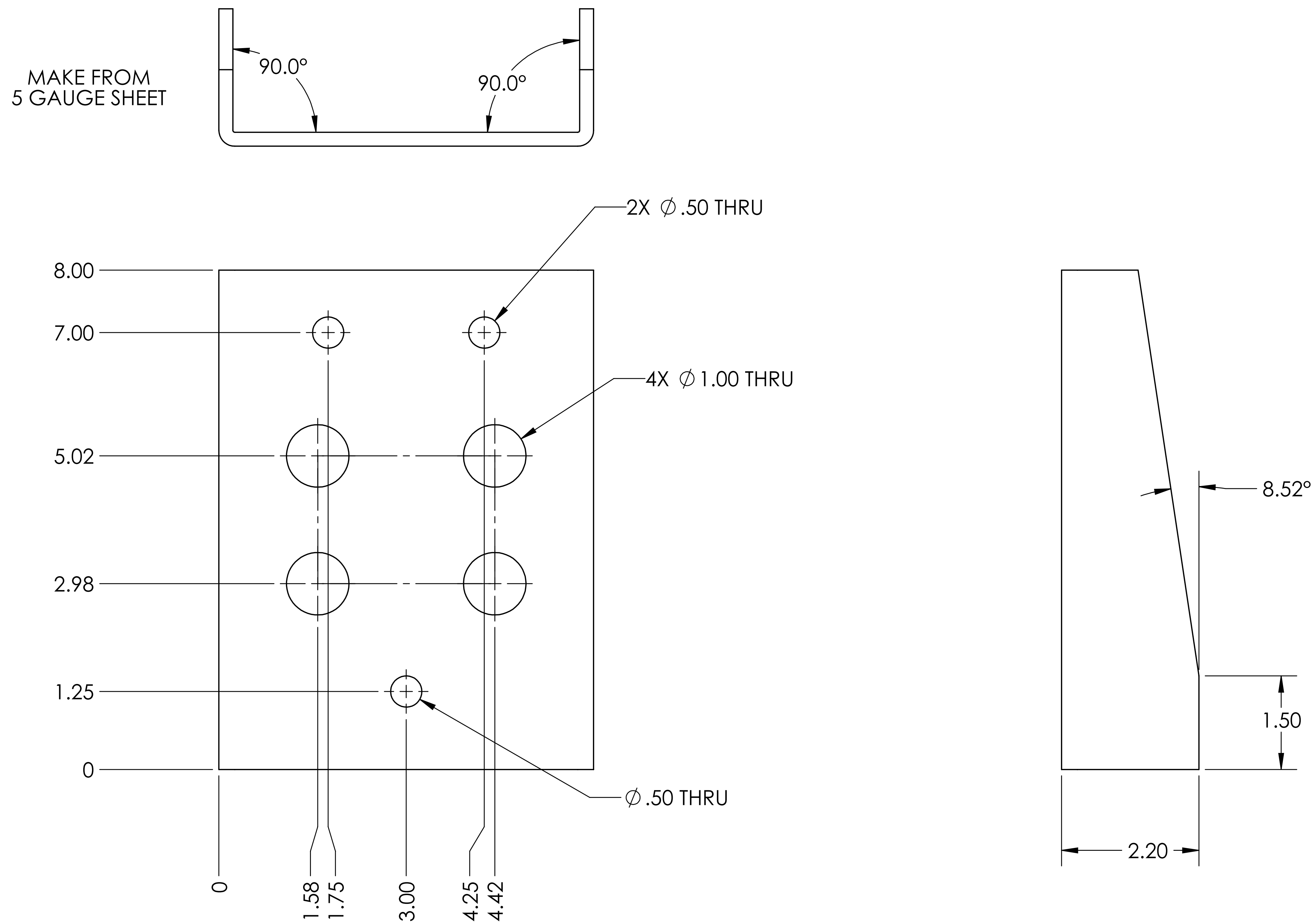
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
CARRIAGE SUPPORT

SIZE B | DWG. NO. PS2M28740 | REV A

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



REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR

DWN:
CKD:

DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

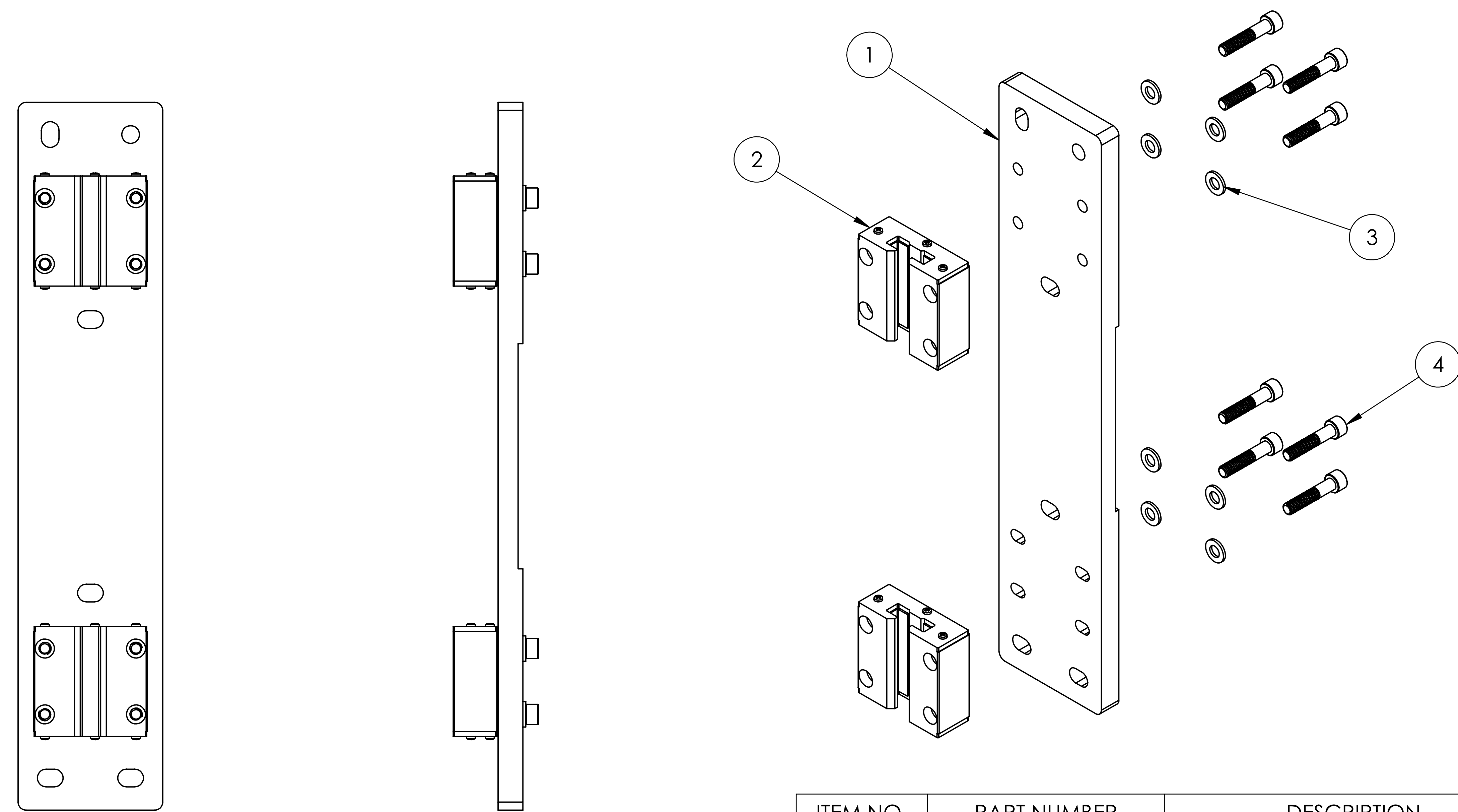
DO NOT SCALE DRAWING

DESCRIPTION:
CARRIAGE MOUNT

SIZE B | DWG. NO. PS3M36730 | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PL1M27341	BEARING PLATE, FIXED	1
2	TW-02-30	LINEAR BEARING CARRIAGE	2
3	90965A200	M10 SS WASHER	8
4	91290A532	M10X1.5X50LG SOCKET CAP SCREW	8

REV	DATE	DESCRIPTION	
A	16/12/04	DRAFT	DWN: AKERR CKD: DDY
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL SEE BOM

FINISH N/A

	NAME	DATE
DRAWN	AKERR	16/12/03
CHECKED	DDY	16/12/04
ENG APPR.		

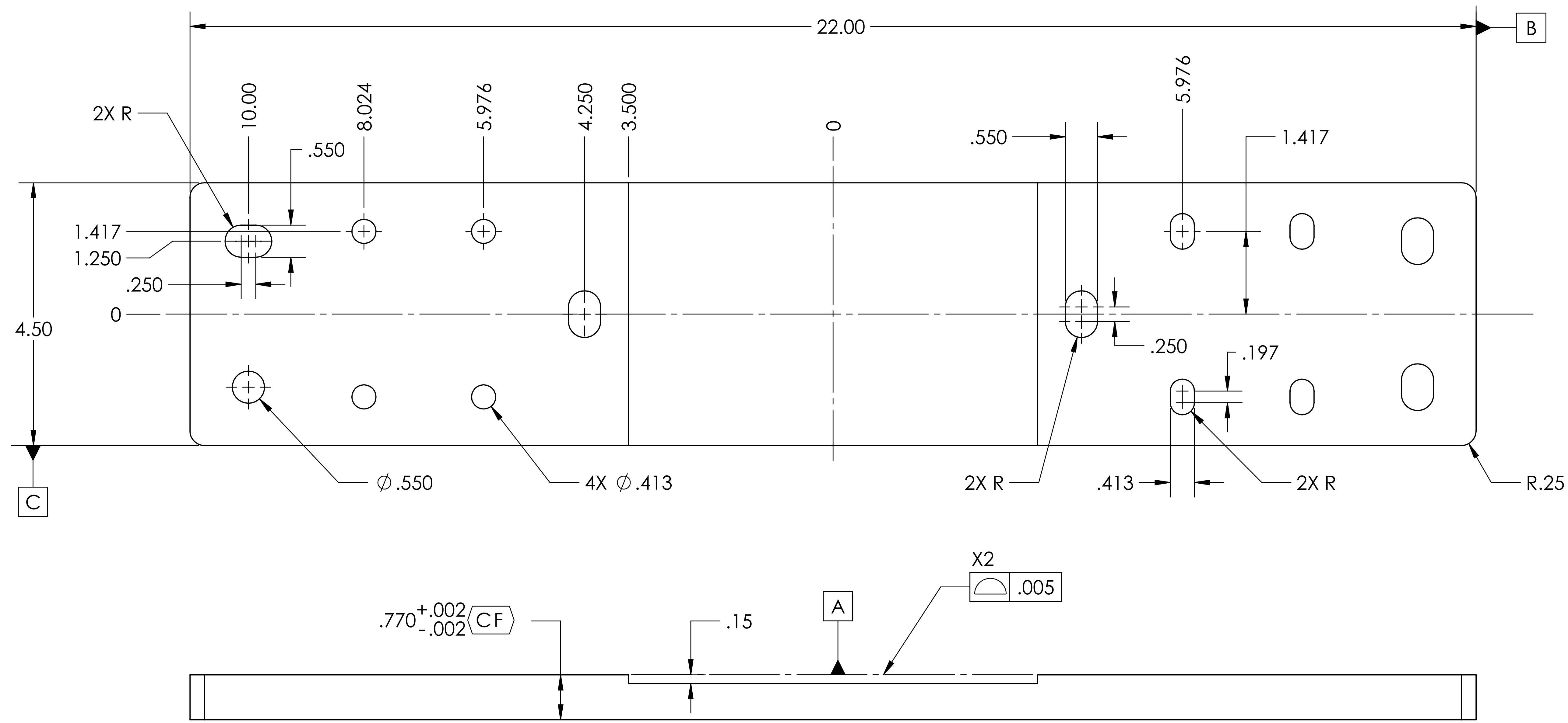
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
BEARING CARRIAGE ASSEMBLY

SIZE B | DWG. NO. AL11000042 | REV A

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



REV	DATE	DESCRIPTION		UNLESS OTHERWISE SPECIFIED:
A	16/12/03	DRAFT	DWN: AKERR CKD:	TOLERANCES: ANGULAR: MACH ± 1° BEND ± 1° ONE PLACE DECIMAL ± 0.1 TWO PLACE DECIMAL ± 0.01 THREE PLACE DECIMAL ± 0.005 INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009
			DWN: CKD:	MATERIAL AA 6061 T6
			DWN: CKD:	FINISH 0.8 AS MACHINED

	NAME	DATE
DRAWN	AKERR	16/12/03
CHECKED		
ENG APPR.		

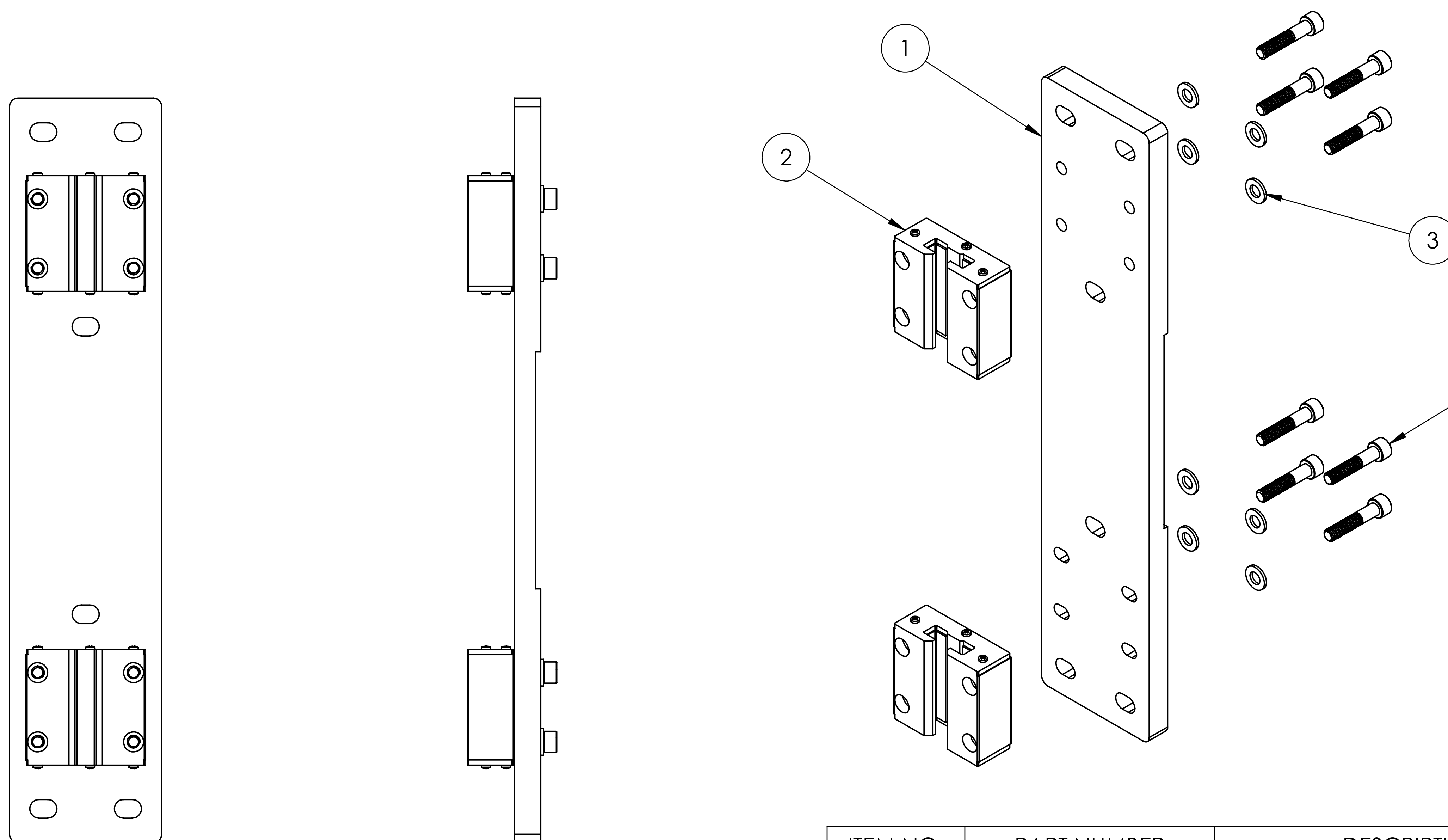
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

TITLE:
BEARING PLATE, FIXED

SIZE	DWG. NO.	REV
B	PL1M27342	A

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



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PL1M27342	BEARING PLATE, FLOATING	1
2	TW-02-30LLZ	LINEAR BEARING CARRIAGE, FLOATING	2
3	90965A200	M10 SS WASHER	8
4	91290A532	M10X1.5X50LG SOCKET CAP SCREW	8

REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: AKERR CKD: DDY
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL SEE BOM

FINISH N/A

	NAME	DATE
DRAWN	AKERR	16/12/03
CHECKED	DDY	16/12/04
ENG APPR.		

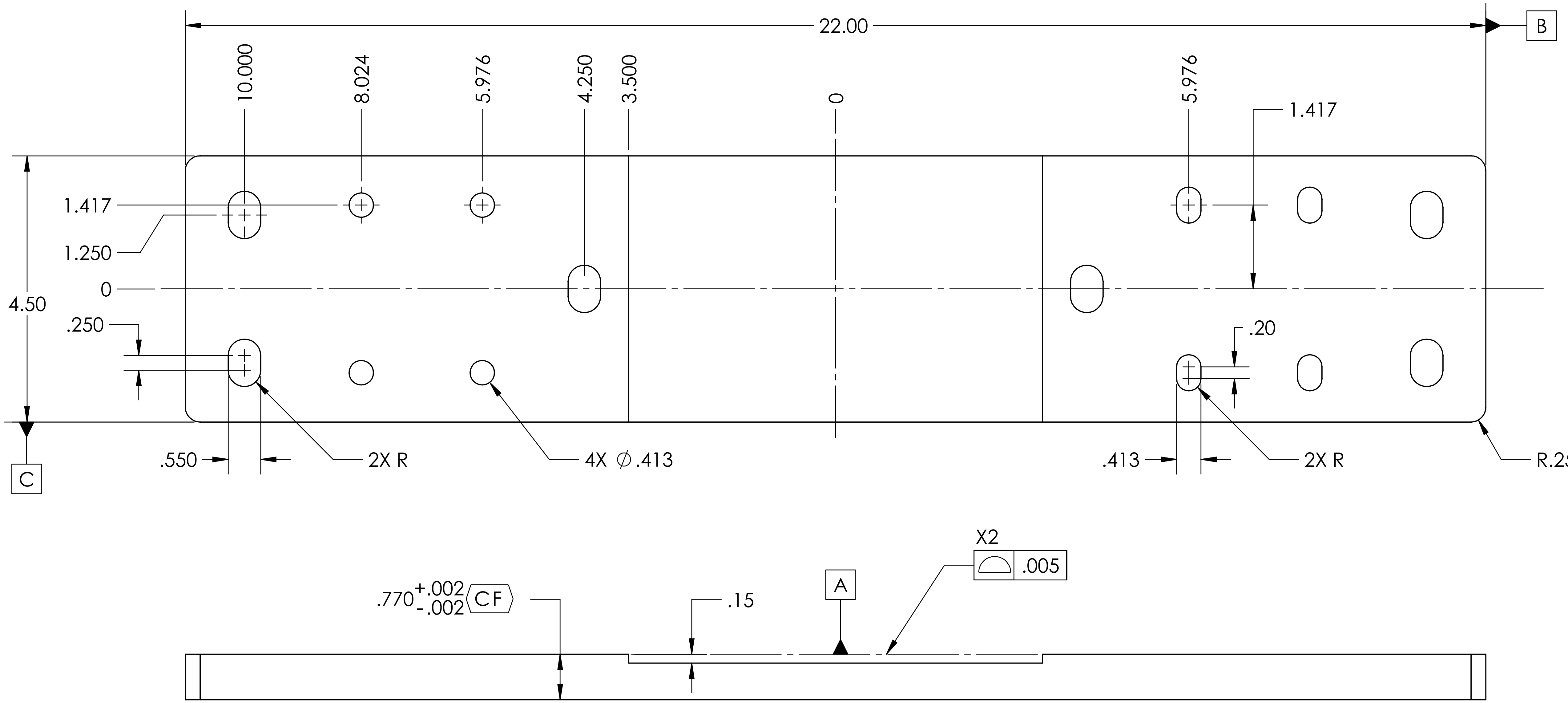
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION: BEARING CARRIAGE ASSEMBLY, FLOATING

SIZE B | DWG. NO. AL11000042 | REV A

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



REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: AKERR CKD:
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± 0.1
TWO PLACE DECIMAL ± 0.01
THREE PLACE DECIMAL ± 0.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AA 6061 T6

FINISH 0.8 AS MACHINED

	NAME	DATE
DRAWN	AKERR	16/12/03
CHECKED		
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

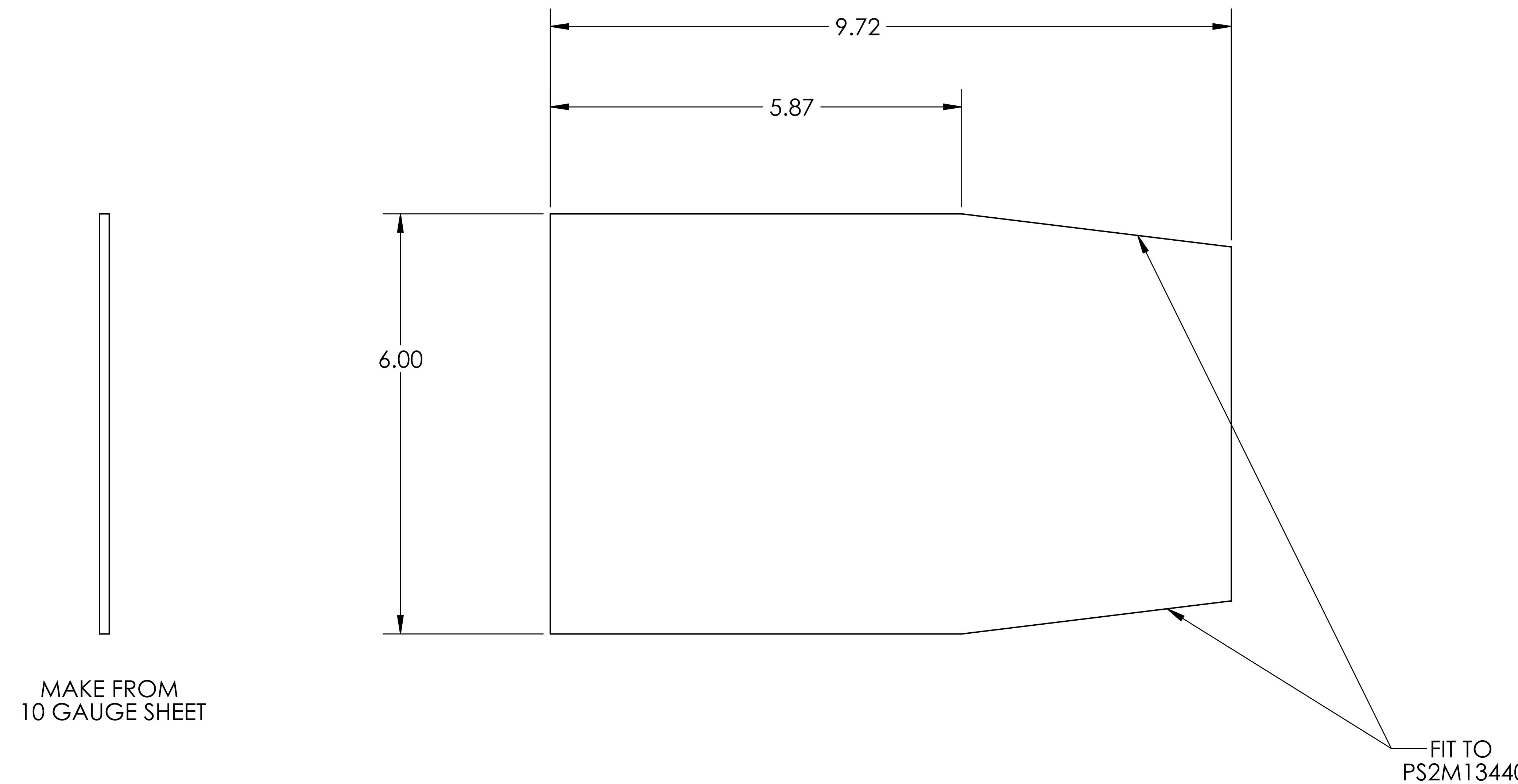
DO NOT SCALE DRAWING

TITLE:
BEARING PLATE, FLOATING

SIZE	DWG. NO.	REV
B	PL1M27342	A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR

DWN:
CKD:

DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

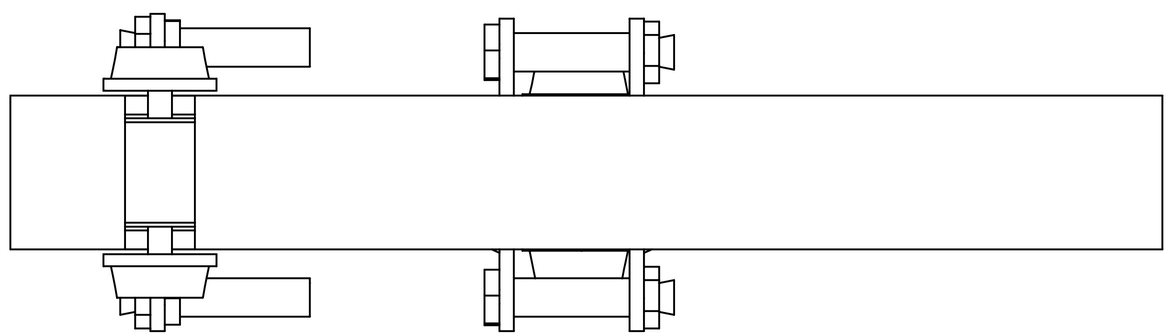
DESCRIPTION:
STIFFENER PLATE

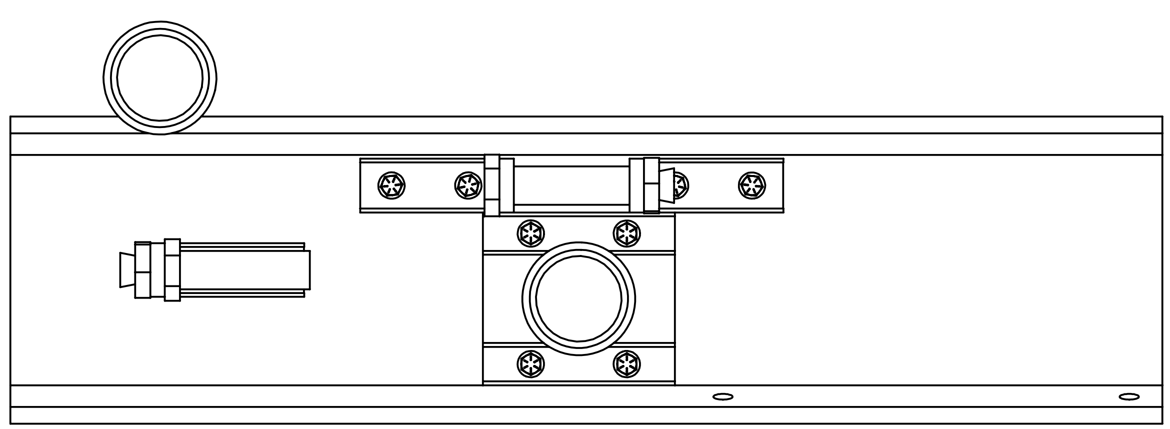
SIZE **B** DWG. NO. STIFFENER PLATER REV A

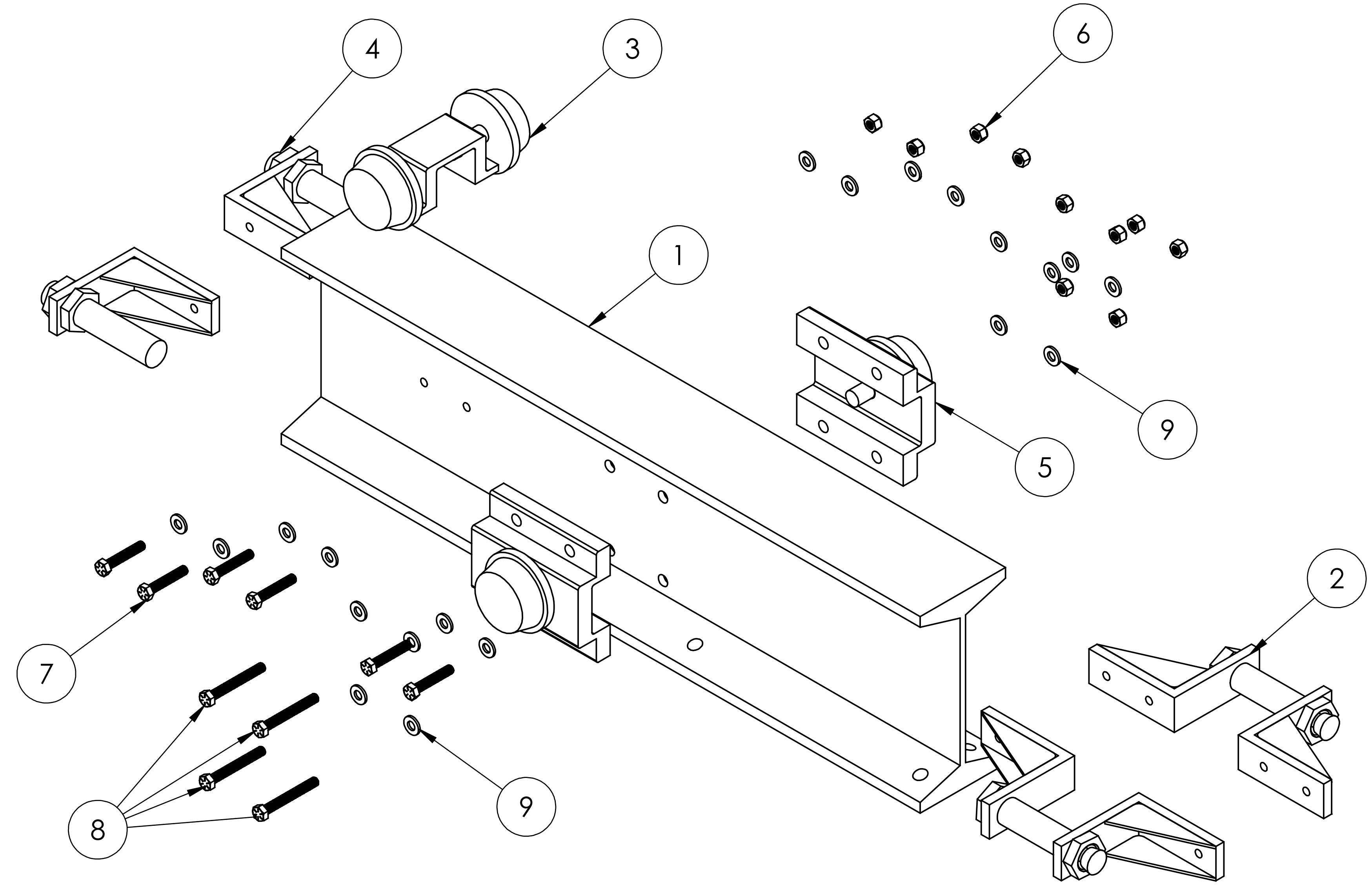
SCALE: 1:2 WEIGHT: N/A SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	PS1P21440	I BEAM	1
2	SMALL DAMPER ASSEMBLY	N/A	2
3	UPPER BEARING MOUNT	N/A	1
4	REAR DAMPER ASSEMBLY	N/A	2
5	TAPER BEARING MOUNT	N/A	2
6	97135A225	NUT	10
7	92620A305	BOLT	6
8	92620A613	BOLT	4
9	93960A140	WASHER	20







REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: YLEE
CKD:
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
SEE BOM

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	YLEE	16/12/06
CHECKED		
ENG APPR.		

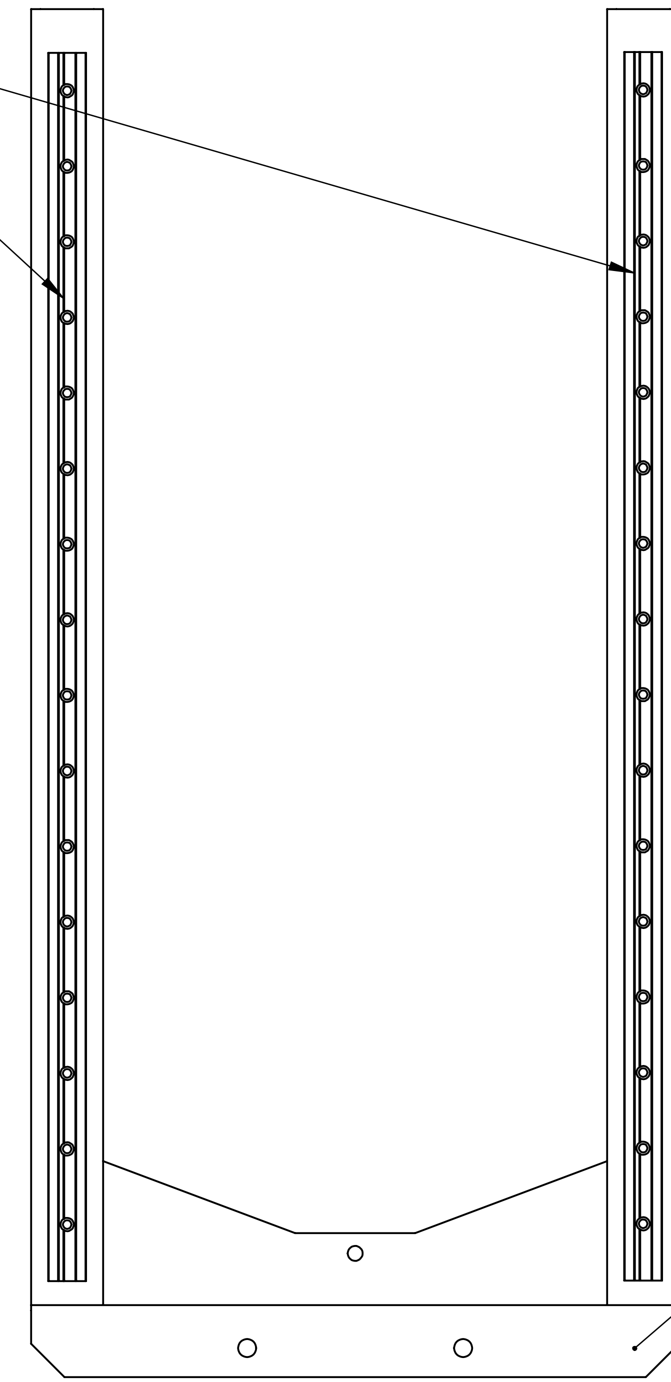
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
I BEAM ASSEMBLY

SIZE B DWG. NO. AL1M00040 REV A

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



REV	DATE	DESCRIPTION	
A	16/12/06	DRAFT	DWN: DDY CKD:AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

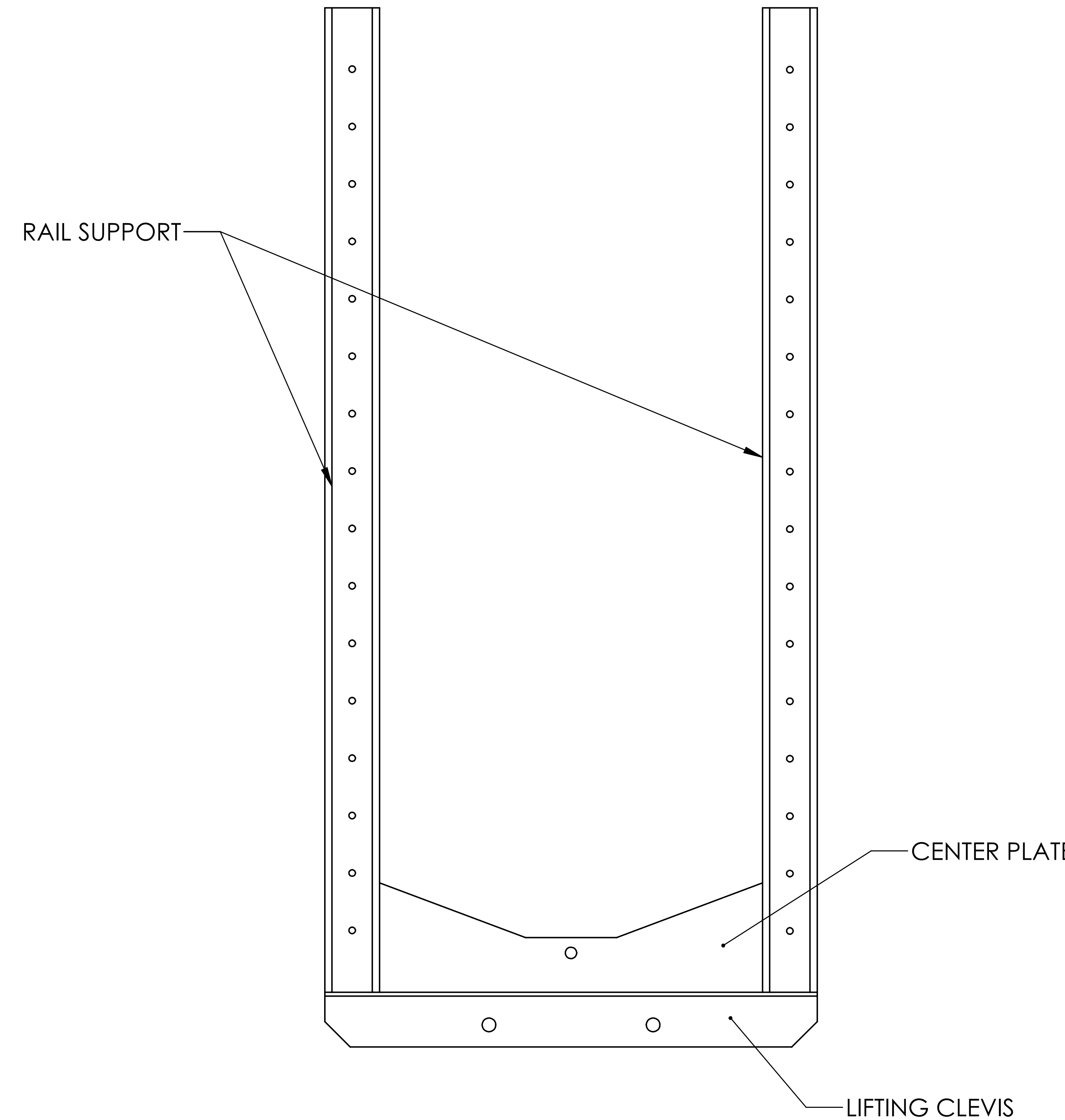
DO NOT SCALE DRAWING

DESCRIPTION:

CARRIAGE RAIL

SIZE **B** DWG. NO. AR1A00040 REV A

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



REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL UNIT: INCHES

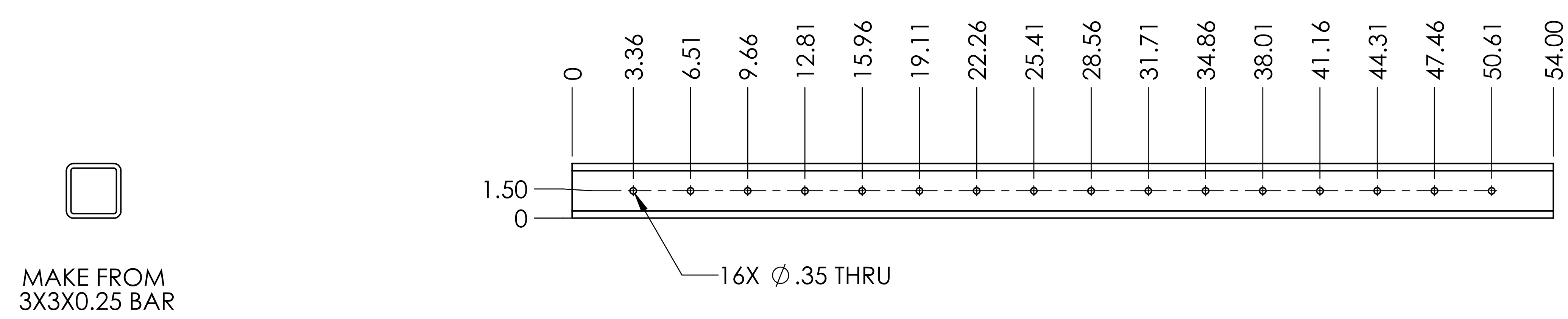
DO NOT SCALE DRAWING

DESCRIPTION:

CARRIAGE FRAME

SIZE B | DWG. NO. AR3A00040 | REV A

SCALE: 1:12 | WEIGHT: N/A | SHEET 1 OF 1



REV	DATE	DESCRIPTION	
A	16/12/06	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

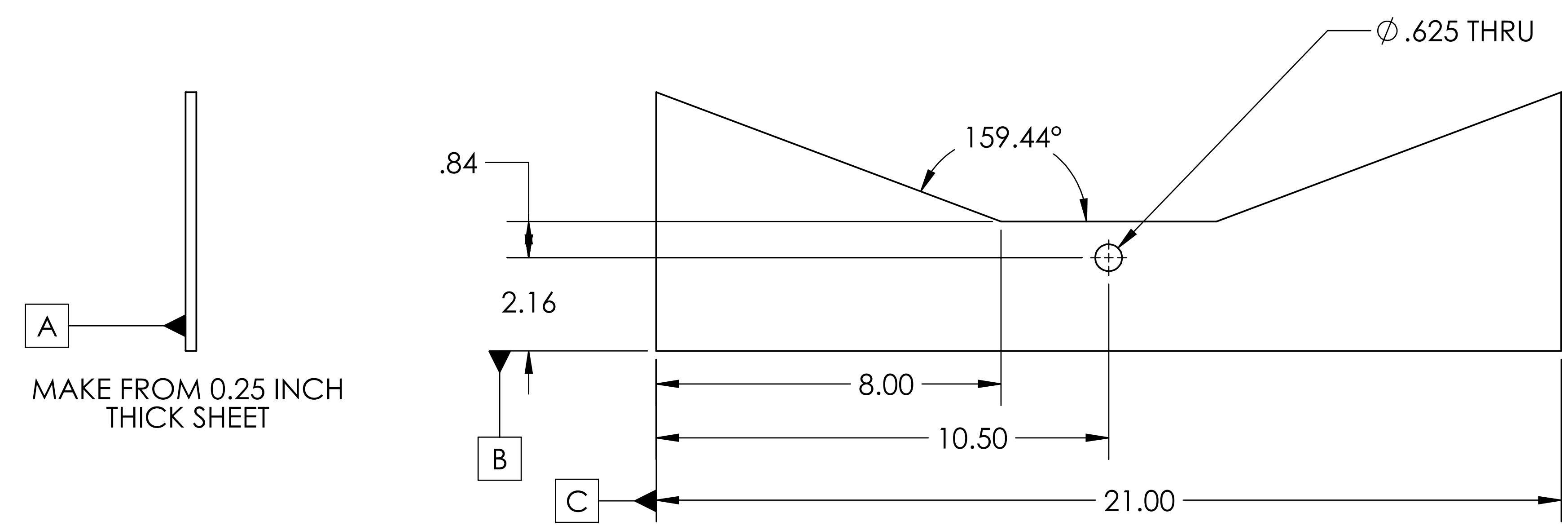
DO NOT SCALE DRAWING

DESCRIPTION:
RAIL SUPPORT

SIZE B | DWG. NO. RAIL SUPPORT | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

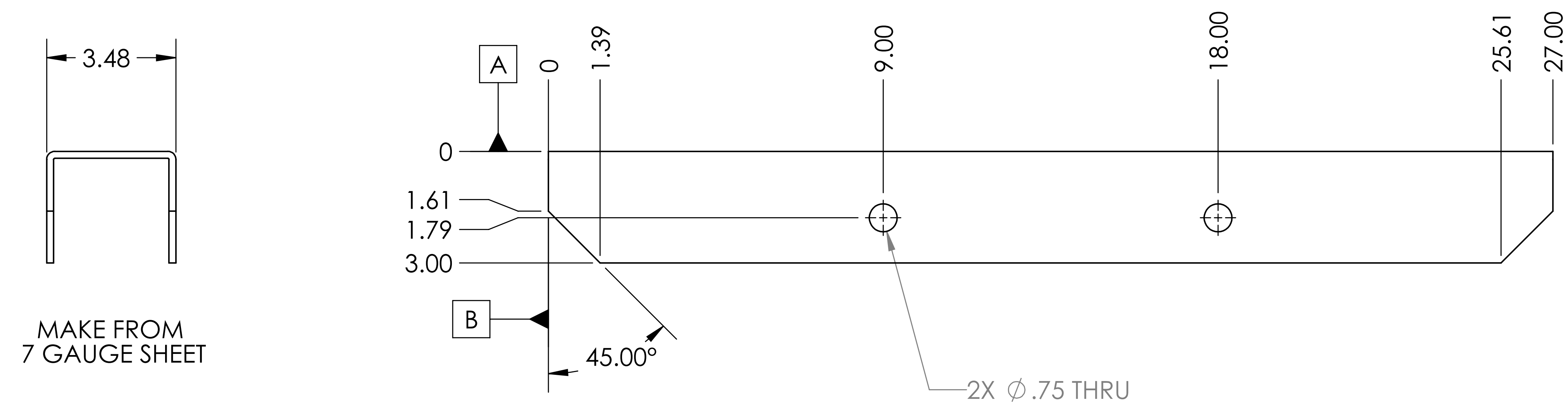
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
CENTER PLATE

SIZE B | DWG. NO. CENTER PLATE | REV A

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



REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

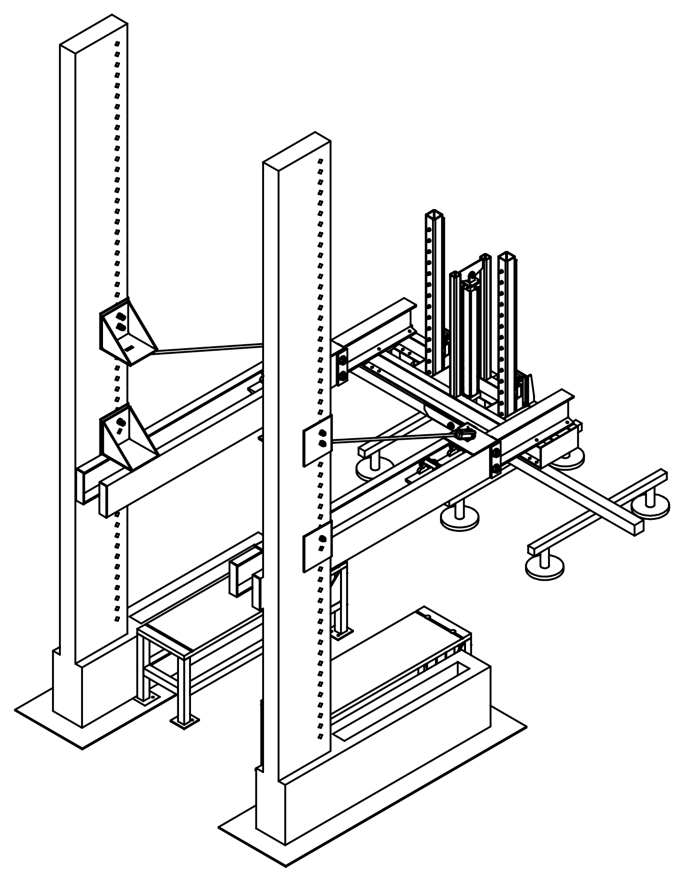
DO NOT SCALE DRAWING

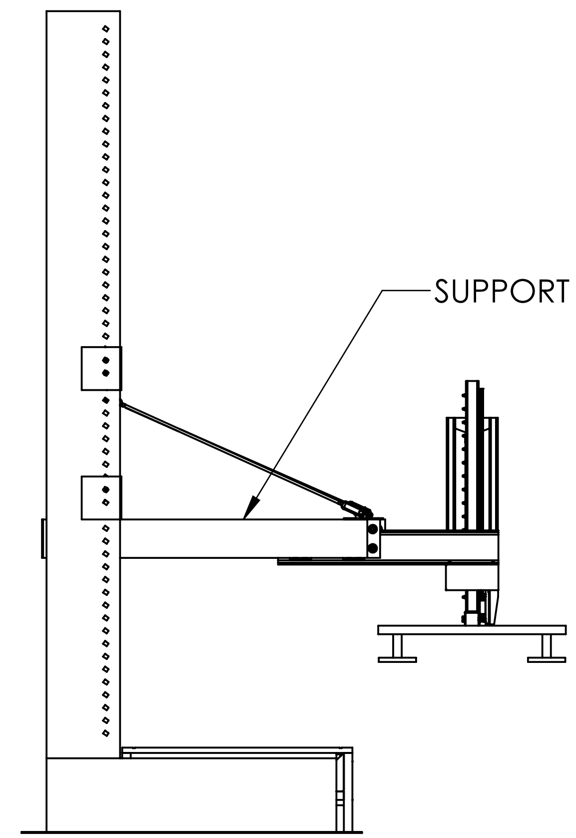
DESCRIPTION:
LIFTING CLEVIS

SIZE B
DWG. NO. LIFTING CLEVIS
REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only





REV	DATE	DESCRIPTION
A	16/12/07	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/07
CHECKED	AKERR	16/12/07
ENG APPR.		

DIMENSIONAL UNIT: INCHES

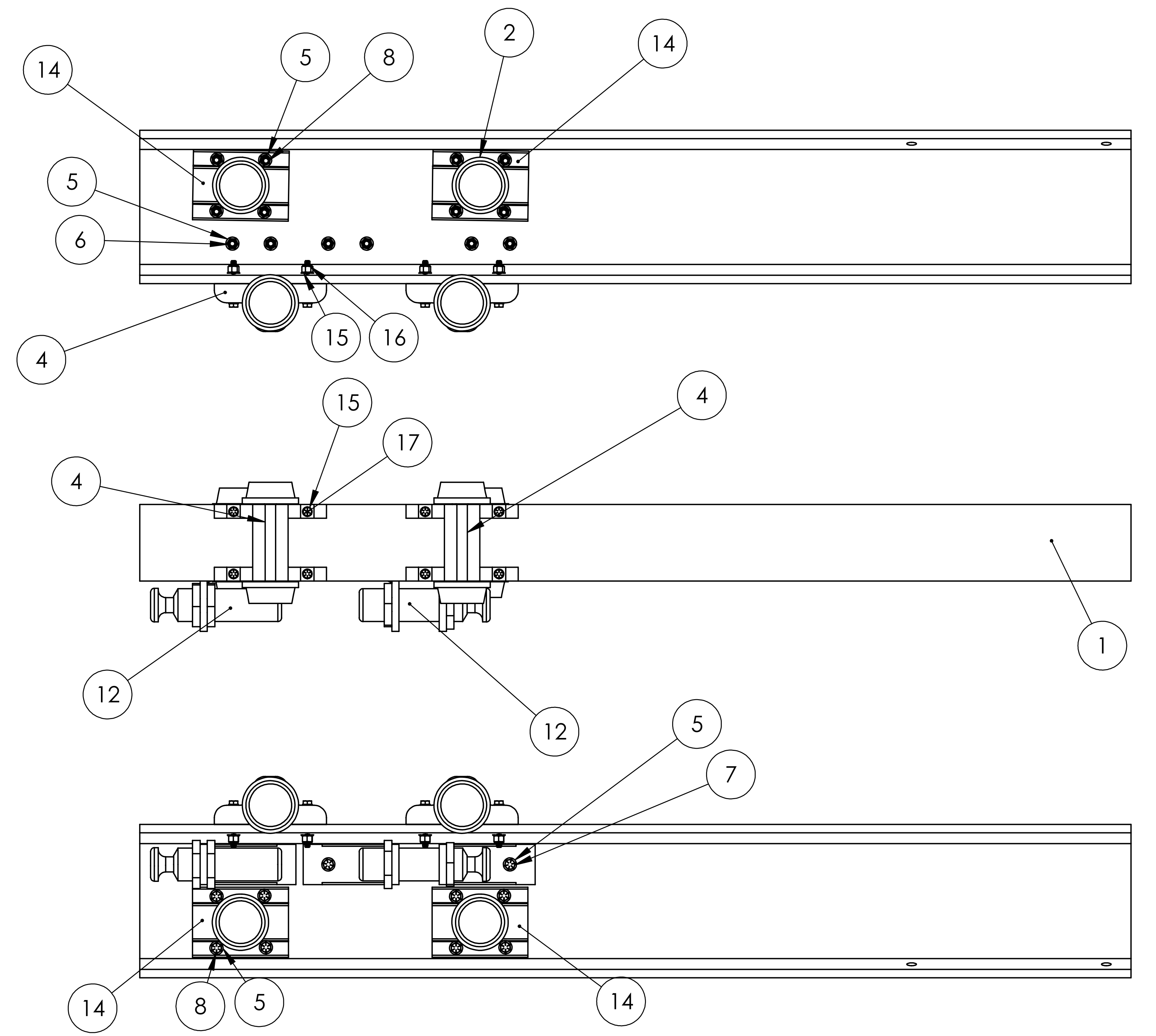
DO NOT SCALE DRAWING

DESCRIPTION:
SUPPORT STRUCTURE

SIZE B DWG. NO. SUPPORT STRUCTURE REV A

SCALE: 1:45 WEIGHT: N/A SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only



ITEM NO.	PART NUMBER	QTY.
1	PS1P21440	1
2	CIR-300-R	8
3	Locknut Roller	8
4	Roller Mount Upper New	2
5	93960A140	28
6	97135A225	14
7	92620A305	6
8	92620A613	8
9	Smaller Damper Assembly	3
10	9530K18	4
11	Damper Gusset	6
12	Damper Body	2
13	Damper Bumper	2
14	Taper Bearing Mount	4
15	92916A355	16
16	90648A029	8
17	91257A551	8

REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN:
CKD:
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

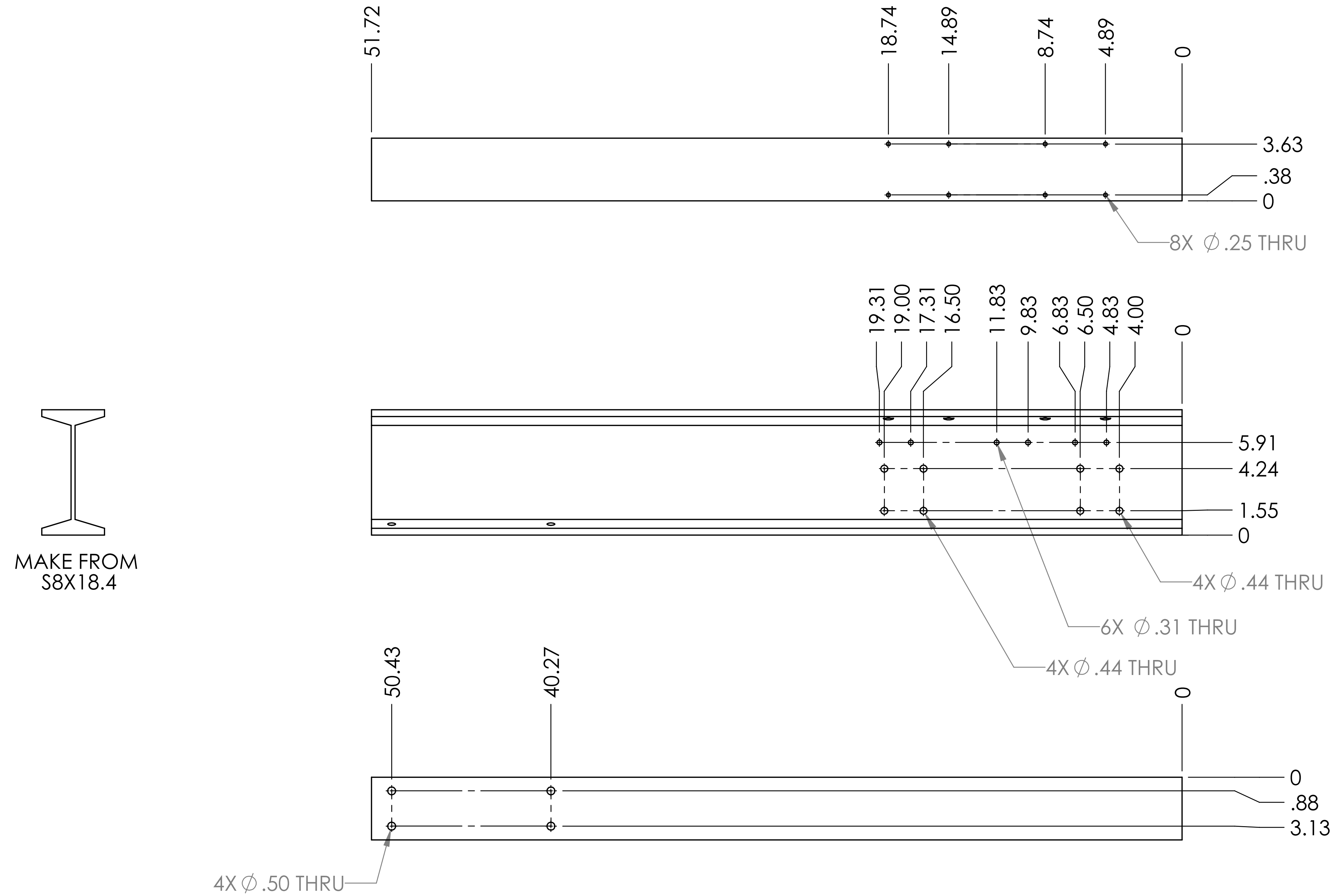
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
SLIDING SUPPORT

SIZE B DWG. NO. AL1M00040 REV A

SCALE: 1:7 WEIGHT: N/A SHEET 1 OF 1



REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN: DDY
CKD: AKERR
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

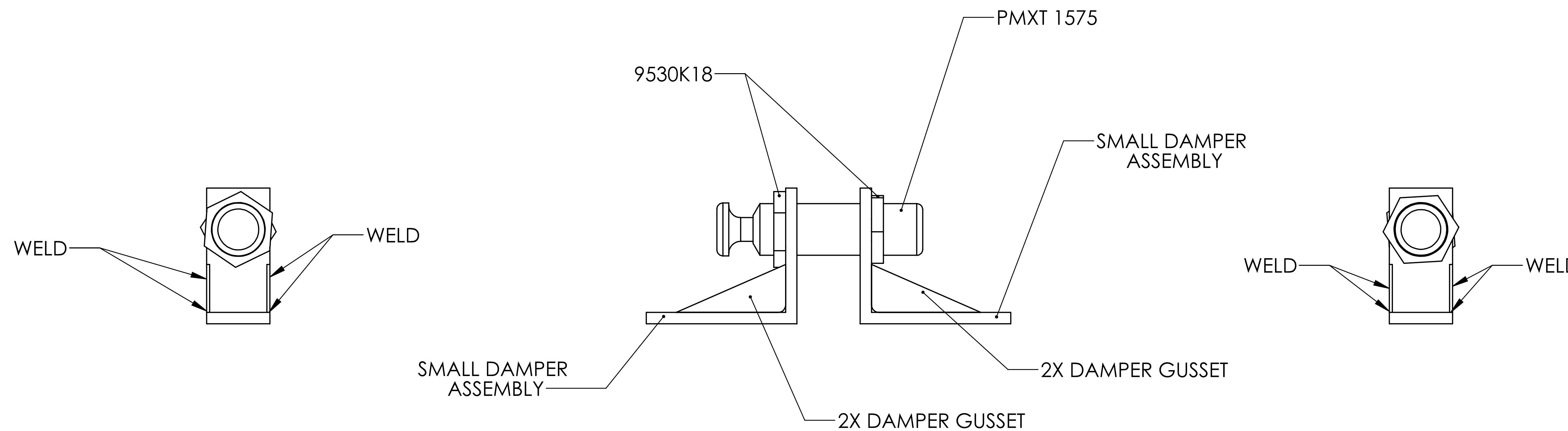
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. PS1P21440 | REV A

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



REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH NO PLATING 3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

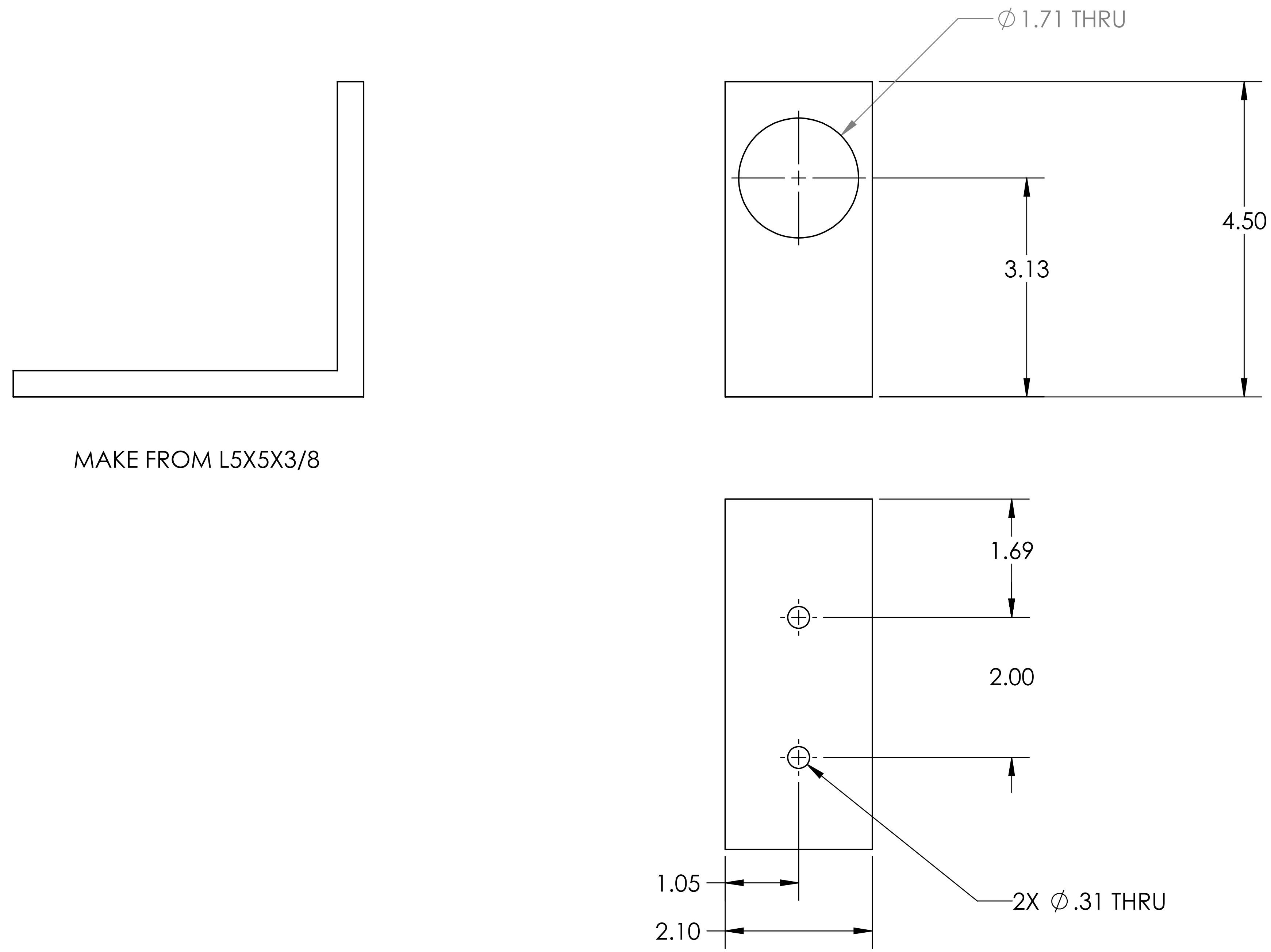
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B DWG. NO. SMALL DAMPER ASSEMBLY REV A

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



DDY

REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

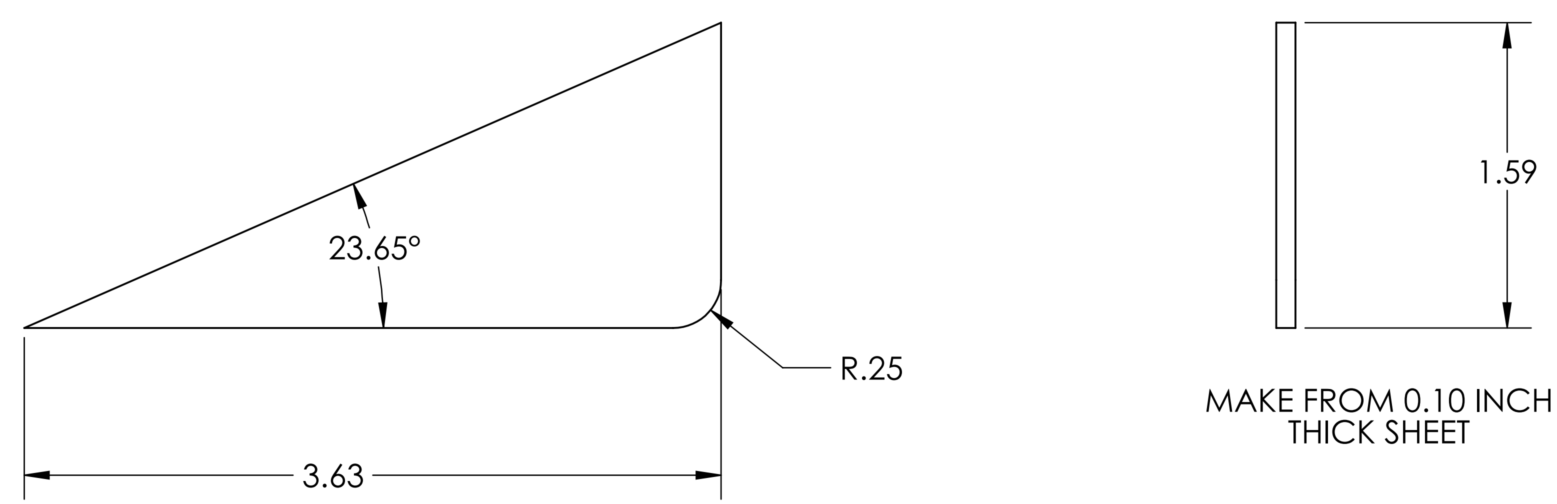
DESCRIPTION:

SIZE B

DWG. NO.
SMALL DAMPER ASSEMBLY

REV A

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



REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

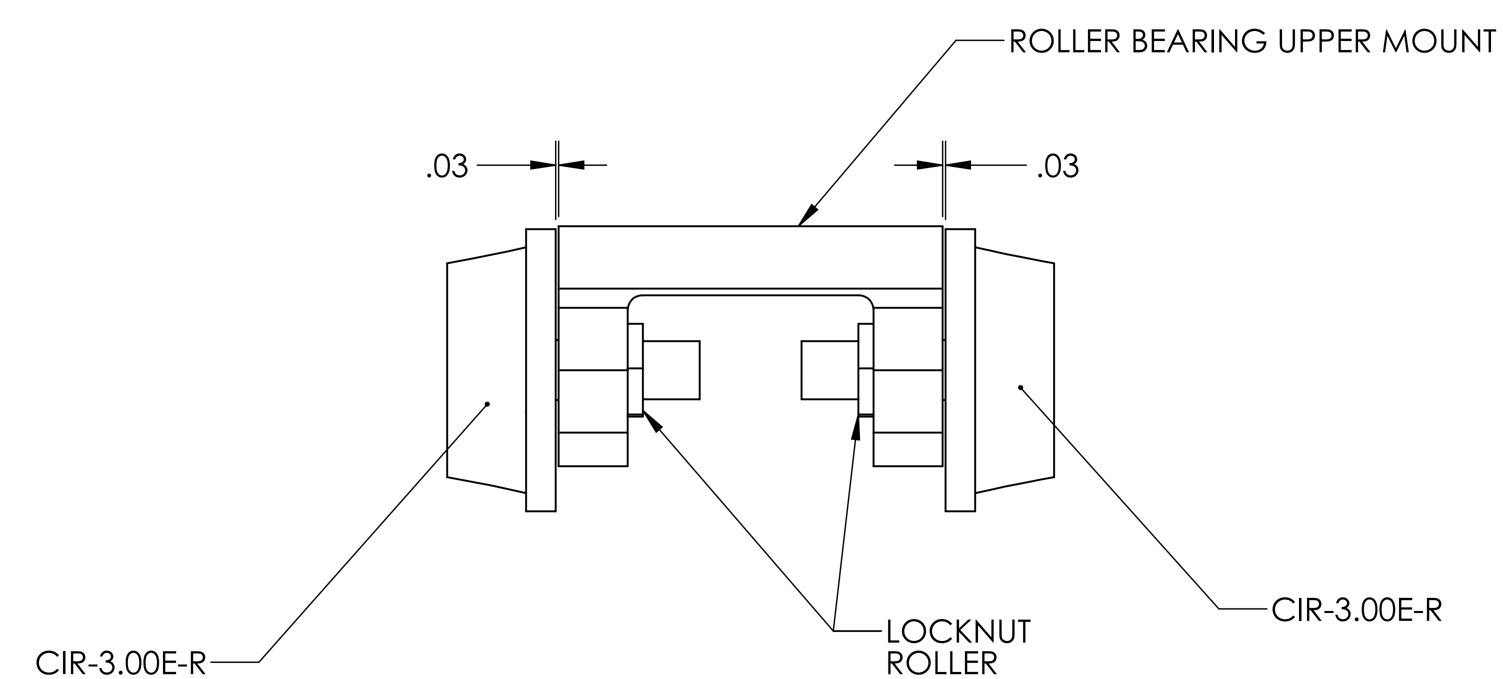
DO NOT SCALE DRAWING

DESCRIPTION:
DAMPER GUSSET

SIZE B | DWG. NO. DAMPER GUSSET | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



SOLIDWORKS Educational Product. For Instructional Use Only

REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH NO PLATING 3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

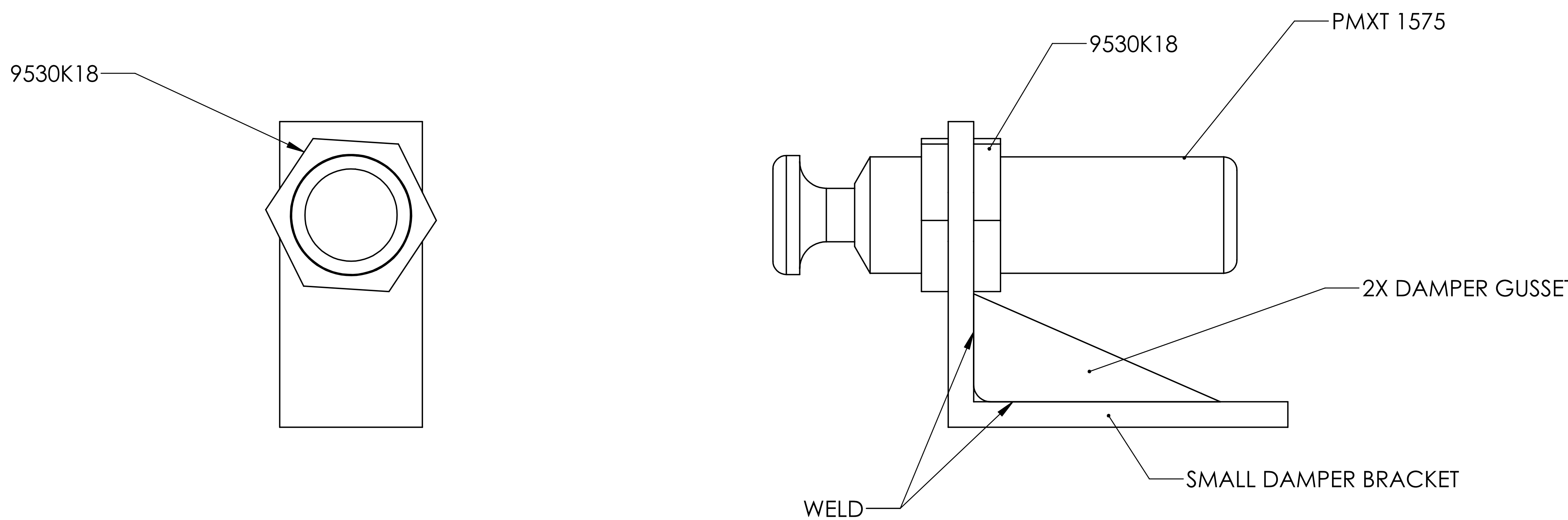
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. UPPER BEARING ASSEMBLY | REV A

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



REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

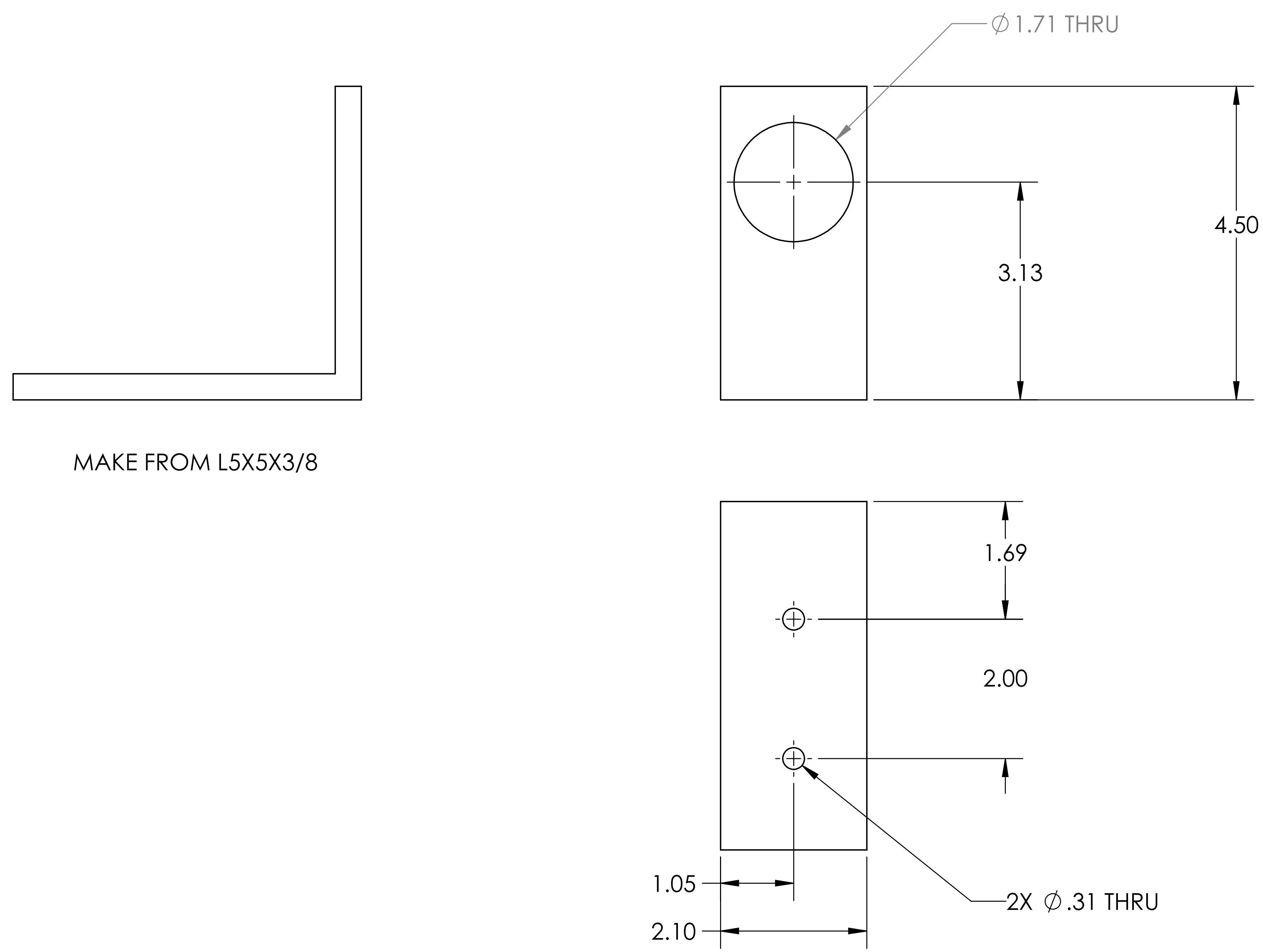
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
REAR DAMPER ASSEMBLY

SIZE B | DWG. NO. REAR DAMPER ASSEMBLY | REV A

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



DDY

REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL UNIT: INCHES

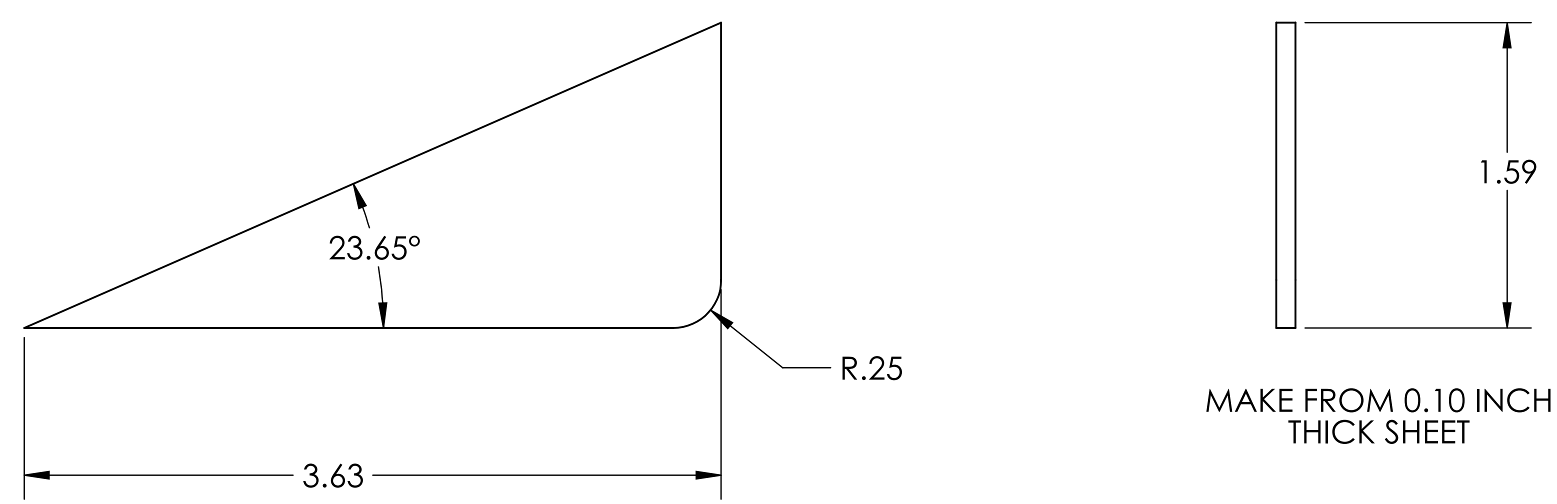
DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. SMALL DAMPER ASSEMBLY | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL: AISI 1020

FINISH: NO PLATING (3.2)

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL UNIT: INCHES

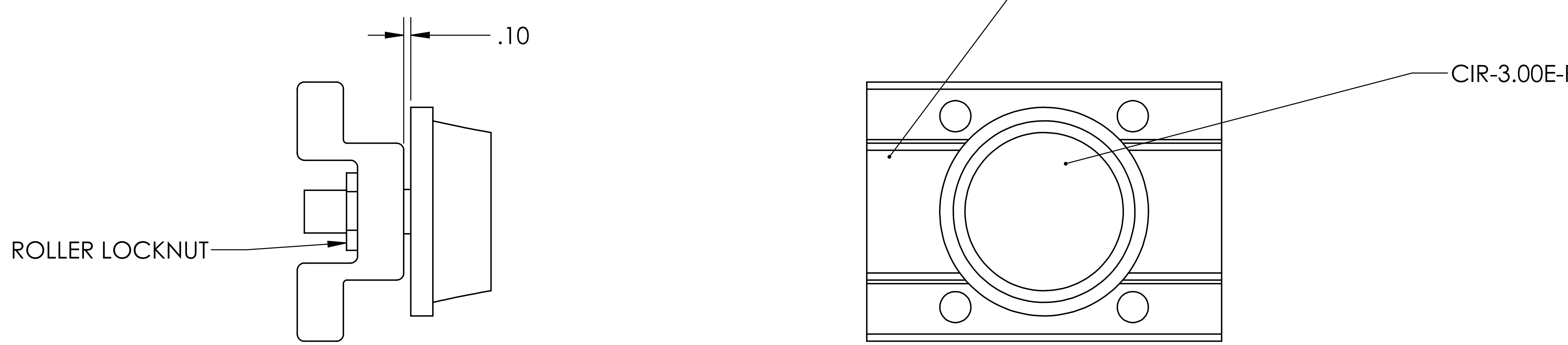
DO NOT SCALE DRAWING

DESCRIPTION: DAMPER GUSSET

SIZE: B | DWG. NO.: DAMPER GUSSET | REV: A

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

180



REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH NO PLATING 3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

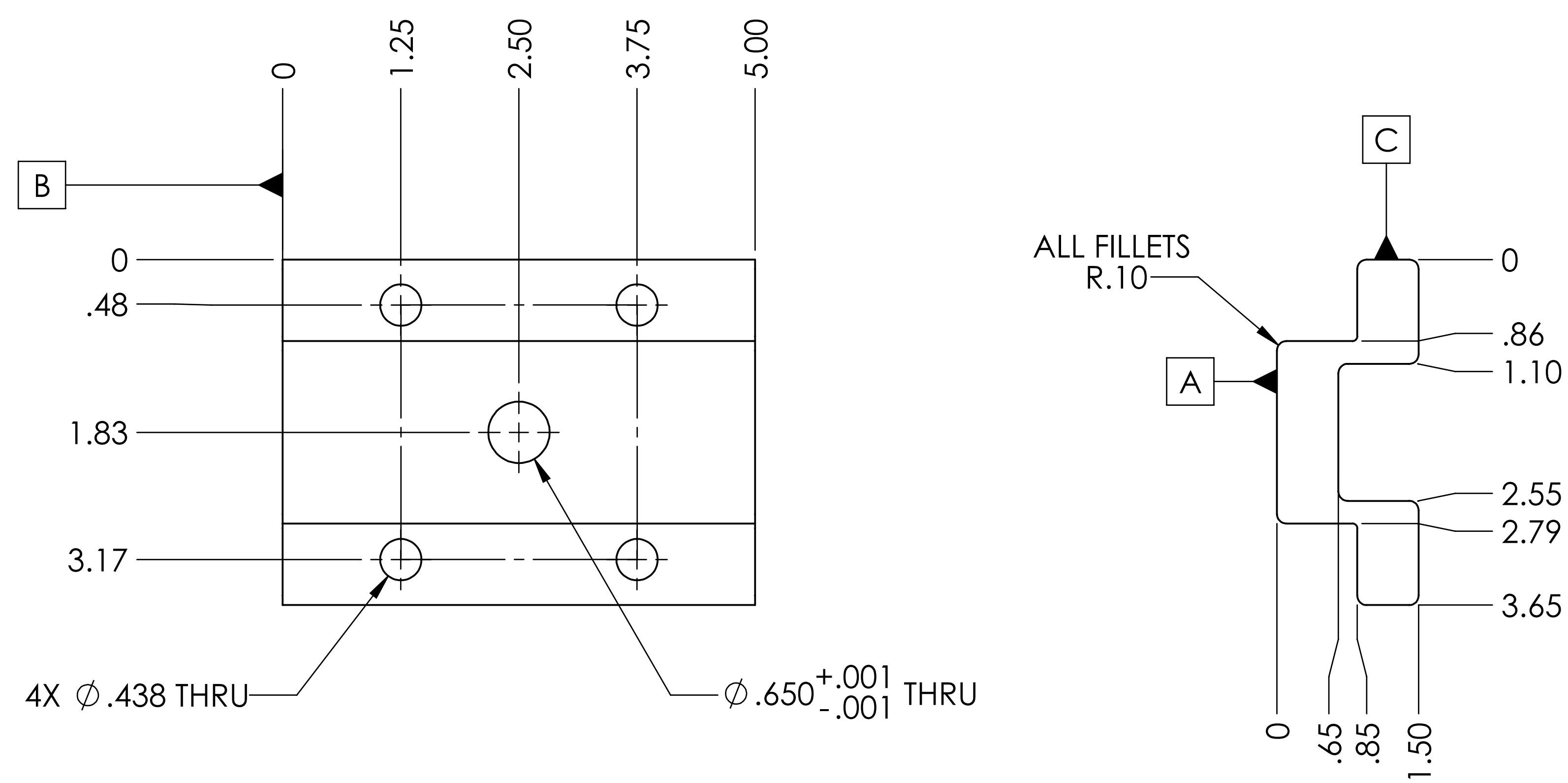
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. TAPER BEARING MOUNT | REV A

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



MAKE FROM 2 INCH
THICK PLATE

REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: DDY
CKD: AKERR

DWN:
CKD:

DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AL 6061-T6

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/06
CHECKED	AKERR	16/12/06
ENG APPR.		

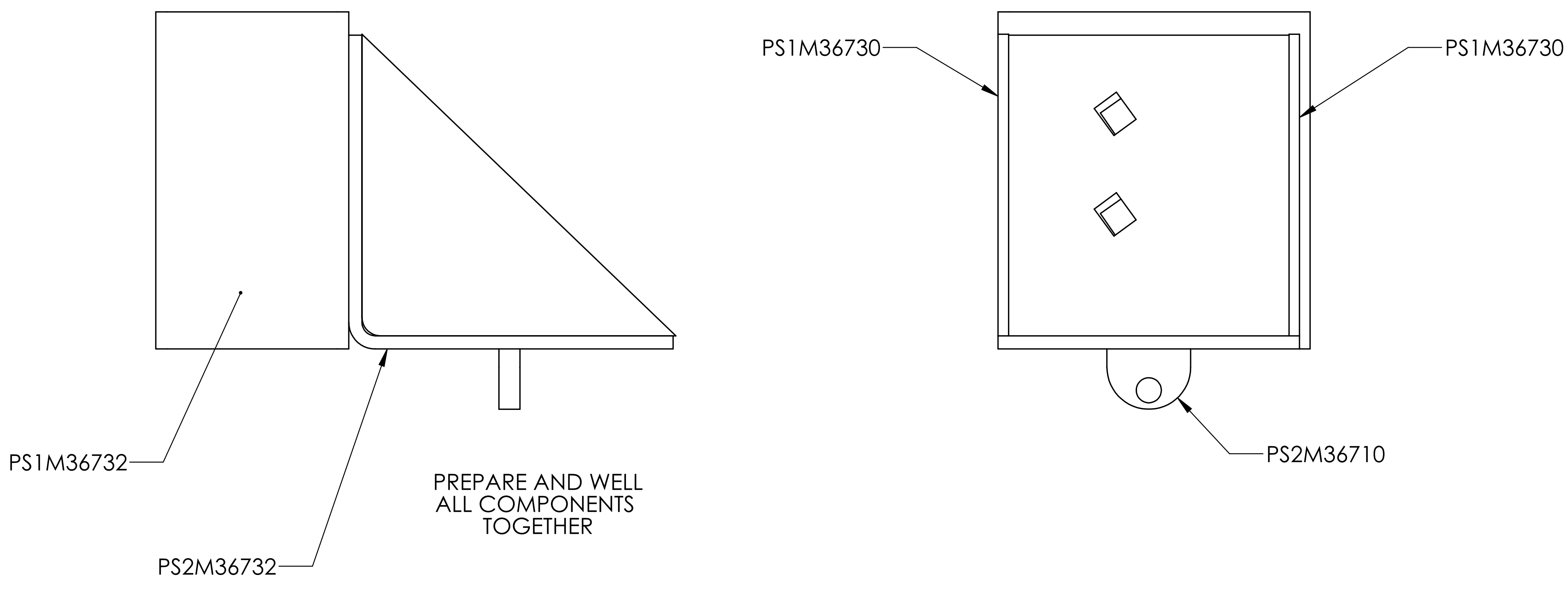
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. TAPER BEARING MOUNT | REV A

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



REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

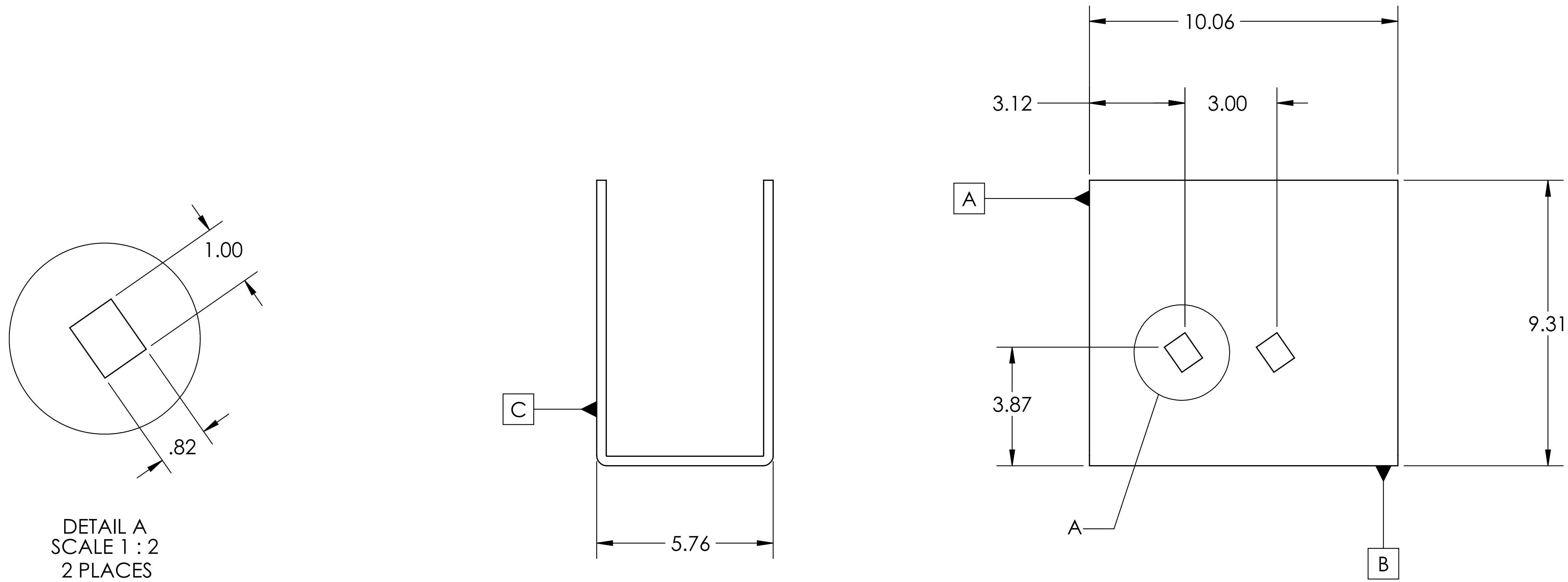
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
TIE ROD RACK BRACKET

SIZE B | DWG. NO. AS0M30042 | REV A

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

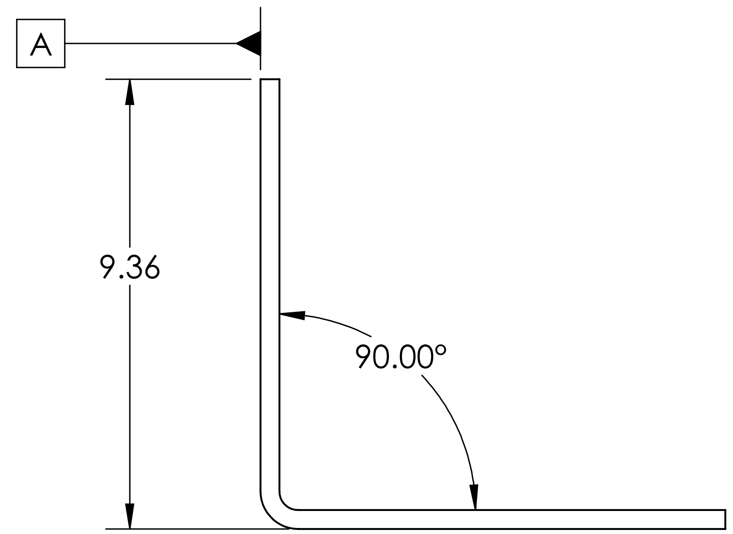


REV	DATE	DESCRIPTION
A		DRAFT

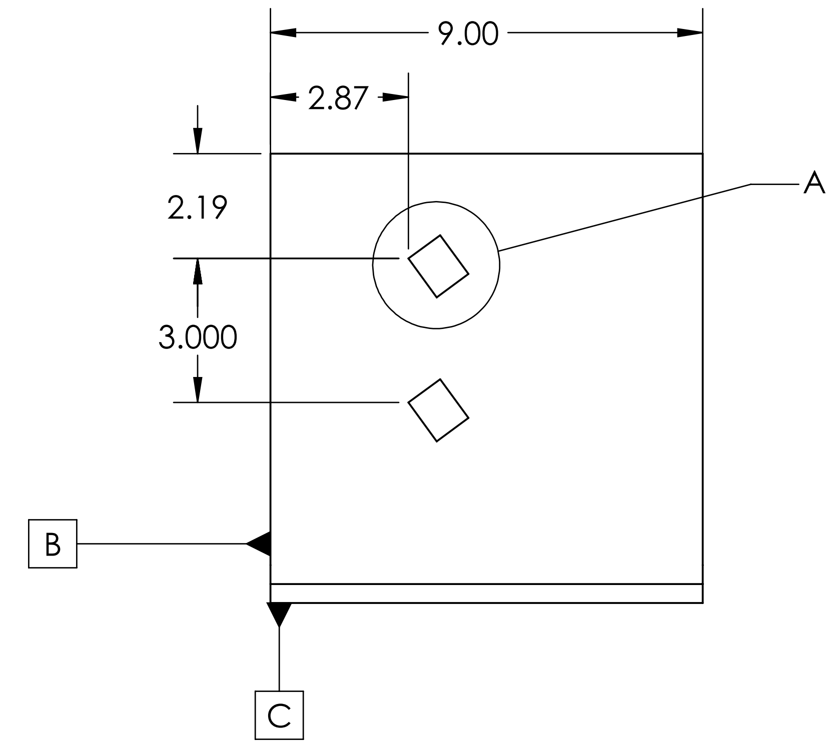
DWN: CKD:	UNLESS OTHERWISE SPECIFIED: TOLERANCES: ANGULAR: MACH ±1° BEND ±1° ONE PLACE DECIMAL ±.1 TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005 INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009		NAME / DATE
		DRAWN	
		CHECKED	
		ENG APPR.	
DWN: CKD:	MATERIAL AISI 1020	DIMENSIONAL UNIT: INCHES	
DWN: CKD:	FINISH 3.2 NO PLATING	DO NOT SCALE DRAWING	

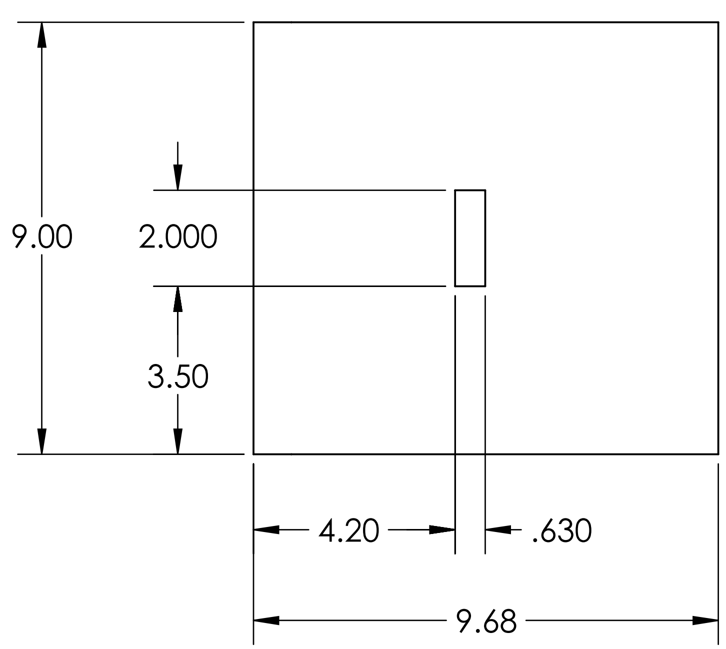
DESCRIPTION: RACK MOUNTING BRACKET		
SIZE B	DWG. NO. PS1M36732	REV A
SCALE: 1:4	WEIGHT: N/A	SHEET 1 OF 1

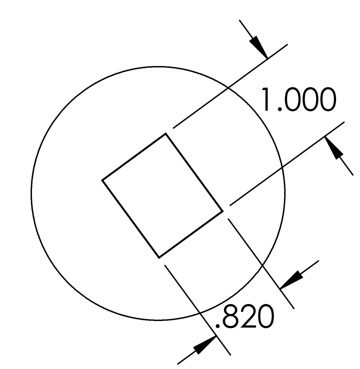
SOLIDWORKS Educational Product. For Instructional Use Only



MAKE FROM
28 GAUGE SHEET







DETAIL A
SCALE 1 : 2
2 PLACES

REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

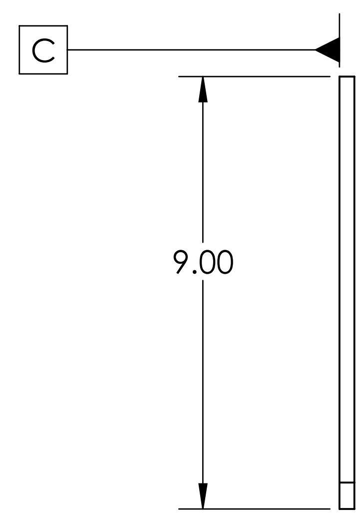
DIMENSIONAL
UNIT: INCHES

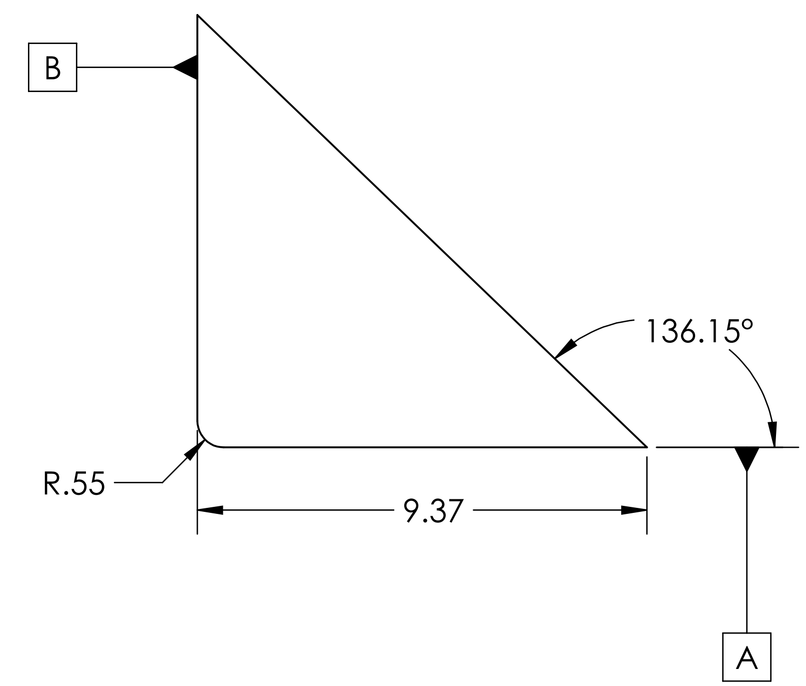
DO NOT SCALE DRAWING

DESCRIPTION:
RACK MOUNT TIE ROD BRACKET

SIZE **B** DWG. NO. PS2M36732 REV A

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





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REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2 √

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

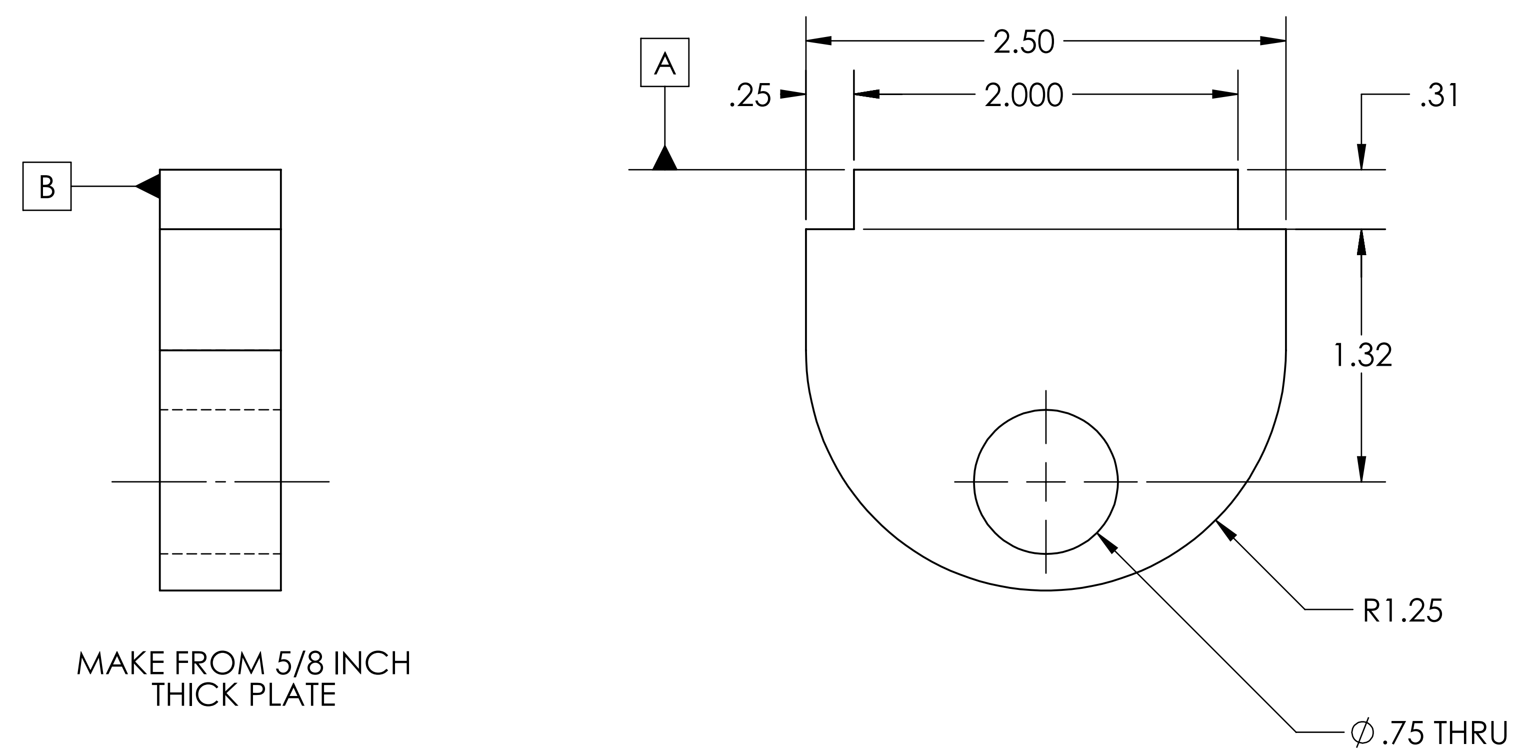
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
GUSSET

SIZE B | DWG. NO. PS1M36730 | REV A

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



REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

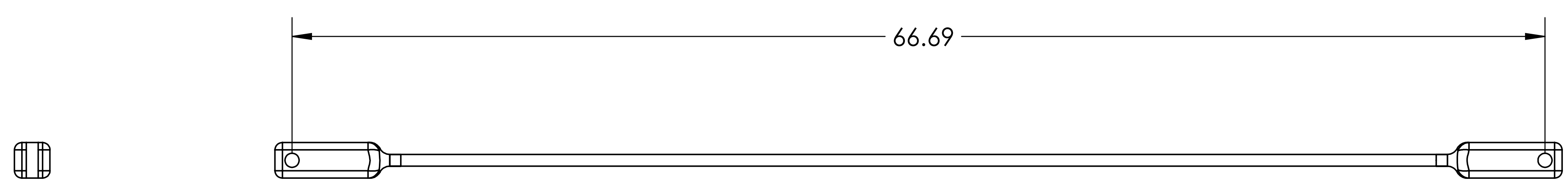
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
TIE ROD CLEVIS PLATE

SIZE B | DWG. NO. PS2M36710 | REV A

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



USED TO ILLUSTRATE
NOMINAL LENGTH

REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

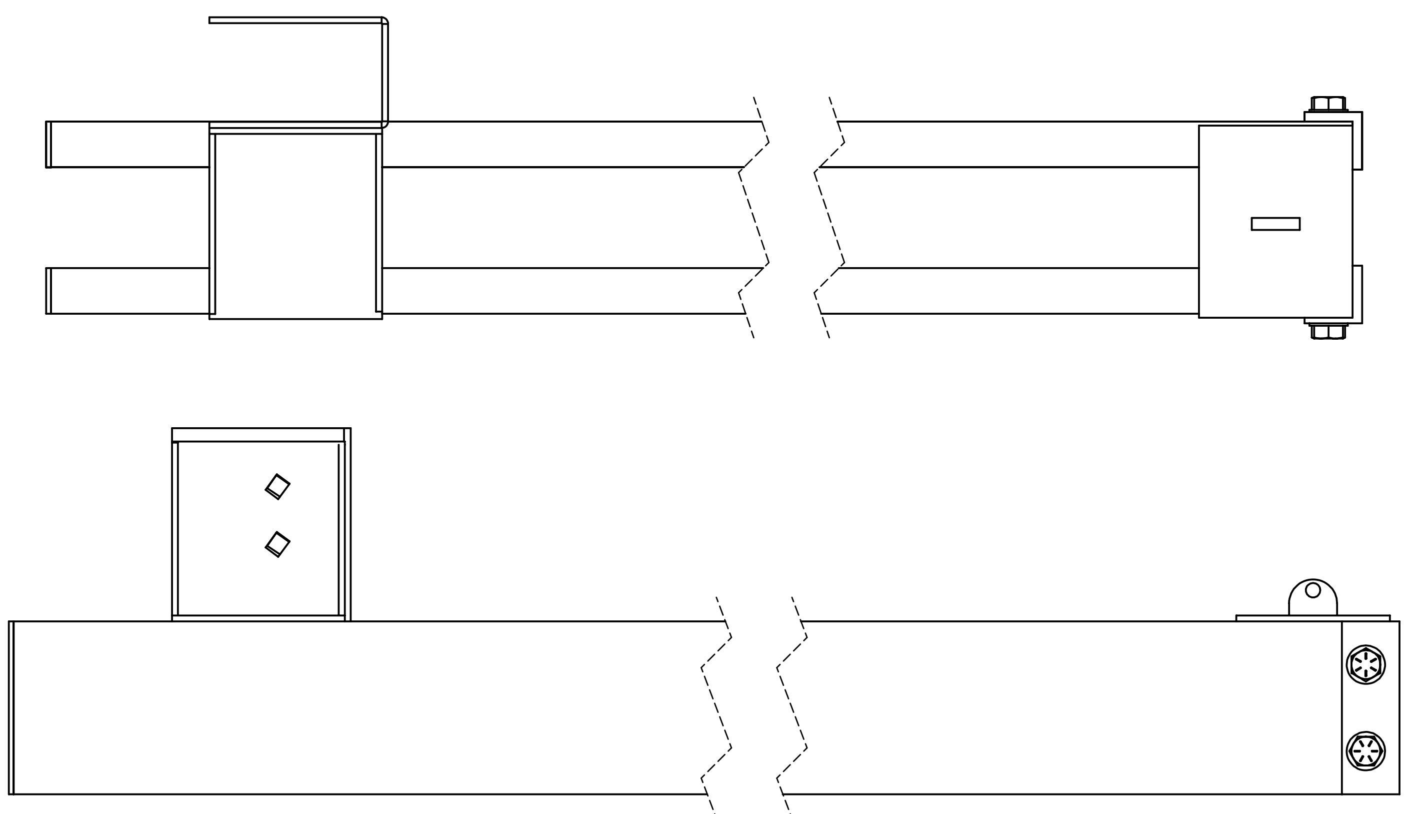
DO NOT SCALE DRAWING

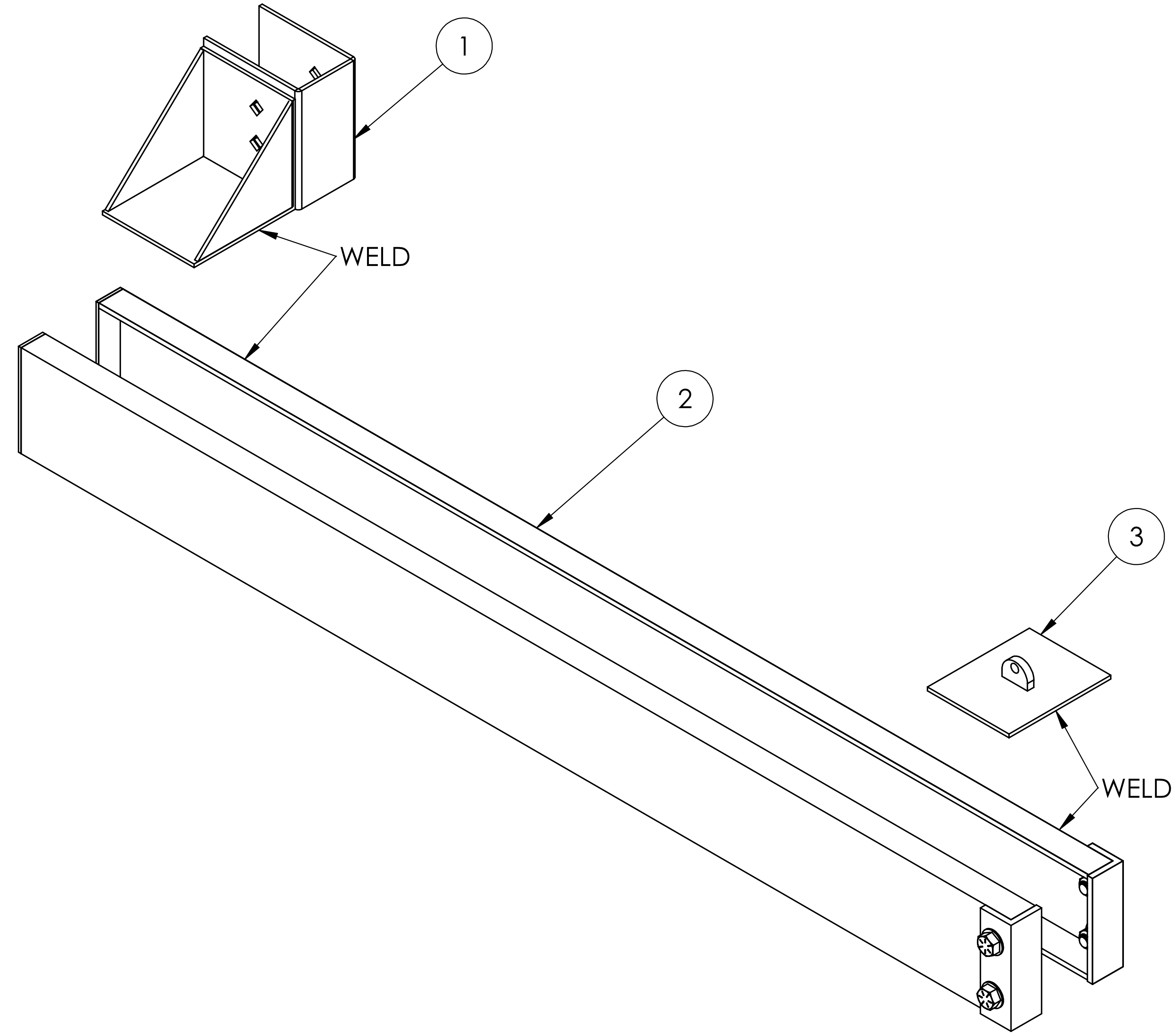
DESCRIPTION:

SIZE B DWG. NO. TIE ROD REV A

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

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	BRACEKT TO RACK		1
2	AS1M00540	C CHANNEL BRACKET	1
3	TIE ROD MOUNTING BRACKET		1





REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: YLEE CKD:
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
SEE BOM

FINISH N/A

	NAME	DATE
DRAWN	YLEE	16/12/05
CHECKED		
ENG APPR.		

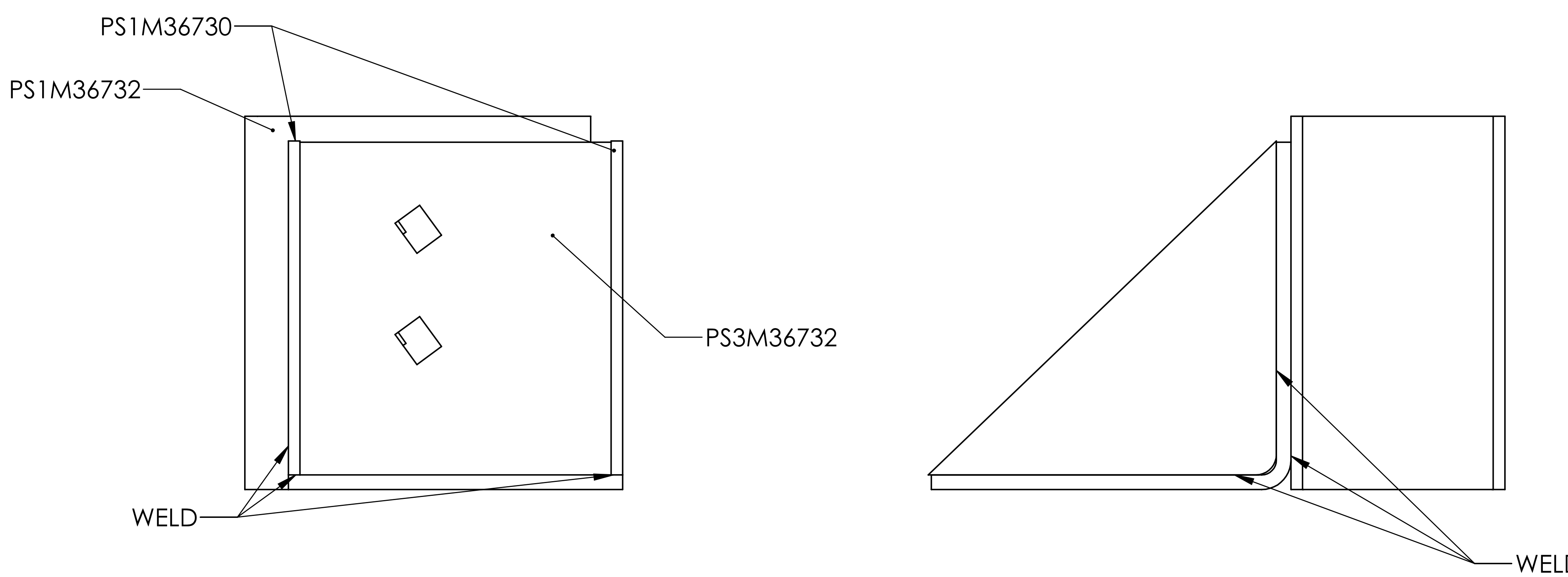
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
C CHANNEL AND BRACKETS ASSEMBLY

SIZE B | DWG. NO. AS2M30041 | REV A

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



REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
N/A

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. BRACKET TO RACK | REV A

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

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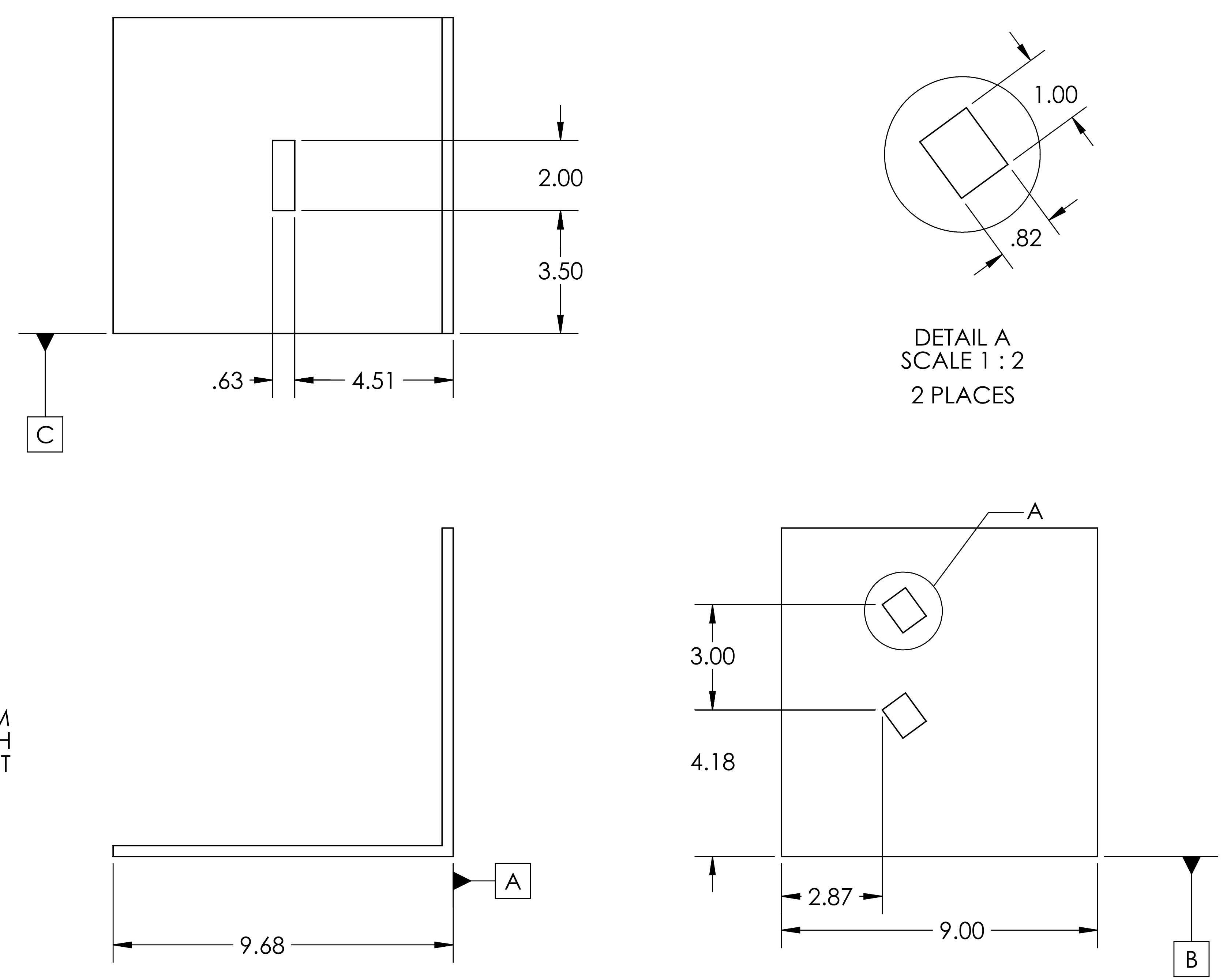


REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN: DDY
CKD: AKERR

DWN:
CKD:

DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

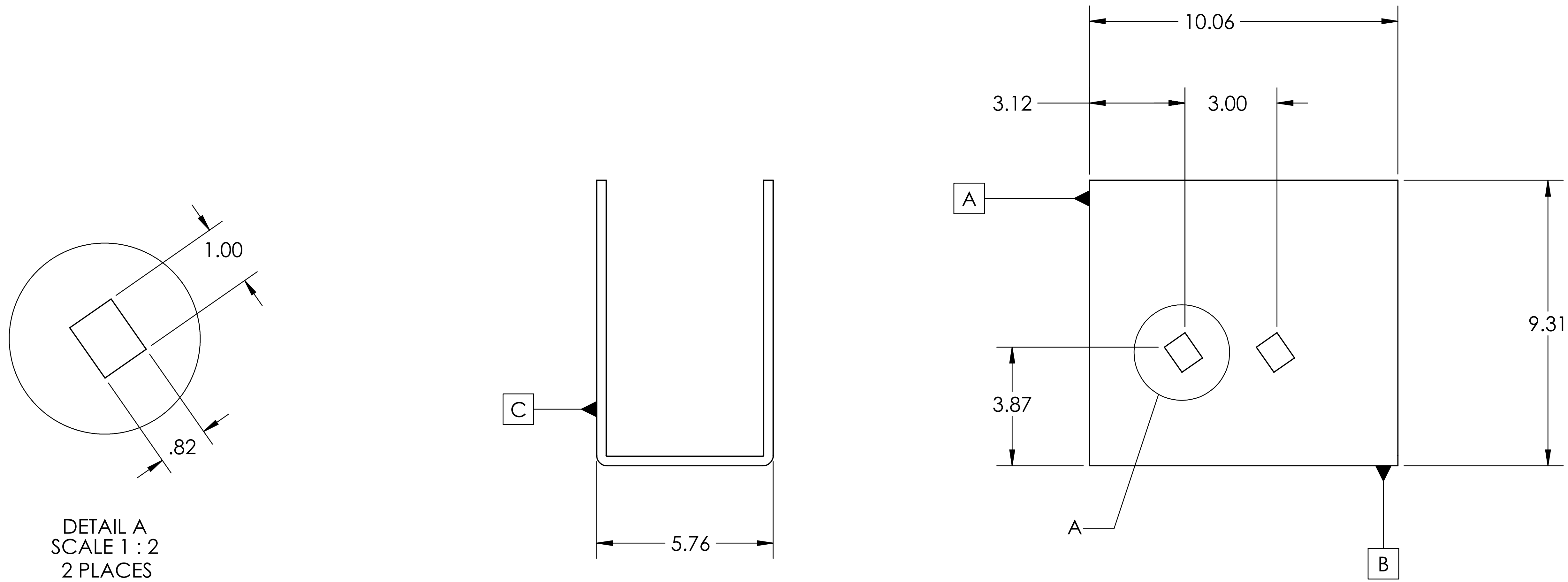
DESCRIPTION:
LEFT SIDE TIE ROD L BRACKET

SIZE B

DWG. NO.
L SUPPORT OTHER SIDE SPAR

REV A

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



DETAIL A
SCALE 1 : 2
2 PLACES

REV	DATE	DESCRIPTION
A		DRAFT

DWN:
CKD:
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN		
CHECKED		
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
RACK MOUNTING BRACKET

SIZE **B** DWG. NO. PS1M36732 REV A

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

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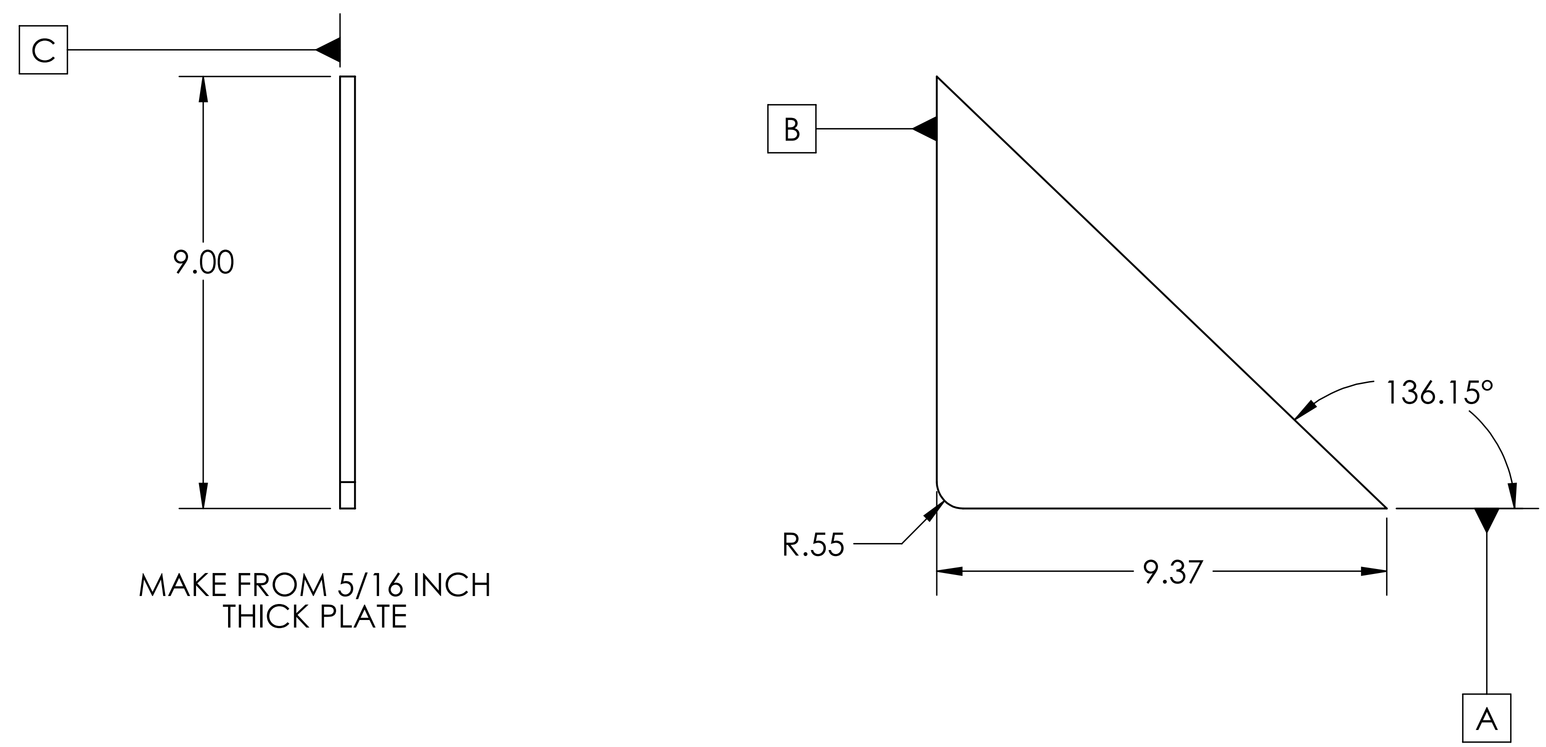

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REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

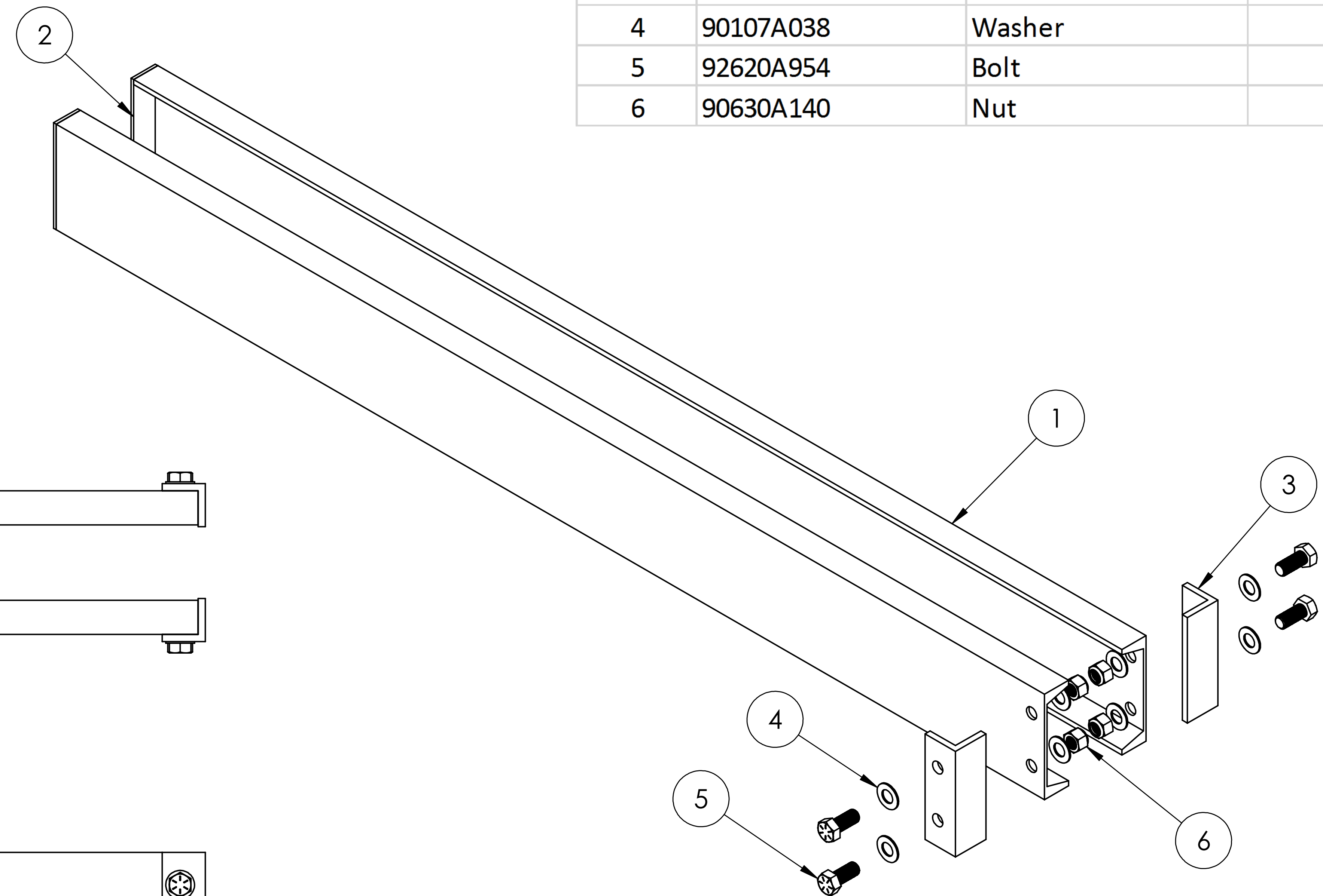
DO NOT SCALE DRAWING

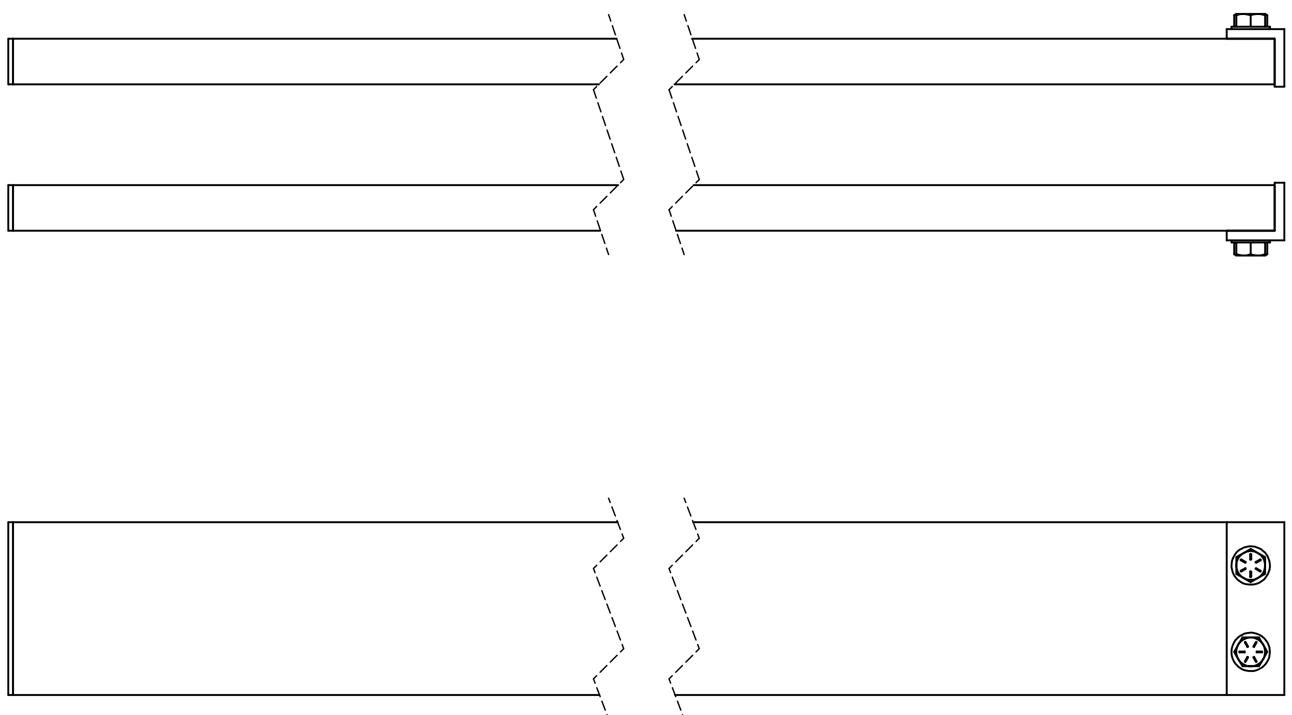
DESCRIPTION:
GUSSET

SIZE	DWG. NO.	REV
B	PS1M36730	A

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

ITEM NO.	PART NUMBER	DESCRIPTION	QTY
1	PS1M23630	C CHANNEL	2
2	PS1M227430	.25 IN THICK	2
3	PS3M37130	L3X3X.5	2
4	90107A038	Washer	8
5	92620A954	Bolt	4
6	90630A140	Nut	4





REV	DATE	DESCRIPTION	
A	16/12/05	DRAFT	DWN: YLEE CKD:
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
SEE BOM

FINISH N/A

	NAME	DATE
DRAWN	YLEE	16/12/05
CHECKED		
ENG APPR.		

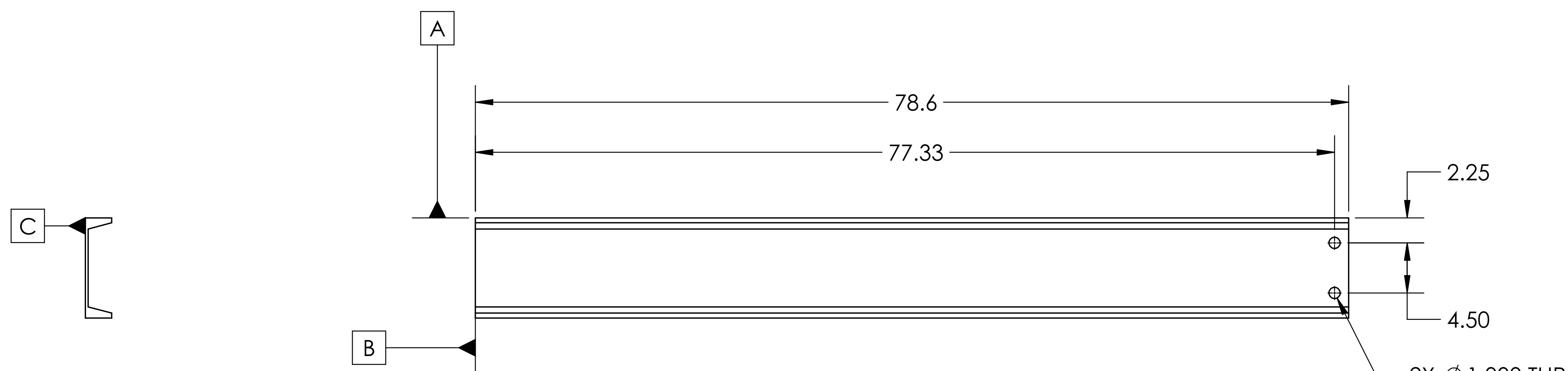
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
C-channel Bracket Assembly

SIZE B | DWG. NO. AS1M00540 | REV A

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



MAKE FROM C9 X 13.4
STRUCTURAL STEEL BEAMS

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REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36 STEEL

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

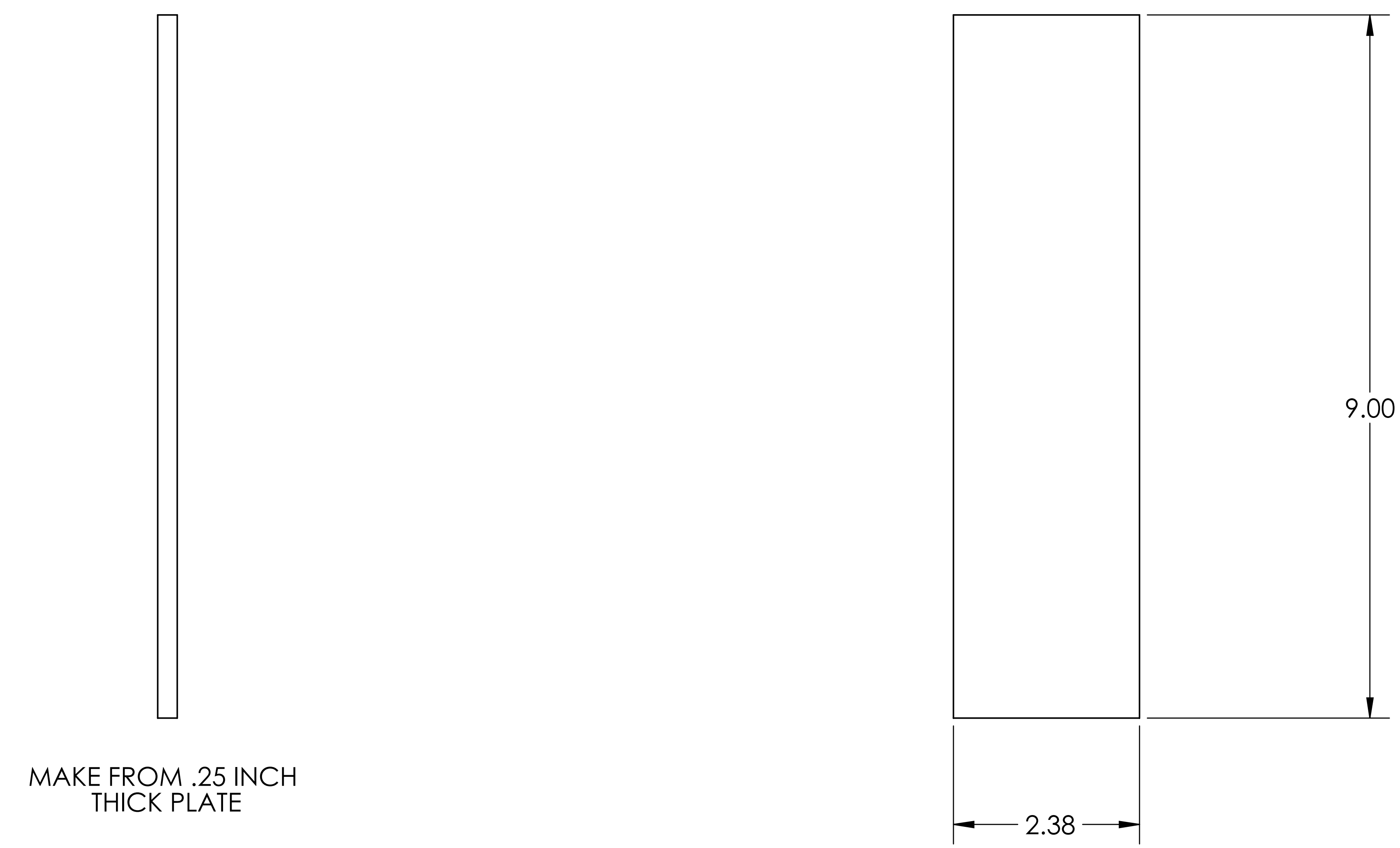
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
SLIDING BEAM RAIL

SIZE B | DWG. NO. PS1M23630 | REV A

SCALE: 1:12 | WEIGHT: N/A | SHEET 1 OF 1



REV	DATE	DESCRIPTION		
A	16/12/04	DRAFT	DWN: DDY CKD: AKERR	
			DWN: CKD:	
			DWN: CKD:	

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/04
CHECKED	AKERR	16/12/04
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

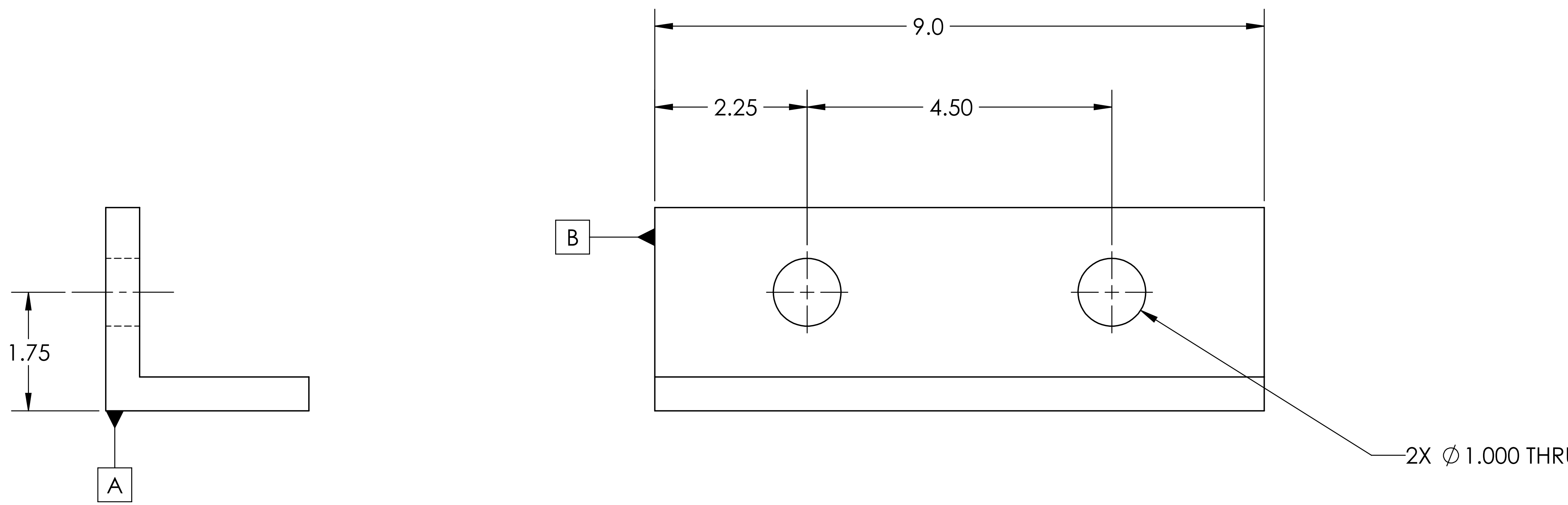
DO NOT SCALE DRAWING

DESCRIPTION:
BACK PLATE

SIZE B | DWG. NO. PS3M37130 | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



MAKE FROM L3X3X0.50

REV	DATE	DESCRIPTION
A	16/12/03	DRAFT

DWN: DDY
CKD: AKERR
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36 STEEL

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

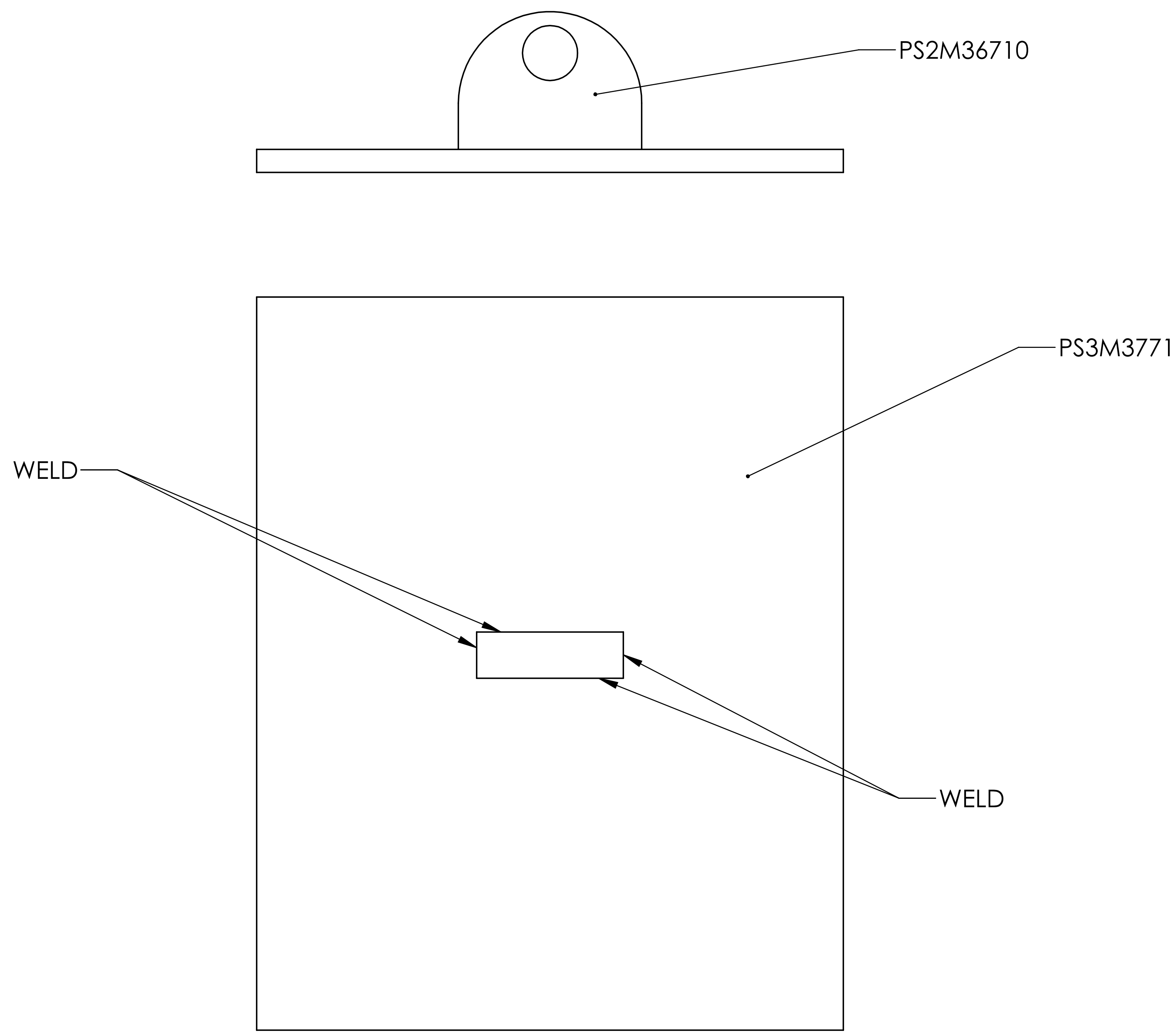
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

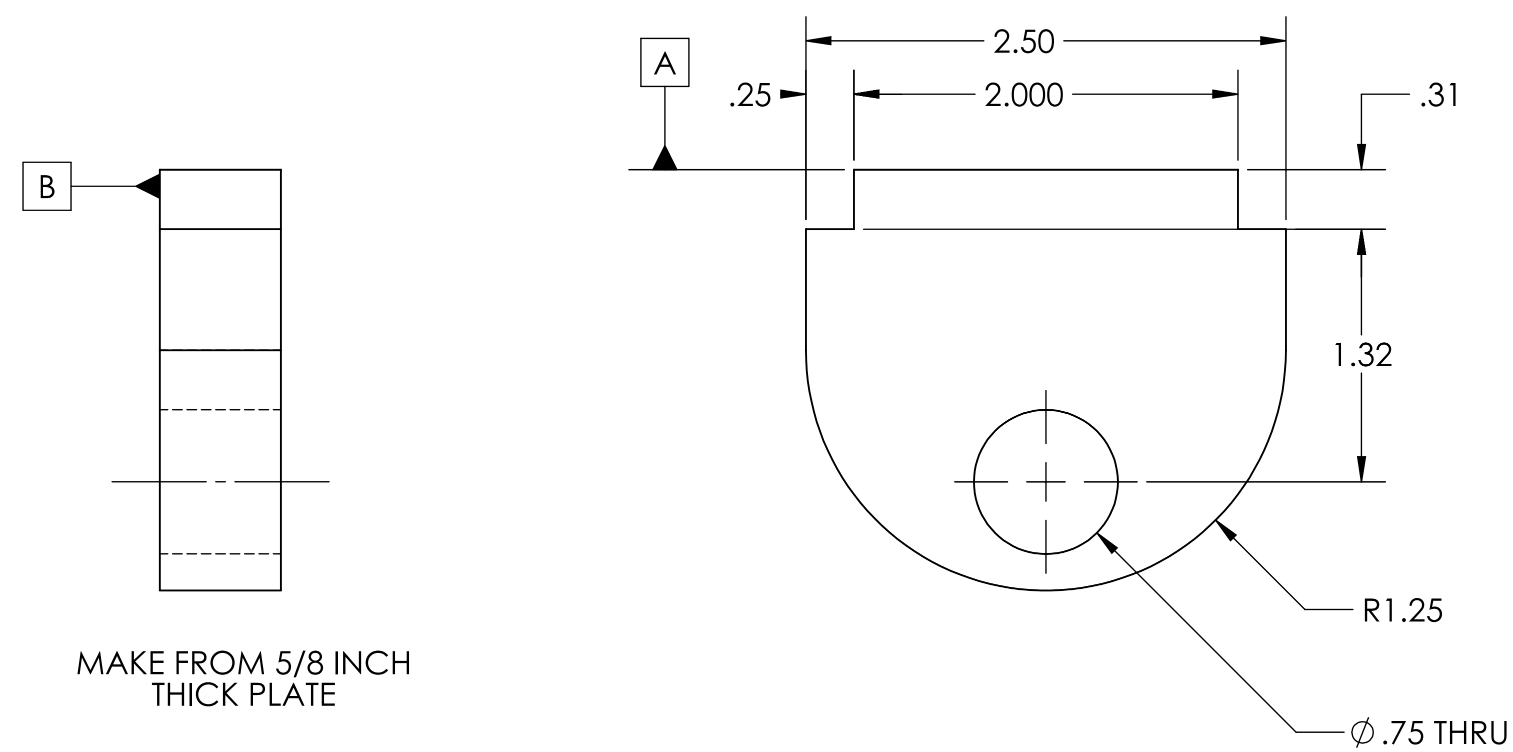
DESCRIPTION:
SLIDING BEAM STOP

SIZE B | DWG. NO. PS1M227430 | REV A

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



REV	DATE	DESCRIPTION		UNLESS OTHERWISE SPECIFIED:		NAME	DATE	DESCRIPTION:
A	16/12/05	DRAFT	DWN: DDY CKD: AKERR	TOLERANCES: ANGULAR: MACH ±1° BEND ±1° ONE PLACE DECIMAL ±.1 TWO PLACE DECIMAL ±.01 THREE PLACE DECIMAL ±.005 INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009	DRAWN	DDY	16/12/05	
					CHECKED	AKERR	16/12/05	
					ENG APPR.			
			DWN: CKD:	MATERIAL N/A	DIMENSIONAL UNIT: INCHES			SIZE B DWG. NO. TIE ROD MOUNTING BRACKET REV A
			DWN: CKD:	FINISH 3.2 NO PLATING	DO NOT SCALE DRAWING			SCALE: 1:2 WEIGHT: N/A SHEET 1 OF 1



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REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

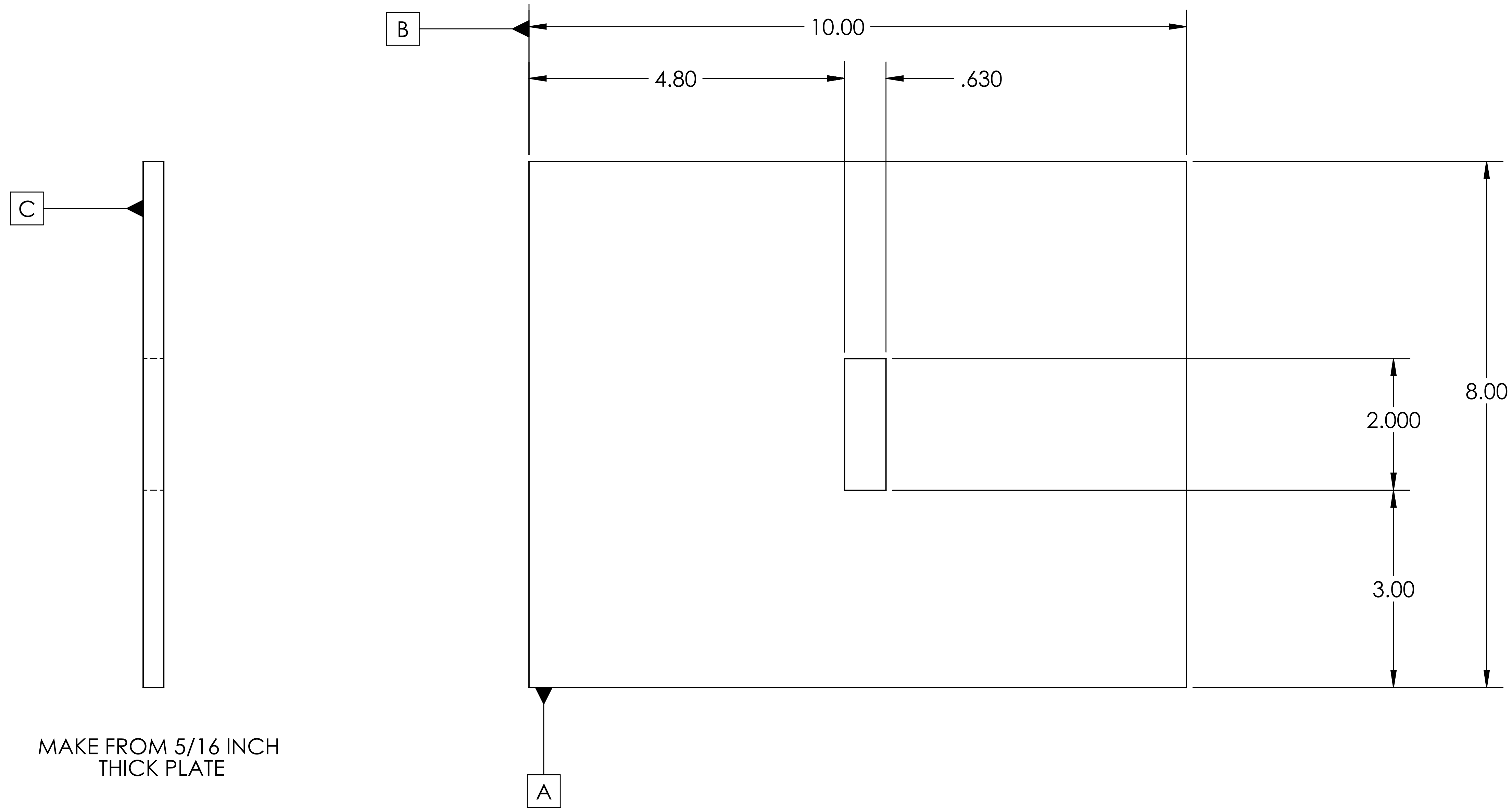
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
TIE ROD CLEVIS PLATE

SIZE	DWG. NO.	REV
B	PS2M36710	A

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



REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

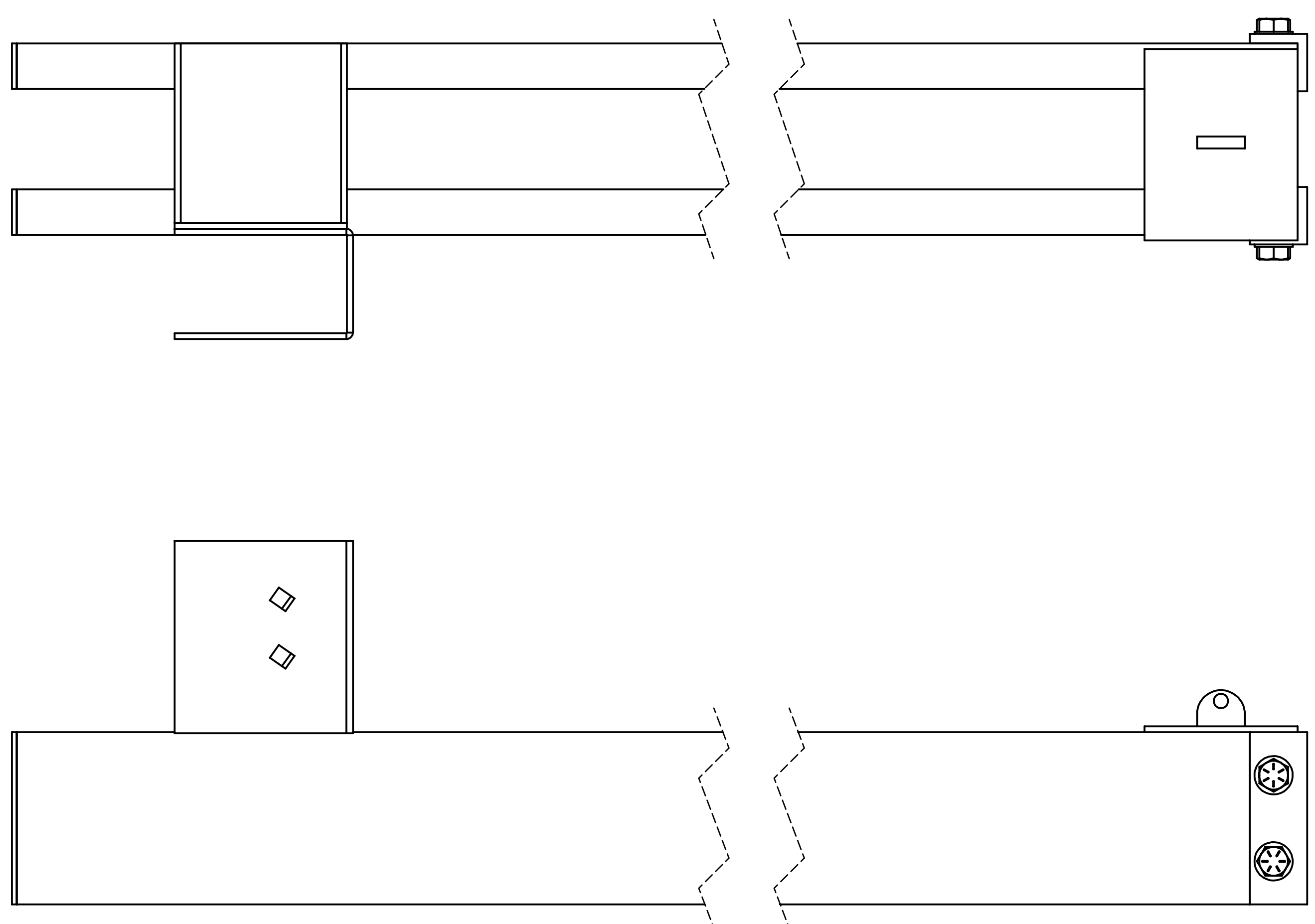
DESCRIPTION:
TIE ROD MOUNTING PLATE

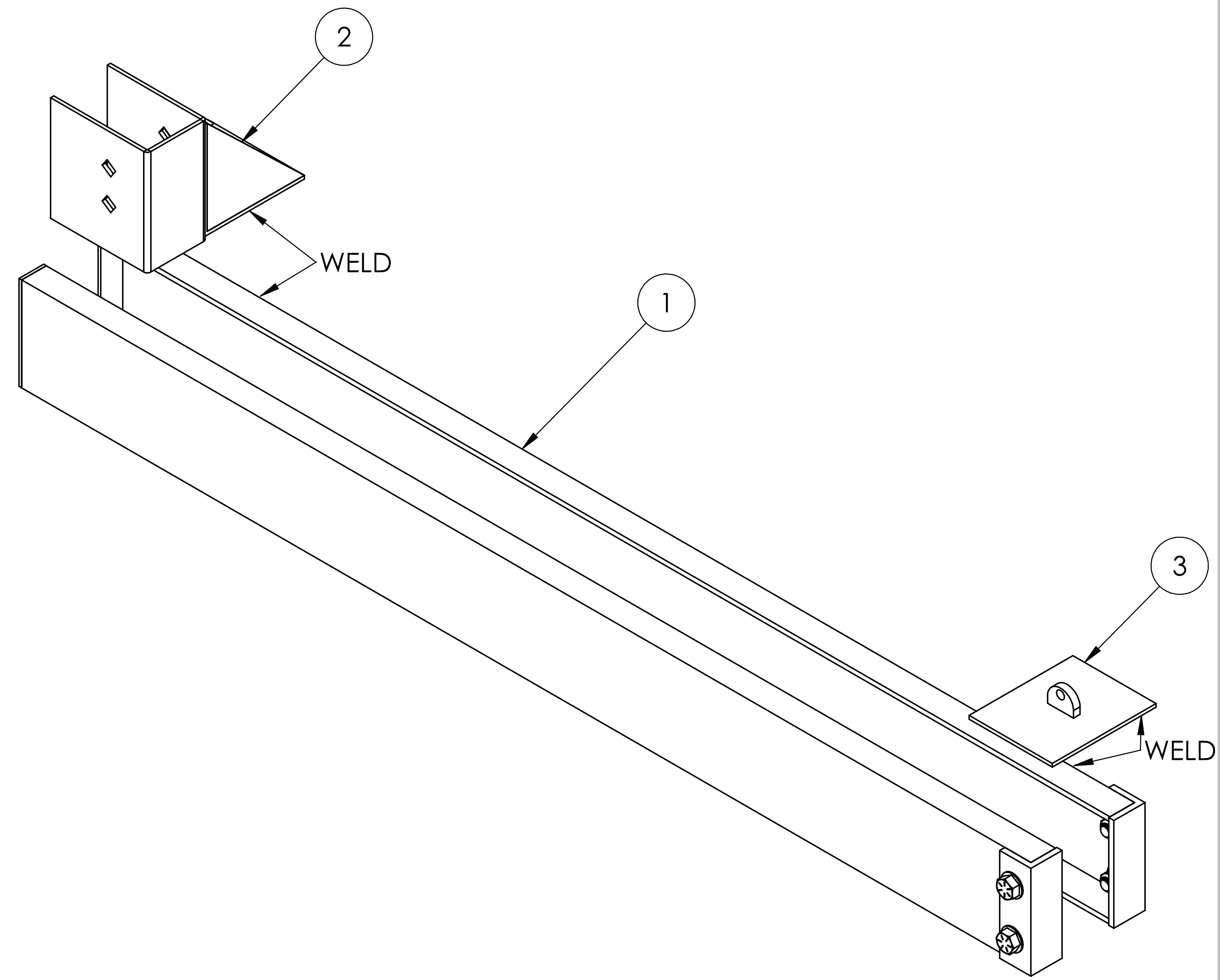
SIZE **B** DWG. NO. PS3M37710 REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	AS1M00540	C CHANNEL BRACKET	1
2	BRACKET TO RACK		1
3	TIE ROD MOUNTING BRACKET		1





REV	DATE	DESCRIPTION
A	16/12/06	DRAFT

DWN: YLEE
CKD:
DWN:
CKD:
DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ADD MATERIAL

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	YLEE	16/12/06
CHECKED		
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

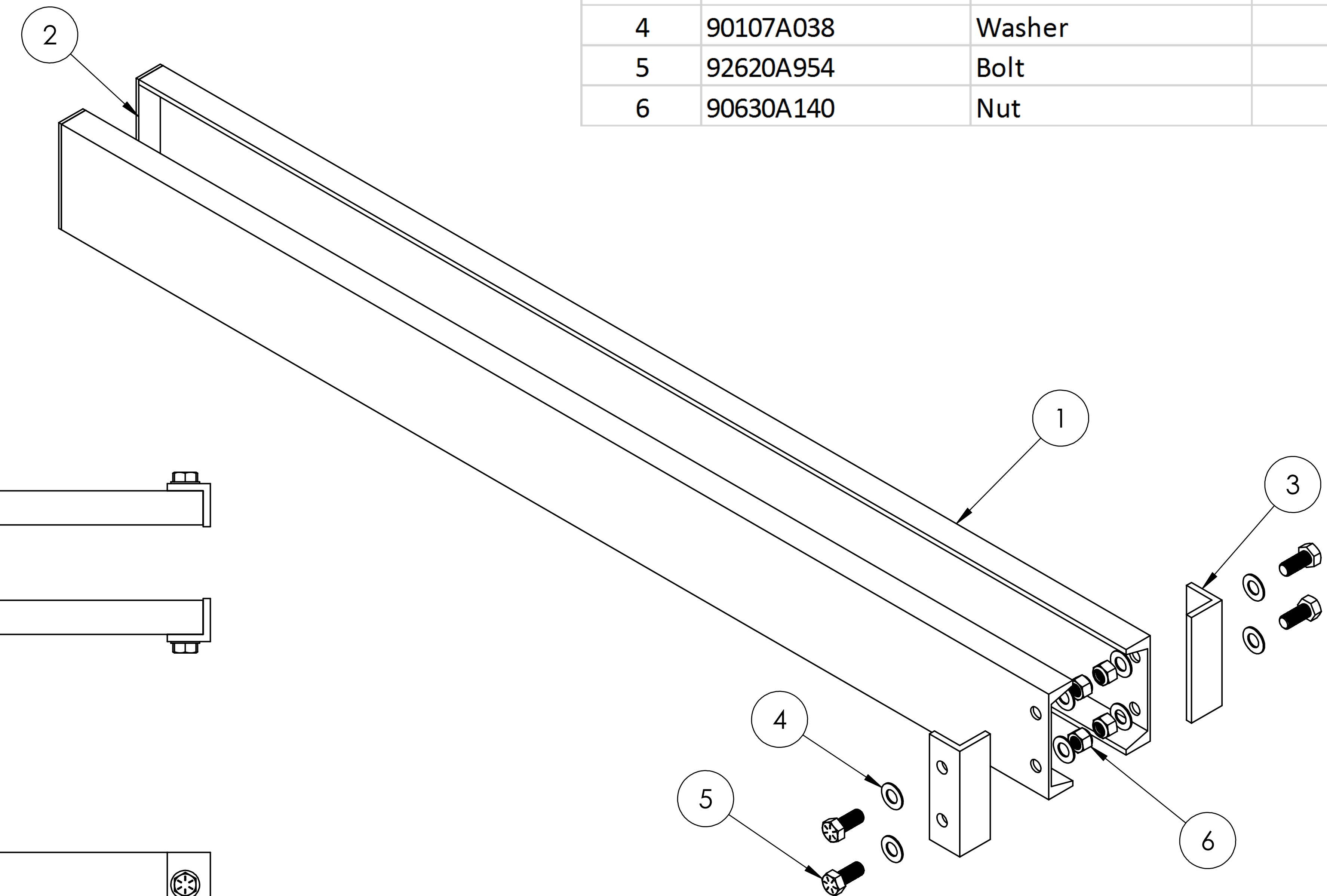
DO NOT SCALE DRAWING

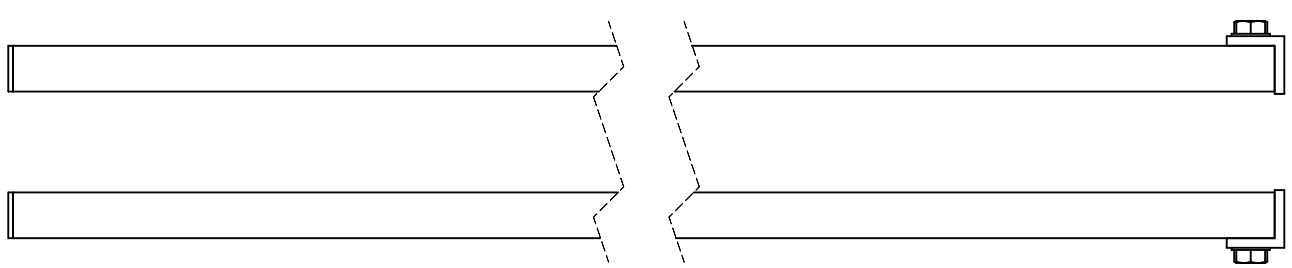
DESCRIPTION:
C CHANNEL AND BRACKETS ASSEMBLY

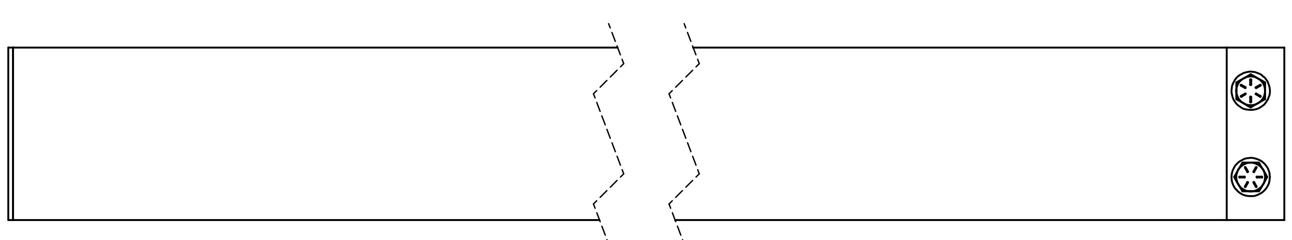
SIZE B DWG. NO. AS2M30042 REV A

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

ITEM NO.	PART NUMBER	DESCRIPTION	QTY
1	PS1M23630	C CHANNEL	2
2	PS1M227430	.25 IN THICK	2
3	PS3M37130	L3X3X.5	2
4	90107A038	Washer	8
5	92620A954	Bolt	4
6	90630A140	Nut	4







REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN: YLEE
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
SEE BOM

FINISH N/A

	NAME	DATE
DRAWN	YLEE	16/12/05
CHECKED		
ENG APPR.		

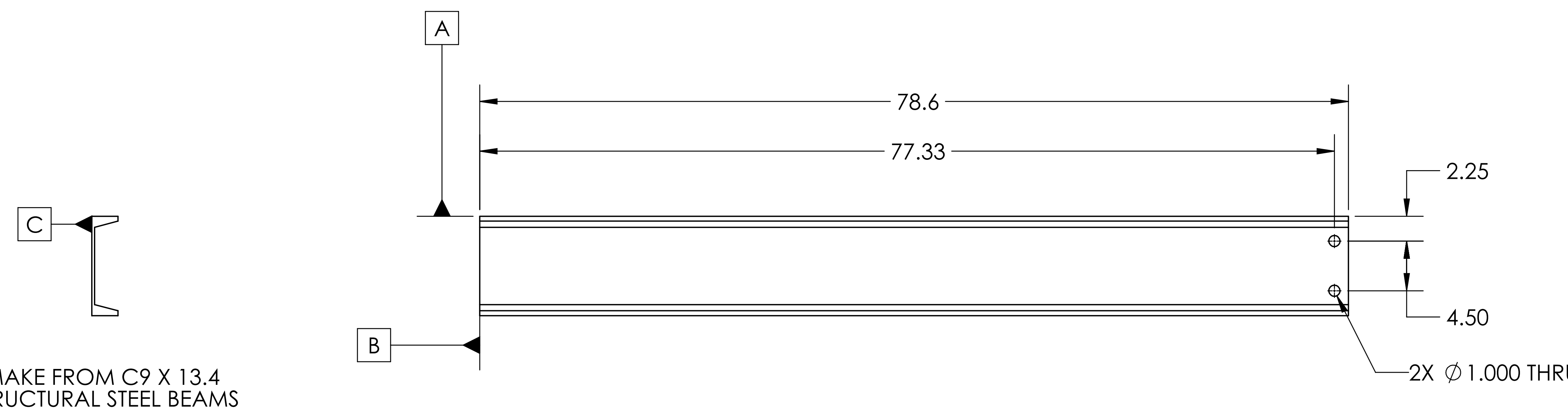
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
C-channel Bracket Assembly

SIZE B | DWG. NO. AS1M00540 | REV A

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



REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36 STEEL

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
SLIDING BEAM RAIL

SIZE **B** DWG. NO. PS1M23630 REV A

SCALE: 1:12 WEIGHT: N/A SHEET 1 OF 1

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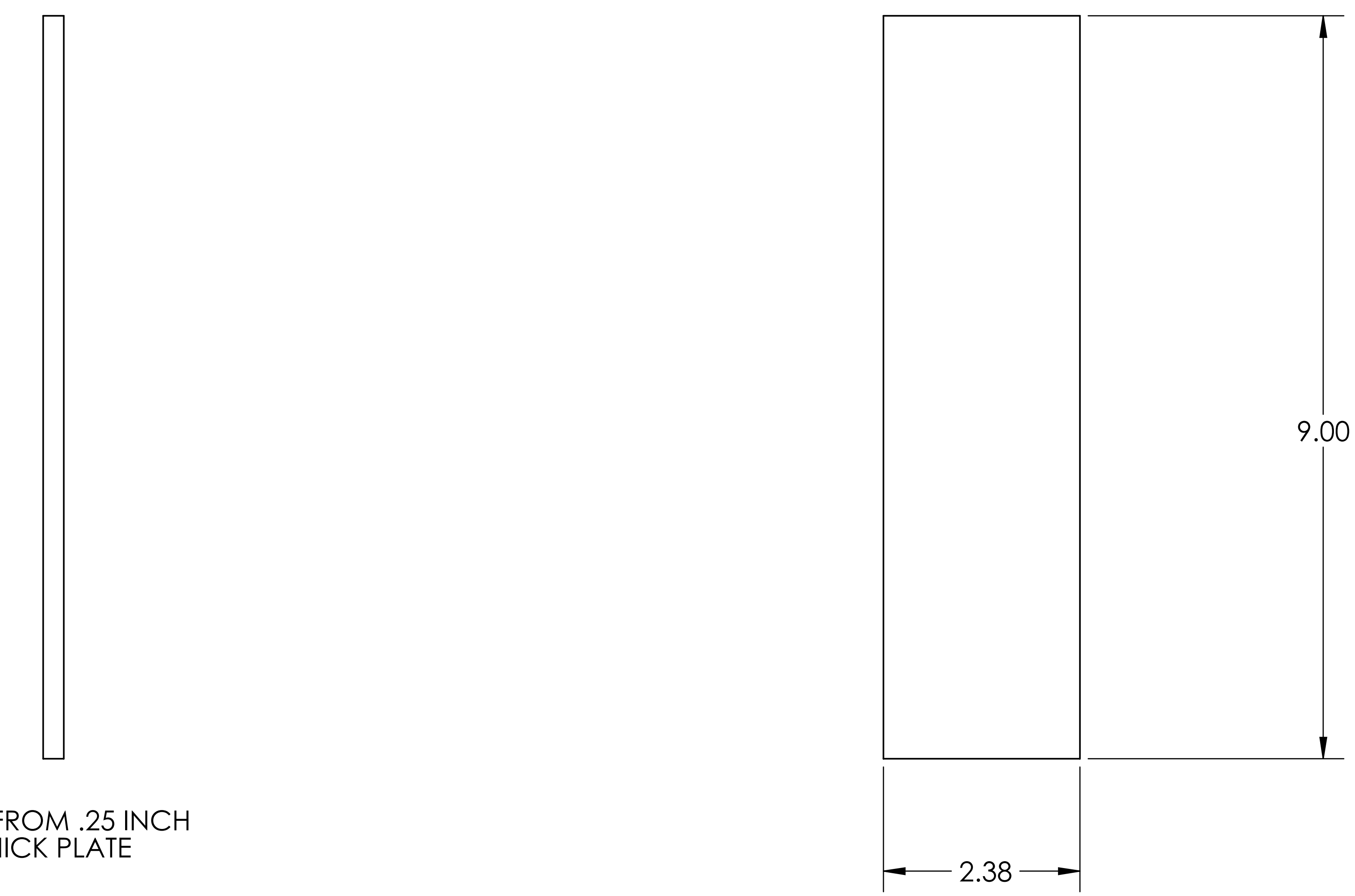


MAKE FROM .25 INCH
THICK PLATE

SOLIDWORKS Educational Product. For Instructional Use Only

REV	DATE	DESCRIPTION	
A	16/12/04	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/04
CHECKED	AKERR	16/12/04
ENG APPR.		

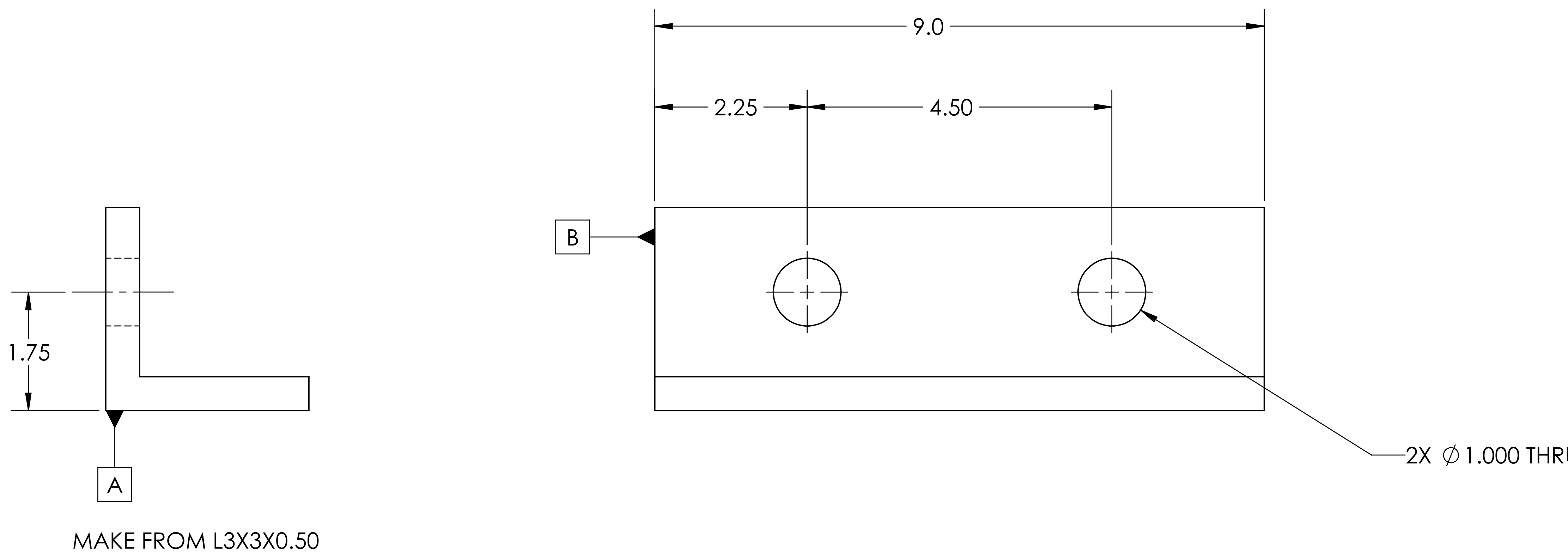
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
BACK PLATE

SIZE B | DWG. NO. PS3M37130 | REV A

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



REV	DATE	DESCRIPTION
A	16/12/03	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
ASTM A36 STEEL

FINISH 3.2 NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

DIMENSIONAL UNIT: INCHES

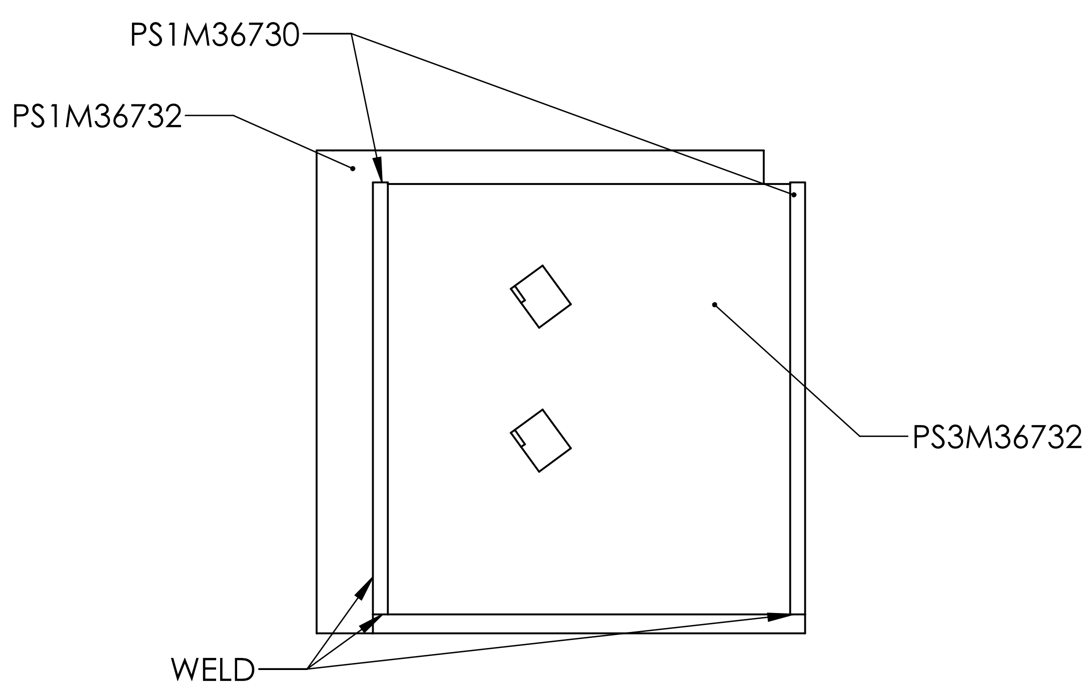
DO NOT SCALE DRAWING

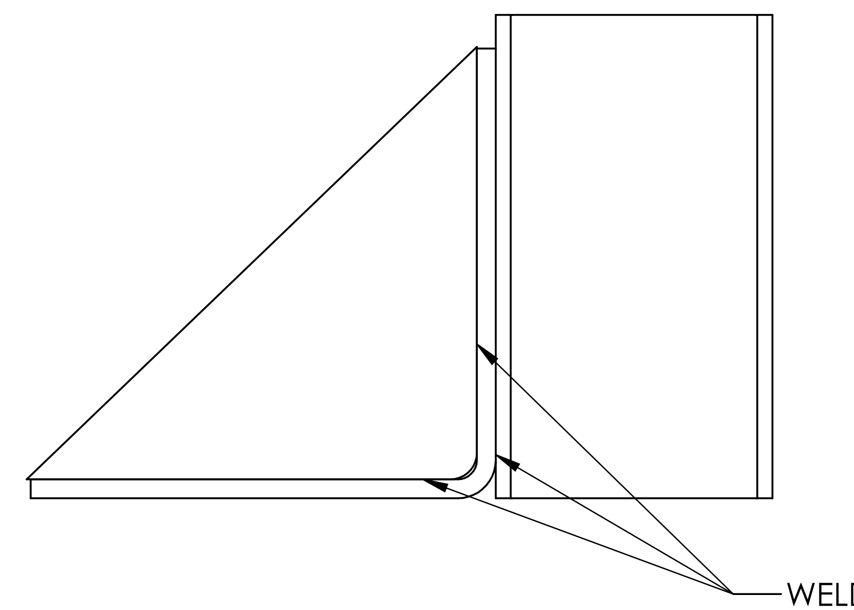
DESCRIPTION:
SLIDING BEAM STOP

SIZE B | DWG. NO. PS1M227430 | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only





REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL UNIT: INCHES

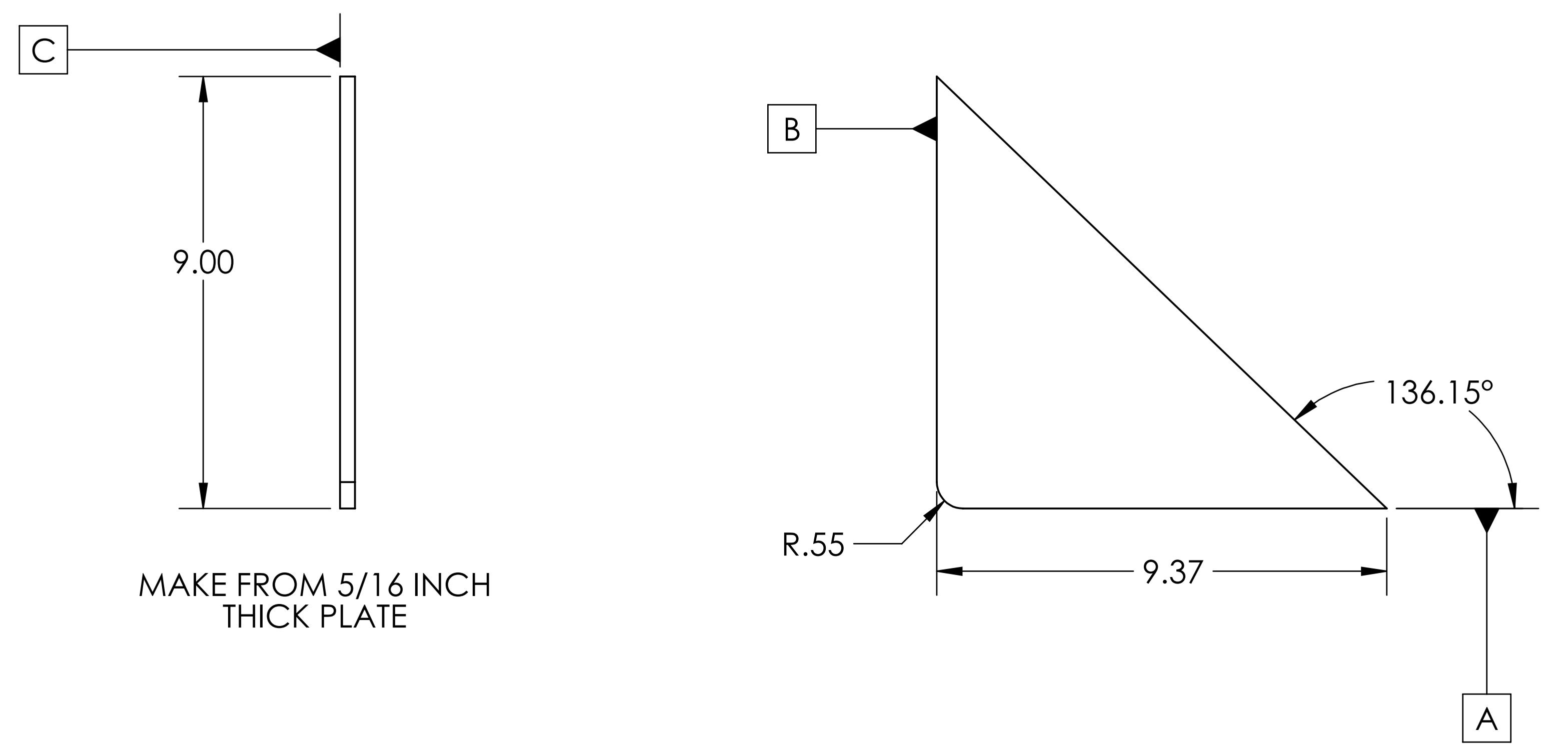
DO NOT SCALE DRAWING

DESCRIPTION:

SIZE B | DWG. NO. BRACKET TO RACK | REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION		
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR	
			DWN: CKD:	
			DWN: CKD:	

UNLESS OTHERWISE SPECIFIED:	
TOLERANCES: ANGULAR: MACH ± 1° BEND ± 1° ONE PLACE DECIMAL ± .1 TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005	
INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009	
MATERIAL	AISI 1020
FINISH 3.2	NO PLATING

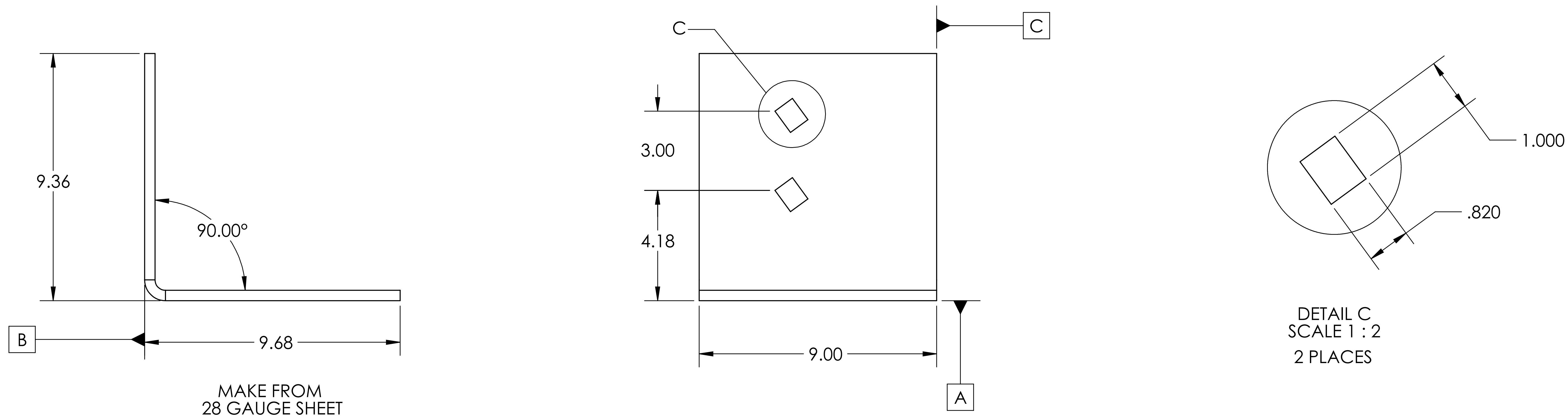
	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION: GUSSET

SIZE	DWG. NO.	REV
B	PS1M36730	A
SCALE: 1:4	WEIGHT: N/A	SHEET 1 OF 1



REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

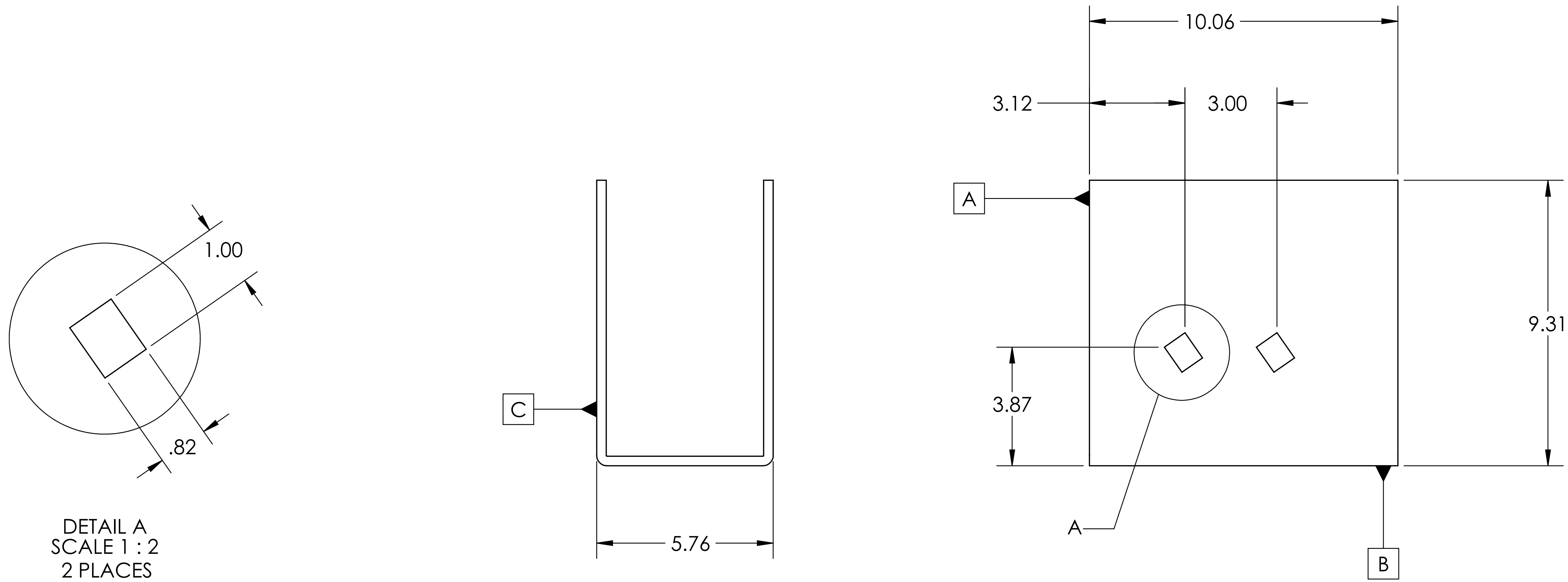
DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
SUPPORT STRUCTURE MOUNTING BRACKET

SIZE	DWG. NO.	REV
B	PS3M36732	A

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



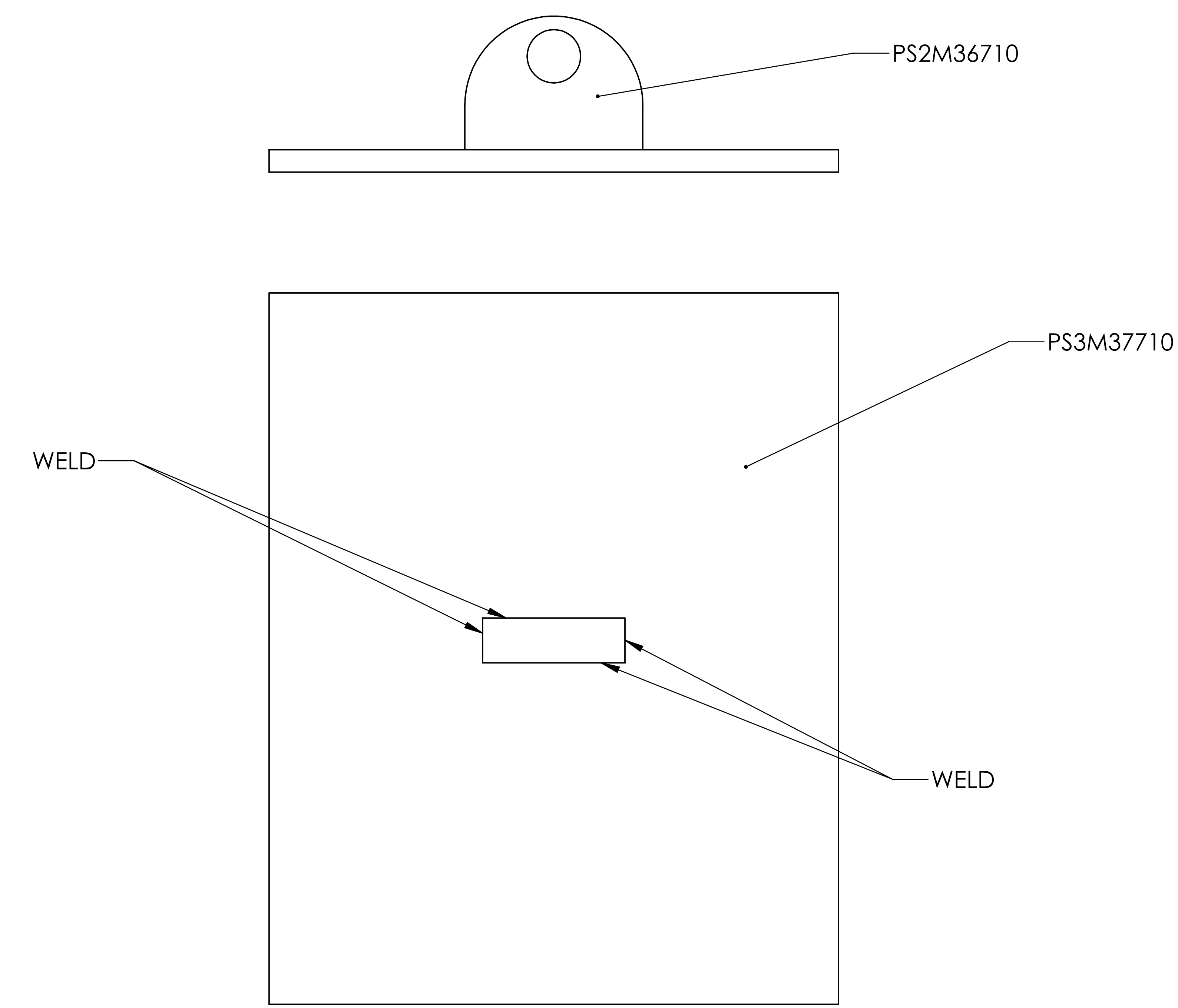
REV	DATE	DESCRIPTION
A		DRAFT

DWN: CKD:	UNLESS OTHERWISE SPECIFIED: TOLERANCES: ANGULAR: MACH ± 1° BEND ± 1° ONE PLACE DECIMAL ± .1 TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005 INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009		NAME	DATE
		DRAWN		
		CHECKED		
		ENG APPR.		
DWN: CKD:	MATERIAL AISI 1020	DIMENSIONAL UNIT: INCHES		
DWN: CKD:	FINISH 3.2 NO PLATING	DO NOT SCALE DRAWING		

DESCRIPTION:
RACK MOUNTING BRACKET

SIZE	DWG. NO.	REV
B	PS1M36732	A
SCALE: 1:4	WEIGHT: N/A	SHEET 1 OF 1

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION
A	16/12/05	DRAFT

DWN: DDY
CKD: AKERR

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL N/A

FINISH NO PLATING

	NAME	DATE
DRAWN	DDY	16/12/05
CHECKED	AKERR	16/12/05
ENG APPR.		

DIMENSIONAL UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:

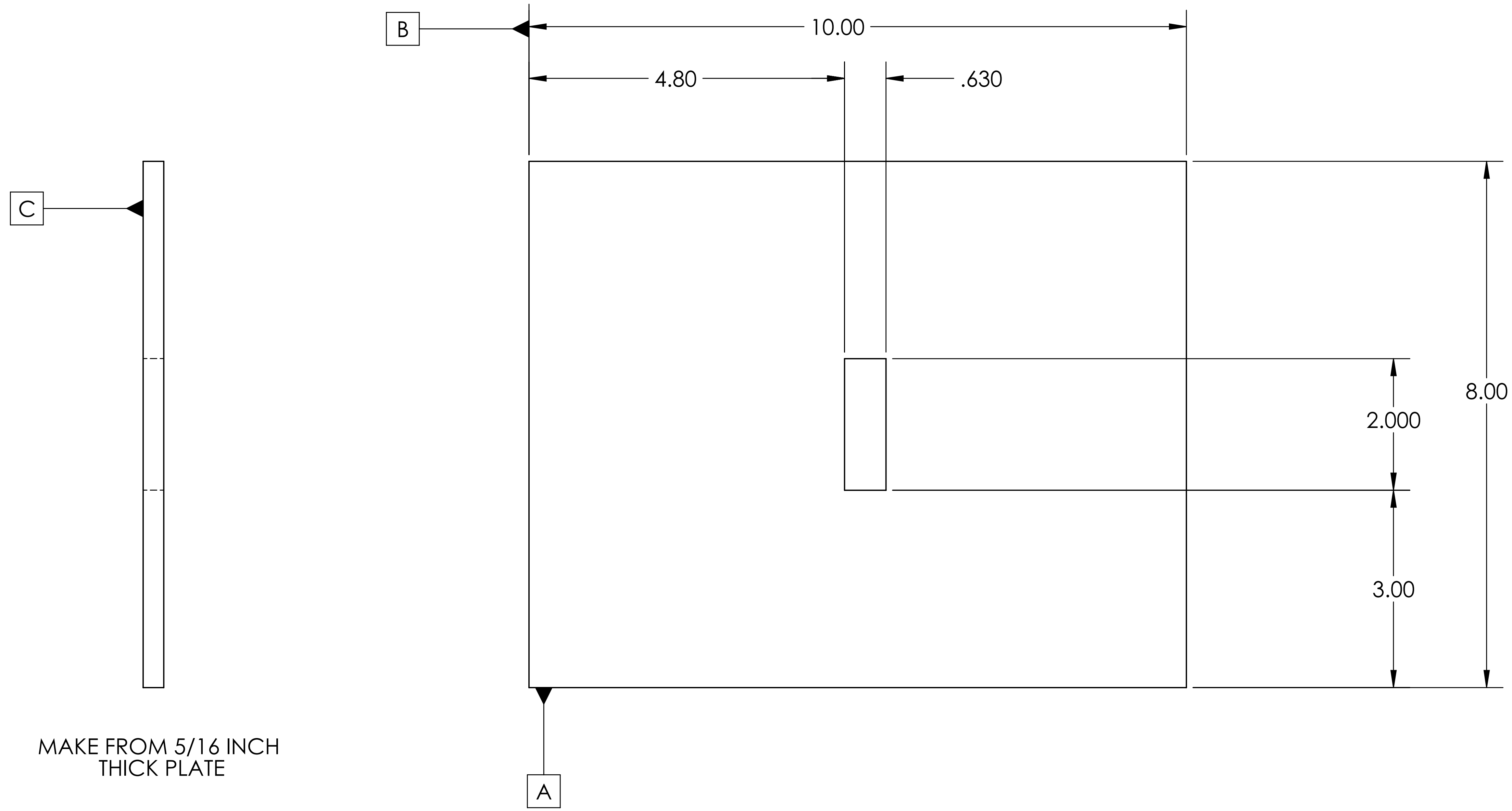
SIZE B

DWG. NO. TIE ROD MOUNTING BRACKET

REV A

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

SOLIDWORKS Educational Product. For Instructional Use Only



REV	DATE	DESCRIPTION	
A	16/12/03	DRAFT	DWN: DDY CKD: AKERR
			DWN: CKD:
			DWN: CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ± 1° BEND ± 1°
ONE PLACE DECIMAL ± .1
TWO PLACE DECIMAL ± .01
THREE PLACE DECIMAL ± .005

INTERPRET GEOMETRIC
TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

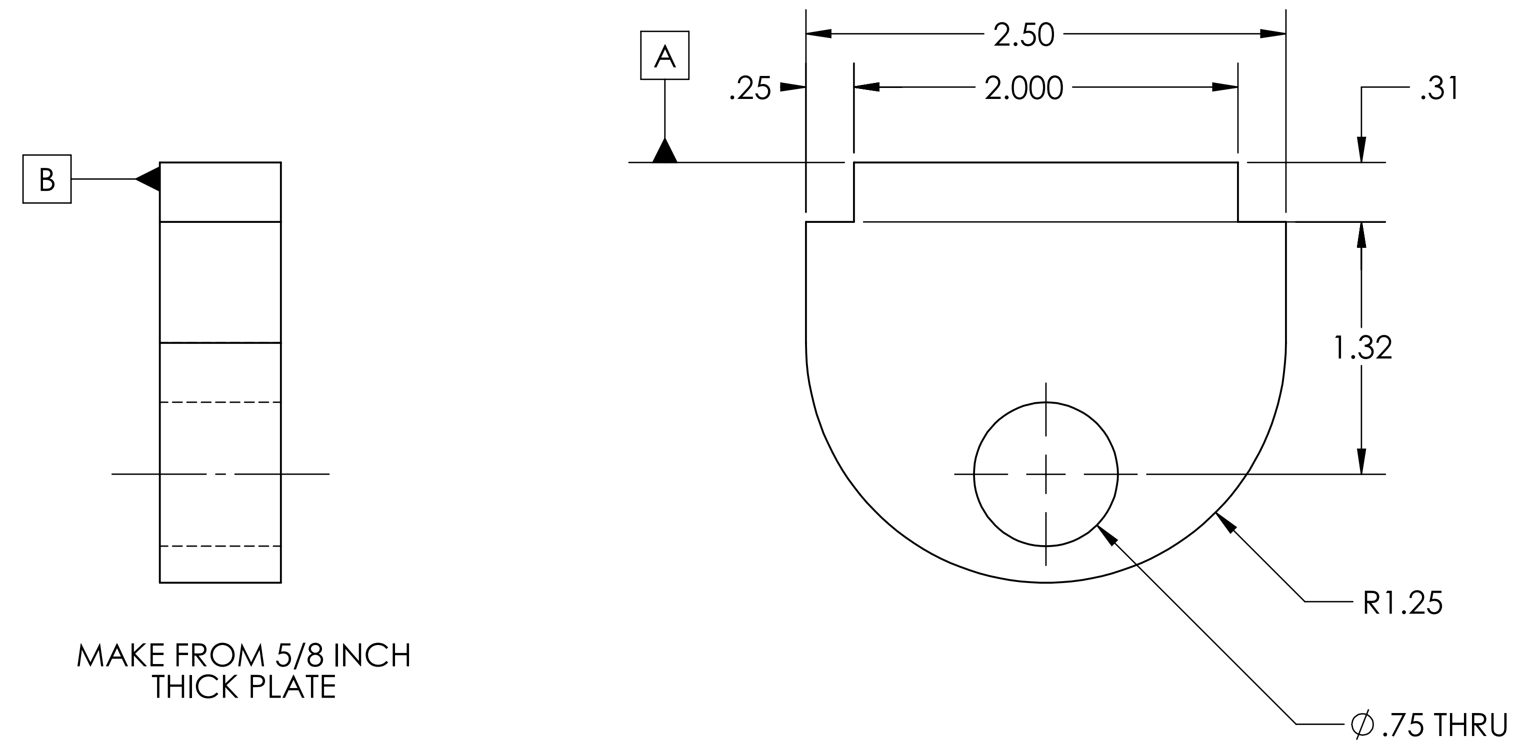
DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
TIE ROD MOUNTING PLATE

SIZE B | DWG. NO. PS3M37710 | REV A

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



REV	DATE	DESCRIPTION
A	16/12/03	DRAFT

DWN: DDY
CKD: AKERR

DWN:
CKD:

DWN:
CKD:

UNLESS OTHERWISE SPECIFIED:
TOLERANCES:
ANGULAR: MACH ±1° BEND ±1°
ONE PLACE DECIMAL ±.1
TWO PLACE DECIMAL ±.01
THREE PLACE DECIMAL ±.005

INTERPRET GEOMETRIC TOLERANCING PER: Y14.5-2009

MATERIAL
AISI 1020

FINISH NO PLATING
3.2

	NAME	DATE
DRAWN	DDY	16/12/03
CHECKED	AKERR	16/12/03
ENG APPR.		

DIMENSIONAL
UNIT: INCHES

DO NOT SCALE DRAWING

DESCRIPTION:
TIE ROD CLEVIS PLATE

SIZE **B** DWG. NO. PS2M36710 REV A

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

3.5.3 Bill of Materials

The preliminary bill of materials for the project can be seen in TABLE LXXXIV. It includes all three systems: support structure, table, and lifter. Each of the system is broken down into various assemblies, sub-assemblies, and parts. All fasteners are priced using information provided by McMaster-Carr, all prices on the website are in USD and are converted to CDN using a conversion of $1 USD is equivalent to $1.30 CDN. Components are priced out by contacting the respective manufacturer and obtaining quotes for the components. These components include: the dampers, rollers, bearing carriages, bearing rails, vacuum lifter, and vacuum generator. For items made out of AISI 1020, 6061-T6 and, plain carbon steel, a calculation is performed to determine the total cost of all items of that material. The calculation for the three listed materials are performed by determining a dollar per gram value and multiplying that value against the total weight of all parts made of that material. The values per gram for AISI 1020, 6061-T6 and, plain carbon steel can be seen in TABLE LXXXII.

TABLE LXXXII: MATERIAL DOLLAR PER GRAM VALUES

Material	Dollar per Gram
AISI 1020	0.01399
6061-T6	0.0256
Plain Carbon Steel	0.014

The 530 parts for the total machine can be broken down into six categories that can be seen in TABLE LXXXIII.

TABLE LXXXIII: PART TYPE AND PART COUNT

Part Type	Part Count
ASTM A36	30
AISI 1020	65
6061-T6	10
Plain Carbon Steel	4
Fasteners	367
Components	54

TABLE LXXXIV: BILL OF MATERIALS FOR THE SHEET TRANSFER STATION

MAIN ASSEMBLY	TIER 1	TIER 2	TIER 3	TIER 4	QUANTITY	MATERIAL	MAKE FROM	WEIGHT	DESCRIPTION	COST	EXTENDED COST	SUPPLIER
TABLE	AIR KNIFE	PT2M36732			2	AISI 1020	1/4 PLATE	1924.25	AIR KNIFE MOUNTING BRACKET	SEE CALCULATION		
		PT2M36731			2	AISI 1020	1/4 PLATE	1924.25	AIR KNIFE MOUNTING BRACKET	SEE CALCULATION		
		92865A714			8	FASTENER				$ 0.33	$ 2.64	MCMASTER CARR
		91251A712			4	FASTENER				$ 1.20	$ 4.80	MCMASTER CARR
		94895A823			4	FASTENER				$ 0.21	$ 0.84	MCMASTER CARR
		AKF12			2	COMPONENT				$1,200.00	$ 2,400.00	INDUSTRIAL MAGNETIC INC
	AT1M00640	PT1M37140			2	ASTM A36		134007	TABLE TOP	$ 611.92	$ 1,223.84	
		PT2M34640			8	AISI 1020	SQUARE TUBE	8978	TABLE LEGS	SEE CALCULATION		
		TABLE BOTTOM			8	ASTM A36	1/4 PLATE	710.96		$ 0.25	$ 2.00	
		TABLE BRACKET			2	ASTM A36				$ 14.69	$ 29.38	
		TABLE BRACKET REAR			2	ASTM A36				$ 40.85	$ 81.70	
		PS1M36730			2	AISI 1020	.3125 IN SHEET	1.70E+03	GUSSET	SEE CALCULATION		
LIFTER	CYLINDER	6491K673			1	FASTENER			CUSHIONED TIE ROD AIR CYLINDE	$ 397.79	$ 397.79	MCMASTER CARR
		6491K74			1	FASTENER			CLEVIS BRACKET	$ 47.51	$ 47.51	MCMASTER CARR
		6491K63			1	FASTENER			ROD CLEVIS WITH PIN	$ 33.18	$ 33.18	MCMASTER CARR
	PR1A00040	PS2M13440			2	ASTM A36	C CHANNEL		LIFTER FRAME	$ 175.50	$ 351.00	
		PR2M37740			2	AISI 1020	.1875 SHEET		END PLATES	SEE CALCULATION		
		PS3M27140			1	AISI 1020	.25 SHEET	104.46	CLEVIS RECIEVER PLATE	SEE CALCULATION		
		PS2M28740			2	ASTM A36	2X2X0.023	5579		$ 31.05	$ 62.10	
		PS3M36730			4	AISI 1020	.2242 INCH THICK	1983		SEE CALCULATION		
		AL11000041	PL1M27341		1	6061-t6	.77 in thick	3042	MACHIED CARRIAGE HOLDER	SEE CALCULATION		
			TW-02-30LLZ		2	COMPONENT			CARRIAGE	$ 90.13	$ 180.26	Igus
			90965A200		2	FASTENER			BOLT	$ 0.30	$ 0.60	MCMASTER CARR
			91290A532		2	FASTENER			BOLT	$ 0.52	$ 1.04	MCMASTER CARR
		AL11000042	PL1M27342		1	6061-t6	.77 in thick	3042	MACHINED CARRIAGE HOLDER	SEE CALCULATION		
			TW-02-30LLZ		2	COMPONENT			CARRIAGE	$ 90.13	$ 180.26	Igus
			90965A200		2	FASTENER			FASTENER	$ 0.30	$ 0.60	MCMASTER CARR
			91290A532		2	FASTENER			FASTENER	$ 0.52	$ 1.04	MCMASTER CARR
		90107A032			24	FASTENER			WASHERS	$ 0.48	$ 11.52	MCMASTER CARR
		97135A255			12	FASTENER			NYLOCK BUT	$ 0.57	$ 6.84	MCMASTER CARR
		SPREADER STIFFINER			4	AISI 1020	.14 IN THICK	1023.02	SPREADER STIFFENER	SEE CALCULATION		
		92620A748			12	FASTENER			BOLT	$ 6.37	$ 76.44	MCMASTER CARR
	AL1M00040	L80M8-110-4/44			1	COMPONENT			ANVER 8 CUP LIFTER	$2,728.70	$ 2,728.70	ANVER
		VPF-57R-AC			1	COMPONENT			REMOTE VACUUM GENERATOR	$2,077.40	$ 2,077.40	ANVER
		90965A190			32	FASTENER			BOLT	$ 0.14	$ 4.58	MCMASTER CARR
		94645A210			32	FASTENER			LOCK NUT	$ 0.22	$ 7.04	MCMASTER CARR
		91290A470			32	FASTENER			NETRIC BOLT	$ 0.83	$ 26.62	MCMASTER CARR
		91257A854			2	FASTENER				$ 7.38	$ 14.76	
		90107A121			2	FASTENER				$ 1.26	$ 2.52	
		97135A280			2	FASTENER				$ 1.47	$ 2.94	
		AR1A00040	TK-02-30-2-2		2	COMPONENT			BEARING RAILS	$ 341.51	$ 683.02	Igus
			AR3A00040	RAIL SUPPORT	2	ASTM A36	3X3X.25	18200	LIFTER BEARING RAILS	$ 87.76	$ 175.52	
				CENTER PLATE	1	AISI 1020			CLEVIS PLATE	SEE CALCULATION		
				LIFTING CLEVIS	1	AISI 1020			CLEVIS PLATE	SEE CALCULATION		

MAIN ASSEMBLY	TIER 1	TIER 2	TIER 3	TIER 4	QUANTITY	MATERIAL	MAKE FROM	WEIGHT	DESCRIPTION	COST	EXTENDED COST	SUPPLIER
SUPPORT STRUCTURE	AL1M00040	PS1P21440			2	AISI 1020	.25 IN THICK	2795		$ 140.87	$ 281.74	
		SMALL DAMPER ASSEMBLY	SMALL DAMPER BRACKET		4	ASTM A36	L5X5X.3750	806.2		$ 12.07	$ 48.28	
			9530K18		4	FASTENER			DAMPER NUT	$ 15.43	$ 61.72	MCMASTER CARR
			DAMPER GUSSET		8	AISI 1020	.1 IN THICK	37.19		SEE CALCULATION		
			PMXT1575		2	COMPONENT					$ -	
		UPPER BEARING MOUNT	CIR-300E-R		8	COMPONENT					$ -	
			LOCKNUT FOR CIR-300E-R		8	COMPONENT					$ -	
			ROLLER MOUNT UPPER		4	PLAIN CARBON		1846.76		SEE CALCULATION		
		REAR DAMPER ASSEMBLY	Small Damper Bracket		2	ASTM A36	L5X5X.3750	806.2		$ 12.07	$ 24.14	
			DAMPER GUSSET		4	AISI 1020	.1 IN THICK	37.19		SEE CALCULATION		
			PMXT1575		2	COMPONENT					$ -	
			9530K18		4	FASTENER			DAMPER NUT	$ 15.43	$ 61.72	MCMASTER CARR
		TAPER BEARING MOUNT	CIR-300E-R		8	COMPONENT					$ -	
			TAPER BEARING MOUNT		8	6061-T6		610		SEE CALCULATION		
			LOCKNUT FOR CIR-300E-R		8	COMPONENT					$ -	
		90648A029			16	FASTENER			NUT	$ 0.11	$ 1.76	MCMASTER CARR
		91257A551			16	FASTENER			BOLT	$ 0.26	$ 4.16	MCMASTER CARR
		92620A305			12	FASTENER			BOLT	$ 4.90	$ 58.80	MCMASTER CARR
		92620A613			16	FASTENER			BOLT	$ 5.44	$ 87.04	MCMASTER CARR
		92916A355			32	FASTENER			WASHER	$ 0.11	$ 3.52	MCMASTER CARR
		93960A140			54	FASTENER			WASHER	$ 0.56	$ 30.24	MCMASTER CARR
	AS0M30042	PS1M36732			2	AISI 1020	.3125 IN PLATE	9.47E+03	RACK BRACKET	SEE CALCULATION		
		PS2M36732			1	AISI 1020	.3937 IN SHEET	8.23E+03	L BRACKET	SEE CALCULATION		
		PS1M36730			2	AISI 1020	.3125 IN SHEET	1.70E+03	GUSSET	SEE CALCULATION		
		PS2M36710			1	AISI 1020	.63 IN PLATE	3.43E+02	TIE ROD BRACKET	SEE CALCULATION		
	AS0M30041	L SUPPORT OTHER SIDE			1	AISI 1020	.3125 IN PLATE	9.47E+03	RACK BRACKET	SEE CALCULATION		
		PS1M36732			1	AISI 1020	.3937 IN SHEET	8.23E+03	L BRACKET	SEE CALCULATION		
		PS1M36730			2	AISI 1020	.3125 IN SHEET	1.70E+03	GUSSET	SEE CALCULATION		
		PS2M36710			1	AISI 1020	.63 IN PLATE	3.43E+02	TIE ROD BRACKET	SEE CALCULATION		
	TIE RODS	TIE ROD CLEVIS			4	COMPONENT					$ -	
		TIE ROD TUBE			4	COMPONENT					$ -	
		92390A953			4	FASTENER			CLEVIS PIN WITH COTTER PIN	$ 16.25	$ 65.00	MCMASTER CARR
	AS2M30041	BRACKET TO RACK	L SUPPORT OTHER SIDE		1	AISI 1020	.3937 IN SHEET	8.23E+03	L BRACKET	SEE CALCULATION		
			PS1M36732		1	AISI 1020	.3125 IN PLATE	9.47E+03	RACK BRACKET	SEE CALCULATION		
			PS1M36730		2	AISI 1020	.3125 IN SHEET	1.70E+03	GUSSET	SEE CALCULATION		
		AS1M00540	PS1M23630		2	ASTM A36	C CHANNEL	5.36E+04		$ 299.70	$ 599.40	Metaldepot
			PS3M37130		2	AISI 1020	.25 IN THICK	6.91E+02		SEE CALCULATION		
			PS1M227430		2	ASTM A36	L3X3X.5	3.08E+03		$ 20.25	$ 40.50	
			90107A038		8	FASTENER			WASHERS	$ 2.43	$ 19.44	MCMASTER CARR
			92620A954		4	FASTENER			BOLT	$ 5.29	$ 21.16	MCMASTER CARR
			90630A14		4	FASTENER			NUT	$ 2.07	$ 8.28	MCMASTER CARR
		TIE ROD MOUNTING BRACKET	PS2M36710		1	AISI 1020	.4345 IN THICK	343	ROD END RECIEVER	SEE CALCULATION		
			PS3M37710		1	AISI 1020	.3125 THICK	3.17E+03		SEE CALCULATION		
	AS2M30042	AS1M00540	PS1M23630		2	ASTM A36	C CHANNEL	5.36E+04		$ 175.57	$ 351.14	
			PS3M37130		2	AISI 1020	.25 IN THICK	6.91E+02		SEE CALCULATION		
			PS1M227430		2	ASTM A36	L3X3X.5	3.08E+03		$ 20.25	$ 40.50	
			90107A038		8	FASTENER			WASHERS	$ 2.43	$ 19.44	MCMASTER CARR
			92620A954		4	FASTENER			BOLT	$ 5.29	$ 21.16	MCMASTER CARR
			90630A140		4	FASTENER			NUT	$ 2.07	$ 8.28	MCMASTER CARR
		BRACKET TO RACK	PS1M36730		1	AISI 1020	.3937 IN SHEET	8.23E+03	L BRACKET	SEE CALCULATION		
			PS3M36732		1	AISI 1020	.3125 IN PLATE	9.47E+03	RACK BRACKET	SEE CALCULATION		
			PS1M36732		1	AISI 1020	.3125 IN SHEET	1.70E+03	GUSSET	SEE CALCULATION		
		TIE ROD MOUNTING BRACKET	PS3M37710		1	AISI 1020	.4345 IN THICK	343	ROD END RECIEVER	SEE CALCULATION		
			PS2M36710		1	AISI 1020	.3125 THICK	3.17E+03		SEE CALCULATION		

The weights found in TABLE LXXXIV, are determined by using the Solidworks mass properties. With all the weights determined, the resulting values are multiplied by the values listed in TABLE LXXXII. The resulting costs for AISI 1020, 6061-T6, and plain carbon steel can be found in TABLE LXXXV.

TABLE LXXXV: COST BREAKDOWN

Items	COST	
ASTM A36	$	3,029.50
AISI 1020	$	1,499.73
6061-T6	$	171.37
Plain Carbon Steel	$	25.85
Fasteners	$	1,115.02
Components	$	10,296.20
Sub-Total:	$	16,137.67
Shipping Fees	$	1,400.00
Duty Fees	$	1,381.38
Taxes	$	2,097.90
Total:	$	21,016.95

The total cost for the project is estimated to be $21 016.95 based on the material costs alone. The estimated cost for the project does not include labor costs for assembly, manufacturing, or testing time. The Team believes that Ryerson is capable of sourcing raw material for a cheaper cost than what the Team used to quote thus leading the project estimate to be on the conservative side for material costs.

3.6 Project Recommendations

Unfortunately, the product being submitted to Ryerson Canada is not a final product and the Team has a list of 24 recommendations that Ryerson should consider prior to the product being implemented and tested in the field. The 24 number of recommendations can be categorized into four different sections: analysis, function, manufacturing and safety. An important general recommendation is to obtain qualification for the machine from a registered engineer to ensure the structure is safe and proper legal due diligence has been completed.

3.6.1 Analysis Recommendations

The team believes that more attention be provided to stress analysis as it is still completed at a moderately high level. The Team recommends performing further stress analysis by utilizing more finite element analysis tools and studies to verify and validate calculations performed by the Team. Fracture mechanics should also be explored at this time to ensure evidence of pending failure is visible prior to catastrophic failure occurring.

The most critical recommendation for the project would be to design a standalone frame to support the machine. The Team established this as the most important recommendation due to unknowns of the current racking system being utilized at Ryerson. The Team believes that the requirements to qualify and customize each rack system to support machine outweighs the cost and space requirements for a standalone structure. After the completion of the design prior to the implementation of the machine, Ryerson should explore tooling and jigging that will be required for manufacturing and assembling the machine.

The Team also recommends that Ryerson include a motorized drive system to move the sliding I beam inwards and outwards of the c-channel. The most cost effective route would be to implement a chain drive or belt drive system to move the sliding I beam. The Team determined this to be a recommendation due to it limiting the velocity of the apparatus thus mitigating one of the failure modes identified in the FMEA. The Team also recommends consulting a qualified engineer to determine sufficient welding specifications to be utilized in the structure to ensure the final design is safe for implementation.

Another recommendation from the Team is to perform a full modal analysis on the structure to ensure vibrations due to the movement of the lifter will not affect the integrity of the structure. The structure and frame that support the lifter assembly should also be revisited to increase the factor of safety on the assembly. The Team has determined that this is a key recommendation for the project due to the calculations yielding an approximate factor of safety of 2.3. The rest of the machine has been designed to a factor of safety of 8 and the Team recommends that the whole structure should be held to the same safety standard.

The final major analysis recommendation the Team would like to provide is that all the analysis should be repeated on the cantilever columns once more data on those specific columns become available. The Team encourages this as an important recommendation as all analysis performed has been based on estimated dimensions and assumed material properties for the cantilever rack system.

3.6.2 Functional Recommendations

The Team believes that the design should be explored further for lateral length adjustments or columns of different widths. The table presented as the final design should also be improved upon by adding guards to prevent damage to the air knifes mounted underneath the tables.

Ryerson stressed that they would like to have the machine lift 5 ft. wide sheets. The Team believes this is possible with a few minor design changes provided the weight of the sheets remain at 250 lbs or less. The Team has constantly ensured that the machine maintains stability and believes that Ryerson should explore utilizing double base columns which would help improve the stability of the machine and increase its overall performance.

3.6.3 Manufacturing Recommendations

A major recommendation for manufacturing would be to utilize precision structural members manufactured from an aluminum alloy. The Team has determined this as a recommendation due to the structural beams having poor tolerances that can lead to binding and tolerance stack up issues. Utilizing precision structural members would also be beneficial to ensure the machine is level across its entirety.

In addition, the machine should have its stiffness improved in areas of concern where high levels of deflection are possible. This increase in stiffness should be accomplished with the addition of stiffening plates, which would also help reduce localized buckling in members experiencing compressive loads.

3.6.4 Safety Recommendations

The Team recognizes Ryerson's commitment to safety, so it is recommended that they investigate and implement a guard rail to create a physical keep out zone for the machine. From a safety perspective, the Team would also like to recommend implementing both visual and audible warnings indicating that the machine is in use. These warnings could consist of hazard strobes, warning lights, and audible tones suitable for the facility. The Team highly recommends that Ryerson does not store material above the lifter on the cantilever racks as it has not been considered as a loading scenario during the design process. The Team has identified one critical failure mode as the sheets could be knocked off the top by the oscillations created by the machine being in operation.

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