



# UNIVERSITY OF MANITOBA

## MECH4860 FINAL DESIGN REPORT

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## **EXECUTIVE SUMMARY**

The problem presented by our client is to come up with a design in order to remove waste packaging film from a steel core, which is created during extrusion of multibarrier packaging at Winpak LTD. The film is currently removed by hand using a utility knife. This process is not very time efficient, but more importantly is causing employee injury and fatigue.

The objective of the project is to bring an engineering solution to the problem described above. The client has outlined needs associated with cost, safety, required time, quality and mobility.

In the first phase of the project we defined the project scope and established a schedule and Gantt chart (see appendix A). From there we used a brainstorming method to come up with a concept design that we filtered and refined to end up with the top 2 concepts – a handheld machine and a fully automated machine for film removal. We then merged some features from both ideas in order to establish our final design.

The final design is a spring-loaded blade that is driven along the length of the roll by an electric motor. The assembly is mounted onto a frame that sits on wheels to allow mobility. The total weight of the design is 740 pounds, and it has a final cost of \$10,011.75. This new method of film removal will reduce the time required to remove the film from 30 min to 11 for a full roll. It incorporates a safety screen, which protects the operator from the moving parts. The entire design process spanned from September 2014 to December 2014, and our group is confident that our final design will accurately meet the needs and specifications of our client.

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## 1. INTRODUCTION: PROBLEM DESCRIPTION

Winpak is a company based in Winnipeg MB. The company manufactures and distributes packaging material for the food and healthcare industries. Winpak was founded in 1977 in Winnipeg as part of a global group, and operates 10 plants in Canada, USA and Mexico. Winpak converts raw material (resin), which is processed and turned into rolls of packaging film. Figure 1 below shows a render of a roll of extruded packaging film.

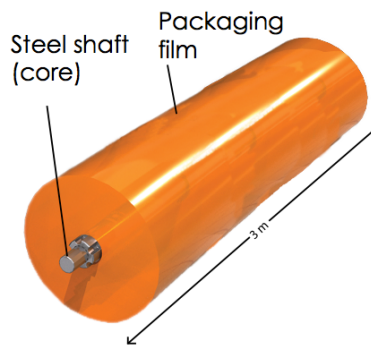


Figure 1: Render of extruded packaging film

Rolls of extruded film go through further processes including slitting, printing, finally being transformed into three categories of products:

- Flexible packaging: used for food packaging and includes vacuum-sealed packaging
- Rigid packaging: includes plastic hard trays
- Pharmaceutical packaging: includes blister foil packaging used for prescription drugs

Our project focuses on the long extruded rolls as shown in Figure 1. When the packaging film is defective due to the presence of air bubbles, non-uniform winding or off-specification thickness, it becomes scrap material and needs to be removed from the steel core. This scrap plastic roll can reach a diameter of up to 1 meter. Currently this process of cutting back scrap material is done manually using a utility knife. This manual cutting is dangerous for the workers and is time consuming. Our goal is to come up with an engineering solution to

the problem, designing a method of removing the scrap film that will be safer and faster than the existing manual operation.

Our main project deliverable is to have a set of drawings showing all the components of our design. The design will have to meet the following specifications as defined by the client.

a. Safety

The safety of the operators is the main issue that this design is trying to solve. The current manual method of cutting is done with the roll sitting on a pallet at ground level with the operator bent over the roll to cut along the length of the roll. This causes discomfort, and creates the risk of both cutting injuries and strain injuries for the workers. The client has stated that the primary specification for the project is safety, namely exceeding existing regulatory standards while maintaining acceptability of the process to workers performing the operation.

b. Mobility

Due to limited space available on the shop floor, the client has requested that we make the design mobile. This will make the design much more convenient to use, and will give operators and managers options for placement of the final design.

c. Time Efficiency

The design needs to cut away an entire roll of scrap film faster than the current method. The current manual cutting method takes 30 minutes to cut away the scrap film from the widest rolls, and as such this will be our maximum time limit.

d. Budget

The client has specified a budget cap of \$50,000. This amount would include mechanical parts as well as material and labor costs.

e. Other specifications

The other specifications outlined by the client include the design being able to accept varying thicknesses of film. This is due to the fact that the scrap

material quantity on the rolls is not controlled but the maximum roll diameter to be considered is 1 meter.

The project objective is to come up with an engineering solution to the problem described earlier. The project deliverable is to provide detailed drawings for all the parts used for the construction of the design by December 1<sup>st</sup> 2014.



## **2. DESIGN DETAILS**

### **2.1 FINAL DESIGN INTRODUCTION**

Our team selected a sliding slicer design for our project, which will consist of a knife blade sliding back and forth across the roll of waste plastic film. The blade will be mounted on a spring-loaded mechanism, allowing the height of the sliding rails to stay constant for a number of passes of the blade. This mechanism will be driven by an electric motor and will be fully enclosed with a suitable guard in order to maintain safety. The entire machine is to be mounted on casters in order to allow it to be moved around the plant and into the work area, whilst a staging frame currently in use at Winpak supports the roll.

#### **2.1.1. FINAL DESIGN [HIGH LEVEL] DESCRIPTION**

Our final selected design consists of a sliding blade car, two guide rails for the sliding car, a motorized mechanism for driving the car and blade, and a frame to hold the mechanism in a suitable position relative to the frame, which supports the plastic film roll. A render of this machine is shown in Figure 2, with the safety screening on the frame hidden for clarity:



Figure 2: Slicing machine render, with screening hidden

This frame has a capacity sufficient to handle expected forces created from the moving blade car and the static weight of the assembly. All components are to be constructed using standard steel plate and angle sections, as well as miscellaneous fastening hardware and bearings. A513 steel was specified for all components that are to be manufactured by the client. The slicing blade is a pair of standard OLFA safety knife blades fastened using machine screws, in order to facilitate maintenance. The motor will require 120vac single-phase power, and the machine will be mounted on casters, allowing for the entire machine to be moved throughout the plant as required. The overall cost of the machine is \$10 011, which is far lower than the maximum budget given by the client of \$50 000.

#### **2.1.2. FINAL DESIGN VS INITIAL CONCEPTS**

The final design that our group selected for development is based on one of our initial concepts for a solution to the design problem being considered. In our initial concept generation and selection process, we used a concept-screening matrix in order to determine our top concepts. Our top three concepts involved the redesign of the core upon which the plastic film is wound, a method for lining the core with a disposable material that would allow for the core to be

removed without cutting, and a single-slice design that would drive a blade back and forth across the roll of film in order to remove several layers of film on each pass. These conceptual designs are pictured as follows:

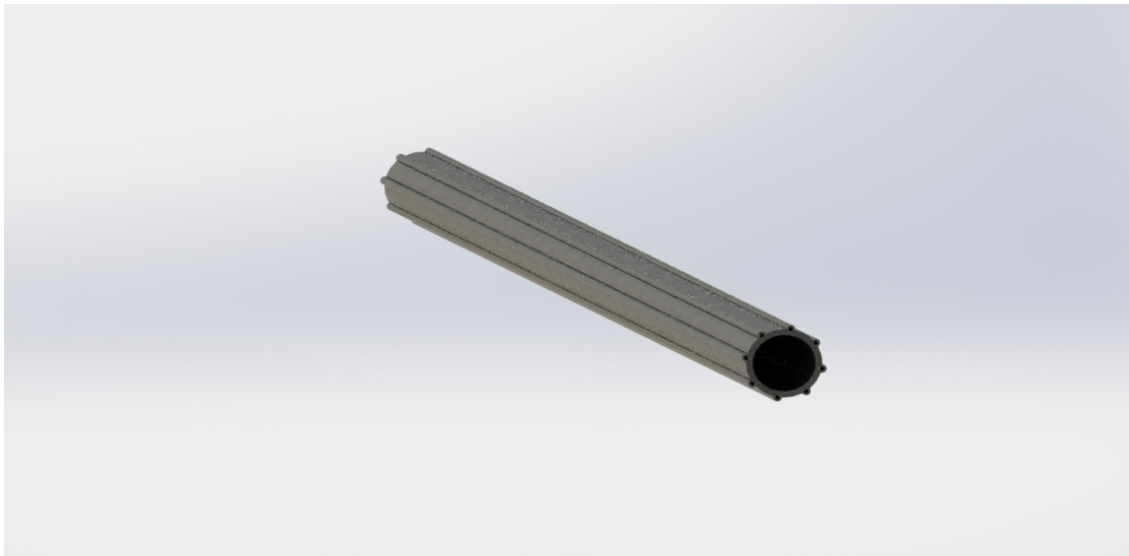


Figure 3: Modified core with hollow cylinders



Figure 4: Core liner model

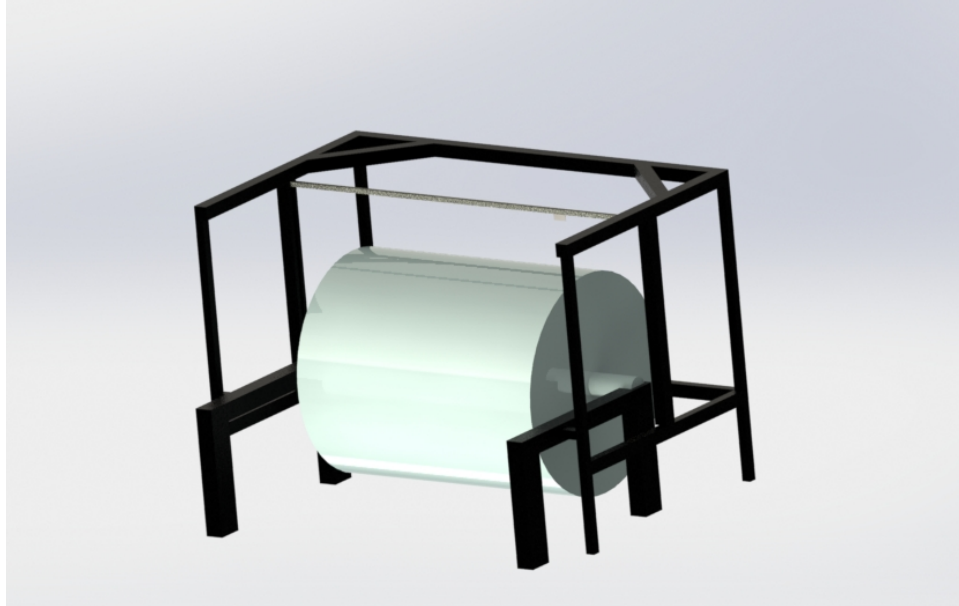


Figure 5: Single slice model isometric view

These three developed concepts were then presented to the client in order to obtain their feedback on which one is most suitable for use at Winpak. Unfortunately, the designs involving modification to the core were deemed to be unacceptable to the client due to the structural and quality requirements of the core. This left the single slice model as the only design among the three that was found to be acceptable to the client. As this meeting occurred immediately prior to submitting this conceptual design report, our group was unable to develop additional designs for the client to include in the concept design report. However, we did create a rough CAD model of a handheld slicing design, our fourth choice, in order to allow for a final selection to be done. This design involves a handheld device that incorporates a sharp blade, as well as safety features to prevent injury due from the blade and to reduce the force required to cut the roll. This CAD model is shown as follows:

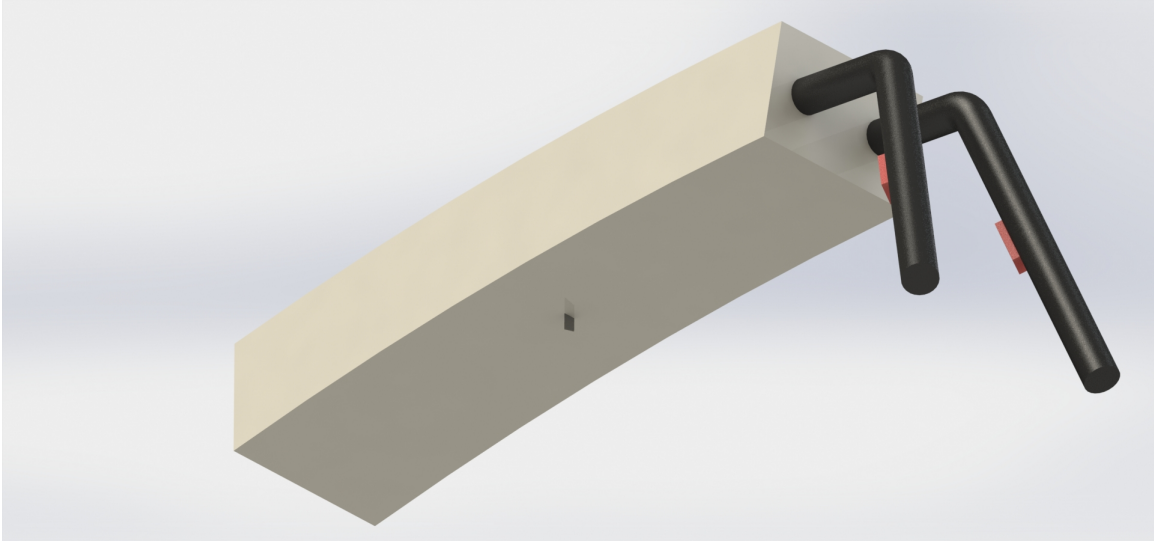


Figure 6: Handheld design render – bottom view

In this preliminary design, there is two buttons on the handles of the device required to retract the cutting blade and the device is shaped in such a way as to guide the cutting motion along the top of the roll of scrap plastic. This design and the single slice design were ultimately selected as final concepts for further development.

After further considerations such as the difficulty of manufacturing and impact upon employee morale were considered, our team selected the single slice design for further development. By reducing the labor time required to perform film removal, the single slice design is expected to improve morale of employees performing the operation. This design will also be far simpler to manufacture, as it can incorporate many off-the-shelf parts and will not involve complex hidden mechanisms in order to protect the blade. By using metal screening on this design, safety can be provided without compromising maintenance possibility. Additionally this design had a significantly higher weighted score than the handheld design in our original concept-scoring matrix, with a score of 3.57 compared to 3.33 for the handheld design.

## **2.2 DESIGN COMPONENTS**

In order to develop our concept of a slicing machine, we divided the overall design into three sub-systems. The Slicer system involves the sliding car, spring, rails and rocker arm and blade holder which perform the cutting

operation on the plastic and allow for small automatic adjustment for the height of the roll as the operation progresses. The Frame consists of the steel frame supporting the mechanism, the hinged door, and the safety mesh, as well as the casters, which allow the design to be mobile. Finally, the drive mechanism consists of the winch and pulley system, which allows for large-scale adjustment of the cutting head height, as well as the motor, sprocket, bearings and chain that drive the cutting head from side to side during the operation. Further description of each of these systems is shown in the sub-sections below.

### **2.2.1. SLICER**

The slicing mechanism of the design was created to accommodate a 6" travel distance, applying cutting force using a compression spring and rocker bar mechanism. This device rests on two sets of three ball bearing assemblies in order to ensure a consistent position relative to the two track rails. The slicing device is designed to remove plastic from the roll in increments of 6" diameter, after which point the relative position of the rails is to be adjusted by the operator using the winch in order to allow for the removal of the remainder of the diameter of plastic. Cutting is performed using two OLFA SKB-2 blades, which were selected on the basis of their proven performance in cutting this multibarrier plastic, in addition to the fact that this blade is already available at Winpak. These blades are held in place using shoulder bolts on a drilled and tapped blade support allowing for a 4mm gap between the blade and the holder. The compressive spring surrounds a rod, which prevents buckling of the spring, and ensures that the force applied to the lever follows the angle of rotation of the lever. An isometric view of this mechanism is shown in Figure 7, with the spring removed for clarity:



Figure 7: Isometric view of slicing mechanism

All hardware used was specified from McMaster Carr in order to facilitate the design, modeling and preliminary costing of this aspect of our final design. These components, however, are standardized and can be replaced with a suitable equivalent part if desired. Apart from the fastening hardware, the other parts of the slicer are manufactured from standard steel sections using typical machining operations - sawing, milling, drilling and tapping. These sections include L3x3x1/2" angle sections, as well as 1/8", 1/4", and 1/2" steel plate. All bolts are to be locked using a suitable thread-locking compound as available at the Winpak Ltd. shop.

The sliding mechanism was designed by first considering the compression spring, specifying a commercially available spring for use, and then specifying mounting hardware and designing the travelling car to carry the cutting lever. The design space created for the car was that it had to fit within a 12" clearance above the maximum diameter roll, it must have two blades in order to allow for bidirectional cutting, and that it must apply a force within the experimentally determined cutting range of 15-25 pounds-force. Imperial measures were used for the design of this mechanism, in order to facilitate calculations involving commonly available parts produced for the North American market.

The preliminary calculations for the spring were based on determining a spring constant for a spring providing the desired force throughout the distance range desired, assuming vertical deflection and using Hooke's law for simple springs:

$$F = -k \cdot \Delta x$$

Where  $F$  is the reaction force of the spring,  $k$  is the linear spring constant, and  $x$  is the linear position of the spring relative to its undeformed state. This calculation suggested a suitable spring constant was 1.65 lbf./in., though the undeformed length of the spring was found to be 17.13 inches. This length was deemed to be excessive, resulting in the decision to use a lever mechanism instead of a simple vertical deflection mechanism. A lever length of 6" was selected, allowing for a deformation of 3" in each direction relative to the neutral axis of the cutting tip of the lever, with a corresponding rotation in each direction of 30 degrees in each direction. A compression spring with a total length of 6" and a spring ratio of 13.96 lbs./in. was selected from the McMaster Carr catalogue – part no. 9657k212. In order to obtain the maximal applied force at the tip of the lever at the maximal deformation of the spring, an initial value problem of Hooke's law was solved for, resulting in a position of 1.76 in. from the pivot of the blade lever. This resulted in a maximal moment of 105.18 in-lb. on the lever. This moment was used in order to solve for the minimum thickness of the lever assuming that it is manufactured from 1/2" plate steel. The minimum thickness was found to be 0.225", and 1/2" was selected in order to provide a factor of safety and to simplify the design process by allowing for ample stress concentrations at fastening points.

This lever was used as the basis for the design of the sliding car and rail system. Two L-section rails spaced 6" apart were selected for use, and sized based on an assumed rolling mass of 25 pounds. Using the maximum shear force and bending moment generated by this mass, a minimum second moment of inertia was selected. This moment of inertia was compared to those tabulated in [1]. Our team selected L2x2x1/8" steel sections for the rails owing to their structural sufficiency and commercial availability. A plate of 6"x6" constructed of 1/4" steel was selected to hold the spring and lever mechanism, as well as the connection to the rails, as the stresses within this plate will be less than those within the lever. The plate slides along the rails on 6, 3/16" bearings (McMaster Carr Part no. 6384k88.) These bearings are held in place by machine screws



installed with a washer into two trimmed, drilled and tapped L3x3x1/2" steel sections welded to the metal plate. Three bearings were selected for each side of the plate in order to prevent lifting of the car off of the track, in a manner similar to that commonly used for roller coasters. The chain driving the lateral motion of the slicing car is connected to the car through the use of a small steel block drilled and tapped to connect to corresponding holes on the car using shoulder bolts. Two pins are to be removed from the roller chain, and the resulting holes used to connect the chain to the sliding car. This mechanism is detailed in the exploded view in Figure 8. Further assembly and manufacturing drawings are shown in 6. Appendix B – Machine Drawings.

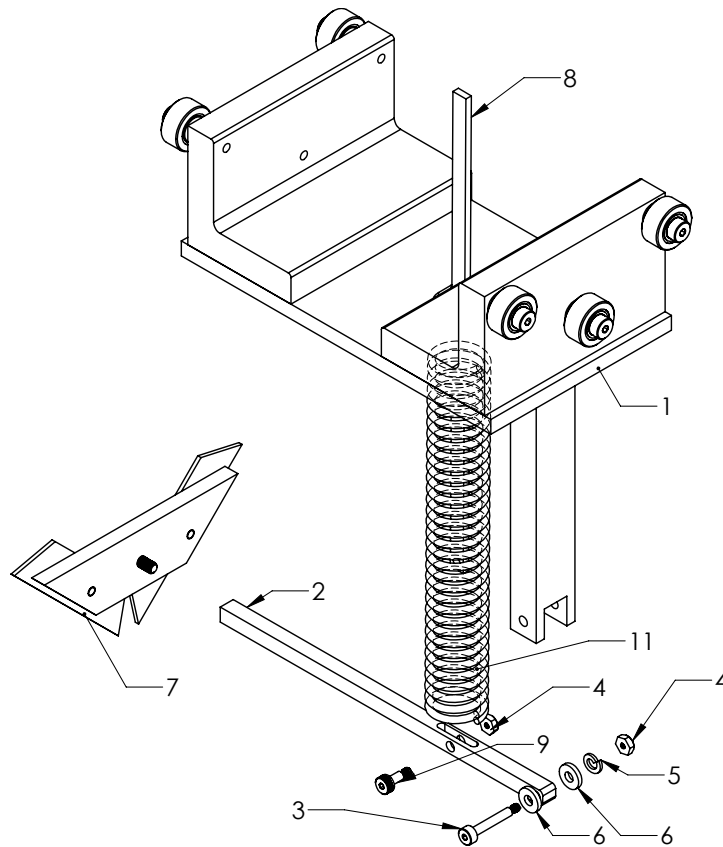


Figure 8: Exploded view of sliding car

### 2.2.2. FRAME

The frame of this design is a rectangular shaped structure that sits on 4 wheels located at each corner of the frame, allowing it to be easily moved.

Safeguarding grilles are mounted on two sides to prevent operator contact with the blade and to allow the operator to supervise the cutting process. The frame

uses 2 x 2 inch square steel tubing with a thickness of 1/8 inch. The following figure shows the frame, along with all of the welded parts and the winch.

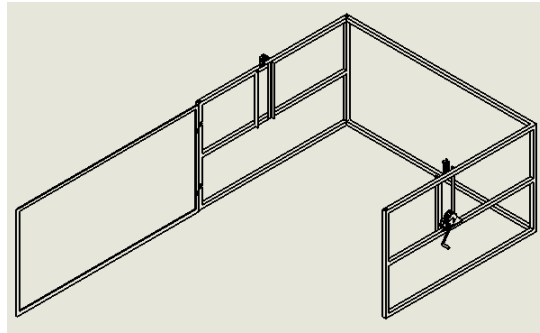


Figure 9: Frame with welded parts

One side of the frame is hinged to enable it to swing open or closed therefore allowing it to be positioned around the staging rack and then secured. On each side of the frame, two parallel vertical rails are mounted to guide the vertical motion of the cutter assembly. These rails are made using L2x2x1/2" angle sections and will be welded onto the frame with a distance of 6.63 inches between them. A winch is mounted onto a support plate on the outside of the frame. The support plate is ergonomically positioned at a distance of about a meter from the floor. The winch specified in this design has the following specifications.

Table I: WINCH SPECIFICATIONS

Capacity, lbs.		Rope		Base		Winch	
Maximum	1200	Maximum Length	106 ft.	Length	5 3/8"	Winch Height	6 1/2"
Fully Wound	480	Diameter	3/16"	Width	3 1/2"	Handle Length	7"

A modification was made to the winch, that is, a split was welded into the winding drum of the winch in order to allow two separate cables to be wound at the same rate. This is seen in the following figure.

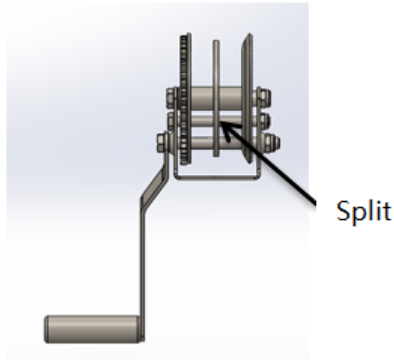


Figure 10: Modified winch

The winch controls a pulley system, which uses 3/16” steel cable to lift the track on which the slicer is mounted. Two different pulleys are used to support the cable. The first is a mounted double pulley with a workload limit of 600lbs. Its dimensions are shown in Figure 11:

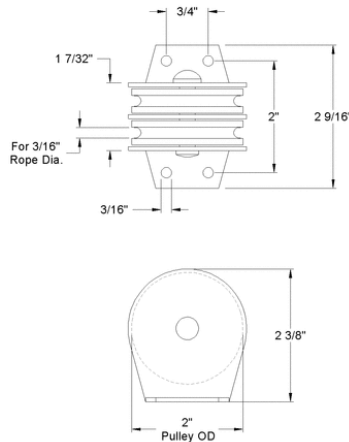


Figure 11: Double pulley dimensions [2]

The second pulley is similar to the first except that it only has one cable mount. It also has a workload limit of 600lbs. The following figure shows the associated dimensions.

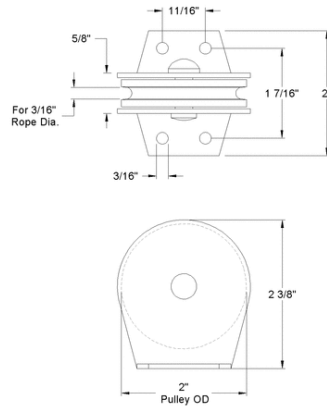


Figure 12: Pulley dimensions [2]

Supports for the pulleys are machined and then welded to the frame. The following figures show the dimensions of the two supports.

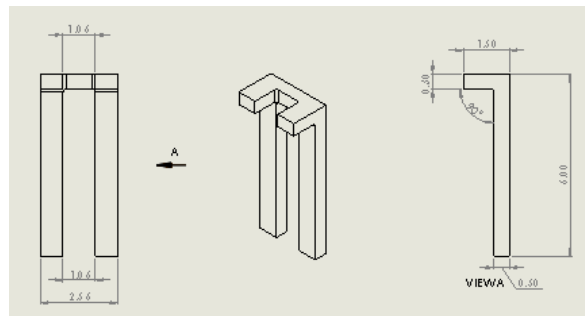


Figure 13: Support for double pulley

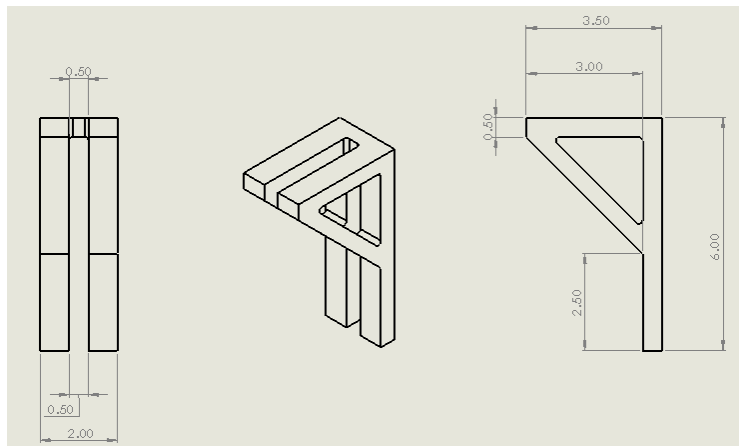


Figure 14: Support for single pulley

Two cables are used to lift each side of the rails as described above. These cables have a diameter of 3/16 inch, which has a breaking strength of 3000lbs. The cable going across the frame will require a minimum cable length of 15 feet and the shorter cable requires a minimum length of 5 feet.

### 2.2.3. DRIVE MECHANISMS

In order to cut the plastic a driving mechanism must be used to push the blade through the material. In this case a relatively slow speed is required since we do not want to put too much energy into the material. Excess energy would cause the plastic to melt and bind together, jamming the mechanism. Therefore a slow speed and high torque drive is required. The best method for applying this motion to the sliding car was determined to be a chain drive mechanism.

When designing a chain drive mechanism, first the force exerted onto the chain must be determined. Once this value is known, an appropriate chain size can be chosen. Then given the chain size a sprocket size must be determined according to the output speed required to cut. A speed of 0.4 meters per second was determined as our target speed. Through experimentation it was found that a 4-millimeter cut through the plastic required a force of 20 pounds.

Since this force is quite small, a small commercially available chain was chosen for this design. The chain number used will be ANSI 25, which has the following characteristics. [3]

Chain Number	Pitch		Working Load	
	In.	mm	In.	N
25	0.25	6.35	88	390

The working load of ANSI 25 chain is about four times the calculated required force to cut the plastic therefore failure of the chain is not an issue.

A sprocket must be specified in order to carry the chain and to drive it. Since the chain will travel a long distance, a larger sprocket is preferred in order to prevent contact of the slack side and the tensioned side of the chain. This could happen due to the chain moving unexpectedly while it is driven back and forth.

Therefore the largest diameter available on the McMaster Carr catalog was chosen. This sprocket has the following specifications.

Table II: SPROCKET SPECIFICATIONS [4]

Number of Teeth	36
Bore Size	¾ in
Outer Diameter	3.01 in
Width	¾ in
Diameter Hub	1-1/2 in

The required torque from the motor can now be determined using the pitch diameter of the sprocket and the determined force required to cut the plastic film. The pitch diameter is calculated using the following equation.

$$P_d = \frac{P}{\sin\left(\frac{180}{N}\right)}$$

Where  $P$  is the pitch of the chain and  $N$  is the number of teeth. A pitch diameter of 2.87in is calculated. The torque can now be calculated using the following equation.

$$T = F \cdot P_d$$

Where  $F$  is the force required to cut 4mm of plastic. The required torque is found to be 66lb-in. The required rotational speed of the motor can also be found using the sprocket pitch diameter. The following equation is used to find the rotational speed of the sprocket to obtain a slicer speed of 40 cm/s.

$$n = \frac{v \cdot 19100}{P_d}$$

Where  $v$  is the chain speed in m/s and  $P_d$  is the pitch diameter in mm. A value of 105 rpm is the result. Therefore the motor must have an output torque of at least 66lb-in at a rotational speed of 105 rpm.

The motor in this design will need to be able to run in both directions since the blade will be travelling back and forth along the roll. It will also require the ability to be stopped at a precise point. For these two reasons a stepper motor was chosen. A stepper motor can be controlled to stop at a precise location and can run in both directions. Stepper motors are commonly used in designs that require position control. A high torque, low speed motor will be preferred in this case, since we do not want the blade to travel too quickly through the plastic. The following motor was chosen from Kollmorgen's catalog.

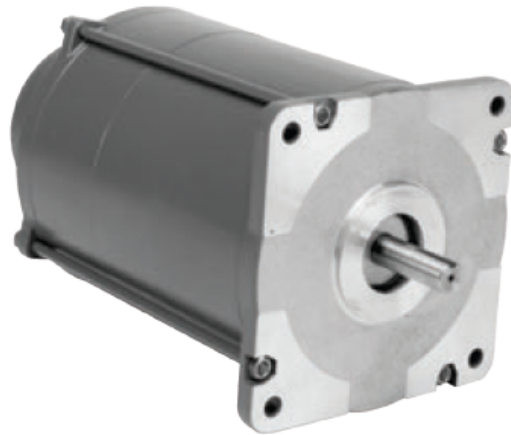


Figure 15: MH172 stepper motor

The specifications for this motor are found in the Kollmorgen catalog. A table was produced to list the important information related to this MH172 series stepper motor.

Table III: SPECIFICATIONS FOR MH172 STEPPER MOTOR [4]

Phases	2
Full Steps per Revolution	200
Step Angle	1.8°
Step Accuracy(of one full step, no load)	± 5%
Operating Temperature	-20°C to +40°C
Insulation Class	Class F, 155°C
Insulation Voltage Rating	340 Vdc
Insulation Resistance	100 Megohms

The following table shows the performance data of the motor as shown in the Kollmorgen catalog.

Table IV: PERFORMANCE DATA FOR MH172 STEPPER MOTOR [5]

Motor Model Number	Config.		Holding Torque (2 phases on) oz-in (Nm) +/-10%	Rated Current/ Phase Amps DC	Phase Resistance Ohms +/-10%	Phase Inductance mH Typical	Detent Torque oz-in (Nm)	Thermal Resistance Mounted °C/Watt	Rotor Inertia oz-in-s <sup>2</sup> (kg-m <sup>2</sup> x 10 <sup>-3</sup> )	Weight lb (kg)	Shaft Loading	
	Parallel	Series									Radial Force lb (N)	Axial Force lb (N)
	MH172	•		6120 (42.2)	21	0.077	2.1	50 (0.35)	1.1	0.87 (6.1)	53 (24)	25 (111)
MH172		•	10.6		0.31	8.5						

The catalog also provides the following performance curves for the parallel and the series configuration.

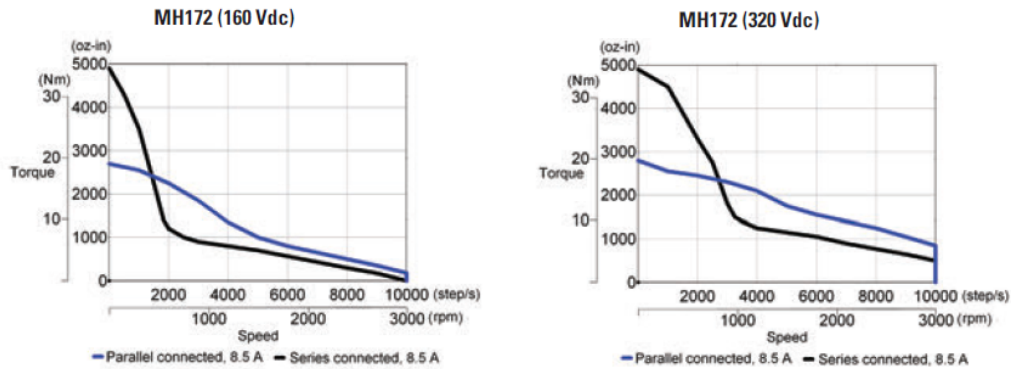


Figure 16: Performance curves for MH172 stepper motor [5]

From Figure 16: Performance curves for MH172 stepper motor, it is clear that at 105 rpm the stepper motor will supply more than 66lb-in of torque. The actual torque on the motor will be more, due to friction and the inertia of the chain-sprocket setup but the motor can supply more than twice the calculated torque, therefore lack of torque is not a concern.

With the motor a position drive must be purchased to control the position of the blade moving along the plastic. Kollmorgen has 3 different position drives, which are available in the catalog.



The motor shaft has a key slot, which will be used to mount the sprocket. A standard 3/16-inch square parallel key is used to transmit the torque from the motor to the sprocket. The key has the following dimensions.

Table V: KEY STOCK DIMENSIONS [4]

<b>Size</b>	<b>3/16 in x 3/16 in</b>
<b>Size Tolerance</b>	$\pm 0.008$ in
<b>Length</b>	12 in
<b>Length Tolerance</b>	-0.125 in

The free sprocket at the other end of the frame is the same size as the driving sprocket described above. It is mounted onto a similar shaft as the motor shaft with the same key stock described above. A bearing is used inside the support structure to allow the sprocket to spin freely. The following figure shows the assembly of the free sprocket.

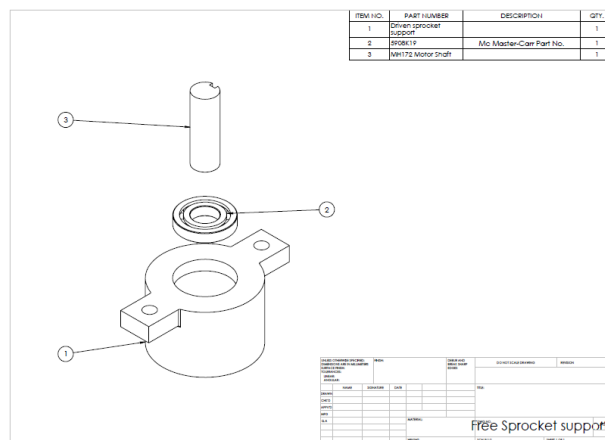


Figure 17: Free sprocket assembly

## 2.3 DESIGN ANALYSIS

The preliminary analysis of the design is based on engineering mathematical calculations and CAD analysis, as well as confirmation of load capacities of load bearing components based on mathematical specifications. The preliminary analysis will show that the critical components in the design can support the loads applied to them. We will also approximate the amount of user force required for the use of the design showing that we do not exceed acceptable workplace health and safety standards.

The following equations and derivations are used in correspondence with computer software to validate the theoretical performance of the design. We will use the equations as a form of validation for the results of the CAD FEA. To show convergence of the FEA models, h-Adaptive Convergence Graphs are shown. This convergence test is based on FEA results of the total strain energy in the part analyzed. The results for each member or part analyzed will be compared with the yield strength of the material in order to show that the part is of sufficient strength. Material properties of the steel used in the design, A513 Steel, are show in Table VI below.

Table VI - MATERIAL PROPERTIES OF ASTM A513 STEEL [6]

Property	Metric	Imperial
Density	7.85 g/cc	0.284 lb./in <sup>3</sup>
Yield Strength	435 MPa	63100 psi
Modulus of Elasticity	205 Gpa	29700 ksi
Shear Modulus	80 GPa	11600 ksi

### 2.3.1 CROSS SECTIONAL PROPERTIES

The cross sections of the members used in our design dictate properties, which affect the stress loads in the design. Cross sections and applicable properties are listed below. We have specified standard structural shapes for use in the design, allowing for structural properties to be obtained from AISC (American Institute of Steel Construction) Tables [7], and need not be calculated. The theory for these calculations is provided for clarification.

#### Square Tube Member

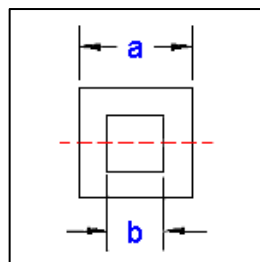


Figure 18 - Cross section of frame members [8]

Figure 18 above represents a square cross section, the following equations define the cross sectional properties.

Moment of Inertia:

$$I = (a^2 - b^2)/12$$

Section Modulus:

$$Z = (a^2 - b^2)/6a$$

L Angle Member

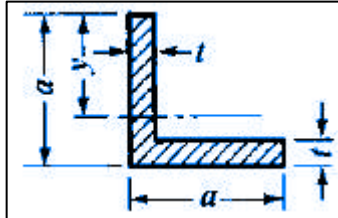


Figure 19 - L Angle cross section [8]

Figure 19 above depicts an L Angle cross-section; the formulae of sectional properties for this section are shown below:

Moment of inertia:

$$I = \frac{1}{3} [ty^3 + a(a - y)^3 - (a - t)(a - y - t)^3]$$

Section Modulus:

$$Z = \left\{ \frac{1}{3} [ty^3 + a(a - y)^3 - (a - t)(a - y - t)^3] \right\} / \left\{ a - \left[ \frac{a^2 + at - t^2}{2(2a - t)} \right] \right\}$$

Solid Rectangular Bar Member

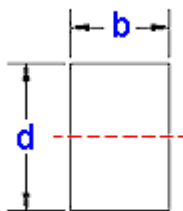


Figure 20 - Rectangular cross section [8]

For the critical component of the structural frame being considered, we will analyze  $b = 1$  in and  $d = .5$  in.

Moment of Inertia [8]:

$$I = \frac{bd^3}{12} \rightarrow \frac{1(.5)^3}{12} \rightarrow 0.010412 \text{ in}^4$$

Section Modulus [8]:

$$Z = \frac{bd^2}{6} \rightarrow \frac{1(.5)^2}{6} \rightarrow 0.04167 \text{ in}^3$$

The Sectional Properties are summarized below in Table VII.

Table VII - SECTIONAL PROPERTIES OF MAIN MEMBERS USED [7]

Cross Section	Moment of Inertia I	Section Modulus Z	Area	Weight Per ft.
Square 2"x2"x1/8"	0.486 in <sup>4</sup> 202288.47 mm <sup>4</sup>	0.584 in <sup>3</sup> 9570.05 mm <sup>3</sup>	.84 in <sup>2</sup> 531.93 mm <sup>2</sup>	3.05 lb.
L 2"x2"x1/8"	0.189 in <sup>4</sup> 78667.74 mm <sup>4</sup>	0.230 in <sup>3</sup> 3769.02 mm <sup>3</sup>	.491 in <sup>2</sup> 316.77 mm <sup>2</sup>	1.65 lb.
Rectangular Bar 1"x1/2"	0.010412 in <sup>4</sup> 4334.00 mm <sup>4</sup>	0.04167 in <sup>3</sup> 682.84 mm <sup>3</sup>	.5 in <sup>2</sup> 322.58 mm <sup>2</sup>	N/A

### 2.3.2 PRELIMINARY ANALYSIS OF DESIGN COMPONENTS IN BENDING

The loading scenario considered in the analysis of the identified critical members in the design is bending. We will approximate stresses using a double supported cantilever beam under a single point load. This scenario is pictured in Figure 21:

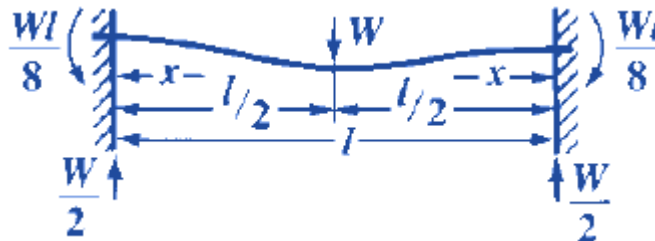


Figure 21 - Load scenario double supported cantilever beam [8]

Figure 21 above illustrates the basic loading scenario for a double supported cantilever beam under a single point load. There are three design components

that experience this type of loading. It is important to note that the bending stresses calculated below are higher than the stresses that members will actually carry in reality. This effect is due to the fact that this loading scenario assumes that the two ends of the beam are infinitely rigid and are unable to move at all, when in reality they will deflect slightly. The following equations will govern the process:

Stress at load point in the center of the beam [8]:

$$\sigma = W\left(\frac{1}{4}l - x\right)/2Z$$

Stress at the endpoints [8]:

$$\sigma = Wl/8Z$$

The highest stress will be seen under the worst-case load scenario. For this type of loading the highest stress occurs when the load is applied at the middle of the beam. Both of the stresses at the endpoints and in the middle should be the same with the mathematical model.

## Square Tube

The first member we will analyze using this method is the square tube member, which supports the pulley carrying the weight of the entire cutting mechanism. Two pulleys located at the ends of the frame support this mechanism. The total weight of the cutting mechanism is 64kg (141.1 lbf). We will assume the absolute worst case in which only one end of the frame supports the entire weight even though in reality both ends play a role. In proving that one member can support the entire load we can be confident that with both ends supporting the load the design will be sound. Figure 22 shows the location of the part in the design for clarification. It is also important to note the presence of two vertical L members, which will carry load. Again, by proving the square member can support the weight it is sufficient to say that with the added strength of the L members the design is sound.

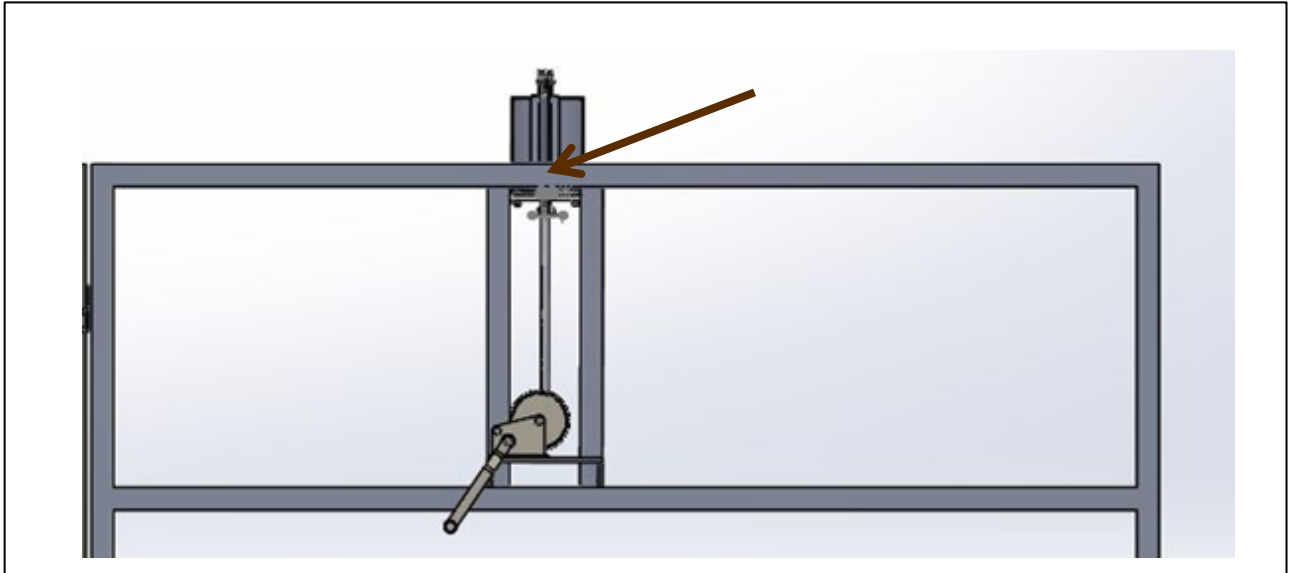


Figure 22 - Load area for square tube

Stress in the center [8]:

$$\text{Parameters: } l = 92in, W = 141.1 \text{ lbf}, x = 46in, Z = .584 \text{ in}^3$$

$$\sigma = \frac{W \left( \frac{1}{4} l - x \right)}{2Z} \rightarrow \frac{141.1 \left( \frac{1}{4} (92) - 46 \right)}{2(.584)} \rightarrow -2778.5 \text{ psi} \rightarrow 2778.5 \text{ psi} \rightarrow 19.16 \text{ MPa}$$

Stress at end points [8]:

$$\text{Parameters: } l = 92in, W = 141.1 \text{ lbf}, x = 46in, Z = .584 \text{ in}^3$$

$$\sigma = \frac{Wl}{8Z} \rightarrow \frac{141.1(92)}{8(.584)} \rightarrow 2778.5 \text{ psi} \rightarrow 19.16 \text{ MPa}$$

The maximum stress seen in the square tube is 19.16 MPa. This is far below the yield strength of the material of 435 MPa, this result yields a safety factor of approximately 22, showing that even in the absence of the vertical L members the and applying a larger load than will be exerted in reality this section of the frame is sufficiently strong. To further support this calculation Solid Works FEA results are show below in Figure 23 and Figure 24.

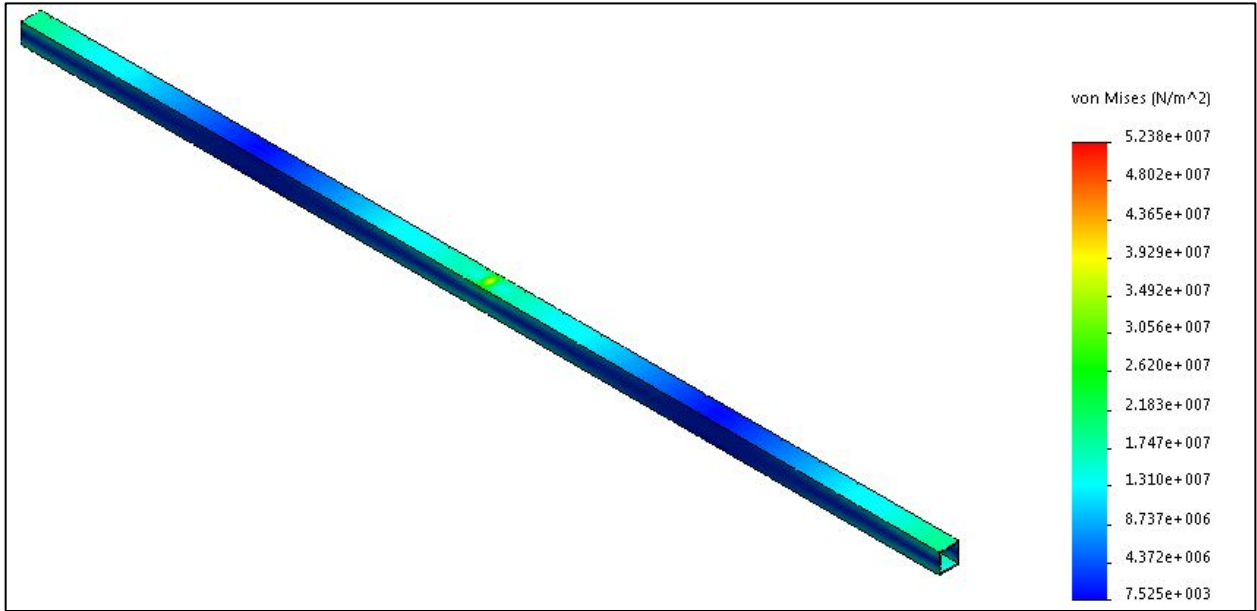


Figure 23 - Max stress CAD FEA of square tube member

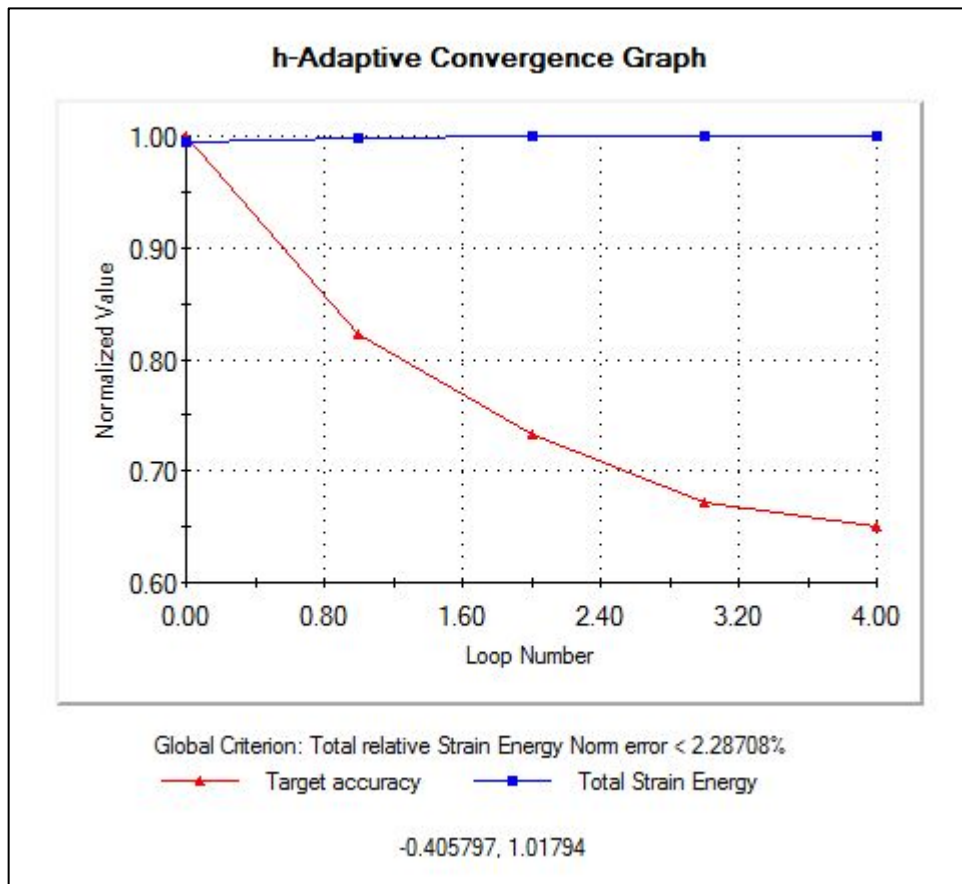


Figure 24 - h-Adaptive convergence plot for square tube member

In Figure 23 we see that by using FEA in Solid Works on the square tube member, the maximum stress in that member is 52.8 MPa. The stress computed using FEA is 2.6 times greater than that found in the mathematical analysis. This is likely the result of the increased constraints that result from the loads applied in the FEA. The FEA stress of 52.8 MPa is far below the yield strength of 435 MPa, resulting in a safety factor of 8.2. Again this is without considering the two L shaped support members. Figure 24 shows that the FEA study converged within 2.28 % error for the total strain energy. This shows that the FEA study is a good approximation of the stress in the member.

### L-member cutter mechanism support rail

To approximate the stresses in the L-member, which supports the cutter mechanism, we will follow the same approach used for the square tube member. The area of interest is indicated in Figure 25. Notice that the cutter mechanism is attached to two L members. Once again the loading consideration will be a worst-case scenario, with the load in the middle of the member and only one member supporting the load, while in fact there are two L members that were not considered in the analysis. The applied load is 58.76 lbf (261.4 N), based on the reaction force from the static state for the rocker bar, which calculated in the cutting mechanism design section.

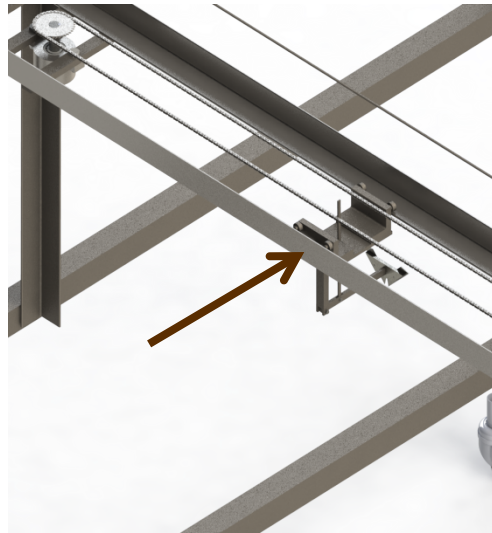


Figure 25 - Load area for L member

Stress in the center [8]:

$$\text{Parameters: } l = 116.1in, W = 58.76 \text{ lbf}, x = 58.05in, Z = .230 \text{ in}^3$$



$$\sigma = \frac{W \left( \frac{1}{4}l - x \right)}{2Z} \rightarrow \frac{58.76 \left( \frac{1}{4}(116.1) - 58.05 \right)}{2(.230)} \rightarrow -3707.6 \text{ psi} \rightarrow 3707.6 \text{ psi}$$

$$\rightarrow 25.56 \text{ MPa}$$

Stress at end points [8]:

*Parameters:*  $l = 116.1 \text{ in}, W = 58.76 \text{ lbf}, x = 58.05 \text{ in}, Z = .230 \text{ in}^3$

$$\sigma = \frac{Wl}{8Z} \rightarrow \frac{58.76(116.1)}{8(.230)} \rightarrow 3707.6 \text{ psi} \rightarrow 25.56 \text{ MPa}$$

The calculated maximum stress seen in the L member is 25.56 MPa. This is far below the yield strength of the material of 435 MPa with a safety factor of approximately 17, showing that even in the absence of the second L member our design is sufficiently strong. To further support this calculation Solid Works FEA results are show below in Figure 26 and Figure 27.

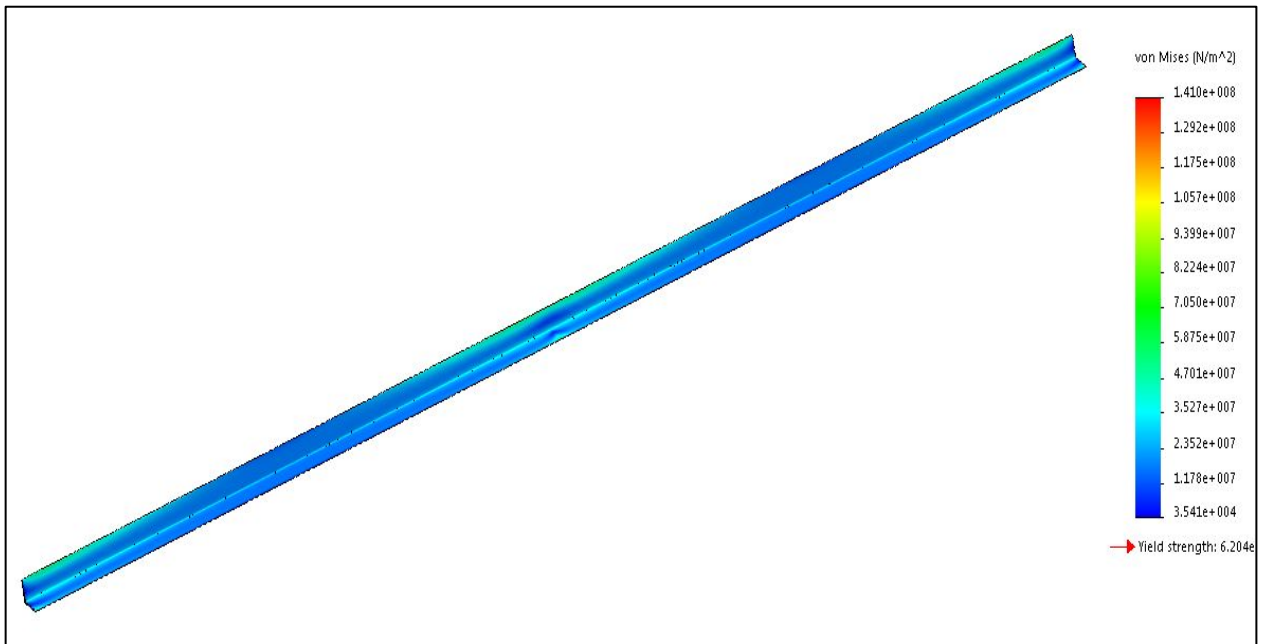


Figure 26 - Max stress CAD FEA of L member

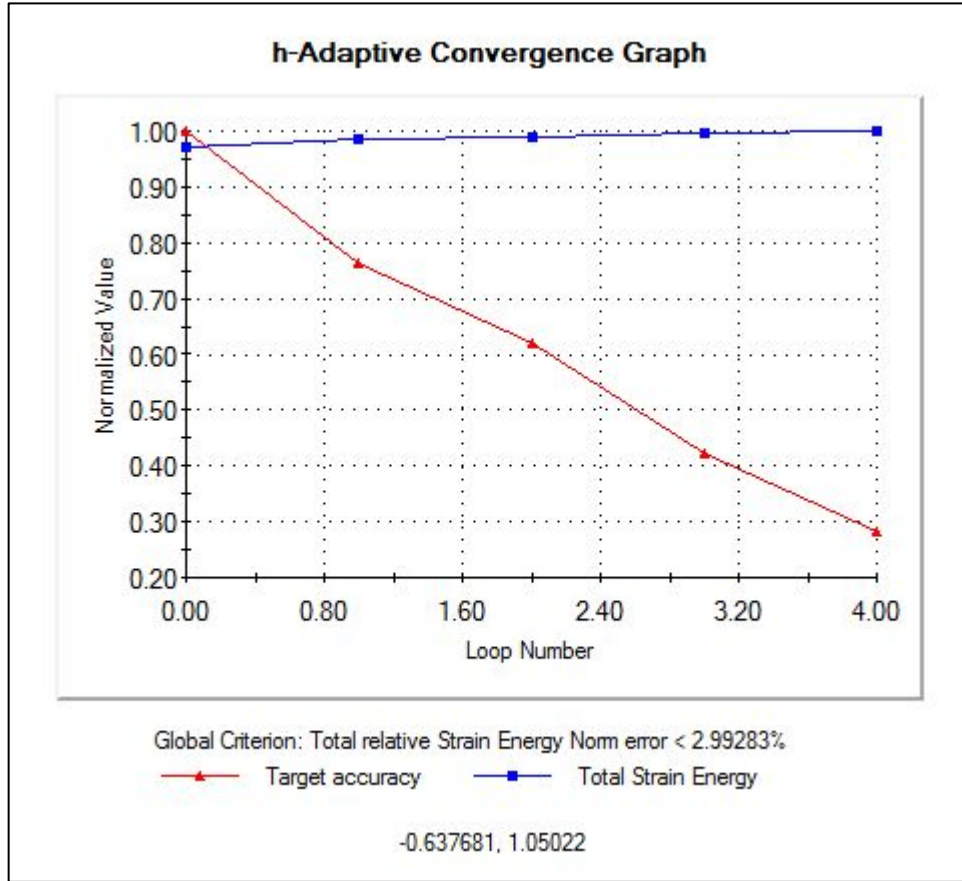


Figure 27 - h-Adaptive convergence plot for L member

In Figure 26 we see that by conducting FEA in Solid Works on the L member, the maximum stress in the member is 14.1 MPa. The stress computed using FEA is 1.8 times smaller than that of the mathematical analysis. This is likely the result of the way the load is applied in the FEA as the point load was applied using a split line resulting in some load distribution. The FEA stress of 14.1 MPa is far below the yield strength of 435 MPa resulting in a safety factor of approximately 30.

Figure 27 shows that the FEA study converged within 2.99 % error for the total strain energy. This shows that the FEA study is a reasonable approximation of the stress in the member.

## Cable Attachment Point

The cable attachment point is another critical point in the design. Once again the analysis is conducted using the load scenario in Figure 21. The point force the cable support will see is related to the tension in the cable created by the load of the entire cutting mechanism, 64kg (141.1 lbf). We will assume the worst-case load, which would be that one cable attachment point supports the load of the entire cutting mechanism. In reality there is a cable attachment and cable at each end of the cutting assembly. The area of interest is shown in Figure 28 for clarification.

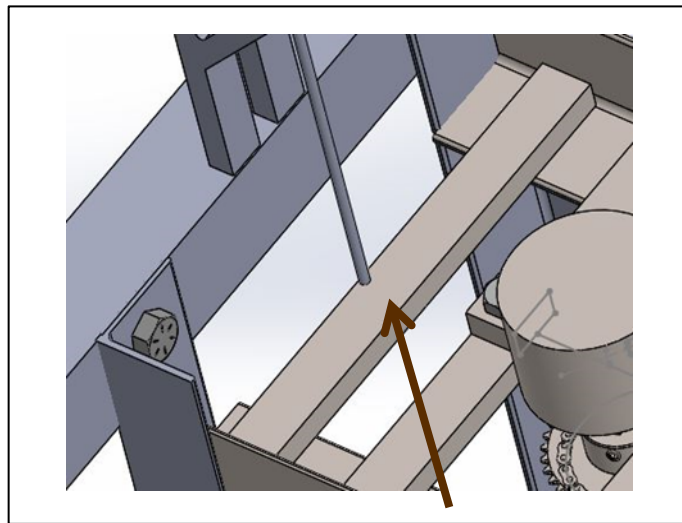


Figure 28 - Load area for cable attachment

Stress in the center [8]:

$$\text{Parameters: } l = 9.5in, W = 141.1 \text{ lbf}, x = 4.75in, Z = .04167 \text{ in}^3$$

$$\sigma = \frac{W \left( \frac{1}{4}l - x \right)}{2Z} \rightarrow \frac{141.1 \left( \frac{1}{4}(9.5) - 4.75 \right)}{2(.04167)} \rightarrow -4021.03 \text{ psi} \rightarrow 4021.03 \text{ psi}$$
$$\rightarrow 27.72 \text{ MPa}$$

Stress at end points [8]:

$$\text{Parameters: } l = 9.5in, W = 141.1 \text{ lbf}, x = 4.75in, Z = .04167 \text{ in}^3$$

$$\sigma = \frac{Wl}{8Z} \rightarrow \frac{141.1(9.5)}{8(.04167)} \rightarrow 4021.03 \text{ psi} \rightarrow 27.72 \text{ MPa}$$

The calculated maximum stress seen in the cable attachment is 27.72 MPa. This is far below the yield strength of the material of 435 MPa, yielding a safety factor

of approximately 15.7, showing that even in the absence of the other cable support carrying any load our design is sufficiently strong. To further support this calculation Solid Works FEA results are show below in.

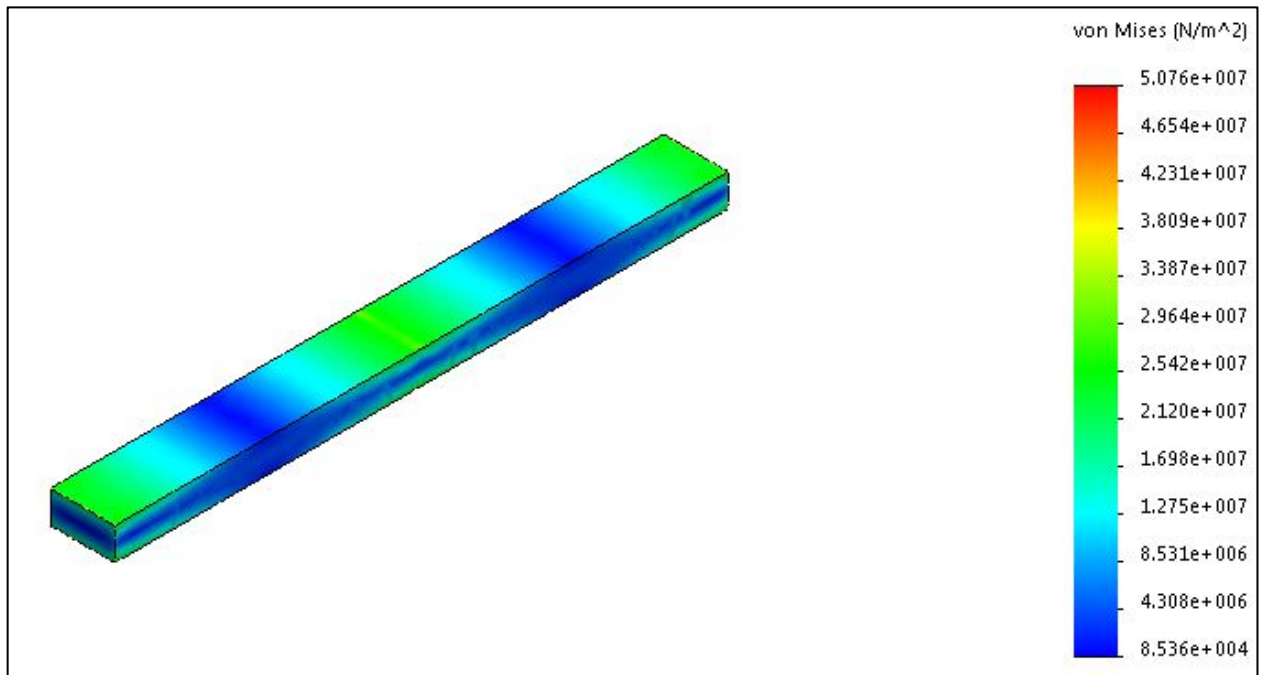


Figure 29 - Max stress CAD FEA of cable attachment

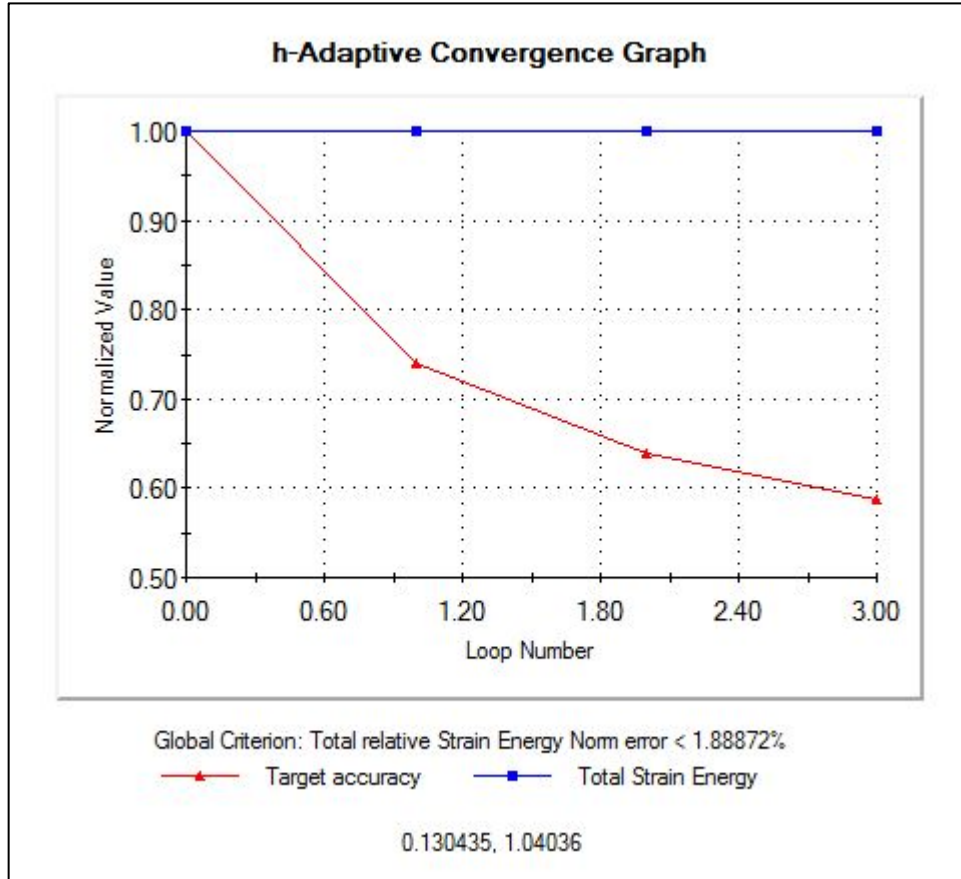


Figure 30 - h-Adaptive convergence graph of cable attachment

In Figure 29 we see that in computation of FEA in Solid Works on the cable attachment the maximum stress in the member is 50.76 MPa. The stress computed using FEA is 1.8 times larger than that of the mathematical analysis. The difference in the results may be the appearance of stress concentrations at the fixtures in the FEA. The FEA stress of 50.76 MPa is far below the yield strength of 435 MPa resulting in a safety factor of approximately 8.6. Figure 30 shows that the FEA study converged within 1.89 % error for the total strain energy. This shows that the FEA study is a reasonable approximation of the stress in the member.

### FEA only components

The following members were only analyzed using CAD FEA. This is due to the complicated nature of the shapes and the stress concentrations associated with them.

## Rocker Bar

The rocker bar connects the blade to the cutting assembly. The forces applied were the same as those used in the Cutting Mechanism design section. In order to compute a FEA solution we had to apply a fixture to the hinge point. In doing this we introduce a higher bending stresses than would be seen in the actual design as there is a moment created by fixing the support. As this effect is known, if we show that the rocker bar is sufficiently strong in the FEA we can be confident that the rocker bar will be sufficiently strong in the designed use. Figure 31 below shows where the forces are applied on the rocker bar. The forces applied are 25 lbf upwards [red arrow] at the blade end and 83.76 lbf downwards [blue arrow] at the spring point. The green arrow shows the fixed point.

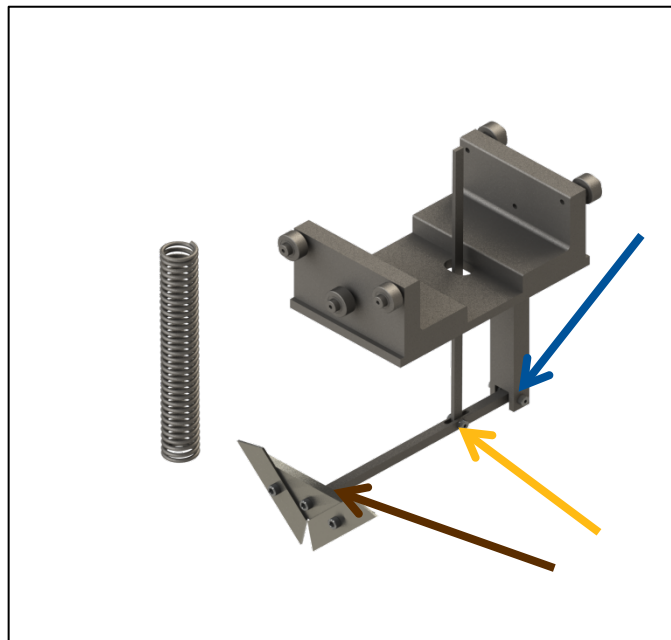


Figure 31 - Load area of rocker bar

Figure 32 and Figure 33 show the results of the CAD FEA computed on the rocker bar.

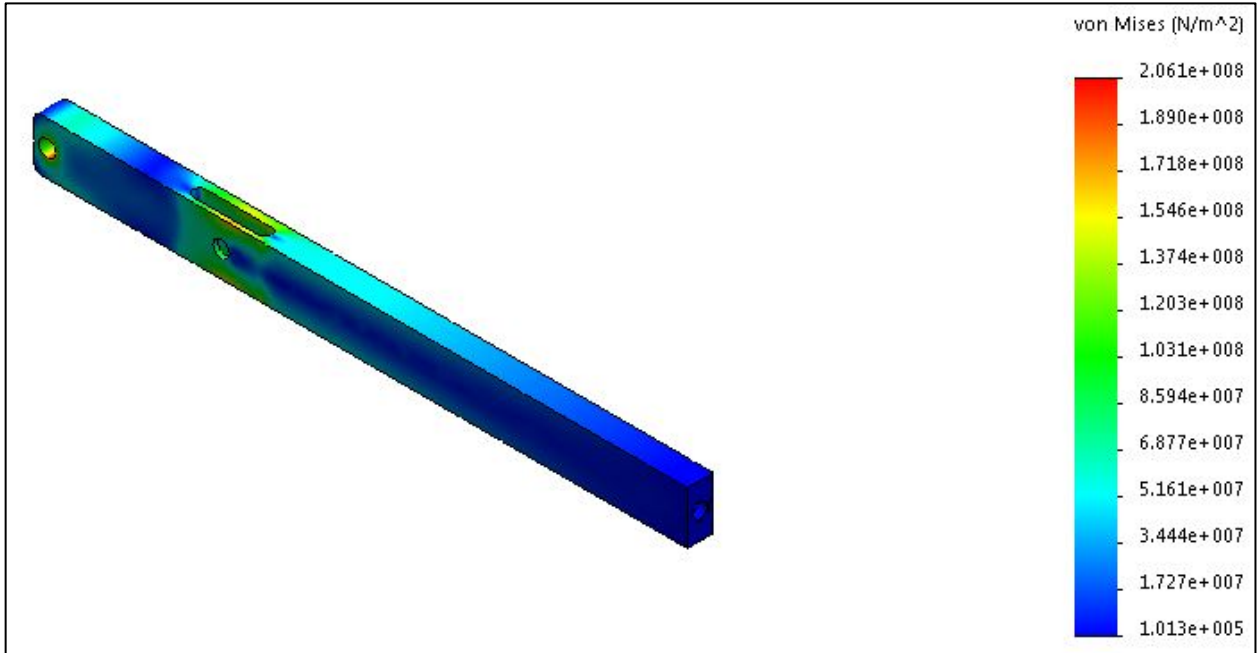


Figure 32 - Max stress FEA of rocker bar

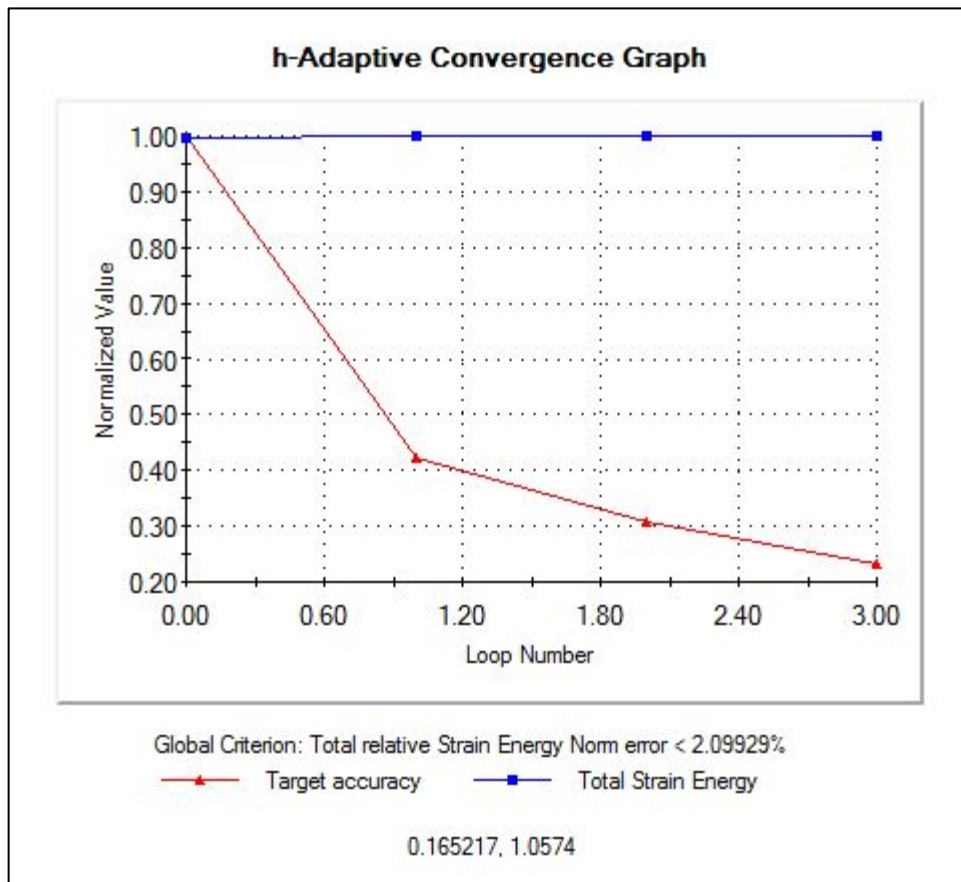


Figure 33 - h-Adaptive convergence graph of rocker bar

From Figure 32 it can be seen that the maximum stress in the rocker bar under the designed loading is 20.6 MPa. This is far below the yield stress of 435 MPa resulting in a safety factor of 21. Knowing that the constraints used in the FEA theoretically cause higher stresses to appear, the rocker bar is found to be sufficiently strong.

Figure 33 shows the convergence of the computed FEA to within 2.1% error with respect to strain energy. This convergence shows that the FEA is a good approximation of the maximum stress in the rocker bar.

### **Pulley Supports**

The pulley supports are attached at the ends of the frame to mount the pulleys, which carry the load of the entire cutting mechanism. Using the worst-case scenario loading each support was analyzed based on having to carry the entire weight of the cutting mechanism, 64 kg. In reality, each support would see a portion of that load. Figure 34 and Figure 35 show the area of interest with respect to pulley support 1 and pulley support 2.

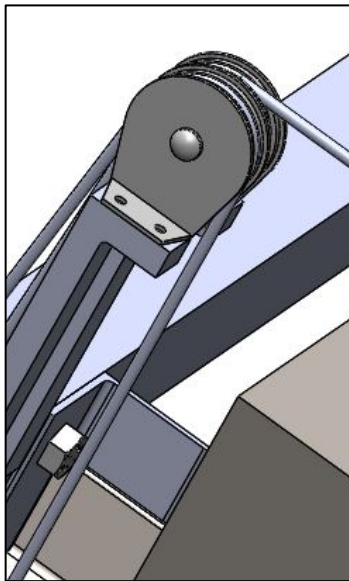


Figure 34 - Load area of pulley support 1



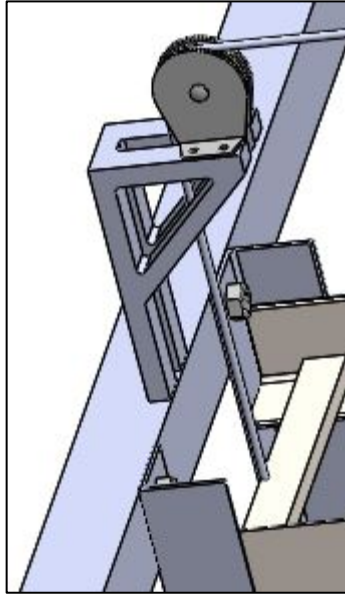


Figure 35 – Load area of pulley support 2

Figure 36 and Figure 37 show the results of the computed CAD FEA below.

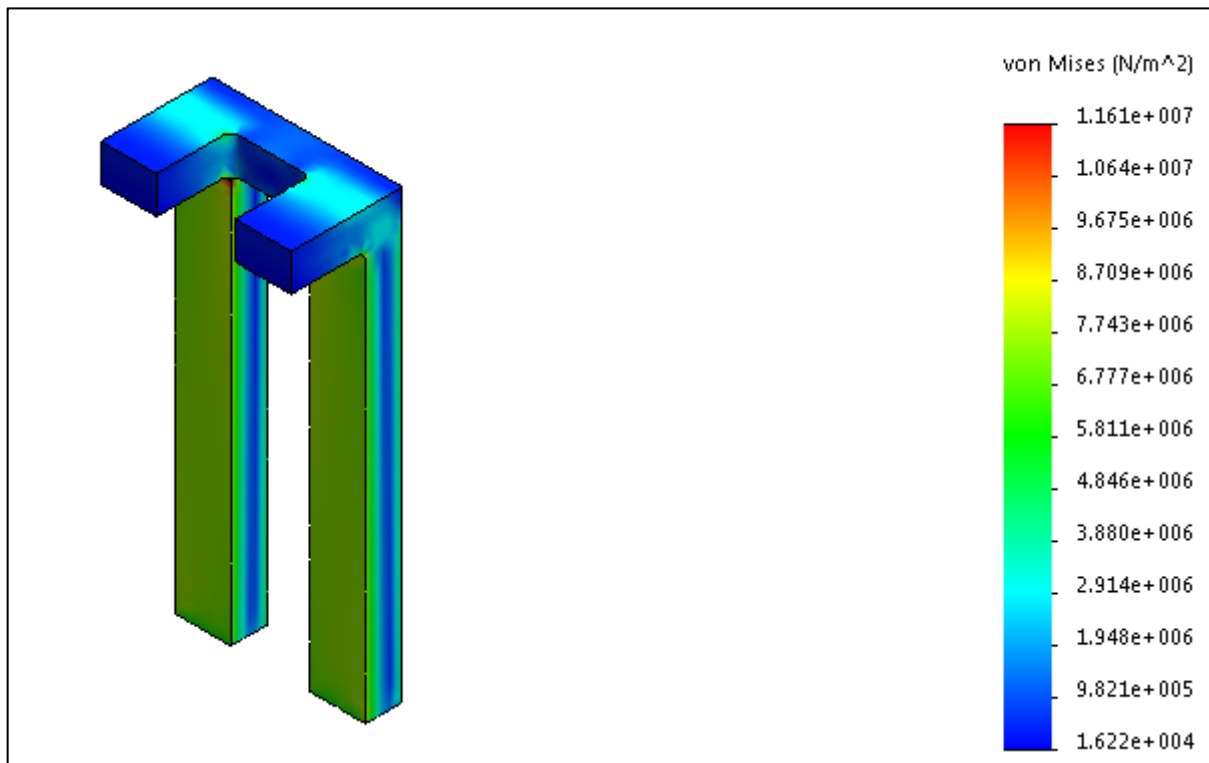


Figure 36 - Max stress FEA of pulley support #1

Figure 36 above shows the computed maximum stress for pulley support #1 based on FEA. The computed maximum stress is 11.6 MPa, far below the yield stress of the material of 435 MPa, resulting in a safety factor of 37.5. Even with

the worst-case scenario of double loading, the pulley support is sufficiently strong for the design.

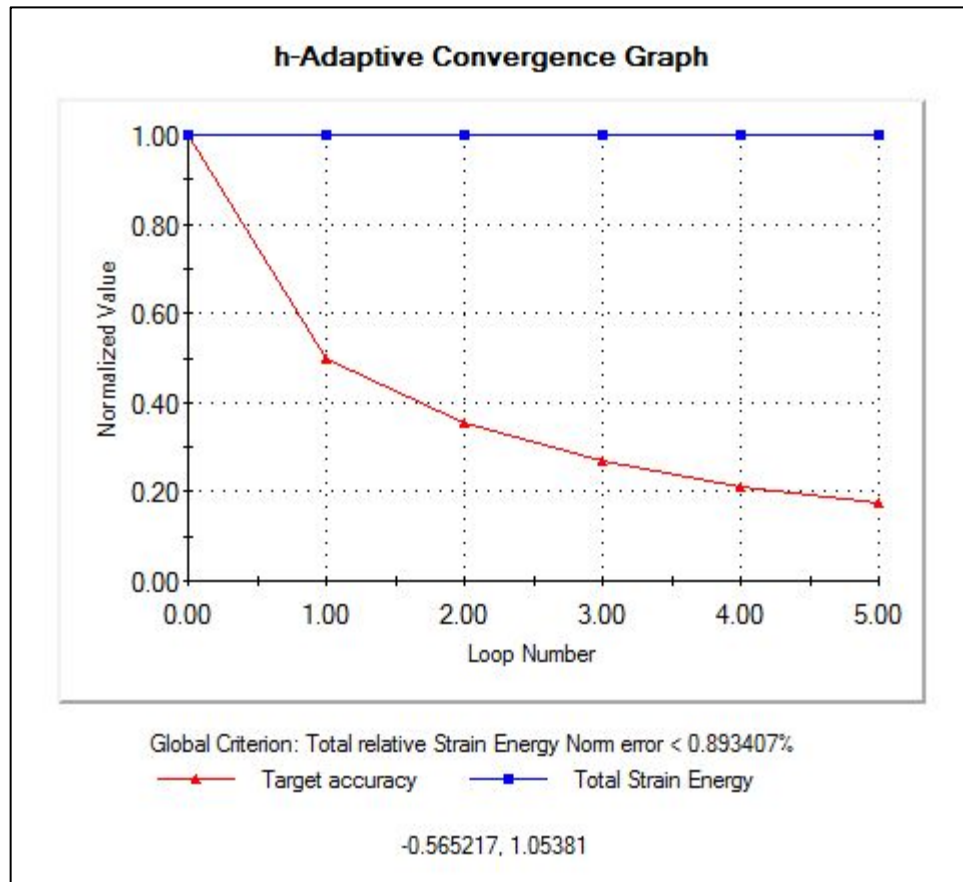


Figure 37 - h-Adaptive convergence graph for pulley support #1

Figure 37 shows the convergence of the computed FEA of within 1% error with respect to strain energy. The convergence shows that the FEA is a good approximation of the maximum stress in pulley support 1.

### Pulley Support #2

Figure 38 and Figure 39 show the results of the computed CAD FEA below.

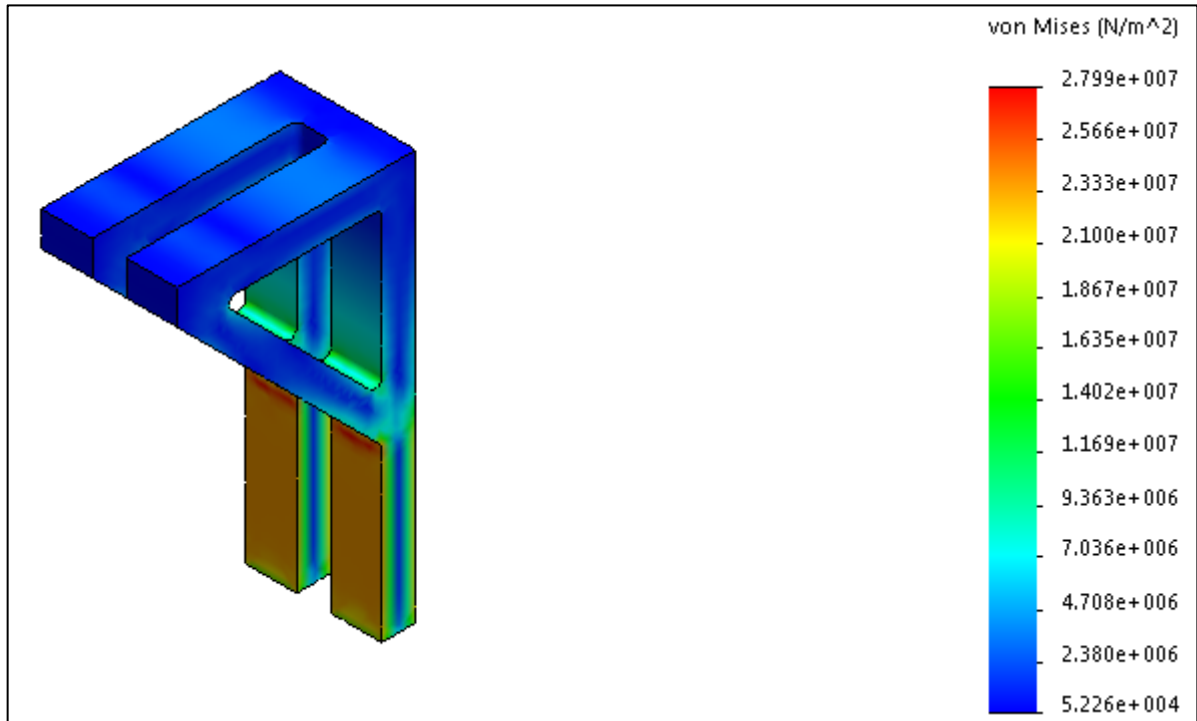


Figure 38 - Max stress FEA of pulley support #2

Figure 38 above shows the computed maximum stress for pulley support #2 based on FEA. The computed maximum stress is 27.9 MPa, far below the yield stress of the material of 435 MPa, resulting in a safety factor of 15.6. Even with the worst-case scenario of double loading, the pulley support is sufficiently strong for the design.

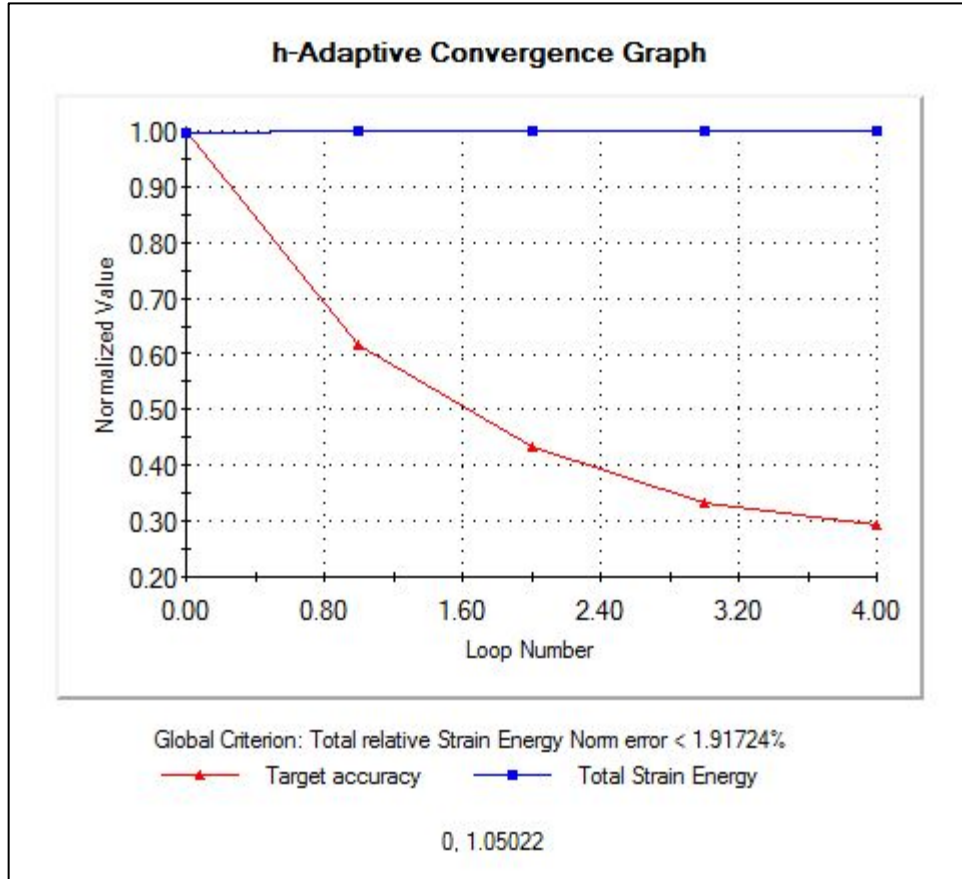


Figure 39 - h-Adaptive convergence graph for pulley support 2

Figure 39 shows the convergence of the computed FEA of within 2% error with respect to strain energy. As such, the convergence shows that the FEA is a good approximation of the maximum stress in pulley support #2.

A summary table of the stress analysis is located below in Table VIII. Note that these results are preliminary in nature, and should be verified prior to manufacturing. The results overall make sense as the loadings seen in each member are small, resulting in large safety factors. This is acceptable as the design weight was not a driving criterion and standard structural shapes were chosen based on manufacturer availability to reduce cost. The additional bulk of the design may also serve well as it may be bumped into on the shop floor and has the potential to withstand incidental impact.

Table VIII - STRESS ANALYSIS SUMMARY

Component	Math Model Stress [MPa]	CAD FEA Stress [MPa]	Yield Strength [MPa]	Resultant Safety Factor
Square Tube	19.16	52.8	435	22/8.2
L Member	25.56	14.1	435	17/30
Cable Attachment	27.72	50.76	435	15.7/8.6
Rocker Bar		20.6	435	21
Pulley Support 1		11.6	435	37.5
Pulley Support 2		27.9	435	15.6

### Critical Off the Shelf Components

Confirmation of off the shelf components with respect to their load capacity was based on manufacturer. The critical components investigated were the castors, hinges, pulleys, cables, and winch. Detailed below is the load each component will carry along with the specified load capacity. All off the shelf components and capacities in this section were selected from a single distributor, McMaster-Carr.

### Castors

Four castors are required to allow the design to be mobile. The castors are mounted on the bottom of the frame on the four corners as such they must be able to support the entire weight of the design. The total weight of the design was calculated to be 740 pounds. Due to the approximate symmetry of the frame we can approximate that each castor will support a quarter of the weight, 185 lbs. The bracket-mounted castors from McMaster-Carr were available in two capacities 100lbs and 500lbs [9]. As such the 500 lb. castor was selected.

### Door Hinges

Two door hinges were specified to attach the door to the rest of the support frame. The total weight of the door was calculated to be 146 lbs. As such the hinge selected has a load capacity of 150 lbs. [9] meaning that even if all the

force is applied to the single hinge it is strong enough to carry the load. Additionally the hinges we specified are lift off hinges. This allows the door to be easily lifted off the frame should the need arise.

**Cable**

Steel cable was specified to support the entire cutting and drive mechanisms. The total weight of the cutter and drive mechanisms is 141.1 lbs. The smallest diameter cable available on McMaster-Carr recommended for vertical lifting is 3/16” diameter cable. The load capacity of the cable is 600 lbs. [9], which is sufficiently strong to support the 141.1 lb. load.

**Pulleys**

Like the cable, the pulleys must support a 141.1 lb. load - the entire weight of the cutting and drive mechanism. Additionally the pulleys must be compatible with 3/16” diameter steel cable. The lowest load double channel pulley available from McMaster-Carr has a 600 lb. [8] capacity, and is suitable for the design. The lowest load rated single channel pulley for 3/16” diameter cable available from McMaster-Carr has a load capacity of 600 lbs. There for both pulleys are suitable for use with our design.

**Winch**

The winch must be rated to carry the entire weight of the cutting and drive mechanism, 141.1 lbs. The lowest rated winch compatible with 3/16” diameter steel cable is rated to 1,200 lbs. [8], and therefore is suitable for the design.

Table IX - OFF THE SHELF COMPONENT SUMMARY [8] [10]

Component	Estimated Max Load	Specified Load Capacity
Castors	740 lbs. over 4 castors	500 lbs. each
Hinges	146 lbs. over 2 hinges	150 lbs. each
3/16” Steel Cable	141.1 lbs.	600 lbs.
Double Channel Pulley	141.1 lbs.	600 lbs.
Single Channel Pulley	141.1 lbs.	600 lbs.
Winch	141.1 lbs.	1,200 lbs.

Table IX above is a summary of the load capacities and expected loads each component will carry. The load capacities of all specified components are substantially greater than the expected loads they will be required to support.

## Force required by worker to move and operate the device

There are two types of forces required to move and operate the device. A pushing or pulling force is required to move the device on the shop floor and a crank force is required to raise and lower the cutting and driving mechanism. Both required forces must not exceed the maximum force of 50lbs as outlined by Workplace Health and Safety Standards in Canada [11].

### Push Force

To calculate the required force to push the design, we must consider the rolling resistance of the castors on the smooth concrete floor at Winpak. The force required to push a load on castors is defined by the force required to overcome the rolling resistance, which is defined by the following formula [10]:

$$F = \frac{fW}{R}$$

*Where: f is the coefficient of rolling friction*

*W is the load per wheel*

*R is the wheel radius*

The coefficient of rolling resistance, f, is dependent on the material of the wheel and the material of the floor. In our case the castor wheel is made from polyurethane 85A as specified through McMaster-Carr and the floor is smooth cement. We were unable to find data on the coefficient of rolling resistance between these two materials; we were able to find data on the polyurethane 85A on a steel floor. The rolling resistance of polyurethane 85A on Steel was found to be .047 in. [10]. To compensate for this we will scale the coefficient of rolling resistance based on the relative smoothness of steel and concrete.

The surface roughness of concrete was found to be .3 mm [12] while the smoothness for steel was found to be .07mm [12]. Therefore the concrete surface is approximately 4.3 times rougher than the steel sheet. As such to provide a reasonable estimate for the coefficient of rolling friction is calculated to be 0.2021 in.

Calculating the force required to push the design continues as follows:

$$\text{Parameters: } f = 0.2021 \text{ in, } R = 2 \text{ in, } W = 185 \text{ lbs}$$

$$F = \frac{fW}{R} \rightarrow \frac{0.2021(185)}{2} \rightarrow 18.7 \text{ lbs}$$

The force required to overcome the rolling resistance of one castor is 18.7 lbs., as there are four castors supporting the frame a total force of 74.8 lbs. is required to move the entire design. This force is above the limit of allowable force for a single employee in the workplace. As such it is recommended that whenever the design is to be moved two workers move it as a team. This is likely the way the design would be maneuvered on the shop floor in practice, as the design is large and would be difficult for a single person to maneuver.

### Cranking Force Required

The hand winch used to raise and lower the drive and cutting mechanism is to be operated by a single worker. Therefore the mechanical advantage gained by the winch must be such that the force imparted on the crank handle is less than 50lbs to raise the weight of the cutting and drive mechanism, 141.1 lbs.

The ratios of the crank handle radius, small gear, and large gear can be shown as a simple gear train design shown below in Figure 40.

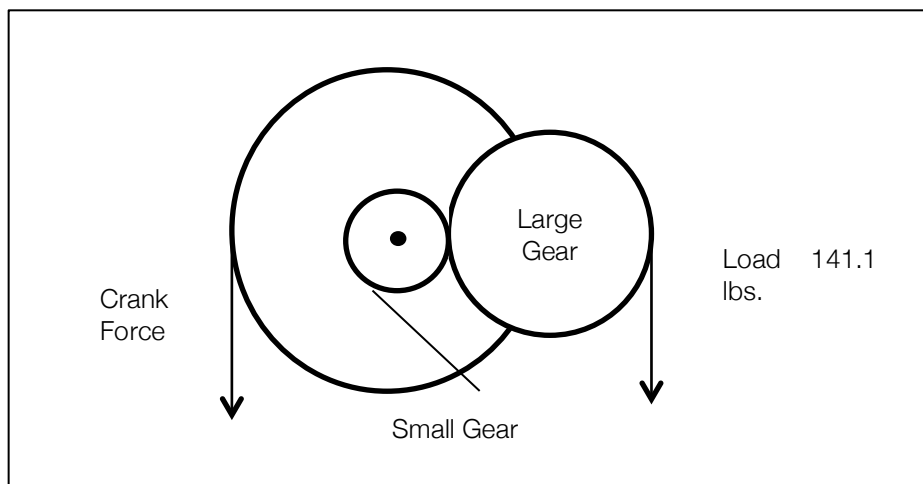


Figure 40 - Gear train approximation for hand winch

The crank force required can be calculated using the following formula:

$$F = \frac{r_s L}{C_l}$$

*Where:  $r_s$  is the small gear radius*



*L is the load*

*C<sub>l</sub> is the crank handle length*

Knowing the radius of the small gear is .846 in, the crank length is 7 inches and the load to be lifted is 141.1 lbs., the crank force required can be calculated as follows:

*Params: r<sub>s</sub> = .846 in, L = 141.1 lbs in and C<sub>l</sub> = 7in*

$$F = \frac{r_s L}{C_l} \rightarrow \frac{.846(141.1)}{7} \rightarrow 17.05 \text{ lbs}$$

The operator force required to raise the load is 17.05 lbs. well below the limit of 50 lbs. Therefore the required force meets the specification for a single operator.

Table X - SUMMARY OF OPERATOR FORCES REQUIRED

Operation	Required Force	Limit	Recommendation
Moving Device	74.8 lbs.	50 lbs.	Two people required to move device on shop floor
Operating Crank	17.05 lbs.	50 lbs.	One operator required

Table X above summarizes the required forces for both the operation of the winch and moving the device around the shop floor. It is recommended that two workers be used to move the device around the shop floor, due to both the size of the device and force required to move it. One worker can operate the winch.

## **2.4 DETAILED BILL OF MATERIALS**

The following sections will outline the materials required for the assembly of the design and the associated cost of the design. We will show that we were able to meet the budget requirements given to us by the client. Additionally we will show that the use of our design will save the client money and present a corresponding payback period.

Listed below in Table XI is the required list of the raw materials required for the assembly of the design. Notice that cost has not been included for the OLFA Blades as Winpak already has them in stock on the shop floor. Additionally only one 1'X1'X1/4" plate is required as it provides enough raw material to complete three separate parts; it is however listed three times. Any material that has an "M" designation in the description requires additional cutting, welding or tooling prior to installation. Due to minimum supply quantities, certain parts must be ordered in larger lots than required for the final design. For clarity the unit price for these types of materials have been designated as price/pkg. or price/length.

Kollmorgen, the manufacturer of the drive motor was contacted to provide an MSRP of the motor, however they were unable to provide the cost of the motor. As such the cost was conservatively approximated as being \$3000.00 dollars.

<u>Assembly</u>	<u>Sub Assembly</u>	<u>Part Number</u>	<u>Supplier</u>	<u>Description</u>	<u>Quantity</u>	<u>Unit Price</u>	<u>Total</u>
Sliding Car	Bumper Rail	A12238	Metals Depot	L 2"X2"X1/2" [M]	2'	14.10	14.10
Sliding Car	Bearing Assembly	6384K880	McMaster-Carr	Double Sealed Bearing	6	7.47	44.82
Sliding Car	Bearing Assembly	92415A760	McMaster-Carr	Zinc Plated Spacer	6	1.28	7.68
Sliding Car	Bearing Assembly	92012A571	McMaster-Carr	Hex Inset Shoulder Screw	6	7.44	44.64
Cutting Mechanism	Blade Holder	91251A191	McMaster-Carr	Hex Inset Screw 8-32 5/16"	100 (Req 3)	8.84 /pkg	8.84
Cutting Mechanism	Blade Holder	91251A194	McMaster-Carr	Hex Inset Screw 8-32 1/2"	100 (Req 1)	10.08 /pkg	10.08
Cutting Mechanism	Blade Holder	OLFA BLADE	OLFA SKB-2	Cutting Knife	2	0	0
Cutting Mechanism	Blade Holder	P114	Metals Depot	1' X 1' X 1/4" Plate [M]	1	17.86	17.86
Cutting Mechanism	Rocker Bar	P114	Metals Depot	1' X 1' X 1/4" Plate [M]	1	0.00	0.00
Sliding Car	Slicer Base Plate	P114	Metals Depot	1' X 1' X 1/4" Plate [M]	1	0.00	0.00
Sliding Car	Slicer Base Plate	F21234	Metals Depot	1/2" X 3/4" Bar [M]	2'	5.76	5.76
Cutting Mechanism	Guide Rod	S112	Metals Depot	1'X2'X.105" (12GA) Sheet [M]	1	18.20	18.20
Cutting Mechanism		92012A506	McMaster-Carr	Hex Inset Shoulder Screw 4-40 1/8"	1	6.67	6.67
		90480A005	McMaster-Carr	Hex Nut 4-40 1/4" X 3/32"	100 (Req 2)	0.87 /pkg	0.87

		91102A735	McMaster-Carr	Split Lock Washer	100 (Req 1)	0.87 /pkg	0.87
		90126A509	McMaster-Carr	Flat Washer	100 (Req 2)	1.35 /pkg	1.35
		91259A155	McMaster-Carr	Hex Insert Shoulder Screw 4-40 1/8"	1	1.88	1.88
Frame	Support Member	T12211	Metals Depot	2"X2"X1/8" Square Tube 6' [M]	4	30.36	121.44
	Support Member	T12211	Metals Depot	2"X2"X1/8" Square Tube 8' [M]	4	40.48	161.92
	Support Member	T12211	Metals Depot	2"X2"X1/8" Square Tube 12' [M]	8	50.64	529.52
	Protective Screening	9302T631	McMaster-Carr	4'X8' Sheet .28"X1" Opening [M]	7	87.89	615.23
	Support Member	A12218	Metals Depot	L 2"X2"X1/8" [M]	12'	16.32/6'	32.64
	Castors	87875T55	McMaster-Carr	Castors	4	172.64	690.56
	Door Hinge	1849A91	McMaster-Carr	Hinge	2	16.40	32.80
Drive Assembly		98535A140	McMaster-Carr	3/16" x 3/16" Key	2	2.77	5.54
		2737T264	McMaster-Carr	Sprocket 1/4" Pitch, 36 Teeth	2	19.81	39.62
		5908K190	McMaster-Carr	Steel Bearing 3/4"	1	40.39	40.39
		90108A033	McMaster-Carr	Steel Flat Washer 1/2"	50 (Req 8)	6.67 /pkg	6.67
		90591A171	McMaster-Carr	Hex Nut M10x1.5	100 (Req 10)	10.48 /pkg	10.48
		90965A200	McMaster-Carr	Flat Washer M10	50 (Req 10)	9.59 /pkg	9.59
		91280A626	McMaster-Carr	Cap Screw M10x1.5 20mm	25 (Req 4)	8.1 /pkg	8.1
		91280A636	McMaster-Carr	Cap Screw M10x1.5 35mm	25 (Req 6)	9.74 /pkg	9.74
		92620A637	McMaster-Carr	Cap Screw 3/8"-16, 1-1/8"	20 (Req 16)	29.38 /pkg	29.38
		92620A714	McMaster-Carr	Cap Screw 1/2"-13, 1-1/4"	25 (Req 8)	14.67 /pkg	14.67
		94895A031	McMaster-Carr	Hex Nut 3/8"-16	100 (Req 16)	7.65 /pkg	7.65
		94895A823	McMaster-Carr	Hex Nut 1/2"-13	50 (Req 8)	7.82 /pkg	7.82
	Motor	MH172	Kollmorgen	Stepper Motor	1	3000.00	3000.00
	Chain	6261K108	McMaster-Carr	ANSI 25 1/4" Pitch Chain	18"	5.14	92.52
	Chain	6261K106	McMaster-Carr	ANSI 25 1/4" Pitch Chain Add Link	2	0.62	1.24
Lifting Mechanism	Steel Rope	3440T55	McMaster-Carr	3/16" Steel Rope [M]	25'	1.18 /Foot	29.50
	Rail Members	A12218	Metals Depot	L 2"X2"X1/8" [M]	20'	20.40 /10'	40.80

	Winch	3644T53	McMaster-Carr	Hand Winch for 3/16" Steel Rope	1	36.87	36.87
	Double Pulley	3099T44	McMaster-Carr	Double Pulley 3/16" Steel Rope	1	11.97	11.97
	Single Pulley	3099T34	McMaster-Carr	Single Pulley 3/16" Steel Rope	1	6.67	6.67
	Support Member	P112	Metals Depot	1'X2'X3/4" Steel Plate [M]	1	101.06	101.06
	Winch Support	P134	Metals Depot	1'X2'X1/2" Steel Plate [M]	1	55.12	55.12
Misc.	Clamp	5037A52	McMaster-Carr	12" C-Clamp	2	39.31	78.62
						<b>Total</b>	6011.75

Table XI- BILL OF MATERIALS [4] [5]

## 2.5 DETAILED DESIGN COST

The cost of our design consists of two main components: raw materials and labor. The cost of the raw materials is provided in Table XI above. Summing the values in the total column we noted that the total cost of the required raw material is \$5933.13. The assembly labor cost is more difficult to account for, as manufacturing time will vary depending on the skill of the workers. We feel that a good conservative estimate would be 40 man-hours, or one standard working week for one worker. Assuming the laborer works at a contract rate of \$100/hr., the assembly cost would be \$4000.00.

Table XII - COST OF DESIGN

Component	Cost
Raw Material	\$6,011.75
Assembly Labor	\$4,000.00
Total	\$10,011.75

Table XII above summarizes the approximate total cost of the design including assembly. We were originally given a target of \$25,000 - \$50,000 dollars by the client. We feel that we have been very successful in meeting that goal with a total cost of \$10,011.75. Note that this is an approximation with respect to the labor cost. Total assembly cost will likely be lower if resources from Winpak are

used for manufacturing. We did not account for shipping costs, though we are confident that they will represent a very minor part of the final cost.

## 2.6 COST SAVINGS

We can approximate the time saved using our design in terms of the time required to trim a full roll based on information provided by the client. At most, the required time to cut back a full roll using the current process is 30 minutes, using two employees. Based on our single pass cut depth of 4mm and assuming a full roll with thickness of 417.5 mm, along with a 2.54 meter pass length and the specified pass speed of .4 m/s the time required to trim back the roll is calculated as follows.

$$t = \frac{t_R}{c} \frac{L}{V60}$$

*Where:  $t_R$  is roll thickness*

*$c$  is cut depth*

*$L$  is pass length*

*$V$  is pass speed*

*$t$  is the total time*

Using the above equation the time required is calculated to be:

$$t = \frac{t_R}{c} \frac{L}{V 60} \rightarrow \frac{417.5(2.54)}{4(.4)(60)} \rightarrow 11.04 \text{ min} \sim 11 \text{ min}$$

Assuming the workers are being paid just above minimum wage at \$12/hour, the current cost to cut the roll is \$12. Using a single worker to cut the same full size roll with our design the cost is \$2.20. Thus, the total labor savings per full roll is \$9.80. Based on this estimate, the number of full rolls that would need to be processed to payback the initial investment is 1022 rolls. Assuming there is one full roll to cut back per week the payback period is approximately 19.7 years. While this payback period may seem large it does not consider additional cost savings that would result from the implementation of this design, such as reduced time off work due employee injury and increased productivity as a result of increased moral.

Table XIII below provides a summary of the cost analysis and findings.

Initial Budget	\$25,000 - \$50,000
Total Material and Assembly Cost	\$10,011.75
Savings per Roll Processed	\$9.80
Initial Time to Process Full Roll	30 min
Time to Process Roll with Design	11 min
Approximate Payback Period	19.7 Years

Table XIII - COST ANALYSIS SUMMARY

## **3. CONCLUSIONS & SUMMARY**

### **3.1 FINAL DESIGN SUMMARY**

Our final design was developed according to the specifications introduced in the project definition report and the feedback obtained from the client through the concept report. A budget of \$25 000 - \$50 000 was provided by the client to manufacture the design. The final cost, including purchasing, assembling and machining of the design parts is just over \$10 000. The time required to cut a full roll of 1-meter diameter is 11 minutes using our design. This is substantially quicker than the current method, which requires 30 minutes. A force of 17 pounds is required to turn the winch in order to operate the vertical motion of the design. This is less than half the maximum force of 50 pounds for safety standards. A force of 75 pounds is required to push the design on the shop floor; therefore two people are required to safely move the design to a desired location. The design uses the same cutting blades as were used in the current method, as such no excess particulates will be produced in the process. Steel safety screens were mounted onto the open sections of the design to prevent any contact of the operator with sharp edges or moving parts. These screens have sufficiently small meshing so that fingers cannot go through. The design is easily moved due to the 4" caster wheels mounted at each corner of the frame. Operators using the design are no longer required to bend since the design was made ergonomically, enabling the operator to be upright in a comfortable position. Any roll size up to 1 meter in diameter can be cut using this design.

### 3.2 FINAL DESIGN VS. PROJECT OBJECTIVES & USER REQUIREMENTS

The overall results of our project are summarized as follows:

Table XIV: CONCLUSION SUMMARY TABLE

Need	Target	Result
Cost	\$25 000 – \$50 0000	\$10 011.75
Cutting time for full roll	Less than 30mins	11 mins
Force required for operation	50lbf	17lbf
Force required to move	50lbf	75lbf (2 people required)
Cleanliness	No fine particulates	Clean cut on straight path
Safety	Operator is protected from sharp edges and moving parts	Steel safety screen with small openings to prevent fingers going through
Mobility	Design must be easily moved	4" caster wheels installed on frame
Ease of use	Can be operated comfortably with minimal force	Operator no longer required to bend over for extended periods
Accepts all roll sizes	Max roll diameter of 1m	Accepts rolls up to 1m diameter

Given these results our team is confident that our design will exceed the client's needs. All of the target values set by the client were achieved or exceeded.



## 4. WORKS CITED

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## 5. APPENDIX A: GANTT CHART

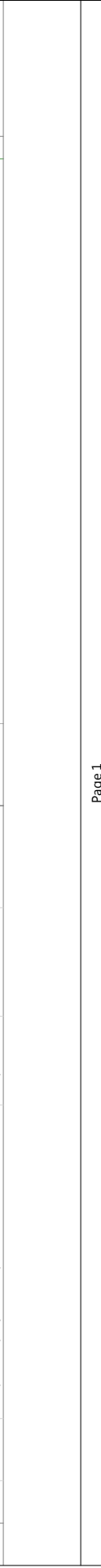
From the first phase of the project we recognized the need to have a schedule of all the tasks to be completed in order to be able to achieve our target of completing all deliverables by December 2<sup>nd</sup> 2014. Within this context this Gantt chart has been followed and updated all along. From the 2<sup>nd</sup> report, we have updated the Gantt chart for the sections pertaining to the final design report (FDR) as well as the final oral presentation and poster.

These three items had to progress in parallel as opposed to being in sequence. This is due to the limited time available from the time we finished designing and modeling our concept picked from the 2<sup>nd</sup> phase.

As soon as the team split the tasks for the individual written report sections, the concept for the poster was underway and the PowerPoint was in construction as well. The team feels that this method was very efficient as it allowed us to achieve our target without major modifications to the initial plan.

Notes: In terms of completion, by the time of submission of this report on December 1<sup>st</sup> 2014, all the items are 100% complete except for the oral presentation and poster presentations held December 2<sup>nd</sup> 2014. This is why the completion bar is not full on the "Capstone Project" item bar as this occurrence accounts for the entire project.

ID	Task Mode	Task Name	Duration	Start	Finish
1	✓	<b>Engineering Design Capstone project</b>	<b>57 days</b>	<b>Mon 14-09-15</b>	<b>Tue 14-12-02</b>
2	✓	phase 1: project definition	11 days	Mon 14-09-15	Mon 14-09-29
3	✓	develop initial Gantt chart	7 days	Mon 14-09-15	Tue 14-09-23
4	✓	prepare meeting minutes for week 1	7 days	Mon 14-09-15	Tue 14-09-23
5	✓	<b>PDR report</b>	<b>8 days</b>	<b>Thu 14-09-18</b>	<b>Mon 14-09-29</b>
6	✓	prepare initial project documents	2 days	Thu 14-09-18	Fri 14-09-19
7	✓	complete individual report sections	5 days	Fri 14-09-19	Thu 14-09-25
8	✓	compile PDR	3 days	Thu 14-09-25	Sun 14-09-28
9	✓	submit PDR	2 days	Sun 14-09-28	Mon 14-09-29
10	✓	PDR complete	0 days	Mon 14-09-29	Mon 14-09-29
11	✓	<b>Oral presentation</b>	<b>4 days</b>	<b>Wed 14-09-24</b>	<b>Mon 14-09-29</b>
12	✓	plan slides	2 days	Wed 14-09-24	Thu 14-09-25
13	✓	create slides	2 days	Thu 14-09-25	Fri 14-09-26
14	✓	create presentation notes	2 days	Thu 14-09-25	Fri 14-09-26
15	✓	practice Oral Presentation	2 days	Fri 14-09-26	Mon 14-09-29
16	✓	<b>CDR report</b>	<b>16 days</b>	<b>Fri 14-10-03</b>	<b>Fri 14-10-24</b>
17	✓	develop detailed design concept	5 days	Fri 14-10-03	Thu 14-10-09
18	✓	develop concepts & scoring matrix	3 days	Fri 14-10-03	Tue 14-10-07
19	✓	prepare details of selected design	3 days	Tue 14-10-07	Thu 14-10-09
20	✓	create CAD drawings	4 days	Thu 14-10-09	Tue 14-10-14
21	✓	perform preliminary analysis	4 days	Fri 14-10-10	Wed 14-10-15
22	✓	write executive summary	1 day	Tue 14-10-14	Tue 14-10-14
23	✓	write individual report sections	6 days	Tue 14-10-14	Tue 14-10-21
24	✓	compile CDR	3 days	Wed 14-10-22	Fri 14-10-24
25	✓	print & submit CDR	0 days	Fri 14-10-24	Fri 14-10-24
26	✓	CDR complete	0 days	Fri 14-10-24	Fri 14-10-24
27	✓	meeting minutes october	0 days	Wed 14-10-29	Wed 14-10-29
28	✓	draft report for peer editing	19 days	Wed 14-10-29	Mon 14-11-24
29	✓	complete peer editing	3 days	Mon 14-11-24	Wed 14-11-26
30	✓	meeting minutes for november	4 days	Mon 14-11-24	Thu 14-11-27
31	✓	<b>FDR report</b>	<b>4 days</b>	<b>Wed 14-11-26</b>	<b>Mon 14-12-01</b>
32	✓	assign individual tasks	1 day	Wed 14-11-26	Wed 14-11-26
33	✓	complete individual write-up sections	3 days	Thu 14-11-27	Mon 14-12-01
34	✓	compile and master edit	1 day	Mon 14-12-01	Mon 14-12-01
35	✓	print & submit FDR	1 day	Mon 14-12-01	Mon 14-12-01
36	✓	<b>poster creation</b>	<b>7 days</b>	<b>Mon 14-11-24</b>	<b>Tue 14-12-02</b>
37	✓	compile visuals	4 days	Mon 14-11-24	Thu 14-11-27
38	✓	plan space usage on poster (3x4 ft)	1 day	Mon 14-11-24	Mon 14-11-24
39	✓	place items on poster	2 days	Tue 14-11-25	Wed 14-11-26
40	✓	review and edit	2 days	Thu 14-11-27	Fri 14-11-28
41	✓	test print poster	1 day	Mon 14-12-01	Mon 14-12-01
42	✓	make changes and final print	1 day	Mon 14-12-01	Mon 14-12-01
43	✓	<b>Final Oral Presentation</b>	<b>7 days</b>	<b>Mon 14-11-24</b>	<b>Tue 14-12-02</b>
44	✓	plan slide sequence	1 day	Mon 14-11-24	Mon 14-11-24
45	✓	compile visuals from CAD model	2 days	Tue 14-11-25	Wed 14-11-26
46	✓	dedicate powerpoint creation to a team	1 day	Tue 14-11-25	Tue 14-11-25
47	✓	compile information from FDR report	2 days	Thu 14-11-27	Fri 14-11-28
48	✓	rehearse for presentation	2 days	Mon 14-12-01	Tue 14-12-02
49	✓	deliver oral defense	1 day	Tue 14-12-02	Tue 14-12-02
50	✓	Capstone project complete	0 days	Tue 14-12-02	Tue 14-12-02

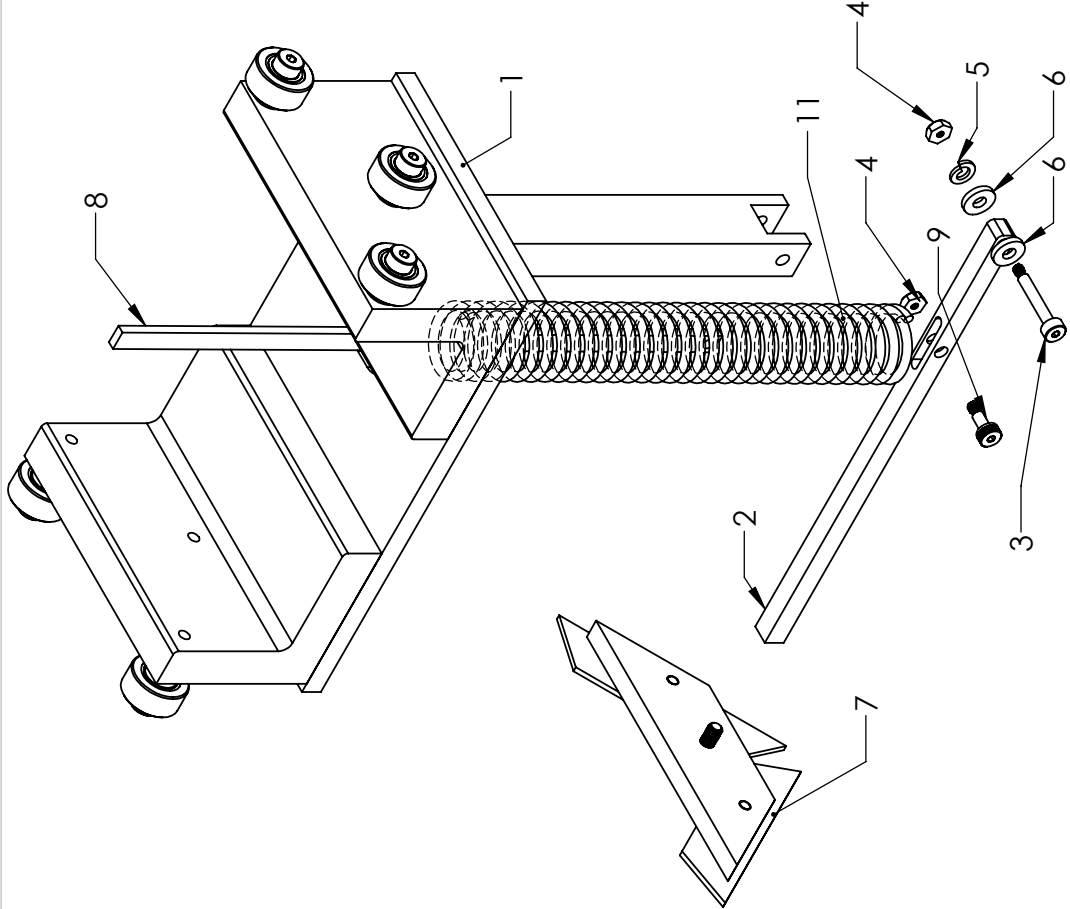


## 6 APPENDIX B – MACHINE DRAWINGS

### 6.1 LIST OF DRAWINGS

Drawing 1: Cutting Mechanism Assembly Drawings.....	IV
Drawing 2: Sliding Car Assembly Drawing.....	V
Drawing 3: Sliding Car Assembly Drawing.....	VI
Drawing 4: Slicer Base Plate.....	VII
Drawing 5: Rocker Bar.....	VIII
Drawing 6: Guide Rod.....	IX
Drawing 7: Bearing Assembly.....	X
Drawing 8: Bumper Rail.....	XI
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Drawing 13: Final Full Assembly.....	XVI
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Detailed drawings for purchasing, manufacturing and assembly of the slider car mechanism are provided below. These drawings indicate the dimensions of relevant features for manufacturing of custom parts, as well as the assembly order and arrangement of all parts. The design was subdivided into several subassemblies in order to facilitate the work within the manufacturing shop used, as well as to create a computer model that is easy to work with for future revision of the design. For many standard fasteners, bearings etc., CAD models were obtained via McMaster Carr [6] and these parts specified in assembly drawings. These parts are to be used, or replaced with an equivalent alternative from another supplier at the discretion of the client.



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Sliding Car		1
2	Rocker Bar		1
3	92012A506	McMaster Carr Part no.	1
4	90480A005	McMaster Carr Part no.	2
5	91102A735	McMaster Carr Part no.	1
6	90126A509	McMaster Carr Part no.	2
7	Blade Holder		1
8	Guide Rod		1
9	91259A155	McMaster Carr Part no.	1
11	9657K212	McMaster Carr Part no.	1

UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES		
TOLERANCES:		
FRACTIONAL: ±		
ANGULAR: MACH: ± BEND ±		
TWO PLACE DECIMAL ±		
THREE PLACE DECIMAL ±		
INTERPRET GEOMETRIC TOLERANCING PER:		
MATERIAL		
FINISH		
DO NOT SCALE DRAWING		

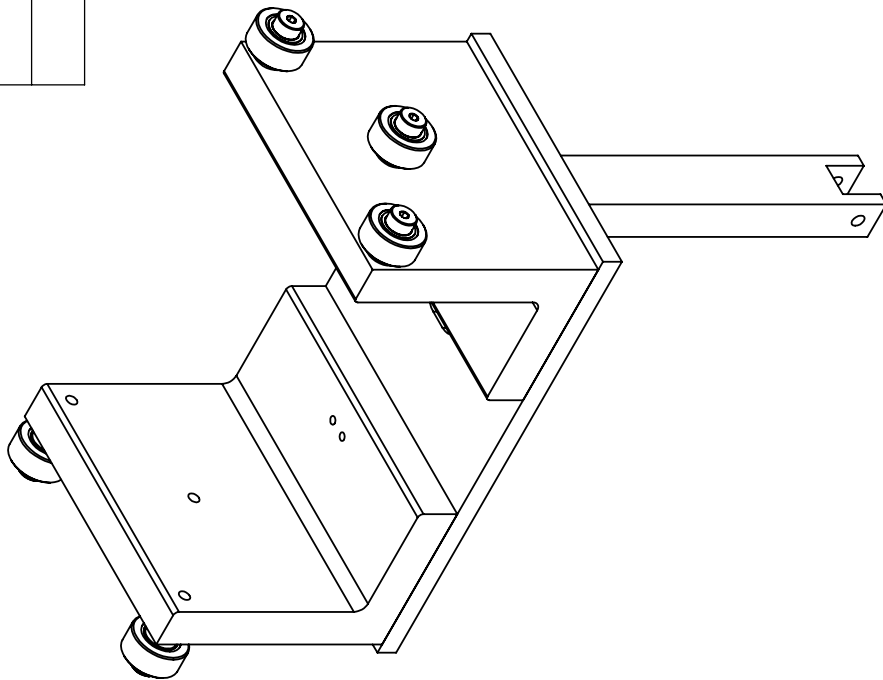
TITLE:  
**CUTTING MECHANISM  
 ASSEMBLY DRAWING**

SIZE DWG. NO. REV  
**A**

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

**PROPRIETARY AND CONFIDENTIAL**  
 THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF <INSERT COMPANY NAME HERE>. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF <INSERT COMPANY NAME HERE> IS PROHIBITED.

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Slicer Base Plate		1
2	Slider Rail		2

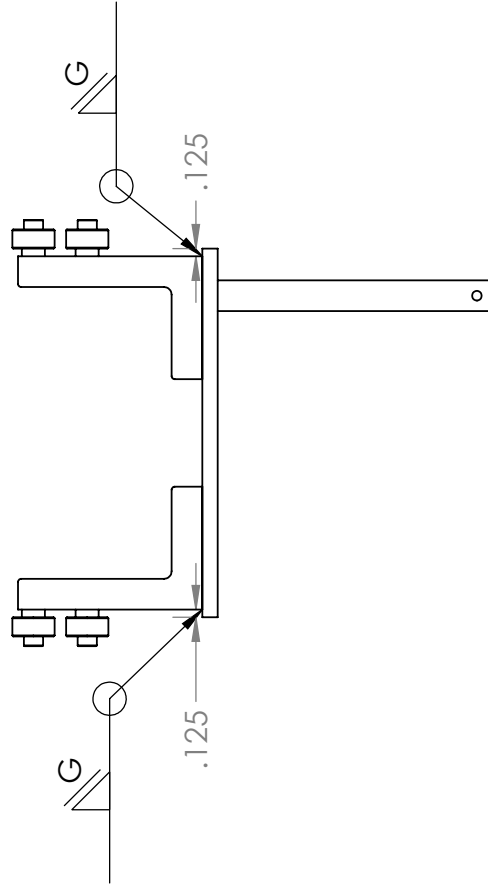
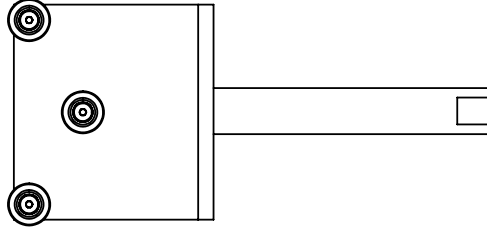
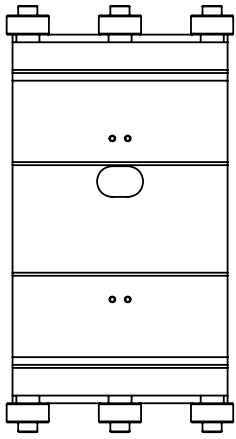


UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES		DRAWN	
TOLERANCES:		CHECKED	
ANGULAR: MACH+ 0.5		ENG APPR.	
TWO PLACE DECIMAL ± 0.01		MFG APPR.	
THREE PLACE DECIMAL ± 0.005		G.A.	
MATERIAL		COMMENTS:	
A513 STEEL, FASTENERS AS SUPPLIED			
FINISH			
AS-MACHINED			
DO NOT SCALE DRAWING			

TITLE:  
**SLIDING CAR ASSEMBLY  
DRAWING**

SIZE DWG. NO. REV  
**A**  
SCALE: 1:2 WEIGHT: SHEET 1 OF 2

**PROPRIETARY AND CONFIDENTIAL**  
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UNLESS OTHERWISE SPECIFIED:	DRAWN	NAME	DATE
DIMENSIONS ARE IN INCHES	CHECKED		
TOLERANCES:	ENG APPR.		
ANGULAR: MACH+ 0.5	MFG APPR.		
TWO PLACE DECIMAL ± 0.01	G.A.		
THREE PLACE DECIMAL ± 0.005	COMMENTS:		
	MATERIAL		
	A513 STEEL, FASTENERS AS SUPPLIED		
	FINISH		
	AS-MACHINED		
	DO NOT SCALE DRAWING		
	APPLICATION		
	NEXT ASSY		
	USED ON		

TITLE:

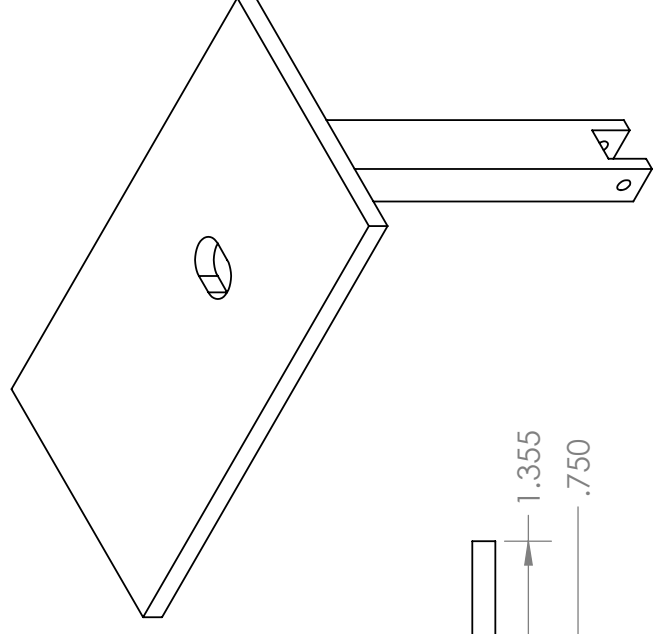
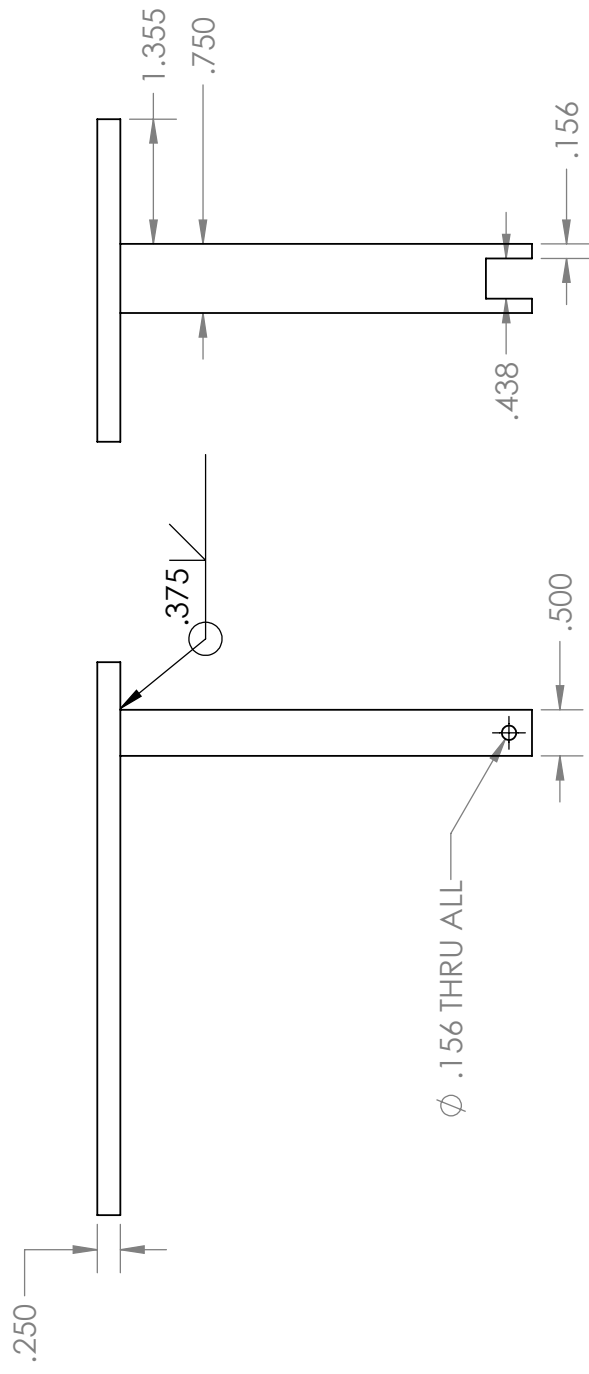
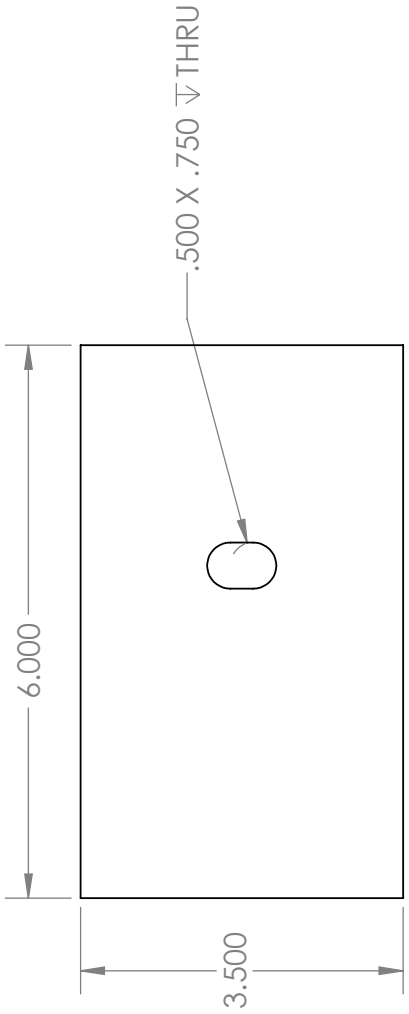
# SLIDING CAR ASSEMBLY DRAWING

SIZE DWG. NO. REV

**A**

SCALE: 1:2 WEIGHT: SHEET 2 OF 2

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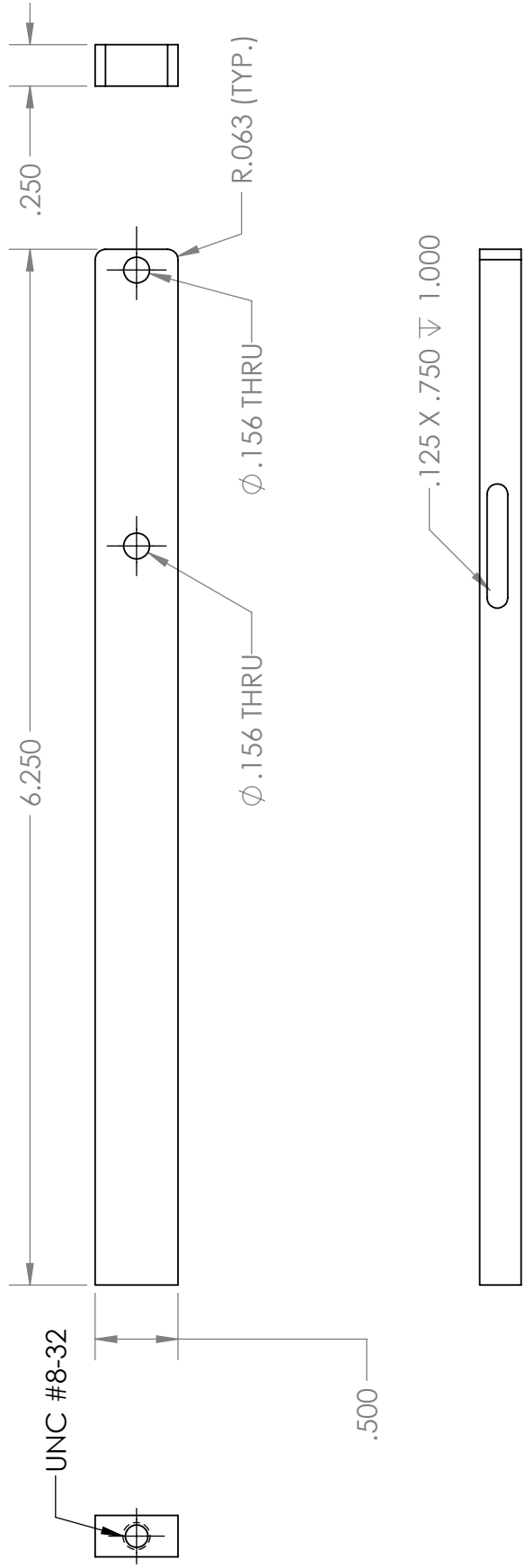
UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES		
TOLERANCES:	DRAWN	
ANGULAR: MACH+ 0.5	CHECKED	
TWO PLACE DECIMAL ± 0.01	ENG APPR.	
THREE PLACE DECIMAL ± 0.005	MFG APPR.	
	G.A.	
	COMMENTS:	
MATERIAL	A513 STEEL, FASTENERS AS SUPPLIED	
FINISH	AS-MACHINED	
USED ON	DO NOT SCALE DRAWING	
APPLICATION		

TITLE: **SLICER BASE PLATE**

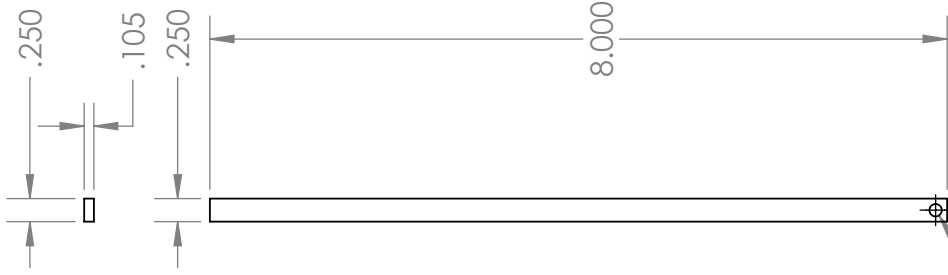
SIZE **A** DWG. NO. REV

SCALE: 1:2 WEIGHT: SHEET 1 OF 1





UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES		DRAWN	
TOLERANCES:		CHECKED	
ANGULAR: MACH ± 0.5		ENG APPR.	
TWO PLACE DECIMAL ± 0.01		MFG APPR.	
THREE PLACE DECIMAL ± 0.005		G.A.	
COMMENTS:		TITLE: <b>ROCKER BAR</b>	
MATERIAL		SIZE	DWG. NO.
A513 STEEL, FASTENERS AS SUPPLIED		<b>A</b>	REV
FINISH		SCALE: 1:1 WEIGHT: SHEET 1 OF 1	
AS-MACHINED			
DO NOT SCALE DRAWING			
APPLICATION			
NEXT ASSY	USED ON		
<p><b>PROPRIETARY AND CONFIDENTIAL</b>          THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF WINPAK. ANY REPRODUCTION IN PART OR AS A WHOLE WITHOUT THE WRITTEN PERMISSION OF WINPAK IS PROHIBITED.</p>			



Ø .136 THRU

UNLESS OTHERWISE SPECIFIED:	DRAWN	NAME	DATE
DIMENSIONS ARE IN INCHES	CHECKED		
TOLERANCES:	ENG APPR.		
ANGULAR: MACH+ 0.5	MFG APPR.		
TWO PLACE DECIMAL ± 0.01	G.A.		
THREE PLACE DECIMAL ± 0.005	COMMENTS:		
	MATERIAL		
	A513 STEEL, FASTENERS AS SUPPLIED		
	FINISH		
	AS-MACHINED		
	DO NOT SCALE DRAWING		
	APPLICATION		
	NEXT ASSY		
	USED ON		

TITLE:

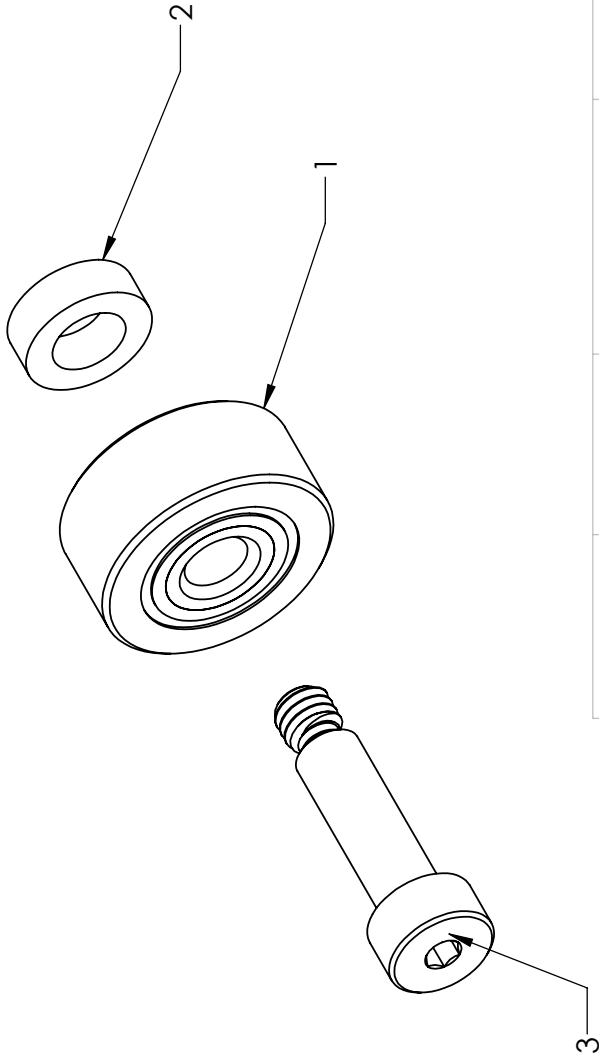
# GUIDE ROD

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SIZE	DWG. NO.	REV
<b>A</b>		

SCALE: 1:5	WEIGHT:	SHEET 1 OF 1
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ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	6384K880	McMaster Carr Part no.	1
2	92415A760	McMaster Carr Part no.	1
3	92012A571	McMaster Carr Part no.	1



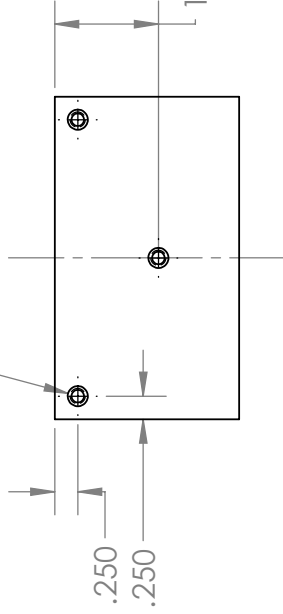
UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES		
TOLERANCES:	DRAWN	
ANGULAR: MACH ± 0.5	CHECKED	
TWO PLACE DECIMAL ± 0.01	ENG APPR.	
THREE PLACE DECIMAL ± 0.005	MFG APPR.	
	G.A.	
	COMMENTS:	
	MATERIAL	
	A513 STEEL, FASTENERS AS SUPPLIED	
	FINISH	
	AS-MACHINED	
	USED ON	
	APPLICATION	
	DO NOT SCALE DRAWING	

# BEARING ASSEMBLY

SIZE	DWG. NO.	REV
<b>A</b>		
SCALE: 2:1	WEIGHT:	SHEET 1 OF 1

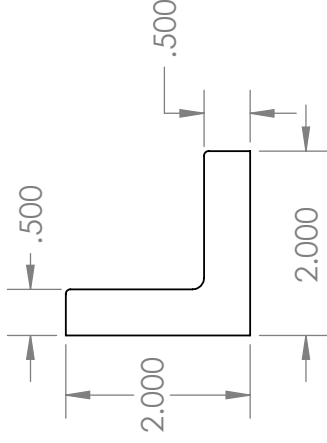
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3X Ø .219 ± .188

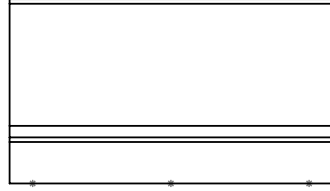
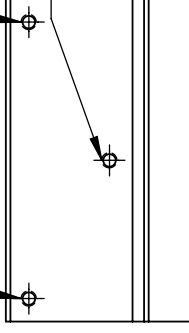


#8-36 Machine Threads

#8-36 Machine Threads



#8-36 Machine Threads



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UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES		
TOLERANCES:	DRAWN	
FRACTIONAL ±	CHECKED	
ANGULAR: MACH ± BEND ±	ENG APPR.	
TWO PLACE DECIMAL ±	MFG APPR.	
THREE PLACE DECIMAL ±	G.A.	
INTERPRET GEOMETRIC TOLERANCING PER:	COMMENTS:	
MATERIAL		
FINISH		
USED ON		
APPLICATION		
DO NOT SCALE DRAWING		

TITLE:

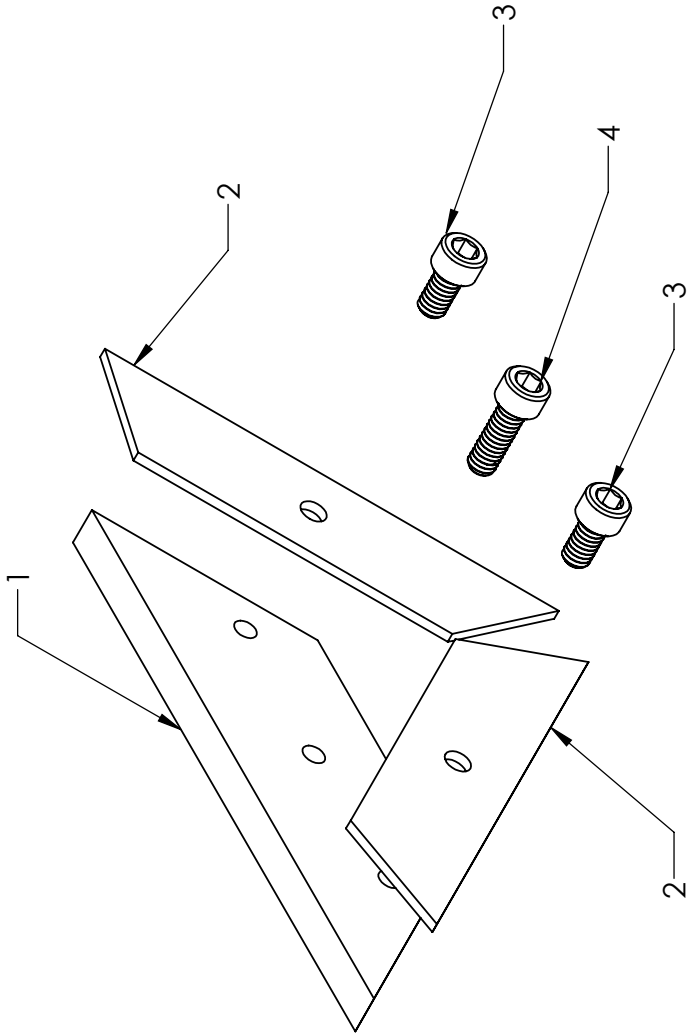
# BUMPER RAIL

SIZE DWG. NO. REV

**A**

SCALE: 1:2 WEIGHT: SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Blade Holder Plate		1
2	OLFA Blade	OLFA SKB-2 Blade	2
3	91251A191	McMaster Carr Part no.	2
4	91251A194	McMaster Carr Part no.	1



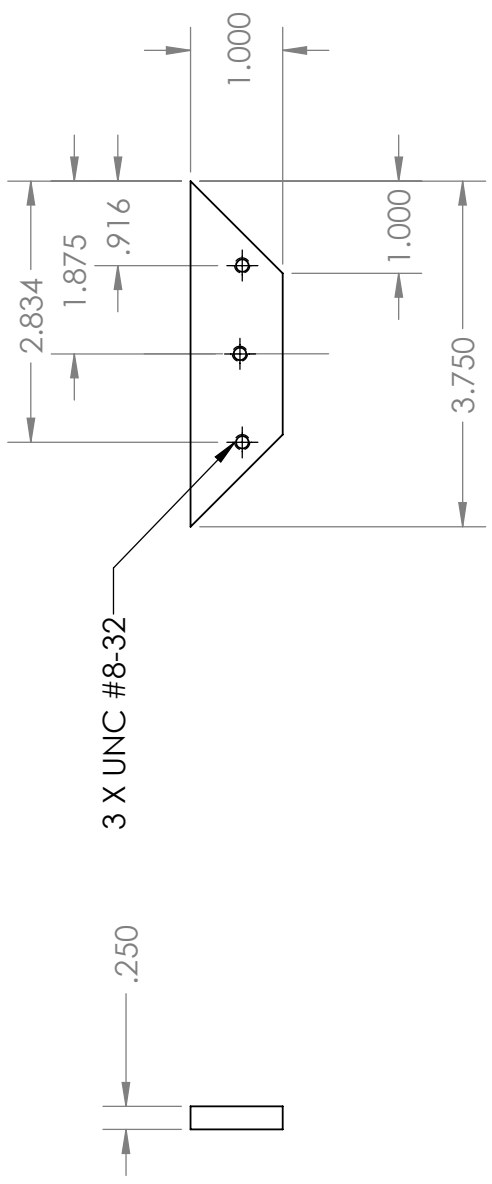
UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES		
TOLERANCES:	DRAWN	
ANGULAR: MACH ± 0.5	CHECKED	
TWO PLACE DECIMAL ± 0.01	ENG APPR.	
THREE PLACE DECIMAL ± 0.005	MFG APPR.	
	G.A.	
	COMMENTS:	
	MATERIAL	
	A513 STEEL, FASTENERS AS SUPPLIED	
	FINISH	
	AS-MACHINED	
	USED ON	
	APPLICATION	
	DO NOT SCALE DRAWING	

TITLE:  
**BLADE HOLDER ASSEMBLY**

SIZE DWG. NO. REV  
**A**

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

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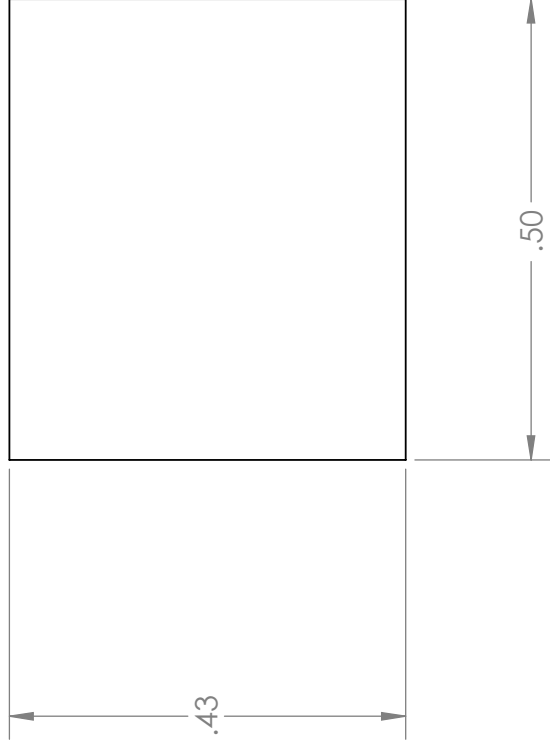
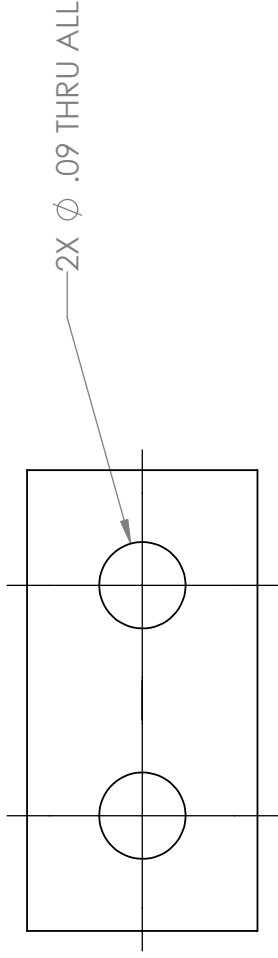


UNLESS OTHERWISE SPECIFIED:		NAME	DATE
DIMENSIONS ARE IN INCHES		DRAWN	
TOLERANCES:		CHECKED	
ANGULAR: MACH ± 0.5		ENG APPR.	
TWO PLACE DECIMAL ± 0.01		MFG APPR.	
THREE PLACE DECIMAL ± 0.005		G.A.	
MATERIAL		COMMENTS:	
A513 STEEL, FASTENERS AS SUPPLIED			
FINISH			
AS-MACHINED			
DO NOT SCALE DRAWING			
APPLICATION			
NEXT ASSY	USED ON		

TITLE:  
**BLADE HOLDER**

SIZE **A** DWG. NO. REV  
SCALE: 1:2 WEIGHT: SHEET 1 OF 1

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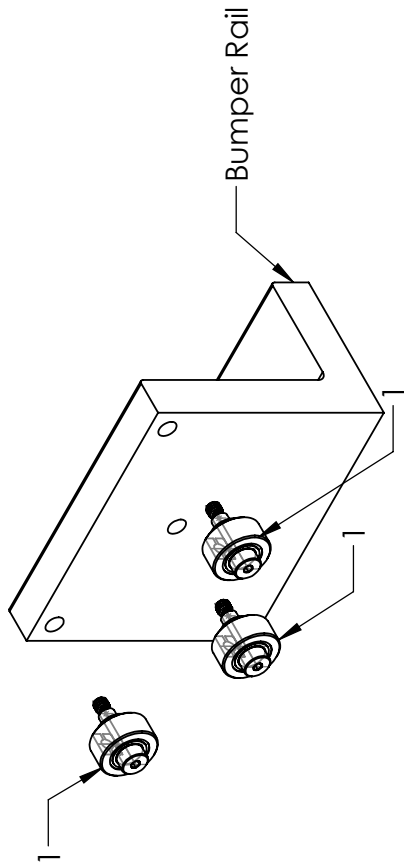
UNLESS OTHERWISE SPECIFIED:	DRAWN	NAME	DATE
DIMENSIONS ARE IN INCHES	CHECKED		
TOLERANCES:	ENG APPR.		
ANGULAR: MACH ± 0.5	MFG APPR.		
TWO PLACE DECIMAL ± 0.01	G.A.		
THREE PLACE DECIMAL ± 0.005	COMMENTS:		
	MATERIAL		
	A513 STEEL, FASTENERS AS SUPPLIED		
	FINISH		
	AS-MACHINED		
	DO NOT SCALE DRAWING		
	USED ON		
	APPLICATION		

TITLE:  
**CHAIN SUPPORT**

SIZE **A** DWG. NO. REV

SCALE: 5:1 WEIGHT: SHEET 1 OF 1

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Bearing Assembly		3
2	Bumper Rail		1



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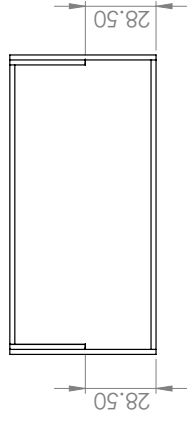
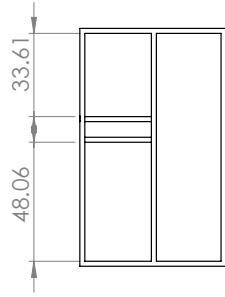
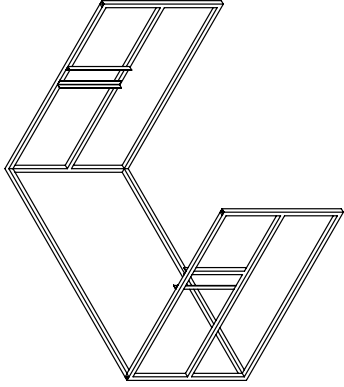
UNLESS OTHERWISE SPECIFIED:	NAME	DATE
DIMENSIONS ARE IN INCHES		
TOLERANCES:	DRAWN	
ANGULAR: MACH ± 0.5	CHECKED	
TWO PLACE DECIMAL ± 0.01	ENG APPR.	
THREE PLACE DECIMAL ± 0.005	MFG APPR.	
	G.A.	
	COMMENTS:	
MATERIAL		
A513 STEEL, FASTENERS AS SUPPLIED		
FINISH		
AS-MACHINED		
USED ON		
APPLICATION		
DO NOT SCALE DRAWING		

# SLIDER RAIL ASSEMBLY

TITLE: SLIDER RAIL ASSEMBLY  
 SIZE: **A**  
 DWG. NO.:  
 REV:  
 SCALE: 1:2 WEIGHT: SHEET 1 OF 1

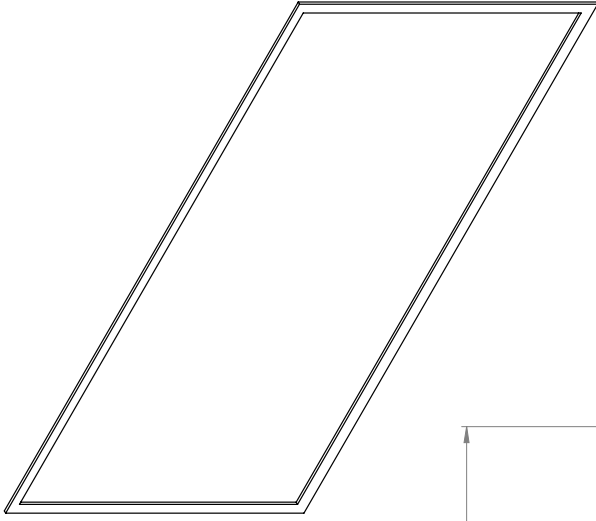






UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS		FINISH:		USE UP AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
SURFACE FINISH:		TOLERANCES:		MATERIAL:		TITLE:			
MATERIAL:		WEIGHT:		SCALE: 1:50		SHEET 1 OF 1		A3	
NAME		SIGNATURE		DATE		DRAWN			
DRAWN		CHECKED		APPROVED		MFG		Q.A.	

# Rail & Frame



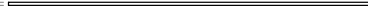
112.25

55.00

59.00

116.25

0.50



UNLESS OTHERWISE SPECIFIED:  
DIMENSIONS ARE IN MILLIMETERS  
SURFACE FINISH:  
TOLERANCES:  
ANGULAR:

USE UP AND  
BREAK SHARP  
EDGES

DO NOT SCALE DRAWING

REVISION

TITLE:

NAME	SIGNATURE	DATE
DRAWN		
CHEK'D		
APP'VD		
MEG		
Q.A.		

MATERIAL:

DWG NO.

door frame

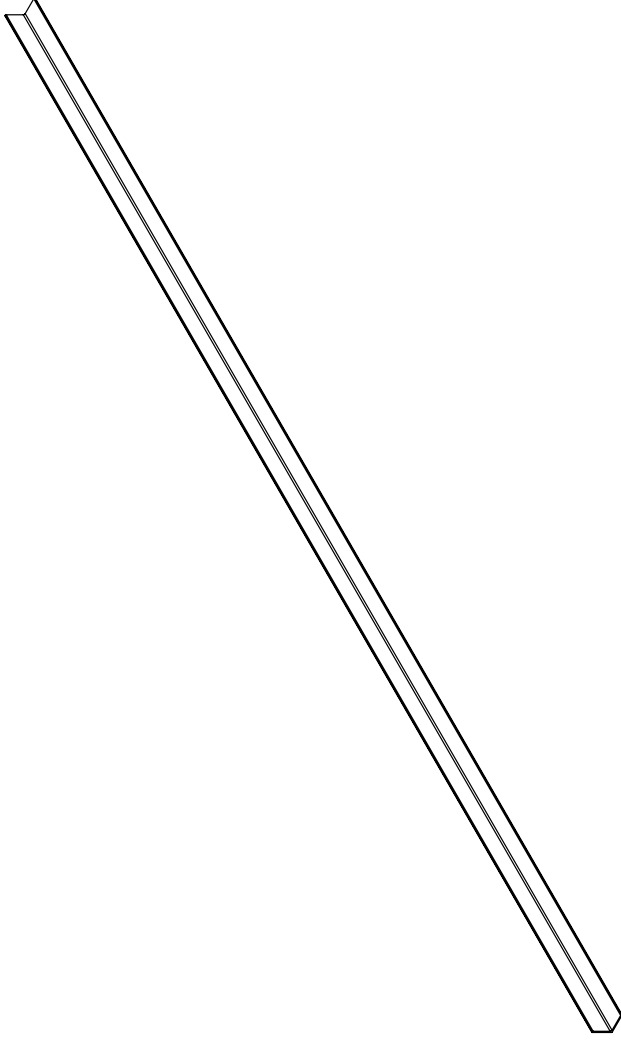
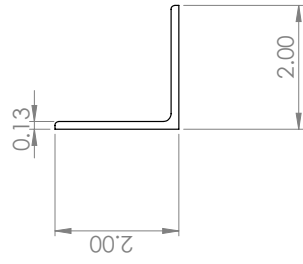
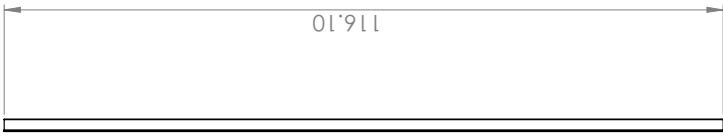
A3

SCALE: 50

SHEET 1 OF 1

WEIGHT:

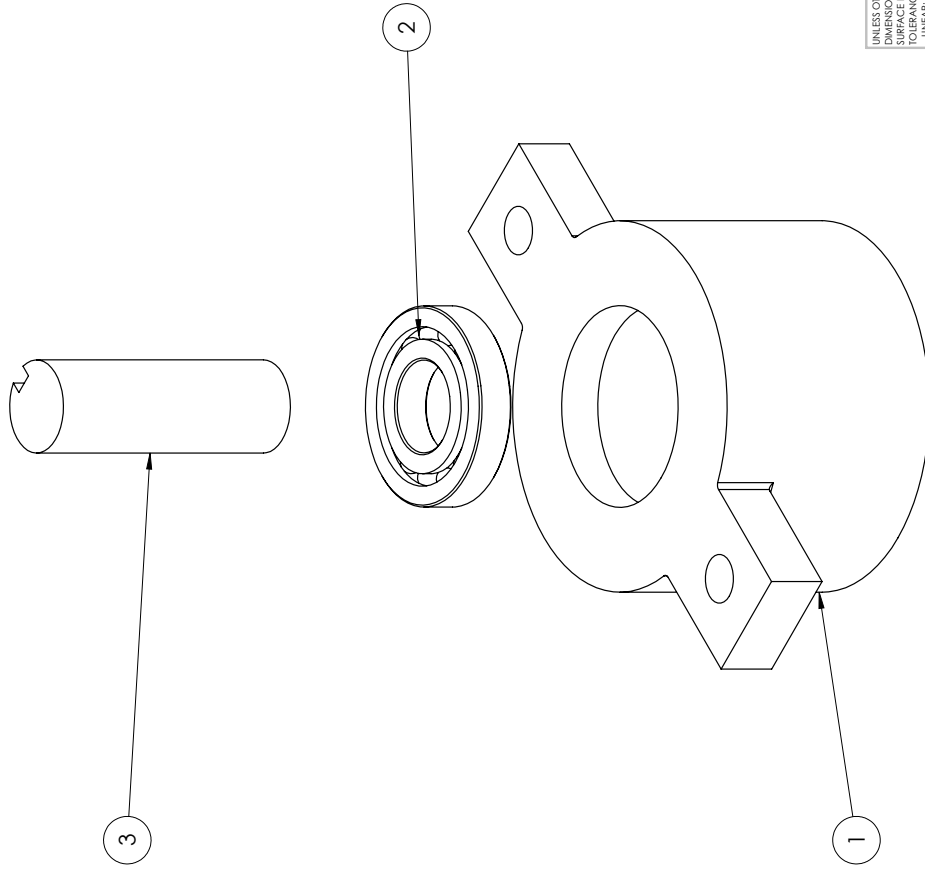




UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS		FINISH:		USE UP AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
SURFACE FINISH:		TOLERANCES:		MATERIAL:		DRAWG. NO.		A3	
ANGULAR:		NAME		SIGNATURE		DATE		TITLE:	
DRAWN		CHKD		APPVD		MEG		Q.A.	
WEIGHT:		SCALE: 20		SHEET 1 OF 1		RAIL			





ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Driven sprocket support		1
2	5908K19	Mc Master-Carr Part No.	1
3	MH172 Motor Shaft		1

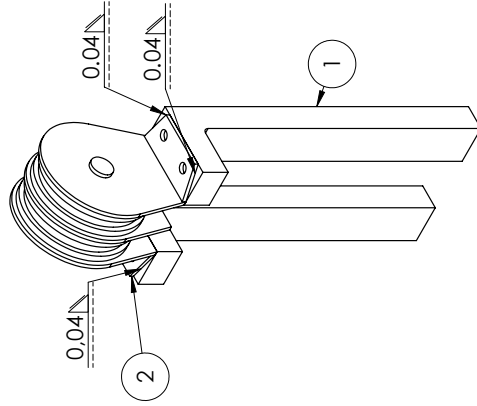


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS		FINISH:		BREAK AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
SURFACE FINISH:		TOLERANCES:		MATERIAL:		TITLE:			
MATERIAL:		WEIGHT:		SCALE: 1/2		SHEET 1 OF 1			
NAME	SIGNATURE	DATE							
DRAWN									
CHEK'D									
APP'VD									
MEG									
Q.A.									

Free Sprocket support

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Pulley Support		1
2	3099T44	McMaster-Carr Part No.	1

ITEM NO.	WELD SIZE SYMBOL	WELD LENGTH	WELD MATERIAL	QTY.
1	0.04 	0.68		2
2	0.04 	0.7		2

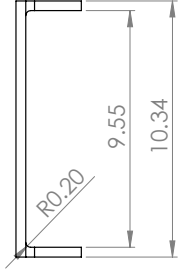
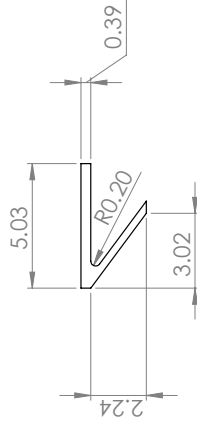
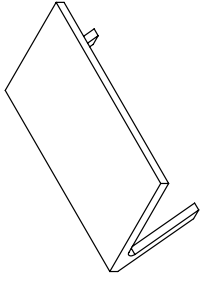


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS		FINISH:		RUB AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
SURFACE FINISH:		TOLERANCES:		HORIZONTAL:		VERTICAL:		ANGULAR:	
DRAWN		NAME		SIGNATURE		DATE		TITLE:	
CHECKED									
APPROVED									
MFG									
Q.A.									
								MATERIAL:	
								WEIGHT:	
								SCALE: 1:2	
								SHEET 1 OF 1	
								DRAWING: <b>Pulley #1 Drawing</b> A3	



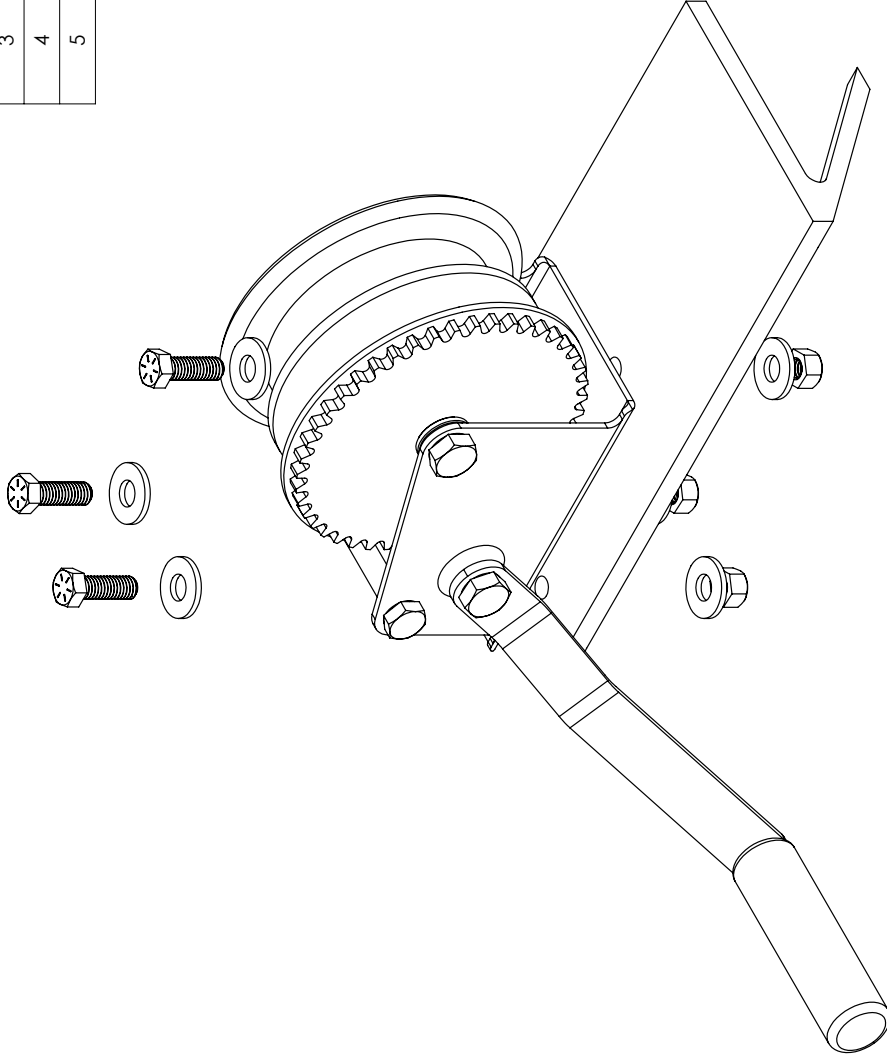






UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS		FINISH:		USE UP AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
SURFACE FINISH:		TOLERANCES:		DRAWN		NAME		SIGNATURE	
ANGULAR:		ANGULAR:		CHK'D		DATE		DATE	
				APP'VD					
				MFG					
				Q.A.					
				MATERIAL:					
				WEIGHT:					
				SCALE: 1:5					
				SHEET 1 OF 1					
				TITLE:					
				DWG NO:					
				A3					
				Stand for winch					

ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Winch Assembly	Käsivints 400-800kg	1
2	Stand for winch		1
3	92620A637	McMaster-Carr Part No.	3
4	90108A417	McMaster-Carr Part No.	6
5	94895A031	McMaster-Carr Part No.	3



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: ANGULAR:		FINISH:		BREAK AND BREAK SHARP EDGES		DO NOT SCALE DRAWING		REVISION	
NAME	SIGNATURE	DATE	TITLE:						
DRAWN									
CHEK'D									
APP'VD									
MEG									
Q.A.									
MATERIAL:			DWG. NO. <b>Winch Assembly exploded</b>						
WEIGHT:			SCALE: 1:5						
			SHEET 1 OF 1						