Team 24

ME 350 W13 Final Report

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1.0 Introduction & Requirements

1.1 Introduction

College students that are constrained to a wheelchair face difficulties when attending class. Lecture halls and classroom usually have little space to move around in, making it difficult to maneuver in a wheelchair. To ease mobility, it is common to place the student's backpack at the back of the wheelchair. However, in this position the backpack is not easily accessible. Consequently, the goal of this project is to design, build and test a powered mechanism to carry a secured backpack from the back of the wheelchair to the side. The full device has to be stored as compactly as possible on the back of the wheel chair and power draw from the battery that will eventually power it has to be minimized. The device must also meet safety requirements, including moving smoothly and at an appropriate speed while not pinching anyone if they have their hand in the way of operation. Additionally, this mechanism needs to be simple to manufacture, compact, reliable, safe, and compatible with a Quickie 2 Wheelchair to be applicable to the real-world. It is also desirable to have a maximum forward and minimum horizontal offset when the mechanism is at its final position, on the right side of the wheelchair.

1.2 System Requirements

1.2.1 Hard Requirements

	Motion Generator Requirements
MGR-01	Volume of entire mechanism has to be minimized and be less than 3000in^3
MGR-02	Angle of the backpack holder in the final position, relative to the armrest of the wheelchair has to be 0 degrees
MGR-03	Horizontal distance from the armrest (outside edge) to the farthest point on the backpack holder has to be minimized between 2in to 10in
MGR-04	Perpendicular distance from the centerline to the farthest point on the backpack holder has to be maximized between 6in to 17in
	Energy conversion & transmission
ETR-01	Total travel time of the mechanism from the starting position to the ending position has to be 9 seconds
ETR-02	Speed of input link during the last 30 degrees of travel has to be less than 10 degrees per sec
ETR-03	Power used by motor has to be minimized
ETR-04	Use provided Pololu motor model 1447 at 9V
ETR-05	Transmission ratio should able modify operating region to allow to operate at high, average, and low torque values from previous years

Table 1 Hard requirement of entire project

1.2.2 Soft Requirements

	Design Requirements
DSR-01	Keep the cost of the entire machine under $70
DSR-02	Design accordingly using materials provided or readily available
	Craftsmanship Requirements
CFR-01	Mechanism is well-made and can operate many cycles

CFR-02	Device should be strong, reliable and predictable
CFR-03	Device has to be well-machined to have a good surface finish
CFR-04	System must look professional, no use of duct tape or glue.
CFR-05	Device has to be sturdy
CFR-06	All parts should be in alignment with each other
Safety & Motor Requirements	
SMR-01	Operator should be able to control motion of mechanism with a rocker switch
SMR-02	IR sensors should stop mechanism if they detect object on the mechanism's motion path
SMR-03	Limit switches should not act as hard stops but as motor counter reset
SMR-04	Entire mechanism mas prevent have redundant system to prevent harm to operator
SMR-05	Motor should provide enough power to move a 12lb backpack

Table 2 Soft requirements for entire project

1.3 Executive summary (results/outcomes preview)

At the manufacturing stage one of the main goals was to maintain axial and translational friction at the lowest level with the main focus on the driving mechanism where several mechanical combinations were evaluated. Solid components were evaluated to have a better fit for the project under friction management premises as better alignment and fit could be provided to pivoting points, where shoulder bolts was decided to be used, also evaluated considering their respective axial loads. Craftsmanship was perhaps our second most important goal, ensuring that all the visible surfaces had the same finishing. After manufacturing was completed, extensive analysis was performed to determine dynamic and motion values to achieve full 175 degrees of deployment in 9 seconds under load condition. It was determined that we needed 4.028 RPM at the link with an induced torque of 760 oz-in. For our 9V Pololu motor we calculated a No Load speed of 60 RPM, a Stall Torque of 187.5 oz-in, a stall current of 3750 mA and torque constant of 50 oz-in/A . The required power was 136.7W and only achievable with the usage of 8:1 transmission reducing the power required to 17.1W. This ratio introduced the main change design as it required the usage of external transmission instead of the internal as initially designed. The needle bearings from the original design were replaced with bushings that demonstrated a better fit with the axis. The stress experienced by the set screws were 3870psi and 5150psi, shaft torques of 24.5lbf, and normal stress due to bending of 8940 psi and 13705 psi on the gears. All components of transmission had safety factors greater than 4.74. In all the cases our expected values we were extremely close to the final results. We were able to complete the cycle in 8.907 sec and achieve a deceleration speed of 9.1 degrees/second. The safety systems in place worked flawlessly and presented no problems during trials and testing.

2.0 Motion Generator

2.1 Design process

Each team member used a combination of Lincages 2000, SolidWorks, and ADAMS to design, model, and simulate a unique four-bar mechanism from the linkage design and conceptualization stage. Using

Lincages 2000, each member was able to set a desired start, middle, and final position for the mechanism and determine the necessary location of the ground points, as well as the ratio between links. We were also able to use Lincages to determine the transmission angle, and ensure that it was above 30 degrees, through the mechanisms path of motion. Each team member then used the information obtained from Lincages 2000 to create a model of their respective mechanism, with the proper dimensions and intended material, in SolidWorks. Each team member then exported their SolidWorks model into ADAMS to conduct a force and torque analysis on their mechanism. Based on the project requirements and the results obtained from each software, our team reviewed and selected the best design out of the five proposed mechanisms (see appendix A for individual designs). The statistics of all five different designs are depicted in Table 3.

Criteria	Design 1 Value	Design 2 Value	Design 3 Value	Design 4 Value	Design 5 Value
Volume, A (in^3)	359.6	363.5	287.3	311	371.5
Absolute Angle Offset, |Ω| (deg)	0	0	6	0	0
Horizontal Offset, B (in)	6.907	6.907	3.52	3.4	4.82
Forward Offset, C (in)	10.33	10.33	8.5	8.95	14.15
Max. Force, F (lb)	40	40	45	125	15

Table 3 Statistics of all 5 different design concepts

2.2 Final design

2.2.1 Design focus

When designing, manufacturing, and testing our mechanism, we always focused on ensuring that it was simple to manufacture, compact, reliable, and safe. We strived for a compact design, as we believed it would provide for easier use and adaptability with the wheelchair, especially given the daily space constraints a wheelchair-based student would experience. We also wanted to ensure our design was easy to manufacture, as we believed it would reduce both implementation and operational problems in the future. We strived to design and manufacture our mechanism to be reliable and robust to ensure reduced risk of failure, as well as the ability to operate under higher than expected loads. Furthermore, we added multiple sensors and hard stops to ensure that our mechanism was safe and reliable.

2.2.2 Design selection and matrix

The selection of our final design was obtained through the use of a Pugh chart. We first scored all designs according to six different criteria (see Table 4). We then compared all designs relative to one another (see Table 6). Based on our scoring criteria, we believed that the design with the highest score would be able to perform the best, according to both the team's goals, as well as the projects specifications. In the event of a tie, we would have picked the design that would have been the simplest to manufacture. Criteria and weighting of the scores are depicted in Table1. The total score for each criterion was calculated by multiplying the associated weight by the numbers of achieved points. The weight of each criteria was distributed over a scale from 1 to 3, were 1 was the least important, 2 important, and 3 the most important. After comparison through the Pugh chart, we found design 5 to be the best choice. Although all designs showed a large potential, design 5 was overall the simplest to manufacture, had lowest weight, and the smoothest transition throughout its operation. However, we believed design 5 could still use additional modifications to ensure that it was as robust, efficient, and safe as possible. The design's low weight allowed us to increase the cross section of the bars by 25%. By enlarging the bars, we were also able to install larger bearings, improving the fitting and limiting the deflections in the linkage system.

Criteria	Weight	Points 1	2	3
Volume, A	1	2000 in^3 < V ≤ 3000in^3	1000 in^3< V ≤ 2000in^3	0 < V ≤ 1000 in^3
Absolute Angle Offset, |Ω|	3	10° < |Ω| ≤ 15°	5° < |Ω| ≤ 10°	0° < |Ω| ≤ 5°
Horizontal Offset, B	3	8 in < B ≤ 10 in	5 in < B ≤ 8 in	2 in < B ≤ 5 in
Forward Offset, C	2	6 in < B ≤ 9 in	9 in < B ≤ 13 in	13 in < B ≤ 17 in
Max. Force, F	2	100 lb < F	50 lb < F ≤ 100 lb	0 < F ≤ 50 lb

Table 4 Weighting of criteria used to generate a Pugh chart to determine best design

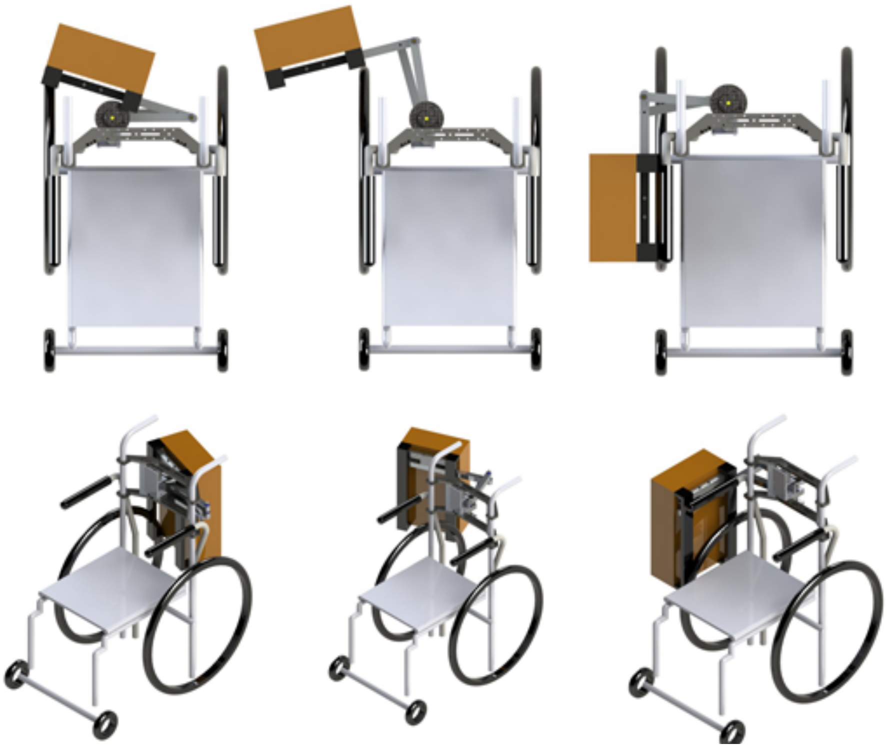
Criteria	Design 1 Points	Design 2 Points	Design 3 Points	Design 4 Points	Design 5 Points
Volume, A (in^3)	**3**	3	3	3	3
Absolute Angle Offset, |Ω| (deg)	9	9	6	9	9
Horizontal Offset, B (in)	6	6	9	9	9
Forward Offset, C (in)	4	4	2	2	6
Max. Force, F (lb)	6	6	6	2	6+
Total Score	**28**	**28**	**26**	**25**	**33**

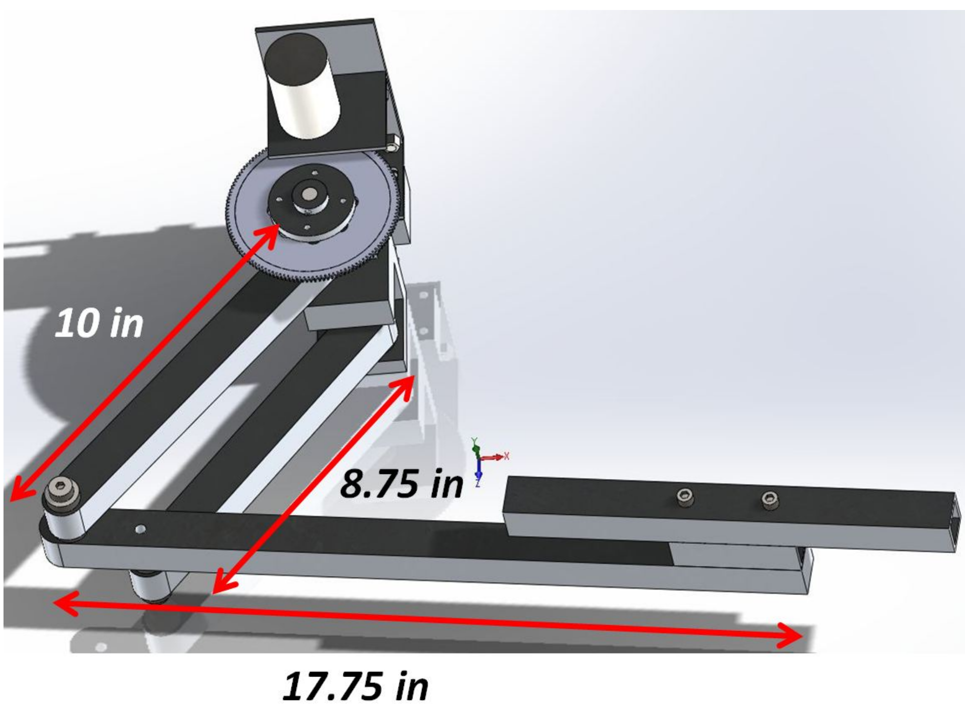
Table 5 Chart to score the 5 different designs. Design 5 was scored the highest. Hence, picked for further development as it scored the highest.

Description	Weight	Design 4	Design 5	Design 2	Design 3	Design 1
Horizontal Offset	3	0	1	1	1	1
Vertical offset	3	0	1	1	1	1
Offset angle	3	0	1	1	1	1
Manufacturability	1	0	1	-1	1	-1
Torque needed	2	0	0	-1	0	-1
Force needed	2	0	0	1	1	1
Volume	1	0	0	1	0	1
Cost	1	0	1	-1	1	-1
Weight	1	0	-1	-1	1	-1
Deflection	3	0	1	-1	-1	-1
Collapsibility	1	0	-1	1	-1	1
TOTAL		**0**	**12**	**7**	**10**	**7**

Table 6 Pugh chart to score the 4 different designs relative to one other design. Design 5 was scored the highest. Hence was picked for further development as it scored the highest.

2.2.3 Final design CAD





2.2.4 Links deflection

We used the multi-body dynamics software, MSC Adams, to conduct loading analysis on our linkage system. Throughout the analysis, we took into account the weight of every component, friction in the joints, the use of aluminum material, and the restriction of the hard stops. Our results yielded that there would be a deflection of approximately 4.35e^-11 inches (see Figure 2). We thus concluded that deflection would be negligible in our mechanism.

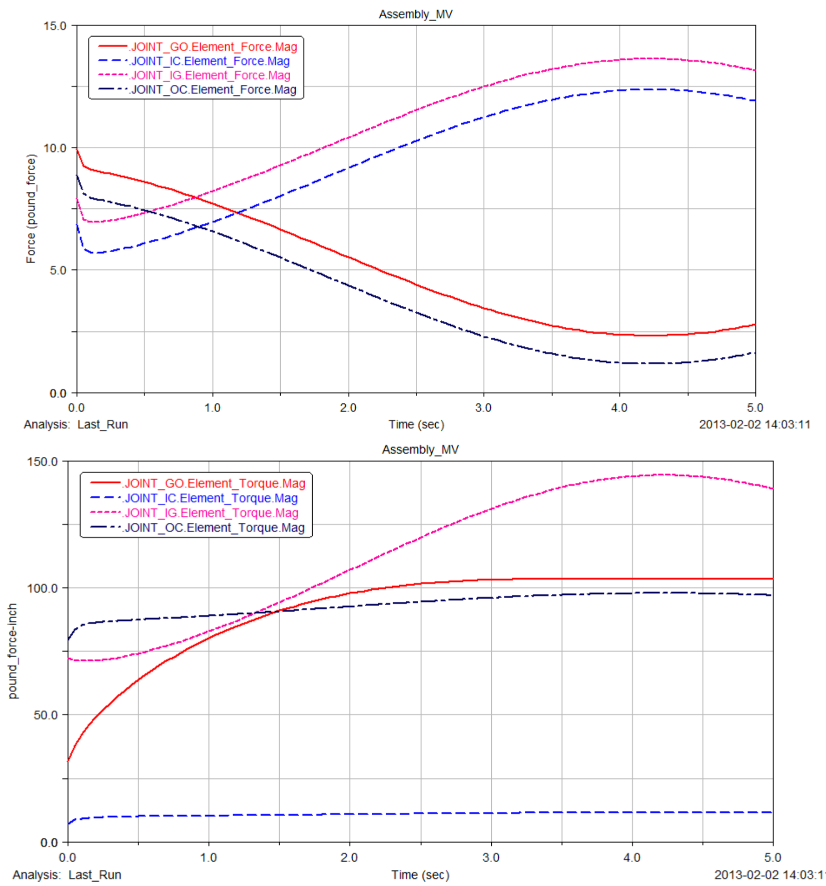


Figure 2: Plots from ADAMS deflection analysis

To calculate the bending moments, we simplified the model by assuming each beam would act as either a cantilever or simply supported beam. Our results showed that although bending moments existed, they were negligible, as they would cause no deflection in our links. Furthermore, we concluded that the material, geometry, and joints chosen for our mechanism were more than sufficient to ensure our design would operate smoothly and without failure during operation.

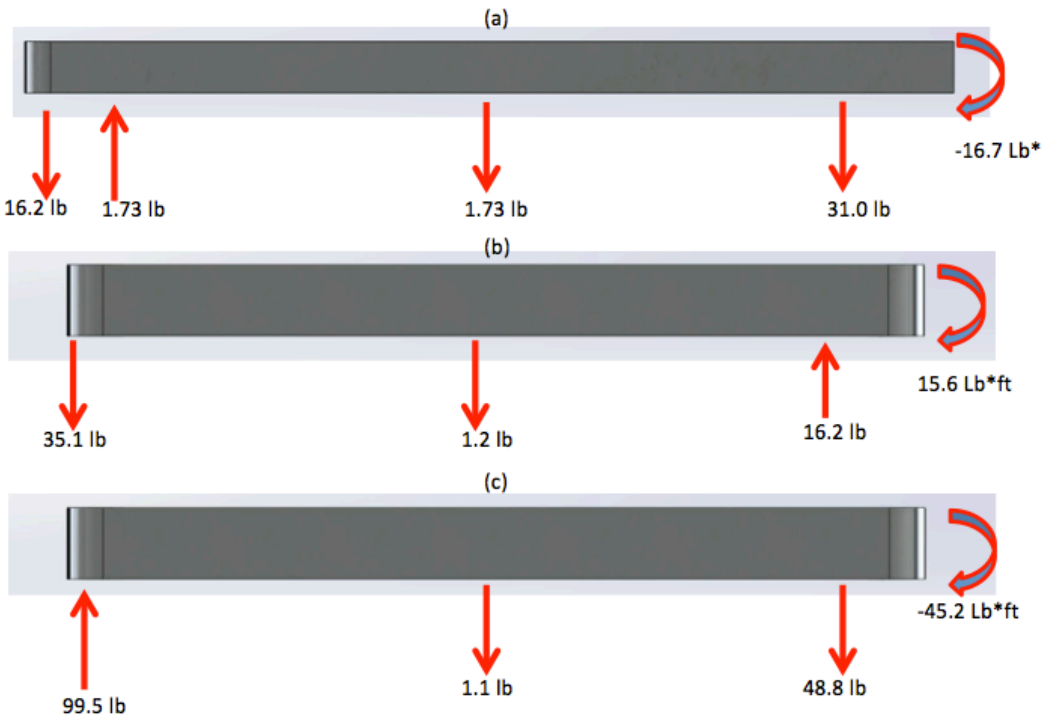


Figure 3: Free body diagrams for (a) coupler, (b) input, and (c) output links. Moments at end of links are considered to be negligible as they won't bent them

2.2.5 Friction minimizing joints

From experience, we knew that a forced pin, or misaligned connection between links would introduce high opposing torques and thus high friction. To address this issue we focused on the ground link, where two of our links would be connected and the highest forces is applied. We manufactured the ground link as one solid piece of aluminum. A solid ground link ensured that the alignment of the axis for the input and output links would only depend on the machining precision and not on the alignment of several assembled components.

Further, all joints were designed to reduce friction as much as possible to meet one of our goals. To properly address and managed friction within our system all joints have sleeve bearings and thrust bearings to reduce friction between links. In addition, all joints were lubricated to allow for a smoother motion. The input-coupler and the output-coupler joints have the same layout. There are sleeve bearings pressed into the input, output, and coupler links. Each joint has a shoulder bolt through the sleeve bearing with a thrust bearing on each side of the link as a spacer (see Figure 4). The shoulder bolt is then threaded into the coupler joint.

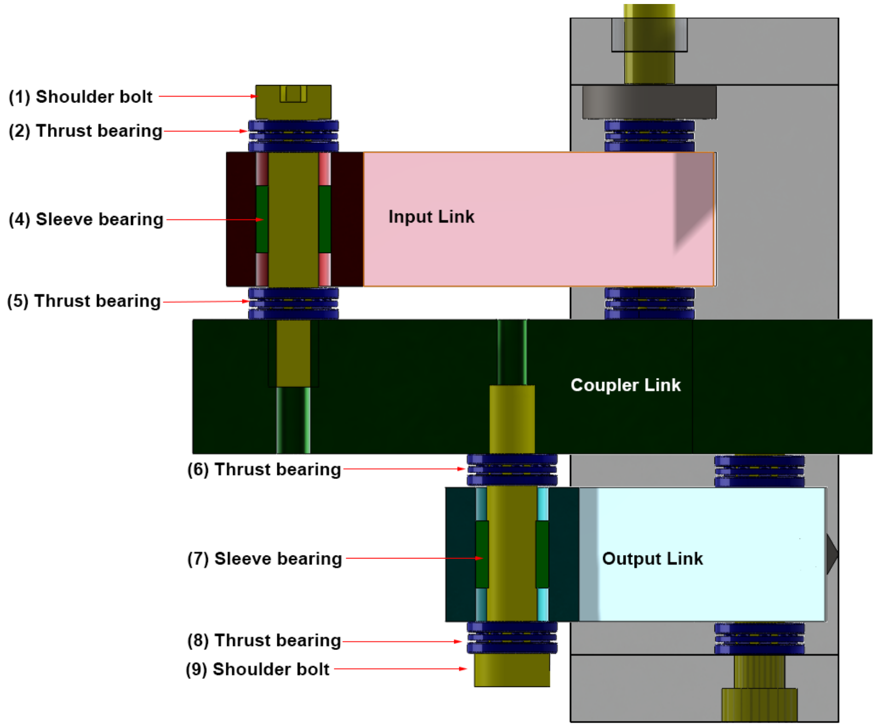


Figure 4: Cross-section view of coupler joints

The joints with the ground link were double-supported to reduce the deflection of the mechanism arm and ensure perfect alignment of axis of rotation. The ground-output joint has sleeve bearings in both links. The output link is secured between this double-supported joint with a shoulder bolt and threaded into the ground link. There are also thrust bearings on each side of the output link along with washers to ensure correct alignment (see in Figure 5). The ground-input joint has a shaft running through them and is fixed to the input link with a dowel pin. Each side of the input link has a thrust bearing to minimize friction and the vertical load on the joint. There are two sleeve bearings pressed into the ground link so that the friction of the shaft is minimal. Additionally, two c-clips on the input shaft restrict the movement of the shaft in the vertical direction.

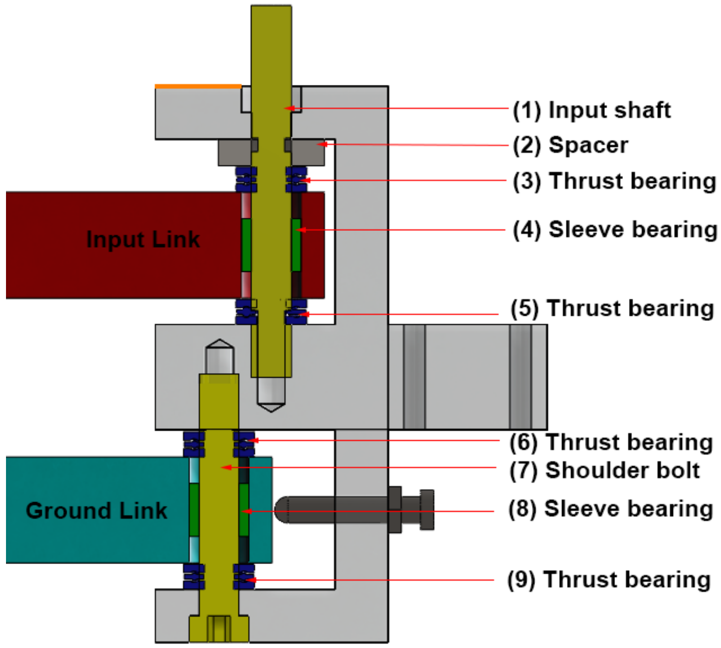


Figure 5: Cross-section view of ground joints

2.2.6 Bolt Load Rating Comparison

In this linkage mechanism we are using 3/8" shoulder bolts. These bolts have a cylindrical shoulder under the head that serves as a bearing surface. Bolts are made from alloy steel, with a Class 3A thread fit, a minimum tensile strength of 140,000 psi, and shear strength of 84,000 psi. Shoulder diameter tolerance is –0.004"; shoulder length tolerance is ±0.005". We used these high precision bolts to aligning moving parts within our linkage system, and were thus a critical component to our design. With a tight fit around these bearings, the deflection at the end of the linkage system was approximately zero. Based, on our calculations these bolts are strong enough to withstand our maximum (100 lbs.) load and the use of bushings will allow for smooth rotation during operation.

2.2.7 Transmission Angle

The transmission angle is a measure of the drivability of a linkage. More specifically, it defines how much of the driving torque can be transmitted to the output. A good linkage mechanism has a minimum transmission of 30 degrees. Analysis of the final design, using Lincages 2000, can be seen in Figure 6. Our mechanism travels 175 degrees from the initial to final position. At the final position of 175 degrees, the transmission angle has a value of 50 degrees. Therefore, the mechanism complies with requirement previously mentioned.

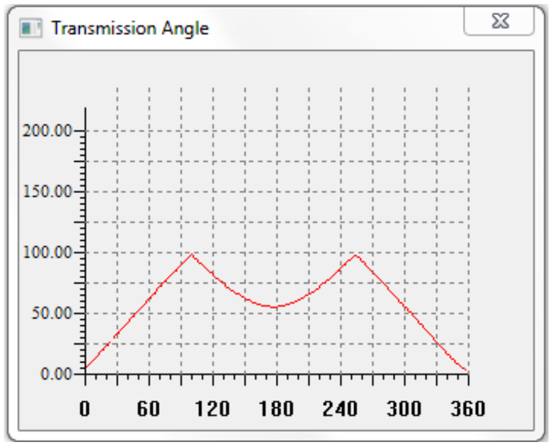


Figure 6: Transmission angle analysis results for final design. Linkage complies with the minimum transmission angle of 30 degrees.

2.2.8 Craftsmanship

The team ensured craftsmanship of the mechanism was of top quality, we ensured every detail was covered to have a sturdy and robust device and at the same time have a professional look. The finish of the aluminum stock surface used for the links was very dull. Since painting the device was not an option, we opted for a machined finish. Using a ½" end mill, we machined a very small layer off of the entire linkage to give it a nice move and texturized finish to the surface. This process also ensured that the all links were exactly 1" by 1". Another process to improve the finish of the device was to make the links' ends rounded. We achieved this by using a radius end mill provided in the shop. This eliminated all possible sharp corners that could harm the operator or others. The open sides of the backpack holder fabricated from square aluminum tubing were sealed with plastic caps to improve the esthetic of the mechanism. To make the mechanism sturdy we fabricated the ground link from a solid aluminum block instead of using a water-jet to cut out several pieces to form the ground. This also reduces the risk of having misalignment issues. All wiring of electronic components was layout tightly and to precise wire sizes to prevent wires from tangling with each other. Also, shrink tubing was used to secure groups of several long wires to have a more professional look.

2.3 Changes to final design

Several changes were made to the final design of the mechanism. On the original manufacturing plans it specified to use needle bearings and shoulder bolts to reduce friction in all joints. With this combination the mechanism assembly was not sturdy. Therefore, we changed the needle bearings for sleeve bearings. Also, we originally designed for our ground link to house a gear up to two inches in diameter. After our transmission analysis we realized that we need a gear of at least four inches to achieve the required transmission ratio. Therefore, we moved the gear from the inside of the ground link to the outside top of the ground link and used a spacer to fill the space that the original gear placement occupied. Also, due to some error in the machining of the ground link we were forced to add washers between the output and ground link.

2.4 Final testing results

The motion generator component of the mechanism was tested on total volume (A) in starting position, and the angle (Θ), horizontal (B), and forward (C) offset at final position (see Figure 7). The total volume was measured as the smallest rectangular box (length, width, and height), while parallel to the connection points, which could encase our device including all elements. Our final testing results were very similar to the values we anticipated based on our 3D CAD Model, which provides accurate measurements (see Table 7). Differences could be a result of the accuracy limits of the analog measuring instruments, as well as human error. All final results, with the exception of the forward offset were either exact or very close to the required target values. Our mechanism's final volume was very small; reaching only 42% of the maximum allowable volume of 3000 in^3. The angle offset was only one-degree away from the target value. This deviation can be accounted for as an accuracy error in the measurement method. The horizontal offset was minimized to be zero inches. Although our forward offset was not maximized

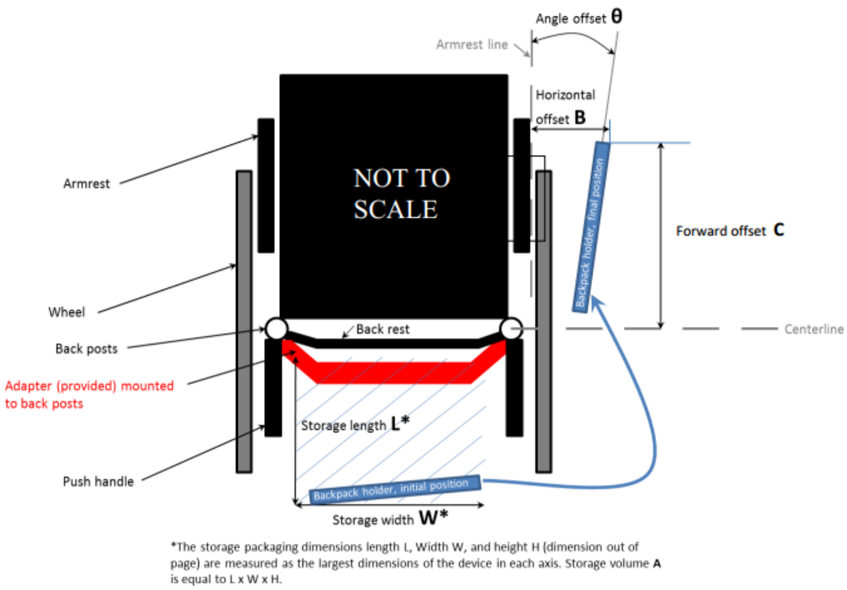


Figure 7: Schematic of wheelchair (Top View) showing start and end positions of backpack holder and measurements for grading criteria. (Not to scale).

Results	**Volume (Height x Length x Width) (Inches^3)**	**Angle Offset, Θ (degrees)**	**Horizontal Offset, B (Inches)**	**Forward Offset, C (Inches)**
Expected		0	0	11.5
Final	(8.25x17.25x9) 1280	1	0	11.375

Table 7: Expected and final results of final motion generator testing

3.0 Energy Conversion & Transmission

3.1 Introduction

When designing our linkage transmission system, we set out to achieve a 175-degree rotation within the desired operation time of 9 seconds. In order to complete our mechanism motion within the desired time of 9 seconds, our linkage system would have to rotate at a minimum RPM of 4.028 RPM and provide 760 oz-in of torque. Also, the last 30 degrees of motion should be no faster than 10 degrees per second. However, In order to achieve these settings, we had to use a transmission, as our stock motor operates at a no load speed of 60 RPM and a stall torque of 188 oz-in. Our calculations showed that we would require an 8:1 transmission on our stock to transform them into our needed operational settings. Although there were multiple transmission options, we decided to use a single gear train, as we believed it would be the most reliable and efficient option.

3.2 Energy Conversion and Transmission Requirements

Energy conversion &transmission	
ETR-01	Total travel time of the mechanism from the starting position to the ending position has to be 9 seconds
ETR-02	Speed of input link during the last 30 degrees of travel has to be less than 10 degrees per sec
ETR-03	Power used by motor has to be minimized
ETR-04	Use provided Pololu motor model 1447 at 9V
ETR-05	Transmission ratio should able modify operating region to allow to operate at high, average, and low torque values from previous years

Table 8: Energy conversion and transmission requirements

3.3 Speed/Time Discussion

To achieve and comply with the set requirements we identified we need to operate the motor at two different speeds. Knowing the total travel time of 9 seconds, the total travel distance of 175 degrees, and the required angular speed for the last 30 degrees of travel we can calculate the required speed for the other operating speed using equation 1-3 below.

Known: t= 9 sec., Θ_T =175 deg., Θ_2 =30 deg. W_2=10 deg/sec
Unknown: W_1

$$\theta_1 = \theta_T + \theta_2 = 175 - 30 = 45\, degrees \qquad eq.1$$

$$t = \frac{\theta_1}{w_1} + \frac{\theta_2}{w_2} = 9 = \frac{145}{w_1} + \frac{30}{10} \qquad eq.2$$

$$\therefore w_1 = \frac{\theta_1}{t - \frac{\theta_2}{w_2}} = \frac{145}{9 - \frac{30}{10}} = 24.17\, deg/\sec = \mathbf{4.028\, rpm} \qquad eq.3$$

3.4 Calculating Torque Requirements

The torque required to move the linkage through the full range of motion was measured using a force gauge kept perpendicular to the input link through the entire motion. The experimental values obtained can be used to approximate the motor torque required to move the linkage, taking into account factors such as friction, alignment, and construction quality. The force gauge measured the force required to move the linkage, to calculate the torque required we used equation 4 below, where L = 8 inches and Θ = 90 degrees.

$$T = LFsin\theta = LF \qquad eq.4$$

For the forward and backward motions we measured a maximum peak force of 95 and 90 ounces respectively. Plugin known values into equation 4 we obtained the maximum torque required for the forward and backward directions.

$$T_{Lforw.} = F_{forw.} \times L = 95[oz] \times 8[in] = \mathbf{760\ oz - in}$$

$$T_{Lback.} = F_{back.} \times L = 90[oz] \times 8[in] = \mathbf{720\ oz - in}$$

3.5 Transmission Ratio Selection

3.5.1 Calculating Motor Specifications at 9V

The motor to be used to power the mechanism motion is a Pololu motor model 1447. It is a DC permanent magnet gearmotor with 131:1 gear ratio. For this project the motor must run at 9V. The specifications provided by the manufacturer (see Table 9) are not at our operating voltage. Therefore, by interpolating and assuming a linear relationship we can obtain the stall torque and no-load speed at 9V using equations 5-8.

Voltage [V]	No-load speed [rpm]	Stall Torque [oz-in]	Stall Current [mA]
6	40	125	2500
12	80	250	5000

Table 9: Manufacture specification of Pololu motor model 1447

$$\text{No} - \text{Load speed: } W_0(9V) = \frac{9[V]}{12[V]} \times W_0(12V)[rpm] = \frac{9}{12} \times 80 = \mathbf{60\ rpm} \qquad eq.5$$

$$\text{Stall Torque: } T_s(9V) = \frac{9[V]}{12[V]} \times T_s(12V)[rpm] = \frac{9}{12} \times 250 = \mathbf{187.5\ oz \cdot in} \qquad eq.6$$

$$\text{Stall Current: } I_s(9V) = \frac{9[V]}{12[V]} \times I_s(12V)[rpm] = \frac{9}{12} \times 5000 = \mathbf{3750\ mA} \qquad eq.7$$

$$\text{Torque Constant: } K_t = \frac{T_s[oz \cdot in]}{I_s[A]} = \frac{187.5}{3.750} = \mathbf{50}\ \frac{\mathbf{oz \cdot in}}{\mathbf{A}} \qquad eq.8$$

3.5.2 Torque/Speed Requirement Analysis

For our mechanism motion to comply with project requirements we found that we need to operate at 4.028 rpm and 760 oz-in torque. However, it is clear from motor's torque/speed curve (see Figure 8) that our desired location to operate lays outside the motors capabilities. Therefore, a transmission needs to be implemented to modify the motors area of operation.

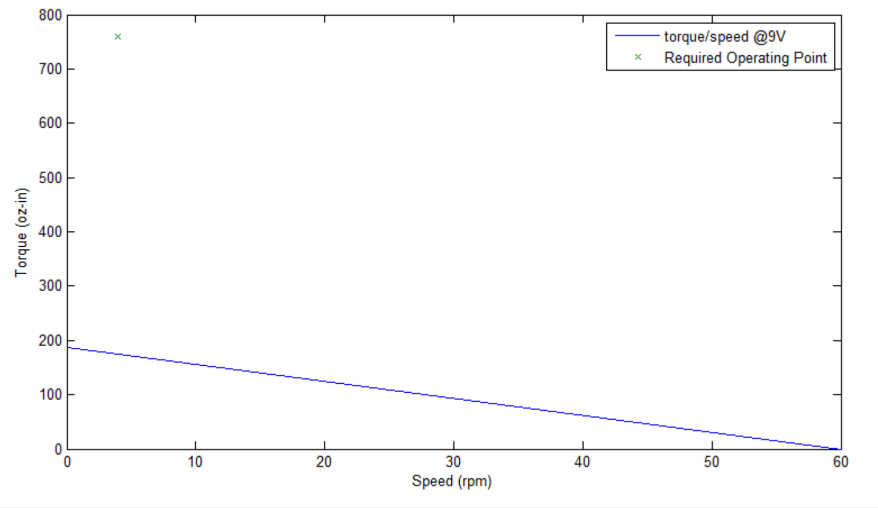


Figure 8: Pololu motor model 1447 torque/speed curve at 9V assuming linear relationship

By using equations 9-10 we calculated the new outputted stall-torque and no-load speed of the motor after the addition of a transmission with a gear ratio of N (see table xx for results). Additionally, using these results and assuming a linear relationship we can plot a torque/speed curve for each ratio (see Figure 9).

$$\text{Stall} - \text{Torque with tranmission: } T_{st} = N \times T_s \qquad eq. 9$$

$$\text{No} - \text{load speed with tranmission: } w_{0t} = \frac{w_0}{N} \qquad eq. 10$$

Transmission Ratio	**Stall-Torque [oz-in]**	**No-load Speed [rpm]**
6:1	6 x 187.5 =1125	7.5
7:1	1312.5	8.571
8:1	1500	7.5
9:1	16875	6.667

Table 10: Stall-torque and No-load speed values for Pololu motor model 1447 with different transmission ratios.

From the torque/speed curves we determined that an 8:1 transmission ratio is required to modify the original operating region in order to operate at 4.028 RPM and 760 oz-in torque. This transmission also allows us to operate at an angular velocity of 4.028 RPM and a low torque of 290 oz-in and an average torque of 560 oz-in. While the transmission ratio does not allow us to operate at a high torque of 1120 oz-in, after discussing with experts in the topic, we concluded that satisfying this point is not of crucial importance.

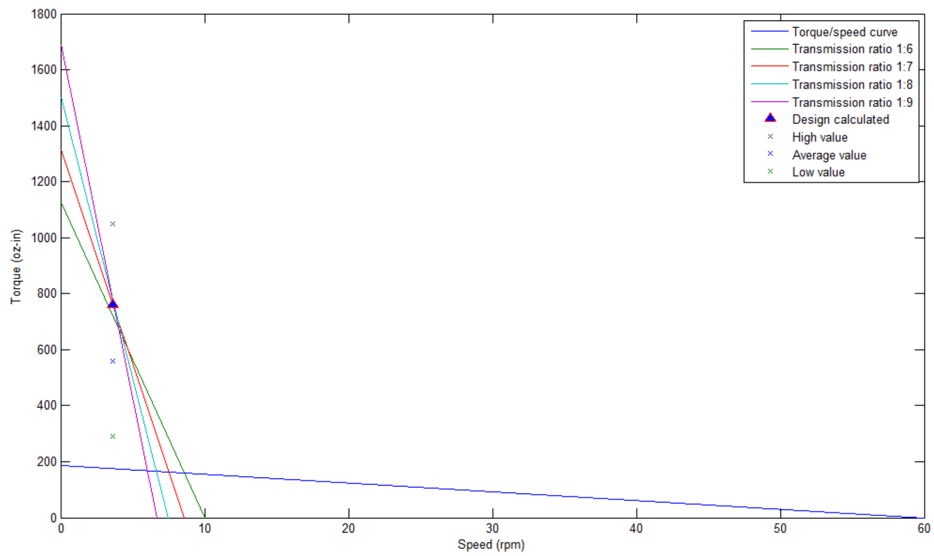


Figure 9: Torque/Speed curves for Pololu motor model 1447 with different transmission ratios.

3.5.3 Power Requirement Analysis

The mechanism must be able to function correctly under the 9 volts supplied to the Pololu motor. Knowing the motor's characteristics, linkage torque, and speed requirements, found previously, we determined the power input, power output, and power required.

$$P_{in} = I_s[A] \times V[V] = 3.75 \times 9 = \mathbf{33.75\ W}$$

$$P_{out} = T_{max.linkage} \times w_{linkage} = 760[oz \cdot in] \times 4.028[rpm] = 3061.28\ oz \cdot in \cdot rpm$$

$$\therefore P_{out} = 3061.28\ [oz \cdot in \cdot rpm] \times \frac{1\ W}{oz \cdot in \cdot rpm} = 3061.28 \times 7.3948 \times 10^{-4} = \mathbf{2.264\ W}$$

$$P_{req.} = \frac{T_L}{K_t} \times V = \frac{760\ oz \cdot in}{50 \frac{oz \cdot in}{A}} \times 9\ V = \mathbf{136.8\ W}$$

Clearly without a transmission the power requirement is beyond the capabilities of the motor. Adding our selected transmission ratio of 8:1 to the system the maximum power requirement decreases to 17.56 W (see equation 11). Therefore, with the selected transmission ratio the motor meets the power requirement imposed by the load on the mechanism.

$$P_{req.} = \frac{T_L}{N \times K_t} \times V = \frac{760\ oz \cdot in}{8 \times 50 \frac{oz \cdot in}{A}} \times 9\ V = \mathbf{17.1\ W} \qquad \boldsymbol{eq.11}$$

Similarly, using the equation 11 above we can calculate the power required if we operated at the high, average, low torque requirements from previous years and use an 8:1 transmission ratio. The calculation results in table 11 show that with our selected transmission ratio we are able to provide sufficient power to all three cases.

Case	Torque [oz-in]	Required Power [W]
High	1050	23.625
Average	560	12.6
Low	290	6.525

Table 11: Power required for the high, average, and low cases using an 8:1 transmission ratio

We required a transmission that would minimize the power drawn, while still achieving a complete motion of the mechanism. The only parameter we could modify to change the amount of power required was the current. The current is a function of the torque required and the torque constant ($I = {}^{T_L}/_{K_t}$). Thus, by reducing the torque load we were able to reduce the current and consequently the power drawn. To reduce the torque we required joints between links to have negligible friction, and zero deflection of links and mechanism. More detail on how to reduce deflection and friction is covered in sections 2.2.4 and 2.2.5.

3.6 Evaluation Transmission Options

3.6.1 Transmission selection

Several options for transmission exist; they are divided into two main categories, flexible and rigid drives. To select our transmission we went through a rigorous down-selection process. Step 1 we analyzed the pros and cons of two flexible and two rigid drives and selected the two best options for our mechanism (see Table 12)

Drive	Pros	Cons
Simple gear train (rigid drive)	• Speed ratio is constant • High efficiency • Durable • Reliable • Lower maintenance	• Does not absorb shock • Relative expensive • Require lubrication
Compound gear train (rigid drive)	• Achieve gear ratio great than 1:10 • High efficiency • Durable • Reliable	• Occupy bigger volume • Relative complex mounting • More prom to failure due to misalignment
Timing belt (flexible drive)	• Relative lower cost • Shock absorbent • Clutching function • Low noise	• Short life • Lower efficiency
Chain (flexible drive)	• Large torque • No slippage • Flexible position • Durable	• Noise • Lower speeds • Lubrication required • Bulky • Hazardous

Table 12: Table of pros and cons for 4 different possible transmissions. Simple gear train and timing belt were selected as the two best options from the four analyzed.

We further analyzed the previously selected simple gear train and timing belt to select our final transmission. We analyzed the feasibility of using them by considering parts availability, cost, assembly, and manufacturability (see Table 13).

Specification	Simple Gear Train	Timing belt
Components	1 – 16 teeth low carbon steel gear 1 – 128 teeth plastic gear	1 – 12 grooves pulley 1 – 100 grooves pulley 1 – Timing belt
Availability	Gears available in stock from www.amazon.com	Pulleys and timing belts available in stock from www.sdps-si.com
Total Cost	$65.85 for gears No cost for manufactured parts	$18.29 for pullers and belt No cost for manufactured parts
Assembly	Assembly would require less space. More restriction on location to prevent interference with linkage. Less probability of transmission interfering with linkage.	Assembly would require more space but it will be easier to adapt it to the link. Has to be precise to have correct tension.
Manufacturability	Requires manufacturing of a motor shaft to link the input link to gear shaft.	Requires manufacturing of a motor shaft to link the input link to gear shaft.
Size	Relative compact transmission, size depends on diameter of gears. Sum of gears diameter is 4.5".	Large transmission due to belt length. Diameter of pulleys does have significant effect on size.
Safety	Potential damage to user by direct contact between gears and skin or hair getting tangled. Very safe transmission if placed correctly.	Not harmful to user if going at moderate speeds. Very safe transmission.

Table 13: Thorough analysis of a simple gear train and timing belt

The final transmission chosen is a simple gear train with a 16 teeth low carbon steel gear and a 128 teeth low carbon steel gear. The simple gear train, despite its higher cost, allowed for an easier assembly, and ensured optimum performance. Also, the durability of this transmission was a crucial selection factor, as we wanted the mechanism to be able to operate for the greatest amount of time, without the need of repairs. The specifications for each component of the transmission are found in Table 14 and more detail about the gear can be found in Appendix C. All the parts for the component are in stock and available from www.amazon.com and will have a total cost of $65.85 including shipping.

	Gear 1	Gear 2	Motor shaft
Part Number	H3216	H32128	N/A
Provider	Boston Gear www.bostongear.com	Boston Gear www.bostongear.com	In house manufactured
Specification	No. of teeth: 16 Diametral Pitch: 32 Material: Steel Hub: Pin Hub Pressure Ang.: 14.5 deg. Diameter: 0.562"	No. of teeth: 128 Diametral Pitch: 32 Material: Steel Hub: Pin Hub Pressure Ang.: 14.5 deg. Diameter: 4.062"	
Cost	$18.83	$47.02	$0.00

Table 14: Specifications of final transmission components.

3.6.2 Transmission Interference

We have a limited selection on gears to achieve our desired gear ratio of 8:1. With the Pololu motor shaft having a diameter of 6mm (.236”) our options dwindled. To overcome this obstacle we chose to go with a gear with a bore of .25”and secure it with a set screw. This will allow us to positions the gear properly on the shaft and give us more possible sizes. The bore of the gear we selected is .5” and we need to achieve .375” bore. We will tap into the gear and secure it to the coupler and then use a set screw in the side of the coupler to tighten down on the shaft. This will give us the proper rigid fit we desire. Due to the change in the placement of the larger gear it is now exposed, this is can be harmful to the user. To overcome this issue we are going to use a proximity sensor so that if an object approaches the gear the entire system with shut down.

3.7 Mounting

3.7.1 Motor mounting

The supplied motor bracket cannot withstand the high forces and will bend. Therefore, the motor will be mounted to the mechanism using a custom manufacture bracket. The bracket will be composed of an aluminum sheets bend at 90 degrees and an aluminum angle hold together by 4 ¼”bolts and nuts to form a C shape (see Figure 10).

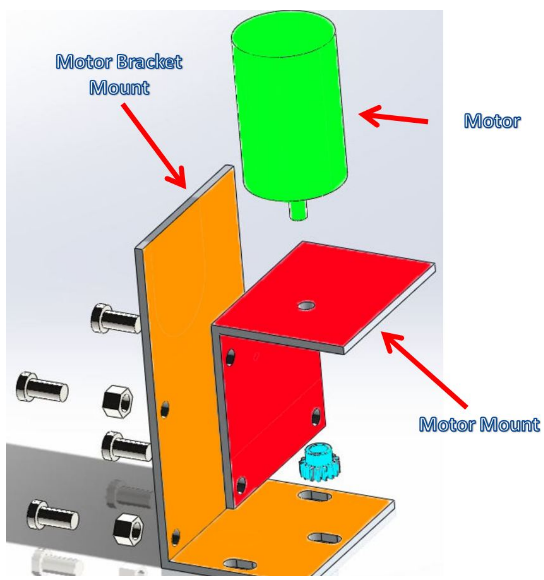


Figure 10: (a) Exploded view of motor mounting bracket

The bracket has been designed to be fully adjustable in both the horizontal and vertical directions to ensure a perfect gear mesh. The bracket will be secured to the motor through small bolts attached at the motors front face. The entire motor bracket will be mounted to the ground link and will be secured with bolts threaded all the way into wheelchair adapter (see Figure 11).

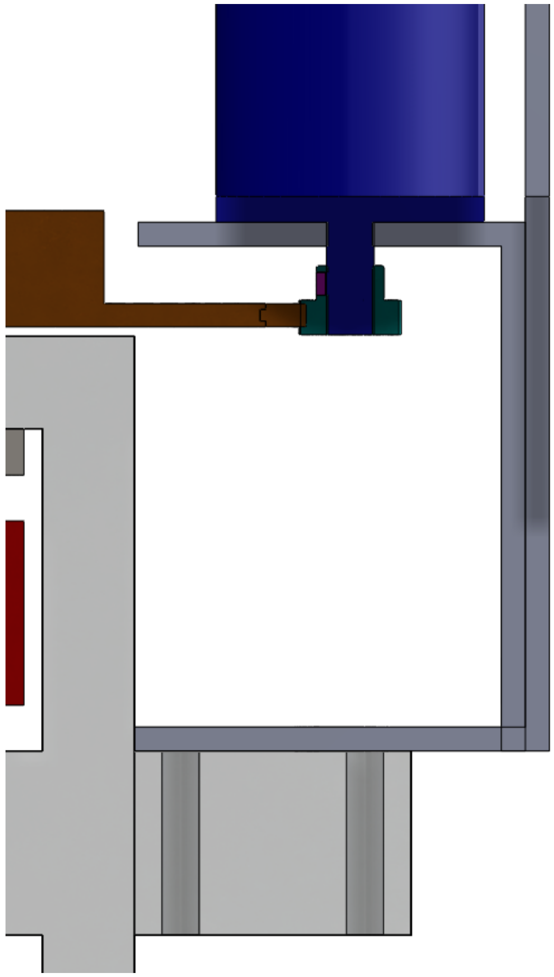


Figure 11: Motor mounting bracket cross-section of mounting

3.7.2 Gear mounting

We mounted our 16 teeth gear directly to the motor shaft and secured it with two set screws to ensure that the gear did not slip and that torque was not diminished. However, to use the two set screws, we had to machine an additional threaded hall in the gear's hub. The 128 teeth gear was press fitted into the input link shaft and secured with a set screw. The input shaft was pressed into the input link, with a dowel pin securing the shaft to the link. This resulted in a fixed gear that was able to effectively turn our mechanism (see Figure 12).

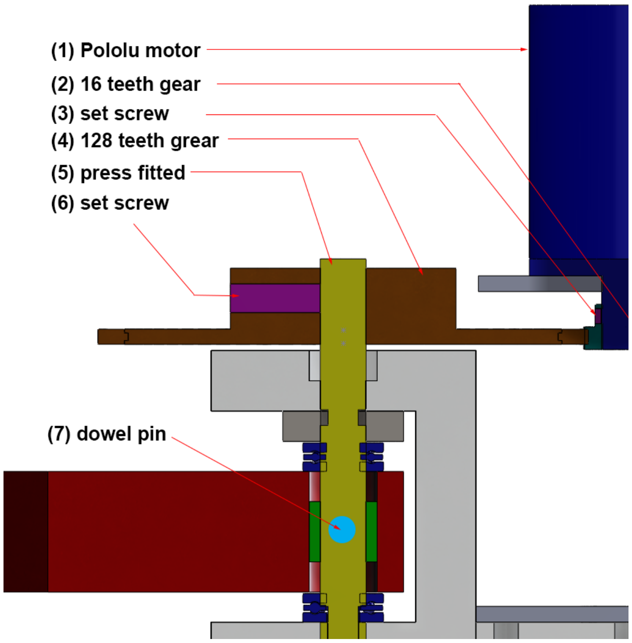
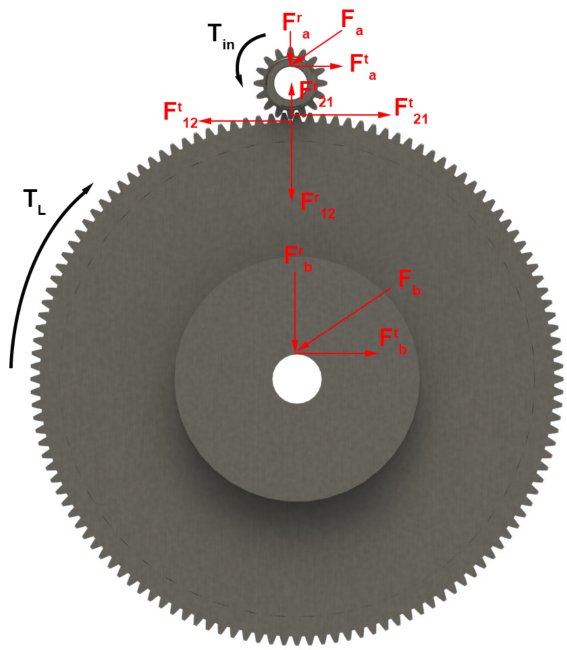


Figure 12: Cross-section view of gear mounts



3.7.3 Effect on volume and offset

The addition of the motor mounting bracket does have a significant effect on the volume of the entire mechanism. Differently the gears do not affect the volume. Because we are mounting it on top of the ground link it increases the total height of the mechanism by 3.5" Consequently, the total volume of the mechanism increased. Despite the increase the volume is still significantly smaller than the maximum allowable. Neither the motor mount nor the gears affects the horizontal, vertical, or angle offset of the mechanism.

3.8 Transmission Loading

3.8.1 Gear's stress analysis

The two spur gears that make up the transmission will experience different torques and consequently will experience different stresses. The safety factor of the transmission failing due to a failure in the set screw of either gear, or failure due to bending of gear teeth was determined. In all cases the safety factor was beyond the 2 value commonly used. Therefore, we do not expect any failures of the transmission. To find the stress due to bending the Lewis equation was used.

3.8.1.1 Stress on 16 teeth gear set screw

Known: N = 8, T_L = 760 oz-in, ø = 14.5 deg. , d_1 = 0.5", d_{hub} =0.4",d_{bore}=0.25", $d_{set\ screw}$ =1/8", N_1 = 16, $\sigma_{allow.\ set\ screw}$ = 33000psi

$$T_{in} = \frac{T_L}{N} = \frac{760\ oz \cdot in}{8} = 95oz \cdot in = 5.9375lbf$$

$$F_a^t = \frac{T_{in}}{d_{bore}/2} = \frac{5.9375}{0.25/2} = 47.5\ lbf$$

$$\sigma_{set\ screw} = \frac{F_a^t}{A_{set\ srew}} = \frac{F_a^t}{\frac{1}{4}\pi d_{set\ scew}{}^2} = \frac{47.5}{\frac{1}{4}\pi\left(\frac{1}{8}\right)^2} = \mathbf{3870\ psi\ (3s.f.)}$$

$$Safety\ factor = \frac{\sigma_{allow\ set\ screw}}{\sigma_{set\ screw}} = \frac{33000}{3870} = \mathbf{8.5}$$

3.8.1.2 Torque on motor shaft, F_m

$$F_{21}^t = \frac{T_{in}}{d_1/2} = \frac{5.9375}{0.5/2} = 23.75\ lbf$$

$$F_{21}^r = F_{21}^t \tan\varphi = 23.75 \tan 14.5 = 6.14\ lbf\ (3s.f.)$$

Since $\sum F_x = 0$ and $\sum F_y = 0$

$$F_a^x = F_{21}^t = 23.75\ lbf \text{ and } F_a^y = F_{21}^r = 6.14\ lbf\ lbf$$

$$\therefore F_m = F_a = \sqrt{23.75^2 + 6.14^2} = \mathbf{24.5\ lbf\ (3sf)}$$

3.8.1.3 Stress due to bending on 16 teeth gear

Known: F =0.188", P = 32, w_1 = 32.224 rpm, Y = 0.296 (table 14-2), $\sigma_{allow.}$= 65000psi assume cut or milled profile

$$\text{Pitch} - \text{line Velocity, V} = \frac{\pi d_1 w_1}{12} = \frac{\pi \times 0.5 \times 32.224}{12} = 4.22\ ft/\min\ (3sf)$$

$$K_v = \frac{1200 + V}{1200} = \frac{1200 + 4.22}{1200} = 1.004$$

$$\sigma_{1bending} = \frac{K_v F_{21}^t P}{FY} = \frac{1.004 \times 23.75 \times 32}{0.188 \times 0.296} = \mathbf{13705\ psi}$$

$$Safety\ Factor = \frac{\sigma_{allow}}{\sigma_{1bending}} = \frac{65000}{13705} = \mathbf{4.74}$$

3.8.1.4 Stress on 128 teeth gear set screw

Known: d_2 = 4", d_{hub} =0.4",d_{bore}=0.375", $d_{set\ screw}$ =1/4", N_2 = 128, $\sigma_{allow.\ set\ screw}$ = 55030 psi

Assuming no slip and perfect mesh

$$F_{12}^t = F_{21}^t = 23.75\ lbf$$

$$F_{12}^r = F_{21}^r = 6.14\ lbf$$

$$\therefore \text{Force on input shaft} = F_b = F_a = \mathbf{24.5\ lbf}$$

If this is correct then $F_{12}^{t}\frac{d_2}{2} = T_L$ should be true

$$F_{12}^{t}\frac{d_2}{2} = 23.75\frac{4}{2} = 47.5\ lbf\ = 760\ oz \cdot in\ \triangleq T_L$$

$$F_b^t = \frac{T_L}{d_{bore}/_2} = \frac{47.5}{0.375/_2} = 253\ lbf$$

$$\sigma_{set\ screw} = \frac{F_b^t}{A_{set\ srew}} = \frac{F_b^t}{\frac{1}{4}\pi d_{set\ scew}{}^2} = \frac{253}{\frac{1}{4}\pi\left(\frac{1}{4}\right)^2} = \mathbf{5150\ psi\ (3s.f.)}$$

$$Safety\ factor = \frac{\sigma_{allow\ set\ screw}}{\sigma_{set\ screw}} = \frac{55030}{5150} = \mathbf{10.7}$$

3.8.1.5 Stress due to bending on 128 teeth gear

Known: F =0.188", P = 32, w_2 = 4.028 rpm, Y = 0.454 (table 14-2), $\sigma_{allow.}$= 65000psi assume cut or milled profile

$$\text{Pitch} - \text{line Velocity}, \text{V} = \frac{\pi d_2 w_2}{12} = \frac{\pi \times 4 \times 4.028}{12} = 4.22\ ft/\min\ (3sf)$$

$$K_v = \frac{1200 + V}{1200} = \frac{1200 + 4.22}{1200} = 1.004$$

$$\sigma_{1bending} = \frac{K_v F_{12}^t P}{FY} = \frac{1.004 \times 23.75 \times 32}{0.188 \times 0.454} = \mathbf{8940\ psi}$$

$$Safety\ Factor = \frac{\sigma_{allow}}{\sigma_{1bending}} = \frac{65000}{8940} = \mathbf{7.27}$$

3.8.1 Motor mounting bracket deflection

Assuming bracket as a cantilever beam and that force acting on the motor shaft is equal to the force acting on the mounting we can estimate the deflection and normal stress on it. See figure 13 for modeling diagram of motor mounting bracket.

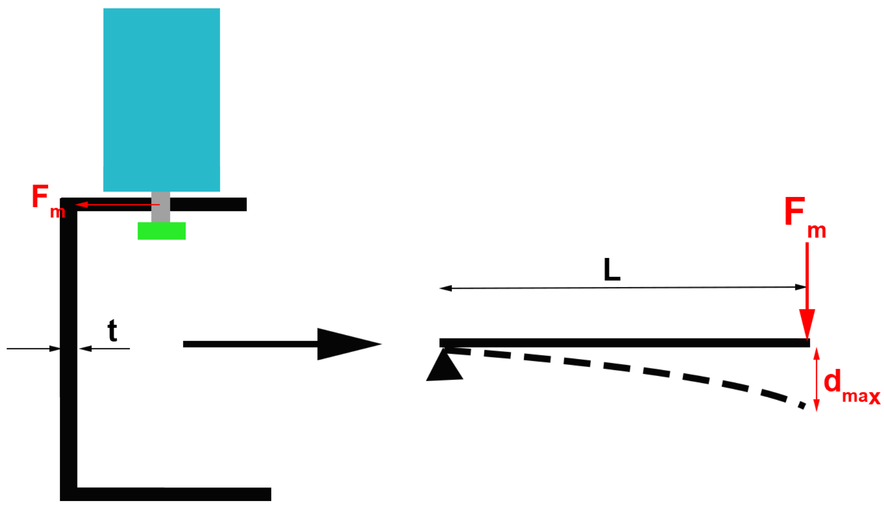


Figure 13: Representation of motor mount as cantilever beam

Known: L = 2.5”, t =1/4” , a = 0.6, F_m =24.5lbf, E = 69 GPa = 10Kpsi,

$$I = \frac{t \times L^3}{12} = \frac{0.25 \times 2.5^3}{12} = 0.3255\ in^4$$

$$\delta_x = \frac{F_m a^2}{6EI}(3L - a) = \frac{24.5 \times 0.6^2}{6 \times 10 \times 10^6 \times 0.3255}(3 \times 2.5 - 0.6) = \mathbf{3.12 \times 10^{-6} in}$$

Maximum normal stress

$$\sigma_{max} = \frac{My_{max}}{I} = \frac{F_m(L-a)\,{}^{t}/_{2}}{I} = \frac{24.5(2.5-0.6)\frac{0.25}{2}}{0.3255} = \mathbf{17.9\ psi}$$

From our calculations we determined that both the normal stress and the deflection on the motor mounting bracket are negligible. Hence, we do not expect our motor mount to fail during operation.

3.9 Final testing results

The energy conversion and transmission of the mechanism was tested on total travel time (t), decelerated speed (w_d), and power (P). Results were acquired using LabView, we measured a travel time of 8.907 seconds, decelerated speed of 9.1 degrees per second, 7.5 Watts peak power and 4.671Watts average power. Figures 14 shows plots of the results obtained in the final testing using LabView.

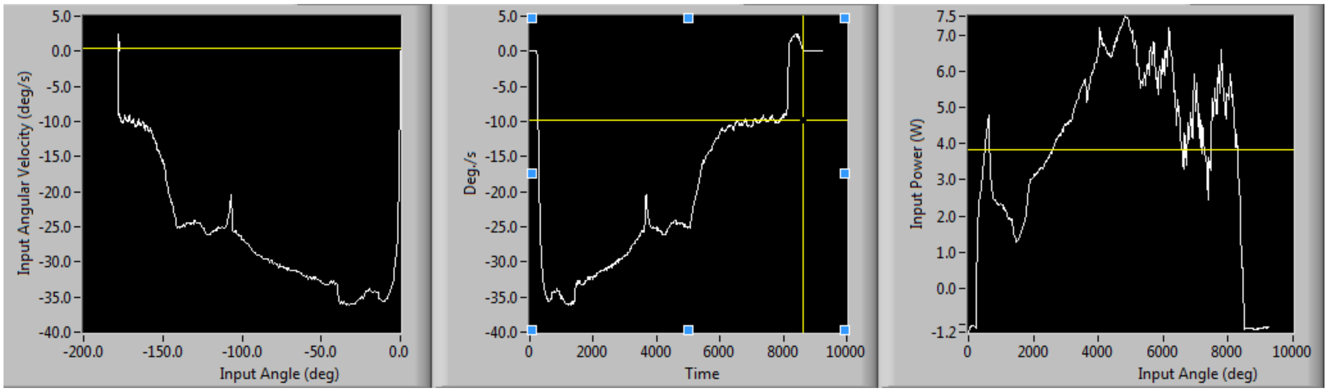


Figure 14: Plot of final energy conversion and transmission test results

The final results were very similar to the values we expected (see Table 15). The travel time was a little faster than expected by 0.1 seconds that can be considered insignificant. Our decelerated speed was 0.9 seconds slower than the maximum allowable by the project requirements. The difference between the expected and the final results is not very large. One of the major reasons for them to be different is that the values we expected were estimated assuming the ideal case were the motor instantaneously decelerated and friction was minimized. With the PI controller we had greater control over the speed, but did not eliminated the error between the desired decelerated speed and actual speed. The final power drawn was significantly lower than the expected power drawn. This difference is due to the fact that we evaluated the expected value using the peak maximum torque measured during the mechanism motion, and in the final testing we measured the average power drawn. If instead we calculated the average torque throughout the entire motion and used it to calculate the expected power drawn we would get a similar value to the final testing result. Overall the final results meet our expectations and the project requirements.

Results	Travel Time (seconds)	Decelerated Speed (deg/sec)	Power (watts)
Expected	9	10	17.1
Final	8.907	9.1	4.671

Table 15: Expected and final results of energy conversion and transmission testing

4.0 Safety & Motor Controls

4.1 Introduction

The motion of the backpack is controlled with precise timing and a robust safety system, as well as a design driver to reduce the power drawn. The motion of the mechanism is controlled using an Arduino Uno microcontroller board. The user will use a rocker switch to direct the forward and backward motion of the mechanism. The safety system includes two limit switches, which will turn off the motor when they are tripped, as well as two infrared proximity sensors, which will stop the motor if someone or something is in the system's path of motion.

4.2 Sensor & Switches Capabilities

4.2.1 Infrared Proximity Sensors

The Sharp GP2D120XJ00F infrared proximity sensory (IR Sensor, see Figure 15) is an analog sensor. It detects the distance to a nearby object without touching it. It is capable of detecting objects up to one and a half feet way and off by up to 45 degrees from the sensor's direct line of sight. Upon calibration of the sensor, it was found that the sensor readings fluctuated even if an object was at a constant distance away. However, we determined that the fluctuation was negligible. In addition, we found that the size of the object did not affect the sensor's reading. The sensors' capabilities are sufficient for the mechanism's needs, as we are interested in identifying objects very close in proximity, and thus did not require a large sensory range. We also exploited the fact that the sensor can detect objects at an angle offset to increase the safety region mechanism. Further, we can change the threshold values of our sensors in the Arduino code to manipulate the distance at which it will detect an object (i.e. 10 cm away from the sensor).

Figure 15: Infrared proximity sensor and sensor's beam

4.2.2 Limit Switch

The Omron Subminiature Basic Switch (part number 187733, see Figure 16) can detect the presence of an object within a very short range; it detects physical contact with the object. However, they do not act as hard stops, as the lever and the switch housing are not designed to absorb the load of the moving mechanism. Nonetheless, the sensor can still be used to detect when the mechanism reaches the hard stops. The roller at the end of the lever aids to handle high point forces that may be destructive. The short lever makes the sensor very sensible, allowing the sensor to be activated from a small force. The sensor capabilities allow it to be a good candidate for telling the motor encoder when the mechanism has reached a hard stop.

Figure 16: Limit Switch

4.2.3 Rocker Switch

The Jameco rocker switch (part number 35-695, see Figure 17) is a single-pole, double throw switch. It is a momentary rocker switch – when neither side is pressed, a spring returns it to the default position of center, where it is connected to neither one of the throws. Therefore, the switch has three possible states which can be use to set a forward, backward, and stationary state for the mechanism's motion.

Figure 17: Rocker switch, Single-pole. Double-throw

4.2.4 Encoder

The motor used contains a Hall Effect encoder (see Figure 18) with two channels, A and B. It provides a digital output as a result of a linear or angular displacement. Hence, it will allow us to measure the angular displacement of the mechanism. The two channels allow it to be possible to identify when the motor is rotating clockwise or counterclockwise, in our case we would be able to identify when the mechanism is moving forward or backward. The encoder has two threshold values, lower and upper. The lower value is when the input link is 30 degrees from the home position and the upper value is when the input link is 30 degrees from the final position. With additional control from the Arduino, we were able determine when the motor speed needed to be changed to meet the project requirements.

Figure 18: Hall Effect Encoder

4.3 Threshold values calculations

4.3.1 IR Sensors

No formal numerical calibration of the sensor could be done as the sensor reading fluctuation was small in magnitude by high in speed. Therefore, a trial an error approach was done to determine the optimum threshold value for the sensor. The closer you get to the sensor the smaller the reading gets. The threshold value was set to 60. Also the threshold value was set high enough to ignore noise. This distance was selected so that neither the operator, wheelchair, or mechanism components activated it.

4.3.2 Encoder

The encoder threshold values were chosen to ensure that the mechanism would slow down and rotate at a maximum of 10 degrees/second for the last 30 degrees of motion to meet project requirement. The lower and upper values were set to 2795 and 13042 respectively. The calculations to determine the threshold values are in Appendix D.

LowerCountTreshold calculation, assuming an 8:1 transmission ratio:

$$30\text{ deg} = 30\text{ deg}\left(\frac{1\text{ rev. input link}}{360\text{ deg}}\right)\left(\frac{8\text{ rev. gearbox}}{1\text{ rev. input link}}\right)\left(\frac{131\text{ rev. encoder}}{1\text{ rev. gearbox}}\right) = 87.3\text{ rev. of encoder}$$

$$87.3\text{ rev.} = 87.3\text{ rev.}\left(\frac{32\text{ counts}}{1\text{ rev.}}\right) = \mathbf{2795\ counts}$$

UpperCountTreshold calculation:

$$170\text{ deg} - 30\text{ deg} = 140\text{ deg}$$

$$140\text{ deg} = 140\text{ deg}\left(\frac{1\text{ rev. input link}}{360\text{ deg}}\right)\left(\frac{8\text{ rev. gearbox}}{1\text{ rev. input link}}\right)\left(\frac{131\text{ rev. encoder}}{1\text{ rev. gearbox}}\right) = 407.6\text{ rev. of encoder}$$

$$407.6\text{ rev.} = 407.6\text{ rev.}\left(\frac{32\text{ counts}}{1\text{ rev.}}\right) = \mathbf{13042\ counts}$$

4.4 Changes to Arduino code

The Arduino Code used included a Proportional Integral (PI) Controller as it had superior motor speed control using feedback loops and auto-corrections. The code constantly imported data from the two IR sensors, two limit switches, rocker switch, and the motor's encoder. With all this values the code controlled the motor and set in what direction and speed it should operate to meet project requirements. The main modifications to the code included the previously stated encoder's lower and upper threshold values and the IR sensor's threshold value. In addition, we modified the motor's low and high speed values to achieve a total travel time of 9.0 seconds, and an angular velocity no greater than 10 degrees/second during the last 30 degrees of travel. The speed values had to be sufficiently large to provide enough torque to move the mechanism without any lack. We set the motor's high and low speed values to 340 and 120 degrees/second respectively. To determine these values we used a trial and error approach until we satisfied the project requirements.

4.5 Sensor mounting

4.5.1 Considerations

The placement of all sensors and switched had to ensure safety for the operators and other around, and at the same time make the mechanism easy to operate. Additionally, we wanted to make as little modifications to the original design to prevent any negative effects on the volume or offset values of our mechanism. The limit switches have to be mounted in locations where the lever will be moved by the mechanism just as it contacts the forward and reverse position hard stops. The proximity sensors should be mounted in areas that could be considered hazards or "pinch points". However, at the same time we want locations were it only detects objects in these areas without detecting the operator, wheelchair or mechanism.

4.5.2 Method of mounting

The design of our mechanism had already accounted for the placement of all sensors and switches. Hence, no extra mounting brackets were required to be manufactured. The two limit switches were mounted at each side of the ground link and next to the hard stops. One IR sensor was placed in on top of the bigger metal gear to prevent harm from the gear teeth or the tangle between the gears contact point. The gear teeth rotating at high speeds can cause severe cuts on a human. The other IR Sensor was placed in the inside end of the coupler to detect any objects on the motion path and prevent it from getting squashed against the wheelchair. Rocker switch was placed in the left armrest of the wheelchair for easy access for operator.

4.5.3 Effect on volume and offset

The placement of all sensors and switches had no effect on the volume or offset values of the mechanism.

5.0 Design Critique & Evaluation

While designing, manufacturing, and testing our linkage system, we were met with challenges and successes. One of our main challenges was in our initial gear design. We initially planned to fit a small 4:1 gear transmission inside our mechanism. However, after reviewing our torque and load calculations we found that we instead would need a larger gear ration of 8:1. Consequently, we needed to use larger spur gears with our mechanism, which would not fit in our initial design. We thus had to redesign our method for mounting the motor and the gears to our mechanism. Despite this set back we were able to adapt our design and machine the necessary parts to incorporate the larger gears. This proved to be a great example of one of our strengths. More specifically, we were able to adapt and machine our design easily and with high craftsmanship. We also found that our methods for handling friction and deflection were very successful. Our use of bushings over needle bearings limited frictional affects and our use of solid aluminum bars as links ensured deflection was negligible. Nonetheless, our ability to manufacture parts with high craftsmanship, coupled with our design's adaptability was reflected in our mechanism's strong performance during testing.

Our final design was able to perform relatively close to our models. However, two major differences between our final design and models lied in its ability to transmit power, and the time it took to complete its path of motion. As mentioned above, based on our models we initially believed that a 4:1 gear ratio would suffice. However, after testing, and further analysis we found that we instead needed an 8:1 gear ration. Our final design's rotational time was also less than our modeled rotational time by less than a second. However, we believe this can be accounted for due to systematic error, as well as the inability to completely model all frictional and resistive forces that act on our linkage system during operation.

Our decision to use steel spur gears, and an adjustable motor mount ensured that we not only had a well meshed gear system, but also allowed us to handle large loads. However, as expected, adding a load to our linkage system did change its performance. The addition of a load increased axial friction from the applied torque between the links, as well as the resultant additional translational friction that the load introduced to the system. Initially, we determined that needle bearings were going to be used to reduce axial friction. However, we found that the actual needles had a much larger gap than we had anticipated. We thus decided to change our design to use bushings instead. This switch resulted in low axial friction, improved axial tolerances, and thus allowed our linkage system to operate better than we had anticipated. This also led to an improved alignment throughout the motion of the linkage system during operation. In addition, the translational friction was reduced through the use of thrust bearings, which we had chosen in our initial design stage. Overall we did not find friction to be a performance-hindering factor in our linkage system during operation.

While we were able to control how forces were handled in our linkage system, we were limited in our ability to control the actual motion of our linkage system. More specifically, we were limited to the algorithm used in pair with the provided Arduino. While we were able to fine tune some of the constraints in the algorithm, we still believed that our motion could have been further improved through a different algorithm. More specifically, it would have served us well if we had been able to better incorporate the electronic positioning and resetting of the motion cycle within the encoder. In addition, the physical position of the external infrared sensors and limit switches further inhibited the performance of our linkage system. Consequently, consistently and accurately re-zeroing the position of the linkage system within the program became a difficulty. This thus affected the overall performance of our linkage system during operation.

In addition to considering the performance and reliability of our linkage system in comparison to the requirements set before our team, we also needed to consider the safety of the operator and those around the operator. We addressed safety through the use of hard stops and infrared sensors. We used hard stops to ensure that the linkage system could not rotate outside our intended path of motion and used infrared sensors to stop the motion of our linkage system in the event that an object or person moved into the system's path of motion or near the gear system.

Having used the allowed sensors in the best manner possible, we believe our mechanism, if ranked on a scale of 1-5, 5 being the best, would receive a safety rating of 4.5. Our mechanism's safety rating could be improved through the use of a guard over our gear system, ensuring that both a bystander and the gear itself would be protected from any possible contact. We could have also used additional infrared sensors or a different kind of sensor, such as a photovoltaic sensor. This would have allowed us to cover a wider range of our linkage system, reducing the chances of harmful contact with the mechanism.

The process of designing, manufacturing, and testing our mechanism has taught our team some valuable lessons. When initially designing our linkage system, we did not think to incorporate enough room for a large gear ratio. Consequently, after finding that we did in fact need a larger gear transmission, we were forced to modify our design and manufacture additional parts to fit our needed 8:1 transmission. This was a major initial flaw in our design. However, as our design was easy to adapt, we were able to make the necessary modifications and maintain a sleek and effective design, without limiting any other aspects of the design. Having initially designed a mechanism that was relatively simple to modify, allowed us to adapt our design and solve problems that arose during the manufacturing and testing of our mechanism. We also learned that deflections did not pose as large of a problem as we thought. Knowing this, we would have used the provided aluminum stock to manufacture our links instead of purchasing solid aluminum to manufacture our mechanism, saving on material, weight, and cost.

Although our linkage system was able to successfully operate and meet the prescribed requirements, we still would not recommend its commercial yet, as our design could use a couple modifications and improvements. In addition to addressing the safety measures listed above, the design could be optimized to reduce the amount of torque on the gear teeth. This is important, as we noticed that during the operation of the linkage system, the torque caused the motor to shift in position when loaded with large torques. Although this did not prevent our system from working, we believe it should still be addressed. In addition, the starting position of our linkage system is extremely close to the gear, this, coupled with the position of the backpack holder itself, makes it difficult to attach a backpack to the backpack holder. We also believe that our design should be modified to hold the mechatronics system better. Nonetheless, while our current linkage system operates exceptionally well, we believe it could use a few more modifications before being put into commercial use.

Despite having been able to successfully meet the prescribed requirements for this project with the given materials, we believe that the addition of some other materials would have been better. It would have been helpful if we had been given some of the transmission pieces in the beginning, as teams would have been able to test different transmissions at an earlier stage. This would have also saved materials and money for each team. In addition, if teams were given more labs set up for testing and building their designs, we believe more teams would have been able to see problems in their designs earlier, and thus addressed them with more time, leading to a better final design.

Appendix A – Individual Designs

Initially, each team member came up with an individual design to solve the problem of moving a backpack on a wheelchair. Each of the five designs was synthesized through Lincages, modeled using Solidworks, and analyzed using Adams.

Design #1: Michael Kamen

This design implemented a simple four-bar mechanism to move the backpack from the rear of the wheelchair to the right side, while having the backpack facing the user the entire time. The main problem was ensuring that the linkage was long enough to ensure the user would not have to reach linkage system them to get his or her belongings while in class.

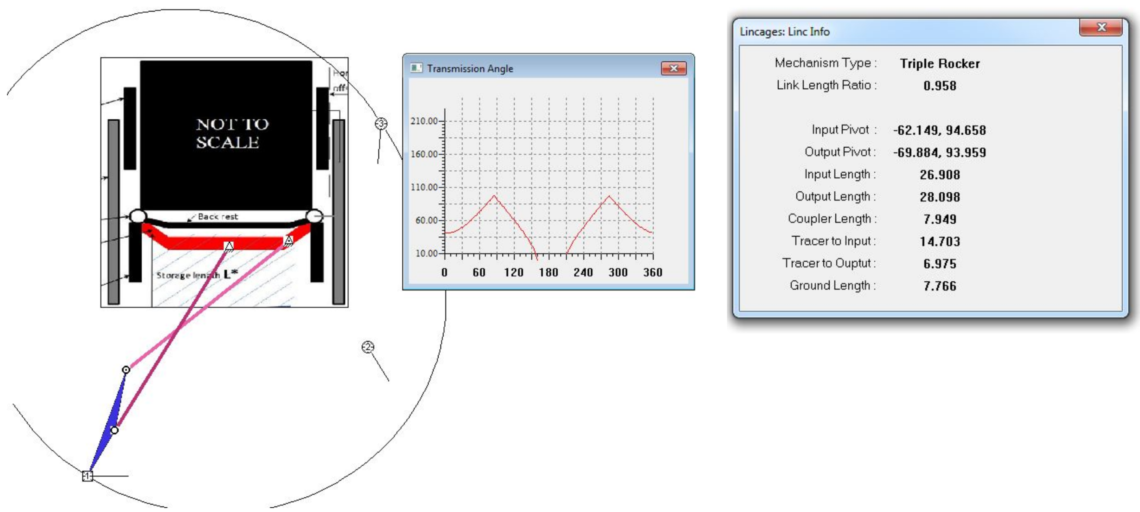


Figure A.1 Lincages 2000 results of mechanism synthesis analysis. Transmission angle and mechanism dimensions

The initial design was created using Lincages. Basically, this was a simple model just to get a feel for what needed to be made. Although the model was not to scale, it gave some valuable information as to the general motion of the needed mechanism.

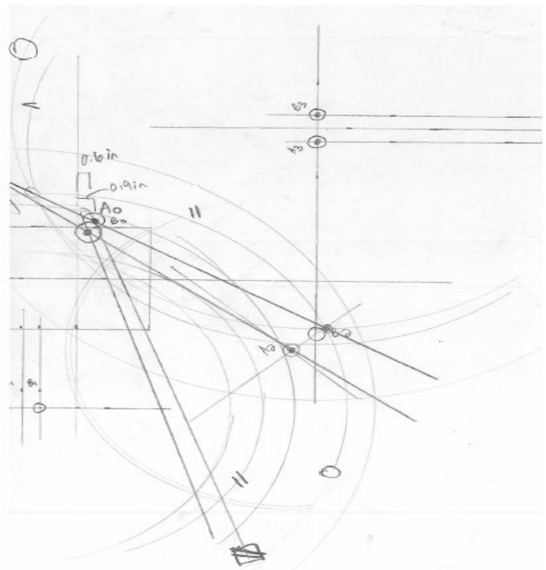


Figure A.2 Manual Synthesis

Next, a manual synthesis was generated to get some idea of the dimensions of each link and how far each link had to move. Using three precision points, and a scaled drawing, the needed forward offset and lateral offset were achieved so that the backpack could be used effectively. The conclusion from this synthesis was the location of the precision points of the coupler arm and the link length.

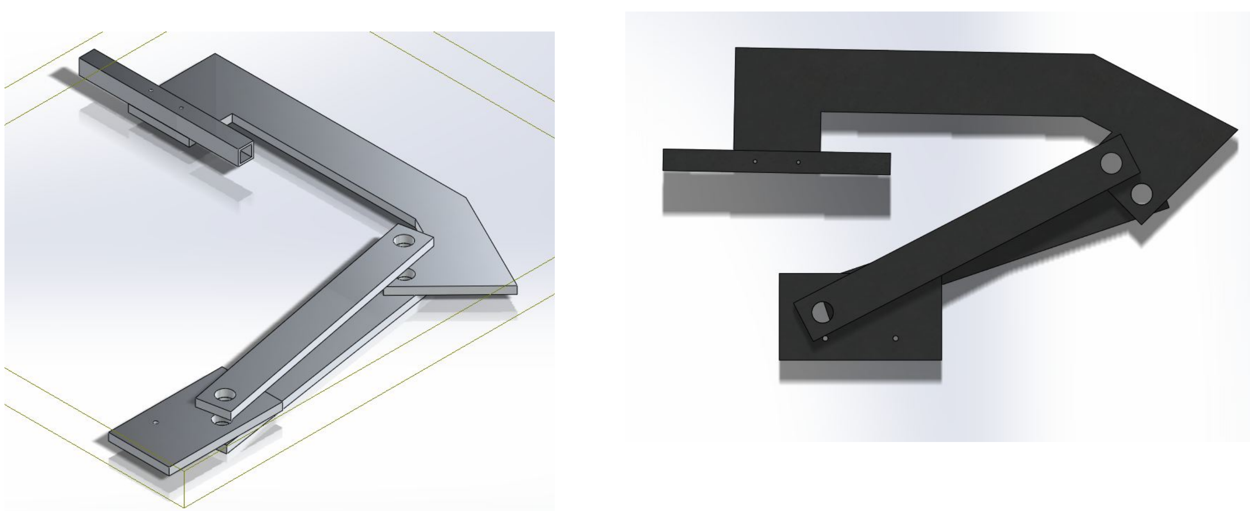


Figure A.3 CAD Model

Next, a Solidworks model was generated to show an actual design solution. Using the precision points from the last synthesis, it was determined that the coupler would have to be oddly shaped to satisfy the needed forward offset. In this model, the backpack faces the user the entire time to make the backpack easily accessible. Some problems with this design included a large moment exerted on the mounting plate due to the extension and some of the links overlapping the joint holes when it turns. This could pose difficulty when adding hardware such as bearings, shoulder bolts, etc. When the mechanism is in the starting position, it has a relatively small footprint. There is still enough room to put the backpack on the backpack holder, and the links don't extend past the boundaries of the wheelchair.

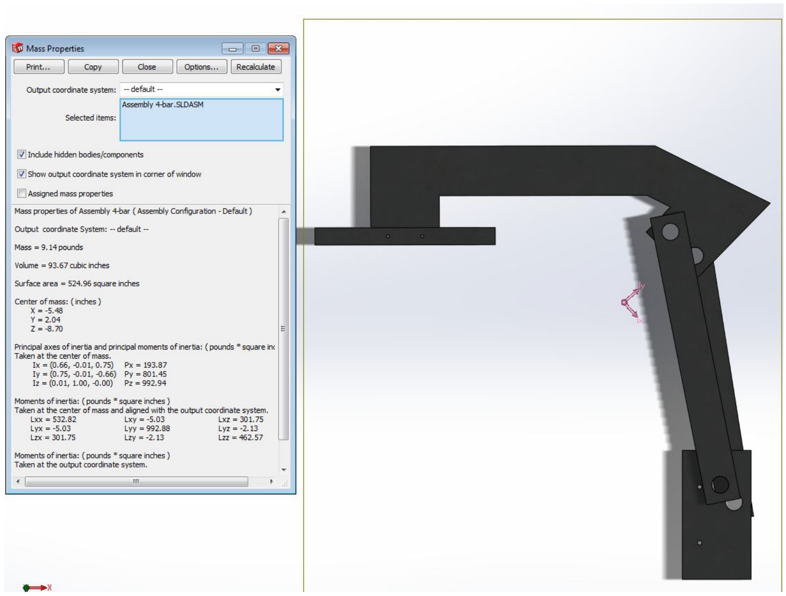


Figure A.4 Overview

When the linkage is fully extended, it reaches the desired length and forward offset without hitting the wheelchair or the wheels of the wheelchair. The final weight of the entire assembly was 9.14 pounds, which is rather heavy. After the backpack is attached, the entire assembly will be over 20 pounds.

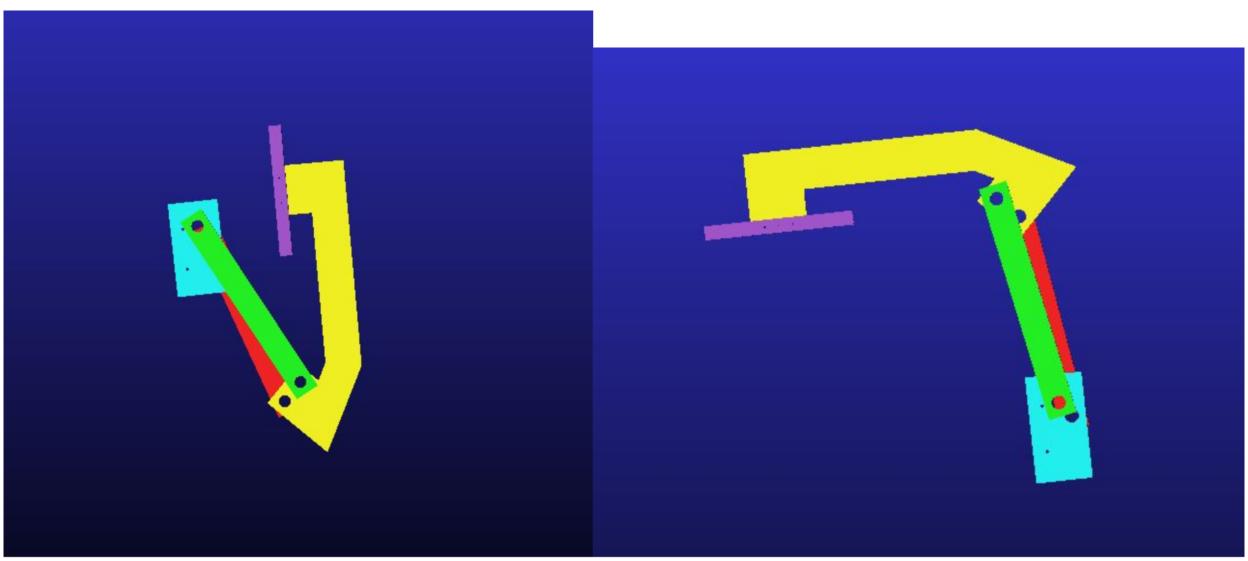
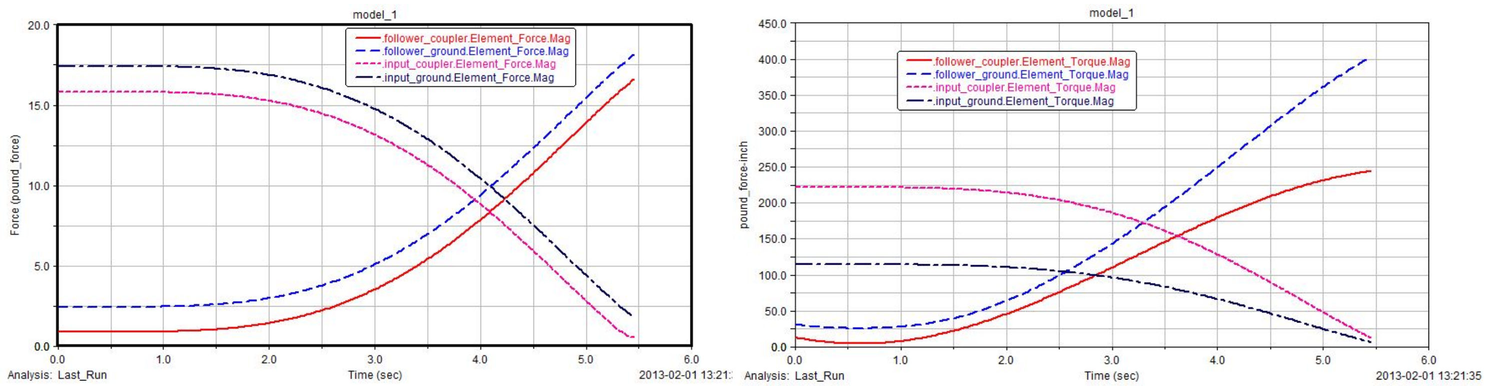


Figure A.5 Mechanism in Adams - view

Once the model was complete, it was uploaded into Adams, an analysis program for mechanisms. We used this to see the forces and torques on each of the links in the device. Each joint was created and named. Then a simulated motion was added to the input link to make the arm go through the desired motion.



(a) (b)

Figure A.6 Load analysis of design 3. (a) Plot of force magnitudes on joints, max. force of 18 lb (b) Plot of torque magnitude on joints, max. torque of 400 lb*ft.

After the simulation, plots representing the magnitudes of the forces and the torques on each link were generated to see the maximum torque and force that would be applied to the respective joints. Although there was large torques acting on the joints, this is mostly due to the overall distance the backpack is from the axis of rotation at the ending point. The moment at the end point is the greatest. To reduce the forces on the bearings that will be added, it would need 2 thrust bearings per joint. This will constrain the joint and the thrust bearings will enable it to take more loads in the downward direction. The bearings will also be rather large to reduce the strain on the bearing race.

Design #2: Joe Klosowski

The goal of this design was to make a sleek simple design that was collapsible in the starting position. This would make the linkage less bulky from behind when traveling from place to place. I started out by examining the wheelchair adapter that was provided and chose two ground points that would be both easily accessible, as well as sturdy. With this information I opened Lincages 2000 and planned the desired travel route that I desired.

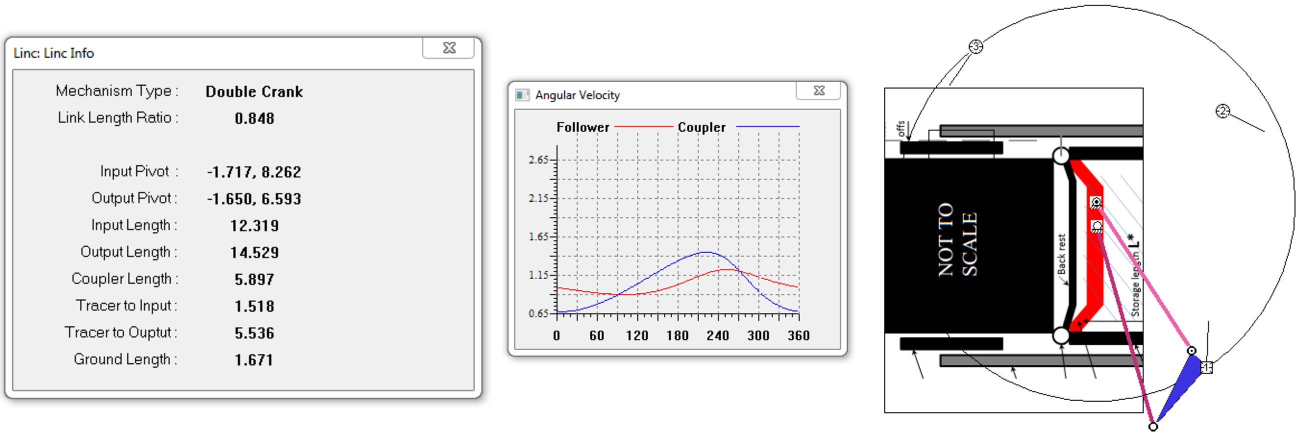


Figure A.8 Lincages 2000 results of mechanism synthesis analysis. Transmission angle and mechanism dimensions

Once I obtained the correct dimensions in Lincages, I was able to move forward with my design in Solidworks. The dimensions of the links and ground points were critical. However, I had the freedom of choosing a coupler design that would be collapsible and long enough for the user to obtain their backpack items. As a lot of weight would be hanging far away from the chair creating a large moment arm, the material had to be both light and strong. I choose 6061 Aluminum based on its material properties, as it would minimize the weight without sacrificing strength.

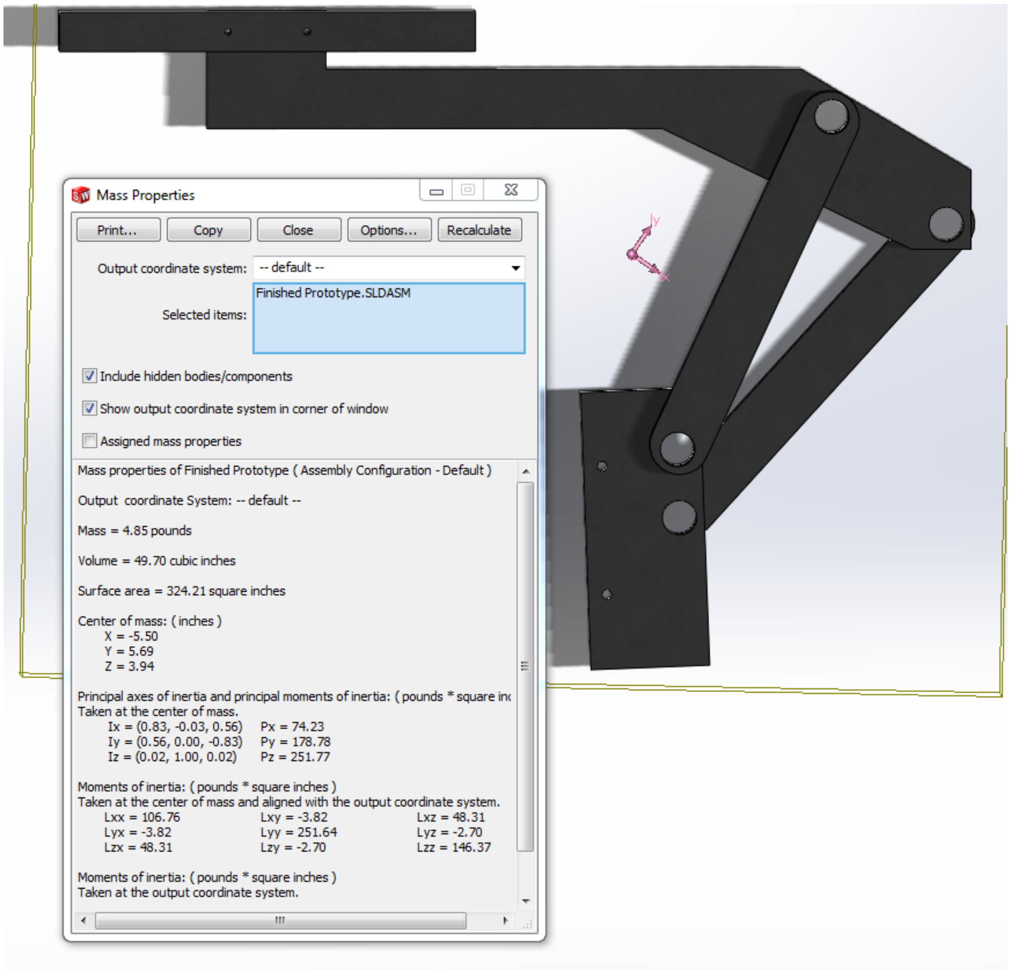


Figure A.9 Overview

Surprisingly, the mechanism only weighed 4.85lbs and had a low volume of 49.70in^3.

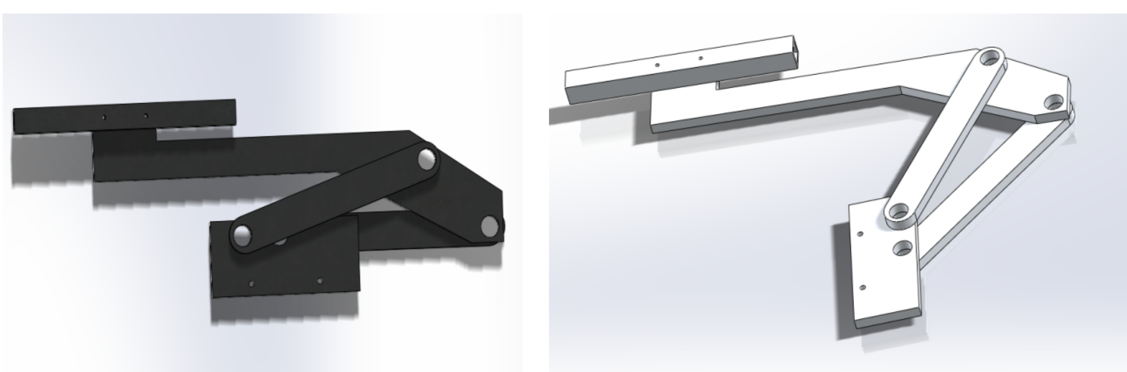


Figure A.10 CAD Model in starting and ending position

The Solidworks assembly simulated the rotation of the desired path. At this point I was very satisfied with my design. The only drawback of the design was that the user would have to reach over the linkage to obtain his or her items. The rotation and translation of the backpack was successful. In order to further analyze the linkage, I uploaded the mechanism into ADAMS. This way I could run an actual simulation and see the forces on the joints. Adding a motor motion proved that the design was probable and could actually run while supporting a load.

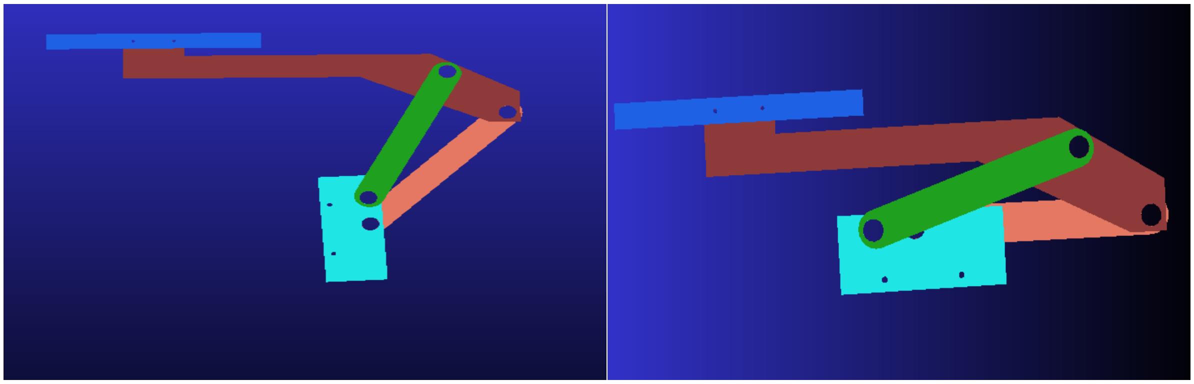
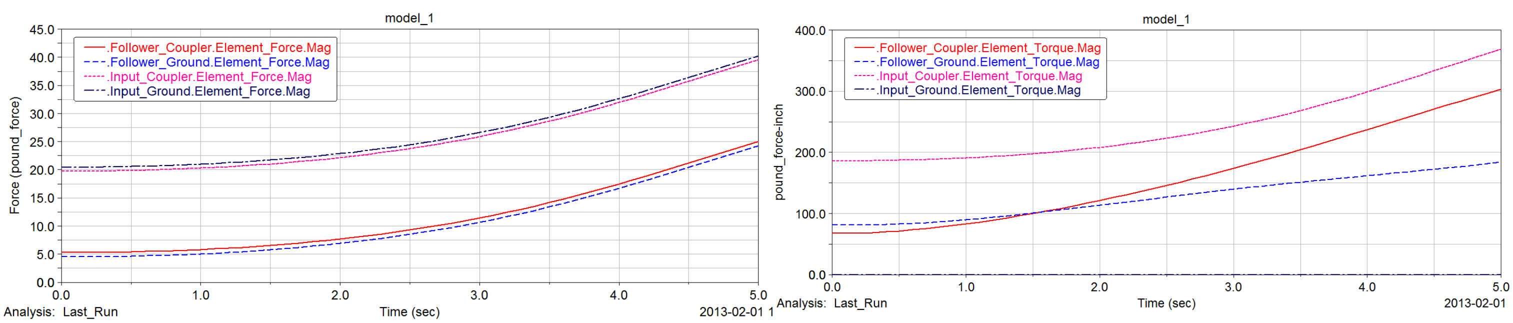


Figure A.11 Adams Simulation

The ADAMS software provided useful information such as the forces and torques on the joints. This information is a useful factor when determining which design to go with. If the torques at the joints were too large the design would have to be reconfigured.



(a) (b)

Figure A.12 Load analysis of design 3. (a) Plot of force magnitudes on joints, max. force of 40 lb (b) Plot of torque magnitude on joints, max. torque of 380 lb*ft.

The next step I faced was to configure the style of joints I would use to perform this task with the given loads. The major concern was managing the load and performing the motion numerous times. After running an ADAMS simulation, I found that the loads on the joints were large. The large overhanging coupler design would be responsible for this. However, these loads could be easily managed y selecting and using the correct joints. I will use large thrust bearings at the joints to absorb the vertical force exerted by the backpack. There will be double supported joints for improved robustness. This will also help reduce the friction and make the linkage rotate easily.

This design meets the requested standards and performs the motion well. I think this will be a good design to base off or use as a final design. The manufacturing, motion, and main objectives are simple and can be easily repeated. Overall, using multiple computer software's following the design process provides us with a viable product that would be an asset to a handicapped individual.

Design #3: Sam Kapor

The purpose of this design was to use a simple four-bar linkage system and to use the least amount of parts as possible. It was also intended to be easily adaptable and constructed.

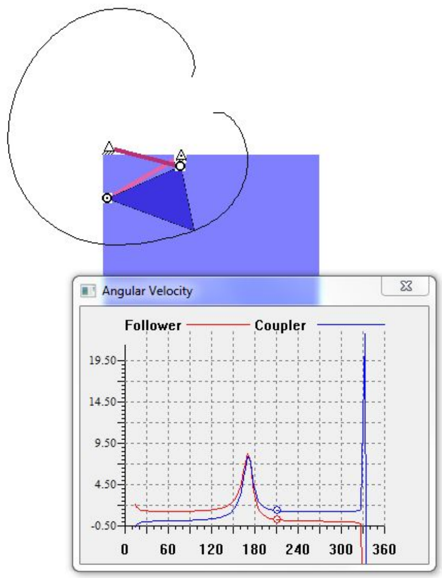


Figure A.13 Lincages 2000 results of mechanism synthesis analysis, mechanism dimensions

Using Lincages 2000 software, I found a four-bar linkage system that would allow for simple motion that would start with the backpack behind the wheelchair and end with the backpack facing outwards on the right side of the wheelchair. After checking that the angular velocities based of this system were not too high, I began to construct a model. I then created a model in SolidWorks, using the size and ratios given by Lincages 2000. I first designed a simple mounting plate that would allow the linkage system to be easily attached to any point on the mounting bracket of the wheelchair. I then designed the coupler to allow the backpack holder to be easily attached and moved around as necessary, making it adaptable around the wheelchair. To avoid large bending moments, forces, and torques, I designed the input and output links with relatively small lengths. This also was intended to ensure that there were minimal deflections throughout the linkage system.

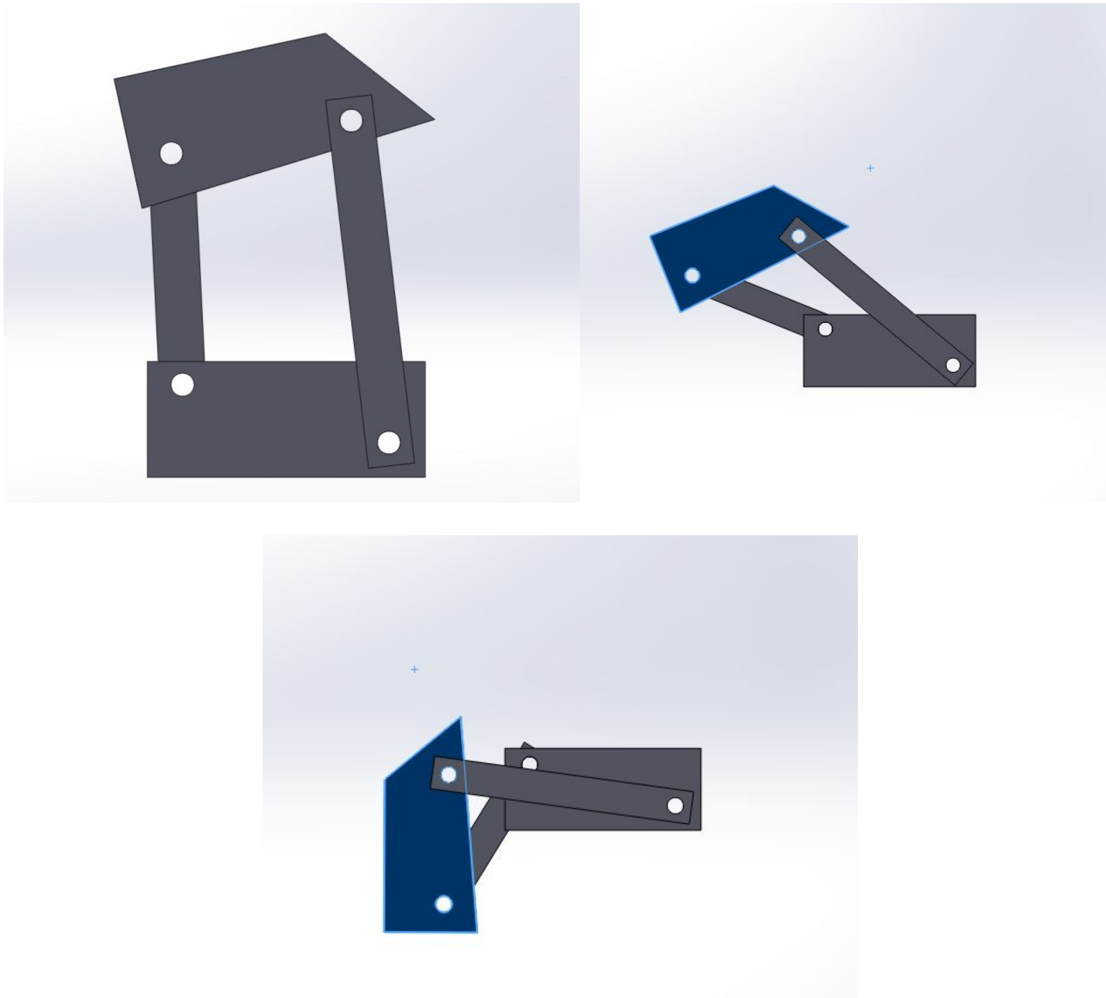


Figure A.14 Top view of design 3 in starting, half-way and end positions.

In addition, the links, mounting plate, and coupler, were all designed such that most parts could be easily machined without any complicated angles or dimensions. Most of the parts resemble either rectangles or a combination of rectangles and triangles. The lengths and dimensions of the parts have been chosen to minimize volume and thus the weight of the design. In addition, this has resulted in smaller joint forces and torques, as well as bending moments.

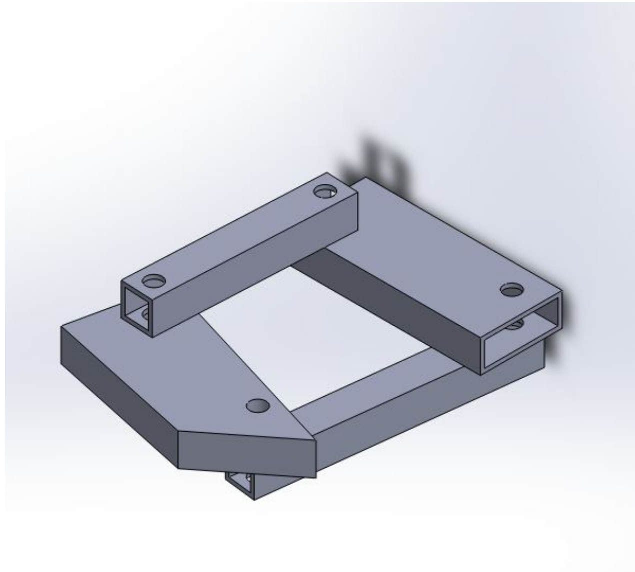
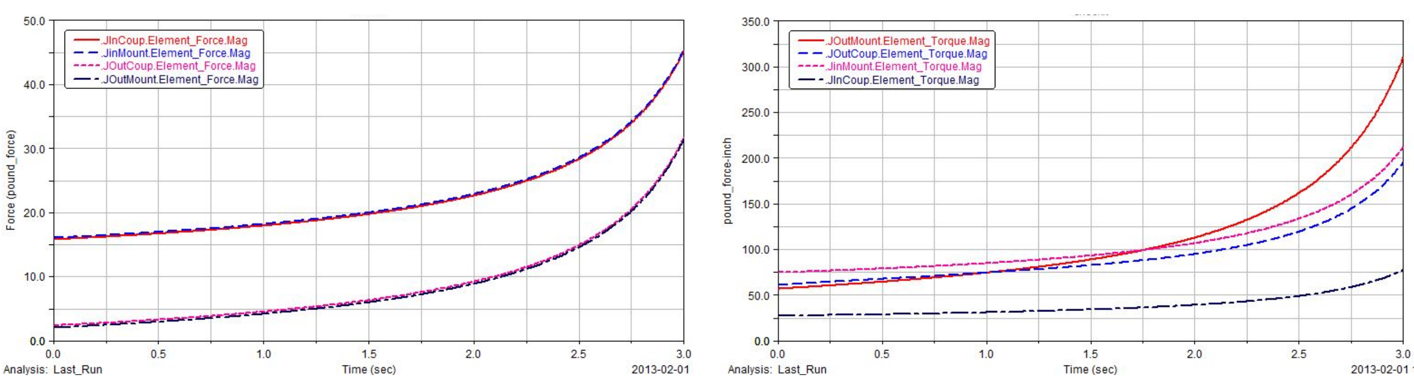


Figure A.15 Isometric view of CAD model of design 3

After creating a model in SolidWorks, I then took the design and used ADAMS to determine the forces that the design would undergo if it were to be used. After running a Force and Torque analysis in Adams, the results did not prove to be too high.



(a) (b)

Figure A.16 Load analysis of design 3. (a) Plot of force magnitudes on joints, max. force of 45 lb (b) Plot of torque magnitude on joints, max. torque of 225 lb*ft.

After running an analysis of the motion of this linkage system it was found that managing the loads and torques on the joints would be relatively easy as the resulting torques and loads were relatively low. However, due to the simple design, the end position of the linkage system is not optimal. To address this, the linkages could be lengthened. However, this would result in greater moment arms, and thus a greater load on the joints. Nonetheless, currently the loads and torques on the joints are manageable, however if the links were to be lengthened, larger bearings and thus different joint mechanisms would have to be considered.

Design #4: Roberto Shu

This four-bar linkage was designed around the idea to have a mechanism simple to manufacture and assemble. It was desired to be a minimalist design that operates with a reduced weight, volume, and probability of failure.

The first step was to design a successful four-bar-linkage that could achieve the desire motion from the back of the wheelchair to the right side using Lincages 2000. Figure A.17 shows the linkage design in the software package, with the coordinate of each link. Using the link-length ratio, I developed a CAD design using SolidWorks that would be compatible with the dimensions of the wheelchair.

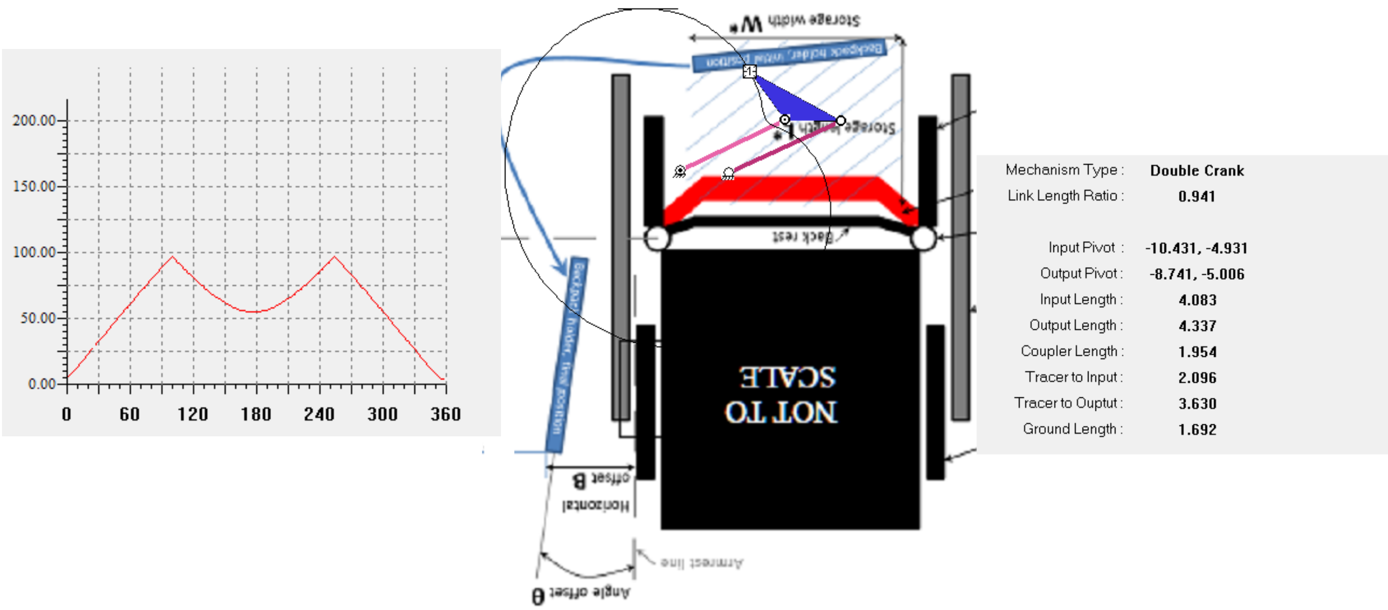


Figure A.17 Linkage design with coordinates in Lincages 2000

To simplify the manufacturing and assembling of the system when designing it in SolidWorks, I used square aluminum tubing for the input, output, and coupler link. The ground link design is made out of a simple aluminum plate, which makes the ground link easy to adjust in size, volume, and support constraints. Not depicted on the CAD is the backpack holder, which will go on top of the coupler. This was removed to simplify ADAMS analysis at it will not affect results.

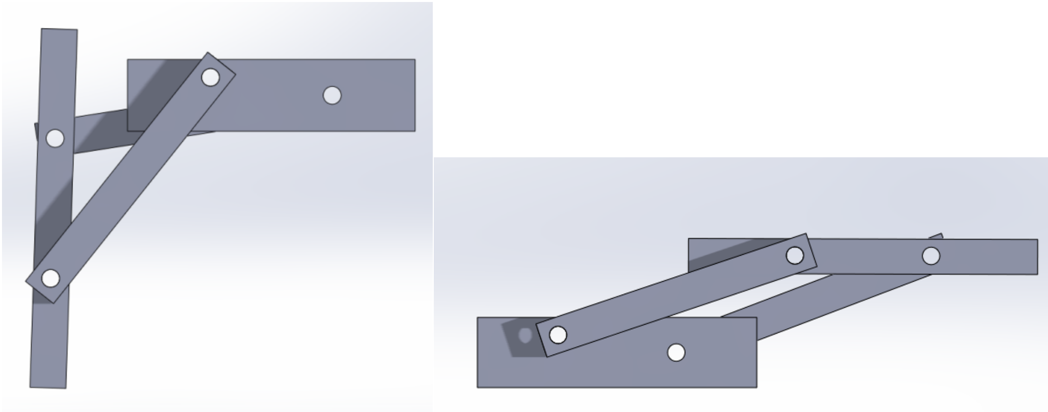


Figure A.18 Top view of design (a) initial and (b) final position.

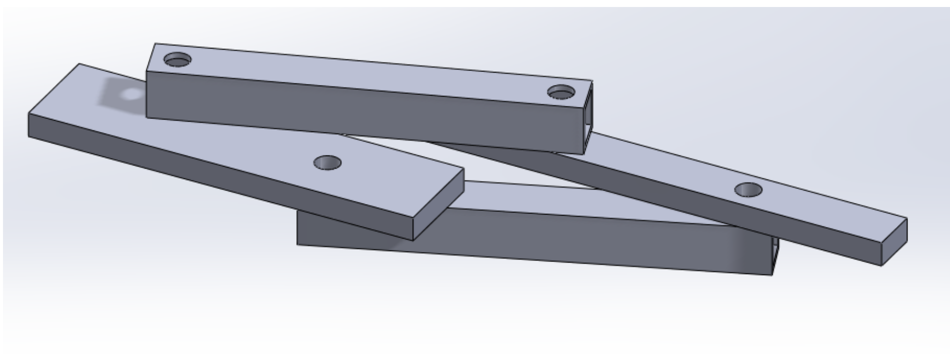


Figure A.19 Isometric view of CAD model of design 3

ADAMS analysis results are depicted in Figure A.20, forces on all joints picked to roughly 125 pound-force at the middle of the trajectory. This is not desirable and will be hard to handle. However, as the force is experience for a very short time the material will be able to support it. Also, the joints will be double supported to reinforce them.

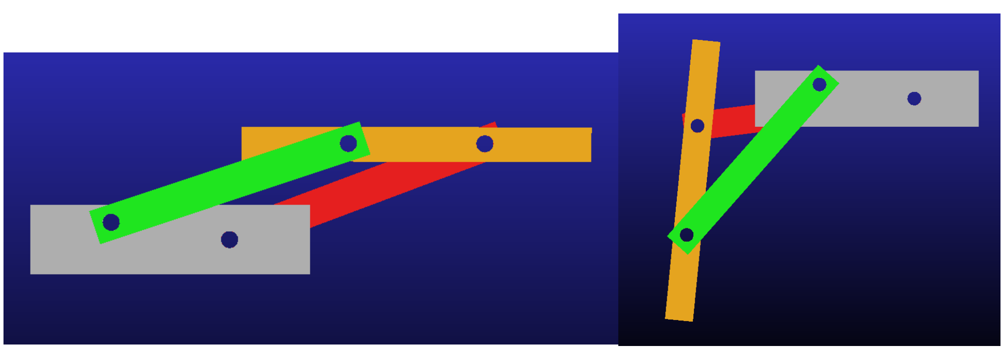
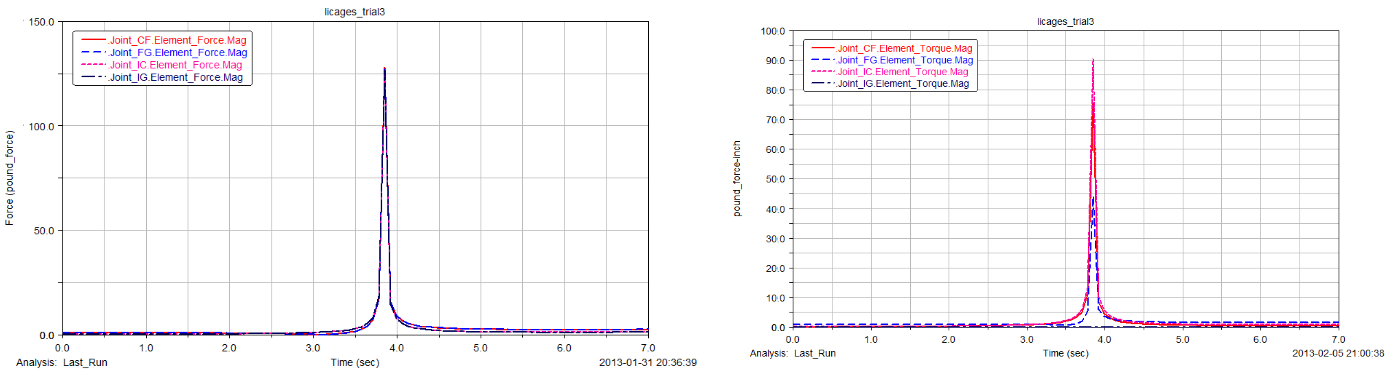


Figure A.20 Screenshot of design 3 in Adams



(a) (b)

Figure A.21 Load analysis of design 4. (a) Plot of force magnitudes on joints, max. force of 125 lb (b) Plot of torque magnitude on joints, max. torque of 90 lb*ft.

Design #5: Miguel Vidal

The main focus for this design was to obtain a simple and lightweight linkage system so we could keep low torques due to the carried mass. It was possible to create the system with straight bars therefore keeping weight for the components and overall system at minimum. The benefit of having an initial lightweight system allows the use of solid bars, resulting in a more robust system. Solid bars serve as a better support, which in-turn provides for stronger fittings reducing deflection in the linkage system. The first step was to create the motion with Lincages 2000. See figure A.22

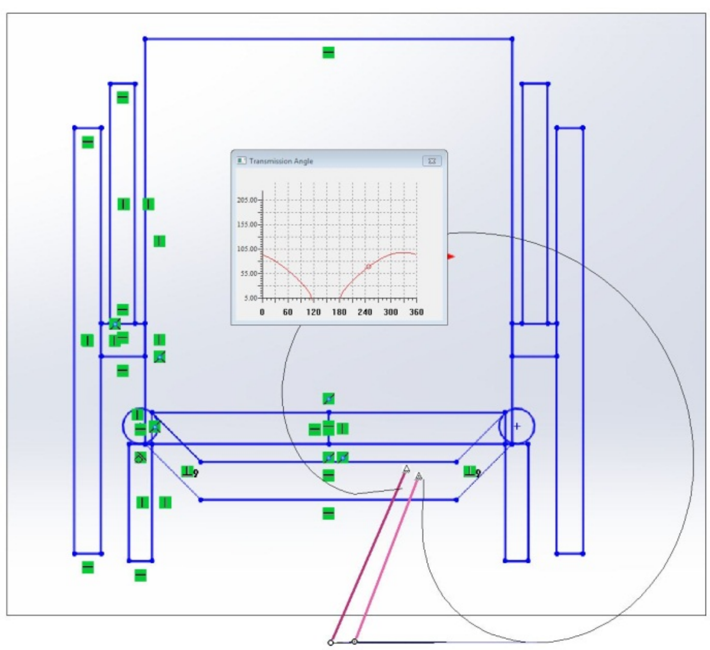


Figure A.22 Lincages 2000 results of mechanism synthesis analysis, transmission angle

With the right link ratios, link lengths, and pin locations, the next step was to design it in Solidworks to allow a visualization of the system mechanics a model motion. Different steps of the design's motion can be seen below in figure A.23.

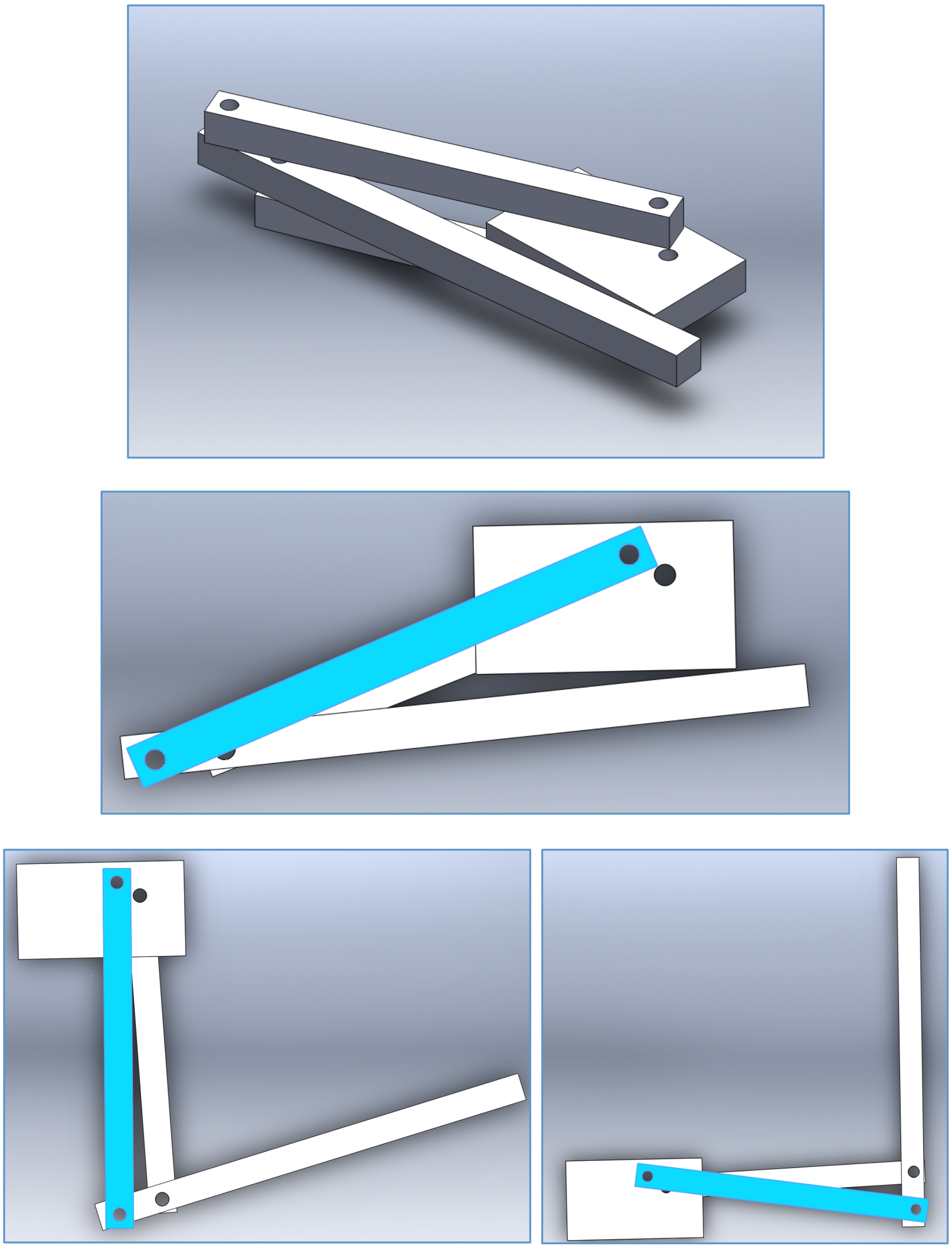


Figure A.23 CAD Model

The next step was to import the design into Adams and understand what the dynamic loads were and thus prevent further issues by improving the design.

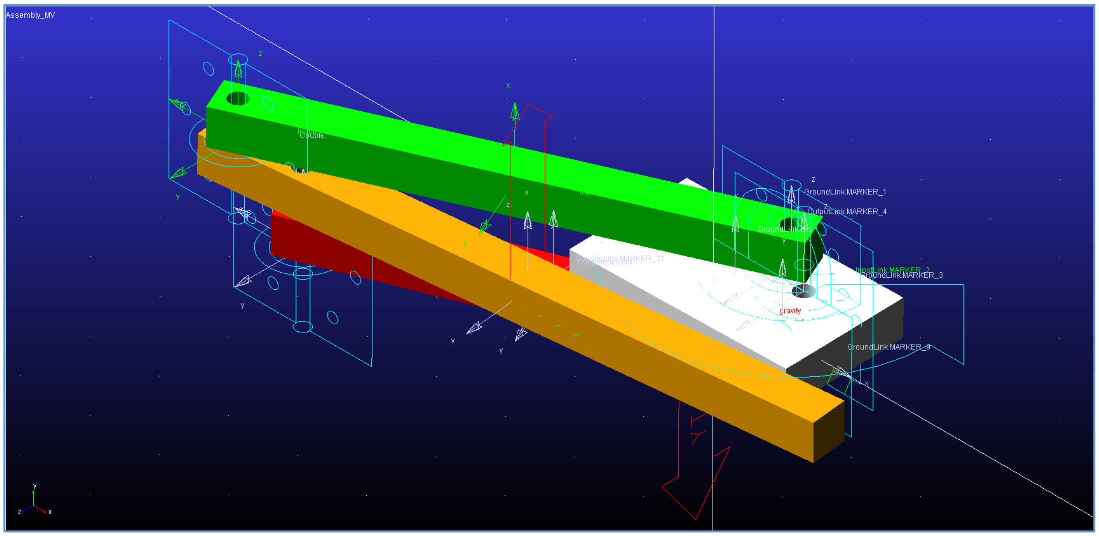


Figure A.24 Adams Model

Forces and Torques loads developed in Adams have shown very smooth transitions and overall low loads and torques applied while in cycle. We are finding a maximum force of 13.5lb at the junction between input and ground links, also location for the higher torques found to be of 140lb.

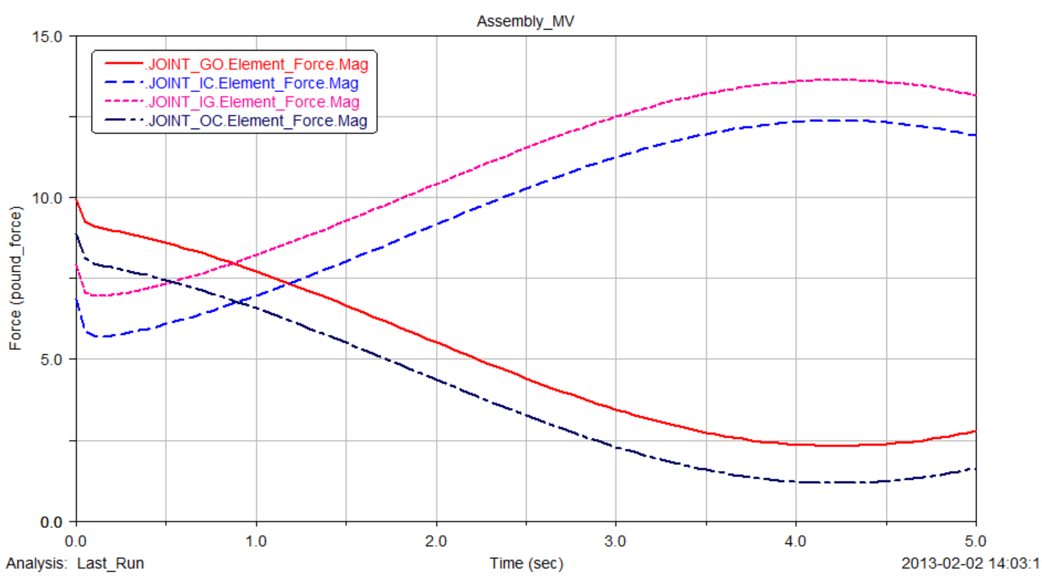


Figure A.25 Adams force graph

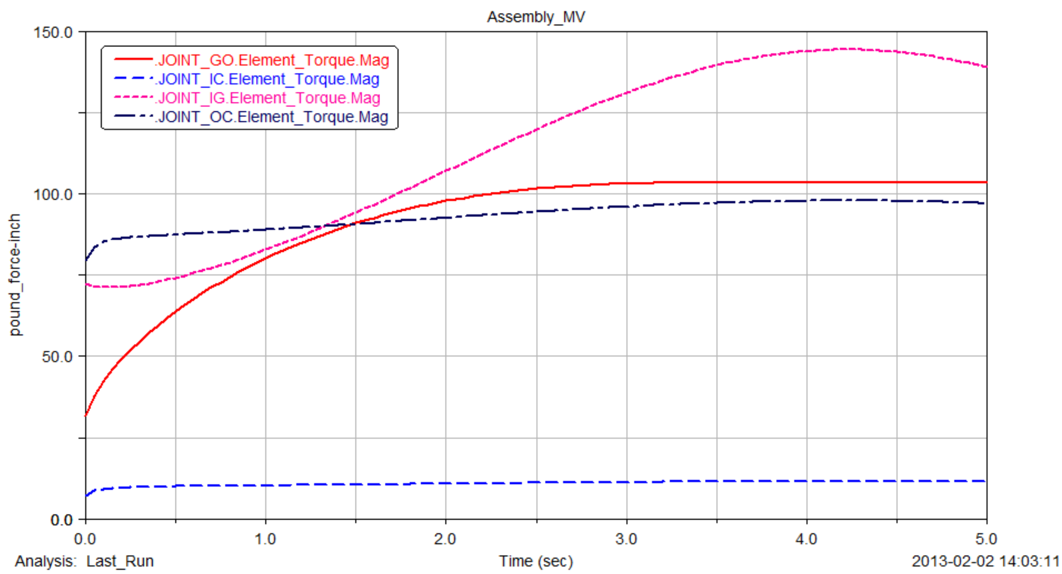
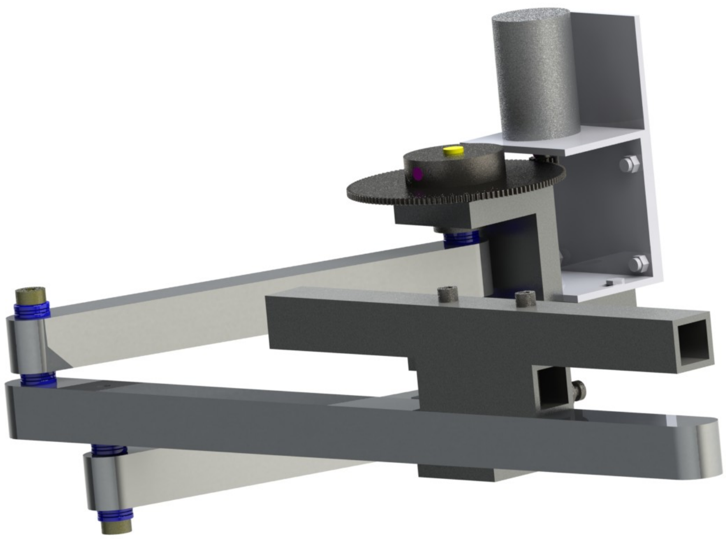


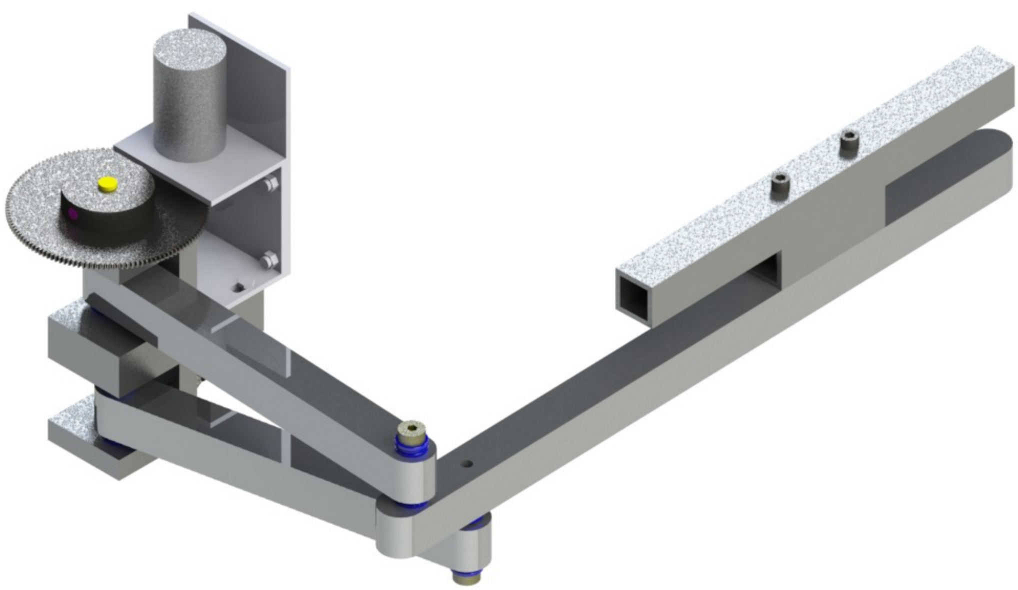
Figure A.26 Adams torque graph

The ratio between the input and output links resulted in a smooth rotational motion and minimized the loads endured by the joins in the linkage system, as seen in Figure A.26. Also, due to the simple shapes of the linkages we have been able to feature solid bars that not only will support higher loads but also will allow us to press in large flanged bearings fitted with shoulder bolts that will tightly fit the bearings, thus eliminating horizontal misalignment.

Appendix B – Motion generator Manufacturing

Final Design





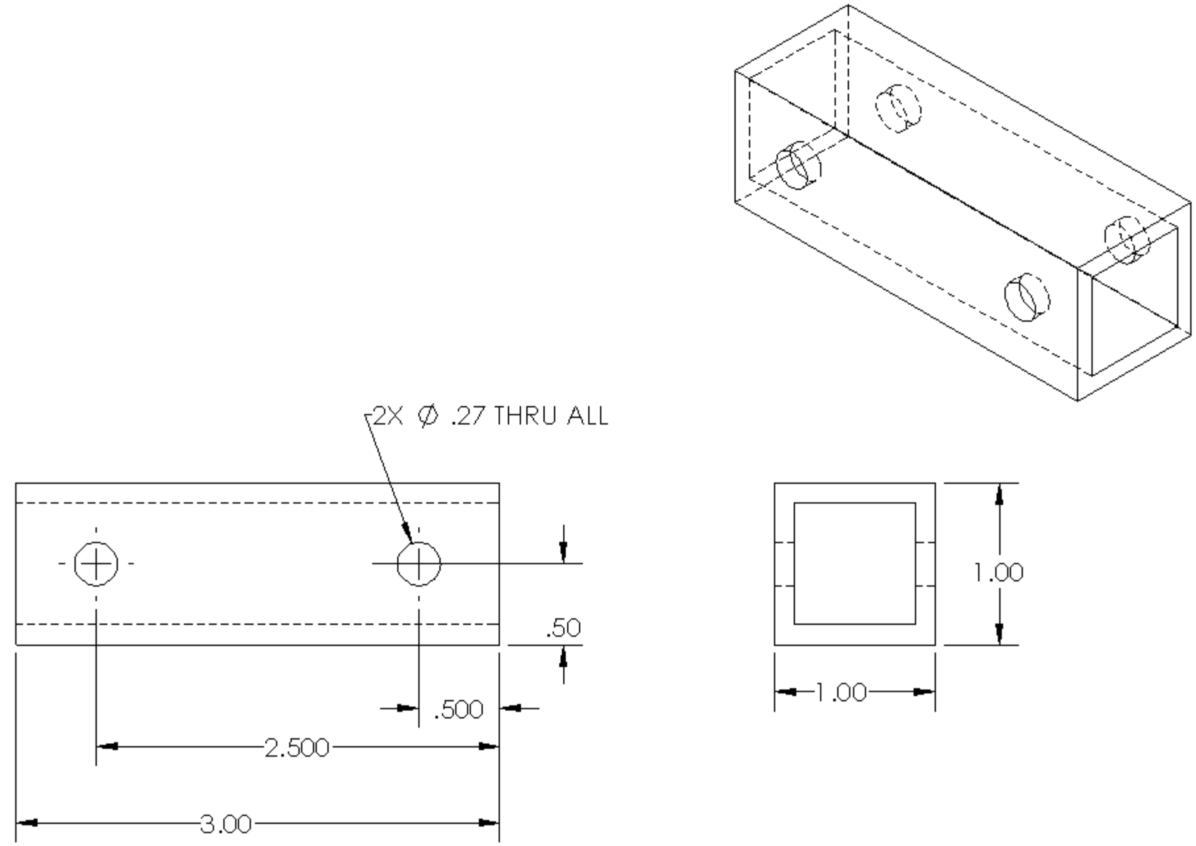
Manufacturing Plans

Backpack holder connector

Backpack Holder Lifter Manufacturing Plan

Step #	Process Description	Machine	Fixtures	Tool(s)	Speed (RPM)
1	Hold part in vise	Mill	Vise		
2	Mill one end of part, just enough to provide a fully machined surface.	Mill	Vise	¾" 2 flute endmill, collet	840
3	Remove part from vise. Break all edges by hand.			file	
4	Place part in vise to machine other end of part. Mill the part to 3" length, taking several passes at .05" per pass. Turn off the spindle, and measure part.	Mill	Vise	¾" 2 flute endmill, collet, calipers	840
5	Remove part from vise. Break all edges by hand.			file	
6	Zero the axes of the machine in order to drill accurate holes.	Mill	Vise	Drill chuck, edge finder	100
7	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	3000
8	Drill .2570" hole in the link center drilled holes.	Mill	Vise	Drill chuck, .2570" drill bit	3000

Table B.1 Backpack Holder connector Manufacturing Plan



UNLESS OTHERWISE SPECIFIED:	
DIMENSIONS ARE IN INCHES TOLERANCES: FRACTIONAL ± ANGULAR: MACH ± BEND ± TWO PLACE DECIMAL ± .010 THREE PLACE DECIMAL ± .005	
INTERPRET GEOMETRIC TOLERANCING PER:	
MATERIAL	Sq. Al. Tube
FINISH	
DO NOT SCALE DRAWING	

	NAME	DATE
DRAWN	RSM	02/04/13
CHECKED		
ENG APPR.		
MFG APPR.		
Q.A.		
COMMENTS:		

NEXT ASSY	USED ON
APPLICATION	

TITLE:

Backpack Connector

SIZE	DWG. NO.	REV
A		0

SCALE: 1:1 WEIGHT: SHEET 1 OF 1

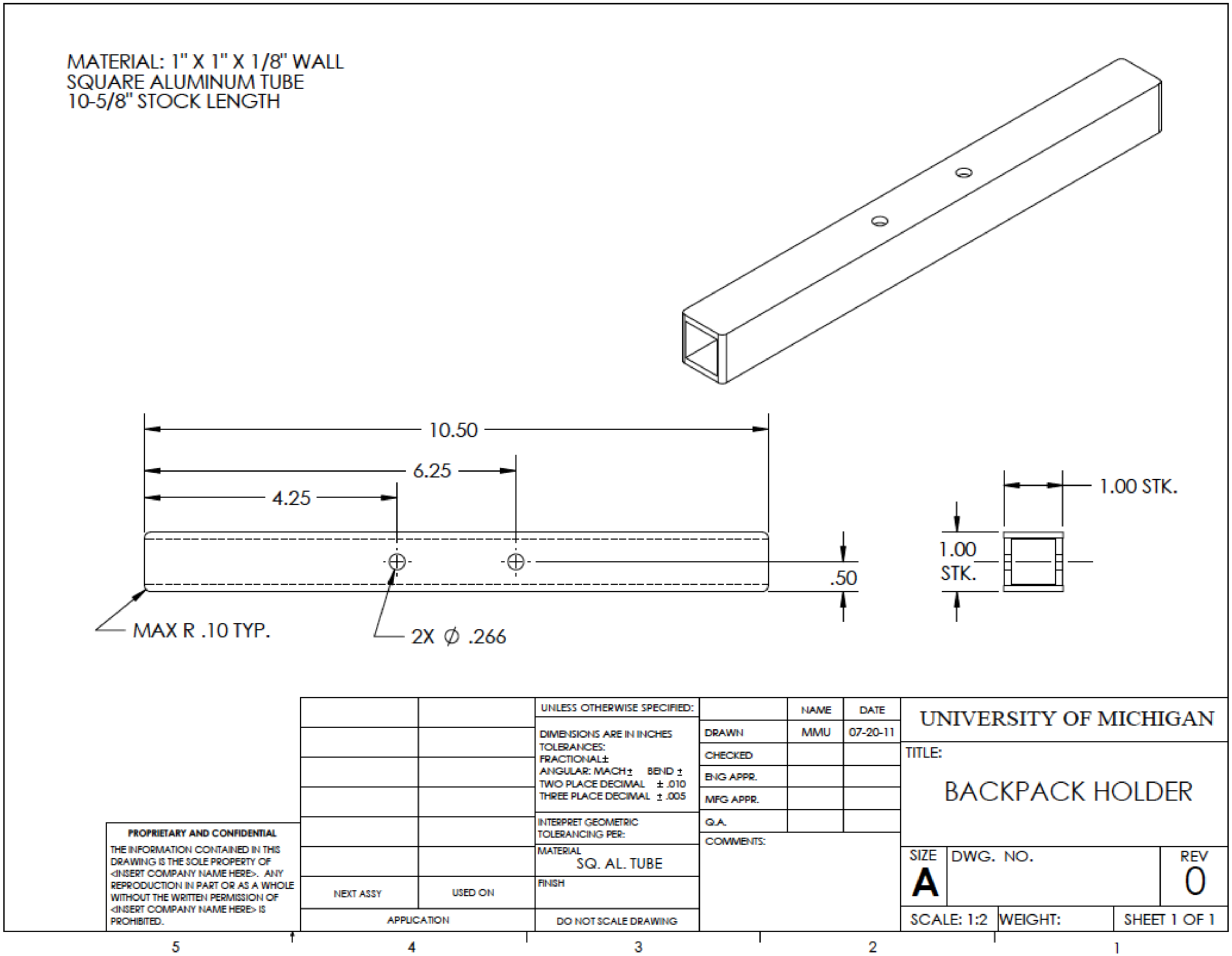
5 4 3 2 1

Backpack holder

Backpack Holder Manufacturing Plan

Step #	Process Description	Machine	Fixtures	Tool(s)	Speed (RPM)
1	Hold part in vise	Mill	Vise		
2	Mill one end of part, just enough to provide a fully machined surface.	Mill	Vise	¾” 2 flute endmill, collet	840
3	Remove part from vise. Break all edges by hand.			file	
4	Place part in vise to machine other end of part. Mill the part to 10.5” length, taking several passes at .05” per pass. Turn off the spindle, and measure part.	Mill	Vise	¾” 2 flute endmill, collet, calipers	840
5	Remove part from vise. Break all edges by hand.			file	
6	Zero the axes of the machine in order to drill accurate holes.	Mill	Vise	Drill chuck, edge finder	100
7	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	1600
8	Drill .2570” hole in the link center drilled holes.	Mill	Vise	Drill chuck, .2570” drill bit	1600

Table B.2 Backpack Holder Manufacturing Plan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SQUARE ALUMINUM TUBE
10-5/8" STOCK LENGTH
10.50
6.25
4.25
.50
1.00 STK.
1.00
STK.
MAX R .10 TYP.
2X ∅ .266
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL±
ANGULAR: MACH± BEND ±
TWO PLACE DECIMAL ±.010
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC
TOLERANCING PER:
MATERIAL
SQ. AL. TUBE
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
MMU
07-20-11
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
UNIVERSITY OF MICHIGAN
TITLE:
BACKPACK HOLDER
SIZE
A
DWG. NO.
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SCALE: 1:2
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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.
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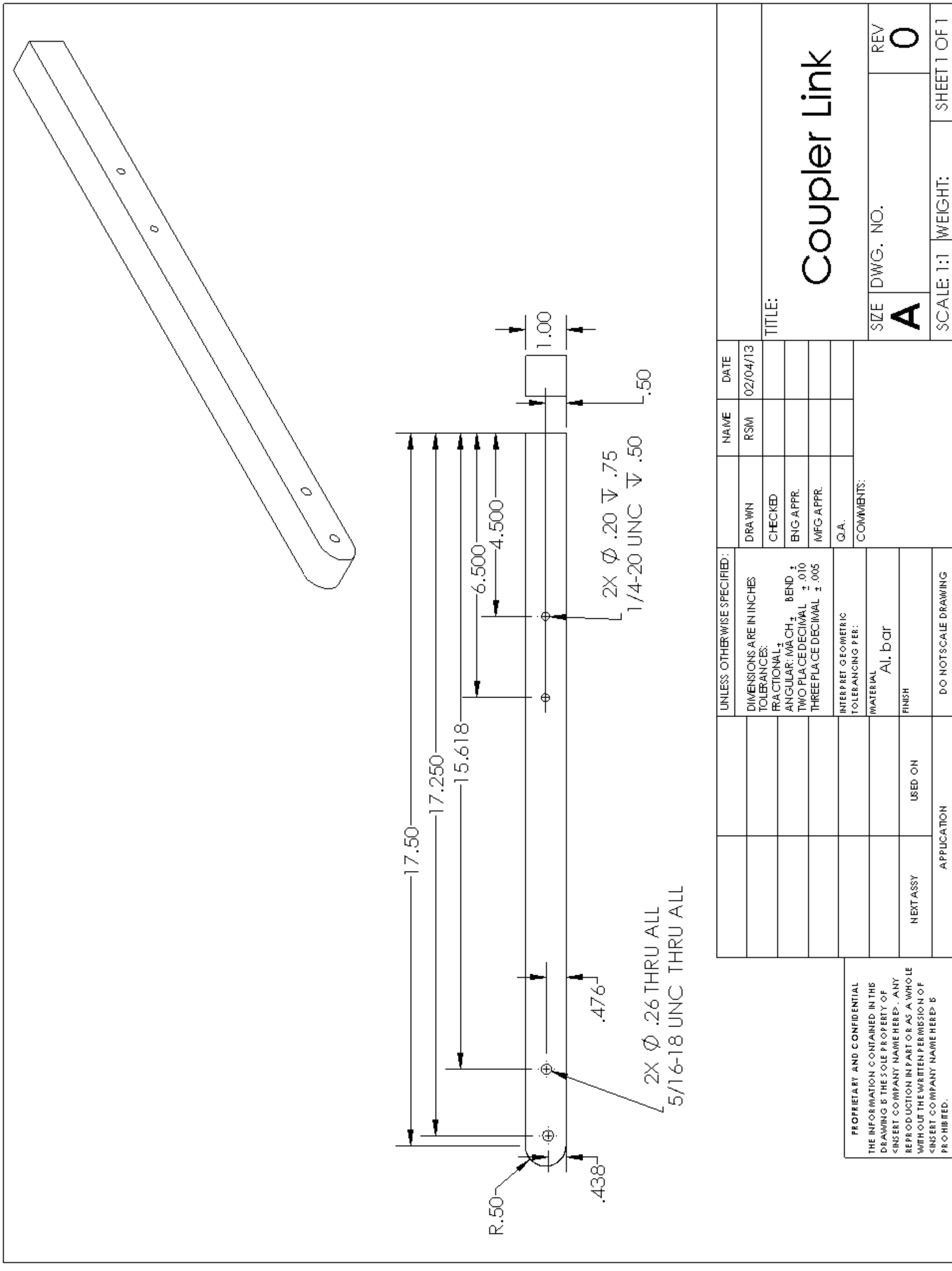
Coupler Link

Coupler Link Manufacturing Plan

Step #	Process Description	Machine	Fixtures	Tool(s)	Speed (RPM)
1	Hold part in vise	Mill	Vise		
2	Mill one end of part, just enough to provide a fully machined surface.	Mill	Vise	¾" 2 flute endmill, collet	840
3	Remove part from vise. Break all edges by hand.			file	
4	Place part in vise to machine other end of part. Mill the part to 16" length, taking several passes at .05" per pass. Turn off the spindle, and measure part.	Mill	Vise	¾" 2 flute endmill, collet,calipers	840
5	Remove part from vise. Break all edges by hand.			file	
6	Zero the axes of the machine in order to drill accurate holes.	Mill	Vise	Drill chuck, edge finder	100
7	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	1600
8	Drill ¼" hole in the link center drilled holes.	Mill	Vise	Drill chuck, ¼" drill bit	1600
9	Drill .5156" hole in the ¼" drilled holes.	Mill	Vise	Drill chuck, .5156" drill	1600
10	Ream a .5624" hole in the previous drilled holes.	Mill	Vise	Drill chuck, .5624" drill	120
11	Drill13/64" hole in the link center drilled holes.	Mill	Vise	Drill chuck, 13/64" drill bit	1600
12	Tap the 13/64"holes with ¼"-20 tap	Mill	Vise	Drill chuck, center bit, tap handle, ¼"-20	
13	Round link edges of desired radius	Water Jet			

Table B.3 Coupler Link Manufacturing Plan



Coupler Link
TITLE:
SIZE A
DWG. NO.
REV 0
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
NAME
DATE
DRAWN RSM 02/04/13
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ± .010
THREE PLACE DECIMAL ± .005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
Al. bar
FINISH
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
PROPRIETARY AND CONFIDENTIAL
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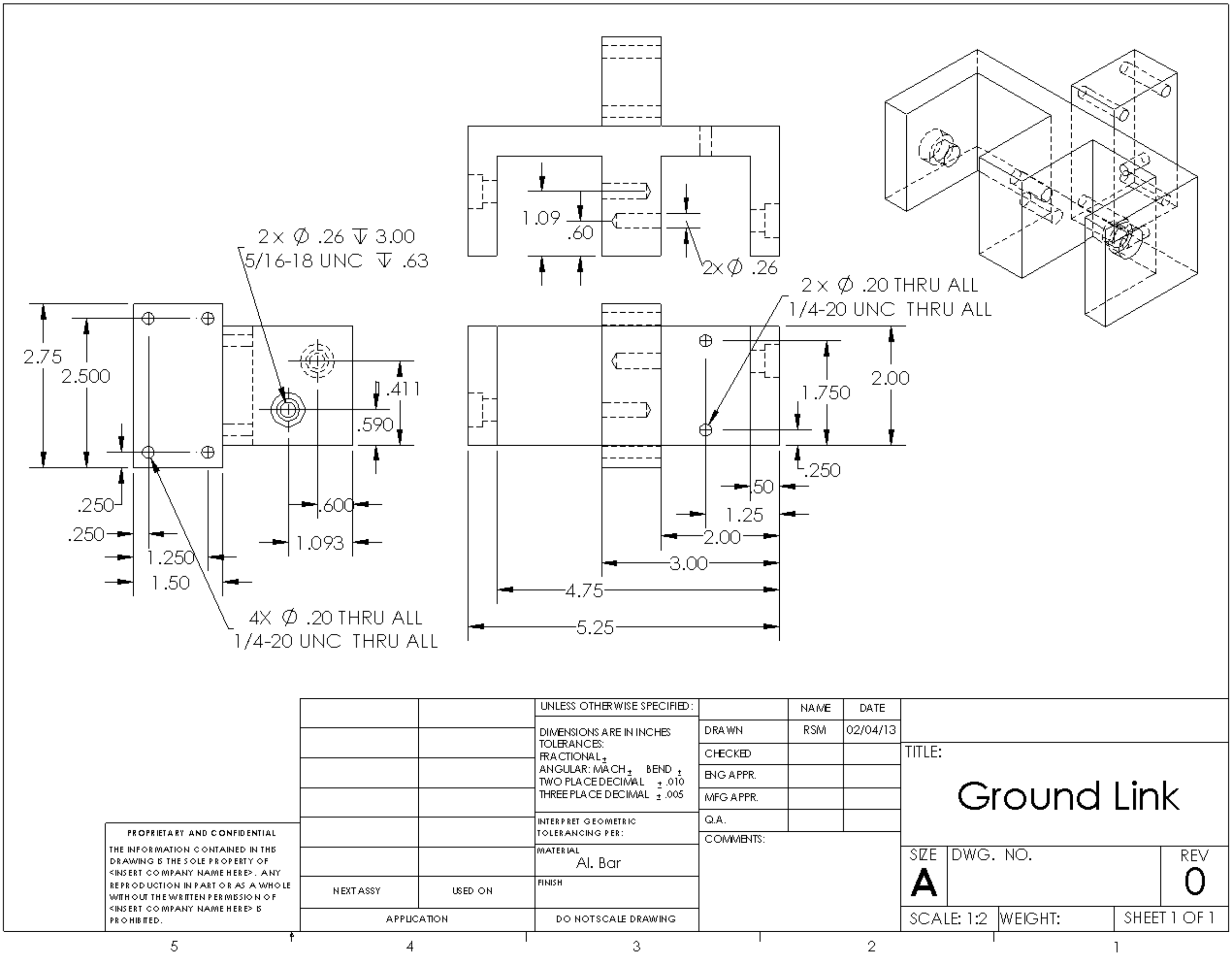
17.50
17.250
15.618
6.500
4.500
1.00
.50
.476
.438
R.50
2X Ø .20 ↧ .75
1/4-20 UNC ↧ .50
2X Ø .26 THRU ALL
5/16-18 UNC THRU ALL
1
2
3
4
5

Ground plate

Ground Plate Manufacturing Plan

Step #	Process Description	Machine	Fixtures	Tool(s)	Speed (RPM)
1	Hold part in vise	Mill	Vise		
2	Mill one end of part, just enough to provide a fully machined surface.	Mill	Vise	¾" 2 flute endmill, collet	840
3	Remove part from vise. Break all edges by hand.			File	
4	Place part in vise to machine other end of part. Mill the part to 5.25" length, taking several passes at .05" per pass. Turn off the spindle, and measure part.	Mill	Vise	¾" 2 flute endmill, collet, calipers	840
5	Place part in vise to machine other end of part. Mill the part to 3.691" wide, taking several passes at .05" per pass. Turn off the spindle, and measure part.	Mill	Vise	¾" 2 flute endmill, collet, calipers	840
6	Zero the axes of the machine in order to drill accurate holes.	Mill	Vise	Drill chuck, edge finder	100
7	Mill out selected voids in the front face of the part	Mill	Vise	¾" 2 flute endmill, collet, calipers	840
8	Remove part from vise. Break all edges by hand.			File	
9	Zero the axes of the machine in order to drill accurate holes.	Mill	Vise	Drill chuck, edge finder	100
10	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	1600
11	Drill 17/64" hole in the plate center drilled holes for the tapped holes.	Mill	Vise	Drill chuck, 17/64" drill bit	1600
12	Drill .3367" holes in the upper and lower levels of the bracket	Mill	Vise	Drill chuck, .3367" drill bit	1600
13	Ream the previous holes .3749"	Mill	Vise	Drill chuck, .3749 reamer	120
14	Drill recessed holes 5/8" in previous holes ¼" deep	Mill	Vise	Drill chuck, 5/8" drill bit	120
15	Tap the correct holes with 5/16" -18 tap	Mill	Vise	Drill chuck, center pin, tap handle, 5/16"-18 tap	
16	Drill the holes for the hard stops with 13/64"	Mill	Vise	Drill chuck,13/64" drill bit	1600
17	Tap the hard stop holes with ¼"-20 tap	Mill	Vise	Drill Chuck, center pin, tap handle, ¼"-20 tap	

Table B.4 Ground Plate Manufacturing Plan


2 × Ø .26 ↧ 3.00
5/16-18 UNC ↧ .63
2.75
2.500
.411
.590
.600
1.093
.250
.250
1.250
1.50
4X Ø .20 THRU ALL
1/4-20 UNC THRU ALL
1.09
.60
2× Ø .26
2 × Ø .20 THRU ALL
1/4-20 UNC THRU ALL
1.750
2.00
.250
.50
1.25
2.00
3.00
4.75
5.25
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ± .010
THREE PLACE DECIMAL ± .005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
Al. Bar
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
RSM
02/04/13
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
Ground Link
SIZE
A
DWG. NO.
REV
0
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL
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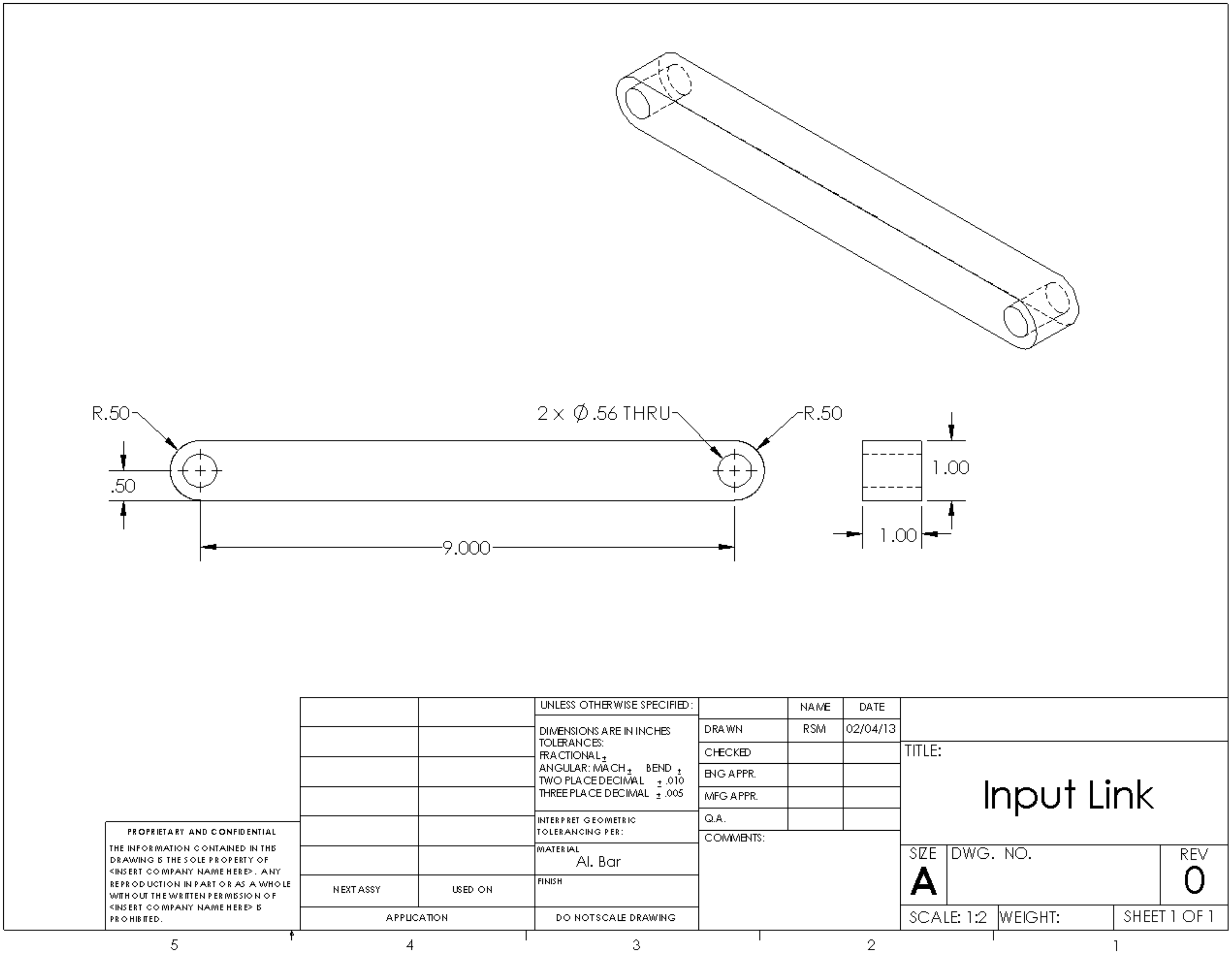
5
4
3
2
1

Input Link

Input Link Manufacturing Plan

Step #	Process Description	Machine	Fixtures	Tool(s)	Speed (RPM)
1	Hold part in vise	Mill	Vise		
2	Mill one end of part, just enough to provide a fully machined surface.	Mill	Vise	¾" 2 flute endmill, collet	840
3	Remove part from vise. Break all edges by hand.			file	
4	Place part in vise to machine other end of part. Mill the part to 9.25" length, taking several passes at .05" per pass. Turn off the spindle, and measure part.	Mill	Vise	¾" 2 flute endmill, collet, calipers	840
5	Remove part from vise. Break all edges by hand.			file	
6	Zero the axes of the machine in order to drill accurate holes.	Mill	Vise	Drill chuck, edge finder	100
7	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	1600
8	Drill ¼" hole in the center drilled holes.	Mill	Vise	Drill chuck, ¼" drill bit	1600
9	Drill .5156" hole in the ¼" drilled holes.	Mill	Vise	Drill chuck, .5156" drill	1600
10	Ream a .5624" hole in the previous drilled holes.	Mill	Vise	Drill chuck, .5624" drill	120
11	Round link edges of desired radius	Water Jet			

Table B.5 Input Link Manufacturing Plan


R.50
2 x Ø.56 THRU
R.50
.50
9.000
1.00
1.00
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ± .010
THREE PLACE DECIMAL ± .005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
Al. Bar
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
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RSM
02/04/13
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
Input Link
SIZE
A
DWG. NO.
REV
0
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
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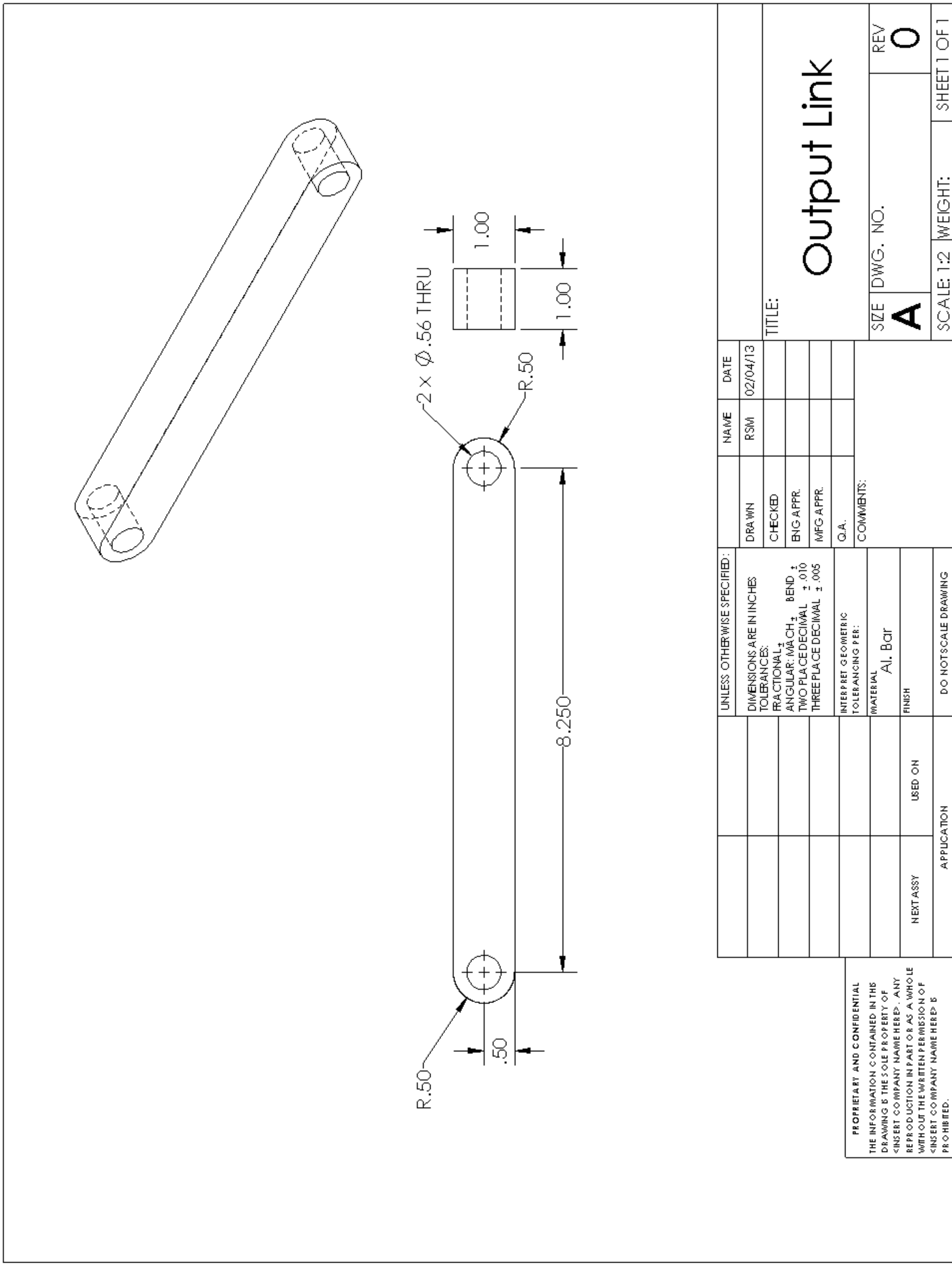
5
4
3
2
1

Output link

Output Link Manufacturing Plan

Step #	Process Description	Machine	Fixtures	Tool(s)	Speed (RPM)
1	Hold part in vise	Mill	Vise		
2	Mill one end of part, just enough to provide a fully machined surface.	Mill	Vise	¾" 2 flute endmill, collet	840
3	Remove part from vise. Break all edges by hand.			file	
4	Place part in vise to machine other end of part. Mill the part to 10" length, taking several passes at .05" per pass. Turn off the spindle, and measure part.	Mill	Vise	¾" 2 flute endmill, collet, calipers	840
5	Remove part from vise. Break all edges by hand.			file	
6	Zero the axes of the machine in order to drill accurate holes.	Mill	Vise	Drill chuck, edge finder	100
7	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	1600
8	Drill ¼" hole in the center drilled holes.	Mill	Vise	Drill chuck, ¼" drill bit	1600
9	Drill .5156" hole in the ¼" drilled holes.	Mill	Vise	Drill chuck, .5156" drill	1600
10	Ream a .5624" hole in the previous drilled holes.	Mill	Vise	Drill chuck, .5624" drill	120
11	Round link edges of desired radius	Water Jet			

Table B.6 Output Link Manufacturing Plan


Output Link
2 X Ø.56 THRU
R.50
R.50
1.00
1.00
8.250
.50
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ±
ANGULAR: MACH ± BEND ±
TWO PLACE DECIMAL ±.010
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
Al. Bar
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
RSM
02/04/13
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
SIZE
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DWG. NO.
REV
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SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
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1
2
3
4
5

Assembly sequence

Final Design Assembly Plan				
Step #	**Process Description**	**Machine**	**Fixtures**	**Tool(s)**
1	Press sleeve bearings into the end of each link.	Hand Press		9/16"Round stock, flat stock
2	Ream the bearings to fit shoulder bolts	Mill	Vise, Drill Chuck, .374" reamer	
3	Support the Coupler link in a vise with the linkage connection out where it would be easy to operate on.		Vise	
4	Attach the ¼"-20 bolts into the coupler to hold the backpack holder and holder lifter secure.		Vise	1/8" Allen wrench
5	With 3/8"x1.5" shoulder bolt in hand slide on thrust bearing, the input link, thrust bearing, and then thread into the outermost hole on the coupler link. Tighten down to desired specs.		Vise	3/16"allen wrench
6	With 3/8"x1.5" shoulder bolt in hand slide on thrust bearing, the output link, thrust bearing, apply Loctite on threads, and then thread into the inner hole on bottom of the coupler link (reverse side of input link). Tighten down to desired specs.		Vise	3/16"allen wrench, Loctite
7	Press bushings into the top and middle section of the ground in the appropriate holes.(input void)	Hand press		9/16"Round stock, flat stock
8	Press the rod through the input while inside the ground link and aligned with the bushings	Hand Press		9/16"Round stock, flat stock
9	Insert C clips in slots		Vise	Flathead Screwdriver
10	Insert the output link into the bottom void of the ground with a thrust bearing on top and bottom of the link. Then apply Loctite onto threads, oil shoulder and thread in a 3/8" x 1.75" shoulder bolt into the hole (aligning the bearings as you proceed down the hole). Tighten down to desired specs.		Vise	3/16"allen wrench, lube, Loctite
11	Insert the ¼"-20 bolts into the ground plate in appropriate holes and adjust to starting and stopping locations			1/8" Allen wrench

Table B.7 Assembly Plan

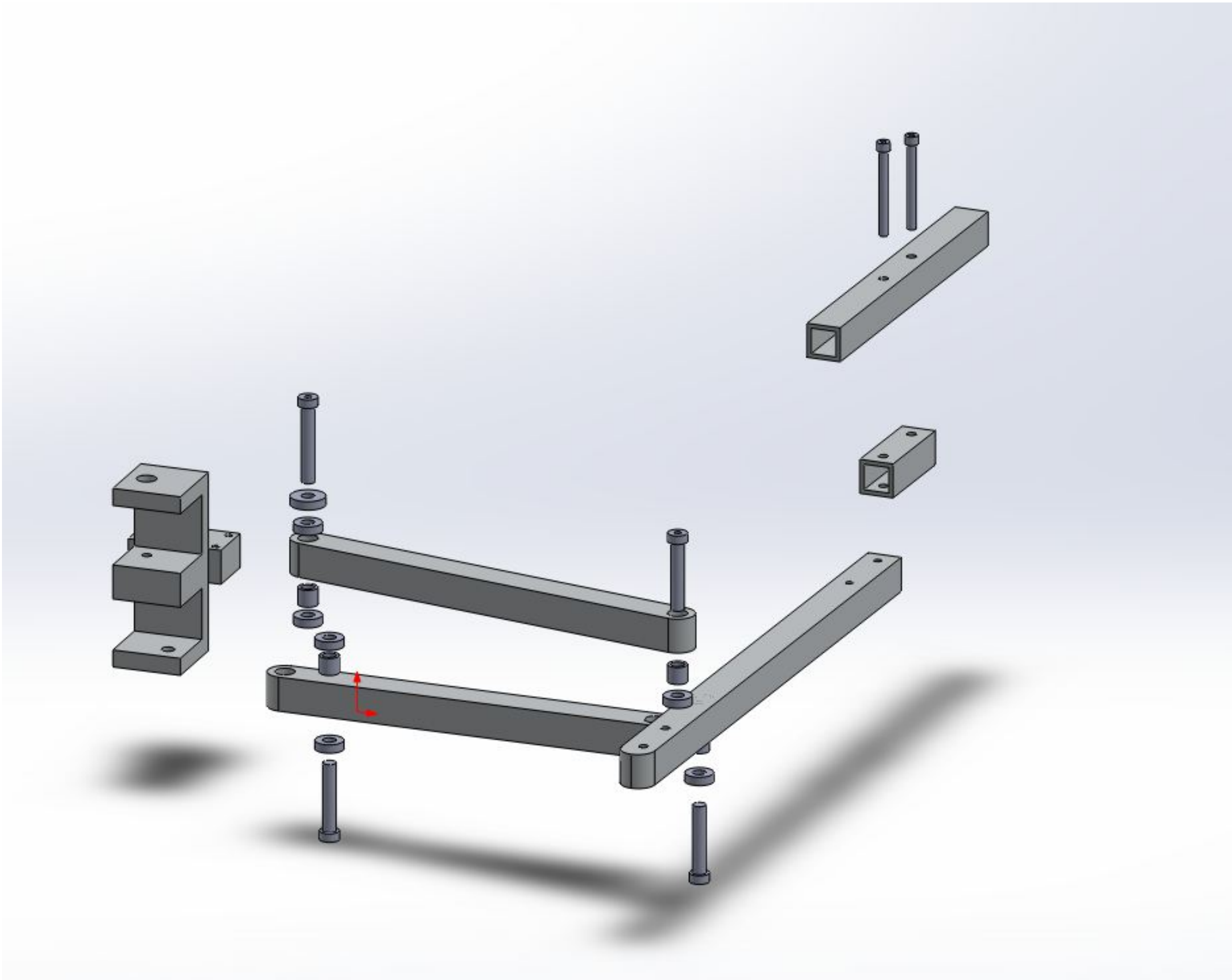


Figure B.1 Assembly Plan Diagram-Exploded View

Bill of Materials

#	Description	Use	Dimension	Supplier	Part#	Comments
1	Square Aluminum Tubing	Backpack Holder	1”x1”x10.5”	McMaster-Carr	88875K33	Kit
2	Aluminum Bar	Coupler	1”x1”x16”	Bob’s shop	N/A	Trade
3	Aluminum Bar	Input Link	1”x1”x9.25”	Bob’s shop	N/A	Trade
4	Aluminum Bar	Output Link	1”x1”x10”	Bob’s shop	N/A	Trade
5	Aluminum Stock	Ground Plate	5.5”x4”x3”	Alro Steel	N/A	$35.00
6	Square Aluminum	Backpack Holder	1”x1”x 3”	McMaster-Carr	88875K33	Kit

	Tubing	Lifter				
7	(8) Thrust Bearings	Reduce friction at joints	ID=3/8” OD=13/16” TH=.249”	McMaster-Carr	6655K15	$19.76
8	(4) Sleeve Bearing	Reduce friction inside joints	ID=3/8” OD=1/2” TH=1”	McMaster	6391K133	Kit
9	(2) 3/8” Shoulder Bolts	Connect Coupler Joints	1.5” Shoulder x 5/16”-18	X50 shop	N/A	
10	(2) 3/8” Shoulder Bolt	Connect Ground Joints	1.75” shoulder x 5/16”-18	X50 shop	N/A	
11	Loctite Red	Adhesive for stronger connection	1 tube	Do-It Hardware	N/A	$6.49
12	(2)1/4” -20 bolts	Hard stops	(1)-2” (1)-1.5”	X50 shop	N/A	
13	(2)1/4”-20 Nuts	Secure hard stops	¼”-20	X50 shop	N/A	
14	(2) ¼”-20 Flathead Screws	Secure Backpack Holder	(2)-2.5”	X50 shop	N/A	
15	Black Tubing covers	Sleek look, safety	1”x1”x ½”	Do- It Hardware	N/A	$3.96
16	Pololu Gear motor	Power the Linkage	37mmD x 57mmL	Pololu	1447	
17	Pololu Motor Hub	Attach motor to Assembly	37mm	Pololu	1038	
Total						**$65.21**

Table B.8 Motion Generation Bill of Material

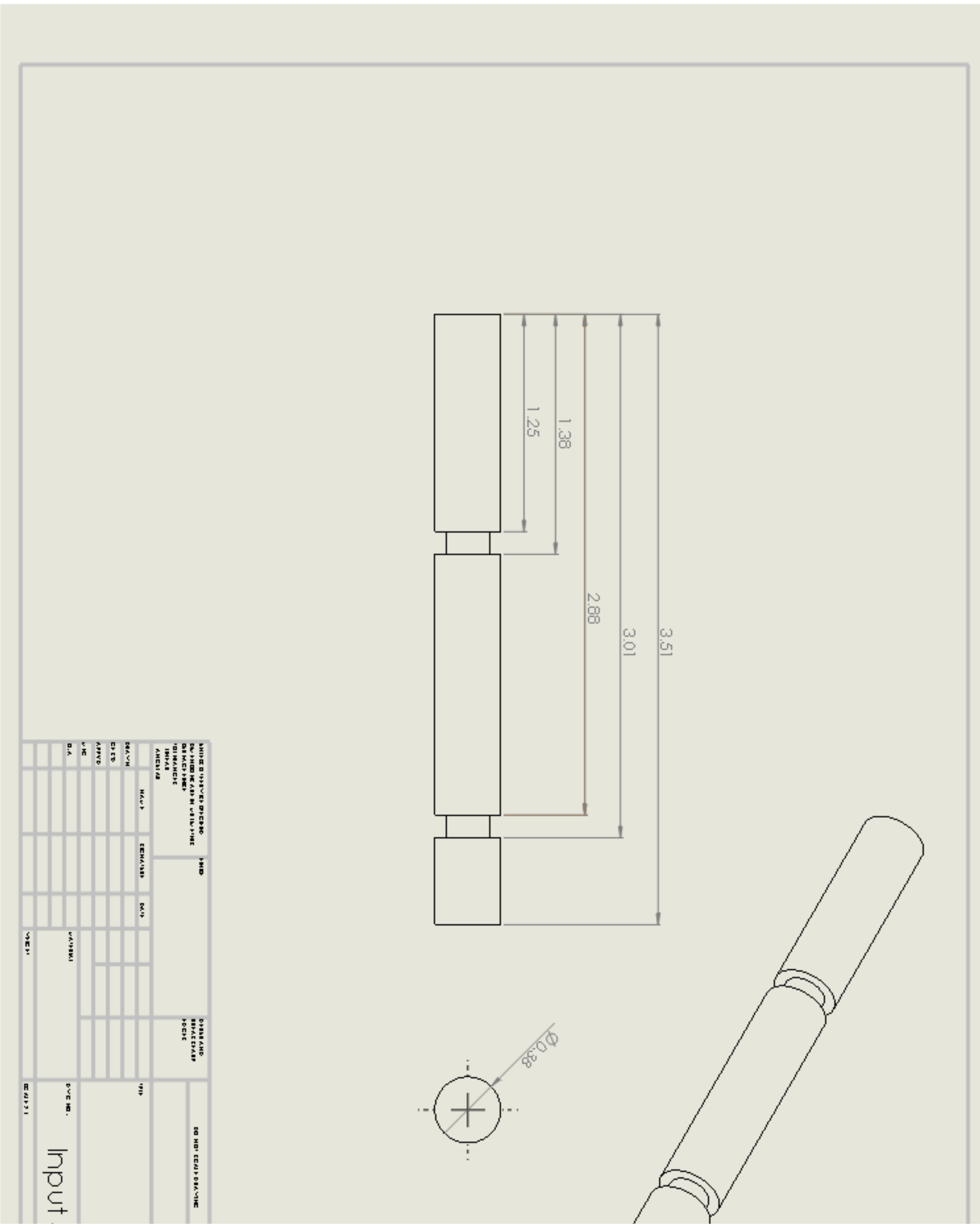
Appendix C – Energy Conversion

Manufacturing Plans

Input shaft

Input Shaft Manufacturing Plan					
Step #	**Process Description**	**Machine**	**Fixtures**	**Tool(s)**	**Speed (RPM)**
1	Measure out a 5” long 3/8” steel shaft		Table	Scribe caliper	
2	Cut to length	Band saw			150
3	Insert shaft into lathe chuck, install tool post, place tool holder on tool post	Lathe		Appropriate chuck, turning/facing tool	
4	Face cut end of part	Lathe		Turning/facing tool	662
5	Begin to cut the C clip slots into the shaft, at appropriate locations	Lathe		Turning/facing tool	662
6	Remove part from chuck once you have made slots and turn it around with rough cut end out			Chuck	
7	Locate the cut off location and proceed slowly, cutting off the unwanted length of material	Lathe		Parting tool	662

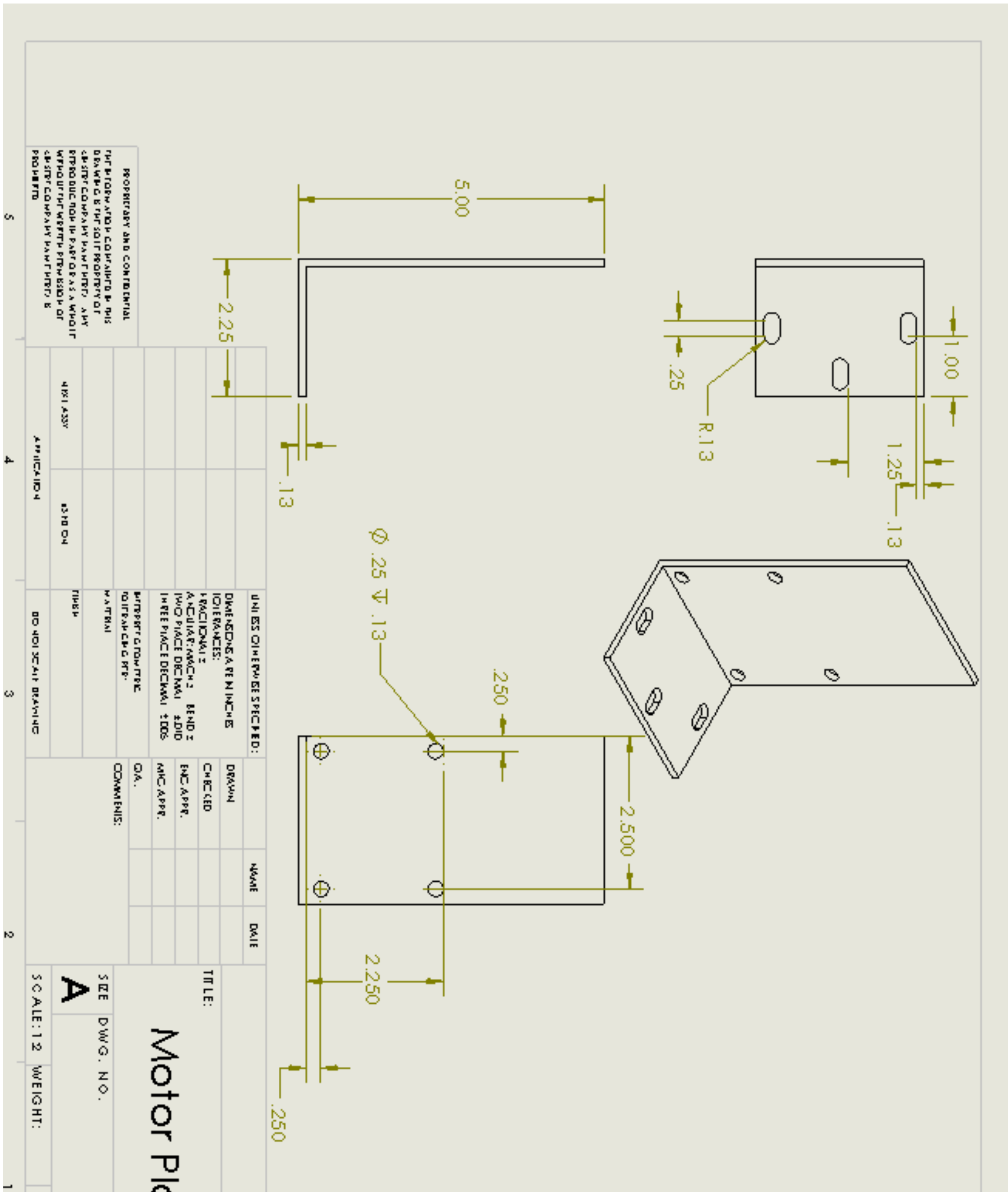
Table C.1 Input Shaft Manufacturing Plan


1.25
1.38
2.88
3.01
3.51
Ø0.38
Input

Motor Bracket

Motor Bracket Manufacturing Plan					
Step #	**Process Description**	**Machine**	**Fixtures**	**Tool(s)**	**Speed (RPM)**
1	Measure out an 4" by 2" piece of 1/8" Aluminum sheet metal		Table	Scribe caliper	
2	Shear the sheet metal on appropriate lines	Shear			
3	Bend a 90º angle in the sheet metal to provide a mounting location	Pan Brake			
4	Zero the axes of the machine in order to drill accurate holes	Mill	Vise	Drill chuck, edge finder	1000
5	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	1600
6	Drill ¼" hole in the sheet metal for mounting in the center drilled holes	Mill	Vise	Drill chuck, 1/4"drill bit	1600

Table C.2 Motor Bracket Manufacturing Plan



5.00
2.25
.13
.25
R.13
1.00
1.25
.13
Ø .25 ↧ .13
.250
2.500
2.250
.250
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
DRAWN
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
NAME
DATE
TITLE:
Motor Pl
SIZE
A
DWG. NO.
SCALE:1:2
WEIGHT:
MATERIAL
FINISH
NEXT ASSY
USED ON
APPLICATION
DO NOT SCALE DRAWING
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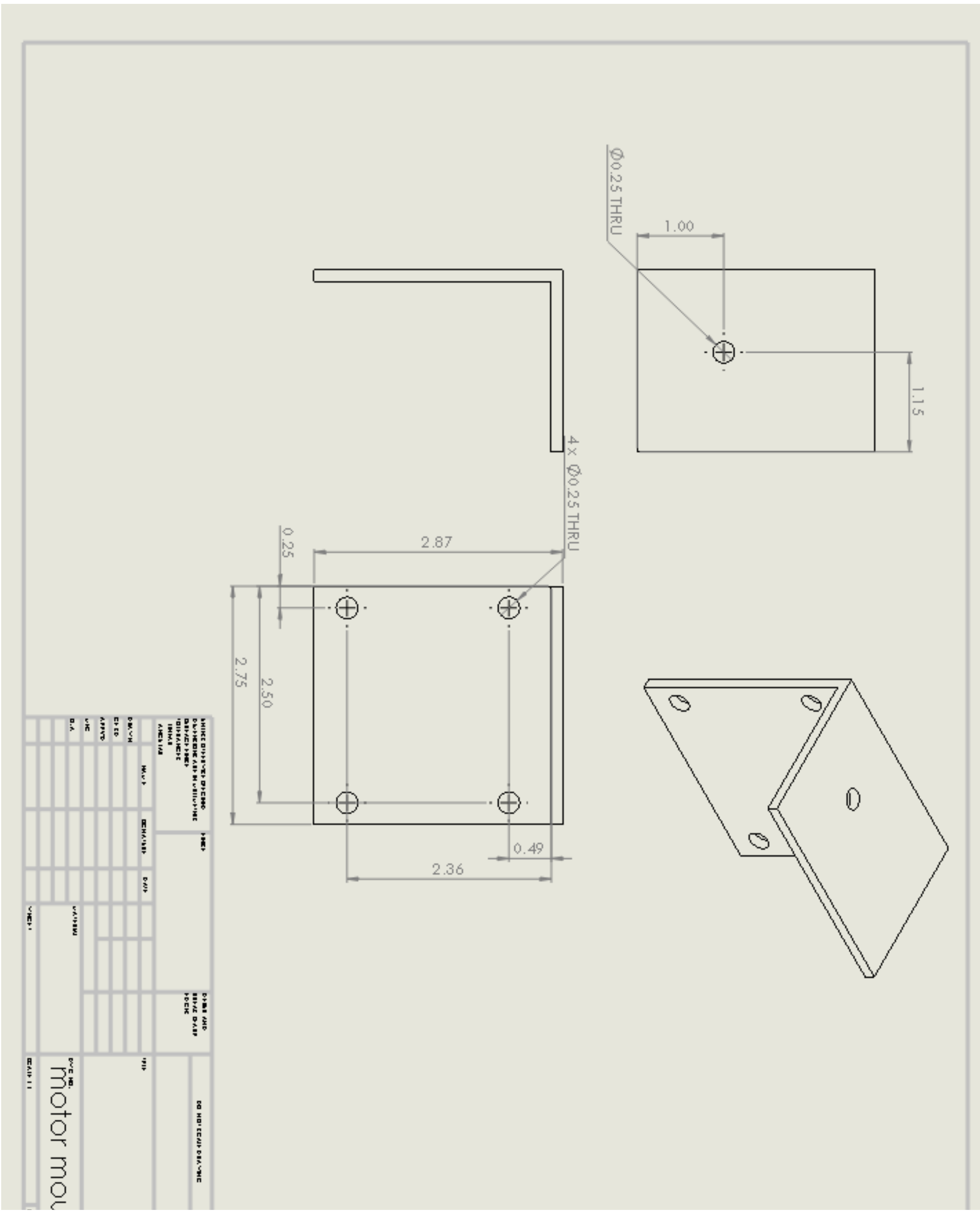
Motor Assembly Bracket

Motor Assembly Bracket Manufacturing Plan					
Step #	**Process Description**	**Machine**	**Fixtures**	**Tool(s)**	**Speed (RPM)**
1	Measure out an 8" by 2" piece of 1/8" Aluminum sheet metal		Table	Scribe caliper	
2	Shear the sheet metal on appropriate lines	Shear			
3	Bend a 90° angle in the sheet metal to provide a mounting location	Pan Brake			
4	Zero the axes of the machine in order to drill accurate holes	Mill	Vise	Drill chuck, edge finder	1000
5	Center drill the desired holes in correct spots	Mill	Vise	Drill chuck, center drill	1600
6	Drill ¼" hole in the sheet metal for mounting in the center drilled holes	Mill	Vise	Drill chuck, 1/4"drill bit	1600
7	Mill out the adjustment holes	Mill	Vise	1/4" endmill, collet	1600

Table C.3 Motor Assembly Manufacturing Plan


Ø0.25 THRU
1.00
1.15
4x Ø0.25 THRU
2.87
0.25
2.75
2.50
0.49
2.36

Assembly sequence

Transmission Assembly Plan				
Step #	**Process Description**	**Machine**	**Fixtures**	**Tool(s)**
1	Now that the linkage is assembled, attach the Input Gear to the gear coupler		Table	Philips screwdriver
2	Slide the gear down the top of the shaft, with 1/8" clearance above ground and tighten the setscrew securing the gear assembly to the shaft.		Linkage in vise	3/16" Allen wrench
3	Attach the motor gear to the motor, same as previous			3/16" Allen wrench
4	Secure the Motor to the motor bracket			3/16"allen wrench
5	Secure the motor bracket and motor to the motor assembly bracket			3/16" Allen wrench
6	Secure and fine adjust the motor assembly to the back of the ground to ensure an accurate meshing of the gears		Linkage in Vise	1/8" Allen wrench

Table C.4 Transmission Assembly Plan

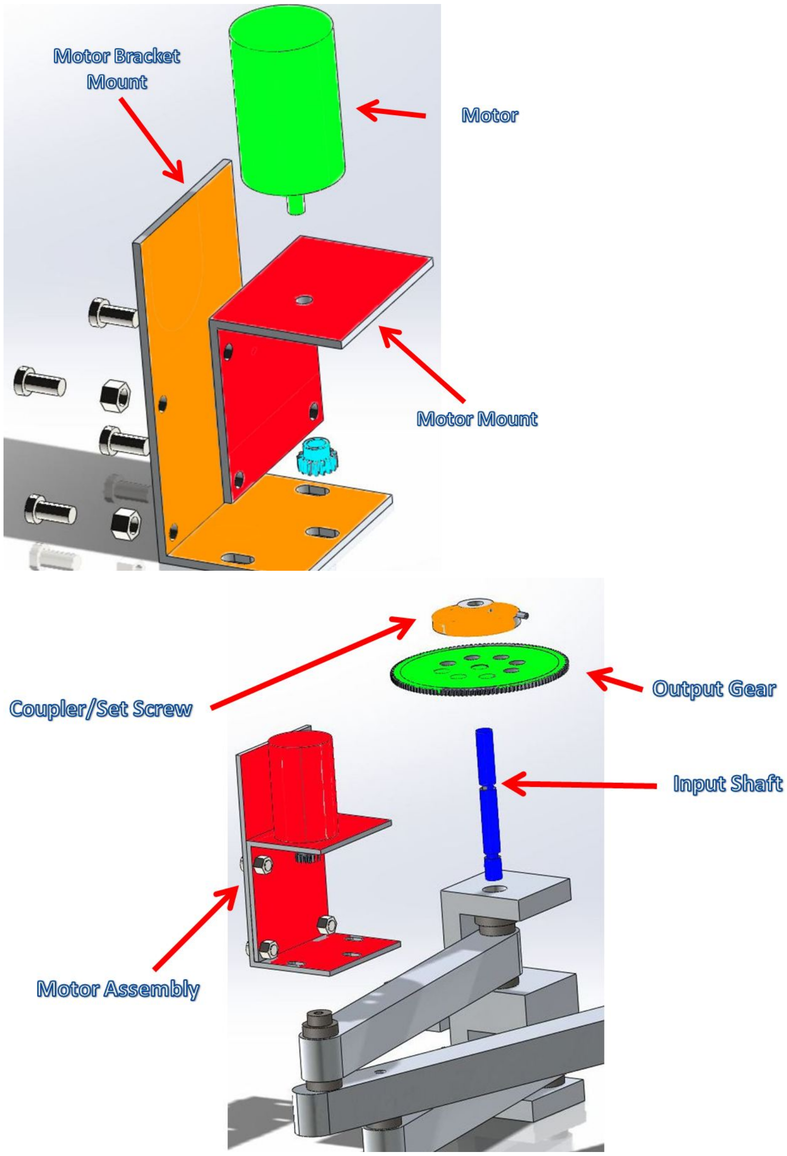


Figure C.1 Transmission Assembly Plan Diagram-Exploded View

Bill of materials

#	Description	Use	Dimension	Supplier	Part#	Comments
16	Pololu Gear motor	Power the Linkage	37mmD x 57mmL	Pololu	1447	
17	Pololu Motor Hub	Attach motor to Assembly	37mm	Pololu	1038	
18	1/8"Aluminum L-Bracket	Hold Motor stable	4"x2"	Bob's shop	N/A	Trade
19	1/8" Aluminum Sheet Metal	Motor Assembly L-Bracket	8"x2"	Bob's Shop	N/A	Trade
20	Steel 16 teeth Gear	Power input gear	½"x 1/8"	Amazon	H3216	$18.83
21	Steel 128 teeth Gear	Move linkage	4"x1/8"	Amazon	H32128	$47.02
22	½"Aluminum Round stock	Gear coupler	1/2"	Bob's Shop	N/A	Trade
23	3/8" Steel shaft	Input shaft	3.5"	Bob's Shop	N/A	Trade
24	Set Screw	Hold coupler to shaft	3/8"L x 8-32	X50 shop	N/A	
25	(4)Gear Attachment Screws	Hold gear to coupler	3/8"L x 8-32	X50 shop	N/A	
Total						**$65.85**

Table C.5 Energy Conversion Bill of Material

Gear specification

Click for larger image and other views

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Image is representative of the product family.

+ **Downloads**
CAD Models, MSDS, Manuals

See all 28 in this Product Family

Boston Gear H32128 Spur Gear, 14.5 Pressure Angle, Steel, Inch, 32 Pitch, 0.375" Bore, 4.062" OD, 0.188" Face Width, 128 Teeth

By Boston Gear
Be the first to review this item

List Price: ~~$51.00~~
Price: $47.02 Prime
You Save: $3.98 (8%)

Only 3 left in stock (more on the way).
Ships from and sold by **Amazon.com**.

Want it tomorrow, April 24? Express delivery is available to 60637. Order within **13 hrs 48 mins**, and choose **Local Express Delivery** at checkout.

Quantity: 1
Add to Cart
or
Sign in to turn on 1-Click ordering.
Add to Wish List

More Buying Choices
Have one to sell? Sell on Amazon
Share

Product Specifications

Part Number	UPC	Bore Diameter	Hub Diameter	Hub Projection
H32128	781711095669	0.375 inches	1.870 inches	0.500 inches
Number of Teeth	**Diameter**	**EAN**	**Number of Items**	
128	4.062 inches	0781711095669	1	

Specification for this product family (See all 28 products)

Brand Name	Material Type	Bore Diameter Tolerance	Face Width	Pitch
Boston Gear	Steel	+/-0.0005 inches	0.188 inches	32

Click for larger image and other views

Share your own related images

Image is representative of the product family.

+ **Downloads**
CAD Models, MSDS, Manuals

See all 28 in this Product Family

Boston Gear H3216 Spur Gear, 14.5 Pressure Angle, Steel, Inch, 32 Pitch, 0.187" Bore, 0.562" OD, 0.188" Face Width, 16 Teeth

By Boston Gear
Be the first to review this item

List Price: ~~$20.40~~
Price: $18.83 Prime
You Save: $1.57 (8%)

Only 2 left in stock (more on the way).
Ships from and sold by **Amazon.com**.

Want it Thursday, April 25? Order within **23 hrs 48 mins**, and choose **One-Day Shipping** at checkout. Details

Quantity: 1
Add to Cart
or
Sign in to turn on 1-Click ordering.
Add to Wish List

More Buying Choices
Have one to sell? Sell on Amazon
Share

Product Specifications

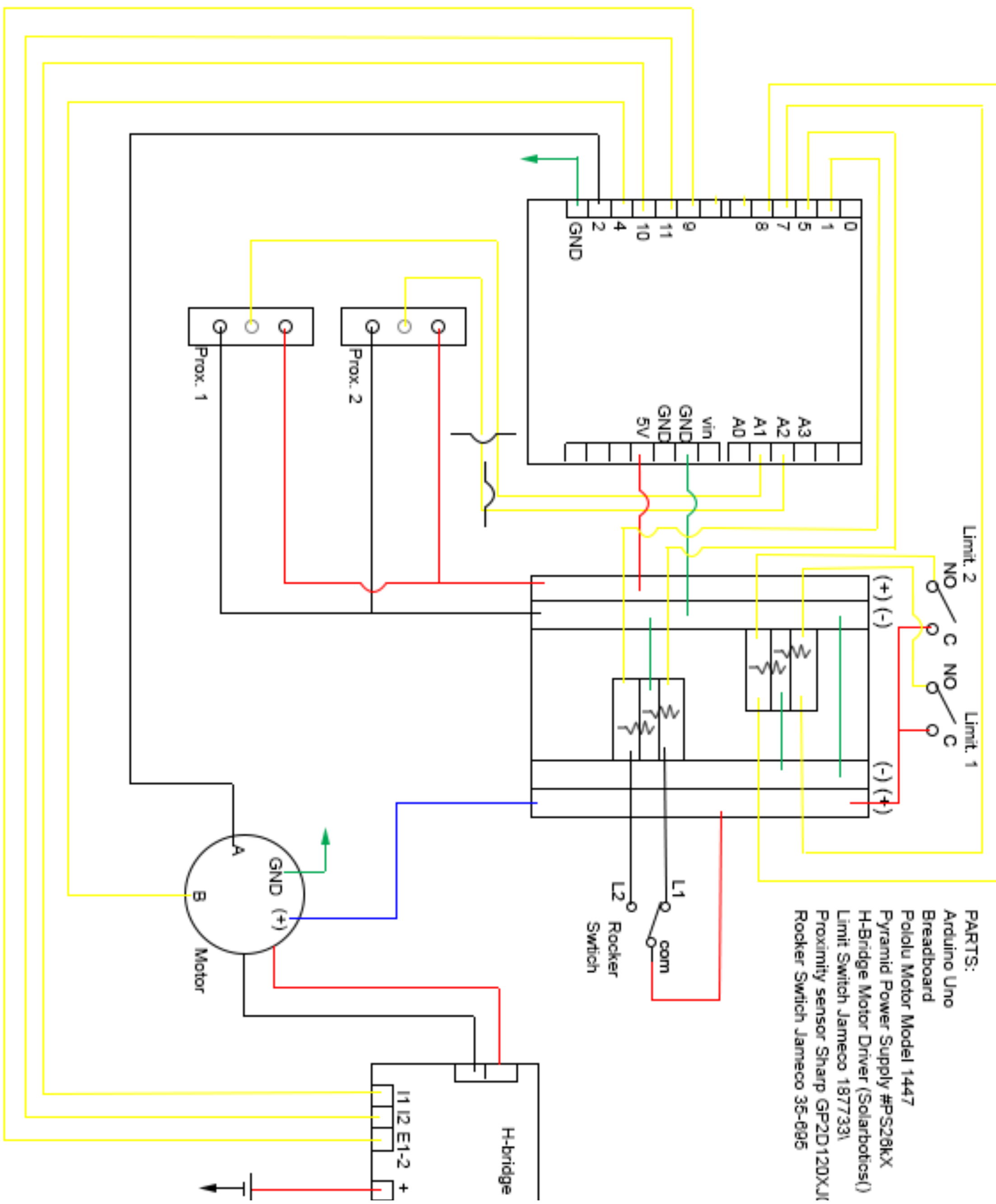
Part Number	UPC	Bore Diameter	Hub Diameter	Hub Projection
H3216	781711095362	0.187 inches	0.400 inches	0.310 inches
Number of Teeth	**Diameter**	**EAN**	**Number of Items**	
16	0.562 inches	0781711095362	1	

Specification for this product family (See all 28 products)

Brand Name	Material Type	Bore Diameter Tolerance	Face Width	Pitch
Boston Gear	Steel	+/-0.0005 inches	0.188 inches	32

Appendix D – Mechatronics

Wiring Diagram



Controller code w/ comments

```
// ME350 Project Sketch_Red H-bridge
// sketch for mobile backpack carrier
// set the values of variables that affect the performance
int lowerCountThreshold=3000;
int upperCountThreshold=13042;
int proxSwitch1Threshold=60;  // a value between 0 and 1024
int proxSwitch2Threshold=60;  // a value between 0 and 1024
float motorSpeedHigh=350;     // reference high speed (unit deg/sec)
float motorSpeedLow=120;      // reference low speed  (unit deg/sec)

//set the initial values of variables
int proxSwitch1=0;
int proxSwitch2=0;
int rockerSwitchForward=0;
int rockerSwitchReverse=0;
int limitSwitchForward=0;
int limitSwitchReverse=0;
volatile long encoderCount = 0;

long encoderCountCurr=0;  // current encoder count
long encoderCountLast=0;  // encoder count last loop
long encoderStep=0;   //encoder count change since last loop

unsigned long tic;
unsigned long toc;
unsigned long timeDelay;
float spd=0;   // motor actual speed
int spdComd=200;  //motor PWM command signal
float integrate=0;  // integration of error

//set the values of variables that represent pin numbers
const int proxSwitch1Pin = 1;
const int proxSwitch2Pin = 2;
const int rockerSwitchForwardPin = 5;
const int rockerSwitchReversePin = 6;
const int limitSwitchForwardPin = 7;
const int limitSwitchReversePin = 8;
const int pwmPin = 9;
const int i1Pin = 10;
const int i2Pin = 11;
const int encoderPinA = 2;
const int encoderPinB = 4;

void setup(){
  //declare which digital pins are inputs and outputs
  pinMode(rockerSwitchForwardPin,INPUT);
  pinMode(rockerSwitchReversePin,INPUT);
  pinMode(limitSwitchForwardPin,INPUT);
  pinMode(limitSwitchReversePin,INPUT);
```

```
  pinMode(encoderPinA,INPUT);
  pinMode(encoderPinB,INPUT);
  pinMode(pwmPin,OUTPUT);
  pinMode(i1Pin,OUTPUT);
  pinMode(i2Pin,OUTPUT);

  //turn on pullup resistors
  digitalWrite(encoderPinA,HIGH);
  digitalWrite(encoderPinB,HIGH);
  // (the other sensors already have physical resistors on the breadboard)

  //set interrupt for encoder pins
  attachInterrupt(0,doEncoder,CHANGE);

  //set initial speed of motor to 0
  analogWrite(pwmPin,0);

  //begin serial communication for display of variable states
  Serial.begin(9600);
  Serial.println("start");
}

void loop(){
  // determine time interval
  toc=tic;
  tic=millis();
  timeDelay=tic-toc; //timeDelay in milli sec

  // determine encoder count change
  encoderCountLast=encoderCountCurr;
  encoderCountCurr=encoderCount;
  encoderStep=encoderCountCurr-encoderCountLast;

  //determine speed
  if (encoderStep>=0){
    // 14.31298 deg/sec   == 1 count / milli sec
    spd=float(encoderStep)/float(timeDelay)*85.8779; //rotate forward unit deg/sec
  }
  else{
    spd=float(-encoderStep)/float(timeDelay)*85.8779;//rotate backward unit deg/sec
  }
// aquire signals from pins
  proxSwitch1=analogRead(proxSwitch1Pin);
  proxSwitch2=analogRead(proxSwitch2Pin);
  rockerSwitchForward=digitalRead(rockerSwitchForwardPin);
  rockerSwitchReverse=digitalRead(rockerSwitchReversePin);
  limitSwitchForward=digitalRead(limitSwitchForwardPin);
  limitSwitchReverse=digitalRead(limitSwitchReversePin);
```

```
  if ((proxSwitch1<proxSwitch1Threshold)&&(proxSwitch2<proxSwitch2Threshold)) {  // if both prox
switches are not made
    if (rockerSwitchForward==HIGH){    // if rocker switch is being pressed forward
      digitalWrite (i1Pin,LOW);      // set direction of motor to be forward (i1 pin)
      digitalWrite (i2Pin,HIGH);      // set direction of motor to be forward (i2 pin)
      if (limitSwitchForward==HIGH){   // if the forward limit switch is made
        analogWrite (pwmPin,0);       // stop the motor
        integrate=0;                 // reset controller
        Serial.print("The forward limit switch is made.   Count:");  // print this to the serial monitor
        Serial.println(encoderCount);    // print the encoder count to the serial monitor
      }
      else if (encoderCount < upperCountThreshold){  // if the encoder count is below the upper count
threshold
        integrate=integrate-abs(encoderStep)+timeDelay*motorSpeedHigh*0.0116444;    // integration of
error
        // time 0.069869 to convert degree into encoder count
        spdComd=piControl(motorSpeedHigh, spd, integrate);      //determine motor command 0-255
        analogWrite(pwmPin,spdComd);             // run motor at high speed
        Serial.print("Direction: Forward    Speed: High   Count:");  // print this to the serial monitor
        Serial.println(encoderCount);    // print the encoder count to the serial monitor
        Serial.print("Speed:    ");
        Serial.println(spd);
    }
      else{                                    // otherwise
        integrate=integrate-abs(encoderStep)+timeDelay*motorSpeedLow*0.0116444;   // integration of
error
        spdComd=piControl(motorSpeedLow, spd, integrate);      // determine motor command 0-255
        analogWrite(pwmPin,spdComd);           // run the motor at low speed
        Serial.print("Direction: Forward    Speed: Low   Count:");  // print this to the serial monitor
        Serial.println(encoderCount);    // print the encoder count to the serial monitor
        Serial.print("Speed:    ");
        Serial.println(spd);
    }
    }

    else if (rockerSwitchReverse==HIGH){    // if rocker switch is being pressed backward
      digitalWrite (i1Pin,HIGH);      // set direction of motor to be reverse (i1Pin)
      digitalWrite (i2Pin,LOW);      // set direction of motor to be reverse  (i2Pin)
      if (limitSwitchReverse==HIGH){   // if the back limit switch is made
        analogWrite (pwmPin,0);       // stop the motor
        integrate=0;                 // reset the controller
        Serial.print("The back limit switch is made.   Count:");  // print this to the serial monitor
        Serial.println(encoderCount);    // print the encoder count to the serial monitor
      }
      else if (encoderCount>lowerCountThreshold){  // if the encoder count is above the lower count
threshold
```

```
      integrate=integrate-abs(encoderStep)+timeDelay*motorSpeedHigh*0.0116444;    //integration of
error
      spdComd=piControl(motorSpeedHigh, spd, integrate);     //determine motor command 0-255
      analogWrite(pwmPin,spdComd);         // run motor at high speed
      Serial.print("Direction: Reverse    Speed: High   Count:");  // print this to the serial monitor
      Serial.println(encoderCount);    // print the encoder count to the serial monitor
      Serial.print("Speed:    ");
      Serial.println(spd);
     }
     else{                                   // otherwise
      integrate=integrate-abs(encoderStep)+timeDelay*motorSpeedLow*0.0116444;   //integration of
error
      spdComd=piControl(motorSpeedLow, spd, integrate);      // determine motor command 0-255
      analogWrite(pwmPin,spdComd);         // run the motor at low speed
      Serial.print("Direction: Reverse    Speed: Low   Count:");  // print this to the serial monitor
      Serial.println(encoderCount);    // print the encoder count to the serial monitor
      Serial.print("Speed:    ");
      Serial.println(spd);
    }

    }
    else {
     analogWrite(pwmPin,0);        // stop the motor
     integrate=0;                  //reset controller
     Serial.println("Rocker switch is not being pressed.");  // print this to the serial monitor
    }

  }
  else{                                       // if at least one of the prox switches is made
   analogWrite(pwmPin,0);                       // stop the motor
   integrate=0;                                //reset controller
   Serial.println("There is an object in the way.");  // print this to the serial monitor
  }

  //Serial.println(spdComd);
  //Serial.println(integrate);
}

void doEncoder(){
 if( digitalRead(encoderPinA) == digitalRead(encoderPinB) ){
   encoderCount--;
 }
 else{
   encoderCount++;
 }
}

int piControl(float targetSpd, float currentSpd, float integrate){
```

```
 // tergetSpd is the speed reference in deg/sec
 // currentSpd is actual motor speed measured
 // integrate is the integration of error
 float Kp=1;  // proportional control constant
 float Kd=0;    // differential control constant (not used)
 float Ki=0.25;  // integration control constant
 int baseCmd=150;  // base motor PMW control signal
 int spdComd;    // closed loop motor PMW control signal
 spdComd=int(Kp*(targetSpd-currentSpd)+Ki*integrate+baseCmd);
 if (spdComd>255){
  spdComd=255;
 }
 else if(spdComd<0){
  spdComd=0;
 }
 return spdComd;
}
```

Calculations for encoder count

LowerCountTreshold calculation, assuming an 8:1 transmission ratio:

$$30\text{ deg} = 30\text{ deg}\left(\frac{1\text{ rev. input link}}{360\text{ deg}}\right)\left(\frac{8\text{ rev. gearbox}}{1\text{ rev. input link}}\right)\left(\frac{131\text{ rev. encoder}}{1\text{ rev. gearbox}}\right) = 87.3\text{ rev. of encoder}$$

$$87.3\text{ rev.} = 87.3\text{ rev.}\left(\frac{32\text{ counts}}{1\text{ rev.}}\right) = \mathbf{2795\ counts}$$

UpperCountTreshold calculation:

$$170\text{ deg} - 30\text{ deg} = 140\text{ deg}$$

$$140\text{ deg} = 140\text{ deg}\left(\frac{1\text{ rev. input link}}{360\text{ deg}}\right)\left(\frac{8\text{ rev. gearbox}}{1\text{ rev. input link}}\right)\left(\frac{131\text{ rev. encoder}}{1\text{ rev. gearbox}}\right) = 407.6\text{ rev. of encoder}$$

$$407.6\text{ rev.} = 407.6\text{ rev.}\left(\frac{32\text{ counts}}{1\text{ rev.}}\right) = \mathbf{13042\ counts}$$

Manufacturing plans

Nothing was manufactured for safety/motor control

References

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