Crane Design Report

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1. Design Description

To fulfill the specifications of the design, we had to lift a ten-pound weight one (1) inch off the ground within two (2) minutes using a motor with a stall torque of 12 oz-in without deflecting more than half an inch. Because of this low torque input, we had to develop a **structural design** that could withstand deflection specifications and a **power transmission system** that gave the motor enough mechanical advantage to lift the weight.

1.1. Power Transmission and Lifting Systems

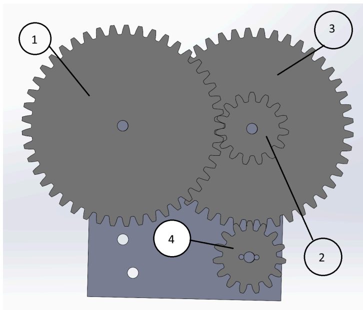


Figure 1: The gear transmission system, each gear labeled 1-4. Gears 1 and 3 have 45 teeth, and gears 2 and 4 have 15 teeth.

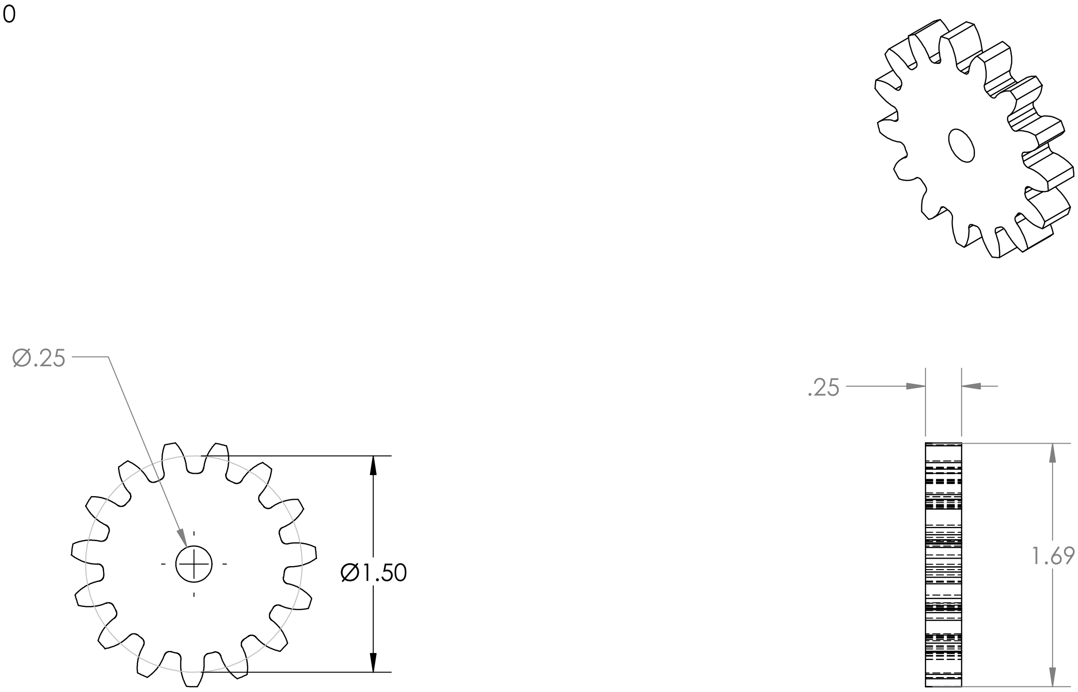
To amplify the motor torque, we used a transmission system that consists of two components: a gear transmission system (Figure 1), and a pulley that splits the tensions. The gear transmission system uses a compound-gear configuration to prevent interference between the gears (our gear ratio would have too much size difference between the gears). According to our theoretical calculations, we needed a 9-to-1 gear ratio to amplify the motor torque enough to lift the 10-pound weight. In the end, to prevent our gearbox from displacing (and the shafts for the gears) too much, we added a pulley that would halve the effective weight carried by the gearbox. The wire used is tied onto the boom arm to split the tension. We used the steel wire instead of the fishing line, as the fishing line would require additional torque, as the fishing line deforms substantially. This steel wire is tied into a hole in the aluminum shafts.

1.2. Boom and Structural Support Systems

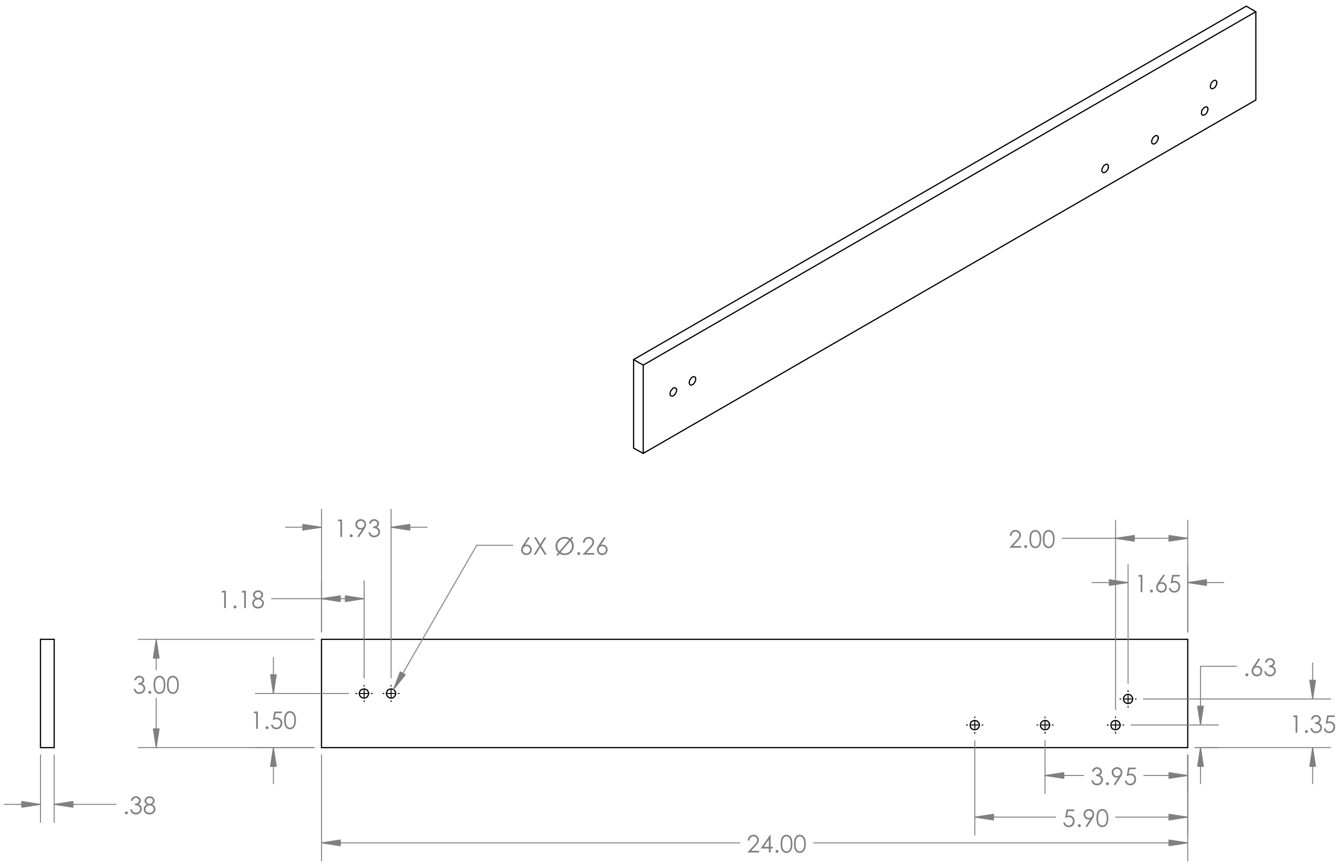
Our boom arm is made of a ⅜″ x 3″ x 24″ sheet of acrylic, with a pulley attachment on the free end, and with supports that consist of aluminum brackets. The pulley attachment is made with two acrylic plates with a plastic shaft that houses the pulley. The two plates are fastened onto the boom arm using screws. The gearbox, made of 0.25″ acrylic walls, is clamped to the mounting plate using aluminum brackets. Aluminum shafts are clearance fitted between the walls to allow for the acrylic gears (press fitted onto the shafts) to rotate. To ensure the aluminum shafts don't shift laterally, we used plastic tubing on the ends that hold onto the shafts with friction. To make sure the pulley is always aligned with the wire that is attached to the aluminum shafts, a threaded rod is used that connects the gearbox to the boom arm, using nuts to secure each in a specific location. Spacers keep the wire within the region that prevents the boom arm from deforming laterally. The gear motor assembly is attached to one side of the gearbox using nuts.

2. CAD Drawings

NOTES:
PRESSURE ANGLE: 20
NUMBER OF TEETH: 15
PITCH: 10

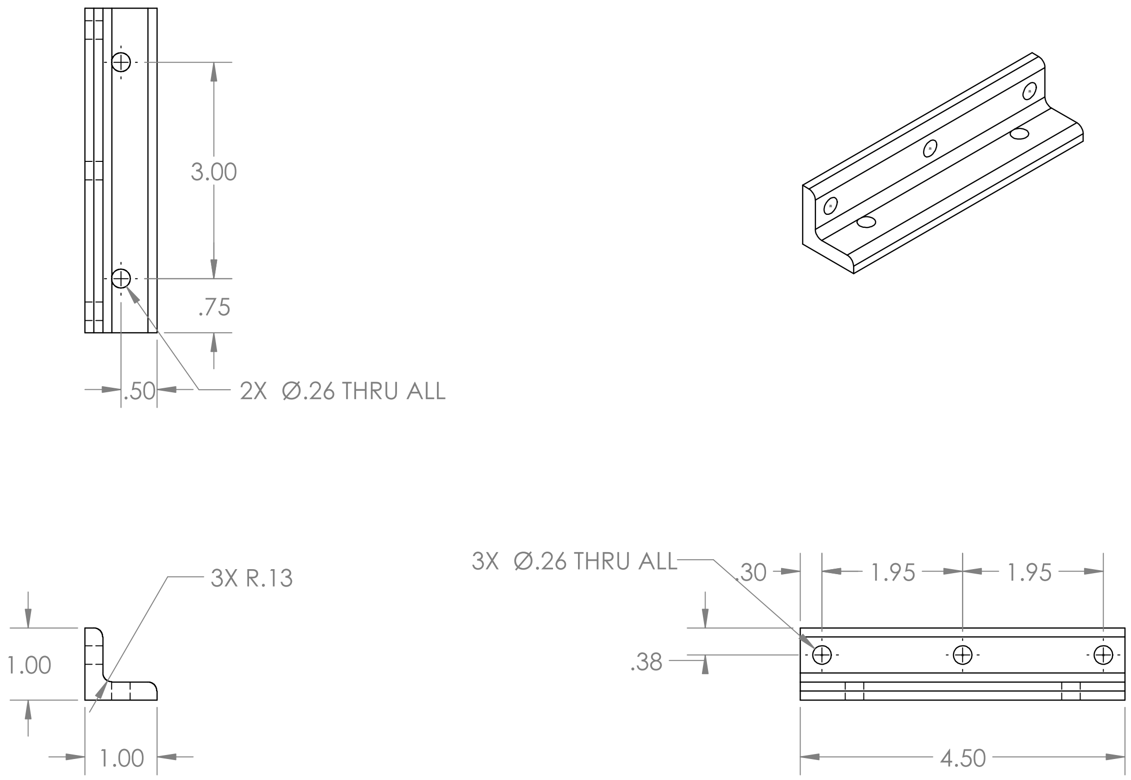


530.216 MBD LAB	TITLE 15 Tooth Gear		UNITS INCHES	SCALE 1:1	MATERIAL Acrylic		QTY. REQ. 2
	TEAM PGH Inc.	DRAWN BY Ronald Garcia			DATE 4/29/2024	DWG. NO. 1	REV. NO. 1



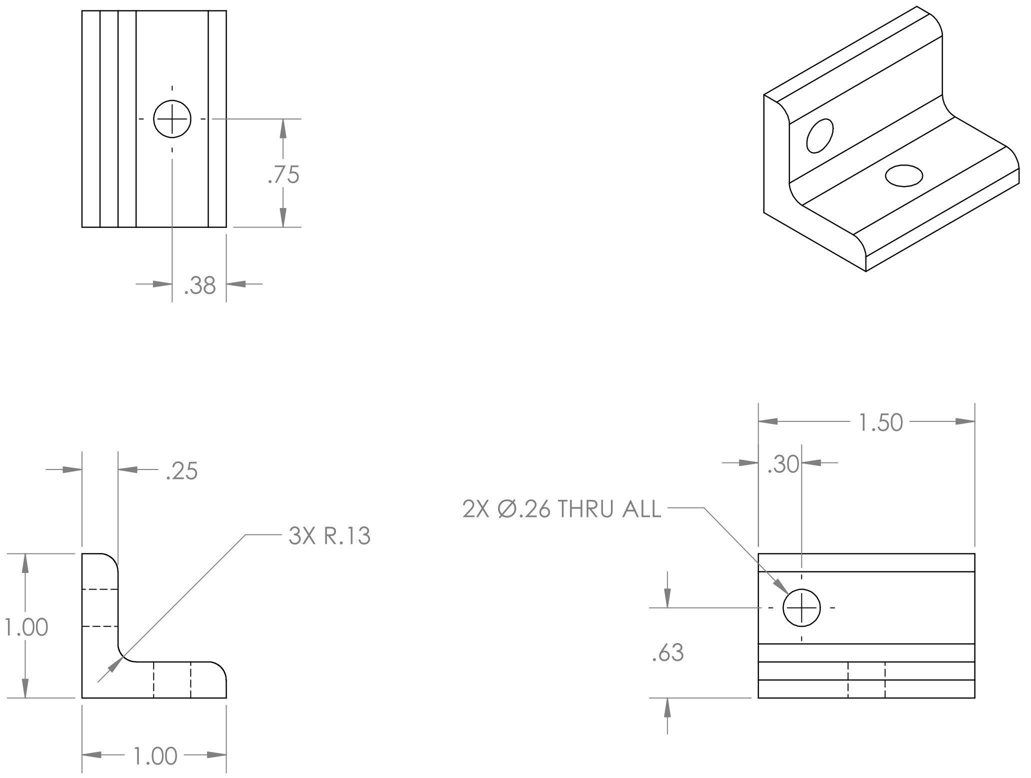
530.216 MBD LAB	TITLE Boom Arm	UNITS INCHES	SCALE 1:4	MATERIAL Acrylic	QTY. REQ. 1
	TEAM PGH inc	DRAWN BY Lizbeth Peralta	DATE 4/29/24	DWG. NO. 2	REV. NO. 1

5 4 3 2 1



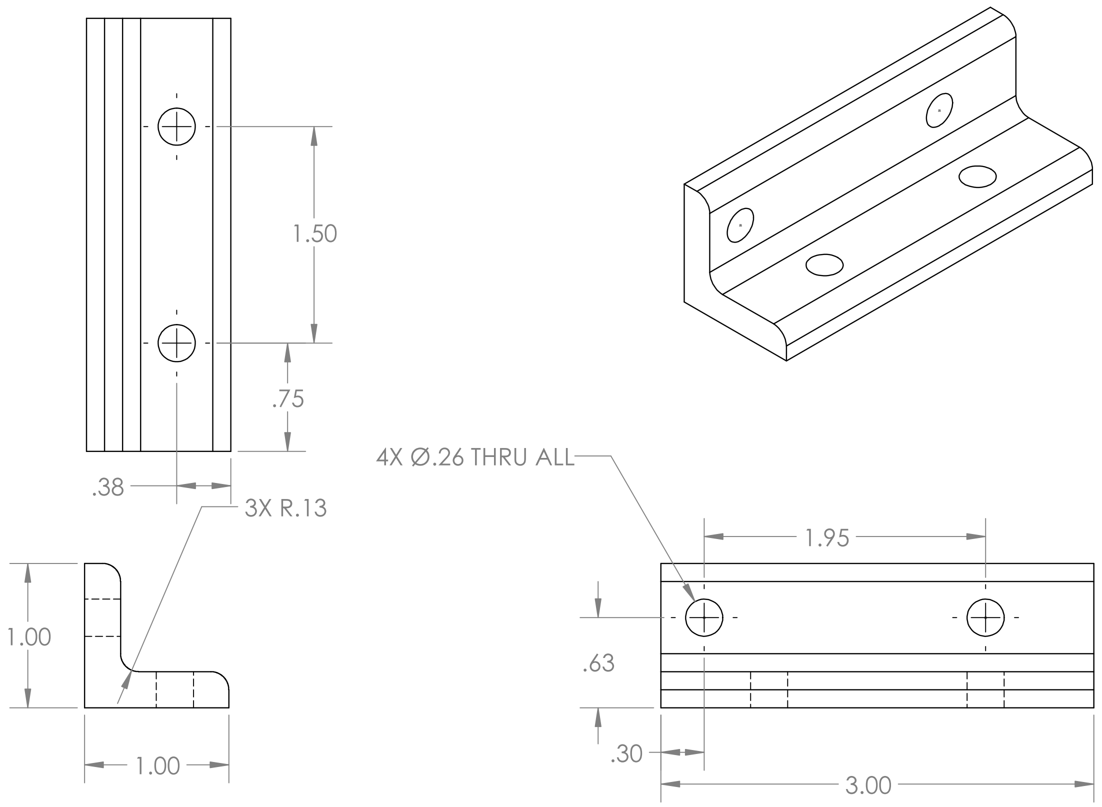
530.216 MBD LAB	TITLE Long Left Bracket		UNITS INCHES	SCALE 1:2	MATERIAL Aluminum		QTY. REQ. 1
	TEAM PGH Inc.	DRAWN BY Ronald Garcia			DATE 4/29/2024	DWG. NO. 3	REV. NO. 1

5 4 3 2 1



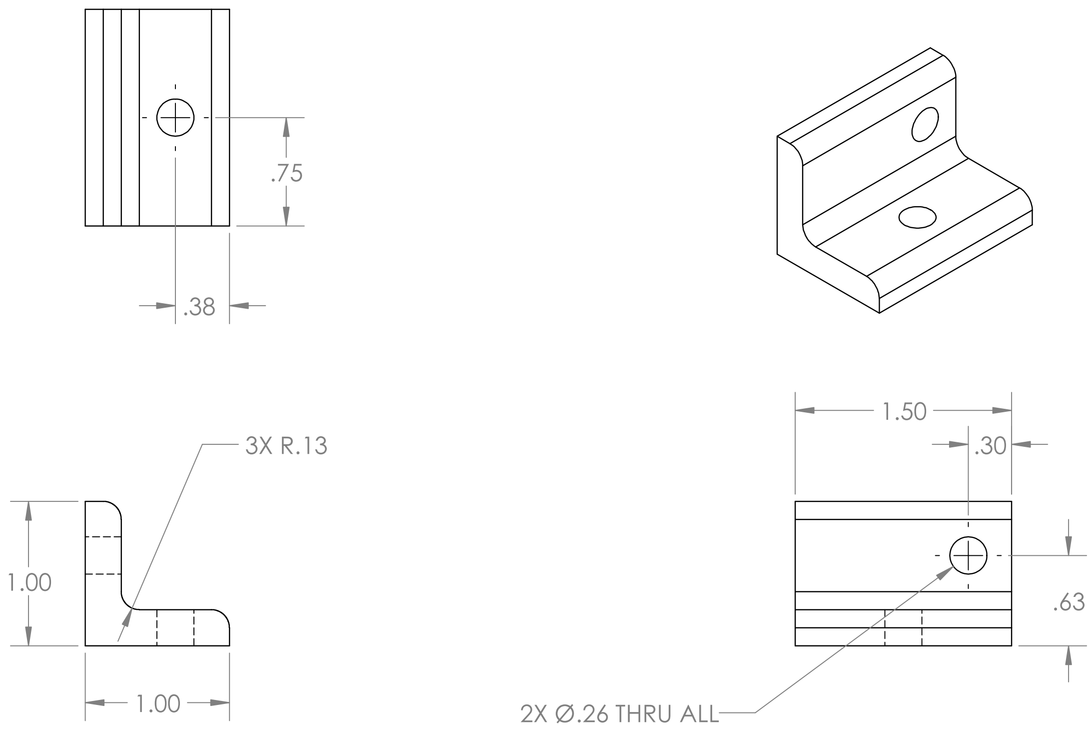
530.216 MBD LAB	TITLE Short Left Bracket		UNITS INCHES	SCALE 1:1	MATERIAL Aluminum		QTY. REQ. 2
	TEAM PGH Inc.	DRAWN BY Ronald Garcia			DATE 4/29/2024	DWG. NO. 4	REV. NO. 2

5 4 3 2 1



530.216 MBD LAB	TITLE Long Right Bracket	UNITS INCHES	SCALE 1:1	MATERIAL Aluminum	QTY. REQ. 1
	TEAM pgh inc.	DRAWN BY Lizbeth Peralta	DATE 4/29/2024	DWG. NO. 5	REV. NO. 1

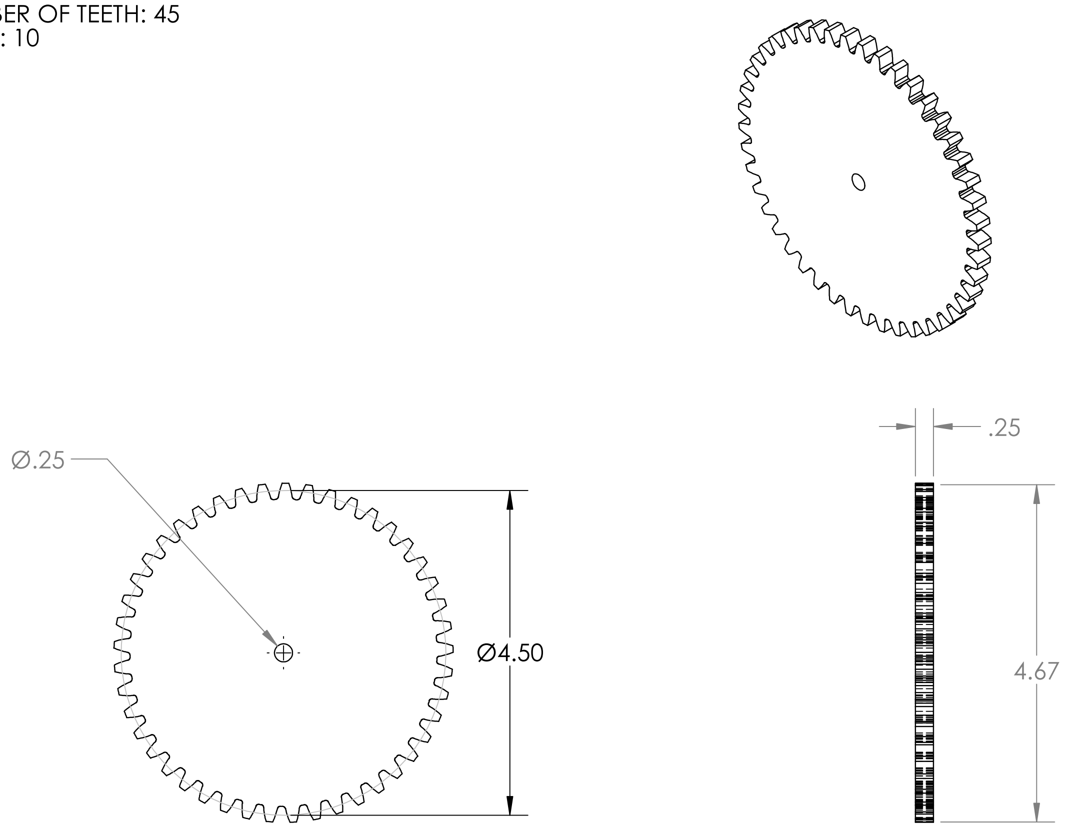
5 4 3 2 1



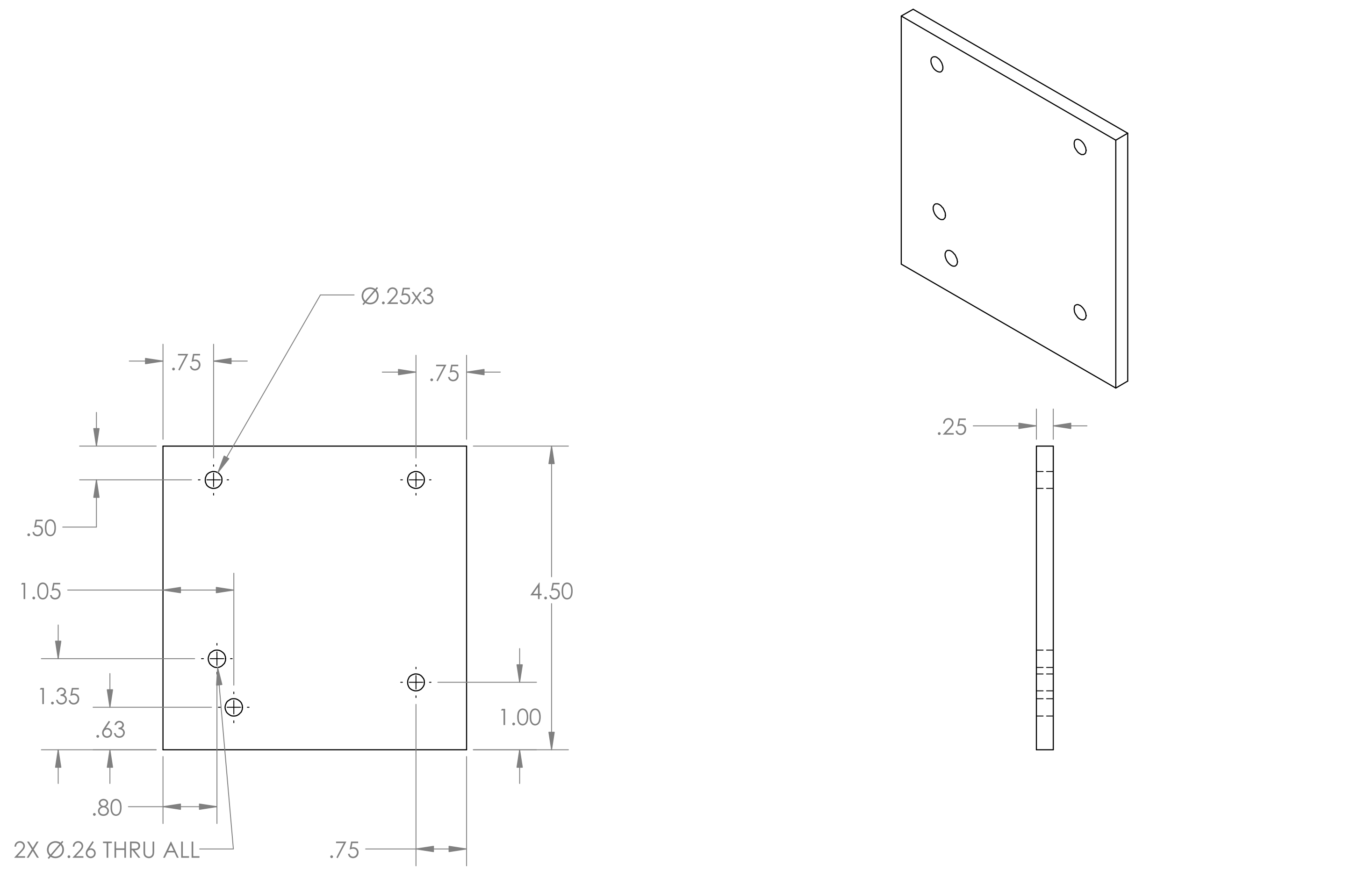
530.216 MBD LAB	TITLE Short Right Bracket		UNITS INCHES	SCALE 1:1	MATERIAL Aluminum	QTY. REQ. 2
	TEAM PGH Inc.	DRAWN BY Lizbeth Peralta		DATE 4/29/2024	DWG. NO. 6	REV. NO. 1

5 4 3 2 1

NOTE:
PRESSURE ANGLE: 20
NUMBER OF TEETH: 45
PITCH: 10

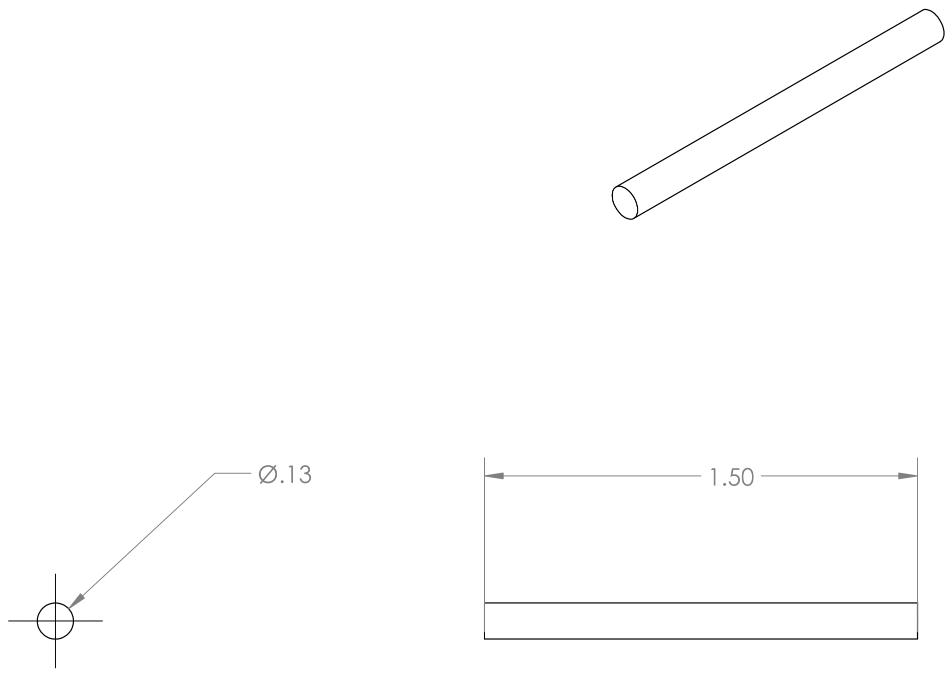


530.216 MBD LAB	TITLE: 45 Tooth Gear	UNITS: INCHES	SCALE: 1:2	MATERIAL: Acrylic	QTY. REQ.: 2
	TEAM: PGH inc	DRAWN BY: Lizbeth Peralta	DATE: 4/29/24	DWG. NO.: 7	REV. NO.: 1



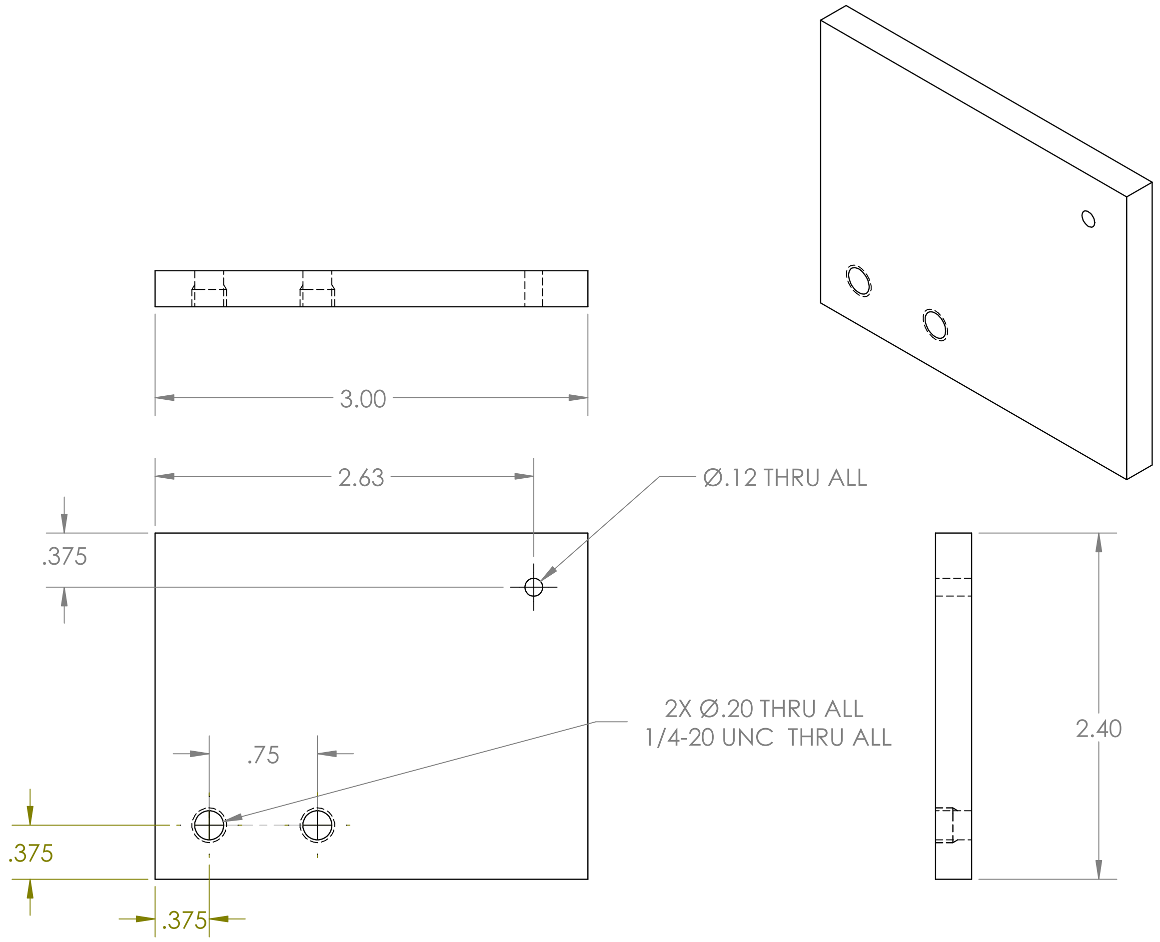
530.216 MBD LAB	TITLE: Gearbox Wall	UNITS: INCHES	SCALE: 1:2	MATERIAL: Acrylic	QTY. REQ.: 1
	TEAM: PGH Inc.	DRAWN BY: Ronald Garcia	DATE: 4/29/2024	DWG. NO.: 8	REV. NO.: 1

5 4 3 2 1



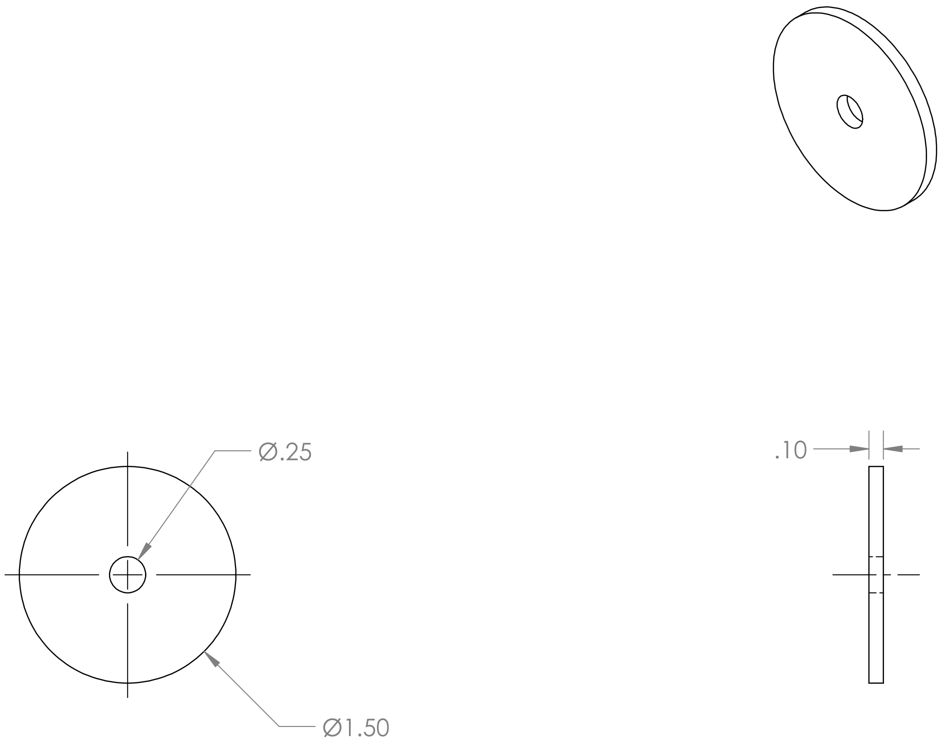
530.216 MBD LAB	TITLE Pulley Shaft		UNITS INCHES	SCALE 2:1	MATERIAL Acrylic	QTY. REQ. 1
	TEAM PGH inc	DRAWN BY Lizbeth Peralta		DATE 4/29/24	DWG. NO. 9	REV. NO. 1

5 4 3 2 1



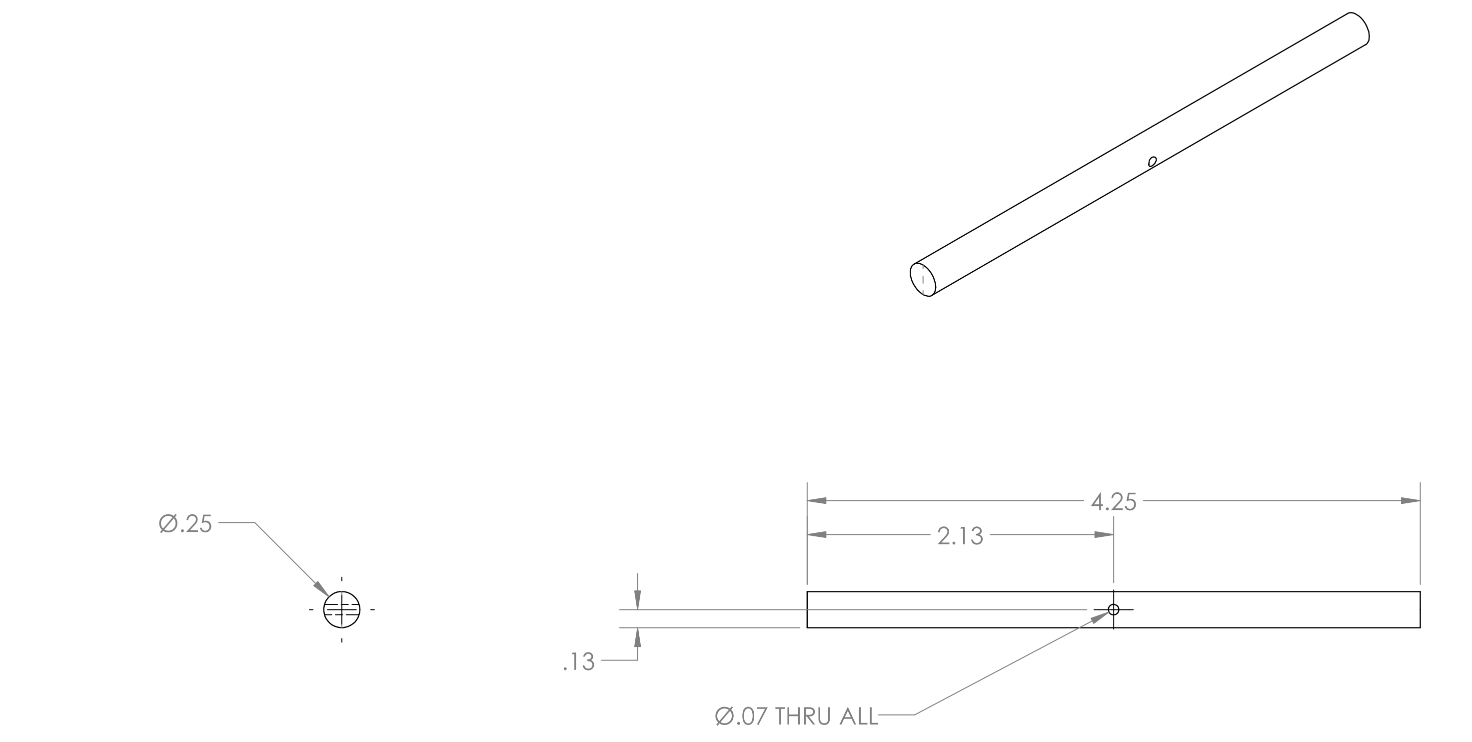
530.216 MBD LAB	TITLE: Pulley Attachment Plate		UNITS: INCHES	SCALE: 1:1	MATERIAL: Acrylic		QTY. REQ.: 2
	TEAM: PGH inc	DRAWN BY: Lizbeth Peralta			DATE: 4/29/2024	DWG. NO.: 10	REV. NO.: 1

5 4 3 2 1



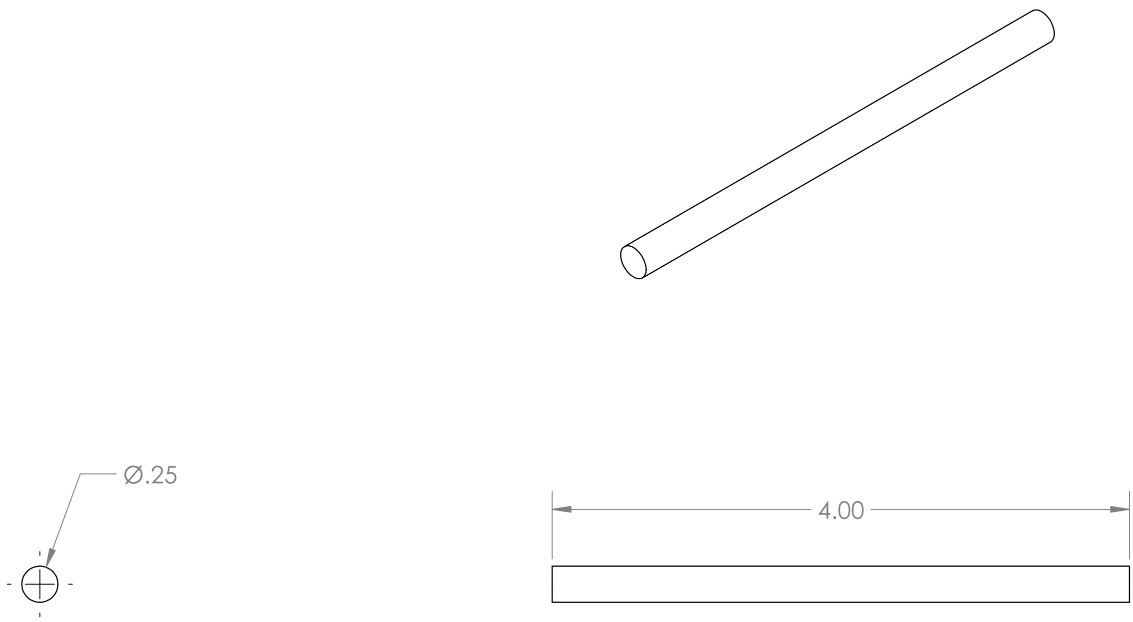
530.216 MBD LAB	TITLE Crane Spacer	UNITS INCHES	SCALE 4:1	MATERIAL Acrylic	QTY. REQ. 2
	TEAM PGH Inc.	DRAWN BY Ronald Garcia	DATE 4/29/2024	DWG. NO. 11	REV. NO. 1

5 4 3 2 1



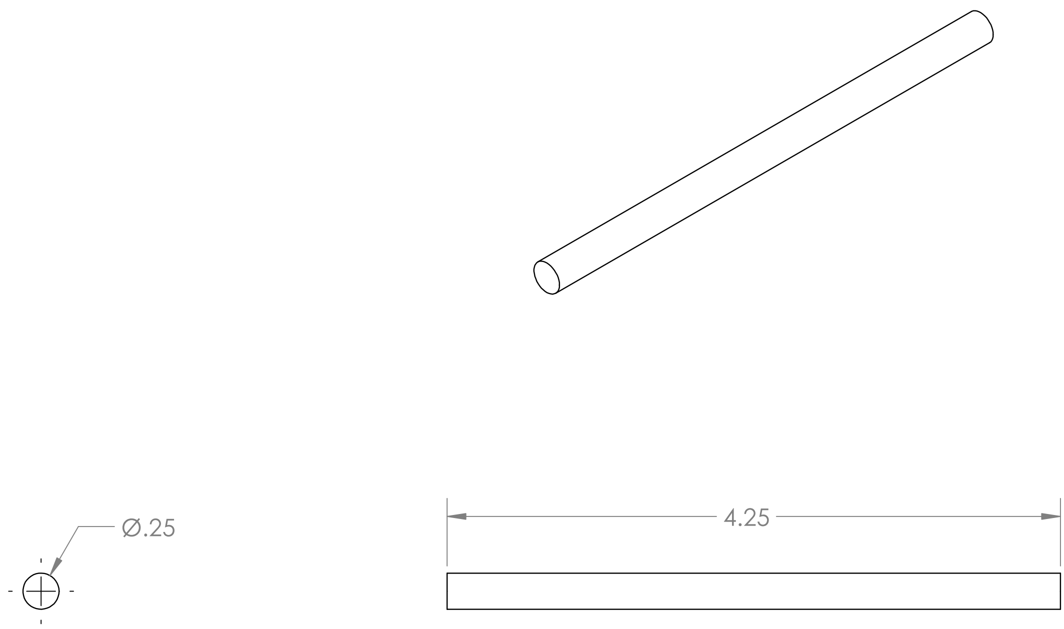
530.216 MBD LAB	TITLE: Output Shaft	UNITS: INCHES	SCALE: 1:1	MATERIAL: Aluminum	QTY. REQ.: 1
	TEAM: PGH Inc.	DRAWN BY: Ronald Garcia	DATE: 4/29/2024	DWG. NO.: 12	REV. NO.: 1

5 4 3 2 1



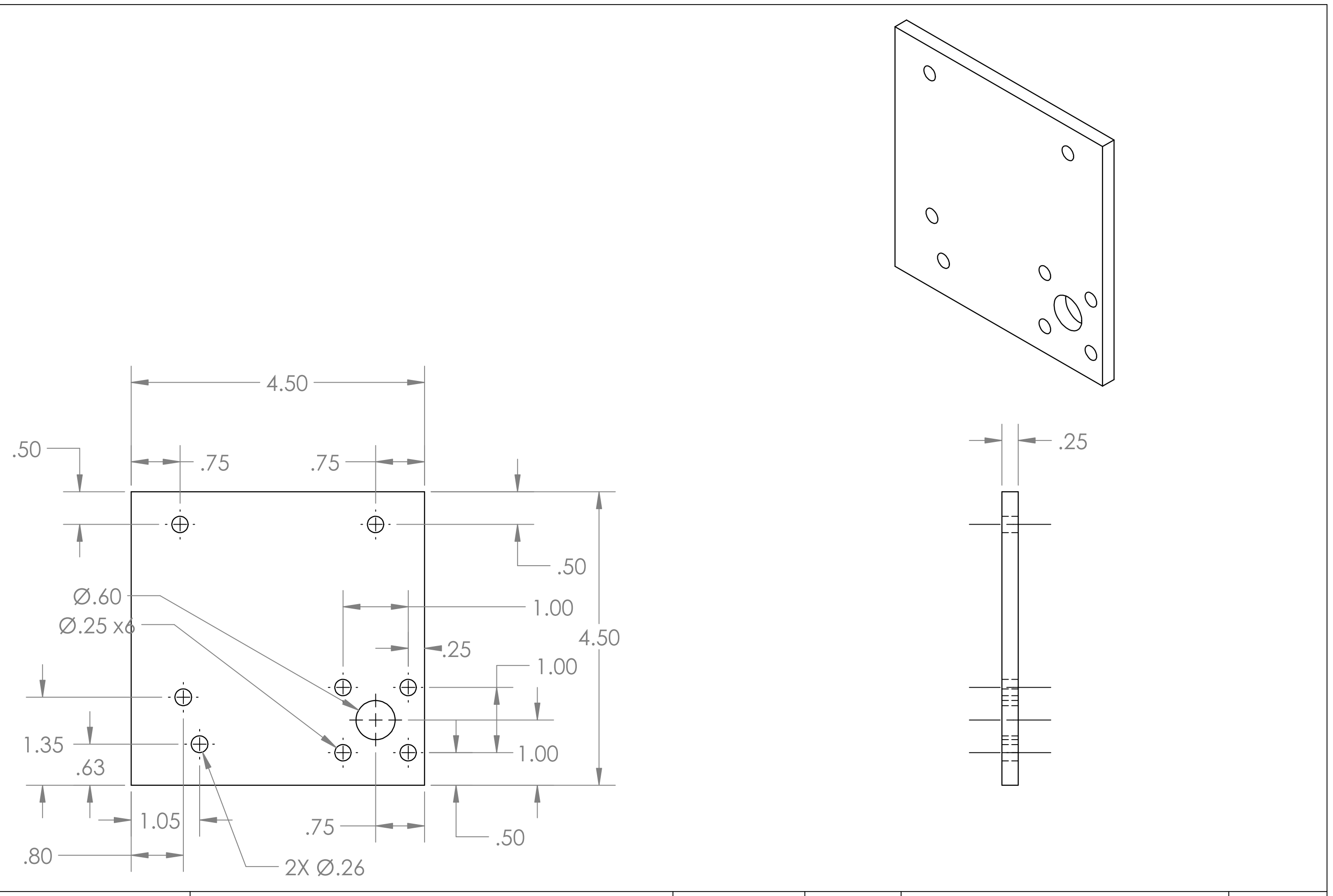
530.216 MBD LAB	TITLE Motor Shaft		UNITS INCHES	SCALE 1:1	MATERIAL Aluminum	QTY. REQ. 1
	TEAM PGH Inc.	DRAWN BY Ronald Garcia		DATE 4/29/2024	DWG. NO. 13	REV. NO. 1

5 4 3 2 1



530.216 MBD LAB	TITLE Crane Shaft	UNITS INCHES	SCALE 1:1	MATERIAL Aluminum	QTY. REQ. 2
	TEAM PGH Inc.	DRAWN BY Ronald Garcia	DATE 4/29/2024	DWG. NO. 14	REV. NO. 1

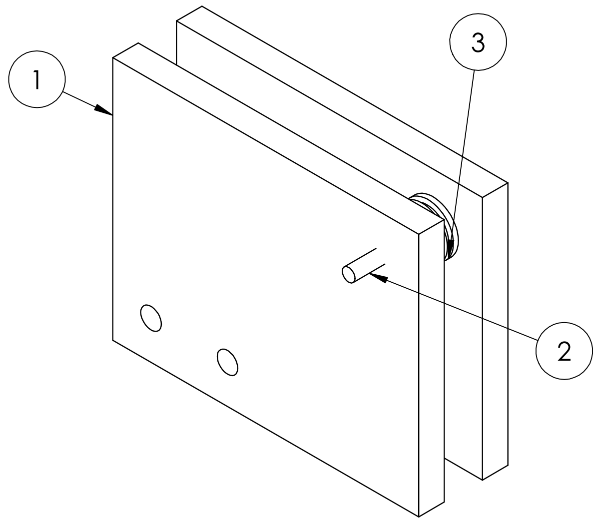
5 4 3 2 1



530.216 MBD LAB	TITLE Crane Gearbox Motor		UNITS INCHES	SCALE 1:2	MATERIAL Acrylic	QTY. REQ. 1
	TEAM PGH Inc.	DRAWN BY Ronald Garcia		DATE 4/29/2024	DWG. NO. 15	REV. NO. 1

5 4 3 2 1

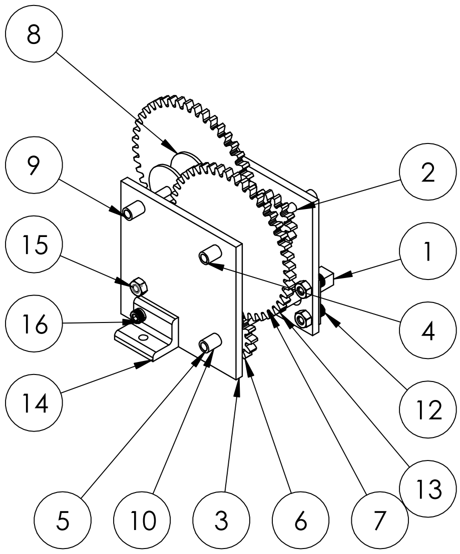
ITEM NO.	PART NUMBER	DRAWING NUMBER	QTY.
1	PULLEY ATTACHMENT PLATE	10	2
2	PULLEY SHAFT	9	1
3	3434T21	N/A	1



530.216 MBD LAB	TITLE Pulley Attachment Assembly	UNITS INCHES	SCALE 1:1	MATERIAL Mixed	QTY. REQ. 1
	TEAM PGH Inc.	DRAWN BY Ronald Garcia	DATE 4/29/2024	DWG. NO. 16	REV. NO. 1

5 4 3 2 1

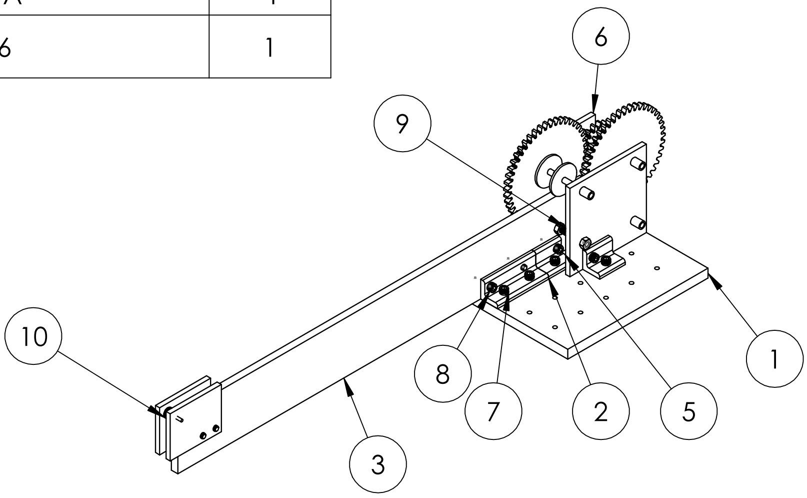
ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	GEARMOTORSUBASSEMBLY	N/A	1
2	CRANE_GEARBOX_MOTOR	15	1
3	CRANE_GEARBOX	8	1
4	CRANE_SHAFT	14	1
5	CRANE_SHAFT_MOTOR	13	1
6	CRANE_15RUSHGEARS_2020_1128-3395	1	2
7	CRANE_45RUSHGEARS__1176-9718	7	2
8	CRANE_SPACER	11	2
9	CRANE_SHAFT_OUTPUT	12	1
10	5233K56	19	5
11	8982K191	6	1
12	91251A537	N/A	4
13	95462A029	N/A	10
14	8982K191	4	1
15	90322A652	N/A	1
16	91251A540	N/A	2



530.216 MBD LAB	TITLE Gearbox Assembly	UNITS INCHES	SCALE 1:4	MATERIAL Mixed	QTY. REQ. 1
	TEAM PGH Inc.	DRAWN BY Lizbeth Peralta	DATE 4/29/2024	DWG. NO. 17	REV. NO. 1

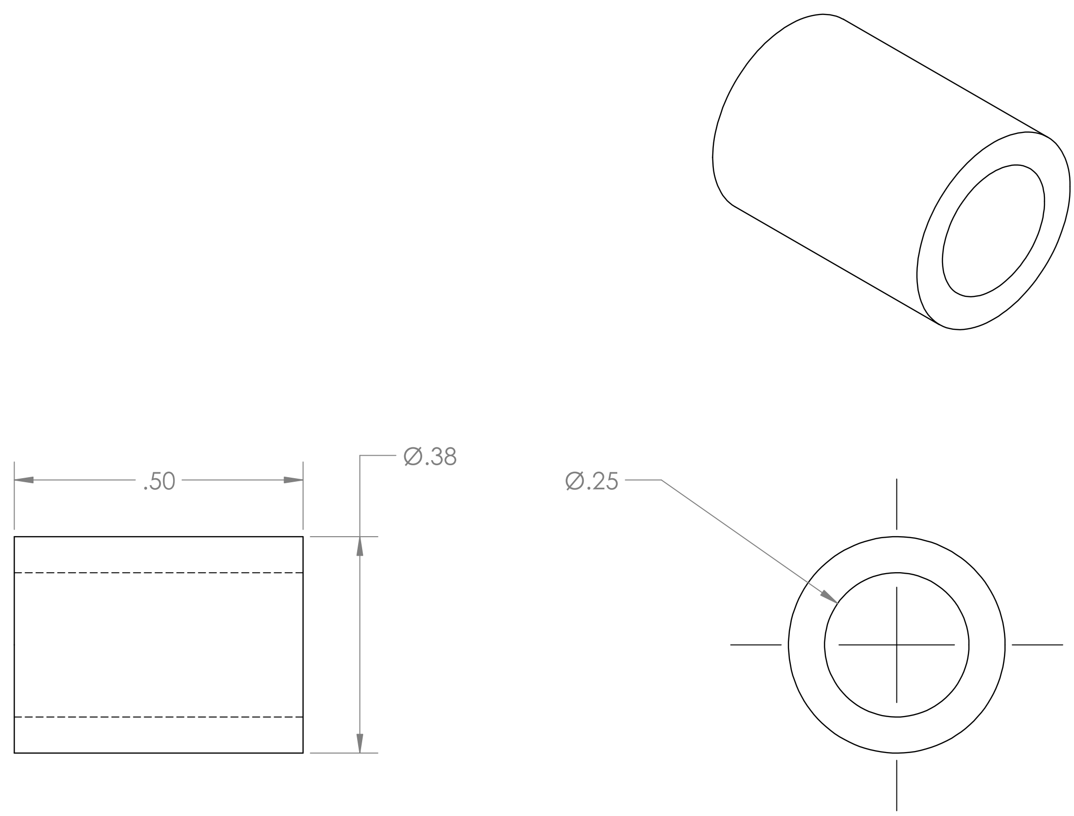
5 4 3 2 1

ITEM NO.	PART NUMBER	DRAWING NUMBER	QTY.
1	CRANE MOUNTING PLATE	N/A	1
2	LONG RIGHT BRACKET	5	1
3	CRANE BOOM	2	1
4	LONG LEFT BRACKET	3	1
5	SHORT RIGHT BRACKET	6	1
6	GEARBOX ASSEMBLY	17	1
7	91251A540	N/A	7
8	91251A542	N/A	5
9	95462A029	N/A	4
10	PULLEY ATTACHMENT ASSEMBLY	16	1



530.216 MBD LAB	TITLE: Crane Assembly	UNITS: INCHES	SCALE: 1:6	MATERIAL: Mixed	QTY. REQ.: 1
	TEAM: PGH Inc.	DRAWN BY: Victoria Hernandez	DATE: 4/29/2024	DWG. NO.: 18	REV. NO.: 1

5 4 3 2 1



530.216 MBD LAB	TITLE Plastic Tubing	UNITS INCHES	SCALE 4:1	MATERIAL Plastic	QTY. REQ. 5
	TEAM PGH Inc.	DRAWN BY Victoria Hernandez	DATE 4/29/2024	DWG. NO. 19	REV. NO. 1

5 4 3 2 1

3. Structural Design Factor Decisions

The first thing to consider, before any calculation, is our chosen design factor.

Consider the equation (the equation, along with the reasoning used for each term, is taken from [1]):

$$DF = DF_{\text{material}} \cdot DF_{\text{strain/displacement}} \cdot DF_{\text{geometry}} \cdot DF_{\text{failure analysis}} \cdot DF_{\text{reliability}}$$

- Choosing DF_{material}

Because the material properties are known from a handbook or are manufacturer's values, we choose $DF_{\text{material}} = 1.1$

- Choosing $DF_{\text{strain/displacement}}$

Because the load is well defined (a ten-pound weight, with specific distances away from the crane), but there can be some deviations, we choose $DF_{\text{strain/displacement}} = 1.1$

- Choosing DF_{geometry}

Because, ideally, our tolerances when manufacturing are average, but we need a few tight tolerances, we choose $DF_{\text{geometry}} = 1.1$.

- Choosing $DF_{\text{failure analysis}}$

Because the failure analysis is of bending, so we choose $DF_{\text{failure analysis}} = 1.1$

- Choosing $DF_{\text{reliability}}$

Because we want the crane to be 100% reliable, we choose $DF_{\text{reliability}} = 1.3$

Therefore, we have a design factor equal to:

$$\boxed{DF_{\text{struct}} = 1.1 \times 1.1 \times 1.1 \times 1.2 \times 1.3 = 1.9033}$$

4. Power Transmission Design Factor Decisions

The first thing to consider, before any calculation, is our chosen design factor.

Consider the equation (the equation, along with the reasoning used for each term, is taken from [1]):

$$DF = DF_{\text{motor specs}} \cdot DF_{\text{required torque}} \cdot DF_{\text{friction}} \cdot DF_{\text{motor power}} \cdot DF_{\text{reliability}}$$

- Choosing $DF_{\text{motor specs}}$

Because the motor specs are known from a handbook or are manufacturer's values, we choose $DF_{\text{motor specs}} = 1.2$

- Choosing $DF_{\text{required torque}}$

Because the required torque is well defined (a torque that depends on the diameter of the winch we are using), but there can be some deviations, we choose $DF_{\text{required torque}} = 1.1$

- Choosing DF_{friction}

Because we are expecting a good amount of friction between our shafts, we can say $DF_{\text{friction}} = 1.2$.

- Choosing $DF_{\text{motor power}}$

Because the motor power given by the manufacturer, we choose $DF_{\text{failure analysis}} = 1.1$

- Choosing $DF_{\text{reliability}}$

Because we want the crane to be 100% reliable, we choose $DF_{\text{reliability}} = 1.3$

Therefore, we have a design factor equal to:

$$\boxed{DF_{\text{power}} = 1.1 \times 1.1 \times 1.2 \times 1.1 \times 1.5 = 2.2651}$$

5. Design Calculations

After deciding on our design factors, we now began designing our boom arm dimensions, power transmission system, and possible stress concentrations.

5.1. Mechanical Calculations

5.1.1. Loading and Deformations

Knowns:
- $DF_{\text{struct}} = 1.903$
- $W = 10$ lb
- $W_{\text{DF}} = 10 \times 1.903 = 19.03$ lb
- $\ell = 21$ in
- $a = 3$ in
- $E_{\text{acrylic}} = 464$ ksi [2]
- $b = \frac{3}{8}$ in
- $h = 3$ in
- $I = \frac{1}{12}bh^3 = 0.84375$ in^4
- $\sigma_{y,C,\text{ acrylic}} = 17$ ksi [3]
- $\sigma_{y,T,\text{ acrylic}} = 10.75$ ksi [3]

Want-to-Finds:
- σ_{max}
- y_{max}

Assumptions:
- This beam can be modeled as a simply supported beam (as described in [4]).
- There is only bending stress causing deflection, and there is axial stress from the pulley attachment plates.
- This beam is slender, ($3 \ll 21$), so the shear stress can be ignored.
- Linear elastic, as well as isotropic material.
- The weight being lifted is 10 pounds.

We want y_{max}, so, using deflection equations from [4],

$$y_{\text{max}} = -\frac{lW_{\text{DF}}(l-a)^2}{3EI}$$

$$y_{\text{max}} = -\frac{21 \times 19.03 \times (21-3)^2}{3 \times 464000 \times 0.84375}$$

$$\boxed{\Rightarrow y_{\text{max}} = 0.110 \text{ in}}$$

We want σ_{max}. This stress has two components, the stress caused by the bending moment σ_B, and the stress caused by the pulley pulling on the weight σ_A.

- Finding σ_B

We know the stress caused by a bending moment $M(x)$ is given by:

$$\sigma(x,y) = -\frac{M(x)y}{I}$$

Where x is the distance along the beam, y is the vertical distance away from the centroid of the cross-section, and I is the moment of inertia of the cross-section.

The maximum vertical distance away from the centroid for a rectangular cross-section would be half the height (positive or negative). In this case, because we know *a priori* that σ_A is negative, we should choose the negative distance, as that would add to the σ_A (versus subtracting from it).

We know, from [4], that the moment $M(x)$ can be described as:

$$M(x) = \begin{cases} -\frac{W_{\text{DF}}(\ell-a)x}{a} & \text{for } x < a \\ W_{\text{DF}}(x-\ell) & \text{for } x \geq a \end{cases}$$

The maximum moment can be found at $x = a$, with a value $M_{\text{max}} = M(a) = W_{\text{DF}}(a - \ell)$. Therefore:

$$\sigma_B = -\frac{W_{\text{DF}}(a - \ell) \times -\frac{h}{2}}{I} = -\frac{19.03 \times (3 - 21) \times -\frac{3}{2}}{0.84375} = -608.96 \text{ psi}$$

- Finding σ_A

The axial stress caused by the pulley is simply:

$$\sigma_A = -\frac{W_{\text{DF}}}{A} = \frac{19.03}{\frac{3}{8} \times 3} = -16.92 \text{ psi}$$

Now, $\sigma_{\text{max}} = \sigma_B + \sigma_A \Rightarrow -16.92 + -608.96 = -625.88$ psi. Because $|\sigma_{\text{max}}| \ll 17000$ psi, the boom arm will not yield in compression. We can also say the boom arm will not yield in tension, as $|\sigma_{\text{max, tensile}} < \sigma_{\text{max, compression}} \ll 10750$ psi.

Summary:

$$y_{\text{max}} = 0.110 \text{ in}$$

$$\sigma_{\text{max}} = -625.88 \text{ psi}$$

This boom arm will not yield, and it will meet the specifications.

Knowns:

- $DF_{\text{struct}} = 1.903$
- $W = 10$ lb
- $W_{\text{DF}} = 10 \times 1.903 = 19.03$ lb
- $\ell = 21$ in
- $a = 3$ in
- $b = \frac{3}{8}$ in
- $h = 3$ in
- $d_{\text{hole}} = 0.257$ in
- $I = \frac{1}{12} b h^3 = 0.84375 \text{ in}^4$
- $\sigma_{y,C,\text{ acrylic}} = 17$ ksi [3]
- $\sigma_{y,T,\text{ acrylic}} = 10.75$ ksi [3]
- $\sigma_{\text{ax}} = \frac{W_{\text{DF}}}{b \times h} = -16.92$ psi

Want-to-Finds:

- $\sigma_{\text{concentration}}$

Assumptions:

- This beam can be modeled as a simply supported beam (as described in [4]).
- There is only bending stress causing deflection, and there is axial stress from the pulley attachment plates.
- This beam is slender, $(3 \ll 21)$, so the shear stress can be ignored.
- Linear elastic, as well as isotropic material.
- The weight being lifted is 10 pounds.

From [4], we have the following equation:

$$M(x) = \begin{cases} -W_{\text{DF}} \frac{\ell - a}{a} x \text{ for } x < a \\ W_{\text{DF}(x-\ell)} \text{ for } x \geq a \end{cases}$$

$$\Rightarrow M_{\text{boom}(x)} = \begin{cases} -19.03 \times \frac{21-3}{3} x \text{ for } x < 3 \\ 19.03(x - 21) \text{ for } x \geq 3 \end{cases}$$

NOTE: because there is both axial and bending stress, we need to find both components and add them together.

- Finding $\sigma_{\text{conc, b}}$

Using the tables on page 1042 (Figure A-15-2) [4], we have:

$$d \sim d_{\text{hole}} = 0.257 \text{ in}$$
$$w \sim h = 3 \text{ in}$$
$$h \sim b = \frac{3}{8} \text{ in}$$

Therefore,

$$\frac{d}{w} = 0.0857$$

$$\frac{d}{h} = 0.6853$$

$$\Rightarrow K_t \approx 2.3$$

Now, because the bending stress is different across the boom arm, we need to calculate the stress at each hole in the boom arm.

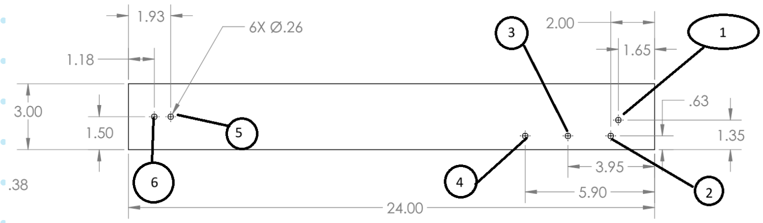


Figure 2: The positions of the holes on the boom arm, labeled from 1 to 6.

Table 1 : The stress concentrators on the boom arm. We have 6 holes, but because we are over-defined, we can consider the boom arm with only two holes that contribute to the support (only x_4 and x_2). Positions (x, c), Bending moment, stress, and stress concentration for all holes are shown below. Because the last support is at x_2, that will be where $x = 0$ for the bending stress equation, where leftward is positive (as shown in Figure 2). Because hole 1 is to the right of the rightmost support, it can be ignored.

Position (x_i, c_i) (in)	**Bending Moment (lb-in)**	$\sigma_0 = \frac{Mc}{I}$ **(psi)**	**Stress Concentration (psi)**
$(0, -0.87)$	$M_{\text{boom}}(x_2) = 0$	$\frac{0 \times -0.87}{0.84375} = 0$	$0 \times 2.3 = 0$
$(1.5, -0.87)$	$M_{\text{boom}}(x_3) = -171.27$	$\frac{-171.27 \times -0.87}{0.84375} = 176.6$	$176.6 \times 2.3 = 406.176$
$(3, -0.87)$	$M_{\text{boom}}(x_4) = -342.54$	$\frac{-342.54 \times -0.87}{0.84375} = 353.197$	$353.197 \times 2.3 = 812.35$
$(19.07, 0)$	$M_{\text{boom}}(x_5) = 36.7279$	$\frac{36.7279 \times 0}{0.84375} = 0$	$0 \times 2.3 = 0$
$(19.82, 0)$	$M_{\text{boom}}(x_6) = 22.4554$	$\frac{-22.459 \times 0}{0.84375} = 0$	$0 \times 2.3 = 0$

The maximum seems to be at hole 3, with a stress $\sigma_{\text{conc, b}} = 812.35$ psi.

- Finding $\sigma_{\text{conc, a}}$

Using the tables on page 1042 (Figure A-15-1) [4], we have:

$$d \sim d_{\text{hole}} = 0.257 \text{ in}$$
$$w \sim h = 3 \text{ in}$$
$$t \sim b = 0.375 \text{ in}$$

Therefore,

$$\frac{d}{w} = 0.0857$$

$$\Rightarrow K_t \approx 2.7$$

Now,

$$\sigma_0 = -\frac{19.03}{(3 - 0.257) \times 0.375} = -18.50 \text{ psi}$$

$$\Rightarrow \sigma_{\text{conc, a}} = |2.7 \times -18.5| = 49.95 \text{ psi}$$

Now,

$$\boxed{\sigma_{\text{conc}} = \sigma_{\text{conc, b}} + \sigma_{\text{conc, a}} = 812.35 + 49.95 = 862.3 \text{ psi}}$$

Because $\sigma_{\text{conc}} \ll \sigma_{\text{y, C, acrylic}}, \sigma_{\text{y, T, acrylic}}$, the boom arm will withstand the stress concentrations.

5.1.2. Gear Tooth Bending Stress Analysis

Knowns:

- $DF_{\text{struct}} = 1.903$
- $W = \frac{10}{2} = 5$ lb
- $W_{\text{DF}} = 5 \times 1.903 = 9.515$ lb
- $b = 0.25$ in
- $\sigma_{y,C,\text{ acrylic}} = 17$ ksi [3]
- $\sigma_{y,T,\text{ acrylic}} = 10.75$ ksi [3]
- $\varphi = 20°$
- $P = 10$
- $N_{\text{teeth}} = 15, 45$
- $d_{\text{small}} = \frac{15}{10} = 1.5$ in
- $d_{\text{big}} = \frac{45}{10} = 4.5$ in
- $Y_{\text{small}} = 0.29$ [5]
- $Y_{\text{big}} = 0.399$ [5]
- $d_{\text{shaft}} = 0.25$ in

Want-to-Finds:

- σ_{bending}

Assumptions:

- The full load is applied at the tip of a single tooth.
- The radial force component is negligible.
- Uniformly distributed load across tooth face.
- Sliding friction forces are negligible.
- Stress concentration at fillet is negligible.
- No slippage of the gears on the shaft (the power is conserved within a shaft).
- The weight being lifted is half of the 10 pound weight (a pulley halves the weight exactly).

We want σ_{bending}, so we can use the Lewis stress equation (found in [5]):

$$\sigma_{\text{bending}} = \frac{F^t P}{bY}$$

Where F^t is the tangential force applied to the gear, P is the pitch of the gears, b is the thickness of the gears, and Y is the Lewis Form Factor.

To find F^t, we can use the following fact (found in [5])

$$F^t = \frac{T_{\text{transmitted}}}{r}$$

Where $T_{\text{transmitted}}$ is the torque transmitted to the gear, and r is the radius of the gear.

Table 2: Calculation of the bending stresses on each of the gears in the gear train of our design. We are taking advantage of the fact that the tangential force at the tip between two meshing gears is the same. Gear labels from Figure 1.

Gear	r	F^t calculation	σ_{bending}
Output Gear (Gear 1)	$\frac{4.5}{2} = 2.25$ in	$\frac{9.515 \times \frac{0.25}{2}}{2.25} = 0.529$ lb	$\frac{0.529 \times 10}{0.25 \times 0.399} = 53.03$ psi
Transmission Gear (Gear 2)	$\frac{1.5}{2} = 0.75$ in	0.529 lb	$\frac{0.529 \times 10}{0.25 \times 0.29} = 72.97$ psi
Tranmission Gear (Gear 3)	$\frac{4.5}{2} = 2.25$ in	$\frac{0.529 \times 0.75}{2.25} = 0.176$ lb	$\frac{0.176 \times 10}{0.25 \times 0.399} = 17.64$ psi
Input Gear (Gear 4)	$\frac{1.5}{2} = 0.75$ in	0.176 lb	$\frac{0.176 \times 10}{0.25 \times 0.29} = 24.28$ psi

Summary:

Based on Table 2, we can see that the stress from each gear $\sigma_i \lll \sigma_{y,C,\text{ acrylic}}, \sigma_{y,T,\text{ acrylic}}$. Therefore, the gears will not yield.

5.2. Power Transmission and Lifting Calculations

Knowns:

- $N_{\text{s, teeth}} = 15$
- $N_{\text{b, teeth}} = 45$

Want-to-Finds:

- R_{eff}

Assumptions:

- There are no significant frictional losses (so our theoretical gear ratio is our actual gear ratio).
- No slippage of the gears on the shaft (the power is conserved within a shaft).
- The gears mesh perfectly, with no backlash.
- The pulley at the end perfectly splits the tension in half.

Going off the gear labeling shown in Figure 1, the gear ratio will be calculated step by step.

Table 3: The gear ratio calculations between each adjacent gear.

Current Stage	Gear Ratio
$4 \to 3$	$R_{4\to3} = \frac{45}{15} = 3$
$3 \to 2$	No mesh, $R_{3\to2} = 1$
$2 \to 1$	$R_{2\to1} = \frac{45}{15} = 3$

$$R_{4\to1} = R_{4\to3} \times R_{3\to2} \times R_{2\to1} = 3 \times 1 \times 3 = 9$$

Because the pulley perfectly divides the tension in half, that is an effective “gear ratio” $R_{\text{pulley}} = 2$.

Therefore,

$$\boxed{R_{\text{eff}} = R_{4\to1} \times R_{\text{pulley}} = 9 \times 2 = 18}$$

Knowns:

- $DF_{\text{power}} = 2.2651$
- $W = 10$ lb
- $W_{\text{DF}} = 10 \times 2.2651 = 22.651$ lb
- $R_{\text{eff}} = 9 \times 2 = 18$
- $d = 0.25$ in
- $T_{\text{stall}} = \frac{3}{4}$ lb-in
- $\omega_{\max} = 140$ RPM

Want-to-Finds:

- T_{motor}
- $\Rightarrow \omega_{\text{motor}}$
- $\Rightarrow \omega_{\text{out}}$
- $\Rightarrow t_{\text{lift}}$

Assumptions:

- There are no significant frictional losses (so our theoretical gear ratio is our actual gear ratio).
- The weight is exactly 10 pounds.
- No slippage of the gears on the shaft (the power is conserved within a shaft).
- There is a linear relationship between motor speed and motor torque.

We know:

$$T_{\text{motor}} = \frac{W_{\text{DF}} \times \frac{d}{2}}{R_{\text{eff}}}$$

Where W_{DF} is the weight of the load, including the design factor, d is the diameter of the shaft, and R_{eff} is the effective gear ratio of our transmission system. Therefore,

$$\boxed{T_{\text{motor}} = \frac{22.651 \times \frac{0.25}{2}}{18} = 0.157 \text{ lb-in}}$$

Now, because we assume there is a linear relation between motor speed and motor torque,

$$\omega(T) = -\frac{\omega_{\max}}{T_{\text{stall}}}(T - T_{\text{stall}})$$

Where $\omega(T)$ is the motor speed as a function of the torque, $\omega_{\max}$ is the maximum motor speed (assuming 0 torque), T_{stall} is the maximum motor torque (assuming 0 motor speed), and T is the current motor torque.

$$\boxed{\Rightarrow \omega(T_{\text{motor}}) = \omega_{\text{motor}} = -\frac{140}{\frac{3}{4}}\left(0.157 - \frac{3}{4}\right) = 110.69 \text{ RPM}}$$

Now, to get the output shaft speed, ω_{out}, we must consider our gear ratio.

$$\omega_{\text{out}} = \frac{\omega_{\text{motor}}}{R_{\text{eff}}}$$

$$\boxed{\Rightarrow \omega_{\text{out}} = \frac{110.69}{18} = 6.15 \text{ RPM} = 6.15 \times \frac{1}{60} \times 2\pi = 0.644 \text{ rad}/s}$$

Now that we know the rotational speed of the output shaft, we know how much the rope is being pulled up. With this, we can find the velocity of the weight v_{lift}, resulting in the height of the weight y_{lift}.

Because we are assuming no slipping,

$$v_{\text{lift}} = \frac{d}{2} \times \omega_{\text{out}}$$

$$\Rightarrow v_{\text{lift}} = 0.0805 \text{ in}/s$$

And $y_{\text{lift}} = \int v_{\text{lift}} dt$,

$$\boxed{\Rightarrow y_{\text{lift}} = 0.0805t \text{ in}}$$

Because we want $y_{\text{lift}} = 1$ in, we can solve for t_{lift} as shown:

$$1 = 0.0805t_{\text{lift}}$$

$$\boxed{\Rightarrow t_{\text{lift}} = 12.42s}$$

Summary:

Because the specification is to lift the weight within 2 minutes, this power transmission would fulfill the specification.

6. Cost Analysis

The total cost of our crane design is $24.85 (part breakdown shown in Table 3).

Table 4: Breakdown of each individual part used in our crane that was purchased from McMaster-Carr.

Part Number [6]	**Item**	**Quantity Used**	**Total**	**Explanation**
8560K611	¼″ Acrylic	$2 \times \pi(0.75)^2 + 2 \times 4.5^2 + 2 \times \pi(2.25)^2 + 2 \times \pi(0.75)^2 - 7.75^2 = 19.3\text{ in}^2$	$19.3\text{ in}^2 \times \$0.08 = \$1.61$	Unit price of ¼″ Acrylic per in^2 calculated from 48″x48″ Acrylic sheet, excluding the provided acrylic. [6].
8974K22	¼″ Aluminum Rods	$3 \times 4 = 12\text{ in}$	$1\text{ ft} \times \$1.48 = \$1.48$	Unit price of ¼″ aluminum rod calculated from 6 ft of aluminum. [6]
3434T21	3/64″ Pulley w/ Ball Bearing	1	$4.90	Unit price retrieved from [6].
8982K191	Aluminum Brackets	12 in	$1\text{ ft} \times \$6.04 = \$6.04$	Unit price calculated using 8 ft long aluminum brackets from [6].
8531K11	⅛″ Acrylic Rod	2 in	$2 \times 0.01 = \$0.02$	Unit price calculated using 6′ of acrylic rod from [6].
1227T359	⅜″ Acrylic Sheet	24 in	$2\text{ ft} \times \$5.38 = \$10.68$	Unit price calculated from 12″x24″ Acrylic sheet from [6].
5233K56	Plastic Tubing	3 in	$3\text{ in} \times \$0.04 = \$0.12$	Unit price calculated using part 5233K56 from [6].

7. Summary of Performance

7.1. Comparing Results

In our theoretical calculations, we factored in any imperfections of our theoretical models, material used, tools used, etc. with a chosen *design factor*. Our theoretical boom arm deflection, as calculated above, is $y_{\text{max, theo}} = 0.110$ in. In our performance evaluation, however, we had a boom arm deflection of $y_{\text{max, real}} = 0.125$ in. If we look at the percent difference, we have:

$$\%_D = \frac{|y_{\text{max, theo}} - y_{\text{max, real}}|}{y_{\text{max, real}}} = 12\%$$

which is quite compatible. Using a t-test, we have $t = \frac{|y_{\text{max, theo}} - y_{\text{max, real}}|}{\partial y_{\text{max, real}}} = 0.24$, where $\partial y_{\text{max, real}}$ is half the precision of the ruler used to measure the deflection, 1/16″. Because $t < 1.6$, these two values are compatible!

7.2. Performance Discussion

The aluminum shafts were successful in their ability to clamp the boom arm to the mounting plate. But, because we only had one pivot point for the gearbox, we had a lot of issues with the gearbox misaligning. This was because of the weight pulling on the shaft. This misalignment would cause the gears to push against each other radially much higher than intended, allowing for much more torque loss than we wanted. To combat this, we implemented a pulley that would halve the weight being pulled by the gearbox. In our performance evaluation, this idea worked exactly as intended, as the misalignment of the gearbox was negligible, and the torque was able to be transmitted effectively.

We were also having issues with the concentricity of the motor coupling. Because of the misalignment (before we implemented the pulley), the shafts were undergoing a significant amount of stress laterally. As a result, the set screw that secured the motor to the coupling shifted. Because of this, the gear on the input shaft (gear 4) would be pushing against gear 3, causing frictional losses. Again, with the implementation of the pulley, we were able to overcome this issue, as the gearbox was lifting half the weight.

Because the pulley that was used had a ball bearing internally, little frictional losses were coming from the pulley.

The thickness of our gearbox walls, the plastic tubing that prevented the aluminum shafts from laterally moving, and the threaded rod that connected the boom arm to the gearbox gave us a very stable gearbox system.

8. Contributions

<table>
<tr><th>Ronald</th><th>Lizbeth</th><th>Victoria</th></tr>
<tr>
<td>

Produced the calculations needed for our report
Machined our L brackets by cutting them the appropiate size and milling the holes at their respective locations
Milled holes into our boom arm
Fixed the gear motor to make the gear shaft spin concentric
Cut our plastic tubing to size
Wrote Summary, Structral Design Factors,and Power Transmission Design Factors as well as the calculations
Figured out that we needed to use mechanical advantage to lift the 10lb weight. To do this we simply split the tension

</td>
<td>

Laser cutted all of the acrylic pieces needed for the crane (pulley attachment plates, spacers, gear box)
Ordered and found parts for crane
Assembled gear box to make sure the gears were meshing and were aligned
Found the press fit needed for our crane and press fitted our gear shafts
Fixed the tap holes by milling them to be clearance holes
Wrote Growth, Improvement and Advice, and Cost Analysis, Contributions

</td>
<td>

Produced all of our Solidworks modeling
Wrote Design Description, Power Transmission and Lifting Systems, and Boom and Structural Support Systems
Measured and cut our shafts to size

</td>
</tr>
<tr><th colspan="3">Group Work</th></tr>
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Worked together to figure out the design of our crane. We figured out the critical dimension of our crane (boom arm thickness, shaft diameters, pitch radius for gears), identified how many gears we were using, flushed out all logistics
We all contributed in producing CAD drawings
Worked on assembly together
Brainstormed ideas to solve our issues with lateral movement which resulted in us purchasing plastic tubing
Tested our crane together
Troubleshoot until our crane was successfully able to lift the 10lbs and have less than 0.5″ of deflection

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</table>

9. Growth, Improvements, and Advice

9.1. Knowledge and skills obtained

1. We learned that the tolerances of each manufacturing process are important to keep in mind. For instance, we realized that the laser cutter's tolerances were not negligible, so working with it tends to be difficult. As a result, we would spend the whole day laser cutting, which would make all of our parts the same. If we were press fitting something we would do it as soon as it came out of the laser cutter to make sure our press fits worked effectively.
2. We learned that not everything needs to be threaded. Originally our group threaded our aluminum L brackets because we assumed they needed to be since the mounting plate was threaded. However, after testing and assembly, we realized that most of our holes could be clearance holes
3. We became much more comfortable machining using the mill and lathe.
4. We learned to make sure the CAD model being used is maintained to the most recent version of the design to ensure that when fabricating parts, the models of the parts are accurate and are compatible with each other.
5. We learned that collaboration is productive and enhances the quality of the work. Not only within one's team but between other teams. Within one's team, being able to trust each other to complete the task assigned is vital to the productivity of a team. Between other teams, sharing/hearing other people's ideas is vital to the ideation process, and is great when troubleshooting one's or other teams' issues with their design.

9.2. Suggestions for improving crane design post-evaluation

1. For our crane we could have used more mechanical advantage. When testing our crane we were only able to lift 9lb but once we split the tension of the wires our crane was able to lift the required 10lb under 20 seconds. By splitting the tension, we could have done more of this, making our crane lift less than 10 pounds.
2. After our crane manufacturing we did not need 3 pegs on the mounting plate to secute our boom arm. This was excessive and resulted in us having problems with the stability of our gearbox. If we were to rebuild our crane we would shift our boom forward giving us more room to secure our gearbox.
3. Instead of buying additional material to prevent lateral movement we could have turned down the size of our rods and made the holes on our gearbox clearance fit to the turned-down diameter.

9.3. Suggestions for improving the design project

1. It would be beneficial if there were more power supplies available. When trying to test our devices there was often a queue for groups to use the power supply. This added a lot more wait time than needed.
2. Getting access to the lab room and the supplies early. It would be nice to start working on the design project as soon as possible but we were only realistically able to after our second design meeting. This gave us two weeks to work on our crane, and these two weeks were packed with exams/projects for other courses.
3. Establishing how deflection and time for the crane were going to be measured. If we found out how it was going to be measured exactly (lifting the weight for deflection, dropping it then lifting it for time) we would have tested like this from the start.

9.4. Advice for next year's students

1. You are going to press fit a lot of parts on your crane but if you are using acrylic you cannot apply too much pressure or your part will break. Our group used press fits of 0.24 and 0.2425 with a 0.25 diameter. We used tapping fluid and a mallet to carefully pound our parts in place.

References

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