Mechanical Design and Manufacturing II Project A

A Mechanism Paperweight

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1. Goal Statement, Gantt Chart, Background

The purpose of this project is to design, manufacture, and analyze a slider-crank system that may be utilized as a paperweight that will sit on a desk. A slider-crank converts rotational motion into linear motion with a four-bar mechanism.

1.1 Goal Statement

The goal of this project was to design, analyze, and manufacture a planar slider-crank mechanism that converts rotary motion into reciprocating linear motion while meeting the dimensional, kinematic, and manufacturing constraints specified in the project guidelines.

The mechanism was required to:

- Fit within a 6.75 in × 4.0 in × 0.25 in baseplate footprint
- Maintain a transmission angle $\mu \geq 40°$ for all crank angles
- Operate smoothly with minimal noise and friction
- Remain fully constrained when inverted (no slider separation)
- Include one to two 3D-printed components
- Be manufacturable using lathe, mill, and drill processes

Success was defined as achieving full 360° crank rotation, continuous slider motion, and agreement between analytical and experimental performance.

1.2 Background

A slider-crank mechanism is a planar four-bar system with one prismatic joint that converts rotational motion into linear motion. It is widely used in internal combustion engines, compressors, and reciprocating pumps due to its compact design and predictable kinematic behavior.

The mechanism consists of four links:

- Ground frame
- Crank (input link)
- Connecting rod (coupler)
- Slider (output link)

One full revolution of the crank produces one complete reciprocating cycle of the slider.

1.3 Gantt Chart

The project schedule was organized using a Gantt chart to track task dependencies and ensure timely completion of analysis, fabrication, and documentation. Concept development and kinematic analysis were completed prior to the February 13 analysis milestone. Manufacturing tasks were scheduled in parallel with report writing to maximize efficiency during build week. Final assembly and testing were completed before the February 20 prototype deadline, followed by report preparation and presentation development for the February 23 submission.

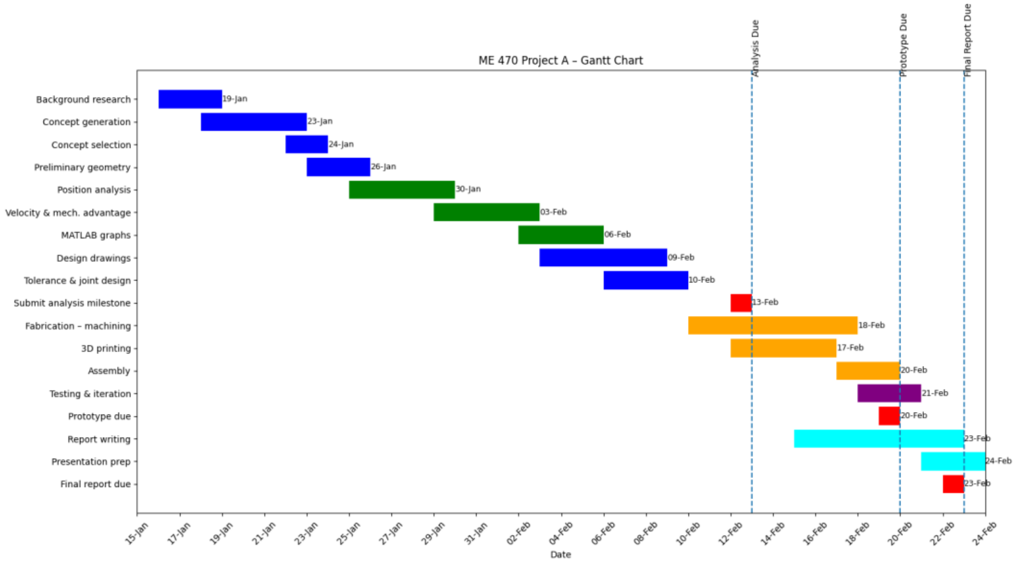

Figure 1 – Gantt Chart.

2. Project Specifications

The project was dedicated to manufacturing a desktop paperweight which could serve as a demonstration of the conversion of rotational motion to linear motion. There were still several manufacturing requirements that the mechanism had to abide by. Those requirements were mentioned above in Chapter 1 but are reiterated below:

- Fit within a 6.75 in × 4.0 in × 0.25 in baseplate footprint
- Maintain a transmission angle $\mu \geq 40°$ for all crank angles
- Operate smoothly with minimal noise and friction
- Remain fully constrained when inverted (no slider separation)
- Include one to two 3D-printed components
- Be manufacturable using lathe, mill, and drill processes

These requirements were considered throughout the entire design and manufacturing process with several team specific specifications regarding creativity and uniqueness being applied as well.

3. Four Design Ideas and Final Selection Rationale

A slider-crank system has multiple components which are joined together to convert rotational motion into its translational equivalent. During the design phase, each component of the system was iterated upon finding balance between smoothness of motion, style, and complexity. There were four main components that needed to be discussed when manufacturing the entire system. Those four categories were: handle, slider, crank, track. Each of these components had multiple iterations which are described below in detail.

3.1 Handle Mechanism

The handle describes the method in which rotational motion is applied to the system. For this, we proposed that a human would be required to apply a moment to the rotational crank. It was also important for us to account for novelty and appeal in the crank mechanism as the purpose of this project was to serve as paperweights.

3.1.1 Linear Handle

Description: A crank would be situated on top of the pin connecting the manufactured wheel of the crank mechanism and the pinned bar.

Advantages: This concept would be easy to manufacture. It would be simple to ensure that the transmission angle fell within the tolerances described in the project

Disadvantages: The sense of novelty involving an actual crank is lost in this idea. A cartoon-like crank design would be preferred by the group. The purpose of this product was to serve as a desk paperweight, which is a very novel idea which should spark curiosity and joy. Changing the cranking mechanism would increase the novelty of the mechanism.

3.1.2 Cartoon Handle

Description: A crank would take on a zigzag “S” shape which would then be fixed to the rotational center of the crank mechanism. This would be used to drive the rotational motion of the slider crank system. A metal rod would have to be machined and bend to create this shape.

Advantages: The novelty of the slider crank system would be present; it would capture a feeling of entertainment and creativity which makes sense for a paperweight.

Disadvantages: It would be difficult to manufacture given the sizes of the baseplate and proportional dimensions of the crank. Bending a thin metal rod often leads to snapping or breakage, which is a significant risk given that the part must be manufactured first.

3.2 Slider Mechanism

The slider is the demonstration of linear motion that is the output of rotational motion. For this project, the two main proposals for the slider revolved around the material in which they were made of. This was significant as it defined the method of manufacturing which was utilized. It is known that for manufacturing beginners, for complicated components it is easier to print the parts compared to traditional manufacturing techniques.

3.2.1 Aluminum Slider

Description: An aluminum slider could be fixed to slide the track. The slider would be manufactured using milling or lathe techniques.

Advantages: This would open the opportunity for another component of this system to be manufactured with 3D printing. According to the project statement, only two components of the slider-crank are allowed to be manufactured using a 3D printer.

Disadvantages: Depending on the complexity of the track, it could become very difficult to machine the slider using a mill or lathe. Given the fact that we are amateur manufacturers, it may not be possible for us to machine a very complicated part.

3.2.2 3D-Printed Slider

Description: A 3D printed slider could be fine-tunned using CAD software to exactly fit the schematics of the design.

Advantages: Since this was the groups second time manufacturing a part in MTL, there was a lack of experience. Group members had more experience using CAD software for personal projects, which meant that complicated designs could be manufactured faster.

Disadvantages: Only two components of the project could be manufactured. This means that using this design we would be limited to designing only one other part using CAD software which could become a problem if there are other more complicated parts that need to be manufactured.

3.3 Crank Mechanism Orientation

The orientation of the crank wheel was design consideration throughout the entire manufacturing process. Two configurations of the crank wheel were proposed, one with the crank situated vertically where the axis of rotation was parallel to the baseplate. The other orientation being where the axis of rotation was normal to the baseplate was classified as the horizontal crank wheel.

3.3.1 Vertical Crank

Description: A wheel would be suspended by two supports. The wheel would be orientated vertically. The wheel could also be built as another shape using 3D printing CAD software to make it more unique.

Advantages: Stylistically, the wheel could be adapted into other shapes using 3D printing. This would create a unique looking mechanism which would look interesting on a desk.

Disadvantages: Added complexity of supporting the crank wheel with milled support on either side of the crank wheel. The support would still have to be manufactured using a mill or a lathe.

3.3.2 Horizontal Crank

Description: A wheel will rest on a pin that will be connected to the baseplate. That wheel will then rotate along the z-axis.

Advantages: A smaller transmission angle due to the lack of suspended support. This configuration would be easier to manufacture since there was a lack of vertical support.

Disadvantages: The wheel shape would be limited to a circular shape. Any unique shape of the wheel would not be noticeable when viewed from a straight on direction of the baseplate. There would be a lack of creativity present in this design.

3.4 Track Mechanism

The track mechanism is defined as the component of the system that restricts the linear motion of the slider. For this project, only one degree of freedom was allowed for the slider which meant that the track's geometry should restrict all types of motion except for linear motion in one direction. It was also defined for the project that the mechanism should work upside down, which meant that the geometry should be manufactured in such a way that the slider is always in contact with the track.

3.4.1 - Trapezoidal Track

Description: As an initial proposal for the track which the slider rested on, an isosceles trapezoidal configuration was proposed. For this, the smaller face was fastened to the baseplate while the slider was able to translate along the bigger face.

Advantages: This mechanism would allow the system to operate upside down. The slider would not be able to fall off the track.

Disadvantages: This track would be difficult to machine and be more time-consuming in the manufacturing lab, which we already have a limited amount of time in. On top of that, the geometry of the slider would be difficult if not impossible to manufacture in a way other than 3D-printing.

3.4.2 - T-Beam Track

Description: After attending the manufacturing lab, it became apparent that the initial idea for the track mechanism was much too complicated to complete in the weeks' worth of time given to manufacture the project. A T-Beam Track involved the fastening of two bars of material with a wider bar situated above a square beam of material with holes which were drilled into the baseplate.

Advantages: This mechanism was extremely easy and fast to manufacture. It only involved the use of a band saw, and the drill press compared to other methods with the added complexity of using the mill for the part.

Disadvantages: It was still very challenging to machine a slider part that could be attached to the track. Therefore, the slider component still needed to be manufactured using 3-D printing.

3.5 Final Selection Rationale

Looking back at the design and manufacturing of this project, it was very apparent that ambition was the driving factor in the design decisions for this project. For this project, creativity was prioritized throughout the design process. It was important to the group to create an eye-catching design that varied from projects done in the past. Besides that, the group decided to consider the size constraints of the baseplate and possible ways to maximize the footprint of the slider-crank mechanism so it could be enjoyed from far away distances. Another consideration for the project was the cost of the slider crank. It is known that 3D printing plastic filament is significantly cheaper than stock aluminum or even steel. Therefore, the group decided to maximize the number of components machined using 3D printing to reduce the total cost of the mechanism.

With those considerations in mind, the group failed to consider the simplicity and manufacturability of the initial proposal in the design phase when first entering the laboratory. The shortcomings of the design were prevalent when approaching the Manufacturing Teaching Laboratory staff on methods to manufacture specific parts and being met with an answer that would prove that the task was nearly impossible. Therefore, it was good that there were several different proposals for each individual component of the slider crank, so if one of the components was deemed too complex, it could easily be exchanged for another design.

In the end, a combination of different concepts was selected to minimize complexity while keeping high amounts of creative design in all aspects of the mechanism. For each of the components mentioned above, the following design concepts were ultimately selected to be manufactured:

- **Linear Handle**
- **3D-Printed Slider**
- **Vertical Crank Wheel**
- **T-Beam Track**

These concepts were selected because they best satisfied the project specifications while minimizing complexity and material cost. It is important to acknowledge that this was not the first iteration of the design initially; the project was too ambitious to realistically manufacture given the week of time in the laboratory to manufacture. It is also important to note the group’s collective lack of experience of manufacturing which resulted in issues in determining what was and was not possible with the given tools provided.

4. Slider Crank Mechanism Fundamentals

A slider crank is a unique type of four bar mechanism which is designed to convert rotational motion into its linear equivalent. The mechanism is comprised of a crank, connecting rod, offset, and input angular velocity. All of these are incorporated into a function that will ultimately describe the overall displacement function denoted by x(t). The total distance traversed by the sider is known as the stroke length.

4.1 Mechanism Description

At its core, a slider crank works by converting angular motion into translational motion. To begin the process, a torque is applied to the crank axle which imposes an angular acceleration on the crank wheel. The torque is typically provided using a motor supplied with a voltage and current; however, in other circumstances like the case of paperweight, the torque may be supplied with a user applying a force at a radial distance away from the axis of rotation. Below, Figure 4.1 depicts a traditional slider crank model with labeled individual components that will be discussed.

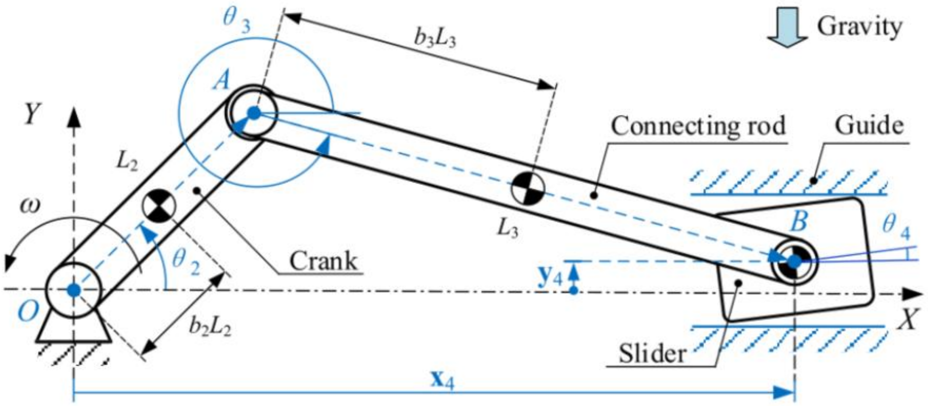

Figure 4.1: Slider-Crank Mechanism with labeled components [1].

Above, the crank mechanism is centered at the origin point labeled as *O* in the drawing. That point is given an angular velocity of ω. The crank has a radius of O*A* given as that is the connection point between the crank and the connecting rod.

As the crank rotates, there is a connected rod attached to a particular point, *A*. That rod is also connected to a slider confined to a linear guide that will be discussed later. The total length of this rod is *AB*. This rod can undergo general plane motion as a rigid body meaning that there is both a translational and rotational component to its kinematic movement. This rod is the fundamental mechanism behind the conversion of motion.

The third part of the mechanism is the slider. On its own, this slider just rotates around the pin at point *B* which ties the connecting rod and slider part together. This however would not be linear motion as the slider part would be able to rotate. Therefore, the linear guide must be used to restrict the motion of the slider. For the slider crank system to be fully defined, the slider must be limited to one degree of freedom (DOF). This means that the slider must not be able to rotate around any axis, nor be able to translate in any direction besides the x-direction.

As the crank rotates one full revolution, the linear motion that is produced is also one full reciprocating cycle. This means that the x-direction motion of the slider is a sinusoidal function like the motion of the crank method. This means that the maximum speed is achieved when the sider is at the center of the entire range of motion when taking the instantaneous rate of change of position with respect to time.

4.2 Identification of Linkages and Joints

A link is a rigid body that connects other bodies and transmits motion. For a standard slider crank mechanism, there are four defined links. Link 1 is defined as the ground. This link is the reference link and is immovable. This variable is defined as d. The second link is the crank which is the location at which rotational torque is applied. This link is subject to pure rotational motion and is defined as a. Link 3 otherwise known as b is the coupler link which connects the crank to the slider. While experiencing general plane motion this Linkages both translates and rotates. Finally, the slider is Linkage 4 otherwise known as c. This link is confined to translational motion in only one direction such that the mechanism has a single degree of freedom.

The other component of the mechanism is the joints between linkages. These allow for relative motion between consecutive links. All Joints are revolute joints besides the slider joint meaning that there is a single DOF for the relative motion. The motion is confined to rotation. The slider on the other hand is a prismatic joint which is confined to pure linear motion. This makes sense as linear motion is the desired output of the mechanism.

4.3 Mobility Calculation

The degree of freedom of the mechanism is verified using Gruebler's equation for planar mechanisms:

$$M = 3(n - 1) - 2j_1 - j_2 \tag{4.3.1}$$

Where:

- n = number of lines
- j_1 = number of lower pair joints
- j_2 = number of higher pair joints

For this slider-crank mechanism:

- n = 4 links
- j_1 = 4 lower pairs (3 revolute + 1 prismatic)
- j_2 =0

Substituting:

$$M = 3(4 - 1) - 2(4) - 0 \tag{4.3.2}$$

$$M = 1 \tag{4.3.3}$$

The mechanism therefore has one degree of freedom, meaning a single input (crank rotation) completely defines the motion of the entire system.

5. Analysis (Corrected Analysis Report)

The following section presents the governing kinematic equations for the slider-crank mechanism derived from geometric compatibility and relative velocity analysis. Numerical evaluation of these expressions at $\theta_2 = 50°$and $\omega_2 = 3\ \mathrm{rad/s}$ is documented in the Appendix.

The total stroke length of the slider is determined from the maximum and minimum geometric positions of the mechanism. These correspond to the extreme configurations of the crank and connecting rod.

$$\text{Stroke} = \sqrt{(b+a)^2 - c^2} - \sqrt{(b-a)^2 - c^2} \tag{5.1}$$

The coupler angle $\theta_3$is obtained from the vertical component compatibility between the crank and connecting rod. Using the geometric constraint of the linkage, the open configuration solution is selected.

$$\theta_3 = \sin^{-1}\left(\frac{-a\sin(\theta_2) + c}{b}\right) + 180° \tag{5.2}$$

The slider displacement is calculated using the horizontal component relationship between the crank and connecting rod. This expression directly relates the crank angle to the linear position of the slider.

$$d = a\cos(\theta_2) - b\cos(\theta_3) \tag{5.3}$$

The transmission angle is computed to evaluate the effectiveness of force transfer between the connecting rod and the slider. This angle measures the deviation from an ideal right-angle force transmission condition.

$$\mu = \left|90° - |\theta_3 - 180°|\right| \tag{5.4}$$

The angular velocity of the connecting rod is determined using relative velocity relationships between the crank and coupler. This is obtained by differentiating the geometric constraint with respect to time.

$$\omega_3 = \frac{a\omega_2}{b} \cdot \frac{\cos(\theta_2)}{\cos(\theta_3)} \tag{5.5}$$

The velocity of point A is determined using rigid body rotation about the crank center. The velocity components are expressed in Cartesian form before conversion to polar representation.

$$\overrightarrow{V_A} = -a\omega_2\sin(\theta_2) + j\,a\omega_2\cos(\theta_2) \tag{5.6}$$

The slider velocity is obtained from the horizontal component of the coupler velocity. Since the slider is constrained to move in the x-direction, only the horizontal velocity component is retained.

$$V_B = -a\omega_2 \sin(\theta_2) + b\omega_3 \sin(\theta_3) \tag{5.7}$$

The mechanical advantage is calculated as the ratio of output linear velocity to input angular velocity scaled by the input radius. This provides a measure of the mechanism's force amplification capability at the given crank position.

$$MA = \left|\frac{\omega_{in} r_{in}}{V_B}\right| \tag{5.8}$$

To better understand the kinematic behavior of the mechanism, the slider displacement d is plotted as a function of the input crank angle θ_2. This relationship illustrates how rotational motion at the crank is converted into translational motion of the slider over one full revolution. Below, Figure 5.1 depicts the variation of slider displacement with respect to the crank angle and highlights the nonlinear geometric relationship governing the mechanism.

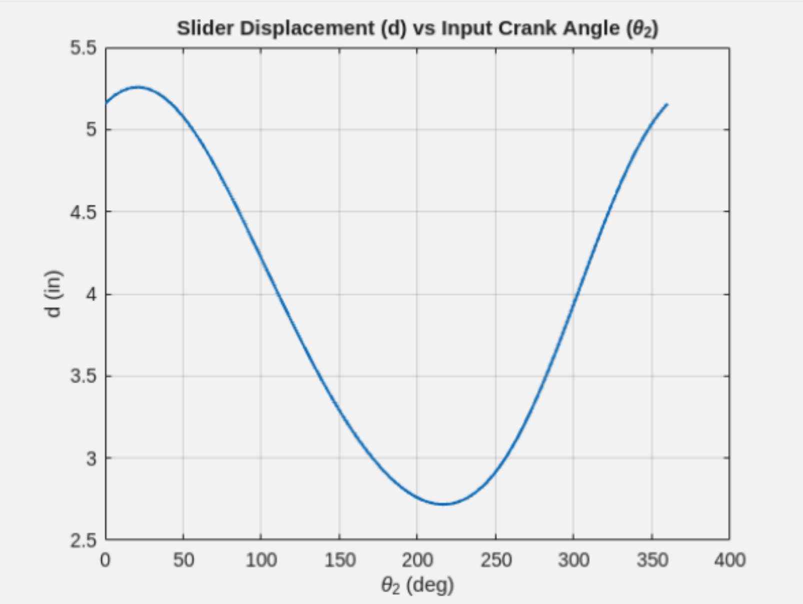

Figure 5.1: Slider Displacement d vs. Input Crank Angle θ_2.

As shown in Figure 5.1, the slider displacement varies smoothly and periodically with crank rotation. The curve exhibits a maximum displacement of approximately 5.2 in and a minimum displacement near 2.7 in, corresponding to the extreme positions of the slider. The non-sinusoidal shape of the curve reflects the geometric constraint imposed by the connecting rod, which causes the forward and return strokes to occur at slightly different rates. This confirms the nonlinear kinematic relationship between $\theta_2$and d.

The transmission angle μ is examined to evaluate force transfer characteristics throughout the cycle. Because the transmission angle directly influences mechanical efficiency, it is important to observe how it varies during crank rotation. Below, Figure 5.2 depicts the transmission angle as a function of the input crank angle and illustrates the regions of optimal and reduced force transmission.

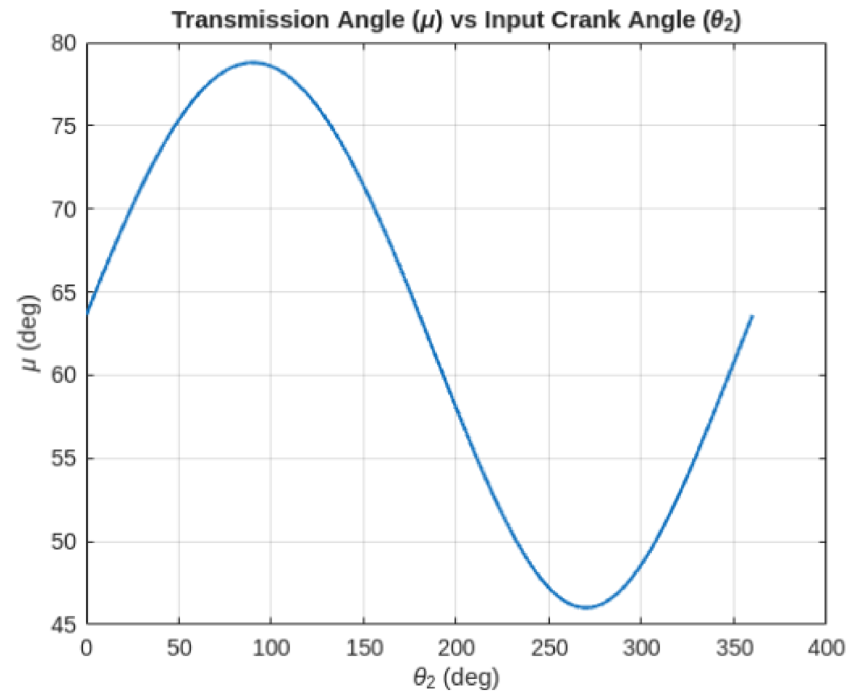

Figure 5.2: Transmission Angle μ vs. Input Crank Angle θ_2.

Figure 5.2 shows that the transmission angle varies between approximately 46° and 79° over one full revolution. The maximum transmission angle occurs near $\theta_2 \approx 100°$, while the minimum occurs near $\theta_2 \approx 270°$. Since transmission angles closer to 90° provide better force transfer, the mechanism operates most efficiently near its peak values and least efficiently near the minimum. However, because the transmission angle never becomes excessively small, the mechanism maintains acceptable force transmission throughout its motion.

The angular position of the connecting rod, represented by θ_3, varies continuously as the crank rotates. Understanding this relationship is essential for analyzing the internal motion of the mechanism. Below, Figure 5.3 depicts the variation of the coupler angle with respect to the input crank angle and shows the constrained rotational behavior of the connecting rod over one full revolution.

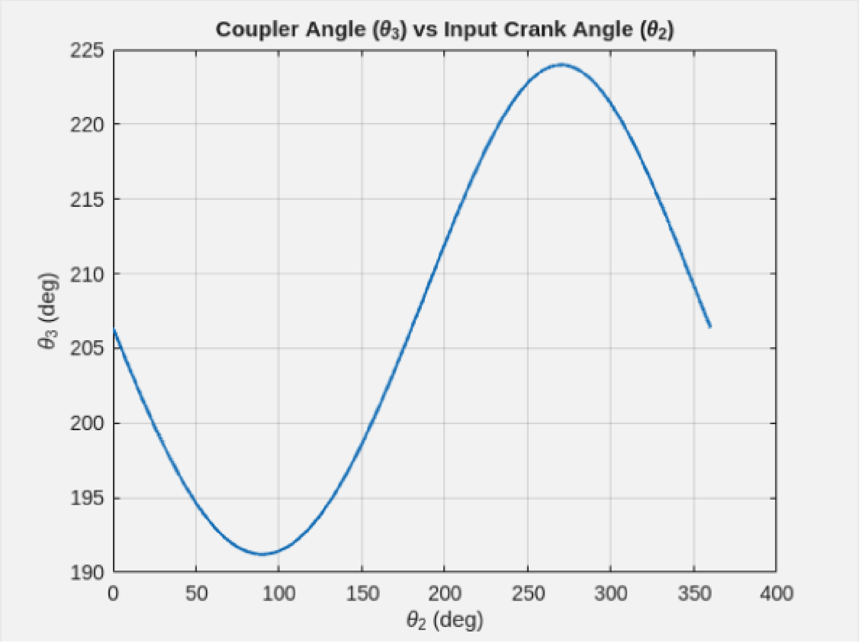

Figure 5.3: Coupler Angle θ_3 vs. Input Crank Angle θ_2.

As shown in Figure 5.3, the coupler angle varies smoothly between approximately 191° and 224° during one full revolution of the crank. The curve demonstrates that the connecting rod undergoes continuous angular motion, but its rate of change is nonlinear due to the offset geometry. The maximum and minimum values correspond to the crank positions where the slider reaches its extreme displacements. This relationship confirms the constrained rotational behavior of the connecting rod within the slider-crank mechanism.

6. Design Drawings with Dimensions

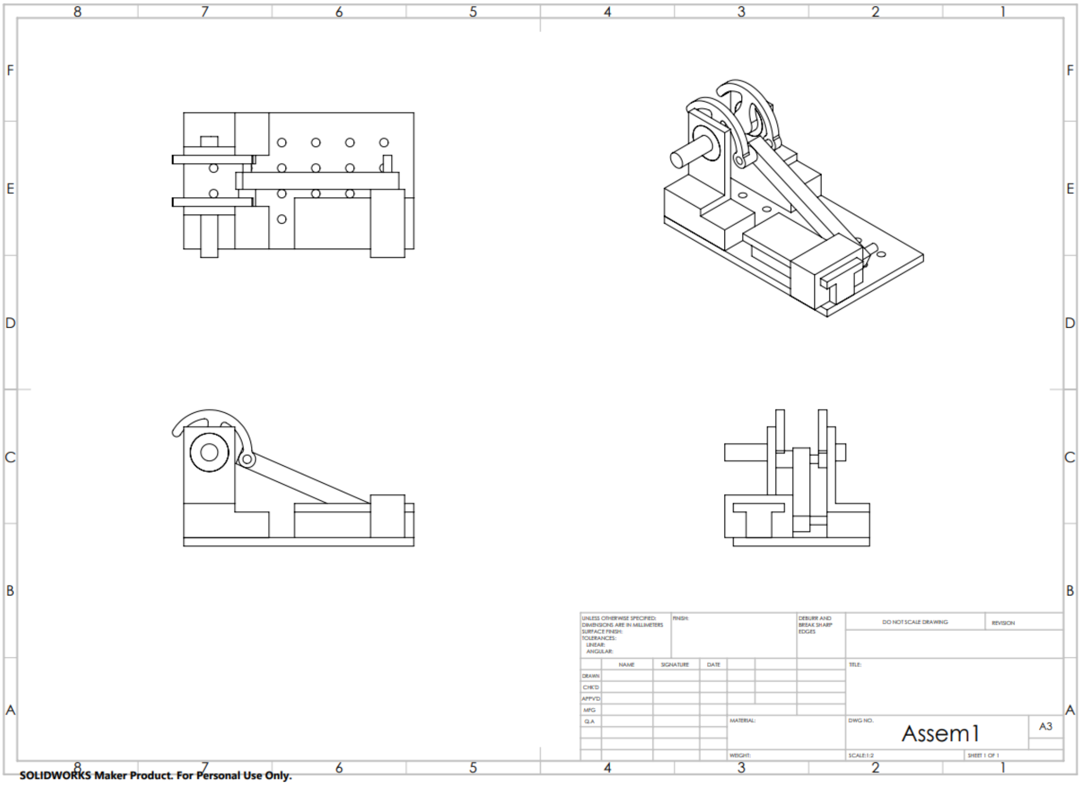

Figure 6.1: Assembly

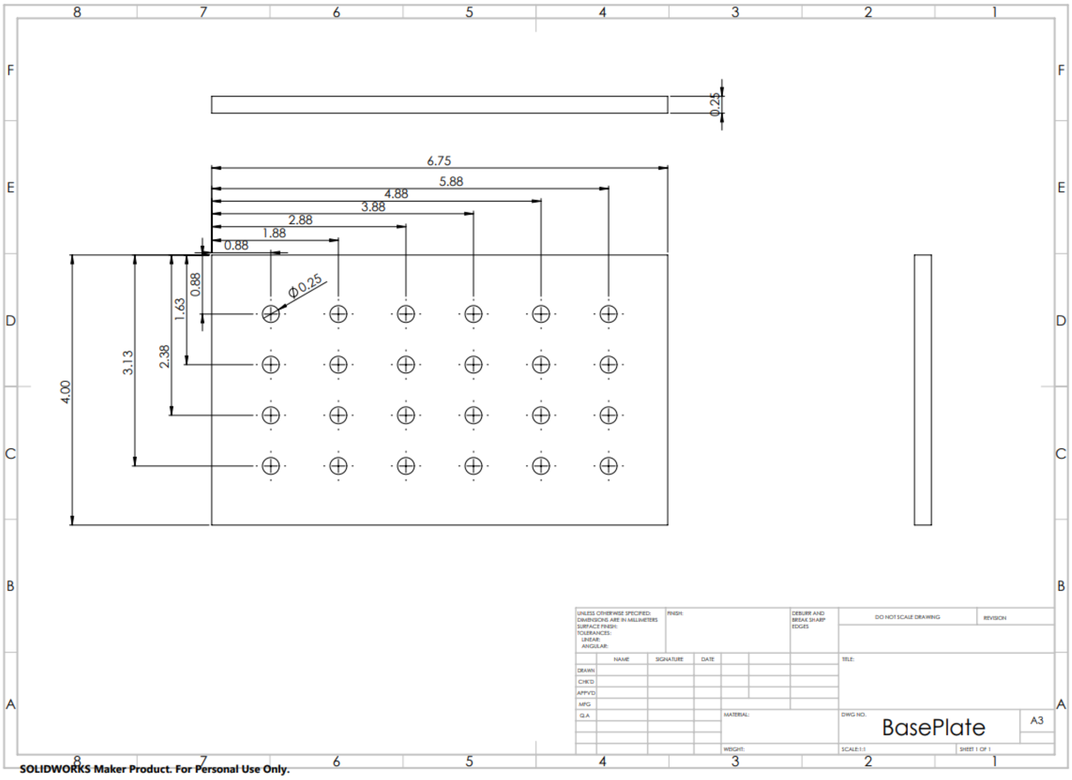

Figure 6.2: Base Plate

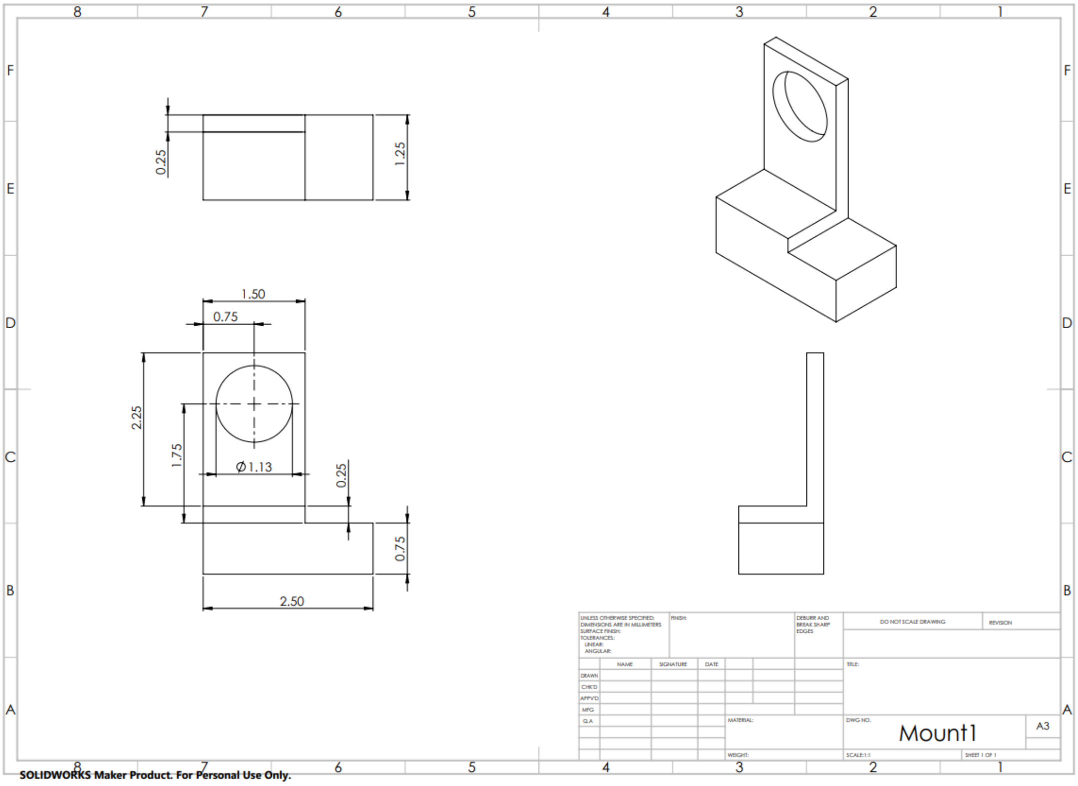

Figure 6.3: Mount 1

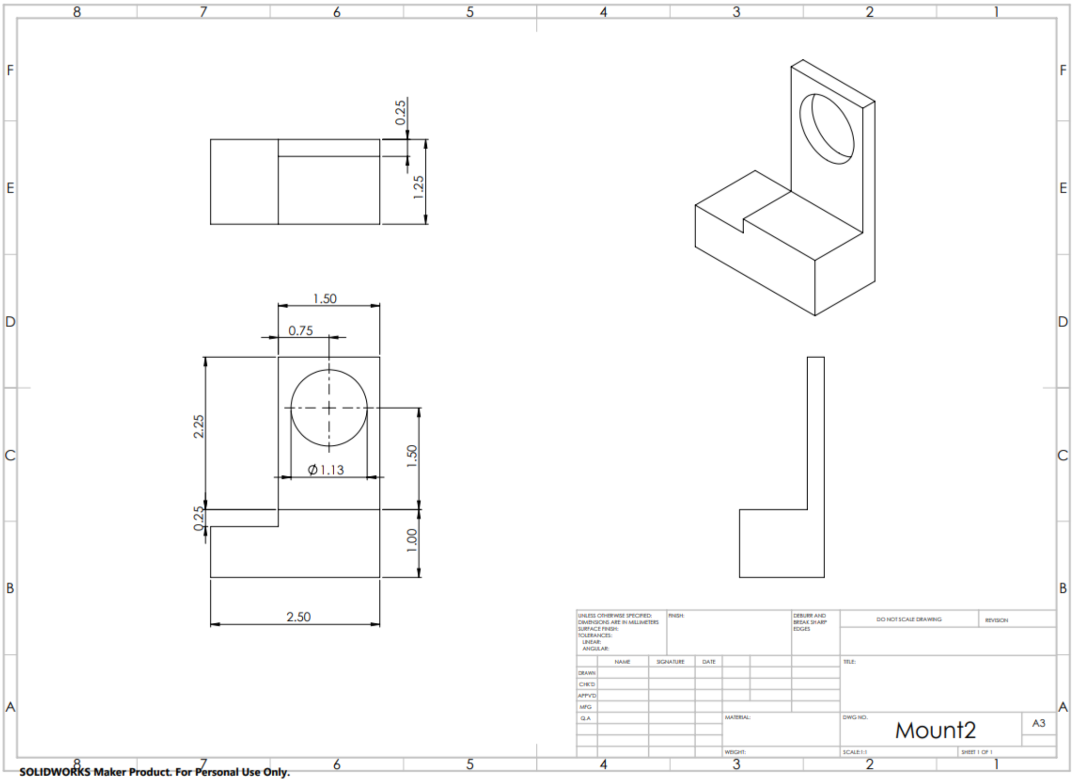

Figure 6.4: Mount 2

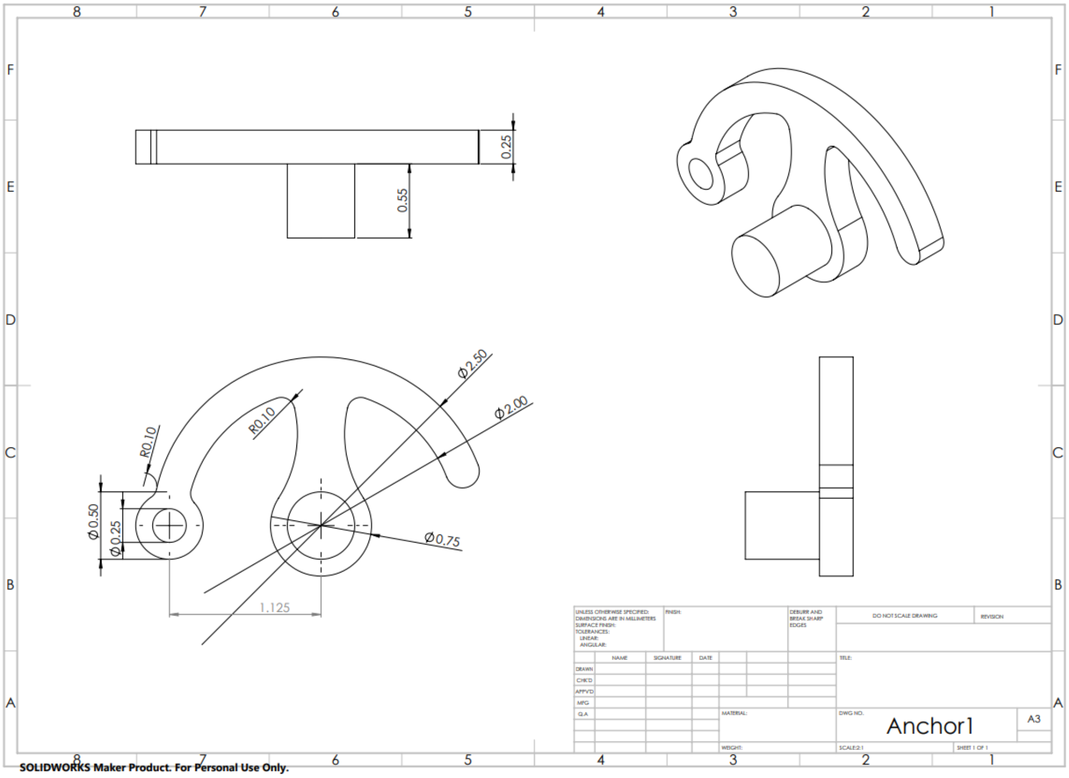

Figure 6.5: Anchor 1

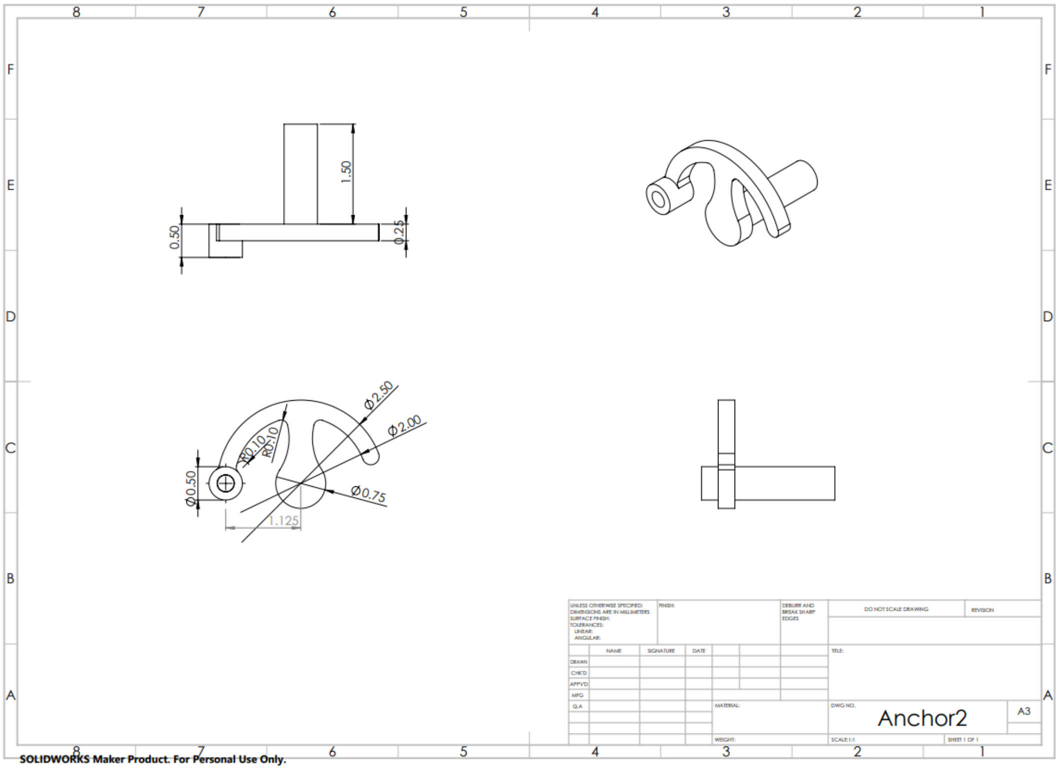

Figure 6.6: Anchor 2

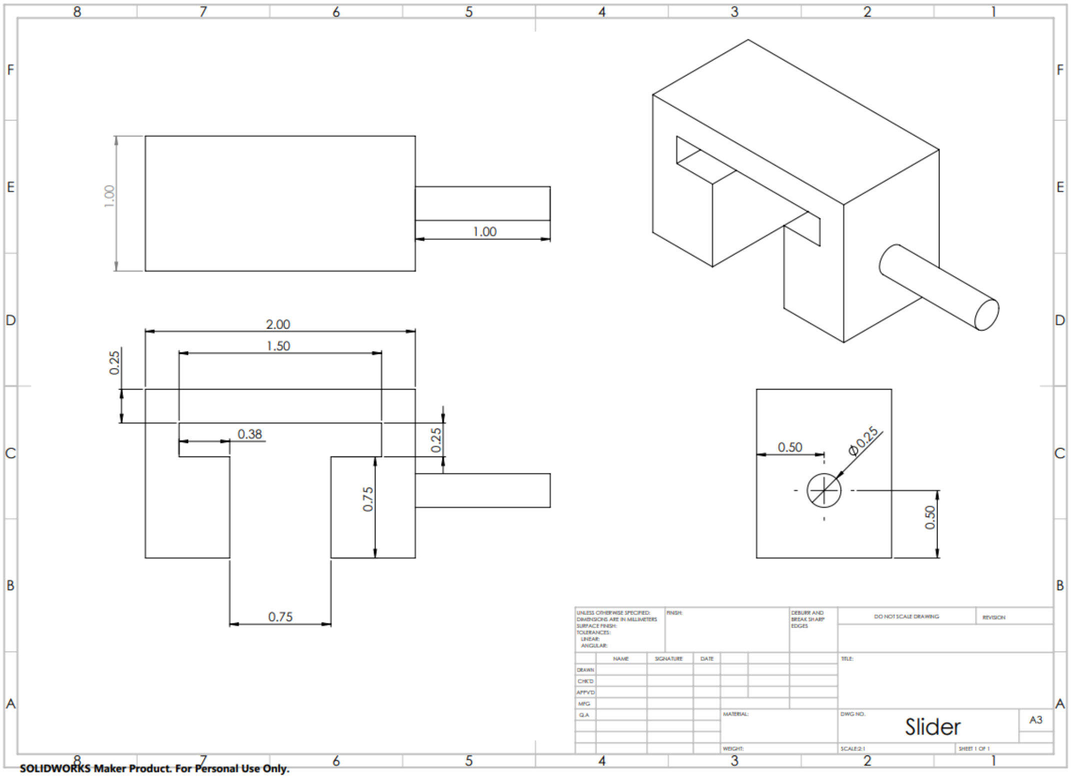

Figure 6.7: Slider

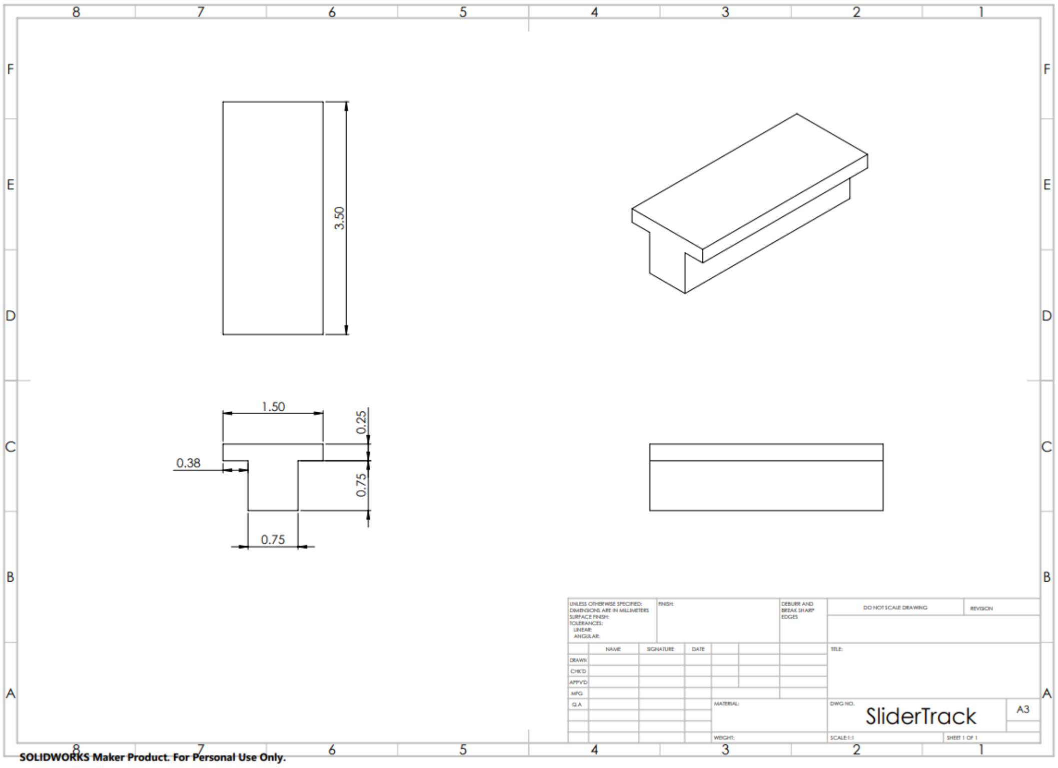

Figure 6.8: Slider Track

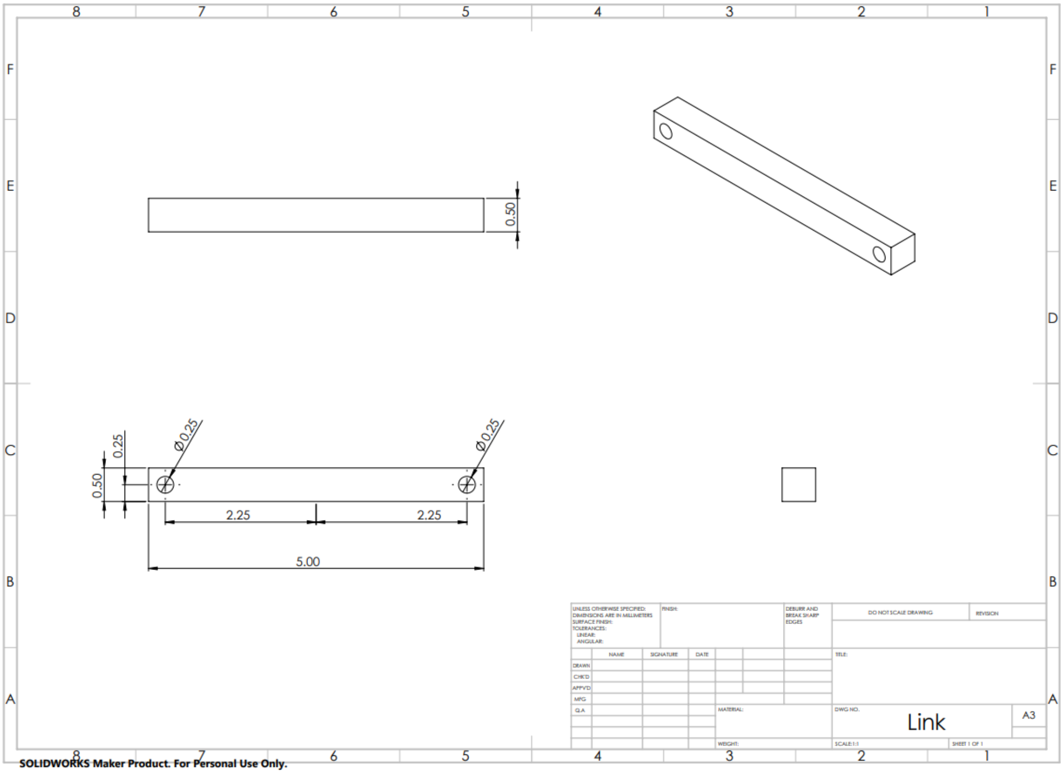

Figure 6.9: Link

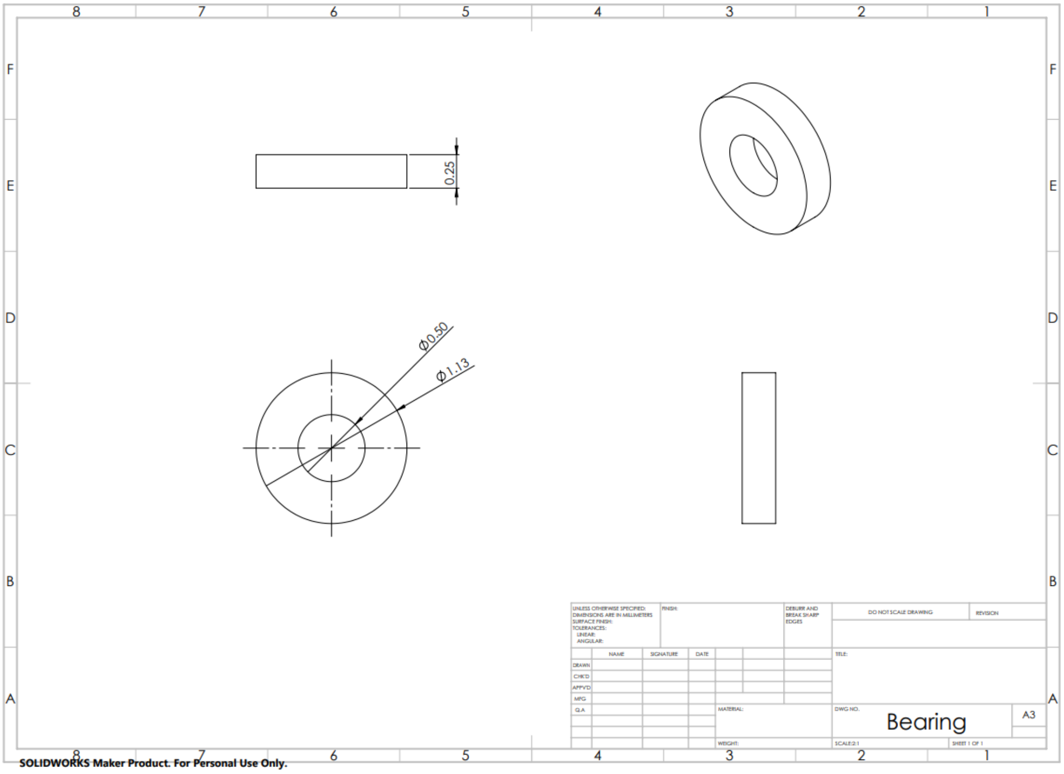

Figure 6.10: Bearing

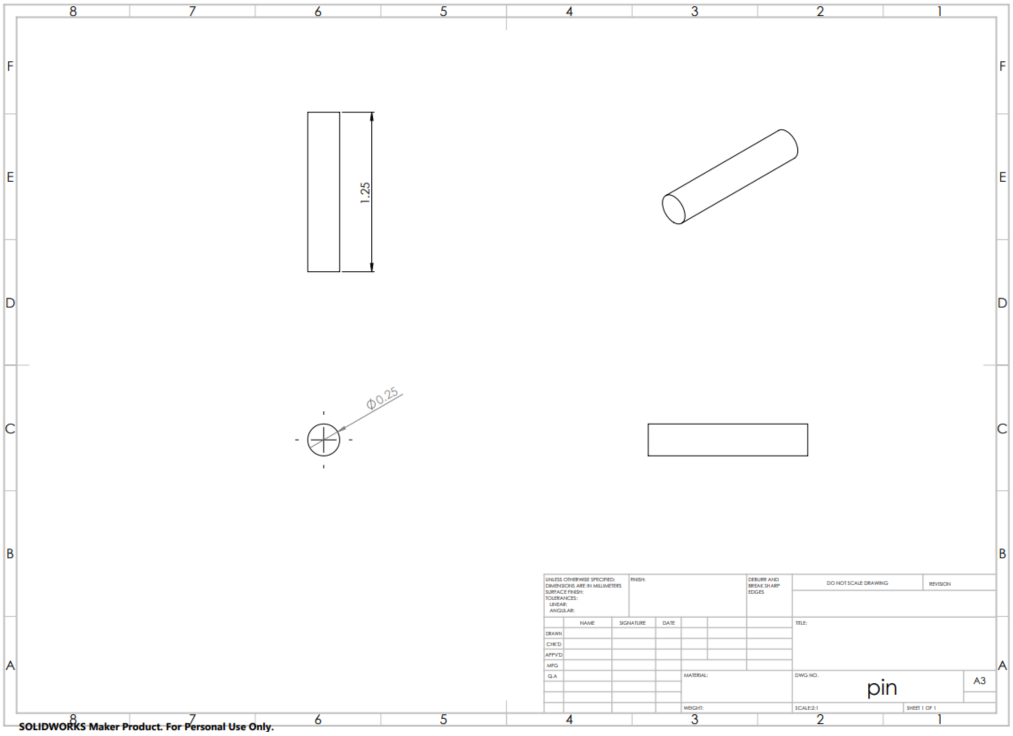

Figure 6.11: Pin

7. Problems, Solutions, Future Improvements

7.1 Problems Encountered

The initial slider retention design incorporated a trapezoidal geometry intended to provide positive vertical constraint and prevent separation from the baseplate during inverted operation. The trapezoidal profile offered strong mechanical capture and appeared to satisfy the functional requirements during conceptual design and CAD modeling.

During fabrication planning, it became evident that the trapezoidal geometry could not be produced using the available manufacturing equipment in the MTL. Machining the angled sidewalls required either a dovetail cutter or a precision tilting fixture setup. These tooling and fixturing options were not available within the constraints of the laboratory environment.

Producing the trapezoidal feature on a standard vertical mill would require multiple complex setups, increasing the likelihood of dimensional inaccuracy and misalignment between the slider and guide surfaces. Maintaining tight tolerances on angled faces without CNC capability would have introduced significant risk of binding or excessive friction.

Because accurate slider guidance is critical to smooth motion and reliable inverted operation, the trapezoidal design was determined to be impractical to manufacture with the available tools and time constraints.

7.2 Solutions

To address these manufacturability limitations, the trapezoidal slider profile was redesigned as a T-shaped configuration. The revised geometry preserved the functional requirement of positive vertical retention while allowing fabrication using standard vertical milling operations.

The T-slot configuration could be produced using straight end mills and orthogonal machining passes, eliminating the need for angled cutting tools or specialized fixturing. This reduced machining complexity and improved dimensional consistency.

The T-shaped design maintained:

- Positive vertical capture during inverted operation
- Adequate lateral guidance
- Reduced risk of binding
- Compatibility with available milling equipment

By modifying the geometry to align with manufacturing capabilities, the final design satisfied both functional and fabrication requirements. This redesign process emphasizes the importance of incorporating manufacturing constraints into early-stage mechanical design decisions.

7.3 Future Improvements

Although the final design satisfied all functional and manufacturing requirements, several improvements could enhance performance, durability, and manufacturability.

First, increasing the input radius slightly would reduce the required input torque and improve ergonomic operation.

Second, tighter dimensional tolerances on the slider-track interface could reduce backlash and improve motion smoothness. Implementing controlled clearance values or surface finishing operations would enhance repeatability.

Third, replacing the printed slider with Delrin or low-friction polymer material could reduce sliding resistance and improve durability under repeated operation.

Finally, incorporating aesthetic refinements such as filleted edges, surface finishing, or anodizing would improve the visual quality of the mechanism while maintaining functionality.

These improvements demonstrate how future iterations could enhance mechanical performance while preserving manufacturability.

References

[1] Wu, Xuze & Sun, Yu & Wang, Yu & Chen, Yu. (2021). Correlation dimension and bifurcation analysis for the planar slider-crank mechanism with multiple clearance joints. Multibody System Dynamics. 52. 10.1007/s11044-020-09769-3.

Appendix A – Hand Calculations

$$2.5288\ \text{in} = \sqrt{(4.5 + 1.125)^2 - 2^2} - \sqrt{(4.5 - 1.125)^2 - 2^2} \tag{A.1}$$

$$194.65^\circ = \sin^{-1}\left(\frac{-1.125\sin(50^\circ) + 2}{4.5}\right) + 180^\circ \tag{A.2}$$

$$5.077\ \text{in} = 1.125\cos(50^\circ) - 4.5\cos(194.65^\circ) \tag{A.3}$$

$$75.35^\circ = \left|90^\circ - |194.65^\circ - 180^\circ|\right| \tag{A.4}$$

$$-0.498\ \text{rad/s} = \frac{1.125(3)}{4.5} \cdot \frac{\cos(50^\circ)}{\cos(194.65^\circ)} \tag{A.5}$$

$$(-2.59\, i + 2.17\, j)\ \text{in/s} = 3.375\ \text{in/s}\ \angle 140^\circ\ (6) = -1.125(3)\sin(50^\circ) + j\, 1.125(3)\cos(50^\circ)$$

$$-2.02\ \text{in/s} = -1.125(3)\sin(50^\circ) + 4.5(-0.498)\sin(194.65^\circ) \tag{A.7}$$

$$0.371 = \left|\frac{3(.25)}{-2.02}\right| \tag{A.8}$$

Appendix B – MATLAB Code

Position Analysis:

```
function out = slidercrank_position_analysis(a, b, c, theta2_deg, branch)
    if nargin < 5 || isempty(branch)
        branch = "open";
    end
    branch = string(lower(branch));

    % --- Stroke limits ---
    dmax   = sqrt((b+a)^2-c^2)
    dmin   = sqrt((b-a)^2-c^2)
    stroke = dmax-dmin

    % --- Solve theta3 ---
    % Preallocate for safety
    theta3 = nan(size(theta2_deg));
    theta3 = asind((-a.*sind(theta2_deg)+c)./b);

    % Branch handling (given)
    if branch == "open"
        theta3 = theta3 + 180;
    elseif branch == "crossed"
        theta3 = 360 - theta3;
    else
        error('branch must be "open" or "crossed".');
    end
```

```
% Valid where theta3 is real and finite (given)
valid = isreal(theta3) & isfinite(theta3);

% Mask invalid entries (given)
theta3(~valid) = NaN;
theta3 = real(theta3);

% --- Slider position dpos ---
dpos = nan(size(theta2_deg));
dpos(valid) = a.*cosd(theta2_deg(valid)) + sqrt(b.^2 - (a.*sind(theta2_deg)-c).^2);

% --- Transmission angle mu ---
theta4_deg = 90*ones(size(theta2_deg));
mu = nan(size(theta2_deg));
mu(valid) = abs(theta4_deg(valid)-abs(theta3(valid)-180));

% Package outputs (given)
out.theta2_deg = theta2_deg;
out.theta3_deg = mod(theta3, 360);
out.d          = dpos;
out.mu_deg     = mu;
out.valid      = valid;

out.stroke.dmax   = dmax;
out.stroke.dmin   = dmin;
out.stroke.stroke = stroke;
```

```
end
%% Slider-Crank Plots (match Position Analysis PDF)
clc; close all;

% Link lengths
a = 1.125;   % inches
b = 4.5;   % inches
c = 2;   % inches (offset)
theta2_deg = 0:360;

% Call provided p-coded function
out = slidercrank_position_analysis(a,b,c,theta2_deg,"open");

% Extract and mask invalid configurations
t2 = out.theta2_deg;
t3 = out.theta3_deg;  t3(~out.valid) = NaN;
d  = out.d;           d(~out.valid)  = NaN;
mu = out.mu_deg;      mu(~out.valid) = NaN;

theta4_deg = 90;
t4 = 90*ones(size(theta2_deg))
% Display stroke values (optional)
fprintf("dmax = %.4f in, dmin = %.4f in, stroke = %.4f in\n", ...
    out.stroke.dmax, out.stroke.dmin, out.stroke.stroke);

% ---- Plots  ----
figure;
```

```
plot(t2, d)
xlabel("Theta 2 (degrees)")
title("d, inches")

figure;
plot(t2, t3)
xlabel("Theta 2 (degrees)")
title("Theta 3 (degrees)")

figure;
plot(t2, mu)
xlabel("Theta 2 (degrees)")
title("Transmission Angle (degrees)")
```