TRIBHUVAN UNIVERSITY
INSTITUTE OF ENGINEERING
PULCHOWK CAMPUS

DESIGN AND FABRICATION OF PAD PRINTING MACHINE CAPABLE OF PRINTING IN UNEVEN SURFACE

SUBMITTED BY
BIBEK BHATTARAI (PUL077BME008)
BIKRAM BHANDARI (PUL077BME010)
AAKASH KHATRI (PUL077BME049)
SOHAN CHHETRI (PUL077BME052)

A PROJECT REPORT
SUBMITTED TO
DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING
IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE DEGREE OF
BACHELOR IN MECHANICAL ENGINEERING

DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING
LALITPUR, NEPAL

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DEPARTMENT OF MECHANICAL AND AEROSPACE ENGINEERING

The undersigned certify that they have read, and recommended to the Institute of Engineering for acceptance, a project report entitled **"DESIGN AND FABRICATION OF PAD PRINTING MACHINE CAPABLE OF PRINTING IN UNEVEN SURFACE"** submitted by Bibek Bhattarai, Bikram Bhandari, Aakash Khatri and Sohan Chhetri in partial fulfillment of the requirement for the degree of Bachelor of Mechanical Engineering.

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Date: 2025/05/23

ABSTRACT

Mass printing on uneven surfaces is a big challenge for small businesses due to the high cost of existing printing technologies. Main focus of this project is designing and fabricating machine capable of printing high-quality designs on irregular surfaces. The main objective is to provide small businesses a cost-effective solution for printing on uneven surfaces. Optimum design was created including mechanisms such as the double cams mechanism and the off-set cam mechanism for smooth and precise vertical and horizontal movements respectively. These designs have been analyzed using SolidWorks simulation and animation to evaluate their functionality and ensure proper synchronization of components. The key aspects is to use locally available materials to balance cost, durability, and performance requirements without compromising the functionality of the machine. The project emphasises practical engineering skills designing, fabrication (uses of different machines like lathe, drill, CNC, Welding) and mechanical assembly, followed by testing to ensure all the objectives are met, while also considering real-world needs for required automation in low-resources environment. This outcomes offers a foundation for future enhancements for full automation, and consider the potential for scalable innovation in basic printing technologies.

ACKNOWLEDGEMENT

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ABBREVIATIONS

3D	3 Dimensional
AS	Automation Studio
CAD	Computer Aided Design
CAE	Computer Aided Engineering
CAM	Computer Aided Manufacturing
CATIA	Computer Aided Three-Dimensional Interactive Application
PP	Pad Printing

CHAPTER 1: INTRODUCTION

1.1 Background

1.1.1 Pad Printing Machine

Pad printing (PP) is an indirect printing process having one ink transferring part. PP machines come in two types: pad transfer printing and rotary pad transfer printing. PP has some advantages over other printing methods. Because it is a gilt-edged technique for printing on non-smooth objects having concave and convex surfaces, it has a competitive advantage for work with 3-dimensional substrates, which have differing shapes, thicknesses and dimensions[1]. PP machines are important tools used in industrial printing, allowing ink to be transferred precisely onto different surfaces. They've changed how things are made and customized in many industries, like cars, electronics, and medical devices. These machines are flexible and can print on tricky surfaces, making them very useful. They help businesses make their products look good and stand out by adding logos, designs, or text. This makes customers notice and remember the brand. Plus, the prints they make last a long time, which is great for brand recognition. With more people wanting personalized products, the demand for PP machines is increasing, especially among small businesses. These businesses can now compete better and meet their customers' needs by using this technology. PP works by transferring ink from a printing plate to a silicone pad, which then stamps the ink onto the desired surface. This process allows printing on irregular shapes and surfaces that are hard to print on using other methods. The ink sticks to the silicone pad and is then pressed onto the product, leaving a clear and precise print. With more people wanting personalized products, the demand for pad PP machines is increasing, especially among small businesses. These businesses can now compete better and meet their customers' needs by using this technology.

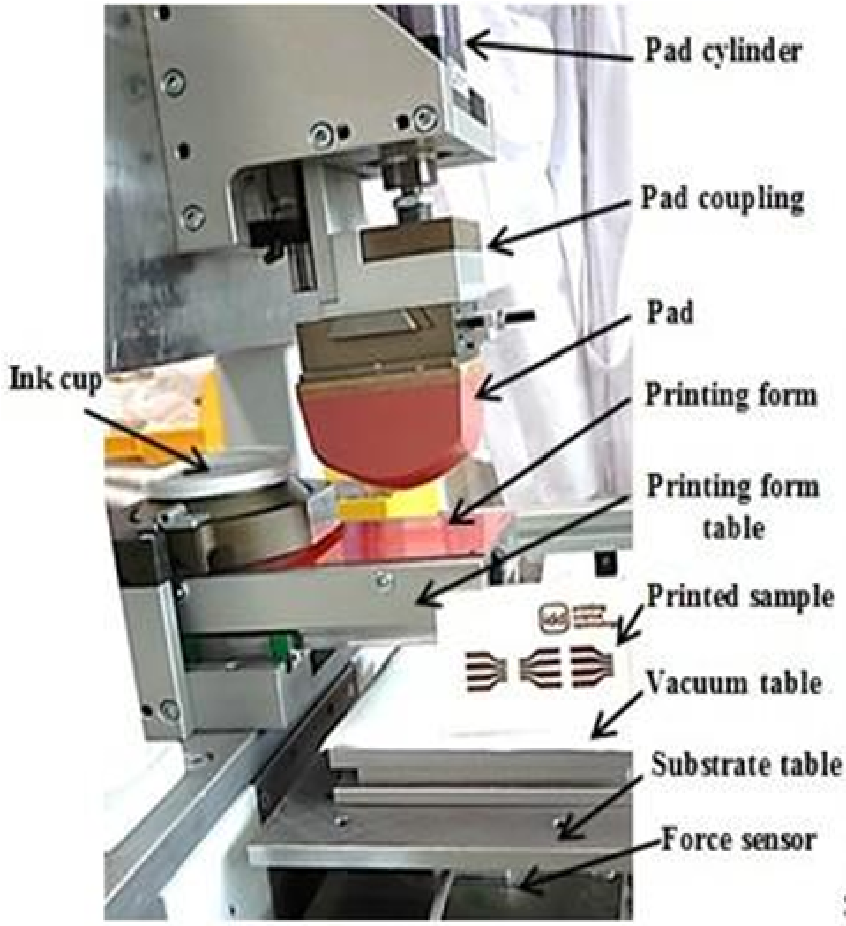

Figure 1.1: Components of PP Machine[1]

Working: The mechanism of PP machines involves several key components working together to transfer ink onto a variety of surfaces with precision and consistency. Here's an overview of the main components and how they function:

- **Printing Plate:** The printing plate, also known as the cliché, is etched with the desired design or image using a chemical or laser process. It acts as the carrier of the ink and determines the final print.

- **Ink Cup:** The ink cup holds the ink and covers the printing plate. It is typically made of steel and has a ceramic or polymer ring to create a seal. The ink cup prevents the ink from drying out and protects it from external contaminants.

- **Doctor Blade**: The doctor blade is a flexible strip made of steel or plastic that removes excess ink from the surface of the printing plate as it passes over it. This ensures that only the etched areas of the plate are filled with ink.

- **Silicone Pad:** The silicone pad, also known as the transfer pad or tampon, is a soft, flexible pad made of silicone rubber. It picks up the ink from the printing plate by pressing against it and then transfers the ink onto the substrate.

- **Substrate:** The substrate is the surface onto which the ink is transferred. It can be made of various materials, including plastics, metals, glass, ceramics, and more.

- **Printing Process:** The pad printing process begins with the ink cup covering the etched printing plate. The doctor blade removes excess ink, leaving ink only in the etched areas. The silicone pad then presses against the plate, picking up the ink. As the pad lifts away from the plate, it transfers the ink onto the substrate, creating the final print.

- **Curing:** Depending on the type of ink used, curing may be required to dry and set the printed image. This can be achieved through air-drying, UV curing, or heat curing methods.

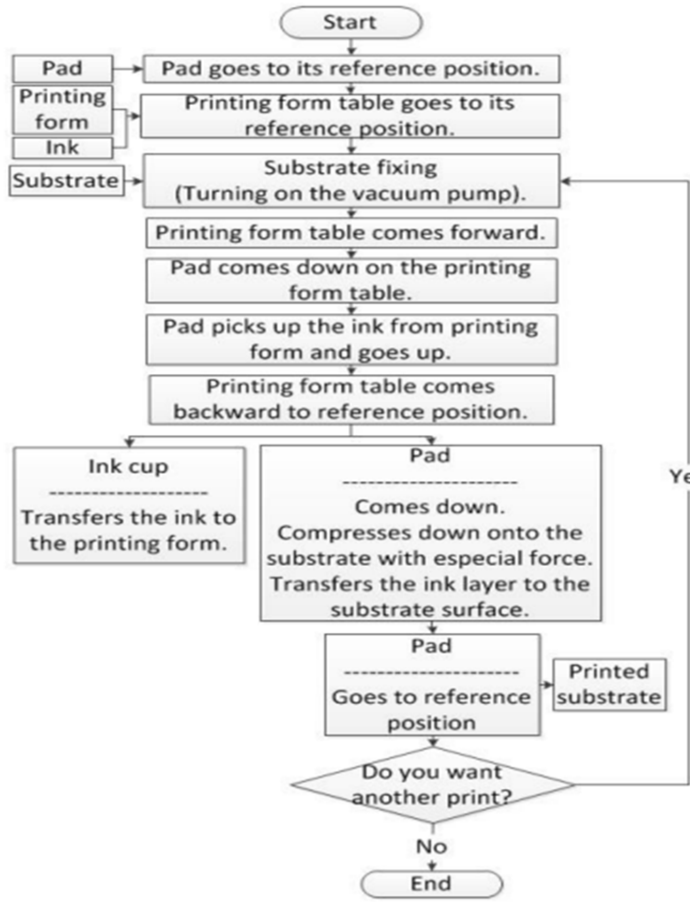

Figure 1.2: Working of PP Machine

1.1.2 Pad Printing(PP) in small business

PP plays a crucial role for small businesses across industries due to its versatility, cost-effectiveness, and ability to produce high-quality prints on various surfaces. This method enables small businesses to provide customized solutions for different materials, from plastics to metals, meeting diverse customer needs and market demands. Its low setup costs and minimal material wastage make it appealing for businesses with limited budgets or operating on a smaller scale. Small businesses utilize pad printing to offer personalized printing solutions, boosting brand visibility and customer engagement. Despite its affordability, pad printing yields durable and precise prints, enduring wear, fading, and environmental factors. Its adaptability to irregular surfaces allows small businesses to expand their offerings and market reach. By integrating pad printing into branding and marketing strategies, small businesses can produce professional printed materials, enhancing competitiveness and leaving a lasting impression on customers. Overall, pad printing is a valuable asset for small businesses, helping to meet customer demands, and increasing brand exposure.

1.2 Problem Statement

Small business faces many difficulties and barriers caused by the existing expensive and complicated printing machines. The machines are uncompetitive for small businesses for two reasons: the materials and components for the machines are often expensive and difficult to get, and most current printing methods, like screen printing, roller printing, are inconsistent while printing on irregularly shaped objects. As a result, the production time will be increased, the labour cost will rise, and the print quality will be variable and limiting, affecting the efficiency and expandability.

Most of the existing pad printing machines are complex and are equipped with several programmable motors to automate and ensure the precision. Although the motors are beneficial and additive to the functions of the machines, they will raise the total cost of the machines while purchasing and during maintenance. Moreover, the complicated structure makes the machines hard to get used to and maintain, which is not suitable for the low-skilled labour often seen in small businesses. This also raises the labor cost of the operation and limits the accessibility of the state-of-the-art pad printing machines.

The issue can be divided into two facts: first, a cheap pad printing machine, which applies materials and components that are easily available, but without compromising on the quality and durability. This type of machine will make the pad printing accessible for small businesses and compete in the market effectively. Second, an automated solution to raise the efficiency and consistency while printing on uneven objects, and reduce the labour and time needed when the objects are placed manually. Also, the fact that how many programmable motors are applied in the machine should be minimized to lower cost and maintenance, and ensure that small businesses and low-skilled labourers can get used to and operate the machine.

Addressing these issues is vital for cost savings, improved efficiency, and consistent print quality for small businesses. This project aims to design and develop a pad printing machine that is affordable, automated, and easy to maintain, making advanced pad printing technology accessible and boosting the competitiveness of small businesses in the market.

1.3 Objectives

1.3.1 Main Objective

Design and develop an affordable and user-friendly pad printing machine that is capable of printing on uneven surfaces.

1.3.2 Specific Objectives

- Integrate a single non-programmable motor to synchronize vertical and horizontal movement (X and Y axis) to produce required pattern of motion.

- Design and fabricate cost-effective machine using inexpensive materials and components without using programmable motors and IC chips.

- Learn to design and use various mechanisms like slider crank, four-bar mechanism, cam mechanism, etc.

- Develop cost-effective alternatives to complex pad printing models and make them available to small-scale industries.

1.4 Scope

1. This Pad printing machine is capable of printing patterns, logos, and designs on uneven surfaces.

2. The machine was developed by focusing on eliminating the use of expensive and complex electronics chips, thus mechanism was synchronized using single non programmable motor and locally available materials. Due to this reason, it can be manufactured in simple workshops.

3. The machine has synchronized multiple mechanisms, including a cam mechanism, to achieve simultaneous x and y motion for precise ink transfer.

4. The machine is capable of printing on a variety of materials, including plastics, metals, and glass, which makes it suitable for applications in industries.

1.5 System Requirements

1.5.1 Hardware Requirements

1.Computer System:

Processor: Intel i5 or equivalent (minimum).

RAM: 8 GB or higher.

Storage: 500 GB HDD or 256 GB SSD (for software and data storage).

Graphics Card: Dedicated GPU (e.g., NVIDIA GTX 1050 or higher) for running SolidWorks smoothly.

2. Fabrication Tools:

CNC machine or manual machining tools (lathe, milling machine, etc.) for fabricating mechanical components.

Measuring tools (calipers, micrometers) for precision assembly.

1.5.2 Software Requirements

MATLAB: Used for mathematical calculations, simulations, and analyzing motion mechanisms.

SolidWorks 2022: Used for 3D modeling, design, and simulation of the pad printing machine components.

Microsoft Excel: Used for data organization, calculations, and creating graphs or charts for analysis.
Google Chrome: Used for research, accessing online resources, and downloading necessary software or documentation.
Additional Tools: Word processing software (e.g., Microsoft Word, Latex) for report writing.

CHAPTER 2: LITERATURE REVIEW

2.1 Pad Printing Machine

A study conducted on Design and simulation of automated pad printing machine using automation studio, CATIA 3D modelling was integrated with Automation Studio (AS). CATIA designs were imported into AS using .igs files which enable visualization and dynamic simulation. During simulation, pneumatic cylinder positions were linked to the virtual model. Sliding between cylinder bores and pistons improved the movement representation. This detail gave engineers an advantage to optimize designs before physical fabrication, which can save time and resources. The pad printing machine's control system offers manual and automatic modes. Transition to auto mode from manual requires the machine to be in its natural position. In auto mod, a program executes a cycle of eight phases. This increases efficiency and accuracy in designing and simulating. This also refined controlling which benefitted manufacturers significantly.[2]

PP, a widely-used method, uses silicon pads to move ink from a design plate to a surface where the pattern is to be printed. The pad's angle and surface quality play a vital role in the quality of good quality prints as it affects the efficiency and precision of ink transfers. Ink transfer systems are categorized as closed or open systems. Closed systems are effective as they reduce solvent evaporation. The ink cup holds and transfers ink to the design plate. The surface receiving the print sits on a table. The design plate table moves using an electric motor and linear axis, and the plate moves vertically simultaneously. The reference position for the pad's movement is responsible for variations in pad height.[3]

2.2 Cam-Follower Mechanism

The cam and follower mechanism has been studied extensively because of its essential role in mechanical systems. A cam is a rotating component that drives a follower, creating a specific motion-either reciprocating or oscillating[4]. The way these two parts interact, through line contact, places them in a category known as higher pair mechanisms. The follower's motion is directly influenced by the shape and design of the cam, which has been a central focus for researchers looking to improve its performance for different uses[5].

Cams are incredibly versatile and have found applications in many industries. For example, they are critical in internal combustion engines, where they control the precise timing of inlet and exhaust valve operations[4]. Researchers have also explored their role in automatic machinery, such as paper-cutting machines and lathe feed mechanisms, where speed and accuracy are key. In textile machinery, advancements in cam design have improved the efficiency of processes like spinning and weaving. Another area of interest in camera research is improving their performance and durability. Since cams usually rotate at a steady speed, wear and tear are significant challenges, especially in high-speed systems[4]. Studies have shown that materials and surface finishes can make a big difference in reducing wear and extending their lifespan. Additionally, modern research is focusing on innovative cam profiles and follower designs to create smoother motions and minimize vibrations.

Overall from [4] and [5], cams and followers are continuously being refined, with ongoing research aimed at solving challenges like wear resistance, better materials, and new designs to meet the needs of various industries.

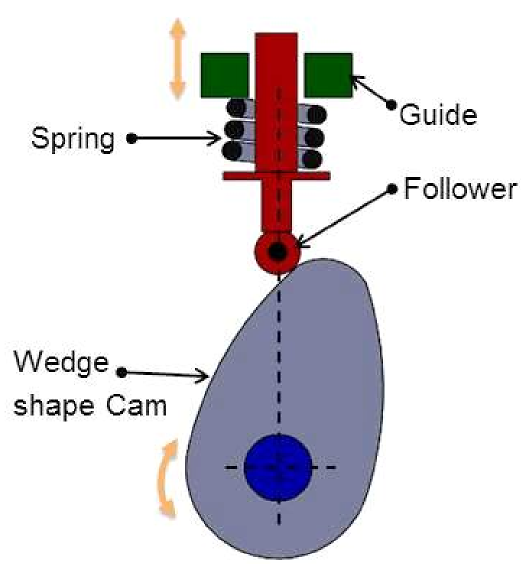

Figure 2.1: Cam and Follower Mechanism[6]

2.2.1 Cam Profile Design

The cam profile determines the motion of the follower. The cam profile determines the motion of the follower. The dwell angle and cam rotation speed play a critical role in the cam profile design for the transmission, which can be determined using a formula derived from [5] and [7].

- Dwell Angle:

$$\theta_{\text{dwell}} = \left(\frac{T_{\text{dwell}}}{T}\right) \times 360 \tag{2.1}$$

  where,

  θ_{dwell}: dwell angle

  T_{dwell}: Time duration of dwell

  T: Total Time Cycle

- Cam Rotation Speed (RPM):

$$N = \frac{60}{T} \tag{2.2}$$

  where,

  T: Period in seconds

2.2.2 Follower Motion Analysis

From book [5], we learnt how to calculate the displacement, velocity, and acceleration for SHM. The cam profile determines the motion of the follower. Key formulas and references include :

- Displacement (Simple Harmonic Motion):

$$S = \frac{h}{2}\left(1 - \cos\left(\frac{\pi\theta}{\theta_{\text{rise}}}\right)\right) \tag{2.3}$$

- Velocity:

$$V = \frac{\pi h}{2\theta_{\text{rise}}} \sin\left(\frac{\pi\theta}{\theta_{\text{rise}}}\right) \tag{2.4}$$

- Acceleration:

$$A = \frac{\pi^2 h}{2\theta_{\text{rise}}^2} \cos\left(\frac{\pi\theta}{\theta_{\text{rise}}}\right) \tag{2.5}$$

  where;

  S: Displacement of follower at any given cam angle

  h: Total lift (or Stroke) of the follower

  θ: Current angular position of the cam

  θ_{rise}: Angle over which follower rises

2.2.3 Force and Torque Analysis

The force analysis is done to calculate the force required to press the material for the printing, which can be calculated using the formula given in [7]. The force and torque required to drive the cam depend on the follower motion and cam profile. Key formulas and references include:

The force which is exerted by the cam to the follower, the torque in the camshaft, and the power obtained from the motor can be calculated by using the formula mentioned in the book [8].

- **Force on Follower:**

$$F = P \cdot A \tag{2.6}$$

  where;

  F: Force exerted by the cam

  P: Pressure applied to the material

  A: Contact area between the follower and the material

- **Torque on Cam Shaft:**

$$\tau = F \cdot r \cdot \sin(\theta) \tag{2.7}$$

where;

τ : Torque on the camshaft

F: Force exerted by the cam on the follower

r: Radius of the cam

θ : Angle at which Force is applied

- **Motor Power:**

$$P = \tau \cdot \omega \tag{2.8}$$

where;

P: Power required from the motor

τ: Torque on the camshaft

ω: Angular velocity of the camshaft

2.2.4 Mathematical Modeling and Simulation

Mathematical modeling and simulation are essential for optimizing cam follower mechanisms.

- **Angular Velocity:** The angular velocity is the key parameter in cam and camshaft during printing because it determines the speed and precision of cam directing the follower. According to Norton [7], the fluctuation in angular velocity affects the dynamic response of printing pads, resulting in different print quality. The angular velocity is mainly dependent upon the speed of rotation of the camshaft. By following [9], we can also analyse the motion of the cam by altering the angular velocity of the camshaft.

$$\omega = \frac{2\pi N}{60} \tag{2.9}$$

where,

ω: Angular velocity in rad/sec

N: Speed in rpm

- **Simulation Tools:** For the design and simulation of Cam, the use of SOLIDWORKS was carried out. Here, we have designed and meshed it with the follower connected to the pad and the overall rotation of the cam with connection with the camshaft was observed.

2.3 Offset-cam Mechanism

Cams are different in configurations, among which off-set cams are the ones whose camshaft axis is intentionally misaligned with the follower's line of action in order to generate a different pattern or a different kinematic behavior. During an extensive research, it was found that offset configuration gives better follower displacement control along with the reduction of pressure angle [10]. Also, calculated offset in a cam can also generate various shape of follower profile without changing the cam size. A research showed how dynamic contact forces vary with the amount of offset resulting into wear prediction [11]. Thus, it is used in applications which requires slightly different linear patterns, precise control, reduced side thrust, smoother motion profiles and also specific mechanical advantage.

Offset cams have a different design parameter to that of a normal cam, which are described below:

1)Offset Distance: This is the distance from the follower's line of motion to the centerline or line of axis of the camshaft. Variation in offset distance results in varied cam geometry.

2) Pressure angle: Offset cams generally allows in reduction of the maximum pressure angle which is a crucial factor to reduce side thrust and wear.

3) Follower type: As in general cam, the type of follower, like knife-edge, roller, flat etc, affects the cam geometry.

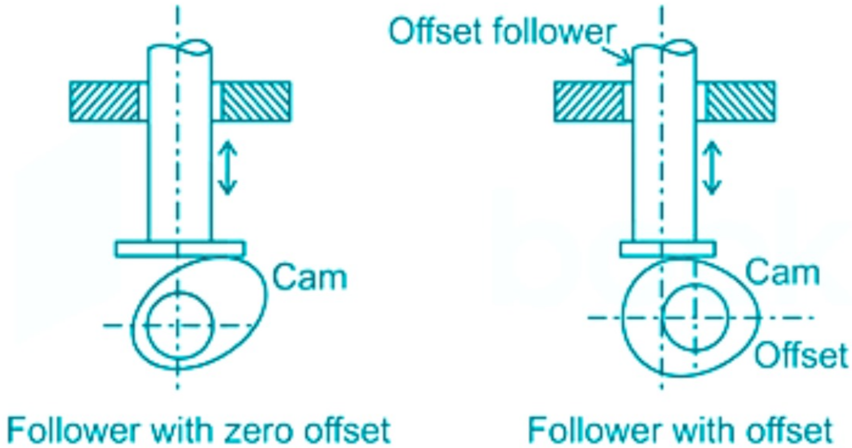

Figure 2.2: Offset cam mechanism [12]

2.4 Bevel Gears for Motion Conversion

Bevel gears are used to transmit power between intersecting shafts, typically at a 90° angle. They are widely used in applications requiring the conversion of horizontal motion to vertical motion, such as in differential drives, printing machines, and robotics.

2.4.1 Gear Ratio and Speed Conversion

The gear ratio determines the relationship between the input and output speeds of the bevel gears.

- **Gear Ratio:**

$$\text{Gear Ratio} = \frac{N_1}{N_2} = \frac{d_2}{d_1} \tag{2.10}$$

  where;

  N_1: Number of teeth of driver

  N_2: Number of teeth of driven

  d_1: Diameter of the driver

  d_2: Diameter of the driven

- **Speed Conversion:**

$$\omega_2 = \omega_1 \times \frac{N_1}{N_2} \tag{2.11}$$

where;

ω_1: Angular velocity of driver

ω_2: Angular velocity of driven

2.4.2 Torque and Power Transmission

The torque and power transmitted by bevel gears depend on the gear ratio and the input power. Key formulas and references include:

- Torque Transmission:

$$\tau_2 = \tau_1 \times \frac{N_2}{N_1} \tag{2.12}$$

where;

N_1: Number of teeth of driver

N_2: Number of teeth of driven

τ_1: Input Torque (Nm)

τ_2:Output Torque (Nm)

- Power Transmission:

$$P_1 = P_2 \times \eta \tag{2.13}$$

where;

P_1: Input power (W or kW)

P_2: Output power (W or kW)

η: Efficiency of the gear system (typically 0.95–0.98 for bevel gears)

2.5 Pad in Pad Printing Machine

Pad printing depends on silicon pads to transfer ink from a cliché to a substrate. The Printing quality is affected by the shape, hardness, and surface finish of the pad[13]. The pad's shape should support a rolling action to pick up ink from the cliché and deposit it onto the substrate, which prevents air from being trapped[13]. Trapping of air can cause print distortion. The hardness of the pad is another essential factor affecting print quality, which is determined by the amount of silicon oil used during moulding. There are four basic hardness levels which are color-coded for easy identification. Harder pads have better performance as they ensure precise ink transfer, but they may be impractical for low-power machines. Low power machine requires softer pads to avoid substrate damage. The surface finish of the pad should have a glossy finish. This is essential for the pad to pick up and transfer ink effectively. The silicone pad has a lifespan of up to 50,000 prints[13]. Optimal pad selection with care, using technical guidelines(pad shape, hardness, and surface finish) with practical experimentation, determines print quality and efficiency across industries, from automotive components to consumer goods, with future advancements.

Table 2.1: Hardness of Pad[13]

Color	**Hardness**
Blue	550 Shore (+2)
Pink	500 Shore (+2)
Green	450 Shore (+2)
White	350 Shore (+2)
Yellow	350 Shore (+2)

2.6 Ink Characteristics and Their Influence on Pad Printing Quality

The printing is highly dependent upon the right ink characteristics to ensure the efficient ink transfer and print durability on various substrates. According to ScreenPrinting Magazine (2023), ink formulation is one of the most critical factors affecting print quality in pad printing, where even minor variations in viscosity, solvent composition, or hardener ratios can lead to failures such as poor adhesion, incomplete transfer, or inconsistent color reproduction[14].

The study emphasizes the key parameters influencing ink performance, including:

a) Viscosity and Thixotropy: The inks in pad printing are usually set to low viscosity for easy transfer of the ink. However, the ink must also exhibit thixotropic behavior to ensure consistent settling and flow during the printing cycle [14].

b) Adhesion and Curing: According to the substrates such as plastics, metals, or glass, the ink should be carefully selected. Reactive or solvent-based 2-component inks are often recommended for high abrasion and chemical resistance[14]. For the optimization of ink durability, curing methods like UV curing or heat curing is responsible for ensuring adhesion strength [14].

c) Solvent Evaporation and Open Time: For the stability of ink viscosity, a properly balanced solvent plays a vital role. The control in the evaporation rate of ink helps to maintain work without drying of ink in the cliché.

d) Additives/ Hardener Mixing Ratios: Hardener percentagesmust be mixed according to ink manufacturer specifications. For instance, inclusion of $5-10\%$ hardener improves adhesion but the pot life gets sacrificed. There are also other additives such as retarders that can assist in extending open time and can be useful for prints at the high end of the scale [14].

2.7 Cliché in Pad Printing Machine

In pad printing machine,Cliché plays an important role as the carrier of the printed image. As noted by Šajgalík and Boršč (2021), "a cliché is made of a photopolymer or steel material, in which the printed image is engraved by an etching process" [15]. This engraved design is the source from which the pad picks up the ink for the printing.

The depth and quality of the etched design are responsible for print output. Specifically, "the depth of the image in the cliché affects the amount of ink that is transferred to the pad and subsequently to the product surface" [15]. If the etching is shallow, it will lead to inadequate ink pickup, while extra depth results in ink overflow or blur image.

The researcher also insisted in maintaining cliché in optimum condition: "The cliché must be cleaned thoroughly and regularly to ensure consistent ink transfer and image clarity" [15]. Additionally, the durability of the cliché material influences production efficiency. Photopolymer clichés are suitable for short runs due to their limited lifespan, whereas steel clichés are more robust and suited for longer production cycles [15].

The study showed that the fluctuation in cliché condition is a major source of error in pad printing process, especially when it is aiming for reproducibility and quality assurance. Therefore, "standardization of cliché preparation and inspection should be a priority in quality-controlled environments" [15].

CHAPTER 3: METHODOLOGY

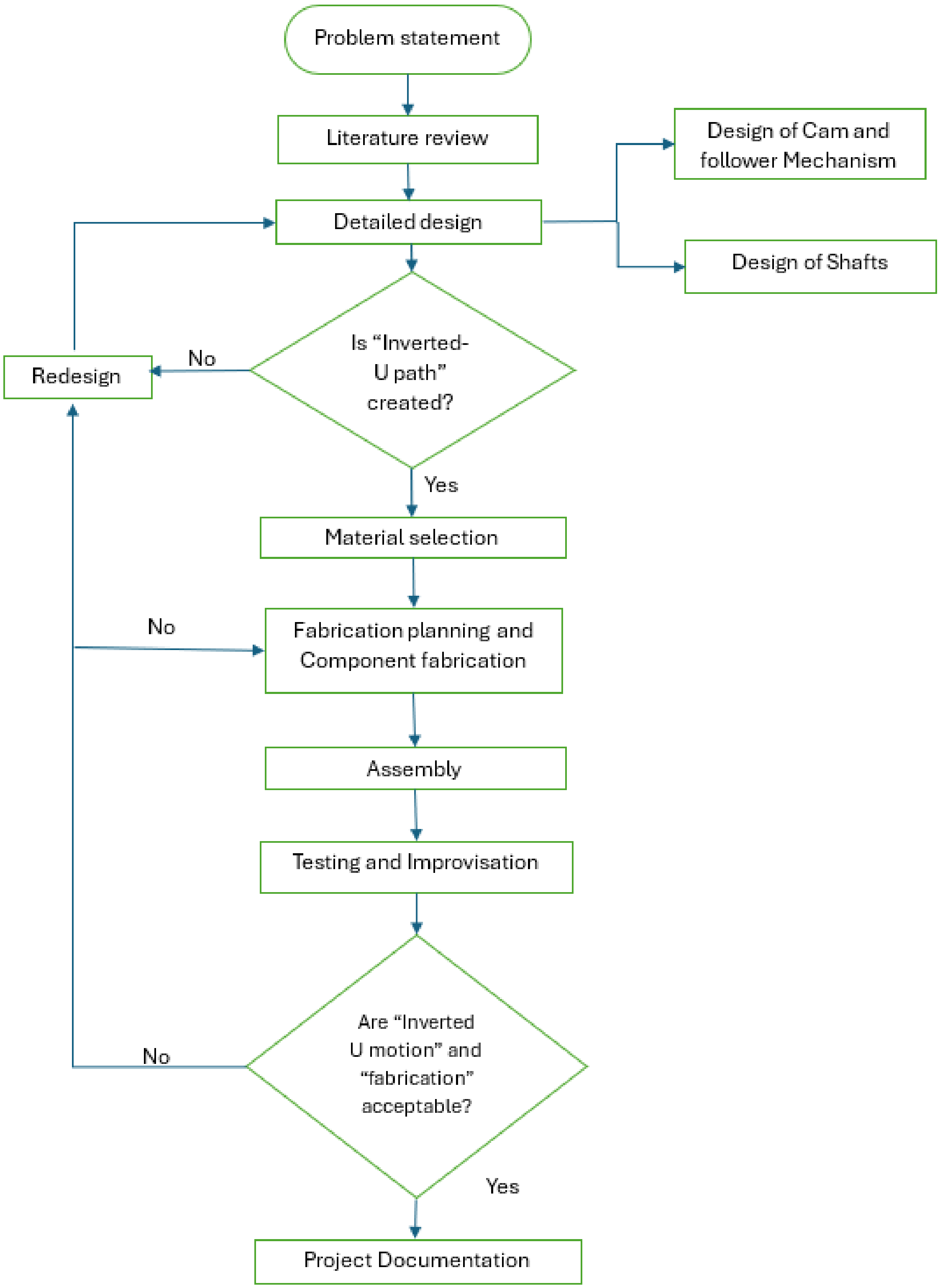

Figure 3.1: Methodology

3.1 Detailed Design

3.1.1 Conceptual Design

Key components along with the functionality of the pad printing machine are identified. Main Components include the printing head, the ink cup, the printing plate (cliché), and the substrate holder. Printing head will transfer image from the plate to the substrate. Ink cup and doctor blade will ensure precise amount of ink.

3.1.2 Design in SOLIDWORKS

Each components are modeled separately using SOLIDWORKS. The components are then assembled with the necessary constraints and relationships, which help simulate real-world interactions.

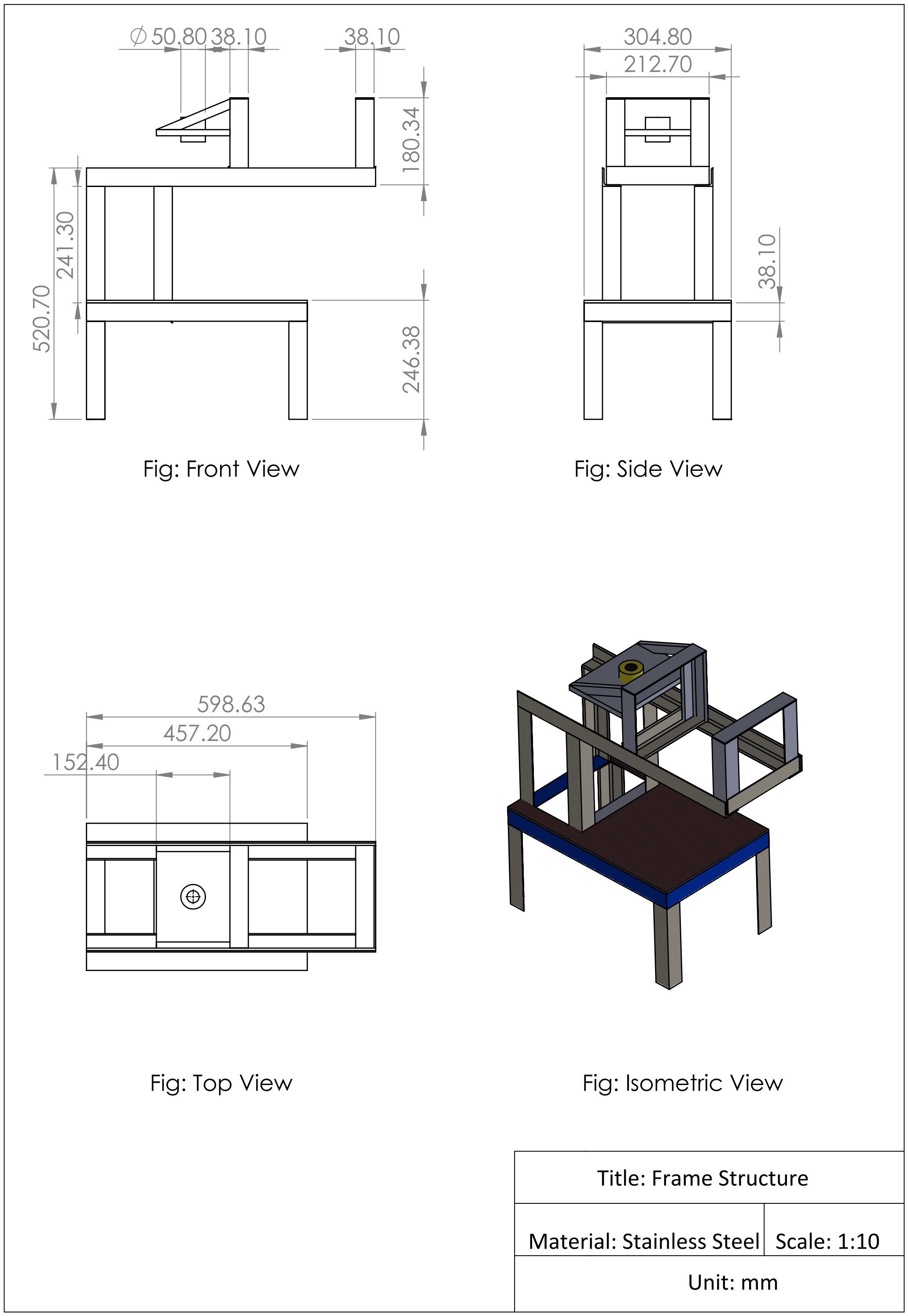

Figure 3.2: Frame Structure

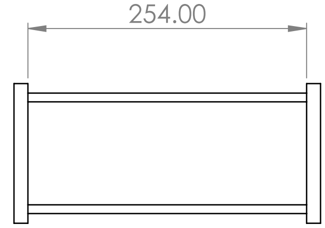

Fig: Front View

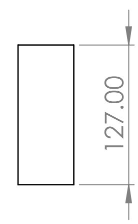

Fig: SIde View

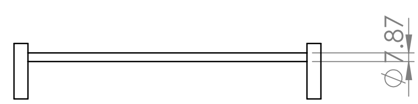

Fig: Top View

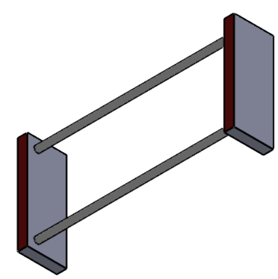

Fig: Isometric View

Title: Horizontal Slider	
Material: Stainless Steel	Scale: 1:5
Unit: mm	

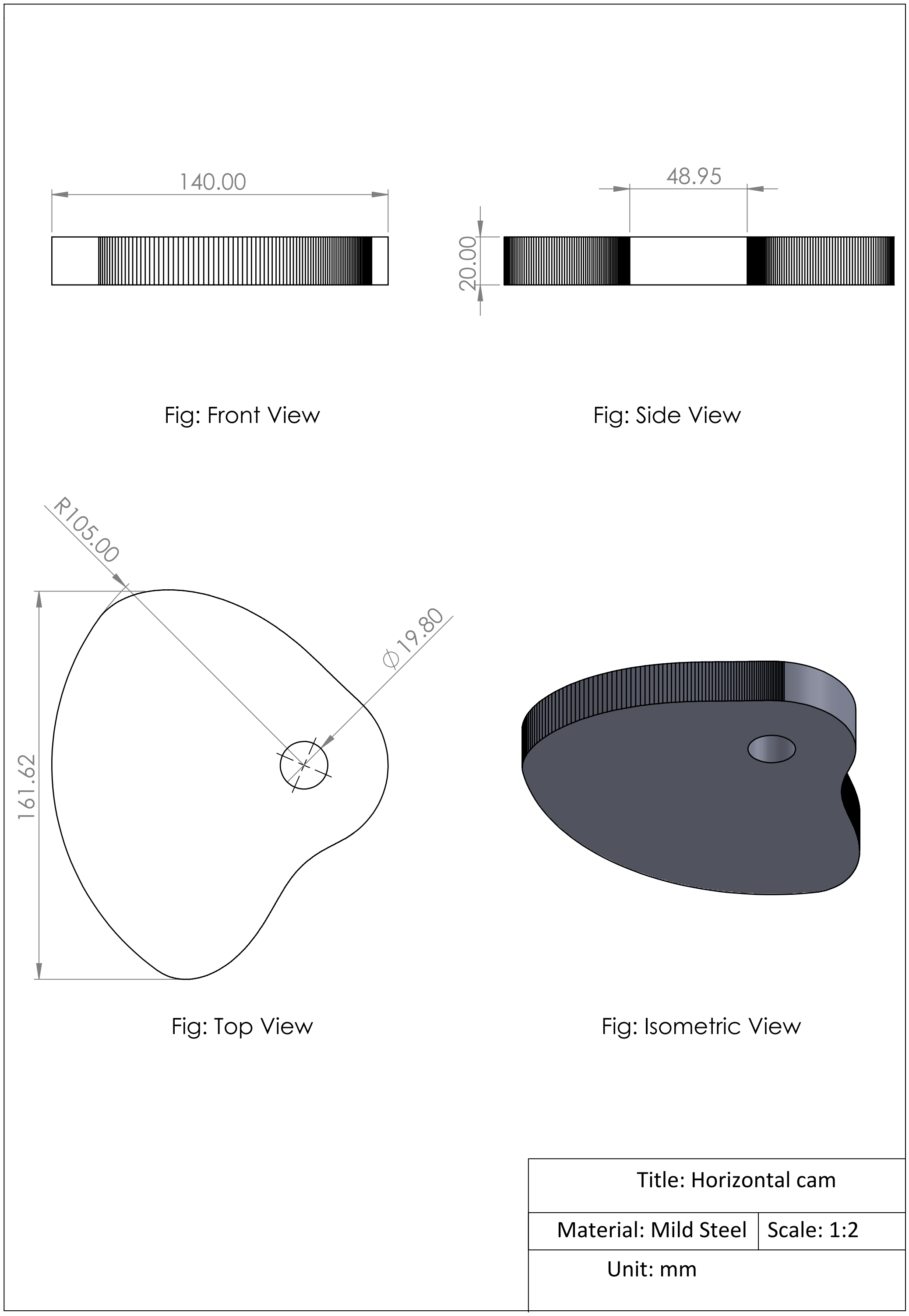

Figure 3.3: Horizontal cam (Offset)

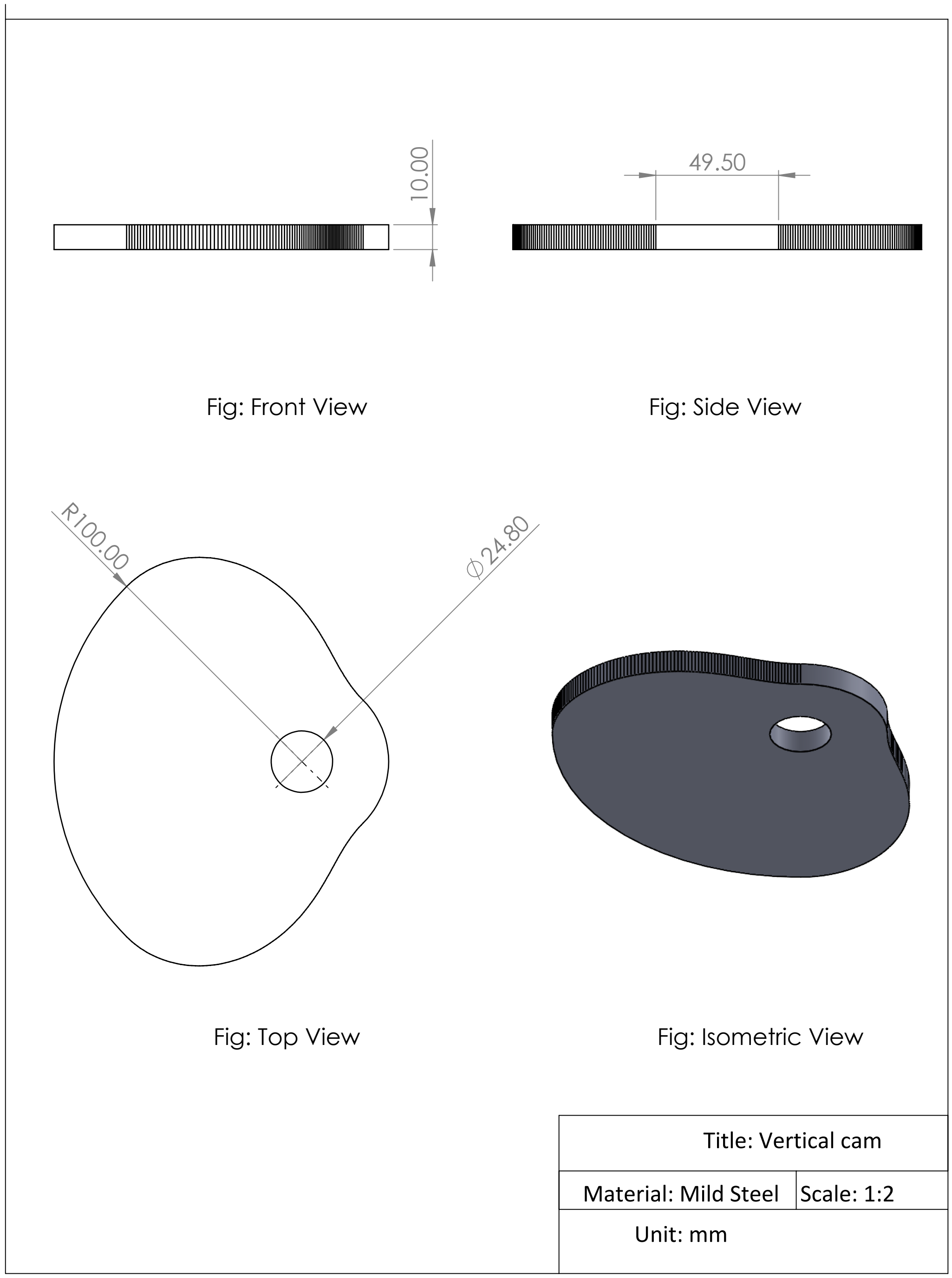

Figure 3.4: Vertical cam

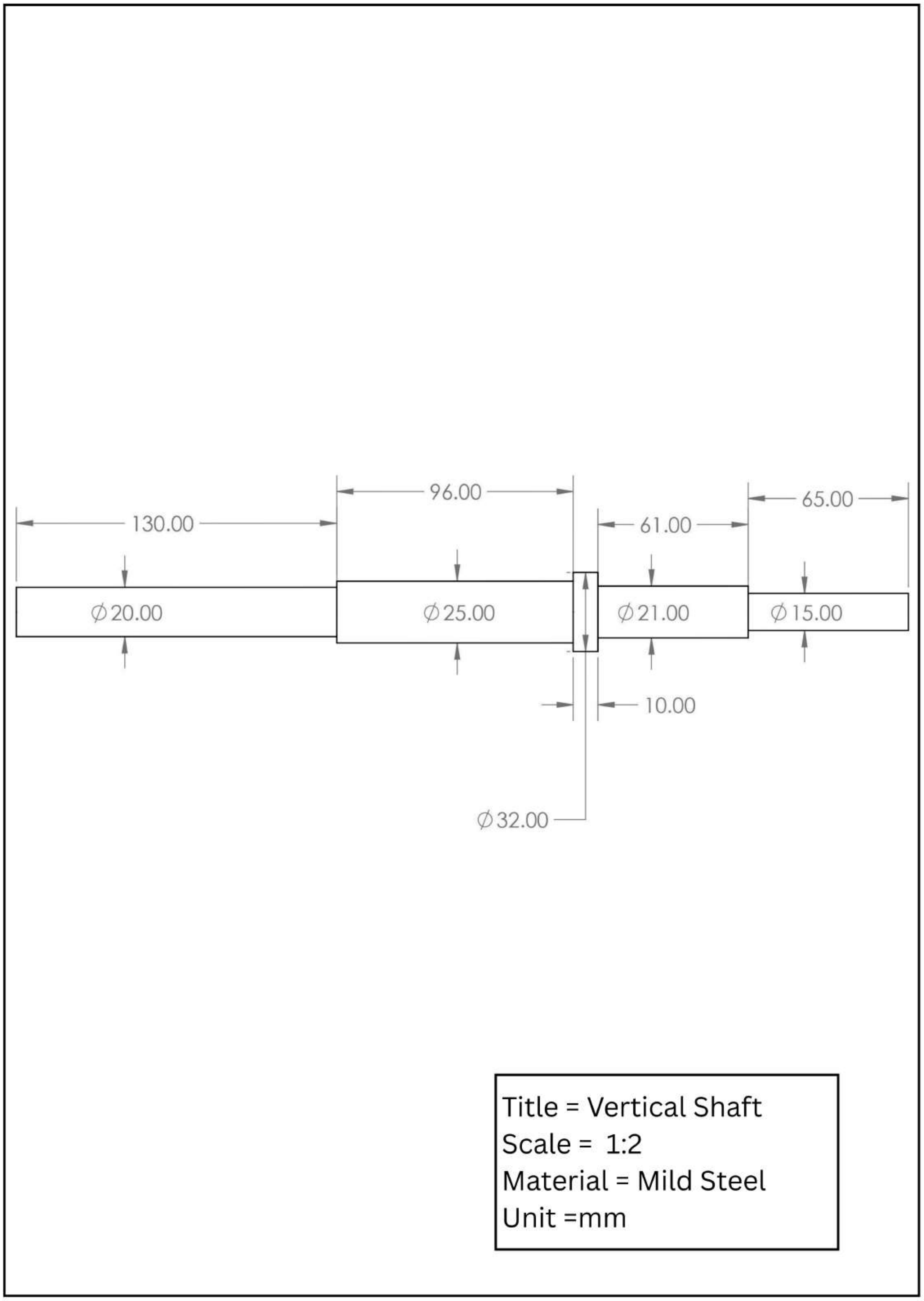

Figure 3.5: Vertical shaft

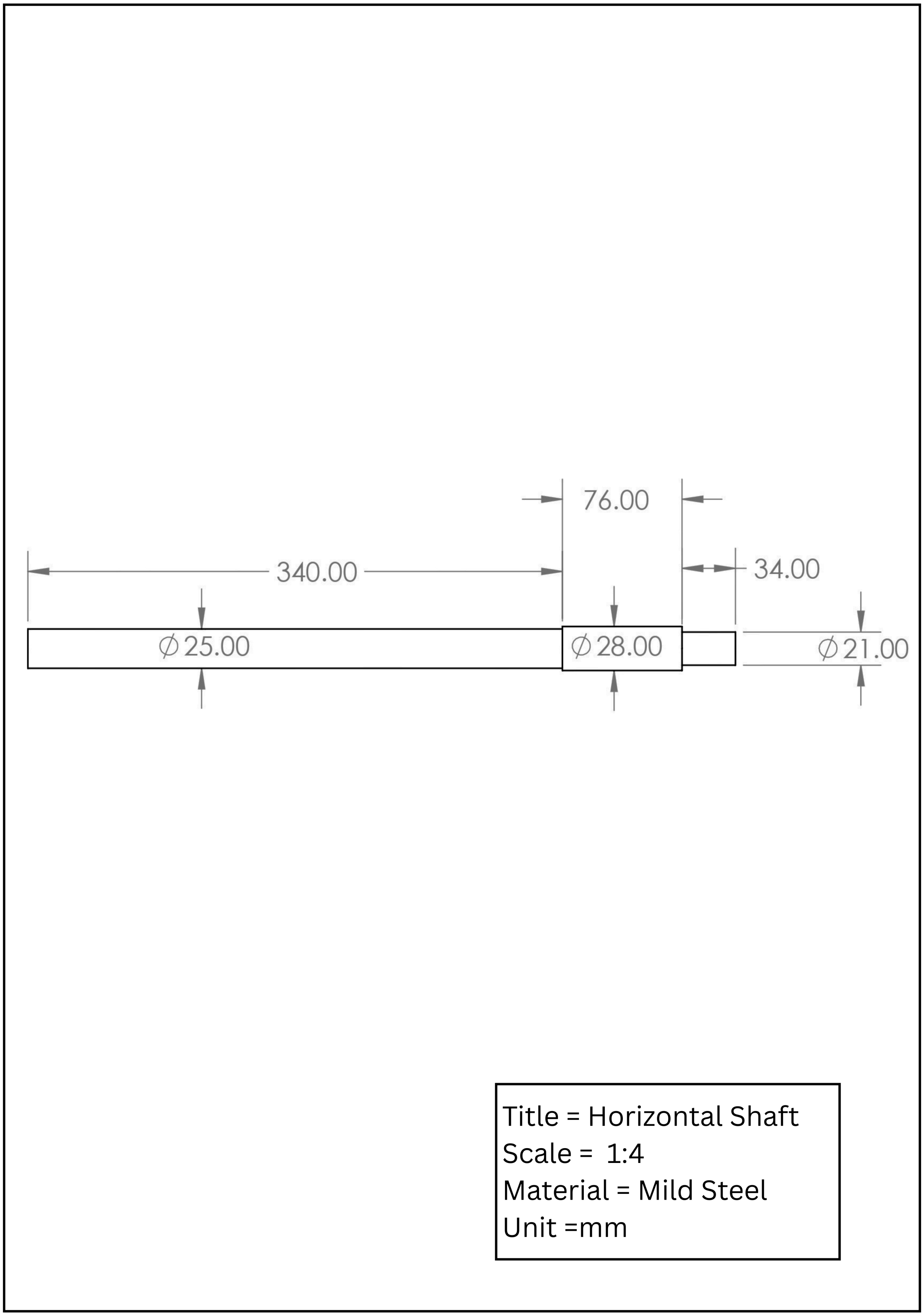

Figure 3.6: Horizontal Shaft

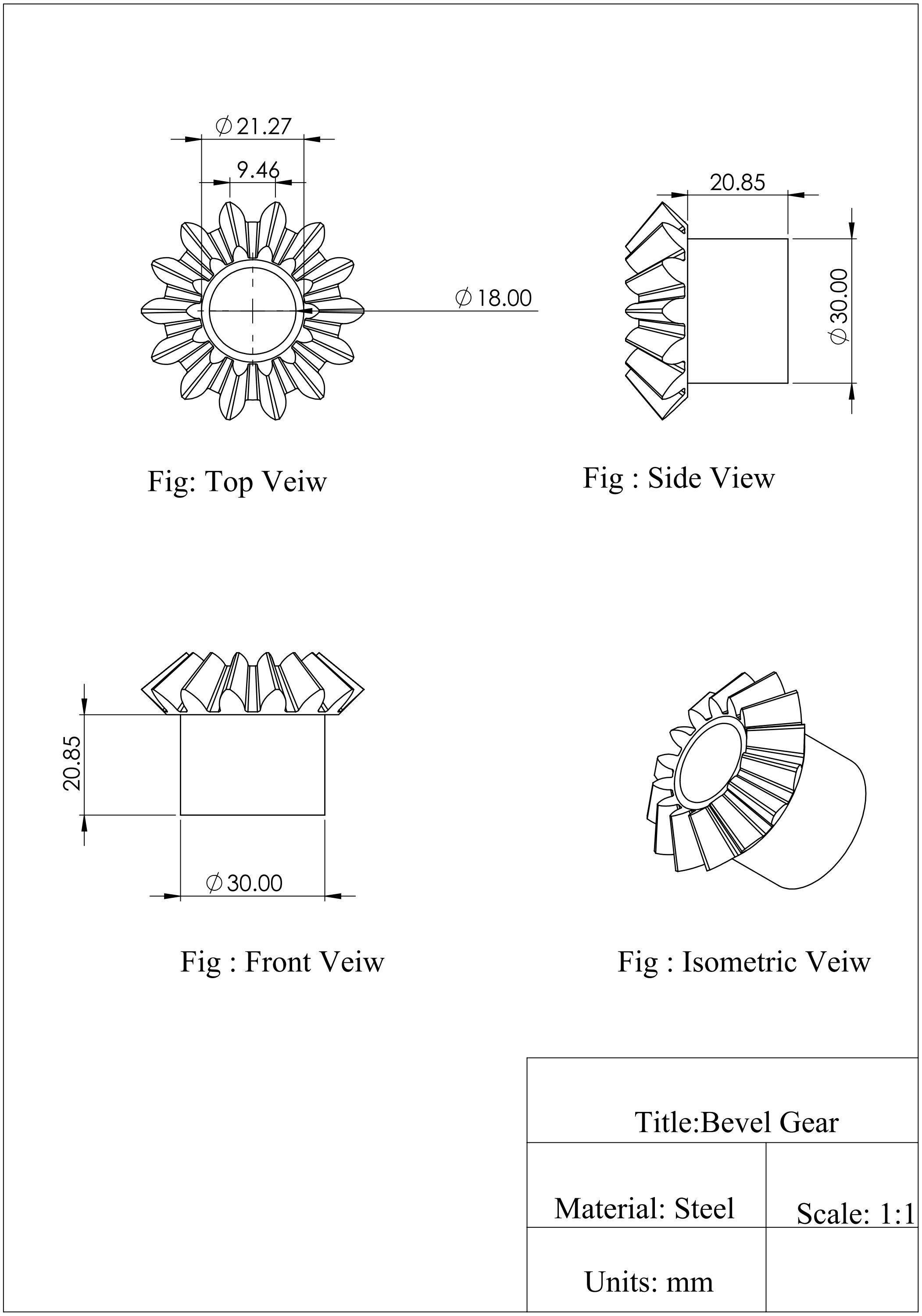

Figure 3.7: Bevel gear

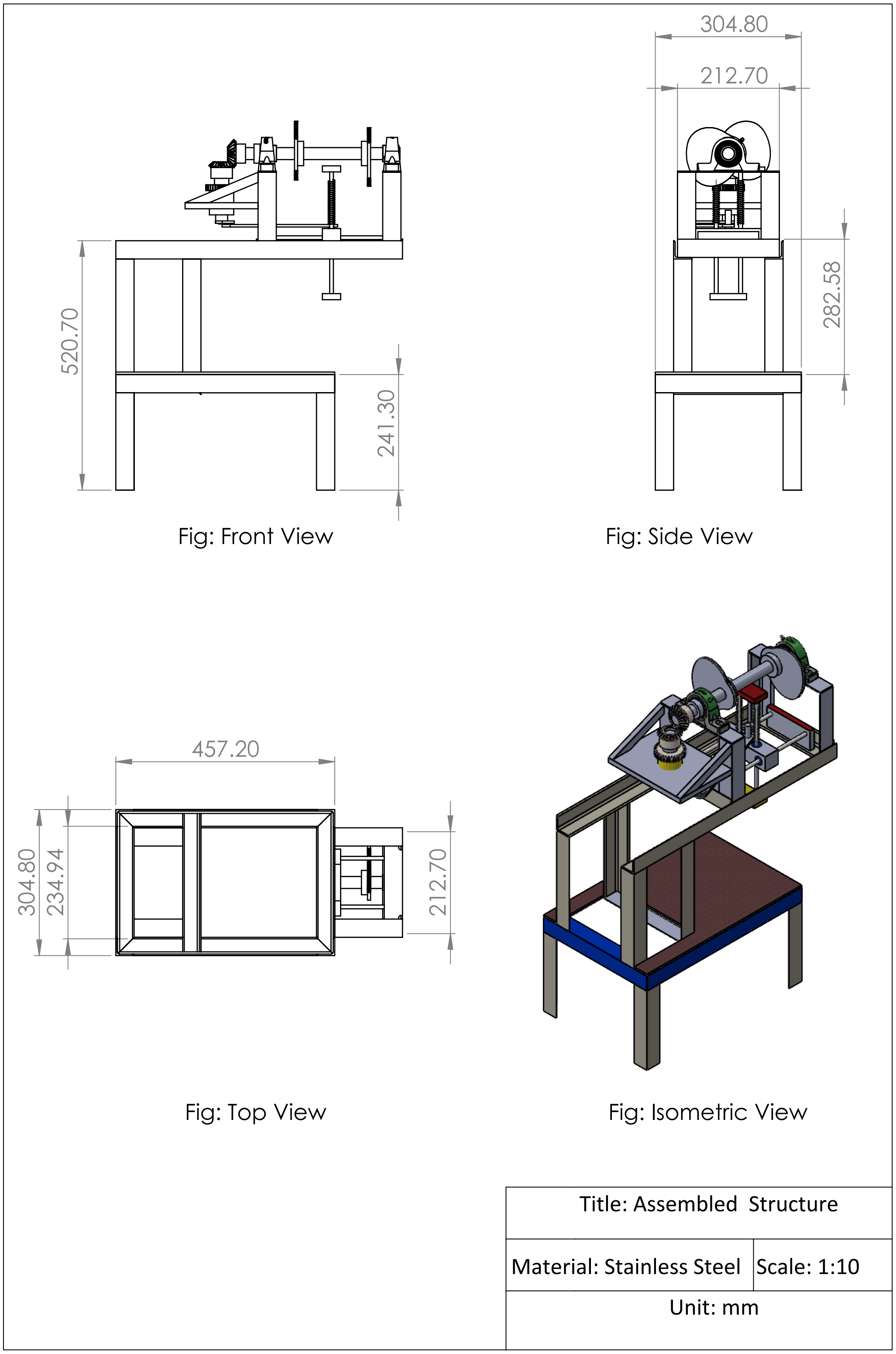

Figure 3.8: Assembled structure

3.1.3 Powering the Mechanisms

A single wiper motor is used to power both mechanisms. The power is transmitted using bevel gears, cam and shafts. The design is simple and has synchronized movement of the mechanism. The motor has the following specifications:

Table 3.1: Motor Specifications

Parameter	**Value**
Torque	15 Nm
Stop Torque	43 Nm
High Speed	60 RPM, 1.3 A
Low Speed	30 RPM, 0.7 A
Power	80 W
Voltage	24 V

3.1.4 Stiffness Calculation of Spring

Compression Spring(Vertically positioned):

The springs are used for the return of the vertical rod carrying the silicon pad. It is compressed by the cam for the printing purpose. We have taken two different springs and the process was carried out. The Stiffness was calculated by placing varying masses over the spring and noting the change in length of the spring. From this, the calculation for the stiffness was carried out resulting in the average stiffness for those two springs. The Original length of the springs during the calculation was 11cm for both of the springs.

Table 3.2: Spring Measurements and Calculations(Compression)

Load (kg)	**Deformed length (L) (m)**		**Change in Length (ΔL)**		**Stiffness**	
	Spring1	**Spring2**	**Spring1**	**Spring2**	**Spring1**	**Spring2**
0.4	0.095	0.1	0.015	0.01	261.6	392.4
0.8	0.08	0.09	0.03	0.02	261.6	392.4
1.2	0.062	0.08	0.048	0.03	245.25	392.4
1.6	0.049	0.067	0.061	0.043	257.3155	365.0233
2	0.029	0.057	0.081	0.053	242.2222	370.1887
2.4	0.025	0.048	0.085	0.062	276.9832	379.7419

These are the data taken by repeating the load 6 times(difference of 400 g) for both springs. Then by averaging the stiffness, we got the average stiffness for spring1 and spring2 as 257.4953 N/m and 382.0256 N/m respectively.

Expansion Spring(Horizontally positioned):

This spring is used to provide the required expansion and return of the slider arrangement. Connects the slider arrangement to the frame. The stiffness was calculated by placing varying masses over the spring and noting the change in length of the spring. From this, the calculation for the stiffness was carried out resulting in the average stiffness. The original length of the spring during the calculation was noted as 76mm.

These are the data that are observed by adding the weight to the spring. The loads were added with an increase of 400gm. It was carried out 6 times and the respective stiffness were also calculated. The average stiffness is 622.240956 N/m.

Table 3.3: Spring Measurements and Calculations(Expansion)

Load (kg)	Deformed Length (L) (m)	Change in Length(ΔL)	Stiffness of Spring
0.4	0.082	0.006	654
0.8	0.089	0.013	603.6923077
1.2	0.095	0.019	619.5789474
1.6	0.102	0.026	603.6923077
2	0.107	0.031	632.9032258
2.4	0.114	0.038	619.5789474

3.1.5 Force Analysis of Cam and Spring Mechanism

a. Force Exerted by the Springs

The force exerted by each spring is given by Hooke's Law:

$$F_{\text{spring}} = k \cdot x \tag{3.14}$$

where:

- k: Spring stiffness (N/m)
- x: Spring deflection (m)

Since there are two springs, the total spring force is:

$$F_{\text{spring, total}} = 2 \cdot F_{\text{spring}} \tag{3.15}$$

b. Forces During Motion

I. **When Springs Push Upward:** The springs push the arrangement upward against gravity. The net upward force is:

$$F_{\text{up}} = F_{\text{spring, total}} - W \tag{3.16}$$

where:

- W: Weight of the arrangement (N)

II. **When Cam Pushes Downward:** The weight of the arrangement acts downward, and the cam does not need to exert any additional force to move the arrangement downward. Thus:

$$F_{\text{cam}} = 0 \tag{3.17}$$

c. Equation of Force Exerted by Cam with Respect to Displacement

The force exerted by the cam (F_{cam}) as a function of the spring displacement (x) is given by:

$$F_{\text{cam}}(x) = 2 \cdot k \cdot x - W \tag{3.18}$$

where:

- k: Spring stiffness (N/m)
- x: Spring displacement (m)
- W: Weight of the arrangement (N)

d. Maximum and Minimum Forces

I. **Maximum Force** The maximum force occurs when the cam is fully extended ($x = 0.1$ m) and must overcome the maximum spring force and weight:

$$F_{\text{cam, max}} = 2 \cdot k \cdot 0.1 - W \tag{3.19}$$

II. **Minimum Force** The minimum force occurs when the cam is at $x = 0$ and does not need to exert any force because the weight itself causes the arrangement to move downward. Thus:

$$F_{\text{cam, min}} = 0 \tag{3.20}$$

3.1.6 Torque Required

The shaft will require a certain torque range so that it can rotate and operate. There are two shafts, i.e., horizontal and vertical shafts. The equations required to calculate the torque are listed below.

Horizontal Shaft:

$$\tau_{\text{fric},h} = \mu F_h r_h \tag{3.21}$$

$$\tau_{\text{spring},h} = F_h r_h \tag{3.22}$$

$$\tau_{\text{inertia},h} = J_h \alpha \tag{3.23}$$

$$\tau_{\text{total},h} = \tau_{\text{fric},h} + \tau_{\text{spring},h} + \tau_{\text{inertia},h} \tag{3.24}$$

Vertical Shaft:

$$\tau_{\text{fric},v} = \mu F_v r_v \tag{3.25}$$

$$\tau_{\text{spring},v} = F_v r_v \tag{3.26}$$

$$\tau_{\text{inertia},v} = J_v \alpha \tag{3.27}$$

$$\tau_{\text{total},v} = \tau_{\text{fric},v} + \tau_{\text{spring},v} + \tau_{\text{inertia},v} \tag{3.28}$$

System Requirement:

The overall torque required by the system for operation will be the sum of both shafts.

$$\tau_{\text{system}} = \tau_{\text{total},h} + \tau_{\text{total},v} \tag{3.29}$$

3.1.7 Equations for Shaft Analysis

Case 1: Cam1 Active

- **Reaction Forces**

$$\sum F_y = R_A + R_B = F_s - 2W_c + W_{\text{shaft}} \tag{3.30}$$

$$\sum M_{B1} = F_s \times 0.122 - W_c \times 0.122 - W_c \times 0.188 + M_f - R_B \times 0.284 = 0 \tag{3.31}$$

- **Bending Moments**

$$M(x) = R_A x - \frac{wx^2}{2} \quad (\text{for } x < 0.249\,\text{m}) \tag{3.32}$$

$$M(x) = R_A x - \frac{wx^2}{2} + (F_s - W_c)(x - 0.249) \quad (\text{for } x \geq 0.249\,\text{m}) \tag{3.33}$$

Case 2: Cam2 Active

- **Reaction Forces**

$$\sum F_y = R_A + R_B = F_s - 2W_c + W_{\text{shaft}} \tag{3.34}$$

$$\sum M_{B1} = -W_c \times 0.122 + F_s \times 0.188 - W_c \times 0.188 + M_f - R_B \times 0.284 = 0 \tag{3.35}$$

- **Bending Moments**

$$M(x) = R_A x - \frac{wx^2}{2} \quad \text{(for } x < 0.315\,\text{m)} \tag{3.36}$$

$$M(x) = R_A x - \frac{wx^2}{2} + (F_s - W_c)(x - 0.315) \quad \text{(for } x \geq 0.315\,\text{m)} \tag{3.37}$$

3.1.8 Shaft Design Equations

1. Fatigue Analysis(Moment Components)

$$M_m = \frac{M_{\max} + M_{\min}}{2} \tag{3.38}$$

$$M_a = \frac{M_{\max} - M_{\min}}{2} \tag{3.39}$$

2. Material Properties

$$\sigma_e' = 0.5\sigma_{\text{ult}} \tag{3.40}$$

$$\sigma_e = k_a k_b \sigma_e' \tag{3.41}$$

3. Stress Calculations

$$\sigma_m = \frac{32M_m}{\pi d^3} \tag{3.42}$$

$$\sigma_a = \frac{32M_a}{\pi d^3} \tag{3.43}$$

$$\tau_m = \frac{16T}{\pi d^3} \tag{3.44}$$

4. Goodman Criterion

$$\frac{\sigma_a}{\sigma_e} + \frac{\sigma_m}{\sigma_{\text{ult}}} + \frac{\tau_m}{0.5\sigma_{\text{ult}}} \leq \frac{1}{n_f} \tag{3.45}$$

5. Diameter Calculation

$$d^3 \geq \frac{32}{\pi}\left(\frac{n_f M_a}{\sigma_e} + \frac{n_f M_m}{\sigma_{\text{ult}}}\right) + \frac{16 n_f T}{0.5\pi\sigma_{\text{ult}}} \tag{3.46}$$

6. Stress Concentration

$$\sigma_{\max} = K_t \sigma_a + \sigma_m \tag{3.47}$$

3.1.9 Bending Moment Equations

Vertical Bending Moment (Mz):

$$\text{Region 1: } M_z(x) = R_{Az}x - \frac{wx^2}{2} \tag{3.48}$$

$$\text{Region 2: } M_z(x) = R_{Az}x - \frac{wx^2}{2} + R_{Bz}(x - x_{B1}) \tag{3.49}$$

$$\text{Region 3: } M_z(x) = R_{Az}x - \frac{wx^2}{2} + R_{Bz}(x - x_{B1}) - W_c(x - x_{B2}) \tag{3.50}$$

Horizontal Bending Moment (My):

$$\text{Region 1: } M_y(x) = R_{Ay}x \tag{3.51}$$

$$\text{Region 2: } M_y(x) = R_{Ay}x + R_{By}(x - x_{B1}) \tag{3.52}$$

$$\text{Region 3: } M_y(x) = R_{Ay}x + R_{By}(x - x_{B1}) - F_s(x - x_{B2}) \tag{3.53}$$

Combined Moment:

$$M_{\text{total}}(x) = \sqrt{M_y(x)^2 + M_z(x)^2} \tag{3.54}$$

3.1.10 Simulation and Analysis

The design has been simulated kinematically to check movements of the parts and the mechanism in SOLIDWORKS and ANSYS. This motion analysis has helped to visualize the timing and path traced by each mechanismeachess, and each force analysis is conducted to ensure that all the parts can withstand the operational load.

3.1.11 Design Iteration

After simulation and analysis, the design was adjusted accordingly to resolve encountered issues. The design was iterated to adjust the mechanism, dimension, materials, and connections. This improved component functioning and machine performance. Refinement with each iteration ensured that the components meet their functional requirement under various real-world conditions.

3.1.12 Documentation and Finalization

The final step involved documentation of technical drawings, which includes dimensions, tolerance,, and material specification. Bill of Materials (BOM) will include all components, quantities, and materials required for fabrication. This documentation will make sure that the design can be manufactured easily using conceptual design.

3.2 Material Selection

Materials for each part were selected carefully, likely considering the parts, their functionality, and durability. Materials that are cost-effective and locally available were given priority without affecting the quality of the machine. The ingredients and materials used.

Table 3.4: Materials and Reasons for Selection in Manufacturing Components

Component	**Material**	**Reason for Selection**
Structural Components (Frame & Parts)	Mild Steel, Aluminium	Mild steel provides strength and stability, whereas aluminium reduces weight and resists corrosion. Used where weight reduction was needed.
Mechanisms (Double Cams & Slider-Crank)	Mild Steel	Provides strength and fatigue resistance, is locally available, and cost effective.
Pad Printing Head	Aluminium	Corrosion-resistant and easy to clean.
Ink Cup	Stainless Steel	Durable and corrosion-resistant.
Printing Plate (Cliché)	Ceramic	High hardness; holds fine details for quality printing.
Substrate Holder	Aluminium	Lightweight and strong.
Bearings and Bushings	Bronze, Steel	Provide wear resistance; can operate without lubrication.
Fasteners and Springs	Stainless Steel, Spring Steel	Stainless steel prevents corrosion; spring steel offers strength and elasticity.

3.3 Component Fabrication Process

3.3.1 Design Translation

The conceptual design was modelled into SOLIDWORKS. Each component and part were modeled separately and assembled. Specification of the machine was transformed into engineering drawings. This drawing consists of detailed dimensions, size, and materials of each part.

3.3.2 Components Selection and Specification

- Linear Bearing: The Linear Bearing is a Linear Bush type of Bearing consisting of a machined aluminum block with a press-fit precision bearing bushing with an 8 mm precision linear shaft. They are used for the operation of the vertical and horizontal movement of the pad. The specifications of the linear bearing that were used are:

Table 3.5: Specifications for Linear Bearing

Name	Linear Bearing
Linear Shaft	8mm
Dynamic Load Rating	882 N
Static Load Rating	1370 N

- Pillow Block Bearing: Pillow block ball bearing unit is with extended inner ring and set screw locking, made up of cast iron. Since the generally available pillow block bearing is of 25mm diameter, the shaft to be attached with this bearing was also chosen to be 25mm.

Table 3.6: Dimensions of the component

Parameters	Dimensions
Attachment bolt diameter	10 mm
Shaft diameter	25 mm
Centre height (pillow block)	36.5 mm
Housing overall width	38 mm
Centre distance between bolt holes	105 mm
Bearing width, total	34 mm

3.3.3 Machining Operation

Machining operations like operations, CNC laser cutting, drills, lathe and grinders were used to create each part from selected materials. Machining operations involves following:
Welding - joining two or more components permanently, making frame arc welding was used.
CNC laser cutting - used to cut the cam to it's precise shape
Lathe Machine - designing stepped shaft (Turning), Facing
Milling - Etching of pattern
Surface grinding - smoothing exposed surface, smooth cliche
Drilling - creating holes for nut bolt fasteners
Threading - cutting threads for bolts
Grinding and cutting - cutting shafts, bars, plates etc

3.3.4 Surface Treatment

Parts were refined protected by smoothing using grinding and other operations. Painting or polishing was applied for surface protection and aesthetics. This improved their appearance and functionality.

CHAPTER 4: RESULTS AND DISCUSSION

4.1 Calculation results

4.1.1 Force exerted by horizontal cam:

The force required to extend the spring by 70 mm is:

$$F = 622.06\text{ N/m} \times 0.07\text{ m} = 43.54\text{ N}$$

4.1.2 Torque required

a. Horizontal Shaft:

$$\tau_{\text{fric},h} = 1.533\,\text{N}\cdot\text{m}$$
$$\tau_{\text{spring},h} = 6.665\,\text{N}\cdot\text{m}$$
$$\tau_{\text{inertia},h} = 0.00198\,\text{N}\cdot\text{m}$$
$$\tau_{\text{total},h} = 8.20\,\text{N}\cdot\text{m}$$

b. Vertical Shaft

$$\tau_{\text{fric},v} = 1.052\,\text{N}\cdot\text{m}$$
$$\tau_{\text{spring},v} = 4.572\,\text{N}\cdot\text{m}$$
$$\tau_{\text{inertia},v} = 0.000706\,\text{N}\cdot\text{m}$$
$$\tau_{\text{total},v} = 5.62\,\text{N}\cdot\text{m}$$

System Requirement:

$$\tau_{\text{system}} = 13.82\,\text{N}\cdot\text{m}$$

Summary

Case 1: Cam1 Active

- Reaction Forces: $R_A = 43.32\,\text{N}, \quad R_B = 24.84\,\text{N}$
- Bending Moments:
  - At B1: $5.14\,\text{N·m}$
  - At Cam1: $10.96\,\text{N·m}$
  - At Cam2: $15.33\,\text{N·m}$

Case 2: Cam2 Active

- Reaction Forces: $R_A = 27.63\,\text{N}, \quad R_B = 40.53\,\text{N}$
- Bending Moments:
  - At B1: $3.15\,\text{N·m}$
  - At Cam1: $5.50\,\text{N·m}$
  - At Cam2: $7.47\,\text{N·m}$

Critical Values:

Parameters	**Value**
Maximum Moment (Cam1 Active)	15.33 N·m
Maximum Moment (Cam2 Active)	7.47 N·m
Frictional Moment (M_f)	1.554 N·m

Bending moment diagram(Cams)

To have a clearer understanding of how the shaft in the pad printing machine behaves under load, bending moment diagrams were constructed for the two loading conditions: Cam1 Active (Cam2 not active) and Cam2 Active (Cam1 not active). The length of the shaft is 41.1 cm with two supports—Bearing 1 at 12.7 cm and Bearing 2 at the end point (41.1 cm). A steady torque of 15 N·m is also applied; however, because torque mainly causes twisting as opposed to bending, this analysis is focused on bending caused by forces along and across the shaft only.

Both cams apply a spring force of 67.57 N when activated and weigh 8.83 N. The shaft also bears its own weight, expressed as a distributed load of 44.4 N/m, totaling 18.25 N. Cam1 is positioned at 24.9 cm, and Cam2 at 31.5 cm along the shaft.

I. Cam1 Active Case:

In the Cam1 Active case, plotted in the top graph with a blue line, the bending moment starts at zero at the gear end (0 cm), due to the fact that it is a pinned support and therefore cannot resist any moment. As you move along the shaft, the moment increases due to the distributed weight, reaching a value of 5.14 N·m at Bearing 1 (12.7 cm). This is in agreement with calculations showing that the reaction force at Bearing 1 is 43.32 N.

The first major peak occurs at 24.9 cm (position of Cam1) where spring upward force (67.57 N) and cam downward weight (8.83 N) are both acting in the same direction, creating a net upward force of 58.74 N. This creates a steep peak in bending moment of 10.96 N·m. Frictional moment of 1.554 N·m is also created here, which assists in this increase. The bending moment continues to increase, reaching a value of 15.33 N·m at Cam2 (31.5 cm), although Cam2 is inactive. The reason for this lies in the persistence of forces applied in the previous stages. The moment eventually decreases to zero at Bearing 2 (41.1 cm) owing to its support condition.

Throughout this case, the shaft bends upwards (positive moment) due to the dominant upward force from the spring of Cam1.

II. Cam2 Active Case:

For the Cam2 Active example (red line in lower figure), the bending moment also starts at zero at the gear end. It increases more slowly, to 3.15 N·m at Bearing 1, since the spring force is applied later along the shaft, and the reaction at Bearing 1 is smaller (27.63 N). At 24.9 cm (Cam1), the moment is at 5.50 N·m, courtesy of the cam weight, since Cam1 is inactive here. There is a steep peak at Cam2 (31.5 cm) where the same spring force upwards and cam weight take effect, causing a net force of 58.74 N and yielding a bending moment of 7.47 N·m (including the frictional moment of 1.554 N·m). The moment then drops back to zero at Bearing 2.

Similar to the first case, the shaft bends upward in general, but the peak moment is smaller since the spring force is nearer the end of the shaft, with less leverage for moment development.

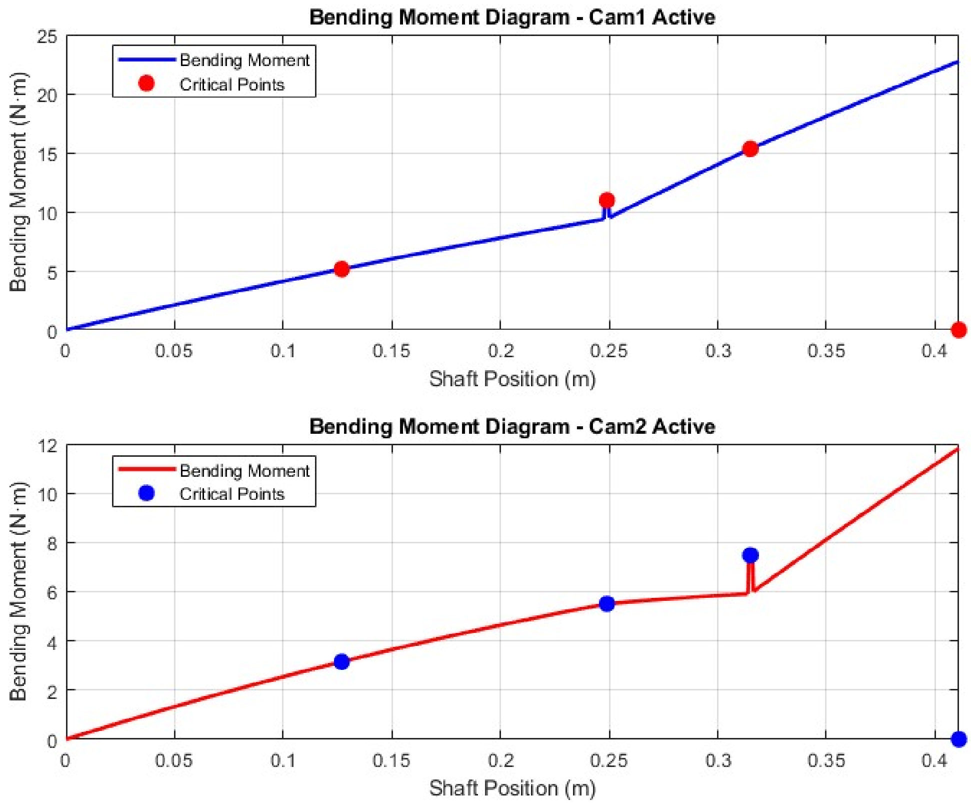

Figure 4.1: Bending moment diagram(Cam)

Horizontal Shaft Design Results

Final Specifications:

Parameter	Value
Material	AISI 1018 Mild Steel
Main Shaft Diameter	16 mm
Gear Mount Diameter	20 mm
Cam Section Diameter	18 mm
Yield Safety Factor	2.5
Fatigue Safety Factor	3.0
Maximum Alternating Stress	46.6 MPa

Key Observations: Both bending moment diagrams exhibit the typical behavior of a simply supported beam loaded in the center. There are steep discontinuities (spikes) in the bending moment at the active cam positions—Cam1 in this first instance and Cam2 in the second—because of the huge net upward forces and the frictional moments added. The Cam1 Active case produces the highest bending moment of 15.33 N·m at Cam2, which is design-critical. Since the spring force is applied farther from the support, it creates more leverage and more stress. This peak value needs to be used in order to make sure that the shaft is designed with an adequate amount of strength and stiffness to withstand operation without failure.

Key Verification Points:

- Critical section: Shoulder fillet (r = 1.6 mm)
- Stress concentration factor: Kt = 1.7
- Goodman criterion ratio: 0.28 (limit = 0.33)

Recommended Dimensions:

Section	Diameter (mm)	Length (mm)
Gear Mount	20	38
Bearing Journal 1	16	20
Cam Section	18	98
Bearing Journal 2	16	30

4.1.3 Bending Moment(Shafts)

In order to understand how the vertical shaft can handle different forces while in operation, bending moment diagrams were drawn. These diagrams are used to show the distribution of internal stresses along the shaft under different loads.

The Vertical Bending Moment (Mz) plot indicates a peak negative moment of -1.33 N·m at the position of the cam, 16.9 cm from the base of the shaft.

This is a negative moment because of the weight of the shaft itself, as a uniformly distributed load of 52.2 N/m, and the point load due to the weight of the cam (8.63 N). The negative sign means the shaft is bending downwards, or sagging, under these vertical gravity forces.

Contrary to the Vertical Bending Moment (My) diagram, it shows a different trend. The highest moment here is 3.71 N·m at Bearing 2, 13.6 cm from the shaft. This moment is essentially caused by a 43.54 N horizontal spring force on the cam. This produces a high sideways bending, especially near the bearing.

When combined with the horizontal and vertical moments, the Combined Bending Moment has a maximum value of 3.93 N·m at Bearing 2. This indicates quite clearly that the horizontal spring force is the predominant contributor to the overall bending stress in the shaft, particularly near the cam and Bearing 2. Although the vertical shaft and cam forces do produce bending, the impact is significantly less than that of the horizontal forces.
Hence, the shaft must be designed to resist the highest combined bending moment of 3.93 N·m. Design must aim at ensuring sufficient strength and stiffness, particularly where the stress is highest, in an attempt to guarantee trouble-free operation of the shaft with no failure during operation.

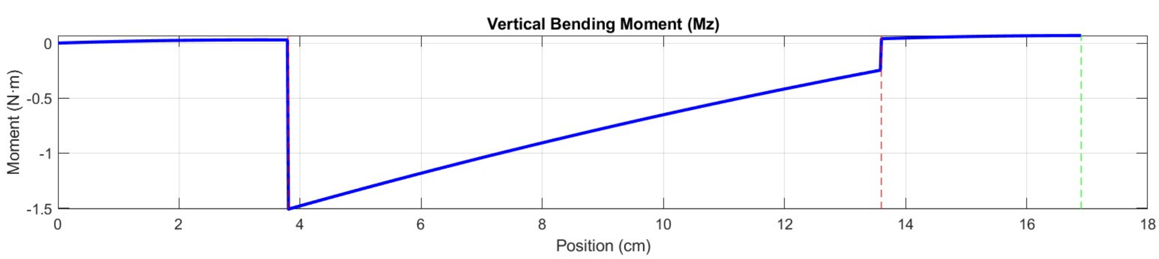

Figure 4.2: Bending Moment (Vertical Shaft)

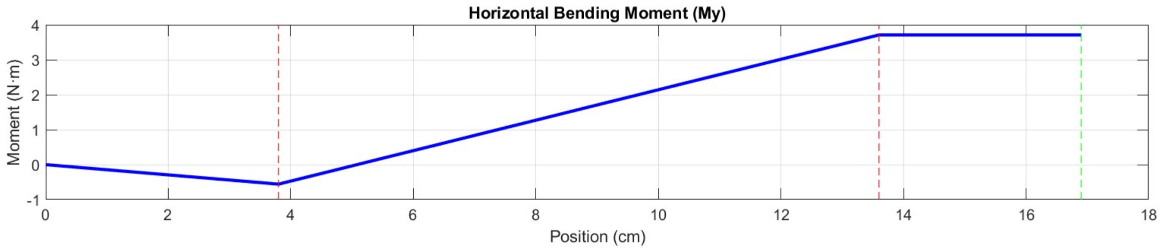

Figure 4.3: Bending Moment (Horizontal Shaft)

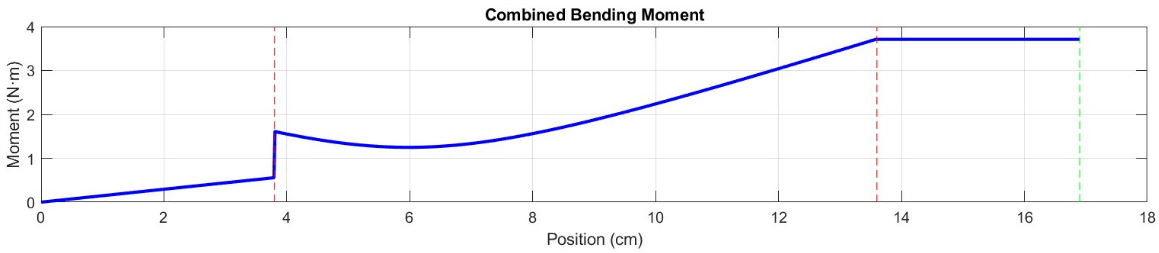

Figure 4.4: Bending Moment (Combined)

Bending Moment Results

Location	Vertical Moment (N·m)	Horizontal Moment (N·m)	Total Moment (N·m)
B1 (3.8 cm)	0.028	-0.56	0.56
B2 (13.6 cm)	1.30	3.71	3.93
Cam (16.9 cm)	1.33	3.70	3.93

Table 4.1: Peak bending moments at critical locations

Key Observations

- Maximum combined moment: 3.93 N·m at B2 and Cam
- Horizontal spring force dominates (contributes 95% of stress)
- Safety factor: 7.6 (AISI 1018, $\sigma_{yield} = 370$ MPa)

Vertical Shaft Design Result:

I. Minimum Diameter

$$d \geq 12.9\,\text{mm}$$

II. Selected Standard Size

$$\boxed{d = 16\,\text{mm}}$$

III. Verification Goodman ratio for $d = 16\,\text{mm}$:

$$0.28 < 0.33 \quad \text{(Passes safety factor)}$$

4.2 Graphs and Explanation

4.2.1 Torque and Power Requirements

This section presents the torque and power graphs along with their corresponding explanations.

I. **Torque Required**

The torque and power required for the cam mechanism are obtained from the torque versus displacement graph. This chart describes the variations in torque as the spring displacement moves from 0 mm to 100 mm. The graph contains three curves. Each of the curves represents horizontal shaft torque (blue), vertical shaft torque (red), and the overall system torque (black). As the shaft remains engaged continuously, the torque on the horizontal shaft continuously rises with distance due to spring force and friction. The vertical shaft torque remains at 0 until the spring displacement reaches 11.6 mm, at the point when the vertical spring force becomes equal to the opposing weight force of 8.83 N. The dashed vertical line on the graph represents this position. After 11.6 mm, vertical torque starts to rise because of the increased height of compression in the spring and some friction.

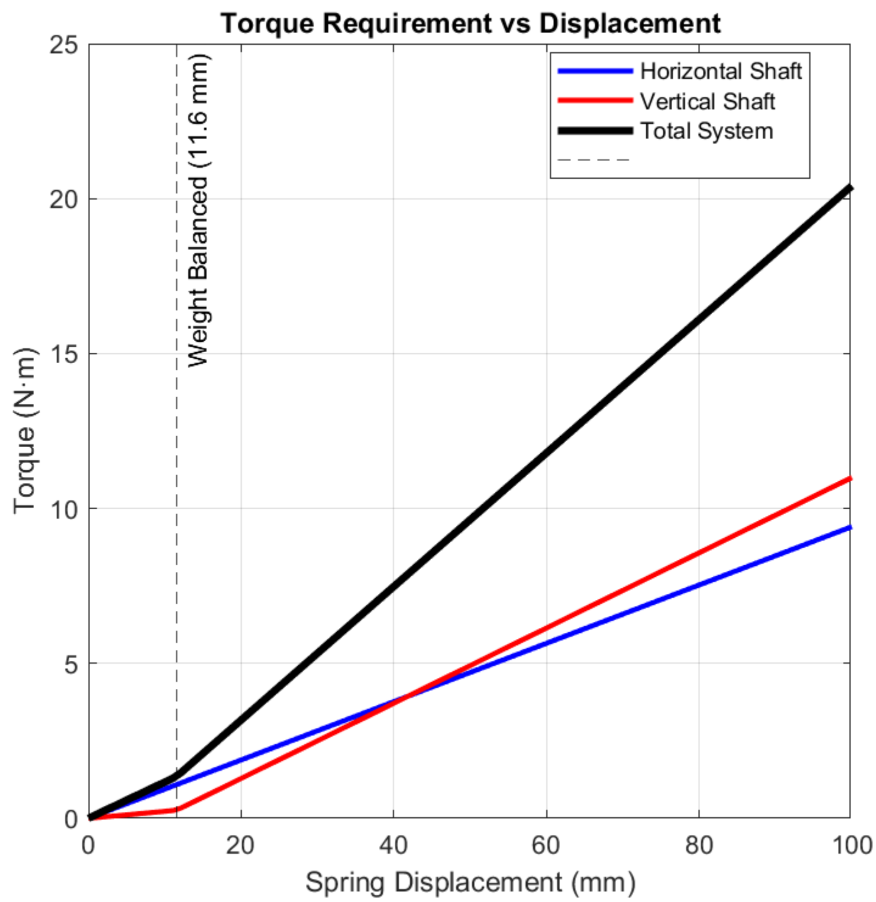

Figure 4.5: Torque required

For combined shafts, the overall torque starts at 1.5 N·m with no displacement because of friction in the horizontal cam and reaches a maximum of 13.8 N·m at 100 mm of displacement. To put it differently, the motor powering the system must be rated for a minimum of 13.8 N·m of torque under maximum load. The torque and power needs would be under 11.6 mm of movement when only the horizontal cam was activated.

II. Power Required

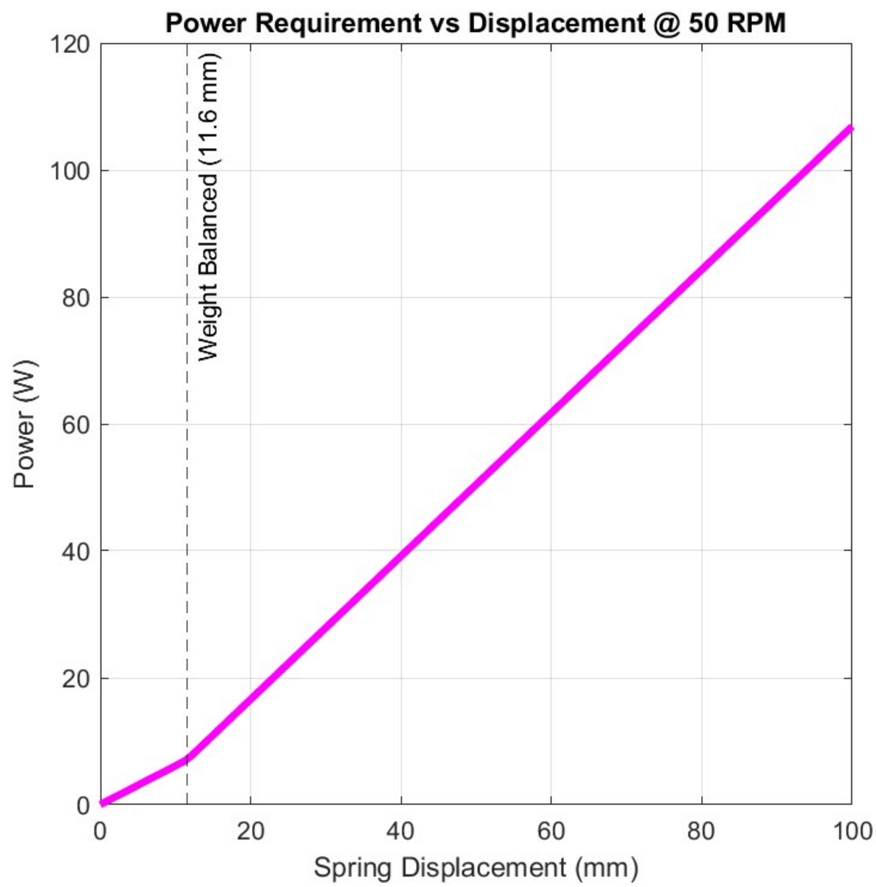

Figure 4.6: Power required

The Power vs. Displacement graph, calculated at 50 RPM, had the same basic shape and profile as the torque curve given that power is the torque multiplied by angular velocity. The demand for power starts with 7 W when the horizontal cam was active. The power demanded increased quickly once the vertical cam became engaged peaking at 72 W at 100 mm of displacement. This indicates that the motor must be provided with at least 72 W of power at 100 mm displacement (max).

One main takeaway from this analysis is that operating the system just below the boundary displacement of 11.6 mm will help ensure that power usage is kept to a minimum.Exceeding this displacement boundary causes an increase in power and torque requirements. The motor should be rated for 13.8 N·m of torque and 72 W of power at 100% load operational characteristics. A safety factor of 1.2-1.5 is preferred to account. By minimizing the friction, adjusting the stiffness of horizontal and vertical spring to balance the condition can increase efficiency.

In conclusion, this analysis shows that the cam mechanism requires a torque of 13.8 N·m and a power of 72 W at full displacement. The 11.6 mm point represents a critical transition that affects the characteristics of loads seen by the cam mechanism. Properly selecting the motor and design of the control system, are critical to ensuring a system that operates efficiently and has reliable performance under realistic operating condition variations.

4.2.2 Displacement, velocity, acceleration and jerk Diagrams

I. **Vertical cam**

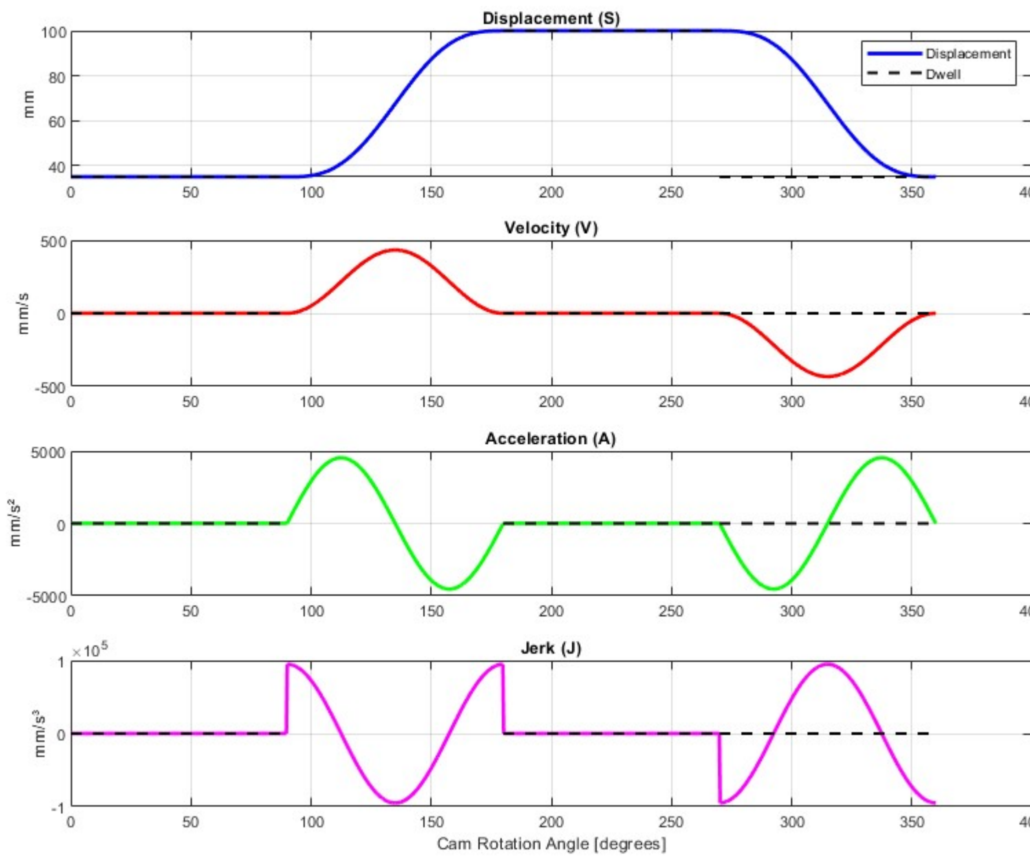

Figure 4.7: SVAJ vertical Cam

SVAJ stands for four quantities: Displacement, Velocity, Acceleration, and Jerk. The overall phenomenon of the cam-follower system's motion simulation for the full 360-

degree rotation of the cam may be seen in the graph. The displacement curve starts at 0 mm to indicate that there was no initial motion. The rise continues from 0 to 150 degrees to 100 mm. It stays steady at 100 mm during 150 to 300 degrees, which is known as the dwell and denoted by the dotted black lines.

The velocity waveform begins from 0 mm/s, increases to a positive peak during the up stroke, falls back to 0 mm/s during the dwell period, falls to a negative value throughout the down stroke, and reaches 0 mm/s once again at the end of a cycle. The dotted lines are when the velocity is constant for the dwell phase. The acceleration curve starts from a positive maximum, it first zero and breaks, becoming a negative value, then zero again and it breaks as negative again and finally it tends to zero during the increasing phase . It is 0 mm/s^2 in the dwell phase, and then follows the description above scaled in reverse for the descent phase also, yet with an acceleration of 0 mm/s^2 in the dwell, as denoted by the broken lines. The rate of change of acceleration begins at 0 mm/s^3 which is known as jerk curve.It first increases to a positive peak ,decreases negatively ,and then once more returns to zero throughout the raising phase.

We can see there is absence of load and smooth trasitions in all four curves that indicate cam follower has Simple Harmonic Motion (SHM) profile.The SAVJ diagram is important as this kind of design method helps in minimizing vibrations ,mechanical wear ,and noise with aim of improving system performance and it's lifespan.

II. **Off-Set cam (Horizontal cam)**

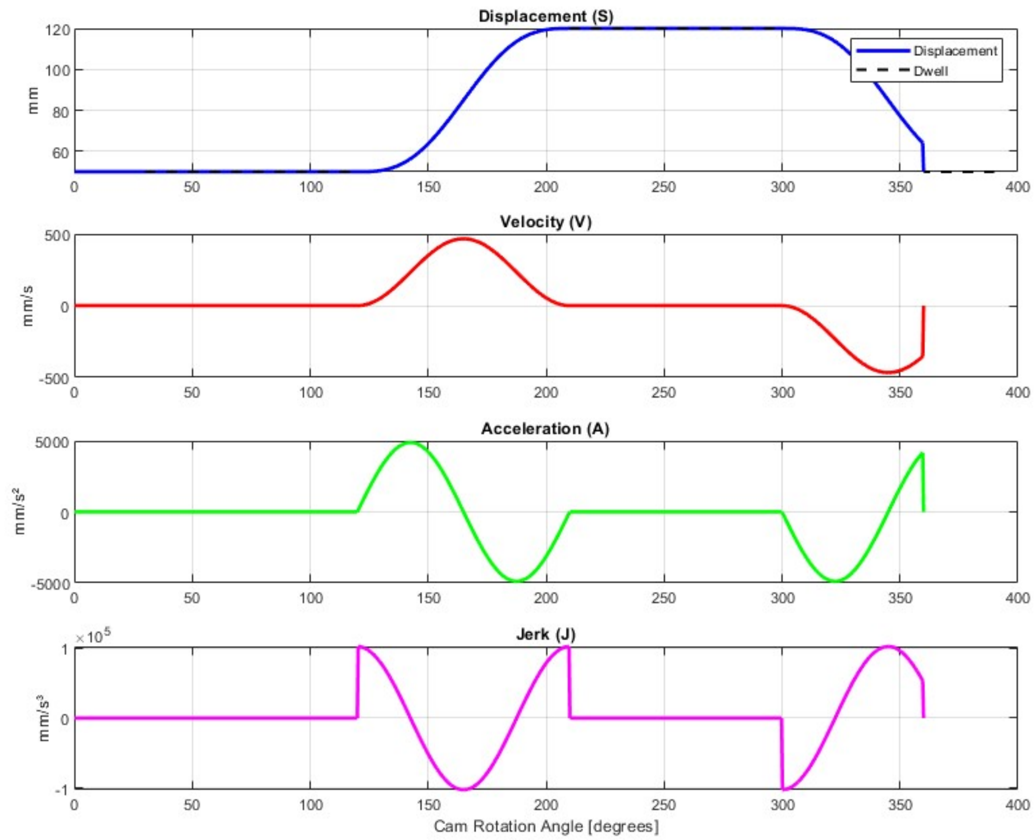

Figure 4.8: SVAJ Off-set Cam

SVAJ stands for four quantities: Displacement, Velocity, Acceleration, and Jerk. The overall phenomenon of the cam-follower system's motion simulation for the full 360-degree rotation of the cam may be seen in the graph.

The displacement curve starts at 0 mm to indicate that there was no initial motion. The rise continues from 0 to 150 degrees to 100 mm. It stays steady at 100 mm during 150 to 300 degrees, which is known as the dwell and denoted by the dotted black lines . The velocity waveform begins from 0 mm/s, increases to a positive peak during the up stroke, falls back to 0 mm/s during the dwell period, falls to a negative value throughout the down stroke, and reaches 0 mm/s once again at the end of a cycle. The dotted lines is when the velocity is constant for the dwell phase.

The acceleration curve starts from a positive maximum, it first zero and breaks, becoming a negative value, then zero again and it breaks as negative again and finally it tends to zero during the increasing phase . It is 0 mm/s² in the dwell phase, and then follows the description above scaled in reverse for the descent phase also, yet with an acceleration of 0 mm/s² in the dwell, as denoted by the broken lines. The rate of change of acceleration begins at 0 mm/s³ which is known as jerk curve.It first increases to a positive peak ,decreases negatively , and then once more returns to zero throughout the raising phase .

We can see there is absence of load and smooth trasitions in all four curves that indicate cam follower has Simple Harmonic Motion (SHM) profile.The SAVJ diagram is important as this kind of design method helps in minimizing vibrations ,mechanical wear ,and noise with aim of improving system performance and it's lifespan.

III. **Force exerted by Cam**

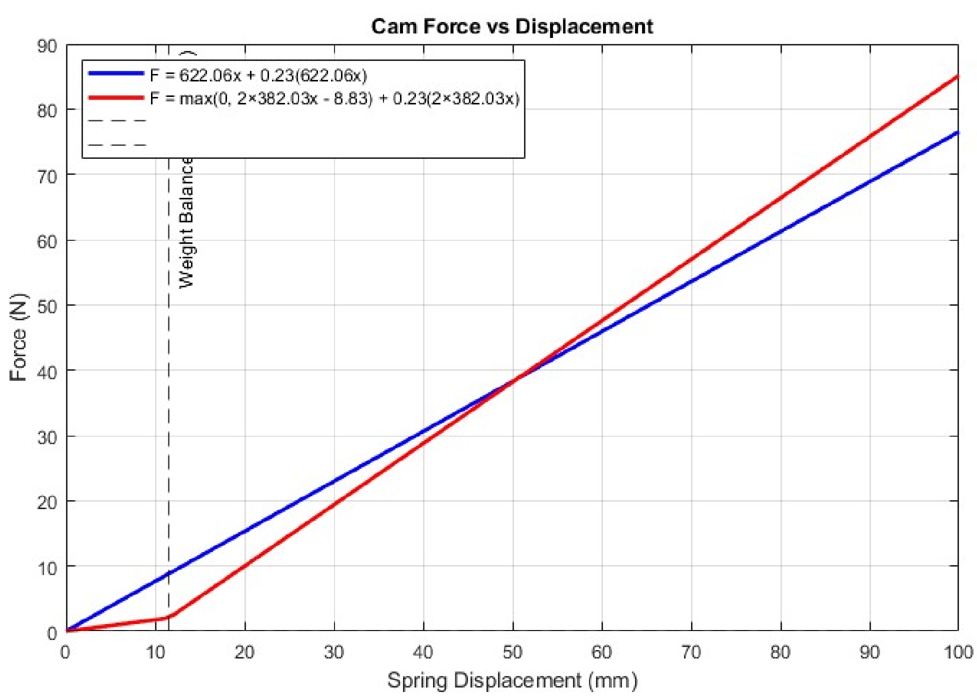

Figure 4.9: Force exerted

In the Force vs. Displacement graph for the cam mechanism presented in the thesis, it plots the forces exerted by horizontal and vertical cams while spring displacement goes from 0-100 mm. The horizontal cam, shown in blue in the plot is defined by the $F_H = 622.06x + 0.23 \times (622.06x)$, where the first term refers to the spring force and the second term is an increase of 23% due to friction. This force starts at 0 N and steadily increases at a constant rate with increasing displacement. You can recall that the cam engages immediately starting at 0 mm and displays a linear response to the applied spring force with stiffness $k_h = 622.06\,\mathrm{N/m}$. The vertical cam force is shown in red,$F = \max(0, 2x \times 382.03 - 8.83) + 0.23\,(2 \times 382.03)$. The vertical cam force $F_v = \max(0, 2 \times 382.03x - 8.83) + 0.23\,(382.03)$, is a little different since this force stays at zero (0 N) until 11.6 mm displacement—this is the point at which the weight of 8.83 N force is overcome by the vertical spring force, represented in the graph by the dashed vertical line. After the vertical cam engagement displacement of 11.6 mm and the subsequent increase in the vertical spring displacement, a linear cam force increment is observed as the springs compress and a incremental offset from friction is active.

The maximum displacement of 100 mm is where the horizontal cam force reaches about 100 N and the vertical cam force is approximately 80 N. The data shows that the horizontal cam is engaged continuously, and the vertical cam is active only after 11.6 mm displacement is surpassed. The friction coefficient of 0.23 adds significantly to both horizontal and vertical forces, indicating that material choice and surface treatments (e.g., lubrication or surface treatments) should contribute to reducing overall force requirements. The fact that the vertical cam is inactive below 11.6 mm could also contribute to less wear in lower load conditions of the operation.

The fact that the spring constants can be changed to design advantage, means that if the horizontal spring constant is changed to have higher stiffness the load will make the system more responsive to the cam position however the amount of generated static force loss of friction increases. Similarly, there is an advantage to changing the vertical spring constant to reduce peak forces; however, it also realized displacement change to the vertical loaded cam beyond 11.6 mm. In terms of design, the 11.6 mm point becomes a relevant point for future energy efficient controls because operating at or near this point may infer substantially less load on the system/energy use. The force profile regarding torque and power requirements is critical in selecting motor size.

In brief, the horizontal cam makes continuous contact with the load, while the vertical cam only makes contact after the load is displaced by greater than 11.6 mm. Friction accounted for 23% of the total force. These findings reaffirm the weight-balance calculation for the system, and provide the information to further develop dynamic loads for cam mechanism design.

4.3 Fabrication:

The main fabricated parts are considered the frame, cams, shafts, and sliding arrangement in our PP machine. These are fabricated and assembled properly for the final operation of the machine.

4.3.1 Cam:

The cams used in horizontal shaft are responsible for the lifting of ink and printing in the substrate. These both cam 1 and cam 2 are same but placed in different position for their particular use. These cams are designed in Solidworks and later it was fabricated by using the CNC-Machine.

Figure 4.10: Vertically aligned Cam 1

Figure 4.10 is the cam that is fabricated for our prototype, responsible for guiding the Slider arrangement (including Silicon Pad), taking ink from the cliche (engraved design) required for the printing. This is the cam located near the gear combination. After our study, we found that the dwell time of this cam, for press, for our currently available displacement is too high. The marked range must be the dwell time so that the silicon pad is properly pressed and released in time.

Figure 4.11 is the cam that is responsible for printing to the substrate. This is located near 1, aligned 180° to it. After certain time of function of cam 1, the function of cam 2 starts. In simple words, it is responsible for the pressing of the silicon pad after the ink is carried with pad.

Figure 4.11: Vertically aligned Cam 2

The cams are fabricated from 6 mm thick, AISI 1018 Mild Steel. The cam profile is defined by a maximum radius of cam is 100 mm and minimum radius is 35 mm, to provide required lift during operation. In addition, a center hole with a diameter of 24.8 mm accommodates the shaft with an appropriate fit. The cam motion cycle is equally divided into three equal angular phases: dwell, return and rise, each occupying an angular span of 90°.

RSOC (Right Side Offset Cam) as shown in figure 4.12, was made by 7" × 7" mild steel plate of 6mm thickness. The chosen material provides a good compromise between machinability, strength and costs, being especially suitable for cam components which have to be durable under cyclical loading but where they aren't subjected to high mechanical stresses.

Figure 4.12: RSOC cam

CNC laser cutting ensures a high level of tolerances: The special cam profile was precision machined using CNC laser cutting technology and allows for very accurate and precise movements. The CNC laser cut allowed highly accurate cam profile shape, which was particularly important for the double harmonic rise and fall motion periods in this design. This approach also preserved small clearances between a base circle (35 mm in radius) and a maximum profile radius of 105 mm, which are necessary to retain smooth velocity and acceleration behaviour of harmonic cam profile.

The 4 motion phases are evenly distributed over 360° and are 90° dwell-in, 90° full rise (double harmonic), 90° dwell-out, 90° full fall (double harmonic) on the continuous cam profile. These motions have been programmed as separate phases into the cam profile, based on computer-developed Solidworks data, and directly into the CNC cutting machine to assure true motions.

4.3.2 Vertical Shaft:

Figure 4.13: Vertical Shaft

A shaft (Mild Steel AISI 1018) was fabricated using lathe machine with the respective dimensions included in the design. The machining process was carried out with proper dimensions and tolerances, resulting in smooth cylinderical surface finish. The bevel gear was also attached to the shaft responsible for the power transmission. The pinhole was precisely designed and drilled for the attachment of the bushes (Cam). Overall, the fabrication met the design requirements, demonstrating the effectiveness of the chosen machining techniques for mild steel.

4.3.3 Horizontal Shaft

Figure 4.14: Horizontal Shaft

A cylindrical mild steel rod (AISI 1018) with an initial diameter greater than 28.0 mm and a length slightly longer than 340.0 mm was prepared to account for machining allowances. It was then mounted into the lathe (step turning was performed), to achieve 25 mm diameter (right section) of length 340 mm, where the cam will be attached. Two holes were drilled into the middle section to attach the bush. A gear was also attached to the shaft so that it can be meshed with another vertical shaft gear. Also, 28 mm diameter with length 76 mm (for locking the bearing) and 21 mm with length 34 mm (for gear) was performed in lathe.

4.3.4 Final Assembly

Figure 4.15: Pad Printing Machine

This is our final assembled prototype of Pad Printing Machine. The fabricated cam, shafts, Slider, gears, bearings, and frame are all assembled to give this final output.

4.4 Motion Result

4.4.1 Expected Working

The cams attached to the horizontal shaft pushes the spring and thus the pad arrangement vertically downward. This results into the pad picking up the ink from the engraved design in the cliche, which is a flat plate which contains the engraved design. During this contact with the cliché, there must not be any horizontal movement because even a small movement will lead to smearing of the ink. Once the ink is collected, the spring pushes the arrangement vertically upward as the cam rotates and disengages. At this stage, a cam located on a vertical shaft is responsible for moving the pad horizontally to the substrate. Upon reaching this position, the pad arrangement stops for certain period due to the dwell angle of the horizontal cam. Then, another cam mounted on the horizontal shaft which is above the printing section pushes the pad vertically downward. This action transfers the ink onto the substrate. During this process, no horizontal movement should occur. This is ensured by the dwell angle of the cam which is used for the vertical movement of the pad. The pad remains in contact with the substrate for a duration determined by the dwell angle of the vertical motion cam. After the dwell angle is completed, the spring pushes the pad section vertically upward. The spring pulls the arrangement back horizontally to its starting point. The pad is now ready to pick up ink, again repeating the cycle.The expected motion of the pad section is inverted U-motion, means pad picking up the ink from cliche, goes vertically upward and moves to the substrate section horizontally and moves vertically downwards for the printing.

4.4.2 Obtained Working

I. Nature of motion of pad:

   Testing showed that the performance varied from the ideal working. This resulted in blurry and smudged prints. When the pad contacted with the cliche, it remained stationary for a certain period. However, it still remained in contact with the cliche even after it started to move horizontally. The case is the same for the printing section, too. This resulted in an inclined path instead of lifting vertically and moving horizontally. This occurred because the pad compresses and deforms slightly, maintaining contact with the cliche longer than required. Then the pad is pushed downward along with the slight horizontal movement in it in the printing section, instead of horizontal to vertical i.e., 90 degrees. After printing, pad moves slightly horizontally while going vertically upward, resulting in the smudged print.

II. Slider Displacement:

While analyzing the performance of the pad printing machine, both horizontal and vertical displacement were evaluated. 70 mm of horizontal displacement was supposed to be produced. But the slider moved only about 50 mm. It could be because of various reasons, like binding or misalignment in the guide rails. Also, the cam in vertical shaft has an offset of 25 mm to the right and a 90-degree dwell time, but if the angle between the follower's motion and the cam surface is too large, it may not push the slider into it's ideal position. This results in limited horizontal movement. Another reason could also be due to loosen parts. The cam in the horizontal shaft was designed to move the slider 70 mm up and down, but it only moved 50 mm. This could be due to various reasons, like mechanical play and backlash. For example, excessive clearance in the bearings, linkages, or slider guide rail. This may result in the improper motion transfer. Assembly errors or vibrations in the machine parts could also lead to restriction of both vertical and horizontal motion.

To fix these issues, some adjustments can be made. Checking and realigning the guide rails to eliminate any binding or misalignment, adjusting the cam's offset or pressure angle might also ensure it pushes the slider more effectively sideways. To reduce the play, the loose parts can be tightened properly. Also, it can be reduced by using pre-loaded or adjustable bearings. The addition of dampers can also help to reduce vibrations. Making these changes may help in the appropriate slider displacement, ensuring the machine run smoothly.

4.4.3 Graphical representation of Motion Profile

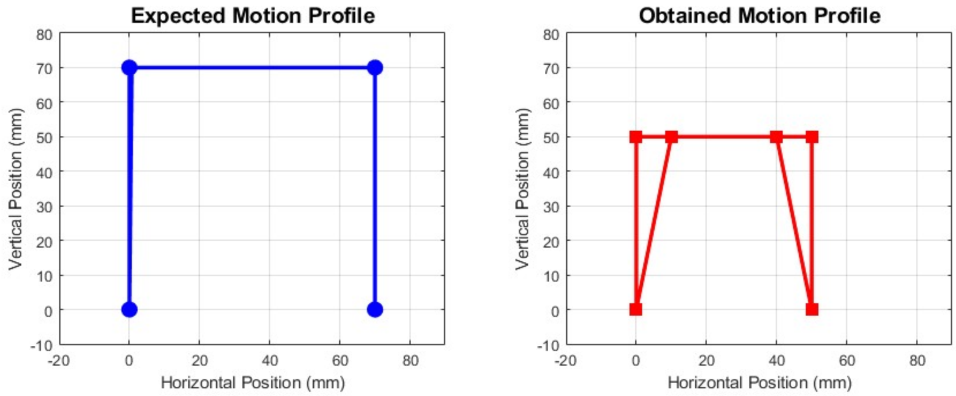

Figure 4.16: Graphical representation of Motion Profile

I. Expected outcome:

Stage 1: At this stage(or coordinate), the silicon pad used to stamp ink is at position (0,0) or origin.

Stage 2: At this stage,the pad travels vertically from (0,0) to (0,70). There is no horizontal displacement.

Stage 3: At this stage,the pad travels horizontally from (0,70) to (70,70) with no vertical displacement.

Stage 4: This is the final stage of printing.At this stage, the pad travels vertically downwards from (70,70) to (70,0).

In this cyclic movement,the mechanism generates an inverted U(sharp corner) motion profile ideally.

(Note: All Units in mm.)

II. Obtained outcome:

The motion profile graph of obtained motion shows that, instead of a straight vertical line, our machine produced a certain slope in 2nd and 4th stages. This was due to the lack of perfect alignment in dwell angle of three individual cams.By reducing the dwell angle of cams placed in horizontal shaft from 90 degrees to 66 degrees (for both dwell in and dwell out), this issue can be solved in further prototype.

The second error is the vertical displacement error where, we achieved 50 mm maximum vertical displacement. This was due to the low stiffness of springs used. This can easily be solved by using a spring of higher stiffness such that the spring pushes upwards upto 70 mm vertical displacement.

Another error was due to manufacturing inaccuracy. This can be solved by using accurate manufacturing method, as well as experienced manpower.

Figure 4.17: Print result

4.4.4 Causes of Deviation in Expected and Obtained Working

The primary problem was the mismatch in dwell times between the cams controlling vertical and horizontal motion which lead to improper timing. The dwell time of the horizontal motion cam ends prematurely. This initiates horizontal motion before the pad has fully lifted from the cliché or substrate, causing the pad to slide and smear the ink. Similarly, the vertical motion cam's dwell time also ended prematurely thus, compounding the issue.

Vibrations were also observed during operation which contributed to the blurriness by causing the pad to shake during ink pickup and transfer. Since the actual stiffness of springs differed from the computed values, the use of non-standard springs with ambiguous stiffness ratings most likely contributed to the difference in dwell periods. Additionally, the plate in the spring-pad arrangement is also wide. This plate should not contact the cam responsible for vertical motion during the horizontal movement of the pad. Its oversized design resulted in unintended contact which introduced unwanted vertical movement. These factors are the primary reasons for the deviation from the ideal inverted U motion, resulting in compromised print quality.

4.4.5 Necessary adjustments to be made

The dwell angles of the cams must be recalibrated to ensure proper synchronization. For the cam controlling horizontal motion, set at a dwell angle of 90 degrees, the parameters for other cams should be adjusted. The dwell angle of the cam that engages to push the pad vertically downward to pick up the pattern should be adjusted to 66 degrees. This angle is currently at 90 degrees. Similarly, the dwell angle of the cam responsible for vertical motion for printing needs to be set to 30 degrees. Its current value is also 90 degrees. These adjustments will ensure the pad remains stationary horizontally during critical contact phases. As a result it produces desired vertical motions for the inverted U path.

The vibrations observed during operation must also be addressed, possibly by using dampening materials or tightening loose components to stabilize the system. The plate in the spring-pad arrangement should also be redesigned to a optimum size thus, preventing it from contacting the vertical motion cam when the horizontal motion cam engages. This maintains purely horizontal motion when required. Implementing these changes will ensure the precise inverted U motion. This will produce clean, sharp prints without smudging, meeting the project's performance objectives.

4.5 Limitations

1. The use of a single non-programmable motor limited the flexibility of the machine in terms of independent control over the x and y axes.

2. The machine had limitations in handling very large or highly irregularly shaped substrates due to the constraints of the cam mechanism and silicone pad size.

3. The project did not include an inbuilt curing system (e.g., UV or heat curing). Thus, the printed substrates might require manual or external curing, affecting production efficiency.

4. Due to mechanical nature of cam , achieving high precision in ink transfer was challenging. Also, the potential for wear and tear over time produced vibration which might produce blurred prints.

5. The project had budget constraints time limitations. This impacted the choice of materials, components.

CHAPTER 5: CONCLUSION AND FUTURE ENHANCEMENT

The design and fabrication of the pad printing machine mainly consist of a double cam mechanism on the horizontal shaft. These cams are responsible for the vertical displacement of the pad and an offset cam on the vertical shaft, which is responsible for the horizontal displacement of the slider. The main focus while designing the mechanism was to achieve synchronized vertical and horizontal movement. This coordinated movement transfers ink from the cliché to the substrate via the pad, tracing an inverted U motion. The primary objective while designing the machine was to provide an economic and feasible printing solution for uneven surfaces for small-scale businesses. However, there were deviations in the functioning of the machine from expected functioning. This resulted in blurry and smudged prints. This was mainly due to mismatched dwell times of the cams. The dwell time of the horizontal cam ended prematurely. As a result, it caused the pad to slide across the cliché or substrate. Similarly, the vertical cam's dwell also terminated too early, causing an inclined motion path instead of the desired vertical lift. As a result, it formed a deviated cycle of motion from the inverted U shape. Furthermore, vibrations during operation also caused print blurriness. Another reason might be the oversized plate in the spring-pad arrangement, which interfered with the vertical motion of the cam. This in turn introduced unwanted vertical movement during horizontal motion phases. The slider displacement analysis showed the cam achieving only 50 mm of horizontal movement and 80 mm of vertical movement instead of the designed 70 mm and 100 mm. This might be due to binding in the guide rails or a larger pressure angle from the offset cam. Other reasons might be mechanical play, backlash, clearance in bearings or guides, and vibrations. Despite these issues, synchronized vertical and horizontal motion was obtained using a single power source.

The calculated torque required to rotate the camshaft and vertical shaft driving the offset cam mechanism was 13.82 Nm. The shaft was designed to provide 15 Nm of torque which is higher than the required torque. The data and graphs obtained align with theoretical trends, validating the design approach. Using proper design tools and guidelines, the pad printing machine was successfully designed, analyzed, and fabricated. The machine was economic as it was fabricated using locally available materials. This also ensured that it can be easily replicated. The use of a single non programmable motor kept the system inexpensive and straightforward to fabricate, resulting in a simple, frugal, and modular pad printing machine, despite various operational challenges.

Following enhancement and adjustment can be made in the future to enhance the machine performance and meet all the objectives.

1. Dwell angles of the cams should be recalibrated for better synchronization. This could be done by adjusting the dwell angle of the vertical motion cam responsible for ink pickup to 66 degrees (from 90 degrees). Similarly, the dwell angle of the cam responsible for printing can be adjusted to 30 degrees (from 90 degrees) while extending the horizontal motion cam's dwell duration. Doing so will prevent premature movement and achieve the intended inverted U motion.

2. Operational vibrations can be minimized by using dampening materials or tightening loose components. This will stabilize the system and reduce blurriness during ink pickup and transfer.

3. The plate in the spring-pad arrangement should be redesigned to an optimal size to prevent contact with the vertical motion cam during horizontal motion. This adjustment will ensure pure horizontal movement when required.

4. Guide rails could be inspected and realigned to solve binding or misalignment of components. Also, the cam's 25 mm offset or pressure angle could be adjusted to achieve the full 75 mm horizontal displacement for the slider.

5. Mechanical play and backlash could be reduced by tightening loose parts, using

preloaded bearings or adjustable bushings to minimize excessive clearance, and conducting a detailed assembly review to correct errors.

6. Vibration dampers can be installed, which could decrease the impact of operational vibrations on both horizontal and vertical displacements.

7. Etching quality could be increased.

8. Materials like aluminium could be used to reduce slider weight. Proper selection of etching plate, doctor blade, pad, and ink could increase print quality.

9. This machine can be scaled up to larger size, which will be capable of printing on larger surfaces.

The following challenges were faced during the design and fabrication of the pad printing machine:

1. Welding Issues: Lack of proper welding equipment, proper electrode selection, and welding skills led to difficult and inconsistent welds.

2. No Surface Treatment Tools: Surface finishing is important to reduce friction and proper surface finishing. This is also important for proper clearance and fit. Unavailability of surface grinders, milling machines, or lathes affected component smoothness and accuracy.

3. Non-Standard Springs: Springs were purchased from the local market. These locally purchased springs lacked known ratings, with material, type, and durability uncertain after lab testing.

4. Imported Printing Pad: The printing pad wasn't locally available. It was imported from India. The durability of the pad was also unknown.

5. Ink Sourcing Difficulty: Suitable pad printing ink was hard to find locally; this impacted on proper printing quality and consistency.

6. Limited Precision Tools: Absence of tools like dial indicators or micrometers made tolerance checks difficult.

7. Uncontrolled Testing Environment: Lack of a controlled setup led to testing difficulty.

8. Material Sourcing Challenges: Finding affordable, local materials with required strength was difficult.

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APPENDIX A

SPRING-CAM MECHANISM FORCE

A1.1 System Parameters

Spring stiffness (k_2)	382.0256 N/m
Spring deflection (x)	0.1 m
Weight of arrangement (W)	9.76 N

A1.2 Calculation Process

1. **Force Exerted by One Spring (Hooke's Law)**

$$\begin{aligned} F_{\text{spring},2} &= k_2 \cdot x \\ &= 382.0256\,\text{N/m} \times 0.1\,\text{m} \\ &= 38.20256\,\text{N} \end{aligned}$$

2. **Total Spring Force (Two Springs)**

$$\begin{aligned} F_{\text{spring,total},2} &= 2 \cdot F_{\text{spring},2} \\ &= 2 \times 38.20256\,\text{N} \\ &= 76.40512\,\text{N} \end{aligned}$$

3. **Net Upward Force When Springs Push Upward**

$$\begin{aligned} F_{\text{up},2} &= F_{\text{spring,total},2} - W \\ &= 76.40512\,\text{N} - 9.76\,\text{N} \\ &= 66.64512\,\text{N} \end{aligned}$$

4. **Cam Force Function (vs. Displacement)**

$$\begin{aligned} F_{\text{cam},2}(x) &= 2 \cdot k_2 \cdot x - W \\ &= 2 \times 382.0256 \times x - 9.76 \\ &= 764.0512x - 9.76\,\text{N} \end{aligned}$$

5. **Maximum Force (At $x = 0.1\,\text{m}$)**

$$\begin{aligned} F_{\text{cam,max},2} &= 2 \cdot 382.0256 \cdot 0.1 - 9.76 \\ &= 76.40512\,\text{N} - 9.76\,\text{N} \\ &= 66.64512\,\text{N} \end{aligned}$$

6. **Minimum Force (At $x = 0\,\text{m}$)**

$$F_{\text{cam,min}} = 0\,\text{N}$$

(No cam force needed - weight causes downward motion)

CAM FORCE CALCULATION

Force Required for Spring Extension

The force (F) required to extend the spring is given by Hooke's Law:

$$F = k \cdot x \tag{1.55}$$

Where:

- F = Force exerted by the cam (N)
- k = Spring constant = 622.06 N/m
- x = Spring displacement (m)

Calculation for 70 mm Extension

For $x = 70 \text{ mm} = 0.07 \text{ m}$:

$$F = 622.06 \text{ N/m} \times 0.07 \text{ m} = 43.54 \text{ N} \tag{1.56}$$

General Force-Displacement Equation

The force as a function of displacement is:

$$F(x) = 622.06 \cdot x \tag{1.57}$$

TORQUE CALCULATION

Given Parameters

$$\text{Angular acceleration } (\alpha) = 10.4\,\text{rad/s}^2$$
$$\text{Friction coefficient } (\mu) = 0.23$$
$$\text{Horizontal shaft:}$$
$$\text{Cam radius } (r_h) = 0.1\,\text{m}$$
$$\text{Spring force } (F_h) = 66.65\,\text{N}$$
$$\text{Vertical shaft:}$$
$$\text{Cam radius } (r_v) = 0.105\,\text{m}$$
$$\text{Spring force } (F_v) = 43.54\,\text{N}$$

Torque Components

1. Horizontal Shaft

$$\tau_{\text{fric},h} = \mu F_h r_h = 0.23 \times 66.65 \times 0.1 = 1.533\,\text{N}\cdot\text{m}$$
$$\tau_{\text{spring},h} = F_h r_h = 66.65 \times 0.1 = 6.665\,\text{N}\cdot\text{m}$$
$$\tau_{\text{inertia},h} = J_h \alpha = 1.90 \times 10^{-4} \times 10.4 = 0.00198\,\text{N}\cdot\text{m}$$
$$\tau_{\text{total},h} = 1.533 + 6.665 + 0.00198 = 8.20\,\text{N}\cdot\text{m}$$

2. Vertical Shaft

$$
\begin{aligned}
\tau_{\text{fric},v} &= \mu F_v r_v = 0.23 \times 43.54 \times 0.105 = 1.052\,\text{N}\cdot\text{m} \\
\tau_{\text{spring},v} &= F_v r_v = 43.54 \times 0.105 = 4.572\,\text{N}\cdot\text{m} \\
\tau_{\text{inertia},v} &= J_v \alpha = 6.79 \times 10^{-5} \times 10.4 = 0.000706\,\text{N}\cdot\text{m} \\
\tau_{\text{total},v} &= 1.052 + 4.572 + 0.000706 = 5.62\,\text{N}\cdot\text{m}
\end{aligned}
$$

System Requirement

$$
\tau_{\text{system}} = \tau_{\text{total},h} + \tau_{\text{total},v} = 8.20 + 5.62 = 13.82\,\text{N}\cdot\text{m}
$$

SHAFT BENDING MOMENT ANALYSIS WITH FRICTION

Case 1: Cam1 Active

Reaction Forces

$$
\begin{aligned}
&\sum F_y = 0: \\
R_A + R_B &= F_s - W_c(\text{Cam1}) - W_c(\text{Cam2}) + W_{\text{shaft}} \\
&= 67.57 - 8.83 - 8.83 + 18.25 = 68.16\,\text{N}
\end{aligned}
$$

Moment Balance About B1

$$\sum M_{B1} = 0:$$
$$F_s \times 0.122 - W_c \times 0.122 - W_c \times 0.188 + M_f = R_B \times 0.284$$
$$67.57 \times 0.122 - 8.83 \times 0.122 - 8.83 \times 0.188 + 1.554 = 0.284 R_B$$
$$R_B = 24.84\,\text{N}, \quad R_A = 43.32\,\text{N}$$

Bending Moments

$$M_{B1} = 43.32 \times 0.127 - \frac{44.4 \times 0.127^2}{2} = 5.14\,\text{N·m}$$
$$M_{\text{Cam1}} = 9.41\,\text{N·m} + 1.554\,\text{N·m} = 10.96\,\text{N·m}$$
$$M_{\text{Cam2}} = 15.33\,\text{N·m}$$

Case 2: Cam2 Active

Reaction Forces

$$R_A + R_B = 68.16\,\text{N}$$
$$\sum M_{B1} = 0:$$
$$-8.83 \times 0.122 + 67.57 \times 0.188 - 8.83 \times 0.188 + 1.554 = 0.284 R_B$$
$$R_B = 40.53\,\text{N}, \quad R_A = 27.63\,\text{N}$$

Bending Moments

$$
\begin{aligned}
M_{B1} &= 3.15\,\text{N·m} \\
M_{\text{Cam1}} &= 5.50\,\text{N·m} \\
M_{\text{Cam2}} &= 5.92\,\text{N·m} + 1.554\,\text{N·m} = 7.47\,\text{N·m}
\end{aligned}
$$

Summary

Case	Moment at Cam1	Moment at Cam2	Max Moment
Cam1 Active	10.96 N·m	15.33 N·m	15.33 N·m
Cam2 Active	5.50 N·m	7.47 N·m	7.47 N·m

COMPLETE BENDING MOMENT ANALYSIS

System Parameters

- Shaft Length: 16.9 cm (0.169 m)
- Bearing Positions:
  - B1: 3.8 cm (0.038 m) from top
  - B2: 13.6 cm (0.136 m) from top
- Cam Position: 16.9 cm (0.169 m) from top
- Forces:
  - Vertical: Shaft weight (52.2 N/m distributed) + Cam weight (8.63 N concen-

trated)

- Horizontal: Spring force (43.54 N concentrated at cam)

Bending Moment Calculations

Vertical Plane (Mz)

$$
\begin{aligned}
\text{Region 1 } (0 \leq x < 0.038\text{m}) &: M_z(x) = R_{Az}x - \frac{wx^2}{2} \\
\text{At } x = 0.038\text{m} &: 1.75 \times 0.038 - \frac{52.2 \times 0.038^2}{2} = 0.028\,\text{N·m} \\
\text{Region 2 } (0.038 \leq x < 0.136\text{m}) &: M_z(x) = R_{Az}x - \frac{wx^2}{2} + R_{Bz}(x - 0.038) \\
\text{At } x = 0.136\text{m} &: 1.75 \times 0.136 - \frac{52.2 \times 0.136^2}{2} + 15.71 \times 0.098 = 1.30\,\text{N·m} \\
\text{Region 3 } (0.136 \leq x \leq 0.169\text{m}) &: M_z(x) = R_{Az}x - \frac{wx^2}{2} + R_{Bz}(x - 0.038) - W_c(x - 0.136) \\
\text{At } x = 0.169\text{m} &: 1.75 \times 0.169 - \frac{52.2 \times 0.169^2}{2} + 15.71 \times 0.131 - 8.63 \times 0.033 = 1.33\,\text{N·}
\end{aligned}
$$

Horizontal Plane (My)

$$
\begin{aligned}
\text{Region 1} &: M_y(x) = R_{Ay}x \\
\text{At } x = 0.038\text{m} &: -14.67 \times 0.038 = -0.56\,\text{N·m} \\
\text{Region 2} &: M_y(x) = R_{Ay}x + R_{By}(x - 0.038) \\
\text{At } x = 0.136\text{m} &: -14.67 \times 0.136 + 58.21 \times 0.098 = 3.71\,\text{N·m} \\
\text{Region 3} &: M_y(x) = R_{Ay}x + R_{By}(x - 0.038) - F_s(x - 0.136) \\
\text{At } x = 0.169\text{m} &: -14.67 \times 0.169 + 58.21 \times 0.131 - 43.54 \times 0.033 = 3.70\,\text{N·m}
\end{aligned}
$$

Location	Vertical Moment (N·m)	Horizontal Moment (N·m)	Total Moment (N·m)
B1 (3.8 cm)	0.028	-0.56	0.56
B2 (13.6 cm)	1.30	3.71	3.93
Cam (16.9 cm)	1.33	3.70	3.93

Results

CALCULATIONS

Step 1: Mean and Alternating Moments

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{15.33 + 5.50}{2} = 10.42\,\text{N·m}$$

$$M_a = \frac{M_{max} - M_{min}}{2} = \frac{15.33 - 5.50}{2} = 4.92\,\text{N·m}$$

Step 2: Rearrange Goodman Criterion

$$\frac{n_f \cdot 32M_a}{\pi d^3 \sigma_e} + \frac{n_f \cdot 32M_m}{\pi d^3 \sigma_{ult}} + \frac{n_f \cdot 16T}{\pi d^3 \cdot 0.5\sigma_{ult}} \leq 1$$

Factor out $\frac{32}{\pi d^3}$:

$$\frac{32}{\pi d^3}\left(\frac{n_f M_a}{\sigma_e} + \frac{n_f M_m}{\sigma_{ult}} + \frac{n_f T}{2\sigma_{ult}}\right) \leq 1$$

Solve for d^3:

$$d^3 \geq \frac{32}{\pi}\left(\frac{n_f M_a}{\sigma_e} + \frac{n_f M_m}{\sigma_{ult}} + \frac{n_f T}{2\sigma_{ult}}\right)$$

Step 3: Plug in Values

$$d^3 \geq \frac{32}{\pi}\left(\frac{3 \times 4.92}{168 \times 10^6} + \frac{3 \times 10.42}{440 \times 10^6} + \frac{3 \times 15}{2 \times 440 \times 10^6}\right)$$

$$d^3 \geq \frac{32}{\pi} \times 21.0 \times 10^{-8} = 2.14 \times 10^{-6}\,\text{m}^3$$

Step 4: Minimum Diameter

$$d \geq \sqrt[3]{2.14 \times 10^{-6}} = 0.0129\,\text{m} = 12.9\,\text{mm}$$

Verification for $d = 16\,\text{mm}$

$$\sigma_m = \frac{32 \times 10.42}{\pi(0.016)^3} = 25.9\,\text{MPa}, \quad \sigma_a = \frac{32 \times 4.92}{\pi(0.016)^3} = 12.2\,\text{MPa}$$

$$\tau_m = \frac{16 \times 15}{\pi(0.016)^3} = 18.6\,\text{MPa}$$

Goodman ratio:

$$\frac{12.2}{168} + \frac{25.9}{440} + \frac{18.6}{220} = 0.28 \quad \text{(Passes, as } 0.28 < 0.33\text{)}$$