UNIVERSITY OF VICTORIA

Department of Mechanical Engineering

MECH 360 - Design of Mechanical Elements

Final Design Report

Students:

(Mina) Carmina Rocheleau - V01021553

(Seb) Sebastian Testart - V00825490

(Julie) Julie Lee - V00916550

Professor: Dr. Flavio Firmani

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Executive Summary

The report begins with an exploration of three concepts—"Planets," "Finding Nemo," and "Flora." Critical considerations include safety, aesthetic appeal, and adherence to constraints like height, car count, and electric motor power. The "Finding Nemo" concept emerges as the preferred choice.

In-depth analyses calculated loads, stress cycles, and component masses, determining specifications such as height, car count, motor power, and gear ratios. The gearbox design takes center stage, emphasizing efficiency and reliability. A 3-stage planetary gearbox is chosen based on torque requirements and power transmission efficiency.

The gear design process involves detailed assessments of specifications and failure analysis. Bending stress, bending-fatigue stress, and surface stress are thoroughly evaluated for each gear stage. Results, presented in tables, demonstrate that safety factors consistently exceed the design objective of 3.0, indicating the gear design meets required criteria.

The shaft assembly is revised to accommodate the small gearbox size, minimizing the need for additional support structures. Fatigue analyses at critical points along the shafts consider bending and shear stresses. Factors of safety above 3.0 are calculated, ensuring shaft reliability. Bearing design involves selecting roller bearings based on load requirements, with calculations demonstrating suitability for the 20-year operational period.

The report concludes with detailed specifications and drawings of gears, shafts, and bearings. The assembly process is outlined step by step for a systematic and efficient construction. The documentation provides a comprehensive overview of the design and analysis processes, emphasizing functionality, reliability, and safety.

Table of Contents

List of Figures

List of Tables

1. Introduction

Ferris wheels were invented in 1893 and are one of the most recognizable structures around [1]. They can be found in malls, amusement parks and scattered around cities as tourist attractions such as the London Eye in London, UK. Given the universal appeal and the whimsical allure of Ferris wheels, introducing one specifically tailored for children can serve as a powerful magnet for family-centered activities and businesses. Such an attraction not only provides an exciting activity for kids, but also keeps families in the vicinity longer, benefiting surrounding establishments. Recognizing this gap and the potential economic boost for local businesses, we are venturing into the design of a Ferris wheel that promises to be the new epicenter of child-centered entertainment and community engagement.

The purpose of the report is to explore potential designs of a mini-Ferris wheel, with an emphasis on the drivetrain of the Ferris wheel itself. Conceptual designs of the gearbox and the theme of the system will be compared using the weighted objectives chart. There is a need for a child-friendly amusement ride in Victoria, B.C., and the goal is to design a fully operational mini-Ferris wheel complete with a gearbox.

2. Background

Ferris wheels are amusement rides that consist of a support structure, axle, rim, passenger cabins, spokes, bearings, counterweights, a braking system, safety measures, and the drive system. Within the drive system, there is a motor, gearbox, and a method of transmitting power to turn the wheel.

Gearboxes are mechanical devices that allow conversions from variable speeds to torques between the power source and the load for a particular system, particularly in vehicles and industrial machinery. Within a gearbox, there exists gears, shafts, bearings, seals/gaskets, and a lubrication system all contained in a housing system. An important consideration when designing a gearbox is the gear ratio, which describes the ratio of the number of teeth on the driven gear to the number of teeth on the driving gear.

There are multiple types of gearboxes used in Ferris wheel applications, such as planetary, worm, bevel, helical, and parallel shaft.

3. Literature Review

3.1 Safety Considerations

Safety considerations for a Ferris wheel, particularly in the absence of restraints, are paramount and must adhere to the stringent standards outlined by organizations like ASTM (American Society for Testing and Materials). While Ferris wheels typically operate at relatively low speeds and offer a gentle, leisurely experience, they can still subject passengers to acceleratory forces. ASTM standards mandate that the maximum g-force experienced by a user should not exceed 1.5 times the force of gravity (1.5g) to ensure safety [2]. This is crucial to prevent discomfort, motion sickness, or any potential injury. Even without restraints, Ferris wheels are designed with secure cabins and railings to keep passengers within the designated seating area, minimizing the risk of accidents or falls. These safety measures, coupled with regular maintenance and inspections, ensure that Ferris wheel rides provide a safe and enjoyable experience for all riders [3].

3.2 Contemporary Drive Mechanisms

The drive mechanisms used in Ferris wheels vary depending on their size and purpose. In the case of large Ferris wheels, such as those found in amusement parks, they are typically powered by electric motors. These motors can draw very large amounts of power directly from a larger energy grid. The transmission method of power is usually a friction gearing mechanism, such as rubber tires, applied to the Ferris wheel’s outer rim. This design reduces the input torque required, which is essential for the operation of a very large Ferris wheels as shown below in

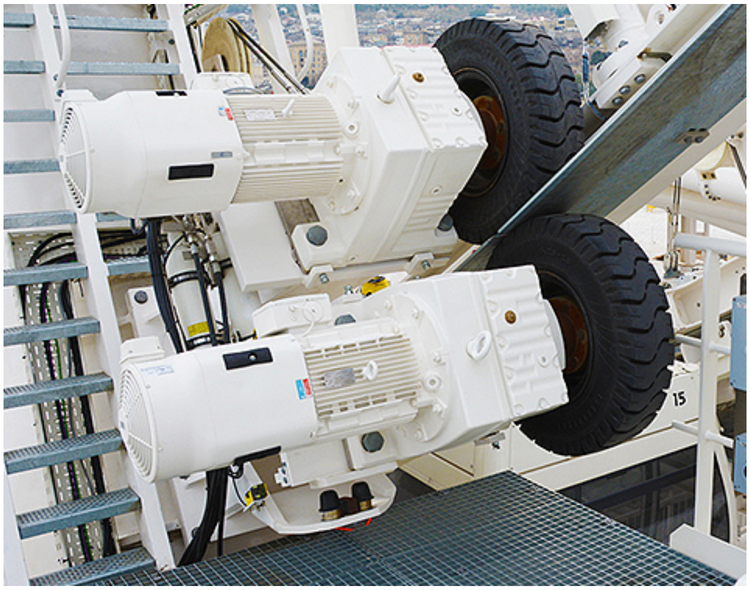
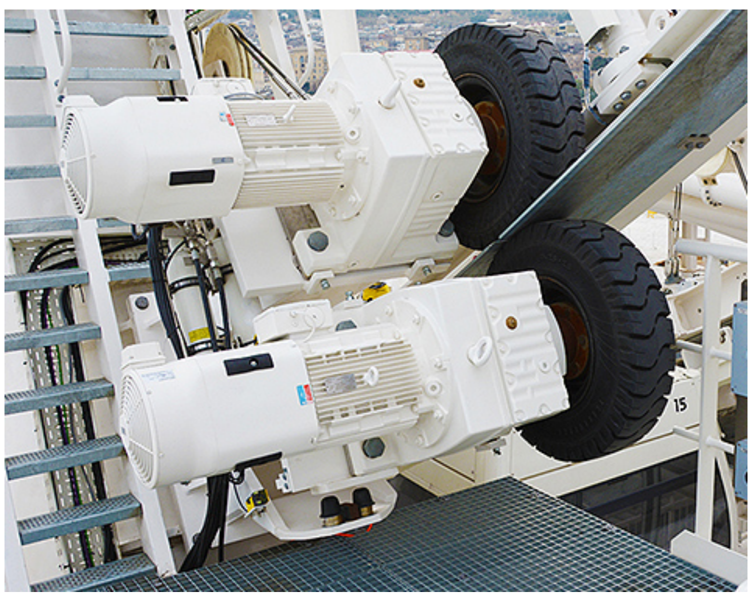


Figure 1.



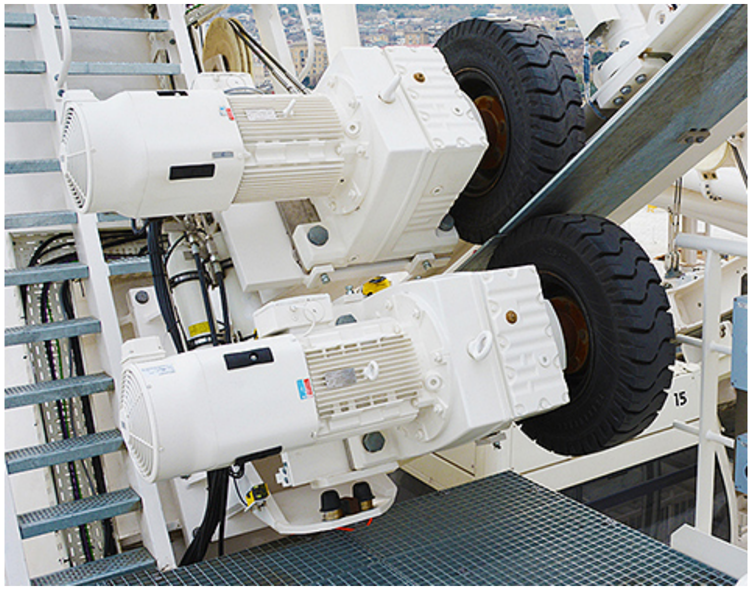


Figure 1. Large Ferris Wheel's Wheel [4]

On the other hand, smaller portable Ferris wheels, like the size being proposed, have lower torque and power requirements. For the sake of mobility and because of the lower power requirement small Ferris wheels are usually powered by combustion engines. A drive system, usually consisting of belts, is often connected directly to the center hub or axle of the Ferris wheel, which simplifies the mechanism and reduces the overall complexity, making it a cost-effective solution for smaller Ferris wheels. A small Ferris wheel is shown below in

Figure 2. Proper design and engineering of the drive mechanism are crucial to ensure the safety and functionality of Ferris wheels of all sizes.

Figure 2. Small Ferris Wheel [5]

3.3 Existing Designs

3.3.1 London eye

Figure 3 London Eye Ferris Wheel [6]

Located in Jubilee Gardens on the South Bank of London, England, the London Eye stands as a local icon and tourist attraction, as shown above in

Figure 3. Completed in March 2000, this Ferris wheel boasts a height of 135 meters and a diameter of 120 meters. It features a robust carbon steel support structure and has 32 passenger gondolas, which can accommodate up to 800 passengers concurrently. The structure, weighing in at 2,100 tons, came at a substantial cost of £75,000,000. The design is credited to Marks Barfield Architects, while the engineering design was a conglomerate effort that brought many small European engineering firms together, coordinated by the firm ARUP. The car of the London Eye is shown below in

Figure 4.

Figure 4 London Eye Car [6]

3.3.2 Singapore Flyer

Figure 5. Singapore Flyer Ferris Wheel [7]

The Singapore Flyer, as shown above in

Figure 5 is another impressive Ferris wheel located on Raffles Avenue, Singapore. It stands tall with a height of 165 meters and a diameter of 150 meters. Completed in 2008, this iconic attraction boasts 28 passenger gondolas and can hold up to 784 passengers simultaneously. A collaboration of firms brought this structure to life: the design was helmed by Kisho Kurokawa Architects and Associates and DP Architects, while engineering expertise was provided by ARUP and Mitsubishi Heavy Industries. The entire project was completed at a significant investment of S$240 million. The Singapore Flyer car is shown below in Figure 6.

Figure 6. Singapore Flyer Car [7]

3.3.3 Ain Dubai

Figure 7. Ain Dubai Ferris Wheel [8]

The Ain Dubai Ferris Wheel is the largest Ferris wheel ever built, as shown above in Figure 7. It towers at an impressive 250 meters in both height and diameter. It currently stands inoperable, but it is located on Bluewaters Island in Dubai. Completed in 2021, this colossal attraction features 48 passenger gondolas with the ability to accommodate 1,750 passengers at once. The ambitious project was another collaborative effort, brought to fruition by leading firms including Hyundai Engineering & Construction, Starneth Engineering, and KCI the Engineers B.V. The Ain Dubai came at a total cost of AED $1 billion. The car of the Ain Dubai is shown below in Figure 8.

Figure 8. Ain Dubai Car [8]

3.4 Market Survey

A 6-meter Ferris wheel can range greatly on price, especially depending on the region it will be shipped from. Typical prices with shipping range from $10,000 to $20,000 [9].

3.4.1 Potential clients

3.4.1.1 Beacon Hill Children's Farm

Benefits: The Children's Farm primarily focuses on farm animals, making it a popular family destination [10]. A Ferris wheel could enhance the overall experience by offering a unique view of the farm and the larger Beacon Hill Park. It could act as a visual anchor, drawing more families to the farm and serve as an additional revenue stream.
Space Feasibility: A small children's Ferris wheel doesn't require much ground area. Given the farm's location within Beacon Hill Park, they have sufficient space in the parking lot or within the farm itself .

3.4.1.2 Royal BC Museum

Benefits: The Ferris wheel can offer a unique view of the museum and the surrounding Inner Harbour area, making the courtyard more attractive and interactive. It could attract more visitors, especially families, and offer an additional revenue source for the museum.

Space Feasibility: The museum’s courtyard is quite spacious; a small children's Ferris wheel can be accommodated without obstructing the entrance or pedestrian pathways [11].

3.4.1.3 Victoria Butterfly Gardens

Benefits: A Ferris wheel could provide an elevated view of the gardens, enhancing visitor experience. It can also serve as a visual attraction, drawing more visitors. With the primary focus being on butterflies, a Ferris wheel can offer variety and an additional activity for children, increasing the time spent at the venue.
Space Feasibility: The garden’s parking lot is large and has the potential to be expanded to the edge of the property lot. Dedicating a portion for the Ferris wheel, while ensuring parking needs are met, could be feasible [12].

4. Alternative Concepts

This section contains mini-Ferris wheel designs brought forth from each individual member. Each design outlines a clear theme, and concept as well as a description of the gearbox selected and a brief explanation as to why it was chosen.

4.1 Seb - Planets

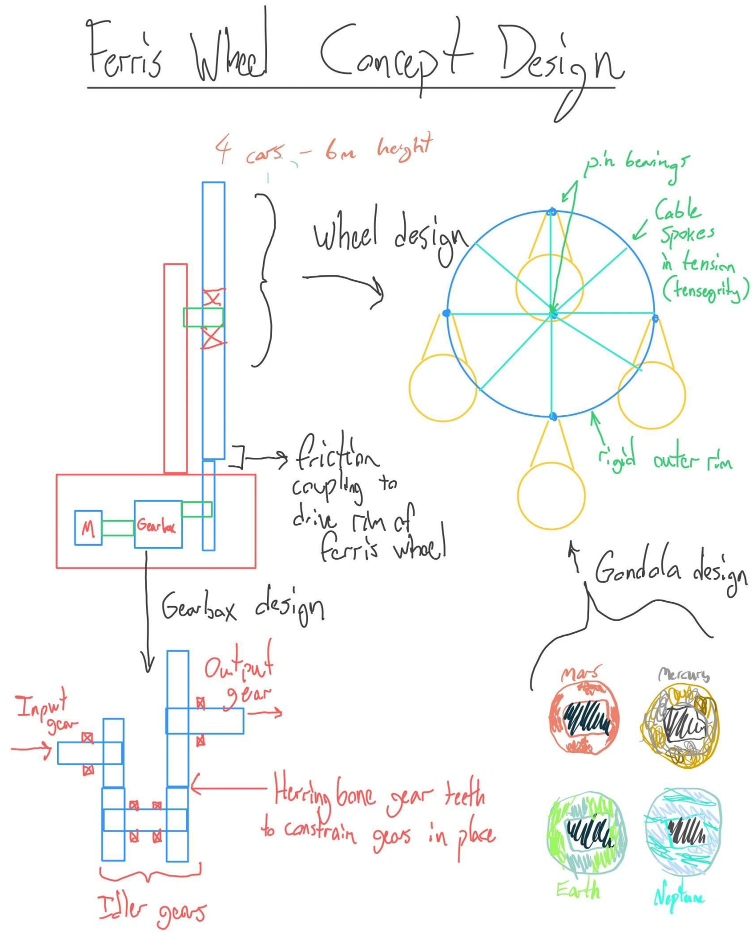


Figure 9. Seb's Planet Ferris Wheel

The planet themed concept design incorporates a gearbox equipped with herringbone gears, as shown in Figure 9. These gears are specifically chosen for their ability to provide axial constraints. The driving mechanism for the design is a friction coupling between the output shaft of the gearbox and the Ferris wheel's outer rim. This feature allows for a controlled braking mechanism via slippage. Structurally, the wheel draws inspiration from a bicycle wheel, utilizing tensegrity principles to maintain rigidity while facilitating smooth rotation about its central axis. Each passenger car is designed to hang from its own free-spinning bearing. Positioned at equidistant points around the wheel, this style of hanging the cars from pivot points ensure that gondolas maintain an upright position and ensure the safety of the passengers. The planet-themed cars are visually appealing while also simplifying the structural model.

4.2 Julie - Finding Nemo

The Finding Nemo concept consists of four cars in the shape of four popular characters in the movie franchise Finding Nemo, shown below in Figure 10. The Finding Nemo Ferris Wheel will be a portable design for ease of transport when needed. The location for the Ferris wheel would be in the parking lot of Hillside Mall located in Victoria, B.C., as a pit stop for children to ride the Ferris wheel during trips to the shopping mall.

The Ferris wheel is a standard spoke-and-hub design for simplicity and ease of portability. The four cars are evenly placed around the mounting stand of the Ferris wheel.

The gearbox design consists of planetary gears for efficient reduction of the incoming motor speed of 1750 RPM. Planetary gears are highly efficient in transmitting power as well as distributing load, which is important in Ferris wheel applications [13].
The planetary gears involve a sun gear that is the main gear in the middle with the planetary gears fixed and the ring gear would be the output gear.

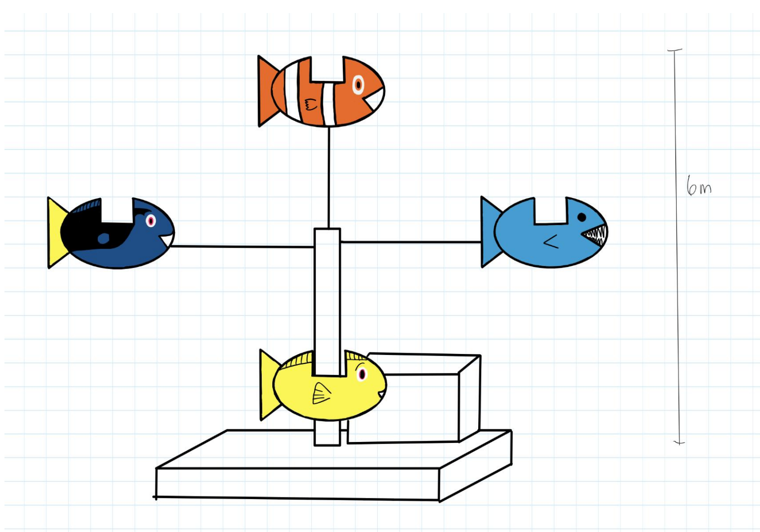


Figure 10. Finding Nemo Concept Ferris Wheel

4.3 Mina - Flora

This design was created with the intent of being able to place the Ferris wheel indoors or outdoors, with almost all year round use in a location with moderate weather such as Victoria B.C. The versatility of this design allows for the variety of target customers. Such as but not limited to Butchart Gardens, City of Victoria, and Malls.

The cars are designed after various potted plants, with an enclosed seating area and fauna that cascades over the top of the car and has lights on the underside, as shown below in Figure 11. This allows for protection against the outdoor elements, use in dark environments, and an attractive aesthetic even at night. The gear box is placed on the pillar in the back with the main driving gear at the center of the Ferris wheel.

The gearbox in this design is based off a worm gear set with a double-throated worm [14]. This gearbox was chosen as it is a compact system with a high-ratio speed reduction. The gearbox consists of a worm with a helical spiral around it that is linked to a worm wheel. A double-throated worm was chosen as it can take a larger load than a single or non-enveloped worm. These gear sets can be commonly found in conveyor belts as it is unidirectional which would be suitable for a mini-Ferris wheel application. The worm gearbox design is compact with a protective cover and implementing it into the design will take less material and space in the overall concept.

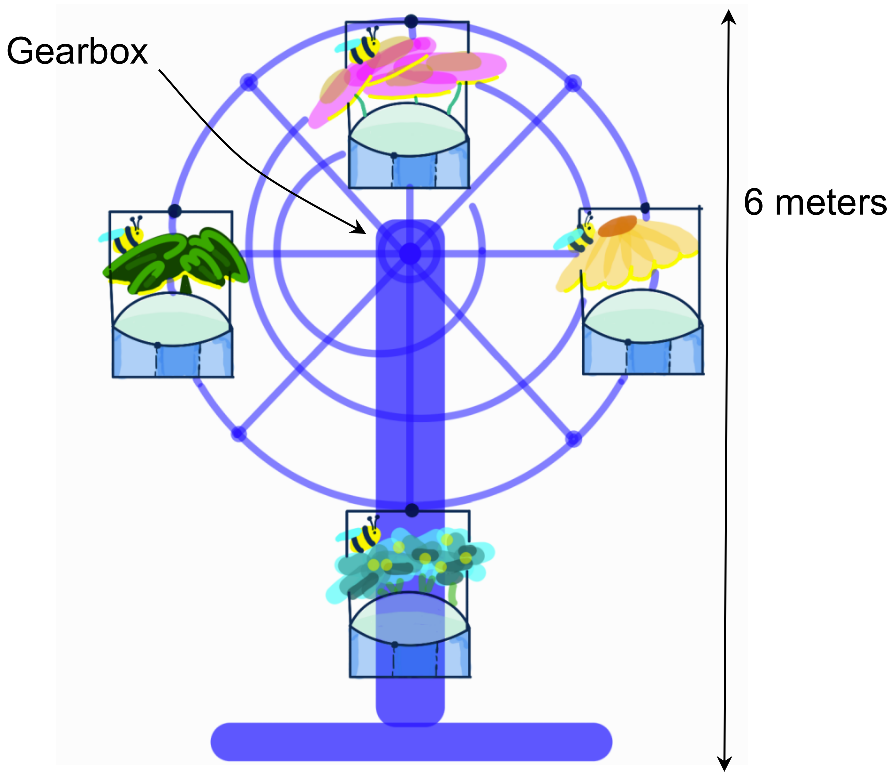


Figure 11. Flora Concept Ferris Wheel

5. Design Objectives and Constraints

The constraints, objectives, and justifications are outlined in this section and will act as an outline and guide for the selection and final design of the Ferris wheel. The design constraints and their justification are outlined below in Table 1.

<u>Goal Statement:</u> Design a mini four-car Ferris wheel that is powered by an electric motor in an efficient manner.

Table 1. Design constraints

constraints	Units	Target Value	Justification

Height	Meters	≤ 6	No higher than 6 meters for safety and available space.
Number of Cars	Cars	4	Four carts as per client specification.
Input Speed	RPM	1750	Powered by a 1750 rpm electric motor as per client specification.
Electric Motor Power	kW	0.5 ≤ x ≤ 5	Motor power requirements must sit within 0.5-5 kW to minimize operating cost.
Gearbox Efficiency	%	≥97	Gearbox must maintain 97% efficiency or higher to minimize construction and operating cost.
System Design Lifetime	years	≥20	The design should last a minimum of 20 years for a reasonable return on initial investment.
Output Velocity of Ferris Wheel	RPM	≤12	Output velocity must be under 1.5 g's as per ASTM international regulation to maintain safety.
Daily Time of Operations	PST	09:00 - 17:00	Must operate between 09:00 - 17:00 seven days a week to conform to clients regular working hours.
Weight of Children	kg	17 ≤ x ≤ 35	Payload must match children within the range requested by the client.

*Riders must experience no more than 1.5 g in any direction for a ride without a headrest or restraints [15]. Any point along the Ferris wheel's edge will encounter the greatest acceleration when the acceleration due to gravity is in addition to the centripetal acceleration.

$$a_T = 1.5g \tag{1}$$

$$a_T = 1.5 \times 9.810$$

$$a_T = 14.715$$

$$a_T = a_c + a_g \tag{2}$$

$$14.715 = a_c + 9.810$$

$$a_c = 4.905$$

$$a_c = \omega^2 \times r \tag{3}$$

$$4.905 = \omega^2 \times 3$$

$$\omega^2 = \frac{4.905}{3}$$
$$\omega \approx 1.278\,\mathrm{rad/s}$$

$$\text{Revolutions per second} = \frac{\omega}{2\pi} \tag{4}$$
$$\text{Revolutions per second} \approx 0.2035$$
$$\text{RPM} \approx 12.21$$

Therefore, we will conservatively set our RPM design constraint as ≤12 RPM.

The design objectives constructed by the team are outlined below in Table 2.

Table 2. Design objectives

No.	Objective Name	Units	Target Value	
1	Risk of Injury	Qualitative	Unlikely	Safety is of utmost importance in any engineering project, especially when the project is to be utilized by children. The Mini-Ferris wheel should have a minimal chance to create any hazards to the operators, users, or those nearby, or else it fails to meet the outlined goals.
2	Potential Injuries	Qualitative	≤ Scrapes/Bruise	No reasonable potential for injuries more harmful than scrapes and bruises.
3	Aesthetic	Qualitative 1-10	8	Captivating to the eye with intentional details that are suitable for the location of use.
4	Resilience to Wind	km/h	>20	Be able to withstand wind without malfunction.
6	Resilience to Rain	Yes/No	Yes	Can withstand rain for the duration of the operational life.
7	Simplicity	Qualitative	8	Function with minimal number of components and assembly if easy to follow.

6. Design Selection

The evaluation scale for the design objectives are outlined below in Table 3.

Table 3. Evaluation scale for design objectives

			Numeric Rating				
No.	**Objective Name**	**Units**	**0**	**2**	**5**	**7**	**10**
1	Risk of Injury	N/A	Guaranteed	likely		unlikely	impossible
2	Potential Injuries	N/A	Potential to cause serious bodily injury or death	Possibility for bodily harm (e.g., broken bones)		Potential for minor injury (e.g., cuts, minor pinches, bruises)	No physical injuries possible
3	Aesthetic	N/A	Poor	Okay	Adequate	Good	Excellent
4	Resilience to Wind	N/A	Operates inside	Operates outside	Operates inside and outside in specific weather conditions	Operates inside and outside in some weather conditions	Operates inside and outside in all weather conditions
5	Resilience to Rain	N/A	Poor	Okay	Adequate	Good	Excellent
6	Simplicity	N/A	Poor	Okay	Adequate	Good	Excellent
7	Meets Design Constraints	N/A	None	Few	Some	Most	All

Table 4. Weighted selection table gear box design

		Planets		Finding Nemo		Flora	
Objective Name	**Real**	**Score**	**Weighted**	**Score**	**Weighted**	**Score**	**Weighted**

	Weight						
Risk of Injury	20%	10	2	10	2	10	2
Potential Injuries	20%	10	2	10	2	10	2
Aesthetics	5%	7	0.35	10	0.5	10	0.5
Resilience to Wind	15%	10	1.5	10	1.5	10	1.5
Resilience to Rain	15%	10	1.5	10	1.5	10	1.5
Simplicity	5%	7	0.35	10	0.5	7	0.35
Meets Design Constraints	20%	10	2	10	2	10	2
		Total:	9.7	**Total:**	10	**Total:**	9.85

The weighted selection table is outlined below in Table 4. The result of the table shows that the Finding Nemo concept was the preferred selection for the design objectives compared to the other concepts with a score of 10.

Table 5. Weighted selection table wheel design

		Planets		**Finding Nemo**		**Flora**	
Objective Name	**Real Weight**	**Score**	**Weighted**	**Score**	**Weighted**	**Score**	**Weighted**
Risk of Injury	20%	7	1.4	7	1.4	7	1.4
Potential Injuries	20%	7	1.4	7	1.4	7	1.4
Aesthetics	5%	7	0.35	10	0.5	10	0.5
Resilience to Wind	15%	10	1.5	10	1.5	10	1.5
Resilience to Rain	15%	10	1.5	10	1.5	10	1.5
Simplicity	5%	7	0.35	10	0.5	7	0.35
Meets Design Constraints	20%	10	2	10	2	10	2

		Total:	8.5	**Total:**	8.8	**Total:**	8.65

According to Table 5, the highest score for the design of the wheel is Finding Nemo with 8.8.

7. Ferris Wheel Preliminary Drawing

The following drawing is the preliminary design based on the concept design that was selected. We are also choosing an overall safety factor of 3. The reasoning behind this is due to our uncertainty of the environmental conditions that our Ferris wheel shall operate in (wind and rain), and due to the uncertainty of the true rider weight. We are reasonably confident of the material properties we shall select for the design of the wheel, and we are also reasonably confident in the mathematical model we shall use to determine the required specifications of the Ferris wheel.

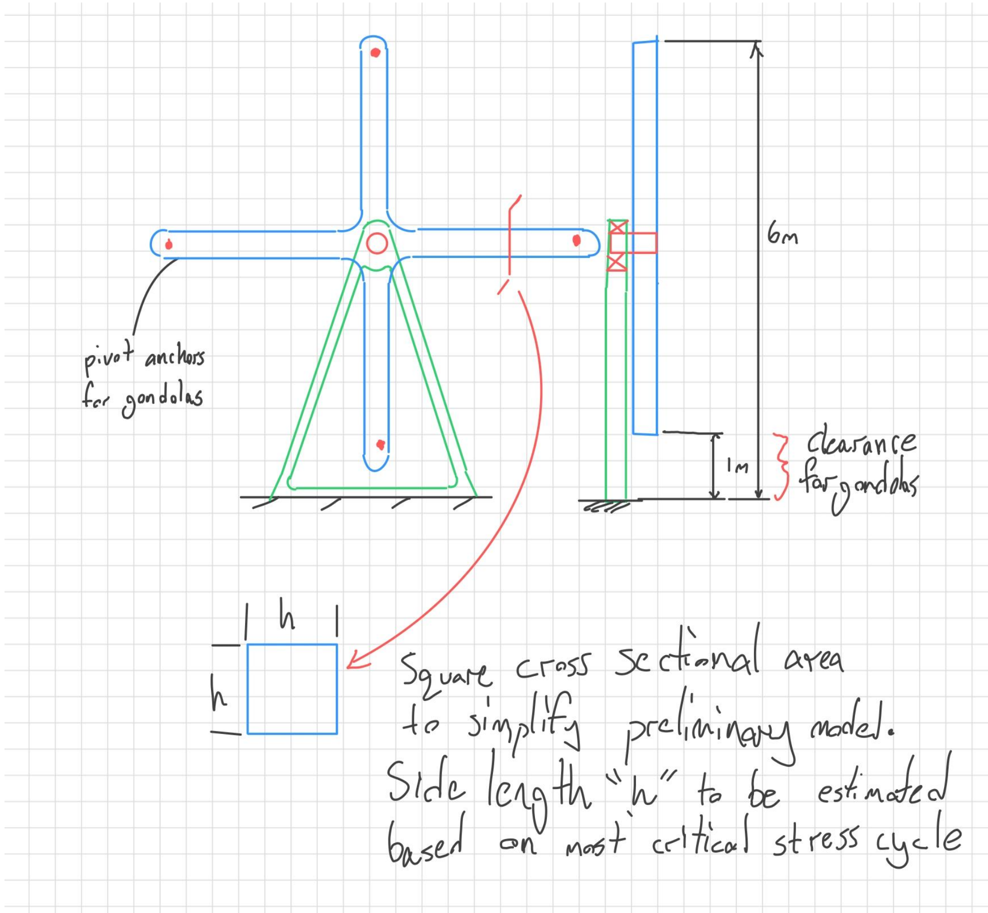


Figure 12. Preliminary Ferris Wheel Design

8. External Load Analysis

The load analysis is conducted on the design shown below in Figure 13. Constant speed and static equilibrium has been assumed. The section is broken down into four subsections to clearly outline the methods used in determining various loads as well as the stress cycle.

The length of the spoke from car to car is proposed to be 5 m in length, with a clearance of 1 meter to the platform. Therefore the shaft that drives the wheel itself is going to be 3.5 m in height. The gearbox dimensions that will house the planetary gearbox is proposed to be 0.5 m in height, 0.5 m in length, and 0.5 m in width.

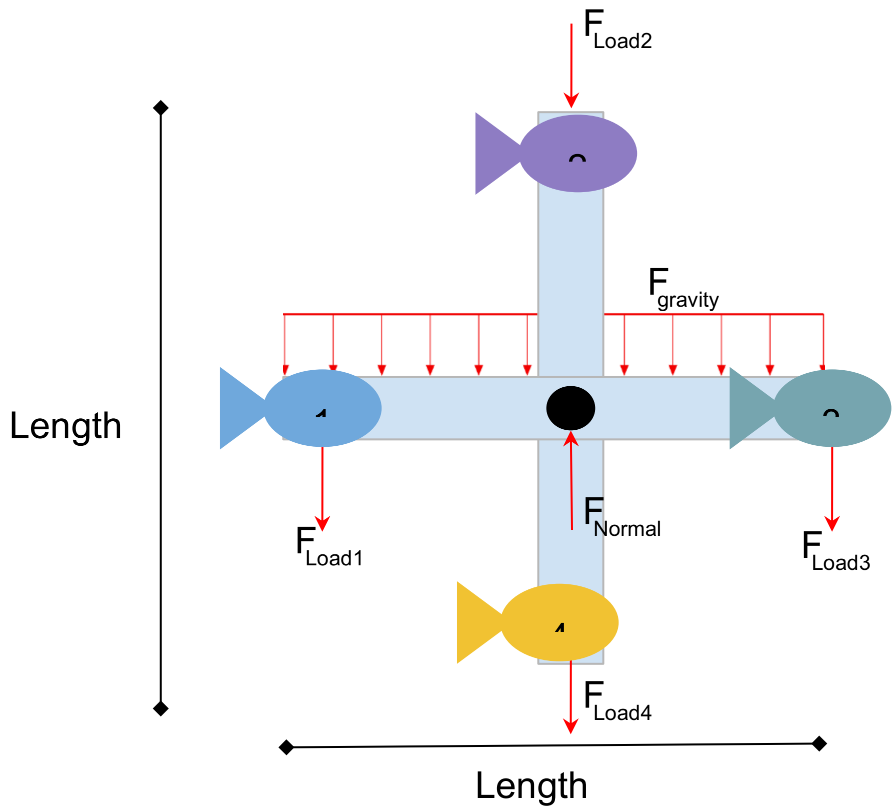


Figure 13. Forces of Ferris Wheel

8.1 Mass of the Children

The masses of the children were determined in the lab section as shown in Table 6.

Table 6. Weight of children

Children	kg	N
1	34	333.54
2	17	166.77
3	35	343.35
4	24	235.44

8.2 Mass of the Cars

The masses of the cars were determined by approximating the weight of the cars from a small Ferris wheel design as shown below in Figure 14.

Figure 14. Car Area Reference

Design parameters of the cars: 1.5m in height; 1m in length; 0.75m in depth

Total tubing dimensions based off of video: 8 x 1m tubes; 21 x 0.75m tubes

Total tube length = 23.75m

Tube diameter = 0.25 inches = 0.00635m

Total volume = $2 * \pi * 0.00635^2 * 23.75 = 0.006017m^3$

Density = 0.284 lb./in^3 [15]

Total mass of 1 car = 7861.0929 kg/m^3 * 0.006017 = 47.3kg

An assumption is made that the weight of the cosmetic material is negligible.

8.3 Most Critical Stress Cycle

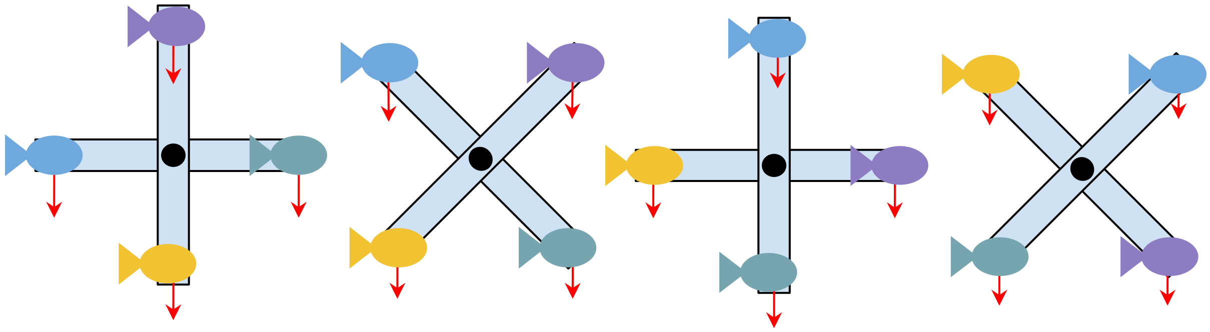


Figure 15. Force and Position Analysis

We created a python script, Figure 16 that would calculate the shear stress in the shaft that supports the weight of the Ferris wheel, we used initial diameter of 2.5 cm to conduct determine the critical stress cycle, further fatigue analysis will be required before a final diameter is selected. We designed the script to iterate through every possible combination of loaded and unloaded values for each car given the predetermined masses of the children. We then iterated the stress calculation across one full rotation of the Ferris wheel in order to account for addition torque caused by an unbalanced wheel and to determine the overall stress cycle.

```python
#Script to determine maximum shear force acting on the shaft that connects to a rotating ferris wheel given variable loading conditions.
#critical point for stress calculation is determined to be point on side of circular cross section of shaft connected to ferris wheel.


import math
import matplotlib.pyplot as plt


# Function to calculate maximum shear stress across one full rotation of ferris wheel
def tau360(comb):
    tau_max = None
    angle_max= None
    tau_min = None
    angle_min = None

    # Additional list to hold (angle, Tau) pairs for graphing
    tau_theta_pairs = []

    Mass_wheel_cars = 216.4
    Mass_kids = sum(x for x in comb)
    Diam = 0.025
    Area = math.pi*Diam**2/4
    # shear force on shaft of wheel is constant/independant of rotation
    V = 9.81 * (Mass_kids + Mass_wheel_cars)

    J = math.pi*Diam**4/64

    # Calculate torque acting on shaft of wheel for every angle 0.1 degree increments
    for theta in range(3600):
        angle = theta/10

        T = 2.5 * (sind(angle) * comb[0] + sind(angle + 90) * comb[1] + sind(angle + 180) * comb[2] + sind(angle + 270) * comb[3])
        Tau = (T * Diam / (2 * J)) + V / (3 * Area)
        # Store each value for graphing
        tau_theta_pairs.append((angle, Tau))
        # Update Tau_max and Tau_min
        if tau_max is None or Tau > tau_max:
            tau_max = Tau
            angle_max = angle
        if tau_min is None or Tau < tau_min:
            tau_min = Tau
            angle_min = angle

    return tau_max, angle_max, tau_min, angle_min, tau_theta_pairs, max_v, max_T

# Function to calculate sine with degree input
def sind(angle_in_degrees):
    # Convert the angle from degrees to radians
    angle_in_radians = math.radians(angle_in_degrees)
    # Return the sine of the angle in radians
    return math.sin(angle_in_radians)
```

```python
# Possible values for each loaded car (including 0 for unloaded cars)
values = [0,17,24,34,35]

# List to hold all combinations
combinations = []

# Generating combination and permutations of possible loading configurations
for a in values:
    for b in values:
        for c in values:
            for d in values:
                # Check if the combination is valid (0 can repeat, others cannot)
                if (a == 0 or a not in [b, c, d]) and \
                   (b == 0 or b not in [a, c, d]) and \
                   (c == 0 or c not in [a, b, d]) and \
                   (d == 0 or d not in [a, b, c]):
                    combinations.append((a, b, c, d))

# Optionally, print all load combinations to verify all configurations are included
#for combo in combinations:
#    print(combo)

# Iterate through every combination to determine which has the maximum possible peak stress
max_value = None
max_combination = None
for combo in combinations:
    value = tau360(combo)
    if max_value is None or value[0] > max_value[0]:
        max_value = value
        max_combination = combo

print("Most Critical Cycle occurs with load combination:")
print(max_combination)
print(f"Maximum Stress During Cycle: {max_value[0]} Pa at {max_value[1]} Degrees")
print(f"Minimum Stress During Cycle: {max_value[2]} Pa at {max_value[3]} Degrees")

angles, taus = zip(*max_value[4])  # Unzip the list of pairs

# plot shear stress vs angle of rotation and mark the maximum and minimum values
plt.scatter([max_value[1]], [max_value[0]], color='red', zorder=5, label=f'Max Tau at {max_value[1]}°')
plt.scatter([max_value[3]], [max_value[2]], color='green', zorder=5, label=f'Min Tau at {max_value[3]}°')
plt.plot(angles, taus)
plt.xlabel('Angle (θ)')
plt.ylabel('Tau (τ)')
plt.title('Tau vs Theta')
plt.legend()
plt.grid(True)
plt.show()
```

```
Most Critical Cycle occurs with load combination:
(0, 0, 34, 35)
Maximum Stress During Cycle: 81425493.81874458 Pa at 224.2 Degrees
Minimum Stress During Cycle: -77623058.64741515 Pa at 44.2 Degrees
```

Figure 16. Script Output

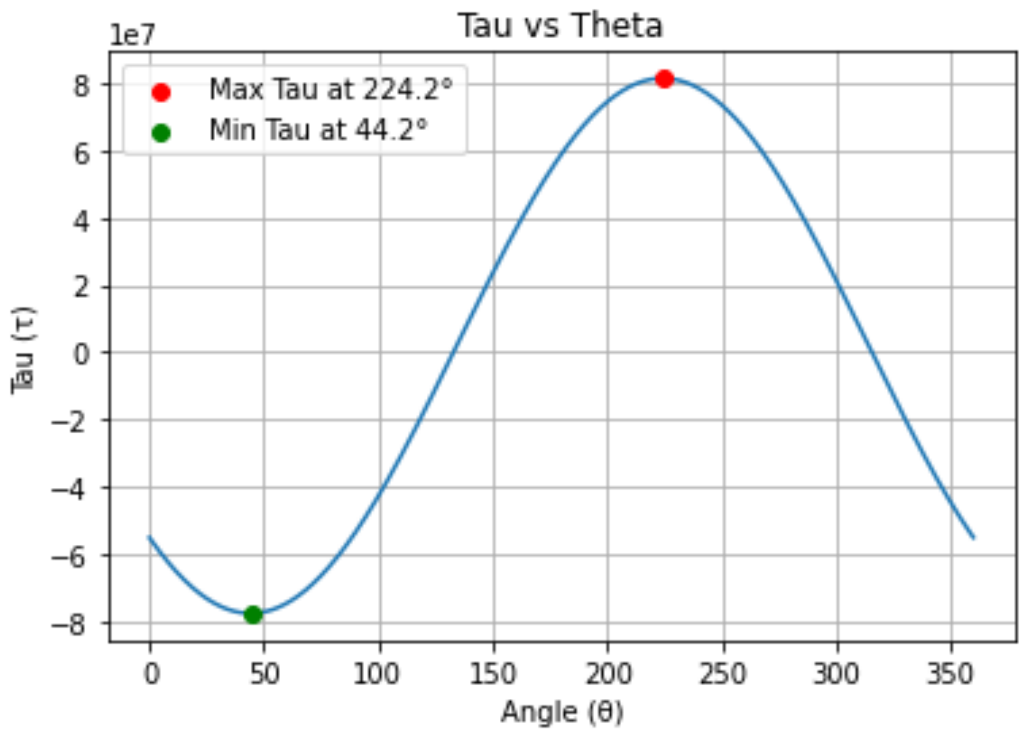


Figure 17. Stress Cycle with Worst Case Load Combination

8.4 Mass of the Wheel

We determined the mass of the Ferris wheel beams by designing a cross section large enough to withstand the maximum possible bending moment induced by the weight of two largest children at either end of one beam on the Ferris wheel.

The maximum stress in the beam will occur at the midpoint, and it will be equal to the stress of the child and car as a point load plus the stress due the distributed load of the weight of the beam, as shown below in
Table 7.

Table 7. Values for beam analysis

Name	Value	Units
Force 1 (F1)	797.553	N

Force 2 (F2)	807.363	N
Length (L)	5	m
Half of the Length (L/2)	2.5	m
Resulting Force (Fr)	Unknown	N

For the analysis the beam is assumed to be fixed at the center unable to rotate creating a static system that is then analyzed as shown in Figure 18 below.

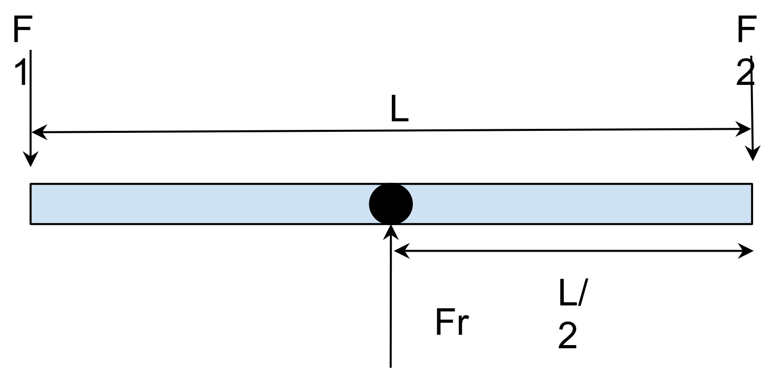


Figure 18. Free Body Diagram Beam Force

Using Newton's Third Law of action and reaction we can calculate the Fr:

$$Fr = F1 + F2 \tag{5}$$

Fr= $797.553N + 807.363N = 1604.916\ N$

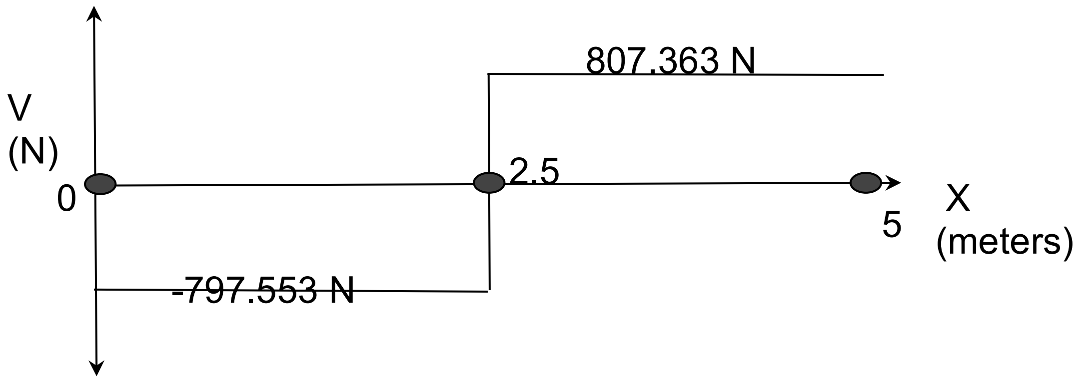


Figure 19. Shear Force Diagram

Using a Shear Force diagram as shown in Figure 19 above, the resulting moment (Mr) can be calculated using the area.

$$Mr = F1 * Area \tag{6}$$

$$Mr = -797.553N * 2.5m = 1993.8825\ N.m$$

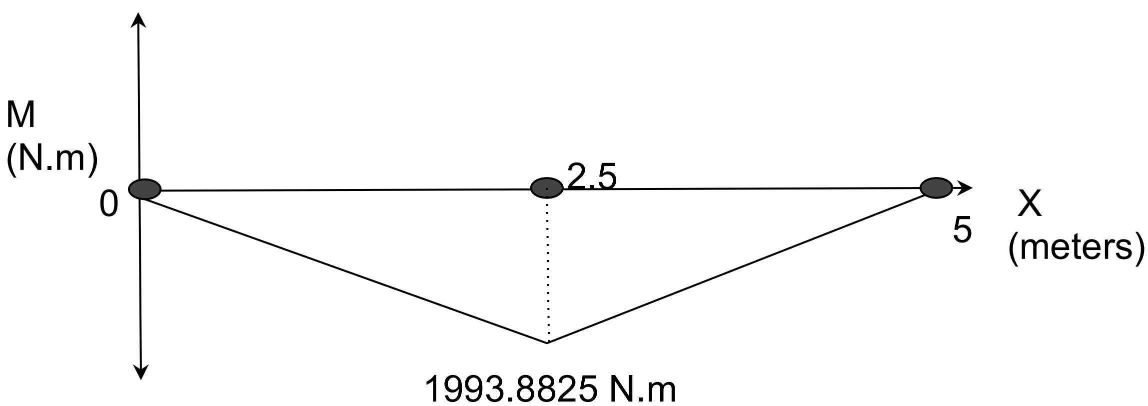


Figure 20. Bending Moment Diagram

Observing Figure 20, the maximum moment is determined as 1993.8825 Nm. The maximum moment is equivalent to the maximum stress being placed in a cycle on the Ferris wheel. Table 8 outlines definitions of mathematical symbols and constants for the following determination of the beam cross section..

Table 8. Mathematical symbols and constants

Name	Symbol	Value	Units
Acceleration due to gravity	g	9.81	m/s^2
Moment	M	T	N*m
Stress	σ	TBD	Pa
Cross section area side length	h	TBD	meters
Volume of beam	V	TBD	m^3
Length of beam	L	5	meters
Mass of car	m_{car}	47.3	kg
Mass of child	m_{child}	34; 17; 35; 24	kg
Mass of beam	m_{beam}	TBD	kg
Moment of inertia	I	TBD	RPM
Height from cross section center of area	y	TBD	meters
Density of 4041 steel	ρ_{steel}	7861	kg/m^3
Weight of beam	w	TBD	N
Ultimate tensile failure strength of 4041 steel	σ_{ult}	655,000,000	Pa
Safety Factor	S_F	3	#

The maximum stress due to the point loads can be solved given:

$$\sigma = \frac{M \times y}{I} \tag{7}$$

$$M = g \times (m_{child} + m_{car}) \times L/2 \tag{8}$$

$$I = \frac{b \times h^3}{12} \tag{9}$$

$$y = h/2 \tag{10}$$

Also, due to square cross section: $b = h$

Therefore:

$$\sigma_{max} = \frac{g \times (m_{child} + m_{car}) \times (L/2) \times h/2}{h^4/12} \tag{11}$$

$$\sigma_{max} = \frac{3g(m_{child} + m_{car}) \times L}{h^3} \tag{12}$$

The maximum stress due to the distributed load can be solved given:

Acceleration due to gravity: $g = 9.81 m/s^2$

$$\rho_{steel} = 7861 kg/m^3$$

$$\sigma = \frac{M \times y}{I}$$

$$M = w \times (L/2) \times (L/4)$$

$$w = g \times \rho_{steel} \times h^2$$

$$M = \frac{g \times \rho_{steel} \times h^2 \times Ł^2}{8}$$

$$I = \frac{h^4}{12}$$

$$y = h/2$$

$$\sigma_{\max} = \frac{\left(\frac{g \times \rho_{\text{steel}} \times h^2 \times L^2}{8}\right) \times \left(\frac{h}{2}\right)}{\frac{h^4}{12}}$$

$$\sigma_{\max} = \frac{3g \times \rho_{steel} \times Ł^2}{4h}$$

We can add these two stresses together to find the overall maximum stress in our beam for the most critical loading cycle of the beam:

$$\sigma_{max} = \frac{3g(m_{child} + m_{car}) \times L}{h^3} + \frac{3g \times \rho_{steel} \times Ł^2}{4h} \tag{7}$$

We can then set the maximum stress equal to our chosen safety factor multiplied with the ultimate tensile stress of 4041 steel (655 MPa) to find the required side length of the square cross-sectional area of the beam:

$$S_F \times \sigma_{\text{ult}} = \frac{3g(m_{child} + m_{car}) \times L}{h^3} + \frac{3g \times \rho_{steel} \times \L^2}{4h} \tag{8}$$

We can rearrange the equation in terms of h to create a cubic polynomial:

$$S_F \times \sigma_{\text{ult}} h^3 - \frac{3}{4} \times g \times \rho_{steel} \times \L^2 h^2 - 3g(m_{child} + m_{car}) \times L = 0 \tag{9}$$

This polynomial only has 1 positive real solution:

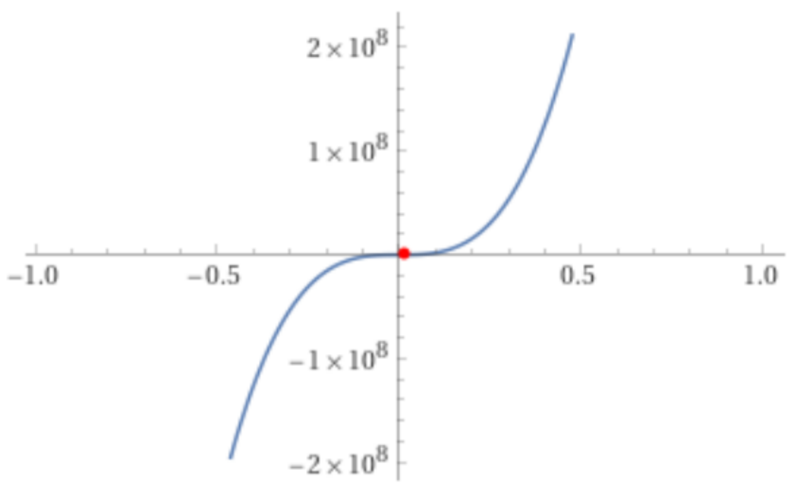


Figure 21. Resulting Polynomial

It can be concluded that the required side length is the positive nonzero root, or h = 1.86 cm which will ensure that maximum stress in the beam does not exceed the chosen failure limit.

The mass of each structural beam that constructs the wheel can be calculated, where the square cross section side length h = 1.86 cm, and the total length of each beam is 5 m.

$$m_{beam} = \rho V = \rho h^2 L$$
$$m_{beam} = 13.6kg$$

9. Gearbox Design

Before determining the preliminary gearbox design, the required power to start spinning the Ferris wheel is calculated.

The total rotational moment of inertia of the wheel is estimated by simplifying each car and child to a point mass and adding the beams rotating about their midpoint.

$$m_{kids+cars} = (34 + 17 + 35 + 24)_{\text{children}} + 4_{\text{cars}} \times 47.3$$
$$m_{kids+cars} = 299.2kg$$

$$I_{wheel} = 2_{beams} * \frac{1}{12}(m_{beam})L^2 + \frac{1}{4}(m_{kids+cars})L^2$$
$$I_{wheel} = 2_{beams} * \frac{1}{12}(13.6)(5)^2 + \frac{1}{4}(299.2)(5)^2$$
$$I_{wheel} = 1926.67kgm^2$$

The required power as the torque is defined as the torque needed to accelerate the wheel up to the desired rotational speed while the tangential acceleration does not exceed the maximum safe value of 1.5 g as defined earlier.

A rotational speed of 2 RPM is chosen as the target for the output speed of the Ferris wheel, as it is well below the constraint.

$$DesiredSpeed = \omega = 2RPM = 0.20944rad/s$$
$$r = \frac{L}{2}$$
$$a_t = 9.81 * 1.5$$

$$a_t = r \cdot \alpha$$
$$\alpha \approx 5.8924\,\mathrm{rad/s^2}$$
$$\tau = I_{wheel}\alpha$$
$$P_{out} = \tau \cdot \omega$$
$$P_{out} = (1926.67)(5.8924)(0.20944) = 2377.7W$$
$$P_{req} = \frac{P}{Efficiency\%}$$

It is assumed that the overall efficiency is equal to the gearbox efficiency, 97%, to simplify the preliminary analysis:

$$P_{req} = \frac{2377.7}{0.97} = 2451.2W$$

The potential designs for the different gearboxes are herringbone gears, planetary gears, and worm gears. The planetary gears for the gearbox design were chosen according to TABLE 4. The desired output velocity is 2 RPM, according to the specifications of a similar-sized Ferris wheel. Therefore, with the output angular velocity of 2 RPM and the incoming electric motor speed of 1750 RPM, the desired gear ratio is

$$Gear\ ratio\ = \frac{rotations\ of\ driver\ gear}{rotations\ of\ driven\ gear} = \frac{1750}{2} = 875$$

In general, planetary gears cannot have a gear ratio higher than 10:1 [16], therefore, to produce the required gear ratio of 875, a 3-stage planetary gearbox is proposed.

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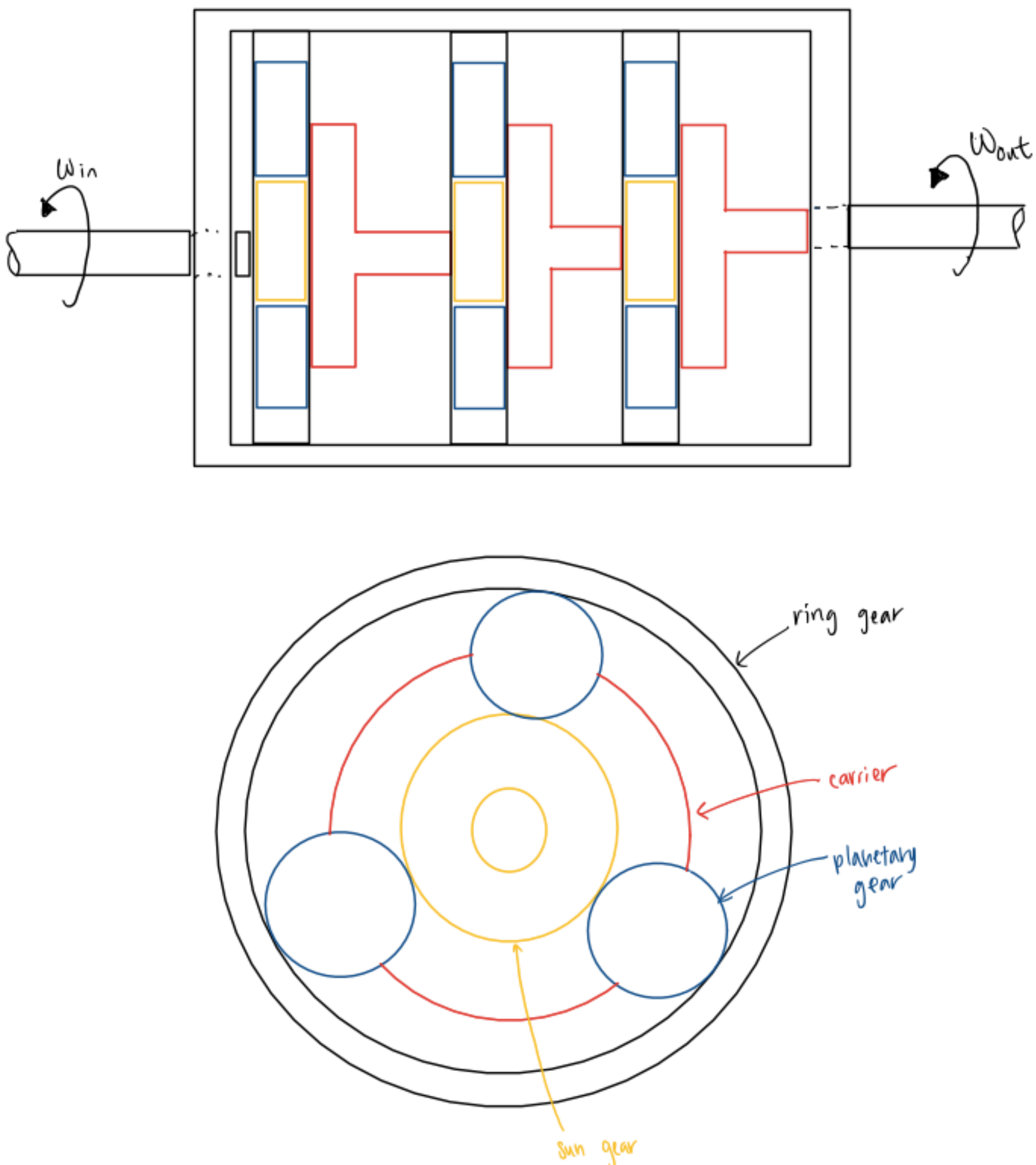


Figure 22. Gear Box Design

Figure 22 above shows the cross-section of the proposed three-stage planetary gearbox. The front view of a planetary gear is shown below the cross-sectional gearbox, consisting of the sun gear, ring gear, and planetary gears along the carrier. The proposed planetary gearbox will have the inputs as the sun gear with a fixed ring gear, and the output as the planetary gear for maximum gear ratio.

The calculations for the angular velocities, torques and gear ratios are shown below with a schematic of the three-stage planetary gearbox shown in Figure 23.

$$P1, \omega 1, T1 \ = \ input$$
$$P2, \omega 2, T2 \ = \ output\ of\ 1st\ stage\ planetary\ gear$$
$$P3, \omega 3, T3 \ = \ output\ of\ 2nd\ stage\ planetary\ gear$$
$$P4, \omega 4, T4 \ = \ output\ of\ 3rd\ stage\ planetary\ gear$$

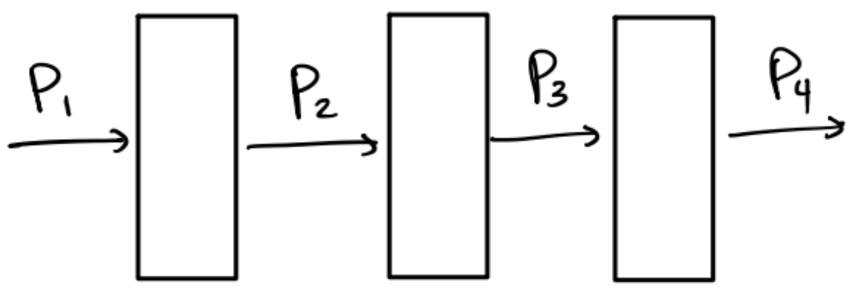


Figure 23. Schematic for Power and Stages of Gearbox

For the gear ratio that is calculated to be 875, it is assumed to have an equal reduction among the three stages of the planetary gears, as shown below.

$$Gear\ ratio\ across\ each\ stage \ = \ \sqrt[3]{875} \ = \ 9.565$$

In order to maintain 97% efficiency, the efficiency is divided equally across each stage of efficiency as shown below.

$$Efficiency\ across\ each\ stage\ =\ \sqrt[3]{0.97}\ =\ 0.9898983$$

The equation to calculate torque and power, the equations are shown below.

$$Torque\ =\ torque * gear\ ratio * efficiency$$
$$Power\ =\ torque * speed$$

The table of all the powers, torques, speeds, and gear ratios are shown below in Table 9.

Table 9. Power, speed, torque, and gear ratio for each stage

		1st Stage		2nd Stage		3rd Stage	
	Electric Motor	**Sun gear**	**Planet carrier**	**Sun gear**	**Planet carrier**	**Sun gear**	**Planet carrier**
Power [W]	2451.2	2451.2	2426.44	2426.44	2401.93	2401.93	2377.7
Speed [RPM]	1750	1750	182.97	182.97	19.13	19.13	2
Speed [rad/s]	183.26	183.26	19.16	19.16	2.00	2.00	0.21
Torque [Nm]	13.38	13.38	126.64	126.64	1199.04	1199.04	11352.68
Gear ratio			9.56		9.56		9.56

10. Wheel Specifications

Based off the preliminary external load analysis, gearbox design, design goals, and design constraints, it is determined that the Ferris wheel shall have the following specifications listed below in Table 10.

Table 10. Ferris wheel specifications

Specifications	Value	Units

Safety Factor	3	#
Height	6	Meters
Width	5	Meters
Number of Cars	4	#
Mass of wheel (cars + beams)	216.4	kg
Electric Motor Speed	1750	RPM
Electric Motor Power	2451.2	Watts
System Design Lifetime	20	Years
Output RPM of Ferris Wheel	2	RPM
Output Speed of Ferris Wheel	0.524	m/s
Gearbox Efficiency	97	%
Gearbox Gear Ratio	875	#
Operational Hours Per Day	8	Hours
Weight of Children	34; 17; 35; 24	kg

11. Gearbox Design

The preliminary gearbox layouts are shown below in Figure 24. The initial concept assumed three planetary gears, however, to satisfy the condition checks, the number of planet gears that are needed is four.

As mentioned previously, the gearbox dimensions are proposed to be 0.5 m in height, 0.5 m in length, and 0.5 m in depth. The height of 0.5 m would provide clearance for the housing and the maximum size of the ring gear according to Table 11. The length of 0.5 m would be for the three stages of the planetary gearbox. The depth of 0.5 m would suffice for the diameter of the maximum size of the ring gear.
The length of the shafts connecting the planet gears and the planet carrier are proposed to be 10 cm each. It is proposed that the output shaft connecting the planet carrier to the sun gear across each stage will be 10 cm as well.

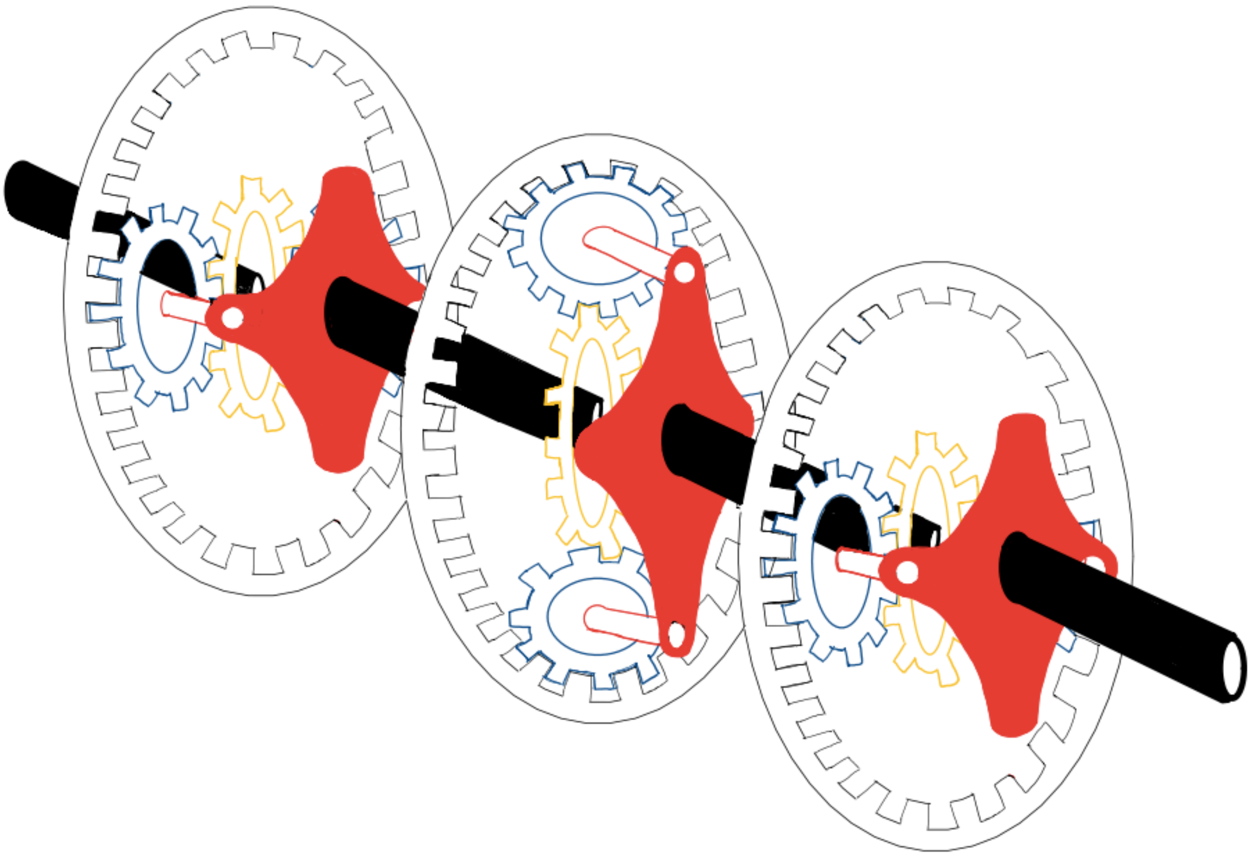


Figure 24. Isometric View of Gearbox

Above in Figure 24, it is shown that the planet carrier is attached to the holes of the planet gears revolving around the sun gear, with the output shaft of the planet carrier attaching to the subsequent sun gear of the next stage. The first shaft is transmitting the speed of the motor to the first sun gear, and the last shaft is the final output speed of the eventual Ferris wheel. Due to the change in number of planets, the conceptual design includes 2 planet gears revolving around the sun gear rather than the initial concept of 4 planet gears.

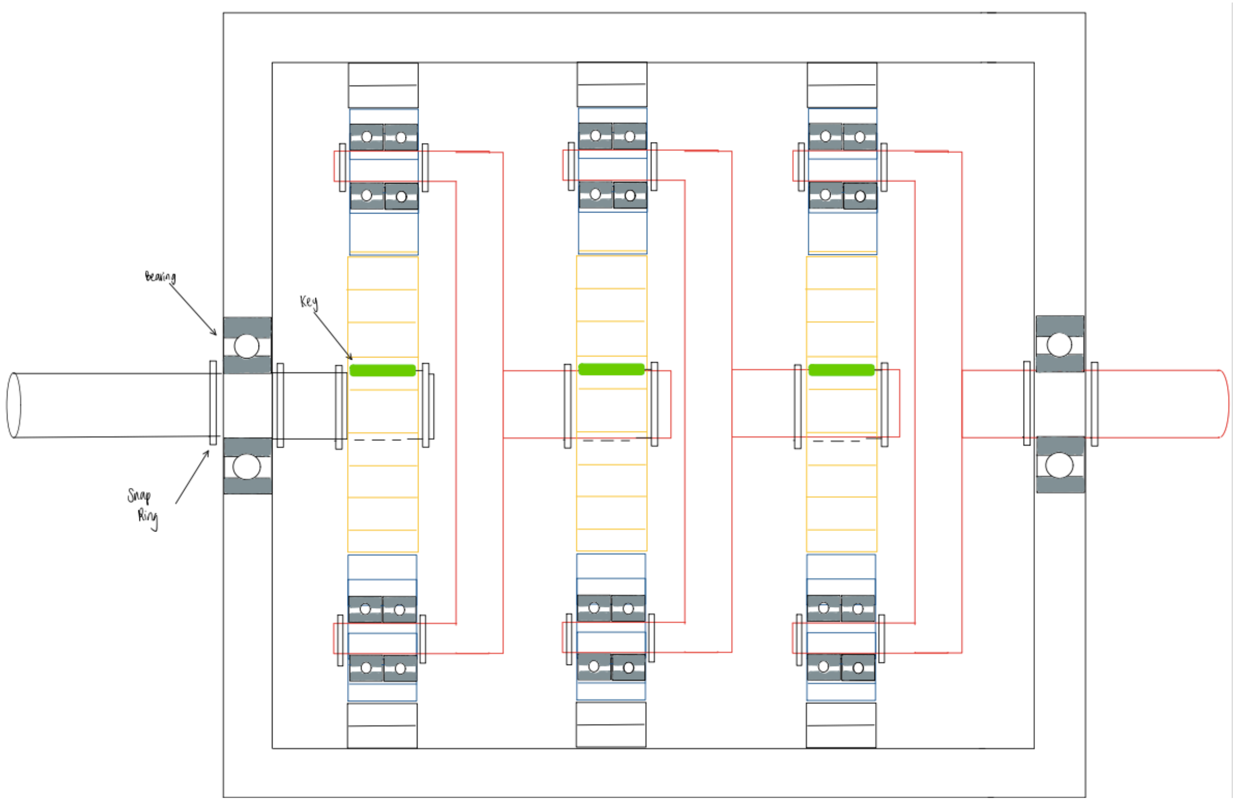


Figure 25. Cross-Sectional View of Gearbox

The cross section above shown in Figure 25 shows the proposed locations of the bearings, keys, snap rings, and the grooves on the shafts. It is proposed to have 2 bearings to fit the width of the planet gears for the connection to the planet carrier. Bearings will be inserted on either side of the gearbox for the input and output shaft rotations. The keys will be inserted between the shafts and the sun gears for all stages of the gearbox. The grooves of the shafts are located where snap rings and keys will be placed. Shoulders on the shafts would be located where the bearings come into contact with the shaft.

The critical moments of the shafts will be at the grooves on the shafts due to the concentration factors that increase the stress experienced by the shafts. The grooves from the keyways and the snap rings will dictate the critical moments on the shafts. Additionally, the shoulders on the shafts where the bearings reside would indicate a critical moment.

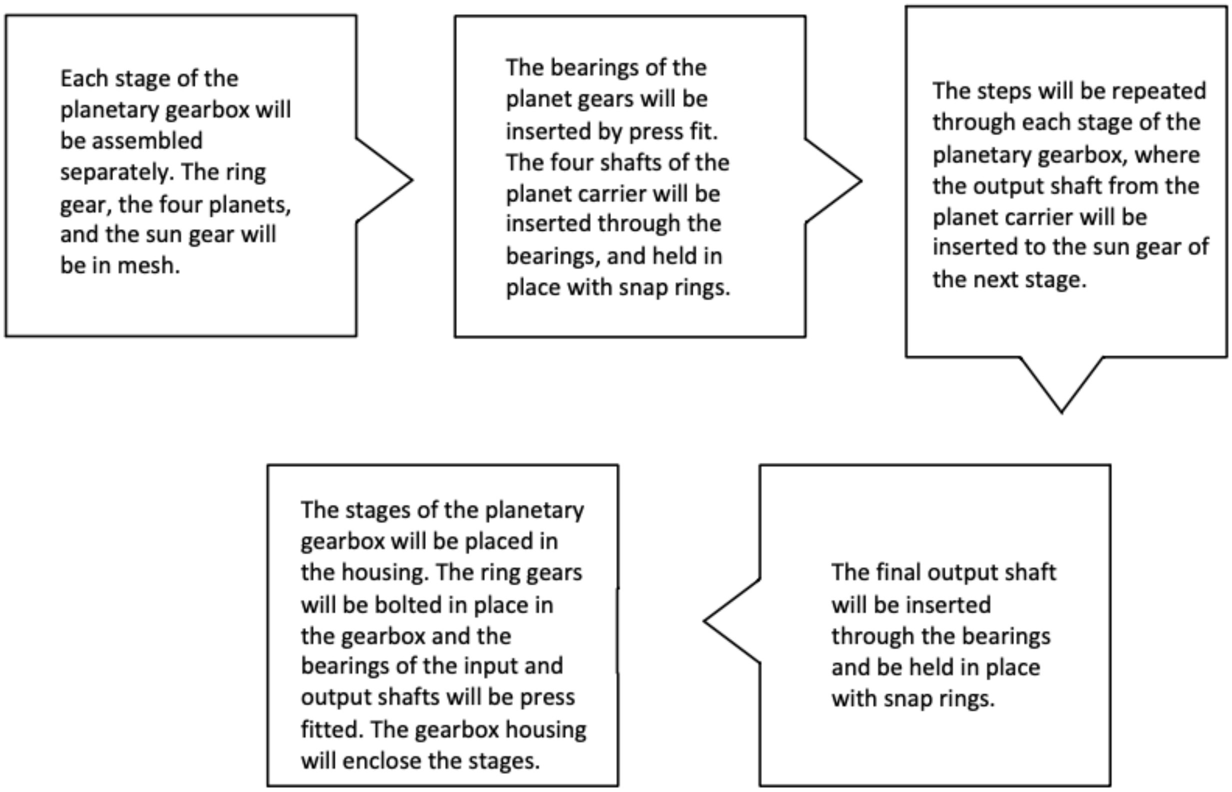


Figure 26. Flow Chart of Assembly

The flow chart for the assembly of the gearbox is shown above in Figure 24. Nominal fasteners will be used to fasten the ring gear to the outer housing of the gearbox, as well as fastening the ring gear to the gearbox housing itself.

12. Gear Design

Hard steel has been selected for the material of choice due to its greased friction factor of 0.029 [17]. Using the values found in Section 9, Table 9 for angular speed and torque for the ideal system, specifications of the system can be determined. Using the provided values as the input and output speed at each stage, a diametral pitch of 0.40, and Gear 4 with the number of teeth of 100, the specifications of the gears were determined. Using the tabulated method shown in Appendix A for each all three stages of the planetary gear, the base equations are listed below.

1. $W1 = X + Y$
2. $W2 = X$
3. $W3 = X - \frac{N1}{N3}Y$
4. $W4 = X - \frac{N1}{N4}Y$

Equation 2 is the carrier gear's (Gear 2) angular speed (ω_2) which is also the output speed. Therefore, X is equal to the output angular speed, and Y's angular speed is the input speed of the sun gear (Gear 1) subtracting X found from rearranging Equation 1. The angular speed (ω_4) of the ring gear (Gear 4) is fixed therefore ω_4=0, allowing for Equation 4 to be rearranged to find the number of teeth (N1) on the sun gear (Gear 1).

Knowing the number of teeth and diameter of the sun gear (Gear 1) and the ring gear (Gear 4) the number of teeth and the diameter of the planet gear (Gear 3) can be determined using Equation 1 and 2 shown below. Once the diameter of the planet gear (Gear 3) is found, the radius of the carrier (Gear 2) can be determined using Equation 3 below.

1. $N3 = \frac{N4 - N1}{2}$
2. $D3 = \frac{D4 - D1}{2}$
3. $R2 = R1 + R3$

With all the radius of the gears determined, each torque of the system can be determined by using the equation below.

$$T2 = \frac{R2}{R1} \cdot T1$$

The specifications of the gears are outlined in the Table 11 below.

Table 11. Gear Specifications

Planetary Gear Specifications				
Material	Hard Steel			
Coefficient of Friction - Greased	0.029			
	Gear 1 (Sun) - Input	Gear 2 (Carrier) - Output	Gear 3 (Planets)	Gear 4 (Ring)
All Stages Number of Teeth - (N)	12	-----	44	100
All Stages Gear Diameter [mm] - (d)	30.00	140.00	110.00	250.00

Stage 1 Angular Speed [RPM] - (w)	1750	182.97	-244.40	0.00
Stage 2 Angular Speed [RPM] - (w)	182.97	19.13	-25.55	0.00
Stage 3 Angular Speed [RPM] - (w)	19.13	2	-2.67	0.00
Stage 1 Torques [Nm.] - (T)	13.38	126.64	49.06	111.50
Stage 2 Torques [Nm.] - (T)	126.64	1199.04	464.35	1055.33
Stage 3 Torques [Nm.] - (T)	1199.04	11352.68	4396.48	9992.00
Stage 1 Power [W] - (P)	2451.20	2426.44	-----	-----
Stage 2 Power [W] - (P)	2426.44	2401.93	-----	-----
Stage 3 Power [W] - (P)	2401.93	2377.70	-----	-----
Diametral Pitch [mm] - (dp)	0.40	-----	0.40	0.40
Gear Ratio (e i/j)	9.56	9.56	9.56	9.56

To check the how many planet gears (Gear 3) can be in the system, the equation below is used with alpha equal to 1 to find the maximum number of planets.

$$n < \frac{180}{\sin^{-1}\left(\frac{N3+2\cdot\mathrm{ALPHA}}{N1+N3}\right)}$$

The check indicated that 3 planets was the maximum number of planets that can be placed in the system. As the original design of the planetary gear had 3 planets using the check given by the equation below, it was found that 3 planet gears did not provide an integer. When 2 planets are used, the equation provides an integer of 56.

$$\frac{N1+N4}{n} = \text{integer.}$$

Table 12. Gear Specifications Check

Max Number of Planets	3.26
Condition Check (2 PLANETS)	56

The base pitch (PB) is determined using the equation below addendum (A) for full teeth at a 20-degree pressure angle shown as below.

$$A = \frac{1}{P}$$

$$PB = \frac{\pi\cdot\cos(\mathrm{PHI})}{\mathrm{PD}}$$

Using the Base Pitch, the contact ratio can be determined. The equation for contact ratio of external to external is shown below using the circumference (C).

$$C = R1 + R3$$
$$Z = \sqrt{(RP + A)^2 - (RP \cdot \cos(\phi))^2} + \sqrt{(RG + A)^2 - (RG \cdot \cos(\phi))^2} - C \cdot \sin(\phi)$$
$$Mp = \frac{Z}{PB}$$

The equation for contact ratio of external to internal is shown below using the circumference (C).

$$C = R4 - R3$$
$$Z = \sqrt{(RP + A)^2 - (RP \cdot \cos(\phi))^2} - \sqrt{(RG + A)^2 - (RG \cdot \cos(\phi))^2} + C \cdot \sin(\phi)$$
$$Mp = \frac{Z}{PB}$$

Due to bracketing errors in EXCEL the final calculated contact ratios values are shown in Table 13 below.

Table 13. Gear Contact Ratios

Contact Ratio External – External (Gear 1 - Gear 3) - (m_p)	1.58
Contact Ratio External - Internal (Gear 3 - Gear 4) - (m_p)	6.55

The efficiency is then calculated for the external to external and external to internal using the equation below.

$$n = 1 - \frac{\mu \cdot m_p \cdot P_B}{4 \cdot \cos(\beta) \cdot \cos(\phi) \cdot \left(\frac{1}{R_P} + \frac{1}{R_G}\right)}$$

Table 14. Gear Efficiencies

External - External (Gear 1– Gear 3) Efficiency (η)	97%
External – Internal (Gear 3 - Gear 4) Efficiency (η)	98%
Total Efficiency (η)	95%

The new efficiency found is lower than the 97% desired due to the size of the teeth, as shown above in Table 14. The down of course teeth mean larger gears, adding bulk and weight to the system. They generally generate more noise and vibration as well as lower precision. However, coarse gear teeth have several advantages. Using larger sized teeth and robust tooth profile allow for a higher load capacity and increased durability under heavy stress, making them well-suited for demanding tasks. Therefore, although the efficiency is lowered the advantages out way the 2% lost.

13. Shaft Design and Load Analysis

The vertical height of the shaft from the platform is proposed to be 3.5 m. The length of the shaft connecting from the vertical shaft through bevel gears to the Ferris wheel is 0.8 m. The overall shaft assembly can be seen in Figure 27, showing locations of all gears, bearings, keys, and snap rings. Figure 27 below shows the cross-sectional view of the vertical shaft connecting to the horizontal shaft with the wheel, connected through bevel gears. The keys and keyways are shown on the horizontal shaft, one to connect to the bevel gear and the other to connect to the wheel. The key and keyway on the vertical shaft connect the vertical shaft to the bevel gear. The grooves on the horizontal shaft are indicated by the location of the snap rings to keep the elements in lateral place.

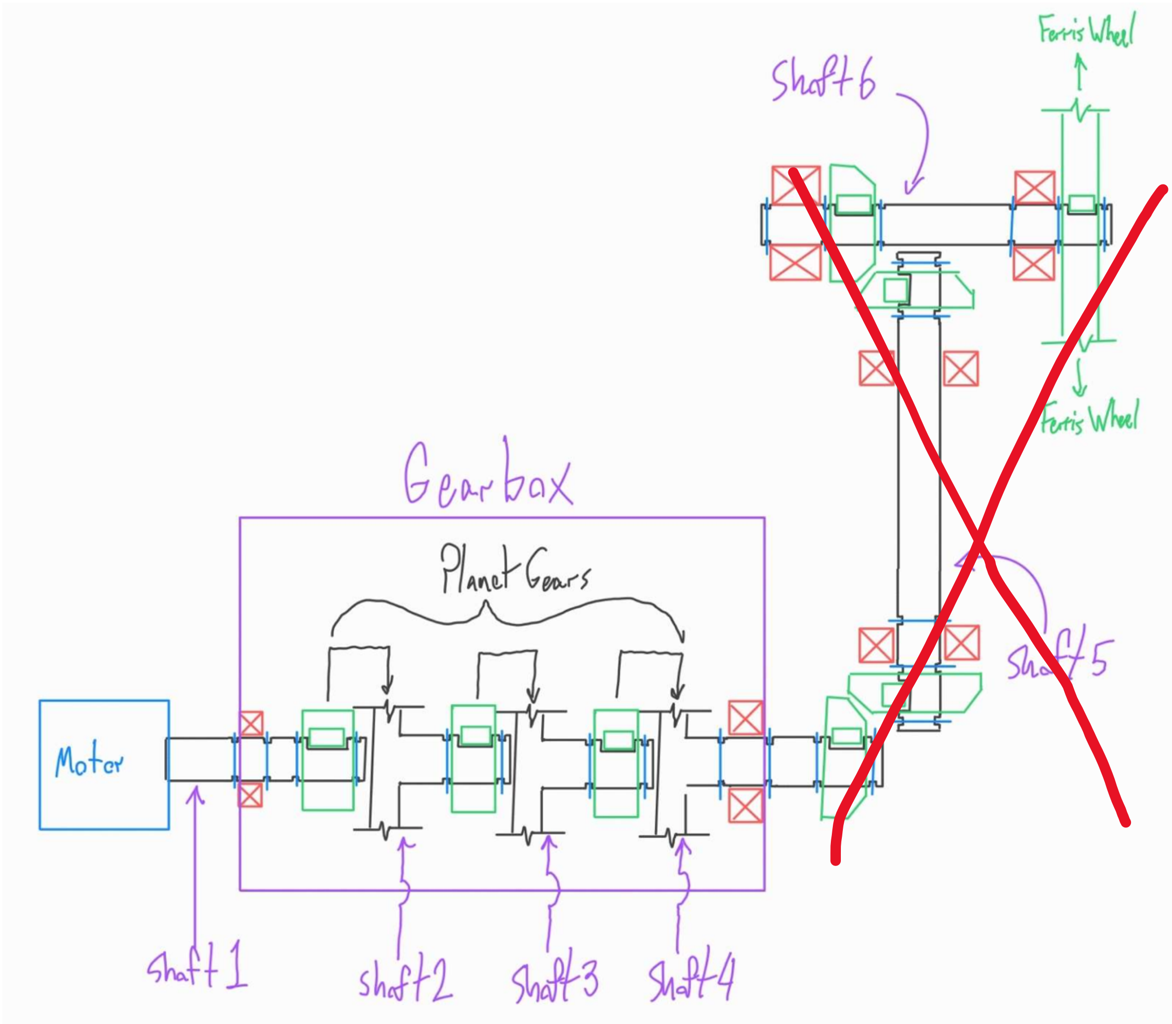


Figure 27. Shaft Assembly Details

A shear moment analysis was done for the output transmission shaft as it supports the weight of the Ferris wheel, for the intermediate vertical shaft and the gearbox shafts a torsional shear analysis was done, as they were not subjected to linear forces or moments. After determining the size of the gearbox, shaft 5 and shaft 6 are not needed, as the gearbox can be directly connected to the wheel of the Ferris wheel. The elimination of shaft 5 and shaft 6 lowers the cost and material used, as well as the potential failure of the other gears and shafts.

13.1 Determination of critical points of stress

The following point in Figure 28 was selected along a snap ring groove of each shaft as the most critical point of failure for the fatigue analysis. This point was selected after comparing the stress raisers due to the keyways, to the stress raisers due to the snap ring grooves, and the stress raisers due to changes in diameter along the shaft.

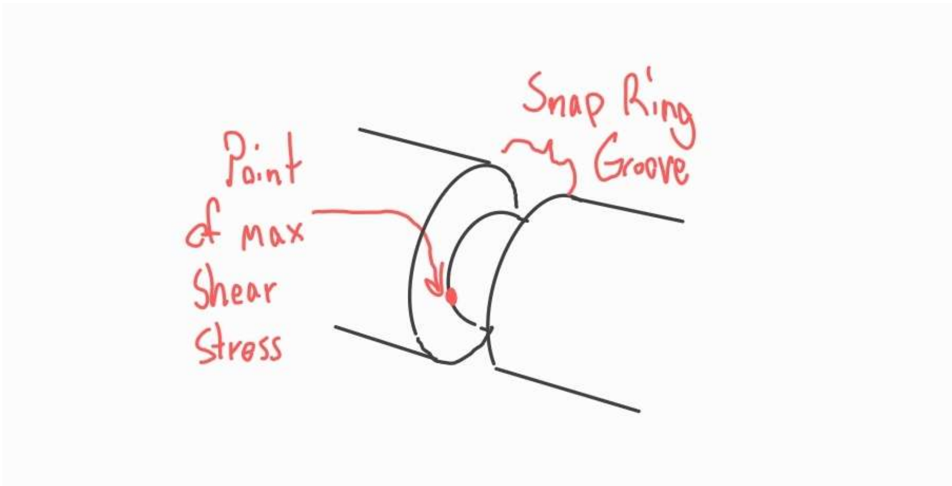


Figure 28. Critical Point of Failure

The rest of this section covers the fatigue analysis of each shaft the diameter of each shaft was the minimum value required to maintain the calculated safety factor above 3, note the torques used for this analysis are the static loading torques, not the torques induced from starting and stopping the Ferris wheel. A deeper analysis must be conducted later to account for the higher stresses at start up and stopping. The Following theory was used to calculate the following tabulated values:

The corrected bending stress is calculated from the equation below, where the moment of inertia must be calculated as well for the shape of the shaft. As the shafts are all cylindrical in nature, the equation is used for all shafts.

$$\sigma_{bend} = \frac{M \cdot y}{I} \cdot K$$

$$I = \frac{\pi D^4}{64}$$

The corrected shear stresses were calculated as:

$$\tau_{torsion} = \frac{T \cdot D}{2J} \cdot K$$

$$\tau_{shear} = \frac{V}{A} \cdot K$$

$$J = \frac{\pi D^4}{32}$$

The area calculated for the shear stress is shown below:

$$\text{Area} = \frac{\pi D^2}{4}$$

$$S_e = S_{ut} \times 0.5$$

The total shear stress was calculated:

$$\tau_{total} = \tau_{torsion} + \tau_{shear}$$

Since there are no loads acting in line with the shafts in our design, the Von mises stress is simplified as:

$$\sigma' = \sqrt{(\sigma_{bend})^2 + 3(\tau_{total})^2}$$

In order to design our shafts with respect to fatigue life, we calculated the amplitude Von Mises stress:

$$\sigma'_A = \sqrt{\left(\frac{\sigma_{bend,max} - \sigma_{bend,min}}{2}\right)^2 + 3\left(\frac{\tau_{total,max} - \tau_{total,min}}{2}\right)^2}$$

And the mean Von Mises stress:

$$\sigma'_m = \sqrt{\left(\frac{\sigma_{bend,max} + \sigma_{bend,min}}{2}\right)^2 + 3\left(\frac{\tau_{total,max} + \tau_{total,min}}{2}\right)^2}$$

We then determine the uncorrected endurance limit of steel and iron:

$$S'_{e(steel)} = S_{ut(steel)} \times 0.5$$

$$S'_{e(iron)} = S_{ut(iron)} \times 0.4$$

We determine all correction factors:

$$C_{load} = 1\ (Torsion\ \ and\ \ Bending)$$

$$C_{surf(steel)} = 0.62\ (Machined)$$

$$C_{surf(iron)} = 1.00\ (Machined)$$

$$C_{size} = 1.189 \cdot (D)^{-0.097}$$

$$C_{temp} = 1\ (T < 450\ \ Celsius)$$

$$C_{reliab} = 0.702\ (99.99\%\ \ Reliability)$$

Then we corrected the endurance limit:

$$S_e = S'_e \cdot C_{load} \cdot C_{surf} \cdot C_{size} \cdot C_{temp} \cdot C_{reliab}$$

We then calculate the factor of safety:

$$N_s = \frac{S_e \cdot S_{ut}}{\sigma'_A \cdot S_{ut} + \sigma'_m \cdot S_e}$$

Cast iron was chosen as the shaft material since material hardness is less important for shafts, and to reduce overall costs. Shaft diameters were selected to keep the safety factor above 3 as previously decided.

13.2 Shaft 1 Analysis and Details

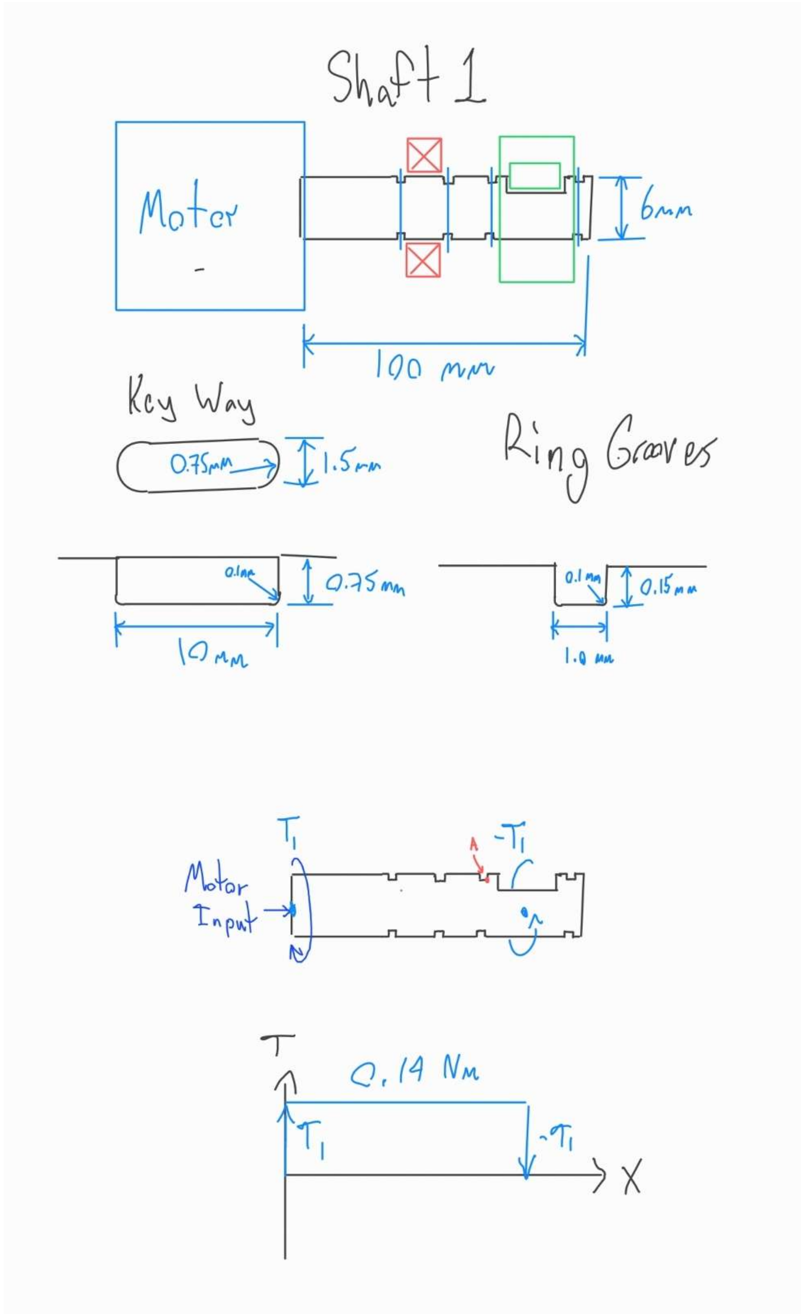


Figure 29: Shaft 1 FBD and Details

Table 15. Shaft Fatigue Analysis at Point A

Diameter	0.006	m^2
Area	0.00003	m^2

J_z	1.27E-10	m^4
V_max	0	N
V_min	0	N
T_xy_max	0.139620939	N*m
T_xy_min	-0.139620939	N*m
tau_xy_max	1.51E+07	Pa
tau_xy_min	-1.51E+07	Pa
tau_xy_m	0.00E+00	Pa
tau_xy_a	1.51E+07	Pa
sigma'_m	0.00E+00	Pa
sigma'_a	2.62E+07	Pa
K (flat groove)	4.6	
Sut	4.00E+08	Pa
Se'	1.60E+08	Pa
Cload	1	
Csize	0.8	
Csurf	1	
Ctemp	1	
Creliab	0.702	
Se	8.99E+07	Pa
Nf	3.43E+00	

13.3 Shaft 2 Analysis and Details

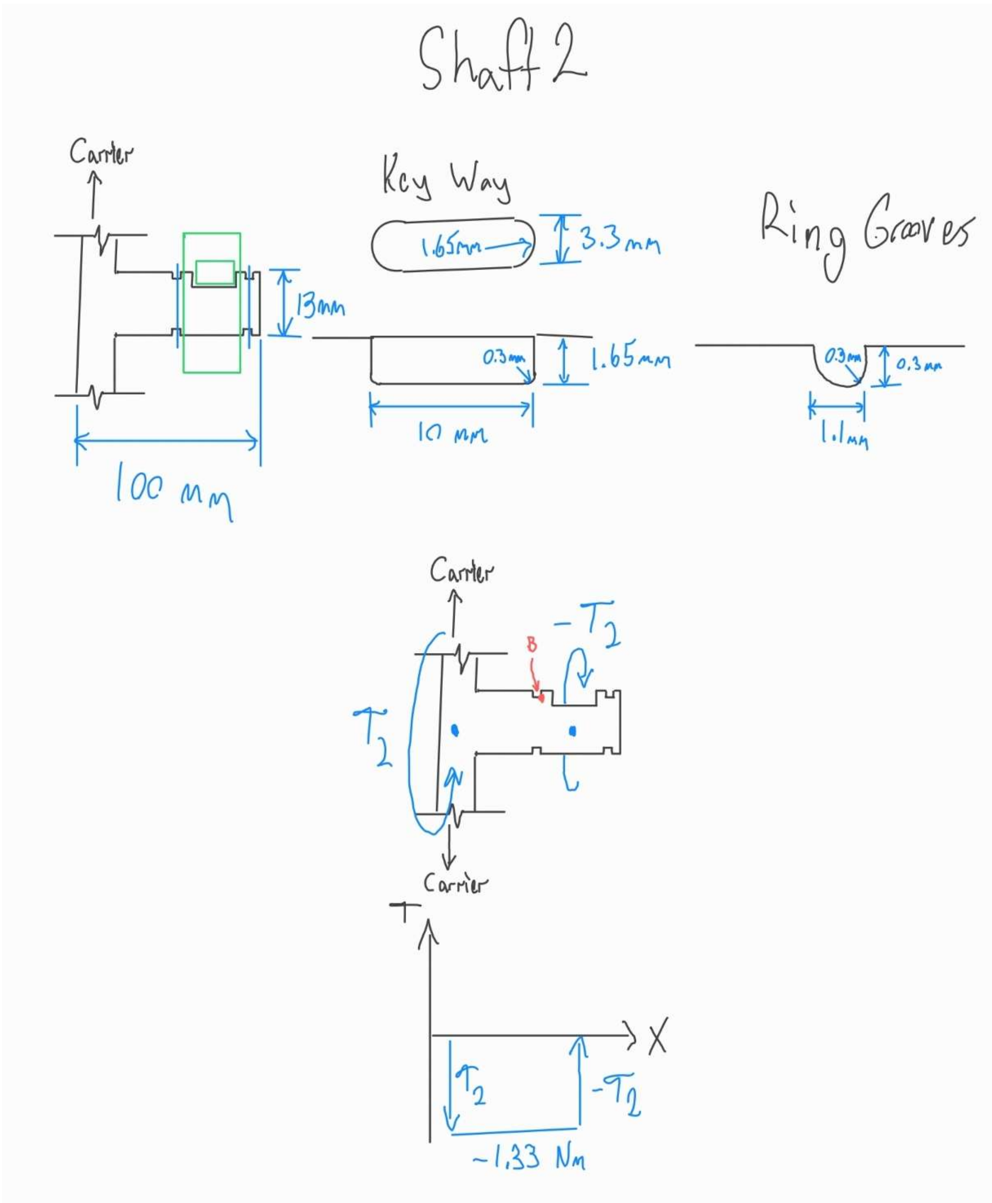


Figure 30. Shaft 2 FBD and Details

Table 16. Shaft Fatigue Analysis at Point B

Diameter	0.013	m^2
Area	0.00013	m^2
J_z	2.80E-09	m^4
V_max	0	N

V_min	0	N
T_xy_max	1.334776177	N*m
T_xy_min	-1.334776177	N*m
tau_xy_max	1.42E+07	Pa
tau_xy_min	-1.42E+07	Pa
tau_xy_m	0.00E+00	Pa
tau_xy_a	1.42E+07	Pa
sigma'_m	0.00E+00	Pa
sigma'_a	2.47E+07	Pa
K (flat groove)	4.6	
Sut	4.00E+08	Pa
Se'	1.60E+08	Pa
Cload	1	
Csize	0.8	
Csurf	1	
Ctemp	1	
Creliab	0.702	
Se	8.99E+07	Pa
Nf	3.64E+00	

13.4 Shaft 3 Analysis and Details

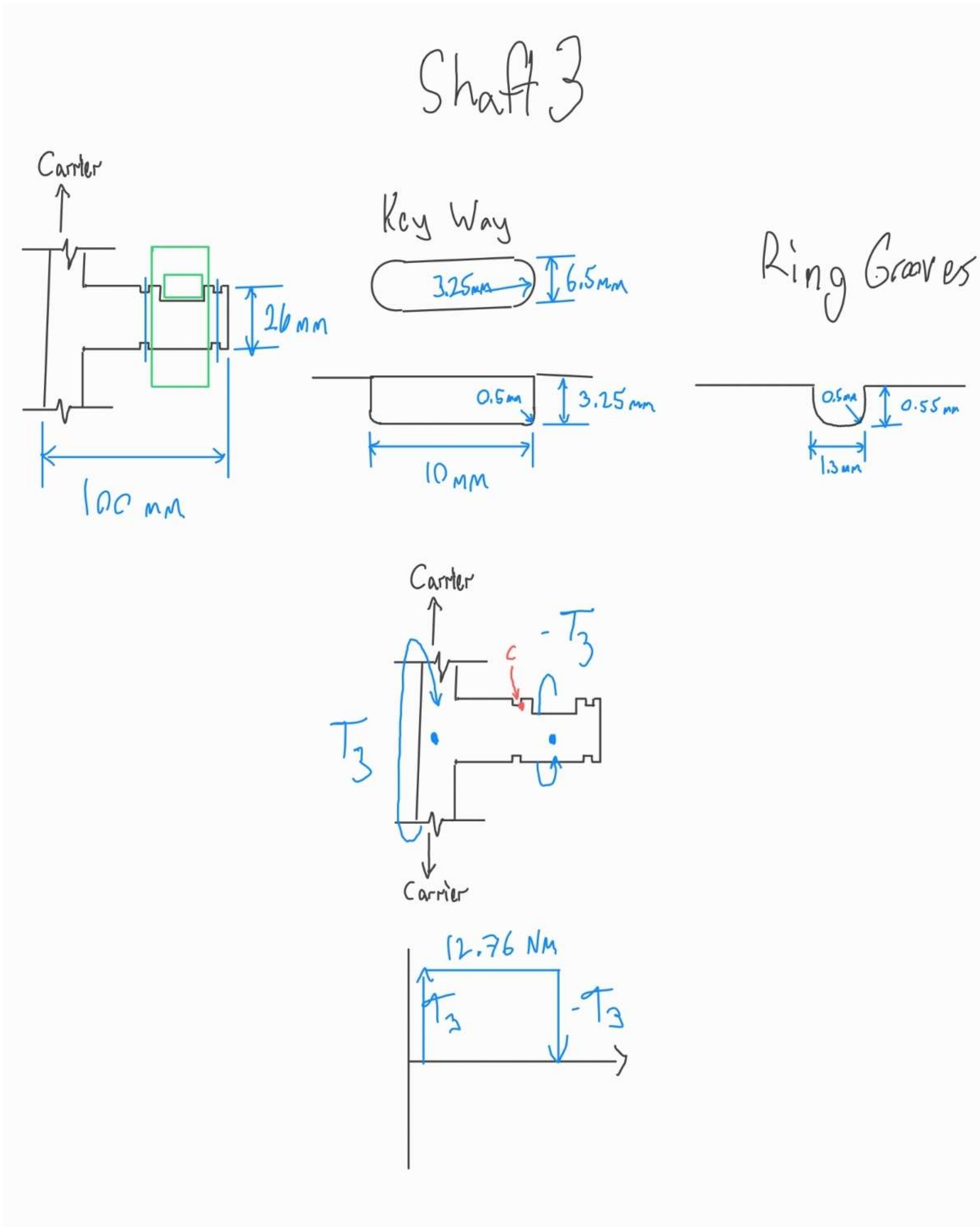


Figure 31. Shaft 3 FBD and Details

Table 17. Shaft Fatigue Analysis at Point C

Diameter	0.026	m^2
Area	0.00053	m^2

J_z	4.49E-08	m^4
V_max	0	N
V_min	0	N
T_xy_max	12.76046025	N*m
T_xy_min	-12.76046025	N*m
tau_xy_max	1.70E+07	Pa
tau_xy_min	-1.70E+07	Pa
tau_xy_m	0.00E+00	Pa
tau_xy_a	1.70E+07	Pa
sigma'_m	0.00E+00	Pa
sigma'_a	2.95E+07	Pa
K (flat groove)	4.6	
Sut	4.00E+08	Pa
Se'	1.60E+08	Pa
Cload	1	
Csize	0.8	
Csurf	1	
Ctemp	1	
Creliab	0.702	
Se	8.99E+07	Pa
Nf	3.05E+00	

13.5 Shaft 4 Analysis and Details

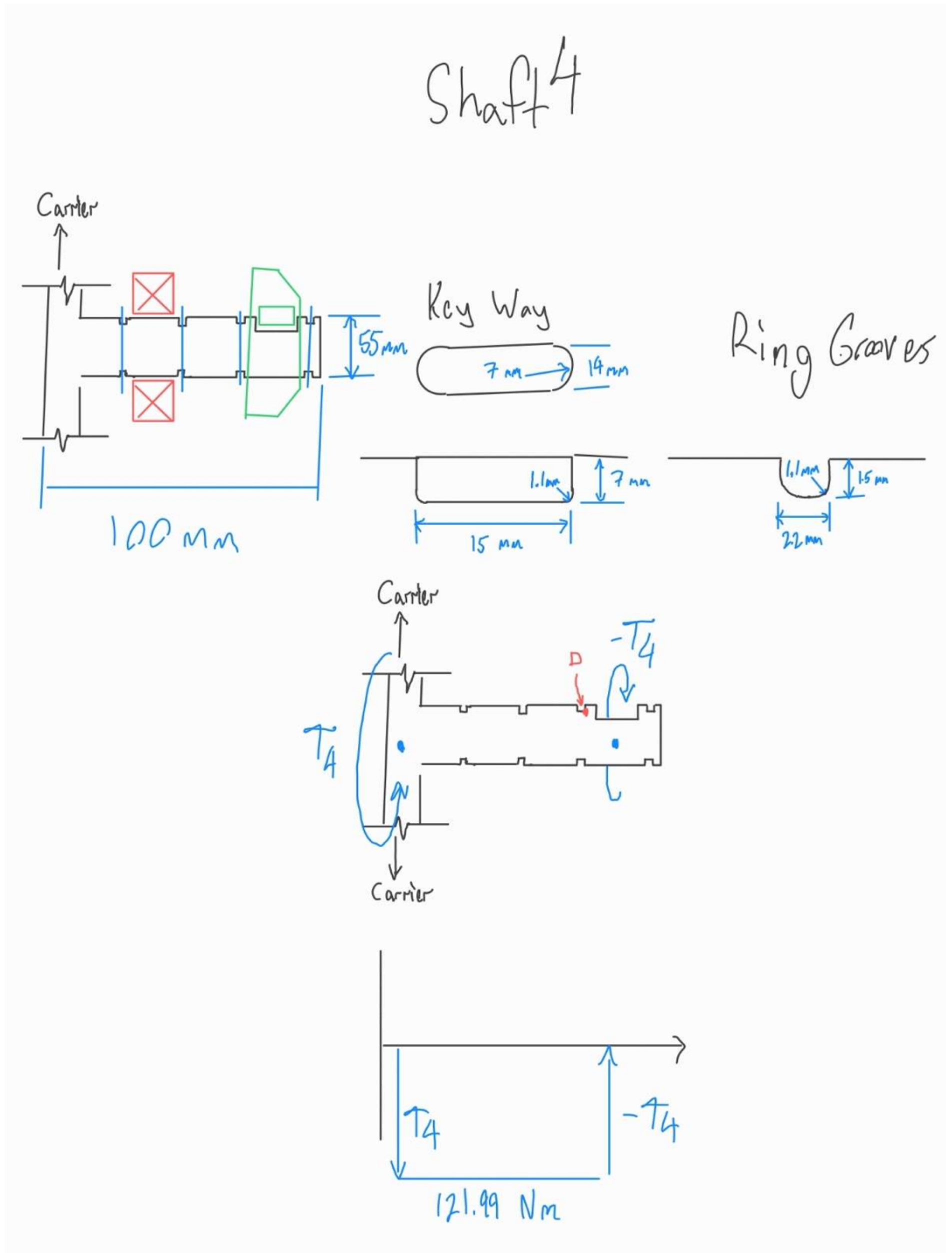


Figure 32. Shaft 4 FBD and Details

Table 18. Shaft Fatigue Analysis at Point D

Diameter	0.055	m^2
Area	0.00238	m^2
J_z	8.98E-07	m^4

V_max	0	N
V_min	0	N
T_xy_max	121.99	N*m
T_xy_min	-121.99	N*m
tau_xy_max	1.72E+07	Pa
tau_xy_min	-1.72E+07	Pa
tau_xy_m	0.00E+00	Pa
tau_xy_a	1.72E+07	Pa
sigma'_m	0.00E+00	Pa
sigma'_a	2.98E+07	Pa
K (flat groove)	4.6	
Sut	4.00E+08	Pa
Se'	1.60E+08	Pa
Cload	1	
Csize	0.8	
Csurf	1	
Ctemp	1	
Creliab	0.702	
Se	8.99E+07	Pa
Nf	3.02E+00	

13.6 Shaft 5 Analysis and Details – NOT NEEDED

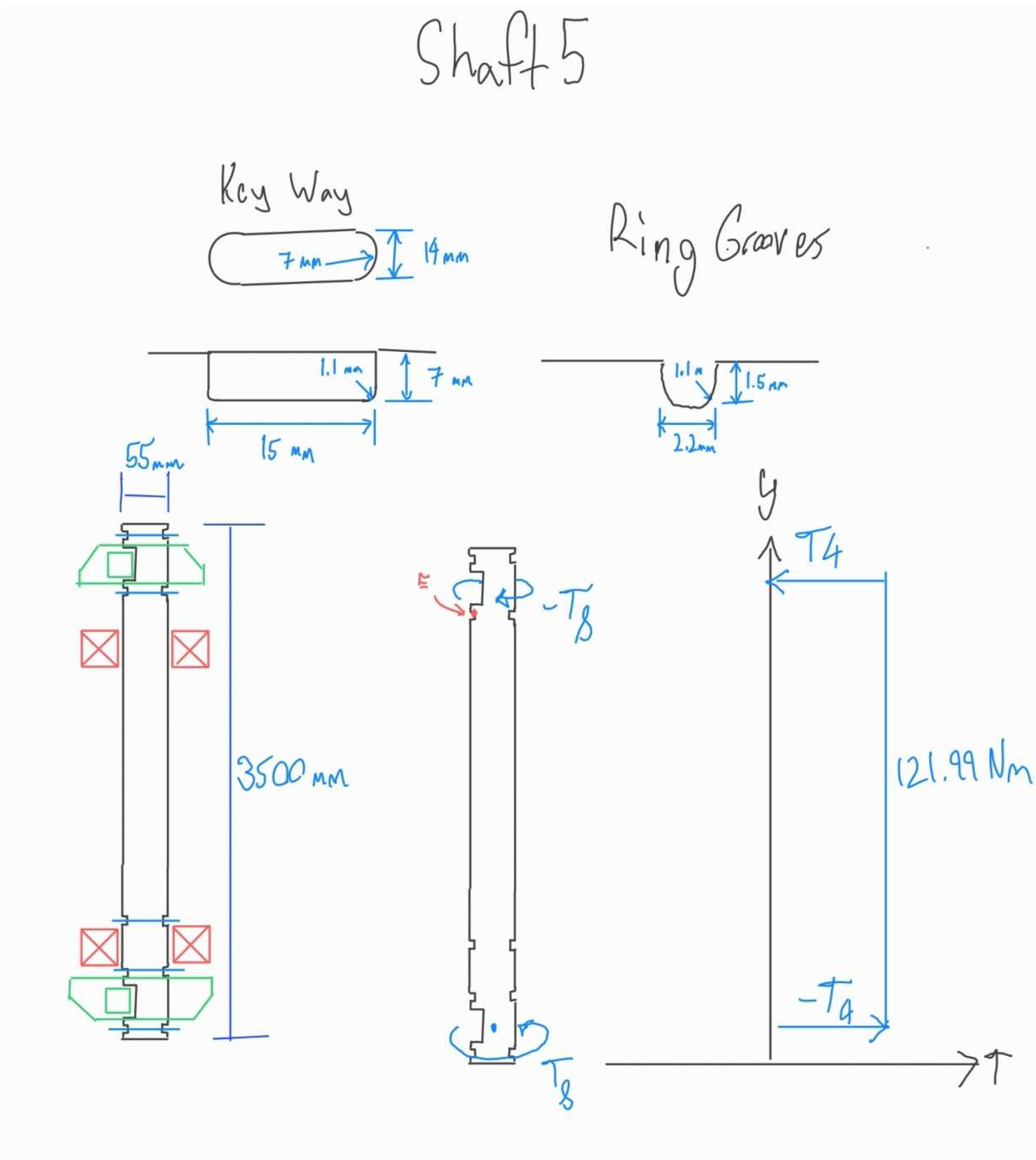


Figure 33. Shaft 5 FBD and Details

Table 19. Shaft Fatigue Analysis at Point E

Diameter	0.055	m^2
Area	0.00238	m^2
J_z	8.98E-07	m^4

V_max	0	N
V_min	0	N
T_xy_max	121.99	N*m
T_xy_min	-121.99	N*m
tau_xy_max	1.72E+07	Pa
tau_xy_min	-1.72E+07	Pa
tau_xy_m	0.00E+00	Pa
tau_xy_a	1.72E+07	Pa
sigma'_m	0.00E+00	Pa
sigma'_a	2.98E+07	Pa
K (flat groove)	4.6	
Sut	4.00E+08	Pa
Se'	1.60E+08	Pa
Cload	1	
Csize	0.8	
Csurf	1	
Ctemp	1	
Creliab	0.702	
Se	8.99E+07	Pa
Nf	3.02E+00	

13.7 Shaft 6 Analysis and Details – NOT NEEDED

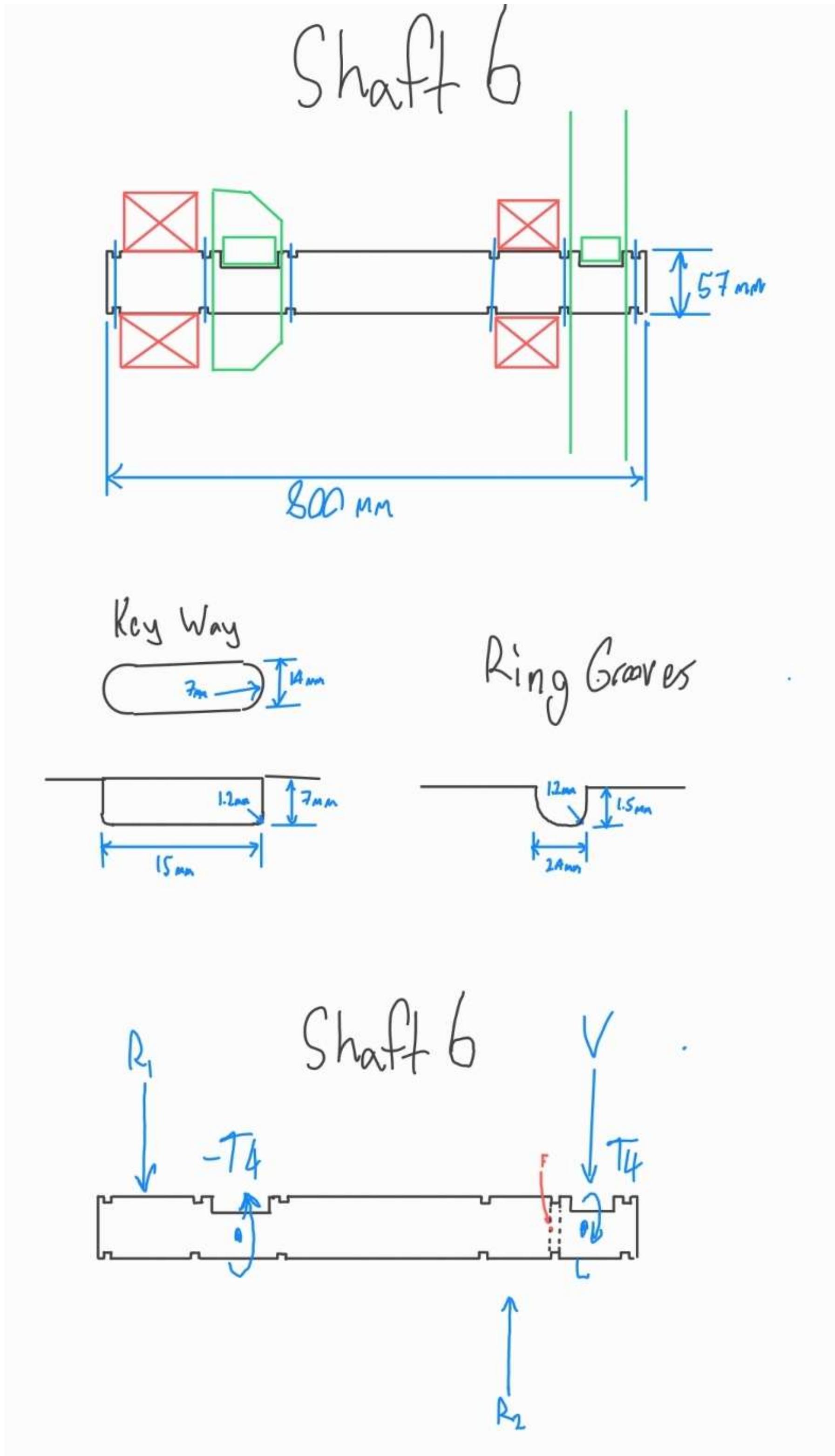


Figure 34. Shaft 6 FBD and Details Part 1

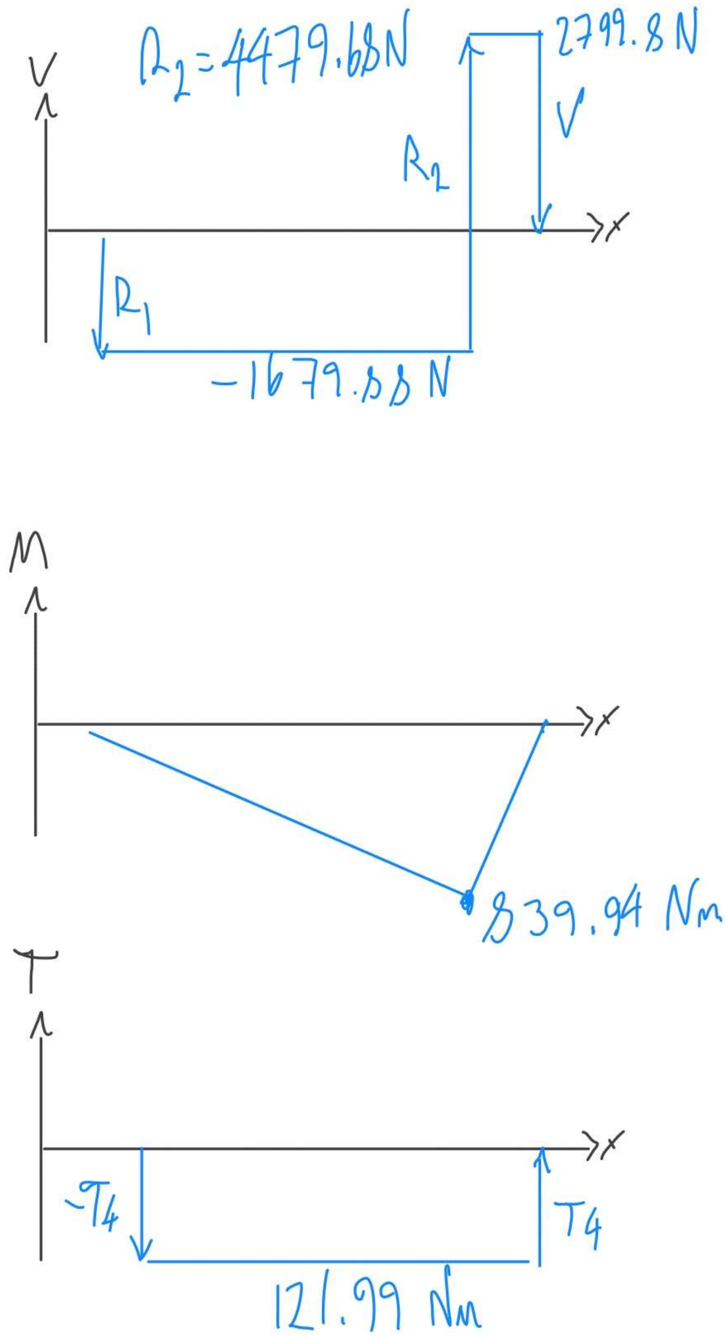


Figure 35. Shaft 6 FBD and Details Part 2

Table 20. Shaft Fatigue Analysis at Point F

Diameter		0.057	m^2
Area		0.00255	m^2
J_z		1.04E-06	m^4

V_max		2799.77	N
V_min		2799.77	N
T_xy_max		121.99	N*m
T_xy_min		-121.99	N*m
tau_xy_max		2.22E+07	Pa
tau_xy_min		-8.70E+06	Pa
tau_xy_m		6.73E+06	Pa
tau_xy_a		1.54E+07	Pa
sigma'_m		1.17E+07	Pa
sigma'_a		2.67E+07	Pa
K (flat groove)		4.6	
Sut		4.00E+08	Pa
Se'		1.60E+08	Pa
Cload		1	
Csize		0.8	
Csurf		1	
Ctemp		1	
Creliab		0.702	
Se		8.99E+07	Pa
Nf		3.06E+00	

14. Gear Design and Analysis

14.1 Gear Specs

The gear specifications are listed in Table 11.

14.2 Gear Failure Analysis

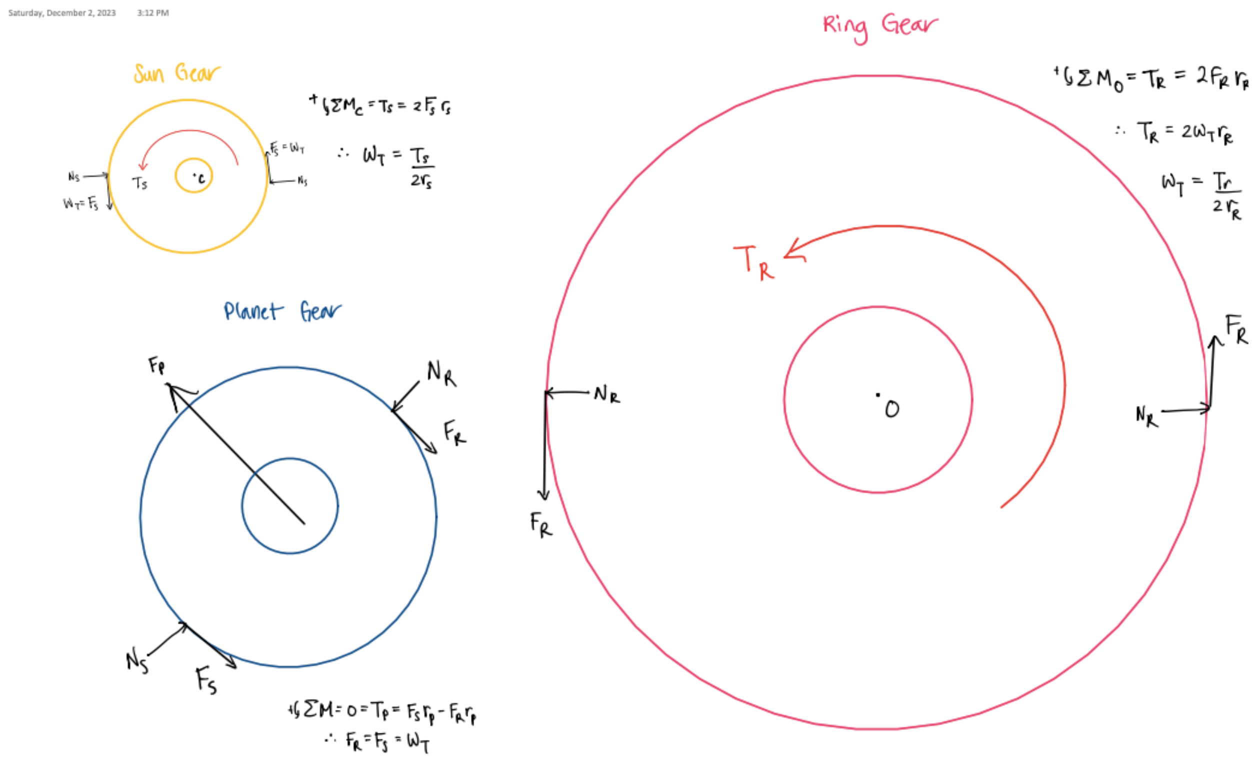


Figure 36. FBDs of Gears

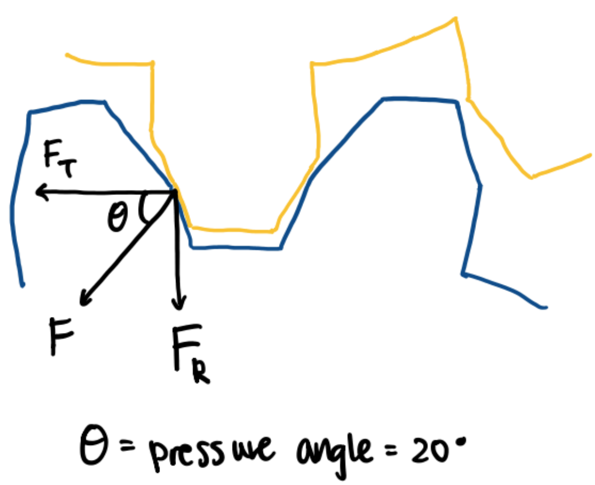


Figure 37. FBD of Teeth Contact

The gear failure analysis was performed for the bending stress, bending-fatigue stress, and the surface stress. Figure 36 shows the free body diagrams of the sun, planet, and ring gears and the equations to calculate the tangential forces experienced by each gear. Figure 37 above shows the free body diagram of the teeth coming into contact with full depth and a pressure angle of 20°. The tangential, radial, and resultant force act equal and opposite to the sun gear as well. The bending force is maximized when the loading occurs at the tip of the tooth.

14.3 Bending Stress

The equation used to calculate the bending stress in SI units is shown below.

$$\sigma_b = \frac{W_t}{FmJ}\frac{K_a K_m}{K_v}K_S K_B K_I$$

The bending strength geometry factor J is determined from the AGMA tables, assuming 20° pressure angle with full depth teeth, by comparing the number of teeth from the pinion and the gear.

The dynamic factor K_v is determined by first calculating the pitch-line velocity.

$$V_t = \omega_p * r_p$$

and then using the equation

SI units

$$K_v = \left(\frac{A}{A + \sqrt{200V_t}}\right)^B$$

$$A = 50 + 56(1 - B) \qquad B = \frac{(12 - Q_V)^{2/3}}{4} \quad \text{for } 6 \leq Q_v \leq 11$$

The Q_v value is determined from Table 11-6 Recommended AGMA Gear Quality Numbers for Various Applications. For the Ferris wheel application, the equivalent is the automotive transmission with a Q_v value of 10.

The load distribution factor K_m is determined from Table 12-16 Load Distribution Factors K_m, which takes into consideration the face width, calculated below.

$$F = \frac{12}{P_d}$$

The application factor K_a is based on the type of shocks experienced by the machine, according to Table 12-17. It is assumed that the driving machine is uniform, and the driven machine is uniform, resulting in the K_a factor as 1.0.

The size factor K_s is assumed to be 1, for general-sized teeth.

The rim thickness factor K_B is known to be 1.0 due to the large thickness of the gears compared to the depth of the teeth for all gears.

The bending fatigue strength, noted as S_{fb}, is the corrected bending fatigue strength calculated from the K factors, shown below.

$$S_{fb} = \frac{K_L}{K_T K_R} S_{fb}'$$

The uncorrected bending fatigue strength, S_{fb}', is obtained from Table 12-20 AGMA Bending-Fatigue Strengths for a Selection of Gear Materials. The gear material is selected as through-hardened steel, with a minimum surface hardness of 300 HB, which gives a bending fatigue strength of 250 MPa.

The life factor K_L is calculated from the equation below, which assumed commercial application and the required number of load cycles for 8 hours a day, 365 days, for 20 years.

$$K_L = 1.3558\, N^{-0.017}$$

The temperature factor K_T is assumed to be 1.0, as the climate will not be above 120°C.

The reliability factor K_R is assumed to be 1.0, with a reliability factor of 99% according to Table 12-19 AGMA Factor K_R.

The safety factor for the bending stress is calculated as below.

$$N_b = \frac{S_{fb}}{\sigma_b}$$

The calculated values are shown below in Table 21.

14.4 Surface Stress

The gears undergo surface stress as well as bending stress due to the friction of the gears moving against each other, causing pitting. The surface stress question is shown below.

$$\sigma_c = C_P \sqrt{\frac{W_t}{FId} \frac{C_a C_m}{C_v} C_s C_f}$$

The correction factors C_a, C_m, C_v, and C_s are equal to the parameters from the bending stress formula.

The correction factor C_f is 1.0, as the gear teeth surfaces are assumed to be smooth.

The elastic coefficient C_p considers the differences in elasticity between the gear teeth. The coefficients for different types of materials are listed in Table 12-18 AGMA Elastic Coefficient C_p. As the chosen material is steel, the C_p factor is 191 $MPa^{1/2}$.

The surface geometry factor I is calculated from the equation below.

$$I = \frac{\cos\phi}{\left(\frac{1}{\rho_p} \pm \frac{1}{\rho_g}\right) d_p}$$

where

$$\rho_g = (r_p + r_g)\sin\phi \mp \rho_p$$

and

$$\rho_p = \sqrt{\left(r_p + \frac{1 + x_p}{P_d}\right)^2 - (r_p \cos\phi)^2} - \frac{\pi}{P_d}\cos\phi$$

As full-depth teeth are being used, $x_p = 0$.

The surface fatigue strength is calculated below.

$$S_{fc} = \frac{C_L C_H}{C_T C_R} S_{fc}'$$

and C_T and C_R are equivalent to K_T and K_R. The uncorrected surface fatigue strength is 850 MPa, from the chosen material through-hardened steel with 300 HB surface hardness.

The correction factor C_L is the surface-life factor, which is calculated from the equation.

$$C_L = 1.4488\, N^{-0.023}$$

where N is the number of cycles required for 8 hours a day, 365 days, for 20 years.

The hardness-ratio factor C_H is calculated to be 1.0 from the equation.

$$C_H = 1 + A(m_G - 1)$$

The safety factor for the surface failure is calculated using the equation below.

$$N_c = \left(\frac{S_{fc}}{\sigma_c}\right)^2$$

The parameters used for calculating the safety factors for each gear are calculated and shown below in Table 21.

Table 21. Gear Failure Analysis for Stage 1,2,3

STAGE 1			
Bending Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
Tangential Force (Wt)	4.65	4.65	4.65
j	0.24	0.24	1.24
Vt (m/s)	5.50	2.82	0.00
Qv	10.00	10.00	10.00
A	83.78	83.78	83.78
B	0.40	0.40	0.40
Kv	0.88	0.91	1.00
F (m)	0.03	0.03	0.03
Km	2	2	2

Ka	1	1	1
Ks	1	1	1
Kb	1	1	1
K1	1.42	1.42	1.42
sigma_b (MPa)	0.838	0.811	0.142
Bending-Fatigue Strengths			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
sfb' (MPa)	250	250	250
number of cycles in 20 years	6.13E+09	8.56E+08	1.28E+09
Kl	0.91	0.94	0.93
Kt	1	1	1
Kr	1	1	1
Sfb (MPa)	226.94	235.04	233.35
Safety Factor			
Nb	270.79	289.96	1642.15
Surface Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
Cv	0.88	0.91	1.00
Diametral Pitch [m] - (dp)	0.0004	0.0004	0.0004
Cm	2	2	2
Ca	1	1	1
Cs	1	1	1
Cf	1	2	3
Pg	0.0189	0.0803	0.1043
Pp	0.0050	0.0188	0.0427
I	1.49E-06	5.72E-06	1.14E-05
Cp	191.0	191.0	191.0
sigma c (MPa)	17.0	6.3	3.5
Surface Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
CL	0.86	0.90	0.89
CH	1	1	1
CT	1	1	1
CR	1	1	1
Sfc' (MPa)	830	830	830
Sfc (MPa)	716.10	749.26	742.34
Safety Factor			

Nc	1776.5	14132.5	46228.9

STAGE 2			
Bending Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
Tangential Force (Wt)	44.49	44.49	44.49
j	0.24	0.24	1.24
Vt (m/s)	0.57	0.29	0.00
Qv	10.00	10.00	10.00
A	83.78	83.78	83.78
B	0.40	0.40	0.40
Kv	0.95	0.97	1.00
F (m)	0.03	0.03	0.03
Km	2	2	2
Ka	1	1	1
Ks	1	1	1
Kb	1	1	1
K1	1.42	1.42	1.42
sigma_b (MPa)	7.36	7.27	1.36
Bending-Fatigue Strengths			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
sfb'	250	250	250
number of cycles in 20 years	6.41E+08	8.95E+07	6.70E+07
Kl	0.95	0.98	0.98
Kt	1	1	1
Kr	1	1	1
Sfb (MPa)	236.25	244.68	245.94
Safety Factor			
Nb	32.08	33.66	181.01
Surface Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
Cv	0.95	0.97	1.00
Diametral Pitch [m] - (dp)	0.0004	0.0004	0.0004
Cm	2	2	2
Ca	1	1	1
Cs	1	1	1

Cf	1	2	3
Pg	0.0189	0.0803	0.1043
Pp	0.0050	0.0188	0.0427
I	1.49E-06	5.72E-06	1.14E-05
Cp	191.0	191.0	191.0
sigma c	50.4	18.9	10.7
Surface Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
CL	0.91	0.95	0.96
CH	1	1	1
CT	1	1	1
CR	1	1	1
Sfc' (MPa)	830	830	830
Sfc (MPa)	754.27	789.21	794.48
Safety Factor			
Nc	224.31	1748.55	5537.84

STAGE 3			
Bending Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
Tangential Force (Wt)	425.33	425.33	425.33
j	0.24	0.24	1.24
Vt (m/s)	0.06	0.03	0.00
Qv	10.00	10.00	10.00
A	83.78	83.78	83.78
B	0.40	0.40	0.40
Kv	0.98	0.99	1.00
F (m)	0.03	0.03	0.03
Km	2	2	2
Ka	1	1	1
Ks	1	1	1
Kb	1	1	1
K1	1.42	1.42	1.42
sigma_b	68.20	67.89	12.99
Bending-Fatigue Strengths			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
sfb'	250	250	250

number of cycles in 20 years	6.70E+07	9.36E+06	7.01E+06
Kl	0.98	1.02	1.02
Kt	1	1	1
Kr	1	1	1
Sfb (MPa)	245.94	254.71	256.03
Safety Factor			
Nb	3.61	3.75	19.71
Surface Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
Cv	0.98	0.99	1.00
Diametral Pitch [mm] - (dp)	0.0004	0.0004	0.0004
Cm	2	2	2
Ca	1	1	1
Cs	1	1	1
Cf	1	2	3
Pg	0.0189	0.0803	0.1043
Pp	0.0050	0.0188	0.0427
I	1.49E-06	5.72E-06	1.14E-05
Cp	191.0	191.0	191.0
sigma c (MPa)	153.3	57.7	33.0
Surface Stress			
	Gear 1 (Sun) - Input	Gear 3 (Planets)	Gear 4 (Ring)
CL	0.96	1.00	1.01
CH	1	1	1
CT	1	1	1
CR	1	1	1
Sfc'	830	830	830
Sfc (MPa)	794.48	831.28	836.83
Safety Factor			
Nc	26.87	207.70	642.71

As shown above in the Table 21 above, the safety factors are greater than the design objective of 3.0, therefore the gear design satisfy the requirements.

15. Shaft Design and Analysis

15.1 Final Shaft Design

The final shaft assembly was revised from the preliminary design, after determining the gearbox required was quite small, altered the transmission method so that the gear box will directly drive the Ferris wheel. This reduces the number of machined parts required, and the additional structure to support the gearbox and motor at a higher elevation is minimal due to the small size of the gearbox.

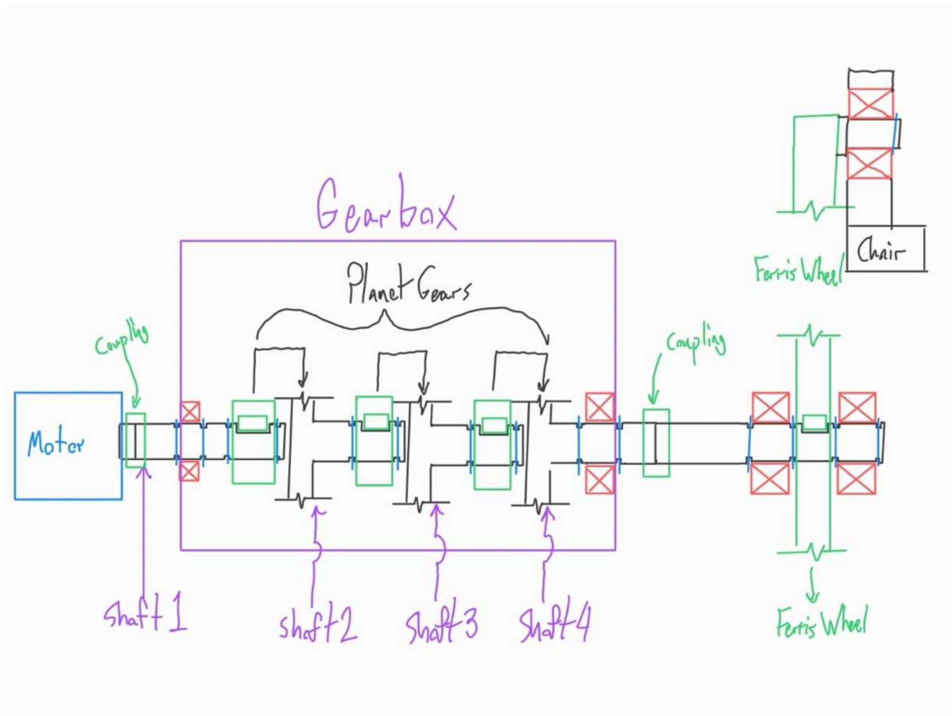


Figure 38. Revised shaft assembly design

The final shafts from shaft 1 through 4, and the Chair Hanger Shaft, and the Wheel Shaft are designed according to the specifications shown in Table 22 and Table 23.

Table 22. Fatigue Analysis with Torsion

Fatigue Analysis at Marked Critical Points						
	Shaft 1	Shaft 2	Shaft 3	Shaft 4	Shaft Wheel	
Diameter	0.009	0.013	0.016	0.055	0.055	m

Area	0.00006	0.00013	0.00020	0.00238	0.00238	m^2
J_z	6.44E-10	2.80E-09	6.43E-09	8.98E-07	8.98E-07	m^4
V_max	0	0	0	0	1400	N
V_min	0	0	0	0	1400	N
T_xy_max	0.139620939	1.334776177	12.76046025	121.99	121.99	N*m
T_xy_min	-0.139620939	-1.334776177	-12.76046025	-121.99	-121.99	N*m
tau_xy_max	4.39E+06	1.42E+07	7.30E+07	1.72E+07	2.03E+07	Pa
tau_xy_min	-4.39E+06	-1.42E+07	-7.30E+07	-1.72E+07	-1.33E+07	Pa
tau_xy_m	0.00E+00	0.00E+00	0.00E+00	0.00E+00	3.54E+06	Pa
tau_xy_a	4.39E+06	1.42E+07	7.30E+07	1.72E+07	1.68E+07	Pa
sigma'_m	0.00E+00	0.00E+00	0.00E+00	0.00E+00	6.12E+06	Pa
sigma'_a	7.60E+06	2.47E+07	1.26E+08	2.98E+07	2.91E+07	Pa
K (flat groove)	4.6	4.6	4.6	4.6	4.5	
Sut	4.00E+08	4.00E+08	1.40E+09	4.00E+08	4.00E+08	Pa
Se'	1.60E+08	1.60E+08	7.00E+08	1.60E+08	1.60E+08	Pa
Cload	1	1	1	1	1	
Csize	0.961	0.927	0.909	0.806	0.806	
Csurf	1	1	0.62	0.8	1	
Ctemp	1	1	1	1	1	
Creliab	0.702	0.702	0.702	0.702	0.702	
Se	1.08E+08	1.04E+08	2.77E+08	7.24E+07	9.05E+07	Pa
Nf	1.42E+01	4.22E+00	2.19E+00	2.43E+00	2.97E+00	

Table 23. Fatigue Analysis with Bending and Shear

Fatigue Analysis at Marked Critical Points									
	Stage 1 Planet Shafts		Stage 2 Planet Shafts		Stage 3 Planet Shafts		Chair Hanger Shafts		
	Point 1	Point 2	Point 1	Point 2	Point 1	Point 2	Point 1	Point 2	
Diameter	0.015	0.015	0.02	0.02	0.04	0.04	0.04	0.04	m
Area	0.00018	0.00018	0.00031	0.00031	0.00126	0.00126	0.00126	0.00126	m^2
J_z	4.97E-09	4.97E-09	1.57E-08	1.57E-08	2.51E-07	2.51E-07	2.51E-07	2.51E-07	m^4
V_max	9.3	0	88.98	0	850.66	0	807.36	0	N
V_min	-9.3	0	-88.98	0	-850.66	0	-807.36	0	N

T_xy_max	0	0	0	0	0	0	0	0	N*m
T_xy_min	0	0	0	0	0	0	0	0	N*m
M_x_max	0	0.14	0	1.33	0	12.76			N*m
M_x_min	0	-0.14	0	-1.33	0	-12.76			N*m
tau_xy_max	1.75E+05	0.00E+00	9.06E+05	0.00E+00	2.08E+06	0.00E+00	1.97E+06	0.00E+00	Pa
tau_xy_min	-1.75E+05	0.00E+00	-9.06E+05	0.00E+00	-2.08E+06	0.00E+00	-1.97E+06	0.00E+00	Pa
tau_xy_m	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	Pa
tau_xy_a	1.75E+05	0.00E+00	9.06E+05	0.00E+00	2.08E+06	0.00E+00	1.97E+06	0.00E+00	Pa
sigma_x_max	0.00E+00	1.69E+06	0.00E+00	4.06E+06	0.00E+00	4.67E+06	0.00E+00	4.43E+06	Pa
sigma_x_min	0.00E+00	-1.69E+06	0.00E+00	-4.06E+06	0.00E+00	-4.67E+06	0.00E+00	-4.43E+06	Pa
sigma'_m	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	0.00E+00	Pa
sigma'_a	3.04E+05	1.06E+06	1.57E+06	4.06E+06	3.60E+06	4.67E+06	3.41E+06	4.43E+06	Pa
K	2.5	2.5	2.4	2.4	2.3	2.3	2.3	2.3	
Sut	4.00E+08	4.00E+08	4.00E+08	4.00E+08	4.00E+08	4.00E+08	4.00E+08	4.00E+08	Pa
Se'	1.60E+08	1.60E+08	1.60E+08	1.60E+08	1.60E+08	1.60E+08	1.60E+08	1.60E+08	Pa
Cload	1	1	1	1	1	1	1	1	
Csize	0.9143258586	0.9143258586	0.8891641518	0.8891641518	0.8313464287	0.8313464287	0.8313464287	0.8313464287	
Csurf	1	1	1	0.62134	1	1	1	1	
Ctemp	1	1	1	1	1	1	1	1	
Creliab	0.702	0.702	0.702	0.702	0.702	0.702	0.702	0.702	
Se	1.03E+08	1.03E+08	9.99E+07	6.21E+07	9.34E+07	9.34E+07	9.34E+07	9.34E+07	Pa
Nf	3.38E+02	9.72E+01	6.36E+01	1.53E+01	2.60E+01	2.00E+01	2.74E+01	2.11E+01	

15.2 Shaft Failure Analysis

The shaft failure analysis was performed using the stress-life analysis at each critical point noted on the shafts within the free body diagrams:

Figure 39. Wheel Shaft FBD

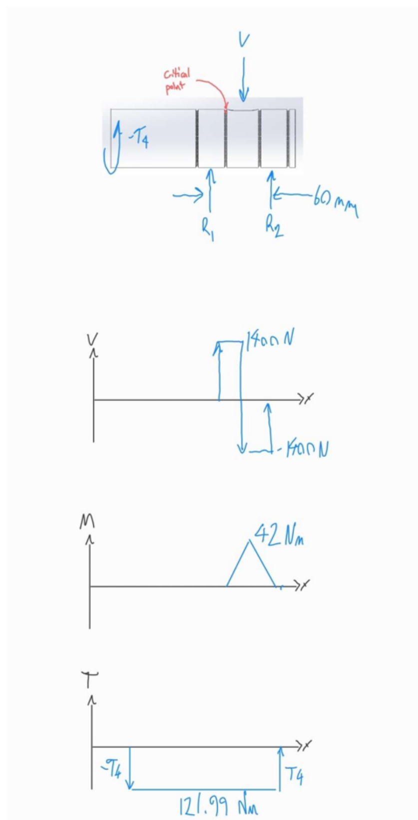


Figure 40. Chair Hanger Shaft FBD

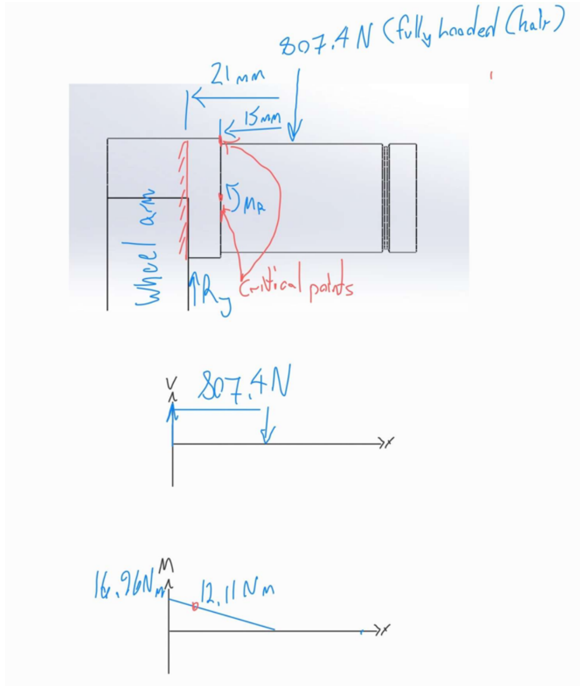


Figure 41. Shaft 1 FBD

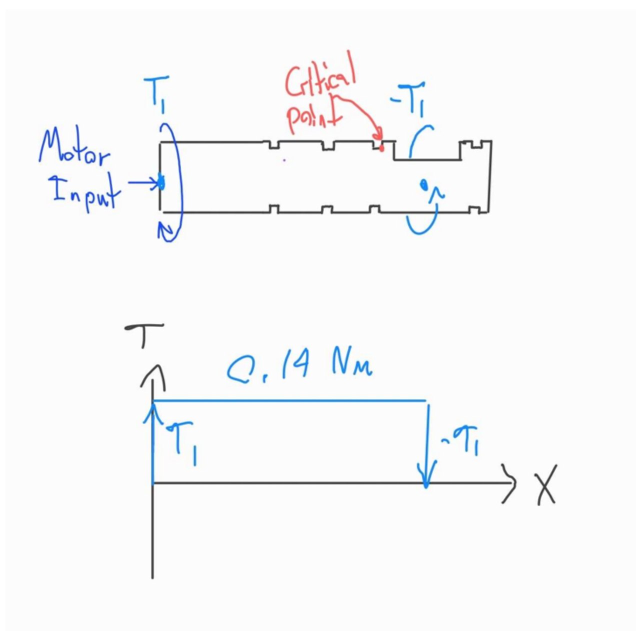


Figure 42. Shaft 2 FBD

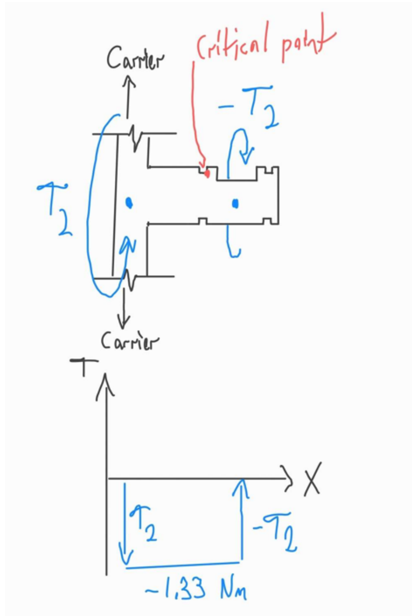


Figure 43. Shaft 3 FBD

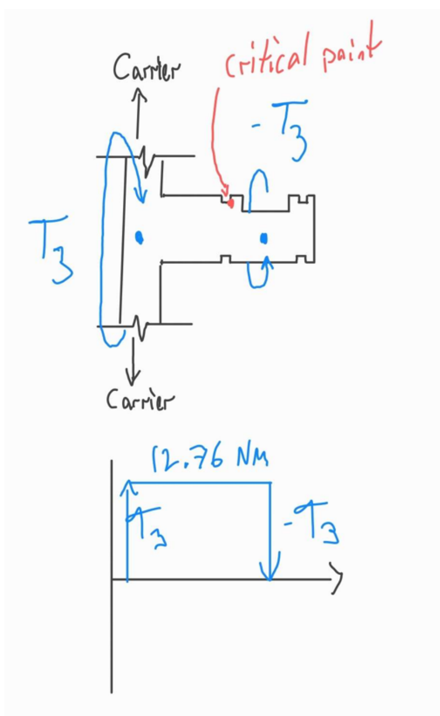


Figure 44. Shaft 4 FBD

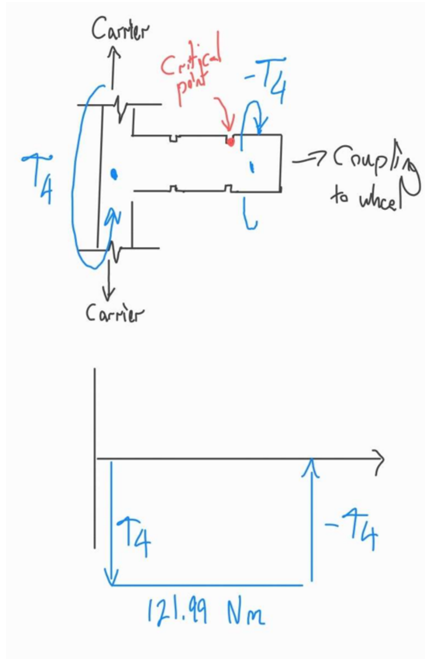


Figure 45. Stage 1 Planet Gear Shaft FBD

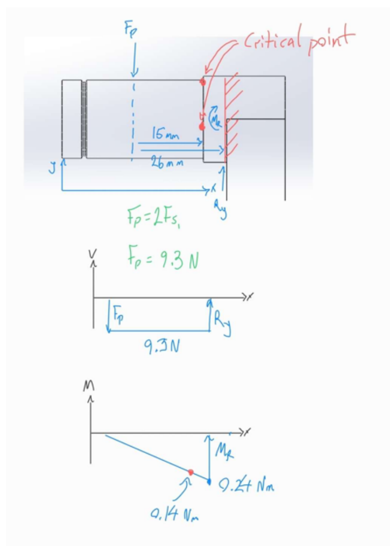


Figure 46. Stage 2 Planet Gear Shaft FBD

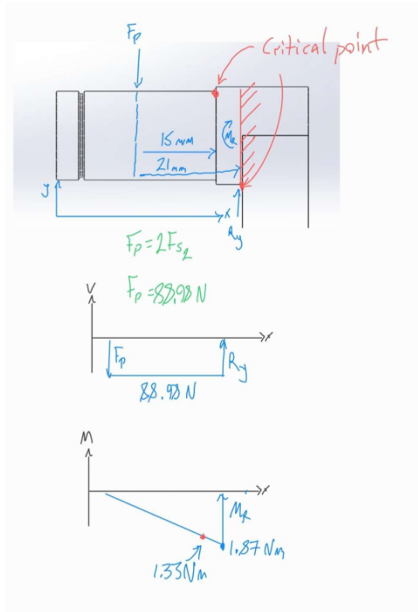
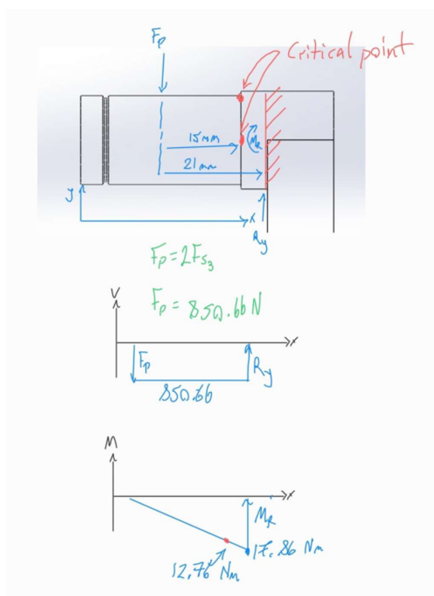
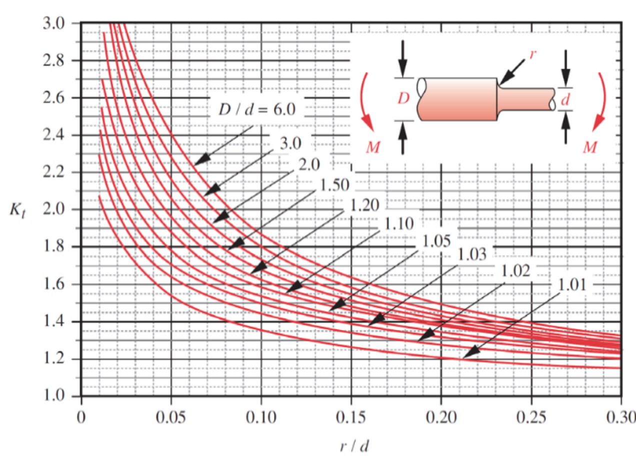


Figure 47. Stage 3 Planet Gear Shaft FBD



A K factor for each critical point was determined from the following charts:



$K_t \cong A\left(\frac{r}{d}\right)^b$

where :

D / d	A	b
6.00	0.878 68	–0.332 43
3.00	0.893 34	–0.308 60
2.00	0.908 79	–0.285 98
1.50	0.938 36	–0.257 59
1.20	0.970 98	–0.217 96
1.10	0.951 20	–0.237 57
1.07	0.975 27	–0.209 58
1.05	0.981 37	–0.196 53
1.03	0.980 61	–0.183 81
1.02	0.960 48	–0.177 11
1.01	0.919 38	–0.170 32

Figure 48. Shoulder Fillet in Bending stress concentration

Figure A–15–17

Round shaft with flat-bottom groove in torsion.

$\tau_0 = \dfrac{16T}{\pi d^3}$

Source: W. D. Pilkey, *Peterson's Stress-Concentration Factors*, 2nd ed. John Wiley & Sons, New York, 1997, p. 133

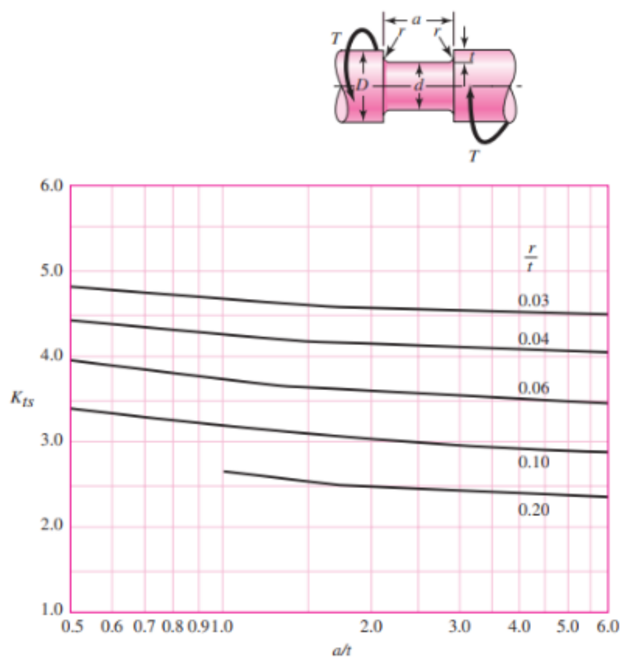
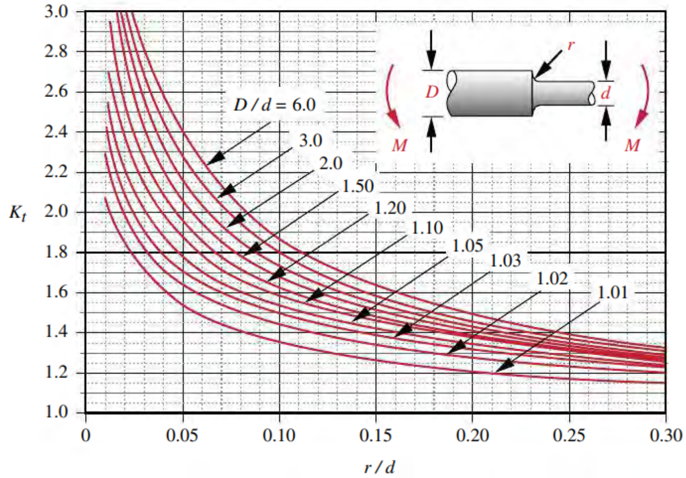


Figure 49. Snap-Ring Groove stress concentration



$K_t \cong A\left(\frac{r}{d}\right)^b$

where:

D/d	A	b
6.00	0.878 68	−0.332 43
3.00	0.893 34	−0.308 60
2.00	0.908 79	−0.285 98
1.50	0.938 36	−0.257 59
1.20	0.970 98	−0.217 96
1.10	0.951 20	−0.237 57
1.07	0.975 27	−0.209 58
1.05	0.981 37	−0.196 53
1.03	0.980 61	−0.183 81
1.02	0.960 48	−0.177 11
1.01	0.919 38	−0.170 32

Figure 50. Change in shaft diameter stress concentration

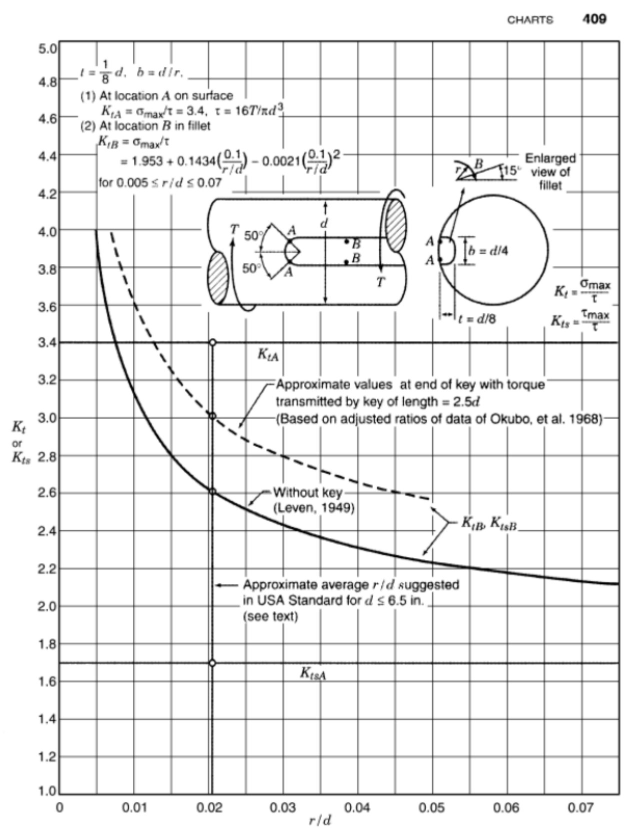


Chart 5.2 Stress concentration factors K_t, K_{ts} for a torsion shaft with a semicircular end keyseat (Leven 1949; Okubo et al. 1968).

Figure 51. Gear keyway stress concentration

The corrected bending stress is calculated from the equation below, where the moment of inertia must be calculated as well for the shape of the shaft. As the shafts are all cylindrical in nature, the equation is used for all shafts.

$$\sigma_{bend} = \frac{M \cdot y}{I} \cdot K$$

$$I = \frac{\pi D^4}{64}$$

The corrected shear stresses were calculated as:

$$\tau_{torsion} = \frac{T \cdot D}{2J} \cdot K$$

$$\tau_{shear} = \frac{V}{A} \cdot K$$

$$J = \frac{\pi D^4}{32}$$

The area calculated for the shear stress is shown below:

$$S_e = S_{ut} \times 0.5$$

$$\text{Area} = \frac{\pi D^2}{4}$$

$$\tau_{total} = \tau_{torsion} + \tau_{shear}$$

Since there are no loads acting in line with the shafts in our design, the Von mises stress is simplified as:

$$\sigma' = \sqrt{(\sigma_{bend})^2 + 3(\tau_{total})^2}$$

In order to design our shafts with respect to fatigue life, we calculated the amplitude Von Mises stress:

$$\sigma'_A = \sqrt{\left(\frac{\sigma_{bend,max} - \sigma_{bend,min}}{2}\right)^2 + 3\left(\frac{\tau_{total,max} - \tau_{total,min}}{2}\right)^2}$$

And the mean Von Mises stress: $\sigma'_m = \sqrt{\left(\frac{\sigma_{bend,max}+\sigma_{bend,min}}{2}\right)^2 + 3\left(\frac{\tau_{total,max}+\tau_{total,min}}{2}\right)^2}$

We then determine the uncorrected endurance limit of steel and iron:

$$S'_{e(steel)} = S_{ut(steel)} \times 0.5$$

$$S'_{e(iron)} = S_{ut(iron)} \times 0.4$$

We determine all correction factors:

$$C_{load} = 1\ (Torsion\ and\ Bending)$$

$$C_{surf(steel)} = 0.62\ (Machined)$$

$$C_{surf(iron)} = 1.00\ (Machined)$$

$$C_{size} = 1.189 \cdot (D)^{-0.097}$$

$$C_{temp} = 1\ (T < 450\ Celsius)$$

$$C_{reliab} = 0.702\ (99.99\%\ Reliability)$$

Then we corrected the endurance limit:

$$S_e = S'_e \cdot C_{load} \cdot C_{surf} \cdot C_{size} \cdot C_{temp} \cdot C_{reliab}$$

We then calculate the factor of safety:

$$N_s = \frac{S_e \cdot S_{ut}}{\sigma'_A \cdot S_{ut} + \sigma'_m \cdot S_e}$$

Every shaft was assessed at its critical points and designed so the factor of safety never fell below 2. We did not meet our goal of a safety factor of 3 with every component, however this goal can be met or changed with further revisions. As revisions to the design were made, some shafts were either increased in diameter, reduced in diameter or their material was changed from iron to steel in order to maintain a high safety factor. Our safety factor was prioritized above weight, size, and cost for the design of the shafts.

Each gear key was also verified to meet our safety factor goal given the maximum shear stress due to torsion applied by the gear:

$$N_s = \left(\frac{T_{max}}{0.5 \cdot D}\right) \cdot \frac{1}{Area_{key}} \cdot S_{ut}$$

16. Bearing Design and Analysis

16.1 Bearing Selection

The bearing type selected is roller bearings due to their sustainability in high-load applications, which the Ferris wheel application would be categorized under. Specifically, the type of bearing used for the input shaft coming into the gearbox housing would be a deep groove ball bearing, due to the high speed and lower tolerances experienced by the input shaft in the first stage relative to the subsequent bearings. The bearing for the planet gears and the output shaft in the 1st stage are selected to be the cylindrical ball bearings due to the higher loads and moderate speeds expected from the shafts. In the 2nd and 3rd stages, the bearings in the planet gears are selected to be cylindrical bearings, and the last bearing is a spherical bearing for their large load capacity. The bearing types and specifications are listed below Table 24. The bearings were selected from the SKF catalogue [18].

Table 24. Bearing Specifications for Stage 1,2,3

STAGE 1	Gear 1 (Sun) - Input	Gear 3 (Planets)
Stage 1 Angular Speed [RPM] - (w)	1750	-244.4
Stage 1 Torques [N.m] - (T)	0.14	0.51
Shaft Diameter [m]	0.009	0.015
Radial Load P [kN]	0.03	0.03
Bearing Type	Deep groove ball bearing	Single row cylindrical roller bearing
Refrence Code	W 618/9	NJ 202 CEP
URL	https://www.skf.com/ca/en/p	https://www.skf.com/ca/en/products
Outside Diameter [mm]	17	35
Width [mm]	4	11
Dynamic Load Rating C	0.761	12.5
C/P	24.53080229	366.3063298
L10 = (C/P)^n; n=3 ball; n=3/10 roller	14761.66199	351680329.6
L10 E06	1.48E+10	3.52E+14
Desired Lifetime	6.13E+09	8.56E+08

STAGE 2	Gear 3 (Planets)
Stage 1 Angular Speed [RPM] - (w)	-25.55
Stage 2 Torques [N.m] - (T)	4.89
Shaft Diameter [m]	0.02
Radial Load P [kN]	0.24
Bearing Type	Single row cylindrical roller bearing
Refrence Code	NJ 204 ECP
URL	https://www.skf.com/ca/en/product
Outside Diameter [mm]	47
Width [mm]	14
Dynamic Load Rating C	28.5
C/P	116.4628
L10 = (C/P)^n; n=3 ball; n=3/10 roller	7714197.57
L10 E06	7.71E+12
Desired Lifetime	8.95E+07

STAGE 3	Gear 3 (Planets)	Gear 2 (Carrier) - Output
Stage 3 Angular Speed [RPM] - (w)	-2.67	2
Stage 3 Torques [N.m] - (T)	46.79	121.99
Shaft Diameter [m]	0.04	0.055
Radial Load P [kN]	1.17	2.22
Bearing Type	Single row cylindrical roller bearing	Spherical roller bearing
Refrence Code	NU 1008 ML	22211 EE
URL	https://www.skf.com/ca/en/products/ro	https://www.skf.com/ca/en/produc
Outside Diameter [mm]	68	100
Width [mm]	15	25
Dynamic Load Rating C	28.5	129
C/P	24.36591622	58.10810811
L10 = (C/P)^n; n=3 ball; n=3/10 roller	41938.13902	759957.1747
L10 E06	4.19E+10	7.60E+11
Desired Lifetime	9.36E+06	7.01E+06

16.2 Bearing Analysis

The lifetime cycles of the bearings were calculated using the formula

$$L_{10} = \left(\frac{C}{P}\right)^n * 10^6$$

Where n = 3 for ball bearings and n = 10/3 for roller bearings. The dynamic load ratings C are obtained from the website SKF catalogue where the bearings were selected, based on the required diameters. The radial load P is calculated from

$$P = \frac{Torque}{radius}$$

The lifetime cycles are compared to the desired lifetimes based on the rotational speed of the gears and the required operational time of 20 years. The results are posted above in Table 24. From the observed lifetime of the bearings, it could be inferred that the bearings for the all planets in all stages and the output carrier bearing can be changed for cheaper bearings as the lifetime of the chosen bearings are higher than the required lifetime based on a 20-year operational period. However, because costs are not included in the scope, the bearings chosen are suitable for the application.

17. Final Gearbox Design

17.1 Specifications and Drawings of Gears, Shafts, Bearings

The specifications of the gears are shown in Table 11 and the drawings are in Appendix L. The specifications and failure analysis of the shafts are shown in Table 22 and Table 23, and the drawings are shown in Appendix L. The bearing specifications are shown in Table 24 and the printouts are shown in Appendix L. The Bill of Materials is shown in Appendix K Bill of Materials.

The overall gearbox assembly is shown in Figure 52 and Figure 53.

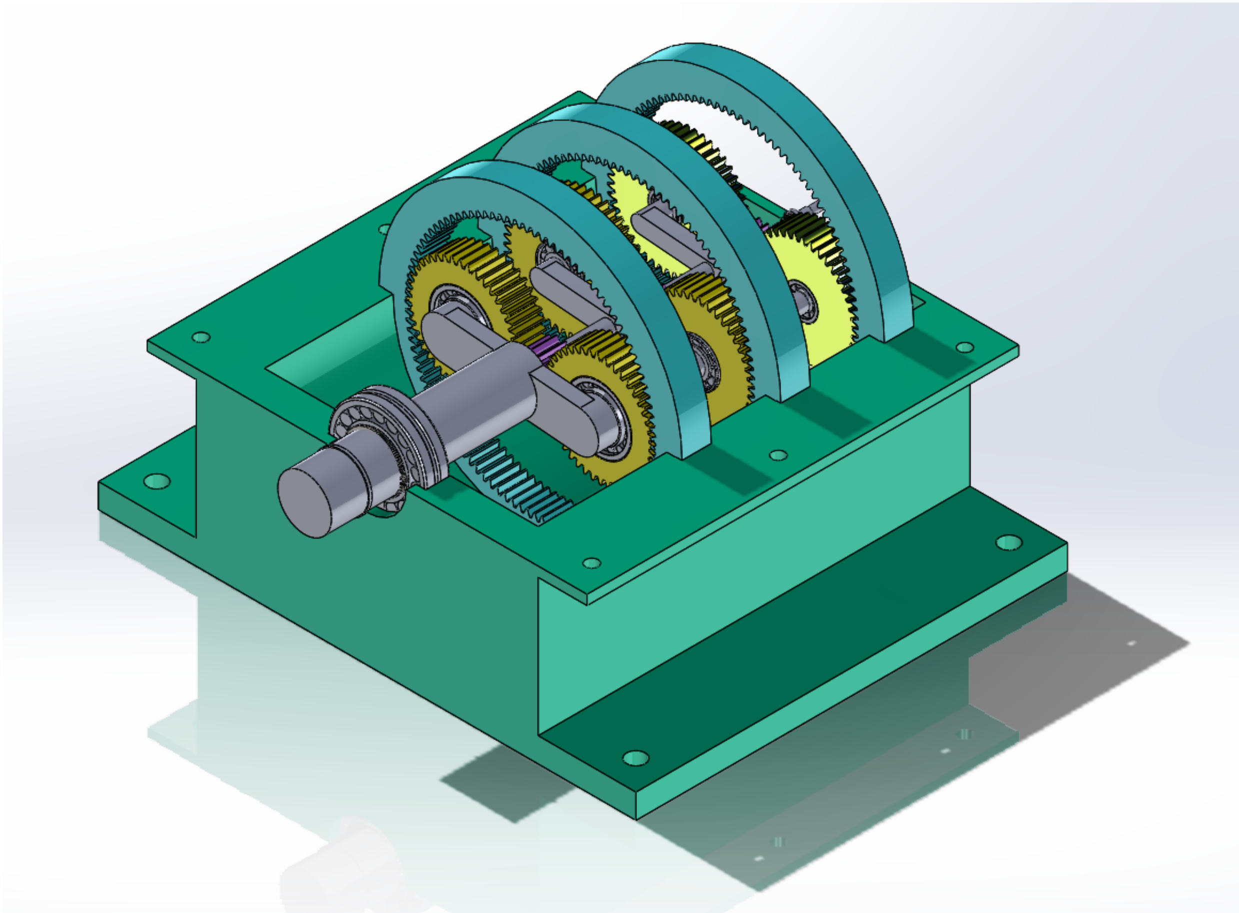


Figure 52. Front Isometric View of Gearbox Assembly

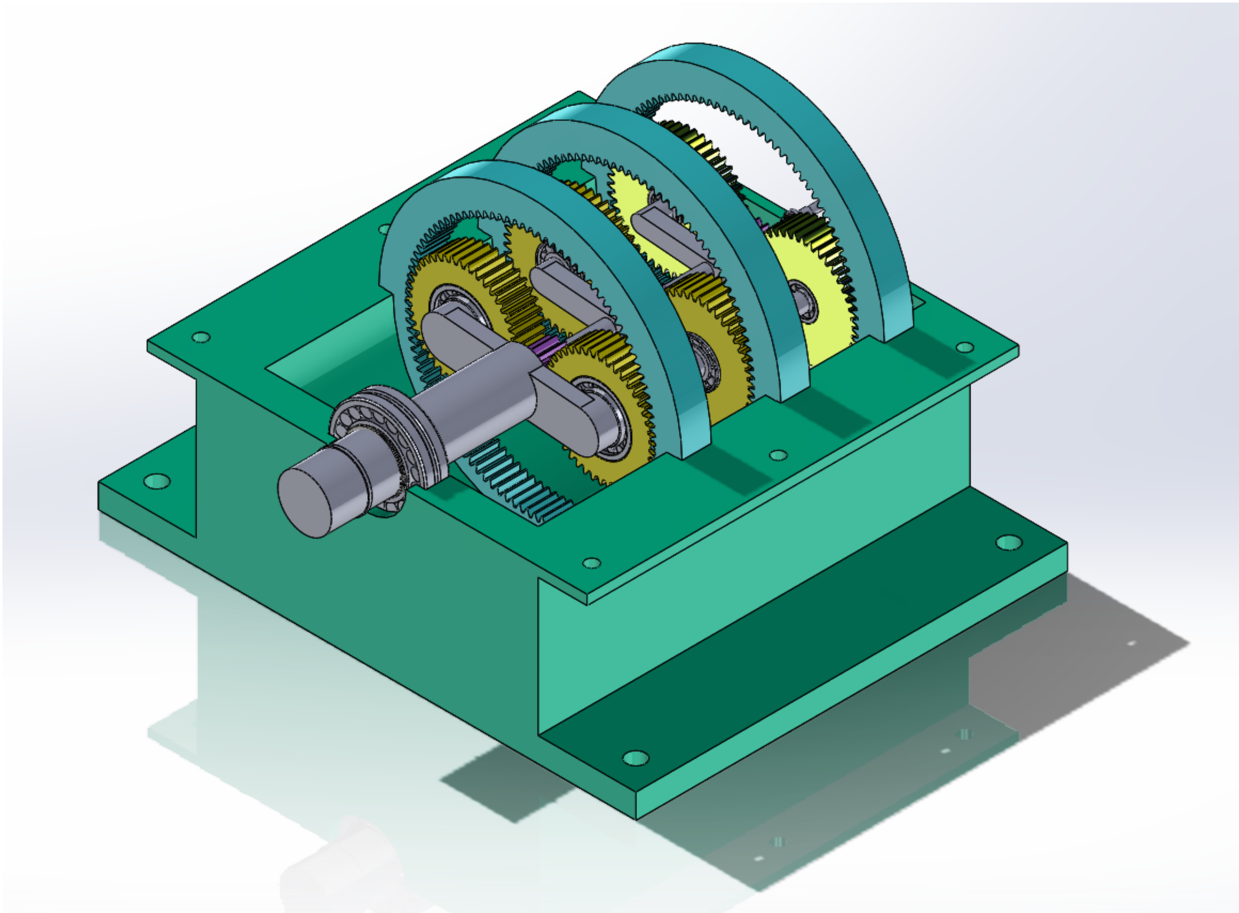


Figure 53. Back Isometric View of Gearbox Assembly

17.2 Gearbox Assembly Process

Step 1: Prepare Stage 3 Carrier

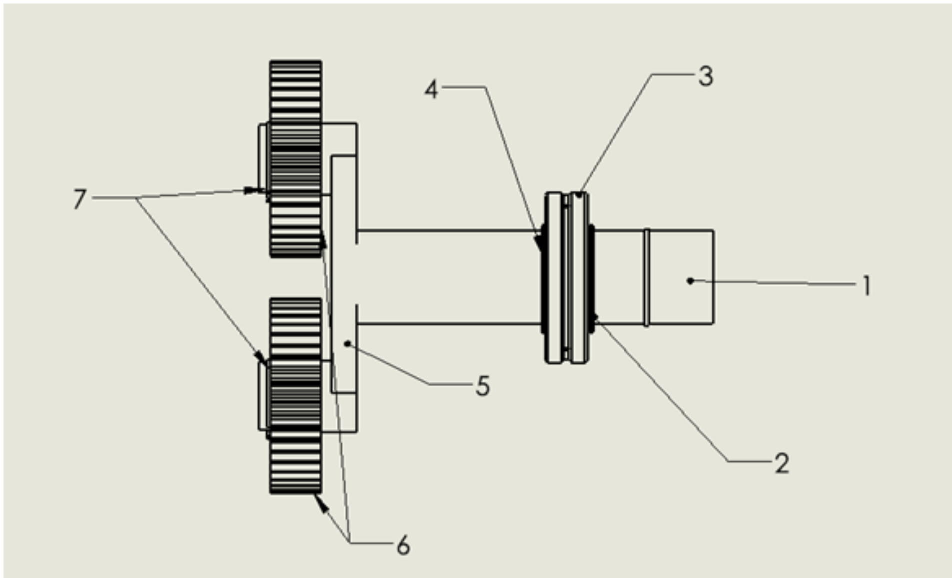


Figure 54. Assembly Process of Stage 3 Carrier

1. Retrieve shaft 4.
2. Install locating snap ring for bearing.
3. Slide bearing into place.
4. Install retaining snap ring for bearing.
5. Fasten shaft 4 carrier to end of shaft.
6. Slide planet gear bearings onto carrier, then slide planet gears onto bearings.
7. Install retaining snap ring for bearing & gear.

Step 2: Prepare Shaft 3

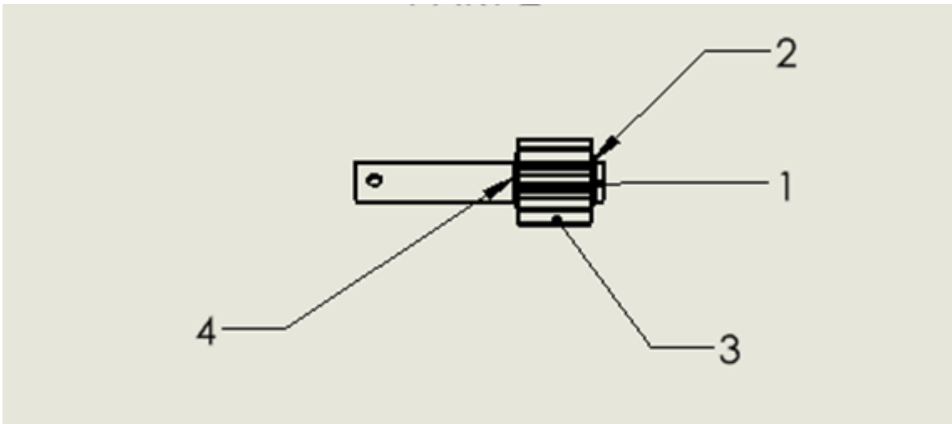


Figure 55. Assembly Process of Shaft 3

1. Retrieve shaft 3.
2. Install locating snap ring for gear.
3. Insert shaft 3 key into slot, then slide sun gear on until it stops at the snap ring.
4. Install retaining snap ring for gear.

Step 3: Assemble Stage 3

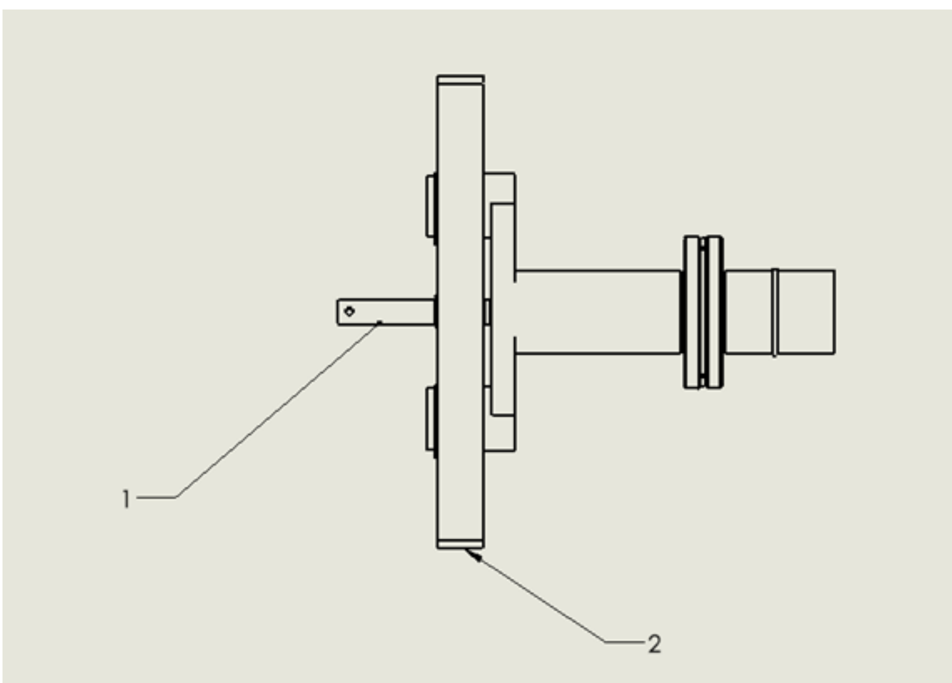


Figure 56. Assembly Process of Stage 3

1. Connect assemblies from step 2 and step 3 by sliding the sun gear between both planet gears, ensure the shafts are aligned.
2. Slide a ring gear over the planet gears

Step 4: Prepare Stage 2 Carrier

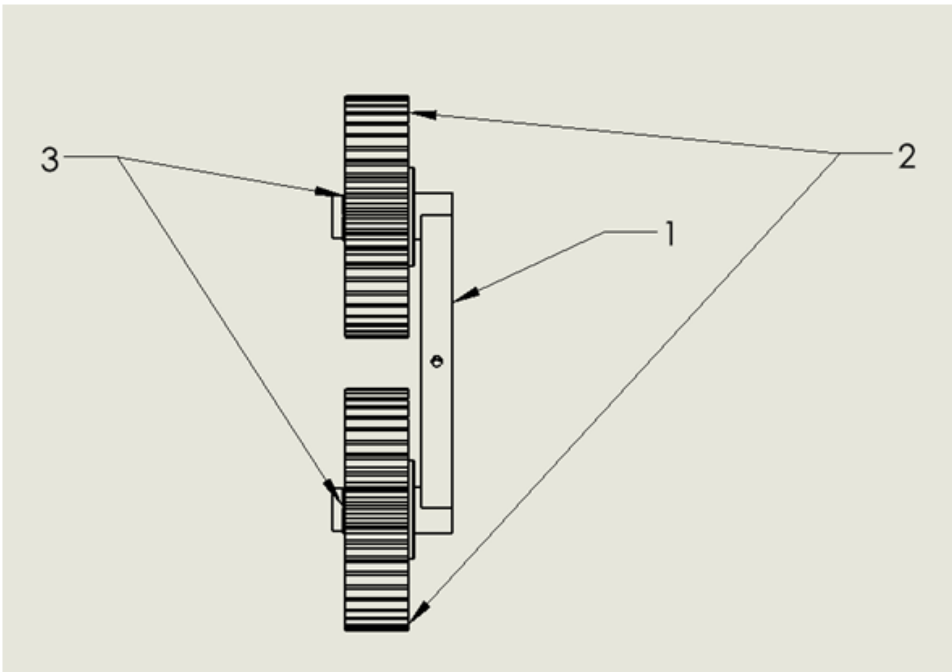


Figure 57. Assembly Process of Stage 2 Carrier

1. Retrieve shaft 3 carrier.
2. Slide planet gear bearings onto carrier, then slide planet gears onto bearings.
3. Install retaining snap rings for bearings & gears.

Step 5: Prepare Shaft 2

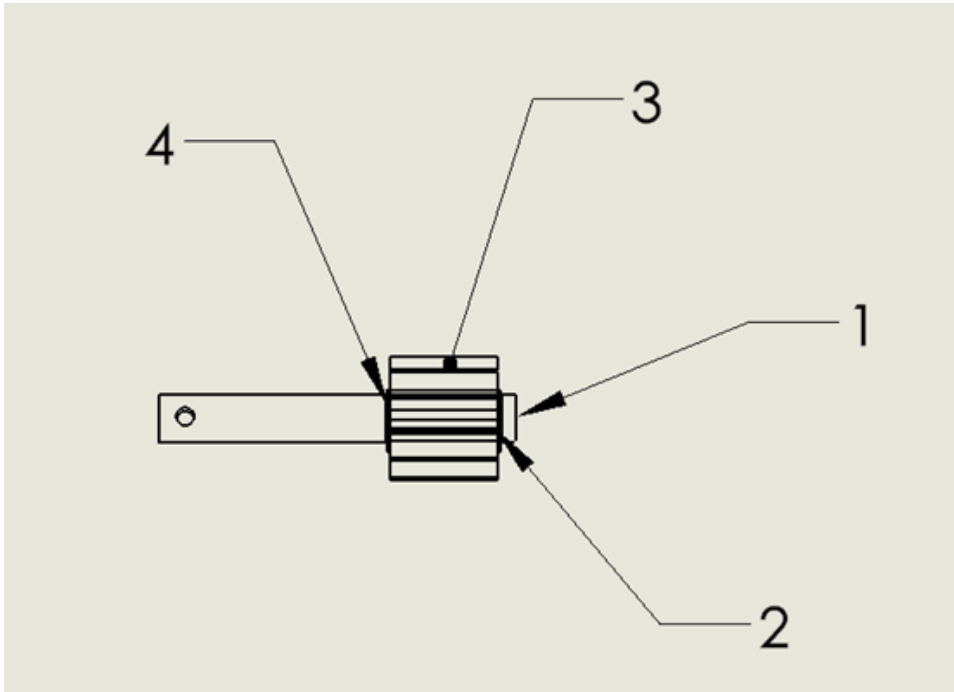


Figure 58. Assembly Process of Shaft 2

1. Retrieve shaft 2.
2. Install locating snap ring for gear.
3. Insert shaft 2 key into slot, then slide sun gear on until it stops at the snap ring.
4. Install retaining snap ring for gear.

Step 6: Assemble Stage 2

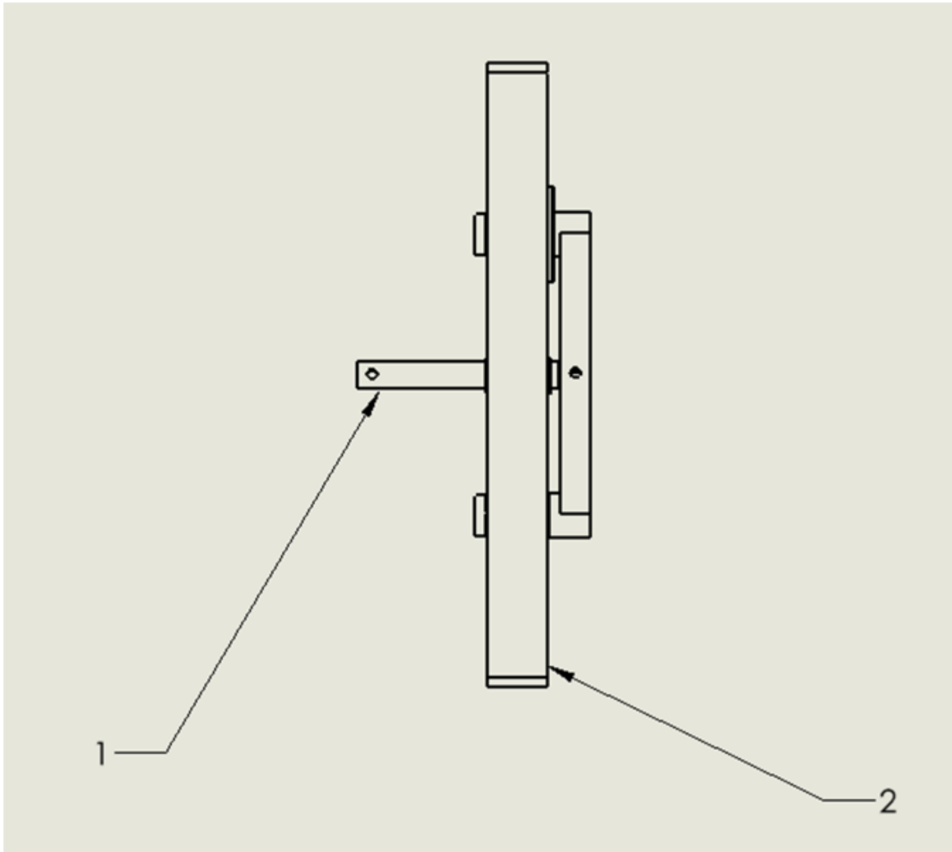


Figure 59. Assembly Process of Stage 2

1. Connect assemblies from step 4 and step 5 by sliding the sun gear between both planet gears, ensure the shafts are aligned.
2. Slide a ring gear over the planet gears.

Step 7: Prepare Stage 1 Carrier

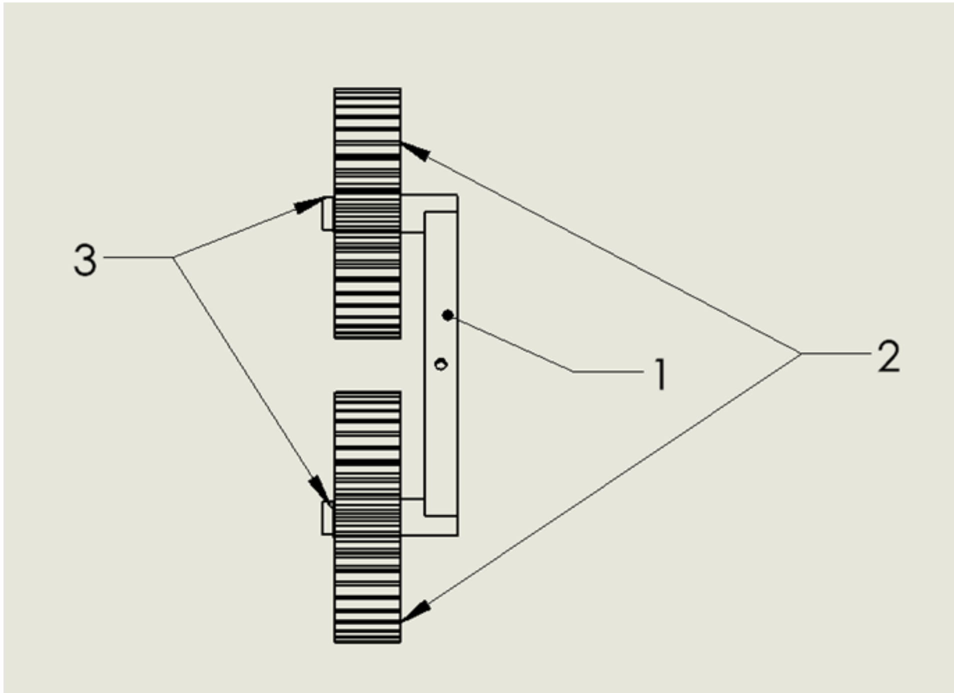


Figure 60. Assembly Process of Stage 1 Carrier

1. Retrieve shaft 2 carrier.
2. Slide planet gear bearings onto carrier, then slide planet gears onto bearings.
3. Install retaining snap rings for bearings & gears.

Step 8: Prepare Shaft 1

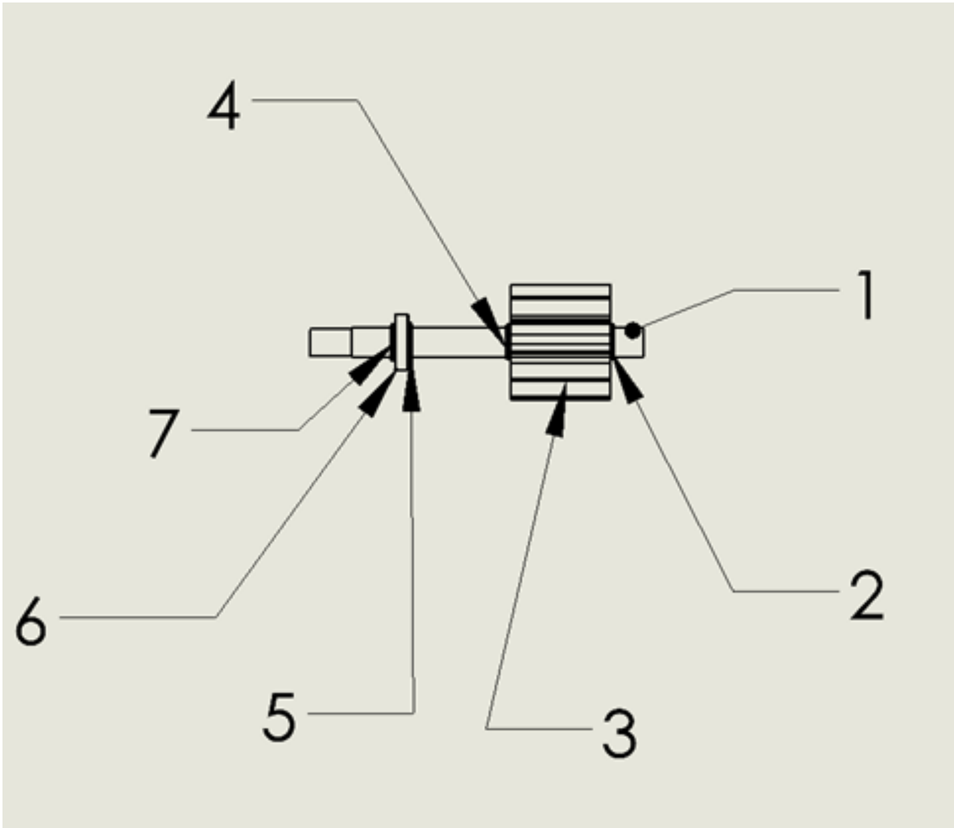


Figure 61. Assembly Process of Shaft 1

1. Retrieve shaft 1.
2. Install locating snap ring for gear.
3. Insert shaft 2 key into slot, then slide sun gear on until it stops at the snap ring.
4. Install retaining snap ring for gear.
5. Install locating snap ring for bearing.
6. Slide bearing into place.
7. Install retaining snap ring for bearing.

Step 9: Assemble Stage 1

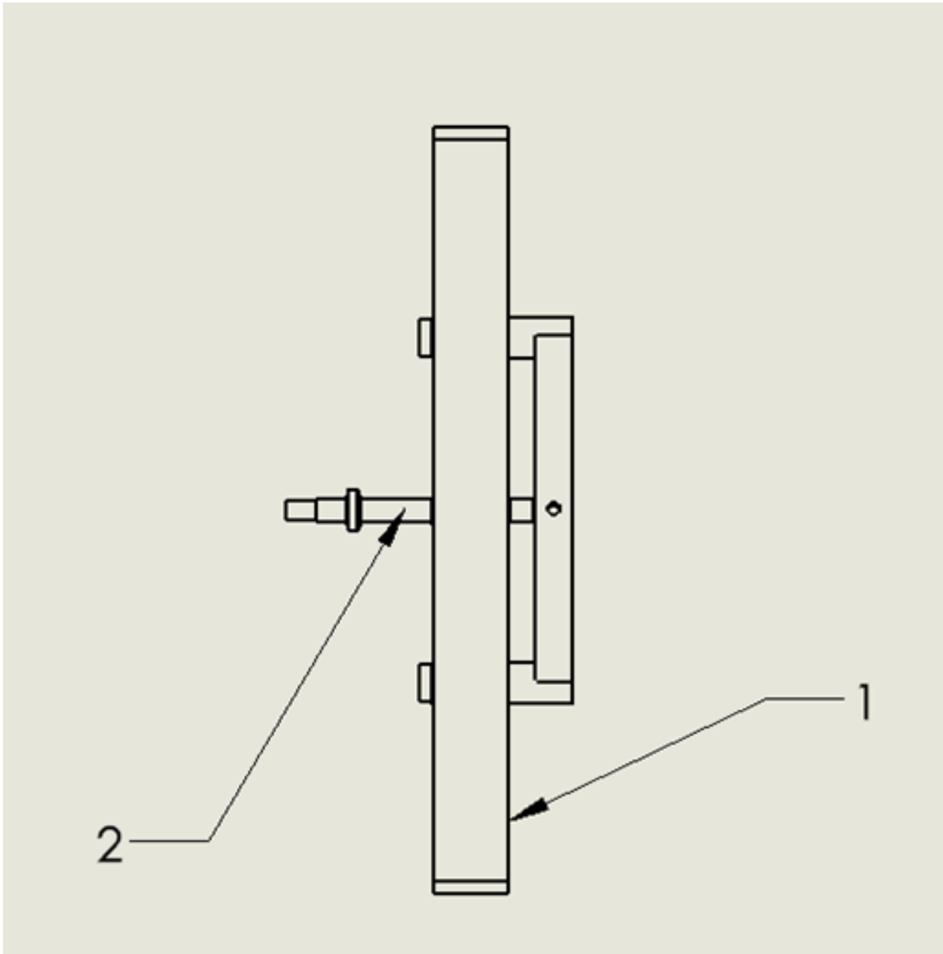


Figure 62. Assembly Process of Stage 1

1. Connect assemblies from step 4 and step 5 by sliding the sun gear between both planet gears, ensure the shafts are aligned.
2. Slide a ring gear over the planet gears.

Step 10: Assemble Gearbox

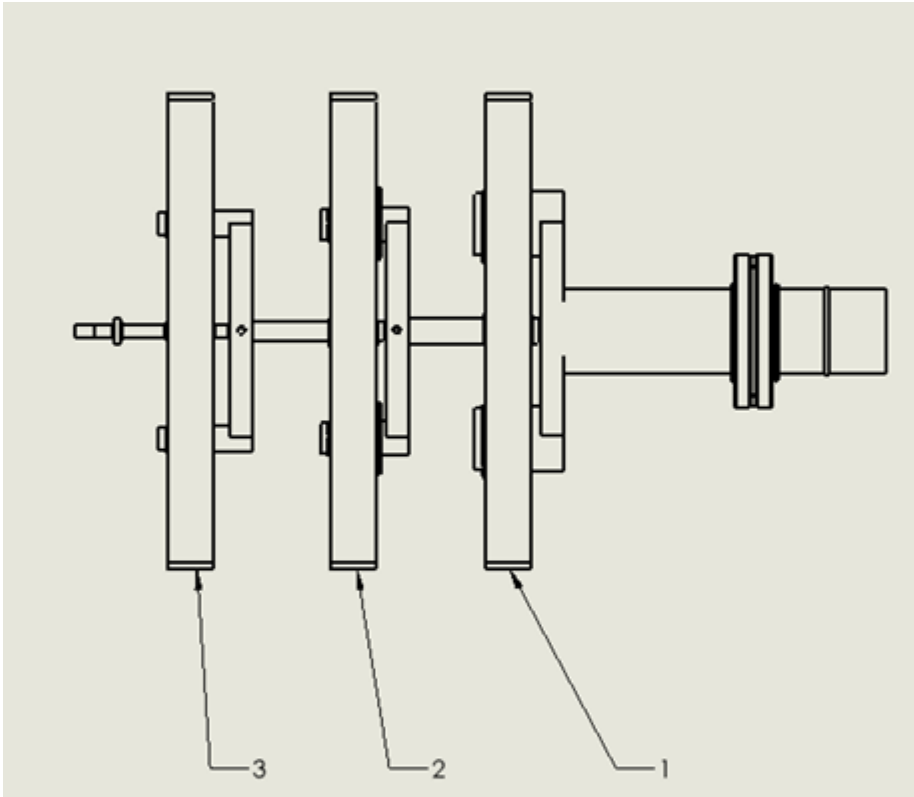


Figure 63. Assembly Process of Gearbox

1. Retrieve assembly prepared in step 3.
2. Connect assembly from step 6 by sliding the sun gear shaft from step 3 into the carrier from step 6 Do not fasten into place yet.
3. Connect assembly from step 9 by sliding the sun gear shaft from step 6 into the carrier from step 6 Do not fasten into place yet.

Step 11: Align Gearbox within Housing

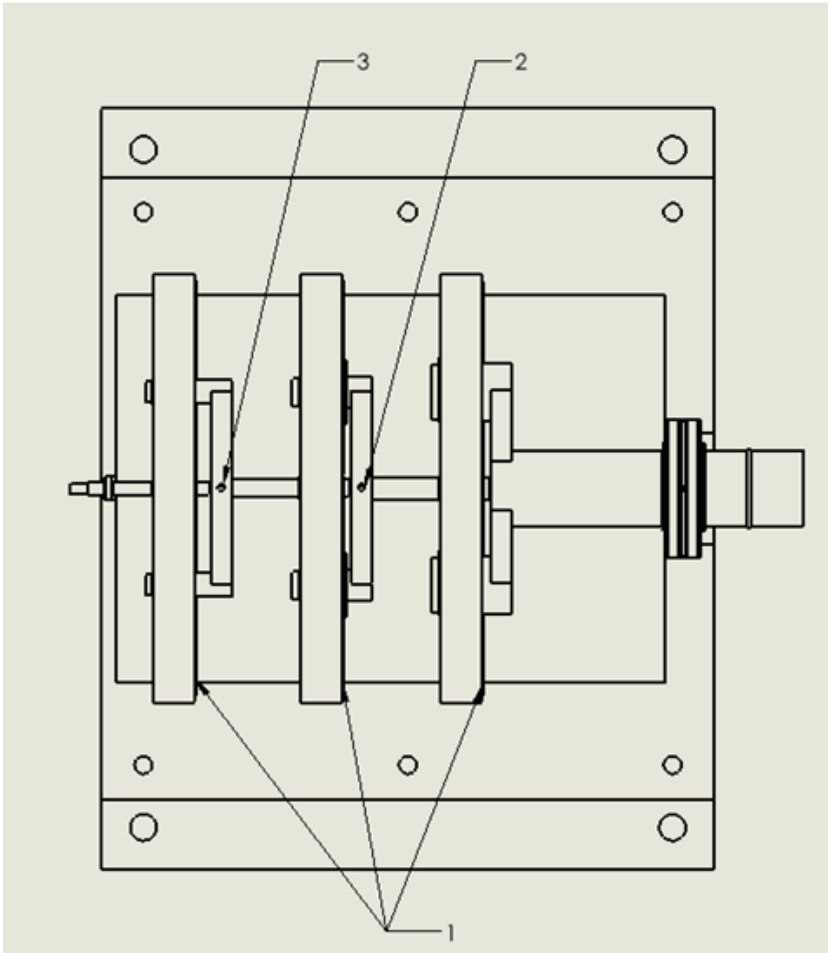


Figure 64. Assembly Process of Gearbox Housing

1. Rotate each ring gear until it’s outside profile aligns with the gearbox bottom housing and it slide into place.
2. Rotate shaft 3 until until the fastener through holes align with the through holes in the carrier, now secure both parts with a fastener.
3. Repeat previous step with shaft 2, then lubricate all moving parts while rotating input shaft.

Step 12: Assembly Housing and Install Gearbox to Ferris Wheel

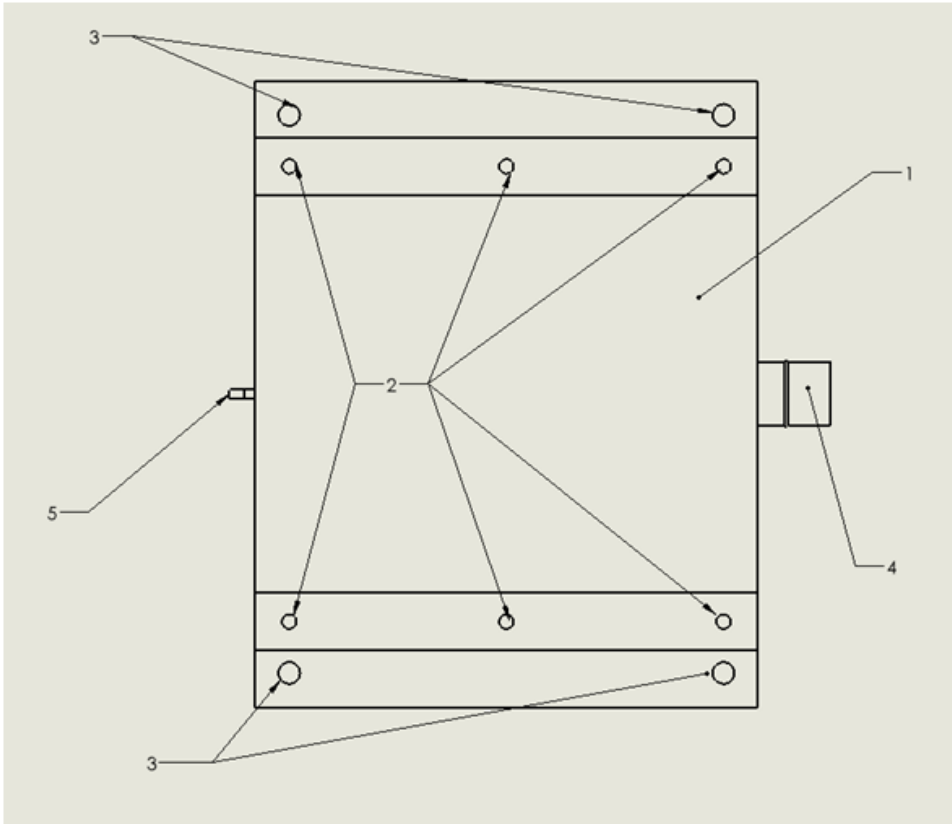


Figure 65. Assembly Process of Gearbox and Ferris Wheel

1. Place top housing onto assembly from previous step, ensure gasket is installed between housing flanges.
2. Fasten housing flange, ensure gasket compression is even and to specified value.
3. Fasten housing to Ferris wheel frame.
4. Fasten coupling between gearbox output shaft and Ferris wheel shaft.
5. Fasten coupling between motor and gearbox input shaft.

17.3 Lubrication and Maintenance Process, Couplings, Fasteners, Seals

The lubrication system used for the planetary gearbox is the oil bath method. The bottom of the gearbox housing would contain reservoirs where the gears would be in contact with lubricating oil. Oil bath lubrication is used for ease of maintenance and lack of intervention. The lubricating oil would be replaced every year. Regular inspection of the oil colour would be performed every three months to ensure safe operation and prevention of accidents due to potentially defected oil. The top of the gearbox housing would be unbolted, and the lubricating oil would be checked.

The maintenance process includes visual inspection, bolts and fasteners checks, structural checks, electrical components, and gearbox check. The maintenance would be performed every three months. The visual inspection includes checks for any signs of visible cracks and deformities with the Ferris wheel and the gearbox. All bolts and fasteners for the gearbox and the Ferris wheel will be performed, checking for stripped and any potential locations for wear. The structural checks would be performed on the spokes of the Ferris wheel, watching for signs of over bending, cracks, or general deformities. The electrical components would be checked, making sure the power from the motor is sufficient to run the Ferris wheel, as well as loose connections, wear, and damage.

Couplings are used in the gearbox to ensure smooth transmission from the motor to the final output to the Ferris wheel. The coupling chosen for the final output shaft of the gearbox to the Ferris wheel is the Series C600M-55 keyless rigid coupling [19]. The coupling selected is based on the size of the size of the shaft from the output and the shaft for the Ferris wheel. Coupling from the electrical motor to the first input shaft of the gearbox would be used and it was selected as item number 2469K2 from McMaster-Carr [20], as well as couplings between each stage of the gearbox.

Fasteners would be used in the gearbox for holding the components in place and closing the housing itself. Bolts would be used to anchor the ring gears to the housing, as well as bolts around the perimeter of the gearbox housing. The method to connect the output shaft of each stage of the planet carrier to the input of the next stage are keys and keyways, as specified in the previous section.

Seals are essential in gearboxes to prevent dust, dirt, and other contaminants from entering important processes in the gearbox. The gearbox housing would have an outer seal to protect all planetary stages as well as keeping the lubricating oil inside the gearbox. Seals would also be employed around the input and final output shaft of the gearbox, as they are also entry points into the gearbox. The bearings chosen have seals within them, maintaining smooth operation between the inner and outer movement of the bearings.

18. Design Reflection

Reflecting on the design, there are numerous aspects that should be improved to ensure safe operation for the lifetime of the Ferris wheel.

Due to the exaggerated safety factors for the surface stresses in all stages of the planetary gears, a cheaper material may be chosen for future consideration for the first and second stages. The third stage has adequate safety factors for the bending stress; therefore, the material would be the same. For ease of manufacturing and bulk purchasing, the same material is used for all gears in the current design.
An important consideration for a future design would be improving the stability of the shafts and gears and keeping them in place, while reducing friction. There could be an improved gearbox housing and have hollow extrusions or plates to keep the shafts and gears in place. Additionally, the gearbox housing would ideally have reservoirs for the lubrication requirement.

To reduce potential downtime and reduce the number of times the gearbox is opened, the lubrication method can be more efficient by incorporating an oil pump to specifically target the gears and shafts undergoing the most friction. Utilizing the method along with the oil reservoirs would minimize the intervention of the operator checking the gearbox and the lubricating oil.

There must be additional calculations and safety verifications for the snap rings and fasteners for the gearbox, as well as further verification for the torques involved in the shafts. Additionally, friction is deemed negligible in the calculations, and although friction would be reduced through the lubrication process, friction is still present and would affect the speed and performance of the gearbox.

Despite the considerations for future design, the high safety factors from the material chosen and the specifications for the gears and shafts guarantee safe operation for the lifetime of the Ferris wheel of 20 years. Additionally, the simple design of the gearbox and placing the gearbox at the top of the vertical shaft to directly drive the Ferris wheel reduces potential failure and increase in maintenance of components.

19. Conclusion

In conclusion, the conceptual prototype of the Ferris wheel has undergone a successful preliminary design phase, with a particular focus on the gearbox and shaft components. The preliminary gearbox layout, featuring four planetary gears, has been meticulously considered to meet all necessary condition checks, and dimensions have been optimized for housing and maximum ring gear size.

The gear design, incorporating hard steel for its advantageous friction properties, has been carefully calculated based on angular speed, torque, and ideal system specifications. The proposed three-stage planetary gear system, despite a slight decrease in efficiency, offers benefits such as higher load capacity and increased durability.

The shaft design and load analysis have provided critical insights into structural integrity, with thorough assessments of stress points and fatigue analyses ensuring a safety factor above 3. Cast iron was selected as the shaft material, and diameters were chosen to meet safety requirements.

The gear design and analysis process included comprehensive evaluations of specifications, failure analysis, and safety factor calculations. The safety factors consistently surpass the design objective of 3.0, indicating that the gear design meets or exceeds requirements.

Shaft design and analysis involved the revision of final shaft assembly designs based on gearbox requirements, with fatigue analysis conducted at critical points. While the safety factor goal of 3 was not met for every component, revisions were made to maintain a safety factor above 2.

The bearing design and analysis involved the selection of roller bearings suitable for high-load applications, with calculations ensuring their compatibility with the desired 20-year operational lifetime. Some bearings could potentially be replaced with cheaper alternatives without compromising performance.

The final gearbox design, with detailed specifications, drawings, and assembly processes, has been outlined systematically. The overall design and analysis processes have met or exceeded safety factor requirements, though further revisions may be needed to optimize for factors such as cost, weight, and size. Moving forward, additional analyses will refine shape characteristics of structural beams, focusing on deflection, buckling, and optimizing strength-to-weight ratios. Further refinement of the gearbox design will concentrate on tooth profiles, shaft diameters, and overall assembly and manufacturing processes to meet torque and service lifetime requirements.

Appendix A Tabulated method for the Planetary Gear

	Stage 1			
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
RBR @ x rpm	x	x	x	x
Fixed C (2), S(1) at y rpm	Y	0	"-(N1/N3) *Y"	"-(N1/N4) *Y"
Total	W1=X+Y	W2=X	W3=X-(N1/N3) Y	W4=X-(N1/N4) Y
	Stage 1			
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
RBR @ x rpm	182.97	182.97	182.97	182.97
Fixed C (2), S(1) at y rpm	1567.03	0	-427.3718182	-182.97
Total	1750	182.97	-244.4018182	0
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
NUMBER OF TEETH	12	N/A	44	100
RADIUS (mm)	15	70	55.00	125
DIAMETER (mm)	30	140.00	110.00	250
DIAMETRAL PITCH (MM)	0.40	N/A	0.40	0.40
	N3	D3	R2	N1
MATH	N3=(N4-N1)/2 N3 = 44	D3=(D4-D1)/2 D3 = 110.00	R2=R1+R3 R2 = 70	N1=((X-W4)/Y) N4 N1 = 12

	Stage 2			
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
RBR @ x rpm	x	x	x	x
Fixed C (2), S(1) at y rpm	Y	0	"-(N1/N3) *Y"	"-(N1/N4) *Y"
Total	W1=X+Y	W2=X	W3=X-(N1/N3) Y	W4=X-(N1/N4) Y
	Stage 2			
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
RBR @ x rpm	19.13	19.13	19.13	19.13

Fixed C (2), S(1) at y rpm	163.84	0	-44.68363636	-19.13
Total	182.97	19.13	-25.55363636	0
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
NUMBER OF TEETH	12	N/A	44	100
RADIUS (mm)	15	70	55.00	125
DIAMETER (mm)	30	140.00	110.00	250
DIAMETRAL PITCH (MM)	0.40	N/A	0.40	0.40
MATH	N3	D3	R2	N1
	N3=(N4-N1)/2	D3=(D4-D1)/2	R2=R1+R3	N1=((X-W4)/Y) N4
	N3 = 44	D3 = 110.00	R2 = 70	N1 = 12
	Stage 3			
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
RBR @ x rpm	x	x	x	x
Fixed C (2), S(1) at y rpm	Y	0	"-(N1/N3) *Y"	"-(N1/N4) *Y"
Total	W1=X+Y	W2=X	W3=X-(N1/N3) Y	W4=X-(N1/N4) Y
	Stage 3			
	Sun (input) (1)	Carrier (output)(2)	Planets (3)	Ring (fixed) (4)
RBR @ x rpm	2	2	2	2
Fixed C (2), S(1) at y rpm	17.13	0	-4.671818182	-2
Total	19.13	2	-2.671818182	0
	Sun (input) (1)	Carrier (output) (2)	Planets (3)	Ring (fixed) (4)
NUMBER OF TEETH	12	N/A	44	100
RADIUS (mm)	15	70	55.00	125
DIAMETER (mm)	30	140.00	110.00	250
DIAMETRAL PITCH (MM)	0.40	N/A	0.40	0.40
MATH	N3	D3	R2	N1
	N3=(N4-N1)/2	D3=(D4-D1)/2	R2=R1+R3	N1=((X-W4)/Y) N4
	N3 = 44	D3 = 110.00	R2 = 70	N1 = 12

Appendix B Condition Check for Number of Planets

CONDITION CHECK		
n<180/(SIN-1((N3+2*ALPHA)/(N1+N3))	3.26	MAX NUMBER
(N1+N4)/n	56	

Appendix C Contact Ratio Calculation

CONTACT RATIO		
Pressure angle	20	DEGREESS
Pressure angle	0.3	RADIANS
PB=PI*COS(PHI)/PD	7.4	
A=1/P	2.5	
B=1.25/P	3	
EXTERNAL - EXTERNAL		
C=R1+R3	C= 70	
Z=SQRT((RP+A) ^2-(RP*COS(PHI)) ^2) +SQRT((RG+A) ^2-(RG*COS(PHI)) ^2) - C*SIN(PHI)		
Z =	50.25	
CONTACT RATION = Z/PB	6.81	
EXTERNAL - INTERNAL		
C=R4-R3	C=70	
Z=SQRT((RP+A) ^2-(RP*COS(PHI)) ^2)-SQRT((RG+A) ^2-(RG*COS(PHI)) ^2) + C*SIN(PHI)		
Z =	93.93	
CONTACT RATION = Z/PB	12.73	

Appendix D Efficiency Calculations

EFFIENCY	
coefficient of friction for Hard Steel Greased	0.029

n=1-((MU*MP*PB)/(4COS(BETA)*COS(PHI)) (1/RP+1/RG)	
EFFIENCY EXTERNAL - EXTERNAL	0.97
EFFIENCY EXTERNAL - INTERNAL	0.98

Appendix E Torque of the Gears Calculation

Torque of gears	Gear 1 (Sun)	Gear 2 (Carrier)	Gear 3 (Planets)	Gear 4 (Ring)
All Stages Gear radius [mm]	15	70	55.00	125.00
Stage 1 Torques [Nm.]	13.38	126.64	49.06	111.50
Stage 2 Torques [Nm.]	126.64	1199.04	464.35	1055.33
Stage 3 Torques [Nm.]	1199.04	11352.68	4396.48	9992.00

Appendix F Bearing for Stage 1 Gear 1 (Sun)

W 618/9Stainless steel deep groove ball bearing

Stainless steel deep groove ball bearing

Stainless steel single row deep groove ball bearings provide greater chemical and corrosion resistance. As with deep groove ball bearings generally, they are particularly versatile, have low friction and are optimized for low noise and low vibration, which enables high rotational speeds. They accommodate radial and axial loads in both directions, are easy to mount, and require less maintenance than many other bearing types.

- Greater chemical and corrosion resistance
- Simple, versatile and robust design
- Low friction and high-speed capability
- Accommodate radial and axial loads in both directions
- Require little maintenance

Overview

Dimensions

Bore diameter	9 mm
Outside diameter	17 mm
Width	4 mm

Performance

Basic dynamic load rating	0.761 kN
Basic static load rating	0.335 kN
Reference speed	85 000 r/min
Limiting speed	53 000 r/min

Properties

Filling slots	Without
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Sheet metal
Matched arrangement	No
Radial internal clearance	CN
Tolerance class	Normal
Material, bearing	Stainless steel
Coating	Without
Sealing	Without
Lubricant	None

Appendix G Bearing for Stage 1 Gear 3 (Planets)

NJ 202 ECPSingle row cylindrical roller bearing, NJ design

Single row cylindrical roller bearing, NJ design

Single row cylindrical roller bearings are designed to accommodate high radial loads in combination with high speeds. Having two integral flanges on the outer ring and one on the inner ring, NJ design bearings can accommodate axial displacement in one direction. An important feature is the separable design, which facilitates mounting and enables the bearing components to be interchanged.

- High radial load carrying capacity
- Low friction
- Long service life
- Locate the shaft axially in one direction
- Separable design

Overview

Dimensions

Bore diameter	15 mm
Outside diameter	35 mm
Width	11 mm

Performance

Basic dynamic load rating	12.5 kN
Basic static load rating	10.2 kN
Reference speed	22 000 r/min
Limiting speed	26 000 r/min

Properties

Bearing part	Complete bearing
Axial displacement capability	In one direction
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Non-metallic
Number of flanges, outer ring	2
Number of flanges, inner ring	1
Loose flange	None
Radial internal clearance	CN
Tolerance class	Normal
Coating	Without
Sealing	Without

Appendix H Bearing for Stage 2 Gear 3 (Planets)

N 204 ECPSingle row cylindrical roller bearing, N design

Single row cylindrical roller bearing, N design

Single row cylindrical roller bearings are designed to accommodate high radial loads in combination with high speeds. Having two integral flanges on the inner ring and no flanges on the outer ring, N design bearings can accommodate axial displacement in both directions. An important feature is the separable design, which facilitates mounting and enables the bearing components to be interchanged.

- High radial load carrying capacity
- Low friction
- Long service life
- Accommodate axial displacement in both directions
- Separable design

Overview

Dimensions

Bore diameter	20 mm
Outside diameter	47 mm
Width	14 mm

Performance

Basic dynamic load rating	28.5 kN
Basic static load rating	22 kN
Reference speed	17 000 r/min
Limiting speed	19 000 r/min
SKF performance class	SKF Explorer

Properties

Bearing part	Complete bearing
Axial displacement capability	In both directions
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Non-metallic
Number of flanges, outer ring	0
Number of flanges, inner ring	2
Loose flange	None
Radial internal clearance	CN
Tolerance class	Normal
Coating	Without
Sealing	Without

Appendix I Bearing for Stage 3 Gear 3 (Planets)

NU 1008 MLSingle row cylindrical roller bearing, NU design

Single row cylindrical roller bearing, NU design

Single row cylindrical roller bearings are designed to accommodate high radial loads in combination with high speeds. Having two integral flanges on the outer ring and no flanges on the inner ring, NU design bearings can accommodate axial displacement in both directions. An important feature is the separable design, which facilitates mounting and enables the bearing components to be interchanged.

- High radial load carrying capacity
- Low friction
- Long service life
- Accommodate axial displacement in both directions
- Separable design

Overview

Dimensions

Bore diameter	40 mm
Outside diameter	68 mm
Width	15 mm

Performance

Basic dynamic load rating	28.5 kN
Basic static load rating	26 kN
Reference speed	12 000 r/min
Limiting speed	18 000 r/min
SKF performance class	SKF Explorer

Properties

Bearing part	Complete bearing
Axial displacement capability	In both directions
Number of rows	1
Locating feature, bearing outer ring	None
Bore type	Cylindrical
Cage	Machined metal
Number of flanges, outer ring	2
Number of flanges, inner ring	0
Loose flange	None
Radial internal clearance	CN
Tolerance class	Normal
Coating	Without
Sealing	Without

Appendix J Bearing for Stage 3 Gear 2 (Carrier)

22211 ESpherical roller bearing with relubrication features

Spherical roller bearing with relubrication features

Spherical roller bearings can accommodate heavy loads in both directions. They are self-aligning and accommodate misalignment and shaft deflections, with virtually no increase in friction or temperature. The design includes features to facilitate relubrication. The bearings can be used in a modular system, including housings, sleeves and nuts.

- Accommodate misalignment
- High load carrying capacity
- Relubrication features
- Low friction and long service life
- Increased wear resistance

Overview

Dimensions

Bore diameter	55 mm
Outside diameter	100 mm
Width	25 mm

Performance

Basic dynamic load rating	129 kN
Basic static load rating	127 kN
Reference speed	6 300 r/min
Limiting speed	8 500 r/min
SKF performance class	SKF Explorer

Properties

Number of rows	2
Locating feature, bearing outer ring	Without
Bore type	Cylindrical
Cage	Sheet metal
Radial internal clearance	CN
Tolerance class	Normal
Tolerance class for dimensions	Normal
Tolerance class for run-out	P5
Sealing	Without
Lubricant	None
Relubrication feature	With

Appendix K Bill of Materials

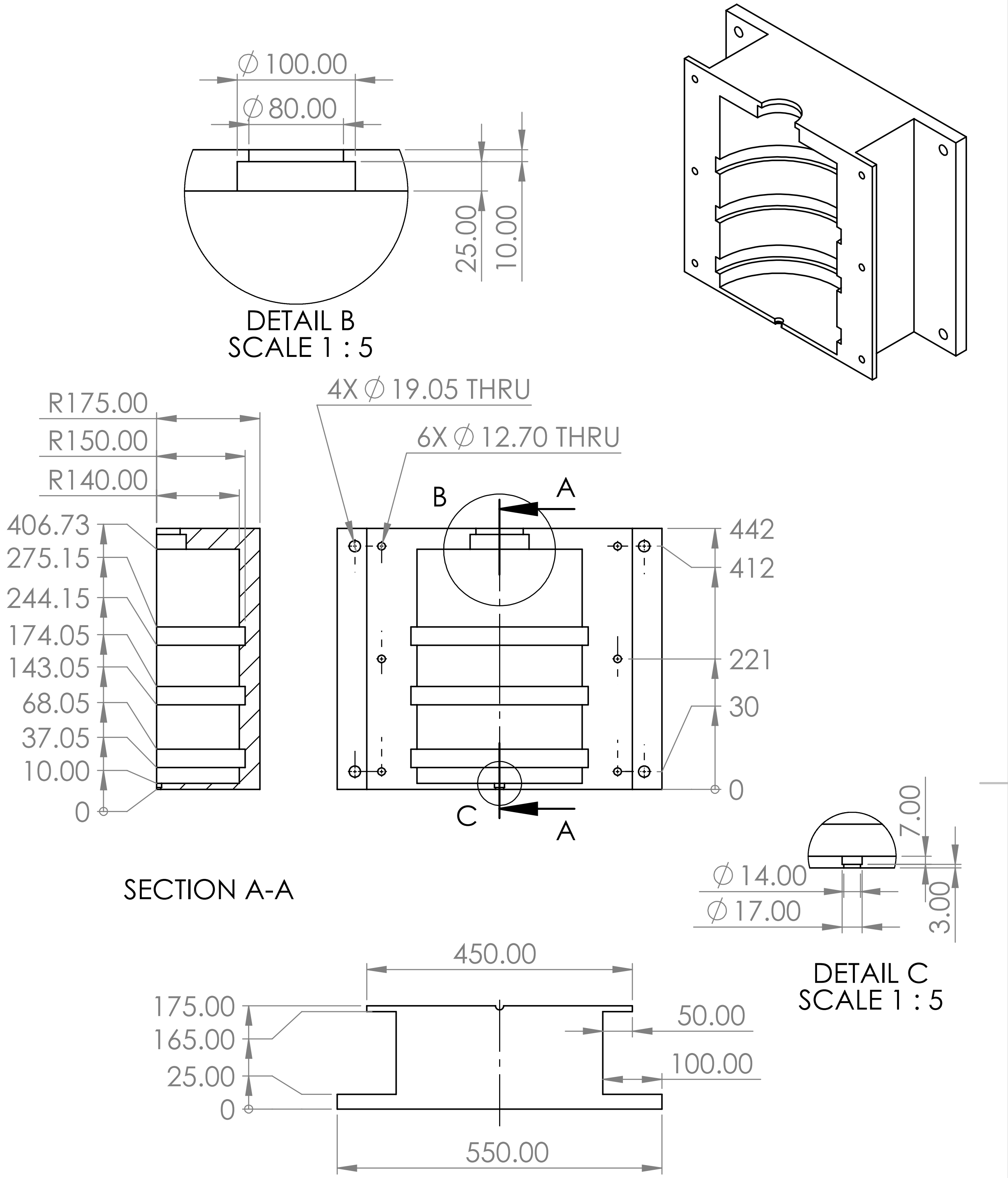
SUB ASSEMBLY A	NAME	QTY.
	Gear assembly Stage 1	**1**
ITEM NO.	PART NUMBER	QTY.
1	Metric - Spur gear 2.5M 12T 20PA 30FW ---S12N75H50L9S1	1
2	Metric - Spur gear 2.5M 44T 20PA 30FW ---S44N75H50L70N	2
3	Metric - Internal spur gear 2.5M 100T 20PA 30FW ---S100S300OD 1AF	1
SUB ASSEMBLY B	NAME	QTY.
	Stage 1 shaft	**1**
ITEM NO.	PART NUMBER	QTY.
1	Shaft_1	1
2	Shaft_1_key	1
3	90967A155 External Retaining Ring	4
SUB ASSEMBLY - A AND B	NAME	QTY.
	Stage 1 gear and shaft	1
ITEM NO.	PART NUMBER	QTY.
1	gear assembly stage 1	1
2	skf_bearing_nj_202_ec p_2	2
3	stage 1 shaft	1
4	Shaft_2_Carrier_V2	1
5	98541A410 External Retaining Ring	2
SUB ASSEMBLY C	NAME	QTY.
	Gear assembly Stage 2	**1**

ITEM NO.	PART NUMBER	QTY.
1	Metric - Spur gear 2.5M 12T 20PA 30FW ---S12N75H50L14S1	1
2	Metric - Spur gear 2.5M 44T 20PA 30FW ---S44N75H50L70N	1
3	Metric - Spur gear 2.5M 44T 20PA 30FW ---S44N75H50L48N	1
4	Metric - Internal spur gear 2.5M 100T 20PA 30FW ---S100S300OD 1AF	1
SUB ASSEMBLY D	NAME	QTY.
	stage 2 shaft	**1**
ITEM NO.	PART NUMBER	QTY.
1	Shaft_2_key	1
2	Shaft_2	1
3	90967A170 External Retaining Ring	2
SUB ASSEMBLY - C AND D	NAME	QTY.
	stage 2 gear and shaft	**1**
ITEM NO.	PART NUMBER	QTY.
1	skf_bearing_n_204_ec p_2	2
2	gear assembly stage 2	1
3	stage 2 shaft	1
4	98541A123 External Retaining Ring	2
5	Shaft_3_Carrier_V2	1
SUB ASSEMBLY E	NAME	QTY.
	Gear assembly stage 3	**1**
ITEM NO.	PART NUMBER	QTY.

1	Metric - Spur gear 2.5M 12T 20PA 30FW -- -S12N75H50L16S1	1
2	Metric - Spur gear 2.5M 44T 20PA 30FW -- -S44N75H50L70N	2
3	Metric - Internal spur gear 2.5M 100T 20PA 30FW ---S100S300OD 1AF	1
SUB ASSEMBLY F	NAME	QTY.
	Stage 3 shaft	**1**
ITEM NO.	PART NUMBER	QTY.
1	90967A205 External Retaining Ring	2
2	Shaft_3	1
3	Shaft_3_Key	1
SUB ASSEMBLY - E AND F	NAME	QTY.
	stage 3 gear and shaft	**1**
ITEM NO.	PART NUMBER	QTY.
1	skf_bearing_nu_100 8_ml_2	2
2	gear assembly stage 3	1
3	stage 3 shaft	1
4	stage 4 shaft	1
5	Shaft_4_Carrier_V2	1
6	98541A156 External Retaining Ring	2
MAIN ASSEMBLY	NAME	QTY.
	full assembly gear	**1**
ITEM NO.	PART NUMBER	QTY.
1	BOX bottom	1
2	3 planet gear assembly	1

3	skf_bearing_22211_e _2	1
4	skf_bearing_w_618_9 _2	1
5	BOX top	1

Appendix L Drawing Package



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SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

DWG NO.

BOX bottom

A4

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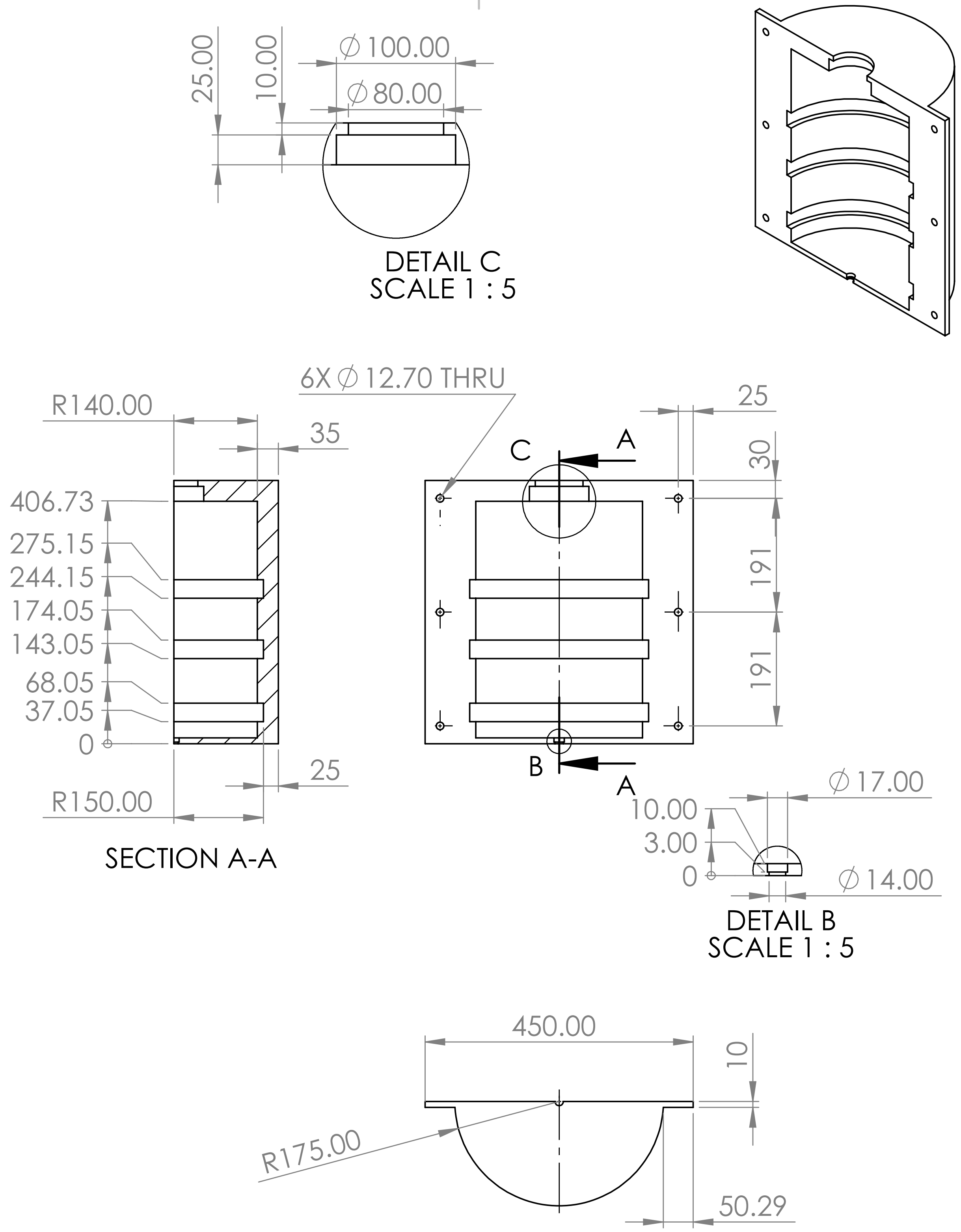
SHEET 1 OF 1



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CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL:		DWG NO. BOX top	A4
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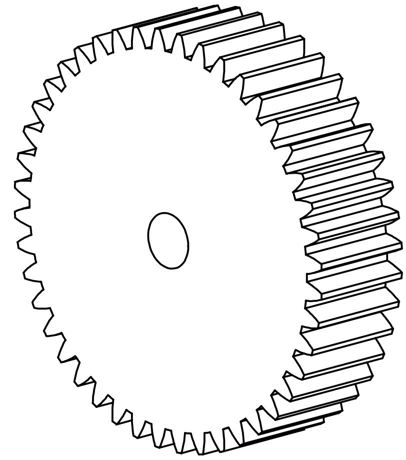

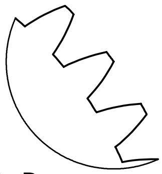

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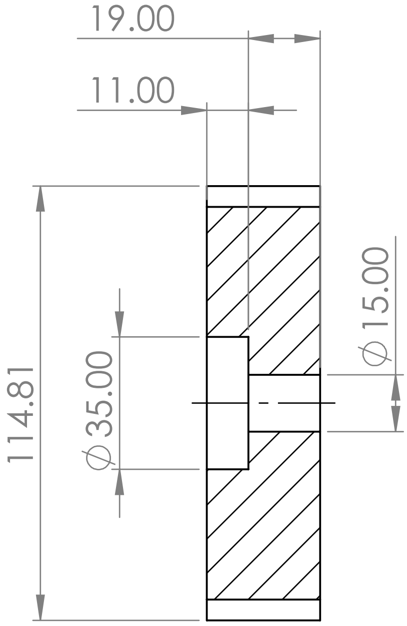
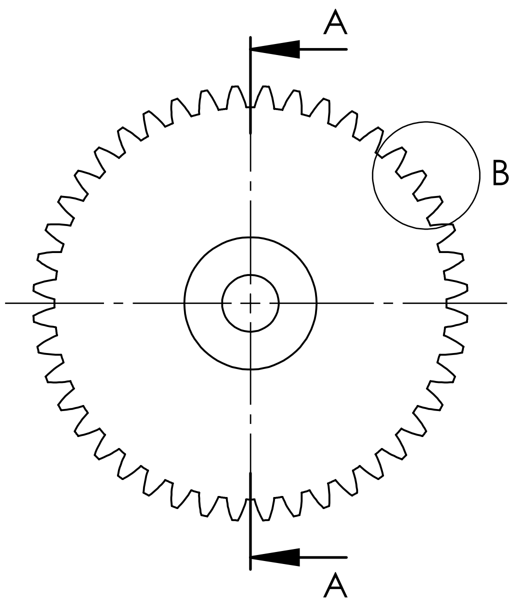
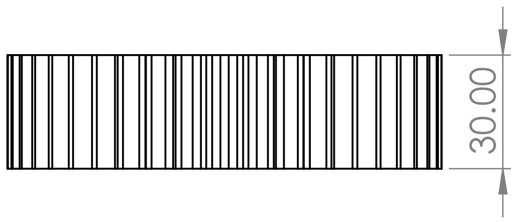


SECTION A-A





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SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

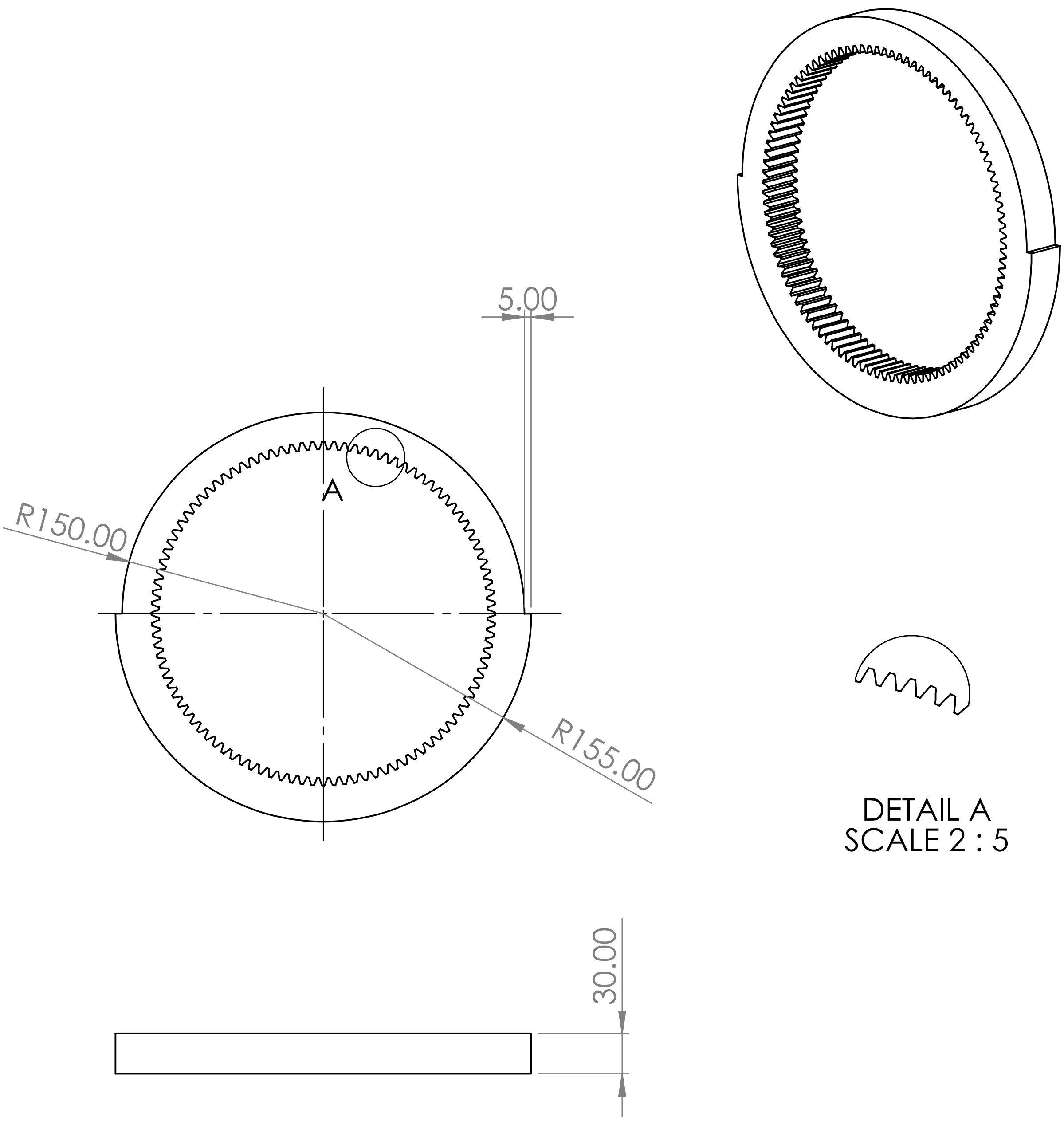
Planet-gear-1

A4

WEIGHT:

SCALE:1:2

SHEET 1 OF 1



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DRAWN							
CHK'D							
APPV'D							
MFG							
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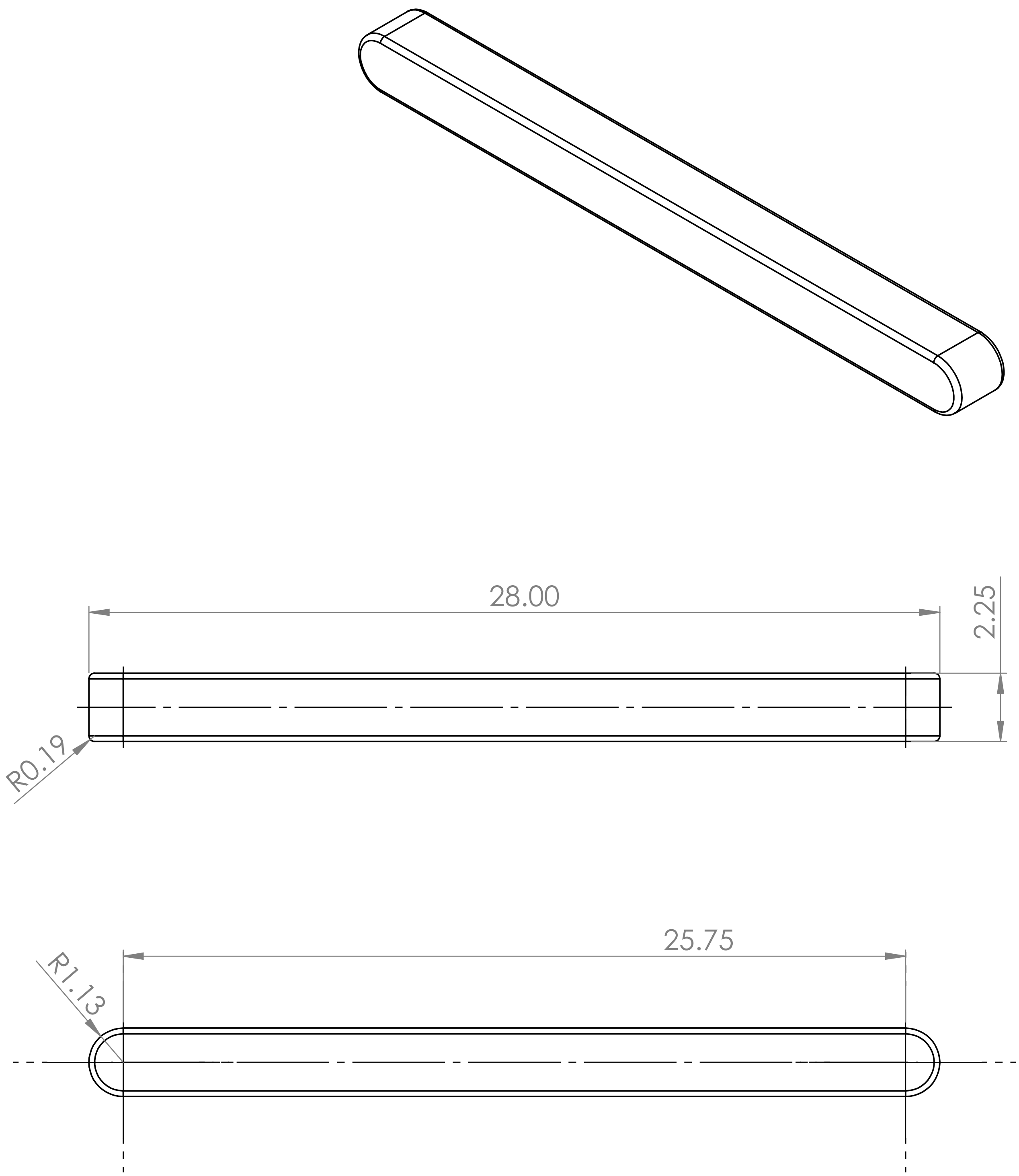

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CHK'D							
APPV'D							
MFG							
Q.A				MATERIAL:		DWG NO. Shaft_1_key	A4
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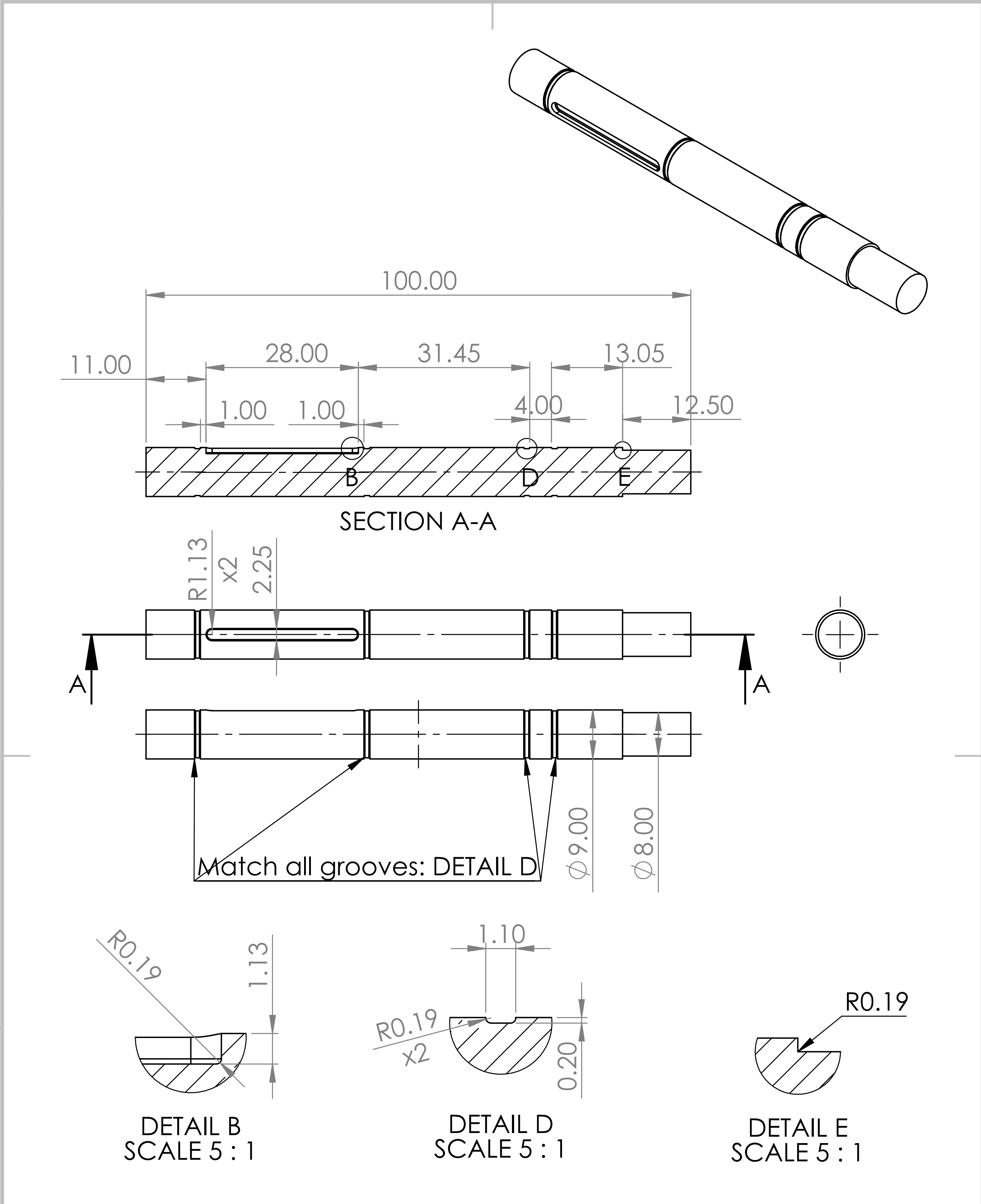

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	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

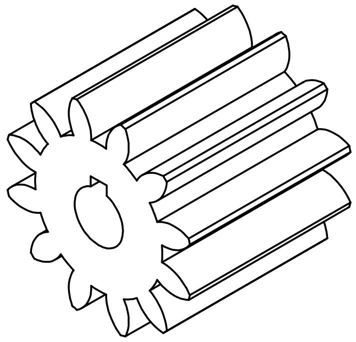
Shaft-1

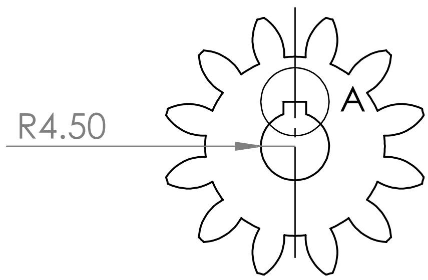
A4

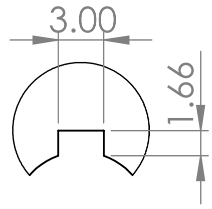
WEIGHT:

SCALE:1:1

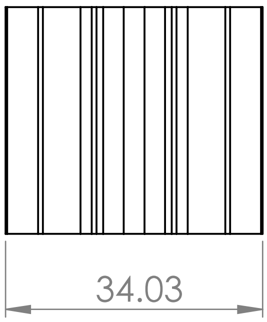
SHEET 1 OF 1







DETAIL A
SCALE 2 : 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

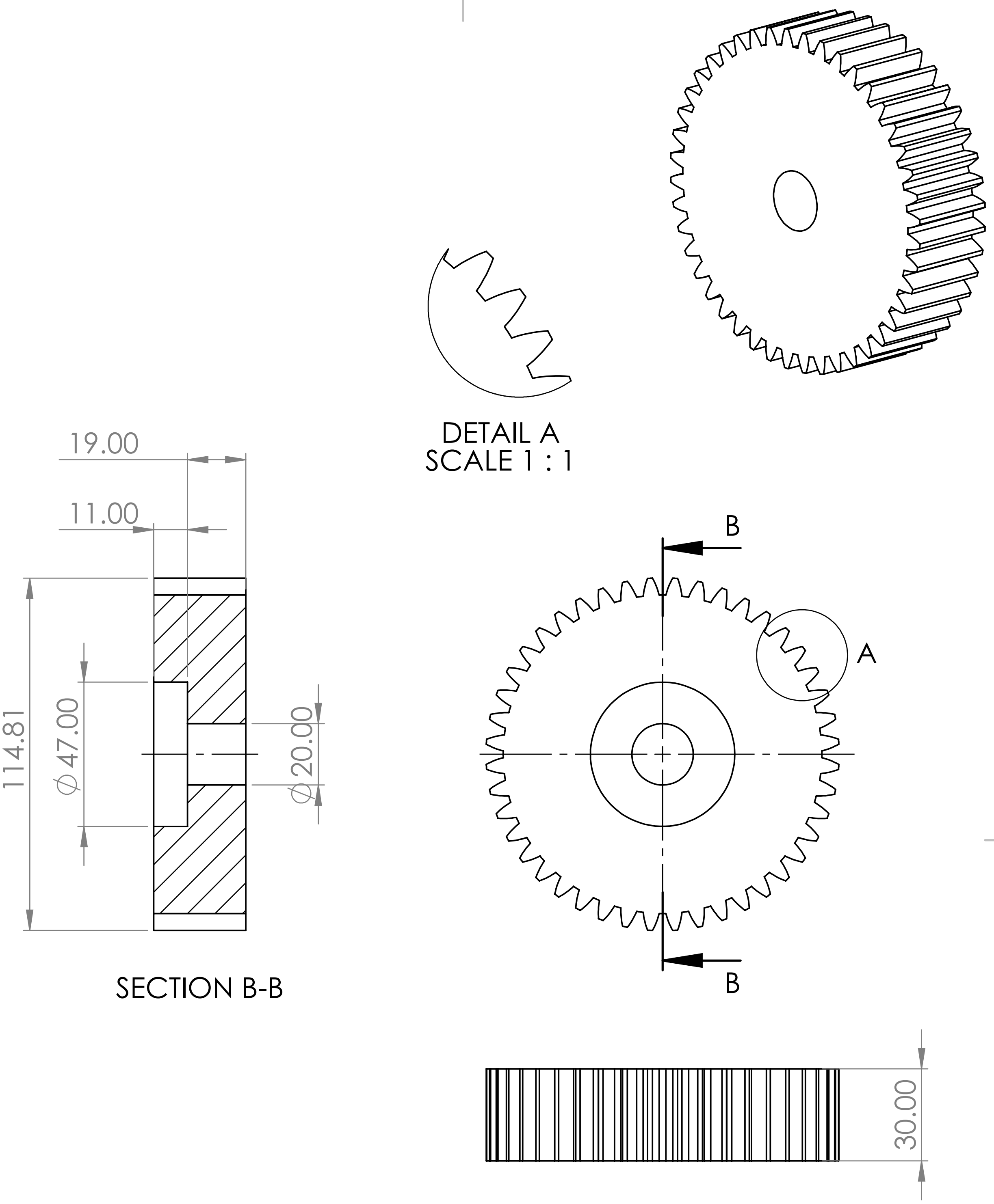
Sun-gear-1

A4

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING
			REVISION

	NAME	SIGNATURE	DATE		
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL:	
				WEIGHT:	

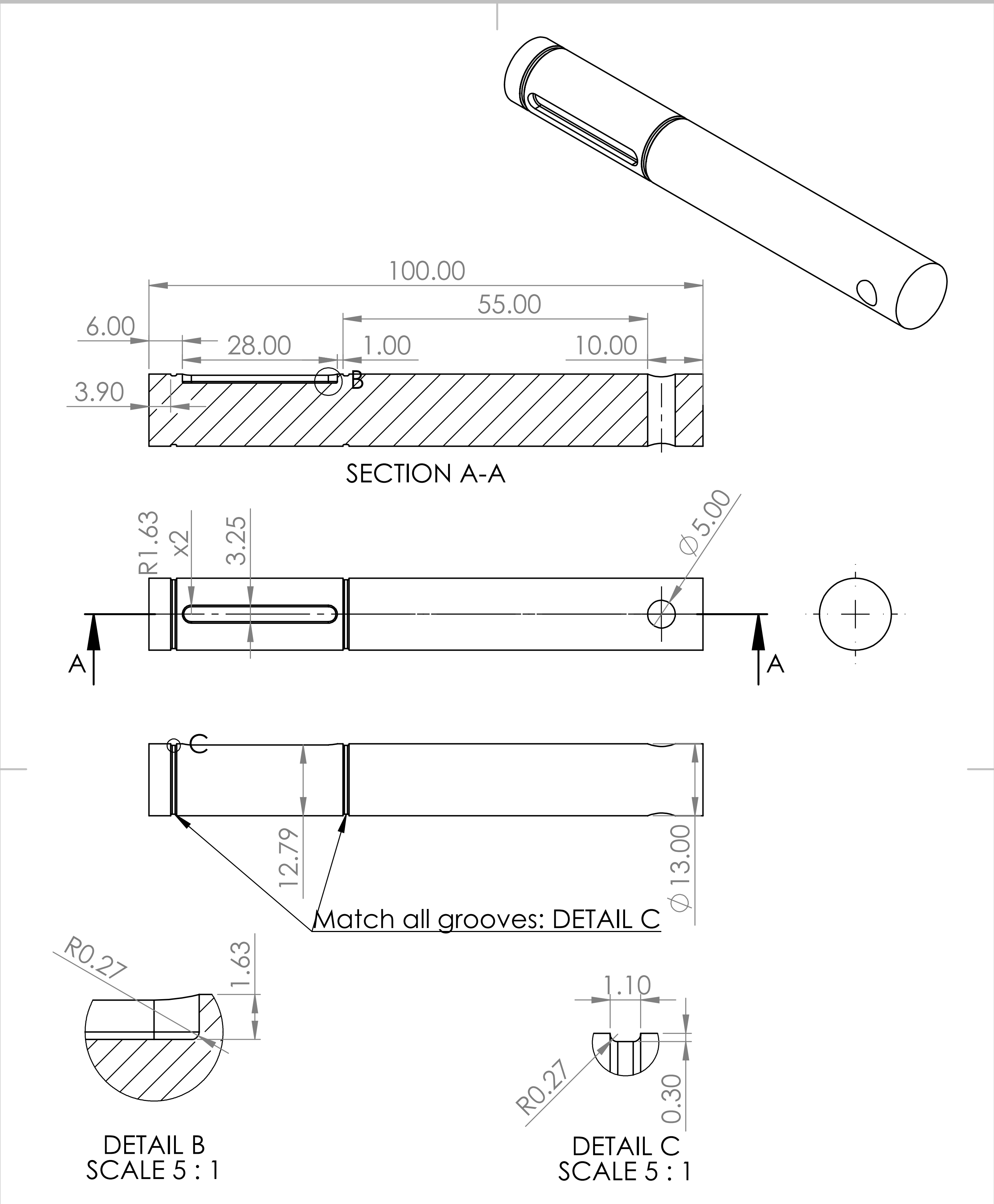
TITLE:

DWG NO.
Planet-gear-2

A4

SCALE:1:2

SHEET 1 OF 1



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SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

WEIGHT:

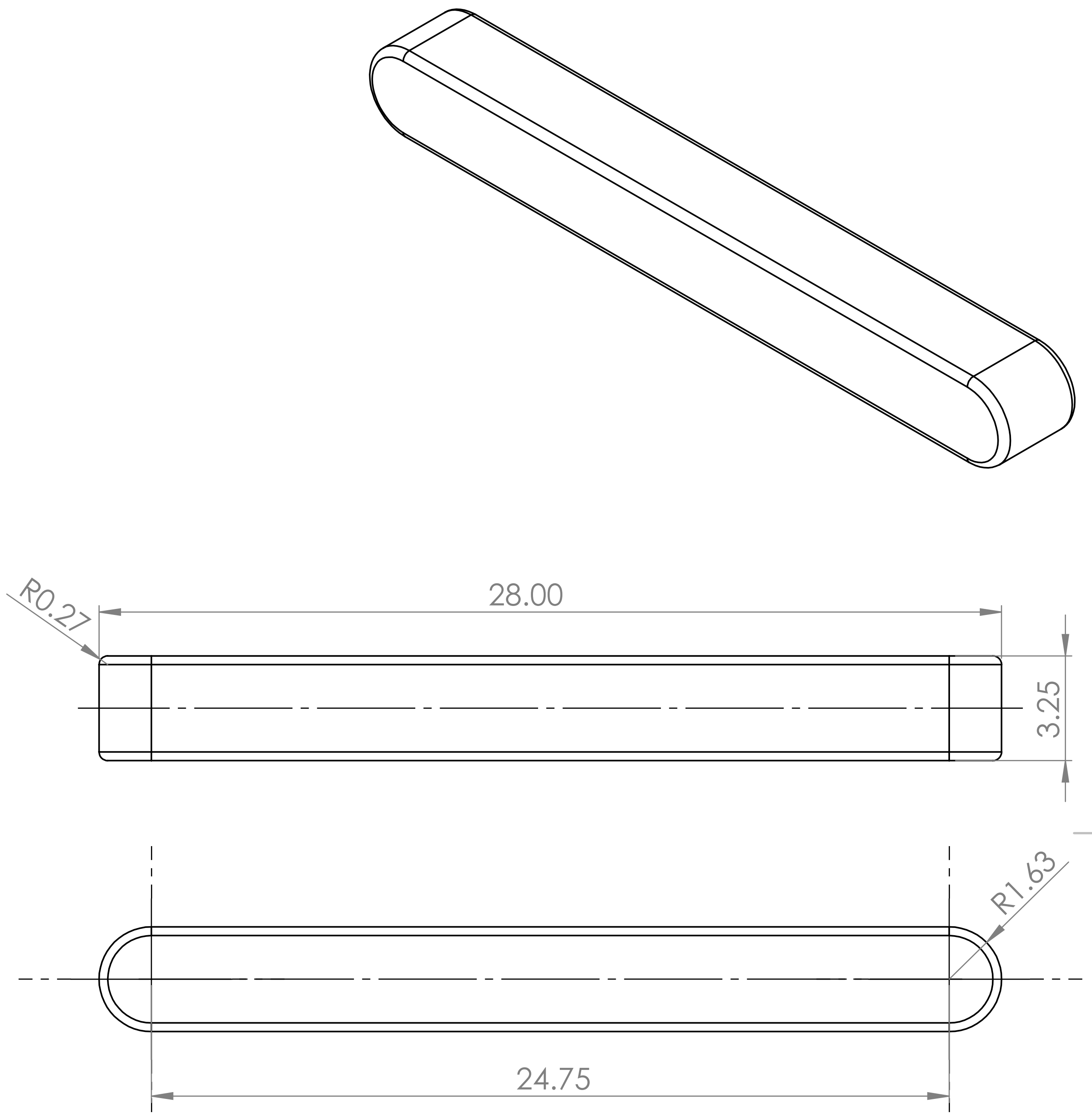
DWG NO.

Shaft-2

A4

SCALE:1:1

SHEET 1 OF 1



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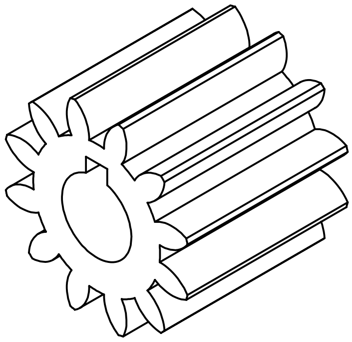
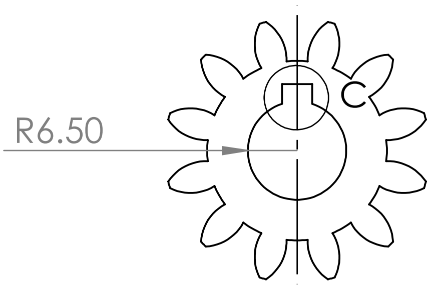
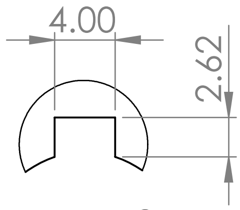
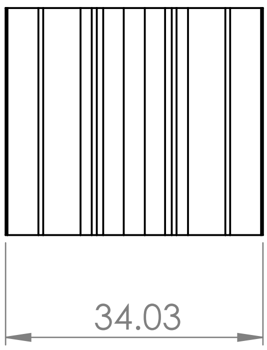
	NAME	SIGNATURE	DATE		TITLE:
DRAWN					
CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL:	DWG NO. Shaft-2-key
					A4
				WEIGHT:	SCALE:5:1
					SHEET 1 OF 1







DETAIL C
SCALE 2 : 1



UNLESS OTHERWISE SPECIFIED:
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SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

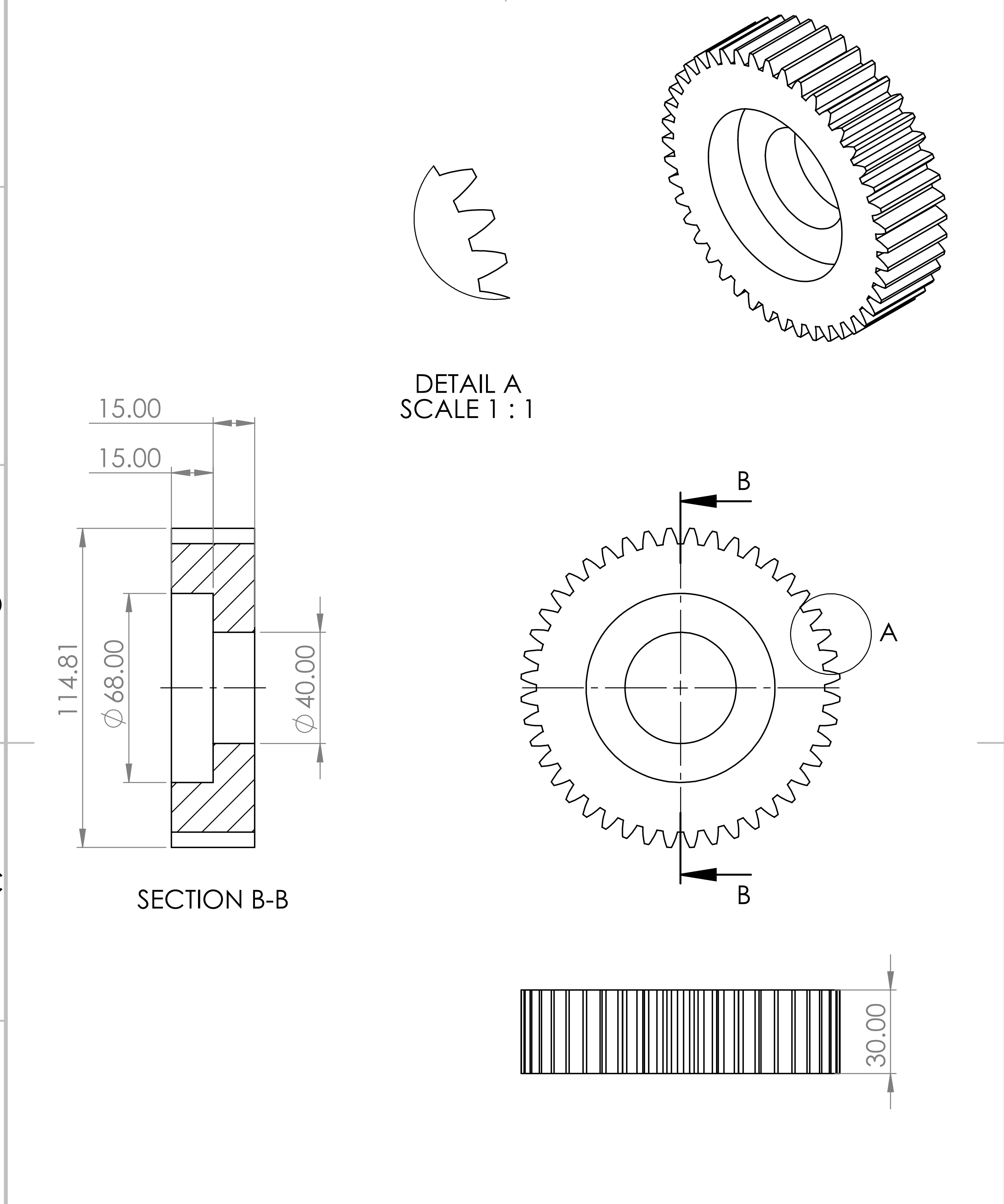
Sun-gear-2

A4

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

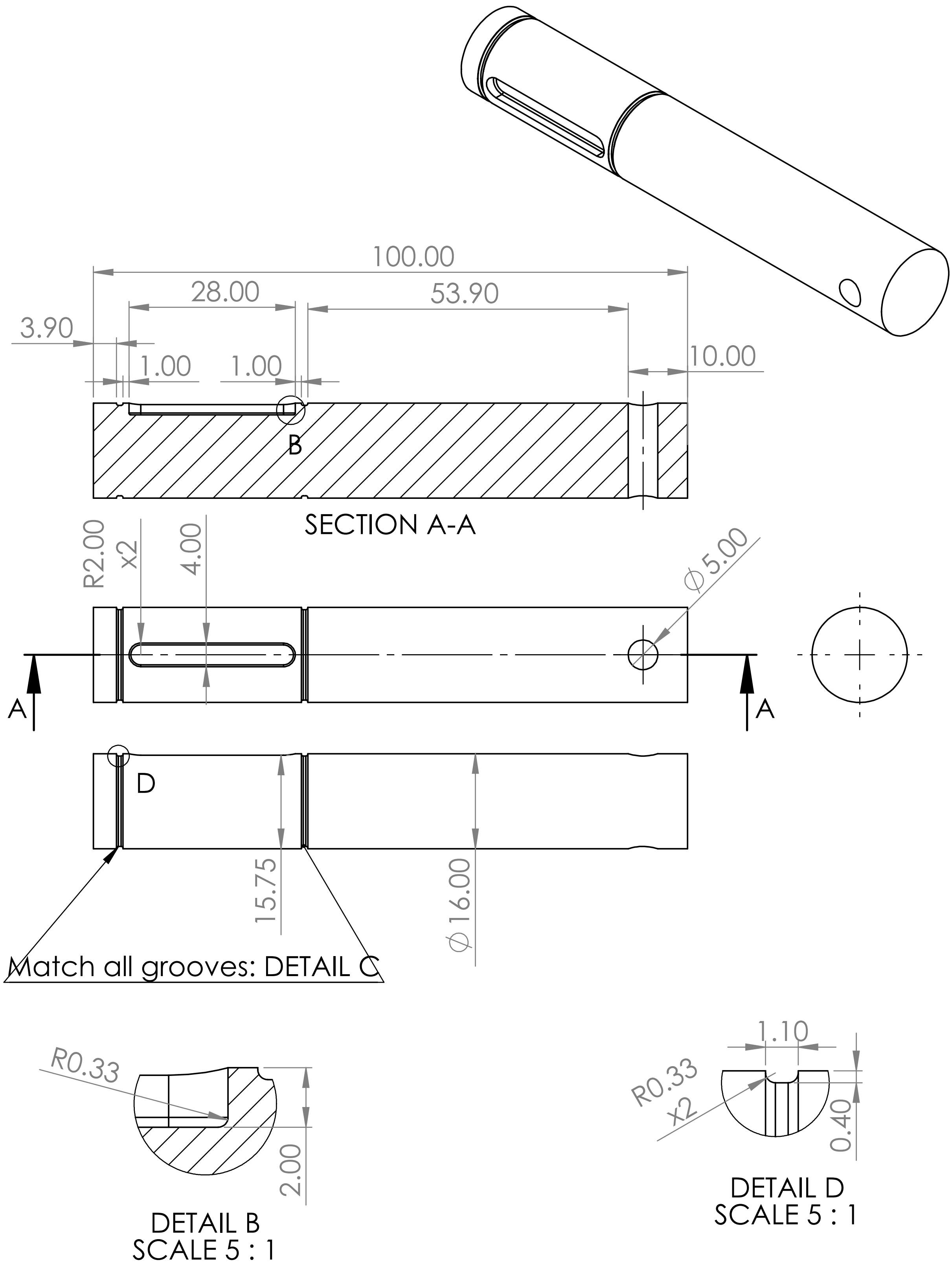
Planet-gear-3

A4

WEIGHT:

SCALE:1:2

SHEET 1 OF 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

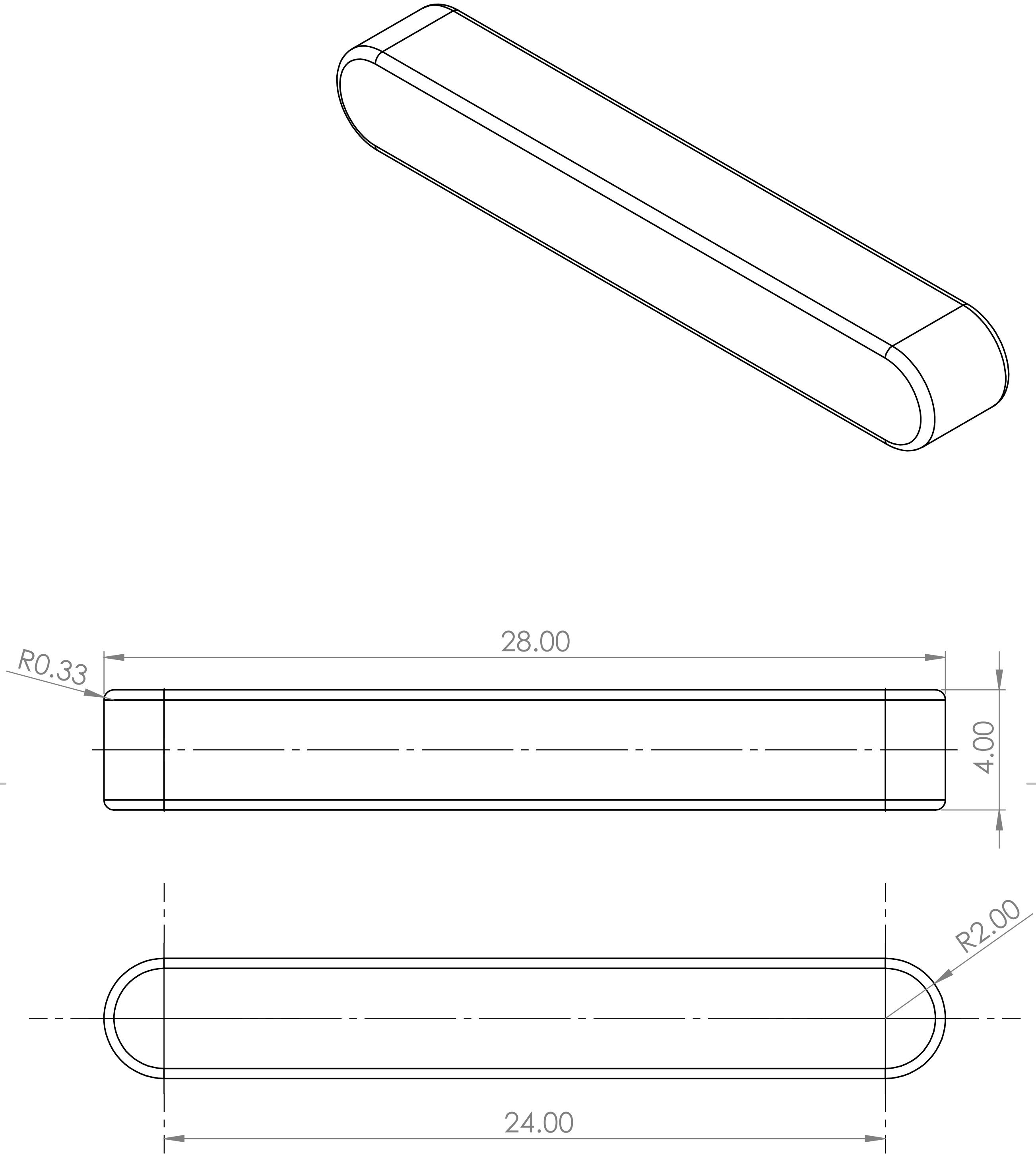
Shaft-3

A4

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



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DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

WEIGHT:

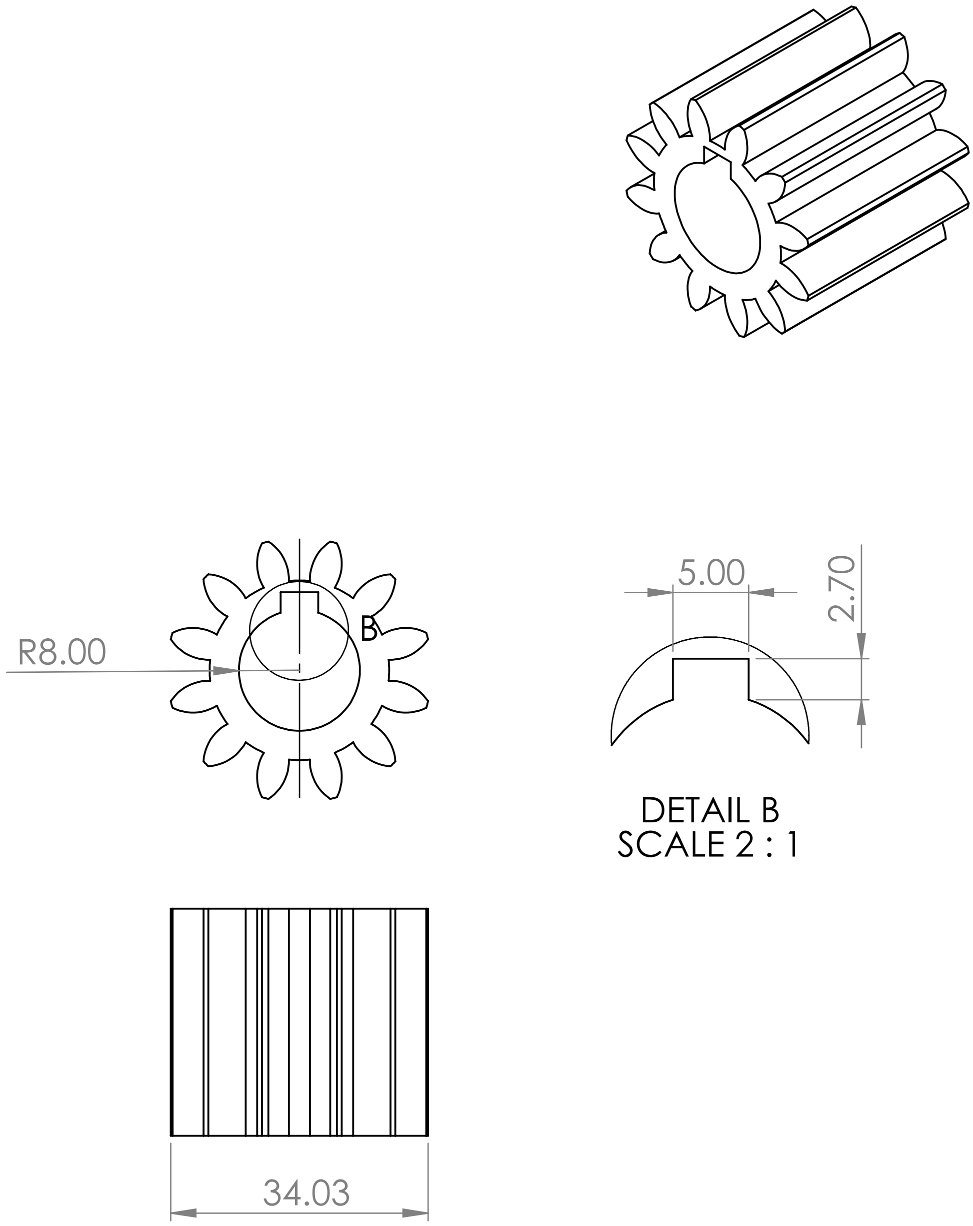
DWG NO.

Shaft-3-key

A4

SCALE:5:1

SHEET 1 OF 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

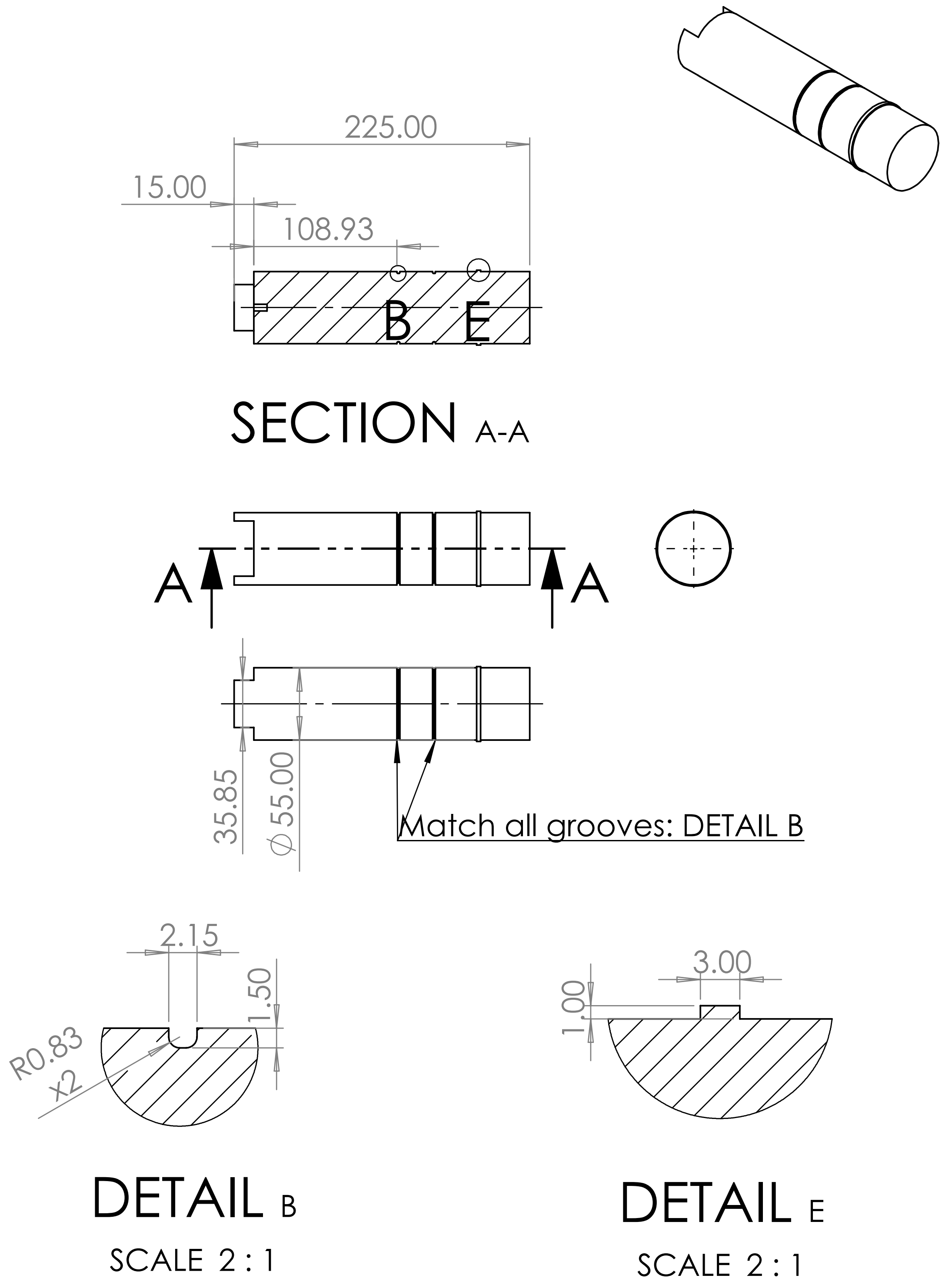
Sun-gear-3

A4

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

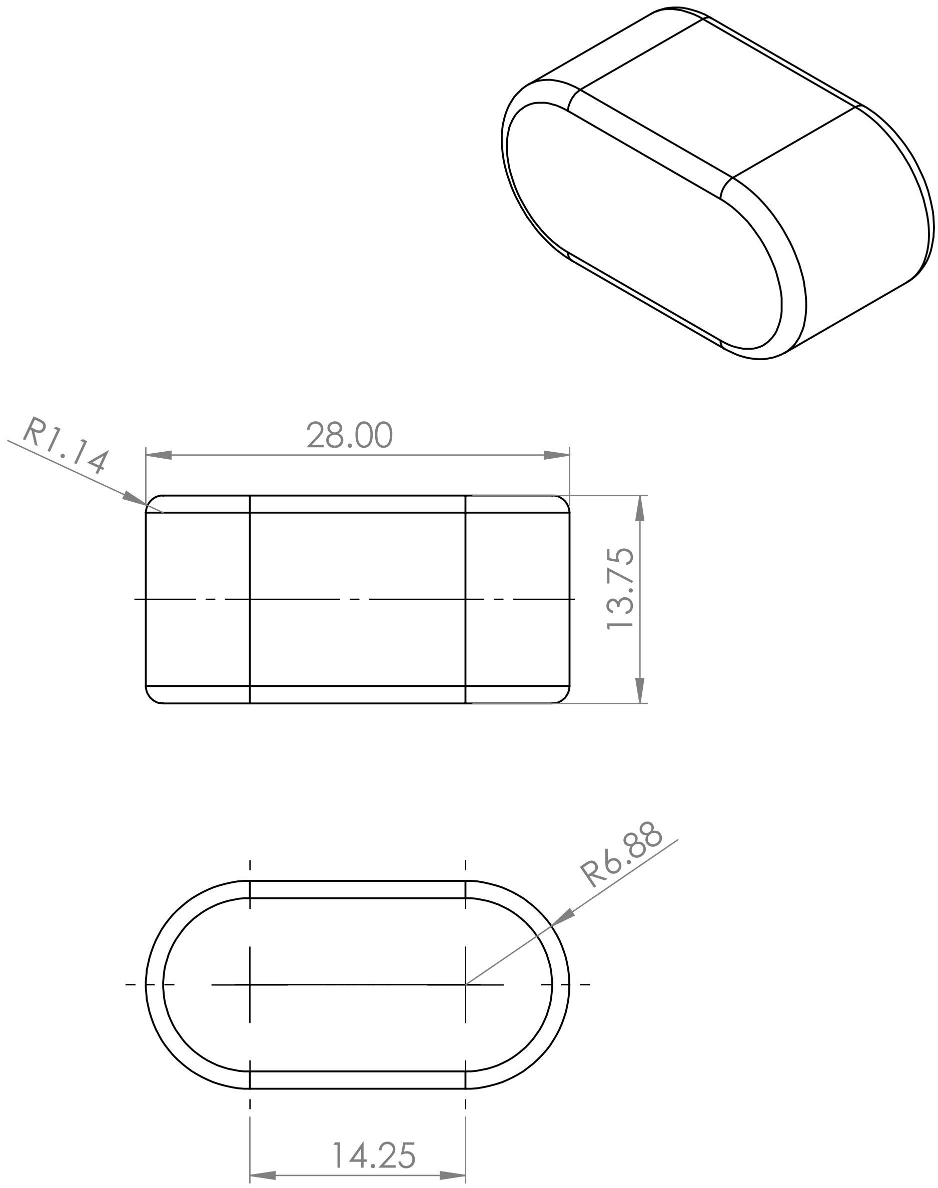
Shaft-4

A4

WEIGHT:

SCALE:1:5

SHEET 1 OF 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

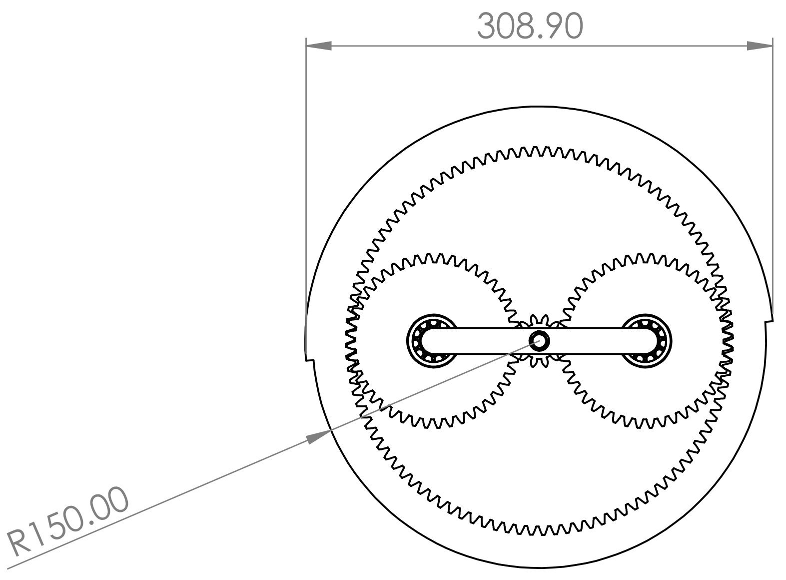
Shaft-4-key

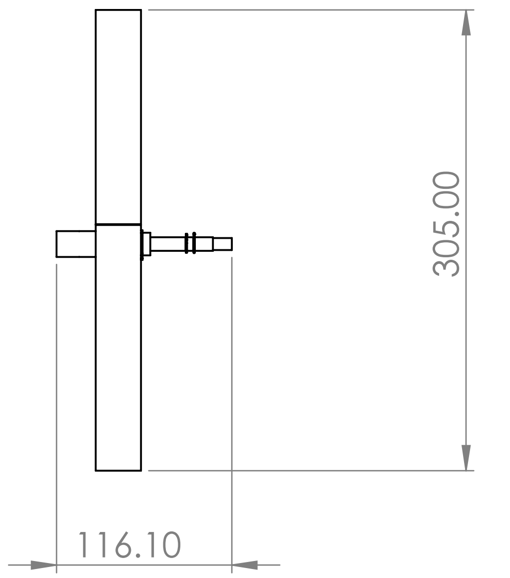
A4

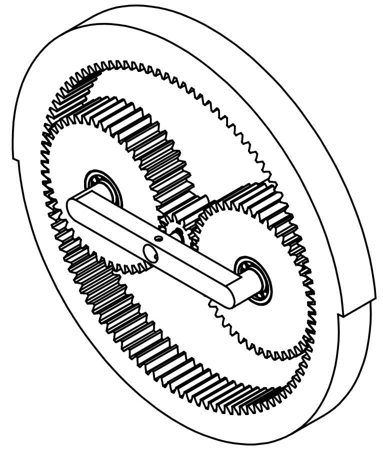
WEIGHT:

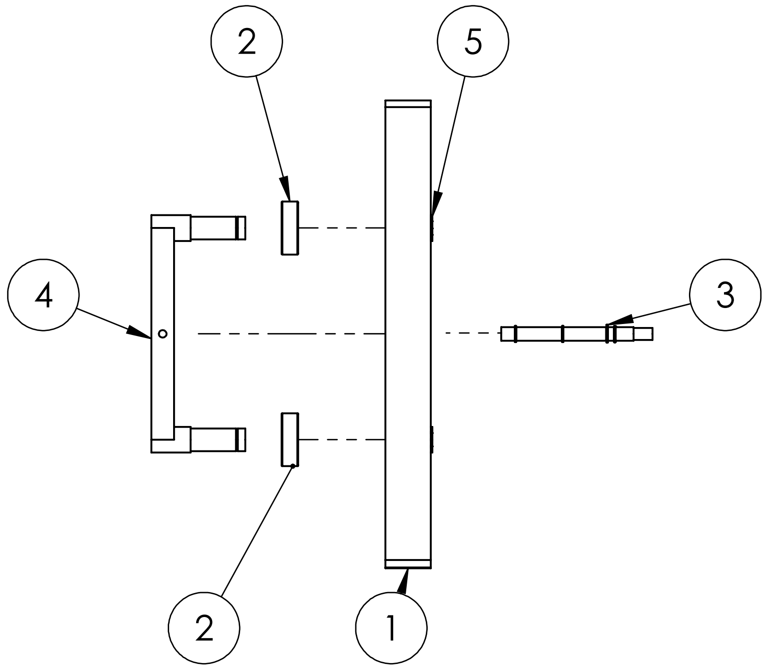
SCALE:2:1

SHEET 1 OF 1









ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	gear assemboly stage 1		1
2	skf_bearing_nj_202_ecp_2		2
3	stage 1 shaft		1
4	Shaft_2_Carrier_V2		1
5	98541A410	External Retaining Ring	2

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

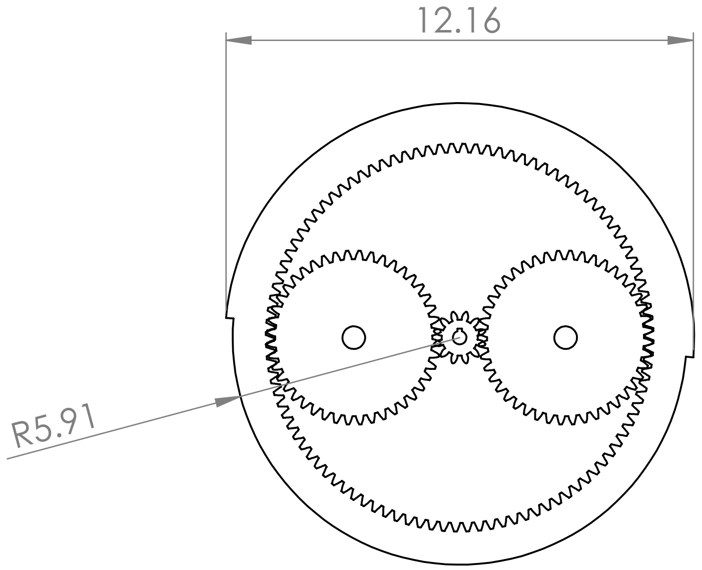
stage 1- gear and shaft

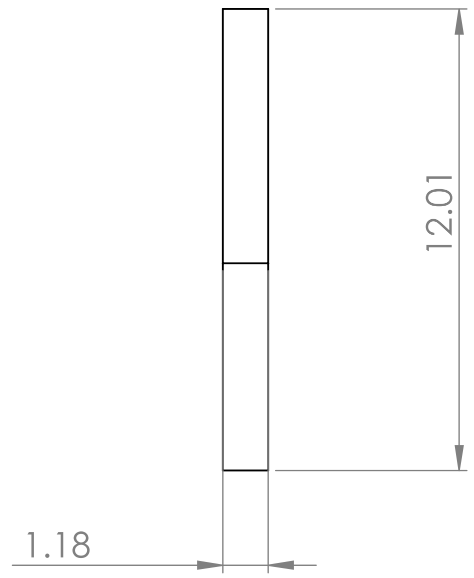
A3

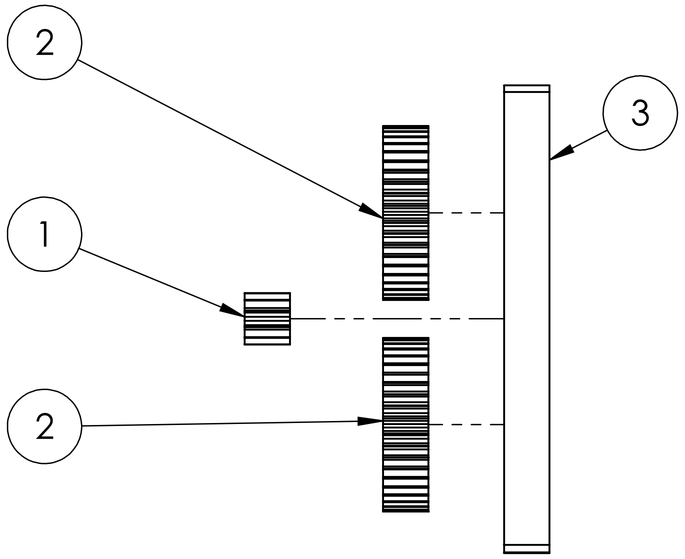
WEIGHT:

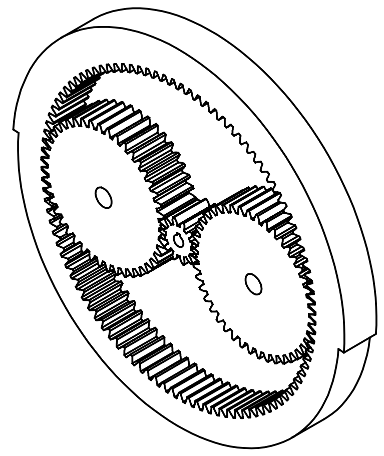
SCALE:1:5

SHEET 1 OF 1









ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Metric - Spur gear 2.5M 12T 20PA 30FW ---S12N75H50L9S1		1
2	Metric - Spur gear 2.5M 44T 20PA 30FW ---S44N75H50L70N		2
3	Metric - Internal spur gear 2.5M 100T 20PA 30FW ---S100S300OD 1AF		1

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

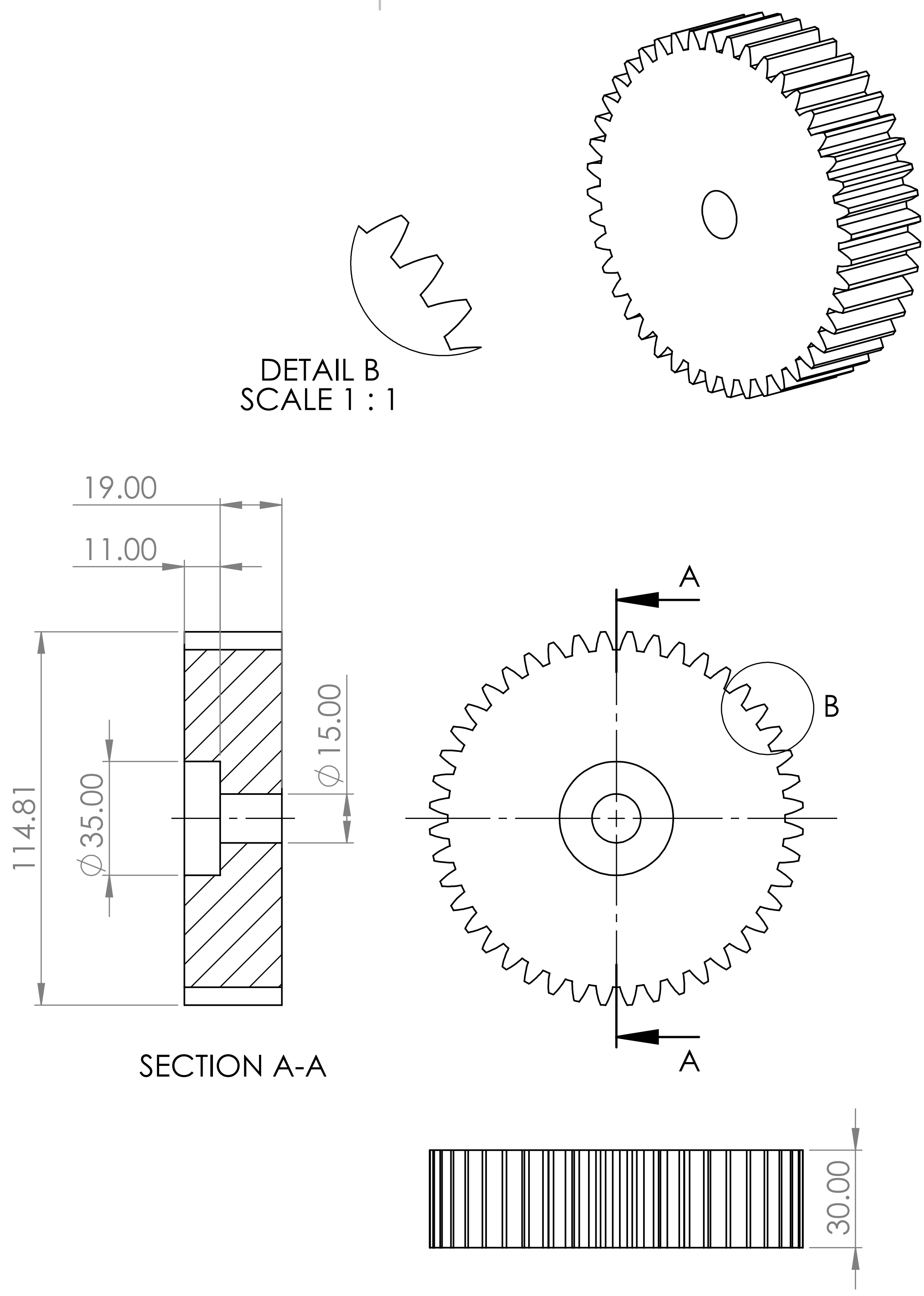
gear assemboly stage 1

A3

WEIGHT:

SCALE:1:5

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

WEIGHT:

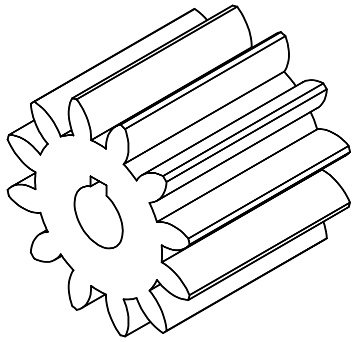
DWG NO.

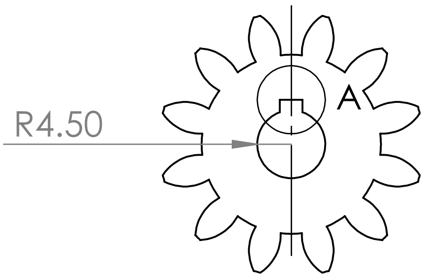
Planet-gear-1

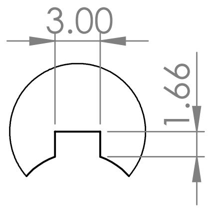
A4

SCALE:1:2

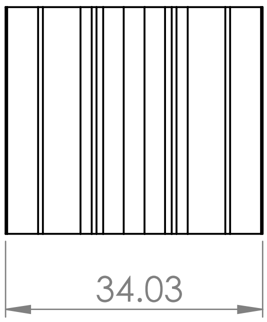
SHEET 1 OF 1







DETAIL A
SCALE 2 : 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

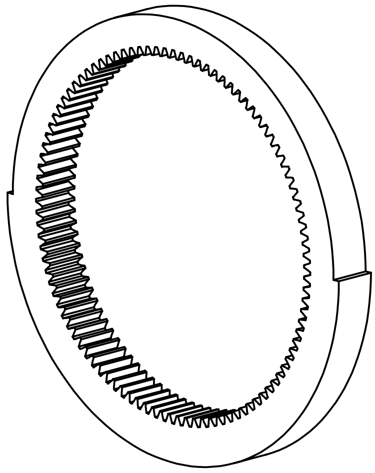
Sun-gear-1

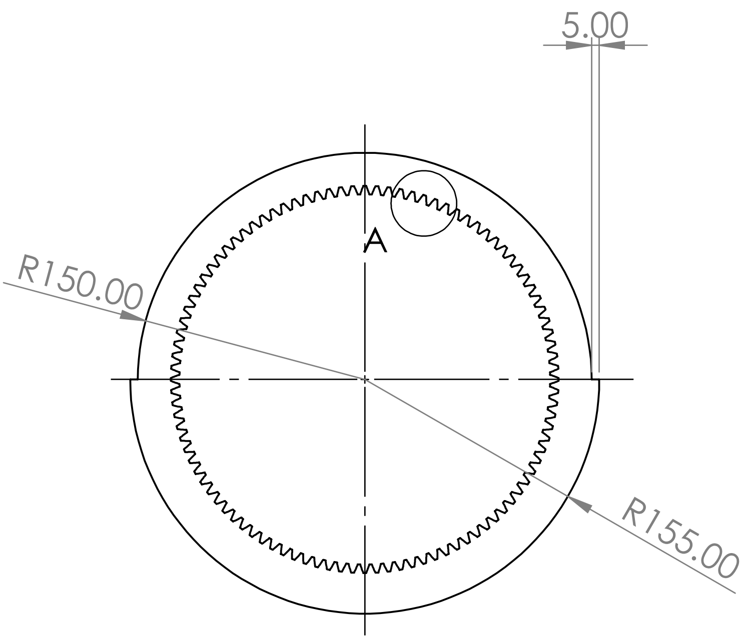
A4

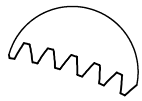
WEIGHT:

SCALE:1:1

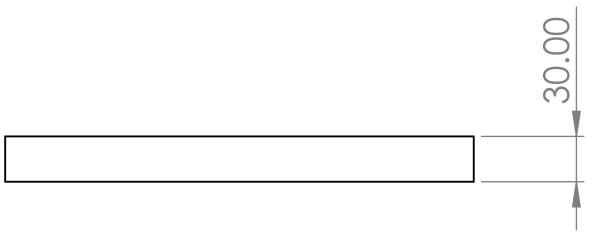
SHEET 1 OF 1







DETAIL A
SCALE 2 : 5



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DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

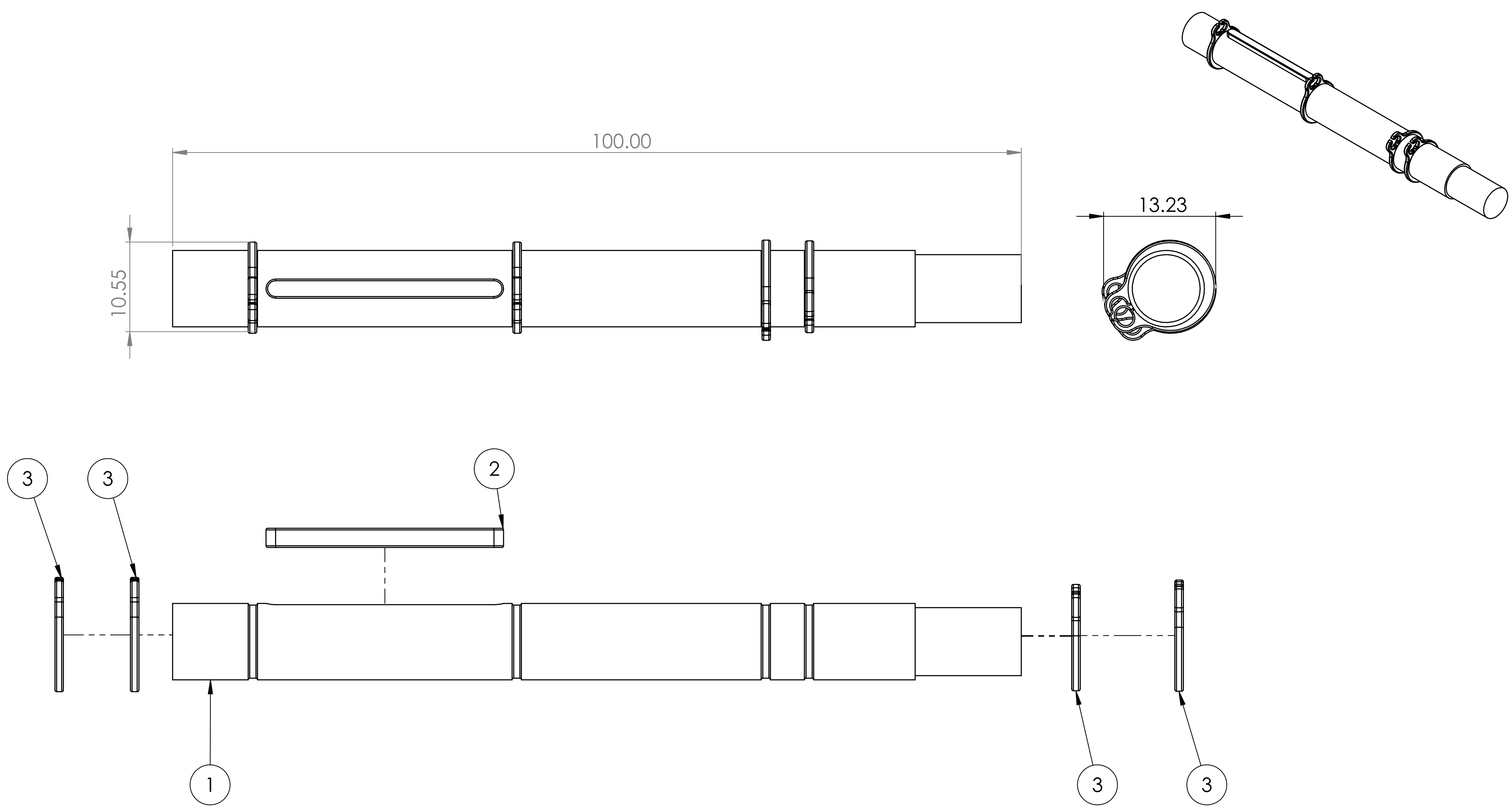
DWG NO. Ring-gear-1

A4

WEIGHT:

SCALE:1:5

SHEET 1 OF 1



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Shaft_1		1
2	Shaft_1_key		1
3	90967A155	External Retaining Ring	4

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

WEIGHT:

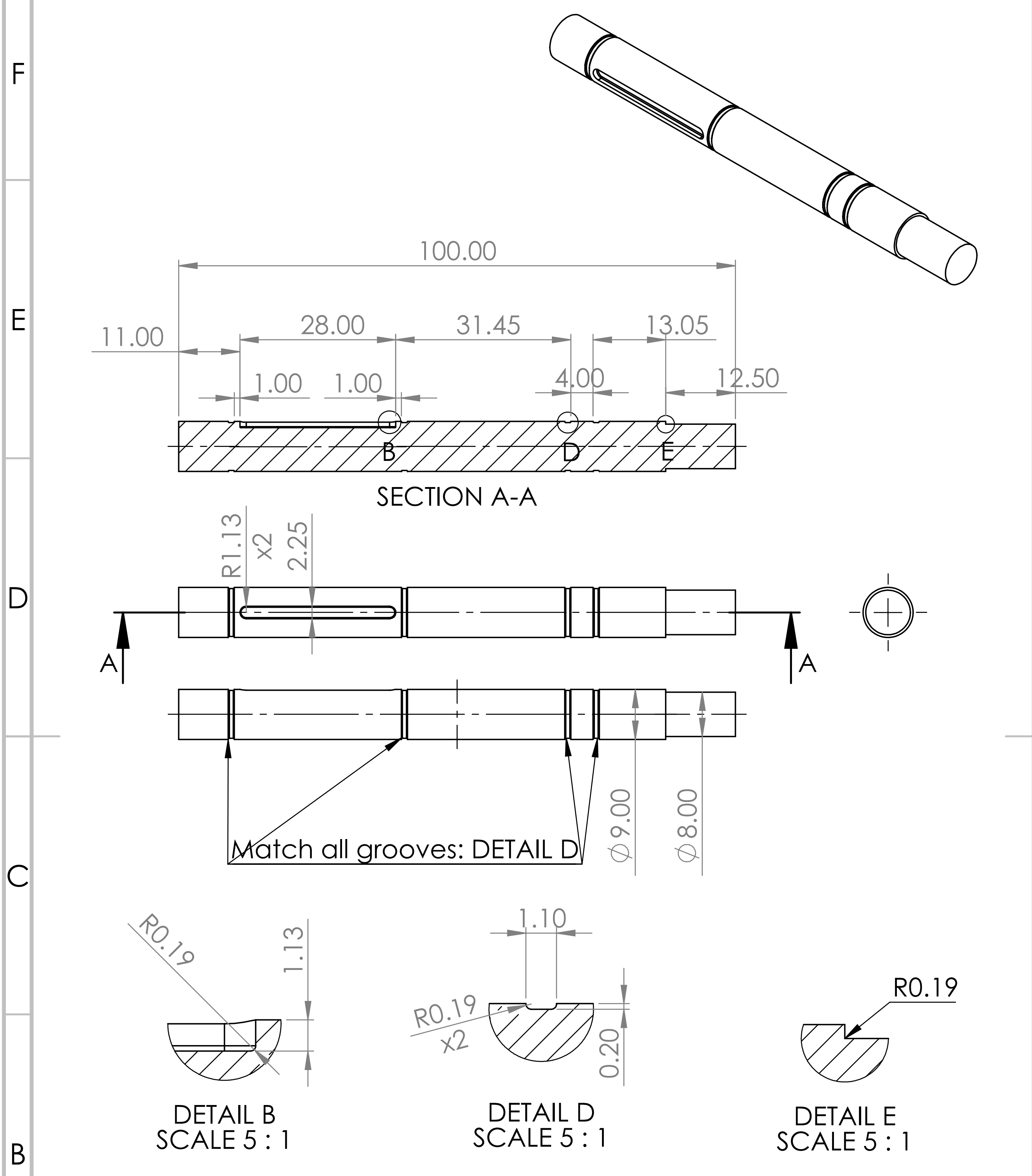
DWG NO.

stage 1 shaft

A3

SCALE:2:1

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
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SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

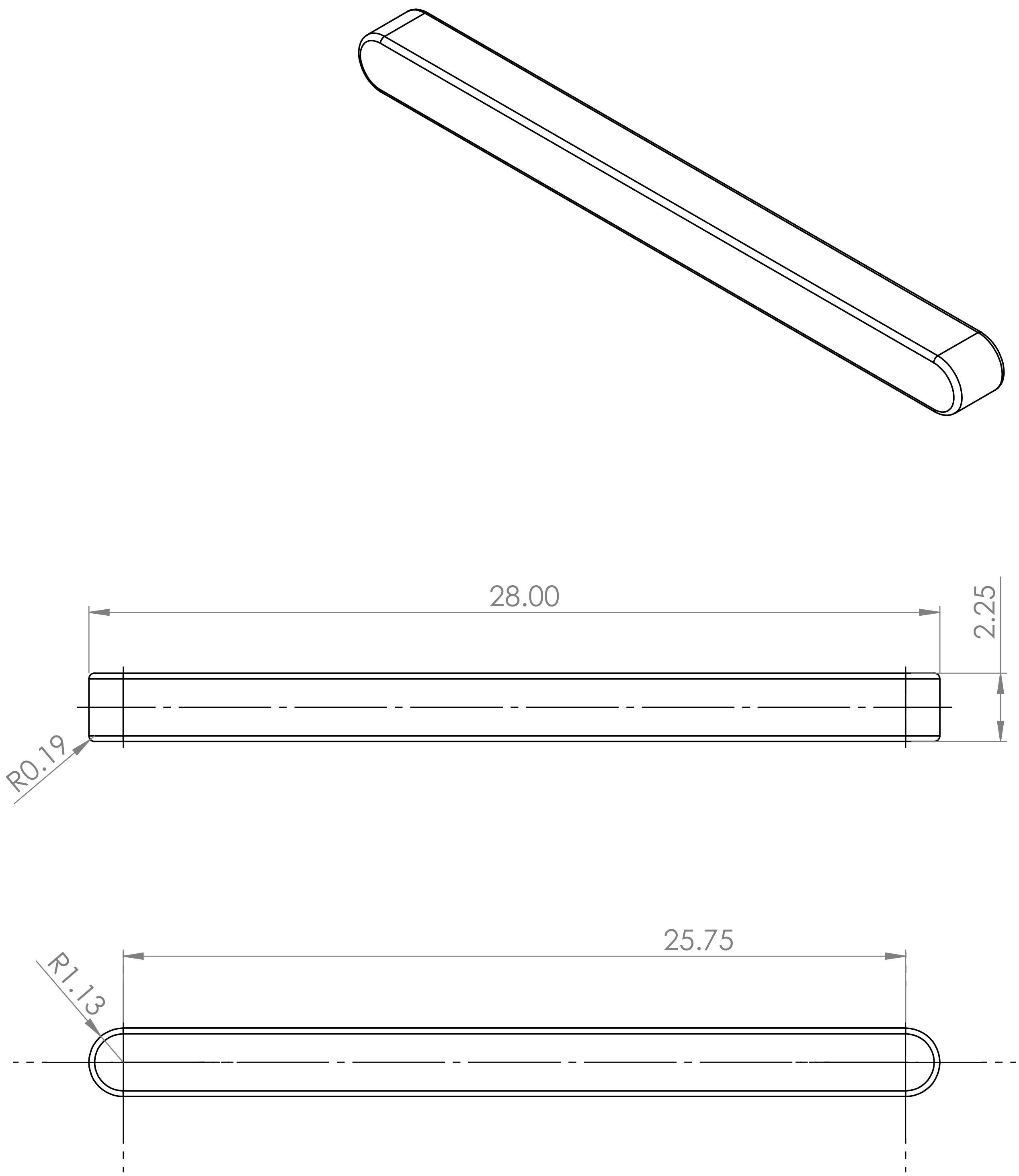
DWG NO.

Shaft-1

A4

SCALE:1:1

SHEET 1 OF 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

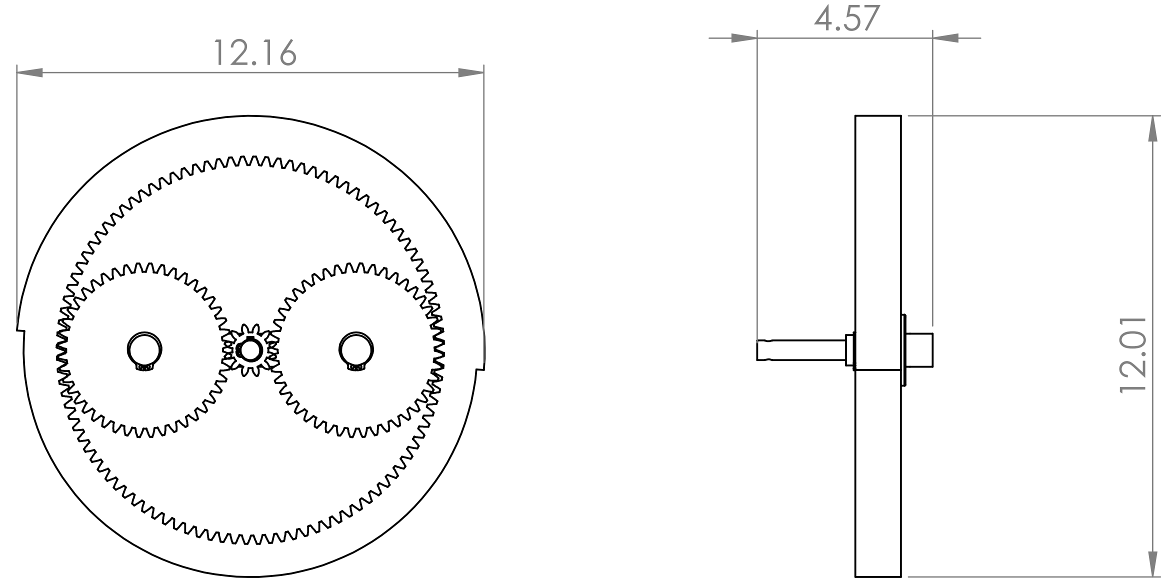
Shaft_1_key

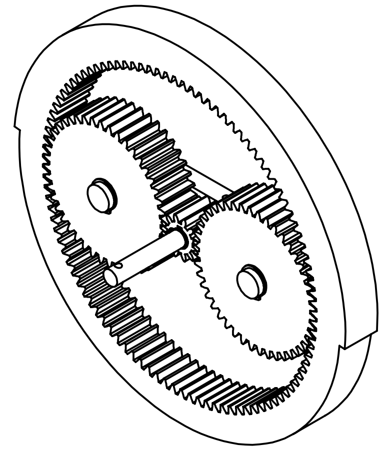
A4

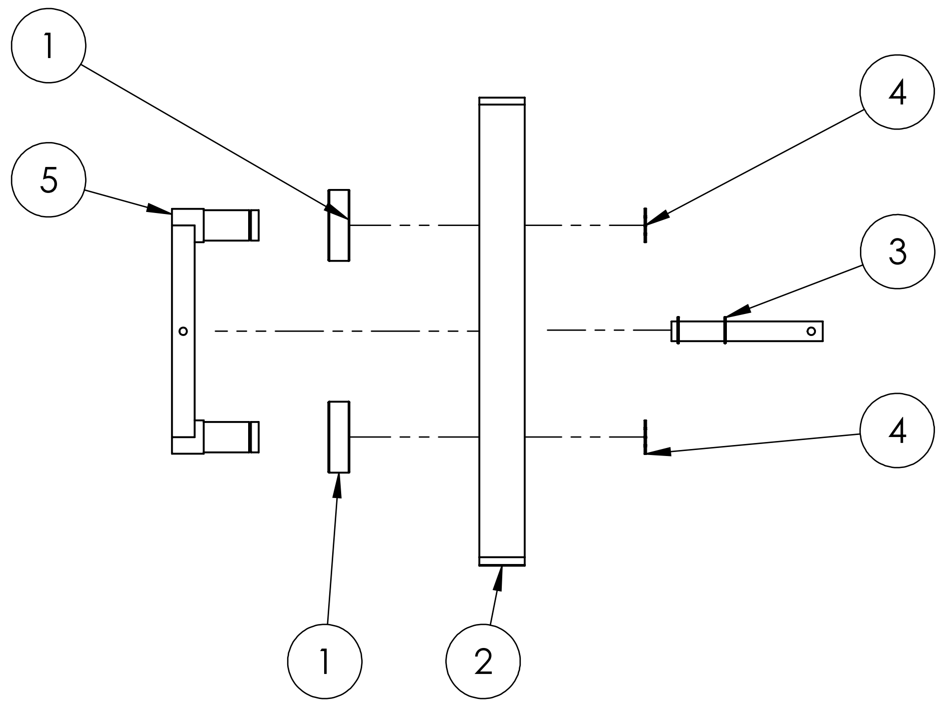
WEIGHT:

SCALE:5:1

SHEET 1 OF 1







ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	skf_bearing_n_204_ ecp_2		2
2	gear assemboly stage 2		1
3	stage 2 shaft		1
4	98541A123	External Retaining Ring	2
5	Shaft_3_Carrier_V2		1

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

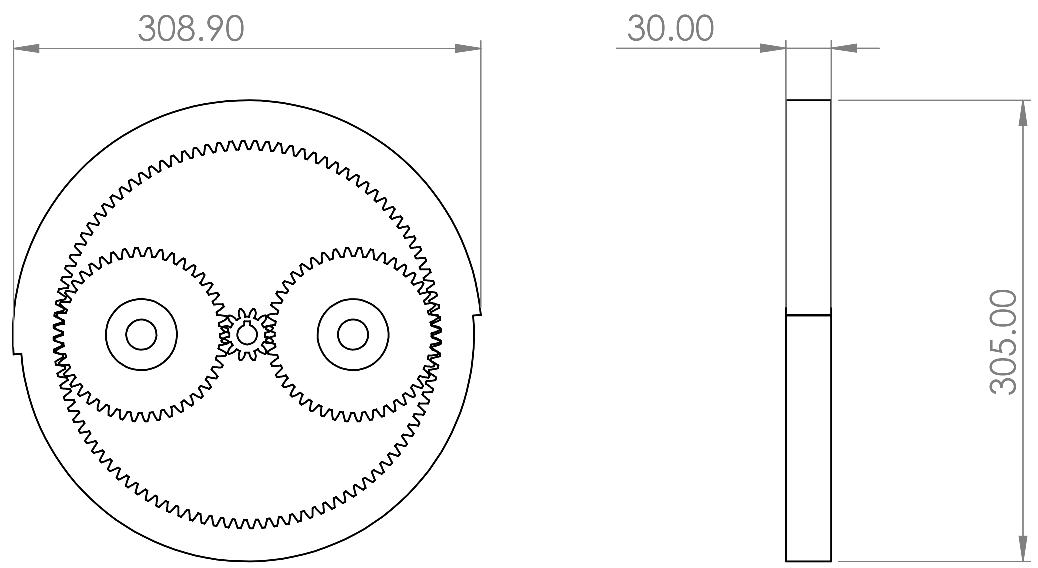
stage 2- gear and shaft

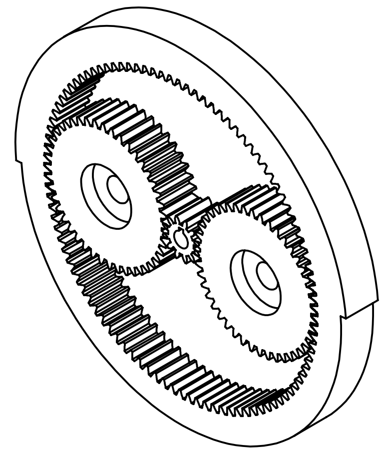
A3

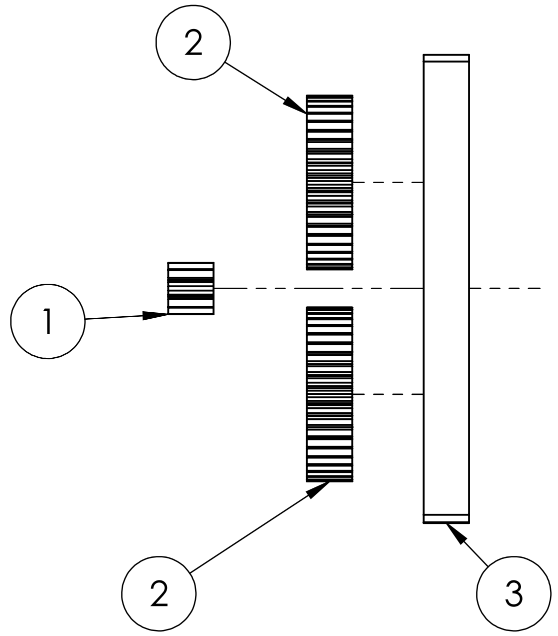
WEIGHT:

SCALE:1:5

SHEET 1 OF 1







ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Metric - Spur gear 2.5M 12T 20PA 30FW --- S12N75H50L14S1		1
2	Metric - Spur gear 2.5M 44T 20PA 30FW --- S44N75H50L48N		1
3	Metric - Internal spur gear 2.5M 100T 20PA 30FW --- S100S300OD 1AF		1

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

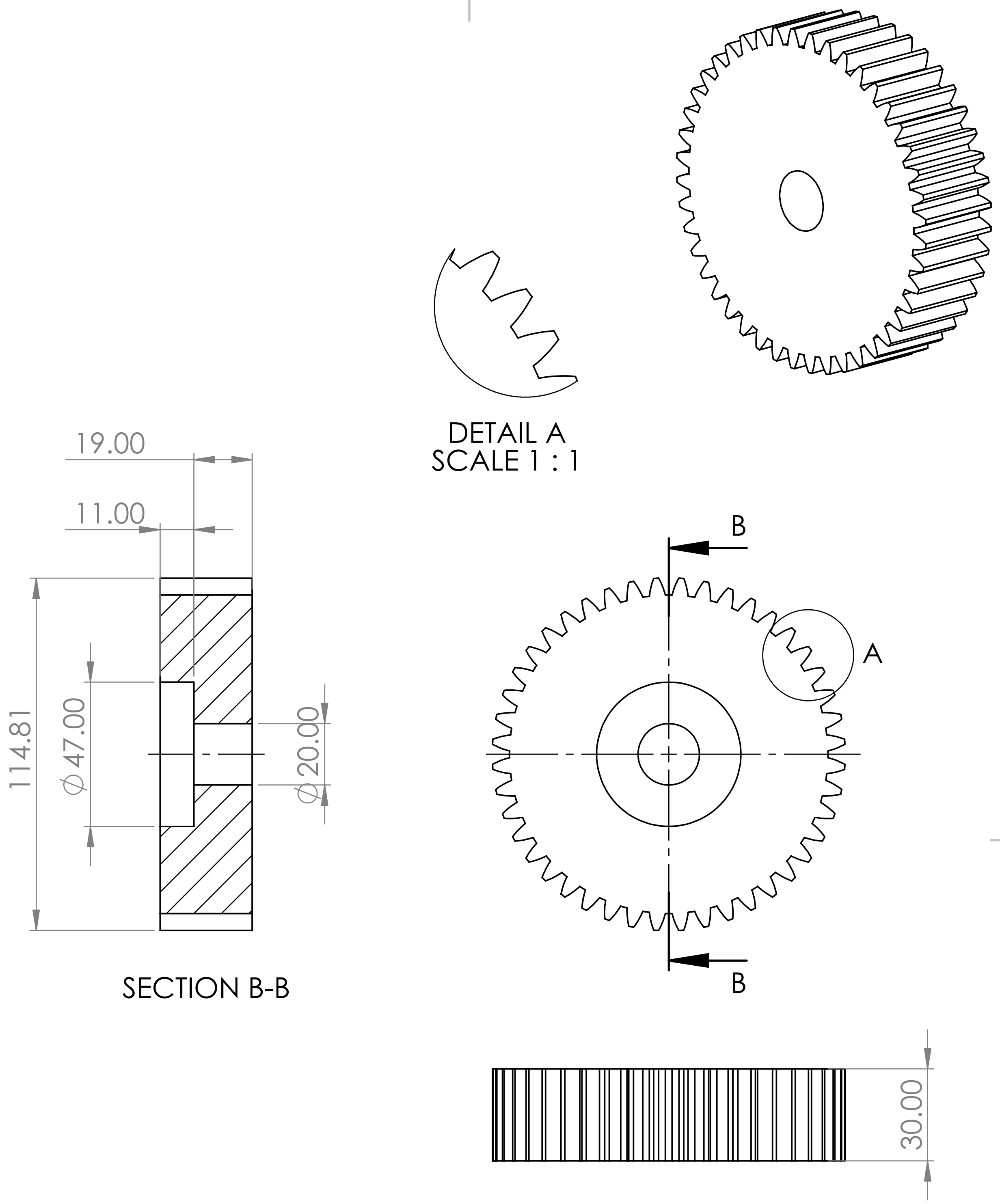
gear assemboly stage 2

A3

WEIGHT:

SCALE:1:5

SHEET 1 OF 1



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REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

MATERIAL:

WEIGHT:

TITLE:

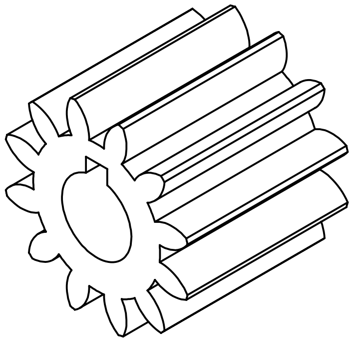
DWG NO.

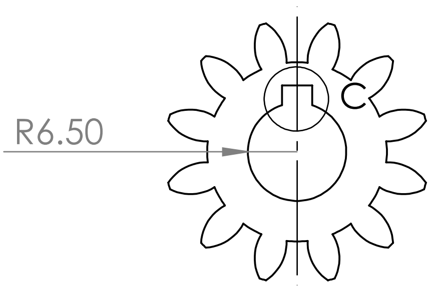
Planet-gear-2

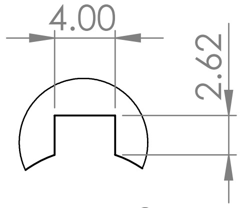
A4

SCALE:1:2

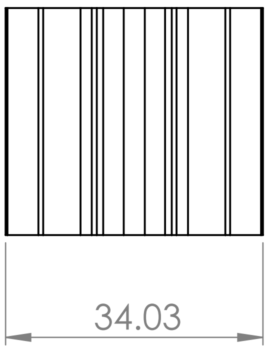
SHEET 1 OF 1







DETAIL C
SCALE 2 : 1



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SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

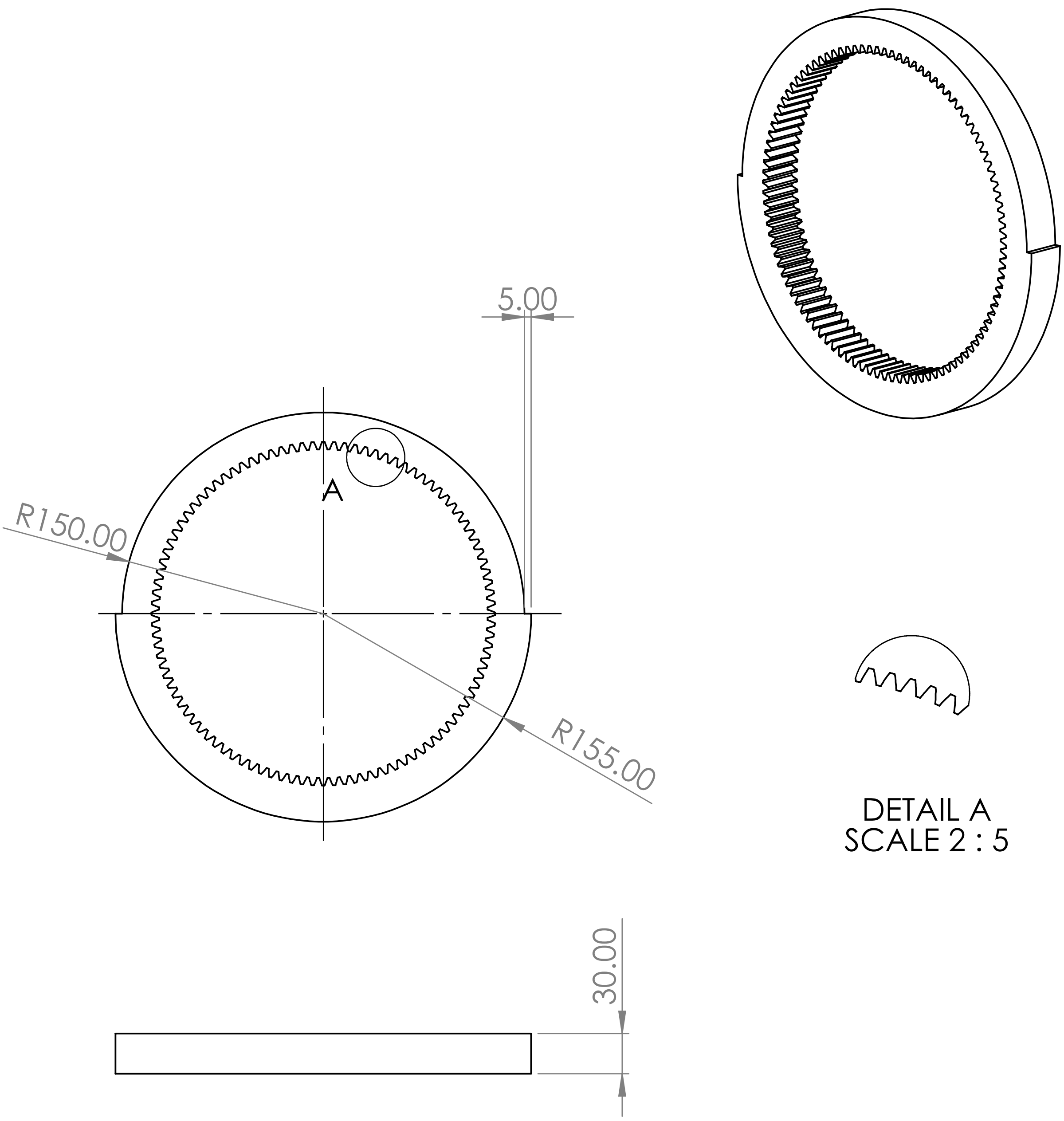
Sun-gear-2

A4

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



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TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

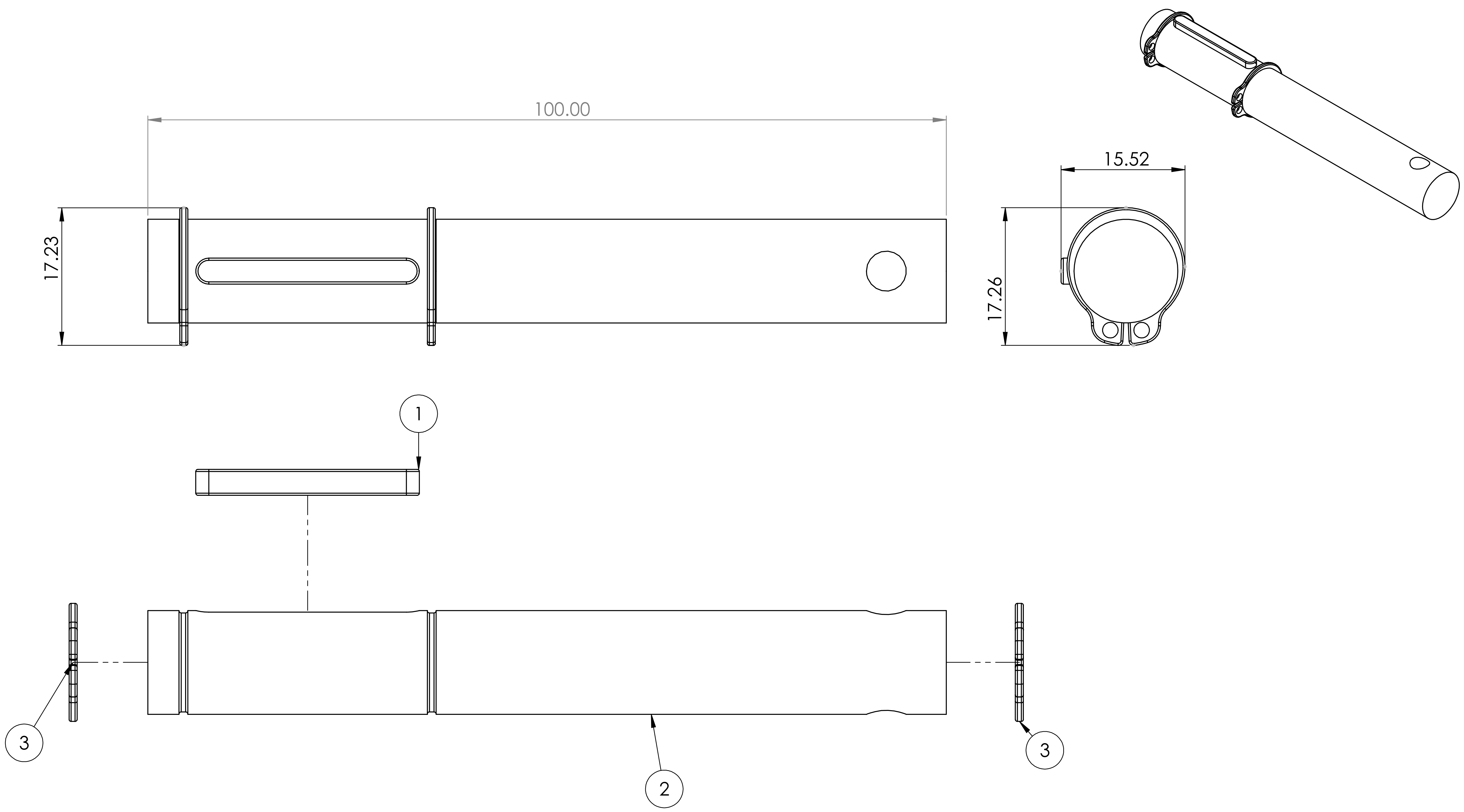
Ring-gear-1

A4

WEIGHT:

SCALE:1:5

SHEET 1 OF 1



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Shaft_2_key		1
2	Shaft_2		1
3	90967A170	External Retaining Ring	2

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

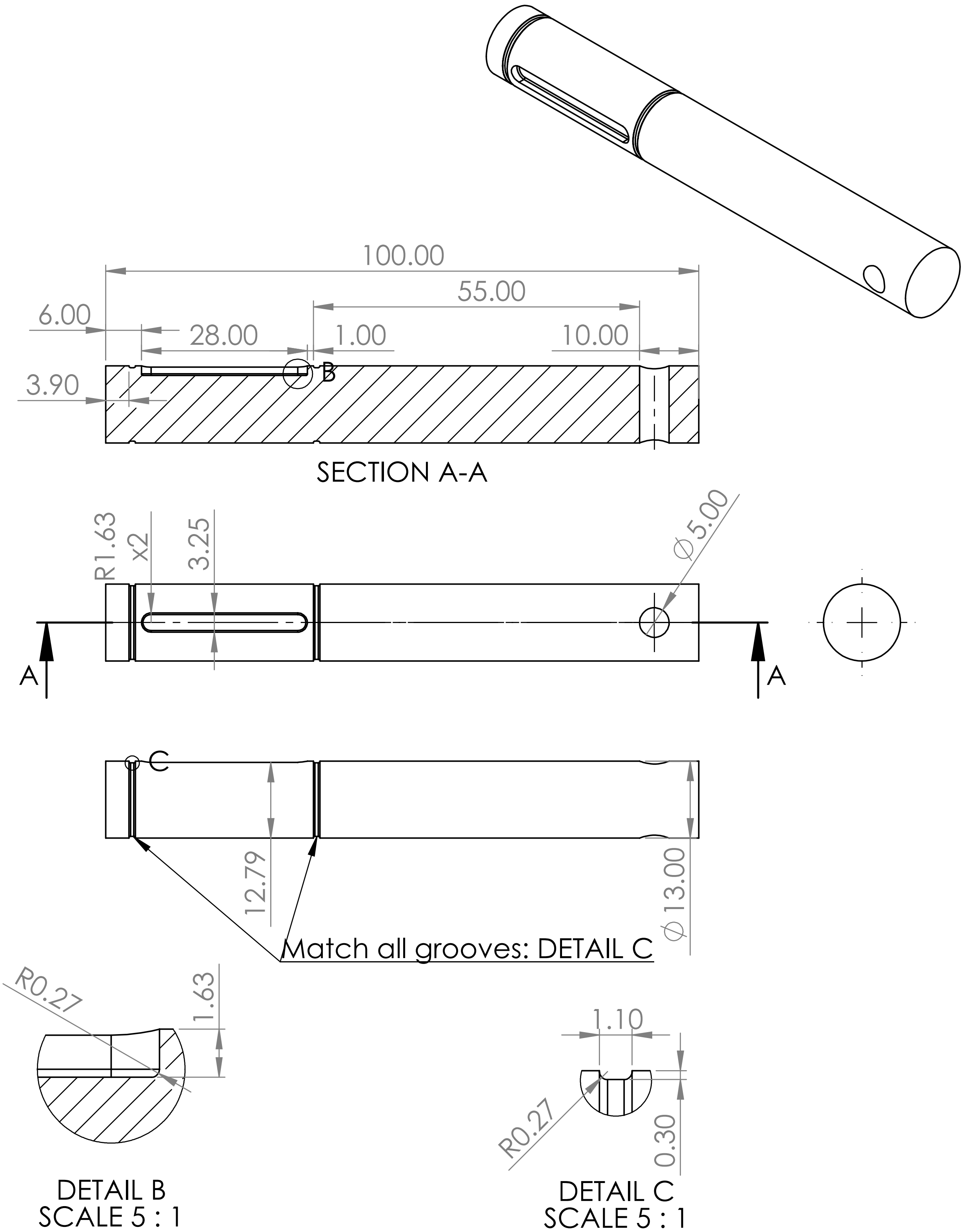
stage 2 shaft

A3

WEIGHT:

SCALE:2:1

SHEET 1 OF 1



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DIMENSIONS ARE IN MILLIMETERS
SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

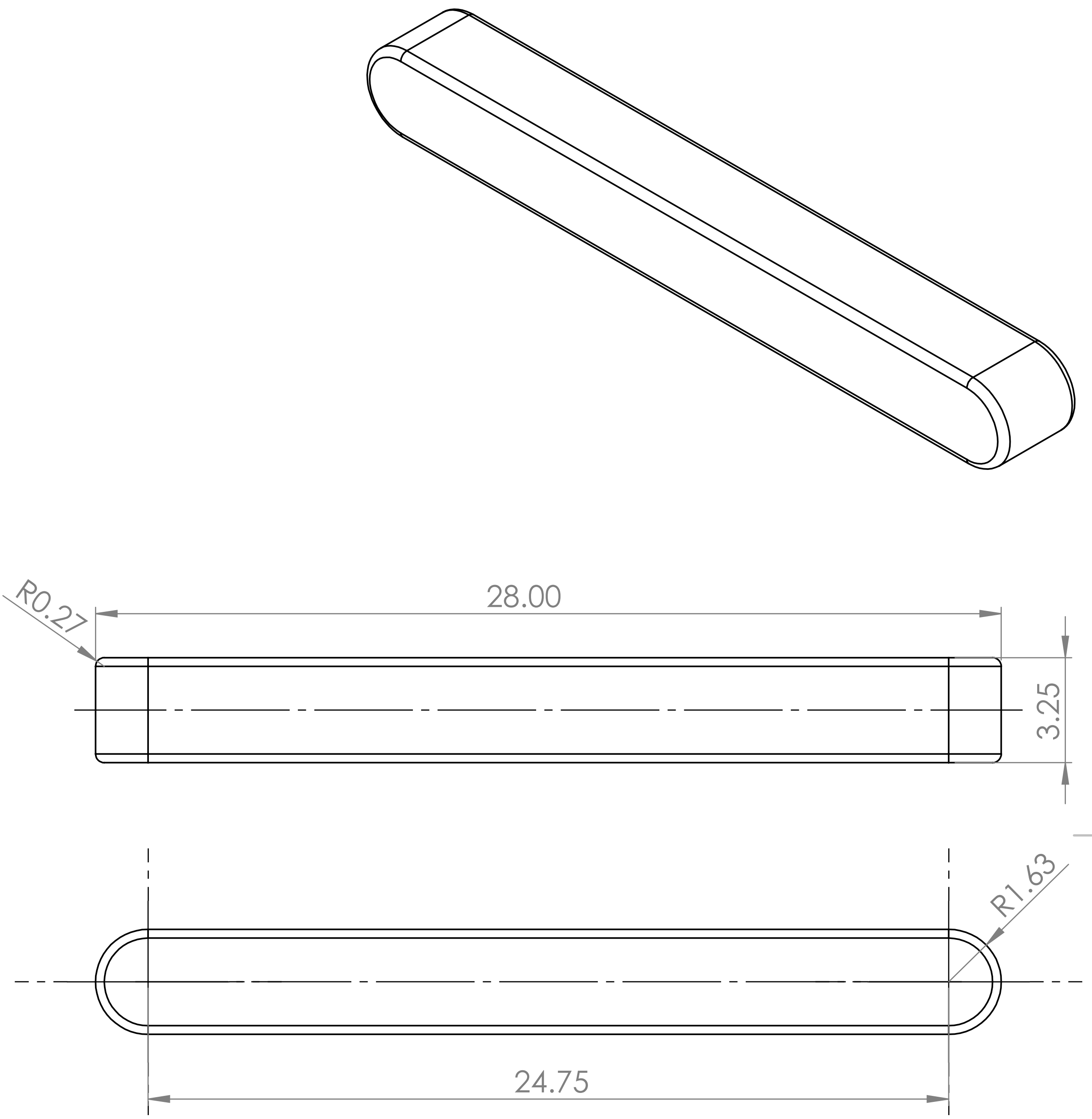
Shaft-2

A4

WEIGHT:

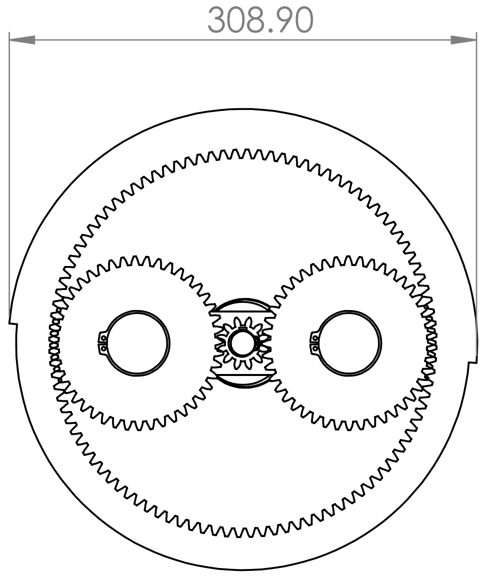
SCALE:1:1

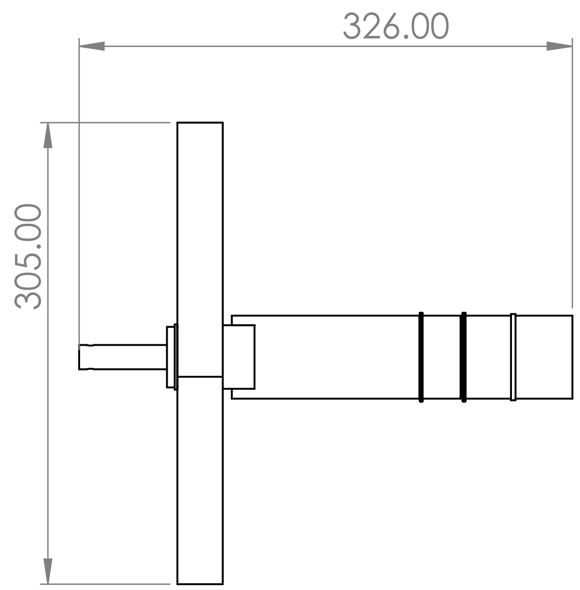
SHEET 1 OF 1

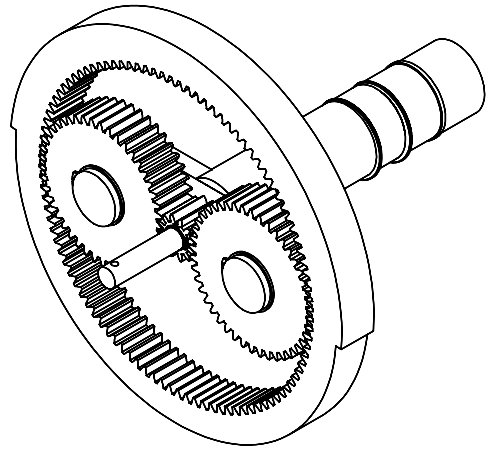


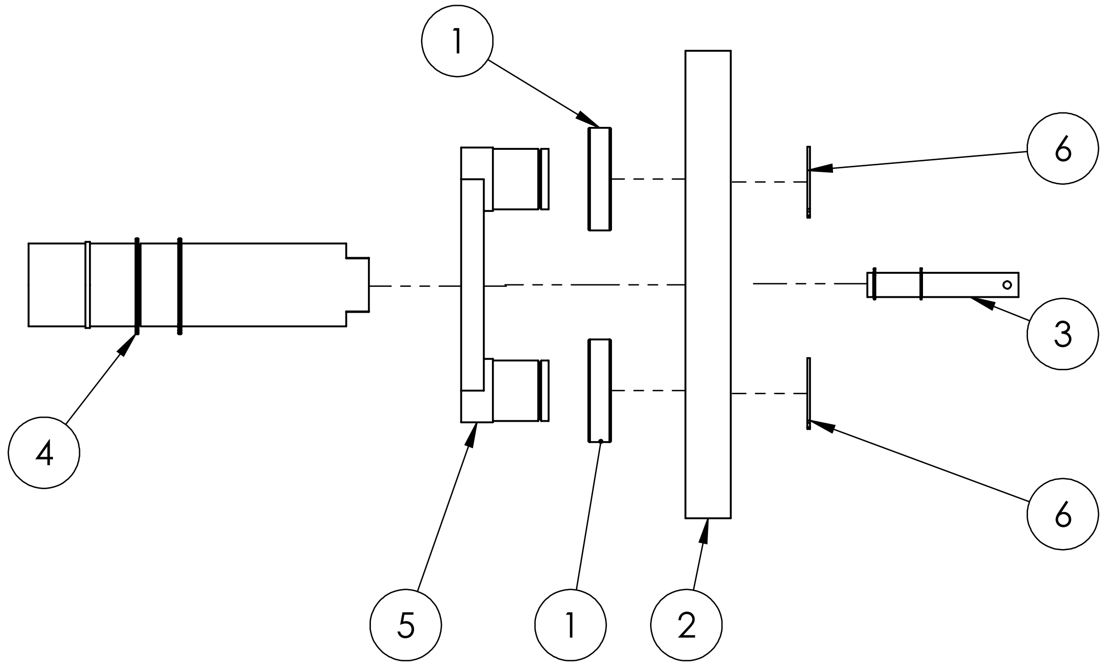
UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

	NAME	SIGNATURE	DATE		
DRAWN					TITLE:
CHK'D					
APPV'D					
MFG					
Q.A				MATERIAL:	DWG NO. Shaft-2-key — A4
				WEIGHT:	SCALE:5:1 — SHEET 1 OF 1









ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	skf_bearing_nu_1008_ml_2		2
2	gear assemboly stage 3		1
3	stage 3 shaft		1
4	stage 4 shaft		1
5	Shaft_4_Carrier_V2		1
6	98541A156	External Retaining Ring	2

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

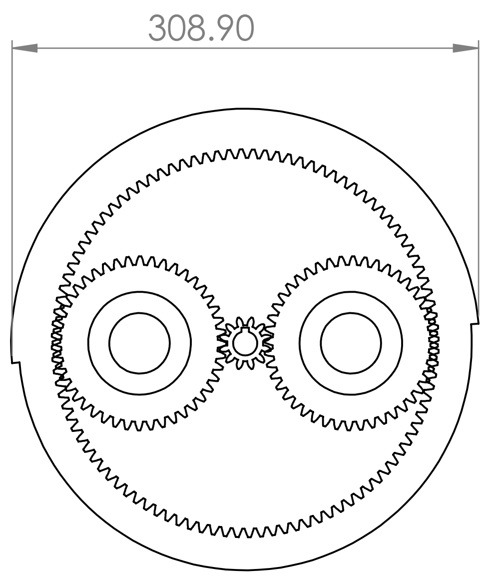
stage 3- gear and shaft

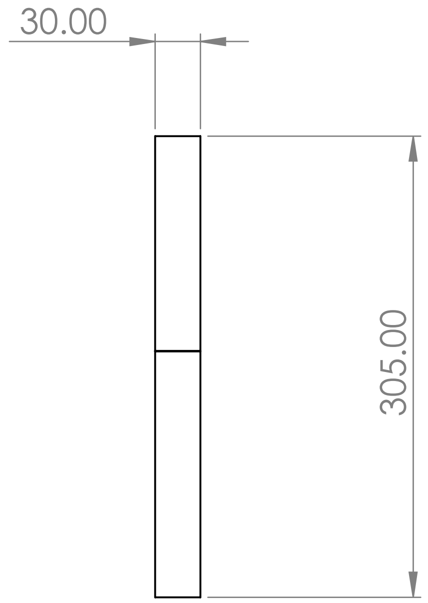
A3

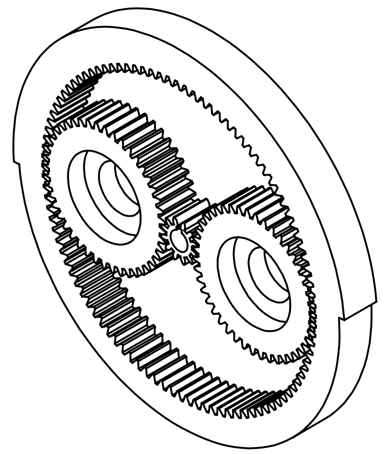
WEIGHT:

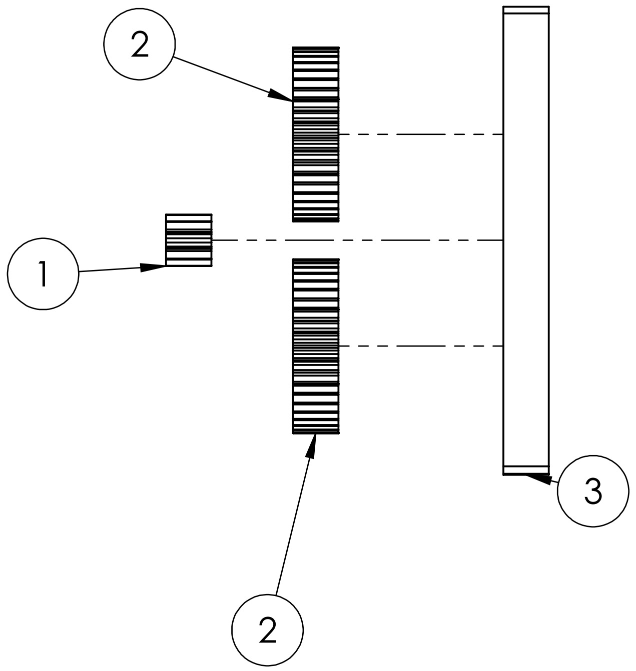
SCALE:1:5

SHEET 1 OF 1









ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	Metric - Spur gear 2.5M 12T 20PA 30FW ---S12N75H50L16S1		1
2	Metric - Spur gear 2.5M 44T 20PA 30FW ---S44N75H50L70N		2
3	Metric - Internal spur gear 2.5M 100T 20PA 30FW --- S100S300OD 1AF		1

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

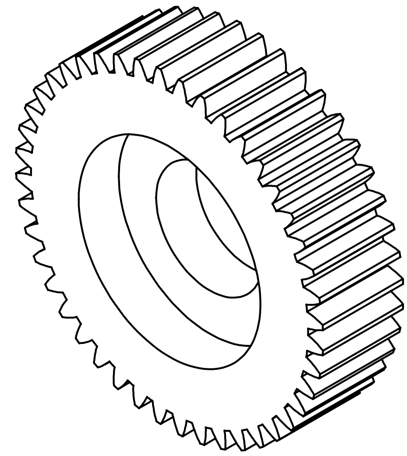
gear assemboly stage 3

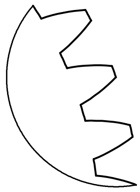
A3

WEIGHT:

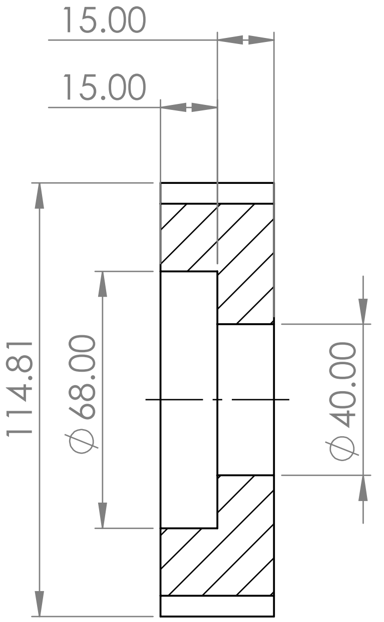
SCALE:1:5

SHEET 1 OF 1

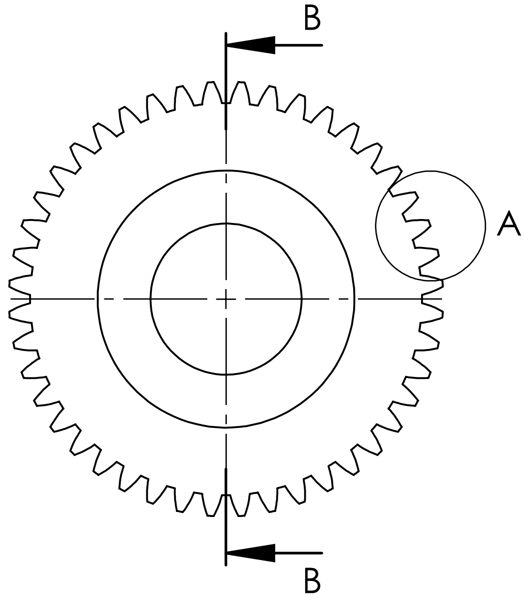


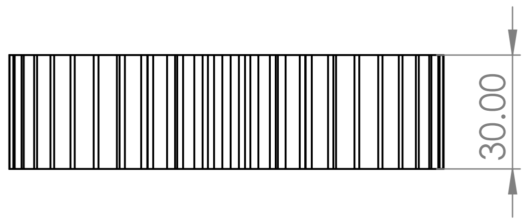


DETAIL A
SCALE 1 : 1



SECTION B-B





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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

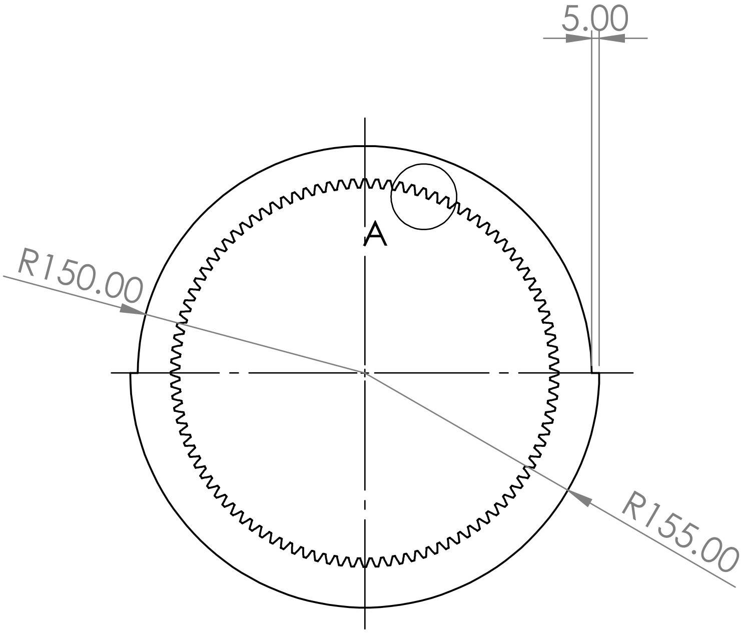
Planet-gear-3

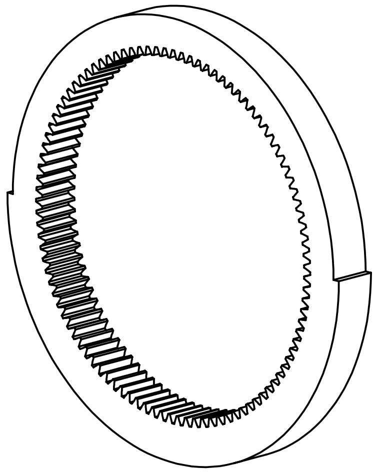
A4

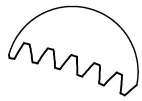
WEIGHT:

SCALE:1:2

SHEET 1 OF 1







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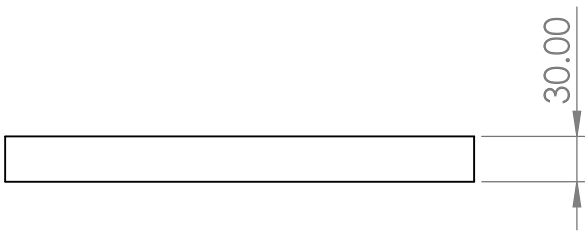
SCALE 2 : 5



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

WEIGHT:

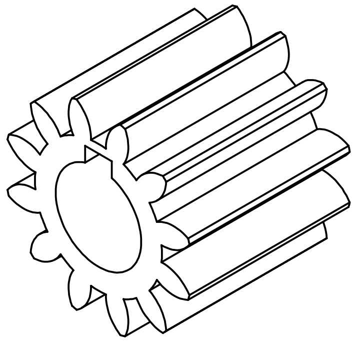
DWG NO.

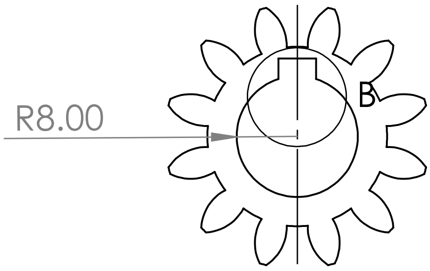
Ring-gear-1

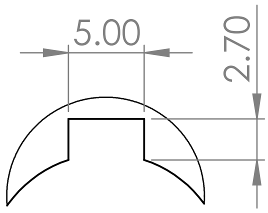
A4

SCALE:1:5

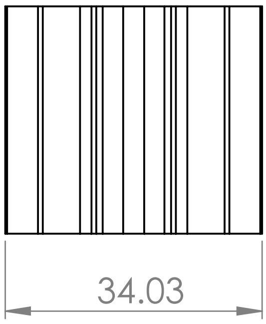
SHEET 1 OF 1







DETAIL B
SCALE 2 : 1



UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN MILLIMETERS SURFACE FINISH: TOLERANCES: LINEAR: ANGULAR:	FINISH:	DEBURR AND BREAK SHARP EDGES	DO NOT SCALE DRAWING	REVISION

	NAME	SIGNATURE	DATE	
DRAWN				
CHK'D				
APPV'D				
MFG				
Q.A				

TITLE:

MATERIAL:

DWG NO.

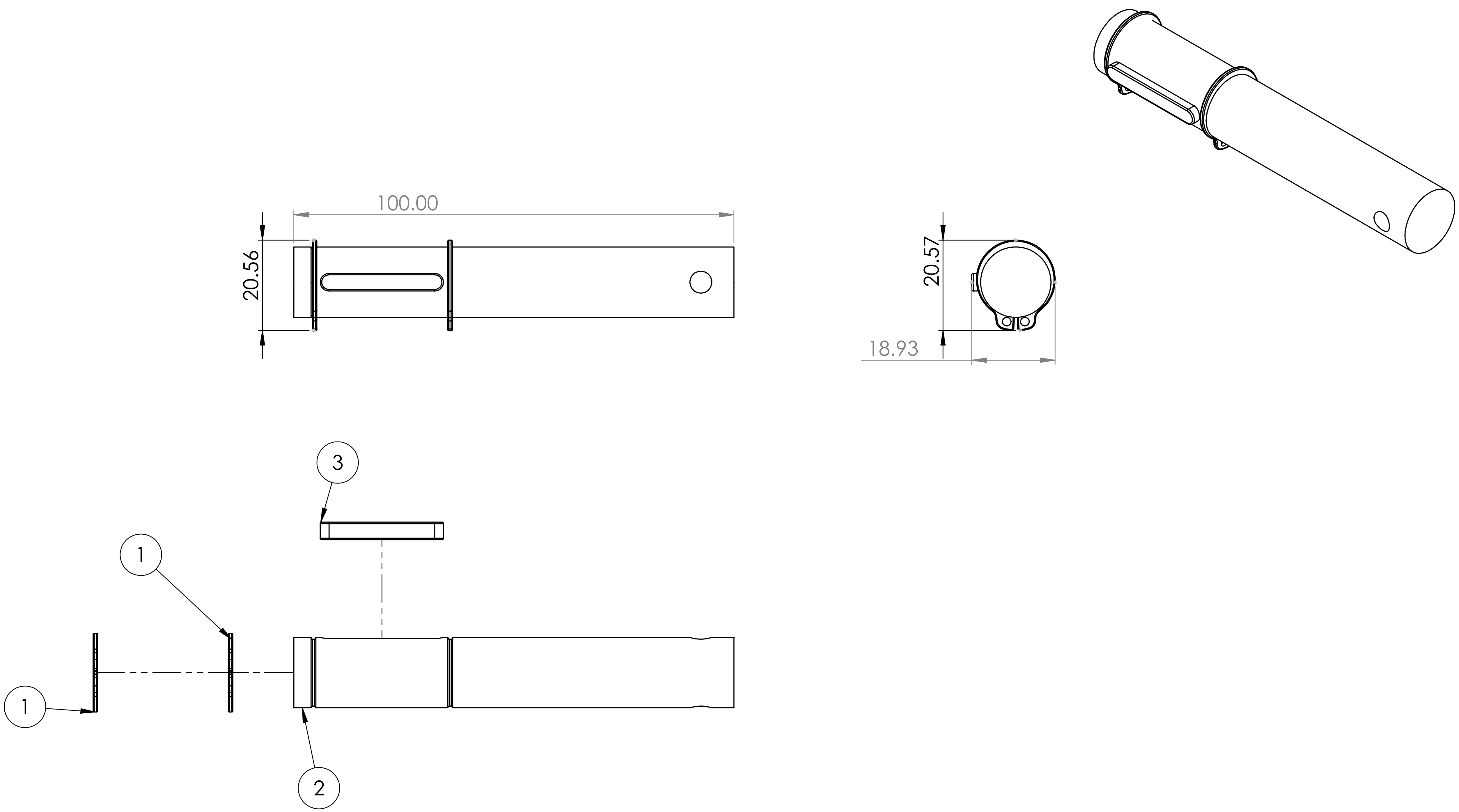
Sun-gear-3

A4

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



ITEM NO.	PART NUMBER	DESCRIPTION	QTY.
1	90967A205	External Retaining Ring	2
2	Shaft_3		1
3	Shaft_3_Key		1

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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

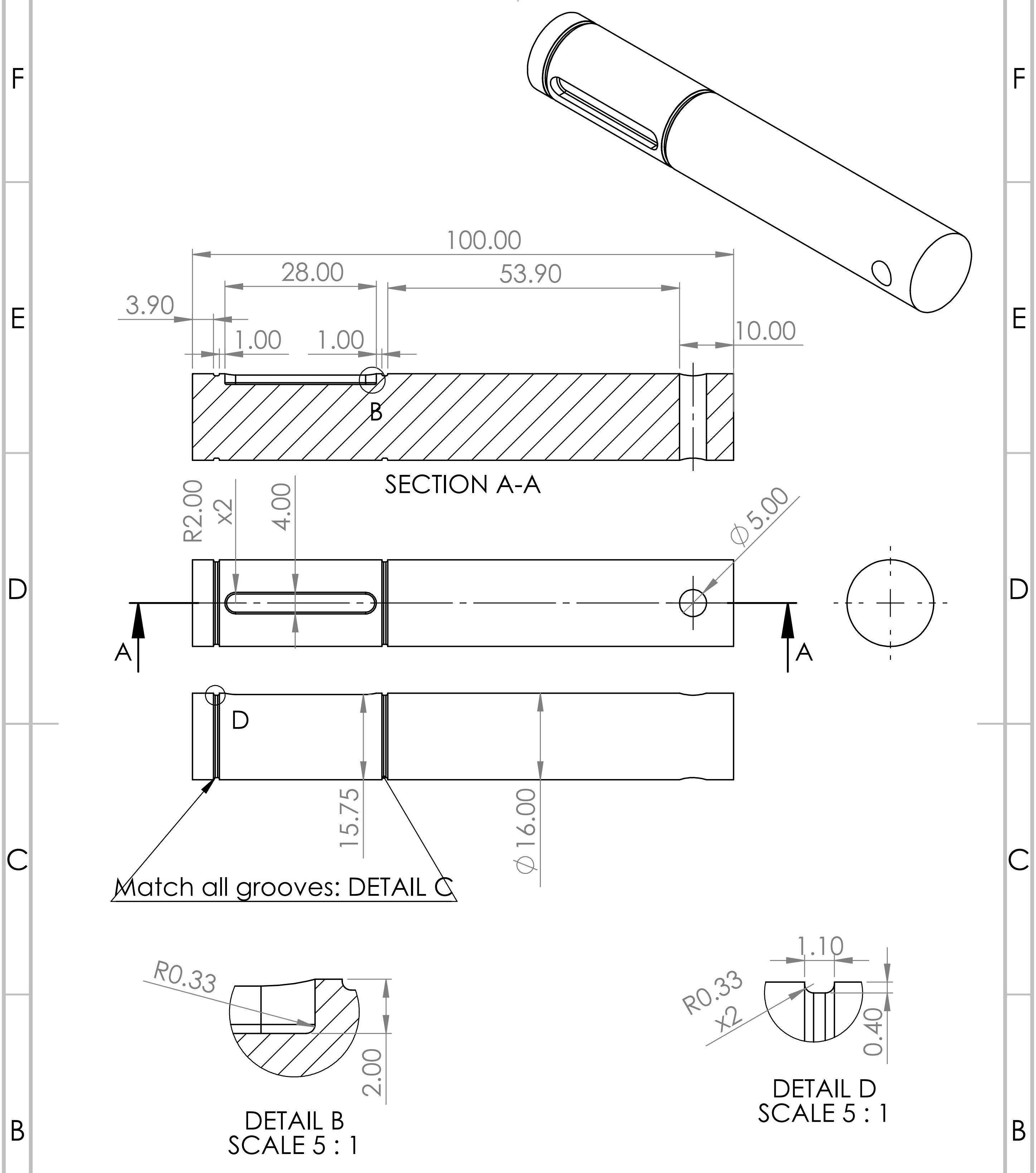
stage 3 shaft

A3

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



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TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

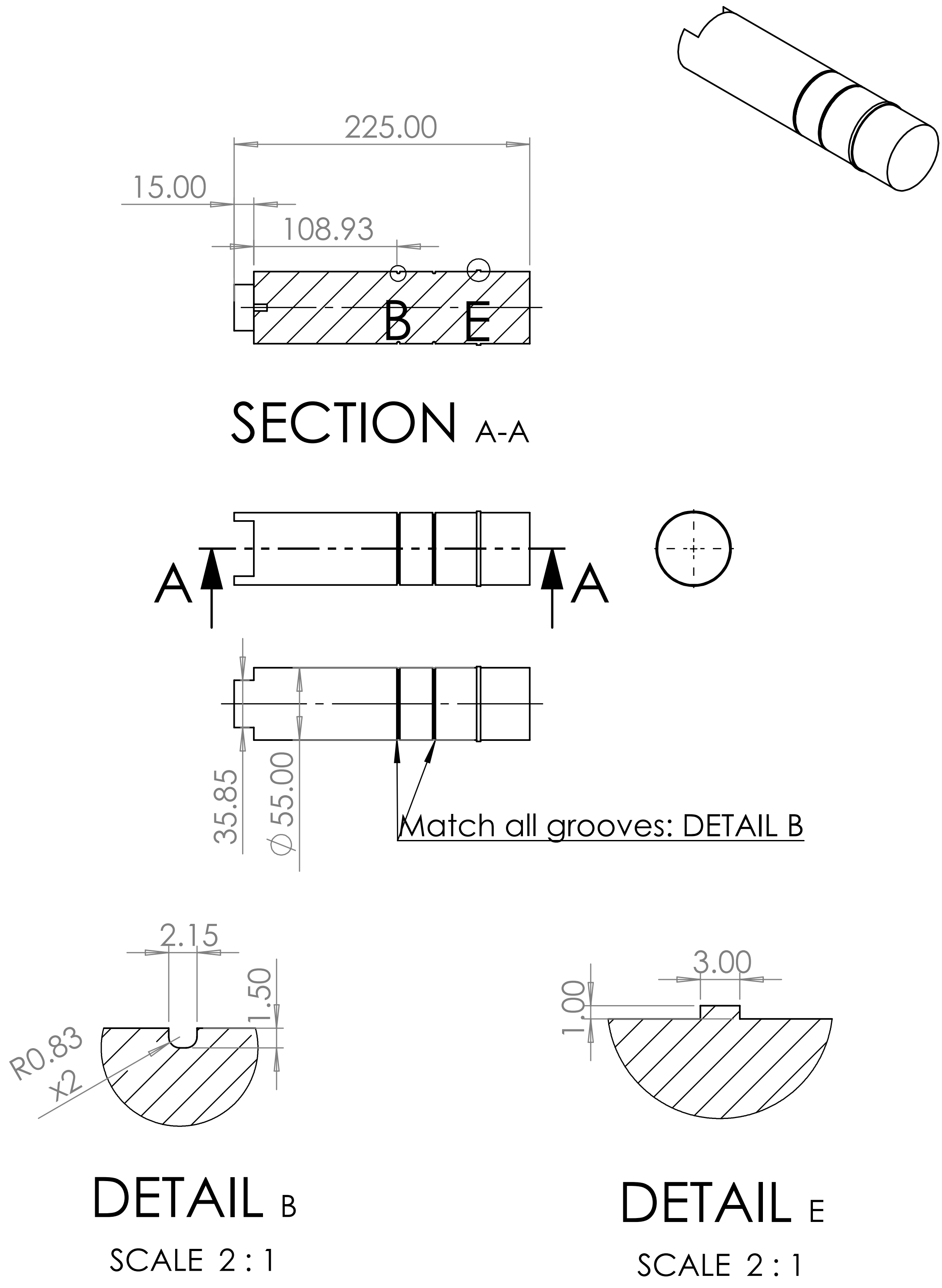
Shaft-3

A4

WEIGHT:

SCALE:1:1

SHEET 1 OF 1



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

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

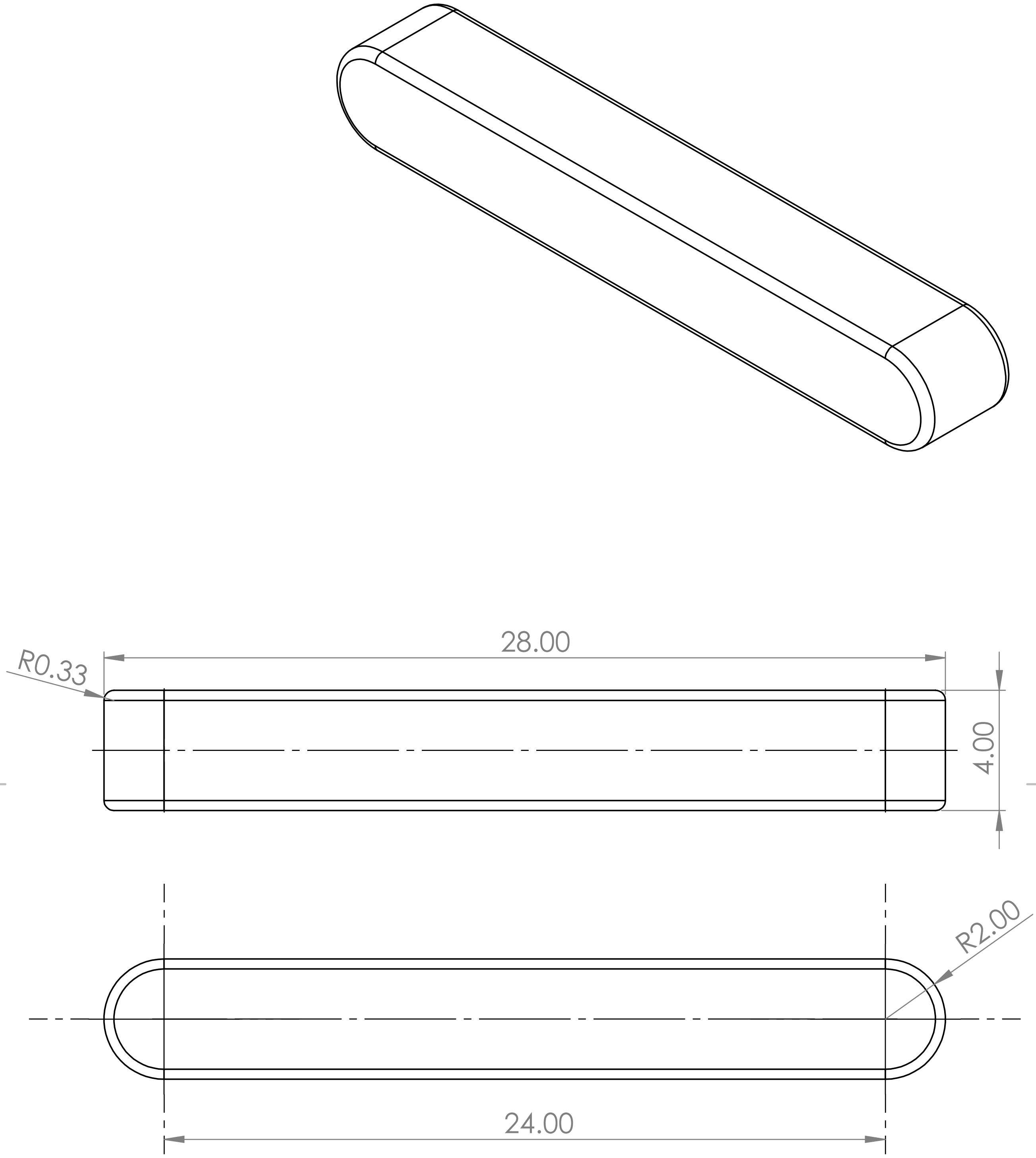
Shaft-4

A4

WEIGHT:

SCALE:1:5

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
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SURFACE FINISH:
TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND
BREAK SHARP
EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO.

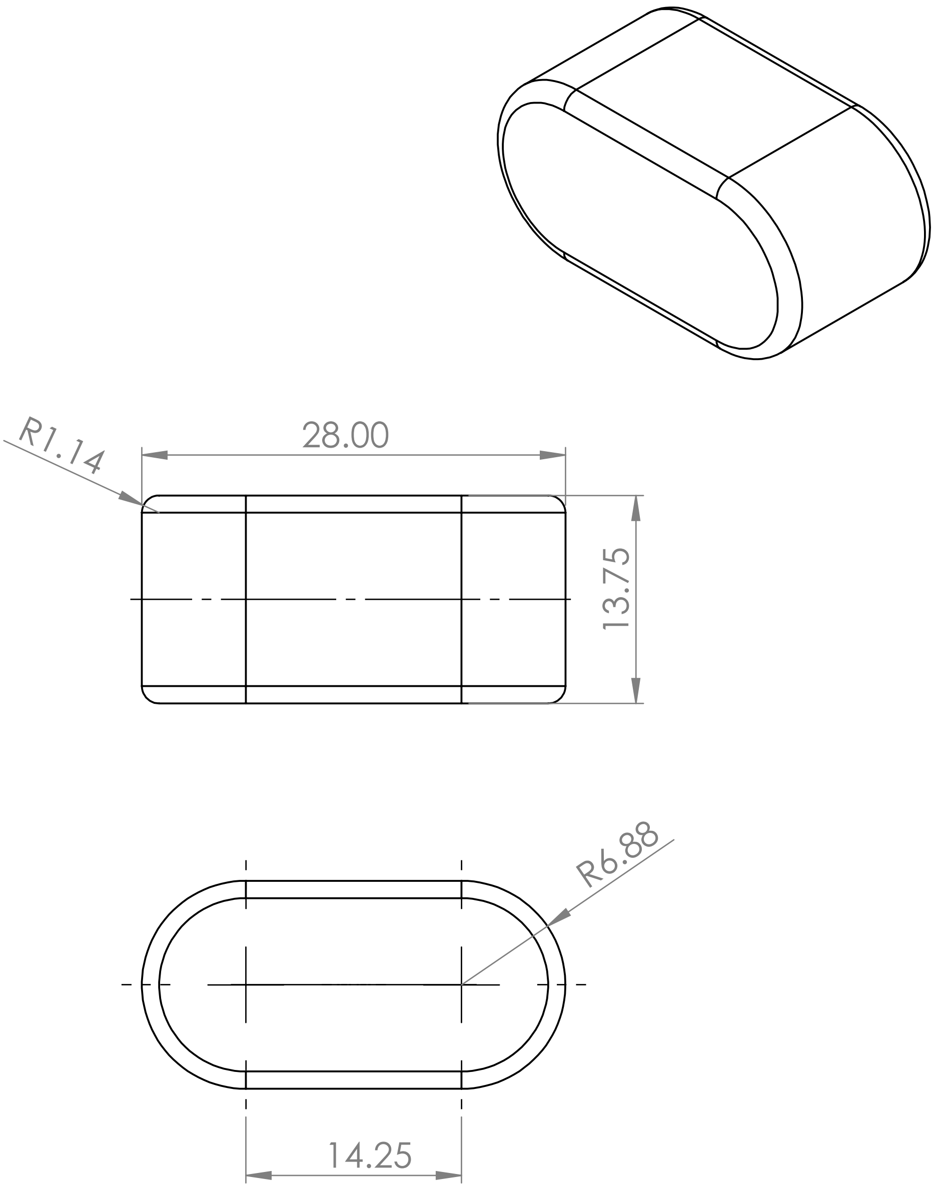
Shaft-3-key

A4

WEIGHT:

SCALE:5:1

SHEET 1 OF 1



UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN MILLIMETERS
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TOLERANCES:
LINEAR:
ANGULAR:

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

	NAME	SIGNATURE	DATE
DRAWN			
CHK'D			
APPV'D			
MFG			
Q.A			

TITLE:

MATERIAL:

DWG NO. Shaft-4-key

A4

WEIGHT:

SCALE:2:1

SHEET 1 OF 1

References

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