

Experimental Comparative Analysis of Cycloidal and Planetary Gearboxes for Compact Actuator Applications

University of Wales Trinity St David

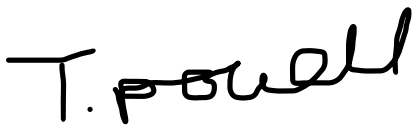
School of Engineering

Tyler Powell

Mechanical Engineering

2026

This thesis is the result of the author's original research. It has been composed by the author and has not been previously submitted for examination which has led to the award of a degree.

Signed: 

Date: 24/04/2026

I acknowledge the use of ChatGPT (<https://chatgpt.com/>) on 24/04/2026 to proofread my report, make grammatical suggestions to make the report clearer and help condense the report into a suitable word count.

I confirm that I have not presented any AI generated text as my own work. I confirm I have copies of my drafts, notes, and other resources I used before, and after the use of AI tools which I may be asked to provide in evidence.

Table of Contents

Table of Figures	5
1. Introduction	6
1.1. Aim and Objective	7
1.1.1. Aim	7
1.1.2. Objective	7
1.2. Typical Applications	7
1.3. Project Timing Plan	8
2. Literature Review	9
2.1. Chapter Overview/Introduction	9
2.2. Broad context and importance	9
2.3. Thematic Review of key topics	9
2.3.1. Planetary Gear Systems	9
2.3.2. Cycloidal Drive Systems	10
2.3.3. Performance Evaluation.....	11
2.4. Critical Synthesis and Identification of Research Gap.....	13
2.5. Summary of the Chapter	14
3. Methodology	15
3.1. Experimental Approach	15
3.2. Gearbox Design Specification	15
3.3. Gearbox Manufacture.....	22
3.4. Test Rig Design	24
3.5. Measurement Procedures	28
3.5.1. Efficiency Measurement.....	28
3.5.2. Backlash Measurement	29
3.5.3. Torsional Stiffness.....	30
3.5.4. Efficiency as a Function of Speed.....	30
3.6. Experimental Control and Repeatability.....	30

3.6.1.	Control of Variables	31
3.6.2.	Repeatability of Measurements.....	31
3.6.3.	Calibration of Measurement Systems	32
3.7.	Sources of Error and Uncertainty.....	32
4.	Results and Discussion	34
4.1.	Transmission Efficiency	34
4.2.	Backlash.....	36
4.3.	Torsional Stiffness	37
4.4.	Comparative Summary.....	39
5.	Critical Evaluation	41
5.1.	Design Evaluation	41
5.2.	Material and Manufacturing Limitations	41
5.3.	Experimental Method Limitations.....	42
5.4.	Test Rig Limitations	43
5.5.	Comparative Validity of Results	43
5.6.	Project Reflection and Management.....	43
6.	Design Improvements and Future Work	46
6.1.	Gearbox Design Improvements	46
6.1.1.	Cycloidal Gearbox	46
6.1.2.	Planetary Gearbox.....	46
6.2.	Experimental and Test Rig Improvements	47
6.3.	Improvements to Experimental Methodology	48
6.4.	Future Work	48
7.	Conclusion	49
	References	51
	Appendix 1.....	53
	Appendix 2.....	56

Table of Figures

Figure 1: Project Gantt Chart	8
Figure 2: Planetary drive diagram - TEC-SCIENCE.com	10
Figure 3: Cycloidal drive diagram - TEC-SCIENCE.com	11
Figure 4: Final iteration of planetary gearbox with colour coded components	16
Figure 5: Final iteration of internal planetary gearbox mechanism with colour coded components ..	16
Figure 6: Final iteration of cycloidal gearbox with colour coded components	17
Figure 7: Final iteration of internal cycloidal gearbox mechanism with colour coded components...	17
Figure 8: Cycloidal output pin carrier reinforcement	18
Figure 9: Planetary output shaft and second phase planet carrier reinforcement	18
Figure 10: Metal component labelling in cycloidal	19
Figure 11: Metal component labelling in planetary	19
Figure 12: Cycloidal section analysis highlighting bearing positions.....	20
Figure 13: Planetary section analysis highlighting bearing positions	21
Figure 14: Cycloidal top view highlighting eccentric offset between discs	22
Figure 15: Test rig isotropic front view with colour coded components	24
Figure 16: Test rig isotropic back view with colour coded components	25
Figure 17: Input shaft side of test rig, highlighting key components	26
Figure 18: Output shaft side of test rig, highlighting key components.....	27
Figure 19: Measured transmission efficiency at varying input speeds	34
Figure 20: Graph comparing measured transmission Efficiency at Varying Input Speeds	34
Figure 21: Measured backlash for planetary and cycloidal gearboxes	36
Figure 22: Graph showcasing torque-twist relationship in the cycloidal gearbox	37
Figure 23: Graph showcasing torque-twist relationship in the planetary gearbox	38
Figure 24: Final project Gantt chart.....	45

1. Introduction

Compact, high-torque, and high-precision transmission systems are fundamental to modern engineering applications, particularly in robotics, automated machinery, and aerospace actuation (American Gear Manufacturers Association, 2015; International Organization for Standardisation, 2018). In such systems, the performance of an actuator is frequently governed not by the electric motor, but by the transmission stage, which defines the effective output characteristics of the system (Norton, 2011). Gearbox characteristics directly influence positional accuracy, controllability, and mechanical robustness, with parameters such as backlash, torsional stiffness, and efficiency determining both steady-state accuracy and dynamic response (Harris & Kotzalas, 2006; Budynas & Nisbett, 2015).

Among compact, coaxial reduction mechanisms, planetary gearboxes and cycloidal drives represent two widely used solutions for achieving high reduction ratios within a limited volume. Planetary gear systems are widely adopted due to their high efficiency, load sharing capability, and favourable torque density (Budynas & Nisbett, 2015; American Gear Manufacturers Association, 2015). However, their performance is sensitive to manufacturing tolerances and assembly errors, which may lead to uneven load distribution between planet gears, increased backlash, and progressive wear (Kahraman, 1994; Parker & Lin, 2004). Cycloidal drives, by contrast, employ multi-tooth engagement and rolling contact geometry, providing high torsional stiffness, strong shock-load tolerance, and the potential for near-zero backlash. These advantages are offset by increased sliding contact, which can reduce mechanical efficiency and introduce speed-dependent losses.

Although the theoretical behaviour of both transmission types is well documented, much of the available literature relies on analytical models or simulations assuming idealised geometry, material properties, and assembly conditions. Consequently, the practical performance of physically manufactured gearboxes, particularly those produced with realistic tolerances and low-cost fabrication methods, is not fully represented. Direct experimental comparisons between planetary and cycloidal gearboxes designed to equivalent specifications and evaluated under identical test procedures are limited. This study therefore evaluates experimentally measurable transmission behaviour using actuator-relevant metrics, specifically backlash, torsional stiffness, mechanical efficiency, and efficiency variation with operating speed, providing a practical basis for comparison between gearbox topologies.

1.1. Aim and Objective

1.1.1. Aim

This dissertation aims to design, manufacture, and experimentally evaluate a planetary gearbox and a cycloidal gearbox developed to a common performance specification representative of a compact actuator system. A dedicated experimental test rig was developed to quantify transmission behaviour under controlled conditions. Due to practical constraints, testing was conducted at reduced torque and power levels, enabling repeatable measurements while maintaining valid comparative analysis.

1.1.2. Objective

The primary objective of the study is to generate directly comparable experimental measurements of transmission performance through the development of a controlled testing methodology. This includes the design and implementation of a purpose-built test rig capable of measuring key actuator-relevant characteristics as defined in 2.3.3:

- Backlash
- Torsional stiffness
- Mechanical efficiency
- Efficiency as a function of rotational speed

The test rig provides consistent boundary conditions for both gearbox types, allowing the influence of gearbox topology on performance to be isolated. In addition, both a planetary gearbox and a cycloidal gearbox are chosen and designed and manufactured to a shared specification so that measured differences arise from transmission principles rather than operating conditions. The reasoning for this selection is discussed in 1.2.

The study does not seek to validate absolute load capacity at the design limit; instead, it evaluates the behaviour of each gearbox as a precision actuator component. By combining gearbox development with experimentally obtained measurements, the work will bridge the gap between theoretical gearbox analysis and practical implementation, providing evidence to support informed transmission selection in precision mechanical and mechatronic systems.

1.2. Typical Applications

Compact transmission systems such as planetary and cycloidal gearboxes are widely used in robotics, automated machinery, and aerospace actuation, where high torque density, precision, and reliability are critical. In these applications, characteristics such as backlash, torsional stiffness, and

efficiency directly influence positioning accuracy, dynamic response, and energy consumption (Budynas & Nisbett, 2015).

Planetary and cycloidal gearboxes were selected for this study as they represent two fundamentally different yet directly comparable compact, coaxial reduction mechanisms capable of achieving similar reduction ratios within a constrained volume. Planetary systems are commonly used for their high efficiency and balanced load distribution, whereas cycloidal drives offer high stiffness, shock-load resistance, and near-zero backlash. Their contrasting operating principles make them well-suited for evaluating how transmission topology influences actuator performance in precision mechanical systems.

1.3. Project Timing Plan

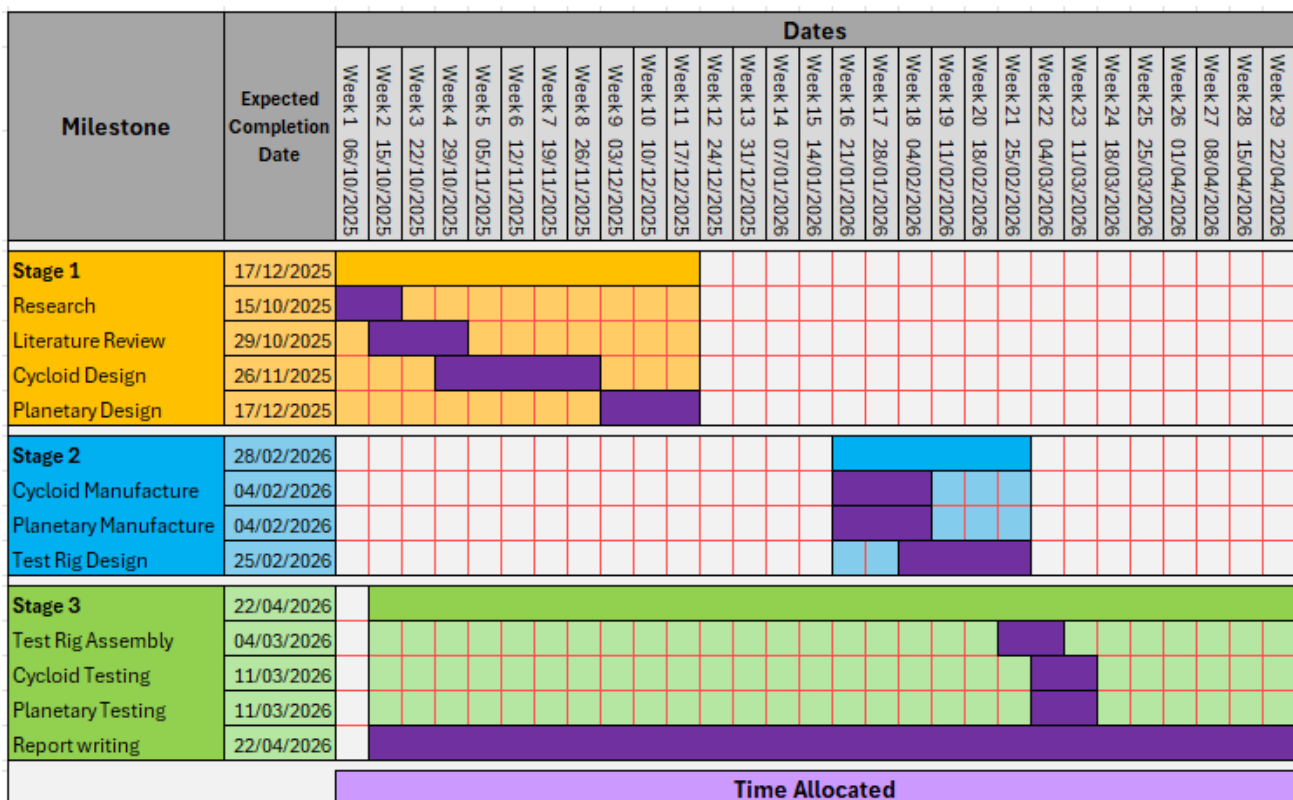


Figure 1: Project Gantt Chart

2. Literature Review

2.1. Chapter Overview/Introduction

This chapter reviews the operating principles, design considerations, and experimentally relevant performance characteristics of planetary and cycloidal gear reducers, establishing the research gap addressed by this study.

2.2. Broad context and importance

The evolution of advanced robotics, automation, and aerospace systems is critically dependent on the performance of precision actuators. The selection of a motor-gearbox unit is a systems-level decision where the transmission's mechanical characteristics, specifically torsional stiffness, operational efficiency, and backlash, directly constrain the overall system performance (Gilberti, et al., 2010). In robotic joints and precision positioning systems, low stiffness produces angular deflection under load, reducing positional accuracy, while backlash degrades repeatability and control stability.

These requirements are particularly important in industries such as industrial robotics, high-precision manufacturing, and aerospace actuation. In robotic manipulators and CNC systems, transmission characteristics directly influence positioning accuracy, surface quality, and repeatability, while in aerospace applications, high stiffness and reliability are essential under dynamic loading conditions (Spong, et al., 2006; Altintas, 2012; National Aeronautics and Space Administration, 2020).

As a result, high-ratio gear reducers such as planetary and cycloidal drives are widely used to achieve compact, high-torque actuation. However, their differing mechanical characteristics present a key design trade-off, making the selection of an appropriate transmission critical to system performance.

2.3. Thematic Review of key topics

2.3.1. Planetary Gear Systems

Planetary gear systems distribute transmitted torque among multiple planet gears, allowing smooth motion transmission and high mechanical efficiency. (Kahraman, 1999) demonstrated that small carrier manufacturing errors can significantly increase load sharing imbalance between planets, accelerating fatigue and reducing gearbox reliability. This sensitivity to manufacturing tolerances represents a key design limitation.

To mitigate this, designs often incorporate "float," allowing components like the sun gear or ring gear to self-align. Furthermore, the bearing life of the planet gears, which are subjected to complex orbital and rotational motions, is often the limiting factor for the gearbox's service life. Despite these limitations, planetary gearboxes remain widely used due to high efficiency and manufacturability.

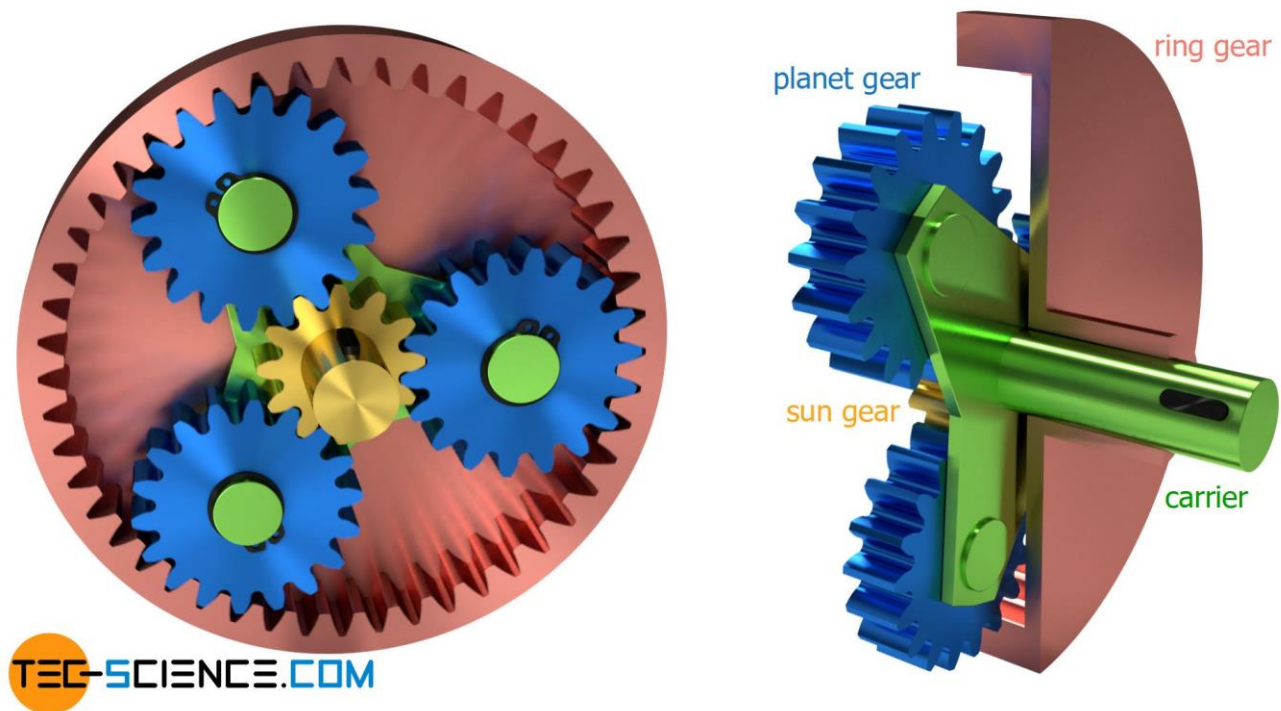


Figure 2: Planetary drive diagram - TEC-SCIENCE.com

2.3.2. Cycloidal Drive Systems

Cycloidal drives operate on a fundamentally different principle from involute gears. The kinematics are based on the rolling of a cycloidal disc, driven by an eccentric cam, against a ring of stationary pins. The cycloidal disc profile strongly influences transmission behaviour. (Blagojevic, et al., 2014) detailed the use of a "generated" or "conjugate" lobe profile, which is not a pure hypotrochoid but is modified to optimize the motion transmission. This optimization is critical for improving mechanical efficiency, which has historically been a weakness of cycloidal drives.

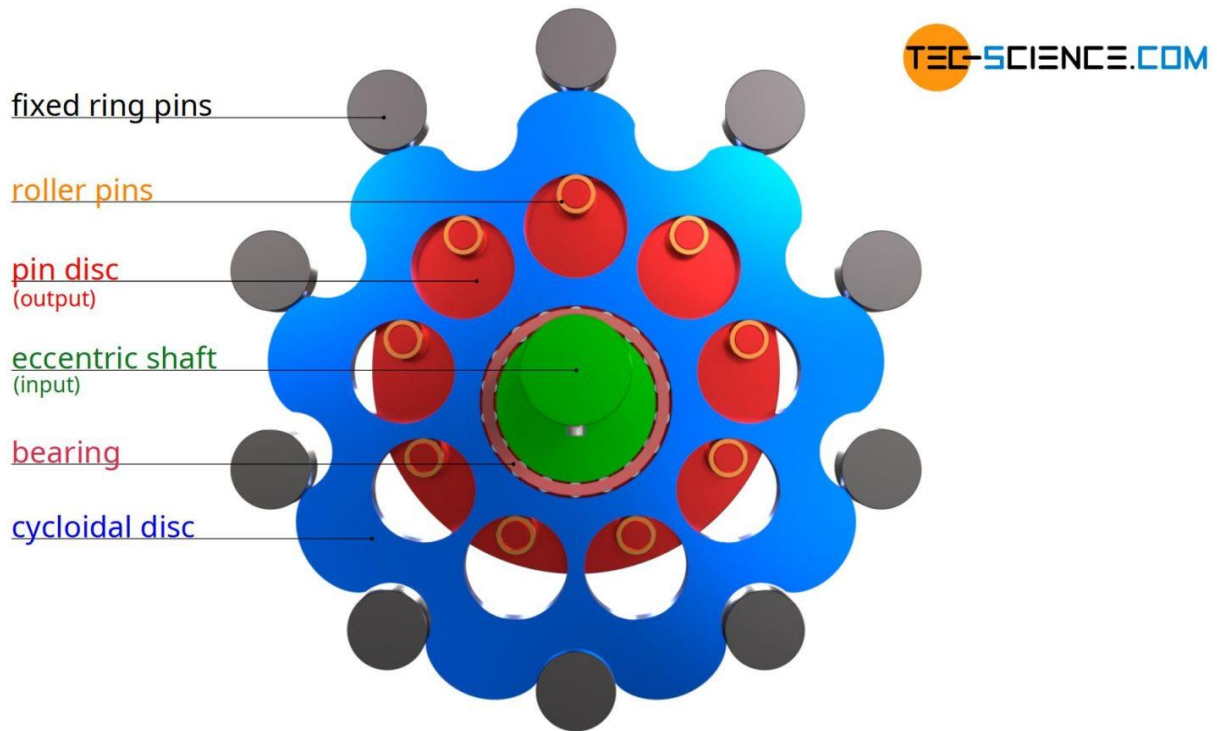


Figure 3: Cycloidal drive diagram - TEC-SCIENCE.com

A critical analysis by (Hsieh, 2015) provides deep insight into the efficiency characteristics, which have historically been a point of comparison with planetary systems. Hsieh's study demonstrates that the primary source of power loss in a cycloidal drive is sliding friction at the interface between the cycloidal disc and the pins. The paper establishes that the efficiency is highly dependent on the design of the disc profile and the number of teeth in contact, with higher reduction ratios generally leading to lower efficiency due to increased sliding action. This provides a quantitative foundation for the trade-off between the high reduction capability and the efficiency of these drives.

Despite these losses, multi-lobe contact results in exceptionally high torsional stiffness and shock load capacity. Furthermore, backlash can be eliminated through slight oversizing of the cycloidal disc and the use of a two-disc design with a phase shift, which compensates for the eccentric motion. The critical point of failure is typically the eccentric cam bearing, which must withstand high dynamic loads from the precession of the cycloidal disc. Consequently, cycloidal drives are widely used in high-precision robotic actuators where stiffness and minimal backlash are prioritised over peak efficiency.

2.3.3. Performance Evaluation

2.3.3.1. Efficiency Measurement

(Lin & Shih, 2013) designed a back-to-back test rig in which two identical gearboxes are mechanically coupled, with one acting as the driver and the other as the load. By measuring the input torque required to maintain a constant rotational speed, the combined losses of both

gearboxes can be determined, enabling the efficiency of an individual unit to be calculated. Their results demonstrated that this method provides a reliable means of measuring gearbox efficiency under steady-state conditions while minimising external load variability.

However, the back-to-back approach requires two identical gearboxes and a more complex mechanical arrangement, which may not be practical within the constraints of this study. As a result, this work adopts a single-gearbox testing methodology using a controlled load applied via a lever arm and load cell. While this approach may introduce greater sensitivity to measurement error, it allows efficiency to be evaluated across a range of operating speeds using a simpler and more adaptable experimental setup.

An alternative approach, more suitable for low-power laboratory test systems, is the determination of transmission efficiency through measurement of rotational speed using rotary encoders combined with a known applied load torque. With a known applied load torque, measurement of input and output angular velocity allows mechanical power to be determined using the relationship $P = T\omega$ (Norton, 2020). Where T is torque and ω is angular velocity (Norton, 2020). When combined with a known applied load torque, the ratio of output power to input power provides gearbox efficiency. Optical or magnetic encoders allow high-resolution angular position and speed measurement without introducing additional mechanical loading, making them widely used in servo actuator and robotics transmission characterisation (Bolton, 2015; Siciliano, et al., 2016). This method avoids the need for a second matched gearbox and instead isolates transmission losses by comparing input and output power across the same unit, making it particularly appropriate for prototype gearboxes and reduced-scale experimental rigs.

2.3.3.2. Backlash

Backlash is the relative angular displacement between input and output shafts that occurs when the direction of transmitted torque is reversed. It arises from clearance between mating gear tooth flanks as well as bearing play, manufacturing tolerances, and assembly misalignment. In precision actuators this lost motion directly affects positioning accuracy and repeatability. (Gilberti, et al., 2010) demonstrated that even small transmission clearances can produce measurable end-effector error in robotic joints, particularly at low load.

A standardized definition is provided in ISO 6336-1:2019, which describes backlash as the angular displacement between mating gears at zero load corresponding to tooth flank clearance. The recommended measurement method applies equal alternating torques to the output shaft and records the angular displacement required before torque transmission occurs in the opposite direction. This approach isolates mechanical play from elastic deformation and provides a consistent basis for comparing different transmission systems.

Dynamic behaviour, however, differs from static clearance alone. (Tsai, et al., 2015) measured transmission error using non-contact techniques and identified a “dead zone” where input rotation produces no proportional output torque, showing that backlash depends on load level and engagement conditions. The effect also varies between gearbox architectures: planetary gear systems inherently retain backlash due to tooth clearance and load-sharing tolerances, whereas cycloidal drives can approach near-zero backlash through simultaneous multi-lobe contact and phase-shifted disc arrangements.

2.3.3.3. Torsional Stiffness

Torsional stiffness is the resistance of a transmission to angular deflection under applied torque and is typically expressed as the ratio between transmitted torque and the relative angular displacement of the input and output shafts. The compliance arises from elastic deformation of gear teeth, shafts, and bearings, meaning that even without backlash, load changes can produce measurable positioning error (Norton, 2020).

In servo-driven systems, transmission stiffness influences motion accuracy and control stability. (Siciliano, et al., 2016) show that drivetrain compliance reduces achievable control bandwidth because the motor position and load position differ under varying torque, introducing lag and potential oscillatory behaviour in closed-loop control. As a result, backlash and stiffness describe different aspects of precision: backlash governs lost motion during load reversal, whereas torsional stiffness governs load-dependent angular deflection.

Gearbox architecture strongly affects stiffness behaviour. Planetary gear systems rely on localized tooth contact and load sharing between planets, so elastic deformation accumulates across multiple meshing interfaces. In contrast, cycloidal drives employ simultaneous multi-lobe engagement, distributing load over several contact points and increasing resistance to angular deformation (Blagojevic, et al., 2014). This structural difference suggests stiffness should differ between the two transmission types and therefore warrants experimental evaluation.

2.4. Critical Synthesis and Identification of Research Gap

Existing studies primarily rely on analytical and numerical modelling, including load sharing models for planetary systems (Kahraman, 1999) and dynamic analyses of cycloidal mechanisms (Blagojevic, et al., 2014). These approaches necessarily assume ideal manufacturing tolerances, material properties, and assembly alignment, and therefore cannot fully represent the behaviour of gearboxes subject to real manufacturing tolerances and assembly errors.

Consequently, there is limited experimental work directly comparing planetary and cycloidal gearboxes designed to equivalent specifications and evaluated under identical testing conditions. This dissertation addresses this gap through the design, manufacture, and experimental testing of both gearbox types on a common test platform, producing directly comparable measurements of backlash, torsional stiffness, and transmission efficiency.

2.5. Summary of the Chapter

This chapter reviewed the operating principles and performance characteristics of planetary and cycloidal gear reducers and examined established approaches for evaluating transmission behaviour. A lack of controlled experimental comparison between equivalent prototypes was identified. The following chapter presents the methodology developed to design the gearboxes and experimentally measure backlash, torsional stiffness, and transmission efficiency.

3. Methodology

3.1. Experimental Approach

This study adopts an experimental comparative methodology to evaluate the performance of planetary and cycloidal gear transmissions. Rather than validating theoretical load capacity, the objective is to measure functional transmission behaviour under controlled laboratory conditions. Both gearbox types were designed to a common reduction ratio and tested on the same apparatus to ensure that any differences in measured performance arise from transmission architecture rather than operating conditions.

Testing was conducted within a reduced torque range relative to the intended design capacity. This allowed safe operation of prototype components while preserving the validity of relative performance comparison between gearbox types. The experiment therefore evaluates transmission characteristics representative of actuator operation rather than maximum structural capability.

3.2. Gearbox Design Specification

The planetary and cycloidal gearboxes were designed as alternative transmission solutions for the same intended application: a compact robotic actuator. To ensure that observed performance differences are attributable primarily to gearbox topology rather than design quality, both systems were developed to a shared specification, including a target reduction ratio of approximately 20:1, a coaxial input–output configuration, and a comparable external envelope. Both gearboxes utilise the same motor, mounting interface, and test conditions, ensuring consistent boundary conditions across all experiments.

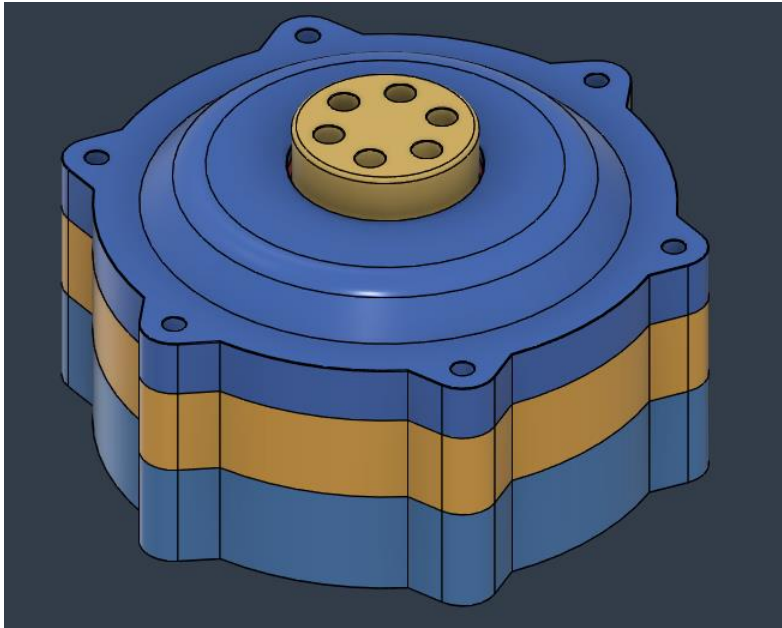


Figure 4: Final iteration of planetary gearbox with colour coded components

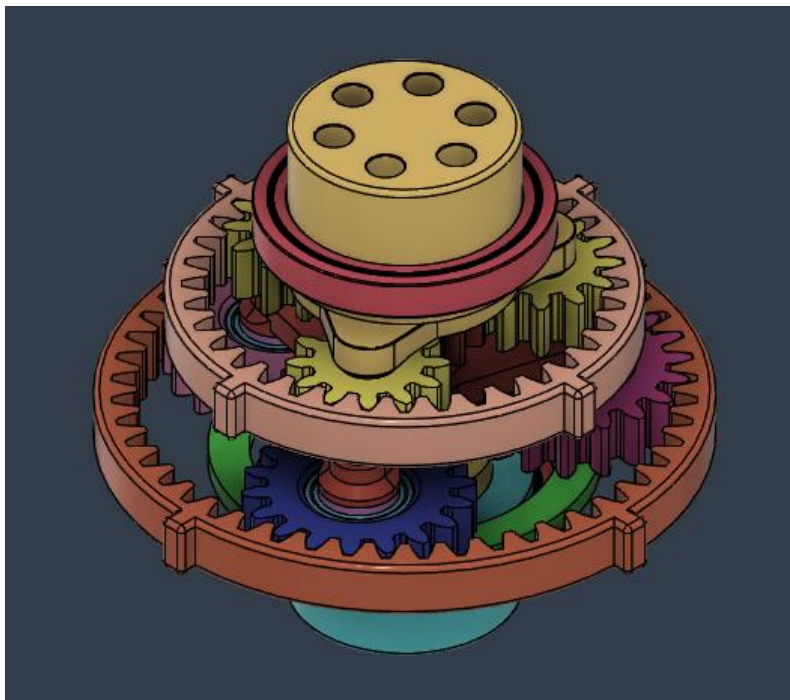


Figure 5: Final iteration of internal planetary gearbox mechanism with colour coded components

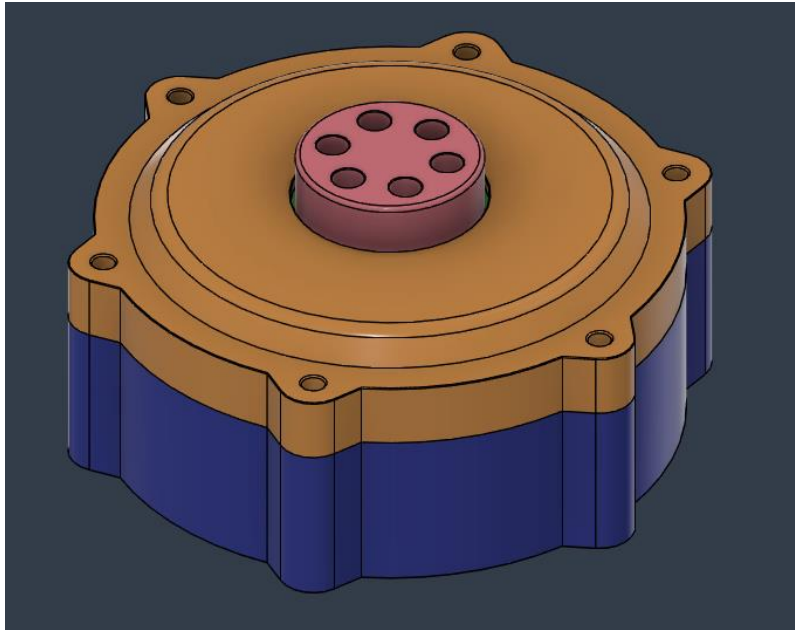


Figure 6: Final iteration of cycloidal gearbox with colour coded components

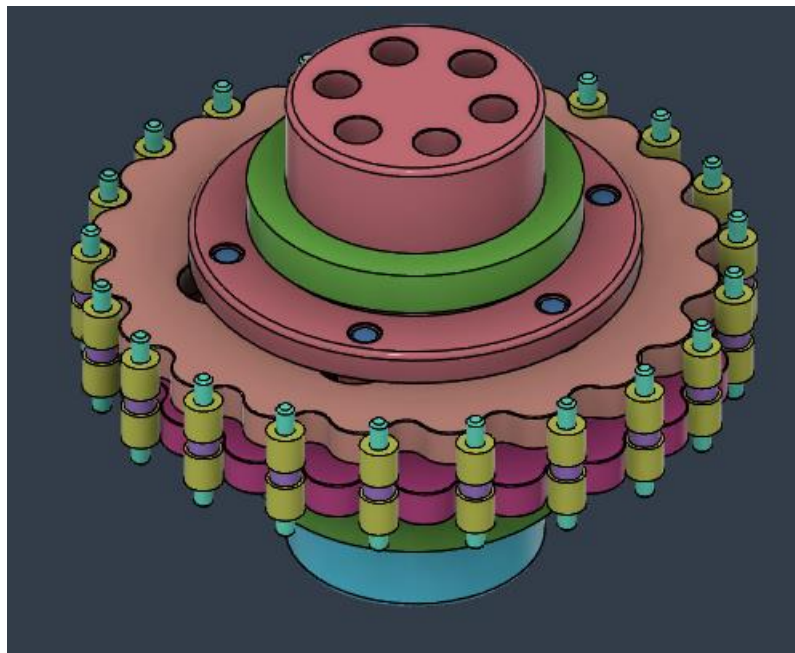


Figure 7: Final iteration of internal cycloidal gearbox mechanism with colour coded components

The initial design intent was to manufacture both gearboxes from metallic materials; however, cost and manufacturing constraints required a transition to polymer-based fabrication using resin and FDM (PLA) processes. Due to the significantly lower strength and stiffness of these materials, the designs were scaled and reinforced to maintain structural integrity. This included increasing section thicknesses, enlarging load-bearing features, and reducing slender unsupported geometries that would be acceptable in metal but prone to failure in polymer. As a result, the final designs represent materially adapted versions of the same functional specification.

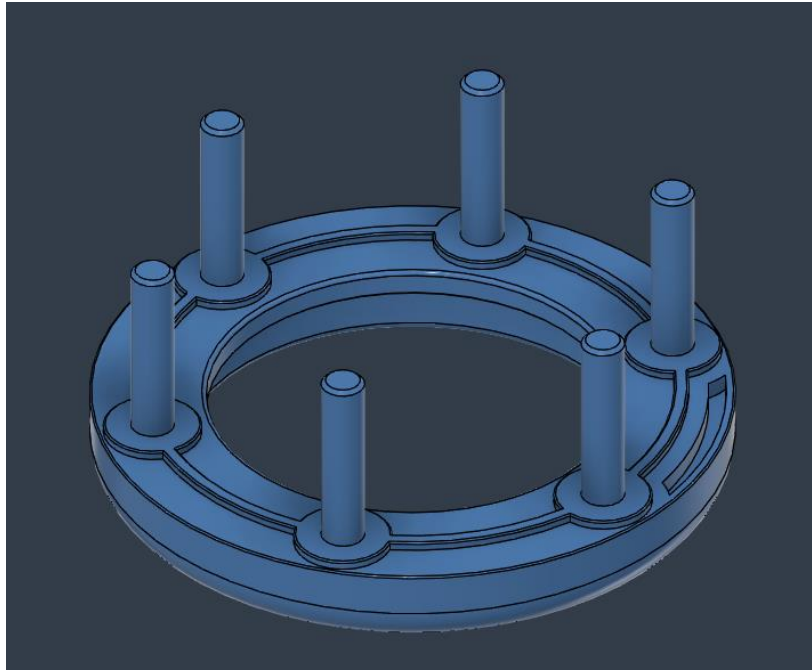


Figure 8: Cycloidal output pin carrier reinforcement

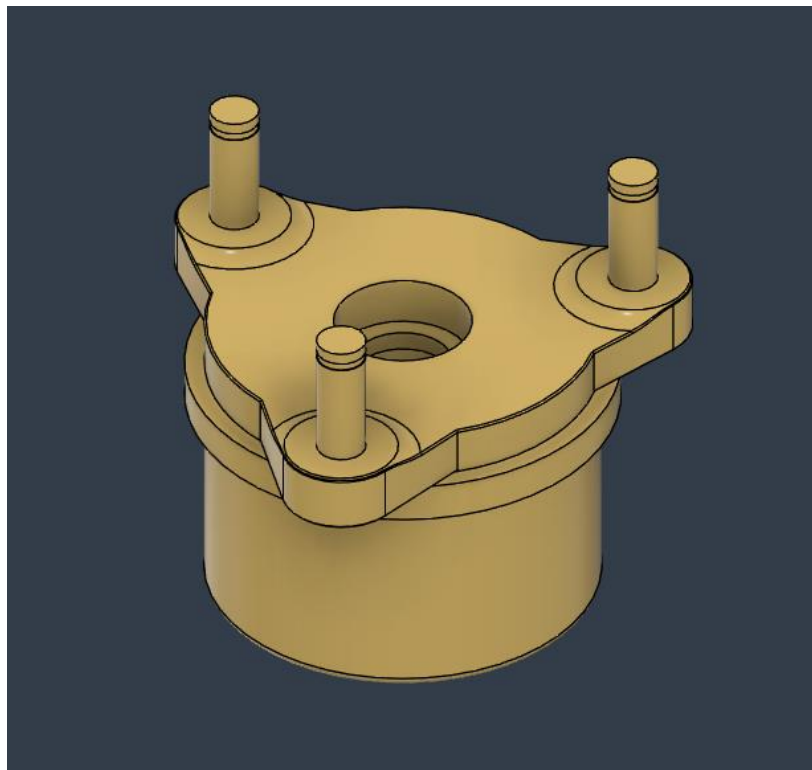


Figure 9: Planetary output shaft and second phase planet carrier reinforcement

To mitigate the limitations of polymer components, critical load bearing and wear-prone features were selectively manufactured using metal components. In the cycloidal gearbox, self-lubricating bronze bushings were implemented at the disc and output pin interfaces to reduce friction and wear under repeated rolling contact. These bushings were mounted on metal pins, which were introduced

due to the impracticality of manufacturing sufficiently strong thin pin features in polymer. The disc pins were identified as critical stress regions subjected to bending and contact forces, and the use of metal pins significantly improved durability and load capacity.

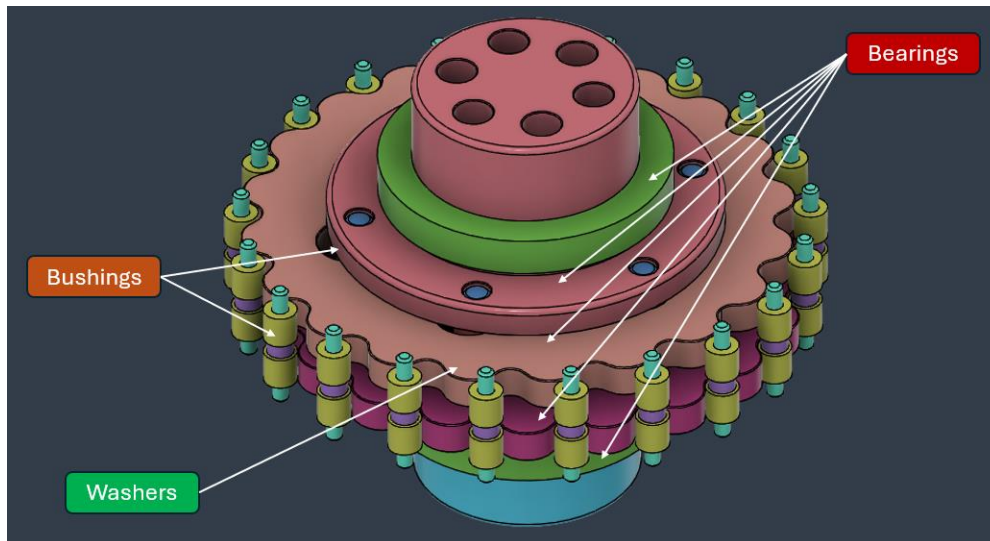


Figure 10: Metal component labelling in cycloidal

In contrast, the planetary gearbox offered fewer opportunities for hybridisation due to the geometric complexity of gear teeth and carrier features, which would be prohibitively expensive to manufacture in metal at this scale. As a result, the majority of planetary components were retained as polymer, with design adaptations focused instead on increasing support and reducing stress concentrations in critical regions such as the first- and second-stage planet pins.

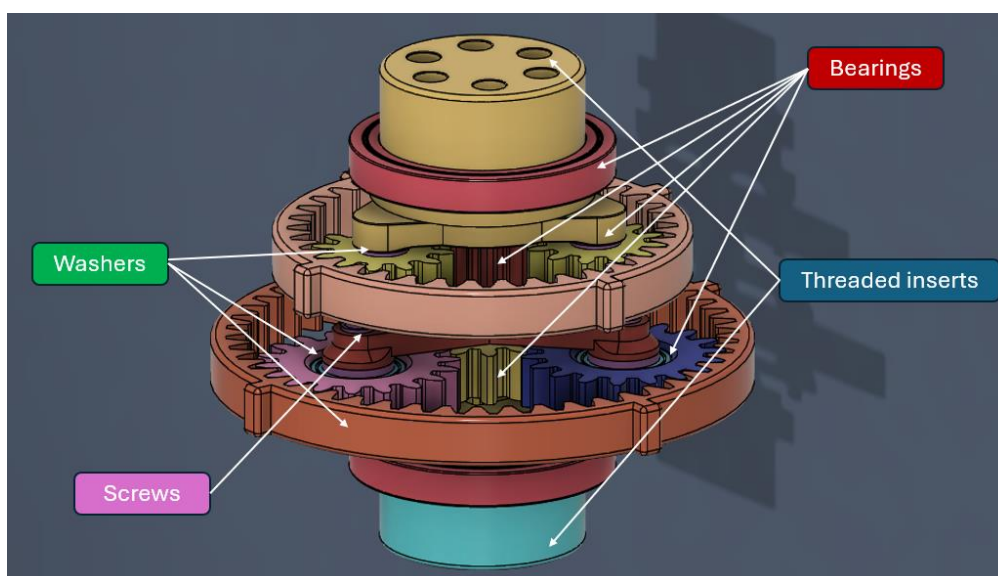


Figure 11: Metal component labelling in planetary

A key design requirement for both systems was that all rotating components were fully supported by rolling element bearings. No moving component was left unsupported, and all bearings used were of a consistent type, specifically rubber-sealed deep groove ball bearings, selected to provide low friction, contamination resistance, and ease of integration. Bearings were used to support both input and output shafts, as well as intermediate rotating elements where required, ensuring proper alignment, reducing frictional losses due to misalignment, and increasing overall system stiffness. The housing was therefore designed as a structural component, providing accurate bearing alignment and support at multiple locations along each shaft.

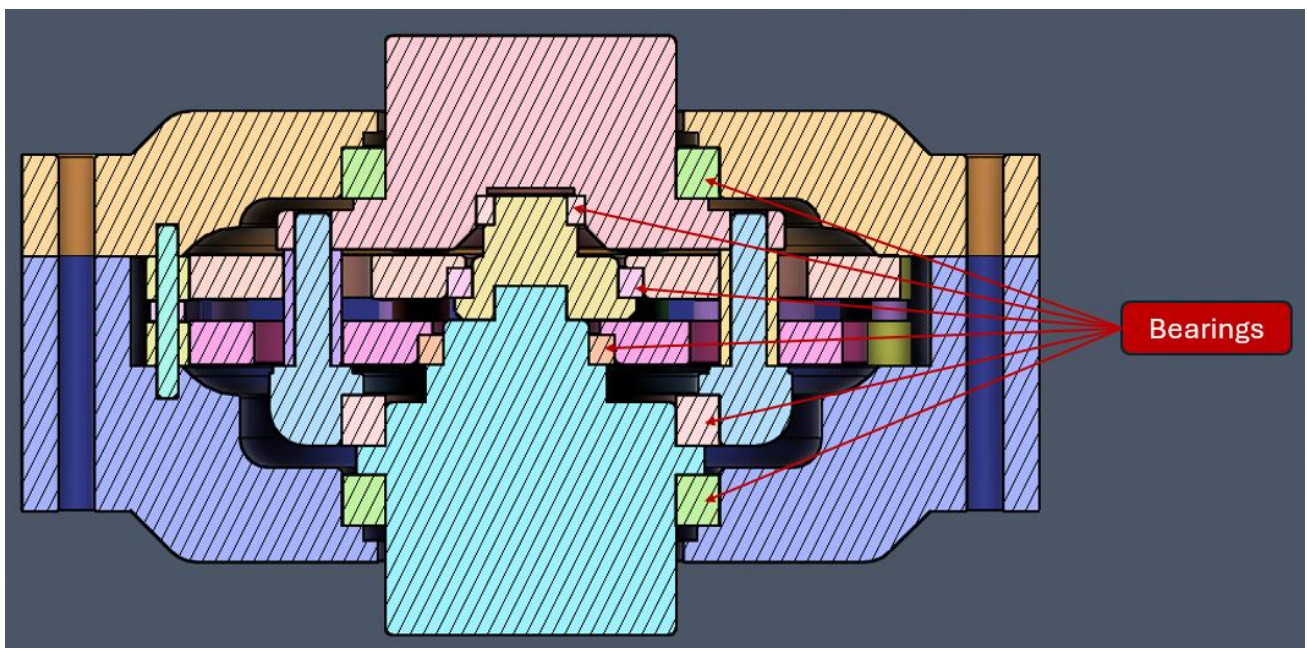


Figure 12: Cycloidal section analysis highlighting bearing positions

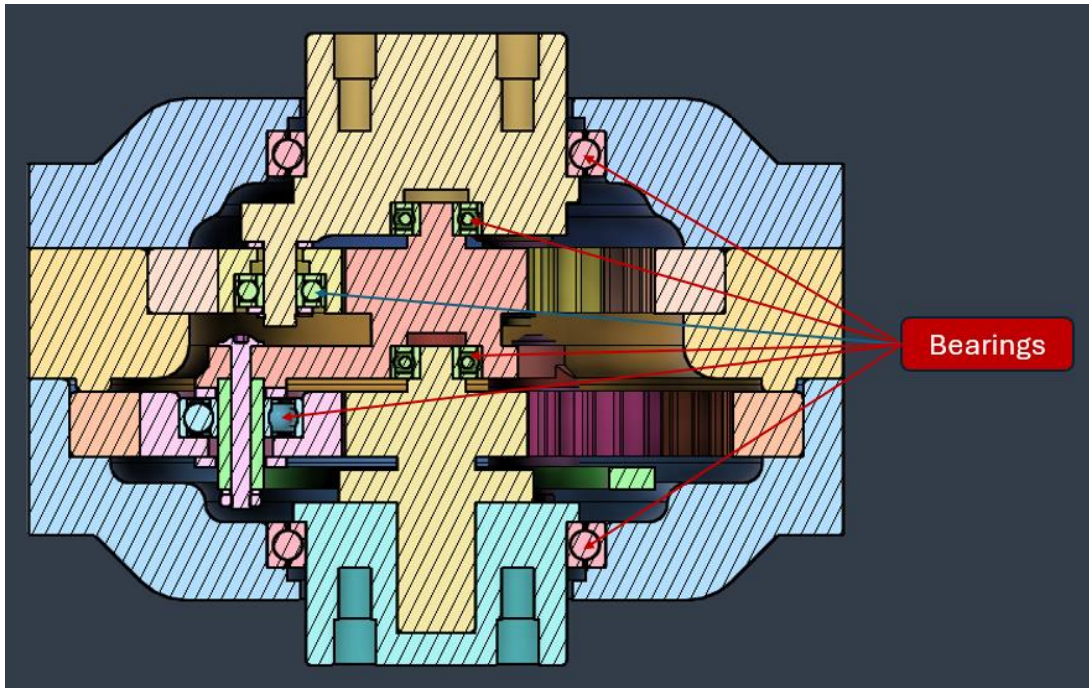


Figure 13: Planetary section analysis highlighting bearing positions

The distribution of reduction across each gearbox was also influenced by mechanical feasibility. While an initial objective was to maintain a consistent number of reduction stages between both designs, this was not practical for the planetary gearbox. Achieving a ~20:1 ratio in a single planetary stage would require extreme gear ratios and impractically large geometry. Therefore, a two-stage planetary configuration was adopted, allowing the required reduction to be achieved within the desired size constraints while maintaining realistic gear proportions as seen in Figure 7. The cycloidal gearbox, by contrast, achieves the required reduction within a single stage due to its inherent operating principle.

The cycloidal gearbox utilises two discs offset by 180° to balance dynamic forces generated by the eccentric input motion. This configuration reduces vibration, improves load distribution, and enhances overall operational smoothness compared to a single-disc arrangement. The cycloidal disc profile itself was generated using MATLAB to parametrically define the conjugate cycloidal curve. The geometry is derived from the kinematics of a rolling circle within a fixed pin ring, producing a hypocycloidal-type profile governed by the pin circle radius, eccentricity, and the difference in tooth count between the disc and ring. By expressing the profile as a function of angular position, coordinate points were generated and iteratively refined to control clearances, avoid interference, and ensure manufacturability at prototype scale before being exported for CAD modelling.

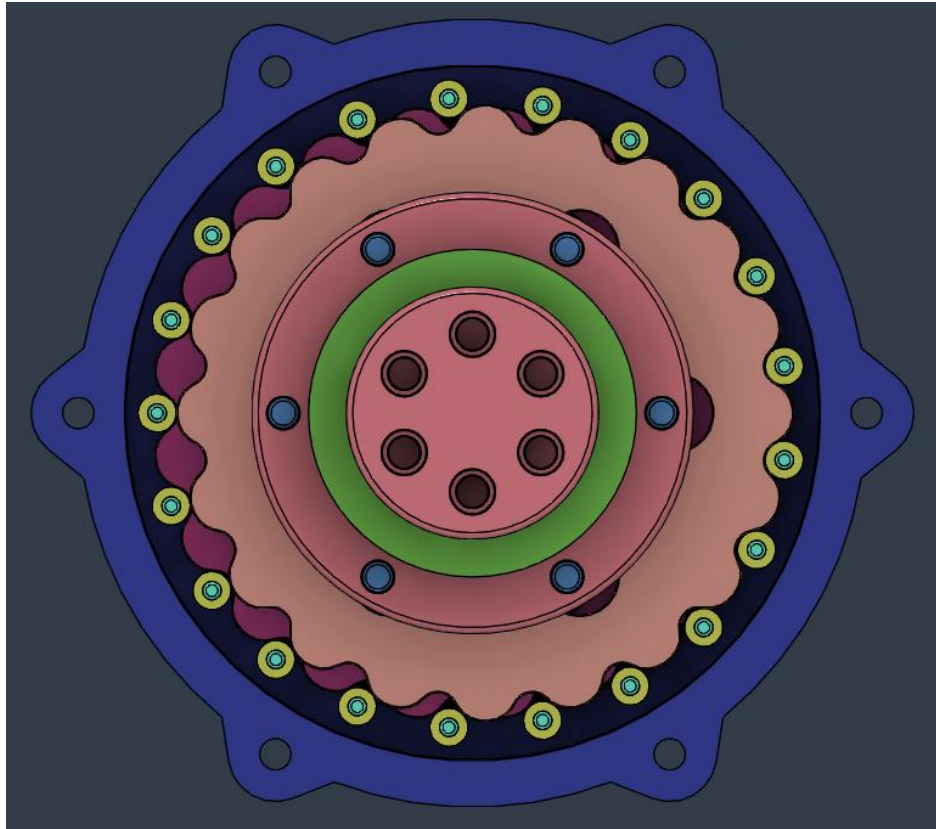


Figure 14: Cycloidal top view highlighting eccentric offset between discs

Manufacturing limitations imposed tolerance constraints of approximately ± 0.05 – 0.1 mm for resin components and ± 0.1 – 0.3 mm for FDM parts. Clearances between mating components were therefore iteratively adjusted to balance backlash reduction with the need to prevent interference and excessive friction. Bearing seat geometry was also refined to ensure sufficient wall thickness to withstand press-fitting forces during assembly, particularly for FDM components where anisotropic behaviour increases the risk of cracking.

Cycloidal Gearbox Exploded and Labelled Image

Planetary Gearbox Exploded and Labelled Image

3.3. Gearbox Manufacture

The gearboxes were manufactured as functional prototypes based on the designs outlined in 3.2. A hybrid manufacturing approach was adopted, combining additive manufacturing with commercially sourced components to achieve the required geometry, accuracy, and structural performance within cost and accessibility constraints.

Structural components, including the gearbox housings and mounting features, were produced using fused filament fabrication (PLA) on Bambu Lab printers. A 0.2 mm layer height “strength” profile was used, incorporating seven perimeter walls and approximately 15% infill. These components were designed to provide alignment, bearing support, and structural integrity, rather than transmit primary loads. Print orientation was selected to improve strength in critical regions such as bearing seats and mounting interfaces.

Motion-transmitting components were manufactured using resin additive manufacturing, as described in 3.2. Resin was selected due to its improved dimensional accuracy and surface finish compared to PLA, which are critical for maintaining consistent meshing conditions and minimising backlash. Components such as gears and cycloidal discs were printed as solid parts and post-cured prior to assembly to maximise material strength.

During assembly, resin components were found to be sensitive to stress concentrations, particularly during press-fitting operations. In early iterations, several parts fractured when installing bearings, requiring refinement of bearing seat geometry and more controlled assembly procedures. These modifications ensured sufficient strength while maintaining the required interference for secure fits.

Commercially sourced metal components were integrated where required, as outlined in 3.2, to improve wear resistance and durability in critical regions. Standard rubber-sealed deep groove ball bearings were used throughout both gearboxes to support all rotating components, ensuring consistent alignment and minimising radial and axial play. Bearings were press-fitted into the housing and used to support input shafts, output shafts, and intermediate rotating elements.

Coaxial alignment was achieved through integrated orientation features within the design, including slot-based geometries that constrained component positioning during assembly. Additionally, spacers were incorporated between the cycloidal discs in the form of washers to maintain consistent axial separation and ensure proper alignment. This prevented contact between adjacent discs and helped maintain parallelism during operation.

Assembly was carried out using primarily mechanical fits, including press-fitted bearings, pins, and dowel elements. Threaded inserts were bonded into holes at the ends of the resin input and output shafts to provide durable fastening points and prevent thread degradation in the printed material. Lubrication was applied to all relevant contact surfaces prior to final assembly, particularly within the cycloidal gearbox where sliding contact is more significant.

Following assembly, each gearbox was manually tested to verify smooth operation, correct alignment, and absence of binding. Minor adjustments were made where required to account for

manufacturing tolerances. This process ensured that both prototypes operated reliably within the constraints of the selected materials and manufacturing methods, enabling consistent experimental evaluation.

3.4. Test Rig Design

The experimental test rig was designed to enable controlled measurement of gearbox performance characteristics, including efficiency, backlash, and torsional stiffness, under consistent operating conditions. The rig configuration was informed by established gearbox testing approaches identified in the literature review, which emphasise the importance of controlled loading, accurate torque measurement, and minimisation of external influences on measured performance. This approach was selected in line with the limitations of back-to-back testing methods identified in the literature (Lin & Shih, 2013), allowing a simplified and adaptable configuration suitable for prototype-scale evaluation.

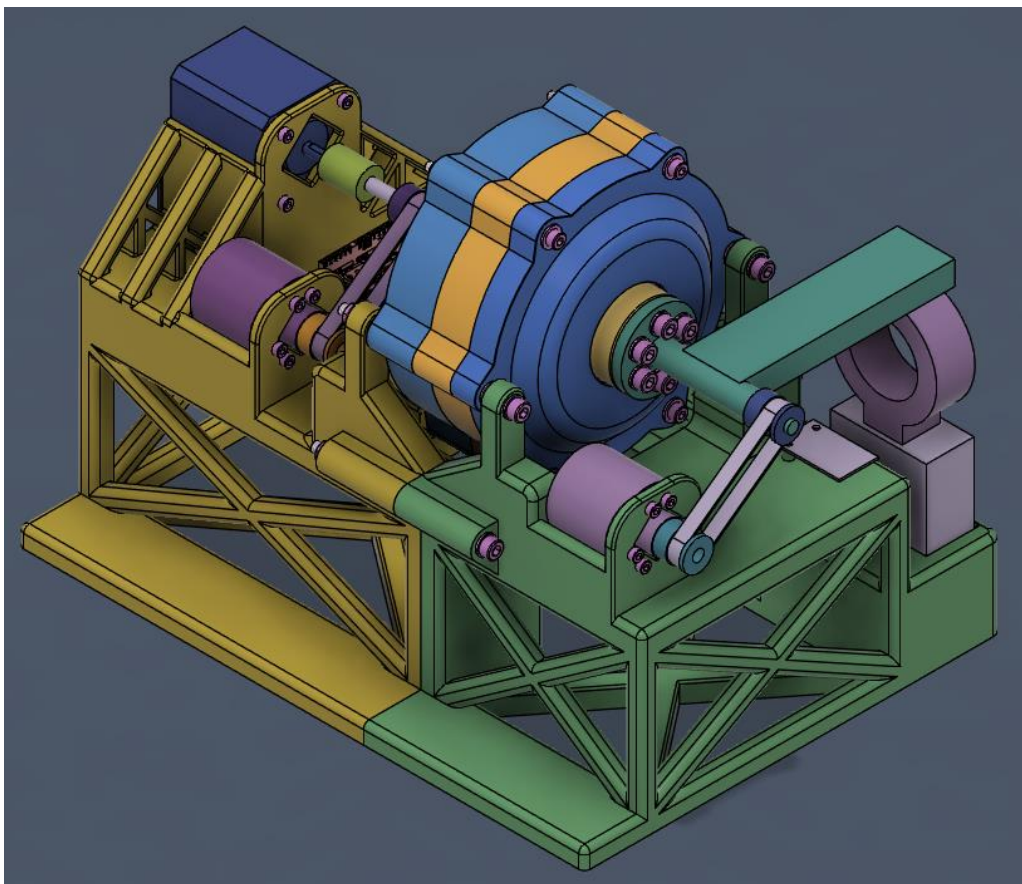


Figure 15: Test rig isotropic front view with colour coded components

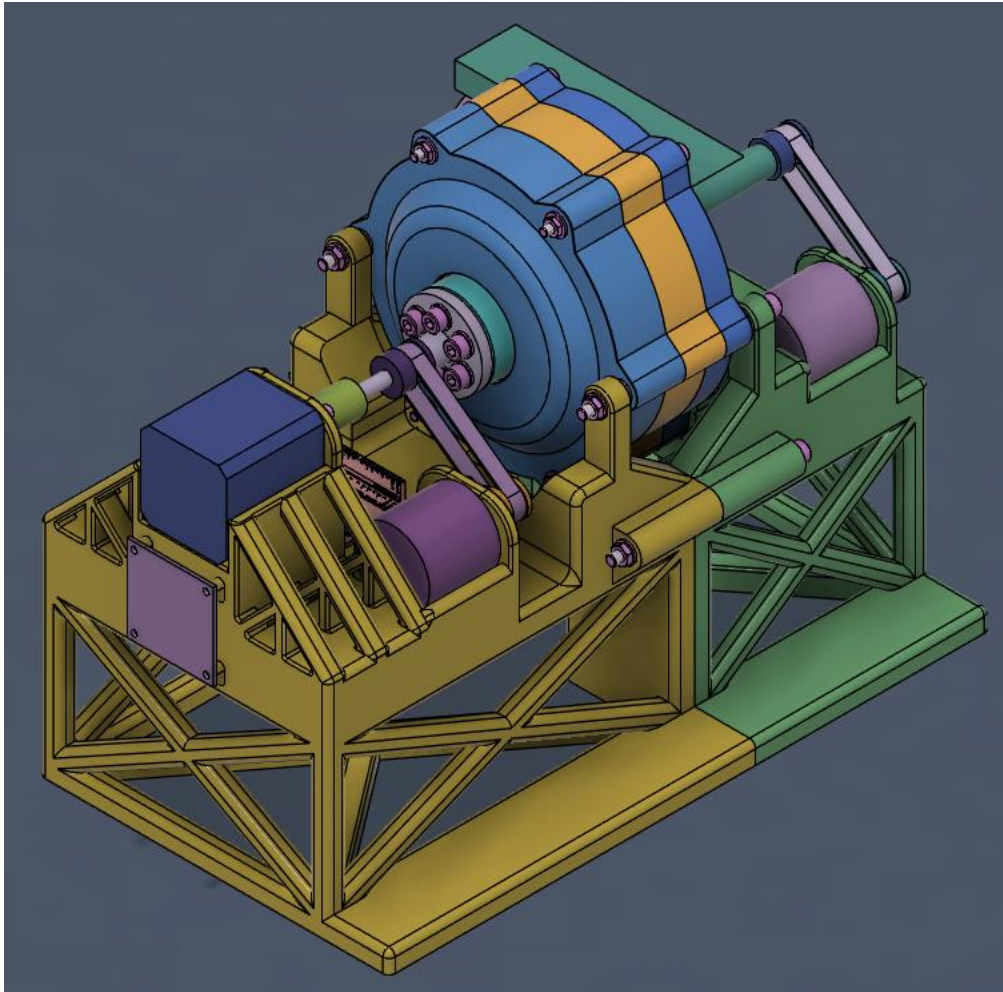


Figure 16: Test rig isotropic back view with colour coded components

The rig consists of a centrally mounted gearbox, with the input shaft connected to a stepper motor via a flexible shaft coupling. This coupling was selected to accommodate minor misalignment between the motor and gearbox shafts, reducing the introduction of parasitic loads that could influence measurement accuracy. Rotational speed at the input is measured using an encoder driven via a belt connected to the input shaft, allowing non-intrusive measurement without affecting shaft loading.

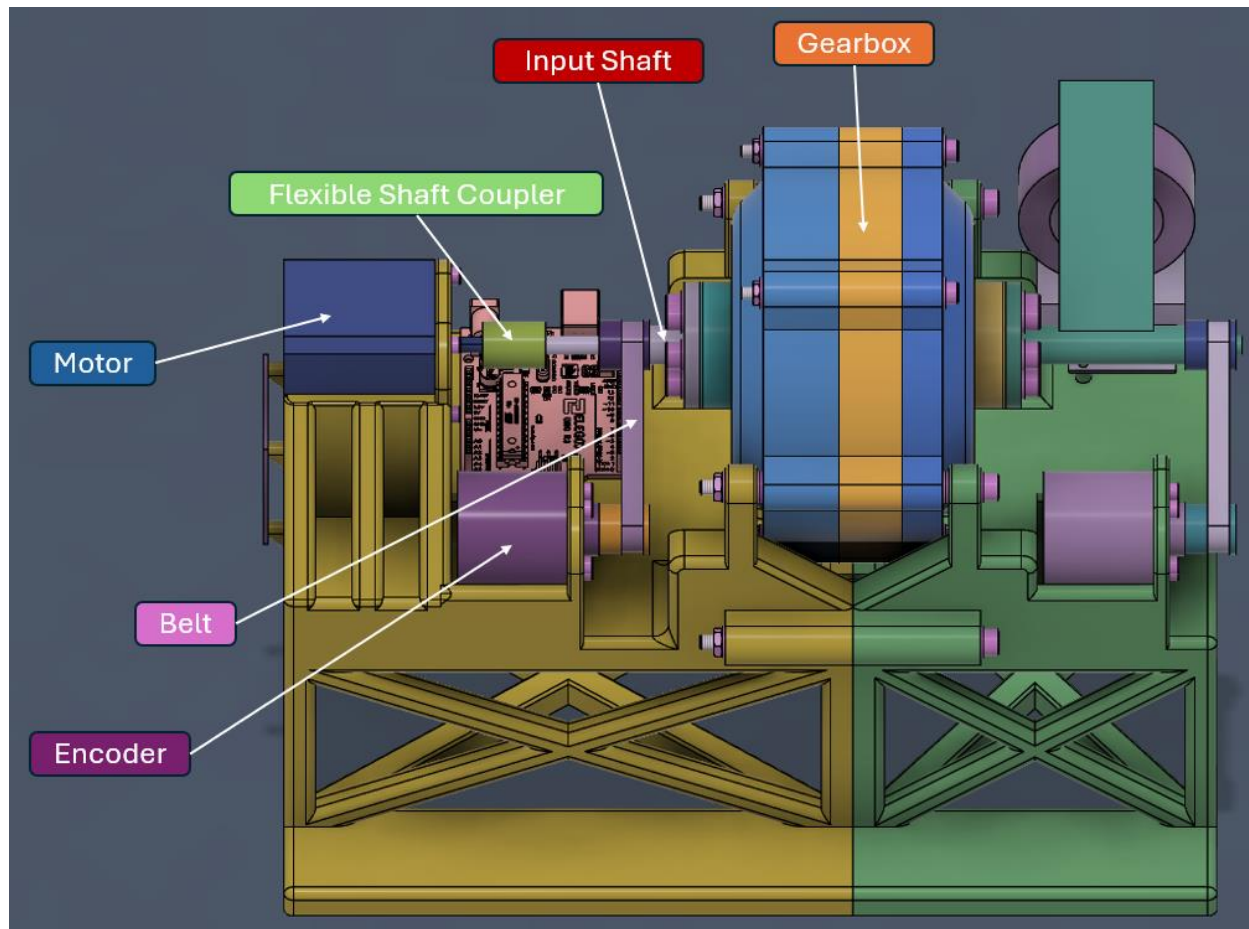


Figure 17: Input shaft side of test rig, highlighting key components

At the output, torque is applied through a rigid lever arm connected to the gearbox output shaft. The lever arm is aligned with a load cell positioned such that the force is applied perpendicular to the arm at a fixed distance of 100 mm from the gearbox centre. This arrangement enables torque to be determined directly using the relationship $T = F \times r$, while ensuring consistent and repeatable loading conditions. Output rotational speed is measured using a second encoder, also connected via a belt-driven system to avoid interference with the load application.

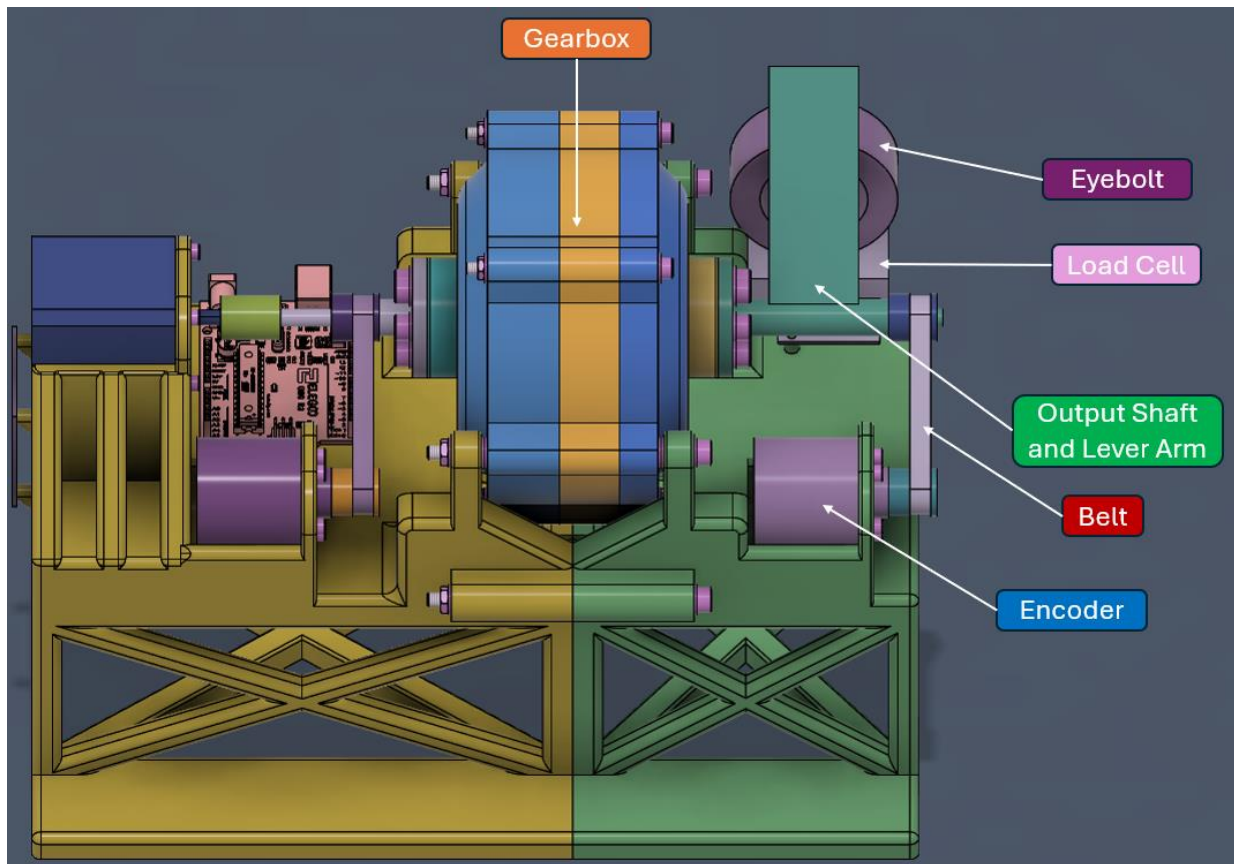


Figure 18: Output shaft side of test rig, highlighting key components

To ensure measurement accuracy, particular attention was given to alignment within the rig. Dedicated geometric features were incorporated to maintain coaxial alignment between the motor, gearbox, and output shaft, while also ensuring correct positioning of the encoder belt drives. The load application point was fixed at an exact radius of 100 mm through the design of the lever arm, ensuring consistent torque calculations across all tests. These measures reduce systematic errors arising from misalignment and inconsistent geometry, which are identified in the literature as significant sources of error in experimental gearbox testing.

The use of belt-driven encoders allows measurement of rotational speed without introducing additional stiffness or resistance into the system, preserving the natural behaviour of the gearbox under test. The encoders used have a resolution of 600 pulses per revolution, providing sufficient accuracy to capture variations in rotational speed and enable reliable calculation of efficiency, backlash, and torsional response. The belt-driven arrangement also isolates the measurement system from the primary load path, reducing the influence of measurement hardware on the observed gearbox performance.

Both planetary and cycloidal gearboxes were tested using the same rig configuration, with identical mounting interfaces, couplings, and measurement systems. This ensures that all boundary

conditions remain constant, allowing direct comparison between gearbox types and isolating the effect of transmission topology on performance.

As discussed in the literature, accurate efficiency measurement requires knowledge of both input and output torque. As direct motor torque data was not available, a preliminary calibration test was conducted using a simplified version of the rig. In this configuration, the motor was directly coupled to a lever arm with the same 100 mm radius used in the main test rig. By operating the motor at a range of speeds and measuring the corresponding force at the load cell, a torque–speed relationship was established. This calibration enabled estimation of motor input torque during subsequent gearbox testing, allowing efficiency to be calculated.

All sensor data, including encoder signals and load cell measurements, were processed using a program implemented on an Arduino platform. This enabled consistent data acquisition and real-time calculation of key parameters while maintaining repeatability across all tests. For Arduino programs used, please view Appendix 1.

The operating range of the test rig was limited to ensure safe operation of both the gearbox components and measurement system. Due to the use of polymer-based components, excessive loading could result in deformation or failure, particularly in critical regions such as pins and bearing interfaces. As a result, testing was conducted within a reduced torque range that prevented structural damage while still allowing meaningful measurement of transmission behaviour.

3.5. Measurement Procedures

This section outlines the experimental methods used to evaluate gearbox performance, specifically efficiency, backlash, and torsional stiffness. The procedures were developed based on established methods identified in Chapter 2, with adaptations made to suit the constraints of a reduced-scale experimental test rig. All measurements were conducted under identical boundary conditions for both gearbox types, ensuring that observed differences are attributable to transmission characteristics rather than experimental variation.

3.5.1. Efficiency Measurement

Transmission efficiency was evaluated using an input–output power comparison approach, consistent with experimental methods described in the literature for low-power systems. While back-to-back testing rigs provide high accuracy by minimising external losses (Lin & Shih, 2013), such methods require two identical gearboxes and complex mechanical arrangements, making them impractical for this study.

Instead, efficiency was determined by measuring torque and rotational speed at both the input and output of a single gearbox. Mechanical power was calculated using the relationship:

$$P = T\omega$$

where T is torque and ω is angular velocity (Norton, 2011). Rotary encoders were used to measure angular velocity, in line with their widespread use in actuator characterisation due to their high resolution and minimal influence on system dynamics (Bolton, 2015; Siciliano, et al., 2016).

A known load torque was applied at the gearbox output using a fixed-radius lever arm and load cell. As direct motor torque data was not available, a preliminary calibration test was conducted in which the motor was directly coupled to an identical lever arm configuration. By measuring force at a fixed radius of 100 mm across a range of rotational speeds, a torque–speed relationship for the motor was established. This enabled estimation of input torque during gearbox testing, allowing efficiency to be calculated as the ratio of output power to input power.

While this method introduces potential uncertainty in input torque estimation, it remains suitable for comparative analysis, as both gearbox types were tested under identical conditions.

3.5.2. Backlash Measurement

Backlash was evaluated as the angular displacement between input and output shafts during reversal of torque direction, consistent with the standard definition (International Organization for Standardisation, 2018). This method isolates mechanical clearance from elastic deformation by measuring the “dead zone” before torque transmission occurs.

The procedure involved applying small alternating rotations to the input shaft and measuring the corresponding output shaft displacement using rotary encoders. The difference in angular position between the input and output during reversal represents the backlash within the system. This approach is supported by findings from Gilberti et al. (2010), who demonstrated that even small transmission clearances can significantly affect positioning accuracy in robotic systems.

This method also reflects observations by Tsai et al. (2015), who identified that backlash manifests as a non-linear “dead zone” in transmission response, reinforcing the validity of encoder-based measurement techniques.

3.5.3. Torsional Stiffness

Torsional stiffness was evaluated by measuring the angular deflection between the input and output shafts under an applied torque. This follows the definition of stiffness as the ratio of torque to angular displacement (Norton, 2011).

A controlled torque was applied using the lever arm and load cell, while angular displacement was measured using the input and output encoders. By incrementally increasing the applied load and recording the corresponding angular deflection, a torque–displacement relationship was obtained. The gradient of this relationship represents the torsional stiffness of the gearbox.

This method aligns with established approaches in servo system analysis, where drivetrain compliance is evaluated based on the difference between motor and load position under varying torque (Siciliano, et al., 2016).

3.5.4. Efficiency as a Function of Speed

Efficiency was evaluated across a range of operating speeds to capture the influence of frictional losses and dynamic effects. This is consistent with findings from studies such as Hsieh (2015), which demonstrate that gearbox efficiency, particularly in cycloidal systems, is strongly dependent on operating speed due to sliding contact behaviour.

The motor was operated at discrete speed intervals (30, 60, 90, 120, and 150 rpm), and efficiency was calculated at each point using the method described in 3.5.1. This range was selected to reflect typical operating conditions for small actuator systems while remaining within the safe operating limits of the experimental setup.

By maintaining consistent loading and measurement conditions across all tests, the resulting efficiency curves provide a direct comparison of performance between the planetary and cycloidal gearboxes.

3.6. Experimental Control and Repeatability

Reliable comparison of gearbox performance requires that measured differences arise from transmission behaviour rather than variation in the experimental setup. As identified in 2.3.3, gearbox testing is highly sensitive to external influences such as misalignment, inconsistent loading, and measurement interference. The test rig and procedures were therefore designed to minimise these effects and maintain consistent operating conditions across all tests.

3.6.1. Control of Variables

Control of variables was achieved by standardising all boundary conditions between tests. Both gearboxes were tested using the same motor, couplings, encoder arrangement, load application method, and data acquisition system, ensuring that no changes to the surrounding system influenced the results. This approach directly addresses the limitation identified in the literature review, where comparative studies are often affected by differing test conditions.

Geometric control was a key strategy in maintaining measurement consistency. The load application point was fixed at a radius of 100 mm through the physical design of the lever arm, ensuring that torque calculations remained consistent. Alignment features were incorporated into the rig to constrain the position of the gearbox, encoder drives, and load cell interface, ensuring repeatable positioning and reducing errors caused by misalignment. This is particularly important given that even small deviations in geometry can introduce significant variation in measured torque and angular displacement.

The use of belt-driven encoders and a flexible shaft coupling further reduced the influence of the measurement system on gearbox behaviour. As discussed in 2.3.3, measurement techniques should minimise interference with the system being evaluated. The selected configuration allows rotational speed and displacement to be measured without introducing additional stiffness or loading into the drivetrain.

3.6.2. Repeatability of Measurements

Repeatability was achieved through a combination of fixed experimental setup and consistent testing procedures. The rig configuration was not altered between tests, and both gearboxes were installed using identical interfaces, allowing them to be exchanged without affecting alignment or measurement geometry. This ensured that each test was performed under the same physical conditions.

In addition, measurements were taken using consistent operating parameters, including identical speed increments and loading conditions. Maintaining these conditions ensures that any variation in results is due to gearbox performance rather than differences in test execution.

Although minor variation is unavoidable due to manufacturing tolerances and material behaviour, the combination of controlled geometry, fixed setup, and consistent procedures provides a stable basis for comparative measurement.

3.6.3. Calibration of Measurement Systems

Calibration was used to ensure that measurement outputs were consistent and suitable for comparative analysis. The load cell was used in conjunction with a fixed radius lever arm, allowing torque to be determined directly from measured force. By maintaining a constant lever arm radius, torque calculations were standardised across all tests.

As direct motor torque data was not available, a calibration procedure was implemented to establish the torque–speed characteristics of the motor. This was achieved using a simplified test configuration in which the motor was directly coupled to a lever arm with the same 100 mm radius used in the main rig. Measuring force at known speeds allowed a torque-speed relationship to be obtained, providing a consistent basis for estimating input torque during gearbox testing.

Rotational measurements were obtained using 600 pulse-per-revolution encoders mounted via belt drives. This configuration provides sufficient resolution for detecting angular displacement and speed variations while avoiding direct mechanical interference with the system. Consistent mounting of the encoders ensured reliable signal acquisition throughout testing.

3.7. Sources of Error and Uncertainty

Despite efforts to control experimental conditions, several sources of error and uncertainty are inherent within the test rig and measurement approach. These arise from limitations in measurement systems, mechanical design, and assumptions made during data analysis.

Measurement uncertainty is introduced through the load cell and encoder systems. The load cell is subject to calibration accuracy, signal noise, and sensitivity to alignment, particularly as force must be applied perpendicular to the lever arm to ensure accurate torque calculation. Any deviation from this condition introduces error in the measured torque. Similarly, the rotary encoders, with a resolution of 600 pulses per revolution, introduce quantisation error in angular displacement and velocity measurement. While this resolution is sufficient for the scale of the experiment, it limits the precision of small angular measurements, particularly in backlash and stiffness evaluation.

Mechanical sources of error are primarily associated with alignment and compliance within the test rig. Although alignment features and a flexible coupling were used to minimise misalignment, small deviations between the motor, gearbox, and output shaft may still introduce parasitic loads and affect measured performance. The flexible coupling itself introduces additional torsional compliance, which may influence stiffness measurements by contributing to overall angular deflection. Additionally, the use of belt-driven encoders introduces potential for minor slip or elastic deformation, which may affect the accuracy of rotational measurements.

Manufacturing variability also contributes to uncertainty. As discussed in 3.3, components were produced using additive manufacturing processes with tolerances of approximately ± 0.05 – 0.1 mm for resin and ± 0.1 – 0.3 mm for FDM parts. These variations can affect gear meshing, clearances, and alignment, leading to differences in measured backlash and efficiency. Furthermore, the use of polymer materials introduces additional compliance and deformation under load, which influences the measured stiffness values compared to equivalent metallic systems.

Uncertainty is also introduced through the estimation of motor input torque. As direct torque data was not available, input torque was derived from a separately obtained torque–speed relationship. This assumes consistent motor behaviour between calibration and testing conditions, which may not fully account for variations such as temperature effects or driver performance. As a result, efficiency calculations may include systematic error associated with this assumption.

Finally, external factors such as friction within bearings, couplings, and other non-transmission components contribute to measured losses. While these effects are present in both gearbox tests and therefore partially cancel in comparative analysis, they introduce uncertainty in absolute efficiency values.

Despite these limitations, the experimental approach remains valid for comparative evaluation. Since both gearboxes were tested under identical conditions, the influence of systematic errors is consistent across measurements, allowing meaningful comparison of performance characteristics such as backlash, torsional stiffness, and efficiency.

4. Results and Discussion

4.1. Transmission Efficiency

The measured transmission efficiencies for both gearboxes at each operating speed are summarised in Figure 19 and presented graphically in Figure 20. Each data point represents the average of three repeated measurements, with error bars indicating the associated experimental uncertainty.

Input Speed (RPM)	Planetary Efficiency (%)	Cycloidal Efficiency (%)
30	89.1	81.8
60	91.3	83.7
90	92.1	81.6
120	88.3	78.3
150	85.7	76.2

Figure 19: Measured transmission efficiency at varying input speeds

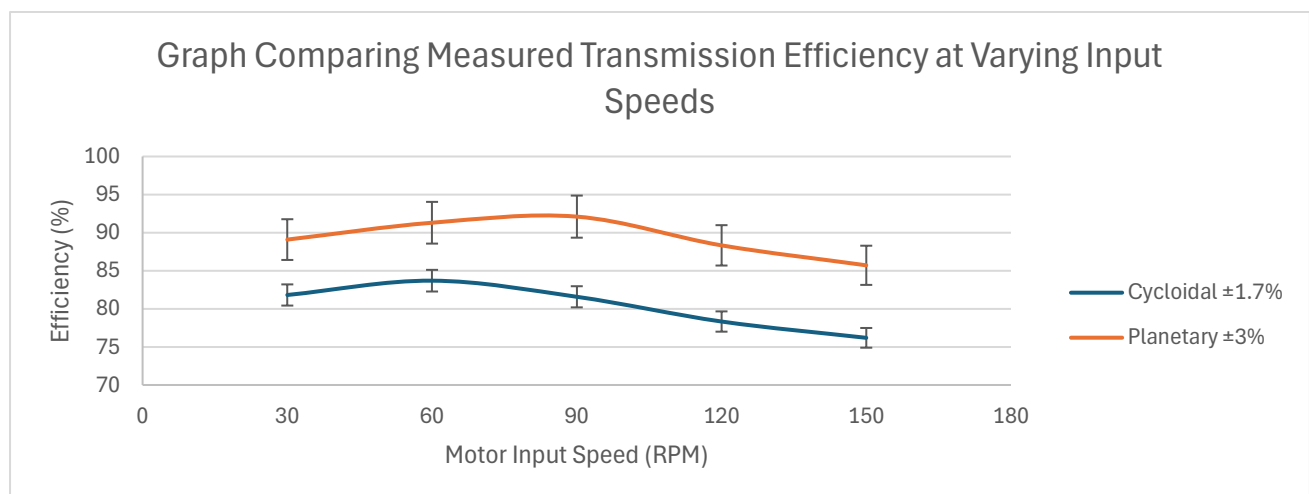


Figure 20: Graph comparing measured transmission Efficiency at Varying Input Speeds

The planetary gearbox demonstrates consistently higher efficiency than the cycloidal gearbox across the full operating range. Its efficiency increases from 30 rpm to a peak at 90 rpm before decreasing at higher speeds. In contrast, the cycloidal gearbox reaches its maximum efficiency at 60 rpm and then exhibits a continuous decline as speed increases. The difference in efficiency between the two gearboxes remains significant at all operating points, ranging from approximately 7–8 percentage points at lower speeds to around 9–10 percentage points at higher speeds.

The uncertainty shown in Figure 20 represents the combined experimental error, estimated from the variation between repeated measurements and measurement system limitations. Each data point is the average of three tests, with the error bars reflecting the variation observed between runs. Additional sources of uncertainty include load cell calibration, encoder resolution (600 pulses per

revolution), and the estimation of input torque from the motor calibration curve. Given that the experimental uncertainty is approximately $\pm 3\%$ for the planetary gearbox and $\pm 1.7\%$ for the cycloidal gearbox, at all operating speeds, the difference in measured efficiency exceeds the associated uncertainty, indicating that the observed trends are not attributable to measurement variation.

The trends observed are consistent with the operating principles discussed in the Literature Review. Cycloidal drives are known to exhibit increased power losses due to sliding contact between the cycloidal disc and ring pins (Hsieh, 2015), with losses becoming more pronounced at higher speeds. In contrast, planetary gear systems primarily rely on rolling contact between gear teeth, resulting in reduced frictional losses and higher efficiency.

The reduction in efficiency at higher speeds for both gearboxes can also be attributed to the motor torque–speed relationship established during calibration which can be viewed in Appendix 2. As rotational speed increases, frictional losses within the transmission become more significant relative to the transmitted power. This is particularly evident in the cycloidal gearbox, where sliding contact dominates and energy dissipation increases with velocity. This contributes to the observed decline in efficiency at higher operating speeds for both systems.

It is also important to consider the influence of the experimental setup and material choices on the measured results. The use of polymer-based components introduces additional compliance and friction compared to equivalent metallic systems, which may reduce absolute efficiency values. Furthermore, losses within the test rig itself, including friction in bearings, couplings, and belt-driven encoder systems, contribute to the measured input power and may slightly underestimate true gearbox efficiency.

Despite these limitations, the comparative validity of the results remains strong. Both gearboxes were tested under identical conditions, meaning that systematic losses and experimental constraints affect both systems equally. As a result, the observed difference in efficiency can be attributed primarily to the inherent transmission characteristics of the gearbox designs rather than artefacts of the test rig.

Overall, the results demonstrate that the planetary gearbox provides superior efficiency across the tested operating range, while the cycloidal gearbox exhibits lower efficiency and greater sensitivity to increasing speed. These findings align with established literature and reinforce the trade-off between efficiency and other performance characteristics such as stiffness and backlash.

The processed data for this can be found in Appendix 2.

4.2. Backlash

The measured backlash values for both gearboxes are summarised in Figure 21. Each value represents the average of three repeated measurements obtained using the procedure described in 3.5.2, with negligible variation observed between trials.

Gearbox Type	Total Backlash Input (°)	Total Backlash Output (°)
Planetary	4.20	0.21
Cycloidal	2.25	0.1125

Figure 21: Measured backlash for planetary and cycloidal gearboxes

The planetary gearbox exhibits significantly higher backlash than the cycloidal gearbox, with approximately 4.2° of input rotation required before output motion occurs, compared to 2.25° for the cycloidal system. This corresponds to an output backlash of 0.21° for the planetary gearbox and 0.1125° for the cycloidal gearbox. The cycloidal gearbox therefore demonstrates approximately a 46% reduction in backlash relative to the planetary design.

The backlash behaviour observed in both systems is consistent with the definition provided in the literature review, where backlash is characterised as the angular displacement occurring before torque transmission during reversal (International Organization for Standardisation, 2018). The measured “deadband” region is clearly visible in the recorded data, where input motion occurs without a corresponding output response until engagement is re-established.

The higher backlash in the planetary gearbox is expected due to its reliance on discrete gear tooth engagement. Clearance between mating tooth flanks, combined with manufacturing tolerances and bearing play, results in an inherent loss of motion during reversal. This behaviour aligns with the findings of Gilberti et al. (2010), who demonstrated that even small clearances in geared systems can significantly affect positioning accuracy.

In contrast, the cycloidal gearbox exhibits reduced backlash due to its multi-lobe contact mechanism and dual-disc configuration. As discussed in the literature review, the use of two cycloidal discs offset by 180° allows for compensation of clearances within the system, effectively reducing lost motion. The results obtained in this study support this behaviour, showing a clear reduction in backlash compared to the planetary gearbox.

However, it is notable that the cycloidal gearbox does not achieve zero backlash. This can be attributed to practical limitations in manufacturing and assembly. The use of additive manufacturing introduces dimensional tolerances, and the need to ensure free movement of components requires small clearances to prevent binding. Additionally, compliance in polymer materials and slight misalignment within the assembly contribute to residual backlash.

The consistency of the measured results across repeated trials indicates that backlash measurement was highly repeatable under the chosen experimental conditions. The minimal variation between tests suggests that measurement uncertainty is low relative to the magnitude of the observed differences, reinforcing the reliability of the comparison.

Overall, the results demonstrate that the cycloidal gearbox provides significantly lower backlash than the planetary gearbox, confirming the expected advantage of cycloidal mechanisms in applications requiring high positional accuracy. While the presence of non-zero backlash reflects the influence of manufacturing tolerances and material limitations inherent to prototype-scale implementations, the consistent performance difference between the two systems indicates that backlash behaviour is primarily governed by transmission architecture rather than material selection alone.

The processed data for this can be found in Appendix 2.

4.3. Torsional Stiffness

The torsional stiffness of both gearboxes was evaluated by measuring the relationship between applied torque and angular deflection (twist), with results presented in Figure 22 and Figure 23. Each data point represents the average of three repeated measurements, with negligible variation observed between trials.

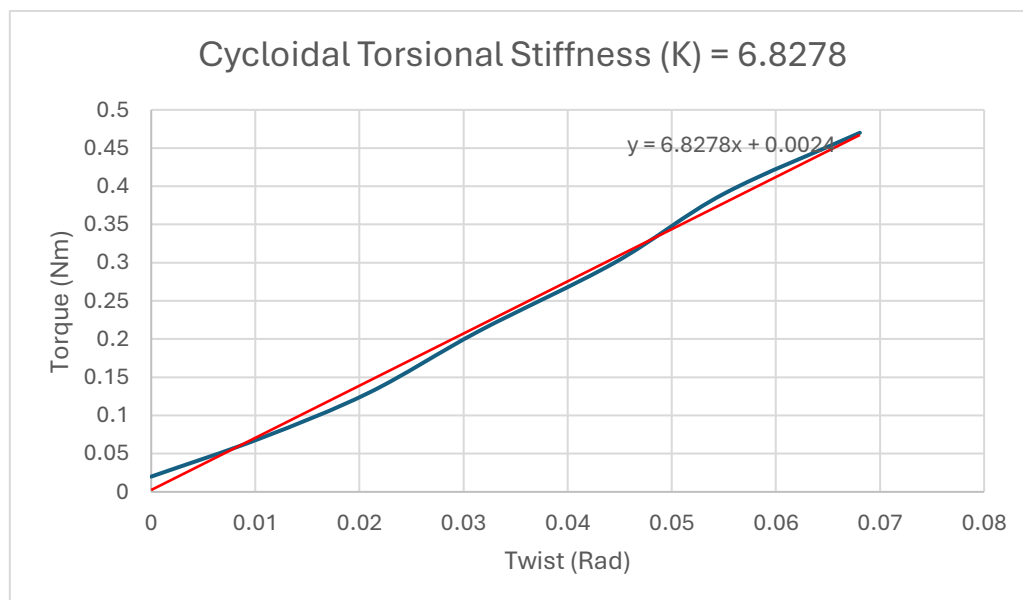


Figure 22: Graph showcasing torque-twist relationship in the cycloidal gearbox

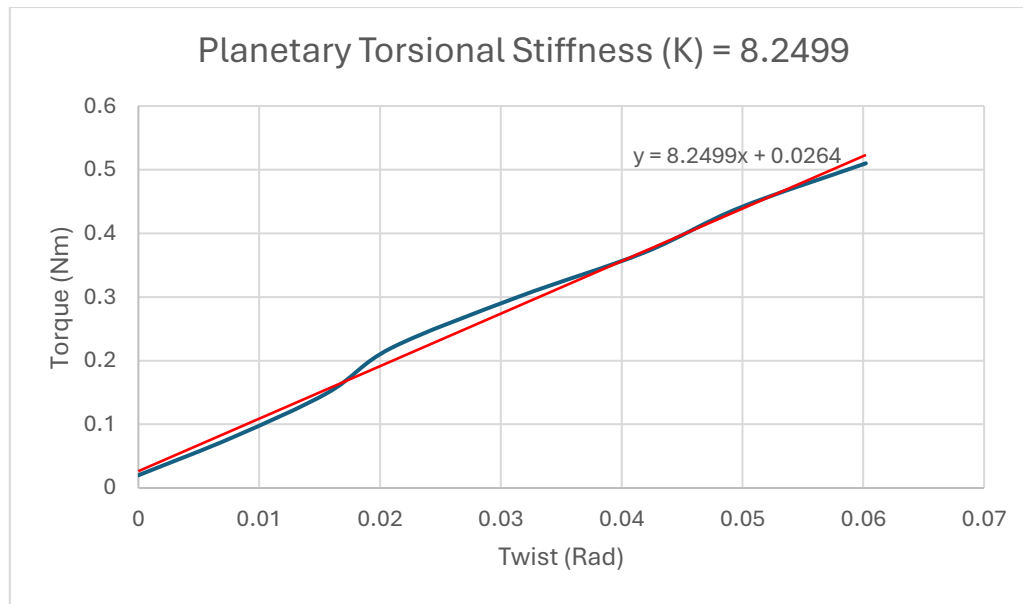


Figure 23: Graph showcasing torque-twist relationship in the planetary gearbox

For both gearboxes, a near-linear relationship between torque and angular deflection is observed, indicating elastic behaviour within the tested load range. Linear regression was applied to the data to determine torsional stiffness, defined as the gradient of the torque–twist relationship (Norton, 2020).

The cycloidal gearbox exhibited a stiffness of 6.83 Nm/rad, while the planetary gearbox demonstrated a higher stiffness of 8.25 Nm/rad. This indicates that, under the tested conditions, the planetary gearbox is approximately 21% stiffer than the cycloidal gearbox.

The linearity of both datasets suggests that deformation is dominated by elastic effects rather than backlash or non-linear contact behaviour within the tested range. However, minor deviations from perfect linearity can be observed, particularly at lower torque levels, which may be attributed to residual backlash, measurement resolution limits, or compliance within the test rig.

The observed stiffness behaviour differs from the idealised expectation presented in the literature review, where cycloidal drives are typically associated with higher torsional stiffness due to multi-lobe contact distributing load across multiple engagement points (Blagojevic, et al., 2014). In contrast, planetary gear systems rely on discrete tooth contact and distributed load sharing, which would generally be expected to result in greater compliance.

The higher measured stiffness of the planetary gearbox is primarily attributed to several practical implementation factors. Firstly, the cycloidal gearbox incorporates multiple interfaces, including disc-to-pin contact and eccentric motion, which introduce additional compliance under load. Secondly, the use of polymer materials increases deformation within these contact regions, reducing overall

stiffness. Additionally, the presence of self-lubricating bushings and clearances required for smooth operation may contribute to increased deflection.

In contrast, the planetary gearbox benefits from a more rigid load path through its gear teeth and carrier structure, which may result in reduced overall compliance under the tested conditions. The simpler geometry and fewer intermediate interfaces also reduce the number of locations where elastic deformation can accumulate.

It is important to note that the measured stiffness represents the combined stiffness of the gearbox and test rig, rather than the gearbox in isolation. Contributions from the flexible shaft coupling, encoder belt systems, and structural compliance within the rig may influence the measured values. However, as both gearboxes were tested under identical conditions, these effects are consistent across both datasets and do not invalidate the comparative analysis.

Overall, the results indicate that the planetary gearbox exhibits higher torsional stiffness than the cycloidal gearbox under the conditions tested. Although cycloidal drives are often associated with high torsional stiffness due to multi-point contact, the measured behaviour reflects the influence of practical implementation factors. The presence of multiple interfaces, including disc–pin contact and eccentric motion, introduces additional compliance, which is further amplified by the use of polymer materials. In contrast, the planetary gearbox exhibits a more direct and continuous load path, resulting in reduced cumulative deformation under load.

The processed data for this can be found in Appendix 2.

4.4. Comparative Summary

The experimental evaluation of both gearbox types highlights clear trade-offs between transmission efficiency, backlash, and torsional stiffness. The planetary gearbox exhibited consistently higher efficiency across the tested speed range, which is consistent with the predominance of rolling tooth contact and lower frictional losses. By contrast, the cycloidal gearbox exhibited lower efficiency and greater sensitivity to increasing speed, reflecting the increased sliding losses associated with disc–pin contact.

In backlash performance, the cycloidal gearbox outperformed the planetary gearbox. This is attributable to its multi-lobe engagement and dual-disc configuration, which compensates for internal clearance and reduces lost motion during reversal. The planetary gearbox, relying on discrete tooth engagement, inherently retained greater backlash due to necessary tooth flank clearance and accumulated manufacturing tolerances.

For torsional stiffness, the planetary gearbox exhibited higher measured stiffness under the tested conditions. Although this differs from the idealised expectation presented in the literature, the result reflects the influence of practical implementation factors, including polymer compliance, multiple contact interfaces, and the presence of bushings and operational clearances within the cycloidal design.

Taken together, the results indicate that the planetary gearbox performed more favourably in efficiency and stiffness, whereas the cycloidal gearbox performed more favourably in backlash. However, these findings should be interpreted in the context of the specific design and manufacturing constraints of this study, particularly the requirement for both gearboxes to operate within a comparable envelope and shared test conditions. This distinction is critical when interpreting the results, as it highlights the difference between theoretical capability and practically realised performance.

5. Critical Evaluation

This chapter critically evaluates the design, manufacture, and experimental methodology used in this study, with particular focus on the limitations influencing the measured performance of the planetary and cycloidal gearboxes. While the results presented in Results and Discussion provide a direct comparison between the two transmission types, it is necessary to consider the extent to which these results reflect inherent gearbox behaviour as opposed to constraints imposed by design choices, material selection, and experimental setup.

5.1. Design Evaluation

Both gearboxes were designed to operate within a shared envelope and achieve a comparable reduction ratio of approximately 20:1. While this approach was necessary to enable direct comparison under identical test conditions, it introduced limitations in the extent to which each gearbox could be optimised for its intended operating principle.

The planetary gearbox required a two-stage configuration to achieve the target reduction ratio within the available space, resulting in a more complex internal structure. However, this configuration allowed relatively conventional gear proportions to be maintained, supporting efficient power transmission and structural rigidity.

In contrast, the cycloidal gearbox inherently allows high reduction ratios within a single stage and is typically advantageous in compact actuator applications. The requirement to match the overall form factor of the planetary design limited the ability to fully exploit this advantage. As a result, the cycloidal gearbox tested in this study may not represent the most compact or optimised implementation of this transmission type.

Additionally, the cycloidal design incorporates multiple interacting components, including eccentric motion, disc-pin contact, and output pin engagement. While these features contribute to reduced backlash, they also introduce additional interfaces where compliance and energy losses can occur. The planetary gearbox, by comparison, benefits from a more direct load path, which contributes to its observed performance in stiffness and efficiency.

5.2. Material and Manufacturing Limitations

A significant limitation of the study was the transition from an initially intended metal design to polymer-based manufacturing using resin and FDM processes. While this approach enabled rapid

prototyping and reduced cost, it introduced material properties that differ substantially from those assumed in theoretical models.

Polymer materials exhibit lower stiffness and strength compared to metals, resulting in increased deformation under load. This is particularly relevant in the cycloidal gearbox, where multiple contact interfaces amplify the effect of material compliance. As a result, the measured torsional stiffness is likely lower than would be expected for an equivalent metal design.

Manufacturing tolerances associated with additive manufacturing also contributed to performance variation. Resin components provided improved accuracy compared to FDM parts; however, dimensional variation remained present, requiring additional clearances to ensure reliable assembly. These clearances directly influence backlash and may have prevented the cycloidal gearbox from achieving near-zero backlash.

Furthermore, assembly challenges were encountered, including failure of resin components during press-fitting of bearings. This required iterative redesign of bearing seats and structural features, demonstrating the sensitivity of the design to material limitations and stress concentrations.

5.3. Experimental Method Limitations

The experimental methodology used to evaluate gearbox performance introduced several sources of uncertainty. Efficiency measurement relied on estimation of motor input torque using a separately derived torque–speed relationship, rather than direct measurement. While this approach was necessary due to equipment constraints, it introduces potential error, particularly if motor behaviour varies between calibration and testing conditions.

The use of a lever arm and load cell for torque measurement assumes ideal force application perpendicular to the arm. Any deviation from this condition affects the accuracy of torque calculation. Additionally, encoder-based measurement of angular displacement is limited by resolution, particularly when measuring small deflections for stiffness and backlash.

Despite these limitations, the chosen methods remain appropriate for comparative analysis, as both gearboxes were evaluated under identical conditions using approaches consistent with small-scale experimental practice. While this introduces a systematic uncertainty in the calculated efficiency values, it primarily affects the absolute results and does not compromise the validity of the comparative trends between the two gearbox types.

5.4. Test Rig Limitations

The measured stiffness and efficiency values represent the combined behaviour of the gearbox and the test rig, rather than the gearbox in isolation. Components such as the flexible shaft coupling, encoder belt drives, and structural elements of the rig introduce additional compliance and friction.

The flexible coupling, while necessary to accommodate misalignment, contributes to torsional compliance, potentially reducing the measured stiffness of both systems. Similarly, belt-driven encoder systems may introduce minor slip or elastic deformation, affecting the accuracy of angular measurements.

While alignment features were incorporated to minimise these effects, perfect alignment cannot be guaranteed, and small deviations may introduce parasitic loads. These factors highlight that the test rig is not a perfectly rigid system and therefore influences the measured performance.

5.5. Comparative Validity of Results

Despite the limitations identified, the comparative validity of the results remains strong. Both gearboxes were tested under identical conditions, with the same test rig, measurement systems, and operating parameters. As a result, systematic errors introduced by the experimental setup affect both gearboxes equally.

This consistency ensures that the relative differences observed in efficiency, backlash, and stiffness are primarily attributable to the gearbox designs rather than external factors. While the absolute values may be influenced by material properties and experimental constraints, the trends and comparisons between the two systems remain meaningful.

5.6. Project Reflection and Management

The project was initially planned using a structured timeline, with distinct phases for design, manufacture, and testing. However, several deviations from the original plan occurred due to practical constraints related to material availability, manufacturing access, and iterative design requirements.

Lead times on materials and components had a significant impact on project progression. Delays in sourcing resin and other manufacturing materials, as well as electronic components required for the test rig, extended the development timeline. In addition, procurement of components for revised gearbox designs introduced further delays, as multiple iterations were required before functional prototypes were achieved.

Access to manufacturing facilities also contributed to schedule disruption. Waiting for approval to use university manufacturing equipment resulted in a delay of approximately one month, while a university holiday closure during the test rig manufacturing phase led to an additional week of lost time. These factors highlight the dependency of project timelines on external resource availability, particularly within shared laboratory environments.

Several technical challenges during development further impacted progress. The initial preliminary test rig exhibited poor dimensional accuracy due to tolerancing issues, requiring redesign and reprinting once additional filament was sourced. Similarly, incorrect dimensions in sourced threaded inserts necessitated the remanufacture of input and output shafts, introducing additional rework. Component failures during assembly, particularly in resin parts, also required redesign of critical features such as bearing seats and load-bearing elements.

An unexpected interruption occurred during testing due to an electrical fault, which resulted in damage to components and required replacement before testing could continue. This further emphasised the need for robustness in both electrical design and test setup validation.

Despite these challenges, the project was successfully completed by adapting the development approach and prioritising key objectives. The experience highlighted the importance of early validation of purchased components, careful consideration of manufacturing tolerances, and the need to account for lead times and facility access in project planning.

Overall, the project demonstrated the iterative nature of engineering development, where design, manufacture, and testing are closely interlinked. It reinforced the importance of flexibility in project management and the value of anticipating practical constraints when working with complex mechanical systems.

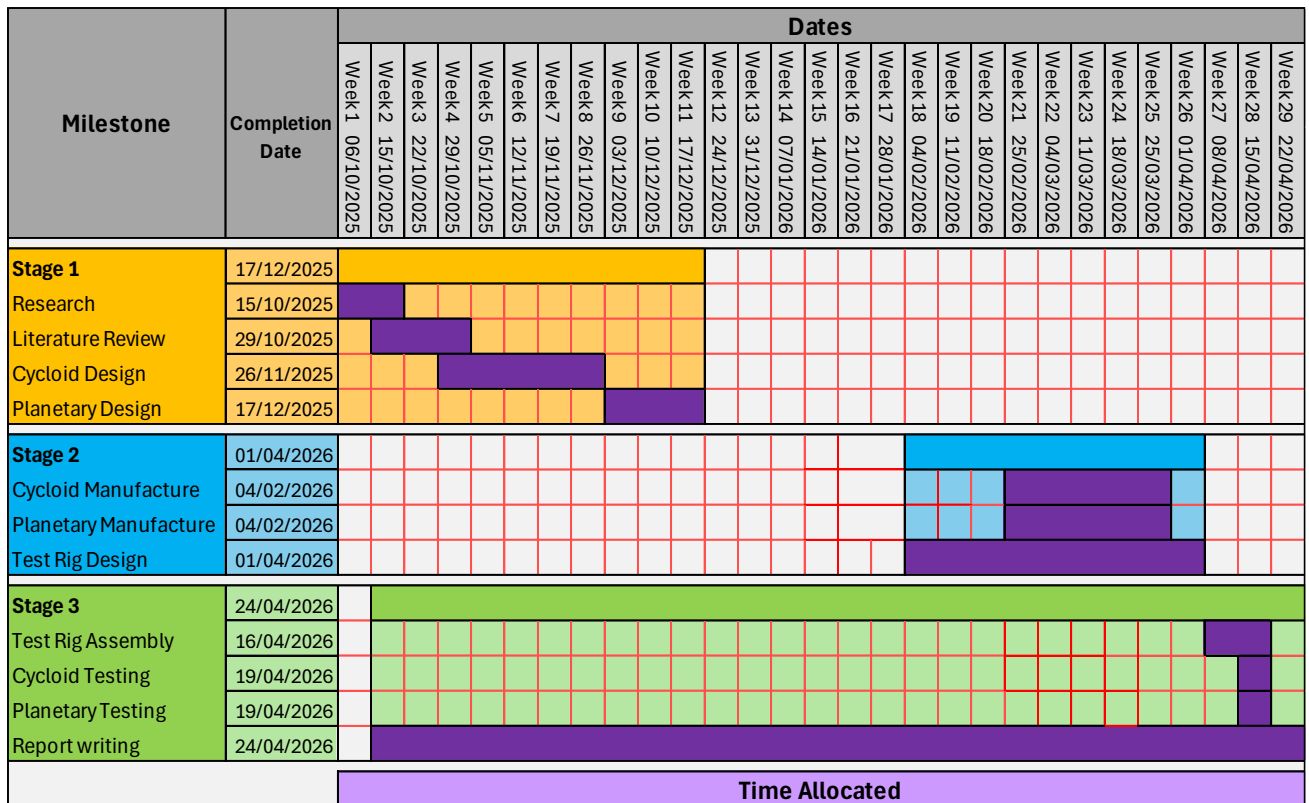


Figure 24: Final project Gantt chart

6.Design Improvements and Future Work

6.1. Gearbox Design Improvements

6.1.1.Cycloidal Gearbox

The cycloidal gearbox demonstrated strong backlash performance; however, its overall performance was limited by design constraints imposed during development. A key limitation was the requirement to match the form factor of the planetary gearbox, which restricted the ability to fully exploit the inherent compactness of cycloidal mechanisms. In practice, cycloidal drives are capable of achieving high reduction ratios within significantly smaller volumes, and future designs should prioritise compactness as a primary optimisation objective rather than enforcing geometric equivalence between transmission types.

Material selection also had a significant impact on performance. The use of polymer-based components introduced additional compliance, particularly at disc–pin interfaces, reducing torsional stiffness and increasing energy losses. Transitioning to metallic components, as originally intended, would significantly improve stiffness, wear resistance, and load capacity, while also allowing tighter tolerances to be achieved. This would likely result in both improved efficiency and further reduction in backlash.

The design of critical load-bearing features, such as output pins and disc pins, could also be improved. These components were identified as high-stress regions and were susceptible to deformation and failure. Increasing pin diameter, improving support conditions, and using hardened metallic components with precision bushings would reduce compliance and improve durability. Additionally, refinement of the cycloidal disc profile, including optimisation of eccentricity and lobe geometry, could reduce sliding losses and improve efficiency without compromising backlash performance.

6.1.2.Planetary Gearbox

The planetary gearbox achieved high efficiency and stiffness; however, several design decisions limited its potential optimisation. The requirement to achieve a reduction ratio of approximately 20:1 within a constrained envelope necessitated a two-stage configuration. While effective, this introduced additional interfaces, increasing complexity and potential sources of loss.

An alternative design approach would involve reducing the overall gear ratio to enable a single-stage planetary system. This would simplify the design, reduce internal losses, and improve

manufacturability. Although this would reduce torque multiplication, it would provide a more direct evaluation of planetary gearbox performance and may result in improved efficiency and stiffness characteristics.

As with the cycloidal gearbox, material selection played a critical role. The use of additively manufactured polymer gears limited achievable precision and surface finish, contributing to frictional losses and wear. Replacing these components with machined metal gears would improve load distribution, reduce friction, and enhance durability.

Further improvements could be made to load sharing between planet gears. Incorporating design features such as floating sun gears or carriers would allow for better load distribution, reducing localised stresses and improving overall gearbox performance.

6.2. Experimental and Test Rig Improvements

The experimental setup introduced several limitations that could be addressed to improve measurement accuracy and reliability.

A primary limitation was the use of motor torque estimation rather than direct measurement. Future work should incorporate an inline torque sensor at the input shaft, eliminating reliance on calibration curves and improving the accuracy of efficiency calculations.

The structural stiffness of the test rig also influenced measured results, particularly for torsional stiffness. The current setup measures combined system stiffness, including contributions from the gearbox, coupling, and supporting structure. To isolate gearbox stiffness, a redesigned test rig with significantly increased rigidity is required. This could include the use of machined metal frames and rigid shaft couplings, reducing compliance within the system.

The use of belt-driven encoder systems introduced additional sources of compliance and potential measurement error. Direct shaft-mounted encoders would improve measurement fidelity by eliminating belt deformation and slip. Additionally, the use of higher-resolution encoders would improve sensitivity when measuring small angular displacements, particularly for backlash and stiffness evaluation.

Manufacturing accuracy of the test rig itself was also a limiting factor. The use of additively manufactured components introduced tolerancing errors that affected alignment and measurement consistency. Future designs should utilise machined components or higher-precision manufacturing methods to improve geometric accuracy and reduce systematic error.

6.3. Improvements to Experimental Methodology

The experimental methodology adopted in this study provided a practical approach for comparative evaluation; however, several improvements could be implemented to enhance measurement accuracy and align the approach more closely with established methods.

For efficiency measurement, a more rigorous approach would be the adoption of the back-to-back testing methodology proposed by (Lin & Shih, 2013), as discussed in 2.3.3. This would remove the need for motor torque estimation and provide a more direct evaluation of transmission losses. While not feasible within the constraints of this study, its implementation would significantly improve the reliability of efficiency measurements.

The current methodology evaluates gearbox behaviour as part of a coupled system, meaning that measured stiffness includes contributions from both the gearbox and the test rig. A refined approach would aim to isolate gearbox behaviour by measuring relative displacement directly across the transmission, reducing the influence of external compliance and allowing stiffness to be defined more accurately as a property of the gearbox itself.

In addition, the experimental procedure could be extended beyond steady-state measurements to include transient analysis. Evaluating system response during load application or reversal would provide further insight into the influence of backlash and compliance, particularly in applications involving dynamic loading conditions.

Finally, the integration of experimental results with computational modelling would provide a more robust framework for validation. Comparing measured behaviour with analytical or numerical predictions would allow discrepancies to be identified more clearly and attributed to specific design or material factors, strengthening the overall interpretation of results.

6.4. Future Work

Further investigation is required to extend the findings of this study.

Testing at higher torque levels would provide a more representative assessment of gearbox performance under realistic operating conditions. In addition, long-term durability testing would allow evaluation of wear mechanisms, particularly in cycloidal pin interfaces and planetary gear teeth.

Exploration of alternative design configurations, including different reduction ratios and scaling effects, would provide further insight into how performance varies with geometry. Finally, comparative testing of polymer and metal implementations of the same designs would allow the direct influence of material properties to be quantified.

7. Conclusion

This study set out to design, manufacture, and experimentally evaluate planetary and cycloidal gearboxes developed to a shared specification representative of compact actuator applications. A dedicated test rig was successfully designed and implemented to enable controlled measurement of key transmission characteristics, including efficiency, backlash, and torsional stiffness under consistent operating conditions. This approach addressed a gap identified in the literature, where direct experimental comparison under equivalent conditions is limited.

The results demonstrate that transmission performance is strongly dependent on gearbox architecture, with clear and consistent differences observed between the two systems. The planetary gearbox exhibited superior efficiency across the tested speed range, with peak performance occurring at intermediate speeds before declining at higher rotational speeds. This behaviour is consistent with rolling contact-dominated power transmission, where frictional losses are minimised. In contrast, the cycloidal gearbox exhibited lower efficiency and increased sensitivity to speed, reflecting the energy dissipation associated with sliding contact between the cycloidal disc and ring pins.

Backlash measurements showed that the cycloidal gearbox achieved significantly lower values, with approximately 46% lower input backlash than the planetary design. This confirms that backlash performance is primarily governed by engagement mechanics rather than material selection alone, with multi-lobe contact and phase-shifted disc configurations effectively reducing lost motion. However, the presence of measurable backlash in both systems highlights the influence of manufacturing tolerances and material compliance in practical implementations.

Torsional stiffness results indicated that the planetary gearbox exhibited higher stiffness under the tested conditions. While this differs from idealised expectations, the observed behaviour reflects the influence of real-world design factors, including polymer material compliance, cumulative deformation across multiple interfaces, and the presence of operational clearances within the cycloidal system. This demonstrates that theoretical performance advantages may not be fully realised without appropriate material selection and manufacturing precision.

Collectively, these findings demonstrate that gearbox selection represents a trade-off between efficiency, stiffness, and positional accuracy, rather than optimisation of a single parameter. The planetary gearbox is more suitable for applications requiring efficient power transmission and structural rigidity, such as continuous-duty or load-bearing systems. In contrast, the cycloidal gearbox is better suited to applications where reduced backlash and high positional accuracy are critical, such as robotic joints and precision positioning systems.

Importantly, the results reflect the performance of the specific prototype implementations tested, rather than the theoretical limits of each gearbox topology. Constraints imposed by material selection, additive manufacturing processes, and the requirement for a shared design envelope limited the extent to which each gearbox could be fully optimised. This highlights the distinction between theoretical capability and practically realised performance and reinforces the need to consider implementation constraints during the design process.

Beyond the specific findings, this project demonstrates the importance of experimental validation in mechanical system design. The integration of design, manufacture, and testing revealed that performance is influenced not only by transmission principles, but also by tolerances, assembly conditions, and test methodology. These factors must be accounted for when translating analytical designs into functional systems.

Overall, this work provides a controlled and directly comparable experimental evaluation of planetary and cycloidal gearboxes, contributing to a more practical understanding of transmission behaviour in compact actuator applications. The findings support informed engineering decision-making by demonstrating how theoretical performance characteristics are modified by real-world constraints, and by identifying the conditions under which each gearbox type offers the greatest advantage.

References

- Altintas, Y., 2012. *Manufacturing Automation: Metal Cutting Mechanics, Machine Tool Vibrations, and CNC Design*. 2nd ed. Cambridge: Cambridge University Press.
- American Gear Manufacturers Association, 2015. *Gear classification and design standards*. s.l.:s.n.
- Blagojevic, M. et al., 2014. A new design of a two-stage cycloidal speed reducer. *Journal of Mechanical Design*.
- Bolton, W., 2015. *Mechatronics: Electronic Control Systems in Mechanical and Electrical Engineering*. 6th ed. Harlow: Pearson.
- Budynas, R. & Nisbett, J., 2015. *Shigley's Mechanical Engineering Design*. 10th ed. McGraw-Hill: s.n.
- Gilberti, H., Cinquemani, S. & Legnani, G., 2010. Effects of transmission mechanical characteristics on the choice of motor-reducer. *Mechanism and Machine Theory*.
- Harris, T. & Kotzalas, M., 2006. *Rolling Bearing Analysis*, s.l.: CRC Press.
- Hsieh, C. F., 2015. Efficiency Analysis of a Cycloidal Reducer Considering the Profile Modification and Force Distribution. *Journal of Mechanical Science and Technology*.
- International Organization for Standardisation, 2018. *ISO gear accuracy standards*. s.l.:s.n.
- Kahraman, A., 1994. *Load sharing characteristics of planetary transmissions*, s.l.: Mechanism and Machine Theory.
- Kahraman, A., 1999. Load sharing characteristics of planetary transmissions. *Mechanism and Machine Theory*.
- Lin, W. & Shih, H., 2013. Design of a test rig for gearbox efficiency measurement. *Journal of Advanced Mechanical Design, Systems, and Manufacturing*.
- National Aeronautics and Space Administration, 2020. *Aerospace actuator and control system design guidelines*. Washington: NASA.
- Norton, R., 2011. *Machine Design: An Integrated Approach*, s.l.: Prentice Hall.

Norton, R., 2020. *Design of Machinery: An Introduction to the Synthesis and Analysis of Mechanisms and Machines*. 6th ed. New York: McGraw-Hill.

Parker, R. & Lin, J., 2004. Mesh phasing relationships in planetary and epicyclic gears. *Journal of Mechanical Design*.

Siciliano, B., Sciavicco, L., Villani, L. & Oriolo, G., 2016. *Robotics: Modelling, Planning and Control*. London: Springer.

Spong, M., Hutchinson, S. & Vidyasgar, M., 2006. *Robot Modeling and Control*, New Jersey: John Wiley & Sons.

Tsai, S., Huang, K. & Huang, C., 2015. Dynamic measurement of the backlash in a precision speed reducer. *Journal of the Chinese Society of Mechanical Engineers*.

Appendix 1

Motor Torque Preliminary Test Code

```
#include "HX711.h"

// =====
// PIN DEFINITIONS
// =====
const byte PIN_HX_DT = 6;
const byte PIN_HX_SCK = 7;
const byte PIN_STEP = 8;
const byte PIN_DIR = 9;

// =====
// MOTOR / TEST SETTINGS
// =====
const int MOTOR_FULL_STEPS_PER_REV = 200; // NEMA 17
const int MICROSTEPPING = 1; //
const long MOTOR_STEPS_PER_REV = MOTOR_FULL_STEPS_PER_REV * MICROSTEPPING;

// 5 calibration speeds
float testRPMs[5] = {30.0, 60.0, 90.0, 120.0, 150.0};

// Lever arm
const float LEVER_ARM_M = 0.100f; // 100 mm
const float G_CONST = 9.80665f;

// HX711 calibration factor
// Placeholder for now - replace after actual calibration
float loadCellCalibrationFactor = -1000.0f;

// Test timing
const unsigned long SETTLE_TIME_MS = 4000; // let force stabilise
const unsigned long SAMPLE_TIME_MS = 5000; // average over this time
const unsigned long SAMPLE_INTERVAL_MS = 100;

// Step pulse timing
const unsigned int STEP_PULSE_HIGH_US = 5;

// =====
// GLOBALS
// =====
HX711 scale;

float commandedRPM = 0.0f;
unsigned long stepIntervalUs = 0;
bool stepPinHigh = false;
unsigned long nextStepRiseUs = 0;
unsigned long stepPulseOffUs = 0;

// =====
// HELPER FUNCTIONS
// =====
void updateStepInterval(float rpm) {
    commandedRPM = rpm;
}

if (rpm <= 0.0f) {
    stepIntervalUs = 0;
    return;
}

float stepsPerSecond = (rpm * MOTOR_STEPS_PER_REV) / 60.0f;
stepIntervalUs = (unsigned long)(1000000.0f / stepsPerSecond);
}

void updateStepper() {
    if (stepIntervalUs == 0) {
        digitalWrite(PIN_STEP, LOW);
        stepPinHigh = false;
        return;
    }
}

unsigned long nowUs = micros();

if (!stepPinHigh && (long)(nowUs - nextStepRiseUs) >= 0) {
    digitalWrite(PIN_STEP, HIGH);
    stepPinHigh = true;
    stepPulseOffUs = nowUs + STEP_PULSE_HIGH_US;
    nextStepRiseUs = nowUs + stepIntervalUs;
}

if (stepPinHigh && (long)(nowUs - stepPulseOffUs) >= 0) {
    digitalWrite(PIN_STEP, LOW);
    stepPinHigh = false;
}
}

float readLoadCellUnits(byte samples = 5) {
    return scale.get_units(samples);
}

float unitsToForceN(float unitsValue) {
    return unitsValue * G_CONST;
}

float forceToTorqueNm(float forceN) {
    return forceN * LEVER_ARM_M;
}

void waitWithStepper(unsigned long durationMs) {
    unsigned long startMs = millis();
    while (millis() - startMs < durationMs) {
        updateStepper();
    }
}

void printInstructions() {
    Serial.println("=== Motor Torque Calibration ===");
    Serial.println("Commands:");
    Serial.println("T -> tare load cell");

    Serial.println("R -> run full 5-speed calibration sequence");
    Serial.println("C<number> -> set HX711 calibration factor");
    Serial.println("D -> print current settings");
    Serial.println("H -> help");
    Serial.println();
    Serial.println("CSV output columns:");
    Serial.println("rpm,avg_units,avg_force_N,avg_torque_Nm");
    Serial.println();
}

void printSettings() {
    Serial.print("Load cell calibration factor: ");
    Serial.println(loadCellCalibrationFactor, 4);

    Serial.print("Lever arm (m): ");
    Serial.println(LEVER_ARM_M, 4);

    Serial.print("Microstepping: ");
    Serial.println(MICROSTEPPING);

    Serial.println("Test RPMs:");
    for (int i = 0; i < 5; i++) {
        Serial.print(" ");
        Serial.print(i + 1);
        Serial.print(" ");
        Serial.println(testRPMs[i], 2);
    }
    Serial.println();
}

void runCalibrationSequence() {
    Serial.println("Starting calibration sequence...");
    Serial.println("rpm,avg_units,avg_force_N,avg_torque_Nm");

    for (int i = 0; i < 5; i++) {
        float rpm = testRPMs[i];
        Serial.print("Settling at ");
        Serial.print(rpm, 2);
        Serial.println(" RPM...");

        updateStepInterval(rpm);
        waitWithStepper(SETTLE_TIME_MS);

        unsigned long startMs = millis();
        unsigned long lastSampleMs = 0;

        float unitsSum = 0.0f;
        float forceSum = 0.0f;
        float torqueSum = 0.0f;
        int sampleCount = 0;

        while (millis() - startMs < SAMPLE_TIME_MS) {
            updateStepper();

            if (millis() - lastSampleMs >= SAMPLE_INTERVAL_MS) {
                lastSampleMs = millis();

                float unitsVal = readLoadCellUnits(3);
                float forceN = unitsToForceN(unitsVal);
                float torqueNm = forceToTorqueNm(forceN);

                unitsSum += unitsVal;
                forceSum += forceN;
                torqueSum += torqueNm;
                sampleCount++;
            }
        }

        float avgUnits = 0.0f;
        float avgForceN = 0.0f;
        float avgTorqueNm = 0.0f;

        if (sampleCount > 0) {
            avgUnits = unitsSum / sampleCount;
            avgForceN = forceSum / sampleCount;
            avgTorqueNm = torqueSum / sampleCount;
        }

        Serial.print(rpm, 2);
        Serial.print(",");
        Serial.print(avgUnits, 5);
        Serial.print(",");
        Serial.print(avgForceN, 5);
        Serial.print(",");
        Serial.print(avgTorqueNm, 5);
        Serial.println();

        // small pause between speeds
        updateStepInterval(0.0f);
        waitWithStepper(1500);
    }

    Serial.println("Calibration sequence complete.");
    Serial.println();
}

void handleSerial() {
    if (!Serial.available()) return;

    String cmd = Serial.readStringUntil('\n');
    cmd.trim();

    if (cmd == "T") {
        scale.tare();
        Serial.println("Load cell tared.");
    }
    else if (cmd == "R") {
        runCalibrationSequence();
    }
}
```

```

else if (cmd == "D") {
  printSettings();
}
else if (cmd == "H") {
  printInstructions();
}
else if (cmd.startsWith("C")) {
  float newCal = cmd.substring(1).toFloat();
  if (newCal != 0.0f) {
    loadCellCalibrationFactor = newCal;
    scale.set_scale(loadCellCalibrationFactor);
    Serial.print("New calibration factor set to: ");
    Serial.println(loadCellCalibrationFactor, 4);
  }
}
}

// =====
// SETUP
// =====
void setup() {
  Serial.begin(115200);

  pinMode(PIN_STEP, OUTPUT);
  pinMode(PIN_DIR, OUTPUT);

  digitalWrite(PIN_STEP, LOW);

```

```

digitalWrite(PIN_DIR, HIGH); // fixed direction

scale.begin(PIN_HX_DT, PIN_HX_SCK);
scale.set_scale(loadCellCalibrationFactor);
scale.tare();

updateStepInterval(0.0f);
nextStepRiseUs = micros() + 1000;

printInstructions();
printSettings();
}

// =====
// LOOP
// =====
void loop() {
  updateStepper();
  handleSerial();
}

// How to use it
// Open Serial Monitor at 115200 baud.
// Use: T
// To tare the load cell.
// Then: R
// To run the full 5-speed calibration sequence.

```

Full Data Collection Test Code

```

#include "HX711.h"

// =====

// Pin definitions
// =====
const int ENC_IN_A = 2;
const int ENC_OUT_A = 3;
const int ENC_IN_B = 4;
const int ENC_OUT_B = 5;

const int HX_DT = 6;
const int HX_SCK = 7;

const int STEP_PIN = 8;
const int DIR_PIN = 9;

// =====
// Encoder settings
// =====
const float INPUT_ENCODER_PPR = 600.0;
const float OUTPUT_ENCODER_PPR = 600.0;
const float QUAD_FACTOR = 4.0;

// =====
// Load cell
// =====
HX711 scale;

// Replace this with your real calibration factor
float LOADCELL_SCALE = -1000.0;

// If HX711 returns kg instead of N, set this true
bool loadcellReadsKg = false;

// =====
// Globals
// =====
volatile long inputCount = 0;
volatile long outputCount = 0;

// =====
// ISR routines
// =====
void isrInputA() {
  bool a = digitalRead(ENC_IN_A);
  bool b = digitalRead(ENC_IN_B);
  if (a == b) inputCount++;
  else inputCount--;
}

void isrOutputA() {
  bool a = digitalRead(ENC_OUT_A);
  bool b = digitalRead(ENC_OUT_B);
  if (a == b) outputCount++;
  else outputCount--;
}

// =====
// Utility
// =====
void setMotorDirection(bool forward) {
  digitalWrite(DIR_PIN, forward ? HIGH : LOW);
}

void stepMotorRPM(float rpm, unsigned long durationMs) {
  if (rpm <= 0) return;

  const float motorStepsPerRev = 200.0; // 1.8 degree stepper
  float stepFreq = (rpm * motorStepsPerRev) / 60.0;
  float periodUs = 1000000.0 / stepFreq;

  unsigned long startMs = millis();
  unsigned long lastStepUs = micros();

  while (millis() - startMs < durationMs) {
    unsigned long nowUs = micros();
    if (nowUs - lastStepUs >= periodUs) {
      digitalWrite(STEP_PIN, HIGH);
      delayMicroseconds(3);
      digitalWrite(STEP_PIN, LOW);
    }
  }
}

```

```

    lastStepUs = nowUs;
  }
}

float getLoadN(int samples = 5) {
  float val = scale.get_units(samples);
  if (loadcellReadsKg) return val * 9.81;
  return val;
}

void resetEncoders() {
  noInterrupts();
  inputCount = 0;
  outputCount = 0;
  interrupts();
}

void printCSVHeader() {
  Serial.println("test,time_ms,input_count,output_count,load_N,dir,target_rpm");
}

void logDataRow(const char* testName, int dirValue, float targetRPM) {
  long inC, outC;
  noInterrupts();
  inC = inputCount;
  outC = outputCount;
  interrupts();

  float loadN = getLoadN(3);

  Serial.print(testName);
  Serial.print(",");
  Serial.print(millis());
  Serial.print(",");
  Serial.print(inC);
  Serial.print(",");
  Serial.print(outC);
  Serial.print(",");
  Serial.print(loadN, 4);
  Serial.print(",");
  Serial.print(dirValue);
  Serial.print(",");
  Serial.println(targetRPM, 2);
}

// =====
// Backlash data collection
// =====
void runBacklashRaw() {
  resetEncoders();

  // Forward slowly
  setMotorDirection(true);
  for (int i = 0; i < 80; i++) {
    stepMotorRPM(5.0, 20);
    logDataRow("BACKLASH", 1, 5.0);
  }

  delay(500);

  // Reverse slowly
  setMotorDirection(false);
  for (int i = 0; i < 80; i++) {
    stepMotorRPM(5.0, 20);
    logDataRow("BACKLASH", -1, 5.0);
  }
}

// =====
// Stiffness data collection
// User applies torque manually while motor holds
// =====
void runStiffnessRaw() {
  resetEncoders();

  Serial.println("INFO,Apply load gradually by hand or fixture");

  for (int i = 0; i < 30; i++) {
    logDataRow("STIFFNESS", 0, 0.0);
    delay(1000);
  }
}

```

```

}
}

// =====
// Efficiency vs speed raw collection
// =====
void runEfficiencyRaw() {
  float speeds[] = {30, 60, 90, 120, 150};
  int nSpeeds = sizeof(speeds) / sizeof(speeds[0]);

  for (int s = 0; s < nSpeeds; s++) {
    float rpm = speeds[s];
    resetEncoders();

    setMotorDirection(true);

    // settle
    stepMotorRPM(rpm, 2000);

    // record for 5 seconds
    unsigned long startMs = millis();
    while (millis() - startMs < 5000) {
      stepMotorRPM(rpm, 50);
      logDataRow("EFFICIENCY", 1, rpm);
    }

    delay(1000);
  }
}

// =====
// Setup / loop
// =====
void setup() {
  Serial.begin(115200);

  pinMode(ENC_IN_A, INPUT_PULLUP);
  pinMode(ENC_OUT_A, INPUT_PULLUP);
  pinMode(ENC_IN_B, INPUT_PULLUP);
  pinMode(ENC_OUT_B, INPUT_PULLUP);

  pinMode(STEP_PIN, OUTPUT);
  pinMode(DIR_PIN, OUTPUT);

  attachInterrupt(digitalPinToInterrupt(ENC_IN_A), isrInputA, CHANGE);
  attachInterrupt(digitalPinToInterrupt(ENC_OUT_A), isrOutputA, CHANGE);

  scale.begin(HX_DT, HX_SCK);
  scale.set_scale(LOADCELL_SCALE);
  scale.tare();

  printCSVHeader();

  Serial.println("READY");
  Serial.println("Send:");
  Serial.println("B for backlash raw data");
  Serial.println("S for stiffness raw data");
  Serial.println("E for efficiency raw data");
}

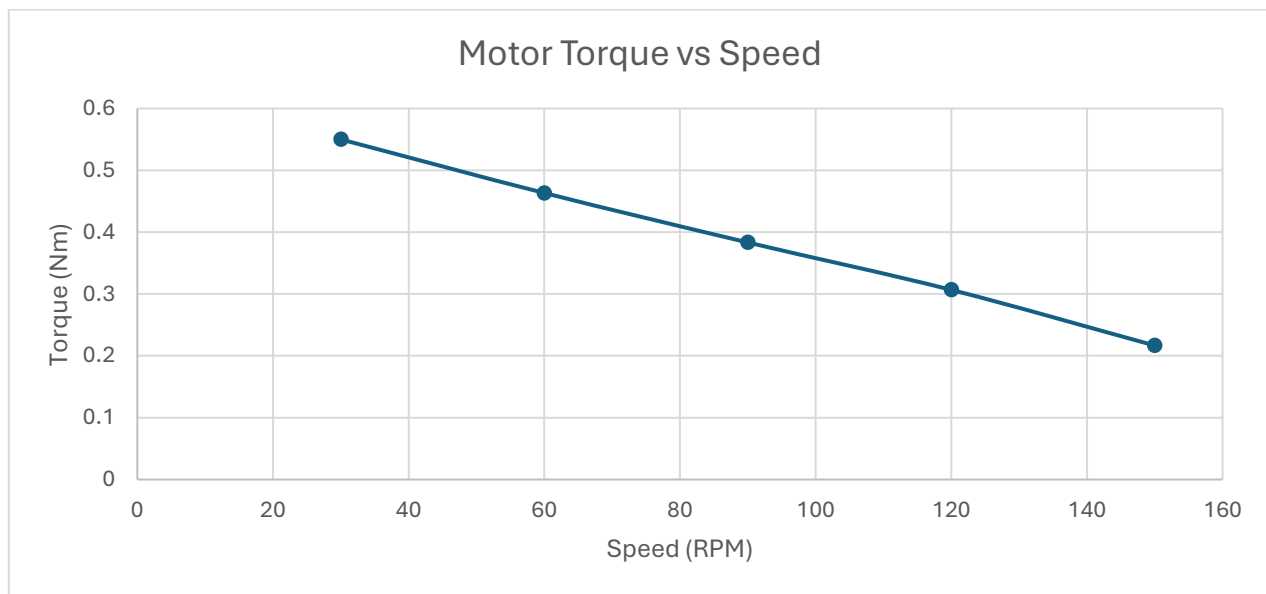
void loop() {
  if (Serial.available()) {
    char cmd = Serial.read();

    if (cmd == 'B' || cmd == 'b') {
      runBacklashRaw();
    }
    else if (cmd == 'S' || cmd == 's') {
      runStiffnessRaw();
    }
    else if (cmd == 'E' || cmd == 'e') {
      runEfficiencyRaw();
    }
  }
}

```

Appendix 2

Preliminary Motor Torque Test							
Speed (RPM)	Raw Data (N)			Converted Torque Data (Nm)			
	Test 1	Test 2	Test 3	Test 1	Test 2	Test 3	Average Data
30	5.6	5.4	5.5	0.56	0.54	0.55	0.55
60	4.6	4.7	4.6	0.46	0.47	0.46	0.463333333
90	3.9	3.8	3.8	0.39	0.38	0.38	0.383333333
120	3.1	3	3.1	0.31	0.3	0.31	0.306666667
150	2.2	2.1	2.2	0.22	0.21	0.22	0.216666667



Cycloidal											
Torsional Stiffness											
Average Data											
Time (ms)	Input Count	Output Count	Load (N)	Direction	Target RPM	Input Angle (deg)	Output Angle (deg)	Output Corrected	Calculated Twist (deg)	Twist (Rad)	Torque (Nm)
0	0	0	0.2	0	0	0	0	0	0	0	0.02
1000	4	0	0.7	0	0	0.6	0	0	0.6	0.01047198	0.07
2000	8	0	1.3	0	0	1.2	0	0	1.2	0.02094395	0.13
3000	13	1	2.1	0	0	1.95	0.15	3	1.8	0.03141593	0.21
4000	18	1	3	0	0	2.7	0.15	3	2.55	0.0445059	0.3
5000	23	2	3.9	0	0	3.45	0.3	6	3.15	0.05497787	0.39
6000	28	2	4.7	0	0	4.2	0.3	6	3.9	0.06806784	0.47

Planetary											
Torsional Stiffness											
Average Data											
Time (ms)	Input Count	Output Count	Load (N)	Direction	Target RPM	Input Angle (deg)	Output Angle (deg)	Output Corrected	Calculated Twist (deg)	Twist (Rad)	Torque (Nm)
0	0	0	0.2	0	0	0	0	0	0	0	0.02
1000	3	0	0.8	0	0	0.45	0	0	0.45	0.00785398	0.08
2000	6	0	1.5	0	0	0.9	0	0	0.9	0.01570796	0.15
3000	9	1	2.2	0	0	1.35	0.15	3	1.2	0.02094395	0.22
4000	13	1	3	0	0	1.95	0.15	3	1.8	0.03141593	0.3
5000	17	1	3.7	0	0	2.55	0.15	3	2.4	0.0418879	0.37
6000	21	2	4.4	0	0	3.15	0.3	6	2.85	0.04974188	0.44
7000	25	2	5.1	0	0	3.75	0.3	6	3.45	0.06021386	0.51

							Planetary Backlash					
Average Data												
Time (ms)	Input Count	Output Count	Direction	Target RPM	Input Angle (deg)	Output Angle (deg)	Forward Deadband	Reverse Deadband	Total Backlash	Total Backlash Input (deg)	Total Backlash Output (deg)	
0	0	0	1	5	0	0	16	12	28	4.2	0.21	
50	4	0	1	5	0.6	0						
100	8	0	1	5	1.2	0						
150	12	0	1	5	1.8	0						
200	16	1	1	5	2.4	0.15						
250	20	1	1	5	3	0.15						
300	24	2	1	5	3.6	0.3						
600	24	2	-1	5	3.6	0.3						
650	20	2	-1	5	3	0.3						
700	16	2	-1	5	2.4	0.3						
750	12	1	-1	5	1.8	0.15						
800	8	1	-1	5	1.2	0.15						
850	4	0	-1	5	0.6	0						

Cyclotidal																	
Target RPM	Start Time / End Time (Delta Time)	Start Input Count	End Input Count	Delta Input Count	Input RPM	Start Output Count	End Output Count	Delta Output Count	Output RPM	Load (N)	Output Torque (Nm)	Input Torque	Input Angular Speed (rad/s)	Output Angular Speed (rad/s)	Input Power	Output Power	Efficiency (%)
30	0	5000	5	0	6000	6000	30	0	300	300	1.5	90	0.55	0.15709633	1.72789596	1.413187684	81.8138182
60	6000	11000	5	0	12000	12000	60	0	600	600	3	77	0.46	0.263185307	0.514192065	2.419326543	83.6965217
90	12000	17000	5	0	18000	18000	90	0	900	900	4.5	6.2	0.42	0.474723898	3.581461463	3.581461468	95.57084737
120	18000	23000	5	0	24000	24000	120	0	1200	1200	6	4.7	0.32	0.125665706	0.628318531	3.76901118	2.953097094
150	24000	29000	5	0	30000	30000	150	0	1500	1500	7.5	3.2	0.21	0.157096327	0.788398163	3.29867229	2.51274123
Planetary																	
Target RPM	Start Time / End Time (Delta Time)	Start Input Count	End Input Count	Delta Input Count	Input RPM	Start Output Count	End Output Count	Delta Output Count	Output RPM	Load (N)	Output Torque (Nm)	Input Torque	Input Angular Speed (rad/s)	Output Angular Speed (rad/s)	Input Power	Output Power	Efficiency (%)
30	0	5000	5	0	6000	6000	30	0	300	300	1.5	98	0.55	0.15709633	1.72789596	1.5393804	89.0900009
60	6000	11000	5	0	12000	12000	60	0	600	600	3	84	0.46	0.263185307	0.514192065	2.63887829	91.3044783
90	12000	17000	5	0	18000	18000	90	0	900	900	4.5	7.0	0.38	0.474723898	3.581461463	3.28672386	92.1026316
120	18000	23000	5	0	24000	24000	120	0	1200	1200	6	5.3	0.21	0.125665706	0.628318531	3.76901118	3.33333333
150	24000	29000	5	0	30000	30000	150	0	1500	1500	7.5	3.6	0.21	0.157096327	0.788398163	3.29867229	4.71428571