ME 14 TRANSMISSION PROJECT

DESIGN NOTEBOOK

SPRING 2025

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Group 7

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PROJECT OVERVIEW

May 4th – May 5th

1.1 Objectives and Problem Statement

The team's goal was to design and fabricate a mechanical transmission that would be coupled to the rotational power of a brushed DC motor and a bicycle wheel. The transmission was required to fit within strict physical and budgetary constraints, deliver the required torque and speed ratio, and be integrated with the provided testing apparatus.

1.2 Requirements and Constraints

A $200 budget was allocated to source materials and components, mainly from McMaster-Carr and SDP/SI. The transmission had to be housed in an acrylic enclosure, be designed for manufacturability within our available machine shop resources and be tested in a timed competition setting that lasted a week and a half. Accuracy, efficiency, compactness, and durability were considered critical.

1.3 Background and Prior Work

After multiple transmission types were explored by the team, a planetary gear system was settled on due to its compact design and its ability to provide a high gear ratio in a small footprint. Planetary gears have been widely used in applications requiring efficient power transmission and torque multiplication, such as automotive transmissions and robotics.

1.4 Some applications of transmission systems

Applications for this kind of transmission included electric bicycles, robotics, and any system where space-efficient power transmission was needed.

** More information is laid out in the contents below.*

** Transmissions were scheduled to be tested and ranked on Wednesday, May 21 in GTL 135 during class time.*

Conceptual Design (Preliminary Design Review) *May 4th – May 8th*

2.1 Brainstorming and Initial Sketches *May 4th*

Various mechanical transmission options were considered to convert the brushed DC motor's high-speed, low-torque output into appropriate torque and speed for the bicycle wheel. Initial sketches focused on compactness, manufacturability, and performance within the project's size and budget constraints.

2.2 Functional Requirements and Subsystems

The transmission was required to:

- Convert motor speed into increased torque for effective wheel drive.
- Maximize angular velocity while maintaining torque within an effective range.
- Fit within the physical constraints of the acrylic enclosure (6" wide x 6" tall x 12 mm thick walls).
- Stay within the $200 budget.
- Be manufacturable using the available machine shop resources.

The system was divided into subsystems including the gear train (sun, planet, and ring gears), gearbox enclosure with mounting and alignment features, and shaft support components.

2.3 Design Alternatives, Selection and Justification *May 5th*

Three primary design options were evaluated:

- Two-Stage Spur Gear Transmission: Efficient and simple, though larger and more complex to assemble.
- Chain and Sprocket Transmission: Easy to implement with reliable torque transmission, but less compact and requiring tension.
- Planetary Gear Transmission: Compact, capable of high torque, and relevant to many real-world applications despite increased complexity.

The Planetary Gear Transmission was selected because:

- It provided an optimal balance of compact size, torque output, and manufacturability.
- It aligned with practical applications such as automotive and robotics transmissions, providing valuable experiential learning.
- Efficiency loss was minimal and acceptable within the project scope.
- The design fit within the enclosure and machining constraints.

With epicyclic transmissions, multiple methods existed to convert input rotation into output rotation. The selected configuration fixed the ring gear, designated the sun gear as the input, and the carrier (attached to two planetary gears) as the output. This arrangement was consistent with the objective of maximizing torque transmission and angular velocity within physical constraints.

The transmission ratio and output torque were derived considering the non-interference condition expressed as:

$$G = \frac{\theta_M}{\theta_L} = \frac{N_S + N_P}{N_S} = \frac{D_S + D_r}{D_S} \qquad \text{(Transmission ratio)}$$

$$\tau_c = \mu \tau_s G \qquad \text{(Output torque)}$$

$$\frac{N_s + N_p}{N_s} \in \mathbb{N} \qquad \text{(Non-interference condition)}$$

Where variables correspond to gear teeth numbers(N), pitch diameters(D), and torque coefficients(τ_c), ensuring smooth gear meshing without interference.

An online planetary gear simulator (https://www.thecatalystis.com/gears/) was used to determine individual gear pitch diameters while considering both score optimization and availability of gears from suppliers SDP/SI. Planetary gear systems tend to be highly efficient; a conservative efficiency of 90% is assumed for the design, slightly lower than typical ballpark values to account for real-world losses.

2.4 System Architecture Diagram and Rough Dimensions

May 6th - May 7th

- Preliminary enclosure dimensions are approximately 6" wide x 6" long x 6.84" tall, fitting within the allowed 7" x 6" x 8" envelope.
- Acrylic housing with 12 mm thick sidewalls includes a center hole and standoff holes to constrain the ring gear.
- The baseplate contains four 0.266 in diameter holes arranged in a 4" x 4" pattern for alignment and mounting.
- Internal standoffs are designed to support the ring and planet gears.
- Planetary Gear Specifications included:
  - Sun Gear: 48 teeth
  - Planet Gears: 84 teeth
  - Ring Gear: 216 teeth
  - Approximate Gear Ratio: 5.5
  - Diametral Pitch: 72
  - Face Width: 0.125 in
  - Ring Gear Outer Diameter: 4 in

Lithium grease was selected to reduce overall friction and improve transmission efficiency. It was applied to the contacting gear surfaces and moving parts to ensure smoother operation and prolong component lifespan.

The gears were specified with a diametral pitch of 72, which was chosen to balance strength and compactness within the design constraints. Consideration was given to using a lower diametral pitch to increase load capacity, but space and material availability restricted the final selection.

Accurate alignment was achieved through datum management using precisely flat machined surfaces on the enclosure and baseplate. This approach ensured consistent positioning of gears and shafts and maintained tight tolerances during assembly.

For rotational support, quality bearings were chosen over bushings. Bearings with an ABEC-1 tolerance rating were used, with consideration given to upgrading to ABEC-3 for better precision. It was noted that approximately 98% of bearings sold fell between ABEC-1 and ABEC-3. The choice was made to balance performance with budget constraints, selecting a cost-effective option that maintained reliability.

This conceptual design was prepared for the Preliminary Design Review (PDR) held on May 7. Feedback from the PDR informs several refinements to improve manufacturability and assembly, which are incorporated in the subsequent Critical Design Review (CDR) on May 9th.

Detailed Design

May 8th - May 10th

3.1 Mechanical Design

After a critical review and further refinement of the previous design, the following gear specifications calculated and acquired from extensive research into optimal planetary gear configurations were used:

- Sun Gear: **28 teeth**
- Planet Gears: **64 teeth**
- Ring Gear: **156 teeth**
- Net Gear Ratio: **~6.5714**

Key dimensions and design features were incorporated:

- Diametral Pitch: **48**
- Face Width: **0.125 in**
- Ring Gear OD: **4 in**
- Enclosure Dimensions: **5.5" wide x 6" long x 5.89" tall**
- Overall design was ensured to fit within the required **7" tall x 6" wide x 8"** long envelope

Mounting features included:

- **6 x 10-32** counterbored holes on bottom
- Two **0.05"** deep slots with locational clearance fits for side walls
- **4 x ¼-20** clearance holes for test rig mounting
- Baseplate slot depth was decreased to reduce the total assembly height
- Ring gear was face-mounted without standoffs
- Two bearing holes were provided for the planet gears
- **A ⅜"** diameter keyed hole was used for a press fit shaft interface
- A **0.25"** diameter hole was press fit with **a 0.125"** bore bearing to axially constrain the **⅜"** keyed shaft

Tolerances and Fits Datum management were applied through machined flat surfaces. The base plate and side walls were to be machined flat on x, y, and z surfaces, and holes were measured from a defined datum for consistency. Tolerances were controlled to accommodate bearing and shaft fits.

Assembly Design Shaft collars were to be used to constrain the carrier axially, allowing flexibility in positioning the planet gears. Retaining rings were also to be incorporated on the opposite side to add additional axial constraint. These rings required machined grooves, so placement would be finalized before machining.

Shafts:

- Thin **1/8"** support shafts connected the input and output rods.
- Thicker **⅜"** shaft was used for input.

- A **1/8"** bore bearing was press fit into the carrier.
- A clearance hole was drilled into the output shaft to prevent collisions.
- The output shaft was keyed to the carrier, while the sun and planet gears were pinned.

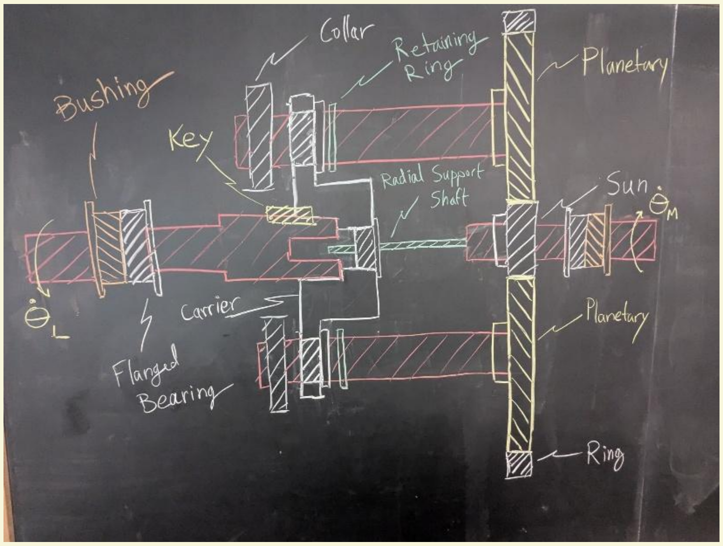


Figure 1: *Drawing of the complete system assembly with all parts labelled.*

3.2 Material and Component Selection Flanged bearings were selected:

- Unshielded types were used to reduce friction.
- Shielded bearings were avoided as they introduced lubrication-related efficiency losses.
- ABEC-3 and ABEC-5 rated bearings were selected where appropriate.

A conservative efficiency of 90% was assumed based on research.

Steel was selected over aluminum due to its higher modulus of elasticity **(30,000 ksi vs. 10,000 ksi)**, which reduced deflection and the associated risk of misalignment in the planetary gear system. Hardness was maintained below **Rockwell 43-46** to permit machining and pinning into shafts.

3.3 Design Calculations The transmission ratio was derived using: Gear Ratio (G):

The same calculations were used since equations and formulas for acquiring gear ratios and conditions are standard across any gear system applications.

3.4 Design for Manufacturing and Assembly (DFMA) A machining plan was established:

- Baseplate: Milled flat on x and z faces, holes drilled and counterbored, slots milled to position walls
- Walls: Milled flat on x and y surfaces, holes measured from x-datum and tapped (10-32 threads)
- Carrier: Cut using waterjet (if allowed), or band sawed and milled; holes drilled and reamed; keyed hole created; bearings press fit

Spindle grease was to be applied sparingly to limit viscosity and energy loss. Components were chosen for manufacturability using available machine shop resources. The assembly process was planned to allow flexibility where possible, while ensuring alignment and stability during operation.

Strengths:

- High torque density
- Compact form factor
- Coaxial input and output shafts
- Load sharing between planets enhances torque output

Weaknesses:

- Complex assembly process
- Sensitivity to gear alignment and backlash
- Higher bearing loads resulting in increased friction

Fixing gear constraints (axial and radial), finalizing bearing types (flanged/unflanged, ABEC rating), and optimizing the carrier interface were ongoing tasks addressed during the transition from preliminary to critical design

Additional actions taken

- Gear meshing clearance and backlash were confirmed.
- Torque/speed ratios were re-evaluated under revised efficiency assumptions (65–75%).
- Simulator outputs were compared against calculated gear ratios.
- Final material justification was completed for shafts (steel over aluminum).
- Bearing specifications were finalized based on performance vs. cost.
- A final bill of materials (BOM) was edited and checked against part availability from SDP/SI and McMaster-Carr. The materials included:
    - MCMASTER-CARR
        - 2 Shaft Collars – $3.70
    - SDP_SI
        - Ring Gear – $75.16
        - Planet Gears – $33.01
        - Sun Gear – $23.12
        - 0.25 ID Flanged Ball bearings - $38.48
        - 0.125ID Small flanged ball bearing - $7.21
- Backup components were selected from the shop inventory in case of order delays or stockouts.
- Lubrication type was discussed with shop supervisors (Trent and Paul); a final decision between lithium grease, machine grease, and spindle grease was deferred pending test fit.
- CDR slides were completed and reviewed as a team, including an overview schedule and finalized renderings.

Prototyping and Fabrication

May 12th - May 20th

4.1 Manufacturing Process Plans

May 12th

The parts were machined based on a plan reviewed with Trent to make sure all tolerances and fits made sense. Focus was on getting the most critical features done first—things like bearing bores and shaft fits—so everything would go together smoothly.

4.2 Tooling and Machining Setup and Task Allocation

Baseplate (Ethan/Jason)

- The aluminum stock (5.5" × 6") to be squared off on the mill.
- Two flat slots milled for alignment with the side walls.
- Six 10-32 counterbored holes and four ¼-20 through holes drilled.

Side Plates (Alex/Jimmy)

- The plates squared to 5" × 5.5" as a pair.
- A ⅜" hole drilled and reamed in the center of each for a press fit shaft.
- Tapped 10-32 holes added to the top and bottom.
- Four 8-32 holes drilled for mounting the ring gear.
- A 3.4" diameter pocket machined using CNC.
- A 0.45" bearing hole counterbored to a depth of 0.035".

Top Plate (Jorge)

- The stock squared to 3.25" × 5.5".
- Six 10-32 holes drilled to match the base.

Carrier (Deon/Jimmy)

- The shape milled from rectangular stock.
- Three bearing press fit holes reamed.
- A counterbore added for the keyed output shaft.
- Material removed from the wings to reduce rotational inertia.
- A possible redesign considered to simplify the geometry.

Shafts

- A ⅜" keyed output shaft turned down to 0.25" on one end.
- A 0.2" hole drilled into the opposite end, and a D-chamfer added for easier coupler fitting.
- The input shaft turned and tapered down to fit snugly, then pinned to the sun gear.
- Two carrier shafts cut and D-chamfered.

Bushings (Ethan)

- Brass bushings lathed to size, and flange thickness was adjusted.

- Holes drilled, and slip fit tolerances based on Shigley's tables.

CAD and GD&T(Everyone)

- Every member allocated a task on a part was responsible for making the CAD and GD&T for the part they were to work on and then a group review was to be done before machining began. ***Refer to A2-A3***

4.3 Part Fabrication Logs

Baseplate(ACER Mill) *May 13th*

- The 5.5" × 6" acrylic plate was squared using a ½" end mill via climb milling for smoother finish.
- Locational clearance slots were milled using the same end mill.
- 6 ¼-20 through holes were drilled using a size 7/8 drill bit in the slots 0.75" away from one end and 2" from each other.
- These same holes were counterbored(10-32) 0.19" deep.
- 4 0.266 through holes were drilled 0.75 away from one end and 4" from each other.

Side Plates(ACER Mill)

- Helped square off two 5" × 5.5" acrylic plates using a ½" end mill via climb milling for smoother finish.
- Confirmed that the center holes were drilled and reamed to ⅜" for press-fit bearings.
- 10-32 and 8-32 holes were tapped and drilled using size 10 drill bits and taps.
- A 3.4" pocket and bearing counterbore were CNC-machined.

Top Plate(ACER Mill) *May 14th*

- Helped square off the 3.25" × 5.5" plate and 10-32 through holes were drilled size 10 drill bits.

⅜" Keyed Output Shaft(ACER Lathe)

- The Aluminum shaft was cut using a vertical bandsaw to a size of 2.5".
- Turned down to 0.25" (1.61" of the shaft from one end in sections) and drilled 0.2" into the other using a size 7 drill bit going a depth of 3/8" into the part.

Carrier Shafts (Lathe)

- Shafts were cut to length and D-chamfered.
- Planet gears were pinned in place.

Brass Bushings (Lathe) *May 15–16th*

- Flanges were reduced, holes drilled and slip fit modeled/checked with some of the machined parts like the shafts and bearings.

Carrier (ACER Mill and Lathe)

- Final two-part design of the carrrier was milled carefully using a v block since its geometry was not optimal for the lathe.

- Press-fit bearing holes were reamed.
- Excess material was removed to reduce inertia.

Ring Gear Holes (ACER Mill)

- Clamped the gear to an aluminum plate with socket head screws to prevent deflection.
- Used the mill to properly edge find and machine part with respect to datums.
- Mounting holes were drilled using a size 7 drill bit.

Spacers (ACER Mill) *May 17th*

- Dimensions were confirmed in CAD.
- Machined to have an inner diameter of 0.2510 slightly bigger than the input shafts.

Carrier Finish(ACER Mill) *May 19th*

- Finalized proper keyed hole press fit.
- Verified part alignment and fitment on the shaft.

Bushings

- Finished boring the inner diameter using a size for an oversized slip fit.
- Press fit tested and installed into side plates.

Output Shaft Redo *May 20th*

- Shaft remade using the same process as before but with more care to improve 0.25" fit with minimal play.
- Verified diameter along full length using calipers.

¼" Input Shaft (ACER Lathe)

- The Aluminum shaft was cut using a vertical bandsaw to a size of 3".
- One end was turned to 0.125" creating the radial support shaft and pinned with gear.
- Shaft was tapered 0.2" from this end using the lathe where the tool was angled at -12.56 degrees from the horizontal.
- The coupler alignment was verified in CAD.

D-Chamfer Output Shaft

- Applied final D-chamfer to updated output shaft.
- Checked fit with coupler and bearings.

Pin Sun Gear

- Measured axial distance to carrier precisely for gear alignment.
- Carefully pinned sun gear in place using press fit and retaining ring.

4.4 Assembly Procedure and Notes

Parts were dry-fit before final assembly to confirm shaft alignments, fits, and gear clearances. Bearings and shafts were inserted by hand where no hammering required was possible. Grease type was selected(spindle) after checking with Trent. Ring gear holes were drilled with the plate clamped to a rigid block to avoid bending. Everything was test-assembled before moving forward. Assembly was done on a granite surface plate to ensure everything was assembled on a level surface.

4.5 Fabrication Challenges and Solutions

- **Tight fits**: A few reamed holes were undersized, so light sanding was used. Applied to input shafts too.
- **Scheduling**: Some machines have inevitable drift when cutting/drilling/milling, so early morning and first two mills were prioritized since they machine accurately better.

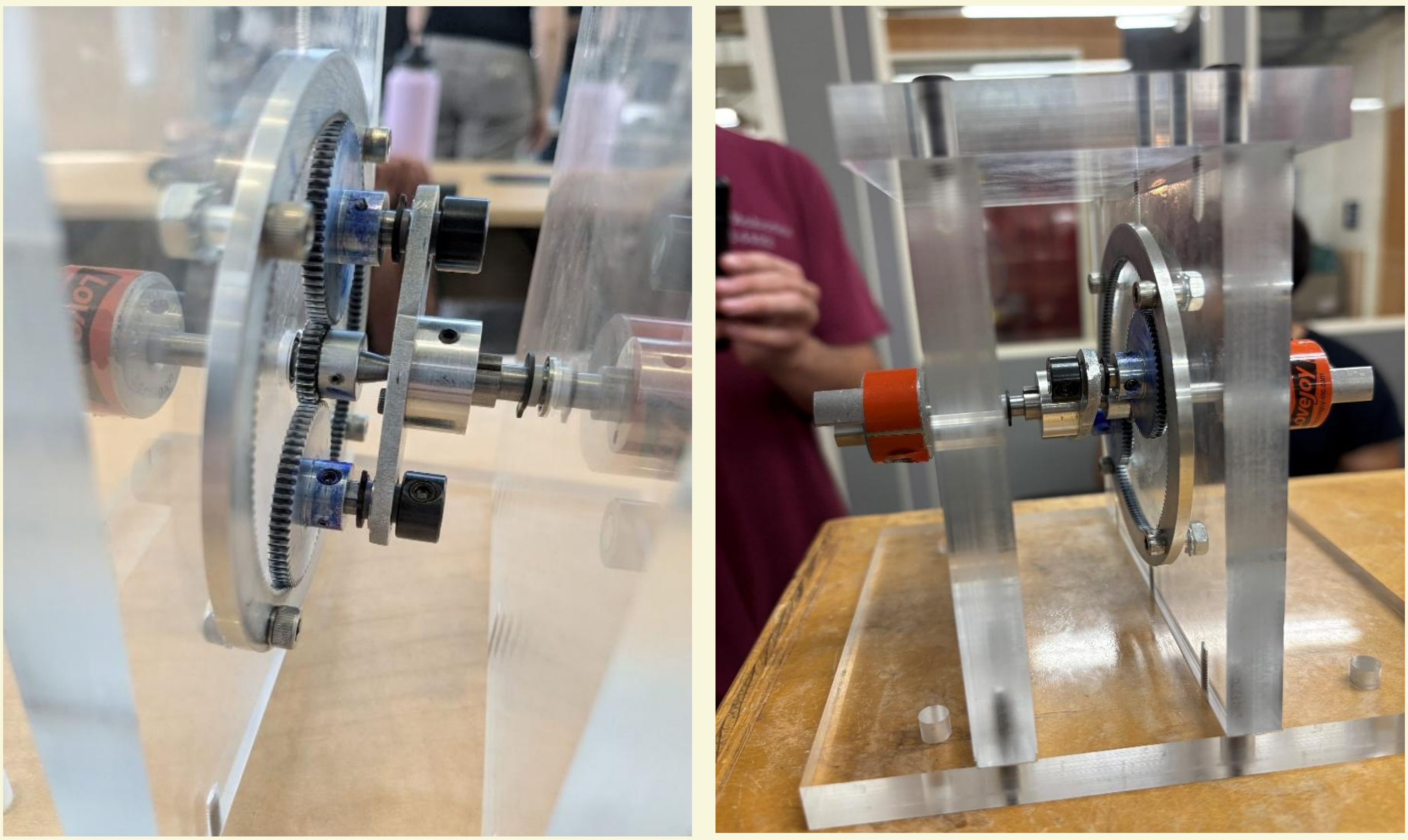


Figure 2(a) and (b) *Complete Setup of the Planetary Gear System in between the two acrylic plates with couplers connected to the ends.*

TESTING AND VALIDATION + CONTEST DAY

May 20th

5.1 Test Plan and Procedures

After assembly, the machine was taken to the test rig where it was connected to the motor and bicycle wheel via couplers. Bushings were considered from the initial design to support the input and output shafts radially and reduce deflections. It turned out, however, that the addition of the bushings introduced significant frictional losses. During testing in the rig, acceleration was reduced, and the bike wheel very quickly decelerated after the motor was turned off. Moreover, there was visible wear on the input shaft due to contact with the bushing. Upon further inspection of the design, the conclusion was the addition of the bushing, specifically on the input side wall, over constrained the input shaft, as it already had two points of support from the flanged bearing mounted on the side wall and the bearing in the carrier. The bushing was ultimately removed from the final assembly, which greatly improved performance and significant losses were no longer apparent. Furthermore, it was noted that securing the base super tightly also caused significant reduction in speed and acceleration, so screws were fastened lightly to allow the shafts to spin without having the reaction forces from the bearing constrain their movements.

Figure 3: Transmission system *connected to the test rig through couplers.*

5.2 Contest Day

Each transmission was tested in the final contest using a constant 10V DC power supply and a standard bicycle wheel rig. The wheel started from rest and was spun up by the motor through the transmission until it reached its top speed. All entries were evaluated under the same conditions.

During testing, torque output from the transmission and the angular speed of the wheel were continuously measured using a Futek FSH02563 rotary torque sensor with a built-in encoder. Real-time plots of torque and speed were displayed as each system ramped up. Contestants were able to watch their transmission's performance live while the system reached steady state. Two test trials were run for each transmission system to ensure each system was performing at peak conditions and smoothen the data.

5.3 Data Collection and Processing

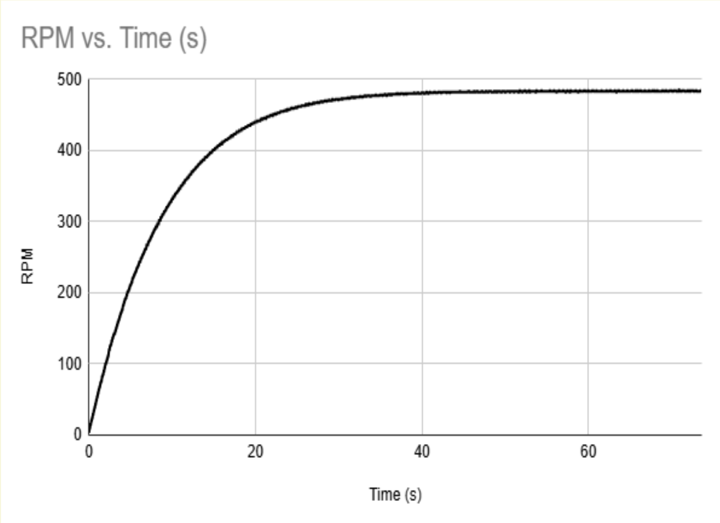
Torque and angular velocity data were automatically recorded throughout each run. The system logged these values over time, and results were plotted live. Once each test finished, the collected data was used to calculate a score using a provided cost function.

The score reflected how efficiently torque was delivered at high speeds and how fast it took to get to 250rpm, balancing both force and rotational velocity. Each team's score was determined based on their final output performance as measured by the sensor. No manual processing was required — the system handled all calculations.

5.4 Comparison to Theoretical Predictions

The MATLAB analysis assumed 70% power transmission efficiency and a transmission ratio of 6.5. Under these parameters, a maximum angular velocity of ~466 RPM was predicted, while the system's maximum RPM during the first trial run was 485 as seen in Fig. 4. Additionally, the MATLAB simulation predicted a maximum power output of ~12 W, which the transmission system also surpassed with a maximum power of ~13 W; see Fig. 5.

Comparing our transmission to the planetary transmission held in the shop, we were also pleasantly surprised that our gears turned with much less resistance. This could be attributed to a variety of differences between the two systems like potential tolerancing errors, the difference in the number of planetary gears (2 versus 3), the overall mass, moment of inertia, quality of bearings, etc. Notably, the gears used in our transmission were lighter and thinner, corresponding to less overall inertia in our system.



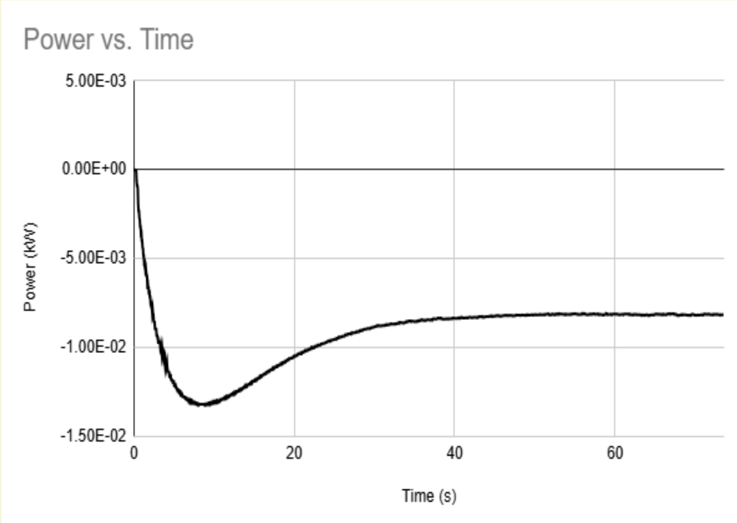


Figure 4: *RPM vs. Time (s) for our first trial run. Our max RPM was 485.*

Figure 5: *Power (Kilowatts) vs. Time (s) for our first trial run. Our max power output was 0.013 Kilowatts.*

Discussion and Reflection

6.1 Lessons Learned

- Our MATLAB analysis assumed a 70% efficiency, but actual performance was better - highlighting that real systems, when well-built, can outperform simplified or conservative models.
- Adding both bushings and bearings over-constrained the shafts, introducing unnecessary friction and wear. This emphasized the importance of understanding when added components help vs. hurt performance.
- The low resistance and smooth operation were likely due to careful machining, lightweight components, and good alignment. Small details in tolerancing and assembly had a big impact on efficiency.
- The axial play in the test rig coupler affected engagement and may have skewed performance data. This showed that even external factors, like test setup, can influence system behavior.
- Issues like misalignment in the ring gear were solved by straightforward machining approaches - clamping and drilling carefully on the mill - which reinforced the value of hands-on problem-solving and using the tools well.

6.2 Design Tradeoffs and Impact

Several design tradeoffs ended up influencing our transmission's performance. The original decision to include both bushings and bearings for radial support was made with the intention of increasing stiffness and reducing shaft deflection. However, this over constraint led to frictional losses and wear. Removing the bushings improved performance significantly, highlighting that simpler support strategies can sometimes be more effective if well-executed.

Effort was also put into minimizing misalignment - the radial support on the input shaft helped maintain coaxial rotation with the carrier, and careful assembly of the housing on a granite block using common datums ensured better alignment between the side plates. These efforts likely contributed to the smooth torque and RPM curves observed in testing.

Another key tradeoff was in gear design. Lighter, thinner gears were chosen to reduce inertia, which appeared to pay off. The result was a system that spun up faster and experienced less resistance than expected.

6.3 Limitations and Sources of Error

A few limitations affected our testing and design process. The MATLAB model made conservative assumptions about power losses, particularly in bearings and gear interfaces, which likely caused it to underpredict performance. While the model estimated a final score of 32.2, our actual score during the contest was 76.14. This discrepancy may have resulted from underestimating system efficiency or ignoring certain factors that proved less significant than expected.

There were also physical limitations. Time constraints due to crowded machine schedules limited how much we could iterate on component fits and tolerances. The over constraint from the bushings, initially unnoticed, introduced significant losses during early testing. The test rig's output coupler also introduced uncertainty, as the loose fit occasionally led to inconsistent engagement. These factors may have introduced variability in our measured performance.

6.4 Recommendations for Future Work

Several improvements would be prioritized for a future model. First, more rigorous testing of bushing and bearing configurations would be done earlier in the design process to avoid over constraint and reduce frictional losses. Shaft lengths would also be extended slightly to ensure full engagement with the test rig’s flex couplers, preventing energy losses and improving mechanical robustness.

Further refinements in shaft and bore tolerances could be pursued to reduce play and increase transmission precision. Additional test iterations could help identify and minimize sources of vibration and misalignment. The assembly process could also be made more repeatable with improved jigs and fixturing based on key datums.

Finally, the effects of lubrication (e.g., the amount and type of spindle grease used) could be studied in more detail. Optimal lubrication might further reduce friction, wear, and energy loss - especially during extended runs or under higher loads.

Appendices

A.1 Bill of Materials (BOM)

	MEMBERS	TEAM #	PART #	DESCRIPTION	QTY	INDIVIDUAL COST	SUBTOTAL	TOTAL
MCMASTER-CARR ORDER	Ethan, Pichon Deon, Petrizzo Jason, Kamau James, Muren Jorge, Elias Alexis, Dias	7	9414T6	Shaft Collar	2	$1.85	$3.70	$3.70
							$0.00	
							$0.00	
							$0.00	
							$0.00	
							$0.00	
							$0.00	
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							$0.00	
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							$0.00	

Table 1: *Orders from MCMASTER-CARR*

	MEMBERS	TEAM #	PART #	DESCRIPTION	QTY	INDIVIDUAL COST	SUBTOTAL	TOTAL
SDP_SI ORDER	Ethan, Pichon Deon, Petrizzo Jason, Kamau James, Muren Jorge, Elias Alexis, Dias	7	S1E64Z-048A156	Ring Gear	1	$75.16	$75.16	$209.99
			S1066Z-048A064	Planetary Gears	2	$33.01	$66.02	
			S1066Z-048A028	Sun Gear	1	$23.12	$23.12	
			S9912Y-E2537FS0	Flanged Ball bearings	4	$9.62	$38.48	
			A 7Y55-F2512	Flanged Ball Bearing	1	$7.21	$7.21	
							$0.00	
							$0.00	
							$0.00	
							$0.00	
							$0.00	
							$0.00	

Table 2: *Orders from SDP_SI*

A.2 Full Assembly CAD

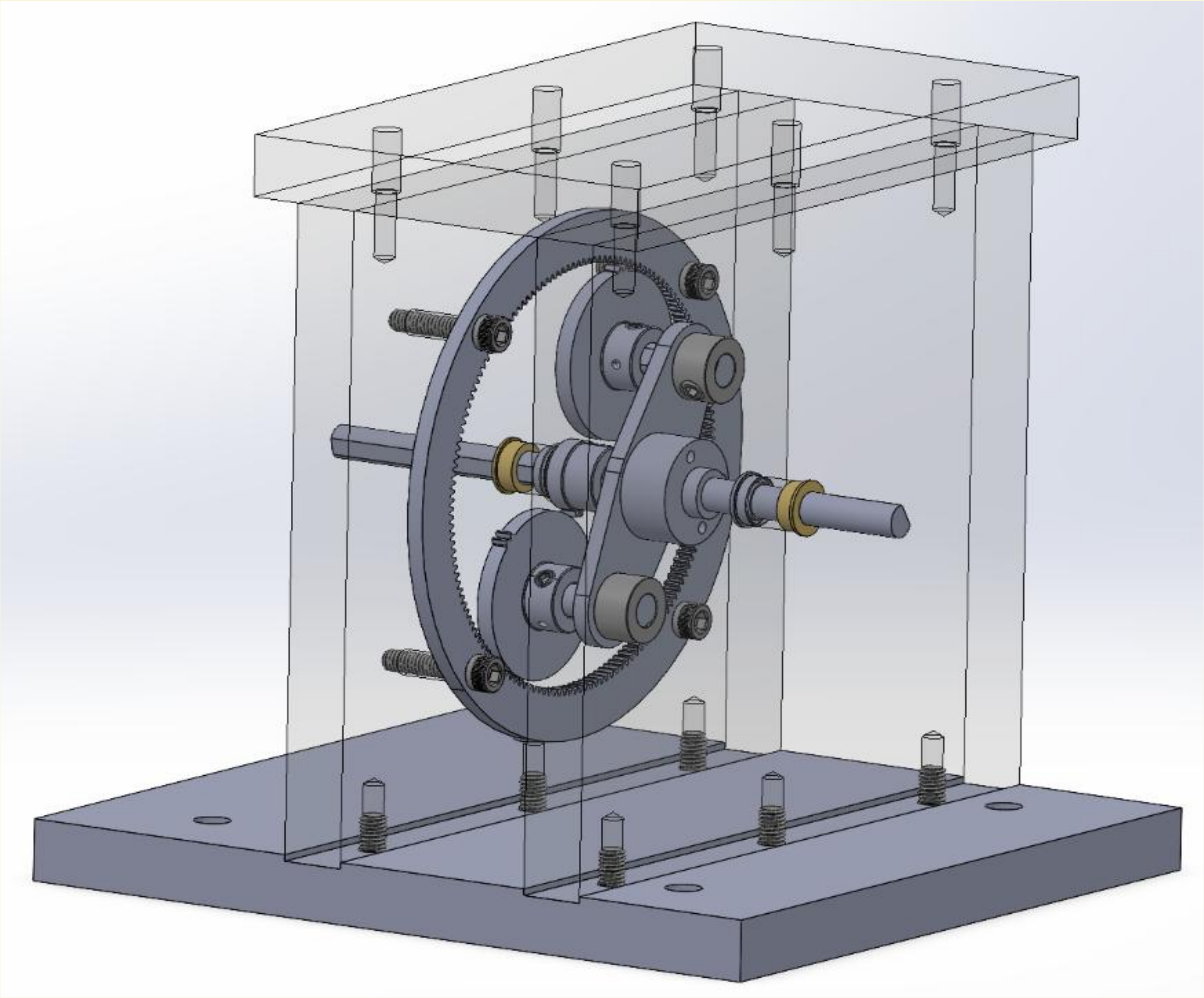

Figure 6: *Full Assembly of the transmission system showcasing the acrylic walls and base, the gears and shafts and the collars, bearings and screws.*

A.3 GD&T

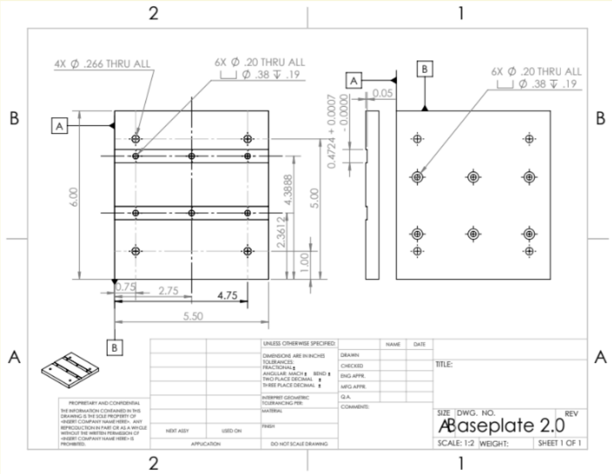

Figure 7: *Base Plate Drawing*

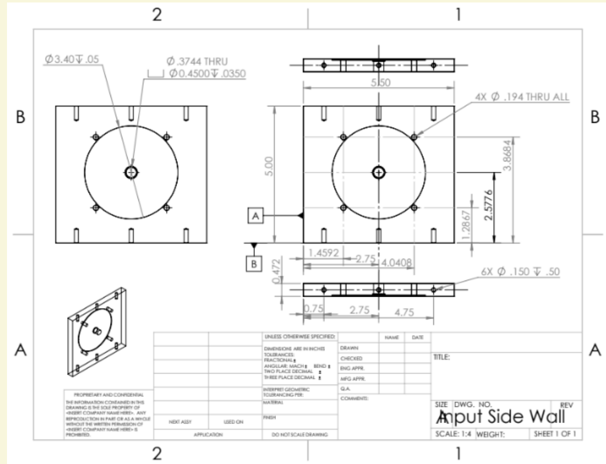

Figure 8: *Input Side Wall Drawing*

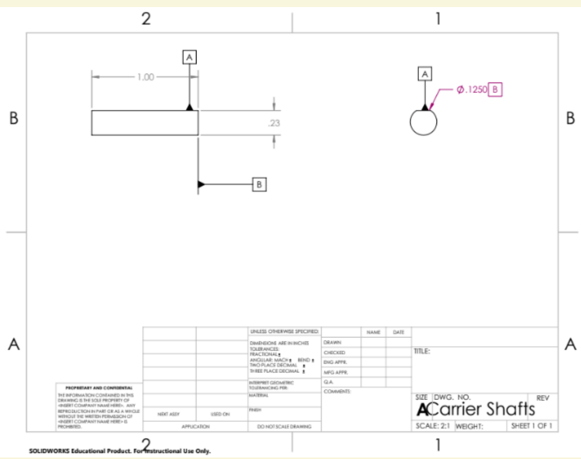

Figure 9: *Carrier Shaft Drawing*

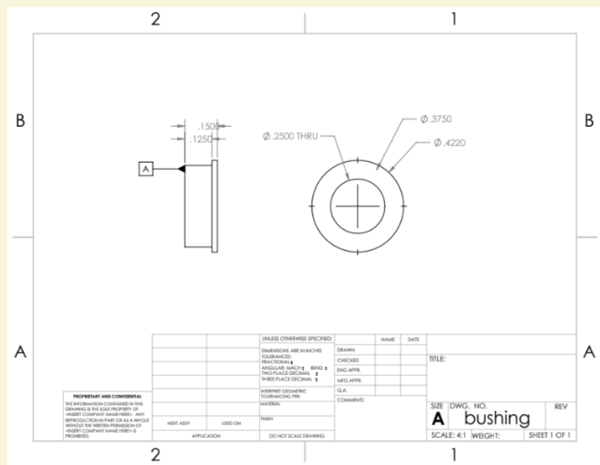

Figure 10: *Bushing Drawing*

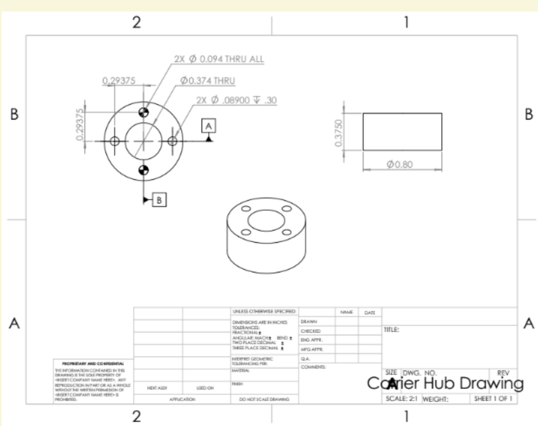

Figure 11: *Carrier Hub Drawing*

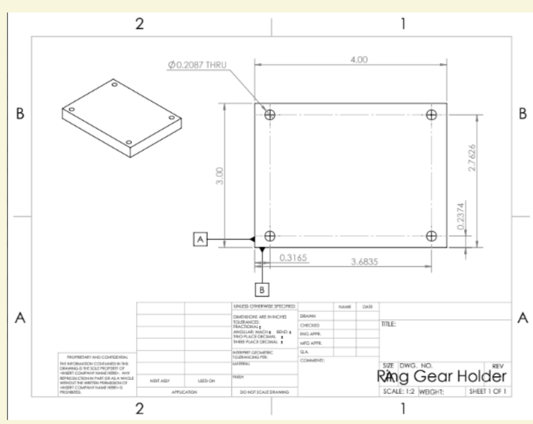

Figure 12: *Ring Gear Holder Drawing*

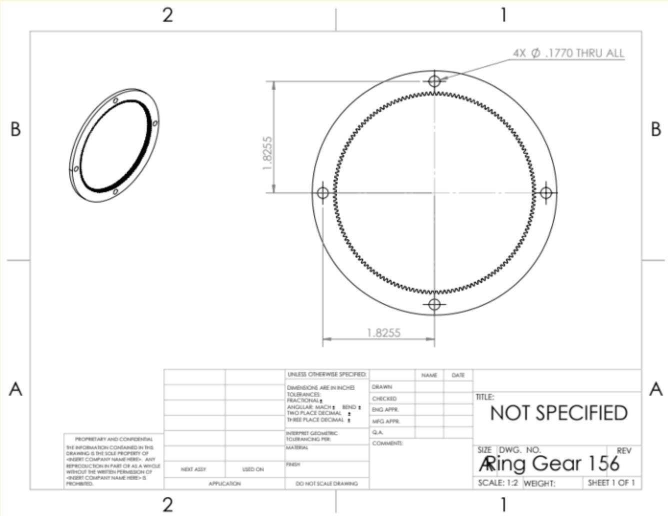

Figure 13: *Ring Gear Drawing(unspecified)*

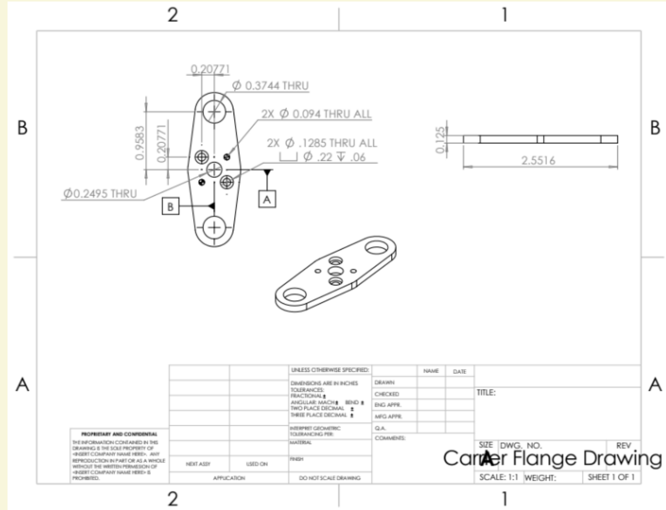

Figure 14: *Carrier Drawing*

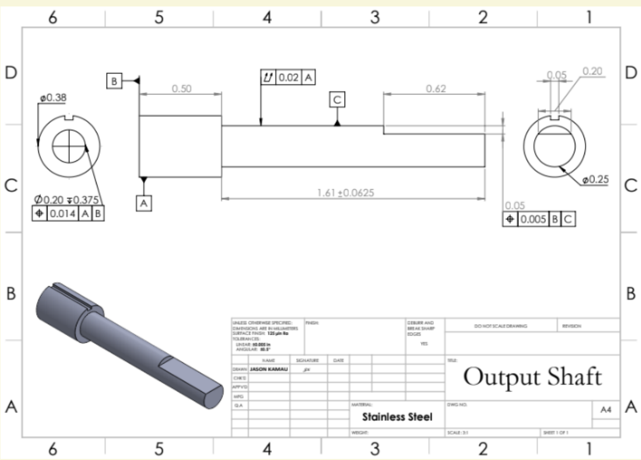

Figure 15: *Output Shaft Drawing*

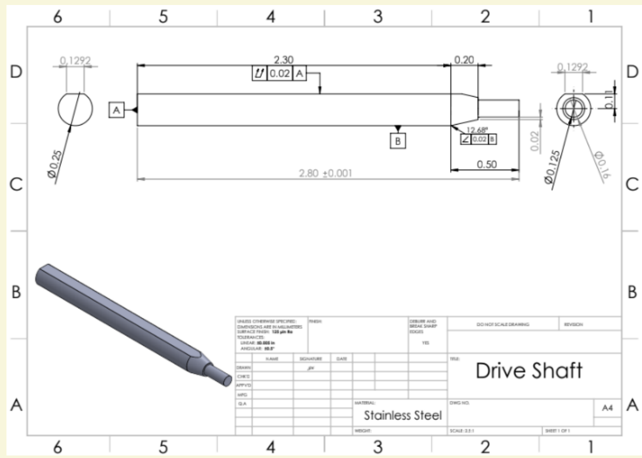

Figure 16: *Input Shaft Drawing*