Design Project

MECH 323 Machine Design (Winter 2024)

Team Number	**33**

Phase Number	**Phase 4**

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1. Overall Gearbox Design

1.1 Housing Design

The gearbox housing is designed to hold and protect the moving components of the gearbox, ensuring proper spacing of the input and output shafts, and gear meshing. The housing consists of two separate parts, each with its CAD drawing seen in Figures 4 and 5. These parts interlock using three dovetail connections on each corner. The larger side panel is shown in Figure 5 features one large cutout between the bushings. The larger plate is 5.5 mm thick to ensure it is strong enough to withstand the stresses present during the event. Additionally, the casing holds the bushing with a counterbore, allowing the shafts to spin freely. The other housing part shown in Figure 4 connects the main side plates and features a cutout in the center to provide clearance for the gears.

1.2 Key Performance Metrics

A single-stage gearbox was chosen for better performance in both top speed and hill climb events, avoiding the drawbacks of a multi-stage gearbox. This choice allows more space for complex shaft geometries and gear shifting. An iterative Python script was developed to determine the optimal gear ratios for both events while meeting functional and dimensional constraints. The hill climbing gear reduction ratio is 2.24, and the top speed gear ratio is 0.61, both using an 8 mm face width for proper gear shifting and accommodating complex shaft geometries.

1.2.1 Hill Climb Gears

Using the 2.24 gear reduction ratio, the hill climb pinion was designed with a module of 1.75 mm, allowing for 15 full depth gear teeth and a pressure angle of 20°. From those parameters, the pitch diameter of the pinion was determined to be 26.25 mm. Similarly, using a module of 1.75 mm and pressure angle of 20°, it was determined that the hill climb gear required 33 full depth gear teeth, and a pitch diameter of 58.75 mm. These gear sizes allow for the motor to run just below the warning torque (2 Nm) at 1.921 Nm and a speed of around 43 RPM. Using the output torque of 1.921 Nm, the vehicle is estimated to be able to climb a distance of 0.5107 m up the ramp.

1.2.2 Top Speed Gears

The gears used for the top speed event were designed to reach top speed within the first 3 m of the track. Using a gear reduction ratio of 0.61, the top speed pinion has a module of 1.25 mm, 43 full depth gear teeth, and a pressure angle of 14.5°. The pitch diameter was determined to be 52.5 mm. Similarly, the top speed gear has 26 gear teeth, with a gear diameter of 32.50 mm. The gear sizes allow the motor to run at a speed of 100.71 rpm at an operating torque of 1.21 Nm, allowing for the car to reach a top speed of 1.24 m/s in 2.49 seconds.

1.2.3 Shaft Design

The gearbox design includes an input shaft receiving torque from the motor, attaching to both hill climb and top speed pinions. The output shaft mirrors the input shaft as seen in Figure 3. Both shafts end has a diameter of 9.53 mm to fit a 3/8th in McMaster-Carr bushing. The shafts were also designed to be hex shafts to prevent gear slippage. The hill climb pinion sits on a 12.70mm hex portion, while the speed test pinion rests on the 9.60 mm portion. Similarly, the hill climb gear sits on the 9.60 mm portion of the output shaft and the speed test gear sits on the 12.70 mm portion of the output shaft. Both shafts feature 17 mm diameter shoulders, each 5 mm thick with the 12.70 mm diameter hex portion acting as a shoulder for the gears sitting on the 9.60 mm portion, reducing print time and stress concentrations. Additionally, 1 mm fillets were added at each diameter change to further reduce stress concentrations in the shafts.

1.3 Significance of Metrics

The four main metrics that have been analyzed throughout all design iterations are operating torque, motor operating speed, top speed/ fastest race time and distance climbed. Maximizing torque at the start of the speed event, and through the hill climb, when it's most crucial will allow the car to preform optimally. The operating speed provides the ability to maximize engine power and reach the highest speed. Balancing torque and operating speed is important to be successful in both events.

Between Phases 3 and 4 There was a change in gear ratio from 0.58 to 0.61 in the speed event and a change of 2.32 to 2.24 in the hill climb. The difference slightly changed the forces experienced in the gearbox and the predicted performance metrics for each event. The summary of the primary metrics can be seen in Table 1 below. For the top speed event, the higher ratio allows for a lower operating torque, decreasing the chance of contact and/or bending fatigue failure on the 3D printed assembly, or complete engine failure.

Table 1: Metric changes throughout the phases.

Predicted Event Performance	Metrics	Phase 1	Phase 2-4
Speed Event	3m Top Speed Time (s)	2.45	2.49
	Top Speed (m/s)	1.25	1.24
	Motor Operating Torque (Nm)	1.28	1.21
	Motor Operating Speed (RPM)	95.3	100.71
Hill Climb Event	Distance (m)	0.5035	0.5107
	Motor Operating Torque (Nm)	1.85	1.921

This change in gear ratio was to allow the gearbox to meet all design constraints. Although this caused adverse effects in the speed event, there was a slight improvement in the hill climb. With the difference of top speed being negligible, it was deemed justifiable in the design process. Using the Python scripts, it was found that keeping the gear ratios reasonably similar to the values from Phase 1, the results would be around 98% similar, assuring that the changes would keep the model valid.

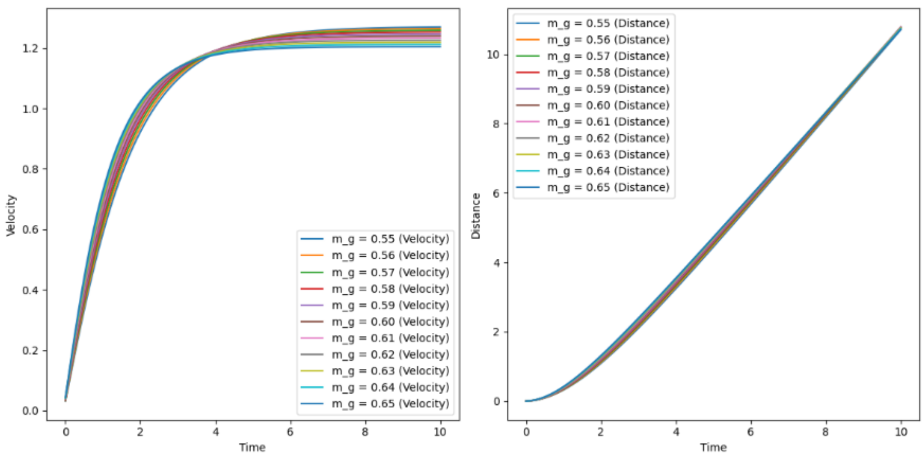


Figure 1: Proof of insignificance of slight gear ratio changes.

The 3D printed material contains different material properties than steel, the small changes in a top speed time or a millimeter improvement on the hill climb is negligible. If this project was properly machined and had the functional capabilities of a real gearbox, then the calculations would be more accurate, although this would be ignoring that fact that they are all very rudimentary and using equations from an introductory level course. For any real predictions to be made, simulations would need to be run to properly analyze the motor similar to the automotive industry.The developed gearbox allows for the gears to mesh smoothly and deliver efficient transmission of motor power through the hex-shafts allowing for optimal performance in both competitions and real-world applications. The fine-tuned design with proven reliability and efficiency allows for the gearbox to offer exceptional performance compared to other designs.

2. Evolution of Gearbox Design

2.1 Phase I

To start a preliminary model of the gearbox was designed to meet the given parameters. First, the max angle the car could reach was calculated, and the required output torque from the gearbox needed to reach that angle was found. The desired torque for the hill climbs event was found to be 4.25Nm. A contingency of 0.05Nm was added to ensure the motor would not stall during the duration of the hill climb test. Python scripts were used to calculate the required gear ratios in the most efficient and accurate manner. The Python script iterates through a comparison of the required output torque and the motor's speed-torque curve. The Python script is then able to determine the required gear ratio, pitch diameter, and number of teeth. Using this information preliminary number of teeth, gear ratios, and pitch diameters.

For the speed challenge, another Python script was written where the code would run through multiple gear ratios to test which would provide the fastest speed. A momentum update was used to see how the car changed position over time using the torque provided by the motor as an accelerating force. Comparing the gear ratios generated by the Python script the most optimal gear ratio was chosen ensuring it fits within the design parameters.

Using the determined gear ratios, further calculations were performed to determine the properties of the gears and shafts needed for the gearbox. Initially in the first phase of the project arbitrary face widths for the gears and shaft holes were determined. Additionally, the distance between the shafts required for the speed and hill climb challenges were marginally different. In phase 1, the shafts were circular and featured keyways to secure the gears and featured arbitrary diameters to match the inner diameters of the gears. Finally, a preliminary CAD model of the gearbox casing was created in SolidWorks featuring 2 different parts. Each part would be printed twice and featured cutouts to lower the weight of the gearbox and shorten the print times. Both parts featured dovetail connections to secure the parts and ensure a stable casing for the gearbox. The gearbox followed all given design parameters, however, the hill climb gear was protruding out of the gearbox by more than the allowable 10mm. Consequently, a corrected gear ratio was determined to ensure the gears would all fit within the design parameters.

2.2 Phase II

Over the duration of Phase II, the gears were redesigned to operate against bending and contact fatigue failure. A theoretical continuous hill climb event lasting 5 years was created. The gear ratio of 2.32 from Phase I was used to calculate the stresses and safety factors under the conditions of the theoretical hill climb test.

It was assumed that a tangential transmitted load of 142.54 N acting on both the pinion and the gear was created and was constant over the duration of the theoretical test. An overload factor of 1.00 was assumed as it was assumed that the motor would act as a uniform power source, and the driven machine would undergo uniform shock on an assumed smooth track. To determine the factor of safety against bending and contact failure, various factors were considered to account for different conditions that otherwise would not have been accounted for using a standard safety factor calculation.

The dynamic factor was determined to be around 1.03, assuming that the motor used in the vehicle was similar to that of a small electric power drill. The size factor was determined to be 1.00 as the size of the pitch diameters of both the gear and pinion are small enough to have no detrimental size effect on the part. The load distribution factor was determined to be 1.6 as the face widths of the gears would not exceed 50 mm, due to size constraints of the project and gearbox assembly. The rim thickness factor was determined to be 1 as the rim thickness of the gears were found to be able to provide full support of the tooth roots. The bending strength geometry factor was taken as a modified form of the Lewis form factor. Considering this gearbox's spur gears with a pressure angle of 20°, assuming full depth teeth, the bending strength geometry factor was determined to be 0.25 for the pinion and 0.36 for the gear of the hill climb system. The stress cycle for bending was determined from the assumption that the gearbox would be running the theoretical hill climb event for 5 years, non-stop at 53 rpm. It was found that the stress cycle factor for bending was 0.97 and 0.99 for the hill climb pinion and gear respectively. It was assumed that the gearbox would run at a constant operating temperature of 100°C, yielding a temperature factor of 1.02. It was also assumed that the gearbox had a reliability of 99%, yielding a reliability factor of 1. As the gears at this point were assumed to be machined from steel, it was assumed that the elastic coefficient of the hill climb pinion and gear was $191\ \sqrt{MPa}$. The surface condition factor accounts for unusually rough surface finishes. Since AGMA does not have a standard for surface finishes, they recommend using a value of 1 for a surface finish with no known detrimental effects. For the hill climb pinion and gear, it was assumed that the parts would be machine finished, with no detrimental surface effects, allowing for the factor to be 1. The geometry factor for contact stress accounts for geometry effects on pitting resistance. It was found that the factor against contact stress was 0.1123 for both hill climb pinion and gear.

Using these factors and a minimum allowable face width of 7 mm, it was found that the safety factor against bending failure was 16.44 for both pinion and gear. Additionally, the safety factor against contact failure was determined to be 1.009 for the pinion and 1.383 for the gear.

To be more conservative, a face width of 8 mm was chosen to be the best face width, allowing for a safety factor against bending failure of 5.378 and 7.860 for the pinion and gear. The safety factors against contact failure were found to be 1.199 and 1.837 for the pinion and gear respectively, higher than the safety factors found using the 7 mm face width. The 8 mm face width also allows for enough room within the gearbox housing to move without interference with other gears and allows for the addition of larger shoulders on both input and output shafts.

2.3 Phase III

2.3.1 Gearbox Housing

During phase 3 the casing received several modifications from the original design in phase 1. First with the new gear and shaft designs, the distance between the shafts had been decreased. Consequently, the gearbox case had to be altered to incorporate this new length. A second potential case side piece was designed to potentially be used in phase 4 shown in Figure 12 in section 5 of the report based off a topology analysis. The results of this topology analysis can be seen in Figure 13 in section 5. The cutout triangles in the face plate resemble the removed material in the topology analysis. This design featured a more overall area of removed material, however due to it have more complex geometry the print time was higher than the original case design. Therefore, the original case design shown in Figure 5 in section 5 was chosen. Furthermore, the thickness of larger casing side pieces with the holes for the shaft was increased by 0.5mm to ensure the casing would not break during testing.

2.3.2 Gears and Pinions

While designing the final CAD to be printed, it was found that the hill climb gear was protruding out of the 10 mm x 10 mm slots at the top of the housing. To ensure adhered to all constraints, a redesign of the hill climb gear system was performed. Due to the differences in the shaft distances, a redesign of the speed test gear system was also redesigned. By measuring the distance between the center of the shaft hole to the edge of the slot, it was found that the diameter of the gear could not exceed 59 mm. Using the previously mentioned Python scripts for both the hill climb and speed events, gear ratios similar to what was used in Phase II were found, with the added constraints that the largest gears could not exceed 59 mm and the distance between the input and output shafts were required to be at the same distance between the hill climb and the top speed systems. The Python script determined that a gear ratio of 2.238, pressure angle of 20°, with a module of 1.75 mm for the hill climb event yields a pinion diameter of 26.25 mm, and a gear diameter of 58.75 mm, just under the 59 mm limit. The Python script also determined that from a gear ratio of 0.61, pressure angle of 14.5°, and a module of 1.25 mm, the top speed pinion has a pitch diameter of 52.50 mm for the pinon and 32.50 mm for the gear. Both gear ratios yielded a shaft distance of 42.50 mm.

To ensure that the new gears would not fail, the calculations from Phase II were reevaluated with the new values of the pinion and gear of the hill climb event. It was found that with the same face width (8 mm), the safety factor against bending failure was 5.378 and 7.86 for the pinion and gear respectively, while the safety factor against contact failure increased to 1.52 for both pinion and gear.

2.3.3 Shafts

The shafts were overhaled to incorporate the new gear design. The diameters of the new input shaft starting from the side opposite the sprocket are:

Table 2: Input Shaft diameter variation points.

Input Shaft	
Location [mm]:	**Diameter [mm]:**
2.5	6.35
5	14.83
17.5	14.83
30	10.6
42.5	10.6
55	6.35
80	6.35

Subsequently the diameters of the new output shaft starting from the side opposite the sprocket are also seen below.

Table 3: Input Shaft diameter variation points.

Output Shaft	
Location [mm]:	Diameter **[mm]**:
0	6.35
5	6.35
17.5	10.6
30	10.6
42.5	14.83
55	14.83
80	6.35

3. Technical Specifications

3.1 Assembly instructions

3.1.1 Parts List

Part	Description	Quantity
Bushing	McMaster-Carr Oil-embedded sleeve bushing, with an outer diameter of 9.5mm and a length of 4.8mm. The bushings were provided by the course.	4
Large Gear Box Housing	Large casing piece feature holes with shoulder to house the bushing. Also features a large cut out to reduce weight and print time. Connects with other casing parts using dovetail connections.	2
Side Gear Box Housing	Smaller casing piece connecting the two larger pieces using dove tail connections. Also features a large cut out in the middle allowing gears to allow for the proper clearance for the gears to protrude through	2

Input Shaft	Receives the input torque from the motor and transfers it to the pinion for both challenges	1
Output Shaft	Receives torque from the gears for both challenges, and outputs it to the sprocket.	1
Hill Climb Pinion	Pinion for the hill climb challenge. Features 34 teeth, and a pitch diameter of 26.25mm	1
Hill Climb Gear	Gear for the hill climb challenge. Features 15 teeth, and a pitch diameter of 58.75mm	1
Speed Challenge Pinion	Pinion for the speed challenge. Features 45 teeth, and a pitch diameter of 55mm	1
Speed Challenge Gear	Gear for the speed challenge. Features 26 teeth, and a pitch diameter of 32.5mm	1

3.1.2 Tools Required

In order to design the gearbox several digital tools were used. Primarily SolidWorks was used to create a successful CAD assembly featuring all the gear box parts. SolidWorks also grants the ability to use built in FEA and Topology tools for future optimization of the gear box. CatalystEX software was used in order to generate an accurate time estimate for 3D printing of all gearbox components as well as estimates using different construction materials. Finally, 3D printers were used to print all gearbox components excluding the bushings.

3.1.3 Brief Assembly Procedure

The assembly of the gear box is fairly straight forward. First, all bushings should be pressed into the holes on the inside of the gearbox casing shown in Figure 5 in section 5. Once all the bushings have been pressed in, the gears must be place on their respective shafts. Ensuring that the gears are properly aligned on the proper shafts and correctly mating the shafts can be inserted into the bushings on one of the case side pieces. Check that all the gears properly align with the hill climb gear and pinion on separate shafts and correctly mating, and the Speed challenge Gear and pinion also on separate shafts properly mating. If all the gears are properly mating slide the other casing side piece with bushing onto the shafts. Finally, connect the other case side pieces shown in Figure 4 in section 5 using the dove tail connections. Check that the gears on the input shaft are able to move allowing the gears to shift for the hill climb and speed challenges. If the gears properly shift the assembly can be attached to the car using the 10mm by 10mm cut aways on the side pieces with the bushings. Then the sprockets can be slid onto the shafts protruding out of the case and the gear box will be fully assembled.

3.2 Technical Specifications Summary

3.2.1 Technical Specifications Summarized

Table 4 below summarizes the technical specifications of the gearbox, showing the gearbox type, estimated lifetime, estimated motor operating torque for each event, lowest safety factor for the pinion and gear, determined gear ratios, print time, and gearbox footprint.

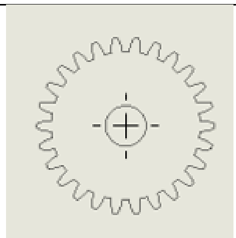
Table 4: Gearbox technical specifications.

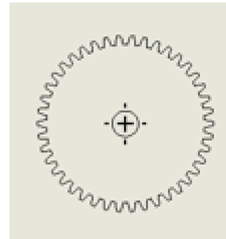
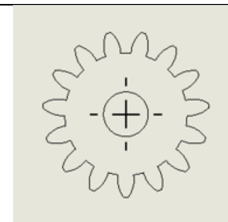
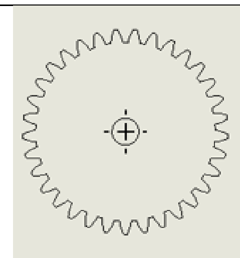
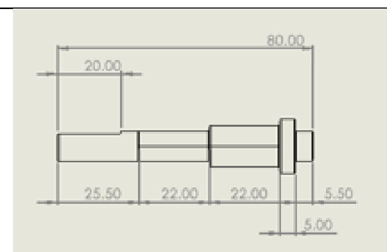
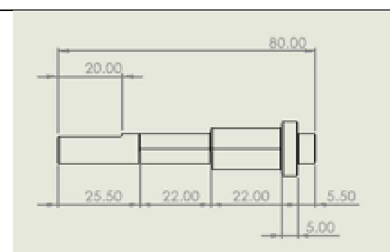
Gearbox Technical Specifications	Value	Units
Gearbox Type	1 stage, shifting	-
Gearbox Design Lifetime	43800	Hours
Motor Operating Torque – Climb	1.921	Nm
Motor Operating Torque – Speed	1.21	Nm
Pinion Lowest Safety Factor	1.52	-
Gear Lowest Safety Factor	1.52	-
Gear Ratio - Climb	2.238:1	-
Gear Ratio - Speed	0.61:1	-
Print Time	6.62	Hours
Gearbox Footprint (L x W x H)	92.50 x 60.00 x 80.00	mm

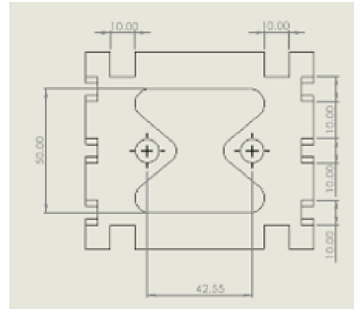
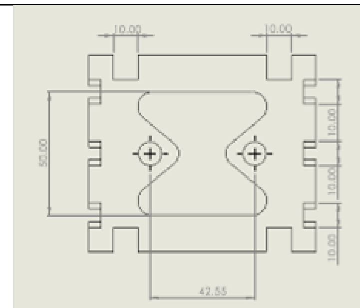
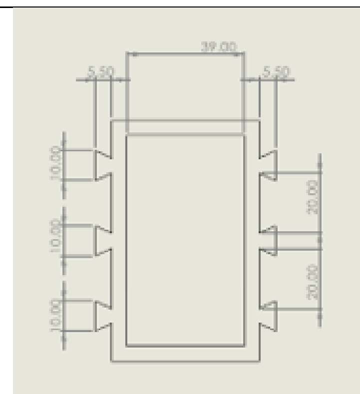
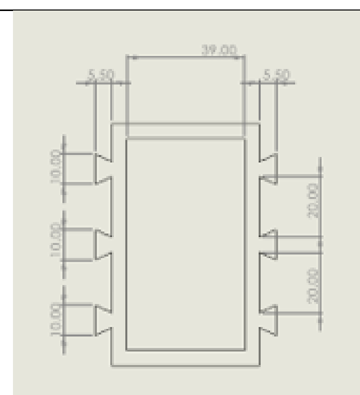
3.2.2 Parts List

A summary of the printed parts can be seen in the table below.

Table 5: Detailed parts List. Note: The gears are not dimensioned in this table, although are in the detailed drawing package.

Part #	Name	Description	Drawing
1	**15 tooth gear**	Hill climb small gear for hill climb.	

2	**33 tooth pinon**	Hill climb small pinon for transmitting torque.	
3	**43 tooth pinon**	Speed test large pinon for transmitting torque.	
4	**26 tooth gear**	Speed large gear for speed test.	
5	**Input Shaft**	For receiving torque from the motor.	
6	**Output Shaft**	For outputting torque to the wheels.	

7	**Left Housing**	Left side gearbox housing.	
8	**Right Housing**	Right side gearbox housing.	
9	**Front Housing**	Front side gearbox housing.	
10	**Back Housing**	Back side gearbox housing.	

4. Future work

A main focus for future improvement would be decreasing the amount of material used in the larger casing side pieces with the bushings. Utilizing a topology analysis of the structure would be the most efficient method of reducing excess material. In order to perform a topology analysis, the expected loads from each shaft, dovetail connections, and connections to the car would have to be estimated. Then by setting the proper constraints and using these estimated loads the topology analysis could be completed on the casing in its basic form without any unnecessary cutouts. The topology analysis would provide data on where to remove martial without sacrificing the structural integrity of the casing. However, the raw topology data would most likely require minor alterations to improve the manufacturing feasibility. The largest difficulty in performing a topology analysis is the processing power required to complete the analysis. A more powerful computer would be required to efficiently perform the topology analysis. By decreasing the amount of material used the print times could be lowered, allowing for more efficient manufacturing of the gearbox. Additionally, by removing unnecessary material the overall design will be lighter resulting in increased performance. Similarly, cut outs could have been added to the speed test pinion and hill climb gear. Both these gears are large enough to incorporate a cut out without sacrificing the structural integrity of the gear.

In terms of the vehicle assembly, a large improvement that could have been better optimized is the weight distribution in the vehicle. Although this vehicle had the gearbox printed its relative mass was not as meaningful, and for this project the emphasis on weight reduction was to improve print time. If the system's mass was considerable, one of the best things that could have been done to improve vehicle performance would have been to lower the center of mass and bring it slightly towards the rear of the vehicle. The reasoning for this is that lowering the center of mass helps to control stability, improve handling, and streamline the aerodynamics of the vehicle. With this change, less emphasis would have been needed on the torque requirement for both events and a higher operating speed could have been used. Similarly, moving the center of mass slightly back in a rear wheel car would allow for the wheels to generate slightly better torque. By taking advantage of the physics behind this, more raw speed and power can be made. This is seen in many applications, especially in racing. All racing vehicles are very close to the ground, and some even use a spoiler to create downforce. Using similar principles this could have been met. As mentioned, an improvement that can be made in a vehicle is aerodynamics. Although we had no control of the chassis and body of the vehicle, there were no real changes that could have been made, but making the vehicle less boxy and more streamlined would have been beneficial in not only both events but in long term motor efficiency.

Improvements can be made past just efficiency, by ensuring reliability and longevity of the gearbox. Making sure that it can withstand internal and external stresses along with thermal and fatigue effects further than just preliminary calculations makes the study much more credible. This can be done using software utilizing finite element analysis such as ANSYS. This much more comprehensively analyzes the gearbox much more accurately than any hand calculation can do. It can also much better simulate thousands of different scenarios and provide a much clearer understanding of risk factors.

Finally, throughout all four phases there was an absence of proper prototyping. There will never be a real gearbox made for production that is not tested in the prototype phase. Being able to prototype different versions to not only compare different designs, but to confirm the calculations of theoretical expectations is just as important if not more than taking the time to properly analyze the system. If this could be done in future this would yield much more precise results.

5. Drawing Package

5.1 Assembly View

Below is an overview of the gearbox assembly. The drawing shows the assembly from a side, top and trimetric view to show the project as a whole. Further views including exploded, and parts views, can be found in the following sections.

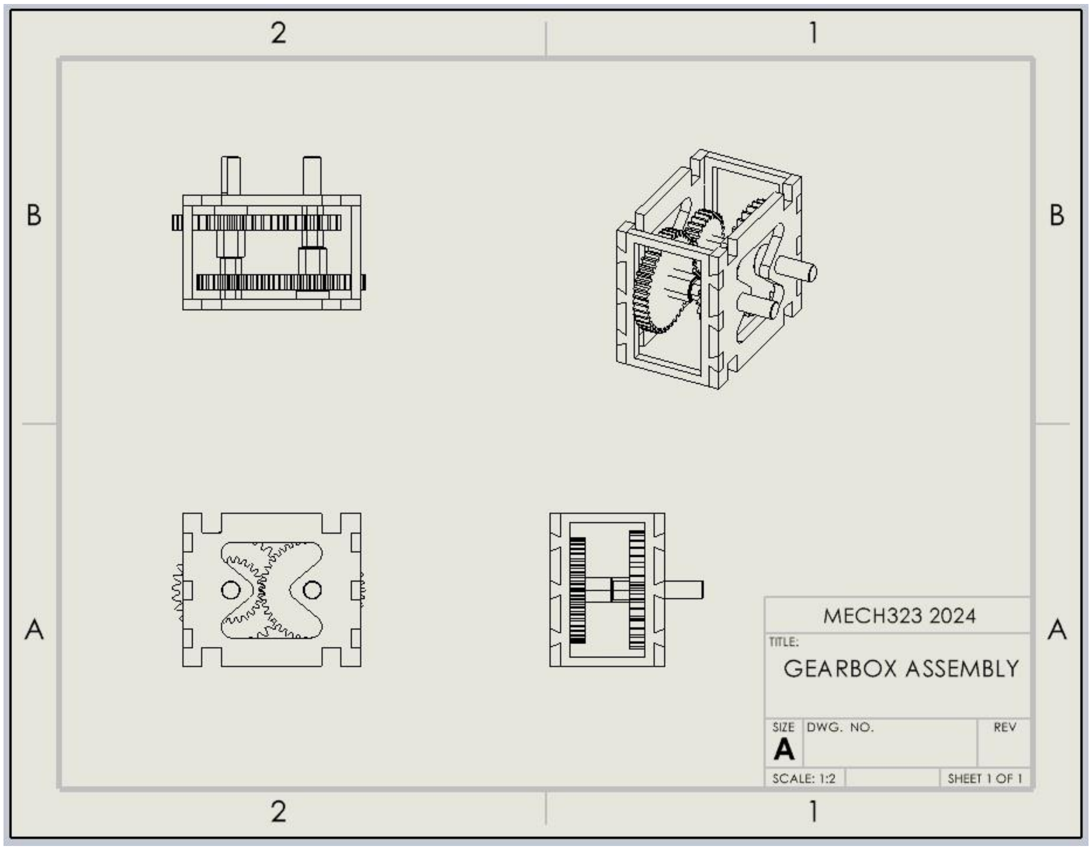


Figure 2: CAD drawing of the gearbox assembly from top, side, and trimetric views.

5.2 Exploded View and Parts List

The exploded view always for a better understanding of the inner composition of the gearbox assembly. It is also accompanied with a bill of materials which specifies the name and placement of each component.

ITEM NO.	PART NAME	DESCRIPTION	QTY.
1	Side Casing	Side parts of the gearbox casing	2
2	Front and Rear Casing	Front and rear parts of the gearbox casing	2
3	Input Shaft	Shaft receiving input torque from motor	1
4	Output Shaft	Shaft outputing torque	1
5	Bushings	Oil-Embedded Bronze Sleeve Bearing	4
6	Speed Gear	Gear component of speed gear pair	1
7	Speed Pinion	Pinion component of speed gear pair	1
8	Hill Climb Gear	Gear component of hill climb pair	1
9	Hill Climb Pinion	Pinion component of hill climb pair	1

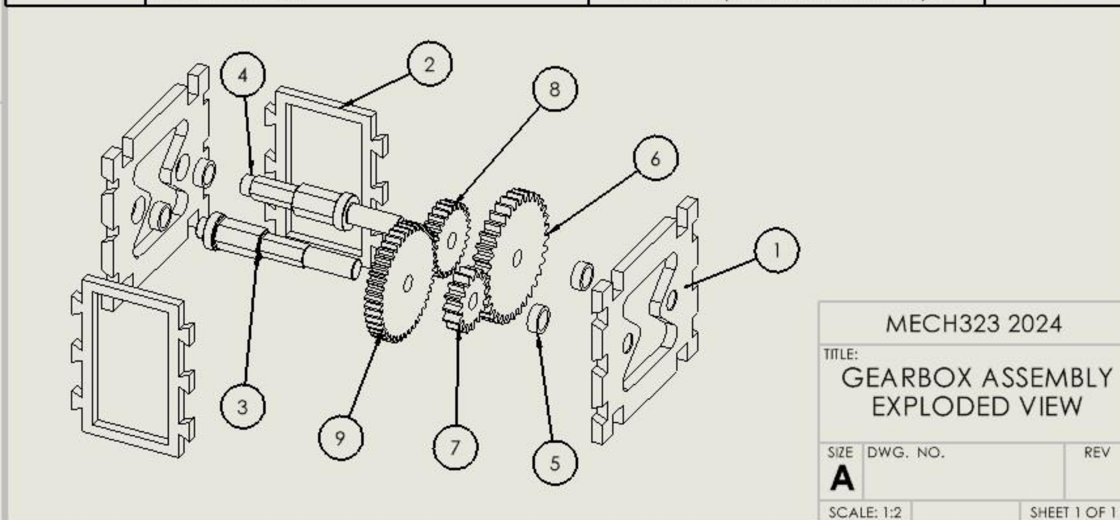


Figure 3: CAD drawing of the gearbox assembly from an exploded view including bill of materials.

5.3 Detailed Parts Drawings

Below can be found the drawings for every component used in the gearbox assembly. They each consist of top front and isometric views so that a better understanding of dimensions of each part is achieved.

Front/Rear Casing

The drawing below shows the component which was used for both the front and rear parts of the gearbox casing. Details to note are the dove-tail extrusion around the edge of the casing part which fits into its sister component.

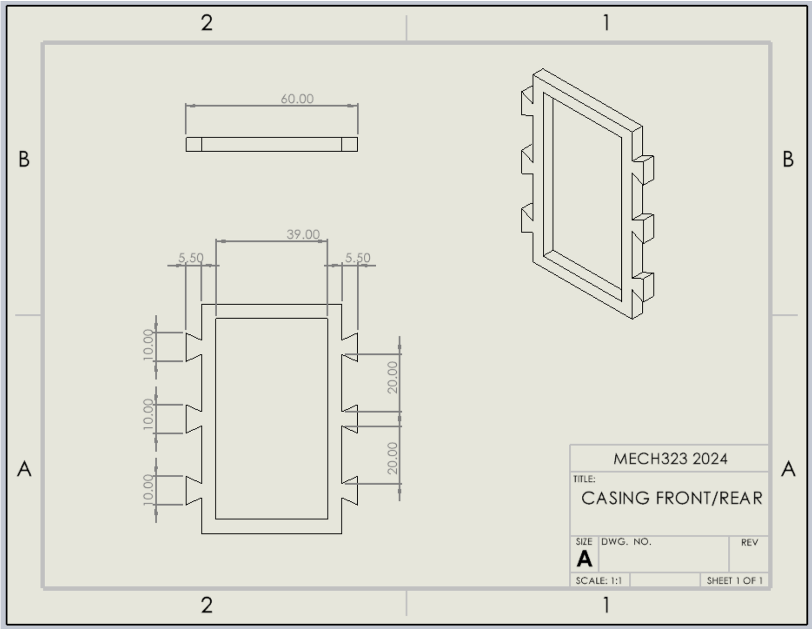


Figure 4: CAD drawing of the side component of the gearbox casing.

Side Casing

The side component of the gearbox casing is shown in the drawing below. The dove-tail groves were made to fit into the front/rear casing parts.

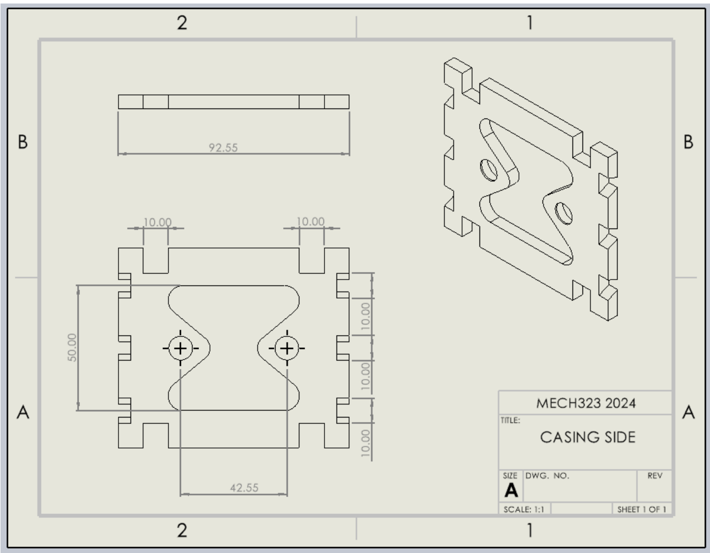


Figure 5: CAD drawing of the side component of the gearbox casing.

Hill Climb Gear Pairing

The two-stage gear has a gear pairing dedicated to the hill climb trial. Drawings of the pinion for that pairing can be seen below.

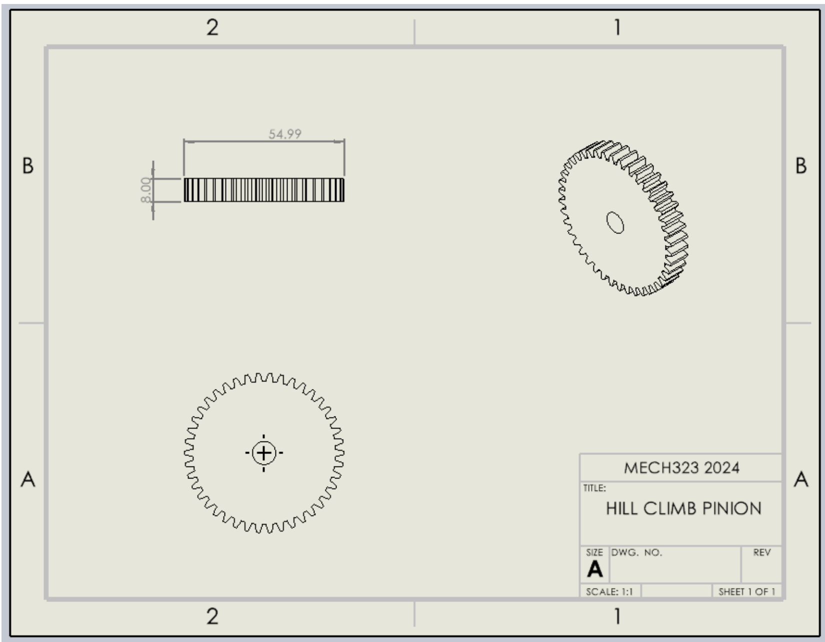


Figure 6: CAD drawing of the pinion component of the hill climb gear pairing.

The gear component of the pairing which is linked to the output shaft is shown in the figure below.

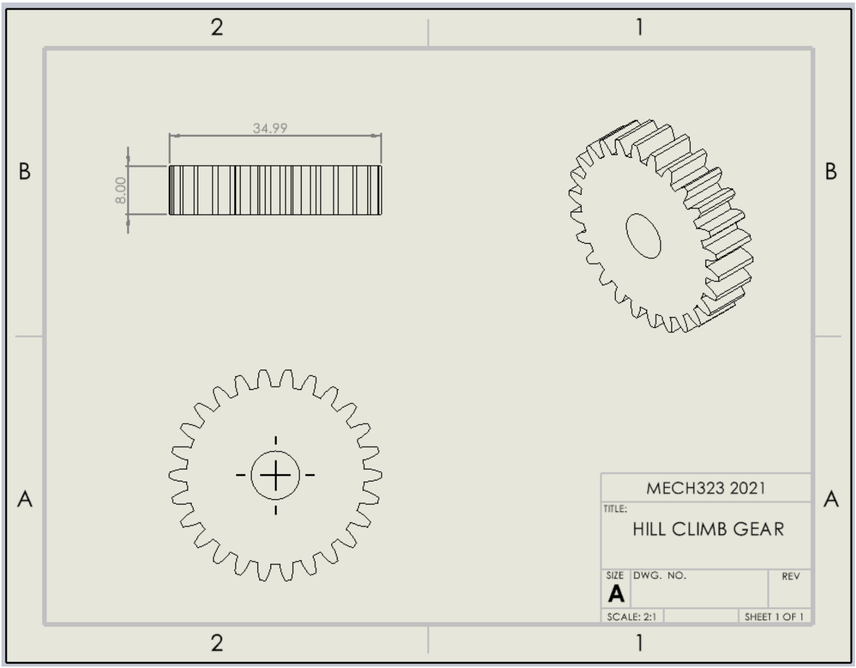


Figure 7: CAD drawing of the gear component of the hill climb gear pairing.

Input and Output Shafts

Both shafts were designed accounting for the different inner bore diameters of each gear pairing, as well as gear face widths. Details to note are the shoulders at the extremities of the shaft, the hexagonal shaped body, the keyways, and filets at each edge. Below is a drawing of the input shaft receiving torque from the motor.

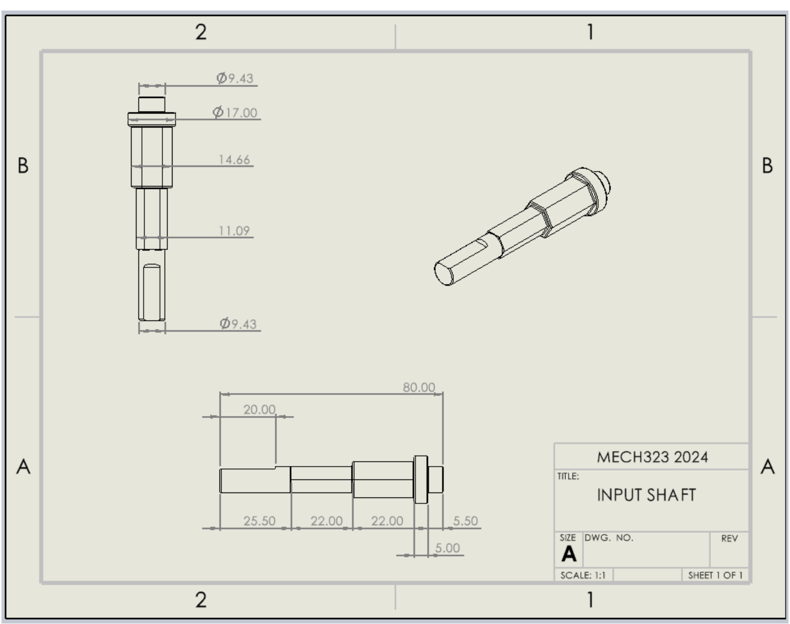


Figure 8: CAD drawing of the input shaft.

The other shaft seen below is the output shaft. It also shares similar features as the input shaft but slightly offset to account for the different positions of the gear inside the casing.

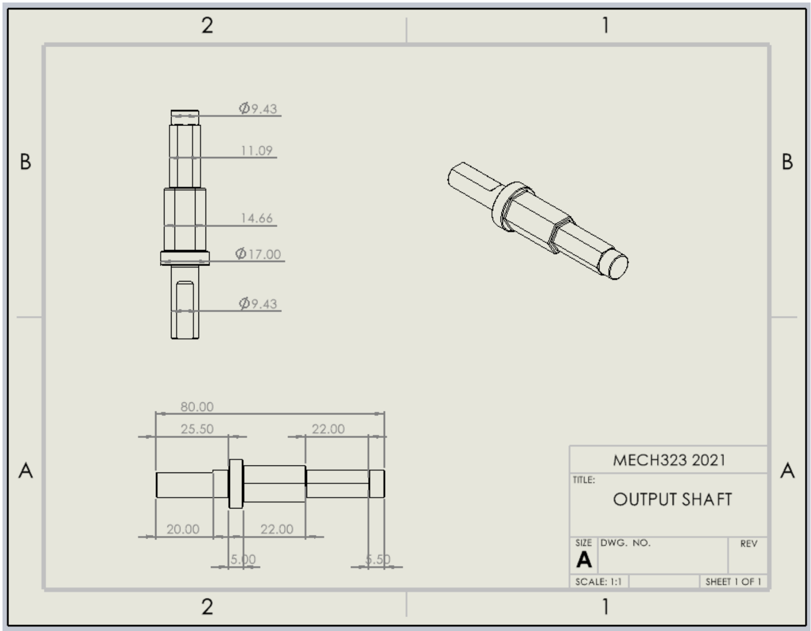


Figure 9: CAD drawing of the output shaft.

Speed Gear Pairing

Similarly to the hill climb pairing, the speed pair was designed for the speed trial the vehicle will face. Below is a drawing of the gear component of the speed pairing.

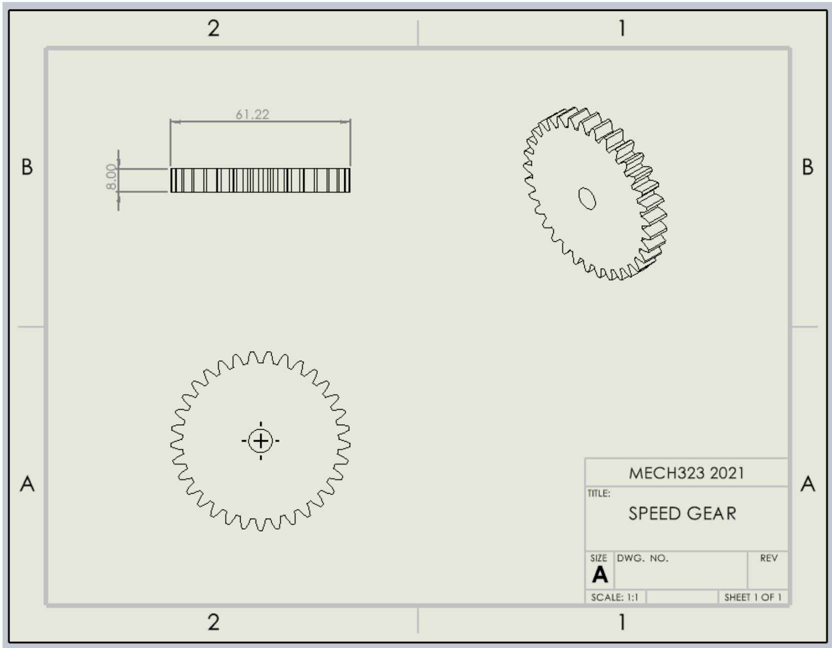


Figure 10: CAD drawing of the gear component of the speed gear pairing.

The pinion component of the speed gear pairing can be seen below. The combination of these two gears will allow for the fastest gear set of the two pairings in the assembly.

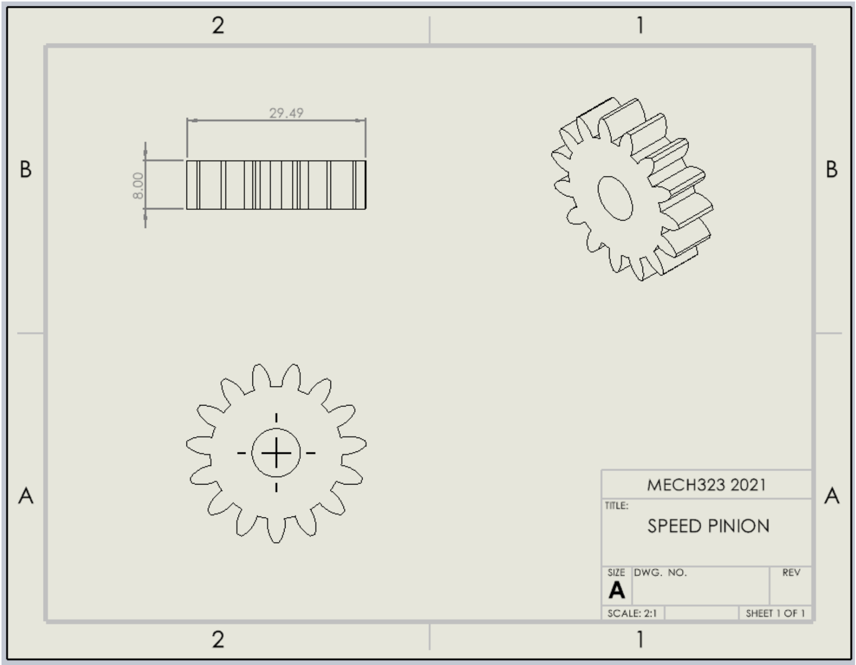


Figure 11: Pinion component of the speed gear pairing.

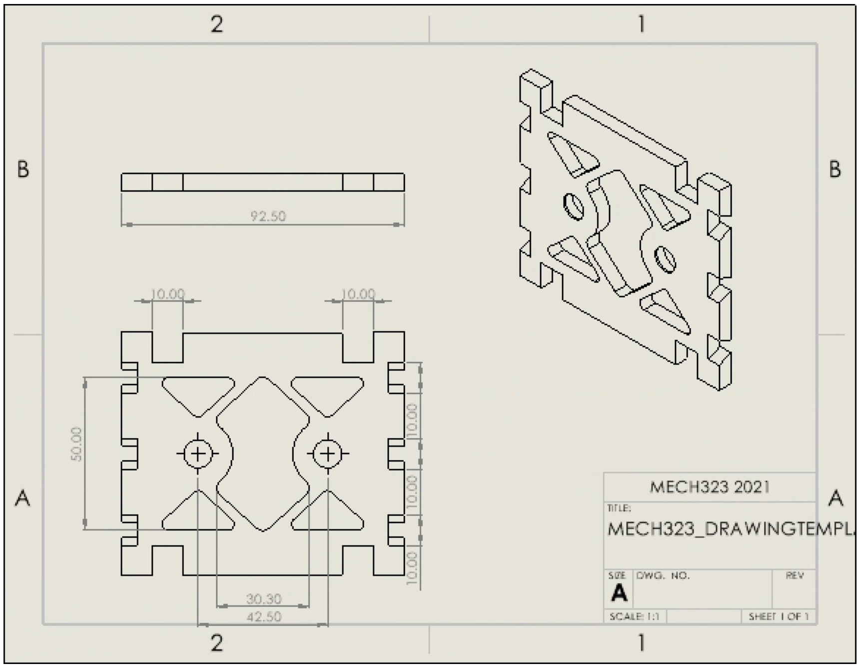


Figure 12: Unused Casing Design based off Topology Analysis

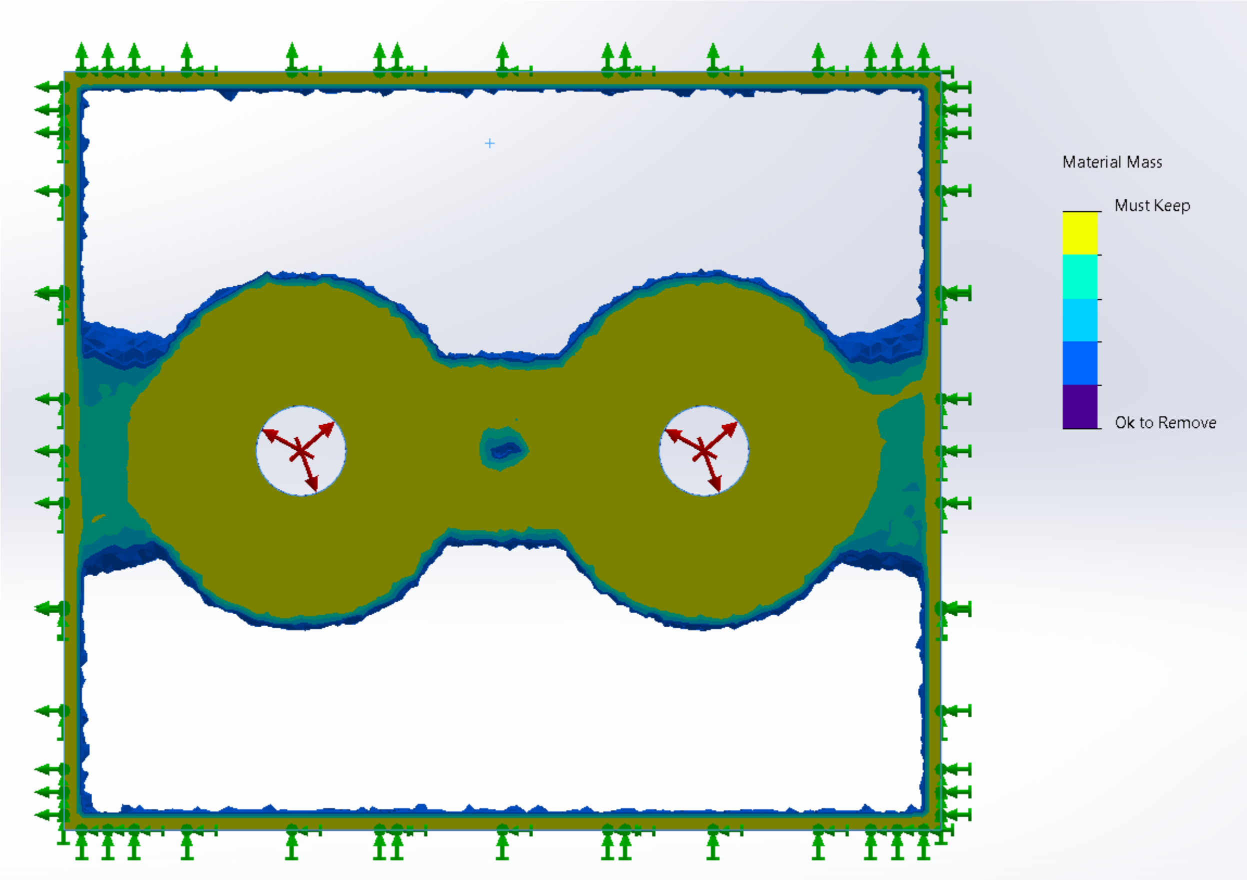


Figure 13: Topology Analysis of Gearbox Casing

5.4 Catalyst Print Time

Using a BST printing case design, a predicted print time of 6 Hours 39 Minutes was estimated using the CatalystEX software in AppsAnywhere. The detailed summary page is seen below.

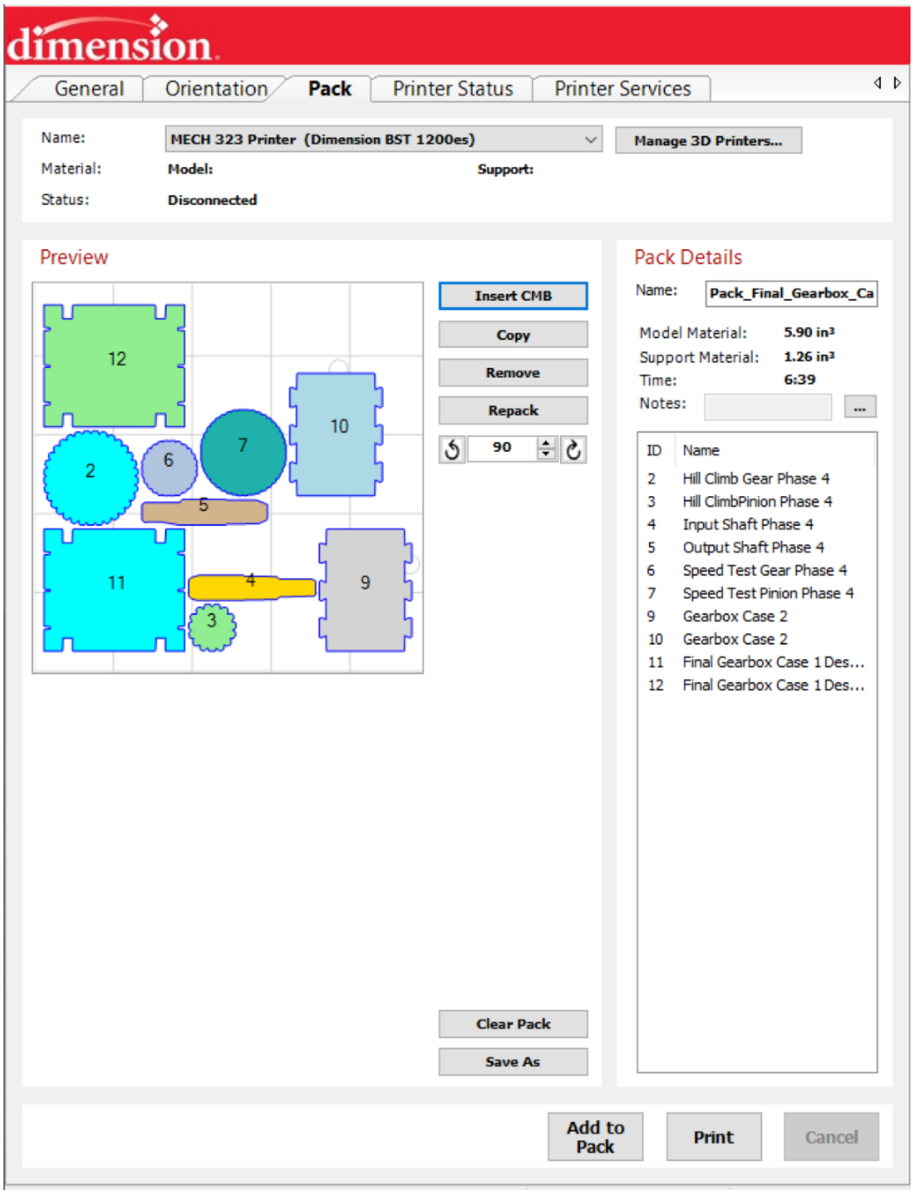


Figure 14: Shortest CatalystEX print time design summary screen.

Phase 4 Summary Page

(Attach to the end of your Phase 4 Report on a separate page)

Team Number	**33**

Phase Number	**Phase 4**

Global Design Characteristics		
Gear Box Parameters	**Number of Stages**	1
	Speed Gear Ratio	0.61
	Hill Climb Gear Ratio	2.238
Vehicle Weights	**Total Number of Weights**	0

Predicted Event Performance		
Speed Event	**3m Top Speed Zone Time (s)**	2.49
	Top Speed (m/s)	1.24
	Motor Operating Torque (Nm)	1.21
	Motor Operating Speed (rpm)	100.71
Hill Climb Event	**Distance (m)**	0.5107