Sustainable Energy Group Design Project

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1 Abstract

In this report, I detail how I built the differential and housing for the turbine rotors, which achieve the following:

- Allow blade pair rotation to be translated perpendicular to the generator.
- Multiply the rotational speed of the blade pair in a 1:4 ratio with the generator.
- Houses all rotating components, protecting them from impacts and allowing them to spin freely using embedded bearings.

This report also includes the design and justifications behind the shafts used to rotate the rotors and generators, which was a shared effort.

2 Introduction

My contribution to the project involved the design, calculation, and manufacturing of the differential and the main housing. I also helped with the selection of the required shafts for our generator and rotors.

These were critical parts that bridged the blades to the generators powering the lightbulb, multiplied the speed required for the generator's specific parameters, and provided the structure which held them together. The design of the differential was largely based on the required parameters of the generator to power the lightbulb at 6V and the rotational speed of the blades obtained from Q-Blade simulations.

3 Design Methodology and Simulation

3.1 Requirements

The need for a differential arose from the brief specifying the need for a pair of independently rotating rotor shafts. The speed produced by the turbines alone would not be enough to power the generator to the required 1.8W to power the lightbulb. Thus a gearbox was required to produce a reasonable generator shaft speed.
From the simulations in QBlade and calculations done by Yahya and Callum, we estimated the blades to rotate between $100 - 500$ RPM. Furthermore, an estimate produced by Dudu working on the generator, places the speed required by the generator's rotors to be at $750 - 1000$RPM. Taking an average of these two in Equation 7.2.1 gives us the average input speed as 300 RPM.

3.2 Differential

Using the gear ratio relations, I determined the required gear ratio to be 1:3, however, 1:4 was chosen to overcompensate for any reasonable uncertainty in the average of the speed calculations. The ratio was calculated using Equation7.2.1, and I went with a module of 2mm, as this gives us a larger diameter of the pitch circle of 96mm, which gives us good tolerances from the 3D printer at this size. Spider gears had a ratio of 1:1 with axle gears and had 20 teeth with the same module of 2mm. All gears had a pressure angle of 20 degrees, as this ensures efficient transmission but results in a slightly higher radial force component, which may cause stress on the teeth. I also ensured to remain within the recommended gear ratios for bevel gears, ranging from 1:1 to 1:8 in extreme cases [1]. Since the gears are made of PLA, I also took into consideration that higher gear ratios would increase stress on the teeth, so I decided to stay on the safe side with a lower transmission ratio. The alternative would be to make a two-stage gearbox, considerably increasing costs and weight. An open differential, such as the most typical kind of differential in automobile powertrains, suited the design very well, and this is what would be implemented. It was able to take two inputs rotating at different torques and speeds and transmit it to one generator shaft.

3.3 Shafts and Bearings

The most cost effective option for the shafts is using the threaded rod. The shafts are made of stainless steel and cost half as much as the stainless steel bar. There are issues with this, as the major diameter of the threaded rod is anywhere up to 0.25mm less than the diameter of the bearing , according to the ISO Metric Thread dimensions, which are standard in the UK. Along with the 626-2RS 6mm I.D. bearings, this would make a H7/f6 clearance fit. Ball bearings were selected as we assume that there is no axial loading, only radial load, so ball-bearings are suited to provide low friction to our shaft. The bearings are housed in a 2-piece laser cut and 3D printed housing that is fixed to the differential casing with 4 bolts on a circular flange shown in Figure 7.3.3.

The shafts were simulated in Fusion 360, performing a 3-point bending test with rotational load on a stainless steel shaft, 130mm long and a nominal diameter of 6mm. I calculated the torque the shaft experiences upon start-up, assuming the start up to a constant speed of 300RPM takes 2 seconds. From Equation 7.2.5, I found the maximum torsional stress experienced by the shaft to be 14.97MPa. Comparing this to the critical shear stress equation, based on the Tresca Yield Criterion [2] and calculated from Equation 7.2.6, we have a safety factor of 5.67. Conservative safety factors are around 3, so this shaft is suitable for these conditions. These dimensions were taken from the CAD drawings that I made on OnShape for the whole differential assembly. The test results, as in Figures 7.3.1, 7.3.2 shows a safety factor of 15 with a maximum displacement of 0.015mm under a rotation of 300RPM. This verifies the ability of the shaft to withstand normal operating conditions.

3.4 Housing

The housing was worked on by myself and Maciek and. Originally, we were making it out of 1.5mm thick aluminium sheet, as it provides ample stiffness for its density, and we prioritised stiffness rather than strength in this design, as a small displacement in the casing would misalign the gears and potentially lose output. It can be hand machined using the facilities in G20, being made of sheet metal. The dimensions were obtained from the CAD drawing I made, shaped to fit the differential and to absorb any impacts and disperse it away from the gears in the durability test. I ran an FEA test in Fusion, shown in Figure 7.3.6 and the maximum stress experienced (0.031 MPa) for a small element of the housing is well under aluminium's yield stress, 30-500 MPa, according to ANSYS Granta, used in Table 7.4.1 [3], the material selection software of choice. The FEA analysis also outputted a safety factor of 15, which was more than enough for this turbine.

4 Build Phase

The differential gears were created in the first session, using the 3D printing facilities in G20. The design of the gear had to be changed to thicken the arms supporting the spider gears, as they broke with an insignificant force being applied to them. It was reprinted in the following week and it seemed to be a sturdy support. The direction of the print meant that the weakest points were at the arms under any lateral loads being applied. I made sure to increase the perimeters of the print which acts as a solid shell, making the teeth and the arms much stronger than the previous print. This is more effective than simply increasing the infil, both for saving filament and for strength. It was also changed to be able to fit the bearing by press-fitting it and then securing it using two laser-cut acrylic disks, as seen in Figure 7.3.4 and Figure 7.3.5

The casing was all prepared in the third session, shown in Figure 7.3.8, where I marked out a rectangular net onto the aluminium sheet, following the dimensions I could easily grab from my CAD model. I then cut it to shape using the metal guillotine, which required very minimal post-processing, other than deburring the edges. The holes for the bearings were scribed and punched by hand and then cut using the pillar drill. It was then folded using the metal folding machine. We later created corner supports using laser-cut plywood with interlocking finger joints to hold the shape of the casing. These supports were cut and bolted into the aluminium by Maciek and held the shape of the casing together.

I spent the next weeks cutting the shafts to length and creating the bearing housing assembly , in Figures 7.3.9, 7.3.7. Originally, I wanted to fix the gears to the shaft using Nylock nuts on the threaded rod, but I had failed to consider spatial constraints that were not visible on our CAD model. I was able to thread the 3D-printed gears and use epoxy resin to lock the bearings into place effectively. In order to keep the gears meshing, it was necessary to keep them in place axially, and this was done using two nuts shearing into each other on either side of the casing. This prevented the nuts from moving in either direction while the shaft rotated, as they were shearing into each other with a high amount of friction. This also helped with the issue of the shafts rotating at an angle, due to the blades weighing the shaft down, but this was still an issue. The blades were also locked onto the shaft using two opposing nuts and washers on either end of the blade pairs. This allowed us to easily take off the blades in the test, as they were the largest components. I spent the final week assembling the differential and attaching the generator to the shaft on which the pinion gear is mounted ,shown in Figure 7.3.10.

5 Results, Analysis and Discussion

Before the test, we predicted that the shaft misalignment would cause periodic misalignment of the differential gears. This would cause problems with the generator as it would disconnect from the differential and stop spinning. The loss of power transmission is not the problem, as the generator's inertia would keep it spinning, but it may cause the teeth to not mesh, which would cause problems, as the gears could be damaged at that speed. We saw this happening when we spun the generator using a drill. In future, I think we should consider using solid steel shafts, which would produce a tighter clearance fit with the bearing and prevent this eccentric spinning.

In the test, the blades did not spin, and we would attribute this to the weight of the generators. The blades required a large initial torque, due to the loads on the gears, to spin them up to the steady state. We predict that it would have worked after the initial spin-up and generated a higher voltage than we needed. In this regard, we would have reduced the size of the discs on the generator, and selected a different material to 3mm steel.

The drop test resulted in the fracturing of the differential spider gear support arms. The blades then spun as the differential was not connected to the generator, which verifies that the weight of the generator was the issue. Needless to say, the differential did not serve its purpose, and the turbine failed to generate power. The failure could be attributed to a crack that propagated between the print layers, which were the weakest point of the gears.

This is a unique property of 3D printed parts, which extrude material layer by layer and utilise the low melting point to create an interface 'weld' by partially remelting the layers. The strength of this bond is weaker than the material properties of polylactic acid (PLA), and can be strengthened by several means. Researchers have found that annealing a print below its crystallisation temperature reduces thermal stresses between layers, increasing the overall tensile strength of the material [4], which can be done by using a heat gun or a plastics oven. This interface can also be strengthened by reducing the travel speed of the extruder, decreasing layer height, and increasing extruder nozzle temperature. However, it is important to note that although changing these parameters can increase tensile strength by 20%, it still lies around 60MPa, which is relatively low and likely would not have survived on its own.

To remedy this, a stronger design of the supporting arms would be required. To reduce stresses at the base of the arms, I would have printed it as a separate component and used fasteners to attach it to the worm gear, likely two bolts, each with Loctite. This way, any lateral stresses are experienced by the fastener against the ring gear itself. Since the interface is now against the grain of the filament, the material is a lot stronger in this direction, and the print would have a better chance of surviving the drop test. I could have also chosen a more suitable material, such as PETG, which has better shock absorption properties due to its ability to deform [5]. However, the design change is needed in either case, as layer adhesion is an inherent property of all FFF 3D printed parts.

6 Conclusion

In conclusion, there were several changes we would make to our design to make the turbine work, including the generator and the shafts. To make sure the differential survived the crash test, I would make changes to the design so that the spider arms are an external component that is secured using a pair of fasteners. I would also make changes to the differential housing and shaft to prevent any misalignment of the gears and produce better meshing and, therefore, a more consistent output generator.

7 Appendix

7.1 References

[1] B. T. Concepts, “What kind of gears should i use?,” 2004.

[2] ScienceDirect, “Tresca criterion,” 2025. Accessed: 2025-04-01.

[3] M. Ashby, *Material Property Data for Engineering Materials.* ANSYS, Inc., 5th ed., 2021.

[4] T. Yang, Y. Chen, Y. Wang, Q. Liu, N. Zhao, X. Du, and J. Xu, “Influence of porosity, crystallinity, and interlayer adhesion on tensile strength of 3d-printed polylactic acid (pla),” *Rapid Prototyping Journal*, vol. 27, no. 1, pp. 207–216, 2021.

[5] All3DP, “PETG vs PLA: 3D Printing Filaments Compared,” 2025. Accessed on: 2025-03-31.

7.2 Equations

$$Input = \frac{100+500}{2} = 300RPM, \quad Output = \frac{750+1000}{2} = 875RPM \tag{7.2.1}$$

$$\frac{N_1}{N_2} = \frac{T_2}{T_1} \quad \text{where} \quad N = \text{Speed of the gear,}, \quad T = \text{Number of teeth,} \tag{7.2.2}$$

$$T = \frac{1}{2}mr^2 \cdot \frac{\Delta\omega}{\Delta t} = \frac{1}{2}(0.150\,\text{kg})(0.006\,\text{m})^2 \cdot \frac{31.4\,\text{rad/s}}{2\,\text{s}} = 5.08 \times 10^{-6}\,\text{Nm} = 5.08\,\mu\text{Nm} \tag{7.2.3}$$

$$J = \frac{\pi d^4}{32} = \frac{\pi(0.012)^4}{32} = 5.44 \times 10^{-9}\,\text{m}^4 \tag{7.2.4}$$

$$\tau = \frac{T \cdot r}{J} = \frac{5.08 \times 10^{-6} \times 0.006}{5.44 \times 10^{-9}} = 14.97\,\text{MPa} \tag{7.2.5}$$

$$\tau_{\text{max}} = \frac{\sigma_{\text{yield}}}{2} = \frac{170}{2} = 85MPa \tag{7.2.6}$$

7.3 Figures

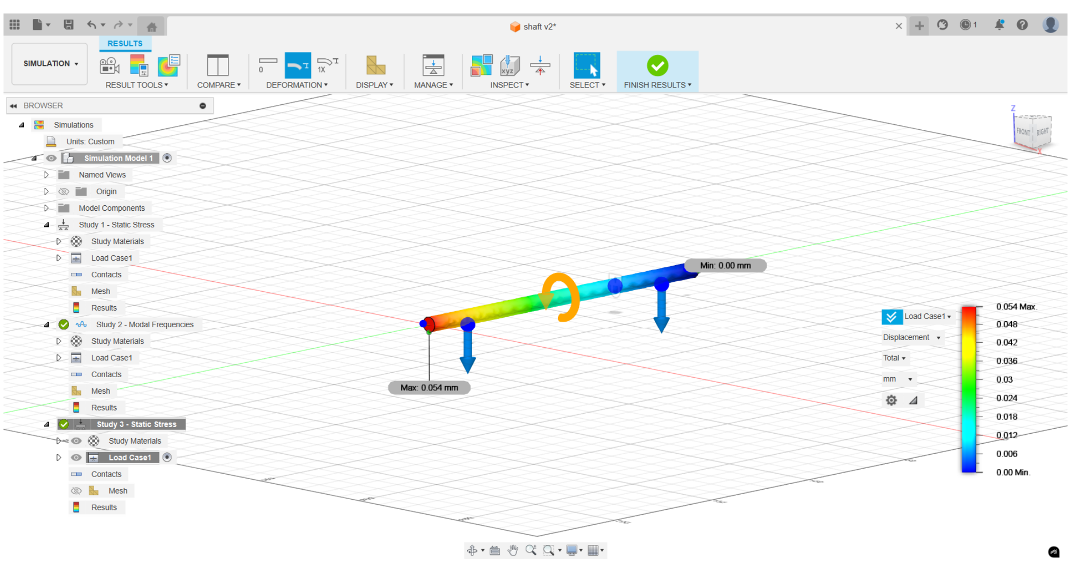


Figure 7.3.1: FEA Test - Dispacement Results

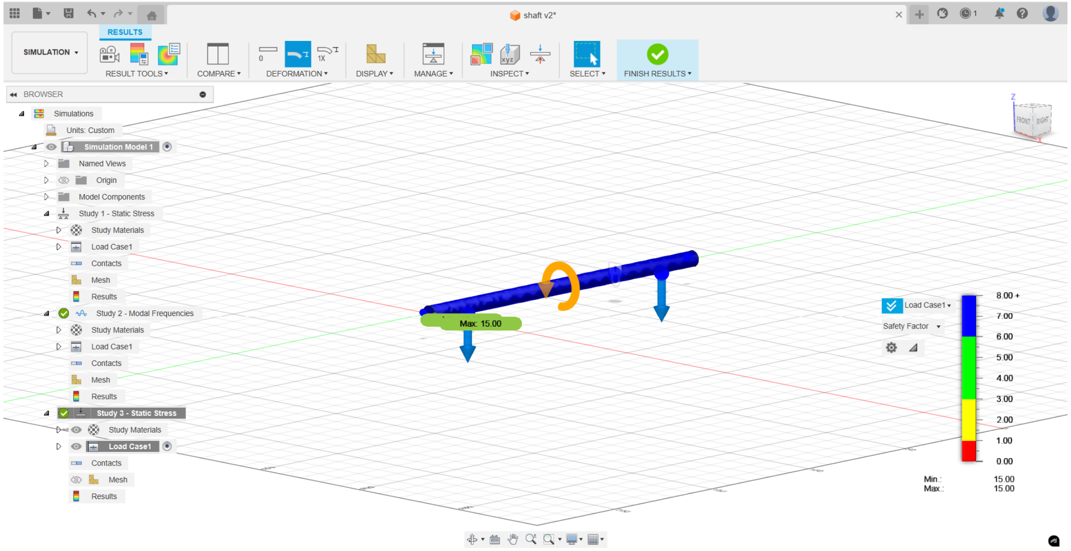


Figure 7.3.2: FEA Test - Safety Factor Results

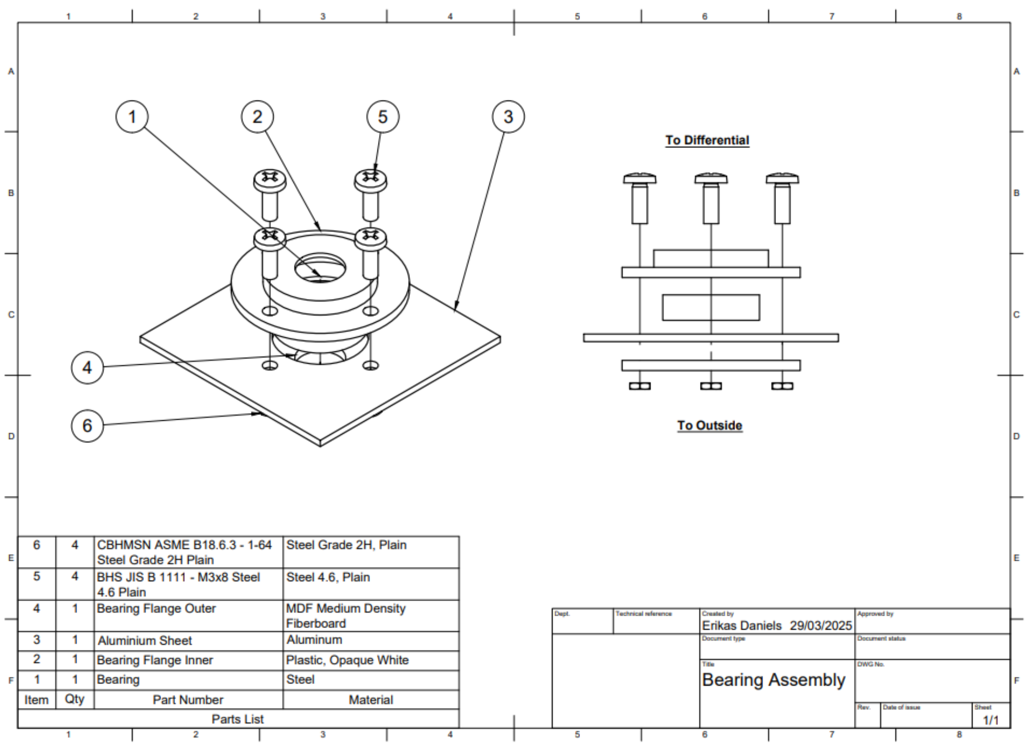
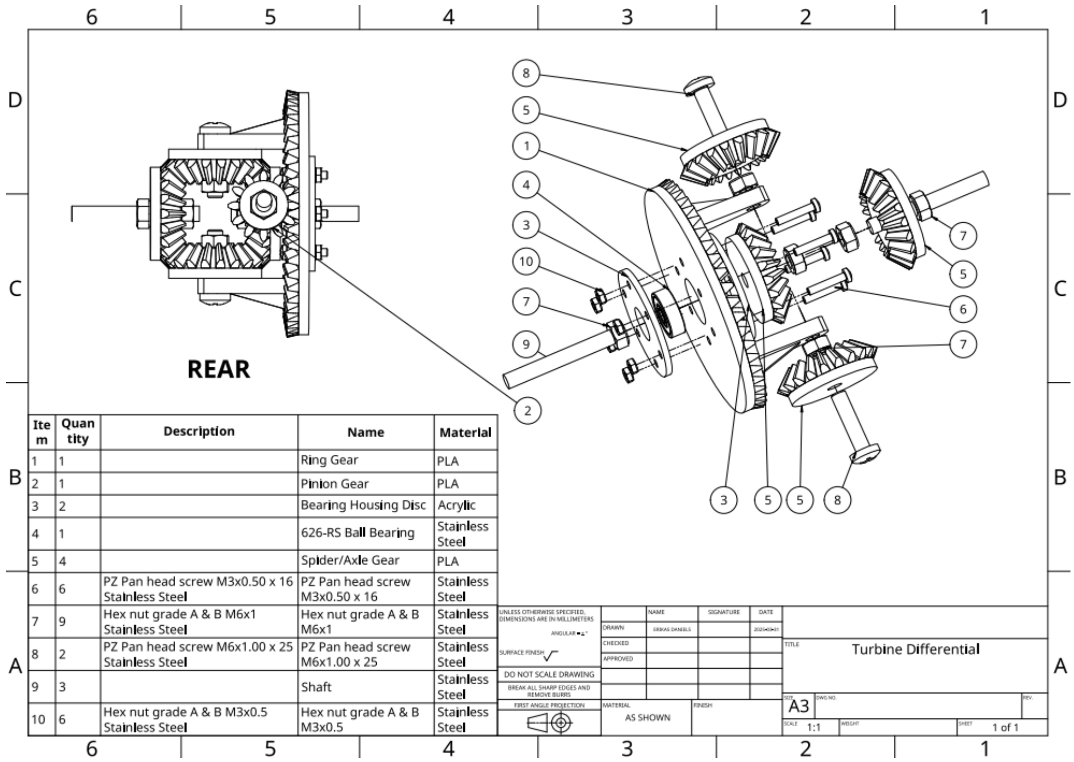


Figure 7.3.3: Assembly of the Bearing Housing



Item	Quantity	Description	Name	Material
1	1		Ring Gear	PLA
2	1		Pinion Gear	PLA
3	2		Bearing Housing Disc	Acrylic
4	1		626-RS Ball Bearing	Stainless Steel
5	4		Spider/Axle Gear	PLA
6	6	PZ Pan head screw M3x0.50 x 16 Stainless Steel	PZ Pan head screw M3x0.50 x 16	Stainless Steel
7	9	Hex nut grade A & B M6x1 Stainless Steel	Hex nut grade A & B M6x1	Stainless Steel
8	2	PZ Pan head screw M6x1.00 x 25 Stainless Steel	PZ Pan head screw M6x1.00 x 25	Stainless Steel
9	3		Shaft	Stainless Steel
10	6	Hex nut grade A & B M3x0.5 Stainless Steel	Hex nut grade A & B M3x0.5	Stainless Steel

Figure 7.3.4: Differential Assembly - Exploded View with BOM

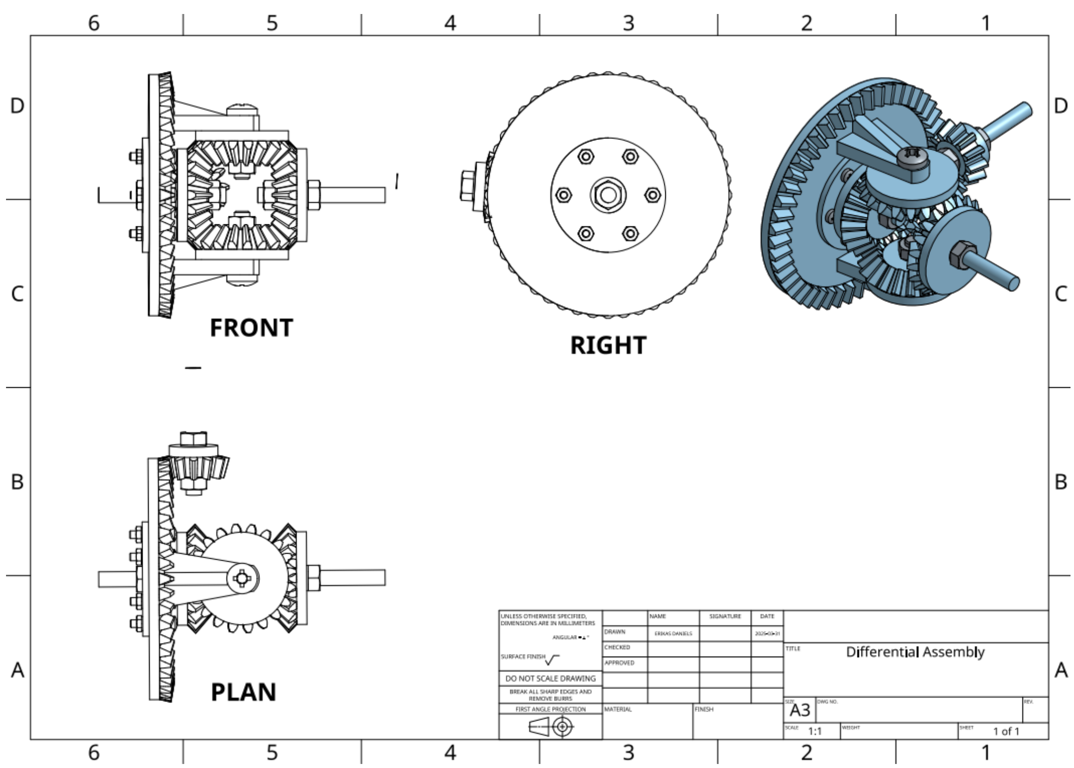


Figure 7.3.5: Differential Assembly - First Angle Projection

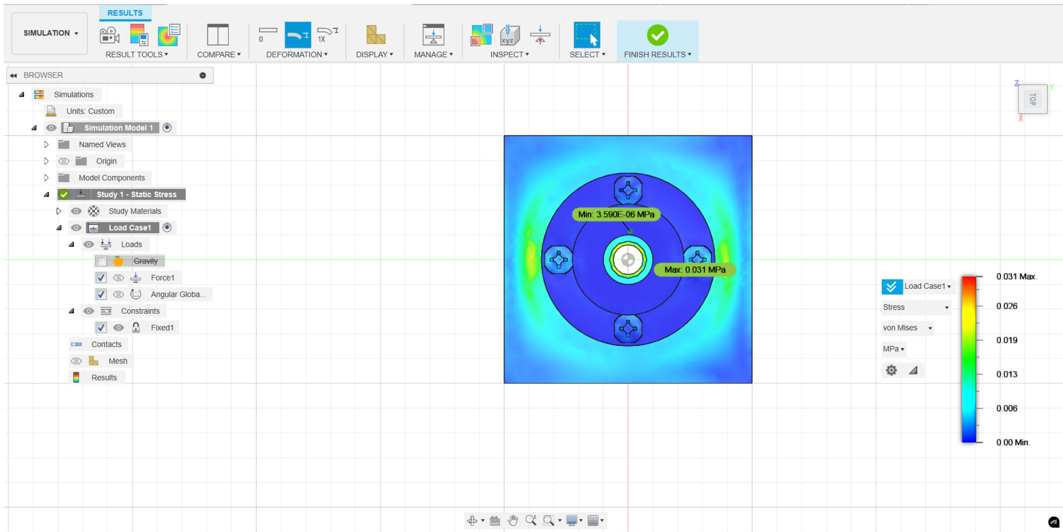


Figure 7.3.6: FEA Analysis for loads on the bearing assembly

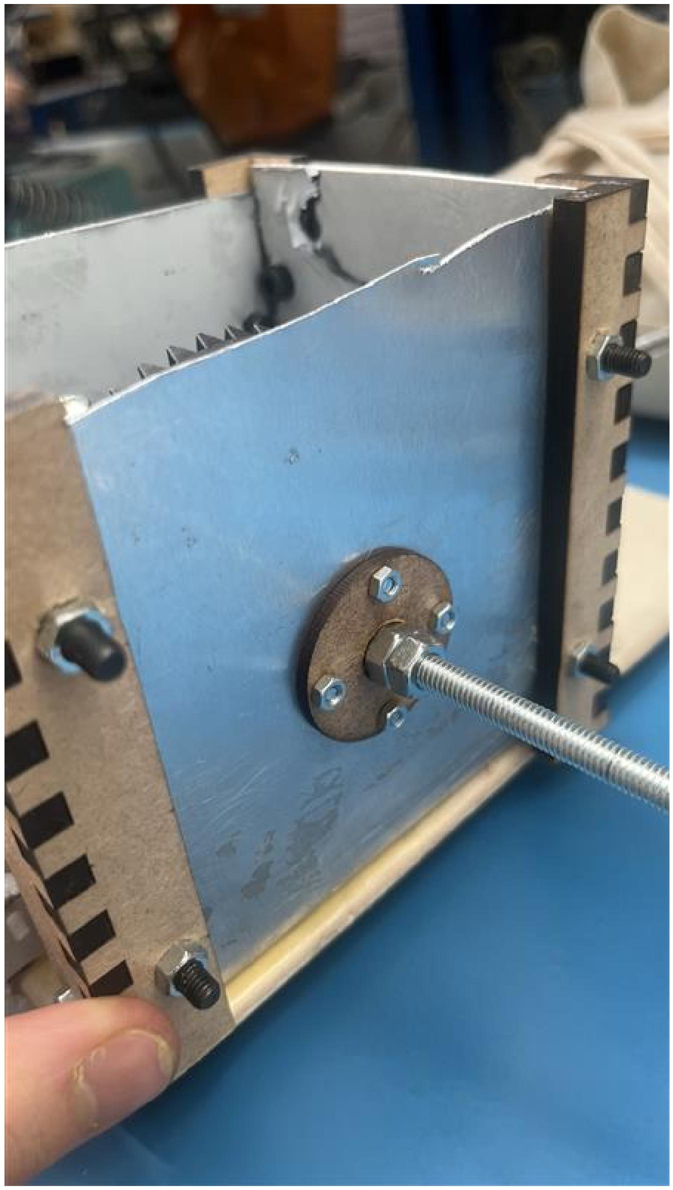


Figure 7.3.7: Outer Bearing Housing

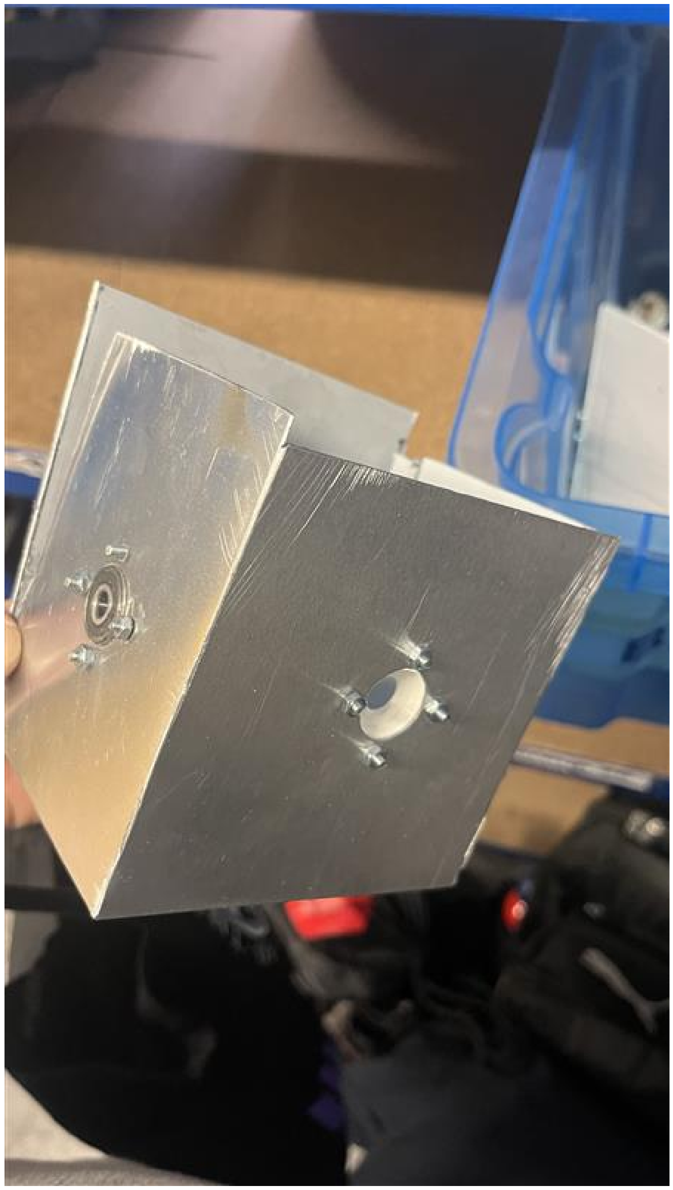


Figure 7.3.8: Initial version of Housing

Figure 7.3.9: Inner Differential Casing with Bearing Housing

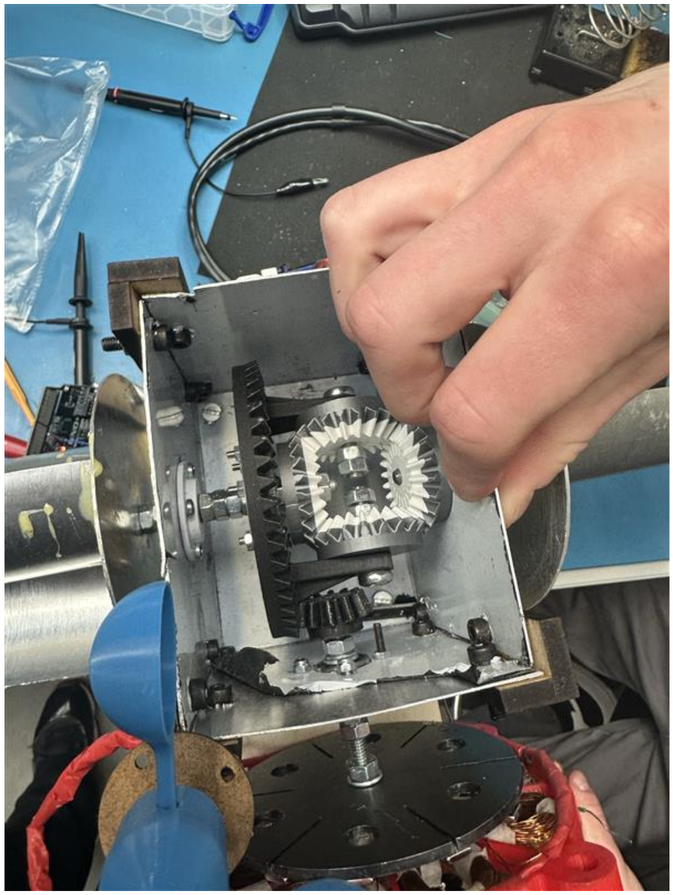


Figure 7.3.10: Final Stages of Assembly

7.4 Tables

Material	Density[kgm^{-3}]	Price [£/100g]	Young's Modulus [GPa]	Yield Strength [MPa]
Aluminium (1.5mm)	2500-2900	1.00	68-82	30-500
Stainless Steel (3mm)	7600-8100	1.00	190-210	100-170
Medium Density Fibreboard (4mm)	700-900	2.75	3.6	36
Plywood (4mm)	660-800	1.75	4.4-6.3	34-43
Acrylic	1180	2.50	3.04	54-73

Table 7.4.1: Material Properties of Stock Material used for Housing