Final Design Report

Team 1

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March 20th, 2018

EME 150B

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Executive Summary

The design revolved around meeting the times at maximum power. Powertrain values, shaft diameters, and spool diameters were determined based on ideal conditions. This would enable the gearbox to achieve times close to the times calculated at maximum power. However, it was found that the calculated diameters would deflect too far under the load conditions. Some shaft diameters were then increased, the design was modified to include parts to support the loads, and fabrication began. Once the parts were made, the gearbox was made and tuned to ensure proper gear meshing. Some load conditions were tested, and the times for the large load cases were too long. The spool diameters were decreased and the shaft was used as the spool for the last load case. This resulted in faster times. As a result, this team achieved 5.15, 13, 26.35, and 56 seconds for load case 1, 3, 5, and 7, respectively. These times are 30.71%, 24.40%, 0%, and 18.12% slower than the fastest times achieved during the competition.

Design Phase

Project Overview

In the following project, teams had to develop a gearbox to lift textbooks up a ramp. The gears and motor were provided, and the load casing varied from heavy loads on a steep ramp to light loads on shallow ramps. The primary focus is to design the gear box to lift the various loads as quickly as possible. Thus, analysis was performed to ensure that the gearbox has the ability to meet this requirement.

Motor Power and Loading

The torque curve for the motor was given as a function of the motor speed. To determine the maximum power, the power function was generated by multiplying the torque function by the speed. The maximum point was then calculated using the first derivative with respect to speed of the power function. The results yielded a torque of 63.5 mN*m at 4600 RPM at maximum power, which was calculated to be 3.05 W. Figure 1 shows the calculations in detail.

Torque Calc

$H = Tn$ (Alternative units)

$$\Rightarrow H = T_s\left(1 - \frac{n}{n_o}\right)n \text{, where } \begin{cases} T_s = \text{stall torque} = 0.0127\ N\cdot m \\ n_o = \text{no load speed} = 9200\ RPM \end{cases}$$

$$\Rightarrow \frac{dH}{dn} = T_s\left(1 - \frac{2n}{n_o}\right) \Rightarrow \frac{d^2H}{dn^2} = -T_s\frac{2}{n_o} \Rightarrow \frac{dH}{dn} = 0 \text{ @ max } H$$

$$T_s\left(1 - \frac{2n}{n_o}\right) = 0 \Rightarrow n = \frac{n_o}{2} \Rightarrow n = 4600\ RPM$$

$$\Rightarrow T = 63.5\ mN\cdot m$$

$$P_{max} = T\omega \Rightarrow P_{max} = (63.5\ mN\cdot m)\left(\frac{1N}{10^3 mN}\right)(4600\ RPM)\left(\frac{1 min}{60 s}\right)\left(\frac{2\pi rad}{1 rev}\right)$$

$$\Leftrightarrow P_{max} = 3.05W = H_{max}$$

Figure 1: Derivations for motor max power calculations

The load cases were modeled as a frictionless ramp in the shape of a right triangle with an angle θ and an object with a mass m. It was determined that the amount of force required to pull the mass up the slope would be **mgsinθ** from statics. Figure 2 shows the free body diagram and corresponding derivation in more detail. Two spool sizes were considered. Note that T_1 was calculated using a spool radius of 7mm while T_2 was calculated using a spool radius of 12.5mm. The use of two spool diameters is explained later. Column 6 in Table 1 also shows the theoretical time to lift the books at the motor's maximum power. This was done using the relationship P = F*v, and then dividing the ramp distance of 3ft by the calculated velocity.

Table 1: Loading case calculations

Test	Angle	Number of books	F (N)	v (m/s)	t (s)	T_1 (N*m)	T_2 (N*m)
1	20	1	6.54	0.47	1.95	0.046	0.0817
2	30	1	9.55	0.32	2.86	0.067	0.119
3	30	2	19.11	0.16	5.71	0.134	0.239
4	40	3	36.85	0.08	11.02	0.258	0.461
5	40	4	49.13	0.06	14.69	0.344	0.614
6	60	4	66.20	0.05	19.79	0.463	0.827
7	60	5	82.75	0.04	24.74	0.579	1.03

The torque values highlighted in green were used for the gearbox kinematic calculations to help design the gearbox.

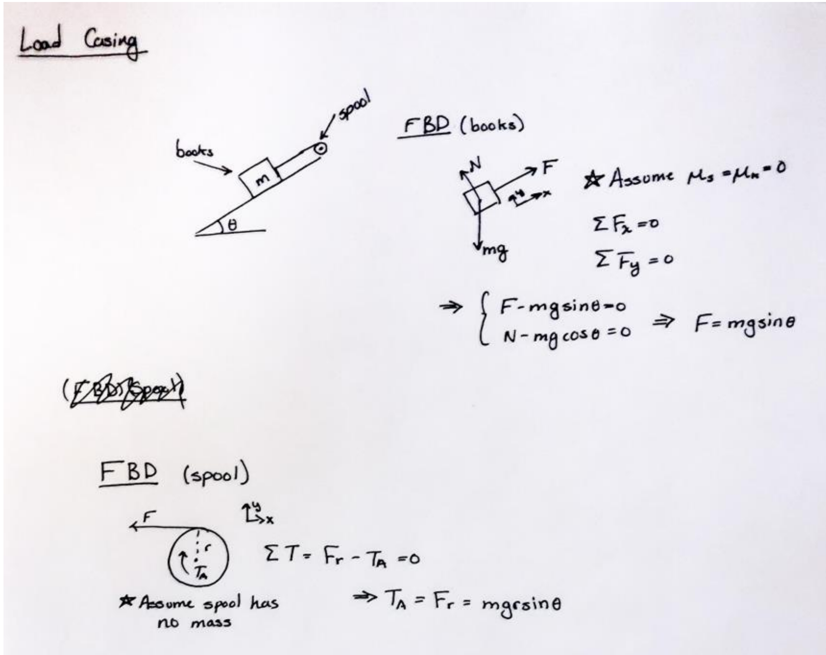

Figure 2: Load Case Derivation

Gearbox Kinematics

To achieve the desired train values at maximum power, the gear meshing combinations had to be determined. Based on external data, it was known that the 10T gear did not mesh well with the 50T gears, thus this meshing combination was avoided. The 10T gears meshed with the 40T gears, and the 20T with the 50T. This produced a maximum gear ratio of 0.01 at the shaft farthest from the motor (shaft d), and a gear ratio of 0.025 on the shaft second farthest from the motor (shaft c). Figure 3 shows the meshing combinations.

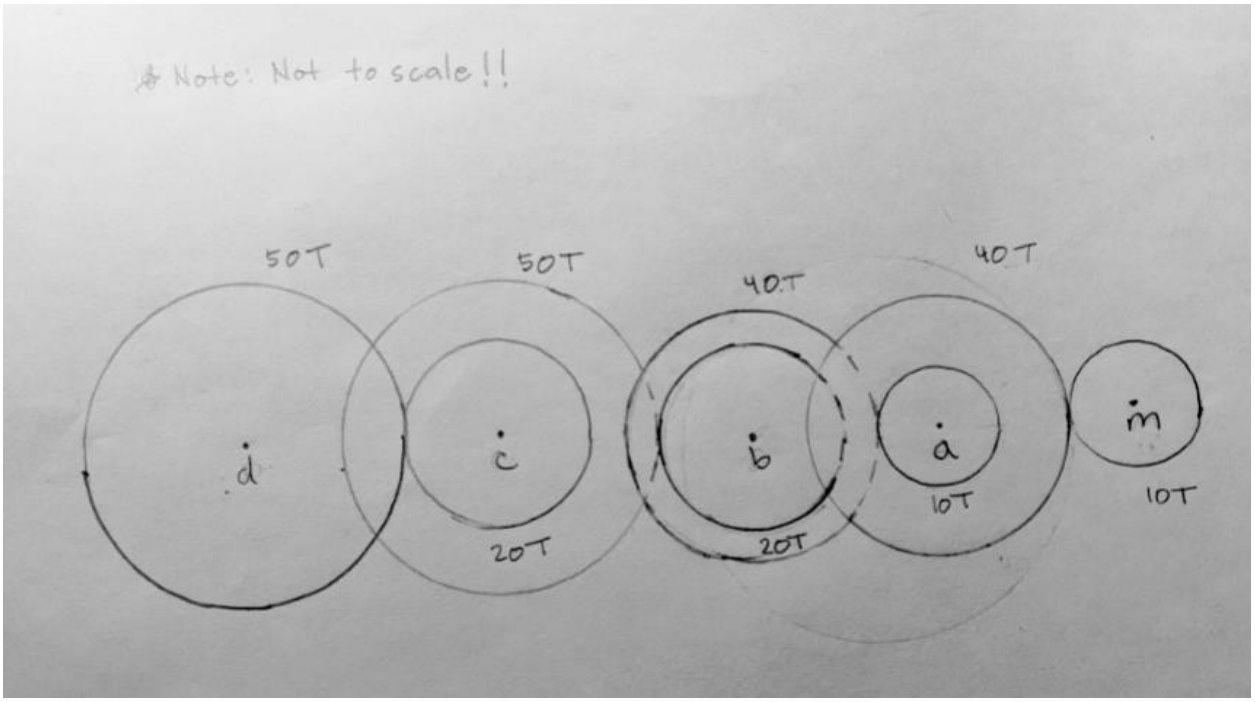

Figure 3: Gear and shaft layout to meet needed ratios

The desired train values were calculated by dividing the torque output of the motor at maximum power with the torque required to lift the loads (T_1 and T_2 from Table 1). From the ideal ratio, the design ratios (e_{des}) could be determined from the ratios available. From the design ratio, the torque load (T_{load}) on the motor could be calculated by multiplying the lifting torque by the design ratio. Once the motor torque was known, the speed of motor could be calculated (n_{des}), and thus the total power the motor produced. From the power, the lifting velocity (v_{des}) and the design lift time (t_{des}) could then be calculated. Figure 4 shows the calculations in more detail.

Table 2: Design ratios and other calculated values

Case	e_{calc}	e_{des}	T_{load}(N*m)	n_{des}(rpm)	v_{des}(m/s)	t_{des}(s)	t_{diff}(s)
1	0.077724	0.025	0.002042	7720.402	0.25265	3.619	1.66
2	0.053166	0.025	0.002986	7036.972	0.230284	3.971	1.11
3	0.026583	0.01	0.002389	7469.578	0.097777	9.352	3.634
4	0.024617	0.025	0.006449	4528.365	0.082987	11.019	0.00267
5	0.018462	0.01	0.003439	6708.462	0.049176	18.595	3.91
6	0.013703	0.01	0.004634	5843.159	0.042833	21.348	1.56

7	0.010963	0.01	0.005792	5003.949	0.036681	24.928	0.192

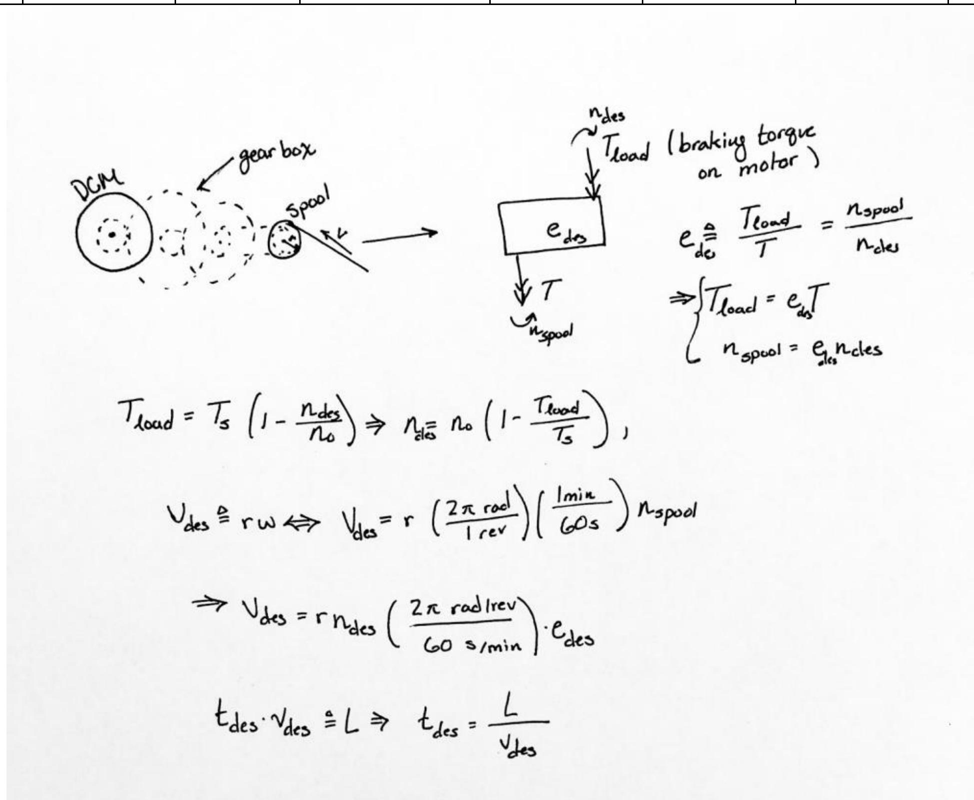

Figure 4: Lift times at chosen train value

Design Concept

When coming up with the gearbox design, there were three important design considerations: adjustability of the shafts, ease of assembly, and torque transmission between the gears and shafts. To allow for shaft adjustability, drift blocks were designed to be bolted into slots in the frame so the blocks could be moved fore and aft. The drift block was designed to have a bearing pressed into it to hold the shaft. Furthermore, the drift block had tapped holes to accept a 6-32 machine screw. This prevented the need for nuts on the inside of the frame which could potentially interfere with moving parts. The shafts were designed to be shouldered, allowing the shafts to be axially constrained between the drift blocks. Figure 5. shows the drift block concept.

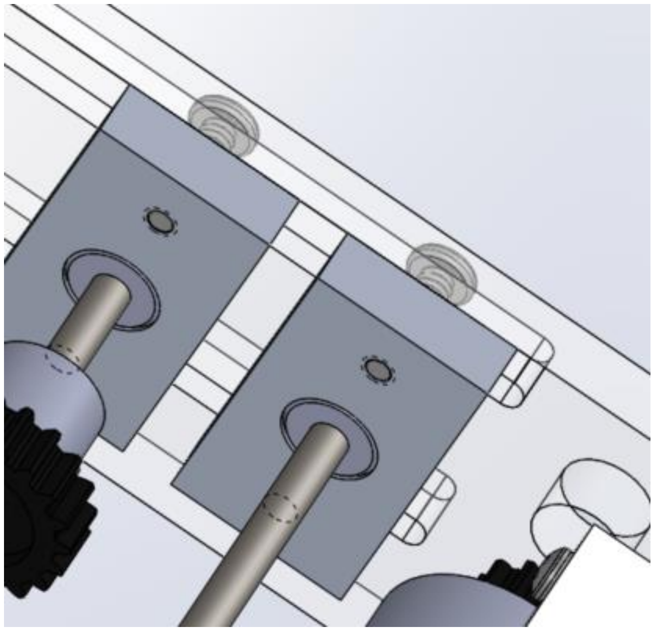

Figure 5: Drift block with shaft. Gear omitted for clarity.

To ensure the gears would not slip on the shaft, gear hubs were designed. The hubs would bolt onto the gears with machine screws, into tapped holes in the hub itself. Then, the hub would attach to the shaft with a set screw. The gears on shafts d, c, and b had hubs designed, because the torques on the these shafts would most likely be too high for a press fit to be feasible. Shaft a didn't have hubs, because the low torque on the shaft made a press fit solution reasonable. Figure 6 shows a gear with a hub.

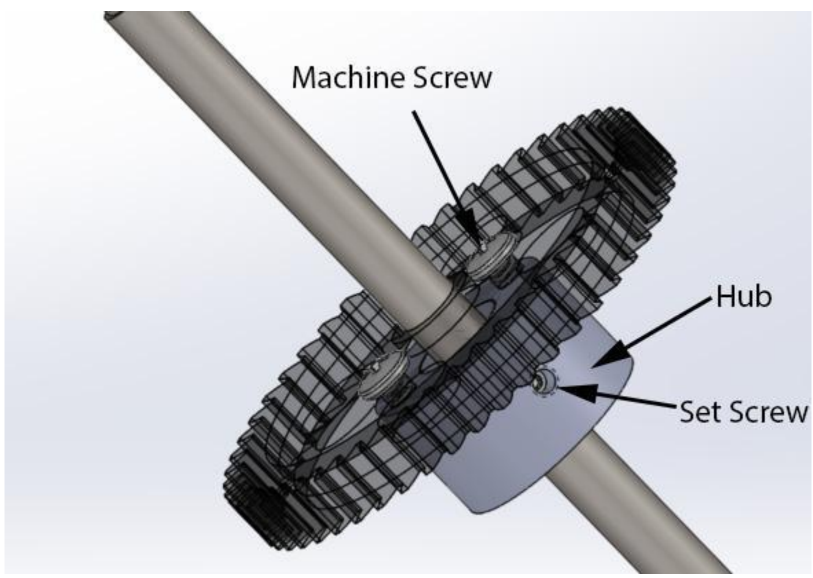

Figure 6: Gear attached to shaft via a hub. Gear shown transparent for clarity

To meet the ease of assembly requirement, the gearbox was designed such that every adjustable element was bolted to the side plates. This meant, that one plate could be loaded with drift blocks, the shaft could then be inserted with their gears, and the other side plate could then be bolted on top. Figure 7 shows an exploded view of how the gearbox can be assembled.

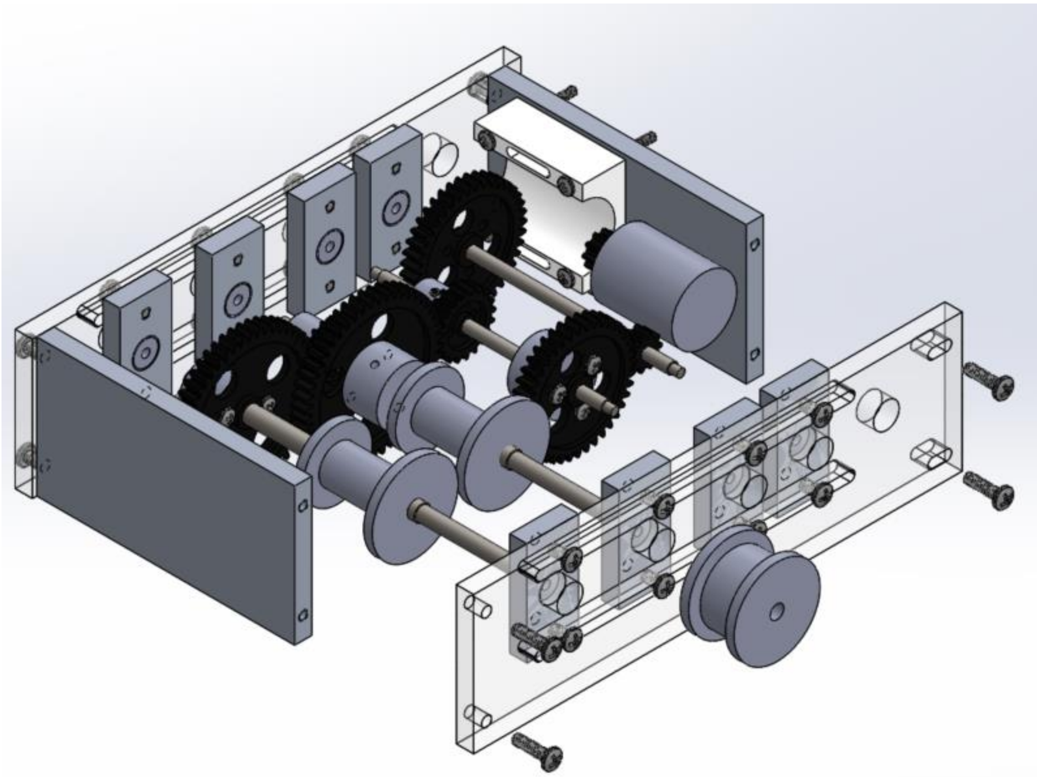

Figure 7: Exploded view of assembly.

Frame Analysis

The deflection of the frame due to the cart and shaft loads is negligible due to the axes that the force is applied on have very high moments of inertia, causing the frame to be very resistant to deflection in those directions. For this reason, a more detailed analysis was not performed. Instead the frame was made overly thick to prevent failure.

Shaft Analysis

The forces in each shaft were calculated using moment and force balances in MATLAB. The force from the cart load was multiplied by the spool radius to find the applied torque on the last shaft. Two critical loads were used in this analysis: first with the 3rd loading scenario using the second to last shaft as the output, and second with the 7th loading scenario and the last shaft as an output. Since the forces were found to be much lower using the parameters for the 3rd shaft, this case was neglected from the rest of the analysis. Using the gear ratios, this torque was propagated back to the motor. The tangential force for each gear was found using this torque and corresponding pitch diameter. With a pressure angle of 20 degrees, the radial force component for each gear was calculated; the results from these calculations are shown below.

Table 3: Torque and force calculation results

	F_t [N]	F_r [N]	F_{total} [N]	No. teeth	Torque [Nm]	Position [in]
Shaft A						
bearing 1	-0.404021	-0.14705	-0.42995	x	0	0
bearing 2	3.9297006	1.430294	4.1819	x	0	3.25
input Gear	4.713463	1.71556	5.0159626	40	0.579237454	2.7625
Output Gear	1.187791	0.432321	1.2640208	10	0.579237454	0.21
Shaft B						
bearing 1	5.2470369	1.909765	5.58378	x	0	0
bearing 2	-0.825595	-0.30049	-0.87858	x	0	3.25
input Gear	9.411346	3.42545	10.015345	40	0.231694982	1.1095
Output Gear	4.751164	1.729282	5.0560831	20	0.231694982	2.7625
Shaft C						
bearing 1	12.578537	4.578213	13.3858	x	0	0
bearing 2	1.3537118	0.492711	1.44059	x	0	3.25
input Gear	23.52836	8.563623	25.038358	50	0.092677993	0.6395
Output Gear	9.596091	3.492691	10.211947	20	0.092677993	1.1095
Shaft C (out)						
bearing 1	-10.582	-2.3046	10.830046	x	0	0
bearing 2	-18.118	-1.195	18.157366	x	0	3.25
input Gear	9.6	3.5	10.218121	50	0.2317	0.6395
Small spool	19.1	0	19.1	20	0.2317	2.525
Shaft D						
bearing 1	-37.718	7.012205	38.364287	x	0	0
bearing 2	-68.97	-1.7177	68.991386	x	0	3.25
input Gear	23.990228	8.731729	25.529867	50	0.023169498	0.6395
Small spool	82.74821	0	82.74821	x	0.023169498	2.525

The shaft furthest from the motor is expected to have the highest deflection due to the high applied loads and incurred bending moment. A deflection analysis was performed using the equations from table A-9[1] in MATLAB. The deflection in m/m was plotted against the distance from the left bearing and is shown in Figure 7 below.

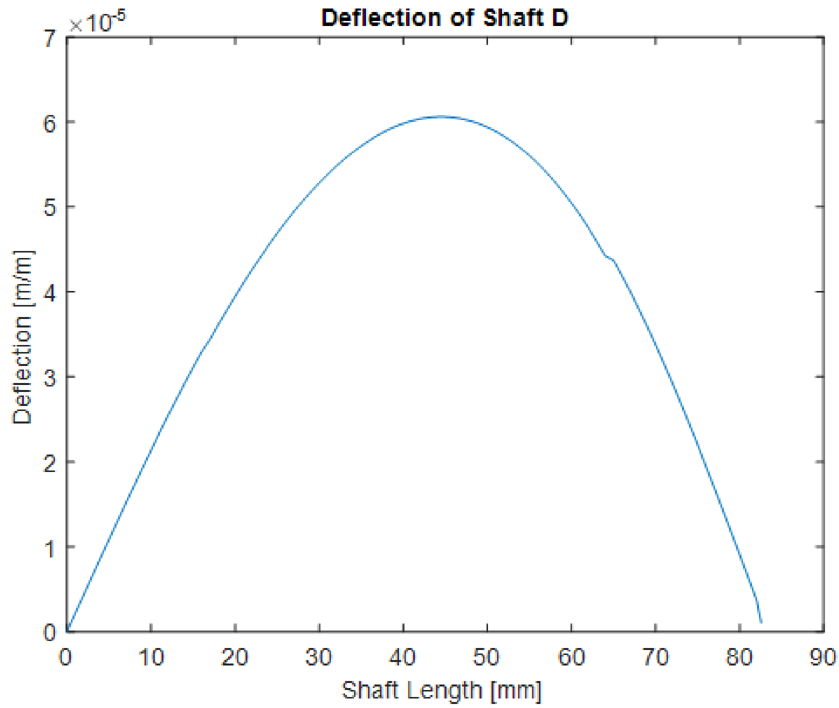

Figure 7: Shaft D deflection plot

The calculated deflection was compared to values in Table 7-2. The highest deflection that occurs in a shaft originally was .0118 in, which when compared to values in the Table 7-2 was too large for the gears on this shaft. The diameter of this critical shaft was increased to 6 mm to compensate for this high deflection. This resulted in a deflection of 0.00007 m (0.002755 in).

The shaft furthest from the motor carries the highest torque. Therefore, it has the highest potential for failure. All of the shafts are shouldered only near the bearings because moment and torsion are small. Stress concentration does not play a factor in risk analysis. This maximum shear and bending stress on this shaft for the worst load case, 7, were found and compared with the yield strength of 1020 cold-rolled steel. Max shear stress was calculated to be 2.45 Mpa while the max bending stress was found to be 59.895 MPa. The yield strength of 1020 steel is 350 MPa which is significantly larger than either of these stresses. The Von Mises stress was calculated for the combined loading giving a value of 60.1 MPa which gave a factor of safety of 5.8.

[1] R.G. Budynas, J.K. Nisbett, *Shigley's Mechanical Engineering Design*, 10th ed., McGraw Hill, 2015

Materials

The materials used to manufacture the gearbox are shown in Table. Most parts were ordered online. However, some screws and old totems were donated by group members or the EFL.

Table 4: This table outlines what materials were ordered, where it was bought, and the cost.

Part/Material	QTY	Vendor	Cost
Aluminum Rod 0.8" DIA	4"	Online metals	$2.56
Aluminum Rod 0.5" DIA	4"	Online metals	$1.44
Aluminum PlAte .25" x 2"	8"	Online metals	$2.76
Aluminum Plate .25" x .75"	24"	Online metals	$2.12
6-32 1/2" Machine Screws	50	Amazon	$10.46
Clear Cast Acrylic 1/4"	12"x12"	Amazon	$14.99
6mm x 200m Steel Shaft	2	Amazon	$6.42
4mmx150mm Steel Shaft	2	Amazon	$6.49
# 6 washers	100	Ace Hardware	$3.01
3mm x 10mm x 4mm Ball Bearings	5	Amazon	$4.32
5mm x 10mm x 4mm Ball Bearings	10	Amazon	$7.99
4-40 set screws	10	Amazon	$5.99
DPDT Switch	1	Amazon	$1.60
2AA Battery Holder	1	Amazon	$1.75
Zip-Tie	1	N/A	$0.00
4-40 1.25" Machine Screw	4	N/A	$0.00
4-26 1" Machine Screw	4	N/A	$0.00
4-40 hex nut	4	N/A	$0.00
Motor Mount	1	N/A	$0.00
Old Totems	2	EFL	$0.00
Total	**N/A**	**N/A**	**$71.90**

Fabrication

All the parts were manufactured in the EFL except the motor mount which was 3D printed by the ESSC. Parts were gradually made in the EFL over two weeks. This happened because we did not receive all of the stock material and manufactured what did come. First, the acrylic panels and spool caps were laser cut in the EFL. This process was fast and easy since the desired shape was programmed in the computer. Next came making the gear hubs, front panel, back panel, shafts, drift blocks, and spool rounds. The gear hubs, front panel, back panel, drift blocks, and spool rounds were easily manufactured using simple turning and parting operations for the spool rounds and hubs and the mill for the drift blocks, front panel, and back panel. The shafts were the most difficult parts to manufacture. Hardened steel shafts were accidentally ordered as the amazon product page did not specify that the shafts were hardened. In order to compensate for this, the shafts were turned at a lower speed than listed and passes of 2-10 thousands of an inch were made until the desired dimension was reached. This was a tedious process, but it prevented the turning tool from being damaged. After everything was cut, the holes were tapped for the machine and set screws.

Part Machining

The frame is composed of two materials, aluminum and acrylic. The side plates are cut from ¼" acrylic. The front, rear and drift blocks were machined out of aluminum. The side plates were laser cut as this process was quick and easy to do. The drift blocks were first drilled out of one long piece of 0.75" x 0.25" aluminum before being cut into 1.5" piece. This was done for speed, since drilling out one long piece meant pieces did not need to be taken in and out of the milled and zeroed. The hole to accept the bearing was reamed to 10mm. The front and rear plate were machined out of aluminum plate. The plates were first machined down to the proper width, the, the holes in the edges to accept 6-32 machine screws were drilled. The face of the rear plate was machined to include bolt holes for the motor mount, and slots for clearance so the motor could be zip-tied to the frame.

The shafts we turned to create the shouldering features. The shafts were turned down in stages to reduce the effect of deflection as the turning tool cut into the shafts. Unfortunately, the shafts were slightly hardened, which was not specified when they were bought. This led to some machining difficulty as the depth of cut had to be very small to ensure decent surface finish.

The gear hubs were machined out of the the rod stock that was purchased. For the 50 tooth and 40 tooth gear hubs, the 0.8" diameter stock was used, and a clearance pocket was drilled in the center to clear the gear hubs. The bolt holes required to bolt the gears to hubs were drilled and tapped. The 2 tooth gear hubs were machined from the 0.5" diameter stock. All hubs were drilled out to the correct shaft diameter. A hole for the set screw was drilled into the side of the hubs and tapped.

Assembly

The drift bocks and the frame are held in place using 6-32 stainless machine screws. Bearings were pressed into the drift blocks using the arbor press, and the hubs/spools were fixed to the shafts using set screws. There were some problems with screw head clearances when attaching the hubs to the 20 tooth gears. For this reason, the heads on the 2-56 screws used to bolt the hub to the gear were turned down. The gears on the first shaft (shaft a) were simply pressed on due to the low torque and consequently low slipping risk. A certain amount of adjustment was needed to get the gears meshing properly, however the drift blocks made the procedure simple. Unfortunately, the center to center distance of the gears was underestimated, such that shaft a was closer to the motor than anticipate. When mounting the motor to the motor mount, the 40 tooth gear on shaft a and the 10 tooth gear on the motor gear were too close. The rear plate has to be milled down where the motor mount attached. Figure X. shows this feature. The sides of the spools were pressed on with the arbor press. Some of the spool sides were not made to tolerance, so an adhesive was used to ensure they did not come apart. Figure X. shows the final design fully assembled and as used on the competition day.

Figure 8: Milled out section of the rear plate

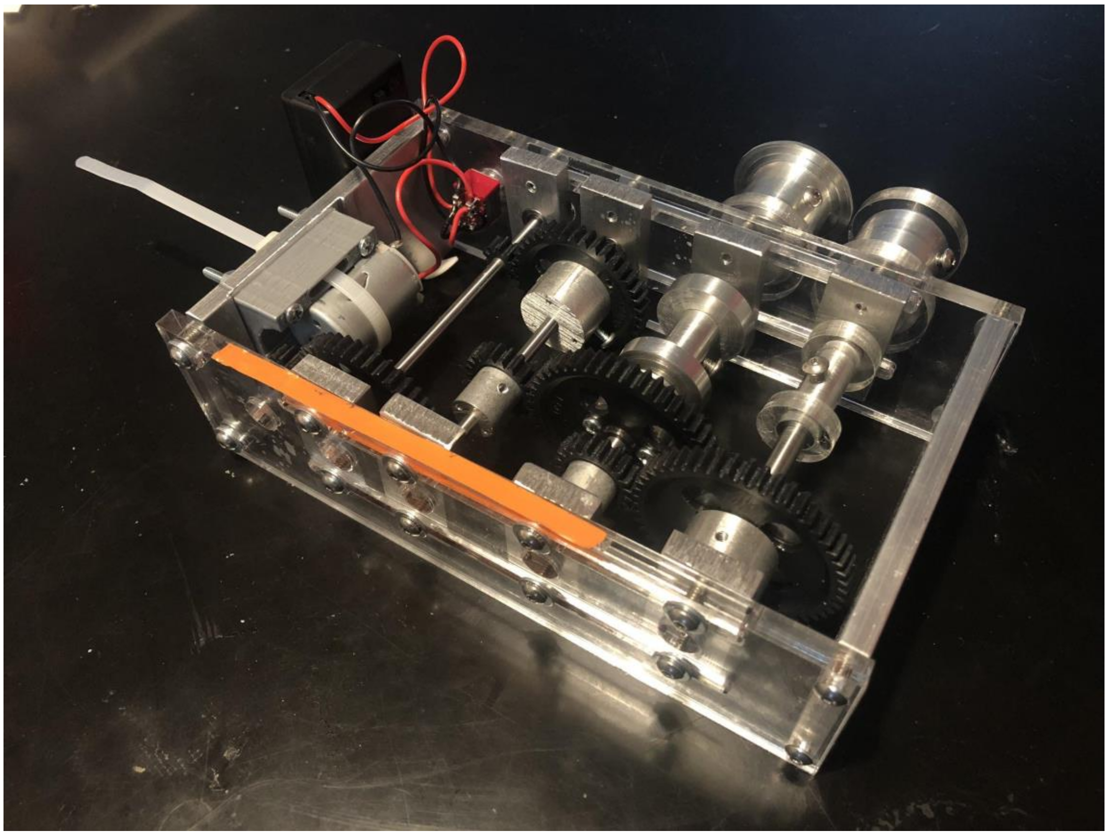

Figure 9: Alternate view of the gearbox.

Testing

Test Report

The gearbox was immediately tested after it was tuned to give proper meshing. This resulted in the values found in Table 5 .

Table 5: Timed Results for 4 different configurations

Number of Books	Angle (deg)	Time (s)
1	30	8.93
1	40	9
3	40	36.83

4	60	DNF

Modifications

The design was slightly modified so that better times could be achieved. For instance, a metal plate was added to a 50T gear to increase its rigidity. The gear would warp under the load conditions and the plate provided enough stiffness to the gear to prevent warping. Washers were added between the side plate, and the rear plate (Figure X.), since the shaft closest to the motor is slightly longer than specified, resulting in binding. The spools also became a three piece assembly rather than a two piece assembly to reduce manufacturing time. An aluminum rod was parted to the right length then shouldered on both ends to press fit acrylic caps on.

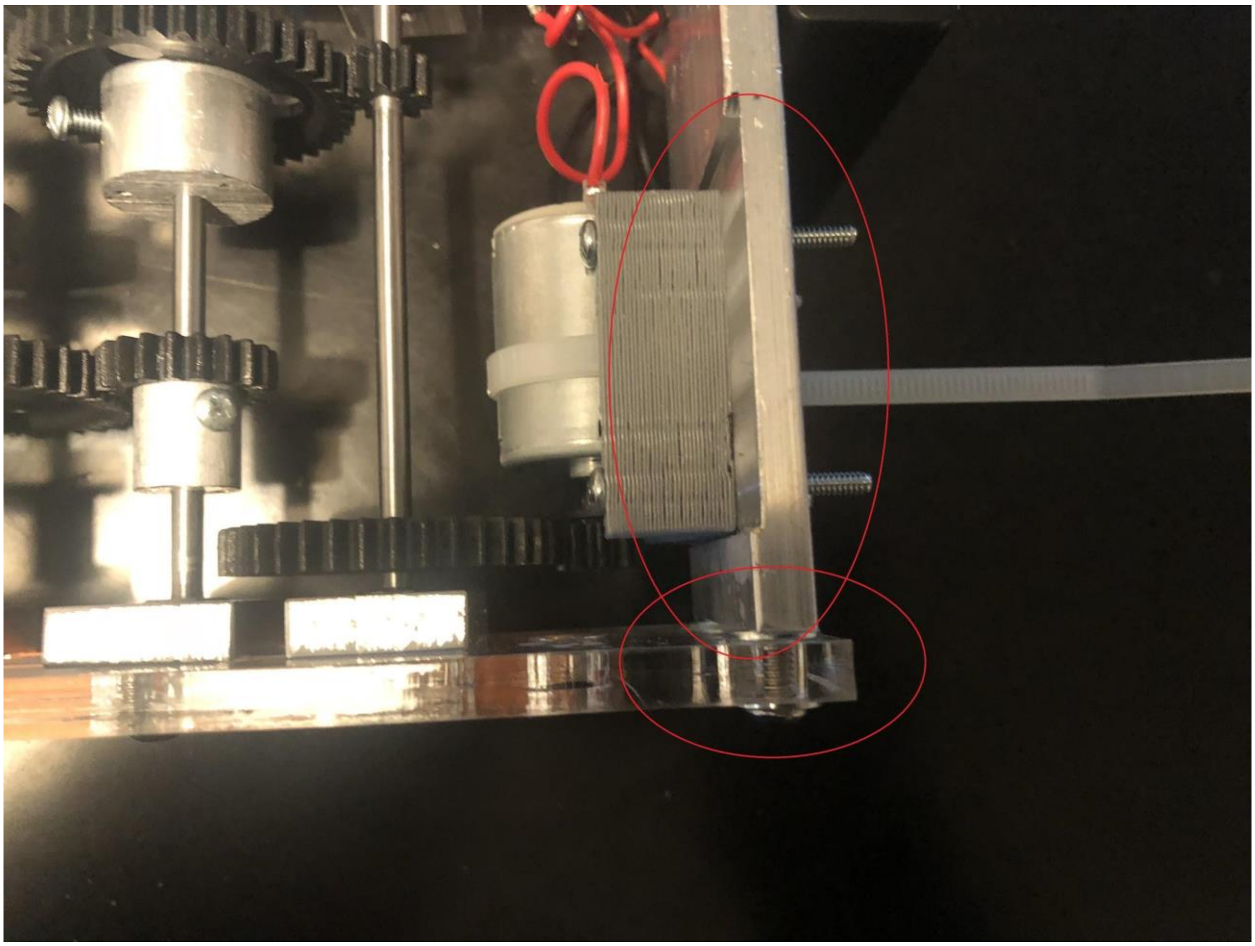

Figure 10: Washers between acrylic plate and rear plate.

A cart was not made during the initial testing. However, one was made before the final competition. Four wheels were attached to a plastic bin and holes were drilled for a rope to thread through. This rope connected the fishing line that connected to the spools on the gearbox. The wheels had a swivel to them so they were glued in place so that the wheels would be parallel to the direction they were pulled in. This design was used because it was the simplest. The bin was big enough to hold 5 books. The wheels also made it so the cart would roll up the ramp rather than slide up the ramp. The rope also acted like a harness for the fishing line to attach to. The spools were also turned down to a smaller diameter before a second testing run and before the competition. It was also decided to use the shaft for the last load case rather than the spools that was designed. This would theoretically increase the speed of the pull rate. It was determined that the difference in torque between the spool and shaft would not outweigh the increase in speed.

Final Competition

Overall, the gearbox performed well with each load case and even achieved the fastest time for one of the load cases. Table 6 shows each teams performance, Table 7 shows the fastest times for each of the load cases, and Table 8 shows Team 1's relative performance against the fastest time of each load case.

Table 6: Each teams performance during the competition.

Load	1	2	3	4	5	6	7	8	9	10	11	12	13
1	5.15	12.5	8.5/ 4.03	4.03	3.94	8.50	7.5	21.75	6.94	5.5	6.04	7.21	8.6
3	16.25/ 13.00	26.69	23.41/ 10.45	53.97/ 38.16	12.38	18.69	20.92	33.44	20.90/ 16.13	21.72	13.1	19.22	34.65/ 22.32
4		34.56						34.12					
5	26.35	52	1:03.13/ 27	55.85	36.1	45.4	34.1	1:03.40	55.85 /47.15/ 38.09	53.53	34.65	54.7	1:10.00/ 1:02.75
6				2:30									
7	59.56/ 56	3:30	1:54.13/ 55.94		47.41	1:04	1:03.25		1:45.00/ 1:00.7	1:44.31	53.3	1:09.05	1:48.00/ 1:04.9

Table 7: Fastest times for each load case.

Case 1	Case 3	Case 5	Case 7
Team 5: 3.94s	Team 3: 10.45s	Team 1: 26.35s	Team 5: 47.21s

Table 8: The relative performance against the fastest time for each load case.

Case 1	Case 3	Case 5	Case 7
30.71% slower	24.40% slower	0% (fastest time)	18.12% slower

Faster times could have been achieved for load case 3 and load case 7. During load case 3, the acrylic cap for the spool broke off and interfered with the fishing line for both runs. As a result, the line was not able to properly wrap around the spool. This is shown with the 3 second difference in two runs. For the 16 second run, the spool cap broke off when the cart was in the middle of the ramp. For the 13 second run, the spool cap broke off right before the cart reached the top.

During load case 7, there was not enough shaft length for the fishing line to wrap around the shaft. This caused the fishing line to wrap around the machine screws that connected the gear to the hub. This action caused a momentary stall in the gearbox and overall slower times. The spool that would have been used to pull the last load case was still attached to the shaft. If that was removed, there would have been enough room for the fishing line to wrap around the shaft without any interference. The spool was left on because the entire gearbox would need disassembling to remove the spool. It took too much time to properly tune the meshing that the spool was left on to prevent the tuning process again.

Even though there were some small hiccups with the gearbox, it still performed very well. All four load cases were done under a minute and the problems that occurred were from small manufacturing mistakes. In a future design, the spools would have better dimensions to give better press fits and the design would include better modularity. The increase in modularity would prevent complete disassembly to remove individual components.

The times recorded in Table 6 differ greatly from the calculated values in Table 2; all the values in Table 6 were greater than those in Table 2. This can be explained by the losses in the gearbox itself. When calculating the design lift times, it was assumed that the mechanical power from the motor would be transferred completely to the spools to lift the loads. This does not occur as the gears are not ideal. Also a significant amount of power is wasted through the gear box. Friction on the cart also increases the effective lifting load as the hauler not only has to overcome gravity, but the friction of the cart on the ramp as well. This is more likely a less significant effect than the power lost in the drivetrain of the gearbox.

Conclusion

The Shigley hauler project reinforced manufacturing ideas learned in EME 50, concepts learned in EME 150A, and developed new ideas learned in EME 150B. An initial concept of the hauler was developed after determining how to best lift the loads. However, this idea was refined after performing deflection analysis. Manufacturing began and was quickly finished within two weeks, which left plenty of time to tune the hauler. The hauler was tuned to comply with its physical performance and produced amazing results. It was the fastest performer in one load case and was not too much slower than the fastest times in the other load cases. Overall, the project developed how to connect theoretical performance with physical limitations with an exciting competition.

The design could be improved however. Instead of using acrylic side plates, aluminum could be used to give the frame more rigidity. Since acrylic is brittle, the hauler could only be clamped to the ramp on an area that did not have slots for the drift blocks. In fact, during testing, part of the frame cracked. Further improvements to the shafts could also be made, such as using non-hardened steel for better machinability. This project taught that material selection and machining tolerance play a big role in the effectiveness of the overall design.