Mechanical Design
Shigley Loader

Final Report

EME 150B

John Betando

Erik Contreras

Jeffrey Lee

Juan Jara

Zac Shyvers

Professor Barbara Linke

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I. INTRODUCTION

The Shigley Hauler is a gear-driven winch used to haul heavy loads (Shigley books) up an incline plane at varying angles. This device features a Mabuchi RE280 DC motor energized by a 3V power source, a gearbox with a 1:100 and 1:50 gear ratio (depending on which shaft one places the spool), a spool with the best diameters for each test setup, and a light aluminum cart used to transport the books up the ramp. This report will describe the designs of each component, the manufacturing process for each component, the testing scheme for the prototype, and the results from the final competition. Because there was a large discrepancy between the theoretical lift-times and competition lift-times, this report will also go over possible causes for these discrepancies.

II. DESIGN

A. Concept

The design consists of three plates mounted to a base plate to constrain the gear shafts. A high level of accuracy and testing was necessary to ensure proper gear tooth contact. Gears are fixed to the shafts with set screw hubs and shaft collars. The motor is mounted in a 3D-printed housing that is bolted to a slot on the base plate. The shafts interface with the side plates using nylon bushings to reduce friction when spinning. Shaft collars constrain the gear shafts axially.

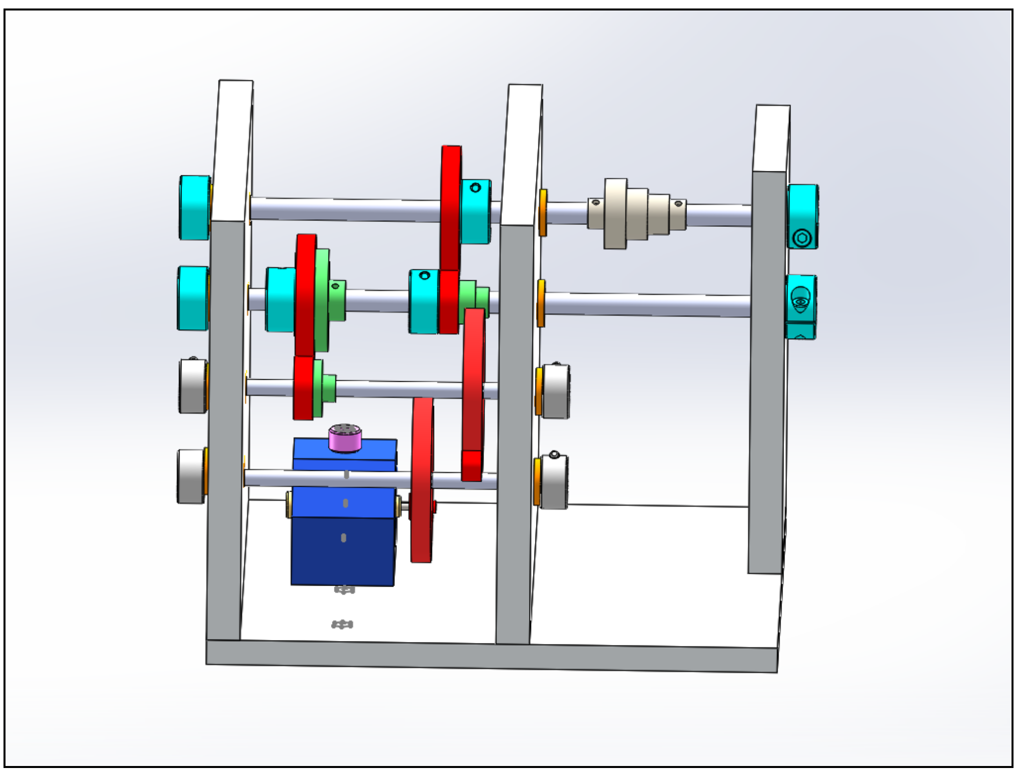


Figure 1: Isometric View of the Shigley Hauler

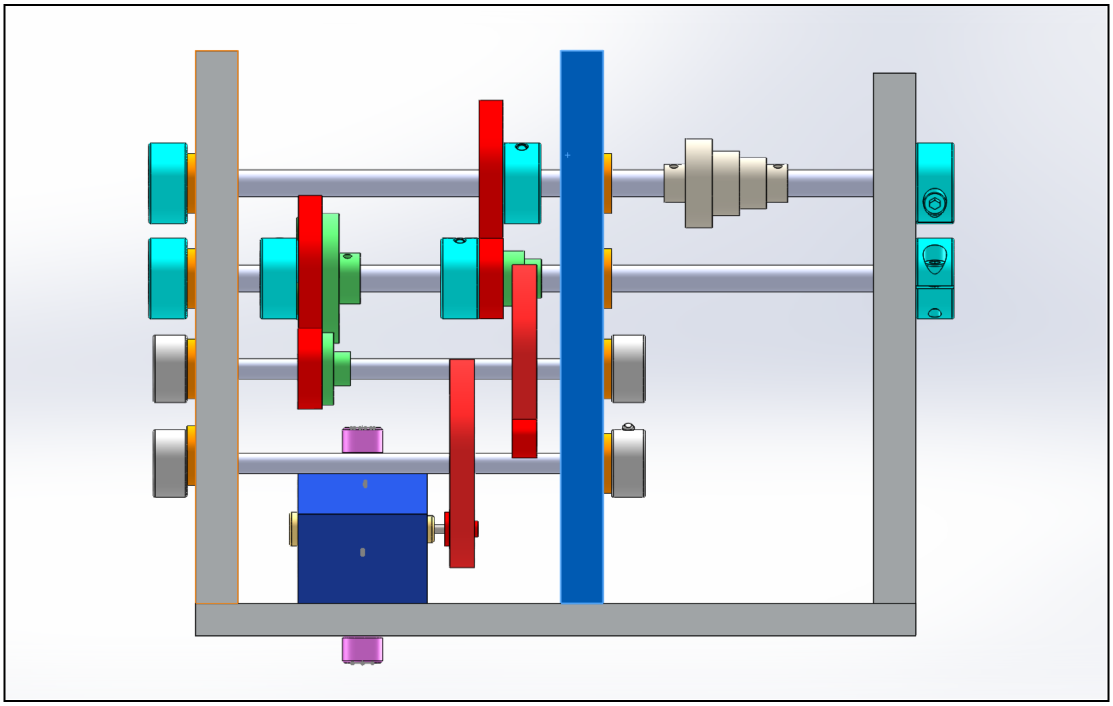


Figure 2: Front view of the Shigley Hauler

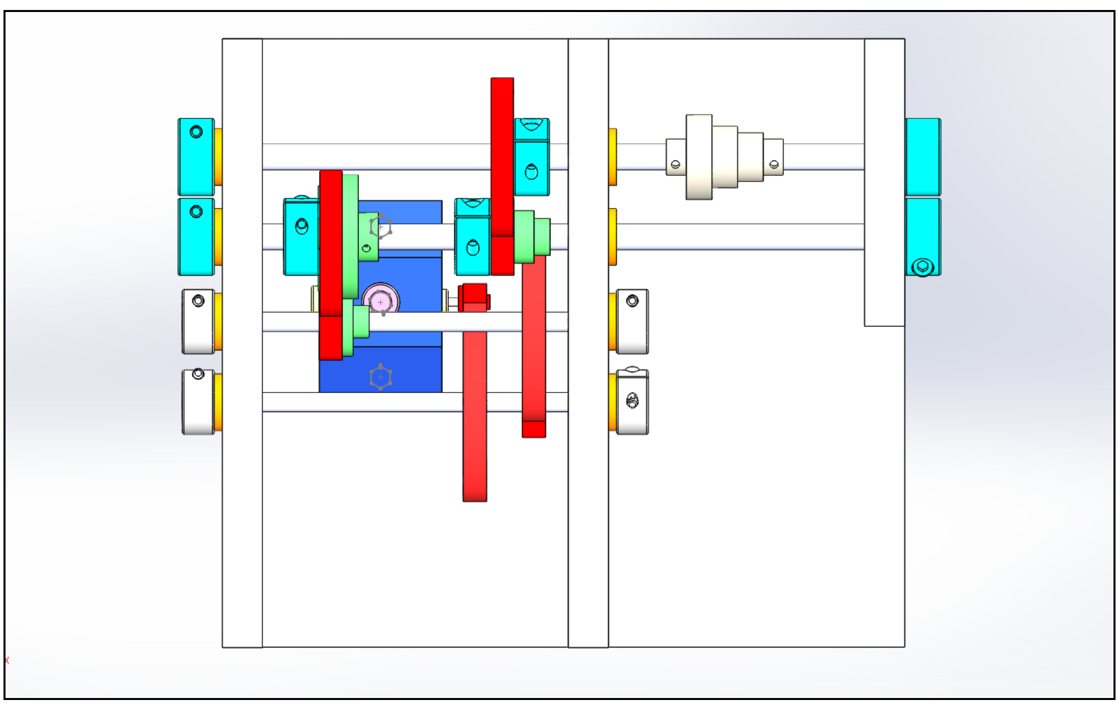


Figure 3: Top view of the Shigley Hauler

B. Gearbox

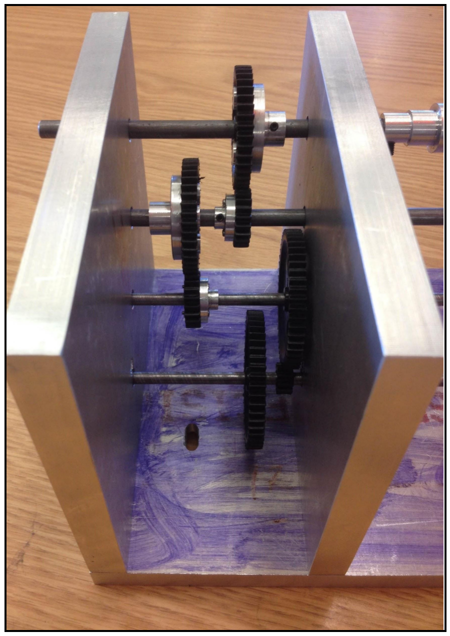


Figure 4: The assembled gearbox

Materials

- Stock aluminum plates and cylinders
- 3/16" and ¼" diameter steel rod
- Nylon string

The team's gearbox is capable of a 1:100 gear ratio. The bottom two shafts have a diameter of 3/16" and the upper two shafts have a diameter of 1/4". A larger diameter rod was used to support the higher forces acting on the last two gear-sets of the gearbox. The first two gear sets had a ratio of 1:5 using a 10 tooth pinion driving a 50 tooth gear. The last two gear sets had a ratio of 1:2 using a 20 tooth pinion driving a 40 tooth gear. With the given gear sets, the team wanted to maximize the amount of torque from the motor. There is a multi-diameter spool that will be used with the 1:50 and 1:100 gear ratio, hence the longer upper shafts.

C. Powertrain

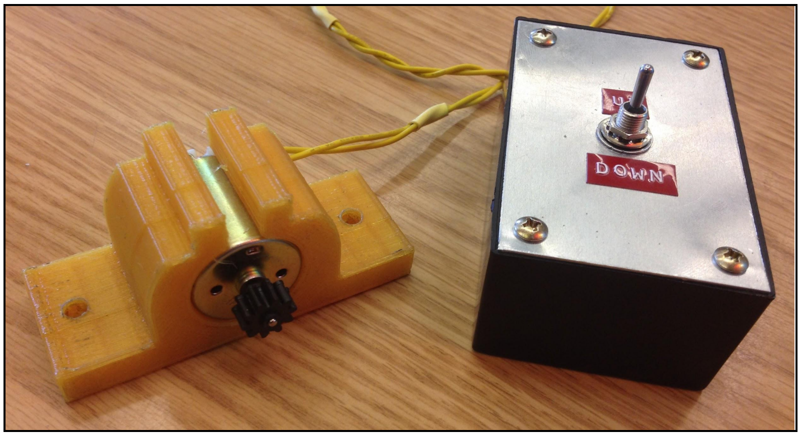


Figure 5: The motor housing and battery container/switch.

Materials

- Mabuchi RE280 DC motor
- 3D-Printed, ABS motor housing
- T10 Nylon pinion
- 2 AA Batteries + Holder (3V power source)
- DPDT switch
- 2”x3”x1.5” Project Box

The Shigley Hauler is driven by a Mabuchi RE280 DC motor that was powered by two AA batteries (3 Volts). The motor was press-fitted into the ABS motor housing to prevent the motor from rotating during operation. A 10-tooth nylon pinion connected to the shaft of the motor. For convenience, a double-pole, double-throw switch was used to allow the operators to wind or unwind the spool, which was useful during testing and the final competition.

III. STRENGTH AND RIGIDITY ANALYSIS

The team chose to use cold-drawn steel rods of diameters 3/16” and 1/4”. The choice of material was based on the material’s high yield strength and inexpensiveness; the material properties can be seen in Table 1 below. The 3/16” diameter rods were used on the first two gear shafts (in respect to the motor) where the bending forces are the lowest, and the 1/4” diameter rods were used on the last two gear shafts where torques, forces, and bending moments were the highest. During all the tests and competition, the shafts did not show any signs of bending. See Appendix C for a rough analysis of the first three shafts under no load. An FEA on the upper shaft that holds the spool can be seen in Figure 6, the calculations for these forces can be seen in Appendix, Figure C5. The FEA was done solely on the shaft that would hold the spool since it would experience the largest loads. THe team made the assumption that the other rods would experience lighter loads. Thus if the shaft that held the spool did not deflect substantially the other rods would not deflect. Figure 6 shows the von Mises stress of this rod under the heaviest load condition of five Shigley’s. In this image it can be seen that the largest stress would occur at the location of the spool. This result was expected due the large load of five Shigley's acting upon the shaft, which produced a force of 108.05 N in addition to the weight of the cart. The stress at this location produced a maximum displacement on the shaft of 0.0399679mm. Figure 6 shows an over exaggeration of this displacement. Figure 7 shows the parameters and assumptions that were made to produce a FEA simulation of the stresses on the shaft.

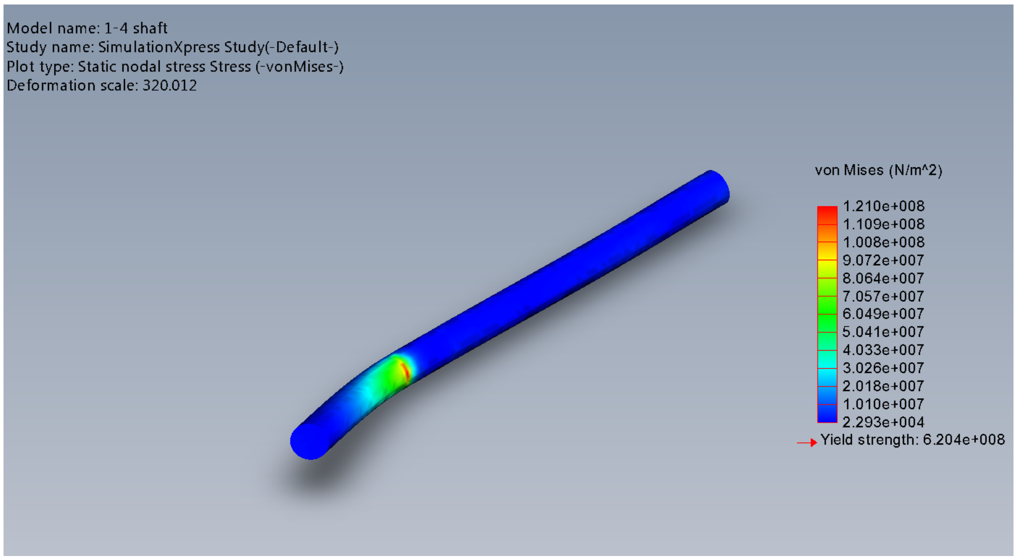


Figure 6: FEA of the upper 1/4" shaft when the Hauler is pulling 5 Shigley's at 60 degrees (worst case scenario)

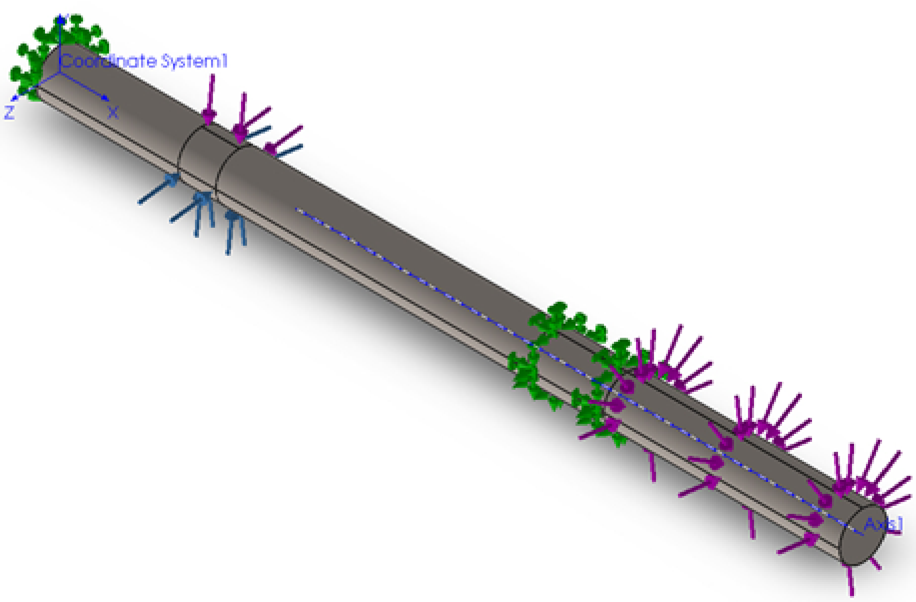


Figure 7: FEA of the upper 1/4" shaft when the Hauler is pulling 5 Shigley's at 60 degrees (worst case scenario)

Table 1: Properties of Cold Drawn Steel

Properties of Cold Drawn Steel				
Yield Strength kpsi	**Tensile Strength kpsi**	**Elongation in 2 inches %**	**Reduction in Area %**	**Brinell Hardness H_b**
44	**53**	**20**	**40**	**105**

In addition, strength and rigidity analysis was done for the aluminum cart, to ensure good performance during the operation. The cart was made out of Multipurpose 6061 Aluminum; the material properties can be seen in Table 2: Using SolidWorks, a FEA analysis was done on the linkages. The displacement results can be seen below in Figure 8. The results from the FEA showed the maximum displacement of 0.04765 mm was satisfactory for the carrying Shigley's during the competition.

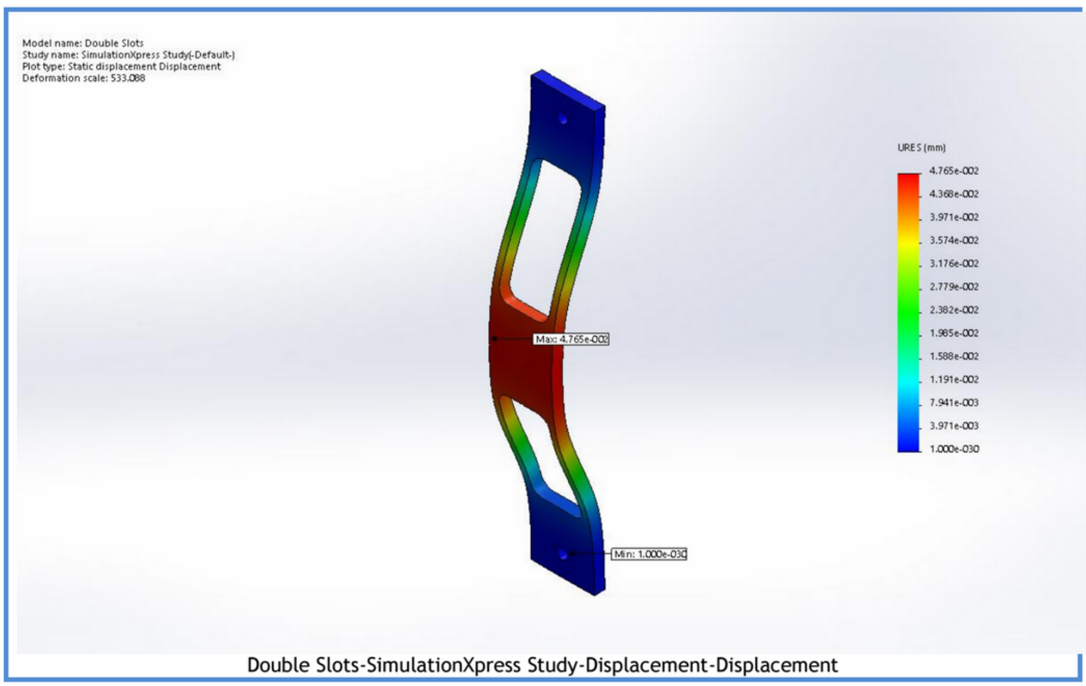


Figure 8 : FEA of one of the cart linkages for the heaviest load, 5 Shigley's -- i.e. 24 lbs.

Properties of Multipurpose 6061 Aluminum				
Yield Strength kpsi	**Tensile Strength kpsi**	**Elongation in 2 inches %**	**Reduction in Area %**	**Brinell Hardness H_b**
40	**45**	**17**	**35**	**95**

Below are the fundamental equations that will be used in the shaft analysis of the gearbox.

Symbol	Variable
W_t	Transmitted Load, lbf
H	Power, hp
V	Pitch-line velocity, ft/min
V	Pitch diameter, in
n	Angular velocity, rpm
M	Moment, lb in
σ	Stress, psi

$$W_t = 33000\frac{H}{V} \tag{1}$$

$$V = \pi dn/12 \tag{2}$$

$$W_{radial} = W_t tan(pitch\ angle) \tag{3}$$

$$M = W_r * distance \tag{4}$$

$$\sigma = \frac{32M}{\pi d^3} \tag{5}$$

$$\sigma_{allow} = \frac{\sigma_{yield}}{F.O.S} \tag{6}$$

IV. FABRICATION / MANUFACTURING

The design calls for precision holes in the side plates for functional distance between the gear sets. Initial pitch diameters were estimated with a caliper and designed in Solidworks. To ensure those measurements were correct, an acrylic prototype was fabricated with a 2-axis laser cutter (Figure 11). The CNC mill was used to cut the side plates to equal size, and the precision holes were drilled with the tested coordinates (found using the estimated pitch diameters of the gears). To reduce the friction on the shaft as it rotates, nylon sleeve bearings were press-fitted into the precision cut holes. Using this technique, the team was able to test the estimated pitch diameters of gears -- i.e the best center-to-center distances between gears.

As the gear ratio increases, the amount of torque increases. For the higher torque gears, hubs were designed and manufactured to secure the gear to its respective shaft. Cylindrical aluminum stock was lathed down to create the nipple for the set screw and the hubcap to precise specifications. Next, the CNC mill was used to cut the islands that would extend from the hub through the gear. Finally, the center and set screw holes were drilled and tapped. The spool was also manufactured at the mill. Our spool shape is designed to have many 4 different diameters so that the spool radius can be changed easily depending on the load.. The lathe was used to create equally spaced, varied diameters, with nipples for set screws.

For user friendliness, a project box was used to house the 3V power source (two AA batteries) and a Double Pole Double Throw (DPDT) switch. The DPDT switch allowed the user to determine the rotational direction of the spool -- i.e. determine whether the cart is moving up or down the ramp.

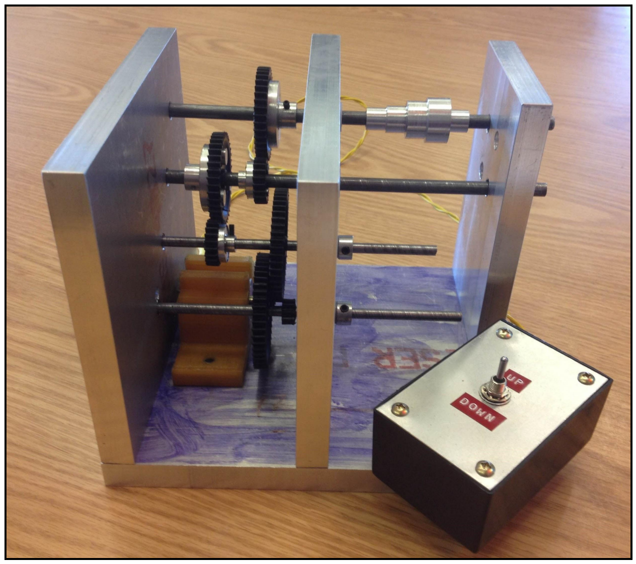


Figure 9: The fully-assembled Shigley Hauler (before testing)

The cart (shown in Figure 10) was made from aluminum flat bars. The bar dimensions were 1½ in wide with a ¼ in thickness. The material, aluminum, was chosen because it would be light yet strong enough to hold the weight of the Shigley books. The main design feature for the cart was to fabricate two L bars using the aluminum flat bars and connect the two with another aluminum flat bar in between them. To construct the L bars, both screws and added hardware were considered to connect the two aluminum bars. However, due to the fact that the bars had a thickness of ¼ in, it would be difficult to find suitable screws or bolts strong enough to sustain the weight of the books. If brackets were to be used, the overall weight of the cart would increase. Thus, to prevent these issues, the team opted to TIG weld the two bars together.

The TIG weld would be more cost efficient and would produce a stronger support than the bolts. However, even without using hardware or bolts, the stock material was heavier than the team's original assumptions. In order to reduce the weight of the aluminum flat bars, the mill was used to cut out pockets within the bars. Based on the cart design, it was concluded that the bottom aluminum flat bar of the L-bracket would sustain the greatest force. A stress analysis calculation and FEA was done for this member of the cart, based on the material properties of 6061 and using the weight of five Shigley books (a force of 52.5 N). The FEA calculated the maximum deflection in the bar to be 0.0476469 mm. Since the cart only needed to perform a total of seven runs, this deflection is within the allowable range of constraints that the team had designated for the cart.

The fabricated aluminum flats bars (the L-beams and the connect beam) were then attached to wheels. The wheels of the cart are composed of bearings that are press-fitted to a rod. Bearings were chosen to be used for wheels since they were very light, had the least resistance, and could handle the load of five Shigley books. Since the inner diameter of the bearings were larger than the rod's diameter, a hub was designed to allow it to be press-fitted to the rod. A lathe was used to create the hub (which could be press-fitted on the bearing and rod). The rod is connected to the cart using eye bolts that are screwed directly into the underside of the cart. The hub-bearing assembly is positioned on the rod so that minimal side movement is allowed.

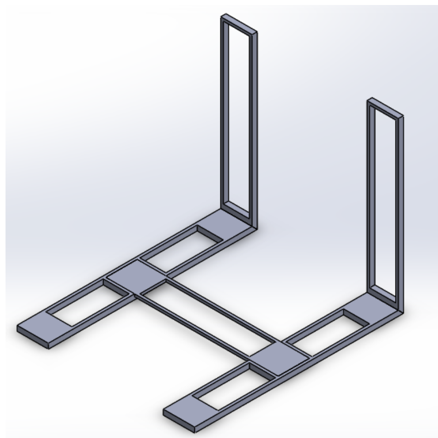


Figure 10: CAD model of the cart made out of 5 linkages

V. TESTING PROCEDURE

Before construction was done, a laser cutted prototype of the gearbox was made to ensure the estimated pitch circles were correct and whether the overall layout fit the needs of the design criteria. The prototype can be seen below in Figure 11.

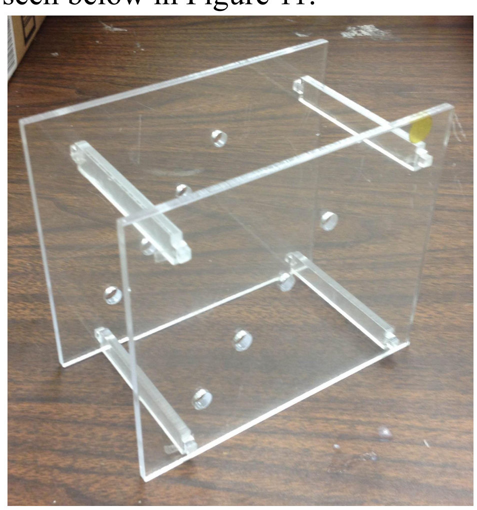


Figure 11: Laser-cutted gearbox prototype

The team tested the Shigley Hauler with test runs 1-4 (seen in Table 1) using the final gearbox assembly. Unfortunately, there were only two Shigleys between the teammates and alternative weights were used. This was done by renting out textbooks from the UC Davis Engineering Library that came within 4.2 lbs -- i.e. the weight of one Shigley. Each of the tests used the competition ramp to gather the best results. The variables that the team analyzed were:

1. String Performance (deformation and failure)
    a. Is there any deformation?
    b. Does failure occur?
2. Gearbox performance
    a. Are there any key points of friction/failure?
    b. Should lubrication be used (if so what kind?)
3. Spool performance
    a. Is the spool leading the string along the correct path -- i.e. is there any skipping?
4. Cart performance
    a. Is the cart holding the weight?
    b. Are there critical points of bending?
5. Overall performance
    a. Is the shigley hauler getting close to the theoretical time (See Appendix)

Table 2: The load case for each test run the shigley loader will perform under.

Test Run	Inclination Angle, θ_k	Load, *j*
1	20°	1 shigley
2	30^0	1 shigley
3	30^0	2 shigley
4	40°	3 shigley
5	40°	4 shigley
6	60°	4 shigley
7	60°	5 shigley

It was expected that the theoretical lift-times (found in Appendix A) would be faster than the real results seen in the competition. This is because the calculations made for the theoretical lift times did not take into account: 1) the friction caused by the nylon inserts, 2) the friction between the ramp and the cart wheels, 3) the weight of the cart.

VI. RESULTS

A. Test Results and Modifications

The team only ran tests on the gearbox and cart for tests 1-4 because of issues with finding loads equivalent to the heavier load cases and attaching them to the cart.

Since single-strand fishing line was cheaper than braided (and since it was rated for the loads), the team chose this type in the beginning. While heavier weights were tested on this string (equal to runs 5+), the single strand fishing line broke. When the team switched to a braided fishing line, the string stretched, but did not break for any load cases.

For the gearbox, the shafts and gears held up to all stresses from the loads. During the heavier load cases, however, the fourth gear in the gear train began slipping from its mounted position. The team found the reason for this was because the gear was not drilled through fully straight and therefore not mounted on the shaft fully straight when glued. When this gear was straightened out, the gear train could handle the higher loads much easier.

The cart held the weight of all 5 books very easily. The team noticed the cart tipping when pulling up higher loads due to the position on the cart where the string was attached. To counter this, the team manufactured a pole to be attached to the front of the cart with different notches for string attachment. The team could then attach the string to the cart at a height so that it matched the center of mass of the cart and books.

B. Final Performance in the Competition

Our gearbox was able to lift 3 out of the 4 tested loads in the competition. A pulley should have been used to lift the heaviest load in order to prevent stalling of the motor. The times were not optimal due to the lack of testing and finalizing our spool diameters.

For the first load case, 1 shigley at 20 degrees, our gearbox was able to pull it up in 18.6 seconds. This timing was fairly good compared to the other teams' timing for this load case. The second load case tested was the case for 2 Shigleys at 30 degrees. For this case our gearbox was able to pull up the Shigleys in 54.7 seconds. Compared to the other teams times, our timing was a little slower than the average. The next load case was 4 Shigleys at 40 degrees, where the gearbox pulled it up at an average time of 2 minutes and 11 seconds. Lastly, for the last case of 5 Shigleys at 60 degrees, our gearbox was not able to pull the Shigleys up as our system stalled.

C. Comparison of Theoretical and Real Lift Times

Based on the theoretical times and the final competition results, it was very apparent that the design did not follow the theoretical expectation. This comparison can be seen in the Table 3 below. It should be noted that the weight of the cart was taken into account for the theoretical times.

A possible reason for this large discrepancy could be in the use of nylon bushings to support the gear shafts. These bushings could provide a high amount of friction during operation and allowed the shafts to move due to the loose fit between the shaft and the bushing. This loose fit would cause inefficiencies in the transfer of energy between the gears. A better alternative to using bushings would be ball-bearings. Unfortunately, cost for these components made them unfeasible to implement for the final design.

An additional reason for the large time discrepancy was in the DC motor. During the testing the motor became noticeably weaker and starting sounding “weird”. Even after the team replaced the batteries that power the motor, the performance was still sub-par. If the team had a fresh-motor the competition, the lift-times would have been faster.

The plastic gears could also be the cause of the slower lift time since their flexible material would: 1) act as a damper when transmitting energy between gears and/or 2) cause misalignment between two gears. This was very apparent with the 50 tooth gears, which had a tendency of flexing during operation, even at low rpm. Metal gears would be the best solution, since their material will allow for rigid performance that would minimize the issues stated earlier.

Since friction between cart and the ramp were not taken into account for the theoretical time calculations, this would also add to the discrepancy between the theoretical and competition times. The team tried their best to reduce the friction between the cart and the ramp by having the cart rolling on a set of cartridge bearings. Having brand new bearings for the competition would have helped reduced the lift-time as the bearings on the cart were already pre-used.

Table 3: Comparison between the theoretical times and their respective lift times

Theoretical vs. Competitions Lift Times			
Runs	**Theoretical Time (s)**	**Competition Time (s)**	**Time Difference**
20°, 1 Shigley	2.20	18.6	16.4
30°, 1 Shigley	3.22	NOT TESTED	--
30°, 2 Shigleys	6.45	54.7	48.25
40°, 3 Shigleys	12.45	NOT TESTED	--
40°, 4 Shigleys	16.60	130	113.4
60°, 4 Shigleys	22.37	NOT TESTED	--
60°, 5 Shigleys	27.97	DNF	--

VII. CONCLUSION

The Shigley Hauler project demonstrated the practical application of a gearbox and the importance of testing known working conditions. In hindsight, designing the gearbox with required precision holes was time consuming and tedious. A more modular and adjustable slotted design might have been easier to manufacture. More parts would have meant an easier time delegating parts for team members to machines. The heaviest load case was not tested and therefore did not work on competition day. The gearbox would have lifted the necessary load if a pulley system had been implemented before the testing day. Also, the fastest lift times were not achieved because the team didn't optimize our spool. If the team had optimized the spool diameter for each load case, the times would have been much faster. All in all, the project was a great learning experience for mechanical design, manufacturing, and testing.

APPENDIX A - Data Tables

Table A1. Theoretical Times for the Shigley Hauler

Maximum Power (T = 0.00635 Nm & n = 4600 rpm)			
Torque Ratio/R	**Angular Velocity Ratio*R**	**Pitch Velocity (m/s)**	**Time (s)**
1163.94	8.59E-04	0.4138602573	2.20
1701.57	5.88E-04	0.283097089	3.22
3403.14	2.94E-04	0.1415485445	6.45
6562.50	1.52E-04	0.07340348112	12.45
8750.00	1.14E-04	0.05505261084	16.60
11788.85	8.48E-05	0.04086154514	22.37
14736.07	6.79E-05	0.03268923611	27.97

Table A2. Competition Times for the Shigley Hauler. Note: DNF (Did Not Finish)

Competition Results	
Runs	**Time (s)**
20°, 1 Shigley	18.6
30°, 1 Shigley	NOT TESTED
30°, 2 Shigleys	54.7
40°, 3 Shigleys	NOT TESTED
40°, 4 Shigleys	2:11
60°, 4 Shigleys	NOT TESTED
60°, 5 Shigleys	DNF

APPENDIX B - Motor Performance Sheet

WEIGHT：42g (APPROX)

RE-280RA/SA

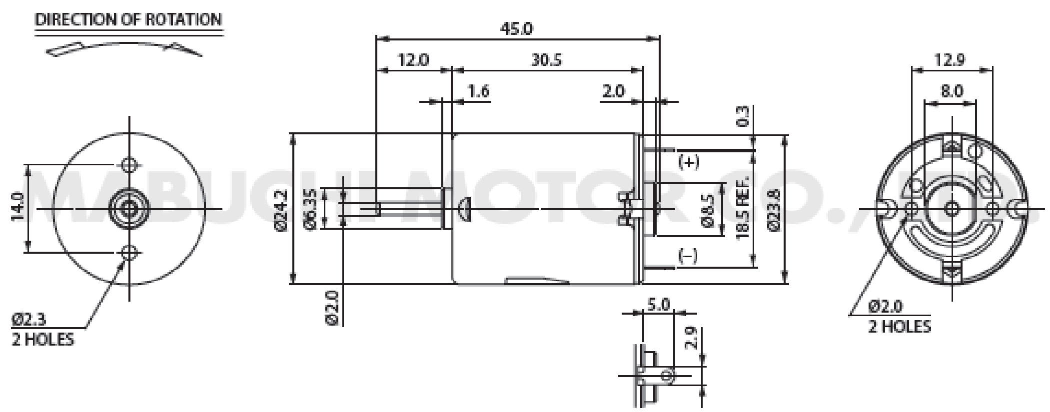
MABUCHI MOTOR

OUTPUT：0.8W～4.5W (APPROX)

金属ブラシ | Metal-brush motors | 金属电刷

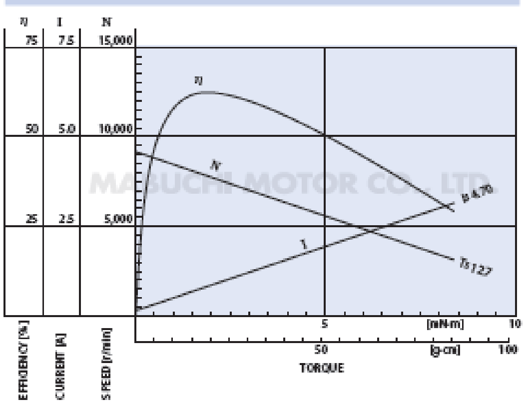
代表的用途	家電機器：マッサージャー／バイブレーター 玩具・模型
Typical Applications	Home Appliances：Massager / Vibrator Toys and Models
主要用途	家电机器：按摩棒 玩具、模型

MODEL	VOLTAGE		NO LOAD		AT MAXIMUM EFFICIENCY					STALL		
	OPERATING RANGE	NOMINAL	SPEED	CURRENT	SPEED	CURRENT	TORQUE		OUTPUT	TORQUE		CURRENT
			r/min	A	r/min	A	mN·m	g·cm	W	mN·m	g·cm	A
RE-280RA-2865	1.5～3.0	3V CONSTANT	9200	0.16	7770	0.87	1.98	20.2	1.61	12.7	129	4.70
RE-280SA-2865	1.5～4.5	3V CONSTANT	7100	0.16	6000	0.88	2.58	26.3	1.62	16.7	170	4.80
RE-280SA-2295	3～6	6V CONSTANT	9600	0.14	8150	0.78	3.27	33.3	2.79	21.6	220	4.40

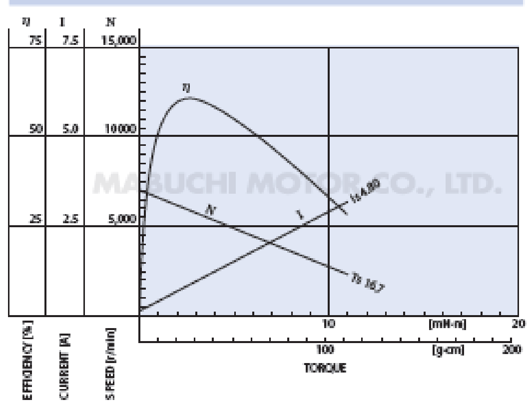


UNIT：MILLIMETERS

RE-280RA-2865 3.0V



RE-280SA-2865 3.0V

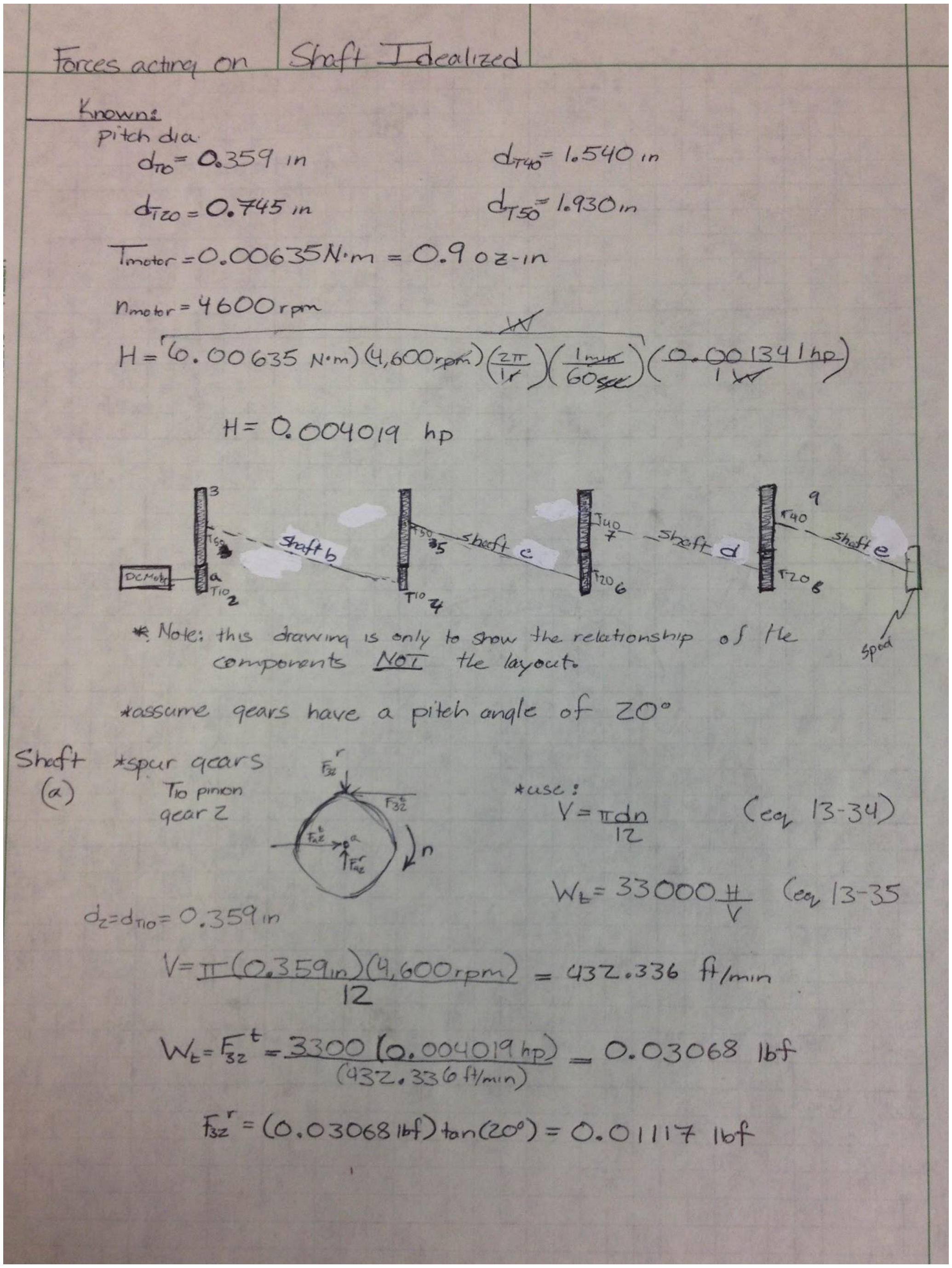


マブチモーター株式会社（本社営業部）
MABUCHI MOTOR CO., LTD.
（Headquarters Sales Dept. | 总公司 营业部）

千葉県松戸市松飛台430番地　〒270-2280　Tel. 047-710-1106　Fax. 047-710-1132
430 Matsuhidai, Matsudo City, Chiba 270-2280, Japan　Tel. 81-47-710-1106　Fax. 81-47-710-1132
E-mail : slsinq@mabuchi-motor.co.jp

APPENDIX C - Calculations for Shaft FEA Analysis

Note: The Calculations done in Figure C1, C2, C3 and C4 were for an unloaded scenario. The forces used for the “worst case” FEA were calculated using the methods found in Figure C5.



Forces acting on Shaft Idealized

Knowns:

pitch dia

$d_{T10} = 0.359$ in $d_{T40} = 1.540$ in

$d_{T20} = 0.745$ in $d_{T50} = 1.930$ in

$T_{motor} = 0.00635\,N\cdot m = 0.9$ oz-in

$n_{motor} = 4600$ rpm

$$H = \overbrace{(0.00635\,N\cdot m)(4{,}600\,rpm)\left(\frac{2\pi}{1r}\right)\left(\frac{1\,min}{60\,sec}\right)}^{W}\left(\frac{0.001341\,hp}{1\,W}\right)$$

$$H = 0.004019\ hp$$

* Note: this drawing is only to show the relationship of the components NOT the layout.

*assume gears have a pitch angle of 20°

Shaft (a) *spur gears

T10 pinion

gear 2

*use:

$$V = \frac{\pi d n}{12} \quad \text{(eq 13-34)}$$

$$W_t = 33000\frac{H}{V} \quad \text{(eq 13-35}$$

$d_2 = d_{T10} = 0.359$ in

$$V = \frac{\pi(0.359\,in)(4{,}600\,rpm)}{12} = 432.336\ ft/min$$

$$W_t = F_{32}^t = \frac{3300\,(0.004019\,hp)}{(432.336\,ft/min)} = 0.03068\ lbf$$

$$F_{32}^r = (0.03068\,lbf)\tan(20°) = 0.01117\ lbf$$

Figure C1: Calculations for the forces acting on the 10 tooth pinion from the DC motor

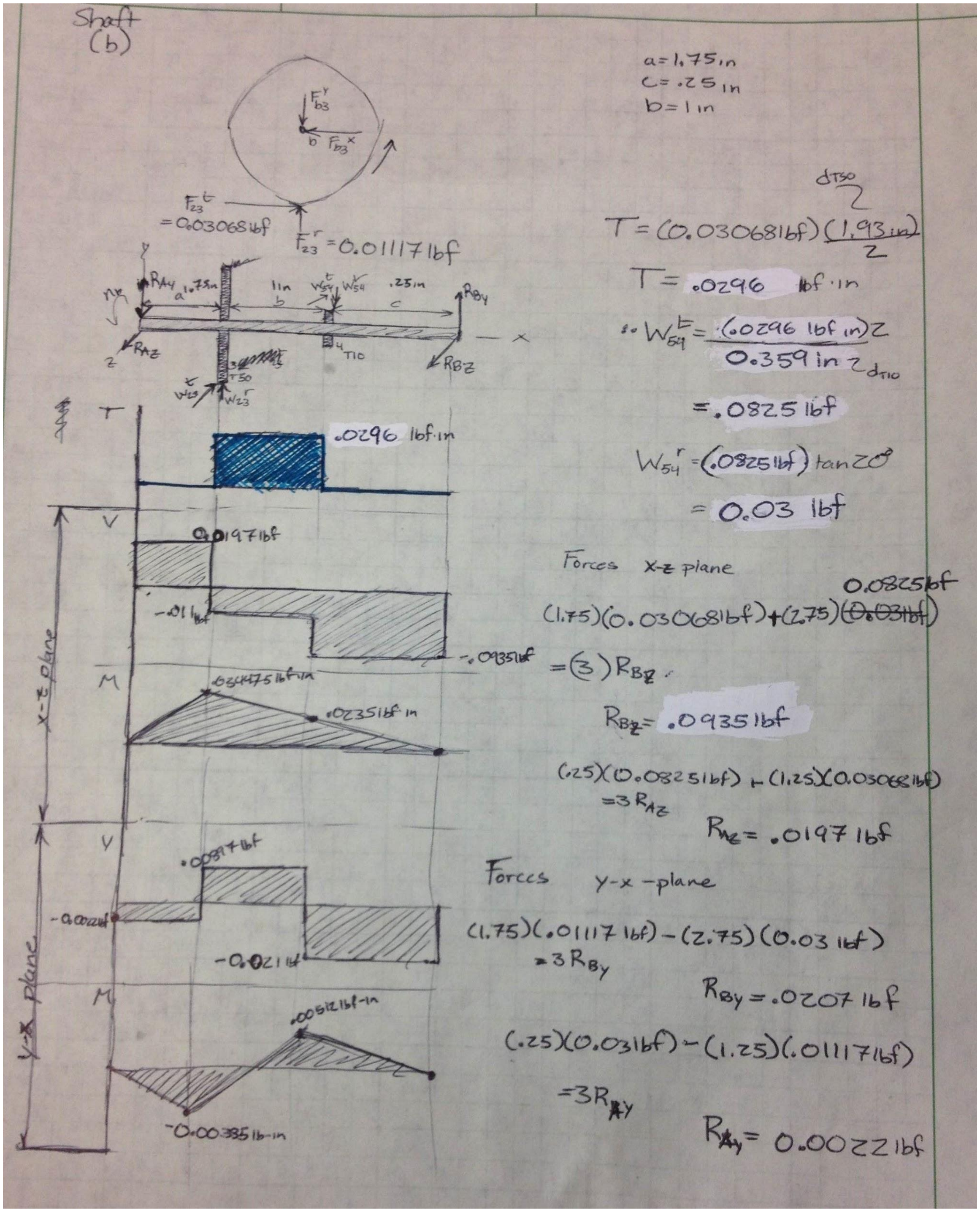


Figure C2: Calculations for the forces acting on shaft b, no-load

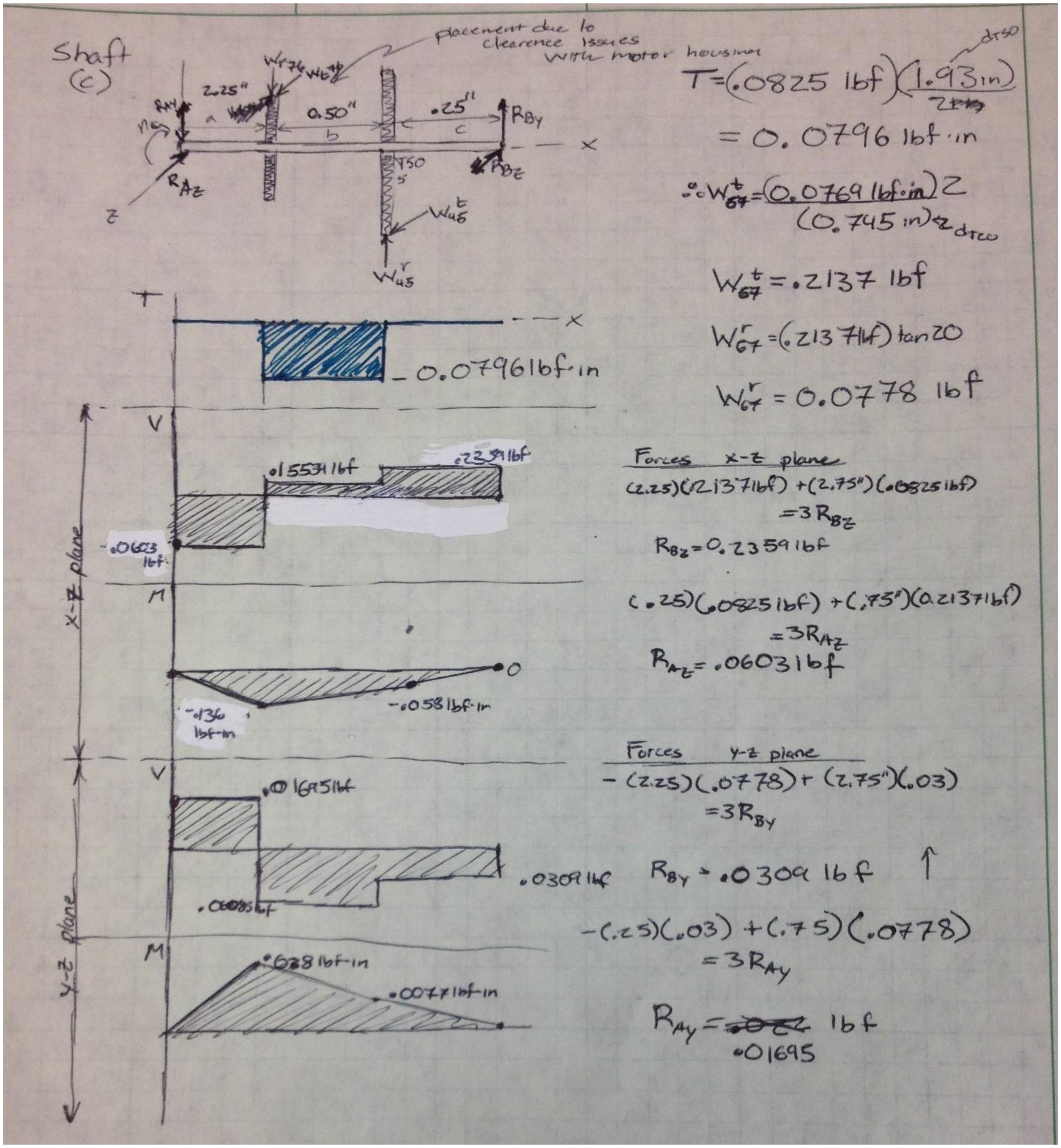


Figure C3: Calculations for the forces acting on shaft c, no-load

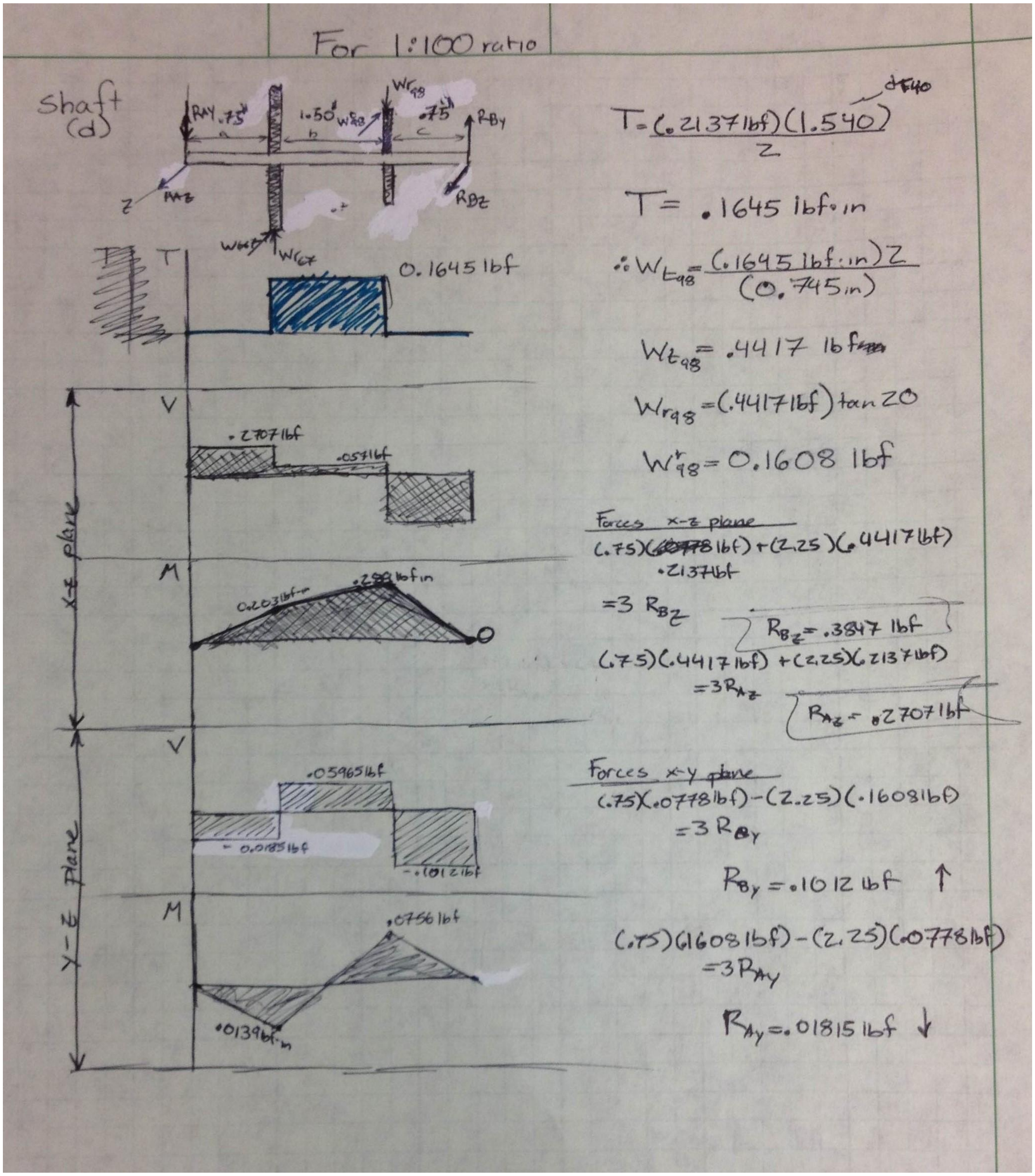
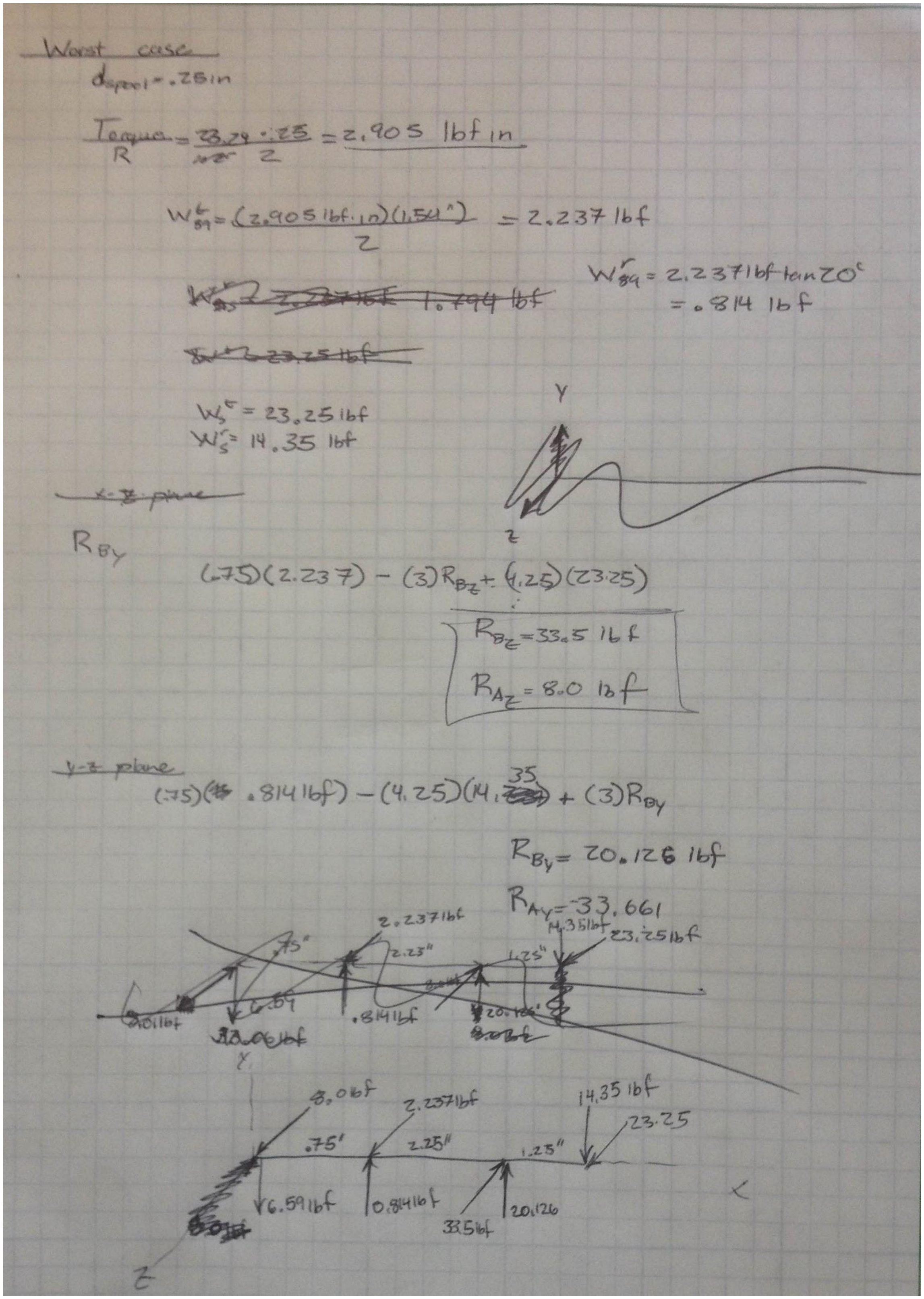


Figure C4: Calculations for the forces acting on shaft d, no-load



Worst case

$d_{spool} = .25 in$

$\frac{Torque}{R} = \frac{33.24 \cdot .25}{2} = 2.905 \text{ lbf in}$

$W^t_{89} = \frac{(2.905 \text{ lbf} \cdot \text{in})(1.54")}{2} = 2.237 \text{ lbf}$

$W^r_{89} = 2.237 \text{ lbf} \tan 20° = .814 \text{ lbf}$

$W^t_s = 23.25 \text{ lbf}$

$W^r_s = 14.35 \text{ lbf}$

~~x-z plane~~

R_{By}

$(.75)(2.237) - (3)R_{Bz} + (4.25)(23.25)$

$\boxed{R_{Bz} = 33.5 \text{ lbf} \quad R_{Az} = 8.0 \text{ lbf}}$

y-z plane

$(.75)(.814 \text{ lbf}) - (4.25)(14.35) + (3)R_{By}$

$R_{By} = 20.126 \text{ lbf}$

$R_{Ay} = -33.661$

Figure C5: Calculations used in the FEA for the worst-case scenario (high load and high ramp angle)