Advanced Mechanical Design, University of Rochester
Final Design Report
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DRILL CART: ALPHA

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ABSTRACT

Throughout the semester, a drill powered cart was designed, manufactured, assembled, and tested. This project consisted of multiple subsystems that included drivetrain, steering, ergonomics & braking, and frame. The purpose was to develop a mode of transportation that was not reliant on fossil fuels to be driven. During the development of the cart, there were several components to consider for the project's success such as identifying the requirements, specifications, and deliverables. For the mechanical analysis of the designs, Siemens NX was used by all subsystems to conduct thorough Finite Element Analysis (FEA) of cart parts. For manufacturing, various tools were used such as the Shop-Bot, mill, lathe, bandsaw, along with many others in the Rettner Machine Shop. After manufacturing, the cart was assembled by integrating all of the different systems into one product. Tests were then performed to determine the success of the initial requirements & specifications that were laid out prior to the start of the project.

Although the cart was assembled, through more time, the cart would have been better developed, driving smoother and efficiently.

PROBLEM DEFINITION

The American transportation system is heavily dependent on fossil fuels, which contribute to carbon emissions and consequently environmental degradation. Addressing this dependency is crucial for both the planet and its inhabitants. In terms of carbon emissions, the reduction of greenhouse gas has been particularly pivotal. This project aims to help fight the growing climate change problem by designing a car that is powered using an electrically powered drill. Inventions like this and other renewable technologies are a crucial step in reducing the world's carbon footprint. This approach not only supports current environmental priorities, but also positions the project within the context of future advancements in transportation.

REQUIREMENTS AND SPECIFICATIONS

Requirements:

- The cart **cannot** have **two wheels**.
- The cart will be able to safely hold one driver, who will **steer with their feet**.
- The accelerator, clutch, and brake will all be **hand activated** and **reachable** by the driver.
- The **clutch** will be **designed in-house** by the team.
- The **frame will be made of plywood** and must be **cut on the Shop-Bot in Rettner**.
- The cart will be operated with a **single drill**, mounted on the frame at the **rear** to transmit power to the wheels.
- The cart will have **proper safety precautions**, such as a seatbelt, helmet, horn, and guarding, outlined by the sponsor
- The cart will be able to complete **multiple** laps of the required course.
- Each cart team will use the **same tires and drill**.
- The cart will use **Ackerman steering** angles.
- The drill will lock and stay on.

Specifications:

- The cart cannot have a **breaking distance exceeding 15ft**.
- The cart must have a **turning radius smaller than 10ft**.
- The difference in **weight of the cart and driver** between both teams **will not exceed 5lbf**. **Sandbags shall be added** to the cart of the lighter team to account for weight differences.
- The maximum speed of the cart cannot exceed **25 mph**.
- **10 in. diameter wheels** will be used by **both teams**.
- The **same drill** will be used between both teams.
- The **plywood used will be 0.5 in.** in thickness.
- During the race the **driver must be switched every 2 laps**.
- The bottom of the **frame** must be at **least 8 in. above the ground.**
- The **length** of the cart must be *less* than **5 ft**.
- The **width** of the cart must be *less* than **4 ft**.

DELIVERABLES

- Gate A
- Gate B
- Gate C
- Prototype Drill Cart
- Safety Report
- Final Report with Test Data & Gate D
- Poster
- Website

CONCEPTS

Drive Train/Clutch:

Designs	Adjusting Belt	Friction Plate	Drill Activator
Ease of Assembly	-	-2	-1
Cad Modeling	-	0	0
Number of Parts	-	0	0
Ease of Use	-	1	-1
Size	-	1	0
Battery Efficiency	-	0	2
Total	-	0	0

Table 1: Drive Train Pugh Matrix

The drive train and clutch system utilized a two-belt system with two main pulleys, a 3 in. idler and a 2 in. driven pulley that was directly connected to the drill. These two pulleys were placed parallel of one another. A 5(a) in. driven pulley was placed below the 2 in. driver pulley. Lastly there was a 5 in. idler pulley used as a tensioner placed between the 3 in idler and 2 in driven pulleys.

All of these pulleys were inside of one 4L belt with a 37 in circumference. Attached to the 5 in. driven pulley was a second 5(b) in driven pulley in parallel on the same shaft. Attached to the second parallel pulley was a smaller 30 in belt that connected a 5(c) in driven pulley to the rear axle.
The clutch system works by allowing slack in the main 37 in belt. With the drill running the belt is able to travel between the 3 in idler pulley and the 2 in driven pulley, not the 5(a) in driven pulley. Upon pressing the tensioner pulley, the belt tightens, driving the 5(a, b, and c) in driven pulleys propelling the cart forward.

To hold the whole clutch/drive train system, 3 plywood frames were made to hold the aforementioned pulleys in place, using 3 aluminum rods as the pulley axles. Two of the frames were exactly the same with the third being slightly smaller as it only needed to hold two of the rods.

Ergonomics:

Seating Position	Normal (Upright)	Reclined Position
Cost	-	0
Ease of Assembly	-	0
Drag Coefficient Efficiency	-	1
Stability	-	-1
Ease of Use	-	1
Comfort Analysis	-	1

Table 2: Ergonomics Pugh Matrix

In the development of the drill cart, the ergonomic design of the driver's seat was paramount to ensuring both comfort and operational efficiency (Figure B1). After evaluating three potential seating positions, “Upright (~85º)” , “Reclined” (~55º), and “Further Reclined” (~30º), the “Reclined Position” was selected as the optimal choice for the driver's seat design (Figure B2). This decision was primarily influenced by comprehensive flow analysis results, which suggested a significant reduction in aerodynamic drag and improved driver comfort in the “Reclined Position” compared to the other options.

The flow analysis, detailed in the “Mechanical Analysis” section of this report, utilized a simulated environment where the human models were substituted with cylindrical proxies due to technical constraints in the chosen software. This analysis was pivotal in demonstrating that the “Reclined Position” not only minimizes the drag coefficient but also reduces the strain on the driver’s back, enhancing driver endurance.

In terms of materials, plywood was chosen for constructing the driver seat due to its balance of strength, flexibility, and ease of manufacturing. Plywood offers considerable advantages in terms of customization and shaping, which are critical in achieving the ergonomic requirements for the “Reclined Position”. The choice of material also supports a sustainable manufacturing practice and ensures that the seat can be economically fabricated without compromising on durability or driver comfort.

Frame:
When considering concepts for frame, it was essential that the number of wheels was predetermined in order to create sketches for possible designs. Once the number of wheels were set, four designs were sketched that contained various geometrical components. The four designs are shown in Appendix C, FC1. These designs were then modeled in Siemens NX and briefly analyzed using a 2D mesh with specified material properties, boundary conditions, and applied loads. Once these designs were individually analyzed such as maximum displacement, total mass, and “stiffness to weight ratio”, a pugh matrix was constructed to easily compare one another. Table 3 shows that the design with the least displacement is Design 3, a rectangular shape with curved end sections.

The design with the smallest total mass was Design 2. All of the designs were relatively the same in terms of their stiffness to weight ratios. Considering that minimizing displacement was crucial for the frame analysis, Design 3 was chosen to be optimized through topology optimization.

Designs	Total Mass	Displacement	Stiffness/Weight Ratio
1 (Baseline)	0	0	0
2	+	-	0
3	-	+	0
4	+	-	0

Table 3: Frame Pugh Matrix

Steering:

Steering Method	Steering Column	Direct Pedals
Cost	-	+

Ease of Assembly	-	+
Handling	-	+
Turning Radius	-	+
Ergonomics	+	-
Total	-3	3

Table 4: Steering Pugh Matrix

In the initial design of the steering system, two steering systems were considered. These choices, found in the Appendix D, FD3, are the “Direct Pedal Concept” and the “Pitman Arm Concept”. The first concept, the direct pedals, involves the steering pedals being mounted separately at each joint close to the outsides of the central bar. With a “Pitman Arm”, the angle of the Ackerman mechanism is controlled with a “Pitman Arm” which moves as a singular arm in the center of the central bar when turned. A Pitman arm design would be more difficult to produce and attach to the frame, but would have a greater possibility of ergonomic improvement, since the angle could be changed. It also had significant issues in other areas. Due to spatial considerations, the pedal placement was moved to the front of the system, directly connected to the steering knuckles.

One of the specifications for the cart was a turning radius of 10 feet. To produce this turning radius, the inner wheel of the cart had to move to an angle of around 23º relative to the back wheels which were on a single connected axel. The goal was to produce an angle of around 25º degrees, minimizing the force and movement of the driver as possible. A mechanism model of the situation was created, which found that the direct pedal design required significantly less movement from the driver to create this amount of movement, as shown in the Appendix D, FD4. With the results of this model, the “Direct Pedal” design was chosen.

Brakes:

Breaking Method	Mechanical Disc	Hydraulic Disc
Cost	+	-
Ease of Assembly	+	-
Required Force	-	+
Effectiveness	-	+
Thermal Eff.	-	+
Total	-1	1

Table 4: Brake Selection Pugh Matrix

For this project disc brakes were chosen. These come in two forms, mechanical and hydraulic. While mechanical brakes are significantly cheaper and easier to implement, a hydraulic braking system was chosen due to concerns over the amount of force required to fully stop our vehicle. Furthermore, hydraulic brakes were noted to also perform better in harsher weather conditions.

MECHANICAL ANALYSIS

Fatigue Analysis, Drive Rod:

As shown in Figure, using the properties of Aluminum 6061 from Siemens NX, the Goodman and Soderberg analyses were performed. Using an alternating stress of 262 psi and mid-range of 131 psi, both methods predicted an infinite life as the calculate Se value was far higher at 1632.810 psi. This makes sense as the torsional strength of aluminum is expected to be extremely high relative to any torque expected.

FEA Steering Knuckle Stress Analysis:

FEA of the steering knuckle design was done to determine the most efficient size for the steering knuckle in terms of height.

It was clear that bending would be a major issue. To combat this the design involved a one inch diameter tube with a half inch inner diameter as the main shaft of the knuckle.

This allowed connections for the wheel, pedal, and Ackerman steering system to be directly welded to it. This design was found to fit within a factor of safety of 2 with a margin of safety of 0.19. The analysis can be found in the Appendix D, FD2.

Flow Analysis (Fluids Simulation):

In the ergonomics section, a flow analysis was performed using Siemens NX to compare the aerodynamic profiles of two driving positions: the “Upright” and “Reclined” positions. The primary goal was to establish which position offered better aerodynamic efficiency, directly impacting the vehicle's performance and the driver's comfort.

A cylindrical model replaced the human model to simulate the driver's body for both driving configurations. Both simulations operated under a uniform wind speed of 25 mph, and the meshing was meticulously configured, with an air mesh size of 1.5 inches and a steel mesh size of 1.28 inches. The chosen tetrahedral (TET10) mesh type provided the necessary detail to accurately represent complex airflow patterns.

The analysis yielded notable differences in airflow velocity between the two positions. The “Upright” position encountered a maximum airflow velocity of 165.33 inches per second (Figure B3), while the “Reclined” position slightly improved this figure with a maximum velocity of 164.98 inches per second (Figure B4). More significantly, the minimum flow velocities revealed that the “Reclined” position had reduced aerodynamic resistance, with a smaller velocity of 483.77 inches per second compared to the "Upright" position value of 513.6 inches per second.

These results showed the selection toward the "Reclined" position with aerodynamic advantages. A lower drag coefficient was inferred for the "Reclined" position, suggesting it would be more efficient at higher speeds, potentially enhancing the drill cart's energy efficiency. This aerodynamic consideration played a critical role in determining the final driver seat design, ensuring the project aligns with the objectives of reducing drag and optimizing the driver's experience.

Steering Material Selection

Given the high forces involved, steel was chosen as the material used in the steering system. Steel also has the benefit of being easily "MIG" welded, which allows for the strange angles and ridges of our design to be manufactured while retaining maximum strength. Aluminum, another possible material, lacked the required strength, and is only weldable through "TIG" welding, which would have required extensive training

Frame Solid Mechanics:

The mechanical analysis of the frame consisted of using the calculated material properties to conduct FEA analysis on the chosen frame design. To calculate the mechanical properties such as elastic modulus and density, a flexural bending test was conducted using solid mechanics principles. Figure 2 in the appendix displays the setup of the flexural test along with the proper equations and calculations and also the material dimensions. The tests consisted of lying the piece of plywood on top and between two tables and allowing the weight of itself to be distributed equally. Once the test was conducted, the displacement at the middle was measured and as expected it had the highest value relative to any other position along the frame. Since the dimensions and weight of the plywood were already measured, the elastic modulus and density were then calculated. These values were significant in terms of implementing them in Siemens NX to create a new material property. Creating a new material was important as we wanted the FEA models to best represent real life.

NX Topology Optimization:

Once the material properties were determined and the frame design was selected, a topology optimization was conducted in NX by decreasing weight and keeping the stiffness constant. First, the frame design was set up using 2D mesh with the properties, boundary conditions, dimensions, and loads using solution 101 which is linear statics. Using these results, solution 200 Topology optimization was then implemented in order to determine which aspects of the current frame design can be removed to minimize weight. Using this analysis, the frame was updated to include the sketch of which parts can be removed. Although the outer shape of the frame was finalized, the interior was still subject to change depending on the concepts of other subsystems such as drivetrain, steering, and ergonomics.

Tolerancing, Drive Train:

The tolerance for the drive train were crucial because any misalignment of the pulleys or axles could create a large amount of unnecessary friction and stress to the drive train and cart as a whole. When initially designed, each hole was perfectly cut to fit the rods that they were meant for. However, even the smallest misalignment of these holes created a lot of friction. Furthermore, the placement of the frames themselves were also crucial for similar reasoning. Extensive sanding and filing of the frames were performed to ensure that little to no friction occurred when operating the drive train.

Bearing Analysis:

Selecting bearings for the drill cart involved a detailed analysis to ensure that the rear axle would be well-supported and aligned. The bearings needed to withstand the combined weight of the cart and the driver, totaling around 350 pounds. The choice fell on sealed steel ball bearings, capable of supporting dynamic loads up to 1600 pounds and static loads up to 1000 pounds, providing a safety margin well above the operational requirements. This robust specification led to a streamlined design approach for the rear axle, negating the need for additional structural support and contributing to an efficient assembly process. These bearings not only ensure the structural soundness of the cart but also simplify manufacturing and maintenance, which are key for the practicality and longevity of the design in a racing environment.

Fastener Analysis

The majority of fasteners used in this design were all SAE grade 5, meaning that they have a minimum tensile strength of around 105,000 psi. Since the bolts used were ½ inch, we do not anticipate that this design will produce the 52,500 lbsF required to break these bolts. Based on our calculations, the bolts connecting the steering system to the frame should only experience around 300 lbsF, which means that they are more than strong enough.

MANUFACTURING

Drive Train:

The primary part of manufacturing the drive train was creating the frames themselves. For this process the drawings of one of the larger frames was printed three times. It was then taped to the plywood so they could be easily cut out. One of the three was cut to length to be the smaller frame, as it just needed to have two of the same holes as the larger ones. Once cut to size. The frames had the holes cut by hand with hand drill. Once cut, sanding and filing were done extensively to ensure the placement of the holes lined up well as to prevent any friction of the rotating rods.

The rods that the pulleys spun upon were made from preexisting 0.5 in and 0.75 in diameter aluminum rods. These were cut to length of 5 in (0.5 in), 10 in (0.5 and 0.75 in diameter), and 16 in (0.5 in). The 0.5 in rods were then placed on the lathe to reduce the diameter so the pullies could be placed on them.

The 0.75 in diameter rod was then cut on the mill to have a keyway as this needed to power the rear axle. The rear axle was made out of a steel threaded rod and a keyway was cut at various points for the wheels, rear pulley, and brake disk. Lastly, keyway plates were cut to be placed on the wheels so they could also be powered.

Ergonomics:

The manufacturing report for the ergonomics subsystem covers the production processes of three key components: the driver seat, the brake system, and the rear axle mount.

The driver seat fabrication utilized plywood, chosen for its durability and comfort after shaping. A primary 12 x 12-inch plywood plate was supported by two additional plywood pieces (Figure B5). These components were precisely cut using a Shop-Bot for accuracy and assembled using wood glue and wood screws. To mitigate the risk of cracking during direct screwing, random wooden blocks served as connectors between the plywood layers. The completed seat assembly was then securely fastened to the frame using wood screws, ensuring a stable and comfortable seating area for the driver.

For braking, a cable disc system from a bicycle was procured. The manufacturing focus was on the caliper mounting plate and the brake disc plate, which required custom fabrication. An aluminum plate was cut with a band saw and further shaped with a mill to drill holes of 0.2 inches and 1/4 inches for the caliper and frame attachment, respectively. The caliper mounting plate was positioned under the frame to align with the brake disc. Likewise, the brake disc plate, also cut from aluminum, was milled for precision hole placement and then affixed to the brake disc, ensuring a reliable braking mechanism (Figure B7).

Plywood was again the material of choice for the rear axle mount, providing a balance of strength and ease of manufacturing. The design included a central bearing mount on a 6 x 5-inch main plate, flanked by support plates that connect to the frame. To avoid the limitations of metal brackets, wooden boxes were used (Figure B8). The mount components were cut with a band saw, drilled for bearing installation, and assembled using a combination of wood glue and a nail gun. This careful construction culminated in a secure and robust rear axle mount, which was then installed beneath the frame.

Frame:

Manufacturing of the frame consisted of using the Shop Bot to cut the 8ft by 4ft piece of plywood. Early stages of manufacturing consisted of only cutting the border of the frame and not the interior until the other subsystems were set on designs. The cad of the frame was used in NX to model the path the Shop Bot drill would take to cut the plywood. After a few more weeks of the other subsystems developing their design, the final frame would be cut in the Shop Bot that included three slots used for drivetrain and brakes. There were also cutouts needed for the wheels to move effectively. In addition to those, the topology optimization that was conducted earlier was used to cut out certain geometrical shapes in the frame. The stiffness supports were made of plywood as well and also cut using the Shop Bot where there are two bars and a support in the middle to stiffen the frame and minimize displacement. The stiffness was then added to the bottom of the frame after all systems were integrated.

Steering

The main steering knuckle is constructed out of a steel pipe with an outer diameter of 1 inch and an inner diameter of ½ inches, threaded rods of ½ inches for the Ackerman assembly and pedal attachment, and a threaded rod with a diameter of ⅝ inches for the wheel. It was manufactured by cutting the central pipe and threaded rods to size, adding holes and threading to the central pipe, attaching the threaded rods, and then welding the assembly in place. Spacers and connective pieces were cut out with the CNC machine and then connected to the steering knuckle. The assembly was then constructed with bolts and ball joint end caps for the threaded rods.

There was some torque on the ball joints teering knuckle to the central bar of the Ackerman steering system, which caused these pieces to spin around slightly. This caused a lock up when the cart was turned to a large angle. While we do not anticipate the system moving to this strong of an angle, two of the end caps were replaced with non-ball jointed end caps, in order to avoid this.

Benjamin Smoker	39
Karla Giron	61
Ryan Choi	36
Mallorie Plevyak	43

Table 5: Development Time (Hours)

Benjamin Smoker	30	$3,000
Karla Giron	33	$3,300
Ryan Choi	55	$5,500
Mallorie Plevyak	35	$3,500

Table 6: Manufacturing Time (Hours)

TEST PLAN AND RESULTS

Turn Radius Test: The test was devised to measure the turning capability of the drill cart. The cart is able to make a turn of radius 8 to 10 ft. This result confirms that the cart could successfully navigate a turn radius less than 10 feet.

Minimizing the Weight Difference: A crucial factor for fairness and performance in racing is minimizing the weight difference between competing teams. After weighing the drivers and accounting for the average driver weight of 219.31 pounds, coupled with the cart's weight of around 100 pounds, it was necessary to utilize sandbags to equalize the total weight with that of other teams. This test passed, ensuring that the weight difference between each team's driver and cart did not exceed 5 pounds.

Bottom Clearance Test: This test was to ensure that the car would not get stuck going up the ramp near Meliora and Rush Rhees Library. While we did not exactly meet the required 8 inches,

Vehicle Dimensions: The cart's dimensions were carefully measured to ensure they fell within the maximum allowed size. The length was recorded at 4.8 feet and the width at 3.5 feet, both well within the required dimensions, leading to a pass in this test category.

Specifications Tests	**Result (Pass/Fail)**
Turn radius less than 10 ft.	Pass
Minimizing the weight difference of each team’s driver	Pass
The bottom of the frame must be at least 8 in. above the ground	Pass
Vehicle dimensions	Pass
Max speed of 25 mph	Pass
Braking distance less than 15 ft	Pass
Plywood with 0.5 in. thickness	Pass
Wheels with 10 in. diameters	

INTELLECTUAL PROPERTY

Is the design or process patentable? Cite relevant existing patents and companies and individuals that are working in the area. Use the portion of the IP assignment related to your personal project for guidance on necessary content.

SOCIETAL AND ENVIRONMENTAL IMPLICATIONS

One of the major advantages of building a drill powered cart is that it’s not dependent on gas as a source of fuel. This greatly reduces the use off fossil fuels which mitigates carbon emissions and limits the detrimental impacts of climate change. This project adds to more tests and development when it comes to analyzing alternative forms of transportation that’s more environmentally beneficial. In terms of drawbacks, most of the cart parts are made of plywood. If mass production were to occur, this would increase the need for plywood which would probably contribute to the issue of deforestation since wood would be in higher demand. In terms of transportation, the cart would be a advantage since it’s easier to maneuver around

considering it’s small shape and it would be relatively cheap in comparison to modern cars.

RECCOMENDATIONS FOR FUTURE WORK

If given more time, there are many more changes that could have been implemented. A more complex and cost-effective braking system could have been designed. Further NX optimization could have been performed on the frame and the steering knuckles. The control systems for the cart could have been refined. Weight could have been removed from some of the structural parts of the drivetrain.
unless the reference starts a sentence in which case Eqn. should

ACKNOWLEDGMENTS

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REFERENCES

Shigleys

APPENDIX A: DRIVETRAIN

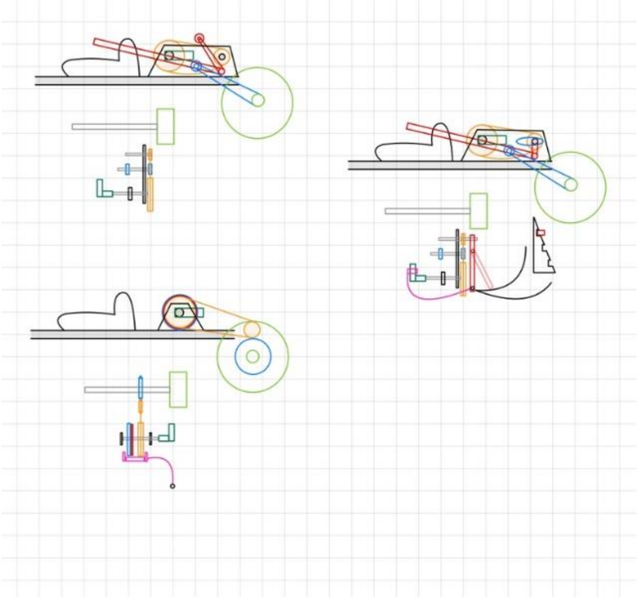
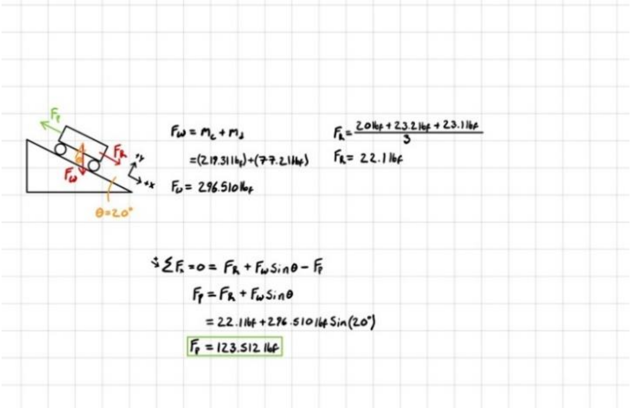


Figure A1: Clutch Design Concepts



$F_\omega = m_c + m_d$

$= (219.311 lb_f) + (77.2 lbf)$

$F_\omega = 296.510 lbf$

$F_R = \frac{20 lbf + 23.2 lbf + 23.1 lbf}{3}$

$F_R = 22.1 lbf$

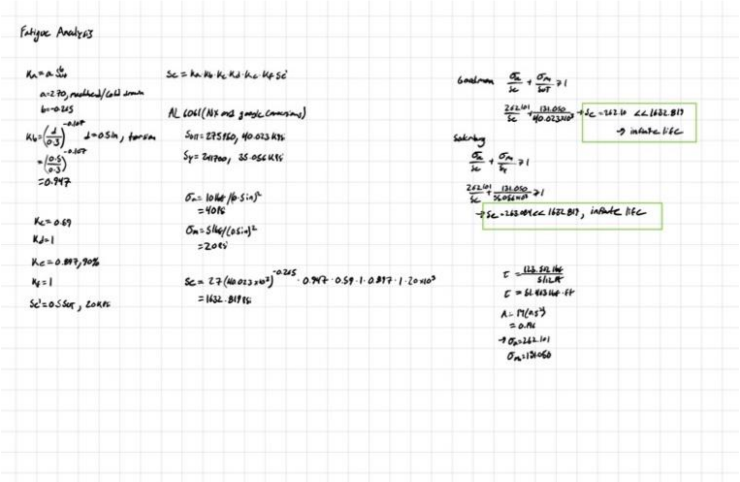
$\sum F_x = 0 = F_R + F_\omega \sin\theta - F_f$

$F_f = F_R + F_\omega \sin\theta$

$= 22.1 lbf + 296.510 lbf \sin(20°)$

$\boxed{F_f = 123.512 lbf}$

Figure A2: Incline Statics Analysis for Minimum Force Output



Fatigue Analysis

$K_a = a \cdot S_{ut}^{b}$

$a = 2.70$, machined/cold drawn

$b = -0.265$

$K_b = \left(\frac{d}{0.3}\right)^{-0.107}$ $d = 0.5 in$, torsion

$= \left(\frac{0.5}{0.3}\right)^{-0.107}$

$= 0.947$

$K_c = 0.59$

$K_d = 1$

$K_e = 0.897, 90\%$

$K_f = 1$

$S_e' = 0.5 S_{ut}$, 20 kpsi

$S_e = K_a \cdot K_b \cdot K_c \cdot K_d \cdot K_e \cdot K_f \cdot S_e'$

AL 6061 (NX and google Consensus)

$S_{ut} = 275960$, 40.023 kpsi

$S_y = 241700$, 35.056 kpsi

$\sigma_a = 10 lbf/(0.5 in)^2$

$= 40 psi$

$\sigma_m = 5 lbf/(0.5 in)^2$

$= 20 psi$

$S_e = 2.7(40.023 \times 10^3)^{-0.265} \cdot 0.947 \cdot 0.59 \cdot 1 \cdot 0.897 \cdot 1 \cdot 20 \times 10^3$

$= 1632.819 psi$

Goodman $\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{uT}} \geq 1$

$\frac{262.101}{S_e} + \frac{131.050}{40.023 \times 10^3} \rightarrow \boxed{S_e = 262.10 \ll 1632.819 \rightarrow \text{infinite life}}$

Soderberg

$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} \geq 1$

$\frac{262.101}{S_e} + \frac{131.050}{35.056 \times 10^3} \geq 1$

$\rightarrow \boxed{S_e = 263.084 \ll 1632.819, \text{infinite life}}$

$\tau = \frac{123.512 lbf}{5/12 ft}$

$\tau = 51.463 lbf \cdot ft$

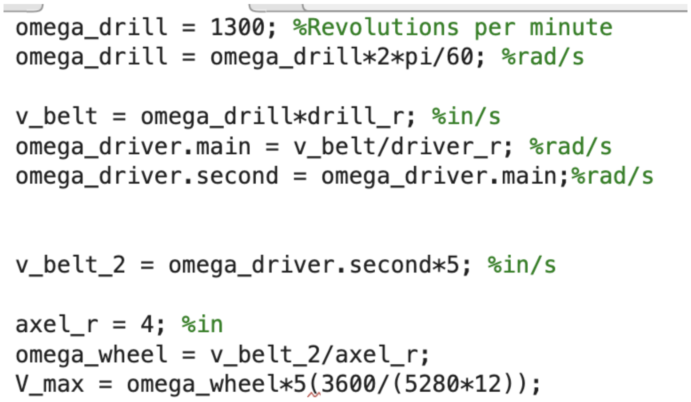
$A = \pi (0.5^2)$

$= 0.196$

$\rightarrow \sigma_a = 262.101$

$\sigma_m = 131.050$

Figure A3: Fatigue Analysis



```
omega_drill = 1300; %Revolutions per minute
omega_drill = omega_drill*2*pi/60; %rad/s

v_belt = omega_drill*drill_r; %in/s
omega_driver.main = v_belt/driver_r; %rad/s
omega_driver.second = omega_driver.main;%rad/s


v_belt_2 = omega_driver.second*5; %in/s

axel_r = 4; %in
omega_wheel = v_belt_2/axel_r;
V_max = omega_wheel*5(3600/(5280*12));
```

Figure A4: Maximum Speed Pulley/Gear Ratio Calculations

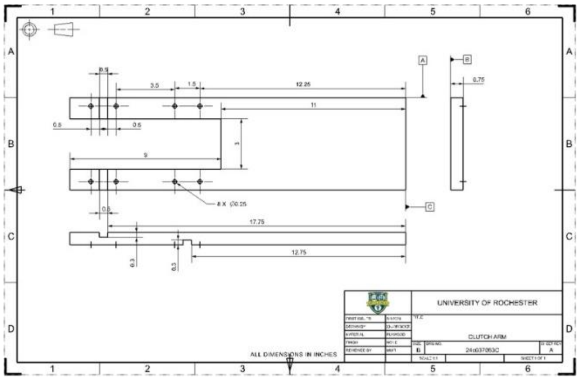


Figure 5: Final Clutch Arm Dr

APPENDIX B: ERGONOMICS & BRAKE

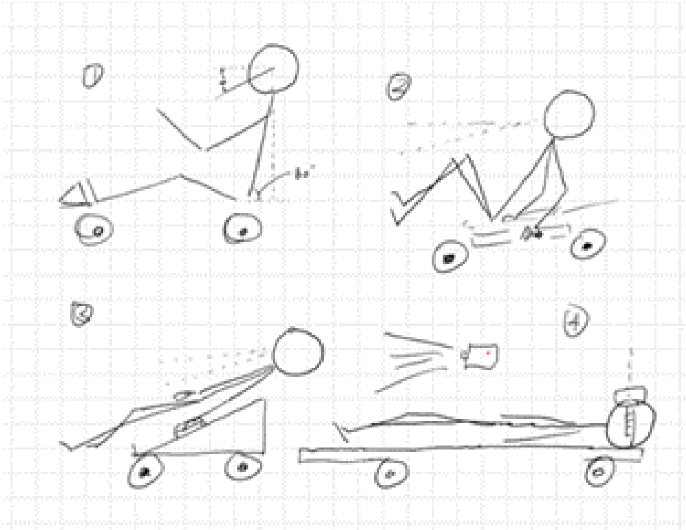


Figure B1: Seat Concept Sketches

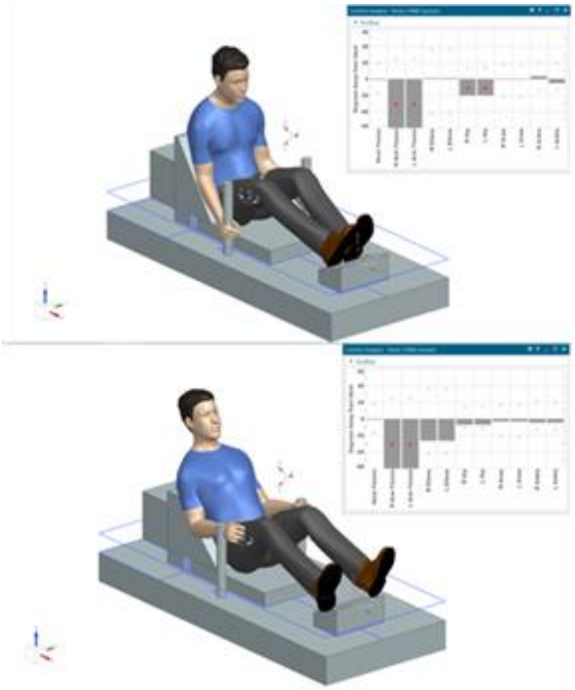
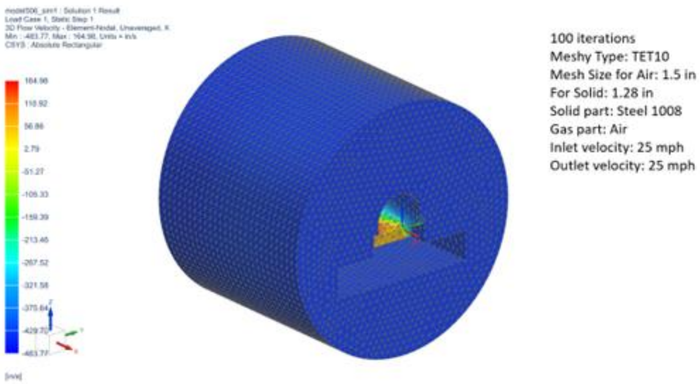


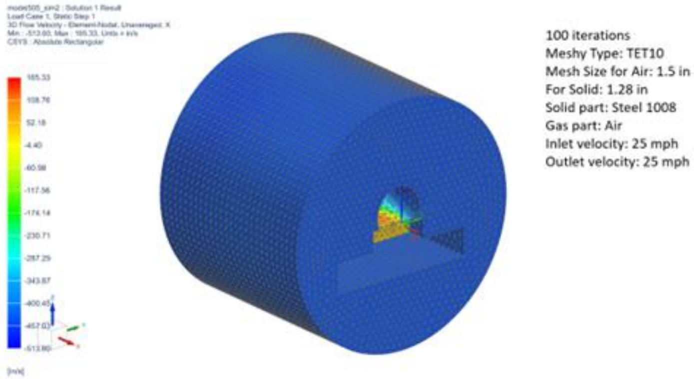
Figure B2: Comfort Analysis of seating position



model506_sim1 : Solution 1 Result
Load Case 1, Static Step 1
3D Flow Velocity - Element-Nodal, Unaveraged, X
Min : -483.77, Max : 164.98, Units = in/s
CSYS : Absolute Rectangular

100 iterations
Meshy Type: TET10
Mesh Size for Air: 1.5 in
For Solid: 1.28 in
Solid part: Steel 1008
Gas part: Air
Inlet velocity: 25 mph
Outlet velocity: 25 mph

Figure B3: Flow Analysis for Upright Seating Position



model505_sim2 : Solution 1 Result
Load Case 1, Static Step 1
3D Flow Velocity - Element-Nodal, Unaveraged, X
Min : -513.60, Max : 165.33, Units = in/s
CSYS : Absolute Rectangular

100 iterations
Meshy Type: TET10
Mesh Size for Air: 1.5 in
For Solid: 1.28 in
Solid part: Steel 1008
Gas part: Air
Inlet velocity: 25 mph
Outlet velocity: 25 mph

Figure B4: Flow Analysis of Reclined Seating Position

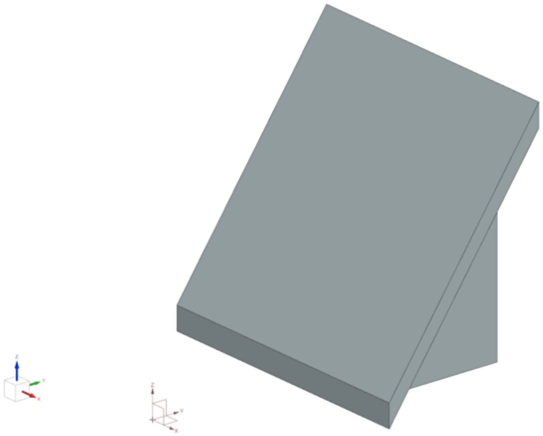


Figure B5: CAD Render of One Rear Axle Mount

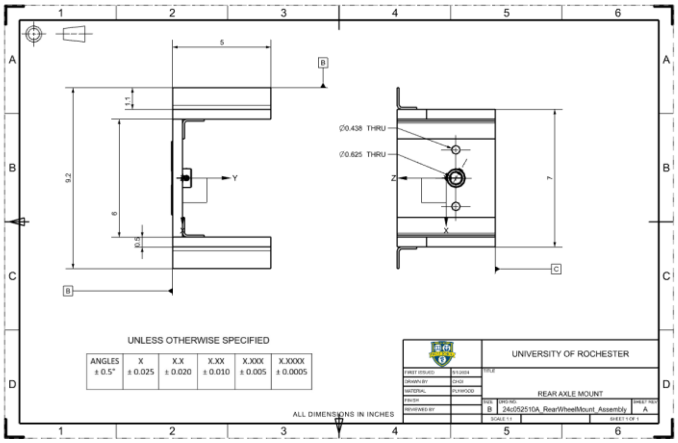


Figure B6: Rear Axle Mount Drawing

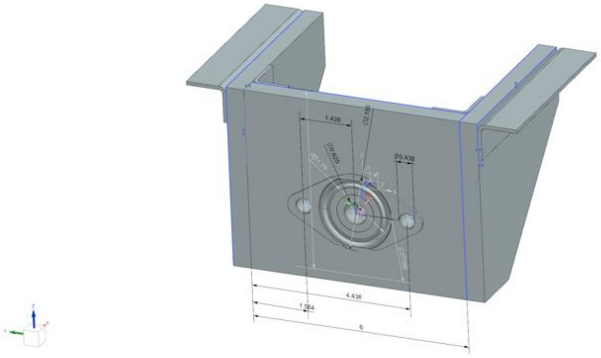


Figure B7: Final Seat CAD Render

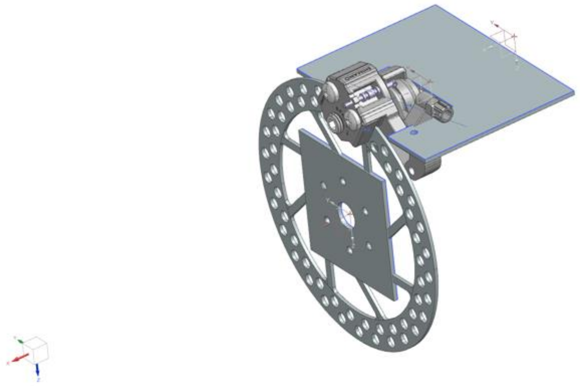


Figure B8: Final Brake Mount

APPENDIX C: FRAME

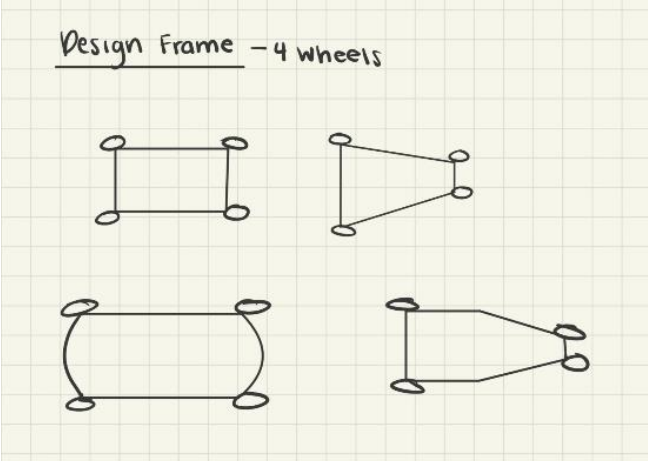
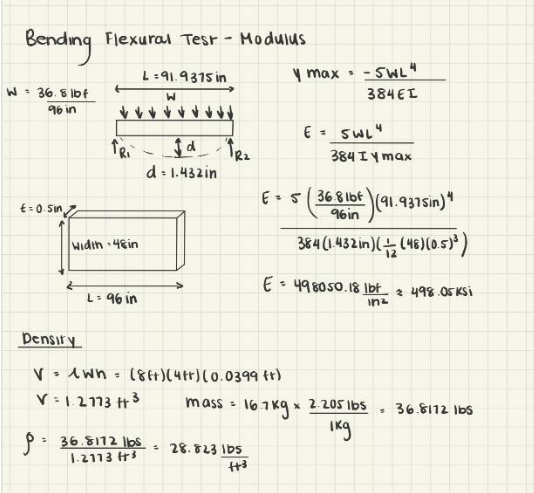


Figure C1: Frame Concept Sketches



Bending Flexural Test – Modulus

W = 36.8 lbf / 96 in

L = 91.9375 in

d = 1.432 in

R_1 R_2

t = 0.5 in

width = 48 in

L = 96 in

$$y_{max} = \frac{-5WL^4}{384EI}$$

$$E = \frac{5WL^4}{384 I y_{max}}$$

$$E = \frac{5\left(\frac{36.8\,\text{lbf}}{96\,\text{in}}\right)(91.9375\,\text{in})^4}{384(1.432\,\text{in})\left(\frac{1}{12}(48)(0.5)^3\right)}$$

$$E = 498050.18\,\frac{\text{lbf}}{\text{in}^2} \approx 498.05\,\text{ksi}$$

Density

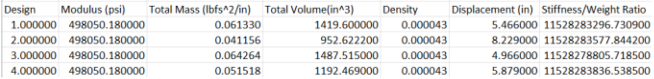
$V = lwh = (8\,\text{ft})(4\,\text{ft})(0.0399\,\text{ft})$

$V = 1.2773\,\text{ft}^3$

$$\text{mass} = 16.7\,\text{kg} \times \frac{2.205\,\text{lbs}}{1\,\text{kg}} = 36.8172\,\text{lbs}$$

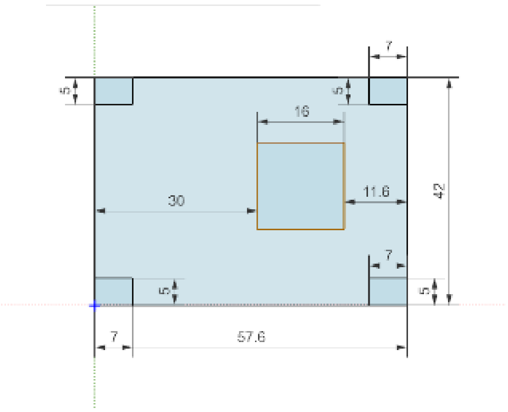
$$\rho = \frac{36.8172\,\text{lbs}}{1.2773\,\text{ft}^3} = 28.823\,\frac{\text{lbs}}{\text{ft}^3}$$

Figure C3C2: Modulus and Density Calculations

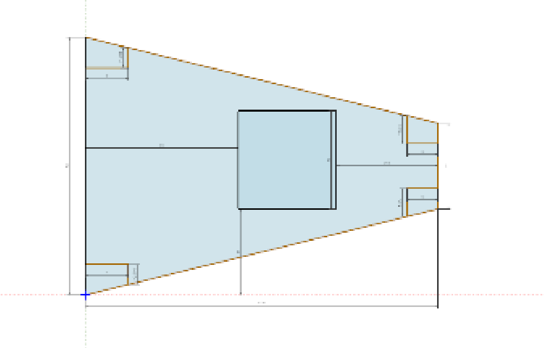


Design	Modulus (psi)	Total Mass (lbfs^2/in)	Total Volume(in^3)	Density	Displacement (in)	Stiffness/Weight Ratio
1.000000	498050.180000	0.061330	1419.600000	0.000043	5.466000	11528283296.730900
2.000000	498050.180000	0.041156	952.622200	0.000043	8.229000	11528283577.844200
3.000000	498050.180000	0.064264	1487.515000	0.000043	4.966000	11528278805.718500
4.000000	498050.180000	0.051518	1192.469000	0.000043	5.879000	11528283836.538500

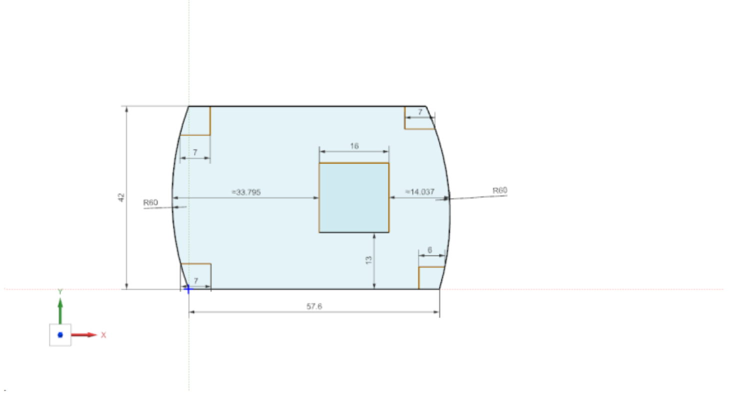
Figure C2: NX Values for Non-Optimized Frame Designs



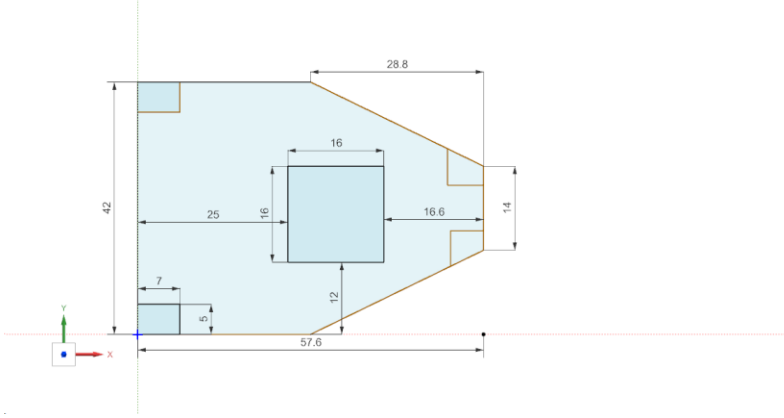
Design 1



Design 2



Design 3Desig



Design 4
Figure C4: NX Frame Concepts

Figure C3: Final Frame Technical Drawing

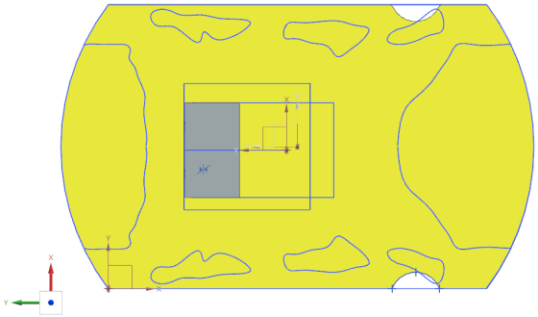


Figure C5C4: Topology Frame Optimization

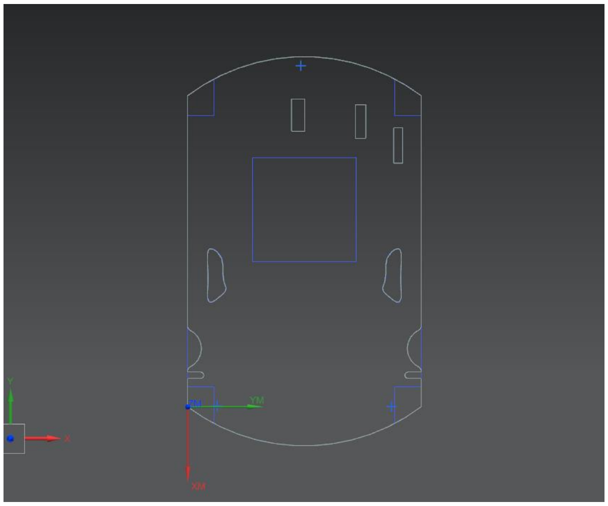


Figure C6: FrameC5: ShopBot Setup

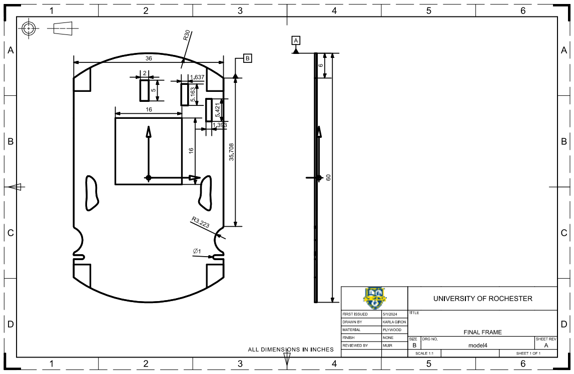


Figure C7C6: Final Frame Technical Drawing

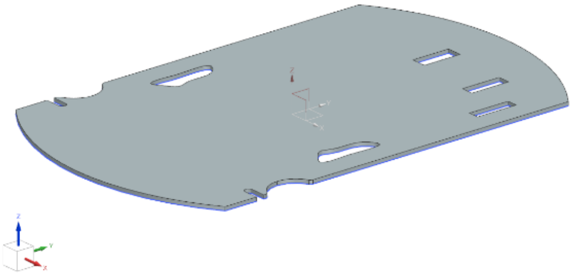
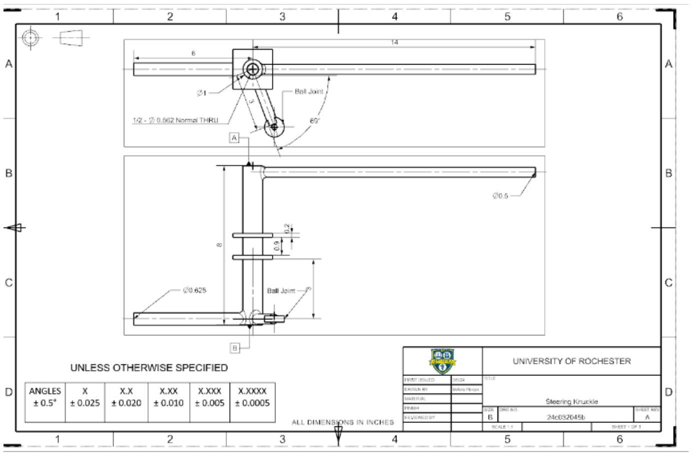


Figure C8C7: Final Frame CAD Render

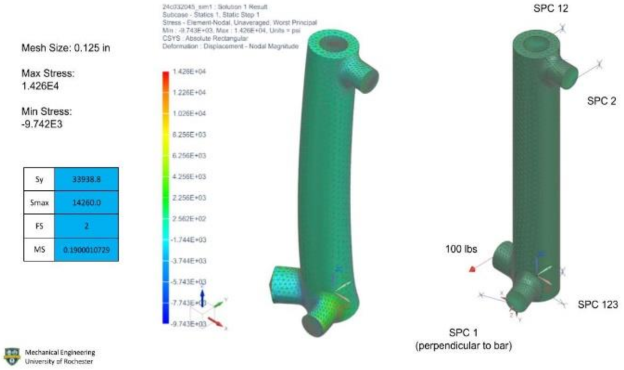
APPENDIX D, STEERING



UNLESS OTHERWISE SPECIFIED

ANGLES	X	X.X	X.XX	X.XXX	X.XXXX
± 0.5°	± 0.025	± 0.020	± 0.010	± 0.005	± 0.0005

Figure D1: Steering Knuckle Drawing



Mesh Size: 0.125 in

Max Stress:
1.426E4

Min Stress:
-9.742E3

Sy	33938.8
Smax	14260.0
FS	2
MS	0.1900010729

Figure D2: FEA Analysis of Steering Knuckle

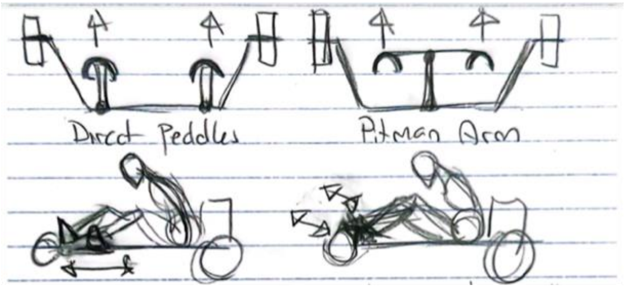


Figure D3: Initial Steering Designs

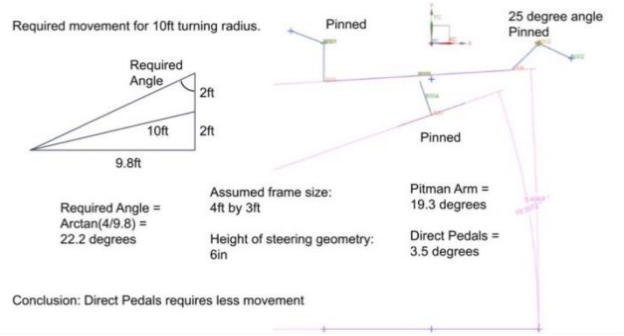


Figure D4: Turning Radius Analysis

APPENDIX E, DRILL CART ASSEMBLY

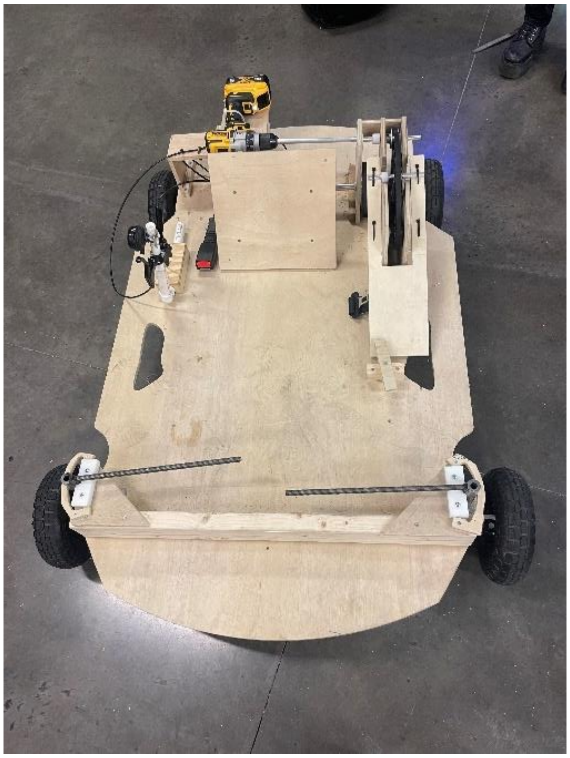


Figure E1: Front View of Cart

Figure E2: Side Profile of Cart Design