Final Design Report
Spring 2023

DRILL CART ALPHA

Brian Valerio **Peter Brodnik** **Sanjeev K C** **Walter Cruz**

ABSTRACT

The drill-powered cart allows for an alternative, more affordable and accessible option to that of your standard internal combustion engines. Through manufacturing and testing, this specific cart design is very limiting in the amount of load that it will carry. Using a hand drill powered by battery allows for only so much power/torque delivery to the drivetrain. There are many improvements that could be made to make this cart a more optimized version of what is it now. With more time permitted, parts could be manufactured and tested differently allowing for this optimization. This project has met all the requirements and still needs to be testing under specifications such as top speed and braking distance. Nonetheless, the drill-powered cart is an environmentally friendly alternative for countries and people that might need economically friendly options of transportations.

PROBLEM DEFINITION

There are negative effects of internal combustion engines (ICEs) in vehicles on the environment and daily lives, specifically in terms of pollution and carbon emission. There is a need for cost and energy efficient transportation alternatives that produce fewer emissions.

REQUIREMENTS, SPECIFICATIONS, DELIVERABLES

Requirements:

- The vehicle will be powered by a single electric drill, each team will use the same drill
- The vehicle body will be made from plywood
- The vehicle can be optimized by each individual team
- The vehicle must have a lap style safety belt and horn for safety
- Pinch points must be guarded and pass inspection by Professor Muir
- Payload will be standardized
- Cart must sustain the weight of the driver
- The vehicle must utilize a non-traditional steering system
- The vehicle must be able to maneuver the course
- Each team will use the same wheels

Specifications:

- The vehicle cannot exceed 25 mph
- Payloads must be within 5 lbf of each other
- Maximum brake distance of 15 feet at maximum speed
- The vehicle must have more than or less than 4 wheels
- Turn radius of less than 11 ft.
- Vehicle dimensions
- Cart must travel up a slope of 4.3 degrees

Deliverables:

- Drill cart operational on design day ready to race
- Technical report including test results
- Theory of operation manual
- Project presentation/website

CONCEPTS

Frame:

When exploring the different possibilities of a frame around an 8ft x 4ft x 0.5in plywood sheet, the first objective was to specify the number of wheels on the kart. After a discussion with the team, there was a census of using 3 wheels to satisfy the specification of the kart having more than or less than 4 wheels. Once that was determined, different concept ideas were explored for the shape of the frame as well as different approaches in executing the concepts. Furthermore, a Pugh Matrix to organize the concepts for the shape of a frame around a 3 wheeled kart was made to gather a better sense of which concept should be explored further. Four main concepts were considered in the Pugh Matrix. The first concept was a simple tadpole shape, usually in recumbent bikes, second concept as the tadpole shape but topologically optimized to further reduce weight in the frame, and the other two concepts being the tadpole shape facing the opposite way, as well as the sheet of plywood being the baseline (Reference Annex A). To explore even more concept ideas, a simple NX model was created to simulate the loads of a driver sitting on the frame supported at the wheel mounts. This was very rudimentary since the concept explore a general area being fully constrained and a concentrated load over an area to simulate the driver and a small dynamic load, as if they were bouncing of a bump in the road. Lastly, a flexural test was done to collect data points relating to a known load and the deflection of the plywood sheet in a 3-point bend/ simple supported beam configuration. (Reference Figure 1 and 2 in Annex A)

Frame Criterion 2.0	Simple Plank Rectangle (baseline)	Tadpole shape _/ (3WD)	2D topology with plank running along frame (front steer)	Inverted Tadpole shape /_\ (3WD)	Inverted 2D topology with plank running along frame
Time (design)	0	-	-	-	-
Weight	0	+	+	+	+
Flexibility	0	-	+	-	+
Manufacturability	0	=	=	=	=
Driver Comfort	0	=	=	=	=
Steering Space	0	+	+	-	-
Drivetrain space	0	-	-	=	=
Seat and brakes space	0	-	=	=	=
Total	0	-2	1	-2	0

Table (1): Pugh concept selection matrix for frame

Steering:

The requirement of a non-traditional steering system eliminated the opportunity to design steering around a central steering column. The most important aspect in the design process for steering was to incorporate an Ackermann steering linkage. This increases the efficiency of turning capabilities of the kart. There were two conceived options to control the linkage for the kart: one central controller i.e., a laterally moving joystick and two handheld levers controlling either end of the linkage close to the wheel. According to the Pugh concept selection matrix in Table (2) below, the hand levers was the obvious choice. With greater perceived ease of manufacturing and handling, this selection would assist in the time constraints of the semester. Ideas of steering mechanisms that incorporated hand levers were found from go kart forums and YouTube searches. (Reference Figure 3, 4 and 5 in Annex A)

Steering	Baseline (Steering Column)	Joystick	Hand Levers
Manufacturing	0	-	+
Assembly	0	0	0
Handling	0	-	+
Control	0	0	-
Turning Radius	0	0	+
Ergonomics	0	+	-
Total	0	-	+

Table (2): Pugh concept selection matrix for steering

Drivetrain:

The concept selection for the number of wheels was based on simplicity and easiness to manufacture and three wheels were finalized as a team. The Pugh selection matrix was created with complexity, maneuvering, and assembly as three criteria as shown in Table (3) below for the selection of drivetrain. Complexity refers to the difficulties in manufacturing various parts, maneuvering refers to the ease of driving the cart with less difficulty in the given course, and assembly refers to how easily are the individual parts assembled and how long it takes to do that. All four design choices are sketched as shown in the Figure 6 in Annex A. The four design choices were 1RWD 2FWS, 1RWS 2FWD, 2RWD 1FWS and 2RWS 1FWD.

Based on these criteria, one rear wheel drive and two front wheel steering cart design was selected. The team decided to move forward with this design because one rear-wheel drive provides stability and better weight distribution, and the front wheels being the steering wheels allow for tighter turns and greater precision. A detailed design on the drivetrain concept for the rear wheel is shown in Figure 7 in Annex A. It shows that chain system is used to connect the drill to the rear wheel without any rear axle or drive shaft. The team also decided to move forward with single gear system to avoid increased complexity as well as reduced reliability. The sprocket/chain mechanism was used to transfer the power from the drill shaft to rear shaft.

For the rear shaft, the rear wheel was bolted to the rear shaft preventing it from acting as a free wheel and the whole shaft would be rotated to move forward.

	1) 1 RWD 2 FWS	2) 1 RWS 2 FWD	3) 2 RWD 1 FWS	4) 2 RWS 1 FWD
Complexity	0	-	0	0
Maneuvering	0	-	-	-
Assembly	0	-	0	-
Total	0	-3	-1	-2

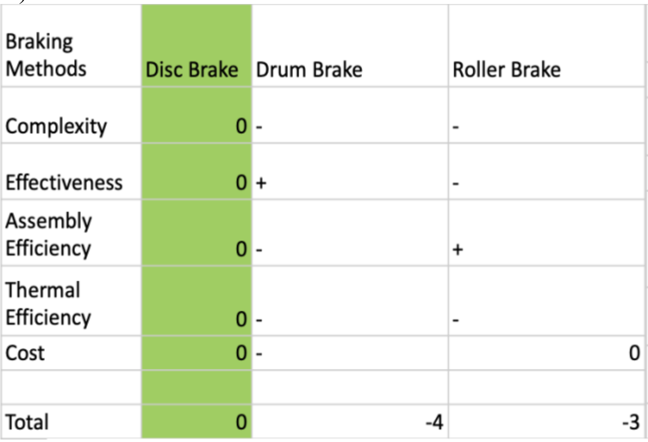
Table (3): Pugh concept selection matrix for drivetrain

Usability:

The first component of Usability is the braking system, the final decision made for the braking system based on the Pugh matrix in Table (4) is a mechanical disc brake. This braking system is a good iteration compared to the others because it is a good balance between complexity, effectiveness, and overall cost. In addition to this a deceleration of -7.073 ft/s^2 is considered for an effective braking system within 15 ft at 25 mph top speed.

The mounting of this braking system consisted of different iterations and designs. Driver engagement is considered when deciding if the system would require foot braking or hand braking. A handle braking system was decided for the braking system requiring about 9 lbf to come to a hard stop with cable braking system that was chosen. Using hand brakes allowed for easy incorporation of braking handles to steering mechanism.

In addition, there is a seating component which has 2 iterations. The final design iteration requires significantly less material and simplified design. Side components were created with slots in the material to allow for the back rest of the seat to be supported at an angle. This allowed for a heavier load to be placed on the back portion. An additional force is also created when adding L-brackets connected both the side panels and the back rest with wood screws. (Reference Figure 8 and 9 in Annex A)



Braking Methods	Disc Brake	Drum Brake	Roller Brake
Complexity	0	-	-
Effectiveness	0	+	-
Assembly Efficiency	0	-	+
Thermal Efficiency	0	-	-
Cost	0	-	0
Total	0	-4	-3

Table (4): Pugh concept selection matrix for braking method

MECHANICAL ANALYSIS

NX Stress Analysis:

When modeling the frame, the team needed to determine the mechanical properties of the plywood. This is crucial in modeling the computer simulation and design of the frame. One factor to consider is that this year's plywood material changed from Baltic Birch plywood to American Yellow Birch. A clear observation was the number of plies that the sheet was made of. Baltic being 6 compact layers and American being 3 thicker layers. This of course was taken into consideration give the global implications and the cost of Baltic Birch Wood when comparing it to American Yellow Birch. After this realization, one of the biggest issues that the team faced when modeling the Finite Element Model (FEM) of the frame was that plywood is not consistent and predicting its mechanical properties are not going to be accurate. To not only rely on internet data of American Yellow Birch Plywood; a flexure test was conducted to determine the Young's Modulus of the wood and model the wood properly on NX.

Setting up the frame FEM under a simple supported statics load case where the holes on the frame served as the constrained points. The rear holes simulated the rear wheel resting on the ground, constrained in the upward Z- direction. The suspension connection points were modeled as RBE2s to simulate the wheels outside of the frame and not mounted directly on the frame, similarly, constrained in the upward Z-direction. These constraints enabled the connection points to rotate about each axis to see the behavior of the frame once load was applied. As previously discussed, there was a fundamental analysis and there is a Structural FEA Analysis of the frame design that the team decided on. Based on the fundamental results and the primary design of the frame, it was intuitive to place supports underneath the frame to stiffen the frame and reduce flexure. (Reference Annex B)

Dynamic/Fundamental Analysis:

One of the challenges was to convert the torque from the hand drill to the torque that will push the cart forward using the right gear ratio. For this a fundamental analysis was done to figure out the torque required to move forward as well as the torque required for the cart to go through the ramp of 4.3 degrees. The gear ratios were then based off this two torque and the ratio required to go at allowable high speed. After deciding to use a single gear system with centrifugal clutch to overcome stalling of the hand drill, right gear ratio based on these three conditions were chosen. The baseline equation for this analysis is,

$$\tau_{output,\ normal} = W_{cart} * \mu_s * r$$

$$\tau_{output,\ ramp} = [(W_{cart} * \cos(\theta) * \mu_s) + (W_{cart} * \sin(\theta))] * r$$

$$Gear\ ratio = \frac{\tau_{output}}{\tau_{input}}$$

where W_{cart} is the weight of the cart, μ_s is the friction between the tire and concrete, r is the radius of the rear wheel, θ is the ramp angle and τ is the torque. The equation for gear ratio was used to calculate the gear ratio for normal condition and the ramp. Based on these calculations made in MATLAB (Reference Annex C), gear ratio of 2.8 was used with the maximum achievable speed of around 14 mph.

Fastener Torque Analysis:

The torque of the two nuts on the preload bar varied based upon the weight of the driver. The heavier driver required a greater torque. The higher torque of these two nuts creates a greater preload within the bar. The load forces the wheels inward creating a positive camber so that when the load of the driver is induced the wheels return to be as neutral as possible. Without the preload bar the wheels would have negative camber that greatly decreased the turning radius. Ideally there would be a spring placed in between the nut and the support block. The spring would have a known force of compression over the distance of the bar the exact torque of the nuts could be fine-tuned. This would be like the way coil overs are preloaded in modern cars.

Material Selection:

When designing the steering knuckle, there were two main deciding factors in the selection of AISI 4130 steel. The first was ease of manufacture. Knowing that there would be three separate pieces needing to be assembled, an easily and strongly weldable material was determined to be steel. Additionally with a Young's modulus of more than double aluminum this was the obvious choice. The steering knuckle would be under considerable stress (each experiencing 1/3rd the load of the driver and cart) and cannot fail. Despite steel being almost three times as dense as aluminum, the weight of the knuckles was not a significant enough factor to consider using it as the material of choice.

Bearing Analysis:

For this assembly self-aligning bearings were used for the drivetrain system. This meaning that the bearings had to be placed carefully on the shaft at the angle that was required to keep the shaft straight. This was a challenge seeing as though since it is self-aligning it needed to be tightened in place at the exact location, we need it. We did this by mounting the system completely all the way through to ensure that the shafts were straight and then tightened the bearings in place. Since it is such a vital piece in the drive train system, it needs to be properly lubricated and inspected to be sure that it is not out of alignment from where it is placed. This will help keep its desired life up for it to be used during a race.

Tolerance Analysis:

The largest trouble team came across during the assembly of steering system into the frame as the misalignment of wood cutouts held the steering knuckles out of place with the frame. The frame was cut on the ShopBot CNC machine with cut out semicircles of 1.25 inch to match flush with the steering knuckle steel tubing. This did not initially match and after painting the frame the connection was even worse. Extreme efforts of sanding and using a Dremel tool allowed for the tubing to fit flush to the frame. The tolerance of the ShopBot CNC drill bit was not considered before the cut. This heavily influenced the extra work in the make-it-work phase of assembly of the steering to the frame.

The next tolerance consideration was that of the 5/8-24 threaded steel rod that was used to fasten the frame and cutouts. These sandwiched the steering knuckles in place where the frame had the semicircles cut out. When aligning the wood cutouts, the holes were marked for drilling. The wood cutout pieces were not fastened together so that the holes could be drilled all at once. This would have alleviated any tolerance issues when putting the threaded rod through the five wood cutout pieces and frame. The 5/8th inch wood drill bit did not create holes that were aligned enough for the 5/8th in threaded rod to easily slide through. Extra steps were taken, specifically sanding, using the Dremel and widening the holes with the drill bit to ensure the holes aligned. This tolerance issue was greatly affected by user error and not at all by the tools themselves. Although a threaded rod may not perfectly fit through a wood cut out of the same size due to splintering, the appropriate precautions were taken to avoid similar issues with the assembly.

Fatigue Analysis:

The fatigue analysis was performed on the rear shaft made from 1045 Carbon Steel which had a diameter of 5/8 inch. According to the calculation made based on modified Goodman, the part had an infinite life. The stress analysis on the rear shaft was also performed to acquire the data. (Reference: Annex D)

Spring Analysis:

The spring analysis can be done on the spring used for the trigger to the drill. The spring will be in tension when the drill is engaged and will be within the elastic limit so that when the drill is disengaged, it goes back to its original position with very little to no displacement. The governing equation was F=kx, where k is the spring constant, x is the displacement of the spring of 38.1mm and F is the restoring force which is 40.043N.

MANUFACTURING

Frame:

Once the final design was decided, the frame needed to be cut. For this, the CAD model was programmed on NX Manufacturing to prepare the Shop Bot CNC Router. With the help of Professor Muir, we were able to cut the frame from our stock sheet of plywood. When the kart was being assembled, the team quickly realized that the frame would not be able to support the wheels and a driver, so, a plan to reinforce the frame from underneath was devised. The first iteration was running aluminum U-brackets underneath the frame with a bolted connection. Unfortunately, that did not provide enough stiffness and support, so it was decided to remove the U-brackets and implement the second iteration of using a system of 2x4 wood planks running underneath the entirety of the car to provide support to the rear wheels in a triangular configuration. The frame was then painted and assembled the rest of the systems. During initial testing, the frame sustained a lot of damage to the drivetrain side of things due to the ½" wood screws ripping out of the L-brackets attached to the frame. Therefore, an additional piece of plywood was placed below the drivetrain system where the drill mounts that allows the system to be one single unit that can be placed as one piece.

Steering:

The most manufacturing for the steering subsystem was the building of the steel steering knuckle. Steel was selected as the material for the knuckle because the front two wheels would need to withstand two thirds of the weight of the cart. Additionally, the magnitude of the moments on the two front wheels would be considerable in comparison to the single centrally located wheel in the rear. A 1/8th inch AISI 4340 low carbon steel sheet and tube was purchased to cut out the components for the two knuckles. The steel plate was cut using plasma cutting and the steel tube was cut using the horizontal band saw. With each of the individual parts cut out, run through the deburring wheel, and bent, they were ready to be TIG welded together. The bent pieces created the most difficulty for the welding. A flush connection was needed to create a secure weld and the initial way the pieces were bent did not satisfy this. The 90-degree bends in the drawing are not accurate to the actual steering knuckle. This affected the metrology of the assembly, which was foreseen but neglected due to a rush of time. The hole size cut from the plasma cutter was increased on the mill to the correct tolerance of the drawings so that the 5/16th screws would fit.

The next piece that was manufactured were the preload bar supports. These pieces were cut from scrap aluminum on the mill. Aluminum was selected for ease of manufacture on the mill. The stresses in the block would not be great enough to require a different material. All the other pieces were cut on either the vertical or horizontal band saw.

Drivetrain:

Aluminum 6061 was used for the drill shaft and 1045 Carbon Steel was used for the rear shaft because it supports one-third the weight of the whole cart along with driver as well as transmits the torque. It needed to be strong enough to hold the weight and stiff so that it does not twist while transmitting the torque from the drill. The keyed rear and drill shaft with the correct dimension were bought. The only manufacturing change was made in the drill shaft was reduced the diameter using lathe machine and made the triangle in the front with the help of Bill Mildenberger as it required special machine to make it so that the drill grips properly onto it. The holes were made simultaneously through the two rear shafts and the tire support so that there is a tight fit tolerance and the hole for the bolts were made using the milling machine.

The other manufacturing changes were made to the centrifugal clutch where the spring was removed so that the clutch engages at lower speed of around 100rpm. The sprocket and the clutch came with the key so that it fitted perfectly with the keyed shaft respectively in rear and drill shaft. The gear ratio was chosen based on the calculation made with the given values. The MATLAB code for it is available in the Annex.

The housing support for the drill shaft along with the drill trigger mechanism were manufactured from birch plywood as the wood is strong enough to withstand force and it is easy to work with. These supports were screwed properly so that it supports the drill and drill shaft movement during the race.

Usability:

When manufacturing the parts for usability most of the material used was aluminum 6061. Parts such as the caliper mounting system, handle tubes, and adapter pieces for the handles. Most of these parts were manufactured on the mill and on the lathe. These parts were possible to make without external help. For the caliper mounting system, the sheets of aluminum were cut a sized down with the holes made on the lathe machine. Prior testing of this part was done using a 3D printed part to verify sizing and adjustments. With this testing we were able to see that this part had to be angled. This angle for the mount was made from sized down plywood.

Tubes were cut to sizes to create the handles for steering, this part was created very easily. In addition, there needed to be a part that would help strengthen the steering tubes since they are relatively thin. To also mount the handles for braking and accelerating there needed to be a piece with the same inner diameter as the steering tubes and an outer diameter of the handles. This mechanism was 3D printed since it was a simple mechanism that would need a very specific sizing and would be able to get clamped on properly using the handles.

Another component to this subsystem is the seating. This was all manufactured from birched plywood, 3 pieces were cut out, 2 side panels, and a back rest. Slots were cut out from the side panels for the back rest to slide in and be supported from both sides. In addition, the seats were laser engraved to add designs on all the parts of the seating.

Development Time:

Walter Cruz	70 h
Peter Brodnik	33 h
Sanjeev K C	110 h
Brian Valerio	80 h
Total	293 h

Manufacturing Time:

Walter Cruz	34 h
Peter Brodnik	33 h
Sanjeev K C	57 h
Brian Valerio	49 h
Total	173 h

Purchased Hardware	$997.83
Manufacture Time ($100/hr)	$17300.00
Total	$18297.83

TEST PLAN AND RESULTS

Maximum Speed Test: The gear ratio was chosen so that the maximum allowable speed was around 14 mph so that it is withing 25mph range. Further tests were done by driving a certain distance by all the team members (to account for variability) and averaging the speed it took to cover the same distance and the average speed was below 25mph.

Turn Radius Test: The cart was able to make a turn of radius 6 ft. This was tested on the track, the turn was done around the George Eastman statue and completely passed.

Payload Test: The cart was tested up to 230 lbf weight of driver along with the weight of the cart and there was no significant displacement in the frame and was also tested through the NX simulations.

Brake Distance Test: The braking distance was measured to be 14ft overall. This meets the 15ft window, although it was close it did pass.

Number of wheels Test: The specification was passed by inspection as the cart was supported by three wheels.

Slope Test: The cart was visually inspected and was able to go up the ramp without the stop. After going down the ramp and around the trees the cart was able to fully go up the slope with enough clearance height.

Vehicle Dimensions: The dimensions of the cart were measured with the measuring tape and was withing the initial requirements of 6ft x 4ft x 4ft for the dimensions of the cart.

Specifications Tests	**Result (Pass/Fail)**
Maximum speed less than 25 mph	Pass
Turn radius less than 11ft.	Pass
Payload within 5 lbf	Pass
Maximum brake distance less than 15 ft.	Pass
No. of wheels in vehicles less or more than 4	Pass
Travel up the slope of 4.3 degrees	Pass
Vehicle Dimensions	Pass

INTELLECTUAL PROPERTY

All the processes involved in concept selection as well as manufacturing parts were not patentable. The frame structure, braking, drivetrain, or steering system were already used in some of the available designs earlier and hence are not patentable.

SOCIETAL AND ENVIRONMENTAL IMPLICATIONS

The drill powered cart project could have several impacts on public health, safety, and welfare. On the one hand, the drill powered cart helps the environment by encouraging people to use this cart over your standardized internal combustion engines in vehicles, which could have positive health benefits. With a drill that is rechargeable this becomes an alternative to gas powered cars that are not cost effective and are not environmentally friendly.

In addition, materials such as plywood reduce the material waste that usually goes into a standard car. Although the use of aluminum and steel might not be completely environmentally friendly, compared to a standard vehicle the quantity of material used does not produce as much carbon emission and waste. One additional thing that might still be considered is using a source of energy that is more environmentally friendly compared to the drill. That is something that can be done to improve it if there is additional time available.

With respect to the welfare and relevant global, cultural, and social implications the drill cart is a more cost-effective method to a vehicle. This being a cost-effective method of transportation could benefit developing countries and communities with lack of resources. One downside to this would be the amount of load that it will hold will not be as much as a vehicle, this could be considered the limiting factor. In addition, as the model is designed this vehicle is a one-person cart which would be another limiting factor.

Overall, the drill powered cart has the potential to counter the standardized internal combustion engines. This has a significant impact on the environment and for more affordable transportation opportunities. Although there might be some safety, ethical, and environment concerns this does create a different opportunity that once optimized can address all these concerns.

RECCOMENDATIONS FOR FUTURE WORK

If presented with the opportunity to continue working and improving this design, there could be additional analysis on the actual movement of the assembly. In addition to simulation, there should be more testing in each subsystem. In drivetrain, multiple gear system could be used that would allow for a change of necessary torque if needed. For frame, looking at a better way to support the 0.5 in plywood from displacing. 2x4 lumber wood was used to support the frame but this can be bulky and could drop your clearance height, looking for an alternative would be the best recommendation. For usability, using a different braking system like hydraulic brakes instead of mechanical brakes could create a better braking distance. Although this may be a costly option it would prove more effective than mechanical brakes. For steering, more analysis supports the flexing of the frame that causes the steering mechanism and wheels to cave in. Find some alternative to make sure that the wheels have the proper support and alignment. These recommendations could help optimize other iterations of this project.

ACKNOWLEDGMENTS

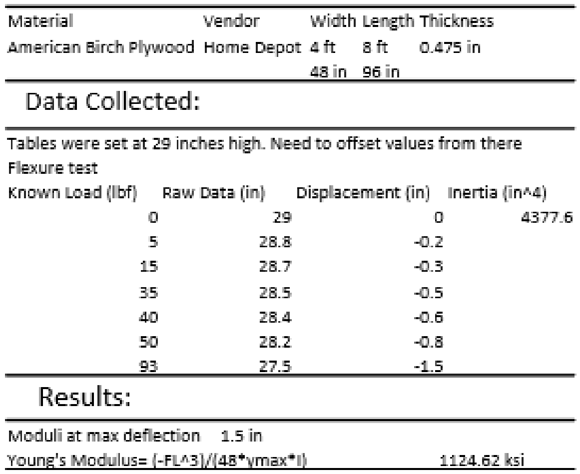
We would like to acknowledge the project manager Christopher Muir for this opportunity. We would also like to acknowledge Chris Pratt, Mike Pomerantz, Jim Alkins, Peter Miklavcic, Bill Mildenberger, Conor McCole and our TA, Seungju Yeo for their assistance in the shop and their support answering questions. They were all a huge help to helping guide us with our development and manufacturing process.

REFERENCES

1. R. G. Budynas and K. J. Nisbett, Shigley's Mechanical Engineering Design, Eleventh. New York: McGraw Hill Education, 2020.

ANNEX A

CONCEPT SKETCHES



Material	Vendor	Width	Length	Thickness
American Birch Plywood	Home Depot	4 ft	8 ft	0.475 in
		48 in	96 in	

Data Collected:

Tables were set at 29 inches high. Need to offset values from there

Flexure test

Known Load (lbf)	Raw Data (in)	Displacement (in)	Inertia (in^4)
0	29	0	4377.6
5	28.8	-0.2	
15	28.7	-0.3	
35	28.5	-0.5	
40	28.4	-0.6	
50	28.2	-0.8	
93	27.5	-1.5	

Results:

Moduli at max deflection	1.5 in	
Young's Modulus= (-FL^3)/(48*ymax*I)		1124.62 ksi

Figure 1: Data collection to determine Young's Modulus

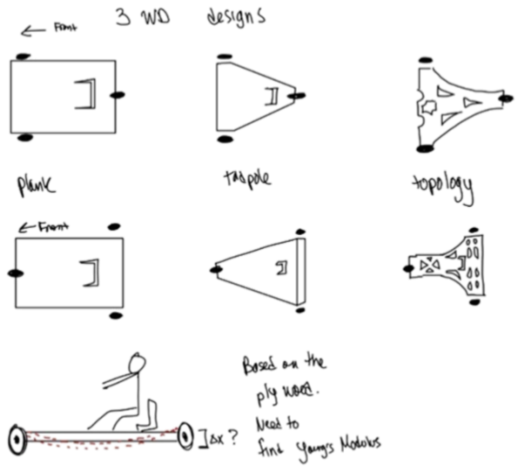


Figure 2: Frame concept drawings

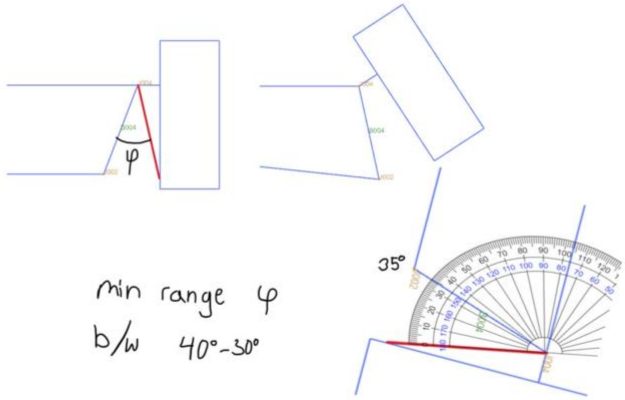


Figure 3: Minimum phi range to accomplish a 5ft turning radius

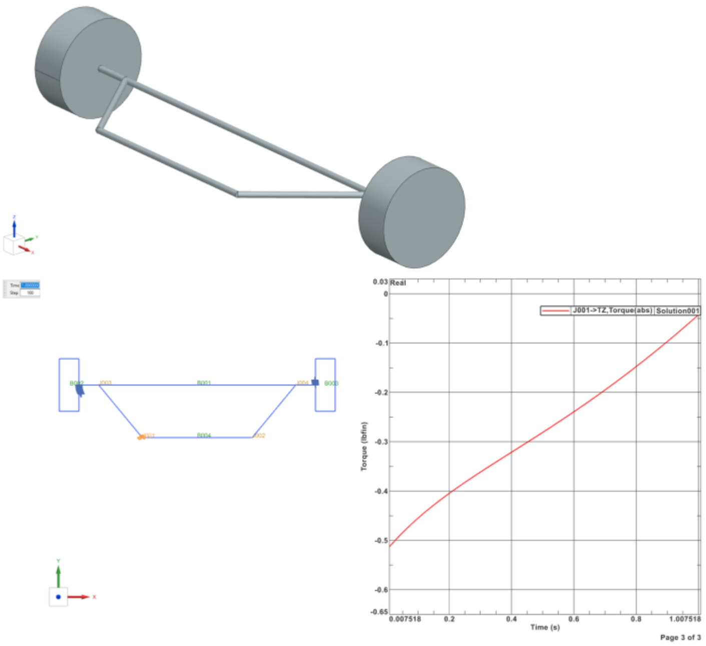
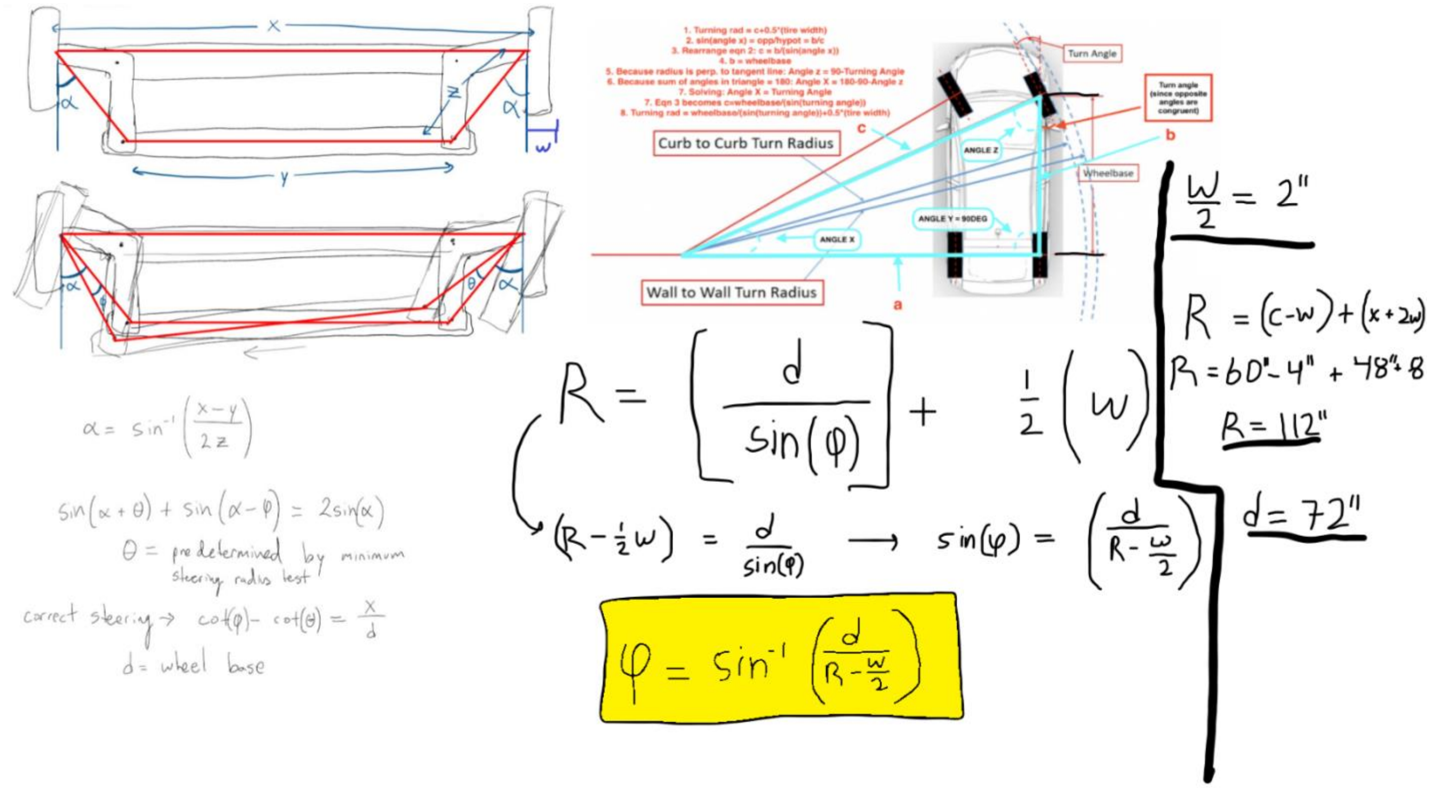


Figure 4: CAD mechanism rendering of basic Ackermann link graph of of basic Ackermann link graph of TZ, Torque around motion driver.



1. Turning rad = c+0.5*(tire width)
2. sin(angle x) = opp/hypot = b/c
3. Rearrange eqn 2: c = b/(sin(angle x))
4. b = wheelbase
5. Because radius is perp. to tangent line: Angle z = 90-Turning Angle
6. Because sum of angles in triangle = 180: Angle X = 180-90-Angle z
7. Solving: Angle X = Turning Angle
7. Eqn 3 becomes c=wheelbase/(sin(turning angle))
8. Turning rad = wheelbase/(sin(turning angle))+0.5*(tire width)

$$\alpha = \sin^{-1}\left(\frac{x-y}{2z}\right)$$

$$\sin(\alpha+\theta) + \sin(\alpha-\varphi) = 2\sin(\alpha)$$

θ = predetermined by minimum steering radius test

correct steering → $\cot(\varphi) - \cot(\theta) = \frac{x}{d}$

d = wheel base

$$R = \left[\frac{d}{\sin(\varphi)}\right] + \frac{1}{2}(w)$$

$$\left(R - \frac{1}{2}w\right) = \frac{d}{\sin(\varphi)} \longrightarrow \sin(\varphi) = \left(\frac{d}{R-\frac{w}{2}}\right)$$

$$\boxed{\varphi = \sin^{-1}\left(\frac{d}{R-\frac{w}{2}}\right)}$$

$$\frac{W}{2} = 2''$$

$$R = (c-w) + (x+2w)$$

$$R = 60'' - 4'' + 48'' + 8$$

$$\underline{R = 112''}$$

$$\underline{d = 72''}$$

Figure 5: Ackermann steering calculation to determine phi angle for a 5ft turning radius with a wheel base of 6ft.

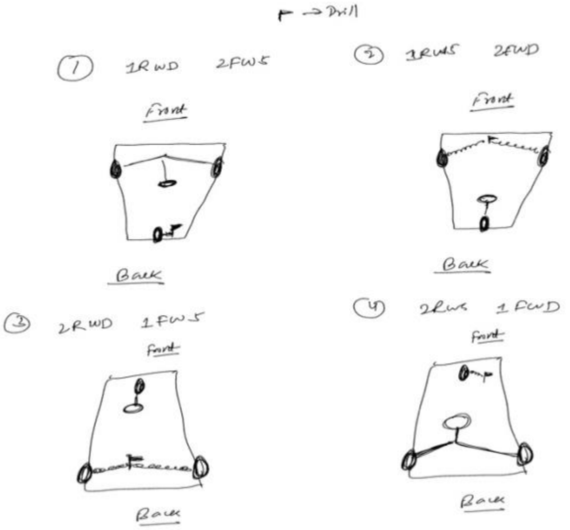


Figure 6: Different configurations of drill and wheel for three-wheeled cart

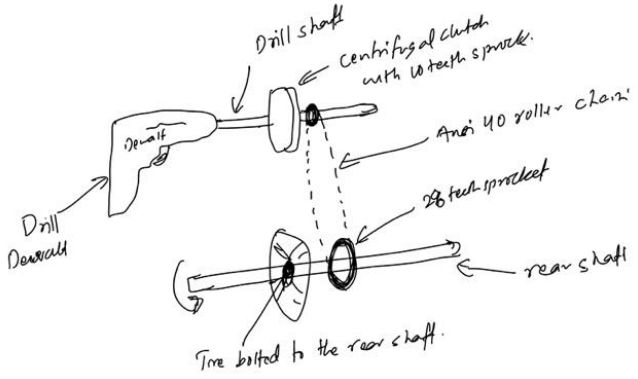


Figure 7: Drivetrain assembly with centrifugal clutch

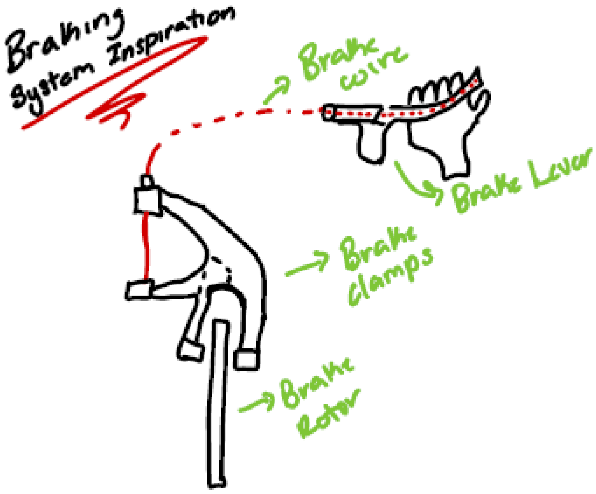
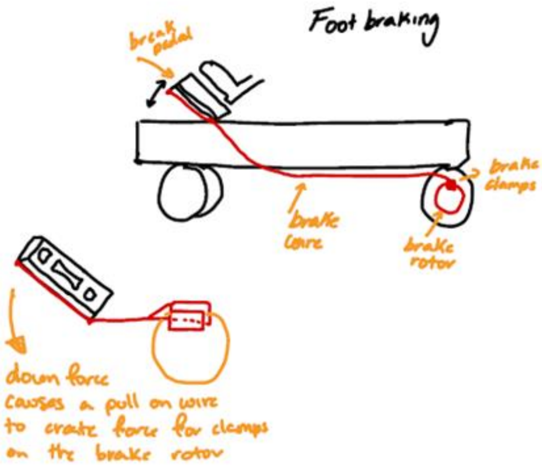


Figure 8: Hand lever concept braking system

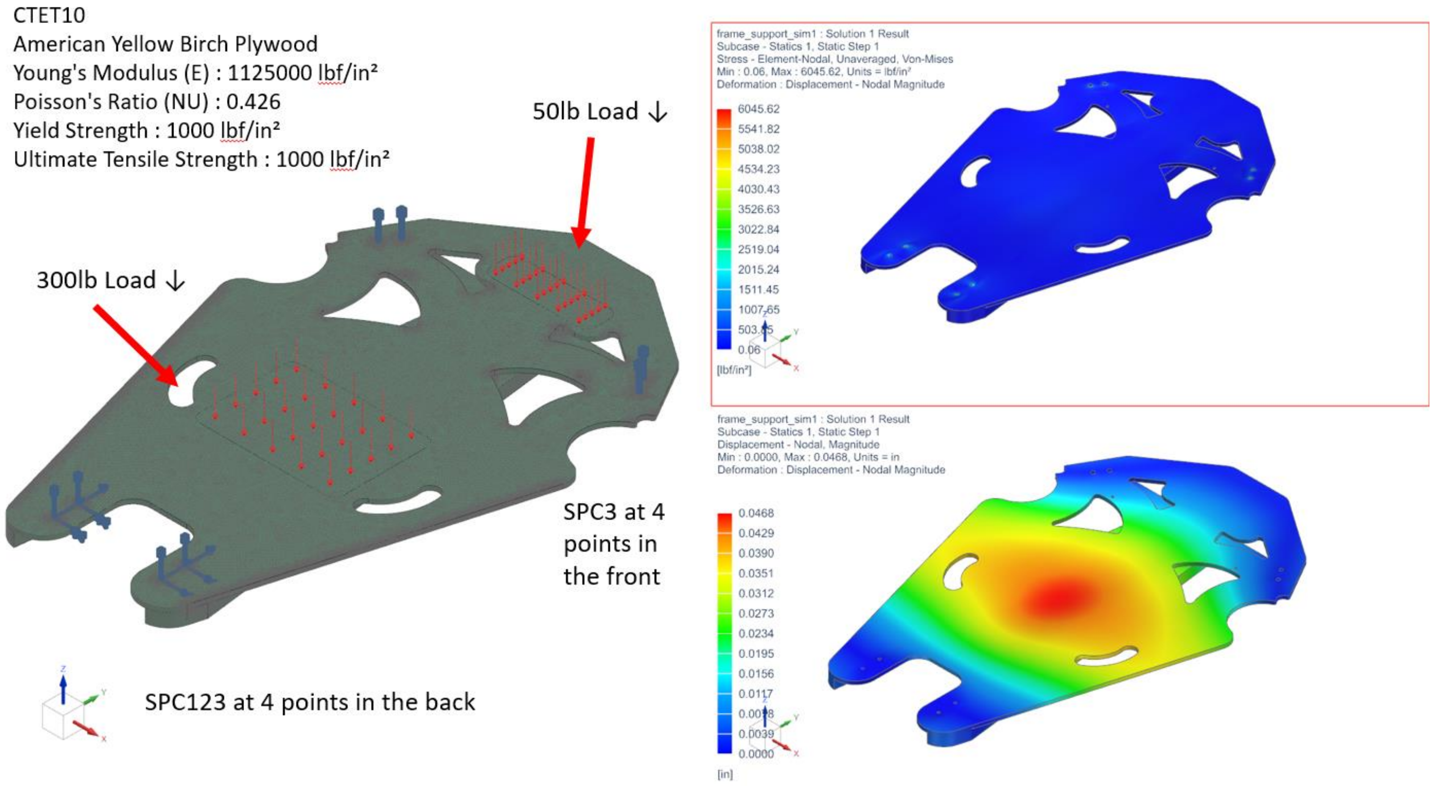


down force causes a pull on wire to create force for clamps on the brake rotor

Figure 9: Foot braking concept system

ANNEX B

MECHANICAL ANALYSIS



CTET10
American Yellow Birch Plywood
Young's Modulus (E) : 1125000 lbf/in²
Poisson's Ratio (NU) : 0.426
Yield Strength : 1000 lbf/in²
Ultimate Tensile Strength : 1000 lbf/in²

frame_support_sim1 : Solution 1 Result
Subcase - Statics 1, Static Step 1
Stress - Element-Nodal, Unaveraged, Von-Mises
Min : 0.06, Max : 6045.62, Units = lbf/in²
Deformation : Displacement - Nodal Magnitude

frame_support_sim1 : Solution 1 Result
Subcase - Statics 1, Static Step 1
Displacement - Nodal, Magnitude
Min : 0.0000, Max : 0.0468, Units = in
Deformation : Displacement - Nodal Magnitude

Figure 1: FEA Results of Frame under the load

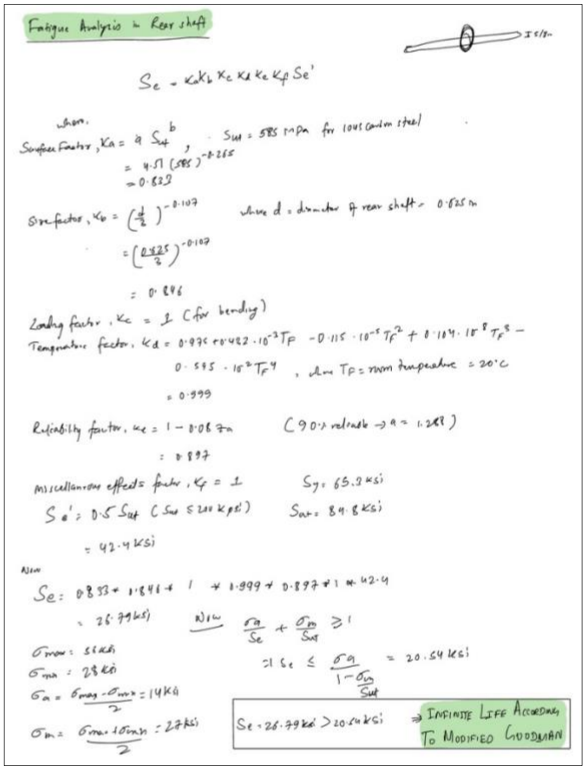
ANNEX C

MATLAB CODE FOR GEAR RATIO

```
clear all; close all;
%% Given values
m_kart = 45;  %Mass of the cart in kg
m_driver = 110; %Mass of the driver in kg, assumed
g = 9.81; %Acceleration due to gravity in m/s^2
W_kart_and_driver = (m_kart + m_driver) * g; %Total weight of kart with driver
W_total = W_kart_and_driver; %Total weight of kart with drill and driver
mu = 1.2/1.7; %friction coefficient of hard rubber on dry concrete
v_max = 11.176; %Maximum allowable speed = 25 mph = 11.176m/s
r = 0.127; %Radius of wheel 5in = 0.127m
T_stall = [3.389545 2.711636 2.033727]; %Initially tested Stall torque for drill at 3 diff speeds
P_max = 300; %Max power of drill in watts
RPM_Drill = [450 1300 2000]; %Available speeds in drill in rpm
pi = 22/7;
alpha = 5; %inclined angle was 4.3 degrees, assumed 5 degrees for factor of safety
F_start = 2.82 * g; %Measured = 9.5g force minimum required to move the kart with driver
F_test = (7.7+6.8+7)/3; %Average of three trials
T_stall_2 = F_test * g * .02; %Measured in the lab and used for calculation

%% Provided we want to only operate in 2nd setting
fprintf('Based on the 2nd setting operation of the drill \n');

%Gear ratio required to move at 25mph
Speed_input_rpm = RPM_Drill (2); %Taking the maximum speed of the drill in rpm
angluar_velocity_output = v_max / r; %Formula to calculate angular velocity in rad/s
Speed_output_rpm = (angluar_velocity_output * 60) / (2 * pi); % Formula to calculate the rpm output speed
Gear_ratio_1 = Speed_input_rpm / Speed_output_rpm; %Gear ratio
fprintf('Gear ratio to move kart with maximum speed is %1.4f \n',Gear_ratio_1);

%% Finding the gear ratio that overcomes stall in the start of the kart
T_start_output = F_start * r; %Torque required to start the kart in Nm
T_input = T_stall_2; %T_stall(2);
Gear_ratio_2= T_start_output/T_input ;%T_stall (2); %Input stall torque of the drill running in with F.S of 1.5
fprintf('Gear ratio to start kart is %1.4f \n',Gear_ratio_2);

%% Use this torque to find corresponding gear ratio needed to go up the ramp
Torque_input_ramp = T_stall_2;
Torque_output_ramp = ((F_start*cosd(alpha)) + ((F_start/mu) * sind(alpha))) * r;
Gear_ratio_3= Torque_output_ramp/Torque_input_ramp;
fprintf('Gear ratio to move up ramp is %1.4f \n',Gear_ratio_3);

%% Chosen gear ratio and torque and speed according to it
%for speed
Gear_ratio_chosen= 2.8; %28 teeth sprocket in rear and 10 teeth sprocket in drill shaft
fprintf('Chosen gear ratio is 2.8 \n');
output_Speed = (Speed_input_rpm/Gear_ratio_chosen)*((2*pi)/60)*r;
Speed_mph= output_Speed * 2.23694; %conversion from m/s to mph
fprintf('Output speed in mph is %1.4f\n',Speed_mph);
fprintf('Checking if drill stalls or not \n');

%for straight path
input_torque_normal = T_start_output/Gear_ratio_chosen;
fprintf('Input torque required in straight path in Nm is %1.4f\n',input_torque_normal);
if input_torque_normal < T_stall_2 %Comparing with the testing
    fprintf('Drill does not stall while moving in straight path \n');
else
     fprintf('Drill stalls while moving in straight path as required is 1.4061 \n');
end

%for ramp
input_torque_ramp = Torque_output_ramp/Gear_ratio_chosen;
fprintf('Input torque required in ramp in Nm is %1.4f\n',input_torque_ramp);
if input_torque_ramp < T_stall_2 %Comparing with the initial testing
    fprintf('Drill does not stall while moving up ramp \n');
else
     fprintf('Drill stalls while moving up ramp as required is 1.4061 \n');
end
```

ANNEX D

FATIGUE ANALYSIS



Fatigue Analysis in Rear shaft

1 5/8 in

$$S_e = k_a k_b k_c k_d k_e k_f S_e'$$

where,

Surface Factor, $k_a = a S_{ut}^{b}$, $S_{ut} = 585$ MPa for 1045 carbon steel

$= 4.51 (585)^{-0.265}$

$= 0.833$

Size factor, $k_b = \left(\frac{d}{3}\right)^{-0.107}$ where d = diameter of rear shaft = 0.825 in

$= \left(\frac{0.825}{3}\right)^{-0.107}$

$= 0.846$

Loading factor, $k_c = 1$ (for bending)

Temperature factor, $k_d = 0.975 + 0.422 \cdot 10^{-3} T_F - 0.115 \cdot 10^{-5} T_F^2 + 0.104 \cdot 10^{8} T_F^3 -$

$0.595 \cdot 10^{2} T_F^4$, where T_F = room temperature = 20°C

$= 0.999$

Reliability factor, $k_e = 1 - 0.08 z_a$ (90% reliable → a = 1.288)

$= 0.897$

miscellaneous effects factor, $k_f = 1$

$S_y = 65.3$ ksi

$S_e' = 0.5 S_{ut}$ ($S_{ut} \le 200$ kpsi)

$S_{ut} = 84.8$ ksi

$= 42.4$ ksi

Now

$S_e = 0.833 * 0.846 * 1 * 0.999 * 0.897 * 1 * 42.4$

$= 26.79$ ksi

Now $\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} \geq 1$

$\Rightarrow S_e \le \frac{\sigma_a}{1-\frac{\sigma_m}{S_{ut}}} = 20.54$ ksi

$\sigma_{max} = 56$ ksi

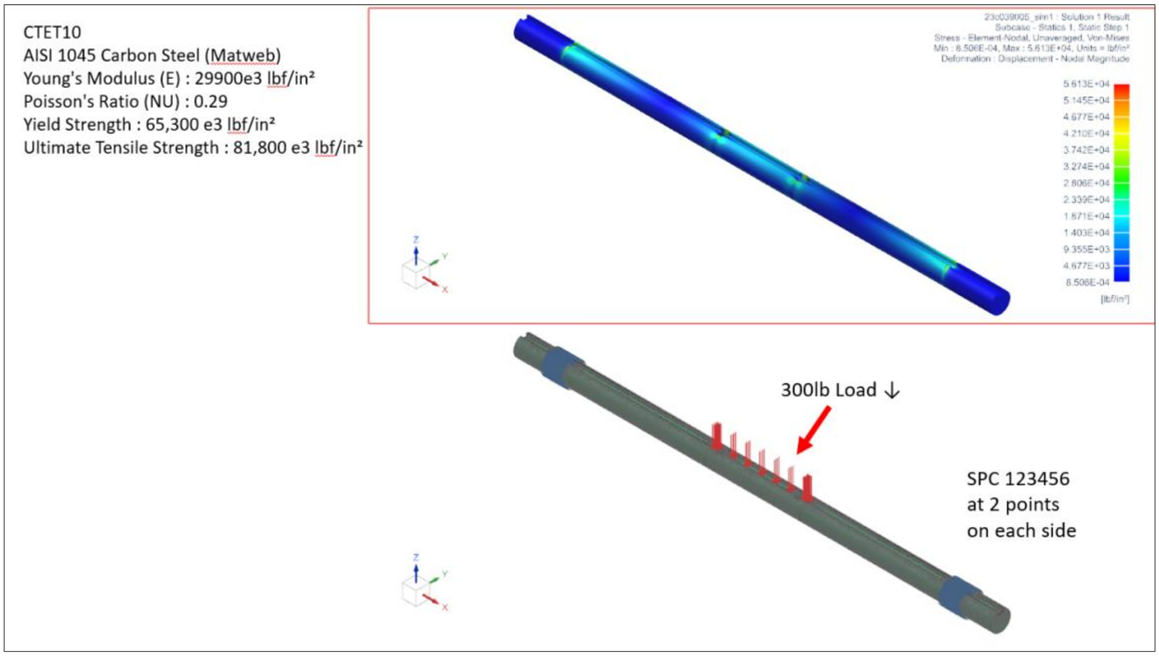
$\sigma_{min} = 28$ ksi

$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = 14$ ksi

$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = 27$ ksi

$\boxed{S_e = 26.79 \text{ ksi} > 20.54 \text{ ksi}}$ ⇒ INFINITE LIFE ACCORDING TO MODIFIED GOODMAN

Figure 1: Hand Calculations



CTET10
AISI 1045 Carbon Steel (Matweb)
Young's Modulus (E) : 29900e3 lbf/in²
Poisson's Ratio (NU) : 0.29
Yield Strength : 65,300 e3 lbf/in²
Ultimate Tensile Strength : 81,800 e3 lbf/in²

23c039005_sim1 : Solution 1 Result
Subcase - Statics 1, Static Step 1
Stress - Element-Nodal, Unaveraged, Von-Mises
Min : 8.506E-04, Max : 5.613E+04, Units = lbf/in²
Deformation : Displacement - Nodal Magnitude

Figure 2: NX FEA Analysis check

Source: https://www.matweb.com/search/datasheet.aspx?matguid=193434cf42e343fab880e1dabdb143ba

ANNEX E

WHOLE CART ASSEMBLY

Figure 1: Trimetric View

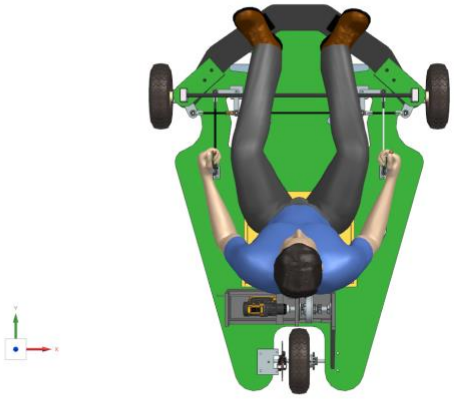


Figure 2: Top View

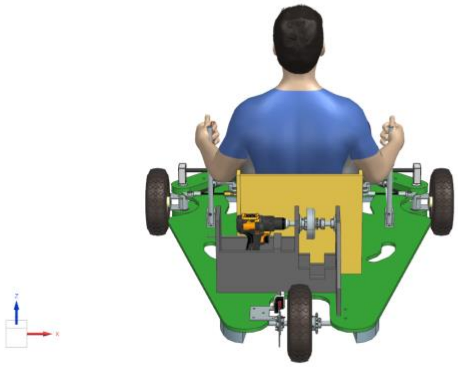


Figure 3: Rear View

Figure 4: Right View

ANNEX F

BILL OF MATERIALS (BOM)

Part Number	Part Owner	Item Description
FRAME		
22c022005A	WC	2 x 1/2" American Birch Plywood
USABILITY		
23c062004A	BV	Handle Adapter
8975K83	BV	Multipurpose 6061 Aluminum Sheet 1/8" thick x 3" wide 1/2 ft long
23c062007A	BV	Brake Handles
23c062008A	BV	Rotor
23c062009A	BV	Wire
23c062010A	BV	Caliper
89965K491	BV	General Purpose Aluminum Tubing 1/2" OD, 0.035" Wall Thickness 3ft
92461A200	BV	Medium-Strength Steel Nylon-Insert Flange Locknut Class 8, Zinc-Plated, M5 x 0.8 mm Thread (Pack 100)
90327A128	BV	Alloy Steel Low-Profile Socket Head Screws Hex Drive, Zinc Plated, M5 x 0.8 mm Thread, 20 mm Long (50 pack)
23c062011A	BV	Lap Belt
89015K239	BV	Multipurpose 6061 Aluminum Sheet
89965k367	BV	General Purpose Aluminum Tubing, 3ft
91251a634	BV	Black-Oxide Alloy Steel Socket Head Screw
91608A318	BV	Slotted Oval Head Screws for Wood
90630a121	BV	High-Strength Steel Nylon-Insert Locknut
DRIVETRAIN		
23c039018A	SKC	BRAVEX Centrifugal Clutch
23c039019A	SKC	Engagement Spring 1000rpm
23c039005A	SKC	Fully Keyed, 5/8" Diameter, 18" Long, 1045 carbon steel - 1497K954
23c039006A	SKC	1 1/2" Al-6061 rod - 8974K18, 1ft
23c039008A	SKC	3/4" AL-6061 rod - 1570K61, 6" long, keyed
23c039020A	SKC	ANSI 40 Steel Roller Chain, 1/2" pitch, 5 ft long
6435K16	SKC	Clamping one piece shaft collar - 3/4in for drive shaft
6436K15	SKC	Clamping one piece shaft collar - 5/8 in for rear shaft
6280K701	SKC	ANSI 40 Steel Roller Chain Sprocket, 1/2" pitch, 16 teeth, keyed
5913K73	SKC	Low Profile Mounted Sealed Steel Bearing 3/4" shaft diameter
5913K62	SKC	Low Profile Mounted Sealed Steel Bearing 5/8" shaft diameter
1570K62	SKC	2024 Aluminum Keyed Rotary Shaft Fully Keyed, 3/4" Diameter, 12" Long
6236K155	SKC	ANSI 40 Steel Roller Chain Sprocket, 1/2" pitch, 28 teeth, keyed
6261K193	SKC	Connecting link chain 40
SUSPENSION		
4459T148	PB	Easy-to-Weld 4130 Alloy Steel 6"x12"x1/8"
2458K331	PB	Right Hand Thread Lube-Free Ball Joint End Rod 5/16"-24
91257A609	PB	Zinc Yellow Plated Grade 8 Steel 5/16"-24 (Lg 1 1/2")
93083A112	PB	Low-Strength Steel Serrated Flange Locknuts 5/16"-24
90322A692	PB	High-Strength Steel Threaded Rod 5/16"-24 (Lg. 6')
89955K179	PB	Easy-to-weld 4130 Alloy Steel Round Tubes 1ft. Lg.
98911A035	PB	Low-Strength Steel Threaded Rod 5/8"-18 (Lg. 6')
90566A235	PB	Thin Steel Nylon-Insert Locknut 5/8"-18 (50ct)
	TM	Drill plus battery