2024 Final Design Briefing

Penn Hyperloop

Not-A-Boring Mini Competition

Team Members + Advisors

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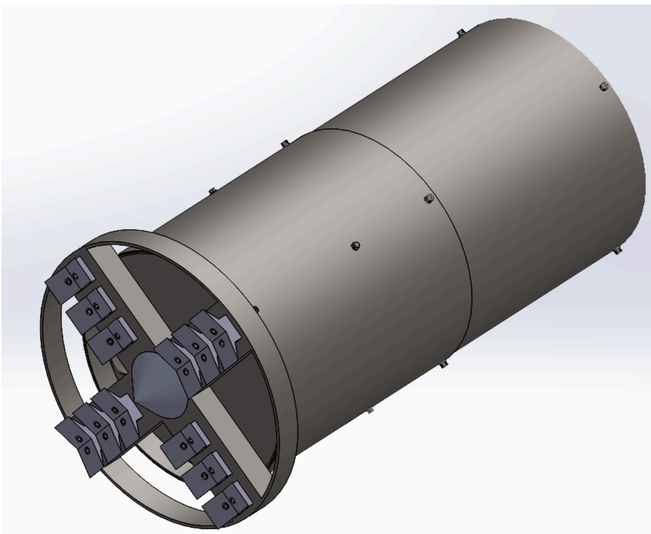
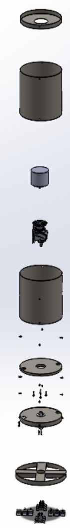
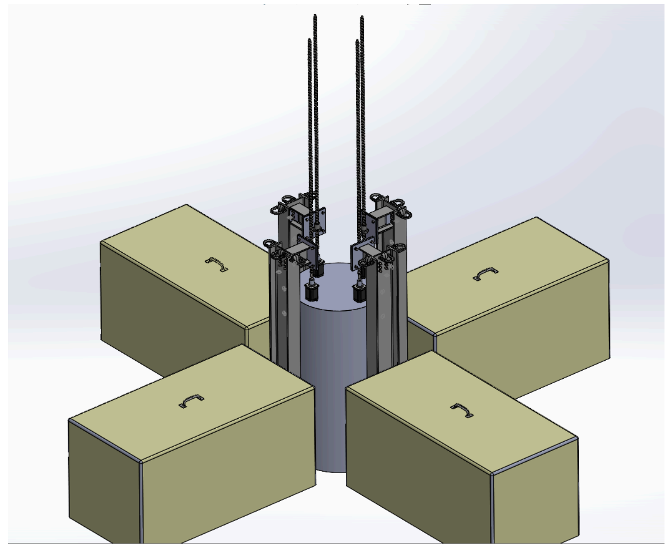
Tunnel Boring Machine and Propulsion System High-Level Overview

<u>Design Constraints + Specifications</u>

- Tunnel Diameter
  - 0.572 m
- Tunnel Length
  - 1 m
- Machine Power Source
  - 208Y 3-Phase Generator
- Peak Power Consumption
  - 27.5 kW

Subsystem	Power [kW]
Excavation	9.6 - continuous 20 - 1 minute
Propulsion	.576
Telemetry	<2
Muck Removal	4.8

- Cutterhead Targets
  - 0.8 kN*m
  - 33 RPM
- Tunnel Dig Time - ~=10 minutes

- In milling, feed rate = chip load * number of cutting edges * rotational speed
  - From Machinery’s Handbook 31 pg.1158 and Table 14 on pg. 1120

    Chip load is commonly used interchangeably with feedrate for a cutting tool with multiple cutting edges (teeth) such as in milling or drilling. Chip load is defined as tool feed distance for each tooth and represents the chip size forming for each tooth. Chip load can also be interpreted as the radial depth of cut for each tooth in milling. The following equation converts chip load of a cutting edge to feedrate of a multiple-edge cutting tool:

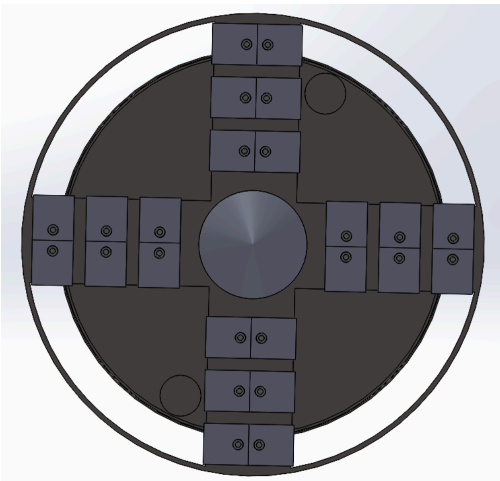
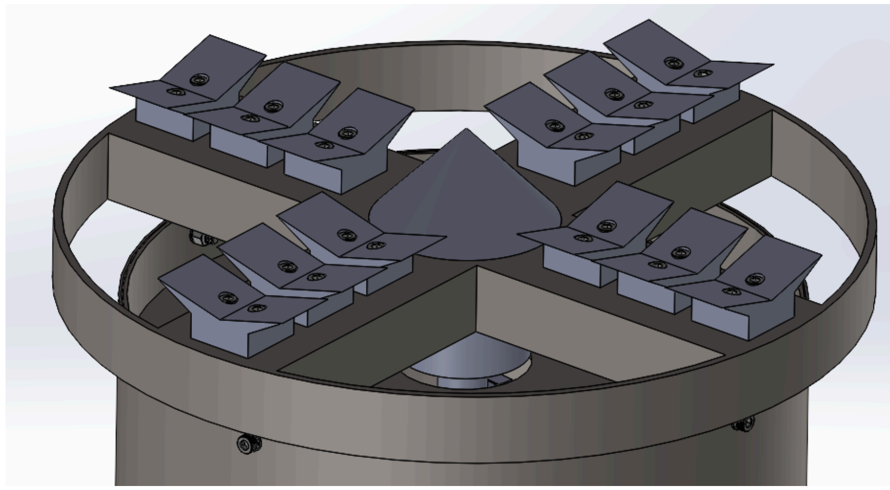
    $$f = c_L n N$$

    where f = feedrate of tool (mm/min, in/min)
    c_L = chip load of a cutting edge (mm/tooth, in/tooth)
    n = number of cutting flutes or cutting edges (#teeth/rev)
    N = rotational speed (rpm)

  - Extremely rough calculation
    Assume c_L of 5 * 25.4 mm/tooth = 127 mm/tooth from ferrous cast metal (softer than steel). Because the hardness of ferrous cast metal of 120 is off by 60x for ~ 2 hardness of clay, assuming linear scaling, then divide 127/60 = 2.1167 mm/tooth for our TBM going through clay.
  - N = 12 RPM (revolutions per minute)
  - n = 4 “flutes”/teeth/cutting edges (spokes in our case, assuming continuous cutting edge) per rev)in tpm (teeth per rev)
  - c_L = 2.1167 mm/tooth
  - $f = c_L$ * n * N = 12 rpm * 4 tpr * 2.1167 mm/tooth = 101.6 mm/min
  - a 1 meter tunnel should take approximately 10 minutes. (ROUGH PRELIMINARY ESTIMATE)

System	Mass (kg)	Dimensions (Bounding Box)
TBM Structure	73.8	Cylinder - 0.5m diameter, 1.3m height
Main Drive System	138.6	Cylinder - 0.4m diameter, 1m height
Propulsion System	65.6	Shaft - 1.55m length Footprint - 3m x 3m
Cutterhead	119.5	0.572m diameter 0.03m height
Total (except prop structure)	397.5	Cylinder - 0.572m diameter, 1.33m height

Cutterhead and Main Drive System Design

In our Preliminary Design Briefing, we did not have access to the Bastrop Geotechnical Analysis Report. At the time, from brief Googling for soil conditions around 130 Walker-Watson Rd, Bastrop, TX (TBC HQ), we found nominally excavating **Bosque Series** soil from the 1977 Soil Service Report. Our cutter design hinged upon this and the fact that Bosque soil seemed to be made up of clay loam which is largely sticky and little rocks. Post-PDB acceptance, we were provided with the Geotechnical Analysis Report of the Cutterhead and confirmed much of our initial research – at least to the extent that mattered for the Mini-event.

<u>Cutterhead Design Rationale</u>

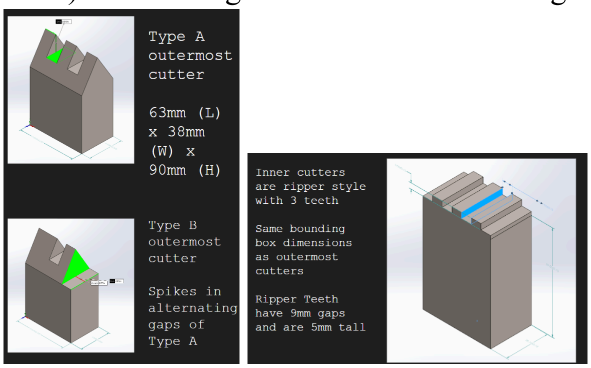
Facing softer ground and clay will fail more in shear vs compression (for the rocks) because failure mode for the soil is in shear. Originally, we were trying to size the cutters from pure first principles by trying to find the shear strength of the clay in some unit volume (assuming uniform density) then based on our propulsion thrust force and target torque of 6kN*m, wanted to size shear force acting on unit volume of clay and cross sectional area to size each cutter. **Then, after TBC Q/A Session 1, we realized that we should just be a little more experimental as modeling the Earth is useless at least for mini-event. Decided to take more simple approach then test and iterate as needed (and by manufacturing the cutters ourselves we can reduce lead time).** We also want to maximize the openings in the face as we are excavating soft ground with higher advance rate so need larger openings for more soil removal.

Our TBM is akin to that of an end mill. Taking inspiration from gauging the number of teeth on an end mill, end mills typically have more teeth for harder material, so we want less for softer. 4 spokes is a good middle ground that maximizes opening area along with soil being excavated. We formed 3 concentric drilling circles with our cutter layout.

With soft ground, we want longer teeth as it is easier to “puncture” into the soil and scrape around the soil; additionally, with the thrust force downwards, our teeth should be able to compress and break any soft rocks.

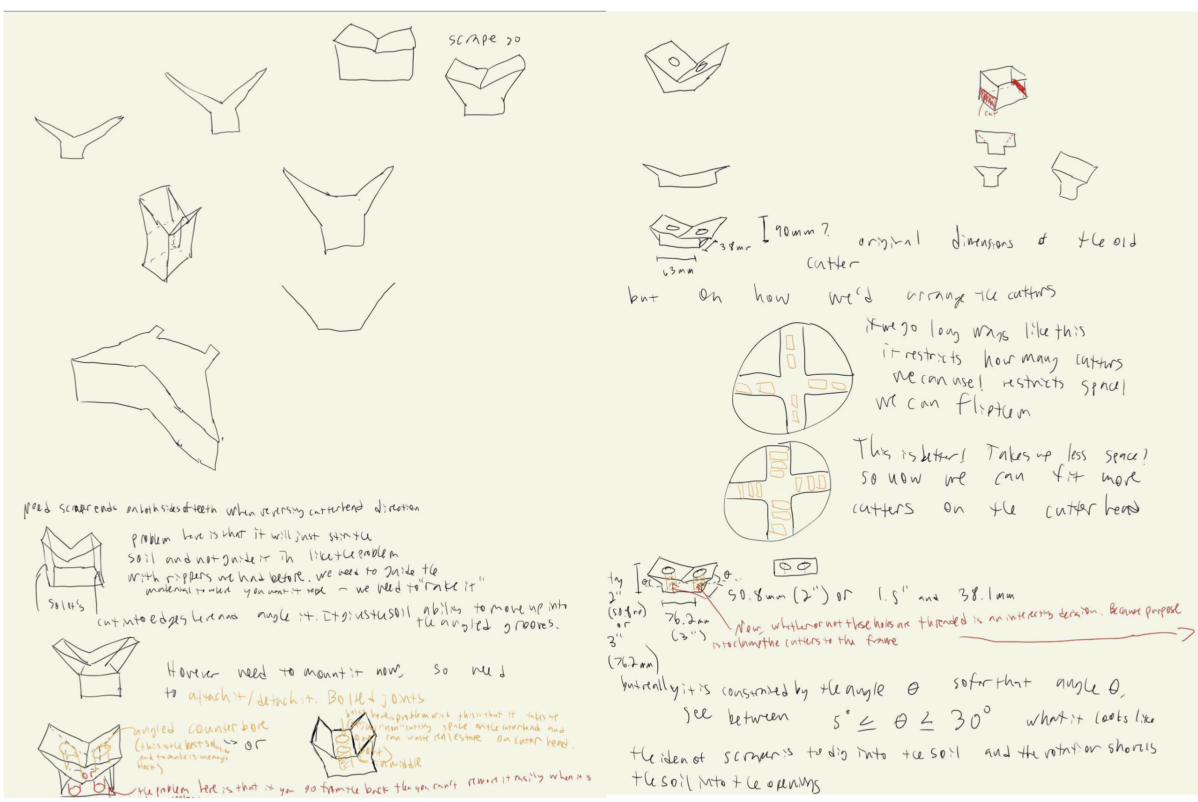
We did not want outermost cutters to be super fine (more teeth, smaller gaps) like a comb, otherwise there is concern about the wear on individual teeth and it distributes load which can make compressive stress lower (so if faced with any rocks, we may not be able to break them up)

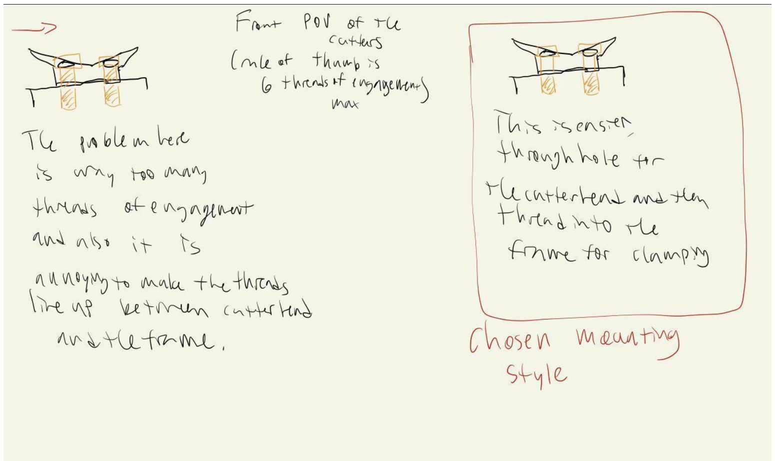
Initially, we had ripper teeth geometry (with alternating teeth to help guide the soil through gaps of the teeth on the next spoke in rotation) with lots of thought put into its design (see our PDB) with a design similar to the following:



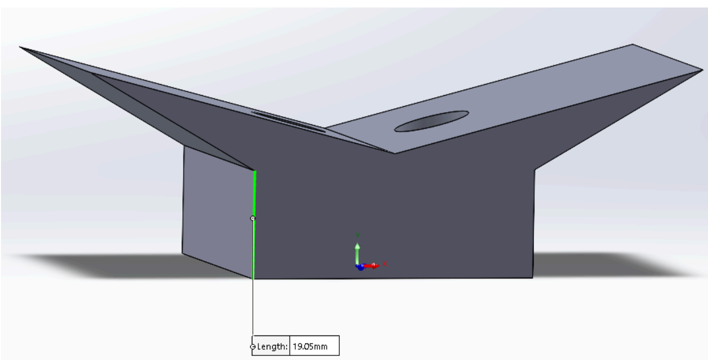
From PDB feedback, we were advised to think about a cutter that would guide the soil into the open excavation sectors for muck removal. This made a lot of sense because with those rippers, we'd essentially just be "stirring" the soil and not guiding it into the open face.

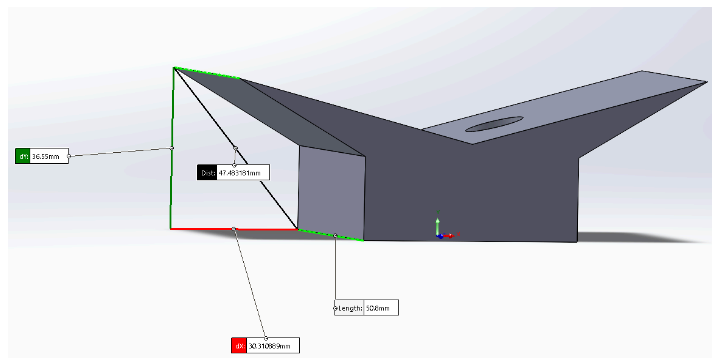
Therefore, we went through some design iterations with a scraper type cutter. Below are some drawings we made throughout the process:



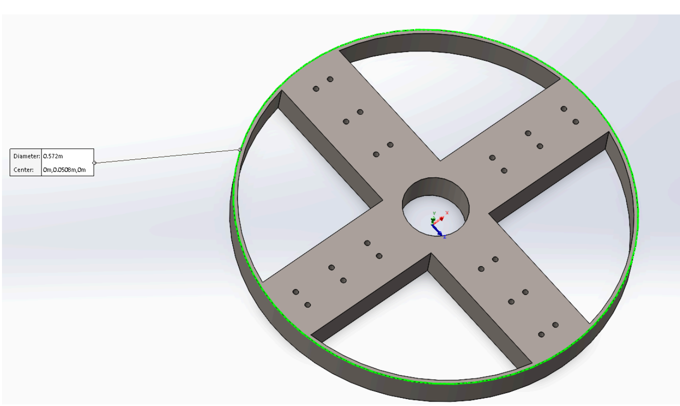


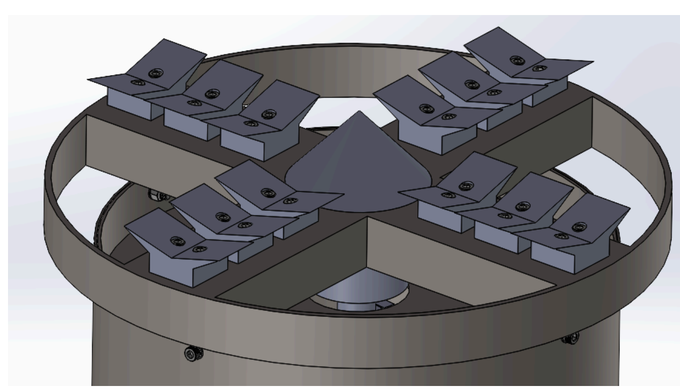
We eventually prototyped steel scrapers to have a rake angle to not only stir the soil, but that angle would dig up into the soil and guide it towards the open face as the cutterhead rotated.



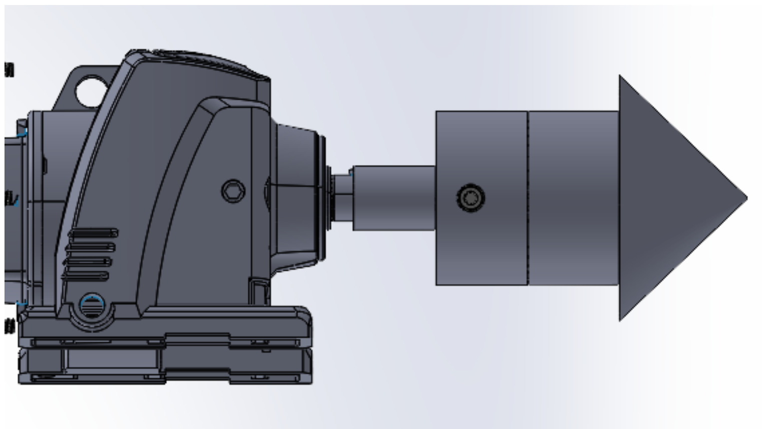


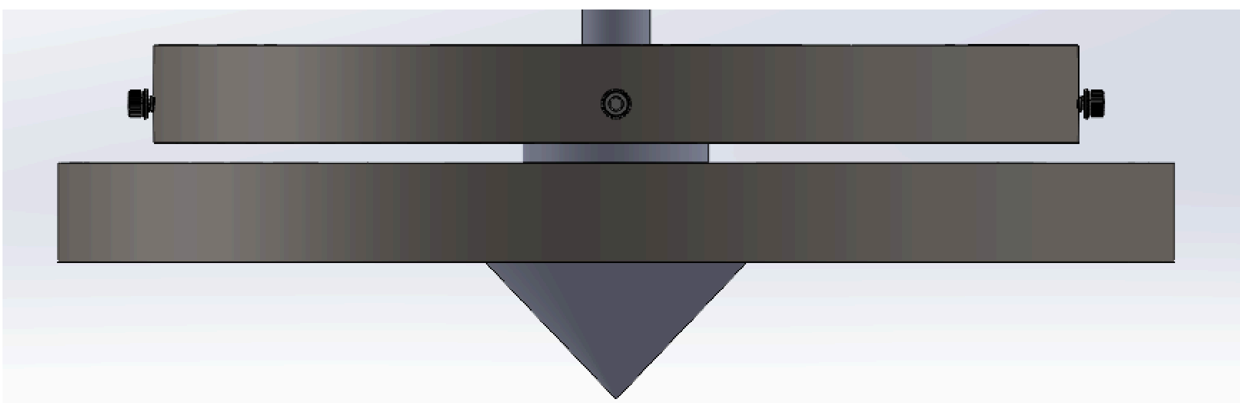
We still continued with the 3 concentric circles of “milling”. The aluminum cutterhead frame will be partitioned into 4 segments (bolted together), as will the spokes (see below). We will weld the steel ring frame together but set screws and double nuts to clamp the spokes to the outer steel cutterhead ring frame.We use fasteners to clamp the cutters to the cutterhead – our cutters have an angled counterbore with a through-hole and we will tap a threaded hole into the cutterhead frame spoke (exact screws cannot be found to match our desired partially threaded length of the screws in cutters, so will modify cutters slightly to accommodate fastener supply from McMaster).





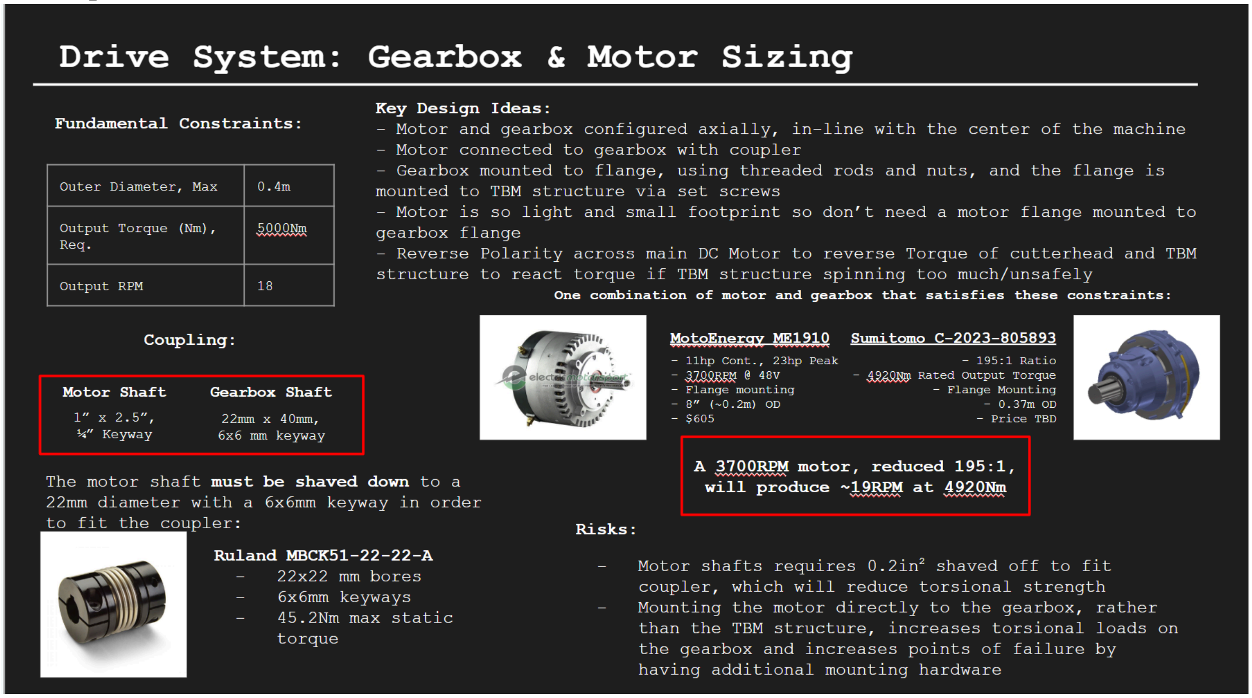
Another update to our PDB was the addition of the aluminum conical bit at the center of the cutterhead, which is welded to the aluminum ring frame, as the conical bit helps puncture the soil in the middle and helps push the soil in the middle to the sides to be guided by the cutters. The conical bit might be replaced with a different cutter variation if necessary for better soil guiding, but as of now, changing the conical bit design is the least of our concerns. The conical bit has a base of 95mm (3.74”) and height of 50.8mm (2”).





Main Drive System

In our Preliminary Design Briefing, our main design constraints were a 0.572m diameter cutterhead, a target output torque of 5kN*m, and baseline target RPM of 18 revolutions/minute of the cutterhead, as described in the slide from our PDB below with the described transmission stack-up.



However, this presented a couple issues. The main issue was the price of the Sumitomo C-2023-805893 speed reducer was almost $6k which was obviously much out of our price range. In fact, any speed reducer that could fit in our TBM Structure (diameter-wise) while still leaving gaps for our muck removal hoses, as well as height-wise, while also providing an extremely high speed reduction to produce higher output torque (conservation of power), would either just violate those size constraints or be well out of our price range.

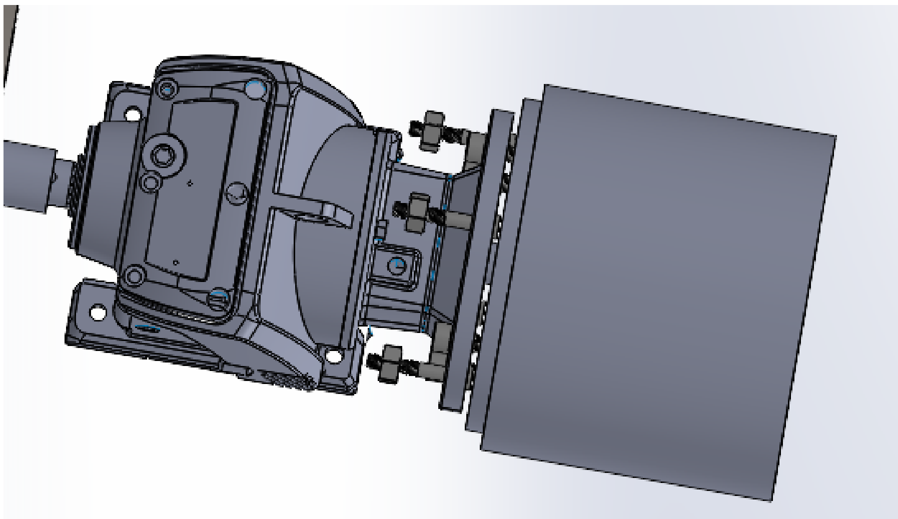
For this reason, we had to relax our output torque constraints and settle for a gearbox with a lower speed reduction ratio that also cost less than $1.5k and didn't have a super high lead time, while fitting our size constraints, and had good documentation on operation and maintenance.

Therefore, the inline speed reducer that matched best we could find, was a WINL77-100/1-145TC (INLINE HELICAL GEAR REDUCER, BOX SIZE 77, 100:1 RATIO, 145TC FRAME) with a 100:1 speed reduction ratio and thus almost 0.8kN*m of output torque (versus our ideal of 5kN*m) for the cutterhead, at an RPM of ~33 revolutions per minute.

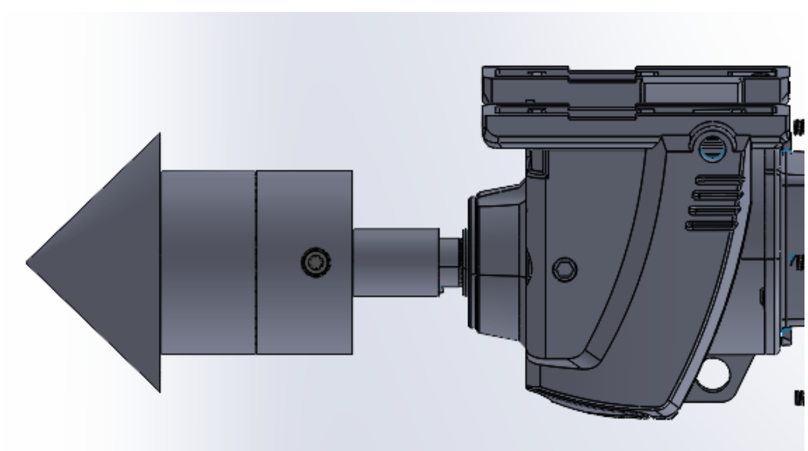
The motor we chose was a Motenergy ME-1602 Brush-Type Permanent Magnet DC Motor, which had similar ratings as our motor in the PDB, but a different shaft size that would fit into the new gearbox's borehole for its input shaft, without needing a coupler. The Brushed DC motor was simplest and easiest to control, and suited our relatively short digging time. The ME-1602 has a max output RPM speed of 3360 RPM at 48V, 200A (9.6kW or 12.87 Hp output power) draw. This motor has a NEMA143T C-face mounting that we can mount directly to the new gearbox's NEMA145T C-frame (flange) face. Because we didn't have the time and bandwidth to design and build our own motor controller, we had to use a COTS solution that had the motor controller and a pair of reversing contactors because we needed the capability to reverse the cutterhead rotation whenever we would feel that it would help excavation during digging (if it gets stuck or just stirs around in one direction). We were able to buy such a full COTS package that included the ME-1602 Motor and an Alltrax SR48400 Controller, and handheld physical throttle control

Main Drive System Transmission Assembly

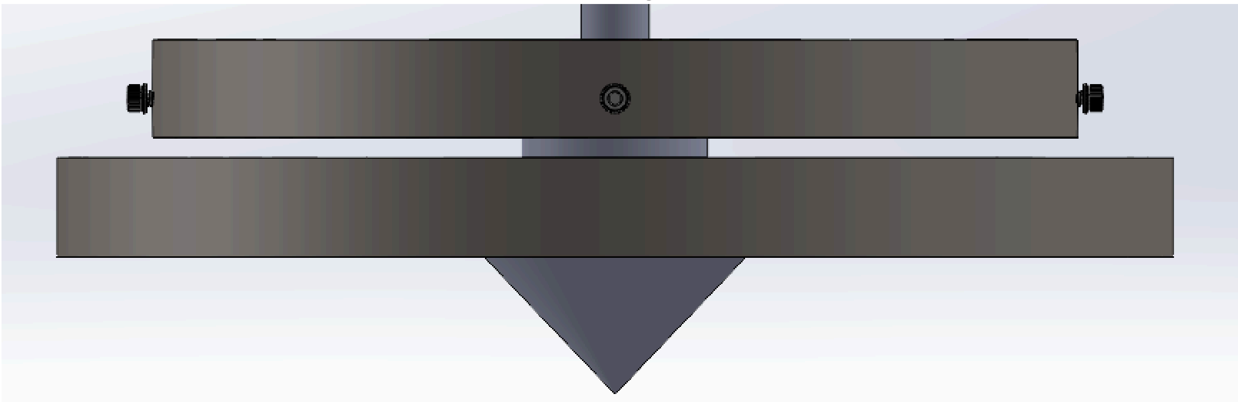
The input bore for the WINL77-100/1-145TC has a diameter of 0.88" as the machine was in metric units while the ME1602 shaft has a 7/8" diameter size and in SI units - the gap is very tiny, only 0.005 inches, so this poses no problem as the motor shaft should fit right into the gearbox's input bore.



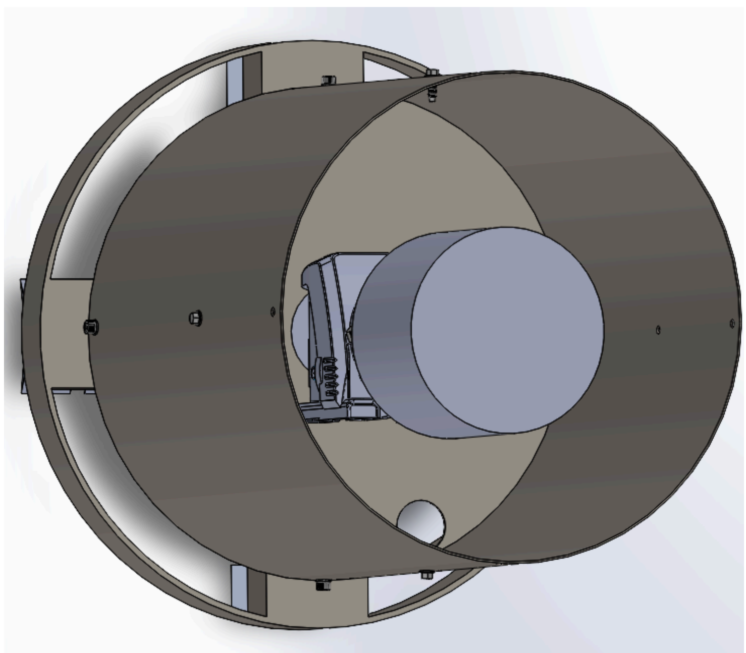
Since the output shaft length from the gearbox is not long enough to attach to the cutterhead directly, we need to pick a shaft that will get welded to the bottom of the conical bit of the cutterhead and a shaft coupling for torque transmission between the gearbox output shaft and this new shaft – not chosen/accurately modeled yet but looks something like this:



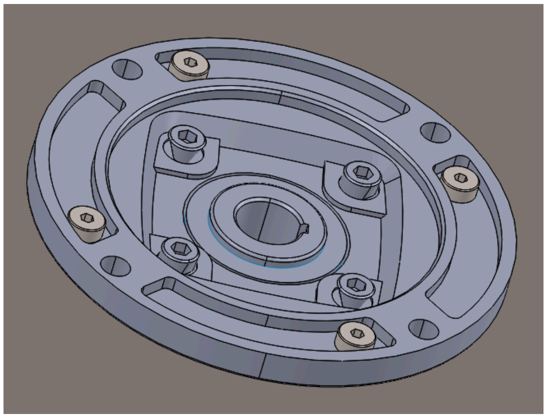
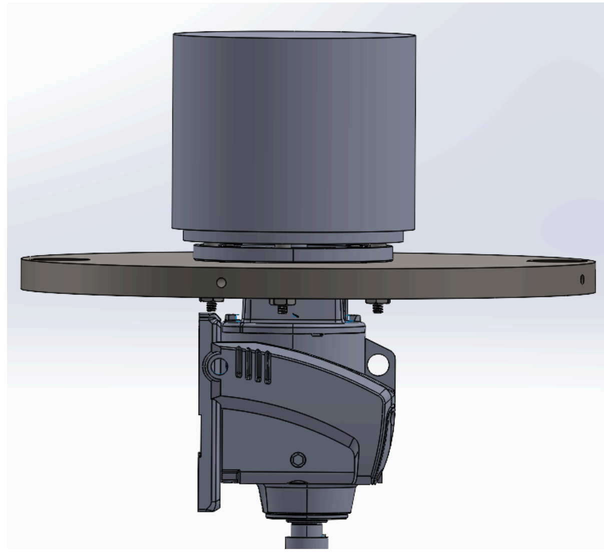
with the cutterhead and conical bit fixed together as



And as shown below, there is a lid at the bottom of the TBM with holes the muck removal hoses go through.



Now, because the reaction torque from cutterhead rotation is transferred through the conical bit, output shaft stack-up, and to the gearbox – the gearbox would normally be spinning on its own – therefore, we designed a flange mounting solution that would fix the gearbox to the TBM structure so that all of the axial reaction load (normal force from the soil) and reaction moment gets transferred through the main shaft stack-up, to the gearbox, and thus to the TBM itself. We are fine with this TBM rotation because we will merely reverse the cutterhead direction if we notice the TBM's rotation getting too out of hand and creating issues either for the TBM itself or to the propulsion system.

We have a steel flange that clamps to the TBM outer skin/structure using screws. The NEMA145T-C Face Flange mount on the gearbox is shown above, with through-holes that are already on the speed reducer – we use bolted joints (bolt and a nut, likely double-nut to prevent self-loosening from any vibrations) to mount the ME1602 motor to the gearbox.

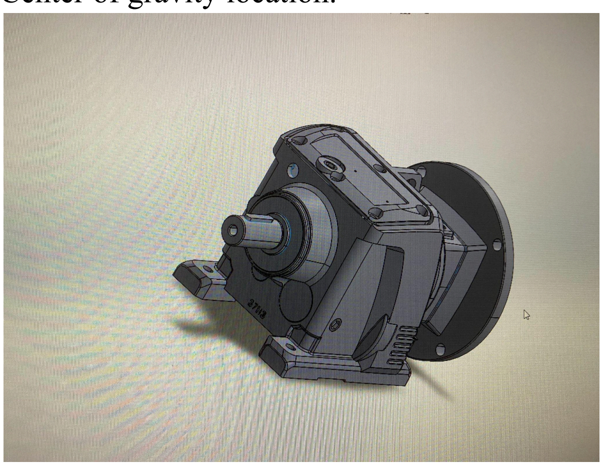
We will mount the gearbox to the flange by drilling extra holes through the cast iron plate on the top of the gearbox, having the gearbox "sit" on top of the flange, fitting into it, and use bolted joints (bolt and a nut, likely double-nut to prevent self-loosening from any vibrations) to clamp the gearbox to the flange.

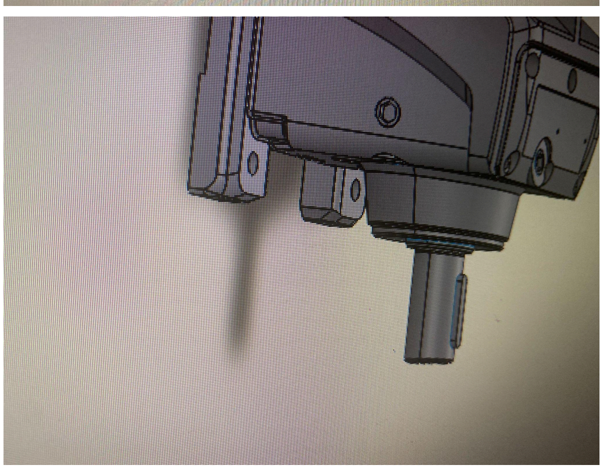
Main Drive System Risks/Concerns and Actions (Tests + Further Analysis)

With regards to risks, there are quite a few concerning risk items ever since delivery of the new gearbox and actually have it in our hands and with the mounting solution we devised so far

Risks or Design Concerns (is there a better way?)	Actions
The welds that are fixed supports in this transmission stack up – between the new output shaft (coupled to the gearbox output shaft) and the conical bit (which is welded to the cutterhead) is worrisome and can pose troubles during development + testing because for any mistakes, we'd have to cut and grind through which will just make the material worse especially if we have to weld again – and that will lead to a terrible weld.	Mechanical Engineers on the team (Rishu, Michael) need to spend more time coming up with a smarter way (that is also not expensive) to transmit torque from the gearbox to the cutterhead without any welding – looking into bearings or some type of swivel eye-joint. These are immediate actions we are working on and will coordinate updates with TBC.

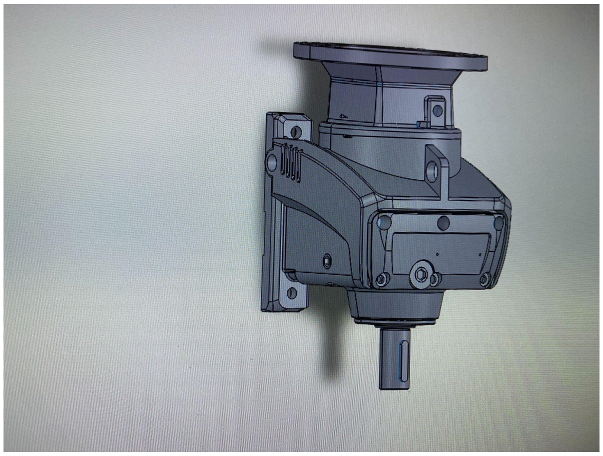
The speed reducer came to us with some surprises – particularly on the location of the center of gravity and how big the footplate on the gearbox was. We couldn't find an inline gearbox that met all of our aforementioned constraints that was also meant to be mounted radially as well – and we thought we could just use this gearbox and make our own flange to fit the top plate. However, the CG seems to be "lower" in the gearbox and can be closer to those rectangular bars on the footplate. We tried to locate the CG in the CAD model but the model did not have any center of mass/material properties. This poses a big risk that since we orient the gearbox in the -z direction, if the CG is not in the neutral axis, then that would create a moment about the neutral axis and cause the TBM to tip. We might have to update the mounting strategy of the gearbox to the TBM structure based on the Center of gravity location.





- We need to perform an experiment to figure out its CG. Two options (second option is way easier)
    - We hang the gearbox from some contraption or a table and let the gearbox pivot around some point attached to a cable, and freely hang. Use a plumb line to mark the line going through the pivot. Then, redo the experiment by letting the gearbox pivot around another point on the machine and redo. Then, take the intersection of both lines as the centroid of the gearbox
    - We push the gearbox slowly off the edge of a table right until it is about to fall – then do it again but oriented 90 degrees away from original trial, and push again – then find the intersection point of the first trial's edge and the second trial's edge
- Once we figure out the CG location, we need to either
    - add a counterweight (which can be determined using basic statics hand-calcs)
    - if that is infeasible then buy an entirely new gearbox
    - or design a support structure (kind of like a suspension system) fixing the gearbox to the TBM

Therefore, due to time constraint, this risk item is imperative for us to figure out and is an immediate concern. We will coordinate with TBC with updates ASAP.

	
A possible failure mode is those fasteners that clamp the flange to the TBM structure itself – if those fasteners fail in shear from the reaction through our designed flange which mounts the gearbox to the TBM structure If this happens, then the gearbox and motor are free to spin and that is not good.	● Basic hand-calcs for shear failure mode of each fastener and size fasteners accordingly ● If necessary, do FEA
TBM Muck Removal Cavity failure - if the fasteners that clamp the muck removal bottom cavity of the TBM to the outer structure skin, fail in shear from the axial loading upwards across the fastener head – this can cause failure of mission and also reduce the torque transmission from the motor greatly as the entire TBM will be full of muck and that can corrode the shaft transmissions, and the muck vacuum hoses won't be able to suction soil out of its vicinity and above it	● Basic hand-calcs for shear failure mode of each fastener and size fasteners accordingly ● If necessary, do FEA

Muck Removal System

Muck Transport System Overview:

Two hoses are fixed to a bulkhead behind the cutterhead in the TBM. Each hose will feed out to a joint that splits off into two 2” hoses, each connected to one of four 16-gallon Shop Vacs (which will be placed on the ground’s surface approximately 5 meters away from the TBM and hole). Currently, as mentioned in our Power systems section, we are trading on doing three 20-gallon Shop Vacs to have a balanced three-phase line but the general system is the same.

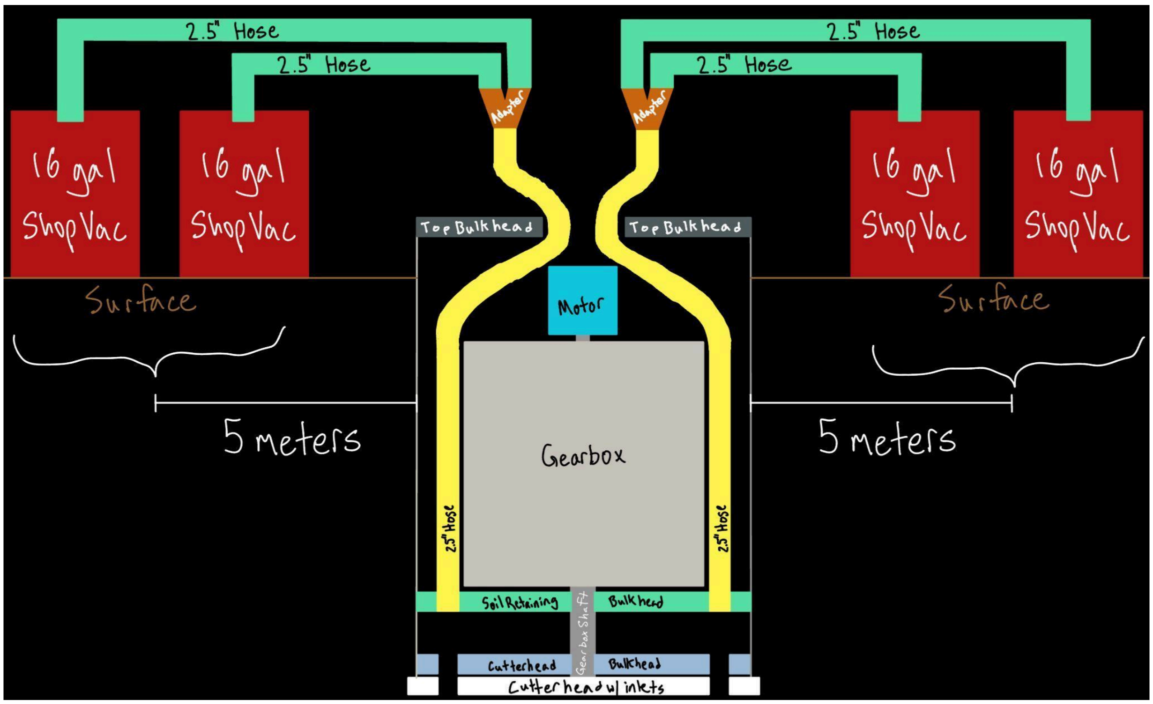


Figure 1: Diagram of muck removal transport system.

Estimated Throughput:

$$Excavation\ Volume\ =\ \pi r^2 h\ =\ \pi(\frac{.5m}{2})^2(1m)\ =.19635m^3\ =\ 51.87034gal$$

$$Total\ Chamber\ Volume\ =\ 2\frac{Shop\ Vacs}{Connection}\times\ 2\ Connections\ \times\ 16\ gal\ =\ 64\ gal$$

Minimum vacuum total Airflow

Max Linear Velocity of TBM - 31.4 meters / minute (Calculation below, based on Machinery’s Handbook 31 p. 1084)

$N\ =\ \frac{1000\cdot v}{\pi D}$, Where N is spindle speed in rev/min, v is linear velocity in m/min, and D is the cutter diameter in mm. Assume a conservative N of 20 rev/min. D is .5 m or 500mm.

Max Linear velocity is $v\ =\frac{\pi ND}{1000}=\frac{\pi(20)(500)}{1000}=31.4\frac{m}{min}$,

$$Excavated\ Soil\ Accumulation\ Rate\ =\ \pi r^2 \times\ Linear\ Velocity\ =\ \pi(\frac{.5m}{2})^2 \times\ 31.4\frac{m}{min}\ =\ 6.17\frac{m^3}{min}$$

Therefore, our estimated throughput is **6.17 m^3/min**. We can calculate the minimum airflow as follows:

$$4\ Shop\ Vacs \times \frac{Airflow}{1\ Shop\ Vac}\ =\ Excavated\ Soil\ Accumulation\ Rate \rightarrow$$

$$Minimum\ Airflow\ =\frac{Excavation\ Soil\ Accumulation\ Rate}{4} = 1.54\frac{m^3}{min} = 54.3846\frac{ft^3}{min}$$

Muck Removal V0 Test Protocol (Test Completed on 1/27):

Purpose: Testing muck removal simulates on-site events and conditions which may not be entirely foreseen with theoretical modeling and design, giving essential information on how to improve this particular subsystem and our overall tunnel boring machine. Because of the volume constraints of our 3D-printers, we created a CAD file of our TBM muck cavity with dimensions closest to our full-scale TBM. Our cavity has a capacity of 1066.806 cm^3—17 cm inner diameter (~0.25 scale) and 4.7 cm height (full-scale).

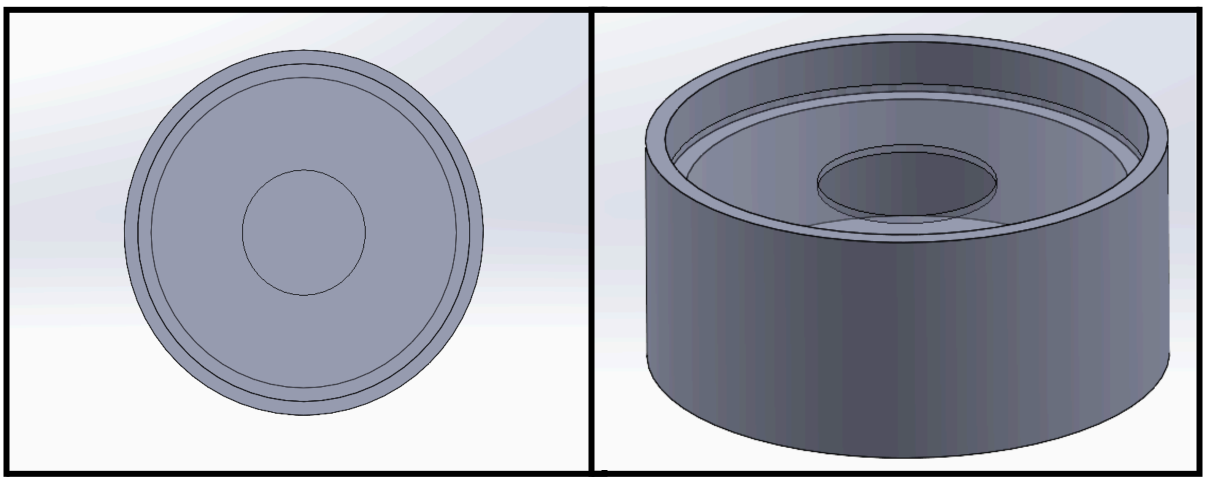


Figure 2: Top view (left) and side view (right) of muck cavity and lid CAD assembly.

Equipment:

Product	**Quantity**
Power outlet, single phase 120VAC	1
Husky 16-gallon Shop Vacuum	1
RIDGID 16-gallon Shop Vacuum	1
13 feet RIDGID 2.5in extension hose	1

Muck cavity with 17 cm inner diameter, 4.7cm height for ¼ scale V0 test	1
Muck cavity lid	1
Ultisol red clay soil	≥ 0.282 gallons
Sand	≥ 0.053 gallons
5 gallon bucket for soil mixing	1
Measuring cups to determine volume	1
Timer	1
Safety goggles for soil dust protection	1

Procedure:

1. Create mixture (80% ultisol clay soil, 15% sand, 5% water).
2. Fill the muck container with mixture that has ¼ volume of the container (container volume is (π * (8.5 cm)^2 * 4.7 cm = 1066.806 cm^3 * (1 gallon / 3785.41 cm^3), so container volume is *0.282 gallons*).
3. Connect the shop vacuum to the extension hose.
4. Connect the extension hose to the clear acrylic lid and seal the connection.
5. Connect the vacuum's cord to the power outlet. While pressing the lid to fix it in place, start the shop vacuum and timer.
6. Continue pressing the lid and stop the timer once all the soil has been vacuumed. Record results and observations.
7. Repeat Steps 1-5 with ½, ¾, and full volume of the container.

Figure 3: Muck cavity filled with soil (left), Full V0 muck removal test setup (right)

Observations/Results:

1. The vacuum hose begins shaking after it is turned on, so some structure in the TBM is needed to constrain hose movement.

2. However, vacuuming is extremely slow with little hose movement. During one vacuum test with little to no hose movement, we turned off the vacuum after 10 minutes—and ~12.5% of the original soil was still remaining.

3. Therefore, because of **Observations 1** and **2**, the constraining structure must not completely constrain hose movement—it must allow *some* hose movement so it can move around within the opening of the muck cavity (the diameter of the muck cavity opening, 2.7", is greater than that of the hose, 2.5"). This hose movement enables it to vacuum more effectively for a larger area.

4. Shop vacuums operate on the principles of *pressure difference* and *pressure equalization*. The motor in the shop vac creates a pressure difference—generating a lower pressure inside the vacuum than outside of it. The carrier fluid – in our case air – will flow from high to low pressure, the air outside of the shop vacuum will flow into it, acting as the medium which carries soil particles into the vacuum. Therefore, to increase the soil to be vacuumed through the hose, we must enlarge the exposed openings in the vacuum exposed to the atmosphere to allow the creation of a pressure difference. Otherwise, there will be zero air flow to carry the soil particles into the vacuum. Our exposed openings in the test weren't big enough.
5. When the vacuum hose is moved rapidly up and down, pressure from the hose pushes the compressed air to the side of the finite volume cavity when it moves down during the

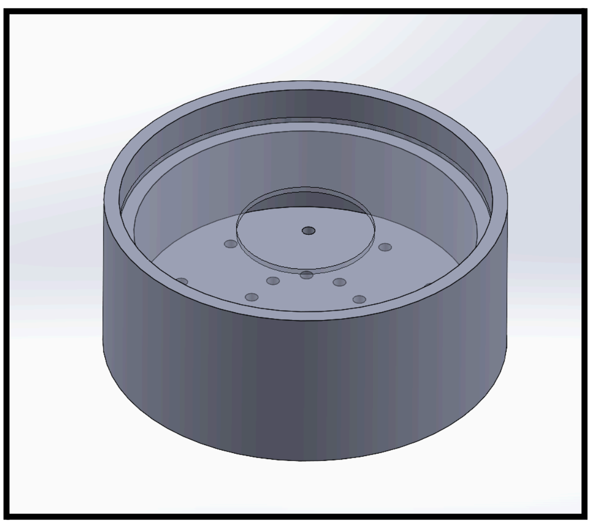
downward motion, and that air carries the soil hitting the finite volume walls. That impulse reaction force on the soil from the walls carries the soil away back towards the middle closer to the hoses.

6. We definitely need to use a soil conditioner to aerate or liquify the soil more.
7. We need to increase hose diameters to allow for increased suction surface area coverage

Future Testing - Muck Removal V1 Test Protocol:

Based on our observations from our V0 test campaign, we are redesigning our muck cavity to create the most accurate simulation possible of true TBM digging conditions. We have now added 20 holes—2 concentric circles with 10 holes each—to simulate air gaps resulting from excavation due to the rotational cutterhead motion. Because of the circular motion of the cutterhead, a circular pattern of particulation and air gaps is created in the soil, which is why we specifically chose to design our new cavity with concentric circles.

Due to the same volume constraints of our 3D-printers (see Muck Removal V0 Test Protocol for more details), the new CAD file of our TBM muck cavity has the same dimensions as before: a capacity of 1066.806 cm^3, 17 cm inner diameter (~0.25 scale) and 4.7 cm height (full-scale).



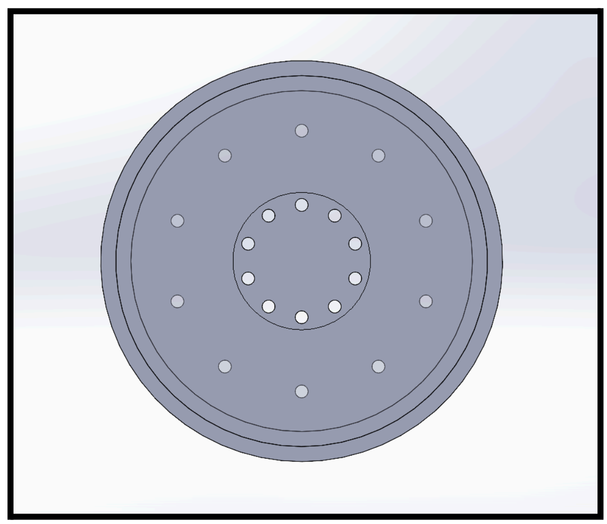


Figure 4: Top view (left) and side view (right) of modified V1 muck cavity container and lid CAD assembly.

Equipment:

Product	Quantity
Power outlet, single phase 120VAC	1

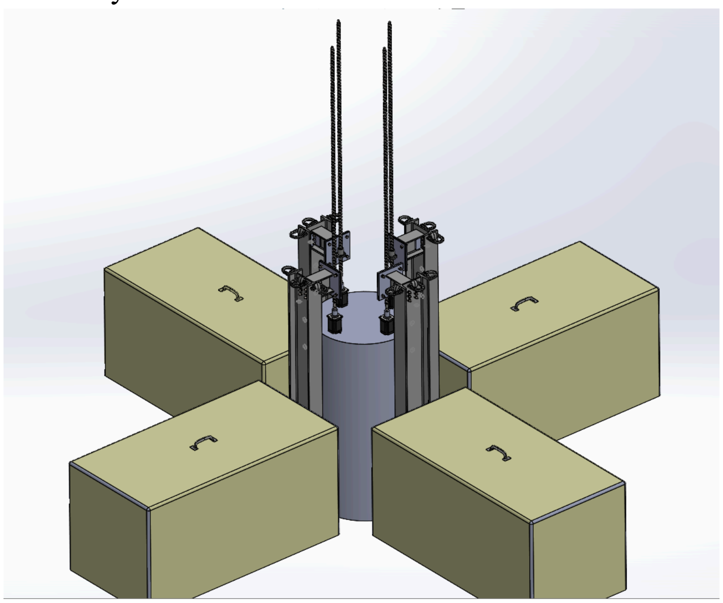
Husky 16 gallon Shop Vacuum	1
RIDGID 16 gallon Shop Vacuum	1
13 feet RIDGID 2.5in extension hose	1
Muck container with 17 cm inner diameter, 4.7 cm height, 20 holes	1
Muck container clear acrylic lid	1
Ultisol red clay soil	≥ 0.282 gallons
Sand	≥ 0.053 gallons
5-gallon bucket for soil mixing	1
Measuring cups to determine volume	1
Timer	1
Safety goggles for soil dust protection	1

Procedure:

1. Create mixture (80% ultisol clay soil, 15% sand, 5% water).
2. Fill the muck container with mixture that has ¼ volume of the container (container volume is (π * (8.5 cm)^2 * 4.7 cm = 1066.806 cm^3 * (1 gallon / 3785.41 cm^3), so container volume is *0.282 gallons*).
3. Connect the shop vacuum to the RIDGID extension hose.
4. Connect the extension hose to the clear acrylic lid and seal the connection.
5. Connect the vacuum's cord to the power outlet. While pressing the lid to fix it in place, start the shop vacuum and timer. Shake the tip of the vacuum hose up and down to simulate the vibration of the actual TBM during excavation.
6. Continue pressing the lid and stop the timer once all the soil has been vacuumed. Record results and observations.
7. Repeat Steps 1-5 with ½, ¾, and full volume of the container.

Propulsion System

Propelling the TBM downwards will require a thrust force that is dependent on friction, soil-density, excavation volume, and other relevant factors. We plan to use 4 actuators placed around the top head of the TBM to push the machine into soil. The force output and reliability of this subsystem is critical for success. This document describes this subsystem in its entirety.



The layout of this section of the technical report is as follows:

- Parameters & Assumptions
- Technique Trade
- Force Analysis
- Shaft Sizing
- Support Structure Overview
- Circuitry & Control
- Thrust Ring
- Testing Campaign
- Mass, Dimensions, and Bill of Materials (BOM)

Required Thrust Calculations (Predicted Loads)

<u>Parameters & Assumptions</u>

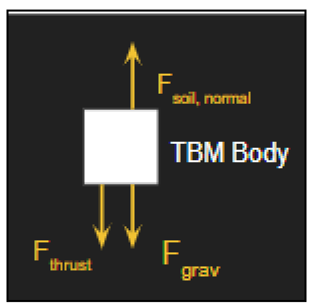
Excavation Radius	0.25m
Excavation	1m

Depth	
Soil Density [1]	1550 kg/m3
TBM Mass	50kg
F.O.S	2

We used F.O.S (Factor of Safety) in the sense that we want to design our system to withstand a higher thrust force than lower, so that we could push the propulsion system further and increase digging speed.

We used a F.O.S of 2 mainly due to the assumptions we made in our initial thrust calculations below and line of reasoning using the amount of soil excavated and normal force from it – which invariably has some errors. Additionally, by using a FOS of 2, we design the system for the “worst-case” scenario with the highest thrust force from the propulsion system. The actual thrust force we can push it to, is determined by our testing (which we describe later in this section).

Thrust Calculations



Therefore $\mathbf{F_{thrust}} = \mathbf{F_{soil,normal}} - \mathbf{F_{grav}}$

Volume of Soil Excavated (m^3)

0.20

Mass of Soil Excavated (kg)

304.34

Weight of All Soil Excavated (N)

2985.59

Mass of TBM (kg)

50

Weight of TBM (N)

490.5

Fnet (Fsoil - Ftbmweight) (N)

2495.10

Fnet, w/ FOS (N)
4990.2

Force Required per Actuator, w/ FOS (N)
1247.55N

Note: Frictional forces between the TBM body and soil are negligible since the cutterhead diameter is greater than the TBM diameter and can therefore be ignored.

Propulsion: Technique Trade

In determining methods of high-force propulsion, the two strongest electric methods are premade linear actuators and custom ball screw assemblies.

Linear Actuators		Ball Screws	
Pros	**Cons**	**Pros**	**Cons**
High reliability	Capital inefficient	Capital efficient	Reliability Risks
Simple implementation	Limited customization	High customizability	Less force exertion
High force exertion	Complex support stand	Simple support stand	Highly vulnerable to contaminants
Time efficient	Involves pipe-jacking (Slow)		Time inefficient

With $F_{thrust, req}$= 624N, and F_{thrust} = 1248N per actuator,
we believe **Ball Screws** are superior for this application

Their benefits of high-customizability and low-cost outweigh their downsides in reliability and force-exertion.

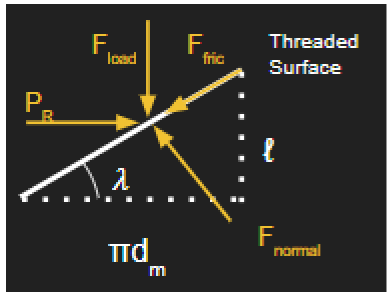
Force Analysis

The torque of a motor shaft that is required in order to apply a certain amount of force on a ball screw nut is dependent on shaft and nut parameters

$$T_{LOWER} = \frac{Fd_M}{2}(\frac{\pi F d_m - l}{\pi d_m + Fl}) \, T_{RAISE} = \frac{Fd_M}{2}(\frac{l + F\pi d_m}{\pi d_m - Fl})$$

Where:
d_M = major diameter (mm)
λ = lead angle (°)
ℓ = lead / pitch (mm)
F = force of nut on shaft (N)

Derived using force equilibrium of shown FBD and Newton's second law:



Assumptions:

d_M	16mm
ℓ	10mm
$\mu_{shaft,ball\text{-}nut}$	0.01
F_{load}	1247.55N

Torque to Raise TBM (Nm)*
0.32
Torque to Thrust TBM (Nm)*
0.99

When raising the TBM, the net force that must be counteracted is only the weight of the TBM. When thrusting the TBM into the soil, the net force that must be counteracted is the force from the soil minus the weight of the TBM. The above torque values are derived using our hand-calc spreadsheet.

**These torque calculations caused some confusion on our PDB. Here the ball nut is being raised (we fix it to the structure) and as the shaft goes down it pushes down on the TBM. When the ball nut is lowered, the shaft is coming up and no longer is applied on the TBM. The lowering calculation is not the value we are looking for - it is the raise calculation that has been restated here as torque to thrust TBM.*

Shaft Sizing Calculations

Ultimate Tensile Strength (UTS) (MPa)	310
Shaft Radius (m)	0.008 (Assuming sfu1605 ball screw)
Max. Shear Strength/ Tau (MPa)	155 (UTS x 0.5 is common practice)
Max. Torque allowed through shaft (Nm)	166.21
with FOS = 2 (Nm)	**83.11**

$T = 2/3 \cdot \pi \cdot r3 \cdot \tau$

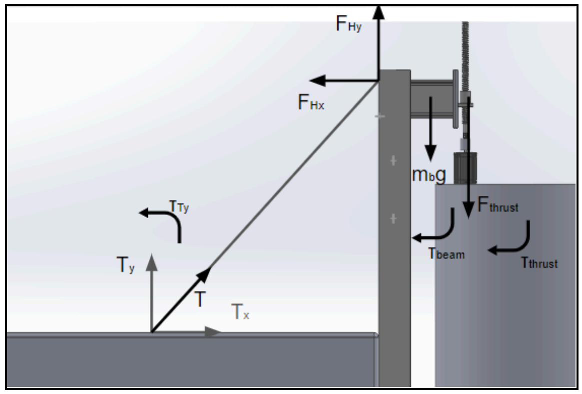
Formula from D'Souza et.al, The Limit Torque of Regular Polygonal Shafts)

Torque acting on each shaft from Thrust calcs = 2.46 Nm < 83.1 Nm
Therefore, the ball screw can handle the torque being put through it.

Support Structure Overview

The system uses I-beams and tensioned cables (connected to concrete blocks), in order to force the motor to drive the ball screw down when running.

- Steel A36 I-beams are chosen since they are easily sourced, high load-bearing capacity and high moment of inertia. This is "Because an i-beam is rolled into that "I" shape, the i-beam will have more flange surface area than a solid beam. For the same amount of steel used and the same overall weight, you'll get a higher load-bearing capacity and a high moment of inertia with an i-beam because of its ability to redistribute that weight." as per Bushwickmetals.com.
- I-beam to I-beam connection using stainless steel clip angle, welded to one beam, bolted to other
- I-beam to ball nut connection using stainless steel plates welded to components, M8 bolts together
- All bolted joints use double nut lock
- Main center i-beam has tie-downs welded for anchoring to ground and concrete block
- Secured to ground using pikes for any lateral disturbance (by team members or staff)



Finding mb	
Density of I-beam steel (kg/m^3)	7850
Volume (m^3)	0.00022
Mass (kg)	1.71
Radius for torques determine by dimensions of given concrete block, spatial arrangement of support structure and TBM	
Angle between F and block	58.44 degrees

Solving Tension in Cable	
$\boldsymbol{\tau_{beam} + \tau_{thrust} = \tau_{Ty}}$	
Tsin(1.0233)(0.61m) = 1247.55(0.15m)+(1.71)(0.076m)	
Tension (N)	**365.52**

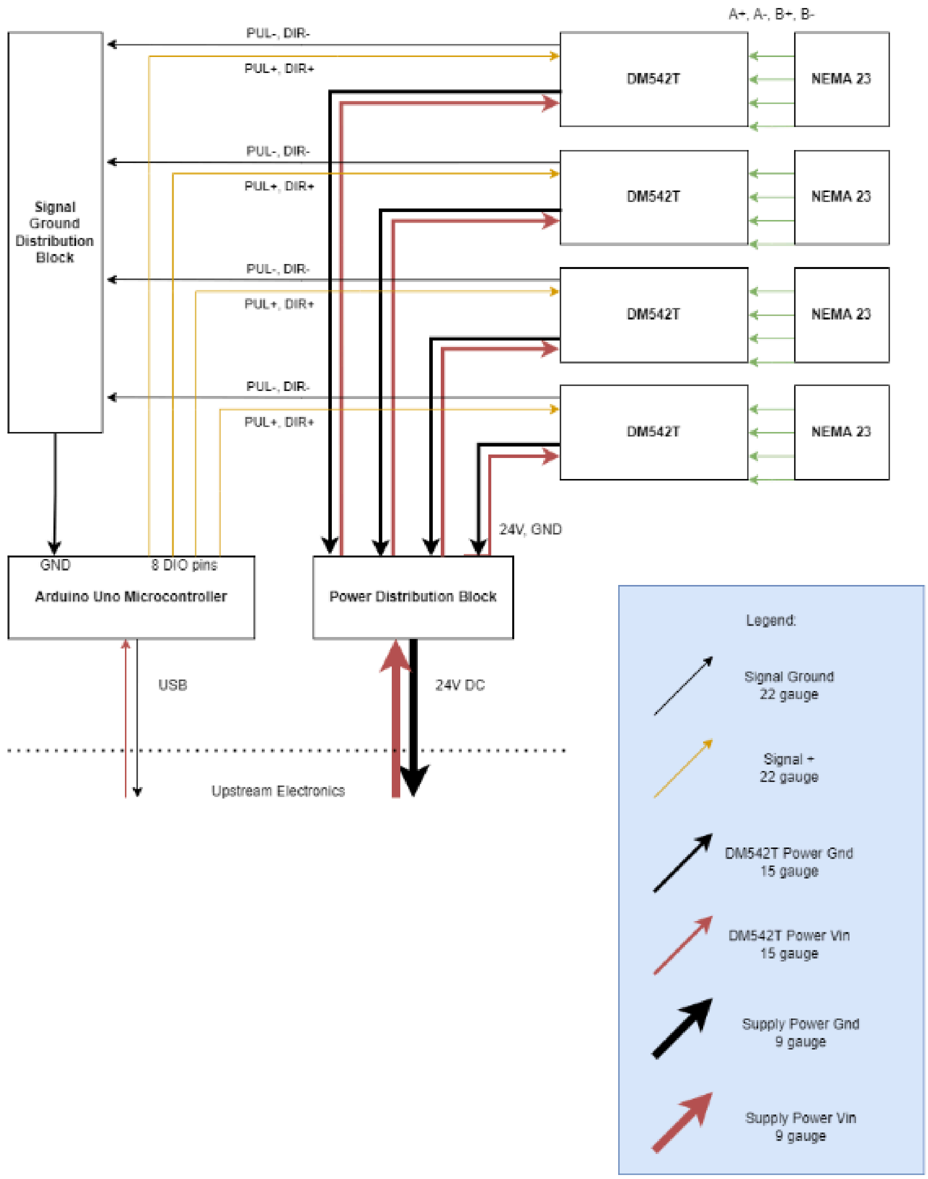
Based on this number a cable was chosen with the following properties:

- T304 stainless steel
- 1/16” OD, 200ft long
- 480 lbs breaking strength

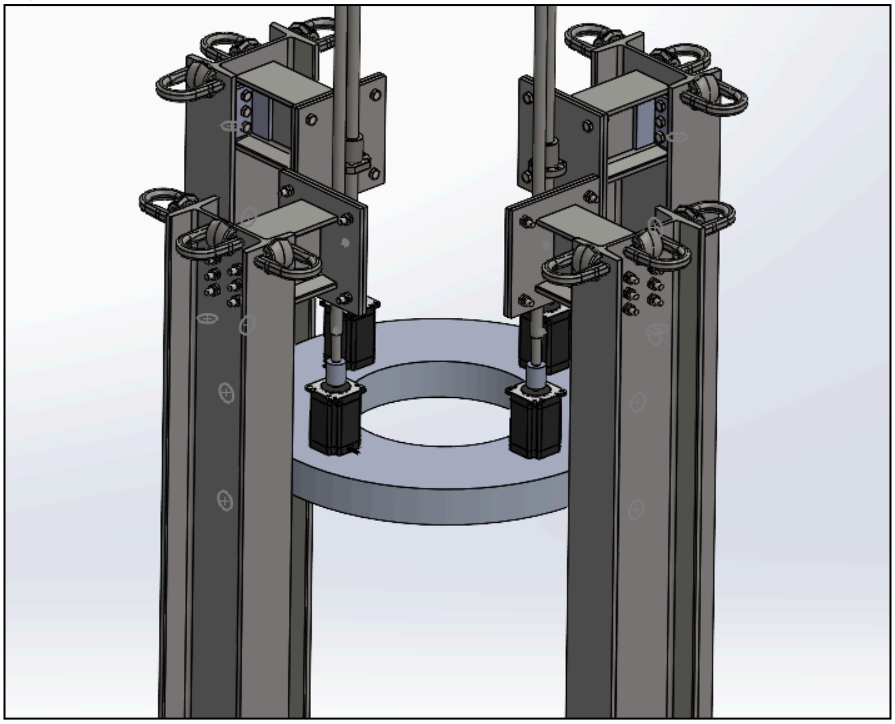
-

Circuitry & Control

The propulsion system requires microcontroller signals for control of actuator motors. Below is the circuit diagram for the entire propulsion system.



Thrust Ring - Overview



Updated propulsion structure with thrust ring in center

Annulus Dimensions:
Outer Diameter: 20 inches
Inner Diameter 10 inches
Thickness: 2 inch

The reaction torque transferred to the TBM structure (which we allow to happen, and reverse cutterhead if we witness the TBM structure rotating too fast and if there are any other issues) causes the TBM structure to rotate. With our previous propulsion approach, the DC Brushed Motors were mounted to the top of the TBM using bolted joints, and the lead screw shaft would push down on the motors -- pushing down on the TBM -- as it is rotated by the Motor operation because the ball screw nut on the shaft is fixed to the external propulsion structure.

One of the major concerns raised in the PDB feedback was that because the propulsion system is pushing down on the TBM continuously -- and since the TBM is allowed to rotate -- this reactive torque on the TBM gets transferred to the propulsion structure and would cause the motor + shaft + propulsion structure to rotate and collapse.

Our solution is a new propulsion system where instead of continuous thrust down on the TBM, we periodically apply thrust on the TBM through a thrust ring that makes contact with the TBM structure periodically and transfers axial force down onto it.

This new system brings about its own risks such as any torque transferred to the thrust ring and full structure during that contact time period, how long that contact time period will be and thus what are the predicted dynamics here. Tests will be conducted to determine the time period.

Further analysis will be completed to value the dynamics before moving to full tests with a functional TBM prior to the final competition date. We shall update TBC with the final test plan and results accordingly.

Failure Mode Analysis:
In order of importance

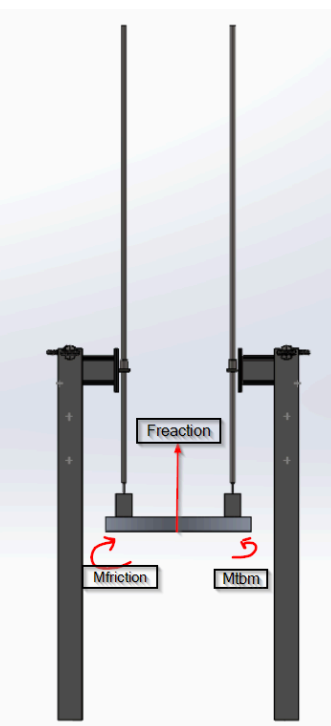
Failure Description	**Risk Mitigation**
The thumping mechanism is the biggest point of failure in this system. This is because it is crucial to the system digging and also digging quickly. In the short point of contact between the ring and the TBM head the torque transferred to the system must be absorbed by the components of each stand without collapsing.	Post testing the total force produced by the stem (last phase in test plans below), a test to determine the duration of contact will be conducted. The test may take place either with a functional TBM that rotates. Else we plan to test using a rotating setup. The team will slowly ramp up both rotation of the TBM head and duration of contact with code. This will help determine when maximal disturbance is observed in the system. More concrete and precise test plans will be drawn up and shared with TBC if needed.
The stepper motors may malfunction - causing an **imbalance in the movement of the thrust ring** and misalignment of the ball shaft with the ball ring.	Phase 2 of our propulsion sub-system tests detailed below) looks to test the reliability of the linear movement using a vertical test setup similar to that on the actual TBM setup. This will help us identify and troubleshoot motor errors and alignment. In the case of an unexpected failure when testing with the TBM under load, the team will use the manual controls to turn off all motors and adjust the malfunctioning motor until the thrust ring returns back to a flat position. However repeated and rigorous testing should ensure this does not happen.
Thrust ring may deform since load onto TBM is being applied at 4 specific points and may not actually evenly spread throughout the ring's bottom surface onto the TBM head.	Specific FEA on the ring will be conducted under different max load conditions with the motors emulating the force to observe if any deformities may occur. Due to the lack of time, the team has not performed this yet. This analysis will help us determine if a stronger (and inherently heavier) material needs to be used instead of the aluminum ring prior to purchase of the material.
The **threads on the ball screw may endure wear** since the aluminum thrust ring and motors are hanging off	Prior to every individual test with the thrust ring, the ring will be supported by a table or platform such that the weight does not lie on the ball screw until the

the few threads that are threaded into that nut. This may cause the actual runs (post testing) to have a less structurally sound ball screw.	start of the test. This procedure ensures that any weight time while testing and iterating does not damage the screw. In the worst case scenario, a second set of the chosen ball screws can be ordered given their short lead time (3-7 days).

FEA Results

We first started off our Propulsion Structures design, using the basic first-principles hand-calcs as described earlier in this section. However, due to the new thrust mechanism development and to get a better understanding of the deformations from the reaction forces transferred from the TBM to the Propulsion system, we had to resolve to Finite Element Analysis (FEA). FEA is conducted on the propulsion structure as a whole by importing a simplified CAD model from Solidworks into ANSYS workbench and using the Static Structural tool. The purpose of this FEA is twofold - to understand the dynamics when the TBM makes contact with the thrust ring, and to view the main points of stress on the system to understand what fixtures/ welds to improve if needed.

Below is the force body diagram for the system that will help determine where to place the loads. Two actuators have been removed since they block the view to draw an appropriate force body diagram. The FBD does not include reaction forces at the joints - this is because on the FEA these are assumed to be fixed. The reaction forces are featured in a previous section.



Here, Freaction is the same as the full thrust force calculated above of 2250 N (without FOS). Mtbm is the maximum reaction moment expected from the TBM of 0.8 kNm during point of contact. Mfriction is the moment from the friction acting between the aluminum thrust ring and the steel TBM head. The value is derived as follows:

$$M = \frac{2}{3}\mu_k F_{load}\left(\frac{R_0^3 - R_i^3}{R_0^2 - R_i^2}\right)$$

Source

μk = 0.45
Fload = 2250 N
R0 = 20 in. = 0.508 m
Ri = 10 in. = 0.254 m

Therefore,

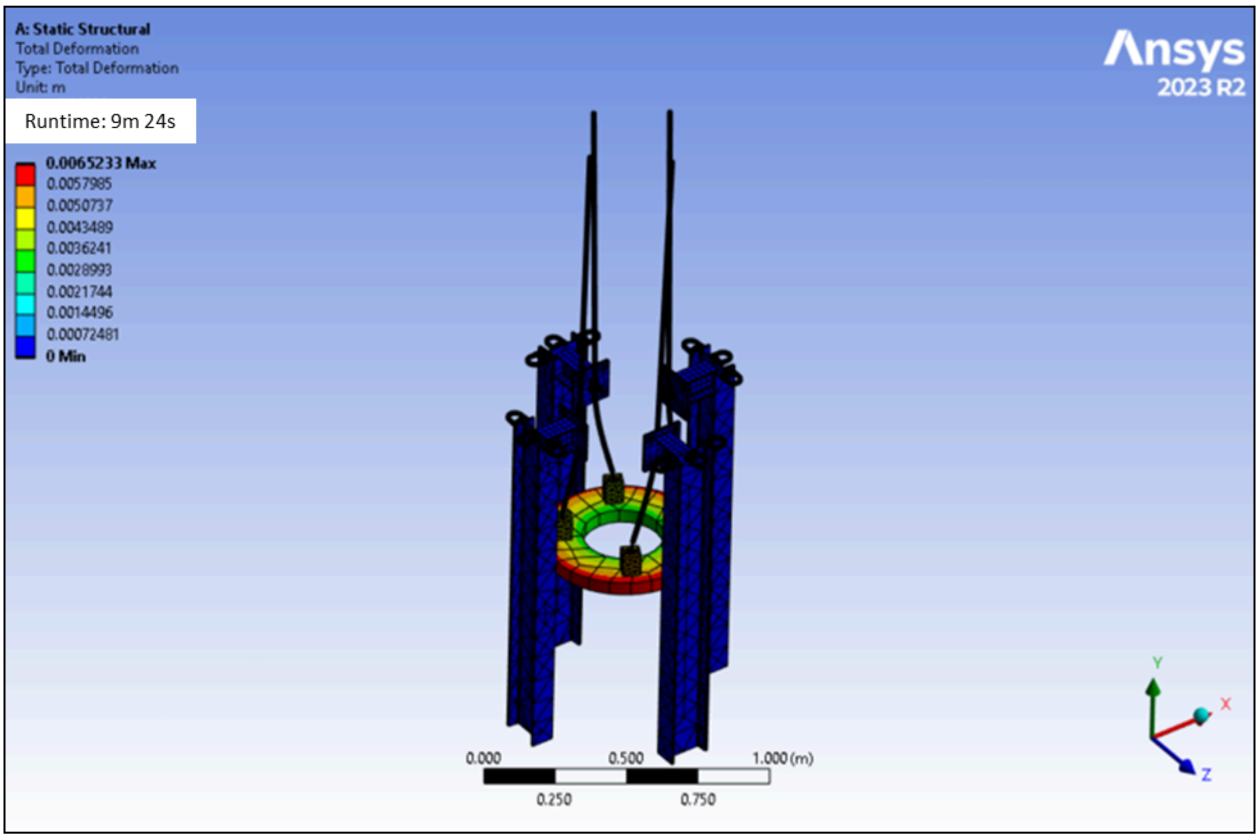
Mtbm = 400 Nm

With these values and noting that Mfriction and Mtbm act in opposite directions, **a force load of 2250 N and Moment of 400 Nm** is applied onto the bottom surface of the thrust ring in the FEA simulation.

The material properties are set to the appropriate materials as per information from the manufacturer. Certain simplifications were made to the model prior to the FEA for quick simulation.

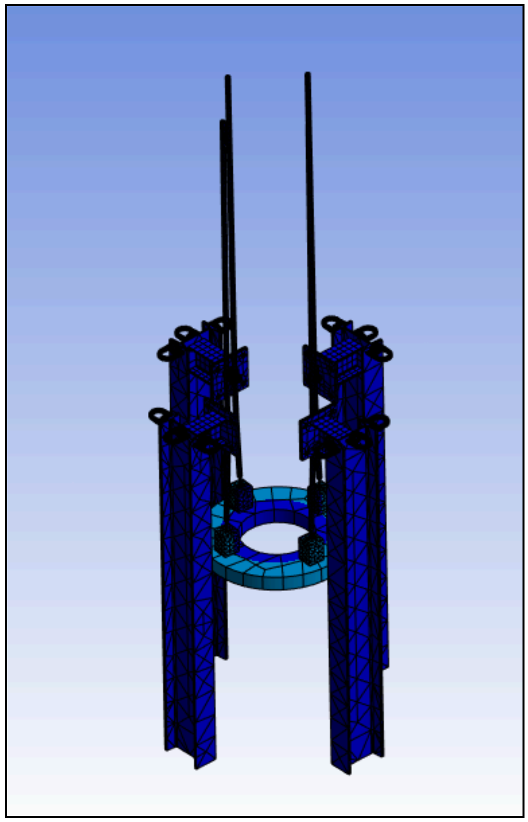
- All welds and clip angle joints are set to be simple connections.
- The bottom, and side of the I-Beam have been made fixed supports as we ignore the concrete block and tension running through the wire.
- Ball screws with threads have been simplified to be cylinders to ease meshing
- NEMA 23 stepper motors converted to match the shape and density - not inner workings.

FEA Result 1



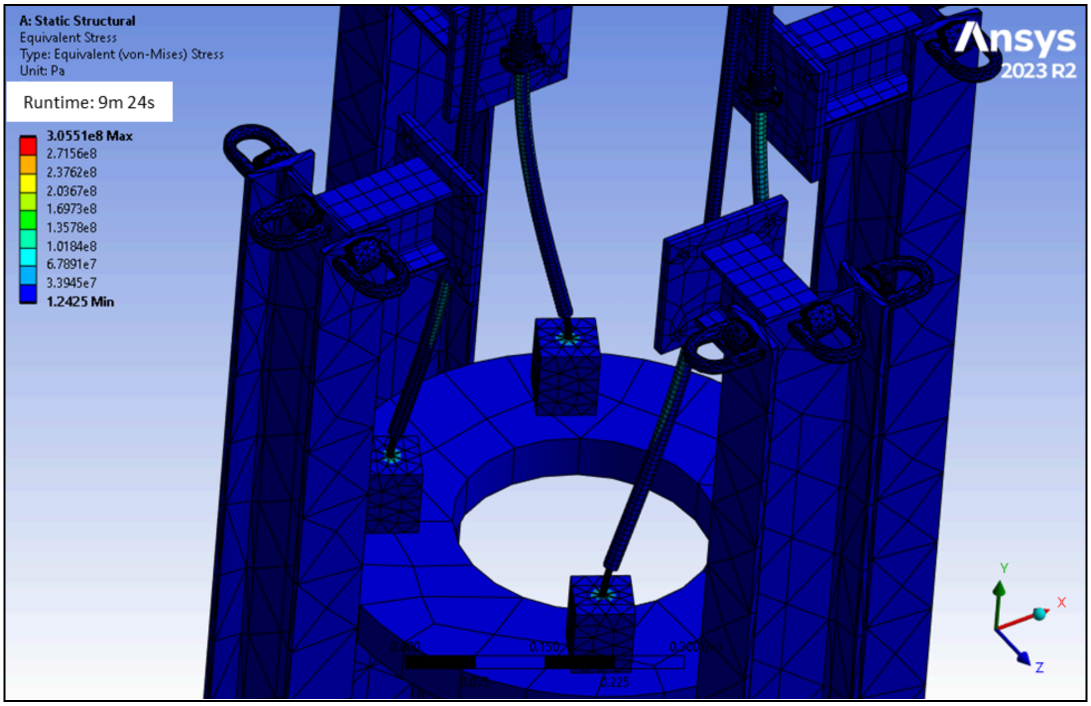
The analysis results reveal crucial insights into the system's structural behavior under the influence of the cutterhead-induced moment onto the thrust ring through the TBM head. The observed deformation of the thrust ring, reaching a maximum of 6mm which is a consequence of the axial loading and torsional forces induced by the net moment acting on its cross-sectional face.

In the first plot, it is evident that the support structures withstand the applied forces, while the deformation in the thrust ring is a result of the dynamic interaction with the cutterhead during the 1-second contact period. This interaction induces significant movement in the motors, leading to the buckling of the attached ball screw.



This second plot of the same FEA shows the deformation after 0.15 seconds. This helps visualize the difference that short interval thumping can make to improve the structural integrity of the system through minimal deformation on the ball screws.

FEA Result 2



This stress heatmap from the same FEA results show a distribution of internal stresses within the structure. The connection between the motor shaft and ball screw, facilitated by the coupler is a key concern. The stresses in this region, while not excessively high, show the importance of the coupling mechanism. The analysis of the ball screw material reveals that it can withstand the obtained pressure values, suggesting that the chosen aluminum couplers and ball screw material are adequate for the current loading conditions.

However, the buckling is still present due to the axial loading from the reaction thrust force onto the TBM structure, through the thrust ring.

This comprehensive understanding of the system's response to axial loading and torsional forces provides valuable insights for optimizing its structural design and ensuring its reliability under operational conditions.

Our propulsion system thus needs significant design changes. In addressing the observations made with the help of FEA, the primary concern is the effect of the moment transferred to the thrust ring from the TBM. These are urgent concerns and the team will update TBC when needed to ensure safety and completeness. The team looks to investigate imminently on other mechanisms to fix this issue. Some of these include a mechanism similar to that of a swashplate in helicopter rotors that allows a top plate to rotate while keeping the bottom one fixed or a swivel-eye joint. The ball screw+nut may also be sized up to better resist the lateral motion they will encounter. Another method to reduce the moment is to increase friction between the two surfaces that are in contact perhaps by introducing ridges in either of the surfaces or coating them with substances that increase friction. Finally the motors will be finally tested to look for minimal contact time <1 second.

Other notes:

The PDB feedback also noted a request to decrease the **number of bunker blocks** used if possible. Although many different orientations were considered. Using four actuators as per our current design requires for separate blocks to hook onto. The design does not require blocks of the size offered and can make do with blocks about half the size of the current (24”x12”x12”) blocks shared with our team by TBM.

All major fixtures that make use of fasteners with a nut on any side make use of double nut locks to increase the strength of the connections. The TBM will vibrate as per various analyses done by other sub-teams. The thumping mechanism will also induce vibrations that would loosen a nut on these fasteners. Therefore, a double nut lock with a thin nut screwed in first before a thick nut. The procedure described here will be employed while torquing these nuts.

Testing Campaigns:

In order to rigorously test the propulsion system in order to mitigate risks of failure, we separated our testing phase into four distinct phases:

A. (Concurrently with remaining tests) Structure testing
   - Assembly and testing of propulsion system structure
B. Out of box testing
   - Ensures components were not damaged during manufacturing or transportation
C. Actuator velocity testing
   - Tests that each propulsion actuator translates at a uniform rate
D. Actuator force testing
   - Measures the max thrust that each actuator can exert across RPMs
E. Complete system testing
   - Tests the thrust and mechanical integrity of the entire propulsion system

Phase A: Structure Testing

a. Objective

To assemble the propulsion stands in their entirety. This will involve welding and drilling various ordered components and also fabricating certain components at nextfab. The structure must be able to be assembled when required and immediately connected to an appropriate test setup and motor further testing.

b. Safety

I-Beams are heavy, keep them on stable surfaces and not slanted surfaces.
General caution to be exercised when using heavy machinery at Nextfab.
No safety concerns when assembling parts with fasteners

c. Materials Required

All required components to be ordered for the structure with quantity and sizing where applicable have been detailed in the **propulsion BOM (below)**.

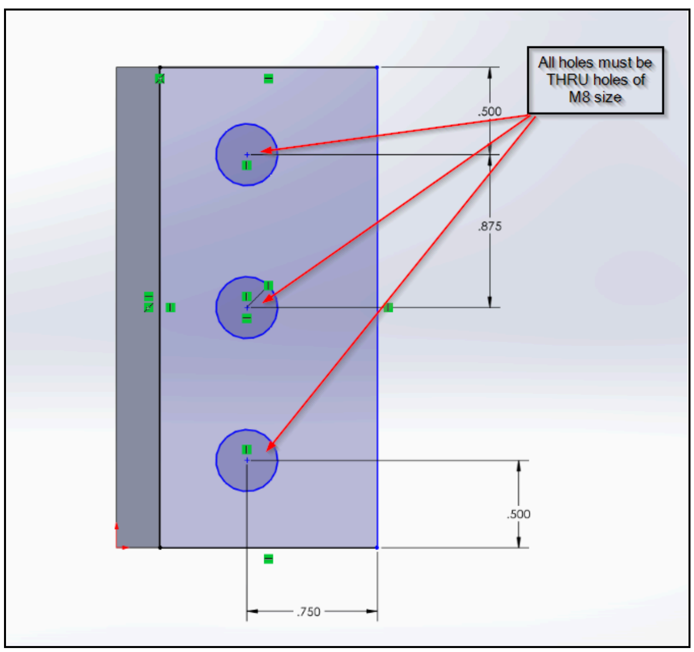
Nextfab equipment is required to:

1. Bend steel plates - Metal Stock Bender
2. Drill holes into the I-beam, Steel plates - Drill Press
3. TIG-weld various components (described below) - TIG Welder
4. Bandsaw to cut threaded rod
5. Cut steel sheet into ring - CNC Mill
6. Tapping into steel sheet - Tap

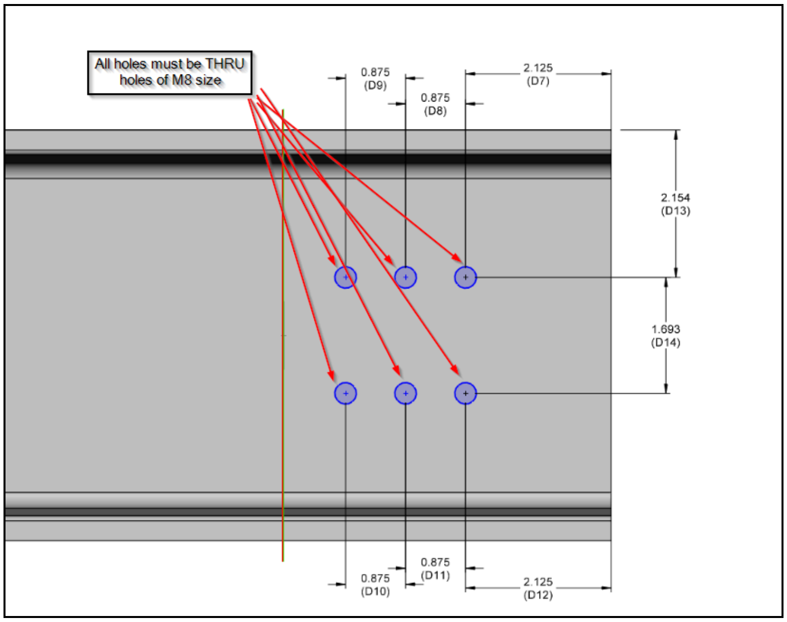
All other assembly steps are to be done by hand/ using basic tools available in-house.

d. Steps

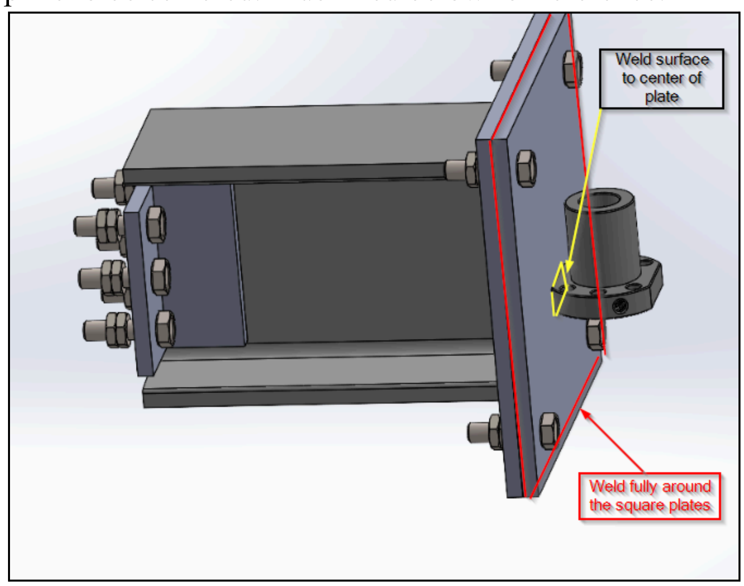
1. Steel Plates (2.75” x 3” plates) for Clip angles to be **bent** using the Metal Stock Bender at Nextfab across its length by 90 degrees (two sections of 2.75” x 1.5” each.
2. Remaining Steel Plates to be used as Connection Plates have M8 holes **drilled** on all 4 corners using Drill Press at Nextfab. Position: Center to be 0.8” away from two edges at every corner.
3. 3 M8 Holes **drilled** into one side of each Clip Angle using Drill Press at Nextfab. Position: drawing below.



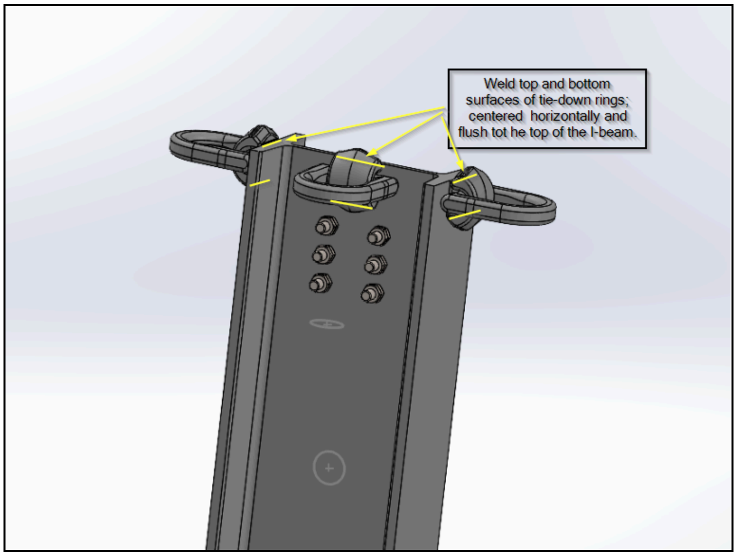
4. **Drill** 6 M8 holes in each Main I-Beam using a drill press. Drawing below for position of holes.



5. **Weld** every connection plate to the edge of the flat side of both sides of the 4 Side I-Beams. Rishu will weld at NextFab
6. **Weld** 4 connection plates to the other side of every side I- Beam. Connection plate to be centered. Illustrated below for reference.



7. **Weld** all tie-down rings to the I-Beams as shown below. Position: centered from both sides, flush to the top of the I-Beam.



8. **Cut** threaded rods to 16 6” pieces.
9. Use the CNC Mill to **cut** into the aluminum sheet to make a ring of outer diameter 20” and inner diameter 10”.
10. **Tap** 4 sets of 4 M4 holes 1” deep into 4 symmetrical edges of the sheet to moun the motors. Each set of 4 holes need to be exactly 47.14 mm apart from each other in a square orientation.
11. **(Once electronics testing is done,) Weld** the other 4 connection plates to the 4 ball nuts ensuring the ball nut is centered with respect to the square plate. Illustration below
12. Attach main I-Beams to side I-Beams with M8 fasteners and double nut locks. Also attach connection plates with M8 fasteners and double nut locks.
13. Thread the ball screws through all the ball nuts. Attach the motor to the coupler to the bottom of each ball screw. Attach motors to the tapped holes using the threaded rods and M4 nuts.

Phase B: Out of Box Testing

a. Test Objective

High level objectives are to establish the functionality of the:
- (x4) NEMA23 stepper motors
- (x4) DM542T motor drivers
- Power distribution block
- Signal ground block

So any defective products can be identified and returned ASAP.

The following specific objectives must be met:
- The DM542T drivers must send accurate and uniform commutation signals to the NEMA23 motors while not exceeding 40 degrees celsius (maximum allowable value as specified in the datasheet)

- The motor shafts must rotate and change direction without the motor bodies exceeding 130 degrees celsius (maximum allowable value as specified in the operating manual)
- The wires connected to the distribution blocks must be sufficiently rigid such that they will not lose contact during operation.

Since dig time is ~10 minutes, a 30 minute test for driver and motor functionality should suffice. To complete the specified objective, this test include:

- Acceleration
- Deceleration
- Direction changing

b. Outcomes / Dependent Variable

Variable	Value Constraints	Reason
NEMA23 temperature (C)	130C max	Maximum allowable value as specified in datasheet
DM542T temperature (C)	40C max	Maximum allowable value as specified in operating manual

c. Independent Variables

The knobs of control for this test is:

- The rotational velocity of the motor shafts.

The motor shafts are spun by using the four signal pins on the DM542T. Below are each of the signal and power lines, their purpose, and how they are connected:

Name	Purpose	Connection
Motor Wires (A+, A-, B+, B-)	Commutate the stepper motor phases	A+, A-, B+, B- pins on DM542T
PWR+, PWR-	24V power supply for the DM542T	Power Distribution Block terminals
PUL+, DIR+	Signal input pins on the DM542T for specifying motor direction and pulses	Arduino UNO or Teensy (test microcontroller)
PUL-, DIR-	Signal ground pins	Signal Ground Block terminals

Power Distribution Block	Supplies 24V for the four DM542Ts	Upstream 24V Full Bridge Rectifier supplies 24V to the four DM542Ts in parallel
Signal Distribution Block	Creates a common signal ground for all the DIR- and PUL- pins	All four pairs of the DM542T's PUL- and DIR- pins are connected to the Arduino Uno's ground pin.

The AccelStepper library will be used for programming. The sequence of commands that will be sent to the NEMA 23's for testing can be found in part (g).

d. Data Gathering

Variable being Measured	Measurement Device	Reason
NEMA23 temperature (C)	IR Thermometer	Our NEMA23s do not have built-in thermal sensors.
DM542T temperature (C)	IR Thermometer	Our DM542Ts do not have overcurrent, overvoltage, or overheating detection.

e. Safety

There are very few safety risks associated with this stage of the test. However, the following precautions should be taken when the system is energized:

- Safety glasses worn
- No hands-on contact with components

f. Materials Required

Assembly Materials:

Name	Description
NEMA 23 Motors (x4)	Stepper motors
DM542T	Stepper drivers that control the NEMA 23s
Power Distribution Block	Distributes the 24V supply power (from upstream rectifier) to all the DM542Ts in

	parallel
Signal Ground Block	Creates a unified ground for all the DIR- and PUL- pins, requiring only one ground pin used on the Arduino
2x2' MDF board	For mounting the DM542Ts, Power Distribution Block, and Signal Ground block
3.5mm screws	Mounts the DM542T, Power Distribution Block, and Signal Ground block to MDF board
9 AWG copper wire	Connects 24V supply to the Power Distribution Block supply terminal
15 AWG copper wire	Connects the Power Distribution Block outputs to the DM542Ts
22 AWG copper wire	Connects the eight PUL+ and DIR+ pins to the Arduino Uno DIO pins. Connects the PUL- and DIR- to the signal ground block outputs Connects the Signal Ground Block input to the Arduino Uno ground pin

Testing Materials:

Name	**Notes**
IR thermometer	Penn most likely has one. Required for testing NEMA23 and DM542T temperatures.

g. Test Plan Steps

The electronics must be mounted to the test board in the following sequence:

1. Mounting holes for 4x DM542T, Power Distribution Block, and Signal Ground Block are marked onto MDF board using pen
2. 2mm diameter pilot holes drilled into MDF board using handheld drill
3. 4x DM542T, Power Distribution Block, and Signal Ground Block screwed into MDF board using 3.5mm self-threading screws
4. 24V power line wired to Power Distribution Block by screwing in 9 AWG (19A capacity) or thicker wire using included ring terminals

5. Eight wires of 15 AWG (4.7A capacity) or thicker are screwed into the Power Distribution Block terminals and connect to their respective PWR- and PWR+ terminals on the DM542Ts
6. Stepper motor wires (A+, A-, B+, B-) are screwed into their respective terminals on the DM542Ts
7. Each of the PUL- and DIR- for all the DM542Ts are screwed into Signal Ground Block using 22 AWG wire attached with ring terminals
8. One wire connects the Signal Ground Block to the test microcontroller's GND pin using a jumper wire
9. Each of the PUL+ and DIR+ for all the DM542Ts are connected to DIO pins on Teensy or test microcontroller using 22 AWG wire

A program will be written using the AccelStepper library, similar to this example program.

Data will be recorded in the following spreadsheet:

NEMA 23 Temperatures					
	Motor:	Motor 1	Motor 2	Motor 3	Motor 4
Test Cycle:					
1					
2					
3					
4					

DM542T Temperatures					
	Driver:	Driver 1	Driver 2	Driver 3	Driver 4
Test Cycle:					
1					
2					
3					
4					

The test sequence will be be programmed and executed as follows:

Speed Testing Section:

(5 secs) Accelerate to 1125 clockwise RPM
(15 secs) Remain at 1125 clockwise RPM
(5 secs) Reverse to 1125 counterclockwise RPM
(15 secs) Remain at 1125 counterclockwise RPM
(5 secs): Decelerate to 150 counterclockwise RPM
(15 secs) Remain at 150 counterclockwise RPM
(5 secs) Accelerate to 750 clockwise RPM
(15 secs) Remain at 750 clockwise RPM

Endurance Testing Section:

(5 secs): Accelerate to 1125 clockwise RPM

(300 secs): Remain at 1125 clockwise RPM

Measurement Section:

Measure and record all NEMA23 and DM542T temperatures using IR thermometer. Record data in spreadsheet.

Repeat the above cycle four more times for a total test period of ~31 minutes and five total measurements of motor temperature.

Phase C: Actuator Velocity Testing

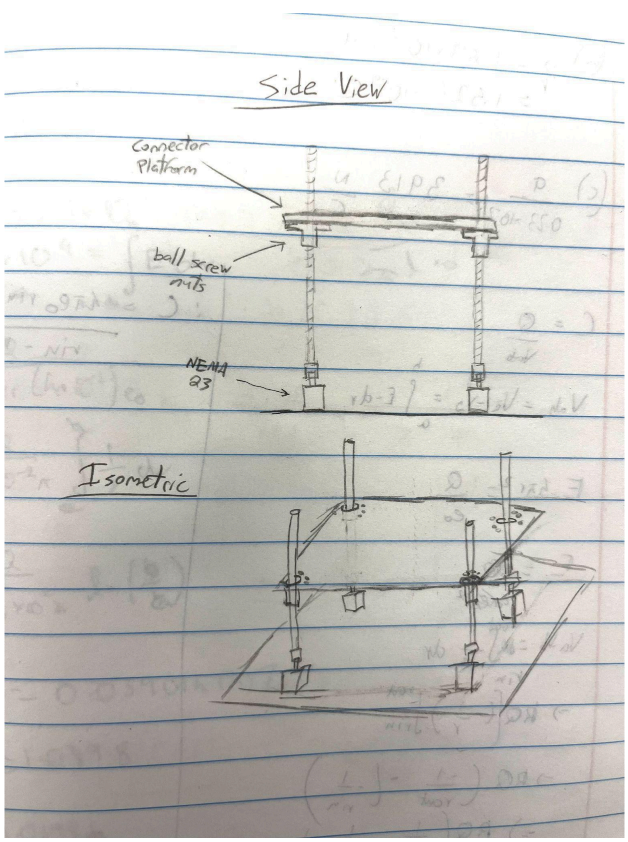
a. Test Objective

High-level Objectives:

- Establish that the linear actuators move smoothly and at the uniformly by connecting all four ball nuts to a machined wooden board

Specifically, we must test that:

- The actuators move all the nuts at the same rate when the motors are spun at the same speed by measuring the level of a platform attached to all four actuators
- The system maintains linear movement accuracy under increasing loads



b. Outcomes / Dependent Variable

Variable	Value Constraints	Reason
Platform Level	0.5 degrees max	If the actuators are not moving in unison, the platform connecting the nuts will not be level, which will cause strains on the propulsion system and will lead to wear and failure in later phases.
Platform Movement Distance	Within 1% of desired distance max	For SFU1605 shafts, one motor revolution should correspond to 5mm of linear movement. We must test that this to ensure that the TBM is thrusted at correct distances and does not overshoot/undershoot even when there is load on the actuators

c. Independent Variables

The wiring is exactly the same: nothing from the previous phase of testing has to be removed or changed.

Knob of Control	Purpose	Associated Dependent Variable
Weight added on top of connecting platform	The propulsion system must have the (relatively) same movement behavior regardless of the load applied to it (up to 75kg). It cannot overshoot, undershoot, or jerk. We must simulate increased loads by adding weights onto the platform that connects the ball nuts.	Platform Movement Distance

d. Data Gathering

Variable being Measured	Measurement Device	Reason
Platform level	Digital level	A non-perfect platform level

		indicates non-uniform movement of the actuators. Placing a digital level on top of the connecting platform will give quantifiable data about how accurate the shaft movements are.
Platform movement distance	Measuring tape & post-it notes	The entire propulsion system must not overshoot or undershoot. To measure the platform's displacement, post-it notes will mark the initial position, the system will move, and a measuring tape will measure the difference.

e. Safety

Same as Phase B.

f. Materials Required

Assembly Materials:

Quantity	Material Name	Description
4	1500mm SFU1605 & Ballnut	Shaft and corresponding ball nut for assembling the actuator
4	8mm-10mm Coupler	Connects the SFU1605 shaft to the NEMA23 motor shaft
1	2x2' MDF board (½ ")	The connecting platform for all four actuators.
1	2x2' MDF board (¼" thick)	The base board to secure the NEMA 23 motors
24	5.5mm diameter screws and nuts	Mounts the 4x ball nuts to the connecting platform

Testing Materials:

Material Name	Description
Digital Level	Placed on top of the connecting platform to measure how uniform each of the actuators move. Penn likely has one somewhere.
Quick-grip clamp	Clamps the 4x NEMA 23 motors to the testing table

g. Test Plan Steps

First, the connecting platform (MDF board) needs to be machined into a circle, with holes for the shafts, and screw holes for mounting to the ball nuts. This will happen in the following sequence:

1. Laser cut ¼" MDF with holes for the four SFU1605 shafts and ball screw nut holes

Then, the individual actuators must be assembled. This will happen in the following sequence:

1. All four of the 10mm to 8mm coupler will be attached to their respective NEMA 23 shafts. They will be secured using the builtin tightening screws
2. All four of the SFU1605 shafts will be mounted to their respective couplers and secured using the coupler's tightening screws
3. All four ball nuts will be marked with a line using a sharpie (in order to keep track of their orientation)
4. All four ball nuts will be rotated 40 revolutions downwards on the shaft. The marks will be used to keep track such that the nuts are at the exact same height

Lastly, the individual actuators must be connected together:

1. Arrange the motors in a square pattern with the outside faces being 0.45m apart
2. Slide the connecting platform onto the 4x ball nuts
3. Using 5.5mm screws and 5.5mm nuts, attach the connecting platform to the 4x ball nuts
4. Using x4 quick-grip clamps, clamp all four NEMA 23 motors to a table.

The movement part of the test is conducted by rotating the motors clockwise and counterclockwise, which will move the plate vertically up and down. The digital level is placed on the platform. The following sequence will be executed:

- Mark platform position with post-it note
  (480s) - Lower platform 1m (real dig length)
- Distance traveled is recorded using measuring tape and level value is recorded

(240s) - Raise platform 1m (simulate retraction)
- Distance traveled is recorded using measuring tape and level value is recorded
- Measure the platform position relative to the post-it note position

If all of the actual distances traveled are within 1% of the respective desired distance, and the level of the platform remains under 0.5 degrees over the entirety, then this component of the test is a pass.

Phase D: Actuator Force Testing

a. Test Objective

High Level Objectives:

- Measure the max thrust that one actuator can exert
- Ensure the motor and ball screw can hold their own weight and move in a controlled fashion

Specifically, we will be measuring the maximum thrust that each actuator can exert at various RPMs, in order to validate that the entire propulsion system is capable of thrusting the TBM during real operation.

b. Outcomes / Dependent Variable

Variable	**Value Constraints**	**Reason**
Force generated	Should reach or exceed 1125N of force	According to thrust hand calcs, the thrust force of the TBM is ~1125N per actuator (with an FOS of 2). Each actuator should meet or exceed this estimated value for reliable TBM propulsion.
(Qualitative) Observe movement of ball screw through ball nut.	Look for smoothness and ensure ball screw threads can handle the weight of the motor itself.	On the final competition setup, all 4 ball screws should be able to hold their own weight (plus the thrust ring) if needed.

c. Independent Variables

Knob of Control	**Purpose**	**Associated Dependent**

		Variable
RPM of NEMA 23 shaft	We desire to create a torque vs RPM curve. To do this, we must vary the RPM of the NEMA23 motors and measure the corresponding max force.	Force generated

d. Data Gathering

Variable being Measured	**Measurement Device**	**Reason**
Force generated	Load cell, something like this	The actuator will thrust itself into a load cell in order to determine the maximum thrust that the actuator can generate at a certain RPM.
(Qualitative) Observe movement of ball screw through ball nut.	Qualitative observation	The ball screw nut and shaft must be able to withstand the loads during operation. A qualitative assessment of the backlash, auditory response, and mechanical rigidity must be considered

e. Safety

Same as prior phases.

f. Materials Required

Quantity	**Material Name**	**Description**
1	Hardwood Board	Used to hold the ball nut in place.
6	5.5mm diameter screws and nuts	Mounts the ball nuts to the hardwood board
2	Quick-grip clamp	Clamps the wood board between two test tables. Must

		have a clamping range of at 2”.

g. **Test Plan Steps**

Setup

1. Cut the wood board into a 14 by 4 inch plank. Cut a 20mm diameter hole through the center of the block with a drill press. Use the ball screw to mark and drill 6 holes to secure the ball nut.
2. Use two quick grip clamps to secure the wood board to the edge of two tables such that there is a gap in between them.
3. Attach the coupler to the first motor’s shaft - tighten bottom bolts on the coupler. Attach the ball screw to the other end of the coupler.
4. Place the ball screw through the hole in the wood plank such that the motor is under the board, shaft facing up. Screw the ball nut in from the top such that it lines up with the holes on the wooden board. Secure the ball nut using appropriate fasteners.
5. Place the digital weighing scale directly underneath the motor on the floor.

First, the actuator must be manually adjusted such that the motor is right above the scale, without applying any pressure on it. This will be done with a program using the serial command-line on the Arduino IDE. The motor will be lowered using this program.

Then, the test must be conducted. All data will be recorded in the Phase III data recording spreadsheet. The following steps will be executed:

(2s) Motor shaft is rotated at 20 RPM such that the entire system moves downwards. Contact is made with the load cell

- The maximum force from the actuator is observed and recorded

(2s) The motor rotates in the reverse direction and the system returns to its initial position

Repeat these steps for 20 RPM, 40 RPM, 60 RPM, 80 RPM, 100 RPM, and 120 RPM for all four actuators.

Phase E: Complete System Testing

a. **Test Objective**

Validate the structural soundness of the propulsion system and the load-producing ability of each motor-support structure group.

This will be done once the support structures are built and motor control is verified. A maximum of 1125 Newtons per motor will be tested - this was the calculated force required per motor with an FOS of 2.

b. Outcomes / Dependent Variable

Variable	Value Constraints	Reason
Force generated	Independent variable (below) stepped up to test various loads up till 1250 Newtons	It is important to see if the Thrust calculations are correct and the propulsion system can produce the required level of thrust.

c. Independent Variables

This test will look to alter RPM of all 4 motors at the same time. This involves using data from Phase 2 and Phase 3 of actuator testing. This will mean the recorded RPM values for various levels of force from Phase 3 will be scaled up to estimate the expected level of thrust. We will use the minimum RPM to generate 10% of our max force and test with an increment of 10% to find an appropriate range of force to apply for the final testing with the TBM.

d. Data Gathering

Variable being Measured	Measurement Device and Reason
Force generated	The dependent variable is to be measured using an industrial weighing scale with sufficient range to correctly read the maximum of 1250 newtons of force. We expect to source this from one of the labs or loading docks on the Penn campus.
(Qualitative) Observe movement of thrust ring before and while applying force onto the scale.	Qualitative observation to look for any deformation on the thrust ring to validate stress calculations.

e. Safety

Follow all directions as per prior phases along with the following:

Exercise caution with team members placing themselves or body parts directly under the thrust ring diameter. Also exercise caution when around the propulsion structure.

f. Materials Required

Apart from the assembled structure + motor system, the following equipment is needed to test.

Quantity	Material Name	Description
1	Industrial Weighing Scale (from Penn)	To measure force
4	Steel Cable (present on BOM)	Used to tension the propulsion structures to concrete blocks behind them.
4	L - Cored Concrete Blocks (present on BOM)	Used to provide a place for the steel cable to attach to and support the I-Beam

g. Test Plan Steps

1. Use the setup at the end of the assembly work order with the weighing scale directly under the thrust ring.
2. Place the 4 concrete blocks behind each propulsion stand. Use the tie-down point on the stand to run a steel cable through the point and the center of the concrete block. Use the tightening mechanism on the clamp on the cable by adjusting a bucket screw on the clamp.. Use another clamp if needed on both ends
3. Run the motors at the various RPMs as described in the Independent Variable section above. Note down observed force values.

Bill of (Assembly) Materials:

Electronics

Category	Name	Unit Cost	Quantity	Total Cost	Lead Time
Motors	NEMA23 340 oz in	$26	4	$104	~1 week
Motor drivers	DM542T	$28	1	$28	<1 week
Ball Screw + Nut	SFU 1605	$75	4	$300	~1 week
Coupler	10mm to 8 mm	$3.6	4	$14.4	2 weeks
Power	Power	$18	1	$18	< 1 week

Distribution	Distribution Block				
Power Distribution	Signal Ground Block	$25	1	$25	< 1 week
			TOTAL COST	**$489.4**	

Structure

Category	**Name**	**Unit Cost**	**Quantity**	**Total Cost**
Main I-Beam	S 6 x 12.5 lb (6.00" x .232" x 3.332") A36/A572-50 Standard Steel I Beam **B16125, Custom size - 4.75 feet**	$178.26	4	$713.04
Side I-Beam	S 4 x 7.7 lb (4.00" x .193" x 2.663") A36/A572-50 Standard Steel I Beam **B1477, Custom Size - 6 inches**	$18.09	4	$72.36
Tension Cable	304 Stainless Steel Wire Cable	$34	1 (4 included)	$34
Fasteners	M8 Bolt 30mm length	$13.74	40 (2 packs of 25)	$27.48
Fastener Nut	M8 Nut	$9.67	40 (1 pack of 100)	$9.67
Fastener Nut	M8 Larger Nut	$11.58	40(1 pack of 50)	$11.58
Pikes + Rope	Ground Anchor with Rope Pack of 8	$28	1 (4 sets included)	$28
Sheet Metal for Clip Angles	0.25" Carbon Steel Plate A36 Hot	$9.78	8	$78.24

	Rolled custom size 6”x6”			
Sheet Metal for Connection Plates	0.25" Carbon Steel Plate A36 Hot Rolled custom size 2.75”x3”	$3.07	8	$24.56
Clip Angle	Manufactured in Nextfab Machine Shop by Rishu - Minimal Cost			
Connection Plate				
Tie-downs	Weld-On Tie-Down Rings	$12.56	12	$150.72
Threaded Rod	Threaded Rod	$16.46	3 1m long rods (16 sets to be cut out)	$49.38
Nut	M4 Nut	$3.81	1 pack of 100	$3.81
Nut	M4 Bigger Nut	$7.14	2 packs of 50	$14.28
Thrust Ring*	Aluminum 6061 Plate	$1250	1	$1250
		TOTAL COST	**$2467.12***	

Structure

Category	**Name**	**Unit Cost**	**Quantity**	**Total Cost**
Concrete Blocks	L-Core Concrete Block	$2.50	4	$10
		TOTAL COST	**$10**	

Shipping cost from MetalsDepot, Online Metals and McMaster of $130, $25 and maybe another $100 respectively

****The thrust ring is a major cost in this system that was not anticipated prior to the PDB. We are looking to source the aluminum through other methods such as using scrap aluminum.***

Mass

Per actuator components

Motor	~1.2kg
Motor Driver	~200g
Shaft + Ball Nut	~4.9kg
Mounting	5kg
Motor Coupler	~0kg
Support Structure	~5kg
TOTAL	~16.4kg

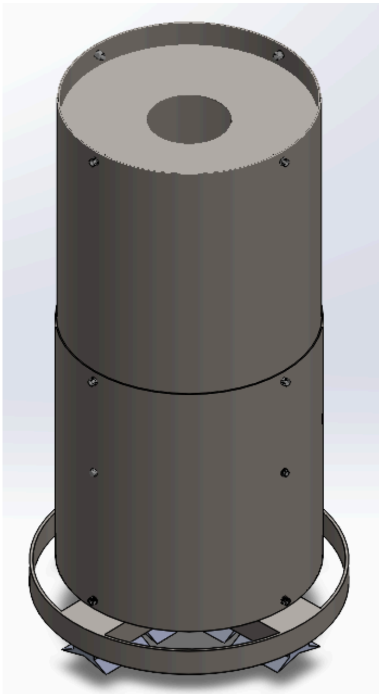
Overall mass

Actuators	~65.2 kg
Thrust ring	~83 kg
TOTAL	148 kg

Dimensions

Shafts - 1.55m length
Overall footprint - 3m x 3m

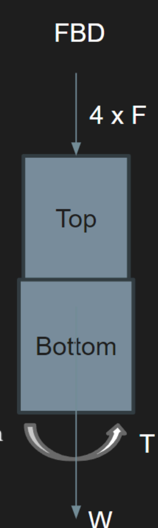
TBM Structure

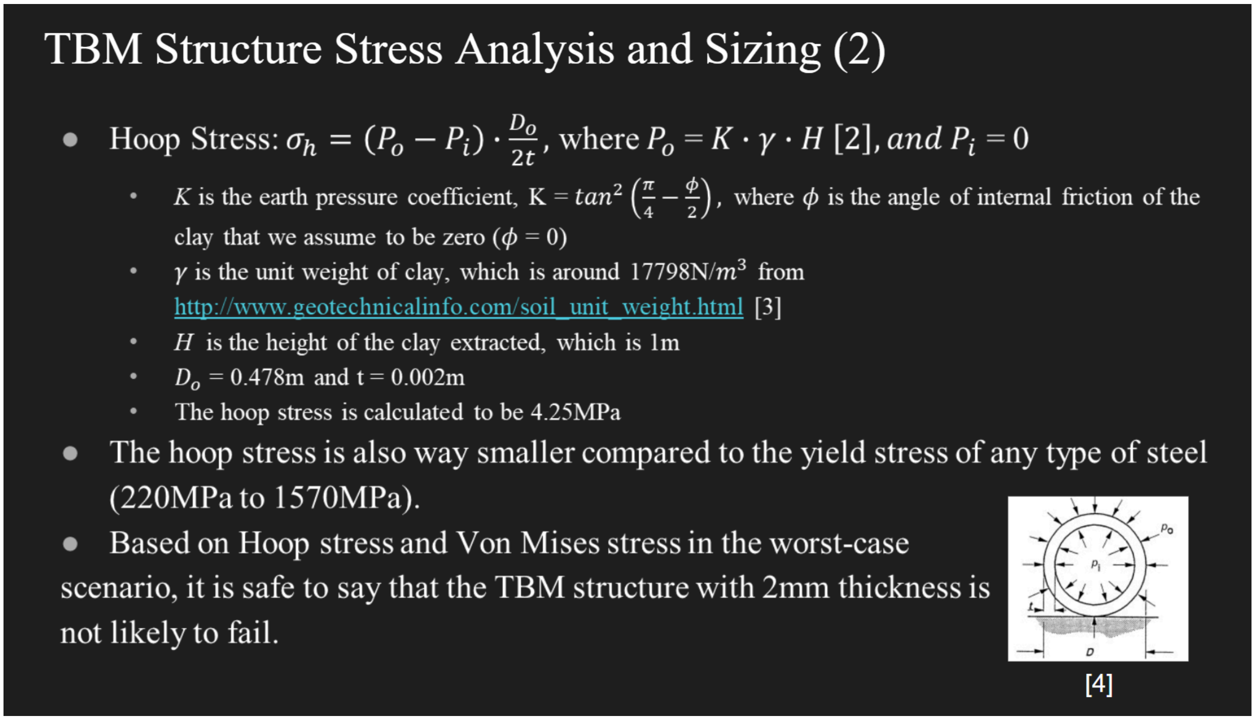


From our PDB, we had presented the following slides

TBM Structure Stress Analysis and Sizing (1)

- Worst case scenario: The motor and gearbox stop working and the TBM structure falls on the cutterhead. The TBM is now under a torque of $6KN \cdot m$ from the cutterhead.
  - Shear Stress: $\tau = \frac{T \cdot r}{J}$, where $J = \frac{\pi}{32}(D_o^4 - D_i^4)$, $T = 6KN \cdot m$
    - For top segment, $D_o = 0.474\text{m}, D_i = 0.470\text{m}, r_{max} = 0.474m$. $\tau_{max} = 17.22MPa$
    - For bottom segment, $D_o = 0.48\text{m}, D_i = 0.476\text{m}, r_{max} = 0.48m$. $\tau_{max} = 16.58\ MPa$
    - The maximum shear stress in the structure is around 17.22 MPa
  - Normal Stress: $\sigma = \frac{4 \cdot F + W}{A}$, where $\text{F} = 1247N, \text{W} = 490.5\text{N}$, and $\text{A} = \pi \cdot \frac{D_o^2}{4} = \pi \cdot \frac{0.474^2}{4} = 0.176m^2$
    - Assume the TBM structure is moving downward at a constant speed, which gives the equilibrium. The reaction force from the clay is then assumed to be equal to the forces provided by the actuators and the weight of the TBM structure.
    - The normal stress in the structure is around $31.13\text{N}/m^2$
  - Von Mises Stress: $\sigma_{VM} = \sqrt{}(0.5 \cdot ((\sigma_x - \sigma_y)^\wedge 2 + (\sigma_y - \sigma_z)^\wedge 2 + (\sigma_z - \sigma_x)^\wedge 2 + 3 \cdot \sqrt{}(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)))$
    - Von Mises Stress is calculated to be around 17.22 MPa
    - It is way smaller than the yield stress of any type of steel (220MPa to 1570MPa) [1]. The reason may be that the forces provided by the actuators and the torque exerted on the structure is are really small in our case.





with the following manufacturing and integration details:

- 2mm thick steel; 1.3m total height split between two steel barrels, 0.5m ID
- Optimize for ease of integration and testing by eliminating welds where possible
- Split TBM into separate barrels for integration and testing ease
- For the top and bottom cylinders, we plan to purchase steel sheets from *COREMARK METALS* and use a Slip Roll at NextFab to form the steel ourselves
- For the top and bottom lids, we plan to buy steel plates from *COREMARK METALS* and use a CNC milling machine ourselves for material removal.
- For the bolts/screws, set screws, nuts, and washers, we will order them from McMaster-Carr.
- Each "lid" has a lip that allows us to clamp the lid to the TBM structure
- Top Lid's hole (152.4mm OD) has room for two vacuum hoses and harnessing

Although the loads for the TBM structure were relatively low in this Mini-Event digging format, especially with our cutterhead being larger than our TBM structure in diameter which eliminated any lateral Earth pressure acting on the TBM, we decided to perform some basic FEA for the Final Design Briefing

We continued investigating our TBM structure and making sure it will not fail during operations by moving from the preliminary analysis with hand calculations to finite element analysis (FEA) which could show us more information and details about the deformations and stresses of the structure. Also, we could compare the results from FEA to the ones from hand calculations and see the differences between the two. From hand calculation using *Eq.1*, *Eq.2*, and *Eq.3* based on the *Figure.1 TBM Free Body Diagram (FBD)*, we determined the equivalent stress to be 17.22MPa, from which we drew the conclusion that the TBM was not likely to fail since the stress was way smaller than the yield stress of any type of steel (220MPa to 1570MPa).

$$\text{Eq.1: Shear Stress: } \tau = \frac{T \cdot r}{J}$$

$$\text{Eq.2: Normal Stress: } \sigma = \frac{4 \cdot F + W}{A}$$

$$\text{Eq.3: Von Mises Equivalent Stress: } \sigma_{VM} = \sqrt{}(0.5 \cdot ((\sigma_x - \sigma_y)^{\wedge}2 + (\sigma_y - \sigma_z)^{\wedge}2 + (\sigma_z - \sigma_x)^{\wedge}2 + 3 \cdot \sqrt{}(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)))$$

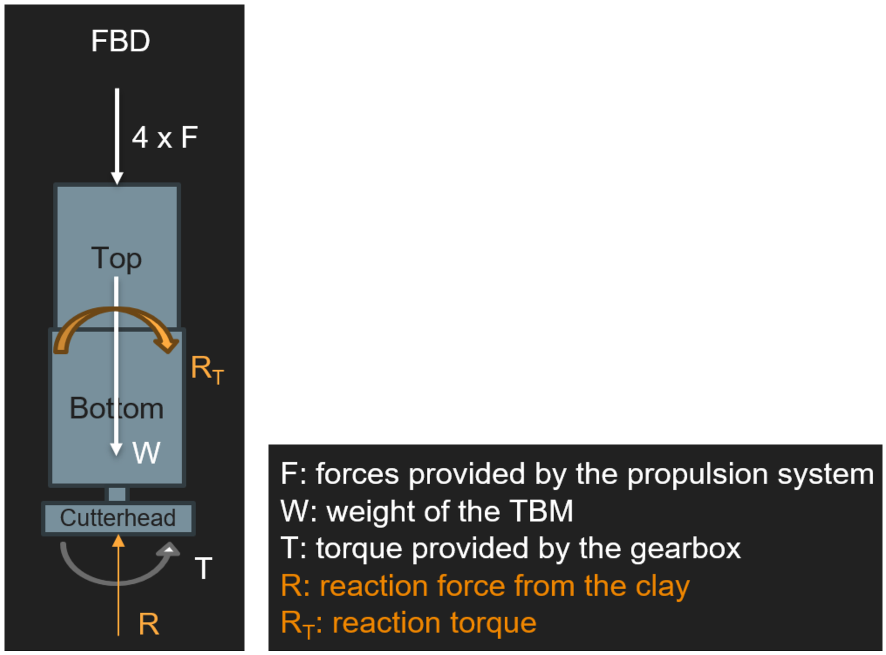
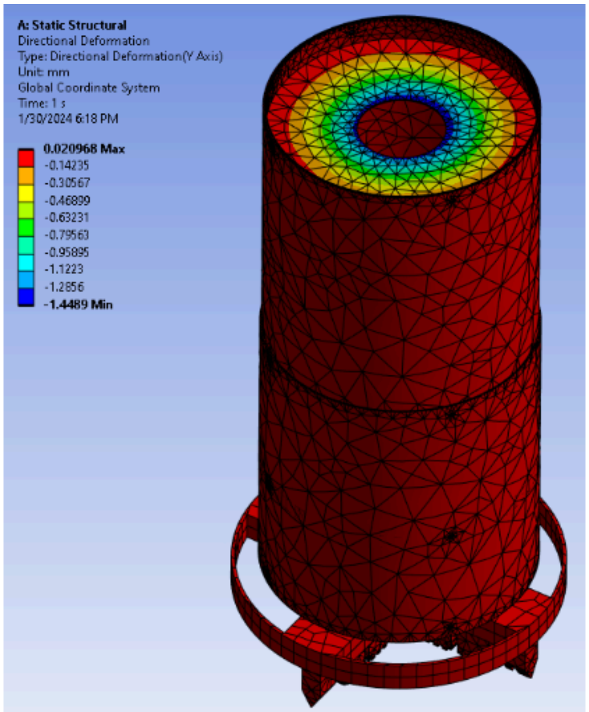


Figure.1 TBM Free Body Diagram (FBD)

In this FBD, the forces and torque exerted on the TBM were labeled in white while the reaction forces and torque were highlighted in orange.

Then we used ANSYS Workbench to perform FEA with the forces and moment inserted for solutions based on the FBD. Structural steel is the material assigned to the TBM structure, and aluminum is assigned to the cutterhead plate, which is a “cross” shown in *Figure.6*. We have designed new cutters for our cutterhead model but did not use them in FEA since the new cutters’ model will not necessarily change the FEA results. Nonetheless, we included our updated TBM model with new cutters in this report.

The deformation and equivalent stresses are shown in *Figure.2 TBM Vertical Deformation* and *Figure.3 TBM Equivalent Stress*. From *Figure.2*, we knew that the maximum deformation of the structure was at the top around the opening hole. The deformation was about 1.4489mm, which was small and acceptable for the operation. From *Figure.3*, we observed that the maximum stress in the structure was about 73.775MPa, which is much greater than the result from hand calculations. However, since the yield stress of typical steels are from 220MPa to 1570MPa, our structure will be safe.



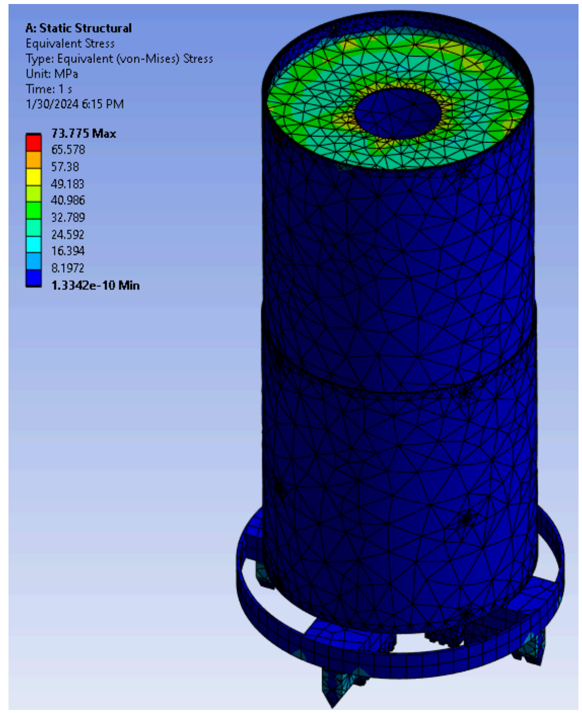


Figure 2: TBM Vertical Deformation

Figure 3: TBM Equivalent Stress

In addition to the whole TBM structure analysis, we closely examined the worst-case scenario that we mentioned in PDB, which is that the vertical deformation/displacement of the cutterhead is too large, causing the cutterhead to contact the TBM structure. To know if it would be the case, we once again performed FEA based on *Figure.4* which presents the FBD of the cutterhead which was connected to the gearbox through the output shaft.

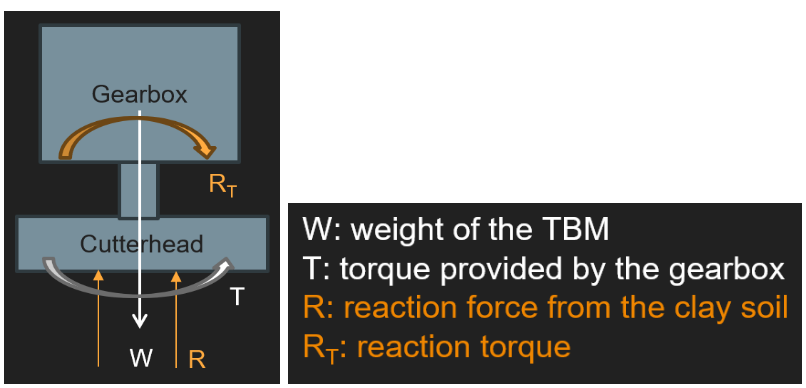


Figure.4: Cutterhead, Shaft, & Gearbox FBD

We assumed that the TBM would be moving downward at a constant speed during the operation so that the cutterhead will be subject to a force of 4*1247 N + 490 N = 5478 N which is the reaction force from the clay soil and is equal to the force provided by the actuators and the weight of the TBM structure. The reaction moment was equal to 6000 Nm which the motor and gearbox provided. After conducting FEA overall TBM structure, we had more experience in shortening the time for finding solutions in ANSYS and making the simulation process more efficient by simplifying the motor, gearbox, and cutter models. The results are shown in *Figure.5: Deformation of Cutterhead* and *Figure.7: Equivalent Stress of Cutterhead*.

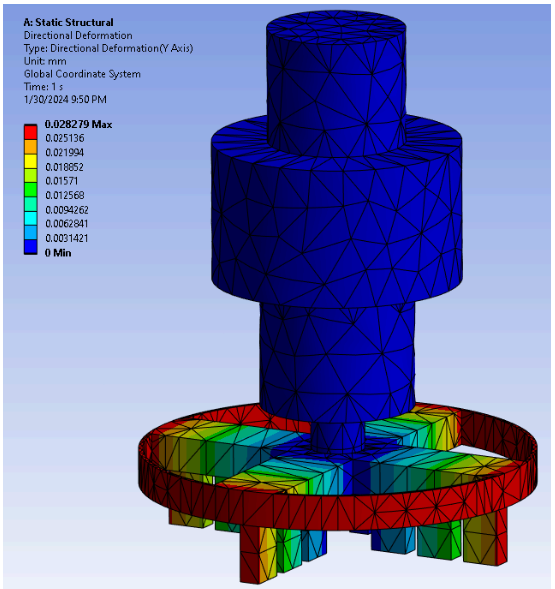


Figure.5: Deformation of Cutterhead

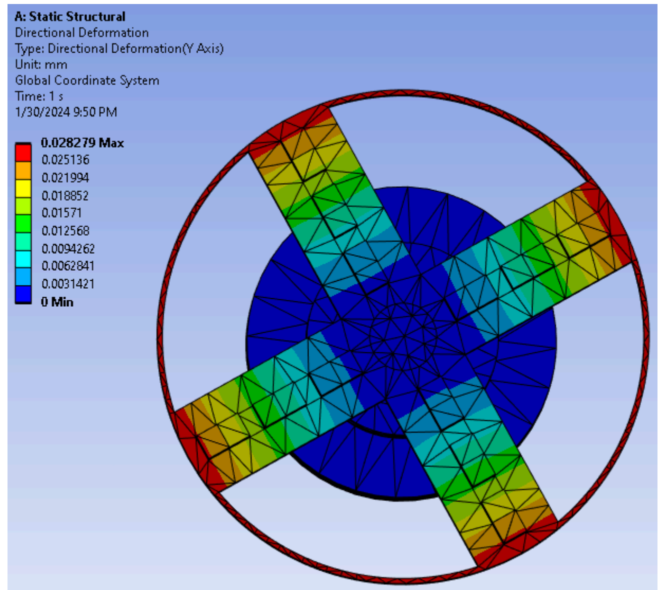


Figure.6: Bottom View of Deformation

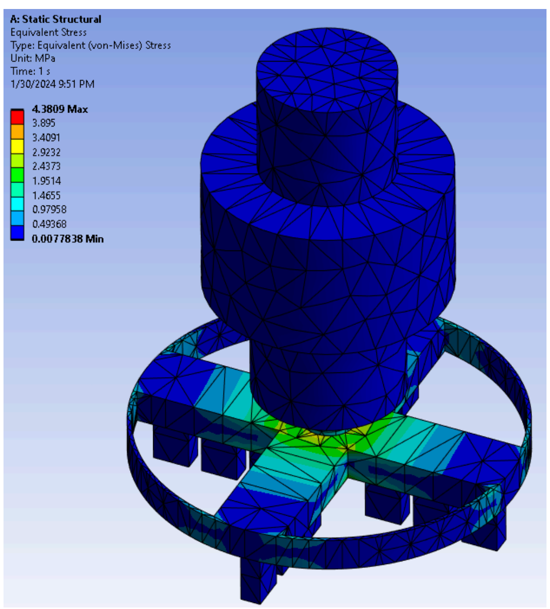


Figure.7: Equivalent Stress of Cutterhead

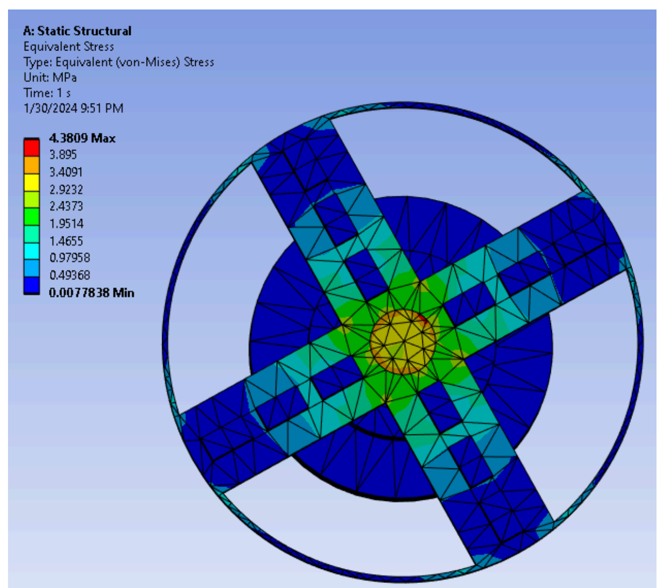
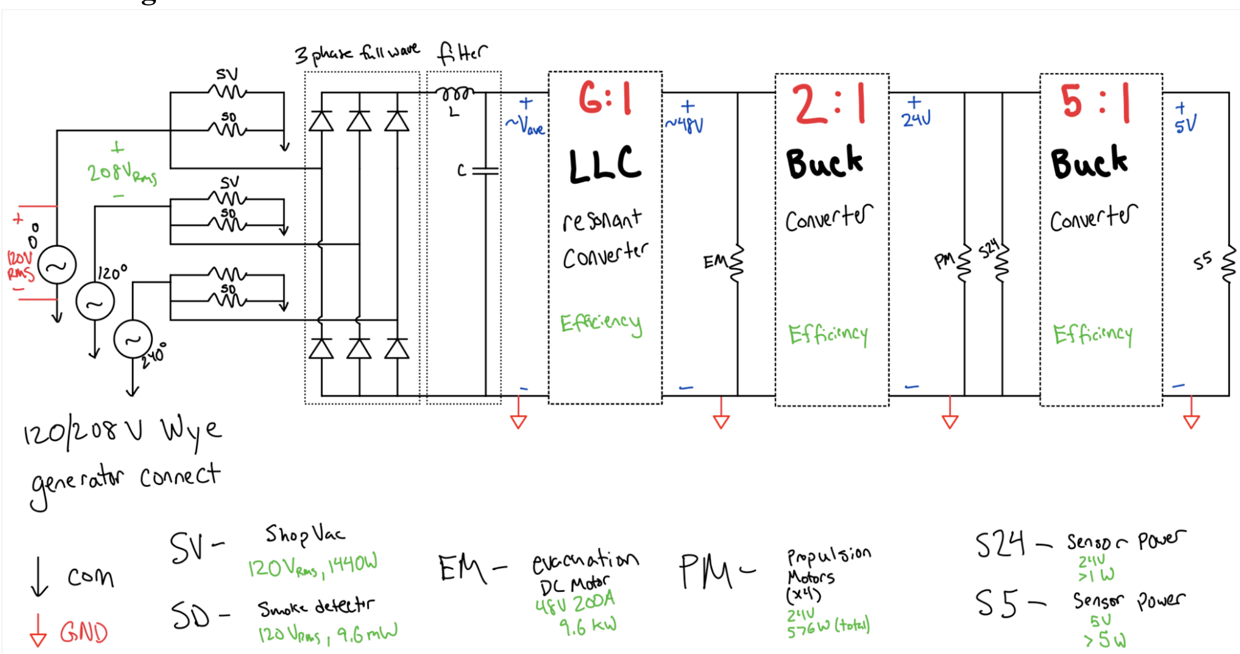


Figure.8: Bottom View of Equivalent Stress

From *Figure.5*, we observed that the vertical upward deformation of the cutterhead was around 0.028mm which was the maximum deformation concentrated in the outer wall of the cutterhead, which made sense since it was the thinnest part of this entire structure and deformed the most. However, 0.028mm was a really small number and could be neglected. Also, the maximum equivalent stress in this structure was only 4.38MPa which is way below the yield stress of either steel or aluminum. Therefore, it is safe to say that the worst-case scenario is not going to happen, and the cutterhead will function well without touching the TBM structure.

Power System, Safety Interlocks, Telemetry, Sensors, Failure Modes, and Hazards

Power Diagram



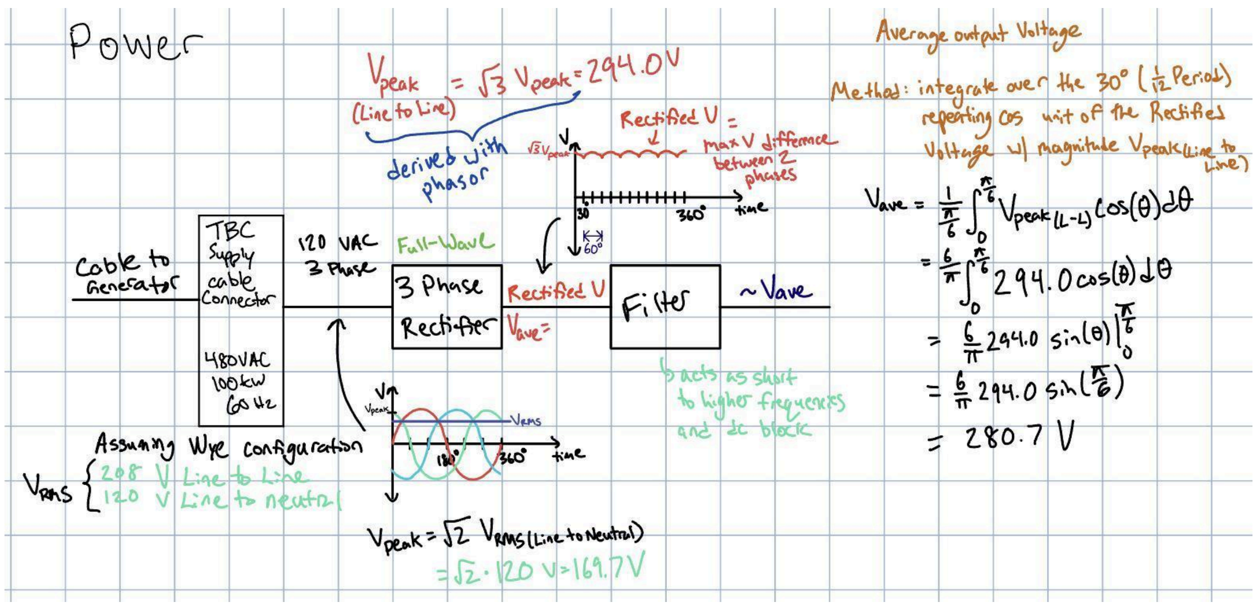
The current system design has been reviewed in a conversation with Professor Lei Gu, who teaches Power electronics at the University of Pennsylvania.

We plan on using the generator in the 120/208 three phase output setting (based on the datasheet about last year's provided generator) so that we can power components that require 120VAC single phase as balanced split loads rather than needing to include a DC-AC pure sine wave inverter after our 3 phase rectification and downconverting. This allows us to decrease the power requirement in these sections, making the implementation cheaper and more efficient. The split loads are of the same devices (3 identical shop vacuums and 3 identical smoke detectors, each split one per line), which will ensure that the power remains balanced between the phases. This is an improvement over one of our old design options

We recently modified our power design with the choice of using a resonant converter rather than a step down transformer or other dc-dc step down options as designing a resonant converter is more financially viable and easily customizable to our requirements than other commercial options. We confirmed this design choice with Prof Gu as well.

Our physical implementation plan includes mounting the rectifier, filter, and LLC components all on heatsink(s) and using insulated wire connections of reasonable gauge for each section of the power distribution. We plan on using commercial buck converter modules to increase safety and efficiency, as well as to decrease our design implementation complexity. If we can find a commercial resonant converter or 3 phase rectifier meeting our requirements, we will also consider purchasing those depending on cost implications on our budget (previous searching for high power commercial modules has found that most do not meet our particular requirements and are very expensive).

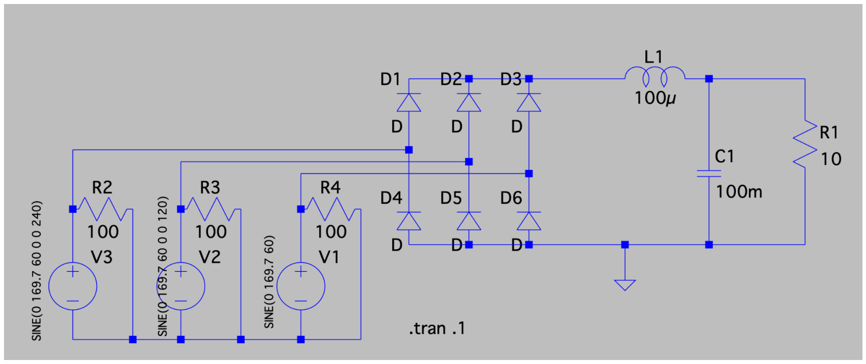
Calculations:

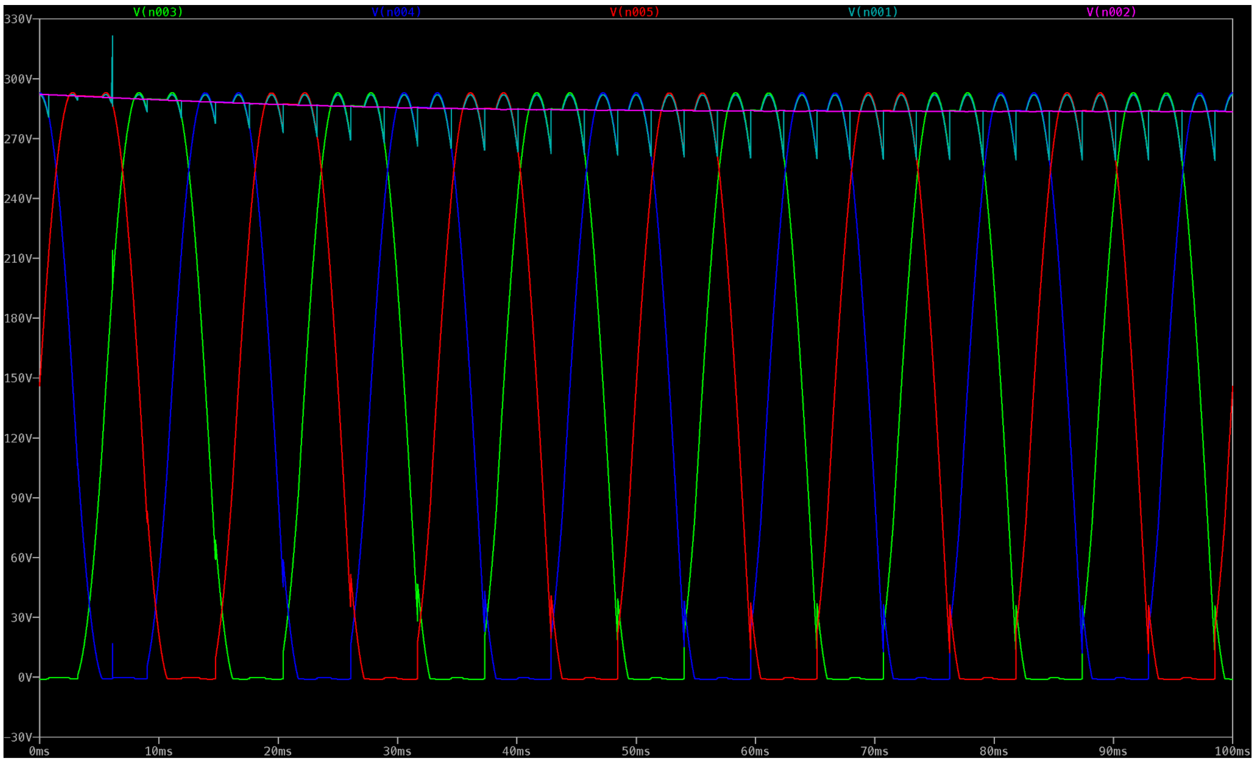


As we source buck converter modules and design our LLC resonant converter, another important calculation is considering our power losses in each section of the power distribution system. We plan on doing this by back calculating through the system, starting with the power consumption at the 5V loads, calculating the power losses of the 5:1 buck converter, then finding the total power after the 2:1 buck, calculating the loss through that buck, etc. all the way until the beginning of the system to find our total expected power draw with losses. We will also use this information in determining the necessary input power rating of each converter, as well as ratings for components in the rectifier, filter, and LLC converter.

LTSPICE initial simulations:

In addition to calculations, we are also using LTSPICE to simulate our circuits during the design and implementation phases. For now, the below simplified simulation has been used to confirm the planning math done for the rectification and filtering components of the system.

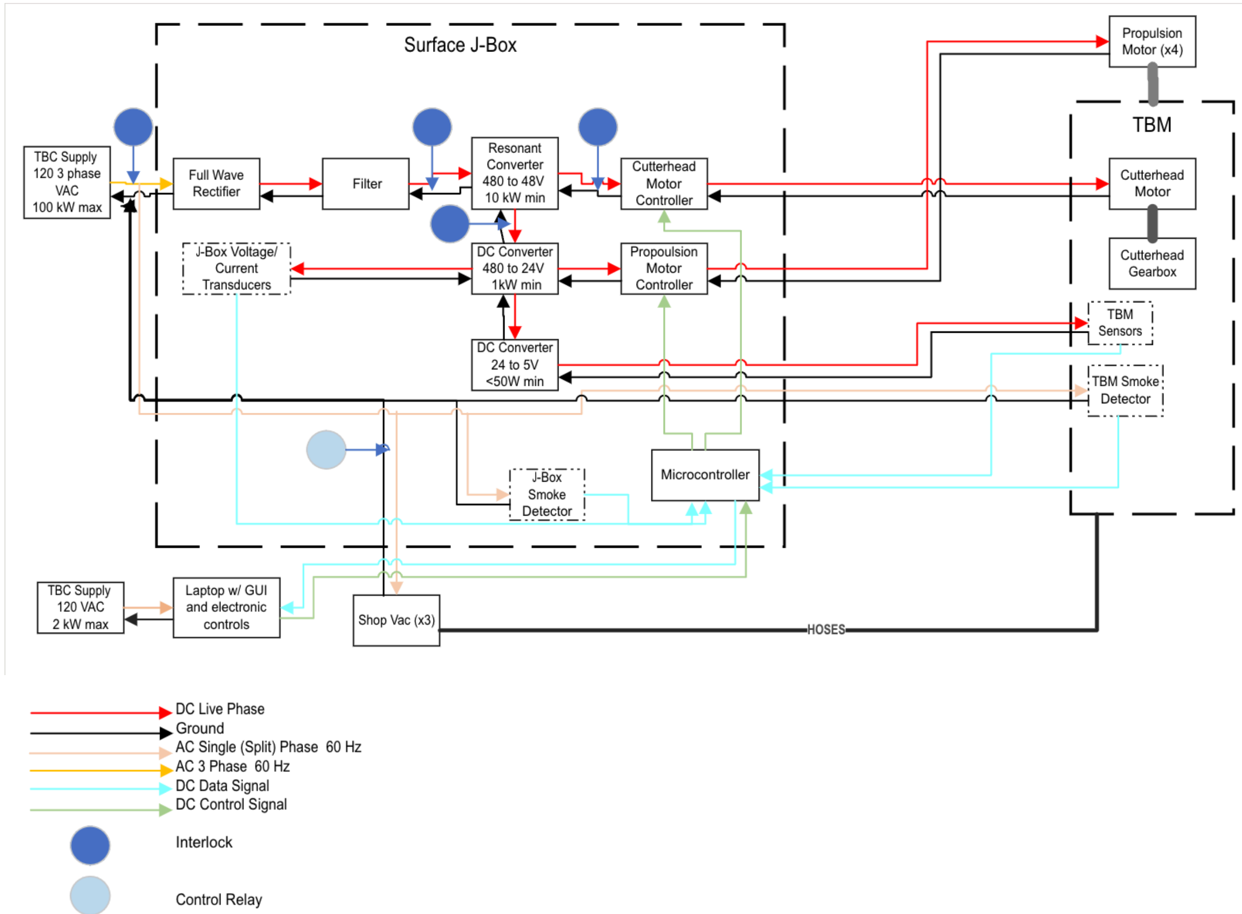




V(n003) is the voltage at the positive terminal of source V3
V(n004) is the voltage at the positive terminal of source V2
V(n005) is the voltage at the positive terminal of source V1
V(n001) is the rectified voltage before the LC filter
V(n002) is the rectified and filtered voltage at the positive terminal of the load resistor R1

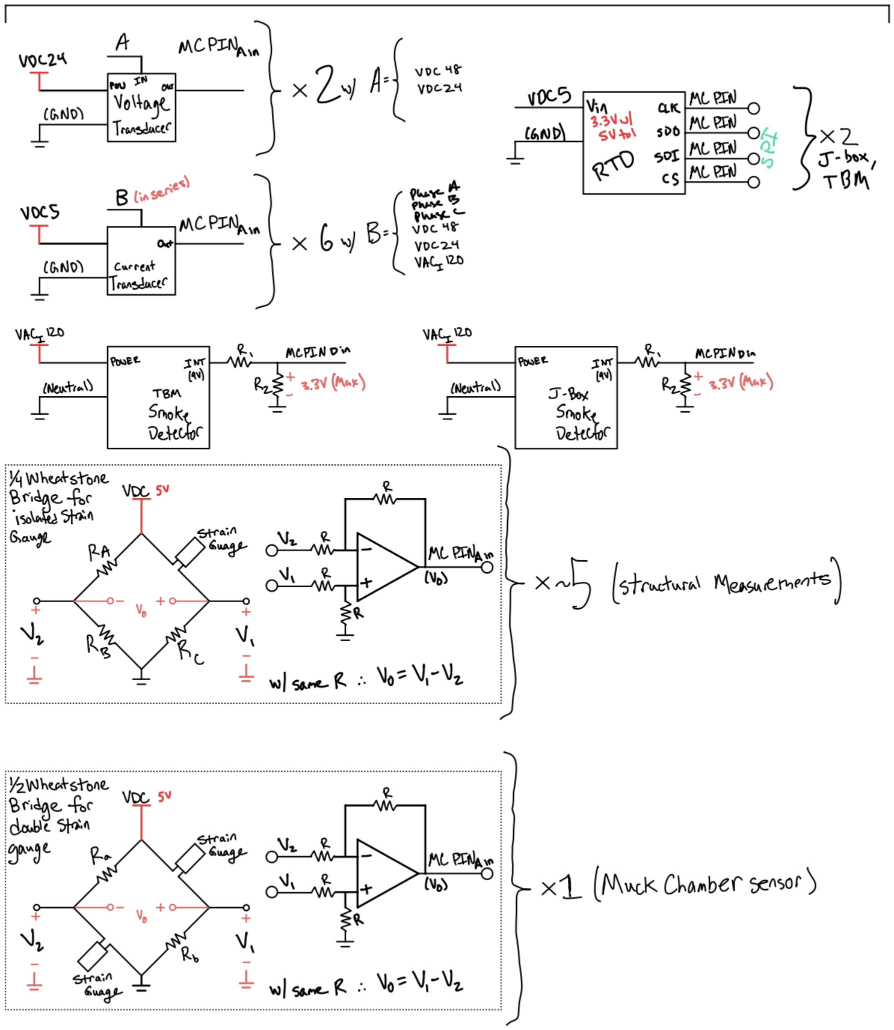
For this simulation, the diodes are ideal, the L, C, and R values are arbitrary for the sake of testing a simplified version of our intended topology, but the sources are accurate to represent the 3 phase 120/208 Volt Wye configuration we intend to use from the generator. From the results, we can see that the simulated rectified+filtered voltage is within a few Volts of the calculated 280.7 Volts.

Extended Electronics Diagrams

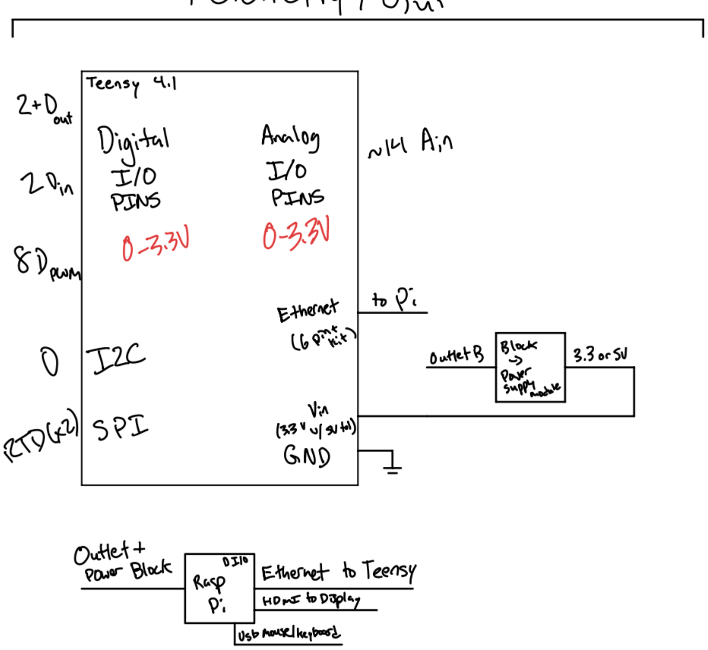


See the next page for a more detailed draft schematic for the sensors, telemetry/GUI, and actuator controls electronics (see associated FDB sections for related details).

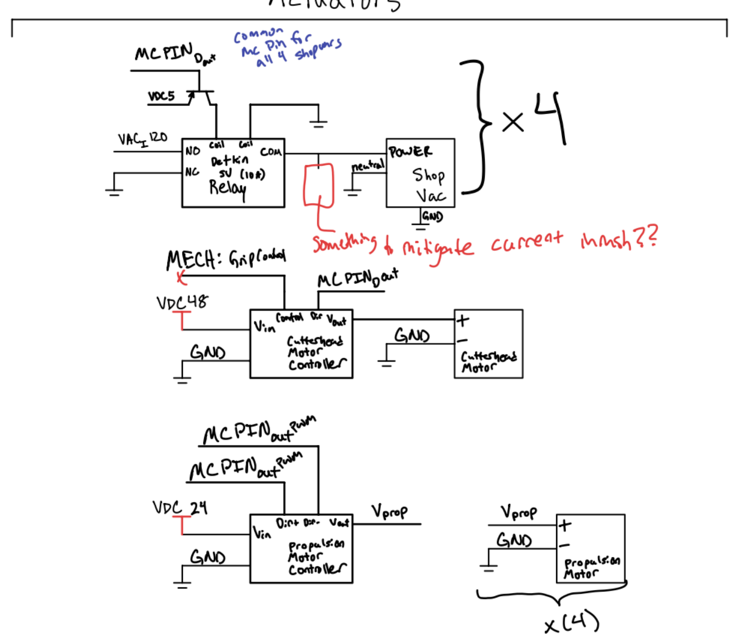
Sensors



Telemetry / Gui



Actuators



POWER BOM

Product	Quantity	Source	Price	Budget	Lead Time
Inductors and Capacitors	Determined by LLC design + 1 each for filter	We should be able to mostly use inductors and capacitors from our EE lab.		$50	
High Voltage Diodes	6	Fast Recovery Standard Recovery Both of the above options are available through Mouser	$20 -$40	$120 - $240	< 1 week
Buck Converter Modules	2	We will be using fairly low power buck converter modules. If we cannot find modules within our budget, we will use electronics available in our EE lab and design a PCB to be printed for free by local sponsor Flash PCB.		~ $60	

GUI Overview:

Our GUI both displays sensor values and indicates whether the value is within the expected range by the color of the box containing the value. Green is within expectations, Yellow is small error, and Red is beyond critical limits. Our GUI is divided into sections:

1. Sensor Data Display
   a. Voltage and Current Monitor
   b. Muck Removal Performance
   c. Motor Shaft Torque, Toxic Gas, Strain Gauge, RTD
2. Safety Interlocks Status
3. Propulsion Controls
4. Motor & Shop Vac Control

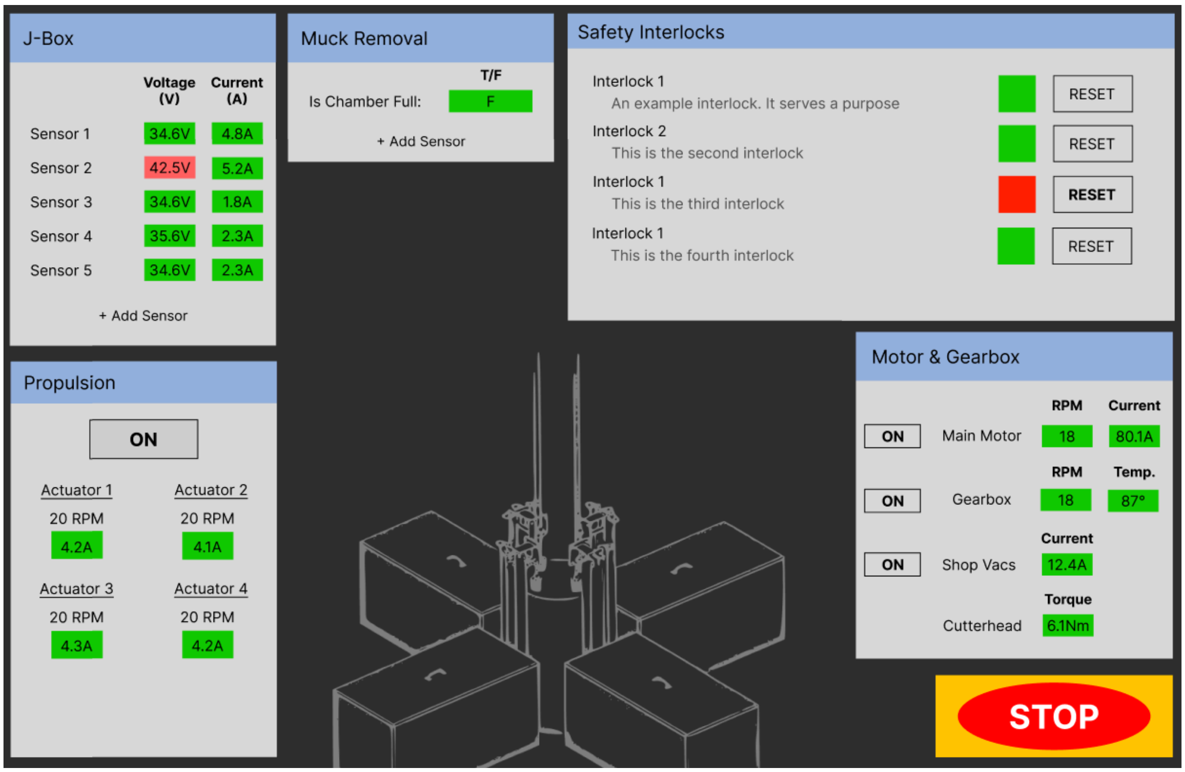


Figure 1: GUI diagram of telemetry and sensors.

Microcontroller

We decided to use a Teensy 4.1 microcontroller with an ARM Cortex-M7 processor and Ethernet chip. The Teensy has many io pins and can communicate to all of our sensors over serial communication protocols. With diligent component selection we were able to eliminate the need to use a 4-20 mA signal DAQ, which both saves money and decreases our system complexity. The Teensy can be coded via the Arduino IDE, with which all of our electrical engineering team members are familiar. Additionally, we can use the ethernet chip for communication over an ethernet cable with a Raspberry Pi running our GUI. Data for the display will be sent to the Pi and electronic control inputs selected through the GUI will be sent to the Teensey.

Product	Quantity	Vendor / Part Number	Price ($)	Total ($)	Lead Time
Teensy 4.1 MCU (with Ethernet chip and ARM Cortex-M7)	1	Pjrc.com / TEENSY41	31.50	31.50	5-7 days
Ethernet Kit for Teensy 4.1 MCU	1	pjrc.com / ETHERNET_KIT	3.90	3.90	5-7 days

Total: $35.40

Communication:

- 10Gbps Ethernet using Cat-6A twisted pair cable between Teensey microcontroller and Raspberry Pi (GUI device)
- Propagation delay at 6m: 30.8 ns
- Distance between ground station and deepest component in TBM will be ~6m
- Harnessing between sensors and electronics in TBM and surface J-Box electronics
    - The selected RTDs communicate over SPI, which can communicate of harnessing up to 10 m long without a repeater module

Sensors Bill of Materials (BOM) - *Changes from PDB to FDP:*

Sensor	Location	PDB Quantity	FDP Quantity
Voltage Transducer	J-Box circuits	3	2
Current Transducer	J-Box circuits	5	6
Load Cell	Muck chamber	2	0
Strain Gauge	Throughout TBM, checking for mechanical failures	3	1
RTD	Near Gearbox, In J-box	2	2
Smoke Detector	TBM, J-box	2	2

Current Sensors BOM:

Product	Quantity	Vendor/Part Number	Price	Total	Lead Time

			($)	($)	
RTD Sensor	2	Adafruit / PT100	11.95	23.90	3 business days
RTD Sensor Board	2	Adafruit / MAX31865	14.95	29.90	3 business days
Voltage Transducer (0-50VDC)*	2	ato.com / ATO-WBV151S01	169.96	339.92	6-12 days
Current Transducer (400A DC for Main Gearbox Motor)	1	Digikey / L37S400S05M	18.76	18.76	1-5 days
Current Transducer (24A DC for 4 Propulsion Motors)	1	Digikey / ACS713ELCTR-30A-T	4.58	4.58	1-5 days
Current Transducer (32/40A AC for Shop Vacuums)	1	Digikey / F02P050S05L	16.73	16.73	1-5 days
Current Transducer (600A AC for each phase)	3	Digikey / L01Z600S05	18.93	56.79	1-5 days
Strain Gauge (pack of 10)	1	McMaster-Carr / 7245n24	147.43	147.73	1-2 days
Smoke Detector	2	Home Depot / i12040	16.97	33.94	2 days

*Due to the relatively expensive price of this Voltage Transducer (buying 2 Voltage Transducers costs more than 50% of the total Sensor BOM cost), we will consider finding a more affordable substitute after submitting this FDP.

Total: $671.95

Safety Features

1. Real-time sensor monitoring to determine the safety state of the TBM

2. Safety Interlocks for automatic shutdown when critical parameter limits are exceeded
   - Refer to the Power section (Extended Electronics Diagrams) for locations
   - Isolate high power electronics and ensure dissipation time for energy storage components
   - Two stage restart
     - Parameter returns within normal limits
     - Physical rest button pressed
3. E-Stops for manual backup system shutdown in case of emergency
   - Minimal dissipation time
4. Electronic control through GUI on laptop

Failure Modes (Top 6):

Failure Mode	**Single Point of Failure**	**Result**
Rectifier failure (480 VDC)	Yes	Full system shutdown. Since our entire system depends on converting 480 VAC to 480 VDC power, this failure mode would cause our entire system to lose functionality.
Filter circuit failure (480 VAC to 480 VDC)	Yes	Full system shutdown. The filter circuit filters 480 VDC power from our rectifier into stable DC output. Thus, this failure mode would lead to unstable or noisy DC output, causing potential damage to *all* TBM components, so it would warrant a full system shutdown.
Buck converter failure (480 to 24 VDC)	Yes	Full system shutdown. The propulsion motor controller and sensors require proper functionality of this buck converter. In case of a failure, both the sensor and propulsion subsystems would also fail. Because proper TBM functionality requires proper propulsion motor controller functionality, and because sensors are critical for safe TBM operation, this failure mode would lead to a full system shutdown.
Buck converter failure (24 to 5 VDC)	Yes	Full system shutdown. Sensors, which receive power from this buck converter, will be isolated if it fails.

		Again, because sensors are critical for safe TBM operation, this failure mode would lead to a full system shutdown.
Motor/motor controller failure	No	Shut down and isolate motors and motor controllers. Failure modes in this category include voltage jumps, inrush current, temperature spikes in electronics or at the motor shaft, and mechanical motor failures.
Inverter failure (12 VDC to 120 VAC)	No	Isolate shop vacuums, which depend on this inverter for power.

Hazards, Risks, and Mitigation Mechanisms:

Hazard/Risk	**Hazard/Risk Type**	**Mitigation Mechanism(s)**
High Power Electronics	Complete loss of machine power	E-stops, Automatic shutdown, Sensors for current and voltage Monitoring, Electronic control through laptop GUI, Manual controls, RTD for J-Box electronics
Fire	Leaks/environmental contamination	Smoke Detector and fire extinguisher
Main Motor Temperature increase	Mechanical	We don’t need a cooling system as of now, but will gauge the necessity of adding one throughout testing
Main Shaft Failure, Cutterhead & TBM collision	Mechanical	Shear stress (limit torsional load) on shaft is less than the yield of the shaft
TBM cutting is uneven, TBM tilting occurs	Mechanical	Center of gravity is along main neutral axis of rotation which allows nominal digging to be straight Will add guide rails if deemed

		necessary during testing
Lateral earth pressure on TBM structure from soil falling on TBM even with cutterhead OD > TBM structure OD	Mechanical	We are using 2 mm thick steel, which is sufficient to deal with any lateral earth pressure
Welding main shaft to cutterhead	Mechanical	We are currently looking into a better mounting method to avoid welding. Strong welding would work, but limits our ability to rapidly test and iterate
Soil cavity size may be too small (height of ~1.85 inches)	Mechanical	During muck removal testing we will update our design to achieve a sufficient cavity

Project Schedule

Week	High-Level Actions + Test Campaigns
02/04 - 02/10	Muck Removal Test, Main Drive System Motor Calibration, Gearbox + Motor Integration, Power Systems full designed (ready to order all parts including those for testing all subsystems), Order Sensors, Update Propulsion System and order parts
02/11 - 02/17	Cutter Manufacturing, TBM Structure Manufacturing, Update Mounting Strategies in TBM for gearbox as necessary per CG Experiment, Power Systems Build and Testing, Sensor Calibration, Propulsion System Testing and Integration
02/18 - 02/24	Sensor Calibration, Write controls code, develop telemetry GUI, Integrate Propulsion System with TBM and controls
02/25 - 03/02	TEST DIG, coordinate logistics
03/03 - 03/09	Update as needed per test dig
03/10 - 03/16	Update as needed per test dig
03/17 - 03/23	Update as needed per test dig
03/24 - 03/30	COMPETITION, coordinate logistics

All the machining and manufacturing is done by Rishu using the manual mill, lathe, CNC mill, and slip roller (for the TBM structure steel barrel forming) equipment at NextFab in South Philly. Engineering Drawings and proper GD&T will be enforced.

Machine Cost Breakdown and Funding Plan

Overview:

Subsystem	**Subsystem Cost**
Drive System & Cutterhead	$4785.16
Muck Removal	$428.29
Propulsion	$2966.52
Structure	$174.61
Power, Electronics, Sensors, Interlocks	$672.25
TOTAL MACHINE COST:	**$9026.83**

Drive System & Cutterhead

Description	**Link to Product**	**Unit Cost**
100:1 In-Line Helical Gear Reducer with a NEMA145TC Flange Face Mount	100:1 In-Line Helical Gear Reducer with a NEMA145TC Flange Face Mount	$1,038.31
Conversion Kit - Motenergy ME1602 Motor (DC Brush Motor with NEMA143C Face Mount), Alltrax SR48400 Controller, Contactor & Accessories (48V, Domino Throttle, Cooling Fan, DC Reverse Contactor)	https://evdrives.com/conversion-kit-motenergy-me1602-motor-alltrax-sr48400-controller-contactor-accessories/	$1517
High-Torque Set Screw Flexible Shaft Coupling	https://www.mcmaster.com/3565N1-3565N14/	$63.2
Clamping Two-Piece Shaft Collar	https://www.mcmaster.com/3357K26/	$26.78
Clamping Shaft Coupling Steel, for 7/8" x 7/8" Diameter Keyed Shaft	https://www.mcmaster.com/61005K155/	$82.14

High-Angular-Misalignment Flexible Shaft Coupling Hub, 1-3/4" Overall Length, 2-3/16" OD	https://www.mcmaster.com/60635K9-60635K91/	$49.6
6000 rpm Acetal Plastic Disc for 2-3/16" OD High-Angular-Misalignment Flexible Shaft Coupling	https://www.mcmaster.com/60635K99/	$36.44
NEMA 143T C-Face .125-inch Weldable Steel Slotted Motor Mounting Plate	https://www.electricmotorsport.com/125-inch-weldable-cold-rolled-steel-slotted-motor-plate-nema-c-face.html	$60
Easy-to-Machine 1215 Carbon Steel Bar, 2" Thick, 2" Wide, 3 Feet Long	https://www.mcmaster.com/catalog/129/4198/4416T55	$167.2
Multipurpose 6061 Aluminum 4" Thick x 6" Wide, 3 Feet Long	https://www.mcmaster.com/catalog/129/4162/8975K293	$759.64
APPROVED VENDOR Sealed Lead Acid Battery: 12V DC, 100 Ah Capacity, 8.46 in Ht, 12.95 in Wd, ABS	https://www.grainger.com/product/APPROVED-VENDOR-Sealed-Lead-Acid-Battery-12V-2UKL8	$486.85
Rishu Mohanka NextFab Membership for Machining (First Month Cost - Student)		$319
Rishu Mohanka NextFab Membership for Machining (Student Monthly Fee)		$179
	TOTAL:	**$4785.16**

Muck Removal

Product	**Qty**	**Cost (before tax)**	**Total**	**Link**
16 Gallon 5.0 Peak HP NXT Wet/Dry Shop Vacuum with Filter, Locking Hose and Accessories	1	119	119	Home Depot
16 Gal. Poly Cart-Design Wet/Dry Vac with a Cartridge Filter, Hose, and Accessories\	1	129	129	Home Depot
13 ft RIDGID 2.5in extention hose	1	24.97	24.97	Home Depot
Wet Debris Application Foam Wet/Dry Vac Cartridge Filter for 5 Gallon and Larger RIDGID Shop Vacuums (1-Pack)	1	24.97	24.97	Home Depot
2 gallon ultisol red clay soil	2	29.95	59.9	Ebay
2-cup and 1-cup measuring cups	1	15.02	15.02	Amazon
5 gallon bucket	1	4.48	4.48	Home Depot
16 Gal Shop vac	3			

Extension hoses	1	24.97	24.97	
Shop Vacuum Y adapter	2	12.99	25.98	Amazon
Wet application filters				
Dust Bag?				
			TOTAL:	**$428.29**

Propulsion

Category	**Name**	**Unit Cost**	**Quantity**	**Total Cost**	**Lead Time**
Motors	NEMA23 340 oz in	$26	4	$104	~1 week
Motor drivers	DM542T	$28	1	$28	<1 week
Ball Screw + Nut	SFU 1605	$75	4	$300	~1 week
Coupler	10mm to 8 mm	$3.6	4	$14.4	`2 weeks
Power Distribution	Power Distribution Block	$18	1	$18	< 1 week
Power Distribution	Signal Ground Block	$25	1	$25	< 1 week
				TOTAL	**$489.4**

Category	**Name**	**Unit Cost**	**Quantity**	**Total Cost**
Main I-Beam	S 6 x 12.5 lb (6.00" x .232" x 3.332") A36/A572-50 Standard Steel I Beam **B16125, Custom size - 4.75 feet**	$178.26	4	$713.04
Side I-Beam	S 4 x 7.7 lb (4.00" x .193" x 2.663") A36/A572-50 Standard Steel I	$18.09	4	$72.36

	Beam **B1477, Custom Size - 6 inches**			
Tension Cable	304 Stainless Steel Wire Cable	$34	1 (4 included)	$34
Fasteners	M8 Bolt 30mm length	$13.74	40 (2 packs of 25)	$27.48
Fastener Nut	M8 Nut	$9.67	40 (1 pack of 100)	$9.67
Fastener Nut	M8 Larger Nut	$11.58	40(1 pack of 50)	$11.58
Pikes + Rope	Ground Anchor with Rope Pack of 8	$28	1 (4 sets included)	$28
Sheet Metal for Clip Angles	0.25" Carbon Steel Plate A36 Hot Rolled custom size 6”x6”	$9.78	8	$78.24
Sheet Metal for Connection Plates	0.25" Carbon Steel Plate A36 Hot Rolled custom size 2.75”x3”	$3.07	8	$24.56
Clip Angle	Manufactured in Nextfab Machine Shop by Rishu - Minimal Cost			
Connection Plate				
Tie-downs	Weld-On Tie-Down Rings	$12.56	12	$150.72
Threaded Rod	Threaded Rod	$16.46	3 1m long rods (16 sets to be cut out)	$49.38
Nut	M4 Nut	$3.81	1 pack of 100	$3.81
Nut	M4 Bigger Nut	$7.14	2 packs of 50	$14.28
Thrust Ring*	Aluminum 6061 Plate	$1250	1	$1250
Concrete Blocks	L-Core Concrete	$2.50	4	$10

	Block			
			TOTAL COST	**$2477.12***

The thrust ring is a major cost in this system that was not anticipated prior to the PDB. We are looking to source the aluminum through other methods such as using scrap aluminum.

Structure

Product	**Quantity**	**Unit Mass**	**Unit Price**	**Total Price**
Steel Barrels	2	12kg	$12	$24
Steel Cutters	12	0.076kg	$0.076	$0.912
Aluminum Frame	1	43kg	$107.5	$107.5
Gearbox Mounting Flange Steel	1	32kg	$32	$32
Bottom muck cavity lid steel	1	5.5kg	$5.5	$5.5
Top TBM Lid Steel	1	4.7kg	$4.7	$4.7
			TOTAL:	**$174.61**

Assuming steel price of $0.5/kg, aluminum price of $1.90/kg (source). Increased estimates to $1/kg for steel and $2.5/kg for aluminum to factor in shipping and tax.

Power, Electronics, Sensors, Interlocks

Product	**Quantity**	**Vendor/Part Number**	**Price ($)**	**Total ($)**	**Lead Time**
RTD Sensor	2	Adafruit / PT100	11.95	$23.90	3 business days
RTD Sensor Board	2	Adafruit / MAX31865	14.95	$29.90	3 business days
Voltage Transducer (0-50VDC)*	2	ato.com / ATO-WBV151S01	169.96	$339.92	6-12 days

Current Transducer (400A DC for Main Gearbox Motor)	1	Digikey / L37S400S05M	18.76	$18.76	1-5 days
Current Transducer (24A DC for 4 Propulsion Motors)	1	Digikey / ACS713ELCTR-30A-T	4.58	$4.58	1-5 days
Current Transducer (32/40A AC for Shop Vacuums)	1	Digikey / F02P050S05L	16.73	$16.73	1-5 days
Current Transducer (600A AC for each phase)	3	Digikey / L01Z600S05	18.93	$56.79	1-5 days
Strain Gauge (pack of 10)	1	McMaster-Carr / 7245n24	147.43	$147.73	1-2 days
Smoke Detector	2	Home Depot / i12040	16.97	$33.94	2 days
				TOTAL:	**$672.25**

Soil Conditioner

Based on the results from our Muck Removal Testing, we concluded that we needed soil conditioner to help improve muck removal throughput.

We will be reaching out to Master Builders Solutions (https://www.master-builders-solutions.com/en-us/products/mining-and-tunneling/tunnel-boring-machine) to discuss the best and economically feasible soil conditioner for us to use based on Bastrop, TX conditions.

Scalability to Main Event 2024-2025

Below are our scalability plans to Main Event 2024-2025

- Assuming the tunnel diameter constraint of 0.5 meters stays the same next year, our cutterhead will largely be the same – including any updates from testing and from the actual Mini-Event this year – but almost everything else changes.
- For muck removal, instead of ShopVacs using suction excavation, we'd need to partner with a hydro vacuum excavation truck company for primary muck removal.
- We need to develop a full TBM stability structure, given that we will be digging a horizontal tunnel so now we have lateral earth pressures on the cutterhead and the full TBM.
- Propulsion system will be a hexapod actuation system – very complex electromechanical system that will require heavy dedication and analysis + testing
- Need to add in tunnel linings
- Power Systems will likely be similar but with some extra protection for higher power drawn over longer periods of time
- Need to perform more structural analysis with settling analysis from geotechnical studies. So now, soil properties and the geotechnical data would matter a lot more towards the design of the TBM
- GNC is now something we have to care about
- Launch Structure would need a ramp to propel the TBM down
- In order to support the main-event, team would need to at least double in size with 4 more mechanical engineers, 2 more electrical engineers (one of whom has solid software chops)
- Funding would need to increase significantly to likely $35k+ (including equipment for the new workspace, in addition to the full system cost)
- Need to get a larger dedicated workspace – so need to coordinate with UPenn Engineering or Weitzmann School of Design

I (Rishu Mohanka) started this team one day before the Mini-event application deadline in November, and was only able to recruit (through much difficulty) very few engineers with no real engineering experience, with 0 money, and very little time. Below is a list of engineering philosophies I imparted to the team throughout our entire journey. Part of the scalability challenge is scaling up good engineers, and this philosophy will continue to be utilized by the team as everyone improves on becoming the best engineers they possibly can be.

Part of my challenge here is to ensure long-term team success which means teaching everyone extreme ownership + accountability, how to do good engineering (easier said than done, so a continuous improvement process to be practiced over and over again), and guiding rookie engineers rather than giving them the answer straight up and giving room to make mistakes by giving them extreme ownership/responsibility…that is the best way to learn as far as I know.

Engineering Philosophy

- Optimize for simplicity and less parts. Trade on cost, DFM, lead time

- Design, Test, iterate as needed
- Mission requirements (which should always be questioned) dictate implementation – not the other way around
- Always dive deep into first principles and the extremely low-level details
- Do more critical thinking on your own before asking for help
- The right tradeoff decision depends on optimization goal and context/problem being solved – this usually helps one make a very clear decision from a set of pros/cons
- ALWAYS draw free-body diagrams first and feel the physics going on
- For electronics design, start from load requirements and input power source constraints, and work backwards.
- Hand-calcs for initial sizing; if seeing variation in limit loads then FEA. Use FEA when necessary
- Follow the Elon Algorithm (question the requirements, delete the part, simplify and optimize, accelerate). Wake up the next day, follow it again.

I (Rishu Mohanka) graduate this Spring 2024 and head back to SpaceX Starship in August 2024 full time so I will no longer be with the team – however, I will help advise and guide them when necessary. Therefore, it is up to the existing team underclassmen to dedicate much of their time towards building up this team and securing the talented engineers available at Penn (of which there are already so few) and funding to compete in 2024-25 Main Event successfully