

Penn Hyperloop Final Design Package

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Penn Hyperloop Final Design Package

Team Description

Penn Hyperloop is a dynamic team composed of innovative engineering and business students from the University of Pennsylvania. Founded in 2023, the team first made its mark by participating in the Mini Event of The Boring Company's Not-A-Boring Competition in 3 months with 4 engineers. Since its inception, Penn Hyperloop has grown rapidly, attracting talent from diverse disciplines to design and construct advanced tunneling technology. Now entering the main event as a rookie team, we bring fresh perspectives and ambitious goals to the competition. Below is an overview of our dedicated team members, their specific roles, and responsibilities, as well as our esteemed advisors and their titles, whose guidance has been invaluable to our journey.

Team Member Name	Role
Gabriel Zhang	Team Lead; Business Operations RE; Electronics & Controls
Adam Shamash	Electronics & Controls
Alexander Meija	Structures RE; Cutterhead; Propulsion
Audrey Awong	Business Operations
Ben Saxon	Electronics & Controls RE; Muck Removal; Business Operations
Fady Fahmy	Muck Removal, Structures
Guiherme Ricci Combe	Business Operations, Cutterhead
Ilya Kozhelskiy	Business Operations
Jason de Gentile	Main Drive
Kawin Leephakpreeda	Business Operations, Cutterhead
Manya Gauba	Business Operations, Electronics & Controls
Mehul Vemareddy	Project Manager; Cutterhead RE; Main Drive RE
Nami Lindquist	Business Operations, Electronics & Controls
Odysseas Kotzampasis	Electronics & Controls
Sophie Li	Business Operations
Tej Panigrahi	Propulsion RE; Structures

Tom Huang	Electronics & Controls
Tony Tian	Muck Removal RE
Zeno Dancanet	Strategy

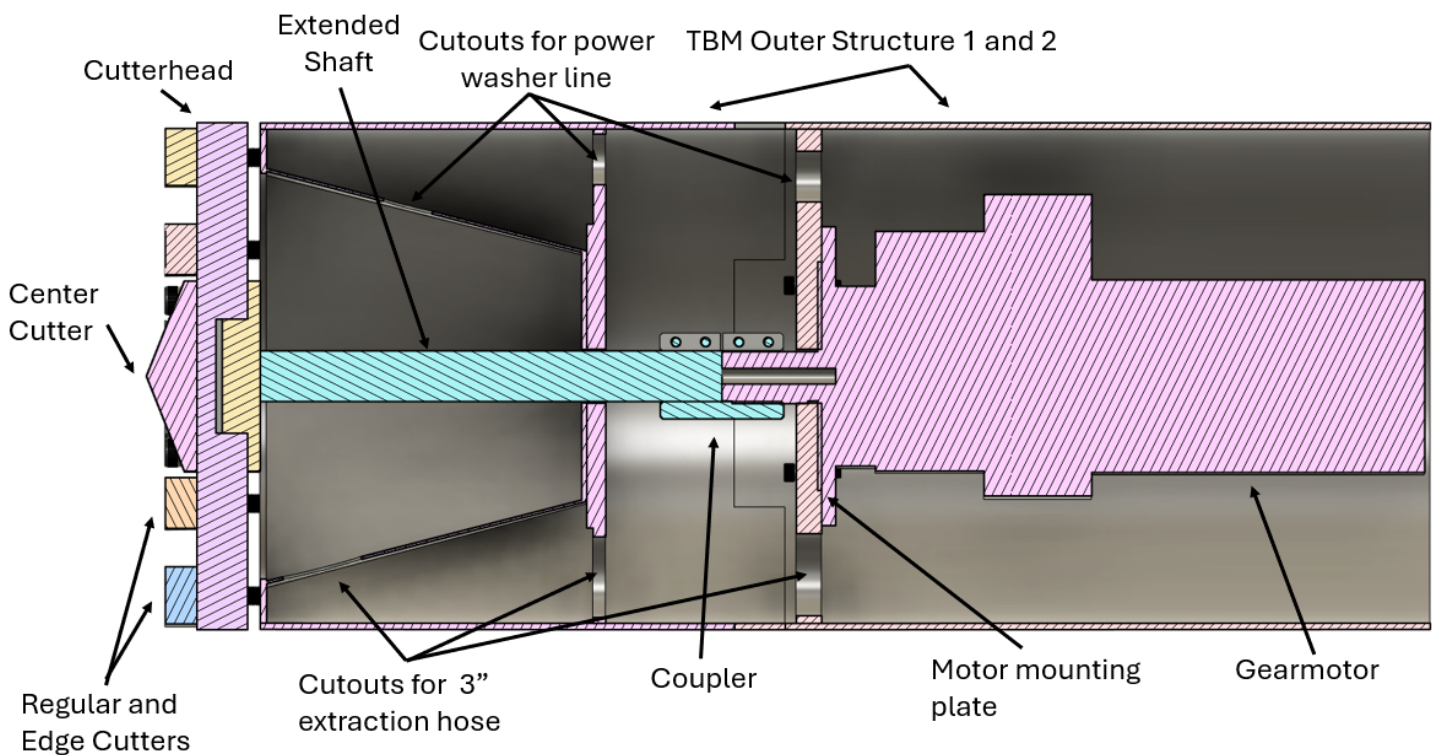
Advisor Name	Role
Douglas Jerolmack	University Faculty Geophysics Advisor
Lei Gu	University Faculty Power Electronics Advisor
Paul Nicholas	AECOM - VP of Tunneling & Trenchless Technology
Rick Lovat	President of Lovat Inc., a wholly-owned subsidiary of Caterpillar
Rishu Mohanka	Team Founder, currently works at SpaceX
Siddharth Deliwala	University Faculty Advisor
Tom Luca Reinhardt	Founder & CEO, Elara Aerospace

Design Description of TBM

Top-Level Summary

Key Design Specifications

- Tunnel Diameter
 - 0.508 m
- Tunnel Length
 - 30 m
- Machine Power Source
 - 240V Three Phase & 120V Single Phase
- Peak Power Consumption
 - 19 kW
- Cutterhead Targets
 - 2700 Nm
 - 10 RPM
- Tunnel Dig Time - 11 hours
- Features two sections attached together like a jigsaw puzzle. Since TBM is always under compression only, there is no attachment mechanism. Duct tape will be used to protect interiors from soil.



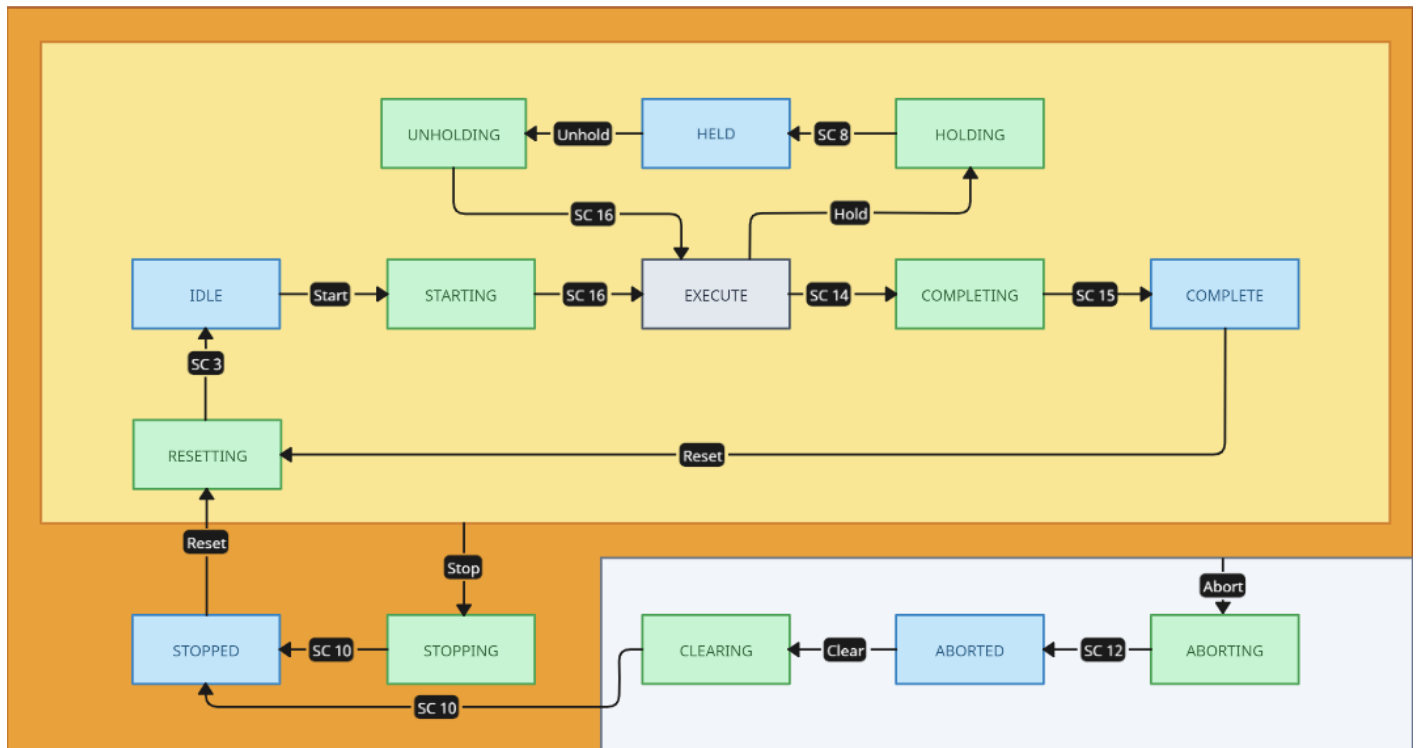
TBM Dimensions and Mass

Subsystem	Mass (kg)	Dimension (Bounding Box LxWxH)
Cutterhead	70	4" x 21" x 21"
Drive System	250	30" x 12" x 12"
Muck Removal	90	16" x 20" x 20"
Structures	90	3'10" x 20" x 20"
Total	500 kg	4'4" x 21" x 21"

TBM Power Source and Consumption

System	Power Draw (kW)	Power Source	Notes/ Details
Main Drive	7.5	240V Three Phase	
Propulsion	11	240V Three Phase	
Muck Removal	0	N/A	Gas powered vacuum truck and gas-powered pressure washer
PLC, Circuits, Navigation	0.5	120V Single Phase	Rectified to 24VDC. Expect absolutely no more than 20A
Total	19		

We are well under the 100kW threshold.

TBM State Diagram

State Descriptions:

SC #	State	Description	Transitions
1	STOPPED	The TBM is fully stopped and de-energized via shunt trip breaker. The machine is in a safe state for maintenance or inspection. If it's the beginning of an operational cycle, the next pipe segment is manually loaded and wires and muck lines are rerouted. If the machine is in the middle of an operational cycle (i.e there was an abort somewhere during it), then adjustments to the physical system can be made.	Awaiting an operator's RESET command to energize the system.
3	IDLE	The TBM is energized but not performing any active operations. This occurs 1) after startup, and 2) after completing the previous cycle and the system has been reset to default positioning. Interlock checks are performed to ensure safe energization. Adjustments to motor parameters (RPM, torque, etc.) for the upcoming operation cycle can be made by the operator. Stack light indicates that the machine is energized.	Awaiting the operator's START command to begin pipe jacking.
2	STARTING	The system performs final interlock and safety checks. Stack light indicates that the machine is operating.	Transitions to EXECUTE state upon

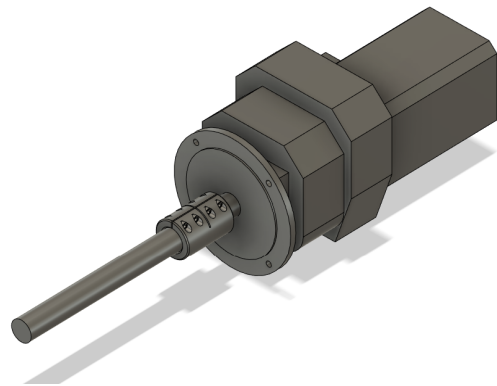
			successful safety checks.
16	EXECUTE	Both main drive motor and propulsion motor are active based on parameters set during the IDLE state. The TBM is actively excavating soil and thrusting forwards.	Transitions to COMPLETING state once 1m is bored.
14	COMPLETING	Triggered when the actuator has extended 1m. The TBM begins procedures to end the active operation cycle. Motor RPMs are slowly reduced to zero.	Transitions to COMPLETED when the RPM of main drive and propulsion motors are both zero.
15	COMPLETE	Both motors are energized but not active. The TBM has successfully inserted the pipe into the ground. The TBM can now be manually controlled by the operator in the event that small adjustments have to be made.	Transitions to RESETTING when pipe-insertion is confirmed successful by the operator.
10	RESETTING	The TBM is energized and resets itself to default positioning. The propulsion actuator returns to its retracted position. All interlocks are checked.	Transitions to IDLE when safety checks are passed and the machine is in a retracted position.
11	ABORTING	Initiated when an error, fault, or emergency stop condition occurs. The shunt trip is broken, and the high-power system is immediately deenergized. All other state transitions are overridden to address urgent issues. The main indicator light will turn off, indicating system deenergization, and the stack light will turn red, indicating an ABORT.	Transitions to ABORTED when the system is deenergized and indicator lights reflect the new state.
12	ABORTED	The TBM remains in a deenergized, non-operational state following an abort. Operator intervention is required to diagnose and resolve the issue. The machine is prevented from restarting until faults are cleared.	Awaits the operator's CLEAR command to enter the STOPPED state.
13	STOPPING	The PLC trips the shunt breaker, deenergizing the system.	
7	HOLDING	The operator initiates a pause in operation. The TBM begins procedures to temporarily stop active processes. Motor RPMs are reduced to 0. Prepares to enter the Held state without completing the full cycle.	Transitions to HELD once RPM of both main drive and propulsion motors are reduced to 0.
8	HELD	The TBM is paused but remains ready to resume operations quickly. Both RPMs of the main drive and propulsion motor are zero, but the system maintains energization and operational readiness.	Transitions to UNHOLDING when inspections or adjustments are made

		Used for short-term interruptions, such as when brief inspections or adjustments need to be made.	and the operator issues the UNHOLD command.
9	UNHOLDING	The TBM reverses the holding procedures to resume operation. The main drive and propulsion motors become active.	Transitions back to the EXECUTE state upon full reactivation.

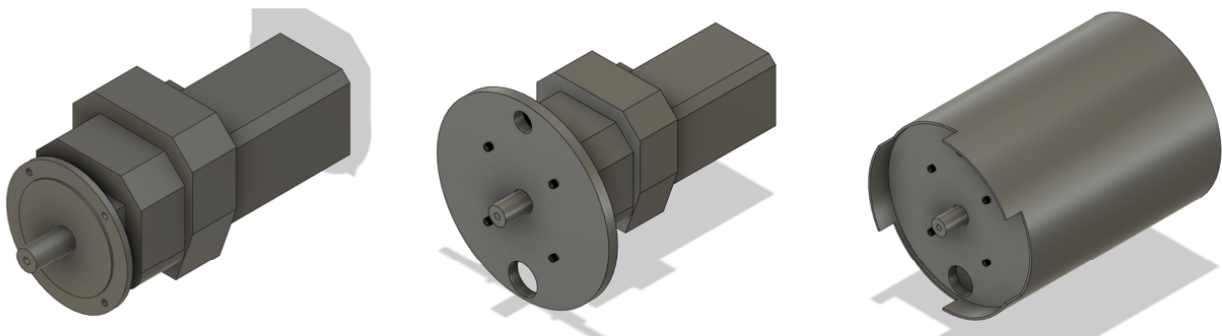
TBM Excavation System

Our excavation system involves the use of an in-line gearmotor with extended shaft to deliver torque to our cutterhead.

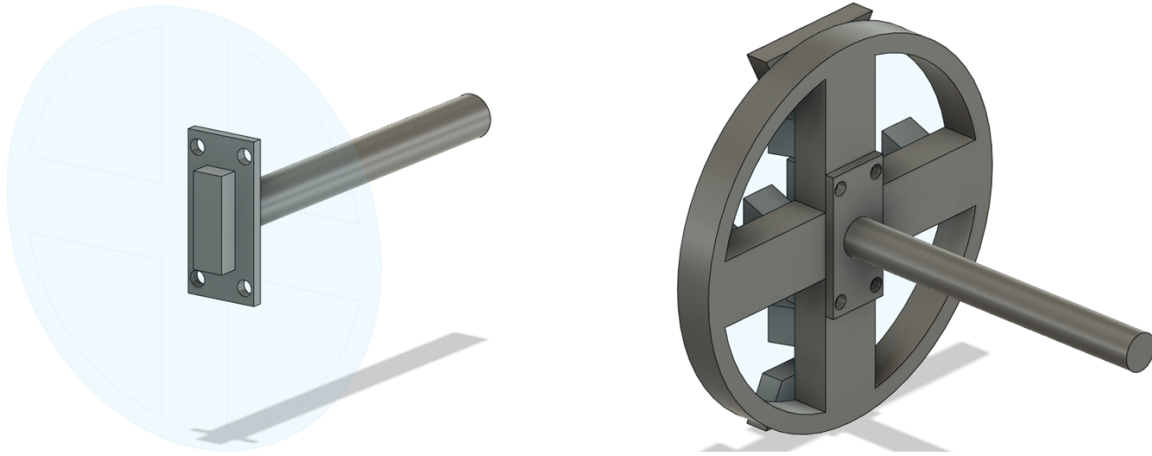
Our main drive system features a 2700 Nm max torque gearmotor that runs at 10 RPM. We plan to run the motor at ~2400 Nm operationally. The calculations justifying this can be found in TBM Structural Analysis. The gearmotor features a 53mm keyed shaft, onto which we will use a coupler to add an extended shaft of the same diameter. The length of this shaft was determined based on the length of the muck chamber with room for error. Further analysis for this is also in the TBM Structures Section.



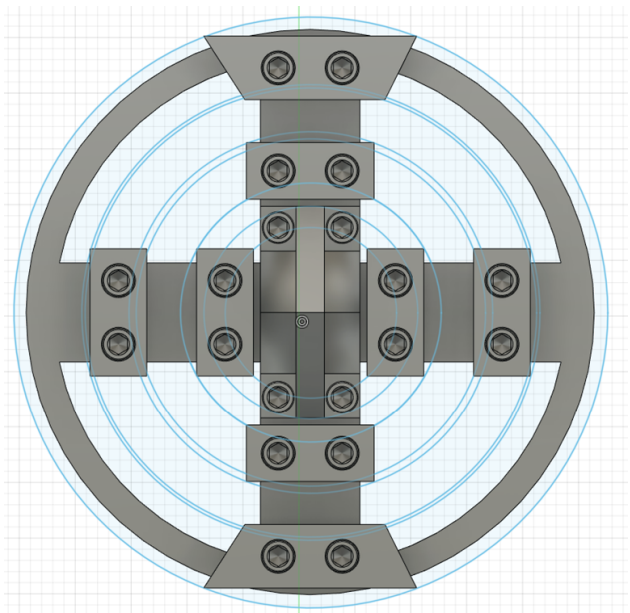
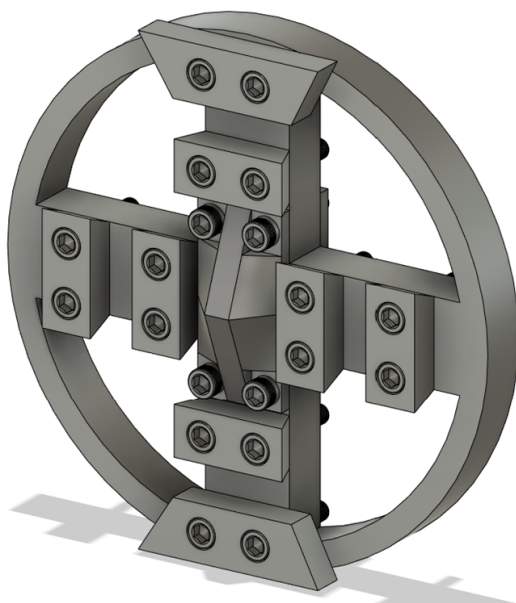
Below is a mockup of the gearmotor based on manufacturer drawings. The gearmotor is attached to our structure using a mounting plate that features through holes corresponding to those on the motor faceplate. This annular mounting plate is welded onto our TBM outer structure. This can also be seen below.



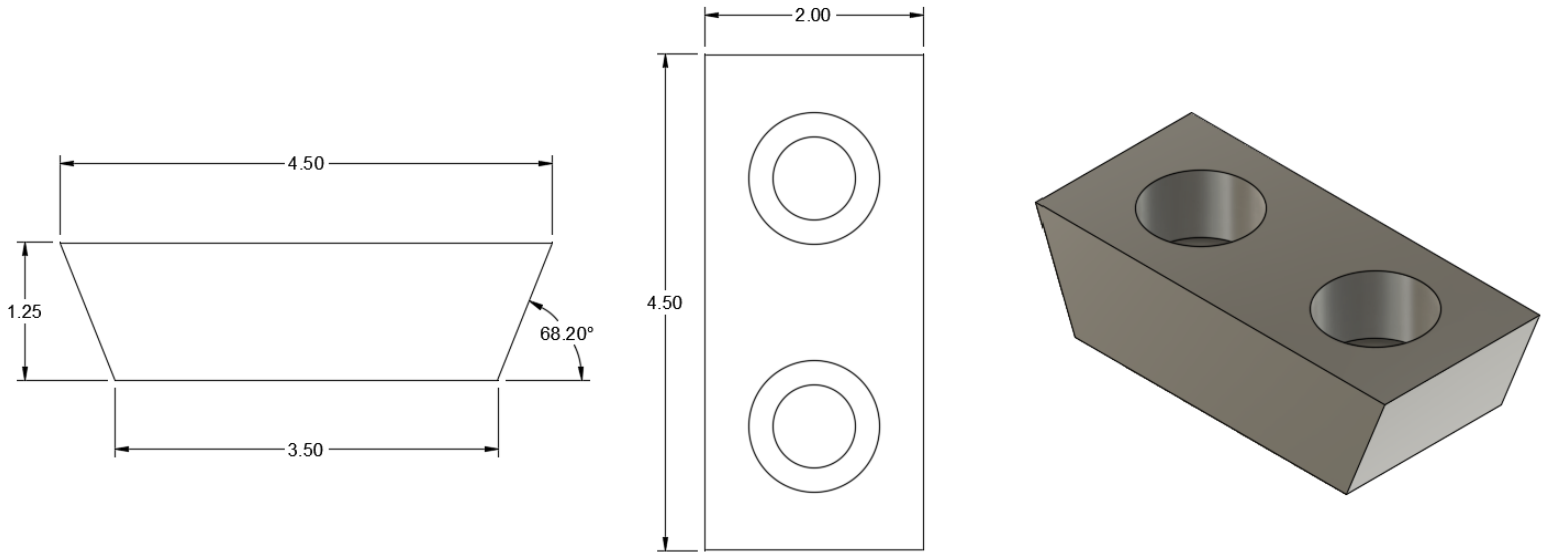
The extended shaft bolts to the back of the cutterhead using an alignment piece. This rectangular piece is used to slot into the back of the cutterhead frame in order to line up the bolt holes while also serving a dual purpose of ensuring efficient torque transfer as illustrated below.



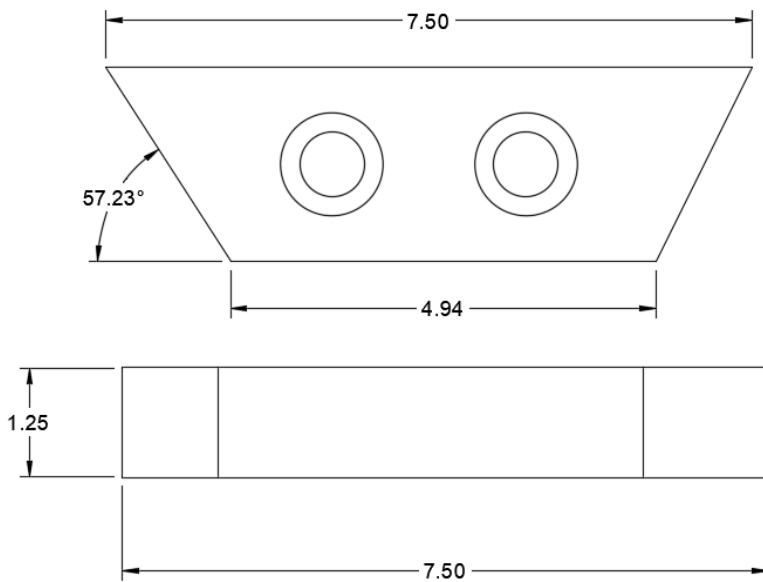
Our cutterhead itself is designed with 6 “regular” cutters, 2 “edge” cutters and one center cutter. The cutterhead has been designed based on advice from previous entrants of the competition and our technical advisors in industry based on our experience with Bastrop soil conditions. The images below show our cutterhead with cutters bolted on it along with a front view which shows concentric circles of cutting that proves that the alternating cutters on parallel spokes will cut the entire surface area of the cutterhead. The image also shows a circle larger than the frame diameter. This is because our edge cutters scrape away more than the diameter of our tunnel to produce an industry-standard overcut. The extent of this overcut is a $\frac{1}{2}$ ” thick ring. This puts the effective cut diameter to 21”.



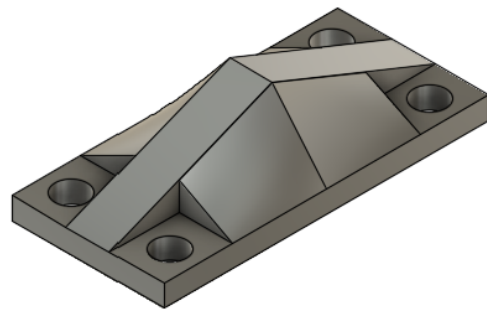
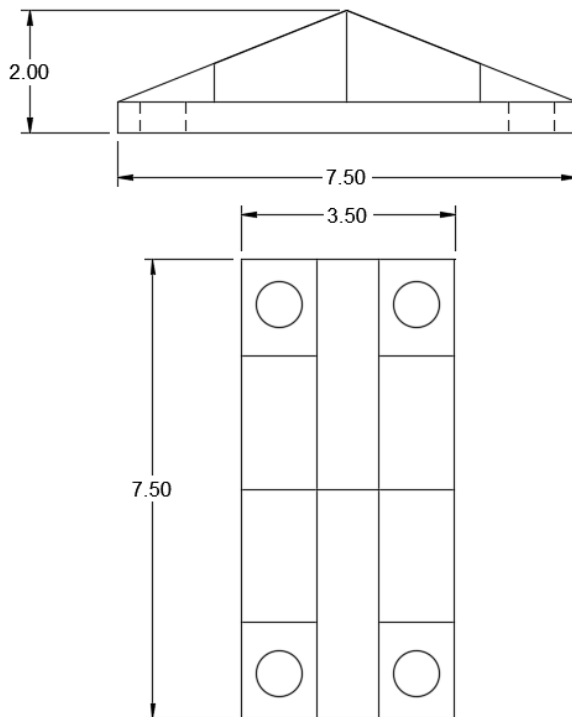
Taking a closer look at the different types of cutters, the regular cutters are 1.25" in height past the cutterhead frame. Its base spans the cutterhead spoke width exactly, and a 68.2 degree attack angle is used to cut into the ground and force the excavated muck into the gaps in our frame and into the muck chamber behind it. The cutters are bolted in using $\frac{3}{4}$ " fasteners.



The edge cutters are also 1.25" in height past the frame and bolted using $\frac{3}{4}$ " fasteners. The attack angle in this case is 57.23 degrees and the cutters are sloped again to force excavated muck into the muck chamber. All cutters are to be manufactured in house and made of Mild Steel.

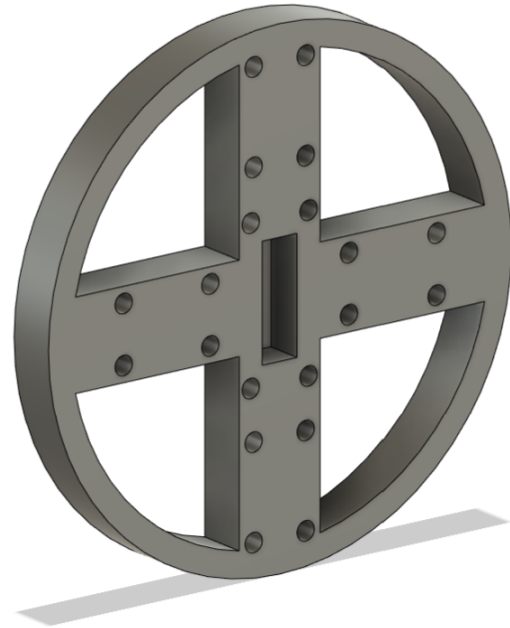
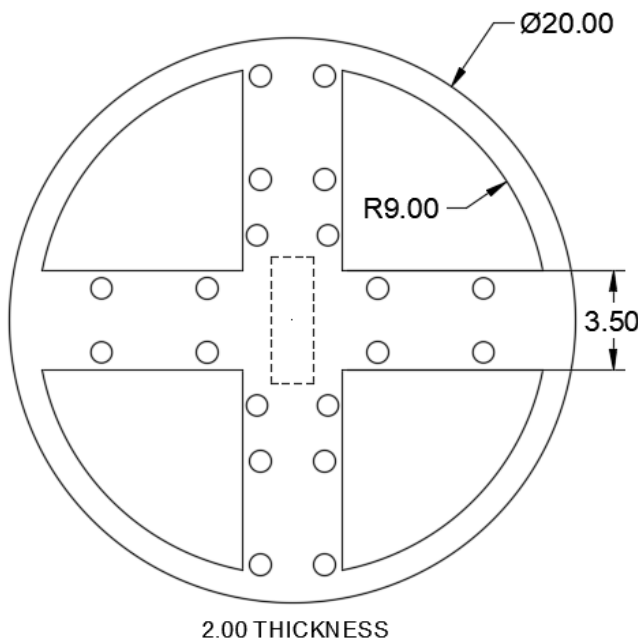


The center cutter serves to scrape the muck that is close to the center of the cutterhead but also to act as a bearing supporting the TBM and its main drive system. It does so using its pointed end structure. Additionally, this point is the first part of the TBM to contact any uncut soil with a height of 2" which is larger than the regular and edge cutters. The center cutter is fastened through the cutterhead frame, into



the alignment piece behind it using the $\frac{3}{4}$ " fastener as well. All cutters are manufactured in house and made of mild steel.

Finally, the cutterhead frame itself is designed with an opening face ratio of ~50%. Below are the dimensions of our frame along with an image of the back of the frame. The back features a cut for the shaft alignment piece.



dimensions, shaft dimensions, cutterhead bolt dimensions and other calculations behind all design decisions are present in TBM Structural Analysis Section.

Estimated Tunneling Time

The tunneling time is limited by three subsystems: propulsion, muck line extensions, and muck extraction. Regarding propulsion, the advance rate of our screw jack largely determines our tunneling speed. According to our supplier, we expect the screwjack to operate at 90mm/min for the majority of our tunneling until increased load from jacking pipe additions. This alone corresponds to a tunneling time of 5.5 hours assuming that we tunnel the full 30m.

Since we are using a pipe-jacking tunneling method, we expect the process of jacking pipe additions to be a time intensive process. This requires us to de-energize our electronic components, install our clay jacking pipes using an excavator, interface the pipes with our jacking plate, install additional muck extraction lines, then re-energize our system. Given that our jacking pipes are each 1 m long and

our TBM is 4m long, this corresponds to 30 pipe additions. Assuming each addition takes a minimum of 10 minutes to complete, this increases our estimated tunneling time to 11 hours.

Lastly, we expect to unclog our muck chamber every 5 minutes in advance. This entails halting our advance while continuing our chamber conditioning and vacuum extraction processes until our extraction line pressure sensor reads atmospheric pressure. We expect this to increase our tunneling time to 11 hours in total.

More analysis is yet to be done into the specifics of our pipe change routine, and de-clogging and our tunneling time we change accordingly. We aim to be fully prepared with all checks and testing done prior to dig day to maximize time available for digging. Additionally, we will look to increase the speed of our actuation system since the screw jacks have a higher advance rate than 90 mm/min for the first few meters.

TBM Propulsion

Force Analysis:

In order to find the propulsion force required to push our system forward for 30 m, we had to find out how the Earth Pressure Balance and Frictional force of soil would counteract our motion. The below equations were found in a paper, “Determining the optimal thrust force of EPB Shield Machine by Analytical Solution” (Shannugan et. al, 2008), published by the Electronic Journal of Geotechnical Engineering.

$$F = F_p + F_f$$

Where F is the total thrust force required, F_p is the force to oppose the Earth Pressure Balance and F_f is the force to oppose the frictional resistance.

$$\begin{aligned} F_{pmax} &= 2 \int_0^R \int_0^\pi [\gamma z k_p + 2c\sqrt{k_p}] r d\theta dr \\ &= 2 \int_0^R \int_0^\pi [\gamma(H + R + r \cos \theta) k_p + 2c\sqrt{k_p}] r d\theta dr \\ &= \pi R^2 [\gamma(H + R) k_p + 2c\sqrt{k_p}] \end{aligned}$$

$$\begin{aligned} F_f &= 2 \int_0^L \int_0^\pi f(\sigma_z + \sigma_x) R d\theta dy \\ &= 2 \int_0^L \int_0^\pi f[\gamma(2 - \sin \phi)(H + R + R \cos \theta)] R d\theta dy \\ &= 2\pi f \gamma L R (2 - \sin \phi)(H + R) \end{aligned}$$

Where the above variables indicate:

Description	Symbol	Unit	Value
Radius	R	m	0.25
Unit weight of soil	γ	N/m ³	18500
Overburden depth	H	m	0.7
Cohesion of Soil	c	N/m ²	15000
Coefficient of Passive Earth Pressure	k_p	-	3
Internal Friction Angle	ϕ	Degrees	30
Composite force of earth pressure	F_p	N	20555.15097
Soil Friction Coefficient	f	-	0.3
Frictional Force of Earth pressure per meter of pipe	F_f	N/m	12423.03545
Meters of pipe	L	m	30
Total Force	F	kN	393.2462145

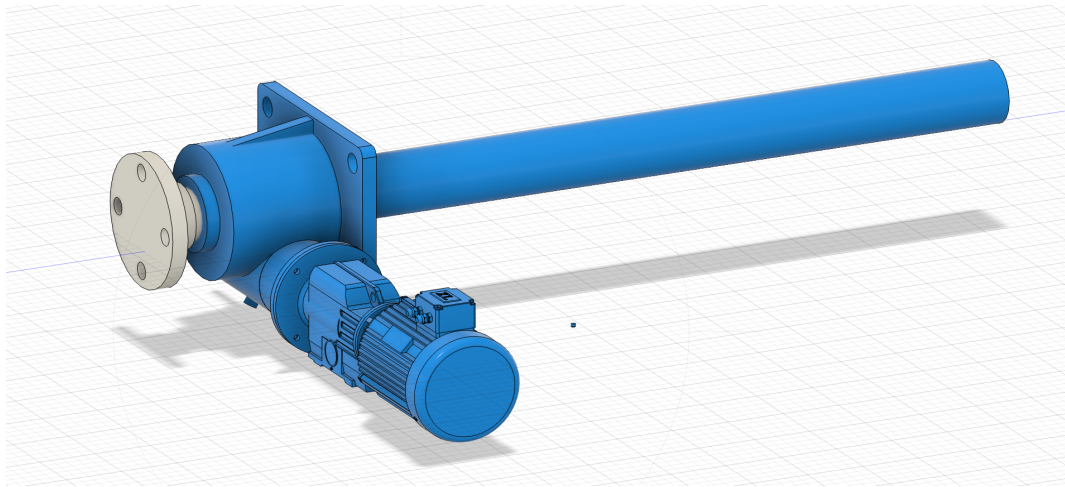
The calculations show that the force needed to dig 30m into the ground is about 393kN. However, this doesn't take into account the overcut we have with our cutterhead and our TBM, which reduces the force required to push into the ground. Moreover, the cutterhead will be ejecting a lubrication fluid to make it easier to cut through the soil, which also makes it easier to push through the soil. The effect of this, however, is very difficult to quantify or calculate with any level of certainty. Thus, after discussing with industry experts, we were recommended to design our propulsion system for **300kN** of thrust force. Even if this assumption is wrong, our original model predicts that we should be able to dig about 22.5 m into the ground with 300kN of force, which is still a valiant attempt.

Propulsion Mechanism:

Currently, for the machinery that will drive our propulsion system, we are looking at a pipe-jacking system, where electric jacks propel the TBM from behind, allowing tunnel lining segments to be added continuously as the machine advances from a fixed position. Although we considered the gripper method, which relies on the TBM gripping the sidewalls to push forward, we ultimately chose pipe-jacking due to its seamless integration with the tunnel lining process, enhanced precision, reduced surface disruption, and superior stability given the ground conditions in Texas.

While we've considered it, we have decided against an indexing method to reduce points of failure along the pipe-jacking method and/or ease the manufacturability of our system.

In particular, our propulsion system will be using a single screw jack that will interface with our thrust plate to push our TBM and the pipes forward. The design for the thrust plate and the remainder of the launch structure will be explored in more detail in section 4. The screwjack we are proposing is going to be an upright translating trapezoidal screw. The translating screw allows our screwjack to operate as a linear propulsion method, by not rotating and providing an upward thrust push. We also chose a specifically trapezoidal screw as those are better designed for low speed and high force applications, rather than the alternative ball screw which is more for high speeds and low forces. The screwjack is run by a 4 kW inline gear motor rated for 480V 60Hz 3-phase power.



- Current Potential Screw Jack Supplier: Jacton Jacks (Model # JTW-50T-M1)
- Specifications: 50 ton force maximum static load, stroke length of 1500 mm
- Maximum Advance Rate: 90 mm/min



Tunnel Liner Material and Mechanism

Since our design uses a pipe-jacking mechanism, we have picked pre-prepared pipes to serve as our tunnel lining. Specifically, we are using NO-DIG Vitrified Clay Jacking Pipes (VCP-J) from the company Logan Clay. These clay pipes are high strength, corrosion resistant pipes that have 316 stainless steel collars on the spigot and bell end to transfer forces. This product is typically used in industry for sewage tunnels that are dug using the pipe-jacking propulsion mechanism. We are in talks for sponsorship for 30 one-meter long, 18” diameter pipes from the company for our use.

Factor of Safety & Testing

The failure modes in question for this component are failure under compression from buckling or axial stress. The pipes will fail in buckling before axial stress due to the thickness of our pipes (~3” thickness) and strength of vitrified clay.

These failure modes will not occur however since Logan Clay has provided us test results for how the pipes withstand compressive force testing. Our chosen size of pipe can withstand 109 Tons at and SF of 2.5 and 79 Tons at 3.5 SF.

We plan to apply 300kN of thrust force which is about 30 tons. This is significantly below the suggested safe loads and puts our effective SF at 9.

This data eliminates the need for our own calculations or testing on tunnel lining. In fact this industry grade testing is stronger in certain versions of the test as per conversations with our contact at the National Clay Pipe Institute (NCPI) who works with Logan Clay.

Delivery System

Our system and method does not require lining storage or in-tunnel delivery mechanisms. We plan to rent an excavator and use lifting chains to guide the clay pipe onto the launch structure. Specific pipe-change and lifting routines are yet to be written based on consultation from our contact at NCPI. Our thrust plate (see Launch Structure) is designed to interact with the back of the pipe to transfer axial load effectively once placed into our launch structure in the pit.

Muck Removal System

Given that the muck removal system is high-risk, this system’s requirements incorporate both large safety factors and cost-effective design choices. Specific analyses to be examined are: soil settlement and heave, soil conditioning process, muck extraction process, and muck chamber design.

Soil Settlement and Heave

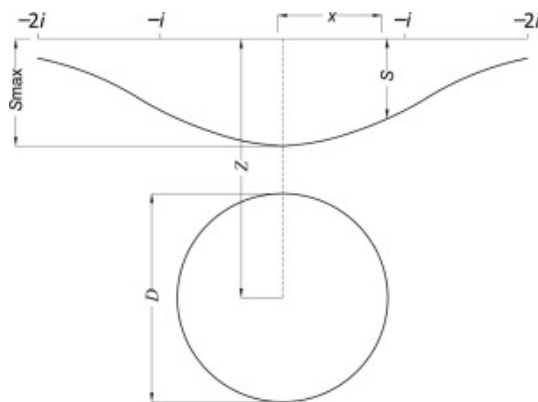
Soil characterization is key as it informs tunnel stability and guides TBM operation settings. Soil settlement during tunneling depends on soil properties like density, moisture, and failure strain, which can be obtained from geotechnical logs and laboratory tests. It importantly concerns the relationship between volume loss and the maximum settlement of soil.

- Volume Loss is calculated using the cutterhead diameter and tunnel diameter, both critical for determining the extent of volume loss and surface settlement.
- Settlement Calculation uses parameters such as volume loss, soil stiffness, and the K values representing soil type. For non-cohesive soils like sand, a lower range of K (0.25–0.5) is applied, whereas cohesive soils like clay have higher K values (0.3–0.9).

By analyzing the soil reports provided in the Geotechnical analysis, we are able to infer some predictions on volume loss, soil stiffness, and settlement.

1. Boring Log B-1: Shows fat clays and lean clay with sand (medium to high plasticity) to about 34.5 feet depth, with sand layers below that depth. The settlement in these clayey layers may be moderate due to higher compressibility.
2. Boring Log B-2: Also includes clayey layers (fat and lean clay) with some well-graded sand and gravel at greater depths, which may induce a stiffer response and lower settlement.
3. Boring Log B-3: Primarily lean clay with sand, but contains silty, clayey sands at different layers, suggesting varying settlement potential depending on depth.

Using [this article published in Underground Space](#) as a reference for understanding settlement and heave by Ahmed, et al, we apply an empirical method developed by [Cording and Hansmire \(1975\)](#), later on further expanded by [O'Reilly and New \(1982\)](#).



$$S = \frac{VL}{i\sqrt{2\pi}}$$

From this model, we understand soil settlement as an inverted normal probability curve where the maximum settlement corresponds to the center of the dug hole. Here, VL corresponds to the volume loss, and i is a factor based on the soil conditions and geometry. For cohesive soils such as those in Bastrop, we determine i using the length of the tunnel and the soil cohesiveness. This relationship is directly taken from the model constructed in [Ahmed et al.](#)

From direct shear testing (e.g., B-1 and B-3), parameters such as effective cohesion (c') and internal friction angle are derived. These parameters, especially cohesion, are critical for predicting settlement and stability. Furthermore, the high plasticity index in clayey soils (e.g., PI of 37 for B-1) indicates the potential for swelling, shrinking, and consolidation under loading. Dry density values from boring logs (e.g., 102-107 pcf for B-1) help predict compressibility under tunnel-induced stresses.

Below is a table compiled given the geotechnical report on values and physical properties of soil that will be taken into consideration in the remainder of the muck removal section.

Parameter	Boring B-1	Boring B-2	Boring B-3
Soil Type	Fat Clay (CH), Lean Clay with Sand (CL)	Lean Clay with Sand (CL), Fat Clay (CH)	Lean Clay with Sand (CL)
Liquid Limit (LL, %)	59	33	38
Plastic Limit (PL, %)	22	18	17
Plasticity Index (PI)	37	15	21
Moisture Content (%)	17.2 - 17.6	13.3	14.7 - 17.7
Dry Density (pcf)	98.8 - 103.0	112	96.2 - 101.1
Effective Cohesion, c' (psi)	0.9 (peak)	Not provided	2.0 (peak)
Void Ratio (e)	0.67 - 0.78	Not provided	0.64 - 0.72
K-Value (Stiff Clay)	0.3 - 0.6	Variable	0.3 - 0.4

Conditioning

Our chosen conditioner is MasterRoc APC 214 Anti-Clay Polymer as it ensures the viscous, clay-like Bastrop soil can be excavated and extracted effectively. This conditioner reduces the soil's permeability and improves cohesion, which provides face support and reduces the risk of cutterhead failure.

In preparing soil for cutterhead interface, we plan to use a pressure washer, *kink*-resistant hose, and high pressure foaming nozzle to distribute conditioning solution onto incoming soil. Once conditioned soil has reached the back of our cutterhead and is approaching our muck chamber, it will be further conditioned using a second high pressure line to prevent muck from sticking inside the chamber.

Our calculations are based on the screwjack's maximum advance rate (90mm/min) and experimental estimates of conditioner-to-soil ratio and conditioned soil compression. Additionally, our goal of digging a 30m long tunnel and frictional losses associated with hose length, material, diameter, and other system components were accounted for. The Bernoulli equation and the following major/minor head loss equations were used in the analysis.

$$z_1 + \frac{P_1}{\rho g} + \frac{V_1^2}{2g} = z_2 + \frac{P_2}{\rho g} + \frac{V_2^2}{2g} + h_L \quad h_{L \text{ major}} = f \frac{\ell}{D} \frac{V^2}{2g} \quad h_{L \text{ minor}} = K_L \frac{V^2}{2g}$$

Description	Unit	Value
Cutterhead Area	m ²	0.22
Advance rate	m/min	0.090
Compressed Clay factor	-	0.7
Soil Intake capacity	gpm	3.63
Hose Inner Diameter	m	0.0064
Hose length	m	46
GCS Concentration (ratio)	-	0.03
Foam to Soil Ratio	-	0.5
Frictional Losses	-	0.078
Minimum Conditioning Solution Flow Rate	gpm	4.65
Minimum Conditioner Requirement	gal	3.6
Minimum Water Tank Requirement	gal	3094

Based on these calculations, we will need a 3000 psi pressure washer that outputs a flow rate of about 10 gpm, a 3000-gallon water tank, and approximately 3.6 gallons of our chosen conditioner.

While the majority of these parts will be assembled directly into the TBM, we have also created an orderly plan for efficiently setting up and operating the system at the dig site.

Safety Precautions:

All personnel on site must be wearing closed-toe boots, long pants, long sleeves. Gloves should be worn when handling the solution, and a long mixing bar should be used. Due to the low concentration of the solution in water and the fact that the dig system will be done in a well-ventilated outdoor space, masks will not be necessary. However, take caution to not overexpose oneself to the conditioner mixture (e.g. by not standing to the IBC tote when the lid is opened for extended durations).

Extreme care should be taken when working with the 3000-gallon IBC tote due to the size, volume, and weight of the tote. It should be deposited and filled on flat ground to avoid motion or sliding. Anyone filling, emptying, or adding conditioner to the tote must have another team member nearby to supervise in case an accident shall occur.

The pressure washer system involves high-pressure water output, strong enough to cut skin and damage surfaces, so protective gear such as gloves, safety glasses, and long clothing need always be worn. The nozzle should be hooked and secured into the TBM system at all times. Place the pressure washer on secure, solid ground, with the anchoring pin inserted. Gas pressure washers also involve the need for working with gasoline. Keep all potential open flames far away from the setup, and a fire extinguishing setup must be within distance as per Texas safety restrictions.

Conditioning Setup:

The IBC tote will be rented from the Austin, Texas area and transported to the dig site. Place the tote on secure, flat ground close to the TBM dig setup.

The 3000-gallon IBC tote should be filled with water by the time of the start of the competition. Filling should begin through nearby water sources 1-2 days ahead of time.

Before the dig begins, add 3.6 gallons of soil conditioner to the IBC tote. Mix well using a long mixing stick. Due to redeposition of the ACP 214

Pressure Washer System Setup:

The pressure washer will be rented from the Austin, Texas area and transported to the dig site. Replacement gas containers or sources should be transported as well and stored in a dry location far from possible open flames.

Verify fuel level. Replace or refill tank if needed.

Connect the IBC tote to the pressure washer securely, checking all hoses and fittings for leaks.

Run a pressure washer for at least 10-15 sections before attaching to confirm that there are no leaks

polymer, the solution should be mixed every 3 / blockages.
hours when the TBM system is running.

Using a hose, attach the outflow line of IBC tote to the pressure washer system. Use the hose adaptor to change pipe diameter from tank to pressure washer. Use an M/M connector if the inflow port of the washer requires an M connector.

Connect the pressure washer outflow pipe to the hose which has one end secured inside the TBM. Secure well.

TBM System Checks:

1. Visually confirm that the pressure washer nozzles are secured within the muck chamber. If nozzles are visibly worn out, more than 1000 gallons of water have been spewed out, or a dig in which more than 2000 gallons of water will exit the system, remove and replace the nozzles with new ones.
2. Confirm that the inflow pipe of the conditioning system is well-secured.
3. Turn on the pressure washer for at least 15 sections to confirm that there are no leaks or blockages anywhere and that the angles of the nozzles are correct. Adjust as needed.

Extraction and Pressure Balance

Our muck extraction system consists of a vacuum truck and steel pipes that will interface with the vacuum truck hose. We plan on pre-installing the steel pipes inside the MTBM and progressively adding pipes as we dig deeper and add our clay jacking pipes.

Our calculations for the vacuum truck requirement incorporate a safety factor of 10 that ensures our extraction line is limited to 10% of its volume (in steady state). There is an additional safety factor of 2 associated with the maximum mass of muck and water in the chamber at any given time. Pipe diameter was set to 2.75in to leave ample room for the necessary pipe thickness, as our TBM's geometry does not permit an extraction line greater than 3.0in outer diameter.

The previously seen head loss equations were once again used to estimate pipe frictional losses. The extraction efficiency is an assumption that accounts for inefficiencies in muck excavation that are a result of clay sticking to the chamber and the back of the cutterhead. In the future, we aim to refine this assumption through experimentation.

Description	Unit	Value
Soil Intake capacity	gpm	3.63
Water density	kg/m ³	997
Clay density	kg/m ³	2000
Pipe Inner Diameter	in	2.75
Pipe length	m	46
Extraction Efficiency	-	0.9
Frictional Losses	-	0.1519
Required Extraction Rate	cfm	59.2

Given our jacking rate of approximately 90mm/min, our required extraction rate from soil intake is low compared to industry-standard vacuum trucks. However, we expect the negative pressure requirement to be high to ensure a pressure balance near the cutterhead- tunnel interface.

To ensure a pressure balance between the cutterhead face and the inside of the TBM (muck chamber) an analysis of the face pressure was completed, accounting for pressure loss due to frictional losses in the extraction line. The minimum pressure required to stabilize the cutterhead face can be described using this equation:

$$P_F = (\gamma \cdot z_0 + q) - N \cdot c_u$$

This corresponds to the undrained condition, since we expect the clay-like soil in Bastrop to have low permeability. Values for parameters seen in the face pressure equation were determined by reviewing validated literature and geotechnical reports for similar soil conditions to that of the dig site in Bastrop.

Description	Unit	Value
Muck Total Unit Weight (γ)	kN/m ³	18.85
Tunneling Depth	m	2.5
Surcharge Loading at Top of Soil Layer (z_0)	kPa	0
Nondimensional Stability Number (N)	-	8
Undrained Shear Strength of Soil Layer (c_u)	kPa	12.0
Face Pressure (P_F)	kPa	48.87
Pressure Losses Due to Friction	kPa	8.59
Required Truck Negative Pressure	in Hg	16.97

This analysis confirms our hypothesis. Our vacuum truck selection is dependent on the required negative pressure of 17 in Hg. For ease of procurement and an additional safety factor, we aim to procure a vacuum truck with specs of 500 cfm and 20 in Hg of negative pressure. Additionally, the use of a pressure sensor in our vacuum line will help us monitor clogs and pressure fluctuation. We will adjust our TBM advance rate accordingly. To ensure this pressure and flow rate do not cause our pipes to fail, our team did a failure analysis due to hoop stress (left) from static weight force ($P = mg / A_{eff}$) and momentum flux ($P = \rho v^2 A / A_{eff}$) as well as elastic buckling pressure (right) from the truck's negative pressure.

$$\sigma_c = \frac{Pd}{2t} \qquad p_{el} = \frac{2E}{1 - \nu^2} \left(\frac{t}{D} \right)^3$$

Description	Unit	Value
Pipe Inner Diameter	in	2.75
Max Mass of Muck + Water in Pipe	kg	3.26
Max Stress	Mpa	250
Poisson Ratio	-	0.3
Elastic Modulus	Gpa	210
Static Stress Hoop Stress Thickness	in	0.0367
Truck Flow Rate	cfm	500
Muck Velocity	m/s	0.377
Momentum Flux Hoop Stress Thickness	in	0.0255
Truck Negative Pressure	in Hg	20
Collapse Pressure Buckling Thickness	in	0.145

These calculations entail that collapse pressure buckling thickness is the limiting factor for pipe failure, requiring a minimum pipe thickness of 0.145in. This includes the previously mentioned safety factor of 2 associated with the maximum mass of muck mixture in the pipes. For muck removal to be set up and ready for operation at the dig site, we have also put together an orderly plan for the effective and safe assembly of the system.

Safety Precautions:

Strict safety precautions are essential to prevent hazards and ensure smooth, secure processes when working with large industrial machines such as the vacuum truck. Park the vacuum truck on secure, solid ground. Operators must wear appropriate personal protective equipment, including hard hats, safety glasses, hearing protection, gloves, and steel-toed boots. All equipment, including hoses and connections, should be thoroughly inspected for leaks, cracks, or damage to avoid exposure to harmful substances or loss of vacuum pressure.

Static discharges may ignite any volatile gasses or materials in the work area, so flammable materials must be kept far away. A fire extinguishing setup must be within distance as per Texas safety restrictions.

The vacuum truck's noise can impair verbal communication, which necessitates hearing protection. Moreover, clear hand signals, in particular one for an emergency stop procedure, must be developed and understood by all operators. There should always be an operator who has the emergency shutdown function within reach.

All the above precautions taken for the soil conditioning system still apply as a large part of the muck excavated is the inflow conditioner.

Vacuum Truck Setup:

The vacuum truck will be rented from the Austin, Texas area and transported to the dig site. Park the truck on secure, flat ground very close to the TBM dig setup. Engage the parking brake and deploy any necessary stabilizers to secure the vehicle.

Verify fuel level of the vacuum truck. If the quantity of gas needs to be topped off, do this immediately at the closest gas station.

Checking that all safety controls, such as emergency shutoff switches and grounding cables, are in place and functional.

Establish clear lines of communication between

Pipe Line Setup:

The pipes and couplers will be brought to the dig site completely separated. All assembly will be done at the site.

Use a 3" to 4" pipe adapter to change the pipeline to the 4" size that will be interfaced to the vacuum truck.

Move the vacuum excavation line to the ground. Use the 4" hose coupler to secure the pipe to the vacuum nozzle. Avoid sharp bends to maintain optimal suction flow.

Conduct an inspection of the setup, checking that all safety controls, such as emergency shutoff switches and grounding cables, are in place and

team members for hand signals that represent actions the vacuum truck will need to take. In particular, an emergency stop hand signal must be clearly known.

Ensure that the area around the vacuum truck is completely clear and free of any obstacles and loose or flammable material.

functional.

Perform an operational test for ≈1 minute to confirm proper suction of the vacuum line before beginning full-scale extraction.

Connect the vacuum line with the pipes inside the TBM using couplers. This coupler will be removed often to add new pipe segments.

TBM System Checks:

1. Install a monitoring station with visibility of muck flow rate and vacuum pressure to ensure continuous operation from the pressure transducer and the vacuum truck measurement instruments.
2. Test emergency stop controls.
3. **Perform an operational test for ≈1 minute to confirm proper suction and operation of the entire vacuum system, identifying leaks and improper pipe couplings**

Chamber

The last element of our muck extraction system is the muck chamber. Our team took inspiration from industry-standard conical structures to ensure a continuous flow of muck, uniform pressure distribution, and optimized use of space. The tapered nature of a cone allows for an incline where gravity is the driving force of the flow of muck and water to the extraction line. This reduces material buildup and controls pressure release within the TBM.

In designing the conical structure for our application, we must consider the angle of repose required for smooth muck flow to the extraction line, the length of the chamber, and the manufacturing process. Due to ease of manufacturing, our team modified the conical shape to a hexahedron structure made of 6 welded steel plates. We plan to have thicker welds at the intersections of the plates to limit sharp edges and ensure that muck does not get stuck.

First, we determine the required chamber length. Our analysis assumes that we are extracting 90% of the muck that enters per minute (extraction efficiency) and that we are declogging our chamber

approximately every 0.5m or 5 minutes. This entails halting our advance while continuing our chamber conditioning and extraction processes until our extraction line pressure sensor reads atmospheric pressure (muck-free). Additionally, we assume that the volume of the chamber can be simplified to that of a cone with a volume correction factor of 1.2.

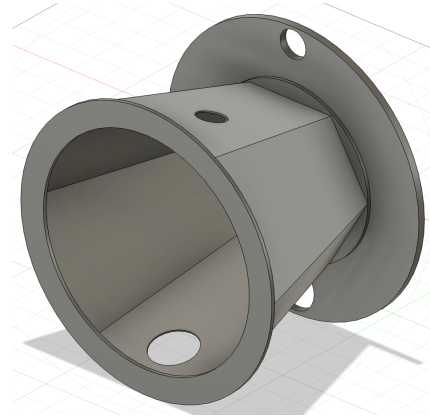
Description	Unit	Value
Soil Intake Capacity	gpm	3.63
Pressure Washer Flow Rate	gpm	10
Extraction Efficiency	-	0.9
Declogging Frequency	1/m	2
Max Volume of Muck + Water in Chamber	m ³	0.0258
Required Chamber Length	m	0.315

Now, we determine the required plate thickness to ensure bending does not occur. Bending is most likely to occur on the bottom-most plate of the muck chamber, as this is the plate that would experience the greatest loading from muck buildup. Gravity will naturally drive the flow of muck and water onto this plate. Given the previously determined chamber length and assuming a 25° angle of repose (with respect to the horizontal), we can determine the effective area of the plate. This angle of repose provides an incline for continuous flow.

The plate's width is constrained by the welding process. We want to avoid welding angles greater than 120 degrees, as those can be challenging to perform. Additionally, since one side of the muck chamber will be welded to an annular plate and then welded to the main TBM pipe, the inner diameter of the main pipe also restricts the plate's width. The same safety factor of 2 previously used regarding the mass of muck and water also applies.

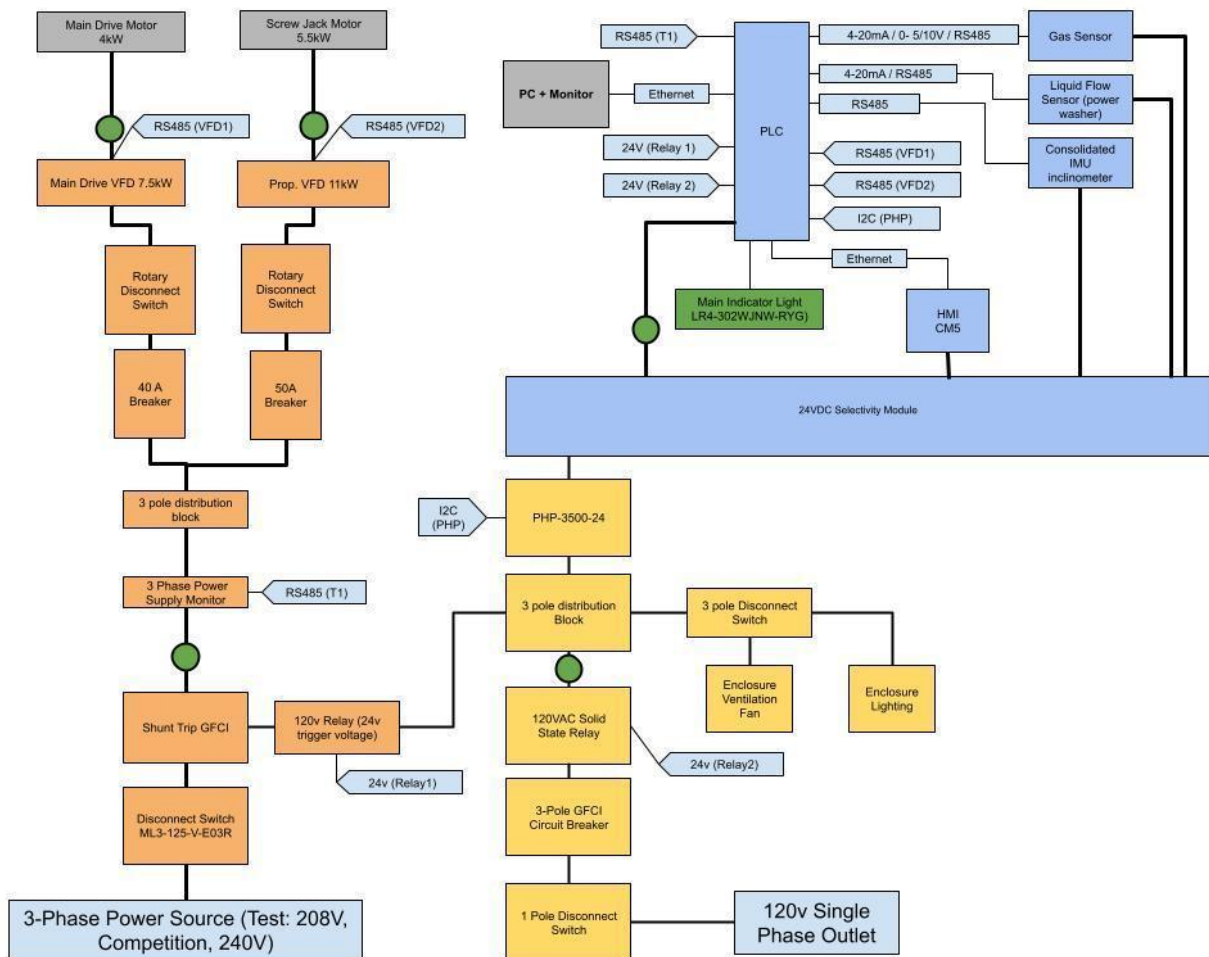
Description	Unit	Value
Water density	kg/m ³	997
Clay density	kg/m ³	2000
Max Mass of Muck + Water in Chamber	kg	32.62
Max Weight Force	N	319.95
Max Stress	Mpa	250
Elastic Modulus	Gpa	210
Plate Width	m	0.20
Plate length	m	0.311
Allowed Bending Displacement	m	0.001
Bending Stress Plate Thickness	in	0.193

The required plate thickness is approximately 0.2in. To the left is a model of our muck chamber, which has holes for the extraction line, the conditioning lines, and 2 annular plates for support. The chamber's dimensions reflect our analytical results, specifically the chamber length and plate thicknesses. The volume is optimized given the required chamber length and desired welding conditions



Electronics System

The electrical system in our TBM is designed to be safe while simple to assemble in an enclosure, test, and operate. Below is our one-line diagram, which highlights the general circuit layout.



Our system is composed of a high-power and low-power circuit. Our high power circuit will be powered from the generator at 240V, 3 phase power at 60 Hz. It will utilize 3 wires, with 2 lines and 1 neutral. This goes to a three-pole physical switch (the physical emergency stop), and to a shunt trip 100A-GFCI breaker. The shunt trip is controlled by a 120V relay that uses a 24V digital output from the PLC to trigger it. This structure allows us to combine the GFCI with a software-based emergency stop, and also removes the need for a solid-state relay on the high-power side to reduce thermal load in the enclosure. After that, we will install a power supply monitor, which gathers voltage and current data using transducers. This data will be communicated to the PLC through RS485. The electricity will then be distributed on a 3-phase distribution block into our motor circuits, which are protected by circuit breakers. We will have two 3-phase motors, one for the main drive, and one for the screw jack propulsion mechanism. Our main drive gearmotor is a 6-pole, 380V, 4kW motor that allows for VFD control. This is done by using a step-up VFD (240v -> 380v). Our propulsion system motor is 4-pole, but the circuit is the same.

On the low-power end, we will use a standard 120VAC, one phase outlet. This voltage will go into a disconnect switch for physical deenergization, GFCI for ground protection, and a solid state relay for software deenergization (a solid state relay can be used because of low power). After this, it's distributed to 1) our PHP-3500-24 power supply, 2) the enclosure ventilation fan, and 3) the relay that triggers the shunt trip for the high-power circuit.

The PHP-3500-24 power supply converts 120VAC to 24VDC, which goes through a selectivity module to provide each 24vdc circuit with overload and short protection. Our selectivity module has 8 outputs, which then connects to each of the DC components in the drawing.

The PLC is the central component of our system, and aggregates all information gathered from sensors to display them on HMI's and monitors, and also takes in commands from the operator of the TBM, adjusting signals sent to VFDs and other modules accordingly. Our sensors, as well as lighting and ventilation, also take in 24V. More information on our PLC system and communications can be found in the sections below.

Our electrical system will be housed in a used Rittal freestanding enclosure that we will purchase off of Ebay and refurbish.



Rittal TS 8005 Enclosure, 40" Width x 78" Height x 20" Depth, Two Door, Key Lock

Updated I/O List in the format specified in the Rules

Sensor Val	Min	Low Warn	High Warn	Max
Shunt Trip Status	N/A	N/A	N/A	N/A
VFD1(screw jack): Fault Code	N/A	N/A	N/A	N/A
VFD2 (main drive): Fault Code	N/A	N/A	N/A	N/A
VFD 1 / VFD 2 Run Command	0	N/A	N/A	1
VFD 1 / VFD 2 Reverse Command	0	N/A	N/A	1
VFD 1 / VFD 2 Running Status	0	N/A	N/A	1
VFD 1 / VFD 2 Emergency Stop	0	N/A	N/A	1
VFD 1 / VFD 2 Overload Warning	0	N/A	N/A	1
VFD 1 / VFD 2 Zero-Speed Indicator	0	N/A	N/A	1
PHP3500 Output Current (A)	0	0	20	140
PHP3500 Temperature (deg C)	-40	0	80	100
IMU Acceleration (X, Y, Z): g	-16	-0.00153	0.00153	16
IMU Gyroscope (ψ , θ , ϕ): $^{\circ}/s$	-2000	-5	5	2000
Inclinometer Z axis: $^{\circ}$	-180	0	2.5	180
Liquid Flow Sensor: m/s	0.5	1	4	5
Combustible Gas Sensor: %LEL	0	0	1	100
VFD1: AC Voltage (V)	0	360	390	400
VFD1: Output Current (A)	0	0	18	24
VFD1: Output Power (kW)	0	0	5.5	11
VFD1: Output Torque	-200%	0	100%	200%

VFD1: Set Frequency (Hz)	0	0	55	500
VFD1: Motor Temperature (°C)	N/A	N/A	110°C	130°C
VFD2: AC Voltage (V)	0	360	390	400
VFD2: Output Current (A)	0	0	11	15
VFD2: Output Power (kW)	0	0	4	7.5
VFD2: Output Torque	-200%	0	100%	200%
VFD2: Set Frequency (Hz)	0	0	55	500
VFD2: Motor Temperature (°C)	N/A	N/A	110°C	130°C

Electronic Control Units

Our only control unit on our machine is the Siemens S7-1200 1214C PLC. As shown in the one-line diagram, the PLC unit aggregates data sent in from each sensor, displays it on an HMI panel and a computer monitor, compares their current values with their expected ranges, and alerts the operator if there's a discrepancy. Based on the operator's input, the PLC reacts accordingly, such as adjusting the speed of the motor through the VFD or shutting down the TBM. The PLC system will also react automatically, such as turning the indicator lights on given current machine state (error, running, de-energized), or shutting down the system by the shunt-trip given unsafe conditions (like if methane detected, for example). The logic flow of our controls will be described in detail within the software section.

Communications Overview

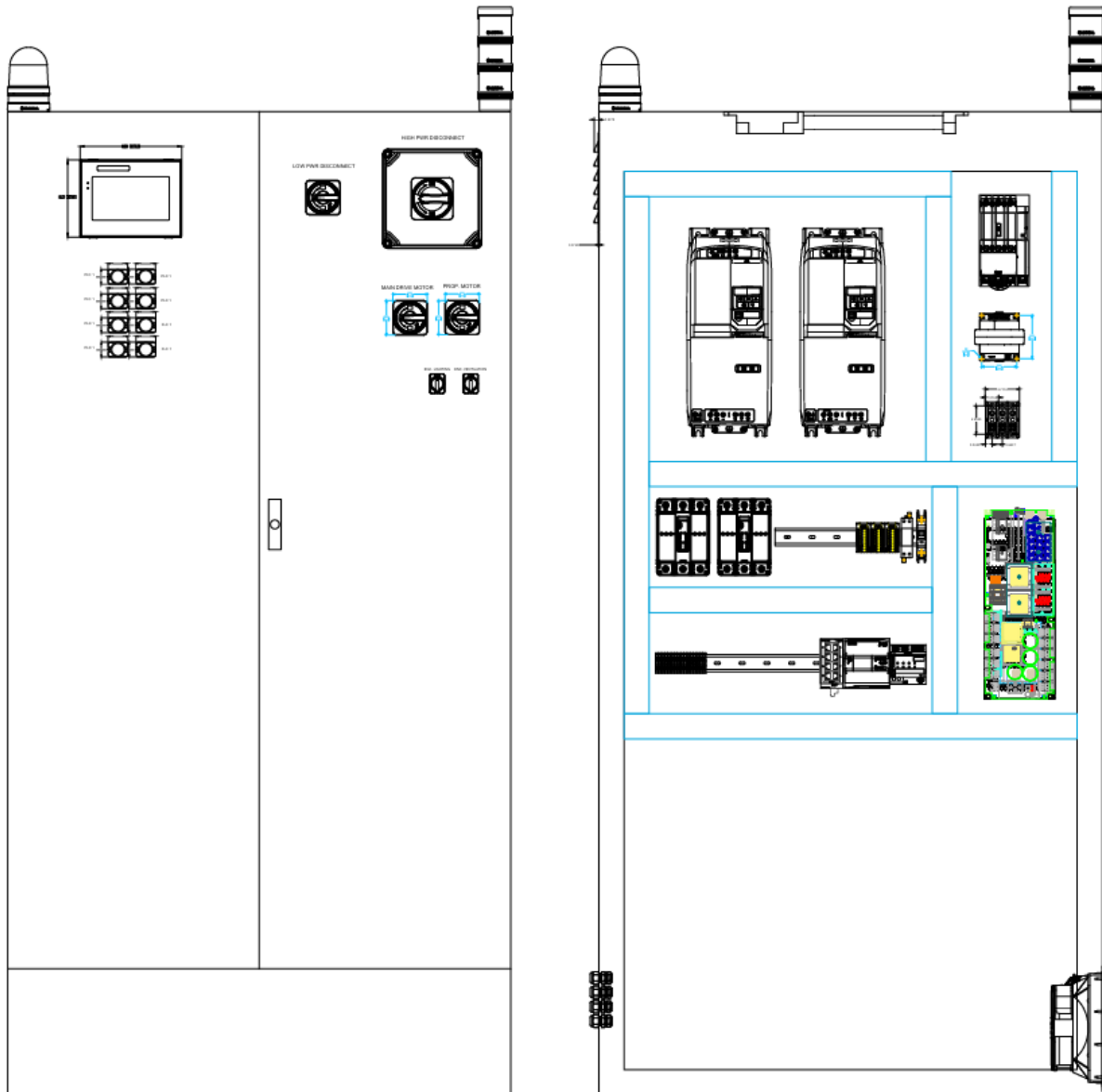
As shown in our one line diagram, most of our modules will be communicating with the PLC via ModBus RTU on RS485 interface. Because Siemens PLCs do not have pre-installed ModBus functionality, we will be using a multi-channel ModBus to Profinet converter, so that the RS485's 2 wire system (excluding power and ground) will be converted to ethernet that the Siemens PLC has natively. To obtain readings from our sensors, the Modbus utilizes a "call-and-response" mechanism, where the PLC sends specific registers of the devices, and the devices respond with data, usually in hexadecimal. We can easily convert this to base-10 and display on HMI and/or PC at the ground station. The only exception is getting readings from the PHP power supply, because it uses PMBus, a type of I2C. To connect that to PLC, we will use an I2C to RS485 adapter, and plug the adapter into our RS485 to Profinet converter, achieving communication even if the PLC does not come with I2C.

Power Consumption of Subsystems

Subsystem	Max Power (kW)
Main Drive	7.5
Propulsion	11
Muck Removal	0 (Vac-truck + gas pressure washer)
PLC, Circuits, Navigation	0.5
Total	19

Our main drive system uses one six-pole AC gearmotor that is powered and controlled by a 7.5kw VFD. Our propulsion system uses one four-pole AC gearmotor that is powered and controlled by an 11kw VFD. Our muck removal system is fully gas-powered, so there is zero power drawn there. The PLC, enclosure lighting, navigation, and other low-voltage loads are powered from a PHP-3500-24 power supply that outputs 24VDC. We estimate that on the 24VDC side, absolutely no more than 20A (480W) will be drawn. This is estimated from the total sum of devices connected, with a safety factor:

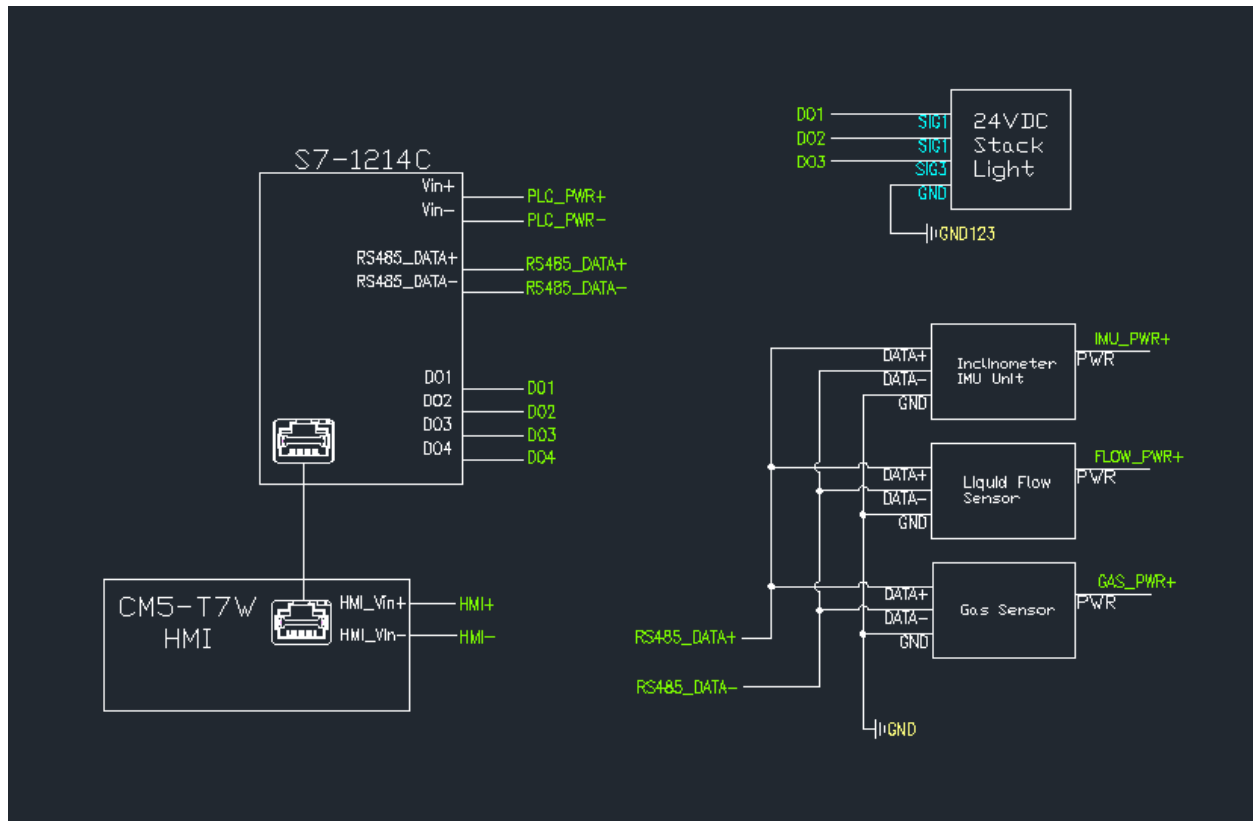
Device	Current (A)
PLC	12
HMI	0.42
Power Monitor	0.041667
IMU, inclinometer	0.04
Flow sensor	0.05
gas sensor	0.0375
lighting	0.208333
Total	12.7975
Total with Safety Margin	20

Schematic and Panel Drawing

The diagram illustrates a 3-phase power monitoring system. At the top, a 3p Power Monitor is connected to a 100A shunt and a 100A main disconnect. The monitor outputs RS485 data signals (RS485_DATA+, RS485_DATA-, RS485_DATA+, RS485_DATA-) to two ODE-3-420300-3F42 modules. Each module is connected to a 40A circuit breaker (CB) and a 40A main disconnect (Disc.). The modules also output signals (L1, L2, L3) to a 3-phase motor (MTR109).

[illegible]

24v PLC circuit



One thing we will absolutely be improving on post-FDP is the quality of our schematic and panel drawing. The reason it lacks detail at the moment is because we were unconfirmed for PLC sponsorship until yesterday (11/12), which is when we received word that Siemens will be donating a S7-1214C to us. Because of this, we did not have enough time to fully map all the signal lines for components like the VFDs. Nevertheless this is the first thing we will be revisiting in order to accurately model and plan out our system before we begin building.

In addition, we will be switching to using EPlan which is a software platform more suited to automation control panel and PLC circuit design. This will give us significantly more clarity about the components, connections, and layout before we begin assembly.

Software System

High-Level Software System

We will be using the Siemens PLC to interface with our TBM. The software system will input readings from our multiple sensors: power monitor, muck flow rate, GCS flow rate, and the consolidated IMU and inclinometer, as well as monitoring and control parameters from the VFDs. The operator will have the ability to adjust the TBM's actions based on the readings from the sensors during the IDLE state.

Our software system allows the operator to 1) adjust whether the machine is energized through a shunt trip, 2) control the speed and direction of the cutterhead, and 3) control the speed and direction of the screw jacks. Our PLC will be programmed with ladder logic and will incorporate safety interlocks that will automatically trigger the system to shut down if there are dangerous readings on our sensors.

We will have two means of HMI: the 7" panel mounted to the front of our enclosure and a computer. The 7" panel will display the machine's state diagram (including the active state), showing whether it is safe to open the enclosure and a software estop. The larger PC monitor will display the state diagram, full IO values (the table presented above), and motor controls. This dual HMI system will allow for full control of the system through the larger monitor, and add redundancy for safety of the operator should he/she decide to work on the machine.

Software Communications/Network Architecture

Our communication systems use Modbus RTU Protocol (RS485), Profinet (Ethernet), and I2C Protocol.

The RS485 protocol is used to communicate between the PLC and both of our VFDs (7.5kW Main Drive VFD and 11kW Propulsion VFD). We also use the RS485 protocol to communicate between the PLC and 1) our power supply monitor, 2) gas sensor, 3) liquid flow sensor, and 4) consolidated IMU inclinometer. Ethernet is used to communicate between the PLC and the HMI. The I2C protocol is used to communicate between the PLC and the PHP power supply.

E-stop Implementation Description

Our system has one physical E-stop and one software E-stop for each the high-power and low-power circuits.

For the high-voltage circuit, our physical E-stop is a 125A disconnect switch that will be mounted to the front of our electrical enclosure. The software E-stop mechanism is implemented with a shunt trip GFCI connected to a 120V relay with a 24V trigger voltage. The 120V relay operates on a 24V trigger voltage and acts as a central control point that can interrupt the power supply when triggered. The GFCI has a shunt trip mechanism that allows it to break the circuit remotely. When the software E-stop is activated on the GUI, it sends a signal through the 24V relay to trip the GFCI, effectively cutting off power to the system, ensuring an immediate shutdown of operations. Finally, the shunt trip GFCI is placed in line with the main 3-phase power source. By interrupting this, the entire system's power supply can be disconnected, halting power to the motors, VFDs, and other critical components, ensuring safety.

For the low-voltage circuit, we have a 40A disconnect switch, which is our physical E-stop, followed by a GFCI breaker, and a solid state relay, which is our software E-stop. This will allow the operator to de-energize the low-power system, either physically or over software.

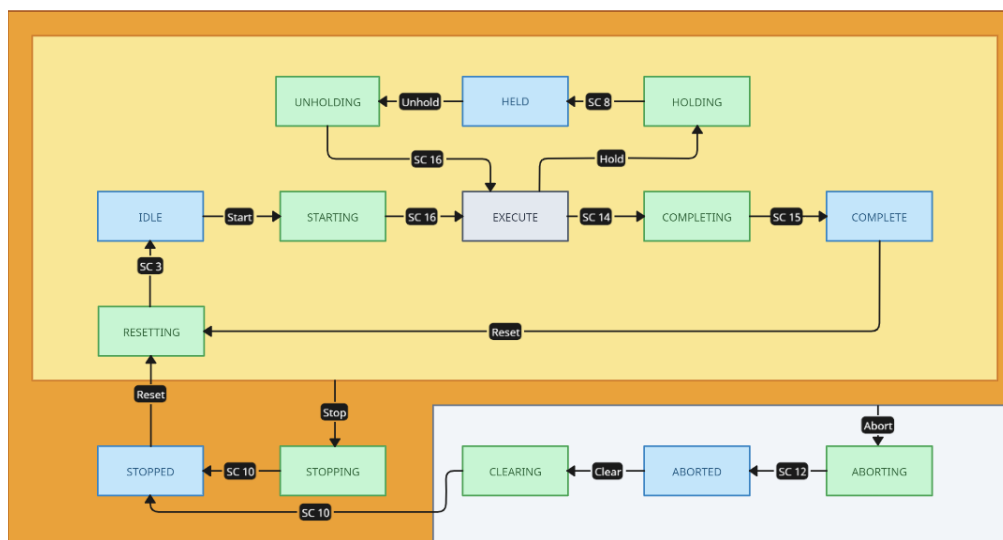
Machine Operator Controls and Interface

Safety Interlocks

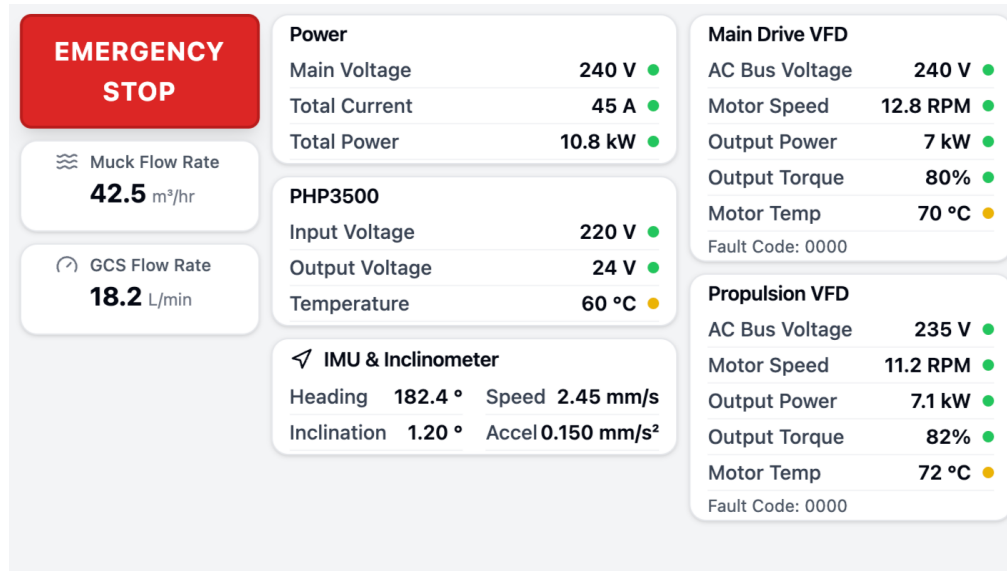
- Non-contact lock on the enclosure door
- High-power 125A disconnect switch
- Low-power 40A disconnect switch
- Independent motor disconnect switches
- High-power GFCI Shunt Trip breaker (software e-stop)
- Low-power GFCI breaker
- 120VAC Solid State Relay
- Physical Barriers to the launch pit area

Mock-Up of The Operator Interface

We will be incorporating data-feedback with state-diagram monitoring and control. This will allow us to control the current state of the machine, as well as view and adjust parameters over the course



of an operational cycle. We will be updating our software layout and UX in the future to optimize for simplicity and ease of programming.



Cooling System

There are two critical areas to consider for thermal management: the main drive system inside the TBM and the electrical enclosure.

TBM

According to our conversation with the manufacturer, given our use case, our motor does not require cooling systems to function because it has internal oils and fans. However, to be sure, we plan on running CFD simulations to investigate if forced convection would be required as we receive parts and conduct testing on heat generation of the motor (explained in the Test Plan section). If so, we can get a standalone, gas-powered fan with flexible ducting to move air through pipes to the back of the TBM where the motor is located, so the electronics do not have to be modified.

Electrical Enclosure

The thermal load for the electrical enclosure is managed by use of a ventilation fan and grill. The following steps were used to select the CFM of the fan:

We assume that the entirety of the thermal load generated inside the enclosure will be from the VFDs. This is realistic since, combined, they account for ~97% of power consumption.

Steps:

- Calculate the internal heat load, Q_i , which is the sum of all the heat generated by the components within the enclosure, in Watts

- Calculate heat transfer load, Q_x , which is the heat gained or lost through the enclosure exterior surface with the surrounding ambient air, in BTUs
 - $Q_x = kA\Delta T$, where
 - k is the heat transfer coefficient. For painted carbon steel, this is 0.97.
 - A is the surface area of the enclosure
 - $\Delta T = T_{\text{max ambient air temp}} - T_{\text{max allowable enclosure air temp}}$
- Convert Q_x to Watts
- Calculate required cooling capacity, $Q_r = Q_i + Q_x$ in Watts
- Calculate fan airflow rate $F_r = -(3.17 \text{ CFM} \cdot ^\circ\text{F/W}) Q_r / \Delta T$

Source: Automation Direct, [Enclosure Cooling Selection](#)

Enclosure Size:

Width	40"
Height	78"
Depth	20"

We assume an average daytime high of $27^\circ\text{C} \approx 81^\circ\text{F}$ in March/April in Bastrop TX. Our components are rated to a max temperature of $40^\circ\text{C} \approx 104^\circ\text{F}$, which is the desired steady state temperature with cooling.

Here is a table that shows the calculation for our fan's CFM with the above assumptions.

Qgenerated (W)	1500
K (Carbon Steel)	0.97
ΔT (F)	-23
A (in ²)	10960
A(ft ²)	76.11
Qbox (BTU)	-1698.0141
Qbox (W)	-497.5181313
Qfan (W)	1002.481869
CFM	138.1681532

For comparison, if no ventilation exists, then we can also calculate the expected steady state temperature assuming the heat taken away by the box is equal to the heat generated by the VFD's:

Q_i (W)	1500
Q_x (W)	-1500
Q_x (BTU)	-5084.745763
k	0.97
A (in ²)	10960
A (ft ²)	76.11
ΔT (F)	-68.87

This means that our enclosure, without cooling, our enclosure will reach 149°F≈65°C. This is a temperature that would even make the enclosure hot to touch, and exceeds the maximum temperature of our components significantly.

To avoid the situation above and corresponding to our calculations, we will be using a 155 CFM intake fan supplied by AutomationDirect alongside a similarly-sized grill for outflow. Our VFDs have integrated fans which should increase the efficiency of our enclosure cooling system.

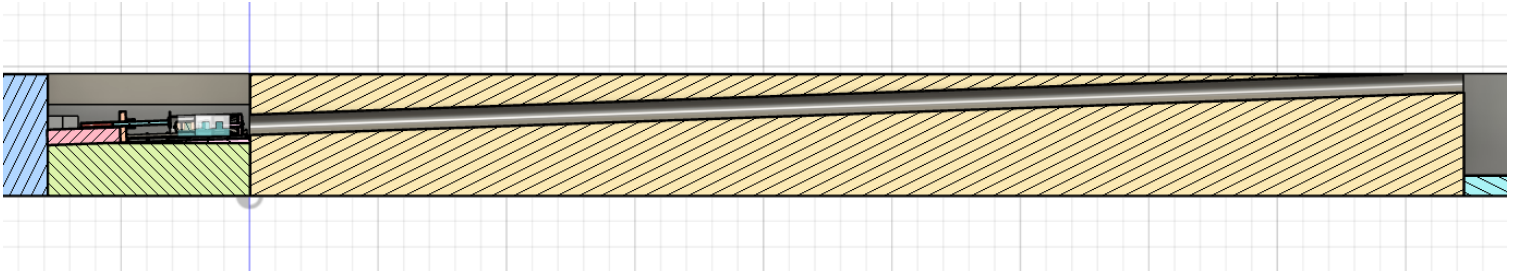


TBM Navigation Mechanism and Alignment Strategy

3D Tunnel Path

Our TBM will aim to dig a path at a 2.5° inclination upward. If no noticeable sinkage occurs, then the resultant path would look similar to that of the image below. We will dig approximately 29m in this idealized case.

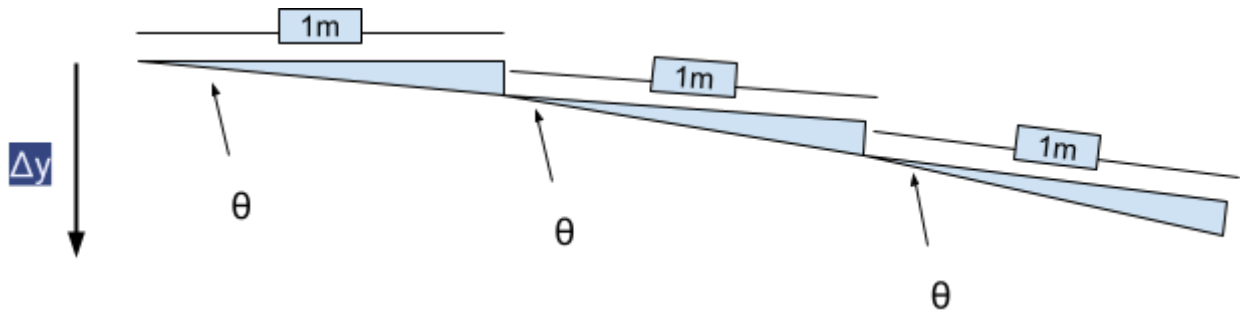
Ideal Scenario: 0° Sinkage



This was calculated using the formula:

$$\int_0^x \tan\left(\frac{(2.5 - 0n)\pi}{180}\right) dn$$

where x represents the horizontal distance traveled by the TBM, and n represents the “degree sinkage per meter” the TBM experiences. This equation is formed by drawing multiple right triangles (where the length is 1m and the height is the coefficient in front of n) that are “stacked” point to point.



For example, if the TBM were to sink an average of 1 cm for every meter it travels horizontally, its “degree sinkage” per meter would be

$$\arctan\left(\frac{0.01}{1}\right) = 0.573^\circ/\text{meter}$$

And our TBM is expected to vertically sink:

$$\int_0^x \tan\left((2.5 - 0.573n) \cdot \frac{\pi}{180}\right) dn$$

$$= -3.22\text{m}$$

Uncertainty Calculations

The following calculations were made for resultant depth due to the following TBM sinkage per meter dug horizontally:



- No sinkage (green)
 - Escape depth: +1.3m
- 0.5 cm sinkage per meter (orange)
 - Escape depth: -0.93m
- 1 cm sinkage per meter (purple)
 - Escape depth: -3.23m
- 2 cm sinkage per meter (red)
 - Escape depth: -8.13m
- 3 cm sinkage per meter (blue)
 - Escape depth: -14.07m

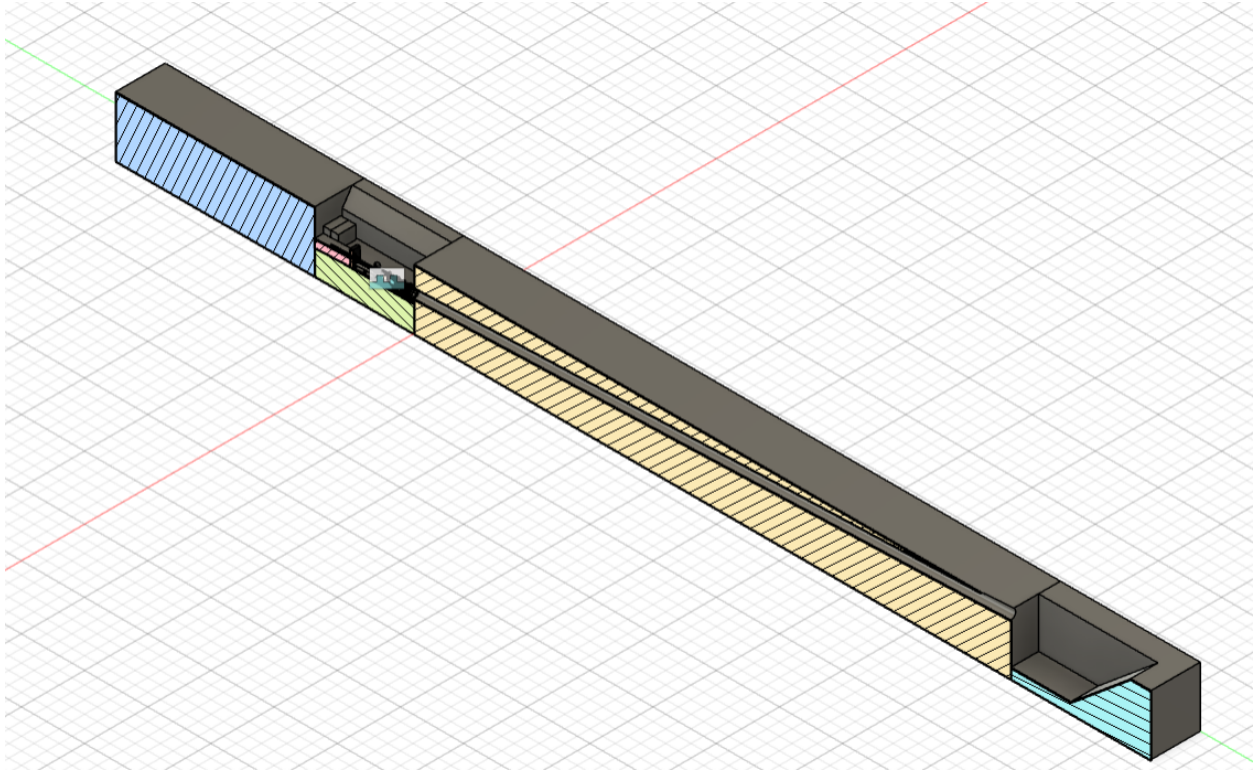
As we further explore what the true expected value is for TBM sinkage per horizontal meter, we can further optimize our initial inclination angle to create a more optimal path for TBM excavation. We currently expect there to be very minimal sinkage, with the orange or purple path (see graph above) being the most realistic occurrence.

Estimate of Error in Breakthrough Location

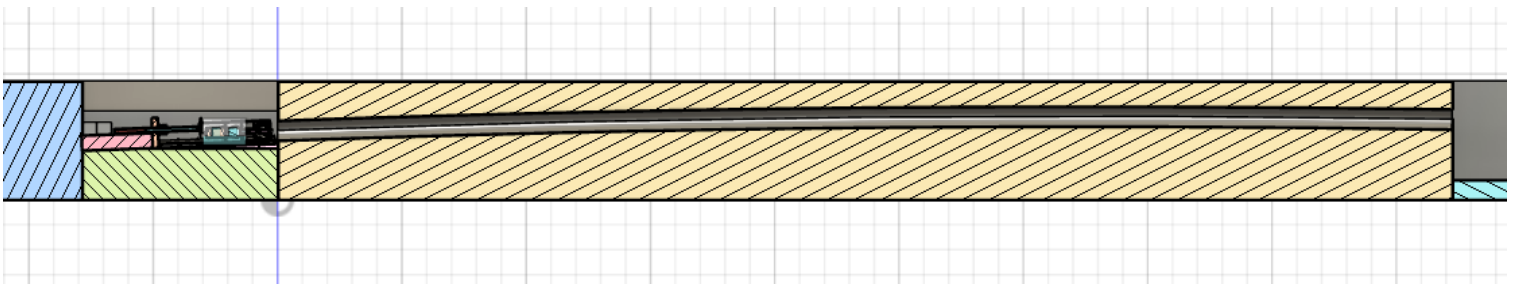
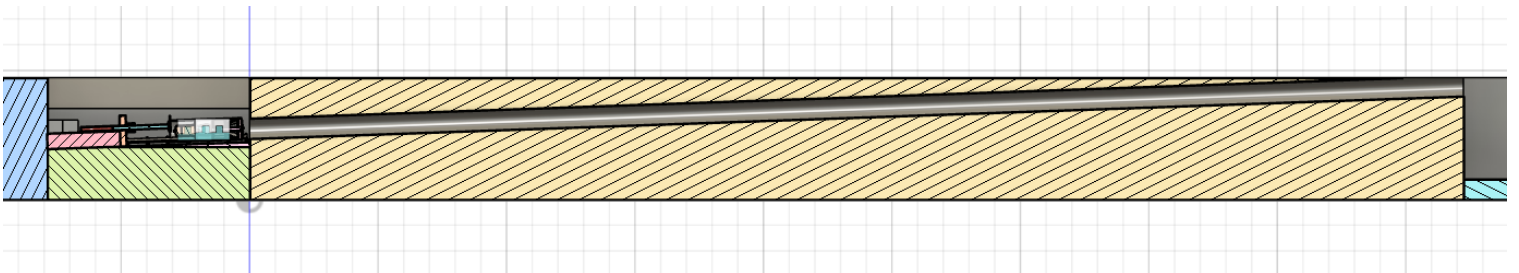
Seeing as our starting excavation depth will be approximately 1.5m below the surface, the estimated breakthrough location ranges from being 0.2m below surface to 14m below the surface, depending on the amount of sinkage experienced by the TBM. The most likely scenario appears to be a breakthrough altitude of approximately 3m below the surface, which should be easily retrievable with an excavator.

We are not confident in the accuracy of our sinking calculations. The tendency of the TBM to sink into the ground is caused by the overcut from the cutterhead, which in our case is 0.05m total. Assuming that the soil on the bottom of the tunnel does not “refill” this overcut, the bottom of the TBM structure will not be supported from the ground. This gravitational force will cause a net force downward on the front of the machine. However, the pipe segments behind the TBM will have a tendency to maintain linear rigidity due to the friction between them and the tunnel wall, along with the compressive force being applied from the propulsion system. The interplay between these two phenomena makes this a challenging problem to model and understand, which is why we are not confident that our (very simple) model is correct.

In order to mitigate this risk and ensure that our TBM will not sink to the point where we cannot retrieve it, we will be 1) discussing this topic more with our advisors to better understand how they model this in industry, and 2) conducting a short test dig in late January to measure the rate and sinkage and adapt our system based on our findings. The length of this test dig will be ~5m. Throughout the course of this, we will be measuring the inclination angle of the machine in order to calculate the rate of change of static inclination. With this rate of change, we will be able to extrapolate it to the longer 30m dig to compute our expected vertical displacement during competition.



Ideal Scenario: 0° Sinkage



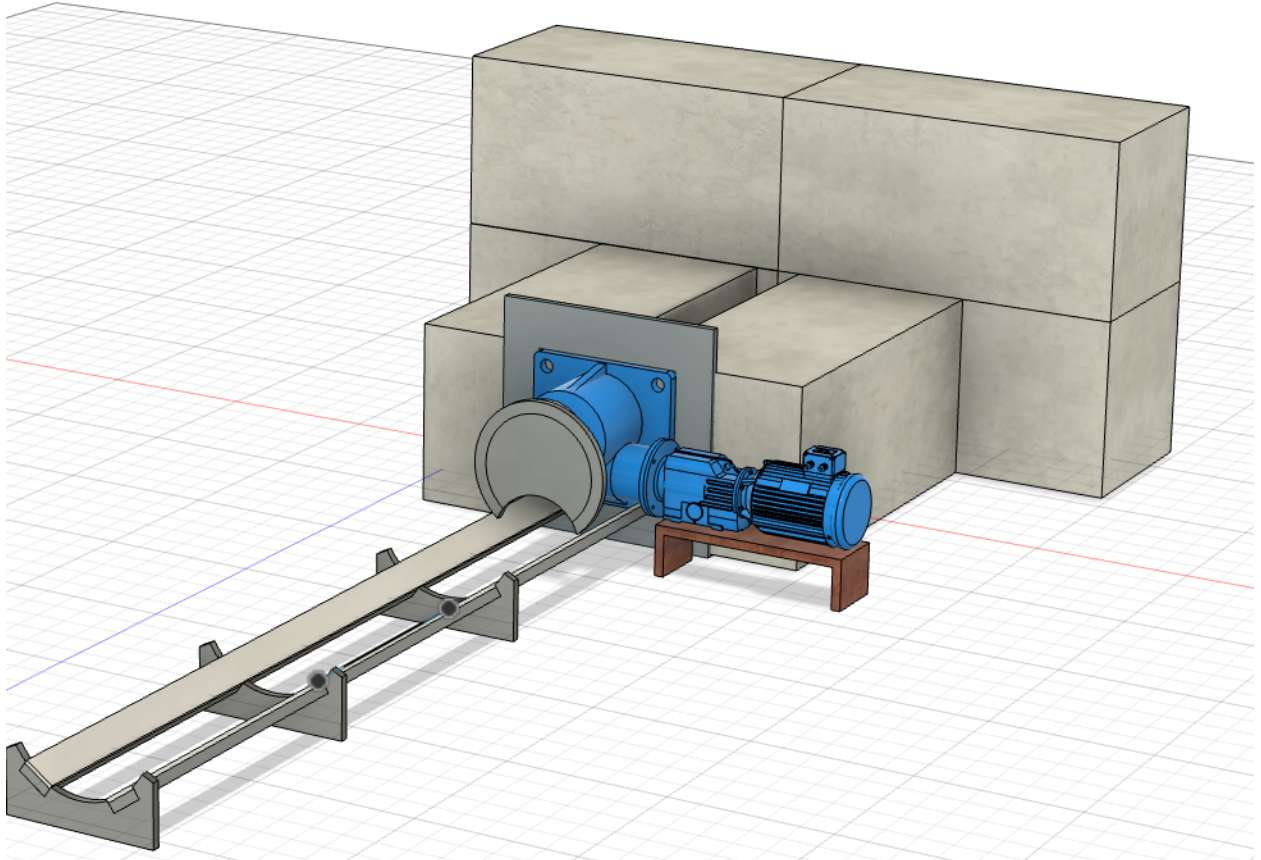
Realistic Scenario: $\sim 1.45^\circ$ average sinkage

Maximum Tolerable Scenario: $\sim 5.44^\circ$ average sinkage

TBM Launch Structure Analysis

Components of Launch Structure:

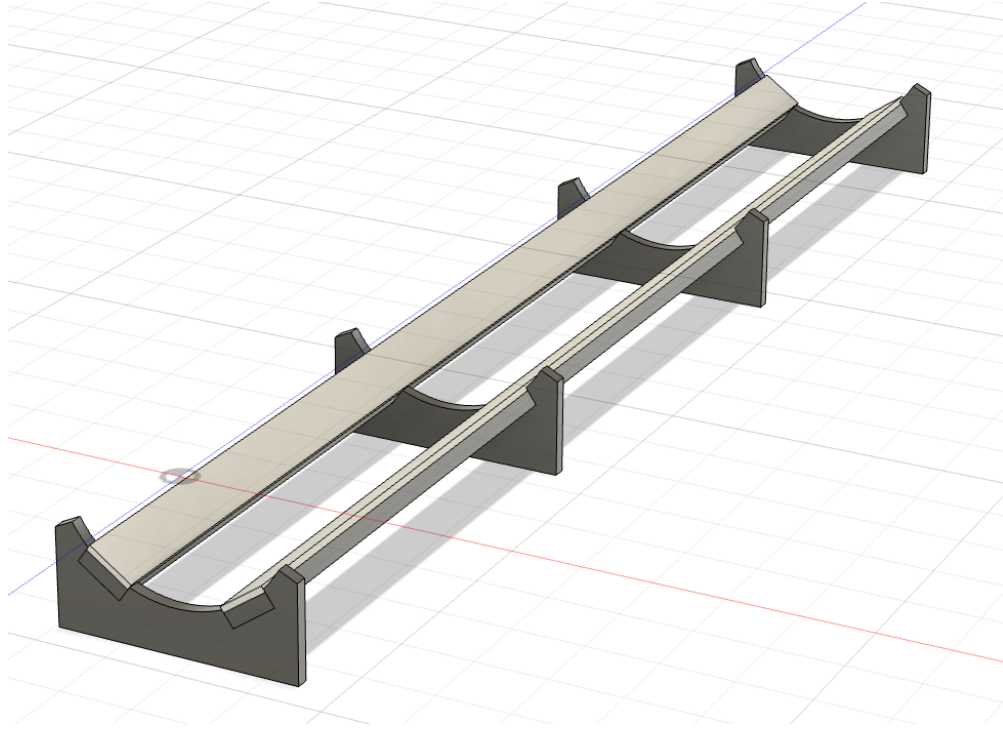
Below is a comprehensive list of all components that will make up our launch structure, which we will follow up with detailed structural analysis and sourcing plans.



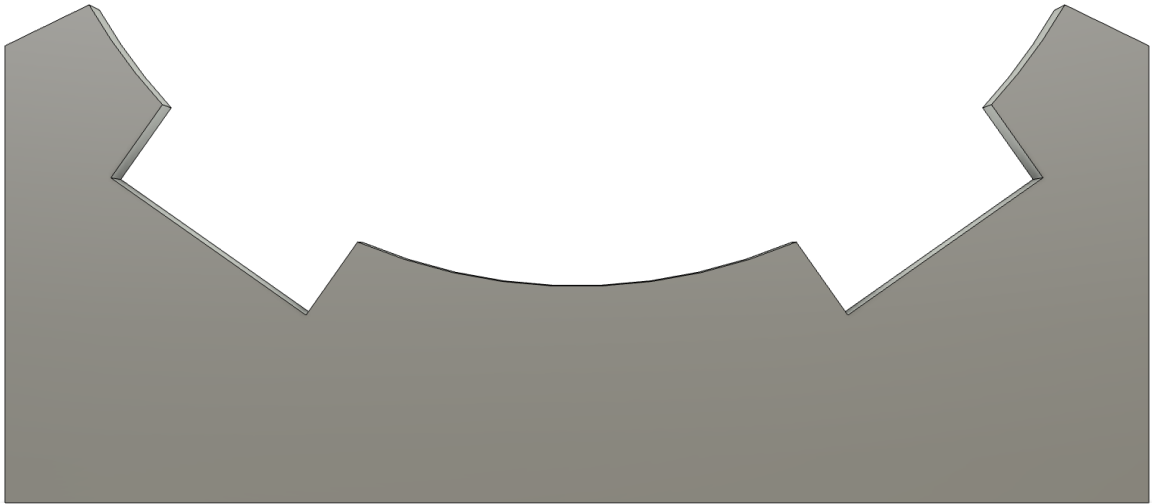
1. Launch Railing:
 - a. Periodic Structural Supports for railing
 - b. Horizontal Metal Sheets
 - c. PTFE Pads
2. Thrust Plate
3. Screwjack (detailed above)
4. Back Support plate
 - a. L-bracket supports for back plate
5. Concrete Blocks

Launch Railing Analysis:

Here is the design of our launch railing as a whole:



Component A: Periodic Supports



- Dimensions: 486.79 mm x 212.14 mm x 25.40 mm, where the circle diameter for support is 479.42 mm and rectangular cutouts are 38.10mm x 101.60 mm.
- Material Selection: AISI 4130 Steel, with a yield strength of 435 MPa.
- Expected Loads: Weight of the TBM applied downwards on the supports, concentrated at rectangular cutout areas.

Assuming uniform normal stress concentrations applied on those areas:

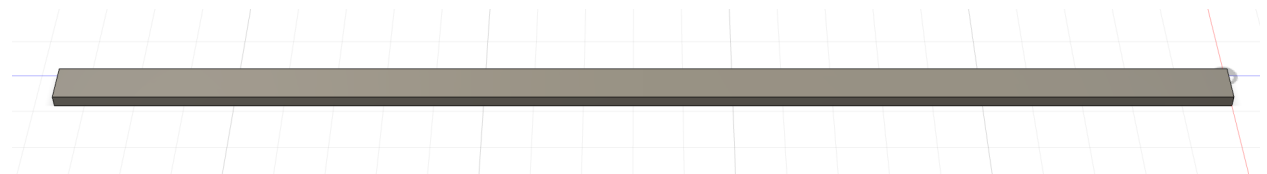
$$\sigma = \frac{F}{A} = \frac{249.48 * 9.81}{0.1016 * 0.0254} = 948369 \text{ Pa} \approx 0.95 \text{ MPa}$$

As an extension, since loads are mostly applied at the metal sheets that run across the periodic supports we can assume they are point loads, and these loads cause the sheets to bend, which when welded and attached to the periodic supports, cause a bending moment in the periodic supports.

The moment created by the force in between two supports is counteracted by the moments at either periodic support. Thus, the maximum moment on either periodic support will be half the moment of the point load exactly in between the two supports, and the bending stress can be calculated as a cantilever beam by $\sigma = -\frac{My}{I}$, and here we will take the point that the moment is being reacted at as the lowest point of the rectangular cutout since this is the shortest possible moment arm, making it a conservative estimate. The force of the weight of the TBM is also split into 2 by the two sheets that it is supported on.

Inputs	Value
Weight of TBM (N)	2447.4
Distance between supports (m)	1
Thickness of support (mm)	25.40
Height to lowest point at cutout (mm)	79.98
Equation Parameters	Value
M (Nm)	305.93
y (m)	0.0127
I (m ⁴)	1.365 x 10 ⁻⁸
Equation Parameters	Value
M (Nm)	305.93

Component B: Horizontal Metal Sheets



- Dimensions: 3000 mm x 100 mm x 19.05 mm
- Material: Mild Steel, with a yield strength of 200 MPa.
- Expected Loads: Weight of the TBM applied on the metal sheets as the TBM slides over the supports. Since the sheet has supports along it, its stress profile can be characterized as bending of beam with supported ends with a point load at the center (for a worst case scenario calculation) where $\sigma = -\frac{My}{I}$, and $M = \frac{FL}{4}$, where F is half the weight of the TBM since there are 2 sheets supporting the TBM.

Inputs	Value
Weight of TBM (N)	2447.4
Distance between supports (m)	1
Thickness of sheet (mm)	19.05
Equation Parameters	Value
M (Nm)	305.93
y (m)	0.009525
I (m ⁴)	5.761 x 10 ⁻⁷
Output Bending Stress σ (MPa)	5.058

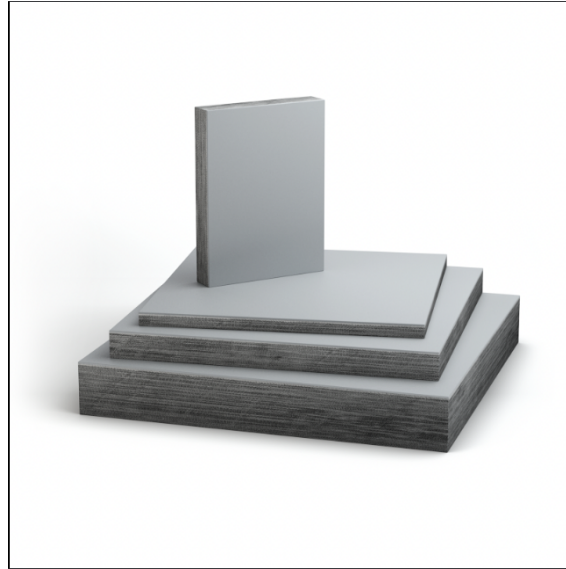
Further, the deflection of the sheet at its center can also be as $\delta = \frac{FL^3}{48EI}$, giving us:

Equation Parameters	Value
F (N)	1223.7
L (m)	1
E (GPa)	210
I (m ⁴)	5.761 x 10 ⁻⁷
Output Deflection δ (mm)	0.211

Thus, in this case, the limiting factor is our beam deflection calculation, which we want to minimize as much as possible, and given that our TBM is not actually a point load, our railing should perform even better in the given load case

Component C: PTFE Pads

- Source: Fabreeka International
- Material: Fabreeka PTFE, Teflon filled base
- Dimensions: 2 strips of 3000mm x 100mm x 6.35 mm

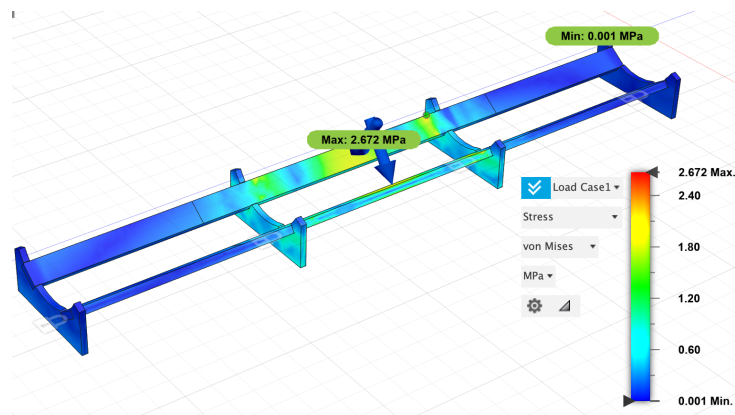


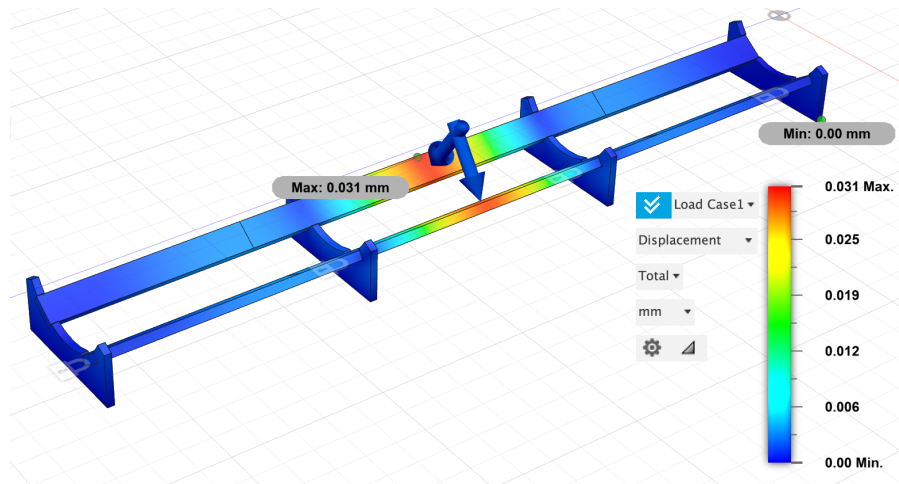
Bonded to the metal sheets using #700 Epoxy Adhesive provided by Fabreeka, which is reported to have a high bond shear strength, for which we are yet to receive specific literature from Fabreeka.

PTFE Pads from Fabreeka are rated for normal stresses far greater than what is expected by the TBM sliding over the launch railing, as tested by Fabreeka, and the PTFE is supported by the metal sheets to limit any large deformations.

Simulations for Launch Structure:

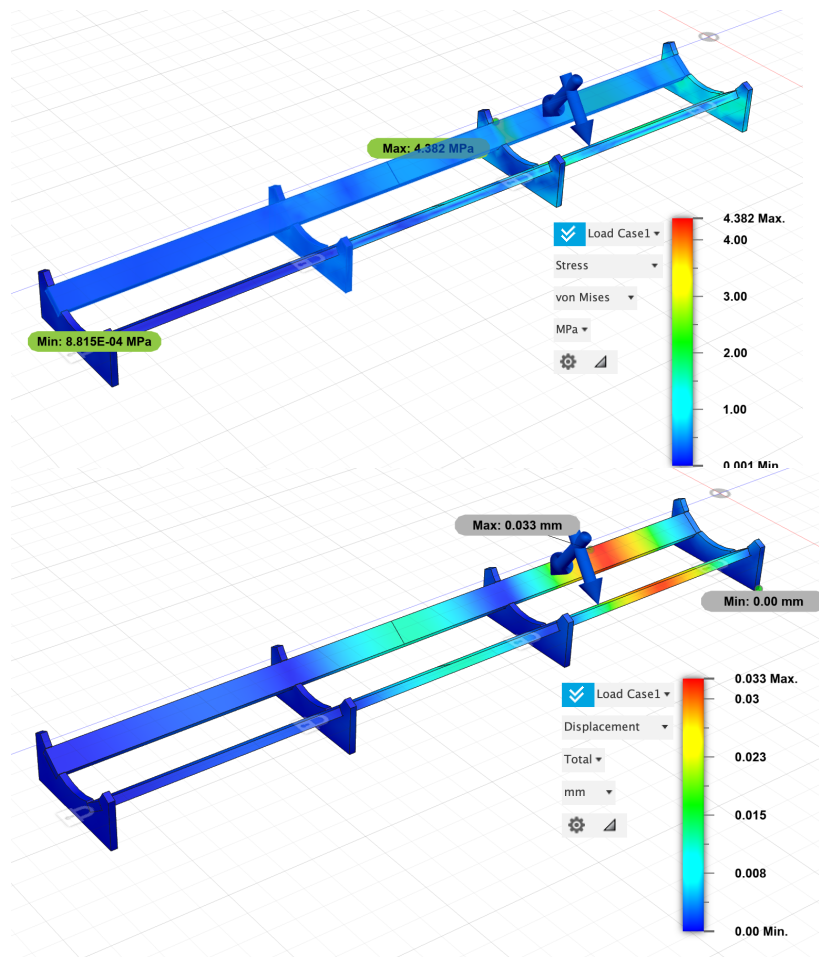
Load Case Set-up: Weight of the TBM Applied to the central part of the launch structure over the length of TBM (1.5m). From this, we can see the stress concentrations along the metal sheets and the periodic supports, and also the max deflection at any point along the metal sheets. Here are the results:





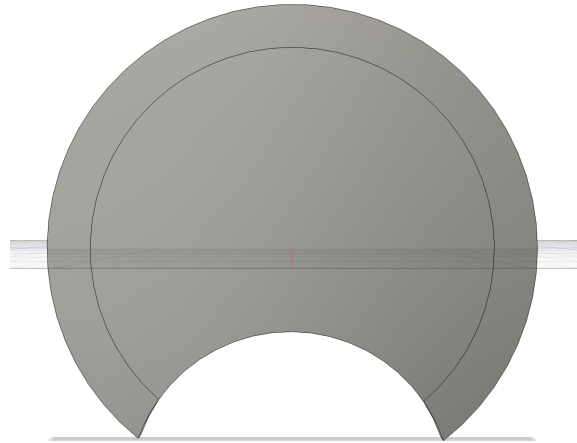
The minimum safety factor found for the model was also **15+**, which makes the model more than safe at every point. The maximum stress and deflection are also less than we had calculated, which is expected given the distributed load.

Just for a verification, we also ran a simulation where the load was applied to one side of the launch railing, rather than at the center area of the launch railing, for the length of the TBM. for



As shown, the stresses get more concentrated and the deflection rises by 0.002 mm when the load moves to the end of the launch railing. However, the minimum safety factor is still **15+** for the entirety of the launch railing, making it a very safe design for our load case.

Thrust Plate Design:



- Dimensions: 462.28 mm OD, 25.40 mm thickness
- Material Selection: AISI 4130 Steel, with a yield strength of 435 MPa.
- Expected Loads: The force applied by the screwjack at the center of the plate and fixed along the edges until the inner diameter of the pipes at about 381 mm. This creates an approximation of a bending moment about the center of the plate, from which we can use the same bending calculations as above and simplify the center cross-sectional area of the plate as almost like a simple rectangular plate area in bending to approximate the stresses near the center, which will be the highest using $\sigma = -\frac{My}{I}$, where $M = \frac{FL}{8}$ this time as we assume the sides are fixed, not just supported.

Inputs	Value
Max. force of screwjack(N)	300000
Distance to side support (mm)	190.50
Thickness of plate (mm)	25.40
Diameter of plate (mm)	462.28
Equation Parameters	Value
M (Nm)	28575
y (m)	0.0127

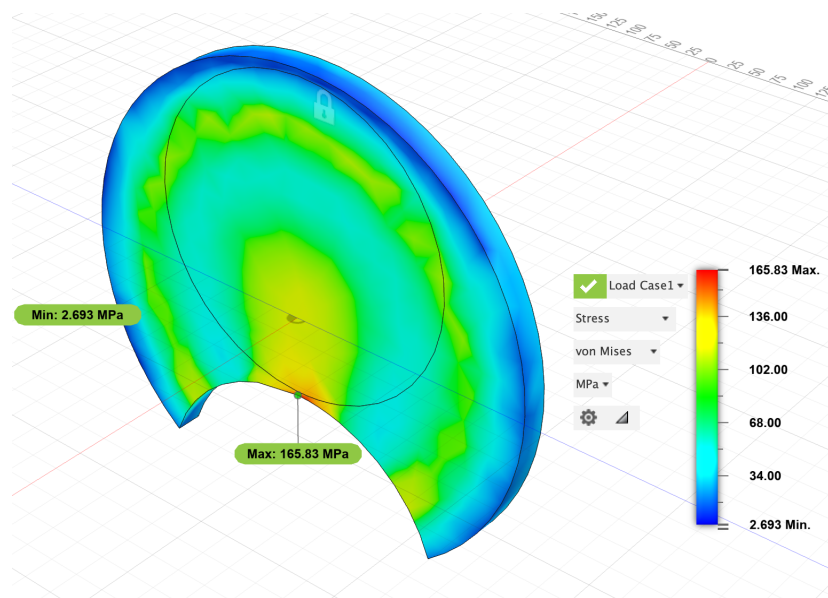
$I \text{ (m}^4\text{)}$	6.31×10^{-7}
Output Bending Stress σ (MPa)	287.4
Factor of Safety	1.51

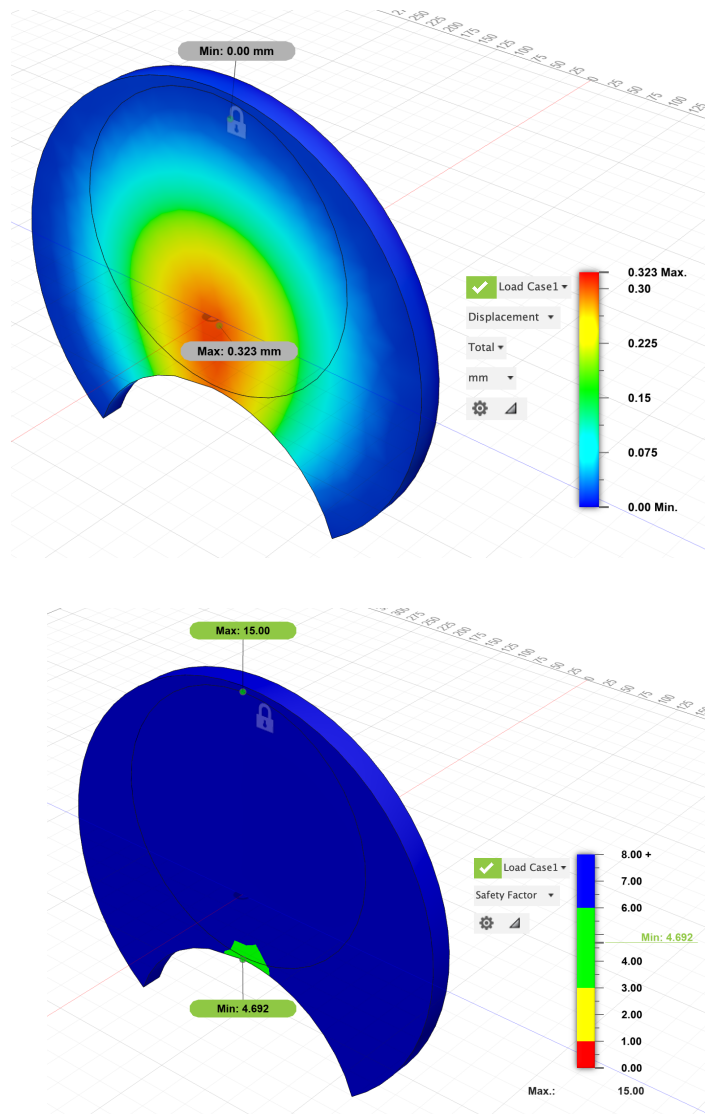
This is still a very conservative estimate for the stresses the thrust plate will experience as our screwjack will not be delivering the force at exactly a point, but rather at a flange plate that encompasses a larger area and distributes the load more evenly, which reduces the bending moment significantly.

Simulation:

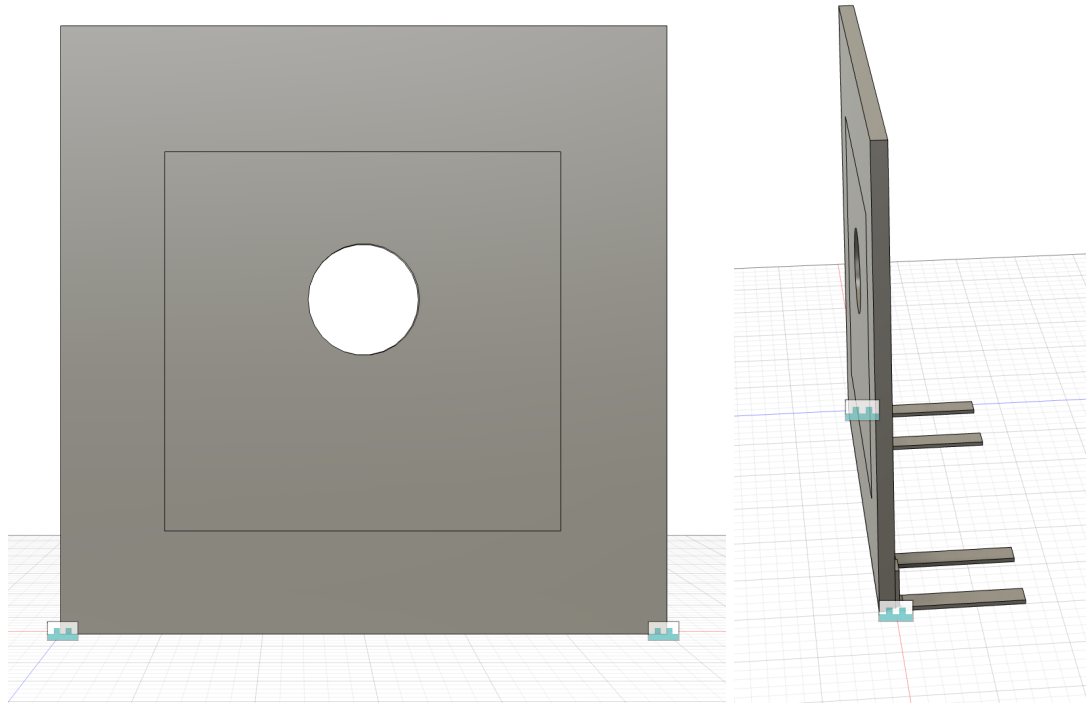
Load Case Set-up: Similar to the calculation load case, the outer sides of one side of the plate are fixed, as they would be to the walls of the pipe they are pushing. There would also be a circular area on the other side of the thrust plate where a 300 kN force is applied simulating the screwjack push and the area over which it pushes.

Results:





The simulation shows that our predicted minimum safety factor is even higher than we had calculated, now showing that it is safe at about **4.69**. The expected maximum stress is also less than what was calculated, which is to be expected as the simulation better depicts the distribution of the load over the screw jack flange, whereas the hand calculation assumed a point load. Finally, the deflection is also just 0.323 mm at most.

Back Plate Design:

- Dimensions: 762 mm x 762mm x 25.40 mm, with a 140mm diameter hole in the center for screwjack.
- Material Selection: AISI 4130 Steel with a yield strength of 435 MPa.
- Expected Loads: The back plate will be responsible for reacting to the loads from the screwjack and transferring it to the concrete blocks. However, since the blocks are not directly behind the line of action of the screw jack, and there is a bit of space between the blocks because of the casing of the screw from the screwjack, there is a small amount of bending that is the primary failure mode with this component.

Here we can assume that it is a fixed support since the concrete blocks will be placed on the L-brackets, causing the back plate's sides to be fixed in place by the concrete blocks. Thus, we can use the bending equation $\sigma = -\frac{My}{I}$, where $M = \frac{FL}{8}$.

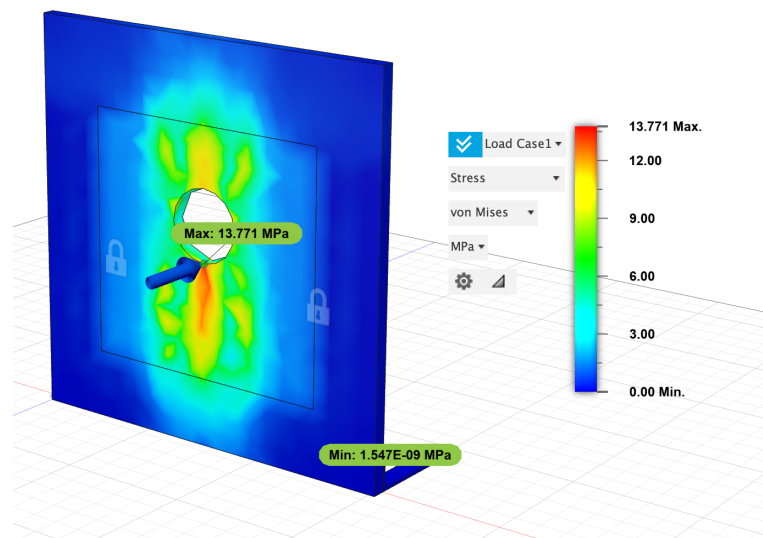
Inputs	Value
Max. force of screwjack(N)	300000
Distance between blocks (mm)	178
Thickness of plate (mm)	25.40

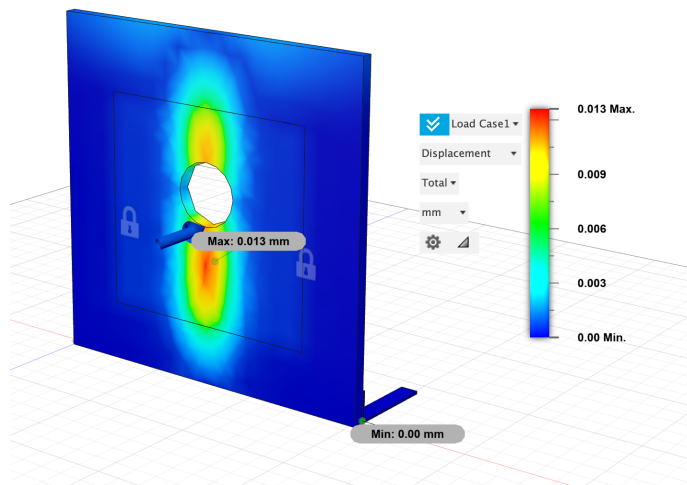
Base width of plate (mm)	762
Equation Parameters	Value
M (Nm)	6675
y (m)	0.0127
I (m ⁴)	1.041 x 10 ⁻⁶
Output Bending Stress σ (MPa)	81.5
Factor of Safety	5.34

Simulation:

Load Case Set-up: The force of the screwjack is applied on one side of the plate along the rectangle of the flange base area of the screw jack. The back of the plate is fixed on two sides as a representation of the blocks, with a space in between for the screw casing to stick out.

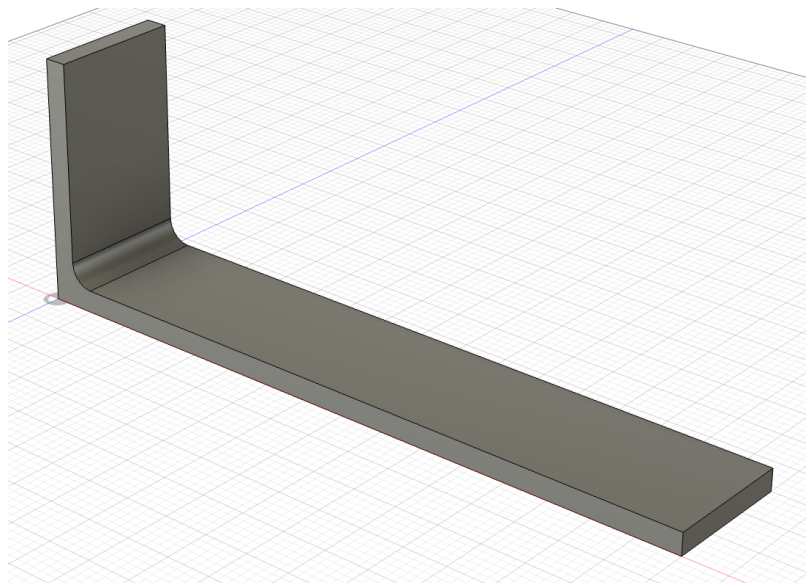
Results:





The safety factor found by the simulation was **15+** at every point along the back plate, making it a very secure design. The maximum stress on the back plate is also less than calculated, which is expected given we had assumed a point load, whereas the simulation distributes the load more evenly over the flange base of the screw jack. Finally, the deflection is only 0.013mm at most, which is acceptable for our design.

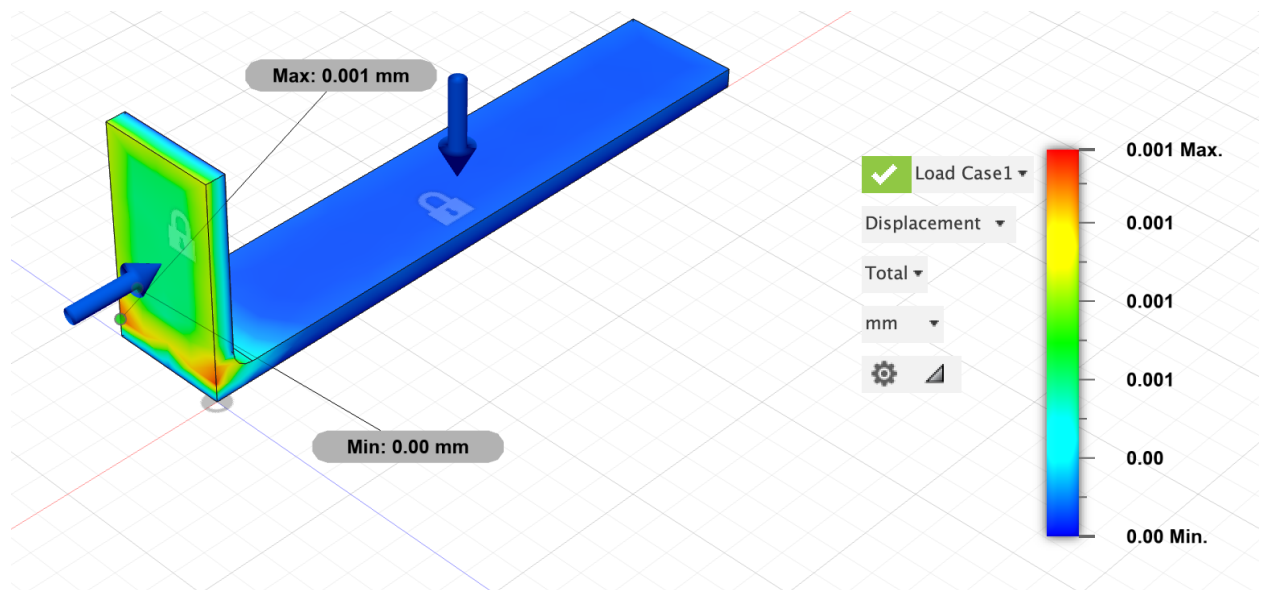
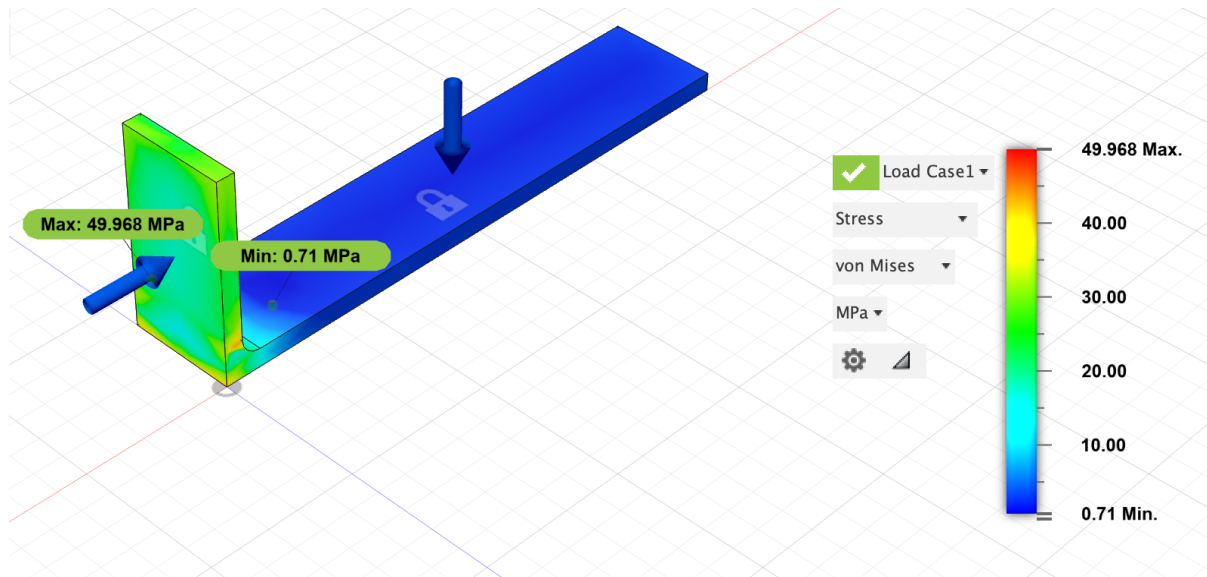
L-Bracket Design

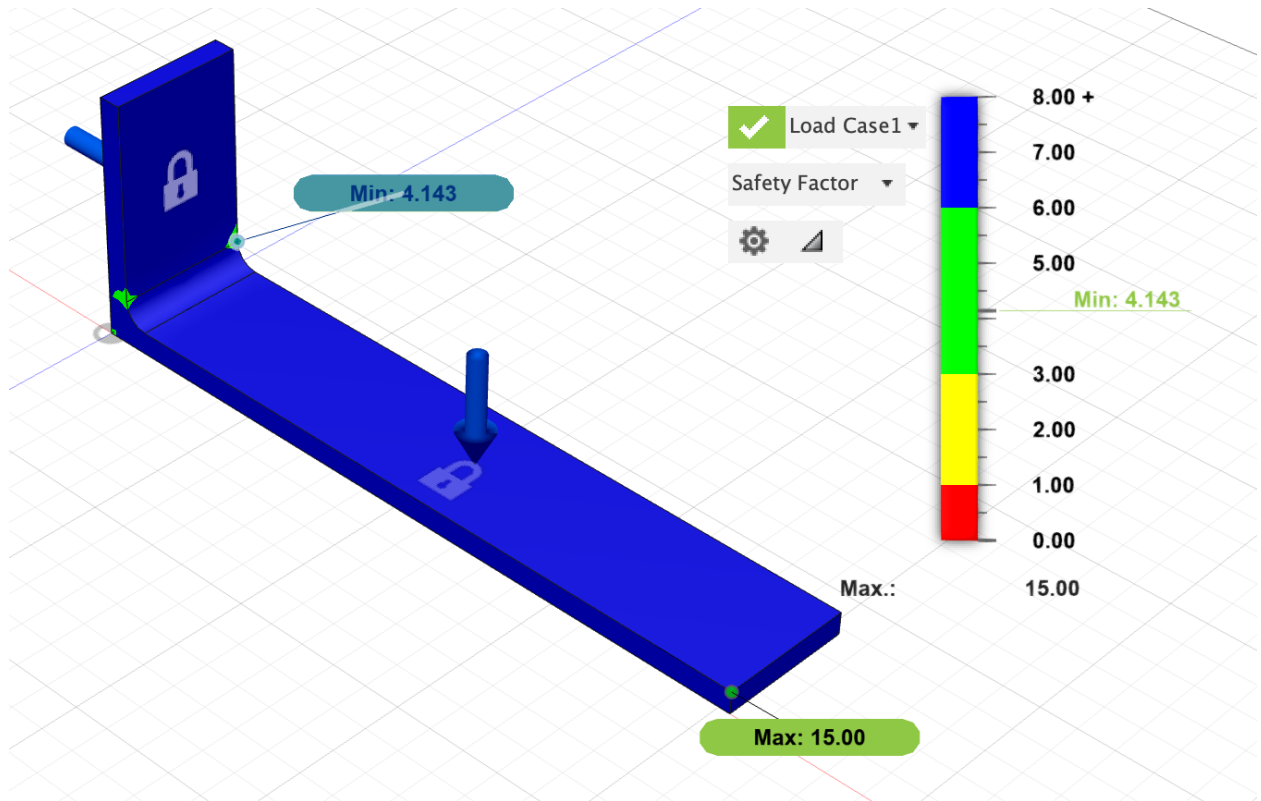


Simulation:

Load Case Set-up: The outer face of the short-side of the L-bracket is applied with the force of the screw jack divided by 4 (since there are 4 L-brackets transferring this force to the concrete blocks. The other face of the short side is fixed against the concrete block. The inner face of the long side of the L-bracket is applied with the weight of the concrete block (as a worst case scenario if only one L-bracket is supporting the weight of the block) and the outer face of the long side is fixed at the ground.

Results:





The minimum safety factor found given the load case is about 4.14 at the corner of the L-bracket, which is expected to have the highest stress concentration, which we see is about 50 MPa from the first picture. This shows that the L-brackets are more than capable of taking the loads and being safe. The maximum displacement is also just 0.001 mm, which is almost negligible.

Bolt Selection:

The screw jack will be connected to the thrust plate and back plate by M12 Class 8.8 bolts (at minimum), where there will be 4 bolts on the thrust plate and 4 bolts on the back plate.

The first failure mode here would be the bolts breaking in shear. Shear capacity of a bolt, which in this case is the weight each bolt can carry, is calculated as $\frac{0.6 * F_{UB} * A}{Safety Factor}$.

Inputs	Value
Ultimate Strength, F_{UB} (MPa)	640
Tensile Area (mm ²)	84.3
Safety Factor	2
Output Shear Capacity (kN)	16.19

This means 8 bolts are capable of carrying ~ 130 kN of shear force in total, which is about 13 tons, even with an overall safety factor of 2. Thus, given our screwjack will be in the range of 100 - 200 kg, we can say that the bolts are safe. Now, to ensure that the bolts don't tear out of the plates they rest in, we need to run a bearing capacity calculation, which calculates the force a plate can withstand from the bolt before tearing, using $\frac{F_{UB} * d * t}{Safety Factor}$. Since the thrust and back plates are both 25.4 mm in thickness and are made of AISI 4130 Steel, we can use the same calculation to solve for the bearing capacity of the plates:

Inputs	Value
Ultimate Strength, F_{UB} (MPa)	435
Bolt diameter, d (mm)	12
Plate thickness, t (mm)	25.4
Safety Factor	2
Output Bearing Capacity (kN)	66.29

Thus, again, each hole can bear up to about 6.63 tons of mass, making it an overall 53 tons that can be equally distributed by the 8 bolt holes, and it will still be able to withstand it with a factor of safety of 2. Thus, M12 Class 8.8 bolts work perfectly for our application.

Concrete Blocks:

Initial Setup:

We are using 6 concrete blocks, given with dimensions 48 in x 24 in x 24 in and a weight of 10.88 kN (using the density of concrete).

These blocks are arranged in 3 layers:

- Layer 1: 2 blocks horizontally with their square faces (24 in x 24 in) against the steel back plate.
- Layer 2: 2 blocks horizontally behind Layer 1, with their long faces (48 in x 24 in) against the blocks in Layer 1, and then 2 more blocks stacked on top of them.

Frictional Force Calculation:

The total weight of all 6 blocks is:

$$\text{Total weight} = 6 \times 10.88 \text{ kN} = 65.28 \text{ kN}$$

Using a conservative friction coefficient of 0.4 for concrete on clay loam soil:

$$\text{Frictional force} = 0.4 \times 65.28 \text{ kN} = \mathbf{26.112 \text{ kN}}$$

Passive Earth Pressure Calculation:

The final layer of 4 blocks interacts with the soil, each with a length of 1.22 m. The passive earth pressure is calculated using Rankine's Earth Pressure Theory:

$$\text{Passive earth pressure (Pp)} = (1/2) \times \gamma \times H^2 \times Kp$$

Where:

γ (unit weight of soil) = 18.5 kN/m³, H (height of the soil) = 1.5 m, Kp (passive earth pressure coefficient for clay loam soil, $\phi = 30^\circ$) = 3

For a total length of 4 blocks:

$$Pp \text{ (total)} = 62.44 \text{ kN/m} \times (4 \times 1.22 \text{ m}) = \mathbf{329.4 \text{ kN}}$$

Total Reaction Force from Concrete Blocks:

$$\text{Total reaction force} = \text{Frictional force} + \text{Passive earth pressure} = 26.112 \text{ kN} + 329.4 \text{ kN} = \mathbf{355.5 \text{ kN}}$$

TBM Retrieval Details

Our retrieval plan features no additional structures. A pit of equivalent depth as our starting pit will be dug at the expected finish point 30 meters from the start position. As we observe the TBM sinkage and track its vertical position, we will adjust the depth of the pit such that the TBM will appear out into the end pit while being supported by soil underneath. Eye bolt lift points will then be added onto the structure (see section "Equipment lifting and transportation structural analysis") and the TBM will be lifted out of the end pit.

TBM Structure Analysis

Before we began component selection we first had to determine our main drive requirements to ensure we had the forces necessary to successfully dig. This brought us to the Geotechnical DATA Report which gave us soil conditions. We then referenced “Automation in Construction: Determination of the cutterhead torque for EPB Shield Tunneling Machine” (Hu, 2011) to determine our minimum torque requirement.

Below we define some of the key inputs and data points we would need for part selection, material selection, and design specifications needed to construct a Micro TBM. These key inputs are selected based on calculations required from the aforementioned paper (Hu, 2011).

Input	Value
Volume Weight (N/m ³)	18500
Overburden Height (m)	2.5
Ko (Lateral Pressure)	0.6
N (Opening Ratio)	0.57
f (coefficient dynamic friction)	0.3
Width (m)	0.05
Fdp	1
Tau (Shear Modulus) Pa	31000
Kq	0.35
Diameter (m)	0.5
Rb (m)	0.19
Nb (#)	4
Lb (m)	0.088
Db (m)	0.055
Theta (Deg)	30
Output	Value
Output Torque (kNm)	2.0933
	6

Performance Requirements:

We have designed the micro TBM to function at 10 RPM, and at that RPM our calculations based on the method from the Automation in Construction paper indicate that we must have a power requirement of around 2.5kW. To meet this power requirement, we have sourced a 6-pole 4kW gear motor from ChangWei (JiangSu) Drives which has a torque of 2.7kN-m at 10RPM. We selected this motor because it would exceed the necessary torque and required power.

Part 1: TBM Body

To effectively design a TBM body, we must consider various failure modes such as normal stress and Euler's buckling. The main force causing these potential failure modes would be a normal compressive load that acts axially along the body of the TBM.

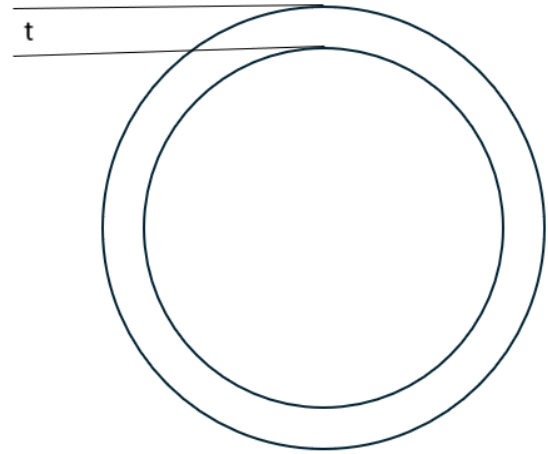
Calculations**TBM Wall Thickness:**

- Normal Compressive Load: 300KN
- Failure Modes: Normal Stress and Buckling

Failure Mode 1: Normal Stress

$$\sigma = F/A$$

Input	Value
Prop Force (KN)	300
Radius(m)	0.25
Max Stress (MPa)	250
FOS	5
Req Thickness (in)	0.15038 67579

**Failure Mode 2: Euler's Buckling**

$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

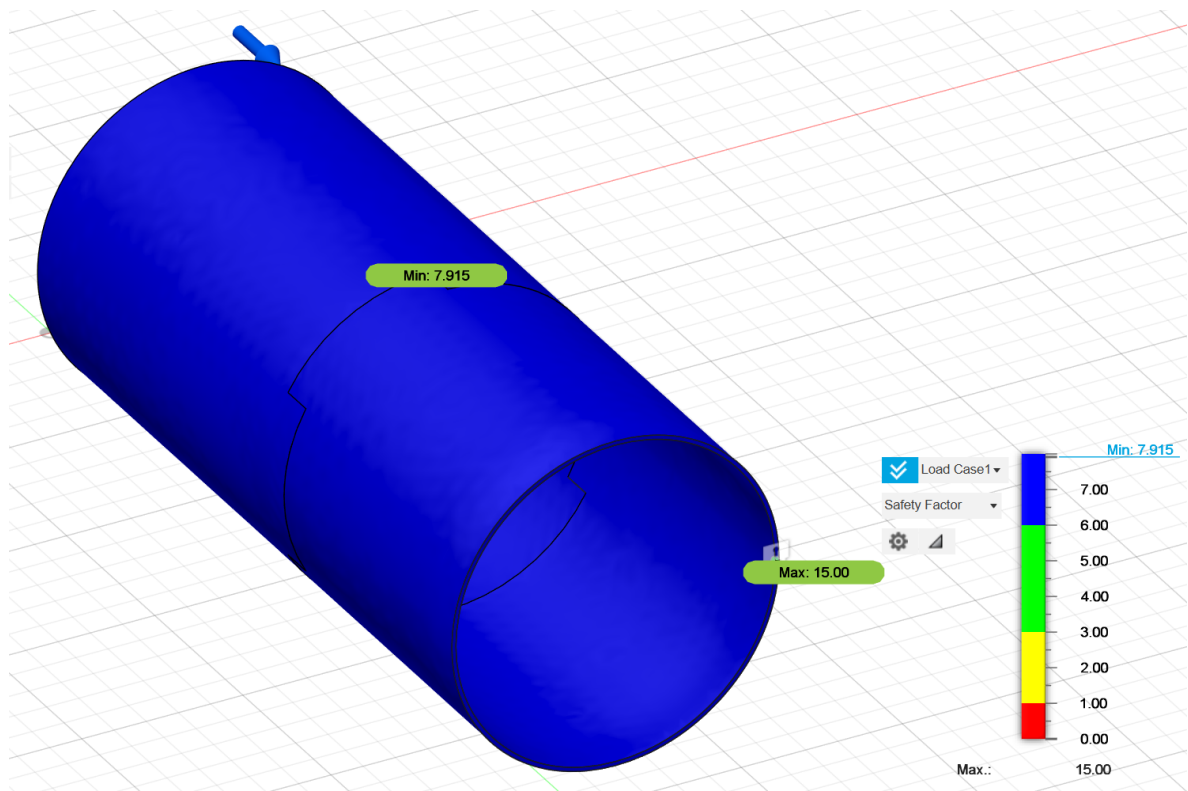
Inputs	Value
Prop Force (KN)	300
Elastic Modulus (GPa)	210

Radius(m)	0.25
Length(m)	30
FOS	2

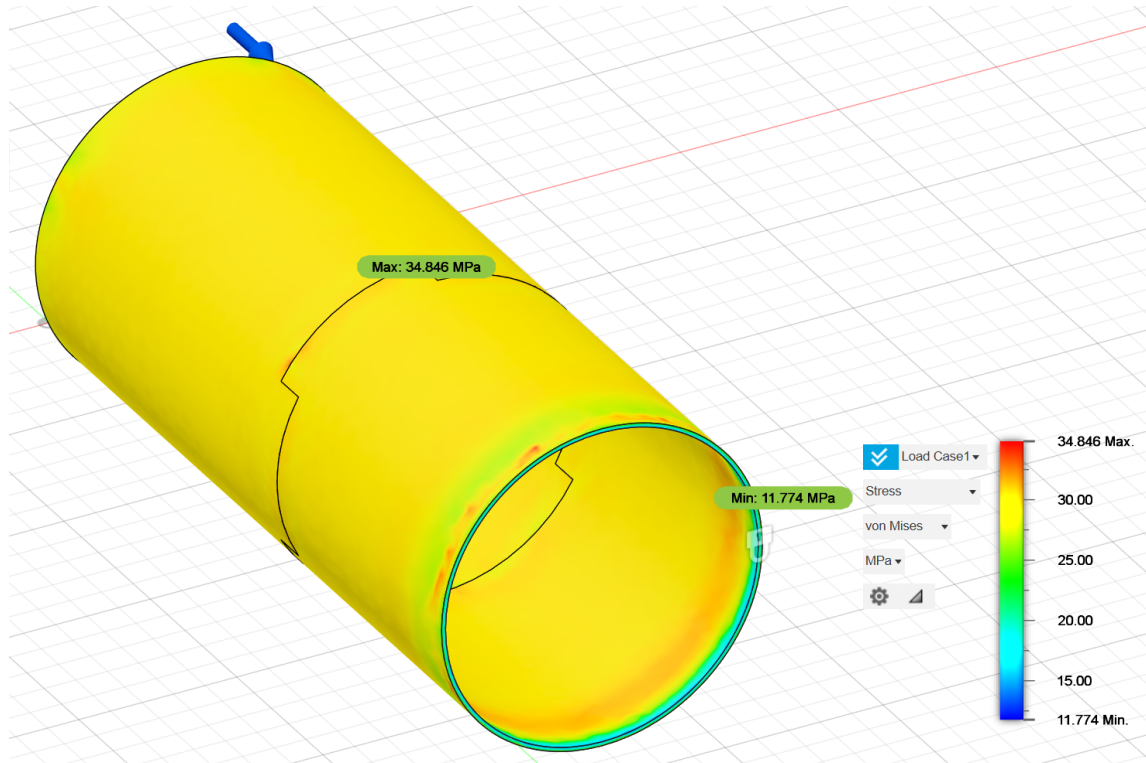
After considering both failure modes, the required thickness obtained from Euler's buckling is larger than the one calculated as a result of normal stress. For this we will consider Euler's buckling as the critical failure mode for the TBM's wall thickness. Thus, in order for the TBM to withstand both normal stress and Euler's buckling we will utilize the thickness of 0.206".

TBM Body FEA:

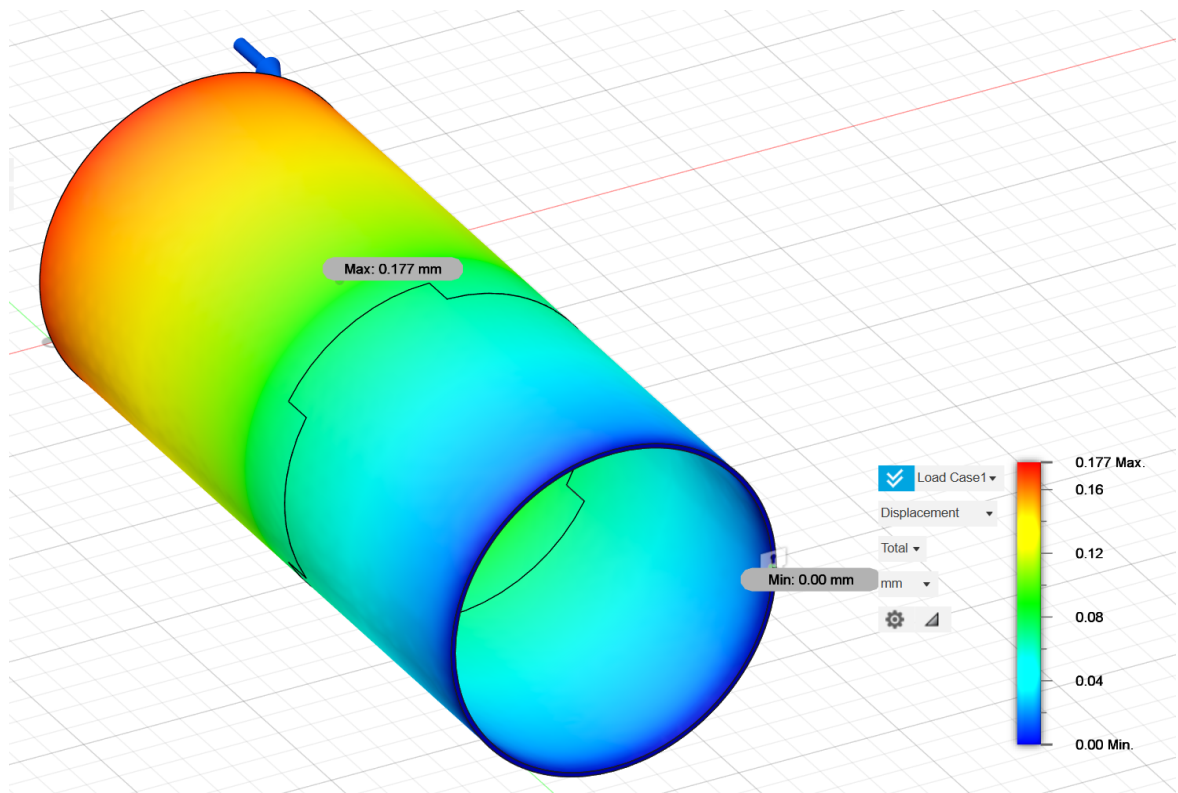
In order to confirm our calculations for the TBM's wall thickness, we performed finite element analysis. As seen in the figure below, we selected the TBM wall materials as steel and applied a 300kN load on one end of the TBM with the force following the axis of the TBM.



FEA analysis of the TBM's body showing the safety factor above the pre-defined maximum stress.



FEA analysis of the TBM's body highlighting the stress distribution.



FEA analysis of the TBM's body highlighting the displacement distribution.

Summary of FEA Analysis on the TBM's Body:

Based on the FEA analysis, our design is capable of withstanding an expected force of 300kN along the axis of the TBM at a thickness of 0.25in. In addition, we were able to confirm that using steel, with a maximum yield strength of 250MPa and elastic modulus of 200GPa, we are consistently at a factor of safety of 7 in all parts of the TBM body.

Part 2: Mounting Plate

To effectively design the mounting plate, we must consider the failure mode of shear stress. The main force causing this potential failure mode would be a shear load that acts on the mounting plate that is fixed by four bolts.

Calculations

Mounting Plate Thickness

- Failure Mode: Shear Stress

Failure Mode: Shear Stress

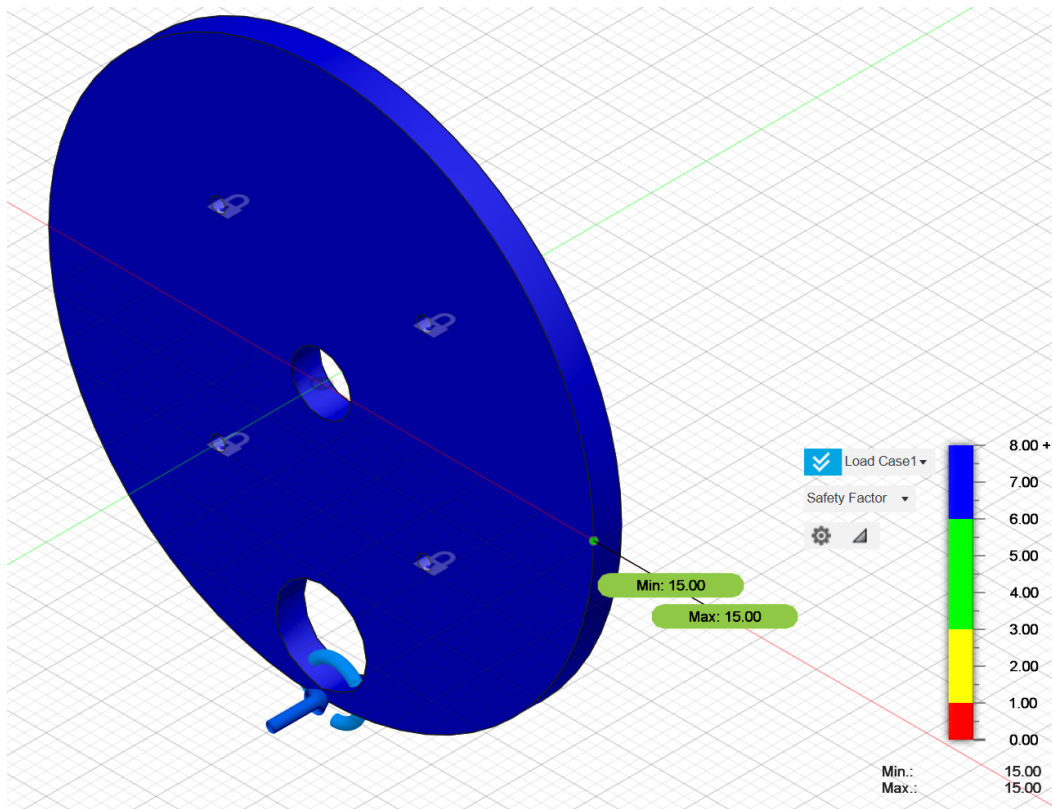
- Shear stress equation: $t = (F/(d*\sigma))$

Inputs	Value
Bolt Diameter (m)	0.0135
Torque (kN-m)	2.7
Max Stress (MPa)	270
Distance From Center	0.1325
FOS	4
Required Plate Thickness (in)	0.89090 14675

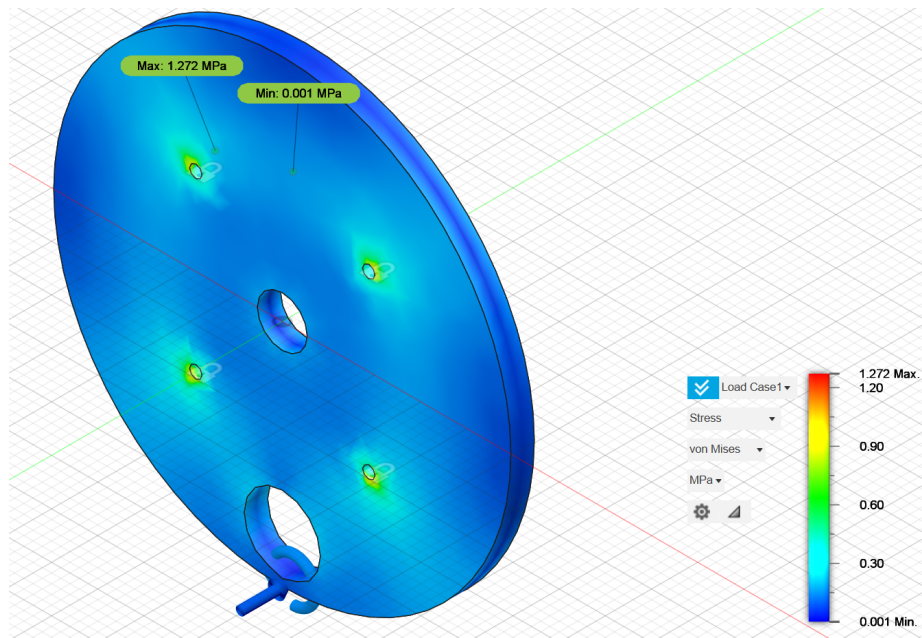
Mounting Plate FEA:

To confirm our design would be able to sustain real-life loads and moments, we used FEA to check for the Factor of Safety, strain, and stress distributions. Applying a 2.7kN-m moment about the

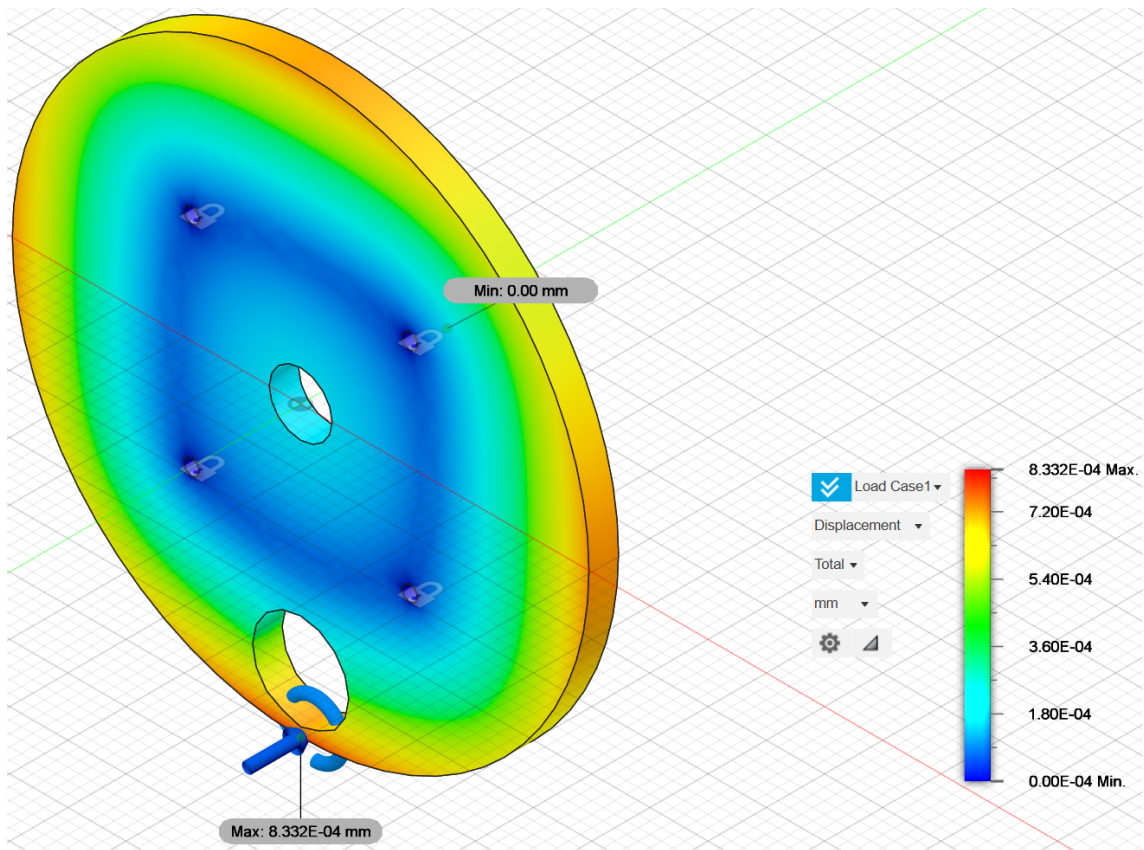
circumference and a 300kN load distributed around the edge while keeping the motor connecting bolt holes fixed, we get the simulations below.



FEA analysis of the TBM's mounting plate showing the safety factor above the predefined maximum stress.



FEA analysis of the TBM's mounting plate highlighting the stress distribution.



FEA analysis of the TBM's mounting plate highlighting the displacement distribution.

The simulation was done with a **1in** thick plate of mild steel with a yield strength of **200MPa** and an elastic modulus of around **200GPa**.

Summary of FEA Analysis on the TBM's Mounting Plate:

Based on our shear stress calculations and FEA analysis, we have confirmed that the mounting plate would have to be fabricated from mild steel at a thickness of 1in. In addition, we were able to confirm that using mild steel, with a maximum yield strength of 200MPa and elastic modulus of 200GPa, we are consistently at a factor of safety of 15 in all parts of the TBM body.

Part 3: Cutterhead

To effectively design the cutterhead, we must consider the failure modes of shear stress, bending stress, and normal stress. The main force causing these potential failure modes would be the propulsion force that drives the TBM forward and the shaft torque.

CalculationsPlate Thickness:

- Failure Modes: Shear Stress and Displacement/Normal Stress

Shear Stress Equation

- $\tau = F/A$

Inputs	Value
Spoke Width (m)	0.089
Prop Force (KN)	300
Max Shear Stress (MPa)	275
FOS	4
Req Thickness (in)	0.4825740 552
Thickness with FOS (in)	1.9303

From this with a FOS of 4 we decided on a cutterhead with a thickness of **2in.**

Spoke Width:

- Failure Mode: Shear and Bending Stress

Failure Mode 1: Shear Stress Equation

- $\tau = F/A$

Inputs	Value
Max Shear Stress (MPa)	275
Torque (KN *m)	3.8
Shaft Diameter (m)	0.09
Cutterhead Thickness	0.0381
FOS	1.5
Required Spoke Width (in)	0.33815

Failure Mode 2: Bending Stress

$$\sigma = -My/I \quad M_{\max} = p \cdot L^2/8$$

Inputs	Value
	167975
Distributed Load (N/m ²)	7.295
Span (m)	0.2
Plate Thickness (m)	0.0381
Max Stress (MPa)	700
FOS	1.5
Required Spoke Width (in)	2.92871 0695

Based on our calculations, the critical failure mode to consider is failure mode 2, bending stress. In order to withstand anticipated bending stress, the required spoke width with a factor of safety of 1.5 must be **3.5"** before bolt calculations which will cause stress concentrations.

Outer Ring Thickness:

- Failure Modes: Euler Buckling and Bending Stress

Euler's Buckling

$$P_{cr} = \frac{\pi^2 EI}{L^2}$$

Inputs	Value
Elastic Modulus (GPa)	210
Diameter (m)	0.5
Torque (kN-m)	2.7
Thickness	0.0381
FOS	5
Min Ring Thickness (in)	1.245

We use Euler's buckling assuming one end of the spoke fixed with a FOS of 5 to allow for a linear beam assumption when in reality the beam is curved. This part of the cutterhead will not fail under bending front to back as it is supported by the TBM body along its length. We chose a thickness of **1.25"**.

Shaft Diameter:

- Failure Modes: Stress and Bending

Failure Mode 1: Stress

$$\tau = Tr/J$$

Inputs	Value
Max Stress (MPa)	740
Torque (KN-m)	2.7
FOS	2
Required Shaft Diameter (in)	2.08561 303

Failure Mode 2: Beam Bending

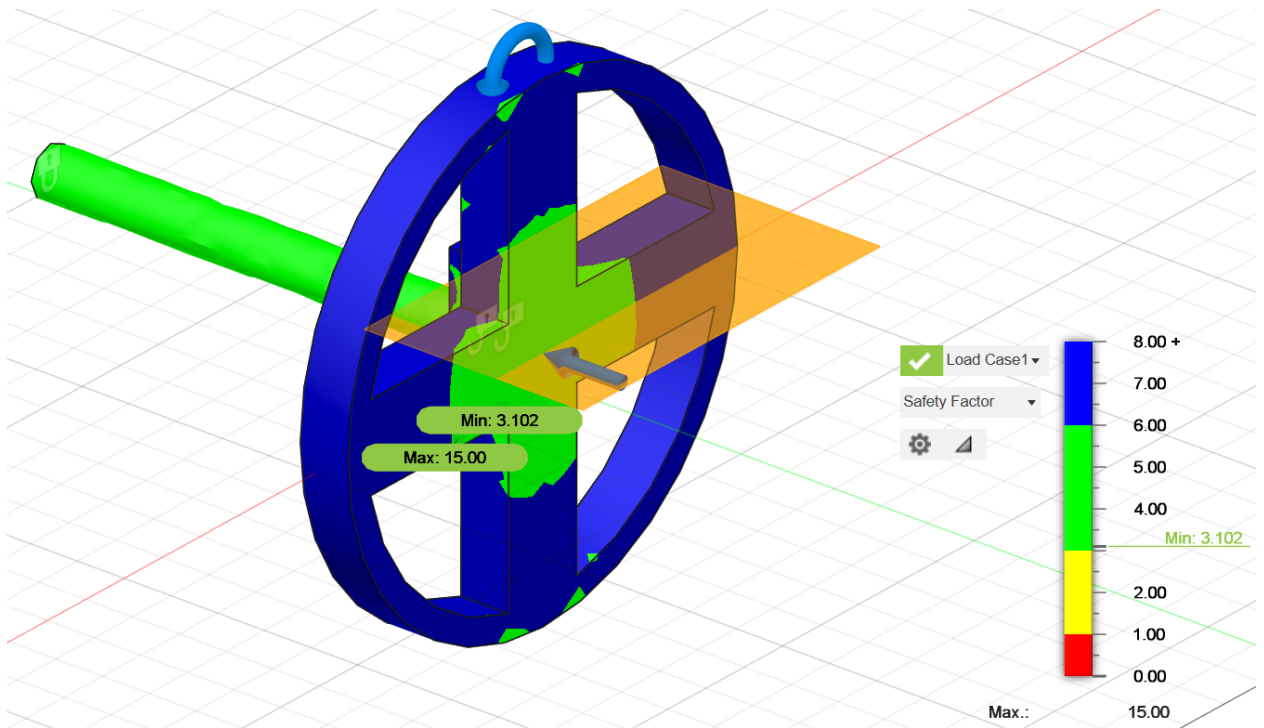
$$U = (PL^3)/(3EI)$$

Inputs	Value
Elastic Modulus (GPa)	210
Allowable Displacement (m)	0.001
Cutterhead Weight (kg)	77
Length after last bearing (m)	0.5
Required Shaft Diameter (in)	1.64573 3632

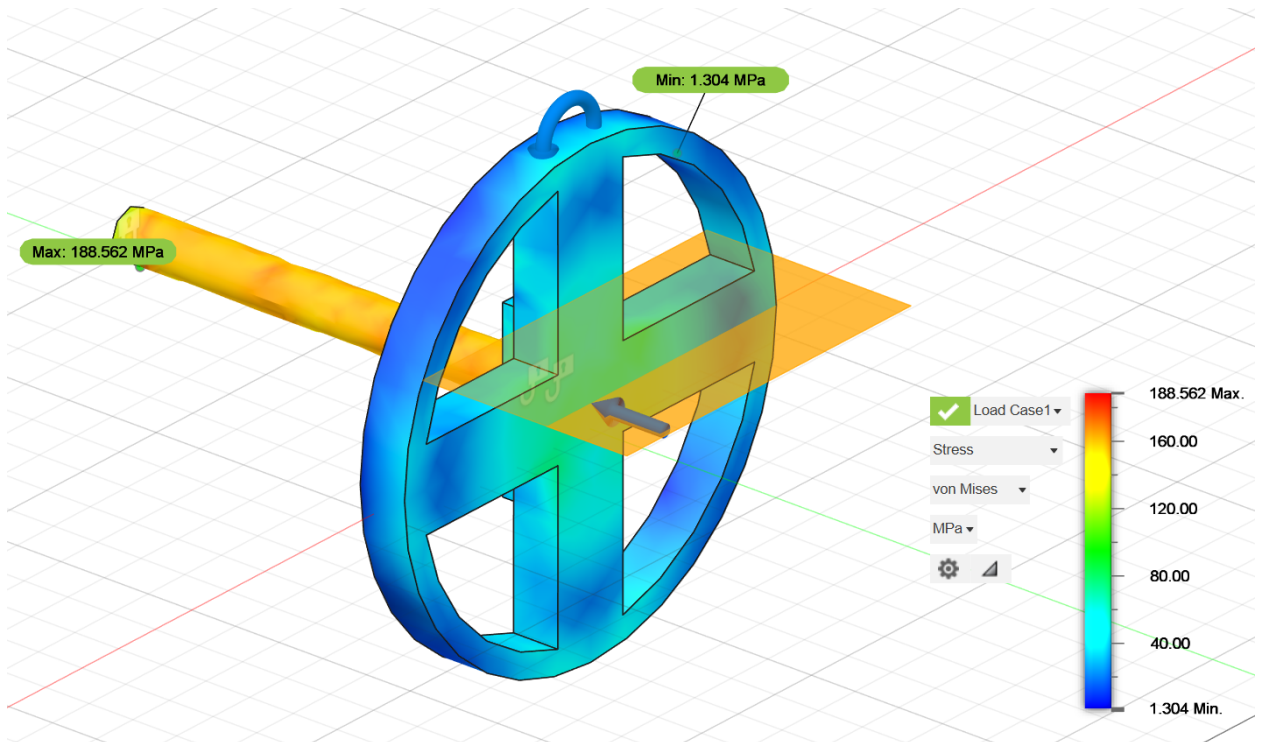
From this we decided on a 2in diameter shaft.

Cutterhead/Shaft FEA:

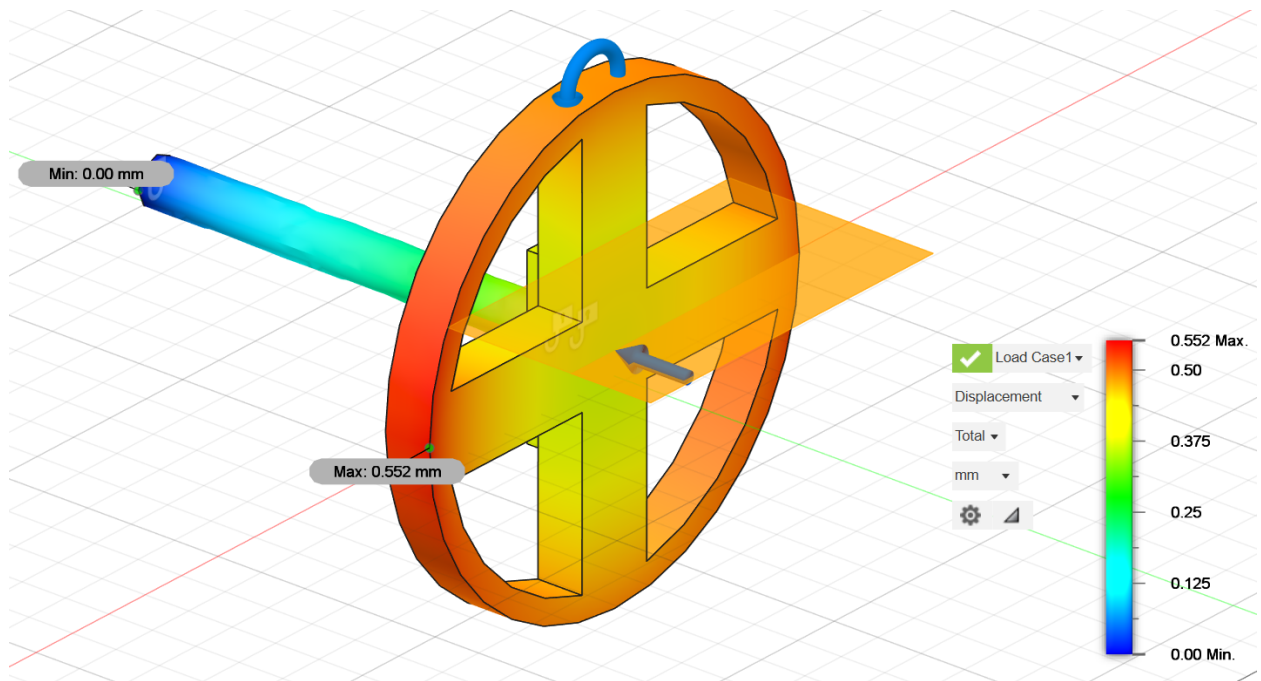
Applying the dimensions above, a 300kN load distributed along the cutterhead face and a 2.7kN-m moment about the cutterhead edge and fixing the end of the output shaft with gravity on.



FEA analysis of the TBM's cutterhead/shaft showing the safety factor above the pre-defined maximum stress.



FEA analysis of the TBM's cutterhead/shaft highlighting the stress distribution.



FEA analysis of the TBM's cutterhead/shaft highlighting the displacement distribution.

This simulation was done with the shaft having yield strength of **635MPa** and the cutterhead/fixture having yield strength of **250MPa** both of which having young's modulus of around **200GPa**. We will be using a higher strength steel on our shaft as it experienced more stress when compared to our cutterhead and mounting fixture.

Summary of FEA Analysis on the TBM's Cutter Head and Shaft:

Based on our FEA analysis and calculations, we have found that the shaft diameter of **2in**, outer ring thickness of **1.25in**, plate thickness of **2.0in**, and a spoke width **3.5in**. Based on the FEA the shaft diameter and cutterhead structure has a minimum factor of safety of 3.1 and therefore it should be able to withstand expected forces and torques. As for spoke width and outer ring thickness, we were able to design for a factor of safety of 1.5 and 5 respectively.

Part 4: Bolt Diameter

It is important to consider shear stress when selecting bolts, especially in rotating systems. For this we have taken into consideration the motor output torque and the shear stress applied to the bolts.

Failure Mode: Shear Stress

$$\tau = F/A$$

Inputs	Value
Motor output torque (kN-m)	2.7
Closest Bolt Radius (m)	0.05
Shear strength (MPa)	700
FOS	4
Bolt Radius (in)	0.558

Based on our calculations, we have selected ¾"-16 bolts. This model of bolt exceeds the bolt radius required to withstand the anticipated shear stress with a diameter of 0.75 in.

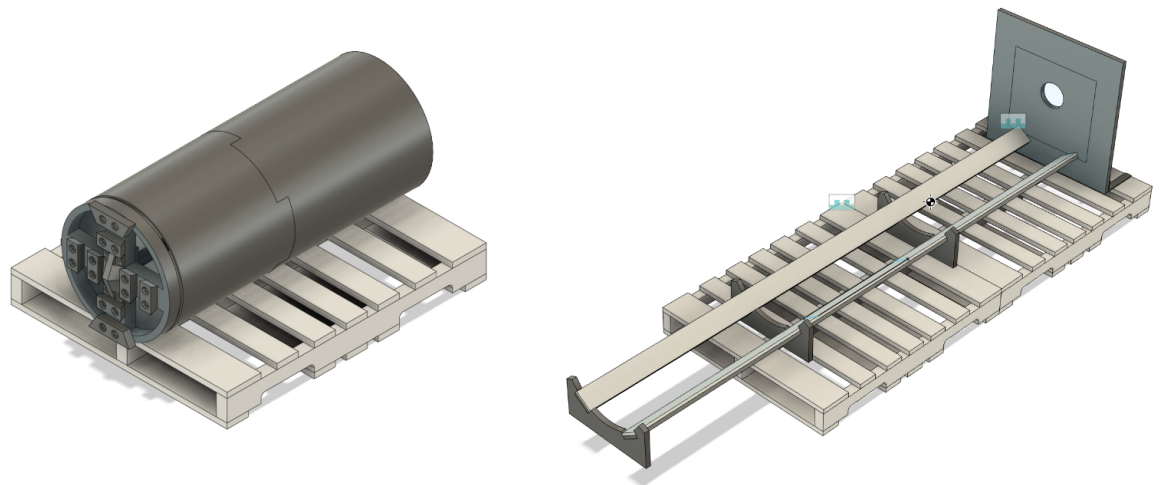
Equipment Lifting and Transportation Structural Analysis

Our TBM and launch structure plan to be transported to, and moved on-site while sitting on standard hardwood pallets. This will be done using forklifts. Lifting is only necessary to place the TBM onto the launch railings and the clay pipes onto the same railings once jacking has begun. This will be done using an excavator.

Transportation Plans

TBM

The TBM will be transported on a 40" x 48" pallet and secured using 2-3 ratchet straps. The following image illustrates how the TBM will fit within the pallet dimensions. The cutterhead will be supported by the pallet but the outer cutters will not since they feature an overcut and protrude beyond the diameter of the outer TBM structure. Any one of the forklifts provided (since they are rated for 5000 lbs or more) will be able to lift the TBM which weighs 500kg/ 1100 lbs.



Launch Structure

Given the length of the launch structure being over 3m long and welded as a single piece, transportation is a slight challenge. For the most part, the launch structure will be transported on a 96" x 40" wooden pallet that allows it to be transported by a forklift.

The launch structure will be placed onto the wooden pallet using a gantry crane. The hand-cranked chain will be routed through the screwjack hole in the back plate and looped back into itself and secure. This side will be lifted by the crane, causing the launch structure to angle up. We will be securing the other side of the launch structure to the ground by holding it in place. Given the length of the launch structure and the standard height of the pallet being about 6.5", we calculated that our crane needs to lift the back plate side of the launch structure about 27" up to accommodate sliding the pallet in place,

and then lowering the back plate side onto the wooden pallet into place. The wooden pallet will also need to be held down by 2 people while lowering to avoid any movement.

To secure the launch structure in place on the pallet, we will be using 4 ratchet straps that will be passed around the top of the launch railing and under the pallet, and tightened until secure. The forklift will then be used to transport the launch structure. The length of the forks of Toyota 8FG25 forklift, which is mentioned as a forklift available at the competition, are specified as 1370mm. The center of mass of the launch structure is found to be about 1140mm from the back of the back plate. This means that, even though the pallet is longer than the forks, the pallet will not be tipping forward, and will be secured against the arms of the fork and there will be no moments about the fork arms that can cause an imbalance. The weight of the entire launch structure unit is about 660 lbs, which is also within the rating of the forklift.

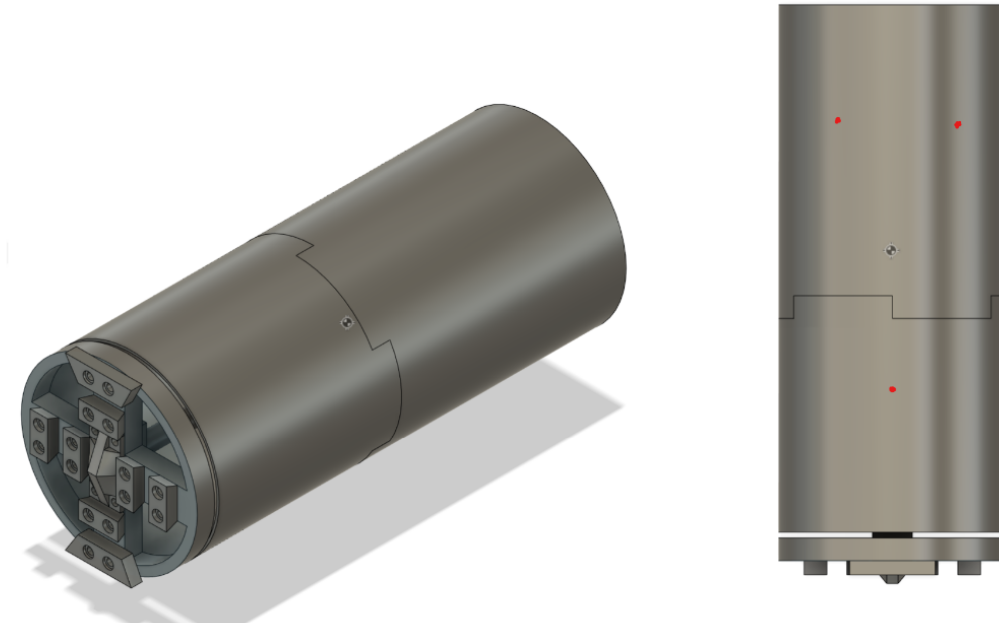
The forklift will transport the structure to and from the U-Haul which will transport the entire machine to the competition. Once at the competition, again the forklift method will be used to remove the pallet from the U-Haul, and can then be transported until the pit on a pallet jack. The pit will likely require a ramp to be placed along a wall for the pallet jack to roll the launch structure down into the pit. Once inside, a gantry crane will be needed to remove the pallet from underneath the launch structure again, similar to how it was placed on.

Lifting Analysis

TBM

The TBM will be lifted in two scenarios - to lift it off the pallet and into the launch railings inside our pit, and to remove it after tunneling is completed. These two scenarios are identical in terms of lifting analysis.

The TBM weighs ~500 kg and the center of mass is along the center radially and fairly balanced leaning slightly towards the gearmotor as can be seen in the screenshot here. We plan to lift with a 5-Ton excavator using chain slings tied to removable eye bolts that are screwed into the outer surface of our TBM which features 3 threaded holes at the locations marked below. These lifting points can handle a maximum load of 9000 lbs.



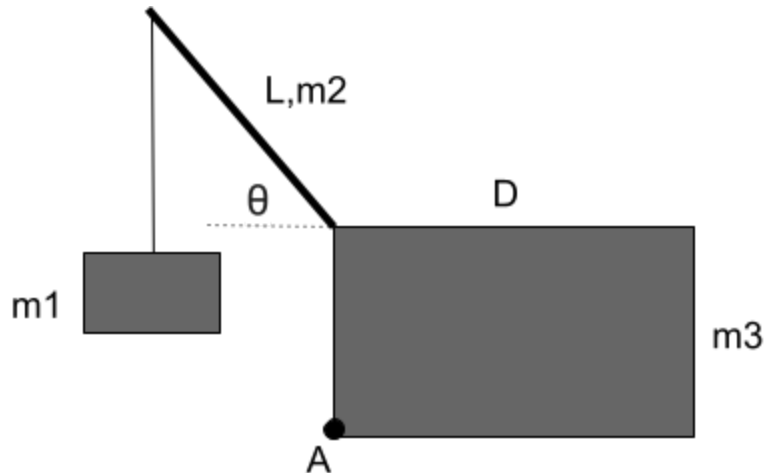
The possible failure modes include the chains snapping from excess load, eye bolt threads stripping through the threaded hole, eye of the eye bolt failing from stress, and the excavator tipping over from a moment caused by the lifting load. We will assume a worst-case scenario for all following calculations that all the load from the TBM is applied on one single cable/ eyebolt vertically down.

Failure Mode 1 -Chain Snapping from Excessive Load:

This failure mode is the least likely since the chain sling we will use is [1/4" proof coil pull chain](#). This is rated for 5000 lbs = 2250 kg equivalent of force thus giving us an FOS of 4.5 even when assuming all load is applied on one chain.

Failure Mode 2 - Excavator tipping from moment caused by load:

The following FBD shows a simplified 5-Ton excavator and its arm with the TBM load applied.



We can neglect the mass of the excavator arm (m_2) by adding a large FOS on the mass of our TBM that is further from the pivot point.

$$\Sigma M_A = 0 = (FOS)m_1gL\cos(\theta) - m_3g\frac{D}{2}$$

$$\theta \geq \cos^{-1}\left(\frac{m_3D}{2m_1L(FOS)}\right)$$

$$m_3 = \text{Mass of Excavator} = 5,185\text{kg}$$

$$m_1 = \text{Mass of TBM} = 300\text{kg}$$

$$D = \text{Excavator Body Length} = 2.59\text{m}$$

$$L = \text{Excavator Max Reach} = 5.99\text{m}$$

$$FOS = \text{Factor of Safety} = 5$$

$$\theta > 41.64^\circ$$

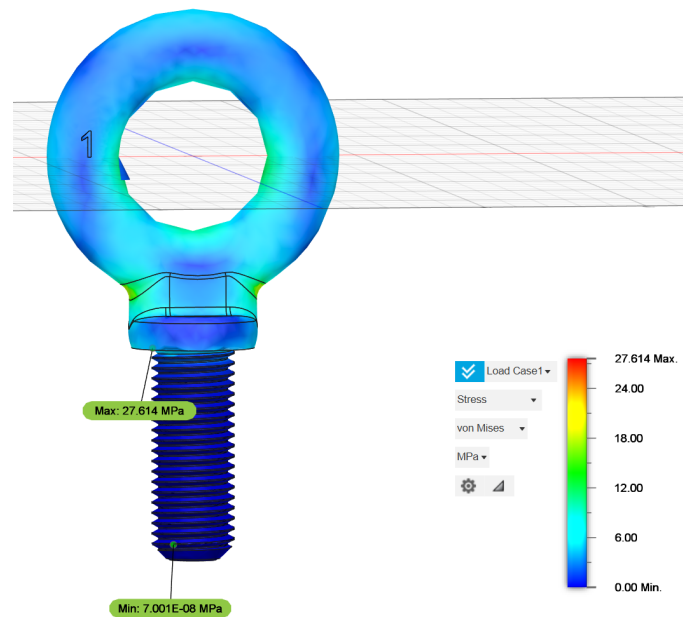
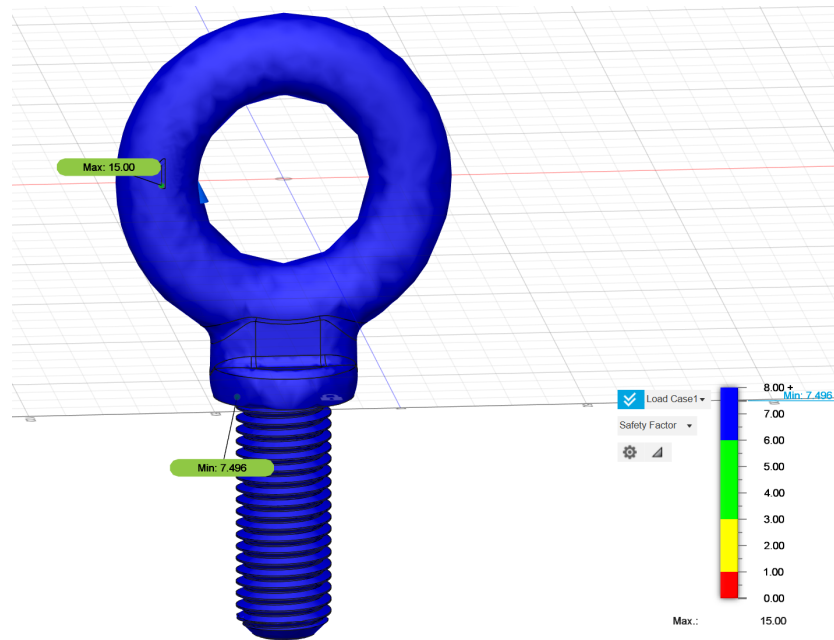
Note: These values are for a fully extended CAT 305 CR. We have yet to find an excavator rental that is sponsored and will re-run this calculation when selected.

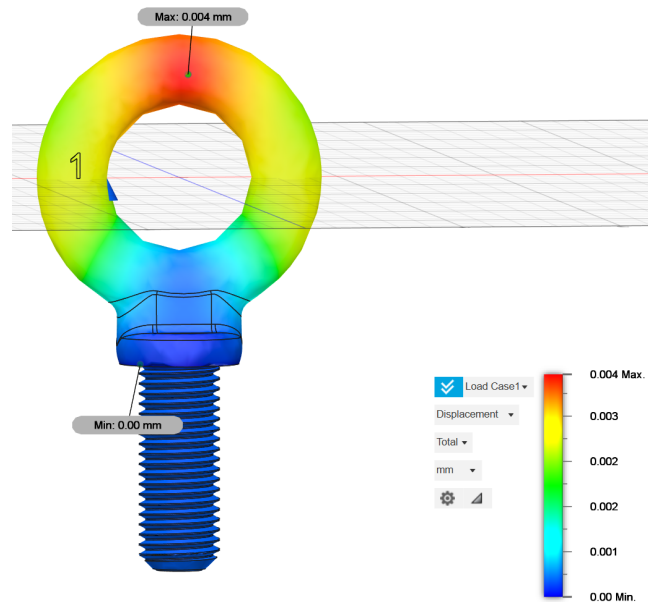
Failure Mode 3 - Eyebolt threads stripping

We plan to use a [1" eye bolt](#) that is rated for 9000 lbs load which is 4000kg. This gives an FOS of 8 even when assuming all load is applied on one chain.

Failure Mode 4: Stress in eye of eyebolt:

FEA analysis on the eyebolt can be conducted. Fixed threads with 300kg load upwards on the eyelet.





The above FEA results show an FOS of 8 overall when tested on CAD given by the supplier. The von Mises stress result shows no cause for concern. The displacement graph also only peaks at a minimal displacement of 0.004mm on the upper portion of the eye. The eye bolt will therefore not fail under loading conditions.

Clay Pipes

The clay pipes we use to conduct pipe jacking are sourced externally. The pipes weigh approximately a ton each. We plan to work with the engineers at Logan Clay who have used these pipes in numerous industrial microtunneling applications regarding appropriate lifting procedures and equipment to use. We have yet to discuss this with the company. All instructions and analysis will be gathered from the company for TBC's review at the competition.

This image below illustrates one possible lifting configuration taken from a video demonstrating the use of the clay pipes.



Test Plans

Electronics Testing

Phase A: Motor and VFD Testing

Objective:

Verify that both gearmotors and corresponding VFDs are functioning properly and ensure it meets manufacturer specifications following shipping. Confirm readiness for operation by checking temperature limits, rotation direction, and stability under basic operating conditions. Conduct tests to determine the ratio between shaft RPM and VFD output frequency.

Tested Components:

- 5.5kW Propulsion Gearmotor
- 4kW Main Drive Gearmotor
- 11kW Propulsion VFD
- 7.5kW Main Drive VFD

Outcome Measurement:

- Motor temperature during low-load operation
- VFD temperature during low-load operation
- Ratio between motor RPM and VFD output frequency
 - This ratio is altered by motor characteristics, so determining this will allow us to calculate motor RPM during competition by scaling the output frequency

Data Collection Devices:

- IR Thermometer: To measure the motor and VFD temperature at regular intervals.
- Smartphone: To verify RPM using slow-motion camera to determine RPM/frequency ratio

Independent Variables:

- Motor speed (RPM)
- Temperature (for thermal effects).

Steps:

1. Visual Inspection:

- Inspect the motor for any visible damage from shipping or manufacturing, including bent shafts, loose connections, or signs of physical stress.
- 2. Motor Function Test:
 - Aim IR thermometer to the body of the motor.
 - Connect the motor to its respective VFD and run it through a sequence: ramp up to 50%, 75%, and full speed
 - Monitor temperature, RPM, and vibrations at each step using data collection devices.
- 3. RPM and output frequency ratio calculation
 - Set motor at 10% output frequency
 - Measure actual output frequency and motor RPM
 - Repeat until 100% output frequency, increasing by 10% each measurement
 - Compute average of the ratio between RPM and actual output frequency
 -

Phase B: Controls and Safety Testing

Objective:

To ensure that the electronics safety systems can properly ensure safety of the operators. In addition, data will be collected regarding TBM navigation

Safety Precautions:

Electrical safety standards will be strictly followed as according to the University of Pennsylvania Power Electronics Laboratory Standard Operating Procedures, which include but are not limited to: PPE, buddy system, one-hand on high-voltage electrical components.

Materials Required:

1. Siemens S7 1214C PLC
2. IMU / inclinometer, liquid flow sensor, methane sensor, relays, RS485-Profinet converter, wires, distribution blocks, physical disconnect switches, circuit breakers
3. Voltmeter

Steps:

1. Assemble everything according to the schematic diagram.

2. Verify the functioning of physical and software disconnects by testing the voltage on the line when the switches are closed vs. open
3. Verify that the reading on sensor data is functioning
4. Test the IMU indoors (without GPS) by lining it on the ground next to marked distance measurements. Accelerate the IMU while taking a video. Use the video and the markings to determine the actual acceleration and compare it to IMU's acceleration measurement. Similar experiments should also be done for angles by placing IMU on a marked arc.

Test Conclusions:

- Verify that safety is working
- Verify the preciseness of the IMU

Propulsion Testing:

Phase A: Structure Testing

Objective:

Assemble and validate the structural components of the propulsion and launch system, including the thrust plate, back support plate, and periodic supports. This phase ensures that each component fits, functions, and aligns properly in preparation for operational testing.

Safety Precautions:

- Ensure stable and flat surfaces for all heavy components.
- Follow all welding safety protocols (PPE, proper ventilation, etc.).

Materials Required:

1. Assembly Components: Thrust plate, screwjack, back support plate, periodic supports, and PTFE pads.
2. Equipment:
 - MIG Welder for welding metal joints.
 - Power tools (saws, drills) for minor adjustments if necessary.
3. Fasteners: M12 bolts, nuts, and threaded rods as needed.

Steps:

1. Fabrication of Key Components:
 - Prepare and inspect all waterjet-cut components (thrust plate, back plate, and periodic supports) for alignment and accuracy based on assembly drawings.
 - Deburr and sand all components to reduce sharp edges and roughness to ease welding
2. Welding Assembly:
 - Weld the metal sheets and structural supports where required, ensuring each joint is secure and aligned correctly.
 - Secure the back plate to the supports with bolts where necessary, and then weld at the joints.
3. Final Assembly:
 - Attach the screwjack to the thrust plate and back plate, ensuring it is positioned correctly for horizontal movement.
 - Position and secure the thrust plate assembly onto the launch structure, ensuring it fits within the launch structure railing.

Test Conclusions:

- Verify dimensions and alignment of all components to ensure proper fit.
- Document any deviations or adjustments made to the assembly.
- Fix any misalignments using saws, drills and angle grinders where possible.

Phase B: Screw Jack Advance Rate and Load Uniformity Testing (with Friction-Based Resistance)

Objective:

Confirm that the screwjack can maintain the expected advance rate of 90 mm/min while pushing the thrust plate under various simulated resistance levels, which replicate soil friction.

Test Setup:

1. Track Design:
 - Mount the thrust plate to screw jack and anchor the screwjack with weights and bolts to a wall/plate for testing to react to the forces
 - Lay out a high-friction rubber sheet track along the ground in a path in front of the thrust plate.

- Ensure the track is horizontal and stable, allowing free movement along the line of propulsion.
2. Adjusting Weight Levels:
- Increase friction resistance by adding rubber coated weights to the track in front of the thrust plate. This adds normal force, which, combined with the frictional properties of the track material, simulates horizontal resistance. In fact, the rubber on rubber friction coefficient is measured to be 1.16, which allows us to require less weights to simulate more resistance.
 - Add weights incrementally to simulate resistance levels from 100 kg to 300 kg

Procedure:

1. Baseline Test (No Added Resistance):
- Operate the screwjack to push along the track with no additional weight, establishing a baseline displacement rate, with the torque and RPM needed to operate at that (specifications given by manufacturer in spec sheet).
 - Measure the average advance rate using tape measure and camera to identify displacement and time, and document any deviations from the expected 90 mm/min.
2. Incremental Resistance Testing:
- Add weights in 50 kg increments, up to 300 kg, to simulate increasing resistance.
 - At each increment, operate the screwjack to push the sled for 1 meter, measuring the advance rate using a camera and tape measure, and ensuring it stays within $\pm 5\%$ of the target rate.
3. Uniformity Check:
- Throughout the test, ensure the screwjack maintains a consistent advance rate without jerking or irregular movement, which would indicate performance issues under load.

Data Collection Devices:

- Tape Measure: Measures the displacement of the thrust plate
- High speed camera: Measures the rate of change of displacement of the thrust plate, using analysis models through image recognition, and also gives us an accurate time of push start and finish to calculate average advance rate.

Expected Outcomes:

- The screwjack should maintain a steady advance rate under each resistance increment, demonstrating its ability to push at a consistent rate against varying loads.
- Allows us to get a baseline understanding of control rate of RPM and torque for varying weights, and allows us to model how that will translate to actual digging time.

Muck Removal Testing:***Phase A: Conditioning Solution System Testing*****Objective:**

To verify the proper functionality of the soil conditioning system, including the IBC tote and pressure washer setup. This test aims to confirm the proper mixing, flow rates, and pressure control needed for optimal soil conditioning. Specific tests will ensure that the IBC tote remains stable under loading, that the soil conditioner disperses evenly, and that the pressure washer operates without leaks or blockages, delivering the expected flow rate for conditioning.

Tested Components:

- 3000-gallon IBC tote
- ACP 214 Soil Conditioner mixed 3% V/V with Water
- Pressure Washer System (including hose and nozzle)

Outcome Measurement:

- Stability of the IBC tote during handling and mixing
- Homogeneity of conditioning solution after mixing
- Flow rate from pressure washer
- Pressure washer stability and leak integrity
- Verification of nozzle attachment and alignment in the TBM system

Data Collection Devices:

- Pressure Gauge: To verify output pressure from the pressure washer
- Timer: To ensure mixing frequency
- Bucket, volumetric container: To measure volumetric flow rate

Steps:

1. Visual Inspection:

- Examine the IBC tote for any signs of damage, such as cracks or leaks.
- Inspect hoses, couplers, and pressure washer components for wear or damage.

2. IBC Tote Stability Test:

- Position the tote on flat ground and fill it gradually to 3000 gallons with water.
- Mix the conditioner by adding 3.6 gallons of ACP 214 and use a long mixing bar, stirring every 3 hours to ensure homogeneity.

3. Pressure Washer Flow and Pressure Test:

- Attach the hose from the IBC tote to the pressure washer, securing all fittings and adaptors.
- Measure initial flow rate using a flow meter and monitor pressure output with a pressure gauge.
- Start the pressure washer, ramping up in stages (50%, 75%, 100%) and measuring temperature and flow rate stability.

4. Nozzle Alignment Check:

- Secure the pressure washer nozzle within the TBM system and confirm it is properly aligned for conditioning solution delivery.
- Run the pressure washer at low speed to observe the spread pattern of the solution.

Phase B: Vacuum Truck and Extraction System Testing**Objective:**

Verify the operation of the vacuum truck and extraction system, ensuring that the system maintains required negative pressure, can sustain continuous muck flow, and effectively handles soil and clay extraction without blockages. The test also aims to check pipeline stability and ensure the entire vacuum line is free from leaks.

Tested Components:

- Vacuum Truck
- PVC Extraction Pipes and Couplings

Outcome Measurement:

- Negative pressure maintained by the vacuum truck

- Continuous flow rate of muck and clay
- Leak integrity of couplings and pipes
- Stability of pipe assembly under negative pressure

Data Collection Devices:

- Pressure Gauge (in Vac Truck system controls): To measure negative pressure in the extraction line
- Flow Meter (in Vac Truck system controls): To monitor muck flow rate
- Visual Inspection: To check for leaks and system integrity

Steps:

1. **Visual Inspection:**

- Inspect all steel pipes, couplings, and clamps for signs of wear, damage, or misalignment. Cracks are major faults that must be replaced immediately. Keep an eye out for all interfaces between pipes or sharp turns and edges.

2. **Vacuum Truck Negative Pressure Test:**

- Park the vacuum truck on stable ground and verify fuel and fluid levels.
- Set the vacuum truck to 50% of full power negative pressure and measure initial performance.
- Run the vacuum truck for 1 minute and monitor pressure and suction to ensure stability and consistency.

3. **Leak and Suction Flow Test:**

- Connect the vacuum truck to the steel pipe assembly inside the TBM and ensure secure connections.
- Begin the suction process with gradual increments in pressure (50%, 75%, 100%) to test for leaks and blockages.
- Measure the flow rate of muck and adjust negative pressure as needed to maintain continuous extraction.

TBM Test Dig

We are planning to conduct a short test dig of our TBM during late February in order to measure our systems performance, identify and detect failure points, and better understand and model certain behaviors like sinkage.

Location:

We are in the process of identifying potential test dig sites. There are some major challenges associated with this. For one, the soil geotechnics in Pennsylvania are significantly different from that of Bastrop TX. Pennsylvania mostly contains Hazleton soil, which is a sandy loam and covers 1.5 million square acres. This is different from the clay loam we expect to encounter at competition. This discrepancy, while likely tolerable, changes our cutterhead and thrust requirements, along with the friction force between the pipe segments and tunnel. It will also affect TBM sinkage, along with ground settlement behavior. This is a concern that we will mitigate as we approach closer by doing more in-depth geotechnical surveys of potential dig-sites. However, this soil discrepancy is likely something we will just have to tolerate and acknowledge.

Logistics:

We will be shipping five pipe segments, the same that we will be using at competition, to the test dig site. In addition, we will be renting a medium-sized excavator that we will have delivered to the site. We will dig the entry and exit pits using this excavator. Then, we will lower the launch structure and the TBM by attaching them with chains to this excavator. **This is an exact replica of what our procedure will be at competition** (assuming we do not adapt after the test-dig).

Test Objectives:

- Measure the average sinkage of the TBM over time
- Measure main drive and propulsion motors' output power
- Validate transportation and logistics procedure for competition
- Come up with standard operating procedures for TBM startup and pipe-reloading
- Ensure integrity of muck extraction systems (no leakages)
- Validate software system for edge cases

Steps:

These are high level steps. We will add specificity and detail to these once we are confirmed to test dig, have the necessary permits and permissions to dig, and agreements with sponsors on major parts/rentals. **This improved test plan will likely be ready before the Final Design Presentation.**

Pre-arrival to dig site,

1. Determine location of dig site and receive relevant permits and permission.
2. Analyze geotechnics of the site, determine soil properties that will govern required thrust force, cutterhead force, and other parameters.

3. Ship NoDig pipes to the site
4. Rent excavator, deliver excavator to the site
5. Rent generator, have it delivered to site
6. Rent vacuum truck, have it delivered to site
7. Rent UHaul, load TBM, electronics, and launch structure into the Uhaul and deliver it to the site

At dig-site,

8. Using the excavator, excavate the launch and retrieval pits
9. Submerge the TBM into the launch pit using chains on the excavator
10. Load first pipe segment into pit using excavator
11. Connect all wires and electronics
12. Connect TBM muck line to vac-truck line
13. Machine is now in STOPPED state, energize it into the IDLE state
14. Test electronics and set parameters for operational cycle
15. START machine, enter EXECUTE state
16. Measure maximum power draw from main drive and propulsion motors
17. Once system enters COMPLETED state, verify that the 1m pipe segment has been fully inserted correctly
18. Record TBM inclination
19. RESET system into the IDLE state
20. Adjust operational parameters if necessary
21. Re-enter EXECUTE state
22. Repeat steps 16 through 20 until all five pipe segments have been inserted and the TBM begins to emerge into the retrieval pit
23. De-energize system by disconnecting wire terminals from generator
24. Use excavator to remove TBM using i-hooks welded onto structure

Business Operations

TBM Production Schedule

Below, we outline a production schedule for our machine for the competition.

Task Name	Timeline
Machining Plans	2nd week, December
Secure Workspace	Early - Mid December
Machining, Assembly, Sub-system Testing	Mid December - Late January
Test Dig	Early-Mid February
Iterations and improvements	Mid February - Mid March
Depart for competition	1 week before competition

TBM Cost Breakdown

The overall budget for the Penn Hyperloop project has been carefully allocated across various systems, with each component contributing to the functionality of the TBM. A detailed breakdown by each component can be found in our BOM in the appendix. The current total expenditure stands at approximately **\$56,158**, distributed among key systems such as propulsion, muck removal, and the mechanical assembly.

The largest expenditure category is the screw jack mechanism, accounting for around 14.4% of total costs. This system is essential for driving the TBM forward with components and set up as compared to hydraulic actuators, justifying its significant share of the budget. Following this, the vacuum truck for muck removal, which is essential for the entire system, represents 8.9% of total expenses. Muck removal is one of our highest-risk systems due to many unknowns and difficulty of modeling, we prefer a high factor of safety using a powerful truck to mitigate clogging.

Another substantial cost is associated with the main drive gearmotor, making up 6.8% of the total budget. This assembly powers the TBM's cutting operations, meaning that we require a motor that has enough torque and RPM, which also has a small form factor so that it can be mounted inside the TBM itself.

Overall, the cost breakdown highlights the project's strategic focus on core mechanical systems by aiming for the most cost-to-performance and simplicity ratio. We are lucky to have significant product donations from the electronics side, with companies like Siemens or Automation Direct supporting us to

lower costs. As we submit additional funding requests and aim to secure more corporate sponsorships, we aim to expand resources for iterative improvements and unexpected expenses as the project advances.

Funding plan

Cash

The Penn Hyperloop team currently holds [XXX] in cash or with written commitments, sourced from:

- [XXX]: [XXX] (as of November 13, 2024)
- [XXX]: [XXX] from the University of Pennsylvania School of Engineering
- **Human Capital Sponsorship:** [XXX] with written commitment
- **Cadence Sponsorship:** [XXX] contract signed
- **Automation Direct Sponsorship:** [XXX] with written commitment

Funding Requests

Total dollar amount of requests for University-affiliated funding submitted is \$76,000. Current funding requests outstanding include:

- [XXX] Draw Down the Lightning Grant
- [XXX] Penn Engineering Entrepreneurship Grant Funding
 - [XXX] Berkman Opportunity Fund
 - [XXX] Miller Innovation Fellowship
- [XXX] Penn Venture Lab Founders Pathway Grant

We also have corporate sponsors such as Siemens or Automation Direct, who will provide us with in-kind donations. The funding drive from Penn alumni and corporate sponsors is ongoing.

Risks & Mitigation

Hazardous Materials

The soil conditioner we use is ACP 214 by MasterBuilder Solutions, which is mixed at 3% V/V with water and pumped into the ground with a pressure washer. The safety data sheet can be found [here](#). All key safety points are accounted for in the on-site setup instructions, briefly summarized below:

- Gloves and goggles will be used when handling the solution
- Solution mixture is kept in a well ventilated area (outside)
- Clean water is nearby and ready in case skin irritation or eye exposure occurs

In addition, the safety data sheet of the material clarifies that:

1. MasterRoc ACP 214 is not classified as hazardous under GHS guidelines, meaning it does not pose significant risks associated with toxicity, flammability, or reactivity.
2. The conditioner contains no hazardous ingredients, which significantly reduces the risks associated with handling and potential exposure.
3. When stored and handled according to instructions, the product remains stable, with no hazardous decomposition products expected, enhancing its safety during various operational conditions.
4. The product is not flammable or explosive, which aligns well with the competition's requirements for avoiding combustion-based hazards.
5. The product does not exhibit significant toxicity toward aquatic life, which suggests it is environmentally benign in controlled usage scenarios.
6. The product is not classified as dangerous for transportation by international standards, making it easier to handle and ship without special restrictions.

To quote the MasterBuilders website:

“Master Builders Solutions is an active participant in the American Association of State Highway & Transportation Officials (AASHTO) National Transportation Product Evaluation Program (NTPEP) Concrete Admixtures (CADD) technical committee. All Master Builders Solutions admixture products are tested through the AASHTO NTPEP program.”

This, along with documentation for agency approval provided on the MasterRoc website, informs us that MasterRoc Solution ACP 214 is safe for use in Texas.

Standard gasoline is used to power the pressure washers and vacuum trucks, and therefore no flames or potentially flammable objects are to be kept nearby. Refill gas containers are to be kept in a dry, secure location far away.

Safety Features

Top Failure Modes

Here, we outline major failure risks of our TBM, as well as methods to mitigate these risks.

Subsystem	Failure Mode Description	Mitigation	Single point of failure? (Y/N)
Electrical	Complete Loss of Supply Power	In the event of a full loss of supply power, likely through a malfunction / overload of the generator, the entire system will rapidly deenergize. This will cause the VFDs, motors, PLCs, and sensors to lose power. If this occurs, the generator will have to be reset / fixed. Once it does, the PLC will be powered on, and the system will enter the STOPPED state. The high-power system will be de-energized but the PLC will be energized, active, and communicating with sensors and peripherals. In this state, manual changes in operational parameters can be made (max power, max torque, motor RPM, etc.). Then, we can enter the IDLE state, where the high-power system is energized but not active, where manual changes in positioning can be made. Finally, the system will enter the EXECUTE state and resume nominal operation.	Y
Electrical	PLC Error / Connection Loss	If the PLC gives an error, or loses connection to the operating station or HMI, then the PLC will enter the system into the ABORT state until the problem is resolved. This would most likely occur if a wire becomes loose somewhere in the enclosure, which would only require some troubleshooting to identify and rectify.	N
Electrical	The inclination sensor becomes shorted, or otherwise outputs an erroneous value	The inclination sensor is the critical output for detecting sinkage angle. Without it, it's possible for our TBM to sink deep enough to where we cannot retrieve our machine. We would have to stop digging if this occurs.	N

		We will ensure that this does not occur through heavy testing of the inclinometer for reliability. We will also be exploring back up options for data transmission in the event of a fault within the main system.	
Electrical	Thermal load causes PLC or VFDs to overheat	The overheating would be communicated to the PLC, which would then enter the system into the ABORT state until the issue is addressed. The system would be given time to cool down by ambient air exposure, then the max power of the VFDs will be reduced to a reasonable amount that we expect will generate tolerable thermal loads.	N
Electrical	Combustible gas is detected within the body of the TBM	If the PPM of methane exceeds 50,000 (the point where it becomes combustible), then the TBM will alert the operator through a warning indication and the operator will perform a software estop of the machine. It is unlikely we will be able to continue digging due to the high risk of internal combustion.	Y
Electrical	A sensor becomes shorted and/or otherwise loses functionality.	The selectivity module will detect this short circuit and disable the shortened portion of the circuit. This change will be communicated to the PLC and the system will either enter the ABORT state if it's a critical short, or simply alert the operator that an error occurred. If the sensor is critical, like the inclination sensor, we will be unable to continue digging.	N
Electrical	A short circuit occurs inside a VFD	Current will spike at the VFD input, and will trip the 40A breaker that is placed in front of it. This will cut power to the VFD, protecting it and causing the PLC to lose connection with it. The system will go into the ABORT state and become deenergize after the shunt trip breaker is tripped. This will allow the operator to fix the issue and resume digging. Refer to the state diagram explanation for this.	N
Electrical	Wires inside the TBM come loose, and a short circuit occurs that results in a current spike on the output of the VFDs	The VFDs we will be using have short circuit and overload protection, so this will not cause damage to the system or operators. It will however cause the system to go into the ABORT state, causing the main shunt trip breaker to trip and de-energizing the high-power system. If this occurs, we will be unable to reattach the wire(s)	Y

		since they are deep in the tunnel, so we will be unable to further dig.	
Muck Removal	Improper seals, attachment of couplers, or cracks in pipes can be detrimental to the machine as muck or conditioner could escape the pipe and leak into the machine or environment. This is a significant failure mode as both the conditioner hose and the muck tube are under very high pressure, one positive and the other negative, and therefore leakages are prone. Leaking into the TBM would be detrimental. Leaking into the environment is not an immediate concern as all materials are bio-safe, but will drastically decrease performance of pressure washer or vacuum truck, dramatically increasing the probability of other failure modes.	All pipes inside TBM will be installed and tested ahead of time (Section 2i) and will never be removed. This ensures that the most critical failure mode of leakages inside the TBM will not occur. Testing of the full TBM pipe and hose system will be done before the TBM goes into the ground to ensure peak pressure washer and vac truck performance. The conditioner delivery hose will be an industry standard 1/4" hose that can handle the pressure requirement and flow rate needed (see BOM, Section 2i). A circular coil mode ensures that no sharp edges or stress points occur as the hose is uncoiled. Industry standard pipe couplers from Victualic or Home Depot will be used to secure pipe lines together. Tests will be done to ensure proper attachment and full vacuum performance with every new pipe attachment (Section 2i).	N
Muck Removal	Muck gets stuck and builds up inside the chamber, failing to reach the extraction line efficiently. This loading could cause the chamber's bottom-most steel plate to bend or collapse entirely. This would cause a pressure imbalance between the chamber and the TBM face, potentially leading to cutterhead failure due to increased torque requirements. Enough buildup could also lead to chamber bending and collapse.	A high-pressure hose line will shoot a wide stream of conditioner solution at the back of the cutter head to reduce clogging and aid muck flow (Section 2i). Welds joining the plates of the muck chamber will be thick to reduce sharp edges and corners where muck can become stagnant (Section 2i). Calculations to determine required plate thickness assuming the maximum amount of muck and water are in the chamber (+ FOS) were completed (Section 2i).	Y
Muck Removal	The muck extraction line becomes clogged with a large buildup of muck, halting the vacuum extraction process and causing the muck chamber to flood with conditioner solution. This would also compromise our chamber and cutterhead face pressure balance. Enough flooding could also lead to chamber bending and collapse.	A pressure sensor will be added to the vacuum extraction line to monitor clogs. The TBM advance rate will be adjusted accordingly to reduce clogging and maintain pressure balance between the chamber and the cutterhead face. Routine "declogging" will be done while digging. This entails halting the TBM advance while continuing chamber conditioning and excavation	Y

		(Section 2i).	
Launch Structure	The screwjack and thrust plate push the TBM at maximum force, and are being met by an equal amount of resistance. Meanwhile, the backplate and concrete blocks have misaligned, leading to insufficient reaction forces, causing either a large bending moment in the backplate, or leading to the entire launch structure being pushed backwards. This could also be a safety issue as the launch structure could be flung backward if not correctly reacted, leading to potential damage to surroundings and individuals in the vicinity.	The back plate has been designed with a significant safety factor of 15+ at every point for the event that the forces it will experience are higher than expected. (Section 4) The back plate also has added L-brackets to avoid misalignment of the backplate from the concrete blocks since the blocks will rest on these brackets and anchor the entire launch structure in place. (Section 4) The concrete blocks are also planned to be arranged in a specific orientation to maximize interface with the back wall of the pit to ensure sufficient reaction forces. (Section 4)	Y
Launch Structure	Given our one screw jack thrusting method, our thrust plate is going to be subjected to significant bending stresses that are the main point of failure. If the thrust force pushing the TBM is close to maximum capacity, and with significant resistive forces against the motion, it can cause the thrust plate to yield.	The design of the thrust plate has been done with a significant factor of safety in place, using both point load assumptions and actual simulations to mitigate any chance of this happening. (Section 4) The screw jack control will also be closely monitored by the team during testing, and any significant deflection of the plate will result in a stoppage of pushing.	Y
TBM Structure	The main drive motor is bolted to a plate by 4 screws. If these bolts were to shear off or shear through the plate the motor would no longer be fixed and would rotate within the body.	We have put a very large factor of safety on our mounting plate thickness. Additionally as we solidify our controls we will incorporate cameras and sensors that will allow us to monitor cutterhead speed while underground	Y
TBM Structure	The TBM body is being thrust forward and the pipe segments that compose our tunnel lining buckle under the load which would not allow us to continue pushing our TBM This would result in us having to excavate our TBM from the place buckling occurred.	In order to mitigate this failure mode our logic will not immediately have the propulsion mechanism push at full strength. The propulsion force will increase steadily as each pipe segment is introduced. That way if buckling does occur it will occur further down the tunnel hopefully allowing for easier extraction.	N
Cutterhead	The cutterhead rotates to cut into the	In order to mitigate this failure mode, the team	

	soil. The outer edge cutters are predicted to undergo more wear than the other cutters since they have a longer effective travel distance per revolution. If the edge cutter wears off to the point where the effective cut diameter is under 20", the cutterhead frame will be scraping against the outer ring of soil thus causing wear on the frame and higher friction = greater propulsion force required.	will observe wear on the edge cutters after the test dig. If we observe damage even just at 5 meters of dig, we will add two more edge cutters on the other two parallel spokes thus reducing the amount of soil cut per edge cutter in half. If significant damage occurs, we will consider re-machining the cutters with stronger steel.	N
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Safety Interlocks Mechanisms

We have multiple safety interlocks, as listed in the software section of TBM description. They are as follows:

- Non-contact lock on the enclosure door
 - This is a magnetic switch that comes in two parts: one for the enclosure body and one for the enclosure door. If the system is energized and this door opens, the system will ABORT
- High-power 125A disconnect switch
 - This is our main emergency stop for the high-power circuit. When switched, it will de-energize the entire high-power system, which includes all motors.
- Low-power 40A disconnect switch
 - This is our main emergency stop for the low-power circuit. When switched, it will de-energize the entire low-power system, which includes the PLC, HMI, and all sensors.
- Independent motor disconnect switches
 - These disconnects will de-energize VFD power independently. This will be useful for testing when we need to isolate each motor.
- High-power GFCI Shunt Trip breaker (software e-stop)
 - This is our main software emergency stop. The shunt trips the circuit breaker when its voltage exceeds 120v. To trigger this from the PLC, we will be using a 120VAC relay with a 24VDC trigger voltage.
- Physical Barriers to the launch pit area

Immovable TBM Recovery Plan

In the case that any of the above failure modes occur causing an immovable state of the TBM, our recovery plan involves de-energizing the TBM and unearthing the TBM based on known inclination values and total pipe segment length inserted.

1. Using dead reckoning from IMU and inclinometer, we will approximate the position of our TBM relative to the surface.
2. We will then use the 5-ton excavator that is used to dig our pit to dig close to the TBM. This digging will stop at a depth ~0.5 meter above our sensor reading for TBM depth.
3. Team members will then dig around our TBM by hand without causing major component damage. No electrical risk is present since TBM is de-energized.
4. The eye bolt lift points are then added back to the TBM Body and it is lifted out of the surface (see section “Equipment lifting and transportation structural analysis”).
5. The excavator will then be used to cover the extraction pit back up.

Final Thoughts

In closing, we would like to express our sincere gratitude to The Boring Company for support, responsiveness, and exceptional organization. Your guidance has been instrumental as we advance toward the Not-A-Boring Competition. As a new team, we are learning and growing rapidly, and your expertise has helped us navigate the complex challenges of this ambitious project.

Despite our progress, we recognize several major risks to our success, including muck removal, electronics failures, and propulsion failures. These represent genuine challenges, and we are addressing them with every resource at our disposal.

We also acknowledge that, as a young team, there may be some discrepancies between what is expected and what we deliver in our Final Design Package (FDP). While our team is composed of talented and hard-working engineers, we are still gaining experience, and each step brings valuable lessons. Above all, we are united in our commitment to make it to the competition, overcoming every obstacle to showcase our capabilities and bring this vision to life.