Assignment Description

A **tensile tester** is a mechanical testing device used to measure how materials respond to uniaxial tension. It applies a controlled pulling force to a specimen while recording the resulting elongation, allowing engineers to determine properties such as **yield strength, ultimate tensile strength, and elongation at break**. This data is essential for understanding a material's mechanical behavior and suitability for design applications.

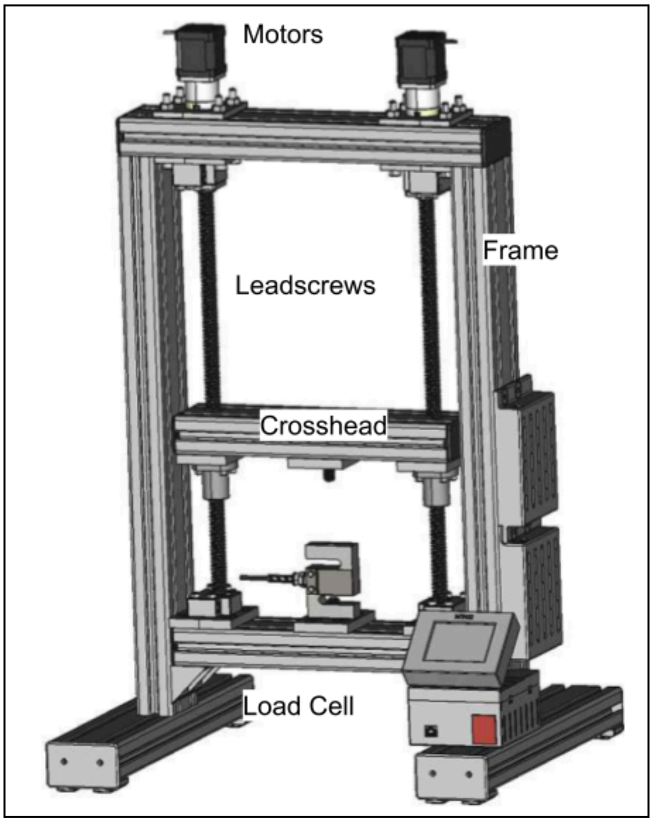


Figure #1. Concept of the Tensile Tester

Develop the motorized drive and structural integration for the 2 kN tensile tester, focusing on motor selection, mounting, frame stability, and clamp design. This phase will ensure that the crosshead moves smoothly under controlled torque and speed while maintaining precise load alignment and secure specimen gripping.

Part 2 Scope of Work

- Select a Motor
- Design Motor Mounts
- Select or design a coupler
- Design a Frame to support a crosshead stability and tensile load
- Select or design a Clamp Analysis and Design
- Provide complete CAD model of the engineered parts of the Tensile Tester
- Deliver a Bill of Materials and Engineering Cost

Key Constraints

- **No electronics, no software**: purely mechanical.
- **Load capacity**: 2 kN (with appropriate safety factor in structure and drivetrain).

- **Speed range**: 1–50 mm/min (achieved via screw lead selection).
- **Form factor**: bench-top footprint favored; manageable overall height; 18 inch leadscrew minimum.
- **Safety Factor**: 4

*Note: Assignment description comes directly from Dr. Fagan's assignment.

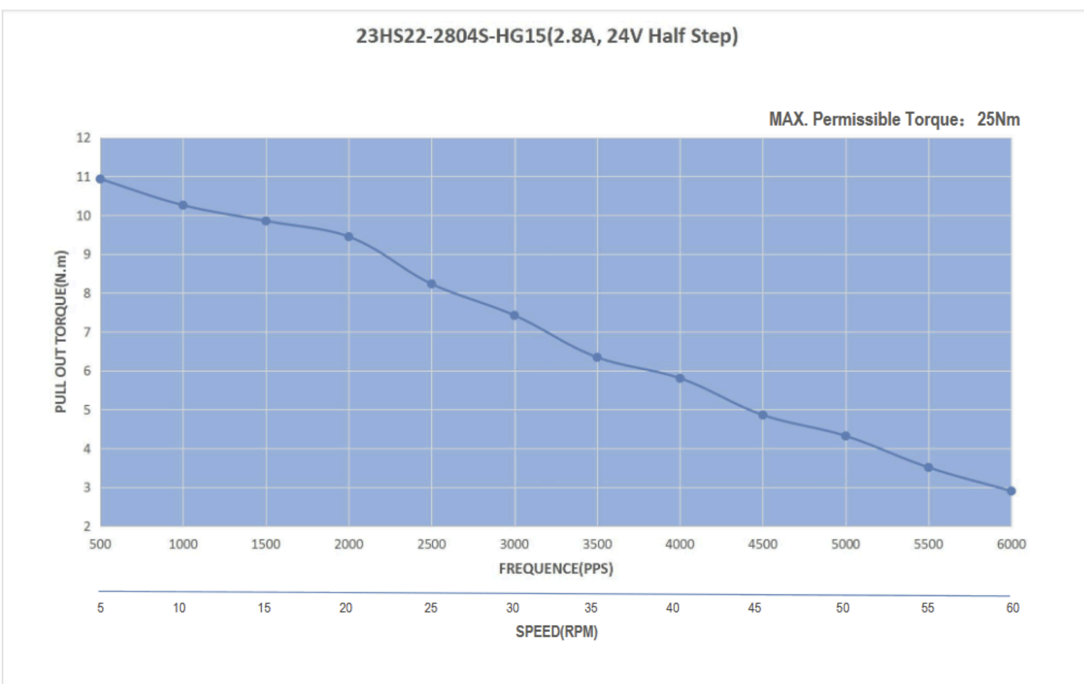
Team Members

Kenny Lee email: klee126@charlotte.edu
Luke DeVries: email: ldevrie2@charlotte.edu

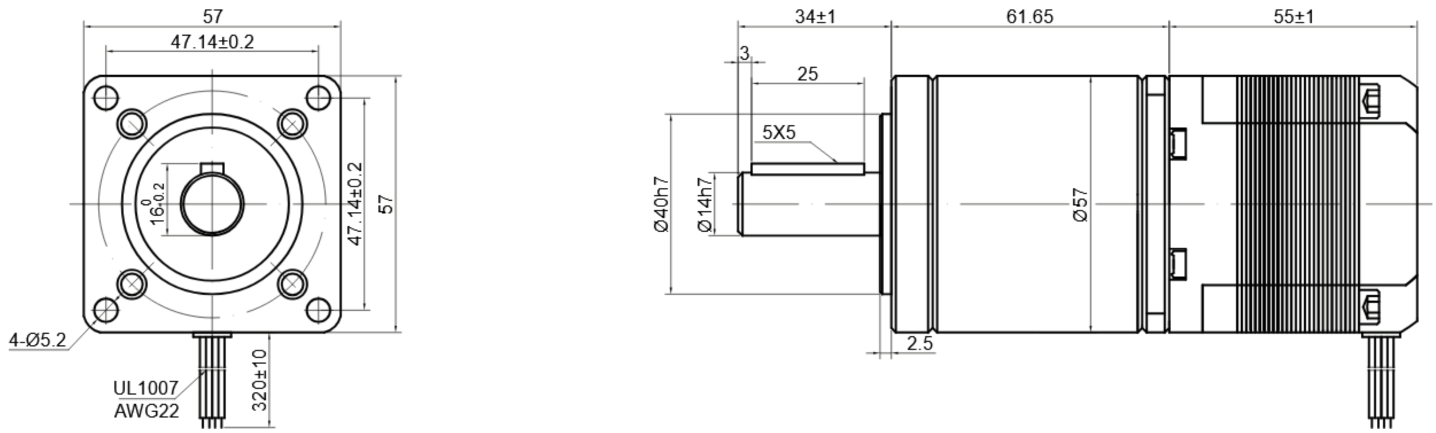
Motor Selection and Attachments

Motor Selection
Team member: Luke

From the previous assignment, it is known that the operational speed range for the lead screws, and therefore the output shaft of the motor, is 0.0318 rpm to 1.592 rpm. We also knew that the required torque for the 2 kN load was 5.408 N-m. I utilized the site omc-stepperonline.com to look for acceptable motors. I started by looking through the geared stepper motors, since they would provide good precision, control, and torque which would all be suitable for a tensile tester. The site utilized the "holding torque w/o gearbox" as one of the searchable criteria. I made sure to use this to check the different motors with their gear ratio and confirm that they could support the required torque of our design. After looking through about 4 or 5 motor spec sheets and torque curves (some brushed DC motors), I eventually settled on the "Nema 23 Stepper motor L=55mm Gear Ratio 15:1 High Precision Planetary Gearbox" found at this link.

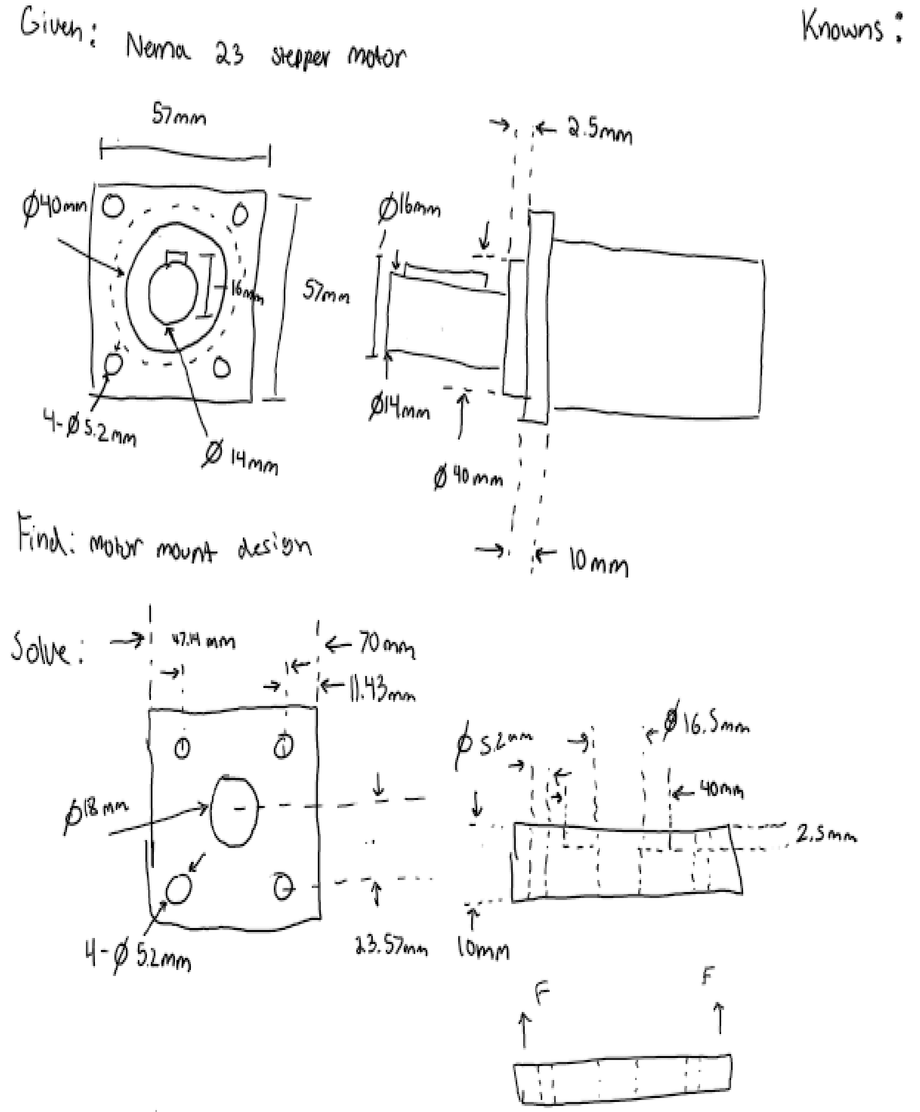


This motor has a holding torque without gearbox of 1.03 N-m, and when accounting for the gear ratio this goes up to 15.45 N-m at the output shaft. The speed range is not listed in the spec sheet, but from the torque-curve, it is graphed from 5 to 60 rpm. Accounting to the 15:1 gear ratio, the max speed needed for our design is about 4.77 rpm which means that this motor will be more than enough to handle the desired speeds. The motor's efficiency is listed at 85%, and the rated current is 2.8 A. The mounting interface is four 5.2mm holes with a nominal distance of 47.14mm between the centers in the x and y directions.



Motor Mount Bracket/Housing
Team member: Kenny

Starting with this part of the project, I started to design the bracket with the given dimensions from the selected "Nema 23 Stepper motor L=55mm Gear Ratio 15:1 High Precision Planetary Gearbox" [motor]. I resketched the motor dimensions onto my document so I could easily access it without switching documents and it helped with trying to design the motor with an ease of comparison. When I started to design the bracket, I started off with the area of the housing that I wanted which was about 70mm^2 and that allowed for the mount to stay within the frame. After that, accounting for the screws and motor mount was already given so I just had to calculate a design that accounts for the motors shaft. Below is the design I went with and the rest of the motor stress equations.



$$\sigma_b = \frac{F}{A} = \frac{2000N}{70mm \times 70mm} = 0.408\ N/mm^2$$

Starting with my work, as I said earlier, I re-drew the motor design so I could look at the dimensions easier. Afterwards, I assumed the area to be 70mmx70mm. Going off of the dimensions, I then assumed the holes to be 47.14mm apart from each other and the diameter of the shaft hole to be 18mm. The 18mm shaft hole accounts for the motor shaft of 14mm plus the additional 2mm that comes from the notch on one side of the shaft. This part of my process is just the design and the assumptions that I made to get it ready to be verified with the math. I also did a little bearing stress calculation for the upcoming thickness calculations. I used the tensile force of 2000N with my assumed 70mmx70mm area for the calculations.

Knowns: $r = 23.57$

$P = 2kN$

$SF = 4$

$T = 20Nm$

$\sigma_b = 0.408$

Unknown: $h = ?$

Solve:

1. $\sigma_b = \frac{F}{dt}$

$t = \frac{F}{d \cdot \sigma_b \cdot SF}$

2. $T = F \times d$

$F = \frac{T}{d} = \frac{T}{n \cdot d}$

3. $F = \frac{T}{n \cdot r}$

$F = \frac{20Nm}{4 \cdot 0.02357m} = 212N$

$t = \frac{212N}{5.2mm \cdot 0.408 N/mm^2 \cdot 4}$

$\boxed{t_{min} = 25mm} > 10mm$

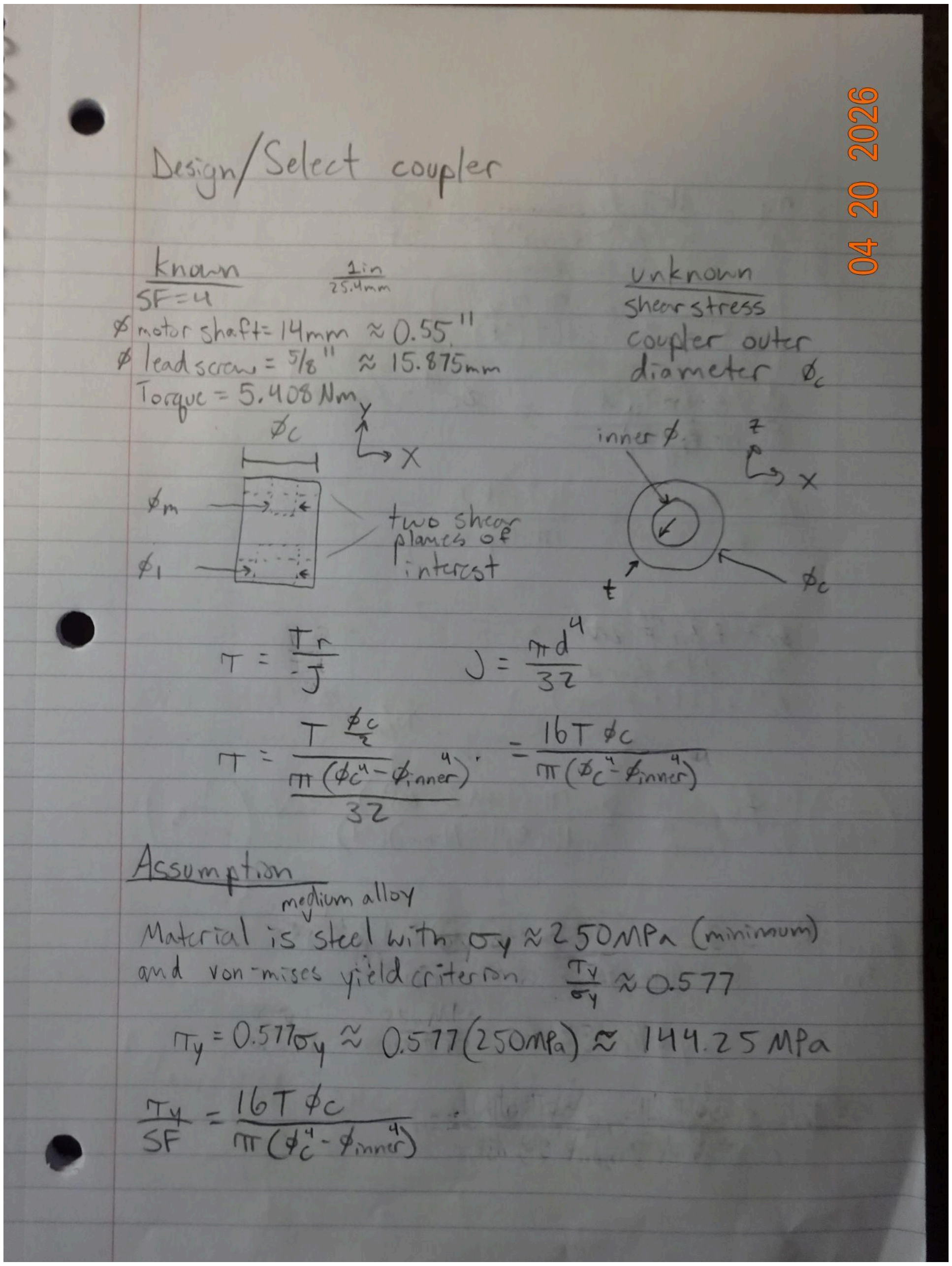
2.5mm

Next was the verification of the thickness. I assumed that I could use a thickness of 10mm and afterwards I started to verify my assumption. After listing my knowns, I planned all the equations I knew that related to the thickness and torque. Using the stress equation, Torque equation, and Force equation with torque, I symbolically solved for the thickness. First, looking at the thickness equation I could see I was missing the force. To get the force, I used our motor torque value with the gear box of 20Nm, the number of screw holes 4 and the r value which is the radius of the circumference from the screw locations. All of these in turn helped me solve for the minimum thickness of 25mm. My assumed thickness of 10mm was too small so I did try the new design with a thickness 25mm to see if the

Select/Design a Coupler

Team member: Luke

As usual, I started the analysis by writing down my known and unknown variables. I knew the motor shaft diameter which was just selected, the lead screw diameter, safety factor, and torque, but I had two unknowns which were shear stress and coupler diameter. Finding the coupler diameter would allow me to calculate the thickness, so I thought it would be simplest to make my calculations in terms of coupler diameter. I modeled the free body diagrams and then began my algebraic calculations of the torsional shear stress. I also made the assumption that the material of the coupler would be steel, I made this assumption after reviewing some of the possible couplers available from McMaster-Carr. I utilized MatWeb to find a rough estimate for the tensile yield strength of a medium alloy steel. I took the lower bound of the range which was about 250 MPa. Since I did not see a provided shear yield strength, I will be making another assumption in the form of the von Mises criterion which estimates the ratio of shear strength to tensile strength as roughly 0.577. This first page of work can be seen below.



Design/Select coupler

Known $\quad \frac{1\,in}{25.4\,mm}$

SF = 4

ϕ motor shaft = 14mm $\approx 0.55''$

ϕ lead screw = $5/8''$ ≈ 15.875mm

Torque = 5.408 Nm

Unknown

shear stress

coupler outer diameter ϕ_c

$$\tau = \frac{Tr}{J} \qquad J = \frac{\pi d^4}{32}$$

$$\tau = \frac{T\frac{\phi_c}{2}}{\frac{\pi(\phi_c^4 - \phi_{inner}^4)}{32}} = \frac{16T\phi_c}{\pi(\phi_c^4 - \phi_{inner}^4)}$$

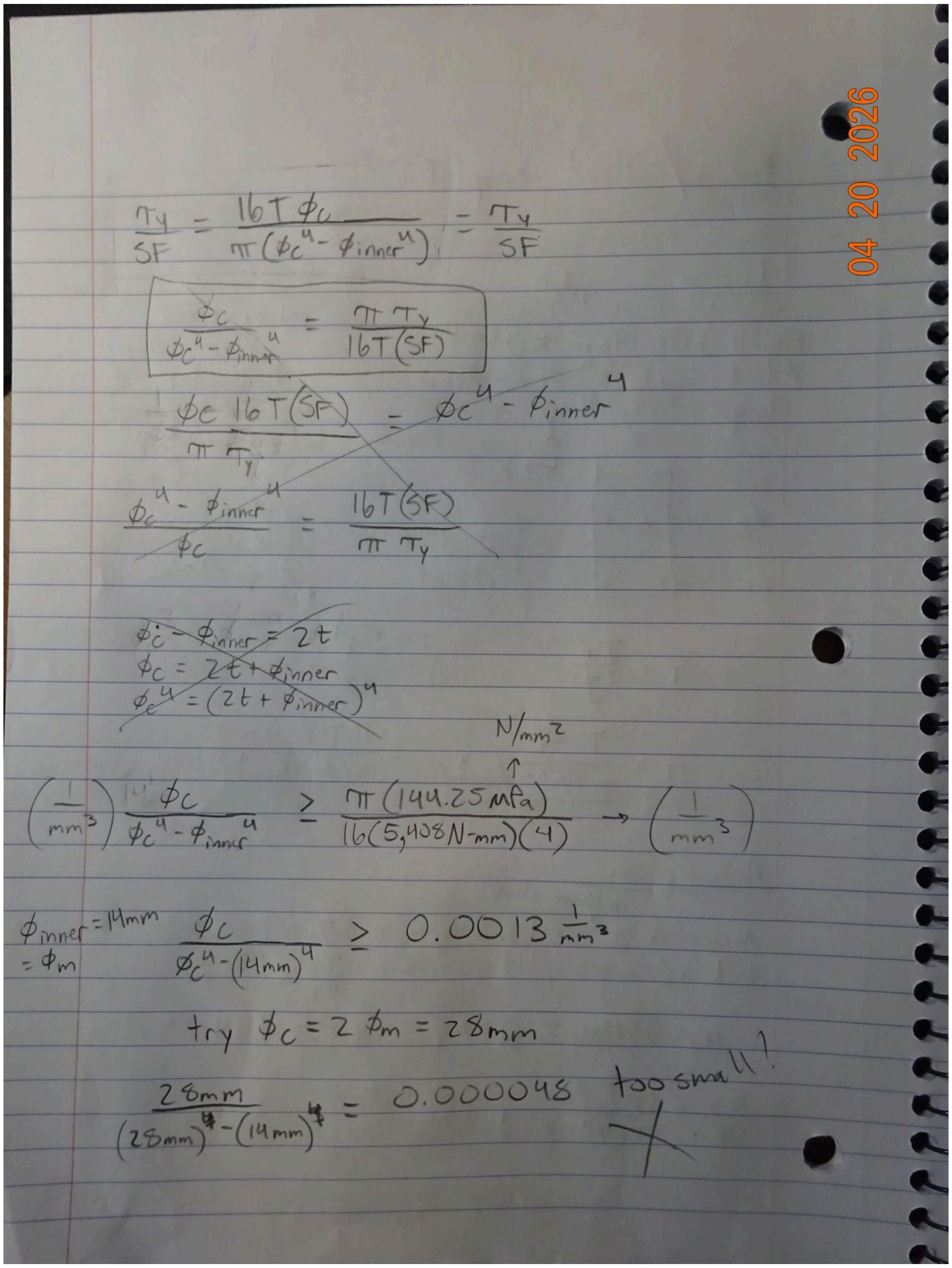
Assumption

Material is medium alloy steel with $\sigma_y \approx 250$MPa (minimum) and von-mises yield criterion $\frac{\tau_y}{\sigma_y} \approx 0.577$

$$\tau_y = 0.577\sigma_y \approx 0.577(250\text{MPa}) \approx 144.25\text{ MPa}$$

$$\frac{\tau_y}{SF} = \frac{16T\phi_c}{\pi(\phi_c^4 - \phi_{inner}^4)}$$

With my assumptions and general equations written down, I began solving algebraically for the diameter of the coupler. Here I ran into my first issue. I quickly discovered that it would take some complex algebra or a difficult to work with expansion to get the diameter of the coupler on its own due to the hollow aspect of the torsional shear stress. After struggling to find a good solution, I decided to simply work with the intermediate values by keeping both the diameter of the coupler and the inner diameter (substituted by the diameter of the motor shaft and/or the lead screw) on one side. Since the inner diameter is already known, I can set up an inequality equation from my algebraic expression and then test individual values for the coupler diameter. During my first attempt at this however, I quickly ran into an issue where no coupler diameter seemed to be possible as the value kept getting further and further away from the inequality expression.



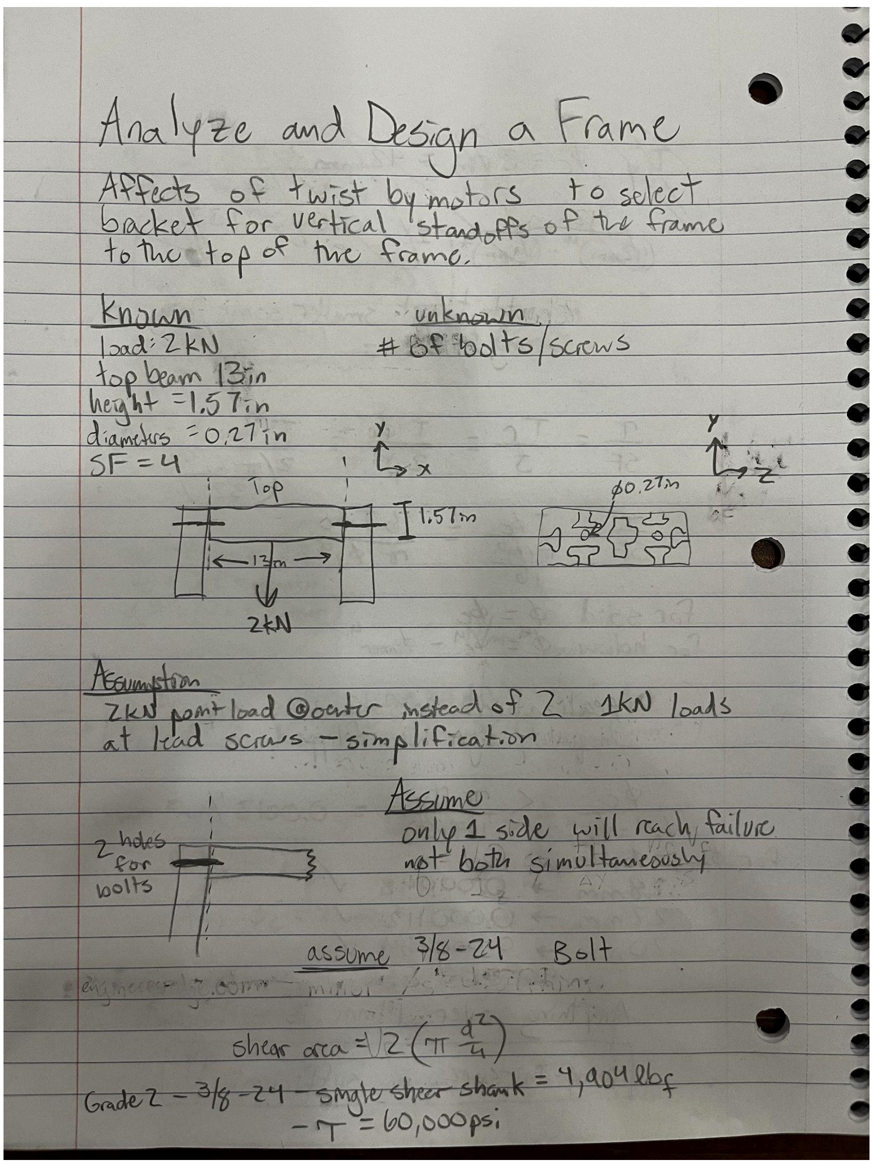
I realized that I must have made an error earlier in the process, so I decided to restart my expression from the beginning. It was through this reworking of the equation that I noticed what my error was. I had simply put the incorrect inequality comparator in my expression, since I had flipped which side was the left of my equation at the top of the page. This meant that I should have been using a less than or equals comparator meaning my calculations were actually correct. I continued plugging in numbers closer and closer to the inner diameter and discovered that almost any coupler diameter should theoretically work! This made me a little uneasy at first, but after thinking about the problem, it seemed reasonable that with the relatively low torque and a fairly large calculated shear strength, even a thickness of 1mm should be able to withstand our design with a safety factor of 4. I am willing to admit that this model could be overly simplified or may miss a critical aspect that could alter the coupler thickness calculation, but I feel confident that any coupler thickness should suffice for our design. This greatly opens up our possibilities for pre-existing couplers as well.

We have decided to select a set screw shaft coupling with a material of black-oxide 1215 carbon steel, for a keyed 14mm diameter shaft from McMaster-Carr which can be found at this link. It has a thickness of 20mm which should be practical for our design and has a keyed slot to fit the motor shaft with key. It is important to note that the lead screw will have to be machined down to a diameter of 14mm to fit within the coupler. The lead screw's nominal diameter of 5/8" can be converted to roughly 15.875 mm, so the turning operation should be quite simple and only take off 0.9375 mm from the radius of the lead screw.

Analyze and Design a Frame

Bracket Connections for the top beam of the frame
Team member: Luke Devries

The frame model was revised after these calculations, but the calculations still hold true for the revised information. I began by writing down my known and unknown variables. I drew a free body diagram where I specified the initial connection type for the top beam which was originally two bolts on each side connected through the vertical portions of the frame. This was later changed to be a corner bracket that still utilized two bolts on each side. The calculations for both situations are shear stress problems with a single shear plane utilizing the cross sectional area of both bolts. My original assumption was to use 3/8" bolts for this connection, and I researched to find the shear force resistance and shear strength for a Grade 2 bolt. This initial page of work can be found below.


Analyze and Design a Frame

Affects of twist by motors to select bracket for vertical standoffs of the frame to the top of the frame.

Known
load: 2 kN
top beam 13 in
height = 1.57 in
diameters = 0.27 in
SF = 4

unknown
# of bolts/screws

Assumption
2 kN point load @ center instead of 2 1 kN loads at lead screws - simplification

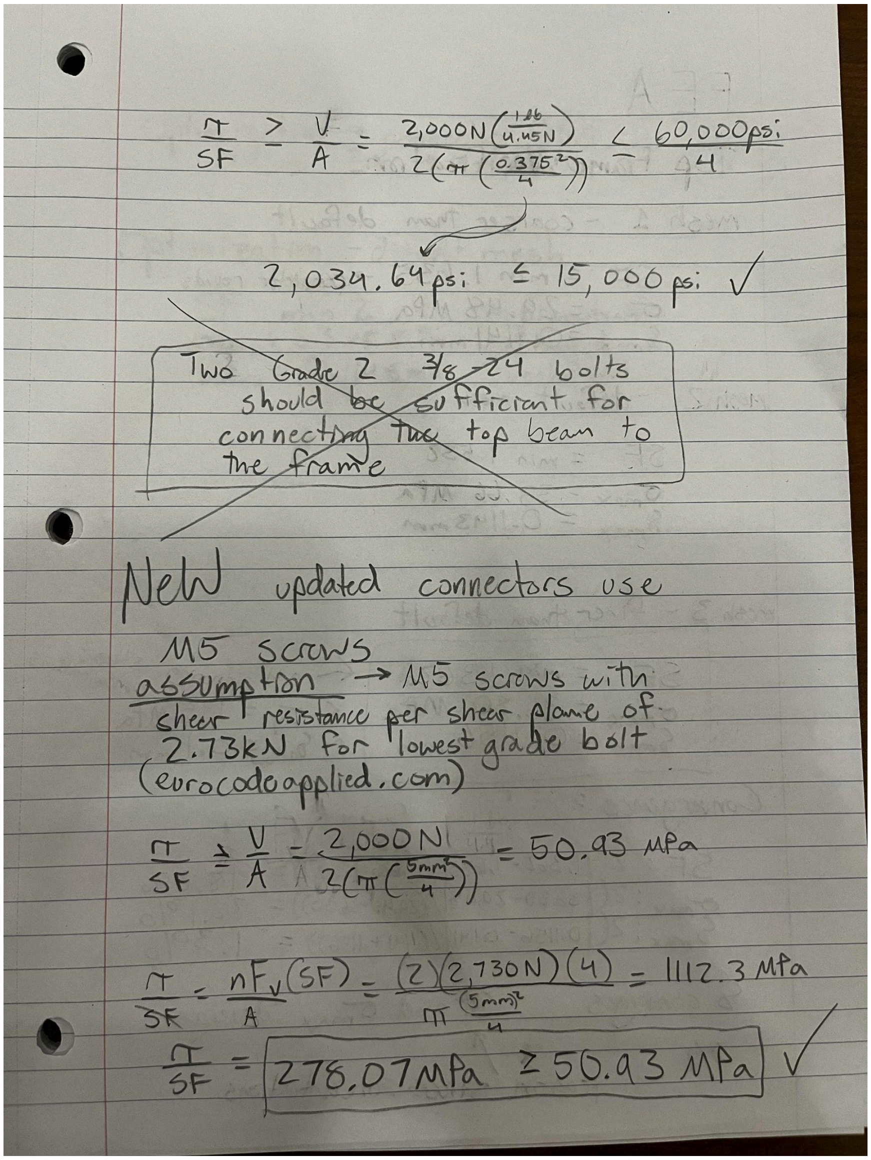
Assume
only 1 side will reach failure not both simultaneously

assume 3/8-24 Bolt

shear area = 1/2 $(\pi \frac{d^2}{4})$

Grade 2 - 3/8-24 - single shear shank = 4,904 lbf
- τ = 60,000 psi

Following this, I made the shear calculations to determine that the two bolts would have a lower shear stress than their allowed maximum utilizing our safety factor of 4. This calculation verified that the bolts would be sufficient. After redesigning the frame to have a longer top beam that would sit on top of the vertical portions of the frame, a corner connector would be utilized. The selected connector allows for a hammer nut to be slid through the t-slot with a M5 bolt to lock the hammer nut and corner connector in place. I redid the calculations for the shear on these bolts utilizing information from eruocodeapplied.com to determine both the allowed shear stress of the bolts and then the shear stress as a result of our 2kN force created by the tensile tester under load. I compared these values and found that our design's maximum shear stress is well below the allowable shear stress with a safety factor of 4. The work for the updated connector style can be found below.



$$\frac{\sigma}{SF} \geq \frac{V}{A} = \frac{2{,}000N\left(\frac{1lb}{4.45N}\right)}{2\left(\pi\left(\frac{0.375^2}{4}\right)\right)} \leq \frac{60{,}000psi}{4}$$

$$2{,}034.64 psi \leq 15{,}000 psi \checkmark$$

~~Two Grade 2 3/8-24 bolts should be sufficient for connecting the top beam to the frame~~

New updated connectors use M5 screws

assumption → M5 screws with shear resistance per shear plane of 2.73kN for lowest grade bolt (eurocodeapplied.com)

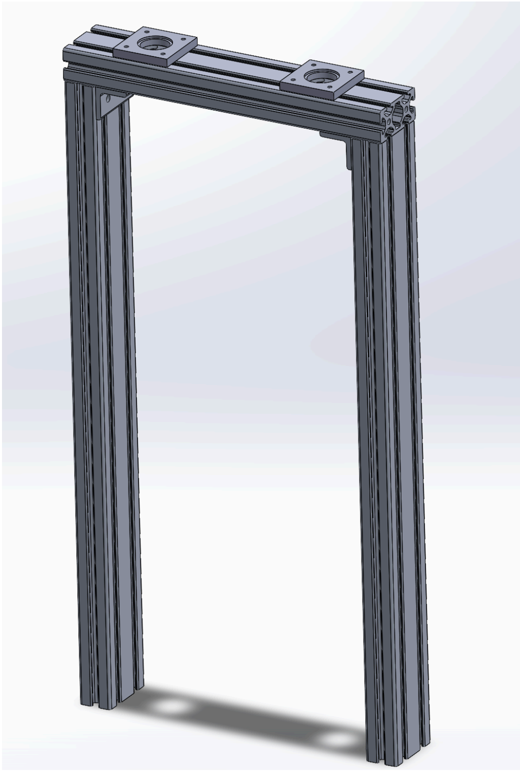
$$\frac{\sigma}{SF} \geq \frac{V}{A} = \frac{2{,}000N}{2\left(\pi\left(\frac{5mm^2}{4}\right)\right)} = 50.93 MPa$$

$$\frac{\sigma}{SF} = \frac{nF_v(SF)}{A} = \frac{(2)(2{,}730N)(4)}{\pi\frac{(5mm)^2}{4}} = 1112.3 MPa$$

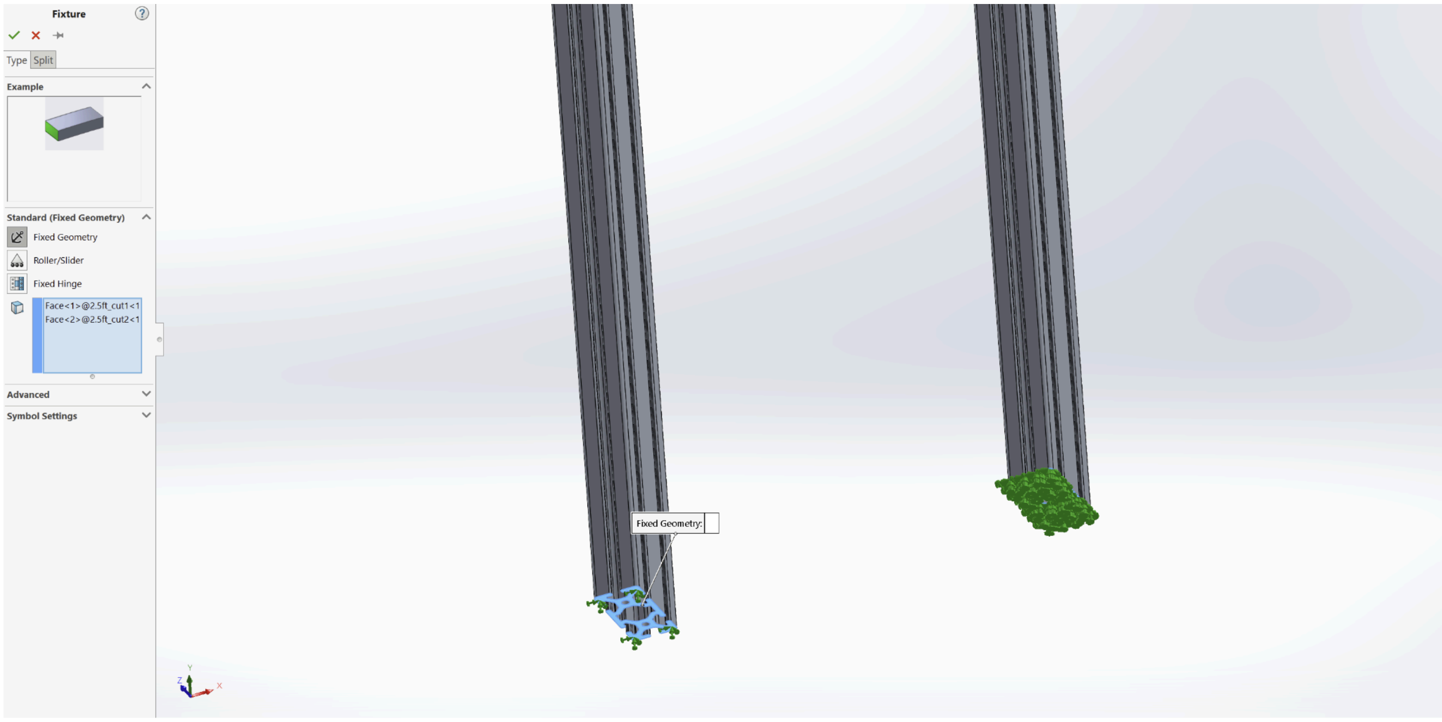
$$\frac{\sigma}{SF} = \boxed{278.07 MPa \geq 50.93 MPa} \checkmark$$

Top Frame Verification (FEA)

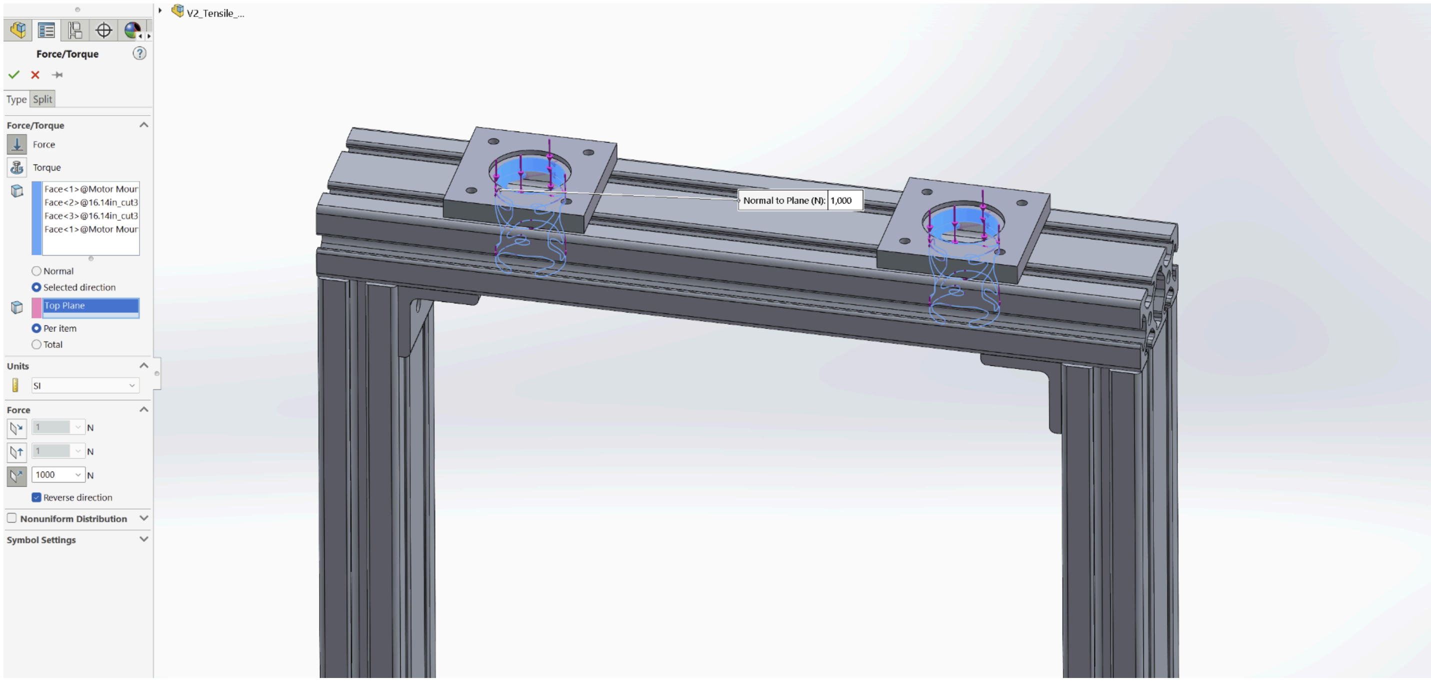
Team member: Luke



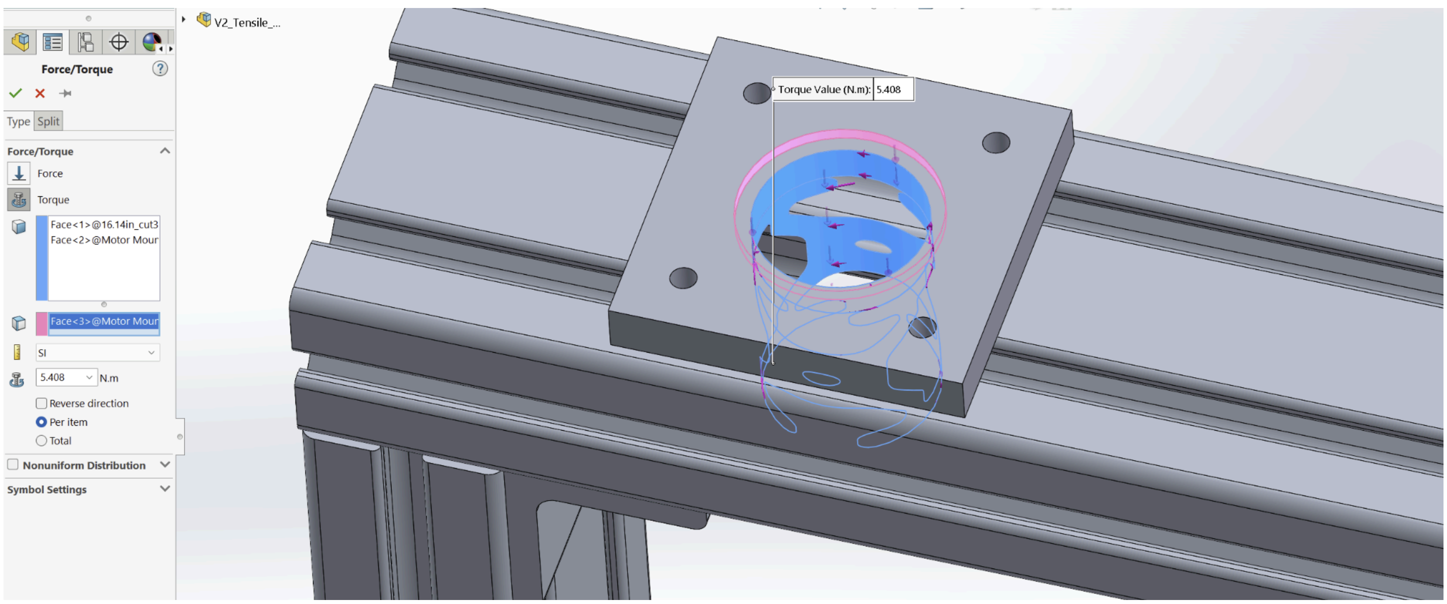
Above is the basic, upper frame design layout. More detail about the assembly can be found under the parametric modeling section. The points of interest for the first verification of finite element analysis were the motor mounts attached and aligned to the top beam, and the two corner connectors. This test was to ensure that the top beam and connectors were sufficient to support the max load of the tensile tester. The targets were to be below the yield stress of the materials, a safety factor near 4, and a displacement below 0.1mm. It should be noted, The connections utilized for the analysis were “bonded” from the constraints and positions created in the assembly.Below are images of the setup for the first FEA.



Fixtures were added to the bottom of both vertical legs of the frame.

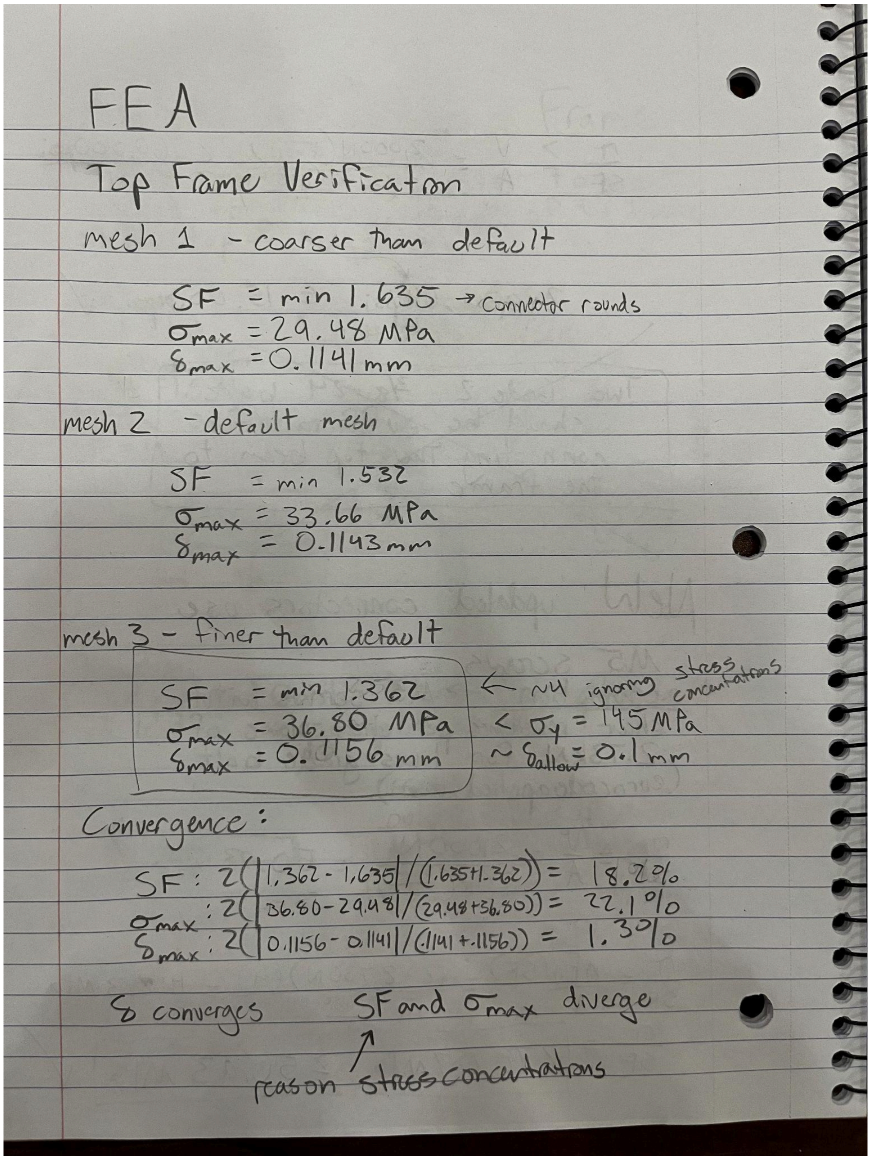


Two 1 kN forces were added to both centers of the lead screw/motor mount holes.



Torques of 5.408 N-m were added to both motor mount/lead screw holes. This value was calculated during the previous assignment.

Three mesh sizes and analyses were run for this first verification step. The first was a coarse mesh, second was a default mesh, and the third was a fine mesh. I tracked the results for the SF, max stress, and max displacement so that I could calculate their convergence. This work can be seen below.



FEA

Top Frame Verificaton

mesh 1 - coarser than default

$SF = \text{min } 1.635$ → Connector rounds
$\sigma_{max} = 29.48$ MPa
$\delta_{max} = 0.1141$ mm

mesh 2 - default mesh

$SF = \text{min } 1.532$
$\sigma_{max} = 33.66$ MPa
$\delta_{max} = 0.1143$ mm

mesh 3 - finer than default

$\boxed{\begin{array}{l} SF = \text{min } 1.362 \\ \sigma_{max} = 36.80 \text{ MPa} \\ \delta_{max} = 0.1156 \text{ mm} \end{array}}$

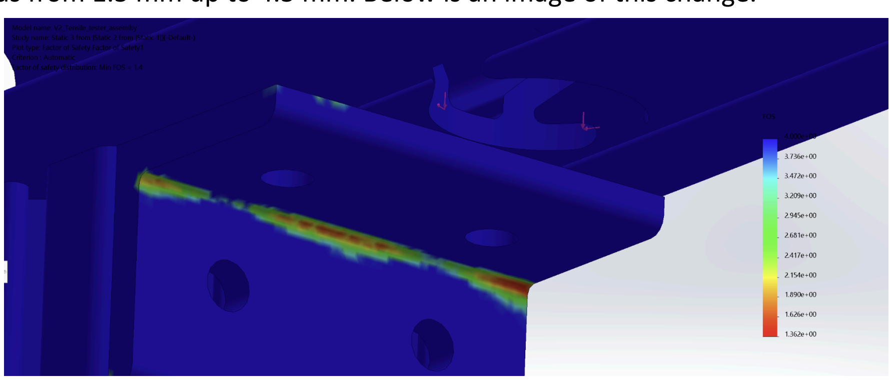
← ~4 ignoring stress concentrations
$< \sigma_y = 145$ MPa
$\sim \delta_{allow} = 0.1$ mm

Convergence:

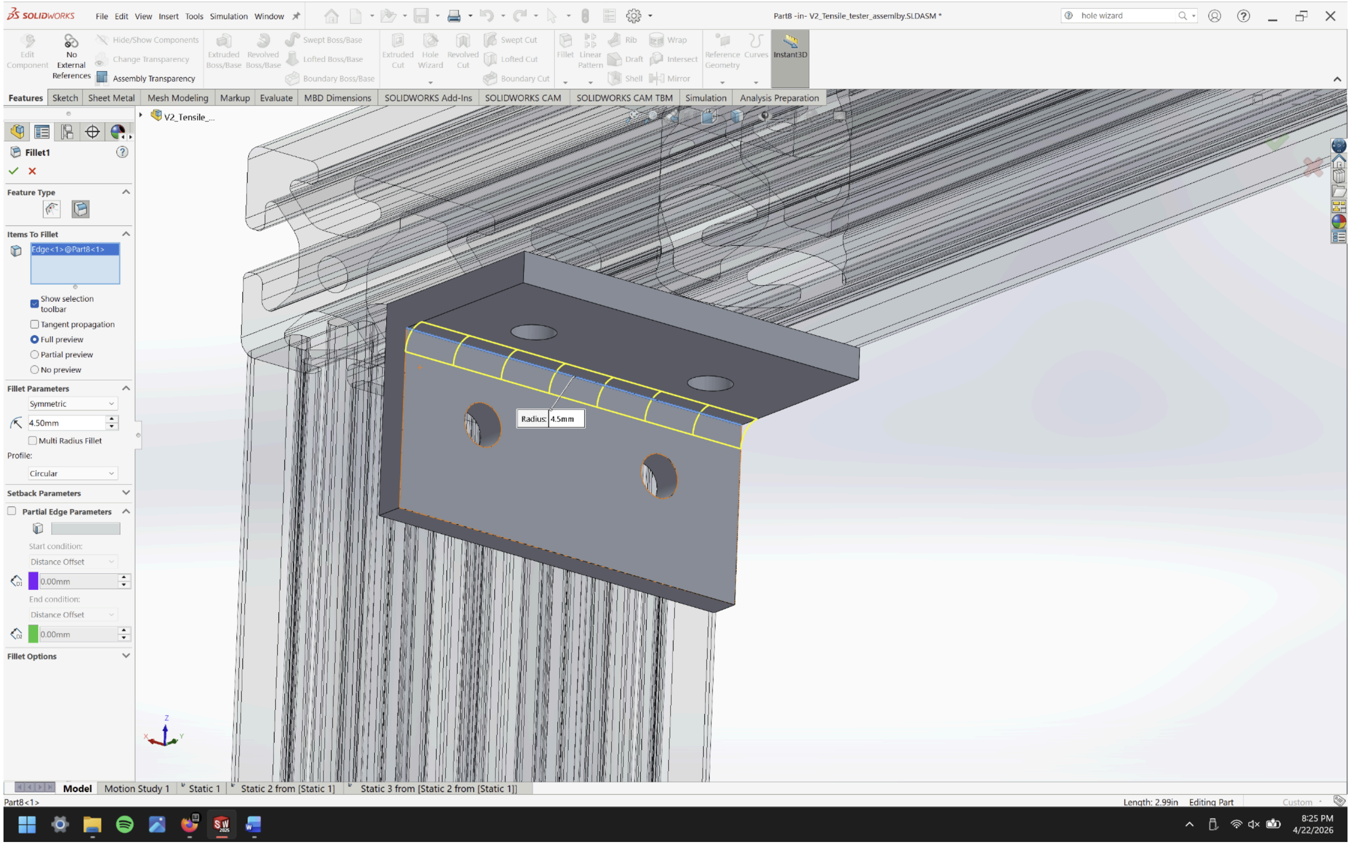
$SF: 2(|1.362 - 1.635| / (1.635 + 1.362)) = 18.2\%$
$\sigma_{max}: 2(|36.80 - 29.48| / (29.48 + 36.80)) = 22.1\%$
$\delta_{max}: 2(|0.1156 - 0.1141| / (.1141 + .1156)) = 1.3\%$

δ converges   SF and σ_{max} diverge
↑
reason stress concentrations

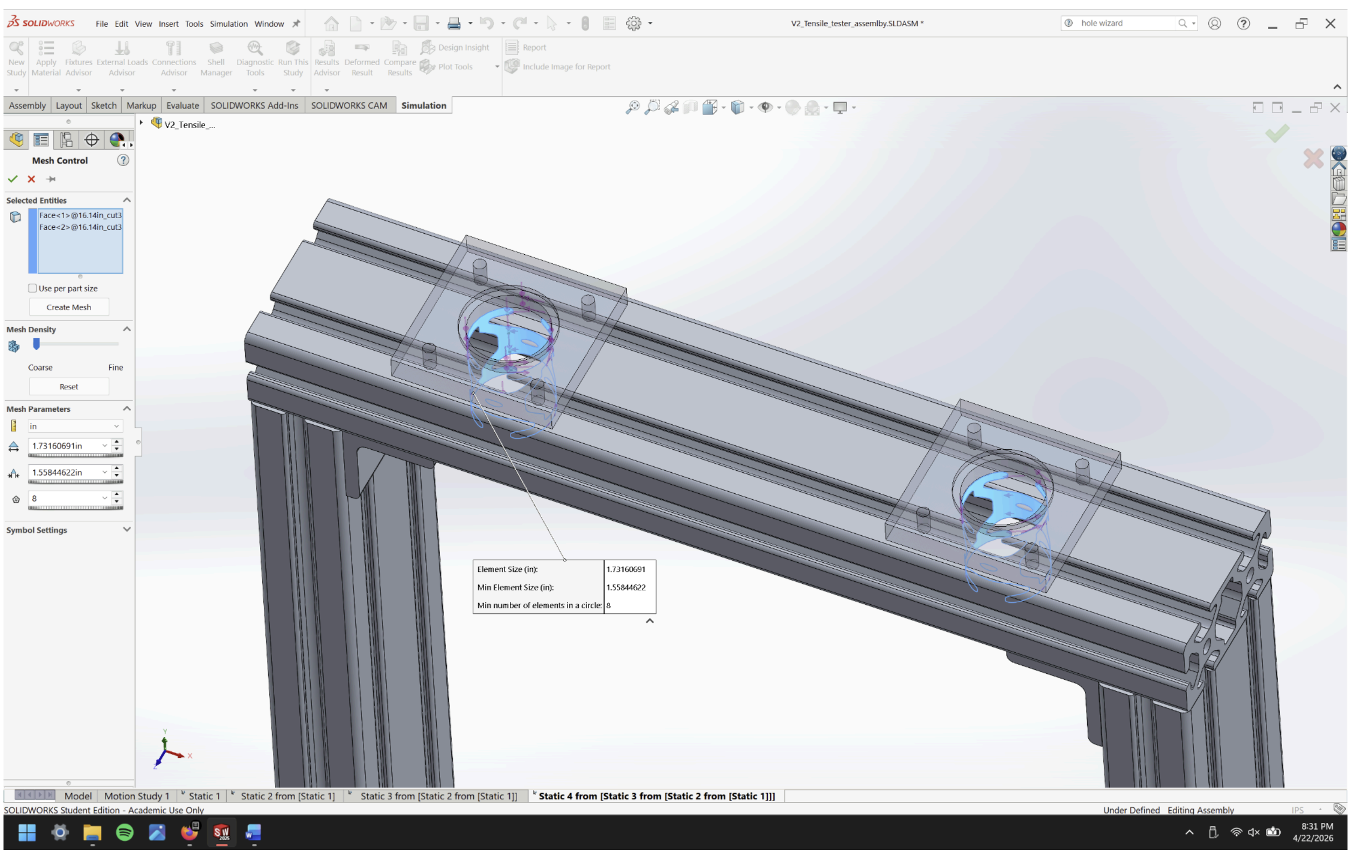
Unfortunately, two of the three criteria did not converge in my tests. By reviewing the results and looking around the model, it seemed that the max stress and minimum safety factor were both concentrated around the fillets in the corner brackets and within the center hole cuts for the lead screws. These are both sources of high stress concentrations, so it is easy for the FEA to inflate these values especially at finer mesh sizes. Understanding this, I continued to work by selecting a few optimization criteria for the model. The first optimization was to change the fillets of the corner brackets to reduce the stress concentrations. After discussing with Kenny, we decided to increase the fillet radius by a factor of 3 which changed the stock radius from 1.5 mm up to 4.5 mm. Below is an image of this change.



Safety Factor at the corner connector fillet

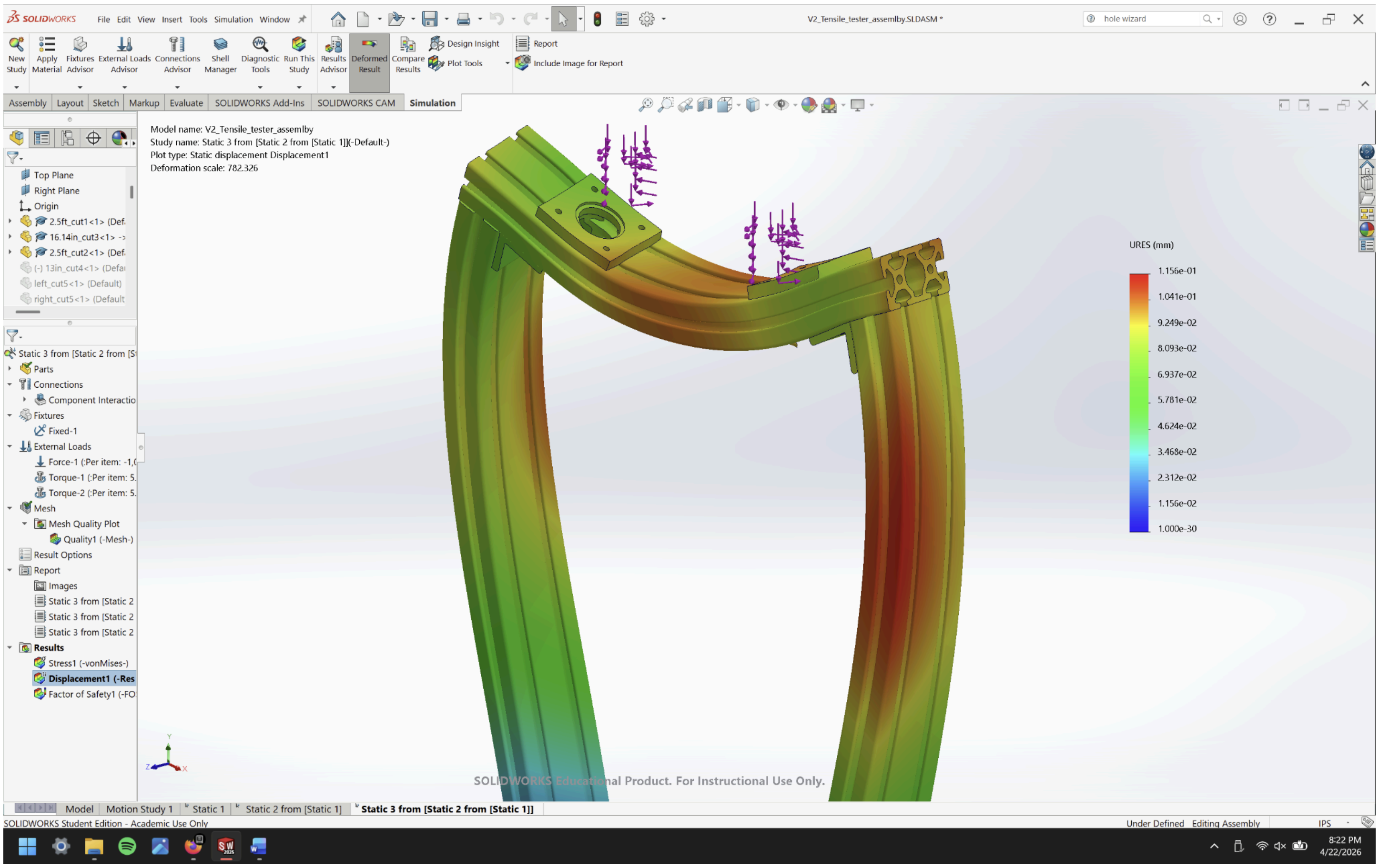


The second optimization strategy was to run the next simulation with the coarsest mesh size around the holes of the lead screw/coupler. The finer mesh should reduce the blown up stress concentrations from the previous analyses and reduce the maximum stress and increase the maximum safety factor. This change can be seen below.

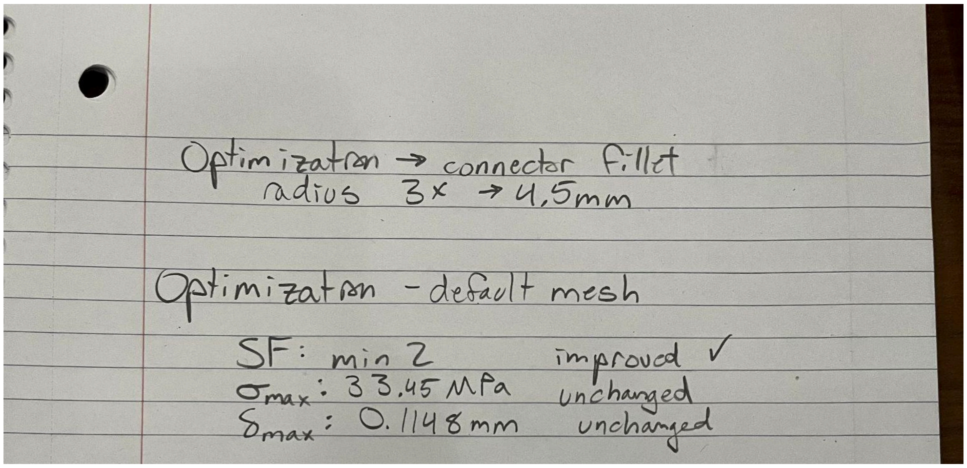


The displacement for the frame was nearly the same as our minimum

With these optimization changes made, we ran a final FEA analysis to see if we improved the values after reviewing the model results, we found that the minimum safety factor was increased to 2, the max displacement stayed the same, and the max stress stayed roughly the same. We can conclude that the important results include the displacement in the center of the top beam which relates the vertical location where the grip line would be of the tensile tester. This can be seen below.



There is also high displacement from twisting in the outer edges of the vertical beams, but all near 0.1mm. The twist was in the direction of the torque generated by the two motors. The max von Mises stress was no longer peaking around the corner connectors, but it was still high within the top beam at the through hole cuts. The concentration point was not as visible as before, but it was still resulting in a high peak stress. Ultimately, the stress was still well below our material yield strength, displacement was close enough to our target, and the safety factor was improved and only lowered due to stress concentrations that would likely not be as high in actuality. With this we can conclude that the optimization pass was successful.



Optimization → connector fillet radius 3x → 4.5mm

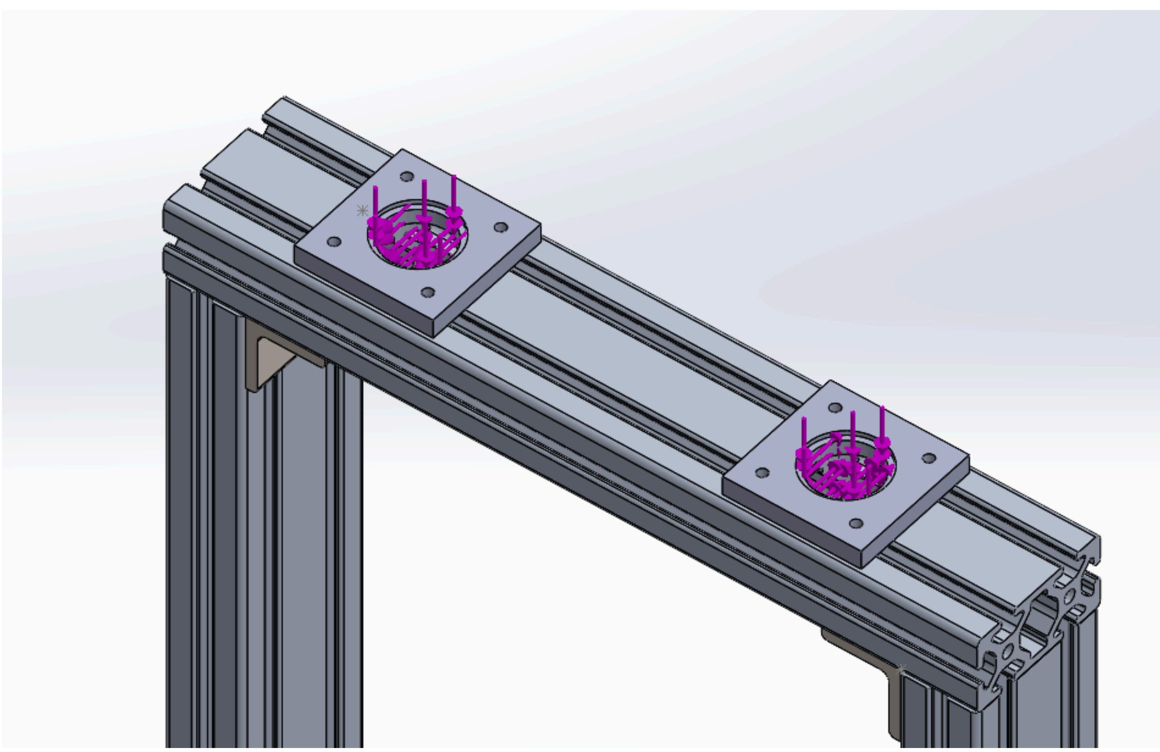
Optimization - default mesh

SF: min 2 improved ✓
σ_{max}: 33.45 MPa unchanged
δ_{max}: 0.1148 mm unchanged

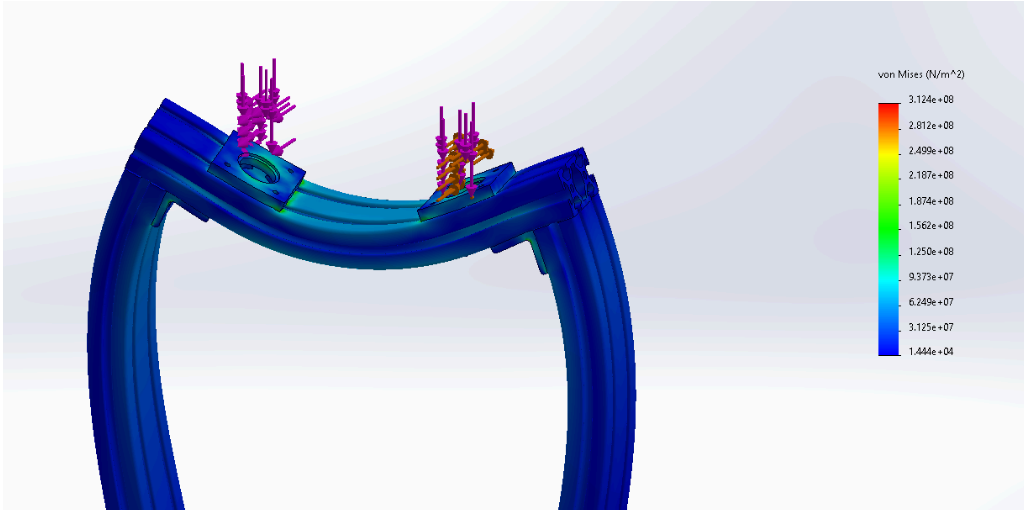
In the downloads section there will be two word document files that contain the SolidWorks Reports for the fine mesh simulation before optimization and then a report for the post optimization simulation. This shows greater detail of the aforementioned results with more information about material, setup, mesh information, and plenty of preprocured images.

Vertical Beam Verification (FEA)
Team member: Kenny

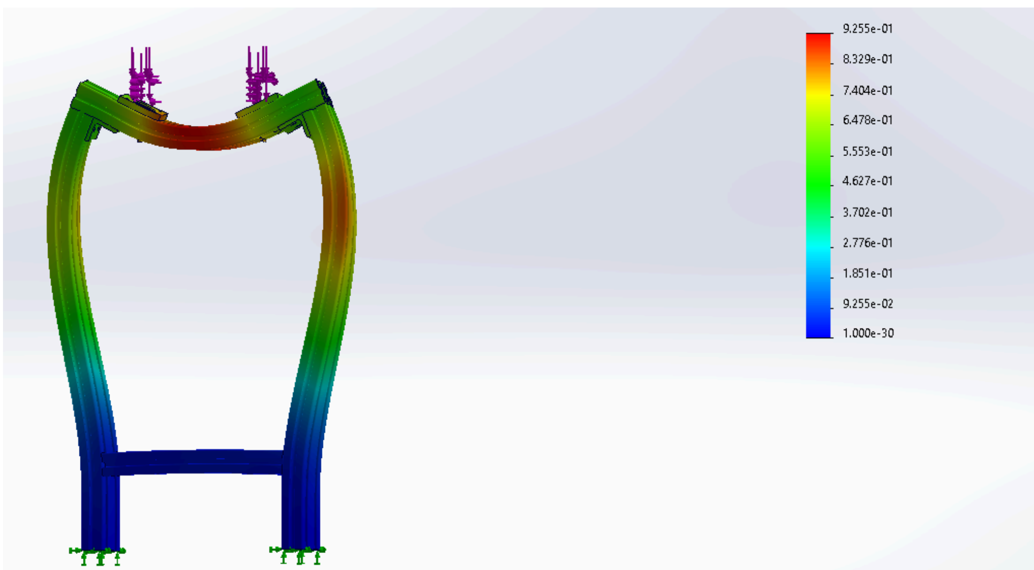
The last and final part was the 2nd portion of the FEA. I was tasked with measuring the vertical beam and verifying its deflection, stress, and safety factor. I started by setting the parameters such as the force inside of the motor mount and coupler to account for the tangential force. I made the tangential forces face the same direction which is a key factor that will be talked about later as well. After calculating the tangential force to be 318N, I started my analysis below.



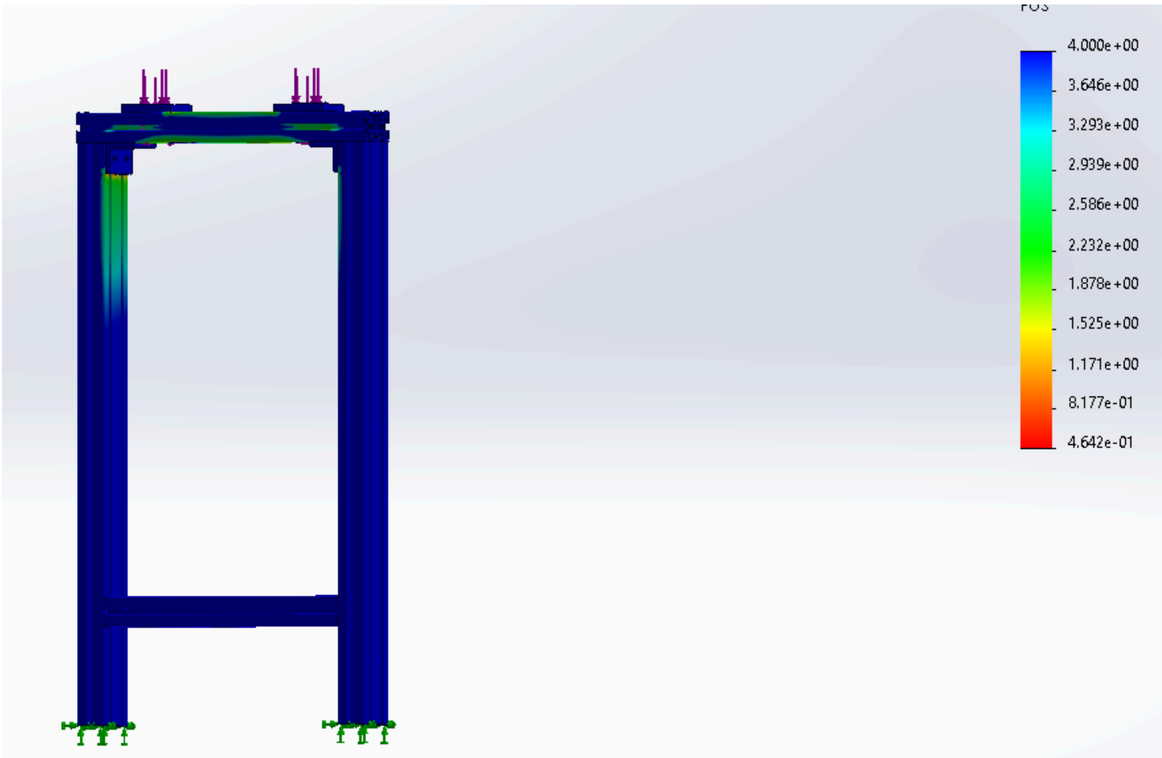
The image describes the forces that are affecting the vertical beams which include the tangential and axial loading.



The first test that was measured was the stress on the beams which came out to be 314 Mpa. There was a slight issue with this data as it was almost double of the yield strength that we measured for the first portion of the FEA. Luke found that our design had a yield strength of about roughly 134Mpa so if my stress value was almost double the yield then there was definitely some issues with the connections of the beam.



Next, was the deflection which came out to be about 0.9255mm. The deflection stayed pretty consistent which allowed for it to converge well where we needed it to be at.



The final measurement was the factor of safety. This value came out to be 0.46 or 0.5. Although not included, the factor of safety tended to stay the same throughout the 3 mesh trials. This is a good thing for us as we had consistent values and the values converged in the end.

Part 6 FEA

Vertical Frame Verification

* Mesh 1 (coarser then default)

δ_{max} = 1.966 mm (center top beam)

σ_{max} = 260 MPa (connectors)

SF = 0.2197

Mesh 2 (default mesh)

δ_{max} = 1.967 mm (center top beam)

σ_{max} = 333 MPa (connectors)

SF = 0.2175

* Mesh 3 (finer then default)

δ_{max} = 1.967mm

σ_{max} = 313 MPa

SF = 0.2132

Convergence:

δ_{max}: $2((|1.967 - 1.966|)/(1.966 + 1.967)) = 5.09\times10^{-4} \times 100$ 0.05%

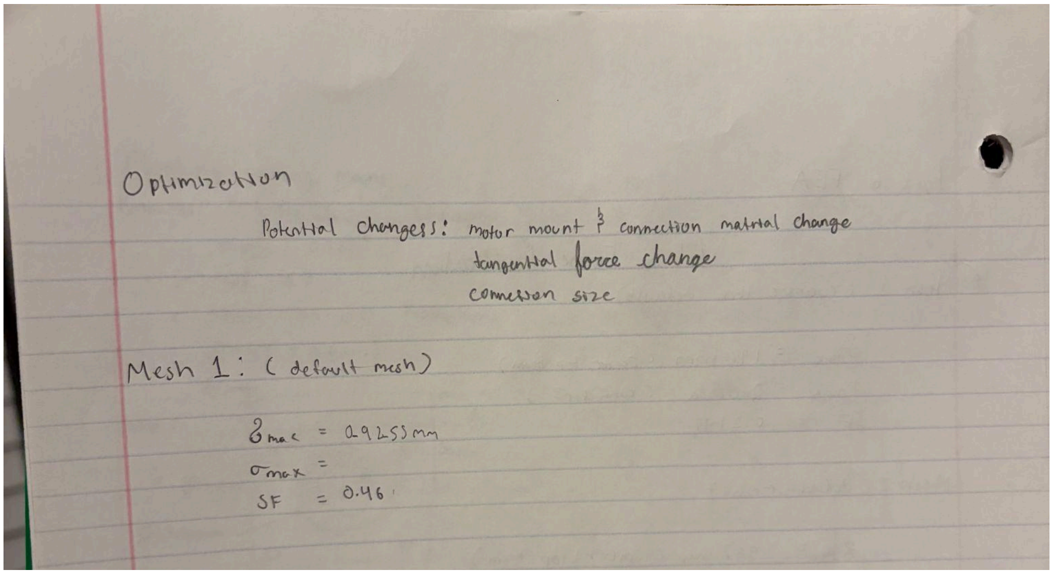
σ_{max}: $2((|313 - 260|)/(260 + 313)) = 0.18 \times 100 = 18\%$

SF: $2((|0.2132 - 0.2197|)/(0.2197 + 0.2132)) \approx 0.03 \times 100 = 3\%$

Converges: δ_{max} & SF

Diverges: σ_{max}

The values for each mesh trial are recorded above. After performing three trials, I took the first mesh and third mesh and calculated the convergence with the formula Covergence%=2((|xf-xi|/(xi+xf))x100. After the percentages, I found that the maximum deflection and safety factor turned out to converge while the maximum stress diverges.



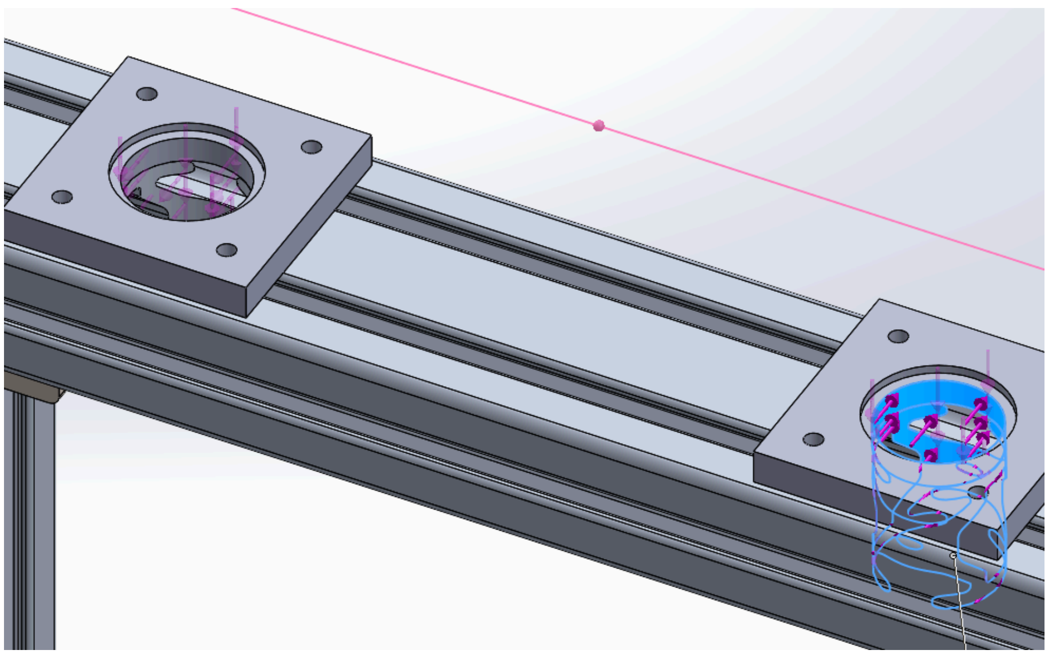
Optimization

Potential changes: motor mount & connection material change
tangential force change
connection size

Mesh 1: (default mesh)

δ_{max} = 0.9253mm
σ_{max} =
SF = 0.46

For optimization my teammate and I concluded that we needed to change the tangential forces so they can cancel each other out and to change the connection material to a steel. I selected an Alloy Steel to hopefully impact and improve the high stress that we measured. The word document has many of the values that are needed to fully grasp the optimization so if you look through it you will find all of our mesh values and elements that were found.



For optimization, again we changed the tangential force direction. This impacted the stress and deflection. There wasn't much twist within the vertical stress so it can be concluded that there were no shifts in critical locations. The critical locations mostly stayed the same, the only change was how impactful each measurement was towards the beam.

Design Floating Bearing Mount

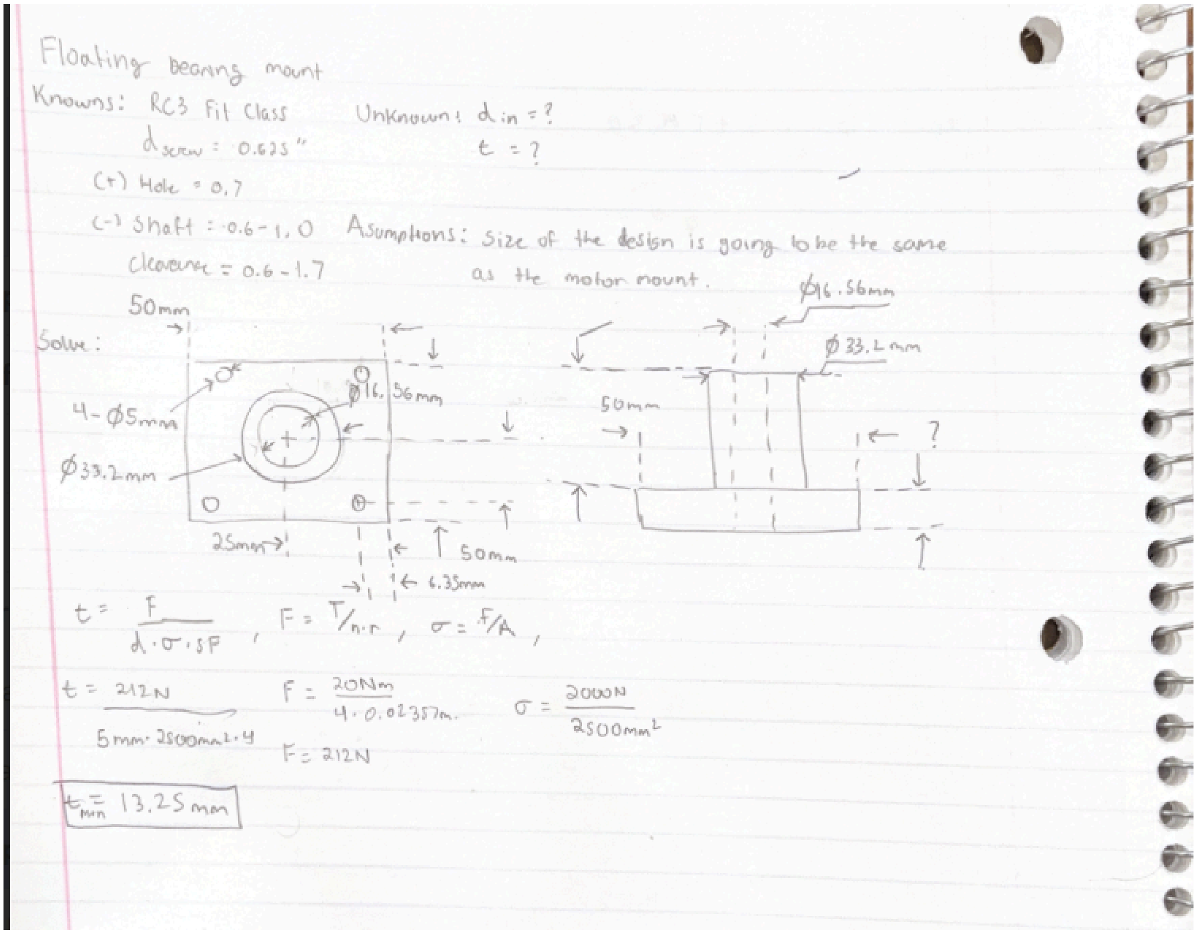
Team member: Kenny Lee

The last part of the design is the floating bearing mount. Again, I wrote down all of my knowns which included the lead screw diameter and the RC3 Fit Class. With a lead screw diameter of 0.625, this gives us the tolerances for the hole to be 0.7, shaft to be 0.6-1 and then a clearance of 0.6-1.7. Based on the machinery's handbook page 654 I found the tolerances for our diameter of 0.625 down below.

Table 8a. American National Standard Running and Sliding Fits *ANSI/ASME B4.1-1967 (R2020*

Nominal Size Range, Inches	Class RC 1			Class RC 2			Class RC 3			
		Standard Tolerance Limits			Standard Tolerance Limits			Standard Tolerance Limits		
	Clearance[a]	Hole H5	Shaft g4	Clearance[a]	Hole H6	Shaft g5	Clearance[a]	Hole H7	Shaft f6	Clearance[a]
Over To	Values shown below are in thousandths of an inch									
0 - 0.12	0.1	+0.2	−0.1	0.1	+0.25	−0.1	0.3	+0.4	−0.3	0.3
	0.45	0	−0.25	0.55	0	−0.3	0.95	0	−0.55	1.3
0.12 - 0.24	0.15	+0.2	−0.15	0.15	+0.3	−0.15	0.4	+0.5	−0.4	0.4
	0.5	0	−0.3	0.65	0	−0.35	1.2	0	−0.7	1.6
0.24 - 0.40	0.2	+0.25	−0.2	0.2	+0.4	−0.2	0.5	+0.6	−0.5	0.5
	0.6	0	−0.35	0.85	0	−0.45	1.5	0	−0.9	2.0
0.40 - 0.71	0.25	+0.3	−0.25	0.25	+0.4	−0.25	0.6	+0.7	−0.6	0.6
	0.75	0	−0.45	0.95	0	−0.55	1.7	0	−1.0	2.3
0.71 - 1.19	0.3	+0.4	−0.3	0.3	+0.5	−0.3	0.8	+0.8	−0.8	0.8
	0.95	0	−0.55	1.2	0	−0.7	2.1	0	−1.3	2.8
1.19 - 1.97	0.4	+0.4	−0.4	0.4	+0.6	−0.4	1.0	+1.0	−1.0	1.0
	1.1	0	−0.7	1.4	0	−0.8	2.6	0	−1.6	3.6
1.97 - 3.15	0.4	+0.5	−0.4	0.4	+0.7	−0.4	1.2	+1.2	−1.2	[illegible]
	1.2	0	−0.7	1.6	0	−0.9	3.1	0	−1.9	[illegible]
3.15 - 4.73	0.5	+0.6	−0.5	0.5	+0.9	−0.5	1.4	+1.4	−1.4	[illegible]
	1.5	0	−0.9	2.0	0	−1.1	3.7	0	−2.3	[illegible]
4.73 - 7.09	0.6	+0.7	−0.6	0.6	+1.0	−0.6	1.6	+1.6	−1.6	[illegible]
	1.8	0	−1.1	2.3	0	−1.3	4.2	0	−2.6	[illegible]
7.09 - 9.85	0.6	+0.8	−0.6	0.6	+1.2	−0.6	2.0	+1.8	−2.0	2.0
	2.0	0	−1.2	2.6	0	−1.4	5.0	0	−3.2	6.6
9.85 - 12.41	0.8	+0.9	−0.8	0.8	+1.2	−0.8	2.5	+2.0	−2.5	2.5
	2.3	0	−1.4	2.9	0	−1.7	5.7	0	−3.7	7.5
12.41 - 15.75	1.0	+1.0	−1.0	1.0	+1.4	−1.0	3.0	+2.2	−3.0	3.0
	2.7	0	−1.7	3.4	0	−2.0	6.6	0	−4.4	8.7
15.75 - 19.69	1.2	+1.0	−1.2	1.2	+1.6	−1.2	4.0	+2.5	−4.0	4.0
	3.0	0	−2.0	3.8	0	−2.2	8.1	0	−5.6	10.5

Again, on page 654 of the Machinery's handbook, I found the RC3 values that I needed to make sure the floating bearing mount that I made was within the given tolerance.



Here are the calculations for the floating bearing mount. Again, I listed my knowns and unknowns to find an equation that could help me. I assumed I could re-use some of the data and equations from the motor mount since the design is relatively the same. The only thing I changed was the diameter of the screws which is 5mm. The height of the cylindrical extrude came out to be 25mm. I assumed the height was negligible for the FEA and negligible in general for the tensile tester. The most important part of the design was adding the tolerances to the different diameters. For the outside diameter, I took off 0.800mm because of the RC3 fit. For the inside diameter, I added 0.7mm for a tighter clearance for the lead screw. After that, I had everything I needed to start the design.

Bill of Materials and Labor Costs

Bill of Materials					
Item Number	Quantity	Name	Cost/Item	Cost	Purchase Link
1	2	Carbon Steel Acme Lead Screw Right Hand, 5/8"-8 Thread Size, 2 Feet Long	$9.79	$19.58	https://www.mcmaster.com/98935A914/
2	2	932 Bearing Bronze Acme Flange Nut	$138.20	$276.40	https://www.mcmaster.com/95120A119/

		Right Hand, 5/8"-8 Thread Size			
3	1	T-Slot Framing System: Rectangle, 40 Series, Smooth, Double, 3 m Nom Lg, 80 mm x 40 mm, 6 Slots, Std	$270.09	$270.09	https://www.grainger.com/product/80-20-T-Slot-Framing-System-Rectangle-804T48
4	2	Set Screw Shaft Coupling Black-Oxide 1215 Carbon Steel, for Keyed 14mm Diameter Shaft	$ 74.49	$ 148.98	https://www.mcmaster.com/3613N15/
5	2	Inside Corner Bracket: Wide Inside-Corner Bracket, 40 Series, 40 mm x 76 mm x 40 mm, 4 Holes, Silver	$ 9.68	$ 19.36	https://www.grainger.com/product/80-20-Inside-Corner-Bracket-Wide-5JRU7
6	8	Hammer Nut: M5 Size, For 8 mm Slot Wd, 15 Series/40 Series, Single Center, Zinc-Plated, Steel, 4 PK	$ 3.66	$ 29.28	https://www.grainger.com/product/FATH-INC-Hammer-Nut-M5-Size-55MT26
7	2	Nema 23 Stepper Motor L=55mm Gear Ratio 15:1 High Precision Planetary Gearbox	$ 91.84	$ 183.68	https://www.omc-stepperonline.com/nema-23-stepper-motor-l-56mm-gear-ratio-15-1-high-precision-planetary-gearbox-23hs22-2804s-hg15
8	4	Inside Corner Bracket: Inside-Corner Bracket, 40 Series, 40 mm x 36 mm x 40 mm, 2 Holes, Silver	$ 6.83	$ 27.32	https://www.grainger.com/product/80-20-Inside-Corner-Bracket-Inside-5JRP7
9	2	Designed Motor Mounts	No Quote Available	No Quote Available	—
10	1	Designed CrossHead	No Quote Available	No Quote Available	—
11	2	Designed Floating Journal Bearings	No Quote Available	No Quote Available	—
12	2	Custom Top Inside Corner Brackets	No Quote Available	No Quote Available	—
13	—	Various Length M5 Hex Bolts	—	—	—
			Total Cost:	$ 974.69	

Labor Cost			
Engineering Student	Hours Worked	Cost/Hour	Cost
Part 1: Lead Screw Design			
Luke DeVries	6.5	$35/hr	$ 227.50
Kenny Lee	4	$35/hr	$ 140.00
Part 2: Motor Selection Design			
Luke DeVries	17.5	$35/hr	$ 612.50
Kenny Lee	16	$35/hr	$ 560.00
		Total Cost:	$ 1540.00

Total Cost of Project: $ 2,514.69

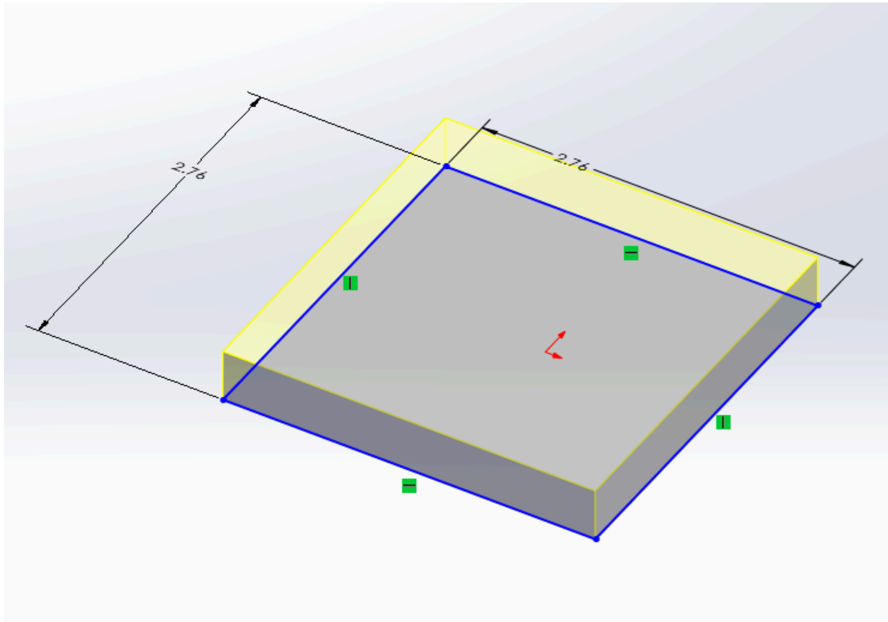
Parametric Modeling

Motor Mount

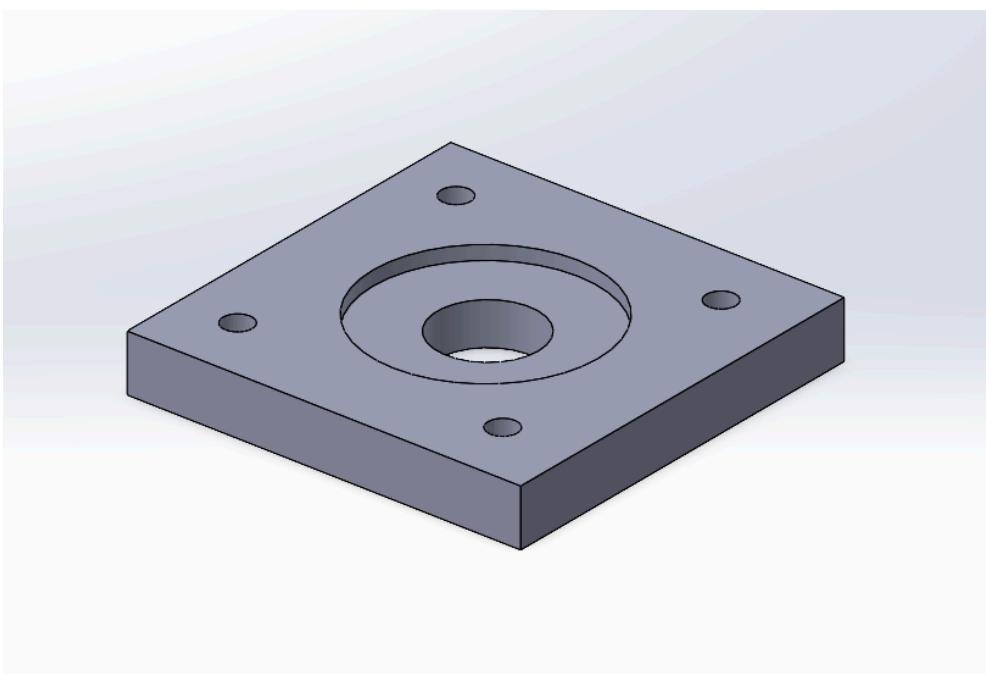
Team member: Kenny Lee

Name	Value / Equation	Evaluates to	Comments
Global Variables			
"Torque"	= 20	20.00	Nm
"Safety Factor"	= 4	4.00	
"n"	= 4	4.00	
"Bearing stress"	= 0.408	0.41	N/mm^2
"db"	= 5.2	5.20	mm
"t"	= ("F") / ("db" * "Bearing stress" * "Safety Factor	25.00	mm
"r"	= 0.02357	0.02	m
"F"	= ("Torque") / ("n" * "r")	212.13	N
Add global variable			
Features			
Add feature suppression			
Dimensions			
D1@Sketch1	2.76in	2.76in	
D2@Sketch1	2.76in	2.76in	
D1@Boss-Extrude1	0.39in	0.39in	
D2@Sketch2	0.2in	0.2in	
D1@Sketch2	0.93in	0.93in	
D5@Sketch2	1.86in	1.86in	
D3@Sketch2	2	2	
D4@Sketch2	2	2	
D1@Sketch3	1.57in	1.57in	
D2@Sketch3	1.38in	1.38in	
D3@Sketch3	1.38in	1.38in	
D1@Cut-Extrude2	0.1in	0.1in	
D1@Sketch4	0.71in	0.71in	
D2@Sketch4	1.38in	1.38in	

Listed above are the dimensions in inches for the motor mount. Also listed are the equations I listed in millimeter and meter where I used to calculate the thickness of the mount. I tried to CAD with the minimum of 25mm but the design of it was too big so I assumed that I could rely on my 10mm thickness like I had already planned.

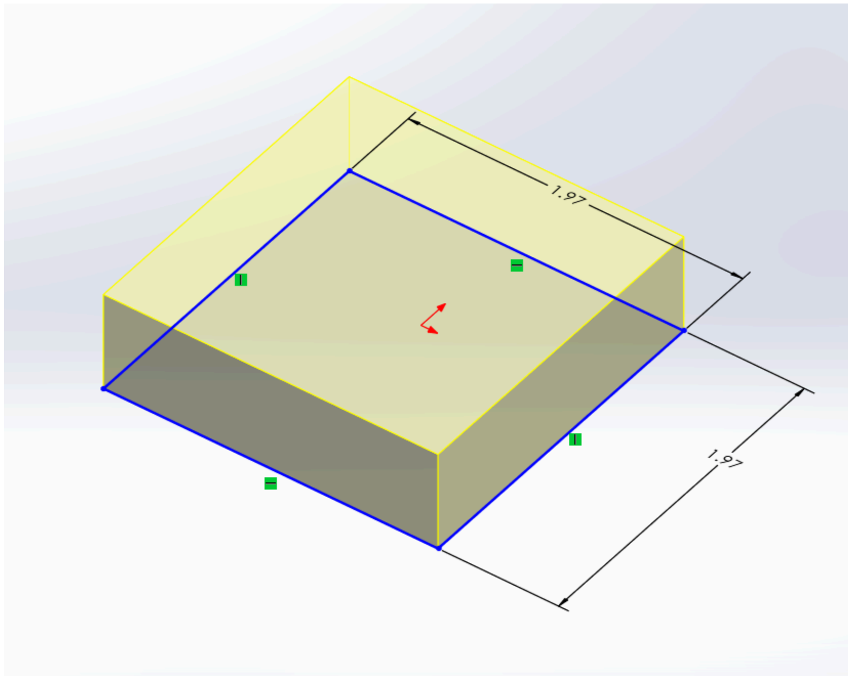


After designing, and validating my thickness I began to sketch my chosen area of 70mmx70mm. I am not too familiar with how to convert mm to inches within a modeling session so all of my millimeter values were automatically converted to inches by default. The next steps were to extrude the thickness value I choose by a value of 10mm to create the final design shown below.

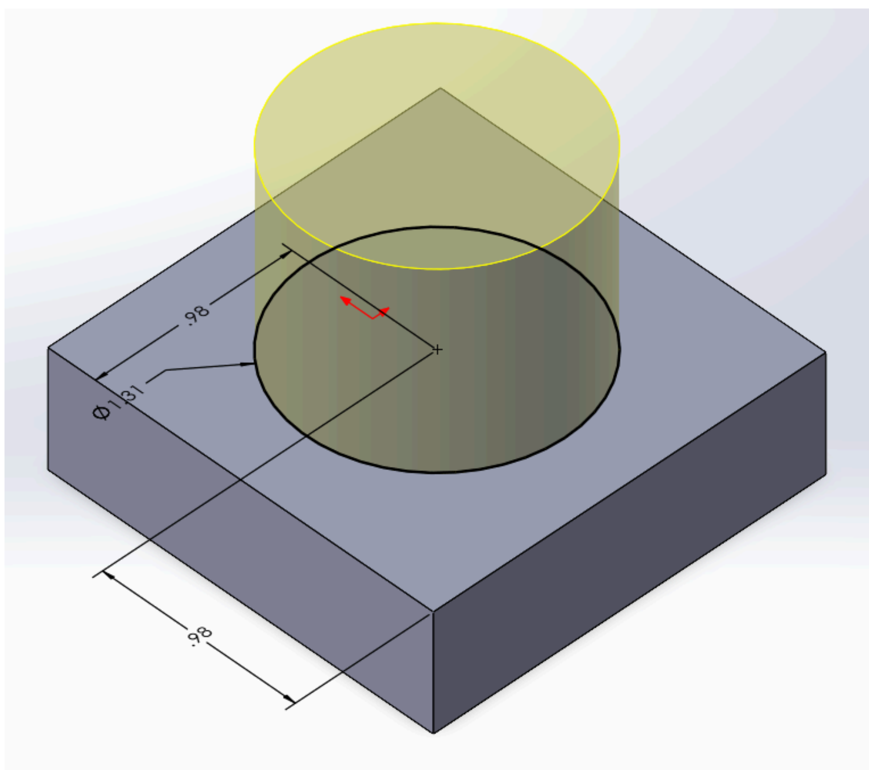


This design accounts for the 4 5.2mm bolts that are in each corner of the motor, the shaft diameter of 16mm, and the diameter of the support under the motor. There isn't anything extreme about the design, it gets the dimensions done and supports the motor well enough.

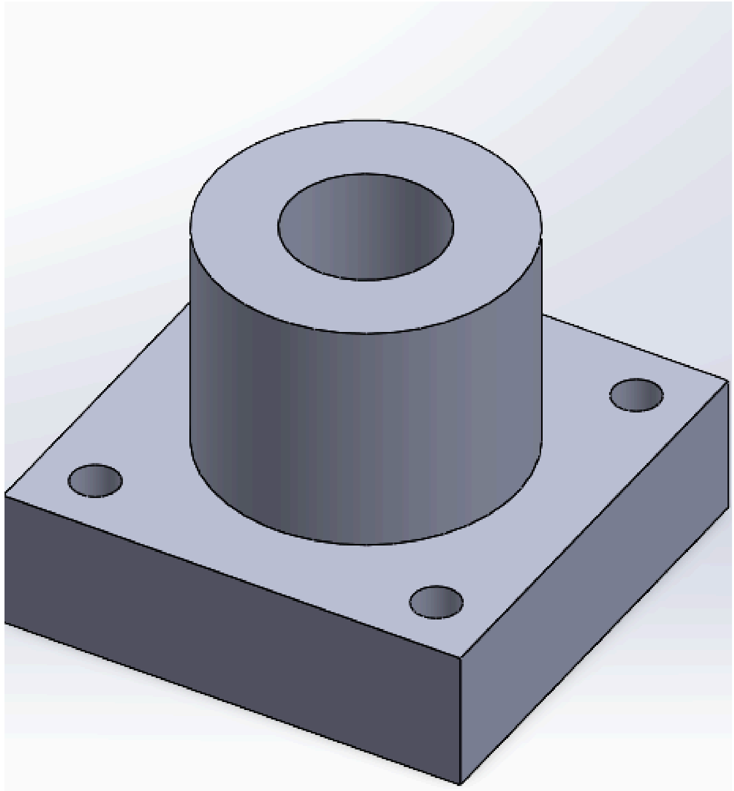
Floating Bearing Mount
Team member: Kenny Lee



The design of the floating bearing mount is relatively the same. With a couple of assumed dimensions for the sketch and height, the measurements turned out to be about 50mmx50mm with a height of 25mm which can be shown below.



After the extrusion of the base, the shaft diameter was a simple extrude that compared to the dimensions that I calculated in the design tab above.



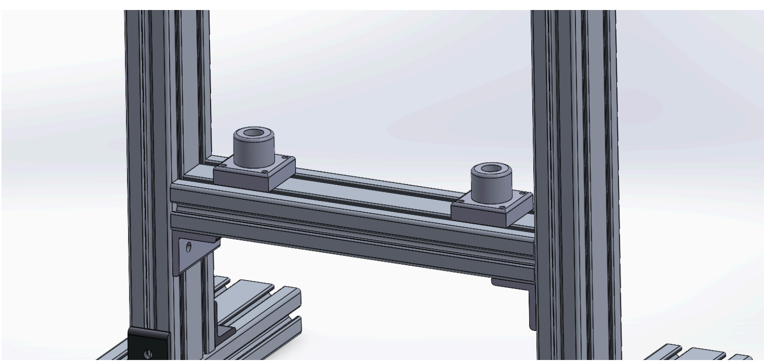
The final design is finally made, with extrusions being used throughout the model. Most of the measurements were taken from the calculations so please refer back to the paper work and sketch this is based off of.

Full Tensile Tester Assembly
Team member: Luke

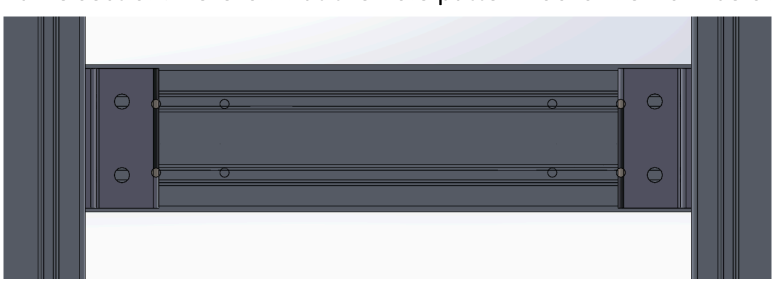
The first step of the assembly was the base frame.

Following the frame, I added all the necessary supporting corner brackets.

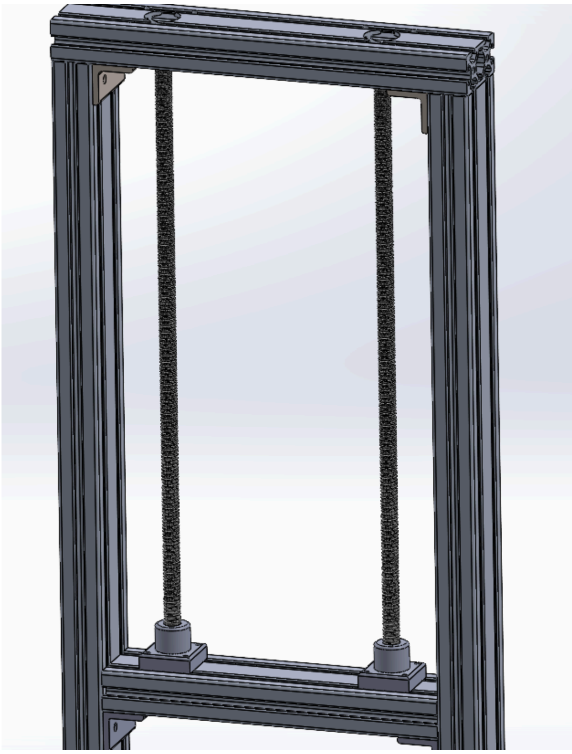
One important note at this stage is that the assembly does not include the hammer nuts or the M5 screws. There is an available CAD model download for the selected hammer screws, but adding all of them to the model is quite tedious and they are not visible from the majority of angles. Similarly, this assembly uses a variety of M5 hex bolts with differing lengths. These are possible to include in the assembly, but become quite intensive while not affecting the overall design in a significant way. For these reasons, we decided to intentionally leave out the hammer nuts and M5 screws for time saving reasons. Following the corner brackets, I added the designed floating bearing mounts.



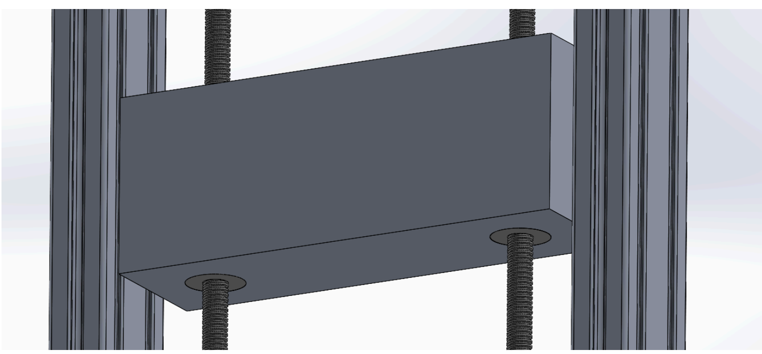
With the addition of the journal bearings, we needed to add through holes for the bolts to pass through the bottom frame section. Here is what the hole pattern looks like from below the bottom frame.



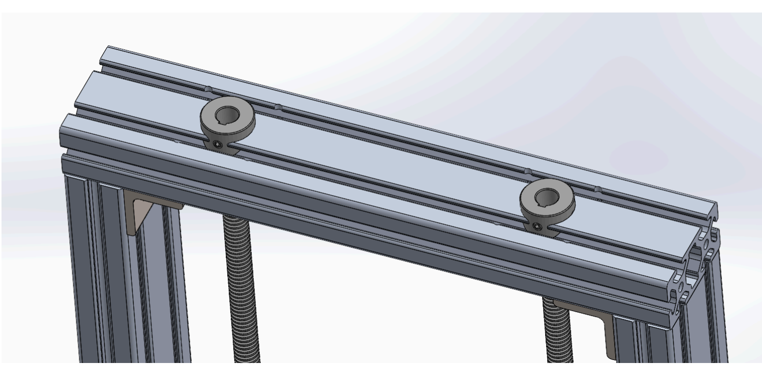
Next, we have the lead screws positioned in their correct locations.



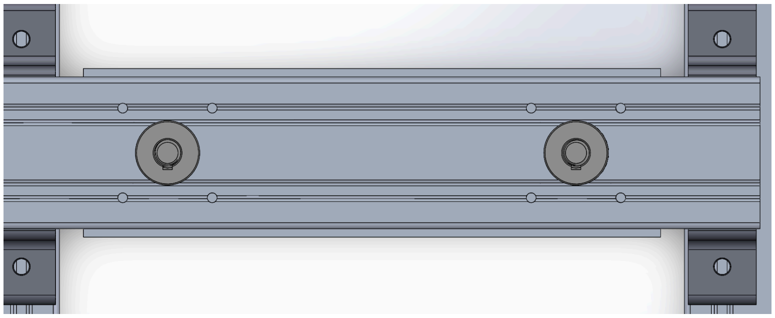
On each lead screw, there is a flange nut, in the model these are both added using a screw mate so that the model dynamically reacts to the movement of the crosshead/lead screws when using the drag tool. Additionally, a motion study could capture a short animation of the tensile tester as the crosshead moves up and down. Next, we add the designed crosshead.



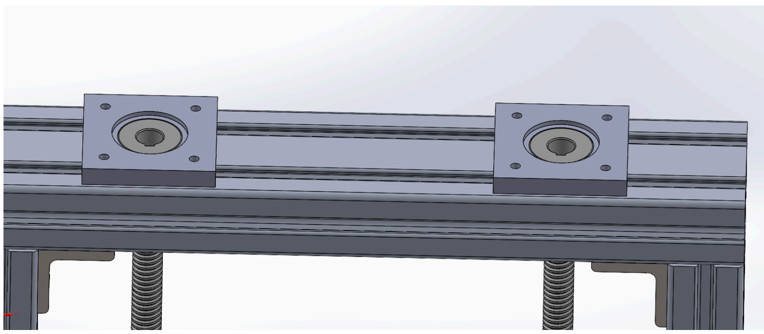
At the top of the frame, we can add the Couplers that attach to the lead screws and connect them to the output shaft of the motor.



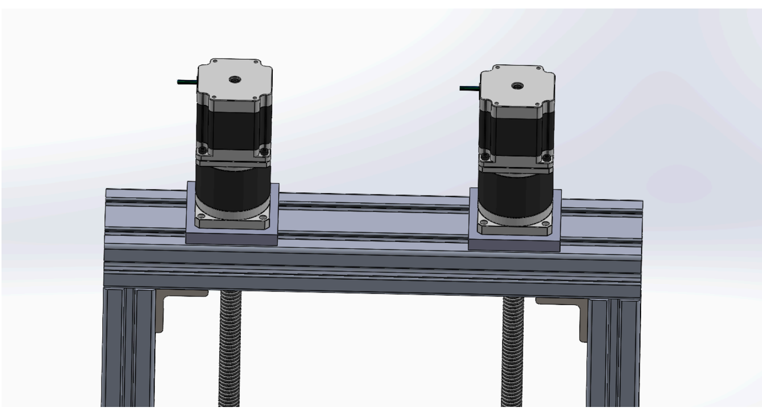
We will need to fasten the motor mounts to the frame, so again we need to utilize the bolt pattern on the motor mounts to create holes on the top frame piece. Here is an image of that bolt pattern.



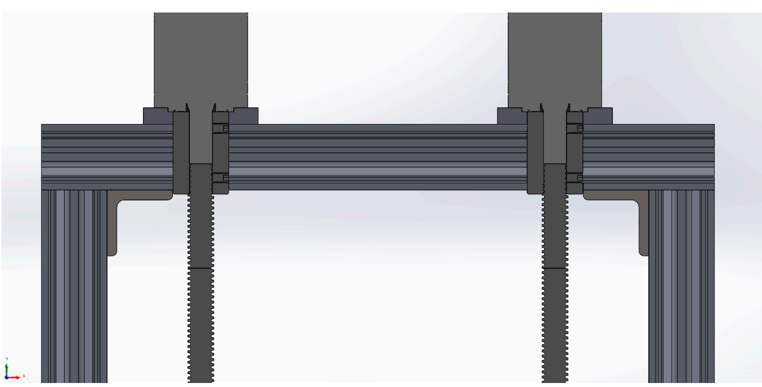
Next we can add the designed motor mounts.



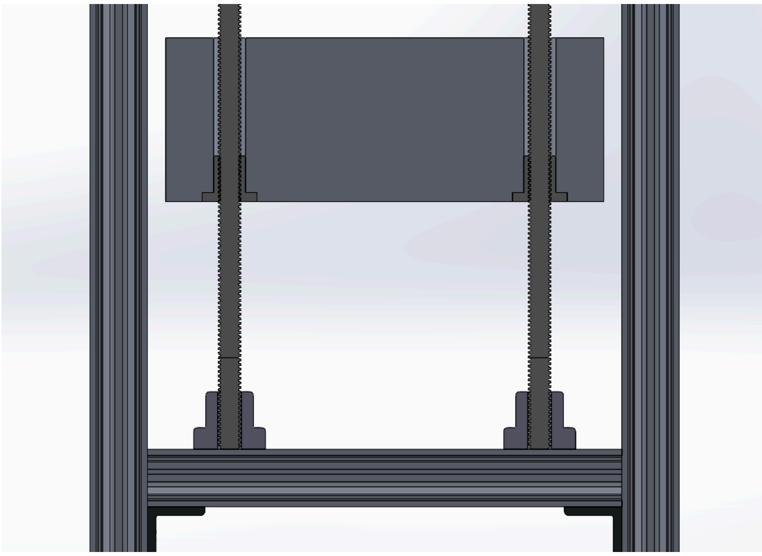
Finally, we can add the two motors to the top motor mounts.



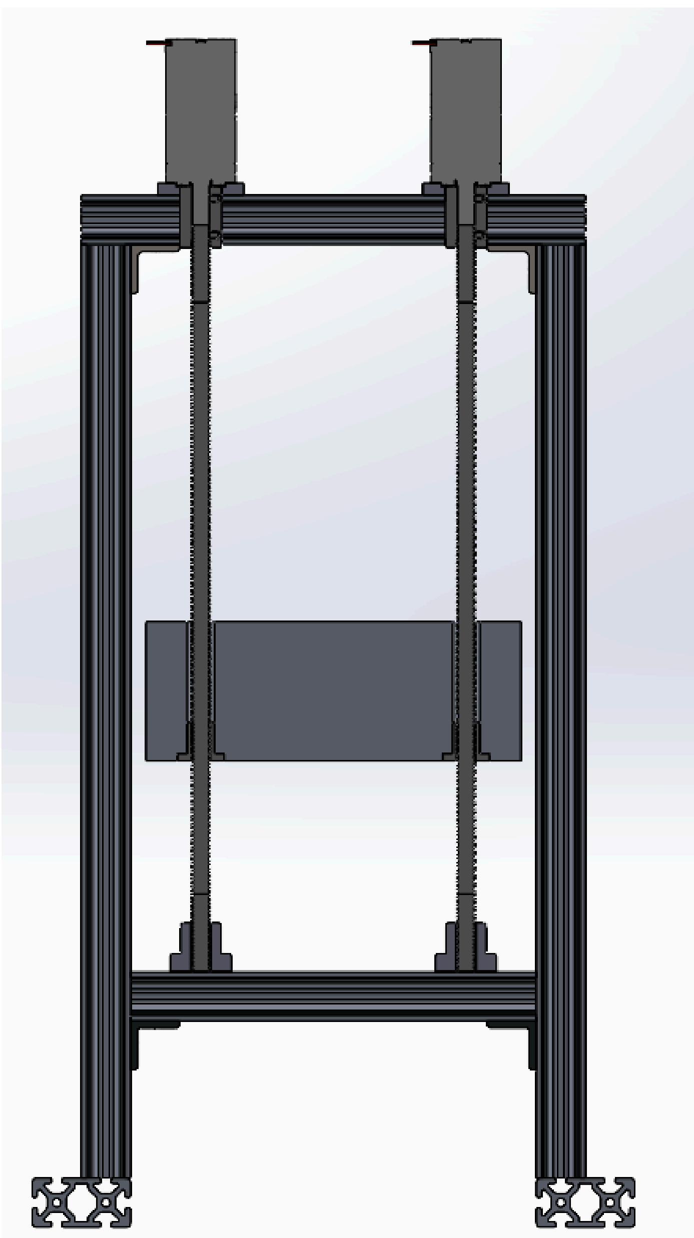
In addition to these photos of the assembly, there are images of the cross sectional view to better understand the inner connections within the frame and the crosshead. First is a cross sectional view of the motor-lead screw connection.



The next section view is of the crosshead and lower portion of the frame.



Finally, we have a full cross sectional view followed by the completed assembly render.



The final note for the full assembly is the lack of load cell and grips for the specimen. Due to time constraints, we did not find a model load cell or design the connection geometry for the grips that would hold the tensile test specimen. This was not explicitly asked for the assignment description, so we feel it is fair to leave this off of our designed assembly. There should be sufficient room to include the load cell and grips within the working portion of our designed tensile tester.

Lessons Learned

Kenny Lee

This assignment made me try harder in different areas of engineering I never thought I would have done before. From the extensive research to the intricate FEA optimization, there was a lot of problem solving and also teamwork that really helped me finish this project up. The biggest lesson I learned from this project was teamwork. Having a teammate really helped out with filling the gaps of my knowledge with theirs. With problem solving sessions as well, having another brain to help think with you helps the project go smoother. I also learned the importance of FEA and how the convergence plays a key role in optimizing your model. When optimizing your model, there is a lot of extra thought into what could be changed, affected, accepted, and most importantly, how much more time it would take to optimize the model. Overall, this project took what I knew and how I solved problems and further solidified my understanding of each and more areas of engineering such as advanced CAD modeling, intensive research, and the importance of teamwork. This assignment took about 15 hours to complete.

Luke DeVries

This assignment has pushed me to learn a lot about CAD assemblies and utilizing patterns/sketches between different parts. I also gained more experience through FEA setup and analysis as well as a better understanding of how to go about optimization and test the results. This assignment as a whole has pushed me a lot deeper into understanding design intent and the importance of research to guide information that is otherwise not provided. Many steps in this assignment had multiple unknown variables and/or vague descriptions that forced me and Kenny to utilize various resources to guide our design intent and process. I learned the importance of utilizing already existing information and real world tensile testers to piece together information that would help in our design and assembly. There were also many instances where the design had to be verified using pre-existing parts which meant that the design calculations were often constrained to the available resources and/or available information. This added up to a substantial bill of materials and taught me the importance of being resourceful in part selection while making sure not to compromise safety or design requirements. Overall, this assignment was a great culmination of all previous assignments as a true test of the design process and the ability to work alongside another engineering student. In total, the motor selection design half of the tensile tester project took me a total of ~17.5 hours of work.

Download Links:

- Complete handwritten work .pdf by Luke: here
- SolidWorks Report for the Top Beam Verification Pre-Optimization .docx: here
- SolidWorks Report for the Top Beam Verification Post-Optimization .docx: here
- SolidWorks Report for the Vertical Frame Verification Pre-Optimization .docx: here
- SolidWorks Report for the Vertical Frame Verification Post-Optimization .docx: here
- Full Tensile Tester Assembly .zip (direct download): here
- Full Tensile Tester Assembly .zip (Google Drive Download): here