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.

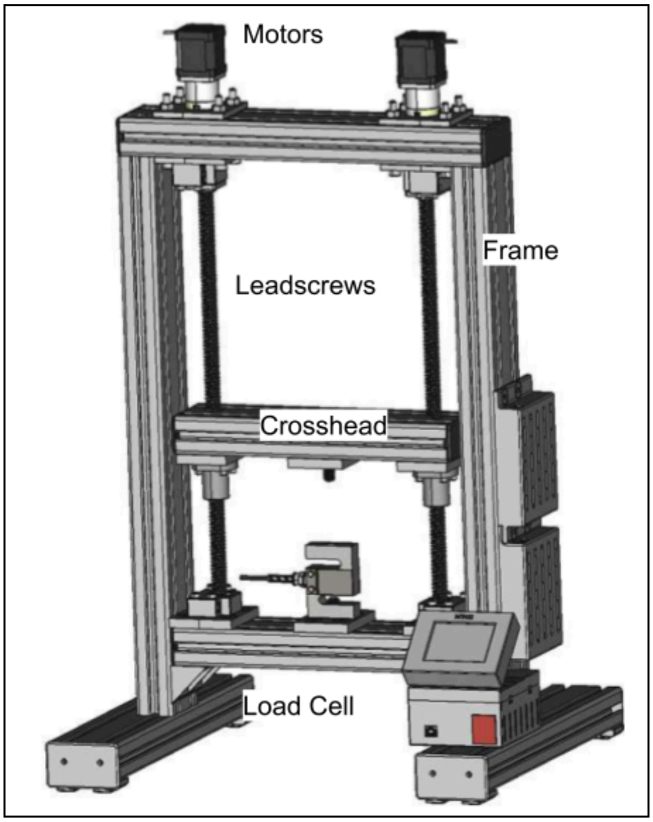


Figure #1. Concept of the Tensile Tester

Design and build a mechanically driven tensile tester capable of applying up to 2 kN axial tensile load with a controllable crosshead speed from 1 to 50 mm/min. Each team is responsible for the mechanical components outlined below. The system will provide consistent, well-aligned loading for polymers and light metals, suitable for instructional use and basic materials evaluation.

Part 1 Scope of Work

- Develop a **lead-screw–driven crosshead**
- Ensure **accurate, repeatable load application** to 2 kN with minimal crosshead deflection and misalignment.
- Provide **robust specimen gripping**, simple calibration provisions (mechanical), and safe, ergonomic operation.

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

Lead Screw Diameter

Team member: Luke DeVries

Determining the Minimum Diameter
To start the lead screw selection, the minimum lead screw diameter needed to be found. I began by writing down some of the given information from the assignment and then wrote down my known and unknown variables. One alteration I made at the beginning was to set the lead screw length up to 24" instead of the given minimum of 18". I had made this change to provide more space within the tensile tester's dimensions, and later found out that this was also beneficial for the selection of the lead screw since an 18" length is not a standard length to order. One of the unknown variables was the tensile yield strength for the lead screw, which comes from its material properties as well as its dimensions. Since I was calculating the dimensional parameter, I needed to research the material property. Using the information from Roton's website, I was able to find two material types listed for their acme screws, the page can be found at this link. I copied down the information, and the most valuable data was that for lead screws with a diameter less than 1 ¾" the material is a special quality low carbon steel. I then searched for "low carbon steel" on MatWeb.com and went to the overview data for low carbon steel. The page can be found at this link, and the important information was copied down as a tensile yield strength of 20300–347000 psi and a modulus of elasticity of 26500–30900 ksi. I later came back and also needed the metric units, so I copied down the tensile yield strength as 140–240 MPa and the modulus of elasticity as 183–213 GPa. It is also important to not that the following calculations will be using the approximation that for ductile materials, compressive strength is equal to tensile strength. This approximation is stated in the "Notes" section of this webpage by MechaniCalc. Below is the first page of work.

A10 Lead Screw Design

Tensile Tester : 2kN axial tensile load
crosshead speed : 1-50 mm/min
lead screw min : 18 in → 24" → 2' reasonable
SAFETY FACTOR : 4

Leadscrew Diameter

Determining min diameter through axial stress.
★ 2 lead screws, SF=4

known	unknown
2kN maxload	σ_{max}
SF=4	A → diam
L = 24"	$F_{leadscrew}$

www.roton.com/products/acme-lead-screws-nuts/general-information

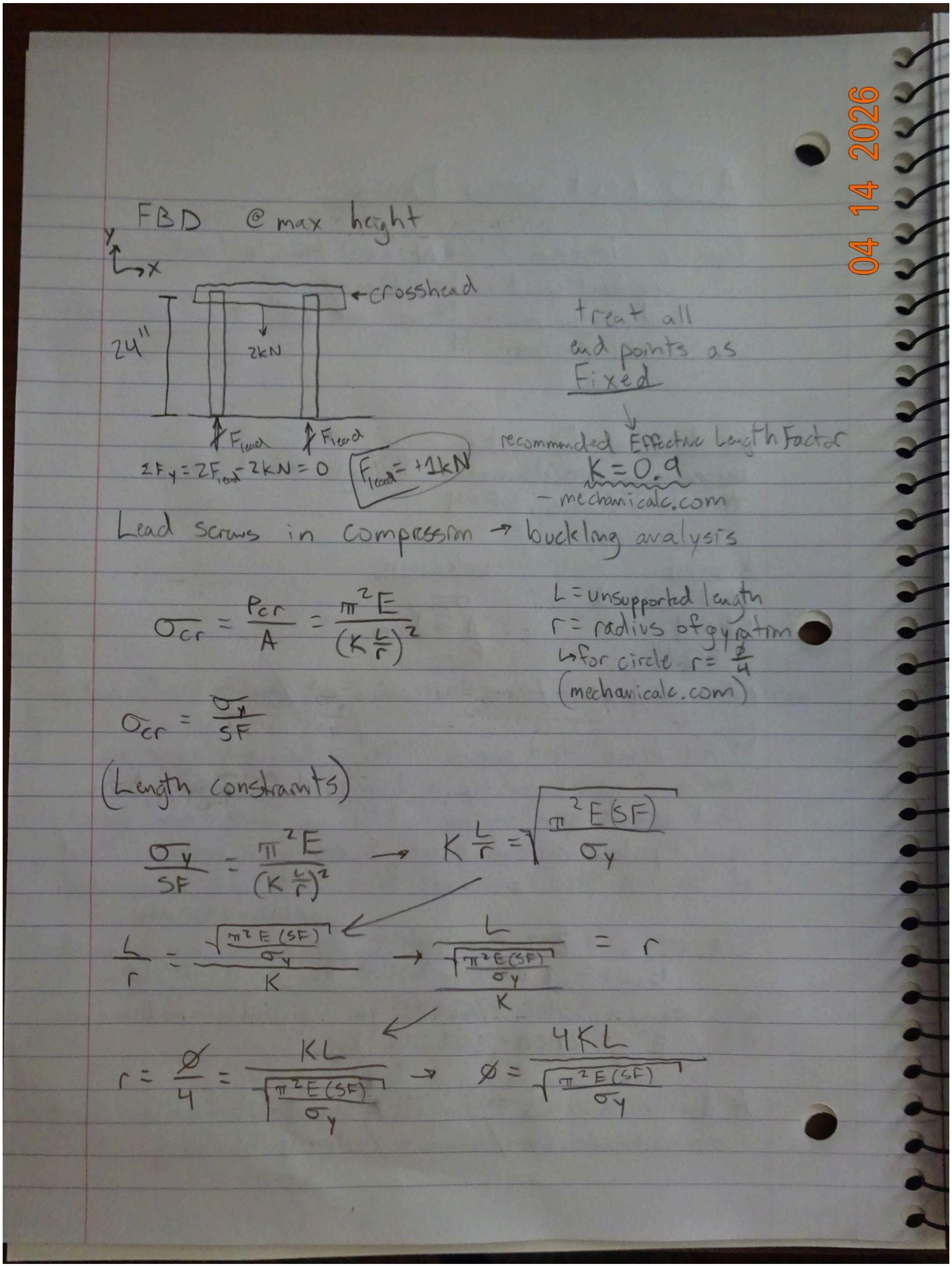
↳ "All Acme lead screws 1 3/4" in diam and larger, are made of special quality medium carbon steel. Acme lead screws smaller than 1 3/4" are made from special quality low carbon steel. Sizes with material listed as "Stainless Steel" are made from 304 type stainless steel."

tensile yield σ_{small} = 20300 - 347000 psi or 140-2400 MPa (matweb - low carbon steel)
E_{small} = 26500 - 30900 ksi or 183-213 GPa

tensile yield σ_{large} = 33400 - 252000 psi (matweb - medium carbon steel)
E_{large} = 27100 - 30900 ksi

★ ductile materials → compressive strength ≈ tensile strength
- mechanicalc.com/reference/column buckling

The next step in the calculations was to draw a free body diagram and algebraically solve the model equations I would be using. The free body diagram allowed me to calculate the force per lead screw as 1 kN, and then I started my analysis of the lead screws under compression/buckling. I treated both endpoints as fixed and got the recommended effective length factor K=0.9 from [MechaniCalc.com](). I also utilized MechaniCalc's [radius of gyration]() for a circle which was equal to the diameter divided by 4. With this, I started algebraically solving the equation that utilized the unsupported length parameter. The final symbolic equation can be seen at the end of the next page of work.



After modeling the equation, I made the assumption that my first round of numerical solutions would assume that the diameter would be less than 1 ¾”. This assumption allowed me to use the minimum values taken from MatWeb for the low carbon steel of the Acme lead screw as stated by Roton. The first minimum calculated diameter was 0.381 inches which is between ⅜” and ½”. I then utilized the other critical buckling stress equation with the load parameter to see if the minimum diameter was different. For this model I chose to utilize metric units since the force was provided as kN. In hindsight, it would have been preferable to convert the load into pounds to reduce the number of conversions I made which are all possible points of rounding error. The calculated value I got was 0.237 inches which was lower than the other minimum diameter. From this I was able to determine that my actual minimum diameter was the first value I calculated of 0.381 inches. I converted this to mm as well just in case I needed to use the metric value when selecting my lead screw. The work for the minimum diameter can be seen below.

Assuming $\phi \le 1\,3/4'' \rightarrow \sigma_y$ small, E_{small}

04 14 2026

$$\phi = \frac{4(0.9)(24\,in)}{\sqrt{\frac{\pi^2(26{,}500{,}000\,psi)(4)}{20{,}300\,psi}}}$$

$$\phi = 0.381\,in$$

$3/8'' < 0.381\,in < 7/16'' \quad < < 1\,3/4''$

(load constraint)

$$\frac{\sigma_y}{SF} = \frac{P_{cr}}{A} \quad . \quad A = \frac{P_{cr}(SF)}{\sigma_y}$$

$$\pi\frac{d^2}{4} = \frac{P_{cr}(SF)}{\sigma_y} \rightarrow \phi = \sqrt{\frac{P_{cr}(SF)4}{\pi\sigma_y}}$$

$P_{cr} = 1\,kN$

$$\phi = \sqrt{\frac{(1{,}000\,N)(4)(4)}{\pi(140\,MPa)}} = 6.03\,mm$$

$$\phi = 6.03\,mm\left(\frac{1\,in}{2.54\,cm}\right)\left(\frac{1\,cm}{10\,mm}\right) = 0.237\,in < 1\,3/4''$$

$\phi_{length} > \phi_{load}$

$$\boxed{\text{minimum diameter} \approx 0.381\,in \text{ or} \approx 9.677\,mm}$$

$$0.381\,in\left(\frac{2.54\,cm}{1\,in}\right)\left(\frac{10\,mm}{1\,cm}\right) = 9.6774\,mm$$

Determining the Range of Rotational Speed Given the Linear Speed

Next, I had to determine the range of rotational speed. Earlier we were provided with a linear speed, so I copied down my known and unknown variables. I also included the minimum diameter from the previous calculations to use here. The linear speed was given in metric units, so I utilized a rounded value of 10mm for the calculation. I drew my free body diagram and found the rotational speed in revolutions per minute. After selecting the lead screw, it may be worth returning to this section and comparing to the TPI values and/or evaluating the rotational speed with Imperial units to see if the same RPM is achieved. The work can be found below for a calculated rotational speed of 0.0318–1.592 RPM.

04 14 2026

Determining range of rotational speed given the linear speed

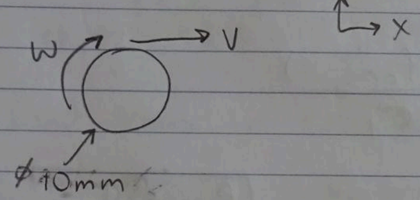
Known

$v = 1\text{–}50\ mm/min$

$\phi_{min} = 9.677\ mm \approx 10\ mm$

Unknown

ω (rpm)



Assumption

diameter rounded from 9.677mm up to a flat 10mm

$$v = \frac{mm}{min} \qquad \omega = \frac{rev}{min}$$

$$1\,rev = \pi\phi\ mm \rightarrow \frac{1\,rev}{\pi\phi\,mm}$$

$$\omega\,(rpm) = v\left(\frac{mm}{min}\right)\left(\frac{1\,rev}{\pi\,\phi\,mm}\right)$$

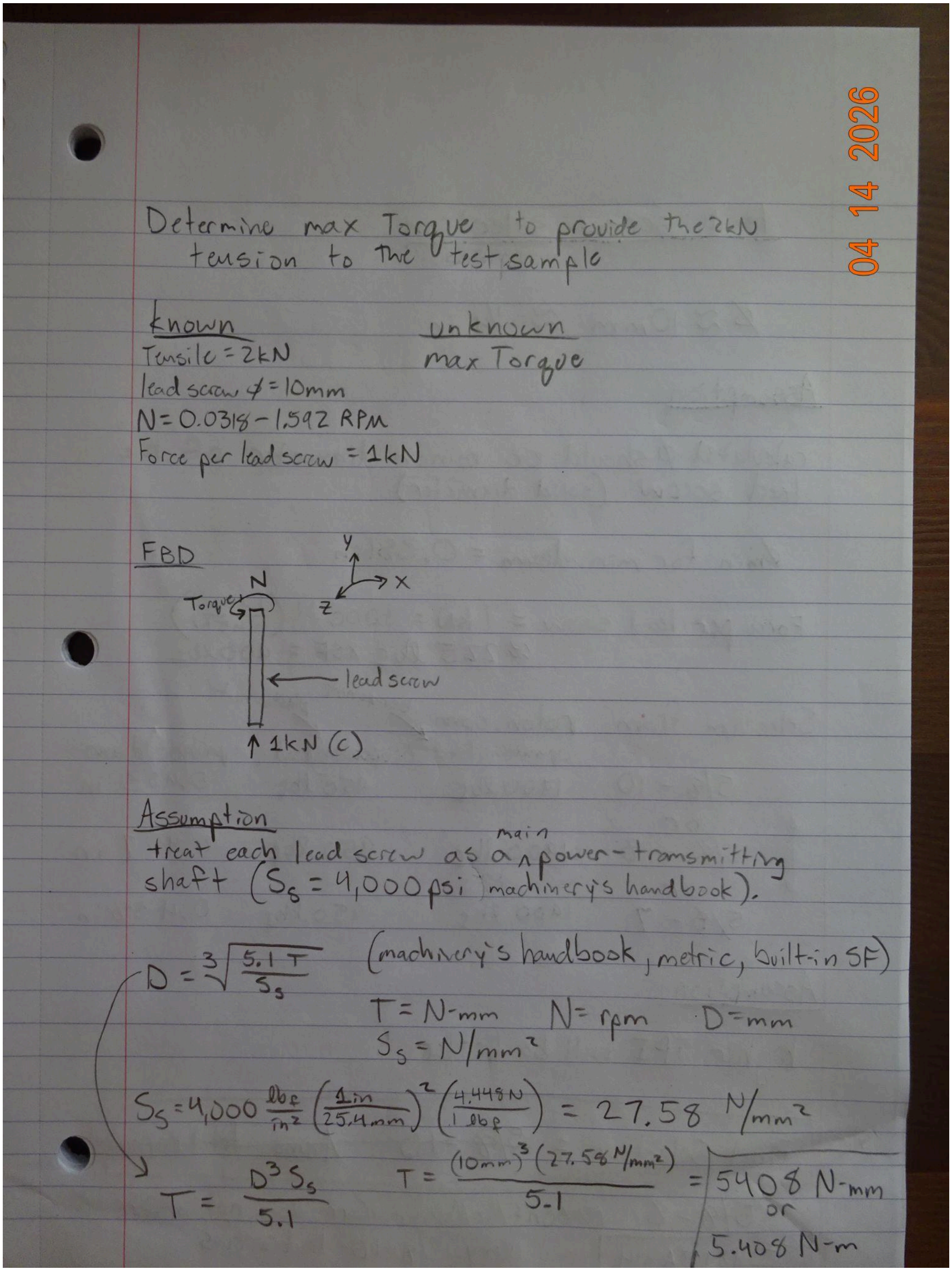
$$\omega_{low} = 1\frac{mm}{min}\left(\frac{1\,rev}{\pi(10\,mm)}\right) = 0.0318\ rpm$$

$$\omega_{high} = 50\frac{mm}{min}\left(\frac{1\,rev}{\pi(10\,mm)}\right) = 1.592\ rpm$$

$$\boxed{\omega_{range} = 0.0318 - 1.592\ \text{RPM}}$$

Determining Max Torque to Provide the 2kN Tension to the Test Sample

Next, the torque needed to be calculated. I followed the same process of writing the known and unknown variables and then drew my free body diagram. For this section I made the assumption that each lead screw could be modeled as a main power-transmitting shaft. The machinery's handbook provides equations with built in safety factors to calculate the torque, and so I converted the provided shear strength of 4,000 psi to 27.58 MPa. Again, it may be wise to return to this section and calculate the final torque into Imperial units, but I made my calculations in metric. The maximum torque was calculated to be 5408 N-mm or 5.408 N-m. The work can be found below.



04 14 2026

Determine max Torque to provide the 2kN tension to the test sample

known	unknown
Tensile = 2kN	max Torque
lead screw ϕ = 10mm	
N = 0.0318 – 1.592 RPM	
Force per lead screw = 1kN	

FBD

Assumption

treat each lead screw as a main power-transmitting shaft (S_s = 4,000 psi machinery's handbook).

$$D = \sqrt[3]{\frac{5.1\,T}{S_s}}$$ (machinery's handbook, metric, built-in SF)

T = N-mm N = rpm D = mm

$S_s = N/mm^2$

$$S_s = 4{,}000\,\frac{lbf}{in^2}\left(\frac{1\,in}{25.4\,mm}\right)^2\left(\frac{4.448\,N}{1\,lbf}\right) = 27.58\ N/mm^2$$

$$T = \frac{D^3 S_s}{5.1} \qquad T = \frac{(10\,mm)^3(27.58\,N/mm^2)}{5.1} = 5408\ N\text{-}mm$$

or

5.408 N-m

Lead Screw Selection

The last step was to select a lead screw. I used the previously given and calculated values to browse Roton's list of Acme lead screws. The main values I was looking at were the operating load for the bronze nut and minor diameter. I made two assumptions in this section. The first was that my calculated minimum diameter was better fit to be the minor diameter of the screw as it would essentially be the solid portion of the shaft. The second was that a higher TPI would be more beneficial for the final design as it would allow greater control of position/load. I came to the final selection of a 5/8-8 Acme lead screw. The reason I did not pick the highest TPI 5/8 screw was because McMaster-Carr did not offer any, and our team needed to utilize McMaster-Carr's SolidWorks downloads for our assembly. The part number for the selected two foot length of 5/8-8 Acme lead screw is 98935A914.

Select a Nut and Flange

Team member: Kenny Lee

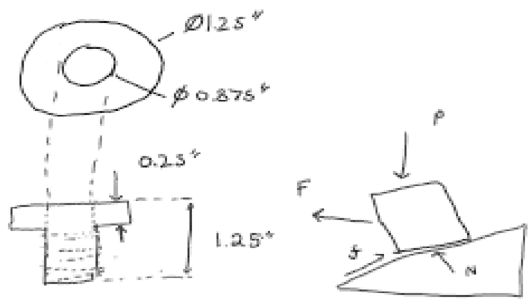
Beginning the calculation for the nut and flange, Luke and I looked over the lecture notes to see what we could use to help validate our selected nut and flange. With the lead screw diameter of 9.677mm, Luke selected the nut and flange 932 Acme Flange Nut. From the lectures slides we found the load and applied load formulas that could help us compare our given load of 2000N to the calculated load. From the websites linked below, we found the pitch dimension and coefficient of friction which helped us determine the calculated load which we then compared to our given load of 2000N. We also divided our given load by 2, getting us our new value of 1000N since there would be two separate flanges. After the math, our calculated load was 1162N which is over our given value but iis close enough to satisfy the safety factor and will slightly prevent shear.

Nut and Flange Stress Calculations

Starting off with the calculations, I wrote down all of the knowns from previous calculations and from the websites that we used and then the unknowns which were the load, applied load, and the circumference of the pitch as we needed it later to help find the applied load. After, I symbolically solved for the applied load with what we were given and the work and answer are attached below for further visualization of the calculations.

Knowns: $T = 5408\ N \cdot mm$ or $5.408\ N \cdot m$ $\quad P_d = 14.288mm$ $\quad \mu = 0.16$

Unknowns: $F = ?,\ P = ?,\ C = ?$



Symbolically:

$F = P(\mu C + L)/(C - \mu L)$ $\quad \mu = \text{Given}$

$C = D \cdot \pi$

$$\frac{F \cdot (C - \mu L)}{(\mu C + L)} = P \qquad F = \frac{2 \cdot T}{P_d}$$

Numerically:

$$F = \frac{2 \cdot 5408\ N \cdot mm}{14.288mm} = 757\ N$$

$$C = 14.288\ mm \cdot \pi = 44.88\ mm$$

$$\mu = 0.16$$

$$P = \frac{F \cdot (C - \mu L)}{(\mu C + L) SF} = \frac{757N\ (44.88mm - 0.16 \cdot 0.125mm)}{((0.16 \cdot 44.88mm) + 0.125mm)4}$$

$$\boxed{P = 1162\ N} > 1000\ N$$

∴ Our calculated P will suffice because the safety factor allows our P to be close to our given Force of 1000N

The webpage MachiningDoctor.com was used for finding some of the thread dimensions of the 5/8-8 size.
The webpage EngineeringToolbox.com was used for finding the static coefficient of friction between the bronze nut and the steel lead screw.

Design the Crosshead

Team member: Luke DeVries

To begin the crosshead design, I looked through some open-source tensile testers to get an idea of how long the crosshead should be for the scale of our design. I used the provided example (Figure 1 here and Figure 14 from this pdf) designed by Stella Demmel, and found that the crosshead length should be around half of the total height of the lead screws. Since our lead screws are 24 inches long, the crosshead length would need to be around 12 inches. I also chose to determine the width of the crosshead by taking the flange diameter of 1.5 inches and adding 1 inch to each side. This gives a crosshead width of 3.5 inches. With this information I was able to start the two analyses that would drive the design of the height and material selection.

Design by Simple Beam Stress/Strength Analysis

To begin this analysis, I wrote down all the known and unknown variables. After this I quickly sketched a free body diagram depicting the general shape of the crosshead along with the known forces acting on it. Below is an image of the first page of setup work for the stress/strength analysis.

Design the Crosshead

Research appropriate length of crosshead

crosshead length ~ $\frac{1}{2}$ lead screw length → 12"

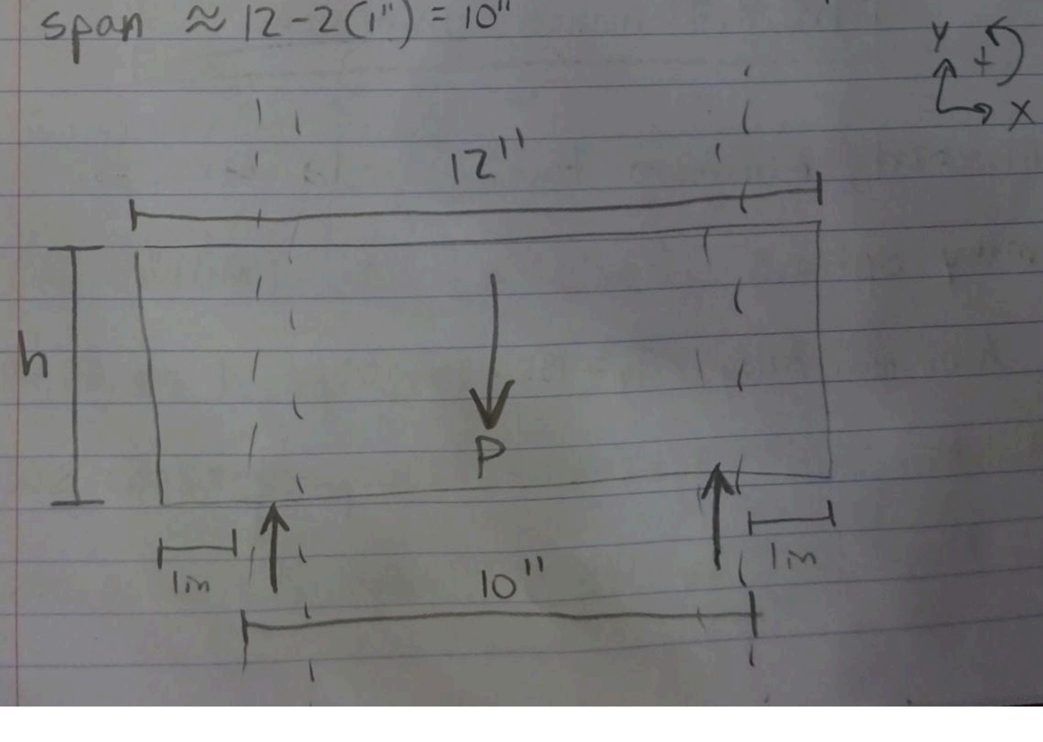
"Design, Manufacturing, and Testing of an Open-Source Universal Testing Machine for Hands-On Learning"
by Stell Demmel

Flange Diameter = 1½"

leave ~1 in of material around the flange cutout

Simple Beam Stress/Strength Analysis

Known	unknown
maxload 2kN	Crossbeam height
SF = 4	σ_{axial}
crosshead length = 12"	$\sigma_{bending}$
crosshead width = 1.5" + 2(1") = 3.5"	E (material properties)
span ≈ 12 − 2(1") = 10"	



For the calculations, I started by doing a simple axial stress calculation utilizing the max load and the cross sectional area using the length and width I set at the start. I then made an assumption that the crosshead could be modeled as a simply supported beam with a load in the center for the beam bending stress analysis. I also made the assumption that in this model the axial stress would be equal to the bending stress in which I could use the calculated axial stress as the bending stress for the second equation. Doing this substitution already accounts for our safety factor of 4, so I made the bending stress calculation without the use of the safety factor. This analysis resulted in a maximum stress of 42.8 psi, a minimum height of 4.48 inches, and a fair amount of materials available for selection. The few materials I listed in the image below were found using the search feature of Matweb.com where I found information for aluminum alloys, medium alloy steel, and low alloy steel, all of which were suitable for the calculated stress. Below is an image of the work for the calculations.

04 15 2026

$$\frac{\sigma_{axial}}{SF} = \frac{F}{A} = \frac{(4)2{,}000N\left(\frac{1\,lb}{4.45N}\right)}{(12")(3.5")} = 42.8\ psi$$

Assumption

Simply Supported, Load at center

$$\frac{\sigma_{bending}}{SF} = -\frac{Wl}{4Z} = -\frac{3Wl(SF)}{bd^3}$$

$$Z = \frac{bd^3}{12} \qquad d = \sqrt[3]{\frac{3Wl(SF)}{b\,\sigma_{bending}}}$$

Assumption

$$\sigma_{axial} = \sigma_{bending} \quad \text{(SF already accounted for)}$$

$$h = d = \sqrt[3]{\frac{3(2{,}000N)\left(\frac{1\,lb}{4.45N}\right)(10")\cancel{(4)}}{(3.5")(42.8\,psi)}}$$

$$\boxed{h = 4.48\ \text{inches minimum}}$$

material, minimum tensile yield 42.8psi

many options here's a few: (MatWeb.com)

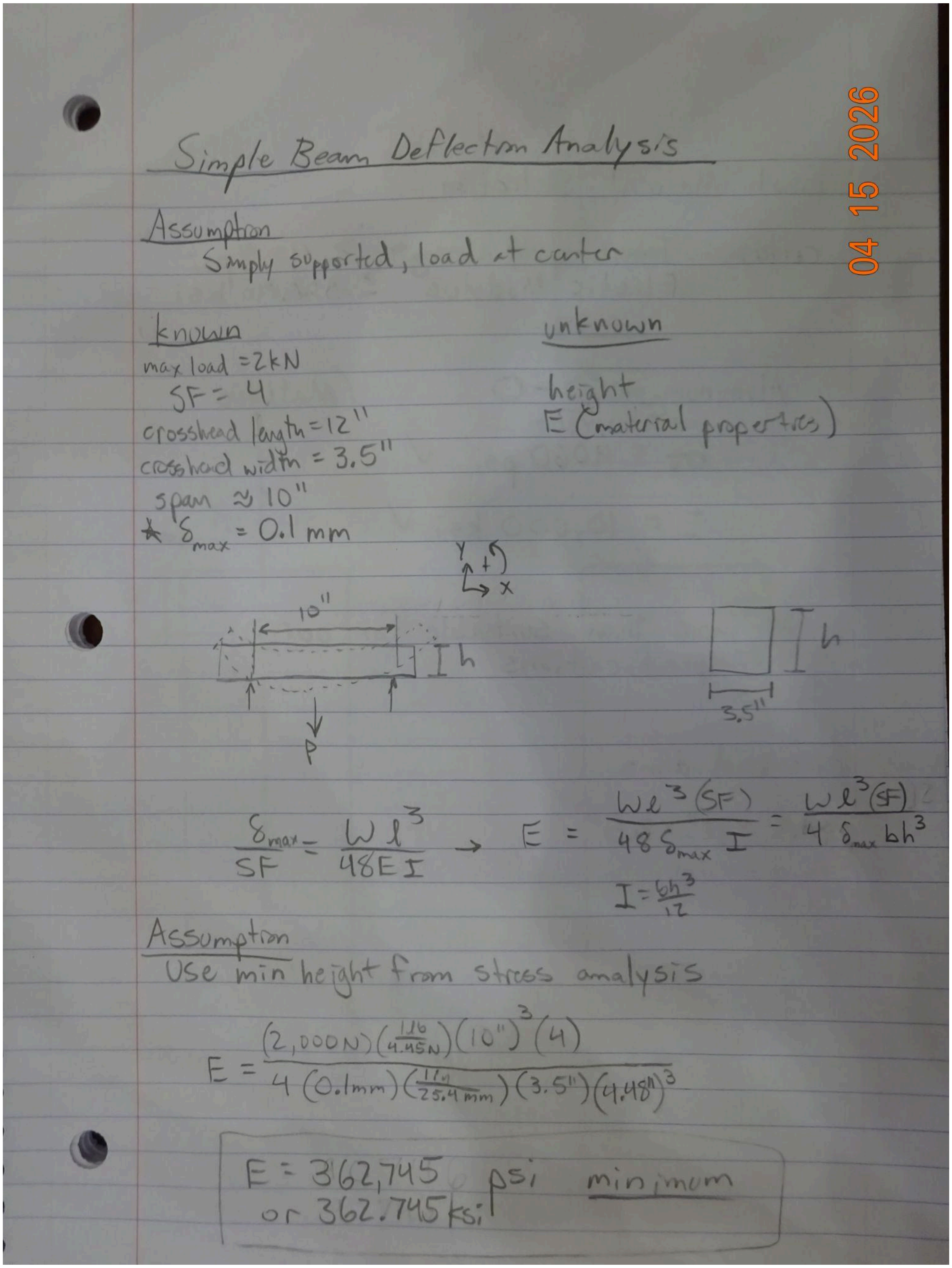
Aluminum Alloy $T_y = 180 - 106000\,psi$ $E = 6.96 - 49600\,ksi$

Medium Alloy Steel $T_y = 24500 - 290000\,psi$ $E = 2900 - 30000\,ksi$

Low Alloy Steel $T_y = 16000 - 347000\,psi$ $E = 26500 - 30400\,ksi$

Design by Simple Beam Deflection Analysis

Next, I moved on to the deflection analysis of the crosshead. We were given a maximum deflection of 0.1 mm, so I included this in my list of known values. My unknown values included the height, but to calculate the required modulus of elasticity, this value needed to be known. To do this, I made an assumption to utilize the height calculated in the stress/strength analysis. After copying over the information, I created a free body diagram and depicted the exaggerated deflection of the crosshead. With this visual guide, I started modeling the deflection equation again for a simply supported beam with a central load. This analysis resulted in a minimum modulus of elasticity equal to 362.745 ksi. The calculation can be seen in the image below.



Simple Beam Deflection Analysis

04 15 2026

Assumption
Simply supported, load at center

known
max load = 2kN
SF = 4
crosshead length = 12"
crosshead width = 3.5"
span ≈ 10"
★ δ_{max} = 0.1 mm

unknown
height
E (material properties)

$$\frac{\delta_{max}}{SF} = \frac{Wl^3}{48EI} \rightarrow E = \frac{Wl^3(SF)}{48\delta_{max} I} = \frac{Wl^3(SF)}{4\delta_{max} bh^3}$$

$$I = \frac{bh^3}{12}$$

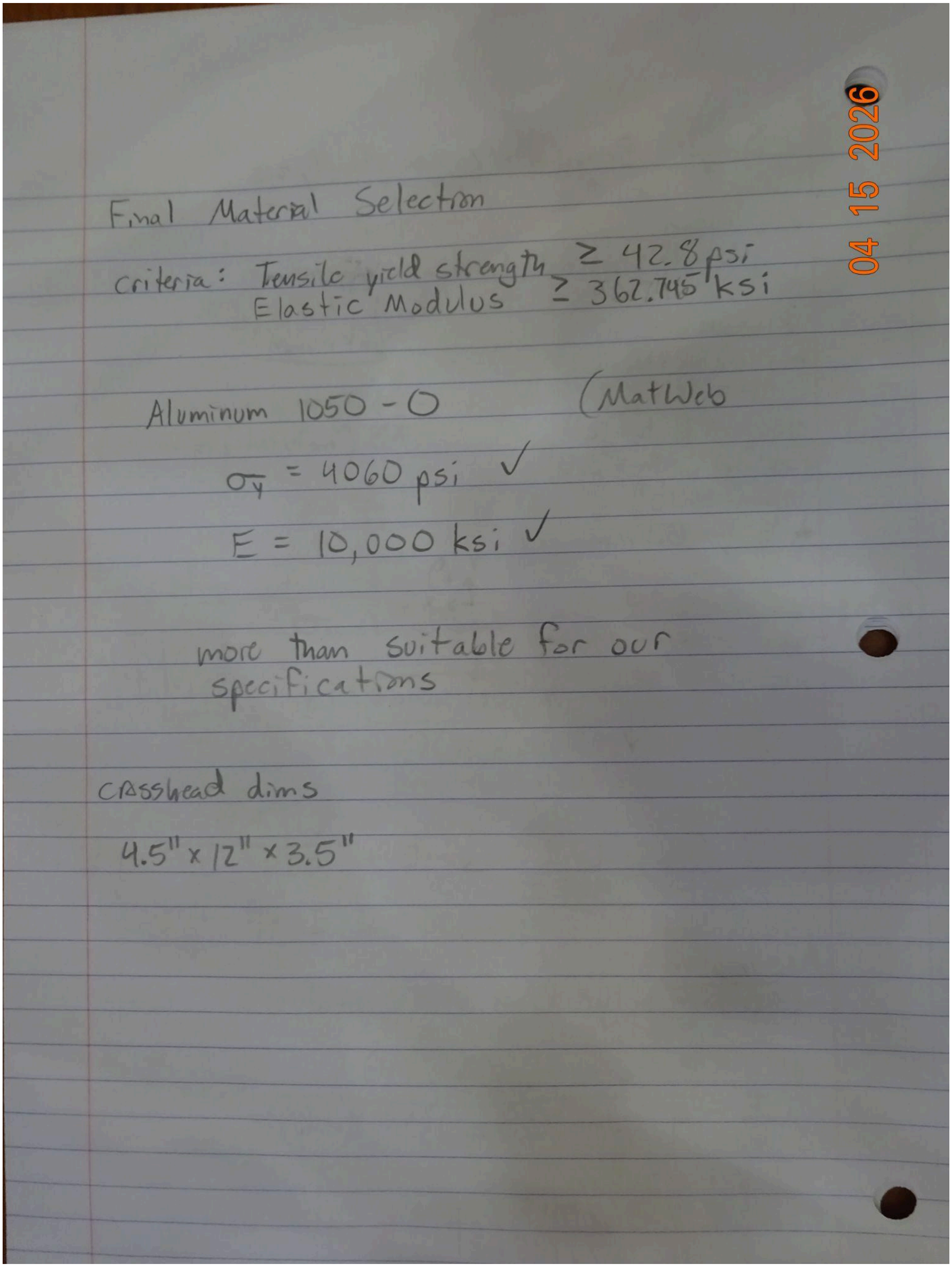
Assumption
Use min height from stress analysis

$$E = \frac{(2{,}000N)\left(\frac{1lb}{4.45N}\right)(10")^3(4)}{4(0.1mm)\left(\frac{1in}{25.4mm}\right)(3.5")(4.48")^3}$$

$$\boxed{E = 362{,}745 \text{ psi minimum or } 362.745 \text{ ksi}}$$

Final Material Selection

With all the dimensional and material properties calculated, I was able to create a list of requirements for the final material selection. I again utilized Matweb.com and chose aluminum 1050-O as its material properties were more than enough to satisfy our design requirements. I also copied down all of the dimensional measurements for the crosshead so that all the information was available on a single page which can be seen below.



04 15 2026

Final Material Selection

criteria: Tensile yield strength ≥ 42.8 psi
Elastic Modulus ≥ 362.745 ksi

Aluminum 1050-O (MatWeb

$\sigma_y = 4060$ psi ✓

$E = 10{,}000$ ksi ✓

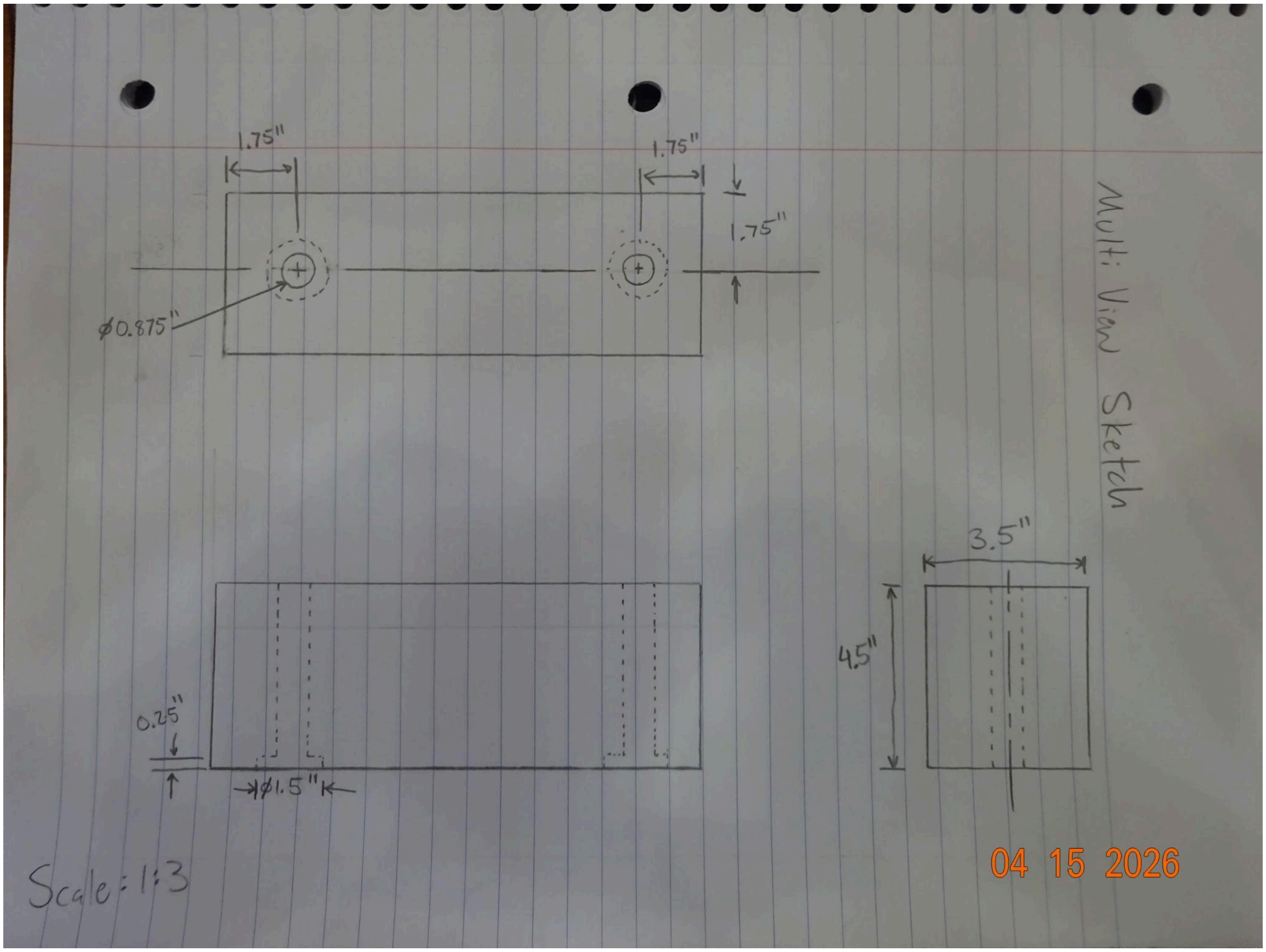
more than suitable for our specifications

crosshead dims

4.5" x 12" x 3.5"

Multi-View Sketch of the Crosshead

The final step before moving into the CAD software was to draw a sketch of the crosshead design. I utilized a ruler to make the drawing a roughly 1:3 scale and included the necessary dimensions for placing the holes for the lead screws and flange nuts. I utilized hidden lines to show the holes and the small counterbore where the flange of the nut will support the crosshead. The sketch can be seen below and is the foundation for the CAD model although new design iterations were also included as the model's design intent progressed.



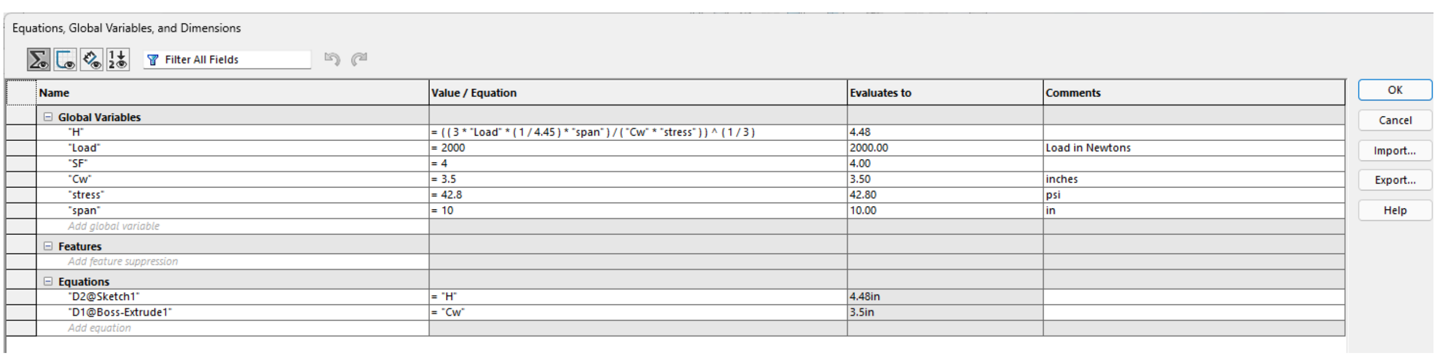
Tabulating Costs

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 Right Hand, 5/8"-8 Thread Size	$138.20	$276.40	https://www.mcmaster.com/95120A119/
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
			Total Cost:	$ 566.07	

Parametric Modeling

Crosshead Model

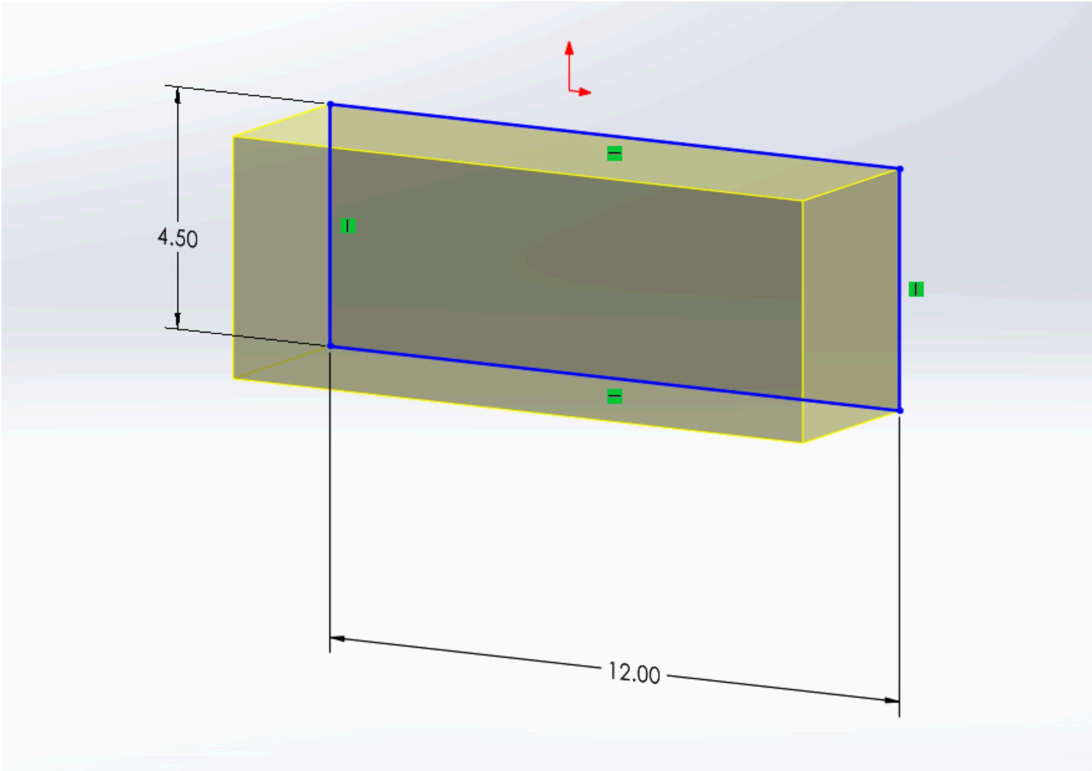
Team member: Kenny Lee



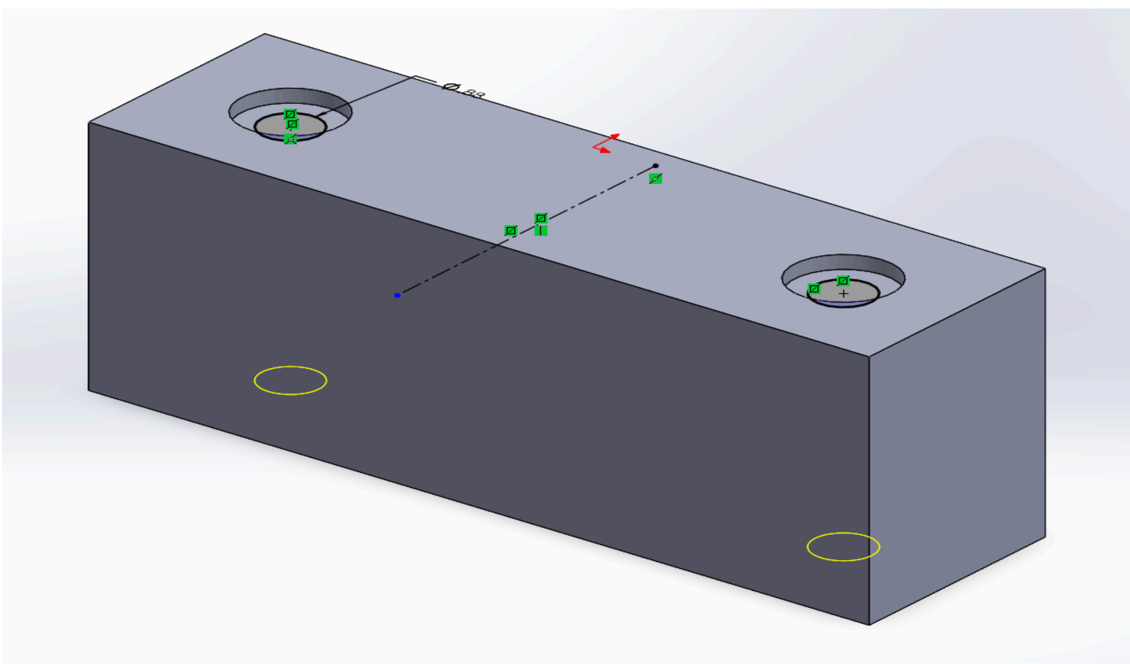
Equations, Global Variables, and Dimensions

Name	Value / Equation	Evaluates to	Comments
Global Variables			
"H"	= ((3 * "Load" * (1 / 4.45) * "span") / ("Cw" * "stress")) ^ (1 / 3)	4.48	
"Load"	= 2000	2000.00	Load in Newtons
"SF"	= 4	4.00	
"Cw"	= 3.5	3.50	inches
"stress"	= 42.8	42.80	psi
"span"	= 10	10.00	in
Add global variable			
Features			
Add feature suppression			
Equations			
"D2@Sketch1"	= "H"	4.48in	
"D1@Boss-Extrude1"	= "Cw"	3.5in	
Add equation			

Taking the dimensions calculated by Luke, I was able to design the cross head with ease. There were no issues with the design, just many extrusions. As shown below, the crosshead used the height of 4.50 in, span of 12 in, and it doesn't show it in the design but I used a width of 3.5 in.



After gathering all the information I needed to design the cross head, I started out with a sketch of the measurements I included and then extruded it by 3.5 in width. After that I had a nice cross head I could extrude holes in for the flanges.

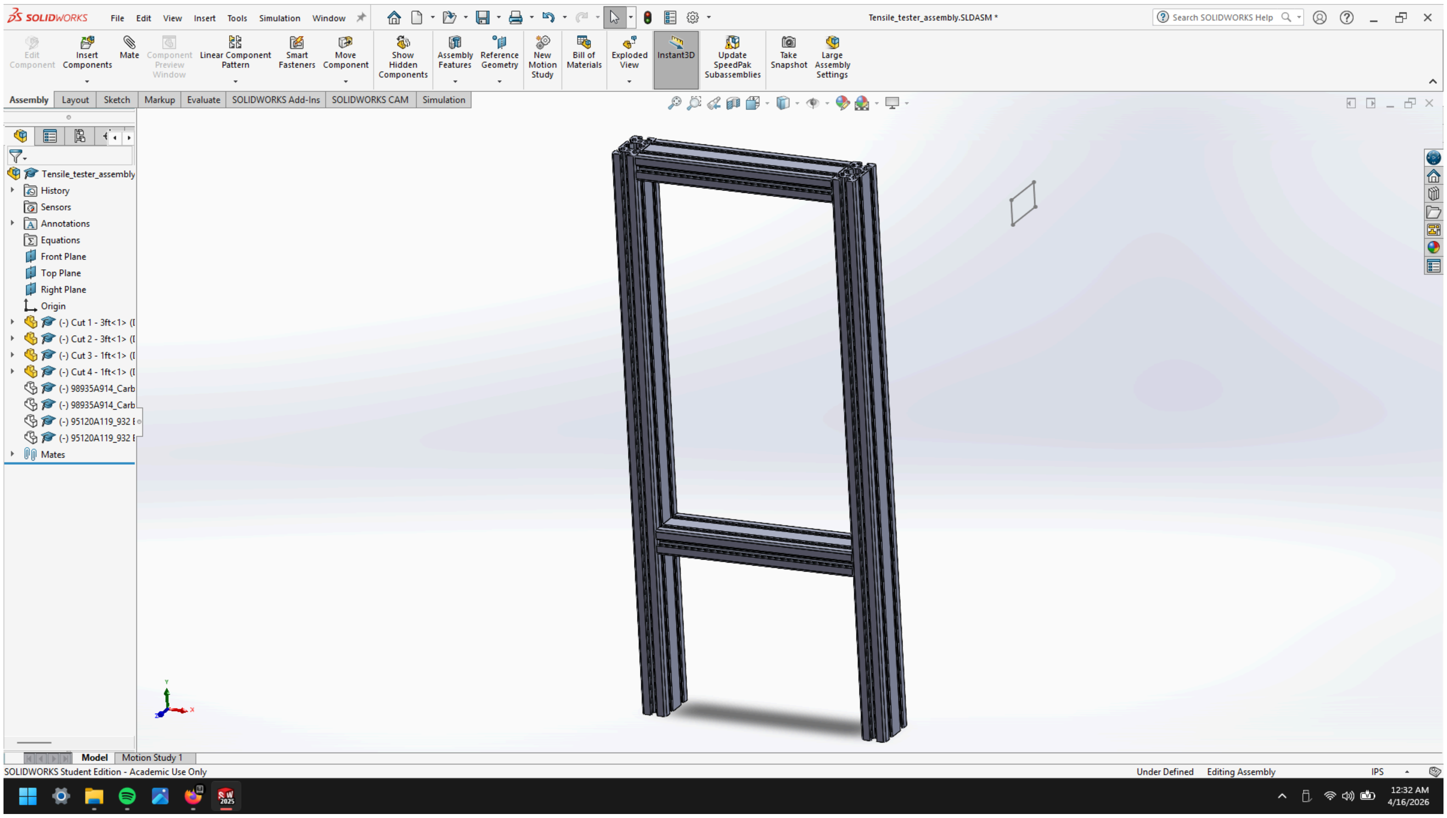


Finally, I extruded my last holes up and measured everything to be the same diameters of the flanges and that concludes the cross head design of our project.

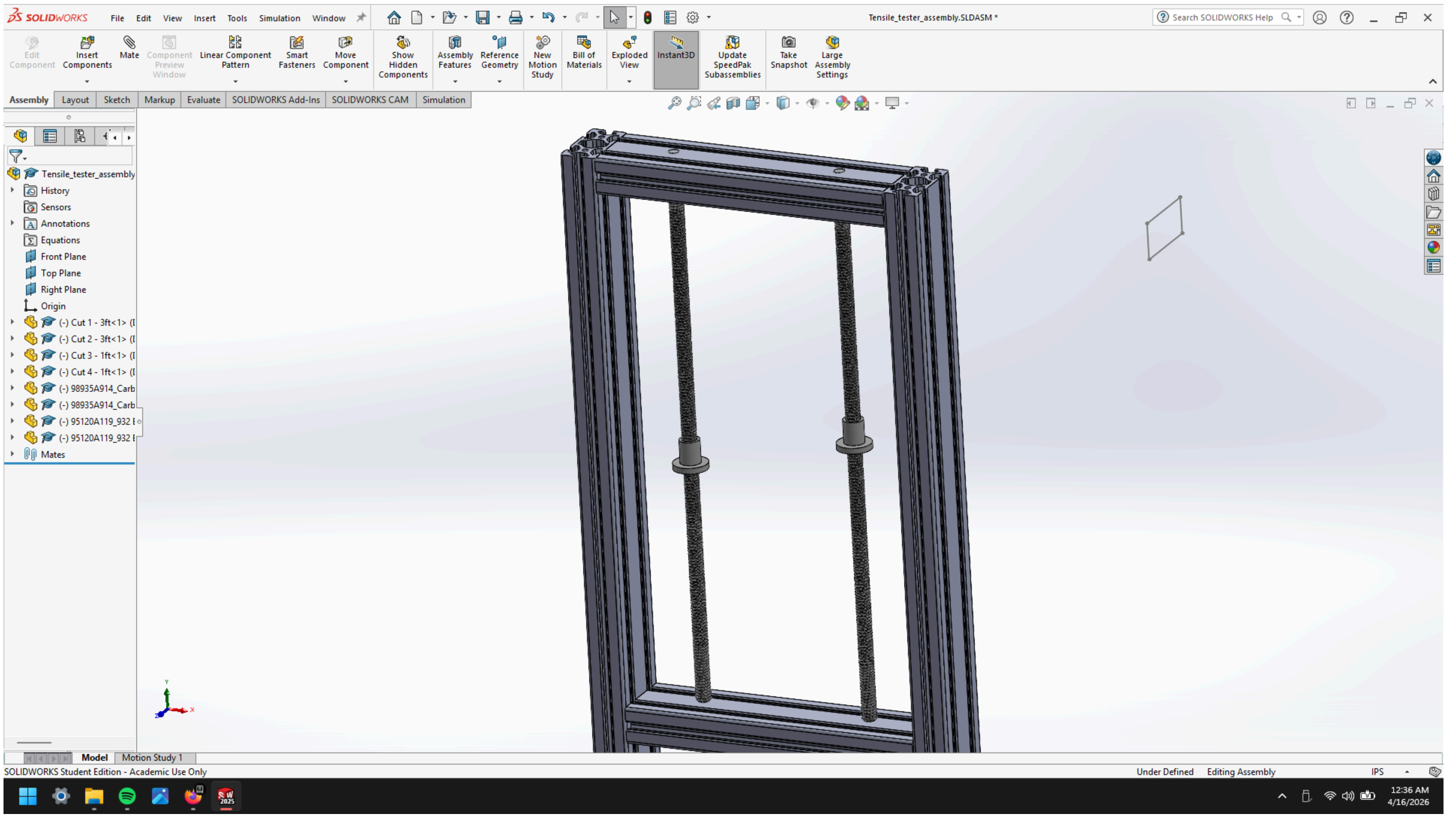
Assembly

Team member: Luke DeVries

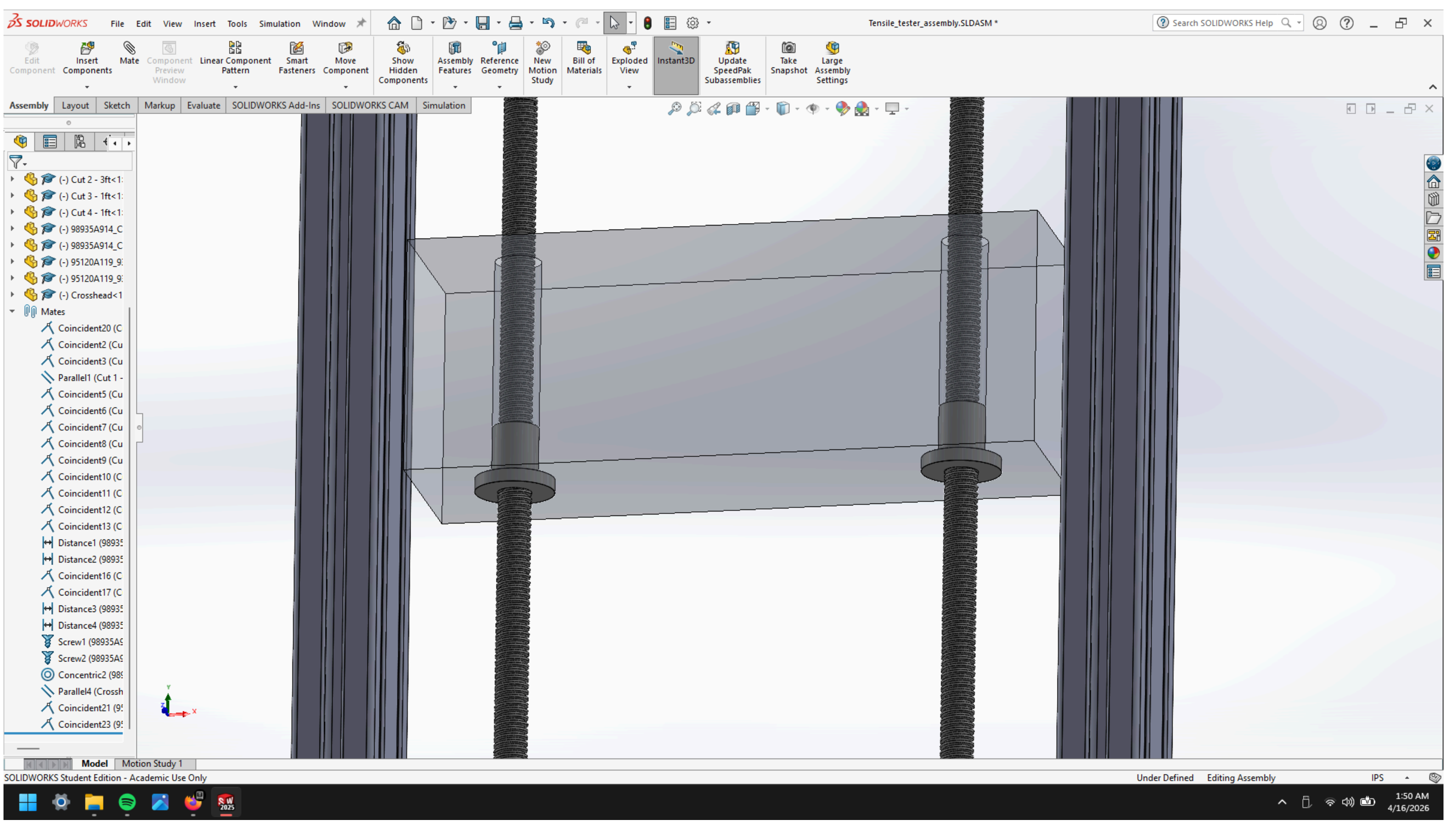
I was not able to find any CAD file downloads for the 1 meter or 2 meter options from Grainger.com, so instead I took a single 3 meter length of frame and made cuts for the necessary lengths of frame. Fortunately, 3 meters of frame is just enough for the two 3 foot vertical frame pieces along with two crossbeams of 1 foot and 1 inch. After making parts for the cut frame pieces, the assembly was started with just a simple shape to house the lead screws and crosshead.



Next, the lead screws and flange nuts were added to the frame. The lead screw and nuts have a screw type mate, so they can be rotated realistically.



Finally, the crosshead was added to the assembly. The .zip file with all included parts can be found at the bottom of this document.



Below is a final render of the assembly.

Lessons Learned

Kenny Lee

This assignment severely strengthened my ability to research, plan, think, and work with team members. Being our first group based project, I underestimated the time it took to work and devise a plan on how to balance out the workload for my team member and I. Not doing so led to the imbalance of workload for my partner and I, but this was not the sole lesson learned here. I also learned the importance of how significant outside factors would help in my calculations. On top of lecture, extra resources and websites were needed to calculate many different problems. Over all, this assignment took 4 hours to complete not including my partners additional hours.

Luke DeVries

In total, I would say that this assignment took around 6.5 hours to complete including all calculations, research, CAD, and documentation. This assignment deepened my understanding of assemblies, design intent, teamwork, and utilizing research to support my understanding of existing designs. It also strengthened my decision making skills when selecting real world parts to fit designed/calculated requirements. I now have a much better understanding of how tensile testers are designed as well as some of the many calculations that are necessary when designing parts that experience large forces and stresses.

Download links:

- Here is the complete hand written work by Luke: [pdf download]
- Here is the link to the CAD Assembly: [zip download]