5.1 Baseline Design

5.1.1 Results

1. Script:

```
l = 16;
h = 0.75;
b = 0.5;
c = 1;
E = 32*10^6;
v = 0.29;
G = E/(2*(1+v));
sigmaUlt = 370*10^3;
I = 1/12*b*h^3;
%%crack propagation
Kic = 20*10^3;
safetyCrack = 2;
a = 0.02;
Tallow = Kic/safetyCrack*b*h^2/6/(1.12*sqrt(pi*a));
disp('Torque Allowable against Crack Propagation:');
disp(Tallow);
%%bending
safetyYield = 4;
Torque = sigmaUlt*I/(h/2)/safetyYield;
P = Torque/l;
F = 600/l;
deflection = F*l^3/(3*E*I);
disp('Torque Allowable against Yield:');
disp(Torque);
disp('Deflection:');
disp(deflection);
disp('Force Allowable against Yield:');
disp(P);
%%fatigue
fatigueStr = 115*10^3;
safetyFatigue = 1.5;
sigAllowF = fatigueStr/safetyFatigue;
FatigueTallow = b*h^2/6*sigAllowF;
disp('Torque Allowable against Fatigue:');
disp(FatigueTallow);
%%strain;
Force = 600/l;
Moment = Force*(l-c);
strain = Moment*h/(2*I)/E;
disp('Strain:');
disp(strain);
%%max normal stress
%%bending
maxBendingStress = Force*l*h/2/I;
disp('Max Bending Stress:');
```

disp(maxBendingStress);

2. Hand Calculation Deflection, Strain, and Maximum Normal Stress

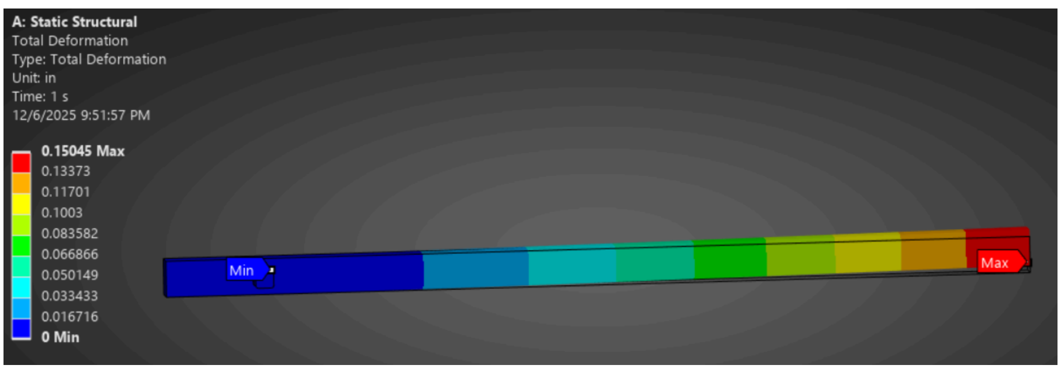
   Deflection:
   0.0910 in

   Strain:
   3.7500e-04

   Max Bending Stress:
   12800 psi

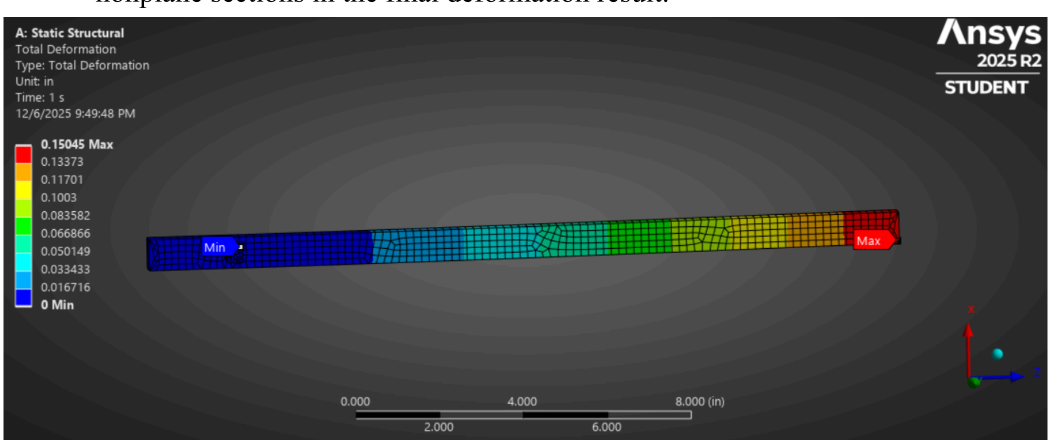
3. FEM Deflection, Strain, and Maximum Normal Stress

   Deflection: 0.15045 in
   Strain: ~-0.00029938
   Maximum Normal Stress: 57702 psi (concentration at drive connection)

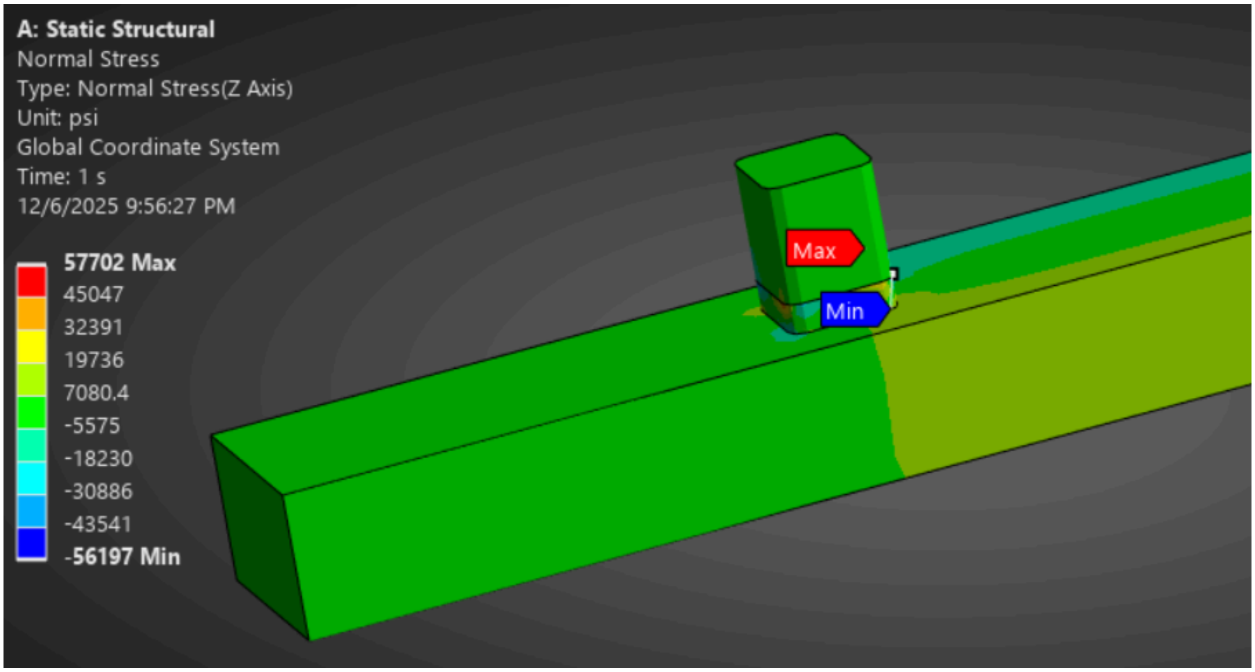


5.1.2 Reflections

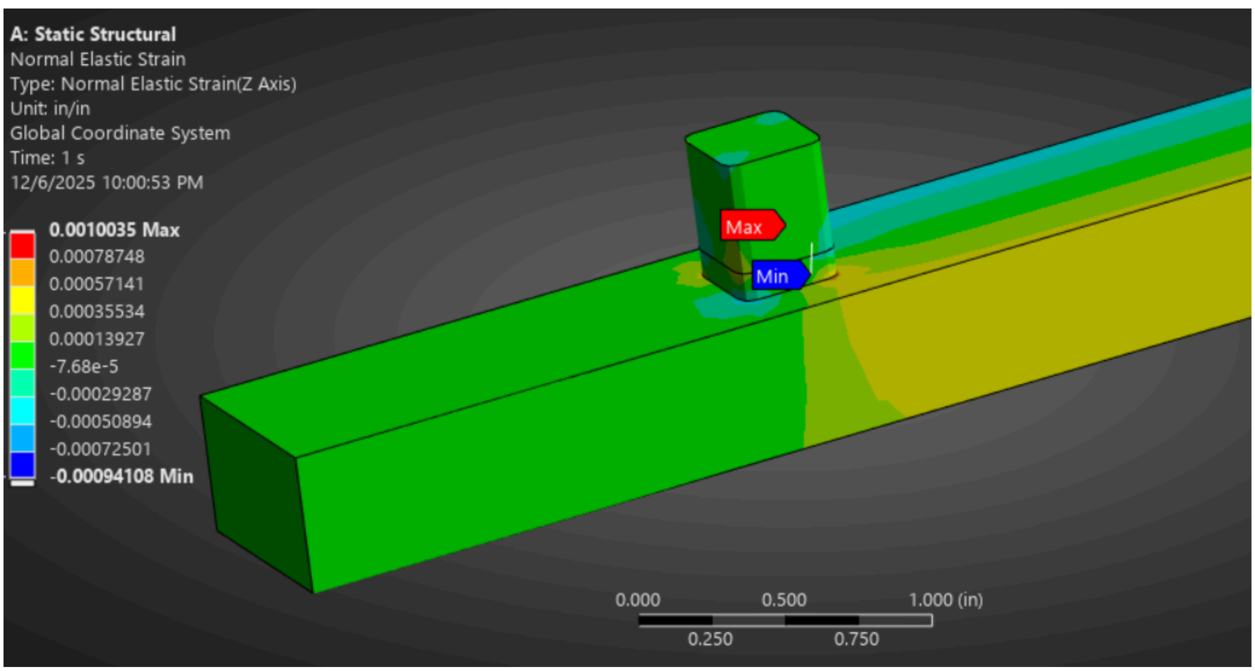
1. In some parts more than others, plane sections remain plane. At the boundaries of where the stress gets incrementally larger along the lever, plane sections are no longer plane. But, overall, I think it's reasonable to still use beam theory considering the visual percentage of plane sections to nonplane sections in the final deformation result.



2. The hand calculated results don't take into account stress concentrations at the drive so it only considers the very end of the cantilevered beam for the highest normal stress location. Thus, the Ansys model predicts a value that's more than four times larger than the hand calculations.



3. The strain at the gauge was predicted to be about 375 microstrain. The Ansys model predicts somewhere around 355 microstrain. This is pretty close. It might be that the strain prediction is lower in the Ansys model because plane sections in that region may not have remained plane.



MODIFIED DESIGN

Summary of our design process:

- Looking at our results for the baseline design (L = 16 in, h = 0.75 in, b = 0.5 in), the only requirement that it fails to meet is the strain sensitivity requirement; the baseline design reads 375 microstrain at the gauge while we are aiming for 1000 microstrain. Additionally, Ansys showed stress concentrations where the drive is attached to the handle. Thus, our two main design goals are **1) increase the strain sensitivity at the strain gauge while not compromising the other properties (yield, crack growth, fatigue)** and **2) reduce stress concentrations.**
- In order to get a strain value of at least 1 mV/V, we need to adjust the stress at the strain gauge since strain is proportional to stress ($\sigma = E\varepsilon$)
    - We did not want to change the E value, since we did not want to change the material and affect the other values; instead, we wanted to change the I value by changing the geometry.
    - By playing around with the geometric parameters in our MATLAB code, we found that b=0.45 would yield a strain of 0.001.

1. **Image(s) of CAD model. Must show all key dimensions.**

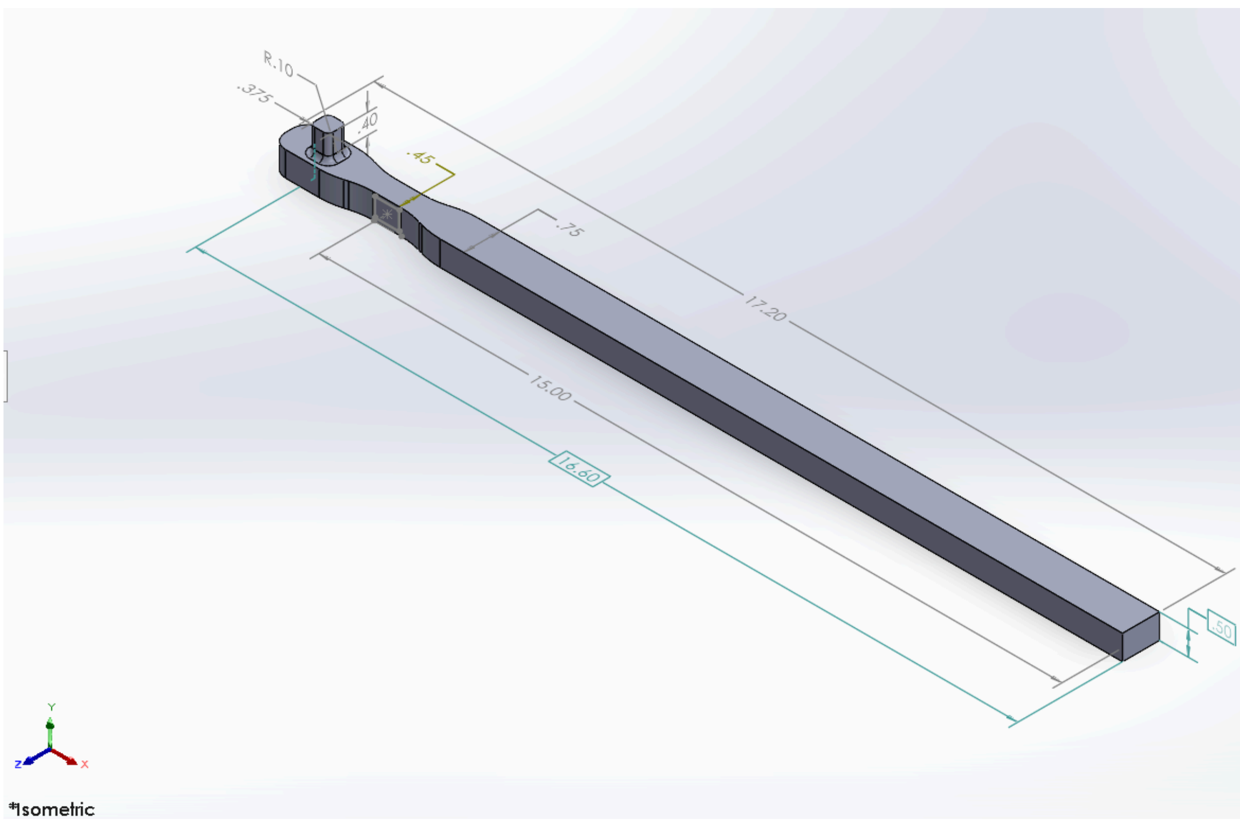


Figure 1: CAD model of the modified torque wrench design

Relevant dimensions:

Drive

- Total height = 0.50 in.
- Height of clamped region = 0.40 in.

- Cross-section = ⅜ in. x ⅜ in.
- Fillet radius (at connection to handle and at 4 edges) = 0.10 in.

Handle

- Total length = 17.20 in.
- Distance between (center of) strain gauge and free end = 15.00 in.
- Distance between (right end of) drive and free end (L) = 16.60 in.
- Distance between drive and strain gauge (c) = 16.60 - 15.00 = 1.6 in.
- Height (b) = 0.50 in.
- Width at widest point (h_2) = 0.75 in.
- Width at strain gauge, i.e. width at narrowest point (h_1) = 0.45 in

2. **Describe material used and its relevant mechanical properties.**
   Material: Tool steel, chromium alloy, AISI H19
   Description:
   Relevant material properties (using upper end values):

   $E = 32 * 10^6$ psi
   $\nu = 0.29$
   $\sigma_o = 260 * 10^3$ psi
   $K_{IC} = 30.4 * 10^3$ psi-in$^{1/2}$
   $\sigma_{fatigue} = 78.3 * 10^3$ psi (for 10^6 cycles)

3. **Diagram communicating how loads and boundary conditions were applied to your FEM model.**

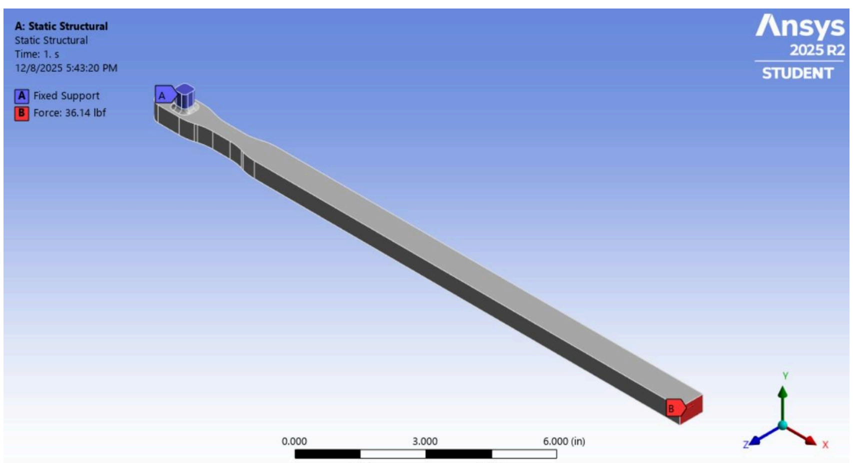


Figure 2: Application of loads and boundary conditions in ANSYS

As shown in the diagram, the body corresponding to the top 0.4” of the drive is fixed. The force applied to the end is **F** = -36.14 lbf z. Since the torque applied at the drive must be 600 lbf-in, and the distance from the end to the drive shaft is 16.6 in, this yields a force with magnitude of 600/16.6 = 36.14.

4. **Normal strain contours (in the strain gauge direction) from FEM**

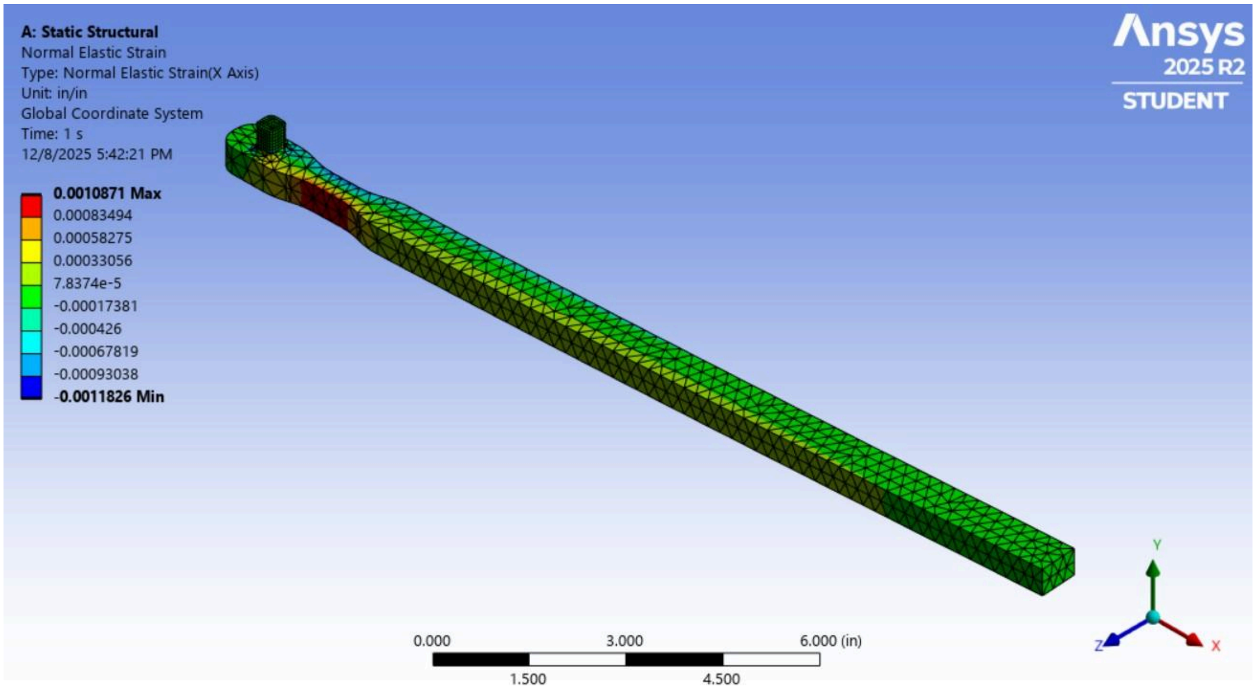


Figure 3a: Normal elastic strain contour, in the x-direction

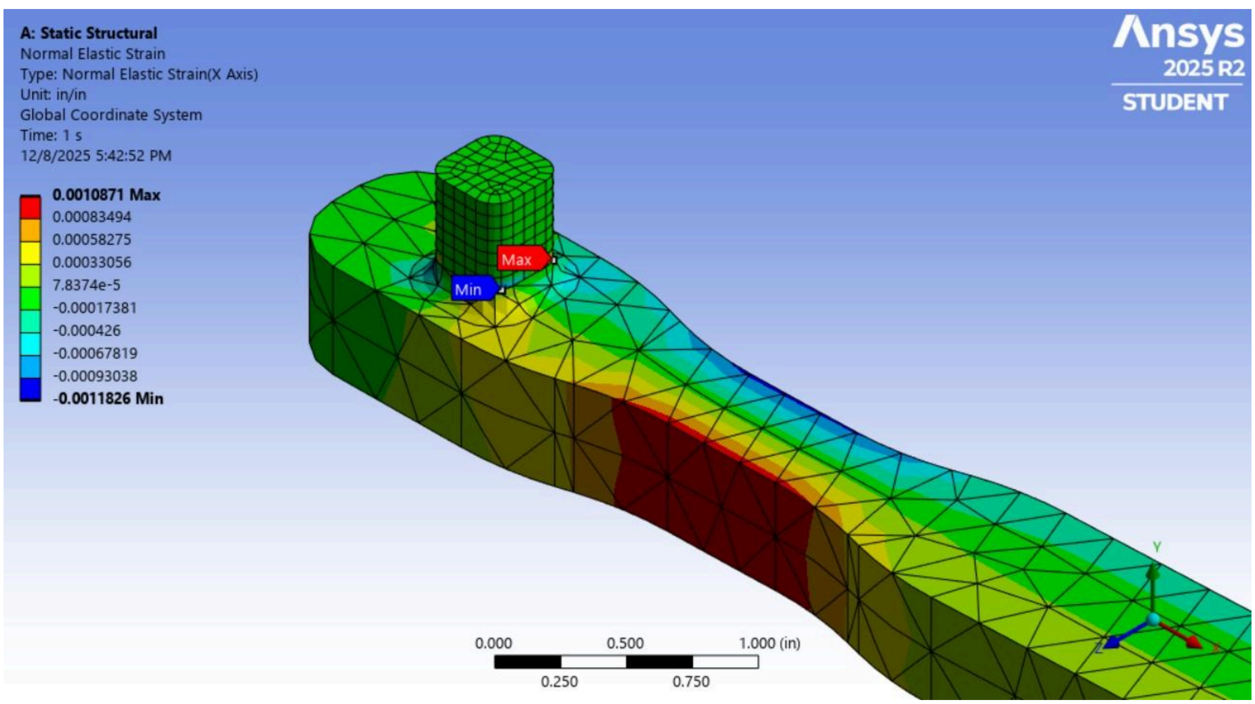


Figure 3b: Zoomed in on locations of maximum and minimum ε_{xx} (0.00109 and -0.00118, respectively)

5. Contour plot of maximum principal stress from FEM

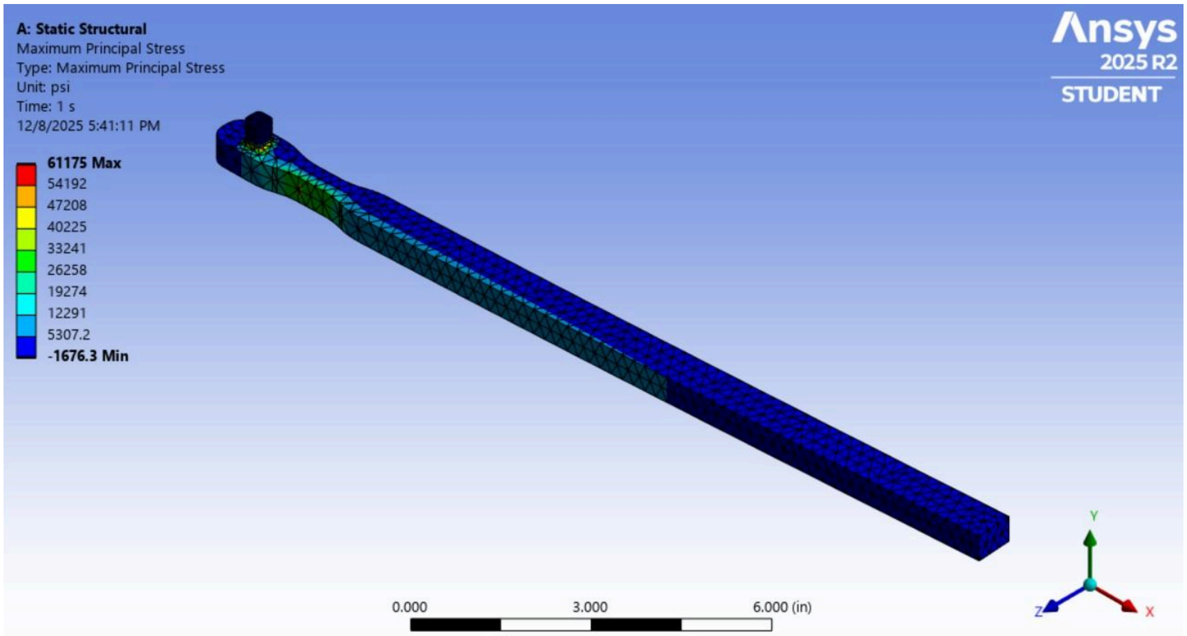


Figure 4a: Maximum principal stress contour

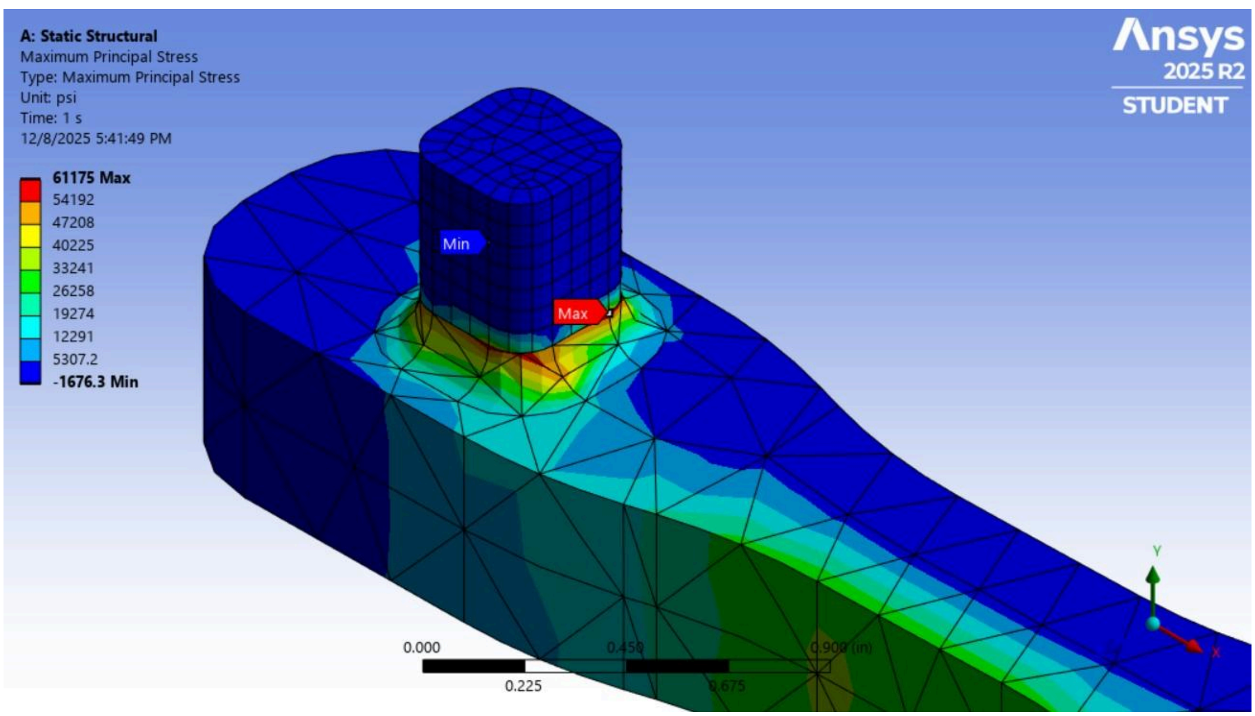


Figure 4b: Zoomed in on locations of maximum and minimum principle stresses (61176 psi and -1676.3 psi, respectively)

6. **Summarize results from FEM calculation showing maximum normal stress (anywhere), load point deflection, strains at the strain gauge locations**

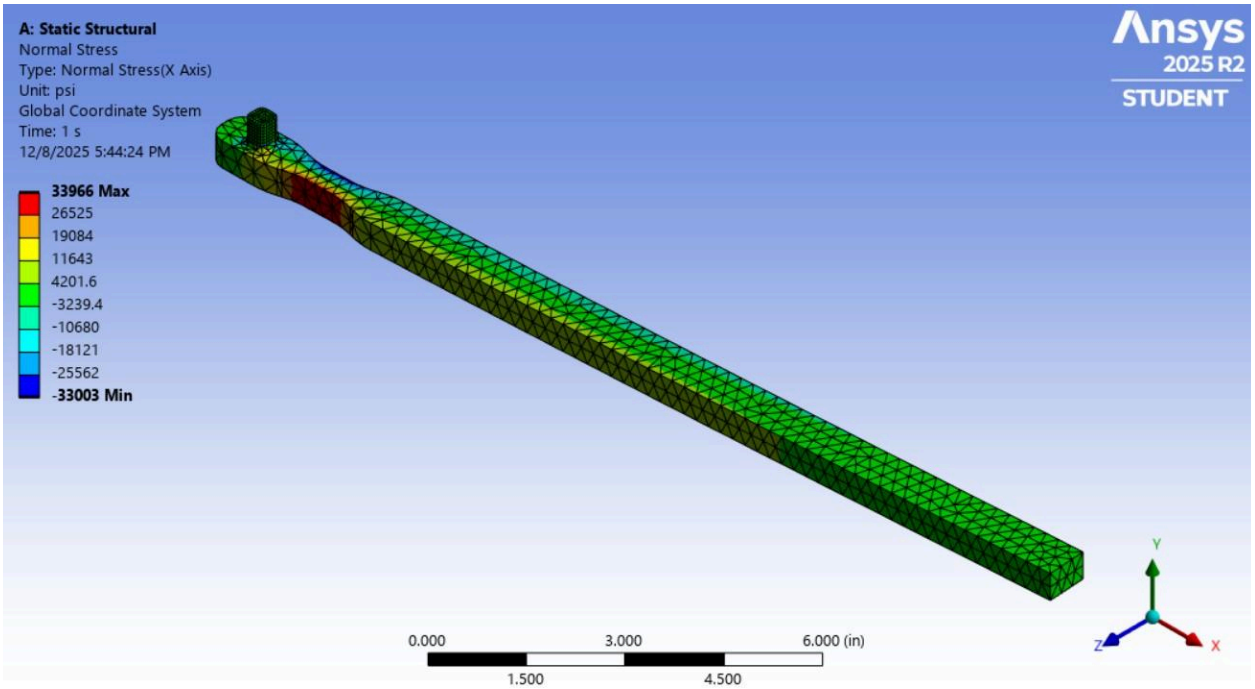


Figure 5a: Normal stress contour, in x-direction

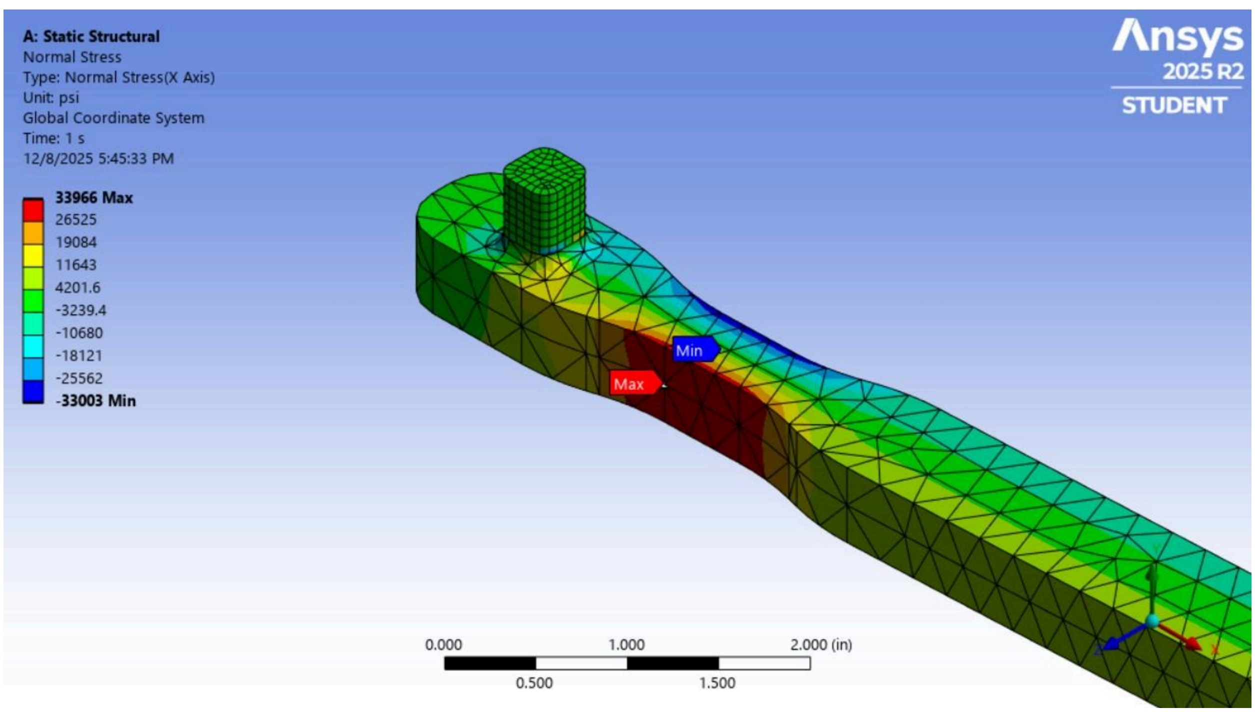


Figure 5b: Zoomed in on locations of maximum and minimum σ_x (33966 psi and -33003 psi, respectively)

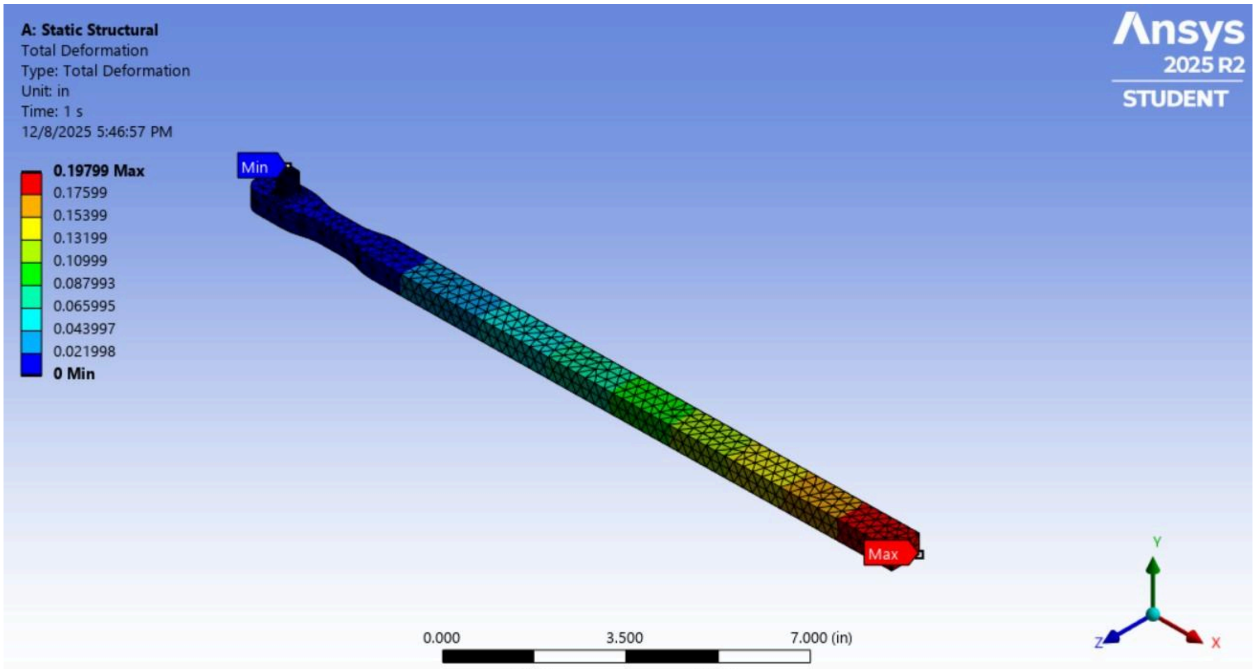


Figure 6: Total deformation contour

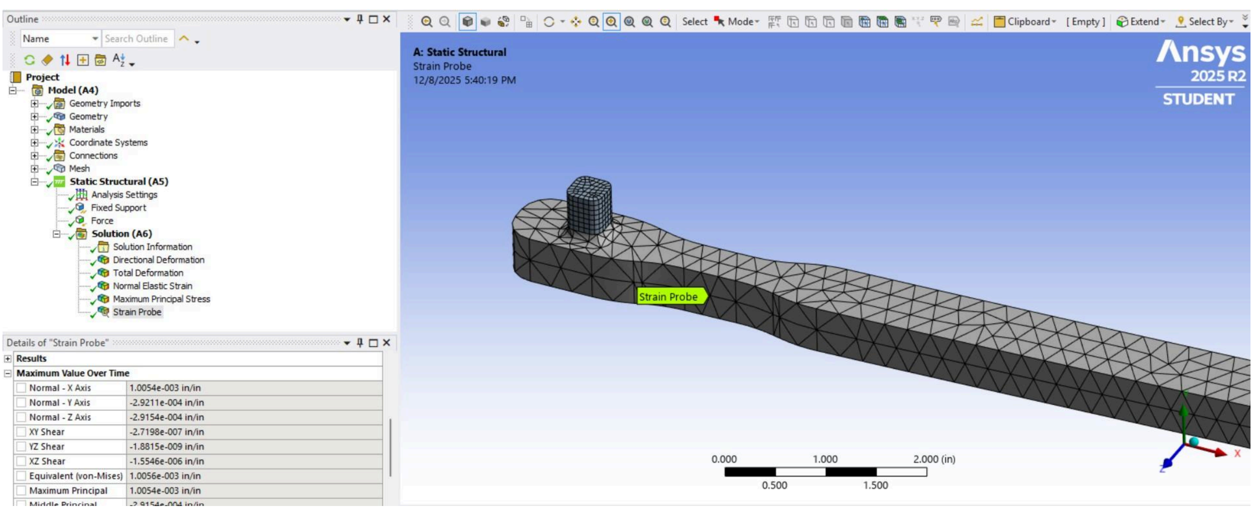


Figure 7: Strains measured at the strain gauge

Summary of Key Results:

- Max normal stress σ_x = 33966 psi, located at or near the strain gauge
- Max deflection = 0.198 in, located at the point of load application
- ε_{xx} at strain gauge = 1.0054 * 10^{-3} in/in

7. **Torque wrench sensitivity in mV/V using strains from the FEM analysis**
   To determine the torque wrench sensitivity, we set up a coordinate system located at the center of the strain gauge and probed the normal strain at that location. We found that the strain was

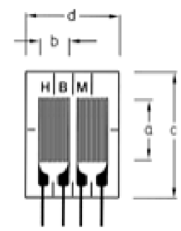
1.0054*10^{-3} in/in, meaning the torque wrench sensitivity is 1.0054 mV/V, meeting the requirement of having a sensitivity of at least 1.0 mV/V.

8. **Strain gauge selected (give type and dimensions). Note that design must physically have enough space to bond the gauges.**
To select a strain gauge, we referenced the following catalogue from HBK:
https://www.hbkworld.com/en/products/support-resources/support-hbm/downloads/product-literature/product-literature-sensors-transducers/product-literature-strain-gauges

First, we selected the geometry of the strain gauge, which depends on the type of measurement being made, such as strain in 1D, determining shear stress, etc. Since we are modeling the torque wrench handle as a beam and using the max bending moment applied to the handle as the torque, the **double linear strain gauge (DY1)** is the most suitable. This geometry comprises two measuring grids arranged in parallel. From there, we chose the DY11, since its temperature response is matched to steel, which is the material that our torque wrench is made out of. This means that the coefficient of thermal expansion for the strain gauge matches that of the material, reducing error from differences in thermal expansion. Specifically, we chose the **1-DY11-6/350,** with the following specs:

Types available from stock		Variants	Noml. resis-tance	Dimensions (mm)				Maximum excitation voltage (*)	Sldr. term-inals
				Measuring grid		Meas. grid carrier			
Steel	Aluminum	Other	Ω	a	b	c	d	V	
1-DY11-3/350	1-DY13-3/350	1-DY1x-3/350	350	3	2.7	9	8	9	LS 7
1-DY11-6/350	1-DY13-6/350	1-DY1x-6/350	350	6	3.2	12.5	9.4	14	LS 7

(*) Maximum excitation voltage for ferritic steel. For other temperature response matchings, the corresponding value is printed on the data sheet included with delivery.



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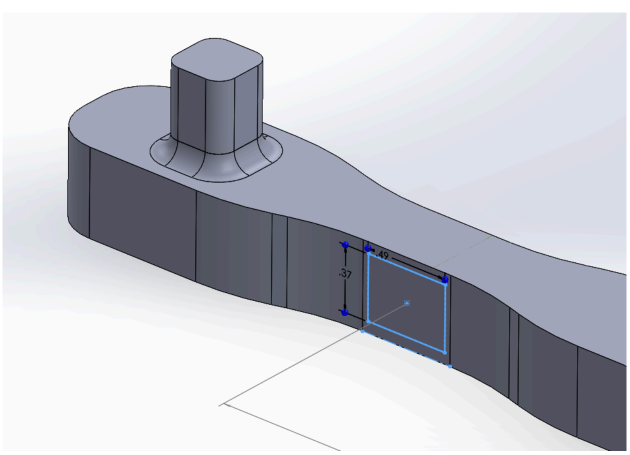
The following sketch shows where the strain gauge would be placed to verify that there is enough room on the torque wrench for the strain gauge.



9.4 mm = 0.37 in.
12.5 mm = 0.49 in

Validation of Functional Requirements

1) Attain at least 1.0 mV/V output at the rated torque of 600 in-lbf
   Met; according to Figure 7, our torque wrench sensitivity is 1.0054 mV/V.

2) Safety factor of $X_o = 4$ for yield
   $$X_o = \frac{\sigma_o}{\sigma_{max}} = \frac{260*10^3}{\sigma_{max}}$$
   Taking σ_{max} as 33966 psi (max normal stress from Figure 5b), $X_o = 7.65$
   → The required safety factor of 4 is exceeded.

3) Safety factor of $X_K = 2$ for crack growth from an assumed crack of depth 0.04 inches
   $X_o = \frac{K_{IC}}{K_{max}} = \frac{30.4*10^3}{F\sigma_{max}\sqrt{\pi a}}$ with F = 1.1 (based on the geometry of an edge crack)
   Taking σ_{max} as 33966 psi, $X_K = 2.30$
   → The required safety factor of 2 is exceeded.

4) Fatigue stress safety factor of $X_S = 1.5$
   $$X_K = \frac{\sigma_{fatigue}}{\sigma_{max}} = \frac{78.3*10^3}{\sigma_{max}}$$
   Taking σ_{max} as 33966 psi (max normal stress from Figure 5b), $X_S = 2.3$
   → The required safety factor of 1.5 is exceeded.