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MAE 3270

Final Homework - Part 1

5.1 Baseline Design

5.1.1 Results

1. **Script used for hand calculation.**

```
M = 600; % max torque (in-lbf)
L = 16; % length from drive to where load applied (inches)
h = 0.75; % width
b = 0.5; % thickness
c = 1.0; % distance from center of drive to center of strain gauge
E = 32.E6; % Young's modulus (psi)
nu = 0.29; % Poisson's ratio
su = 370.E3; % tensile strength use yield or ultimate depending on
material (psi)
KIC = 15.E3; % fracture toughness (psi sqrt(in))
sfatigue = 115.e3; % fatigue strength from Granta for 10^6 cycles
name = 'M42 Steel'; % material name
I=b*h^3/12;
F=M/L;
c_new=h/2;
load_point_deflection= M*L^2/(3*E*I);
max_normal_stress= M*c_new/I;
a=0.04; %crack size of 0.04 in
x = a/h;
Y=1.12; %for a very short crack use Y = 1.12
Kmax = Y*max_normal_stress*sqrt(pi*a);
safety_factor_for_strength=su/max_normal_stress;
safety_factor_for_crack_growth=KIC/Kmax;
safety_factor_for_fatigue=sfatigue/max_normal_stress;
M_bending=F*(L-c);
sigma_bending=M_bending*c_new/I;
strain_at_guage=sigma_bending/E;
output=strain_at_guage*1000;
load_point_deflection
max_normal_stress
safety_factor_for_strength
safety_factor_for_crack_growth
safety_factor_for_fatigue
strain_at_guage
output
```

Screenshot:

/MATLAB Drive/MAE 3270 Final HW/baseline.m

```
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load_point_deflection
max_normal_stress
safety_factor_for_strength
safety_factor_for_crack_growth
safety_factor_for_fatigue
strain_at_guage
output

```

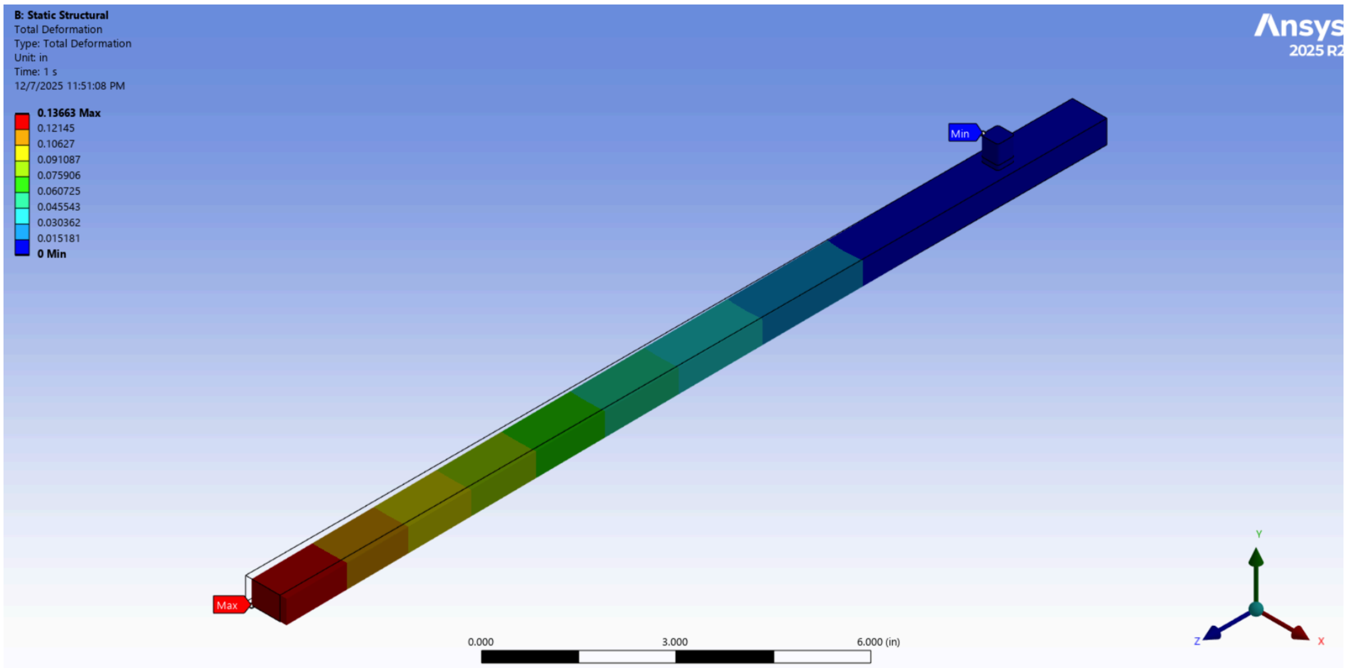
2. **Results from hand calculation of base design showing maximum normal stress (anywhere), strains at the strain gauge locations and deflection of the load point.**

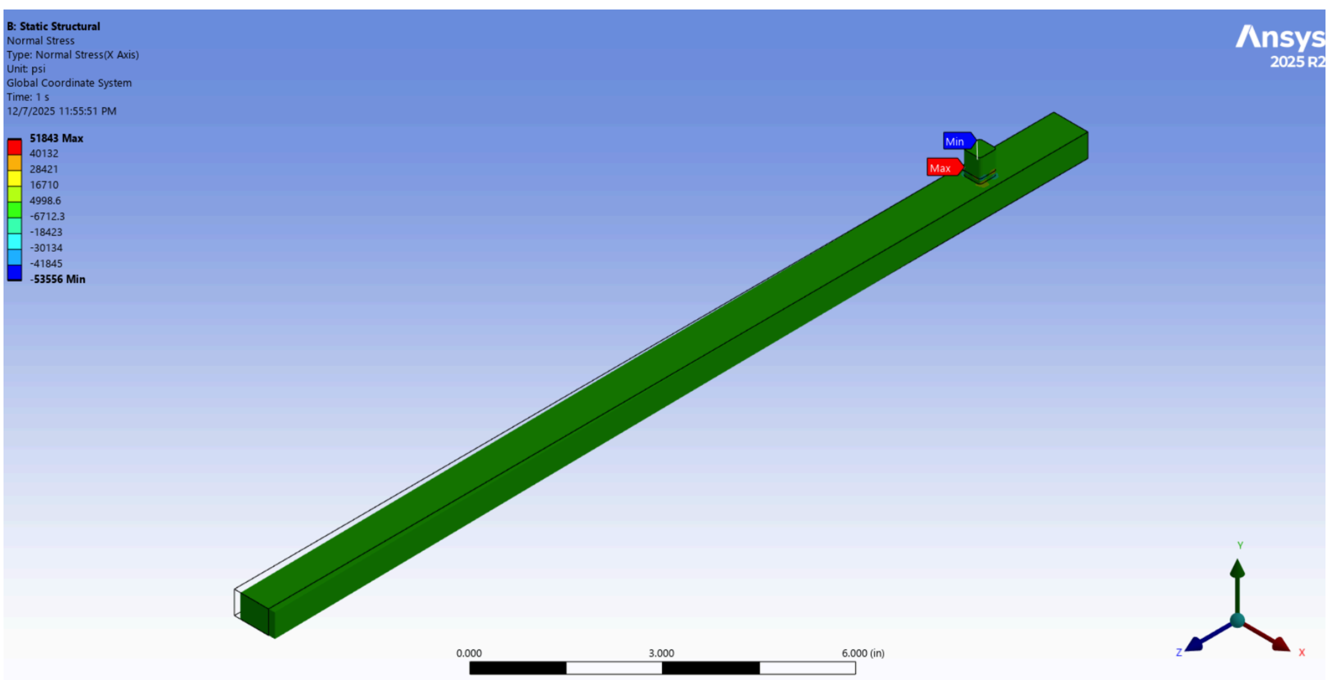
Results from MATLAB script with input baseline parameters shown above:

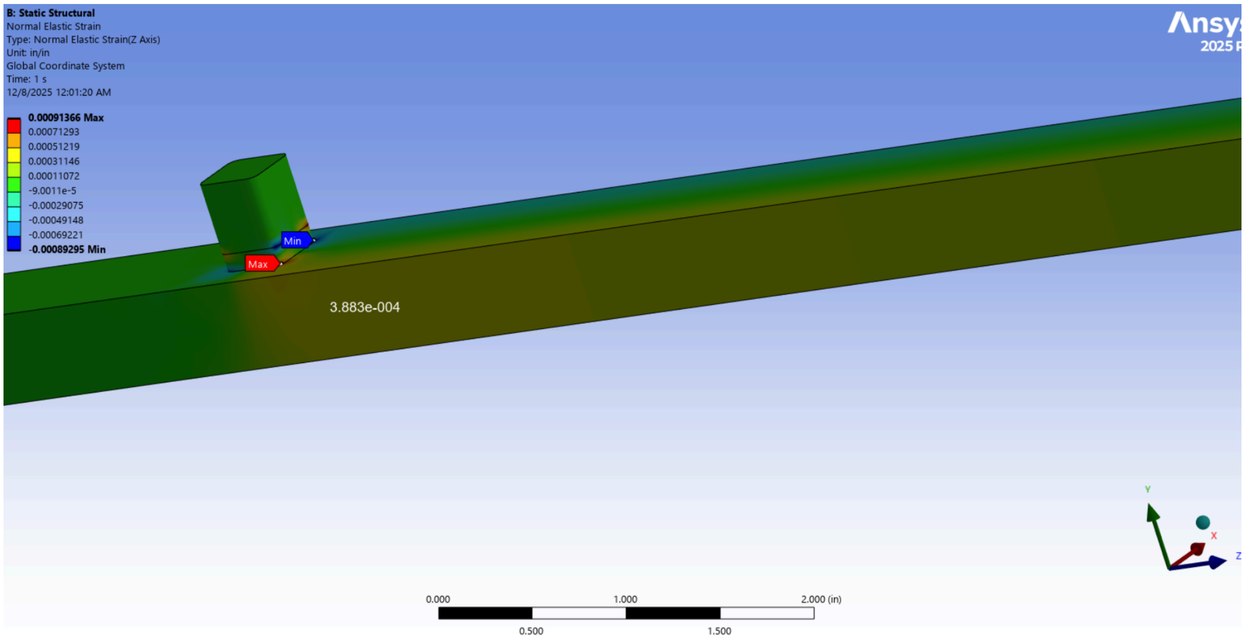
- load_point_deflection = 0.0910 in
- max_normal_stress = 12800 psi = 12.8 ksi
- safety_factor_for_strength = 28.9062
- safety_factor_for_crack_growth = 2.9516
- safety_factor_for_fatigue = 8.9844
- strain_at_guage = 3.7500e-04 = 375 microstrain
- output (torque sensitivity) = 0.3750 mV/V

These results match Professor Zehnder's expected outputs therefore we can conclude that our model represents the torque wrench baseline design closely and reliably.

3. **Results from FEM calculation of base design. From the FEM find the maximum normal stress (anywhere), strains at the strain gauge locations and deflection of the load point.**
    - Deflection of the load point: 0.13663 in
    - Maximum normal stress: 51,843 psi
    - Strain at the strain gauge location: 388.3 microstrain relative to z-axis



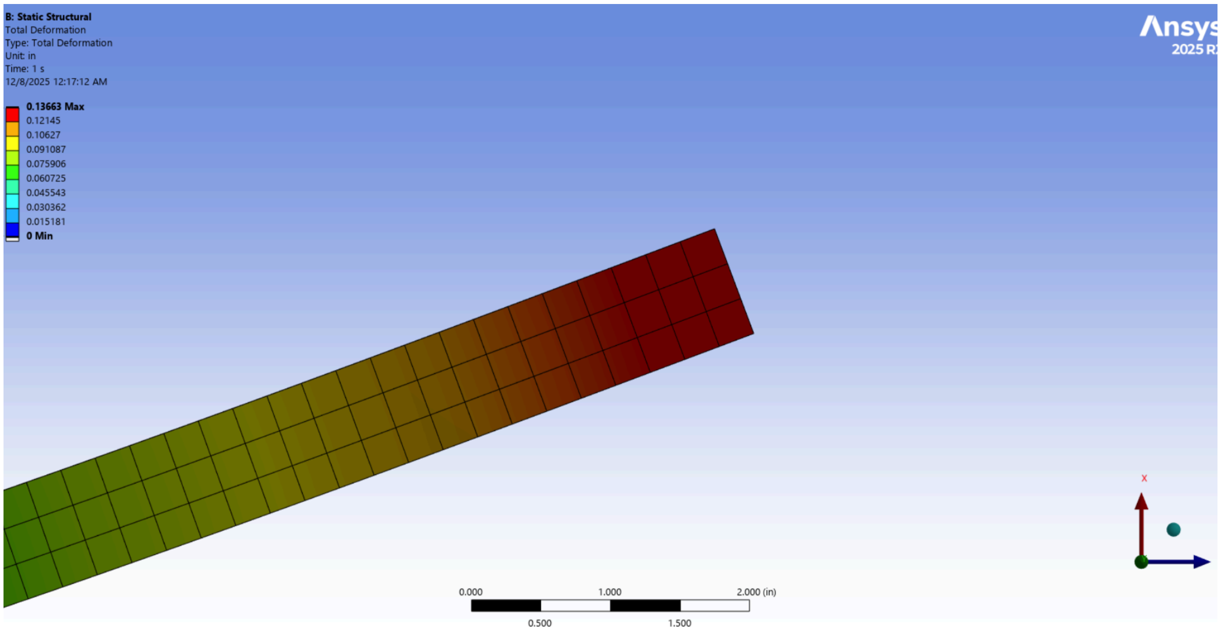




5.1.2 Reflections

1. **Beam theory assumes that plane sections remain plane. View the deformed mesh and check if mesh lines that cut across the beam handle remain as straight lines. Do you think that beam theory is reasonably accurate?**

   When examining the deformed mesh, the mesh lines across the beam remain largely straight and orthogonal throughout most of the beam's length, indicating that plane sections before deformation remain approximately plane after deformation (picture of which is shown below). This observation supports the core assumption of beam theory. Near the drive, however, we see that the mesh begins to show slight distortion, suggesting localized deviations due to stress concentrations or boundary effects. Overall, the deformation pattern implies that beam theory is reasonably accurate for this case, as the beam's geometry, as it is sufficiently long and slender, minimizes shear deformation and warping. If the beam were shorter, thicker, or subjected to more complex loading, these assumptions would likely break down, reducing the accuracy of beam theory in other applications.



2. **How do the FEM and hand calculated maximum normal stresses compare? If they differ significantly, why?**

   The maximum normal stresses predicted by the FEM (51,843 psi) and hand calculations (12,800 psi) differ significantly. This discrepancy is primarily due to the way the drive is clamped relative to the handle, which introduces localized stress concentrations that are not captured in the simplified hand calculations. The hand calculations assume idealized beam theory and a uniform stress distribution, however, the FEM accounts for complex geometry, boundary conditions, and local stress amplifications near the clamp and other constraints. These localized effects lead to the much higher maximum normal stress observed in the FEM. However, it is important to note that the normal stress at the strain gauge location (~12,000 psi) closely matches the beam theory calculation from hand calculation, indicating that, despite the discrepancy in maximum overall stress, the FEM reliably predicts the stress in the regions critical for measurement and design evaluation at the gauge location.

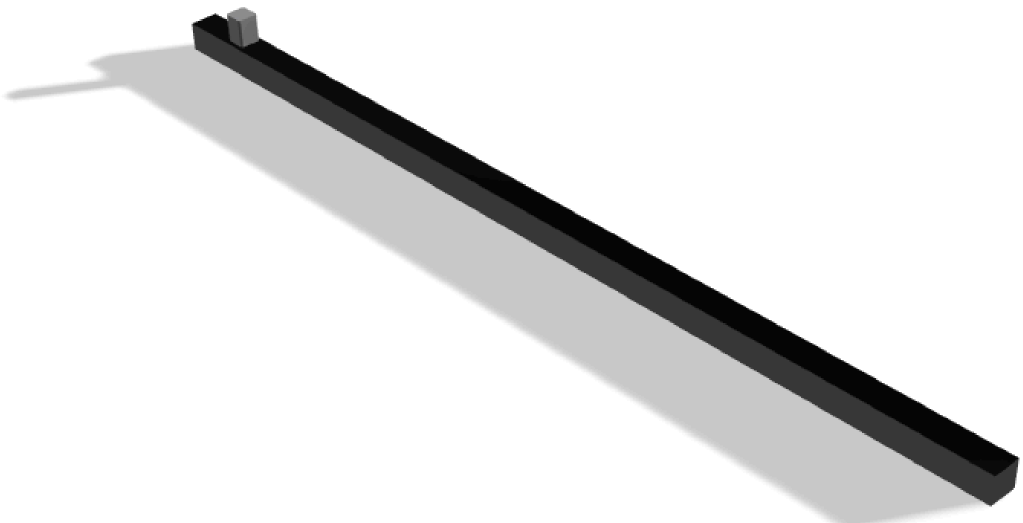
3. **How do the FEM and hand calculated displacements compare? If they differ, why?**

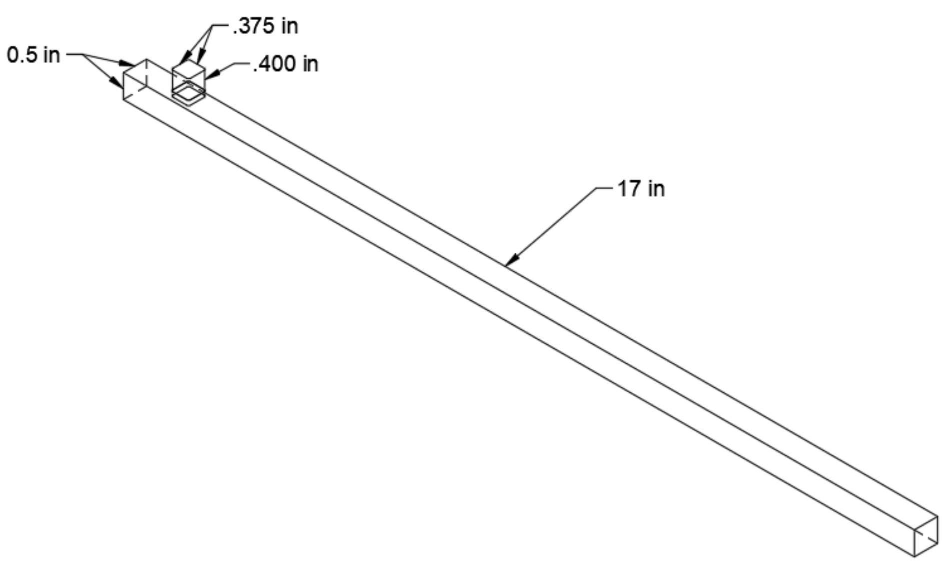
   The FEM and hand-calculated displacements are in close agreement, with only minor differences. The hand calculation predicts a load-point deflection of 0.0910 in, while the FEM predicts 0.13663 in. Both methods are grounded in the same beam theory principles, which govern the overall bending behavior of the torque wrench handle. The hand calculations assume an idealized beam with uniform cross-section, simple boundary conditions, and linear elastic behavior, which leads to a straightforward estimate of deflection at the load point. However, the FEM captures the full geometry of the handle, the detailed clamping of the drive, and the localized effects of stress concentrations and material constraints. These factors account for the slightly higher displacement predicted by the FEM. Overall, the close correspondence between the two approaches indicates that hand calculations are sufficiently accurate for estimating global deflection, while the FEM provides a more precise prediction of local deformations and load responses. This reinforces the validity of beam theory for most of the handle, while also highlighting FEM's advantage in capturing detailed variations near constraints.

5.2 My Design

5.2.1 Results

1. Images of CAD Model. Must show all key dimensions.

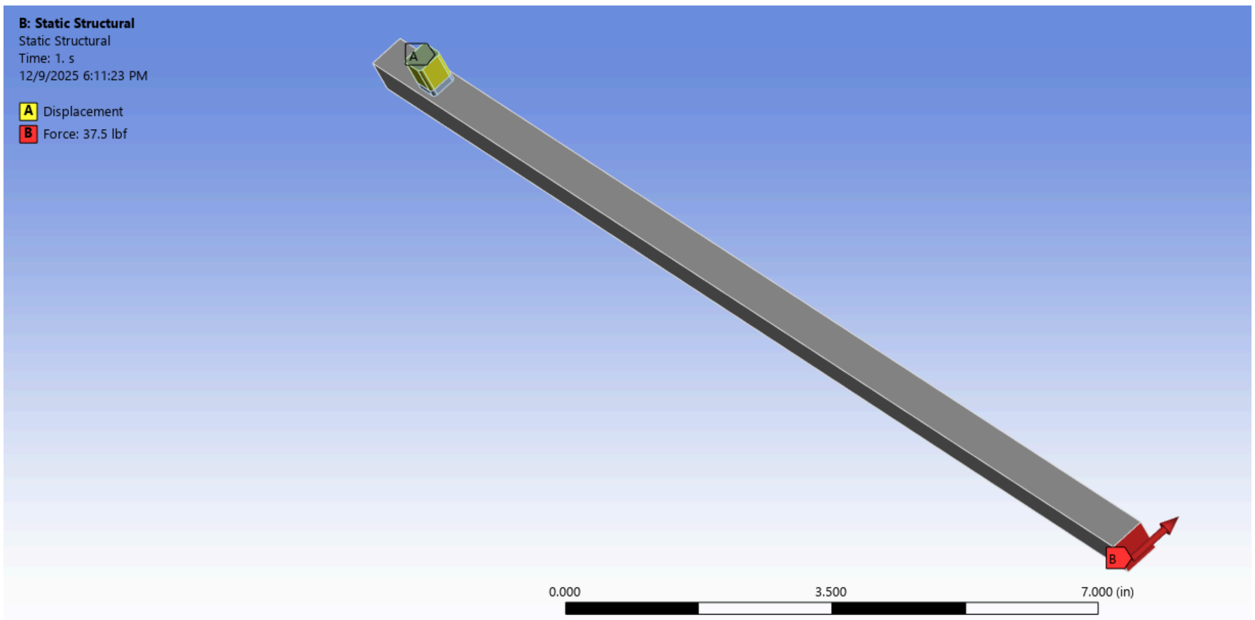




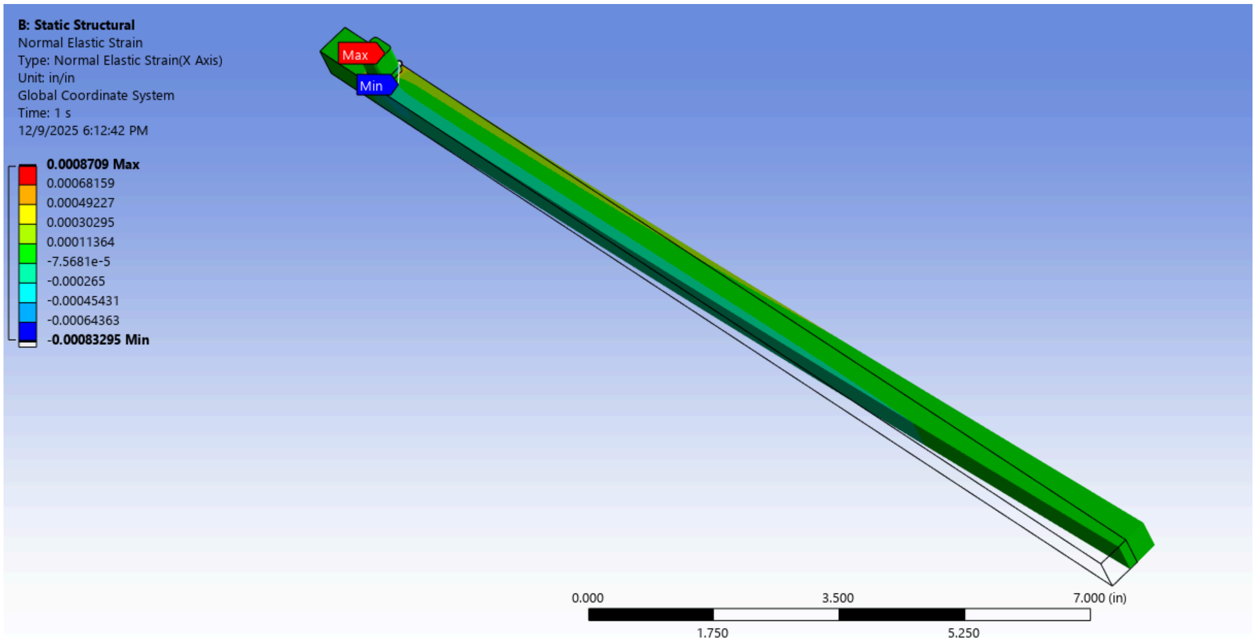
2. Describe material used and its relevant mechanical properties.

For this design, AF1410 Steel was chosen for its strength, hardness and fracture toughness. Typically, AF1410 is utilized for aerospace and defense applications in landing gear, airframes, and armor. The properties of this alloy were found on Granta: Elastic modulus (E) - 31 * 10^6 psi, poisson's ratio- 0.3, yield strength - 238 ksi, fracture toughness - 146 ksi*sqrt(in), and fatigue strength after 10^6 cycles - 109 ksi.

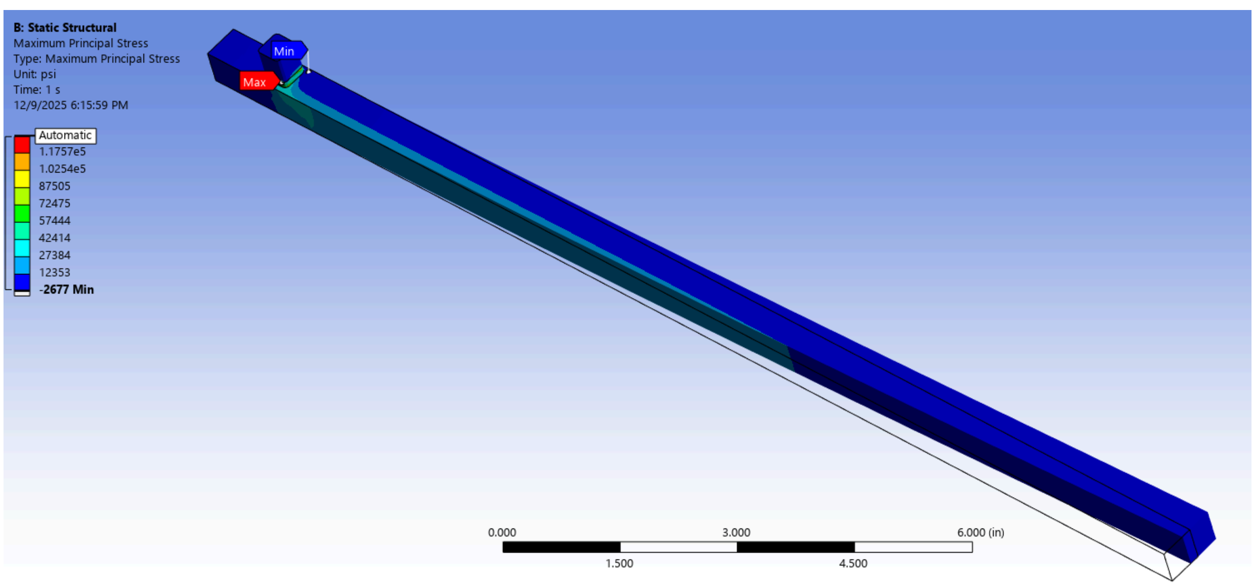
3. Diagram communicating how loads and boundary conditions were applied to your FEM Model.



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



5. Contour plot of maximum principal stress from FEM



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

FEM Results	
Strain at Gauge Location	872.9 Microstrain
Maximum Normal Stress	54292 psi
Max Deflection	0.35505 inches

These results are within a reasonable threshold of my hand-calculations.

```
Strain at Gauge                : 871.0 microstrain
Maximum Normal Stress           : 54200 psi
Tip Deflection                 : 0.3171 in
```

Differences likely come from small errors in CAD and/or computer rounding limitations.

7. Torque wrench sensitivity in mV/V using strains from the FEM analysis

```
Sensitivity                     : 2306 µV/V / 1000
Sensitivity                     : 2.306 mV/V
```

8. Strain gauge selected (give type and dimensions). Note that design must physically have enough space to bond the gauges.

For this design, a uniaxial strain gauge will be used to measure the strain in the 0° direction, since torque wrenches can only be twisted along an axis perpendicular to its shaft. Therefore, we should only be focusing on strains in the x-direction. The dimension of this gauge is typically 3 mm x 1.5 mm, with the carrier dimensions being typically 10 mm x 10 mm. This means that both the gauge and carrier should comfortably fit on the torque wrench design.