Sweksha Mehta and Jason Bak
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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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
```

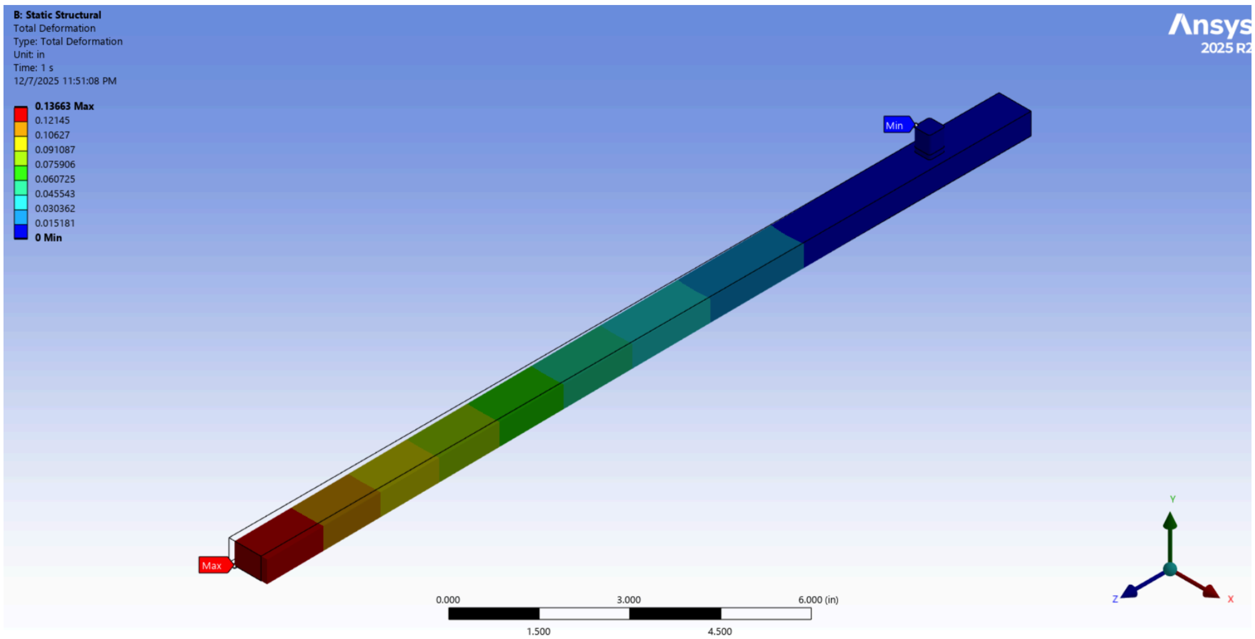
2. **Results from hand calculation of base design showing maximum normal stress (anywhere), strains at the strain gauge locations and deflection of the load point.**

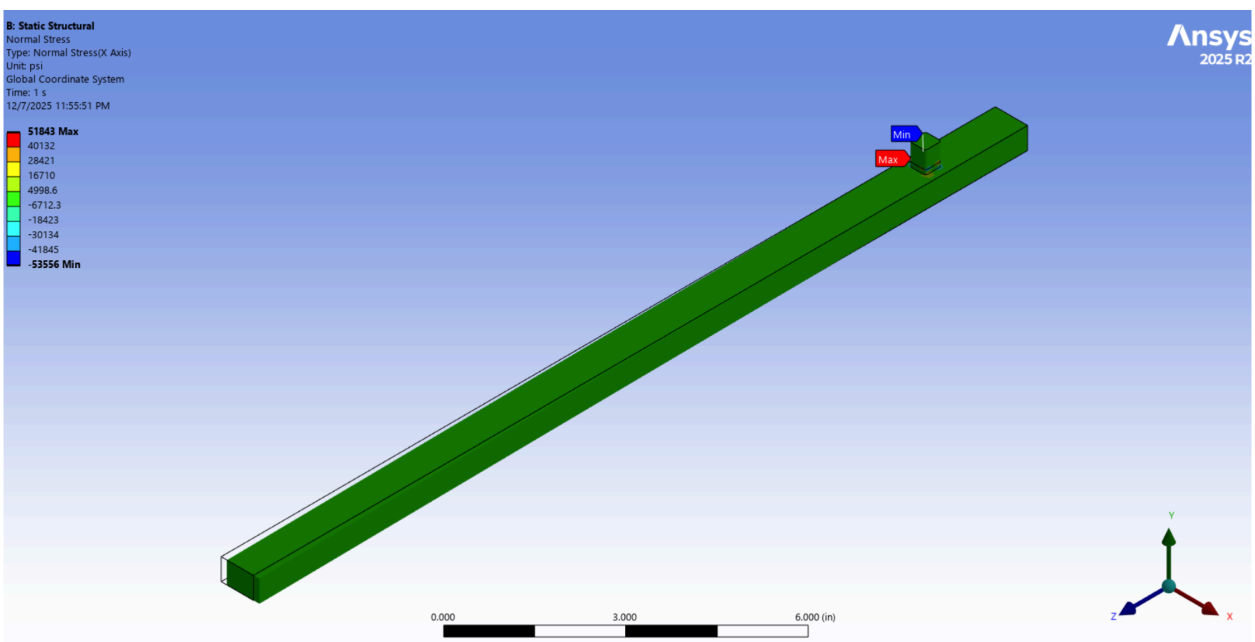
Results from MATLAB script with input baseline parameters shown above:

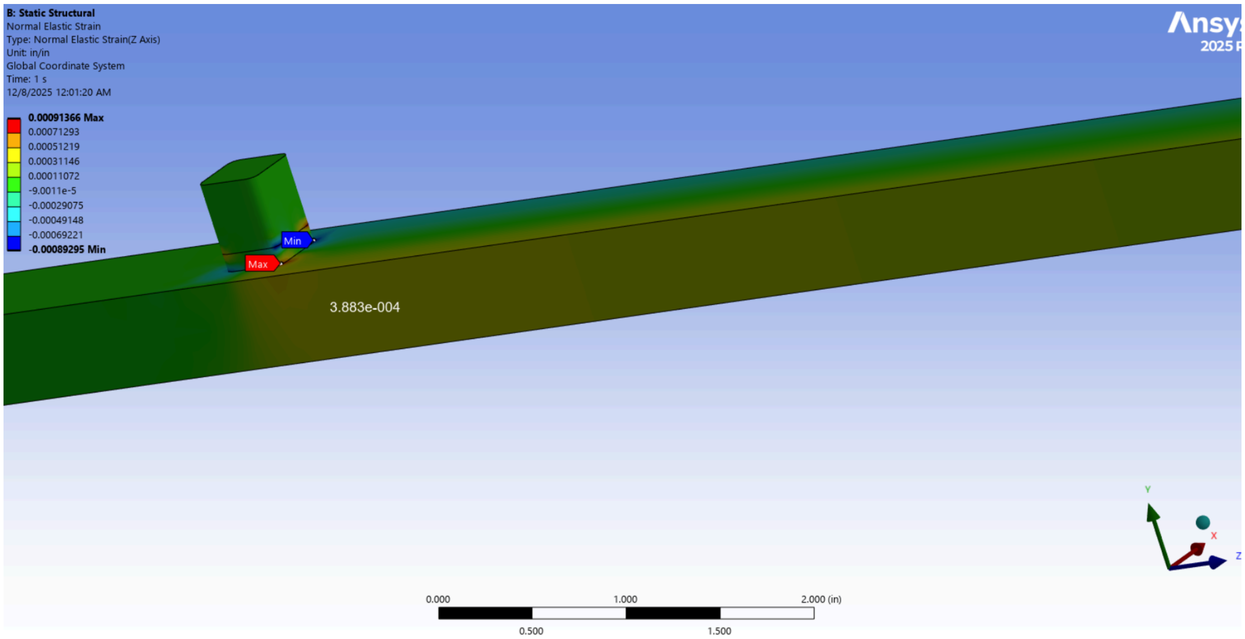
- load_point_deflection = 0.0910 in
- max_normal_stress = 12800 psi = 12.8 ksi
- safety_factor_for_strength = 28.9062
- safety_factor_for_crack_growth = 2.9516
- safety_factor_for_fatigue = 8.9844
- strain_at_guage = 3.7500e-04 = 375 microstrain
- output (torque sensitivity) = 0.3750 mV/V

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



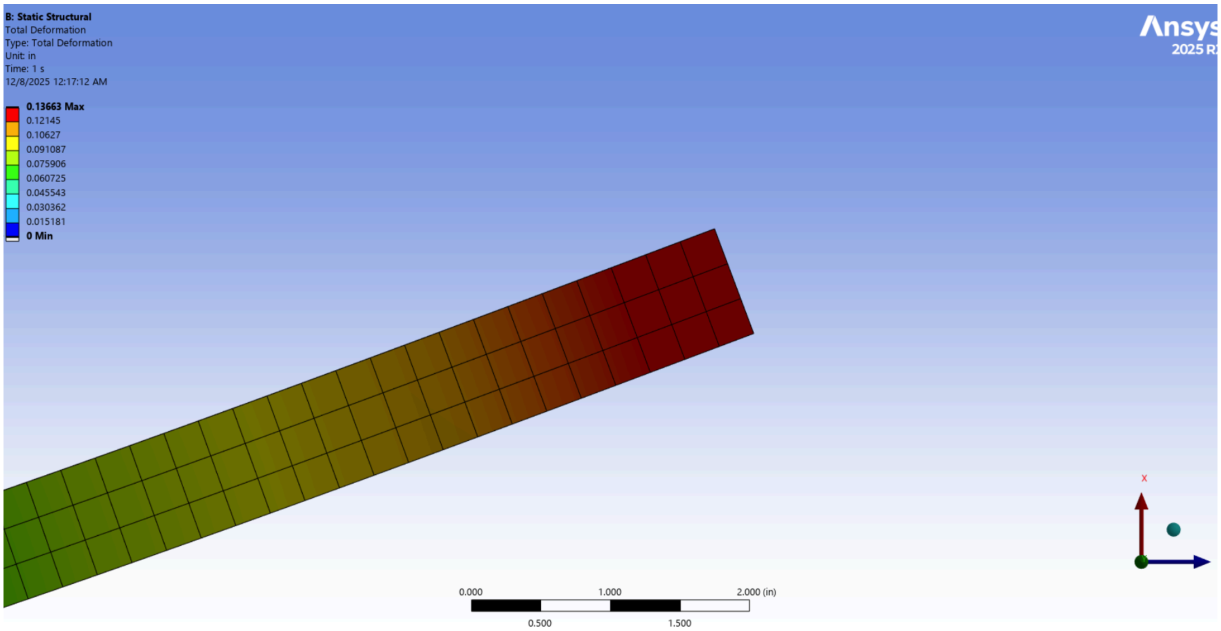






*5.1.2 Reflections*

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.

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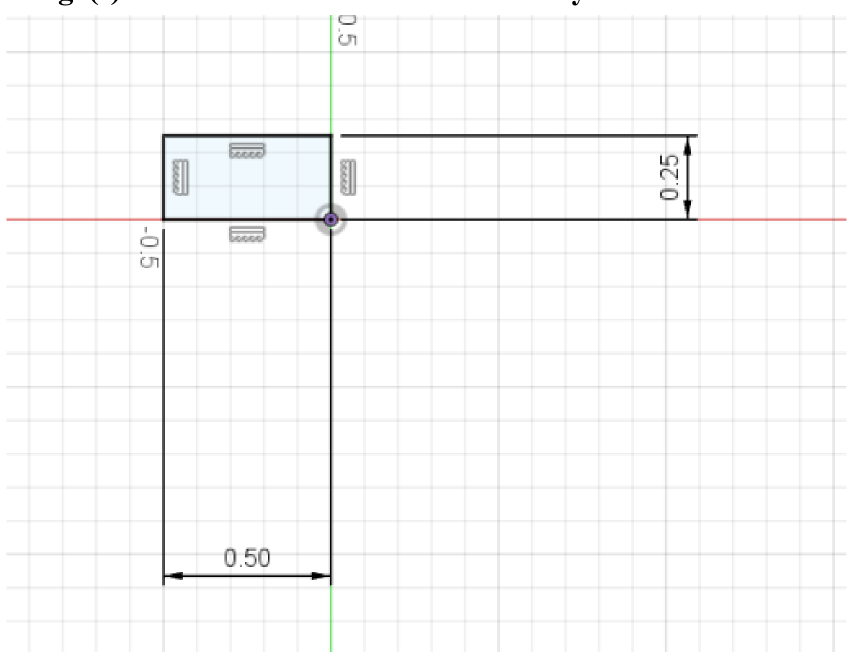
MAE 3270

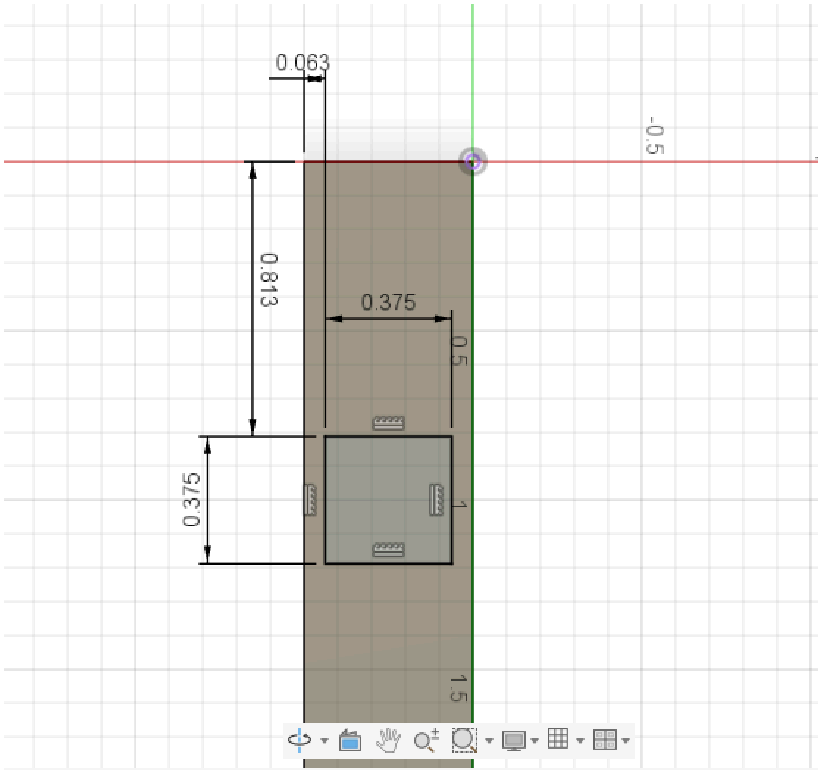
# Final Homework - Part 2

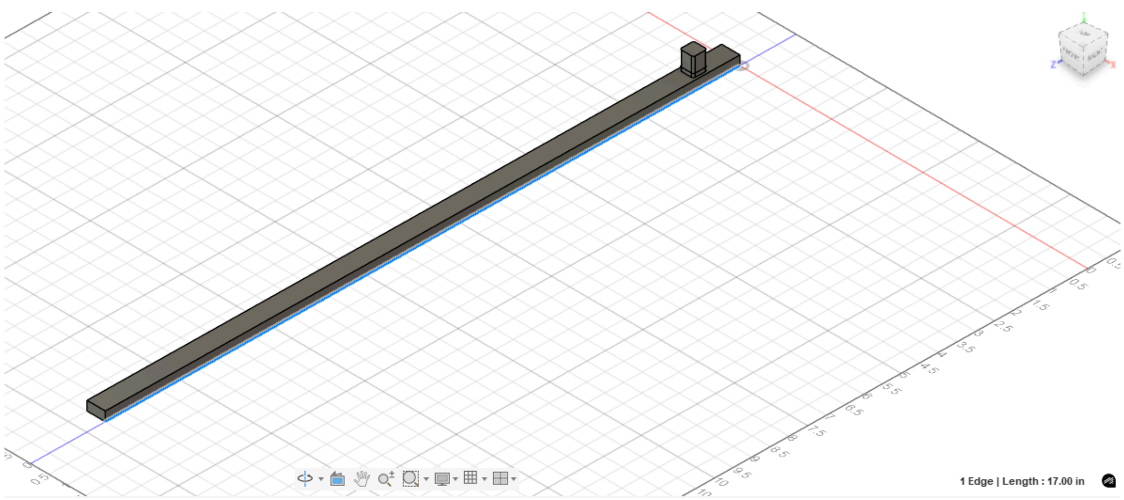
## 5.2 My Design, Upload to portfolio

*5.2.1 Results*

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





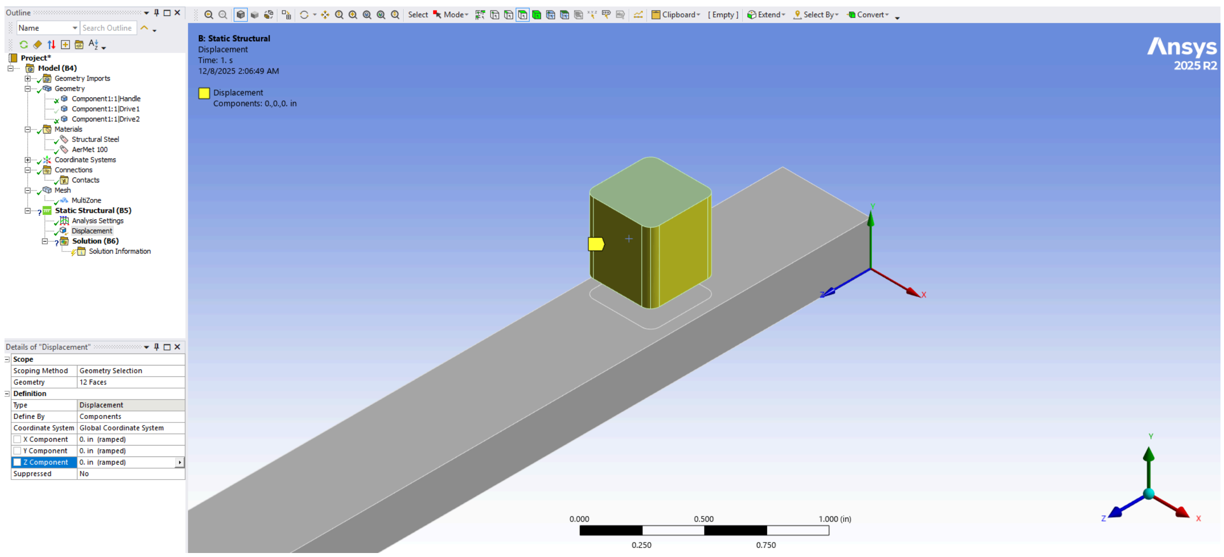

2. **Describe material used and its relevant mechanical properties.**
   The material selected for the torque wrench design is AerMet 100, a high-strength steel alloy widely used in aerospace and defense applications. AerMet 100 is chosen for its exceptional combination of tensile strength, fracture toughness, and fatigue resistance, making it well-suited for components subjected to high torque and cyclic loading like this torque wrench scenario.

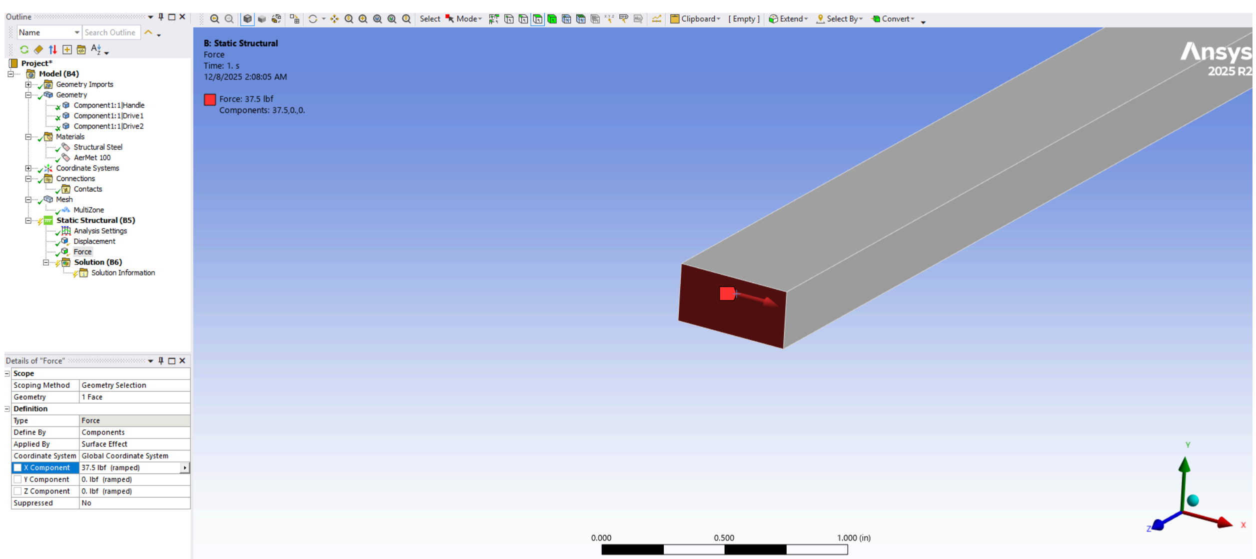
   Mechanical properties of AerMet 100 from Granta are:
   - Young's Modulus (E): $29 \times 10^6$ psi
   - Poisson's Ratio (ν): 0.31

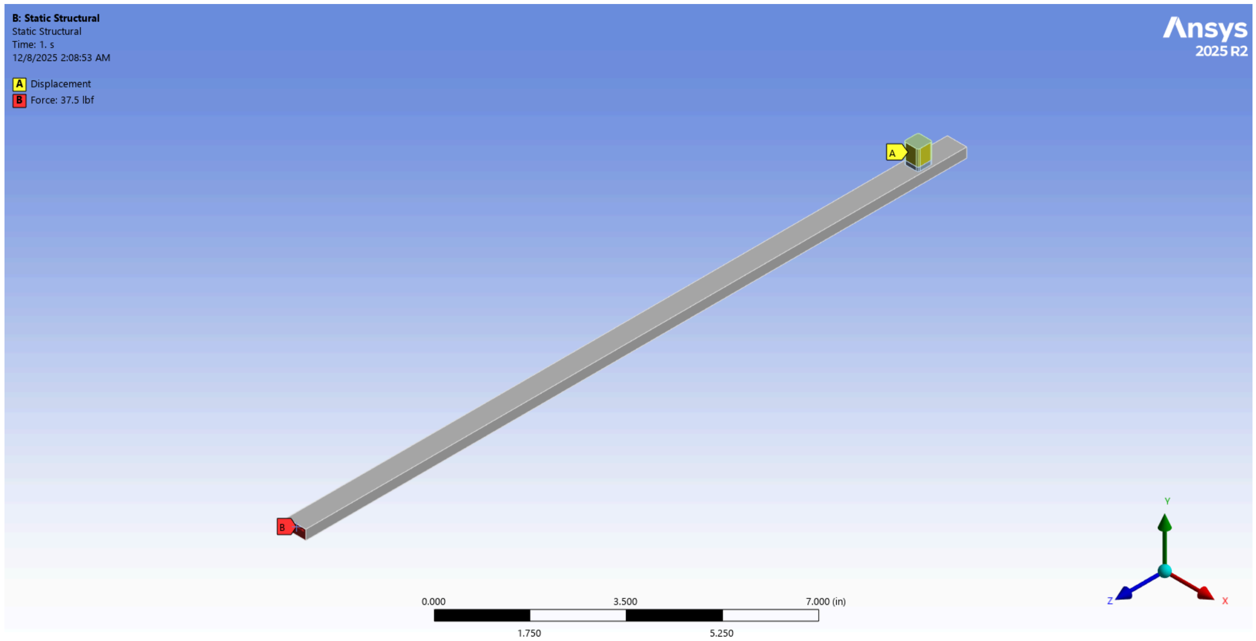
- Ultimate Tensile Strength: $310 \times 10^3$ psi,
- Fracture Toughness): $137 \times 10^3$ psi$\sqrt{\text{in}}$,
- Fatigue Strength: $131 \times 10^3$ psi (for $10^6$ cycles)

Given the design constraints, a maximum applied torque of 600 in-lbf, a handle length of 16 in, cross-sectional dimensions of 0.25 in by 0.5 in, and a strain gauge centered 1 in from the drive, AerMet 100 provides the required strength, stiffness, and fatigue resistance to meet or exceed all safety factor requirements. Its mechanical properties allow the torque wrench to sustain repeated loading cycles while maintaining a high signal output from the strain gauges.

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



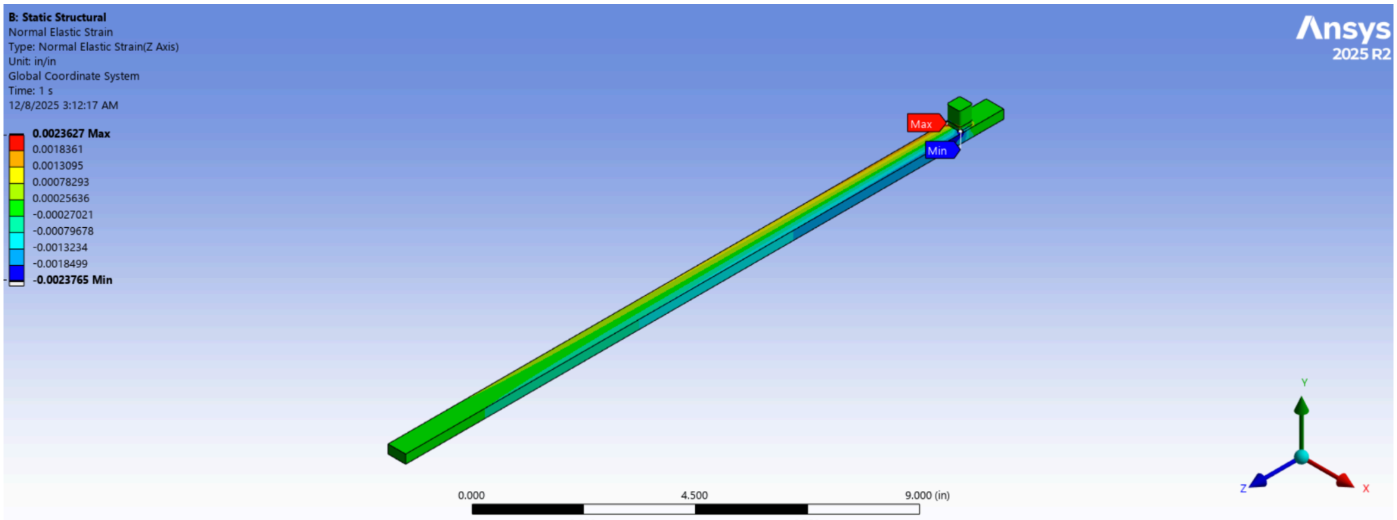



In the finite element model, the torque wrench was analyzed under the rated torque of 600 in-lbf. The setup replicates realistic operating conditions and matches the assumptions used in the hand calculations.

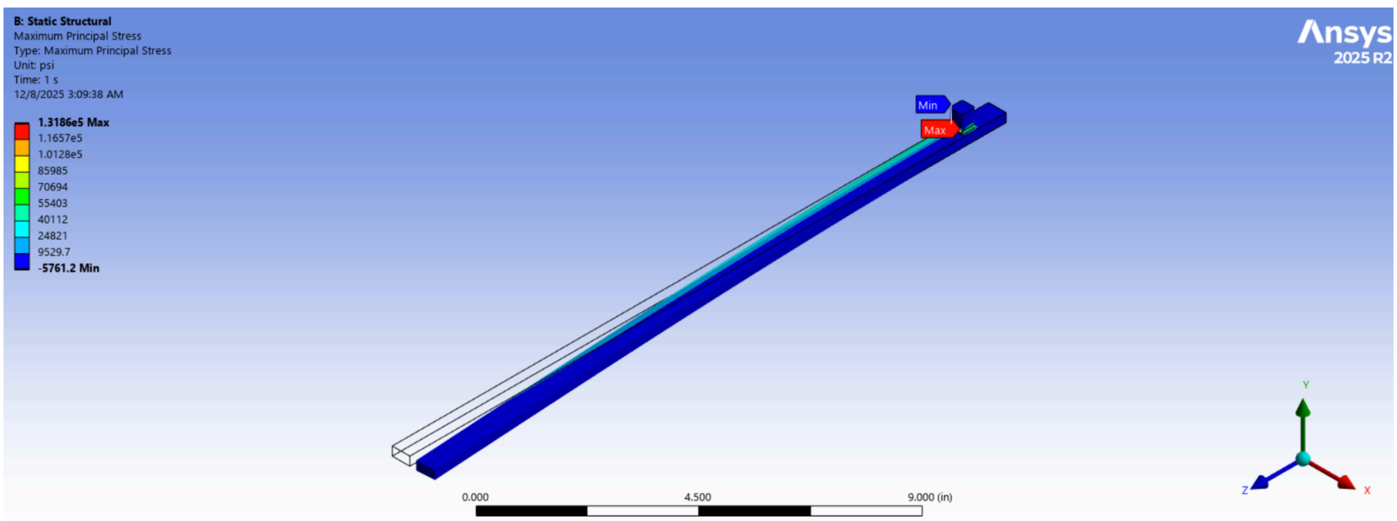
- Region A (Clamped Boundary Condition):
  The upper 0.4 inches of the drive were modeled as fixed supports, representing the portion of the wrench engaged with the socket drive. This region was fully constrained in all translational and rotational degrees of freedom to simulate a rigidly clamped connection.
- Region B (Applied Load):
  A lateral force corresponding to a torque of 600 in-lbf was applied at a distance of 16 inches from the drive. The applied force was calculated as T/L, where T is the applied torque and L is the length from the drive to the point of load application. This force was applied on the handle surface in the direction perpendicular to the wrench axis to generate the desired twisting moment.

  Both of these configurations produce the same loading conditions used in the analytical design calculations and allow direct comparison between the FEM results (strain, stress, and deflection) and hand-calculated values.

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

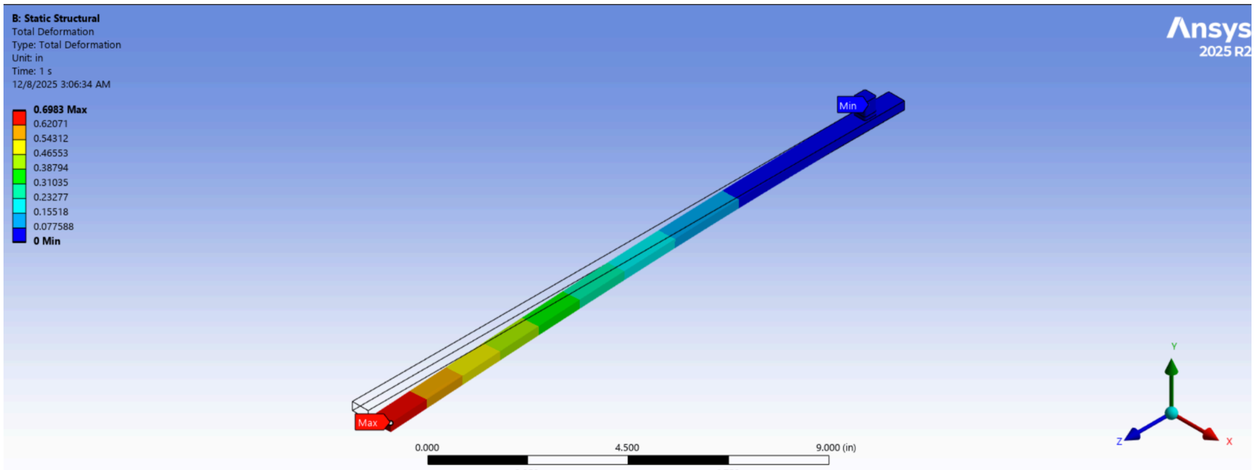


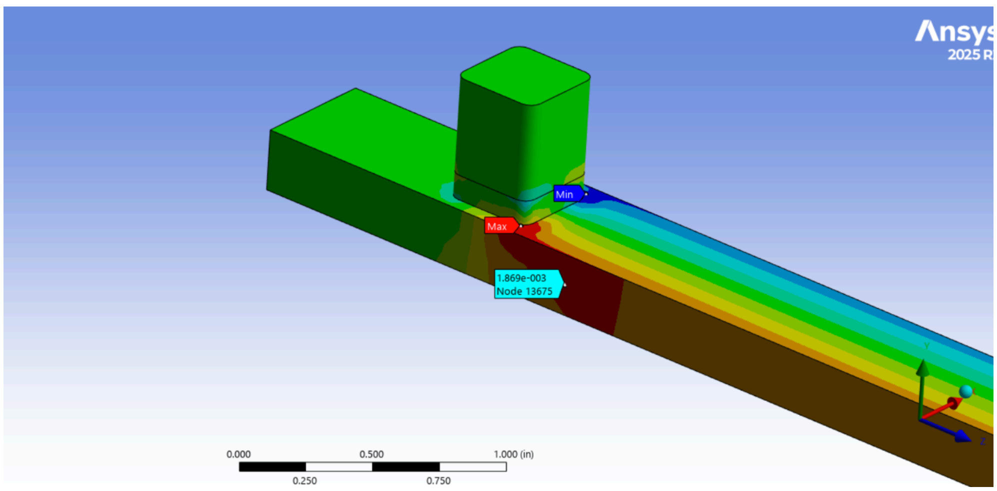

### 5. Contour plot of maximum principal stress from FEM




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

The finite element analysis was conducted on the improved AerMet 100 torque wrench design under an applied torque of 600 in-lbf. The results indicate that the maximum normal stress in the model occurs with a value of 52,114 psi. The load point deflection was found to be 0.6983 inches. At the strain gauge location (1 inch from the drive center), the normal strain was 0.001869. These results confirm that the design performs well under the rated torque, remaining safely below failure criteria and providing sufficient output for accurate torque measurement.




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

Torque sensitivity calculation

$\varepsilon$ = strain @ gauge location: $1869 \cdot 10^{-6}$

half-bridge w/ 2 gauges at opposite strains:

K = 2 (gauge factor)

$$\frac{V_{out}}{V_{in}} = \frac{K(\varepsilon_1 - \varepsilon_2)}{4} = \frac{2 \cdot (1869 \cdot 10^{-6} - (-1869 \cdot 10^{-6}))}{4} = 1869 \cdot 10^{-6} \rightarrow 1.869 \text{ mV/V}$$

8. **Strain gauge selected (give type and dimensions). Note that design must physically have enough space to bond the gauges.**
   The strain gauge selected for this torque wrench design is a CEA-06-125UNA-350 from Vishay Precision Group (Micro-Measurements). This gauge design is a general-purpose uniaxial strain gauge. The gauge has an active length of 0.125 inches, width of 0.095 inches, and a gauge factor of approximately 2.1. Given the torque wrench handle dimensions of 0.25 inches by 0.5 inches, the gauge fits comfortably on the 0.5-inch surface at the specified location 1 inch from the drive center, with ample space for proper adhesion and wiring. For the half-bridge configuration calculations done above, two of these gauges are mounted at the same axial location on opposite surfaces of the handle which would maximize signal output, resulting in the calculated sensitivity of 1.869 mV/V at the rated torque of 600 in-lbf.