Torque Wrench Design and FEM Analysis

MAE 3270 - Mechanics of Materials

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Abstract

My final project in my Mechanics of Materials course (MAE 3270) was to design a torque wrench such that it met certain criteria, then analyze it with FEM to verify my results. Given a baseline design and some required parameters, I changed the design, material, and strain gauge type to meet the safety factor and sensitivity requirements.

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1 Overview

We were initially provided a baseline design of a torque wrench attached to a drive with a 600 in-lbf torque applied. This was modeled as a cantilever beam in bending with a load applied at the end of the beam.

The first part of the assignment involved calculating the maximum normal stress, the strain gauge output, and the safety factors for yield, fracture, and fatigue for the baseline design. We then compared our results to an FEM analysis, where it was apparent that there were stress concentrations where the drive and wrench attached, meaning the actual maximum normal stress is greater than the bending stress predicted by the hand calculations.

The second part of the project involved designing an improved model with a different geometry, material, and/or strain gauge. The requirements we had to meet included using a steel, aluminum or titanium alloy as the material and attaining at least:

- 1.0 mV/V output at the rated torque of 600 in-lbf.
- Safety factor of $X_O = 4$ for yield or brittle failure
- Safety factor of $X_K = 2$ for crack growth from an assumed crack of depth 0.04 inches.
- Fatigue stress safety factor of $X_S = 1.5$

2 CAD Design

Starting with the baseline design, I decided to keep the width and height of the beam the same, since although increasing them will increase the bending stress and therefore the strain gauge sensitivity, they will also decrease the factors of safety. To increase sensitivity, I made the length of the beam a little longer.

My primary change was adding fillets between the drive base and beam to reduce the additional stress due to stress concentrations. Keeping all other variables constant, I tried a few different radii for both these fillets and those of the drive's vertical edges, deciding on the pair that resulted in the smallest maximum normal stress in the FEMs. Figure 1 is a CAD drawing of my final design.

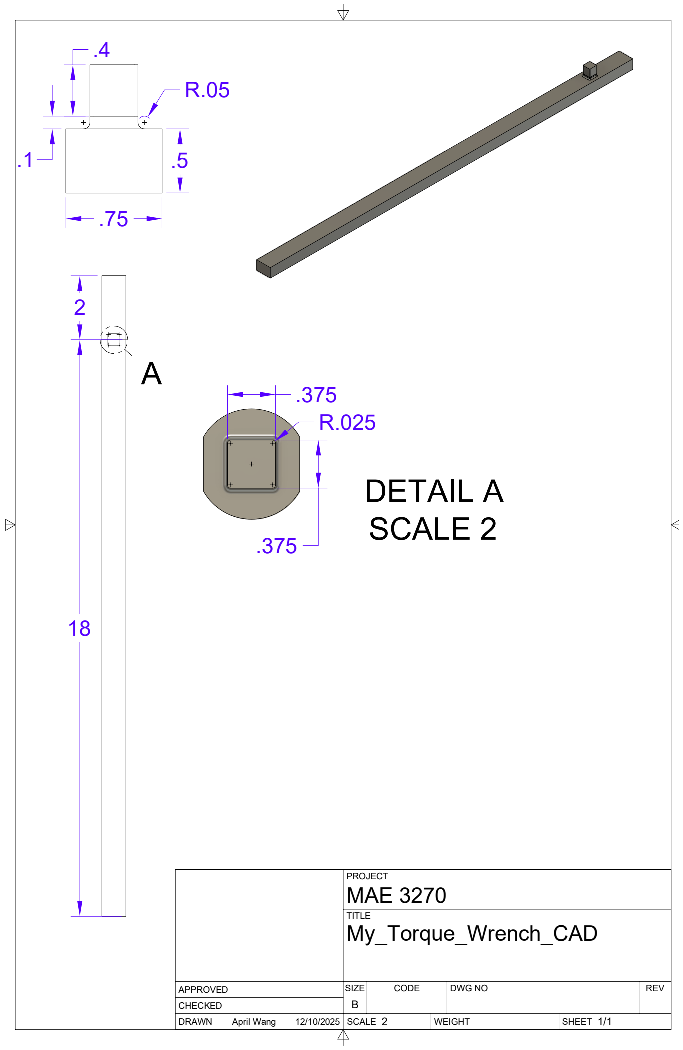

Figure 1: Fusion 360 CAD drawing of torque wrench design with all dimensions labeled.

3 Material Choice

In order to maximize the sensitivity of the strain gauge output while also meeting the required factors of safety, I chose a material that had a lower stiffness (E), but also high yield strength, fracture toughness, and fatigue strength. I did this by plotting each material failure property vs. Young's Modulus on Granta for steel, titanium, and aluminum alloys, using the baseline material values as the minimum cutoff for each material failure property.

Here is one plot with materials I considered using:

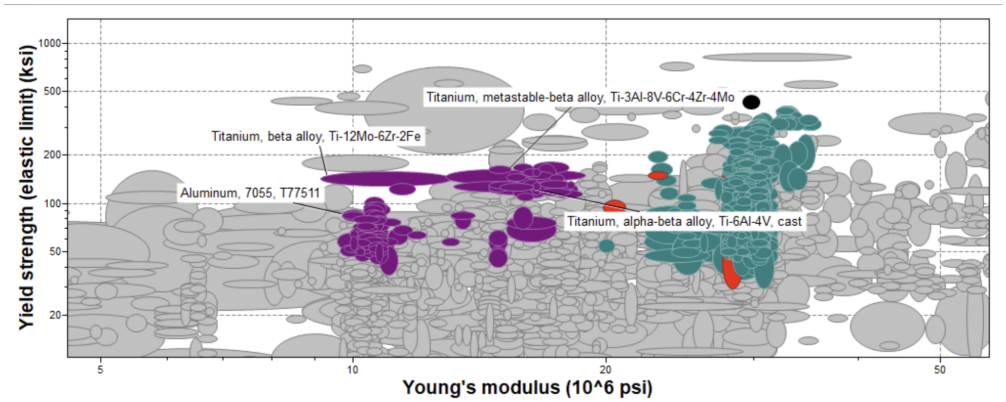

Figure 2: Yield strength vs. Young's Modulus for titanium (purple), aluminum (red), and steel (teal) alloys, plotted using Granta EduPack 2025 R2

The material I chose to use was Titanium, metastable-beta alloy, Ti-3Al-8V-6Cr-4Zr-4Mo. This is a strong and lightweight titanium alloy with the following material properties:

- Young's Modulus = 15.4×10^3 ksi
- Yield strength = 168×10^3 ksi
- Fracture toughness = 64.7 ksi$\sqrt{in}$
- Fatigue strength (10^6 cycles)= 115 ksi

4 Strain Gauge

To maximize the torque wrench sensitivity of the strain outputs, I chose to use a full bridge strain gauge for bending and axial load that integrates two gauges in the principal direction and two gauges in the transverse direction. This type of strain gauge produces the highest output for bending, with the output being $\frac{k}{4}\epsilon_g$, where $k \approx 2$ and $\epsilon_g = 2\epsilon_p - 2\epsilon_t$ for full bridge strain gauges.

The model I chose (figure 3) has dimensions of Length = 0.394, Width = 0.354, which fits on the side of the beam.

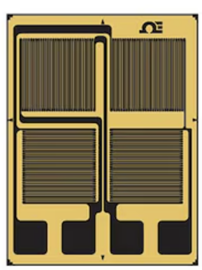

Figure 3: SGT-2-250-FB13-Strain-Gauge.

5 FEM Setup

As seen in figure 4, the upper portion of the drive is clamped, with the displacement of the vertical faces fixed in all directions. A load of 33.3 lbf is applied to the face at the end of the beam in the $+x$ direction to model the 600 in-lbf torque applied about the drive.

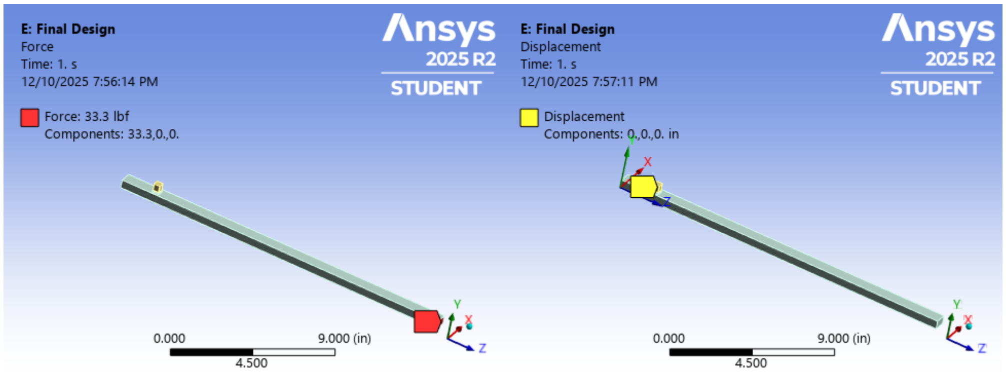

Figure 4: FEM Model with applied loads and boundary conditions.

6 FEM Results

As seen in figure 5, the normal strain varies along the z-axis, with the highest value at about 0.375 in from the center of the drive. Thus, I decided to place the strain gauge at 0.4 in from the drive center.

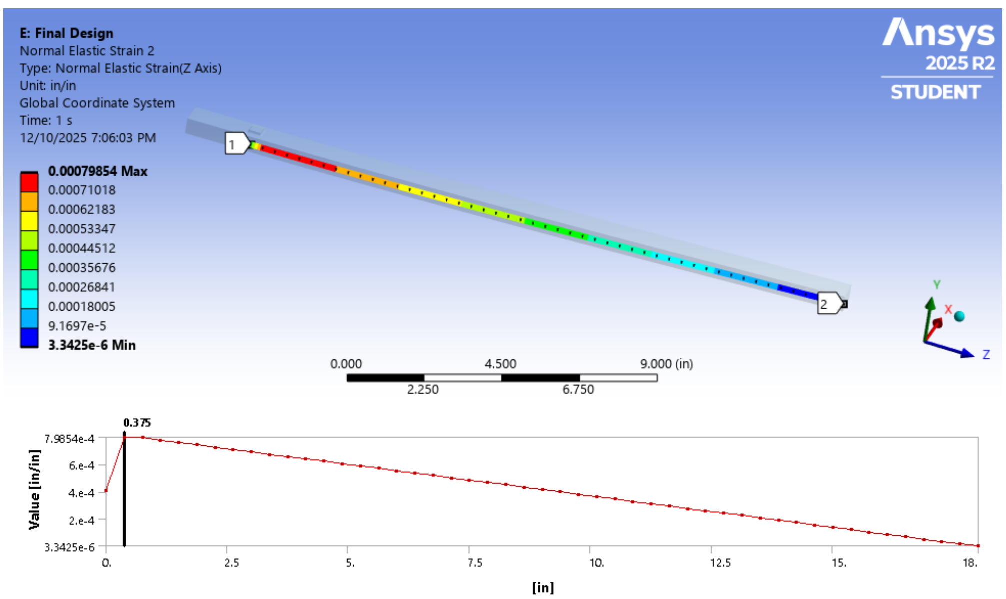

Figure 5: Normal strain contours along the bending axis of the beam.

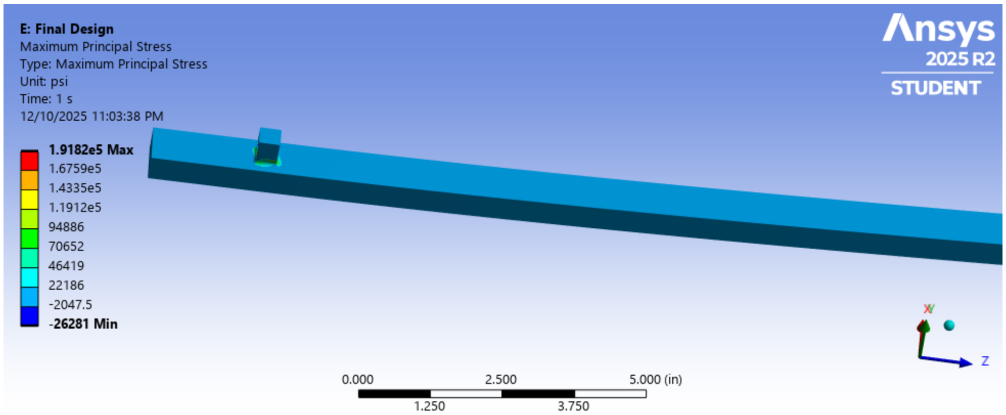

Figure 6: Contour plot of maximum principal stress on section plane where the drive and beam attach.

6.1 FEM Calculation Results

Based on the FEM calculations:

- Maximum normal stress (anywhere): **59 ksi** (7)
- Total deformation: **0.34 in** (8)
- Strain at strain gauge (0.4 in from drive center):
  - Principal stress (bending) direction: $\epsilon_1 = \epsilon_z = 800$ microstrain
  - Transverse direction (across width): $\epsilon_x = -294$ microstrain
  - Strain at gauge (full bridge): $\epsilon_g = 2\epsilon_z - 2\epsilon_x =$ **2188 microstrain**

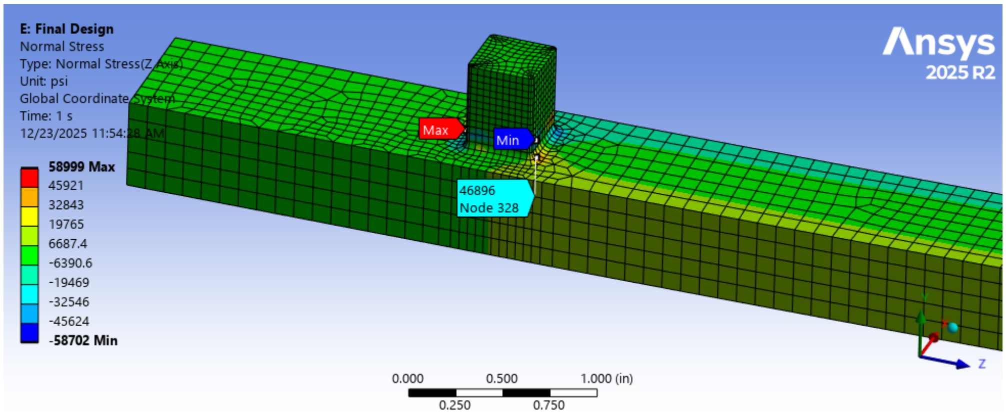

Figure 7: Contour plot of stress normal to z-axis

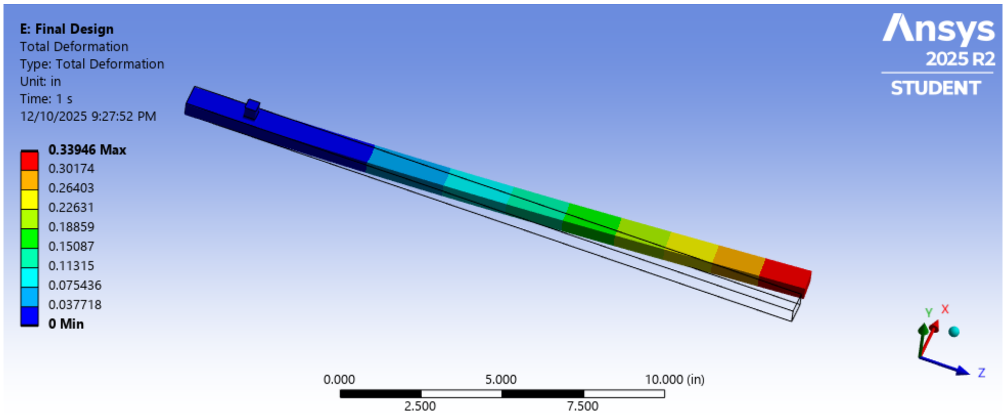

Figure 8: Contour plot of total deformation

Since the FEM creates artificial stress singularities at the edge of the clamped BC, the maximum normal stress there is significantly higher than the calculated value due to bending:

$$\sigma_{bending,max} = \frac{(600 \text{ in-lbf})(h/2)}{I} = 12.8 \text{ ksi}$$

Therefore, instead of using the maximum normal stress anywhere to calculate the factors of safety, the maximum normal stress where the drive meets the beam will be used: $\sigma_{N,max} = 47$ ksi (7). Although this value is also much higher than the calculated bending value, the stress concentrations at the corners do reflect real world conditions and are influenced less by the clamped BC.

The safety factors for each type of failure can be found by dividing the material's properties by their corresponding values calculated from the maximum normal stress (e.g. safety factor for yield: $X_o = \frac{\sigma_o}{\sigma_{N,max}}$.) The final results are:

- Safety factor for yield: $X_O = 7.15 > 4$
- Safety factor for crack growth from an assumed crack of depth 0.04 inches: $X_K = 3.41 > 2$
- Fatigue stress safety factor: $X_S = 2.45 > 1.5$
- Output at the rated torque of 600 in-lbf: 1.0940 mV/V > 1 mV/V

Even though the FEM has a significantly higher maximum normal stress value than that of pure bending, the design still meets all the required safety factors!