Torque Wrench Final Design Project

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Abstract:
This document shows the process of designing and testing a theoretical torque wrench to meet multiple factors of safety and design requirements. The wrench was analyzed both in Matlab using beam theory and in ANSYS using finite element analysis.

Wrench design:
The wrench dimensions were based on multiple design requirements, listed below.
- Factor of safety
    - Against fatigue - 1.5
    - Against yield - 4
    - Against fracture (initial crack of 0.04 in) - 2
- Strain gauge sensitivity of 1 mV/V or higher for a rated torque of 600 lbf-in
- Must use a steel, aluminum or titanium alloy.

To find the final dimensions of the torque wrench a Matlab script was written. The script computes the factor of safety for yield, fracture and fatigue given wrench dimensions as the input. Additionally it displays strain at the location of the strain gauge and the deflection at the end of the wrench. Values for base and height were changed until all requirements were met. AISI low alloy steel was chosen as the material, with its properties being listed as inputs. The full code is attached in the appendix.

Final dimensions:
Base = 0.45 inches
Height = 0.72 inches
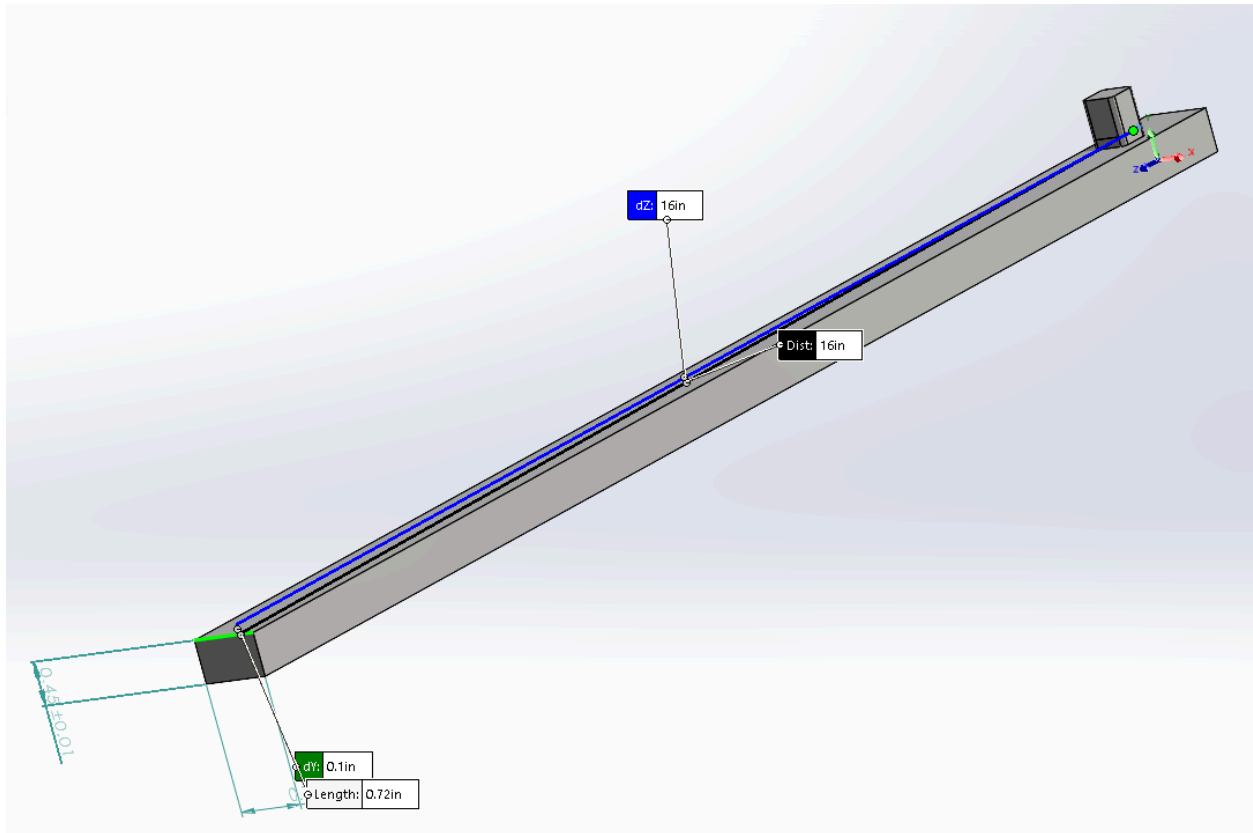Length (from end to drive socket) = 16 inches

Key metrics (for 600 lbf-in torque)
Maximum stress: σ_{max} = 15.43 ksi
Strain at strain gauge location = 497.8 $\mu\varepsilon$
Deflection at load point = 0.118 inches

1. Image of CAD model used for ANSYS analysis

**2. Describe material used and its relevant mechanical properties.**

Our design utilizes AISI 4140 low alloy steel. The Modulus of Elasticity is 3.1 * 10^7 psi and the Poisson's Ratio is 0.29. It was chosen for its strength and resistance to fatigue. One alternative option was to use annealed AISI 4140 steel, which has a lower elastic modulus. This would have helped meet the 1 mV/V strain gauge sensitivity requirement by increasing the overall strain, but it also would have decreased the overall strength of the wrench. In the end, we managed to find a design that fulfilled all requirements using regular 4140 steel.

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

Image 3.1 Capture from ANSYS showing that the geometry was centered and zeroed at the drive socket (marked by yellow pointer). This boundary condition was fixed, while the wrest of the wrench was allowed to flex.
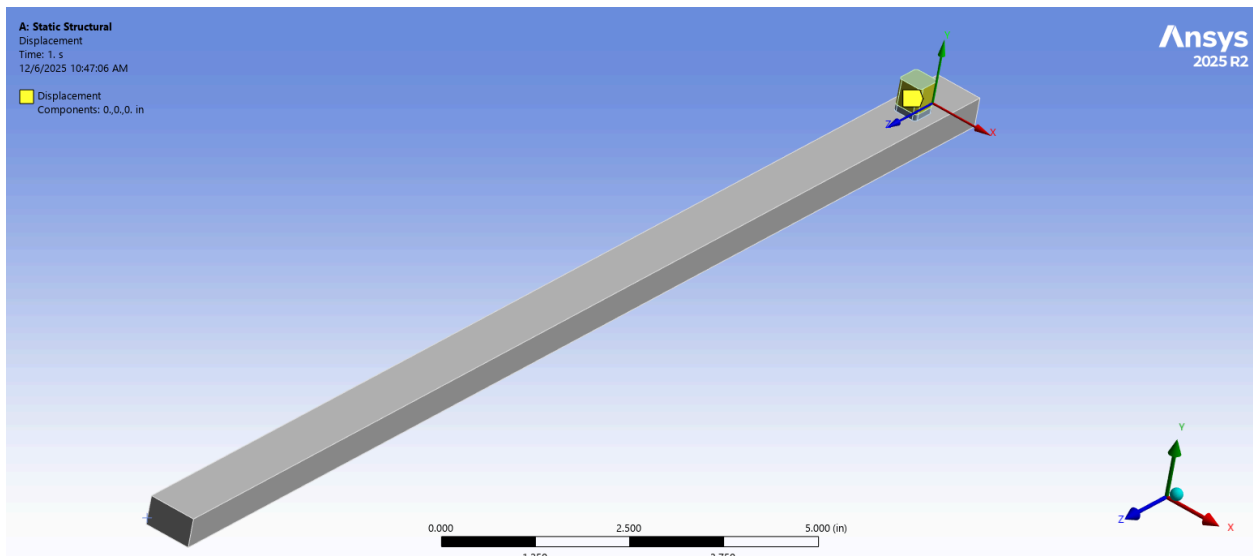

Image 3.2 This image shows that the reaction force of the entire system acts on the socket, in the opposite direction that it is applied to the wrench on the other side. This again shows that the wrench socket is the 0 location of the geometry, which is how we applied boundary conditions to this model.
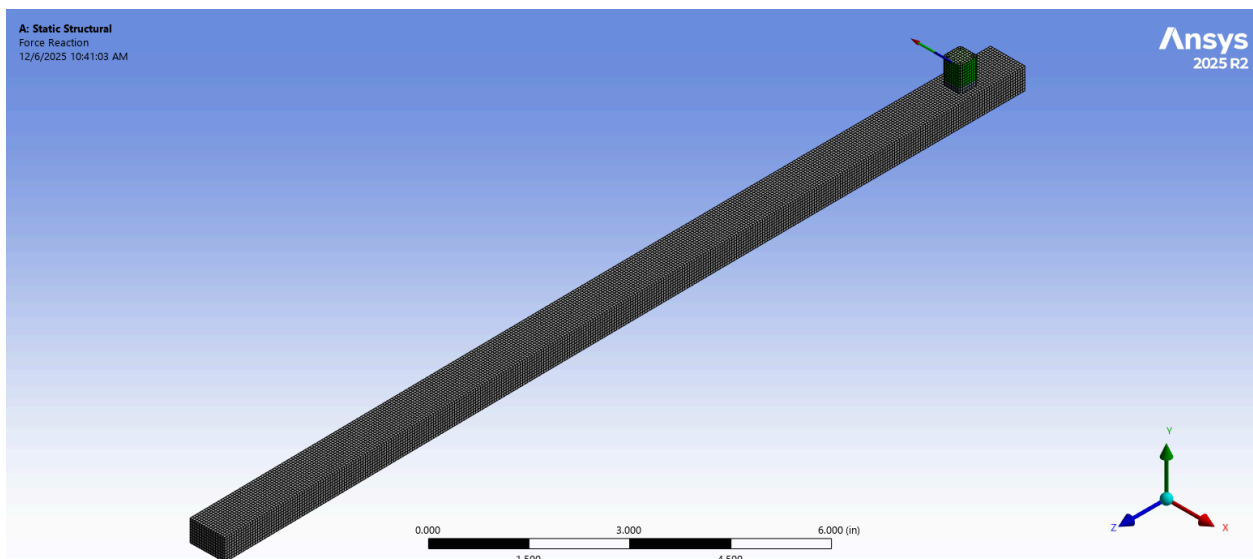
Image 3.3 This image shows the location and direction of the applied loads on the system. To obtain a 600 lb-in torque a 37.5 lb force was applied at the end of the wrench. This is indicated by the red arrow in the bottom left of the image and the magnitude included in the key in the top left.
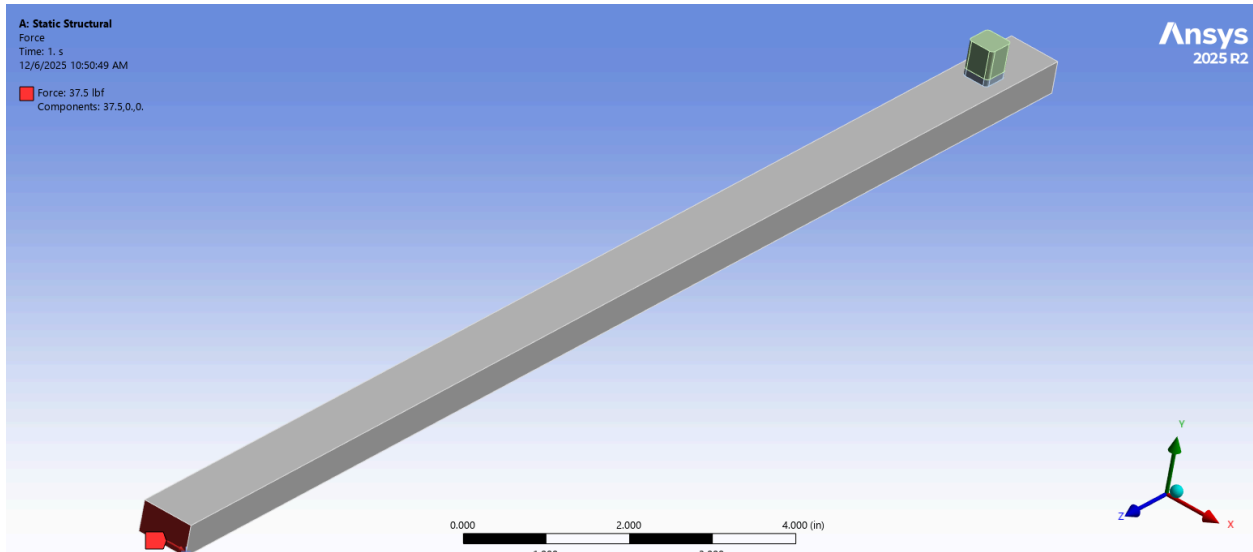
4. Normal strain contours (in the strain gauge direction) from FEM
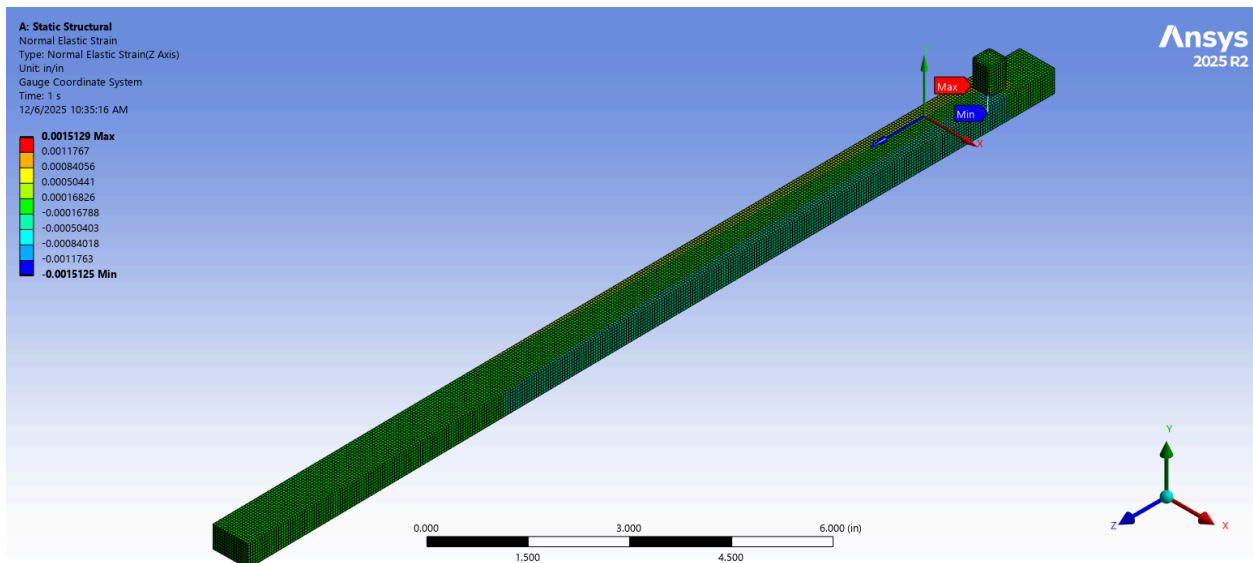
5. Contour plot of maximum principal stress from FEM
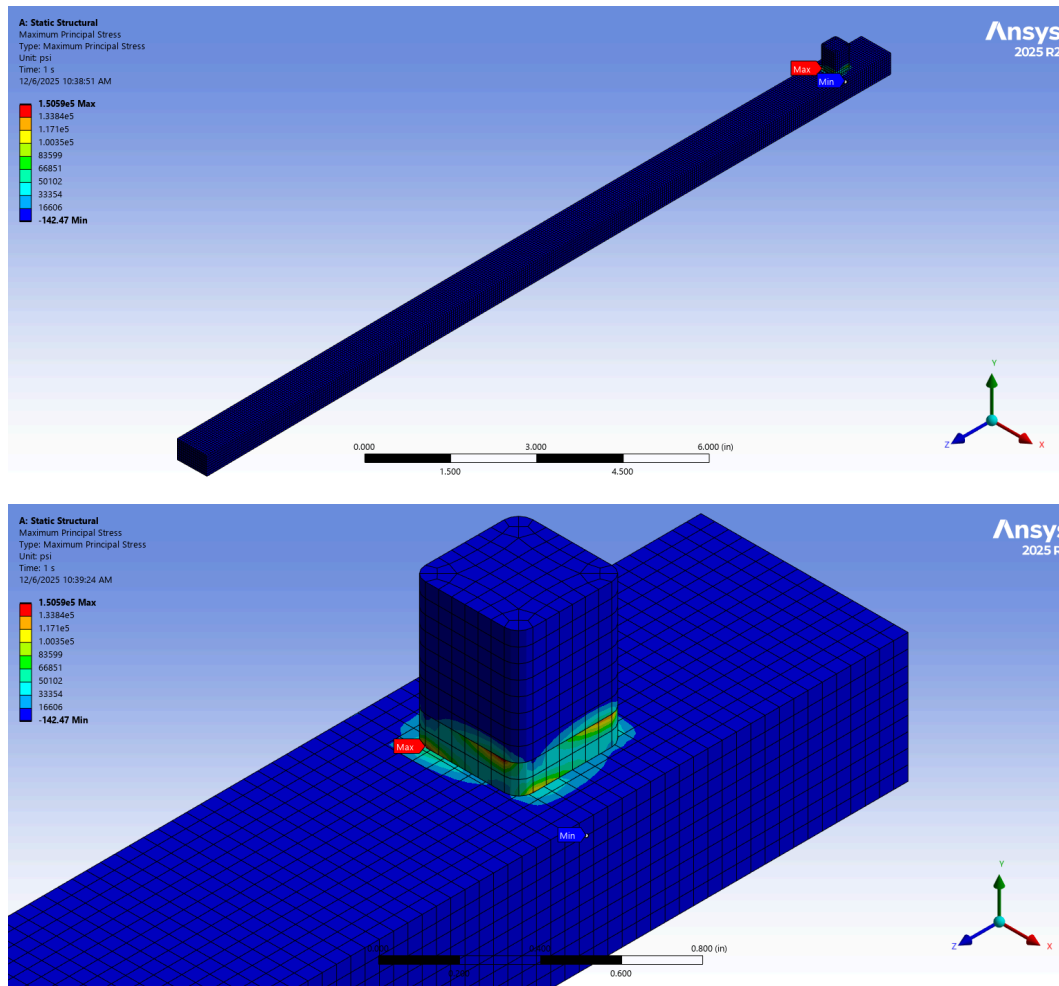
6. Results from FEM calculation showing maximum normal stress (anywhere), load point deflection, strains at the strain gauge locations

Max normal stress = 64959 PSI
Load point deflection = 0.16569 inches
Strain at strain gauge location = 466.7 microstrain
Maximum principal stress: 1.506e5 psi

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

Using the strain at the strain gauge point (1 inch from the drive socket) from the FEM analysis, we get a torque wrench sensitivity of 0.9786 mV/V. This is just barely below the 1 mV/V requirement. The reason for this is that we designed the wrench using beam theory *exactly* to the 1 mV/V requirement. This was so that we wouldn't have to deal with a more complicated geometry. So when the wrench was evaluated using FEM

rather than beam theory, a few variables changed resulting in the strain gauge coming just short of the 1 mV/V requirement.

8. Strain gauge selected

Below is a link to the strain gauges used for the calculation. The important details are that the strain gauge utilizes a full bridge, has a gauge factor of 2.1, and has dimensions of 9.6 x 7.8 mm (0.38 x 0.31 in). This results in a high enough sensitivity to meet the 1 mV/V requirement and is small enough to fit on the chosen geometry.

Appendix:

Link to Strain gauge:

https://www.amazon.com/DKARDU-Strain-BF350-3EB-Precision-Pressure/dp/B0B5G NDK6Z/ref=sr_1_9?dib=eyJ2IjoiMSJ9.7NKNzxH0G_ER9Qn0dg0Lm7ZT_BPvClkKizI VXK5quIGxA7Oh4gqt3QLDxUNkfjd9xSjyQ3_XF275knBfe804MYVB4J0-P1iPQeoiSnp WRIdpvHrFCbBqe9k9NczpXFJ4SHZBGtTWcnlVUvZM1sDV6dejcz8APGg8vZe6q09Er 7-_sYnAnQfvo1ApNPgHA4Xza7dYtpmFb-pGNOpydPuixQqLUgaWAwsCXjxe_YXwb78 .lqxJXLfxamLTbwEZmhLdsJo2wnrL8XYKXg8g5bAEfUE&dib_tag=se&keywords=strai n+gauges&qid=1764690379&sr=8-9

MATLAB code used for beam theory calculations:

```
%Material: AISI 4140 Low Alloy Steel
M = 600; % max torque (in-lbf)
L = 16; % length from drive to where load applied (inches)
h = 0.72; % width (inches)
b = 0.45; % thickness (inches)
A = b*h; %Cross sectional area (inches^2)
c = 0; % distance from center of drive to center of strain gauge (inches)
E = 31E6; % Young's modulus (psi)
v = 0.29; % Poisson's ratio
su = 95E3; % tensile strength use yield or ultimate depending on material
(psi)
KIC = 80E3; % fracture toughness (psi sqrt(in))
sfatigue = 75E3; % fatigue strength from Granta for 10^6 cycles (psi)
I = b * h^3 / 12; %Moment of inertia
y = h/2;  %Distance from center to edge, used in bending stress calculations
GF = 2.1; %Strain gauge factor
%Calculate yield factor of safety due to bending stress
sigma = M*y/I;      %Stress due to bending
YieldFOS = su/sigma      %Factor of safety for yield
if YieldFOS >= 4
   disp('√ Yield FOS met. FOS >= 4');
else
```

```matlab
    disp('✗ Yield FOS not met FOS < 4');
end
disp('Max normal stress (PSI)');
disp(sigma);
%Calculate fracture factor of safety.
a = 0.04;      %Crack length, inches
KI = 1.12 * sigma*sqrt(pi*a);      %Stress
FracFOS = KIC/KI            %FOS for fracture
if FracFOS >= 2
    disp('✓ Fracture FOS met for crack of 0.04 inches. FOS >= 2');
else
    disp('✗ Fracture FOS not met for crack of 0.04 inches. FOS < 2');
end
%Calculate fatigue stress factor of safety.
FatigueFOS = sfatigue/sigma
if FatigueFOS >= 1.5
    disp('✓ Fatigue FOS met. FOS >= 1.5');
else
    disp('✗ Fatigue FOS not met. FOS < 1.5');
end
%Check to see if the 1mv/V requirement is met by strain gauge.
Mc = M * (1 - c / L);      % Moment at the strain gauge location
sigma_gauge = Mc * y / I;  % Bending stress at gauge location
epsilon = sigma_gauge / E; %Strain at the strain gauge
disp('strain at gauge (microstrain) =');
disp(epsilon*1E6);
% WHEATSTONE BRIDGE OUTPUT (Full-bridge))
Vout_over_Vin = (GF) * epsilon   % V/V
required_output = 1e-3;   % 1 mV/V
if Vout_over_Vin >= 0.001
    disp('✓ Sensitivity requirement met: ≥ 1 mV/V');
else
    disp('✗ Sensitivity requirement NOT met.');
end
%This section solves for the deflection when maximum torque is applied.
%Only pure bending is considered here.
deflection = M*L^2 / (2*E*I)     %Deflection of beam tip (treat as cantilever
beam)
disp('Deflection (inches) =');
disp(deflection);
```