Torque Wrench Baseline Design

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1. Results

1.1 MATLAB Scripts for hand calculation

```matlab
M = 600; % max torque (in-lbf)

%%% Material Properties %%%
matl_property = {{32.E6, 0.29, 370.E3, 15.E3, 115.E3, 'M42 Steel'};
                 {16.1E6, 0.35, 148E3, 74.6E3, 90E3, "Ti-6Al-4V, aged"}
                 };
dimensions = {{16, 0.75, 0.5, 1, 'M42 Steel'};
              {16, 0.5, 0.5, 1, "Ti-6Al-4V, aged"}
              };
disp(" ")
disp("REQUIREMENTS: ")
disp("    output>1e-3,     Xo>4,     Xk>2,    Xs>1.5")

for i = 1:2
    % Extract material properties
    E = matl_property{i}{1}; % Young's modulus (psi)
    nu = matl_property{i}{2}; % Poisson's ratio
    sigu = matl_property{i}{3}; % tensile strength use yield or ultimate depending on mate (psi)
    KIC = matl_property{i}{4}; % fracture toughness (psi sqrt(in))
    sfatigue = matl_property{i}{5}; % fatigue strength from Granta for 10^6 cycles
    name = matl_property{i}{6}; % material name

    % Extract beam dimensions
    L = dimensions{i}{1}; % length from drive to where load applied (inches)
    h = dimensions{i}{2}; % width
    b = dimensions{i}{3}; % thickness
    c = dimensions{i}{4}; % distance from center of drive to center of strain gauge

    % Yield/brittle failure
    I = (b*h^3)/12;
    sigmax = M*(h/2)/I; % max normal (psi)
    Xo = sigu/sigmax; % safety factor against brittle failure
    P = M/L; % equivalent load
    umax = (P*L^3)/(3*E*I); % max deflection

    % Fracture failure w crack depth=0.04in
    a = 0.04; % crack depth (in)
    Sg = 6*M/(b*h^2);
    KI = 1.12*Sg*sqrt(pi*a);
    Xk = KIC/KI; % safety factor against fracture

    % Fatigue failure
    Xs = sfatigue/sigmax; % safety factor against fatigue

    % Strain gauge
    Mb = M*(1-c/L);
    sigmaxeps = Mb*(h/2)/I;
    eps = sigmaxeps/E;
    k = 2;
    output = k*eps/2;

    fprintf("TEST of %s \n", name)
    fprintf('    output=%.2e, Xo=%.2f, Xk=%.2f, Xs=%.2f \n', output, Xo, Xk, Xs)
    fprintf('    max normal stress = %.3e, strain at gauge = %.3e, deflection = %.3f \n', sigmax, eps, umax)
end
```

Output:

Command Window

```
REQUIREMENTS:
    output>1e-3,     Xo>4,     Xk>2,    Xs>1.5
TEST of M42 Steel
    output=3.75e-04, Xo=28.91, Xk=2.95, Xs=8.98
    max normal stress = 1.280e+04, strain at gauge = 3.750e-04, deflection = 0.091
TEST of Ti-6Al-4V, aged
    output=1.68e-03, Xo=5.14, Xk=6.52, Xs=3.12
    max normal stress = 2.880e+04, strain at gauge = 1.677e-03, deflection = 0.611
```

Figure 1: Code and Output of MATLAB Calculation

Therefore, from the output of M24 Steel in Figure 1, MATLAB calculation verifies the validity of the code for outputting the same result as the given test case.

Using this code, we also verified the design with dimensions of length L=16in, thickness b=0.5in, width h=0.5in, and using the material of Ti-6Al-4V (aged) to meet all the design criterions.

1.2 Results from hand calculations of base design

The calculated maximum normal stress, strain at the gauge, and maximum deflection are:
max normal stress = 2.700e+04 psi, strain at gauge = 1.677e-03, deflection = 0.611 in

1.3 Results from FEM calculation of base design

The FEM calculation was based on the original model with dimensions of length L=16in, thickness b=0.5in, width h=0.75in, using material M42 Steel.

The hand calculation of the max normal stress, strain at the gauge, and max deflection is outputted as max normal stress = 1.280e+04 psi, strain at gauge = 3.750e-04, and deflection = 0.091 in, as shown in Figure 1.

The FEM results, as shown in Figures 2-4, exhibits a max normal stress of 5.5961e+04 psi, a strain at gauge of 3.7529e-04, and a max deflection of 0.13518 in.

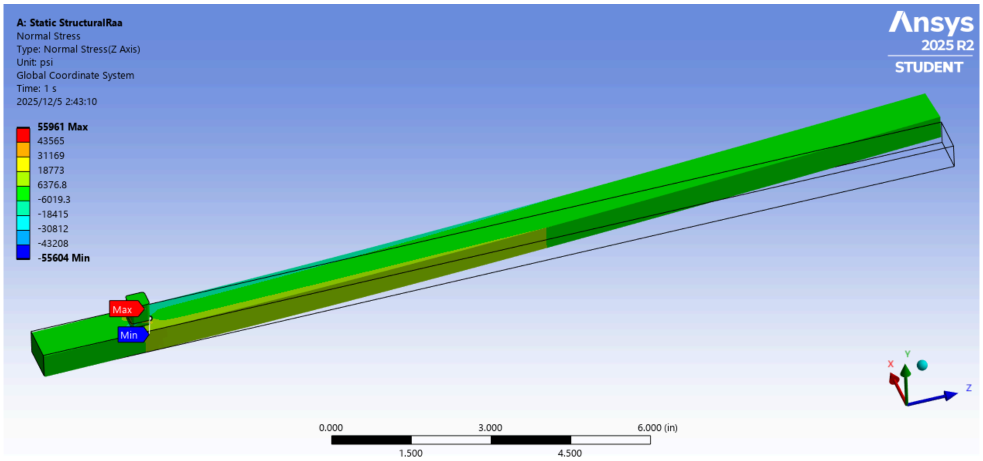

Figure 2: Maximum normal stress occurred in z-axis is 55961psi

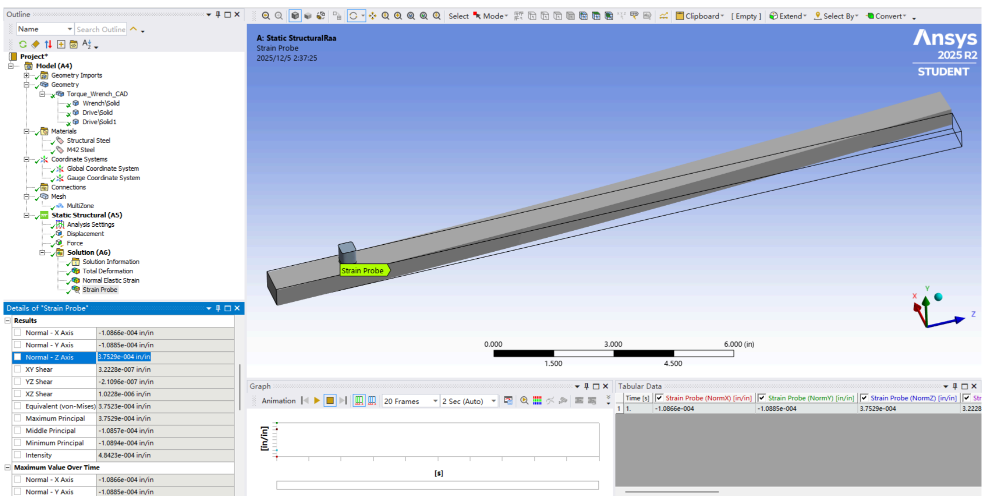

Figure 3: Strain at the gauge location is 375.29με

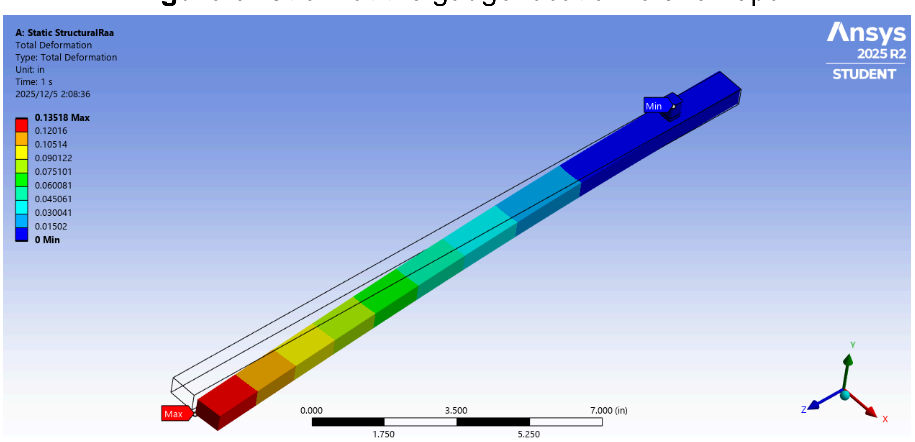

Figure 4: Maximum deflection in the beam is 0.13518in (modeled with fillet)

2. Reflections

2.1. 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?

In the deformed ANSYS model, the mesh lines that cut across the mesh lines seem to stay almost straight and do not show noticeable warping, except very close to the support. It indicates the beam theory is reasonably accurate for this beam.

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

The hand-calculated maximum normal stress is 12.8 ksi, while the FEM result is 53.9ksi, so the FEM stress is roughly four times larger. This difference is expected because the hand calculation uses simple beam theory (assuming a prismatic beam and gives only the nominal bending stress). The FEM model, however, has 3-D geometry, including the transition where the drive meets the handle. The geometric changes create stress concentrations, so the peak stress from FEM is higher than the average stress predicted by beam theory.

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

The hand-scaled tip deflection is 0.091 in, whereas the FEM predicts a larger deflection of 0.13518 in. The difference arises from modeling assumptions. In the hand calculations, the beam is identified as a uniform distribution with simple boundary conditions and pure bending, neglecting shear deformation and local flexibility near the drive. The FEM model includes more detailed information, including the geometry of the handle and drive, the actual way the load and constraints are applied, introducing additional compliance.