BOLTING - EXAMPLE 1

A section of a tension-loaded connection employing a confined gasket is shown by Figure 14-32.a of Norton. This connection has been designed to carry a load which varies between 0 and 12,000 pounds-per-bolt. Determine the factor-of-safety of the connection which has been preloaded to 10,000 pounds. The bolt is a 5/8"-11UNC-Grade 4 bolt, 2.25-inches long with rolled threads.

In working this problem, w*e will assume the thread has a machine finish and will use a stress concentration factor of 3.0 for rolled threads*.

$k_{surface} := 0.73 \quad k_{size} := 0.85 \qquad k_e := \frac{1}{3}$

$S_{ult} := 100000 \quad \text{ksi}$

$S_e := k_{surface} \cdot k_{size} \cdot k_e \cdot \frac{S_{ult}}{2} \qquad S_e = 1.034 \cdot 10^4 \quad \text{ksi}$

The endurance limit has now been determined using techniques from the chapter on fatigue and data given in the problem statement. The next step is determine the proof strength of the bolt, the preload, bolt load, and clamped member loads. Following that, the loads will be checked against the criteria for tensile (positive) total bolt load and compressive (negative) clamped member loads. Finally, the fatigue life will be checked using the factor of safety equation given in your notes.

From the Table 14-6 (page 914 of Norton) we can find the properties for a class 4 bolt:

65ksi	=	*proof strength*
100 ksi	=	*yield strength*
115 ksi	=	*ultimate strength*
0.2260 in^2	=	*stress area*

As a first check, verify the preload stress load is less than the proof stress.

$A_t := 0.2260$ square − inches

$F_i := 10000$ pounds

$\sigma_i := \frac{F_i}{A_t}$ $\sigma_i = 4.425 \cdot 10^4$ psi

This stress is less that the proof stress (65-ksi). We can, therefore, continue with our calculations.

The next step is to evaluate the stiffness of the components, the clamped members and the bolt. For these calculations, note the material of the clamped members is #25 cast iron. We will need to find the stiffness (Young's Modulus) for that material. The clamped length of the joint is 1.5 inches, and the bolt diameter 5/8-inch. Because the modulus of elasticity for steel is different than that for cast iron, we can not make the assumption the member stiffness is eight times that of the bolt.

#25 CI	*E = 12,000,000 psi, member*
Steel	*E = 30,000,000 psi, bolt*
Length	*l = 1.5 inches*
Diameter	*d = 0.625 inch*

$E_b := 30 \cdot 10^6$ psi $E_m := 12 \cdot 10^6$ psi

$d := 0.625$ inch $l := 1.50$ inch

$k_b := \pi \cdot d^2 \cdot \frac{E_b}{4 \cdot l}$ $k_b = 6.136 \cdot 10^6$ $\frac{lb}{in}$

$k_m := 2 \cdot \pi \cdot d^2 \cdot \frac{E_m}{l}$ $k_m = 1.963 \cdot 10^7$ $\frac{lb}{in}$

Next, the minimum (preload), mean, maximum, and alternating stresses are calculated. This is done to check the factor of safety in fatigue as well as compare the maximum stress to the proof stress and ultimate stress of the bolt. As mentioned in class, for now we do not determine and check the clamped member stresses. We need to know more about surface fatigue and compressive stresses in fatigue to do this, which is beyond the scope of our current course.

$P_{min} := 0$ pounds $\qquad P_{max} := 12000$ pounds

$$P_{mean} := \frac{P_{min} + (P_{max})}{2} \qquad P_{alt} := \frac{P_{max} - P_{min}}{2}$$

$$\sigma_{min} := \left[\frac{k_b}{(k_b + k_m)}\right] \cdot \frac{P_{min}}{2 \cdot A_t} + \frac{F_i}{A_t} \qquad \sigma_{min} = 4.425 \cdot 10^4 \text{ psi}$$

$$\sigma_{mean} := \frac{k_b}{k_b + k_m} \cdot \frac{P_{mean}}{A_t} + \frac{F_i}{A_t} \qquad \sigma_{mean} = 5.057 \cdot 10^4 \text{ psi}$$

$$\sigma_{max} := \frac{k_b}{k_b + k_m} \cdot \frac{P_{max}}{A_t} + \frac{F_i}{A_t} \qquad \sigma_{max} = 5.689 \cdot 10^4 \text{ psi}$$

$$\sigma_{alt} := \frac{k_b}{k_b + k_m} \cdot \frac{P_{alt}}{A_t} \qquad \sigma_{alt} = 6.321 \cdot 10^3 \text{ psi}$$

$$F_{bmax} := F_i + P_{max} \cdot \left[\frac{k_b}{(k_b + k_m)}\right] \qquad F_{bmax} = 1.286 \cdot 10^4 \text{ pounds}$$

$$\sigma_{bmax} := \frac{F_{bmax}}{A_t} \qquad \sigma_{bmax} = 5.689 \cdot 10^4 \text{ psi}$$

At this point, we see the maximum bolt stress is less than the allowable proof stress; and, we can continue on with our calculations.
For this case we need to check the minimum external load against the criteria for flange separation. The maximum bolt load is use for strength calculations.

$F_{b,min} > 0$
$F_{m,max} < 0$

$$C := \frac{k_b}{k_b + k_m} \qquad C = 0.238$$

$$F_{bmin} := P_{min} \cdot C + F_i \qquad F_{bmin} = 1 \cdot 10^4 \quad \text{pounds}$$

$$F_{mmax} := P_{max} \cdot (1 - C) - F_i \qquad F_{mmax} = -857.143 \quad \text{pounds}$$

From these calculations, the criteria has been satisfied. There is a positive load in the bolt and a negative (compressive) load in the clamped members.

The next step is to evaluate the factor of safety in fatigue using equation number 14.16 on page 924 of Norton.

$$N_f := S_e \cdot \frac{S_{ult} - \sigma_{min}}{\left[S_e \cdot \left(\sigma_{mean} - \sigma_{min} \right) \right] + S_{ult} \cdot \sigma_{alt}} \qquad N_f = 0.827$$

This factor of safety, which indicates negative margin of safety in fatigue, is unsatisfactory. We have several options to improve the design so we have a positive margin of safety. Among these are:

(1) Use a stronger bolt.

(2) Use a larger bolt.

(3) Use more bolts to decrease the external load/bolt.

(4) Change the clamped member material to a stronger material such as steel.

For this example, we will select option (4), use steel members with a modulus of 30E+06 modulus. This will affect the portion of the external load carried by the members and bolt. We need modify only the equations involving the member stiffness determination; all others will remain the same.

$E_b := 30 \cdot 10^6$ psi

$E_m := 30 \cdot 10^6$ psi

$d := 0.625$ inch $\quad l := 1.50$ inch

$k_m := 2 \cdot \pi \cdot d^2 \cdot \frac{E_m}{l}$ $\quad k_m = 4.909 \cdot 10^7$ $\frac{lb}{in}$

$k_b := \pi \cdot d^2 \cdot \frac{E_b}{4 \cdot l}$ $\quad k_b = 6.136 \cdot 10^6$ $\frac{lb}{in}$

$P_{min} := 0$ pounds $\quad P_{max} := 12000$ pounds

$$P_{mean} := \frac{P_{min} + (P_{max})}{2} \qquad P_{alt} := \frac{P_{max} - P_{min}}{2}$$

$$\sigma_{min} := \left[\frac{k_b}{(k_b + k_m)} \right] \cdot \frac{P_{min}}{2 \cdot A_t} + \frac{F_i}{A_t} \qquad \sigma_{min} = 4.425 \cdot 10^4 \text{ psi}$$

$$\sigma_{mean} := \frac{k_b}{k_b + k_m} \cdot \frac{P_{mean}}{A_t} + \frac{F_i}{A_t} \qquad \sigma_{mean} = 4.72 \cdot 10^4 \quad \text{psi}$$

$$\sigma_{max} := \frac{k_b}{k_b + k_m} \cdot \frac{P_{max}}{A_t} + \frac{F_i}{A_t} \qquad \sigma_{max} = 5.015 \cdot 10^4 \quad \text{psi}$$

$$\sigma_{alt} := \frac{k_b}{k_b + k_m} \cdot \frac{P_{alt}}{A_t} \qquad \sigma_{alt} = 2.95 \cdot 10^3 \quad \text{psi}$$

$$F_{bmax} := F_i + P_{max} \cdot \left[\frac{k_b}{\left(k_b + k_m\right)} \right] \qquad F_{bmax} = 1.133 \cdot 10^4 \quad \text{pounds}$$

$$\sigma_{bmax} := \frac{F_{bmax}}{A_t} \qquad \sigma_{bmax} = 5.015 \cdot 10^4 \quad \text{psi}$$

$$C := \frac{k_b}{k_b + k_m} \qquad C = 0.111$$

$$F_{mmax} := P_{max} \cdot (1 - C) - F_i \qquad F_{mmax} = 666.667 \quad \text{pounds}$$

$$F_{bmin} := P_{min} \cdot C + F_i \qquad F_{bmin} = 1 \cdot 10^4 \quad \text{pounds}$$

This calculation indicates we have a positive load for the clamping force, meaning the joint separates.

The factor of safety will also be checked, just to determine the effect of the material change.

$$N_f := S_e \cdot \frac{S_{ult} - \sigma_{min}}{\left[S_e \cdot \left(\sigma_{mean} - \sigma_{min}\right) \right] + S_{ult} \cdot \sigma_{alt}} \qquad N_f = 1.771$$

This value is acceptable. The design, is, however, unacceptable because the joint separates. Further further calculations, such as increasing the size of the bolts, or increasing the number of bolts are required.

The only remaining step is to run a hand calculation to verify the computer results. Remember, in computer-aide-analysis, we need to verify our program to avoid "GIGO". Which can be a very embarrassing event to incur, especially during a design review.