ME424 Design Project-2

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Purpose

Designing a bolted joint that will have the lowest cost with the given specifications and constraints.

Givens

- Fluid pressure varies between 0 and 14.5 MPa.
- Elastic modulus of the bolt and the cylinder is 207 GPa.
- Poisson's ratio is 0.3 for both members.
- Thickness, outer diameter, length of the cylinder are 6.1, 100, 324 mm respectively.
- Thickness of the end plates are 20 mm.
- Fatigue life is 5 x 10^5 cycles.
- Reliability is 99.9%.
- Confined gaskets are used.
- Price information according to the classes and diameters are provided.

Design Parameters

1. Major diameter
2. Property class
3. Preload factor

Constraints

- Safety factor against all failure modes must be at least 1.6.
- Cost must be minimized.
- Preload factor must be between the specified range: 0.50 – 0.90.

Assumptions and Decisions:

- End plates are rigid.
- Gasket thickness is small so that its effect is neglected.
- Confined gasket is used preventing leakage until $F_c = 0$.
- Thread stripping does not occur.
- Local yielding occurs first in the fatigue failure analysis.

Procedure:

There are 3 design parameters to be decided, thus 3 for loops are needed.

1. Given parameters, standard diameter, pitch and class properties are defined.
2. For loops are introduced.
3. All the geometric variables (diameters and areas) are calculated.
4. Preload is determined.
5. Number of turns are determined by the formula that is found in the part-a.
6. External, bolt and clamped forces (F_e, F_b, F_c) are calculated by the formula introduced in the part-b of the project.
7. Static yielding failure analysis: The pressure that will cause the yielding is found by equating axial stress to the yield strength ($F_b/A_t = S_y$) where F_b is defined as a function of pressure. Dividing this yielding pressure by the maximum pressure applied gives the safety factor against gross yielding. Safety factor check is done.
8. Leakage/separation failure analysis: Since confined gasket is used, leakage and separation failure analyses are the same. The pressure that will cause the separation is found by equating clamped force to zero ($F_b – F_e = 0$) where forces are defined as a function of pressure. Dividing this leakage pressure by the maximum pressure applied gives the safety factor against leakage/separation. Safety factor check is done.
9. Fatigue failure analysis: equivalent alternating and mean stresses are calculated, and fatigue failure analysis procedures are applied. Safety factor check is done.
10. Computed costs of every iteration that passes the checks are compared to find the design with the minimum cost.

Optimum Design Parameters

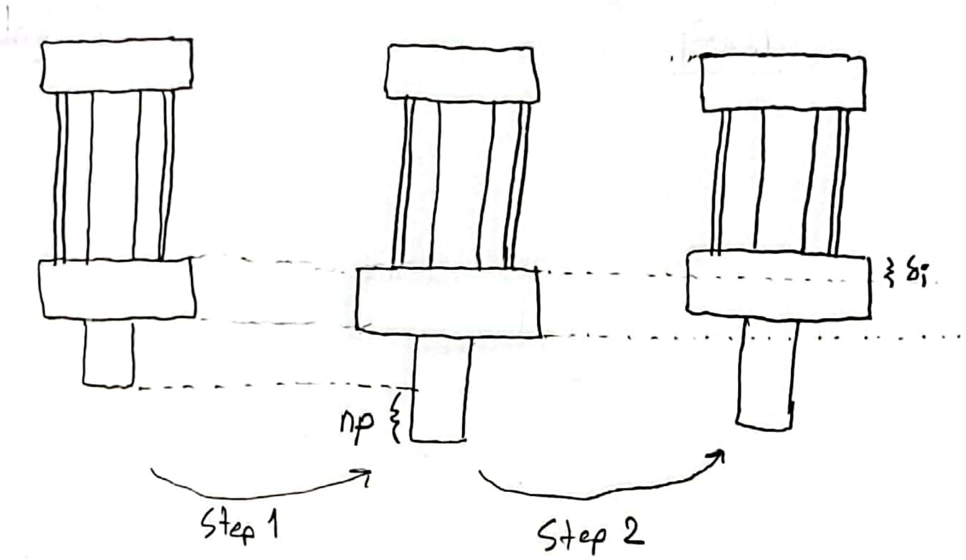
1. Major diameter of the bolt: $d = 33$ mm
2. Property class of the bolt: '4.6'
3. Preload factor: $K_i = 0.739$

Hand Solution

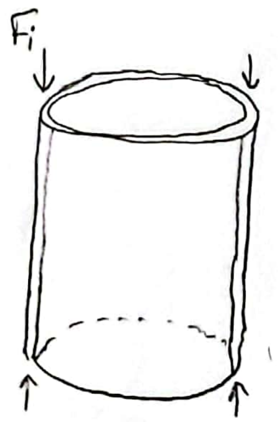
Scanned handwritten solutions to parts a and b are provided after this point. Also, the derivations and calculations of the analysis are described by using optimum design parameters.

(a)

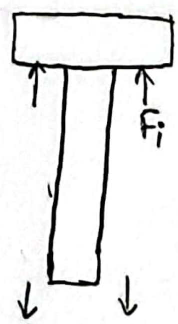
	Step 1	Step 2
Cylinder:	$L \to L$	$L \to L - \delta_i$
Bolt :	$L + 2h - np \to L + 2h$	$L+2h \to L + 2h - \delta_i$



	Initial	Final
Cylinder:	L	$L - \delta_i$
Bolt :	$L+2h-np$	$L+2h-\delta_i$



$$\delta_i = \frac{F_i L}{E_c A_c} \quad (1)$$



$$np - \delta_i = \frac{F_i (L+2h-np)}{E_b A_b} \quad (2)$$

By equation (1) and (2), we have

$$\Rightarrow \quad np - \frac{F_i L}{E_c A_c} = \frac{F_i (L+2h-np)}{E_b A_b}$$

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$$\Rightarrow \quad n p + \frac{F_i}{E_b A_b} n p - \frac{F_i L}{E_c A_c} = \frac{F_i (L+2h)}{E_b A_b} \quad \text{we know } E_b = E_a = E$$

$$\Rightarrow \quad n p = \frac{F_i}{E} \left(\frac{L}{A_c} + \frac{L+2h}{A_b} \right) \cdot \frac{1}{1 + \frac{F_i}{E A_b}}$$

$$\Rightarrow \quad \boxed{n = \frac{F_i}{p E} \left(\frac{L}{A_c} + \frac{L+2h}{A_b} \right) \cdot \frac{1}{1 + \frac{F_i}{E A_b}}}$$

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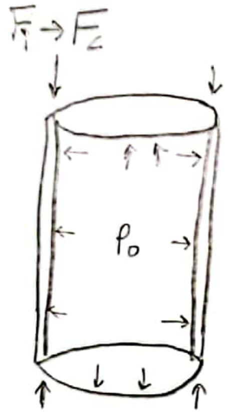
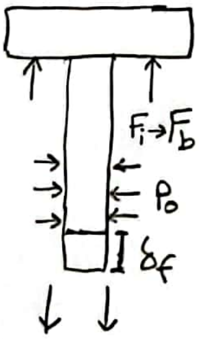
(b)



$$\delta_f = \frac{(F_i - F_c)(L - \delta_i)}{E A_c} - \frac{\nu \sigma_\theta}{E} L \quad (1)$$

where $\sigma_\theta = \frac{pr}{t} = \frac{p_o(D - 2t)}{2t}$

Unknowns: F_c, δ_f



$$\delta_f = \frac{(F_b - F_i)(L + 2h - \delta_i)}{E A_b} + \frac{\nu p_o}{E} L \quad (2)$$

where $F_b = F_c + F_e$ where $F_e = p_o A_s$

Equating (1) and (2), then we have

$$\frac{(F_b - F_i)(L + 2h - \delta_i)}{E A_b} + \frac{\nu p_o}{E} L = \frac{(F_i + F_e - F_b)(L - \delta_i)}{E A_c} - \frac{\nu p_o r}{E t} L$$

$\Rightarrow$

$$F_b \left[\frac{(L + 2h - \delta_i)}{E A_b} + \frac{L - \delta_i}{E A_c} \right] = \frac{F_i (L + 2h - \delta_i)}{E A_b} + \frac{(F_i + F_e)(L - \delta_i)}{E A_c} - \frac{\nu p_o}{E}\left(1 + \frac{r}{t}\right) L$$

$\Rightarrow$

$$F_b = \frac{\frac{F_i (L + 2h - \delta_i)}{A_b} + \frac{(F_i + F_e)(L - \delta_i)}{A_c} - \nu p_o \left(1 + \frac{r}{t}\right) L}{\left[\frac{(L + 2h - \delta_i)}{A_b} + \frac{L - \delta_i}{A_c} \right]}$$

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We can also conclude that

$$F_b = F_c + F_e$$

so

$$F_c = F_b - F_e \quad \text{where} \quad F_e = \rho_0 A_s$$

$$\Rightarrow F_c = F_b - \rho_0 A_s$$

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$a = 7,\ b = 5$

Given: Cylinder outer diameter: $D = 100mm$

Maximum pressure: $P_{max} = 14.5\ MPa$

Minimum pressure: $P_{min} = 0\ MPa$

Modulus of Elasticity: $E = 207\ GPa$

Length of rigid end plate: $h = 20\ mm$

Length of cylinder: $L = 324\ mm$

Length of bolt: $L_{bolt} = 364\ mm$

Poisson ratio: $V = 0.3$

Cylinder thickness: $t = 6.1\ mm$

Life requirement: $Life = 5 \times 10^5$

Design Choices

Material class	Proof strength	Yield strength	Ultimate strength	Major diameter
4.6	$S_p = 225\ MPa$	$S_y = 420\ MPa$	$S_u = 520\ MPa$	33mm

Preload factor

$K_i = 0.739$

Solution

Pitch: $P = 3.5mm$ standard value for $d = 33\ mm$

Pitch diameter: $d_p = d_{bolt} - 3\sqrt{3}p/8$ $\quad d_p = 30.73\ mm$

Root diameter: $d_r = d_{bolt} - 17\sqrt{3}p/24$ $\quad d_r = 28.71\ mm$

Tensile stress area: $A_t = (\pi/4)\left[(d_p + d_r)/2\right]^2$ $\quad A_t = 693.6\ mm^2$

Calculate preload: $F_i = K_i . S_p . A_t$ $\quad F_i = 135608\ N$

Bolt shank area: $A_b = \frac{1}{4} . \pi . d_{bolt}^2$ $\quad A_b = 855.3\ mm^2$

Clamped area: $A_c = \frac{1}{4}\pi(D^2 - (D-2t)^2)$ $\quad A_c = 1799.5\ mm^2$

Surface area: $A_s = \frac{1}{4}\pi((D-2t)^2 - d^2)$ $\quad A_s = 5199.2\ mm^2$

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Cylinder inner radius: $r_c = (D-2t)/2$ $\quad r_c = 43.9\,mm$

External seperating force: $F_e = p_{max}\,A_s$ $\quad F_e = 7.30\times10^4\,N$

Maximum tension bolt: F_b (Using equation in Part (b)) $\quad F_b = 7.159\times10^5\,N$

Minimum compression in the clamped member: $F_c = F_b - F_e$ $\quad F_c = 6.429\times10^5\,N$

Uncorrected endurance limit: $S_n' = 0.5\,S_u$ $\quad S_n' = 200\,MPa$

Load factor, multiaxial loading: $C_{load} = 1$

Temperature factor: $C_{temp} = 1$

Size factor, $8mm < d \leq 250mm$: $C_{size} = 1.189 \times d_{bolt}^{(-0.097)}$ $\quad C_{size} = 0.847$

Surface factor, included in K_f: $C_{surf} = 1$

Reliability factor: $C_{rel} = 0.753$ for $Rel = 99$

Endurance limit: $S_n = C_{load} \cdot C_{size} \cdot C_{surf} \cdot C_{temp} \cdot C_{rel} \cdot S_n'$ $\quad S_n = 127.56\,MPa$

Fatigue strength for 1000 cycles: $S_{f10^3} = 0.75\,S_u$ $\quad S_{f10^3} = 300\,MPa$

Fatigue strength for 10^5 cycles:

$$\frac{6-3}{\log(S_{f10^3}) - \log(S_n)} = \frac{6 - \log(5\times10^5)}{\log(S_f) - \log(S_n)}$$

$$S_f = \exp\left(\frac{2}{3}\cdot\ln(S_n) + \frac{1}{3}\ln\left(S_{f10^3}\right)\right) \qquad S_f = 138.99\,MPa$$

Approximate value for the Brinell hardness: $HB = \dfrac{S_u}{3.45}$ $\quad HB = 115.9$

Fatigue stress concentration factor

$$K_f = \begin{cases} 2.2 & \text{if } HB < 200 \\ 3 & \text{if } HB > 200 \end{cases}$$

for rolled thread class <=4.6

$K_f = 2.2$

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Alternating force in the bolt: $F_a = \frac{F_b - F_i}{2}$ $\quad F_a = 290146 N$

Mean force in the bolt: $F_m = \frac{F_b + F_i}{2}$ $\quad F_m = 425754 N$

Initial stress in the bolt for fatigue: $\sigma_{bif} = K_f \frac{F_i}{A_t}$ $\quad \sigma_{bif} = 430.13 MPa$

Initial compressive stress in the clamped member for fatigue: $\sigma_{cif} = K_f \frac{F_i}{A_c}$ $\quad \sigma_{cif} = 165.79 MPa$

Tensile stress in the bolt after pressure applied: $\sigma_{bff} = K_f \frac{F_b}{A_t}$ $\quad \sigma_{bff} = 376.37 MPa$

Compressive stress in the clamped member after pressure applied: $\sigma_{cff} = K_f \frac{F_c}{A_c}$ $\quad \sigma_{cff} = 52.89 MPa$

Axial alternating stress: $\sigma_{ax} = \frac{\sigma_{bff} - \sigma_{bif}}{2}$ $\quad \sigma_{ax} = 5.28 MPa$

Radial alternating stress: $\sigma_{ay} = K_f \frac{P_{max}}{2}$ $\quad \sigma_{ay} = 15.95 MPa$

Alternating equivalent stress: $\sigma_{ea} = \sqrt{\sigma_{ax}^2 + \sigma_{ay}^2 - \sigma_{ax}\sigma_{ay}}$ $\quad \sigma_{ea} = 14.07 MPa$

Axial mean stress: $\sigma_{mx} = \frac{\sigma_{bi} + \sigma_{bf}}{2}$ $\quad \sigma_{mx} = 168.68 MPa$

Radial mean stress: $\sigma_{my} = \frac{P_{max}}{2}$ $\quad \sigma_{my} = 7.25 MPa$

Mean equivalent stress: $\sigma_{em} = \frac{\sigma_{mx} + \sigma_{my}}{2} + \left|\frac{\sigma_{mx} - \sigma_{my}}{2}\right|$ $\quad \sigma_{em} = 168.68 MPa$

We assume local yielding occurs then

$$S_a = \frac{S_f(S_u - S_y)}{S_u - S_f} \qquad S_a = 160.72 MPa$$

Safety factor against fatigue: $SF_{fatigue} = \frac{S_a}{\sigma_{ea}}$ $\quad \boxed{SF_{fatigue} = 11.42 \geq 1.6}$ ✓

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Leakage occurs when $F_c > 0$ for confined gaskets

We look for critical pressure for $F_c = 0$

We use equation in part (b) for F_c with $p_o = p_{sep}$ which is

$$F_c = 0 = \frac{\frac{F_i}{A_b}\left(L + 2h - \frac{F_i L}{EA_c}\right) + \frac{(F_i + F_e)}{A_c}\left(L - \frac{F_i L}{EA_c}\right) - \nu p_{sep}\left(1 + \frac{r}{t}\right)L}{\left[\frac{L + 2h - \frac{F_i L}{EA_c}}{A_b} + \frac{L - \frac{F_i L}{EA_c}}{A_c}\right]} - p_{sep} A_s$$

$\Rightarrow p_{sep} = 23.206$

Safety factor against leakage: $SF_{leak} = \frac{p_{sep}}{p_{max}}$ $\boxed{SF_{leak} = 1.6004 \geq 1.6}$ ✓

Static Yielding Failure occurs when $\frac{F_b}{A_t} = S_y$

We look for critical pressure

We use equation in part(b) for F_b with $p_o = p_{yield}$

$$\frac{\frac{F_i}{A_b}\left(L + 2h - \frac{F_i L}{EA_c}\right) + \frac{(F_i + F_e)}{A_c}\left(L - \frac{F_i L}{EA_c}\right) - \nu p_{yield}\left(1 + \frac{r}{t}\right)L}{\left[\frac{L + 2h - \frac{F_i L}{EA_c}}{A_b} + \frac{L - \frac{F_i L}{EA_c}}{A_c}\right]} = S_y$$

$\Rightarrow p_{yield} = 222.53$ MPa

Safety factor against yielding: $SF_{yield} = \frac{p_{yield}}{p_{max}}$ $\boxed{SF_{yield} = 15.35 \geq 1.6}$ ✓

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