32.4 Which of the following are the common ways in which threaded fasteners fail during tightening (two best answers): (a) excessive compressive stresses on the head of the fastener due to force applied by the tightening tool, (b) excessive compressive stresses on the shank of the fastener, (c) excessive shear stresses on the shank of the fastener, (d) excessive tensile stresses on the head of the fastener due to force applied by the tightening tool, (e) excessive tensile stresses on the shank of the fastener, and (f) stripping of the internal or external threads?

Answer. (e) and (f).

32.5 The difference between a shrink fit and an expansion fit is that in a shrink fit the internal part is cooled to a sufficiently low temperature to reduce its size for assembly, whereas in an expansion fit, the external part is heated sufficiently to increase its size for assembly: (a) true or (b) false?

Answer. (b). In a shrink fit the external part is heated and then cooled to shrink it onto the internal part. In an expansion fit, the internal part is cooled to contract it for assembly; it then expands to form the interference fit.

32.6 Advantages of snap fit assembly include which of the following (three best answers): (a) components can be designed with features to facilitate part mating, (b) ease of disassembly, (c) no heat affected zone, (d) no special tools are required, (e) parts can be assembled quickly, and (f) stronger joint than with most other assembly methods?

Answer. (a), (d), and (e).

32.7 The difference between industrial stitching and stapling is that the U-shaped fasteners are formed during the stitching process while in stapling the fasteners are preformed: (a) true or (b) false?

Answer. (a).

32.8 From the standpoint of assembly cost, it is more desirable to use many small threaded fasteners rather than few large ones in order to distribute the stresses more uniformly: (a) true or (b) false?

Answer. (b). From the standpoint of assembly cost, it is more desirable to use few large threaded fasteners rather than many small ones because the large fasteners are easier to handle and since there are fewer of them, they require less assembly time.

32.9 Which of the following are considered good product design rules for automated assembly (two best answers): (a) design the assembly with the fewest number of components possible, (b) design the product using bolts and nuts to allow for disassembly, (c) design with many different fastener types to maximize design flexibility, (d) design parts with asymmetric features to mate with other parts having corresponding (but reverse) features, and (e) limit the required directions of access when adding components to a base part?

Answer. (a) and (e). All of the other answers go against design-for-assembly principles.

Problems

Threaded Fasteners

32.1 A 5-mm-diameter bolt is to be tightened to produce a preload = 250 N. If the torque coefficient = 0.23, determine the torque that should be applied.

Solution: $T = C_t DF = 0.23(5.0)(250) =$ **287.5 N-mm = 0.2875 N-m**

32.2 A 3/8-24 UNF nut and bolt (3/8 in nominal diameter, 24 threads/in) are inserted through a hole in two stacked steel plates. They are tightened so the plates are clamped together with a force of 1000 lb. The torque coefficient is 0.20. (a) What is the torque required to tighten them? (b) What is the resulting stress in the bolt?

Solution (a) $T = C_tDF = 0.20(3/8)(1000) =$ **75 in-lb**

(b) $A_s = 0.25\pi(D - 0.9743/n)^2 = 0.25\pi(3/8 - 0.9743/24)^2 = 0.0878$ in^2
$\sigma = F/A_s = 1000/0.0878 =$ **11,386 lb/in^2**

32.3 An alloy steel Metric 10x1.5 screw (10 mm diameter, pitch $p = 1.5$ mm) is to be turned into a threaded hole and tightened to one/half of its proof strength. According to Table 32.2, the proof strength = 830 MPa. Determine the maximum torque that should be used if the torque coefficient = 0.18.

Solution: $A_s = 0.25\pi(10 - 0.9382 \text{ x } 1.5)^2 = 57.99$ mm^2
$\sigma = 0.5$ of 830 MPa = 415 MPa = 415 N/mm^2
$F = \sigma A_s = 415(57.99) = 24{,}066$ N
$T = C_tDF = 0.18(10)(24{,}066) =$ **43,319 N-mm = 43.32 N-m**

32.4 A Metric 16x2 bolt (16 mm diameter, 2 mm pitch) is subjected to a torque of 15 N-m during tightening. If the torque coefficient is 0.24, determine the tensile stress on the bolt.

Solution: $T = 15$ N-m = 15,000 N-mm
$F = T/ C_tD = 15{,}000/(0.24 \text{ x } 16) = 3906$ N
$A_s = 0.25\pi(16 - 0.9382 \text{ x } 2)^2 = 156.7$ mm^2
$\sigma = 3906/156.7 =$ **24.9 N/mm^2 = 24.9 MPa**

32.5 A 1/2-13 screw is to be preloaded to a tension force = 1000 lb. Torque coefficient = 0.22. Determine the torque that should be used to tighten the bolt.

Solution: $T = C_tDF = 0.22(0.50)(1000) =$ **110 in-lb**

32.6 Threaded metric fasteners are available in several systems, two of which are coarse and fine (Table 32.1). Finer threads are not cut as deep and as a result have a larger tensile stress area for the same nominal diameter. (a) Determine the maximum preload that can be safely achieved for coarse pitch and fine pitch threads for a 12 mm bolt. (b) Determine the percent increase in preload of fine threads compared to course threads. Coarse pitch = 1.75 mm and fine pitch = 1.25 mm. Assume the proof strength for both bolts is 600 MPa.

Solution: (a) For standard thread,
$A_s = 0.25\pi(D - 0.9382p)^2 = 0.25\pi(12 - 0.9382 \text{ x } 1.75)^2 = 84.3$ mm^2
$F = A_s\sigma = 84.3(600) =$ **50,560 N**
For fine thread,
$A_s = 0.25\pi(D - 0.9382p)^2 = 0.25\pi(12 - 0.9382 \text{ x } 1.25)^2 = 92.1$ mm^2
$F = A_s\sigma = 92.1(600) =$ **55,243 N**

(b) Percent increase = (55,243 – 50,560)/50,560 = 0.0926 = **9.26 % increase**

32.7 A torque wrench is used on a 3/4-10 UNC bolt in an automobile final assembly plant. A torque of 70 ft-lb is generated by the wrench. If the torque coefficient = 0.17, determine the tensile stress in the bolt.

Solution: $T = 70$ ft-lb = 840 in-lb
$F = T/C_tD = 840/(0.17 \text{ x } 0.75) = 6588$ lb.
$A_s = 0.25\pi(0.75 - 0.9743/10)^2 = 0.334$ in^2
$\sigma = 6588/0.334 =$ **19,697 lb/in^2**

32.8 The designer has specified that a 3/8-16 UNC low-carbon bolt (3/8 in nominal diameter, 16 threads/in) in a certain application should be stressed to its proof stress of 33,000 lb/in^2 (see Table 32.2). Determine the maximum torque that should be used if $C = 0.25$.

Solution: $A_s = 0.25\pi(0.375 - 0.9743/16)^2 = 0.0775$ in^2

$F = \sigma A_s = 33{,}000(0.0775) = 2557.5$ lb
$T = C_t DF = 0.25(0.375)(2557.5) =$ **240 in-lb**

32.9 A 300-mm-long wrench is used to tighten a Metric 20x2.5 bolt. The proof strength of the bolt for the particular alloy is 380 MPa. The torque coefficient is 0.21. Determine the maximum force that can be applied to the end of the wrench so that the bolt does not permanently deform.

Solution: $A_s = 0.25\pi(D - 0.9382p)^2 = 0.25\pi(20 - 0.9382*2.50)^2 = 244.8$ mm^2
Preload force, $F = A_s\sigma = 244.8(380) = 93{,}022$ N
$T = C_t DF = 0.21(20)(93{,}022) = 390{,}690$ N-mm
$T = F_{wrench}L_{wrench}$; $F_{wrench} = T/L_{wrench} = 390{,}690/300 =$ **1302 N**

32.10 A 1-8 UNC low carbon steel bolt (diameter = 1.0 in, 8 threads/in) is currently planned for a certain application. It is to be preloaded to 75% of its proof strength, which is 33,000 lb/in^2 (Table 32.2). However, this bolt is too large for the size of the components involved, and a higher strength but smaller bolt would be preferable. Determine (a) the smallest nominal size of an alloy steel bolt (proof strength = 120,000 lb/in^2) that could be used to achieve the same preload from the following standard UNC sizes used by the company: 1/4-20, 5/16-18, 3/8-16, 1/2-13, 5/8-11, or 3/4-10; and (b) compare the torque required to obtain the preload for the original 1-in bolt and the alloy steel bolt selected in part (a) if the torque coefficient in both cases = 0.20.

Solution: (a) $A_s = 0.25\pi(1.0 - 0.9743/8)^2 = 0.6057$ in^2
$F = \sigma A_s = 0.75(33{,}000)(0.6057) = 14{,}992$ lb.
For the alloy bolt, $\sigma = 120{,}000$ lb/in^2.
$A_s = F/\sigma = 14992/(0.75 \times 120{,}000) = 0.1665$ in^2
$A_s = 0.1665$ in$^2 = 0.25\pi(D - 0.9743/n)^2$
$(D - 0.9743/n)^2 = 0.1665$ in$^2/0.25\pi = 0.212$ in^2
$(D - 0.9743/n) = 0.4605$ in
Possible bolt sizes are: (1) 1/4-20, (2) 5/16-18, (3) 3/8-16, (4) 1/2-13, (5) 5/8-11, (6) 3/4-10
Try (1): $(D - 0.9743/n) = (0.25 - 0.9743/20) = 0.2013$ in. Obviously, none of the D values below 0.4605 will be sufficient.
Try (4): $(D - 0.9743/n) = (0.500 - 0.9743/13) = 0.425$ in < 0.4605 in. Cannot use ½-13 bolt
Try (5): $(D - 0.9743/n) = (0.625 - 0.9743/11) = 0.5364$ in > 0.4605 in. **Use 5/8-11 bolt**

(b) For the original 1-8 bolt, $T = C_t DF = 0.2(1.0)(14{,}992) =$ **2,998 in-lb.**
For the 5/8-11 bolt, $T = C_t DF = 0.2(0.625)(14{,}992) =$ **1,874 in-lb**.

Interference Fits

32.11 A dowel pin made of steel (elastic modulus = 209,000 MPa) is to be press fitted into a steel collar. The pin has a nominal diameter of 16.0 mm, and the collar has an outside diameter of 27.0 mm. (a) Compute the radial pressure and the maximum effective stress if the interference between the shaft OD and the collar ID is 0.03 mm. (b) Determine the effect of increasing the outside diameter of the collar to 39.0 mm on the radial pressure and the maximum effective stress.

Solution: (a) $p_f = Ei(D_c^2 - D_p^2)/D_pD_c^2 = 209{,}000(0.03)(27^2 - 16^2)/(16 \times 27^2) =$ **254 MPa**
Max $\sigma_e = 2p_fD_c^2/(D_c^2 - D_p^2) = 2(254.3)(27^2)/(27^2 - 16^2) =$ **784 MPa**

(b) When $D_c = 39$ mm, $p_f = 209{,}000(0.03)(39^2 - 16^2)/(16 \times 39^2) =$ **326 MPa**
Max $\sigma_e = 2(325.9)(39^2)/(39^2 - 16^2) =$ **784 MPa**

32.12 A pin made of alloy steel is press-fitted into a hole machined in the base of a large machine. The hole has a diameter of 2.497 in. The pin has a diameter of 2.500 in. The base of the machine is 4 ft x 8 ft. The base and pin have a modulus of elasticity of 30 x 10^6 lb/in^2, a yield strength of 85,000 lb/in^2, and a tensile strength of 120,000 lb/in^2. Determine (a) the radial pressure between the pin and the base and (b) the maximum effective stress in the interface.

Solution: (a) $i = 2.500 - 2.497 = 0.003$ in
$p_f = Ei/D_p = 30\text{x}10^6\ (0.003)/2.5 =$ **36,000 lb/in²**

(b) Max $\sigma_e = 2p_f = 2(36{,}000) =$ **72,000 lb/in²**

32.13 A gear made of aluminum (modulus of elasticity = 69,000 MPa) is press fitted onto an aluminum shaft. The gear has a diameter of 55 mm at the base of its teeth. The nominal internal diameter of the gear = 30 mm and the interference = 0.10 mm. Compute: (a) the radial pressure between the shaft and the gear, and (b) the maximum effective stress in the gear at its inside diameter.

Solution: (a) $p_f = Ei(D_c^2 - D_p^2)/D_pD_c^2 = 69{,}000(0.10)(55^2 - 30^2)/(30 \text{ x } 55^2) =$ **161.5 MPa**

(b) Max $\sigma_e = 2p_fD_c^2/(D_c^2 - D_p^2) = 2(161.5)(55^2)/\ (55^2 - 30^2) =$ **460 MPa**

32.14 A steel collar is press fitted onto a steel shaft. The modulus of elasticity of steel is $30 \text{ x } 10^6$ lb/in². The collar has an internal diameter of 2.498 in and the shaft has an outside diameter = 2.500 in. The outside diameter of the collar is 4.000 in. Determine the radial (interference) pressure on the assembly, and (b) the maximum effective stress in the collar at its inside diameter.

Solution: (a) $i = 2.500 - 2.498 = 0.002$ in
$p_f = Ei(D_c^2 - D_p^2)/D_pD_c^2 = 30 \text{ x } 10^6(0.002)(4.000^2 - 2.500^2)/(2.500 \text{ x } 4.000^2) =$ **14,625 lb/in²**

(b) Max $\sigma_e = 2p_fD_c^2/(D_c^2 - D_p^2) = 2(14{,}625)(4.000^2)/\ (4.000^2 - 2.500^2) =$ **48,000 lb/in²**

32.15 The yield strength of a certain metal = 50,000 lb/in² and its modulus of elasticity = $22 \text{ x } 10^6$ lb/in². It is to be used for the outer ring of a press-fit assembly with a mating shaft made of the same metal. The nominal inside diameter of the ring is 1.000 in and its outside diameter = 2.500 in. Using a safety factor = 2.0, determine the maximum interference that should be used with this assembly.

Solution: Max $\sigma_e \leq Y/SF$, use Max $\sigma_e = Y/SF = 50{,}000/2.0 = 25{,}000$ lb/in² Eq. (32.9)
Max $\sigma_e = 2p_fD_c^2/(D_c^2 - D_p^2) = 25{,}000$ lb/in² Eq. (32.6)
Rearranging, $p_f = \sigma_e(D_c^2 - D_p^2)/2D_c^2 = 25{,}000(2.5^2 - 1.0^2)/(2 \text{ x } 2.5^2) = 10{,}500$ lb/in²
$p_f = Ei(D_c^2 - D_p^2)/D_pD_c^2$ Eq. (32.5)
Rearranging, $i = p_f\, D_pD_c^2/E(D_c^2 - D_p^2)$
$i = 10{,}500(1.0)(2.5^2)/(22 \text{ x } 10^6\ (2.5^2 - 1.0^2)) =$ **0.00057 in**

32.16 A shaft made of aluminum is 40.0 mm in diameter at room temperature (21°C). Its coefficient of thermal expansion = $24.8 \text{ x } 10^{-6}$ mm/mm per °C. If it must be reduced in size by 0.20 mm in order to be expansion fitted into a hole, determine the temperature to which the shaft must be cooled.

Solution: $(D_2 - D_1) = -0.20 = 24.8 \text{ x } 10^{-6}(40)(T_2 - 21)$
$T_2 - 21 = -0.20/(24.8 \text{ x } 10^{-6} \text{ x } 40) = -201.6$
$T_2 = -201.6 + 21 =$ **-180.6°C**

32.17 A steel ring has an inside diameter = 30 mm and an outside diameter = 50 mm at room temperature (21°C). If the coefficient of thermal expansion of steel = $12.1 \text{ x } 10^{-6}$ mm/mm per °C, determine the inside diameter of the ring when heated to 500°C.

Solution: $D_2 - D_1 = D_2 - 30 = 12.1 \text{ x } 10^{-6}(30)(500 - 21)$
$D_2 = 30 + 0.174 =$ **30.174 mm.**

32.18 A steel collar is to be heated from room temperature (70°F) to 700°F. Its inside diameter = 1.000 in, and its outside diameter = 1.625 in. If the coefficient of thermal expansion of the steel is = $6.7 \text{ x } 10^{-6}$ in/in per °F, determine the increase in the inside diameter of the collar.

Solution: $(D_2 - D_1) = \alpha D_1(T_2 - T_1) = 6.7 \text{ x } 10^{-6}(1.0)(700 - 70) = 4221 \text{ x } 10^{-6} =$ **0.0042 in**

32.19 A bearing for the output shaft of a 200 hp motor is to be heated to expand it enough to press on the shaft. At 70°F the bearing has an inside diameter of 4.000 in and an outside diameter of 7.000 in.

The shaft has an outside diameter of 4.004 in. The modulus of elasticity for the shaft and bearing is 30×10^6 lb/in^2 and the coefficient of thermal expansion is 6.7×10^{-6} in/in per °F. (a) At what temperature will the bearing have 0.005 of clearance to fit over the shaft? (b) After it is assembled and cooled, what is the radial pressure between the bearing and shaft? (c) Determine the maximum effective stress in the bearing.

Solution: (a) interference $i = 0.004$ in, additional required clearance = 0.005 in
Total expansion = 0.004+0.005 = 0.009 in = $(D_2 - D_1)$
$T_2 = (D_2 - D_1)/\alpha D_1 + T_1 = 0.009/(6.7\text{x}10^{-6} \text{ x } 4.000) + 70 = 336 + 70 =$ **406°F**

(b) $p_f = Ei\ (D_c^2 - D_p^2)/D_pD_c^2 = 30\text{x}10^6(0.004)(7^2 - 4^2)/(4 \text{ x } 7^2) =$ **20,204 lb/in^2**

(c) Max $\sigma_e = 2p_fD_c^2/(D_c^2 - D_p^2) = 2(20{,}204)(7^2)/(7^2 - 4^2) =$ **60,000 lb/in^2**

32.20 A steel collar whose outside diameter = 3.000 in at room temperature is to be shrink fitted onto a steel shaft by heating it to an elevated temperature while the shaft remains at room temperature. The shaft diameter = 1.500 in. For ease of assembly when the collar is heated to an elevated temperature of 1000°F, the clearance between the shaft and the collar is to be 0.007 in. Determine (a) the initial inside diameter of the collar at room temperature so that this clearance is satisfied, (b) the radial pressure and (c) maximum effective stress on the resulting interference fit at room temperature (70°F). For steel, the elastic modulus = 30,000,000 lb/in^2 and coefficient of thermal expansion = 6.7 x 10^{-6} in/in per °F.

Solution: (a) If the clearance = 0.007 in, then the inside diameter of the collar must be
$D_2 = D_p + 0.007 = 1.500 + 0.007.$
$1.507 - D_1 = 6.7 \text{ x } 10^{-6}\ D_1(1000 - 70)$
$1.507 - D_1 = 0.00623\ D_1$
$1.507 = D_1 + 0.00623\ D_1 = 1.00623\ \text{D}_1$
$D_1 = 1.507/1.00623 =$ **1.4977 in**

(b) Interference $i = 1.500 - 1.4977 = 0.00233$ in
$p_f = 30 \text{ x } 10^6\ (0.00233)(3.0^2 - 1.5^2)/(1.5 \text{ x } 3.0^2) =$ **34,950 lb/in^2**

(c) Max $\sigma_e = 2(34{,}950)(3.0^2)/\ (3.0^2 - 1.5^2) =$ **93,200 lb/in^2**

32.21 A pin is to be inserted into a collar using an expansion fit. Properties of the pin and collar metal are: coefficient of thermal expansion is 12.3 x 10^{-6} m/m/°C, yield strength is 400 MPa, and modulus of elasticity is 209 GPa. At room temperature (20°C), the outer and inner diameters of the collar = 95.00 mm and 60.00 mm, respectively, and the pin has a diameter = 60.03 mm. The pin is to be reduced in size for assembly into the collar by cooling to a sufficiently low temperature that there is a clearance of 0.06 mm. (a) What is the temperature to which the pin must be cooled for assembly? (b) What is the radial pressure at room temperature after assembly? (c) What is the safety factor in the resulting assembly?

Solution: (a) $D_2 - D_1 = \alpha D_1(T_2 - T_1) =$
$T_2 = (D_2\text{-}D_1)/(\alpha D_1) + T_1 = ((60.00\text{-}0.06) - 60.03)/(12.3\text{x}10^{-6}*60.03) + 20 =$ **-101.9°C**

(b) $p_f = Ei(D_c^2 - D_p^2)/D_pD_c^2$
$p_f = 209\text{x}10^9(0.03)(95^2 - 60^2)/(60(95^2) = 0.0628(10^9)\ \text{N/m}^2 =$ **62.8 MPa**

(c) Max $\sigma_e = 2p_fD_c^2/(D_c^2 - D_p^2) = 2(62.8)(95^2)/(95^2 - 60^2) = 209$ MPa
If $Y = 400$ MPa and Max $\sigma_e = Y/SF$, then $SF = Y/(\text{ Max } \sigma_e) = 400/209 =$ **1.91**