7 A trial material needs to be selected for the computations. We will first try an inexpensive, low-carbon, cold-rolled steel such as SAE 1020 with S_{ut} = 65 kpsi and S_y = 38 kpsi. Though not exceptionally strong, this material has low notch sensitivity, which will be an advantage given the large stress concentrations. Calculate the uncorrected endurance strength using equation 6.5 (p. 330):

$$S_{e'} = 0.5 S_{ut} = 0.5(65\ 000) = 32\ 500 \text{ psi} \tag{k}$$

This must be reduced by various factors to account for differences between the part and the test specimen.

$$\begin{aligned} S_e &= C_{load}\, C_{size}\, C_{surf}\, C_{temp}\, C_{reliab}\, S_{e'} \\ S_e &= (1)(1)(0.84)(1)(1)(32\ 500) = 27\ 300 \text{ psi} \end{aligned} \tag{l}$$

The loading is bending and torsion, so C_{load} is 1. Since we don't yet know the part size, we will temporarily assume C_{size} =1 and adjust it later. C_{surf} is chosen for a machined finish from either Figure 6-26 (p. 332) or equation 6.7*e* (p. 333). The temperature is not elevated, so C_{temp} = 1, and we assume 50% reliability at this preliminary design stage with C_{reliab} = 1.

8 The material's notch sensitivity is found from either equation 6.13 (p. 345) or Figure 6-36 (pp. 344–345) and is q = 0.5 in bending and q = 0.57 in torsion, assuming a notch radius of 0.01 in.

9 The fatigue stress-concentration factor is found from equation 6.11*b* (p. 343) using the assumed geometric stress-concentration factor noted above. For the bending stress in the step at point C:

$$K_f = 1 + q(K_t - 1) = 1 + 0.5(3.5 - 1) = 2.25 \tag{m}$$

The stress concentration for a step loaded in torsion is less than for the same geometry loaded in bending:

$$K_{fs} = 1 + q(K_{ts} - 1) = 1 + 0.57(2 - 1) = 1.57 \tag{n}$$

From equation 6.17 (p. 364) we find that in this case, the same factor should be used on the mean torsional stress component:

$$K_{fsm} = K_{fs} = 1.57 \tag{o}$$

10 The shaft diameter at point C can now be found from equation 10.6 using the moment magnitude at that point of 63.9 in-lb.

$$\begin{aligned} d_2 &= \left\{ \frac{32 N_f}{\pi} \left[\left(K_f \frac{M_a}{S_f} \right)^2 + \frac{3}{4} \left(K_{fsm} \frac{T_m}{S_y} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} \\ &= \left\{ \frac{32(2.5)}{\pi} \left[\left(2.25 \frac{63.9}{27\ 300} \right)^2 + \frac{3}{4} \left(1.57 \frac{73.1}{38\ 000} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} = 0.531 \text{ in} \end{aligned} \tag{p}$$

If K_{fsm} is set to 1 as ASME recommends, then equation 10.6 gives d = 0.520 in. If the more general equation 10.8 is used, the result is d = 0.557 in. Note that the

10

ASME method is less conservative than equation 10.8 as it gives smaller shaft diameters for the same safety factor. A modified-Goodman diagram for this stress element is shown in Figure 10-8*b*. It predicts failure from fatigue.

11 At point B, under the gear, the moment is less, but the fatigue stress-concentration factors K_f and K_{fs} are greater so should also be calculated. At B:

$$\begin{aligned} K_f &= 1 + q(K_t - 1) = 1 + 0.5(4 - 1) = 2.50 \\ K_{fs} &= 1 + q(K_t - 1) = 1 + 0.57(4 - 1) = 2.70 \end{aligned} \qquad (q)$$

12 The minimum recommended diameter at point B from equation 10.6 is

$$d_1 = \left\{ \frac{32 N_f}{\pi} \left[\left(K_f \frac{M_a}{S_f} \right)^2 + \frac{3}{4} \left(K_{fsm} \frac{T_m}{S_y} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$= \left\{ \frac{32(2.5)}{\pi} \left[\left(2.50 \frac{32.8}{27\,300} \right)^2 + \frac{3}{4} \left(2.71 \frac{73.1}{38\,000} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} = 0.517 \text{ in} \qquad (r)$$

If K_{fsm} is set to 1 as ASME recommends, then equation 10.6 gives $d = 0.444$ in. If the more general equation 10.8 is used, the result is $d = 0.524$ in. Again, the ASME method is nonconservative compared to equation 10.8. A modified-Goodman diagram for this stress element is shown in Figure 10-8*a*. It predicts failure from fatigue.

13 Another location of possible failure is the step against which the sheave seats, at point D. The moment is lower than at C, being about 9.1 lb-in. (See Figure 10-7.) However, the shaft will be stepped smaller there and will have the same order of stress concentration as at point C. (The keyway for the sheave is in a region of zero moment and so will be ignored.) Using those data in equation 10.6 (p. 558) for point D:

$$d_3 = \left\{ \frac{32 N_f}{\pi} \left[\left(K_f \frac{M_a}{S_f} \right)^2 + \frac{3}{4} \left(K_{fsm} \frac{T_m}{S_y} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}}$$

$$= \left\{ \frac{32(2.5)}{\pi} \left[\left(2.25 \frac{9.1}{27\,300} \right)^2 + \frac{3}{4} \left(1.57 \frac{73.1}{38\,000} \right)^2 \right]^{\frac{1}{2}} \right\}^{\frac{1}{3}} = 0.411 \text{ in} \qquad (s)$$

If K_{fsm} is set to 1 as ASME recommends, then equation 10.6 gives $d = 0.360$ in. If the more general equation 10.8 is used, the result is $d = 0.387$ in. A modified-Goodman diagram for this stress element is shown in Figure 10-8*c*. It predicts a yielding failure.

14 From these preliminary calculations, we can determine reasonable sizes for the four step diameters, d_0, d_1, d_2, d_3 of Figure 10-5 (p. 561). The next largest standard ball-bearing diameter to the $d_2 = 0.531$ in calculated for point C is 15 mm or 0.591 in. Selecting this value for d_2, we set $d_3 = 0.50$ in, and $d_1 = 0.625$ in. The stock size d_0 is then 0.75 in, left as-rolled for the outside diameter at the gear flange. These dimen-

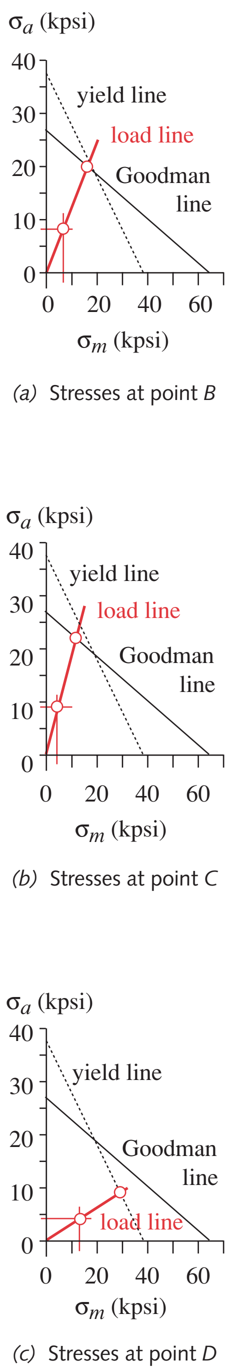

FIGURE 10-8

Modified-Goodman Diagrams for Three Points on Shaft in Example 10-1 sions will give safety factors that meet or exceed the specification. The stresses and safety factors at all three points should now be recalculated using more accurate strength-reduction (e.g., C_{size}) and stress-concentration factors based on the final dimensions.*

EXAMPLE 10-2

Shaft Design for Repeated Torsion with Repeated Bending

Problem **Design a shaft to support the attachments shown in Figure 10-5 (p. 561) with a minimum design safety factor of 2.5.**

Given **The torque and the moment on the shaft are both varying with time in repeated fashion, i.e., their alternating and mean components are of equal magnitude. The mean and alternating components of torque are both 73 lb-in, making the peak torque twice the mean value of Example 10-1. The mean and alternating components of moment are equal in magnitude. Figure 10-9 shows the peak moment and peak torque, which are each twice the value of their fully reversed counterparts of Figure 10-5 and Example 10-1 due to the presence of the mean moment.**

Assumptions **There are no applied axial loads. Steel will be used for infinite life. Assume a stress-concentration factor of 3.5 for the step radii in bending, 2 for the step radii in torsion, and 4 at the keyways. Since the torsional load is not steady and the bending moment is not fully reversed, the ASME method of equation 10.6 (p. 558) should not be used.**

Solution **See Figures 10-5 (p. 561), 10-9, 10-10 and Table 10-1.**

1 For comparison purposes, we will keep all factors except the loading configuration the same as in the previous example. The same low-carbon, cold-rolled steel SAE 1020, which has $S_{ut} = 65$ kpsi, $S_y = 38$ kpsi, and a corrected $S_e = 27.3$ kpsi, is used. Its notch sensitivity is 0.5.

2 There are three points of interest, labeled B, C, and D in Figure 10-5 (p. 561). The fatigue stress-concentration factors are assumed to be the same at C and D and are larger at B. See Example 10-1 (p. 560) for their calculation.

3 The required shaft diameter at point C can be found from equation 10.8 (p. 560).

$$d_2 = \left\{ \frac{32 N_f}{\pi} \left[\frac{\sqrt{\left(K_f M_a\right)^2 + \frac{3}{4}\left(K_{fs} T_a\right)^2}}{S_f} + \frac{\sqrt{\left(K_{fm} M_m\right)^2 + \frac{3}{4}\left(K_{fsm} T_m\right)^2}}{S_{ut}} \right] \right\}^{\frac{1}{3}}$$

$$= \left\{ \frac{32(2.5)}{\pi} \left[\frac{\sqrt{[2.25(64)]^2 + \frac{3}{4}[1.57(73.1)]^2}}{27\,300} + \frac{\sqrt{[2.25(64)]^2 + \frac{3}{4}[1.57(73.1)]^2}}{65\,000} \right] \right\}^{\frac{1}{3}}$$

$$d_2 = 0.614 \qquad (a)$$

* Files EX10-01a, EX10-01b, EX10-01c, and EX10-01d are on the CD-ROM.

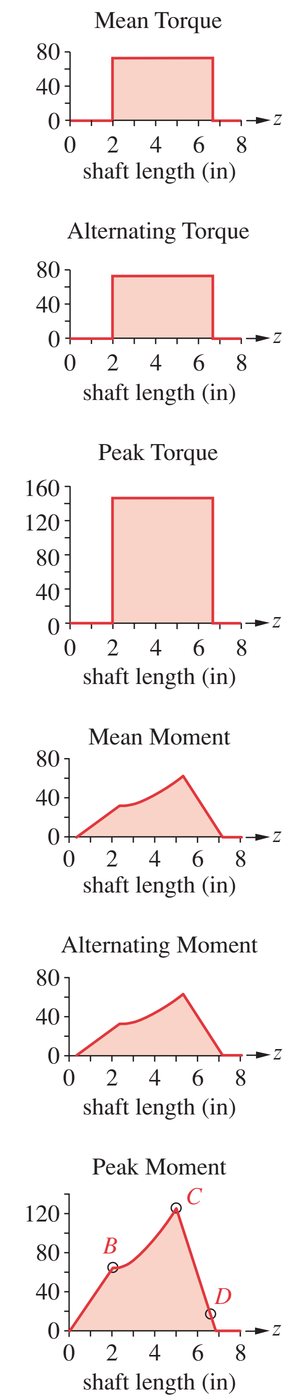

FIGURE 10-9
Torque and Moments in Example 10-2 (lb-in)