**Chapter 6**

**6-1** Eq. (2-21): 

Eq. (6-8): 

Table 6-2: 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**6-2 (a)**  Table A-20: *Sut =* 80 kpsi

Eq. (6-8): 

**(b)**  Table A-20: *Sut =* 90 kpsi

Eq. (6-8): 

**(c)**  Aluminum has no endurance limit. *Ans.*

**(d)** Eq. (6-8): *Sut >* 200 kpsi, 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-3 **

Fig. 6-18: 

Eq. (6-8): 

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-4 **

Fig. 6-18: *Sut*  = 1600 MPa = 232 kpsi. Off the graph, so estimate *f* = 0.77.

Eq. (6-8): *Sut* > 1400 MPa, so *Se* = 700 MPa

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-5** 

Fig. 6-18, point is off the graph, so estimate: *f* = 0.77

Eq. (6-8): *Sut >* 200 kpsi, so 

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-13): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-6** = 160 kpsi

Fig. 6-18: *f* = 0.79

Eq. (6-8): 

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-13): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-7 **

Fig. 6-18: *f* = 0.798

From Fig. 6-10, we note that below 103 cycles on the *S-N* diagram constitutes the low-cycle region, in which Eq. (6-17) is applicable.

Eq. (6-17): 

The testing should be done at a completely reversed stress of 122 kpsi, which is below the yield strength, so it is possible. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-8** The general equation for a line on a log *Sf -* log N scale is *Sf­*  = *aNb*, which is Eq. (6-13). By taking the log of both sides, we can get the equation of the line in slope-intercept form.



Substitute the two known points to solve for unknowns *a* and *b*. Substituting point (1, *Sut*),



From which . Substituting point 



From which 



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-9** Read from graph:  From 



From which







Check:



The end points agree.

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-10** *d* = 1.5 in, *Sut* = 110 kpsi

Eq. (6-8): 

Table 6-2: *a* = 2.70, *b* = − 0.265

Eq. (6-19): 

Eq. (6-20): *kb* = 0.879 *d* −0.107= 0.879(1.5) −0.107 =0.842

Eq. (6-18): *Se* = *kakb* = 0.777(0.842)(55) = 36.0 kpsi *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-11** For AISI 4340 as-forged steel,

Eq. (6-8): *Se* = 100 kpsi

Table 6-2: *a* = 39.9, *b* = − 0.995

Eq. (6-19): *ka* = 39.9(260)−0.995 = 0.158

Eq. (6-20): 

Each of the other modifying factors is unity.

*Se* = 0.158(0.907)(100) = 14.3 kpsi *Ans.*

For AISI 1040:



Each of the other modifying factors is unity

 *Ans.*

Not only is AISI 1040 steel a contender, it has a superior endurance strength. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-12** *D* = 1 in, *d* = 0.8 in, *T* = 1800 lbf⋅in, *f* = 0.9, and from Table A-20 for AISI 1020 CD,

*Sut* = 68 kpsi, and *Sy* = 57 kpsi.

**(a) **

Get the notch sensitivity either from Fig. 6-21, or from the curve-fit Eqs. (6-34) and

(6-35b). Using the equations,





Eq. (6-32): *Kfs* = 1 + *qs* (*Kts* − 1) = 1 + 0.812(1.40 − 1) = 1.32

For a purely reversing torque of *T* = 1800 lbf⋅in,



Eq. (6-8): 

Eq. (6-19): *ka* = 2.70(68)−0.265 = 0.883

Eq. (6-20): *kb* = 0.879(0.8)−0.107 = 0.900

Eq. (6-26): *kc* = 0.59

Eq. (6-18) (labeling for shear): *Sse* = 0.883(0.900)(0.59)(34) = 15.9 kpsi

For purely reversing torsion, use Eq. (6-54) for the ultimate strength in shear.

Eq. (6-54): *Ssu* = 0.67 *Sut* = 0.67(68) = 45.6 kpsi

Adjusting the fatigue strength equations for shear,

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

**(b)** For an operating temperature of the temperature modification factor,

from Table 6-4 is *kd* = 0.90.

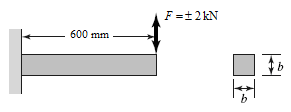
*Sse* = 0.883(0.900)(0.59)(0.9)(34) = 14.3 kpsi





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-13** (Table A-20)

 First evaluate the fatigue strength.





Since the size is not yet known, assume a

typical value of *kb* = 0.85 and check later.

All other modifiers are equal to one.

Eq. (6-18): *Se* = 0.488(0.85)(385) = 160 MPa

In kpsi, *Sut* = 770/6.89 = 112 kpsi

Fig. 6-18: *f* = 0.83

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-13): 

Now evaluate the stress.



Pa, with *b* in m.

Compare strength to stress and solve for the necessary *b.*



*b* = 0.0299 m Select *b* = 30 mm.

Since the size factor was guessed, go back and check it now.

Eq. (6-25): 

Eq. (6-20): 

Our guess of 0.85 was slightly conservative, so we will accept the result of

*b* = 30 mm. *Ans.*

Checking yield,

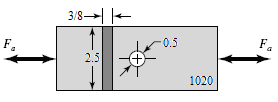




**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-14** Given: *w* =2.5 in, *t* = 3/8 in, *d* = 0.5 in, *nd* = 2. From Table A-20, for AISI 1020 CD,

*Sut* = 68 kpsi and *Sy* = 57 kpsi.



Eq. (6-8): 

Table 6-2: 

Eq. (6-21): *kb* = 1 (axial loading)

Eq. (6-26): *kc* = 0.85

Eq. (6-18): *Se* = 0.88(1)(0.85)(34) = 25.4 kpsi



Get the notch sensitivity either from Fig. 6-20, or from the curve-fit Eqs. (6-34) and

(6-35a). The relatively large radius is off the graph of Fig. 6-20, so we will assume the curves continue according to the same trend and use the equations to estimate the notch sensitivity.





Eq. (6-32): 



Since a finite life was not mentioned, we’ll assume infinite life is desired, so the completely reversed stress must stay below the endurance limit.





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-15** Given: ****

From Table A-20, for AISI 1095 HR, *Sut* = 120 kpsi and *Sy* = 66 kpsi.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-26): 

Eq. (6-18): 

Fig. A-15-14:  

Get the notch sensitivity either from Fig. 6-20, or from the curve-fit Eqs. (6-34) and

(6-35a). Using the equations,





****

****

****

Eq. (6-36): ****



Eq. (6-46): 



A factor of safety less than unity indicates a finite life.

Check for yielding. It is not necessary to include the stress concentration for static yielding of a ductile material.



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-16** From a free-body diagram analysis, the bearing reaction forces are found to be 2.1 kN at the left bearing and 3.9 kN at the right bearing. The critical location will be at the shoulder fillet between the 35 mm and the 50 mm diameters, where the bending moment is large, the diameter is smaller, and the stress concentration exists. The bending moment at this point is *M =* 2.1(200) = 420 kN∙mm. With a rotating shaft, the bending stress will be completely reversed.



This stress is far below the yield strength of 390 MPa, so yielding is not predicted. Find the stress concentration factor for the fatigue analysis.

Fig. A-15-9: *r/d* = 3/35 = 0.086, *D/d* = 50/35 = 1.43, *Kt* =1.7

Get the notch sensitivity either from Fig. 6-20, or from the curve-fit Eqs. (6-34) and

(6-35a). Using the equations, with *Sut* = 470 MPa = 68.2 kpsi and *r* = 3 mm = 0.118 in,





****

Eq. (6-8): 

Eq. (6-19): 

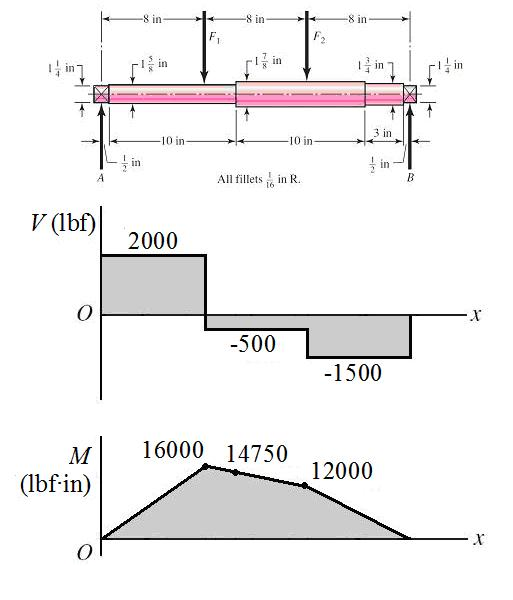
Eq. (6-24): 

Eq. (6-26): 

Eq. (6-18): 



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**



**6-17** From a free-body diagram analysis, the bearing reaction forces are found to be

*RA* = 2000 lbf and *RB* = 1500 lbf. The shear-force and bending-moment diagrams are shown. The critical location will be at the shoulder fillet between the 1-5/8 in and the 1-7/8 in diameters, where the bending moment is large, the diameter is smaller, and the stress concentration exists.

*M* = 16 000 – 500 (2.5) = 14 750 lbf ∙ in

With a rotating shaft, the bending stress will be completely reversed.



This stress is far below the yield strength of 71 kpsi, so yielding is not predicted.

Fig. A-15-9: *r/d* = 0.0625/1.625 = 0.04, *D/d* = 1.875/1.625 = 1.15, *Kt* =1.95

Get the notch sensitivity either from Fig. 6-20, or from the curve-fit Eqs. (6-34) and

(6-35a). Using the equations,



.

Eq. (6-32): 



Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-26): 

Eq. (6-18): 



Infinite life is not predicted. Use the *S-N* diagram to estimate the life.

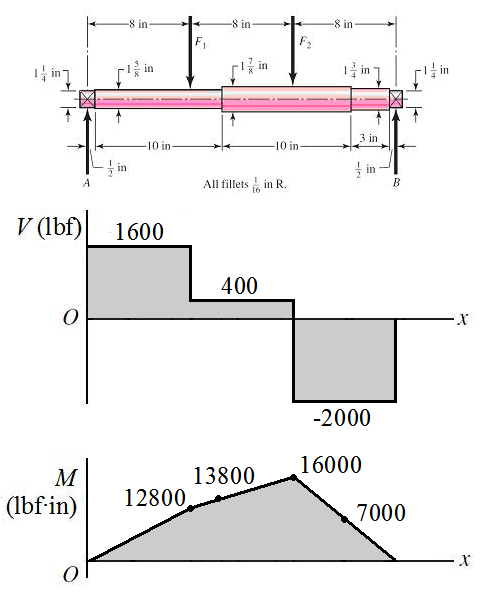
Fig. 6-18: *f* = 0.867





*N* = 4600 cycles *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**



**6-18** From a free-body diagram analysis, the bearing reaction forces are found to be *RA* = 1600 lbf and *RB* = 2000 lbf. The shear-force and bending-moment diagrams are shown. The critical location will be at the shoulder fillet between the 1-5/8 in and the 1-7/8 in diameters, where the bending moment is large, the diameter is smaller, and the stress concentration exists.

*M* = 12 800 + 400 (2.5) = 13 800 lbf ∙ in

With a rotating shaft, the bending stress will be completely reversed. 

This stress is far below the yield strength of 71 kpsi, so yielding is not predicted.

Fig. A-15-9: *r/d* = 0.0625/1.625 = 0.04, *D/d* = 1.875/1.625 = 1.15, *Kt* =1.95

Get the notch sensitivity either from Fig. 6-20, or from the curve-fit Eqs. (6-34) and

(6-35a). Usingthe equations,





Eq. (6-32): 



Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-26): 

Eq. (6-18): 



Infinite life is not predicted. Use the *S-N* diagram to estimate the life.

Fig. 6-18: *f* = 0.867





*N* = 7500 cycles *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-19** Table A-20: 

*N* = (950 rev/min)(10 hr)(60 min/hr) = 570 000 cycles

One approach is to guess a diameter and solve the problem as an iterative analysis problem. Alternatively, we can estimate the few modifying parameters that are dependent on the diameter and solve the stress equation for the diameter, then iterate to check the estimates. We’ll use the second approach since it should require only one iteration, since the estimates on the modifying parameters should be pretty close.

First, we will evaluate the stress. From a free-body diagram analysis, the reaction forces at the bearings are *R*1 = 2 kips and *R*2 = 6 kips. The critical stress location is in the middle of the span at the shoulder, where the bending moment is high, the shaft diameter is smaller, and a stress concentration factor exists. If the critical location is not obvious, prepare a complete bending moment diagram and evaluate at any potentially critical locations. Evaluating at the critical shoulder,





Now we will get the notch sensitivity and stress concentration factor. The notch sensitivity depends on the fillet radius, which depends on the unknown diameter. For now, let us estimate a value of *q* = 0.85 from observation of Fig. 6-20, and check it later.

Fig. A-15-9: 

Eq. (6-32): 

Now, evaluate the fatigue strength.



Since the diameter is not yet known, assume a typical value of *kb* = 0.85 and check later. All other modifiers are equal to one.

*Se* = (0.76)(0.85)(60) = 38.8 kpsi

Determine the desired fatigue strength from the *S-N* diagram.

Fig. 6-18: *f* = 0.82





Compare strength to stress and solve for the necessary *d.*



*d* = 2.29 in

Since the size factor and notch sensitivity were guessed, go back and check them now.

Eq. (6-20): 

From Fig. 6-20 with *r* = *d/*10 = 0.229 in, we are off the graph, but it appears our guess for *q* of 0.85 is low. Assuming the trend of the graph continues, we’ll choose *q* = 0.91 and iterate the problem with the new values of *kb* and *q*.

Intermediate results are *Se* = 36.5 kpsi, *Sf* = 39.6 kpsi, and *Kf* = 1.59. This gives



*d* = 2.36 in *Ans.*

A quick check of *kb* and *q* show that our estimates are still reasonable for this diameter.

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-20 **

Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



****

****

**(a)** Modified Goodman, Table 6-6



**(b)** Gerber, Table 6-7



**(c)** ASME-Elliptic, Table 6-8



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-21 **

Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



****

****

**(a)** Modified Goodman, Table 6-6



**(b)** Gerber, Table 6-7



**(c)** ASME-Elliptic, Table 6-8



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-22 **

Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



****

****

**(a)** Modified Goodman, Table 6-6



**(b)** Gerber, Table 6-7



**(c)** ASME-Elliptic, Table 6-8



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-23 **

Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



****

****

**(a)** Modified Goodman, Table 6-6



**(b)** Gerber criterion of Table 6-7 is only valid for *m* > 0; therefore use Eq. (6-47).



**(c)** ASME-Elliptic, Table 6-8



Since infinite life is not predicted, estimate a life from the *S-N­* diagram. Since *'m* = 0, the stress state is completely reversed and the *S-N*diagram is applicable for *'a.*

Fig. 6-18: *f* = 0.875

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-24 **

Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.





****

****

**(a)** Modified Goodman, Table 6-6



**(b)** Gerber, Table 6-7



**(c)** ASME-Elliptic, Table 6-8



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-25** Given: . From Table A-20, for AISI 1040 CD,

Check for yielding





Determine the fatigue factor of safety based on infinite life

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-21): 

Eq. (6-26): 

Eq. (6-18): 

Fig. 6-20: *q* = 0.83

Fig. A-15-1: 





Note, since *σm* = 0, the stress is completely reversing, and



Since infinite life is not predicted, estimate the life from the *S-N­* diagram. With *m* = 0, the stress state is completely reversed, and the *S-N*diagram is applicable for *a.*

*Sut* = 590/6.89 = 85.6 kpsi

Fig. 6-18: *f* = 0.87

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

*N* = 34 000 cycles *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-26** 

Check for yielding





Determine the fatigue factor of safety based on infinite life.

From Prob. 6-25: 





Modified Goodman criteria:





Gerber criteria:







ASME-Elliptic criteria:



= 1.54 *Ans*.

The results are consistent with Fig. 6-27, where for a mean stress that is about half of the yield strength, the Modified Goodman line should predict failure significantly before the other two.

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-27** 

**(a)** 

Check for yielding





From Prob. 6-25: 



For the modified Goodman criteria,





Since infinite life is not predicted, estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress. For the modified Goodman criteria, see Ex. 6-12.



Fig. 6-18: *f* = 0.87

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

**(b)** 

The maximum load is the same as in part (a), so





Factor of safety based on infinite life:







**(c)** 

The compressive load is the largest, so check it for yielding.





Factor of safety based on infinite life:



For *σm* < 0, 

Since infinite life is not predicted, estimate a life from the *S-N­* diagram. For a negative mean stress, we shall assume the equivalent completely reversed stress is the same as the actual alternating stress. Get *a* and *b* from part (a).

Eq. (6-16): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-28** Eq. (2-21): *Sut* = 0.5(400) = 200 kpsi

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-25): 

Eq. (6-20): 

Since we have used the equivalent diameter method to get the size factor, and in doing so introduced greater uncertainties, we will choose not to use a size factor greater than one. Let *kb* = 1.

Eq. (6-18): 





**(a)** Modified Goodman criterion





Since infinite life is not predicted, estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress (See Ex. 6-12).



Fig. 6-18: *f* = 0.775

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

**(b)** Gerber criterion, Table 6-7



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-29 **

**(a) **

****

****

****

**(b)** Get the fatigue strength information.

Eq. (2-21): *Sut* = =3.4*HB* = 3.4(490) = 1666 MPa

From problem statement: *Sy* = 0*.*9*Sut* = 0*.*9(1666) = 1499 MPa

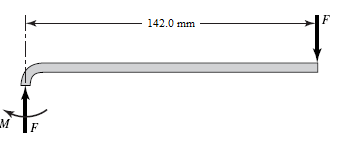
Eq. (6-8): 

Eq. (6-19): *ka* = 1*.*58(1666)-0*.*085 = 0*.*84

Eq. (6-25): *de* = 0*.*808[20(4)]1*/*2 = 7.23 mm

Eq. (6-20): *kb* = 1.24(7.23)-0*.*107 = 1*.*00

Eq. (6-18): *Se* = 0*.*84(1)(700) = 588 MPa



This is a relatively thick curved beam, so use the method in Sect. 3-18 to find the stresses. The maximum bending moment will be to the centroid of the section as shown.

*M* = 142*F* N∙mm, *A* = 4(20) = 80 mm2, *h* = 4 mm, *ri* = 4 mm, *ro* = *ri* + *h* = 8 mm,

*rc* = *ri* + *h*/2 = 6 mm

Table 3-4: 







Get the stresses at the inner and outer surfaces from Eq. (3-65) with the axial stresses added. The signs have been set to account for tension and compression as appropriate.













To check for yielding, we note that the largest stress is –498.6 MPa (compression) on the inner radius. This is considerably less than the estimated yield strength of 1499 MPa, so yielding is not predicted.

Check for fatigue on both inner and outer radii since one has a compressive mean stress and the other has a tensile mean stress.

Inner radius:

Since *m* < 0, 

Outer radius:

Since *m* > 0, using the Modified Goodman line,



Infinite life is predicted at both inner and outer radii. The outer radius is critical, with a fatigue factor of safety of *nf* = 3.33. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-30** From Table A-20, for AISI 1018 CD, 

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-26): 

Eq. (6-18): 

Fillet:

Fig. A-15-5: 

Use Fig. 6-20 or Eqs. (6-34) and (6-35a) for *q*. Estimate a little high since it is off the graph. *q* = 0.85







Since the midrange stress is negative,



Hole:

Fig. A-15-1: 

Use Fig. 6-20 or Eqs. (6-34) and (6-35a) for *q*. Estimate a little high since it is off the graph, *q* = 0.85









Since the midrange stress is negative,



Thus the design is controlled by the threat of fatigue at the hole with a minimum factor of safety of 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-31** 

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-26): 

Eq. (6-18): 

Fillet:

Fig. A-15-5: 

Use Fig. 6-20 or Eqs. (6-34) and (6-35a) for *q*. Estimate a little high since it is off the graph. *q* = 0.85









Using Modified Goodman criteria,





Hole:

Fig. A-15-1: 

Use Fig. 6-20 or Eqs. (6-34) and (6-35a) for *q*. Estimate a little high since it is off the graph. *q* = 0.85









Using Modified Goodman criteria





Thus the design is controlled by the threat of fatigue at the fillet with a minimum factor of safety of 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-32** 

From Prob. 6-30, the fatigue factor of safety at the hole is *nf* = 1.48. To match this at the fillet,



where *Se* is unchanged from Prob. 6-30. The only aspect of *a* that is affected by the fillet radius is the fatigue stress concentration factor. Obtaining *a* in terms of *Kf*,



Equating to the desired stress, and solving for *Kf*,



Assume since we are expecting to get a smaller fillet radius than the original, that *q* will be back on the graph of Fig. 6-20, so we will estimate *q* = 0.8.



From Fig. A-15-5, with *D / d* = 3.5/3 = 1.17 and *Kt* = 2.6, find *r / d*. Choosing *r / d* = 0.03, and with *d* = *w*2 = 3.0,

****

At this small radius, our estimate for *q* is too high. From Fig. 6-20, with *r* = 0.09, *q* should be about 0.75. Iterating, we get *Kt* = 2.8. This is at a difficult range on Fig. A-15-5 to read the graph with any confidence, but we’ll estimate *r / d* = 0.02, giving *r* = 0.06 in. This is a very rough estimate, but it clearly demonstrates that the fillet radius can be relatively sharp to match the fatigue factor of safety of the hole. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-33** 

Inner fiber where 



Table 3-4, p. 135,







Eq. (3-65), p. 133,



where *T* is in lbf∙in and  is in psi.



Eq. (6-8): 

Eq. (6-19): 

Eq. (6-25): 

Eq. (6-20): 

Eq. (6-19): 

For a compressive midrange component, 





Outer fiber where 









**(a)** Using Eq. (6-46), for modified Goodman, we have





**(b)** Gerber, Eq. (6-47), at the outer fiber,

****

****

**(c)** To guard against yield, use *T* of part (b) and the inner stress.



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-34** From Prob. 6-33, 

**(a)** Assuming the beam is straight,



Goodman: 

****

**(b)** Gerber: 



**(c)** 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-35 **

Bending: 

Axial: 

Torsion: 

Eqs. (6-55) and (6-56):



Using Modified Goodman, Eq. (6-46),





Check for yielding, using the conservative,



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-36 **

Bending: 

Torsion: 

Eqs. (6-55) and (6-56):



Using Modified Goodman,





Check for yielding, using the conservative ,



Since the conservative yield check indicates yielding, we will check more carefully with obtained directly from the maximum stresses, using the distortion energy failure theory, without stress concentrations. Note that this is exactly the method used for static failure in Ch. 5.



Since yielding is not predicted, and infinite life is not predicted, we would like to estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress (See Ex. 6-12).



This stress is much higher than the ultimate strength, rendering it impractical for the *S-N* diagram. We must conclude that the stresses from the combination loading, when increased by the stress concentration factors, produce such a high midrange stress that the equivalent completely reversed stress method is not practical to use. Without testing, we are unable to predict a life.

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-37** Table A-20: ****

From Prob. 3-68, the critical stress element experiences ** = 15.3 kpsi and ** = 4.43 kpsi. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 15.3 kpsi, *m* = 0 kpsi, *a* = 0 kpsi, *m* = 4.43 kpsi. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-38** Table A-20: ****

From Prob. 3-69, the critical stress element experiences ** = 263 MPa and ** = 57.7 MPa. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 263 MPa, *m* = 0, *a* = 0 MPa, *m* = 57.7 MPa. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



 Infinite life is not predicted. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-39** Table A-20: ****

From Prob. 3-70, the critical stress element experiences ** = 21.5 kpsi and ** = 5.09 kpsi. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 21.5 kpsi, *m* = 0 kpsi, *a* = 0 kpsi, *m* = 5.09 kpsi. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.







Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-40** Table A-20: ****

From Prob. 3-71, the critical stress element experiences ** = 72.9 MPa and ** = 20.3 MPa. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 72.9 MPa, *m* = 0 MPa, *a* = 0 MPa, *m* = 20.3 MPa. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-41** Table A-20: ****

From Prob. 3-72, the critical stress element experiences ** = 35.2 kpsi and ** = 7.35 kpsi. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 35.2 kpsi, *m* = 0 kpsi, *a* = 0 kpsi, *m* = 7.35 kpsi. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



 Infinite life is not predicted. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-42** Table A-20: ****

From Prob. 3-73, the critical stress element experiences ** = 333.9 MPa and ** = 126.3 MPa. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 333.9 MPa, *m* = 0 MPa, *a* = 0 MPa, *m* = 126.3 MPa. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



The sample fails by yielding, infinite life is not predicted. *Ans.*

The fatigue analysis will be continued only to obtain the requested fatigue factor of safety, though the yielding failure will dictate the life.

Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



 Infinite life is not predicted. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-43** Table A-20: ****

From Prob. 3-74, the critical stress element experiences completely reversed bending stress due to the rotation, and steady torsional and axial stresses.



Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-44** Table A-20: ****

From Prob. 3-76, the critical stress element experiences completely reversed bending stress due to the rotation, and steady torsional and axial stresses.



Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



 Infinite life is not predicted. *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-45** Table A-20: ****

From Prob. 3-77, the critical stress element experiences ** = 68.6 MPa and ** = 37.7 MPa. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 68.6 MPa, *m* = 0 MPa, *a* = 0 MPa, *m* = 37.7 MPa. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-46** Table A-20: ****

From Prob. 3-79, the critical stress element experiences ** = 3.46 kpsi and ** = 0.882 kpsi. The bending is completely reversed due to the rotation, and the torsion is steady, giving *a* = 3.46 kpsi, *m* = 0, *a* = 0 kpsi, *m* = 0.882 kpsi. Obtain von Mises stresses for the alternating, mid-range, and maximum stresses.



Check for yielding, using the distortion energy failure theory.



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



 *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-47** Table A-20: ****

From Prob. 3-80, the critical stress element experiences ** = 16.3 kpsi and ** = 5.09 kpsi. Since the load is applied and released repeatedly, this gives **max = 16.3 kpsi, **min = 0 kpsi, **max = 5.09 kpsi, **min = 0 kpsi. Consequently,*m* = *a* = 8.15 kpsi, *m* = *a* = 2.55 kpsi.

For bending, from Eqs. (6-34) and (6-35a),





Eq. (6-32): 

For torsion, from Eqs. (6-34) and (6-35b),





Eq. (6-32): 

Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).



Check for yielding, using the conservative ,



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-48** Table A-20: ****

From Prob. 3-81, the critical stress element experiences ** = 16.4 kpsi and ** = 4.46 kpsi. Since the load is applied and released repeatedly, this gives **max = 16.4 kpsi, **min = 0 kpsi, **max = 4.46 kpsi, **min = 0 kpsi. Consequently,*m* = *a* = 8.20 kpsi, *m* = *a* = 2.23 kpsi.

For bending, from Eqs. (6-34) and (6-35a),





Eq. (6-32): 

For torsion, from Eqs. (6-34) and (6-35b),





Eq. (6-32): 

Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).



Check for yielding, using the conservative ,



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-49** Table A-20: ****

From Prob. 3-82, the critical stress element experiences repeatedly applied bending, axial, and torsional stresses of *x*,bend = 20.2 kpsi, *x*,axial = 0.1 kpsi, and ** = 5.09 kpsi.. Since the axial stress is practically negligible compared to the bending stress, we will simply combine the two and not treat the axial stress separately for stress concentration factor and load factor. This gives **max = 20.3 kpsi, **min = 0 kpsi, **max = 5.09 kpsi, **min = 0 kpsi. Consequently,*m* = *a* = 10.15 kpsi, *m* = *a* = 2.55 kpsi.

For bending, from Eqs. (6-34) and (6-35a),





Eq. (6-32): 

For torsion, from Eqs. (6-34) and (6-35b),





Eq. (6-32): 

Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).



Check for yielding, using the conservative ,



Obtain the modifying factors and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,





**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-50** Table A-20: ****

From Prob. 3-83, the critical stress element on the neutral axis in the middle of the longest side of the rectangular cross section experiences a repeatedly applied shear stress of **max = 14.3 kpsi, **min = 0 kpsi. Thus, *m* = *a* = 7.15 kpsi. Since the stress is entirely shear, it is convenient to check for yielding using the standard Maximum Shear Stress theory.



Find the modifiers and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

The size factor for a rectangular cross section loaded in torsion is not readily available. Following the procedure on p. 297, we need an equivalent diameter based on the 95 percent stress area. However, the stress situation in this case is nonlinear, as described on p. 116. Noting that the maximum stress occurs at the middle of the longest side, or with a radius from the center of the cross section equal to half of the shortest side, we will simply choose an equivalent diameter equal to the length of the shortest side.



Eq. (6-20): 

We will round down to *kb* = 1.

Eq. (6-26): 

Eq. (6-18): ****

Since the stress is entirely shear, we choose to use a load factor *kc* = 0.59, and convert the ultimate strength to a shear value rather than using the combination loading method of Sec. 6-14. From Eq. (6-54), *Ssu* = 0.67*Su* = 0.67 (64) = 42.9 kpsi.

Using Modified Goodman,



**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-51** Table A-20: ****

From Prob. 3-84, the critical stress element experiences ** = 28.0 kpsi and ** = 15.3 kpsi. Since the load is applied and released repeatedly, this gives **max = 28.0 kpsi, **min = 0 kpsi, **max = 15.3 kpsi, **min = 0 kpsi. Consequently,*m* = *a* = 14.0 kpsi, *m* = *a* = 7.65 kpsi. From Table A-15-8 and A-15-9,



Eqs. (6-34) and (6-35), or Figs. 6-20 and 6-21: *q*bend = 0.78, *q*tors = 0.82

Eq. (6-32):



Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).



Check for yielding, using the conservative ,



Since stress concentrations are included in this quick yield check, the low factor of safety is acceptable.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



****

Since infinite life is not predicted, estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress (See Ex. 6-12).



Fig. 6-18: *f* = 0.9

Eq. (6-14): 

Eq. (6-15): 

Eq. (6-16): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-52** Table A-20: ****

From Prob. 3-85, the critical stress element experiences *x,*bend = 46.1 kpsi, *x,*axial = 0.382 kpsi and ** = 15.3 kpsi. The axial load is practically negligible, but we’ll include it to demonstrate the process. Since the load is applied and released repeatedly, this gives **max,bend = 46.1 kpsi, **min,bend = 0 kpsi, **max,axial = 0.382 kpsi, **min,axial = 0 kpsi, **max = 15.3 kpsi, **min = 0 kpsi. Consequently,*m,*bend = *a,*bend = 23.05 kpsi, *m,*axial = *a,*axial = 0.191 kpsi, *m* = *a* = 7.65 kpsi. From Table A-15-7, A-15-8 and A-15-9,



Eqs. (6-34) and (6-35), or Figs. 6-20 and 6-21: *q*bend = *q*axial =0.78, *q*tors = 0.82

Eq. (6-32):



Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).





Check for yielding, using the conservative ,



Since the conservative yield check indicates yielding, we will check more carefully with with obtained directly from the maximum stresses, using the distortion energy failure theory, without stress concentrations. Note that this is exactly the method used for static failure in Ch. 5.



This shows that yielding is imminent, and further analysis of fatigue life should not be interpreted as a guarantee of more than one cycle of life.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



****

Since infinite life is not predicted, estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress (See Ex. 6-12).



This stress is much higher than the ultimate strength, rendering it impractical for the *S-N* diagram. We must conclude that the fluctuating stresses from the combination loading, when increased by the stress concentration factors, are so far from the Goodman line that the equivalent completely reversed stress method is not practical to use. Without testing, we are unable to predict a life.

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-53** Table A-20: ****

From Prob. 3-86, the critical stress element experiences *x,*bend = 55.5 kpsi, *x,*axial = 0.382 kpsi and ** = 15.3 kpsi. The axial load is practically negligible, but we’ll include it to demonstrate the process. Since the load is applied and released repeatedly, this gives **max,bend = 55.5 kpsi, **min,bend = 0 kpsi, **max,axial = 0.382 kpsi, **min,axial = 0 kpsi, **max = 15.3 kpsi, **min = 0 kpsi. Consequently,*m,*bend = *a,*bend = 27.75 kpsi, *m,*axial = *a,*axial = 0.191 kpsi, *m* = *a* = 7.65 kpsi. From Table A-15-7, A-15-8 and A-15-9,



Eqs. (6-34) and (6-35), or Figs. 6-20 and 6-21: *q*bend = *q*axial =0.78, *q*tors = 0.82

Eq. (6-32):



Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).



Since these stresses are relatively high compared to the yield strength, we will go ahead and check for yielding using the distortion energy failure theory.



This shows that yielding is predicted. Further analysis of fatigue life is just to be able to report the fatigue factor of safety, though the life will be dictated by the static yielding failure, i.e. *N* = 1/2 cycle. *Ans.*

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-18): 

Using Modified Goodman,



****

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-54** From Table A-20, for AISI 1040 CD, *Sut* = 85 kpsi and *Sy* = 71 kpsi. From the solution to Prob. 6-17 we find the completely reversed stress at the critical shoulder fillet to be **rev = 35.0 kpsi, producing *a* = 35.0 kpsi and *m* = 0 kpsi. This problem adds a steady torque which creates torsional stresses of



From Table A-15-8 and A-15-9, *r/d* = 0.0625/1.625 = 0.04, *D/d* = 1.875/1.625 = 1.15, *Kt*,bend=1.95, *Kt*,tors=1.60

Eqs. (6-34) and (6-35), or Figs. 6-20 and 6-21: *q*bend = 0.76, *q*tors = 0.81

Eq. (6-32):



Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).



Check for yielding, using the conservative ,



From the solution to Prob. 6-17, *Se* = 29.5 kpsi. Using Modified Goodman,



****

Since infinite life is not predicted, estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress (See Ex. 6-12).



Fig. 6-18: *f* = 0.867





*N* = 2300 cycles *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-55** From the solution to Prob. 6-18 we find the completely reversed stress at the critical shoulder fillet to be **rev = 32.8 kpsi, producing *a* = 32.8 kpsi and *m* = 0 kpsi. This problem adds a steady torque which creates torsional stresses of



From Table A-15-8 and A-15-9, *r/d* = 0.0625/1.625 = 0.04, *D/d* = 1.875/1.625 = 1.15, *Kt*,bend=1.95, *Kt*,tors=1.60

Eqs. (6-34) and (6-35), or Figs. 6-20 and 6-21: *q*bend = 0.76, *q*tors = 0.81

Eq. (6-32):



Obtain von Mises stresses for the alternating and mid-range stresses from Eqs. (6-55) and (6-56).





Check for yielding, using the conservative ,



From the solution to Prob. 6-18, *Se* = 29.5 kpsi. Using Modified Goodman,



****

Since infinite life is not predicted, estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress (See Ex. 6-12).



Fig. 6-18: *f* = 0.867





*N* = 4000 cycles *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-56 **

Eqs. (6-34) and (6-35*b*), or Fig. 6-21: *qs* = 0.80

Eq. (6-32): 

****

****

****

****

Since the stress is entirely shear, it is convenient to check for yielding using the standard Maximum Shear Stress theory.



Find the modifiers and endurance limit.

Eq. (6-8): 

Eq. (6-19): 

Eq. (6-24): 

Eq. (6-20): 

Eq. (6-26): ****

Eq. (6-18): ****

Since the stress is entirely shear, we will use a load factor *kc* = 0.59, and convert the ultimate strength to a shear value rather than using the combination loading method of Sec. 6-14. From Eq. (6-54), *Ssu* = 0.67*Su* = 0.67 (55) = 36.9 kpsi.

**(a)** Modified Goodman, Table 6-6



**(b)** Gerber, Table 6-7







**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-57** 

From Eqs. (6-34) and (6-35*a*), or Fig. 6-20, with a notch radius of 0.1 in, *q* = 0.9. Thus, with *Kt* = 3 from the problem statement,









From Eqs. (6-34) and (6-35*b*), or Fig. 6-21, with a notch radius of 0.1 in,  Thus, with *Kts* = 1.8 from the problem statement,







Eqs. (6-55) and (6-56):



Eq. (6-8): 

Eq. (6-19): 

Eq. (6-20): 

Eq. (6-18): 

Modified Goodman: 



Yield (conservative): 

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-58** From Prob. 6-57, ****

****

****

****

****

****

****

****

****

****

****

Eqs. (6-55) and (6-56):



Modified Goodman: 

*nf* = 0.79

Since infinite life is not predicted, estimate a life from the *S-N­* diagram. First, find an equivalent completely reversed stress (See Ex. 6-12).



Fig. 6-18: *f* = 0.8







*N* = 67 600 cycles *Ans.*

**\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_**

**6-59** For AISI 1020 CD, From Table A-20, *Sy* = 390 MPa, *Sut* = 470 MPa. Given: *Se* = 175 MPa.

First Loading: ****

Goodman: 



Second loading: ****



(**a**) Miner’s method: 



(**b**) Manson’s method: The number of cycles remaining after the first loading

*N*remaining =145 920 − 80 000 = 65 920 cycles

Two data points: 0.9(470) MPa, 103 cycles

223.8 MPa, 65 920 cycles



\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**6-60** Given: *Se* = 50 kpsi, *Sut* = 140 kpsi, *f* =0.8. Using Miner’s method,







\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**6-61** Given: *Sut* = 530 MPa, *Se* = 210 MPa, and *f* = 0.9.

**(a)** Miner’s method

****

****

****





(**b**) Manson’s method:

The life remaining after the first series of cycling is *NR*1 = 13 550 − 5000 = 8550 cycles. The two data points required to define are [0.9(530), 103] and (350, 8550).











\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**6-62** Given: *Se* = 45 kpsi, *Sut* = 85 kpsi, *f* = 0.86, and *σa* = 35 kpsi and *σm* = 30 kpsi for 12 (103) cycles.

Gerber equivalent reversing stress: 

(**a**) Miner’s method: *σ*rev < *Se*. According to the method, this means that the endurance limit has not been reduced and the new endurance limit is  = 45 kpsi. Ans.

(**b**) Manson’s method: Again, *σ*rev < *Se*. According to the method, this means that the material has not been damaged and the endurance limit has not been reduced. Thus, the new endurance limit is  = 45 kpsi. Ans.

\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**6-63** Given: *Se* = 45 kpsi, *Sut* = 85 kpsi, *f* = 0.86, and *σa* = 35 kpsi and *σm* = 30 kpsi for 12 (103) cycles.

Goodman equivalent reversing stress: 

Initial cycling





(**a**) Miner’s method (see discussion on p. 333): The number of remaining cycles at 54.09 kpsi is *N*remaining = 52 190 − 12 000 = 40 190 cycles. The new coefficients are *b′ = b*, and *a′ =Sf /Nb* = 54.09/(40 190) − 0.070 235 = 113.89 kpsi. The new endurance limit is



(**b**) Manson’s method (see discussion on p. 334): The number of remaining cycles at 54.09 kpsi is *N*remaining = 52 190 − 12 000 = 40 190 cycles. At 103 cycles,

*Sf* = 0.86(85) = 73.1 kpsi. The new coefficients are

*b′* = [log(73.1/54.09)]/log(103/40 190) = − 0.081 540 and *a′* = *σ*1/(*N*remaining) *b′* =

54.09/(40 190) − 0.081 540 = 128.39 kpsi. The new endurance limit is



\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_