**Chapter 14**

**14-1** 

Table 14-2: *Y* = 0*.*331

Eq. (13-34): 

Eq. (14-4*b*): 

Eq. (13-35) : 

Eq. (14-7): 

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**14-2** 

Table 14-2: *Y* = 0.309

Eq. (13-34): 

Eq. (14-4*b*): 

Eq. (13-35) : 

Eq. (14-7): ****

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**14-3** 

Table 14-2: *Y* = 0.309



Eq. (14-6*b*): 

Eq. (13-36): 

Eq. (14-8): 

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**14-4** 

Table 14-2: *Y* = 0.296



Eq. (14-6*b*): 

Eq. (13-36): 

Eq. (14-8): 

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**14-5** 

Table 14-2: *Y* = 0.296



Eq. (14-6*b*): 

Eq. (13-36): 

Eq. (14-8): 

From Table 13-2, use *F* = 11 mm or 12 mm, depending on availability.*Ans.*

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**14-6** 

Table 14-2: *Y* = 0.322



Eq. (14-6*b*): 

Eq. (13-36): 

Eq. (14-8): 

From Table 13-2, use *F* = 28 mm. *Ans.*

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**14-7** 

Table 14-2: *Y* = 0.337

Eq. (13-34): 

Eq. (14-4*b*): 

Eq. (13-35) : 

Eq. (14-7): 

Use *F* = 63.50 mm (2*.*5 in)  *Ans.*

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**14-8** 

Table 14-2: *Y* = 0.296

Eq. (13-34): 

Eq. (14-4*b*): 

Eq. (13-35) : 

Eq. (14-7): 

Use *F* = 63.50 mm (2*.*5 in) *Ans.*

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**14-9** Try *P* = 8 (0.315) which gives *d* = 18*/*0.315 = 57.15 mm and *Y* = 0*.*309*.*

Eq. (13-34): 

Eq. (14-4*b*): 

Eq. (13-35): 

Eq. (14-7): 

Using coarse integer pitches from Table 13-2, the following table is formed.

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| *P* | *d* | *V* | *Kv* | *Wt* | *F* |
| 0.079 | 228.60 | 7.182 | 2.177 | 259.74 | 2.09 |
| 0.118 | 152.40 | 4.788 | 1.785 | 389.61 | 3.86 |
| 0.157 | 114.30 | 3.591 | 1.589 | 519.48 | 6.10 |
| 0.236 | 76.20 | 2.394 | 1.392 | 779.21 | 12.03 |
| 0.315 | 57.15 | 1.795 | 1.294 | 1038.95 | 19.88 |
| 0.394 | 45.72 | 1.436 | 1.235 | 1298.69 | 29.65 |
| 0.472 | 38.10 | 1.197 | 1.196 | 1558.43 | 41.34 |
| 0.630 | 28.58 | 0.898 | 1.147 | 2077.90 | 70.48 |

Other considerations may dictate the selection. Good candidates are *P* = 8 (0.315 mm)

(*F* = 22.23 mm)and  *P* =0.394 (*F* = 31.75 mm)*. Ans.*

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**14-10** Try *m* = 2 mmwhich gives *d* = 2(18) = 36 mm and *Y* = 0*.*309*.*



Eq. (14-6*b*): 

Eq. (13-36): 

Eq. (14-8): 

Using the preferred module sizes from Table 13-2:

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| *m* | *d* | *V* | *Kv* | *Wt* | *F* |
| 1.00 | 18.0 | 0.848 | 1.139 | 1768.388 | 86.917 |
| 1.25 | 22.5 | 1.060 | 1.174 | 1414.711 | 57.324 |
| 1.50 | 27.0 | 1.272 | 1.209 | 1178.926 | 40.987 |
| 2.00 | 36.0 | 1.696 | 1.278 | 884.194 | 24.382 |
| 3.00 | 54.0 | 2.545 | 1.417 | 589.463 | 12.015 |
| 4.00 | 72.0 | 3.393 | 1.556 | 442.097 | 7.422 |
| 5.00 | 90.0 | 4.241 | 1.695 | 353.678 | 5.174 |
| 6.00 | 108.0 | 5.089 | 1.834 | 294.731 | 3.888 |
| 8.00 | 144.0 | 6.786 | 2.112 | 221.049 | 2.519 |
| 10.00 | 180.0 | 8.482 | 2.391 | 176.839 | 1.824 |
| 12.00 | 216.0 | 10.179 | 2.669 | 147.366 | 1.414 |
| 16.00 | 288.0 | 13.572 | 3.225 | 110.524 | 0.961 |
| 20.00 | 360.0 | 16.965 | 3.781 | 88.419 | 0.721 |
| 25.00 | 450.0 | 21.206 | 4.476 | 70.736 | 0.547 |
| 32.00 | 576.0 | 27.143 | 5.450 | 55.262 | 0.406 |
| 40.00 | 720.0 | 33.929 | 6.562 | 44.210 | 0.313 |
| 50.00 | 900.0 | 42.412 | 7.953 | 35.368 | 0.243 |

Other design considerations may dictate the size selection. For the present design,

*m* = 2 mm (*F* = 25 mm) is a good selection. *Ans.*

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**14-11** Given: *Sc* = 2.22 *HB* +200 MPa,*HB* = 200, Pinion *NP* = 16 teeth, *φ* = 20o, *m* = 8 mm,

*F* = 90 mm, *nP* = 150 rev/min, *W t* = 6 kN, gear ratio 4:1. *NG* = 16(4) = 64 teeth.

Eq. (13-2): *dP* = *mNP* = 8(16) = 128 mm, *dG* = *mNG* = 8(64) = 512 mm

Pinion and Gear steel,

Eq. (14-13): 



Eq. (14-6*b*): 

Eq. (14-12): 

Eq. (14-14):



*Sc* = 2.22 *HB* +200 = 2.22(200) + 200 = 644 MPa



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**14-12**



Eq. (14-4*b*): 

Eq. (13-36): 

Table 14-8:  [Note: Using Eq. (14-13) can result in wide variation in *Cp* due to wide variation in cast iron properties.]

Eq. (14-12): 

Eq. (14-14): 



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**14-13**



Eq. (14-4*b*): 

Eq. (13-36): 

Table 14-8: [Note: Using Eq. (14-13) can result in wide variation in *Cp* due to wide variation in cast iron properties.]

Eq. (14-12): 

Eq. (14-14):



Use *F* = 19.05 mm (0*.*75 in) *Ans.*

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**14-14 **

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Eq. (14-6*a*): 



where *H* is in kW and *Wt* is in kN

Table 14-8:[Note: Using Eq. (14-13) can result in wide variation in *Cp* due to wide variation in cast iron properties].

Eq. (14-12): 

Eq. (14-14): 



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**14-15**



Eq. (14-6*a*): 



Table 14-8:[Note: Using Eq. (14-13) can result in wide variation in *Cp* due to wide variation in cast iron properties.]

Eq. (14-12): 

Eq. (14-14):

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**14-16** The pinion controls the design.

*Bending YP*= 0*.*303, *YG*= 0*.*359



Eq. (14-4*b*): 

Eq. (6-10): 

Eq. (6-18): *ka*=*aSutb* = 3.04(524)–0*.*217 = 0*.*781



Eq. (14-3): 

Eq. (b), Sec. 14-1: 

Eq. (6-24): 

Eq. (6-19): 

*kc* = *kd* = *ke* = 1

Account for one-way bending with *kf* = 1.66. (See Ex. 14-2.)

Eq. (6-17): *Se*= 0*.*781(0*.*996)(1)(1)(1)(1*.*66)(262) = 338.55 MPa

For stress concentration, find the radius of the root fillet (See Ex. 14-2).



From Fig. A-15-6,



Approximate *D/d* =∞with *D/d* = 3; from Fig. A-15-6, *Kt*= 1*.*68*.*

From Fig. 6-26, with *Sut* = 524 MPa and *r* = 0.635 mm, *q* = 0.62.

Eq. (6-32): *Kf* = 1 + *q*(*Kt* – 1) =1 + 0.62 (1.68 – 1) = 1.42



*Wear*

*ν*1 = *ν*2 = 0*.*292, *E*1 = *E*2 = 207 GPa

Eq. (14-13): 

Eq. (14-12): 





Eq. (6-79): 

From the discussion and unnumbered equation immediately following Eq. (6-80), 

From Eq. (14-14):





Rating power (pinion controls):

*H*1 = 1.29 kW

*H*2 = 0*.*1 kW

*H*all = (min 1.29, 0*.*1) = 0*.*1 kW *Ans.*

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**14-17** See Prob. 14-16 solution for equation numbers.

Pinion controls: *YP*= 0*.*322, *YG*= 0*.*447

*Bending dP*= 20*/*0.118 = 169.33,mm *dG*= 100/0.118 = 846.66 mm

(P=0.118 teeth/mm)

**

*kf* = 1.66 (See Ex. 14-2.)





*Kt*= 1*.*75, *q* = 0.85, *Kf*= 1*.*64





*Wear*

Eq. (14-13): 

Eq. (14-12): *r*1 = (169.33*/*2) sin 20° = 28.95 mm

*r*2 = (846.66*/*2) sin 20° = 144.76 mm

Eq. (6-68): *SC*= [2.76(262) – 70](106) = 653.12 MPa







For 108 cycles (revolutions of the pinion), the power based on wear is 38.63 kW.

Rating power (pinion controls):



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**14-18** See Prob. 14-16 solution for equation numbers.

Given: *φ*= 20°, *n* = 1145 rev/min, *m* = 6 mm, *F* = 75 mm, *NP*= 16 milled teeth,

*NG*= 30*T*, *Sut*= 900 MPa, *HB*= 260, *nd*= 3, *YP*= 0*.*296, and *YG*= 0*.*359*.*

*Pinion bending*







*kf* = 1.66 (See Ex. 14-2)





*r/d* = *rf /t* = 1*.*8*/*12 = 0*.*15, *Kt*= 1*.*68, *q* = 0.86, *Kf* = 1.58



Eq. (14-8): 

Eq. (13-36): 

*Wear*: Pinion and gear

Eq. (14-12): *r*1 = (96*/*2) sin 20° = 16*.*42 mm

*r*2 = (180*/*2) sin 20° = 30*.*78 mm

Eq. (14-13): 

Eq. (6-79): *SC*= 6*.*89[0*.*4(260) – 10] = 647*.*7 MPa



Eq. (14-14): 



Eq. (13-36): 

Thus, wear controls the gearset power rating; *H* = 20.0 kW*. Ans.*

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**14-19**  *NP*= 17 teeth, *NG*= 51 teeth





Eq. (14-4*b*): *Kv*= (6.1 + 4.22)*/*6.1 = 1*.*692



Table 14-2: *YP*= 0*.*303, *YG*= 0*.*410

Eq. (14-7): 

Eq. (13-35): 

Based on yielding in bending, the power is 50.42 kW.

**(a)** Pinion fatigue

*Bending*

Eq. (2-21): *Sut* = 0.5 *HB* = 0.5(232) = 800 MPa

Eq. (6-10): 

Eq. (6-18): 

Table 13-1: 

Eq. (14-3): 

Eq. (*b*), Sec. 14-1: 

Eq. (6-24): 

Eq. (6-19): 

*kc*= *kd*= *ke*= 1

Account for one-way bending with *kf* = 1.66. (See Ex. 14-2.)

Eq. (6-17): 

For stress concentration, find the radius of the root fillet (See Ex. 14-2).



Fig. A-15-6: 

Estimate *D/d* =∞ by setting *D/d* = 3, *Kt*= 1*.*68*.*

Fig. 6-26: *q* = 0.86

Eq. (6-32): 





**

**(b)** Pinion fatigue

*Wear*

Eq. (14-13): 

Eq. (14-12): 





Eq. (6-79): 

In terms of gear notation

*σC*= [2.76(232) – 70]106 = 570.32 MPa

We will introduce the design factor of *nd*= 2 and because it is a contact stress apply it to the load*Wt*by dividingby. [See unnumbered equation immediately

after Eq. (6-80)].



Solve Eq. (14-14) for *Wt*:



For 108 cycles (turns of pinion), the allowable power is 4.95 kW.

**(c)** Gear fatigue due to bending and wear

*Bending*

Eq. (14-3): 

Eq. (*b*), Sec. 14-1: 

Eq. (6-24): 

Eq. (6-19): 

*kc*= *kd*= *ke*= 1

*kf* = 1.66. (See Ex. 14-2.)

Eq. (6-17): 



Approximate *D/d* =∞ by setting *D/d* = 3 for Fig. A-15-6; *Kt*= 1*.*80*.*

Fig. 6-26: *q* = 0.82

Eq. (6-32): 





**

The gear is thus stronger than the pinion in bending.

*Wear*

Since the material of the pinion and the gear are the same, and the contactstresses are the same, the allowable power transmission of both is the same. Thus,*H*all = 4.95 kW for 108 revolutions of each. As yet, we have no way to establish *SC*for108*/*3 revolutions.

**(d)**

Pinion bending: *H*1 = 50.42 kW

Pinion wear: *H*2 = 4.95 kW

Gear bending: *H*3 = 28.55 kW

Gear wear: *H*4 = 4.95 kW

Power rating of the gear set is thus

*H*rated = min(50.42, 4.95, 28.55, 4.95) = 4.95 kW *Ans.*

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**14-20** Given: *H* = 3.73 kW, *nP* = 300 rev/min, *φ* = 20o, *HB* = 200, *Qv*= 6,*L* = 108 cycles with *R* = 90 percent.

**(a)** Let *F* = 50.8 mm, *dP*=*NP*/*P* = 16*/*0.236 = 67.73 mm, *dG*= 48*/*60.236= 203.2 mm

Eq. (13-34): 

Eq. (13-35): 

Assuming uniform loading, *Ko* = 1. *Qv*= 6,

Eq. (14-28): *B* = 0.25(12 −*Qv*) = 0.25(12 − 6)2/3 = 0.8255

*A* = 50 + 56(1 −*B*) = 50 + 56(1 − 0.8255) = 59.77

Eq. (14-27): 

Table 14-2: 

From Eq. (*a*), Sec. 14-10, with *F* = 50.8 mm



Eq. (14-31): *Cmc*= 1

Eq. (14-32): 

Eq.(4-33) assuming gears are centered between bearings: *Cpm*= 1

Fig. 14-11: *Cma*= 0.16

Eq. (14-35): *Ce*= 1

Eq. (14-30): 

Eq. (14-40) assuming constant gear thickness: *KB*= 1

For the pinion,*N* = 108 cycles, and from Fig. 14-14: 

Fig. 14-6: 

Table 14-10, for *R* = 0.90: *YZ*= 0*.*85

Figs. 14-17 and 14-18: *Y0*= *Cf*= 1

Eq. (14-22): *mG*= *NG/NP*= 48*/*16 = 3

Eq. (14-23): 

Table 14-8: 

*Strength*: Grade 1 steel with *HBP*= *HBG*= 200

Fig. 14-2: (*St*)*P*= 0.533(200) + 88.3 = 194.9 MPa

Fig. 14-5: (*Sc*)*P*= 2.22(200) + 200 = 644 MPa

Fig. 14-15: (*ZN*)*P*= 1*.*4488(108)–0*.*023 = 0*.*948

Sec. 14-12: *HBP/HBG*= 1 ∴*CH*= 1

***Pinion tooth bending***

Eq. (14-15):



Eq. (14-17):



Eq. (14-41): 

***Pinion tooth wear***

Eq. (14-16):

Eq. (14-18): *Ans.*

Eq. (14-42):



**(b)** Only check wear in U.S. customary unit. From Eq. (14-16) in part (a): 

*Sc ZN* / (*KTKR*) = 93 500 (0.948)/[1(0.85)] = 104 280 psi



(69.89 mm)

*p* = π/6 = 0.5235 in, 3*p* = 1.57 in and 5*p* = 2.62. Fails to satisfy requirements since both are less than 2.75 in.

Try *P* = 5 teeth/in. 3*p* = 3π/5 = 1.885 in and 5*p* = 5π/5 = 3.142 in.

*dP* = 16/5 = 3.2 in, *V* = π (3.2) 300/12 = 251.3 ft/min





Keep *Km* = 1.2207 and check later. *KR* = 0.85, *I* = 0.1205, *Cp* = 2300 , (*Sc*)*P* =

93 500 psi, (*ZN*)*P* = 0.948, and *ScZN* / (*KTKR*) = 104.28(103) psi.





(56.388mm)

Use *F* = 2.5 in (63.50 mm) *Ans*.

Rechecking *Km*.



*Cma*= 0.17







*F* = 2.5 in Acceptable. *Ans*.

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**14-21** *dP*= 2*.*5(20) = 50 mm, *dG*= 2*.*5(36) = 90 mm



With no specific information given to indicate otherwise, assume

*KB = Ko*=*Yθ* = *ZR* = 1

Eq. (14-28): *Qv*= 6, *B* = 0*.*25(12 – 6)2*/*3 = 0*.*8255

*A* = 50 + 56(1 – 0*.*8255) = 59*.*77

Eq. (14-27): 

Table 14-2: *YP*= 0*.*322, *YG*= 0*.*3775

Similar to Eq. (*a*) of Sec. 14-10 but for SI units:







 



Fig. 14-14: (*YN*)*P*= 1*.*3558(108)–0*.*0178 = 0*.*977

(*YN*)*G*= 1*.*3558(108*/*1*.*8)–0*.*0178 = 0*.*987

Fig. 14-6: (*YJ*)*P*= 0*.*33, (*YJ*)*G*= 0*.*38

Eq. (14-38): *YZ*= 0*.*658 – 0*.*0759 ln(1 – 0*.*95) = 0*.*885

Eq. (14-23): 

Table 14-8: 

*Strength* Grade 1 steel, *HBP*= *HBG*= 200

Fig. 14-2: (*St*)*P*= (*St*)*G*= 0*.*533(200) + 88*.*3 = 194*.*9 MPa

Fig. 14-5: (*Sc*)*P*= (*Sc*)*G*= 2*.*22(200) + 200 = 644 MPa

Fig. 14-15: (*ZN*)*P*= 1*.*4488(108)–0*.*023 = 0*.*948



Fig. 14-12: 

*Pinion tooth bending*

Eq. (14-15): 



Eq. (14-41) for SI: 

*Gear tooth bending*



*Pinion tooth wear*

Eq. (14-16): 



Eq. (14-42) for SI:

*Gear tooth wear* 



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**14-22**  *Pt = Pn* cos *ψ* = 0.236cos 30o = 0.205 teeth/mm



Eq. (13-34): 

Eq. (13-35): 

Eq. (14-28): *B* = 0.25(12 −*Qv*) = 0.25(12 − 6)2/3 = 0.8255

*A* = 50 + 56(1 −*B*) = 50 + 56(1 − 0.8255) = 59.77

Eq. (14-27): 

From Prob. 14-20:



The pressure angle is:







Eq. (14-25):





Eq. (14-21): 

Eq. (14-23): 

Fig. 14-7: 

Fig. 14-8: Correctionis0.94. Thus, *JP* = 0.45(0.94) = 0.423.

Eq. (14-31): *Cmc* = 1

Eq. (14-32: 

Eq. (14-33) assuming gearsare centered between bearings: *Cpm* = 1

Fig. 14-11: *Cma* = 0.16

Eq. (14-35): *Ce* = 1

Eq. (14-30): 

*Pinion tooth bending*

Eq. (14-15): 

Eq. (14-41): 

*Pinion tooth wear*

Eq. (14-16): 

(Eq. 14-43): 

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**14-23** Given: *R* = 0*.*99 at 108 cycles, *HB*= 232 through-hardening Grade 1, core and case, bothgears. *NP*= 17*T*, *NG*= 51*T*,

Table 14-2: *YP*= 0*.*303, *YG*= 0*.*4103

Fig. 14-6: *JP*= 0*.*292, *JG*= 0*.*396

*dP*= *NP / P* = 17 / 0.236 = 71.96 mm, *dG*= 51 / 0.236 = 215.9 mm.

*Pinion bending*

From Fig. 14-2:



Fig. 14-14: *YN*= 1*.*6831(108)–0*.*0323 = 0*.*928



 

















Eq. (14-15): 



*Pinion wear*

Fig. 14-15: *ZN*= 2*.*466*N*–0*.*056 = 2*.*466(108)–0*.*056 = 0*.*879



Eq. (14-23): 

Fig. 14-5: 





Eq. (14-16): 



The pinion controls, therefore *H*rated = 5.6 kW *Ans.*

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**14-24** *l* = 2.25/ *Pd*, *x* = 3*Y* / 2*Pd*









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**14-25** *YP*= 0*.*331, *YG*= 0*.*422, *JP*= 0*.*345, *JG*= 0*.*410, *Ko*= 1*.*25*.* The service conditionsare adequately described by *Ko.* Set *SF*= *SH*= 1*.*(Use P = 4 teeth/in or 0.157 teeth/mm and Face width 82.55 mm (3.25 in))

*dP* = 22 / 0.157 = 139.7 mm

*dG* = 60 / 0.157 = 381 mm



*Pinion bending*



Eq. (14-17): 





















Eq. (14-15): 



*Gear bending* By similar reasoning, 

*Pinion wear*



















*Gear wear*

Similarly, 

*Rating*



Note differing capacities. Can these be equalized?

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**14-26** From Prob. 14-25:



*Pinion bending*: The factor of safety, based on load and stress, is

 *Ans.*

*Gear bending* based on load and stress

 *Ans.*

*Pinion wear*

based on load: 

based on stress:  *Ans.*

*Gear wear*

based on load: 

based on stress:  *Ans.*

Factors of safety are used to assess the relative threat of loss of function 3.14, 3.86, 1.06,1.18 where the threat is from pinion wear. By comparison, the AGMA safety factors

(*SF*)*P*, (*SF*)*G*, (*SH*)*P*, (*SH*)*G*

are

3*.*14, 3*.*86, 1*.*03, 1*.*09 or 3*.*15, 3*.*86, 1*.*061*/*2, 1*.*181*/*2

and the threat is again from pinion wear. Depending on the magnitude of the numbers,using *SF*and *SH*as defined by AGMA, does not *necessarily* lead to the same conclusionconcerning threat. Therefore be cautious.

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**14-27** Solution summary from Prob. 14-25: *n* = 1145 rev/min, *Ko*= 1*.*25, Grade 1 materials,*NP*= 22*T*, *NG*= 60*T*, *mG*= 2*.*727*, YP*= 0*.*331*,YG*= 0*.*422, *JP*= 0*.*345,*JG*= 0*.*410, *Pd*= 0.157*T* /mm, *F* = 82.55 mm*, Qv*= 6, (*Nc*)*P*= 3(109), *R* = 0*.*99, *KH*= 1*.*240, *Y0*= 1, *KB*= 1,

*dP*= 139.7 mm ,*dG*= 381 mm,*V* = 8.37 m/s,*Kv*= 1*.*534, (*Ks*)*P*= (*Ks*)*G*= 1, (*YN*)*P*= 0*.*832,(*YN*)*G*= 0*.*859, *YZ*= 1

Pinion *HB*: 250 core, 390 case

Gear *HB*: 250 core, 390 case

*Bending*



*Wear*















*Rated power*

*H*rated = min(117.07, 143.90, 78.38, 87.73) = 78.38 kW *Ans.*

Prob. 14-25:

*H*rated = min(117.07, 143.90, 39.33, 44.01) =39.33 kW

The rated power approximately doubled.

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**14-28** The gear and the pinion are 9310 grade 1, carburized and case-hardened to obtain Brinell285 core and Brinell 580–600 case.

Table 14-3: 

Modification of *St*by (*YN*)*P*= 0*.*832 produces



Similarly for (*YN*)*G*= 0*.*859





From Table 14-8, Also, from Table 14-6:



Modification of *Sc*by *YN*produces



and



*Rating*

*H*rated = min(170.09, 211.12, 92.73, 103.10) = 92.73 kW *Ans.*

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**14-29** Grade 2,9310 carburized and case-hardened to 285 core and 580 case in Prob. 14-28.

*Summary:*

Table 14-3: 



and it follows that



From Table 14-8, Also, from Table 14-6:







Consequently,



*Rating*

*H*rated = min(201.42, 249.91, 179.04, 199.18) = 179.04 kW*. Ans.*

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**14-30** Given: *n* = 1145 rev/min, *Ko*= 1*.*25,*NP*= 22*T, NG*= 60*T, mG*= 2*.*727, *dP*= 69.85 mm,*dG*= 190.5 mm,*YP*= 0*.*331*,YG*= 0*.*422*, JP*= 0*.*335, *JG*= 0*.*405, *P* = 0.315*T* /mm,*F* = 41.28 mm, *HB*= 250, case and core, both gears. *Cm*= 1, *F/dP*= 0*.*0591,*ZR*= 0*.*0419*, Cpm*= 1, *Cma*= 0*.*152, *Ce*= 1, *KH*= 1*.*1942, *Y0*= 1,*KB*= 1,*Ks*= 1,*V* = 824 m/s, (*YN*)*P*= 0*.*8318, (*YN*)*G*= 0*.*859,*YZ*= 1,*ZI* = 0*.*117 58



and it follows that



For wear



*Rating*

*H*rated = min(16.39, 20.44, 5.66, 6.34) = 5.66 kW

In Prob. 14-25, *H*rated = 39.54. Thus,



The transmitted load rating is



In Prob. 14-25



Thus

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**14-31** *SP*= *SH*= 1, *Pd*= 0.157, *JP*= 0*.*345, *JG*= 0*.*410, *Ko*= 1*.*25

*Bending*

Table 14-4: 









*Wear*

Table 14-8: 

Table 14-7: 



*Rating*

*H*rated = min(57.14, 70.42, 48.27, 48.27) = 48.27 kW *Ans.*

Notice that the balance between bending and wear power is improved due to CI’s morefavorable *Sc/St*ratio. Also note that the life is 107 pinion revolutions which is (1*/*300) of3(109). Longer life goals require power de-rating.

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**14-32** From Table A-24*a*, *Eav*= 81.3 GPa

For *φ*= 14*.*5° and *HB*= 156



For *φ*= 20°



The first two calculations were approximately 4 percent higher.

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**14-33** Programs will vary.

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**14-34 **

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Table 14-6: 

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Pinion bending is controlling.

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**14-35** 



Table 14-3: *St*= 380 MPa













Table 14-6: *Sc*= 1240 MPa



















Pinion bending controlling

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**14-36** (*YN*)*P*= 0*.*928, (*YN*)*G*= 0*.*962 (See Prob. 14-34)

Table 14-3: *St*= 448 MPa













Table 14-6: *Sc*= 1551 MPa

















The bending of the pinion is the controlling factor.

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**14-37** *P* = 0.079 teeth/mm, *d* = 203.20 mm, *N* = *dP* = 203.20 (0.079) = 16 teeth



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*FB*= 3336 N



*n* = 2400 / 2 = 1200 rev/min



We will obtain all of the needed factors, roughly in the order presented in the textbook.

Fig. 14-2: *St*= 0.703(300) + 113 = 323.90 MPa

Fig. 14-5: *Sc*= 2.41(300) + 237 = 960.00 MPa

Fig. 14-6: *YJ* = 0.27

Eq. (14-23): 

Table 14-8: 

Assume a typical quality number of 6.

Eq. (14-28): 



Eq. (14-27): 

To estimate a size factor, get the Lewis Form Factor from Table 14-2,*Y* = 0.296.

From Eq. (*a*), Sec. 14-10,



The load distribution factor is applicable for straddle-mounted gears, which is not the case here since the gear is mounted outboard of the bearings. Lacking anything better, we will use the load distribution factor as a rough estimate.

Eq. (14-31): *Cmc* = 1 (uncrowned teeth)

Eq. (14-32): 

Eq. (14-33): *Cpm* = 1.1

Fig. 14-11: *C­ma* = 0.23 (commercial enclosed gear unit)

Eq. (14-35): *Ce­*= 1

Eq. (14-30): 

For the stress-cycle factors, we need the desired number of load cycles.

*N* = 15 000 h (1200 rev/min)(60 min/h) = 1.1 (109) rev

Fig. 14-14: *YN* = 0.9

Fig. 14-15: *ZN* = 0.8

Eq. 14-38: 

With no specific information given to indicate otherwise, assume *Ko*=*KB*= *Y0*= *ZR*=1

*Tooth bending*

Eq. (14-15): 



Eq. (14-41): 



*Tooth wear*

Eq. (14-16): 



Since gear *B* is a pinion, *CH* is not used in Eq. (14-42) (see Fig. 14-18), where





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**14-38** *m* = 18.75 mm/tooth, *d* = 300 mm

*N* = *d/m* = 300 / 18.75 = 16 teeth



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*FB*= 22.81 kN



*n* = 1800 / 2 = 900 rev/min



We will obtain all of the needed factors, roughly in the order presented in the textbook.

Fig. 14-2: *St*= 0.703(300) + 113 = 324 MPa

Fig. 14-5: *Sc*= 2.41(300) + 237 = 960 MPa

Fig. 14-6: *J* = *YJ* = 0.27

Eq. (14-23): 

Table 14-8: 

Assume a typical quality number of 6.

Eq. (14-28): 



Eq. (14-27): 

To estimate a size factor, get the Lewis Form Factor from Table 14-2,*Y* = 0.296.

Similar to Eq. (*a*) of Sec. 14-10 but for SI units:





Convert the diameter and facewidth to inches for use in the load-distribution factor equations. *d* = 300/25.4 = 11.81 in, *F* = 236/25.4 = 9.29 in

Eq. (14-31): *Cmc* = 1 (uncrowned teeth)

Eq. (14-32): 

Eq. (14-33): *Cpm* = 1.1

Fig. 14-11: *C­ma* = 0.27 (commercial enclosed gear unit)

Eq. (14-35): *Ce­*= 1

Eq. (14-30): 

For the stress-cycle factors, we need the desired number of load cycles.

*N* = 12 000 h (900 rev/min)(60 min/h) = 6.48 (108) rev

Fig. 14-14: *YN*  = 0.9

Fig. 14-15: *ZN* = 0.85

Eq. 14-38: 

With no specific information given to indicate otherwise, assume *Ko* = *KB*= *KT* = *ZR* =1.

*Tooth bending*

Eq. (14-15): 



Eq. (14-41): 



*Tooth wear*

Eq. (14-16): 



Since gear *B* is a pinion, *CH* is not used in Eq. (14-42) (see Fig. 14-18), where





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**14-39** From the solution to Prob. 13-46, *n* = 191 rev/min, *Wt* = 1600 N, *d* = 125 mm,   
*N* = 15 teeth, *m* = 8.33 mm/tooth.





We will obtain all of the needed factors, roughly in the order presented in the textbook.

Table 14-3: *St*= 65 kpsi = 448 MPa

Table 14-6: *Sc*= 225 kpsi = 1550 MPa

Fig. 14-6: *J* = *YJ* = 0.25

Eq. (14-23): 

Table 14-8: 

Assume a typical quality number of 6.

Eq. (14-28): 



Eq. (14-27): 

To estimate a size factor, get the Lewis Form Factor from Table 14-2,*Y* = 0.290.

Similar to Eq. (*a*) of Sec. 14-10 but for SI units:





Convert the diameter and facewidth to inches for use in the load-distribution factor equations. *d* = 125/25.4 = 4.92 in, *F* = 105/25.4 = 4.13 in

Eq. (14-31): *Cmc* = 1 (uncrowned teeth)

Eq. (14-32): 

Eq. (14-33): *Cpm* = 1

Fig. 14-11: *C­ma* = 0.32 (open gearing)

Eq. (14-35): *Ce­*= 1

Eq. (14-30): 

For the stress-cycle factors, we need the desired number of load cycles.

*N* = 12 000 h (191 rev/min)(60 min/h) = 1.4 (108) rev

Fig. 14-14: *YN*  = 0.95

Fig. 14-15: *ZN* = 0.88

Eq. 14-38: 

With no specific information given to indicate otherwise, assume *Ko* = *KB*= *KT* = *ZR* =1.

*Tooth bending*

Eq. (14-15): 



Since gear is a pinion, *CH* is not used in Eq. (14-42), where





*Tooth wear*

Eq. (14-16): 



Eq. (14-42): 



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**14-40** From the solution to Prob. 13-47, *n* = 2(70) = 140 rev/min, *Wt* = 801.48 N, *d* = 127 mm   
*N* = 15 teeth, *P* = 0.118 (3 teeth/in).





We will obtain all of the needed factors, roughly in the order presented in the textbook.

Table 14-3: *St*= 448 MPa

Table 14-6: *Sc*= 1551 MPa

Fig. 14-6: *YJ* = 0.25

Eq. (14-23): 

Table 14-8: 

Assume a typical quality number of 6.

Eq. (14-28): 



Eq. (14-27): 

To estimate a size factor, get the Lewis Form Factor from Table 14-2,*Y* = 0.290.

From Eq. (*a*), Sec. 14-10,



Eq. (14-31): *Cmc* = 1 (uncrowned teeth)

Eq. (14-32): 

Eq. (14-33): *Cpm* = 1

Fig. 14-11: *C­ma* = 0.32 (Open gearing)

Eq. (14-35): *Ce­*= 1

Eq. (14-30): 

For the stress-cycle factors, we need the desired number of load cycles.

*N* = 14 000 h (140 rev/min)(60 min/h) = 1.2 (108) rev

Fig. 14-14: *YN*  = 0.95

Fig. 14-15: *ZN* = 0.88

Eq. 14-38: 

With no specific information given to indicate otherwise, assume *Ko* = *KB*= *Y0* = *ZR*=1.

*Tooth bending*

Eq. (14-15): 



Eq. (14-41): 



*Tooth wear*

Eq. (14-16):

Since gear *B* is a pinion, *CH* is not used in Eq. (14-42), where





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