**Chapter 15**

**15-1** Given: Uncrowned, through-hardened 300 Brinell core and case, Grade 1, *NC*= 109 rev of pinion at *R* = 0*.*999, *NP*= 20 teeth, *NG*= 60 teeth, *Qv*= 6, *Pd*= 0.236 teeth/mm (6 teeth/in), normal pressure angle = 20°,shaft angle =90°, *np*= 900 rev/min, *b* = 31.75 mm, *SF*= *SH*= 1, *KA*= 1.

Fig. 15-7: *YJP* = 0.249, *YJG* = 0.216

*Mesh dP*= 20*/*0.236 = 84.66 mm, *dG*= 60*/*0.236 = 254 mm

Eq. (15-7): *vt*= **(0.0846)(900*/*60) = 3.99 m/s

Eq. (15-6): *B* = 0*.*25(12 – 6)2*/*3 = 0*.*8255

*A* = 50 + 56(1 – 0*.*8255) = 59*.*77

Eq. (15-5): 

Eq. (15-8): *vt*,max = [59*.*77 + (6 – 3)]2 /200 = 19.7 m/s

Since 3.99 *<*19.7, *Kv*= 1*.*374 is valid. The size factor for bending is:

Eq. (15-10): *Yx*= 0*.*4867 + 0.00839(4.23) = 0*.*5222

For one gear straddle-mounted, the load-distribution factor is:

Eq. (15-11): *KHB*= 1*.*10 + 5.6(10-6)(31.75)2 = 1*.*106

For *KL*, the more conservative “critical” equation will be used here. Though it is acceptable, according to AGMA, to use the less conservative “general” equation for general applications.

Eq. (15-15): (*YNT*)*P*= 1*.*6831(109)–0*.*0323 = 0*.*862

(*YNT*)*G*= 1*.*6831(109*/*3)–0*.*0323 = 0*.*893

Eq. (15-14): (*ZNT*)*P*= 3*.*4822(109)–0*.*0602 = 1

(*ZNT*)*G*= 3*.*4822(109*/*3)–0*.*0602 = 1*.*069

Eq. (15-19): *YZ*= 0*.*50 – 0*.*25 log(1 – 0*.*999) = 1*.*25 (or Table 15-3)



*Bending*

Fig. 15-13: 

Eq. (15-4): 

Eq. (15-3): 



Eq. (15-4): 

**

The gear controls the bending rating.

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

**15-2** Refer to Prob. 15-1 for the gearset specifications.

*Wear*

Fig. 15-12: 

For the pinion, *ZW*= 1*.* From Prob. 15-1, *ZZ*= 1*.*118. Thus, from Eq. (15-2):



For the gear, from Eq. (15-16),



From Prob. 15-1, (*ZNT*)*G*= 1*.*0685*.* Equation (15-2) thus gives



For steel: 

Eq. (15-9): 

Fig. 15-6: *ZI*= 0*.*083

Eq. (15-12): *Zxc*= 2

Eq. (15-1): 



The pinion controls wear: *H* = 8.21 kW *Ans.*

The power rating of the mesh, considering the power ratings found in Prob. 15-1, is

*H* = min(12.15, 10.92, 8.21, 9.39) = 8.21 kW *Ans.*

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

**15-3** AGMA 2003-B97 does not fully address cast iron gears. However, approximate comparisonscan be useful. This problem is similar to Prob. 15-1, but not identical. We will organizethe method. A follow-up could consist of completing Probs. 15-1 and 15-2 withidentical pinions, and cast iron gears.

Given: Uncrowned, straight teeth, *Pd*= 0.236 teeth/mm (6 teeth/in), *NP*= 30 teeth, *NG*= 60 teeth, ASTM30 cast iron, material Grade 1, shaft angle 90°, *F* = 31.75 mm, *nP*= 900 rev/min, *φn*= 20°,one gear straddle-mounted, *KA*= 1,*SF*= 2,

Fig. 15-7: *YJP*= 0*.*268, *YJG*= 0*.*228

*Mesh dP*= 30*/*0.236 = 127 mm, *dG*= 60*/*0.236= 254 mm

*vt* = *π* (0.127)(900 / 60) = 5.98 m/s

Set *NL*= 107 cycles for the pinion. For *R* = 0*.*99,

Table 15-7: 

Table 15-5: 

Eq. (15-4): 

The velocity factor *Kv*represents stress augmentation due to mislocation of tooth profilesalong the pitch surface and the resulting “falling” of teeth into engagement. Equation (5-67)shows that the induced bending moment in a cantilever (tooth) varies directly withof thetooth material. If only the material varies (cast iron vs. steel) in the same geometry, *I* is thesame. From the Lewis equation of Section 14-1,



We expect the ratio *σ*CI*/σ*steel to be



In the case of ASTM class 30, from Table A-24(*a*)

(*E*CI)av= (13 + 16*.*2)*/*2 = 14*.*7 kpsi

Then, 

Our modeling is rough, but it convinces us that (*Kv*)CI*<*(*Kv*)steel, but we are not sure ofthe value of (*Kv*)CI*.* We will use *Kv*for steel as a basis for a conservative rating.

Eq. (15-6): *B* = 0*.*25(12 – 6)2*/*3 = 0*.*8255

*A* = 50 + 56(1 – 0*.*8255) = 59*.*77

Eq. (15-5): 

*Pinion bending* (*σ*F)*P*= *swt*= 15.52 MPa

From Prob. 15-1, *YB*= 1, *KHB*= 1*.*106, *Yx*= 0*.*5222

Eq. (15-3): 





*Gear bending*



The gear controls in bending fatigue.*H* = 3.38 kW *Ans.*

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

**15-4** Continuing Prob. 15-3,

Table 15-5: 



Eq. (15-1): 

Fig. 15-6: *ZI* = 0*.*086

From Probs. 15-1 and 15-2: *Zx*= 0*.*593 75, *Yx*= 0*.*5222, *KHB*= 1*.*106, *Zxc*= 2

From Table 14-8: 

Thus, 

*Ans.*

*Rating*

Side note: Based on results of Probs. 15-3 and 15-4,

*H* = min(3.97, 3.38, 2.42, 2.42) = 2.42 kW

The mesh is weakest in wear fatigue.

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

**15-5** Uncrowned, through-hardened to 180 Brinell (core and case), Grade 1, 109 rev of pinion at*R* = 0*.*999, *NP*= *z*1 = 22 teeth, *NG*= *z*2 = 24 teeth, *Qv*= 5, *met*= 4 mm,

*φn*= 20°,shaft angle90°, *n*1 = 1800 rev/min, *SF*= 1, *F* = *b* = 25 mm, *Ko*= *KA*= *KT*= *Kθ*= 1 and 

Fig. 15-7: *JP*= *YJ*1 = 0*.*23, *JG*= *YJ*2 = 0*.*205

*Mesh dP*= *de*1 = *mz*1 = 4(22) = 88 mm, *dG*= *met z*2 = 4(24) = 96 mm

Eq. (15-7): *vet*= 5*.*236(10–5)(88)(1800) = 8*.*29 m/s

Eq. (15-6): *B* = 0*.*25(12 – 5)2*/*3 = 0*.*9148

*A* = 50 + 56(1 – 0*.*9148) = 54*.*77

Eq. (15-5): 

Eq. (15-10): *Ks*= *Yx*= 0*.*4867 + 0*.*008 339(4) = 0*.*520

Eq. (15-11): with *Kmb*= 1.25(neither member straddle-mounted),

*Km*= *KHβ*= 1.25 + 5*.*6(10–6)(252) = 1*.*2535

From Fig. 15-8,



Eq. (15-12): *Cxc*= *Zxc*= 2 (uncrowned)

Eq. (15-19): *KR*= *YZ*= 0*.*50 – 0*.*25 log (1 – 0*.*999) = 1*.*25



From Fig. 15-10, *CH*= *Zw*= 1

Eq. (15-9): *Zx*= 0*.*004 92(25) + 0*.*4375 = 0*.*560

*Wear of Pinion*

Fig. 15-12: *σH* lim = 2*.*35*HB*+ 162*.*89

= 2*.*35(180) + 162*.*89 = 585*.*9 MPa

Fig. 15-6: *I* = *ZI*= 0*.*066

Eq. (15-2): 



Eq. (15-1): 

The constant 1000 expresses *Wt*in kN.



Eq. (13-36): 

*Wear of Gear*

*σH* lim = 585*.*9 MPa





Thus in wear, the pinion controls the power rating; *H* = 3.92 kW *Ans.*

We will rate the gear set after solving Prob. 15-6.

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

**15-6** Refer to Prob. 15-5 for terms not defined below.

*Bending of Pinion*



Fig. 15-13: *σF* lim = 0*.*30*HB*+ 14*.*48

= 0*.*30(180) + 14*.*48 = 68*.*5 MPa

Eq. (15-13): *Kx*= *Yβ*= 1

From Prob. 15-5: *YZ*= 1*.*25, *vet*= 8*.*29 m/s,



Eq. (5-4): 

Eq. (5-3): 





*Bending of Gear*









Rating of mesh is

*H*rating = min(8.29, 7.42, 3.92, 3.95) = 3.92 kW *Ans.*

with pinion wear controlling.

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

**15-7**

**(a) **



All terms cancel except for *sat*, *KL*, and *J*,

(*sat*)*P*(*KL*)*P JP*= (*sat*)*G*(*KL*)*G JG*

From which



where *β*= –0*.*0178 or *β*= –0*.*0323 as appropriate. This equation is the same as

Eq. (14-44). *Ans.*

**(b)** In bending

 (1)

In wear



Squaring and solving for *Wt*gives

 (2)

Equating the right-hand sides of Eqs. (1) and (2) and canceling terms, and recognizingthat and *PddP*= *NP*,we obtain



For equal *Wt*in bending and wear



So we get



**(c)**



Substituting in the right-hand equality gives



Denominators cancel, leaving

(*sac*)*P*(*CL*)*P*= (*sac*)*G*(*CL*)*GCH*

Solving for (*sac*)*P*gives,



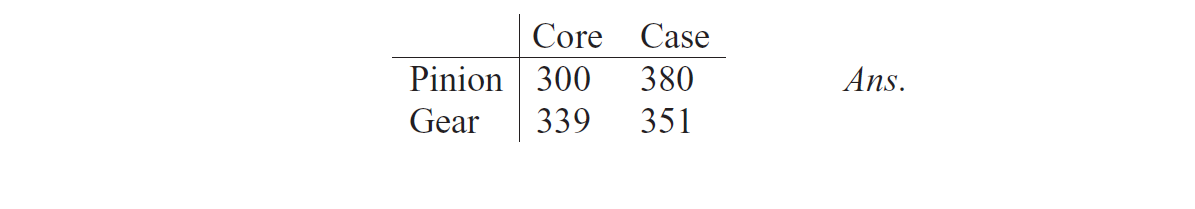
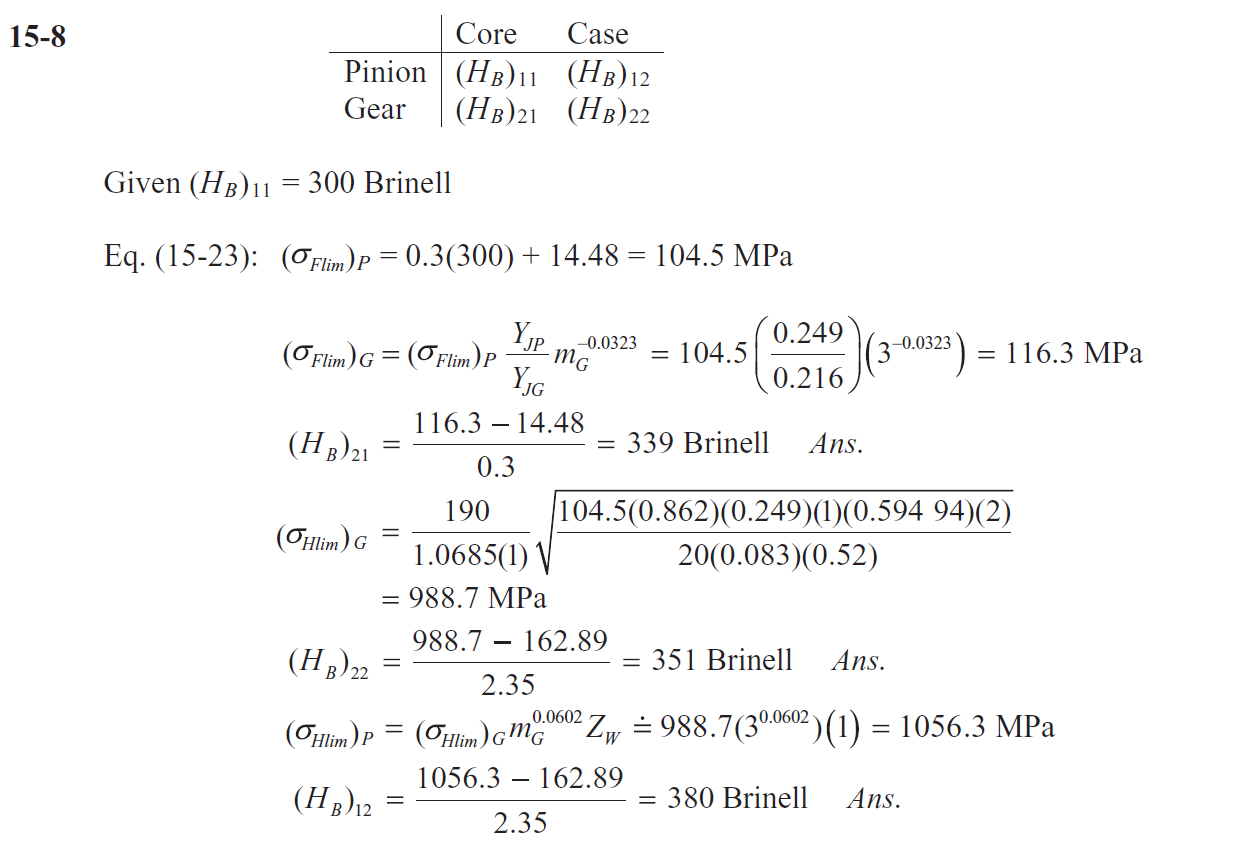
From Eq. (15-14), 

Thus,

 *Ans.*

This equation is the transpose of Eq. (14-45).

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



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

**15-9**

*Pinion core*









*Gear core*







*Pinion case*







*Gear case*







All four transmitted loads are essentially equal, so the equations developed within Prob. 15-7 are effective. *Ans.*

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

**15-10** The catalog rating is 3.88 kW at 1200 rev/min for a straight bevel gearset. Also given:*NP*= 20 teeth, *NG*= 40 teeth, *φn*= 20°, *b* = 18 mm, *YJP*= 0*.*241, *YJG*= 0*.*201,

*Pd*= 10 teeth/in (0.394 teeth /mm) , through-hardened to 300 Brinell-General Industrial Service, and

*Qv*= 5 uncrowned.

*Mesh*

**

**

**

Eq. (15-6): *B* = 0*.*25(12 – 5)2*/*3 = 0*.*9148

*A* = 50 + 56(1 – 0*.*9148) = 54*.*77

Eq. (15-5): 

Eq. (15-10): *Yx*= 0*.*4867 + 0*.*008339(2.54) = 0*.*508

Eq. (15-11): *KHB*= 1*.*25 + 0*.*0036(0*.*71)2 = 1*.*252, where *Kmb*= 1*.*25

Eq. (15-15): (*YNT*)*P*= 1.3558(3∙106)–0*.*0178 = 1.040

(*YNT*)*G*= 1.3558(3∙106*/*2)–0*.*0178 = 1.053

Eq. (15-14): (*ZNT*)*P*= 3*.*4822(3∙106)–0*.*0602 = 1*.*419

(*ZNT*)*G*= 3*.*4822(3∙106*/*2)–0*.*0602 = 1*.*479

Eq. (15-19): *Yz*= 0*.*50 – 0*.*25 log(1 – 0*.*99) = 1*.*0



*Bending*

Pinion:

Eq. (15-23): 

Eq. (15-4): 

Eq. (15-3): 



Gear: 

Eq. (15-4): 

Eq. (15-3): 



*Wear*

Pinion:



Eq. (15-22): 

Eq. (15-2): 

Eq. (15-1): 





Gear:







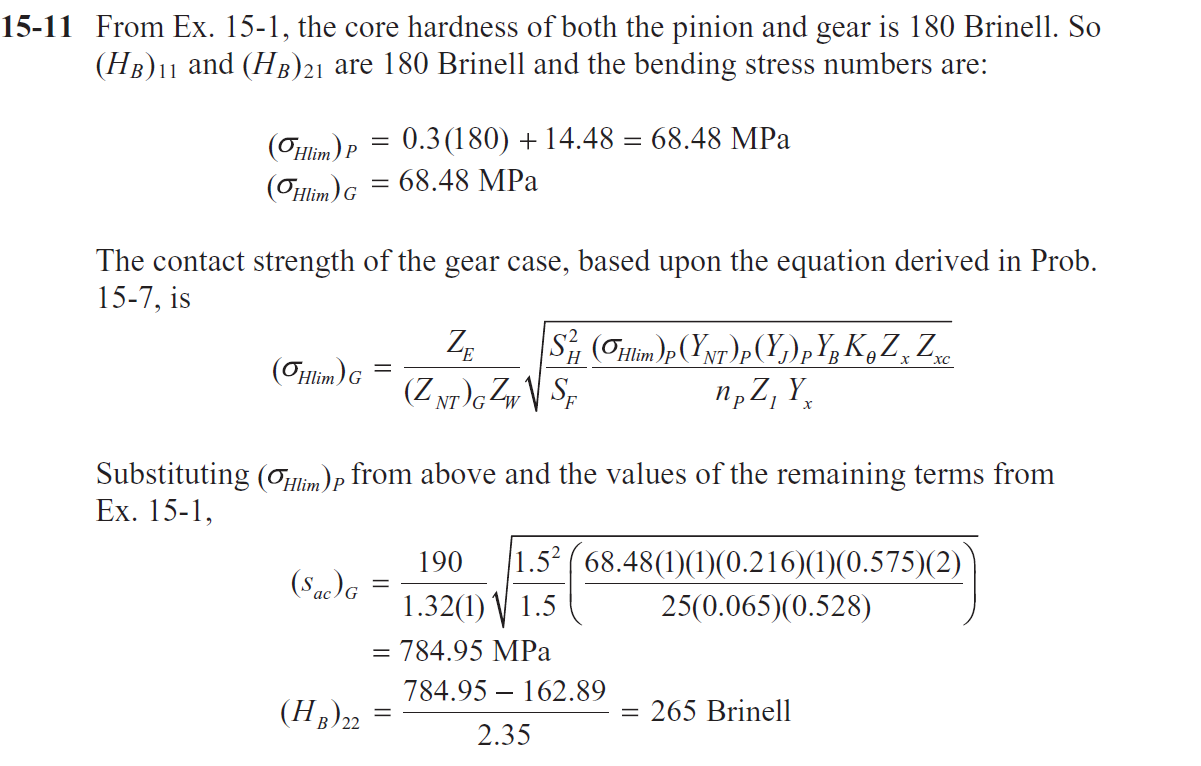


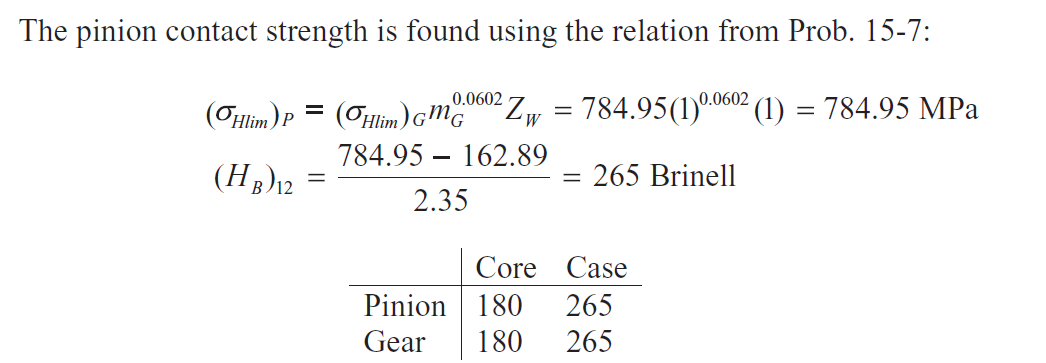
Rating:

*H* = min(4.25, 3.59, 5.08, 5.51) = 3.59 kW

Pinion wear controls the power rating. The catalog rating of 3.59 kW is slightly higher than predicted here, but seems reasonable. *Ans.*

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

****

**

*Realization of hardnesses*

The response of students to this part of the question would be a function of the extentto which heat-treatment procedures were covered in their materials and manufacturingprerequisites, and how quantitative it was. The most important thing is to have the studentthink about it.

The instructor can comment in class when students’ curiosity is heightened. Optionsthat will surface may include:

(a)Select a through-hardening steel which will meet or exceed core hardness in the hot-rolledcondition, then heat-treating to gain the additional 86 points of Brinell hardnessby bath-quenching, then tempering, then generating the teeth in the blank.

(b) Flame or induction hardening are possibilities.

(c) The hardness goal for the case is sufficiently modest that carburizing and case hardeningmay be too costly. In this case the material selection will be different.

(d)The initial step in a nitriding process brings the core hardness to 33–38 RockwellC-scale (about 300–350 Brinell), which is too much.

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

**15-12** Computer programs will vary.

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

**15-13** A design program would ask the user to make the a priori decisions, as indicated inSec. 15-5 of the text. The decision set can be organized as follows:

A priori decisions:

• Function: *H*, *Ko*, rpm, *mG*, temp., *NL*, *R*

• Design factor: *nd*(*SF*= *nd*, )

• Tooth system: Involute, Straight Teeth, Crowning, *φn*

• Straddling: *Kmb*

• Tooth count: *NP*(*NG*= *mGNP*)

Design decisions:

• Pitch and Face: *Pd*, *F*

• Quality number: *Qv*

• Pinion hardness: (*HB*)1, (*HB*)3

• Gear hardness: (*HB*)2, (*HB*)4

First, gather all of the equations one needs, then arrange them before coding. Find the required hardnesses, express the consequences of thechosen hardnesses, and allow for revisions as appropriate.

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | **Pinion Bending** | **Gear Bending** | **Pinion Wear** | **Gear Wear** |
| Load-induced  stress (Allowable  stress) |  |  |  | *s*22 = *s*12 |
| Tabulated  strength |  |  |  |  |
| Associated  hardness |  |  |  |  |
| Chosen  hardness | (*HB*)11 | (*HB*)21 | (*HB*)12 | (*HB*)22 |
| Newtabulated  strength |  |  |  |  |
| Factor of  safety |  |  |  |  |

*Note: *

**15-14** *NW*= 1, *NG*= 56, *Pt*= 0.315 teeth/mm (8 teeth/in), *d* = 38.1 mm, *Ho*= 746 W, *φn*= 20°, *ta*= 21.1°C,*Ka*= 1*.*25, *nd*= 1, *Fe*= 50.8 mm,  *A* = 0.548 m2

**(a)** *mG*= *NG/NW*= 56, *dG*= *NG/Pt*= 56*/*0.315 = 177.8 mm

*px*= */*0.315 = 9.97 mm, *C* = 38.1 + 177.8 = 215.90 mm

Eq. (15-39): *a* = *px/*= 9.97*/*= 3.18 mm

Eq. (15-40): *b*= 0*.*3683*px*= 3.67 mm

Eq. (15-41): *ht*= 0*.*6866*px*= 6.848 mm

Eq. (15-42): *do*= 38.1 + 2(3.18) = 44.45 mm

Eq. (15-43): *dr*= 76.2 – 2(3.67) = 68.86 mm

Eq. (15-44): *Dt*= 177.80 + 2(3.18) = 184.15 mm

Eq. (15-45): *Dr*= 177.80 – 2(3.67) = 170.46 mm

Eq. (15-46): *c* = 3.67 – 3.18 = 0.5 mm

Eq. (15-47): 



Eq. (13-27): 

Eq. (13-28): 



Eq. (15-62): 

**(b)**

Eq. (15-38): 

Eq. (15-54):



Eq. (15-58): 

Eq. (15-57): 

**

**(c)**

Eq. (15-33): *Cs*= 1190 – 477 log 7*.*0 = 787

Eq. (15-36): 

Eq. (15-37): 

Eq. (15-38): (*Wt*)all = 787(177.8/25.4)0*.*8(50.8/25.4)(0*.*767)(0*.*312) = 7948 N

Since the mesh will survive at least 25 000 h.

Eq. (15-61): 

Eq. (15-63): 



The mesh is sufficient *Ans.*







The stress is high. At the rated horsepower,



**(d)**

Eq. (15-52): *A*min = 43*.*2(2.64)(215.9)1*.*7 = 1060027 mm2*<*1096772 mm2

Eq. (15-49): *H*loss = (1 – 0*.*7563)(1626.28) = 396 W

Assuming a fan exists on the worm shaft,

Eq. (15-50): 

Eq. (15-51): 

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

**15-15 to 15-22**

Problem statement values of 25 hp, 1125 rev/min, *mG*= 10, *Ka*= 1*.*25, *nd*= 1*.*1, *φn*= 20°, *ta*= 70°F are not referenced in the table. The first four parameters listed in the table were selected as design decisions.

|  |  |  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- | --- | --- |
|  | 15-15 | 15-16 | 15-17 | 15-18 | 15-19 | 15-20 | 15-21 | 15-22 |
| *px* | 1.75 | 1.75 | 1.75 | 1.75 | 1.75 | 1.75 | 1.75 | 1.75 |
| *dW* | 3.60 | 3.60 | 3.60 | 3.60 | 3.60 | 4.10 | 3.60 | 3.60 |
| *FG* | 2.40 | 1.68 | 1.43 | 1.69 | 2.40 | 2.25 | 2.4 | 2.4 |
| *A* | 2000 | 2000 | 2000 | 2000 | 2000 | 2000 | 2500 | 2600 |
|  |  |  |  |  |  |  | FAN | FAN |
| *HW* | 38.2 | 38.2 | 38.2 | 38.2 | 38.2 | 38.0 | 41.2 | 41.2 |
| *HG* | 36.2 | 36.2 | 36.2 | 36.2 | 36.2 | 36.1 | 37.7 | 37.7 |
| *Hf* | 1.87 | 1.47 | 1.97 | 1.97 | 1.97 | 1.85 | 3.59 | 3.59 |
| *NW* | 3 | 3 | 3 | 3 | 3 | 3 | 3 | 3 |
| *NG* | 30 | 30 | 30 | 30 | 30 | 30 | 30 | 30 |
| *KW* |  |  |  | 125 | 80 | 50 | 115 | 185 |
| *Cs* | 607 | 854 | 1000 |  |  |  |  |  |
| *Cm* | 0.759 | 0.759 | 0.759 |  |  |  |  |  |
| *Cv* | 0.236 | 0.236 | 0.236 |  |  |  |  |  |
| *VG* | 492 | 492 | 492 | 492 | 492 | 563 | 492 | 492 |
|  | 2430 | 2430 | 2430 | 2430 | 2430 | 2120 | 2524 | 2524 |
|  | 1189 | 1189 | 1189 | 1189 | 1189 | 1038 | 1284 | 1284 |
| *f* | 0.0193 | 0.0193 | 0.0193 | 0.0193 | 0.0193 | 0.0183 | 0.034 | 0.034 |
| *e* | 0.948 | 0.948 | 0.948 | 0.948 | 0.948 | 0.951 | 0.913 | 0.913 |
| (*Pt*)*G* | 1.795 | 1.795 | 1.795 | 1.795 | 1.795 | 1.571 | 1.795 | 1.795 |
| *Pn* | 1.979 | 1.979 | 1.979 | 1.979 | 1.979 | 1.732 | 1.979 | 1.979 |
| *C*-to-*C* | 10.156 | 10.156 | 10.156 | 10.156 | 10.156 | 11.6 | 10.156 | 10.156 |
| *ts* | 177 | 177 | 177 | 177 | 177 | 171 | 179.6 | 179.6 |
| *L* | 5.25 | 5.25 | 5.25 | 5.25 | 5.25 | 6.0 | 5.25 | 5.25 |
| *λ* | 24.9 | 24.9 | 24.9 | 24.9 | 24.9 | 24.98 | 24.9 | 24.9 |
| *σG* | 5103 | 7290 | 8565 | 7247 | 5103 | 4158 | 5301 | 5301 |
| *dG* | 16.71 | 16.71 | 16.71 | 16.71 | 16.71 | 19.099 | 16.7 | 16.71 |