# Problem 1

$$d_P = 17/8 = 2.125 \text{ in}$$

$$d_G = \frac{N_2}{N_3} d_P = \frac{1120}{544}(2.125) = 4.375 \text{ in}$$

$$N_G = Pd_G = 8(4.375) = 35 \text{ teeth} \qquad \textit{Ans.}$$

$$C = (2.125 + 4.375)/2 = 3.25 \text{ in} \qquad \textit{Ans.}$$

# Problem 2

The smallest pinion that will mesh with a gear ratio of $m_G$ = 2.5, from Eq. (13-11) is

$$\begin{aligned} N_P &\geq \frac{2k}{(1+2m)\sin^2\phi}\left(m + \sqrt{m^2 + (1+2m)\sin^2\phi}\right) \\ &\geq \frac{2(1)}{[1+2(2.5)]\sin^2 20^\circ}\left\{2.5 + \sqrt{2.5^2 + [1+2(2.5)]\sin^2 20^\circ}\right\} \\ &\geq 14.64 \quad \rightarrow \quad 15 \text{ teeth} \qquad \textit{Ans.} \end{aligned}$$

The largest gear-tooth count possible to mesh with this pinion, from Eq. (13-12) is

$$\begin{aligned} N_G &\leq \frac{N_P^2 \sin^2\phi - 4k^2}{4k - 2N_P \sin^2\phi} \\ &\leq \frac{15^2 \sin^2 20^\circ - 4(1)^2}{4(1) - 2(15)\sin^2 20^\circ} \\ &\leq 45.49 \quad \rightarrow \quad 45 \text{ teeth} \qquad \textit{Ans.} \end{aligned}$$

## Problem 3

Applying Eq. (13-30), $e = (N_2 / N_3)(N_4 / N_5) = 45$. For an exact ratio, we will choose to factor the train value into integers, such that

$$N_2 / N_3 = 9 \qquad (1)$$
$$N_4 / N_5 = 5 \qquad (2)$$

Assuming a constant diametral pitch in both stages, the geometry condition to satisfy the in-line requirement of the compound reverted configuration is

$$N_2 + N_3 = N_4 + N_5 \quad (3)$$

With three equations and four unknowns, one free choice is available. It is necessary that all of the unknowns be integers. We will use a normalized approach to find the minimum free choice to guarantee integers; that is, set the smallest gear of the largest stage to unity, thus $N_3 = 1$. From (1), $N_2 = 9$. From (3),

$$N_2 + N_3 = 9 + 1 = 10 = N_4 + N_5$$

Substituting $N_4 = 5\, N_5$ from (2) gives

$$10 = 5\, N_5 + N_5 = 6\, N_5$$
$$N_5 = 10 / 6 = 5 / 3$$

To eliminate this fraction, we need to multiply the original free choice by a multiple of 3. In addition, the smallest gear needs to have sufficient teeth to avoid interference. From Eq. (13-11) with $k = 1$, $\phi = 20°$, and $m = 9$, the minimum number of teeth on the pinion to avoid interference is 17. Therefore, the smallest multiple of 3 greater than 17 is 18. Setting $N_3 = 18$ and repeating the solution of equations (1), (2), and (3) yields

$$N_2 = 162 \text{ teeth}$$
$$N_3 = 18 \text{ teeth}$$
$$N_4 = 150 \text{ teeth}$$
$$N_5 = 30 \text{ teeth} \qquad \textit{Ans.}$$

# Problem 4

$d_P$ = 16/6 = 2.667 in, $d_G$ = 48/6 = 8 in

$$V = \frac{\pi(2.667)(300)}{12} = 209.4 \text{ ft/min}$$

$$W^t = \frac{33\,000(5)}{209.4} = 787.8 \text{ lbf}$$

Assuming uniform loading, $K_o$ = 1.

Eq. (14-28): $Q_v = 6, \quad B = 0.25(12 - 6)^{2/3} = 0.8255$

$$A = 50 + 56(1 - 0.8255) = 59.77$$

Eq. (14-27): $$K_v = \left(\frac{59.77 + \sqrt{209.4}}{59.77}\right)^{0.8255} = 1.196$$

Table 14-2: $Y_P = 0.296, \quad Y_G = 0.4056$

From Eq. (*a*), Sec. 14-10 with $F$ = 2 in

$$(K_s)_P = 1.192\left(\frac{2\sqrt{0.296}}{6}\right)^{0.0535} = 1.088$$

$$(K_s)_G = 1.192\left(\frac{2\sqrt{0.4056}}{6}\right)^{0.0535} = 1.097$$

From Eq. (14-30) with $C_{mc}$ = 1

$$C_{pf} = \frac{2}{10(2.667)} - 0.0375 + 0.0125(2) = 0.0625$$

$$C_{pm} = 1, \quad C_{ma} = 0.093 \quad \text{(Fig. 14 - 11)}, \quad C_e = 1$$

$$K_m = 1 + 1[0.0625(1) + 0.093(1)] = 1.156$$

Assuming constant thickness of the gears → $K_B$ = 1

$m_G = N_G/N_P$ = 48/16 = 3

With $N$ (pinion) = $10^8$ cycles and $N$ (gear) = $10^8/3$, Fig. 14-14 provides the relations:

$$(Y_N)_P = 1.3558(10^8)^{-0.0178} = 0.977$$

$$(Y_N)_G = 1.3558(10^8 / 3)^{-0.0178} = 0.996$$

Fig. 14-6: $J_P = 0.27, \quad J_G \text{ B } 0.38$

Table 14-10: $K_R = 0.85$

$K_T = C_f = 1$

Eq. (14-23): $$I = \frac{\cos 20^\circ \sin 20^\circ}{2(1)}\left(\frac{3}{3+1}\right) = 0.1205$$

Table 14-8: $$C_p = 2300\sqrt{\text{psi}}$$

*Strength*: Grade 1 steel with $H_{BP} = H_{BG} = 200$

Fig. 14-2: $(S_t)_P = (S_t)_G = 77.3(200) + 12\ 800 = 28\ 260$ psi

Fig. 14-5: $(S_c)_P = (S_c)_G = 322(200) + 29\ 100 = 93\ 500$ psi

Fig. 14-15: $(Z_N)_P = 1.4488(10^8)^{-0.023} = 0.948$

$(Z_N)_G = 1.4488(10^8/3)^{-0.023} = 0.973$

Sec. 14-12: $H_{BP}/H_{BG} = 1 \qquad \therefore\ C_H = 1$

*Pinion tooth bending*

Eq. (14-15): $$(\sigma)_P = W^t K_o K_v K_s \frac{P_d}{F}\frac{K_m K_B}{J}$$

$$= 787.8(1)(1.196)(1.088)\left(\frac{6}{2}\right)\left[\frac{(1.156)(1)}{0.27}\right]$$

$$= 13\,170 \text{ psi} \qquad Ans.$$

Eq. (14-41): $$(S_F)_P = \left[\frac{S_t Y_N\ /\ (K_T K_R)}{\sigma}\right]$$

$$= \frac{28\ 260(0.977)\ /\ [(1)(0.85)]}{13\,170} = 2.47 \qquad Ans.$$

*Gear tooth bending*

Eq. (14-15): $$(\sigma)_G = 787.8(1)(1.196)(1.097)\left(\frac{6}{2}\right)\left[\frac{(1.156)(1)}{0.38}\right] = 9433 \text{ psi} \qquad Ans.$$

Eq. (14-41): $$(S_F)_G = \frac{28\ 260(0.996)\ /\ [(1)(0.85)]}{9433} = 3.51 \qquad Ans.$$

*Pinion tooth wear*

Eq. (14-16): $$(\sigma_c)_P = C_p\left(W^t K_o K_v K_s \frac{K_m}{d_P F}\frac{C_f}{I}\right)_P^{1/2}$$

$$= 2300\left[787.8(1)(1.196)(1.088)\left(\frac{1.156}{2.667(2)}\right)\left(\frac{1}{0.1205}\right)\right]^{1/2}$$
$$= 98\ 760 \text{ psi} \quad \textit{Ans.}$$

Eq. (14-42):

$$(S_H)_P = \left[\frac{S_c Z_N/(K_T K_R)}{\sigma_c}\right]_P = \left\{\frac{93\ 500(0.948)/[(1)(0.85)]}{98\ 760}\right\} = 1.06 \quad \textit{Ans.}$$

*Gear tooth wear*

$$(\sigma_c)_G = \left[\frac{(K_s)_G}{(K_s)_P}\right]^{1/2}(\sigma_c)_P = \left(\frac{1.097}{1.088}\right)^{1/2}(98\ 760) = 99\ 170 \text{ psi} \quad \textit{Ans.}$$

$$(S_H)_G = \frac{93\ 500(0.973)(1)/[(1)(0.85)]}{99\ 170} = 1.08 \quad \textit{Ans.}$$

The hardness of the pinion and the gear should be increased.

Fig. 14-5: $(S_c)_P = (S_c)_G = 322(200) + 29\,100 = 93\,500$ psi

Note that in general the pinion and the gear will have different life as the pinion will experience more number of revolutions than the gear.

Fig. 14-15: $(Z_N)_P = 1.4488(10^8)^{-0.023} = 0.948$

$(Z_N)_G = 1.4488(10^8/3)^{-0.023} = 0.973$

Could also use Z_N =0.948 for both the pinion and gear.

Sec. 14-12: $H_{BP}/H_{BG} = 1 \quad \therefore\ C_H = 1$ for gear

*Pinion tooth bending*

Eq. (14-15): $(\sigma)_P = W^t K_o K_v K_s \dfrac{P_d}{F}\dfrac{K_m K_B}{J}$ =787.8(1.6)(1.196)(6)/2/0.27 = 16750psi

Eq. (14-41): $(S_F)_P = \left[\dfrac{S_t Y_N\ /\ (K_T K_R)}{\sigma}\right]$=(28260)(0.977)/.85/16750=1.9392

*Gear tooth bending*

Eq. (14-15): $(\sigma)_G$=787.8(1.6)(1.196)(6)/2/0.38=11902psi

Eq. (14-41): $(S_F)_G$=(28260)(0.977)/.85/11902=2.7292

If $(Y_N)_G = 0.996$ is used instead, $(S_F)_G$=(28260)(0.996)/.85/11902=2.7822

*Pinion tooth wear*

Eq. (14-16): $(\sigma_c)_P = C_p\left(W^t K_o K_v K_s \dfrac{K_m}{d_P F}\dfrac{C_f}{I}\right)_P^{1/2}$

=2300(787.8(1.196)(1.6)/2.667/2/0.1205)^(1/2)=111390psi

Eq. (14-42):

$$(S_H)_P = \frac{\frac{(93500)(0.948)}{0.85}}{111390} = 0.9362$$

*Gear tooth wear*

$$(S_H)_G = \frac{\frac{(93500)(0.973)}{0.85}}{111390} = 0.9609$$

If $(Z_N)_G = 0.948$ is used instead, $(S_H)_G = 0.9362$

2. $$F = \frac{S_F W^t P K_o K_v K_s K_m K_B K_T K_R}{S_t \cdot J \cdot Y_N} = F(P) \qquad (*)$$

<u>Pinion:</u>

$S_t = 40$ ksi,

$S_F = 2$,

$d_P = \frac{N_P}{P} = \frac{23}{P}$

$$V_t = \frac{\pi n_P d_P}{12} = \frac{\pi (1000) \frac{23}{P}}{12} = V_t(P)$$

$$W^t = \frac{33000 H}{V_t} = \frac{33000 (125)}{V_t} = W^t(P)$$

$K_o = 1$

$B = 0.25(12 - Q_v)^{2/3} = 0.25(12-9)^{2/3} = 0.52$

$A = 50 + 56(1-B) = 76.8788$

$$K_v = \left(\frac{A + \sqrt{V_t}}{A}\right)^B = K_v(P)$$

assume $K_m = 1.6$

$K_B = 1$

$K_T = 1$

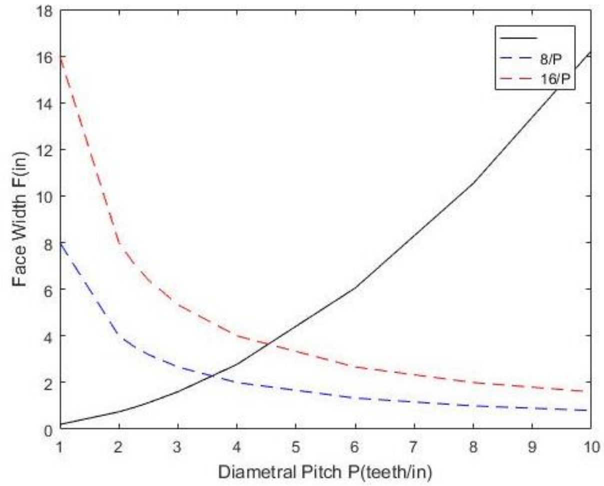
$K_R = 1.25 \quad (R = 99.9\%)$

$J_p = 0.355$

$(Y_N)_P = 3.517(10^5)^{-0.0817} = 1.3730$

plug all variables to eqn $(*)$

and plot $F(P)$ with $\frac{8}{P}$ and $\frac{16}{P}$.




From plot: select $P = 4$ teeth/in.

use $P=4$, calculate $F = 2.7811$ in.

use $F = 2.8$ in.

update $K_m = 1.62$ and iterate.

$P = 4$ teeth/in. $F = 3$ in. updated $K_m = 1.625$.

Gear.

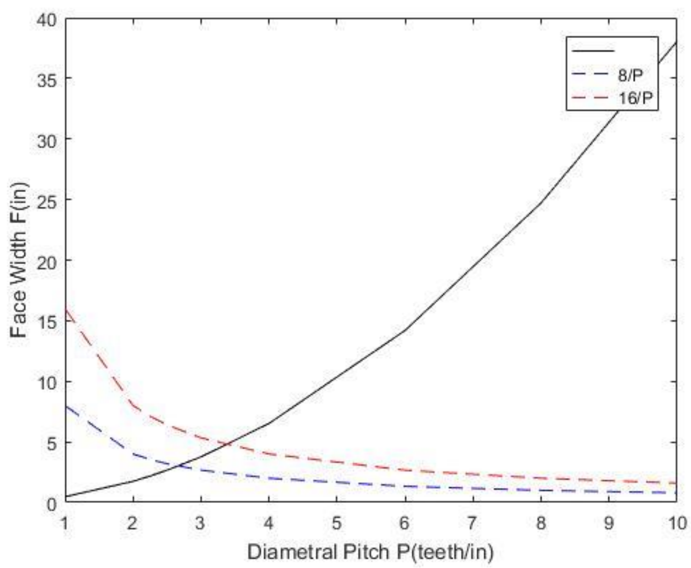
Different parameters:

$J_G = 0.41$.

$(Y_N)_G = 6.1514\,(10^5)^{-0.1192} = 1.5595$

$S_t = 13$ ksi, plug all variables to eqn (*)

based on plot $F(P)$, $\frac{8}{P}$, and $\frac{16}{P}$



select $P = 3$ teeth/in. $F = 3.7672$ in. use $F = 3.8$ in.
update $K_m = 1.645$ and iterate
$P = 3$ teeth/in. $F = 4.0$ in. updated $K_m = 1.65$

For the gearset, use $\boxed{P = 3 \text{ teeth/in.} \quad F = 4.0 \text{ in.}}$

$$\boxed{d_P = \frac{N_P}{P} = 7.67 \text{ in.}} \quad \boxed{d_G = \frac{N_G}{P} = 19 \text{ in.}}$$

realized F.S.s,

pinion:

$$(S_F)_P = \frac{(S_t)_P (Y_N)_P / (K_R K_T)}{\sigma_P} = \frac{(S_t)_P (Y_N)_P / ((K_R)(K_T))}{W^t K_o K_v K_s \frac{P}{F} \frac{K_m K_B}{J_P}}$$

$$= 4.8302 \quad > 2$$

$$(S_F)_G = \frac{(S_t)_G (Y_N)_G / (K_R K_T)}{W^t K_o K_v K_s \frac{P}{F} \frac{K_m K_B}{J_G}}$$

$= 2.0593 > 2$ satisfactory.

3.
$$F = \frac{S_H^2 C_P^2 W^t K_o K_v K_s K_m C_f K_R^2 K_T^2}{d_P \, I \cdot S_c^2 \cdot Z_N^2 \cdot C_H^2} \qquad (**)$$

refer to solutions from problem 2 for most of the parameters,

pinion:
$S_H = 2$.
$C_P = 2100 \sqrt{psi}$
$S_c = 165$ ksi,
$C_H = 1$

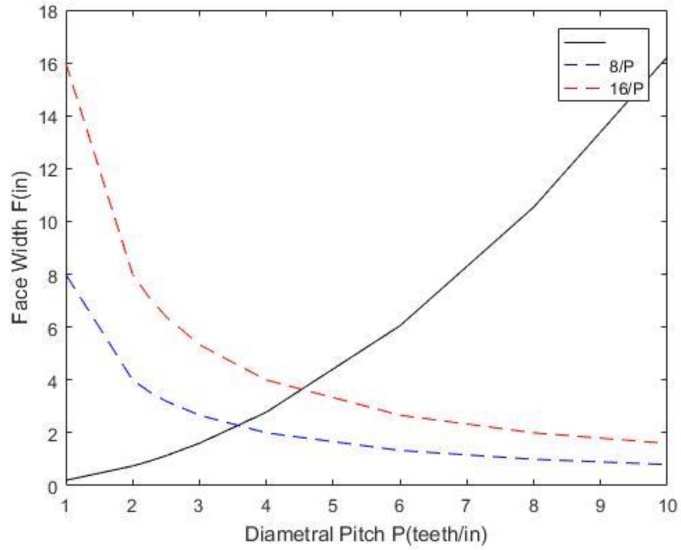
$C_f = 1$

$$I = \frac{\sin 20^\circ \cos 20^\circ}{2} \frac{\frac{57}{23}}{\frac{57}{23} + 1} = 0.1145$$

$$(Z_N)_P = (Z_N)_G = 2.466 (10^5)^{-0.056} = 1.2942$$

plug all variables to eqn (**) to obtain $F(P)$

plot $F(P)$ and $\frac{8}{P}$. $\frac{16}{P}$.




from plot, select $P = 3$ teeth/in. $F = 2.8749$ in.

use $F = 3.0$ in.

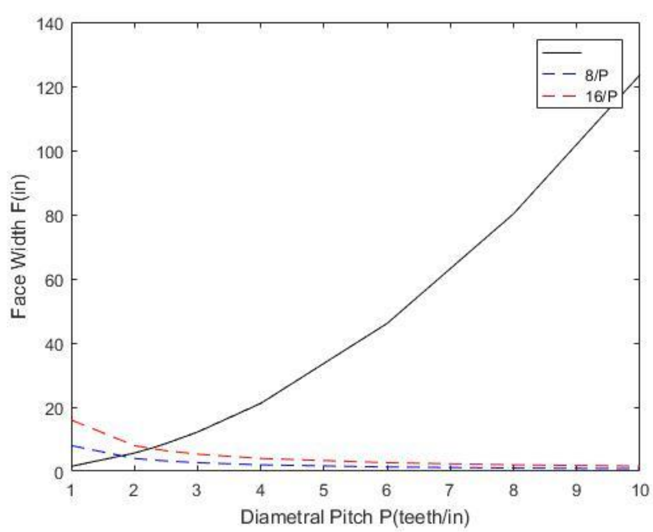
update $K_m = 1.625$ and iterate

$P = 3$ teeth/in. $F = 3.0$ in. updated $K_m = 1.625$

Gear: $S_c = 80$ ksi,

$C_H = 1$ since $A' = 0$ for $\frac{H_{BP}}{H_{BG}} = \frac{230}{200} = 1.15 < 1.2$

repeat the process and plot $F(P)$, $\frac{8}{P}$, $\frac{16}{P}$



from plot, select $P = 2$ teeth/in. , $F = 5.6648$ in.

use $F = 5.8$ in.

update $K_m = 1.695$ and iterate

select $P = 2$ teeth/in. $F = 6.0012$ in.

use $F = 7$ in. , $K_M = 1.725$

For the gearset = $\boxed{P = 2 \text{ teeth/in.} \quad F = 7 \text{ in.}}$ $K_m = 1.725$

$\boxed{d_P = \frac{N_P}{P} = 11.5 \text{ in.}}$ $\boxed{d_G = \frac{N_G}{2} = 28.5 \text{ in.}}$

realized F.S.s:

$$(S_H)_P = \frac{(S_c)_P Z_N C_H/(K_R K_T)}{(\sigma_c)_P} = \frac{(S_c)_P Z_N C_H/(K_R K_T)}{C_P \sqrt{W^t K_o K_v K_s \frac{K_m C_f}{d_P F I}}}$$

$= 4.4162 > 2$

$(S_H)_G = 2.1412 > 2$

**Additional Problems with Solutions:**

1. A spur gearset has 17 teeth on the pinion and 51 teeth on the gear. The pressure angle is $20^0$ and the overload factor $K_o = 1$. The diametral pitch is 6 teeth/in and the face width is 2in. The pinion speed is 1120rev/min and its cycle life is to be $10^8$ revolutions at a reliability R=0.99. The quality number is 5. The material is a through-hardened steel, grade 1, with Brinell hardnesses of 232 core and case of both gears. For a design factor of 2, rate the gearset for these conditions using the AGMA method.

Solutions:

Given: $R = 0.99$ at $10^8$ cycles, $H_B = 232$ through-hardening Grade 1, core and case, both gears. $N_P = 17T$, $N_G = 51T$,

Table 14-2: $Y_P = 0.303$, $Y_G = 0.4103$

Fig. 14-6: $J_P = 0.292$, $J_G = 0.396$

$d_P = N_P / P = 17 / 6 = 2.833$ in, $d_G = 51 / 6 = 8.500$ in.

*Pinion bending*

From Fig. 14-2:

$$\begin{aligned} {}_{0.99}(S_t)_{10^7} &= 77.3H_B + 12\,800 \\ &= 77.3(232) + 12\,800 = 30\,734 \text{ psi} \end{aligned}$$

Fig. 14-14: $Y_N = 1.6831(10^8)^{-0.0323} = 0.928$

$$V = \pi d_P n / 12 = \pi(2.833)(1120 / 12) = 830.7 \text{ ft/min}$$

$$K_T = K_R = 1, \quad S_F = 2, \quad S_H = \sqrt{2}$$

$$\sigma_{\text{all}} = \frac{30\ 734(0.928)}{2(1)(1)} = 14\ 261 \text{ psi}$$

$$Q_v = 5, \quad B = 0.25(12 - 5)^{2/3} = 0.9148$$

$$A = 50 + 56(1 - 0.9148) = 54.77$$

$$K_v = \left(\frac{54.77 + \sqrt{830.7}}{54.77}\right)^{0.9148} = 1.472$$

$$K_s = 1.192\left(\frac{2\sqrt{0.303}}{6}\right)^{0.0535} = 1.089 \Rightarrow \text{use } 1$$

$$K_m = C_{mf} = 1 + C_{mc}(C_{pf}C_{pm} + C_{ma}C_e)$$

$$C_{mc} = 1$$

$$C_{pf} = \frac{F}{10d} - 0.0375 + 0.0125F$$

$$= \frac{2}{10(2.833)} - 0.0375 + 0.0125(2) = 0.0581$$

$$C_{pm} = 1$$

$$C_{ma} = 0.127 + 0.0158(2) - 0.093(10^{-4})(2^2) = 0.1586$$

$$C_e = 1$$

$$K_m = 1 + 1[0.0581(1) + 0.1586(1)] = 1.217$$

$$K_B = 1$$

Eq. (14-15): $$W^t = \frac{FJ_P\sigma_{\text{all}}}{K_oK_vK_sP_dK_mK_B}$$

$$= \frac{2(0.292)(14\ 261)}{1(1.472)(1)(6)(1.217)(1)} = 775 \text{ lbf}$$

$$H = \frac{W^tV}{33\ 000} = \frac{775(830.7)}{33\ 000} = 19.5 \text{ hp}$$

*Pinion wear*

Fig. 14-15: $Z_N = 2.466N^{-0.056} = 2.466(10^8)^{-0.056} = 0.879$

$$m_G = 51/17 = 3$$

Eq. (14-23): $$I = \frac{\cos 20^\circ \sin 20^\circ}{2}\left(\frac{3}{3+1}\right) = 1.205, \quad C_H = 1$$

Fig. 14-5: $${}_{0.99}(S_c)_{10^7} = 322H_B + 29\,100$$

$$= 322(232) + 29\,100 = 103\,804 \text{ psi}$$

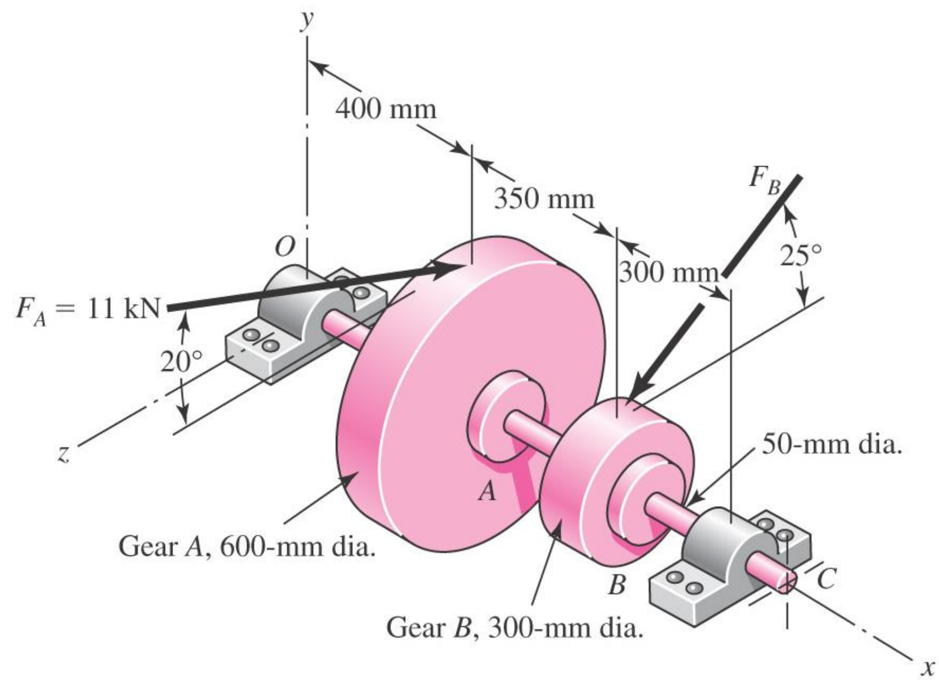
$$\sigma_{c,\text{all}} = \frac{103\,804(0.879)}{\sqrt{2}(1)(1)} = 64\,519 \text{ psi}$$

Eq. (14-16): $$W^t = \left(\frac{\sigma_{c,\text{all}}}{C_p}\right)^2 \frac{Fd_P I}{K_o K_v K_s K_m C_f}$$

$$= \left(\frac{64\,519}{2300}\right)^2 \left[\frac{2(2.833)(0.1205)}{1(1.472)(1)(1.2167)(1)}\right]$$
$$= 300 \text{ lbf}$$
$$H = \frac{W^t V}{33\,000} = \frac{300(830.7)}{33\,000} = 7.55 \text{ hp}$$

The pinion controls, therefore $H_{\text{rated}}$ = 7.55 hp *Ans.*

2. The countershaft in Problem 3-73, is part of a speed reducing compound gear train using $20^0$ spur gears. A gear on the input shaft drives gear A with a 2 to 1 speed reduction. Gear B drives a gear on the output shaft with a 5 to 1 speed reduction. The input shaft runs at 1800rev/min. All gears are to be of the same material. Since gear B is the smallest gear, transmitting the largest load, it will likely be critical, so a preliminary analysis is to be performed on it. Use a module of 18.75 mm/tooth, a face-width of 4 times the circular pitch, a Grade 2 steel through-hardened to a Brinell hardness of 300, and a desired life of 12kh with a 98 percent reliability. Determine factors of safety for bending and wear.




## Solutions:

$m$ = 18.75 mm/tooth, $d$ = 300 mm

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

$$F = b = 4p = 4(\pi m) = 4\pi(18.75) = 236 \text{ mm}$$

$$\sum M_x = 0 = 300(11)\cos 20^\circ - 150F_B \cos 25^\circ$$

$F_B$ = 22.81 kN

$$W^t = F_B \cos 25^\circ = 22.81 \cos 25^\circ = 20.67 \text{ kN}$$

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

$$V = \frac{\pi dn}{60} = \frac{\pi(0.300)(900)}{60} = 14.14 \text{ m/s}$$

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

Fig. 14-2: $S_t$ = 0.703(300) + 113 = 324 MPa

Fig. 14-5: $S_c$ = 2.41(300) + 237 = 960 MPa

Fig. 14-6: $J = Y_J$ = 0.27

Eq. (14-23): $$I = Z_I = \frac{\cos 20^\circ \sin 20^\circ}{2(1)}\left(\frac{5}{5+1}\right) = 0.134$$

Table 14-8: $$Z_E = 191\sqrt{\text{MPa}}$$

Assume a typical quality number of 6.

Eq. (14-28): $$B = 0.25(12 - Q_v)^{2/3} = 0.25(12 - 6)^{2/3} = 0.8255$$

$$A = 50 + 56(1 - B) = 50 + 56(1 - 0.8255) = 59.77$$

Eq. (14-27): $$K_v = \left(\frac{A + \sqrt{200V}}{A}\right)^B = \left(\frac{59.77 + \sqrt{200(14.14)}}{59.77}\right)^{0.8255} = 1.69$$

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:

$$K_s = \frac{1}{k_b} = 0.8433\left(mF\sqrt{Y}\right)^{0.0535}$$

$$K_s = 0.8433\left[18.75(236)\sqrt{0.296}\right]^{0.0535} = 1.28$$

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): $C_{mc} = 1$ (uncrowned teeth)

Eq. (14-32): $$C_{pf} = \frac{9.29}{10(11.81)} - 0.0375 + 0.0125(9.29) = 0.1573$$

Eq. (14-33): $C_{pm} = 1.1$

Fig. 14-11: $C_{ma} = 0.27$ (commercial enclosed gear unit)

Eq. (14-35): $C_e = 1$

Eq. (14-30): $$K_m = K_H = 1 + 1[0.1573(1.1) + 0.27(1)] = 1.44$$

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

$N = 12\,000$ h (900 rev/min)(60 min/h) = $6.48\,(10^8)$ rev

Fig. 14-14: $Y_N = 0.9$

Fig. 14-15: $Z_N = 0.85$

Eq. 14-38: $$K_R = 0.658 - 0.0759\ln(1 - R) = 0.658 - 0.0759\ln(1 - 0.98) = 0.955$$

With no specific information given to indicate otherwise, assume $K_o = K_B = K_T = Z_R = 1$.

*Tooth bending*

Eq. (14-15): $$\sigma = W^t K_o K_v K_s \frac{1}{bm_t}\frac{K_H K_B}{Y_J}$$

$$= 20\,670(1)(1.69)(1.28)\left[\frac{1}{236(18.75)}\right]\left[\frac{(1.44)(1)}{0.27}\right] = 53.9 \text{ MPa}$$

Eq. (14-41): $$S_F = \left[\frac{S_t Y_N \ / \ (K_T K_R)}{\sigma}\right]$$

$$= \frac{324(0.9) \ / \ [(1)(0.955)]}{53.9} = 5.66 \quad \textit{Ans.}$$

*Tooth wear*

Eq. (14-16): $$\sigma_c = Z_E \left(W^t K_o K_v K_s \frac{K_H}{d_{w1} b}\frac{Z_R}{Z_I}\right)^{1/2}$$

$$= 191\left[20\,670(1)(1.69)(1.28)\left(\frac{1.44}{300(236)}\right)\left(\frac{1}{0.134}\right)\right]^{1/2}$$
$$= 498 \text{ MPa}$$

Since gear *B* is a pinion, $C_H$ is not used in Eq. (14-42) (see p. 757), where

$$S_H = \frac{S_c Z_N \ / \ (K_T K_R)}{\sigma_c}$$

$$= \frac{960(0.85) \ / \ [(1)(0.955)]}{498} = 1.72 \quad \textit{Ans}$$