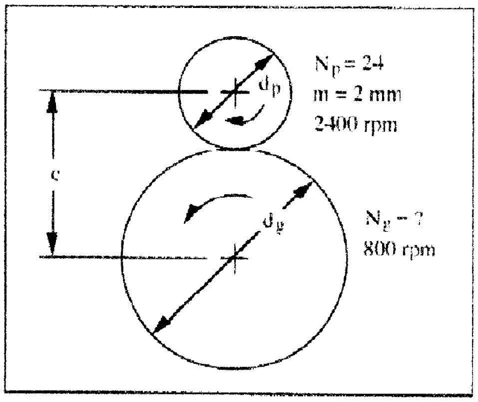
SOLUTION (15.12)

Known: A pinion with known module and number of teeth rotates at 2400 rpm and drives a gear at 800 rpm.

Find: Determine the number of teeth on the gear, circular pitch and theoretical center distance.

Schematic and Given Data:



Assumption: The spur gears mesh along their pitch circles.

Analysis:

1. For the 3:1 velocity ratio, $N_g = 24(3) = 72$ ■
2. $p = \pi m = 2\pi$ mm ■
3. $d = Nm$; Hence $d_p = 48$ mm, $d_g = 144$ mm
4. $c = \dfrac{48 + 144}{2} = 96$ mm ■

Comments:

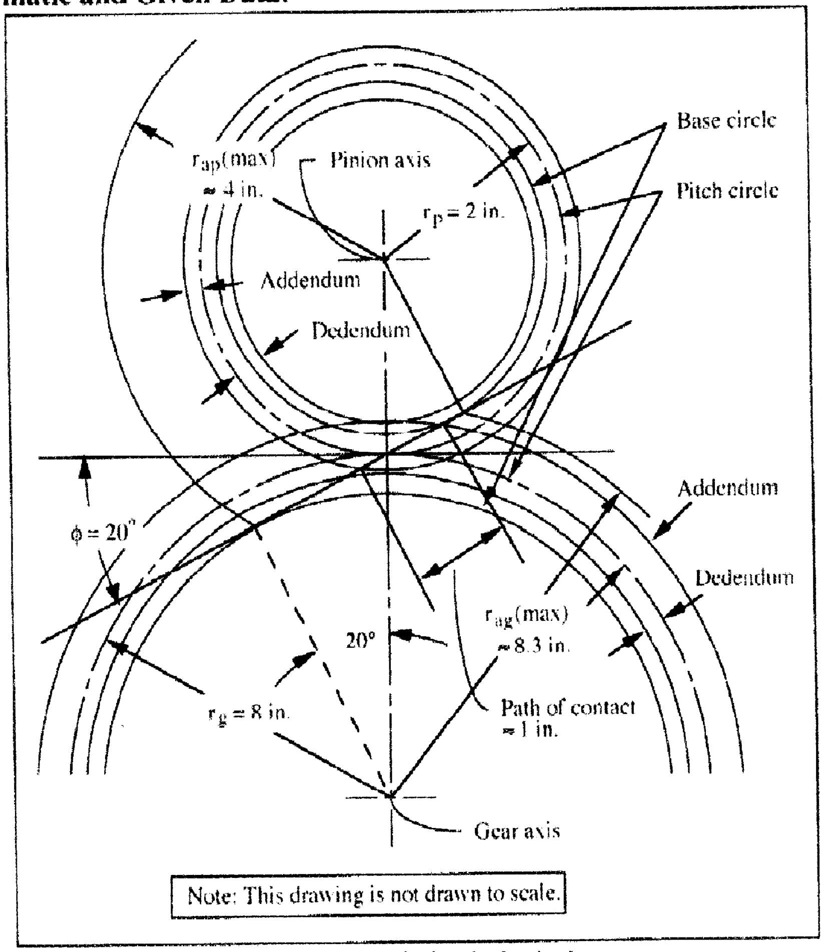
1. Similar to the diametral pitch, the module must be the same for a pair of meshing gears.
2. If the module were a higher value the pitch diameters of the gears and the theoretical center distance would have been higher (other parameter values remaining the same).

SOLUTION (15.22)

Known: A pair of standard spur gears of known pressure angle, center distance and velocity ratio is given. Number of teeth on pinion is specified.

Find: Compute the contact ratio using equations in Section 15.3 and compare with graphical results of Problem 15.21.

Schematic and Given Data:



Assumption: The gears mesh along their pitch circles.

Analysis:

1. From Eq. (15.11), $r_{bp} = r_p \cos\phi = 2.0 \cos 20^\circ = 1.879$ in.
   $r_{bg} = r_g \cos\phi = 8 \cos 20^\circ = 7.517$ in.

From Eq. (15.8),

$r_{ap}(max) = \sqrt{1.879^2 + 10.0^2 \sin^2 20^\circ} = 3.90$ in. ■

$r_{ag}(max) = \sqrt{7.517^2 + 10.0^2 \sin^2 20^\circ} = 8.26$ in. ■

(this agrees with graphical solution)

2. From Eq. (15.9), (with $r_{ap} = 2.2$ and $r_{ag} = 8.2$ from Problem 15.21)

$$CR = \frac{\sqrt{2.2^2 - 1.879^2} + \sqrt{8.2^2 - 7.517^2} - 10 \sin 20^\circ}{0.590}$$

CR = 1.69 (which is more accurate than the graphical solution.) ■

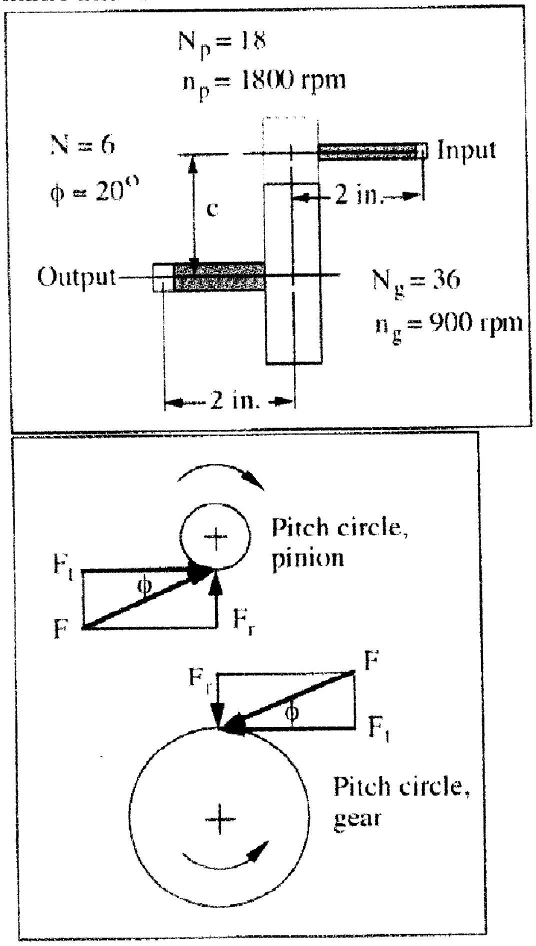
Comment: The contact ratio can be increased by choosing a greater number of teeth on the gears and/or increasing the diametral pitch.

SOLUTION (15.27)

Known: For a single stage speed reducer, gear geometry, overhang and the horsepower transmitted are specified.

Find: Estimate the forces on the pinion, gear and shafts.

Schematic and Given Data:



Assumptions:
1. The gears are spur gears.
2. The gears mesh along their pitch circles.
3. Friction losses in the gears and bearings can be neglected.
4. The shafts are parallel.
5. Bending deflection of each shaft is negligible.
6. The gears are rigidly connected to their shafts.
7. The weight of the gear on its shaft can be neglected.

Analysis:

1. Pitch diameter of the input pinion, $d_p = N/p = 18/6 = 3$ in.
   Pitch line velocity, $V = \pi dn/12 = \pi(3)1800/12 = 1413.7$ ft/min.
2. From Eq. (15.14), the power transmitted, $\dot{W} = (F_t V/33000)$

   $$F_t = \frac{(0.5)(33000)}{1413.7} = 11.67 \text{ lb}$$

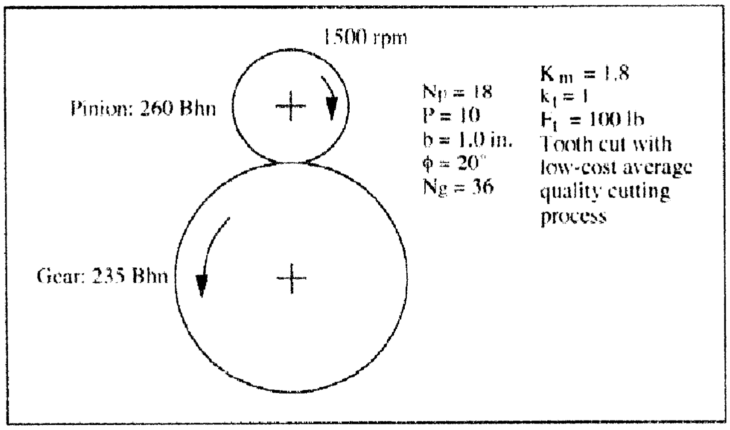
   From Eq. (15.12), $F_r = F_t \tan\phi = (11.67)\tan 20°$. Hence, $F_t = 4.25$ lb.
   Therefore, the force on the pinion, $F = \sqrt{F_t^2 + F_r^2}$
   Hence, $F = \sqrt{11.67^2 + 4.25^2} = 12.42$ lb. ■
3. From Newton's third law, the force on the gear tooth equals the force on the pinion tooth. ■

SOLUTION (15.33)

Known: A spur gear speed reducer is driven by an electric motor and drives a load involving "moderate shock". The gear teeth are standard full depth and of specified geometry and material. Required life is 10^6 pinion revolutions for a specified transmitted load.

Find: Determine an estimate of the reliability of the speed reducer with respect to bending fatigue failure.

Schematic and Given Data:



Assumptions:

1. The spur gears mesh at the pitch circles.
2. Load sharing is not expected since the cutting process is of average quality.
3. The effects corrected by the velocity factor, K_v, correspond to the middle of the range in Fig. 15.24 with manufacture by form cutters.
4. The pinion is driven by a uniform power motor while the gear drives a load involving "moderate shock" (given).
5. The tooth fillet radius is approximately equal to 0.35/P (to enable the use of Fig. 15.23 to estimate geometry factor J).

Analysis:

1. From Fig. 15.23(a), with no load sharing, J = 0.24.
   From Eq. (15.13a),

   $$V = \frac{\pi d n}{12} = \frac{\pi N_p n_p}{12P} = \frac{\pi(18)(1500)}{(12)(10)} = 706.8 \text{ ft/min}$$

   From Fig. 15.24, with V = 706.8 ft/min, $K_v = 2.0$
   From Table 15.1, $K_o = 1.25$
2. From Eq. (15.17) applied to the pinion:

   $$\sigma = \frac{F_t P}{b J} K_v K_o K_m = \frac{100(10)}{1.0(0.24)} (2.0)(1.25)(1.8)$$

   σ = 18,750 psi = 18.75 ksi

3. From Eq. (15.18) applied to the pinion:
$S_n = S_n' C_L C_G C_s k_r k_t k_{ms}$
$S_n = (65)(1)(1)(0.72)k_r(1)(1.4) = 65.52k_r$ ksi
since, $S_u = 500(Bhn) = 500(260)$ psi = 130 ksi.
$S_n' = S_u/2 = 65$ ksi and for bending loads, $C_L = 1.0$, for $P > 5$, $C_G = 1.0$,
from Fig. 8.13, $C_s = 0.72$.
Therefore $18.75 = 65.52k_r$; hence, $k_r = 0.29$
4. Similarly for the gear, $J = 0.27$, $S_n' = 58.75$ ksi.
$C_s = 0.75$; hence, $k_r = 0.30$
5. From Table 15.3, reliability is >> 99.999% ■

Comments:
1. The reliability estimated in this problem is based on considering failure only by bending fatigue. A more accurate estimate of reliability must consider failure by surface fatigue also.
2. Increasing the hardness of the gears will result in new choices in transmitting a higher load and higher rpm or choosing a smaller face width or a larger diametral pitch (i.e., with finer teeth).
3. The choice of a harder material for the pinion gives approximately the same reliability for both the pinion and gear in this case. Thus choice of a harder material for the pinion reflects consistency in the strength design of the gears.