Errata
Shigley's Mechanical Engineering Design
Richard D. Budynas and J. Keith Nisbett
ISBN 978-007-125763-3 or MHID 007-125763-2
Compiled in November 2008

Page 20

Current version

$a = 44.50 \pm 0.08$ mm $b = 19.05 \pm 0.02$ mm

$c = 3.05 \pm 0.13$ mm $c = 22.23 \pm 0.02$ mm

Corrected version

$a = 44.50 \pm 0.08$ mm $b = 19.05 \pm 0.02$ mm

$c = 3.05 \pm 0.13$ mm $d = 22.23 \pm 0.02$ mm

Current version

Thus, both clearance and interference are possible.
(b) If $w_{\min}$ is to be 0.08 mm, then, $\bar{w} = w_{\min} + t_w = 0.08 + 0.025 = 0.105$ mm. Thus,

$$\bar{d} = \bar{a} - \bar{b} - \bar{c} - \bar{w} = 44.50 - 19.05 - 3.05 - 0.105 = 22.30 \text{ mm}$$

Corrected version

Thus, both clearance and interference are possible.
(b) If $w_{\min}$ is to be 0.08 mm, then, $\bar{w} = w_{\min} + t_w = 0.08 + 0.25 = 0.33$ mm. Thus,

$$\bar{d} = \bar{a} - \bar{b} - \bar{c} - \bar{w} = 44.50 - 19.05 - 3.05 - 0.33 = 22.07 \text{ mm}$$

Page 32

Current version

$$f(x) = \frac{1}{17.9\sqrt{2\pi}} \exp\left[-\frac{1}{2}\left(\frac{x - 438.3}{17.9}\right)^2\right]$$

where the mean stress is 438.3 MPa and the standard deviation is 17.9 MPa. A plot of $f(x)$ is included in Fig. 2–5. The description of the strength S_{ut} is then expressed in terms of its statistical parameters and its distribution type. In this case $\mathbf{S}_{ut} = \mathbf{N}(438.3, 17.9)$ MPa.

Corrected version

$$f(x) = \frac{1}{18.16\sqrt{2\pi}} \exp\left[-\frac{1}{2}\left(\frac{x - 445.4}{18.16}\right)^2\right]$$

where the mean stress is 445.4 MPa and the standard deviation is 18.16 MPa. A plot of $f(x)$ is included in Fig. 2–5. The description of the strength S_{ut} is then expressed in terms of its statistical parameters and its distribution type. In this case $\mathbf{S}_{ut} = \mathbf{N}(445.4, 18.16)$ MPa.

Page 92

Current version

However, the maximum stress due to the combined bending and direct shear stresses may be maximum at the point $(76^{-}, 32.9)$ that is just to the left of the applied load, where the web joins the flange. To simplify the calculations we assume a cross section with square corners (Fig. 3–19*c*). The normal stress at section *ab*, with $x = 3$ in, is

Corrected version

However, the maximum stress due to the combined bending and direct shear stresses may be maximum at the point $(80^{-}, 32.9)$ that is just to the left of the applied load, where the web joins the flange. To simplify the calculations we assume a cross section with square corners (Fig. 3–19*c*). The normal stress at section *ab*, with $x = 0.08$ m, is

---

Page 93

Current version

The principal stresses at the point can now be determined. Using Eq. (3–13), we find that at $x = 76^{-}$ mm, $y = 32.9$ mm,

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{-5.24 + 0}{2} \pm \sqrt{\left(\frac{-5.24 - 0}{2}\right) + (-2.67)^2} = 1.12, -6.36 \text{ M}\ \ \text{a}$$

For a point at $x = 76^{-}$ mm, $y = -32.9$ mm, the principal stresses are $\sigma_1, \sigma_2 = 6.36, -1.12$ MPa. Thus we see that the maximum principal stresses are ± 1200 psi, 21 percent higher than thought by the designer.

Corrected version

The principal stresses at the point can now be determined. Using Eq. (3–13), we find that at $x = 80^{-}$ mm, $y = 32.9$ mm,

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$= \frac{-5.24 + 0}{2} \pm \sqrt{\left(\frac{-5.24 - 0}{2}\right) + (-2.67)^2} = 1.12, -6.36 \text{ M}\ \ \text{a}$$

For a point at $x = 80^{-}$ mm, $y = -32.9$ mm, the principal stresses are $\sigma_1, \sigma_2 = 6.36, -1.12$ MPa. Thus we see that the maximum principal stresses are ± 6.36 MPa, 5.1 percent higher than thought by the designer.

Page 106

Current version

As presented in the table, K_t is a decreasing monotone. This rod end is similar to the square-ended lug depicted in Fig. A–15-12 of appendix A.

Corrected version

As presented in the table, K_t is a decreasing monotone. This rod end is similar to the square-ended lug depicted in Fig. A–13-12 of appendix A.

Page 159

Current version

Compare Eqs. (*a*) and (*b*) with Eqs. (4–3) and (4–5). In Example 4–8, the bending strain energy for a cantilever having a concentrated end load was found. According to Castigliano's theorem, the deflection at the end of the beam due to bending is

Corrected version

Compare Eqs. (*a*) and (*b*) with Eqs. (4–3) and (4–5). In Example 4–9, the bending strain energy for a cantilever having a concentrated end load was found. According to Castigliano's theorem, the deflection at the end of the beam due to bending is

---

Page 197

Current version

4–62 The steel beam *ABCD* shown is supported at *C* as shown and supported at *B* and *D* by steel bolts each having a diameter of 8 mm. The lengths of *BE* and *DF* are 50 and 62 mm, respectively. The beam has a second area moment of 20.8×10^{-9} m^4. Prior to loading, the nuts are just in contact with the horizontal beam. A force of 2 kN is then applied at point *A*. Using procedure 2 of Sec. 4–10, determine the stresses in the bolts and the deflections of points *A*, *B*, and *D*. For steel, let E = 207 GPa.

Corrected version

4–62 The steel beam *ABCD* is supported at *C* as shown and supported at *B* and *D* by steel shoulder bolts each having a diameter of 8 mm. The lengths of *BE* and *DF* are 50 and 62 mm, respectively. The beam has a second area moment of 20.8×10^{-9} m^4. Prior to loading, the nuts are just in contact with the horizontal beam. A force of 2 kN is then applied at point *A*. Using procedure 2 of Sec. 4–10, determine the stresses in the bolts and the deflections of points *A*, *B*, and *D*. For steel, let E = 207 GPa.

Page 209

Current version

The rationale can be expressed as follows. The worst-case scenario is that of an idealized non–strain-strengthening material shown in Fig. 5–6. The stress-strain curve rises linearly to the yield strength S_y, then proceeds at constant stress, which is equal to S_y. Consider a filleted rectangular bar as depicted in Fig. A–15–5, where the cross-section area of the small shank is 1 in^2. If the material is ductile, with a yield point of 280 MPa, and the theoretical stress-concentration factor (SCF) K_t is 2,

Corrected version

The rationale can be expressed as follows. The worst-case scenario is that of an idealized non–strain-strengthening material shown in Fig. 5–6. The stress-strain curve rises linearly to the yield strength S_y, then proceeds at constant stress, which is equal to S_y. Consider a filleted rectangular bar as depicted in Fig. A–13–5, where the cross-section area of the small shank is 643 mm^2. If the material is ductile, with a yield point of 280 MPa, and the theoretical stress-concentration factor (SCF) K_t is 2,

---

Page 276

Current version

$(S'_f)_{10^3} = \sigma'_F(2.10^3)^b = f\,S_{ut}$

Corrected version

$(S'_f)_{10^3} = \sigma'_F(2 \cdot 10^3)^b = f\,S_{ut}$

Page 338

Current version

ASME-elliptic $(\sigma_a/S_e)^2 + (\sigma_m/S_{ut})^2 = 1/n^2$ (6–47)

Corrected version

ASME-elliptic $(\sigma_a/S_e)^2 + (\sigma_m/S_y)^2 = 1/n^2$ (6–47)

Page 404

Current version

$$\sigma = \frac{F}{A} = \frac{4F}{\pi d_r^2} \qquad (8\text{–}8)$$

Corrected version

$$\sigma = -\frac{F}{A} = -\frac{4F}{\pi d_r^2} \qquad (8\text{–}8)$$

Page 423

Current version

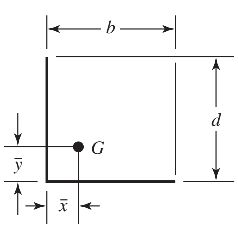
$$T = \frac{F_i d_m}{2}\left(\frac{\tan\lambda + f\sec\alpha}{l - f\tan\lambda\sec\alpha}\right) + \frac{F_i f_c d_c}{2} \qquad (b)$$

Corrected version

$$T = \frac{F_i d_m}{2}\left(\frac{\tan\lambda + f\sec\alpha}{1 - f\tan\lambda\sec\alpha}\right) + \frac{F_i f_c d_c}{2} \qquad (b)$$

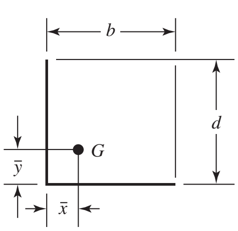
Page 466

Current version



$A = 0.707h(2b + d)$

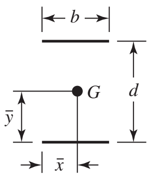
Corrected version



$A = 0.707h(b + d)$

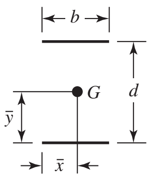
Page 470

Current version



$A = 1.414hd$

Corrected version



$A = 1.414hb$

Page 726

Current version

P_d is the transverse diameteral pitch

Corrected version

P_d is the transverse diametral pitch

Page 831

Current version

$$r_e = \frac{p_a \int_{r_i}^{r_o} r^2\, dr}{p_a \int_{r_i}^{r_o} r\, dr} = \frac{r_o^3 - r_i^3}{3} \frac{2}{r_o^2 - r_i^2} = \frac{2}{3} \frac{r_o^3 - r_i^3}{r_o^2 - r_i^3} \qquad (16\text{–}39)$$

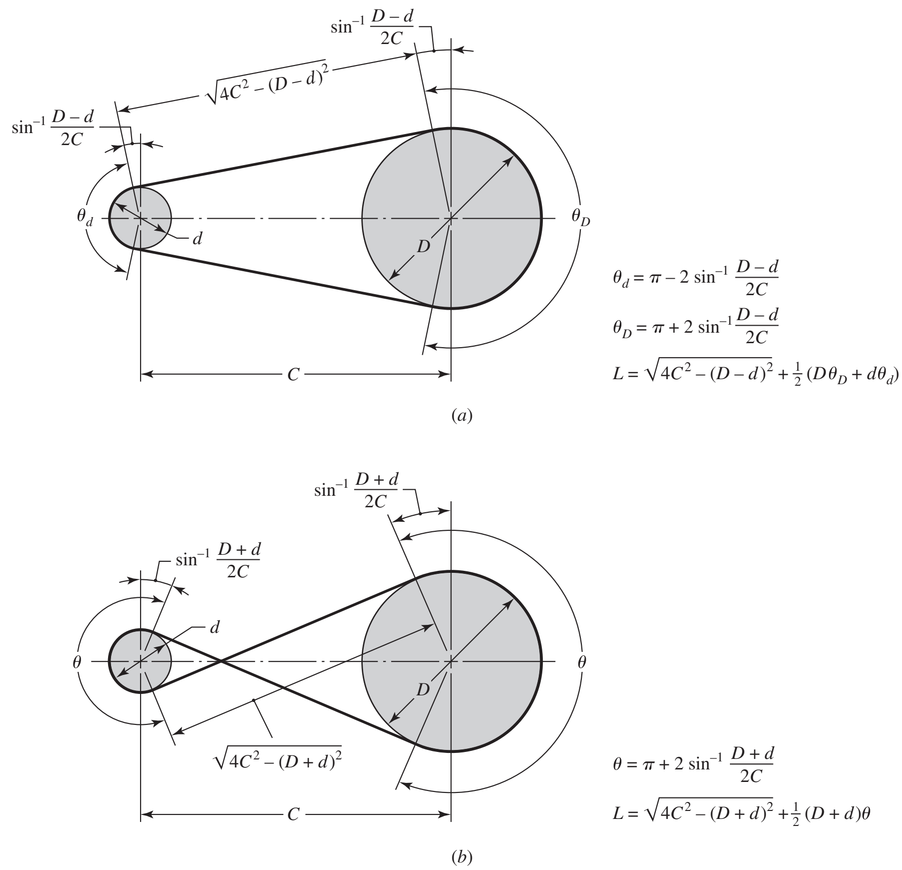
Corrected version

$$r_e = \frac{p_a \int_{r_i}^{r_o} r^2\, dr}{p_a \int_{r_i}^{r_o} r\, dr} = \frac{r_o^3 - r_i^3}{3} \frac{2}{r_o^2 - r_i^2} = \frac{2}{3} \frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} \qquad (16\text{–}39)$$

Current version

Figure 17–1

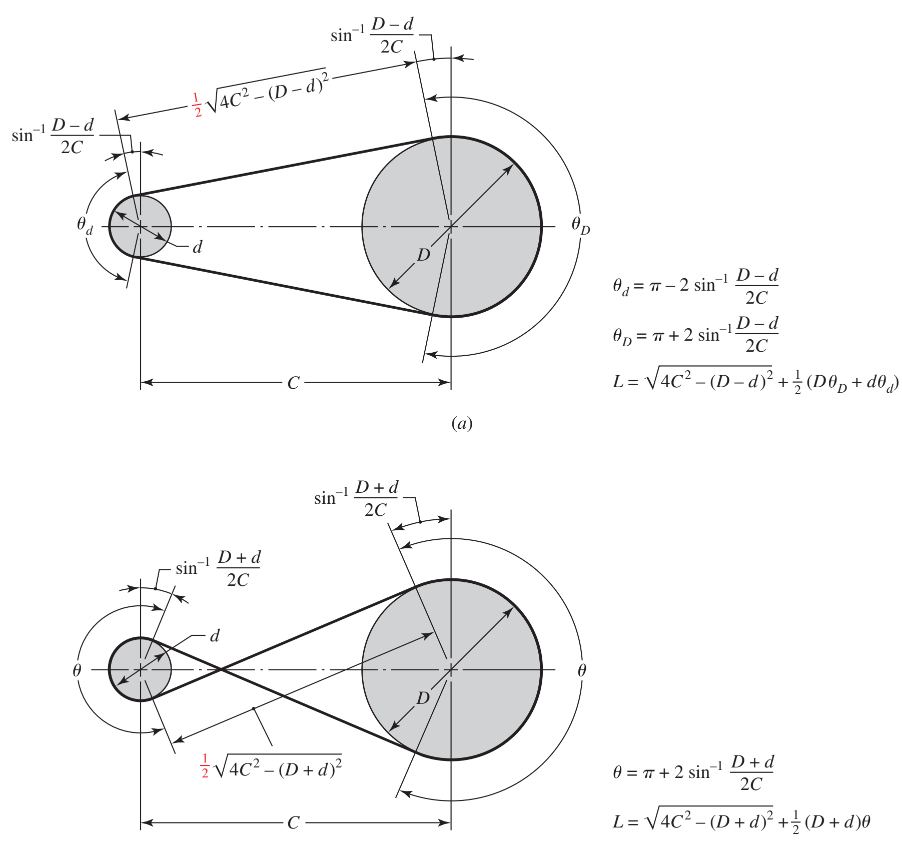
Flat-belt geometry. (*a*) Open belt. (*b*) Crossed belt.



Corrected version

Figure 17–1

Flat-belt geometry. (*a*) Open belt. (*b*) Crossed belt.



Page 865

Current version

$$F_1 = F_i + F_c + \Delta F' = F_i + F_c + T/D \qquad (f)$$

$$F_2 = F_i + F_c - \Delta F' = F_i + F_c - T/D \qquad (g)$$

where F_i = initial tension

F_c = hoop tension due to centrifugal force

$\Delta F'$ = tension due to the transmitted torque T

D = diameter of the pulley

The difference between F_1 and F_2 is related to the pulley torque. Subtracting Eq. (g) from Eq. (f) gives

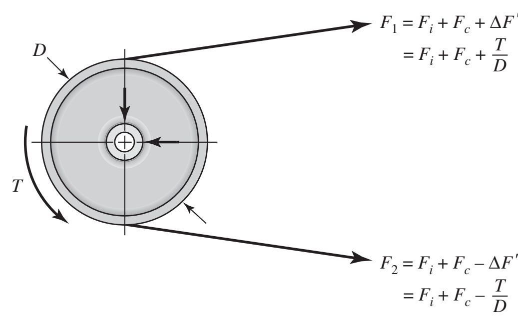
$$F_1 - F_2 = \frac{2T}{D} = \frac{T}{D/2} \qquad (h)$$

Adding Eqs. (f) and (g) gives

$$F_1 + F_2 = 2F_i + 2F_c$$

Figure 17–7

Forces and torques on a pulley.



Corrected version

$$F_1 = F_i + F_c + \Delta F' = F_i + F_c + T/d \qquad (f)$$

$$F_2 = F_i + F_c - \Delta F' = F_i + F_c - T/d \qquad (g)$$

where F_i = initial tension

F_c = hoop tension due to centrifugal force

$\Delta F'$ = tension due to the transmitted torque T

d = diameter of the pulley

The difference between F_1 and F_2 is related to the pulley torque. Subtracting Eq. (g) from Eq. (f) gives

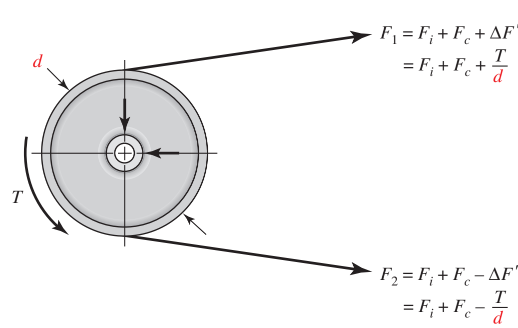
$$F_1 - F_2 = \frac{2T}{D} \qquad (h)$$

Adding Eqs. (f) and (g) gives

$$F_1 + F_2 = 2F_i + 2F_c$$

Figure 17–7

Forces and torques on a pulley.



Page 866

Current version

$$\frac{F_i}{T/D} = \frac{(F_1 + F_2)/2 - F_c}{(F_1 - F_2)/2} = \frac{F_1 + F_2 - 2F_c}{F_1 - F_2} = \frac{(F_1 - F_c) + (F_2 - F_c)}{(F_1 - F_c) - (F_2 - F_c)}$$

$$= \frac{(F_1 - F_c)/(F_2 - F_c) + 1}{(F_1 - F_c)/(F_2 - F_c) - 1} = \frac{\exp(f\phi) + 1}{\exp(f\phi) - 1}$$

from which

$$F_i = \frac{T}{D}\frac{\exp(f\phi) + 1}{\exp(f\phi) - 1} \tag{17–9}$$

Corrected version

$$\frac{F_i}{T/d} = \frac{(F_1 + F_2)/2 - F_c}{(F_1 - F_2)/2} = \frac{F_1 + F_2 - 2F_c}{F_1 - F_2} = \frac{(F_1 - F_c) + (F_2 - F_c)}{(F_1 - F_c) - (F_2 - F_c)}$$

$$= \frac{(F_1 - F_c)/(F_2 - F_c) + 1}{(F_1 - F_c)/(F_2 - F_c) - 1} = \frac{\exp(f\phi) + 1}{\exp(f\phi) - 1}$$

from which

$$F_i = \frac{T}{d}\frac{\exp(f\phi) + 1}{\exp(f\phi) - 1} \tag{17–9}$$

Current version

$$F_1 = F_i + F_c + \frac{T}{D} = F_c + F_i + F_i\frac{\exp(f\phi) - 1}{\exp(f\phi) + 1}$$

$$= F_c + \frac{F_i[\exp(f\phi) + 1] + F_i[\exp(f\phi) - 1]}{\exp(f\phi) + 1}$$

$$F_1 = F_c + F_i\frac{2\exp(f\phi)}{\exp(f\phi) + 1} \tag{17–10}$$

Corrected version

$$F_1 = F_i + F_c + \frac{T}{d} = F_c + F_i + F_i\frac{\exp(f\phi) - 1}{\exp(f\phi) + 1}$$

$$= F_c + \frac{F_i[\exp(f\phi) + 1] + F_i[\exp(f\phi) - 1]}{\exp(f\phi) + 1}$$

$$F_1 = F_c + F_i\frac{2\exp(f\phi)}{\exp(f\phi) + 1} \tag{17–10}$$

Page 866

Current version

$$F_2 = F_i + F_c - \frac{T}{D} = F_c + F_i - F_i \frac{\exp(f\phi) - 1}{\exp(f\phi) + 1}$$

$$= F_c + \frac{F_i[\exp(f\phi) + 1] - F_i[\exp(f\phi) - 1]}{\exp(f\phi) + 1}$$

$$F_2 = F_c + F_i \frac{2}{\exp(f\phi) + 1} \tag{17–11}$$

Corrected version

$$F_2 = F_i + F_c - \frac{T}{d} = F_c + F_i - F_i \frac{\exp(f\phi) - 1}{\exp(f\phi) + 1}$$

$$= F_c + \frac{F_i[\exp(f\phi) + 1] - F_i[\exp(f\phi) - 1]}{\exp(f\phi) + 1}$$

$$F_2 = F_c + F_i \frac{2}{\exp(f\phi) + 1} \tag{17–11}$$

Page 867

Current version

Equation (17–7) is called the *belting equation,* but Eqs. (17–9), (17–10), and (17–11) reveal how belting works. We plot Eqs. (17–10) and (17–11) as shown in Fig. 17–8 against F_i as abscissa. The initial tension needs to be sufficient so that the difference between the F_1 and F_2 curve is $2T/D$. With no torque transmitted, the least possible belt tension is $F_1 = F_2 = F_c$.

The transmitted power is given by

Corrected version

Equation (17–7) is called the *belting equation,* but Eqs. (17–9), (17–10), and (17–11) reveal how belting works. We plot Eqs. (17–10) and (17–11) as shown in Fig. 17–8 against F_i as abscissa. The initial tension needs to be sufficient so that the difference between the F_1 and F_2 curve is $2T/d$. With no torque transmitted, the least possible belt tension is $F_1 = F_2 = F_c$.

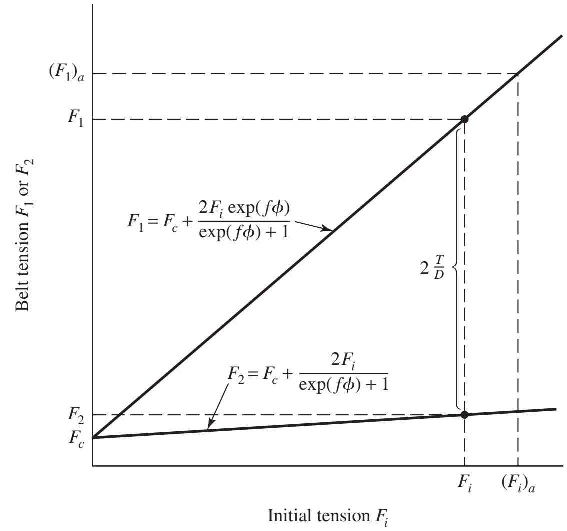
The transmitted power is given by

Page 867

Current version

Figure 17–8

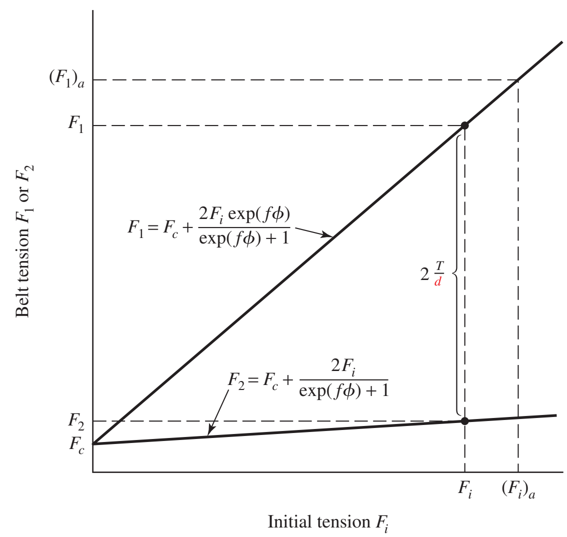
Plot of initial tension F_i against belt tension F_1 or F_2, showing the intercept F_c, the equations of the curves, and where $2T/D$ is to be found.



Corrected version

Figure 17–8

Plot of initial tension F_i against belt tension F_1 or F_2, showing the intercept F_c, the equations of the curves, and where $2T/d$ is to be found.



Page 868

Current version

4 From torque T find the necessary $(F_1)_a - F_2 = 2T/D$

Corrected version

4 From torque T find the necessary $(F_1)_a - F_2 = 2T/d$

Current version

$$d = \frac{L^2 w}{8F_i} \tag{17–13}$$

where d = *dip*, m
L = center-to-center distance, m
w = weight per foot of the belt, N/m
F_i = initial tension, N

In Ex. 17–1 the dip corresponding to a 1240-N initial tension is

$$d = \frac{(2.4)^2\, 5.4}{8(1240)} = 0.0032 \text{ m} = 3.2 \text{ mm}$$

Corrected version

$$dip = \frac{L^2 w}{8F_i} \tag{17–13}$$

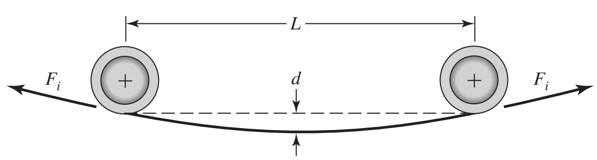
where dip = dip, m
L = center-to-center distance, m
w = weight per unit volume of the belt, N/m³
F_i = initial tension, N

In Ex. 17–1 the dip corresponding to a 1240-N initial tension is

$$dip = \frac{(2.4)^2\, 5.4}{8(1240)} = 0.0032 \text{ m} = 3.2 \text{ mm}$$

Current version

Figure 17–11

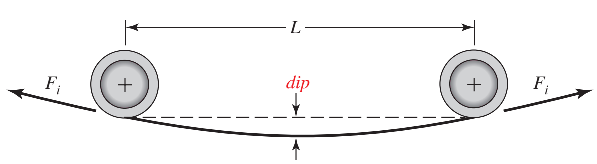


(c)

(c) Catenary-induced tension.

Corrected version

Figure 17–11



(c)

(c) Catenary-induced tension.

Page 874

Current version

$$d = \frac{L^2 w}{8F_i} = \frac{4.8^2(37.6)0.25}{8(2420)} = 0.011 \text{ m} = 11 \text{ mm}$$

Corrected version

$$dip = \frac{L^2 w}{8F_i} = \frac{4.8^2(37.6)0.25}{8(2420)} = 0.011 \text{ m} = 11 \text{ mm}$$

Page 880

Current version

$$H_a = K_1 K_2 H_{\text{tab}} \qquad (17\text{–}17)$$

where H_a = allowable power, per belt, Table 17–12

Corrected version

$$H_a = K_1 K_2 H_{\text{tab}} \qquad (17\text{–}17)$$

where H_a = allowable power, per belt

Page 908

Current version

(*c*) Estimate the rated (allowable) power that would appear in Table 17–20 for a 20 000-h life.

Corrected version

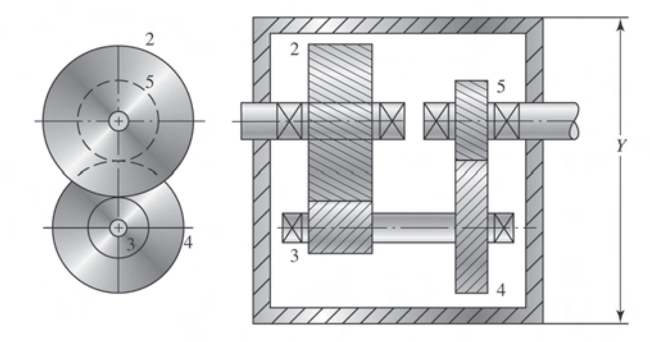
(*c*) Estimate the allowable horsepower for a 20 000-h life.

Page 914

Current version

Figure 18–1

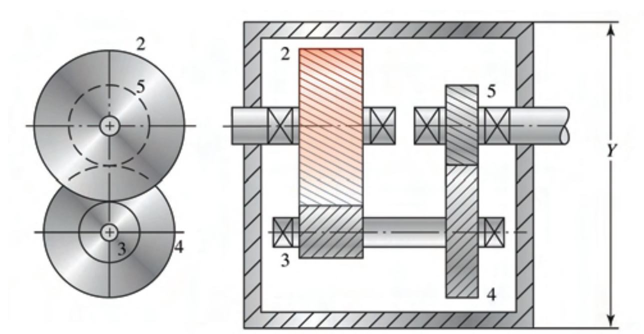
A compound reverted gear train.



Corrected version

Figure 18–1

A compound reverted gear train.



Page 922

Current version

$$\sigma_c = 2300\sqrt{\frac{2431(1.18)(1.21)}{12(2)(0.1315)}} = 76\ 280 \text{ psi}$$

$$\sigma = (2431)(1.18)\left(\frac{6}{2}\right)\left(\frac{1.21}{0.41}\right) = 25\ 400 \text{ psi}$$

Choose a Grade 1 steel, through-hardened to 250 H_B. From Fig. 14–2, p. 727 , $S_t = 32\,000$ psi and from Fig. 14–5, p. 730, $S_c = 110\ 000$ psi.

Corrected version

$$\sigma_c = 2300\sqrt{\frac{2431(1.18)(1.21)}{2.67(2)(0.1315)}} = 161\ 700 \text{ psi}$$

$$\sigma = (2431)(1.18)\left(\frac{6}{2}\right)\left(\frac{1.21}{0.41}\right) = 25\ 400 \text{ psi}$$

Choose Grade 2 carburized and hardened, the same as gear 4.

Current version

Gear 3 Wear and Bending

$$J = 0(41 \qquad Y_N = 0.9 \qquad Z_N = 0.9$$

$$\sigma_c = 2300\sqrt{\frac{(539.7)(1.37)(1.19)}{12(1.5)(0.1315)}} = 44\,340 \text{ psi}$$

$$\sigma = 539.7(1.37)\frac{(6)(1.19)}{1.5(0.41)} = 8584 \text{ psi}$$

Try Grade 1 steel, through-hardened to 200 H_B. From Fig. 14–2, p. 727, $S_t = 28\,000$ psi and from Fig. 14–5, p. 730, $S_c = 90\,000$ psi.

$$n_c = \frac{90\,000(0.9)}{44\,340} = 1.83$$

$$n = \frac{\sigma_{\text{all}}}{\sigma} = \frac{28\,000(0.9)}{8584} = 2.94$$

In summary, the resulting gear specifications are:

All gears, $P = 6$ teeth/in

Gear 2, Grade 1 flame-hardened, $S_c = 170\,000$ psi and $S_t = 45\,000$ psi
$d_2 = 2.67$ in, face width $= 1.5$ in

Gear 3, Grade 1 through-hardened to 200 H_B, $S_c = 90\,000$ psi and $S_t = 28\,000$ psi
$d_3 = 12.0$ in, face width $= 1.5$ in

Gear 4, Grade 2 carburized and hardened, $S_c = 225\,000$ psi and $S_t = 65\,000$ psi
$d_4 = 2.67$ in, face width $= 2.0$ in

Gear 5, Grade 1 through-hardened to 250 H_B, $S_c = 110\,000$ psi and $S_t = 31\,000$ psi
$d_5 = 12.0$ in, face width $= 2.0$ in

Corrected version

Gear 3 Wear and Bending

$$J = 0(41 \qquad Y_N = 0.9 \qquad Z_N = 0.9$$

$$\sigma_c = 2300\sqrt{\frac{(539.7)(1.37)(1.19)}{2.67(1.5)(0.1315)}} = 94\,000 \text{ psi}$$

$$\sigma = 539.7(1.37)\frac{(6)(1.19)}{1.5(0.41)} = 8584 \text{ psi}$$

Try Grade 1 steel, through-hardened to 300 H_B. From Fig. 14–2, p. 727, $S_t = 36\,000$ psi and from Fig. 14–5, p. 730, $S_c = 126\,000$ psi.

$$n_c = \frac{126\,000(0.9)}{94\,000} = 1.21$$

$$n = \frac{\sigma_{\text{all}}}{\sigma} = \frac{36\,000(0.9)}{8584} = 3.77$$

In summary, the resulting gear specifications are:

All gears, $P = 6$ teeth/in

Gear 2, Grade 1 flame-hardened, $S_c = 170\,000$ psi and $S_t = 45\,000$ psi
$d_2 = 2.67$ in, face width $= 1.5$ in

Gear 3, Grade 1 through-hardened to 300 H_B, $S_c = 126\,000$ psi and $S_t = 36\,000$ psi
$d_3 = 12.0$ in, face width $= 1.5$ in

Gear 4, Grade 2 carburized and hardened, $S_c = 225\,000$ psi and $S_t = 65\,000$ psi
$d_4 = 2.67$ in, face width $= 2.0$ in

Gear 5, Grade 2 carburized and hardened, $S_c = 225\,000$ psi and $S_t = 65\,000$ psi
$d_5 = 12.0$ in, face width $= 2.0$ in

Page 940

Current version

$$\begin{Bmatrix} f_{1\cdot 1} \\ f_{2\cdot 1} + f_{2\cdot 2} \\ f_3 \end{Bmatrix} = \begin{Bmatrix} F_1 \\ F_2 \\ F_3 \end{Bmatrix} = \begin{Bmatrix} k_1 & -k_1 & 0 \\ -k_1 & (k_1 + k_2) & -k_2 \\ 0 & -k_2 & k_2 \end{Bmatrix} \begin{Bmatrix} u_1 \\ u_2 \\ u_3 \end{Bmatrix} \qquad (19\text{–}5)$$

Corrected version

$$\begin{Bmatrix} f_{1\cdot 1} \\ f_{2\cdot 1} + f_{2\cdot 2} \\ f_3 \end{Bmatrix} = \begin{Bmatrix} F_1 \\ F_2 \\ F_3 \end{Bmatrix} = \begin{bmatrix} k_1 & -k_1 & 0 \\ -k_1 & (k_1 + k_2) & -k_2 \\ 0 & -k_2 & k_2 \end{bmatrix} \begin{Bmatrix} u_1 \\ u_2 \\ u_3 \end{Bmatrix} \qquad (19\text{–}5)$$

Page 941

Current version

With $u_1 = 0$, $F_2 = 4500$ N and the assumption that $u_3 = \epsilon = 0.05$ mm, Eq. (19.5) becomes

$$\begin{Bmatrix} F_1 \\ 4500 \\ F_3 \end{Bmatrix} = 10^3 \begin{Bmatrix} 17(802 & -17.802 & 0 \\ -17(802 & 40.066 & -22.264 \\ 0 & -22.264 & 22.264 \end{Bmatrix} \begin{Bmatrix} 0 \\ u_2 \\ 0.05 \end{Bmatrix} \qquad (1)$$

Corrected version

With $u_1 = 0$, $F_2 = 4500$ N and the assumption that $u_3 = \epsilon = 0.05$ mm, Eq. (19-5) becomes

$$\begin{Bmatrix} F_1 \\ 4500 \\ F_3 \end{Bmatrix} = 10^3 \begin{bmatrix} 17(802 & -17.802 & 0 \\ -17(802 & 40.066 & -22.264 \\ 0 & -22.264 & 22.264 \end{bmatrix} \begin{Bmatrix} 0 \\ u_2 \\ 0.05 \end{Bmatrix} \qquad (1)$$

Page 995

Current version

$$y_c = -\frac{Fa^2}{3EI}(l + a)$$

Corrected version

$$y_C = -\frac{Fa^2}{3EI}(l + a)$$

Page 1036

Current version

6–12 Yield: $n_y = 1.18$. Fatigue: (a) $n_f = 1.06$, (b) $n_f = 1.31$, (c) $n_f = 1.32$

Corrected version

6–12 Yield: $n_y = 1.67$. Fatigue: (a) $n_f = 1.06$, (b) $n_f = 1.31$, (c) $n_f = 1.32$

Page 1037

Current version

9–8 $F = 49.2$ kN

Corrected version

9–8 $F = 49.7$ kN

Additional Erratum
Shigley's Mechanical Engineering Design
November 2009

Page 505

Current version

can be employed to obtain the torsional yield strength ($S_{ys} = 0.577S_y$). This approach results in the range

$$0.35S_{ut} \leq S_{sy} \leq 0.52\ S_{ut} \qquad (10\text{–}15)$$

for steels.

Corrected version

can be employed to obtain the torsional yield strength ($S_{sy} = 0.577S_y$). This approach results in the range

$$0.35S_{ut} \leq S_{sy} \leq 0.52\ S_{ut} \qquad (10\text{–}15)$$

for steels.