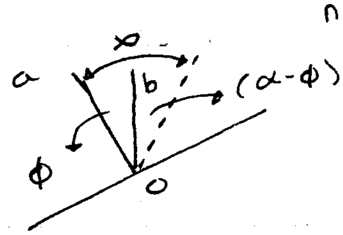
Feb. 27 /18
MACHINE DESIGN

For table (1, not 3)
↪ look up music wire, should bring you to

- Reverse motion : if the rotation of the screw moves the nut in the same direction as the load, then;



$$T = -\frac{Qd}{2}\tan(\alpha-\phi)$$

$$T = \frac{Qd}{2} \times \frac{\pi f d - L}{\pi d + f L}$$

3.2 - Angular or V-thread

- it can be shown that in this case,

$$\boxed{T = Q\frac{d}{2} \times \frac{\pi f d \sec\beta + L}{\pi d - f L \sec\beta}}$$

and

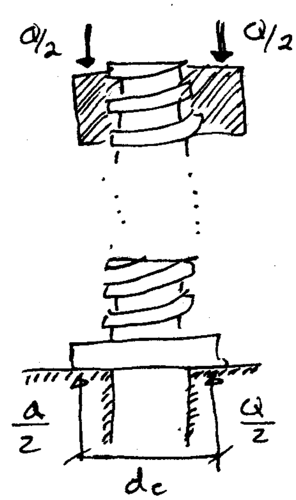
$$\boxed{e = \frac{\tan\alpha\,[1-(f\sec\beta\tan\alpha)]}{\tan\alpha + f\sec\beta}}$$

4 - Collar Friction

- $T_c = F d_c$

$F = f\frac{Q}{2}$

$\therefore \quad T_c = fQ d_c/2$



5 - Stresses in Screws

5.1 - Tensile or Compressive stress

$\sigma = Q/A$

where A = area of minimum cross section

5.2 - Torsional Stress

$$\tau = Tr / J$$

where r = radius of minimum cross-section

6.3 - Shearing Stress (on thread)

$$S_s \text{ (screw)} = Q / n\pi d_r t$$

$$S_s \text{ (nut)} = Q / n\pi d_o t$$

Where, Q = axial load

d_o = outside diameter of thread (major)

d_r = root diameter (minor)

t = width of thread

n = number of engaged threads

5.4 - Bearing Pressure on the Threads

$$S_b = \frac{4Q}{n\pi (d_o^2 - d_r^2)}$$

6 - Coefficient of Friction

1 - For high-grade materials, workmanship, and well run-in and lubricated threads $f \simeq 0.1$

2 - For average grade ; $f \simeq 0.125$

3 - For poor-quality ; $f \simeq 0.15$

4 - For starting conditions ; $f_s = 1\tfrac{1}{3} f$

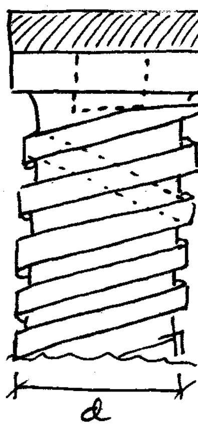
5 - Collar friction may be taken as the same as for thread friction.

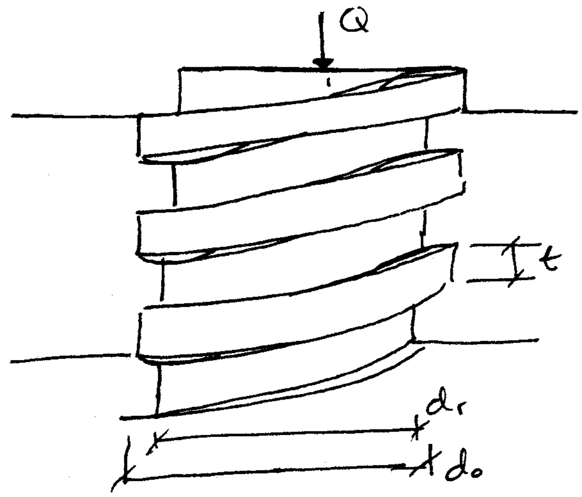
Example 1 - A 10-ton screw jack with a maximum extension of 4 in. is to have double-square threads.

- allowable stress in compression is 5000 psi.

a) Determine, - the size of the screw
- the size of the collar
- length of nut

b) Find the torque required to raise the load, and the efficiency of the jack.





Solution :

a) $\sigma_c = Q/A$

the root area $A = Q/\sigma_c$

$$A = \frac{20000}{5000} = 4 \text{ in}^2$$

From table 12-1 (H.O.), a $2\,^3/_4$ in screw with 2 threads/in is selected.

$\therefore$ $L = 1$ in

$P = {}^1/_2$ in (double thread $\therefore L = 2P$)

Assume an outer diameter for the collar.

$d_o = 3\,{}^1/_2$ in

and $d_i = 1$ in

Then the bearing pressure P_c is

$$P_c = \frac{4Q}{\pi(d_o^2 - d_i^2)} = \frac{4 \times 20000}{\pi(3.5^2 - 1^2)} = 2270 \text{ psi}$$

From table 12-3 (H.O.) P_c is within the safe range

$$h = \tfrac{7}{16} P \simeq \tfrac{1}{2} P = 0.25 \text{ in}$$

$$d = 2\,3/4 - 0.25 = 2.5 \text{ in}$$

Bearing area / Thread $= \pi d h = \pi \times 2.5 \times 0.25 = 1.97 \text{ in}^2$

Using an allowable thread bearing pressure of 2500 psi

$$n\, S_b = \frac{Q}{A} = 20000 / 1.97$$

$$n = \frac{20000}{2500 \times 1.97} = 4.06 \text{ thread}$$

or since we have 2 threads / in the height of the nut is ; $h = 4.06 / 2 \simeq 2$ in

- For stability of screw it is common practice to use its length at least equal to the diameter of the thread.

$\therefore$ $h = 3$ in is a reasonable length.

b -

$$T = \frac{Qd}{2} \times \frac{\pi f d + L}{\pi d - f L}$$

if we take $f = 0.125$

$$T = \frac{20000 \times 2.5}{2} \times \frac{\pi \times 0.125 \times 2.5 + 1}{\pi \times 2.5 - 0.125 \times 1} = 6400 \text{ lb-in}$$

The torque for collar friction is :

$$T_c = \frac{f Q d_c}{2} = \frac{0.125 \times 20000 \times 2.25}{2} \; ; \; T_c = 2810 \text{ in-lb}$$

$$T_t = T + T_c = 6400 + 2810 = 9210 \text{ lb-in}$$

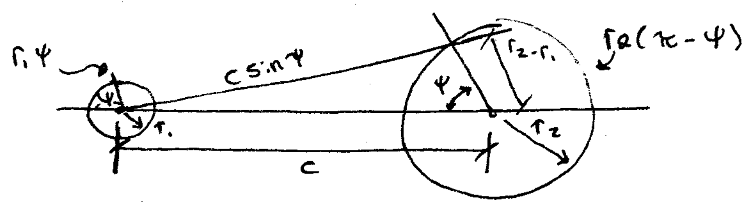
$$e = \frac{QL}{2\pi T} = \frac{20000 \times 1}{2\pi \times 9210} = 0.35$$

MARCH 1/18
MACHINE DESIGN

Belts

Symbols are as listed on pp. 354 - 355 (Spotts)

1 - Center Distance for V-belts Drive



$$\cos\psi = (r_2 - r_1)/c$$

$$l = 2c\sin\psi + 2\pi r_2 - 2\psi(r_2 - r_1)$$

↳ very difficult to solve

The centre distance c is found from the following equation:

$$\tfrac{1}{2} l = \frac{\pi}{2} r_1 + \sqrt{C^2 + (r_2 - r_1)^2} + \frac{\pi}{2} r_2$$

OR $$C^2 = (1/4)\left[l - \pi(r_1 + r_2)\right]^2 - (r_2 - r_1)^2$$

and the half angle of contact is given by $\cos\psi = \frac{r_2 - r_1}{c}$

Example 1 - A v-belt is 87.9 in. long and operates on sheaves of pitch diameters of 12 in and 16 in. Find the centre distance c.

Solution:

$$C^2 = 1/4\left[l - \pi(r_1 + r_2)\right]^2 - (r_2 - r_1)^2$$

$$= 1/4\left[87.9 - \pi(6+8)\right]^2 - (8-6)^2$$

$$= 478.24 \text{ in}^2$$

$$C = 21.9 \text{ in}$$

2 - Fatigue of V-belts

The velocity of the belt is;

$$V = \frac{\pi d n}{12}$$

where V = belt velocity, ft/min
d = pulley diameter, in
n = pulley speed, rpm

The nominal horsepower of the belt is

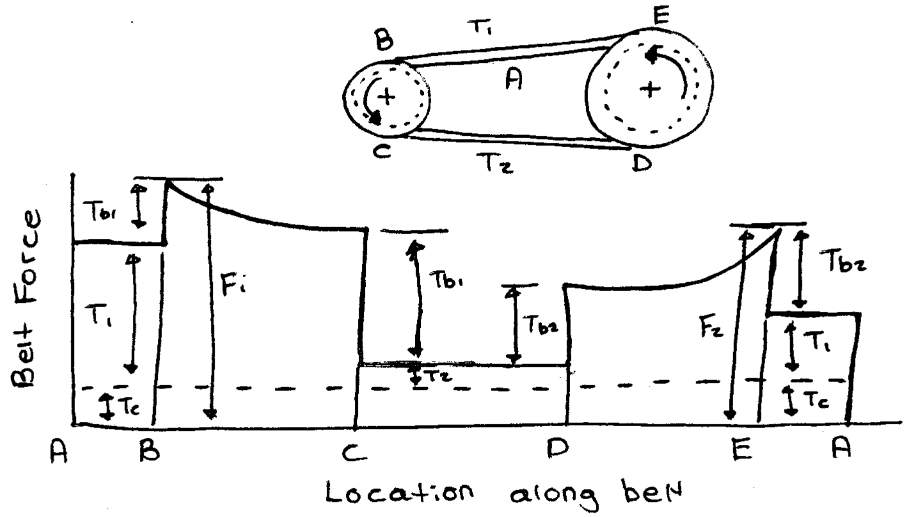
$$hp = \frac{(T_1 - T_2)V}{33{,}000}$$

A service factor from table 6-3 (spotts) must be applied to the nominal horsepower to account for fluctuations in the loading.

The peak force in the belt is given by:

$$F_i = T_1 + T_{bi} + T_c$$

where,
T_1 = tight side tension
T_{bi} = Force caused by bending around pulley i
T_c = Force due to the centrifugal effects



Thus at B $F_1 = T_1 + T_{b1} + T_c$
Thus at E $F_2 = T_2 + T_{b2} + T_c$

Where, $T_b = \dfrac{K_b}{d}$; K_b = constant from table 6-4 (spotts)

and $T_c = K_c\left(\dfrac{V}{1000}\right)^2$; K_c = constant from table 6-4 (spotts)
V = belt speed (Ft/min)

The belt life is given by;

$$M_1 = \left(\frac{Q}{F_1}\right)^x$$

where, M_1 = number of applications of peak force F_1
Q and x are given in table 6-5 (spotts)

For the slack side

$$M_i = \left(\frac{Q}{F_2}\right)^x$$

Let N' = number of belt rotation to Failure

then $\frac{1}{N'} = \frac{1}{M_1} + \frac{1}{M_2}$

and the belt life is ;

$$L = \frac{N' l}{12 v} \quad \text{(minutes)}$$

Where, L = belt life, minutes

N' = belt life, revolutions

l = belt length, in

V = belt velocity, ft/min

Example 2 – A C-section belt 87.9 in long operates on pulleys of pitch diameter 12 in and 16 in. Speed of smaller pulley is 1160 rpm Horsepower is 9, but a service factor of 1.6 must be used. Find the expected life.

Solution : $V = \frac{\pi d n}{12} = \frac{\pi 12 \times 1160}{12} \simeq 3644$ ft/m

$$C = \{(1/4)[87.9 - \pi(6+8)]^2 - (8-6)^2\}^{1/2} = 21.9 \text{ in}$$

$$\cos\psi = (r_2 - r_1)/c = (8-6)/(21.9) = 0.09145$$

From Fig 6-3 (spotts)

$T_1/T_2 = 4.55$

$T_2 = T_1/(4.55) = 0.220\,T_1$

Design hp $= 9 \times 1.6 = 14.4$

$$hp = \frac{(T_1 - T_2)V}{33{,}000} \qquad T_1 - T_2 = T_1 - 0.220\,T_1 = \frac{33000\,hp}{V}$$

$$0.780\, T_1 = \frac{33{,}000 \times 14.4}{3.644} = 130.4 \text{ lb}$$

(K_b, K_c from T6-4)

$$T_1 = 167.2 \text{ lb}$$

$$T_{b1} = \frac{K_b}{d} = \frac{1600}{12} = 133.3 \text{ lb}$$

$$T_c = K_c \left(\frac{V}{1000}\right)^2 = 1.716 \times 3.644^2 = 22.8$$

$$F_1 = T_1 + T_{b1} + T_c = 323.3 \text{ lb}$$

$$M_1 = \left(\frac{Q}{F_1}\right)^x = \left(\frac{2038}{323.3}\right)^{(11.173)} = 859 \times 10^6$$

$$T_{b2} = \frac{K_b}{d} = \frac{1600}{16} = 100 \text{ lb}$$

$$F_2 = T_1 + T_{b2} + T_c = 290 \text{ lb}$$

$$M_2 = \left(\frac{2038}{290}\right)^{(11.173)} = 2895 \times 10^6$$ Force peaks

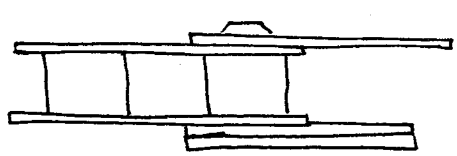
$$\frac{1}{N'} = \frac{1}{M_1} + \frac{1}{M_2} = \frac{1}{10^9}\left(\frac{1}{0.859} + \frac{1}{2.895}\right)$$

$N' = 662{,}700{,}000$ belt passes

$$L = \frac{N' l}{12 v} = \frac{(6.627 \times 10^8)(87.9)}{(12)(3{,}644)}$$

$L = 1332128.8$ min $= 22202.147$ hr

Roller Chains



1- horsepower capacity

At lower speeds, the horsepower capacity is determined by the Fatigue life of the link plates.

$$hp = 0.004 (N_1)^{1.08} (n_1)^{0.9}\, p^{(3.0 - 0.07p)}$$

Where, N_1 = number of teeth in the smaller sprocket
n_1 = speed, rpm, of the smaller sprocket
p = chain pitch, in

For No. 41 chain the constant 0.004 must be replaced by 0.0022.

At higher speeds the horsepower is determined by the roller bushing fatigue life.

$$hp = \frac{1000\, K (N_1)^{1.5}\, p^{0.8}}{(n_1)^{1.5}}$$

Where $K = 29$ for chains Nos. 25 and 35
$K = 3.4$ for chains Nos. 41
$K = 17$ for chains Nos. 40 to 240

Example 1 - For a single-strand No. 60 chain, $p = 3/4$ in and the smaller sprocket has $N_1 = 15$ teeth smooth loading.

a - Find the horsepower capacity at $n_1 = 900$ rpm for the smaller sprocket

b - Find the horsepower capacity if $n_1 = 1400$ rpm

a/ Link plate fatigue :

$$hp = 0.004 \times 15^{1.08} \times 900^{0.9} \times 0.75^{(3 - 0.07 \times 0.75)}$$
$$= 14.60$$

Roller-bushing fatigue :

$$hp = \frac{17000 \times 15^{1.5} \times 0.75^{0.8}}{900^{1.5}} = 29.06$$

$\therefore$ at 900 rpm link plate fatigue controls b/ link plate Fatigue :

$$hp = 0.004 \times 15^{1.08} \times 1400^{0.9} \times 0.75^{(3 - 0.07 \times 0.75)}$$

$$= 21.65$$

roller bushing Fatigue :

$$hp = \frac{17000 \times 15^{1.5} \times 0.75^{0.8}}{1400^{1.5}} = 14.98$$

$\therefore$ at 1400 rpm, roller bushing Fatigue controls