APRIL 3/18
MACHINE DESIGN

Solution:

| | $S_o$ (Table 18-2) | $n$ | $y$ (Table 18-1) | $S_o y$ |
|---|---|---|---|---|
| Pinion | 30000 | 24 | 0.107 | 3210 |
| Gear | 20000 | 3×24 = 72 | 0.136 | 2720 |

$S_o y$ of Gear < $S_o y$ of Pinion

$\therefore$ Gear is weaker, must be basis for design

$$T_G = \frac{63030 \times hp}{rpm} \times 1.3$$

$$T_G = \frac{63030 \times 10 \times 1.3}{(1170/3)} = 2100 \text{ lb-in}$$

$$S_{ind} = \frac{2TP^3}{K\pi^2 n y}$$ Assuming $K = F/p = 3.5$

$$S_{ind} = \frac{2 \times 2100 P^3}{3.5\pi^2 \times 72 \times 0.136} = 12.4 P^3$$

$$S_{allow} = S_o \frac{600}{600+V} = 20000 \frac{(600)}{(600+V)}$$

Trial Solution

$S_{ind}$ = 4260 psi For $P = 7$
= 6350 $P = 8$
= 9040 $P = 9$

From Fig 18-4 (H.O.) $C_v$ changes from 0.5 to 2

Assume $C_v = 1/3$

Then $S_{all} = 20000 \times 1/3$

Take $S_{all} = 7000$ psi

Then choose $P = 9$

$$\therefore D = \frac{n}{P} = \frac{72}{9} = 8 \text{ in}$$

$$V = \frac{\pi D(rpm)}{12} = \frac{\pi \times 8 \times 1170/3}{12} \cong 816 \text{ fpm}$$

$$S_{all} = S_o \times \frac{600}{600+V} = 20000 \times \frac{600}{600+816}$$

$S_{all}$ = 8480 psi

Therefore, $P = 9$ may be satisfactory $S_{ind}$ is close to $S_{allow}$
The computed face width is:

$$K = F/p$$

$$\therefore \quad F = Kp = K\,\pi/P = 3.5\,\pi/9 = 1.22 \text{ in}$$

To bring the induced stress down to allowable stress F is multiplied by a factor:

$$S_{ind} / S_{all} > 1$$

$$\therefore \quad F = \frac{1.22 \times 9040}{8480} = 1.3 \text{ in}$$

Note that if $S_{ind}$ was less than $S_{all}$ then $S_{ind} / S_{all} < 1$

have $p = \frac{\pi}{P} = \frac{\pi}{9} = 0.349$

$\therefore \quad 3 \times 0.349 < F < 4 \times 1.396$

$1.047 < F < 1.396$

$\therefore$ design o.k.

Results $P = 9$ ; $D_P = 8/3 = 2.67$ in

$D_G = 8.0$ in ; $F = 1.3$ in

May use commercial dimension $F = 1.25$ in

Centre distance $= (2.67 + 8)/2 = 5.33$ in

4 - Buckingham Equation

After extensive series of tests it was Found that a closer approximation to actual conditions is achieved by replacing W by $W_d$

$$W_d = W + \frac{0.05V(FC+W)}{0.05V + \sqrt{FC+W}}$$

Where $W_d$ = max dynamic load, lb

$W$ = steady transmitted load, lb

$V$ = pitch-line velocity, fpm

$F$ = width of Face of gears, in

$C$ = deformation Factor (Table 18-6)

① - Find error Factor (Tables 18-4, 18-5) H.O.

② - Find C (Table 18-6) H.O.

5 - Design Formula Under Dynamic Loading

$$S_{ef} = \frac{f W_d}{F p y}$$

where,

$S_{ef}$ = Flexural endurance limit (H.O. Table 18-7)

$W_d$ = dynamic load

$f$ = 1.25 for steady loads

= 1.35 for pulsating loads

= 1.5 for shock loads

6 - Wear of Gear Teeth - Buckingham Equation

$$W_w = DFKQ$$

where,

$W_w$ = limiting load for wear, lb

$Q = 2r/(r+1)$ for external gears

$= 2r/(r-1)$ for internal gears

$r$ = velocity ratio (higher to lower)

$K$ = load-stress factor (Table 18-8, H.O.)
$D$ = pitch diameter of pinion (in)
$S_{es}$ = surface endurance limit, psi (Table 18-7, H.O.)
$\phi$ = pressure angle
$F$ = face width of gears, in
$E_1, E_2$ = moduli of elasticity of materials, psi

and

$$K = \frac{S_{es}^2 \sin\phi}{1.4}\left(\frac{1}{E_1} + \frac{1}{E_2}\right)$$

Example 2 - Determine the Brinell Hardness number (Bhn) for the pinion and gear of example 1 on the basis of:

a- dynamic load
b- wear load

Solution:

From example 1: $T_G = 2100$ lb-in ; $V = 816$ fpm
$p = \frac{\pi}{P} = 0.349$ ; $P = 9$
$D_p = 2.67$ in ; $n = 24$
$D_G = 8$ in ; $n = 72$
$F = 1.25$ in ; $r = 3:1$

a) Dynamic load (For 130% rating)

$W = \frac{2T}{D} = 2 \times 2100/8 = 525$ lb

From table 18-5 for class 2, $P > 6$ ; error = 0.001

From table 18-6 $\left\{\begin{array}{l}\text{steel - steel} \\ 20^\circ \text{ full depth} \\ \text{error} = 0.001\end{array}\right\}$ $c = 1660$

$Q = 2r/(r+1) = 6/4 = 1.5$

$$W_d = W + \frac{0.05V(FC + W)}{0.05V + \sqrt{FC + W}}$$

$$W_d = 525 + \frac{0.05 \times 816 \;(1.25 \times 1660 + 525)}{0.05 \times 816 + \sqrt{125 \times 1660 + 525}}$$

$W_d = 1681$ lb

$$S_{ef} = \frac{f W_d}{F P y}$$

$f = 1.25$ (steady load)

$$S_{ef} = \frac{(1.25)(1681)}{(1.25)(0.349)\,y} = 4817/y$$

For pinion From Table 18-1 H.O. $y = 0.107$

For gear " " $y = 0.137$

$\therefore$ For pinion $S_{ef} = 4817/0.107 \simeq 45000$ psi

For gear $S_{ef} = 4817/0.137 \simeq 35200$ psi

and Table 18-7:

Bhn For pinion = 200

Bhn For gear = 150

b) wear load : (100 % rating)

$$W = \frac{2T}{D} = \frac{2 \times 2100}{1.3 \times 8} = 404 \text{ lb}$$

$$W_d = 404 + \frac{0.05 \times 816\,(1.25 \times 1660 + 404)}{0.05 \times 816 + \sqrt{1.25 \times 1660 + 404}} = 1520 \text{ lb}$$

Pinion : $$K = \frac{W_w}{DFQ} = \frac{1520}{DFQ} = \frac{1520}{2.67 \times 1.25 \times 1.5} = 300$$

From table 18-8 Bhn = 400

Gear : $$K = \frac{1520}{8 \times 1.25 \times 1.5} = 100$$

From table 18-8 Bhn = 200