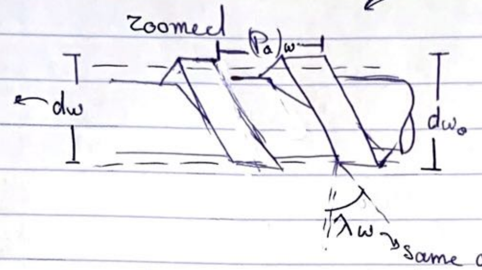
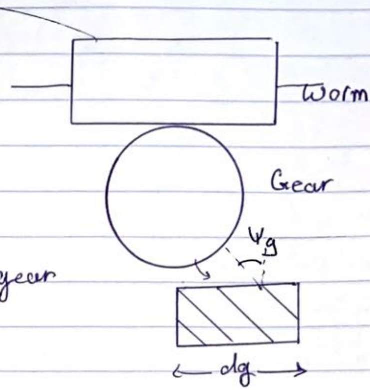
Warm Gear

- Gives high velocity ratio but higher friction (low efficiency)

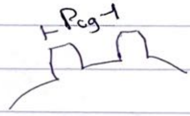
worm : power screw



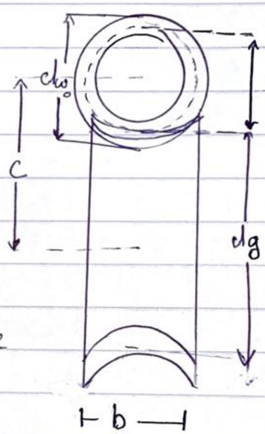


$\lambda_w = \psi_g$ and same hand

$$C = \frac{d_w + d_g}{2}$$



$$(P_a)_w = (P_c)_g$$



$$V_R = \frac{\omega_w}{\omega_g} = \frac{N_g}{N_w} \neq \frac{d_g}{d_w}$$

N_w : عدد السنات اللي على ال worm = Single, double, ...

Number of teeth (N_w) and d_w are not related

But in gear $P_c = \pi \frac{d_g}{N_g}$

$b \leq 0.5 (d_w)_o$, Recommended value

$d_{w_o} = d_w + 2a$

$(d_g)_o = d_g + 2a$

Recommended value: $\frac{C^{0.875}}{3''} < d_w < \frac{C^{0.875}}{1.7''}$ in inch

where: C: center distance

$\frac{C^{0.875}}{2.002} < d_w < \frac{C^{0.875}}{1.068}$ in mm

$N_g > 21$, $N_g + N_w > 40$

$(P_d)_w$ has recommended values (P 809)

Lead Angle for the worm: $\tan^{-1} \frac{L}{\pi d_w} = \lambda_w$

d_w → Average not outside

Lead = $L = N_w P_a$ — N_w: single, double ---

$P_a = \frac{\pi}{P}$ — P → transverse (Given)

Addendum, dedendum from Table 13-5 P690

$d_{dedu} = d_w - 2b$

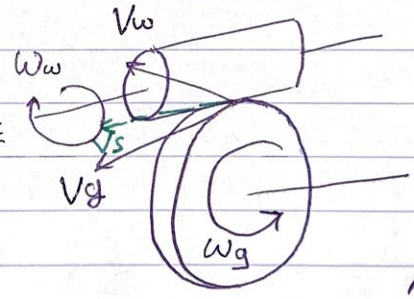
Relation between lead Angle and Normal Pressure Angle from Table 15-9 to prevent interference (The Given is λ_{max} and $\lambda < \lambda_{max}$)

Force Analysis of warm gear

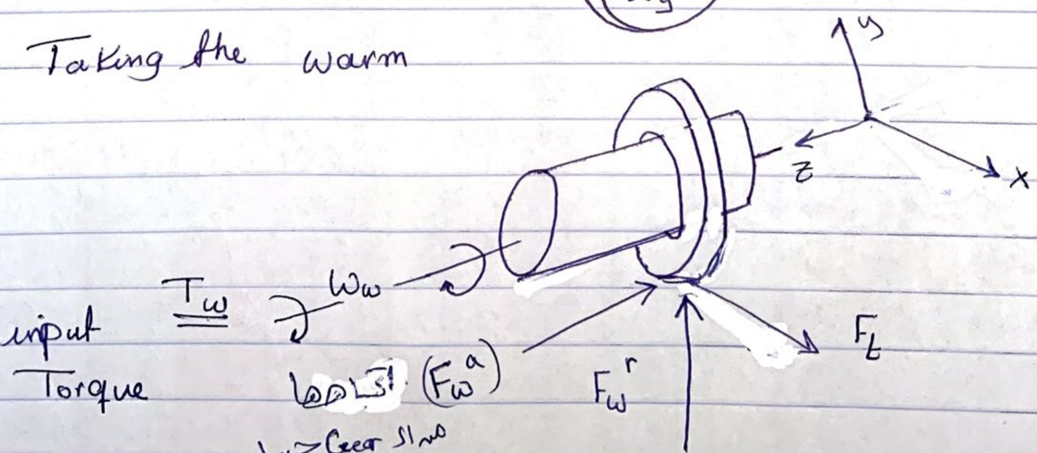
Right hand worm, clock-wise Rotation

Worm is driver

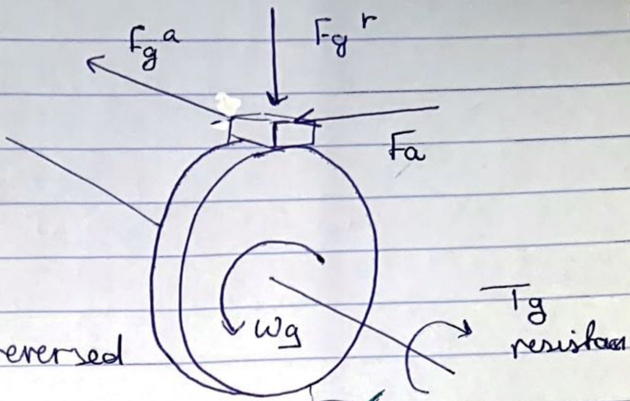
(left hand opposite)



Taking the warm



$F_w^t = F_g^a$
$F_w^r = F_g^r$
$F_w^a = F_g^t$



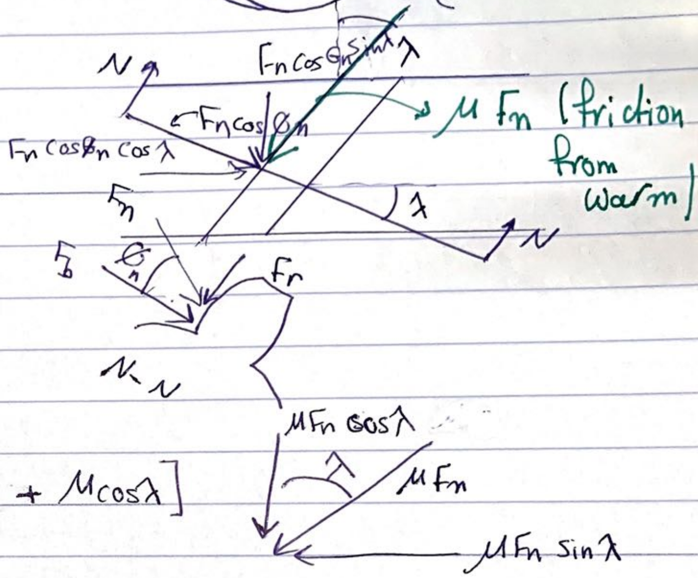
- If left hand, ω_g is reversed and F_g^t is reversed

For the warm gear

warm is Driver

$F_b = F_n \cos\phi_n$

$F_r = F_n \sin\phi_n$



$$F_g^t = F_w^a = F_n\left[\cos\phi_n \cos\lambda - \mu \sin\lambda\right]$$

$$F_g^a = F_w^t = F_n\left[\cos\phi_n \sin\lambda + \mu\cos\lambda\right]$$

$$F_g^r = F_n \sin\phi_n = F_w^r$$

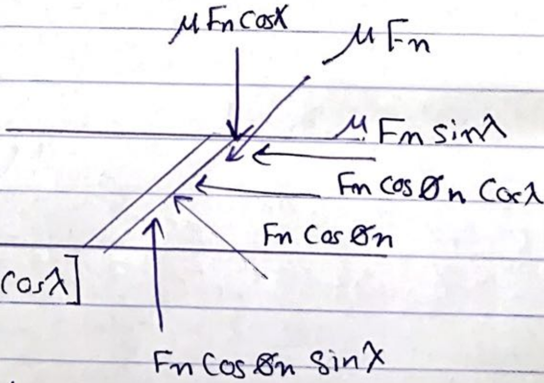
$$\frac{F_g^t}{F_w^t} = \frac{\cos\phi_n\cos\lambda - \mu\sin\lambda}{\cos\phi_n \sin\lambda + \mu\cos\lambda}$$

$$F_g^r = \frac{F_g^t \sin\phi_n}{\cos\phi_n \cos\lambda - \mu\sin\lambda} = \frac{F_w^t \sin\phi_n}{\cos\phi_n\sin\lambda + \mu\cos\lambda}$$

If the gear is the Driver

Does the worm move?

$F_w^t = ?$



$$F_w^t = F_g^a = F_n\left[\cos\phi_n \sin\lambda - \mu\cos\lambda\right]$$

If $F_w^t > 0$: moves

if $F_w^t < 0$: Self locked $\rightarrow \mu > \cos\phi_n \tan\lambda$

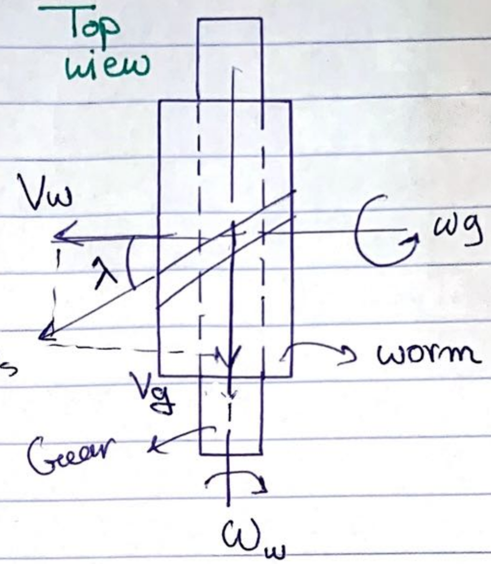
Velocities

$\tan\lambda = V_g / V_w$

$\sin\lambda = V_g / V_s$

$\cos\lambda = V_w / V_s$

V_s : friction جوّه



$$e_w \text{ (efficiency)} = \frac{F_g^{\,t}\ V_g}{F_w^{\,t}\ V_w}$$

$\leftarrow$ output

from Previous

Input

$$= \left(\frac{\cos\phi_n \cos\lambda - \mu \sin\lambda}{\cos\phi_n \sin\lambda + \mu \cos\lambda} \right) \frac{V_g}{V_w} = \frac{\cos\phi_n - \mu \tan\lambda}{\cos\phi_n + \mu \cot\lambda}$$

$\rightarrow \tan\lambda$

If the gear is Driver

$$e_g = \frac{\cos\phi_n - \mu \cot\lambda}{\cos\phi_n + \mu \tan\lambda}$$

Figure 13-42 : used to find friction coefficient
curve B (firiction) < curve A

A, B depends on material

Allowable tangential force on worm gear

Contact stress

$(F_g^t)_{all} = C_s \, C_m \, C_v \, (d_g)^{0.8} \, b$

d_g : diameter of gea

b : face width of gear

at strength = stress يعني أكبر الضغط المسموح به

AGMA so $F_g^t{}_{app} < F_g^t{}_{all}$

C_s : material factor : depends on material (strength)
From Equations : 15-32, 15-33, 15-34 and 15-35

Note D_m = mean diameter of gear = d_g

C_v : velocity factor
From Equation : 15-37

C_m : velocity ratio factor : $m_g = \frac{\omega_w}{\omega_g} = \frac{N_g}{N_w}$

Equation 15-38 → f for gear made of bronze (or use curve)

How to find if it is safe?

$F_g^t = H_g \frac{33000}{V_g} \, n \, K_a$

n : design factor

Ka : application factor

$n = \frac{(F_g^t)_{all}}{(F_g)^t \, K_a}$

$H_{g\,all} = (F_g^{\,t})_{all}\, V_g$

$H_g = (F_g^{\,t} V_g)$

so $H_{g\,all} = \underline{H_{nominal}}\, K_a\, n$

Given or wanted

Wear Stress

Buckingham $(F_g^{\,t})_{all} = \underline{K_w}\, \overset{\text{Given}}{d_g}\, b$

wear factor From Table 15-11

steel 250 – 500 (3rd raw)

Example: Worm steel $H_B = 300$
Gear Chilled Bronze } Go to row 3

1st, 2nd rows : Hardness > 500

This is also compared to $(F_g^{\,t})$ applied

Bending stress

$$\sigma_b = \frac{W_g^{\,t}}{P_n\, b\, y} \qquad W_g^{\,t} \to F_g^{\,t}$$

$P_n = P\cos\lambda$

y : lewis factor

$\phi_n = 14.5 \to y = 0.1$

$\phi_n = 20 \to y = 0.125$

$\phi_n = 25 \to y = 0.15$

$\phi_n = 30° \to y = 0.175$

$\sigma_b = S_e \to (F_{tg})_{all}$

- We obtain F_g^t all from theree ways wear, contact, bending → we take the smallest and Power is calculated

Worm gear thermal capacity

Friction causes heat

$T < 200°F\ (93°C)$ → satisfactory operation

We need to dissipate heat

$H_T = C_t\, A\, (t_o - t_a)$

Heat transfer coefficient

Surface Area

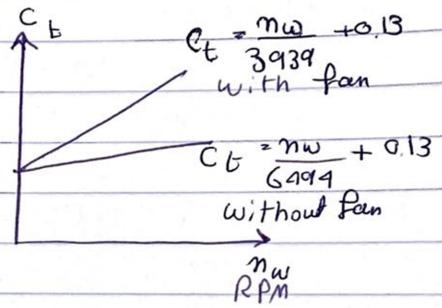
oil temp

ambient temp

$H_T \begin{cases} \frac{ft.lb}{min} \\ W \end{cases}$

$A \begin{cases} ft^2 \\ m^2 \end{cases}$

To find C_t



t_o between: $T_o = 160 - 200F°$

Arear is Given or estimated

$A = 0.3\, C^{1.7}$

In ft^2

center distance in in

$A = 43.2\, C^{1.7}$

In in^2

To calculat friction:

$H_f = H_w - H_g \sim e_w H_w$

$e = \frac{H_{out}}{H_{in}} = \frac{H_{in} - H_f}{H_{in}}$

$H_f \simeq H_T$ (Assumed)

$H_f = F_f V_s$

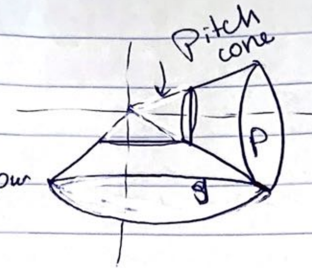
$= \mu F_n = \mu \frac{F_t}{\cos\phi_n \cos\lambda - \mu \sin\lambda}$

Bevel Gears

Geometry:

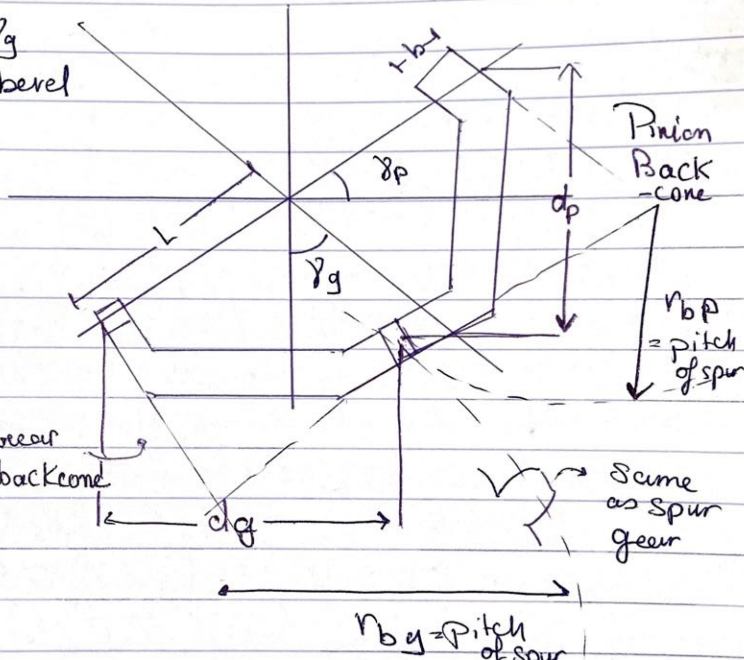
Assumed to be two Cones intersecting

Pitch cones of bevel gear are analogus to pitch cylinder of spur Gear



$d = \frac{N}{P}$

Σ (shafts) $= 90 = \gamma_p + \gamma_g$
in case of straight bevel



$\gamma_p = \tan^{-1}\left(\frac{d_p}{d_g}\right)$

$= \tan^{-1}\left(\frac{N_p}{N_g}\right)$

$\frac{\omega_p}{\omega_g} = \frac{d_g}{d_p} = \frac{N_g}{N_p}$

$= \tan\gamma_g = \cot\gamma_p$

$L = \frac{d_g}{2\sin\gamma_g}$

Involute in back cone similar to spur gea with radius = r_b
Virtual no of teeth = N'

$P = \frac{2\pi r_{bP}}{N'}$

$P = \frac{N_g'}{2r_{bg}} = \frac{N_p'}{2r_{bp}}$

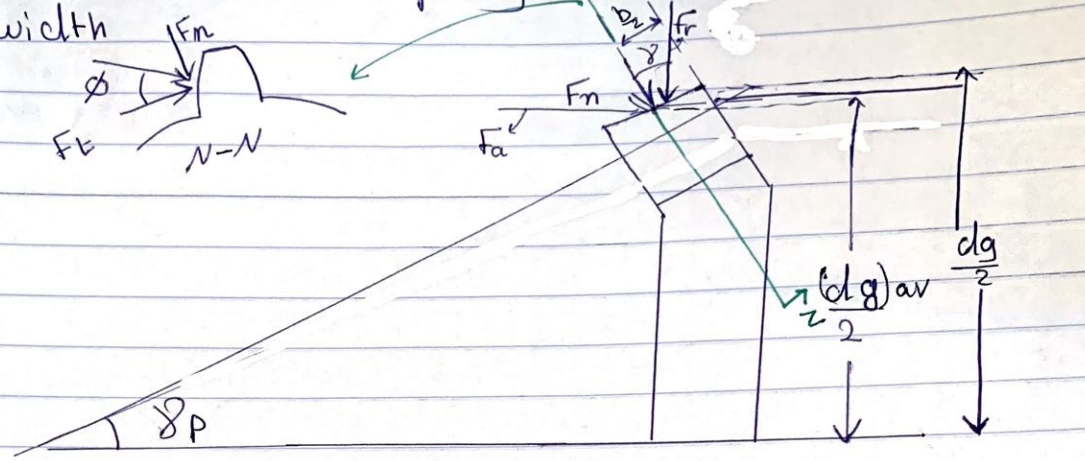
limitations of b

$b \le \frac{L}{3}$ element

$b \le \frac{10}{P}$

Force analysis of Bevel gear

load assumed uniformly distributed along fac width



$d_{av} = d - b \sin\gamma$

$V_{av} = \left(\frac{\pi d_{av} n}{12}\right)$ for hp $= \left(\frac{\pi d_{av} n}{60}\right)$ for W

$F_t \begin{cases} = \frac{W}{V_{av}} \text{(watts)} & \text{lb} \\ \frac{33000 \text{(hp)}}{V_{av}} & \text{N} \end{cases}$

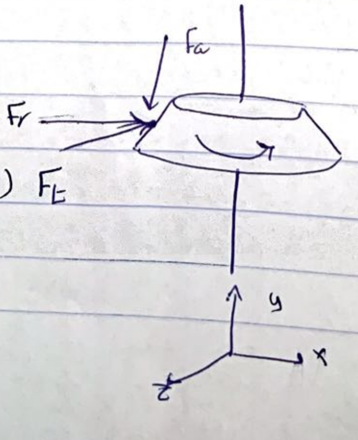
$F_n = F_t \tan\phi$ (lec 3 (last Questio

$F_a = F_n \sin\gamma = F_t \tan\phi \sin\gamma$

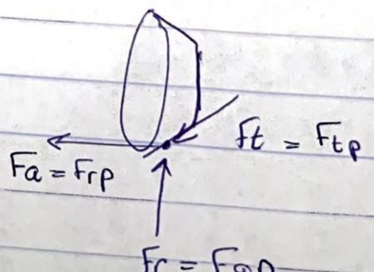
$F_r = F_n \cos\gamma = F_t \tan\phi \cos\gamma$

Direction

(دائماً عكسه) F_t



Fa دائماً تبعد الـ gear لبرا



Design of bevel gears

Bending

$$\sigma = \frac{F_t P}{b J} K_o K_m K_v K_s$$

K_o: over load factor $\longrightarrow$ Table (15-2)

J: Figure [15-7] , $\Sigma = 90$, $\phi = 20$

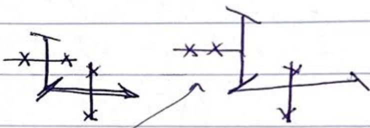
K_m = mounting factor

$K_m = K_{mb} + 0.0036\, b^2$ b = in

$K_m = K_{mb} + 5.6 \times 10^{-6}\, b^2$ b = mm

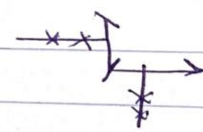
$K_{mb} = 1$ if Both Straddle

$K_{mb} = 1.1$ one gear straddle, other overhang



= overboard — over hung (Both)

$K_{mb} = 1.25$



$K_v \rightarrow$ fig [15-5]

K_s = Size factor   Equation 15-10

Y_s same but metric

Strength =

$$S_e = \frac{K_L K_t S_t}{K_r}$$

at $L = 10^7$, $R = 99\%$

S_t: Table 15-6, fig 15-13 (Through hardened)

K_L: fig 15-9 or Eq 15-15

$K_L \rightarrow Y_N$ metric

K_r: Table 15-3 or Eq 15-19, 20

$K_t = \dfrac{710}{160 + T}$ $T > 250\,°F$

$K_t = 1$ $T < 250\,°F$

$K_t = \dfrac{393}{273 + T}$ $T > 120\,°C$

Contact

$$\sigma_c = C_p \sqrt{\frac{F_t \; C_o C_m C_v C_s C_c}{b \, I \, d_p}}$$

$C_o = K_o$

$C_m = K_m$

$C_v = K_v$

C_s = size factor from Eq 15-9

C_c = crown factor from Eq 15-12

1.5 crowned — 2 uncrowned

I from fig 15-6 (what I used as Pinion here is considered to be d_p)

$C_p = 2290 \sqrt{psi}$ — steel-steel

$= 190 \sqrt{MPa}$

or If not steel-steel $C_p = \sqrt{\dfrac{1}{\pi \left(\dfrac{1-\nu_p^2}{E_p}\right) + \left(\dfrac{1-\nu_g^2}{E_g}\right)}}$

Posons

Strength

$S_{fc} = S_c \dfrac{C_t C_L C_H}{C_R}$ C_H used for gear — same spur as gears taken before

C_L = figure 15-8

S_c Table 15-4 fig 15-12 through hardened

$C_t = K_t$

C_R = Table [15-3] or eq [15-20], [15-19]

$= \sqrt{Y_R}$