load factor $K_c = 1$.

Bending $\rightarrow$ $\sigma_b \rightarrow K_f \cdot \sigma_b$ $\rightarrow$ fatigue SCF

Axial $\rightarrow$ $\sigma_{axial} \rightarrow K_f \cdot \sigma_{axial}$

Torsion/ shear : $\tau \rightarrow K_{fs} \cdot \tau$

$\hookrightarrow$ fatigue SCF in shear

Calculate von mises stress

$$\sigma' = \left\{\left(K_f \sigma_{bending} \pm \frac{K_f \cdot \sigma_{axial}}{0.85}\right)^2 + 3(K_{fs} \cdot \tau)^2\right\}^{1/2}$$

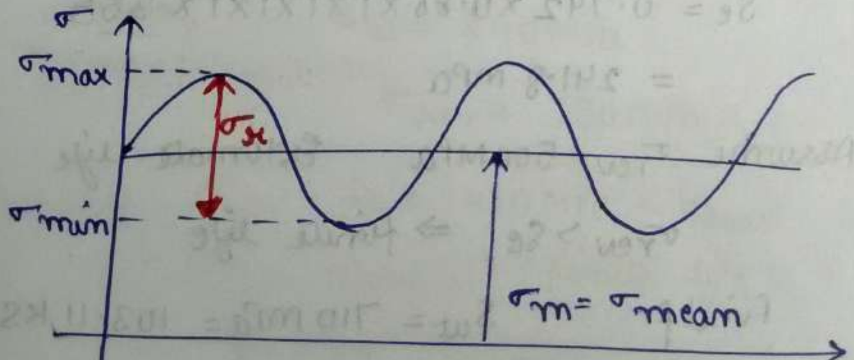
takes into account $K_c = 0.59$ implicitly

Fatigue under fluctuating load

- Load is not completely reversible.

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} \neq 0.$$

- S-N diagram works only when $\sigma_{mean} = 0$.



σ_{max} := maximum stress

σ_{min} := minimum stress

$\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2}$:= amplitude component

$$\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} := \text{midrange component}$$

$$\sigma_r = (\sigma_{max} - \sigma_{min}) = \text{range of stress}$$

Notch Sensitivity:

Practise : Before yielding,

$\sigma_m = \sigma_{m0} \cdot K_f$ — Nominal values

$\sigma_a = \sigma_{a0} \cdot K_f$ — same factor K_f to amplitude and midrange component

After yielding,

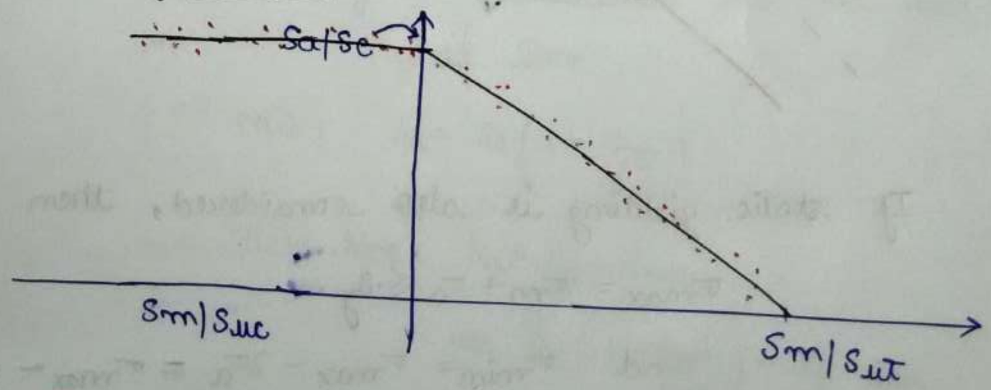
$\sigma_a = K_f \cdot \sigma_{a0}$

$\sigma_m = \sigma_{m0}$ (No K_f on σ_{m0})

Modified Goodman Criterion

Observation: S_a, S_m are limiting values of amplitude and midrange components of stress.

If $N > 10^6$:



M-G eqn.

$$\frac{S_a}{S_e} + \frac{S_m}{S_{ut}} = 1 \quad \text{for } \sigma_m > 0$$

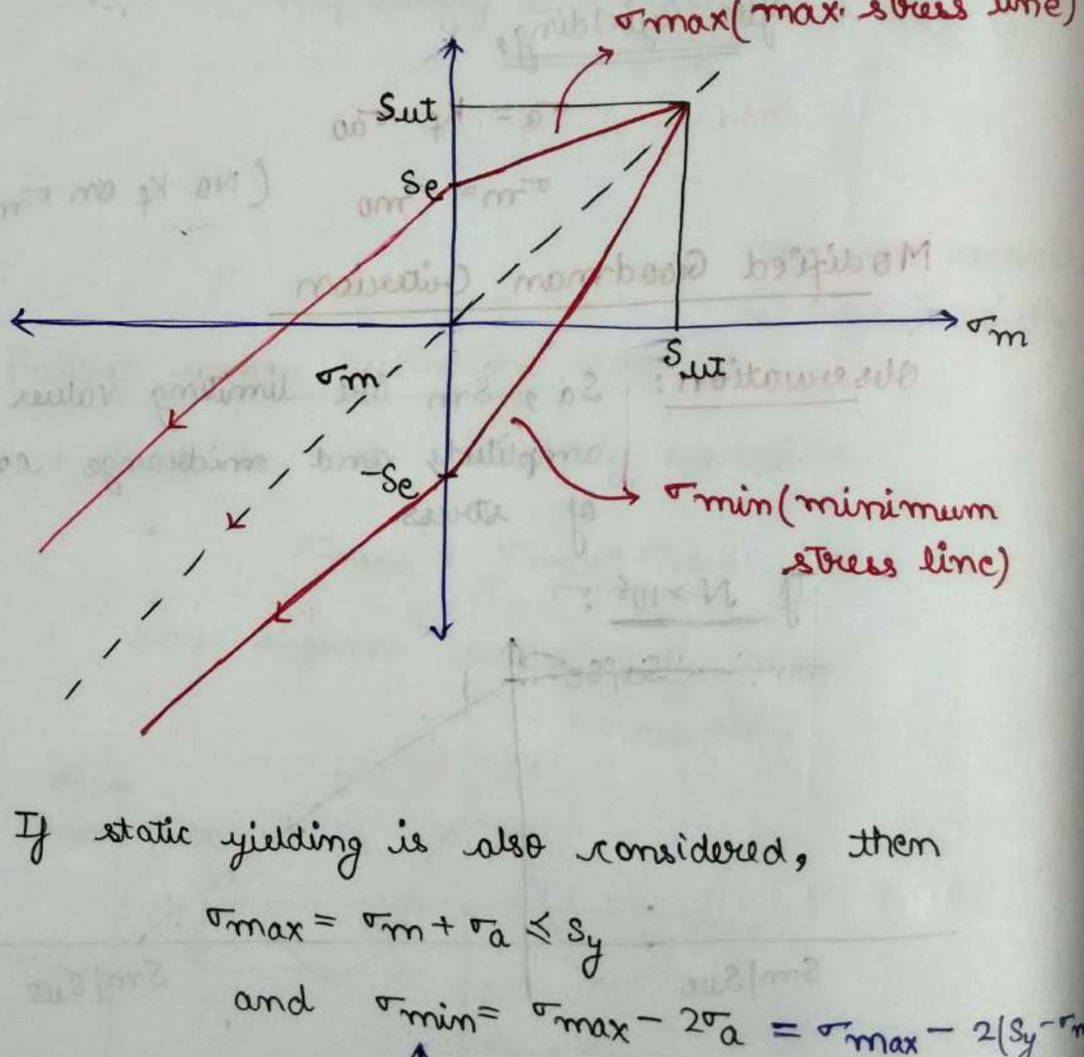
or $$\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = 1 \quad \text{— " —}$$

and $\frac{S_a}{S_e} = 1$ or $\frac{\sigma_a}{S_e} = 1$ for $\sigma_m < 0$.

$$\sigma_a = S_e\left(1 - \frac{\sigma_m}{S_{ut}}\right) \quad \text{for } \sigma_m > 0$$

or $\sigma_{max} = \sigma_m + \sigma_a = S_e + \sigma_m\left(1 - \frac{S_e}{S_{ut}}\right)$

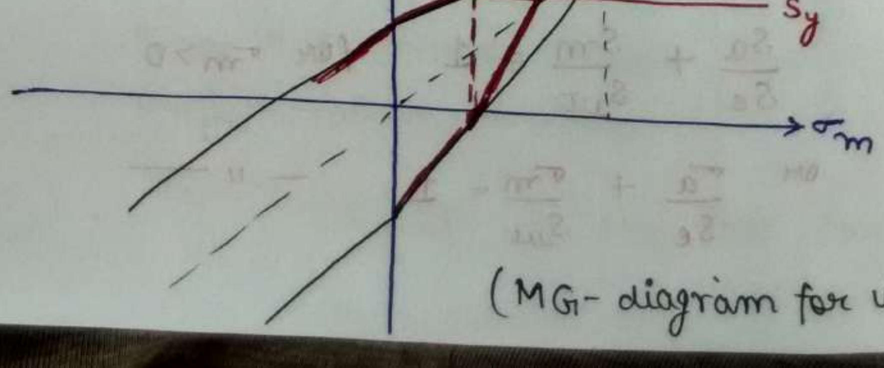
$\sigma_{min} = \sigma_m - \sigma_a = -S_e + \sigma_m\left(1 + \frac{S_e}{S_{ut}}\right)$



If static yielding is also considered, then

$\sigma_{max} = \sigma_m + \sigma_a \leq S_y$

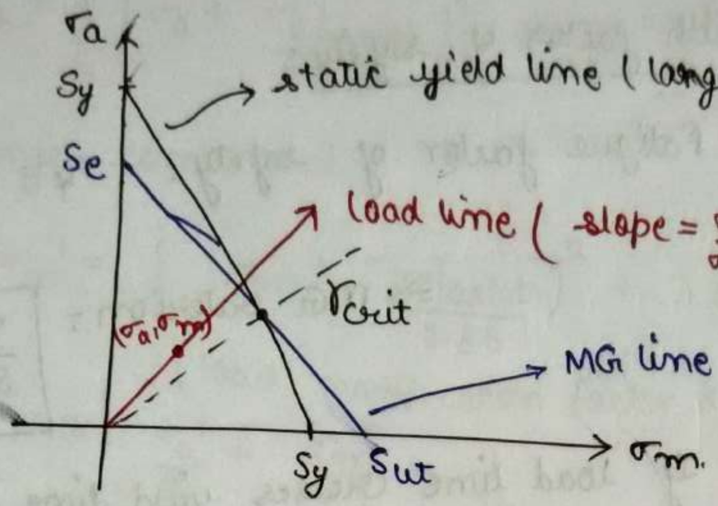
and $\sigma_{min} = \sigma_{max} - 2\sigma_a = \sigma_{max} - 2(S_y - \sigma_m)$



(MG- diagram for infinite life)

For finite life: replace S_e by S_f.

MG diagram combined with static yield (for $\sigma_m > 0$)



Load line slope:

$$r = \frac{\sigma_a}{\sigma_m}$$

If $r > r_{crit} \Rightarrow$ fatigue failure

$r < r_{crit} \Rightarrow$ static yielding

Load line: $r = \frac{\sigma_a}{\sigma_m} = \frac{S_a}{S_m}$

for r_{crit}: find intersection betn MG line and yield line

MG: $S_a = S_e\left(1 - \frac{S_m}{S_{ut}}\right)$

Yield line: $S_a = S_y - S_m$.

$$\Rightarrow S_e\left(1 - \frac{S_m}{S_{ut}}\right) = S_y - S_m$$

$$\Rightarrow S_e - S_y = S_m\left(\frac{S_e}{S_{ut}} - 1\right)$$

$$\Rightarrow S_m = \frac{S_y - S_e}{S_{ut} - S_e} \times S_{ut}$$

$$\Rightarrow S_a = \frac{(S_{ut} - S_y) S_e}{S_{ut} - S_e}$$

$$\boxed{r_{crit} = \frac{S_a}{S_m} = \frac{S_{ut} - S_y}{S_y - S_e} \times \frac{S_e}{S_{ut}}}$$

with factor of safety:

Fatigue factor of safety: $n_f = \frac{\sigma_a}{S_e} = \frac{\sigma_m}{S_m}$

$\Rightarrow$ MG criterion: $\boxed{\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_{ut}} = \frac{1}{n_f}}$

If load line crosses yield line first:

$$\sigma_{max} = \frac{S_y}{n}$$

$$\Rightarrow \boxed{\frac{\sigma_a}{S_y} + \frac{\sigma_m}{S_y} = \frac{1}{n}}$$

For pure torsional loading:

Use MG criterion with

$$\boxed{\begin{aligned} S_{sy} &= 0.577 S_y \\ S_{su} &= 0.67 S_{ut} \\ \text{or } K_c &= 0.59 \end{aligned}}$$

combined loading:

Bending: $\sigma_a = K_f \cdot \sigma_{ao}$ (fatigue SCF)

$\sigma_m = K_f \cdot \sigma_{mo}$

Axial: $(\sigma_a)_{axial} = K_f \cdot (\sigma_{ao})_{axial}$

$(\sigma_m)_{axial} = K_f \cdot (\sigma_{mo})_{axial}$

Torsion/ shear:

$\tau_a = K_{fs} \cdot \tau_{ao}$

$\tau_m = K_{fs} \cdot \tau_{mo}$

Von mises stress:

Alternating component

$$\sigma_a' = \left\{ \left(\sigma_a + \frac{\sigma_{a\,axial}}{0.85} \right)^2 + 3\tau_a^2 \right\}^{1/2}$$

midrange component

$$\sigma_m' = \left\{ \left(\sigma_m + \frac{\sigma_{m\,axial}}{0.85} \right)^2 + 3\tau_m^2 \right\}^{1/2}$$

(load modification factor K_c is incorporated)

MG: $\frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}} = \frac{1}{n_f}$

$\Rightarrow$ choose load factor $K_c = 1$ for calculating S_e

Example → loading

$\sigma_{bending} = 200$ to 260 MPa

$\tau = -120$ to 120 MPa

$\sigma_{axial} = -100$ MPa

} After correcting for notch sensitivity

Material: $S_{ut} = 1200$ MPa $S_y = 900$ MPa

$S_e = 400$ MPa (fully corrected)

Calculate n_f for infinite life

Solution →

	Alternating (MPa)	Mid Range (MPa)
Bending	30	230
Torsion	120	0
Axial	0	-100

Von mises

$$\sigma_a' = \left\{ \left(30 + \frac{0}{0.85}\right)^2 + 3\times 120^2 \right\}^{1/2} = 210\ \text{MPa}$$

$$\sigma_m' = \left\{ \left(230 - \frac{100}{0.85}\right)^2 + 3\times 0^2 \right\}^{1/2} = 112.35\ \text{MPa.}$$

load line : $r = \frac{\sigma_a'}{\sigma_m'} = \frac{210}{112.35} = 1.866$

$$r_{crit} = \frac{S_{ut} - S_y}{S_y - S_e} \times \frac{S_e}{S_{ut}}$$

$$= \frac{1200 - 900}{900 - 400} \times \frac{400}{1200} = 0.2$$

since, $r > r_{crit} \Rightarrow$ fatigue failure

$$\Rightarrow \frac{1}{n_f} = \frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}}$$

$$\Rightarrow n_f = \left(\frac{210}{400} + \frac{112.35}{1200} \right)^{-1}$$

$$\Rightarrow n_f = 1.62$$

Cumulative fatigue damage

Fully reversible

stress level : σ_1 for n_1 cycles $\rightarrow$ cycle of operation

N_1 be the life at stress σ_1.

After n_1 cycles at σ_1 apply stress σ_2 for n_2 cycles

N_2 be the life at stress σ_2.

Example $\Rightarrow$ At σ_1, $N_1 = 10^4$ cycles

$n_1 = 3\times 10^3$ cycles

life remaining at σ_1 : $N_1 - n_1$

$= 7\times 10^3$ cycles

Now let $\sigma_2 = S_e$, $N_2 = 10^6$ cycles

what is the remaining life n_2?

Palmgreen - Miner rule:

Damage gets accumulated at every cycle of operation

- Damage variable D.

stress level σ_1 : $D_1 = \frac{n_1}{N_1}$, stress level: σ_2 $D_2 = \frac{n_2}{N_2}$

stress level σ_i $D_i = \frac{n_i}{N_i}$

Total / cumulative damage

$$D = D_1 + D_2 + \dots + D_i$$

For failure it is fully damaged

$\Rightarrow D = 1.$

$\Rightarrow$ For failure

$$\boxed{D = \frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \dots + \frac{n_i}{N_i} = 1}$$

Eg. $n_1 = 3\times10^3$ $N_1 = 10^4$

$n_2 = ?$ $N_2 = 10^6$

$$\Rightarrow 1 = \frac{n_1}{N_1} + \frac{n_2}{N_2}$$

$$\Rightarrow 1 = \frac{3}{10} + \frac{n_2}{10^6} \quad \Rightarrow n_2 = 7\times10^5 \text{ cycles}$$

(Not $10^6 - 3000$ cycles)

Eg $\rightarrow$ For a machine part

$S_{ut} = 595$ MPa, $f = 0.86$

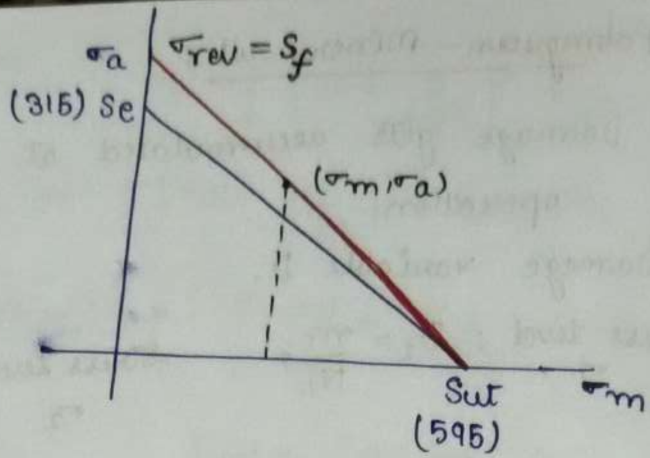
Fully corrected endurance strength

$S_e = 315$ MPa.

Loading: $\sigma_a = 245$ MPa, $\sigma_m = 210$ MPa. for 12×10^3 cycles.

Find remaining life at $\sigma_a = S_e$, $\sigma_m = 0$.

Sol. →



Load point is beyound MG line

$\Rightarrow$ finite life

→ Draw a straight line joining $(S_{ut}, 0)$ & (σ_m, σ_a)

$\Rightarrow$ Inter-section with y-axis gives $\sigma_{rev} = S_f$.

$$\Rightarrow \frac{\sigma_{rev}}{S_{ut}} = \frac{\sigma_a}{S_{ut} - \sigma_m} \Rightarrow S_f = \sigma_{rev} = \frac{\sigma_a}{S_{ut}} \times \frac{1}{(S_{ut} - \sigma_m)}$$

Eg → $S_f = 378.6$ MPa > 315 MPa $= S_e$

$\Rightarrow$ finite life.

$f \times S_{ut} = 0.86 \times 595 = 511.7$ MPa $> S_f$

$\Rightarrow$ High cycle Fatigue

$\Rightarrow N_1 =$ life at $\sigma_{rev} = S_f$ (completely reversible)

S-N diagram

$$N_1 = \left(\frac{\sigma_{rev}}{a}\right)^{1/b} = \left(\frac{S_f}{a}\right)^{1/b}$$

$$a = \frac{(f \cdot S_{ut})^2}{S_e} \qquad b = -\frac{1}{3}\log_{10}\left(\frac{f \cdot S_{ut}}{S_e}\right)$$

$\Rightarrow a = 831.23 \qquad b = -0.07$

$\Rightarrow N_1 = 75 \times 10^3$ cycles

$(\sigma_m, \sigma_a) = (245, 210) \equiv \sigma_1 = S_f = 378.6$ MPa

$n_1 = 12 \times 10^3$

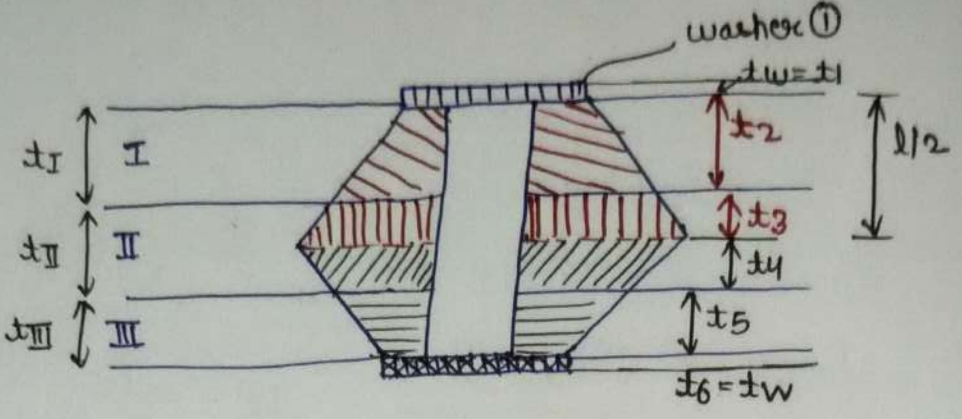
$N_1 = 75 \times 10^3$

Miner: $\frac{n_1}{N_1} + \frac{n_2}{N_2} = 1$ $\Rightarrow \frac{n_2}{N_2} = 1 - \frac{12 \times 10^3}{75 \times 10^3}$

$\Rightarrow n_2 = 8.4 \times 10^5$ cycles.

(remaining life)

Pressure cone



Member 1 (washer)

$D = 1.5\,d$

$t_1 = t_w$

$E_1 = E_w.$

Member 2:

$D_2 =$ calculate $=$ Top dia of seg. 2

$t_2 = t_I$ $\quad = D + 2t_1 * \tan\alpha$

$E_2 = E_I$

Member 3:

$D_3 =$ top dia of seg 3

$=$ bottom dia of seg 2

$= D_2 + 2t_2 \tan\alpha$

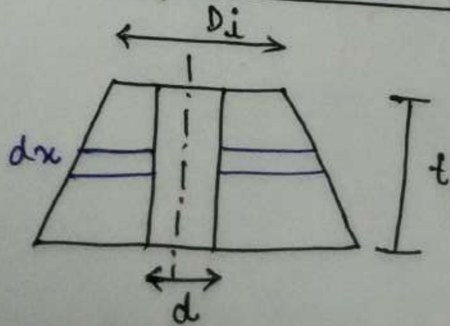
$E_3 = E_{II}$

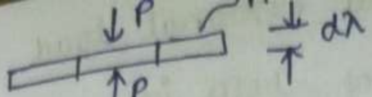
$t_3 = \frac{l}{2} - (t_1 + t_2)$

$= \frac{l}{2} - (t_w + t_I)$

Similarly for members 4, 5, 6.

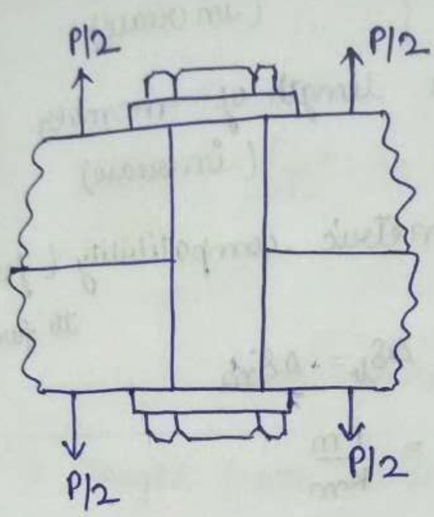
For any member segment - i





$$d\delta = \frac{P}{AE} \times dx$$

Total elongation
$$\delta = \int_0^{ti} \frac{P}{AE}\,dx$$
$$= \frac{P}{E} \int_0^{ti} \frac{dx}{A}$$

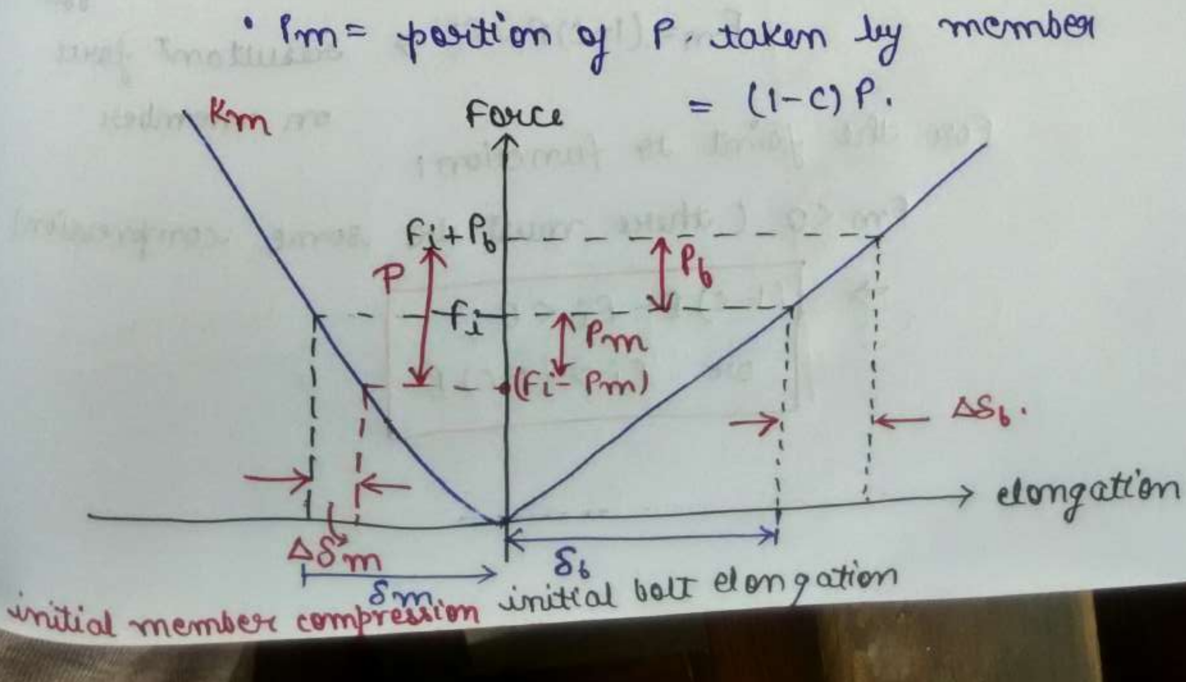


F_i = initial pre load per bolt

K_b = bolt stiffness

K_m = member stiffness

- upon application of external load P
  - bolt tension increases
  - member compr^n decreases.
- Bolts and members → two equivalent springs in parallel.
  - P_b = portion of P taken by bolt = CP
- P_m = portion of P taken by member = (1−C) P.



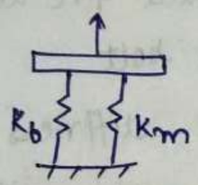
P_{total} = total external load

N = number of bolts in a joint

$P = \frac{P_{total}}{N}$ = external load per bolt
(+ pressure cone)

upon application of P

- $\Delta\delta_b \rightarrow$ change in length of bolt (increase)
- $\Delta\delta_m \rightarrow$ change in length of member (increase)



For geometric compatibility (for joint to function)

$$\Delta\delta_b = \Delta\delta_m$$

$$\Rightarrow \frac{P_b}{K_b} = \frac{P_m}{K_m}$$

and $P_b + P_m = P$

$$\Rightarrow P_b = \underbrace{\frac{K_b}{K_m + K_b}}_{C} \times P \quad , \quad P_m = \underbrace{\frac{K_m}{K_b + K_m}}_{1-C} \times P$$

$C = \frac{K_b}{K_m + K_b}$:= stiffness constant of the joint

$\Rightarrow F_b = F_i + CP \rightarrow$ resultant force on bolt

$F_m = (1-C)P - F_i \rightarrow$ resultant force on member

For the joint to function:

$F_m < 0$ (there must be some compression)

$$\Rightarrow \boxed{\begin{array}{l} (1-C)P - F_i < 0 \\ \text{or} \quad F_i > (1-C)P \end{array}}$$

How to apply preload on bolt?

- Related to bolt tightening torque
- Torque $T = K F_i d_i$

$K \rightarrow$ torque factor (Table 8-15)

Recommended pre load:

$$F_i = \begin{cases} 0.75 F_p & \text{for non-permanent joint} \\ 0.9 F_p & \text{for permanent joint} \end{cases}$$

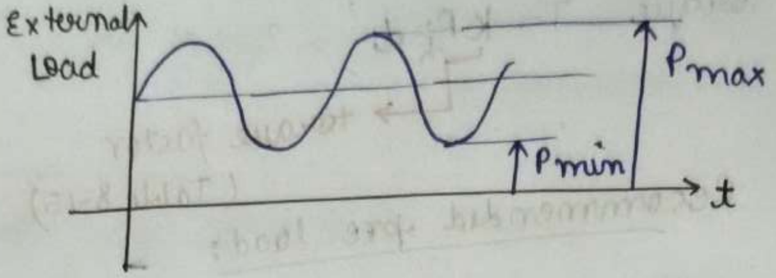
Proof load $\leftarrow F_p = A_t \times S_p \rightarrow$ proof strength (proportional limit)

table 8.9 – 8.11 for steel bolts

$\rightarrow S_p = 0.75 S_y$ for other materials

→ Taught from slides :- Bolt Analysis

Fatigue loading on bolt :



Load on bolt :

$$F_{b\,max} = c P_{max} + F_i$$

$$F_{b\,min} = c P_{min} + F_i$$

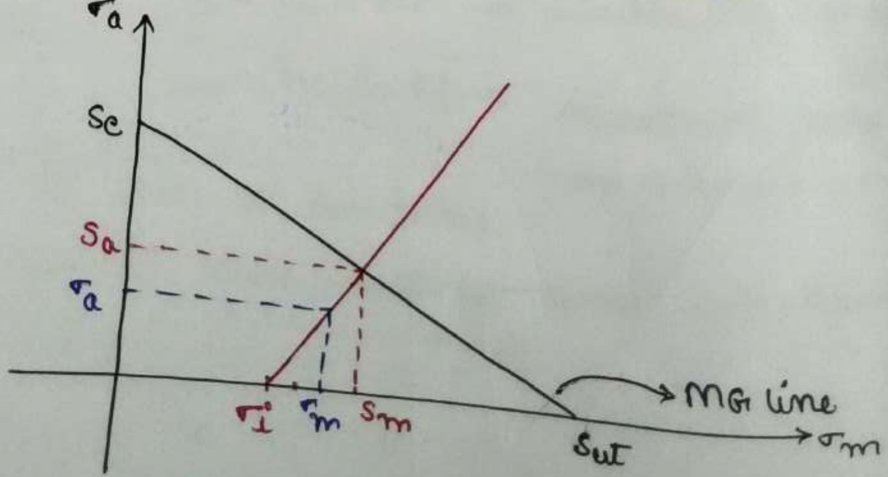
Stresses on bolt :

Alternating : $\sigma_a = \dfrac{F_{b\,max} - F_{b\,min}}{2A_t} = \dfrac{c(P_{max} - P_{min})}{2A_t}$.

Mid Range : $\sigma_m = \dfrac{F_{b\,max} + F_{b\,min}}{2A_t} = \underbrace{\dfrac{F_i}{A_t}}_{\sigma_i} + \dfrac{c(P_{max} + P_{min})}{2A_t}$

When $P_{min} = 0$: $\sigma_a = \dfrac{cP_{max}}{2A_t}$

$$\sigma_m = \sigma_i + \frac{cP_{max}}{2A_t} = \sigma_i + \sigma_a.$$



Fatigue factor of safety

$$n_f = \frac{S_a}{\sigma_a} = \frac{S_m - \sigma_i}{\sigma_m - \sigma_i} \neq \frac{S_m}{\sigma_m}$$

→ increases factor of safety

- Bolt preload improves factor of safety.

- Endurance strength: $S_e = k_a k_b k_c k_d k_e k_f S_e'$

or

$$S_e = \frac{0.85 S_e'}{K_f} \longrightarrow \text{fatigue SCF}$$

K_f ← accounts for notch sensitivity and surface finish

↓

Table 8-16

Eg.

Pressure vessel: internal pressure varies from 0 to p_{max}.

External load on bolt: 0 to P $\left(\begin{matrix} P_{min} = 0 \\ P_{max} = P \end{matrix}\right)$

with preload: $\sigma_a = \frac{CP}{2A_t}$, $\sigma_m = \sigma_a + \sigma_i$

Factor of safety: $n_f = \frac{S_a}{\sigma_a} = \frac{S_m - \sigma_i}{\sigma_m - \sigma_i}$

$$= \frac{S_m - \sigma_i}{\sigma_a}$$

→ Refer to dig. on left side.

MG Line

$$\frac{S_a}{S_e} + \frac{S_m}{S_{ut}} = 1$$

Also, from load line,

$$S_m = \sigma_i + S_a$$

$$\Rightarrow \frac{S_a}{S_e} + \frac{\sigma_i + S_a}{S_{ut}} = 1$$

$$\Rightarrow S_a\left(\frac{1}{S_e} + \frac{1}{S_{ut}}\right) = 1 - \frac{\sigma_i}{S_{ut}} \Rightarrow S_a = S_e\left(\frac{S_{ut} - \sigma_i}{S_{ut} + S_e}\right)$$

$$\Rightarrow n = \frac{S_p}{\sigma_m + \sigma_a} = \frac{S_p A_t}{(F_i + cP)}$$

Let $c = 0.2$, Grade 9-8 bolt M10X1.5

$S_p = 650\,MPa$

$S_{ut} = 900\,MPa$, $A_t = 58\,mm^2$

$S_e = 140\,MPa$ Load: $P = 0$ to $20\,kN$

$$\sigma_a = \frac{cP_{max}}{2A_t} = \frac{0.2 \times 20 \times 10^3}{2 \times 58} = 34.5\,MPa$$

$$\sigma_m = \sigma_i + \sigma_a = 0.9 S_p + \sigma_a$$

$\sigma_i = 0.9 S_p$ (permanent joint)

$$\Rightarrow \sigma_m = 619.5\,MPa$$

$$\Rightarrow S_m = S_{ut} \times \frac{(S_e + \sigma_i)}{(S_{ut} + S_e)}$$

$$S_a = S_m - \sigma_i = 42.4\,MPa > \sigma_a$$

$\Rightarrow$ $\boxed{\text{Infinite life}}$

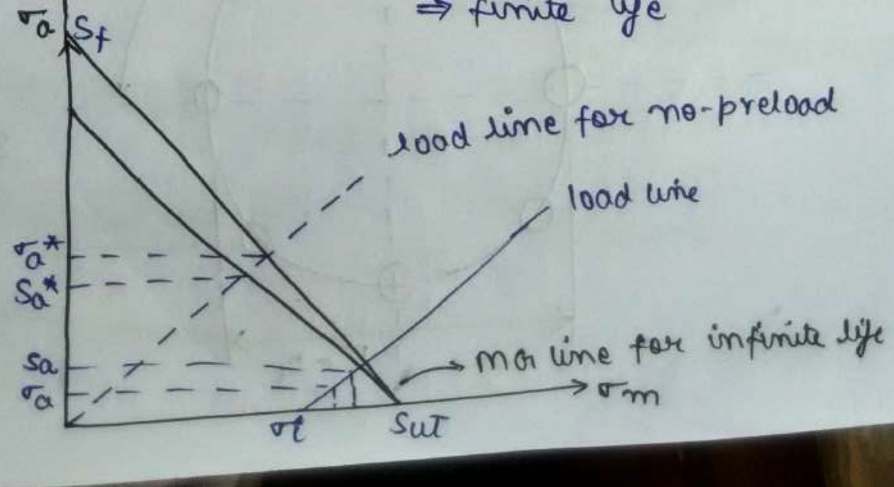
when preload is zero,

$c = 1$, $\sigma_i = 0$.

$$\sigma_m^* = \sigma_a^* = \frac{cP_{max}}{2A_t} = \frac{20 \times 10^3}{2 \times 58} = 172\,MPa$$

$$S_a = \frac{S_e S_{ut}}{S_{ut} + S_e} = 121.2\,MPa < \sigma_a^*$$

$\Rightarrow$ finite life



$$S_a > \sigma_a \Rightarrow \text{infinite life}$$

$$S_a^* < \sigma_a^* \Rightarrow \text{finite life}$$

Eg. Pressure vessel : CI grade 30

Cyl Head : steel

No of bolts : N

Gasket effective dia : D

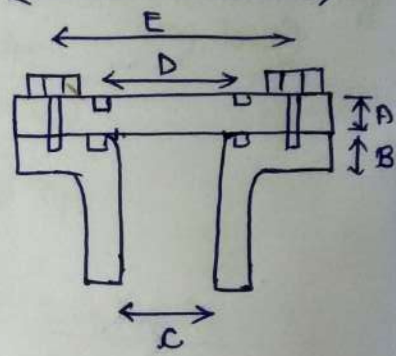
cylinder pressure : P_g

$A = 20$ mm $\quad C = 100$ mm

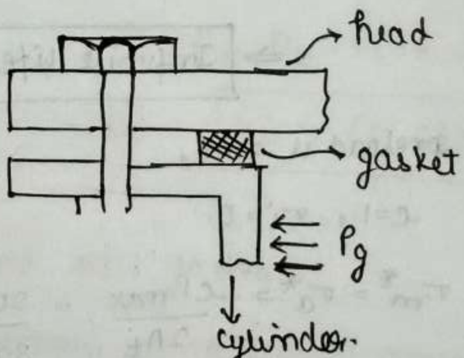
$B = 20$ mm $\quad D = 150$ mm

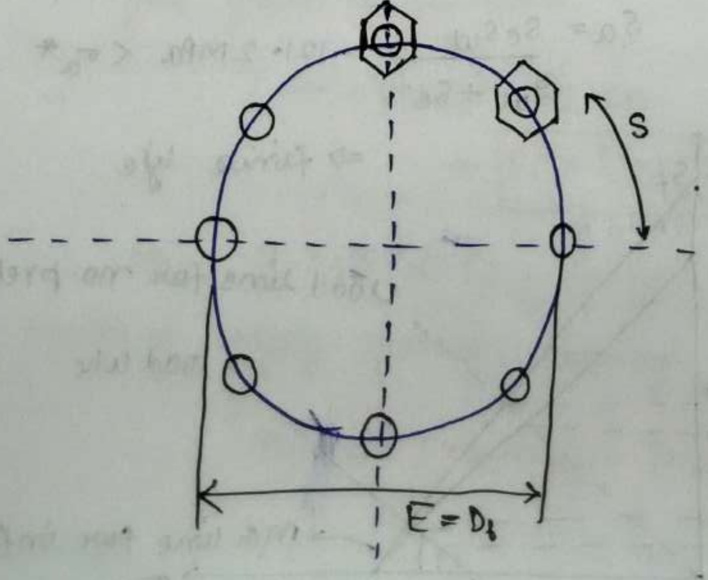
$E = 200$ mm $\quad f = 300$ mm

$N = 10$, $P_g = 6$ MPa.



Gasketed Joint





$D_b = E =$ bolt circle dia.

Bolt spacing

$$S = \frac{\pi D_b}{N} = \frac{\pi E}{N}$$

$$\boxed{3d < \frac{\pi D_b}{N} < 6d}$$ d → dia. of bolt

$3d < \frac{\pi D_b}{N}$: for wrench clearance (tightening)

$\frac{\pi D_b}{N} < 6d$: for uniform gasket pressure.

★ Joint separation $f_m < 0$

At separation, $f_m = 0$

$$\Rightarrow (1-C) P_0 - F_i = 0$$

$$\Rightarrow P_0 = \frac{F_i}{1-C} \Leftarrow \text{load at separation}$$

Factor of safety against joint separation

$$n_0 = \frac{P_0}{P} = \frac{F_i}{(1-C)P}$$

Factor of safety against overloading

$$n_L = \frac{S_p A_t - F_i}{CP}$$

Bolt Length:

grip length: $l = A + B = 40\text{ mm}$

Nut height: $H = 10.8\text{ mm}$ for M12 bolts (Table A31)

$\Rightarrow$ Bolt length: $L > l + H = 50.8\text{ mm}$

Standard value $= 60\text{ mm}$ (Table A17)

Threaded length

$$L_T = 2d + 6 = 30\text{ mm}$$

Unthreaded length $L_d = l_d = L - L_t = 30\text{ mm}$

Threaded length in grip

$$l_t = l - l_d = 10\text{ mm}.$$

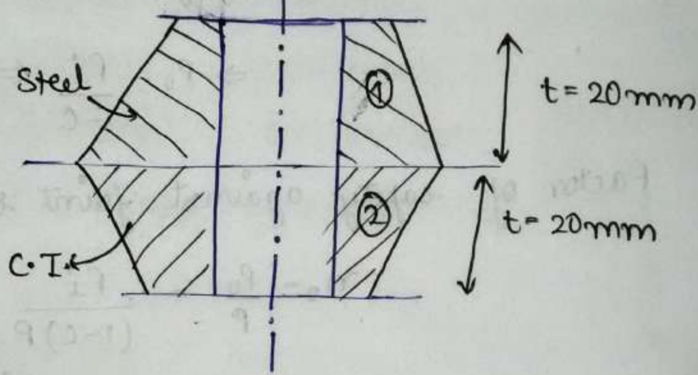
Bolt areas: $A_d = \frac{\pi d^2}{4} = 113.1 \text{ mm}^2$

$A_t = 84.3 \text{ mm}^2$

Bolt stiffness: $K_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$

$= 534 \text{ MN/m}$

Member stiffness:



Pressure-cone

Member 1 →

$$K_1 = \frac{\pi E_1 d \tan\alpha}{\ln\left[\frac{2t\tan\alpha + D - d}{2t\tan\alpha + D + d} \cdot \left(\frac{D+d}{D-d}\right)\right]}$$

$d = 12 \text{ mm}$

$D = 1.5d = 18 \text{ mm}$

$\alpha = 30°$

$t = 20 \text{ mm}$

$E_1 = 205 \text{ GPa}$ (steel)

$\Rightarrow K_1 = 4426.95 \text{ MN/m}.$

Member 2 → Only E_2 is different. $E_2 = 100 \text{ GPa}$

$K_2 = K_1 \frac{E_2}{E_1} = 4426 \times \frac{100}{205} = 2159.48 \text{ MN/m}.$

$\Rightarrow K_m = \left(\frac{1}{K_1} + \frac{1}{K_2}\right)^{-1} = 1451.45 \text{ MN/m}.$

stiffness constant:

$$C = \frac{K_b}{K_b + K_m} = \frac{534}{534 + 1451.45} = 0.268$$

Bolt material: ISO 9.8 grade

$S_p = 650\ \text{MPa}$, $S_{ut} = 900\ \text{MPa}$

$S_y = 720\ \text{MPa}$ (Table 8-11)

Preload: $F_i = 0.75\, S_p A_t$ (non permanent)

$= 41.1\ \text{kN}$

Total external load:

$$P_{total} = p_g \times \frac{\pi c^2}{4}$$

$$= \frac{6 \times 10^6 \times \pi \times (100)^2 \times 10^{-6}}{4} = 47.12\ \text{kN}$$

P = External load per bolt $= \frac{P_{total}}{N} = 4.71\ \text{kN}.$

Factor of safety

Yielding: $n_p = \frac{S_p A_t}{CP + F_i} = 1.14$

Joint separation: $n_0 = \frac{F_i}{P(1-C)} = 13.67$

Load factor: $n_L = \frac{S_p A_t - F_i}{CP} = 6.67$

→ Repeat the calculation when pressure varies from 0 to $p_g = 6\ \text{MPa}$.

$$\Rightarrow \frac{F_A''}{r_A} = \frac{F_B''}{r_B} = \frac{F_C''}{r_C} = \frac{F_D''}{r_D} \rightarrow ②$$

Calculate F_A'', F_B'', F_C'', F_D'' using ① & ②

If bolts have different dia.

$$\tau_A'' = \frac{M r_A}{J} \Rightarrow F_A'' = \tau_A'' A_A = \left(\frac{\pi M}{4J}\right) r_A d_A^2$$

$$\Rightarrow \frac{F_A}{r_A d_A^2} = \frac{F_B}{r_B d_B^2} = \frac{F_C}{r_C d_C^2} = \frac{F_D}{r_d d_d^2} \rightarrow ②'$$

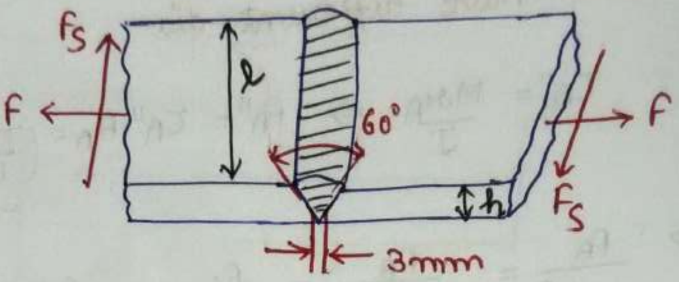
Primary shear: $F_A' = F_B' = F_C' = F_D' = \frac{V}{4}$ (if same dia)

F_A is resultant of F_A' & F_A''
F_B —— " —— F_B' & F_B''
F_C F_C' & F_C''
F_D —— " —— F_D' & F_D''.

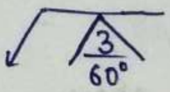
} Design bolt based on resultant shear.

Welded Joints (Butt and Fillet Joints)

Butt joint :-



Symbol:

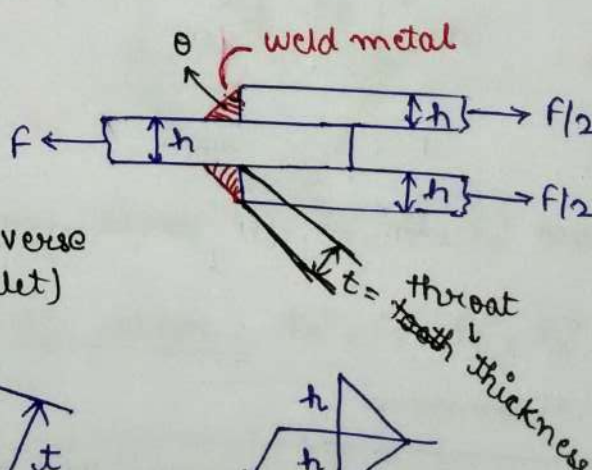


can be loaded in tension or shear

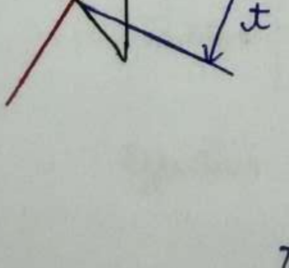
Avg. normal stress: $\sigma = \dfrac{F}{hl}$

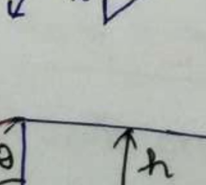
Avg. shear stress: $\tau = \dfrac{F_s}{hl}$

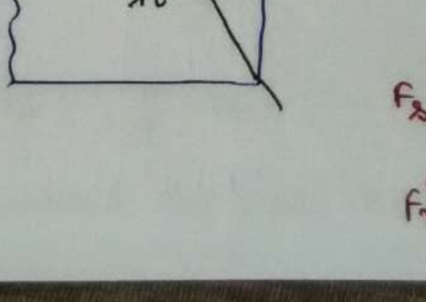
Fillet Joint :→

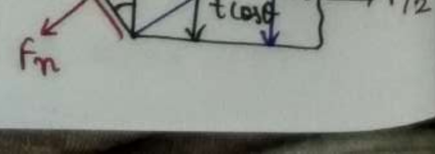


(Transverse fillet)









$$F_n = \frac{F}{2}\cos\theta$$

$$F_s = \frac{F}{2}\sin\theta$$

Throat thickness

$$h = t\cos\theta + t\sin\theta$$

$$\Rightarrow t = \frac{h}{(\cos\theta + \sin\theta)}$$

Throat area : $A_{throat} = t \cdot l = \dfrac{hl}{(\cos\theta + \sin\theta)}$

shear stress: $\tau = \dfrac{F_s}{A_{throat}} = \dfrac{F}{2hl}\sin\theta(\sin\theta + \cos\theta)$

Normal stress: $\sigma = \dfrac{F_n}{A_{throat}} = \dfrac{F}{2hl}\cos\theta(\sin\theta + \cos\theta)$

Von mises stress: $\sigma' = \sqrt{\sigma^2 + 3\tau^2}$

$$= \frac{F}{2hl}(\sin\theta + \cos\theta)\sqrt{\cos^2\theta + 3\sin^2\theta}$$

Maximum von mises stress:

$$\sigma' = 2.16\left(\frac{F}{2hl}\right) \quad \text{at} \quad \theta = 62.5°$$

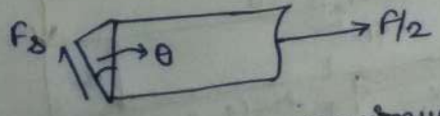
Maximum shear stress:

$$\tau_{max} = 1.207\left(\frac{F}{2hl}\right) \quad \text{at} \quad \theta = 67.5°$$

Conservative Design:

Ignores normal stress at throat.
All the load is taken by shear.

$$\boxed{F_s \sin\theta = \frac{F}{2}}$$



maximum F_s.

$F_s = \ldots 45$

Minimum throat length

$$t_{min} = \frac{h}{\sqrt{2}} \quad at \quad \theta = 45°$$

$$F_s\Big|_{at\, t_{min}} = \frac{F}{(2lt)} \cdot \frac{1}{\sin 45°} = \frac{1.414\, F}{2lt}$$

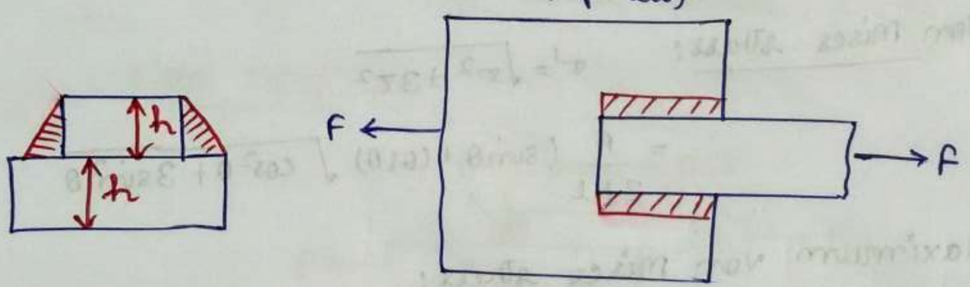
$$\tau\Big|_{\theta=45°} = \left(\frac{F}{2lh}\right) \times 1.414 \longrightarrow (*)$$

$\tau_{max} = \tau\Big|_{\theta=45°}$ is slightly larger than

$$\tau = 1.207\left(\frac{F}{2lh}\right) \quad at \quad \theta = 67.5°$$

Parallel Fillet :

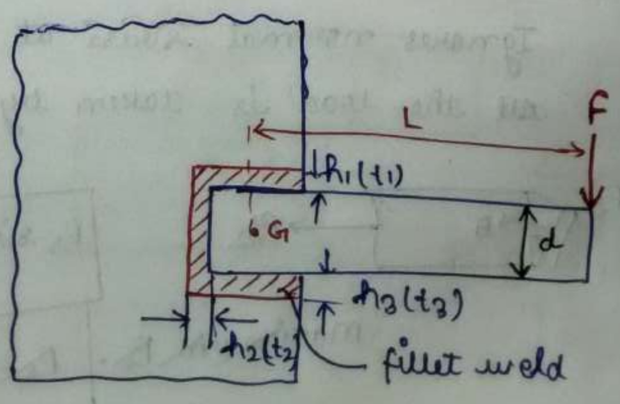
(top view)



- minimum throat area : $A_{throat} = \dfrac{2hl}{\sqrt{2}}$

- $\tau_{max} = \dfrac{F}{\frac{lh}{\sqrt{2}}} = 1.414\left(\dfrac{F}{2lh}\right)$

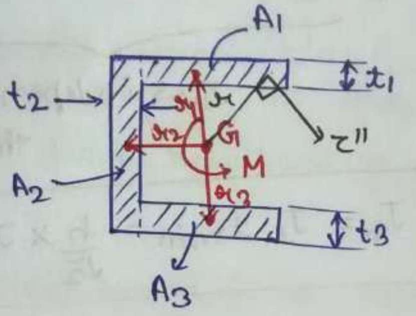
Combined Loading



t_i = throat thickness = $\frac{h_i}{\sqrt{2}}$

Primary shear: $\tau' = \frac{F}{A_{throat}}$

Secondary shear:



G: centroid of weld group

$M = FL$

<u>Throat areas:</u> $A_1 = bt_1$, $A_2 = dt_2$, $A_3 = bt_3$

r_1, r_2, r_3: distance from G to centroids of A_1, A_2, A_3 respectively

Weld group is under torsion due to M

$\Rightarrow \tau = \frac{Mr}{J}$ J → polar M.I. of weld group

$$J_{1G} = \frac{bt_1}{12}(b^2 + t_1^2) + A_1 r_1^2$$

$$J_{2G} = \frac{dt_2}{12}(d^2 + t_2^2) + A_2 r_2^2$$

$$J_{3G} = \frac{bt_3}{12}(b^2 + t_3^2) + A_3 r_3^2$$

$$J_G = J_{1G} + J_{2G} + J_{3G}$$

$$= \frac{1}{12}\left[(b^3 + br_1^2)t_1 + (d^3 + dr_2^2)t_2 + \left(\frac{b^3}{12} + br_3^2\right) \times t_3\right]$$

$$+ \frac{1}{12}(bt_1^3 + \cancel{dt_2^3} + bt_3^3) \rightarrow \text{Ignore}$$

$$\Rightarrow J_G = \left(\frac{b^3}{12} + b r_1^2\right) t_1 + \left(\frac{d^3}{12} + d r_2^2\right) t_2 + \left(\frac{b^3}{12} + b r_3^2\right) t_3$$

If $t_1 = t_2 = t_3 = t$

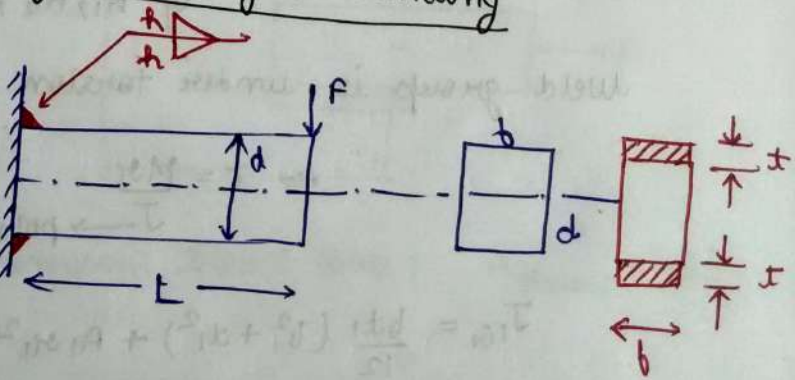
$$\Rightarrow J_G = \Big[\underbrace{\frac{2b^3}{12} + \frac{d^3}{12} + b(r_1^2 + r_3^2) + d r_2^2}_{J_u \rightarrow \text{ independent of weld thickness}} \Big] t$$

$$\Rightarrow \boxed{J_G = J_u \, t_{min} = \frac{h}{\sqrt{2}} \times J_u}$$

$$\Rightarrow \tau'' = \frac{M r}{J_G} \Rightarrow \boxed{\tau''_{max} = \frac{M r_{max}}{J_G}}$$

↓ Calculate resultant of τ' and τ'' for design

Secondary shear for bending



Bending moment = $M = FL$

Area moment of inertia:

$$I = I_u \cdot t = \frac{h}{\sqrt{2}} I_u$$

↓ unit M.I. of weld group about centroid

$$\tau'' = \frac{Mc}{I}$$

For J_u and I_u see table 9-1 & 9-2.

steps for combined loading

1. compute centroid of weld group
2. —"— Bending moment about G
3. —"— J_u (9-1) or I_u (9-2)
4. —"— Find τ''_{max}
5. Primary shear $\tau' = \frac{F}{A_{throat}}$

find resultant of τ' and τ''.

Note:

① when base metal is a cold drawn steel, the cold drawn properties are replaced by hot rolled properties near weld group.

② Always check for base metal failure

③ shear loading - shear stress on base metal is less than $0.4 S_y$ → (of base metal)

P-9-17 →

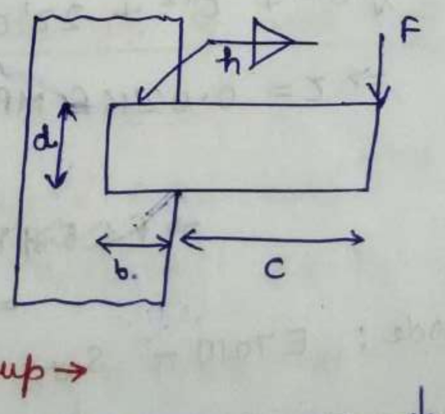
$b = 50$ mm

$d = 50$ mm

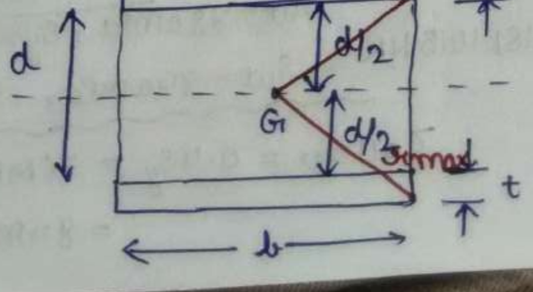
$c = 150$ mm

$h = 5$ mm

$\tau_{allow} = 140$ MPa.



Sol. → Weld group →



$$A_{throat} = 2b \times t$$
$$= (0.707h)\,2b$$
$$= 1.414\,hb$$

$$J_u = \frac{b}{6}(3d^2 + b^2) = 83.33 \times 10^3 \text{ mm}^3$$

$$J_G = (0.707h)\,J_u$$

$$L = c + \frac{b}{2} = 175 \text{ mm}$$
$$M = F.L$$

Primary shear: $\tau' = \frac{F}{A} = \frac{F}{353}$ MPa (F in N)

Secondary shear: $\tau'' = \frac{M.r_{max}}{J_G}$

$$r_{max} = 37.9 \text{ mm}$$

$$\Rightarrow \tau'' = \frac{F \times 175 \times 37.9}{(0.707h)\,J_u} \text{ MPa}$$

$$= 0.0225F \text{ MPa.}$$

Angle between τ' and τ'' is $\sim 45°$.

Resultant

$$\tau = \sqrt{\tau'^2 + \tau''^2 + 2\tau'\tau''\cos 45°}$$

$\Rightarrow \tau = 0.0246\,F$ MPa ≤ 140 MPa (allowable shear)

$\Rightarrow$ ~~F ≤ 5.67 kN~~

$\Rightarrow F \leq 5.69$ kN

(6) Electrode : E7010 – $S_{ut} = 482$ MPa $S_y = 393$ MPa } Table 9-3

$\tau_{allow} = 145$ MPa $(0.3 \times S_{ut})$

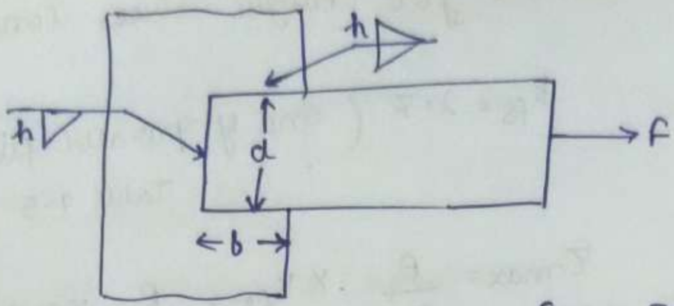
Bar: AISI 1020 HR – $S_{ut} = 380$ MPa, $S_y = 210$ MPa

Support: AISI 1015 HR – $S_{ut} = 340$ MPa, $S_y = 190$ MPa } Table A20

$\tau_{allow} = 0.4 S_y = 76$ MPa for support

$= 84$ MPa for bar

9-25)



Bar and support : AISI 1010 HR $\longrightarrow$ $S_{ut} = 320\,MPa$, $S_y = 180\,MPa$

Electrode: E 6010 $\rightarrow S_{ut} = 427\,MPa$

alternating load F

$b = d = 50\,mm$, $h = 5\,mm$.

Throat area, $A_{throat} = (2b+d)t = 0.707h(b+d)$

$= 530.25\,mm^2$.

Endurance strength of a member

$$S_e = K_a K_b K_c K_d K_e K_f S_e' \quad (S_e' \to 0.5 S_{ut})$$

$K_a = a S_{ut}^{b} = 57.7 \times (320)^{-0.718} = 0.917$ (Hot Rolled)

$K_b = 1.$

$K_c = 0.59$

$K_d = K_e = K_f = 1.$

$$S_e = 0.917 \times 0.59 \times 160 = 83.63\,MPa$$

Endurance strength of weld material

$$S_e = K_a K_b K_c K_d K_e K_f S_e'$$

$K_a = a S_{ut}^{b} = 0.7 \times (427)^{-0.718} = 0.745$

$K_b = 1$ $K_c = 0.59$ $K_d = K_e = K_f = 1.$

$$S_e = 0.745 \times 0.59 \times \left(\frac{427}{2}\right) = 93.84\,MPa.$$

$\Rightarrow$ Member is weaker.

Use factor of safety = 1.

Fatigue SCF ?

Table 9-5 for fatigue-stress concentration factor

$K_{fs} = 2.7$ (end of parallel fillet)

Table 9-5

$$\tau_{max} = \frac{F}{A_{throat}} \times K_{fs} = \frac{F}{530.3} \times 2.7$$

$$\tau_{max} \leq \tau_{allow} = 83.8 \text{ MPa}$$

$$\Rightarrow \frac{F}{530.3} \times 2.7 \leq 83.8 \text{ MPa}$$

$$\Rightarrow F \leq 16.4 \text{ kN}$$

Gears

→ Introduction from slides.

Design of spur gear

Angular velocity: n (rpm)

Pitch line velocity: $V = \frac{\omega d}{2} = \frac{2\pi n}{60} \times \frac{d}{2}$ (m/s)

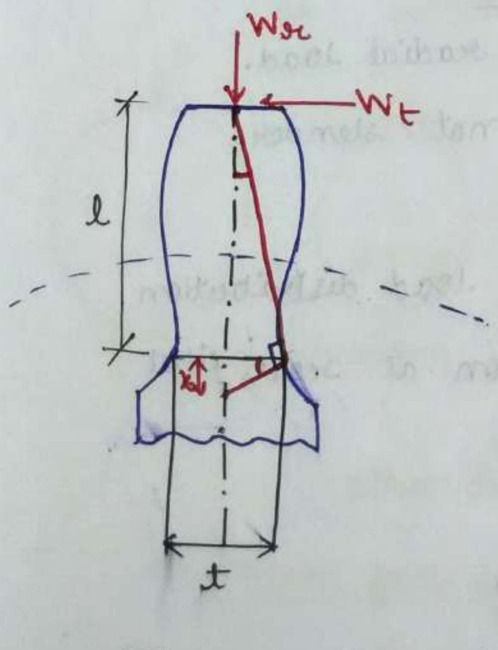
$= \frac{\pi d n}{60000}$ (if d is in mm)

Transmitted load: W_t = Power/Velocity

$= \frac{60000\,H}{\pi d n}$ kN

(if H is in kW)

Bending stress:



a: = point of max. bending stress (at root fillet)

t: = tooth thickness at a.

l: tooth depth

b: width of tooth

$$\sigma_{bending} = \frac{Mc}{I} = \frac{Mt}{2I}, \quad I = \frac{bt^3}{12}, \quad M = W_t l$$

$$\Rightarrow \sigma_{bending} = \frac{W_t l \times t}{\frac{2bt^3}{12}} = \frac{6W_t}{b}\left(\frac{l}{t^2}\right) = \frac{3W_t}{2bx}$$

Lewis bending eqn. :

$$\frac{x}{(t/2)} = \frac{(t/2)}{l} \Rightarrow \frac{t^2}{l} = 4x.$$

$$\Rightarrow \sigma_{bending} = \frac{3W_t}{2bx}$$

$$\sigma_{bending} = \frac{3W_t}{2bx} \cdot \frac{p}{p} = \frac{W_t}{bp} \cdot \left(\frac{3p}{2x}\right)$$

$$\boxed{\sigma_{bend.} = \frac{W_t}{bp} \cdot y}$$

$y := \frac{3p}{2x}$ = Lewis form factor

$p \rightarrow$ pitch (circular pitch)

$\rightarrow$ depends on tooth profile

Note : Not considered

① Compression due to radial load.

② Beam assumed is not slender.

③ Dynamic effects.

④ Non-uniformity of load distribution

⑤ stress - concentration at root fillet

⑥ Rim can be elastic

$$\sigma = \frac{W}{bp} \cdot y$$

with all factors

$$\Rightarrow \sigma = K_O \, K_V \, K_s \left(\frac{W}{b m_t}\right) \cdot \frac{1}{y_J} \cdot K_H \, K_B$$

K_O → overload factor

K_V → Dynamic factor

K_s → size factor

m_t → $\left(\frac{p}{\pi}\right)$

K_H → load distribution factor

K_B → rim thickness factor

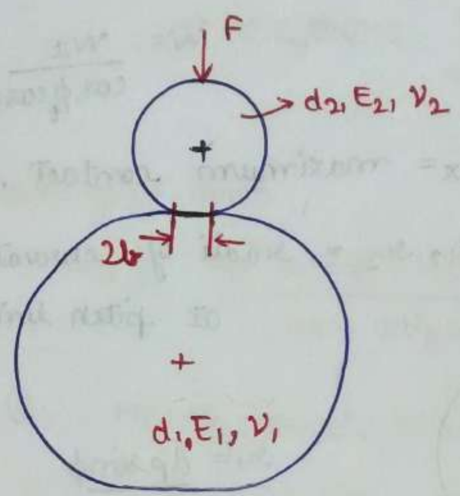
m_t – transverse module

$m_t = m$ for spur gear

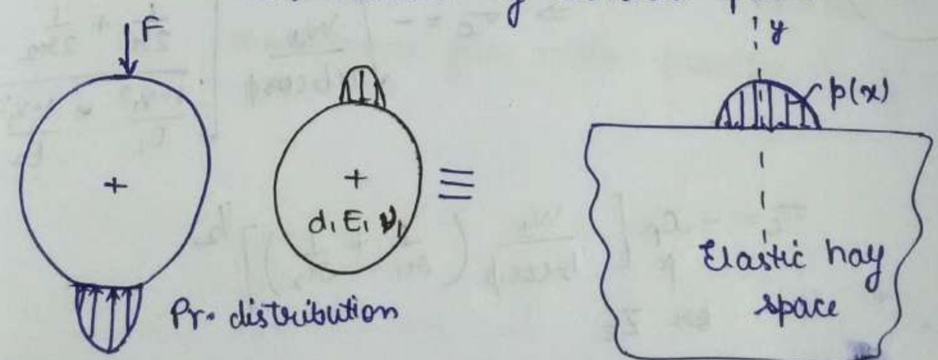
→ See the related slide also.

<u>Hertzian Contact betn two cylinders</u>

$\nu \rightarrow$ poisson ratio.



2b: width of contact patch



when $d_1, d_2 >> b$

Maximum <u>pressure</u>:

$$p_{max} = \frac{2F}{\pi b l} = \sqrt{\frac{2F}{\pi l}} \left\{ \frac{\frac{1}{d_1} + \frac{1}{d_2}}{\frac{(1-\nu_1^2)}{E_1} + \frac{(1-\nu_2^2)}{E_2}} \right\}^{1/2}$$

$$p(x) = p_{max}\left(1 - \frac{x^2}{b^2}\right)$$

$l \rightarrow$ length of cylinder

$$b = \sqrt{\left\{ \frac{2F}{\pi l} \left[\frac{(1-\nu_1^2)/E_1 + (1-\nu_2^2)/E_2}{\frac{1}{d_1} + \frac{1}{d_2}} \right] \right.}$$

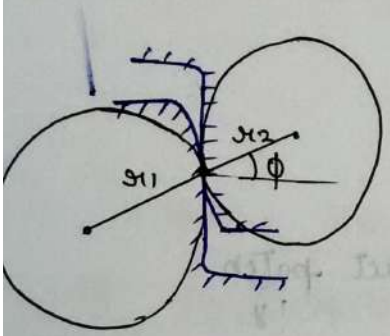
For gear tooth contact

Normal load on tooth : $W = \frac{W_t}{\cos\phi}$ (Spur gear)

$$W = \frac{W_t}{\cos\phi_t \cos\psi} \quad \text{(helical gear)}$$

p_{max} = maximum contact stress = σ_c

r_1, r_2 = radii of curvatures of tooth profile at pitch line.



$$r_1 = \frac{d_P \sin\phi}{2} \qquad r_2 = \frac{d_G \sin\phi}{2}$$

$$\Rightarrow \sigma_c = -\sqrt{\frac{W_t}{\pi b \cos\phi}} \left[\frac{\frac{1}{2r_1} + \frac{1}{2r_2}}{\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}} \right]^{1/2}$$

$$\sigma_c = -C_P \left[\frac{W_t}{b\cos\phi} \left(\frac{1}{r_1} + \frac{1}{r_2} \right) \right]^{1/2}$$

or Z_E

C_P = Elastic constant = $\left[\frac{1}{2\pi \left(\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2} \right)} \right]^{1/2}$

Problem 13-10)

$$N_P \geq \frac{2K}{(1+2m_G)\sin^2\phi} \left[m_G + \sqrt{m_G^2 + (1+2m_G)\sin^2\phi} \right]$$

a) Take $m_G = 1$, $\phi = 20°$, $K = 1$ [full depth tooth]

$N_P = 13, 14, \ldots$

(b) $m_G = 2.5$

$N_P \geq 14.63 \quad \Rightarrow N_P = 15.$

(c) smallest pinion that will run with rack

$m_G = \infty$

as $m_G = \frac{d_G}{d_P} = \frac{N_G}{N_P}.$

$\Rightarrow \cancel{N_P \geq 18.244} \quad N_P \geq 17.097$

$\cancel{\Rightarrow N_P =} \quad \Rightarrow N_P = 18.$

Maximum gear tooth

$$N_G \leq \frac{N_P^2 \sin^2\phi - 4k^2}{4k - 2N_P \sin^2\phi}$$

for (b), $N_P = 15, \ \phi = 20°, \ k = 1.$

$N_G \leq 45.489 \quad \Rightarrow \quad N_{G|max} = 45$

$N_{pinion} = 15.$

maximum gear ratio possible is $\frac{45}{15} = 3.$

Tutorial-9 Problem- Gear design

(a) Pinion and Gear teeth number

Interference condition

$$N_P \geq \frac{2K}{(1+2m_G)\sin^2\phi}\left[m_G + \sqrt{m_G^2 + (1+2m_G)\sin^2\phi}\right]$$

$K = 1$ (full depth)

$m_G = \frac{N_G}{N_P} = 4.$

$\phi = 20°$

$\Rightarrow N_P \geq 15.44$

Choose $\boxed{N_P = 16 \quad \Rightarrow N_G = 64.}$

for $N_P = 16$, $\phi = 20°$
$N_G \leq 101.07$

module:

$d_P = m N_P$

$d_G = m N_G$

center-to-centre distance:

$$c = \frac{d_P + d_G}{2} = 200 \text{ mm}$$

$$\Rightarrow \frac{m(N_P + N_G)}{2} = 200$$

$$\Rightarrow \boxed{m = 5 \text{ mm}}$$

$$\Rightarrow \boxed{\begin{aligned} d_P &= 80 \text{ mm}. \\ d_G &= 320 \text{ mm}. \end{aligned}}$$

(6) Power rating $H = 25$ KW

Pinion speed $n = 2000$ rpm

Transmitted load:

(Transverse) $W_t = \frac{60000 H}{\pi d_p N}$ KN

$$\Rightarrow W_t = \frac{60,000 \times 25}{\pi \times 80 \times 2000} = 2.984 \text{ kN}$$

$$\boxed{W_t = 2.984 \text{ KN}}$$

Pitch line velocity:

$$v = \frac{\pi \overbrace{d_p}^{\text{(in mm)}} n}{60,000} \text{ m/s.} = 8.378 \text{ m/s.}$$

Factors for AGMA bending stress eqn.

overload factor: $\boxed{K_0 = 2.0}$ (fig 14-17)

→ Power source light shock
Driven heavy shock

~~surface~~ size factor: $\boxed{K_S = 1.}$

Dynamic factor: (Eq 14-27, 14-28)

$$K_v = \left(\frac{A + \sqrt{200 v}}{A} \right)^B$$

$$A = 50 + 56(1 - B)$$

$$B = 0.25(12 - Q_v)^{2/3} = 0.25(12 - 10)^{2/3}$$

as $Q_v = 10$.

$$\Rightarrow \left.\begin{array}{l} B = 0.3969 \\ A = 83.776 \end{array}\right\} \Rightarrow \boxed{K_v = 1.171}$$

Load distribution factor

$K_H = 1.3$ (from table)
↓
slides

Bending strength:

$$S_t = 0.7255\, H_B + 153.63 \text{ (MPa)} \quad \text{(fig 14-4)}$$

Material: Nitrided, 2.5 % Chrome steel

$H_B = 250$ (P)

$H_B = 200$ (G)

$(S_t)_P = 335$ MPa (Pinion)

$(S_t)_G = 298.73$ MPa (gear)

Allowable stress: $\sigma_{all} = \dfrac{S_t \cdot Y_N}{S_f \cdot Y_\theta\, Y_z}$

Pinion $\sigma_{all}|_P = \dfrac{335 \times 0.977}{S_f} = \dfrac{327.3}{S_f}$

Gear $\sigma_{all}|_G = \dfrac{298.73 \times 1.009}{S_f} = \dfrac{299}{S_f}$

Factor of safety

Pinion $S_{f_P} = \dfrac{\sigma_{all}}{\sigma_P} = \dfrac{327.3}{139.6} = 2.43$

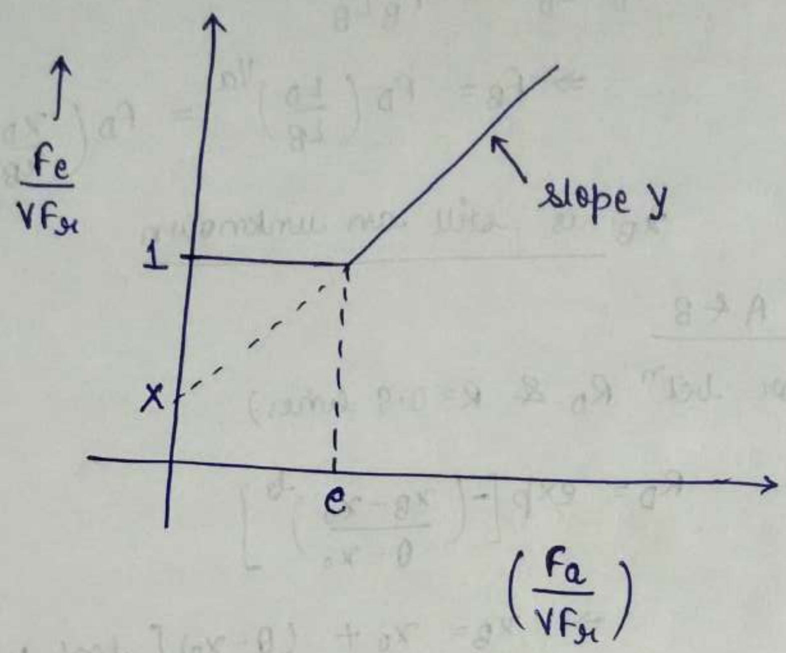
Gear $S_{f_G} = \dfrac{\sigma_{all}}{\sigma_G} = \dfrac{299}{88.6} = 3.373$

Combined Radial + Thrust Load

F_a = Thrust load (along shaft axis)

F_r = Radial load

Find equivalent radial load. → F_e



$$\frac{F_e}{VF_r} = 1 \text{ for } \frac{F_a}{VF_r} \leq e$$

$$= X + Y\left(\frac{F_a}{VF_r}\right) \text{ for } \frac{F_a}{VF_r} \geq e$$

$$V = \begin{cases} 1 & \text{if inner race rotates} \\ 1.2 & \text{if outer race rotates} \end{cases}$$

"e" depends on static load rating C_0

C_0 := basic static load rating

- A load that produces a total permanent deformation of 10^{-4} times that of rolling element size.

→ For e, X, Y see Table 11-1.

→ For C_{10}, C_0 see table 11-2, 11-3.

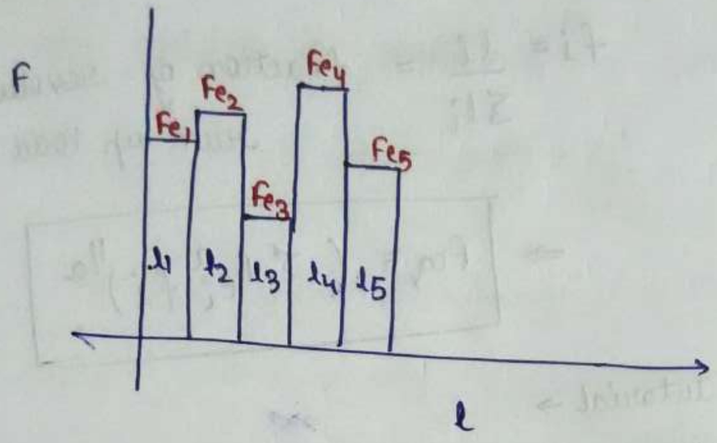
→ Load application factor a_f (table 11-5)

Variable Loading

$Fe_1 \leftarrow (F_{a_1}, F_{r_1})$ load for $- l_1$ cycles

$Fe_2 \leftarrow (F_{a_2}, F_{r_2})$ load for $- l_2$ cycles

$\vdots$



$FL^{1/a} = \text{constant}$

$\quad \longrightarrow$ life to failure

$\Rightarrow \; Fe_1 L_1^{1/a} = Fe_2 L_2^{1/a} = \dots$ (since bearing is same)

L_i is life at load Fe_i

$l_1 < L_1, \dots, l_i < L_i$

$\Rightarrow \; Fe_i^a L_i = \text{constant}$

Linear damage hypothesis:

$\Rightarrow$ Damage : $D_1 = Fe_1^a l_1$

$D_2 = Fe_2^a l_2, \dots$

Total Damage

$$D = D_1 + D_2 + \dots = \sum_{i=1}^{N} Feq_i^a \, l_i$$

Let bearing run for $(l_1 + l_2 + \dots + l_N)$ cycles. at load Feq and cause same damage

$\Rightarrow \; Leq = (l_1 + l_2 + \dots + l_N)$

$$\Rightarrow \; Feq^a \, Leq = \sum Feq^a \, li = D.$$

$$F_{eq}^{a} = \frac{\Sigma F_{e_i}^{a} l_i}{\Sigma l_i} = \Sigma F_{e_i}^{a}\left(\frac{l_i}{\Sigma l_i}\right)$$

$$= \Sigma F_{e_i}^{a} f_i =$$

$f_i = \frac{l_i}{\Sigma l_i}$ = fraction of revolution run up load

$$\Rightarrow \boxed{F_{eq} = \left(\Sigma F_{e_i}^{a} f_i\right)^{1/a}}$$

Q→ from Tutorial →

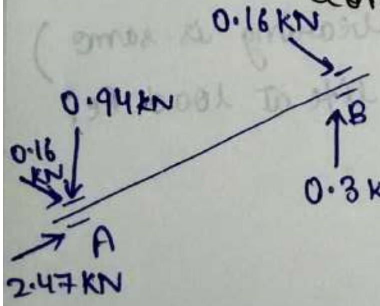
Given: At A: Angular contact bearing, Takes all the axial thrust } 02-Series

At B: Straight roller bearing

combined reliability : 0.99,

Application factor : $a_f = 1.3$

Weibull parameters: $x_0 = 0.02$, $\theta - x_0 = 4.139$, $b = 1.483$



Reliability : $R = R_A R_B \geq 0.99$
↓
combined

Either $R_A = R_B = \sqrt{0.99}$

or choose : $R_A = 0.99$
$R_B = 1$

In fig. loads are in lbf convert to KN.

Selection of Bearing at A:

$$F_r = \sqrt{(0.16^2 + 0.94^2)} = 0.954 \text{ KN}$$

$$F_a = 2.47 \text{ KN}$$

Find equivalent radial load: F_e

Inner race loading : $V = 1$

$$\frac{F_a}{VF_r} = \frac{2.47}{0.954} = 2.59$$

Need to know: e from table 11-1.

→ choose a starting value of $\frac{F_a}{C_0}$ at somewhere in the middle of table 11-1.

Let $\frac{F_a}{C_0} = 0.07 \Rightarrow e = 0.27 < \frac{F_a}{VF_r}$

$X = X_2 = 0.56, \quad Y = Y_2 = 1.63$

$$\Rightarrow F_e = VF_r\left(X + Y.\frac{F_a}{V.F_r}\right) = 4.57\ \text{kN}$$

$$C_0 = \frac{F_a}{0.07} = 35.3\ \text{kN}$$

$$C_{10} = a_f . F_D . \left\{\frac{x_D}{x_0 + (\theta - x_0)\{\ln(\frac{1}{R_D})\}^{1/b}}\right\}^{1/a}$$

$a = 3$ for ball bearing

$C_{10} = 96.93\ \text{kN}$

$L_D = 25 \times 10^3 \times 600 \times 60 = 900 \times 10^6$ cycles

$$x_D = \frac{L_D}{L_{10}} = \frac{900 \times 10^6}{10^6} = 900.$$

Now select a bearing from table 11-2.

Select: 02-series bearing with bore dia 90-mm

$\Rightarrow C_{10} = 106\ \text{kN} > 96.93\ \text{kN}$

$C_0 = 73.5\ \text{kN} > 35.3\ \text{kN}$

Check: $\frac{F_a}{C_0} = \frac{2.47}{73.5} = 0.034$

$\Rightarrow e = 0.23,$ (table 11-1) & linear interpolation)

$X_2 = 0.56, \quad Y_2 = 1.93$

$$F_e = VF_r\left(0.56 + 1.93 \times \frac{F_a}{VF_r}\right)$$

$$= 5.3\ \text{kN} = F_D.$$

$$\Rightarrow C_{10} = 110.2\ \text{kN} > 106\ \text{kN}$$

Not satisfied

Now, select bearing with bore dia 95 mm

$$\frac{F_a}{C_0} = \frac{2.47}{85} = 0.029$$

$$\Rightarrow e = 0.221,\ X_2 = 0.56$$
$$Y_2 = 1.98$$

$$\Rightarrow F_e = 5.425\ \text{kN}$$

$$\Rightarrow C_{10} = 112.8\ \text{kN} < 121\ \text{kN}$$

Satisfied

Bearing at A: 02-series, 95 mm bore dia Angular contact

Bearing at B: $R_D = 1$, straight roller

$$F_r = F_D = \sqrt{(0.16^2 + 0.3^2)} = 0.34\ \text{kN}$$

$$C_{10} = a_f F_D\left(\frac{x_D}{0.02}\right)^{1/a}$$ ($a = 10/3$ for roller bearing)

$$C_{10} = 11\ \text{kN}$$

See bearing from table 11-3.

Choose bore dia = 25 mm (02-series)

$$\Rightarrow \boxed{C_{10} = 16.8\ \text{kN},\ C_0 = 8.8\ \text{kN}}$$

or choose same bore dia as bearing at A

ie 95 mm $\Rightarrow$ $C_{10} = 165\ \text{kN},\ C_0 = 112\ \text{kN}$