Sudden variation in c/s are found in

- ✓ Stepped shafts
- ✓ Holes in plates
- ✓ Threads.
- ✓ Key slots.
- ✓ Grooves.
- ✓ Cracks
- ✓ Blow Holes.

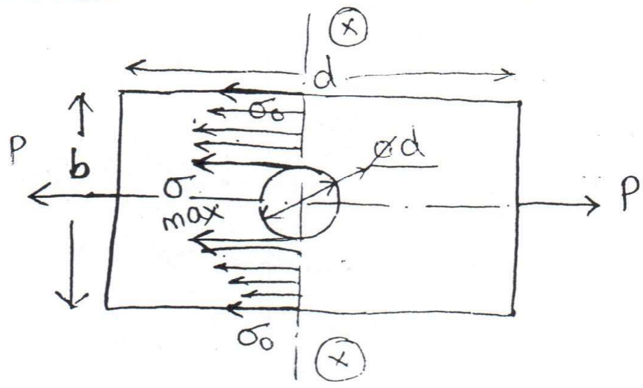
$$\sigma_0 = \frac{P}{\frac{\pi}{4}d^2} = \text{Nominal stress}$$

This is called Nominal stress

Based on Net (or) minimum area of cross-section

According to experimental observation, $\boxed{\sigma_{max} > \sigma_0}$

Stress Concentration: It is found experimentally that the stress nearer to a sudden variation in cross-section is much higher than the stress evaluated using strength of material relation based on net area. This phenomenon of rise in stress is called stress concentration, and the sudden variations are called stress risers.

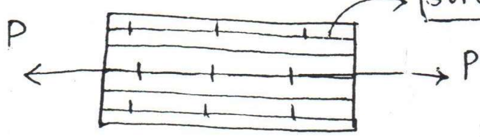


$$\sigma_0 = \frac{P}{(b-d)t}$$

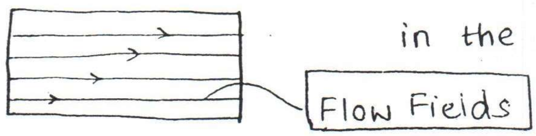
$$\boxed{\sigma_{max} > \sigma_0}$$

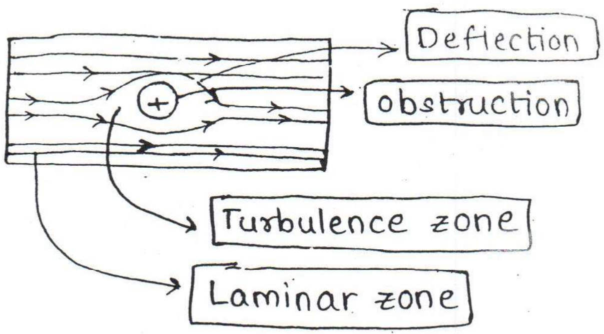
Stress Fields :

→ Joining of points with same stress, in a component.



stress fields → can be similar to the flow through similar cross-sections i.e. There is no disturbance of flow in the stress fields.

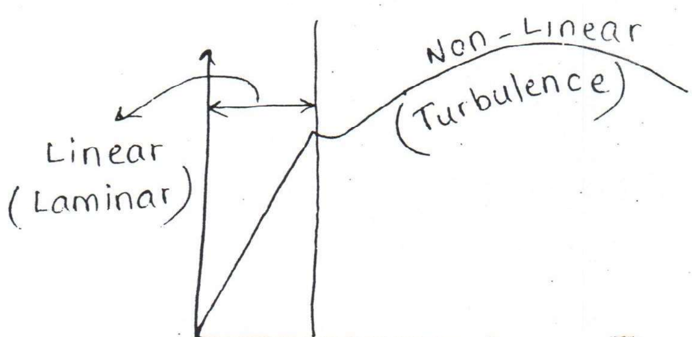




From strength of materials point of view,

Laminar → Below yield point zone

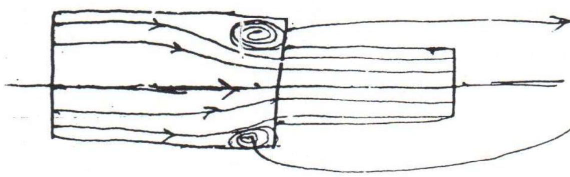
Turbulence → above yield point zone.



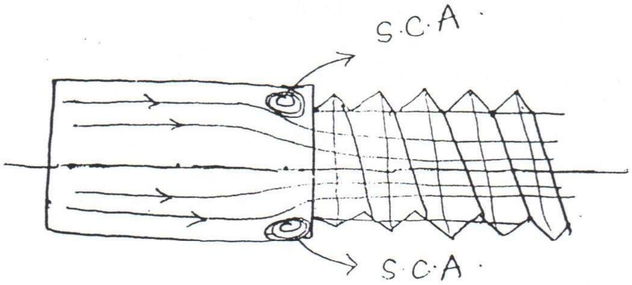
$$\boxed{S \cdot E = \frac{\sigma^2}{2E} \times \text{volume}}$$

If energy is more, it is in Turbulence zone, i.e. The stress increases.

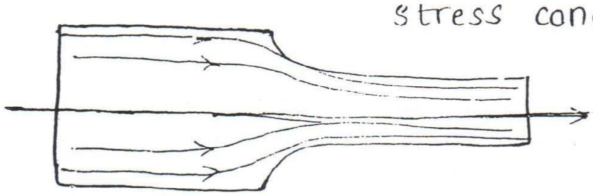
Therefore, stress evaluated using S.O.M relations is less because of the variation in stresses due to Non-Linear flow stress fields.



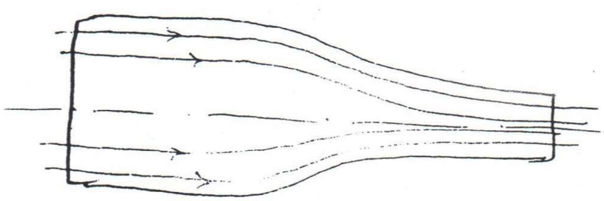
stress- concentration areas (Turbulence zone)



stress fields in a threaded zone.



stress concentrations can be reduced by introducing fillets



✓ As suddenness increases, rise in stress increases.

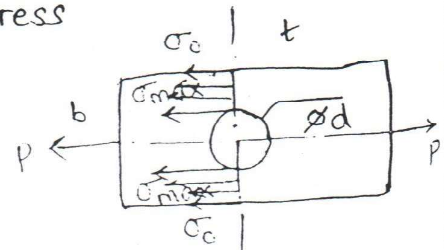
✓ Increase graduality to decrease stress concentration

Stress- concentration Factor (K_t):

$$\sigma_o = \frac{P}{(b-d)\times t}$$

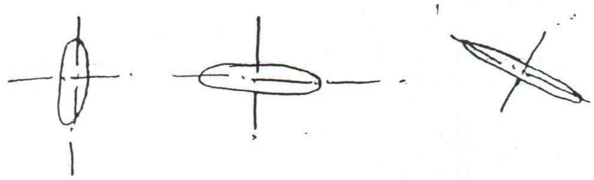
$$K_t = \frac{\sigma_{max}}{\sigma_o} = \frac{\text{maximum stress}}{\text{nominal stress}}$$

$$\boxed{\sigma_{max} = K_t \times \sigma_o}$$

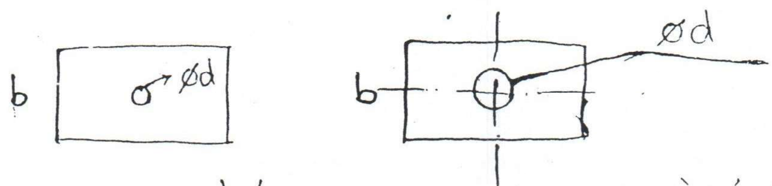


✓ Depends on shape of discontinuity.

✓ depends on orientation of discontinuity.



✓ depends on suddeness of discontinuity



As diametre 'd' increases (or) width 'b' decreases,

suddeness increases ⇒ σ_{max} increases.

✓ depends on types of Load.

(i) Based on direction

- Normal
- shear
- Bending Moment
- Twisting Moment

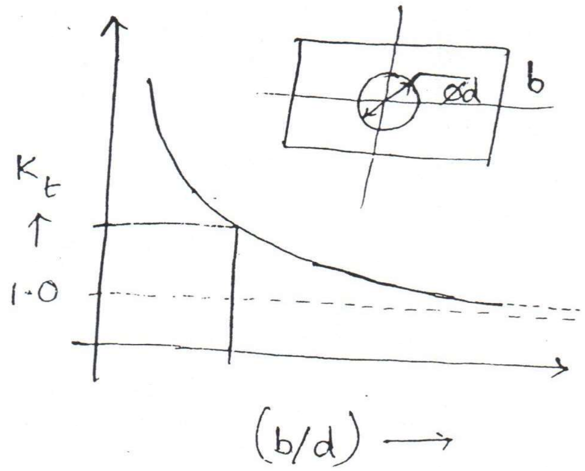
(ii) Based on variation with respect to time

- constant
- varying with respect to time.

✓ doesn't depend on the material.

It is called as Theoritical Stress Concentration Factor.

NOTE: For common discontinuities with loads, Theoritical stress concentration factors are evaluated and available as ① Tables
② Graphs.



As (b/d) increases, suddenness decreases

$\Rightarrow$ K_t decreases

$$\boxed{\begin{gathered} K_t \geq 1 \\ 1 \leq K_t \leq \infty \end{gathered}}$$

If (b/d) ratio is known, corresponding "K_t" value can be obtained from the graph. This "K_t" value is multiplied with nominal stress to give the maximum stress.

i.e. $\boxed{\sigma_{max} = K_t \times \sigma_o}$

✓ If "K_t" value is not mentioned, it is taken as "1".

NOTE: ① Maximum stress occurs nearer to the discontinuity and reduces to nominal stress (σ_o) as moving away from discontinuity.

② ***
$$\boxed{\begin{aligned} \sigma_{max} &= K_t \times \sigma_o && \text{— for constant loads} \\ &= K_f \times \sigma_o && \text{— for varying loads.} \end{aligned}}$$

K_f = fatigue stress concentration factor

$$\boxed{K_f = 1 + q\,(K_t - 1)}$$

q = Notch sensitivity. i.e. $q = \dfrac{(K_f - 1)}{(K_t - 1)} = \dfrac{\sigma_o K_f - \sigma_o}{\sigma_o K_t - \sigma_o}$

"q" → material property.

"q" varies from 0 to 1 i.e. $0 \le q \le 1$ for any material

$\therefore$ If $q = 0 \Rightarrow \boxed{K_f = 1}$

If $q = 1 \Rightarrow \boxed{K_f = K_t}$

$\boxed{1 \le K_f \le K_t}$

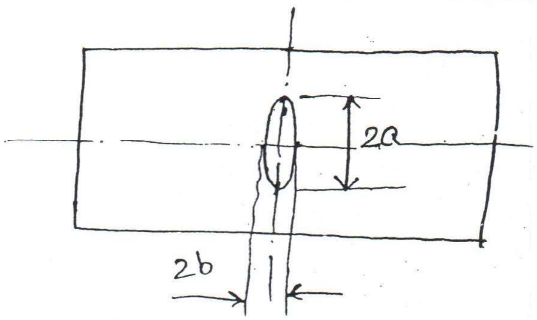
$K_f = 1 \rightarrow$ it is insensitive to cracks.

$K_f = K_t \rightarrow$ Fully sensitive.

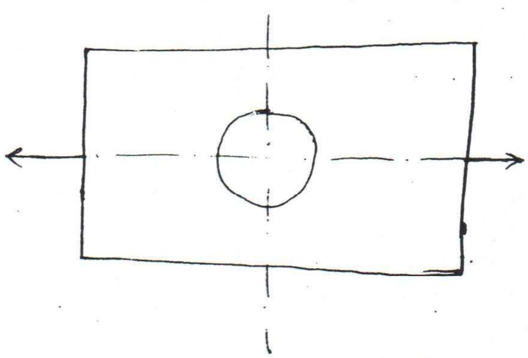
→ For most of the materials, "q" varies from 0.8 to 0.95

→ In the absence of any data, $\boxed{K_t = 1 \ \& \ K_f = K_t}$ ***

③ Plate with elliptical hole:



$$\boxed{K_t = 1 + 2\left(\frac{a}{b}\right)}$$

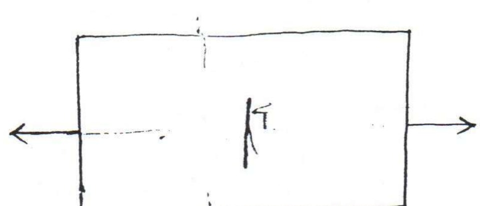


For a plate with circular hole,

$a = b$

$\Rightarrow \boxed{K_t = 3}$ * → This value is maximum. *

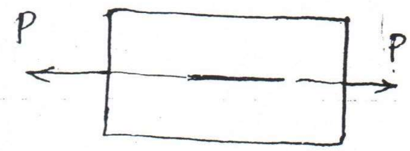
If the graduality is increased, K_t may reduce to "1".



For a transverse crack, $b = 0$

$\Rightarrow \boxed{K_t = \infty}$ * → maximum / highest "K_t" *

Transverse crack is most critical discontinuity ⇒ immediately fails.



In case of parallel cracks, $a = 0$

$\Rightarrow \boxed{K_t = 1.}$

→ Design for varying cross-sections :

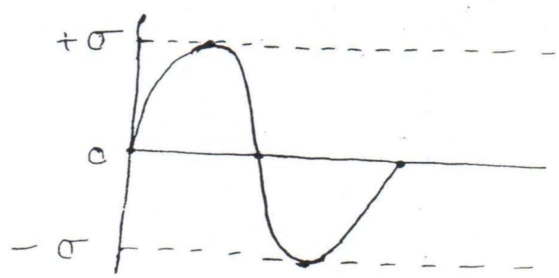
① Gradual variations, $\sigma_{max} = \sigma_0 = \dfrac{\sigma_y \text{ (or) } \sigma_{us}}{F.S.}$

② sudden variations, $\sigma_{max} = K_t \times \sigma_0 = \dfrac{\sigma_y \text{ (or) } \sigma_{us}}{F.S.}$

$= K_f \times \sigma_0 = ?$

FATIGUE : Fatigue is a phenomenon by virtue of which, a material fails at a stress much lower than its failure stress due to the cyclic stresses, (or) cyclic reversal of stresses.

Cyclic stress :



Completely Reversed stress.

Why a material fails at a lower value than failure stress?

Because of :

① Dislocations in the material.

② Movement of dislocations.

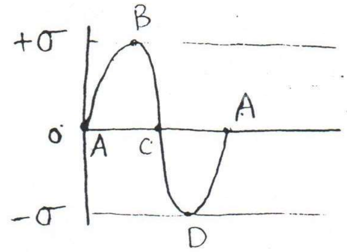
③ Collide and merge to form a nucleus of failure.

④ Nucleus of failure propagates and forms cracks.

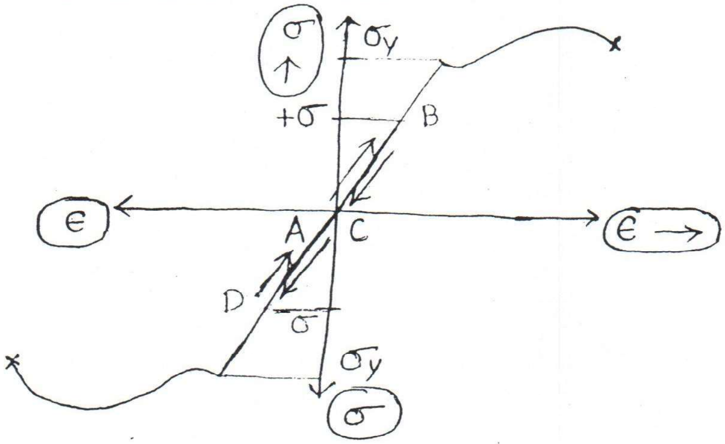
⑤ Propagation of cracks to fracture.

Bauschinger effect :

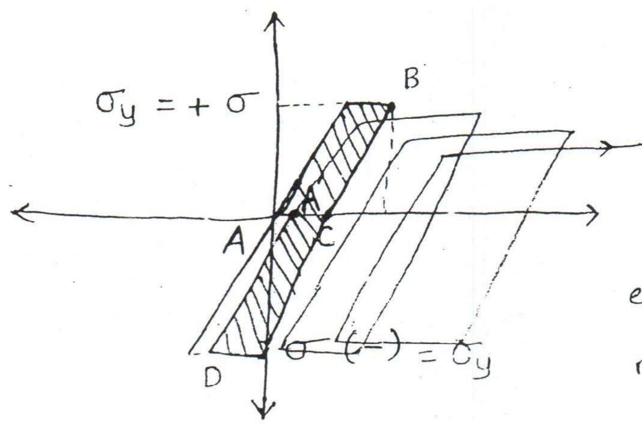
It says that the yield stress of a material reduces with every cyclic stress applied.



→ For a ductile material,



After few cycles, $\sigma_y \cong \sigma$



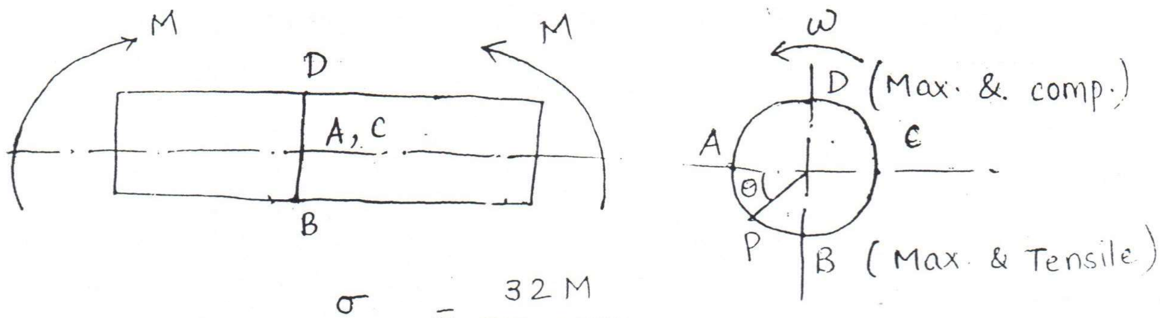
AC → Permanent deformation → AA' in the specimen

Shaded region → the energy stored in the material due to cyclic stresses applied.

→ If applied stress $= \sigma_y$, the material fails in lower number of cycles.

Life : It is the number of stress cycles completed before the failure of the material.

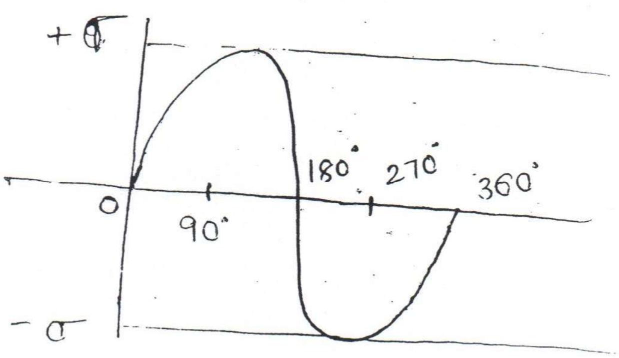
tresses in rotating shafts subjected to pure bending:



$$\sigma_{max.} = \frac{32M}{\pi d^3}$$

- When rotating shafts are subjected to pure bending, stresses induced are completely reversed Bending stresses.

$$\boxed{\sigma = \frac{M}{I} \times y = \frac{M}{I} \times r\sin\theta}$$

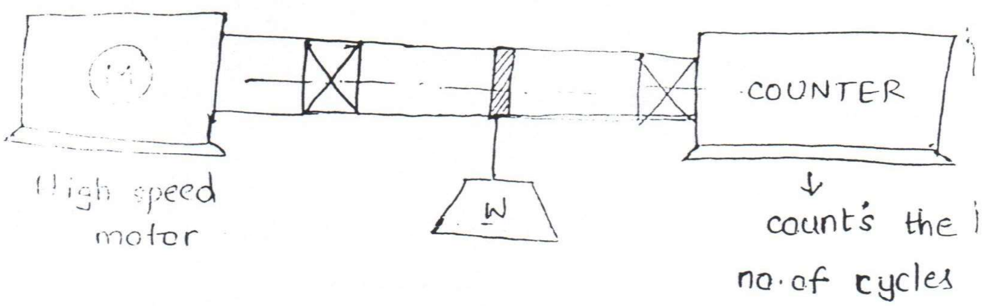


$$\sigma = \frac{32M}{\pi d^3}$$

completely Reversed Bending stresses.

* **Fatigue Test:**

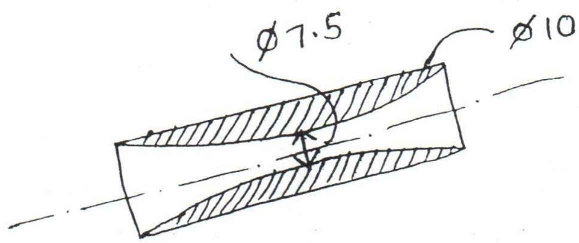
- also called as rotating Beam Test / R.R. Moore Test.



✓ A group of standard specimens are used.

(i) They are mirror finished

(ii) standard size and shape.



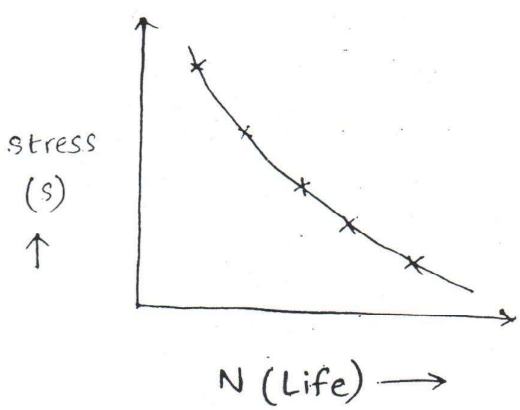
The specimen can be considered as a simply supporte[d] Beam on two Bearings

$$\Rightarrow \text{Max. B.M.} = M = \frac{WL}{4}$$

$$\sigma = \frac{32M}{\pi d^3} = \frac{8WL}{\pi d^3}$$

S.No.	W	σ	"N" rev.
1.	W_1	σ_1	N_1
2.	W_2	σ_2	N_2
⋮	⋮	⋮	⋮

N_1 → No. of cycles before failure



curve is obtained by organising the data of

As stress S increases, Life N of the material decreases.

<u>Endurance Limit (S_e)</u>: It is the maximum completely reversed stress that the material can withstand infinite cycles without fatigue failure.

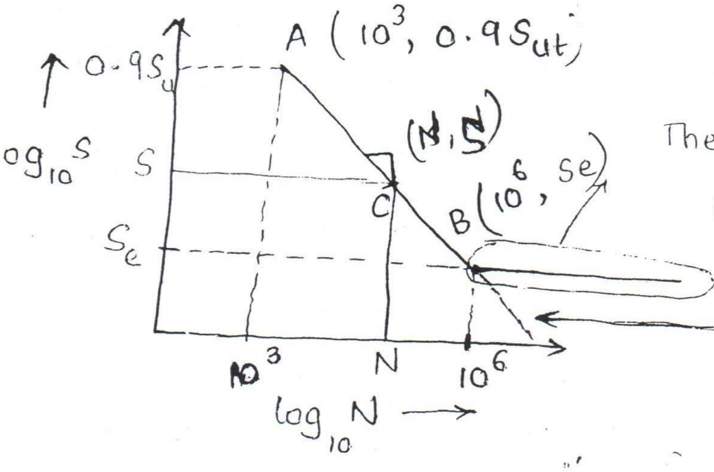
→ Practical life $= 10^6$ cycles

$\doteq$ 1 million cycles

= Endurance Life / Fatigue life.

-N Diagram / curve: It is a double log curve.

i.e. it is $\boxed{\log_{10} S \ \text{vs.} \ \log_{10} N}$ curve.



Theoretically, it is assumed to be a line parallel to x-axis

Practical Behaviour of the material.

⇒ component has infinite life which is not possible practically

$$\dot{m} = \text{slope} = \frac{y_A - y_B}{x_A - x_B} = \frac{y_c - y_B}{x_c - x_B}$$

$$\Rightarrow \frac{0.9 S_{ut} - S_e}{10^3 - 10^6} = \frac{S - S_e}{N - 10^6}$$

$$\therefore \boxed{\text{Allowable stress} = \frac{S \text{ (or) } S_e}{F.S.}}$$

Modified / corrected Endurance Limit:

To suit the given conditions, test endurance limit is modified i.e.

Load, size, surf. finish, Temp, reliability

$$\boxed{\begin{aligned} S_{em} &= S_e * K_L \ K_s \ K_{sf} \ K_t \ K_r \\ &= S_e * K_A \ K_B \ K_C \ K_D \end{aligned}}$$

$\boxed{K \leq 1}$

<u>Note:</u> ① $S_e = 0.5 * S_{ut}$ → for steels at 50 % reliability

② "S" at 1000 cycles $= 0.9 * S_{ut}$

③ Load correction factor $K_L = 1.00$ for Bending

$= 0.5$ for shear

$= 0.6$ to 0.8 for axial loads.

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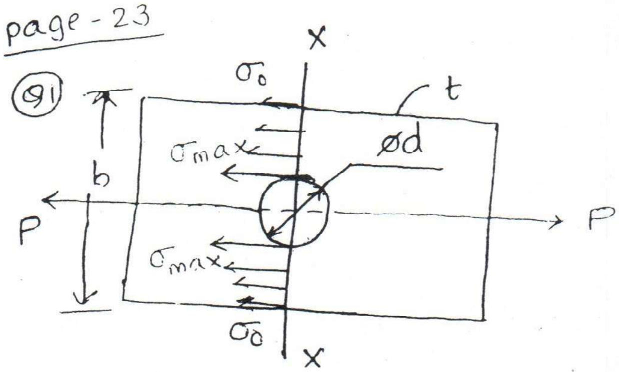
Q1



allowable stress

$$\sigma_{max} = K_t \times \sigma_0$$

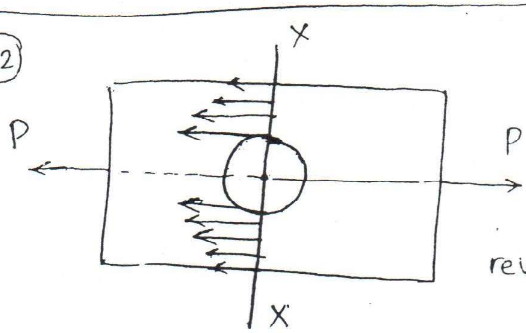
$$= K_t (=1) \times \frac{P}{(b-d)t} = \frac{\sigma_y (\text{or}) \sigma_u}{F.S.}$$

$$\Rightarrow 62.5 \times 10^6 = \frac{P}{(50-10)\times 10}$$

$$\Rightarrow P = 62.5 \times 10^6 \times 400$$

i.e. $\boxed{P = 25\ kN}$

Q2



$P = \pm 16\ kN$ completely reversed Load.

$q = 0.8$

$K_t = 2.35$

$$\sigma_{max} = K_f \times \frac{P}{(b-d)t} = \frac{S_{em}}{F.S.}$$

$$K_f = 1 + q(K_t - 1)$$

$$= 1 + 0.8(2.35 - 1) \Rightarrow \boxed{K_f = 2.08}$$

$$S_e = 0.5\, S_{ut} \text{ for steel.}$$

$$S_{em} = (0.5 S_{ut}) * K_a\, K_b\, K_c\, K_d \qquad S_e = 0.5\, S_{ut}$$

$$= 440 * (0.67 \times 0.85 \times 0.9 \times 0.897) \times 0.5$$

$$S_{em} = 101.146 \text{ MPa}$$

$$\therefore K_f \frac{P}{(b-d)\,t} = \frac{S_{em}}{F.S.} \qquad F.S. = 1.5$$

$$2.08 \times \frac{16 \times 10^3}{(50-10) \times t} = \frac{101.146}{1.5}$$

$$\Rightarrow t = \frac{2.08 \times 16 \times 10^3 \times 1.5}{40 \times 101.146}$$

$$\Rightarrow t = \frac{49{,}920}{4045.84} = \boxed{12.33 \text{ mm}}$$

min. thickness to be designed

$$\therefore \boxed{t = 12 \text{ mm}} \; \times \qquad \boxed{t = 20 \text{ mm}}$$ as per options

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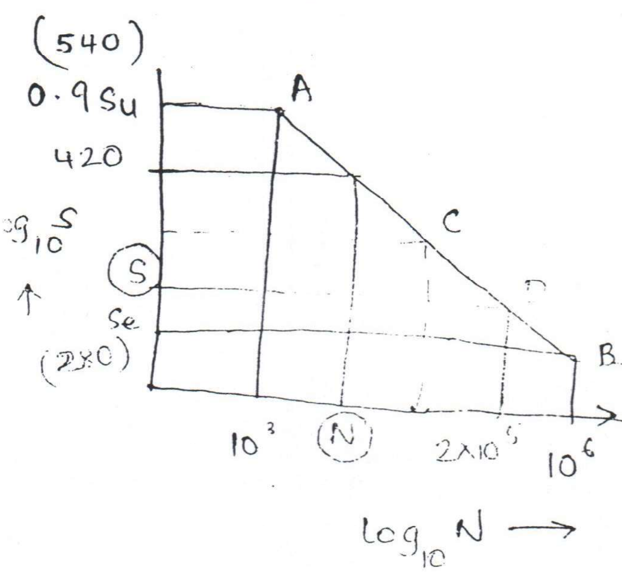
Q13 – Q14 $\quad S_e = 280$ MPa $\quad S_u = 600$ MPa ;



$S_e = 280$ MPa @ 10^6 cycles

$S = 0.9 S_u = 0.9 \times 600$

$= 540$ MPa $\times 10^3$ cycles

Slope of S-N curve.

$$m = \frac{\log_{10} 540 - \log_{10}(280)}{3 - 6} = \frac{0.285}{-3}$$

$$\Rightarrow m = 0.09$$

(31)

$$\dot{m} = \frac{\log_{10} S - \log_{10} 280}{\log_{10}(2\times10^5) - 6} = -0.09$$

$$-0.09 = \frac{\log_{10} S - (2.447)}{5.30 - 6}$$

$$-0.09 = \frac{\log_{10} S - (2.447)}{-0.698}$$

$$0.0628 + 2.447 = \log_{10} S \Rightarrow \boxed{S = 323.44 \text{ MPa}}$$

$$\boxed{S \approx 326 \text{ MPa}}$$

(ii) N = ?

$$-0.09 = \frac{\log_{10}(420) - \log_{10}(280)}{\log_{10} N - 6}$$

$$-0.09 = \frac{(2.623 - 2.447)}{\log_{10} N - 6}$$

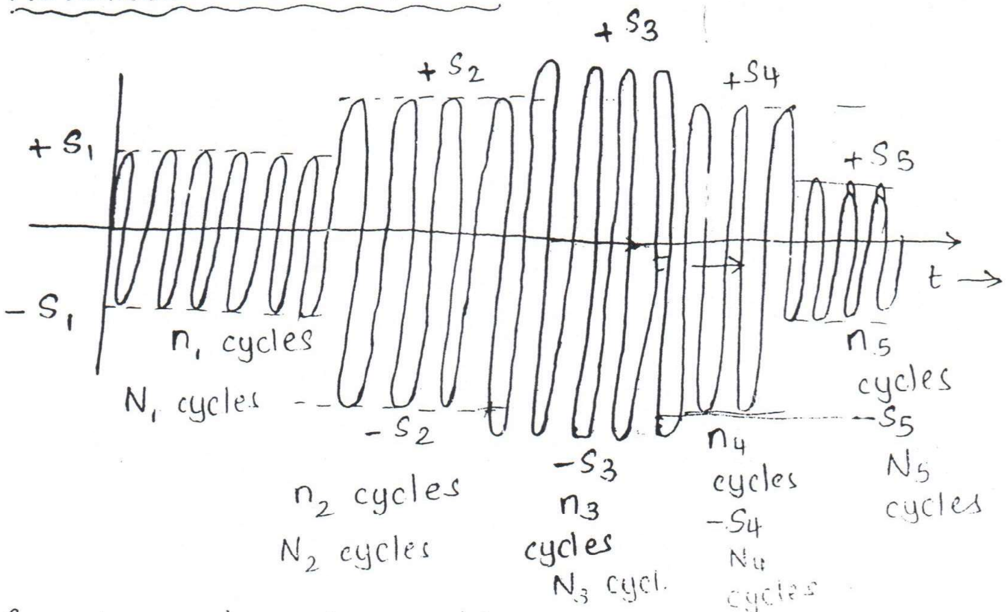
$$-0.09 = \frac{0.176}{\log_{10} N - 6}$$

$$\log_{10} N - 6 = \frac{0.176}{-0.09}$$

$$\log_{10} N - 6 = -1.958$$

$$\log_{10} N = 4.042 \Rightarrow \boxed{N = 1.4 \times 10^4 \text{ cycles}}$$

Automobile Loads / stresses :



Life at varying stress cycles :

✓ can be studied by cumulative damage concept

✓ Miner's rule.

Let, Life at stresses $\pm S_1 = N_1$ cycles (found by SN diagram)

cumulative damage = Fraction of life consumed for 1 cyclic stress applied (say S_1)

$\therefore$ Fraction of Life consumed for

1 revolution at $\pm S_1$ stress $= \dfrac{1}{N_1}$

Due to n_1 cycles @ $\pm S_1$ stress $= \dfrac{n_1}{N_1}$

N_1, N_2, N_3, ⋮ } found from S-N curve

Similarly, $\dfrac{n_2}{N_2}$, $\dfrac{n_3}{N_3}$, … Life consumed at n_2 cycles @ $\pm S_2$, etc

$\therefore$ Failure occurs if,

$$\boxed{\frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} + \cdots = 1} \Rightarrow \text{Miner's Rule}$$

$\therefore$ Life of the component $= (n_1 + n_2 + n_3 + \dots) = N$ revolutions.

(99)

S.No	P	n_i	N_i
1.	HL	9.8 hrs	10 hrs
2.	NL	8.2 hrs	?
		18 hrs	

According to Miner's rule,

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

$$\Rightarrow \frac{9.8}{10} + \frac{8.2}{N_2} = 1$$

$$\Rightarrow \frac{8.2}{N_2} = 1 - \frac{9.8}{10}$$

$$\Rightarrow \frac{8.2}{N_2} = \frac{0.2}{10} \Rightarrow N_2 = \frac{8.2 \times 10}{0.2} \Rightarrow \boxed{N_2 = 410 \text{ hrs}}$$

(912)

N_i	S.No	S_i		%t	n_i
1×10^5 cycl	1.	500 MPa	10 cycles	$\frac{10}{18} \times 100$	$\frac{2 \times 10}{18} \times 18 \times 60H$ $= 1200H$
0.4×10^5 cyc.	2.	600 MPa	5 cycles	$\frac{5}{18} \times 100$	$5 \times 120H$ $= 600H$
0.15×10^5 cyc.	3.	700 MPa	3 cycles	$\frac{3}{18} \times 100$	$3 \times 120H$ $= 360H$
			18 cycles		N cycles

Let, H = Life in hours.

$$N = \frac{18 \text{ cycles}}{\frac{1}{2} \text{ min}} = 36 \text{ cycles/min} \times 60$$

$$= 36 \times 60 \times H \text{ cycles}$$

According to Miner's rule,

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} = 1.$$

$$\Rightarrow \frac{1200\,H}{1\times10^5} + \frac{600\,H}{0.4\times10^5} + \frac{360\,H}{0.15\times10^5} = 1$$

$$\Rightarrow \frac{H}{10^5}\left[\frac{1200}{1} + \frac{600}{0.4} + \frac{360}{0.15}\right] = 1.$$

$$\Rightarrow \boxed{H = 19.6 \text{ hours}}$$

17)

S.No.	S_i	% t	n_i	N_i
1.	± 350	85 %	0.85 N	N_1
2.	± 400	12 %	0.12 N	N_2
3	± 500	3 %	0.03 N	N_3
			N cycles	

✓ $S_e = 280$ MPa $\rightarrow 10^6$ cycles

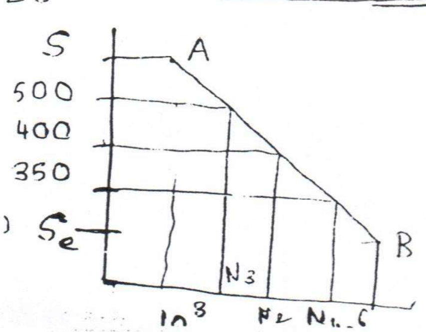
$S_u = 660$ MPa

✓ $S_f = 0.8 \times S_u$

$= 0.8 \times 660 \longrightarrow 10^3$ cycles

$\boxed{S = 528 \text{ MPa}}$

28



$$m = \frac{\log_{10}(528) - \log_{10}(280)}{3-6} = \frac{\log_{10} 350 - \log_{10} 28}{\log_{10} N_1 - 3}$$

$$-0.09 = \frac{0.09}{\log_{10} N_1 - 3} \Rightarrow \cancel{\boxed{N_1 = 106 \text{ cycles}}}$$

$\boxed{N_1 = 8.8\times10^4}$

$$m = \frac{\log_{10} 400 - \log_{10} 280}{\log_{10} N_2 - 6} \Rightarrow N_2 = 2.05 \times 10^4$$

$$= \frac{\log_{10} 500 - \log_{10} 280}{\log_{10} N_3 - 6} \Rightarrow N_3 = 0.18 \times 10^4$$

According to Miner's rule,

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + \frac{n_3}{N_3} = 1$$

$$\Rightarrow \frac{0.85 N}{N_1} + \frac{0.12 N}{N_2} + \frac{0.03 N}{N_3} = 1.$$

$$\Rightarrow \frac{0.85 N}{8.8 \times 10^4} + \frac{0.12 N}{2.05 \times 10^4} + \frac{0.03 N}{0.18 \times 10^4} = 1$$

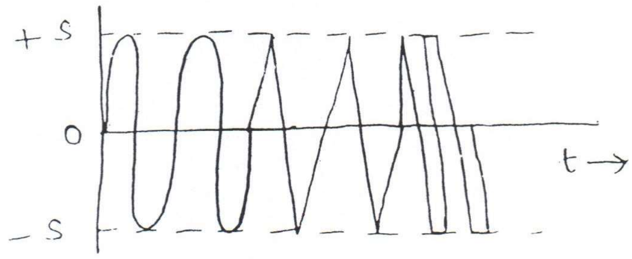
$$\Rightarrow \boxed{N = 31092 \text{ cycles}}$$

Q5 – Q6

S.No	Si	% t	ni		Ni
1.	± 225	50%	0.50 N =	12500 cy	N_1 ✓
2.	± 145	30%	0.30 N	7500 cy	N_2 ✓
3.	± S	20%	0.20 N	5000 cy	N_3

N = 25000 cycles

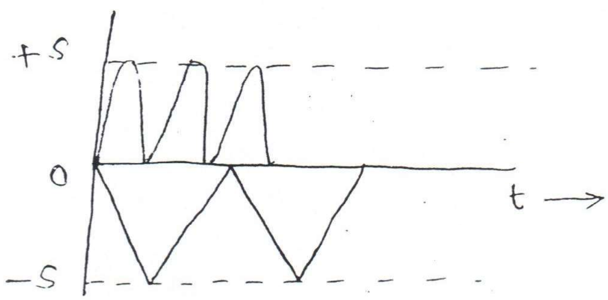
completely reversed load / stress :



Everytime, variation need not be a sinusoical curve. It can be anything which varies from same positive val.e to negative value.

$$\boxed{\text{allowable stress} = \frac{S_{em}}{F.S.}}$$

Repeated Load / stress : varies from 0 to S.



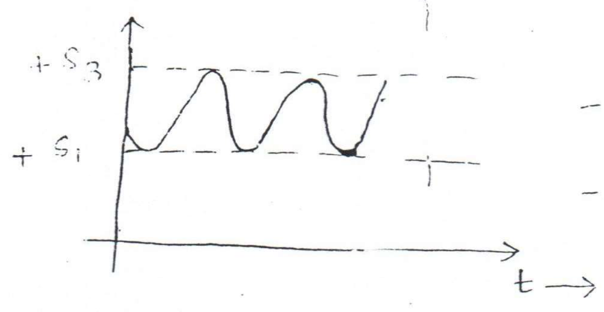
Ex: Punching operation when punch hits the sheet, it becomes +v and then goes to zero and process continues.

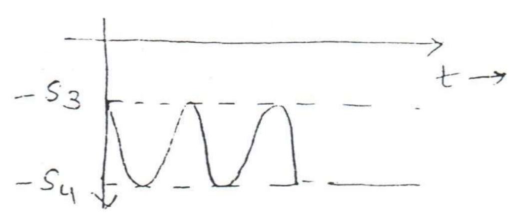
✓ It may be either on positive side, (or) on the negative side.

Fluctuating stress :

→ varies from $+S_1$ to $+S_2$ (or) $-S_3$ to $-S_4$.

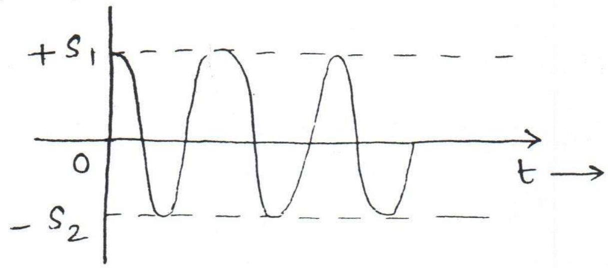
→ varies only on one side.





Alternating stress :

→ varies from $+S_1$ to $-S_2$



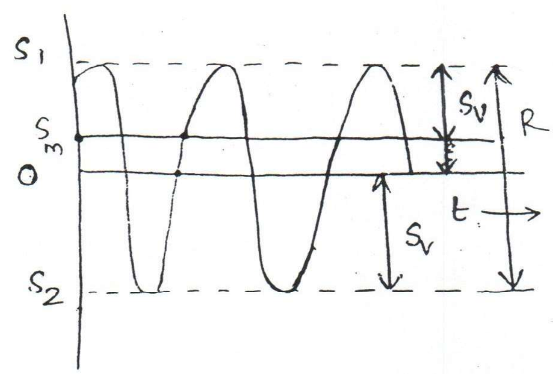
varies between the stresses of unequal magnitudes but of opposite signs.

✓ maximum stresses can be on any one side (or both, but positive maximum and negative maximum are not equal in magnitude, in any case.

* Design for variable stresses / Loads :

Let, S_1 = maximum stress / Load.

S_2 = minimum stress / Load.



✓ $S_1 = S_2$ same
↳ const. load.

✓ $S_1 = S_2$, opposite
↳ comp. reversed load

✓ $S_1 \neq S_2$ → opposite side ⇒ alternating stress

✓ S_1 & $S_2 = 0$ $(S_1 \neq 0)$ ⇒ Repeated load / stress

✓ $S_1 \neq S_2$ → either side ⇒ fluctuating stress.

⇔ S_m = mean (or) average stress $\boxed{S_m = \frac{S_1 + S_2}{2}}$

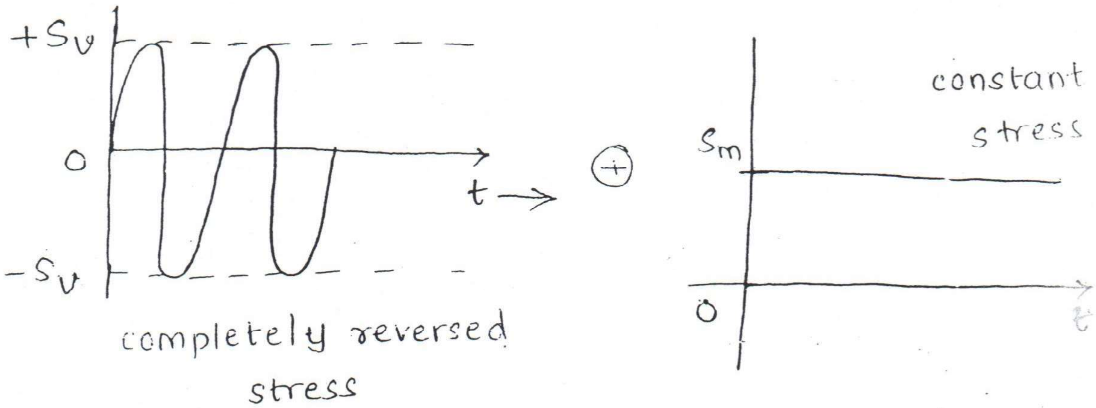
S_v = variable stress
= stress amplitude

$$\boxed{S_v = \frac{S_1 - S_2}{2}}$$

$(S_1 - S_2) = R = \text{Range}$

$$\boxed{\begin{aligned} S_1 &= S_m + S_v \\ S_2 &= S_m - S_v \end{aligned}}$$

$$\boxed{\text{Stress amplitude} = \frac{S_2}{S_1} = \frac{S_{min}}{S_{max}}}$$



∴ A Generalised stress ≅ A completely reversed stress + a constant stress

✓ When two stresses (i.e. a completely reversed stress and constant stress), design is done according to the fatigue theories of failure.

* Fatigue Theories of Failure:

→ These are used to predict the failure condition due to static stress and completely reversed stress.

1. Goodman's criterion
2. Gerber's criterion
3. Soderberg's criterion

① Goodman's straight line Equation:

$$\boxed{\frac{S_m}{S_u} + \frac{S_v}{S_e} = 1}$$

const stress ← S_m

ultimate stress ← S_u

S_v → variable stress

S_e → endurance limit

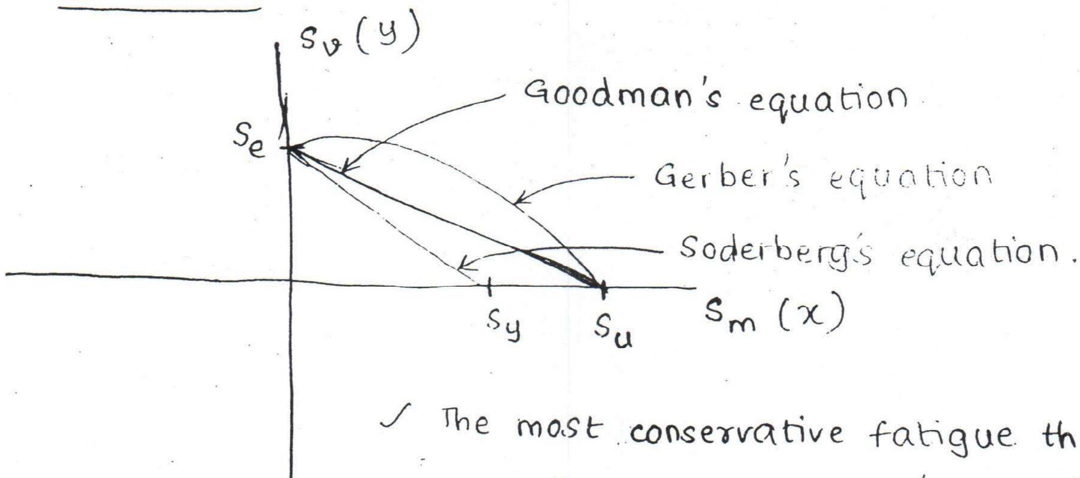
② Gerber's Parabolic equation:

$$\boxed{\left(\frac{S_m}{S_u}\right)^2 + \frac{S_v}{S_e} = 1}$$

③ Soderberg's straight Line Equation:

$$\boxed{\frac{S_m}{S_y} + \frac{S_v}{S_e} = 1.}$$

Safe zones:



✓ The most conservative fatigue theories of failure → soderberg's criteria.

✓ Gives bigger dimensions.

✓ Soderberg's criteria → default criteria.

✓ For Brittle materials, "S_y" is not available, use Goodman's criteria.

✓ Gerber's criteria over estimates the strength. Hence, it is generally not used.

Design for variable stresses:

① Soderberg's equation for safe design is

$$\boxed{\frac{S_m}{S_y} + \frac{S_v}{S_{em}} = \frac{1}{F.S.}}$$

② Soderberg's equation with stress concentration factor is

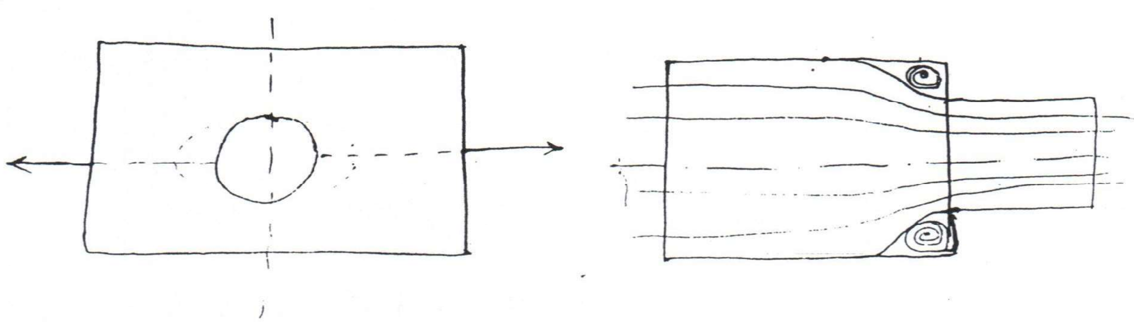
$$\boxed{\frac{K_t \cdot S_m}{S_y} + \frac{K_f \cdot S_v}{S_{em}} = \frac{1}{F.S.}} \rightarrow \text{For Brittle materials.}$$

NOTE: In industrial design,

	K_t	K_f
Brittle	K_t	K_f
Ductile	1	K_f

✓ Brittle material has no yield point

✓ For Ductile material, yield point exists.



Due to yielding, deformation occurs in a ductile material, thereby reducing 'K_t'.

→ Stress concentration factors for ductile materials is applied only on variable stresses.

→ In ductile materials, for constant stresses, the discontinuity changes its suddenness, thereby decreasing K_t to almost equal to 1.

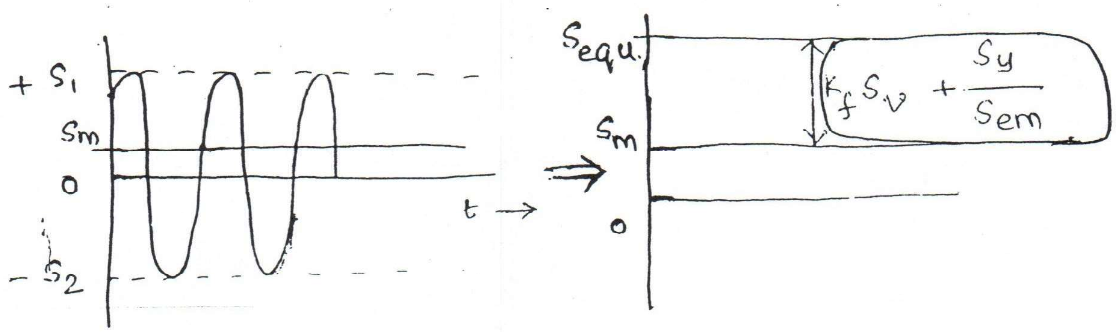
Modified soderberg's equation is

$$\boxed{\frac{S_m}{S_y} + \frac{K_f S_v}{S_{em}} = \frac{1}{F.S}} \rightarrow \text{For ductile materials}$$

when "S_y" is multiplied on both sides,

$$\underbrace{\left[S_m + K_f \cdot S_v \cdot \frac{S_y}{S_{em}}\right]}_{\text{Equivalent static stress to variable stress.}} = \underbrace{\left(\frac{S_y}{F.S}\right)}_{\text{allowable static stress}}$$

$$\boxed{S_{Equ} = S_m + K_f \cdot S_v \cdot \frac{S_y}{S_{em}}}$$



(98) $T_1 = 2\ kN\text{-}m$ $\quad d = 50mm = 50 \times 10^{-3} m$

$T_2 = -0.8\ kN\text{-}m$ $\quad T_{mean} = \frac{T_1 + T_2}{2} = \frac{1.2}{2} = 0.6\ kN\text{-}m$

$$T_v = \frac{(T_1 - T_2)}{2} = 1.4\ kN\text{-}m.$$

according to soderberg's equation,

$$\frac{S_{sm}}{S_{sy}} + \frac{S_{sv}}{S_{se}} = \frac{1}{F.S}$$

$$S_{sm} = \frac{16 T_m}{\pi d^3} = \frac{16 \times 0.6 \times 10^3 \times 10^6}{\pi \times (50)^3} = 24.44\ MPa$$

$$S_{sv} = \frac{16 T_v}{\pi d^3} = \frac{16 \times 1.4 \times 10^3}{\pi \times (50)^3} = 0.05704 MPa.$$

$$\frac{24.44}{225} + \frac{57.04}{150} = \frac{1}{F.S}$$

$$0.4888 = \frac{1}{F.S.} \Rightarrow \boxed{F.S. = 2.04} \approx 2$$

---

(414) $S_1 = 400\ MPa$;

$S_2 = 200\ MPa$; stress amplitude = ?

$$\text{i.e. } S_v = \frac{S_1 - S_2}{2} = \frac{400-200}{2} = 100\ MPa$$

9) $\sigma = \dfrac{8WL}{\pi d^3} = 100$ MPa ;

$$\left.\begin{aligned} S_y &= 300 \text{ MPa} ; \\ S_u &= 500 \text{ MPa} ; \\ S_{em} &= 200 \text{ MPa} ; \end{aligned}\right\} F.S = ?$$

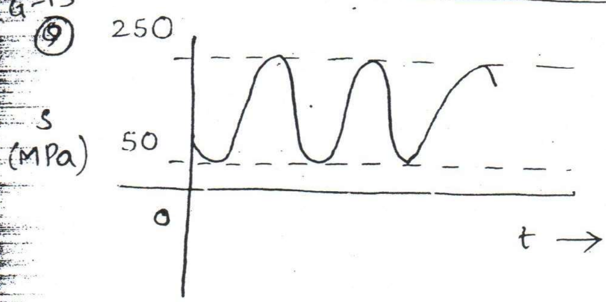
Rotating shaft = completely reversed Bending stress

$$\therefore \sigma_b = \frac{S_{eb}}{F.S}$$

$$\Rightarrow F.S = \frac{S_{eb}}{\sigma_b} = \frac{200}{100} = \underline{\underline{2}}$$

G-15

9



$$S.A. = \frac{S_1 - S_2}{2} = \frac{250 - 50}{2} = 100 \text{ MPa} ;$$

$$S.R. = \frac{S_2}{S_1} = \frac{5\cancel{0}}{25\cancel{0}} = \frac{1}{5} = \underline{\underline{0.2}} ;$$

9

G-87

$S_e = 196$ MPa

$S_y = 294$ mPa

$$S_m = \frac{147 + 49}{2} = 98 \text{ mPa}$$

$K_f = 1.32$

$S_v = 49$ MPa ;

$$\frac{98}{294} + \frac{1.32 \times 49}{196} = \frac{1}{F.S} \rightarrow \text{soderberg's Equation.}$$

$$0.33 + 0.33 = \frac{1}{F.S.} \Rightarrow \underline{\underline{F.S. = 1.51}}$$

$$\therefore \boxed{F.S. \equiv 2}$$

⑨ -07 <u>sphere</u>

$D = 200\,mm$

$t = 1\,mm$

$P_i = 4$ to 8 MPa

$S_y = 600$ MPa

$S_u = 800$ MPa

$S_e = 400$ MPa

Goodman's Equation

F.S. = ?

$$P_{mean} = \frac{4+8}{2} = 6\ MPa \ ; \quad P_v = \frac{8-4}{2} = 2\ MPa$$

$$S_m = S_H = \frac{PD}{4t} = \frac{6 \times \cancel{200}^{50} \times \cancel{10^{-3}}}{\cancel{4} \times 1 \times \cancel{10^{-3}}} = 300\ MPa \ ;$$

$$S_v = \frac{2 \times 200}{4 \times 1} = 100\ MPa \ ;$$

$$\frac{3\cancel{00}}{8\cancel{00}} + \frac{1\cancel{00}}{4\cancel{00}} = \frac{1}{F.S}$$

$$\boxed{F.S. = 1.6}$$

---

⑧ -09

$F_1 = +160$ kN

$F_2 = -40$ kN

$$F_{mean} = \frac{160 - 40}{2} = 60\ \text{kN} \ ;$$

$$F_v = \frac{160 - (-40)}{2} = 100\ \text{kN} \ ;$$

$S_u = 600$ MPa ;

$S_y = 420$ MPa ;

$S_e = 240$ MPa ;

$d = 30\,mm$;

F.S.)

$$\frac{F_m}{A} = S_m = \frac{60 \times 10^3 \times 4}{\pi \times (30)^2} = 84.88\ MPa \ ;$$

$$\frac{F_v}{A} = S_v = \frac{100 \times 10^3 \times 4}{\pi \times (30)^2} = 141.47\ MPa \ ;$$

$$\frac{84.88}{420} + \frac{141.47}{240} = \frac{1}{F.S} \rightarrow \text{soderberg's equation}$$ 38

$$\boxed{F.S. = 1.26}$$

9
G-13

$F_1 = 100$ kN $F_m = \frac{100+20}{2} = 60$ kN $F.S. = 2$

$F_2 = 20$ kN $F_v = \frac{100-20}{2} = 40$ kN

$S_y = 240$ MPa ;

$S_e = 160$ MPa ; $S_m = \frac{F_m}{A_m} \Rightarrow A_m = \frac{F_m}{S_m}$

$$\frac{S_m}{S_y} + \frac{S_v}{S_e} = \frac{1}{2}$$

$$\frac{60 \times 10^3}{A_m(240)} + \frac{40 \times 10^3}{A_m(160)} = \frac{1}{2}$$

$$\frac{1}{A}\left[250 + 250\right] = \frac{1}{2}$$

$$A = 500 \times 2 = 1000 \text{ mm}^2$$

9
G-16

$S_1 = 150$ MPa $S_m = \frac{200}{2} = 100$ MPa

$S_2 = 50$ MPa $S_v = 50$ MPa

$S_e = 200$ MPa ;

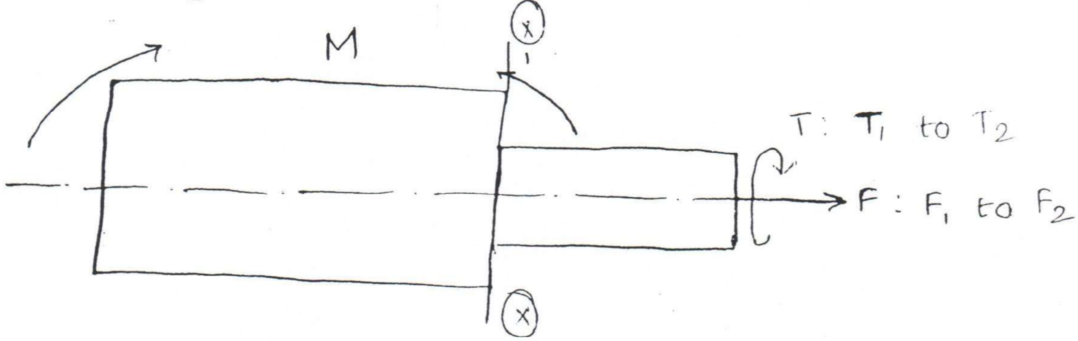
$S_u = 400$ MPa ; $S_y = 300$ MPa ;

Goodman's $\Rightarrow \frac{S_m}{S_u} + \frac{S_v}{S_e} = \frac{1}{F.S}$

$$\Rightarrow \frac{100}{400} + \frac{50}{200} = \frac{1}{F.S} \Rightarrow \boxed{F.S. = 2}$$

Design for combined variable loads :

→ Ex: Rotating shaft may be subjected to Bending moment, Twisting moment and an axial force.

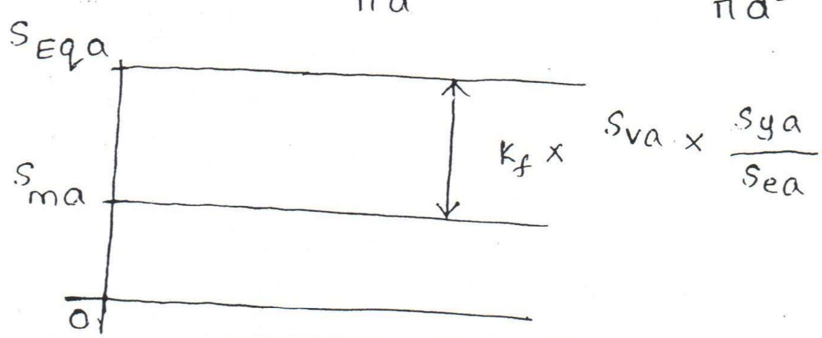


At critical Resisting cross-section xx,

① axial load : F_1 to F_2

$$F_m = \frac{F_1 + F_2}{2} \quad \& \quad F_v = \frac{F_1 - F_2}{2} \; ;$$

$$S_{ma} = \frac{4 F_{mean}}{\pi d^2} \; ; \quad S_{va} = \frac{4 F_v}{\pi d^2} \; ;$$



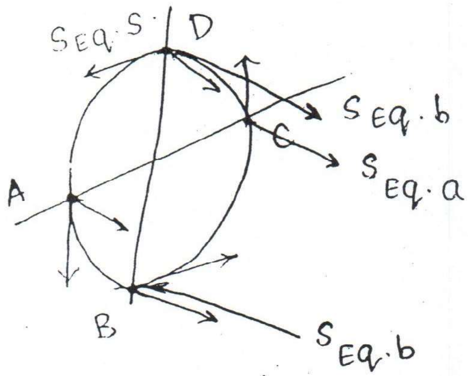
$$\boxed{S_{Eq.a} = S_{ma} + K_f \times S_{va} \cdot \frac{S_{ya}}{S_{ea}}}$$

similarly,

$$\boxed{S_{eq.s} = S_{m.s} + K_f \cdot S_{v.s} \frac{S_{y.s}}{S_{es}}}$$

$$\boxed{S_{eq \cdot b} = S_{mb} + K_f \cdot S_{vb} \cdot \frac{S_{yb}}{S_{eb}}}$$

✓ superimpose the effect of each load at extreme points of critical resisting cross-section



Heavily stressed point = D

$\sigma_x = S_{Eq.a} + S_{Eq.b}$

$\sigma_y = 0$

$\tau_{xy} = S_{Eq.s}$

✓ Find $\sigma_1, \sigma_2, \tau_{max}$

✓ use Theories of Failure to Design.

page-24

(Q4) $\sigma_1 = -50$ MPa to 150 MPa ;

$\sigma_2 = 25$ MPa to 175 MPa ;

$\sigma_{1m} = \frac{-50+150}{2} = 50$ MPa ;

$\sigma_{1v} = \frac{-(-50)+150}{2} = 100$ MPa ;

$\sigma_{2m} = \frac{25+175}{2} = 100$ MPa ;

$\sigma_{2v} = \frac{175-25}{2} = 75$ MPa ;

$S_u = 500$ MPa ;

$S_e = 250$ MPa ;

$K_t = 1.85 = K_f$ — q is not given

F.S. = ? for MDET

$S_{eq1} = \sigma_{1m} + K_f \times \sigma_{1v} \cdot \frac{\sigma_u}{\sigma_e}$

$= 50 + (1.85) \times 100 \times \frac{500}{250}$

$$\boxed{S_{eq1} = 420 \text{ MPa}}$$

$$\boxed{S_{eq\,2} = S_{2m} + k_f \cdot S_{2v} \cdot \frac{Su}{Se}}$$

$$= 100 + (1.85) \times 75 \cdot \frac{500}{250}$$

$$\boxed{S_{eq\,2} = 377.5 \text{ MPa.}}$$

$$\text{MDET} \Rightarrow (S_{eq1})^2 + (S_{eq2})^2 - (S_{eq1} \cdot S_{eq.2}) = \left(\frac{\sigma_y}{F.S.}\right)^2$$

$$(420)^2 + (377.5)^2 - (420 * 377.5) = \left(\frac{500}{F.S.}\right)^2$$

$$400.44 = \frac{500}{F.S.} \Rightarrow F.S. = 1.24$$

$$\therefore \text{MDET} \Rightarrow \boxed{F.S. = 1.24}$$

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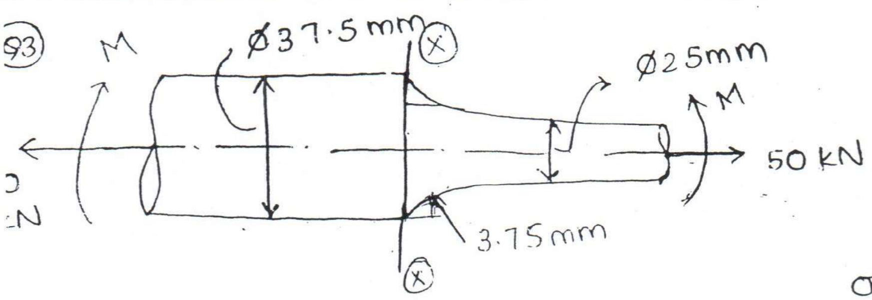
(93)

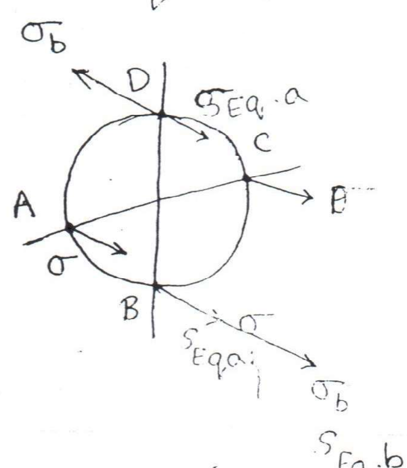


$S_{ut} = 300$ MPa

$S_e = 200$ MPa

$K_t = 1.55$;

$q = 0.9$



$$k_f = 1 + q(k_t - 1)$$

$$k_f = 1 + 0.9(1.55 - 1)$$

$$\boxed{k_f = 1.495}$$

Heavily stressed point is 'B'

$\sigma_m = \sigma$

$\sigma_v = 0$

$$\therefore \quad \sigma_x = \sigma_a + \sigma_b \qquad \sigma_y = 0; \quad \tau_{xy} = 0;$$

$$= \frac{4P}{\pi d^2} + \frac{32M}{\pi d^3}$$

$$S_m = \frac{4 \times 50 \times 10^3}{\pi \times (25)^2} = 101.85 \text{ MPa};$$

$$S_{eq.1} = S_m + K_f \cdot S_v \cdot \frac{S_u}{S_e}$$

$$S_{Eq.a} = S_{ma} + K_f \cdot \cancel{S_{va}} \cdot \frac{S_y}{S_e}$$

$\cancel{S_{va}} \to 0$

$$\boxed{S_{Eq.a} = 101.85 \text{ MPa}}$$

$$S_{Eq.b} = S_{mb} + K_f \cdot S_{mvb} \times \frac{S_{yb}}{S_{eb}}$$

For rotating shafts, stress induced is a completely reversed Bending stress

$(M = N\text{-}m)$

$$\sigma_{bm} = 0 \qquad \sigma_{bv} = \frac{32M}{\pi d^3} = \frac{32\,M}{\pi(0.025 \times 10^6)} = 0.651\,M \text{ MPa}$$

$$S_{Eqb} = 1.495 \times 0.651\,M \times \frac{300}{200}$$

$$\boxed{S_{Eqb} = 1.4598\,M} \text{ MPa}$$

Superimposing effects, $\sigma_x = S_x = S_{Eq.} + S_{Eq.b}$

$$\boxed{\sigma_x = 101.85 + 1.4598\,M}\ ; \sigma_y = 0 ; \tau_{xy} = 0$$

$$\boxed{\sigma_x = \sigma_1}$$

$$\text{MPST} \Rightarrow \sigma_1 = \frac{\sigma_{ut}}{F.S.} \Rightarrow 101.85 + 1.459\,M = \frac{300}{1.}$$

$$\boxed{M = 135.73 \text{ N-m}}$$ (assume F.S = 1)

Q10 – Q11) $\sigma = -130\ MPa$ to $+130\ MPa$

$\tau_T = 16\ MPa$ to $57\ MPa$.

$\sigma_m = S_m = 0 \qquad \sigma_v = S_v = \frac{130-(-130)}{2} = 130\ MPa$

$\tau_{Tm} = S_{sm} = \frac{57+16}{2} = 36.5\ MPa$;

$\boxed{S_{ut} = 1400\ MPa}$

$K_{sur} = 0.76$

$\tau_{TV} = S_{sv} = \frac{57-16}{2} = 20.5\ MPa$;

$K_{size} = 0.85$

$K_{rel} = 0.897$

$K_f = 1 + q(K_t - 1)$

$K_t = 1.85$

$q = 0.95$

$= 1 + 0.95(1.85-1)$

$\boxed{K_f = 1.8075}$

$S_{eb} = 0.5\ S_{ut}$

$S_{Eq.n} = S_{mn} + K_f \cdot S_{mv} \times \frac{S_{ut}}{S_{em}}$

$S_{Eq.n} = 0 + 1.8075 \times 130 \times \frac{0.5(1400)}{S_{em}} = \frac{328965}{S_{em}}$

$\times \quad \boxed{S_{Eq.n} = \frac{328965}{S_{em}}}$ – wrong

$S_{Eq.s} = S_{sm} + K_f \times S_{sv} \times \frac{S_{ut}}{S_e}$

$= 36.5 + (1.8075)(20.5) \times \frac{(1400)0.5}{S_e}$

$\boxed{S_{Eq.s} = 36.5 + \frac{51,875.25}{S_e}}$

$$S_{eb} = 0.5 \times S_{ut} = 700 \text{ MPa};$$

$$S_{em} = S_e \times K_{sf} \times K_s \times K_r$$

$$= 0.85 \times 0.897 \times 0.76 \times 700$$

$$\boxed{S_{em} = 405 \text{ MPa}}$$

$$\therefore \quad S_{Eq.n} = \frac{(328965)0.5}{S_{em}^*}$$

$$\boxed{S_{em}^* = \frac{S_{em}}{K_f} = 225 \text{ MPa}}$$

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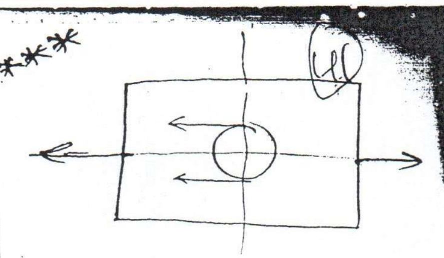


$$\sigma_{max} = K_f \times \frac{P}{(b-d)t} = \frac{\sigma_{em}}{F.S.}$$

$$\frac{P}{(b-d)t} = \left(\frac{\sigma_{em}}{F.S}\right) \times \frac{1}{K_f}$$

$$= \left(\frac{\sigma_{em}}{K_f}\right) \times \frac{1}{F.S}$$

$$\boxed{\sigma_{em} = \frac{S_e * K_A K_B K_C K_D}{K_f}}$$

$$\therefore \quad S_{Eq.n} = \frac{328965}{225} = \frac{1462.06 \text{ MPa}}{2} = 731.03 \text{ MPa}$$

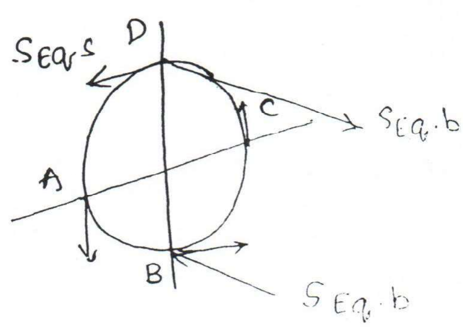
wrong

$$S_{Eq.s} = 36.5 + \frac{51875.25}{225} = 267.05 \text{ MPa}$$

$$S_{Eq.b} = 0 + 1.8075 \times 130 \times \frac{1400}{405} = 812.26 \text{ MPa}$$

correct

$$S_{Eq.s} = 36.5 + 1.8075 \times (20.5) \times \frac{0.5 \times (1400)}{0.5 \times (405)} = 164.58 \text{ MPa}$$



HSP B and D.

$$\sigma_x = S_{Eq.b} = 812.26 \text{ MPa}$$

$$\sigma_y = 0$$

$$\tau_{xy} = 164.58 \text{ MPa},$$

$$= S_{Eq.s}$$

MDET $\Rightarrow \sigma_x^2 + 3\tau_{xy}^2 = \left(\frac{\sigma_y}{F.S.}\right)^2 \rightarrow (\because \sigma_y = 0)$

$$(812.26)^2 + 3(164.58)^2 = \left(\frac{1400}{F.S}\right)^2$$

$$860 = \frac{1400}{F.S} \Rightarrow \boxed{F.S = 1.6}$$

Goodman's equation

$$\Rightarrow \frac{S_m}{S_u} + \frac{S_v}{S_e} = \frac{1}{F.S}$$

DESIGN OF MACHINE ELEMENTS :

DME - I :

Design of Miscellaneous Elements

1. Design of Rivetted Joints.
2. Design of Bolted Joints
3. Design of welded Joints.
4. Springs. $\rightarrow$ S.O.M.

Joint : Joining of two parts together.

JOINTS

- Permanent Joints $\rightarrow$ welded Joints
- Temperary Joints $\rightarrow$ Bolted Joints
- Semi-permanent joints $\rightarrow$ Rivetted Joints