SOLUTION :— (MIDTERM) Ist BIET, SIKAR DESIGN OF MACHINE ELEMENT - IInd (6ME)

Q.(i) What is stress concentration ? Explain causes & mitigation of stress concentration.

Ans. Stress Concentration :- Localisation of stress due to abrupt changes in concentration is called stress concentration. It occurs for all kinds of stresses in the presence of fillet, notches, holes, keyways, splines, surface roughness or scratches etc.

→ The material near the edges is stressed considerably higher than the average value.

→ The max. stress occurs at some points on the fillets and is directed parallel to the boundary at that point.

⇒ Causes of stress Concentration :-

The causes of stress concentration are as follows:

(i) Variation in properties of Materials :- In design of machine components, it is assumed that the material is homogeneous throughout the component. In practice, there is variation in material properties from one end to another due to the following factors:

(a) Internal cracks and flaws like blow holes
(b) Cavities in welds
(c) Air holes in steel components
(d) Non-metallic or foreign inclusions.

(ii) Load Application :- Machine components are subjected to forces. These forces act either at a point or over a small area on the component. Since the area is small, the pressure at these points is excessive. This results in stress concentration. The examples of these load applications are as follows

(a) contact b/w the meshing teeth of the driving and the driven gear.
(b) contact b/w the cam & the follower.
(c) contact b/w the balls & the races of ball bearing.

(iii) Abrupt changes in section:- In order to mount gears, sprockets, pulleys and ball bearings on a transmission shaft, steps are cut on the shaft an shoulders are provided from assembly considerations. Although these features are essential, they create change of the cross-section of shaft.

(iv.) Discontinuities in the component:- Certain features of machine components such as oil holes or oil grooves, keyways and splines, and screw threads result in discontinuities in the cross section of the component. There is stress concentration in the vicinity of these discontinuities.

(v.) Machining Scratches:- Machining scratches, stamp marks of inspection marks are surface irregularities, which cause stress concentration.

→ It can be avoided by

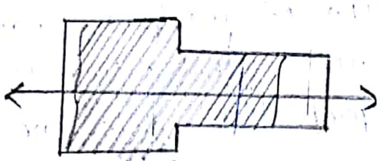
(i) smooth surface produced.

(ii) Load should be uniform on the system.

(iii) sharp edges must be avoided.

(iv) Avoiding fillets, notches, holes, keyways, splines, surface roughness.

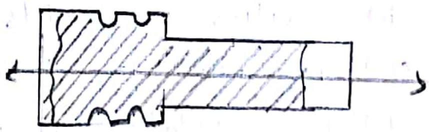
⇒ Mitigation of Stress Concentration:- The mitigation of stress concentration means that the stress flow lines shall maintain their spacing as far as possible in order to improve the situation.



(a) Poor

(b) Good

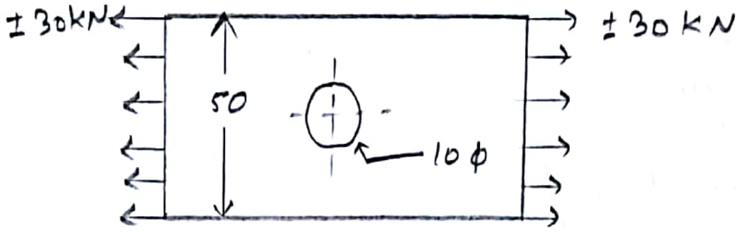
(c) Preferred



(d) Preffered

In fig. (a) we see that stress lines tend to bunch up & cut very close to the sharp re-entrant corner. In order to improve the situation, fillets may be provided, as shown in fig (b) & (c) to give more equally spaced flow lines.

Q.2 A plate made of 20C8 steel ($S_{ut} = 440\ N/mm^2$) in hot rolled and normalised condition as shown in fig. It is subjected to a completely reversed axial load of 30 kN. The notch sensitivity factor 'q' can be taken as 0.8 and the expected reliability is 90%. The size factor is 0.85. The factor of safety is 2. Determine the plate thickness for infinite life.



Sol^n :-

given data

Component material - steel

$S_{ut} = 440\ N/mm^2$

$P = \pm 30 kN = 30{,}000 N$

$q = 0.8$

reliability factor $K_c = .9$

size factor $K_b = .85$

FOS $= 2$

$w = 50 mm$

$a = 10 mm$

Find out

Thickness $(t) = ?$

formula used

Variable stress $\sigma_v = \frac{P}{A}$

$\sigma_v = \frac{P}{(w-a)\,t}$

$\sigma_v = \frac{S_e}{FOS}$

$S_e = K_a K_b K_c K_d S'_e$

$S'_e = 0.5 \times S_{ut}$

$K_d = \frac{1}{K_f}$

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

$K_t =$ from data book

$\therefore \frac{a}{w} = \frac{10}{50} = 0.2$

So $\boxed{K_t = 2.5}$

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

$K_f = 1 + 0.8(2.5 - 1)$

$\boxed{K_f = 2.2}$

$\boxed{K_d = \frac{1}{K_f} = 0.45}$

for hot rolled steel at $S_{ut} = 440\ N/mm^2$

$\boxed{K_a = 0.6}$

& $S'_e = 0.5 \times S_{ut}$

$S'_e = 0.5 \times 440 = 220\ N/mm^2$

$S_e = K_a K_b K_c K_d S'_e$

$S_e = (0.6)(.85)(.90)(0.45)(220)$

$\boxed{S_e = 45.44 \ N/mm^2}$

$\because \quad \sigma_v = \dfrac{S_e}{FOS}$

$\sigma_v = \dfrac{45.44}{2}$

$\boxed{\sigma_v = 22.72 \ N/mm^2}$

$\because \quad \sigma_v = \dfrac{P}{(w-a)t}$

$t = \dfrac{P}{(w-a)\sigma_v}$

$t = \dfrac{30{,}000}{(50-10)22.72}$

$\boxed{t = 33 \ mm}$

Q. Explain soderberg & Goodman lines.

Ans:- When a component is subjected to fluctuating stresses there is mean stress (σ_m) as well as stress amplitude (σ_a). It has been observed that the mean stress component has an effect on fatigue failure when it is present in combination with an alternating component.

In the fatigue diagram, the mean stress is plotted on the abscissa. The stress amplitude is plotted on the ordinate. The magnitude of σ_m & σ_a depend upon the magnitudes of maximum & minimum force acting on the component. When stress amplitude (σ_a) is zero the load is purely static and the criterion of failure is S_{ut} or S_{yt}.

When the mean stress (σ_m) is zero, the stress is completely reversing and the criterion of failure is the endurance limit 'S_e' that is plotted on the ordinate.

⇒ Gerber Curve:- A parabolic curve joining 'S_e' on the ordinate to ultimate stress (S_{ut}) on the abscissa, is called gerber curve.

⇒ Soderberg line:- A straight line joining 'S_e' on the ordinate to yield stress (S_{yt}) on the abscissa, is called soderberg ~~curve~~ line.

⇒ Goodman line:- A straight line joining 'S_e' on the ordinate to ultimate stress (S_{ut}) on the abscissa, is called goodman line.

We will apply form for the eq^n of straight line-

$$\frac{x}{a} + \frac{y}{b} = 1$$

where a & b are the intercepts for the line on the abscissa & ordinate axes respectively.

→ Applying the formula the eqn of Soderberg line is given by.

$$\Rightarrow \frac{\sigma_m}{S_{yt}} + \frac{\sigma_v}{S_e} = 1$$

$$\frac{\sigma_m}{S_{yt}} + \frac{\sigma_v}{S_e} = \frac{1}{FOS}$$

→ Soderberg line eqth.

→ Similarly the eqth of goodman line is given by.

$$\rightarrow \frac{\sigma_m}{S_{ut}} + \frac{\sigma_v}{S_e} = 1$$

$$\frac{\sigma_m}{S_{ut}} + \frac{\sigma_v}{S_e} = 1/FOS$$

Goodman line eqth.

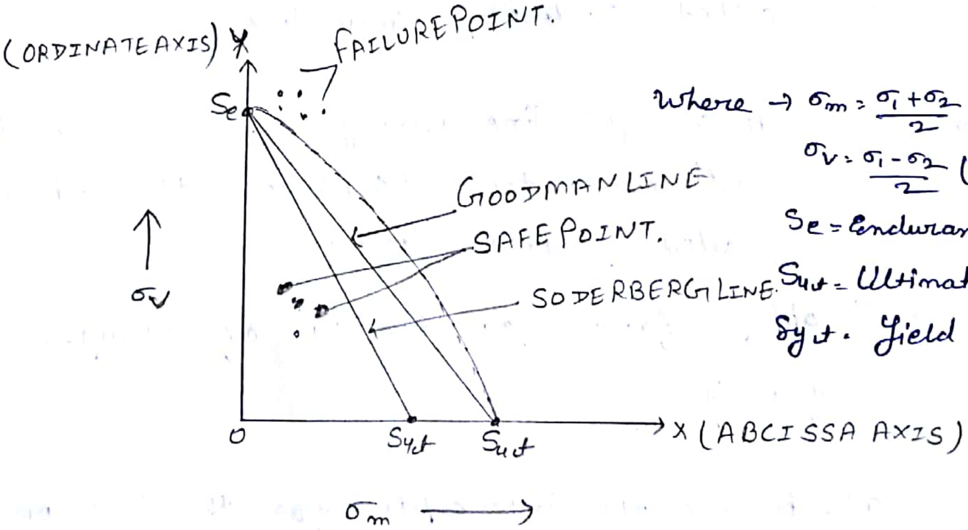
→ We applying following form for eqth of parabola is given by.

$$\left(\frac{\sigma_m}{S_{ut}}\right)^2 + \frac{\sigma_v}{S_e} = 1$$

$$FOS\left(\frac{\sigma_m}{S_{ut}}\right)^2 + \frac{\sigma_v}{S_e} = 1/FOS$$

→ Gerber Curve eqth.

<u>Diagram</u> → For Gerber Curve, Goodman line & Soderberg line.



where → $\sigma_m = \frac{\sigma_1 + \sigma_2}{2}$ (mean Stress)

$\sigma_v = \frac{\sigma_1 - \sigma_2}{2}$ (Variable stress)

S_e = Endurance Strength.

S_{ut} = Ultimate Strength.

S_{yt} = Yield Strength

Q.2 A rotating bar made of steel 45C8 ($S_{ut} = 630\ N/mm^2$) is subjected to a completely reversed bending stress. The corrected endurance limit of the bar is $315\ N/mm^2$. Calculate the fatigue strength of the bar for a life of 90,000 cycles.

Soln.:-

given data	Find out	Formula used
→ Component material - steel → $S_{ut} = 630\ N/mm^2$ → Completely reversed bending stress → Corrected endurance limit $S'_e = 315\ N/mm^2$ → $N = 90{,}000$ cycles	Fatigue strength (S_f) = ?	By using S-N curve. $S_e = 0.9\ S_{ut}$

$$S_e = 0.9 \times S_{ut}$$
$$S_e = 0.9 \times 630$$
$$S_e = 567\ N/mm^2$$

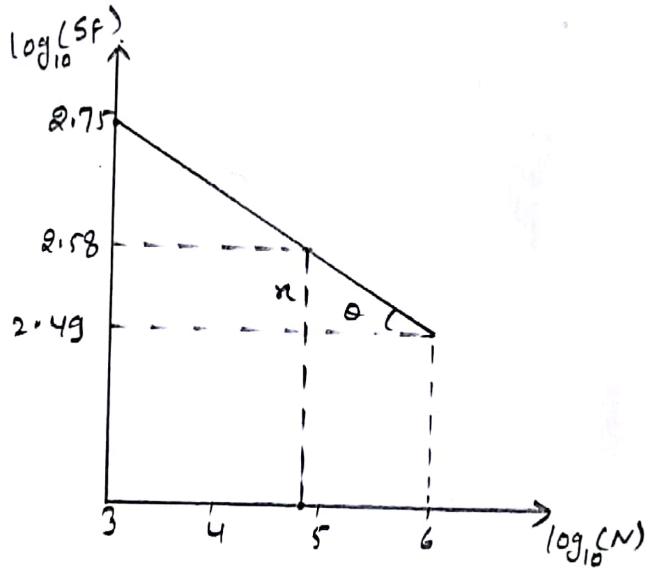
So $\log_{10}(S_e) = \log_{10}(567) = 2.75$

$\because\ S'_e = 315\ N/mm^2$

$\log_{10}(S'_e) = \log_{10}(315) = 2.49$

& $N = 90{,}000$ cycles

$\log_{10}(N) = \log_{10}(90{,}000) = 4.95$



$$\because\ \tan\theta = \frac{2.75 - 2.49}{6-3}$$

$$\boxed{\tan\theta = 0.0867}$$

from another triangle

$$\tan\theta = \frac{x}{6-4.95} = \frac{x}{1.05}$$

$$x = \tan\theta \times 1.05$$

$$\boxed{x = 0.09}$$

$$\log_{10}(S_f) = 2.49 + 0.09 = 2.58$$

$$\boxed{S_f = 380\ N/mm^2}$$

Q 2 Design a cylinder for 4-stroke diesel engine, B.P. = 7.5 kW, speed = 1400 rpm, mean effective indicated pressure = 0.35 MPa, Mechanical efficiency = 80%, max. gauge pressure = 3.5 MPa, cylinder liner & head made of gray cast iron FG 260, $\mu = 0.25$, stud is made of plane carbon steel, $S_{yt} = 380\ N/mm^2$ & FOS = 6.

Solⁿ:- step I :- Dia & length of cylinder

assumptions ⇒ $L = 1.5D$

length of cylinder $L_c = 1.15L$

we know that $I.P. = \frac{P_m L A n K}{60}$ (W)

$$\eta_{mech} = \frac{B.P.}{I.P.}$$

$$I.P = \frac{B.P.}{\eta_{mech}} = \frac{7.5 \times 1000}{0.80} = 9.375 \times 10^3\ W.$$

$$\because\ I.P. = \frac{P_m L A n K}{60}$$

$$9.375 \times 10^3 = \frac{(.35) \times \left(\frac{1.5D}{1000}\right)\left(\frac{\pi}{4} D^2\right) \times (700)(1)}{60}$$

$\left(N = \frac{N}{2} \text{ for 4-stroke}\right)$

$$\boxed{D = 124.93 \approx 125\ mm}$$

so $L = 1.5D$

$$\boxed{L = 187.5\ mm}$$

⇒ step II :-

Thickness of cylinder liner

$$t = \frac{PD}{2\sigma_t} + 3.8$$

$$t = \frac{3.5 \times 125}{2 \times 43.33} + 3.8$$

$$\boxed{t = 8.85 \approx 10\ mm}$$

$\sigma_t = \frac{S_{ut}}{FOS} = \frac{260}{6}$

$\sigma_t = 43.33$

⇒ step III :- Apperent stresses

(i) circumferential stress $(\sigma_c) = \frac{PD}{2t} = \frac{3.5 \times 125}{2 \times 10} = 22\ N/mm^2$

(ii) longitudnal stress $(\sigma_l) = \frac{\sigma_c}{2} = 11\ N/mm^2$

⇒ step IV :-

Thickness of cylinder head $T_h = 0.31 D \sqrt{\frac{P}{\sigma_t}}$

$$T_h = 0.31 \times 125 \times \sqrt{\frac{3.5}{43.33}}$$

$$T_h = 11.01$$

$$\boxed{T_h \approx 12 \text{ mm}}$$

→ No. of studs $i = 0.015 D + 4$

$$i = 0.015 \times 125 + 4$$

$$i = 5.875$$

$$\boxed{i \approx 6}$$

→ Root dia of stud

$$d_r = D\sqrt{\frac{P}{i\sigma_t}} = 125 \times \sqrt{\frac{3.5}{6 \times 63.33}}$$

$\sigma_t = \frac{S_{yt}}{FOS}$

$\sigma_t = \frac{380}{6} = 63.33$

$$\boxed{d_r = 12 \text{ mm}}$$

→ dia of studs

$$d = \frac{d_r}{0.8} = \frac{12}{0.8}$$

$$\boxed{d = 15 \text{ mm}}$$

⇒ step IV :-

Pitch dia of studs $D_p = D + 3d$

$$D_p = 125 + 3 \times 15$$

$$\boxed{D_p = 170 \text{ mm}}$$

$$\text{Pitch} = \frac{\pi D_p}{i} = \frac{\pi \times 170}{6}$$

$$\boxed{\text{Pitch} = 89.01 \text{ mm}}$$

The design is safe. because pitch (89.01) is lie between $19\sqrt{d}$ to $28.5\sqrt{d}$ (i.e. from 73.58 to 110.38)

Q.(2.) The following data is given for the piston of four - stroke diesel engine:

Cylinder bore = 250 mm

Maximum gas pressure = 4 MPa

Allowable bearing pressure for skirt = 0.4 MPa

(i) Ratio of side thrust on liner to max. gas load on piston = 0.1

width of top land = 45 mm

width of ring grooves = 6 mm

Total no. of piston rings = 4

Axial thickness of piston rings = 7 mm

Calculate:

(i) length of skirt; and

(ii) length of piston.

Soln:- given data:-

$D = 250$ mm

$P_{max} = 4$ MPa $= 4$ N/mm^2

$\mu = 0.1$

$P_b = 0.4$ MPa $= 0.4$ N/mm^2

$h_1 = 45$ mm

$h_2 = 6$ mm

$h = 7$ mm

$z = 4$

Step I:- Length of skirt

$$\mu \left(\frac{\pi D^2}{4}\right) P_{max} = P_b D l_s$$

$$(0.1)\left(\frac{\pi (250)^2}{4}\right)(4) = (0.4)(250)\, l_s$$

$$\boxed{l_s = 196.35 \text{ mm}}$$

step II :- Length of piston

length of ring section $= 4h + 3h_2$

$= 4(7) + 3(6)$

$= 46$ mm

length of piston $= h_1 +$ length of ring section $+ l_s$

$= 45 + 46 + 196.35$

$L = 287.35$ mm

$\boxed{L \simeq 288 \text{ mm}}$

Acc. to empirical relationship,

$L = D$ to $1.5D = 250$ to 375 mm

$\therefore D < L > 1.5D$