UNIT –I

STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS

PART –A

1. **What is 'Adaptive design and Optimum design? (Dec 2007, 2011, 2012)**

Adaptive design: It is a design process where a new product is developed just by making small changes to the existing product.

Optimum design: Optimization is the process of maximizing a desired quantity or minimizing an undesired one.

2. **List some factors that influence machine design. (Dec 2010)**

Strength, stiffness, surface finish, tolerances, manufacturability, ergonomics and aesthetics, working atmosphere, cost, safety and reliability.

3. **Describe material properties hardness stiffness and resilience. (Apr 2009, Nov 2009, Dec 2013)**

Hardness: It is the ability of the material to resist scratching and indentation

Stiffness: It is the ability of the material to resist deformation under loading

Resilience: It is the ability of the material to resist absorb energy and to resist shock and impact load.

4. **What is interchangeable manufacture?**

Manufacturing process in which the produced parts that go in to assembly may be Selected at random from a large number of plates.

5. **What are unilateral and bilateral tolerances?(May 2013)**

A unilateral tolerance is tolerance in which variation is permitted only in one direction from, the specified direction. Eg- $1800^{+0.060/-0.060}$

Bilateral tolerance is tolerance in which variartion is permitted in both direction from the specified direction. eg- $1800^{+0.060/-0.060}$

6. **Differentiate between hardness and toughness of materials. (May 2014)**

s.no	Hardness	Toughness
1.	It refers yhe energy required to deform a material	It refer the total energy which can be used before the material breaks
2.	Hardness is the characteristic of a solid material expressing its resistance to permanent deformation	Toughness is the resistance to fracture of a material when stressed
3.	Hardness is the ability to withstand localized deformation at the surface.	Toughness is the measure of a material ability to absorb energy without breaking or fracture

7. **List at least two methods to improve the fatigue strength. (Nov 2008)**
   - Annealing
   - Plastic coating
   - Cold straining

8. **Determine the force required to punch a hole of 20mm diameter in a 5mm thick plate with ultimate shear strength of 250MPa. (Nov 2014)**

Given data:

Diameter, d=20mm

Thickness, t= 5mm

Shear strength, τ = 250MPa=250N/mm^2

Solution: Force F= π d t τ = π.20.5.250 = 78.54KN

9. State the different between straight and curved beams. (Dec 2012)

Feature	Straight beam	Curved beam
Centroidal axis and neutral axis	Are coincident	Are not coincident. Neutral axis is shifted towards the centre of curvature
Stress developed	Same throughout the section	Different at inner and outer radii of the section

10. Give some methods of reducing stress concentration.(Dec 2010)

i. Avoiding sharp corners.

ii. Providing fillets.

iii. Use of multiple holes instead of single hole

iv. Undercutting the shoulder parts.

11. What are the factors that govern selection of materials while designing a machine component? (Dec 2010)

- Required material properties
- Manufacturing easy
- Material availability
- Cost

12. Define stress concentration and stress concentration factor.(Apr 2009, May 2012, 2014)

Stress concentration is the increase in local stresses at points of rapid change in cross section or discontinuities.

Stress concentration factor is the ratio of maximum stress at critical section to the nominal stress. $K_t=\sigma_{max}/ \sigma_o$

13. Explain notch sensitivity. State the relation between stress concentration factor, fatigue stress concentration factor and notch sensitivity.

Notch sensitivity (q) is the degree to which the theoretical effect of stress concentration is actually reached. The relation is, $K_f = 1 + q (K_t - 1)$

14. What are the methods used to improve fatigue strength? (Dec 2013)

- Cold working like shot penning, burnishing.
- Heat treatment such as induction hardening, case hardening, nitrating
- Pre-stressing

15. State Rankine theory of failure and its limitations.

Rankine theory of failure: According to this theory, the failure or yielding occurs at a point in a member when the maximum principal or normal stress in a bi-axial stress system reaches the limiting strength of the material in a simple tension test.

Limitations: Since the maximum principal or normal stress theory is based on failure in tension or compression and ignores the possibility of failure due to shearing stress, therefore it is not used for ductile materials. However, for brittle materials which are relatively strong in shear but weak in tension or compression, this theory is generally used.

16. Define modulus of resilience and proof resilience. (April 2017)

The total strain energy stored in a body is commonly known as resilience. Resilience is also defined as the capacity of a strained body for doing work on the removal of the straining force.

Proof resilience is the maximum amount of strain energy stored in the body, when the body is stressed upto elastic limit.

Modulus of resilience is is the maximum amount of strain energy stored in the body per unit volume, when the body is stressed upto elastic limit.

UNIT-I

STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS

PART-B

1. A hot rolled steel shaft is subjected to a torsional moment that varies from 330N-m clockwise to 110N-m counterclockwise and an applied bending moment at a critical section varies from +440N-m to −220N-m. The shaft is of uniform cross section and no keyway is present at the critical section. Determine the required shaft diameter. The material has an ultimate strength of 550 MN/m² and a yield strength of 410 MN/m². Take the endurance limit as half the ultimate strength, factor of safety as 2, size factor of 0.85 and a surface finish factor of 0.62. [NOV/DEC 2013]

GIVEN DATA:

T_{max} = Maximum Torsional Moment $= 330\,N\text{-}m = 330 \times 10^3\,N\text{-}mm$

T_{min} = Minimum Torsional Moment $= -110\,N\text{-}m = -110 \times 10^3\,N\text{-}mm$.

M_{max} = Maximum Bending Moment $= 440\,N\text{-}m = 440 \times 10^3\,N\text{-}mm$

M_{min} = Minimum Bending Moment $= -220\,N\text{-}m = -220 \times 10^3\,N\text{-}mm$

K_f = Fatigue Stress Concentration Factor

$= 1$ [No keyway is Present]

σ_u = Ultimate Strength

$= 550\,MN/m^2 = 550 \times 10^6\,N/m^2 = 550 \times 10^6 \times \frac{1}{10^6}\,N/mm^2$

$= 550\,N/mm^2$.

σ_y = Yield Strength

$= 410\,MN/m^2$

$= 410\,N/mm^2$

σ_{-1} = Endurance Strength

$= 0.5\,\sigma_u$

$= 0.5 \times 550$

$= 275\,N/mm^2$

n = F.O.S

$= 2$

B = Size Factor = 0.85

C = Surface Finish factor = 0.62

A = Load Correction factor

A = 0.6 [For Torsional Loading]

= 1 [For Bending Load]

τ_{-1} = Endurance Strength in Shear

$= 0.55 \times \sigma_{-1}$

$= 0.55 \times 275$

$= 151.25 \ N/mm^2$

τ_y = Yield Strength in Shear

$= 0.5 \times \sigma_y$

$= 0.5 \times 410$

$= 205 \ N/mm^2$

To Find: Shaft Diameter, d = ?

Formulae Used:

T_m = Mean Torsional Moment

$$= \frac{T_{max} + T_{min}}{2}$$

T_a = Variable Torsional Moment

$$= \frac{T_{max} - T_{min}}{2}$$

τ_m = Mean Shear Stress

$$= \frac{16 \times T_m}{\pi d^3}$$

τ_a = Variable Shear Stress

$$= \frac{16 \times T_a}{\pi d^3}$$

τ_{eq} = Equivalent shear Stress

$$= \tau_m + \frac{\tau_y \tau_a k_f}{\tau_{-1} ABC}$$

M_m = Mean Bending Moment

$$= \frac{M_{max} + M_{min}}{2}$$

M_a = Variable Bending Moment

$$= \frac{M_{max} - M_{min}}{2}$$

σ_m = Mean Bending Stress

$$= \frac{32 \times M_m}{\pi d^3}$$

σ_a = Variable Bending Stress

$$= \frac{32 \times M_a}{\pi d^3}$$

σ_{eq} = Equivalent Bending Stress

$$= \sigma_m + \frac{\sigma_y \sigma_a k_f}{\sigma_{-1} ABC}$$

σ_1 = Principal Stress

$$= \frac{\sigma_{eq}}{2} + \sqrt{\left(\frac{\sigma_{eq}}{2}\right)^2 + \tau_{eq}^2}$$

τ_{max} = Maximum Shear Stress

$$= \sqrt{\left(\frac{\sigma_{eq}}{2}\right)^2 + \tau_{eq}^2}$$

$$\sigma_1 \leq \left(\frac{\sigma_y}{n}\right); \quad \tau_{max} \leq 0.5 \frac{\sigma_y}{n}$$

SOLUTION:

I FOR VARYING TORQUE

$$T_m = \frac{T_{max} + T_{min}}{2} = \frac{330 \times 10^3 + (-110 \times 10^3)}{2} = 110 \times 10^3 \text{ N-mm}$$

$$T_a = \frac{T_{max} - T_{min}}{2} = \frac{330 \times 10^3 - (-110 \times 10^3)}{2} = 220 \times 10^3 \text{ N-mm}$$

$$T = \frac{\pi}{16} \tau d^3; \quad \tau = \frac{T \times 16}{\pi d^3}$$

$$\tau_m = \frac{16 \times T_m}{\pi d^3} = \frac{16 \times 110 \times 10^3}{\pi d^3} = \frac{5,60,225.39}{d^3} \text{ N/mm}^2$$

$$\tau_a = \frac{16 \times T_a}{\pi d^3} = \frac{16 \times 220 \times 10^3}{\pi d^3} = \frac{11,20,450.79}{d^3} \text{ N/mm}^2$$

$$\tau_{eq} = \tau_m + \frac{\tau_y \tau_a k_f}{A B C \tau_{-1}}$$

$$= \frac{5,60,225.39}{d^3} + \frac{205 \times 11,20,450.79}{d^3 \times 151.25 \times 0.6 \times 0.85 \times 0.62}$$

$$= \frac{53,62,968.96}{d^3}$$

II FOR VARYING BENDING MOMENT:

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{440 \times 10^3 + (-220 \times 10^3)}{2} = 110 \times 10^3 \text{ N-mm}$$

$$M_a = \frac{M_{max} - M_{min}}{2} = \frac{440 \times 10^3 - (-220 \times 10^3)}{2} = 330 \times 10^3 \text{ N-mm}$$

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

$$\sigma_m = \frac{32 \times M_m}{\pi d^3} = \frac{32 \times 110 \times 10^3}{\pi d^3} = \frac{1120450.79}{d^3}$$

$$\sigma_a = \frac{32 \times M_a}{\pi d^3} = \frac{32 \times 330 \times 10^3}{\pi d^3} = \frac{3361352.39}{d^3}$$

$$\sigma_{eq} = \sigma_m + \frac{\sigma_y \sigma_a k_f}{\sigma_{-1} ABC}$$

$$= \frac{1120450.79}{d^3} + \frac{3361352.39 \times 410}{275 \times 1 \times 0.85 \times 0.62}$$

$$= \frac{10,62,9883.12}{d^3}$$

III PRINCIPAL STRESSES:

σ_1 = Maximum Principal Stress

$$\sigma_1 = \frac{\sigma_{eq}}{2} + \sqrt{\left(\frac{\sigma_{eq}}{2}\right)^2 + \tau_{eq}^2}$$

$$= \frac{10629883.12}{2d^3} + \sqrt{\left(\frac{10629883.12}{2d^3}\right)^2 + \left(\frac{53,62,968.96}{d^3}\right)^2}$$

$$= \frac{10629883.12}{2d^3} + \sqrt{\frac{2.82 \times 10^{13}}{d^6} + \frac{2.87 \times 10^{13}}{d^6}}$$

$$= \frac{5314941.56}{d^3} + \frac{7543208.86}{d^3}$$

$$= \frac{12858150.43}{d^3}$$

τ_{max} = Maximum Shear Stress

$$= \sqrt{\left(\frac{\sigma_{eq}}{2}\right)^2 + \tau_{eq}^2}$$

$$= \frac{7543208.86}{d^3}$$

IV DIAMETER CALCULATION:

① $\sigma_1 \le \left(\frac{\sigma_y}{n}\right)$

$$\frac{12858150.43}{d^3} \le \frac{410}{2}$$

$$d^3 \ge \frac{12858150.43}{205}$$

$$d \ge 39.73\ mm$$

② $\tau_{max} \le \frac{0.5\,\sigma_y}{n}$

$$\frac{754320 8.86}{d^3} \le \frac{0.5 \times 410}{2}$$

$$d^3 \ge \frac{7543208.86}{102.5}$$

$$d \ge 41.9\ mm$$

Take Highest value among the two values.

$$d \ge 41.9mm$$

From DDB. P.No.7.20 & from R20 Series the Standard value is 45

$\therefore$ d = diameter of shaft

$$\boxed{d = 45mm}$$

2. A steel rod is subjected to a reversed axial load of 180 KN. Find the diameter of the rod for a factor of Safety of 2. Neglect Column action. The material has an ultimate tensile strength of 1070 N/mm² and yield strength of 910 N/mm². The endurance limit in reversed bending may be assumed to be one half of the ultimate tensile strength. Other Correction factors may be taken as: For axial loading = 0.7; For machined surface = 0.8; For size = 0.85; For stress Concentration = 1.0. [MAY/JUNE 2012]

GIVEN DATA:

Reversed Load = Load which vary from one value of Compressive to the same value of tensile.

$W_{min} = -180 KN = -180 \times 10^3 N.$

$W_{max} = 180 KN = 180 \times 10^3 N$

Factor of Safety $= n = 2$

σ_U = Ultimate Tensile Strength = 1070 N/mm²

σ_y = Yield Strength = 910 N/mm²

σ_{-1} = Endurance Strength in shear

$= 0.5\, \sigma_U = 0.5 \times 1070 = 535$ N/mm²

A = Load Correction factor

= 0.7 (For Axial Loading)

B = Size Factor = 0.85

C = Surface Finish Factor = 0.8

To FIND: d = Diameter of the Rod.

FORMULAE USED:

W_m = Mean Axial Load $= \dfrac{W_{max} + W_{min}}{2}$

W_a = Variable Axial Load $= \dfrac{W_{max} - W_{min}}{2}$

Design Data Book (DDB) Page No: 7.4

Soderberg Equation:

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_a K_f}{\sigma_{-1} ABC}$$

Goodman Equation:

$$\frac{1}{n} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_a K_f}{\sigma_{-1} ABC}$$

SOLUTION:

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{180 + (-180)}{2} = 0$$

$$W_a = \frac{W_{max} - W_{min}}{2} = \frac{180 - (-180)}{2} = 180\,kN = 180 \times 10^3\,N.$$

$$\sigma = \frac{W}{A}\ ; \text{ Where } A = \text{Cross Sectional Area}$$

$$= \frac{\pi}{4} d^2.$$

$$\sigma_a = \frac{W_a \times 4}{\pi \times d^2} = \frac{180 \times 10^3 \times 4}{\pi \times d^2} = \frac{229183}{d^2}\ N/mm^2$$

$$\sigma_m = \frac{W_m \times 4}{\pi \times d^2} = 0$$

Soderberg Equation:

$$\frac{1}{n} = \cancel{\frac{\sigma_m}{\sigma_y}}^{0} + \frac{\sigma_a K_f}{\sigma_{-1} ABC}$$

$$\frac{1}{2} = \frac{229183 \times 1}{d^2 \times 535 \times 0.7 \times 0.85 \times 0.8}$$

$$\boxed{d = 42.42\,mm \simeq 45\,mm}$$

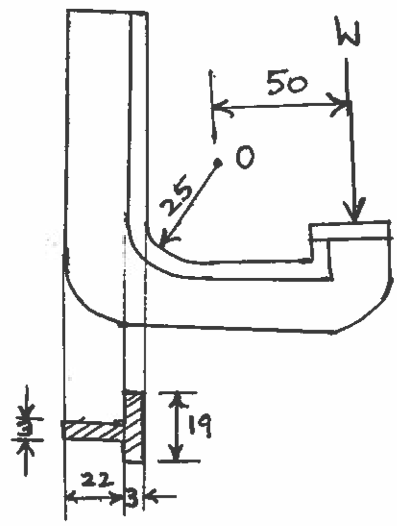
Goodman Equation:

$$\frac{1}{n} = \cancel{\frac{\sigma_m}{\sigma_u}}^{0} + \frac{\sigma_a K_f}{\sigma_{-1} ABC}$$

$$\frac{1}{2} = \frac{229183 \times 1}{d^2 \times 535 \times 0.7 \times 0.85 \times 0.8}$$

$$\boxed{d = 42.42\,mm \simeq 45\,mm}$$

3. A C-Clamp is Subjected to a maximum load of 'W' as shown in fig. If the maximum tensile stress in the clamp is limited to 140 MPa, find the value of load, W. [Nov/Dec 2012]



Given Data:

Load = W

σ_t = Tensile stress = 140 MPa = 140 N/mm²

To Find:

W = Load = ?

Formulae Used:

σ_A = Tensile Stress at inner Section 'A'

$$= \frac{W}{A} + \frac{M y_A}{A e r_i}$$

Where, e = Distance from Centroidal Axis to the Neutral Axis.

$= R - R_n$.

y_A = Distance from the neutral axis to the inner fibre or Section (A).

$= r_n - r_i$

Where, r_n = Radius of Neutral Axis.

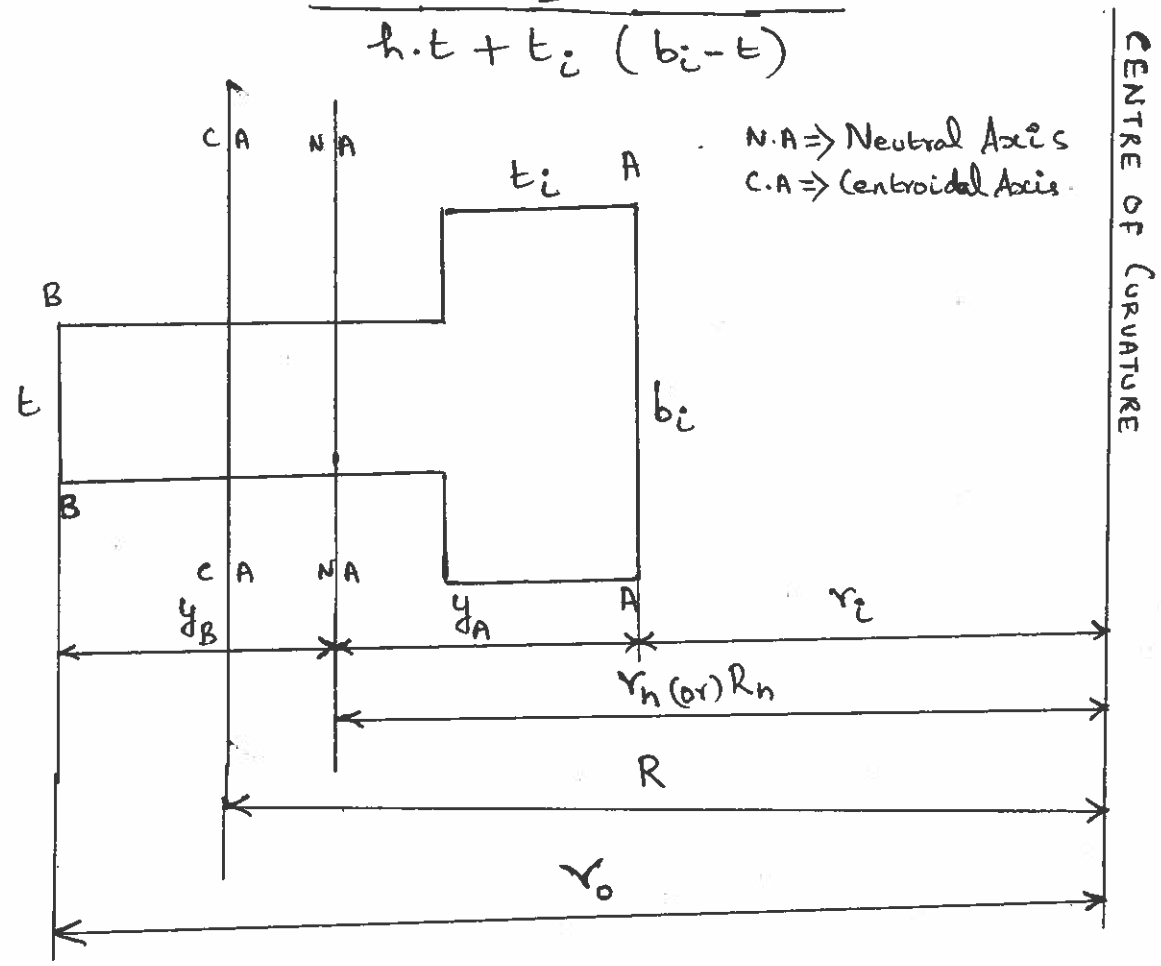
r_i = Radius of Curvature of Inside Fibre (or) Section.

R = Radius of the Centroidal Axis.

Design Data Book Page No: 6.3

$$R_n = \frac{t_i(b_i - t) + t.h}{(b_i - t)\log_e\left(\frac{R_i + t_i}{R_i}\right) + t\log_e\left(\frac{R_o}{r_i}\right)}$$

$$R = R_i + \frac{\frac{1}{2}h^2 t + \frac{1}{2}t_i^2(b_i - t)}{h.t + t_i(b_i - t)}$$



SOLUTION:

$r_o = 25 + 3 + 22 = 50\text{mm}$

$r_i = 25\text{mm}$

$b_i = 19\text{mm}$; $t_i = 3\text{mm}$; $t = 3\text{mm}$; $h = 22 + 3 = 25\text{mm}$

$A = \text{Area} = (3 \times 22) + (3 \times 19) = 123\text{mm}^2$

$$R_n = \frac{t_i(b_i - t) + t.h}{(b_i - t)\log_e\left(\frac{r_i + t_i}{r_i}\right) + t\log_e\left(\frac{r_o}{r_i}\right)}$$

$$= \frac{3 \times (19 - 3) + 3 \times 25}{(19 - 3)\log_e\left(\frac{25 + 3}{25}\right) + 3 \times \log_e\left(\frac{50}{25}\right)} = 31.64\text{mm}$$

$$R = r_i + \frac{\frac{1}{2}h^2 t + \frac{1}{2} t_i^2 (b_i - t)}{h \cdot t + t_i (b_i - t)}$$

$$= 25 + \frac{\frac{1}{2} \times 25^2 \times 3 + \frac{1}{2} 3^2 (19-3)}{25 \times 3 + 3 \times (19-3)}$$

$$= 33.2 \text{mm}$$

$$e = R - R_n = 33.2 - 31.64 = 1.56 \text{mm}$$

$$y_A = R_n - r_i = 31.64 - 25 = 6.64 \text{mm}$$

$$y_B = r_o - R_n = 50 - 31.64 = 18.36 \text{mm}$$

$$M = W \times (50 + R)$$

$$= W \times (50 + 33.2)$$

$$= 83.2 \, W \text{ N-mm}$$

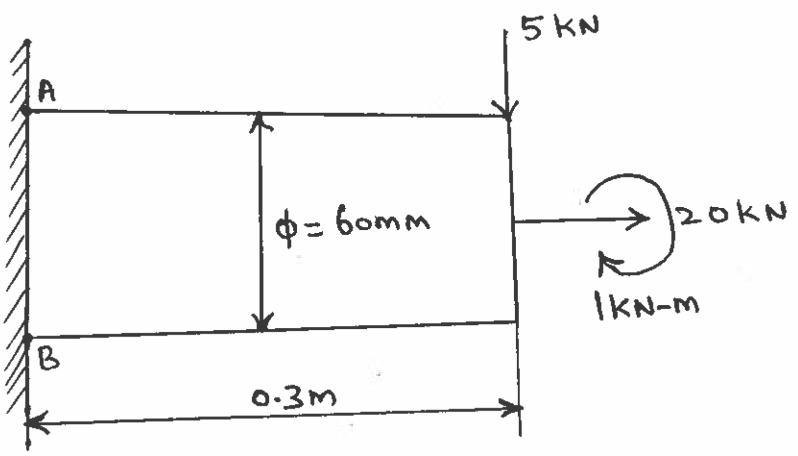
$\sigma_A = \sigma_t$ = Stress at Inner Section is Tensile.

$$\sigma_A = \frac{W}{A} + \frac{M y_A}{A e r_i}$$

$$140 = \frac{W}{123} + \frac{83.2 W \times 6.64}{123 \times 1.56 \times 25}$$

$$\therefore \boxed{W = 1138 \text{N}}$$

4. A 60mm diameter shaft as shown in figure is Subjected to a bending load of 5KN, pure torque of 1KN-m and an axial pulling force of 20KN. Determine the stresses at A & B. [APR/MAY 2010]



Given Data:

P_b = Bending Load $= 5\,kN = 5\times10^3\,N$

P_a = Axial Tensile Load $= 20\,kN = 20\times10^3\,N$

T = Torque or Twisting Moment $= 1\,kN\text{-}m = 1\times10^3\,N\text{-}m$
$= 1\times10^6\,N\text{-}mm$

d = Diameter of Shaft $= 60\,mm$

L = Length of Shaft $= 0.3\,m = 300\,mm$

To Find:

① Stresses at A : $\sigma_1, \sigma_2, \tau_{max}$

② Stresses at B : $\sigma_1, \sigma_2, \tau_{max}$

Formulae Used:

σ_a = Axial Stress $= \dfrac{P_a}{A}$

Where A = Cross Sectional Area of Shaft
$= \dfrac{\pi}{4} d^2$

σ_b = Bending Stress $= \dfrac{M}{Z}$

Where $M = P_b \times L$

$Z = \dfrac{\pi}{32} d^3$

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

T = Torque $= \dfrac{\pi}{16} \tau d^3$

$$\therefore \tau = \frac{16\times T}{\pi d^3}$$

σ_1 = Maximum Principal or Normal Stress

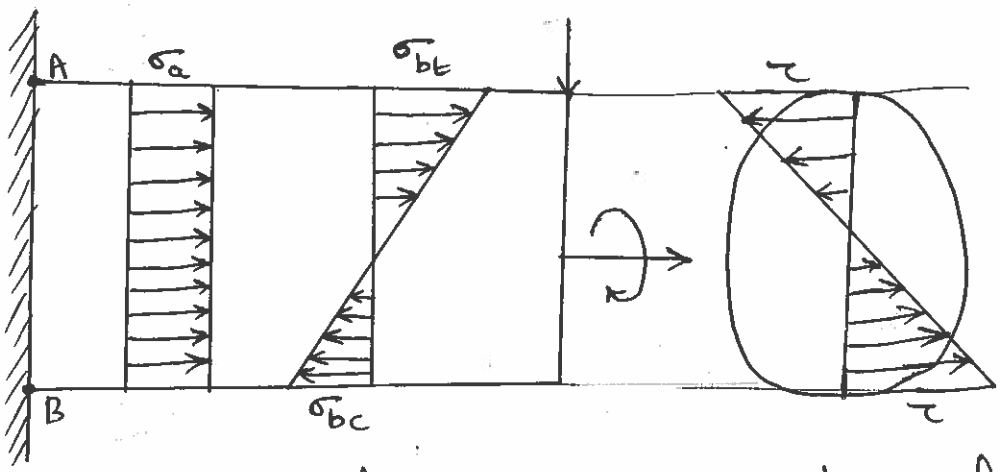
σ_2 = Minimum Principal or Normal Stress

$$\sigma_{1,2} = \frac{\sigma_T}{2} \pm \sqrt{\left(\frac{\sigma_T}{2}\right)^2 + \tau^2}$$

τ_{max} = Maximum Shear Stress

$$= \frac{\sigma_1 - \sigma_2}{2}$$

SOLUTION:



σ_a = Axial Tensile stress due to axial load. all over the cross section

σ_{bt} = Bending stress which is tensile in nature on the upper cross section due to bending or transverse load at the free end.

σ_{bc} = Bending stress which is compressive in nature on the lower cross section due to bending or transverse load at the free end.

τ = Shear stress developed across the cross section due to torque

I Stress due to Axial Load:

$$\sigma_a = \frac{P_a}{A} = \frac{20 \times 10^3}{\left(\frac{\pi}{4} \times d^2\right)} = 7.07\ N/mm^2$$

II Stress due to Bending Load:

M = Bending Moment

$= P_b \times L = 5000 \times 300 = 1500000\ N\text{-}mm$

$$\sigma_{bt} = \sigma_{bc} = \frac{32 \times M}{\pi d^3} = \frac{32 \times 5000 \times 300}{\pi \times 60^3} = 70.7\ N/mm^2$$

III Stress due to Torque:

τ = Shear stress

$$= \frac{16 \times T}{\pi d^3} = \frac{16 \times 1000000}{\pi \times 60^3} = 23.6\ N/mm^2$$

IV STRESSES AT 'A':

σ_T = Total Normal or Principal Stress

$= \sigma_{bt} + \sigma_a = 70.7 + 7.07 = 77.8\ N/mm^2$

$$\sigma_{1,2} = \frac{\sigma_T}{2} \pm \sqrt{\left(\frac{\sigma_T}{2}\right)^2 + \tau^2}$$

$$= \frac{77.8}{2} \pm \sqrt{\left(\frac{77.8}{2}\right)^2 + 23.6^2}$$

$$\boxed{\begin{aligned} \sigma_1 &= 84.4\ N/mm^2 \text{ (Tensile)} \\ \sigma_2 &= -6.6\ N/mm^2 \text{ (Compressive)} \end{aligned}}$$

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} = \frac{84.4 - (-6.6)}{2}$$

$$\boxed{\tau_{max} = 45.5\ N/mm^2}$$

V STRESSES AT 'B'

σ_T = Total Normal or Principal Stress

$= -\sigma_{bc} + \sigma_a$

$= -70.7 + 7.07$

$= -63.6\ N/mm^2$

$$\sigma_{1,2} = \frac{\sigma_T}{2} \pm \sqrt{\left(\frac{\sigma_T}{2}\right)^2 + \tau^2}$$

$$= \frac{-63.6}{2} \pm \sqrt{\left(\frac{-63.6}{2}\right)^2 + 23.6^2}$$

$$\boxed{\begin{array}{l} \sigma_1 = 7.8\, N/mm^2 \text{ (Tensile)} \\ \sigma_2 = -71.4\, N/mm^2 \text{ (Compressive)} \end{array}}$$

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} = \frac{7.8 - (-71.4)}{2}$$

$$\boxed{\tau_{max} = 39.6\, N/mm^2}$$

5. A bolt is subjected to a tensile load of 20 kN and to a shear load of 15 kN. Suggest a suitable size of the bolt according to various theories of failure. Take yield strength = 300 N/mm²; Factor of safety = 2.5; Poisson's ratio = 0.25 [Nov/Dec 2008]

GIVEN DATA:

P_t = Tensile Load = 20 kN = 20×10^3 N

P_τ = Shear Load = 15 kN = 15×10^3 N

$\sigma_y = 300 N/mm^2$; F.O.S = n = 2.5; $\nu = 0.25$

TO FIND:

① d_c = Core diameter of the bolt

② Standard size of the bolt.

FORMULAE USED:

Design Data Book Page No: 7.2

$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

= Principal Stresses.

Design Data Book Page No. 7.3

FAILURE THEORIES

(1) Max. Principal Stress Theory

$$\sigma_1 \leq \frac{\sigma_y}{n}$$

(2) Maximum Shear Stress Theory

$$(\sigma_1 - \sigma_2) \leq \frac{\sigma_y}{n}$$

(3) Maximum Principal Strain Theory

$$(\sigma_1 - \nu\,\sigma_2) \leq \frac{\sigma_y}{n}$$

(4) Maximum Strain Energy Theory

$$\sigma_1^2 + \sigma_2^2 - 2\nu\,\sigma_1\sigma_2 \leq \left(\frac{\sigma_y}{n}\right)^2$$

(5) Maximum Distortion Energy Theory

$$\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 \leq \left(\frac{\sigma_y}{n}\right)^2$$

SOLUTION:

Bolt is subjected to a tensile load and shear load.

∴ Bolt is subjected to a two dimensional state of stress.

Third Direction stress, $\sigma_z = 0$ (i.e, $\sigma_3 = 0$)

I PRINCIPAL STRESSES:

DDB. P.No: 7.2

$$\sigma_{1,2} = \frac{1}{2}\left[(\sigma_x + \sigma_y) \pm \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}\right]$$

$$\sigma_x = \sigma_t = \text{Tensile Stress.}$$

$$\sigma_y = 0 \; ; \; \tau_{xy} = \tau$$

$$\sigma_t = \text{Induced Tensile stress} = \frac{P_t}{A} = \frac{20{,}000}{\left(\frac{\pi}{4} d_c^2\right)}$$

$$= \frac{80{,}000}{\pi d_c^2}$$

Induced Shear Stress, $\tau = \frac{P_\tau}{A} = \frac{15,000}{\left(\frac{\pi}{4} d_c^2\right)}$

$$= \frac{60,000}{\pi d_c^2}$$

$$\sigma_{1,2} = \frac{1}{2}\left[\sigma_t \pm \sqrt{\sigma_t^2 + 4\tau^2}\right]$$

$$= \frac{\sigma_t}{2} \pm \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \frac{\cancel{4}\tau^2}{\cancel{2^2}}}$$

$$= \frac{\sigma_t}{2} \pm \sqrt{\left(\frac{\sigma_t}{2}\right)^2 + \tau^2}$$

$$= \frac{80,000}{\pi d_c^2 \times 2} \pm \sqrt{\left(\frac{80,000}{\pi d_c^2 \times 2}\right)^2 + \left(\frac{60,000}{\pi d_c^2}\right)^2}$$

$$\sigma_1 = \frac{35,685}{d_c^2}\ N/mm^2 \ ; \ \sigma_2 = \frac{-10,220.8}{d_c^2}\ N/mm^2$$

<u>Note:</u> One of the principal stresses is negative (-ve)

$\therefore$ $\sigma_3 = 0$ is not used in calculations.

<u>II Maximum Principal Stress Theory:</u>

$$\frac{\sigma_y}{n} = \frac{300}{2.5} = 120\ N/mm^2$$

$$\sigma_1 \le \frac{\sigma_y}{n}$$

$$\frac{35,685}{d_c^2} \le 120$$

$$\therefore \boxed{d_c \ge 17.24\,mm}$$

III Maximum Shear Stress Theory:

$$(\sigma_1 - \sigma_2) \leq \frac{\sigma_y}{n}$$

$$\left[\frac{35{,}685}{d_c^2} - \left(-\frac{10{,}220.8}{d_c^2}\right)\right] \leq 120$$

$$\frac{45{,}905.8}{d_c^2} \leq 120$$

$$\therefore \boxed{d_c \geq 19.55\,mm}$$

IV Maximum Principal Strain Theory:

$$(\sigma_1 - \nu\sigma_2) \leq \frac{\sigma_y}{n}$$

$$\left[\frac{35{,}685}{d_c^2} - 0.25\left(-\frac{10{,}220.8}{d_c^2}\right)\right] \leq 120$$

$$\frac{38{,}240.2}{d_c^2} \leq 120$$

$$\therefore \boxed{d_c \geq 17.85\,mm}$$

V Maximum Strain Energy Theory:

$$\sigma_1^2 + \sigma_2^2 - 2\nu\sigma_1\sigma_2 \leq \left(\frac{\sigma_y}{n}\right)^2$$

$$\left(\frac{35{,}685}{d_c^2}\right)^2 + \left(-\frac{10{,}220.8}{d_c^2}\right)^2 - 2 \times 0.25 \times \left(\frac{35{,}685}{d_c^2}\right)\left(-\frac{10{,}220.8}{d_c^2}\right) \leq 120^2$$

$$\frac{15.6 \times 10^8}{d_c^4} \leq 14{,}400$$

$$\therefore \boxed{d_c \geq 18.14\,mm}$$

VI Maximum Distortion Energy Theory:

$$\sigma_1^2+\sigma_2^2-\sigma_1\sigma_2 \le \left(\frac{\sigma_y}{n}\right)^2$$

$$\left(\frac{35,685}{d_c^2}\right)^2+\left(-\frac{10,220.8}{d_c^2}\right)^2-\left(\frac{35,685}{d_c^2}\right)\left(\frac{-10,220.8}{d_c^2}\right)\le 120^2$$

$$\frac{17.426\times10^8}{d_c^4}\le 14,400$$

$$\therefore \boxed{d_c \ge 18.65\,mm}$$

Core diameter is maximum according to maximum Shear stress theory

$$d_c = 19.55\,mm$$

For metric thread, Nominal diameter $= \frac{d_c}{0.84}$

$$\boxed{d_c = \frac{19.55}{0.84} = 23.27\,mm}$$

From Design Data Book Page No: 5.49

M24 is the std. size of the Bolt

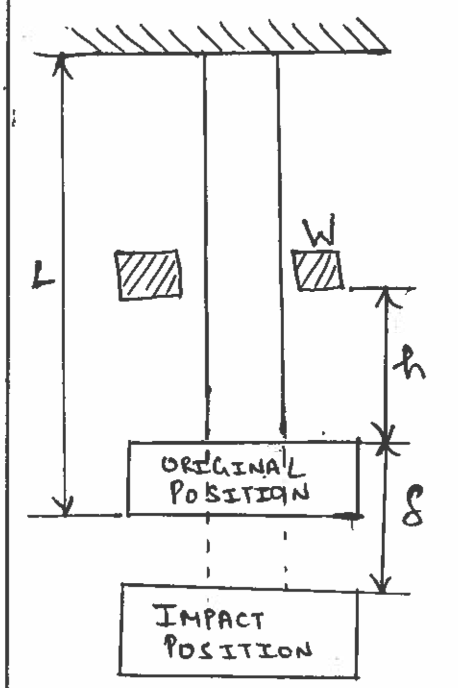
6. An Unknown Weight falls through 10mm on to a Collar rigidly attached to the lower end of a Vertical bar 3m long and 600mm² cross section. The maximum instantaneous extension is 2mm. What is the Corresponding Stress and the Value of the weight. Take $E = 200\,kN/mm^2$. [Nov/Dec 2014]

GIVEN DATA:

h = height = 10mm
δ = Instantaneous Extension = 2mm

L = Length
= 3m
= 3000mm
$A = 600\,mm^2$

$E = 200\,kN/mm^2$
$= 200\times10^3\,N/mm^2$
$= 2\times10^5\,N/mm^2$



To Find: (1) Load, W
(2) Stress, σ

Solution:

Strain Energy Gained by the System } = Potential Energy Lost by Weight

$$\frac{1}{2} P \delta l = W(h+\delta)$$

where P = Equivalent static load

$$= \frac{\delta AE}{L}$$

$$= \frac{2 \times 600 \times 2 \times 10^5}{3000}$$

$$= 80{,}000\ N$$

$$\frac{1}{2} \times 80{,}000 \times 2 = W(10+2)$$

$$W = 6666.7 N$$

$$\sigma = \frac{P}{A} = \frac{80{,}000}{600}$$

$$\sigma = 133.33\ N/mm^2$$

7. Explain Various phases in design using a flow diagram and enumerate the factors influencing the machine design. [MAY/JUNE 2013]

(1) Design Process:

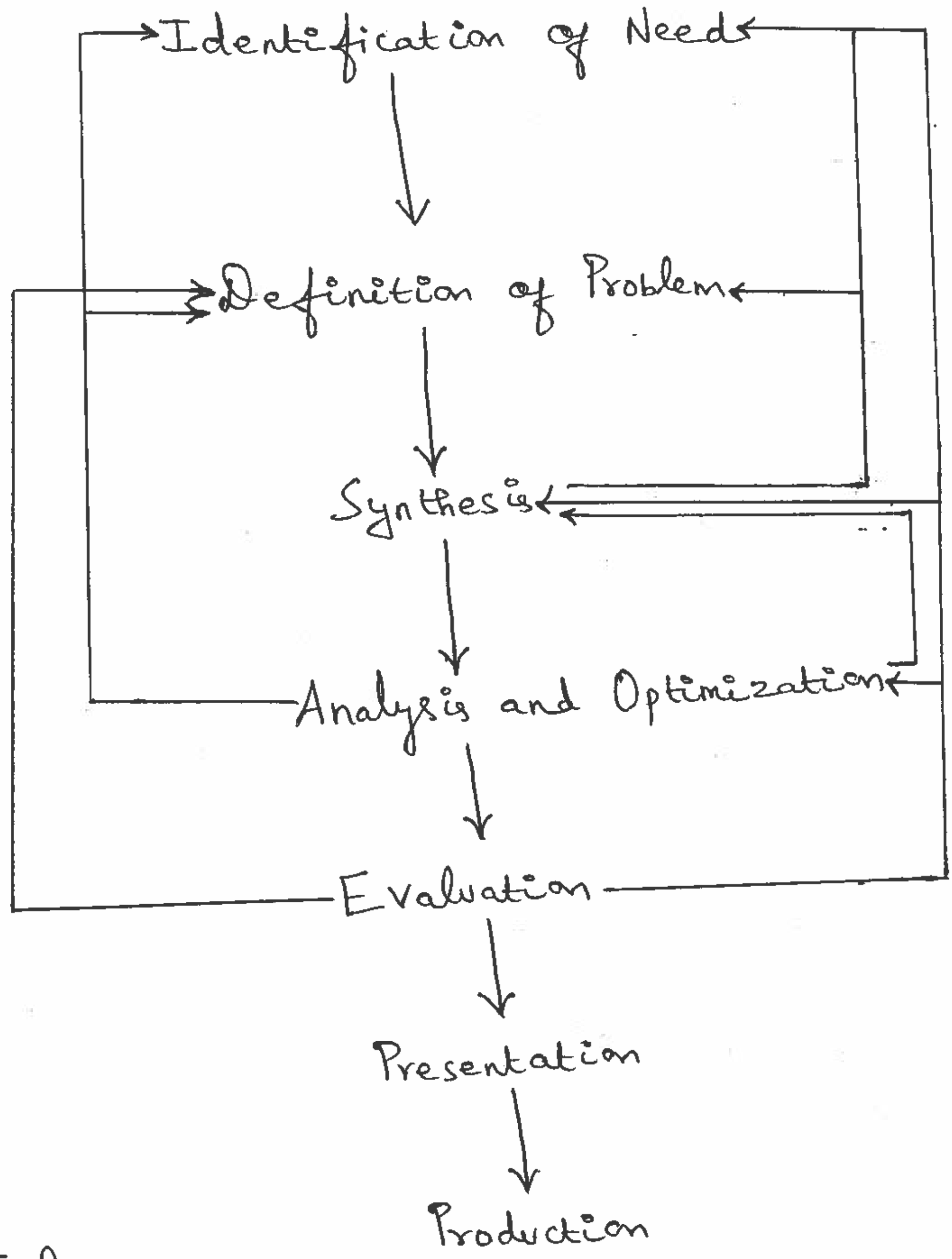
I IDENTIFICATION OF NEED:

Highly creative act.

Need arises from feeling of uneasiness, Criticism on product function and sensing that something is not right.

Aim for which machine to be designed is to satisfy new requirement and to improve the existing design.

Design Process Flow Diagram:



II DEFINITION OF PROBLEM:

Specifications for the object that is to be designed is decided in this step.

Specifications: Input and Output quantities, the characteristics, dimensions of the space that object occupy and limitation of these quantities.

Specification: define the Cost, no. of Components to be manufactured, expected life, the operating temperature and reliability.

III SYNTHESIS :

Selection of mechanism which Consists of Components

Invention of the Concept Design

Select possible mechanism, Investigate it and quantified to get desired motion.

IV ANALYSIS OF FORCES:

Find: ① forces acting on each member of the machine
② Energy transmitted by each member

Mechanism that do not survive analysis are revised, improved or discarded.

Number of mechanisms are analyzed to determine the best performing mechanism: Optimization.

V EVALUATION:

Testing of a prototype in the laboratory to Check whether the mechanism Satisfied the needs?, Reliable?, Economic to manufacture and use? Maintained and adjusted.

VI PRESENTATION :

Detailed drawing of each Component and assembly of machine with complete Specification was prepared and presented to production.

② Factors Influencing the Machine Design:

1. Type of Load:

Internal stresses are developed due to load on a machine Component.

Components to be designed for dynamic and impact load should be stronger than that for Steady load

2. Form and Size of the parts:

Size is inversely proportional to material Strength if the load is kept Constant.

Check that the stresses induced in the designed Cross section are reasonably safe.

3. Selection of Materials:

Properties of the materials and their behaviour under working Conditions.

Designing a lathe bed require Cast iron which is more hard and high Compressive strength.

Spectacles Cover of dial gauge must be highly transparent.

4. Lubrication:

Power loss due to frictional ressistance

Friction at starting is higher than that of running friction

Lubrication applied to all surfaces which move in Contact with others, whether in rotating, sliding or rolling.

5. Use of Standard Parts:

Reduce design Cost by using standard and readily available raw materials and Components:

Exa: Screw, key, Cross Section, Pipes, etc.,

6. Safety of Operation:

Parts should have provisions for safe handling and easy Maintenance

The safety appliances will not interfere with operation of the machine

Any moving part of a machine which is within the zone of a worker is considered an accident hazard.

7. Cost of Construction:

Any designed component should be within peoples buying capacity.

The aim of design engineer under all conditions should be to reduce the manufacturing cost to the minimum

8. Workshop Facilities:

Unless there are suitable workshops available in the nearby places and proper manufacturing methods, designing any component for such situations will become difficult.

Hence the design engineer should be familiar with available workshops and manufacturing methods in his area.

9. Environmental Conditions:

Components operating atmosphere, Corrosive, non Corrosive, Cool or hot climatic Conditions.

Components operated in sea side area should be corrosive ressistant.