DHANALAKSHMI COLLEGE OF ENGINEERING

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DEPARTMENT OF MECHANICAL ENGINEERING

III YEAR MECHANICAL - V SEMESTER

GE6503 – DESIGN OF MACHINE ELEMENTS

ACADEMIC YEAR (2017 - 2018) - ODD SEMESTER

UNIT – 1 (STUDY NOTES)

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UNIT 1 : STEADY STRESSES AND VARIABLE STRESSES IN MACHINE MEMBERS

PART – A

1. What is meant by optimum design? **(N/D – 07/11)**

It is the best design for the given objective function under the specified constraints. It may be achieved by minimizing the undesirable effects.

2. What are the material properties hardness, stiffness and resilience?

(A/M – 09) (N/D - 09)

Hardness is the ability of material to resist scratching and indentation.
Stiffness is the ability to resist deformation under loading.
Resilience is the ability of material to resist absorb energy and to resist shock and impact load.

3. What is Gerber theory? **(N/D - 09)**

Gerber parabola joins endurance stress and ultimate stress (like Goodman line). According to Gerber method, the relationship between σ_m, σ_a, σ_u, σ_{-1} is give by

$$\sigma_a = \sigma_{-1} (1-(\sigma_m / \sigma_u)^2)$$

4. Define – Stress Concentration **(A/M – 09) (M/J - 12)**

Whenever a machine component changes the shape of its cross-section, the simple stress distribution no longer holds good and the neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. It occurs for all kinds of stresses in the presence of fillets, notches, holes, keyways, splines, surface roughness or scratches etc.

5. List the important factors that influence the magnitude of factor safety. **(N/D - 11)**

a) The reliability of the properties of the material and change of these properties during service
b) The reliability of test results and accuracy of application of these results to actual machine parts
c) The reliability of applied load
d) The certainty as to exact mode of failure
e) The extent of simplifying assumptions
f) The extent of localised stresses
g) The extent of initial stresses set up during manufacture
h) The extent of loss of life if failure occurs
i) The extent of loss of property if failure occurs

6. What are mechanical properties of the metals? Write any four mechanical properties.

(M/J - 12)

a) The mechanical properties of the metals are those which are associated with the ability of the material to resist mechanical forces and load
b) These mechanical properties of the metal include strength, stiffness, elasticity, plasticity, ductility, brittleness, malleability, toughness, resilience, creep and hardness

7. State the difference between straight beams and curved beams. **(N/D - 12)**

In straight beams, the neutral axis of the section coincides with its centroidal axis and the stress distribution in the beam is linear. But in case of curved beams, the neutral axis of the cross-section is shifted towards the centre of curvature of the beam causing a non-linear (hyperbolic) distribution of stress.

8. What is 'adaptive design'? Where is it used? Write examples. **(N/D - 12)**

In most cases, the designer's work is concerned with adaptation of existing designs. This type of design needs no special knowledge or skill and can be attempted by designers of ordinary technical training. The designer only makes minor alternation or modification in the existing designs of the product.

9. What are unilateral and bilateral tolerances? **(M/J - 13)**

a) It is the difference between the upper limit and lower limit of a dimension. In other words, it is the maximum permissible variation in a dimension. The tolerance may be unilateral or bilateral.
b) When all the tolerance is allowed on one side of the nominal size, e.g., $20^{+0.004}_{-0.000}$, then it is said to be unilateral system of tolerance.
c) When the tolerance is allowed on both sides of the nominal size, e.g. $20^{+0.002}_{-0.002}$, then it is said to be bilateral system of tolerance. In this case + 0.002 is the upper limit and – 0.002 is the lower limit.

10. What are the various theories of failure? **(M/J - 13)**

a) Maximum principal (or normal) stress theory (also known as Rankine's theory).
b) Maximum shear stress theory (also known as Guest's or Tresca's theory).
c) Maximum principal (or normal) strain theory (also known as Saint Venant theory).

11. Define. 'Design'

Design is a series of activities to gather all the information necessary to realize the designer's idea as a real product.

$$\boxed{\text{Designer's idea}} \xrightarrow{\text{Design process}} \boxed{\text{Real Product}}$$

12. What are the various phases of design process? **(A/M 2006)**

i. Recognition of need.
ii. Definition of problem
iii. Synthesis
iv. Analysis and optimization
v. Evaluation
vi. Presentation

13. List some factors that influence machine design.

i. Strength and Stiffness
ii. Surface finish and tolerances
iii. Manufacturability
iv. Ergonomics and esthetics
v. Working atmosphere

vi. Safety and reliability
vii. Cost

14. Explain Design for Manufacture.

Design for manufacture means designing is done so that manufacturing in the shop floor is possible. It includes prescribing available materials, possible manufacturing methods and achievable.

15. What are the various optimization methods available.

i. Optimization by evaluation
ii. Optimization by intuition
iii. Optimization by trail and error
iv. Optimization by numerical algorithm

16. Identify the steel designated as 50 C 4 as per BIS.

It is plain carbon steel, average % of carbon = 50/100 = 0.5% and average % of Manganese = 4 = 0.04%.

17. Mention some standard codes of specification of steels. (N/D 2008)

Designation of plain carbon steels

These are denoted like: x.C.y

Where x – Number 100 times the average percent of carbon, y – Number 100 times the average percent of Manganese.

Designation of Alloy steels

Alloy steels are denoted by arranging the alloying elements in the descending order of their proportion. And, the average % of each element is shown with its chemical symbol before that number. The letter C is omitted here, and just the number is written to denote carbon percentage. When an alloying element is less than 1%, that element is denoted by an underline, after written up to two decimals places.

Eg.: 40 Cr 14

18. What is an impact load? Give examples.

If the time load application is less than one third of the lowest national period of vibration of the part, the load is called an impact load.

Examples: Punching presses, Hammers, loads exerted on cams during the motion due to eccentricity, loads imposed on gear teeth due to irregular tooth profile.

19. For ductile material, which of the strength is considered for designing a

(a) component subjected to static loading

(b) component subjected to fatigue loading (A/M 2008)

For component subjected to static loading, Yield strength is considered.

For component subjected to fatigue loading Endurance strength is considered.

20. Define Principal plane, principal stress.

A plane where only normal stresses act, with no shear stress acting is called principal plane. The (normal) stress acting in this plane is called principal stress.

21. Give examples for curved beams.

Frames of nibbling machine, riveting machine, boring machine and punching machine, crane hook.

22. State the difference between straight beams and curved beams. (N/D 2012)

In straight beams, the neutral axis of the section coincides with its centroidal axis and the stress distribution in the beam is linear. But in case of curved beams, the neutral axis of the cross-section is shifted towards the centre of curvature of the beam causing a non-linear (hyperbolic) distribution of stress.

23. Where will be the maximum stress developed in a curved beam.

In a symmetrical section (circle, rectangle, I section with equal flanges) maximum bending stress occurs at the inner fibre. If the section is not symmetrical, maximum bending stress may occur at either inner or outer fiber.

24. Differentiate the stress distribution in a bar subjected to axial force and beam subjected to bending. (A/M 2008)

In axial case, the stress is uniform across the section. In bending, the stress is tensile on one side of the neutral axis and compressive on the other.

25. Define 'Factor of Safety'.

The ratio between maximum stresses to working stress is known as factor of safety.

Factor of safety = Maximum stress / Working stress

26. How is factor of safety is defines for brittle and ductile materials?

For ductile material, Factor of safety = Yield stress / Working stress

For ductile material, Factor of safety = Ultimate stress / Working stress

27. A flat bar 32 mm wide and 12 mm thick is loaded by a steady tensile load of 85kN. The material is mild steel with yield point stress of 315 N/mm^2. Find the factor of safety based on the yield point.

Stress induced = Load/Area

$= 85 \times 10^3 / (32 \times 12) = 221.34$ MPa

Factor of safety = Yield stress / Stress induced

= 315/221.34 = 1.42

28. List the important factors that influence the magnitude of factor safety. **(N/D 2011)**

1. The reliability of the properties of the material and change of these properties during service.
2. The reliability of test results and accuracy of application of these results to actual machine parts
3. The reliability of applied load
4. The certainty as to exact mode of failure
5. The extent of simplifying assumptions
6. The extent of localised stresses
7. The extent of initial stresses set up during manufacture
8. The extent of loss of life if failure occurs and
9. The extent of loss of property if failure occurs.

29. Why normal stress theory is not suitable for ductile materials?

Ductile materials are mostly undergone by shearing. But this theory considers only tensile or compressive stresses. So, this is not suitable for ductile materials.

30. State Rankine's theory. **(N/D 2007)**

Failure occurs when the maximum normal stress is equal to the tensile yield strength.

31. State St. Venant theory of failure. **(N/D 2006)**

According to this theory, failure occurs when the maximum strain in the member equals the tensile yield strain.

$\sigma_1 - \gamma\ (\sigma_2 + \sigma_3)$ (or) $\sigma_2 - \gamma\ (\sigma_3 + \sigma_1)$ (or) $\sigma_3 - \gamma\ (\sigma_1 + \sigma_2) = \sigma_y$, whichever is maximum, where γ = Poisson's ratio.

32. Define the term equivalent torque and equivalent moment. **(N/D 2006)**

Equivalent torque, $M_e = \sqrt{M_b^2 + M_t^2}$

Equivalent moment, $M_{be} = \frac{1}{2}\left(M_b + \sqrt{M_b^2 + M_t^2}\right)$

33. What are the methods to reduce stress concentration? **(N/D 2008)**

Avoiding sharp corners, Providing fillets, Use of multiple holes instead of single hole, Undercutting the shoulder parts.

34. Differentiate between static and variable stresses.

Static stress does not change in magnitude and in direction. Variable stress changes in magnitude or direction or both.

35. What are the types of variable stresses?

i. Completely reversed or cyclic stresses
ii. Fluctuating stresses
iii. Repeated stresses
iv. Alternating stresses

36. What is 'Adaptive design'? Where is it used? Give examples. (N/D 2012)

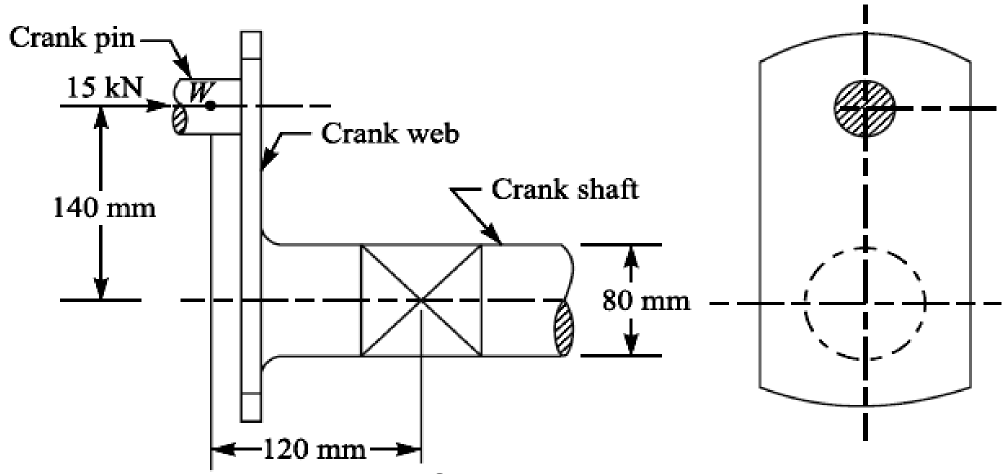
In most cases, the designer's work is concerned with adaptation of existing designs. This type of design needs no special knowledge or skill and can be attempted by designers of ordinary technical training. The designer only makes minor alternation or modification in the existing designs of the product.

37. What is an S-N Curve?

An S-N curve has fatigue stress on Y axis and number if loading cycles in X axis. It is used to find the fatigue stress value corresponding to a given number of cycles.

PART – B

1. **An overhang crank with pin and shaft is shown in Fig. A tangential load of 15 kN acts on the crank pin. Determine the maximum principal stress and the maximum shear stress at the centre of the crankshaft bearing.** **(A/M - 010)**



Solution. Given : $W = 15$ kN $= 15 \times 10^3$ N ; $d = 80$ mm ; $y = 140$ mm ; $x = 120$ mm

Bending moment at the centre of the crankshaft bearing,

$$M = W \times x = 15 \times 10^3 \times 120 = 1.8 \times 10^6 \text{ N-mm}$$

and torque transmitted at the axis of the shaft,

$$T = W \times y = 15 \times 10^3 \times 140 = 2.1 \times 10^6 \text{ N-mm}$$

We know that bending stress due to the bending moment,

$$\sigma_b = \frac{M}{Z} = \frac{32\,M}{\pi\,d^3} \qquad \ldots\left(\because Z = \frac{\pi}{32} \times d^3\right)$$

$$= \frac{32 \times 1.8 \times 10^6}{\pi\,(80)^3} = 35.8 \text{ N/mm}^2 = 35.8 \text{ MPa}$$

and shear stress due to the torque transmitted,

$$\tau = \frac{16\,T}{\pi\,d^3} = \frac{16 \times 2.1 \times 10^6}{\pi\,(80)^3} = 20.9 \text{ N/mm}^2 = 20.9 \text{ MPa}$$

Maximum principal stress

We know that maximum principal stress,

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

$$= \frac{35.8}{2} + \frac{1}{2}\left[\sqrt{(35.8)^2 + 4\,(20.9)^2}\right] \qquad \ldots (\text{Substituting } \sigma_t = \sigma_b)$$

$$= 17.9 + 27.5 = 45.4 \text{ MPa } \textbf{Ans.}$$

Maximum shear stress

We know that maximum shear stress,

$$\tau_{max} = \tfrac{1}{2}\left[\sqrt{(\sigma_t)^2 + 4\,\tau^2}\right] = \tfrac{1}{2}\left[\sqrt{(35.8)^2 + 4\,(20.9)^2}\right]$$

$$= 27.5 \text{ MPa } \textbf{Ans.}$$

2. A rectangular strut is 150 mm wide and 120 mm thick. It carries a load of 180 kN at aneccentricity of 10 mm in a plane bisecting the thickness as shown in Fig. Find the maximum and minimum intensities of stress in the section.

Solution. Given : b = 150 mm ; d = 120 mm ; P = 180 kN $= 180 \times 10^3$ N ; e = 10 mm

We know that cross-sectional area of the strut,

$$A = b.d = 150 \times 120 = 18 \times 10^3 \text{ mm}^2$$

∴ Direct compressive stress,

$$\sigma_o = \frac{P}{A} = \frac{180 \times 10^3}{18 \times 10^3} = 10 \text{ N/mm}^2 = 10 \text{ MPa}$$

Section modulus for the strut,

$$Z = \frac{I_{YY}}{y} = \frac{d\,.\,b^3/12}{b/2} = \frac{d\,.\,b^2}{6} = \frac{120\,(150)^2}{6} = 450 \times 10^3 \text{ mm}^3$$

Bending moment, $M = P.e = 180 \times 10^3 \times 10 = 1.8 \times 10^6$ N-mm

∴ Bending stress, $\sigma_b = \dfrac{M}{Z} = \dfrac{1.8 \times 10^6}{450 \times 10^3} = 4 \text{ N/mm}^2 = 4 \text{ MPa}$

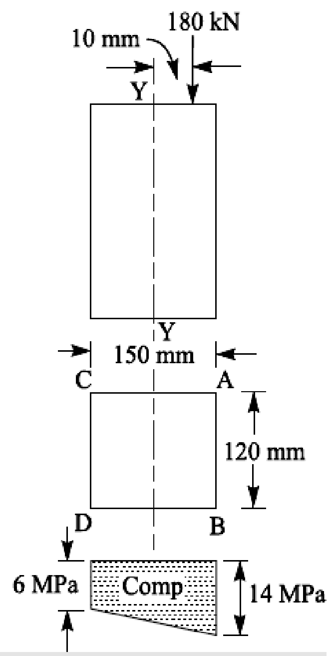


Fig. 5.21

Since σ_o is greater than σ_b, therefore the entire cross-section of the strut will be subjected to compressive stress. The maximum intensity of compressive stress will be at the edge AB and minimum at the edge CD.

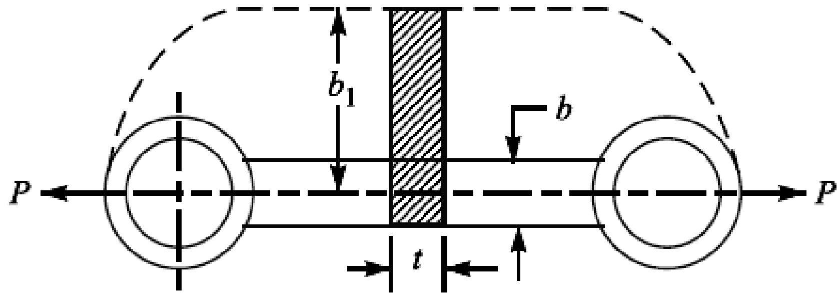
∴ Maximum intensity of compressive stress at the edge AB

$$= \sigma_o + \sigma_b = 10 + 4 = 14 \text{ MPa} \quad \textbf{Ans.}$$

and minimum intensity of compressive stress at the edge CD

$$= \sigma_o - \sigma_b = 10 - 4 = 6 \text{ MPa} \quad \textbf{Ans.}$$

3. A mild steel link, as shown in Fig. by full lines, transmits a pull of 80 kN. Find the dimensions b and t if b = 3t. Assume the permissible tensile stress as 70 MPa. If the original link is replaced by an unsymmetrical one, as shown by dotted lines in Fig., having the same thickness t, find the depth b1, using the same permissible stress as before.



Solution. Given : $P = 80$ kN $= 80 \times 10^3$ N ; $\sigma_t = 70$ MPa $= 70$ N/mm^2

When the link is in the position shown by full lines in Fig. 5.24, the area of cross-section,

$$A = b \times t = 3\,t \times t = 3\,t^2 \qquad ...(\because\ b = 3\,t)$$

We know that tensile load (P),

$$80 \times 10^3 = \sigma_t \times A = 70 \times 3\,t^2 = 210\,t^2$$

$\therefore$ $t^2 = 80 \times 10^3 / 210 = 381$ or $t = 19.5$ say 20 mm **Ans.**

and $b = 3\,t = 3 \times 20 = 60$ mm **Ans.**

When the link is in the position shown by dotted lines, it will be subjected to direct stress as well as bending stress. We know that area of cross-section,

$$A_1 = b_1 \times t$$

$\therefore$ Direct tensile stress,

$$\sigma_o = \frac{P}{A} = \frac{P}{b_1 \times t}$$

and bending stress,

$$\sigma_b = \frac{M}{Z} = \frac{P\,.\,e}{Z} = \frac{6\,P\,.\,e}{t\,(b_1)^2} \qquad ...\left(\because\ Z = \frac{t\,(b_1)^2}{6}\right)$$

$\therefore$ Total stress due to eccentric loading

$$= \sigma_b + \sigma_o = \frac{6\,P\,.\,e}{t\,(b_1)^2} + \frac{P}{b_1 \times t} = \frac{P}{t\,.\,b_1}\left(\frac{6\,e}{b_1} + 1\right)$$

Since the permissible tensile stress is the same as 70 N/mm^2, therefore

$$70 = \frac{80 \times 10^3}{20\,b_1}\left(\frac{6 \times b_1}{b_1 \times 2} + 1\right) = \frac{16 \times 10^3}{b_1} \qquad ...\left(\because\ \text{Eccentricity, } e = \frac{b_1}{2}\right)$$

$\therefore$ $b_1 = 16 \times 10^3 / 70 = 228.6$ say 230 mm **Ans.**

4. A mild steel bracket as shown in Fig., is subjected to a pull of 6000 N acting at 45° to its horizontal axis. The bracket has a rectangular section whose depth is twice the thickness. Find the cross-sectional dimensions of the bracket, if the permissible stress in the material of the bracket is limited to 60 MPa. **(N/D - 07)**

Solution. Given : $P = 6000$ N ; $\theta = 45°$; $\sigma = 60$ MPa $= 60$ N/mm^2

Let t = Thickness of the section in mm, and

b = Depth or width of the section $= 2\,t$ ...(Given)

We know that area of cross-section,

$$A = b \times t = 2\,t \times t = 2\,t^2 \text{ mm}^2$$

and section modulus,

$$Z = \frac{t \times b^2}{6} = \frac{t\,(2t)^2}{6} = \frac{4\,t^3}{6} \text{ mm}^3$$

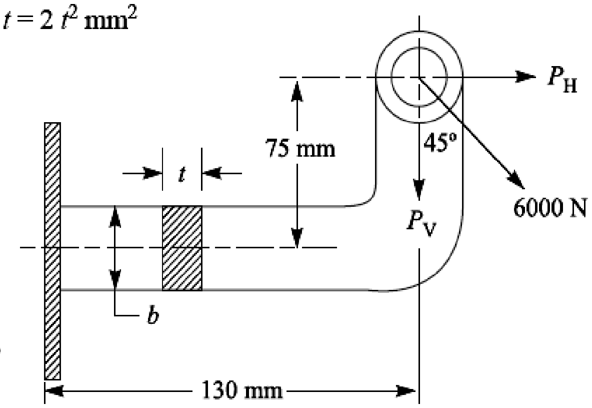


Fig. 5.28

Horizontal component of the load,

$$P_H = 6000 \cos 45° = 6000 \times 0.707 = 4242 \text{ N}$$

∴ Bending moment due to horizontal component of the load,

$$M_H = P_H \times 75 = 4242 \times 75 = 318\,150 \text{ N-mm}$$

A little consideration will show that the bending moment due to the horizontal component of the load induces tensile stress on the upper surface of the bracket and compressive stress on the lower surface of the bracket. ∴Maximum bending stress on the upper surface due to horizontal component,

blades

$$\sigma_{bH} = \frac{M_H}{Z} = \frac{318\,150 \times 6}{4\,t^3}$$

Note : Thi
and is nc

$$= \frac{477\,225}{t^3} \text{ N/mm}^2 \text{ (tensile)}$$

Vertical component of the load,

$$P_V = 6000 \sin 45° = 6000 \times 0.707 = 4242 \text{ N}$$

∴ Direct stress due to vertical component,

$$\sigma_{oV} = \frac{P_V}{A} = \frac{4242}{2t^2} = \frac{2121}{t^2} \text{ N/mm}^2 \text{ (tensile)}$$

Bending moment due to vertical component of the load,

$$M_V = P_V \times 130 = 4242 \times 130 = 551\,460 \text{ N-mm}$$

This bending moment induces tensile stress on the upper surface and compressive stress on the lower surface of the bracket.

∴ Maximum bending stress on the upper surface due to vertical component,

$$\sigma_{bV} = \frac{M_V}{Z} = \frac{551\,460 \times 6}{4\,t^3} = \frac{827\,190}{t^3} \text{ N/mm}^2 \text{ (tensile)}$$

and total tensile stress on the upper surface of the bracket,

$$\sigma = \frac{477\,225}{t^3} + \frac{2121}{t^2} + \frac{827\,190}{t^3} = \frac{1\,304\,415}{t^3} + \frac{2121}{t^2}$$

Since the permissible stress (σ) is 60 N/mm^2, therefore

$$\frac{1\,304\,415}{t^3} + \frac{2121}{t^2} = 60 \quad \text{or} \quad \frac{21\,740}{t^3} + \frac{35.4}{t^2} = 1$$

∴ $t = 28.4$ mm **Ans.** ... (By hit and trial)

and $b = 2t = 2 \times 28.4 = 56.8$ mm **Ans.**

5. The frame of a punch press is shown in Fig. Find the stresses at the inner and outer surface at section X-X of the frame, if W = 5000 N. (N/D - 05)

Solution. Given : $W = 5000$ N ; $b_i = 18$ mm ; $b_o = 6$ mm ; $h = 40$ mm ; $R_i = 25$ mm ; $R_o = 25 + 40 = 65$ mm

We know that area of section at X-X,

$$A = \frac{1}{2}(18 + 6)\,40 = 480 \text{ mm}^2$$

The various distances are shown in Fig. 5.10.

We know that radius of curvature of the neutral axis,

$$R_n = \frac{\left(\frac{b_i + b_o}{2}\right) h}{\left(\frac{b_i R_o - b_o R_i}{h}\right) \log_e \left(\frac{R_o}{R_i}\right) - (b_i - b_o)}$$

$$= \frac{\left(\frac{18 + 6}{2}\right) \times 40}{\left(\frac{18 \times 65 - 6 \times 25}{40}\right) \log_e \left(\frac{65}{25}\right) - (18 - 6)}$$

$$= \frac{480}{(25.5 \times 0.9555) - 12} = 38.83 \text{ mm}$$

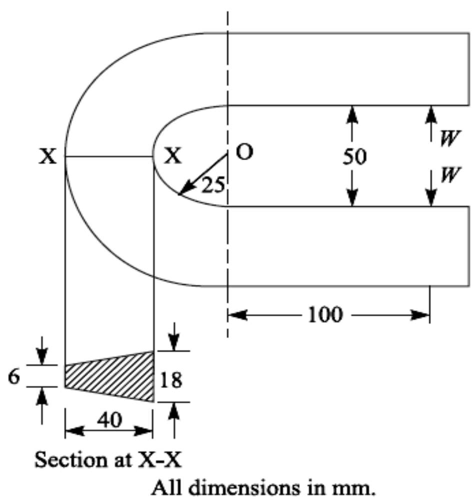


Fig. 5.9

and radius of curvature of the centroidal axis,

$$R = R_i + \frac{h\,(b_i + 2b_o)}{3\,(b_i + b_o)} = 25 + \frac{40\,(18 + 2 \times 6)}{3\,(18 + 6)} \text{ mm}$$

$$= 25 + 16.67 = 41.67 \text{ mm}$$

Distance between the centroidal axis and neutral axis,

$$e = R - R_n = 41.67 - 38.83 = 2.84 \text{ mm}$$

and the distance between the load and centroidal axis,

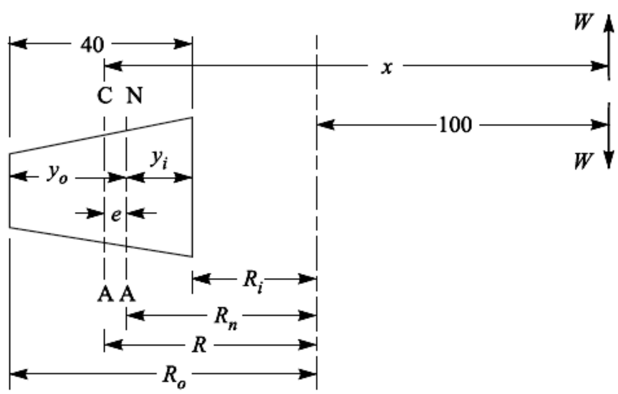
$$x = 100 + R = 100 + 41.67 = 141.67 \text{ mm}$$

∴ Bending moment about the centroidal axis,

$$M = W.x = 5000 \times 141.67 = 708\,350 \text{ N-mm}$$

The section at X-X is subjected to a direct tensile load of W = 5000 N and a bending moment of M = 708 350 N-mm. We know that direct tensile stress at section X-X,

$$\sigma_t = \frac{W}{A} = \frac{5000}{480} = 10.42 \text{ N/mm}^2 = 10.42 \text{ MPa}$$



All dimensions in mm.

Distance from the neutral axis to the inner surface,

$$y_i = R_n - R_i = 38.83 - 25 = 13.83 \text{ mm}$$

Distance from the neutral axis to the outer surface,

$$y_o = R_o - R_n = 65 - 38.83 = 26.17 \text{ mm}$$

We know that maximum bending stress at the inner surface,

$$\sigma_{bi} = \frac{M\,.\,y_i}{A\,.\,e\,.\,R_i} = \frac{708\,350 \times 13.83}{480 \times 2.84 \times 25} = 287.4 \text{ N/mm}^2$$

$$= 287.4 \text{ MPa (tensile)}$$

and maximum bending stress at the outer surface,

$$\sigma_{b0} = \frac{M\,.\,y_o}{A\,.\,e\,.\,R_o} = \frac{708\,350 \times 26.17}{480 \times 2.84 \times 65} = 209.2 \text{ N/mm}^2$$

$$= 209.2 \text{ MPa (compressive)}$$

$\therefore$ Resultant stress on the inner surface

$$= \sigma_t + \sigma_{bi} = 10.42 + 287.4 = 297.82 \text{ MPa (tensile)} \textbf{ Ans.}$$

and resultant stress on the outer surface,

$$= \sigma_t - \sigma_{bo} = 10.42 - 209.2 = -198.78 \text{ MPa}$$

$$= 198.78 \text{ MPa (compressive)} \textbf{ Ans.}$$

6. The crane hook carries a load of 20 kN as shown in Fig. The section at X-X is rectangular whose horizontal side is 100 mm. Find the stresses in the inner and outer fibres at the given section. **(N/D - 06)**

Solution. Given : $W = 20 \text{ kN} = 20 \times 10^3$ N ; $R_i = 50$ mm ; $R_o = 150$ mm ; $h = 100$ mm ; $b = 20$ mm

We know that area of section at *X-X*,

$$A = b.h = 20 \times 100 = 2000 \text{ mm}^2$$

The various distances are shown in Fig. 5.12.

We know that radius of curvature of the neutral axis,

$$R_n = \frac{h}{\log_e\left(\frac{R_o}{R_i}\right)} = \frac{100}{\log_e\left(\frac{150}{50}\right)} = \frac{100}{1.098} = 91.07 \text{ mm}$$

and radius of curvature of the centroidal axis,

$$R = R_i + \frac{h}{2} = 50 + \frac{100}{2} = 100 \text{ mm}$$

$\therefore$ Distance between the centroidal axis and neutral axis,

$$e = R - R_n = 100 - 91.07 = 8.93 \text{ mm}$$

and distance between the load and the centroidal axis,

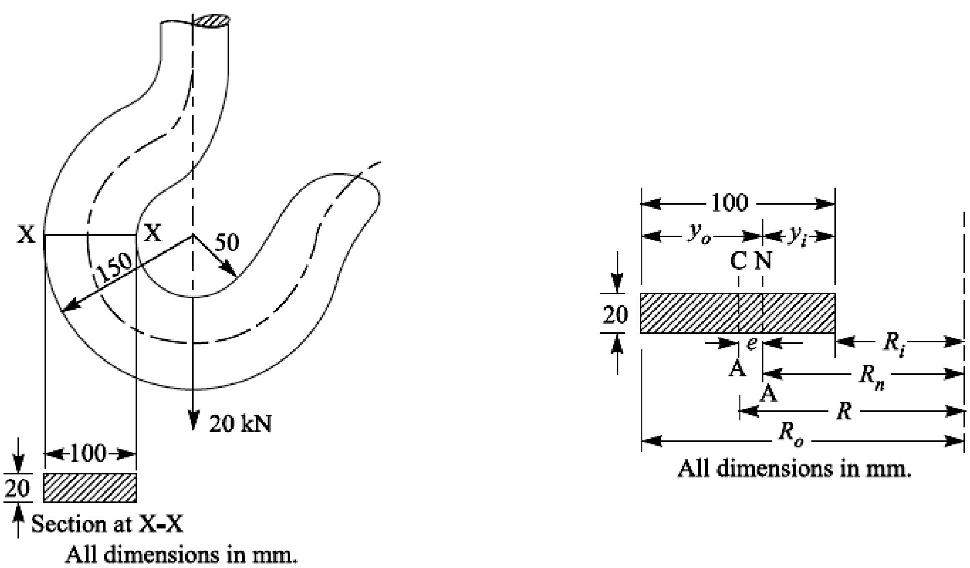
$$x = R = 100 \text{ mm}$$

$\therefore$ Bending moment about the centroidal axis,

$$M = W \times x = 20 \times 10^3 \times 100 = 2 \times 10^6 \text{ N-mm}$$

The section at *X-X* is subjected to a direct tensile load of $W = 20 \times 10^3$ N and a bending moment of $M = 2 \times 10^6$ N-mm. We know that direct tensile stress at section *X-X*,

$$\sigma_t = \frac{W}{A} = \frac{20 \times 10^3}{2000} = 10 \text{ N/mm}^2 = 10 \text{ MPa}$$



We know that the distance from the neutral axis to the inside fibre,

$$y_i = R_n - R_i = 91.07 - 50 = 41.07 \text{ mm}$$

and distance from the neutral axis to outside fibre,

$$y_o = R_o - R_n = 150 - 91.07 = 58.93 \text{ mm}$$

∴ Maximum bending stress at the inside fibre,

$$\sigma_{bi} = \frac{M \cdot y_i}{A \cdot e \cdot R_i} = \frac{2 \times 10^6 \times 41.07}{2000 \times 8.93 \times 50} = 92 \text{ N/mm}^2 = 92 \text{ MPa (tensile)}$$

and maximum bending stress at the outside fibre,

$$\sigma_{bo} = \frac{M \cdot y_o}{A \cdot e \cdot R_o} = \frac{2 \times 10^6 \times 58.93}{2000 \times 8.93 \times 150} = 44 \text{ N/mm}^2$$

$$= 44 \text{ MPa (compressive)}$$

∴ Resultant stress at the inside fibre

$$= \sigma_t + \sigma_{bi} = 10 + 92 = 102 \text{ MPa (tensile)} \textbf{ Ans.}$$

and resultant stress at the outside fibre

$$= \sigma_t - \sigma_{bo} = 10 - 44 = -34 \text{ MPa} = 34 \text{ MPa (compressive)} \textbf{ Ans.}$$

7. A cast iron pulley transmits 10 kW at 400 rpm. The diameter of the pulley is 1.2 m and it has four straight arms of elliptical cross-section, in which the major axis is twice the minor axis. Determine the dimension of the arm if the allowable bending stress is 15 MPa

(8) (M/J - 11)

Solution. Given : $P = 10$ kW $= 10 \times 10^3$ W ; $N = 400$ r.p.m ; $D = 1.2$ m $= 1200$ mm or $R = 600$ mm ; $\sigma_b = 15$ MPa $= 15$ N/mm^2

Let T = Torque transmitted by the pulley.

We know that the power transmitted by the pulley (P),

$$10 \times 10^3 = \frac{2\pi N \cdot T}{60} = \frac{2\pi \times 400 \times T}{60} = 42\, T$$

$$\therefore \quad T = 10 \times 10^3/42 = 238 \text{ N-m} = 238 \times 10^3 \text{ N-mm}$$

Since the torque transmitted is the product of the tangential load and the radius of the pulley, therefore tangential load acting on the pulley

$$= \frac{T}{R} = \frac{238 \times 10^3}{600} = 396.7 \text{ N}$$

Since the pulley has four arms, therefore tangential load on each arm,

$$W = 396.7/4 = 99.2 \text{ N}$$

and maximum bending moment on the arm,

$$M = W \times R = 99.2 \times 600 = 59\,520 \text{ N-mm}$$

Let $2b$ = Minor axis in mm, and

$2a$ = Major axis in mm $= 2 \times 2b = 4b$ ...(Given)

∴ Section modulus for an elliptical cross-section,

$$Z = \frac{\pi}{4} \times a^2 b = \frac{\pi}{4} (2b)^2 \times b = \pi b^3 \text{ mm}^3$$

We know that bending stress (σ_b),

$$15 = \frac{M}{Z} = \frac{59\,520}{\pi b^3} = \frac{18\,943}{b^3}$$

or $b^3 = 18\,943/15 = 1263$ or $b = 10.8$ mm

∴ Minor axis, $2b = 2 \times 10.8 = 21.6$ mm **Ans.**

and major axis, $2a = 2 \times 2b = 4 \times 10.8 = 43.2$ mm **Ans.**

8. A circular bar of 500 mm length is supported freely at its two ends. It is acted upon by a central concentrated cyclic load having a minimum value of 20 kN and a maximum value of 50 kN. Determine the diameter of bar by taking a factor of safety of 1.5, size effect of 0.85 and surface finish factor of 0.9. The material properties of the bar is given by: ultimate strength of 650 MPa, yield strength of 500 MPa and endurance strength of 350 MPa.

(16) (M/J - 11)

Solution. Given : $l = 500$ mm ; $W_{min} = 20$ kN $= 20 \times 10^3$ N ; $W_{max} = 50$ kN $= 50 \times 10^3$ N ; $F.S. = 1.5$; $K_{sz} = 0.85$; $K_{sur} = 0.9$; $\sigma_u = 650$ MPa $= 650$ N/mm^2 ; $\sigma_y = 500$ MPa $= 500$ N/mm^2 ; $\sigma_e = 350$ MPa $= 350$ N/mm^2

Let d = Diameter of the bar in mm.

We know that the maximum bending moment,

$$M_{max} = \frac{W_{max} \times l}{4} = \frac{50 \times 10^3 \times 500}{4} = 6250 \times 10^3 \text{ N-mm}$$

and minimum bending moment,

$$M_{min} = \frac{W_{min} \times l}{4} = \frac{20 \times 10^3 \times 500}{4} = 2550 \times 10^3 \text{ N-mm}$$

∴ Mean or average bending moment,

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{6250 \times 10^3 + 2500 \times 10^3}{2} = 4375 \times 10^3 \text{ N-mm}$$

and variable bending moment,

$$M_v = \frac{M_{max} - M_{min}}{2} = \frac{6250 \times 10^3 - 2500 \times 10^3}{2} = 1875 \times 10^3 \text{ N-mm}$$

Section modulus of the bar,

$$Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3 \text{ mm}^3$$

∴ Mean or average bending stress,

$$\sigma_m = \frac{M_m}{Z} = \frac{4375 \times 10^3}{0.0982\, d^3} = \frac{44.5 \times 10^6}{d^3} \text{ N/mm}^2$$

and variable bending stress,

$$\sigma_v = \frac{M_v}{Z} = \frac{1875 \times 10^3}{0.0982\, d^3} = \frac{19.1 \times 10^6}{d^3} \text{ N/mm}^2$$

We know that according to Goodman's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times K_f}{\sigma_e \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 650} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \quad \text{...(Taking } K_f = 1)$$

$$= \frac{68 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{139 \times 10^3}{d^3}$$

∴ $d^3 = 139 \times 10^3 \times 1.5 = 209 \times 10^3$ or $d = 59.3$ mm

and according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_e \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 500} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \quad \text{...(Taking } K_f = 1)$$

$$= \frac{89 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{160 \times 10^3}{d^3}$$

∴ $d^3 = 160 \times 10^3 \times 1.5 = 240 \times 10^3$ or $d = 62.1$ mm

Taking larger of the two values, we have $d = 62.1$ mm **Ans.**

9. 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.

(16) (A/M – 05) (N/D - 12)

Solution. Given : $\sigma_{t(max)}$ = 140 MPa = 140 N/mm^2 ; R_i = 25 mm ; R_o = 25 + 25 = 50 mm ; b_i = 19 mm ; t_i = 3 mm ; t = 3 mm ; h = 25 mm

We know that area of section at *X-X*,

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

The various distances are shown in Fig. 5.14. We know that radius of curvature of the neutral axis,

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

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

$$= \frac{123}{16 \times 0.113 + 3 \times 0.693} = \frac{123}{3.887} = 31.64 \text{ mm}$$

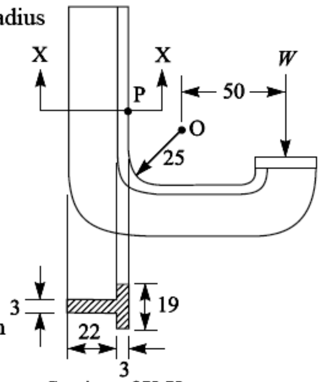


Fig. 5.13

and radius of curvature of the centroidal axis,

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

$$= 25 + \frac{\frac{1}{2} \times 25^2 \times 3 + \frac{1}{2} \times 3^2\,(19-3)}{25 \times 3 + 3\,(19-3)} = 25 + \frac{937.5 + 72}{75 + 48}$$

$$= 25 + 8.2 = 33.2 \text{ mm}$$

Distance between the centroidal axis and neutral axis,

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

and distance between the load *W* and the centroidal axis,

$$x = 50 + R = 50 + 33.2 = 83.2 \text{ mm}$$

∴ Bending moment about the centroidal axis,

$$M = W.x = W \times 83.2 = 83.2\,W \text{ N-mm}$$

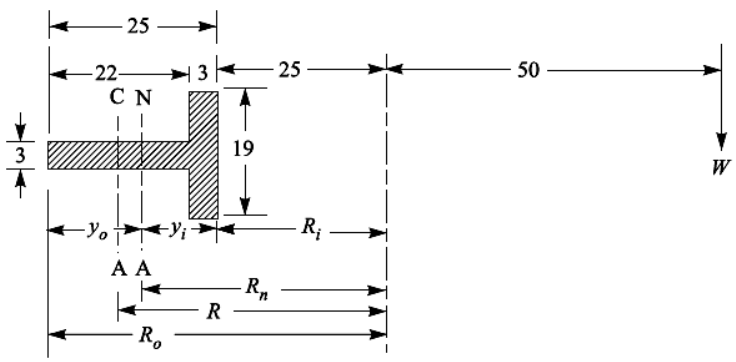


Fig. 5.14

The section at *X-X* is subjected to a direct tensile load of *W* and a bending moment of 83.2 *W*. The maximum tensile stress will occur at point *P* (*i.e.* at the inner fibre of the section).

Distance from the neutral axis to the point *P*,

$$y_i = R_n - R_i = 31.64 - 25 = 6.64 \text{ mm}$$

Direct tensile stress at section X-X,

$$\sigma_t = \frac{W}{A} = \frac{W}{123} = 0.008\ W\ \text{N/mm}^2$$

and maximum bending stress at point P,

$$\sigma_{bi} = \frac{M\ .\ y_i}{A\ .\ e\ .\ R_i} = \frac{83.2\ W \times 6.64}{123 \times 1.56 \times 25} = 0.115\ W\ \text{N/mm}^2$$

We know that the maximum tensile stress $\sigma_{t(max)}$,

$$140 = \sigma_t + \sigma_{bi} = 0.008\ W + 0.115\ W = 0.123\ W$$

$\therefore$ $W = 140/0.123 = 1138$ N **Ans.**

Note : We know that distance from the neutral axis to the outer fibre,

$$y_o = R_o - R_n = 50 - 31.64 = 18.36 \text{ mm}$$

$\therefore$ Maximum bending stress at the outer fibre,

$$\sigma_{bo} = \frac{M\ .\ y_o}{A\ .\ e\ .\ R_o} = \frac{83.2\ W \times 18.36}{123 \times 1.56 \times 50} = 0.16\ W$$

and maximum stress at the outer fibre,

$$= \sigma_t - \sigma_{bo} = 0.008\ W - 0.16\ W = -0.152\ W \text{N/mm}^2$$

$$= 0.152\ W \text{N/mm}^2 \text{ (compressive)}$$

From above we see that stress at the outer fibre is larger in this case than at the inner fibre, but this stress at outer fibre is compressive.

10. A rectangular plate 60 mm × 10 mm with a hole 12 diameter as shown in Fig. 6.13 and subjected to a tensile load of 12 kN. (A/M - 96) (N/D - 08)

Given: b = 60 mm ; t = 10 mm ; d = 12 mm ; W = 12 kN = 12 × 103 N We know that cross-sectional area of the plate,

$$A = (b - d)\ t = (60 - 12)\ 10 = 480 \text{ mm}^2$$

$\therefore$ Nominal stress $= \frac{W}{A} = \frac{12 \times 10^3}{480} = 25 \text{ N/mm}^2 = 25$ MPa

Ratio of diameter of hole to width of plate,

$$\frac{d}{b} = \frac{12}{60} = 0.2$$

From Table 6.1, we find that for $d/b = 0.2$, theoretical stress concentration factor,

$$K_t = 2.5$$

$\therefore$ Maximum stress $= K_t \times$ Nominal stress $= 2.5 \times 25 = 62.5$ MPa **Ans.**

11. A cast iron pulley transmits 10 kW at 400 rpm. The diameter of the pulley is 1.2 metre and it has four straight arms of elliptical cross-section, in which the major axis is twice the minor axis. Determine the dimensions of the arm if the allowable bending stress is 15 MPa. **(A/M - 08)**

Solution. Given : $P = 10$ kW $= 10 \times 10^3$ W ; $N = 400$ r.p.m ; $D = 1.2$ m $= 1200$ mm or $R = 600$ mm ; $\sigma_b = 15$ MPa $= 15$ N/mm^2

Let T = Torque transmitted by the pulley.

We know that the power transmitted by the pulley (P),

$$10 \times 10^3 = \frac{2\,\pi\,N\,.\,T}{60} = \frac{2\pi \times 400 \times T}{60} = 42\,T$$

$$\therefore \quad T = 10 \times 10^3/42 = 238 \text{ N-m} = 238 \times 10^3 \text{ N-mm}$$

Since the torque transmitted is the product of the tangential load and the radius of the pulley, therefore tangential load acting on the pulley

$$= \frac{T}{R} = \frac{238 \times 10^3}{600} = 396.7 \text{ N}$$

Since the pulley has four arms, therefore tangential load on each arm,

$$W = 396.7/4 = 99.2 \text{ N}$$

and maximum bending moment on the arm,

$$M = W \times R = 99.2 \times 600 = 59\,520 \text{ N-mm}$$

Let $2b$ = Minor axis in mm, and

$2a$ = Major axis in mm $= 2 \times 2b = 4b$ ...(Given)

$\therefore$ Section modulus for an elliptical cross-section,

$$Z = \frac{\pi}{4} \times a^2 b = \frac{\pi}{4}\,(2b)^2 \times b = \pi\,b^3 \text{ mm}^3$$

We know that bending stress (σ_b),

$$15 = \frac{M}{Z} = \frac{59\,520}{\pi\,b^3} = \frac{18\,943}{b^3}$$

or $b^3 = 18\,943/15 = 1263$ or $b = 10.8$ mm

$\therefore$ Minor axis, $2b = 2 \times 10.8 = 21.6$ mm **Ans.**

and major axis, $2a = 2 \times 2b = 4 \times 10.8 = 43.2$ mm **Ans.**

12. A hot rolled steel shaft is subjected to a torsional moment that varies from 330 Nm clockwise to 110 Nm counterclockwise and an applied bending moment at a critical section varies from 440 Nm to – 220 Nm. 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/m^2 and yield strength of 410 MN/m^2. 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. **(16) (N/D - 08)**

Solution. Given : T_{max} = 330 N-m (clockwise) ; T_{min} = 110 N-m (counterclockwise) = – 110 N-m (clockwise) ; M_{max} = 440 N-m ; M_{min} = – 220 N-m ; σ_u = 550 MN/m^2 = 550 × 10^6 N/m^2 ; σ_y = 410 MN/m^2 = 410 × 10^6 N/m^2 ; $\sigma_e = \frac{1}{2}\sigma_u$ = 275 × 10^6 N/m^2 ; *F.S.* = 2 ; K_{sz} = 0.85 ; K_{sur} = 0.62

Let d = Required shaft diameter in metres.

We know that mean torque,

$$T_m = \frac{T_{max} + T_{min}}{2} = \frac{330 + (-110)}{2} = 110 \text{ N-m}$$

and variable torque,

$$T_v = \frac{T_{max} - T_{min}}{2} = \frac{330 - (-110)}{2} = 220 \text{ N-m}$$

∴ Mean shear stress,

$$\tau_m = \frac{16\, T_m}{\pi\, d^3} = \frac{16 \times 110}{\pi d^3} = \frac{560}{d^3} \text{ N/m}^2$$

and variable shear stress,

$$\tau_v = \frac{16\, T_v}{\pi\, d^3} = \frac{16 \times 220}{\pi d^3} = \frac{1120}{d^3} \text{ N/m}^2$$

Since the endurance limit in shear (τ_e) is 0.55 σ_e, and yield strength in shear (τ_y) is 0.5 σ_y, therefore

$$\tau_e = 0.55 \times 275 \times 10^6 = 151.25 \times 10^6 \text{ N/m}^2$$

and

$$\tau_y = 0.5 \times 410 \times 10^6 = 205 \times 10^6 \text{ N/m}^2$$

We know that equivalent shear stress,

$$\tau_{es} = \tau_m + \frac{\tau_v \times \tau_y\ K_{fs}}{\tau_e \times K_{sur} \times K_{sz}}$$

$$= \frac{560}{d^3} + \frac{1120 \times 205 \times 10^6 \times 1}{d^3 \times 151.25 \times 10^6 \times 0.62 \times 0.85} \quad \text{...(Taking } K_{fs} = 1)$$

$$= \frac{560}{d^3} + \frac{2880}{d^3} = \frac{3440}{d^3} \text{ N/m}^2$$

Mean or average bending moment,

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{440 + (-220)}{2} = 110 \text{ N-m}$$

and variable bending moment,

$$M_v = \frac{M_{max} - M_{min}}{2} = \frac{440 - (-220)}{2} = 330 \text{ N-m}$$

Section modulus, $Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3 \text{ m}^3$

∴ Mean bending stress,

$$\sigma_m = \frac{M_m}{Z} = \frac{110}{0.0982\, d^3} = \frac{1120}{d^3} \text{ N/m}^2$$

and variable bending stress,

$$\sigma_v = \frac{M_v}{Z} = \frac{330}{0.0982\, d^3} = \frac{3360}{d^3} \text{ N/m}^2$$

Since there is no reversed axial loading, therefore the equivalent normal stress due to reversed bending load,

$$\sigma_{neb} = \sigma_{ne} = \sigma_m + \frac{\sigma_v \times \sigma_y \times K_{fb}}{\sigma_{eb} \times K_{sur} \times K_{sz}}$$

$$= \frac{1120}{d^3} + \frac{3360 \times 410 \times 10^6 \times 1}{d^3 \times 275 \times 10^6 \times 0.62 \times 0.85}$$

...(Taking $K_{fb} = 1$ and $\sigma_{eb} = \sigma_e$)

$$= \frac{1120}{d^3} + \frac{9506}{d^3} = \frac{10\,626}{d^3} \text{ N/m}^2$$

We know that the maximum equivalent shear stress,

$$\tau_{es(max)} = \frac{\tau_y}{F.S.} = \frac{1}{2}\sqrt{(\sigma_{ne})^2 + 4\,(\tau_{es})^2}$$

$$\frac{205 \times 10^6}{2} = \frac{1}{2}\sqrt{\left(\frac{10\,625}{d^3}\right)^2 + 4\left(\frac{3440}{d^3}\right)^2}$$

$$205 \times 10^6 \times d^3 = \sqrt{113 \times 10^6 + 4 \times 11.84 \times 10^6} = 12.66 \times 10^3$$

$$\therefore \quad d^3 = \frac{12.66 \times 10^3}{205 \times 10^6} = \frac{0.0617}{10^3}$$

or $$d = \frac{0.395}{10} = 0.0395 \text{ m} = 39.5 \text{ say } 40 \text{ mm} \quad \textbf{Ans.}$$

13. Determine the diameter of a circular rod made of ductile material with a fatigue strength (complete stress reversal), σ_e = 265 MPa and a tensile yield strength of 350 MPa. The member is subjected to a varying axial load from W_{min} = – 300 × 103 N to W_{max} = 700 × 103 N and has a stress concentration factor of 1.8. Use factor of safety as 2.0. (8) (A/M - 09)

Solution. Given : $\sigma_e = 265$ MPa $= 265$ N/mm^2 ; $\sigma_y = 350$ MPa $= 350$ N/mm^2 ; $W_{min} = -300 \times 10^3$ N ; $W_{max} = 700 \times 10^3$ N ; $K_f = 1.8$; $F.S. = 2$

Let d = Diameter of the circular rod in mm.

$$\therefore \quad \text{Area, } A = \frac{\pi}{4} \times d^2 = 0.7854\, d^2 \text{ mm}^2$$

We know that the mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{700 \times 10^3 + (-300 \times 10^3)}{2} = 200 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{200 \times 10^3}{0.7854\, d^2} = \frac{254.6 \times 10^3}{d^2} \text{ N/mm}^2$$

$$\text{Variable load, } W_v = \frac{W_{max} - W_{min}}{2} = \frac{700 \times 10^3 - (-300 \times 10^3)}{2} = 500 \times 10^3 \text{ N}$$

$$\therefore \text{ Variable stress, } \sigma_v = \frac{W_v}{A} = \frac{500 \times 10^3}{0.7854\, d^2} = \frac{636.5 \times 10^3}{d^2} \text{ N/mm}^2$$

We know that according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_e}$$

$$\frac{1}{2} = \frac{254.6 \times 10^3}{d^2 \times 350} + \frac{636.5 \times 10^3 \times 1.8}{d^2 \times 265} = \frac{727}{d^2} + \frac{4323}{d^2} = \frac{5050}{d^2}$$

$$\therefore \quad d^2 = 5050 \times 2 = 10\,100 \text{ or } d = 100.5 \text{ mm } \textbf{Ans.}$$

14. Determine the thickness of a 120 mm wide uniform plate for safe continuous operation if the plate is to be subjected to a tensile load that has a maximum value of 250 kN and a minimum value of 100 kN. The properties of the plate material are as follows: Endurance limit stress = 225 MPa, and Yield point stress = 300 MPa. The factor of safety based on yield point may be taken as 1.5. **(N/D - 06)**

Solution. Given : $b = 120$ mm ; $W_{max} = 250$ kN; $W_{min} = 100$ kN ; $\sigma_e = 225$ MPa = 225 N/m
$\sigma_y = 300$ MPa = 300 N/mm^2; $F.S. = 1.5$

Let t = Thickness of the plate in mm.

$\therefore$ Area, $A = b \times t = 120\,t$ mm^2

We know that mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{250 + 100}{2} = 175 \text{ kN} = 175 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{175 \times 10^3}{120\,t} \text{ N/mm}^2$$

$$\text{Variable load, } W_v = \frac{W_{max} - W_{min}}{2} = \frac{250 - 100}{2} = 75 \text{ kN} = 75 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Variable stress, } \sigma_v = \frac{W_v}{A} = \frac{75 \times 10^3}{120\,t} \text{ N/mm}^2$$

According to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e}$$

$$\frac{1}{1.5} = \frac{175 \times 10^3}{120\,t \times 300} + \frac{75 \times 10^3}{120\,t \times 225} = \frac{4.86}{t} + \frac{2.78}{t} = \frac{7.64}{t}$$

$\therefore$ $t = 7.64 \times 1.5 = 11.46$ say 11.5 mm **Ans.**

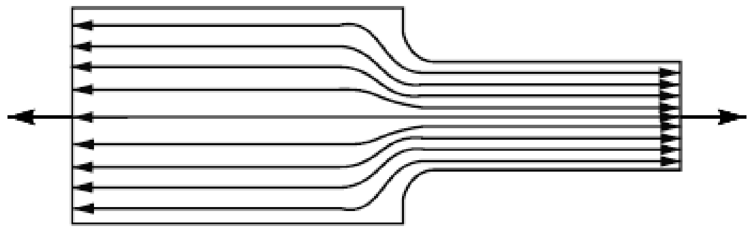
15. Explain Stress concentration. Write some methods to reduce stress concentration. **(M/J - 11)**

Stress concentration

Whenever a machine component changes the shape of its cross-section, the simple stress distribution no longer holds good and the neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called ***stress concentration.***

It occurs for all kinds of stresses in the presence of fillets, notches, holes, keyways, splines, surface roughness or scratches etc. In order to understand fully the idea of stress concentration, consider a member with different cross-section under a tensile load as shown in Fig.

A little consideration will show that the nominal stress in the right and left hand sides will be uniform but in the region where the crosssection is changing, a re-distribution of the force within the member must take place. The material near the edges is stressed considerably higher than the average value. The maximum stress occurs at some point on the fillet and is directed parallel to the boundary at that point.



The theoretical or form stress concentration factor is defined as the ratio of the maximum stress in a member (at a notch or a fillet) to the nominal stress at the same section based upon net area. Mathematically, theoretical or form stress concentration factor,

Kt = Maximum stress / Nominal stress

The value of Kt depends upon the material and geometry of the part.

Methods of reducing stress concentration

Whenever there is a change in cross-section, such as shoulders, holes, notches or keyways and where there is an interference fit between a hub or bearing race and a shaft, then stress concentration results. The presence of stress concentration cannot be totally eliminated but it may be reduced to some extent. A device or concept that is useful in assisting a design engineer to visualize the presence of stress concentration and how it may be mitigated is that of stress flow lines, as shown in Fig 6.8. The mitigation of stress concentration means that the stress flow lines shall maintain their spacing as far as possible.

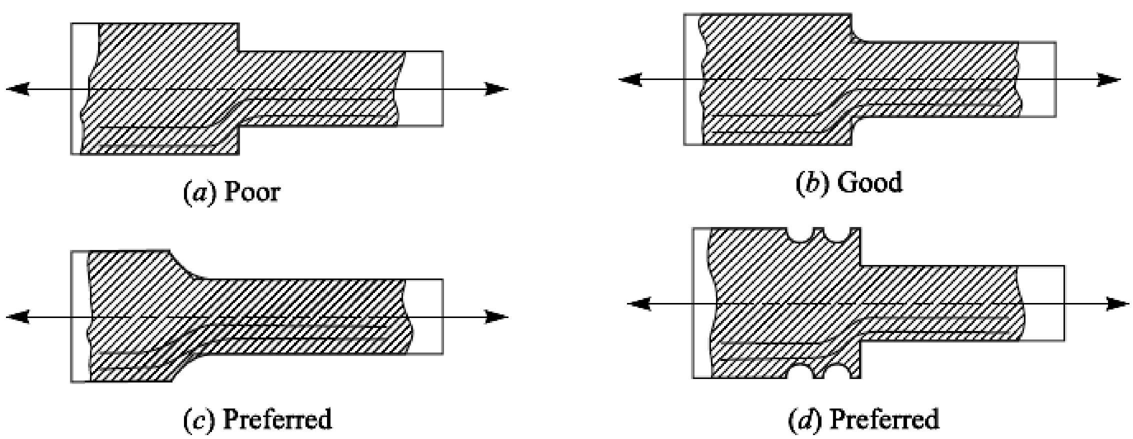


Fig. 6.8

In Fig. 6.8 (*a*) we see that stress lines tend to bunch up and cut very close to the sharp re-entrant corner. In order to improve the situation, fillets may be provided, as shown in Fig. 6.8 (*b*) and (*c*) to give more equally spaced flow lines. Figs. 6.9 to 6.11 show the several ways of reducing the stress concentration in shafts and other cylindrical members with shoulders, holes and threads respectively. It may be noted that it is not practicable to use large radius fillets as in case of ball and roller bearing mountings. In such cases, notches may be cut as shown in Fig. 6.8 (*d*) and Fig. 6.9 (*b*) and (*c*).

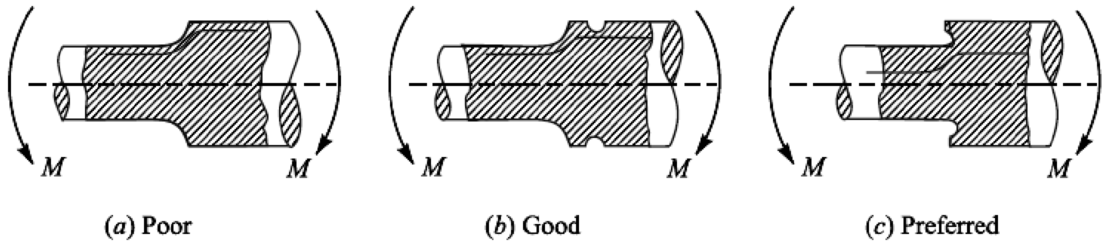


Fig. 6.9. Methods of reducing stress concentration in cylindrical members with shoulders.

Fig. 6.10. Methods of reducing stress concentration in cylindrical members with holes.

The stress concentration effects of a press fit may be reduced by making more gradual transition from the rigid to the more flexible shaft. The various ways of reducing stress concentration for such cases are shown in Fig. 6.12 (*a*), (*b*) and (*c*).

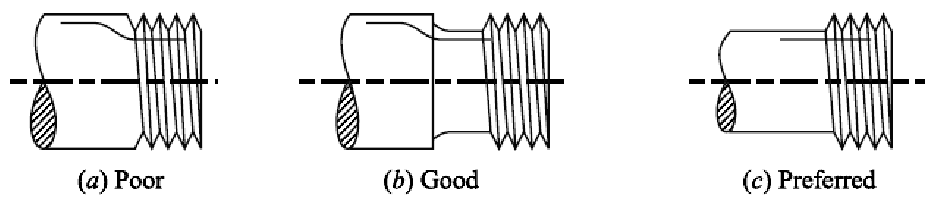


Fig. 6.11. Methods of reducing stress concentration in cylindrical members with holes.

16. Explain in detail, the factors influencing machine design. **(8) (M/J - 12)**

General Considerations in Machine Design

Following are the general considerations in designing a machine component :

1. Type of load and stresses caused by the load. The load, on a machine component, may act in several ways due to which the internal stresses are set up.

2. Motion of the parts or kinematics of the machine. The successful operation of any machine depends largely upon the simplest arrangement of the parts which will give the motion required. The motion of the parts may be :

(a) Rectilinear motion which includes unidirectional and reciprocating motions.
(b) Curvilinear motion which includes rotary, oscillatory and simple harmonic.
(c) Constant velocity.
(d) Constant or variable acceleration.

3. Selection of materials. It is essential that a designer should have a thorough knowledge of the properties of the materials and their behaviour under working conditions. Some of the important characteristics of materials are : strength, durability, flexibility, weight, resistance to heat and corrosion, ability to cast, welded or hardened, machinability, electrical conductivity, etc. The various types of engineering materials and their properties

4. Form and size of the parts. The form and size are based on judgement. The smallest practicable cross-section may be used, but it may be checked that the stresses induced in the designed cross-section are reasonably safe. In order to design any machine part for form and size, it is necessary to know the forces which the part must sustain. It is also important to anticipate any suddenly applied or impact load which may cause failure.

5. Frictional resistance and lubrication. There is always a loss of power due to frictional resistance and it should be noted that the friction of starting is higher than that of running friction. It is, therefore, essential that a careful attention must be given to the matter of lubrication of all surfaces which move in contact with others, whether in rotating, sliding, or rolling bearings.

6. Convenient and economical features. In designing, the operating features of the machine should be carefully studied. The starting, controlling and stopping levers should be located on the basis of convenient handling. The adjustment for wear must be provided employing the various takeup devices and arranging them so that the alignment of parts is preserved. If parts are to be changed for different products or replaced on account of wear or breakage, easy access should be provided and the necessity of removing other parts to accomplish this should be avoided if possible. The economical operation of a machine which is to be used for production, or for the processing of material should be studied, in order to learn whether it has the maximum capacity consistent with the production of good work.

7. Use of standard parts. The use of standard parts is closely related to cost, because the cost of standard or stock parts is only a fraction of the cost of similar parts made to order. The standard or stock parts should be used whenever possible ; parts for which patterns are already in existence such as gears, pulleys and bearings and parts which may be selected from regular shop stock such as screws, nuts and pins. Bolts and studs should be as few as possible to avoid the delay caused by changing drills, reamers and taps and also to decrease the number of wrenches required.

8. Safety of operation. Some machines are dangerous to operate, especially those which are speeded up to insure production at a maximum rate. Therefore, any moving part of a machine which is within the zone of a worker is considered an accident hazard and may be the cause of an injury. It is, therefore, necessary that a designer should always provide safety devices for the safety of the operator. The safety appliances should in no way interfere with operation of the machine.

9. Workshop facilities. A design engineer should be familiar with the limitations of his employer's workshop, in order to avoid the necessity of having work done in some other workshop. It is sometimes necessary to plan and supervise the workshop operations and to draft methods for casting, handling and machining special parts.

10. Number of machines to be manufactured. The number of articles or machines to be manufactured affects the design in a number of ways. The engineering and shop costs which are called fixed charges or overhead expenses are distributed over the number of articles to be manufactured. If only a few articles are to be made, extra expenses are not justified unless the machine is large or of some special design. An order calling for small number of the product will not permit any undue expense in the workshop processes, so that the designer should restrict his specification to standard parts as much as possible.

11. Cost of construction. The cost of construction of an article is the most important consideration involved in design. In some cases, it is quite possible that the high cost of an article may immediately bar it from further considerations. If an article has been invented and tests of hand made samples have shown that it has commercial value, it is then possible to justify the expenditure of a considerable sum of money in the design and development of automatic machines to produce the article, especially if it can be sold in large numbers. The aim of design engineer under all conditions, should be to reduce the manufacturing cost to the minimum.

12. Assembling. Every machine or structure must be assembled as a unit before it can function. Large units must often be assembled in the shop, tested and then taken to be transported to their place of service. The final location of any machine is important and the design engineer must anticipate the exact location and the local facilities for erection.

17. Explain the various phases in Design using flow diagram and the factors influencing the machine design. **(8) (M/J - 13)**

General Procedure in Machine Design

In designing a machine component, there is no rigid rule. The problem may be attempted in several ways. However, the general procedure to solve a design problem is as follows :

1. *Recognition of need.* First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed.

2. *Synthesis* (*Mechanisms*). Select the possible mechanism or group of mechanisms which will give the desired motion.

3. *Analysis of forces.* Find the forces acting on each member of the machine and the energy transmitted by each member.

4. *Material selection.* Select the material best suited for each member of the machine.

5. *Design of elements* (*Size and Stresses*). Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit.

6. *Modification.* Modify the size of the member to agree with the past experience and judgment to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost.

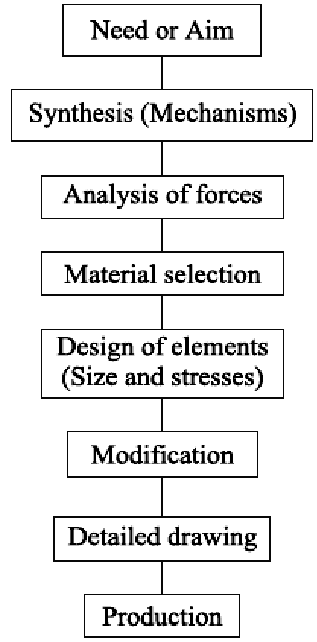


Fig. 1.1. General procedure in Machine Design.

7. *Detailed drawing.* Draw the detailed drawing of each component and the assembly of the machine with complete specification for the manufacturing processes suggested.

8. *Production.* The component, as per the drawing, is manufactured in the workshop.

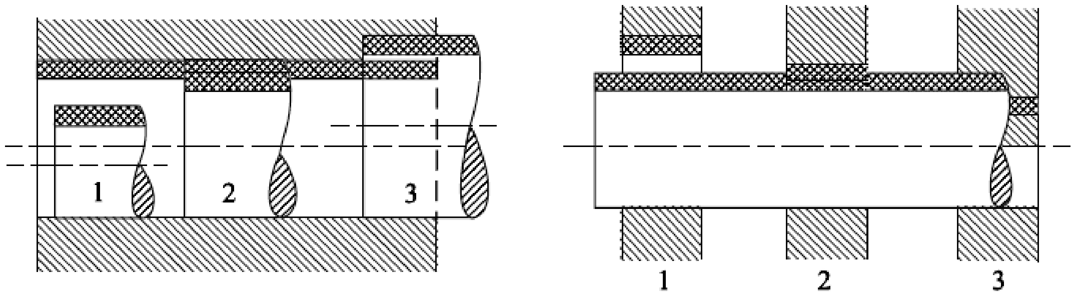
The flow chart for the general procedure in machine design is shown in Fig. 1.1.

18. What is meant by hole basis system and shaft basis system? Which one is preferred? Why? **(8) (M/J - 13)**

The following are two bases of limit system:

1. *Hole basis system.* When the hole is kept as a constant member (*i.e.* when the lower deviation of the hole is zero) and different fits are obtained by varying the shaft size, as shown in Fig. 3.6 (*a*), then the limit system is said to be on a hole basis.

2. *Shaft basis system.* When the shaft is kept as a constant member (*i.e.* when the upper deviation of the shaft is zero) and different fits are obtained by varying the hole size, as shown in Fig. 3.6 (*b*), then the limit system is said to be on a shaft basis.

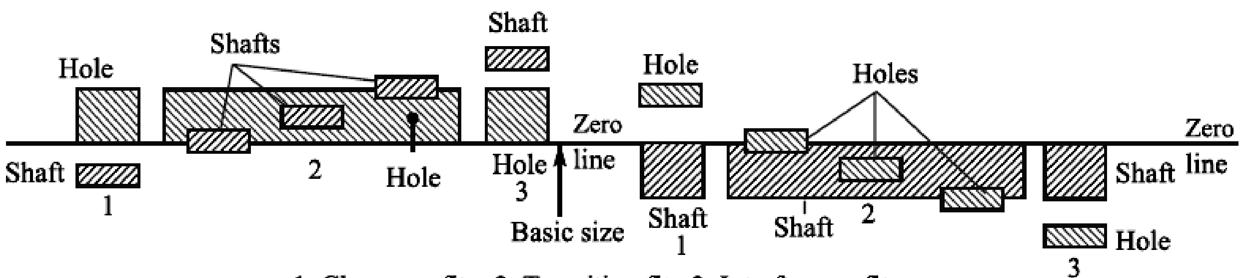


1. Clearance fit. 2. Transition fit. 3. Interference fit.

(*a*) Hole basis system. (*b*) Shaft basis system.

Fig. 3.6. Bases of limit system.

The hole basis and shaft basis system may also be shown as in Fig. 3.7, with respect to the zero line.



1. Clearance fit. 2. Transition fit. 3. Interference fit.

(*a*) Hole basis system. (*b*) Shaft basis system.

Fig. 3.7. Bases of limit system.

19. Write short notes on preferred numbers fits and types of fits. **(16) (M/J - 12)**

PREFERRED NUMBERS

When a machine is to be made in several sizes with different powers or capacities, it is necessary to decide what capacities will cover a certain range efficiently with minimum number of sizes. It has been shown by experience that a certain range can be covered efficiently when it follows a geometrical progression with a constant ratio. The preferred numbers are the conventionally rounded off values derived from geometric series including the integral powers of 10 and having as common ratio of the following factors:

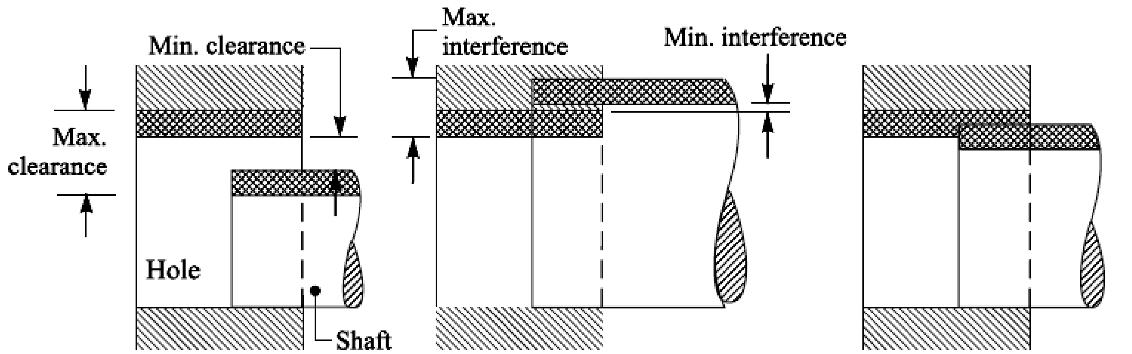
$$\sqrt[5]{10},\ \sqrt[10]{10},\ \sqrt[20]{10} \text{ and } \sqrt[40]{10}$$

These ratios are approximately equal to 1.58, 1.26, 1.12 and 1.06. The series of preferred numbers are designated as *R5, R10, R20 and R40 respectively. These four series are called ***basic series***. The other series called ***derived series*** may be obtained by simply multiplying or dividing the basic sizes by 10, 100, etc. The preferred numbers in the series R5 are 1, 1.6, 2.5, 4.0 and 6.3. Table 3.12 shows basic series of preferred numbers according to IS : 1076 (Part I) – 1985 (Reaffirmed 1990).

FITS

The degree of tightness or looseness between the two mating parts is known as a *fit* of the parts. The nature of fit is characterised by the presence and size of clearance and interference.

The ***clearance*** is the amount by which the actual size of the shaft is less than the actual size of the mating hole in an assembly as shown in Fig. 3.5 (*a*). In other words, the clearance is the difference between the sizes of the hole and the shaft before assembly. The difference must be ***positive.***



(*a*) Clearance fit. (*b*) Interference fit. (*c*) Transition fit.

Fig. 3.5. Types of fits.

The ***interference*** is the amount by which the actual size of a shaft is larger than the actual finished size of the mating hole in an assembly as shown in Fig. 3.5 (*b*). In other words, the interference is the arithmetical difference between the sizes of the hole and the shaft, before assembly. The difference must be ***negative.***

TYPES OF FITS

According to Indian standards, the fits are classified into the following three groups :

1. ***Clearance fit.*** In this type of fit, the size limits for mating parts are so selected that clearance between them always occur, as shown in Fig. 3.5 (*a*). It may be noted that in a clearance fit, the tolerance zone of the hole is entirely above the tolerance zone of the shaft.

In a clearance fit, the difference between the minimum size of the hole and the maximum size of the shaft is known as ***minimum clearance*** whereas the difference between the maximum size of the hole and minimum size of the shaft is called ***maximum clearance*** as shown in Fig. 3.5 (*a*).

The clearance fits may be slide fit, easy sliding fit, running fit, slack running fit and loose running fit.

2. ***Interference fit.*** In this type of fit, the size limits for the mating parts are so selected that interference between them always occur, as shown in Fig. 3.5 (*b*). It may be noted that in an interference fit, the tolerance zone of the hole is entirely below the tolerance zone of the shaft.

In an interference fit, the difference between the maximum size of the hole and the minimum size of the shaft is known as ***minimum interference,*** whereas the difference between the minimum size of the hole and the maximum size of the shaft is called ***maximum interference,*** as shown in Fig. 3.5 (*b*).

The interference fits may be shrink fit, heavy drive fit and light drive fit.

3. ***Transition fit.*** In this type of fit, the size limits for the mating parts are so selected that either a clearance or interference may occur depending upon the actual size of the mating parts, as shown in Fig. 3.5 (*c*). It may be noted that in a transition fit, the tolerance zones of hole and shaft overlap.

The transition fits may be force fit, tight fit and push fit.

REVIEW QUESTIONS: (2 MARKS)

1. What are the various phases of design process? **(A/M - 06)**
2. What do you mean by Optimum design? **(N/D - 07/11)**
3. Describe material properties hardness, stiffness and resilience. **(A/M - 09) (N/D - 09)**
4. Mention some standard codes of specification of steels. **(N/D - 08)**
5. State the difference between straight beams and curved beams. **(N/D - 12)**
6. How is factor of safety is defines for brittle and ductile materials?
7. List the important factors that influence the magnitude of factor safety. **(N/D - 11)**
8. Which are mechanical properties of the metals? List any four mechanical properties. **(M/J - 12)**
9. State Rankine's theory. **(N/D - 07)**
10. Define stress concentration. **(M/J – 09/12)**

16 MARKS

1. The crane hook carries a load of 20 kN as shown in Fig. The section at X-X is rectangular whose horizontal side is 100 mm. Find the stresses in the inner and outer fibres at the given section. **(N/D - 06)**
2. The frame of a punch press is shown in Fig. 5.9. Find the stresses at the inner and outer surface at section X-X of the frame, if W = 5000 N. **(N/D - 05)**
3. A mild steel bracket as shown in Fig., is subjected to a pull of 6000 N acting at 45° to its horizontal axis. The bracket has a rectangular section whose depth is twice the thickness. Find the cross-sectional dimensions of the bracket, if the permissible stress in the material of the bracket is limited to 60 MPa. **(N/D - 07)**

4. A mild steel link, as shown in Fig. by full lines, transmits a pull of 80 kN. Find the dimensions b and t if b = 3t. Assume the permissible tensile stress as 70 MPa. If the original link is replaced by an unsymmetrical one, as shown by dotted lines in Fig. 5.24, having the same thickness t, find the depth b1, using the same permissible stress as before.
5. A pulley is keyed to a shaft midway between two anti-friction bearings. The bending moment at the pulley varies from – 170 N-m to 510 N-m and the torsional moment in the shaft varies from 55 N-m to 165 N-m. The frequency of the variation of the loads is the same as the shaft speed. The shaft is made of cold drawn steel having an ultimate strength of 540 MPa and a yield strength of 400 MPa. Determine the required diameter for an indefinite life. The stress concentration factor for the keyway in bending and torsion may be taken as 1.6 and 1.3 respectively. The factor of safety is 1.5. Take size factor = 0.85 and surface finish factor = 0.88. **(M/J - 12, N/D - 12)**
6. What is the difference between Gerber curve and Soderberg and Goodman lines? **(M/J - 13)**
7. A machine component is subjected to fluctuating stress that varies from 40 to 100 N/mm^2. The corrected endurance limit stress for the machine component is 270 N/mm^2. The tensile strength and yield strength of material are 600 and 450 N/mm^2 respectively. Find the factor of safety using: 1. Gerber theory, 2. Soderberg line, 3. Goodman line and 4. Also, find the factor of safety against static failure. **(M/J - 13)**