CBCS SCHEME

USN 1 C R 1 9 M E 0 3 3 **18ME52**

Fifth Semester B.E. Degree Examination, Feb./Mar. 2022

Design of Machine Elements – I

Time: 3 hrs. Max. Marks: 100

Note: 1. Answer any FIVE full questions, choosing ONE full question from each module.
2. Use of design data handbook is permitted.

Important Note : 1. On completing your answers, compulsorily draw diagonal cross lines on the remaining blank pages.
2. Any revealing of identification, appeal to evaluator and /or equations written eg, 42+8 = 50, will be treated as malpractice.

Module-1

1 a. Explain the factors which influence the selection of engineering materials. **(05 Marks)**

b. Explain codes and standards. List any four organizations who have established specifications for standards and codes. **(05 Marks)**

c. A machine member 60mm diameter is subjected to combined loading as shown in Fig.Q1(c). Determine the maximum principal stress and maximum shear stress at point P.

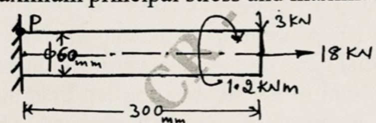

Fig.Q1(c) **(10 Marks)**

OR

2 a. Explain even and uneven materials with the help of Mohr's circles. **(04 Marks)**

b. State and explain the following theories of failure:
(i) Maximum normal stress theory (ii) Maximum shear stress theory
(iii) Distortion energy theory (Hencky Von-Mises theory) **(06 Marks)**

c. A flat bar as shown in Fig.Q2(c) is subjected to an axial pull of 100 kN. Calculate its thickness if allowable tensile stress is 180 MPa.

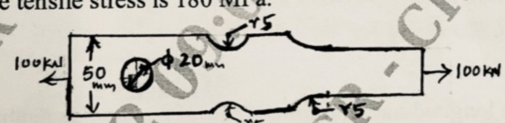

Fig.Q2(c) **(10 Marks)**

Module-2

3 a. Obtain an expression for impact stress induced in a member subjected to axial load. **(05 Marks)**

b. A steel rod 1.5m long has to resist longitudinally an impact of 2.5 kN falling under gravity at a velocity of 0.9925 m/s. The maximum computed stress is to be limited to 150 MPa. Determine the diameter of the round rod. Take E = 210 GPa. **(07 Marks)**

c. A beam of I-section 250mm depth has a moment of inertia of 60×10^6 mm^4. It is simply supported at the ends at a distance of 3m apart. A weight of 3 kN falls at its middle from an unknown height. Determine the safe height 'h' taking the allowable stress as 90 MPa. Take E = 210 GPa. **(08 Marks)**

OR

4 a. Obtain Soderberg's relation for a member subjected to fatigue loading. **(05 Marks)**

b. A steel connecting rod of rectangular cross-section having depth twice that of the width is subjected to a completely reversed axial load of 18 kN. The endurance stress is 300 MPa and yield stress is 420 MPa. Determine suitable cross-sectional dimensions of the connecting rod. Take size factor = 0.9, Load factor = 0.7, Surface factor = 0.85, Stress concentration factor = 1.5, Notch sensitivity = 1. Factor of safety = 1.8. Neglect column effect. **(07 Marks)**

c. A steel rod (σ_y = 400.1 MPa and σ_{-1} = 345.2 MPa) of circular cross-section shown in Fig.Q4(c) is subjected to load varying from 3F to F. Determine the value of F. Use a factor of safety 3. Take stress concentration factor = 1.43, q = 1.

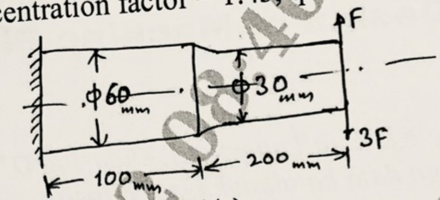

Fig.Q4(c)

Load factor = 1.0 , Size factor = 0.85, Surface factor = 0.85. **(08 Marks)**

Module-3

5 A commercial steel shaft 1m long supported between bearings carries a pulley of diameter 600mm weighing 1 kN located 400mm to the right of the right hand bearing and receives 25 kW at 1000 rpm by a horizontal belt drive. The power from the shaft is transmitted by a spur pinion of 20° pressure angle having pitch circle diameter 200mm to a spur gear such that the tangential force on gear acts vertically upwards. The pinion is keyed to the shaft at a distance of 200mm to the right of the left bearing. Taking the ratio of belt tensions as 3, determine the diameter of the shaft required. Use maximum shear stress theory. Take $\tau_d = 40\ N/mm^2$. **(20 Marks)**

OR

6 a. Select a rectangular sunk key to transmit 9 kW at 300 rpm. The yield stress for the steel used is 310 MPa. Take factor of safety as 2.5 . **(06 Marks)**

b. Design a rigid flange coupling (Un-protected) to transmit 18 kW at 1440 rpm. The allowable shear stress for CI flange is 4 MPa. The shafts, keys and bolts are made of annealed steel having allowable shear stress of 93 MPa. Take allowable crushing stress = 186 MPa for key. Take key way factor $\eta = 0.75$ for shaft. **(14 Marks)**

Module-4

7 a. Design a longitudinal joint for a boiler of 2m diameter subjected to a pressure of 1 MPa. The joint is a triple riveted butt joint with equal covers and efficiency of 85%. The pitch of the outer row is twice the pitch of inner rows. The arrangement is of chain type. Take allowable stress in tension = 117.67 N/mm^2, in shear = 70.6 N/mm^2 and in crushing = 176.50 N/mm^2. Take coefficient $k_1 = 6$ and corrosion allowance of 2 mm. **(12 Marks)**

b. A bracket attached to a vertical column by means of four identical rivets, is subjected to an eccentric force of 25 kN as shown in Fig.Q7(b). Determine the diameter of rivets, if the permissible shear stress is 60 N/mm^2.

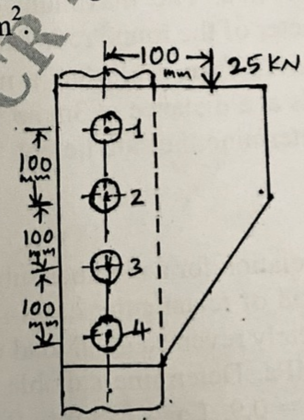

Fig.Q7(b) **(08 Marks)**

OR

8 a. A welded connection as shown in Fig.Q8(a) is subjected to an eccentric force of 60 kN in the plane of the welds. Determine the size of the welds, if the permissible shear stress for the weld is 100 N/mm^2. Assume static conditions.

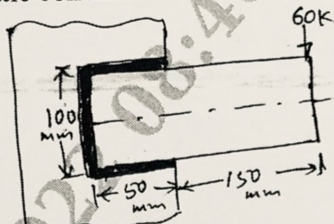

Fig.Q8(a)

(12 Marks)

b. Determine the load carrying capacity of a welded joint loaded as shown in Fig.Q8(b). The allowable shear stress for 10mm weld used is 50 MPa.

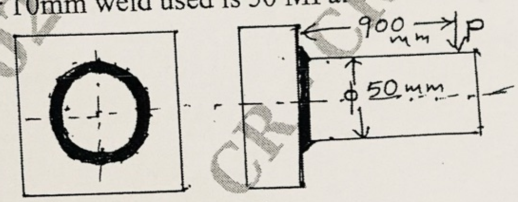

Fig.Q8(b)

(08 Marks)

Module-5

9 a. It is required to design a cottor joint to connect two steel rods of equal diameter. Each rod is subjected to axial tensile force of 50 kN. Design the joint and specify main dimensions. Take permissible stresses for rods in tension = 67 N/mm^2. Crushing = 134 N/mm^2 and for cottor in tension = 100 N/mm^2. **(08 Marks)**

b. Explain self-locking in power screws. **(02 Marks)**

c. A bracket is bolted as shown in Fig.Q9(c). All the bolts are of same size and are made of steel having allowable tensile stress of 90 MPa and allowable shear stress of 52 MPa. Determine the size of the bolts to be used.

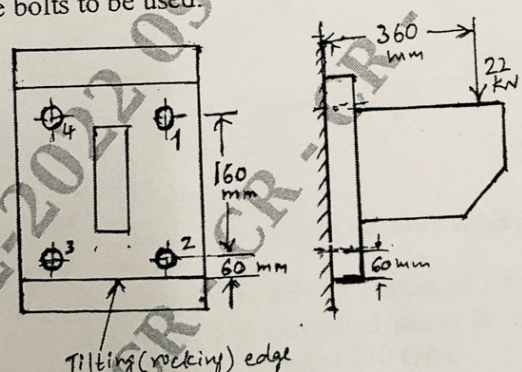

Fig.Q9(c)

(10 Marks)

OR

10 a. Obtain an expression for torque required to raise the load in power screws. **(08 Marks)**

b. Enumerate four typical applications of knuckle joint. **(02 Marks)**

c. A machine weighing 20 kN is to be raised by a single start square threaded screw rod of 50mm diameter, 8mm pitch screw jack at a maximum speed of 6 m/minute. If the coefficient of friction for threads is 0.2, determine the power required to raise (lift) the machine. The inside and outside diameters of the thrust collar are 30 and 60mm respectively. The coefficient of friction for collar is 0.1. **(10 Marks)**

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18ME52 - Design of Machine Elements- 1 Feb 2022
Answer key

Module 1

1a. The choice of materials depends upon the following factors
- a. Availability of the materials.
- b. Suitability of materials for the working conditions in service
- c. The cost of materials.

1b. Codes and standards:

Codes are a set of specifications for the analysis, design, manufacture and construction of something. The main purpose of code is to achieve a specified degree of safety, efficiency and quality.

Standard is the set of specifications for parts, materials or processes intended to achieve uniformity and specified quality.

Some organisations for codes and standards include American Gear Manufacturers Association (AGMA), American Society for Mechanical Engineers (ASME), Society of Automotive Engineers (SAE) and American society of Metals (ASM).

1c.

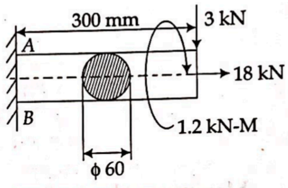


Fig. 1.36

Given data: Axial load, $F_1 = 18$ kN $= 18000$ N

Transverse load, $F_2 = 3$ kN $= 3000$ N

Torque, $M_t = 1.2$ kNm $= 1.2 \times 10^6$ N-mm.

Bending moment due to transverse load, $M_b = F_2 \cdot l$

$$M_b = 3000 \times 300 = 900 \times 10^3 \text{ N-mm}$$

$$\text{Direct tensile stress, } \sigma_d = \frac{F_1}{A} = \frac{18000}{\left(\frac{\pi}{4}60^2\right)} = 6.37 \text{ MPa}$$

This is tensile both A and B.

$$\text{Bending stress due to transverse load, } \sigma_B = \frac{M_b \cdot C}{I}$$

$$C = \frac{d}{2} = \frac{60}{2} = 30 \text{ mm}$$

$$I = \frac{\pi d^4}{64} = \frac{\pi \times 60^4}{64} = 636.17 \times 10^3 \text{ mm}^4$$

$$\therefore \quad \sigma_B = \frac{900 \times 10^3 \times 30}{636.17 \times 10^3} = 42.44 \text{ MPa}$$

This is tensile at A and compressive at B.

Total tensile stress at A.

$$\therefore \quad \sigma_A = \sigma_d + \sigma_B = 6.37 + 42.44 = \sigma_x$$

$$\sigma_x = 48.81 \text{ MPa}$$

$$\text{Shear stress, } \tau = \tau_{xy} = \frac{M_t \cdot r}{J}$$

$$r = \frac{d}{2} = \frac{60}{2} \text{ mm}, \quad J = \frac{\pi d^4}{32} = \frac{\pi (60)^4}{32} \text{ mm}^4$$

$$\therefore \quad \tau_{xy} = \frac{1.2 \times 10^6 \times (60/2)}{\left(\frac{\pi (60)^4}{32}\right)} = 28.29 \text{ MPa}$$

∴ Maximum principal stress,

$$\sigma_1 = \frac{1}{2}\left[\sigma_x + \sqrt{\sigma_x^2 + 4\tau_{xy}^2}\right]$$

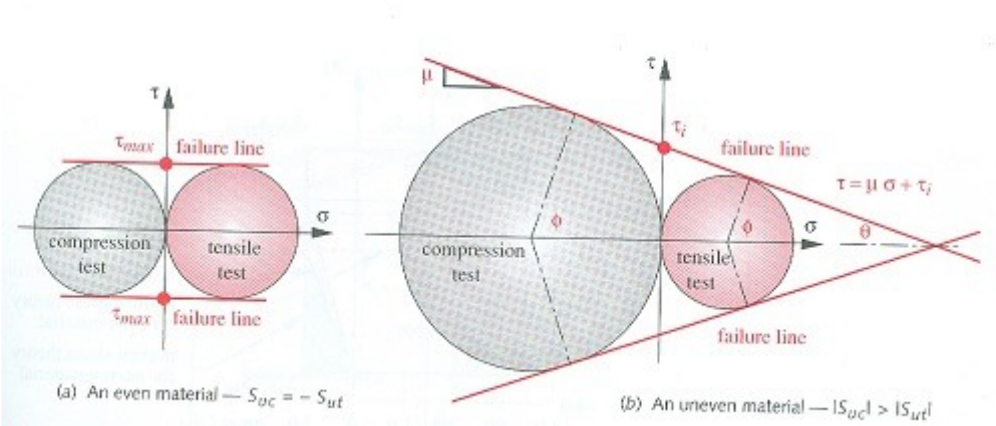
$$= \frac{1}{2}\left[48.81 + \sqrt{48.81^2 + 4(28.29)^2}\right]$$

$$\sigma_1 = 61.77 \text{ MPa}$$

Maximum shear stress, $\tau_{max} = \frac{1}{2}\sqrt{\sigma_x^2 + 4\tau_{xy}^2}$

$$= \frac{1}{2}\sqrt{48.81^2 + 4(28.29)^2} = 37.36 \text{ MPa.}$$

2a.



(a) An even material — $S_{uc} = -S_{ut}$ (b) An uneven material — $|S_{uc}| > |S_{ut}|$

Brittle materials can be classified into two classes: even and uneven brittle materials. Even brittle materials generally have compressive strengths equal to their tensile strengths. (Most ductile materials, which we've discussed earlier, are also even materials.) This can happen with wrought materials, like some tool steels, or other metals which have undergone hardening. (Many ductile metals can be worked to increase their hardness - this frequently leads to them becoming less ductile and more brittle.) Even brittle materials can be evaluated for failure by comparing their maximum normal stress to their tensile strengths.
Uneven materials have compressive strengths greater than their tensile strengths. This is typical of many materials which are cast. These materials normally contain microscopic voids due to the casting process. When they're subjected to tension, these voids can produce stress concentrations and lead to rapid fracture. However, when they're subjected to compression, the flaws of the casting are pushed together and their faces can support each other, allowing the material to survive much more force before failing.
The Mohr's circle on the left hand side shows the stresses seen in tension and compression tests for an even material. The one on the right shows the same two tests for an uneven material. The important takeaway from this is that, for uneven materials, their normal and shear strengths are interdependent.

2b.

1. Maximum Principal Stress Theory (Rankine's Theory)

According to this theory failure of a component takes place if the maximum principal stress at any point exceeds the ultimate or yield point stress.

If the principal stresses acting in the three directions are σ_1, σ_2 and σ_3 and if $\sigma_1 > \sigma_2 > \sigma_3$, then the design equation is:

$$\sigma_1 = \frac{\sigma_y}{FOS} \qquad (\sigma_y = \text{Yield point stress})$$

and $\sigma_1 = \frac{1}{2}[(\sigma_x + \sigma_y) + \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2}$

or $\sigma_1 = \frac{\sigma_y}{FOS} = \sigma_e$ = Equivalent stress or Allowable stress

$$\therefore \sigma_e = \frac{1}{2}[(\sigma_x + \sigma_y) + \sqrt{(\sigma_x - \sigma_y)^2 + 4\tau_{xy}^2} \quad (5.20)$$

2. Maximum shear stress theory

Condition for failure,

Maximum shear stress induced at a critical point under triaxial combined stress > Yield strength in shear under tensile test

$$\tau_{max} \le \frac{\tau_y}{FOS} \text{ or } \tau_{max} = \frac{\tau_y}{FOS}$$

Since $\tau_y = \frac{\sigma_{yt}}{2}$, $\tau_{max} = \frac{\sigma_{yt}}{2 \times FOS} = \sigma_e$ = Equivalent or Allowable stress.

This theory is mostly used for ductile materials.

3. Maximum distortion Energy or Hencky and Mises Theory

According to this theory the cause of failure is not total strain energy. Part of the strain causes uniform extension and the remaining part causes shearing action, known as shear energy or energy distortion.

Thus this theory states that failure occurs when the strain energy of distortion per unit volume at any point in the machine element becomes equal to the strain energy per unit volume in a standard tension test specimen, when yielding start.

In tension test,

$$\sigma_1 = \sigma_x = \sigma_{yt} \,;\, \sigma_2 = \sigma_3 = 0.$$

(a) For Bi-axial Stress state:

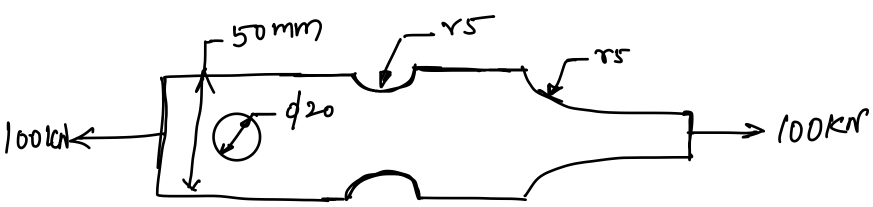
$$(\frac{\sigma_{yt}}{FOS})^2 = \sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2$$

(b) For Tri-axial Stress,

$$(\frac{\sigma_{yt}}{FOS})^2 = \sigma_1^2 + \sigma_2^2 + \sigma_3^2 - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1$$

$$= \frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_1 - \sigma_3)^2}{2}$$

2c.



Consider the hole section

Width of the plate = 50mm = w } $\frac{d}{w} = \frac{20}{50} = 0.4$
dia of hole = 20mm = d

From Fig. 4.5, for $\frac{d}{\omega} = 0.4$, $K_t = 2.25$

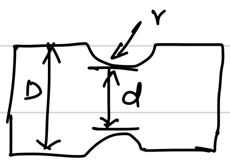
Also $K_t = \frac{\sigma_{max}}{\sigma_{nom}} \Rightarrow \sigma_{nom} = \frac{180}{2.25} = 408\ N/mm^2$

Also, $\sigma_{nom} = \frac{F}{(\omega - d)h}$

$$405 = \frac{100 \times 10^3}{(50-20) \times h}$$

$\Rightarrow h = 8.23\ mm$

(ii) Consider the section with notch.



$d = D - 2r$

$= 50 - 2(5) = 40\ mm$

$\frac{D}{d} = \frac{50}{40} = 1.25$ $\quad \frac{r}{d} = \frac{5}{40} = 0.125$

From Fig. 4.22A, $K_t = 2.27$.

Also $K_t = \frac{\sigma_{max}}{\sigma_{nom}} \Rightarrow 2.27 = \frac{180}{2.27}$

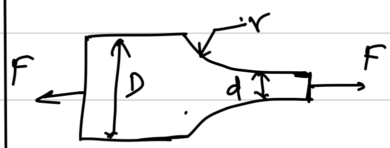
$\Rightarrow \sigma_{nom} = 79.29\ N/mm^2$

Also, $\sigma_{nom} = \frac{F}{h \cdot d}$

$$79.29 = \frac{100 \times 10^3}{h \times 40}$$

$\Rightarrow h = 31.52\ mm$

(iii) Consider the Fillet section



$\frac{r}{d} = \frac{5}{40} = 0.125$

$\frac{D}{d} = \frac{50}{40} = 1.25$

$K_t = 1.78$

$K_t = \frac{\sigma_{max}}{\sigma_{nom}} \Rightarrow \sigma_{nom} = \frac{180}{1.78} = 101.12\ N/mm^2$

Also $\sigma_{nom} = \frac{F}{hd}$

$$101.12 = \frac{100 \times 10^3}{h \times 40}$$

$\Rightarrow h = 24.72\ mm.$

Recommended thickness is greater of all values

$h = 31.52\ mm$

Module 2

3a.

IMPACT STRESS FROM AXIAL LOAD

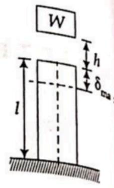


Fig. 5.1

Ref Fig 5.1

W = weight of the falling object, N

h = height of fall, mm

l = length, mm

A = area of c/s, mm^2

δ_{max} = maximum instantaneous deflection under impact, mm

δ_{ST} = static deflection (deflection under static load W), mm

E = modulus of elasticity, MPa

Vol = volume = $A.l$, mm^3

σ_{max} = maximum stress under impact or maximum instantaneous stress or impact stress

σ_{ST} = static stress = $\frac{W}{A}$

U = resilience or energy stored.

Figure 5.1 shows a weight W falls form a height of 'h' on to an element of cross sectional area A and length l

δ_{max} = max instantaneous deflection under impact.

Potential energy, $PE = mg(h + \delta_{max}) = W(h + \delta_{max})$

Equate this resilience, U = unit resilience × Vol

For axial loading, $U = \frac{\sigma^2}{2E} \cdot \text{vol}$

For impact, $U = \frac{\sigma_{max}^2}{2E} \cdot Al$

$$\therefore \quad \frac{\sigma_{max}^2}{2E} \cdot Al = W(h + \delta_{max})$$

$$\sigma_{max}^2 = W\frac{2E}{Al} \cdot (h + \delta_{max})$$

put $\frac{W}{A} = \sigma_{ST}$

For axial loads, deflection = Change in length

$$\delta = \frac{Wl}{AE} = \frac{\sigma l}{E}$$

$$\therefore \quad \delta_{max} = \frac{\sigma_{max} \cdot l}{E}$$

$$\text{Substituting, } \sigma_{max}^2 = \frac{2\sigma_{ST} E}{l}\left(h + \frac{\sigma_{max}\, l}{E}\right)$$

$$\sigma_{max}^2 = \frac{2\sigma_{ST} Eh}{l} + 2\sigma\sigma_{max}$$

$$\sigma_{max}^2 - 2\sigma_{ST}\sigma_{max} = \frac{2\sigma_{ST} Eh}{l}$$

Add σ_{ST}^2 on both sides

$$\sigma_{max}^2 - 2\sigma_{ST}\,\sigma_{max} + \sigma_{ST}^2 = \sigma_{ST}^2 + \frac{2\sigma_{ST} Eh}{l}$$

$$(\sigma_{max} - \sigma_{ST})^2 = \sigma_{ST}^2\left(1 + \frac{2Eh}{\sigma_{ST}\, l}\right)$$

Taking square root on both sides

$$\sigma_{max} - \sigma_{ST} = \sigma_{ST}\sqrt{1+\frac{2Eh}{\sigma_{ST}l}}$$

$$\sigma_{max} = \sigma_{ST} + \sigma_{ST}\sqrt{1+\frac{2Eh}{\sigma_{ST}l}}$$

$$\sigma_{max} = \sigma_{ST}\left(1+\sqrt{1+\frac{2Eh}{\sigma_{ST}l}}\right)$$

Put $\sigma_{ST} = \frac{W}{A}$

$$\sigma_{max} = \frac{W}{A} + \left(1+\sqrt{1+\frac{2hEA}{Wl}}\right)$$

Again substituting $\frac{W}{A} = \sigma_{ST}$

and $\delta_{ST} = \frac{\sigma_{ST}l}{E} \Rightarrow \frac{E}{\sigma_{ST}l} = \frac{1}{\delta_{ST}}$

$$\sigma_{max} = \sigma_{ST}\left(1+\sqrt{1+\frac{2h}{\delta_{st}}}\right)$$

3b. **Given data:** $l = 1.5 \text{ m} = 1500 \text{ mm}$, $W = 2.5 \text{ kN} = 2.5 \times W^3 \text{ N}$

$v = 0.9925 \text{ m/s}$, $\sigma_{max} = 150 \text{ MPa}$

Find $d = ?$

To find height of fall 'h'.

We know that $v^2 - u^2 = 2as$

For falling under gravity, $u = 0$, $a = g$, $s = h$

$\therefore$ $$V^2 = 2gh \Rightarrow h = \frac{V^2}{2g} = \frac{0.9925^2}{2\times 9.81} = 0.0502 \text{ m}$$

$\therefore$ height of fall, $h = 50.2$ mm

Assume, $E = 210 \times 10^3$ MPa

Impact axial stress, $$\sigma_{max} = \frac{W}{A}\left(1+\sqrt{1+\frac{2hAE}{Wl}}\right)$$

$$90 = 4.6875\left(1+\sqrt{1+\frac{2h}{0.1339}}\right)$$

$$\frac{90}{4.6875} - 1 = \sqrt{1+14.93(h)}$$

$$h = 22.12 \text{ mm.}$$

3c.

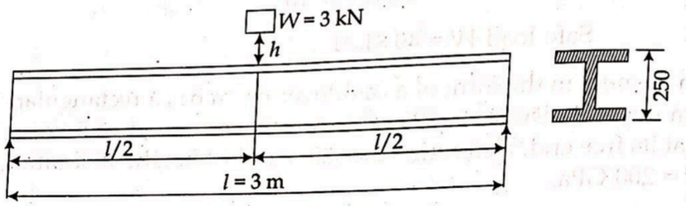


Fig. 5.7

Given data:

$$d = 250 \text{ mm}$$
$$I = 60 \times 10^6 \text{ mm}^4$$
$$l = 3000 \text{ mm}$$
$$W = 3 \text{ kN} = 3 \times 10^3 \text{ N}$$
$$E = 210 \times 10^3 \text{ MPa}$$
$$\sigma_{b\max} = 90 \text{ MPa}$$

For bending, impact stress, $\sigma_{b\max} = \sigma_{bST}\left(1+\sqrt{1+\dfrac{2h}{\delta_{ST}}}\right)$ ...(

$$\sigma_{bST} = \frac{M_b C}{I}$$

For simply supported beam with central load ...(T 2-8/F

$$M_b = \frac{Wl}{4} = \frac{3000 \times 3000}{4} = 2.25 \times 10^6 \text{ N-mm}$$

$$C = \frac{d}{2} = \frac{250}{2} = 125 \text{ mm}$$

$$\sigma_{bST} = \frac{2.25 \times 10^6 \times 125}{60 \times 10^6} = 4.6875 \text{ MPa}$$

Static deflection, $\delta_{ST} = \dfrac{Wl^3}{48EI}$ for simply supported beam with central load

...(T 2-8/P 2

$$\delta_{ST} = \frac{3000 \times 3000^3}{48 \times 210 \times 10^3 \times 60 \times 10^6} = 0.1339 \text{ mm}$$

4a.

4.4 DERIVATION OF SODERBERG'S RELATION

Figure 4.3 shows a graph drawn for σ_a V_s σ_m. Line AB is Soderberg's failure line. Line CD is safe Soderberg's line. Consider any point $P(\sigma_m, k_\sigma\sigma_a)$ on line CD. Drop a perpendicular PE from P.

Δ*ls COD* and *PED* are similar

$$\frac{PE}{OC} = \frac{DE}{OD}$$

But $\quad DE = OD - OE$

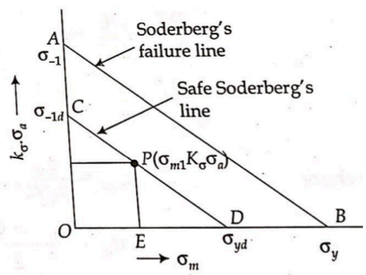


Fig. 4.3

$$\therefore \quad \frac{PE}{OC} = \frac{OD - OE}{OD} = 1 - \frac{OE}{OD}$$

$$\frac{PE}{OC} + \frac{OE}{OD} = 1$$

where $PE = y$ coordinate of $P = k_\sigma \sigma_a$

$OC = \sigma_{-1d}$

$OE = x$-coordinate of $E = \sigma_m$

$OD = \sigma_{yd}$

Substituting

$$\frac{k_\sigma \sigma_a}{\sigma_{-1d}} + \frac{\sigma_m}{\sigma_{yd}} = 1$$

where, $\sigma_a = \dfrac{\sigma_{max} - \sigma_{min}}{2}$, $\quad \sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2}$

$\sigma_{-1d} = \dfrac{\sigma_{-1} k_l k_s k_z}{FOS}$ $\quad \sigma_{yd} = \dfrac{\sigma_y}{FOS}$

FOS = Factor of safety.

4b.

Given data: Rectangular cross-section $d = 2\,b$

Completely reversed load of 18 kN = 18000 N

$\therefore$ $F_{max} = +18{,}000$ N

$F_{min} = -18{,}000$ N

$\sigma_{-1} = 300$ MPa $\quad \sigma_y = 420$ MPa

FOS = 1.8 $\quad$ Size factor, $k_z = 0.9$

For axial loads, load factor, $k_l = 0.7$

Assume surface factor, $k_s = 0.85$

Stress concentration factor, $k_\sigma = 1.5$

Since notch sensitivity is not given, $k_{\sigma s} = k_\sigma = 1.5$

$$\sigma_{yd} = \frac{\sigma_y}{FOS} = \frac{420}{1.8} = 233.33 \text{ MPa}$$

$$\sigma_{-1d} = \frac{\sigma_{-1} k_l k_s k_z}{FOS} = \frac{300 \times 0.7 \times 0.85 \times 0.9}{1.8} = 89.25 \text{ MPa}$$

For axial load, $\sigma = \dfrac{F}{A}$

$$\sigma_{max} = \frac{F_{max}}{A} = \frac{+18 \times 10^3}{A}$$

$$\sigma_{min} = \frac{F_{min}}{A} = \frac{-18 \times 10^3}{A}$$

$\therefore$ Mean stress, $\sigma_m = \dfrac{\sigma_{max} + \sigma_{min}}{2} = 0$

$$\text{Amplitude stress, } \sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{\frac{18 \times 10^3}{A} - \left(\frac{-18 \times 10^3}{A}\right)}{2}$$

$$= \frac{18 \times 10^3}{A}$$

Applying Soderberg's relation

$$\frac{k_{\sigma a}\sigma_a}{\sigma_{-1d}} + \frac{\sigma_m}{\sigma_{yd}} = 1$$

$$1.8 \times \frac{18 \times 10^3}{A \times 89.25} + 0 = 1$$

$$A = 363.025 = bd = b \times 2b$$

width, $b = 13.47 \simeq 14$ mm

depth, $d = 2b = 28$ mm.

4c.

Given data:

$F_{max} = +3P$

$F_{min} = -P$ (opposite direction)

$l = 200$ mm (upto change of c/s)

$D = 60$ mm, $d = 30$ mm $r = 6$ mm

$\sigma_u = 620.8$ MPa, $\sigma_y = 400.1$ MPa, $\sigma_{-1} = 345.2$ MPa

FOS = 3

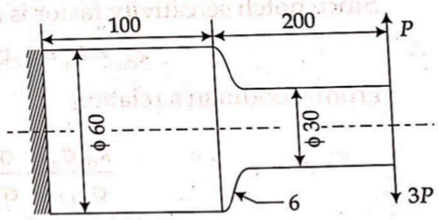


Fig. 4.6

For bending load,

Load factor, $k_l = 1$

Assume size factor, $k_z = 0.85$

Surface factor, $k_s = 0.85$

$$\therefore \quad \sigma_{ud} = \frac{\sigma_u}{FOS} = \frac{620.8}{3} = 206.93 \text{ MPa}$$

$$\therefore \quad \sigma_{yd} = \frac{\sigma_y}{FOS} = \frac{400.1}{3} = 133.37 \text{ MPa}$$

$$\sigma_{-1d} = \frac{\sigma_{-1}k_l k_s k_z}{FOS} = \frac{345.2 \times 1 \times 0.85 \times 0.85}{3} = 83.14 \text{ MPa}$$

$$M_{bmax} = F_{max} \cdot l = 3P \times 200 = 600P$$

$$M_{b\,min} = F_{min}\, l = P \times 200 = 200P$$

$$C = d/2 \text{ and } I = \pi d^4/64$$

For bending loads stress, $\sigma = \dfrac{M_b C}{I}$

$$\sigma_{max} = \frac{M_{b\,max}\,C}{I} = \frac{600p \times \frac{30}{2}}{\left(\frac{\pi 30^4}{64}\right)} = 0.2264P$$

$$\sigma_{min} = \frac{M_{b\,min}\,C}{I} = \frac{-200p \times \frac{30}{2}}{\left(\frac{\pi \times 30^4}{64}\right)} = -0.0755P$$

$\therefore$ Mean stress, $\sigma_m = \frac{\sigma_{max} + \sigma_{min}}{2} = \frac{(0.2264 - 0.0755)P}{2}$

$\sigma_m = 0.0755P$

Amplitude stress, $\sigma_a = \frac{\sigma_{max} - \sigma_{min}}{2} = \frac{[0.2264 - (-0.0755)]P}{2}$

$\sigma_a = 0.1509P$

From Fig. 3.13 stress concentration factor ...(Fig. 4-21 A

$$\frac{r}{d} = \frac{6}{30} = 0.2, \quad \frac{D}{d} = \frac{60}{30} = 2, \quad k_\sigma = 1.43$$

Since notch sensitivity factor is not given, $q = 1$

$\therefore$ $k_{\sigma a} = k_\sigma = 1.43$

From Goodman's relation

$$\frac{k_{\sigma a}\sigma_a}{\sigma_{-1d}} + \frac{\sigma_m}{\sigma_{ud}} = 1$$

$$\frac{1.43 \times 0.0755P}{83.14} + \frac{0.1509P}{206.93} = 1$$

$\therefore$ $P = 493.14$ N

Module 3

5a.

Solution:

The shaft is subjected to combined bending and torsion and hence the shaft diameter is obtained from

$$D = \left[\frac{16}{\pi\tau_{ed}}\left\{(K_b M_b)^2 + (K_t M_t)^2\right\}^{\frac{1}{2}}\right]^{\frac{1}{3}} \quad \text{...E(14.12)}$$

Where $\tau_{ed} = 40$ MPa, *for commercial steel according to ASME code*

$K_b = 1.5, \quad K_t = 1$

To find M_t, the torque
Consider the pulley

Ratio of belt tensions $= \frac{T_1}{T_2} = e^{\mu\theta} = e^{0.3\times180\times\frac{\pi}{180}} = 2.57$

Here $T_1 = T_{max} = 3000$ N

Therefore, $T_2 = \frac{T_1}{e^{\mu\theta}} = \frac{3000}{2.57} = 1167.32$ N

Hence torque, $M_t = (T_1 - T_2) r_{\text{Pulley1}}$

$$= (3000 - 1167.32)\frac{600}{2}$$

$$= 549804 \text{ N-mm}$$

To find M_b, the maximum bending moment on shaft

Consider pulley 1

Horizontal load on pulley 1 = 0.

Vertical load on pulley 1 $= (T_1 + T_2 + W_p)\downarrow$

$$= (3000 + 1167.32 + 800)\downarrow$$

$$= 4967.32 \text{ N}\downarrow$$

Consider pulley 2

Horizontal load on pulley 2 $= (T_1' + T_2') \rightarrow$

To find T_1' and T_2'

Ratio of belt tensions $\frac{T_1'}{T_2'} = 2.57$

***Note:** Torque transmitted by both pulleys is the same.*

Therefore, $M_t)_{\text{Pulley1}} = M_t)_{\text{Pulley2}}$

$$549809 = (T_1' - T_2') r_{\text{Pulley2}}$$

$$= (2.57T_2' - T_2')\frac{800}{2}$$

$$T_2' = 875.5 \text{ N}$$

$$T_1' = 2626.48 \text{ N}$$

Horizontal load on pulley 2 $= 3501.98 \text{ N} \rightarrow$

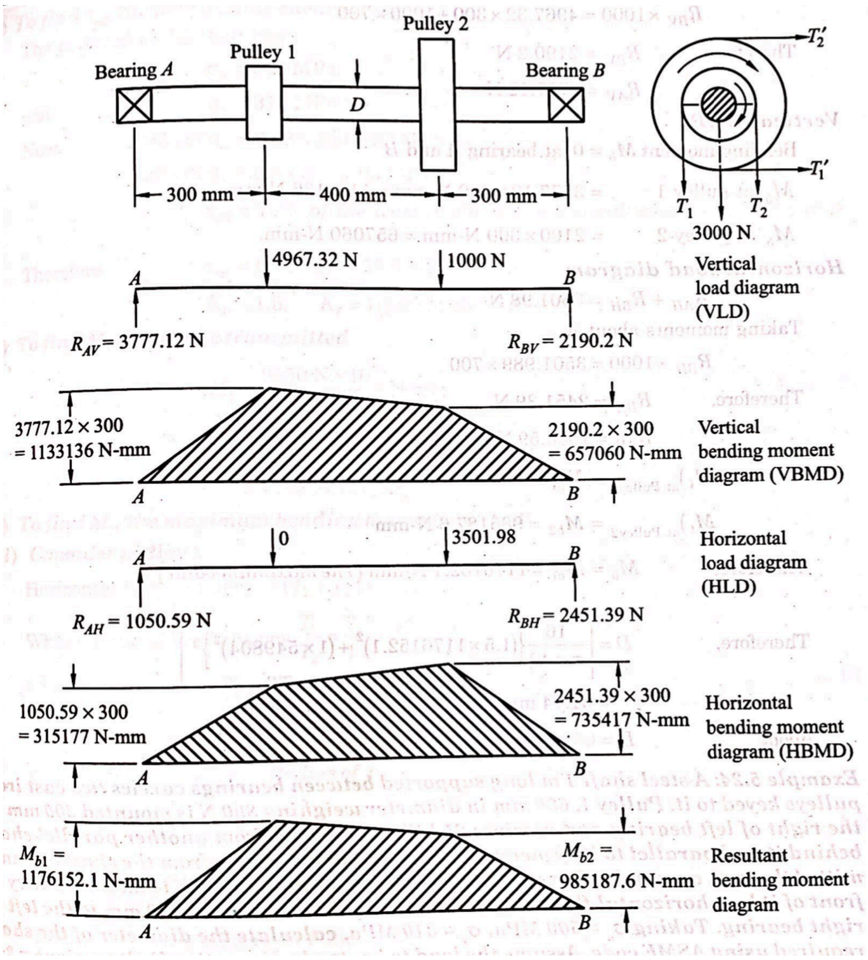
Vertical load on pulley 2 $= Wp_2 \downarrow = 1000 \text{ N} \downarrow$

Vertical load diagram

$$R_{AV} + R_{BV} = 4967.32 + 1000$$

$$= 5967.32 \text{ N} \uparrow$$

Taking moments about A, and equating the sum of clockwise moments to sum of anticlockwise moments,



$$R_{BV} \times 1000 = 4967.32 \times 300 + 1000 \times 700$$

Therefore, $R_{BV} = 2190.2$ N

$R_{AV} = 3777.12$ N

Vertical BMD

Bending moment $M_b = 0$ at bearing A and B

M_b at pulley 1 $= 3777.12 \times 300$ N-mm $= 1133136$ N-mm

M_b at pulley 2 $= 2190 \times 300$ N-mm $= 657060$ N-mm

Horizontal load diagram

$$R_{AH} + R_{BH} = 3501.98 \text{ N}$$

Taking moments about A,

$$R_{BH} \times 1000 = 3501.989 \times 700$$

Therefore, $R_{BH} = 2451.39$ N

$R_{AH} = 1050.59$ N

$M_t)_{\text{at Pulley1}} = M_{b1} = 1176152.1$ N-mm

$M_t)_{\text{at Pulley2}} = M_{b2} = 985187.6$ N-mm

Therefore, $M_b = M_{b1} = 1176152.1$ N-mm (*The maximum value*)

Therefore, $$D = \left[\frac{16}{\pi \times 40}\left\{(1.5 \times 1176152.1)^2 + (1 \times 549804)^2\right\}^{\frac{1}{2}}\right]^{\frac{1}{3}}$$

$= 61.74$ mm

Adopt $D = 65$ mm

6a.

Given data: Power, $P = 9$ kW, Speed, $n = 300$ rpm

$\sigma_y = 310$ MPa, FOS = 2.5

$\therefore$ Allowable stress, $\sigma = \frac{\sigma_y}{FOS} = \frac{310}{2.5} = 124 \text{ MPa} = \sigma_b'$

assume $\tau = 0.5\sigma = 62 \text{ MPa} = \tau_k$

with key way in the shaft assume, $\tau_s = 0.75\tau$

$\tau_s = 0.75 \times 62 = 46.5$ MPa.

Step 1: Torque, $M_t = \frac{9.55 \times 10^6 P}{n} = \frac{9.55 \times 10^6 \times 9}{300} = 286.5 \times 10^3$ N-mm

Step 2: Dia of shaft, $d = \left[\frac{16 M_t}{\pi \tau_S}\right]^{1/3}$

$$d = \left[\frac{16 \times 286.5 \times 10^3}{\pi \times 46.5}\right]^{1/3} = 31.54 \simeq 32 \text{ mm}$$

Step 3: Ref to Table 7.1, select standard key ...(Table 17-1/P 17.10)

For shaft dia, $d = 32$

Select, $b = 10$ mm

$h = 8$ mm

Step 4: To find length of the key

(a) Width, $b = \frac{2M_t}{dl\tau_K}$

$$10 = \frac{2 \times 286.5 \times 10^3}{32 \times l \times 62} \Rightarrow l = 28.88$$

(b) Thickness, $h = \frac{4M_t}{dl\sigma_b'}$

$$8 = \frac{4 \times 286.5 \times 10^3}{32 \times l \times 124} = 36.1 \text{ mm}$$

Recommended length, $l = 36.1 \simeq 37$ mm (bigger value).

6b.

Step 1: Torque, $M_t = \frac{9.55 \times 10^6 P}{n} = 119.4 \times 10^3$ N-mm

Step 2: Diameter of shaft, $d = \left[\frac{16Mt}{\pi \eta \tau_s}\right]^{\frac{1}{3}}$ $d = \left[\frac{16 \times 119.4 \times 10^3}{\pi \times 0.75 \times 93}\right]^{1/3} = 20.58$

Step 3: No. of bolts, $i = 0.02d + 3 = 3.44 \simeq 4$ bolts

Step 4: Bolt circle diameter, $D_1 = 2d + 50 = 94$ mm

Step 5: Bolt diameter, $d_1 = \frac{0.5d}{\sqrt{i}} = 5.5$ mm

Standard diameter from table, $d_1 = 6$ mm ($M6$ bolt)

Step 6: Check for shear stress in bolt:

Torque capacity of bolts, $M_t = i\left(\frac{\pi}{4}d_1^2\right)\tau_b \cdot \frac{D_1}{2}$

$$119.4 \times 10^3 = 4\left(\frac{\pi}{4}6^2\right)\tau_b \cdot \frac{94}{2}$$

$\therefore$ $\tau_b = 22.46 < \tau_b$ allowable (93 MPa) safe.

Step 7: Design of hub:

(a) Hub diameter, $D_2 = 1.5d + 25$

$$D_2 = 1.5 \times 22 + 25 = 58 \text{ mm}$$

(b) Hub length, $l = 1.25d + 18.75 = 46.25 \simeq 47$ mm

Step 8: Design of flange:

(a) Outside diameter of flange, $D = 2.5d + 75 = 130$ mm

(b) Flange thickness, $t = 0.25d = 5.5 \simeq 6$ mm

Step 9: Check for shear stress in flange

Torque capacity of flange, $M_t = t(\pi D_2)\,\tau_f\left(\frac{D_2}{2}\right)$

$$119.4 \times 10^3 = 6\,(\pi \times 58)\tau_f \times \left(\frac{58}{2}\right)$$

$\therefore$ Shear stress in flange, $\tau_f = 3.77 < \tau_f$ allowable (4 MPa) safe.

Step 10: Design of key

(a) Length of key = Length of hub $l = 47$ mm

(b) Width of key, $b = \frac{2M_t}{dl\tau_k}$

$$b = \frac{2 \times 119.4 \times 10^3}{22 \times 47 \times 93} = 2.48 \simeq 3 \text{ mm}$$

(c) Thickness of key, $h = \frac{4M_t}{dl\sigma_b'} = \frac{4 \times 119.4 \times 10^3}{22 \times 47 \times 186} = 2.48 \simeq 3$ mm

Select standard key from Table 7.1

$b = 3$ mm, $h = 3$ mm.

Module 4

7a.

Given data: Inside diameter of boiler, $D_i = 2$ m = 2000 mm

Internal pressure, $p_f = 1$ MPa

Efficiency, $\eta = 0.85$

Yield stress, $\sigma_y = 353$ MPa

FOS = 3 for tension

= 5 for shear and

= 2 for crushing

$$\sigma_\theta = \frac{\sigma_y}{FOS} = \frac{353}{3} = 117.67 \text{ MPa}$$

$$\tau = \frac{\sigma_y}{FOS} = \frac{353}{5} = 70.6 \text{ MPa}$$

$$\sigma_c = \frac{\sigma_y}{FOS} = \frac{353}{2} = 176.5 \text{ MPa.}$$

Triple riveted double cover butt joint with equal covers and pitch of the outer rwo is twice the pitch of inner rows.

The joint is shown in Figure 11.25.

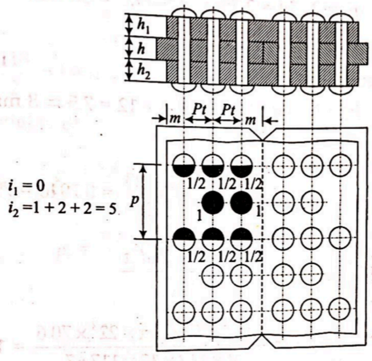


Fig. 11.25

From Fig. $i_1 = 0$ (no signal shear)

$i_2 = 1 + 2 + 2 = 5$ rivet/pitch

Step 1: Thickness of the plate $h = \dfrac{P_f D_i}{2\eta\sigma_\theta} + C$

Assume corrosion allowance, $C = 2$ mm

$$h = \frac{1 \times 2000}{2 \times 0.85 \times 117.67} + 2 = 11.998 \simeq 12 \text{ mm}$$

Step 2: Diameter of the rivet

$$d = 6\sqrt{h} \text{ to } 6.3\sqrt{h}$$

$$d = 6\sqrt{12} \text{ to } 6.3\sqrt{12}$$

$$= 20.75 \text{ to } 21.82.$$

From Table 11.1, select standard diameter of the rivet, $d = 22$ mm and rivet hole diameter, $d_h = 23$ mm.

Step 3: Pitch

(a) $p = \dfrac{(i_1 + 1.875 i_2)\pi d^2 \tau}{4h\sigma_\theta} + d_h = \dfrac{(0 + 1.875 \times 5)\pi \times 22^2 \times 70.6}{4 \times 12 \times 117.67} + 23 = 201.18$ mm

(b) $p = Kh + 40$ for double cover butt joint with 5 rivets/pitch,

From Table 11.2, $K = 6$,

$$p = 6 \times 12 + 40 = 112 \text{ mm}$$

$\therefore$ Pitch, $p = 112$ mm

Step 4: Transverse pitch, $p_t = 2.25d = 2.25 \times 22 = 49.5 \simeq 50$ mm

Step 5: Margin $m = 1.5d = 1.5 \times 22 = 33$ mm

Step 6: Thickness of cover plates,

For equal covers, $h_1 = h_2 = 0.625h = 0.625 \times 12 = 7.5 \simeq 8$ mm

Step 7: Efficiency

(a) For the plate, $\eta_p = \dfrac{p - d_h}{p} = \dfrac{112 - 23}{112} = 0.7946 = 79.46\%$

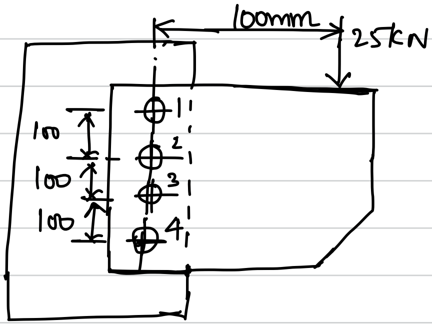
(b) For the rivet in shear, $\eta_\tau = \dfrac{(i_1 + 1.875 i_2)\pi d^2 \tau}{4ph\sigma_\theta}$

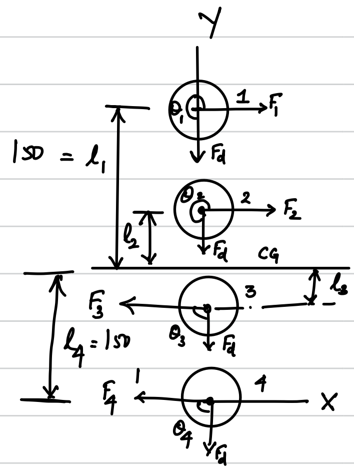
$$= \frac{(0 + 1.875 \times 5)\pi \times 22^2 \times 70.6}{4 \times 112 \times 12 \times 117.67} = 1.5909 = 159.09\%$$

(c) For the rivet in crushing, $\eta_c = \dfrac{\left(i_2 + i_1 \dfrac{h_2}{h}\right)\sigma_c}{\left(i_2 + i_1 \dfrac{h_2}{h}\right)\sigma_c + \sigma_\theta}$

$$\eta_c = \frac{(5 + 0)176.5}{(5 + 0)176.5 + 117.67} = 0.8824 \simeq 88.24\%$$

7c)





1. CG of the rivets

$$\bar{x} = \frac{x_1 + x_2 + x_3 + x_4}{\text{No. of rivets}} = 0$$

$$\bar{y} = \frac{y_1 + y_2 + y_3 + y_4}{\text{No. of rivets}} = \frac{300 + 200 + 100 + 0}{4} = 150\text{ mm}$$

2. To Find $l_{1,2,3,4}$ and $\cos\theta_{1,2,3,4}$ and 'e'

$l_1 = 150mm$, $l_2 = 50mm$, $l_3 = 50mm$, $l_4 = 150mm$

$\theta_1 = 90°$, $\theta_2 = 90°$, $\theta_3 = 90°$ (or) $270°$, $\theta_4 = 90°$ (or) $270°$

$e = 100mm$ (From Figure)

3. Direct shear load on each rivet

$$F_d = \frac{\text{Total load}}{\text{No. of rivets}} = \frac{25 \times 10^3}{4} = 6250N$$

4. Secondary shear load on rivets

$$F_1 = \frac{F \cdot e \cdot l_1}{l_1^2 + l_2^2 + l_3^2 + l_4^2} = \frac{(25\times10^3)(100)(150)}{(150)^2 + (50)^2 + (50)^2 + (150)^2} = 7500\text{ N} = F_4$$

$$F_2 = F_1 \cdot \frac{l_2}{l_1} = 2500N = F_3$$

5. Resultant shear load $\{ \because F_1 > F_3 \}$

$$F_R = \sqrt{F_1^2 + F_d^2 + 2F_1F_d\cos\theta_1}$$

$$F_R = 9762.81N$$

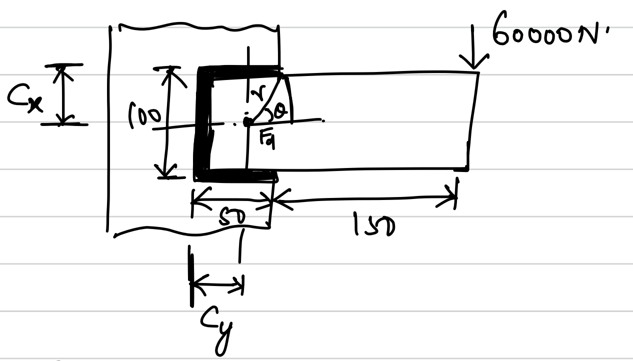
6. Diameter of Rivet

$$F_{R_{max}} = \frac{\pi}{4} d^2 \times \tau$$

$$\Rightarrow d = \sqrt{\frac{4 \times F_{R_{max}}}{\pi \times \tau}} = 14.39mm$$

standard dia $d = 15mm$

8a) $F = 60000\,N$; $\tau = 100\,N/mm^2$



(i) CG of weld

$C_x = \frac{100}{2} = 50$ mm

$C_y = \frac{b^2}{2b+d} = \frac{50^2}{2(50)+100} = 12.5$ mm

2) Find r, e and Cos θ.

$e = 150 + (50 - 12.5) = 187.5$ mm

$r = \sqrt{x^2 + y^2} = \sqrt{50^2 + 37.5^2} = 62.5$ mm

$\cos\theta = \frac{x}{r} = \frac{37.5}{62.5} = 0.6$

3) Direct shear load per unit length of the weld

$P_d = \frac{P}{l} = \frac{60000}{(50+50+100)} = 300\ N/mm$

4) Load due to torsional moment per unit length of weld

$P_n = \frac{P \cdot e \cdot r}{J}$

$J = J_w = \frac{(2b+d)^3}{12} - \frac{b^2(b+d)^2}{2b+d}$

$= 385416.67\ mm^3$

$P_n = \frac{60000 \times 187.5 \times 62.5}{385416.67}$

$P_n = 1824.32\ N/mm$

5) Resultant load per unit length of weld

$P_R = \sqrt{P_d^2 + P_n^2 + 2P_d P_n \cos\theta}$

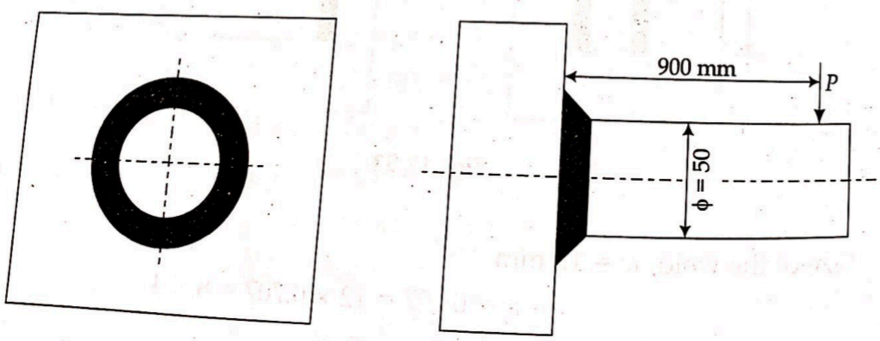
$= 2018.64\ N/mm$

6) Size of the weld

$\tau = \frac{P_R}{0.707\,w\,l} \Rightarrow 100 = \frac{2018.64}{0.707 \times w \times 1} \Rightarrow w = 28.55\,mm$

$w \simeq 30\,mm$

8b.



Given data: Size of weld $w = 10$ mm

∴ $t = w \times 0.707 = 10 \times 0.707$ mm

$e = 900$ mm

$d = 50$ mm

$\tau = 50$ MPa

length of the weld $= \pi d = \pi(50)$

Step 1: Direct load/unit length of weld

$$P_d = \frac{P}{l} = \frac{P}{\pi d}$$

$$P_d = \frac{P}{\pi \times 50} = 6.366 \times 10^{-3} \text{ P}$$

Step 2: Normal load/unit length of the weld

$$P_n = \frac{Pey}{I} = \frac{P.e}{z}$$

Ref. Fig. 12, Table 12.3,

$$z = \frac{\pi d^2}{4} = \frac{\pi \times 50^2}{4} = 1963.5 \text{ mm}^3/\text{mm}$$

∴ $$P_n = \frac{P \times 900}{1963.5} = 0.4584P$$

Step 3: Resultant load/unit length of weld

$$P_R = \sqrt{P_d^2 + P_n^2} = \sqrt{(6.366 \times 10^{-3} P)^2 + (0.4584 P)^2}$$

$$P_R = 0.4584 \text{ P}$$

Step 4: Thickness of the weld $t = \frac{P_R}{\text{stress}}$

$$7.07 = \frac{0.4584P}{50}$$

∴ Load capacity, $P = 771.16$ N.

9a) Cotter joint

$F = 50000N$; $(\sigma_t)_{rod} = 67 N/mm^2$, $\sigma_c = 134 N/mm^2$

$(\sigma_t)_{Cotter} = 100 N/mm^2$

1. Design of rod

Axial stress in rod $\sigma = \frac{4F}{\pi d^2}$

$$67 = \frac{4\times 50000}{\pi d^2}$$

$\Rightarrow d = 30.83mm \simeq 32mm$

2. Design of Cotter and Enlarged end of rod

(i) Crushing strength, $F = d_1 t \sigma_c$ (17.69)

$50000 = (d_1 t)(134)$

$d_1 t = 373.13$

(ii) Axial stress along slot of rod

$$\sigma = \frac{4F}{\pi d_1^2 - 4d_1 t}$$

$$67 = \frac{4\times 50000}{\pi d_1^2 - 4(373.13)}$$

$$\pi d_1^2 = \frac{4\times 50000}{67} + 4(373.13)$$

$\Rightarrow d_1 = 37.76 mm \simeq 38 mm$

dia of spigot.

Also, $d_1 t = 373.13$

$$t = \frac{373.13}{38} = 9.819 \simeq 10mm$$

$t = 10mm \leftarrow$ Thickness of cotter.

(iii) Shear strength of cotter

$F = 2bt\tau$

$50000 = 2\times b\times 10\times 50$ $\{\tau = 0.5\times 100 = 50 N/mm^2\}$

$\Rightarrow b = 50mm$

(Mean width of cotter)

3. Design of Spigot collar

(i) Bearing stress $\sigma_c = \frac{4F}{\pi(d_2^2 - d_1^2)}$

$$134 = \frac{4\times 50000}{\pi(d_2^2 - 38^2)}$$

$$\Rightarrow d_2 = \sqrt{\left(\frac{4\times 50000}{134\pi}\right) + 38^2}$$

$d_2 = 43.8 \simeq 44mm$

(ii) Shear stress induced in collar

$$\tau = \frac{F}{\pi d_1 e}$$

$$50 = \frac{50000}{\pi(38)e} \Rightarrow e = 8.38mm$$

$e \simeq 9mm$

4) Design of socket

(i) Tensile stress in socket $\sigma = \frac{4F}{\pi(d_3^2 - d_1^2) + 4t(d_3 - d_1)}$

$$67 = \frac{4\times 50000}{\pi(d_3^2 - 38^2) + 4(10)(d_3 - 38)}$$

$$(d_3^2 - 38^2) + \frac{40}{\pi}(d_3 - 38) = \frac{4\times 50000}{67}\times\frac{1}{\pi}$$

$$d_3^2 + 12.73(d_3 - 38) = 950.67$$

$$d_3^2 + 12.73 d_3 - 1434.41 = 0$$

$d_3 = 32.03 \simeq 32mm$

(ii) $\sigma_c = \frac{F}{(d_4 - d_1)t} \Rightarrow d_4 = \frac{50000}{137\times 10} + 38$

$\Rightarrow d_4 = 74.49 \simeq 75mm$

(iii) $\tau = \frac{F}{2c(d_4 - d_1)} \Rightarrow c = \frac{50000}{2\times 50(75-38)}$

$c = 13.51 \simeq 14mm$

9b. **Self locking and over hauling of screws**

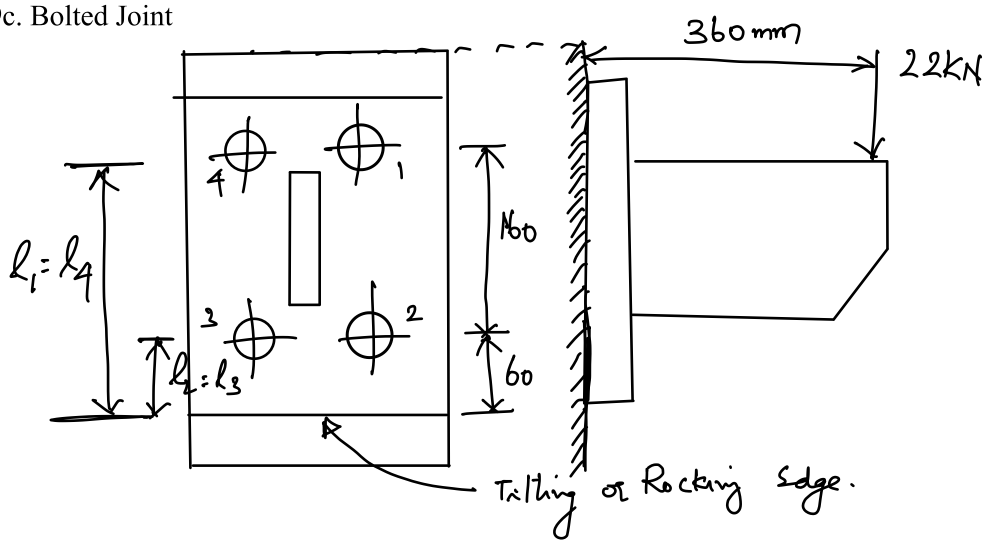
The torque required to lower the load

$$M_{t_s} \quad = \quad W.\frac{d_2}{2}.tan\ (\phi - \alpha)$$

In the above expression., if $\phi < \alpha$, then the torque required to lower the load will be negative i.e the load will start moving downward without the application of any torque, such a condition is known as overhauling of screws.

If $\phi > \alpha$, then the torque required to lower the load will be positive i.e., an effort is applied to lower the load, such a screw is known as self locking screw.

9c. Bolted Joint



$\sigma_{t_{all}} = 90 MPa$

$\tau_{all} = 52 MPa.$

$F = 22 \times 10^3 N$

$l_1 = l_4 = 220 mm$; $l_2 = l_3 = 60 mm$; $i = 4$ (No: of Bolts)

Assume there are no pre-tension on bolts.

Direct shear load on each bolt $F_b = \frac{F}{i} = \frac{22\times10^3}{4} = 5.5\times10^3 N$

The maximum load occurs on the bolts 1 and 4 as they are far from the tilting edge.

Normal load on bolt 1 $= F_1 = \frac{F.l.l_1}{l_1^2+l_2^2+l_3^2+l_4^2}$

$\Rightarrow F_1 = \frac{22\times10^3 \times 220 \times 160}{220^2+60^2+220^2+60^2}$

$F_1 = 7446.15 N$

According to max'm shear stress theory

Max'm shear load on bolt 1

$F_\tau = \frac{1}{2}\sqrt{F_1^2+4F_b^2} = \frac{1}{2}\sqrt{7446.15^2+4(5500)^2}$

$F_\tau = 6641.63 N$

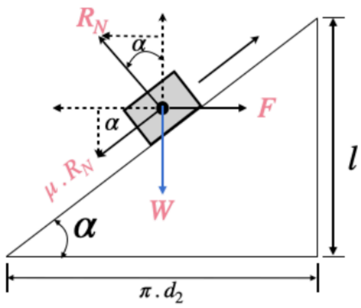
∴ Shear stress area of bolt,

$A_c = \frac{F_\tau}{\tau} = \frac{6641.63}{52} = 127.72 mm^2$

From Table 18.7, Nearest core area $A_c = 157 mm^2$

Selected bolt = M16

10a.



If one complete turn of the screw three be, unwound from the body and developed, it will form an inclined plane as shown.

$$tan\alpha = \frac{l}{\pi . d_2}$$

Since the working principle of screw jack is similar to that of an inclined plane., From above figure.,

R_N = Normal reaction
μR_N = Frictional force

For equilibrium of horizontal forces.,

$$F = \mu R_N \ cos\alpha \ + \ R_N \ sin\alpha \quad \text{———————> (i)}$$

For equilibrium of vertical forces.,

$$W = R_N cos\alpha \ - \ \mu R_N \ sin\alpha \quad \text{———————> (ii)}$$

Equation $\frac{(1)}{(2)}$.,

$$\frac{F}{W} = \frac{\mu cos\alpha \ + sin\alpha}{cos\alpha \ - \ \mu sin\alpha}$$

$$F = W . \left[\frac{\mu cos\alpha \ + sin\alpha}{cos\alpha \ - \mu sin\alpha}\right]$$

$$F = W\left[\frac{\mu \ + \ tan\alpha}{1 \ - \mu tan\alpha}\right] \qquad \text{(Equation 18.27 DDB)}$$

Torque required to raise the load.,

$$M_{t_s} \ = \ F . \frac{d_2}{2} = W . \frac{d_2}{2}\left(\frac{\mu \ + \ tan\alpha}{1 \ - \ \mu tan\alpha}\right)$$

Also., $\mu \ = \ tan\phi$

$$\therefore M_{t_s} \ = \ W\frac{d_2}{2}\left[\frac{tan\phi \ + \ tan\alpha}{1 \ - \ tan\phi tan\alpha}\right]$$

$$M_{t_s} \ = \ W . \frac{d_2}{2} . tan \ (\phi + \alpha)$$

10b. Applications of Knuckle joint

1. The joint between the tie rod joint of a roof truss.
2. The joint Tension link in bridge structure.
3. Tie rod joint of the jib crane.
4. Link of roller chain, bicycle chain, and Chain straps of watches.

10c.

Solution:

To calculate power required (N)

Torque $M_t = 9550 \times \frac{N}{n}$ N-m

Therefore, $N = \frac{M_t n}{9550}$

Where M_t is in N-m

n = *Speed in rpm*

N = *Power in kW*

For square threads, torque required to raise load W is given by

$$M_t = W\left[\frac{d_2}{2}\left(\frac{\tan\alpha + \mu}{1-\mu\tan\alpha}\right) + \frac{\mu_c d_c}{2}\right]$$

Where $W = load = 20\text{ kN} = 20\times 10^3\text{ N}$

d_2 = *Mean diameter of screw* $= \frac{d + d_1}{2}$

For square threads,

$d = d_1 + p$

i.e., $50 = d_1 + 8$

$d_1 = 42$ mm

Therefore, Mean diameter

$$d_2 = \frac{50+42}{2} = 46\text{ mm}$$

$$\tan\alpha = \frac{lead}{\pi d_2}$$

Lead = pitch (since single start thread) = 8 mm

Therefore, $\tan\alpha = \frac{8}{\pi \times 46} = 0.05536$

μ = *Coefficient of thread friction* = 0.2

μ_c = *Coefficient of collar friction* = 0.1

d_c = *Mean diameter of collar*

$$= \frac{d_c)_{\text{outside}} + d_c)_{\text{inside}}}{2}$$

$$= \frac{60+30}{2} = 45\text{ mm}$$

Therefore, $M_t = 20\times 10^3\left[\frac{46}{2}\left(\frac{0.055356+0.2}{1-0.2\times 0.05536}\right) + \frac{0.1\times 45}{2}\right]$

$= 163780$ N-mm

$= 163.78$ N-m