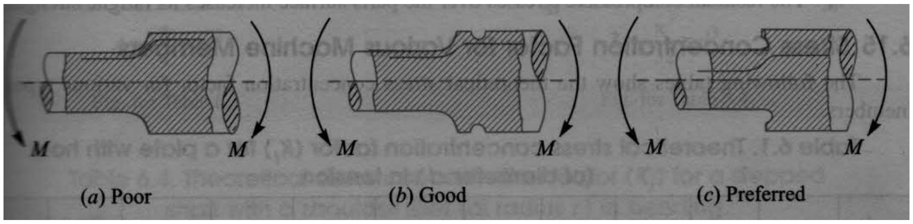
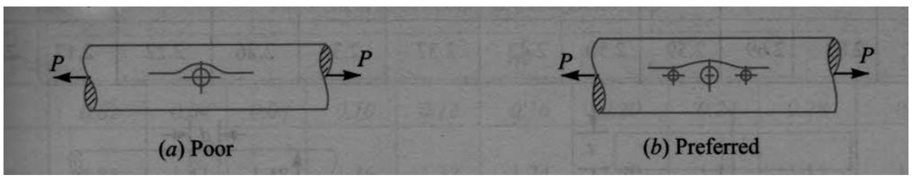
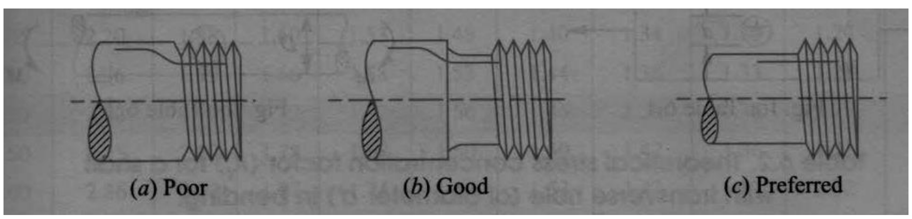
WINTER– 16 EXAMINATION
Model Answer

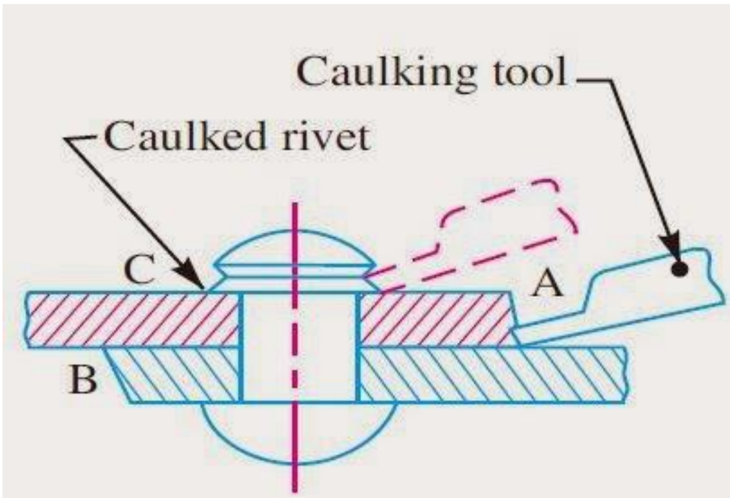
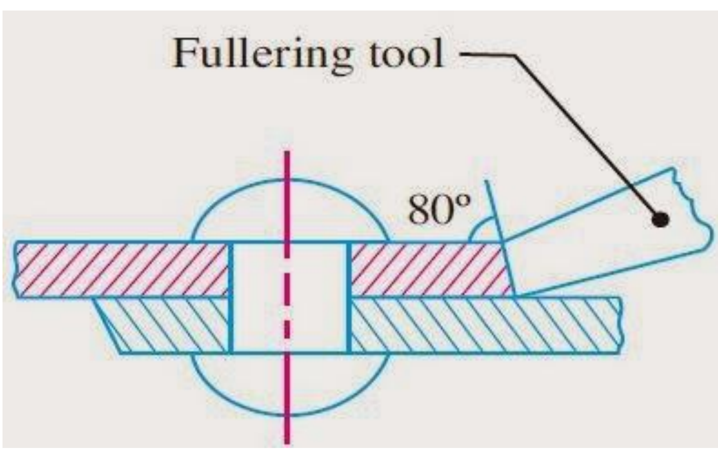
Subject Code: **17553**

Important Instructions to examiners:

1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling errors should not be given more Importance (Not applicable for subject English and Communication Skills.
4) While assessing figures, examiner may give credit for principal components indicated in the figure. The figures drawn by candidate and model answer may vary. The examiner may give credit for any equivalent figure drawn.
5) Credits may be given step wise for numerical problems. In some cases, the assumed constant values may vary and there may be some difference in the candidate's answers and model answer.
6) In case of some questions credit may be given by judgement on part of examiner of relevant answer based on candidate's understanding.
7) For programming language papers, credit may be given to any other program based on equivalent concept.

Q. No.	Sub Q. N.	Answer	Marking Scheme
1		**Attempt any FIVE of the following:**	**5X4=20**
	a)	Design data book:- When a designer wants to design & develop a product he requires lot of information such as material specifications, physical & mechanical properties of materials, standards, different manufacturing processes, empirical relations, tool data etc. so a book which contains all this information is called data book. **Use:-** 1) The standard dimensions of nut, bolts, shafts, bearings etc. are available in the design data book. 2) Use of design data book makes easy for a designer engineer to collect the above data from a single book. 3) Also it saves the time of a designer, so the design process becomes simple & easy	2m 2m (any 2)
	b)	**Causes:- It may occurs due to-** 1) Change in cross section such as stepped axle, grooves, keyways, threaded holes etc. 2) Concentrated load applied at minimum areas of machine parts such as contact between gear teeth. 3) Variation in mechanical properties of materials from point to point due to cavities, cracks etc.	2m (any2)

		4) Surface irregularities or poor surface finish. **Remedies:-** 1) By fillets, undercutting & notches  2) Additional notches & holes  3) Reducing stress concentration in threaded members  	2m (any2)
	c)	•Keyway is a slot machined either on the shaft or in the hub to accommodate the key. • It is cut by vertical or horizontal milling cutter. • The keyway cut into the shaft reduces the load carrying capacity of shaft. • This is due to stress concentration near the comers of the keyway and reduction in the cross sectional area of shaft. • In other words, the torsional strength of shaft is reduced. • The following relation of reduction factor is used to analyze the weakening effect of keyway is given by H. F. Moore. $e = 1 - 0.2 (w/d) - 1.1(h/d)$ Where, e = shaft strength factor = Strength of shaft with keyway/Strength Of shaft Without keyway w = Width of keyway, d = Diameter of shaft h = Depth of keyway = 112 x thickness of key = 1/2 x t	4m

		• It is usually assumed that strength of keyed shaft is 75% of solid shaft. • Thus, after finding out dimensions of key, the reduction factor 'e' is Calculated and for safe design, its value should be less than 0.75.	
	d)	Caulking:-  In order to make the joints leak proof or fluid tight in pressure vessels like steam boilers, air receivers and tanks etc. a process known as **caulking** is employed. In this process, a narrow bunt tool called caulking tool about 5 mm thick and 38 mm in breadth is used. The edge of the tool is ground to an angle of 80°. **Fullering:-**  A more satisfactory way of making the joints staunch is known as **fullering** which has largely superseded caulking. In this case, a fullering tool with a thickness at the end equal to that of the plate is used in such a way that the greatest pressure due to the blows occur near the joint, giving a clean finish, with less risk of damaging the plate.	2m 2m
	e)	following stresses are induced in a bolt, screw or stud when it is screwed up tightly	4m

1. Tensile stress due to stretching bolt

Since none of the above mentioned stresses are accurately determined, therefore bolts are designed on the basis of direct tensile stress with a large

Factor of safety in order to account for the indeterminate stresses. The initial tension in a bolt, based on experiments, may be found by the relation Pi = 2840dN

Pi = Initial tension in a bolt, and

d = Nominal diameter of bolt, in mm.

2.Torsional shear stress caused by the frictional resistance of the threads during its tightening

The torsional shear stress caused by frictional resistance of the threads during its tightening may be obtained by using the torsion equation. We know that

$T/J = T_s/r$

$T_s = T/J \times r = \{ T/(\pi/32) \times d_c^4 \} \times \{ d_c /2\} = 16\ T/ \pi(d_c)^3$

Where T_s = Torsional shear stress,

T = Torque applied, and

d_c= Minor or core diameter of thread

3.Shear stress across the threads. The average thread shearing stress for the screw(T_s) is obtained by using the relation:

$T_s = p/(\pi\ d_c \times b \times n)$

Where b = Width of the thread section at the root.

The average thread shearing stress for the nut is

$T_n = p/(\pi\ d \times b \times n)$

Where d = Major diameter.

4. Compression or crushing Stress on threads. The compression or crushing stress between the threads (σ_c) may be obtained by using the relation :

$\sigma_c = p/ \pi[d^2 – (d_c)^2]n$

Where d = Major diameter,

d_c = Minor diameter, and

n = Number of threads in engagement.

5. Bending stress if the surfaces under the head or nut are not perfectly parallel to the bolt axis.When the outside surfaces of the parts to be connected are not parallel to each other, then

		the bolt will be subjected to bending action. The bending stress (σ_b) induced in the shank of the bolt is given by σ_b = x.E/2l where where x = Difference in height between the extreme corners of the nut or head, I = Length of the shank of the bolt, and E = Young's modulus for the material of the bolt.	
	f)	**Advantages:-** 1. The welded structures are usually lighter than riveted structures. This is due to the reason that in welding, gussets or other connecting components are not used. 2. The welded joints provide maximum efficiency (may be 100%) which is not possible in case of riveted joints. 3. Alterations and additions can be easily made in the existing structures 4. As the welded structure is smooth in appearance, therefore it looks pleasing. 5. In welded connections, the tension members are not weakened as in the case of riveted joints. 6. A welded joint has a great strength. Often a welded joint has the strength of the parent metal itself. 7. Sometimes, the members are of such a shape (i.e. circular steel pipe) that they afford difficulty for riveting. But they can be easily welded. 8. The welding provides very rigid joints. This is in line with the modern trend of providing rigid frames. 9. It is possible to weld any part of a structure at any point. But riveting requires enough clearance. 10. The process of welding takes less time than the riveting. **Disadvantages:-** 1. Since there is an uneven heating and cooling during fabrication, therefore the member may get distorted or additional stresses may develop. 2. It requires a highly skilled labour and supervision. 3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it. 4. The inspection of welding work is more difficult than riveting work.	2m (any2) 2m (any2)
	g)	**Perfect frame :** A pin-jointed frame which has got just sufficient number of members to resist the loads	2m

without undergoing appreciable deformation in shape is called rigid or perfect frame. The perfect frame obeys the following condition viz.

n = 2 j – 3

where, n= no. of links and j= no. of joints

Deficient frame:

A frame is said to be deficient if the number of members in it is less than that required for a perfect frame. Such frames can’t retain their shape when loaded. — 2m

2)		**Attempt any TWO of the following:**	**2x8=16**

a) — 6M (procedure)

Let

D_1 = effective diameter of circle touching bolt holes

d_1 = diameter of bolt hole

D_p = pitch circle diameter

$$D_1 = D_p - d_1$$

∴ Force trying to seperate two flanges

$$F = \frac{\pi}{4}(D_1)^2 \cdot p \quad \ldots (1)$$

Let

n = number of bolts

d_c = Core diameter of the bolts

σ_t = Permissible stress for bolt material

∴ Resistance of tearing bolts

$$= \frac{\pi}{4}(d_c)^2 \sigma_t \times n \quad \ldots (2)$$

the value of n may be obtained by equating (1) & (2)

The circumferencial pitch of the bolts

$$P_c = \frac{\pi D_p}{n}$$

Following emperical relations

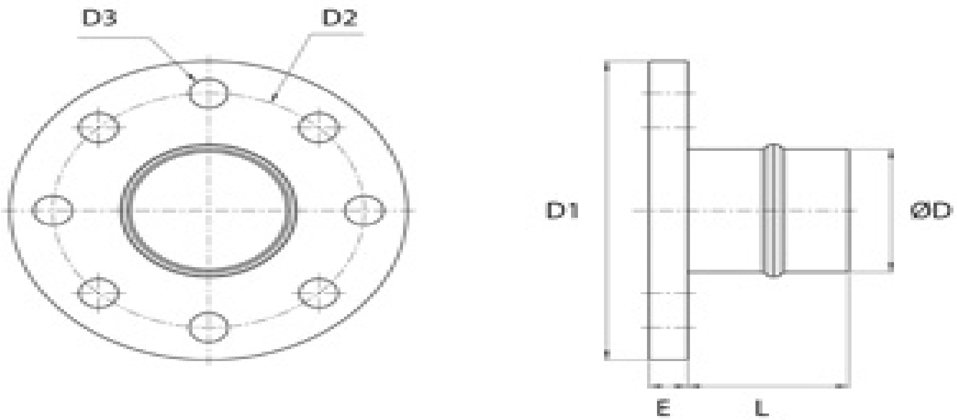
Nominal dia of bolts $d = 0.75t + 10$ mm

No of bolts $n = 0.0275D + 1.6$

Thickness of flange $t_f = 1.5t + 3$ mm

width of flange $B = 2.3d$

outside dia of flange $D_o = D + 2t + 2B$ mm

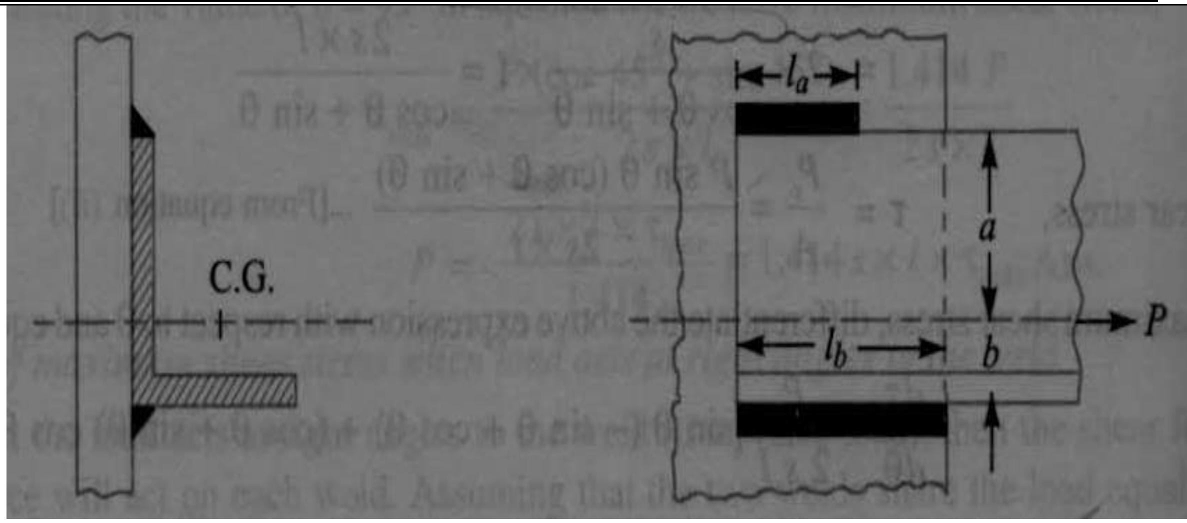
			2m dia
	b)	**The general procedure of machine design is as follows:** 1. Recognition of need: First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed. 2. Synthesis (Mechanisms): Select the possible mechanism or group of mechanisms 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 judgement to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost. 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.	4m
		Velocity Factor:- This factor is considered in design of gears . It depends upon velocity of operating geras It is represented by $C_{v.}$ Permissible working stress is obtained as follows. $\sigma_{wp=} \sigma_{p\ x}\ C_v$ Where, $\sigma_{wp\ =}$ permissible working stress $\sigma_{p=}$ permissible stress	2m

		$c_{v=}$ Velocity factor **Service Factor:-** It is the ratio of maximum load (torque) to the average load (torque). T_{max} = Service factor x $T_{average}$	2m
	c)	Given $d = 50\ mm$ $\tau = 42\ N/mm^2$ $\sigma_{ck} = 70\ N/mm^2$ 1) To Find Torque Transmitted by shaft $T = \frac{\pi}{16} \tau d^3$ $= \frac{\pi}{16} \times 42 \times 50^3$ $T = 1.03 \times 10^6\ N.mm$	2m
		2) Thickness of key $= t = \frac{d}{6} = \frac{50}{6} = 8.33 = 10\ mm$ width of key $= w = \frac{d}{4} = \frac{50}{4} = 12.5 = 14\ mm$ $\therefore$ $t = 10\ mm$, $w = 14\ mm$	2m
		3) Considering shear failure of key. $T = l \times w \times \tau \times \frac{d}{2}$ $1.03 \times 10^6 = l \times 14 \times 42 \times \frac{50}{2}$ $l = 70.06\ mm \approx 72\ mm$	2m
		4) Considering crushing failure of key $T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ $1.03 \times 10^6 = l \times \frac{10}{2} \times 70 \times \frac{50}{2}$ $l = 117.7 \approx 120\ mm$ Taking larger value $\therefore$ $l = 120\ mm$	2m

3)		**Attempt any TWO of the following:**	**2x8=16**
	a)	Given $T = 200 N.m = 200 \times 10^3 N.mm$ $M = 350 N.m = 350 \times 10^3 N.mm$ $\tau_{yield} = 300 N/mm^2$ FOS = 3 $K_m = 2 \quad K_t = 1.5$	Given 1m
		1) To Find working Shear Stress $\tau = \frac{\tau_{yield}}{FOS} = \frac{300}{3}$ $\tau = 100 N/mm^2$	2m
		2) To Find Equivalant Twisting Moment $T_e = \sqrt{K_m . M^2 + K_t . T^2}$ $= \sqrt{2 \times (350 \times 10^3)^2 + 1.5 \times (200 \times 10^3)^2}$ $= \sqrt{(2.45 \times 10^{11}) + (6 \times 10^{10})}$ $T_e = 552.26 \times 10^3 N.mm$	3m
		We also know that $T_e = \frac{\pi}{16} \tau d^3$ $552.26 \times 10^3 = \frac{\pi}{16} 100 \times d^3$ $\therefore d = 30.41 mm$ $d \approx 35 mm$	for dia -2m

	b)	d = diameter of shaft D = diameter of hub = 2d d_1 = Nominal dia of bolt D_1 = diameter of bolt circle = 3d n = no of bolts t_f = thickness of flange = 0.5d $\tau_s, \tau_b, \& \tau_k$ = Allowable shear stresses for shaft, bolt & key. τ_c = Allowable shear stress for flange material $\sigma_{ck} \& \sigma_{cb}$ = Allowable crushing stress for bolt & key. 1) <u>Design of hub</u> $T = \frac{\pi}{16} \tau_c D^3 (1 - K^4) \quad \cdots \text{①}$ where $K = \frac{D}{d}$ here D = 2d & L = 1.5d. From equation ① the diameter of hub can be checked If $\tau_c < \tau_{given}$ design is safe 2) <u>Design of key</u> $w = \frac{d}{4}$ $t = \frac{d}{6}$ $l = L = 1.5d$. 3) <u>Design for flange</u> $T = \pi \times D \times t_f \times \tau_c \times \frac{D}{2}$ here $t_f = 0.5d$. In above equation if $\tau_c < \tau_{given}$ design safe 4) <u>Design of bolts</u> load on each bolt = $\frac{\pi}{4} \times (d_1)^2 \times \tau_b$ ∴ Total load on bolts = $n \times \frac{\pi}{4} (d_1)^2 \times \tau_b$ ∴ Torque transmitted $T = n \times \frac{\pi}{4} (d_1)^2 \times \tau_b$ From above equation d_1 can be calculated. checking of bolt under crushing. $T = n \times d_1 \times t_f \times \sigma_{cb} \times \frac{D_1}{2}$ If $\sigma_{cb} < \sigma_{cb\,given}$ design is safe	Design of hub 2m Design of key 2m Design of flange 2m Design of bolts 2m

c) | Fig 2m



Derivation 6m

Sometimes unsymmetrical sections such as angles, channels-sections etc., welded on the flange edges are loaded axially as shown in Fig. In such cases, the lengths of weld should be proportioned in such a way that the sum of resisting moments of the welds about the gravity axis is zero. Consider an angle section as shown in Fig.

Let l_a = Length of weld at the top,
l_b = Length of weld at the bottom,
l = Total length of weld = $l_a + l_b$
P = Axial load,
a = Distance of top weld from gravity axis,
b = Distance of bottom weld from gravity axis, and
f = Resistance offered by the weld per unit length
Moment of the top weld about gravity axis
= l_a x f x a
and moment of the bottom weld about gravity axis
= l_b x f x b
Since the sum of the moments of the weld about the gravity axis must be zero, therefore,
l_a x f x a - l_b x f x b
or l_a X a = I_b x b ……..(i)
We know that
l = $l_a + l_b$ …….(ii)
From equations (i) and (ii), we have
l_a= l x b /a + b and l_b = l x a /a + b

4)		**Attempt any TWO of the following:**	**2x8=16**
	a)	Given $w = 80\ mm$ $t = S = 10\ mm$ $\sigma_t = 70\ N/mm^2$ $\tau = 50\ N/mm^2$ $W = 55\ kN = 55 \times 10^3\ N$	Given 1m
		1) To Find length of single transverse fillet weld $l_1 = w - 12.5 = 80 - 12.5$ $l_1 = 67.5\ mm$	1m
		2) Load carried by single transverse fillet weld $W_{\sigma t} = W_1 = 0.707 \times S \times l_1 \times \sigma_t$ $= 0.707 \times 10 \times 67.5 \times 70$ $W_1 = 33.40 \times 10^3\ N$	2m
		3) Load carried by double parallel fillet weld $W_\tau = W_2 = 2 \times 0.707 \times S \times l_2 \times \tau$ $= 2 \times 0.707 \times 10 \times l_2 \times 50$ $W_2 = 707\ l_2$	2m
		We know that total load carried by plate $W = W_1 + W_2$ $55 \times 10^3 = 33.40 \times 10^3 + 707\ l_2$ $55 \times 10^3 - 33.40 \times 10^3 = 707\ l_2$ $21.6 \times 10^3 = 707\ l_2$ $\therefore l_2 = 30.55\ mm \approx 31\ mm$ For starting & stopping of weld run 12.5 mm should be added	2m

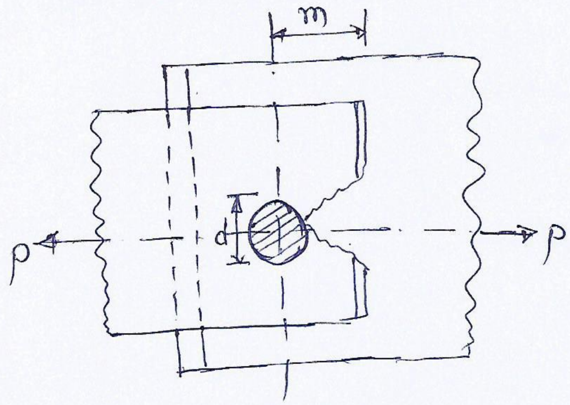
b)

Failure 6m

(any 3 types)

Failure of a Rivetted Joint:

1) Tearing of the plate at an edge:-



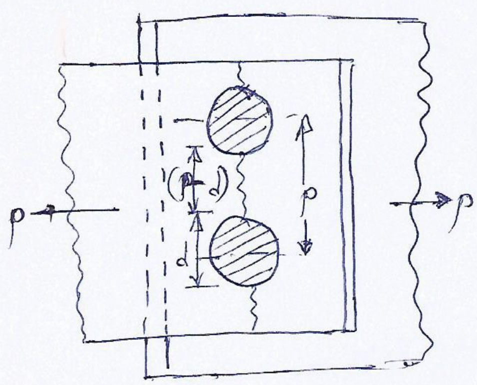
A joint may fail due to tearing of the plate at an edge shown in fig.

This can be avoided by keeping the margin

$m = 1.5d$

where d is the diameter of the rivet hole.

2) Tearing of the plate across a row of rivets:-



The resistance offred by the plate against tearing is known as tearing resistance of the plate

Let p = pitch of rivets

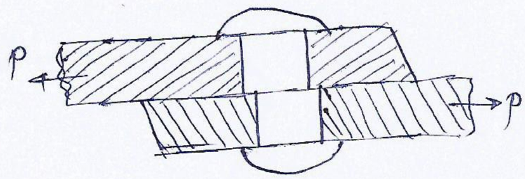
d = diameter of rivet hole

t = thickness of plate

f_t = permissible tensile stress

$\therefore P_t = (p - d) \cdot t \cdot f_t$

3) Shearing of the rivets



Let

d = diameter of the rivet hole

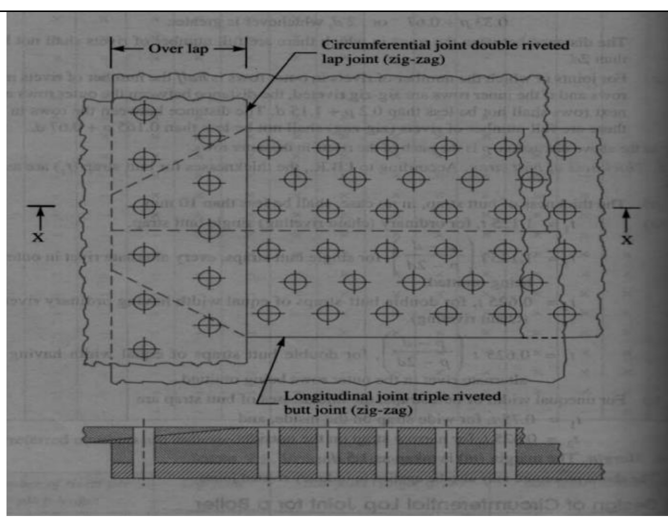
τ = Safe permissible stress

n = number of rivets

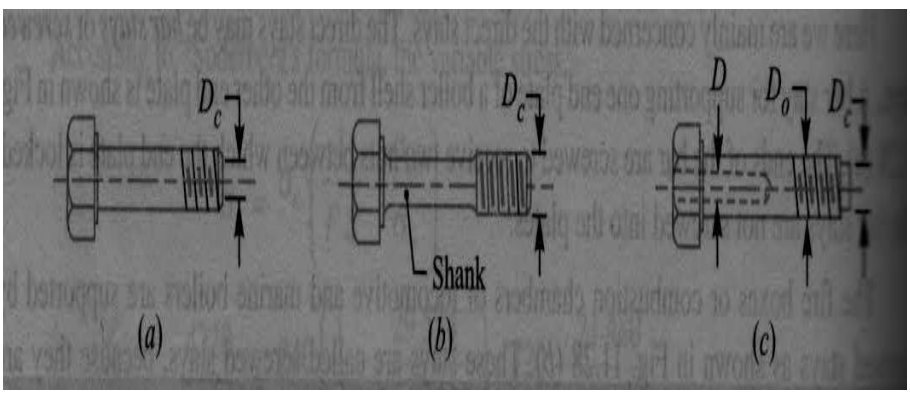
Shearing resistance offred by plate

$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$ .. Single shear

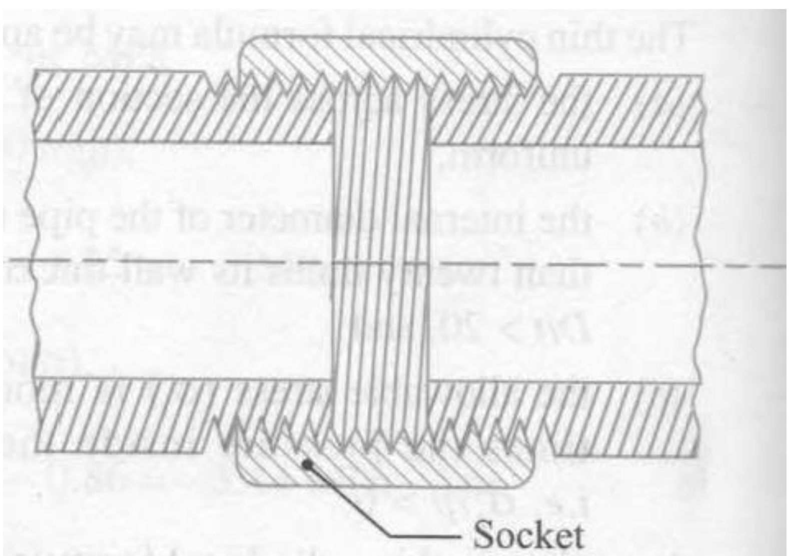
$= n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau$ -- Double shear

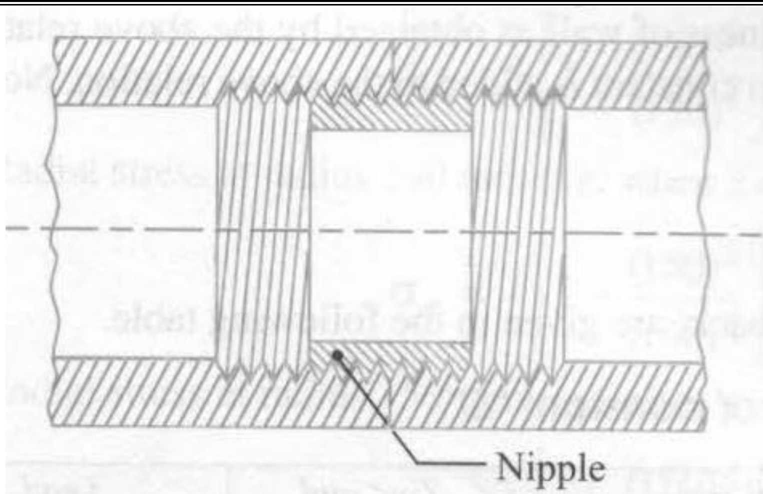
		4) Crushing of the plate :- Sometimes the rivets do not actually shear off under tensile stress but are crushed as shown. Due to this rivet becomes oval shaped & hence the joint becomes loose. ∴ Crushing resistance of the rivet let d = diameter of rivet hole t = thickness of plate σ_c = Crushing stress n = no. of rivets per pitch $\therefore P_c = n \cdot d \cdot t \cdot \sigma_c$ Efficiency of rivetted joint :- $\eta = \frac{\text{Least of } P_t, P_s \text{ and } P_c}{\text{Actual strength of Plate}}$ $\therefore \eta = \frac{\text{least of } P_t, P_s \text{ or } P_c}{p \times t \times \sigma_t}$ where p = pitch of rivet t = thickness of plate σ_t = permissible tensile stress of the plate	2m
	c)		2m (dia)

		1) Thickness of boiler shell $t = \frac{P \cdot D}{2 \cdot \sigma_t \times \eta_l} + 1\ mm$ t = thickness of boiler shell P = Steam pressure in boiler D = Internal diameter of boiler shell σ_t = Permissible tensile stress η_l = Efficiency of the joint 2) Diameter of rivets $d = 6\sqrt{t}$ -- when $t > 8$ mm If $t < 8$ m then equating $P_s = P_c$ 3) Number of Rivets $P_s = n \times \frac{\pi}{4} d^2 \times \tau$ 4) Pitch of Rivets $\eta_c = \frac{P_1 - d}{P_1}$ P_1 Can be obtained from this relationship & $P_{max} = C \times t + 41.28$ mm t = thickness of shell plate c = Constant 5) Distance between rows of rivets:- $0.33p + 0.67d$ --- Zig Zag Rivetting $2d$ ------ For chain Riveting 6) Margin:- $m = 1.5d$.	6m derivation

5)		**Attempt any TWO of the following:**	**2x8=16**
	a)	**Advantages:-** 1) Screwed joints are highly reliable in operation. 2) Screwd joints are convenient for assembling & dissembling. 3) A wide range of scred joints may be adopted to various operating conditions. 4) Screws are relatively cheap to produce. **Disadvantages**:- 1) The main disadvantage of the screwed joints is the stress concentration in the threaded portions. 2) It becomes loose because of machine vibrations. 3) Its strength can not be compared with other joints. **Bolts of Uniform strength:-** If the shank of the bolt is turned down to a diameter equal or even slightly less than the core diameter of the thread (D) as shown in Fig. (b), then shank of the bolt will undergo higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the sections near the thread. The bolt, in this way, becomes stronger and lighter and it increase shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us **bolts of uniform strength**. The resilience of a bolt may also be increased by increasing its length.  	2m (any2) 2m (any2) 4m

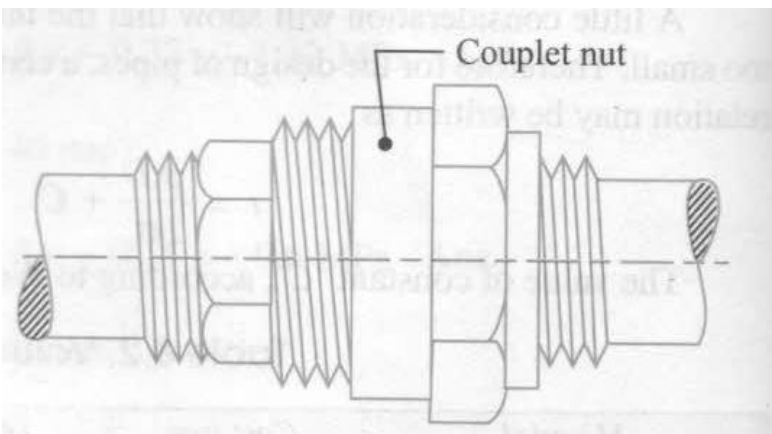
	b)	Given $d_o = 20$ mm $D = 300$ mm $p = 0.65$ N/mm^2 $\sigma_t = 20$ N/mm^2 1) Core diameter of bolt $d_c = 0.84 d_o$ $d_c = 16.8$ mm 2) Total vertical load acting on the Cylinder $W_n = p \times \frac{\pi}{4} \times D^2$ $= 0.65 \times \frac{\pi}{4} \times (300)^2$ $W_n = 45.94 \times 10^3$ N 3) Load acting on each bolt $\therefore W = \frac{W_n}{n}$ ..... eqn (1) 4) We also know that $\sigma_t = \frac{W}{\frac{\pi}{4}(d_c)^2}$ $\therefore W = \sigma_t \times \frac{\pi}{4}(d_c)^2$ $= 20 \times \frac{\pi}{4}(16.8)^2$ $W = 4.43 \times 10^3$ N ... (2) Equating eqn (1) & (2) $n = \frac{W_n}{W} = \frac{45.94 \times 10^3}{4.43 \times 10^3}$ Therefore **n= 12**	Given 1m 1m 2m 2m 2m

	c)	**Stresses in Pipes**: The stresses in pipes due to the internal fluid pressure are determined by Lame's equation. According to Lame's equation, tangential stress at any radius x $\sigma t = \{[p\,(ri)^2] / [(ro)^2 - (ri)^2]\} / \{1 + [(ro)^2 / x^2]\}$ And Radial stress at any radius x $\sigma r = \{[p\,(ri)^2] / [(ro)^2 - (ri)^2]\} / \{1 - [(ro)^2 / x^2]\}$ where p = Internal fluid pressure in the pipe, ri = Inner radius of the pipe, and ro = Outer radius of the pipe **The various types of pipe joints are as follows.** 1. Socket or a coupler joint.  2. Nipple joint.	04 marks for stresses 04 marks for sketch any 2 joints



Nipple joint.

3.Union joint.



Union joint

4. Spigot and socket joint.

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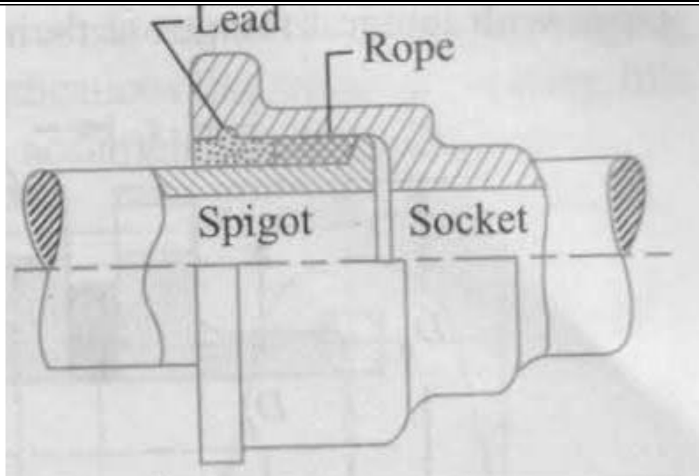
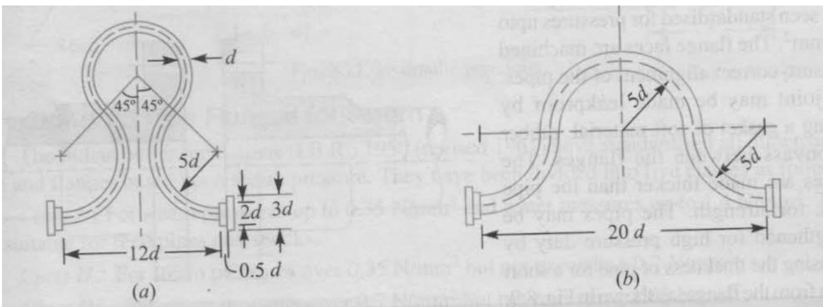
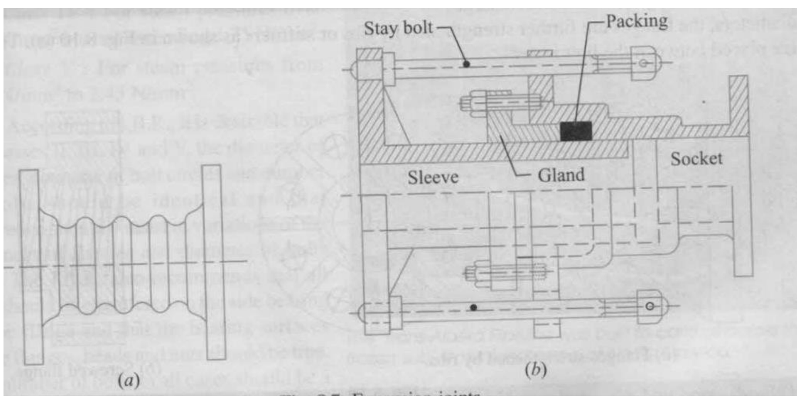


Spigot and socket joint

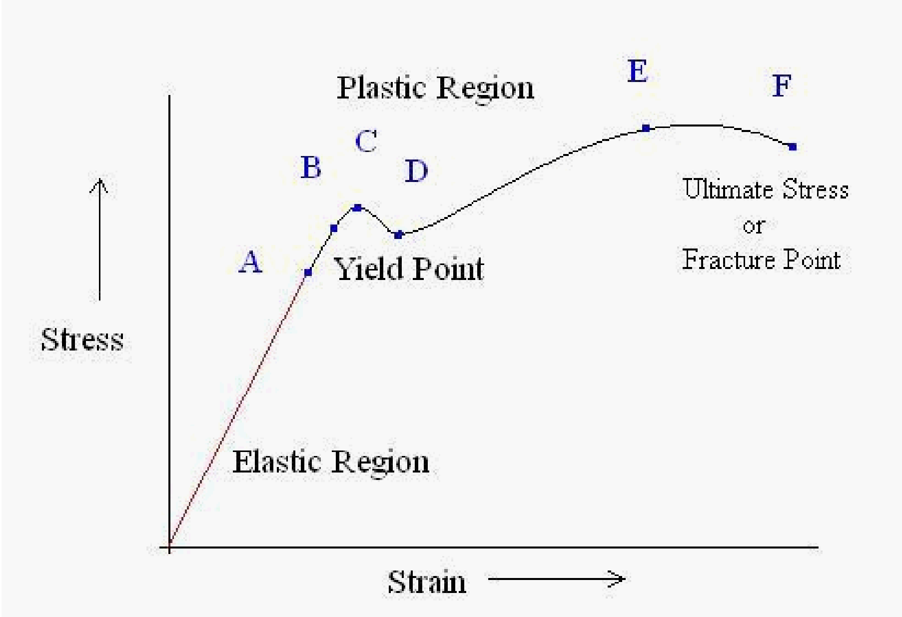
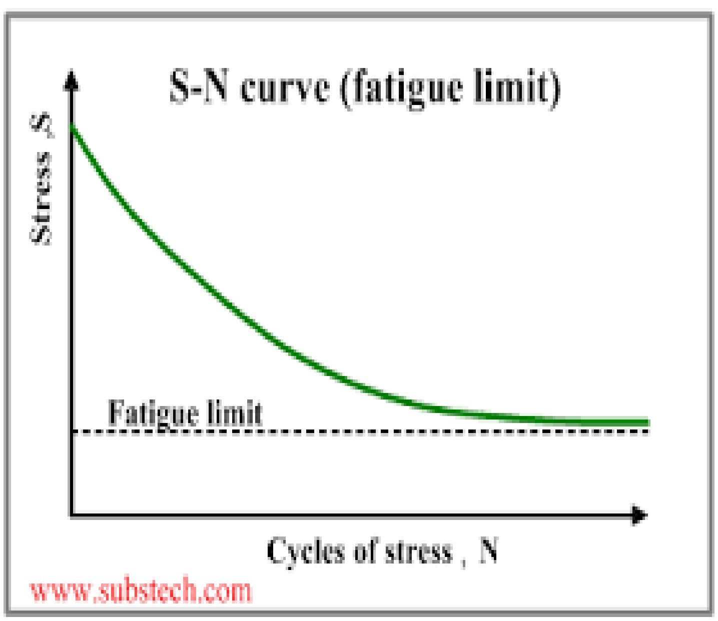
5. Expansion joint.

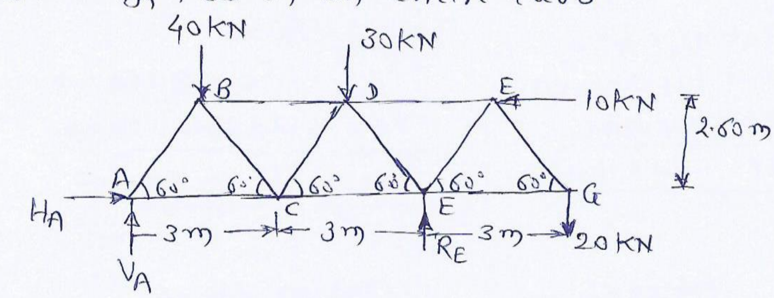
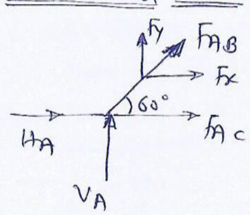
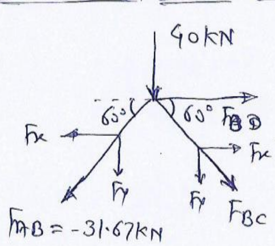


Expansion bends

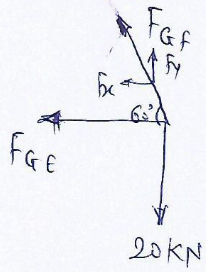


Expansion joints

6.		**Attempt any TWO of the following:**	**2x8=16**
	a) i)	 Description:- Region O-A:- Proportional Limit Point B:- Elastic Limit Point C:- Upper yield point Point D:- Lower Yield Point Point E:- Ultimate stress point Point F:- Breaking stress point	2m (dia) 2m
	a) ii)	**S-N curve:-**  Consider a standard mirror polished specimen rotating in a fatigue testing machine &	1m dia

		loaded in a bending. The specimen is subjected to completely reversed stresses. A record is kept of number of cycles required to produce a failure & results are plotted on Stress-cycle graph as shown in fig. **Endurance Limit:-** Endurance or fatigue limit is defined as the maximum value of completely reversed bending stress, which a standard specimen can withstand without failure for infinite number of cycles of loads.	1m 2m
	b)	Drawing FBD of an entire truss  Taking $\Sigma f_x = 0$ $H_A - 10 = 0$ $H_A = 10\,kN$ Taking $\Sigma f_y = 0$ $V_A - 40 - 30 - 20 + R_E = 0$ $V_A + R_E = 90\,kN$ ... (1) Taking $\Sigma M_A = 0$ $(40 \times 1.5) + (30 \times 4.5) - (6 \times R_E) - (10 \times 2.60) + (20 \times 9) = 0$ $R_E = 58.16\,kN$ From eq (1) $V_A = 31.83\,kN$ Isolating Joint A  Taking $\Sigma f_y = 0$ $F_{AB} \sin 60 + V_A = 0$ $F_{AB} \sin 60 = -31.83$ $\therefore F_{AB} = -36.75\,kN$ C Taking $\Sigma f_x = 0$ $F_{AB} \cos 60 + F_{AC} + H_A = 0$ $-36.75 \cos 60 + F_{AC} + 10 = 0$ $F_{AC} = 8.40\,kN$ T Isolating Joint B  Taking $\Sigma f_y = 0$ $-40 - F_{BC} \sin 60 - F_{AB} \sin 60 = 0$ $-40 - F_{BC} \sin 60 + 31.67 \sin 60 = 0$ $F_{BC} = -14.50\,kN$ C Taking $\Sigma f_x = 0$ $F_{AC} + F_{BC} \cos 60 - F_{AB} \cos 60 = 0$ $F_{BD} - 14.5 \cos 60 + 31.67 \cos 60 = 0$ $F_{BD} = 8.40\,kN$ C	FBD 1m Support reactions 2m Forces in any five members 5m (1m for each)

Isolating Joint G



Taking $\Sigma F_y = 0$

$20 + F_{GF}\sin 60 = 0$

$20 = -F_{GF}\sin 60$

$F_{GF} = -23.09\,kN$ (C)

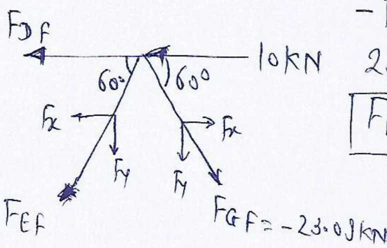
Taking $\Sigma F_x = 0$

$-F_{GE} - F_{GF}\cos 60 = 0$

$-F_{GE} + 23.09\cos 60 = 0$

$F_{GE} = -11.54\,kN$ (C)

Isolating Joint F



Taking $\Sigma F_y = 0$

$-F_{GF}\sin 60 - F_{EF}\sin 60 = 0$

$23.09\sin 60 = F_{EF}\sin 60$

$F_{EF} = 23.09\,kN$ (T)

Taking $\Sigma F_x = 0$

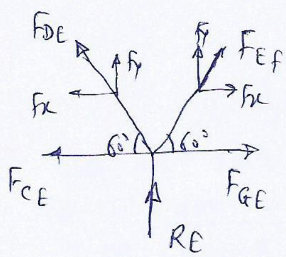
$-10 - F_{DF} - F_{EF}\cos 60 + F_{GF}\cos 60 = 0$

$-10 - F_{DF} - 23.09\cos 60 - 23.09\cos 60 = 0$

$-F_{DF} - 32.9 = 0$

$F_{DF} = -32.9\,kN$ (C)

Isolating Joint E



Taking $\Sigma F_y = 0$

$R_E + F_{DE}\sin 60 + F_{EF}\sin 60 = 0$

$58.16 + F_{DE}\sin 60 + 23.09\sin 60 = 0$

$F_{DE} = -90.29\,kN$ (C)

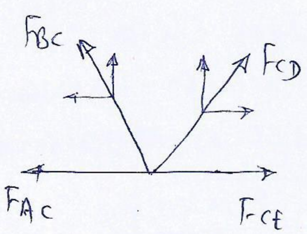
Taking $\Sigma F_x = 0$

$F_{GE} - F_{CE} - F_{DE}\cos 60 + F_{EF}\cos 60 = 0$

$-11.54 - F_{CE} + 90.24\cos 60 + 23.09\cos 60 = 0$

$F_{CE} = 45.13\,kN$ (T)

Isolating Joint C



Taking $\Sigma F_x = 0$

$F_{CE} - F_{AC} + F_{CD}\cos 60 - F_{BC}\cos 60 = 0$

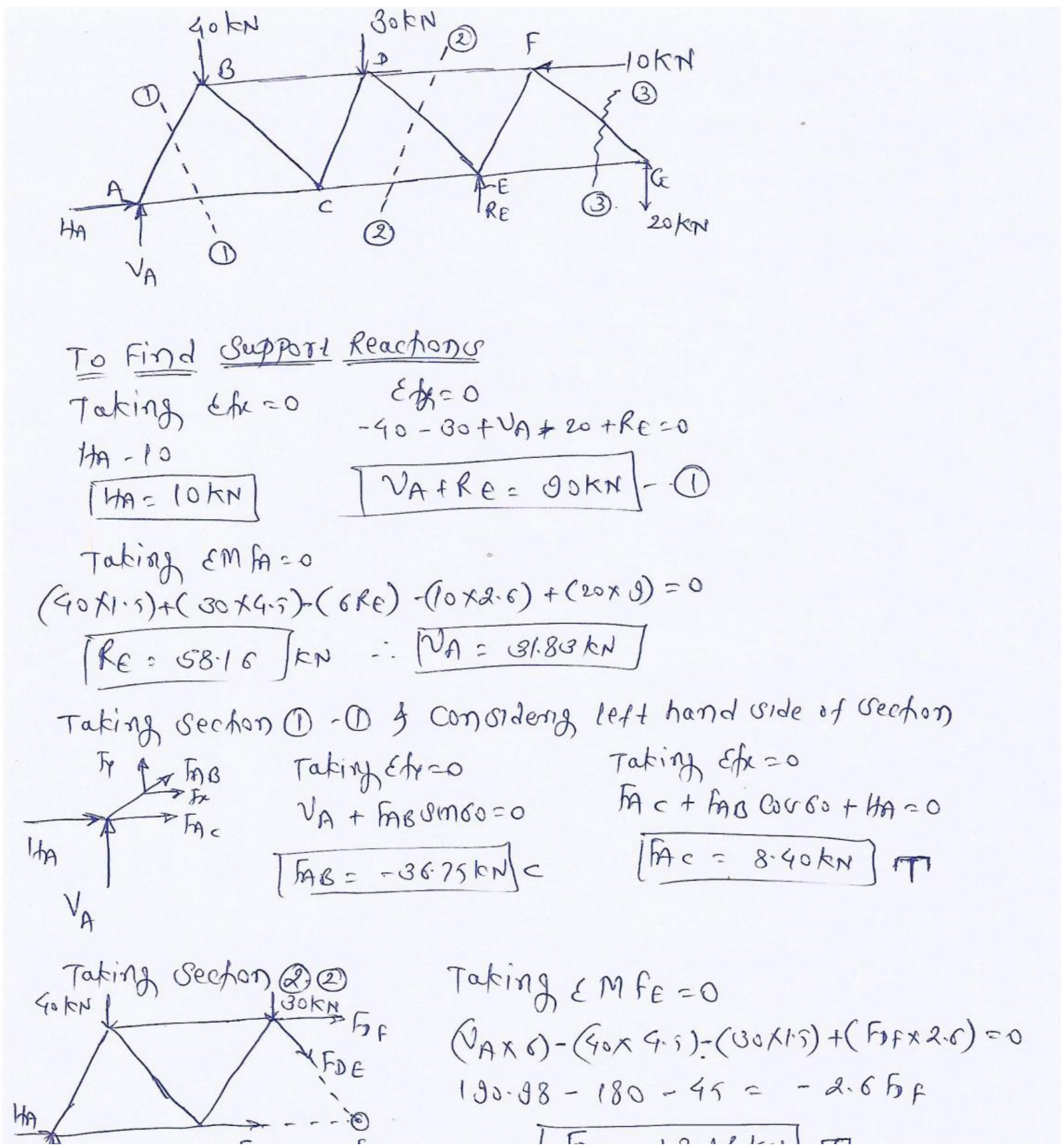
$45.13 - 8.40 + F_{CD}\cos 60 + 14.50\cos 60 = 0$

$F_{CD} = -87.96\,kN$ (C)

	c)	The Method of Sections In the method of sections, a frame is divided into two parts by taking an imaginary "cut" (shown here as a-a) through the frame. Since frame members are subjected to only tensile or compressive forces along their length, the internal forces at the cut member will also be either tensile or compressive with the same magnitude. This result is based on the equilibrium principle and Newton's third law. Steps for Analysis 1. Decide how you need to "cut" the frame. This is based on: a) Where you need to determine forces, and, b) where the total number of unknowns does not exceed three (in general).	8M

2. Decide which side of the cut frame will be easier to work with (minimize the number of forces you have to find).
3. If required, determine the necessary support reactions by drawing the FBD of the entire frame and applying the COE.
4. Draw the FBD of the selected part of the cut truss. You need to indicate the unknown forces at the cut members. Initially we assume all the members are in tension, as we did when using the method of joints. Upon solving, if the answer is positive, the member is in tension as per your assumption. If the answer is negative, the member must be in compression. (Please note that you can also assume forces to be either in tension or compression by inspection as was done in the figures above.)
5. Apply the COE to the selected cut section of the truss to solve for the unknown member forces. Note that in most cases it is possible to write one equation to solve for one unknown directly.

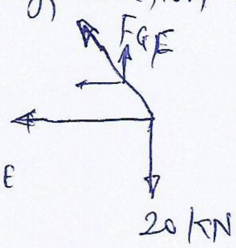
OR



FBD 1m

F_{DF} = 13.08 KN

Support reactions

		Taking $\Sigma f_y = 0$ $V_A - 40 - 30 - F_{DE} \sin 60 = 0$ $31.83 - 40 - 30 = F_{DE} \sin 60$ $\therefore \boxed{F_{DE} = -44.07\,kN}$ C Taking $\Sigma f_x = 0$ $H_A + F_{CE} + F_{DF} + F_{DE} \cos 60 = 0$ $10 + F_{CE} + 13.08 - 44.07 \cos 60 = 0$ $\boxed{F_{CE} = -32.94\,kN}$ C Taking Section ③ - ③  Taking $\Sigma f_y = 0$ $-20 + F_{GF} \sin 60 = 0$ $\therefore \boxed{F_{GF} = 23.09\,kN}$ T $\Sigma f_x = 0$ $-F_{GE} - F_{GF} \cos 60 = 0$ $-F_{GE} = F_{GF} \cos 60$ $\boxed{F_{GE} = -11.54\,kN}$ C	2m Forces in members any 5 (1m for each)