WINTER – 17 EXAMINATIONS

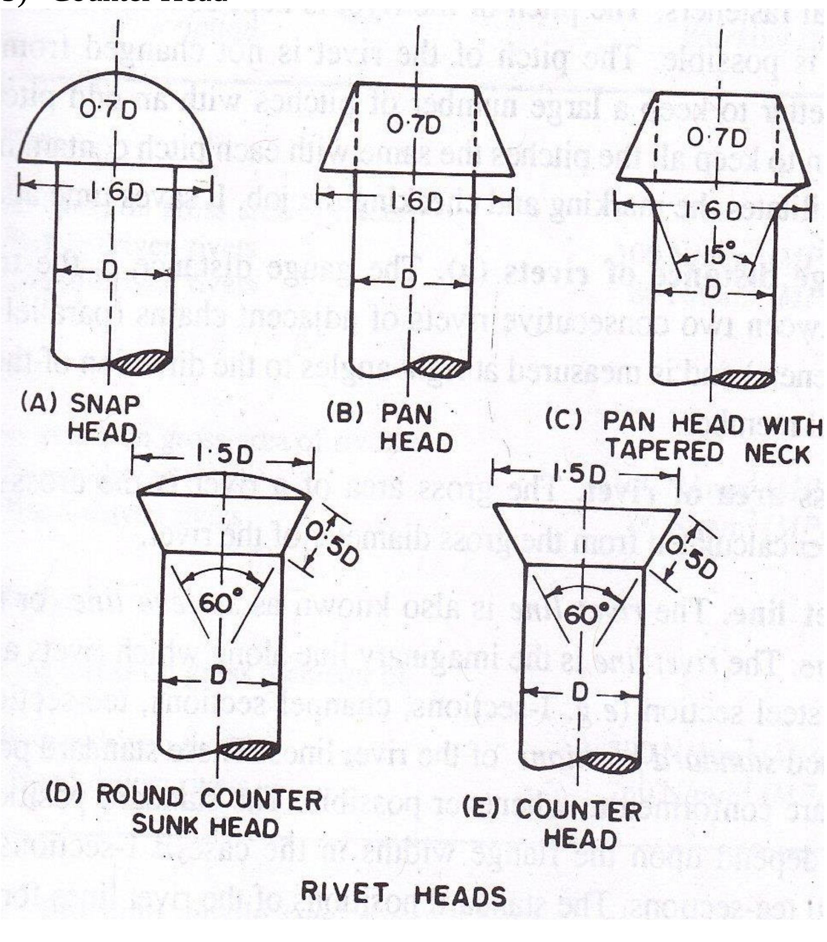
Subject Code: **17553** **Model Answer** Page No: ____/ N

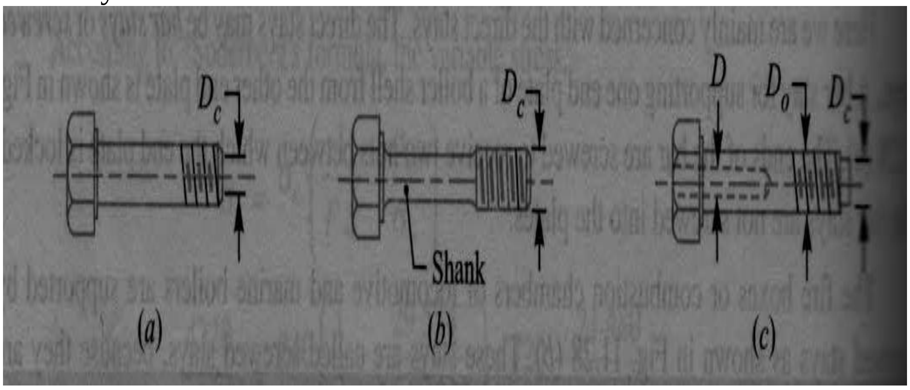
Important Instructions to examiners:

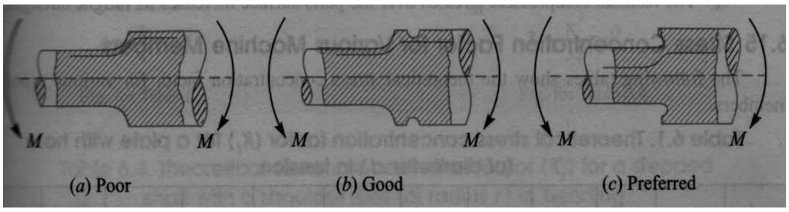
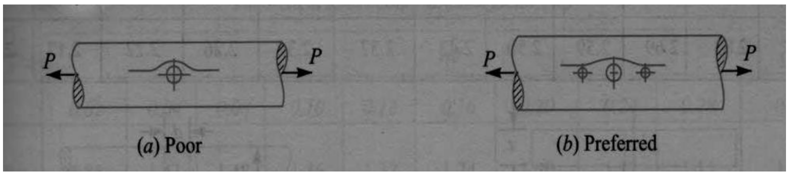
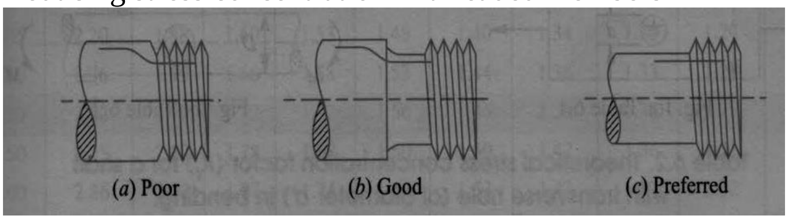
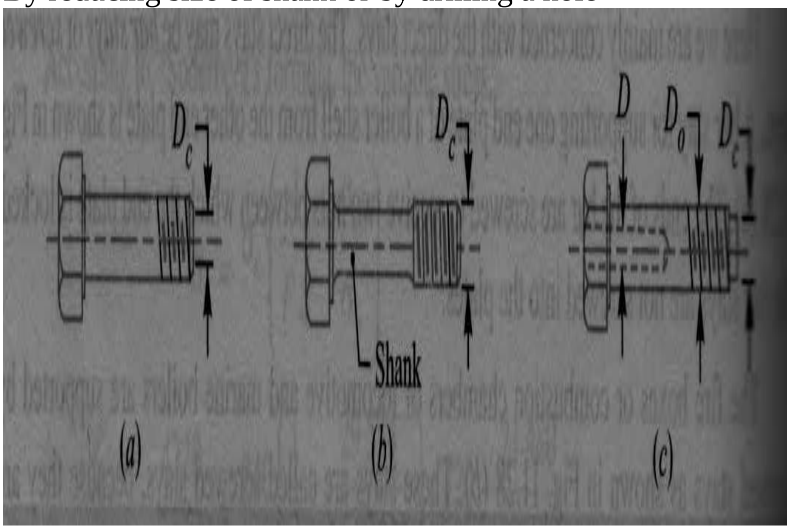
1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling 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 judgment 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.	MODEL ANSWER	MARKS	TOTAL MARKS
1	**Attempt any FIVE of the following:**		**5X4=20**
a	**Factor of Safety** It is defined, in general, as the ratio of the maximum stress to the working stress. Mathematically, Factor of safety = Maximum stress / Working or design stress In case of ductile materials e.g. mild steel, where the yield point is clearly defined, the factor of safety is based upon the yield point stress. In such cases, Factor of safety = Yield point stress / Working or design stress **Factors affecting selection of FOS:-** 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.	2 marks 2 marks. Any 2	**4M**
b	FG 300- It is a greay cast iron having a minimum tensile strenghth of 300 N/mm^2 40C4:- It is a plain carbon steel having 0.4 % carbon & 0.4% tungsten.	02 mark each	**04 marks**

c	We know for a square key $w = t = \frac{d}{4}$ Shear strength of key $T = l \times w \times \tau \times \frac{d}{2}$ ... (1) Crushing strength of key $T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ ... (2) If the key is equally strong in shearing & crushing eq^n (1) = eq^n (2) $\therefore l \times w \times \tau \times \frac{d}{2} = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ $w \times \tau = \frac{t}{2} \times \sigma_{ck}$ $\frac{d}{4} \times \tau = \frac{d/4}{2} \cdot \sigma_{ck}$ $\frac{d}{4} \times \tau = \frac{d}{8} \cdot \sigma_{ck}$ $\therefore \boxed{\sigma_{ck} = 2\tau}$		**4m**
d	**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	2m Any 2	**4m**

	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 ANY 2	
e	**Following are the types of riveted heads:-** 1) Snap Head 2) Pan Head 3) Pan Head with Taperd Neck 4) Round Countersunk head 5) Counter Head 	02 m Any 4 2m Any 2 diag	**4m**
f	**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	2m	**4m**

	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 be increased by increasing its length.of an increased modulus of resilience. This gives us **bolts of uniform strength**. The resilience of a bolt may also  	2m diag	
g	**Perfect frame :** A pin-jointed frame which has got just sufficient number of members to resist the loads without undergoing appreciable deformation in shape is called rigid or perfect frame. The perfect frame obeys the following condition viz. $n = 2j - 3$ where, n= no. of links and j= no. of joints	4 marks	**4m**
2.	**Attempt any TWO of the following:**		8X2=16
a	**Stress Concentartion:-** 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 etc. **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. 4) Surface irregularities or poor surface finish. **Remedies:-** 1) By fillets, undercutting & notches	2m 2m (any 2 causes) 4m	**8m**

	 (a) Poor (b) Good (c) Preferred 2) Additional notches & holes  (a) Poor (b) Preferred 3) Reducing stress concentration in threaded members  (a) Poor (b) Good (c) Preferred 4) By reducing size of shank or by drilling a hole  (a) (b) (c)	(any 2 remedies with diag)	

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) Design of hub $T = \frac{\pi}{16} \tau_c D^3 (1 - K^4) \cdots (1)$ 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) Design of key $w = \frac{d}{4}$ $t = \frac{d}{6}$ $l = L = 1.5d$ 3) Design for flange $T = \pi \times D \times t_f \times \tau_c \times \frac{D}{2}$ here $t_f = 0.5d$. In above equation if $\tau_c < \tau_{given}$ design safe	Design of hub 2m Design of key 2m Design of flange 2m Design of bolts 2m	**08M**

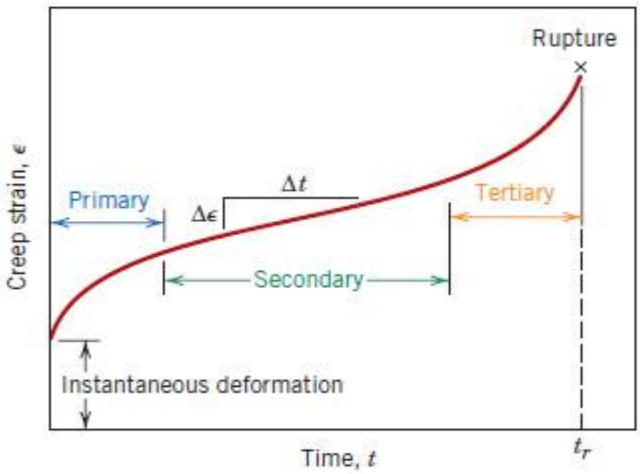
	4) Design of bolts load on each bolt $= \frac{\pi}{4} \times (d_1)^2 \times \tau_b$ $\therefore$ Total load on bolts $= n \times \frac{\pi}{4} (d_1)^2 \times \tau_b$ $\therefore$ Torque transmitted $T = n \times \frac{\pi}{4} (d_1)^2 \times \tau_b$ From above equation d_1 can be calculated. checking 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		

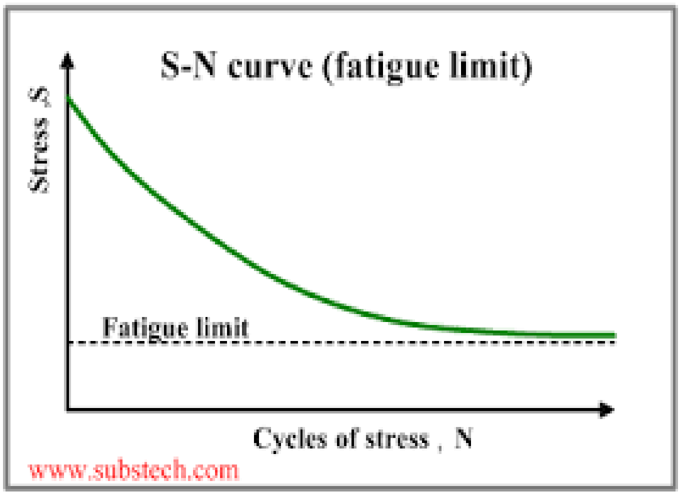
c)	Given $W = 80$ mm, $S = t = 10$ mm, $\sigma_t = 70$ N/mm² $\tau = 50$ N/mm² $W = 60$ kN $= 60\times10^3$ N. To find the effective length of single transverse fillet weld. i.e. l_1 $\therefore l_1 = W - 12.5$ $= 80 - 12.5$ $l_1 = 67.5$ mm Tensile strength of plate ($W_{\sigma t}$):- $W_{\sigma t} = 0.707 \times S \times l_1 \times \sigma_t$ $= 0.707\times10\times67.5\times70$ $W_{\sigma t} = 33.40\times10^3$ N Shearing strength of plate (W_τ):- $W_\tau = 2\times0.707\times S\times l_2\times\tau$ $= 2\times0.707\times10\times l_2\times50$ $W_\tau = 707 l_2$ We know total load carried by plate $W = W_{\sigma t} + W_\tau$ $60\times10^3 = 33.40\times10^3 + 707 l_2$ $\therefore l_2 = \frac{60\times10^3 - 33.40\times10^3}{707}$ $l_2 = 37.62$ mm For starting & stopping of weld run 12.5 mm is to be added $\therefore l_2 = 37.62 + 12.5$ $l_2 = 50.12$ mm .. length of Parallel fillet weld.	1m(L_1) 2m($W_{\sigma t}$) 2m(W_τ) 2m(L_2) 1m	08 marks

3.	**Attempt any TWO of the following:**		**2X8=16**
	1) Thickness of Boiler shell:- It can be determined by using thin Cylindrical Formula $t = \frac{P \cdot D}{2\sigma_t \times \eta_l} + 1\,mm$ 2) Diameter of Rivet If $t > 8$ mm then $d = 6\sqrt{t}$ But If $t < 8$ mm then the diameter of rivet can be found by equating shearing resistance with crushing resistance $P_s = P_c$ 3) Pitch of Rivet:- $P_{max} = C \cdot t + 41.28\,mm$ where C = Constant taken from standard table Also pitch can be found by equating $P_t = P_s$ & the least amongst the value is taken. 4) Back Pitch (P_b): $P_b = 0.33P + 0.67d$. 5) Margin (m): $m = 1.5\,d$. 6) Strap Thickness (t_1): $t_1 = 0.625t$	2m thickness 2m for dia of rivet 2m for pitch or rivet 1m back pitch 1m for margin **Strap thickness is extra data.**	**08m**

b	3b **Given** $d_1 = 50mm,\ d_2 = 375mm,\ W = 12\times10^3 N$ $d = 400mm\quad n = 4,\quad \sigma_t = 84 N/mm^2$ Direct shear load $W_s = \frac{W}{n} = \frac{12\times10^3}{4} = 3000 N$ The maximum tensile load $W_t = \frac{W\cdot d\cdot d_2}{2[d_1^2+d_2^2]} = \frac{12\times10^3\times400\times375}{2[50^2+375^2]}$ $= \frac{360\times10^6}{286.25\times10^3} = 1.25\times10^3 N$ $W_t = 1.25\times10^3 N$ **Equivalent load** $W_{te} = \frac{1}{2}[W_t + \sqrt{(W_t)^2+4(W_s)^2}$ $= \frac{1}{2}[1.25\times10^3 + \sqrt{(1.25\times10^3)^2+4(3000)^2}$ $W_{te} = 3.68\times10^3$ **Size of bolt** We know $\sigma_t = \frac{W_{te}}{\frac{\pi}{4}(d_c)^2}$ $\therefore 84 = \frac{3.68\times10^3}{\frac{\pi}{4}(d_c)^2}$ $d_c^2 84 = 4.68\times10^3$ $\therefore d_c = 6.61mm$ $d_c \approx 8mm$ We will use bolt of size M10.	02 m for direct shear load 2 m for tensile load 2m equivalent load 2m for size of bolt	**8m**

C	3c Given $D = 50mm$, $R = 25mm$, $P = 7 N/mm^2$, $\sigma_t = 20 N/mm^2$ $\sigma_b = 60 N/mm^2$ To find thickness of pipe t $t = R\left[\sqrt{\frac{\sigma_t + P}{\sigma_t - P}} - 1\right] = 25\left[\sqrt{\frac{20+7}{20-7}} + 1\right]$ $t = 11.03 \approx 12mm$ Outside dia of Packing (D_1) $D_1 = D + 2 \times$ (width of Packing) $= 50 + 2\times10$ ---- ∵ assuming Packing width = 10mm $D_1 = 70mm$ Force Trying to seperate the flange $F = \frac{\pi}{4}(D_1)^2 \times P = \frac{\pi}{4}(70)^2 \times 7$ $F = 26943N$ Since the flange is secured by two bolts $\therefore F_b = \frac{F}{2} = \frac{26934}{2} = 13471.5 N$. Let d_c = Core dia of bolts we know $F_b = \frac{\pi}{4}(d_c)^2 \times \sigma_b$ $13471.5 = \frac{\pi}{4} \times (d_c)^2 \times 60$ $d_c = 16.9 \approx 17mm$ we know $d_c = 0.84 d_o$ $\therefore d_o = 22mm$ Outer diameter of flange $D_o = D + 2t + 4.6d$ $= 50 + 2\times12 + 4.6\times22$ $D_o = 175.2mm \approx 180mm$ Pitch circle diameter $D_p = D_o - (3t + 20mm)$ $= 180 - (3\times12 + 20)$ $D_p = 124 mm$	1m given 1m thickness 1m D_1 1m for F 1m for F_b 1m d_o 1m D_o 1m D_p	**8M**

<table>
<tr><td>4.</td><td>Attempt any TWO of the following:</td><td></td><td>2X8=16</td></tr>
<tr><td>a i)</td><td>Creep Curve:- Fig shows an idealized creep curve.
When load is appilied at the beginning of creep test, Instanteneous deformation occurs.
This deformation followed by Creep curve. It occurs in three stages.
1) First stage:- it is called primary creep during which creep rate decreases.
2) Second stage:- It is called secondary creep during which creep rate remains constant.This stage occupies major portion of curve.
3) Third Stage:- it is called tertiary creep during which creep rate increases & neck formation will takes place finly reslutls in fracture.

</td><td>2m

2m
diag</td><td>4m</td></tr>
<tr><td>a ii)</td><td>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.
S-N curve:-


Consider a standard mirror polished specimen rotating in a fatigue testing machine & loaded in a bending.</td><td>2m

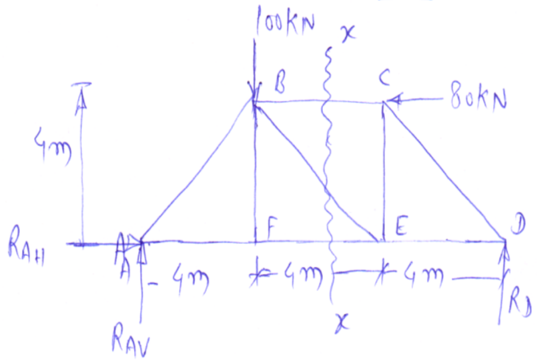
2m</td><td>4m</td></tr>
</table>

	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.		
b	Given, $M = 3000\ N\cdot m = 3\times10^6\ N\cdot mm$ $T = 1000\ N\cdot m = 1\times10^6\ N\cdot mm$ $\sigma_{tu} = \sigma_{bu} = 700\ N/mm^2$ $\tau_u = 500\ N/mm^2$, $FOS = 6$. 1) To Find σ_b & τ $\therefore \sigma_b = \frac{\sigma_{bu}}{FOS} = \frac{700}{6} = 116.67\ N/mm^2$ $\tau_1 = \frac{\tau_u}{FOS} = \frac{500}{6} = 83.33\ N/mm^2$ 2) Accordding to Max Shear Stress Theory Equivalant Twisting Moment $T_e = \sqrt{M^2+T^2} = \sqrt{(1\times10^6)^2 + (3\times10^6)^2}$ $T_e = 3.16\times10^6\ N\cdot mm$ Equating with $T_e = \frac{\pi}{16}\tau d^3 = \frac{\pi}{16}\times 83.33\ d^3$ $\therefore 3.16\times10^6 = 16.36\ d^3$ $d = 57.80\ mm$ ≈ 58 mm ≈ 60 mm 3) Accordding to Maximum Normal Stress Theory $M_e = \frac{1}{2}\left[M + \sqrt{M^2+T^2}\right] = \frac{1}{2}\left[3\times10^6 + 3.16\times10^6\right]$ $M_e = 3.08\times10^6\ N\cdot mm$ Equating with $M_e = \frac{\pi}{32}\times \sigma_b \times d^3 = \frac{\pi}{32}\times 116.67\times d^3$ $\therefore 3.08\times10^6 = 11.45\ d^3$ $d = 64.54\ mm$ ≈ 65 mm $\therefore$ Selecting the larger value $d = 65\ mm$	given1m first step 1m Second step 3m Third step 3m	**8m**

c

4c.

Drawing FBD of an entire truss



To Find Support Reactions

$\Sigma M_A = 0$

$+(100\times4) - (80\times4) - (R_D\times12) = 0$

$400 - 320 = 12 R_D$

$R_D = 6.67\,kN$

Taking

$\Sigma F_x = 0$

$R_{AH} - 80 = 0$

$R_{AH} = 80\,kN$

$\Sigma F_y = 0$

$R_{AV} - 100 + R_D = 0$

$R_{AV} =$ ~~122.3 kN~~ $93.33\,kN$

Let us consider the equilibrium of truss to the left of section x-x

Taking moment about B.

$(F_{FE}\times4) + (80\times4) = 93.33\times4$

$F_{FE} = 13.33\,kN$ T

Taking moments about E

$(F_{BC}\times4) + (100\times4) = 93.33\times8$

$F_{BC} = -86.66$ C

$\Sigma F_x = 0$

$80 - F_{BC} + F_{EF} - F_{BE}\cos45 = 0$

$F_{BE} = -9.43\,kN$ C

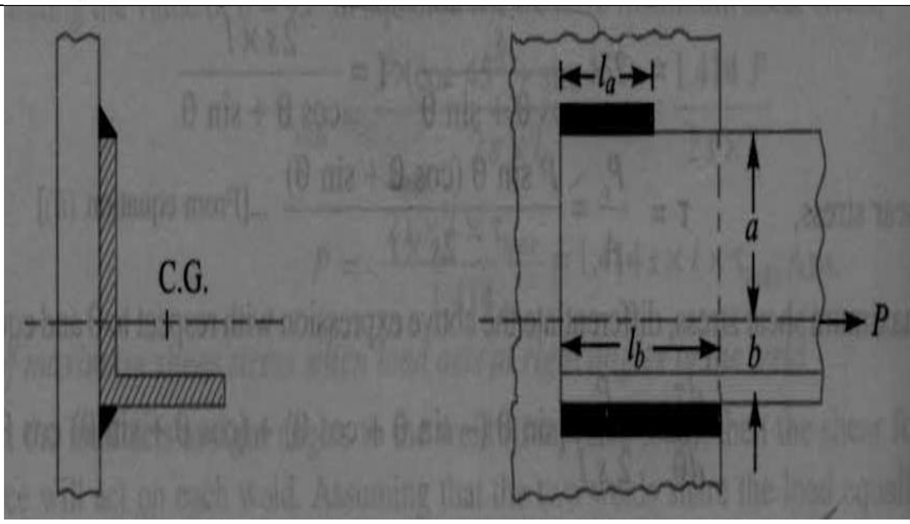
Sr No	Member	Mag	Nature
1	FE	13.33	T
2	BC	86.66	C
3	BE	9.43	C

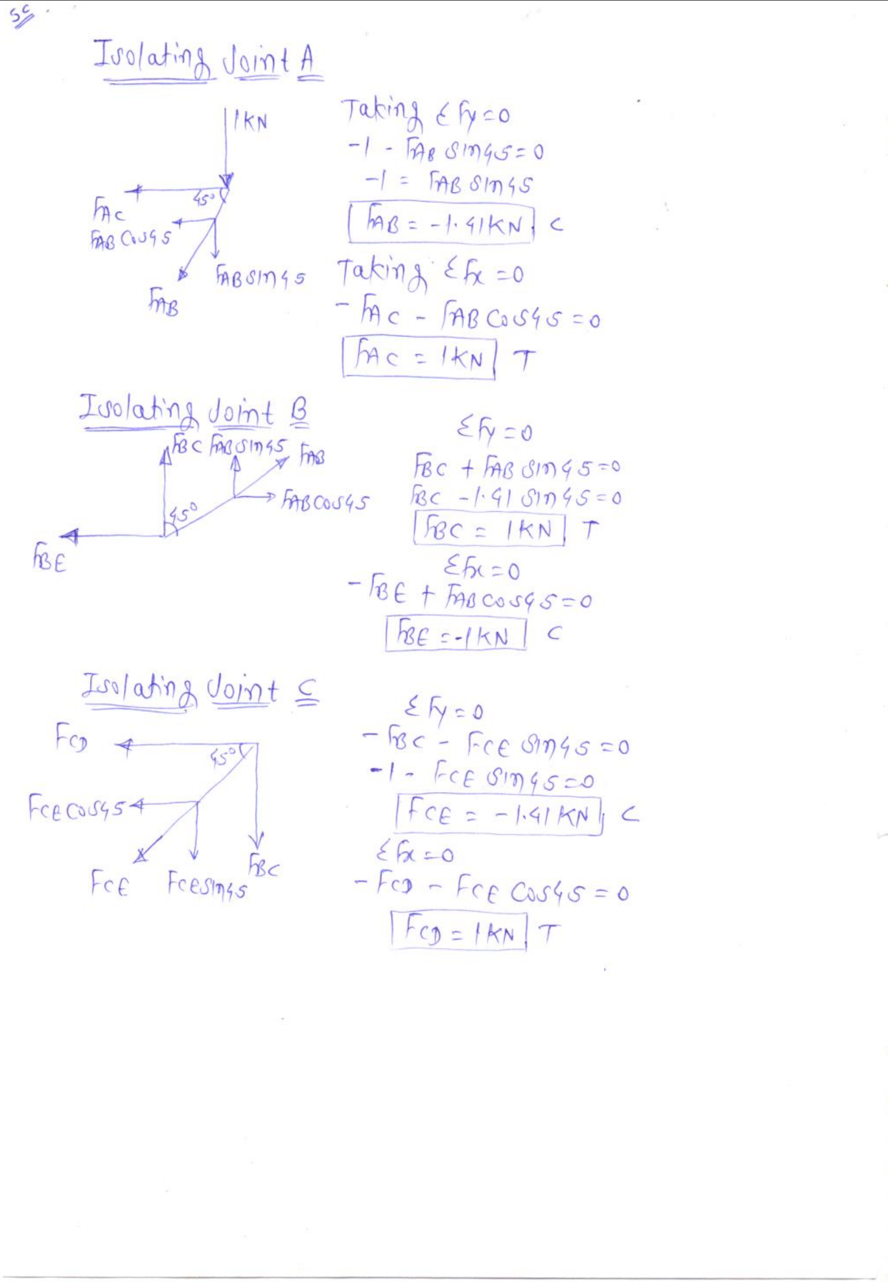
02 m to find support reaction

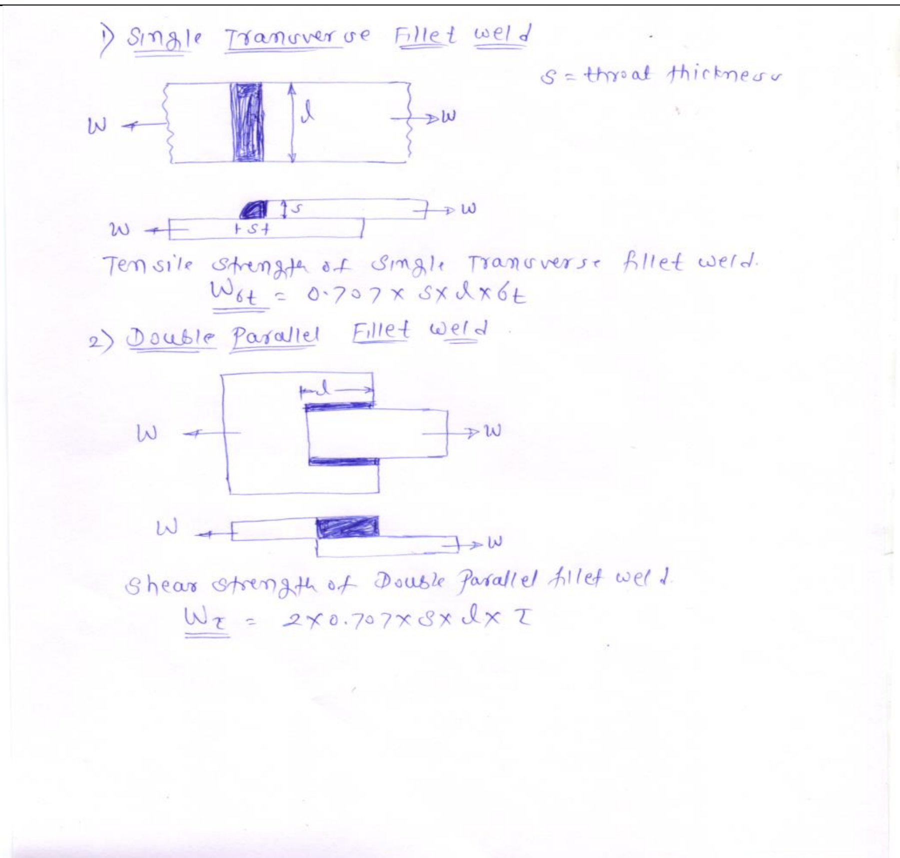
2m for each member

8m

5.	Attempt any TWO of the following:		2X8=16
a	5a Given $P = 40\ kw = 40\times10^3\ W$, $N = 350\ rpm$ $\tau_{shaft} = 40\ N/mm^2$, $\tau_{key} = 40\ N/mm^2$, $\tau_{muff} = 15\ N/mm^2$ $\sigma_{ck\ key} = 80\ N/mm^2$ 1) To Find Torque $P = \frac{2\pi N T}{60}$ $\therefore T = \frac{60P}{2\pi N}$ $\therefore T = \frac{60\times40\times10^3}{2\times\pi\times350}$ $\therefore T = 1.09\times10^3\ N.m$ $\boxed{T = 1.09\times10^6\ N.mm}$ 2) Diameter of shaft $T = \frac{\pi}{16}\tau d^3 \Rightarrow \therefore 1.09\times10^6 = \frac{\pi}{16}\times40\times d^3$ $\therefore \boxed{d = 51.77\ mm \approx 55\ mm}$ OR $\boxed{d \simeq 52\ mm}$ 3) Design of Muff/Sleeve :- By using emperical relations outer dia of muff $D = 2d+13 = 123\ mm$ OR $117\ mm$ length of Muff $L = 3.5d = 192.5 = 194\ mm$ OR $182\ mm$ checking of sleeve under shearing $T = \frac{\pi}{16}\tau D^3(1-k^4)$ $1.09\times10^6 = \frac{\pi}{16}\tau(123)^3\cdot(1-0.44^4)$ $1.09\times10^6 = 351.68\times10^3\ \tau$ $\therefore \boxed{\tau = 3.10\ N/mm^2}$ here $\tau < \tau_{given}$ ie $15\ N/mm^2$ design is safe 4) Design of key By using Emperical Relations $w = \frac{d}{4} = \frac{55}{4} = 14\ mm$ $l = \frac{L}{2} = 97\ mm$ $t = \frac{d}{6} = \frac{55}{6} = 10\ mm$	First step 1m Second step 1m Third step 3m Fourth step 3m	**8m**

	Shearing of key checking $T = l \times w \times \tau \times \frac{d}{2}$ $1.09 \times 10^6 = 97 \times 14 \times \tau \times \frac{55}{2}$ $\tau = 29.18\ N/mm^2$ here $\tau < \tau_{given}$ ie $40\ N/mm^2$ design safe checking of key under crushing $T = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2}$ $1.09 \times 10^6 = 97 \times \frac{10}{2} \times \sigma_{ck} \times \frac{55}{2}$ $\sigma_{ck} = 79.47\ N/mm^2$ here $\sigma_{ck} < \sigma_{ck\,given}$ ie $80\ N/mm^2$ design safe		
b	 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 $= l_a \times f \times a$ and moment of the bottom weld about gravity axis $= l_b \times f \times b$ Since the sum of the moments of the weld about the gravity axis must be zero, therefore,	Fig 2m Derivation 6m	**8M**

	l_a x f x a = l_b x f x b or l_a X a = l_b x b ……..(i) We know that l = l_a + l_b …….(ii) From equations (i) and (ii), we have $l_a = \frac{lxb}{(a+b)}$ & $l_b = \frac{lxa}{(a+b)}$		
C		02 Marks for all FBD of isolated joints 1m each for each member with nature.	**8M**

6.	Attempt any FOUR of the following:		4X4=16
a	•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 crosssectionalarea 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 WIithout keyway w = Width of keyway, d = Diameter of shaft h = Depth of keyway = 112 x thickness of key = 1/2 x t • 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.	4m	**4m**
b		2m 2m	**4m**

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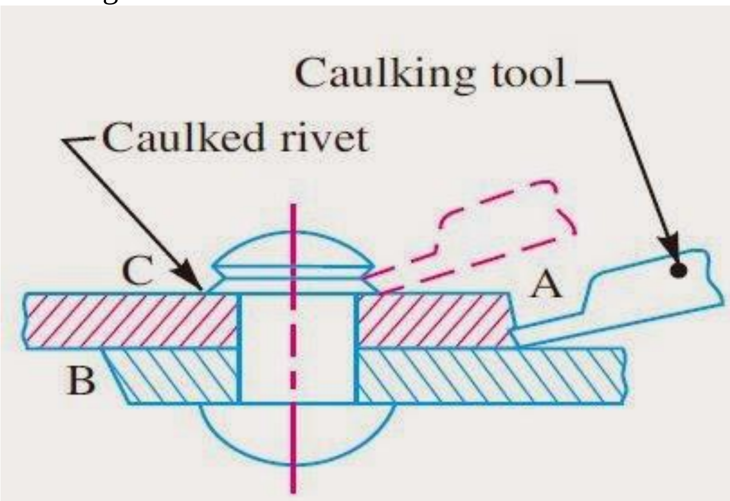
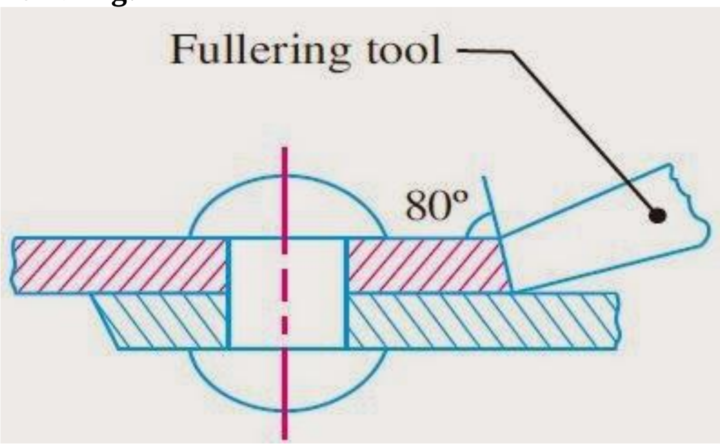
<tr><td></td><td></td><td></td><td></td></tr>
<tr><td>c</td><td>Caulking:-


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.</td><td>2m

2m</td><td>4m</td></tr>
<tr><td>d</td><td>following stresses are induced in a bolt, screw or stud when it is screwed up tightly
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</td><td>4m any 4</td><td>4m</td></tr>
</table>

	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 $\sigma_b = x.E/2l$ 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.		
e	**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**	4m	**4m**

	$\sigma r = \{[p (ri)^2] / [(ro)^2 - (ri)^2]\} / \{1 - [(ro)^2 / x^2]\}$ where p = Internal fluid pressure in the pipe, ri = Inner radius of the pipe, and ro = Outer radius of the pipe		
f	**Assumptions in the analysis of truss:-** 1)The frame is a perfect one ie the relation n=2j-3 must be satisfied. 2) All the members are hinged or pin jointed at the ends. 3) The loads are acting only at the joints. 4) The self weight of the member is neglected.	4m 1m each point	**4m**