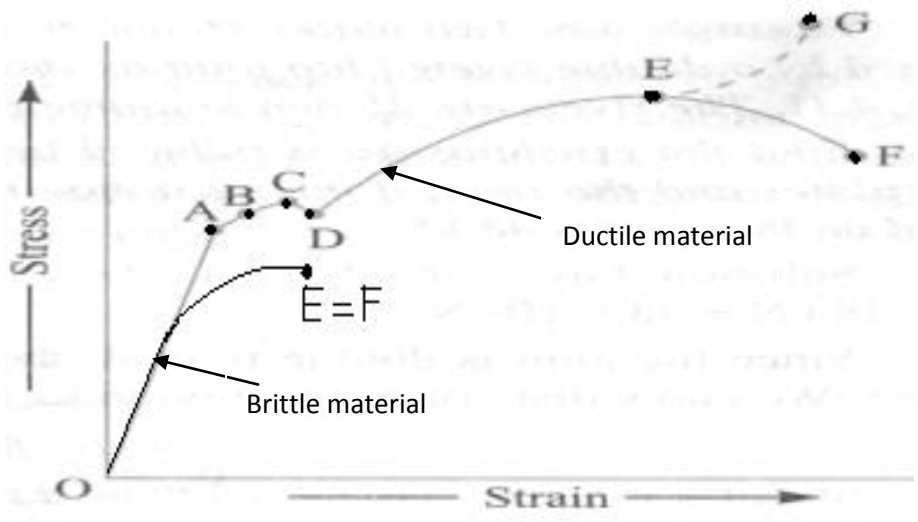
WINTER – 14 EXAMINATIONS

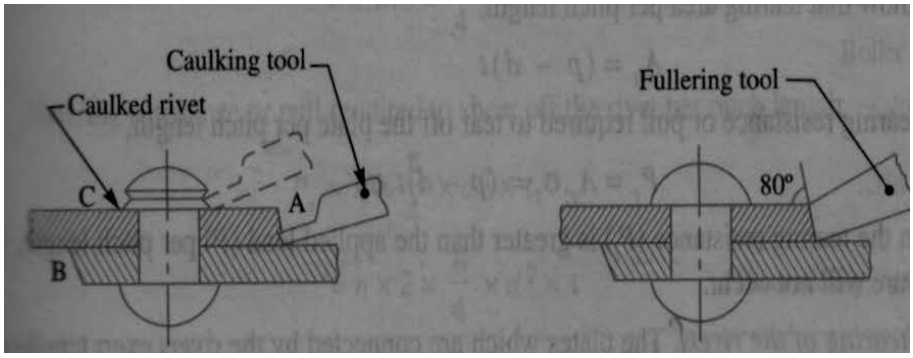
Subject Code: **17553** **Model Answer** Page No: ____/ N

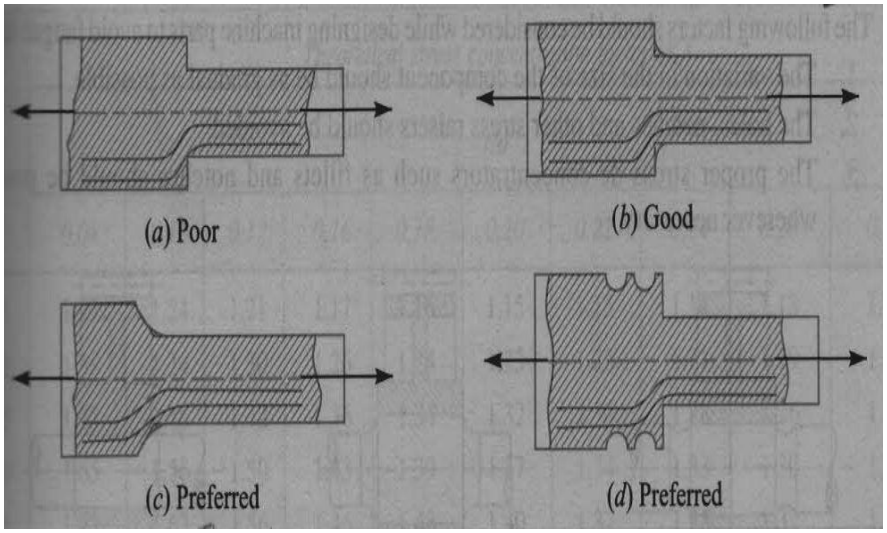
Important Instructions to examiners:

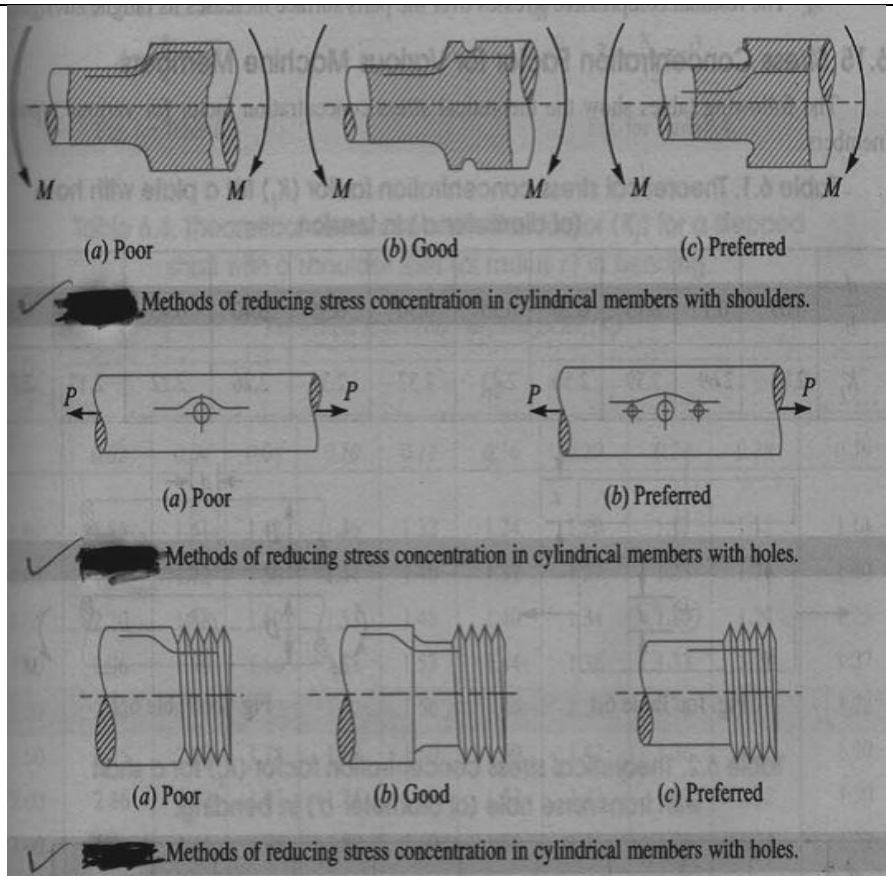
1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling 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.

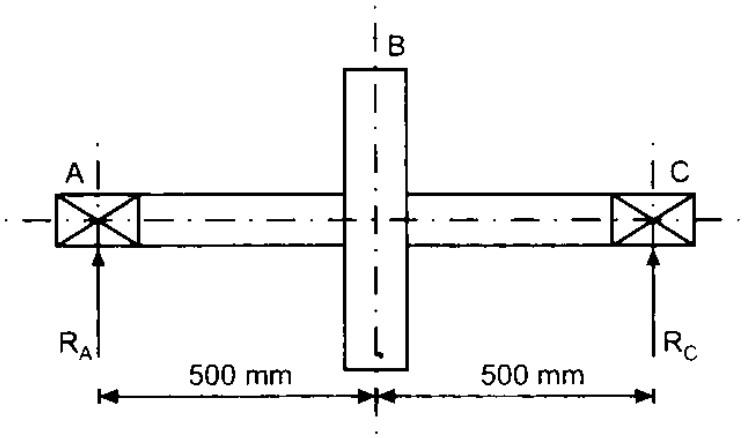
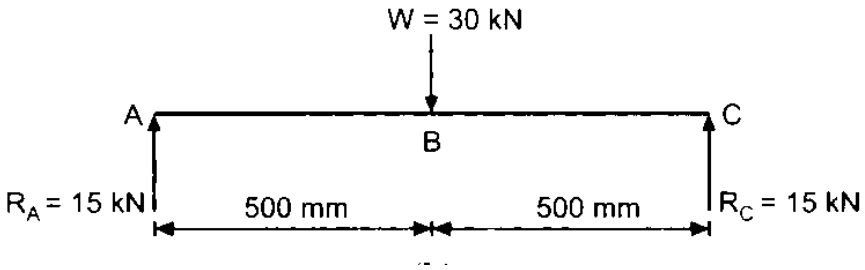
Q. NO.	MODEL ANSWER	MARKS	TOTAL MARKS
1	**Attempt any five**		**20**
a i	Grey Cast Iron	1	4
Ii	(Alloy Steel) Nickel Stells,Brass 35Mn2Mo45/40C10S18	1	
Iii	Chrome Nikel Steels 40Cr1Mo28(Alloy Steels)	1	
Iv	Carbon Steel (30C8Mn Avg)	1	
b	 A. Proportional limit: Hooke's law holds good up to point A and it is known as proportional limit. It is defined as that stress at which the stress-strain curve begins to deviate from the straight B. Elastic limit: The material has elastic properties up to the point B. This point is known as elastic limit. It is defined as the stress developed in the material without any permanent set C & D. Yeild Point: There are two yield points C and D. The points C and D are called the upper and lower yield points respectively. E. Ultimate stress: At E, the stress, which attains its maximum value is known as ultimate stress. F. Breaking strength: Failure is complete	02 (for figure) 02 (for denotions)	4
c	• The keyway cut into the shaft **reduces the load carry capacity of shaft.** • This is due to **stress concentration** near the corners of the keyway and reduction in the crossectional 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 keywayis given by H. F. Moore; e = 1 - 0.2 (w/d) - 1.1(h/d) where, e = (shaft strength factor) /(Streng th of shaft Wlithout keyway) w = Width of keyway, d = Diameter of shaft h = Depth of keyway = 1/2 x thickness of key = 112 x t • It is usually assumed that **strength of keyed shaft is 75% of solid shaft.**	4 (any 4 effects)	4

d	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 Integral weld neck flanged joint: • Overturning moment and Internal hydrostatic pressure (annular ring section) • Shear force and bending moment (tapered hub section) • Discontinuity shear force and bending moment at shell/hub interface (shell ring section)	2 2 (for any one type of joint)	4
e	Following are the advantages and disadvantages of welded joints. • 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.	2 (any 4) 2	4
f	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	2	4

	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°. The tool is moved after each blow along the edge of the plate, which is planned to a bevel of 75° to 80° to facilitate the forcing down of edge. 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.  	2 (Diags.)	
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	2 2	4
2	**Attempt any two**		**16**
A	The general procedure to solve a design problem is as follows: 1. Recognition of need: First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed. 2. Synthesis (Mechanisms): Select the possible mechanism or group of mechanisms which will give the desired motion. 3. Analysis of forces: Find the forces acting on each member of the machine and the energy transmitted by each member. 4. Material selection: Select the material best suited for each member of the machine. 5. Design of elements (Size and Stresses):Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit. 6. Modification: Modify the size of the member to agree with the past experience and judgement to facilitate manufacture. The modification may	4 (1/2 mark each)	8

	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. General procedure in Machine Design: Following are the general considerations in designing a machine component: 1. Type of load and stresses caused by the load 2. Motion of the parts or kinematics of the machine. 3. Selection of materials 4. Form and size of the parts 5. Frictional resistance and lubrication. 6. Convenient and economical features 7. Use of standard parts 8. Safety of operation 9. Workshop facilities 10. Number of machines to be manufactured 11. Cost of construction. 12. Assembling.	4 (1/2 mark each for any 8)	
b i	The presence of stress concentration i.e. say, a presence of abrupt change/s in cross sectional area/s cannot be totally eliminated but it may be reduced to some extent. A device or concept that is useful in assisting a design engineer to visualize the presence of stress concentration and how it may be reduced is that of stress flow lines, as shown in Figs. The reduction of stress concentration means that the stress flow lines shall maintain their spacing as far as possible.  	2 2 (any 2 remedies)	8

			
ii	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 In case of brittle materials e.g. cast iron, the yield point is not well defined as for ductile materials. Therefore, the factor of safety for brittle materials is based on ultimate stress Factor of safety = Ultimate stress /Working or design stress Selection of Factor of safety: Before selecting a proper factor of safety, a design engineer should consider the following points : 1. The reliability of the properties of the material and change of these	2 2 (any 4 points)	

	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.		
c	Given data: WB = 30 kN = 30 X 10^3 N, бb = 60 MPa = 60 N/mm^2  (a)  Considering the vertical load diagram, RA + Rc = 30 x 10^3 N To find Rc, taking moment about point A, Σ MA = O 30 X 10^3 X 500 - Rc x 1000 = 0 Rc = 15 X 10^3 N and RA = 30 X 10^3 - Rc = 30 X 10^3 - 15 X 10^3 RA = 15 X 103 N Bending moment at point B = MB = 15 X 10^3 X 500 = 7500 X 10^3 N-mm Axles are subjected to bending moment only. Therefore, the axle can be designed by using the equation; M =(π/ 32)' бb X d^3 7500 x 10^3 = 32 x 60 x $(d)^3$ d = 108.38 mm ~ 110 mm (say)	4 (Sketch) 4	8

3	Attempt any two		16
a	Given: P = 15 kW = 15 X 10^3 W ; N = 900 r.p.m. ; Service factor = 1.35,ד s =ד b = ד k = 40 MPa = 40 N/mm^2 бcb = σck= 80 MPa = 80 N/mm^2,ד c = 8 MPa = 8 N/mm^2 1.*Design for hub* First of all, let us find the diameter of the shaft *(d).* We know that the torque transmitted by the shaft T= (P x 60)/ (2πN) = (15 X 10^3 x 60) /(2π x 900) = 159.13 N-m Considering torsional shear stress; d= 164.36mm *2. Design for bolts* Let *di*= Nominal diameter of bolts. Since the diameter of the shaft is 35 mm, therefore let us take the number of bolts n = 3 and pitch circle diameter of bolt , *D1* = *3d* = 3 x 35 = 105 mm The bolts are subjected to hear stress due to the torque tran mitted. We kn maximum torque transmitted (T max) 215 x 10^3 = (π/4) x $(d1)^2$ x ד b X n X(D/ 2) =(π/4) x $(d1)^2$ x 40 X *3* X(105/ 2) =4950$(d1)^2$ SS $(d1)^2$ = 215 x 10^3/4950 = 43.43 or *d1*= 6.6 mm Assuming coarse threads, the nearest standard size of bolt is M 8. Ans	4 4 (design & no. of bolts)	8
b	Let T = Torque transmitted by the shaft, F = Tangential force acting at the circumference of the shaft, d = Diameter of shaft, l = Length of key, w = Width of key. t=Thickness of key, and ד and бc = Shear and crushing stresses for the material of key. A little consideration will show that due to the power transmitted by the shaft, the key may fail due to shearing or crushing. Considering shearing of the key, the tangential shearing force acting at the circumference of the shaft, F = Area resisting shearing x Shear stress = l x w X ד Torque transmitted by the shaft, $T = F$ x (d/2) = l x w X ד X (d/2) ... *(i)* Considering crushing of the key, the tangential crushing force acting at the circumference of the shaft F = Area resisting crushing x Crushing stress = l x t/2 x бc Torque transmitted by the shaft, $T = F$ x *d*/2 = l x t/2 x d/2 x бc x d/2 ------*(ii)* The key is equally strong in shearing and crushing, if l x w X ד X (d/2) = l x t/2 x d/2 x бc x d/2 ...[Equating equations *(i)* and *(ii)]*	3 3 2	8

	Or (w/t) = (бc /2τ) The permissible crushing stress for the usual key material is atleast twice the permissible shearing stress. Therefore from equation *(iii),* we have w = *t.* In other words, a square key is equally strong in shearing and crushing]		
c	Given: Width = 120 mm; Thickness = 15 mm In Fig. AB represents the single transverse weld and AC and BD represents double parallel fillet welds. 1. Length of the weld run for a single transverse weld The effective length of the weld run (l1) for a single transverse weld may be obtained by subtracting 12.5 mm from the width of the plate. l1 = 120 - 12.5 = 107.5 mm Ans. 2. Length of the weld run for a double parallel fillet weld subjected to variable loads Let l2 = Length of weld run for each parallel fillet, and s = Size of weld = Thickness of plate = 15 mm Assuming the tensile stress as 70 MPa or N/mm^2 and shear stress as 56 MPa or N/mm^2 for static loading. We know that the maximum load which the plate can carry is P = Area x Stress = 120 x 15 x 70 = 126 X 10^3 N we find that the stress concentration factor for transverse weld is 1.5 and for parallel fillet welds is 2.7. Permissible tensile stress, бt = 70/ 1.5 = 46.7 N/mm^2 and permissible shear stress, τ = 56 / 2.7 = 20.74 N/mm^2 Load carried by single transverse weld, P1 = 0.707 s x l1X бt = 0.707 x 15 x 107.5 x 46.7 = 53 240 N and load carried by double parallel fillet weld, P2 = 1.414 s x 12 X τ= 1.414 x 15 x l2 x 20.74 = 440 l2 N Load carried by the joint (P), 126 x 10^3 = P1 + P2 =53240 + 440 l2 or l2= 165.4 mm Adding 12.5 mm for starting and stopping of weld run, we have l2 = 165.4 + 12.5 = 177.9 say 178 mm Ans	2 2 2 2	8
4	**Attempt any two**		**16**
a	 Let	4 (Diag.)	8

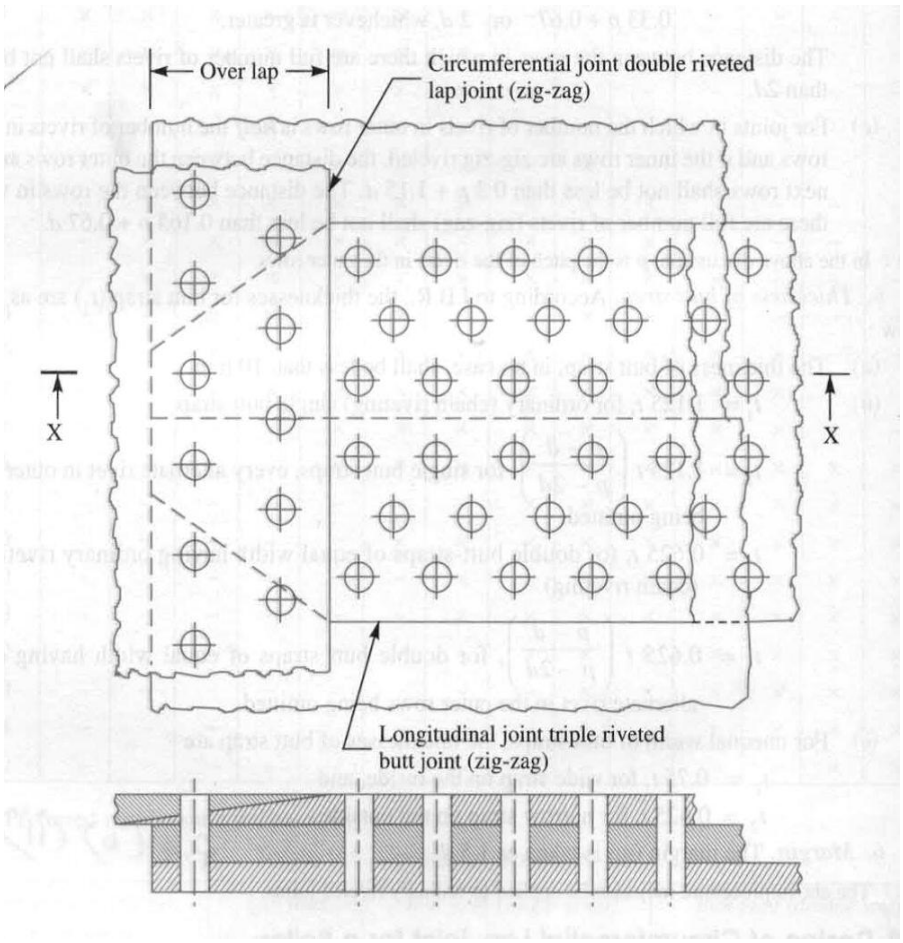
	la = Length of weld at the top, Ib = Length of weld at the bottom, l= Total length of weld = la + lb P = Axial load, a = Distance of top weld from gravity axis, b = Distance of bottom weld from gravity axis, and f = Resistance offered by the weld per unit length. Moment of the top weld about gravity axis = la x f x a and moment of the bottom weld about gravity axis = Ib xf x b Since the sum of the moments of the weld about the gravity axis must be zero, therefore, la xfx a -lb xfx b = 0 or la X a = Ib X lb ----------i We know that l1 = la + lb ----------ii :. From equations (i) and (if), we have *la* =(l x b)/(a + b) and lb =(l x a) / (a + b)	4 (Proced ure)	
b i	Design of longitudinal Butt joint for boiler. According to Indian Boiler Regulations (I.B.R), the following procedure should be adopted for the design of longitudinal butt joint for a boiler. 1. Thickness of boiler shell. First of all, the thickness of the boiler shell is determined by using the thin cylindrical formula i.e. t = (P. D)/(2бt x ηl) where; t = Thickness of the boiler shell, P = Steam pressure in boiler, D = Internal diameter of boiler shell, бt = Permissible tensile stress, and ηl = Efficiency of the longitudinal joint. The following points may be noted: (a) The thickness of the boiler shell should not be less than 7 mm. (b) The efficiency of the joint may be taken from the following table Indian Boiler Regulations (I.B.R.) allow a maximum efficiency of 85% for the best joint. (c) According to I.B.R., the factor of safety should not be less than 4. The following table shows the values of factor of safety for various kind of joints in boilers. 2. Diameter of rivets. After finding out the thickness of the boiler shell (t), the diameter of the rivet hole (d) may be determined by using Unwin's empirical formula, i.e. d = 6. √ t (when t is greater than 8 mm) But if the thickness of plate is less than 8 mm, then the diameter of the rivet hole may be calculated by equating the shearing resistance of the	4	8

rivets to crushing resistance. In no case, the diameter of rivet hole should not be less than the thickness of the plate, because there will be danger of punch crushing. The following table gives the rivet diameter corresponding to the diameter of rivet hole as per IS : 1928 - 1961 (Reaffirmed 1996).
According to IS : 1928 - 1961 (Reaffirmed 1996), the table in the Design data Book gives the preferred length and diameter combination for rivets.

3. Pitch of rivets. The pitch of the rivets is obtained by equating the tearing resistance of the plate to the shearing resistance of the rivets. It may noted that
(a)The pitch of the rivets should not be less than 2d, which is necessary for the formation of head.
(b)The maximum value of the pitch of rivets for a longitudinal joint of a boiler as per I.B.R. is
Pmax = Cx t + 41.28 mm
where t = Thickness of the shell plate in mm, and
C = Constant.
The value of the constant C is given in Table 9.5.

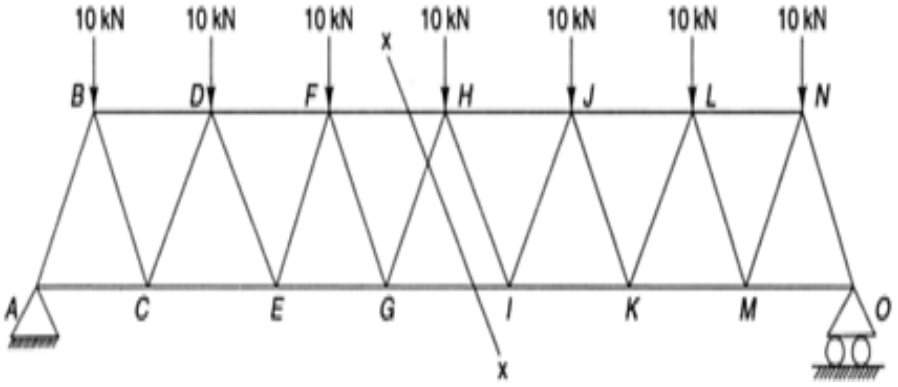
4. Distance between the rows of rivets. The distance between the rows of rivets as specified by
Indian Boiler Regulations is as follows:
(a) For equal number of rivets in more than one row for lap joint or butt joint, the distance between the rows of rivets (Pb) should not be less than
0.33 P + 0.67 d, for zig-zig riveting, and
2 d, for chain riveting.
(b) For joints in which the number of rivets in outer rows is half the number of rivets in inner rows and if the inner rows are chain riveted, the distance between the outer rows and the next rows should not be less than
0.33 P + 0.67 or 2 d, whichever is greater.
The distance between the rows in which there are full number of rivets shall not be less than 2d.
(c) For joints in which the number of rivets in outer rows is half the number of rivets in inner rows and if the inner rows are zig-zig riveted, the distance between the outer rows and the next rows shall not be less than 0.2 P + 1.15 d. The distance between the rows in which there are full number of rivets (zig-zag) shall not be less than 0.165 P + 0.67 d.
Note: In the above discussion, P is the pitch of the rivets in the outer rows.

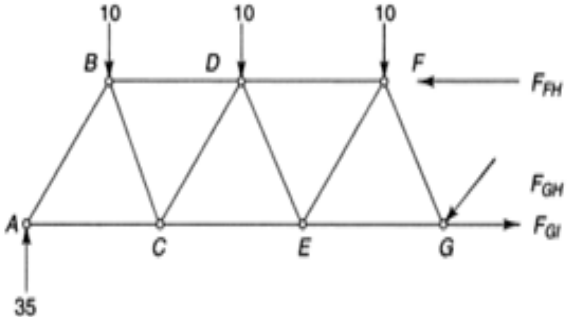
5. Thickness of butt strap. According to I.B.R., the thicknesses for butt strap (t1) are as given below:
(a) The thickness of butt strap, in no case, shall be less than 10 mm.
(b) t1 = 1.125 t, for ordinary (chain riveting) single butt strap.
Tl1 = 1.125 t[(p - d)/(p – 2d)]for single butt straps, every alternate rivet in outer rows being omitted.
t1= 0.625 t, for double butt-straps of equal width having ordinary

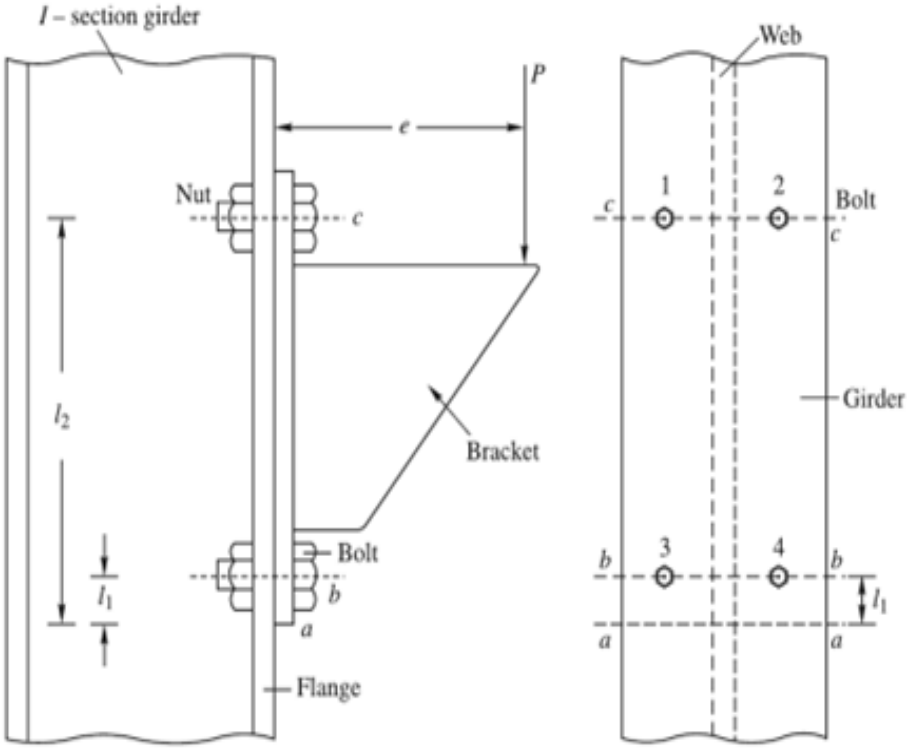
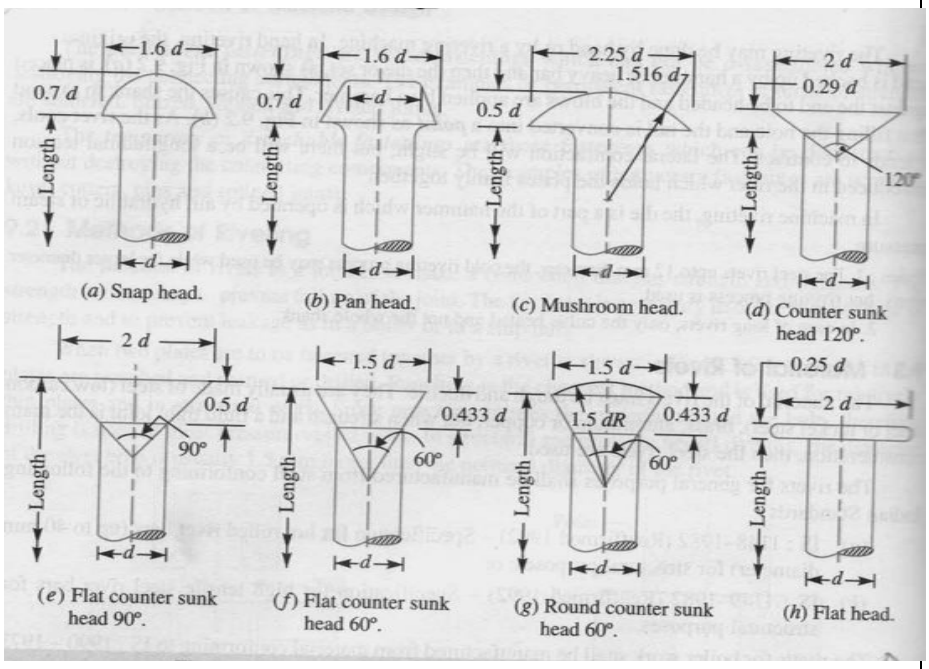
	riveting(chain riveting). T1 = 0.625 t /[(p - d)/(p – 2d)] for double butt straps of equal width having every alternate rivet in the outer rows being omitted. (c) For unequal width of butt straps, the thicknesses of butt strap are t1 = 0.75 t, for wide strap on the inside, and t2 = 0.625 t, for narrow strap on the outside. 6. Margin. The margin (m) is taken as 1.5 d. Note: The above procedure may also be applied to ordinary riveted joints		
Ii	 Design of Circumferential Lap Joint for a Boiler The following procedure is adopted for the design of circumferential lap joint for a boiler. 1. Thickness of the shell and diameter of rivets. The thickness of the boiler shell and the diameter of the rivet will be same as for longitudinal joint. 2. Number of rivets. Since it is a lap joint, therefore the rivets will be in single shear. Shearing resistance of the rivets, $P_s = n \times \pi/4 \times d^2 \times \tau$ ... (i)	2 (diagm.) 2 (Proced ure)	

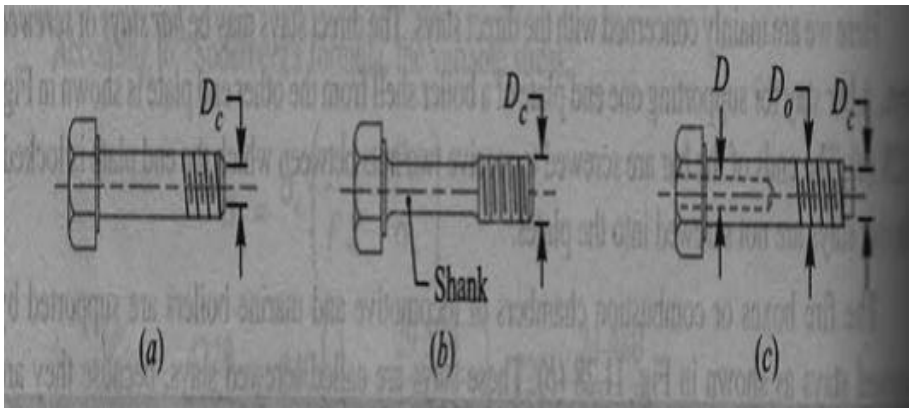
	where n = Total number of rivets. Knowing the inner diameter of the boiler shell (D), and the pressure of steam (P), the total shearing load acting on the circumferential joint, $W_s = \pi /4 \times D^2 \times P$ -----(ii) From equations (i) and (ii), we get $n \times \pi /4 \times d^2 \times \tau = \pi /4 \times D^2 \times P$ $n = (D/d)^2 \times (P/\tau)$ 3. Pitch of rivets. If the efficiency of the longitudinal joint is known, then the efficiency of the circumferential joint may be obtained. It is generally taken as 50% of tearing efficiency in longitudinal joint, but if more than one circumferential joints is used, then it is 62% for the intermediate joints. Knowing the efficiency of the circumferential lap joint (TIc)' the pitch of the rivets for the lap joint (P1) may be obtained by using the relation: $\eta c = (P1 - d)/P1$ 4. Number of rows**.** The number of rows of rivets for the circumferential joint may be obtained from the following relation: Number of rows = Total number of rivets / Number of rivets in one the number of rivets in one row $\pi (D + t) /p1$ where D = Inner diameter of shell. 5. After finding out the number of rows, the type of the joint (i.e. single riveted or double riveted etc.) may be decided. Then the number of rivets in a row and pitch may be re-adjusted. In order to have a leak-proof joint, the pitch for the joint should be checked from Indian Boiler regulations. 6. The distance between the rows of rivets (i.e. back pitch) is calculated by using the relations discussed in the previous article. 7.After knowing the distance between the rows of rivets (Pb)' the overlap of the plate may be fixed by using the relation, Overlap = (No. of rows of rivets - 1) Pb + m where m = Margin		
c	Given: $p = 0.7$ N/mm^2; $n = 12$; $D = 300$ mm; $\sigma t = 100$MPa $= 100$ N/mm^2 We know that the total force (or the external load) acting on the cylinder head i.e. on 12 bolts. $= \pi/4 (D)^2 P = = \pi/4 (300)^2 0.7 = 49490$ N External load on the cylinder head per bolt, $P2 = 49490/ l2 = 4124$ N d = Nominal diameter of the bolt, and dc= Core diameter of the bolt. We know that initial tension due to tightening of bolt, $P1 = 2840\ d$ N ... (where d is in mm) we find that for soft copper gasket with long through bolts, the minimum value of K = 0.5. Resultant axial load on the bolt, $P = P1 + K.\ P2 = 2840\ d + 0.5 \times 4124 = (2840\ d + 2062)$ N We know that load on the bolt (P), $2840\ d + 2062 = \pi/4 (dc)^2 \sigma t = \pi/4 (0.84d)^2 100 = 55.4\ d^2$ (Taking dc = 0.84)	2 2 2	8

	55.4 d2 - 2840d - 2062 = 0 d2-51.3d-37.2 =0 d = [51.3 ± √ {(51.3)^2 + 4 x 37.2}] /2 = (51.3± 52.7) /2 = 52 mm .. (Taking + ve sign) Thus, we shall use a bolt of size M 52 Ans.	2	
5	**Attempt any two**		**16**
a	Design of Circular Flanged Pipe Joint the effective diameter on which the fluid pressure acts, just at the point of leaking, is the diameter of a circle touching the bolt holes. Let this diameter be D1. If d1 is the diameter of bolt hole and Dp is the pitch circle diameter, then D1 = Dp-d1 Force trying to separate the two flanges, F = π/4 (D1)^2 P ----------(i) Let n = Number of bolts, dc = Core diameter of the bolts, and бt = Permissible stress for the material of the bolts. Resistance to tearing of bolts = π/4 (dc)^2 бt x n --------(ii) Assuming the value of d, the value of n may be obtained from equations (i) and (ii). The number of bolts should be even because of the symmetry of the section. The circumferential pitch of the bolts is given by Pc =πDp/n In order to make the joint leakproof, the value of Pe should be between 20 √ d1 to 30 √ d2 where d., is the diameter of the bolt hole. Also a bolt of less than 16 mm diameter should never be used to make the joint leakproof. The thickness of the flange is obtained by considering a segment of the flange In this it is assumed that each of the bolt supports one segment. The effect of joining of these segments on the stresses induced is neglected. The bending moment is taken about the section X-X, which is tangential to the outside of the pipe. Let the width of this segment is x and the distance of this section from the centre of the bolt is y. Bending moment on each bolt due to the force F = F/n x y ----------(iii) and resisting moment on the flange = бb X Z ...(iv) where бb = Bending or tensile stress for the flange material, and Z = Section modulus of the cross-section of the flange = 1/6 x(x) (tf)^2 Equating equations (iii) and (iv), the value of 'imay be obtained. The dimensions of the flange may be fixed as follows: Nominal diameter of bolts, d = 0.75 t + 10 mm Number of bolts, n = 0.0275 D + 1.6 ...(D is in mm) Thickness of flange, tf = 1.5 t + 3 mm Width of flange, B = 2.3 d	2 2 2 2	8

	Outside diameter of flange, D = D+2t+2B Pitch circle diameter of bolt Dp = D + 2t +2d +12mm		
b	Start with Joint C: ΣV = 0 = BC sin60 – 40 ΣH = 0 = BC cos60 – DC Get values of BC & DC Take Joint D: ΣV = 0 = BD – 40 ΣH = 0 = ED – DC Get values of BD & ED Take Joint B: Apply for the members AB & EB (unknown), BD & DC (now known) the equations for equilibrium; ΣV = 0 ΣH = 0 Get values of AB & EB	2 (Diagm.) 2 2 2	8
c	The method of section is preferred over the method of joints when; 1. If forces in a few members are to be determined in a large truss. 2. In solving for forces in members for those trusses where the method of joint cannot be applied. Consider the truss as shown below as an example. Applying the method of section x-x, to find force in members FH, GH & GI;  	2	8

	Due to symmetry $R_A = R_0 = \frac{1}{2} \times 10 \times 7 = 35$ kN Since force in members *FH*, *HG* and *GI* is to be determined, take section x–x cutting these three members which divide the truss into two parts. We can take any part. Let us take the left part of the truss.  Now $\Sigma M_G = 0$ $-F_{FH} \times 4 \sin 60 + 35 \times 12 - 10 \times 10 - 10 \times 6 - 10 \times 2 = 0$ $\therefore \quad F_{FH} = 69.2$ kN $\Sigma P_y = 0$ $35 - F_{GH} \sin 60 - 10 - 10 - 10 = 0$ or $\quad F_{GH} = 5.7$ kN $\Sigma P_x = 0,$ $F_{GI} - F_{GH} \cos 60 - F_{FH} = 0$ or $\quad F_{GI} = 72.05$ kN	2 2 2	
6	**Attempt any two**		**16**
a	Consider tensile stress area of each bolt, A_t and determine; • Direct shear stress on each bolt • Bending stresses in upper bolts • Final stresses in upper bolts • Principal stresses in upper bolts Using the given safe working stress in tension, calculate the tensile stress area A_t.	1 1 1 1	8

		4	
b	 The efficiency of a riveted joint is defined as the ratio of the strength of riveted joint to the strength of the un-riveted or solid plate. We have already discussed that strength of the riveted joint	4 2	8

	= Least of Pt, Ps and Pc Strength of the un-riveted 01: solid plate per pitch length: P = p x t X бt η = Least of Pt, Ps and Pc / p X t X бt where p = Pitch of the rivets, t = Thickness of the plate, and бt = Permissible tensile stress of the plate materials Materials may be any ductile material; ferrous/non ferrous	2	
c i	Bolt 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. 	2 2 (diagm.)	8
ii	1.The shafts are usually cylindrical, but may be square or cross-shaped in section. They are solid in cross-section but sometimes hollow shafts are also used. 2. An axle, though similar in shape to the shaft, is a stationary machine element and is used for the transmission of bending moment only. It simply acts as a support for some rotating body such as hoisting drum,a car wheel or a rope sheave. 3. A spindle is a short shaft that imparts motion either to a cutting tool (e.g. drill press spindles) or to a work piece (e.g. lathe spindle) The material used for ordinary shafts is carbon steel of grades 40 C 8, 45 C 8, 50 4 and 50 C12	2 (any 2) 2	