WINTER – 18 EXAMINATIONS

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

Important Instructions to examiners:

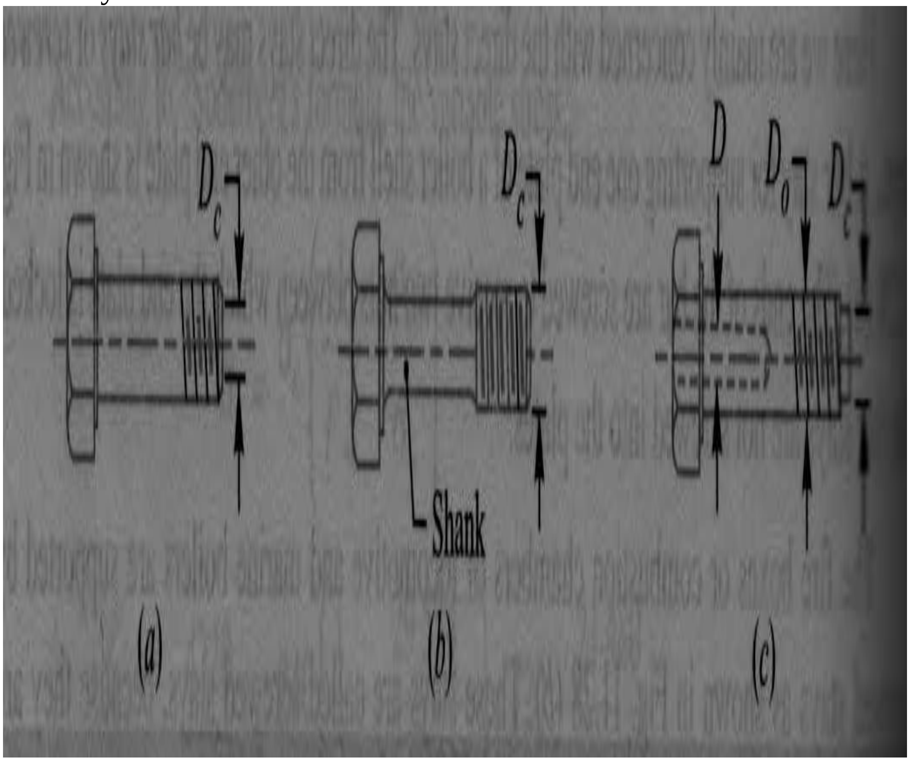
1) The answers should be examined by key words and not as word-to-word as given in the model answer scheme.
2) The model answer and the answer written by candidate may vary but the examiner may try to assess the understanding level of the candidate.
3) The language errors such as grammatical, spelling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

<table>
<tr><th>Q. NO.</th><th>MODEL ANSWER</th><th>MARKS</th><th>TOTAL MARKS</th></tr>
<tr><td>1</td><td>Attempt any FIVE of the following:</td><td></td><td>5X4=20</td></tr>
<tr><td>a</td><td>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.</td><td>4m
Any 4
1m each</td><td>4</td></tr>
<tr><td>b</td><td>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.

Effect:-
• 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.</td><td>01 mark

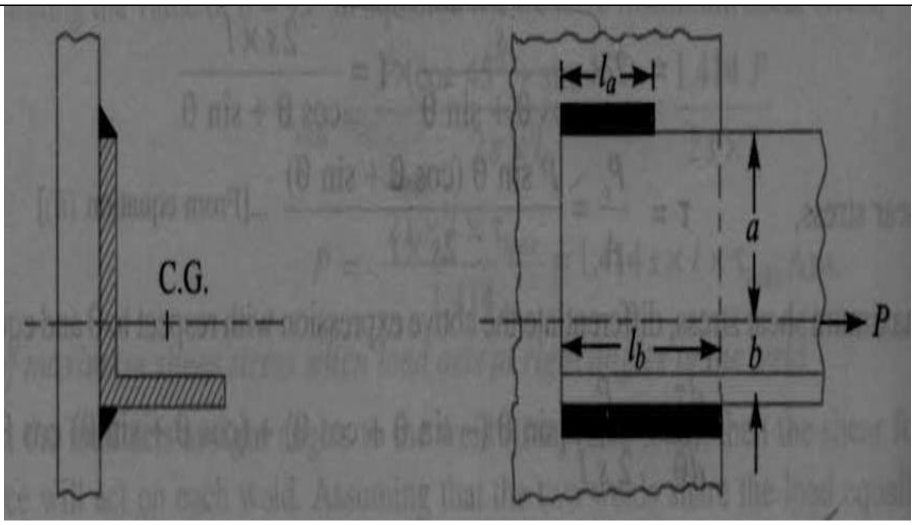
3 mark effect</td><td>04 marks</td></tr>
<tr><td>c</td><td>Advantages:-
1. The welded structures are usually lighter than riveted structures. This is due to the reason that in welding, gussets or other connecting components are not used.
2. The welded joints provide maximum efficiency (may be 100%) which 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</td><td>2m
Any 2</td><td>4m</td></tr>
</table>

	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 Any 2	
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. **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. **Materials:-** The material of the rivets must be tough and ductile. They are usually made of steel (low carbon steel or nickel steel), brass, aluminum, or copper, but when strength and a fluid tight joint is the main concern then the steel rivets are used.	1.5m 1.5m 1m	**4m**
e	**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 expaina tion	**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 diagram	
f	Stresses in Pipes:The stresses in pipes due to the internal fluid pressure are determined by Lame's equation. Following are the stresses 1)According to Lame's equation, tangential stress at any radius x $\sigma t = \{[p\,(ri)^2] / [(ro)^2 - (ri)^2]\} / \{1 + [(ro)^2 / x^2]\}$ 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	2m 2m	**4m**
g	**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	4 marks 1M EACH POINT	**4m**

2.	**Attempt any TWO of the following:**		8X2=16
a	Given $W = 15 kN = 15\times10^3 N$ $d = 80 mm$, $y = 140 mm$, $x = 120 mm$ Bending Moment at the centre of the Crankshaft bearing $M = W\times x = 15\times10^3\times120 = 1.8\times10^6 N\cdot mm$ Torque transmitted $T = W\times y = 15\times10^3\times140 = 2.1\times10^6 N\cdot mm$ Bending stress induced. $\sigma_b = \frac{M}{Z} = \frac{1.8\times10^6}{\frac{\pi}{32}d^3}$ $= \frac{32\times1.8\times10^6}{\pi(80)^3}$ $\boxed{\sigma_b = 35.8 N/mm^2}$ Shear stress $\tau = \frac{T}{\frac{\pi}{16}d^3} = \frac{16\times2.1\times10^6}{\pi(80)^3}$ $\boxed{\tau = 20.9 N/mm^2}$ Maximum Principal Stress $\sigma_{t max} = \frac{\sigma_t}{2} + \frac{1}{2}\left[\sqrt{\sigma_t^2 + 4\tau^2}\right]$ $= \frac{35.8}{2} + \frac{1}{2}\left[\sqrt{(35.8)^2 + 4(20.9)^2}\right]$ $\sigma_{t max} = 45.4 N/mm^2$	1m for bending moment 1m for torque 1m for bending stress 1 m for shear stress 2m	**8m**

	Maximum shear stress $\tau_{max} = \frac{1}{2}\left[\sqrt{\sigma_t^2 + 4\tau^2}\right]$ $= \frac{1}{2}\left[\sqrt{(35.8)^2 + 4(20.9)^2}\right]$ $\tau_{max} = 27.5\ N/mm^2$	2m	
b)	d = diameter of shaft D = diameter of hub = 2 d d_1 = Nominal dia of bolt D_1 = diameter of bolt circle = 3 d n = no of bolts t_f = thickness of flange = 0.5 d $\tau_s, \tau_b, \& \tau_k$ = 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 $T = \frac{\pi}{16} \tau_c D^3 (1 - K^4)$ ... ① where $k = \frac{D}{d}$ here D = 2 d & L = 1.5 d. From equation ① the diameter of hub can be checked If $\tau_c < \tau_{given}$ design is safe 2) Design of key $w = \frac{d}{4}$ $t = \frac{d}{6}$ $l = L = 1.5 d$ 3) Design for flange $T = \pi \times D \times t_f \times \tau_c \times \frac{D}{2}$ here t_f = 0.5 d. In above equation if $\tau_c < \tau_{given}$ design safe 4) Design of bolts 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	**08M**

<table>
<tr><td>c) i)</td><td>

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,
$l_a \times f \times a = l_b \times f \times b$
or $l_a \times a = l_b \times b$ ……..(i)
We know that
$l = l_a + l_b$ …….(ii)
From equations (i) and (ii), we have
$l_a = \frac{l x b}{(a+b)}$ &

$l_b = \frac{l x a}{(a+b)}$</td><td>1m dia

3m derivation</td><td>04 marks</td></tr>
</table>

C ii)	Given $a+b = 200\,mm$ $\tau = 75\,N/mm^2$ $W = P = 200\times10^3 N$ $t = S = 10\,mm$ For single Tranuverse Fillet weld. $W = 0.707 \times S \times l \times \tau$ $200\times10^3 = 0.707\times10\times l\times75$ $200\times10^3 = 530.25\,l$ $\therefore \boxed{l = 378\,mm}$ $\therefore l_a + l_b = l = 378\,mm$ To Find position of centroidal axis. As per given diagram $b = 55.3\,mm$ $\therefore a = 200 - 55.3$ $a = 144.7\,mm$ Weld length at the top section $l_a = \frac{l\times b}{a+b}$ $= \frac{378\times55.3}{200}$ $= 104.51\,mm$ $\boxed{l_a \approx 105\,mm}$ $\therefore$ weld length at the bottom section $l_b = l - l_a$ $= 378 - 105$ $\boxed{l_b = 273\,mm}$	1m 1m 1m 1m	04

3.	Attempt any TWO of the following:		2X8=16
a	1) Thickness of boiler shell $t = \frac{P \cdot D}{2.6t} + 1\ mm$	1m	08m
	2) Diameter of Rivet $d = 6\sqrt{t}$	1m	
	3) Number of Rivet Shearing Resistance of rivet $P_s = n \times \frac{\pi}{4} \times d^2 \times \tau \quad \cdots (1)$ Total shearing load $W_s = \frac{\pi}{4} \times D^2 \times P \quad \cdots (2)$ equating (1) + (2) no of rivets can be found.	2m	
	4) Pitch of Rivet $\eta_c = \frac{p_1 - d}{p_1}$	1m	
	5) Number of Rows $= \frac{\text{Total no. of Rivets}}{\text{No. of Rivets in one row}}$	1m	
	6) Margin $m = 1.5d$	1m	
	7) Strap Thickness: 0.625 t	1m	

b	Given $l = 500\,mm$, $l_1 = 50\,mm$ $l_2 = 400\,mm$ $W = 30\times10^3 N$ $\sigma_t = 70 N/mm^2$ $n = 4$. 1) Direct shear load. $W_s = \frac{W}{n} = \frac{30\times10^3}{4} = 7.5\times10^3 N$. 2) Maximum tensile load. $W_t = \frac{W \cdot l \cdot l_2}{2[l_1^2 + l_2^2]} = \frac{30\times10^3 \times 500 \times 400}{2[50^2 + 400^2]} = \frac{6\times10^9}{325\times10^3}$ $W_t = 18.46\times10^3 N$ 3) Equivalant load $W_{te} = \frac{1}{2}\left[W_t + \sqrt{(W_t)^2 + 4(W_s)^2}\right]$ $= \frac{1}{2}\left[18.46\times10^3 + \sqrt{(18.46\times10^3)^2 + 4(75\times10^3)^2}\right]$ $= \frac{1}{2}\left[18.46\times10^3 + 151.13\times10^3\right]$ $W_{te} = 84.79\times10^3 N$ 4) Core diameter & Nominal diameter of bolt. $\sigma_t = \frac{W_{te}}{\frac{\pi}{4} d_c^2}$ $\therefore W_{te} = \sigma_t \cdot \frac{\pi}{4} d_c^2$ $\Rightarrow 84.79\times10^3 = 70\times\frac{\pi}{4}\cdot d_c^2$ $\boxed{d_c = 39.27 \approx 40\,mm}$ $d_c = 0.84\, d_o$ $\therefore d_o = \frac{d_c}{0.84} = \frac{40}{0.84}$ $\boxed{d_o = 47.61 \approx 48\,mm}$	02 m for direct shear load 2 m for tensile load 2m equivalent load 2m for size of bolt	**8m**

C	Given $D = 50mm$, $R = 25mm$, $p = 7\ N/mm^2$ $\sigma_t = 20\ N/mm^2$, $\sigma_{tb} = 60\ N/mm^2$ Using Lame's equation $t = R\left[\sqrt{\frac{\sigma_t + p}{\sigma_t - p}} - 1\right] = 25\left[\sqrt{\frac{20+7}{20-7}} - 1\right]$ $\boxed{t = 11.03 \simeq 12\ mm}$ Assuming width of Packing = 10 mm $\therefore D_1 = D + 2 \times (\text{width of Packing})$ $= 50 + 2\times10$ $\boxed{D_1 = 70\ mm}$ Force Trying to Seperate the flange $F = \frac{\pi}{4}(D_1)^2 . p = \frac{\pi}{4}(70)^2 . 7$ $\boxed{F = 26943 N}$ Load on each bolt. $F_b = \frac{F}{2} = 13471.5 N$ $\therefore F_b = \frac{\pi}{4}(dc)^2 \times \sigma_{tb} = \frac{\pi}{4}(dc)^2 \times 60$ $\therefore dc = 16.9 mm$ $\boxed{dc \approx 17 mm}$ & Nominal diameter $d = \frac{dc}{0.85} = \frac{17}{0.85}$ $\boxed{d = 20.2 = 22 mm}$	1m given 2m 1m 1m 2m 1m	**8M**
4.	**Attempt any TWO of the following:**		2X8=16

a)	**Procedure:-** 1. Find the angles between the members at required points. 2. In case of simply supported frame find support reactions by using COE. 3. Isolate the joint from its parent truss in such a way that it should not carry more than two unknown members. 4. Draw the Free body diagram of that joint showing nature of member forces as pull type. 5. Assume that if the entiret russ is in equilibrium then that isolated joint must be in equilibrium. 6. Apply COE to that joint and find the nature and magnitude of forces. 7. Repeat the same process for all the joints. **8.** Tabulate the result.	1m for each step (students may club the points.. kindly give marks accordingly)	**8m**
b **i)**	**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.	2m 2m Any 2	**4m**
b ii)	Stress Concentration. Whenever a machine component changes the shape of its cross-section, the simple stress distribution no longer holds good and the neighbourhood of the discontinuity is different. This irregularity in the stress distribution caused by abrupt changes of form is called stress concentration. It occurs for all kinds of stresses in the presence of fillets, notches, holes, keyways, splines, surface roughness 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.	2m 2m any 2	**4m**

c	Given $W_p = 200N$ $D_p = 200mm$ $R_p = 100mm$ $P = 1\times10^3 W.$ $\tau = 35\ N/mm^2$ $N = 120rpm$ $\theta = 180^\circ = 1\ Rad.$ $\mu = 0.3$ $K_m = 1.5$ $K_t = 2$  1) To Find Twisting Moment $P = \frac{2\pi N T}{60}$ $\therefore T = \frac{60P}{2\pi N}$ $T = \frac{60\times1\times10^3}{2\times\pi\times120} = 79.57\ N.m = 79.57\times10^3\ N.mm$ 2) To Find T_1 & T_2 We know $\frac{T_1}{T_2} = e^{\mu\theta}$ $\therefore \frac{T_1}{T_2} = e^{0.3.1}$ $\frac{T_1}{T_2} = 1.34$ $\therefore T_1 = 1.34\,T_2$ We also know Torque Transmitted by pulley $T = (T_1 - T_2).R_p$ $\therefore 79.57\times10^3 = (1.34T_2 - T_2).100$ $T_2 = 2.34\times10^3 N$ $T_1 = 1.34\times2.34\times10^3 = 3.13\times10^3 N$ $\therefore$ Total load acting on pulley W. $W = W_p + T_1 + T_2$ $= 200 + 2.34\times10^3 + 3.13\times10^3$ $W = 5670N.$	4.c 1m for torque 2m for T_1 & T_2 1m for total load	**8m**

	9.c 3) To Find Bending Moment here $M = W \times 800 = 5670 \times 800$ $M = 1.70 \times 10^6$ N.mm 4) Equivalant Twisting load. $T_e = \sqrt{k_m . M^2 + k_t . T^2}$ $= \sqrt{1.5 . (1.70 \times 10^6)^2 + 2 (79.57 \times 10^3)^2}$ $= \sqrt{4.33 \times 10^{12} + 1.26 \times 10^{10}}$ $T_e = 2.08 \times 10^6$ Equating $T_e = \frac{\pi}{16} \tau d^3$ $2.08 \times 10^6 = \frac{\pi}{16} \times 35 . d^3$ $d = 67.14$ mm $\boxed{d \approx 68 \text{mm.}}$	1m for bending moment 2m 1m	

5.	Attempt any TWO of the following:		2X8=16
a	Given $P = 15\times10^3\,W$, $N = 960\,rpm$, $d = 40\,mm$ $l = 75\,mm$, $\tau = 56\,N/mm^2$ $\sigma_{ck} = 112\,N/mm^2$	1m given	**8m**
	Torque Transmitted $P = \frac{2\pi N T}{60}$ $\therefore T = \frac{60P}{2\pi N}$ $T = \frac{60\times15\times10^3}{2\cdot\pi\times960} = 149\,N.m$ $T = 149\times10^3\,N.mm$	1m torue	
	Width and thickness of key As $\sigma_{ck} = 2\tau$ the key must be square For square key $w = \frac{d}{4} = \frac{40}{4} = 10\,mm$ $t = \frac{d}{4} = \frac{40}{4} = 10\,mm$	2m	
	Accordding to H.F Moore $e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$ $e = 1 - 0.2\left(\frac{10}{40}\right) - 1.1\left(\frac{10}{40}\right)$ $e = 0.8125$	1m	
	Shear strength of key. $= l\times w\times\tau\times\frac{d}{2} = 75\times10\times56\times\frac{40}{2}$ $= 840000\,N$	1m	
	Normal strength of shaft. $= \frac{\pi}{16}\tau d^3\cdot e = \frac{\pi}{16}\times56\times(40)^3\times0.8125$ $= 571844$	1m	

	$\frac{\text{Shear Strength of key}}{\text{Normal Strength of shaft}} = \frac{840000}{571844} = 1.47$	1m	
b	5.6 Given Width = 75mm, $t = S = 12.5$mm, $\sigma_t = 70$ N/mm² $\tau = 56$ N/mm². Total load Carried by plate W = Area × Max Stress $= (75 \times 12.5) \cdot 70$ $\boxed{W = 65.62 \times 10^3 N}$ For Static Loading - Case No. 1 here l_1 = width - 12.5 $l_1 = 75 - 12.5$ $l_1 = 62.5$ mm Tensile Strength of plate $W_{\sigma t} = 0.707 \times S \times l_1 \times \sigma_t$ $= 0.707 \times 12.5 \times 62.5 \times 70$ $W_{\sigma t} = 38.66 \times 10^3 N$. Shear strength of plate $W_S = 2 \times 0.707 \times S \times l_2 \times \tau$ $= 2 \times 0.707 \times 12.5 \times l_2 \times 56$ $= 989.8\, l_2$ We know Total load $W = W_{\sigma t} + W_T$ $65.62 \times 10^3 = 38.66 \times 10^3 + 989.8\, l_2$ $\therefore l_2 = \frac{65.62 \times 10^3 - 38.66 \times 10^3}{989.8}$ $\boxed{l_2 = 27.23 \text{ mm}}$ Adding 12.5mm for Starting & stopping weld run $\therefore l_2 = 27.23 + 12.5 = 39.73$ mm	1m 1m 1m 1m	8M

For Dynamic Loading - Case II

Actual working stresses.

$$\sigma_{t\,actual} = \frac{\sigma_{t\,given}}{S.C.F} = \frac{70}{1.5} = 46.66\ N/mm^2$$

1m

$$\tau_{actual} = \frac{\tau_{given}}{S.C.F} = \frac{56}{2.7} = 20.74\ N/mm^2$$

Tearing strength

$$W_{\sigma t} = 0.707 \times s \times l_1 \times \sigma_{t\,actual}$$
$$= 0.707 \times 12.5 \times 62.5 \times 46.66$$
$$W_{\sigma t} = 25.77 \times 10^3\ N.$$

1m

Shearing strength

$$W_\tau = 2 \times 0.707 \times s \times l_2 \times \tau_{actual}$$
$$= 2 \times 0.707 \times 12.5 \times l_2 \times 20.74$$
$$W_\tau = 366.57\, l_2$$

1m

Total Strength

$$W = W_{\sigma t} + W_\tau$$
$$65.62 \times 10^3 = 25.77 \times 10^3 + 366.57\, l_2$$
$$l_2 = \frac{65.62 \times 10^3 - 25.77 \times 10^3}{366.57\, l_2}$$

1m

$$\boxed{l_2 = 108.71\,mm}$$

For starting & stopping of weld run

$$l_2 = 108.71 + 12.5$$

$$\boxed{l_2 = 121.21\,mm}$$

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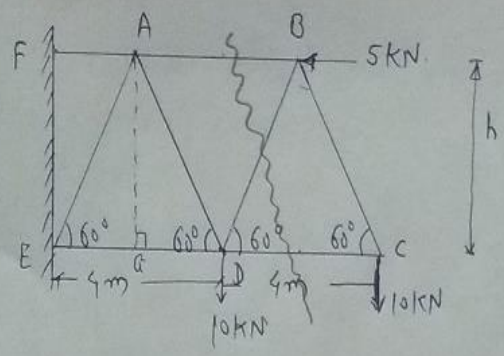
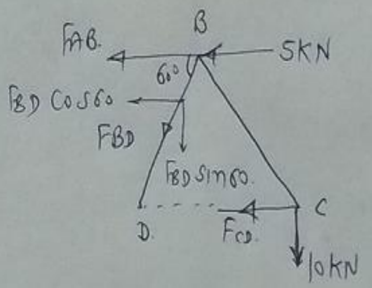
<td>


To Find h.
Consider a Right angle ΔAGE
$\therefore h = \tan 60 \cdot (2)$
$h = 3.46\,m$
Taking a section line which pass through members AB, BD & CD. Draw FBD of truss on R.H.S. of section.


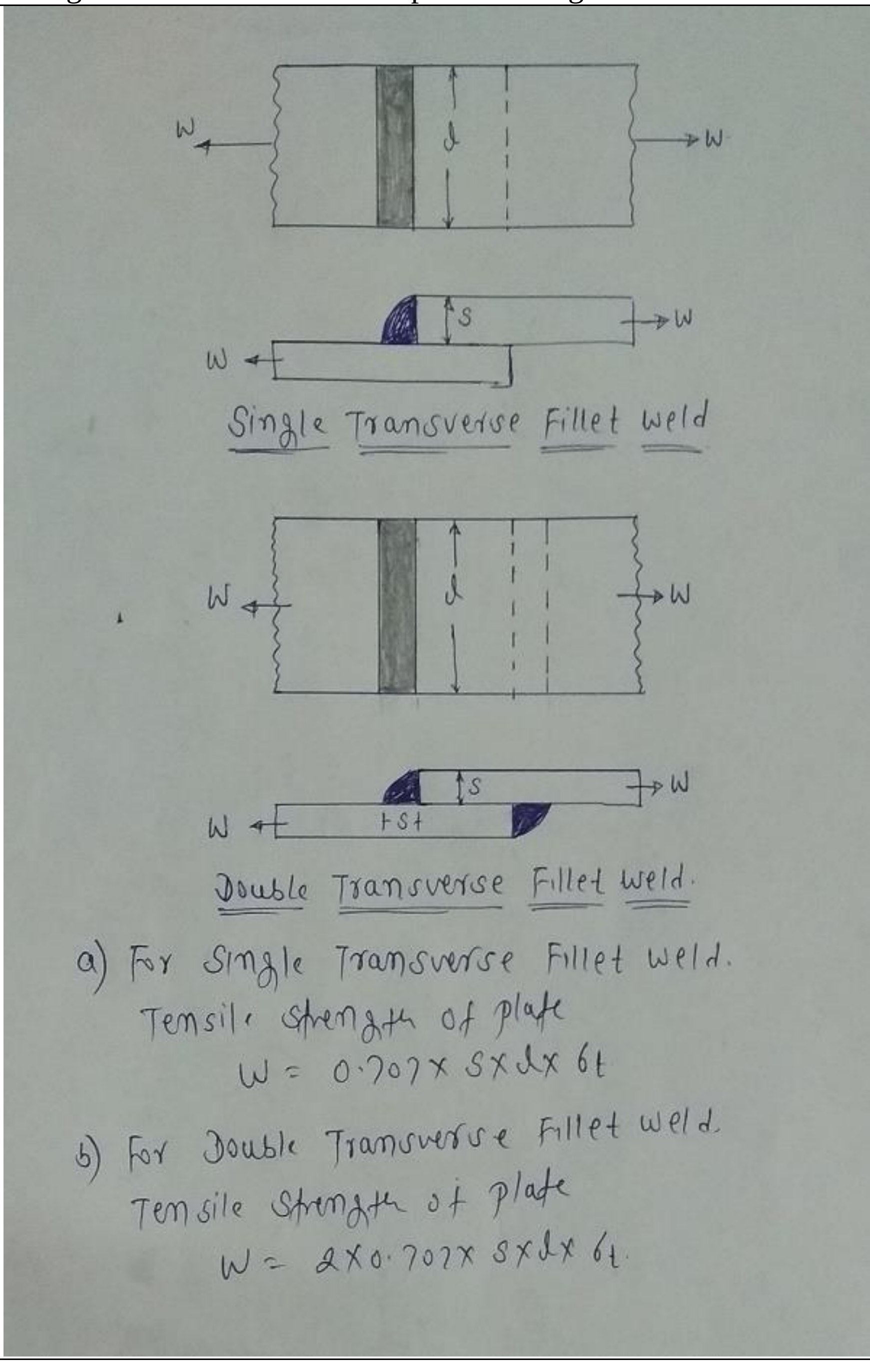
Taking $\Sigma M_D = 0$
$-(F_{AB} \times 3.46) - (5 \times 3.46) + (10 \times 4) = 0$
$\boxed{F_{AB} = 6.55\,kN}$ Tensile
Taking $\Sigma M_B = 0$
$+(10 \times 2) + (F_{CD} \times 3.46) = 0$
$\boxed{F_{CD} = -5.78\,kN}$ Compressive
Taking $\Sigma F_Y = 0$
$-10 - F_{BD} \sin 60 = 0$
$\boxed{F_{BD} = -11.54\,kN}$ Compressive
</td>
<td>1m for height

1m for FBD

2m

2m

2m</td>
<td>8M</td>
</tr>
</table>

6.	Attempt any FOUR of the following:		4X4=16
a	i) 30 Ni4 Cr1:- It is a high carbon steel which contains 0.3 % carbon, 4 % Nickel, and 1% chromium. ii) SG 400/12:- It is a spheroidal graphite cast iron which has a minimum tensile strengrth of 400 N/mm^2 and 12 percent elongation.	2m 2m	**4m**
b		1m 1m 1m 1m	**4m**

c	**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 **Methods ofAnalysis:-** 1) Method of joints 2) Method of sections	2m 2m For method name	**4m**
d	Couplings are used to transmit power when there it is required. Couplings tend to produce unbalanced forces due to misalignments of shafts which cause vibrations in rotating machinery. How much ever the bearing is closer to coupling, the lesser overhang there is . Leading to lower amplitude of vibrations.	4m	**4m**
e	Given $t = 13\,mm$, $\sigma_t = 80\,N/mm^2$ $\tau = 60\,N/mm^2$ $\sigma_{ck} = 120\,N/mm^2$ Diameter of Rivet $d = 6\sqrt{t} = 6\sqrt{13}$ $\boxed{d = 21.6\,mm \approx 23\,mm}$ Pitch of Rivets $P_t = (p-d)\cdot t\cdot\sigma_t$ $= (p-23)\cdot 13\cdot 80$ $= (p-23)\cdot 1040$ --- (1) $P_s = n \times \frac{\pi}{4} \times d^2 \times \tau$ $= 2 \times \frac{\pi}{4} \times (23)^2 \times 60$ $= 49864\,N$ . . . (2) equating (1) & (2) $(p-23)\cdot 1040 = 49864$ $\boxed{p = 71\,mm}$ Maximum pitch $p_{max} = C\cdot t + 41.28$ $p_{max} = 2.62 \times 13 + 41.28$ $\boxed{p_{max} = 75.28\,mm}$ $\therefore$ Pitch $p = 76\,mm$ Tearing Resistance (P_t) $P_t = (p-d)\cdot t\cdot\sigma_t$ $= (76-23)\cdot 13\cdot 80$ $P_t = 55120\,N.$	1m given 1m 1m	**4m**

	Shearing Resistance $\underline{P_s} = \underline{49864\ N.}$ Crushing Resistance $P_c = n.d.t.\sigma_{ck}$ $= 2 \times 23 \times 13 \times 120$ $\underline{P_c} = \underline{71760\ N.}$ Strength of unriveted plate $P = p.t.\sigma_t$ $= 76 \times 13 \times 80$ $\underline{P.} = \underline{79040\ N.}$ Efficiency $\eta = \frac{P_s}{P} = \frac{49864}{79040}$ $\boxed{\eta = 63\%}$	1m	
f	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 initial tension in a bolt, based on experiments, may be found by the relation Pi = 2840dN Pi = Initial tension in a bolt, and	1m for each stress any 4 stresses .	**4m**

	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 =Ts/r Ts = T/J x r ={ T/(π/32) x dc4 } x { dc /2} = 16 T/ π(dc)3 Where Ts = Torsional shear stress, T = Torque applied, and dc= Minor or core diameter of thread 3.Shear stress across the threads. The average thread shearing stress for the screw(Ts) is obtained by using the relation: Ts = p/(π dc x b x n) Where b = Width of the thread section at the root. The average thread shearing stress for the nut is Tn = p/(π d x b x n) Where d = Major diameter. 4. Compression or crushing Stress on threads. The compression or crushing stress between the threads (Ϭc) may be obtained by using the relation : Ϭc= p/ π[d2 – (dc)2]n Where d = Major diameter, dc = 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 (Ϭb) induced in the shank of the bolt is given by Ϭb = 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.		