Temporary Joints

1. Bolted Joint including eccentric loading
2. Cotter Joint
3. Knuckle Joint

Permanent Joints

1. Riveted Joint
2. Welded Joint with including eccentric loading

Bolted Joints:

A steam engine cylinder has an effective diameter of 350 mm and the maximum steam pressure of 1.25 Mpa. Calculate the number and size of studs required to fix the cylinder cover, assuming the permissible stress in the studs as 33 Mpa.

Solution:

\[
D = 350 \text{ mm} \quad P = 1.25 \text{ Mpa} \\
\sigma_b = 33 \text{ Mpa}
\]

Upward force acting on the cylinder cover

\[
F = \frac{\pi}{4} D^2 \times P = \frac{\pi}{4} \times (350)^2 \times 1.25
\]

\[
F = 120256.5 \text{ N}
\]

Assume \( d = 24 \text{ mm} \) (nominal diameter)

Resisting force offered by \( n \) no of studs; \( d_c = 20.32 \text{ mm} \) for 8 g studs

\[
F = \frac{\pi}{4} \times (d_c)^2 \times \sigma \times n
\]

\[
F = \frac{\pi}{4} \times (20.32)^2 \times 33 \times n = 10700 \times n \text{ N}
\]

\[
F = 10700 \times n \text{ N}
\]
now \( 12.0 \times \beta_5 = 10700 \) 
\[ n = 11.24 \]
\[ \therefore \ n = 12 \]

\[ \text{The No. of Sheds Used Are} \]
\[ \n = 12 \]

\[ \text{Pitch Circle Diameter} \]
\[ D_p = D + 2t + 3d \]
\[ \therefore D_p = 285 + \phi \times (50) + 3(25) \]
\[ \therefore D_p = 385 \text{ mm} \]

\[ \text{Circumferential Pitch} \]
\[ P_c = \frac{\pi \times D_p}{n} = \frac{\pi \times 385}{12} = 116.5 \text{ mm} \]

\[ \text{Note: For leak proof Joint} P_c \text{ must be} \ t/4 \]
\[ 20 \phi d_i \leq 30 \sqrt{d_i} \]
\[ 20 \times \phi \times 25 \leq 150 \text{ mm} \]
\[ 20 \times 25 = 150 \text{ mm} \]
\[ \text{we get} \ P_c = 116.5 \text{ mm} \]

\[ \therefore \text{Size of the Shed} = \text{M 84} \]
The cylinder head of a steam engine is subjected to a steam pressure of 0.7 MPa. It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of the cylinder is 300 mm. Find the size of the bolts so that the shear in the bolt is not to exceed 0.1 MPa.

Solution

\[ d = 300 \text{ mm}, \quad P = 0.7 \text{ MPa}, \quad n = 12 \]

\[ \sigma_{\text{permis}} = 0.1 \text{ MPa}. \]

Upward force acting on the cylinder head

\[ F = \frac{\pi}{4} P d^2 \frac{h}{p} = \frac{\pi}{4} \left( \frac{300}{2} \right)^2 \times 0.7 = 49490 \text{ N} \]

\[ \therefore F = 49490 \text{ N} \]

For 1 bolt

\[ F = \frac{49490}{12} = 4124 \text{ N} \]

Critical tension due to tightening of bolt:

\[ F_{\text{crit}} = 2840d \text{ N} \]

\[ k = 0.5 \] (for soft copper base)

Now, resultant Axial load

\[ R = F_{\text{crit}} + k F \]

Also

\[ \frac{\pi}{4} (d_0)^2 \sigma_x = \frac{4k}{u} + \frac{1}{3} (2490d + 0.5 \times 4124) \]

Size of

\[ \frac{\pi}{4} (d_0)^2 \sigma_x = \frac{4k}{u} + \frac{1}{3} (2490d + 0.5 \times 4124) \]

\[ \Rightarrow d_0 = 52 \text{ mm} \]

\[ d = 52 \text{ mm} \]
Bolted joints with eccentric loading:

1. Eccentric load parallel to the axis of the bolts,
2. Eccentric load perpendicular to the axis of the bolts,
3. Eccentric load in the plane containing the bolts.

Eccentric load parallel to the axis of bolt:

A bracket is shown in Figure, supports a load of 30kN. Determine the size of bolts, if the maximum allowable tensile stress in the bolt material is 60 MPa. The distances are: $L_1 = 80\text{mm}$, $L_2 = 250\text{mm}$, $L = 50\text{mm}$.

Solution:

Directed tensile load carried by each bolt:

$$W_{t1} = \frac{W}{n} = \frac{30 \times 10^3}{4} = 7500 \text{ N}$$

Load in a bolt per unit distance:

$$w = \frac{W \times L}{2 \left[ L_1^2 + L_2^2 \right]} = \frac{30 \times 10^3 \times 500}{2 \left[ 80^2 + 250^2 \right]} = 109 \text{ N/mm}$$

Load on heavily loaded bolt:

$$W_{t2} = w \times L_2 = 109 \times 250 = 27250 \text{ N}$$

Max. Tensile load on heavily loaded bolt:

$$W_{t2} = \max. \text{Tensile load on heavily loaded bolt} = 34750 \text{ N}$$
\[ W_t = \frac{\pi}{4} (d_c)^2 \times \delta (\text{tension}) \]

347.50 = \frac{\pi}{4} (d_c)^2 \times 60

\[ d_c = 28.2 \text{ mm} \]

From DBSE,

standard cure dia \( d_c = 28.706 \text{ mm} \)

and the corresponding size of the bolt is M33.

.: Size of the bolt is M33.

**Eccentric Load perpendicular to the axis of bolt:**

For supporting the travelling crane in a workshop, the brackets are fixed on steel columns as shown in figure. The maximum load that comes on the bracket is taken as:

volutically at a distance of 400 mm from the face of the column. The vertical bracket is secured to a column by four bolts. Determine the size of the bolt. Allowable tension stress is 800. Also find the cross-section of the column of the bracket which is rectangular.

\[ 400 \text{ mm} \]

\[ 121 \text{ mm} \]

\[ 375 \text{ mm} \]

\[ 120 \text{ mm} \]
direct shear load on each bolt:

\[ W_s = \frac{W}{n} = \frac{12400 \times 3}{4} = 3000 \text{ N} \]

Net tensile load carried by the bolts:

\[ W_t = \frac{W_xL_xL_2}{2[L_1^2+L_2^2]} = \frac{12400 \times 400 \times 345}{2[53^2+29^2]} \]

\[ W_t = 6290 \text{ N} \]

Equivalent tensile load:

\[ W_{te} = \frac{1}{2} \left[ W_t + \sqrt{W_t^2+4W_s^2} \right] \]

\[ W_{te} = \frac{1}{2} \left[ 6290 + \sqrt{6290^2+4 \times 3000^2} \right] \]

\[ W_{te} = 7490 \text{ N} \]

Size of bolt:

\[ W_{te} = \frac{\pi}{4} \left( d_c \right)^2 \times 84 \]

\[ 7490 = \frac{\pi}{4} \left( d_c \right)^2 \times 84 \]

\[ d_c = 10.65 \text{ mm} \]

From Boeing standards:

- Standard core diameter, \( d_c = 11.545 \text{ mm} \)
- Corresponding bolt size = M10

Cross section of bolt:

\[ \sigma_c \text{ (in mm) } = \frac{M}{2} \times \frac{1}{t b^2} = \frac{12400^2 \times 400 \times 3}{12 \times 2.5^2 \times 84} \]

Assume \( t = 34300 \)

\[ b = 2.5 \text{ mm} \]

When \( b = 2.5 \text{ mm} \), then \( t = 5.5 \text{ mm} \).
Eccentric load in the plane containing the bolts:

The figure shows a solid forged bracket to carry a vertical load of 13.5 kN applied through the centre of the hole. The square flange is secured to the flat side of a vertical stanchion through four bolts. Calculate the suitable diameter \( D \) and \( d \) for the arms of the bracket if the permissible stresses are 110 MPa in tension and 65 MPa in shear.

\[ M = 13500 \times (300 - 25) \]
\[ M = 3712.5 \times 10^3 \text{ Nmm} \]

\[ T = 13500 \times 250 \]
\[ T = 3375 \times 10^3 \text{ Nmm} \]

Equivalent twisting moment
\[ T_e = \sqrt{M^2 + T^2} \]
\[ T_e = \sqrt{3712.5^2 + 3375^2} \text{ Nmm} \]
\[ T = \frac{m}{16} \xi D^3 \]

\[ \text{so that } \text{w} = \frac{m}{16} (\xi D^3) \]

\[ D = 73.24 \text{ mm} \]

Take \( D = 74 \text{ mm} \)

Diameter (d) of the arm of the brace:

\[ M = 13500 \left( 250 - \frac{75}{2} \right) \]

\[ M = 2868.8 \times 10^{-3} \text{ Nmm} \]

\[ \sigma_b = \frac{M_y}{I} \]

\[ 110 = \frac{2868.8 \times 10^{-3} \times d}{\frac{\pi}{64} d^4} \]

\[ d = 64.3 \text{ mm} \]

Take \( d = 64 \text{ mm} \)
Cotton Joints:

1. Square and Spigot Cotton joint
2. Sleeve and Cotton joint

Square and Spigot Cotton joint:

Design a cotton joint to support a load varying from 30 kN (compression) to 30 kN (tension). The material used here is carbon steel.

The following data can be considered:

- Allowable Tensile Stress = 50 MPa
- Allowable Compressive Stress = 50 MPa
- Allowable Shear Stress = 35 MPa
- Allowable Yielding Stress = 90 MPa

**Step 1:**

Diameter of rod (d)

Consider tension in rod

Tensile load = \( \frac{\pi}{4} \times d^2 \times \sigma \) (in kN)

30 \times 10^3 = \( \frac{\pi}{4} \times d^2 \times 50 \)

d = 25.16 mm

Take \( d = 25 \text{ mm} \)
Step 2: Diameter of spigot \( d_2 \) and thickness of cotton:

Consider Tension in spigot.

\[
\text{Tensile load} = \left( \frac{\pi}{4} (d_2^2 - d_2 t) \right) \sigma \text{ (perm)}
\]

\[
20 \pi w_0^3 = \left( \frac{\pi}{4} d_2^2 - d_2 \frac{d_2 t}{4} \right) (50)
\]

\[
t = \frac{d_2}{4}
\]

\[d_2 = 38.4 \text{ mm}\]

Take \( d_2 = 34 \text{ mm} \)

and thickness of cotton

\[t = \frac{d_2}{4} = 8.5 \text{ mm}\]

Check for crushing stress.

\[
\text{Compressive load} = d_2 \times \sigma_c
\]

\[
20 w_0^3 = \frac{\pi}{4} x 8.5 \times \sigma_c
\]

\[
\sigma_c = 103.8 \text{ N/mm}^2
\]

\[
\sigma_c \text{ (perm)} = 90 \text{ mpa}
\]

\[
\text{Crushing stress exceeds}
\]

Find suitable \( d_2 \) and \( t \) by asking \( \sigma_c = 90 \text{ mpa} \)

\[
20 w_0^3 = d_2 \times \frac{d_2 t}{4} \times 90
\]

\[d_2 = 36.5 \text{ mm} \]

\[t = \frac{3d_2}{4} = 9.5 \text{ mm} \]
step 3: Outside diameter of conical \( d_h \)

Tensile load = \[ \left[ \frac{\pi}{4} \left( d_h^2 - d_2 \right) \right] - (d_1 - d_2) \]

so \( \delta \omega = \left[ \frac{\pi}{4} (d_1^2 - 38^2) - (d_1 - 38) \right] \)

\( d_1 = 50.9 \text{ mm} \) \[ \text{true} \]

\( d_1 = 50 \text{ mm} \)

step 4: Width of cotter \( = b \)

cotter is in double shear along the width.

Tensile loading = 2 \((b \times t \times \epsilon \text{ (permissible)})\)

so \( \delta \omega = 2(b \times 9.5 \times 35) \)

\( b = 43.2 \text{ mm} \) \[ \text{true} \]

\( b = 40 \text{ mm} \)

step 5: Diameter of spigot collar \( d_3 \)

Compressive load = \[ \left[ \frac{\pi}{4} (d_3^2 - d_2^2) \right] \]

so \( \delta \omega = \left[ \frac{\pi}{4} (d_3^2 - 38^2) \right] \times 90 \)

\( d_3 = 44.8 \text{ mm} \) \[ \text{true} \]

\( d_3 = 45 \text{ mm} \)
Design a sleeve and cotter joint to resist a tensile load of 60kN. The following values are considered:

- $\sigma_T$ (permissible) = 60 MPa
- $\sigma_E$ (permissible) = 70 MPa
- $\sigma_C$ (permissible) = 125 MPa.

Step 1: Diameter of rod ($d$)

Tensile load = $\frac{\pi}{4} d^2 \times 60$

$60 \times 10^3 = \frac{\pi}{4} d^2 \times 60$

d = 35.7 mm

Take $d = 36$ mm

Step 2: Diameter of enlarged end of rod and thickness of cotter ($t$)

Tensile load = $\frac{\pi}{4} (d_2^2 - d_1^2) \times 60$

$60 \times 10^3 = \left(\frac{\pi}{4} (d_2^2 - d_1^2) \right) \times 60$

$d_2 = 43.2$ mm

$d_1 = 36$ mm

$t = \frac{d_2 - d_1}{2} = \frac{43.2 - 36}{2} = 3.6$ mm

$t = 11$ mm
Design for Clutch

Load = \( d_2 \times r \times \delta_c \)

\[ 60 \times 60^3 = 241 \times 11 \times \delta_c \]

\( \delta_c = 121 \ \text{N/mm} \leq \delta_c \text{ (permissible)} = 125 \ \text{N/mm} \)

Hence design is safe

No modification of \( d_2 \) and \( t \) are needed.

Step 3: Outer diameter of sleeve: \( d_1 \)

\[ \delta \text{allow} = \left[ \frac{121}{\pi} (d_1^2 - d_2^2) - (d_1 - d_2) t \right] \delta_c \text{ (permissible)} \]

\[ 60 \times 60^3 = \left[ \frac{121}{\pi} (d_1^2 - 41^2) - (d_1 - 41) t \right] 60 \]

\( d_1 = 59.4 \ \text{mm} \)

\[ d_1 = 60 \ \text{mm} \]

Step 4: Width of collet is:

Double shear occurs

\[ \delta \text{allow} = 2 \left( 5 \times t \times \delta_c \text{ (allow)} \right) \]

\[ \text{Copper} = 2 \left( 2 \times 11 \times 70 \right) \]

\( b = 39.2 \ \text{mm} \)

\[ b = 40 \ \text{mm} \]
Step 5: Distance of rod from beginning to the cut-off hole.

\[ \text{Double shear:} \]

\[ L_{\text{cut-off}} = 2 \left( a \times d_1 \times \gamma_{\text{Gross}} \right) \]

\[ 60 \times 10^3 = 2 \times a \times 44 \times 70 \]

\[ a = 9.74 \text{ mm} \]

\[ \text{Tune: } a \leq 60 \text{ mm} \]

Step 6: Distance of rod from end to the cut-off hole.

\[ \text{Double shear:} \]

\[ L_{\text{cut-off}} = 2 \left( c \times d_1 \times \gamma_{\text{Gross}} \right) \]

\[ 60 \times 10^3 = 2 \left( 60 \times 44 \right) \times 70 \]

\[ c = 27.609 \text{ mm} \]

\[ \text{Tune: } c \leq 28 \text{ mm} \]
Crib and Court Joint:

Design a Crib and Court Joint to carry a mean load of 85 kN.

\[ \sigma_C = 20 \text{ MPa} \quad I = 15 \text{ MPa} \quad \sigma_C = 50 \text{ MPa} \]

\[ \text{Step 1: Side of the Square Rod: (mm)} \]

\[ \text{Load} = \pi^2 \times \sigma_t \]

\[ 85 \times 10^3 = \pi^2 \times 20 \]

\[ \pi = 4.44 \text{ mm} \]

\[ s = \text{Side of the Square Rod} = 44 \text{ mm} \]

**Other dimensions**

- Width of strap, \( B_1 = s = 44 \text{ mm} \)
- Thickness of cotter, \( t = \frac{B_1}{4} = \frac{44}{4} = 10.5 \text{ mm} \)
- Thickness of crib = Thickness of cotter = 11 mm

**Step 2: Width of Crib and Cotter**

\[ \text{Load} = 2 \times B \times t \times I \]

\[ 85 \times 10^3 = 2 \times 44 \times 11 \times 15 \]

\[ B = 94.2 \text{ mm} \]

\[ t = \text{Width of Cotter} = 0.45 \times 44 = 19.8 \text{ mm} \]

\[ b_1 = \text{Width of Crib} = 0.55 \times 36 = 19.8 \text{ mm} \]
Step 3: Thrust & shear (l1)

Load = 2 (x(t1 - t1) 5) - 35 x 10^3

\[ 2 \left[ 42 \times t_1 - t_1 \times 12 \right] \times 20 \]

\[ t_1 = 29.1 \text{ mm} \]

Load [\( t_1 = 30 \text{ mm} \)]

Check for yielding

Load = 2 x 6 x 6 x 6C

\[ 35 x 10^3 = 2 x 30 x 12 x 6C \]

\[ 6C = 48.6 \text{ N/mm}^2 \]

\[ \leq 50 \text{ MPa} \]

stress (design is safe)

Step 4: Length of the rod (l2)

Double shear

Load = 2 x 6 x 6 x 6C

\[ 35 x 10^3 = 2 x 30 x 4x15 \]

\[ l = 27.7 \text{ mm} \]

\[ \text{Torque} \ f = 28 \text{ mm} \]
Knuckle Joint

Design a knuckle joint for a 40 mm rod of a circular section to sustain a maximum load of 70 kN. The UTS of the material of the rod is 420 MPa. The UTS and ultimate shearing strength of the pin are 510 MPa and 396 MPa, respectively. The factor of safety is 6.

Solution:

Step 1: Failure of the rod in tension:

\[ \sigma = \frac{P}{\pi d^2} \]

\[ 70 \times 10^3 = \frac{\pi d^2 \times (420)}{6} \]

\[ d = 35.7 \text{ mm} \]

True \[ \frac{d}{d} = 36 \text{ mm} \]

Diameter of knuckle pin

\[ d_1 = d = 36 \text{ mm} \]

Outer dia. of eye

\[ d_2 = 2d_1 = 72 \text{ mm} \]

Diameter of knuckle pin head and collar

\[ d_3 = 1.5d_2 = 108 \text{ mm} \]

Thickness of rod and

\[ t = 1.25 d_1 = 1.25 \times 36 = 45 \text{ mm} \]

Thickness of bush

\[ t_1 = 0.75 d_1 = 0.75 \times 36 = 27 \text{ mm} \]
Step 2: Failure of knuckle pin in shear

\[ \text{Load} = \frac{d}{2} \left( \frac{a}{6} \cdot d^2 \times \tau \right) \]

\[ \tau_{\text{max}} = \frac{2 \times \frac{\pi}{5} \times (3b^2) \times \tau}{\pi} \]

\[ \tau = \frac{34.1}{6} \text{ N/mm}^2 < \frac{39.6}{6} = 6.6 \text{ N/mm}^2 \]

Hence design is safe.

Step 3: Failure of rod end (single eye) in tension

\[ \text{Load} = (d_2 - d_1) \times 6 \times \sigma_t \]

\[ \tau_{\text{max}} = (\tau_2 - \tau_1) \times \mu \times 6 \]

\[ \sigma_t = 43.2 \text{ N/mm}^2 < \frac{420}{6} = 70 \text{ N/mm}^2 \]

Hence design is safe.

Step 4: Failure of forged end in tension

\[ \text{Load} = (d_2 - d_1) \times 6 \times \sigma_t \]

\[ \tau_{\text{max}} = (\tau_2 - \tau_1) \times \mu \times 6 \]

\[ \sigma_t = 36 \text{ N/mm}^2 < 70 \text{ N/mm}^2 \]

Hence design is safe.
Welded Joints

Determine the length of the weld run for the plate of size 120 mm wide and 15 mm thick to be welded to another plate by means of:

1. A single transverse weld
2. Double parallel fillet weld when the joint is subjected to variable loads.

\[ \text{Solution:} \]

Let AB represents the single transverse weld. Ac and BD represents the double parallel weld.

Length of weld run for a single transverse weld:

The effective length of the weld run \( l_1 \) may be obtained by subtracting 12.5 mm from the width of the plate weld.

\[ \begin{align*}
  l_1 &= 120 - 12.5 \\
  l_1 &= 107.5 \text{ mm}
\end{align*} \]

Length of the weld run for double parallel fillet weld subjected to variable loads:

\[ l_2 = \text{Length of double parallel weld} \]

\[ S = \text{size of weld} = \text{thickness of plate} = 15 \text{ mm} \]
the maximum load which the plate can carry

\[ \sigma = \frac{P}{bh} \]

\[ P = 120 \times 15 \times 70 \]

\[ = \frac{120 \times 15 \times 70}{8 \times 6} \]

\[ = 126 \times 10^3 \text{ N} \]

Let stress concentration factor for transverse weld = 1.5
and for parallel weld, it is 2.7.

Permissible Tensile Stress

\[ \sigma_{\text{perm}} = \frac{450}{1.5} = 300 \text{ N/mm}^2 \]

Permissible Shear Stress

\[ \tau_{\text{perm}} = \frac{56}{2.7} = 20.7 \text{ N/mm}^2 \]

Load carried by single transverse weld

\[ P_1 = 0.705 \times 5 \times 15 \times \sigma_{\text{perm}} \]

\[ = 53,240 \text{ N} \]

Load carried by double parallel fillet weld

\[ P_2 = 1.414 \times 5 \times \frac{15}{2} \times \tau_{\text{perm}} \]

\[ = 440 \text{ N} \]

Max load = \( P_1 + P_2 \)

\[ 126 \times 10^3 = 53,240 + 440 \]

\[ b_2 = 14.7 \text{ mm} \]

\[ t = 19.8 \text{ mm} \]
A 200 x 150 x 10 mm angle is to be welded to a steel plate by fillet welds as shown in Fig. If the angle is subjected to a static load of 200 kN, find the length of the weld at the top and bottom. The allowable shear stress for static loading may be taken as 75 MPa.

\[
a + b = 200 \text{ mm}
\]

\[
P = 200 \times 10^3 \text{ N}
\]

\[
T = 75 \text{ MPa}.
\]

\[l_a = \text{length of weld at top}\]

\[l_b = \text{length of weld at bottom}\]

\[l = l_a + l_b\]

Thickness of weld: \(s = t = 10 \text{ mm}\)

Maximum load: \(2 \times 10^3 \times 200 \times 10^3 = 750 \times 10^6 \times 75\)

\[l = 377 \text{ mm}\]

\[l_a + l_b = 377 \text{ mm}\]
Let $b =$ distance up centroidal axis from bottom.

\[ b = \frac{(200-10) \times 95 + (150 \times 60 \times 5)}{(90 \times 10) + (150 \times 60)} \]

\[ b = 55.3 \text{ mm} \]

\[ a = 200 - 55.3 = 144.7 \text{ mm} \]

\[ l_a = \frac{d \times b}{a + b} = \frac{37.7 \times 55.3}{200} = 101.2 \text{ mm} \]

\[ l_b = l - l_a = 37.7 - 104.2 = 272.8 \text{ mm} \]

\[ \begin{align*}
  l_a &= 104.2 \text{ mm} \\
  l_b &= 272.8 \text{ mm}
\end{align*} \]
Welded Joint subjected to eccentric loading.

12. A 50mm diameter solid shaft is welded to a flat plate as shown in figure. If the size of the weld is 15 mm, find the maximum normal and shear stress in the weld.

\[
\begin{align*}
D &= 50 \text{ mm} \\
S &= 15 \text{ mm} \\
P &= 10 \times 10^3 \text{ N} \\
e &= 200 \text{ mm} \\
t &= \text{ throat thickness}.
\end{align*}
\]

Throat area, \( A = t \pi D = 0.707 \times 5 \times 50 \)

\[
A = 1666 \text{ mm}^2
\]

Direct shear stress.

\[
\tau = \frac{P}{A} = \frac{10 \times 10^3}{1666} = 6 \text{ N/mm}^2
\]

Bending moment.

\[
M = Pe = 10 \times 10^3 \times 200 = 2 \times 10^6 \text{ Nmm}
\]

Now \( \sigma = \frac{My}{I} = \frac{2 \times 10^6 \times 50}{5 \times 10^7} \geq 96 \text{ N/mm}^2 \)
Maximum Normal Stress

\[ \sigma_{(\text{max})} = \frac{\sigma_b}{2} + \frac{1}{2} \sqrt{\sigma_b^2 + 4\tau^2} \]

\[ = \frac{96}{2} + \frac{1}{2} \sqrt{(96)^2 + 4(6)^2} \]

\[ \sigma_{(\text{max})} = 96.4 \text{ N/mm}^2 \]

Maximum Shear Stress

\[ \tau_{(\text{max})} = \frac{1}{2} \sqrt{\sigma_b^2 + 4\tau^2} \]

\[ = \frac{1}{2} \sqrt{(96)^2 + 4(6)^2} \]

\[ \tau_{(\text{max})} = 48.21 \text{ N/mm}^2 \]

Result

\[ \sigma_{(\text{max})} = 96.4 \text{ N/mm}^2 \]

\[ \tau_{(\text{max})} = 48.21 \text{ N/mm}^2 \]
Riveted Joints for Structures:  [Lozenge Joint]

Two lengths of mild steel tie rod having width 200mm and thickness 12.5 mm are to be connected by means of a butt joint with double cover plates. Design the joint if the permissible stresses are 80 MPA in tension, 65 MPA in shear and 150 MPA in crushing...

Step 1: Diameter of rivet:

Let the diameter of rivet hole be

\[ d = d_{sc} = 6\sqrt{t} = 6\sqrt{12.5} = 21.2 \, \text{mm} \]

Standardised, \( d = 21.5 \, \text{mm} \) and the corresponding diameter of rivet, \( d_r = 20 \, \text{mm} \).

Step 2: Number of rivets:

Max. pull acting on the joint:

\[ P_t = (d - d_r) \times r_t = (200 - 21.5) \times 12.5 \times 80 \]

\[ P_t = 138500 \, \text{N} \]

Shearing resistance of 1 rivet:

\[ P_s = 1.735 \pi t \sqrt{d_r} = 1.735 \times \frac{\pi}{4} \times 21.5 \times 65 \]

\[ P_s = 41300 \, \text{N} \]

Crushing resistance of 1 rivet:

\[ P_c = d_r^2 t_c = 21.5 \times 12.5 \times 1.5 = 43000 \, \text{N} \]
no. of sheets \( n \) = \( \frac{P_L}{P_s} \) = \( \frac{178500}{41300} \) = 4.3 \( \text{Say, } n = 5 \)

Step 3: Thickness \( t_l \) butt strip

\( t_1 = 0.75 \times 6 = 0.75 \times 12.5 \)

\( t_1 = 9.375 \text{ mm} \)

\( |t_1| = 9.4 \text{ mm} \)

Step 4: Efficiency of the joint:

\( \eta = \frac{\text{Strength of the joint}}{\text{Strength of the Unirated plate}} \)

Strength of the joint \( \geq (b - d) \times 6 = 178500 \text{ N} \) (already found)

Strength of the unirated plate

\( \geq 6 \times 6 \times 6 = 200 \times 12.5 \times 80 \)

\( \geq 200000 \text{ N} \)

\( \eta = \frac{178500}{200000} = 0.8925 \)

\( \therefore \eta = 89.25\% \)
Design a lap joint for a mild steel flat tie 100 x 70mm using 24mm diameter rivets. Allowable stresses are 112 Mpa in tension, 200 Mpa in compression and shear stress of the rivets as 84 Mpa. Take diameter of rivet hole as 25.5 mm.

**Solution**

**Step 1: Number of Rivets**

\[ n = \frac{P_t}{P_s} \]

Max pull \( P_t = (b - d) t \sigma_t = (200 - 25.5) \times 100 \times 112 \)

\[ P_t = 195,440 \text{ N} \]

Shearing resistance, \( P_s = \frac{n \pi}{4} d^2 \sigma_s = \frac{n \pi}{4} (25.5)^2 \times 84 \)

\[ P_s = 42905 \text{ N} \]

\[ n = \frac{195,440}{42905} = 4.5 \]

Say \( n = 5 \)

**Step 2: Thickness of Cover Plate**

\[ t_1 = 1.25 t = 1.25 \times 10 = 12.5 \text{ mm} \]

\[ t_1 = 12.5 \text{ mm} \]
Step 3: Efficiency of the joint

\[ \gamma = \frac{\text{Strength of the joint}}{\text{Strength of unnotched plate}}. \]

\[ \text{Strength of Joint} = (b - 3d + 0.5) + Ps \]
\[ = \left( 200 - 3 \times (25.5) \right) + 42055 \]
\[ = 181288 \text{ N} \]

\[ \text{Strength of Joint} = 181288 \text{ N} \]

\[ \text{Strength of Unnotched plate} = 65 \sigma_t = 200 \times 10 \times 12 \]
\[ = 24000 \text{ N} \]

\[ \therefore \gamma = \frac{181288}{240000} \]
\[ \gamma = 0.754 \]

\[ \gamma = 80.74 \]