4.1.Basics of Power Screw

1) What is Power screw? State the Applications of Power screws.

A power screw is a mechanical device used for converting rotary motion into linear motion and transmitting power. The main applications of power screws are,

(i) To raise the load. e.g. screw-jack,
(ii) To obtain accurate motion in machining operations, e.g. lead-screw of lathe,
(iii) To clamp a workpiece. e.g. vice, and
(iv) To load a specimen, e.g. universal testing machine.

2) What are the advantages and disadvantages of Power screw?

Power screws offers following advantages:

(i) Power screws have large load carrying capacity.
(ii) The overall dimensions of power screw are small,resulting in compact construction.
(iii) Power screws are simple to design.
(iv) The manufacturing of power screws is easy without requiring special machinery.
(v) Power screw provides large mechanical advantage. Heavy loads can be lifted with small amount of efforts.
(vi) Most power screws are self locking..they retain the load at its position even when the effort is removed

The disadvantages of power screw are as follows :

(i) Power screw has very low efficiency as low as 40%. Power screws are mainly used for intermittent motion that is occasionally required for lifting the load or actuating the mechanism.
(ii) High friction in threads causes rapid wear of the screw or the nut. Therefore, wear is serious problem in power screws.

3) What are various forms of threads used in power screws? Sketch any two. List properties and application of each type

In power screws following types of threads are used

1. Square threads
2. Trapezoidal threads
3. ACME threads
4. Buttress threads

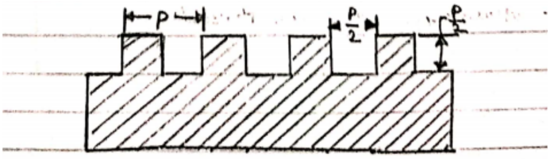


Fig: Square Threads.

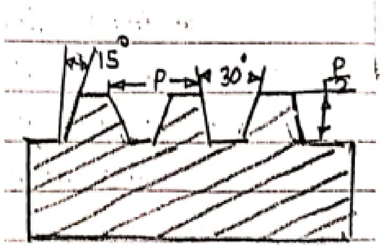


Fig: Trapezoidal Threads.

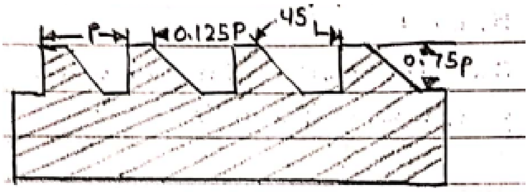


Fig: Buttress Threads.

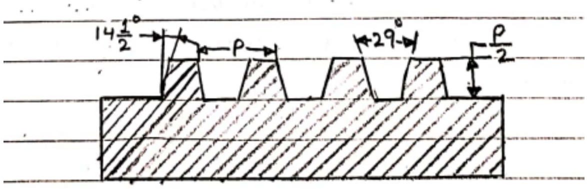
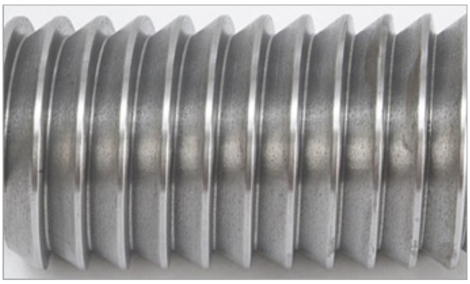
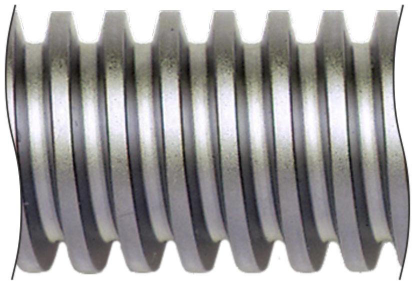


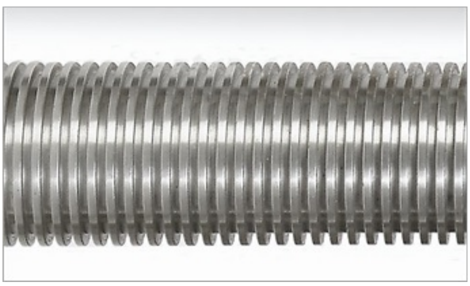
Fig: Acme Threads.



BUTTRESS THREADS



ACME THREADS



SQUARE THREADS

TRAPEZOIDAL THREADS

Properties and applications of various forms of threads

Screw Form	Characteristic	Application
Square Thread	No side thrust Higher efficiency	Used for general purpose power transmission
Trapezoidal Threads	Stronger than square threads Easy to manufacture Wear compensation	Used for higher power transmission
ACME threads	Stronger than square threads Easy to manufacture Wear compensation	Used for higher power transmission
Buttress threads	Can bear very heavy load in one direction	Used to handle heavy forces in one direction, like in truck jack

4) State the Advantages and disadvantages of Square threads over Trapezoidal threads.

The advantages of square threads are as follows:

1) Efficiency of square threads is more than trapezoidal threads
2) There is no side thrust or radial pressure.

The disadvantages of square threads are,

1) Square threads are difficult to manufacture than trapezoidal threads.
2) The wear of square threads can not be compensated as it can be done in trapezoidal.
3) The thread thickness at core is less than trapezoidal, hence square threads have less load carrying capacity.

5) State the Advantages and disadvantages of Trapezoidal threads over Square threads.

The advantages of Trapezoidal threads are as follows:

1) Trapezoidal threads are easy to manufacture.
2) Trapezoidal threads have more thread thickness at root, hence stronger than square threads
3) The axial wear on the surface can be compensated by means of split type nut.

The disadvantages of square threads are,

1) Efficiency of trapezoidal threads is less than square threads.
2) The trapezoidal threads result in side thrust or radial pressure on the nut.

6) Compare the Square and trapezoidal threads used in power screws

Point	Square threads	Trapezoidal threads
Efficiency	More	Less
Side thrust	No	Yes
Manufacturing	Difficult {single point tool}	Easy {Multi Point tool}
Strength	Less {less area at root}	More {more area at root}
Wear	Can not be compensated	Can be compensated

7) Why square threads are preferred over V threads for power transmission.

Square threads are preferred in power transmission over v threads because fo the following reasons,

1) Square threads have maximum efficiency in power transmission
2) Square threads have noiseless operation.
3) Square threads have no side thrust.

So these advantages are the main requirements of application of square threads in power transmission. Hence square threads are preferred over v or trapezoidal threads.

8) What do you mean by self locking and Overhauling.(VVIMP)

The torque required to lower the screw is given by the relation,

$$T = W \tan(\theta - a) \times \frac{d}{2}$$

where W= load on screw

ϕ=Angle of friciton

α= Screw helix angle

d= mean diameter of screw

Now here are two possibilities,

1) $\phi > \alpha$ {Friction angle greater than Helix angle}

In this case the value of T will be positive, means torque is required to be applied to move the load down. Means load will not come down by itself, Such condition is called Self locking condition, and such screw is called SELF LOCKING SCREW. Self locking of screw is not possible when the coefficient of friction is low or the lead of screw is large. Self locking condition is desirable in certain applications like Screw Jack.

2) $\phi<\alpha$ {Friction angle less than Helix angle}

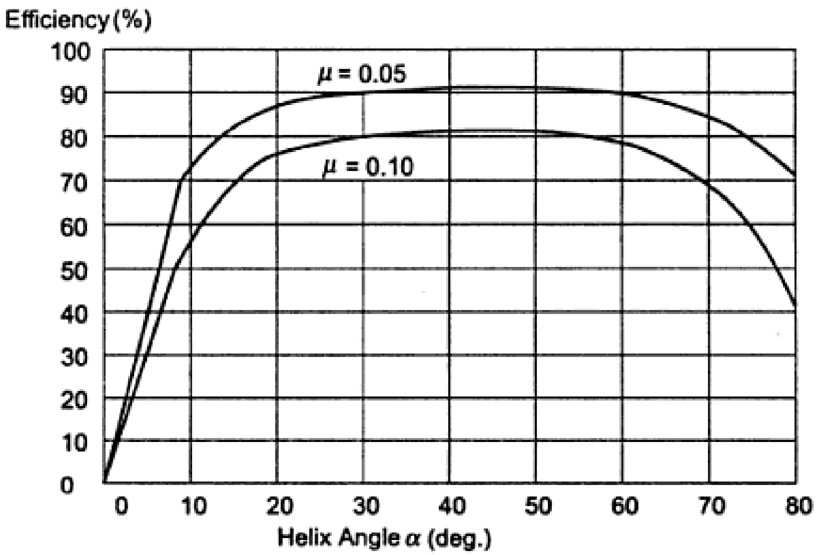
In this case the value of T will be Negative, means torque is not required to be applied to move the load down. Means load will come down by itself, Such condition is called Overhauling and such screw is called overhauling.

9) State and explain the effect of helix angle on efficiency of square threaded screw?

Efficiency of Square threaded screw is given by,

$$n=\frac{\tan a}{\tan(\theta+a)}$$

From equation it is clear that the efficiency of the square threaded screw depends upon the helix angle. The graph below shows the variation in efficiency with change in the helix angle.



From graph it is clear that,

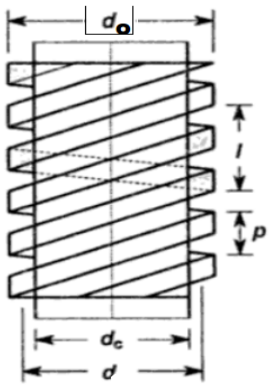
1) Efficiency increases rapidly upto helix angle 20^0.

2) Efficiency is maximum when angle is 40^0-45^0

3) The efficiency decreases rapidly after helix angle exceeds 60^0.

Numericals on Power screw

Formulas and Steps



do= Nominal diameter {Outside diameter of screw}
dc= Core diameter of screw
d= Mean Diameter of the screw
p= Pitch of the screw.

1) Mean diameter of Screw (d)

Mean diameter $d = d_0 - \frac{P}{2}$

core diameter $d_c = d_0 - p$

2) Thread angle (α)

$$\alpha = \tan^{-1}\left(\frac{p \text{ or } 2p \text{ or } 3p}{\pi d}\right)$$

p for single start, 2p for double start 3p for triple start

3) Angle of Friction (φ)

For Square Threads……... $\varphi = \tan^{-1}(\mu)$

For Trapezoidal threads .. $\varphi = \tan^{-1}\left(\frac{\mu}{\cos 15}\right)$

For ACME threads …… $\varphi = \tan^{-1}\left(\frac{\mu}{\cos 14.5}\right)$

4) Torque Required

Torque required to raise load

$$T_s = W\left[\tan(\varphi+\alpha)\frac{d}{2}\right]$$

Torque required to Lower load

$$T_s = W\left[\tan(\varphi-\alpha)\frac{d}{2}\right]$$

5) Collar Friction

$T_c = \mu_c . W . \left(\frac{r_1+r_2}{2}\right) N-mm$ where $\left(\frac{r_1+r_2}{2}\right)$ is mean radius of collar ...Uniform wear

$T_c = \mu_c . W . \left[\frac{2}{3}\left(\frac{r_1^3 - r_1^3}{r_1^2 - r_2^2}\right)\right] N-mm$ ...Uniform Pressure

{If anything is not mentioned we assume uniform wear}

6) Total Torque

$T_{raise} = T_s + T_c$ $T_{Lower} = T_s + T_c$

7) Force/Power required to operate

$T_{raise} = F \times Radius(length\, of\, handle)$ Find force F

$T_{raise} = \frac{60P}{2\pi N} \times 10^6$ Find Power P

Cutting speed =Pitch ×RPM mm/min

8) Efficiency

$$Screw\ Efficiency = \frac{Torque\ required\ without\ considering\ friction}{Actual\ torque\ required\ for\ screw \wedge collar} = \frac{W \times \tan a \times d/2}{T_s + T_c} \times 100$$

NUMERICAL PROBLEMS

1] A vertical two start square threaded screw of a 100 mm mean diameter and 20 mm pitch supports a vertical load of 18 kN. The axial thrust on the screw is taken by a collar bearing of 250 mm outside diameter and 100 mm inside diameter. Find the force required at the end of a lever which is 400 mm long in order to lift and lower the load. The μ for the screw and nut is 0.15 and that for collar is 0.20.

{Ans : T raise =569150 N-mm, Force=1423 N, T lower = 3x35315 N-mm, Force=838.8 N}

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2) A vertical screw with single start square threads of 50 mm mean diameter and 12.5 mm pitch is raised against a load of 10 kN by means of a hand wheel, the boss of which is threaded to act as a nut. The axial load is taken up by a thrust collar which supports the wheel boss and has a mean diameter of 60 mm. The coefficient of friction is 0.15 for the screw and 0.18 for the collar. If the tangential force applied by each hand to the wheel is 100 N, find suitable diameter of the hand wheel.

{T raise =112200 N-mm, Dia of wheel =1122 mm}

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3)The nominal diameter of a Triple threaded screw is 50mm & pitch 8mm .It is used with collar 100mm outer dia & 65mm inner dia coefficient of friction for threads as well as collars is 0.15 . Screw is used to raise of load of 15 KN calculate Using uniform wear theory i)Torque required to lift the load. ii)Torque required to lower the load. iii)Force required at radius 500mm.

{Ans : T raise=204643.56 N-mm, T lower =87404.87 N-mm, Force to raise = 409.3 N }

4]An electric motor driven power screw moves a nut in a horizontal plane against a force of 75 kN at a speed of 300 mm / min. The screw has a single square thread of 6 mm pitch on a major diameter of 40 mm. The coefficient of friction at screw threads is 0.1. Estimate power of the motor.

{T raise = 211.45 $\times 10^3 N - mm$**,Power=1.108 kW }**

5] The cutter of a broaching machine is pulled by square threaded screw of 55 mm external diameter and 10 mm pitch. The operating nut takes the axial load of 400 N on a flat surface of 60 mm and 90 mm internal and external diameters respectively. If the coefficient of friction is 0.15 for all contact surfaces on the nut, determine the power required to rotate the operating nut when the cutting speed is 6 m/min. Also find the efficiency of the screw.

{T raise =4410 N-mm, Power=0.277 kW, efficiency 14.4%}

6] A Machine vice has single start square threads with nominal diameter 22 mm and pitch 5 mm. Collar dia are 55mm & 45mm. μ for threads 0.15 & u for collar 0.17

The machinist can comfortably apply a force of 125 n on handle of mean radius 150 mm. Assuming uniform wear theory calculate,1) The clamping force developed between the jaws, 2) The overall efficiency of screw

{ W=2868.73 N, eff=12.18%}

7] The lead screw of a lathe has Acme threads of 50 mm outside diameter and 8 mm pitch. The screw must exert an axial pressure of 2500 N in order to drive the tool carriage. The thrust is carried on a collar 110 mm outside diameter and 55 mm inside diameter and the lead screw rotates at 30 r.p.m.Find (a) the power required (b) the efficiency of the lead screw.of 0.15 for the screw and 0.12 for the collar.

{Ans : Total torque 24565 N-m, Power=0.077 Kw, Efficiency =13%}

8] The lead screw of a lathe has single start I.S.O. metric trapezoidal threads of 52 mm nominal diameter and 8 mm potch. The screw is required to exert an axial force of 2 kN in order to drive the tool carriage during the turning operation. the thrust is carried on a collar 100 mm outer diameter and 60 mm inner diameter. the values of coefficient of friction at the screw threads and the collar are 0.15 and 0.12 respectively. The screw rotates at 30 RPM. Calculate;1) The power required to drive the lead screw and 2) The overall efficiency of the screw.

{ P=0.0618 kW, overall efficiency =12.94%}

Problems on Stresses in Screw and Nut.

Stresses in screw

Torsional shear stress

$$T=\frac{\pi}{16}\times\tau\times dc^3 .... Find\ \tau$$

Compressive Stress

$$\sigma_c=\frac{load}{area}=\frac{W}{\frac{\pi}{4}d_c^2} .... Find\ \sigma_c$$

$$\boldsymbol{\tau\ combined=\frac{1}{2}[\sqrt{\sigma_c{}^2+4\tau^2}]}$$ ....This should be less than allowable τ

Stresses in NUT

Bearing Pressure in nut is given by

$$P_b = \frac{W}{\frac{\pi}{4}[d_o^2 - d_c^2] \times n}$$

where n= number of threads in contact with screw

h= Height of Nut = n × p

t= Thickness of screw = p/2

Shear stress induced in the nut is given by ,

$$\tau = \frac{load}{area\ of\ one\ thread \times no\ of\ threads} = \frac{W}{\pi \times d_o \times p/2 \times n}$$ ......Direct Shear stress induced in nut { Note that screw fails at do in shear}

NUMERICALS

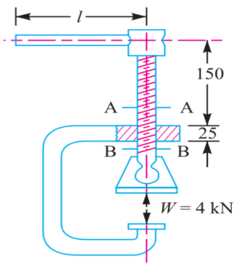
1) A power screw having double start square threads of 25 mm nominal diameter and 5 mm pitch is acted upon by an axial load of 10 kN. The outer and inner diameters of screw collar are 50 mm and 20mm respectively. The coefficient of thread friction and collar friction may be assumed as 0.2 and 0.15 respectively. The screw rotates at 12 r.p.m. Assuming uniform wear condition at the collar and allowable thread bearing pressure of 5.8 N/mm2,

find: 1. the torque required to rotate the screw;

2. the stress in the screw; and

3. the number of threads of nut in engagement with screw.

{Ans T raise = 65771 n-mm, Stress=44.8 Mpa,number of th=9.76}



All dimensions in mm.

Fig. 17.10

2) A 'c' clamp as shown in figure has trapezoidal athreads of 12 mm outside diameter and 2 mm pitch. The coefficient of friction for screw thread is 0.12 and for collar is 0.25. the mean radius of collar is 6 mm. If the force exerted by the operator at the end of the handle is 80 mm. Find 1. Length of handle 2. Maximum shear stress in the body of screw 3. Bearing pressure on threads

{Ans : T raise =10033 N-mm, Length of handle =125.4 mm, Max shear stress =39.83 Mpa, bearing pressure=9.26 Mpa}

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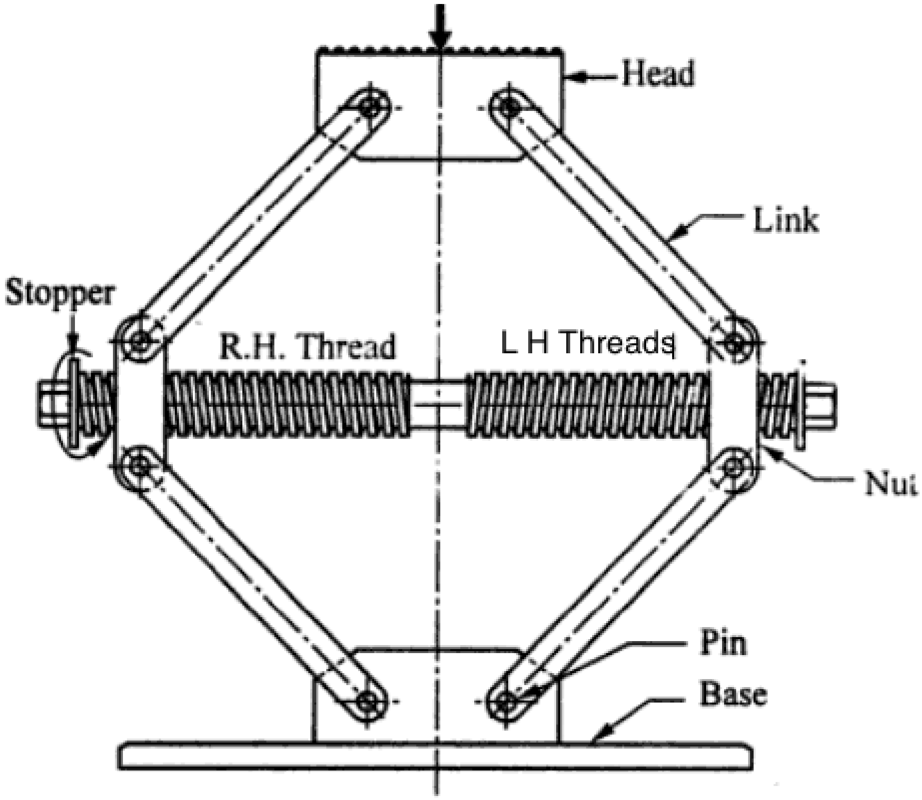
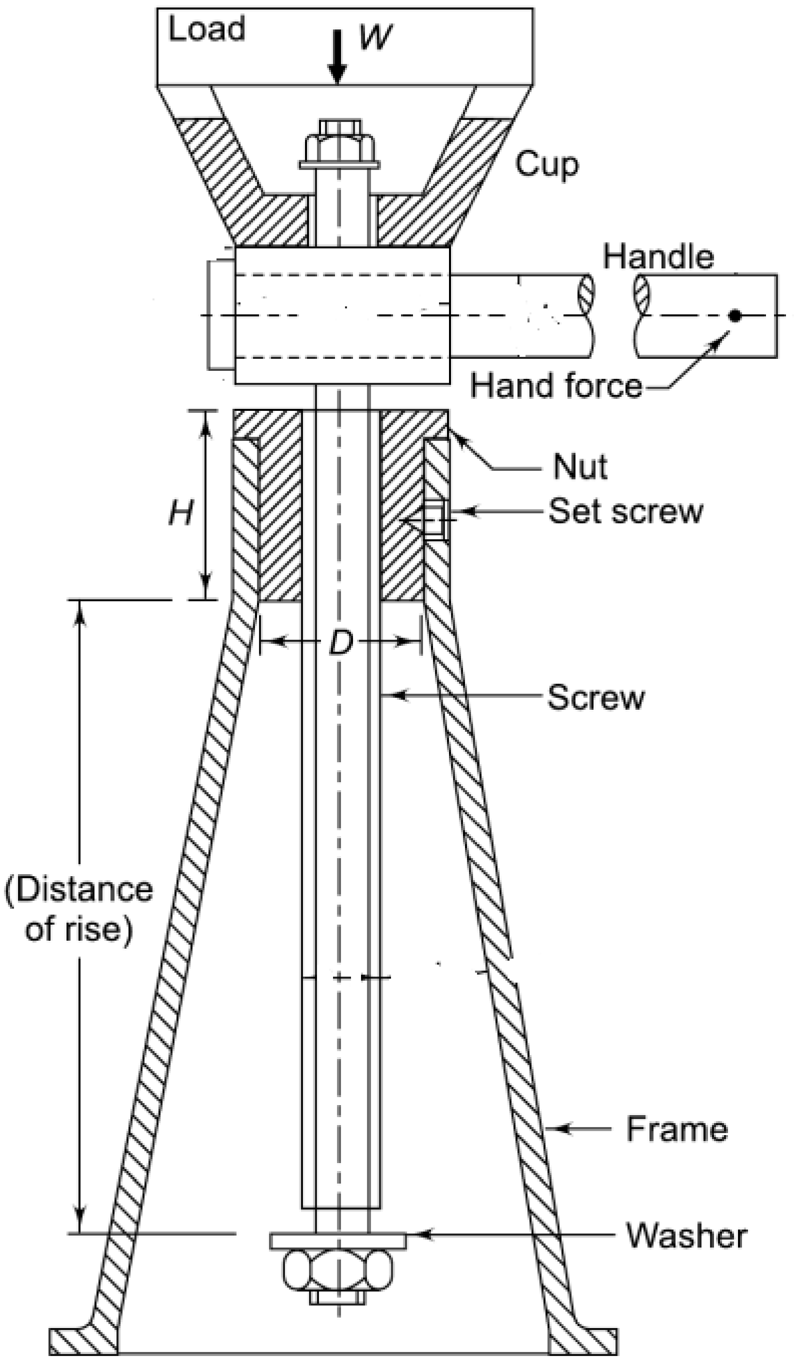
3) A power screw on a machine has single start square thread with a non rotating bronze nut. Axial force on the screw is 15 kN. Allowable stresses for screw material in compression and shear are 85 MPa and 37 MPa respectively. Allowable bearing pressure for the screw nut pair is 5 MPa. Find (I) Core diameter of screw (ii) Length of the nut(iii) Efficiency of power screw in coefficient of friction between screw and nut is 0.12. (iv) Shear stresses in the threads of screw and nut.

4.2 Desing of Screw Jack and Toggle screw

10) Draw a neat sketch of TOGGLE JACK and label its parts



11) Draw a neat sketch of SCREW JACK and lable its parts



4.3 Design of Screw Jack (Screw and Nut)

Step I] Design of screw on basis of compressive failure

Compressive Failure of screw $\sigma_c = \frac{load}{area} = \frac{W}{\frac{\pi}{4} d_c^2}$ Find dc from this equation

Nominal diameter $d_o = \frac{d_c}{0.84}$ {Take round even number and about 20% more size to account for additional shear stress}

Pitch of Screw $P = d_0 - d_c$ {Take Round figure }

Torque required to raise the load

$$\alpha = \tan^{-1}\left(\frac{p}{\pi \times d}\right) \qquad \varphi = \tan^{-1}[\mu] \qquad T_s = W \times \tan(\varphi + \alpha) \times \frac{d}{2}$$

Step II) Check Stresses in Screw

Torsional shear stress

$$T = \frac{\pi}{16} \times \tau \times dc^3 ....Find\ \tau$$

Compressive Stress

$$\sigma_c = \frac{Load}{Area} = \frac{W}{\frac{\pi}{4} \times d_c^2} ......\text{Find } \sigma_c$$

$$\tau\, combined = \frac{1}{2}\left[\sqrt{\sigma_c^2 + 4\tau^2}\right]$$ ....This should be less than allowable τ

Step III) Check Stresses in NUT

Bearing Pressure in nut is given by

$$P_b = \frac{W}{\frac{\pi}{4}[d_o^2 - d_c^2] \times n}$$

where n= number of threads in contact with screw, h= Height of Nut = n $\times$ p, t= Thickness of screw = p/2 ,Shear stress induced in the nut threads is given by ,

$$\tau = \frac{load}{area\, of\, one\, thread \times no\, of\, threads} = \frac{W}{\pi \times d_o \times p/2 \times n}$$ ....Shear stress induced

Numerical Problems

A screw jack carries a load of 25 kN. If the coefficient of friction between screw and nut is 0.15, Design the screw and nut. Neglect collar friction and column action take c=42 N/mm2 and =30 N/mm2 for screw and nut=30 N/mm2 for nut, Allowable bearing pressure = 14 N/mm2 (Use single start thread).

4.3 Design of Screw Fasteners

1) State the advantages and limitations of threaded joints.

Advantages of threaded joints:

(i) The parts are held together by means of a large clamping force. There is no loosening of the parts.

(ii) Threaded joints have small overall dimensions resulting in compact construction.

(iv) The threads are self-locking. Therefore, threaded joints can be placed in any position—vertical, horizontal or inclined.

(v) Threaded fasteners are economical to manufacture. Their manufacturing is simple. High accuracy can be maintained for the threaded components.

(vi) The parts joined together by threaded joints can be detached as and when required. This requirement is essential in certain applications for the purpose of inspection, repair or replacement.

(vii) Threaded fasteners are standardized and a wide variety is available for different operating conditions and applications.

Disadvantages of threaded Joints :

(i) Threaded joints require holes in the machine parts that are to be clamped. This results in stress concentration near the threaded portion of the parts. Such areas are vulnerable to fatigue failure.

(ii) Threaded joints loosen when subjected to severe vibrations.

(iii) Threaded fasteners are considered as a major obstacle for efficient assembly.

2) State the advantages and limitations of Welded joints.

Advantages of Welded joints:

(i) Welded steel structures are lighter than the corresponding iron castings or steel castings .

(ii) The cost of welded assembly is lower than that of bolted/riveted joints.

(iii) The design of welded assemblies can be easily and economically modified to meet the changing product requirements. Alterations and additions can be easily made in the existing structure by welding.

(iv) Welded assemblies are tight and leak-proof as compared other assemblies.

(v) The production time is less for welded assemblies.

(vi) When two parts are joined by the bolted joints method, holes are drilled in the parts to accommodate the bolts. The holes reduce the cross-sectional area of the members and result in stress concentration. There is no such problem in welded connections.

(vii) The strength of welded joint is high. Very often, the strength of the weld is more than the strength of the plates that are joined together.

Disadvantages of Welded Joints :

(i) Compared to cast iron structures,the capacity of welded structure to damp vibrations is poor.

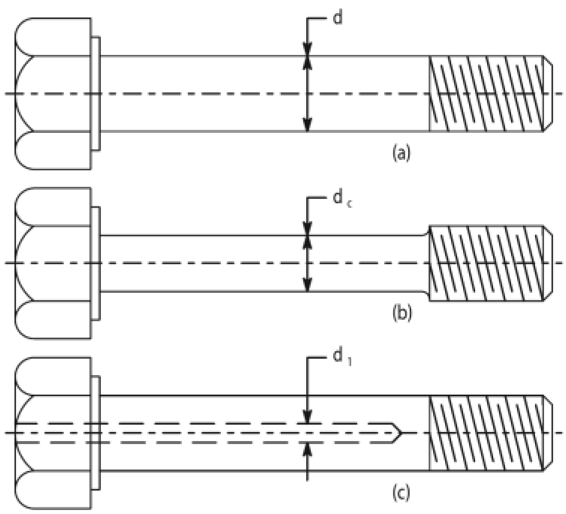
(ii) Welding results in a thermal distortion of the parts, thereby inducing residual stresses. In many cases, stress-relieving heat treatment is required to relieve residual

stresses.
(iii) The quality and the strength of the welded joint depend upon the skill of the welder. It is difficult to control the quality when a number of welders are involved.
(iv) The inspection of the welded joint is more specialized and costly compared with the inspection of bolted or cast structures.

3) What do you mean by Bolts of uniform strength? How it is achieved? (V V Imp)

When a bolt is subjected to shock loading, as in case or a cylinder head bolt of an internal combustion engine, the resilience of the bolt should be considered in order to prevent breakage at the thread. In an ordinary bolt shown in figure a, the effect of the impulsive loads applied axially is concentrated on the weakest part of the bolt i.e. cross sectional area at the root of the threads. In other words, the stress in the threaded part of the bolt will be higher than that in the shank. Hence, a great portion of the energy will be absorbed at the region of the threaded part which may fracture the threaded portion because of its small length.

If the shank of the bolt turned down to a diameter equal or even slightly less than the core diameter of the thread (Dc) as shown in figure b, then shank of the bolt will undergo the higher stress. This means that a shank will absorb a large portion of the energy, thus relieving the material at the section near the thread. The bolt in this way, become stronger and lighter and it increases the shock absorbing capacity of the bolt because of an increased modulus of resilience. This gives us the bolt of uniform strength. The resilience of the bolt may be increased by increasing its length.
A second alternative method of obtaining the bolts of uniform strength is shown in figure c.
In this method, an axial hole is drilled through the head as far as the thread portion such that the area of the shank becomes equal to the root area of the threads.



Various Methods of making bolt of Uniform strength.

d1= Diameter of the hole

d = Outer diameter of the thread, and

dc= Root or core diameter of the thread.

$$\frac{\pi}{4} \times d_1^2 = \frac{\pi}{4} \times (d^2 - d_c^2)$$

$$\frac{\pi}{4} \times d_1^2 = \frac{\pi}{4} \times (d^2 - d_c^2)$$

$$d_1^2 = (d^2 - d_c^2)$$

$$d_1^2 = (d^2 - d_c^2)$$

$$d_1 = \sqrt{(d^2 - d_c^2)}$$

$d_1 = \sqrt{(d^2 - d_c^2)}$ This formula gives the size of hole to be drilled.

4. What are the various stresses induced in a threaded joint?

When a nut is tightened over a screw following stresses are induced:

1.Tensile or shear stress due to external load.

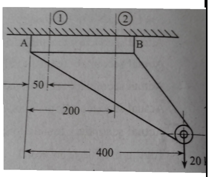
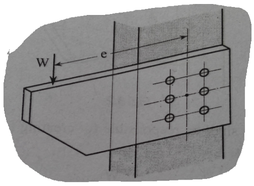
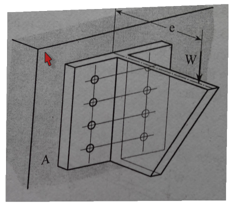
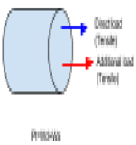
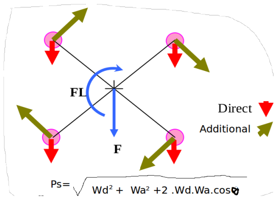
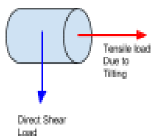
2. Tensile stresses due to stretching of the bolt

3. Torsional shear stress due to frictional resistance at the threads.

4.Shear stress across threads

5. Compressive or crushing stress on the threads

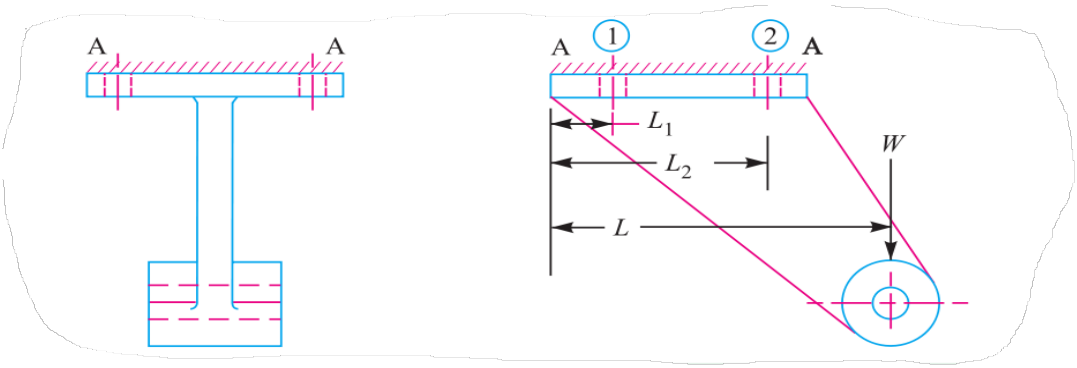
Eccentrically Loaded Bolted Joints

	Bolt axis parallel to load	Bolt axis in the plane of load	Bolt axis perpendicular to load
Diagram			
Direct Load	$W_{t1}=\frac{W}{No.of\ bolts(n)}$ (Tensile)	$W_{s1}=\frac{W}{No.of\ bolts(n)}$ (Shear)	$W_s=\frac{W}{No.of\ bolts(n)}$ (Shear)
Additional load due to Tilting	$W_{t2}=\frac{W.e.l_{max}}{l_1^2+l_2^2+l_3^2+l_4^2}$ (Tensile) l= Distance of bolts from tilting edge	$W_{s2}=\frac{W.e.l_{max}}{l_1^2+l_2^2+l_3^2+l_4^2}$ (Shear) l= Distance of bolts from Centroid	$W_t=\frac{W.e.l_{max}}{l_1^2+l_2^2+l_3^2+l_4^2}$ (Tensile) l= Distance of bolts from tilting edge
Forces			
Equivalent Load	Total equivalent load $P_t=W_{t1}+W_{t2}$ (Tensile)	Total Equivalent load $P_s=\sqrt{W_d^2+W_a^2+2W_dW_a cos\theta}$ (Shear) θ =Angle made by line joining centroid and centre of bolt having max radius	Total Equivalent Tensile load $P_t=\frac{1}{2}[W_t+\sqrt{W_t^2+4.W_s^2}]$ Total Equivalent Shear $P_s=\frac{1}{2}[\sqrt{W_t^2+4.W_s^2}]$
Bolt Size calculation	$P_t=\frac{\pi}{4}d_c^2\times f_t$ $d=1.25\times d_c$	$P_s=\frac{\pi}{4}d_c^2\times f_s$ $d=1.25\times d_c$	$P_t=\frac{\pi}{4}d_c^2\times f_t$ $P_s=\frac{\pi}{4}d_c^2\times f_s$ $d=1.25\times d_c$

Type 1: Bolt axis parallel to load

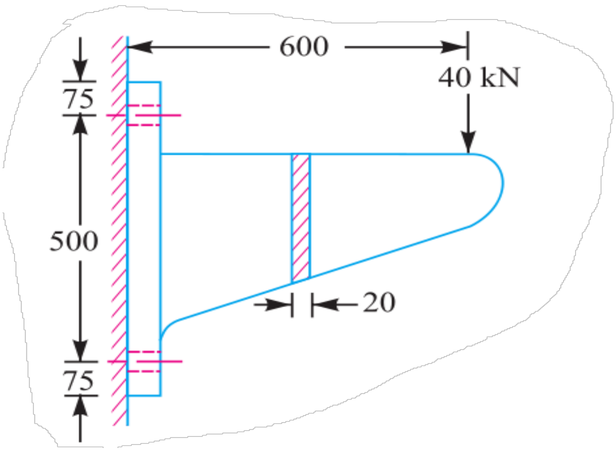
1. A bracket is fitted to the ceiling with the help of two bolts, the bracket carries a load of 25 kN and the allowable tensile stress for bolt material is 90 Mpa. The distance of first bolt from the leftmost edge is 100 mm, the distance between two bolts is 120 mm and the distance between leftmost edge and the line of application of load is 150 mm. Find the size of the bolt to take this load. the load direction is parallel to the bolt axis.
2. A bracket, as shown in figure below supports a load of 30 KN with the help of FOUR bolts, Determine the size of the bolts if the maximum allowable tensile stress in the bolt material is 60 MPA. The distances are

L1=80mm, L2=250mm and L=500mm



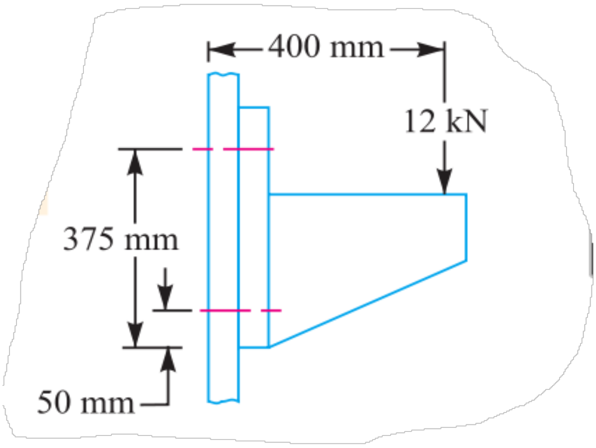
{Ans:dc=27.2 and d=33.75 The bolts should be of size M34.}

Type 2: Bolt axis Perpendicular to the Direction of load.



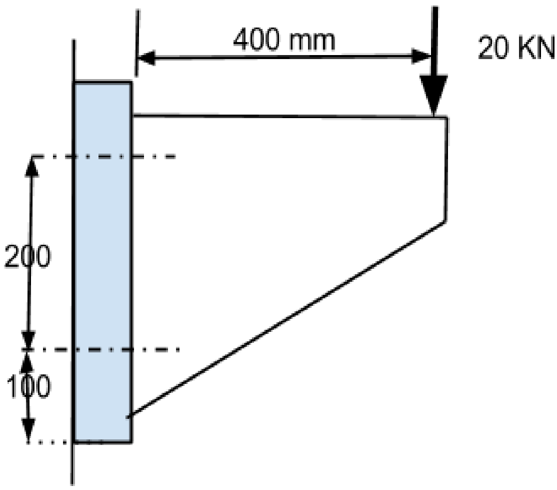
1.A wall bracket, as shown in Fig. , is fixed to a wall by means of four bolts. Find the size of the bolts .The safe stress in tension for the bolt may be assumed as 70 MPa.

[Ans:dc=21.14, M28]



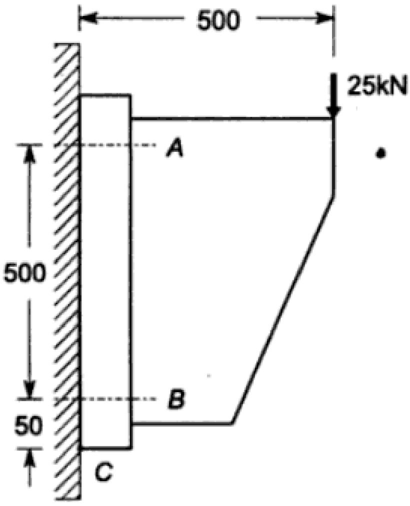
2.The brackets are fixed on steel columns as shown in Fig Vertical face of the bracket is secured to a column by four bolts, in two rows (two in each row) at distance of 50 mm from the lower edge of the bracket. Determine the size of the bolts if the permissible value of the tensile stress for the bolt material is 84 MPa.

{Ans: dc=10.65}



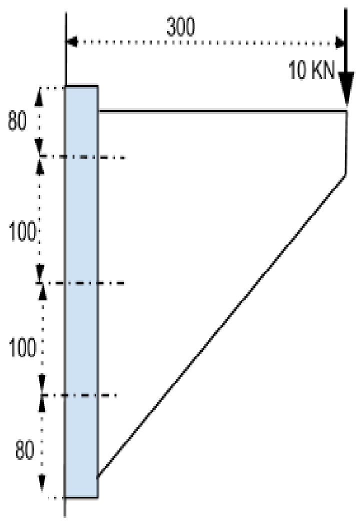
3. A bracket is fitted to wall by means of four bolts as shown in figure
Find the size of the bolt if
Allowable tensile stress = 50 MPa
Allowable Shear stress = 30 Mpa

{By tensile strength M24, By shear strength M22, FInal size M24}



4. A wall bracket is attached to a wall by means of four identical bolts, two at A and two at B as shown in figure below.If allowable stress is 35 N/mm2, determine the size of the bolts on the basis of Maximum principal stress theory.

{dc=22.61,M30}



5..A bracket is fitted with the help of six bolts of identical size bolts to wall as shown in figure. Find the size of bolt if ft=42 MPa and Fs=35 MPa.

{dc=11.32,M16 and dc=9.42, M12..Final size M16}

Type 3: Bolt axis in the plane of load

Type 3: Bolt axis in the plane of load

$$W s_1 = \frac{W}{No} of\ bolts(n)$$ …......................... Shear Stress

$$W s_2 = \frac{W.e.l_{max}}{I_1^2 + I_1^2 + I_1^2 + I_1^2 ....}$$ …......................... Shear Stress

l= Distance of bolts from Centroid of lamina connecting all bolts

Total Equivalent load

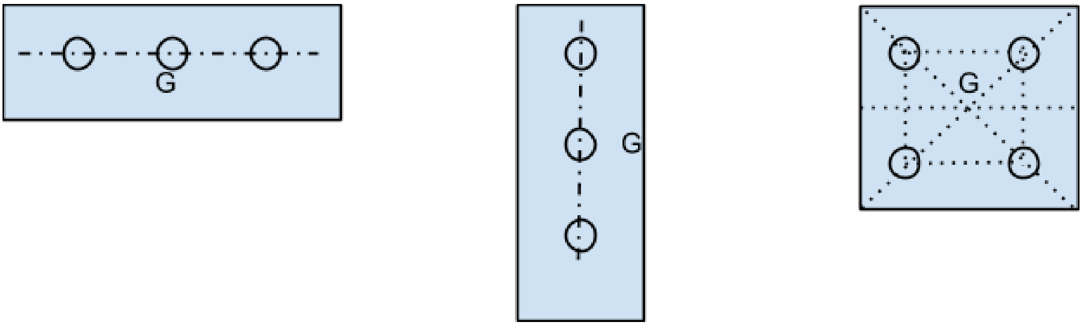
$$P_s = \sqrt{W_{s1}^2 + W_{s2}^2 + 2 \times W_{s1} \times W_{s2} \times \cos\theta} \text{ .... } \textit{Shear Stress}$$

$P_s \sqrt{w^2 s1 + w^2 s2 + 2 \times W s1 \times w^2 s2 \times \cos 0}$ .... Shear Stress

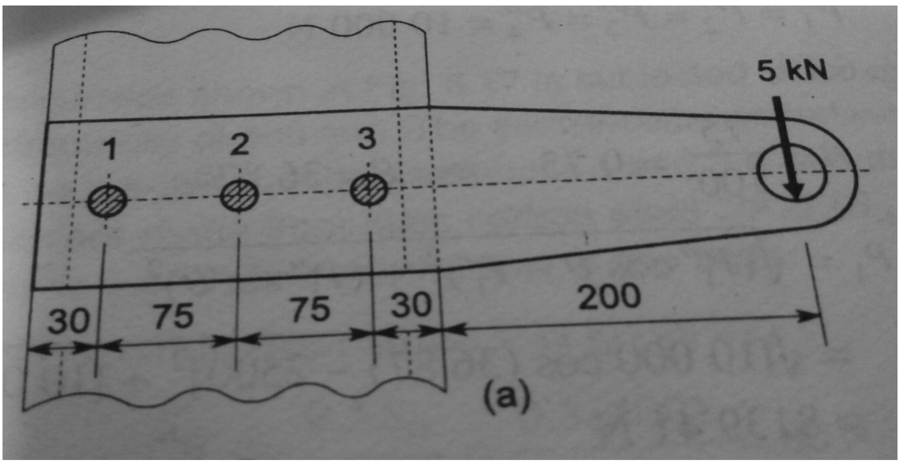
$$\theta = \textit{Angle made by the line joining centroid} \wedge \textit{the centre of farthest bolt}$$

θ = Angle mode by the line joining centroid the Centre of farthest bolt

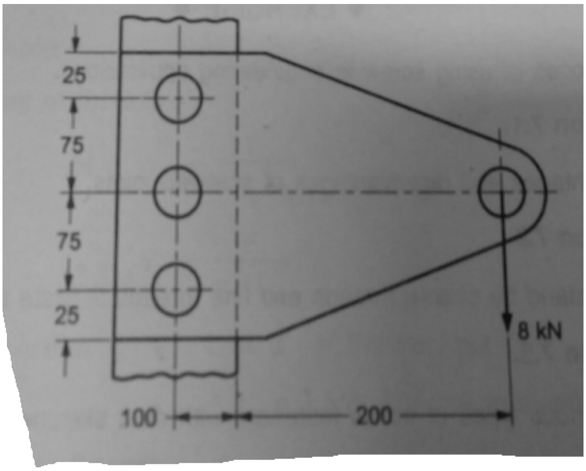
Location of centroid for the various configuration of bolts



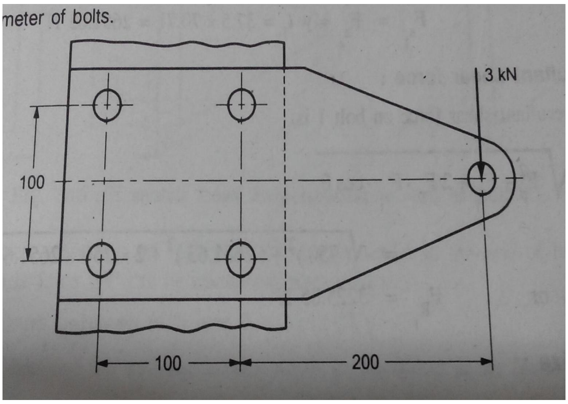
1) A steel plate is subjected to a force of 5 kN and fixed to a channel by means of three identical bolts as shown below. The bolts are made up of plain carbon steel 45c8 (syt = 380 N/mm2) and the factor of safety is 3. Specify the size of the bolt.



2) A steel plate subjected to force of 8 kN is fixed to a channel by means of three indicating bolts as shown in Fig 13 The are made of 45C8 (S = 380 N / mm^2 ***) If The required factor of safety is 2.5 determine the size of the bolts.***



3) A steel Plate Subjected to force of 3kN and fixed to a vertical chancel by means of four identical bolts is shown in Fig 7.4.1 The bolts are made of plain carbon steel 45C8 with yield strength of 380 N / mm^2 ***If the required factor of safety is 2, determine the diameter of bolts.***



4.4 Problems on Design Of Welded Joints

1.Design of a butt joint:

The main failure mechanism of welded butt joint is tensile failure. Therefore the strength of a butt joint is

$$P = s_T l t$$

where s_T =allowable tensile strength of the weld material.

t =thickness of the weld

l =length of the weld.

For a square butt joint t is equal to the thickness of the plates. In general, this need not be so (see figure 1).

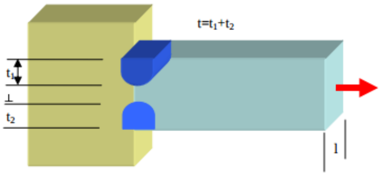


Figure 10.4.1: Design of a butt joint

2.Design of transverse fillet joint:

Consider a single transverse joint as shown in figure 10.4.2. The general stress distribution in the weld metal is very complicated. In design, a simple procedure is used assuming that entire load *P* acts as shear force on the throat area, which is the smallest area of the cross section in a fillet weld. If the fillet weld has equal base and height, (*h*, say), then the cross section of the throat is easily seen to be $\frac{hl}{\sqrt{2}}$. With the above consideration the permissible load carried by a transverse fillet weld is

$$P = s_s A_{throat}$$

where s_s -allowable shear stress

A_{throat} =throat area.

For a double transverse fillet joint the allowable load is twice that of the single fillet joint.

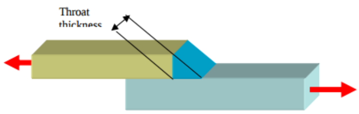


Figure 10.4.2: Design of a single transverse fillet

3.Design of parallel fillet joint:

Consider a parallel fillet weld as shown in figure 10.4.3. Each weld carries a load $P/2$. It is easy to see from the strength of material approach that the maximum shear occurs along the throat area (try to prove it). The allowable load carried by each of the joint is $s_s A_t$ where the throat area $A_t = \frac{lh}{\sqrt{2}}$. The total allowable load is

$$P = 2 s_s A_t .$$

Figure 3: Design of a parallel fillet joint

In designing a weld joint the design variables are h and l. They can be selected based on the above design criteria. When a combination of transverse and parallel fillet joint is required (see figure-10.4.4) the allowable load is

$$P = 2s_s A_t + s_s A_t'$$

where A_t =throat area along the longitudinal direction.

A_t' =throat area along the transverse direction.

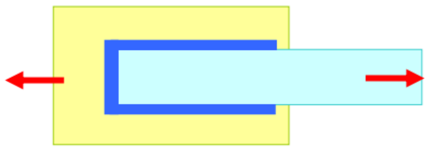
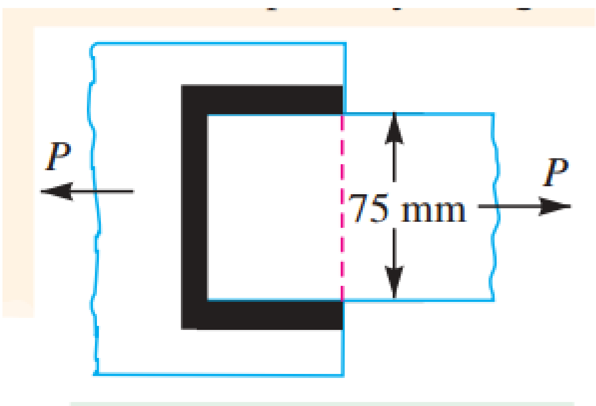


Figure 10.4.4: Design of combined transverse and parallel fillet joint

Problems

1) A Plate 75 mm wide and 12.5mm thick is joined with another plate by a single transverse weld and a double parallel fillet weld as shown in Fig 10.15 The maximum tensile and shear stresses are 70MPa and 56 MPa respectively.
Find the length of each parallel filler weld if the joint is subjected to both static and fatigue loading.



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

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