EXPERIMENT-1

AIM- Design and drawing of Cotter Joint

FUNCTION- A cotter joint is used to connect one end of a rod is provided with a socket type of end and the other end of the rod is inserted to a socket. The end of the rod which goes into a socket is also called Spigot.

APPLICATIONS-

1. Joints between the piston rod and the cross need of the steam engine.
2. Joints between the side spindle and the fork of the valve mechanism.
3. Joint between the piston rod and the tail or pump foundation bolts.

ASSUMPTIONS-

1. The rods are subjected to axial tensile force the effect of steels concentrationdue to shaft is neglected.

NOTATION-

F= Axial load carried by the rod

d=Diameter of the rod

d_1= diameter of spigot

d_2=diameter of collar

d_3=outside diameter of sleeve

d_4=diameter of sleeve or socket

a= width at rod end

b= width of collar

c= width of socket end

e=Thickness of collar

t= thickness of cotter

σ=allowable shear stress

σ_c= allowable crushing stress

PROCEDURE:

STEP 1: Design of rods

For the rods under axial load,

Axial stress in the rods, $\sigma = \frac{4F}{\pi d^2}$

STEP 2: Design of the spigot and the Cotter

a.) Crushing strength of the cotter, $F = d_1 t \sigma_c$

b.) Axial stress across the slot of the rod

$$\sigma = \frac{4F}{\pi (d_1)^2 - 4 d_1 t}$$

STEP 3: Design of the cotter

Strength of the cotter in double shear $F = 2bt\tau$

STEP 4: Design of the collar

a.) Bearing stress in the collar, $\sigma_c = \frac{4F}{\pi\{(d_2)^2 - (4d_1)^2\}}$

b.) Shear stress in the collar, $\tau = \frac{F}{\pi d_1 e}$

STEP 5: Design of the sleeve

a.) Axial stress across the slot of the sleeve

$$\sigma = \frac{4F}{\pi[(d_3)^2-(d_1)^2]-4t(d_3-d_1)}$$

b.) Crushing strength of the socket

$$F = (d_4-d_1)t \times \sigma_c$$

STEP 6: Design of the rod end

a.) Shear stress at the rod end due to double shear

$$\tau = \frac{F}{2ad_1}$$

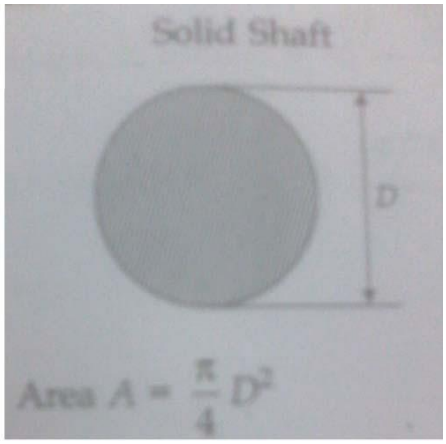
STEP 7: Design of the socket end

Shear stress at the socket end, $\tau = \frac{F}{2c(d_4-d_1)}$

EXPERIMENT-2

AIM-Design of shafts subjected to Torsion, Bending Moment and Combined Bending & Torsion

INTRODUCTION- Shaft usually a round member, solid or hollow cross section that rotates and transmits power. It carries machine elements like gears, pulleys, cams, sprockets, couplings etc. Its design primarily consists of finding the size in order to satisfy the strength and rigidity, while transmitting power. Shafts are usually subjected to torsion, bending etc.



NOTATIONS

P = Power in kW

n= speed in rpm

M_b = Maximum bending moment on shaft

M_t = Maximum torque on shaft

D = Diameter of shaft

F = axial or thrust load

K_b = Combined shock and Endurance factor in Bending

K_t = Combined shock and Endurance factor in torsion

For rotating shafts with gradually applied loads

K_b =1.5 and K_t = 1.0

Suddenly, applied loads with minor shocks, K_b = 1.5 to 2.0 andK_t = 1 to 1.5

Suddenly, applied loads with heavy shocks, K_b = 2.0 to 3.0 andK_t = 1.5 to 3.0

σ = allowable tensile stress for shaft material

τ = allowable shear stress for shaft material

SHAFTS SUBJECTED TO PURE TORSION

Solid Shaft: Diameter of shaft, $D = \left[\frac{16K_t M_t}{\pi\tau}\right]^{1/3}$

SHAFTS SUBJECTED TO PURE BENDING

Solid Shaft: Diameter of shaft, $D = \left[\frac{32K_b M_b}{\pi\sigma}\right]^{1/3}$

SHAFTS SUBJECTED TO COMBINED BENDING AND TORSION

Solid Shaft:

i. According to maximum normal stress theory
Diameter of Shaft, $D = \left[\frac{16}{\pi\sigma}\left\{K_b M_b + \sqrt{(K_b M_b)^2 + (K_t M_t)^2}\right\}\right]^{1/3}$

ii. According to maximum shear stress theory
Diameter of Shaft, $D = \left[\frac{16}{\pi\sigma}\sqrt{(K_b M_b)^2 + (K_t M_t)^2}\right]^{1/3}$

EXPERIMENT-3

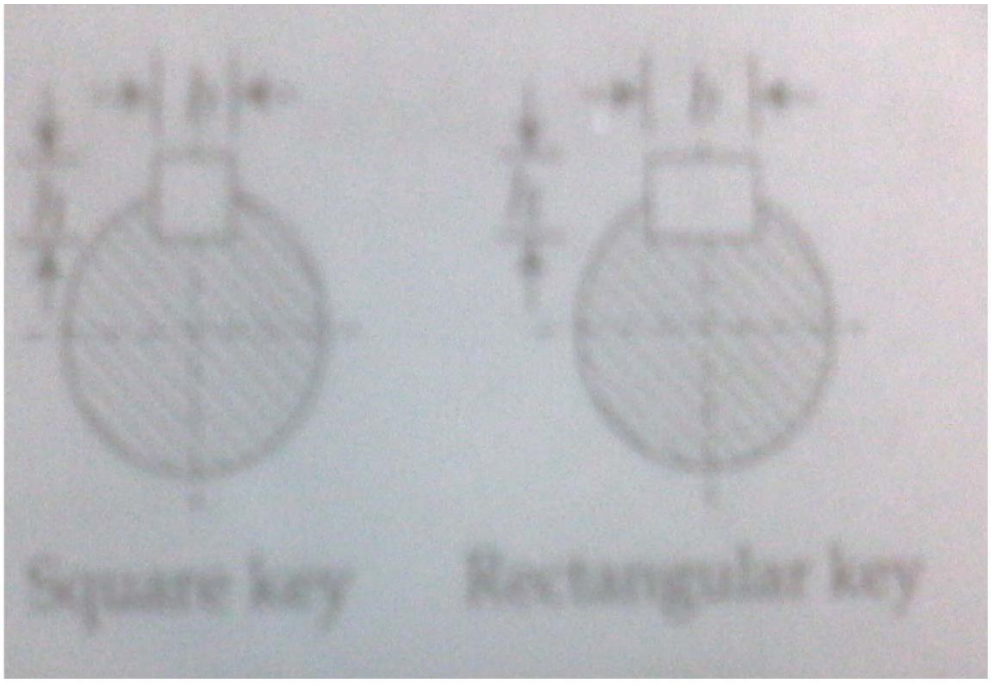
AIM- Design of Flat and Square Keys

INTRODUCTION

Keys are used to prevent relative motion between a shaft and the connected member through which torque is being transmitted. Common types of Keys are: Square key, Rectangular key, Feather key, Round key, Gib key, Head key, Taper key, Barth key, Kennedy key, Saddle key and Woodruff key etc.

DESIGN OF SQUARE AND FLAT KEY

The keys are subjected to shear and crushing or bearing stress while transmitting torque.



NOTATIONS

b= width of key

h= thickness

l=length of key

Considering the shearing of key

Shear stress, $\tau = \frac{F}{\text{Area of shear}} = \frac{F}{bl}$

a.) Where F= Force $= \frac{\text{Torque}}{\text{Radius of shaft}} = \frac{M_t}{r} = \frac{M_t}{(d/2)} = \frac{2M_t}{d}$

$\tau = \frac{2M_t}{\ }$ =>width of the key $b = \frac{2M_t}{\ }$

Considering crushing of key

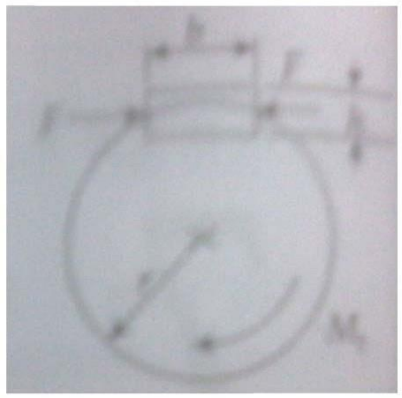
Crushing stress, $\sigma_c = \frac{F}{\text{Area of crushing}} = \frac{F}{(h/2)l}$

$\sigma_c = \frac{2M_t}{d(h/2)l} = \frac{4M_t}{dhl}$ => thickness of key $h = \frac{4M_t}{dl\sigma_c}$

For Square key b=h

$b = \frac{2M_t}{dh\tau}$ and $h = b = \frac{4M_t}{dl\sigma_c}$

$M_t = \frac{bdl\tau}{4}$ in shear and $M_t = \frac{bdl\sigma_c}{4}$ in crushing



Procedure Type 1: Length of Key

STEP 1: Find Torque, $M_t = \frac{9.55 \times 10^6 \times P}{N}$ P=Power in kW, n=speed,rpm

STEP 2: Diameter of shaft, $d = \sqrt[3]{\frac{16M_t}{\pi\tau_s}}$

τ_s = allowable shear stress for shaft material

STEP 3: Find length of key from

a.) Width of key, $b = \frac{2M_t}{dl\tau_k}$

τ_k = allowable shear stress for key

b.) Thickness of key, $h = \frac{4M_t}{dl\sigma_b'}$

Recommend the bigger one as length.

σ_b' = allowable crushing stress or bearing stress for key.

Procedure – Type 2: Length of key

STEP 1: Torque, $M_t = \frac{9.55 \times 10^6 \times P}{N}$

STEP 2: Diameter of shaft, $d = \sqrt[3]{\frac{16M_t}{\pi\tau_s}}$

STEP 3: Design of Key

a.) Length of key = length of hub = l

b.) Width of key, $b = \frac{2M_t}{dl\tau_k}$

c.) Thickness of key, $h = \frac{4M_t}{dl\sigma_b'}$

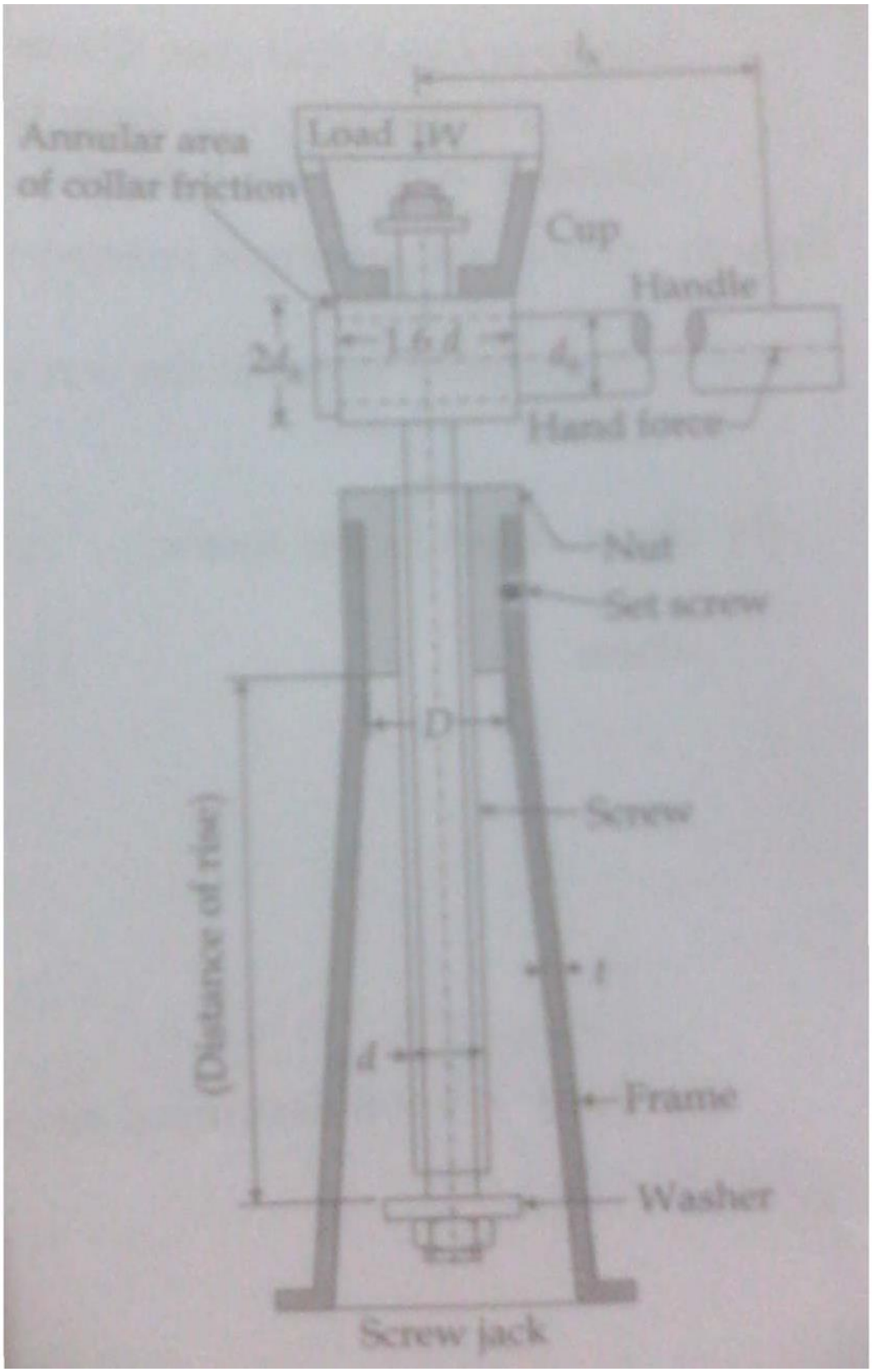
EXPERIMENT-4

AIM- Design and drawing of Screw Jack

INTRODUCTION

Screws used for power transmission are known as power screws. They provide a means for obtaining a large mechanical advantage, in such applications as screw jacks, c-lamps, hand presses land screw on lathe machines, hoisting machines etc.

SCREW JACK


Annular area
of collar friction
Load W
Cup
Handle
1.6d
4
Hand force
Nut
Set screw
D
Screw
(Distance of rise)
t
d
Frame
Washer
Screw jack

NOTATIONS

W = load to be lifted

d= outside diameter or nominal diameter of screw

p= pitch

d_h = diameter of handle

l_h = length of handle

D = outside diameter of nut

D_i, D_o = inside and outside the diameter of thrust collar

d_c= mean diameter of the thrust collar = $\frac{D_i + D_o}{2}$

l_n= nut length

t= thickness of body

σ_{max}= allowable normal stress in the screw

τ_{max}= allowable shear stress in the screw

σ_b= allowable bending stress in the screw

σ_b'=allowable bearing pressure in the threads

τ_n= allowable shear stress in the nut

μ = coefficient of thread friction

μ_c= coefficient of collar friction

Design Procedure

STEP 1: Design of screw

a.) Considering the screw under axial compression

$\sigma_c = \frac{W}{A_c}$ => core area, $A_c = \frac{W}{\sigma_c}$

σ_c= allowable compressive stress in screw = σ_{max}

b.) Considering the screw as a column loaded in axial compression, when fully extended, Rankine' equation for columns

$$\sigma = \frac{W}{A_c}\left(1 + a\left(\frac{l_e}{k}\right)^2\right)$$

σ = allowable stress in the screw = σ_{max}

$A_c = \pi d_1^2/4$

a= Rankine's Constant = 1/7500 for steels

l_e = equivalent length of the column

Screw Jack isa columnwith one end fixed at nut and other end is free where the load is sitting.

So, Equivalent length, l_e= 2 x (lift or extension)

K = radius ofgyration=$\frac{I}{A} = \sqrt{\frac{\left(\frac{\pi d_1^4}{64}\right)}{\left(\frac{\pi d_1^2}{4}\right)}} = \frac{d_1}{4}$

STEP 2: Check for principal stresses in the screw

a.) Compressive stress in the screw, $\sigma_c = \frac{W}{A_c}$

b.) Shear stress in the screw, $\tau_s = \frac{M_{ts} r_1}{J}$

Where M_{ts}= frictional torque in the screw only

$$= \frac{W d_2}{2}\left(\frac{\tan\alpha + \mu}{1 - \mu\tan\alpha}\right)$$

$d_2 = d - \frac{1p}{2}$, $\tan\alpha = \frac{l}{\pi d_2}$

(l= lead = 1xp assuming single start)

$r_1 = \frac{d_1}{2}$

$J = \frac{\pi d_1^4}{32}$

c.) Combined or principal stresses

i. Maximum normal stress

$$\sigma_{max} = \frac{1}{2}\left(\sigma_c + \sigma_c^2 + 4\tau_s^2\right)^{1/2}$$

ii. Maximum shear stress, $\tau_{max} = \frac{1}{2}\left(\sigma_c^2 + 4\tau_s^2\right)^{1/2}$

STEP 3: Design of screw head and collar

Assume pin head diameter, D_o = 1.6d = outside the diameter of the collar

Pin diameter, D_i = 0.5d = Inside the diameter of the collar

So, Mean Diameter of the collar, $d_c = \frac{D_i + D_o}{2}$

STEP 4: Design of nut:

Assuming bronze for nut,

a.) Length of nut, $l_n = \dfrac{4Wp}{\sigma_b' \pi (d^2 - d_1^2)}$

b.) Tensile stress in the nut = $\sigma_{nut} = \dfrac{W}{\pi/4(D^2 - d^2)}$

Assume, $\sigma_b' = 15MPa$ and

$\sigma_{nut} = 40$ to $50MPa$

STEP 5: Frictional torque

Total frictional torque including the collar friction $M_t = W\left\{\dfrac{d_2}{2}\left(\dfrac{\tan\alpha + \mu}{1 - \mu\tan\alpha}\right) + \dfrac{\mu_c d_c}{2}\right\}$

STEP 6: Design of handle

a.) Length of handle, $l_h = \dfrac{M_t}{F}$

b.) To find the diameter of handle, the handle is subjected to bending moment

WKT $\dfrac{M_b}{I} = \dfrac{\sigma_b}{c}$

Where M_b=bending moment = $Fl_h = M_t$

I = moment of inertia for handle, $I = \dfrac{\pi d_h^4}{64}$

$C = d_h$, σ_b = allowable bending stress

STEP 7: Efficiency, $= \dfrac{d_2 \tan\alpha}{\tan\alpha + \mu d_2 + \mu_c d_c \left(\dfrac{}{1 - \mu \tan\alpha}\right)}$

STEP 8: Self locking condition

The screw jack to be of self locking type,

Hence check for self locking.

$$\tan\alpha < \frac{\mu d_2 + \mu_c d_c}{d_2 - \mu\mu_c d_c}$$

EXPERIMENT-5

AIM- Design and drawing of Riveted Joints

INTRODUCTION

Two or more plates are joined by means of rivets and such joints are called riveted joints. These joints are permanent in nature unlike bolted joints. These are used in bridges, ships, boilers, tanks, etc.

Riveted Joints are classified into two types viz, lap joint and butt joint.

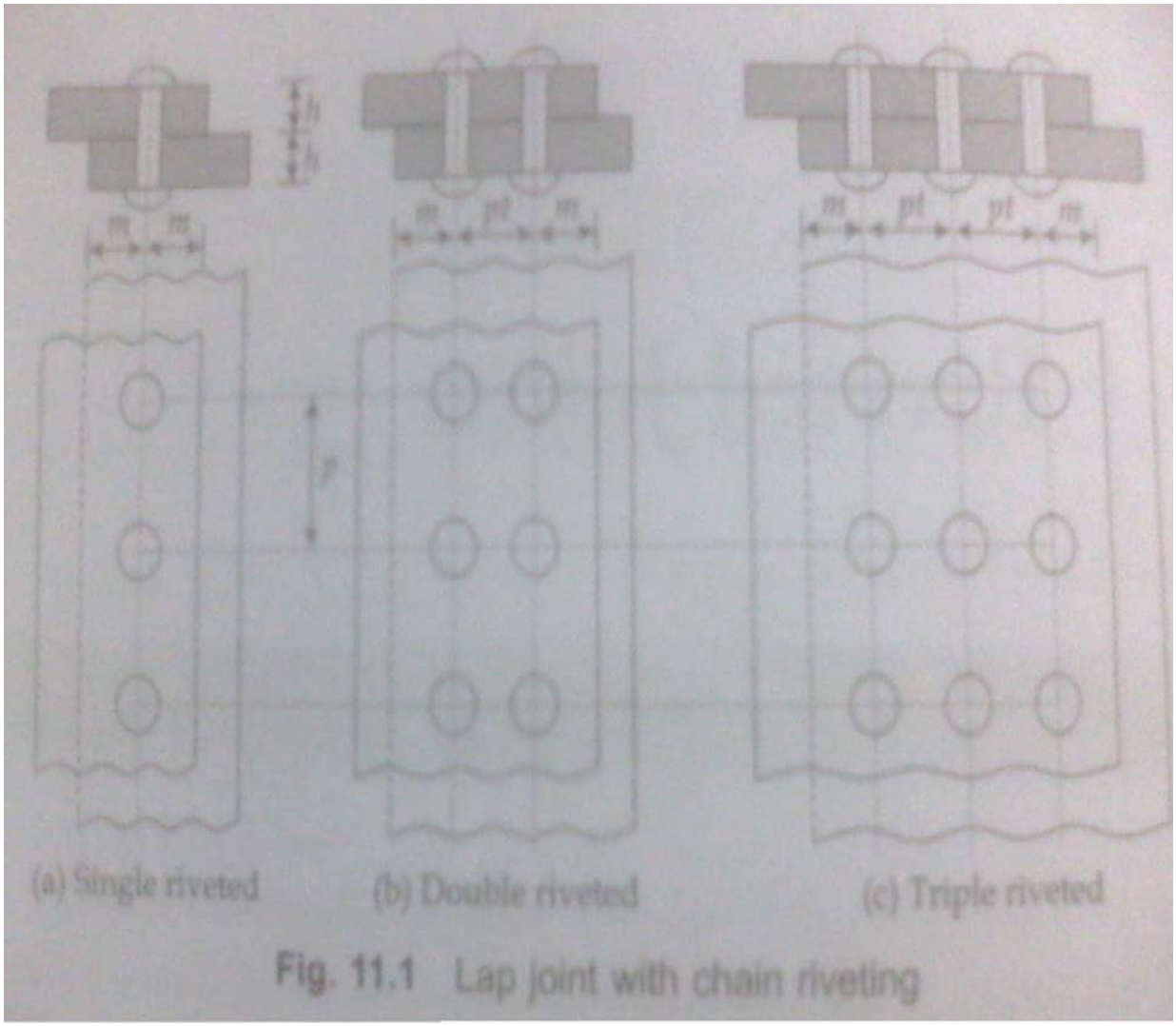


Fig. 11.1 Lap joint with chain riveting

NOTATIONS

Pitch, p = It is the distance between centres of two consecutive rivets in a row

Margin, m = It is the distance from end of plate to first low of rivets.

Transverse pitch, p_t = Distance between two consecutive rows of rivets.

Diagonal pitch,p_d = In zigzag riveting, the distance between centres of adjacent rivets inside by side rows

F = tensile or compressive load on plates

d= diameter of the rivet

d_h= diameter of rivet hole

h = thickness of main plate

h_1,h_2 = thickness of cover plates

i_1 = number of rivets in single shear in one pitch length or repetitive length of joint

i_2 = number of rivets in double shear in one pitch length

C= Corrosion allowance 1 to 3mm

K = factor as per boiler code

= efficiency

b = width of plate

σ_t= σ_θ= allowable tensile stress for plates

τ = allowable shear stress for rivets

σ_c= allowable crushing stress for rivets

RIVETED JOINTS FOR BOILERS

D_1= Inside diameter of the boiler

P_f = Fluid pressure or steam pressure

DESIGN PROCEDURE FOR BOILER JOINTS

Part-I Longitudinal joint

Step 1: Thickness of plate $h = \frac{P_f D_i}{2\,\sigma_\theta} + C$

= Efficiency of joint = 0.7 to 0.8

C = Corrosion allowance = 1 to 3mm

Step 2: Diameter of the rivet, $d = 6\sqrt{h}$

Step 3: Pitch of the rivets:

a.) $p = \frac{(i_1 + 1.875 i_2)\pi d^2 \tau}{4ph\sigma_\theta} + d_h$ and

b.) $p = Kh + 40$

Step 4: Transverse pitch $p_t = 2.25d$

Step 5: Margin $m = 1.5d$

Step 6: Thickness of cover plates:

- For equal covers $h_1 = h_2 = 0.625h$
- For unequal covers $h_1 = 0.625h$ and $h_2 = 0.625h$

Step 7: Efficiency

a.) Efficiency of plate, $\eta_p = \frac{p-d}{P}$

b.) Efficiency of rivets in shear, $\eta_\tau = \frac{i_1 + 1.875 i_2\, \pi d^2 \tau}{4ph\sigma_\Theta}$

c.) Efficiency of rivets in crushing, $\eta_c = \frac{i_2 + \frac{i_1 h_2 \sigma_c}{h}}{i_2 + \frac{i_1 h_2 \sigma_c}{h} + \sigma_\Theta}$

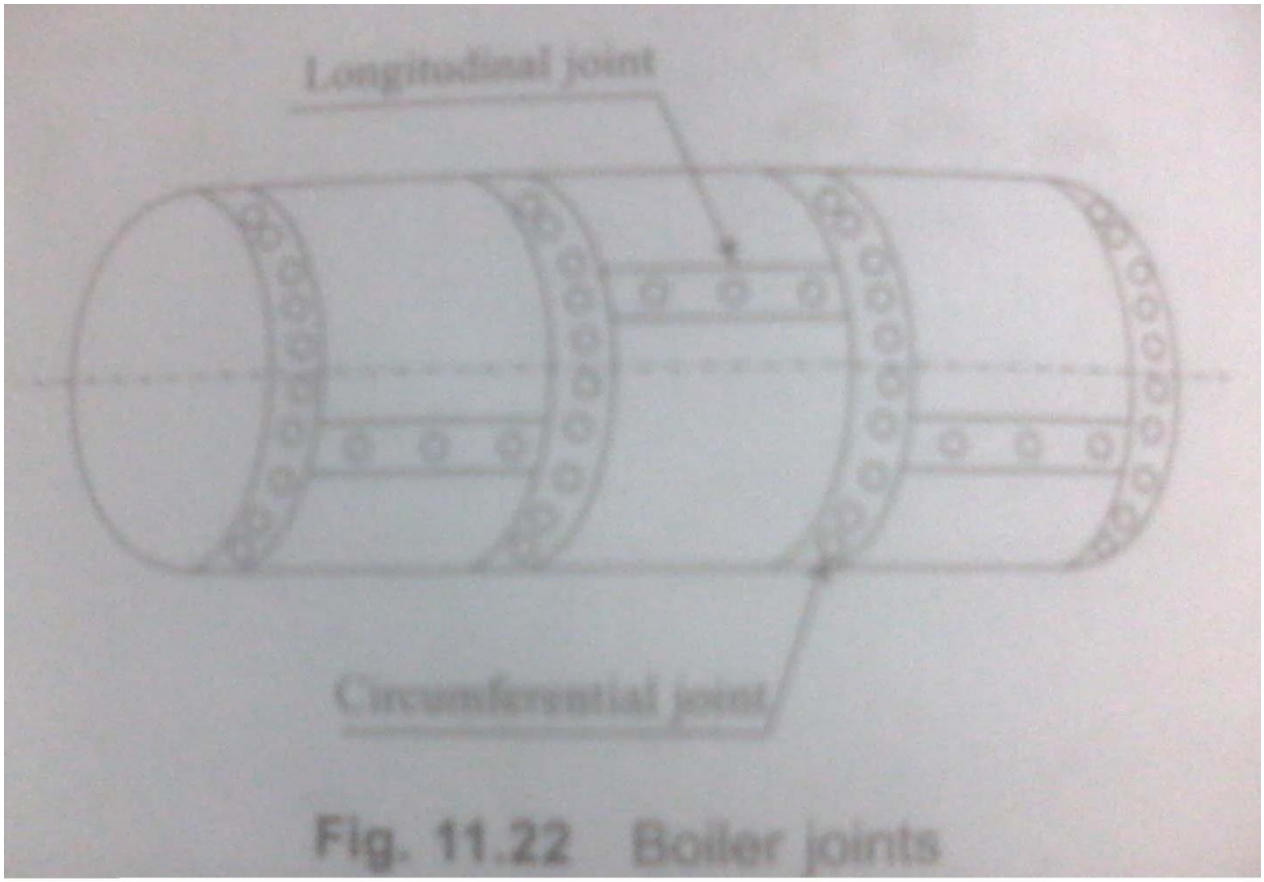


Fig. 11.22 Boiler joints

PART-II Circumferential joints

The plates to be riveted being same in both longitudinal and circumferential joints (h being same), the diameter of rivets, transverse pitch and margin are same.

h, d, p_t and m are same as above.

Step 1: Total steam load $F = \frac{\pi D i^2}{4}$

STEP 2: Strength of each rivet

a.) In shear $F_\tau = \frac{\pi d^2 \tau}{4}$

b.) In crushing $F_c = dh\sigma_c$

STEP 3: Number of rivets required = $\frac{\text{Steam Load}}{\text{Minimum strength of rivet}}$

$$i = \frac{F}{F_1}$$

Step 4: Rivets/Rows = i/row = $\frac{\text{Total number of rivets}}{\text{Number of rows of rivets}}$

Number of rows = 1 for single riveted joint

= 2 for double riveted joint

Step 5: Pitch for circumferential joint

$P_c = \frac{\text{Circumference}}{\text{Rivets/Row}} = \frac{\pi D_i}{\text{i/row}}$

Step 6: Efficiency

a.) Efficiency of plate, $\eta_p = \frac{p_c - d_h}{P_c}$

b.) For rivets in shear, $\eta_\tau = \frac{(i_1 + 1.875 i_2)\pi d^2 \tau}{4ph\sigma_\Theta}$

c.) For rivets in crushing, $\eta_c = \dfrac{i_2 + i_1 \dfrac{h_2}{h} \sigma_c}{i_2 + i_1 \dfrac{h_2}{h} \sigma_c + \sigma_\Theta}$