

# Mechanical Engineering

## Machine Design

---

Comprehensive Theory

with Solved Examples and Practice Questions

---



**MADE EASY**  
Publications



### **MADE EASY Publications Pvt. Ltd.**

Corporate Office: 44-A/4, Kalu Sarai (Near Hauz Khas Metro Station), New Delhi-110016

E-mail: infomep@madeeeasy.in

Contact: 011-45124660, 8860378007

Visit us at: [www.madeeasypublications.org](http://www.madeeasypublications.org)

### **Machine Design**

© Copyright by MADE EASY Publications Pvt. Ltd.

All rights are reserved. No part of this publication may be reproduced, stored in or introduced into a retrieval system, or transmitted in any form or by any means (electronic, mechanical, photo-copying, recording or otherwise), without the prior written permission of the above mentioned publisher of this book.

**First Edition: 2015**

**Second Edition: 2016**

**Third Edition: 2017**

**Fourth Edition: 2018**

**Fifth Edition: 2019**

**Sixth Edition: 2020**

**Seventh Edition: 2021**

****Eighth Edition: 2022****

# Contents

# Machine Design

## Chapter 1

### Design Against Fluctuating Load ..... 1

1.1	Introduction .....	1
1.2	Types of Stresses.....	1
1.3	Endurance Strength ( $\sigma_e$ ) .....	3
1.4	Stress Concentration/Stress Concentration Factor	3
1.5	Gerber Line/Soderberg Line/Goodman Line .....	8
1.6	Low & High Cycle Fatigue; Finite and Infinite Life problem .....	13
	<i>Objective Brain Teasers</i> .....	16
	<i>Student's Assignments</i> .....	19

## Chapter 2

### Cotter and Knuckle Joint..... 20

2.1	Cotter .....	20
2.2	Construction of a Cotter Joint.....	21
2.3	Knuckle Joint.....	24
	<i>Objective Brain Teasers</i> .....	29
	<i>Student's Assignments</i> .....	30

## Chapter 3

### Bolted, Welded and Riveted Joints ..... 31

3.1	Introduction.....	31
3.2	Bolted Joint.....	31
3.3	Welded Joints.....	37
3.4	Riveted Joint.....	44
	<i>Objective Brain Teasers</i> .....	51
	<i>Student's Assignments</i> .....	52

## Chapter 4

### Screw Threads and Power Screw ..... 53

4.1	Threads .....	53
4.2	Power Screw .....	57
4.3	Self-Locking Screw .....	59
4.5	Coefficient of Friction for Thrust Collar Sliding Contact.....	61
	<i>Objective Brain Teasers</i> .....	66
	<i>Student's Assignments</i> .....	67

## Chapter 5

### Shaft, Key and Coupling..... 68

5.1	Shaft .....	68
5.2	Shaft Design.....	69
5.3	Shaft Design on Torsional Rigidity Basis.....	70
5.4	Design of Hollow Shaft on Strength Basis .....	71
5.5	Design on the Basis of Maximum Principal Stress Theory .....	72
5.6	Design on the Basis of Maximum Share Stress Theory .....	72
5.7	Design of Hollow Shaft on Torsional Rigidity Basis .....	72
5.8	Keys .....	74
5.9	Material of Keys .....	74
5.10	Heavy Duty Keys.....	79
5.11	Couplings.....	81
5.12	Rigid Flange Coupling.....	84
	<i>Objective Brain Teasers</i> .....	87
	<i>Student's Assignments</i> .....	89

## **Chapter 6**

### **Chain and Belt Drive.....90**

6.1	Chain Drive .....	90
6.2	Roller Chains .....	90
6.3	Terms Used in Chain Drive .....	91
6.4	Nomenclature of Chain Drive .....	91
6.5	Relation between pitch and pitch circle diameter and speed of sprockets, length of chain .....	91
6.6	Difference between Gear and Sprocket .....	93
6.7	Rules for Design of Chain Drive.....	93
6.8	Belt Drive.....	93
6.9	Types of Belt Drives .....	94
6.10	Types of Belts .....	95
6.11	Belt Joints.....	96
6.12	Efficiencies of Belt Joints .....	97
6.13	Velocity Ratio of a Belt Drive .....	97
6.14	Slip of Belt .....	97
6.15	Length of an Open Belt Drive .....	98
6.16	Length of a Cross Belt Drive .....	98
6.17	Power Transmitted by a Belt.....	98
6.18	Ratio of Driving Tensions for Flat Belts .....	99
6.19	Centrifugal Tension.....	99
6.20	Maximum Tension in the Belt .....	99
6.21	Condition for the Transmission of Maximum Power .....	100
6.22	Tension in the Belt .....	100
6.23	Flat Belt Pulleys .....	100
	<i>Objective Brain Teasers .....</i>	104
	<i>Student's Assignments .....</i>	106

## **Chapter 7**

### **Clutches.....107**

7.1	Introduction .....	107
7.2	Location of a clutch.....	107
7.3	Type of Clutches.....	108

7.4	Principle of Friction Clutches.....	108
7.5	Cone Clutches .....	113
7.6	Centrifugal Clutch.....	114
	<i>Objective Brain Teasers .....</i>	117
	<i>Student's Assignments .....</i>	120

## **Chapter 8**

### **Brakes .....121**

8.1	Introduction.....	121
8.2	Types of brakes .....	121
	<i>Objective Brain Teasers.....</i>	131
	<i>Student's Assignments .....</i>	133

## **Chapter 9**

### **Gears .....134**

9.1	Gears .....	134
9.2	Spur Gears .....	134
9.3	Helical Gear .....	141
9.4	Worm and Worm Gear .....	144
9.5	Bevel Gears.....	145
	<i>Objective Brain Teasers.....</i>	150
	<i>Student's Assignments .....</i>	151

## **Chapter 10**

### **Bearing .....152**

10.1	Bearing.....	152
10.2	Classification of Bearing .....	152
10.3	Type of Rolling Contact Bearing.....	153
10.4	Needle Bearing .....	156
10.5	Sliding Contact bearing.....	156
	<i>Objective Brain Teasers .....</i>	164
	<i>Student's Assignments .....</i>	166



# Cotter and Knuckle Joint

## 2.1 Cotter

A cotter is a flat wedge-shape piece of rectangular cross-section and its width is tapered from one end to another for an easy adjustment. It is usually made of mild steel or wrought iron.

### Use of Cotter Joint

A cotter joint is used to connect **two co-axial rods**, which are subjected to either axial tensile force or axial compressive force. It is not used for connecting shafts that rotate and transmit torque.

### Application of Cotter Joints

- Joints between the piston rod and the cross-head of a steam engine.
- Joint between the piston rod and the tail or pump rod.
- Joint between the slide spindle and the fork of the valve mechanism.
- Foundation Bolt.

### Principle of Cotter Joint

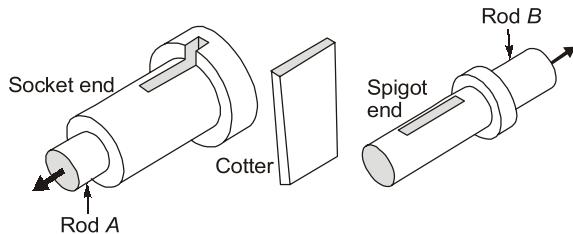
- The principle of wedge action is used in a cotter joint.
- The joint is tightened and adjusted by means of a wedge action of the cotter.
- Cotter has uniform thickness and the width dimension has a slight taper.
- The taper is usually 1 in 24. Due to taper shape, it is easy to remove the cotter and dismantle the joint.
- The taper of the cotter as well as slots is on one side because machining a taper on two side is more difficult.
- A clearance of 1.5 mm to 3 mm is provided between the slots and the cotter.

**Remember :** **Draw of Cotter:** The amount by which the two rods are drawn together is called the draw of the cotter.

### Advantage of Cotter Joint

- (a) Assembly and dismantling the parts of the cotter joint is quick and simple.
- (b) The wedge action develops a very high tightening force, which prevents loosening of parts in service.
- (c) Joint is simple to design and manufacture.

## 2.2 Construction of a Cotter Joint



**Figure 2.1**

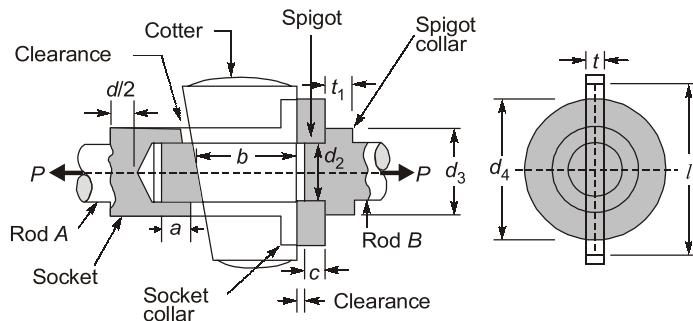
The construction of cotter joint used to connect two rods *A* and *B* is shown figure 2.1. Rod-*A* is provided with a socket and while Rod-*B* is provided with a spigot end. The socket end of Rod-*A* fits over the spigot end of Rod-*B*. The socket as well as the spigot is provided with a narrow rectangular slot. A cotter is tightly fitted in this slot passing through the socket and the spigot.

**NOTE :** Taper in the cotter is provided to facilitate its removal when it fails due to shear.

### 2.2.1 Design procedure for Cotter Joint

**Assumption for stress analysis:**

- The rods size subjected to axial tensile force.
- Effect of stress concentration due to slot is neglected.
- Stress due to initial tightening of the cotter are neglected.



**Figure 2.2**

Here,

- $P$  = Tensile force acting on rod  
 $d$  = Diameter of each rod  
 $d_1$  = Outside diameter of socket  
 $d_2$  = Diameter of spigot or inside diameter of socket  
 $d_3$  = Diameter of spigot collar  
 $d_4$  = Diameter of socket collar  
 $a$  = Distance from end of slot to the end of spigot on Rod-B  
 $b$  = Mean width of cotter  
 $c$  = Axial distance from slot to end of socket collar  
 $t$  = Thickness of cotter  
 $t_1$  = Thickness of spigot-collar  
 $l$  = Length of cotter

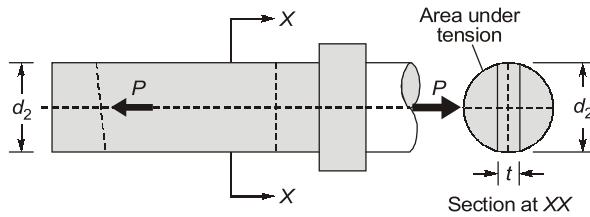
**(i) Tensile Failure of Rods,**

Permissible tensile stress

$$\sigma_t = \frac{P}{\frac{\pi}{4}d^2} \Rightarrow d = \sqrt{\frac{4P}{\pi\sigma_t}}$$

**(ii) Tensile Failure of Spigot**

The weakest cross-section is at X-X of spigot end, which is subjected to tensile stress.

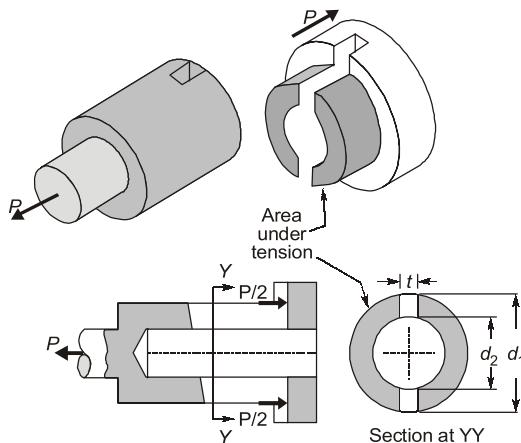
**Figure 2.3**

$$\text{Area of section at } X-X = \frac{\pi}{4}d_2^2 - d_2t$$

$$\therefore \text{Permissible tensile stress, } \sigma_t = \frac{P}{\frac{\pi}{4}d_2^2 - d_2t}$$

Thickness of cotter is usually determined by empirical relationships

$$t = 0.31 d_2$$

So, we can find inner diameter ( $d_2$ ) of socket.**(iii) Tensile failure of Socket****Figure 2.4**

From the above figure, weakest section is Y-Y

$$\text{Area of section at } Y-Y = \left[ \frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t \right]$$

$$\therefore \text{Permissible tensile stress, } \sigma_t = \frac{P}{\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t}$$

Because ' $d_2$ ' and 't' is known, we can find outside diameter of socket ( $d_1$ ).

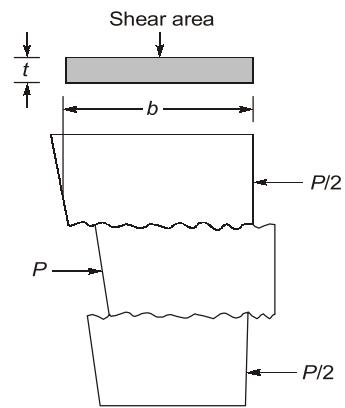
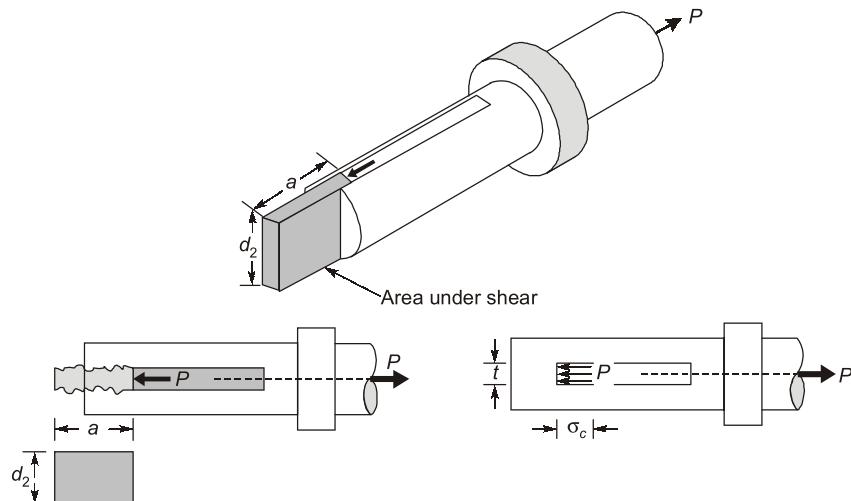
**(iv) Shear Failure of Cotter**

The cotter is subjected to double shear.

$$\text{Area of shear plane} = bt$$

$$\therefore \text{Permissible shear stress, } \tau = \frac{P}{2bt}$$

$\therefore$  Mean width of cotter ( $b$ ) can be calculated.


**Figure 2.5**
**(v) Shear failure of Spigot End**

**Figure 2.6**

Spigot end is subjected to double shear. Area of plane that resist shear failure =  $a.d_2$

$$\therefore \text{Permissible shear stress, } \tau = \frac{P}{2(ad_2)}$$

Thus, we can calculate ' $a$ ' from above equation.

**(vi) Shear failure of Socket End**

Socket end is subject to double shear.

$$\text{Area subjected to shear} = (d_4 - d_2)C$$

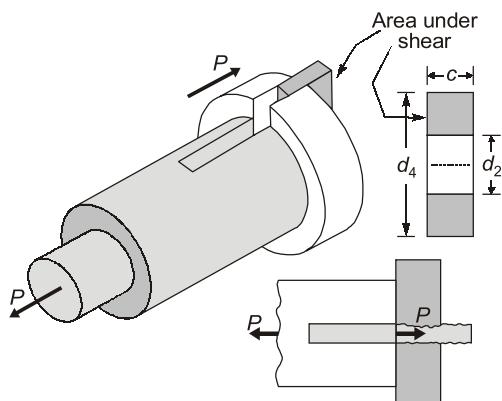
$$\therefore \text{Permissible shear stress, } \tau = \frac{P}{2(d_4 - d_2)C}$$

Thus we can find out  $C$ .

**(vii) Crushing failure of Spigot End**

Area, subjected to compressive stress =  $\tau \times d_2$

$$\therefore \text{Permissible compressive stress, } \sigma_c = \frac{P}{\tau \times d_2}$$


**(viii) Crushing failure of Socket End**

Area subjected to compressive stress =  $(d_4 - d_2) \times t$

$$\therefore \text{Permissible compressive stress} = \sigma_c = \frac{P}{(d_4 - d_2) \times t}$$

**Figure 2.7**

**(ix) Calculation of  $a, c, d_3, d_4, t_1$  by empirical relationship**

$$\begin{aligned}d_3 &= 1.5 d & d_4 &= 2.4 d \\a &= c = 0.75 d & t_1 &= 0.45\end{aligned}$$

**(x) If material of Rod is not given**

Take material - Plain carbon steel [3068]

$$\sigma_{yt} = 400 \text{ N/mm}^2$$

**2.3 Knuckle Joint**

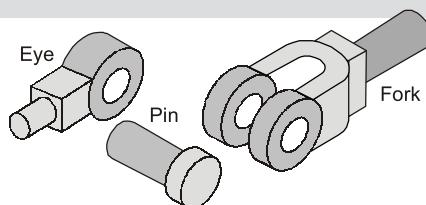
It is used to connect two rods whose axes either coincide or intersect and lie in one plane and used to transmit only axial tensile force. This joint permits limited angular movement between rods, about the axis of the pin. It is not used to connect rods, that rotate and transmit torque.

**Application of Knuckle Joint**

- (i) Joint between tie bar and roof trusses.
- (ii) Joint between link of suspension bridge.
- (iii) Joints in valve mechanism of reciprocating engine.
- (iv) Fulcrum for the levers.
- (v) Joint between link of a bicycle chain.



In rare application, a knuckle joint is used to connect three rods - two with forks and third with eye.

**Advantage of Knuckle Joint**

- (a) Joint is simple to design and manufacture.
- (b) Due to few part, cost is less and high reliability.
- (c) Assembly and dismantling is quick and simple.

**2.3.1 Design procedure of Knuckle Joint****Assumption for stress analysis**

- (i) Rods are subjected to only axial tensile force.
- (ii) The effect of stress concentration due to hole is neglected.
- (iii) Force is uniformly distributed in various parts.

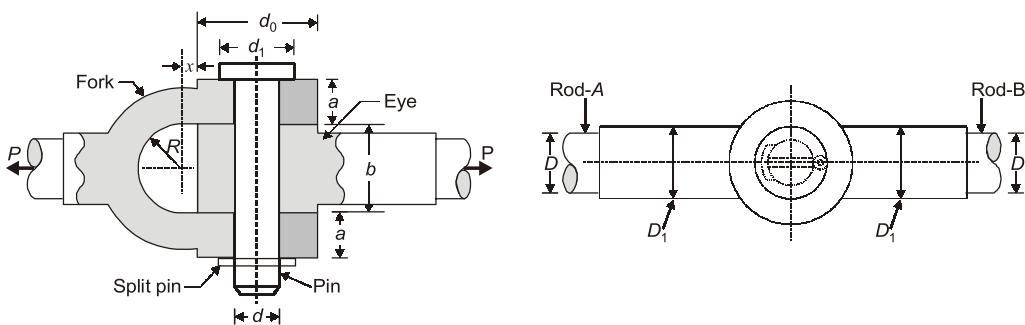


Figure 2.8

- Here,
- $D$  = Diameter of each rod
  - $D_1$  = Enlarged diameter of each rod
  - $d$  = Diameter of knuckle pin/Inside diameter of fork/eye
  - $d_0$  = Outside diameter of eye or fork.
  - $a$  = Thickness of each eye of fork (Rod-A)
  - $b$  = Thickness of eye of Rod-B
  - $d_1$  = Diameter of pin head
  - $x$  = Distance of centre of fork radius R from the eye.

### (i) Tensile Failure of Rod

Permissible tensile stress,

$$\sigma_t = \frac{P}{\pi D^2} = \frac{4P}{\pi D^2}$$

For calculation of  $D_1$ , use empirical relationship

$$D_1 = 1.1 D$$

### (ii) Shear Failure of Pin

The pin is subjected to double shear

Permissible shear stress,

$$\tau = \frac{P}{2 \times \left( \frac{\pi}{4} d^2 \right)}$$

$$\Rightarrow d = \sqrt{\frac{2P}{\pi \tau}}$$

Standard diameter of pin( $d$ ) should be

$$d = D$$

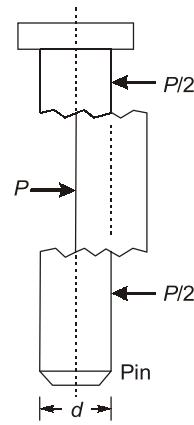


Figure 2.9

### (iii) Crushing Failure of Pin in Eye

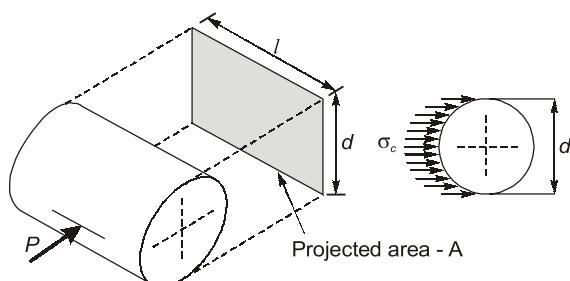


Figure 2.10

When a cylindrical surface is subjected to a force along its periphery, its projected area (not actual area) is taken into consideration.

$$\therefore \text{Permissible crushing stress, } \sigma_c = \frac{P}{l \times d}$$

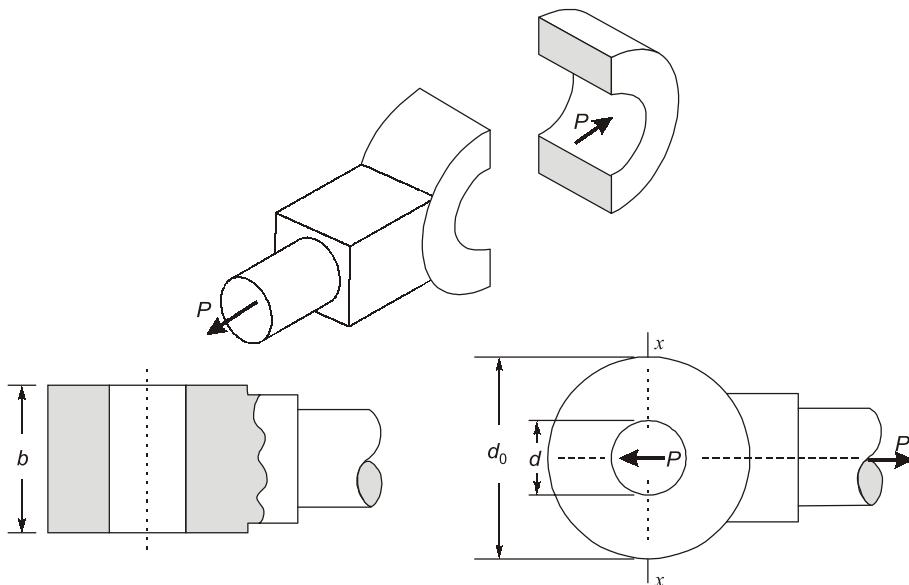
The projected area of the pin in the eye is  $b \times d$ , thus, permissible crushing stress,

$$\sigma_c = \frac{P}{b \times d}$$

**(iv) Crushing Failure of Pin in the Fork**

Total projected area of pin in fork =  $2ad$

$$\therefore \text{Permissible crushing stress, } \sigma_c = \frac{P}{2ad}$$

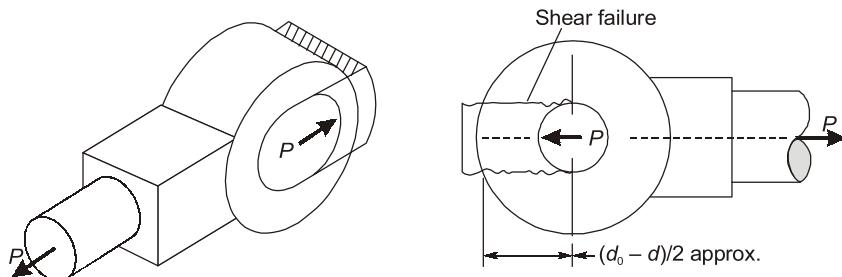
**(v) Tensile Failure of Eye**

**Figure 2.11**

In figure,  $X-X$  section is the weakest section of the eye.

$$\text{Area at } X-X = b(d_0 - d)$$

$$\therefore \text{Permissible tensile stress, } \sigma_t = \frac{P}{b(d_0 - d)}$$

**(vi) Shear Failure of Eye**

**Figure 2.12**

Eye is subjected to double shear

$$\therefore \text{Area of each shear plane} = \frac{b(d_0 - d)}{2}$$

$$\therefore \text{Permissible shear stress, } \tau = \frac{P}{2 \left[ \frac{b(d_0 - d)}{2} \right]} = \frac{P}{b(d_0 - d)}$$

$$\text{Standard proportion, } d_0 = 2d$$

### (vii) Tensile Failure of Fork

Fork has a double eye

$$\text{Area of weakest section} = a(d_0 - d)$$

$$\text{Permissible tensile stress in the fork, } \sigma_t = \frac{P}{2a(d_0 - d)}$$

### (viii) Shear Failure of Fork

Each eye of fork subjected to double shear.

$$\therefore \text{Permissible shear stress, } \tau = \frac{P}{2b(d_0 - d)} \quad [\text{from 6}]$$

Standard proportion for 'a' and 'b' and 'd<sub>1</sub>', 'x'

$$a = 0.75D \quad b = 1.75D$$

$$d_1 = 1.5d \quad x = 10 \text{ mm (usually)}$$

### (ix) Bending Failure of Pin

- When the pin is tight in the eye and the fork, failure occur due to shear, when the pin is loose failure occur due to bending moment.
  - Load acting on the pin is uniformly distributed in the eye, but uniformly varying in the two part of fork.
- $\therefore$  Permissible bending stress

$$\sigma_b = \frac{M_b y}{I} = \frac{\frac{P}{2} \left[ \frac{b}{4} + \frac{a}{3} \right] d}{\frac{\pi d^4}{64}} = \frac{32}{\pi d^3} \times \frac{P}{2} \left[ \frac{b}{4} + \frac{a}{3} \right]$$

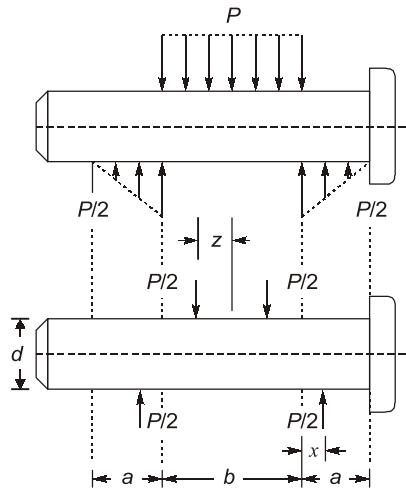


Figure 2.13

**Example 2.1** Design a socket and spigot cotter joint for transmission of 25 kN tensile forces for given allowable stress,  $\sigma_t = 50 \text{ MPa}$ ,  $\sigma_c = 80 \text{ MPa}$ ,  $\tau = 40 \text{ MPa}$ .  $P = 25 \text{ kN}$ .

**Solution :**

Rod:

$$\sigma_t = \frac{P}{\pi / 4 \times d^2} = 50$$

$$d = 25.23 = 30 \text{ mm}$$

$$t = \frac{d}{4} = \frac{30}{4} = 7.5 \text{ mm}$$

Spigot:

$$\sigma_t = \frac{P}{\left[ \frac{\pi}{4} d_1^2 - d_1 t \right]} = 50$$

$\Rightarrow$ 

$$d_1 = 30.45 \approx 35 \text{ mm}$$

$$\sigma_c = \frac{P}{d_1 t} = 95.23 > 80 \text{ MPa design fails.}$$

$$80 = \frac{25 \times 10^3}{35 \times t} \Rightarrow t = 8.9 \text{ mm} \approx 10 \text{ mm}$$

Socket

$$\sigma_t = \frac{P}{\pi / 4 (d_2^2 - d_1^2) - (d_2 - d_1)t} = 50$$

 $\Rightarrow$ 

$$d_2 = 44.5 \approx 45 \text{ mm}$$

$$\sigma_c = \frac{P}{(d_3 - d_1)t} = 80 \Rightarrow d_3 = 66.25 \approx 70 \text{ mm}$$

Cotter

$$\tau = \frac{P}{2bt} \Rightarrow b = 31.25 \text{ mm} \approx 35 \text{ mm}$$

$$L = (D_3 + 20) = 90 \text{ mm}$$

**Example 2.2** Design a gib and cotter joint for 2 rods of square cross-section carrying an axial tensile load of 40 kN. Cotter and rods (same material –  $\sigma_t = 60 \text{ MPa}$ ,  $\sigma_c = 125 \text{ MPa}$ ,  $\tau = 45$ ).

**Solution :**

Rod

$$\sigma_t = \frac{P}{a(a-t)} = 60$$

$$a = 31.62 \approx 35 \text{ mm}$$

 $\therefore$ 

$$t = \frac{35}{3} = 11.66 \approx 12 \text{ mm}$$

$$\sigma_c = \frac{P}{at} = 95.23 < 125 - \text{design is safe}$$

$$\tau = \frac{P}{2aL_1}$$

 $\Rightarrow$ 

$$L_1 = 12.69 \text{ mm} \approx 15 \text{ mm}$$

Strap

$$\sigma_t = \frac{P}{(a-t)t_1/2} = 60$$

 $\Rightarrow$ 

$$t_1 = 14.49 \approx 15 \text{ mm}$$

$$\sigma_c = \frac{P}{2tt_1} = 11411 < 125 \text{ safe}$$

$$\tau = \frac{P}{4t_1L_2} = 45$$

 $\Rightarrow$ 

$$L_2 = 14.81 \approx 15 \text{ mm}$$

$$t = 12 \text{ mm}$$

$$\tau = 45 = \frac{P}{2(b+b_1)t}$$

$$b = 16.46 \approx 18 \text{ mm}$$

 $\therefore$ 

$$b_1 = 22.5 \approx 24 \text{ mm}$$

and

$$L_3 = 0.79 = 24.5 \text{ mm} \approx 25 \text{ mm}$$



## Objective Brain Teasers

- Q.1** A cotter joint is used when no relative motion is permitted between the rods joined by the cotter.

It is capable of transmitting

- twisting moment
- an axial tensile as well as compressive load
- the bending moment
- only compressive axial load

- Q.2** In a gib and cotter joint, the gib and cotter are subjected to

- Single shear only
- Double shear only
- Single shear and crushing
- Double shear and crushing

- Q.3** Permissible shear stress in socket end

$$(a) \tau = \frac{P}{2(d_o - d_i)C}$$

$$(b) \tau = \frac{P}{(d_o + d_i)C}$$

$$(c) \tau = \frac{2P}{(d_o + d_i)C}$$

$$(d) \tau = \frac{P}{(d_o + d_i)2C}$$

where,  $C$  = Axial distance from slot to end of socket collar

- Q.4** The taper on cotter is usually

- 1 in 24
- 1 in 8
- 1 in 100
- 1 in 48

- Q.5** A taper is provided for cotter

- to ensure tightness in operating condition
- to provide wedge action
- to ease the removal of cotter during dismantling
- for all three reasons

- Q.6** The joint between the piston rod and the cross head of steam engine is

- Kunkle joint
- Universal joint
- Cotter joint
- Key joint

- Q.7** Cotter joint is used for the joint between

- piston rod and cross-head of steam engine
- slide spindle and fork of valve mechanism
- piston rod and tail rod or pump rod
- for all three applications

- Q.8** A knuckle joint is used to transmit

- axial tensile force only
- axial tensile or compressive force
- axial compressive force only
- combined bending and torsional moment

- Q.9** The pin in knuckle joint is subjected to

- double shear stress
- torsional shear stress
- axial tensile stress
- axial compressive stress

### Answers

- |        |        |        |        |        |
|--------|--------|--------|--------|--------|
| 1. (b) | 2. (d) | 3. (a) | 4. (a) | 5. (d) |
| 6. (c) | 7. (d) | 8. (a) | 9. (a) |        |

### Hints and Explanations:

- 1. (b)**

Note that cotter joint is not used for connecting two shafts which are rotating and transmitting torque. It is used to transmit axial tensile as well as compressive load.


**Student's  
Assignments**

- Q.1** Two rods are connected by means of a cotter joint. The inside diameter of the socket and outside diameter of the socket collar are 50 and 100 mm respectively. The rods are subjected to a tensile force of 50 kN. The cotter is made of steel 30C8 ( $S_{yt} = 400 \text{ N/mm}^2$ ) and the factor of safety is 4. The width of the cotter is five times of thickness. Calculate:
- width and thickness of the cotter on the basis of shear failure; and
  - width and thickness of the cotter on the basis of bending failure.

- Q.2** A gib and cotter joint is used for strap-type big end of the connecting rod. The rod is subjected to a maximum pull of 50 kN and the diameter of circular part of the rod adjacent to strap end is 60 mm. The strap, gib and cotter are made of plain carbon steel of grade 30C8 ( $S_{yt} = 400 \text{ N/mm}^2$ ) and the factor of safety is 10. There are three reason for recommending a rather high factor of safety, namely – (i) Big end of connecting rod is critical part in engine design; (ii) The analysis used for calculating dimensions of the joint is too elementary; and (iii) There is stress concentration due to oil hole in the strap which is neglected. Calculate:
- the width of strap;
  - the thickness of strap at the thinnest cross-section and at cross-section passing through the slot; and
  - thickness and widths of gib and cotter at mid section.

