DARBHAMGA COLLEGE OF ENGINEERING

DARBHANGA, BIHAR

COURSE FILE

OF

DESIGN OF MACHINE ELEMENTS

(02 1615)

FACULTY NAME:

PRASHANT KUMAR SINGH

ASSISTANT PROFESSOR,

DEPARTMENT OF MECHANICAL ENGINEERING

Vision of the Mechanical Engineering Department:

To bring forth quality engineers embodying societal ethics to serve national and multinational organisations as well as harping on higher studies.

Mission of the Mechanical Engineering Department:

1. To create a modern ambiance focusing on advanced pedagogy and tools for mechanical engineers.
2. To collaborate with domain industry and research institutes to enhance the skills and knowledge of the graduates.
3. To inject necessary professional skills to serve the industry and the nation.
4. To inculcate humanitarian ethical values in graduates through various social-cultural activities.

Program Educational Objectives (PEOs) :

PEO 1	The graduates will be able to demonstrate knowledge and skills of mechanical engineering to obtain solution to engineering problems.
PEO 2	The graduate will able to apply the mechanical engineering concepts while pursuing academic and research activities.
PEO 3	The graduates will be able to showcase professional skill and expertise.

Program Outcomes (POs) :

PO 1	**Engineering knowledge:** An ability to apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization to get the solution of the engineering problems.
PO 2	**Problem analysis:** Ability to Identify, formulates, review research literature, and analyze complex engineering problems.
PO 3	**Design/development of solutions**: Ability to design solutions for complex engineering problems by considering social, economic and environmental aspects
PO 4	**Conduct investigations of complex problems:** Use research-based knowledge to design, conduct analyze experiments to get valid conclusion.
PO 5	**Modern tool usage:** ability to create, select, and apply appropriate techniques, and to model complex engineering activities with an understanding of the limitations.
PO 6	**The engineer and society:** Ability to apply knowledge by considering social health, safety, legal and cultural issues.
PO 7	**Environment and sustainability:** Understanding of the impact of the adopted engineering solutions in social and environmental contexts.
PO 8	**Ethics:** Understanding of the ethical issues of the Mechanical engineering and applying ethical principles in engineering practices.
PO 9	**Individual and teamwork:** Ability to work effectively as an individual or in team, as a member or as a leader.
PO 10	**Communication:** An ability to communicate clearly and effectively through different modes of communication.
PO 11	**Project management and finance:** Ability to handle project and to manage finance related issue
PO 12	**Life-long learning:** Recognize the need for, and have the preparation and ability to engage in independent and life-long learning.

Program Specific Outcomes (PSOs) :

PSO 1	Students will be oriented towards research in engineering technologies like Advance Manufacturing, 3 D Printing, Alternative Fuels to contribute the evolving research and development in the field of Mechanical Engineering.
PSO 2	Students will be able to learn and apply software like AutoCAD, Ansys, Catia for various applications.

Course Description

In this course students will probably start with basic understanding of design of machine element, whether it is an automobile or other consumer products. In this course we will study to design an existing machine or new machine with the help of knowledge of scientific principle, technical information and imagination.

Course Objectives

The objectives of the course are to:

• Cover the basics of machine design, including the design process, engineering mechanics and materials, failure prevention under static and variable loading, and characteristics of the principal types of mechanical elements.

• Offer a practical approach to the subject through a wide range of real-world applications and examples.

• Encourage students to link synthesis and analysis.

• Encourage students to link fundamental concepts with practical component specification.

Course Outcomes

At the end of the course students will be able to

1. Understand the various components used for assembly.
2. Apply the scientific principle and technical information to analyse the existing components.
3. Relate the manufacturing with design.
4. Calculate the dimensions of the mechanical component by using fundamental equations.
5. Select suitable mechanism from given alternative mechanism.

SYLLABUS FOR DESIGN OF MACHINE ELEMENTS

1. Introduction : Engineering material and their properties, Manufacturing consideration in machine design, factor of safety.
2. Simple stresses in machine parts, torsional and bending stresses, dynamic loads, stress concentration.
3. Design of riveted joints, welded joints, bolted joint, cotter joint, knuckle joint, pressure vessels and pipe joints.
4. Design of keys, couplings, shafts levers, columns, studs, power screw, belt drive, pulley.
5. Springs, clutches and brakes.

GATE SYLLABUS FOR DESIGN OF MACHINE ELEMENTS

Design for static and dynamic loading; failure theories; fatigue strength and S-N diagram; principles of the design of machine elements such as bolted, riveted, and welded joints; shafts, gears, rolling and sliding contact bearings, brakes and clutches, springs.

DARBHANGA COLLEGE OF ENGINEERING

COURSE FILE

OF

DESIGN OF MACHINE ELEMENT

(02 1615)

Mr. Prashant Kumar Singh

Assistant Professor

Department of Mechanical Engineering

College Name	**Darbhanga College of Engineering**		
Program Name	**B.Tech Mechanical Engineering**		
Course Name	Design of Machine Element		
Course Code		Course Credit	5
Lecture/Tutorial Per Week	03/00		
Course Coordinator Name	Mr. Prashant Kumar Singh		

1.Scope and Objectives of the Course

In this course students will probably start with basic understanding of design of machine element, whether it is an automobile or other consumer products. In this course we will study to design an existing machine or new machine with the help of knowledge of scientific principle, technical information and imagination.

The objectives of the course are to:

• Cover the basics of machine design, including the design process, engineering mechanics and materials, failure prevention under static and variable loading, and characteristics of the principal types of mechanical elements.

• Offer a practical approach to the subject through a wide range of real-world applications and examples.

• Encourage students to link synthesis and analysis.

• Encourage students to link fundamental concepts with practical component specification.

2.Text Books

TB 1 : Design of Machine Element by V B Bhandari, Third Edition ,Mc Graw Hill publication

TB 2 : Singley's Mechanical Engineering Design by Richard G. Budyans , Ninth Edition,

3. Reference books

RB 1: Machine Design by Timothy H. Wentzell, P. E.

RB 2: Machine Design by R. S. Khurmi and J. K Gupta, S Chand publication.

3. Other readings and relevant websites :

S. No.	Link of websites
1	http://nptel.ac.in/downloads/112105125/
2	https://www.youtube.com/watch?v=nqhyCzrFp1s&list=PLHpC4_VH4uh0bIKMtFg0hXFckep6sBzwi

4. Course Plan

Lecture No.	Date of Lecture	Topics	Web links for video lectures	Text Books, Reference Books and other reading materials	Page numbers of the text books
1- 4		**Introduction**		TB 1, RB 2	1- 75
		Machine Design, basic procedure of machine design, design of machine element, Engineering materials, Manufacturing considerations in machine design	https://www.youtube.com/watch?v=mzWMdZZaHwI&list=PL3D4EECEFAA99D9BE		
		Assignment 1			
5- 12		**Stress in Machine Parts**		TB 1, RB 2	76- 84 and 101- 177
		Simple stresses, bending stresses, torsional stresses, eccentric axial loading, static load, factor of safety, stress concentration, Design against fluctuating loading, fatigue failure, endurance limit, Reversal stresses- design for finite and infinite life	https://www.youtube.com/watch?v=2xLHFiBOA4M&index=7&list=PL3D4EECEFAA99D9BE		
		Assignment 2			
13- 22		**Design of riveted joint, welded joint, bolted joint, pressure vessels**			231- 235, 272- 325 and 768- 791
		Bolted joint- simple analysis, eccentrically loaded bolted joint in shear, welded joints, strength of butt and fillet joint, maximum shear stress in parallel fillet weld, riveted joint, type of riveted joint, strength equations, efficiency of joint, thin cylinders, thick	https://www.youtube.com/watch?v=C5ZPaCvoigw&list=PL3D4EECEFAA99D9BE&index=22		

		cylinders- principal stresses, Lames Equation,Clavarino’s and Birnie’s equations, Autofrettage, compound cylinder			
		Assignment 3			
22- 30		**Design of keys, coupling and shafts, columns and studs**		TB 1,TB 2 and RB 2	TB 1 (330-376), TB 2 (181- 190)
		Solid and hollowShafts designed on strength and torsional rigidity basis, ASME code for shaft design,Keys, Types of keys, design of square and flat keys, design of kennedy key, Couplings, Muff coupling design, Rigid flange coupling design,Columns and type of columns, Euler’s equation, Rankine’s theory, Instability of column	https://www.youtube.com/watch?v=dKfriV8H9-8&index=34&list=PL3D4EECEFAA99D9BE		
Assignment 4					
31- 42		**Power screw, belt drive, pulley, springs, clutches and brakes**		TB 1 and RB 2	184- 206, 499-540, 393- 439 and 448- 496
		Forms of threads, multiple threaded screws, terminology of power screw, Torque requirement- lifting and lowering load, self locking screw, Efficiency of screws, trapezoidal and Acme thread, Belt constructions, geometrical relationships, analysis of belt tensions, condition for maximum power, selection of belts from manufacturer catalogue, Pulleys for flat belt and V	https://www.youtube.com/watch?v=PEKfS2Q1WqM&list=PL3D4EECEFAA99D9BE&index=19		

		belts, Torque transmitting capacity of clutches, multi disc clutches, cone clutches, centrifugal clutchges, Energy Equation, Brakes, energy equations, block brake with shoe, Internal expanding brake, band brake and disc brake			
Assignment 5					

5.Evaluation Scheme

Component 1	Mid semester examination	20
Component 2	class test	5
Component 3	TA	5
Component 4	End Semester Examination	70
Total		100

6. Syllabus

Topics	**No. of lectures**	**Weightage**
Introduction: Engineering Materials and their properties, Manufacturing consideration in design, factor of safety	4	9%
Simple stresses in machine parts, torsional and bending stresses, stress concentration	4	10%
Design of riveted joint, welded joint and bolted joint, cotter joint, knuckle joint, pressure vessels and pipe joints	12	28%
Design of keys, coupling and shafts, columns, studs, power screws, elt drive and pulleys	14	33%
Springs, clutches and brakes	8	20%

7. This document is approved by

Designation	**Name**	**Signature**
Course Coordinator	**Prashant Kumar Singh**	
HoD	**Mr. Vishnu Singh**	
Principal	**Dr. Achintya**	

Topics	Lecture Number
Introduction	
Engineering materials and their properties (Ferrous Material)	1
Non ferrous material	2
Manufacturing consideration in machine design	3
Allowable stresses and factor of safety	4
Stresses in machine parts	
Simple stresses in machine parts	5
Torsional stresses in machine parts	6
Fluctuating loading	7
stress concentration	8
Design of joints	
Design of riveted joints	9
Design of riveted joints	10
Design of welded joints	11
Design of welded joints	12
Design of bolted joints	13
Design of bolted joints	14
Design of cotter joints	15
Design of cotter joints	16
Design of knuckle joints	17
Design of knuckle joints	18
Design of Pipe joints	19
Design of pressure vessels	20
Design of shafts	
Design of keys	21
Design of coupling (Rigid flange coupling)	22
Rigid flange coupling	23
Bushed pin flexible coupling	24
Bushed pin flexible coupling	25
Design of shaft levers	26
Column and strut	
column and studs basics	27
Design of column and strut	28
Forms of thread, Terminology of power screw, Torque requirement	29
Design of screw jack	30
Design of screw jack	31
Belt drive and pulley	
Construction, geometrical relationships, analysis of belt tensions	32
Selection of flat belts from manufacturer's catalogue	33
Pulley for flat and v belt	34
Springs	
Type of springs,Terminology of Helical Springs, stress and deflection equation	35
Design of springs	36

Design of springs	37
Clutches	
Type of clutches, friction clutches,Torque transmitting capacity	38
Multi disc clutches, cone clutches	39
Centrifugal clutches and energy consideration	40
Brakes	
Energy equations, block brakes, band brakes	41
Internal expanding brakes	42

DARBHANGA COLLEGE OF ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

Design of Machine Element

Assignment 1

1. Explain the properties of materials briefly.
2. How will you designate plain carbon steel?
3. Explain manufacturing consideration in design and broadly classify manufacturing processes.
4. What are the steps involved in design of machine element?

DARBHANGA COLLEGE OF ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

Design of Machine Element

Assignment 2

1 The frame of a hydraulic press consisting of two identical steel plates is shown in Fig. 4.28. The maximum force P acting on the frame is 20 kN. The plates are made of steel 45C8 with tensile yield strength of 380 N/mm2. The factor of safety is 2.5. Determine the plate thickness.

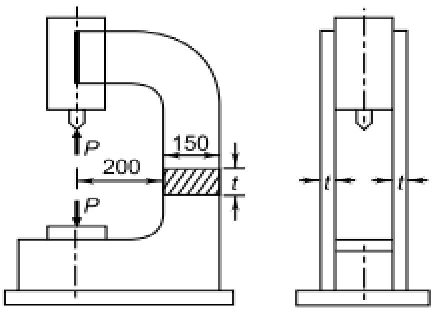


Fig. 1: Frame of hydraulic press

2 The shaft of an overhang crank subjected to a force P of 1 kN is shown in Fig. 2 The shaft is made of plain carbon steel 45C8 and the tensile yield strength is 380 N/mm2. The factor of safety is 2. Determine the diameter of the shaft using the maximum shear stress theory.

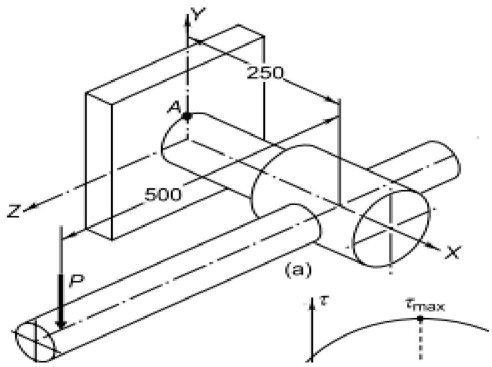


Fig. 2

3 A component machined from a plate made of steel 45C8 (Sut = 630 N/mm2) is shown in Fig. 3. It is subjected to a completely reversed axial force of 50 kN. The expected reliability is 90% and the factor of safety is 2. The size factor is 0.85. Determine the plate thickness t for infi nite life, if the notch sensitivity factor is 0.8.

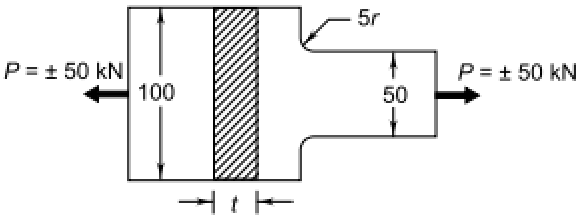


Fig. 4

4 A forged steel bar, 50 mm in diameter, is subjected to a reversed bending stress of 250 N/mm2. The bar is made of steel 40C8 (Sut = 600 N/mm2). Calculate the life of the bar for a reliability of 90%.

DARBHANGA COLLEGE OF ENGINEERING

DEPARTMENT OF MECHANICAL ENGINEERING

Design of Machine Element

Assignment 3

1. A steel plate, 100 mm wide and 10 mm thick, is joined with another steel plate by means of single transverse and double parallel fillet welds, as shown in Fig. 1. The strength of the welded joint should be equal to the strength of the plates to be joined. The permissible tensile and shear stresses for the weld material and the plates are 70 and 50 N/mm2 respectively. Find the length of each parallel fillet weld. Assume the tensile force acting on the plates as static.

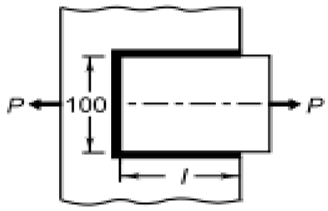


Fig. 1

2. A bracket is attached to a steel channel by means of nine identical rivets as shown in Fig. 2. Determine the diameter of rivets, if the permissible shear stress is 60 N/mm2.

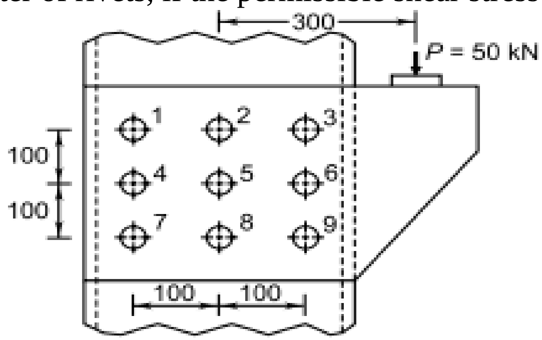


Fig. 2

3. A steel plate subjected to a force of 5 kN and fi xed to a channel by means of three identical bolts is shown in Fig. 3. The bolts are made of plain carbon steel 30C8 (Syt = 400 N/mm2) and the factor of safety is 3. Determine the diameter of the shank.

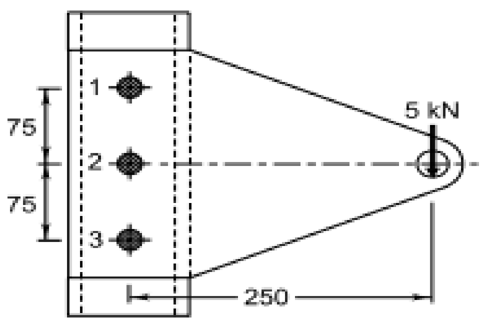
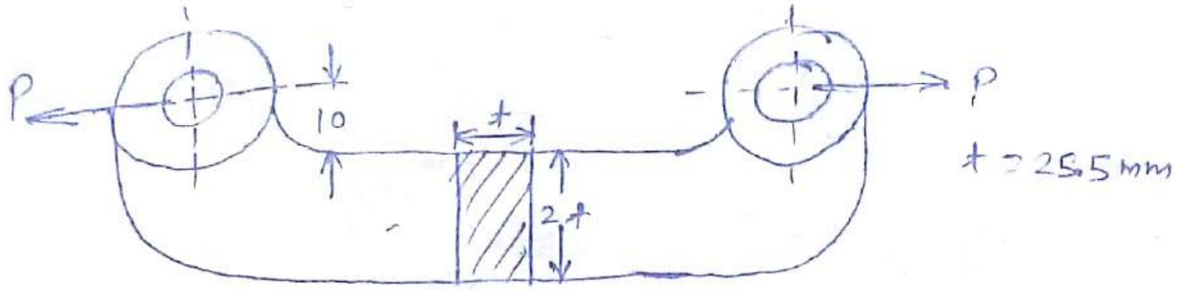
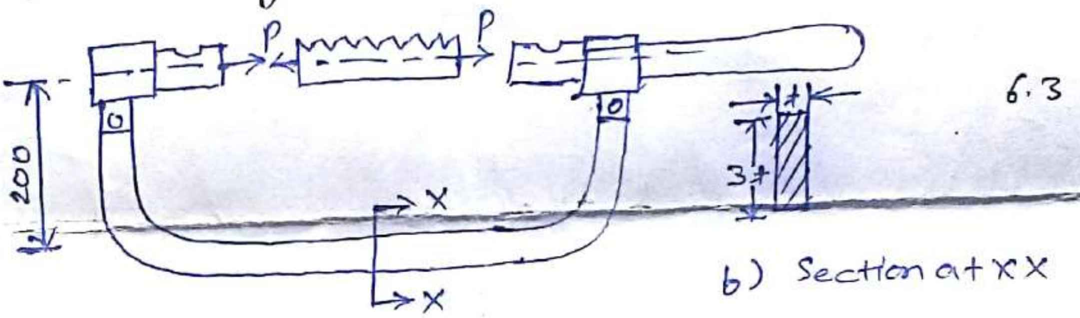


Fig. 3

Q. An offset link subjected to a force of 25KN. It is made of grey cast iron FG300 and fs is 3. Determine the dimensions of the C-S of the link.



Q. The frame of hackshaw is shown in fig. The initial tension P in the blade should be 300N. The frame is made of plain carbon steel 30C8 with a tensile yield strength of 400 N/mm^2 and fs is 2.5. The C-S of the frame is rectangular with a ratio of width to depth as 1/3, as shown in fig. b. Determine the dimensions of the C

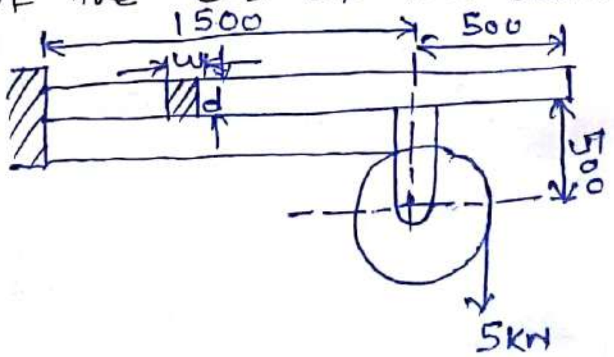


b) Section at XX

Fig(a) frame of Hackshaw.

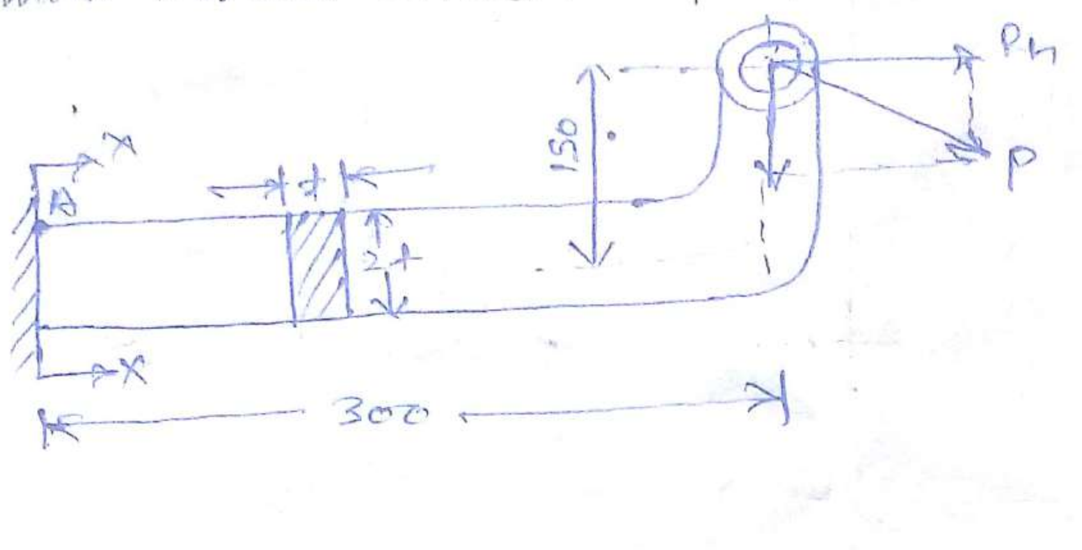
Q. A Cantilever beam of rectangular C-S is used to support a pulley as shown in fig. The tension in the wire rope is 5KN. The beam is made of C.I FG200 and fs is 2.5. The ratio of depth to width of the C-S is 2. Determine the dimensions of the C-S of the beam.

w = 57.r



Q. A wall bracket with a rectangular C-S is shown in f The depth of the C-S is twice of the width. The force P acting on the bracket at 60° to the vertical is 5KN. T Material of the bracket is Grey CI FG200 and fs is 3.5 ... the C-S of the bracket. Assum

Maximum normal stress theory of failure.



Code : 021615

B.Tech 6th Semester Exam., 2018

DESIGN OF MACHINE ELEMENTS

Time : 3 hours *Full Marks* : 70

Instructions :

(i) The marks are indicated in the right-hand margin.

*(ii) There are **NINE** questions in this paper.*

*(iii) Attempt **FIVE** questions in all.*

(iv) Question No. 1 is compulsory.

1. Choose the correct option (any *seven*) : 2×7=14

(a) Steels used for automobile bodies and hoods are

- (i) medium carbon steel
- (ii) mild steel
- (iii) high carbon steel
- (iv) alloy steel

(b) Material used for self-lubricated bearing is

- (i) acetal
- (ii) polyurethane
- (iii) polytetrafluoroethylene (Teflon)
- (iv) Any one of the above

(c) In forged components

- (i) fiber lines are arranged in a predetermined way
- (ii) fiber lines of rolled stock are broken
- (iii) there are no fiber lines
- (iv) fiber lines are scattered

(d) When a circular shaft is subjected to torque, the torsional shear stress is

- (i) maximum at the axis of rotation and zero at the outer surface
- (ii) uniform from axis of rotation to the outer surface
- (iii) zero at the axis of rotation and maximum at the outer surface
- (iv) zero at the axis of rotation and zero at the outer surface and maximum at the mean radius

(e) The thermal stresses are caused due to

- (i) variation in temperature
- (ii) high temperature
- (iii) specific heat
- (iv) latent heat

(f) A stress that varies in sinusoidal manner with respect to time from tensile to compressive (or vice-versa) and with zero mean is called

(i) reversed stress

(ii) fluctuating stress

(iii) repeated stress

(iv) varying stress

(g) In order to find the endurance limit, the rotating beam specimen is subjected to

(i) repeated stresses

(ii) reversed stresses

(iii) fluctuating stresses

(iv) maximum stress

(h) In design of screw jack from buckling considerations, the end conditions are assumed as

(i) both ends are hinged

(ii) both ends are fixed

(iii) one end fixed and other hinged

(iv) one end fixed and other free

(i) In the running condition, the net force acting on the drum of centrifugal clutch is equal to

(i) the centrifugal force on shoe

(ii) the centrifugal force on shoe minus spring force

(iii) the centrifugal force on shoe plus spring force

(iv) the spring force

(j) The maximum shear stress in spring wire is induced at

(i) inner surface of the coil

(ii) outer surface of the coil

(iii) central surface of the coil

(iv) end coils

2. (a) What are the factors to be considered for selection of engineering materials for a machine component? Discuss the important manufacturing considerations in machine design. 7

(b) How will you select direction of fiber lines in forged components? 7

3. The force acting on a bolt consists of two components—an axial pull of 12 kN and a transverse shear force of 6 kN. The bolt is made of steel having $S_{yt} = 310\ \text{N/mm}^2$ and factor of safety is 2·5. Determine the diameter of the bolt using the maximum shear stress theory of failure. 14

4. A rotating bar made of steel having $S_{ut} = 620\ \text{N/mm}^2$ is subjected to a completely reversed bending stress. The corrected endurance limit of the bar is $310\ \text{N/mm}^2$. Calculate the fatigue strength of the bar for a life of 1,00,000 cycles. 14

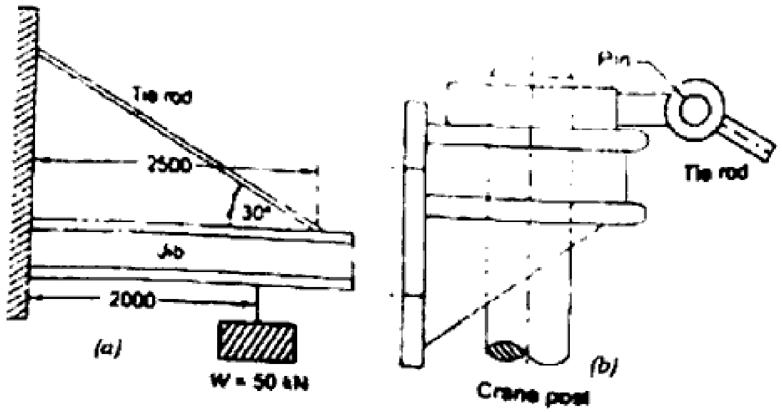
5. A forged steel bar of 55 mm diameter is subjected to a reversed bending stress of $260\ \text{N/mm}^2$. The bar is made of 40C8 steel ($S_{ut} = 610\ \text{N/mm}^2$). Calculate the life of the bar for a reliability of 90%. 14

6. A transmission shaft carries a pulley midway between the two bearings. The bending moment at the pulley varies from 200 N-m to 600 N-m, as the torsional moment in the shaft varies from 70 N-m to 200 N-m. The frequencies of variation of bending and

(Turn Over)

torsional moments of steel FeE 400 ($S_{yt} = 400\ \text{N/mm}^2$ and $S_{ut} = 540\ \text{N/mm}^2$). The corrected endurance limit of the shaft is $210\ \text{N/mm}^2$. Determine the diameter of the shaft using a factor of safety of 2·5. 14

7. The layout of a wall crane and the pin-joint connecting the tie-rod to the crane post is shown in the figures *(a)* and *(b)* respectively. The tension in the tie-rod is maximum, when the load is at a distance of 2 m from the wall. The tie-rod and the pin are made of steel having $S_{yt} = 250\ \text{N/mm}^2$ and factor of safety is 3·0. Determine the diameter of the tie-rod and the pin. 14



8. A gearbox weighing 6 kN is provided with a steel eyebolt for lifting and transporting on the shop floor. The eyebolt is made of 30C8 steel (S_{yt} = 380 N/mm^2) and factor of safety is 5. Determine the nominal diameter of the eyebolt having coarse threads if $d_c = 0 \cdot 8d$ where d_c and d are the core and major diameters respectively. 14

9. A cylindrical pressure vessel with a 0·8 m inner diameter is subjected to an internal steam pressure of 2 MPa. The permissible stresses for cylinder plate and rivets in tension, shear and compression are 80, 60 and 120 N/mm^2 respectively. The efficiency of longitudinal joint can be taken as 80% for calculating the plate thickness. The corrosion allowance is 2 mm. The efficiency of circumferential lap joint should be at least 62%. Design the circumferential lap joint and calculate the thickness of plate, diameter of the rivets, number of rivets and pitch of rivets. 14

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Code : 021615

B.Tech 6th Semester Exam., 2019

DESIGN OF MACHINE ELEMENTS

Time : 3 hours

Full Marks : 70

Instructions :

(i) *All questions carry equal marks.*

(ii) *There are **NINE** questions in this paper.*

(iii) *Attempt **FIVE** questions in all.*

(iv) *Question No. **1** is compulsory.*

(v) *Students are allowed to use design data book.*

1. Choose the correct answer of the following (any *seven*) :

(a) Which of the following parameters can be obtained by tension test of a standard specimen?

(i) Proportional limit

(ii) Yield strength

(iii) Percentage reduction in area

(iv) All of the above

(b) Which of the following is the definition of compliance?

(i) Inverse of rigidity

(ii) Inverse of stiffness

(iii) Proportional to elastic limit

(iv) None of the above

(c) Yield strength is defined as the maximum stress at which a marked increase in elongation occurs without increase in

(i) load

(ii) strength

(iii) toughness

(iv) hardness

(d) Relative density of aluminium is roughly _____ of steel.

(i) one-third

(ii) one-fifth

(iii) one-tenth

(iv) equal

(e) Which of the following are true for aluminium?

(i) Low specific gravity

(ii) Corrosion resistance

(iii) High thermal conductivity

(iv) All of the above

(f) In alloy 4450, 4 represents

(i) silicon

(ii) aluminium

(iii) manganese

(iv) zinc

(g) Ductile cast iron is

(i) nodular cast iron

(ii) spheroidal graphite cast iron

(iii) carbon is present in the form of spherical nodules

(iv) All of the above

(h) Grey cast iron is formed when

(i) carbon content in the alloy exceeds the amount that can be dissolved

(ii) carbon content in the alloy is less than the amount that can be dissolved

(iii) carbon content in the alloy is equal to the amount that can be dissolved in the alloy

(iv) None of the above

(i) Which of the following are true?

(i) Brass is costlier then copper

(ii) Brass has excellent corrosion resistance

(iii) Brass has good machinability

(iv) Brass has poor thermal conductivity

(j) Proof strength is defined as the stress which will produce a permanent extension of how much percentage in the gauge length of the standard test specimen

(i) 0·1

(ii) 0·2

(iii) 0·3

(iv) 0·4

2. Two rods, made of plain carbon steel 40C8 ($S_{yt} = 380\ \text{N/mm}^2$), are to be connected by means of a cotter joint. The diameter of each rod is 50 mm and the cotter is made from a steel plate of 15 mm thickness. Calculate the dimensions of the socket end making following assumptions :

(a) The yield strength in compression is twice of the tensile yield strength

(b) The yield strength in shear is 50% of the tensile yield strength

The factor of safety is 6.

3. A component machined from a plate made of steel 45C8 ($S_{ut} = 630\ \text{N/mm}^2$) is shown in Fig. 1. It is subjected to a completely reversed axial force of 50 kN. The expected reliability is 90% and the factor of safety is 2. The size factor is 0·85. Determine the plate thickness t for infinite life, if the notch sensitivity factor is 0·8 :

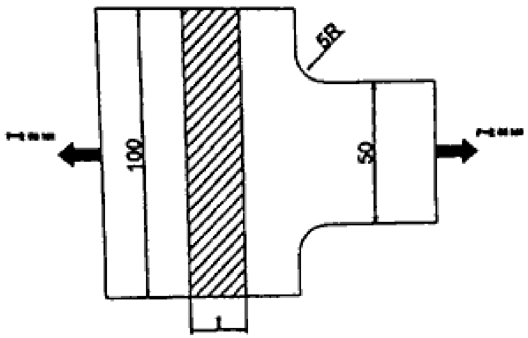


Fig. 1

4. A welded connection, as shown in Fig. 2 is subjected to an eccentric force of 7·5 kN. Determine the size of welds if the permissible shear stress for the weld is $100\ \text{N/mm}^2$. Assume static conditions :

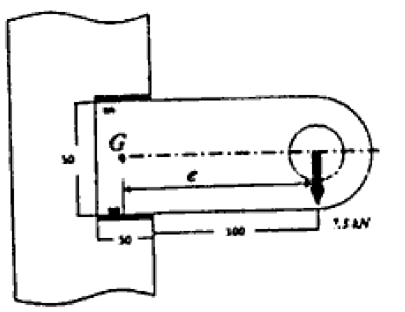


Fig. 2

5. A cylindrical pressure vessel with 1 m inner diameter is subjected to internal steam pressure of 1·5 MPa. The permissible stresses for the cylinder plate and the rivets in tension, shear, and compression are 80, 60 and $120\ \text{N/mm}^2$ respectively. The efficiency of longitudinal joint can be taken as 80% for the purpose of calculating the plate thickness. The corrosion allowance is 2 mm. The efficiency of circumferential lap joint should be at least 62%. Design the circumferential lap joint and calculate—

(a) thickness of the plate;

(b) diameter of the rivets;

(c) number of rivets;

(d) pitch of rivets;

(e) number of rows of rivets;

(f) overlap of the plates.

6. It is required to design a square key for fixing a gear on a shaft of 25 mm diameter. The shaft is transmitting 15 kW power at 720 r.p.m. to the gear. The key is made of steel 50C4 ($S_{yt} = 460\ \text{N/mm}^2$) and the factor of safety is 3. For key material, the yield strength in compression can be assumed to be equal to the yield strength in tension. Determine the dimension of the key.

7. Design a muff coupling to connect two steel shafts transmitting 25 kW power at 360 r.p.m. The shafts and key are made of plain carbon steel 30C8 $(S_{yt} = S_{yc} = 400\ \text{N/mm}^2)$. The sleeve is made of grey cast iron FG200 $(S_{ut} = 200\ \text{N/mm}^2)$. The factor of safety for the shaft and key is 4. For sleeve, the factor of safety is 6 based on ultimate strength.

8. A hard-drawn steel wire extension spring has a wire diameter of 0·9 mm, an outside coil diameter of 6·3 mm, hook radii of $r_1 = 2 \cdot 7$ mm and $r_2 = 2 \cdot 3$ mm, and an initial tension of 5 N. The number of body turns is 12·17. From the given information—

   (a) determine the physical parameters of the spring;

   (b) check the initial preload stress conditions;

   (c) find the factors of safety under a static 23 N load.

9. The following data is given for an open-type V-belt drive :

   Diameter of driving pulley = 150 mm

   Diameter of driven pulley = 300 mm

   Centre distance = 1 m

   Groove angle = 40°

http://www.akubihar.com
http://www.akubihar.com
http://www.akubihar.com

Mass of belt = 0·25 kg/m

Maximum permissible tension = 750 N

Coefficient of friction = 0·2

Plot a graph of maximum tension and power transmitted against the belt velocity. Calculate the maximum power transmitted by the belt and the corresponding belt velocity. Neglect power losses.

★★★

 Code : 021615

B.Tech 6th Semester Exam., 2014

DESIGN OF MACHINE ELEMENTS

Time : 3 hours

Full Marks : 70

Instructions :

(i) *The marks are indicated in the right-hand margin.*

(ii) *There are* **NINE** *questions in this paper.*

(iii) *Attempt* **FIVE** *questions in all.*

(iv) *Question No.* **1** *is compulsory.*

(v) *Use of data books is permitted. Select data, if missing, suitably.*

1. Answer any *seven* of the following as directed : 2×7=14

   (a) Give two examples of bearing pressure and crushing stress in the design consideration of machine elements.

   (b) A hollow shaft and a solid shaft are of equal weight. The hollow shaft has

   (i) lower strength but greater stiffness

   (ii) lower strength and lower stiffness

   (iii) greater strength but lower stiffness

   (iv) greater strength and also greater stiffness

   (Choose the correct option)

(c) If a helical coil spring of stiffness K is cut into two identical half coil springs, the stiffness of each of these half spring will be ——.

(Fill in the blank)

(d) Cast iron is widely used for machine frames. Give two reasons.

(e) Give the composition of $25Cr_4Mo_2$.

(f) The resistance of fatigue of a material is measured by

(i) elastic limit

(ii) proportionate limit

(iii) endurance limit

(iv) ultimate strength limit

(Choose the correct option)

(g) What is the minimum efficiency required for the circumferential boiler joint?

(h) Why are multiple threaded screws not recommended in screw jack?

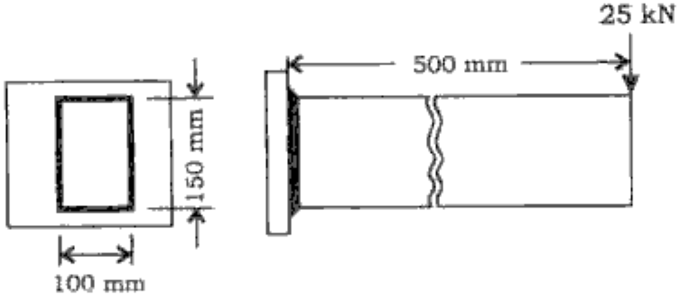
(i) Suggest suitable coupling for shafts with parallel misalignment.

(j) Name the three stresses induced in belt due to power transmission.

2. Design a cotter joint, made of 30C8 steel, to support a load of 50 kN which is subjected to slow reversals of direction. 14

3. Determine the main dimensions of the longitudinal joints of a boiler whose inner diameter is 1·7 m and pressure of steam is 20 bar. The allowable tensile, crushing and shear stresses of mild steel rivet are 80 N/mm^2, 120 N/mm^2 and 65 N/mm^2 respectively. Assume quadruple rivetted, zig-zag butt joint with unequal cover plates. 14

4. Determine the size of the welds to support by means of fillet welds of a beam of rectangular cross-section as shown in the figure below if the permissible shear stress in the weld is limited to 75 N/mm^2. 14



5. A mild steel shaft has to transmit 70 kW at 240 r.p.m. The allowable shear stress in the shaft material is limited to 45 MPa and the angle of twist is not to exceed 1° in a length of 20 times the shaft diameter. Determine the shaft diameter and design a cast iron flange coupling of protected type for the shaft. The shear stress in the coupling bolts is to be limited to 30 MPa. 14

6. Design a screw jack for lifting a load of 20 kN through a distance of 200 mm. 14

7. A safety valve of 60 mm diameter is to blow off at a pressure of 12 bar. It is held on its seat by a close-coiled helical spring. The maximum lift of the valve is 10 mm. Determine main dimensions of a compression spring of spring index 5. Take initial compression of the spring as 35 mm. The maximum shear stress in the material of the spring wire is to be limited to 500 N/mm^2. [Take C = 82 GPa] 14

8. A crossed belt drive is to transmit 10 kW at 1200 r.p.m. of the smaller pulley which is 250 mm in diameter. The velocity ratio is 2 and centre distance is 1·2 m. It is desired to use a 6 mm thick leather belt with coefficient of friction equal to 0·25. If the permissible stress for the belt material is 2 N/mm^2, determine the width of the belt. [Take the mass density of the belt material as 1000 kg/m^3] 14

9. A single-disc clutch is required to resist a maximum torque 500 N-m. The outer radius of the friction lining is 30% more than the inner radius. The permissible intensity of pressure between the contact surfaces is 0·08 N/mm^2. The coefficient of friction is 0·25. Eight helical compression springs are used to provide axial force necessary to engage the clutch. If the stiffness of each spring is 36 N/mm, determine the size of the friction lining and initial compression in the spring. 14

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DARBHANGA COLLEGE OF ENGINEERIG, DARBHANGA

SESSION- (2019- 2020)

MECHANICAL ENGINEERING (6th SEM)

DESIGN OF MACHINE ELEMENTS

(021615)

Question Bank

Unit- 1

Objective Questions:

1.Which of the following material has the maximum ductility?

a) Mild steel (b) Copper (c) Zinc (d) Aluminium

Ans: (a)

2.According to Indian standard specifications, a grey cast iron designated by 'FG 200' means that the
(a) carbon content is 2%

(b) maximum compressive strength is 200 N/mm2

(c) minimum tensile strength is 200 N/mm2

(d) maximum shear strength is 200 N/mm2

Ans.: (c)

3.According to Indian standard specifications, a plain carbon steel designated by 40C8 means that

(a) carbon content is 0.04 per cent and manganese is 0.08 per cent

(b) carbon content is 0.4 per cent and manganese is 0.8 per cent

(c) carbon content is 0.35 to 0.45 per cent and manganese is 0.60 to 0.90 per cent

(d) carbon content is 0.60 to 0.80 per cent and manganese is 0.8 to 1.2 per cent

Ans.: (c)

4.The material commonly used for machine tool bodies is

(a) mild steel (b) aluminium (c) brass (d) cast iron

Ans.: (d)

5.The material commonly used for crane hooks is

(a) cast iron (b) wrought iron (c) mild steel (d) aluminium

Ans.: (b)

6.The steel widely used for motor car crankshafts is

(a) nickel steel (b) chrome steel (c) nickel-chrome steel (d) silicon steel

Ans.: (b)

7.The castings produced by forcing molten metal under pressure into a permanent metal mould is known as

(a) permanent mould casting (b) slush casting
(c) die casting (d) centrifugal casting

Ans.: (c)

8.The metal is subjected to mechanical working for

(a) refining grain size (b) reducing original block into desired shape
(c) controlling the direction of flow lines (d) all of these

Ans.: (d)

9. The temperature at which the new grains are formed in the metal is called

(a) lower critical temperature (b) upper critical temperature
(c) eutectic temperature (d) recrystallisation temperature

Ans.: (d)

10. During hot working of metals

(a) porosity of the metal is largely eliminated

(b) grain structure of the metal is refined

(c) mechanical properties are improved due to refinement of grains

(d) all of the above

Ans.: (d)

11. The parts of circular cross-section which are symmetrical about the axis of rotation are made by

(a) hot forging (b) hot spinning (c) hot extrusion (d) hot drawing

Ans.: (b)

12. The process extensively used for making bolts and nuts is

(a) hot piercing (b) extrusion (c) cold peening (d) cold heading

Ans.: (d)

13. Factor of safety for fatigue loading is the ratio of

(a) elastic limit to the working stress

(b) Young's modulus to the ultimate tensile strength

(c) endurance limit to the working stress

(d) elastic limit to the yield point

Ans.: (c)

Subjective Questions:

1. Enumerate the various manufacturing methods of machine parts which a designer should know.
2. What do you understand by ‘hot working’ and ‘cold working’ processes? Explain with examples.
3. Give the composition of 35 Mn 2 Mo 45 steel. List its main uses.
4. What are the factors to be considered for the selection of materials for the design of machine elements? Discuss.
5. What are the steps involved in in design of a machine elements?
6. Distinguish between design synthesis and design analysis.
7. What is standardization?

Unit- 2

Objective Questions:

1. When a machine member is subjected to torsion, the torsional shear stress set up in the member is

(a) zero at both the centroidal axis and outer surface of the member

(b) Maximum at both the centroidal axis and outer surface of the member

(c) zero at the centroidal axis and maximum at the outer surface of the member

(d) none of the above

2. The stress which vary from a minimum value to a maximum value of the same nature (i.e. tensile or compressive) is called

(a) repeated stress
(b) yield stress
(c) fluctuating stress
(d) alternating stress

3. The endurance or fatigue limit is defined as the maximum value of the stress which a polished standard specimen can withstand without failure, for infinite number of cycles, when subjected to
(a) static load
(b) dynamic load

(c) static as well as dynamic load
(d) completely reversed load

4. Failure of a material is called fatigue when it fails

(a) at the elastic limit
(b) below the elastic limit

(c) at the yield point
(d) below the yield point

5. The resistance to fatigue of a material is measured by

(a) elastic limit
(b) Young's modulus

(c) ultimate tensile strength
(d) endurance limit

6. The yield point in static loading is .... as compared to fatigue loading.

(a) higher (b) lower (c) same

7. If the size of a standard specimen for a fatigue testing machine is increased, the endurance limit for the material will

(a) have same value as that of standard specimen

(b) increase

(c) decrease

8. The residual compressive stress by way of surface treatment of a machine member subjected to fatigue loading

(a) improves the fatigue life
(b) deteriorates the fatigue life

(c) does not affect the fatigue life
(d) immediately fractures the specimen

9. Stress concentration factor is defined as the ratio of

(a) maximum stress to the endurance limit

(b) nominal stress to the endurance limit

(c) maximum stress to the nominal stress

(d) nominal stress to the maximum stress

10. In static loading, stress concentration is more serious in

(a) brittle materials (b) ductile materials

(c) brittle as well as ductile materials (d) elastic materials

11. In cyclic loading, stress concentration is more serious in

(a) brittle materials (b) ductile materials

(c) brittle as well as ductile materials (d) elastic materials

Short- Answer Questions:

1. What is fluctuating stress? Draw a stress–time curve for fluctuating stress.

2. What are the methods of reducing stress concentration?

3. What is the difference between failure due to static load and fatigue failure?

4. What are the factors that affect endurance limit of a machine part?

5. What is modifying factor to account for stress concentration?

Subjective Questions:

1. A rectangular plate, 15 mm thick, made of a brittle material is shown in Fig. 5.58. Calculate the stresses at each of three holes of 3, 5 and 10 mm diameter.

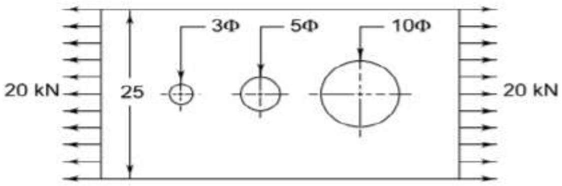

Fig. 1

2. A solid circular shaft, 15 mm in diameter, is subjected to torsional shear stress, which varies from 0 to 35 N/mm2 and at the same time, is subjected to an axial stress that varies from –15 to +30 N/mm2. The frequency of variation of these stresses is equal to the shaft speed. The shaft is made of steel FeE 400 (Sut = 540 N/mm2 and Syt = 400 N/mm2) and the corrected endurance limit of the shaft is 200 N/mm2. Deter mine the factor of safety.

3. A 25 mm diameter shaft is made of forged steel 30C8 (S_{ut} = 600 N/mm^2). There is astep in the shaft and the theoretical stress concentration factor at the step is 2.1. The notch sensitivity factor is 0.84. Determine the endurance limit of the shaft if it is subjected to a reversed bending moment.

4. A rotating bar made of steel having Sut = 620 N/mm^2 is subjected to a completely reversed bending stress. The corrected endurance limit of the bar is 310 N/mm^2 . Calculate the fatigue strength of the bar for a life of 1, 00,000 cycles.

<u>Unit- 3</u>

Objective Questions:

1. A cotter joint is used to transmit

(a) axial tensile load only (b) axial compressive load only

(c) combined axial and twisting loads (d) axial tensile or compressive loads

2. The taper on cotter varies from

(a) 1 in 15 to 1 in 10 (b) 1 in 24 to 1 in 20

(c) 1 in 32 to 1 in 24 (d) 1 in 48 to 1 in 24

3. In a steam engine, the piston rod is usually connected to the crosshead by means of a

(a) knuckle joint (b) universal joint

(c) flange coupling (d) cotter joint

4. In a steam engine, the valve rod is connected to an eccentric by means of a

(a) knuckle joint (b) universal joint

(c) flange coupling (d) cotter joint

5.A rivet is specified by

(a) shank diameter (b) length of rivet

(c) type of head (d) length of tail

6.The rivet head used for boiler plate riveting is usually

(a) snap head (b) pan head

(c) counter sunk head (d) conical head

7. A line joining the centres of rivets and parallel to the edge of the plate is known as

(a) back pitch (b) marginal pitch

(c) gauge line (d) pitch line

8. If the tearing efficiency of a riveted joint is 50%, then ratio of diameter of rivet hole to the pitch of rivets is

(a) 0.20 (b) 0.30 (c) 0.50 (d) 0.60

9. The longitudinal joint in boilers is used to get the required

(a) length of boiler (b) diameter of boiler

(c) length and diameter of boiler (d) efficiency of boiler

10. For longitudinal joint in boilers, the type of joint used is

(a) lap joint with one ring overlapping the other

(b) butt joint with single cover plate

(c) butt joint with double cover plates

(d) any one of these

11. The washer is generally specified by its

(a) outer diameter (b) hole diameter

(c) thickness (d) mean diameter

12. A locking device extensively used in automobile industry is a

(a) jam nut (b) castle nut (c) screw nut (d) ring nut

13.A bolt of uniform strength can be developed by

(a) keeping the core diameter of threads equal to the diameter of unthreaded portion of the bolt

(b) keeping the core diameter of threads smaller than the diameter of unthreaded portion of the bolt

(c) keeping the nominal diameter of threads equal to the diameter of unthreaded portion of bolt

(d) none of the above

Subjective Questions:

1. It is required to design a cotter joint to connect two steel rods of equal diameter. Each rod is subjected to an axial tensile force of 50 kN. Design the joint and specify its main dimensions.

2. 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 and the factor of safety is 4. The width of the cotter is five times the thickness. Calculate:

(i) width and thickness of the cotter on the basis of shear failure; and

(ii) width and thickness of cotter on the basis of bending failure.

3. Design a knuckle joint which is used to connect two rods which are required to withstand a tensile load of 100 kN. The rods and pin are made of plain carbon steel 30C8 (S_{yt} = 400 N/mm^2) and assume suitable factor of safety.

4.A welded connection of steel plates is shown in fig. 1. It is subjected to an eccentric force of 50 kN. Determine the size of weld, if the permissible shear stress in the weld is not to exceed 70 N/mm^2.

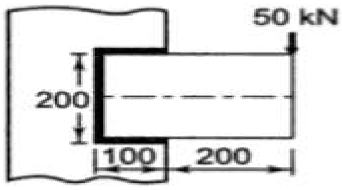

Fig. 1

5.A bracket is attached to a horizontal column by means of three identical rivets as shown in fig. 2. The maximum permissible shear stress for the rivet is 60 N/mm^2.

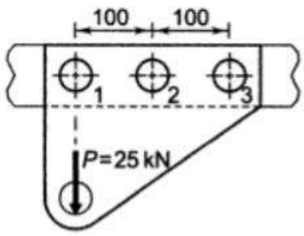

Fig. 2

6. A pressure vessel of the boiler consists of cylindrical shell of 0.8 m inner diameter. It is subjected to internal steam pressure of 2 MPa. Triple-riveted double-strap longitudinal butt joint is used to make the shell. The straps are of unequal width. The pitch of the rivets in outer row is twice of the pitch of rivets in middle and inner rows. A zig-zag pattern is used for arrangement of rivets. The effi ciency of the joint should be at least 80%. The corrosion allowance is 2 mm. The permissible stresses for rivets and shell in tension, shear and compression are 80, 60 and 120 N/mm2 respectively.

Calculate:

(i)thickness of the shell;

(ii) diameter of the rivets;

(iii) pitch of the rivets in outer row;

(iv) distance between outer and middle rows;

(v) distance between middle and inner rows;

(vi) thickness of inner strap;

(vii) thickness of outer strap; and

(viii) effi ciency of the joint.

7.A gearbox weighing 7.5 kN is provided with a steel eye bolt for lifting and transporting on the shop-floor. The eyebolt is made of plain carbon steel 30C8 (Syt = 400 N/mm2) and the factor of safety is 5. Determine the nominal diameter of the eye bolt having coarse threads

if, dc = 0.8d

where dc and d are core and major diameters respectively.

Unit-4

Objective Questions:

1.The usual proportion for the width of key is

(a) d/8 (b) d/6 (c) d/4 (d) d/2

where d = Diameter of shaft.

2. A feather key is generally

(a) loose in shaft and tight in hub

(b) tight in shaft and loose in hub

(c) tight in both shaft and hub

(d) loose in both shaft and hub.

3. The type of stresses developed in the key is/are

(a) shear stress alone

(b) bearing stress alone

(c) both shear and bearing stresses

(d) shearing, bearing and bending stresses

4. For a square key made of mild steel, the shear and crushing strengths are related as

(a) shear strength = crushing strength

(b) shear strength > crushing strength

(c) shear strength < crushing strength

(d) none of the above

5. A keyway lowers

(a) the strength of the shaft

(b) the rigidity of the shaft

(c) both the strength and rigidity of the shaft

(d) the ductility of the material of the shaft

6. The sleeve or muff coupling is designed as a

(a) thin cylinder (b) thick cylinder

(c) solid shaft (d) hollow shaft

7. Two shafts A and B are made of the same material. The diameter of the shaft A is twice as that of shaft B. The power transmitted by the shaft A will be .... of shaft B.

(a) twice (b) four times (c) eight times (d) sixteen times

8. Which of the following loading is considered for the design of axles ?

(a) Bending moment only

(b) Twisting moment only

(c) Combined bending moment and torsion

(d) Combined action of bending moment, twisting moment and axial thrust

9. A connecting rod is designed as a

(a) long column (b) short column (c) strut (d) any one of these

10. The most suitable section for the connecting rod is

(a) L-section (b) T-section (c) I-section (d) C-section

11. Which of the following screw thread is adopted for power transmission in either direction?

(a) Acme threads (b) Square threads

(c) Buttress threads (d) Multiple threads

12. Multiple threads are used to secure

(a) low efficiency (b) high efficiency

(c) high load lifting capacity (d) high mechanical advantage

13. The material suitable for the belts used in agricultural equipments is

(a) cotton (b) rubber

(c) leather (d) balata gum

14. The power transmitted by means of a belt depends upon

(a) velocity of the belt

(b) tension under which the belt is placed on the pulleys

(c) arc of contact between the belt and the smaller pulley

(d) all of the above

15. When the speed of belt increases,

(a) the coefficient of friction between the belt and pulley increases

(b) the coefficient of friction between the belt and pulley decreases

(c) the power transmitted will decrease

(d) the power transmitted will increase

Subjective Questions:

1.It is required to design a square key for fi xing a pulley on the shaft, which is 50 mm in diameter. The pulley transmits 10 kW power at 200 rpm to the shaft. The key is made of steel 45C8 (Syt = Syc = 380 N/mm2) and the factor of safety is 3. Determine the dimensions of the key.

Assume (Ssy = 0.577Syt)

2.A rigid coupling is used to connect a 45 kW, 1440 rpm electric motor to a centrifugal pump. The starting torque of the motor is 225% of the rated torque. There are 8 bolts and their pitch circle diameter is 150 mm. The bolts are made of steel 45C8 (Syt = 380 N/mm2) and the factor of safety is 2.5. Determine the diameter of the bolts.

Assume (Ssy = 0.577Syt)

Assume that the bolts are fi nger tight in reamed and ground holes.

3.A bushed pin type fl exible coupling is used to connect two shafts and transmit 5 kW power at 720 rpm Shafts, keys and pins are made of commercial steel (Syt = Syc = 240 N/mm2) and the factor of safety is 3. The fl anges are made of grey cast iron FG 200 (Sut = 200 N/mm2) and the factor of safety is 6. Assume, Ssy = 0.5Syt and Ssu = 0.5Sut There are 4 pins. The pitch circle diameter of the pins is four times the shaft diameter. The permissible shear stress for the pins is 35 N/mm2. The permissible bearing pressure for the rubber bushes is 1 N/mm2. The keys have a square cross-section.

Calculate: (i) diameter of the shafts;

(ii) dimensions of the key;

(iii) diameter of the pins; and

(iv) outer diameter and effective length of the bushes.

4. The layout of a crossed leather belt drive transmitting 7.5 kW is shown in Fig. 13.30. The mass of the belt is 0.55 kg per metre length and the coeffi cient of friction is 0.30. Calculate (i) the belt tensions on the tight and loose sides, and (ii) the length of the belt.

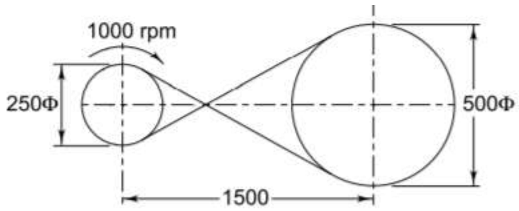

Fig.

5. A V-belt drive is required for a 15-kW, 1440 rpm electric motor, which drives a centrifugal pump running at 360 rpm for a service of 24 hours per day. From space considerations, the centre distance should be approximately 1 m.

Determine

(i) belt specifications;

(ii) number of belts;

(iii) correct centre distance; and

(iv) pulley diameters.

Unit- 5

Objective Questions:

1.A spring used to absorb shocks and vibrations is

(a) closely-coiled helical spring (b) open-coiled helical spring

(c) conical spring (d) torsion spring

2. The spring mostly used in gramophones is

(a) helical spring (b) conical spring

(c) laminated spring (d) flat spiral spring

3. Which of the following spring is used in a mechanical wrist watch?

(a) Helical compression spring (b) Spiral spring

(c) Torsion spring (d) Bellevile spring

4. When a helical compression spring is subjected to an axial compressive load, the stress induced in the wire is

(a) tensile stress (b) compressive stress

(c) shear stress (d) bending stress

5.When two concentric coil springs made of the same material, having same length and compressed equally by an axial load, the load shared by the two springs will be .... to the square of the diameters of the wires of the two springs.

(a) directly proportional (b) inversely proportional (c) equal to

6. A leaf spring in automobiles is used

(a) to apply forces (b) to measure forces

(c) to absorb shocks (d) to store strain energy

7. In leaf springs, the longest leaf is known as

(a) lower leaf (b) master leaf (c) upper leaf (d) none of these

8. A jaw clutch is essentially a

(a) positive action clutch (b) cone clutch

(c) friction clutch (d) disc clutch

9. The cone clutches have become obsolete because of

(a) small cone angles (b) exposure to dirt and dust

(c) difficulty in disengaging (d) all of these

10. A brake commonly used in railway trains is

(a) shoe brake (b) band brake

(c) band and block brake (d) internal expanding brake

11. A brake commonly used in motor cars is

(a) shoe brake (b) band brake

(c) band and block brake (d) internal expanding brake

12. The material used for brake lining should have .... coefficient of friction.

(a) low (b) high

13. When the frictional force helps to apply the brake, then the brake is said to be

(a) self-energizing brake (b) self-locking brake

(c) universal brake (d) none ofthese

14. For a band brake, the width of the band for a drum diameter greater than 1 m, should not exceed (a) 150 mm (b) 200 mm (c) 250 mm (d) 300 mm

Subjective Questions:

1.An automobile vehicle weighing 13.5 kN is moving on a level road at a speed of 95 km/h. When the brakes are applied, it is subjected to a uniform deceleration of 6 m/s2. There are brakes on all four wheels. The tyre diameter is 750 mm. The kinetic energy of the rotating parts is 10% of the kinetic energy of the moving vehicle. The mass of each brake drum assembly is 10 kg 498 Design of Machine Elements and the specifi c heat capacity is 460 J/kg°C.

Calculate

(i)the braking time;

(ii)the braking distance;

(iii)the total energy absorbed by each brake;

(iv) the torque capacity of each brake; and

(v) the temperature rise of brake drum assembly

2. State different types of brakes and give at least one practical application of each.

3. A multi-disk clutch consists of two steel disks with one bronze disk. The inner and outer diameters of the contacting surfaces are 200 and 250 mm respectively. The coeffi cient of friction is 0.1 and the maximum pressure between the contacting surfaces is limited to 0.4 N/mm2. Assuming uniform wear theory, calculate the required force to engage the clutch and the power transmitting capacity at 720 rpm

4. A centrifugal clutch, transmitting 18.5 kW at 720 rpm, consists of four shoes. The clutch is to be engaged at 75% of the running speed. The inner radius of the drum is 165 mm, while the radius of the centre of gravity of each shoe, during engaged position, is 140 mm. The coeffi cient of friction is 0.25. Calculate the mass of each shoe.

5. It is required to design a helical compression spring subjected to a force of 500 N. The defl ection of the spring corresponding to this force is approximately 20 mm. The spring index should be 6. The spring is made of cold-drawn steel wire with ultimate tensile strength of 1000 N/mm2. The permissible shear stress for the spring wire can be taken as 50% of the ultimate tensile strength (G = 81 370 N/mm2). Design the spring and

calculate:

(i) wire diameter;

(ii) mean coil diameter;

(iii) number of active coils;

(iv) total number of coils;

(v) free length of the spring; and

(vi) pitch of the coils.

Assume a gap of 1 mm between adjacent coils under maximum load condition. The spring has square and ground ends.

6. A helical compression spring is required to defl ect through approximately 25 mm when the external force acting on it varies from 500 to 1000 N. The spring index is 8. The spring has square and ground ends. There should be a gap of 2 mm between adjacent coils when the spring is subjected to the maximum force of 1000 N. The spring is made of cold-drawn steel wire with ultimate tensile strength of 1000 N/mm2 and permissible shear stress in the spring wire should be 50% of the ultimate tensile strength (G = 81 370 N/mm2). Design the spring and

calculate:

(i) wire diameter;

(ii) mean coil diameter;

(iii) number of active coils;

(iv) total number of coils;

(v) solid length;

(vi) free length;

(vii) required spring rate; and

(viii) actual spring rate.

Machine Design : M/C designed is defined as the use of scientific principles technical information and imagination in the description of M/C or a mechanical system to perform specific functions with maximum economy and efficiency.

This definition of M/C design contains the following important features:

(i) A designer uses principles of basic and engineering sciences such as physics, mathematics, statics and dynamics - - -. Some of the examples of these principles are..

— — —

— — —

(ii) The designer has techical information of the basic elements of a machine. - - - A m/c is a combination of these basic elements.

(iii) The designer uses his skill and imagination to produce a configuration, which is a combination of these basic elements.

(iv) The final outcome of the design process consists of the description of the m/c. The description is in the form of drawings of assembly and indivisual components.

(v) A design is created to satisfy a recognised need of customer. The need may be to perform a sp. function with maximum economy & efficiency

Design < Analysis / Synthesis

Analysis! — The designer assumes a particular mechanism, a particular material and mode of failure for the component.

⇒ With the help of these information he determines the dimensions of the product.

Synthesis! — It is defined as the process of creating ~~an~~ or selecting configurations, materials, shapes and dimensions for a product.

⇒ It is a decision making process with the main objective of optimisation.

⇒ ~~Synthesis does not permit~~ assumptions.

In design synthesis, a designer has to fix the objectives. The objective can be minimum cost, minimum weight or volume, maximum reliability or maximum life.

→ The second step is mathematical formulation of these objectives and requirements.

and the final step is mathematical analysis for optimisation and interpretation of the result.

Pre-requisite of Machine Design:
→ Idea gathered from Som & Mechanics
Design Phylosophy
— Design is essentially a decision making process and for every problem we need to design a solution. So design is to formulate a plan to satisfy a particular need and to create sth. with a physical reality.
When we face a problem we will try to solve our problem by existing methodology or new ideas.
Let us design a chair:
Factors need to be considered:
1. Purpose for which the chair is designed.
2. Whether the chair is to be designed for an adult person or a child.
3. Material for the chair, strength and cost need to be ~~cost~~ determined. (wrought Iron chair)
4. Aesthetics and ergonomics of the designed chair.
Almost everyone is involved in design in one way or other, in our daily lives because problems are faced & they need to be solved.
cooking — design new recipe
Basic concept of Design.
→ Decision Making: At every stage of design.
→ Consideration of different factors.
→ To draw certain conclusions leading to an optim design
→ Market survey to read People's mind.
→ Study of existing norms.
Design Disciplines

BIS System of designation of Steel:-

Steels are designated by a group of letters or numbers indicating any one of the following three properties.

(i) Tensile strength (ii) carbon content

(iii) Composition of alloying elements. $\boxed{(YS)_{min} = 0.55\% \text{ of } (TS)_{min}}$

Steels on the basis of tensile strength

Fe 360 → minimum tensile strength = 360 N/mm²

FeE 250 → minimum yield strength = 250 N/mm²

The designation of plane carbon steel.

55 C4 → % of carbon — 0.50 to 0.60 %.
% of manganese — 0.4 %.

The designation of unalloyed free cutting steels consists of following quantities

(i) A figure indicating 100 times the average percentage of carbon;

(ii) A letter C;

(iii) A figure indicating 10 times the av. percentage of manganese

(iv) A symbol 'S', 'Se', 'Te' or 'Pb' depending upon the element that is present and which makes the steel free cutting; and

Tellurium

(v) A figure indicating 100 times the av. percentage of the above element that makes the steel free cutting.

As an example, 25C12S14 → 0.25 % C
1.2 % Mn
0.14 % S

The designation of alloy steel: (10%)

(i) A figure indicating 100 times the average percentage of carbon

(ii) Chemical symbol of alloying elements each followed by its percentage content multiplied by a factor

Elements	Multiplying factor
Cr, Co, Ni, Mn, Si and W	4
Al, Be, V, Pb, Cu, Nb, Ti, Ta, Zr and Mo	10
P, S, N	100

the content of Mn is equal to or greater than 1%. The chemical symbols and their figures are arranged in descending order of their percentage content.

25Cr4Mo2 → 0.25% C, 1% Cr, 0.2% Mo

40Ni8Cr8V2 → 0.4% C, 2% Ni, 2% Cr, 0.2% V

→ Consider an alloy steel with the following composition:

2.6 round off

Carbon = 0.12 − 0.18%   15Cr3

Si = 0.15 − 0.35%

Mn = 0.4 − 0.6% } negligible

Cr = 0.5 − 0.8%

C = 0.12 − 0.20%

Si = 0.15 − 0.35%

Mn − 0.6 − 1.00%

Ni − 0.60 − 1.00%

Cr = 0.40 − 0.80%

} 16Ni3Cr2

The term 'high alloy steels' is used for alloy steels containing more than 10% of alloying element

(i) X

(ii) 100 times C

(iii) chemical symbol for alloying elements followed by the figure for its average percentage content

(iv) chemical symbol to indicate a specially added element to attain desired properties, if any.

X15 Cr25 Ni12

Mechanical Properties of Engineering Materials.

1. Strength: Strength is defined as ability of material to resist, without fracture, external force causing various type of stresses.

2. Elasticity:— Elasticity is defined as ability of material to regain its original shape and size after the deformation, when the external force is removed.

3. Plasticity:— Plasticity is defined as ability of material to retain the deformation produced under the load on permanent basis.

... hood and doors for automobile to be stamped.

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Stiffness or rigidity: It is defined as the ability of the material to resist deformation under the action of an external load. Modulus of elasticity is the measure of stiffness.

Resilience is It is defined as the ability of material to absorb energy when deformed elastically and to release energy when unloaded.

Toughness :- Toughness is defined as the ability of the material to absorb energy before fracture takes place. Tough materials has the ability to bend, twist or stretch before failure takes place. All the structural steels are tough materials.
Toughness decreases as the temperature increases.

Malleability:- It is defined as the ability of material to deform to a greater extent before the sign of crack, when it is subjected to compressive force. Ability to be hammered out in thin sections.
Malleable metals can be rolled, forged or extruded because these processes involve shaping under compressive forces. Low carbon steels, copper and aluminium are malleable metals.
Malleability increases with temperature (in general)

Ductility !- Ductility is defined as the ability of a material to deform to a greater extent before the sign to crack when it is subjected to tensile force. Ductile materials are those materials which deform plastically to a greater extent prior to fracture in a tension test.
All ductile material are also malleable but the converge is not also true.
→ The ductility of metal decreases as the temperature increases because metals become weak at increasing temperature.
Some metals are soft but weak in tension, therefore tend to tear apart under tension.

Brittleness: Brittleness is the property of a material which shows negligible plastic deformation prior to fracture. A tensile strain of 5% at fracture in a tension test is considered as the dividing line between ductile and brittle materials

Hardness :- Hardness is defined as the resistance of the material to penetration or permanent deformation. It usually indicates resistance to abrasion, Scratching, cutting ...

Stress-Strain Relationship :—

When a mechanical component is subjected to an external static force, a resisting force is set-up within the component. "The internal resisting force against deformation per unit area is called stress.

Tensile Stress   compressive stress.

· A tension rod subjected to an external force P is shown in the fig.

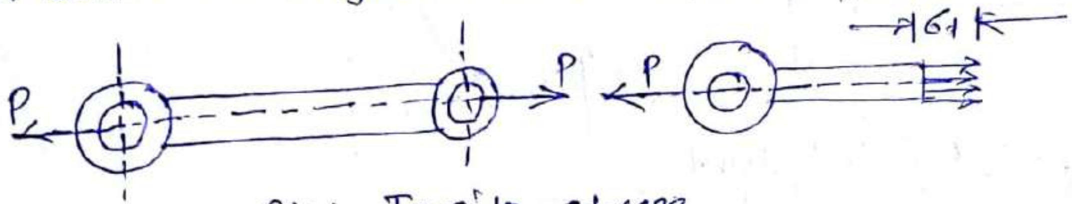


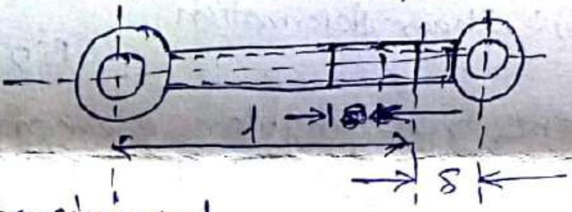
Fig: Tensile stress.

$$\boxed{\sigma_t = \frac{P}{A}} \quad \text{—①}$$

where σ_t = Tensile stress (N/mm²)
P = external force (N)
A = cross-section area (mm

The strain is defined as deformation per unit length.

$$\varepsilon = \frac{\delta}{l} \quad \text{—②}$$



where ε = Strain (mm/mm)
δ = elongation in the tension rod.

According to Hook's law, the stress is directly proportional to strain within elastic limit.

There fore, $\sigma_t \propto \varepsilon$

$$\boxed{\sigma_t = E\varepsilon} \quad \text{—③}$$

For carbon steel $E = 210000$ N/mm2
For grey cast iron $E = 100\,000$ N/mm2

$$\frac{P}{A} = E\frac{\delta}{l}$$

$$\boxed{\delta = \frac{Pl}{AE}}$$

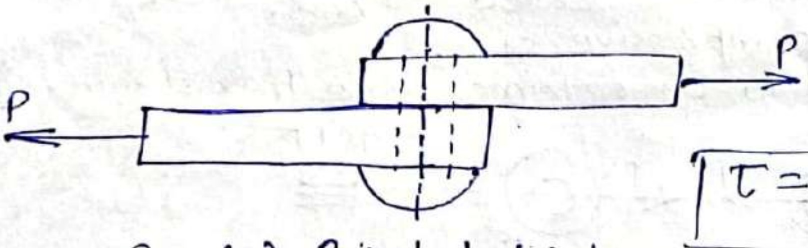
· Symillarily for compressive Stress.

Assumptions are made in the analysis of Stress & Strain

(i) The material is homogeneous.
(ii) The load is gradually applied.
(iii) The line of action of forces passes through the geometric axis of the C-S.
(iv) The C-S is uniform.
(v) There is no stress concentration.

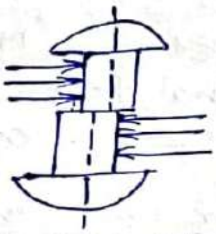
Shear Stress and Shear Strain

When the external force acting on a component tends to slide the adjacent planes w.r.t each other, The resulting stresses on these planes are called direct shear stresses.



$$\boxed{\tau = \frac{P}{A}}$$

fig (a) Riveted Joint.



Fig(b): Shear deformation

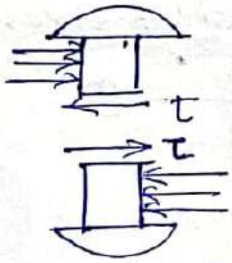
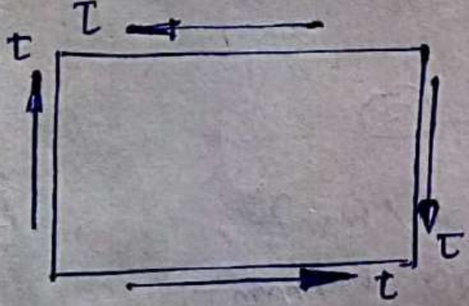
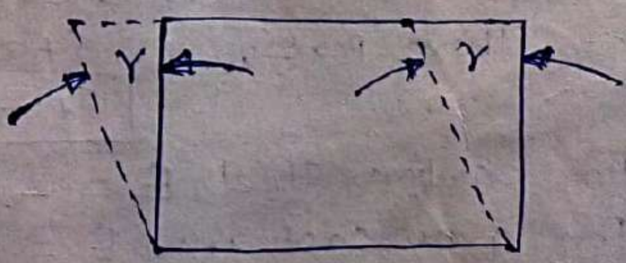


fig (c): Shear Stress.

A plane rectangular element, cut from the component and subjected to shear force. is shown in fig. Shear stresses causes distortion in the original right angles.





The shear strain γ is defined as the change in right angle of a shear element. within the elastic limit, the stress - strain relationship is given by,

$$\boxed{\tau = G\gamma}$$

where

γ = shear strain (radians)

For carbon steel $G = 80$ GPa

For grey cast iron $G = 40$ GPa.

Bending Stresses :— A st. beam subjected to a bending moment M_b is shown is fig. (4)

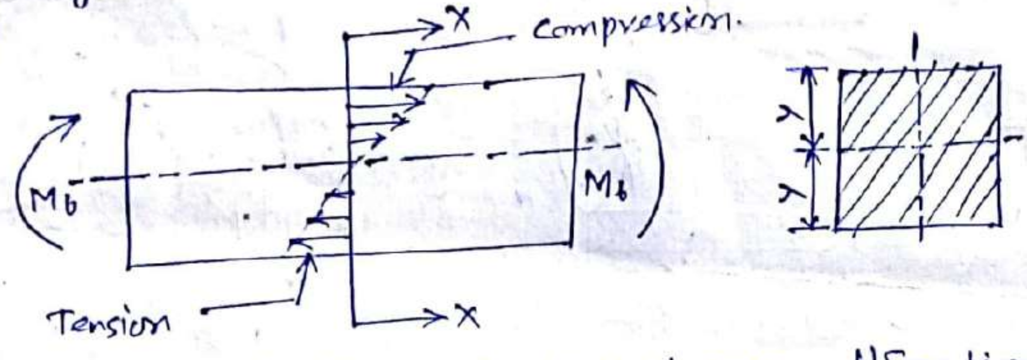


fig: a) distribution of bending stresses   b) Section at XX

The beam is subjected to a combination of tensile stress on one side ~~and~~ of the neutral axis and compressive stress on the other. (Thick leather belt)

The bending stress at any fibre is given by,

$$\boxed{\sigma_b = \frac{M_b y}{I}}$$

Assump

where, σ_b = bending stress at a distance of y from the neutral axis (MPa)

M_b = applied bending moment (N-mm)

I = moment of Inertia of the C-S about the neutral axis (mm^4)

The bending stress is maximum in the farthest fibre from neutral axis. The distribution of stresses is linear and stress is proportional to the distance from the neutral axis.

Stresses due to torsional moment.

A transmission shaft, subjected to an external torque, is shown in fig. ~~2.8(a)~~ The internal stresses which are induced to resist the action of twist, are called torsional shear stresses. The torsional shear stress is given by

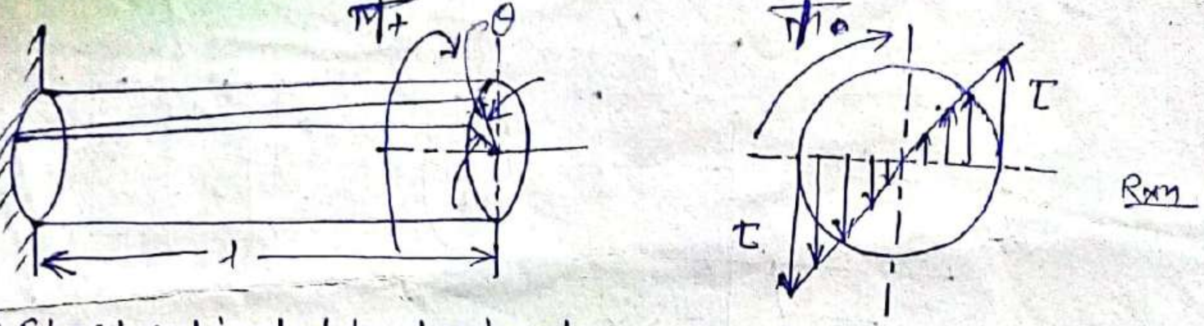
$$\boxed{\tau = \frac{T r}{J}} \quad — (1)$$

where, τ = torsional shear stress at the fibre.

T = applied torque.

r = radial distance of the fibre from the axis of rotation (mm)

J = polar moment of inertia of the c-s about the axis of rotation (mm⁴)



a) Shaft subjected to torsional moment b) Distribution of torsional shear stresses.

The distribution of torsional shear stresses is shown in fig (b). The stress is maximum at the outer fibre are zero at the axis of rotation. The angle of twist is given by.

$$\boxed{\theta = \frac{Tl}{JG}}$$

EI → flexural rigidity

GJ → Torsional rigidity

Assumptions in theory of torsion:

(i) ~~The shaft is straight with circular C-S.~~

(ii) A plane transverse section remains plane after twisting.

(iii) The material is homogeneous, isotropic and obey's Hooke's law.

$$\boxed{P = \frac{2\pi NT}{60}}$$

Eccentric Axial loading: — There are certain mechanical component subjected to an external force, which does not pass through the centroid of the c-s.

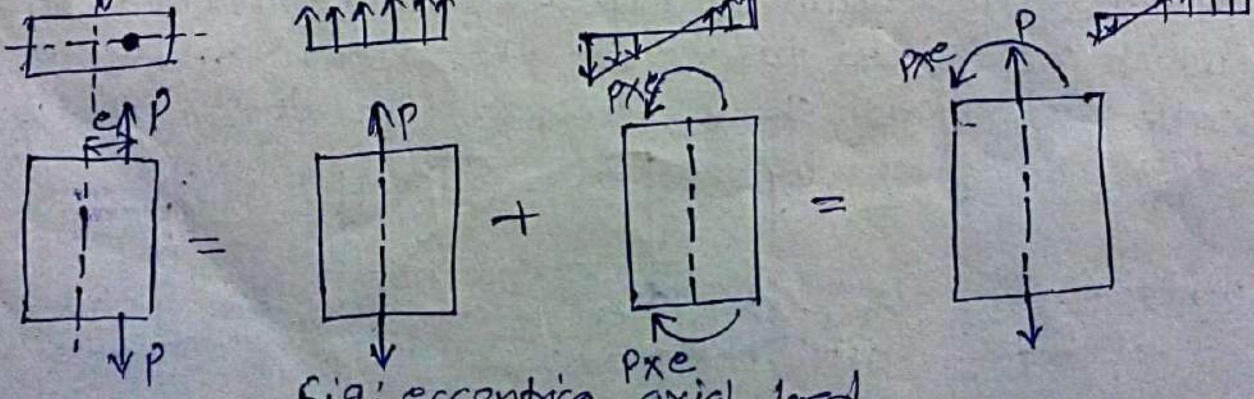


Fig: eccentric axial load

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5.

Factor of Safety: while designing a component, it is necessary to provide sufficient reserve strength in case of an accident. This is achieved by taking a suitable factor of safety.

$$FS = \frac{\text{Failure Stress}}{\text{Allowable stress}}$$

The allowable stress is the stress value, which is used ~~to~~ in design to determine the dimensions of the component. It is considered as a stress, which designer expects will not be exceeded under normal operating conditions.

ductile, brittle

There are no. of factors which are difficult to evaluate accurately in design analysis. Some of the factors are as follows:

(i) Uncertainty in the magnitude of external force acting on the component.

(ii) variations in the dimensions of the component due to imperfect workmanship.

no of assumptions -

The magnitude of FS. depends upon the following factors:

i) effect of failure: in Pr vess

failure of ball bearing in Gear box / Failure of val

ii) Type of load: FS is low (Static) a load which does not vary in magnitude or dirn w.r.t. time.

b) impact load.

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(iii) Degree of Accuracy in Force Analysis:
(i) Precisely determined (ii) uncertain
(iv) material of Component: steel
cast iron (non-homogeneous structure)

(v) Reliability of Component: continuous process equipment, power stations or defense equipment, → high reliability.
(vi) Cost of component: As fs increases, dim of compo. increases, material req. & cost increases. fs low for cheap machine part.

(vii) Testing of m/c element: low f.s. actual condn.

(viii) Service Conditions: When the m/c element is likely to operate in corrosive atm. or high temp. environment, a higher fos is required.

) Quality of manufacture.

3:05 CD

does not twist ... too much under the action of external torque.

does not deflect explanatory

Theories of failure

- The are number of Machine components, which are subjected to different types of loads simultaneously. when the components is subjected to several types of loads, combined stresses are induced.

The design of M/C parts subjected to combined loads should be related to experimentally determined properties of material under symilar conditions. However it is not possible to conduct such tests for dipperent possible ~~condition~~ combination of loads and obtain mechanical properties. In practice the mechanical properties are obtained from a simple tension test. In the tension test, the specimen is axially loaded in tension. Theories of elastic failure provide a relationship between the strength of machine component subjected to complex state of stresses with the mechanica properties obtained in tension test. with the help of these theories, the data obtained in tension test can be used to determine dimensi ns of the component, irrespective of the natur of stresses induced in the component due to complex loads

The Generally accepted theories are:

Ductile Materials (Yield Criteria)

1. Maximum Shear Stress
2. Distortion energy
3. Ductile coulomb - Mohr.

Brittle Materials (Fracture Criteria)

1. Maximum normal stress
2. Brittle Coulomb - Mohr.
3. Modified Mohr.

Maximum Strain theory
Maximum strain energy theory.

I. Maximum Normal Stress Theory :—
(Rankine)

The theory states that the failure of the mechanical component subjected to bi-axial or tri-axial stresses occurs when the ~~mechanical~~ maximum principal stress reached the yield or ultimate strength of the material.

If σ_1, σ_2 and σ_3 are the three principal stresses at a point on the component and

$$\sigma_1 > \sigma_2 > \sigma_3$$

then, the failure occurs whenever,

$$\sigma_1 = S_{yt} \quad \text{or} \quad \sigma_1 = S_{ut}$$

whichever is applicable.

The dimensions of the component are determined by using FOS.

for tensile stresses,

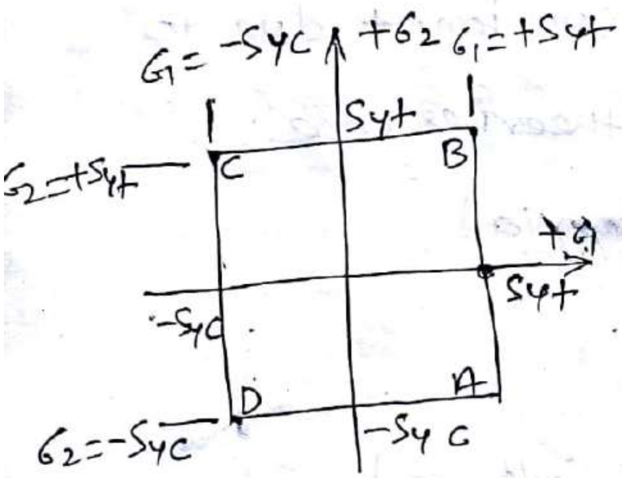
$$\sigma_1 = \frac{S_{yt}}{(FS)} \quad \text{or} \quad \sigma_1 = \frac{S_{ut}}{FS}$$

Symillarily for compressive stresses.

Region of Safety :—

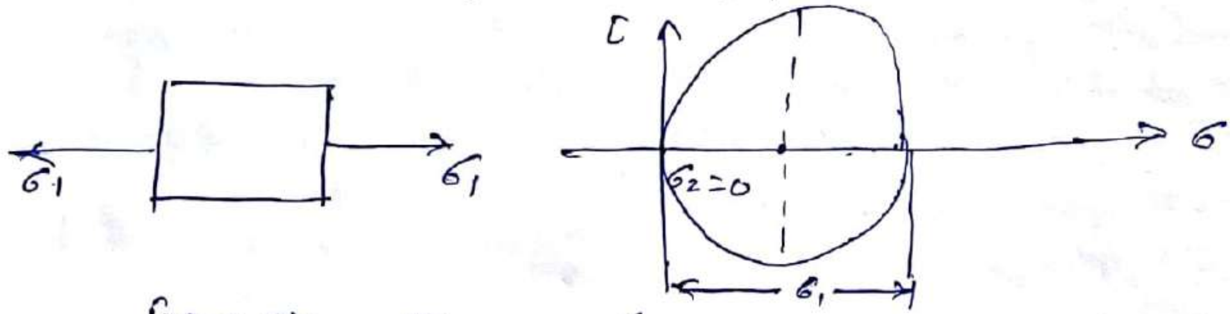
It is assumed that,

$$\boxed{S_{yc} = S_{yt}}$$



2. Maximum Shear Stress Theory :— (Tresca and Guest) The theory states that the failure of a mechanical component subjected to bi-axial or tri-axial stresses occurs when the maximum shear stress at any point in the component becomes equal to the maximum shear stress in the standard specimen of the tension test, when yielding starts.

In the tension test, the specimen subjected to uni-axial stress (σ_1) and ($\sigma_2 = 0$)



from fig, $\tau_{max} = \dfrac{\sigma_1}{2}$

when the specimen starts yielding ($\sigma_1 = S_{yt}$),

$$\tau_{max} = \frac{S_{yt}}{2}$$

Therefore the maximum shear stress theory predicts that the yield strength in shear is half the yield strength in tension, i.e $\boxed{S_{sy} = 0.5\, S_{yt}}$

$$\tau_{12} = \left(\frac{\sigma_1 - \sigma_2}{2}\right) \qquad \tau_{23} = \left(\frac{\sigma_2 - \sigma_3}{2}\right) \qquad \text{—— (1)}$$

--

The largest of these stresses is equated to (τ_{max} or ($S_{yt}/2$)

Considering FOS,

$$\left(\frac{\sigma_1 - \sigma_2}{2}\right) = \frac{S_{yt}}{2(FS)}$$

or $(\sigma_1 - \sigma_2) = \dfrac{S_{yt}}{FS}$

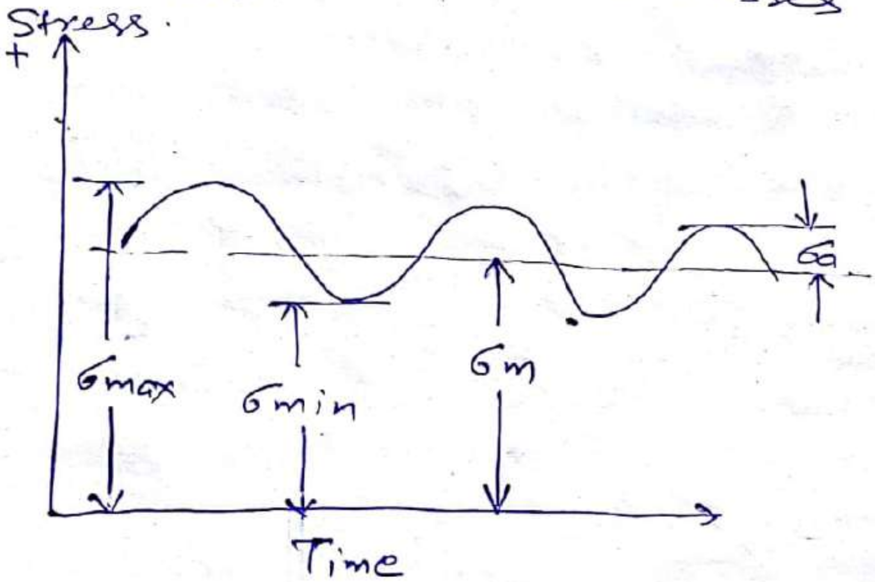
The above relationships are used to determine the dimensions of the component. Refer to eq

Design against fluctuating Load

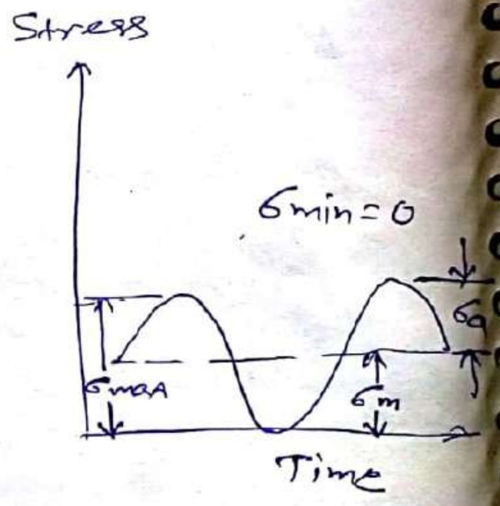
Fluctuating Stresses :— In many applications, the components are subjected to forces which are not static, but vary in magnitude with time. The stresses induced due to such forces are called fluctuating stresses.

It is observed that about 80% of failure of mechanical components are due to fatigue failure resulting from fluctuating stresses.

There are three types of mathematical models for cyclic stresses - fluctuating or alternating stresses, repeated stresses and reversal stresses.



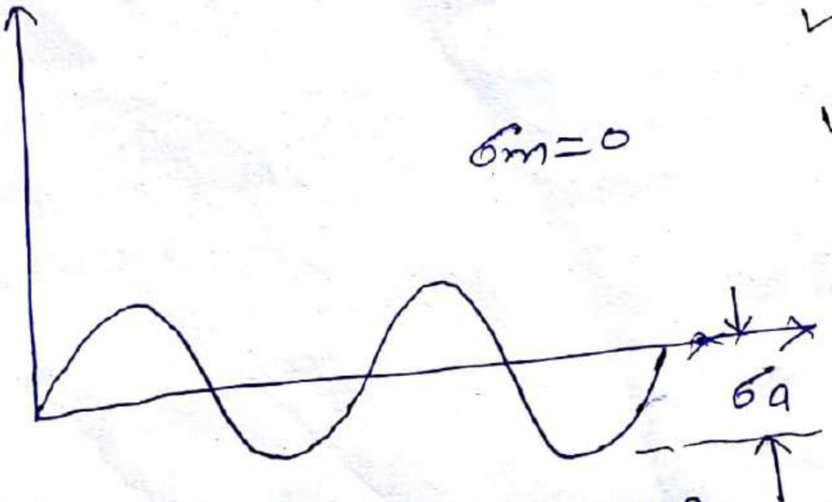
a) Fluctuating Stresses



b) repeated stresses

Stress ratio $= \dfrac{\sigma_{min}}{\sigma_{max}}$

Amplitude ratio $= \dfrac{\sigma_a}{\sigma_m}$



c) Reversal Stresses.

stresses, while σ_m and σ_a are called mean stress and stress amplitude respectively.

$$\sigma_m = \frac{1}{2}(\sigma_{max.} + \sigma_{min})$$

$$\sigma_a = \frac{1}{2}(\sigma_{max} - \sigma_{min})$$

Stress Concentration :—

In design of M/C elements, the following three fundamental equation are used,

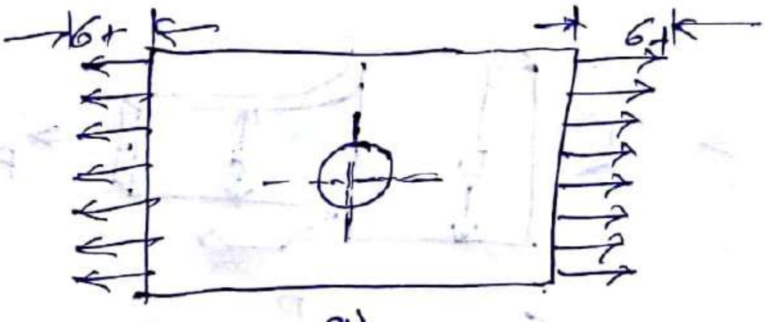
$$\sigma_t = \frac{P}{A} \quad \sigma_b = \frac{M_b y}{I} \text{ and } t = \frac{M_t r}{J}$$

The above equation are called elementary equations. These equation are based on no. of assumptions. One of the assumptions is that there are no discontinuity and irregularities in C-S. of the component.

However, in practice, these discontinuities due to and abrupt change in cross-section are un-avoidable due to certain features of the component such as, oil holes and grooves, keyways and splines, screw threads and shoulders. Under these circumstances, the elementary equations do not give correct results.

Stress concentration is defined as the localisation of high stresses due to irregularities present in the component and abrupt change in C-S.

photo elasticity
plate – epoxy
circular polariscope.



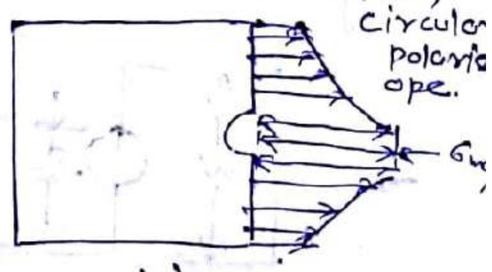


Fig 2.2 : Stress concentration.

In order to consider the effect of stress concentration and find out localised stresses, a factor called stress concentration factor is used

$$K_t = \frac{\text{Higher values of actual stress near discontinuity}}{\text{nominal stress obtained by elementary equation for minimum C-S.}}$$

$$K_t = \frac{\sigma_{max}}{\sigma_o} = \frac{\tau_{max}}{\tau_o}$$ where,

$t \rightarrow$ theoretical.

The magnitude of stress concentration factor depends upon geometry of the component.

The causes of stress concentration are as follows:

i) Variation in properties of material. homoge

ii) Load application point load

iii) Abrupt changes in section.

iv) Discontinuities in the component.

v) Machine Scratches.

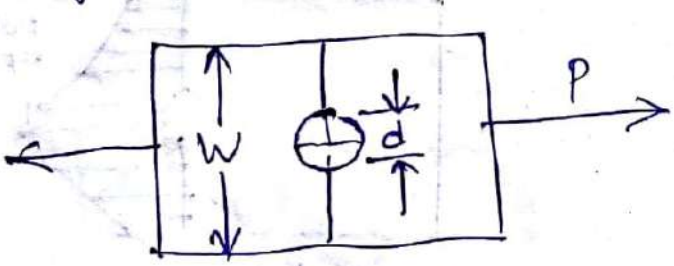
Stress concentration factors:

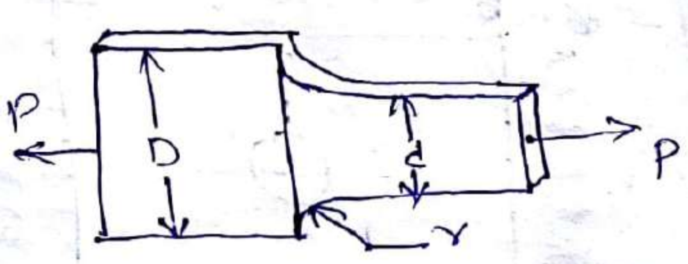
The stress concentration factors are determined by these methods:

(a) the mathematical model based on the theory of elasticity and experimental method like photoelasticity (for simple geometrical shapes)

The chart for different geometrical shape was originally developed by RE Peterson.

At present, FEA Packages are used to find out the stress concentration factor for any geometric shape.

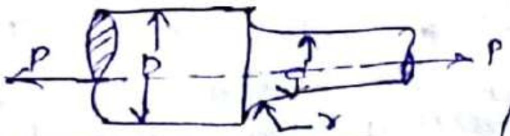




The nominal stress

$$\sigma_o = \frac{P}{(w-d)t}$$

$$\sigma_o = \frac{P}{dt}$$



$$\boxed{\sigma_o = \frac{P}{\frac{\pi}{4}d^2}}$$

(ii) $\sigma_o = \frac{M_b y}{I}$ $\quad I = \frac{\pi d^4}{64}$

and $y = \frac{d}{2}$

(iii) Torsional Moment

$$\tau_o = \frac{M_t r}{J}$$

where, $J = \frac{\pi d^4}{32}$ and $r = \frac{d}{2}$

It is possible to find out the stress concentration facto[r] for some simple geometric shapes using the theory of elasticity.

$$K_t = 1 + 2\left(\frac{a}{b}\right) \quad -(1)$$

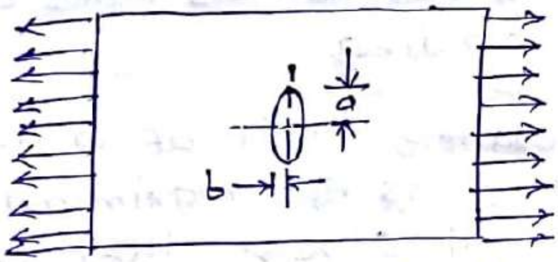


Fig: Stress concentration due to elliptical hole.

where a = half width (or semi-major axis) of the ellip[se] perpendicular to the direction of the loa[d]

b = half width (or semi-minor axis) of the ellip[se] in the direction of load.

As b ~~becomes~~ approaches zero, the ellipse becomes sharpe[r] and sharper. $K_t = \infty$ when $b = 0$

The ellipse becomes circle when $a = b$.

from eq^n (1) $K_t = 1 + 2 = 3$

Fatigue failure :— It has been observed that materials fails under fluctuating stresses a stress magnitude which is lower than the ultimate tensile strength of the material. Sometimes, the magnitude is even lower than the yield strength. Further it has been observed that the magnitude of the stress causing fatigue failure decreases as the no. of stress cycles increases. This phenomenon of decreased resistance of the material to fluctuating stresses is the main characteristics of fatigue failure.

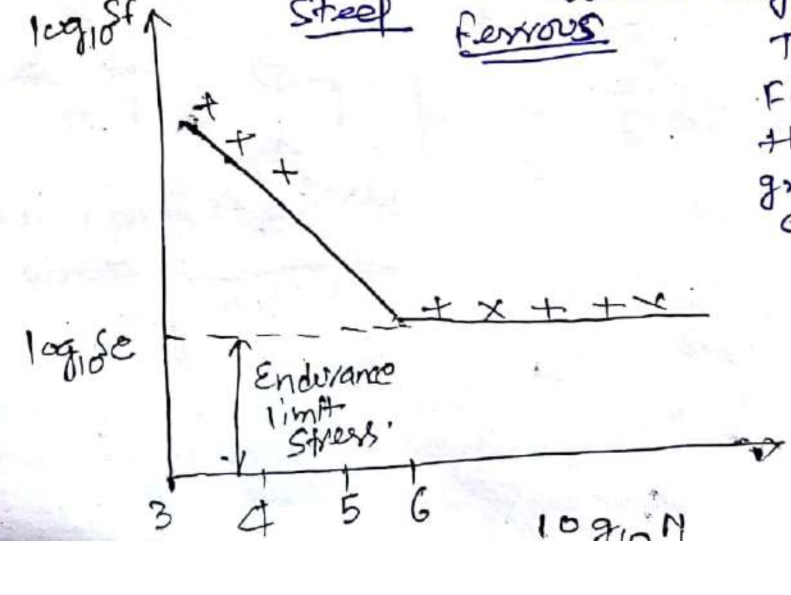
Fatigue failure is defined as time delayed fracture under cyclic loading.

Endurance Limit: Endurance limit of a material is defined as the maximum amplitude of completely reversal stress that a standard specimen sustain for an unlimited no. of cycle without fatigue failure. 10^6 no. of cycles are considered as a sufficient number of cycle to define Endurance limit.

Fatigue life is frequently used with E.L. It is defined as the no. of stress cycle that the St. specimen can sustain during the test before the appearance of first fatigue crack.

The S-N curve is the graphical representation of stress amplitude (S_f) verses number of stress cycle (N) before the fatigue failure on a log-log paper.

Wöhler diagram.



The S-N curve for non-ferrous metals like Al Alloy the S-N curve slopes gradually even after 10^6 cycles.

The endurance limit, is not exactly a property of Material. It is affected by factors such as size of the component, shape of the component, the surface finish, temperature and the notch sensitivity ~~fac~~ of the material.

Notch - Senstivity :— Some materials are not fully sensitive to the presence of notches and hence, for these, a reduced value of K_t can be used. for these, a reduced value of K_t can be used. for these materials, the effective max. stress in fatigue is,

$$\boxed{\sigma_{max} = K_f \sigma_0} \quad \text{or} \quad \boxed{\tau_{max} = K_{fs} \tau_0}$$

where K_f is a reduced value of K_t.

$$K_f = \frac{\text{Maximum Stress in notch specimen}}{\text{Stress in notch free specimen}}.$$

Notch - Senstivity is defined as the susceptibility of a material to succumb to the damaging effect of stress raising notches in fatigue loading.

$$q = \frac{\text{Increase in actual stress over normal stress}}{\text{Increase in theoretical stress over nominal stress}}.$$

actual stress = $K_f \sigma_0$
theoretical stress = $K_t \sigma_0$

Al → less sensitive than steel.
S_{ut} ↑ then q ↑
E → BHN ↑ → q ↑

Therefore

$$q = \frac{\sigma_0 (K_f - 1)}{\sigma_0 (K_t - 1)}$$

or

$$\boxed{q = \frac{(K_f - 1)}{(K_t - 1)}}$$

$$\therefore K_f = 1 + q(K_t - 1)$$

(i) when the material has no sensitivity to notches
$q = 0$ and $K_f = 1$

(ii) when the material is fully sensitive to notch then $\boxed{K_f = K_t}$

Endurance limit — Approximate Estimation

Two seperate notations are used for endurance limit, viz. (S_e') and S_e, where,

S_e' = Endurance limit stress of a rotating beam specimen subjected to reversed bending stress (N/mm²)

S_e = Endurance limit stress of a particular mech. component subjected to reversed bending stress.

Approximate relations.

for Steel $\boxed{S_e' = 0.5 S_{ut}}$

For CI $\boxed{S_e' = 0.4 S_{ut}}$

For wrought Al Alloy $\boxed{S_e' = 0.4 S_{ut}.}$

for cast Al alloy $\boxed{S_e' = 0.3 S_{ut}}$

⇒ These relationship are based on 50 % reliability

⇒ The actual ~~specimen~~ comp. has different specifications and working conditions. To account for these different modifying factors are used which are called derating factors. means these factors reduced the Endurance limit to suit the actual component.

$$\boxed{S_e = K_a K_b K_c K_d S_e'}$$

where K_a = surface finish factor.
K_b = size factor.
K_c = reliability factor
K_d = modifying factor to account for stress concentration.

Reliability factor :— The greater the likelyhood that a part will survive, the more is the reliability and less is the reliability factor. The RF is one for 50% reliability. This means that 50% of the component will survive in given set of condition. To ensure that more than 50% of parts will survive, the stress amplitude on the component should be lower than the tabuled value of Endurance limit.

Modifying factor to account for stress concentration:-
To apply the effect of stress concentration, the designer can either reduce the endurance limit by Kd or increase the stress amplitude by Kf. We will use first approach.

$$\boxed{K_d = \frac{1}{K_f}}$$

$$\boxed{S_{se} = 0.5 S_e}$$

According to distortion energy theory $\boxed{S_{se} = 0.577 S_e}$

→ The endurance limit in axial loading is lower than the rotating beam test.

For axial loading,

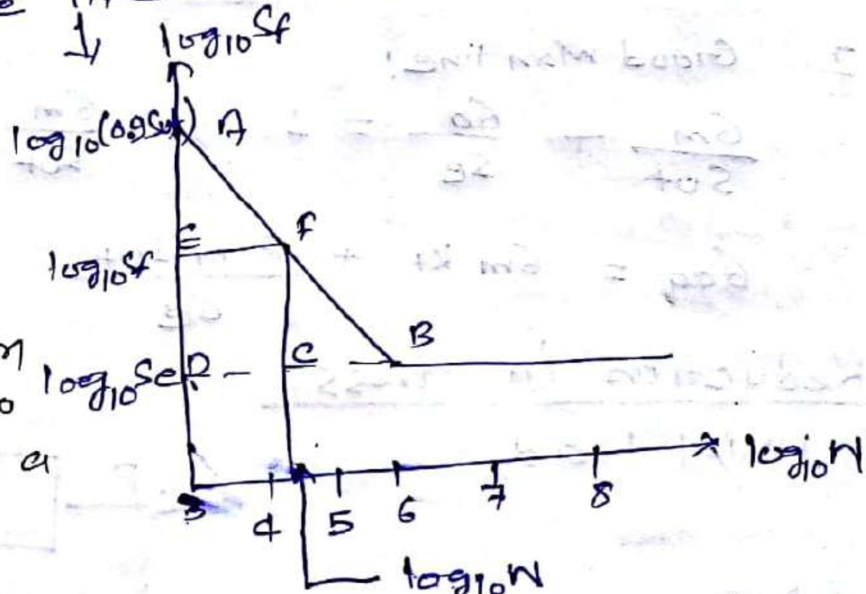
$$\boxed{(S_e)_a = 0.8 S_e}$$

Reversed Stresses:— Design for finite and infinite life:

The design problems for completely reversed stresses are further divided into two groups,

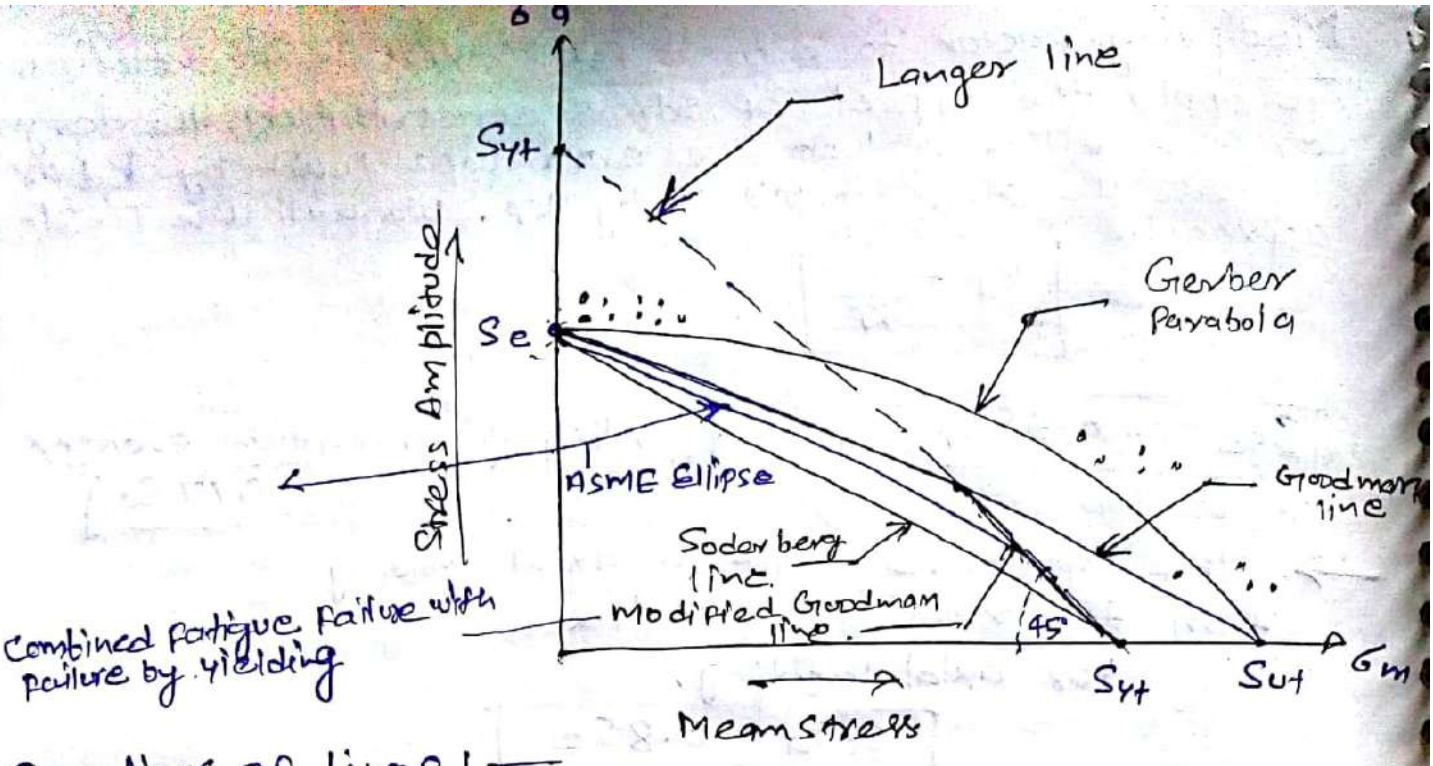
1) Design for infinite life → Endurance limite.
2) Design for finite life → S-N curve.

$$\left\{\begin{aligned} \sigma_a &= \frac{S_e}{fs} \\ \tau_a &= \frac{S_{se}}{fs} \end{aligned}\right.$$



St. line AB drawn from $(0.9 S_{ut})$ at 10^3 cycles to (S_e) at 10^6 cycles on a log-log paper.

→ When a component is subjected to fluctuating stresses as shown in fig. there is mean stress (σ_m) as well as stress amplitude (σ_a). It has been observed that the mean stress component has an effect on fatigue failure when it is present in combination with alternating component.



Equations of lines! —

1. Soderberg line:

$$\frac{\sigma_m}{S_{yt}} + \frac{\sigma_a}{S_e} = 1 \implies \frac{\sigma_m K_t}{S_{yt}} + \frac{\sigma_a K_f}{S_e} = \frac{1}{FOS}$$

$$\sigma_{eq} = \sigma_m K_t + \frac{\sigma_a K_f S_{yt}}{S_e} \implies \text{for more than one load}$$

2. Goodman line!

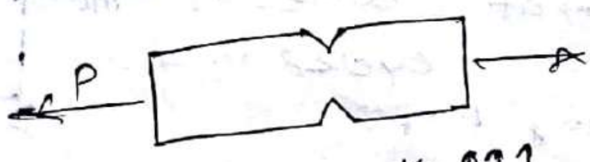
$$\frac{\sigma_m}{S_{ut}} + \frac{\sigma_a}{S_e} = 1 \implies \frac{\sigma_m}{S_{ut}} K_t + \frac{\sigma_a K_f}{S_e} = \frac{1}{FOS}$$

$$\frac{S_a}{S_e} + \left(\frac{S_m}{S_{ut}}\right)^2 = 1$$ — Gerber Parabola.

$$\sigma_{eq} = \sigma_m K_t + \frac{\sigma_a K_f S_{ut}}{\sigma_e}$$

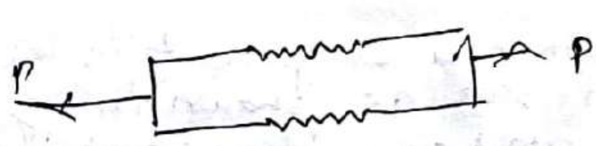
Reduction in Stress

1) Axial load.



$K_t \uparrow\uparrow\uparrow$

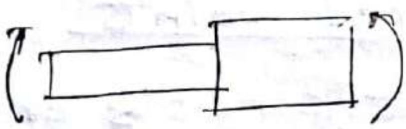
a) Provide more notch



b) Drilling hole.

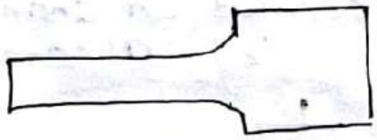
c) Removal of Material.

2) Bending

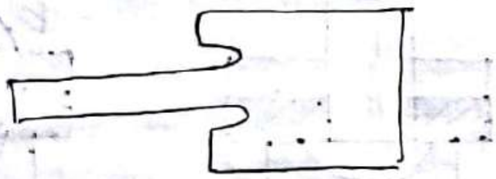


$K_t \uparrow\uparrow\uparrow$

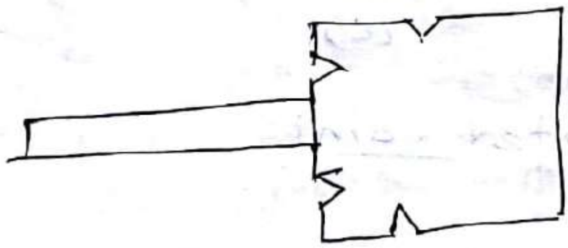
a) Provide $r > 0$



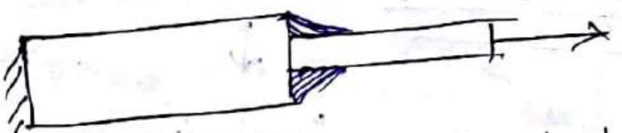
b) Undercutting



c) More notches.



⇒ Effect of K_t is neglected in case of ductile Materi[al]. Because geometry near the discontinuity will be rearranged due to local yielding.



⇒ K_f cannot be neglected for any material.

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Rivet: — A rivet consists of a cylindrical shank with a head at one end as shown in Fig 1(a). This head is formed on shank by an upsetting process in a machine called automatic header.

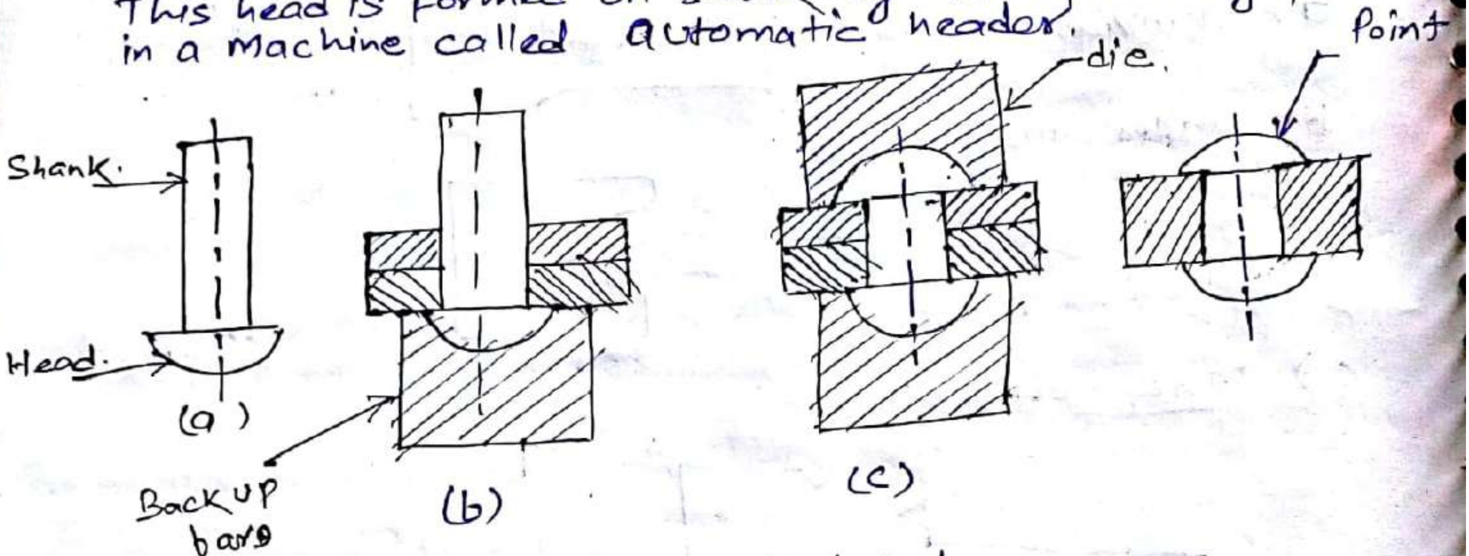


Fig: 1 Riveted Joint

Types of Riveted Joints: —

Riveted Joints

- Lap joint (Depending upon no. of Rows)
  - Single-Riveted lap Joint
  - Double-Riveted lap joint or tripple-riveted lap joint
    - Arrangement
      - Chain Pattern
      - Zig-Zag Pattern.
- Butt joint
  - Depending upon no. of rows of rivets in each plate
    - Single row butt joint
    - Double-row butt joint
  - Depending upon no. of straps.
    - Single strap butt joint
    - Double butt joint

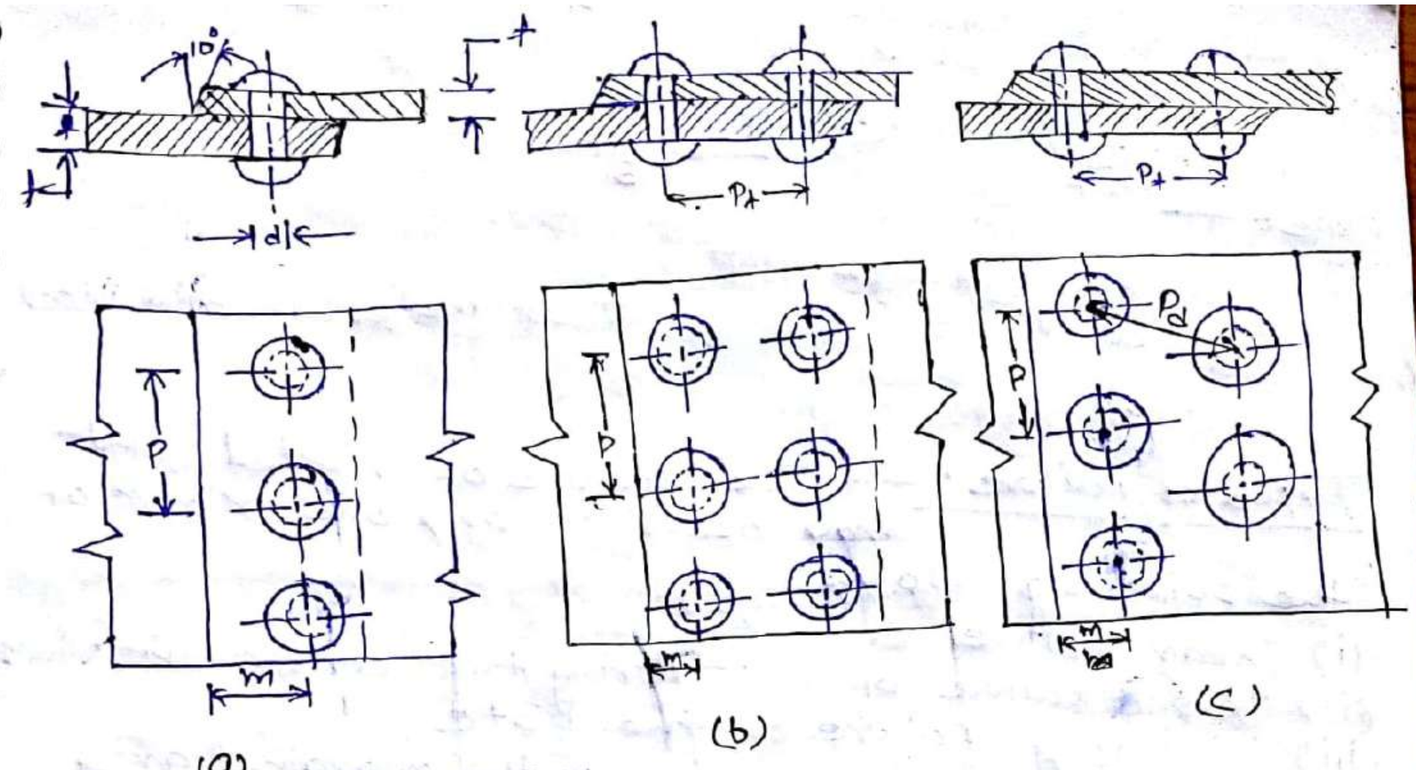


fig: 2 Types of lap joints

The following terms are used in terminology of riveted Joints:

(i) Pitch:— It is defined as the distance between centre of one rivet to centre of adjacent rivet in the same row.

Usually, $p = 3d$. where d = shank diameter of rivet.

(ii) Margin (m) The margin is the distance between the edge of the plate to the centreline of rivets in the nearest row. Usually,

$$m = 1.5d$$

(iii) Transverse Pitch (P_t):— It is the distance between (back or row pitch) two consecutive rows of rivets in the same plate.

usually,
$$P_t = 0.8P \quad (\text{for chain riveting})$$
$$P_t = 0.6P \quad (\text{for zig-zag riveting})$$

(iv) Diagonal Pitch (P_d) Diagonal pitch is the distance between the centre of one rivet to the centre of the adjacent rivet located in the adjacent row.

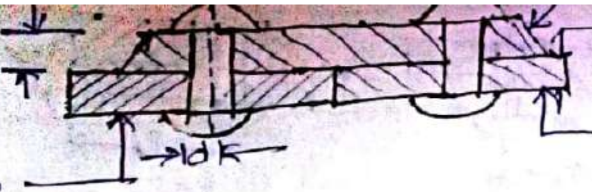
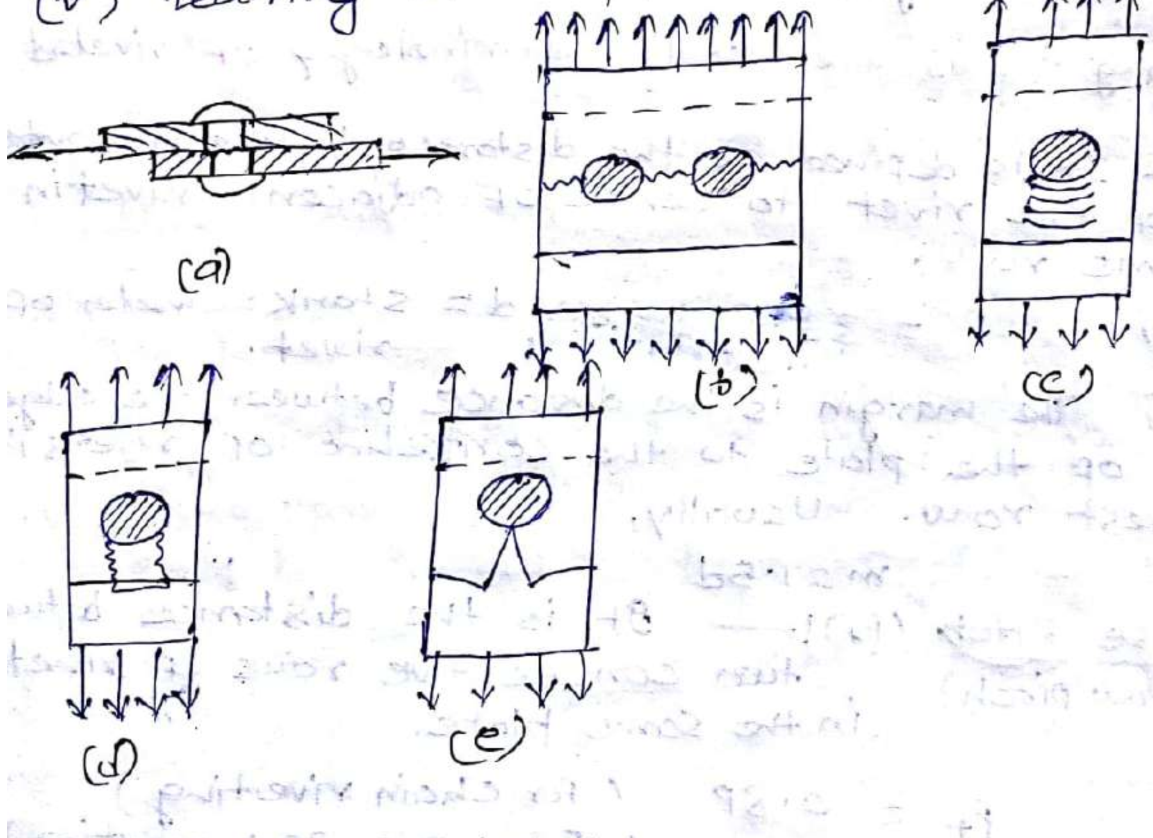


Fig: Single riveted single strap butt joint.

Rivet Material - mild steel.

Types of failure :— The failure of riveted joint may occur in any one or more of the following ways:

(i) Shear failure of rivet.
(ii) Tensile failure of rivet during two consecutive rivets
(iii) Crushing failure of the plate.
(iv) Shear failure of plate in the marginal area.
(v) tearing of the plate. in the marginal area.



Strength Equations :—

The strength of the riveted joint is defined as the force that the joint can withstand without causing failure. When the operating force acting on the joint exceeds this force, failure occurs.

$$P_s = \frac{\pi}{4} d^2 \tau \quad \text{(rivet in single shear)}$$

where P_s = shear resistance of rivet per pitch length (N)

d = Shank diameter of rivet (mm).

t = permissible shear stress of rivet material (N/mm²)

$$P_s = \frac{\pi}{4} d^2 \tau n$$ where n = no. of rivets per pitch length

$$P_s = 2\left[\frac{\pi}{4} d^2 \tau n\right] \quad \text{(for double shear)}$$

(ii) Tensile strength of Plate between rivets.

$$P_t = (P-d)\, t\, \sigma_t \quad \text{two}$$

(iii) Crushing Strength of Plate! —

$$P_c = d\, t\, \sigma_c\, n$$

Efficiency of Joints! — The efficiency of the riveted joint is defined as the ratio of the strength of riveted joint to the strength of unriveted solid plate.

The strength of the ~~riveted~~ solid plate of width, equal to the pitch P and thickness t, subjected to tensile strength σ_t is given by

$$P = p\, t\, \sigma_t$$

Therefore the effeciency is given by,

$$\eta = \frac{\text{Lowest of } P_s, P_t, \text{ and } P_c}{P}$$

Design of Boiler Shell :—

Type of Riveted joint in cylindrical Boiler Shell

- ↓ longitudinal butt Joint (usually double strap tripple riveted)
- ↓ Circumferential lap Joint

The longitudinal Joint is used to increase the diameter of shell while Circumferential joint is used to increase the length of the shell.

$\sigma_h \rightarrow$ causes failure of longitudinal joint.

$\sigma_l \rightarrow$ causes failure of circumferential Joint.

longitudinal joint should be stronger than circumferential joint because hoop stress is twice of longitudinal stress So butt joint is used.

Boiler must conform to the Indian Boiler Regulation Act.

The following procedure is adopted for the design of a longitudinal butt joint for the boiler shell as illustrated in fig.

(i) Thickness of Boiler Shell.

$$t = \frac{P_i D_i}{2 \sigma_i \eta} + CA \quad (1 \text{ or } 2)$$

The permissible tensile stress

$$\sigma_t = \frac{S_{ut}}{FS.}$$

$FS \rightarrow 5$ (taken in case of boiler)

Diameter of rivet :—

when $t > 8$ mm

$d = 6\sqrt{t} \rightarrow$ Unwin's formula.

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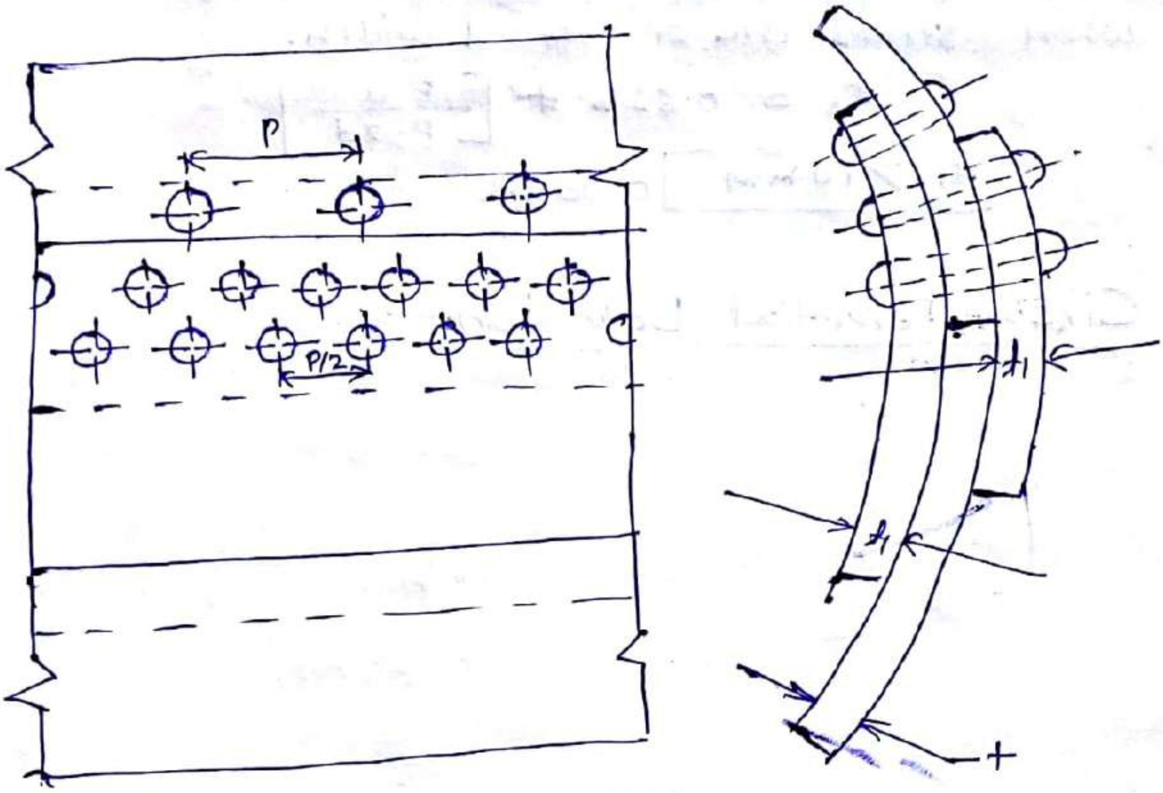
the diameter of rivet hole is slightly more than rivet diameter.

$$\boxed{d' = d + (1 \text{ to } 2\ mm)}$$



The tensile strength of the plate per pitch length in outer row of the rivet is given by.

$$P_t = (P-d)\,t\,\sigma_t \quad \text{—— (1)}$$

$$P_s = \left[\frac{\pi}{4}d^2\tau\right]n_1 + 1.875\left[\frac{\pi}{4}d^2\tau\right]n_2$$

or, $$P_s = (n_1 + 1.875\,n_2)\left[\frac{\pi}{4}d^2\tau\right] \quad \text{—— (2)}$$

By equating eq^n (1) & (2) we can obtain P.

But According to IBR.

min Pitch $P_{min} = 2d$

& Max Pitch $\boxed{P_{max} = Ct + 41.28}$

zig zag Pattern:

$P_t = 0.2P + 1.15d$ —— (3) for outer row & next nearest row

$P_t = 0.165P + 0.67d$ —— (4) for middle & inner row.

Margin(m) :— $m = 1.5d$

Thickness of straps (t_1)

$t_1 = 0.75t$ (for wide strap)

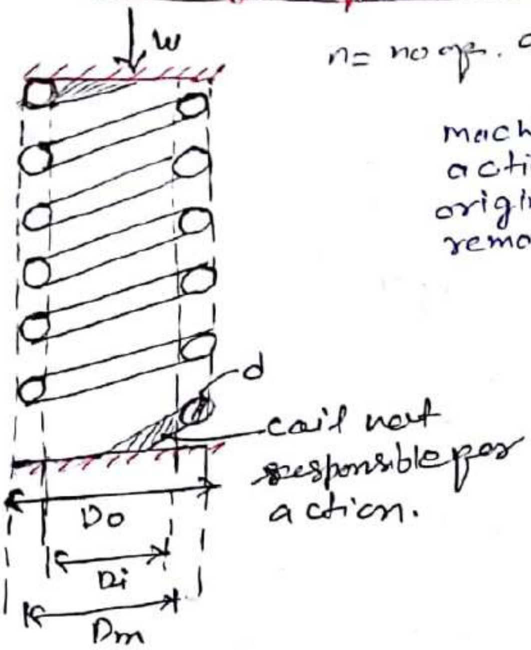
$= 0.625t$ (for narrow strap)

when straps are of equal width.

$$t_1 = 0.625\, t \left[\frac{p-d}{p-2d}\right]$$

$\boxed{t_1 > 10\,mm}$ always.

Design of Helical Spring



n = no of. active coils.

A spring is defined as an elastic machine element, which deflect under the action of the load and returns to its original ~~po~~ shape when the load is removed.

Max. Length or free length of spring :— It is length of spring when there is no load acting on spring.

Compressive length :— It is length of spring under max. deflection condition. when load applying on the spring.

Solid length of spring :— It is the length of the spring when there is no gap b/w the coils

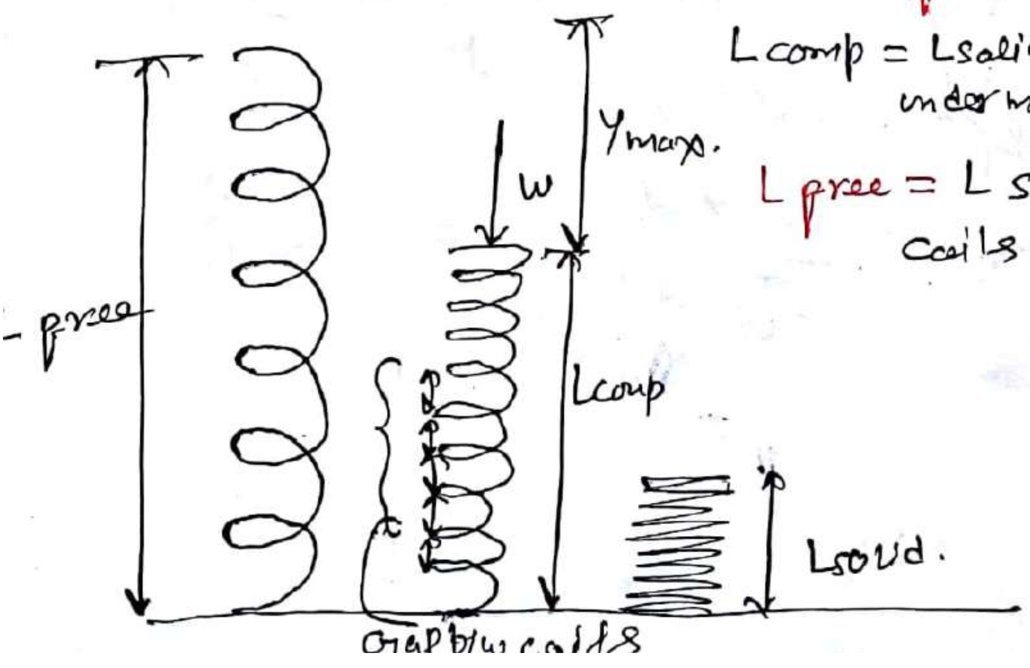
$L_{free} = L_{comp} + Y_{max}$

$L_{comp} = L_{solid}$ + gap b/w the coils under max. def. Cond^n

$L_{free} = L_{solid} + Y_{max}$ + gap b/w coils under max. def^n cond

$\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad \| $

$\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad\quad 0.15 Y_{max}$



$L_{free} = L_{solid} + Y_{max} + \ldots$

$$\boxed{L_{free} = L_{solid} + 1.15\, Y_{max}}$$

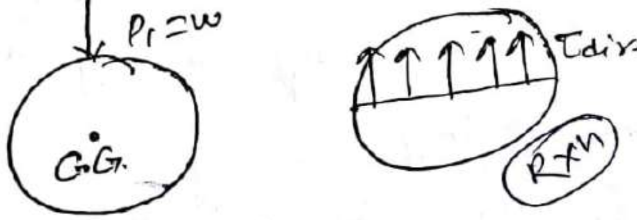
Design !—

Step. I !— Apply two equal & opposite forces P_1 & P_2 passing through the C.G of spring wire. such that $\boxed{P_1 = P_2 = W}$

Step II !—

Effect of P_1 : As P_1 passes through C.G of spring wire it result direct shear stress induced in the spring wire of equal magnitude at each point.

$$\tau_{direct} = \frac{W}{\frac{\pi}{4} d^2}$$



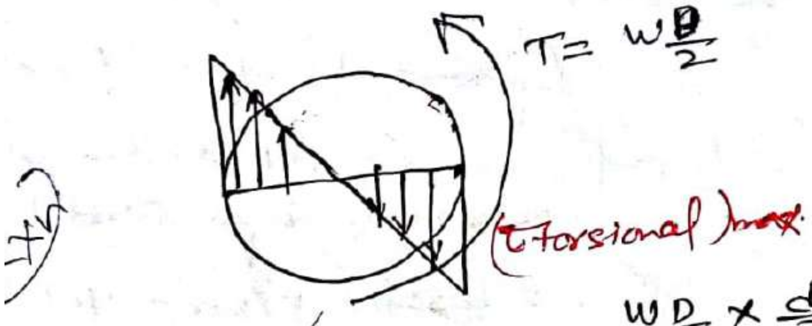
we see RxN.

step III :—

Effect of P_2 & W !— P_2 & W causes a constant twisting couple of magnitude (P×e) or $\{ w \times \frac{D}{2} \}$ {load & eccentricity} in anticlockwise dirn. about C.G of wire.

$$\frac{T}{I_p} = \frac{\tau}{R}$$

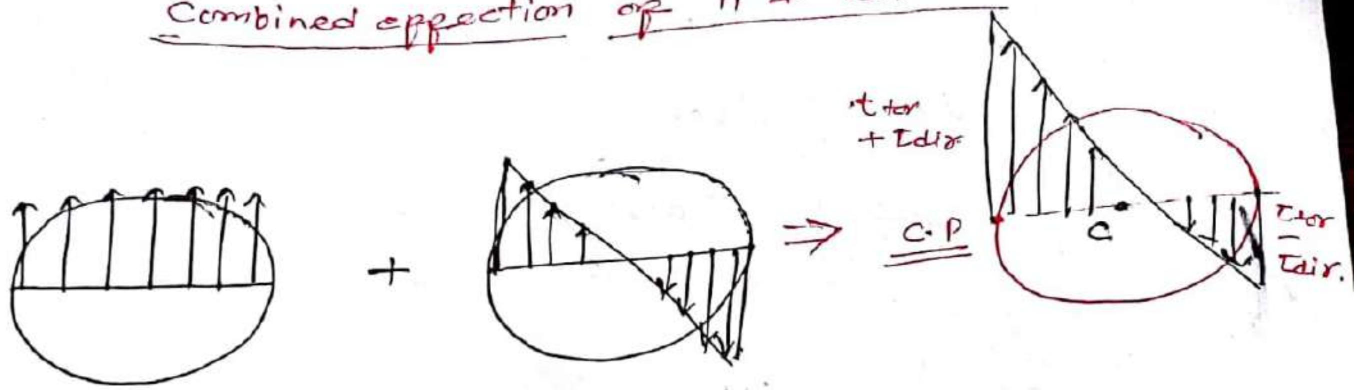
$$\boxed{\tau = \frac{T r}{I_p}}$$



$$(\tau_{max})_{torsional} = \frac{\frac{WD}{2} \times \frac{d}{2}}{\frac{\pi}{32} d^4} = \frac{8 W D}{\pi d^3}$$

Step IV :—

Combined effection of P_1 & P_2, W :—



Hence inner fibre of spring wire subjected to max stress.

$$\tau_{max} = (\tau_{torsional})_{max} + \tau_{direct}.$$

$$\tau_{max} = \frac{8WD}{\pi d^3} + \frac{4W}{\pi d^2}$$

$$\boxed{C = \frac{D}{d}}$$

$$= \frac{8WD}{\pi d^3}\underbrace{\left[1 + \frac{0.5}{C}\right]}$$

K_{sh} ⇒ direct shear stress factor.

$$\tau_{max} = \frac{8WD}{\pi d^3} K_{sh}$$

K_C ⇒ curvature effect factor.

$$\boxed{\tau_{max} = \frac{8WD}{\pi d^3} K_{sh} K_C}$$

K_W (Walh's factor)

where

$$\boxed{K_W = \frac{4C-1}{4C-4} + \frac{0.615}{C}}$$

Safe Cond^n :—

$\tau_{max} \le \tau_{permissible}$

$$\frac{8WD}{\pi d^3} \times K_w \le \tau_{per}$$

$$\boxed{W_{max} = \frac{\pi d^3}{8 D K_w} \tau_{per}}$$

$\Downarrow$

Strength of spring

Expression for deflection & Stiffness :—

$$\text{Strain Energy} = \frac{1}{2} T \theta$$

$$U = \frac{1}{2} T \left(\frac{TL}{JG}\right)$$

$$= \frac{1}{2} \left(\frac{WD}{2}\right) \left(\frac{WD}{2} \cdot \frac{\pi D n}{G \cdot \frac{\pi}{32} d^4}\right)$$

$$\boxed{U = \frac{4 W^2 D^3 n}{G d^4}}$$

$$\delta = \frac{\partial U}{\partial W}$$

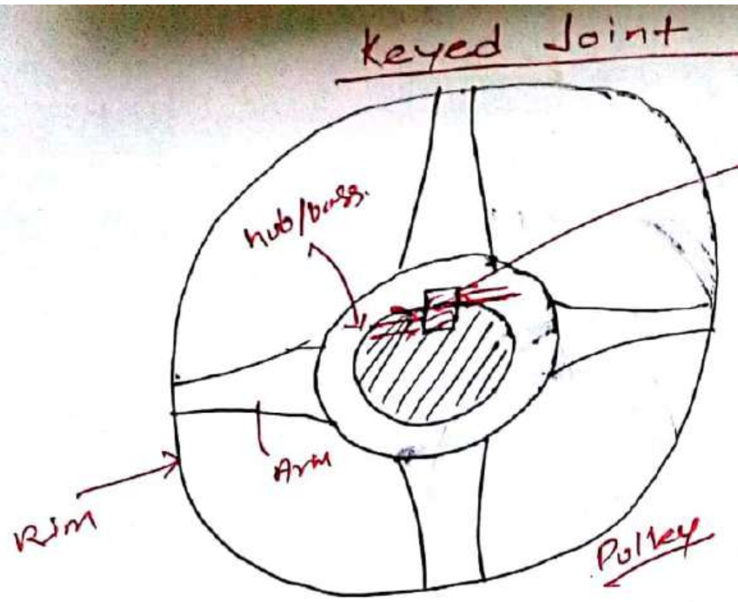
$$\boxed{\delta = \frac{8 W D^3 n}{G d^4}}$$

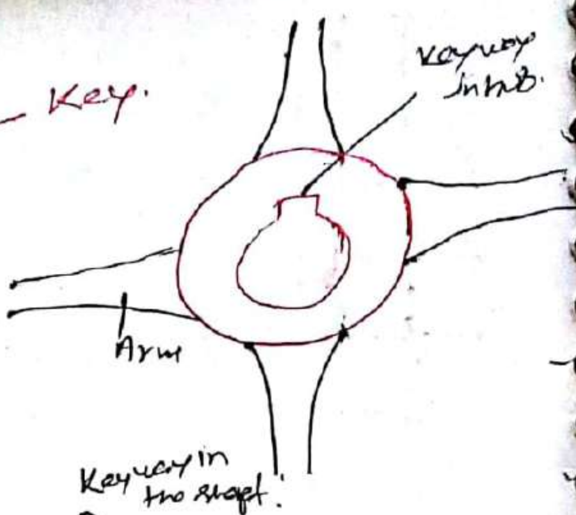
$$\text{Stiffness } K = \frac{\text{load}}{\delta}$$

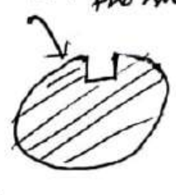
$$\boxed{K = \frac{G d^4}{8 D^3 n}}$$

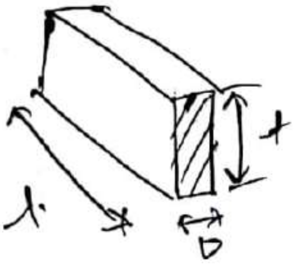
$$\boxed{K \propto \frac{1}{n}}$$

Keyed Joint





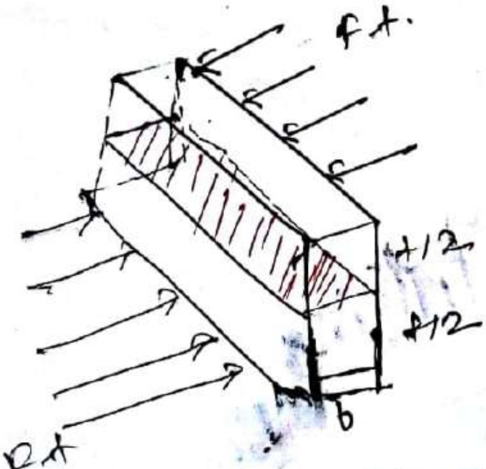




Key is the temporary fastener which is inserted b/w shaft and its assembly (Pulley, gear, flywheel etc) to transmit power b/w them. by preventing relative motion b/w them.

Note! — Key is the weakest element among shaft, assembly & Key.

Shear strength of the key !—



$$\tau_{ind} = \frac{F_t}{b\,l}$$

$$F_t \times \frac{D}{2} = T \qquad F_t = \frac{2T}{D}$$

$$\tau_{ind} = \frac{2T}{D\,b\,l}$$

Safe Cond^n

$$\tau_{ind} \le \tau_{perm.}$$

$$\frac{2T}{D\,b\,l} \le \tau_{per}$$

Shear strength in terms of torque.

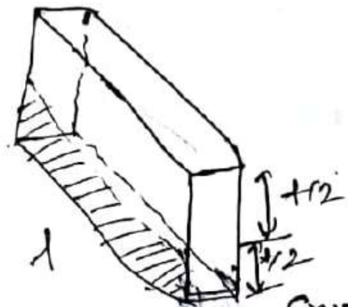
$$\boxed{T_{max} = \frac{D\,b\,l}{2}\,\tau_{per}}$$

Crushing strength of Key :—

compression

$$(\sigma_{ind})_{crush} = \frac{2F_t}{t l}$$

$$(\sigma_{ind})_{crush} = \frac{4T}{t l D}$$



Safe condn.

$$\frac{4T}{t l D} \le \sigma_{per}$$

Crushing strength of key in term of torque

$$\boxed{T_{max} = \frac{t l D \sigma_{per}}{4}}$$

Actual Strength of key :—

min. of $(T_{max})_{crush}$, $(T_{max})_{shearing}$

Type of key :—

Key

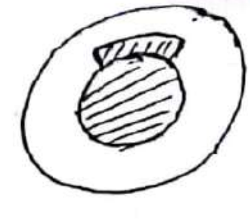
Sunk key

1. Hub part of key in Hub & another part of key in shaft.
2. Hence keyway is present in both hub & shaft
3. Key is responsible to transmit power.
4. Power transmission capacity is more

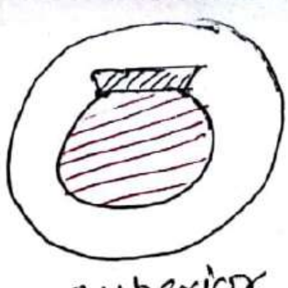
Saddle Key

1. Saddle key is present only in hub.
2. Hence keyway is also present only in the hub.
3. friction force is responsible to transmit power
4. Power transmission capacity is less.
5. Because of no keyway present in the shaft stress conc. factors are less. hence strength of the shaft increases & cost decreases

(1) Hollow saddle key.



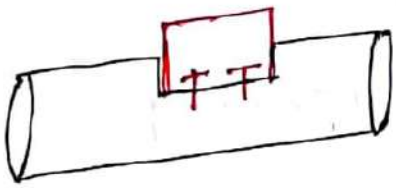
(2) Flat saddle key.

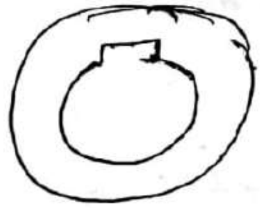


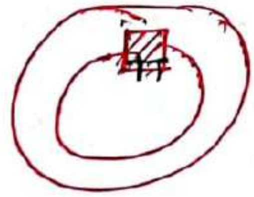
- Flat saddle key is more superior ~~than~~ to Holloy saddle key w.r.t power transmission capacity.

Sunk key :—

(1) Feather key :—

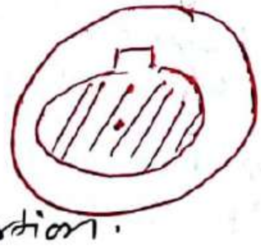






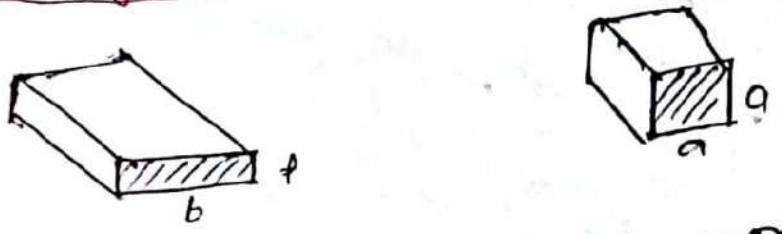
- key is fixed either with shaft or with hub.
- Permit axial relative motion b/w shaft & its assembley.
- It is a type of parallel key.

(2) Woodruff key :—



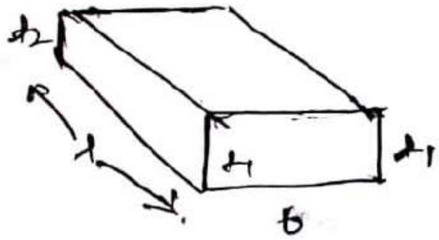
- Key contains semicircular disc type portion.
- Hence keyway is also semi circular disc.
- ✓ It can align itself hence used in tapered shaft
- Extra depth of Key in the shaft provide more power transmission capacity.

- Rectangular & square sunk key



- Rectangular sunk key is more stable than square sunk key & mostly used in Industrial application.

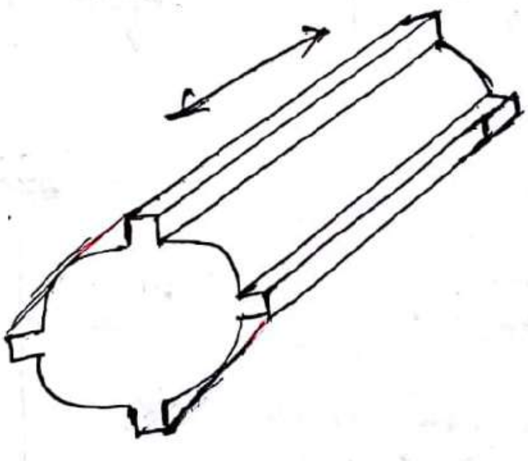
- Parallel key & Tapered key :—



Taper key

- Splined key or splined shaft :—

- Keys are inbuilt with shaft.



- Keyways are inbuilt with shaft
- Permit axial relative motion b/w shaft & hub.
- Used in automobile gearboxes & clutches.

Step I *Permissible shear stress for rivets*

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\ S_{yt}}{(fs)} = \frac{0.5\ (250)}{(2.5)} = 50\ \text{N/mm}^2$$

Step II *Diameter of rivets*

Since there are three rivets,

$$3\left[\frac{\pi}{4}d^2\right]\tau = P \quad \text{or} \quad 3\left[\frac{\pi}{4}d^2\right]50 = 50 \times 10^3$$

$\therefore\ d = 20.60$ or 22 mm (i)

Step III *Permissible tensile stress for plates*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{250}{2.5} = 100\ \text{N/mm}^2$$

Step IV *Thickness of plates*

As shown in Fig. 4.10(b),

$$\sigma_t\,(200 - 3d)\,t = P$$

or $100(200 - 3 \times 22)\,t = 50 \times 10^3$

$\therefore\ t = 3.73$ or 4 mm (ii)

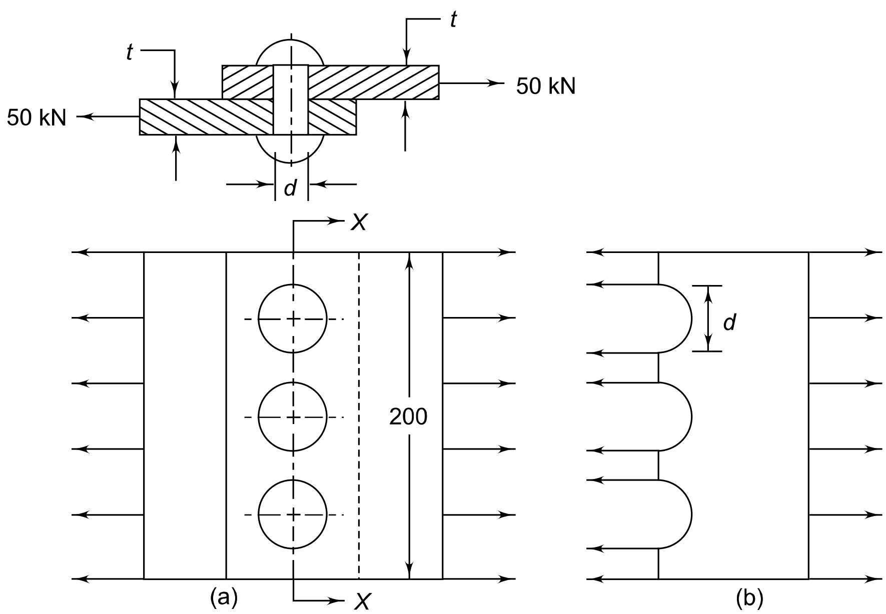


Fig. 4.10 *(a) Riveted Joint (b) Tensile Stress in Plate*

4.9 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 also used to connect a rod on one side with some machine part like a crosshead or base plate on the other side. It is not used for connecting shafts that rotate and transmit torque. Typical applications of cotter joint are as follows:

(i) Joint between the piston rod and the crosshead of a steam engine
(ii) Joint between the slide spindle and the fork of the valve mechanism
(iii) Joint between the piston rod and the tail or pump rod
(iv) Foundation bolt

The principle of wedge action is used in a cotter joint. A cotter is a wedge-shaped piece made of a steel plate. The joint is tightened and adjusted by means of a wedge action of the cotter. The construction of a cotter joint, used to connect two rods *A* and *B* is shown in Fig. 4.11. Rod-*A* is provided with a socket end, 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. The cotter has uniform thickness and the width dimension *b* is given a slight taper. The taper is usually 1 in 24. The taper is provided for the following two reasons:

(i) When the cotter is inserted in the slot through the socket and the spigot and pressed by means of hammer, it becomes tight due to wedge action. This ensures tightness of the joint in operating condition and prevents loosening of the parts.

(ii) Due to its taper shape, it is easy to remove the cotter and dismantle the joint.

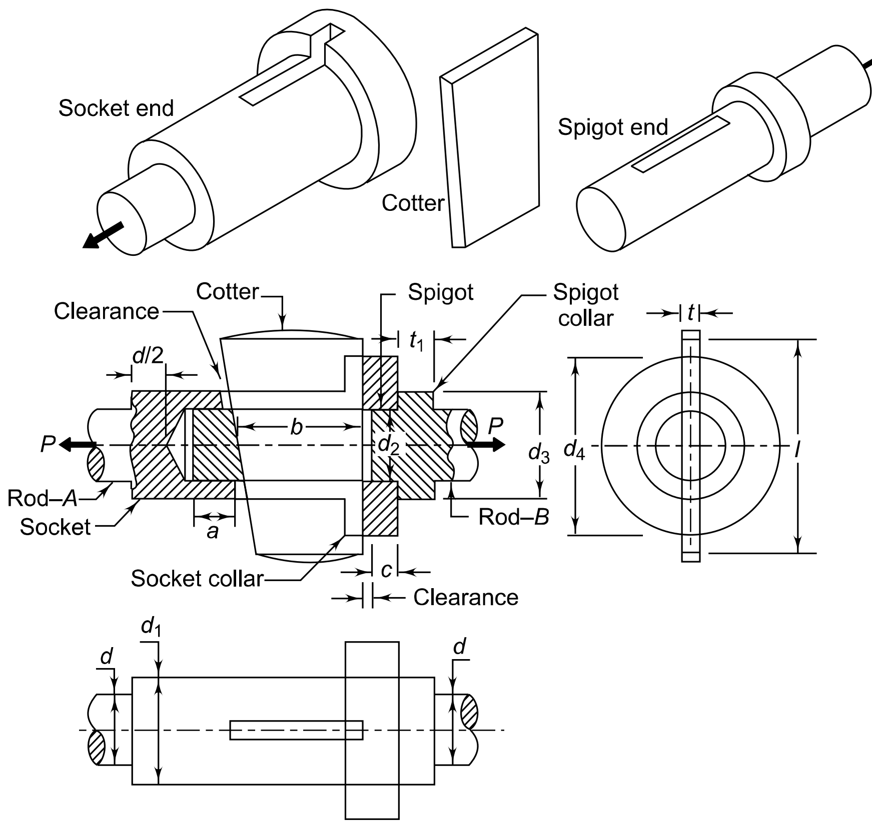


Fig. 4.11 *Cotter Joint*

The taper of the cotter as well as slots is on one side. Machining a taper on two sides of a machine part is more difficult than making a taper on one side. Also, there is no specific advantage of a taper on two sides. A clearance of 1.5 to 3 mm is provided between the slots and the cotter. When the cotter is driven in the slots, the two rods are drawn together until the spigot collar rests on the socket collar. The amount by which the two rods are drawn together is called the *draw* of the cotter. The cotter joint offers the following advantages:

(i) The assembly and dismantling of parts of the cotter joint is quick and simple. The assembly consists of inserting the spigot end into the socket end and putting the cotter into their common slot. When the cotter is hammered, the rods are drawn together and tightened. Dismantling consists of removing the cotter from the slot by means of a hammer.

(ii) The wedge action develops a very high tightening force, which prevents loosening of parts in service.

(iii) The joint is simple to design and manufacture.

Free body diagram of forces acting on three components of cotter joint, viz., socket, cotter and spigot is shown in Fig. 4.12. This diagram is constructed by using the principle that actions and reactions are equal and opposite. The forces are determined in the following way,

(i) Consider rod–A with a socket end. The rod is subjected to a horizontal force P to the left. The sum of all horizontal forces acting on the rod A with socket must be equal to zero. Therefore, there should be a force P to the right acting on the socket. This force is shown by two parts, each equal to (P/2) on the socket end.

(ii) Consider rod-B with the spigot end. The rod is subjected to a force P to the right. The sum of all horizontal forces acting on rod-B must be equal to zero. Therefore, there should be a force P to the left on the spigot end.

(iii) The forces shown on the cotter are equal and opposite reactions of forces acting on the spigot end of rod-B and the socket end of rod-A.

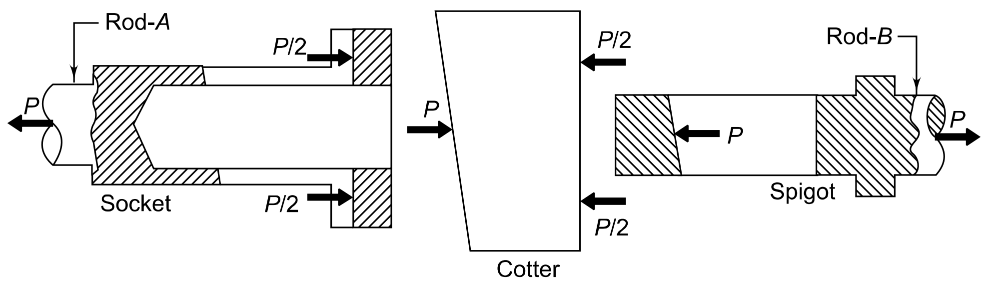


Fig. 4.12 *Free Body Diagram of Forces*

For the purpose of stress analysis, the following assumptions are made:

(i) The rods are subjected to axial tensile force.
(ii) The effect of stress concentration due to the slot is neglected.
(iii) The stresses due to initial tightening of the cotter are neglected.

In Fig. 4.11, the following notations are used

P = tensile force acting on rods (N)
d = diameter of each rod (mm)
d_1 = outside diameter of socket (mm)
d_2 = diameter of spigot or inside diameter of socket (mm)
d_3 = diameter of spigot-collar (mm)
d_4 = diameter of socket-collar (mm)
a = distance from end of slot to the end of spigot on rod-B (mm)
b = mean width of cotter (mm)
c = axial distance from slot to end of socket collar (mm)
t = thickness of cotter (mm)
t_1 = thickness of spigot-collar (mm)
l = length of cotter (mm)

In order to design the cotter joint and find out the above dimensions, failures in different parts and at different cross-sections are considered. Based on each type of failure, one strength equation is written. Finally, these strength equations are used to determine various dimensions of the cotter joint.

(i) Tensile Failure of Rods Each rod of diameter d is subjected to a tensile force P. The tensile stress in the rod is given by,

$$\sigma_t = \frac{P}{\left[\frac{\pi}{4} d^2\right]}$$

or $$d = \sqrt{\frac{4P}{\pi \sigma_t}} \qquad (4.25a)$$

where σ_t is the permissible tensile stress for the rods.

(ii) Tensile Failure of Spigot Figure 4.13(a) shows the weakest cross-section at XX of the spigot end, which is subjected to tensile stress.

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

Therefore, tensile stress in the spigot is given by,

$$\sigma_t = \frac{P}{\left[\frac{\pi}{4} d_2^2 - d_2 t\right]}$$

or $$P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right] \sigma_t \qquad (4.25b)$$

From the above equation, the diameter of spigot or inner diameter of socket (d_2) can be determined by assuming a suitable value of t. The thickness of the cotter is usually determined by the following empirical relationship,

$$t = 0.31d \tag{4.25c}$$

(iii) Tensile Failure of Socket Figure 4.14(a) shows the weakest section at YY of the socket end, which is subjected to tensile stress. The area of this section is given by,

$$\text{area} = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]$$

The tensile stress at section YY is given by,

$$\sigma_t = \frac{P}{\text{area}}$$

or
$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t \tag{4.25d}$$

From the above equation, the outside diameter of socket (d_1) can be determined.

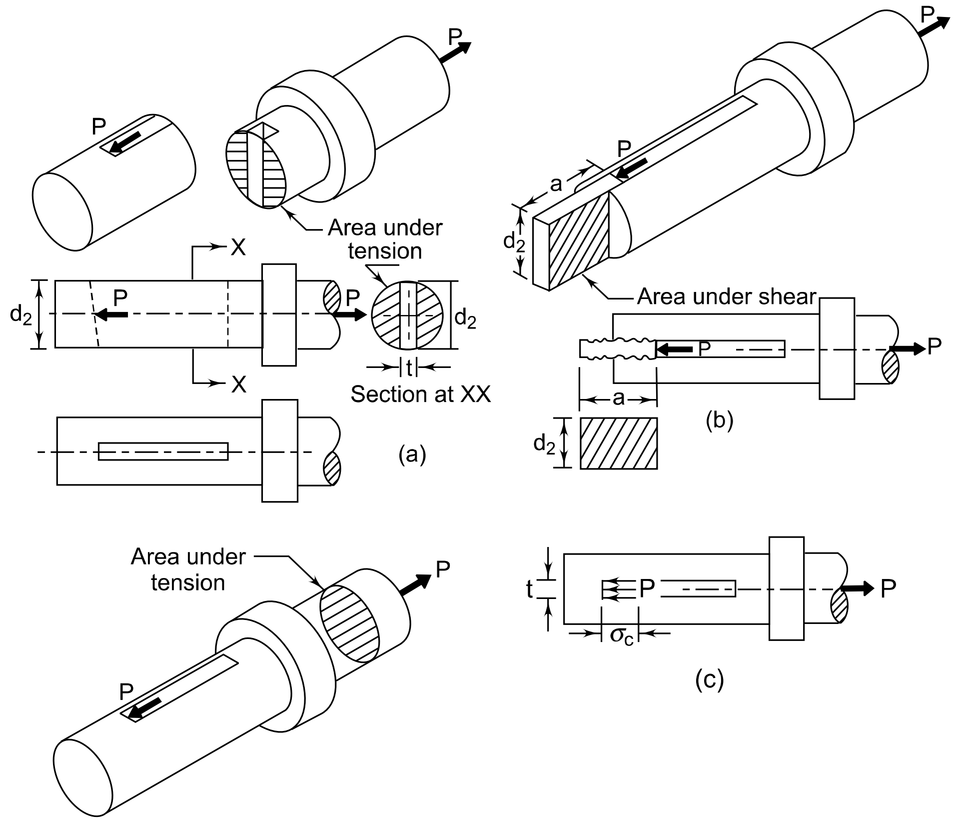


Fig. 4.13 *Stresses in Spigot End (a) Tensile Stress (b) Shear Stress (c) Compressive Stress*

(iv) Shear Failure of Cotter The cotter is subjected to double shear as illustrated in Fig. 4.15. The area of each of the two planes that resist shearing failure is (bt). Therefore, shear stress in the cotter is given by,

$$\tau = \frac{P}{2\,(bt)}$$

or
$$P = 2\,bt\tau \tag{4.25e}$$

where τ is permissible shear stress for the cotter. From Eq. (4.25e), the mean width of the cotter (b) can be determined.

(v) Shear Failure of Spigot End The spigot end is subjected to double shear as shown in Fig. 4.13(b). The area of each of the two planes that resist shear failure is (ad_2). Therefore, shear stress in the spigot end is given by,

$$\tau = \frac{P}{2\,(ad_2)}$$

or
$$P = 2\,ad_2\tau \tag{4.25f}$$

where τ is the permissible shear stress for the spigot. From Eq. (4.25f), the dimension a can be determined.

(vi) Shear Failure of Socket End The socket end is also subjected to double shear as shown in Fig. 4.14(b). The area of each of the two planes that resist shear failure is given by,

$$\text{area} = (d_4 - d_2)\ c$$

Therefore, shear stress in the socket end is given by,

$$\tau = \frac{P}{2\,(d_4 - d_2)\,c}$$

or $$P = 2\,(d_4 - d_2)\ c\ \tau \qquad (4.25g)$$

From the above equation, the dimension c can be determined.

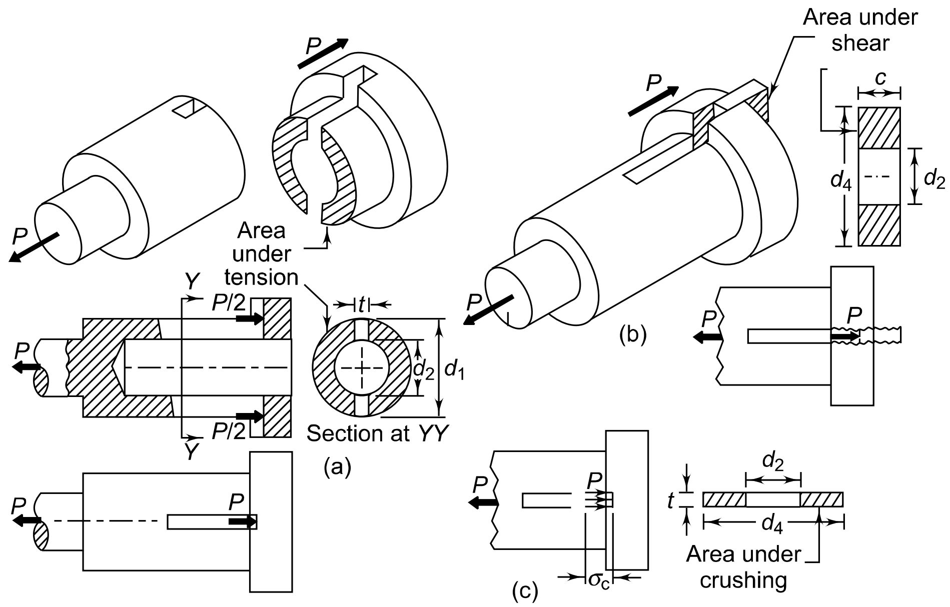


Fig. 4.14 *Stresses in Socket End (a) Tensile Stress (b) Shear stress (c) Compressive Stress*

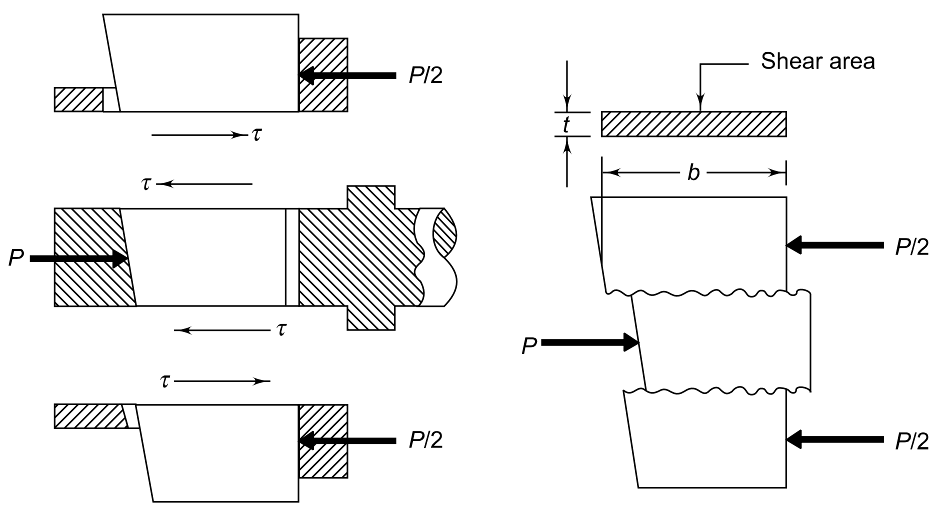


Fig. 4.15 *Shear Failure of Cotter*

(vii) Crushing Failure of Spigot End As shown in Fig. 4.13(c), the force P causes compressive stress on a narrow rectangular area of thickness t and width d_2 perpendicular to the plane of the paper. The compressive stress is given by,

$$\sigma_c = \frac{P}{t d_2} \qquad (4.25h)$$

(viii) Crushing Failure of Socket End As shown in Fig. 4.14(c), the force P causes compressive stress on a narrow rectangular area of thickness t. The other dimension of rectangle, perpendicular to the plane of paper is $(d_4 - d_2)$. Therefore, compressive stress in the socket end is given by,

$$\sigma_c = \frac{P}{(d_4 - d_2)t} \tag{4.25i}$$

(ix) Bending Failure of Cotter When the cotter is tight in the socket and spigot, it is subjected to shear stresses. When it becomes loose, bending occurs. The forces acting on the cotter are shown in Fig. 4.16(a). The force P between the cotter and spigot end is assumed as uniformly distributed over the length d_2. The force between the socket end and cotter is assumed to be varying linearly from zero to maximum with triangular distribution. The cotter is treated as beam as shown in Fig. 4.16(b). For triangular distribution,

$$x = \frac{1}{3}y = \frac{1}{3}\left(\frac{d_4 - d_2}{2}\right) = \left(\frac{d_4 - d_2}{6}\right)$$

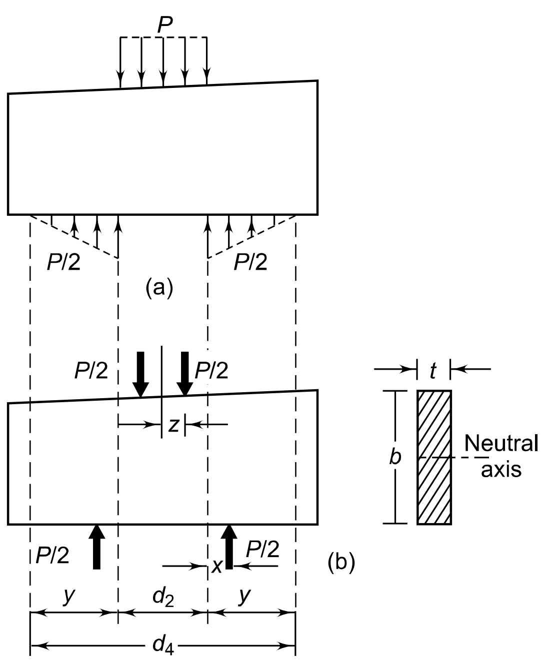


Fig. 4.16 *Cotter Treated as Beam (a) Actual Distribution of Forces (b) Simplified Diagram of Forces*

The bending moment is maximum at the centre. At the central section,

$$M_b = \frac{P}{2}\left[\frac{d_2}{2} + x\right] - \frac{P}{2}(z)$$

$$= \frac{P}{2}\left[\frac{d_2}{2} + \frac{d_4 - d_2}{6}\right] - \frac{P}{2}\left[\frac{d_2}{4}\right]$$

$$= \frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]$$

Also, $\quad I = \dfrac{tb^3}{12} \qquad y = \dfrac{b}{2}$

and $\quad \sigma_b = \dfrac{M_b\, y}{I}$

Therefore,

$$\sigma_b = \frac{\frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]\frac{b}{2}}{\left(\frac{tb^3}{12}\right)} \tag{4.25j}$$

The applications of strength equations from (4.25a) to (4.25j) in finding out the dimensions of the cotter joint are illustrated in the next example and the design project. In some cases, the dimensions of a cotter joint are calculated by using empirical relationships, without carrying out detail stress analysis. In such cases, following standard proportions can be used,

$d_1 = 1.75d$	$d_2 = 1.21d$
$d_3 = 1.5\,d$	$d_4 = 2.4\,d$
$a = c = 0.75\,d$	$b = 1.6\,d$
$t = 0.31\,d$	$t_1 = 0.45\,d$
Clearance = 1.5 to 3 mm	
Taper for cotter = 1 in 32	

4.10 DESIGN PROCEDURE FOR COTTER JOINT

The basic procedure to calculate the dimensions of the cotter joint consists of the following steps:

(i) Calculate the diameter of each rod by Eq. (4.25a),

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

(ii) Calculate the thickness of the cotter by the empirical relationship given in Eq. (4.25c),

$$t = 0.31\,d$$

(iii) Calculate the diameter d_2 of the spigot on the basis of tensile stress. From Eq. (4.25b),

$$P = \left[\frac{\pi}{4} d_2^2 - d_2\, t\right] \sigma_t$$

When the values of P, t and σ_t are substituted, the above expression becomes a quadratic equation.

(iv) Calculate the outside diameter d_1 of the socket on the basis of tensile stress in the socket, from Eq. (4.25d),

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)\, t\right] \sigma_t$$

When values of P, d_2, t and σ_t are substituted, the above expression becomes a quadratic equation.

(v) The diameter of the spigot collar d_3 and the diameter of the socket collar d_4 are calculated by the following empirical relationships,

$$d_3 = 1.5\, d$$
$$d_4 = 2.4\, d$$

(vi) The dimensions a and c are calculated by the following empirical relationship,

$$a = c = 0.75\, d$$

(vii) Calculate the width b of the cotter by shear consideration using Eq. (4.25e) and bending consideration using Eq. (4.25j) and select the width, whichever is maximum between these two values.

$$b = \frac{P}{2\tau t} \qquad \text{or} \qquad b = \sqrt{\frac{3P}{t\sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]}$$

(viii) Check the crushing and shear stresses in the spigot end by Eqs. (4.25h) and (4.25f) respectively.

$$\sigma_c = \frac{P}{td_2}$$
$$\tau = \frac{P}{2ad_2}$$

(ix) Check the crushing and shear stresses in the socket end by Eqs (4.25i) and (4.25g) respectively.

$$\sigma_c = \frac{P}{(d_4 - d_2)\, t}$$
$$\tau = \frac{P}{2\,(d_4 - d_2)\, c}$$

(x) Calculate the thickness t_1 of the spigot collar by the following empirical relationship,

$$t_1 = 0.45\, d$$

The taper of the cotter is 1 in 32.

Example 4.2 *It is required to design a cotter joint to connect two steel rods of equal diameter. Each rod is subjected to an axial tensile force of 50 kN. Design the joint and specify its main dimensions.*

Solution

Given $P = (50 \times 10^3)$ N

Part I *Selection of material*

The rods are subjected to tensile force and strength is the criterion for the selection of the rod material. The cotter is subjected to direct shear stress and bending stresses. Therefore, strength is also the criterion of material selection for the cotter. On the basis of strength, the material of the two rods and the cotter is selected as plain carbon steel of Grade 30C8 ($S_{yt} = 400$ N/mm^2).

Part II *Selection of factor of safety*

In stress analysis of the cotter joint, the following factors are neglected:

(i) initial stresses due to tightening of the cotter; and

(ii) stress concentration due to slot in the socket and the spigot ends.

To account for these factors, a higher factor of safety is used in the present design. The factor of safety for the rods, spigot end and socket end is assumed as 6, while for the cotter, it is taken as 4. There are two reasons for assuming a lower factor of safety for the cotter. They are as follows:

(i) There is no stress concentration in the cotter.

(ii) The cost of the cotter is small compared with the socket end or spigot end. If at all, a failure is going to occur, it should occur in the cotter rather than in the spigot or socket end. This is ensured by assuming a higher factor of safety for the spigot and socket ends, compared with the cotter.

It is assumed that the yield strength in compression is twice the yield strength in tension.

Part III *Calculation of permissible stresses*

The permissible stresses for rods, spigot end and socket end are as follows:

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{6} = 66.67 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{2S_{yt}}{(fs)} = \frac{2(400)}{6} = 133.33 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5\,(400)}{6}$$

$$= 33.33 \text{ N/mm}^2$$

Permissible stresses for the cotter are as follows:

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{4} = 100 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(400)}{4} = 50 \text{ N/mm}^2$$

Part IV *Calculation of dimensions*

The dimensions of the cotter joint are determined by the procedure outlined in Section 4.10.

Step I *Diameter of rods*

$$d = \sqrt{\frac{4P}{\pi\sigma_t}} = \sqrt{\frac{4(50\times10^3)}{\pi(66.67)}} = 30.90 \text{ or } 32 \text{ mm}$$

Step II *Thickness of cotter*

$$t = 0.31\,d = 0.31(32) = 9.92 \text{ or } 10 \text{ mm}$$

Step III *Diameter (d_2) of spigot*

$$P = \left[\frac{\pi}{4}d_2^2 - d_2 t\right]\sigma_t$$

$$50\times10^3 = \left[\frac{\pi}{4}d_2^2 - d_2(10)\right](66.67)$$

or $\quad d_2^2 - 12.73d_2 - 954.88 = 0$

Solving the above quadratic equation,

$$d_2 = \frac{12.73 \pm \sqrt{12.73^2 - 4(-954.88)}}{2}$$

$\therefore \quad d_2 = 37.91$ or 40 mm

Step IV *Outer diameter (d_1) of socket*

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t$$

$$50\times10^3 = \left[\frac{\pi}{4}(d_1^2 - 40^2) - (d_1 - 40)\ (10)\right](66.67)$$

or $\quad d_1^2 - 12.73d_1 - 2045.59 = 0$

Solving the above quadratic equation,

$$d_1 = \frac{12.73 \pm \sqrt{12.73^2 - 4(-2045.59)}}{2}$$

$\therefore \quad d_1 = 52.04$ or 55 mm

Step V *Diameters of spigot collar (d_3) and socket collar (d_4)*

$d_3 = 1.5d = 1.5(32) = 48$ mm

$d_4 = 2.4d = 2.4(32) = 76.8$ or 80 mm

Step VI *Dimensions a and c*

$a = c = 0.75d = 0.75(32) = 24$ mm

Step VII *Width of cotter*

$$b = \frac{P}{2\tau t} = \frac{50\times10^3}{2(50)(10)} = 50 \text{ mm} \qquad \text{(a)}$$

or

$$b = \sqrt{\frac{3P}{t\sigma_b}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]}$$

$$= \sqrt{\frac{3\ (50\times10^3)}{(10)\ (100)}\left[\frac{40}{4} + \frac{80-40}{6}\right]}$$

$$= 50 \text{ mm} \qquad \text{(b)}$$

From (a) and (b),

$b = 50$ mm

Step VIII *Check for crushing and shear stresses in spigot end*

$$\sigma_c = \frac{P}{td_2} = \frac{50\times10^3}{(10)(40)} = 125 \text{ N/mm}^2$$

$$\tau = \frac{P}{2\ ad_2} = \frac{50\times10^3}{2(24)(40)} = 26.04 \text{ N/mm}^2$$

$\therefore \quad \sigma_c < 133.33 \text{ N/mm}^2$ and $\tau < 33.33 \text{ N/mm}^2$

Step IX *Check for crushing and shear stresses in socket end*

$$\sigma_c = \frac{P}{(d_4 - d_2)\,t}$$

$$= \frac{50\times10^3}{(80-40)(10)} = 125 \text{ N/mm}^2$$

$$\tau = \frac{P}{2\,(d_4 - d_2)\,c}$$

$$= \frac{50 \times 10^3}{2(80-40)(24)} = 26.04 \text{ N/mm}^2$$

$\therefore \quad \sigma_c < 133.33$ N/mm^2 and $\tau < 33.33$ N/mm^2

The stresses induced in the spigot and the socket ends are within limits.

***Step X** Thickness of spigot collar*

$$t_1 = 0.45d = 0.45(32) = 14.4 \text{ or } 15 \text{ mm}$$

The taper for the cotter is 1 in 32.

***Part V** Dimensioned sketch of cotter joint*

The dimensions of various components of the cotter joint are shown in Fig. 4.17.

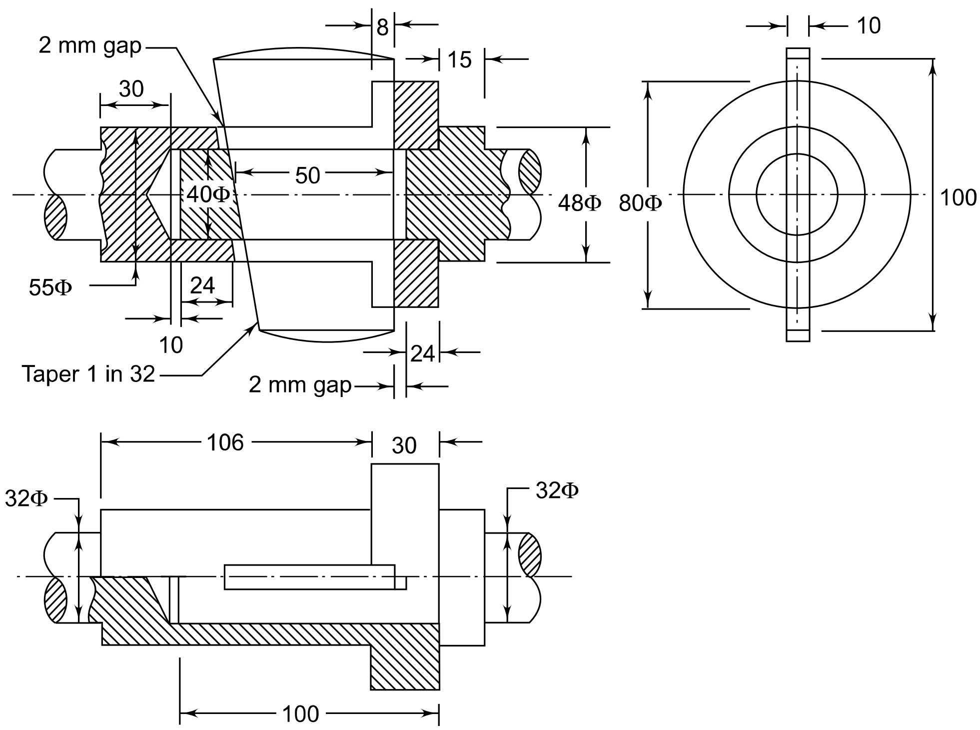


Fig. 4.17 *Dimensions of Cotter Joint*

Example 4.3 *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 N/mm^2) and the factor of safety is 4. The width of the cotter is five times of thickness. Calculate:*

(i) width and thickness of the cotter on the basis of shear failure; and

(ii) width and thickness of the cotter on the basis of bending failure.

Solution

Given $S_{yt} = 400$ N/mm^2 $(fs) = 4$

$P = (50 \times 10^3)$ N $\quad d_4 = 100$ mm $\quad d_2 = 50$ mm

***Step I** Permissible stresses for cotter*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{4} = 100 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(400)}{4} = 50 \text{ N/mm}^2$$

***Step II** Width and thickness on the basis of shear failure*

$$b = 5t$$

From Eq. (4.25e),

$P = 2bt\tau$ or $50 \times 10^3 = 2\,(5t)\,t\,(50)$

$\therefore \quad t = 10$ mm and $b = 5t = 50$ mm (i)

***Step III** Width and thickness on the basis of bending failure*

$$d_4 = 100 \text{ mm} \qquad d_2 = 50 \text{ mm}$$

From Eq. (4.25j),

$$\sigma_b = \frac{\frac{P}{2}\left[\frac{d_2}{4} + \frac{d_4 - d_2}{6}\right]\frac{b}{2}}{\left(\frac{tb^3}{12}\right)}$$

$$100 = \frac{\frac{50 \times 10^3}{2}\left[\frac{50}{4} + \frac{(100-50)}{6}\right]\frac{(5t)}{2}}{\left[\frac{t(5t)^3}{12}\right]}$$

$\therefore$ $t = 10.77$ or 12 mm and $b = 5t = 60$ mm (ii)

Example 4.4 *Two rods, made of plain carbon steel 40C8 (S_{yt} = 380 N/mm²), are to be connected by means of a cotter joint. The diameter of each rod is 50 mm and the cotter is made from a steel plate of 15 mm thickness. Calculate the dimensions of the socket end making the following assumptions:*

(i) the yield strength in compression is twice of the tensile yield strength; and

(ii) the yield strength in shear is 50% of the tensile yield strength.

The factor of safety is 6.

Solution

Given $S_{yt} = 380$ N/mm² $(fs) = 6$ $t = 15$ mm $d = 50$ mm

Step I *Permissible stresses*

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{2S_{yt}}{(fs)} = \frac{2(380)}{6} = 126.67 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5S_{yt}}{(fs)} = \frac{0.5(380)}{6} = 31.67 \text{ N/mm}^2$$

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{380}{6} = 63.33 \text{ N/mm}^2$$

Step II *Load acting on rods*

$$P = \frac{\pi}{4} d^2 \sigma_t \quad \text{or} \quad P = \frac{\pi}{4}(50)^2 (63.33)$$

$$= 124\,348.16 \text{ N}$$

Step III *Inside diameter of socket (d_2)*

From Eq. (4.25b),

$$P = \left[\frac{\pi}{4} d_2^2 - d_2 t\right]\sigma_t$$

$$124\,348.16 = \left[\frac{\pi}{4} d_2^2 - d_2(15)\right](63.33)$$

or $d_2^2 - 19.1 d_2 - 2500 = 0$

Solving the above quadratic equation,

$$d_2 = \frac{19.1 \pm \sqrt{19.1^2 - 4(-2500)}}{2}$$

$\therefore$ $d_2 = 60.45$ or 65 mm (i)

Step IV *Outside diameter of socket (d_1)*

From Eq. (4.25d),

$$P = \left[\frac{\pi}{4}(d_1^2 - d_2^2) - (d_1 - d_2)t\right]\sigma_t$$

$$124\,348.16 = \left[\frac{\pi}{4}(d_1^2 - 65^2) - (d_1 - 65)(15)\right](63.33)$$

or $d_1^2 - 19.1\, d_1 - 5483.59 = 0$

Solving the above quadratic equation,

$$d_1 = \frac{19.1 \pm \sqrt{19.1^2 - 4(-5483.59)}}{2}$$

$\therefore$ $d_1 = 84.21$ or 85 mm (ii)

Step V *Diameter of socket collar (d_4)*

From Eq. (4.25i),

$$\sigma_c = \frac{P}{(d_4 - d_2)t}$$

or $$126.67 = \frac{124\,348.16}{(d_4 - 65)(15)}$$

$\therefore$ $d_4 = 130.44$ or 135 mm (iii)

Step VI *Dimensions a and c*

From Eq. (4.25f),

$$a = \frac{P}{2 d_2 \tau} = \frac{124\,348.16}{2(65)(31.67)} = 30.20 \text{ or } 35 \text{ mm} \quad \text{(iv)}$$

From Eq. (4.25g),

$$c = \frac{P}{2(d_4 - d_2)\tau} = \frac{124\,348.16}{2(135-65)(31.67)}$$

$$= 28.04 \text{ or } 30 \text{ mm} \quad \text{(v)}$$

4.11 KNUCKLE JOINT

Knuckle joint is used to connect two rods whose axes either coincide or intersect and lie in one plane. The knuckle joint is used to transmit axial tensile force. The construction of this joint permits limited angular movement between rods, about the axis of the pin. This type of joint is popular in machines and structures. Typical applications of knuckle joints are as follows:

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

A knuckle joint is unsuitable to connect two rotating shafts, which transmit torque. The construction of a knuckle joint, used to connect two rods *A* and *B* subjected to tensile force *P*, is shown in Fig. 4.18. An eye is formed at the end of rod-*B*, while a fork is formed at the end of rod-*A*. The eye fits inside the fork and a pin passes through both the fork and the eye. This pin is secured in its place by means of a split-pin. Due to this type of construction, a knuckle joint is sometimes called a *forked-pin joint.* In rare applications, a knuckle joint is used to connect three rods—two with forks and a third with the eye.

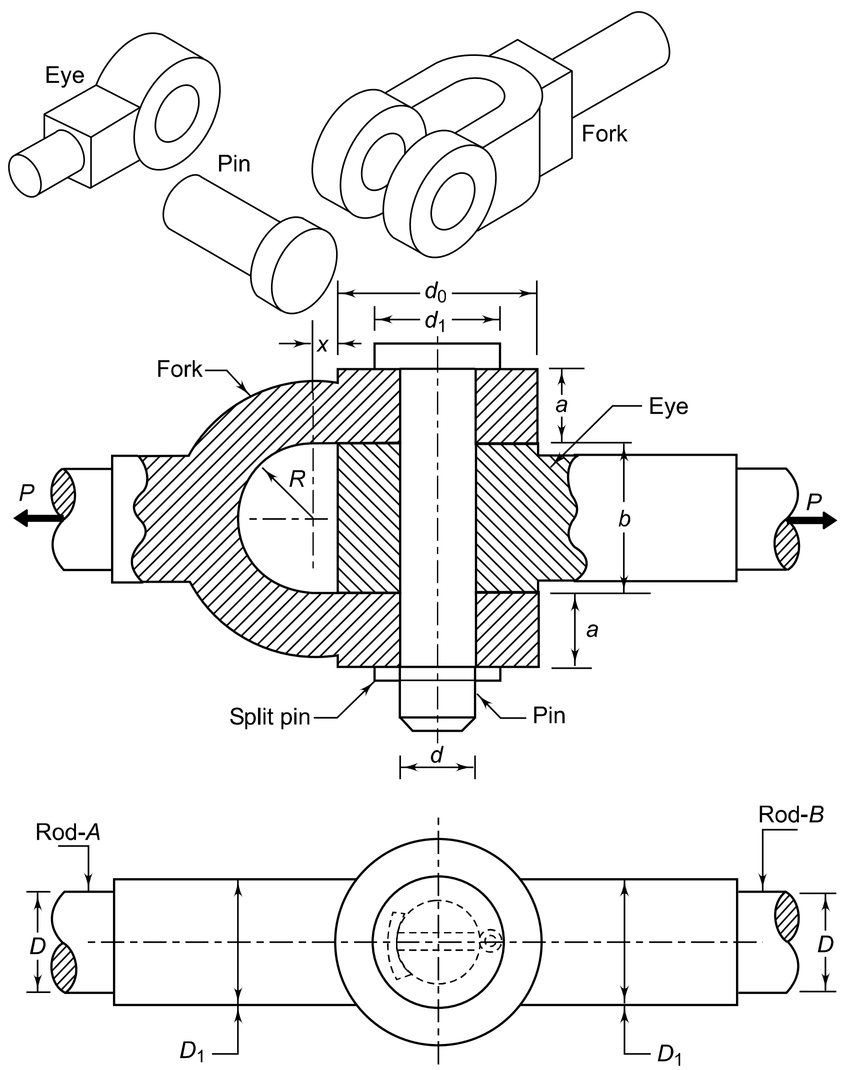


Fig. 4.18 *Knuckle Joint*

The knuckle joint offers the following advantages:

(i) The joint is simple to design and manufacture.

(ii) There are a few parts in the knuckle joint, which reduces cost and improves reliability.

(iii) The assembly or dismantling of the parts of a knuckle joint is quick and simple. The assembly consists of inserting the eye of one rod inside the fork of the other rod and putting the pin in their common hole and finally putting the split-pin to hold the pin. Dismantling consists of removing the split-pin and taking the pin out of the eye and the fork.

In Fig. 4.18, the following notations are used.

D = diameter of each rod (mm)
D_1 = enlarged diameter of each rod (mm)
d = diameter of knuckle pin (mm)
d_0 = outside diameter of eye or fork (mm)
a = thickness of each eye of fork (mm)
b = thickness of eye end of rod-B (mm)
d_1 = diameter of pin head (mm)
x = distance of the centre of fork radius R from the eye (mm)

For the purpose of stress analysis of a knuckle joint, the following assumptions are made:

(i) The rods are subjected to axial tensile force.

(ii) The effect of stress concentration due to holes is neglected.

(iii) The force is uniformly distributed in various parts.

Free body diagram of forces acting on three components of the knuckle joint, viz., fork, pin and eye is shown in Fig. 4.19. This diagram is constructed by using the principle that actions and reactions are equal and opposite. The forces are determined in the following way,

(i) Consider rod-A with the fork end. The rod is subjected to a horizontal force P to the left. The sum of all horizontal forces acting on rod-A must be equal to zero. Therefore, there should be a force P to the right acting on the fork end. The force P is divided into two parts, each equal to (P/2) on the fork end.

(ii) Consider rod-B with the eye end. The rod is subjected to a horizontal force P to the right side. The sum of all horizontal forces acting on rod-B must be equal to zero. Therefore, there should be a force P to the left acting on the eye end.

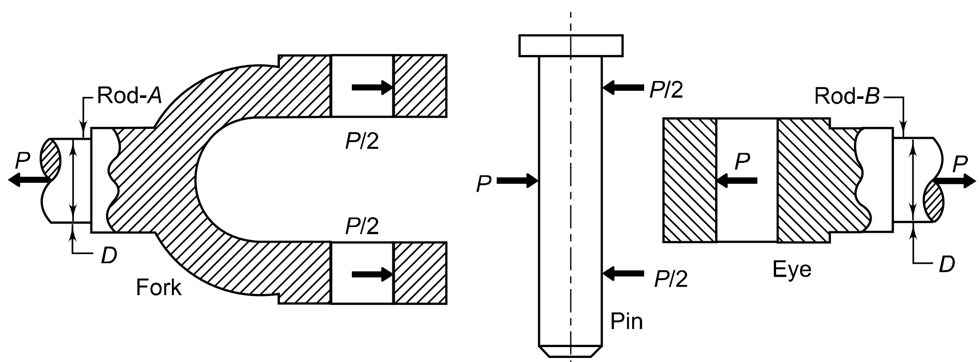


Fig. 4.19 *Free Body Diagram of Forces*

(iii) The forces shown on the pin are equal and opposite reactions of forces acting on the fork end of rod-A and the eye end of rod-B.

In order to find out various dimensions of the parts of a knuckle joint, failures in different parts and at different cross-sections are considered. For each type of failure, one strength equation is written. Finally, these strength equations are used to find out various dimensions of the knuckle joint.

(i) Tensile Failure of Rods Each rod is subjected to a tensile force P. The tensile stress in the rod is given by,

$$\sigma_t = \frac{P}{\left(\dfrac{\pi}{4}D^2\right)} \quad \text{or} \quad D = \sqrt{\frac{4P}{\pi\,\sigma_t}} \tag{4.26a}$$

where σ_t is the permissible tensile stress for the rods. The enlarged diameter D_1 of the rod near the joint is determined by the following empirical relationship,

$$D_1 = 1.1\,D \tag{4.26b}$$

(ii) Shear Failure of Pin The pin is subjected to double shear as shown in Fig. 4.20. The area of each of the two planes that resist shear failure is $\left(\dfrac{\pi}{4}d^2\right)$. Therefore, shear stress in the pin is given by,

$$\tau = \frac{P}{2\left(\dfrac{\pi}{4}d^2\right)} \quad \text{or} \quad d = \sqrt{\frac{2P}{\pi\,\tau}} \tag{4.26c}$$

where τ is the permissible shear stress for the pin. The standard proportion for the diameter of the pin is as follows,

$$d = D \tag{4.26d}$$

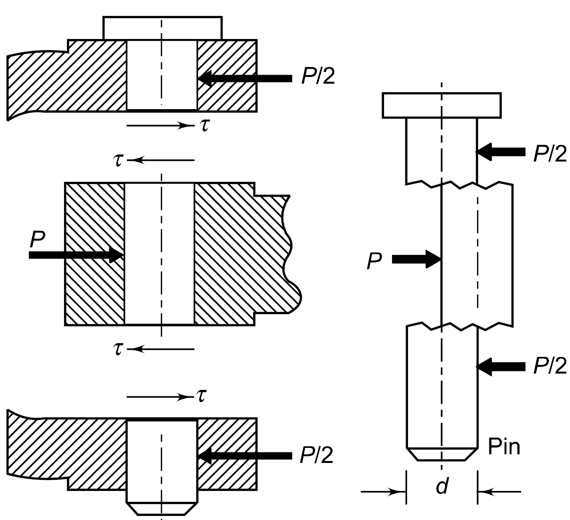


Fig. 4.20 *Shear Failure of Pin*

(iii) Crushing Failure of Pin in Eye When a cylindrical surface such as a pin is subjected to a force along its periphery, its projected area is taken into consideration to find out the stress. As shown in Fig. 4.21, the projected area of the cylindrical surface is ($l \times d$) and the compressive stress is given by,

$$\sigma_c = \frac{\text{force}}{\text{projected area}} = \frac{P}{(l \times d)}$$

As shown in Fig. 4.18, the projected area of the pin in the eye is (bd) and the compressive stress between the pin and the eye is given by,

$$\sigma_c = \frac{P}{bd} \tag{4.26e}$$

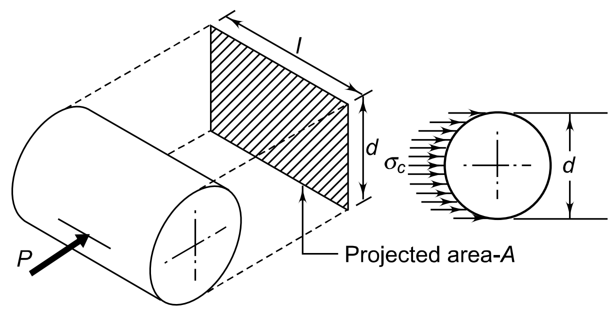


Fig. 4.21 *Projected Area of Cylindrical Surface*

(iv) Crushing Failure of Pin in Fork As shown in Fig. 4.18, the total projected area of the pin in the fork is ($2ad$) and the compressive stress between the pin and the fork is given by,

$$\sigma_c = \frac{P}{2ad} \tag{4.26f}$$

(v) Bending Failure of Pin When the pin is tight in the eye and the fork, failure occurs due to shear. On the other hand, when the pin is loose, it is subjected to bending moment as shown in Fig. 4.22. It is assumed that the load acting on the pin is uniformly distributed in the eye, but uniformly varying in two parts of the fork. For triangular distribution of load between the pin and the fork,

$$x = \frac{1}{3}a \qquad \text{also,} \qquad z = \frac{1}{2}\left(\frac{1}{2}b\right) = \frac{1}{4}b$$

The bending moment is maximum at the centre. It is given by,

$$M_b = \frac{P}{2}\left[\frac{b}{2} + x\right] - \frac{P}{2}(z)$$

$$= \frac{P}{2}\left[\frac{b}{2} + \frac{a}{3}\right] - \frac{P}{2}\left[\frac{b}{4}\right] = \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]$$

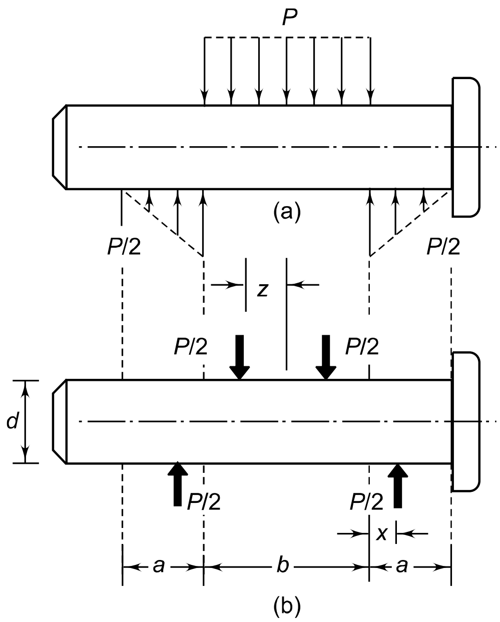


Fig. 4.22 *Pin Treated as Beam (a) Actual Distribution of Forces (b) Simplified Diagram of Forces*

Also, $I = \dfrac{\pi d^4}{64}$ and $y = \dfrac{d}{2}$

From Eq. (4.12),

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

or $$\sigma_b = \frac{32}{\pi d^3} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right] \qquad (4.26g)$$

(vi) Tensile Failure of Eye Section XX shown in Fig. 4.23(a) is the weakest section of the eye. The area of this section is given by,

$$\text{area} = b\,(d_0 - d)$$

The tensile stress at section XX is given by,

$$\sigma_t = \frac{P}{\text{area}} \quad \text{or} \quad \sigma_t = \frac{P}{b\,(d_0 - d)} \qquad (4.26h)$$

(vii) Shear Failure of Eye The eye is subjected to double shear as shown in Fig. 4.23(b). The area of each of the two planes resisting the shear failure is

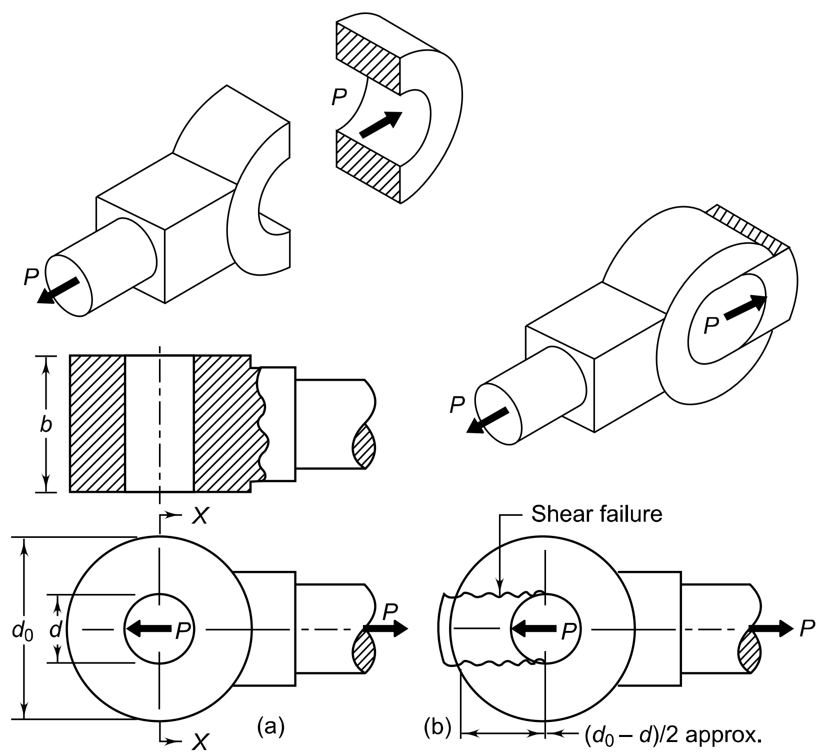


Fig. 4.23 *(a) Tensile Failure of Eye (b) Shear Failure of Eye*

$[b\,(d_0 - d)/2]$ approximately. Therefore, shear stress is given by,

$$\tau = \frac{P}{2\,[b(d_0 - d)/2]}$$

or $$\tau = \frac{P}{b\,(d_0 - d)} \qquad (4.26i)$$

Standard proportion for outside diameter of the eye or the fork is given by the following relationship,

$$d_0 = 2d \qquad (4.26j)$$

(viii) Tensile Failure of Fork Fork is a double eye and as such, Fig. 4.23 is applicable to a fork except for dimension b which can be modified as $2a$ in case of a fork. The area of the weakest section resisting tensile failure is given by

$$\text{area} = 2a\,(d_0 - d)$$

Tensile stress in the fork is given by

$$\sigma_t = \frac{P}{2a\,(d_0 - d)} \qquad (4.26k)$$

(ix) Shear Failure of Fork Each of the two parts of the fork is subjected to double shear. Modifying Eq. (4.26i),

$$\tau = \frac{P}{2a\,(d_0 - d)} \qquad (4.26l)$$

Standard proportions for the dimensions a and b are as follows,

$$a = 0.75\,D \qquad (4.26m)$$
$$b = 1.25\,D \qquad (4.26n)$$

The diameter of the pinhead is taken as,

$$d_1 = 1.5\,d \qquad (4.26o)$$

The gap x shown in Fig. 4.18 is usually taken as 10 mm.

$$\therefore \quad x = 10 \text{ mm} \qquad (4.26p)$$

The applications of strength equations from (4.26a) to (4.26l) in finding out the dimensions of the knuckle joint are illustrated in the next example. The eye and the fork are usually made by the forging process and the pin is machined from rolled steel bars.

4.12 DESIGN PROCEDURE FOR KNUCKLE JOINT

The basic procedure to determine the dimensions of the knuckle joint consists of the following steps:

(i) Calculate the diameter of each rod by Eq. (4.26a).

$$D = \sqrt{\frac{4P}{\pi \sigma_t}}$$

(ii) Calculate the enlarged diameter of each rod by empirical relationship using Eq. (4.26b).

$$D_1 = 1.1\,D$$

(iii) Calculate the dimensions a and b by empirical relationship using Eqs (4.26m) and (4.26n).

$$a = 0.75\,D \qquad b = 1.25\,D$$

(iv) Calculate the diameters of the pin by shear consideration using Eq. (4.26c) and bending consideration using Eq. (4.26g) and select the diameter, whichever is maximum.

$$d = \sqrt{\frac{2P}{\pi \tau}} \quad \text{or} \quad d = \sqrt[3]{\frac{32}{\pi \sigma_b} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

(whichever is maximum)

(v) Calculate the dimensions d_o and d_1 by empirical relationships using Eqs (4.26j) and (4.26o) respectively.

$$d_o = 2d \qquad d_1 = 1.5d$$

(vi) Check the tensile, crushing and shear stresses in the eye by Eqs (4.26h), (4.26e) and (4.26i) respectively.

$$\sigma_t = \frac{P}{b\,(d_0 - d)}$$

$$\sigma_c = \frac{P}{bd}$$

$$\tau = \frac{P}{b\,(d_0 - d)}$$

(vii) Check the tensile, crushing and shear stresses in the fork by Eqs (4.26k), (4.26f) and (4.26l) respectively.

$$\sigma_t = \frac{P}{2a(d_0 - d)}$$

$$\sigma_c = \frac{P}{2ad}$$

$$\tau = \frac{P}{2a(d_0 - d)}$$

The application of the above mentioned procedure is illustrated in the next example.

Example 4.5 *It is required to design a knuckle joint to connect two circular rods subjected to an axial tensile force of 50 kN. The rods are co-axial and a small amount of angular movement between their axes is permissible. Design the joint and specify the dimensions of its components. Select suitable materials for the parts.*

Solution

Given $P = (50 \times 10^3)$ N

Part I *Selection of material*

The rods are subjected to tensile force. Therefore, yield strength is the criterion for the selection of material for the rods. The pin is subjected to shear stress and bending stresses. Therefore, strength is also the criterion of material selection for the pin. On strength basis, the material for two rods and pin is selected as plain carbon steel of Grade 30C8 (S_{yt} = 400 N/mm^2). It is further assumed that the yield strength in compression is equal to yield strength in tension. In practice, the compressive strength of steel is much higher than its tensile strength.

Part II *Selection of factor of safety*

In stress analysis of knuckle joint, the effect of stress concentration is neglected. To account for this effect, a higher factor of safety of 5 is assumed in the present design.

Part III *Calculation of permissible stresses*

$$\sigma_t = \frac{S_{yt}}{(fs)} = \frac{400}{5} = 80 \text{ N/mm}^2$$

$$\sigma_c = \frac{S_{yc}}{(fs)} = \frac{S_{yt}}{(fs)} = \frac{400}{5} = 80 \text{ N/mm}^2$$

$$\tau = \frac{S_{sy}}{(fs)} = \frac{0.5\,S_{yt}}{(fs)} = \frac{0.5(400)}{5} = 40 \text{ N/mm}^2$$

Part IV *Calculation of dimensions*

The dimensions of the knuckle joint are calculated by the procedure outlined in Section 4.10.

Step I *Diameter of rods*

$$D = \sqrt{\frac{4P}{\pi\sigma_t}} = \sqrt{\frac{4(50 \times 10^3)}{\pi\,(80)}} = 28.21 \text{ or } 30 \text{ mm}$$

Step II *Enlarged diameter of rods (D_1)*

$$D_1 = 1.1\,D = 1.1(30) = 33 \text{ or } 35 \text{ mm}$$

Step III *Dimensions a and b*

$$a = 0.75\,D = 0.75(30) = 22.5 \text{ or } 25 \text{ mm}$$
$$b = 1.25\,D = 1.25(30) = 37.5 \text{ or } 40 \text{ mm}$$

Step IV *Diameter of pin*

$$d = \sqrt{\frac{2P}{\pi\,\tau}} = \sqrt{\frac{2\,(50 \times 10^3)}{\pi\,(40)}} = 28.21 \text{ or } 30 \text{ mm}$$

Also,

$$d = \sqrt[3]{\frac{32}{\pi\,\sigma_b} \times \frac{P}{2}\left[\frac{b}{4} + \frac{a}{3}\right]}$$

$$= \sqrt[3]{\frac{32}{\pi\,(80)} \times \frac{(50 \times 10^3)}{2}\left[\frac{40}{4} + \frac{25}{3}\right]}$$

$$= 38.79 \text{ or } 40 \text{ mm}$$

$\therefore \quad d = 40$ mm

Step V *Dimensions d_0 and d_1*

$$d_0 = 2d = 2(40) = 80 \text{ mm}$$
$$d_1 = 1.5d = 1.5(40) = 60 \text{ mm}$$

Step VI *Check for stresses in eye*

$$\sigma_t = \frac{P}{b\,(d_0 - d)} = \frac{(50 \times 10^3)}{40\,(80 - 40)} = 31.25 \text{ N/mm}^2$$

$\therefore \quad \sigma_t < 80$ N/mm^2

$$\sigma_c = \frac{P}{b\,d} = \frac{(50 \times 10^3)}{40\,(40)} = 31.25 \text{ N/mm}^2$$

$\therefore \quad \sigma_c < 80$ N/mm^2

$$\tau = \frac{P}{b\,(d_0 - d)} = \frac{(50 \times 10^3)}{40(80 - 40)} = 31.25 \text{ N/mm}^2$$

$\therefore \quad \tau < 40$ N/mm^2

Step VII *Check for stresses in fork*

$$\sigma_t = \frac{P}{2a\,(d_0 - d)} = \frac{(50 \times 10^3)}{2(25)(80 - 40)} = 25 \text{ N/mm}^2$$

$\therefore \quad \sigma_t < 80$ N/mm^2

$$\sigma_c = \frac{P}{2ad} = \frac{(50 \times 10^3)}{2(25)\,(40)} = 25 \text{N/mm}^2$$

$\therefore \quad \sigma_c < 80$ N/mm^2

$$\tau = \frac{P}{2a\,(d_0 - d)}$$

$$= \frac{(50 \times 10^3)}{2(25)\,(80 - 40)} = 25 \text{N/mm}^2$$

$\therefore \quad \tau < 40$ N/mm^2

It is observed that stresses are within limits.

Part V *Dimensioned sketch of knuckle joint*
Main dimensions of the knuckle joint are shown in Fig. 4.24.

Example 4.6 *A wall-rack, used to store round steel bars, consists of two I-section cantilever beams fixed in the wall. The bars are stacked in a triangular fashion as shown in Fig. 4.25(a). The total weight of the bars is 75 kN. The permissible bending stress for the cantilevers is 165 N/mm^2. Select a standard rolled I-section beam from the following table:*

Designation	*b (mm)*	*h (mm)*	I_{xx} *(mm^4)*
ISLB 150	80	150	688.2×10^4
ISLB 175	90	175	1096.2×10^4
ISLB 200	100	200	1696.6×10^4
ISLB 225	100	225	2501.9×10^4
ISLB 250	125	250	3717.8×10^4

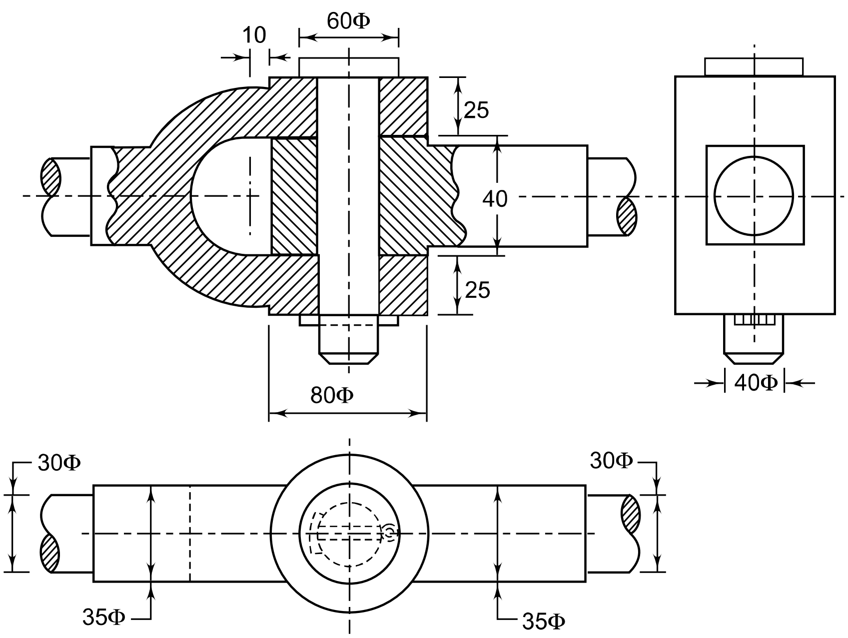


Fig 4.24 *Dimensions of Knuckle Joint*

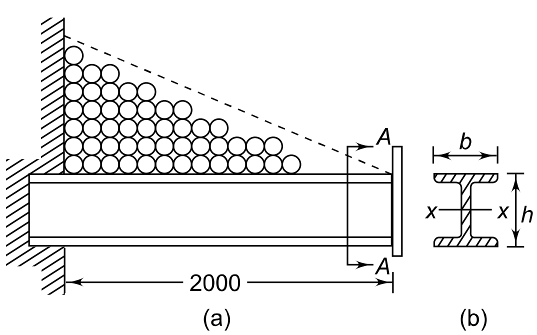


Fig 4.25

Solution

Given $W = 75$ kN $\sigma_b = 165$ N/mm^2

Step I *Calculation of bending moment*
There are two cantilever beams and the load supported by each beam is (75/2) or 37.5 kN. For a triangular load distribution, the centre of gravity of the resultant load is at a distance of (2000/3) mm from the wall. Therefore,

$$M_b = (37.5 \times 10^3)\left(\frac{2000}{3}\right) = 25 \times 10^6 \text{ N-mm}$$

Step II *Calculation of* (I_{xx}/y)
From Eq. (4.12),

$$\frac{I_{xx}}{y} = \frac{M_b}{\sigma_b} = \frac{25 \times 10^6}{165} = 151.51 \times 10^3 \text{ mm}^3$$

Step III *Selection of beam*
The cross-section of the beam is shown in Fig. 4.25 (b), ($y = h/2$)

Couplings:— The coupling can be defined as a mechanical device that permanently joints two rotating shafts to each other.

The shaft to be connected by the coupling may have collinear axes, intersecting axes or parallel axes with a small distance in between. Oldham's Coupling is used to connect two ~~co~~ ~~axial~~ parallel shafts when they are at a small distance apart.

1) Rigid flange Coupling: (Protected Type)

A flange coupling consists of two flanges — one keyed to the driving shaft and other to the driven shaft. The two flanges are connected together by means of four or six bolts arranged on a circle concentric with the axes of the shafts.

Power is transmitted from the driving shaft to left side flange ~~of~~ through the key. It is then transmitted from the left side flange to right side flange through the bolts. finally power is transmitted from right side flange to driven shaft throug the key.

Design Procedure for Rigid flange Coupling:

Step I Shaft diameter: Calculate the shaft diameter by using the following equations:

40C8 ⇓ $S_{yt} = 380$ Material for shaft

$$M_t = \frac{60P}{2\pi N} \quad \text{—— (1)}$$

where P = power transmitted

M_t = torque transmitted by shaft

&

$$\tau = \frac{16 M_t}{\pi d^3} \quad \text{—— (2)}$$

$\boxed{d = ?}$

$\boxed{FOS = 2 \text{ for shaft}}$

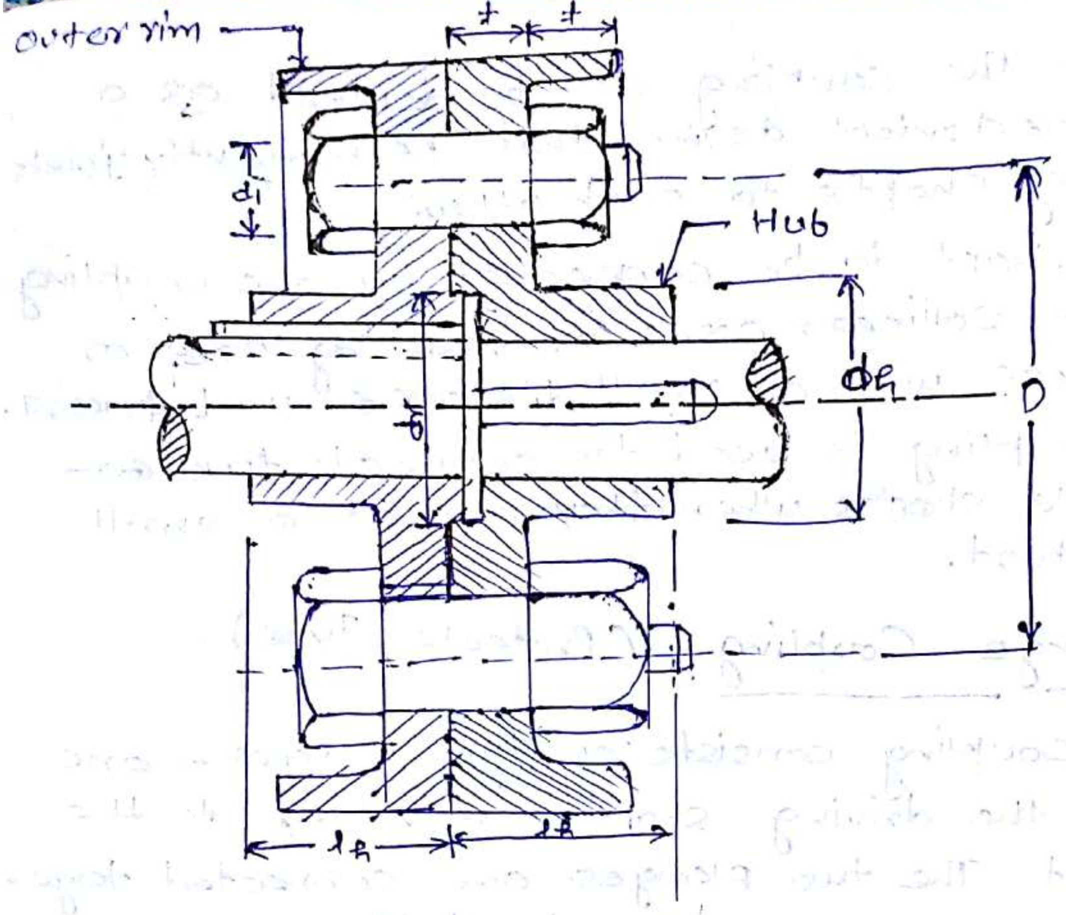
Step II:— Dimensions of flanges:

Empirical relation

⇒ Hub diameter (d_h) = 2d

⇒ length of hub or effective length of key.

$l_h = 1.5d$



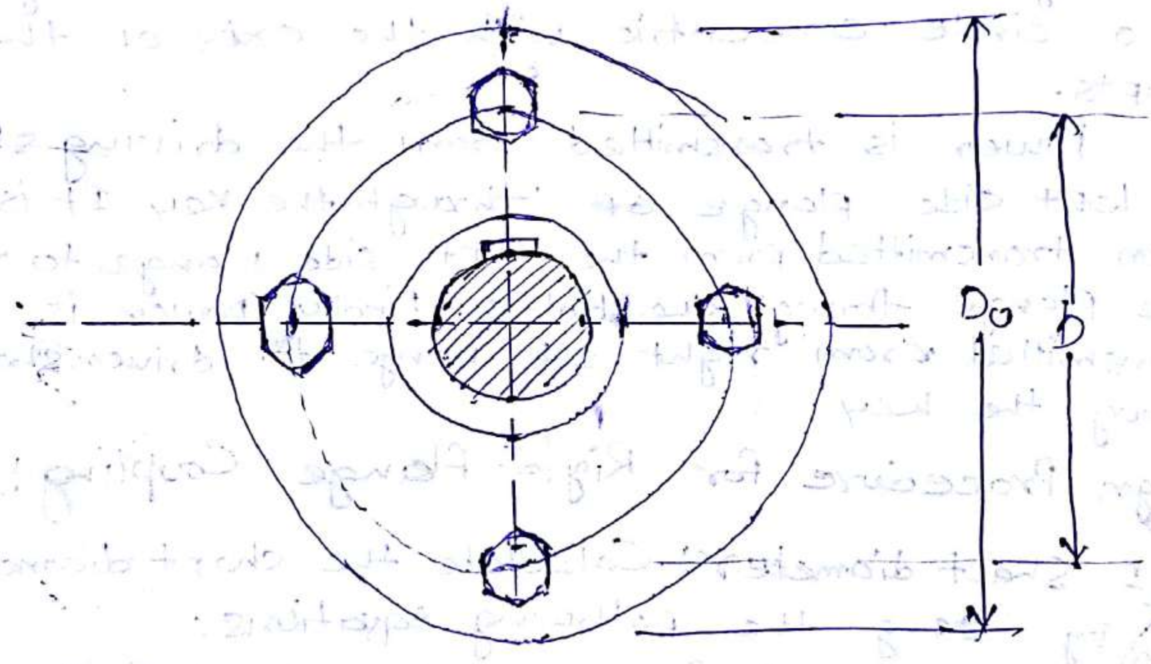


Fig1: Proportion of Rigid Coupling

- pitch circle diameter of bolts $(D) = 3d$.
- thickness of flanges $(t) = 0.5\,d$
- thickness of protecting rim $(t_1) = 0.25\,d$.
- diamer of spigot and recess $(d_r) = 1.5\,d$.
- Outside diameter of flange $(D_o) = (4d + 2t_1)$

No. of bolts (N) is also decided from the shaft diameter in the following ways:

$N = 3$ for $d < 40$

$N = 4$ for $40 < d < 100$

$N = 6$ for $100 < d < 180$

Step III ~~Hub dian~~ Torsional - Shear Stress in Hub :—

$$\tau = \frac{M_t r}{J} \quad \text{where } J = \frac{\pi(d_h^4 - d^4)}{32}$$

$$\& \; r = \frac{d_h}{2}$$

$$\boxed{\tau = \checkmark}$$

Check that τ_{ind} is less than τ permissible for Hub.

$$\boxed{\tau \le \tau_{permissible}}$$

Assume :—
Material for Hub & Flange : Grey Cast Iron FG 200 ($S_{ut} = 200 \, N/mm^2$)

then $\tau_{per} = \frac{S_{us}}{fos} = \frac{0.5 S_{ut}}{FOS}$ (if fos is given)

FOS → 6 for Flange & HUB

The flanges at junction of the hub is under shear while transmitting the torsional moment M_t.

from fig. area under shear $= (\pi d_h) \times t$

Shear Force $=$ Area $\times \tau$

$= \pi d_h t \times \tau$

Resisting torque $M_t =$ Shear Force $\times \frac{d_h}{2}$

$$\boxed{M_t = \frac{\pi d_h^2 t \tau}{2}}$$

$$\therefore (\tau)_{flange} = \frac{2 M_t}{\pi d_h^2 t}$$

$\boxed{\tau_{flange} \le \tau_{permi}}$ for flange.

Step IV :— Diameter of bolts :—
(By assuming bolts fitted in reamed and grounded hole).

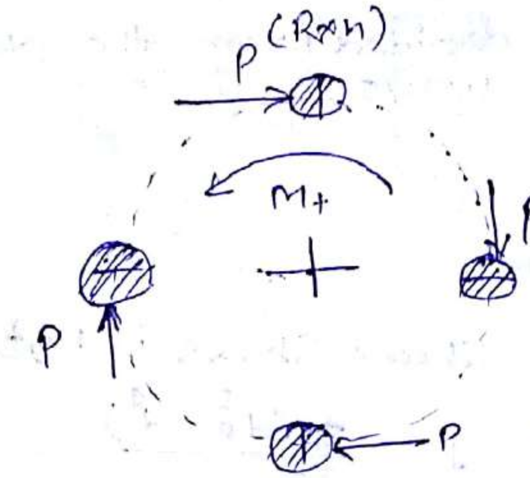


Fig: Shear resistance of Bolts

$(M_t) = (M_t)_r$

$M_t = \left(P \times \frac{D}{2}\right) \times N$

where
$(M_t)_r$ = resisting twisting moment

The direct shear stress on one bolt.

$(\tau)_{bolt} = \frac{P}{\frac{\pi}{4} d_1^2}$

d_1 = diameter of bolt. (nominal)

$\Rightarrow \boxed{(\tau)_{bolt} = \frac{2 M_t}{\frac{\pi}{4} d_1^2 (DN)}}$

obtained by material property

$d_1 = ?$

τ permissible for bolt is given or assumed that material of bolt is Plain Carbon steel 30C8 ($S_{yt} = 400$ N/mm^2)

FOS = 2.5

Step V :- Compressive Stress in the bolt :

As shown in fig 1. bolt area under flange is. $(d_1 \times t)$ for crushing.
(Projected area)

For N no. of bolt crushing area = $N d_1 t$.

Compressive force = $N d_1 t \sigma_c$

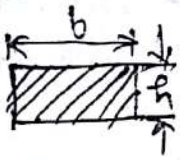
torque $M_t = (N d_1 t \sigma_c) \times \frac{D}{2}$

$\sigma_c = \frac{2 M_t}{N d_1 t D}$

$\boxed{\sigma_c \le (\sigma_{per})_{crushing}}$

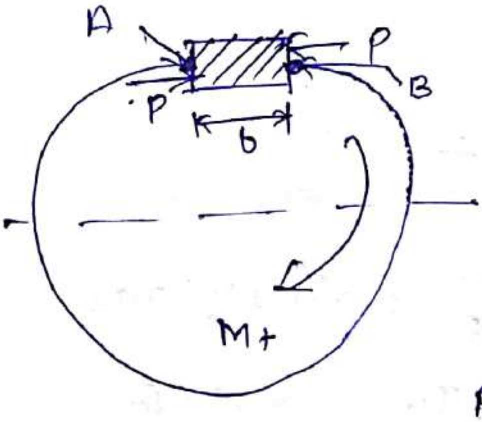
where
(σ_{per}) for crushing $= \frac{1.5 S_{yt}}{FOS}$

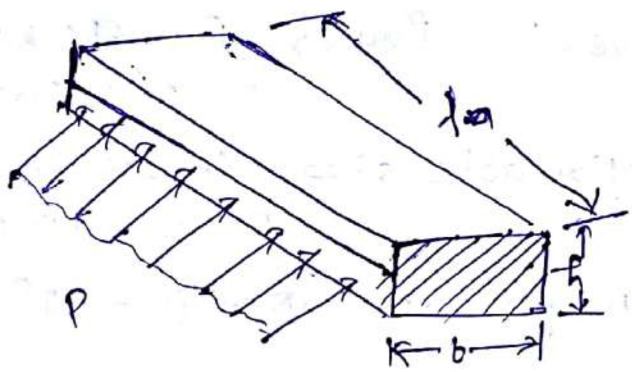
Step VI :— Dimensions of keys



b×h obtained from table for a range of shaft diameter. specific dimension (b×h) of key is given in table.

& length of key(l) is equal to $l_h = 1.5d$.





Area for shear = $b \times l$

Area for crushing = $\left(\frac{h}{2} \times l\right)$

$$\therefore \quad \tau = \frac{P}{bl} \quad \text{—(1)}$$

$$\& \quad \sigma_c = \frac{2P}{hl} \quad \text{—(2)}$$

where $\boxed{P = \left(\frac{M_t}{d/2}\right) = \frac{2M_t}{d}}$

put the value of P in Expression (1) & (2)

$$\boxed{\tau = \frac{2M_t}{dbl} \quad \& \quad \sigma_c = \frac{4M_t}{dhl}}$$

$$\boxed{(\tau)_{key} \leq (\tau_{per})_{key}} \quad \& \quad \boxed{(\sigma_c)_{perm} \geq (\sigma_c)_{key}}$$

Q.1. Design and draw a cast iron flange coupling for a mild steel shaft transmitting 90 KW at 250 rpm. The allowable shear stress in the shaft is 40 MPa, and the angle of twist is not to exceed 1° in a length of 20 diameters. The allowable shear stress in the coupling bolt is 30 MPa.

Solution :—

Given: Power $P = 90\ KW$

$N = 250\ rpm$

Allowable shear Stress in shaft

$\tau_s = 40\ MPa.$

Angle of twist $\theta = 1^\circ = \frac{\pi}{180} = 0.0175\ rad$

Allowable shear stress in bolts $\tau_b = 30\ MPa.$

Step I: Diamete of shaft

$$P = \frac{2\pi N M_t}{60}$$

$$\therefore M_t = \frac{P \times 60}{2\pi N} = \frac{90 \times 10^3 \times 60}{2\pi \times 250}$$

$$= 3440\ N\text{-}m = 3440 \times 10^3\ N\text{-}mm.$$

We know that Torsion Expression.

$$\frac{T}{J} = \frac{\tau_s}{r} = \frac{G\theta}{l}$$

By using $\frac{T}{J} = \frac{\tau}{r}$ we calculate diameter of shaft on strength basis.

By putting all expression

$$\tau_s = \frac{16T}{\pi d^3}$$

$$40 = \frac{16 \times 3440 \times 10^3}{\pi d^3}$$

$$d = 75.94 \simeq 76\ mm$$

By considering,

$$\frac{T}{J} = \frac{G\theta}{l}$$ we can calculate the diameter on rigidity basis.

$$\frac{3990 \times 10^3}{\frac{\pi}{32} d^4} = \frac{80 \times 10^3 \times 0.0175}{20d}$$

$\left\{ \begin{array}{l} l = 20d \text{ (given)} \\ G = 80 \text{ GPa (by material property for steel)} \\ \quad = 80 \times 10^3 \text{ MPa} \end{array} \right.$

$d = 78$ mm

Taking the larger of the two values, we have

$d = 78$ mm

~~Design for hub!~~

Step II : Dimensions of flanges!

Empiral relations

Hub diameter $d_h = 2d = 2 \times 78 = 156$ mm.

and length of hub, or effective length of key.

$l_h = 1.5d = 1.5 \times 78 = 117$ mm.

pitch circle diameter of bolts $D = 3d = 234$ mm

thickness of flanges $(t) = 0.5d = 39$ mm.

thickness of protecting rim $(t_1) = 0.25d = 19.5 \simeq 20$ mm

diameter of spigot and recess $(d_r) = 1.5d$
$= 117$ mm

Outside diameter of flange.

$D_o = (4d + 2t_1) = 4 \times 78 + 2 \times 20$
$= 352$ mm

No. of bolts $N = 4$ for 78 mm diameter shaft

Step III : Torsional Stresses in Hub :

Assume material for Hub and flanges is ~~FG~~ Grey cast iron FG 200 ($S_{ut} = 200$ N/mm²)

~~the~~ and factor of safety ~~of~~ for Hub and flanges is 6.

$\therefore S_{su} = 0.5 S_{ut} = 100$ N/mm²

and $\tau_{permissible} = \frac{S_{su}}{f.o.s} = \frac{100}{6} = 16.67$ N/mm²

$\tau_h = \frac{M_t r}{J}$ where $J = \frac{\pi}{32}(d_h^4 - d^4) = 54509169.38\ mm^4$

& $r = \frac{d_h}{2} = \frac{156}{2} = 78\ mm$

$$\tau_h = \frac{3440 \times 10^3 \times 78}{54509169.38} = 4.92\ N/mm^2$$

$$\boxed{\tau_h \le \tau_{permissible}}$$

Therefore design of the hub is safe.

⇒ The flanges at the junction of hub is under shear while transmitting the torsional moment M_t.

$$\tau_{flange} = \frac{2M_t}{\pi d_h^2 t}$$

$$= \frac{2 \times 3440 \times 10^3}{\pi \times (156)^2 \times 39}$$

$$= 2.307\ N/mm^2$$

$$\boxed{\tau_{flange} \le \tau_{permissible}}\ \cancel{(16\ N/mm^2)}$$

There fore design of the flange is safe.

Step IV: Diameter of bolts:

It is assumed that bolts are fitted in reamed and Ground hole.

~~Assume material for the bolt is plain carbon steel: 30C8 ($S_{yt} = 400\ N/mm^2$) and factor of safety is 2.5.~~

~~$(\tau_b)_{per.} = \frac{S_{sy}}{f.s} = \frac{0.5 S_{yt}}{(f.s)} = \frac{0.5 \times 400}{2.5} = 80\ MPa.$~~

τ_b is given $\tau_b = 30\ MPa$

$$\tau_b = \frac{2M_t}{\frac{\pi}{4} d_1^2 (DN)} \qquad 30 = \frac{8 \times 3440 \times 10^3}{\pi \times d_1^2 \times 234 \times 4}$$

$$\underline{d_1 = 17.66\ mm}$$

Step V: Compressive strength of bolts

$\because$ Allowable shear stress for bolts is given in Question so we can not assume material for bolt.

$\Rightarrow$ There fore neglect this step.

you have not to write this step in solution.

Step VI: Dimensions of key

from table given in data book, the standard Cross - Section of the flat key for a 78 mm diameter shaft is 22 × 14 mm (b × h)

and length of key $l = l_h = 117$ mm

The dimensions of key = 22 × 14 × 117 mm.

Assuming that shaft & key are of same material.

induced shear stress in key.

$$\tau = \frac{2 M_t}{d b l} = \frac{2 \times 3440 \times 10^3}{78 \times 22 \times 117}$$

$$= 34.26 \text{ N/mm}^2.$$

$$\boxed{\tau_{ind} \le (\tau_{per})}$$

$\tau_{per} = 40$ N/mm² given for shaft

There fore the key is safe.

Q.2 It is required to design a rigid flange coupling to connect two shafts. The input shaft transmit 37.5 kW power at 180 rpm. to output shaft through coupling. The service factor for the application is 1.5. Select suitable material for various parts of the coupling Design the coupling and specify the dimensions of its components.

Here, design torque = (service factor) × (M_t)
↓
rated torque

$$P = \frac{2\pi N M_t}{60}$$

M_t ✓

then $\boxed{(M_t)_d = 1.5 \times M_t}$

and use $(M_t)_d$ in designing the coupling.

Here angle of twist is not given then. we ~~find~~ will find the diameter of shaft only on strength basis.

$$\boxed{\frac{M_t}{J} = \frac{\tau}{R}} \Rightarrow \tau = \frac{16 M_t}{\pi d^3}$$

Bushed - Pin flexible Coupling:

Rigid Coupling can be used only when there is perfect alignment between the axes of two shafts and the motion is free from vibration and shocks. In practice it is impossible to obtain perfect ~~mis~~alignment of shafts. To overcome misalignment flexible couplings are used.

The construction of the flexible coupling is shown in fig. 2. It is similar to rigid flange coupling except for the provision of rubber bush and pins in place of bolts.

The coupling consists of two flanges, one keyed to the input shaft and other to the output shaft. The two flanges are connected together by means of four or six pins. At one end pin is fixed to the output flange by means of nut. The diameter of the pin is enlarged in the input flange where a rubber bush is mounted over the pin. The rubber bush is provided with

10.

brass lining at inner surface. The lining reduces wear at of the inner surface of rubber bush. power is transmitted from input shaft to the input flange through key. It is then transmitted from input ~~shaft to output s'~~ flange to the pin through rubber bush. The pin then transmit the power to the output flange by shear resistance. finally power is transmitted from the output flange to the output shaft through the key.

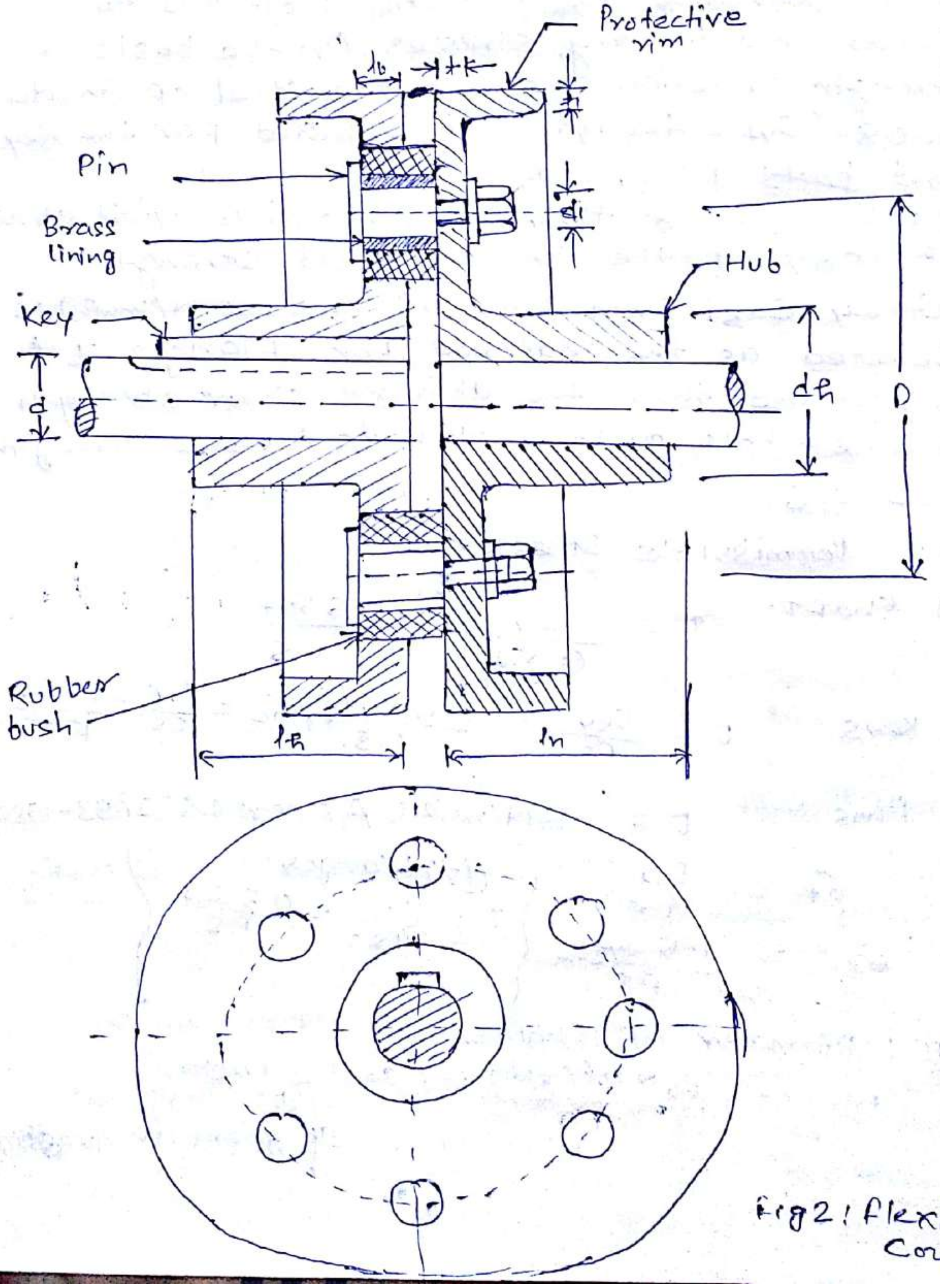


Fig 2: flexible Coupling

Design Procedure for flexible Coupling

Step I: Selection of Materials.

(i) The shafts are subjected to torsional shear stress. On the basis of strength, plain carbon steel of Grade 40C8 ($S_{yt} = 380\ N/mm^2$) is used for the shafts. The factor of safety for the shaft is assumed as 2.

(ii) The keys are subjected to shear and compressive stresses. The pins are subjected to shear and bending stresses. On the basis of strength criterion, plain carbon steel of Grade 30C8 ($S_{yt} = 400\ N/mm^2$) is selected for the keys and ~~bolts~~. pins. FOS = 2.
It is assumed that compressive yield stress is 150% of the tensile yield strength.

(iii) Grey Cast Iron FG 200 ($S_{ut} = 200\ N/mm^2$) is selected as the material for flanges. It is assumed that the ultimate shear strength is one-half of the ultimate tensile strength.
FOS = 6.
also for Hub

Step II: Permissible Stresses:

(i) Shaft. $\tau = \dfrac{S_{sy}}{(FS)} = \dfrac{0.5\,S_{yt}}{F.S.}$

(ii) Keys $\tau = \dfrac{S_{sy}}{FS} = \dfrac{0.5\,S_{yt}}{F.S.}$, $\sigma_c = \dfrac{S_{yc}}{FS} = \dfrac{1.5\,S_{yt}}{FS}$

(iii) Pins $\tau = 35\ N/mm^2$ (As per IS 2693-1980)

$$\sigma_t = \frac{S_{yt}}{FS.}$$

$$\sigma_c = \frac{1.5\,S_{yt}}{FS.}$$

(iv) Flanges

$$\tau = \frac{S_{su}}{(FS)} = \frac{0.5\,S_{ut}}{FS}$$

Step III: Diameter of Shafts.

$$M_t = \frac{60 \times 10^6 (kW)}{2\pi n} \times (\text{Service factor})$$

$\Downarrow$
gp given in question

Scanned by CamScanner

Step IV : Dimension of Flanges!

Hub diameter $d_h = 2d$

effective length of key or length of Hub $l_h = 1.5d$

pitch circle diameter of bolts

$D = 4d$

thickness of flanges $t = 0.5d$.

thickness of protective rim $t_1 = 0.25d$.

The hub is treated as a hollow cylinder subjected to torsional moment.

$$J = \frac{\pi(d_h^4 - d^4)}{32} \qquad r = \frac{d_h}{2}$$

The torsional shear stress.

$$\tau = \frac{M_t r}{J}$$

$$\boxed{\tau < \tau_{permissible}}$$

Stress in the hub is within limit

The shear stress in the flange at the junction of the hub is determined by.

$$\boxed{M_t = \frac{\pi d_h^2 t \tau}{2}}$$ ⟹ for understanding See shear stress in the flange of Rigid Flange coupling at page No. 3.

$$\boxed{\tau < \tau_{per}}$$

Stress in the flange is within limit.

Step V : Diameter of pins.

The no. of pins is generally selected as 6.

Empirical relation.

diameter of pin $d_1 = \frac{0.5d}{\sqrt{N}}$

Determine the shear stress in the pins

$$\tau = \frac{8M_t}{\pi d_1^2 D N}$$

See the explaination in the box

Torque transmitted by coupling.

$$M_t = P \times \frac{D}{2} \times N$$

$$\therefore \quad P = \frac{2M_t}{DN}$$

and direct shear stress on pin.

$$\tau = \frac{P}{\frac{\pi}{4} d_1^2}$$

$$\therefore \quad \tau = \frac{8M_t}{\pi d_1^2 DN}$$

$$\boxed{\tau < \tau_{per} = 35 \text{ N/mm}^2}$$

pin is safe by shear consideration.

Also, determine the bending stresses in the pins and confirm that it is within limit.

- The gap between two flanges is generally taken as 5 to 6 mm.

Step VII → The subassembly of the bush and pin is shown in fig. ~~8~~3.

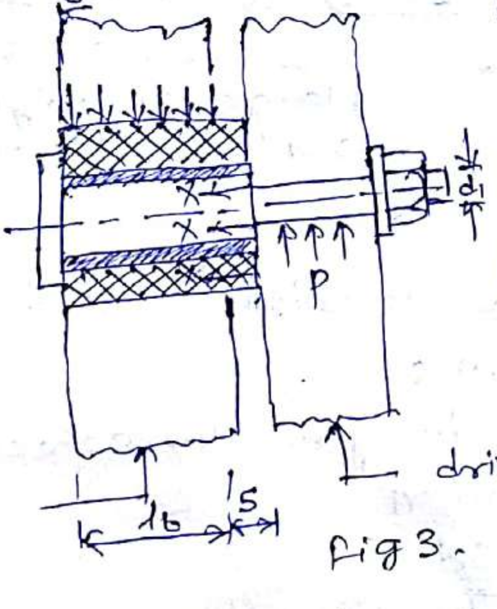


fig 3.

bending moment

$$M_b = P\left(5 + \frac{l_b}{2}\right)$$

unknown.

$$\sigma_b = \frac{32 M_b}{\pi d_1^3}$$ ⟹ Calculate d_1

equal to σ_t

If d_1 is greater than d_1 obtained by shear consideration then take greater d_1.

Step VI: Dimension of bushes:

The permissible intensity of pressure between the rubber bush and the cast Iron flange is taken as 1 N/mm².

Assume $\boxed{\frac{l_b}{D_b} = 1}$

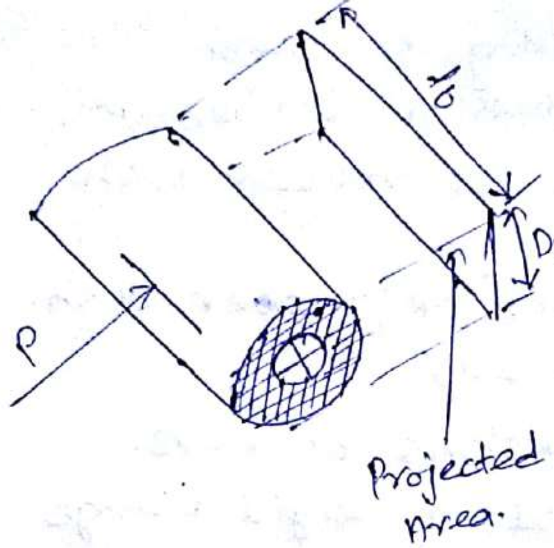


fig: 4

force $P = (D_b l_b) \times P_m$

where P_m = permissible intensity of pressure between the flange and rubber bush.

D_b = outer diameter of bush.

l_b = effective length of bush in contact with input flange.

$M_t = P \times \frac{D}{2} \times N$ & $\frac{l_b}{D_b} = 1.$

$P_m = 1\ N/mm^2.$

$$M_t = (D_b l_b) P_m \times \frac{D}{2} \times N$$

$$M_t = \frac{1}{2} D_b^2 D N$$

$$\therefore \boxed{D_b^2 = \frac{2 M_t}{D N}}$$

$D_b = \sqrt{\ } = l_b$

put l_b in bending equation.
& obtain d_1

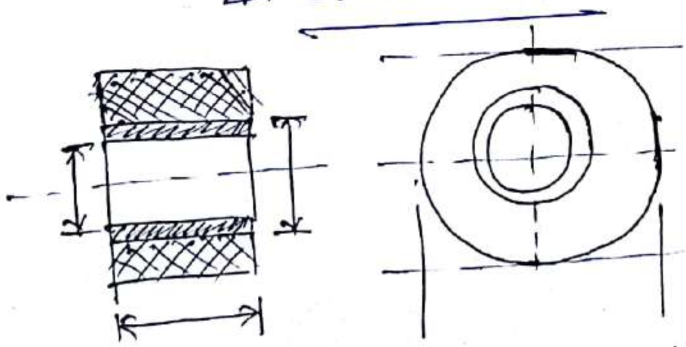


fig 5.

Enlarged diameter at section XX is taken as $(d_1 + 6)$. Other diameter of pin & bushes are shown in fig 5.

thickness of brass lining = 2 mm

the minimum thickness of rubber bush = 10 mm.

then inner diameter of rubber bush.

$= (d_1 + 6 + 4)$

The outside diameter of rubber bush.

$= (d_1 + 6 + 4 + 20)$

Step VII :— Dimensions of keys.

used Step VI of Rigid flange coupling.