Contents

3. VISION AND MISSION OF THE INSTITUTE

VISION

To impart quality technical education with strong ethics, producing technically sound engineers capable of serving the society and the nation in a responsible manner.

MISSION

M1: To provide adequate knowledge encompassing strong technical concepts and soft skills thereby inculcating sound ethics.

M2: To provide a conducive environment to nurture creativity in teaching- learning process.

M3: To identify and provide facilities which create opportunities for deserving students of all communities to excel in their chosen fields.

M4: To strive and contribute to the needs of the society and the nation by applying advanced engineering and technical concepts.

4. VISION AND MISSION OF MECHANICAL ENGINEERING DEPARTMENT

VISION

To achieve excellence in Mechanical Engineering by imparting technical and professional skills along with ethica values to meet social needs via industrial requirements.

MISSION

M1: To offer quality education with the supportive facilities to produce efficient and competent engineers through industry-institute interaction.

M2: To prepare the students with academic excellence, professional competence, and ethical behaviour for a lifelong learning.

M3: To inculcate moral & professional values among the students to cater the needs of the society anc environment.

05. Syllabus

DESIGN OF MACHINE MEMBERS - I

B.Tech. III Year I Sem. L T/P/D C Course
Code: ME501PC 4 1/0/0 4

UNIT – I

Introduction: General considerations in the design of Engineering Materials and their properties – selection – Manufacturing consideration in design. Tolerances and fits –BIS codes of steels. Design for Static Strength: Simple stresses – Combined stresses – Torsional and Bending stresses – Impact stresses – Stress strain relation – Various theories of failure – Factor of safety – Design for strength and rigidity – preferred numbers. The concept of stiffness in tension, bending, torsion and combined situations.

UNIT – II

Design for Fatigue Strength: Stress concentration–Theoretical stress Concentration factor– Fatigue stress concentration factor- Notch Sensitivity – Design for fluctuating stresses – Endurance limit – Estimation of Endurance strength – Gerber's curve– Modified Goodman's line– Soderberg's line.

UNIT – III

Riveted, Welded and Bolted Joints: Riveted joints- methods of failure of riveted jointsstrength equations-efficiency of riveted joints-eccentrically loaded riveted joints. Welded joints-Design of fillet welds-axial loads-circular fillet welds under bending, torsion. Welded joints under eccentric loading. Bolted joints – Design of bolts with pre-stresses – Design of joints under eccentric loading – locking devices – bolts of uniform strength.

UNIT – IV

Keys, Cotters and Knuckle Joints: Design of keys-stresses in keys-cottered joints-spigot and socket, sleeve and cotter, jib and cotter joints-Knuckle joints.

UNIT – V

Shafts: Design of solid and hollow shafts for strength and rigidity – Design of shafts for combined bending and axial loads – Shaft sizes – BIS code. Use of internal and external circlips, Gaskets and seals (stationary & rotary) Shaft Couplings: Rigid couplings – Muff, Split muff and Flange couplings. Flexible couplings – Flange coupling (Modified).

06.PROGRAMME OUTCOMES(PO'S):

1. **Engineering Knowledge**: Apply the knowledge of mathematics, science, engineering fundamentals, and an engineering specialization to the solution of complex engineering problems.
2. **Problem Analysis**: Identify, formulate, research literature, and analyze complex engineering problems reaching substantiated conclusions using first principles of mathematics, natural sciences, and engineering sciences.
3. **Design/development of Solutions**: Design solutions for complex engineering problems and design system components or processes that meet t h e specified needs with appropriate consideration for the public health and safety, and the cultural, societal, and environmental considerations.
4. **Conduct Investigations of Complex Problems**: Use research-based knowledge and research methods including design of experiments, analysis and interpretation of data, and synthesis of the information to provide valid conclusions.
5. **Modern Tool usage**: Create, select, and apply appropriate techniques, resources, and modern engineering and IT tools including prediction and modelling to complex engineering activities with an understanding of the limitations.
6. **The Engineer and Society**: Apply reasoning informed by the contextual knowledge to assess societal, health safety, legal and cultural issues and the consequent responsibilities relevant to the professional engineering practice.
7. **Environment and Sustainability**: Understand the impact of the professional engineering solutions in societal and environmental contexts, and demonstrate the knowledge of, and need for sustainable development.
8. **Ethics**: Apply ethical principles and commit to professional ethics and responsibilities and norms of the engineering practice.
9. **Individual and Team Work**: Function effectively as an individual, and as a member or leader in diverse teams and in multidisciplinary settings.
10. **Communication**: Communicate effectively on complex engineering activities with the engineering community and with society at large, such as, being able to comprehend and write effective reports and design documentation make effective presentations, and give and receive clear instructions.
11. **Project Management and Finance**: Demonstrate knowledge and understanding of the engineering and management principles and apply these to one's own work, as a member and leader in a team, to manage projects and in multidisciplinary environments.
12. **Life-long Learning**: Recognize the need for, and have the preparation and ability to engage in independent and life-long learning in the broadest context of technological change.

7.1 PROGRAM EDUCATIONAL OBJECTIVES(PEO'S):

PEO1. Graduates will apply their engineering knowledge and problem-solving skills to design mechanical systems and processes.

PEO2. Graduates will embrace leadership skills at various roles in their careers and establish excellence in the field of Mechanical Engineering.

PEO3. Graduates will provide engineering solutions to meet industrial requirements there by full fill global and societal needs.

7.2 PROGRAM SPECIFIC OBJECTIVES(PSO'S):

PSO1: Implement new ideas on product design and development with the help of modern computer aided tools, while ensuring best manufacturing practices

PSO2: Impart technical knowledge, ethical values and managerial skills to make successful mechanical engineers.

PSO3: Develop innovative attitude, critical thinking and problem-solving approach for any domains of mechanical engineering.

8. COURSE OBJECTIVES:

1. To understand the general design procedures and principles in the design of machine elements.
2. To study different materials of construction and their properties and factors determining the selection of material for various applications.
3. To determine stresses under different loading conditions.

COURSE OUTCOMES(CO'S):

CO.1. tha student acquires the knowledge about the principles of design ,material Selection,components behavior subjected toloads.
CO.2. understand the concept of principal stresses, stress concentration in m/c components and fatigue loading.
CO.3. design and analysis of riveted ,welded and bolted joints.
CO.4. design and analysis of cotter joint ,knuckle joint, shafts, keys and coupling.

9.MAPPING OF COURSE OUTCOMES WITH PROGRAM OUTCOMES

Course Outcomes	PO 1	PO 2	PO 3	PO 4	PO 5	PO 6	PO 7	PO 8	PO 9	PO 10	PO 11	PO 12
CO1	3	2	3	3	2	3	2		1	0	1	3
CO2	3	2	3	3	2	3	2		1	0	1	3
CO3	3	2	3	3	2	3	2		1	0	1	3
CO4	3	2	3	3	2	3	2		1	0	1	3

10. Pre-requisite:

- Basic Concepts of Engineering mechanics.
- Basic Concepts of mechanics of solids
- manufacturing processes
- metallurgy and material science.

11. Individual Time Table

Acadmic year: 2019-2020 FAZAL MOHAMMED
Course: B.Tech

	9:30 to10:20	10:20 to11:10	11:10 to 12:00	12:00 to12:50	12:50 to1:40	1:40 to2:30	2:30 to 3:20	3:20 to 4:10
MONDAY					LUNCH BREAK	DMM-I		
TUESDAY		MT LAB/ME /TE LAB						DMM-I
WEDNESDAY								
THURSDAY	DMM-I					MT LAB/ME /TE LAB		
FRIDAY						MT LAB/ME /TE LAB		

12. Class Time Table

NAWAB SHAH ALAM KHAN COLLEGE OF ENGINEERING & TECHNOLOGY
DEPARTMENT OF MECHANICAL ENGINEERING
TIME TABLE

Acadmic year: 2019-2020 Year-Semester: III-I-A
Course: B.Tech Room No. : G-LH:4

	9:30 to10:20	10:20 to11:10	11:10 to 12:00	12:00 to12:50	12:50 to1:40	1:40 to2:30	2:30 to 3:20	3:20 to 4:10
MONDAY	TE-I		FOM		LUNCH	DMM-I		LIBRARY
TUESDAY	TE-I	MT LAB/ME /TE LAB				DM		DMM-I
WEDNESDAY	TE-I		FOM			PE		
THURSDAY	DMM-I		MMT			MT LAB/ME /TE LAB		
FRIDAY	MMT		DM		BREAK	MT LAB/ME /TE LAB		

THEORY:

THERMAL ENGINEERING I (TE-I)	Mr.VINAY KULKARNI
DESIGN OF MACHINE MEMBERS -I(DMM-I)	Mr.FAZAL MOHAMMED
METROLOGY AND MACHINE TOOLS(MMT)	Mr.SADAT ALI
FUNDAMENTALS OF MANAGMENT(FOM)	Ms.QIZER UNNISA
DISATER MANAGEMENT(DM)	Mr.MD RAFEEQ

LABS:

METROLOGY LAB	Mr.SADAT ALI /Mrs. TASLEEM BANU
MACHINE TOOLS LAB	Mr.FAZAL MOHAMMED/Mr. AQEEL
TE LAB	Mr.VINAY KULKARNI/Mr MD TAHER
PROFESONAL ETHICS(PE)	Mr.

HOD PRINCIPAL

13 LECTURE SHEDULE WITH METHODOLOGY BEING USED

S.No	Unit NO	Total no of periods	Topic to be covered	Regular/Additional	Teaching adopted (Blackboard/PPT)	Remarks
1.	I	10	**Introduction**,general consideration in Machine design,engineering materials and their properties,	Regular	BB	Completed
2.			Manufacturing consideration,limits system,	Regular	BB	Completed
3.			Tolerances and fits –BIS codes of steels.	Regular	BB	Completed
4.			Simple stresses – Combined stresses	Regular	BB	Completed
5.			Torsional and Bending stresses	Regular	BB	Completed
6.			Impact stresses – Stress strain relation	Regular	BB	Completed
7.			Various theories of failure,	Regular	BB	Completed
8.			Factor of safety, Design for strength and rigidity	Regular	BB	Completed
9.			The concept of stiffness in tension, bending, torsion and combined situations.	Regular	BB	Completed
10.			Numericals	Regular	BB	Completed
11.	II	12	Stress concentration–Theoretical stress Concentration factor	Regular	BB	Completed
12.			Fatigue stress concentration factor	Regular	BB	Completed
13.			Notch Sensitivity	Regular	BB	Completed
			[illegible]	Regular	BB	Completed

16.			Gerber's curve	Regular	BB	
17.			Modified Goodman's line– Soderberg's line.	Regular	BB	Completec
18.			Numericals	Regular	BB	Completec
19.			Numericals	Regular	BB	Completec
20.			Numericals	Regular	BB	Completec
21.			Numericals	Regular	BB	Completec
22.	**III**	**15**	**Riveted joints-** Methods of Riveting, methods of failure of riveted joints	Regular	BB	Completec
23.			Caulking and Fullering. strength equations	Regular	BB	Completec
24.			-efficiency of riveted joints	Regular	BB	Completec
25.			eccentrically loaded riveted joints	Regular	BB	Completec
26.			Numericals	Regular	BB	Completec
27.			**Welded joints-** Advantages and Disadvantages of Welded Joints overRivetedJoints.Welding Processes.	Regular	BB	Completec
28.			Design of fillet welds ,axial loads	Regular	BB	Completec
29.			circular fillet welds under bending, torsion.	Regular	BB	Completec
30.			Welded joints under eccentric loading	Regular	BB	Completec
31.			**Bolted joints** – Advantages and Disadvantages of Screwed Joints Locking Devices	Regular	BB	Completec
32.			Design of bolts with pre-stresses Design of joints under eccentric loading	Regular	BB	Completec
33.			bolts of uniform strength	Regular	BB	Completec
34.			Numericals	Regular	BB	Completec
35.			Numericals	Regular	BB	Completec
36.			Numericals	Regular	BB	Completec
37			Numericals	Regular	BB	Completec
38	**IV**	**09**	Types of Keys Design of keys	Regular	BB	Completec
39.			stresses in keys	Regular	BB	Completec
40.			Numericals	Regular	BB	Completec
41.			**cottered joints**-spigot and socket	Regular	BB	Completec
42.			sleeve and cotter	Regular	BB	Completec
43.			jib and cotter joint	Regular	BB	
44.			Numericals	Regular	BB	Completec
45.			**Knuckle Joint**	Regular	BB	Completec
46.			Numericals	Regular	BB	Completec
47.	**V**	**10**	Design of solid and hollow shafts for strength and rigidity	Regular	BB	Completec
48.			Shafts Subjected to Twisting Moment Only	Regular	BB	Completec
49.			Shafts Subjected to Bending Moment Only	Regular	BB	Completec
50.			Shafts Subjected to Combined	Regular	BB	Completec

			in addition to Combined Torsion and Bending Loads.			
52.			Requirements of a Good Shaft Coupling. Types of Shaft Couplings.	Regular	BB	Completec
53.			Sleeve or Muff Coupling. Clamp or Compression Coupling	Regular	BB	Completec
54.			Design of Flange Coupling	Regular	BB	Completec
55.			Flexible Coupling Bushed Pin FlexibleCoupling. Oldham Coupling. . Universal Coupling.	Regular	BB	Completec
56.			Numericals	Regular	BB	Completec

14 LESSON SHEDULE

S.No	Unit NO	Total no of periods	Topic to be covered	Regular/Additional	Teaching adopted (Blackboard/PPT)	Remarks
	I	10	**Introduction&Design for Static Strength**			
1.		01	general consideration in Machine design,engineering materials and their properties,	Regular	BB	Completed
		01	Manufacturing consideration,limits system,	Regular	BB	Completed
3.		01	Tolerances and fits –BIS codes of steels.	Regular	BB	Completed
4.		01	Simple stresses – Combined stresses	Regular	BB	Completed
5.		01	Torsional and Bending stresses	Regular	BB	Completed
6.		01	Impact stresses – Stress strain relation	Regular	BB	Completed
7.		01	Various theories of failure,	Regular	BB	Completed
8.		01	Factor of safety, Design for strength and rigidity	Regular	BB	Completed
9.		01	The concept of stiffness in tension, bending, torsion and combined situations.	Regular	BB	Completed
10.		01	Numericals	Regular	BB	Completed
	II	12	**Design for Fatigue Strength:**			
11.		02	Stress concentration	Regular	BB	Completed

			sensitivity			
12.		01	Design for fluctuating stresses	Regular	BB	Completed
13.		01	Endurance limit – Estimation of Endurance strength	Regular	BB	Completed
14.		02	Gerber's curve Modified Goodman's line– Soderberg's line.	Regular	BB	Completed
15.		02	Numericals	Regular	BB	Completed
16.		02	Numericals	Regular	BB	
17.		02	Numericals	Regular	BB	Completed
	III	15	**Riveted, Welded and Bolted Joints**			
18.		01	Methods of Riveting, methods of failure of riveted joints	Regular	BB	Completed
19.		01	Caulking and Fullering. strength equations	Regular	BB	Completed
20.		01	efficiency of riveted joints	Regular	BB	Completed
21.		01	Numericals	Regular	BB	Completed
22.		02	eccentrically loaded riveted joints	Regular	BB	Completed
23.		01	Advantages and Disadvantages of Welded Joints overRivetedJoints.Welding Processes.	Regular	BB	Completed
24.		01	Design of fillet welds ,axial loads	Regular	BB	Completed
25.		01	circular fillet welds under bending, torsion.	Regular	BB	Completed
26.		01	Welded joints under eccentric loading	Regular	BB	Completed
27.		01	Numericals	Regular	BB	Completed
[illegible].		01	Advantages and Disadvantages of Screwed Joints Locking Devices	Regular	BB	Completed
29.		01	Design of bolts with pre-stresses Design of joints under eccentric loading	Regular	BB	Completed
30.		01	bolts of uniform strength	Regular	BB	Completed
31.		01	Numericals	Regular	BB	Completed
	IV	09	**Keys, Cotters and Knuckle Joints**			
32.		01	Types of Keys Design of keys, stresses in keys	Regular	BB	Completed
33.		01	Numericals	Regular	BB	Completed
34.		02	spigot and socket, spigot and socket, jib and cotter joint	Regular	BB	Completed
35.		02	Numericals	Regular	BB	Completed
36.		01	Knuckle Joint	Regular	BB	Completed
37		02	Numericals	Regular	BB	Completed
	V	10	**Shafts,Shaft Couplings**			
		[illegible]	[illegible]	Regular	BB	Completed

			Bending Moment Only.			
39.		01	Shafts Subjected to Combined Twisting Moment and Bending Moment.	Regular	BB	Completed
40.		01	Shafts Subjected to Axial Load in addition to Combined Torsion and Bending Loads. Design of Shafts on the Basis of Rigidity.	Regular	BB	Completed
41.		03	Numericals	Regular	BB	Completed
42.		01	Requirements of a Good Shaft Coupling. Types of Shaft Couplings.	Regular	BB	Completed
43.		01	Sleeve or Muff Coupling. Clamp or Compression Coupling, Design of Flange Coupling	Regular	BB	
44.		01	Flexible Coupling Bushed Pin FlexibleCoupling. Oldham Coupling. . Universal Coupling.	Regular	BB	Completed
45.		01	Numericals	Regular	BB	Completed

15. Detailed Class Notes

Introduction

The subject Machine Design is the creation of new and better machines and improving the existing ones. A new or better machine is one which is more economical in the overall cost of production and operation. The process of design is a long and time consuming one. From the study of existing ideas, a new idea has to be conceived. The idea is then studied keeping in mind its commercial success and given shape and form in the form of drawings. In the preparation of these drawings, care must be taken of the availability of resources in money, in men and in materials required for the successful completion of the new idea into an actual reality. In designing a machine component, it is necessary to have a good knowledge of many subjects such as Mathematics, Engineering Mechanics, Strength of Materials, Theory of Machines, Workshop Processes and Engineering Drawing.

Classifications of Machine Design

The machine design may be classified as follows:

1. ***Adaptive design.*** In most cases, the designer's work is concerned with adaptation of existing designs. This type of design needs no special knowledge or skill and can be attempted by designers of ordinary technical training. The designer only makes minor alternation or modification in the existing designs of the product.

2. ***Development design.*** This type of design needs considerable scientific training and design ability in order to modify the existing designs into a new idea by adopting a new material or different method of manufacture. In this case, though the designer starts from the existing design, but the final product may differ quite markedly from the original product.

3. ***New design.*** This type of design needs lot of research, technical ability and creative thinking. Only those designers who have personal qualities of a sufficiently high order can take up the work of a new design. The designs, depending upon the methods used, may be classified as follows:

(a) ***Rational design.*** This type of design depends upon mathematical formulae of principle of mechanics.

(b) ***Empirical design.*** This type of design depends upon empirical formulae based on the practice and past experience.

(c) ***Industrial design.*** This type of design depends upon the production aspects to manufacture any machine component in the industry.

size, it is necessary to know the forces which the part must sustain. It is also important to anticipate any suddenly applied or impact load which may cause failure.

5. ***Frictional resistance and lubrication.*** There is always a loss of power due to frictional resistance and it should be noted that the friction of starting is higher than that of running friction. It is, therefore, essential that a careful attention must be given to the matter of lubrication of all surfaces which move in contact with others, whether in rotating, sliding, or rolling bearings.

6. ***Convenient and economical features.*** In designing, the operating features of the machine should be carefully studied. The starting, controlling and stopping levers should be located on the basis of convenient handling. The adjustment for wear must be provided employing the various take up devices and arranging them so that the alignment of parts is preserved. If parts are to be changed for different products or replaced on account of wear or breakage, easy access should be provided and the necessity of removing other parts to accomplish this should be avoided if possible. The economical operation of a machine which is to be used for production or for the processing of material should be studied, in order to learn whether it has the maximum capacity consistent with the production of goodwork.

7. ***Use of standard parts.*** The use of standard parts is closely related to cost, because the cost of standard or stock parts is only a fraction of the cost of similar parts made to order. The standard or stock parts should be used whenever possible; parts for which patterns are already in existence such as gears, pulleys and bearings and parts which may be selected from regular shop stock such as screws, nuts and pins. Bolts and studs should be as few as possible to avoid the delay caused by changing drills, reamers and taps and also to decrease the number of wrenches required.

8. ***Safety of operation.*** Some machines are dangerous to operate, especially those which are speeded up to insure production at a maximum rate. Therefore, any moving part of a machine which is within the zone of a worker is considered an accident hazard and may be the cause of an injury. It is, therefore, necessary that a designer should always provide safety devices for the safety of the operator. The safety appliances should in no way interfere with operation of the machine.

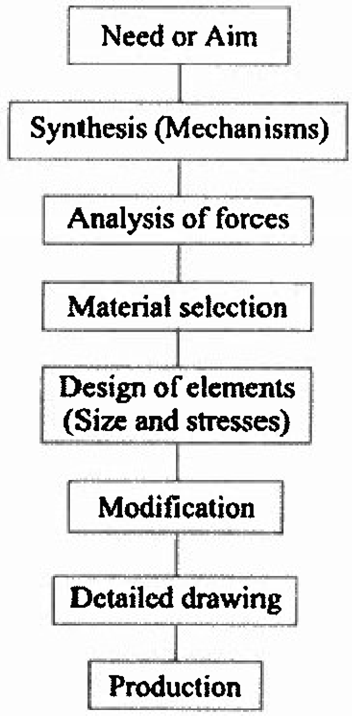


Fig.1. General Machine Design Procedure

1. ***Recognition of need.*** First of all, make a complete statement of the problem, indicating the need, aim or purpose for which the machine is to be designed.

2. ***Synthesis (Mechanisms).*** Select the possible mechanism or group of mechanisms which will give the desired motion.

3. ***Analysis of forces.*** Find the forces acting on each member of the machine and the energy transmitted by each member.

4. ***Material selection.*** Select the material best suited for each member of the machine.

5. ***Design of elements (Size and Stresses).*** Find the size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit.

The ***non-ferrous*** metals are those which have a metal other than iron as their main constituent, such as copper, aluminum, brass, tin, zinc, etc.

Selection of Materials for Engineering Purposes

The selection of a proper material, for engineering purposes, is one of the most difficult problems for the designer. The best material is one which serves the desired objective at the minimum cost. The following factors should be considered while selecting the material:

1. Availability of the materials,

2. Suitability of the materials for the working conditions in service, and

3. The cost of the materials.

The important properties, which determine the utility of the material, are physical, chemical and mechanical properties. We shall now discuss the physical and mechanical properties of the material in the following articles.

Physical Properties of Metals

The physical properties of the metals include luster, colour, size and shape, density, electric and thermal conductivity, and melting point. The following table shows the important physical properties of some pure metals.

Mechanical Properties of Metals

The mechanical properties of the metals are those which are associated with the ability of the material to resist mechanical forces and load. These mechanical properties of the metal include strength, stiffness, elasticity, plasticity, ductility, brittleness, malleability, toughness, resilience, creep and hardness. We shall now discuss these properties as follows:

1. Strength. It is the ability of a material to resist the externally applied forces without breaking or yielding. The internal resistance offered by a part to an externally applied force is called stress.

2. Stiffness. It is the ability of a material to resist deformation under stress. The modulus of elasticity is the measure of stiffness.

10. Resilience. It is the property of a material to absorb energy and to resist shock and impact loads. It is measured by the amount of energy absorbed per unit volume within elastic limit. This property is essential for spring materials.

11. Creep. When a part is subjected to a constant stress at high temperature for a long period of time, it will undergo a slow and permanent deformation called **creep.** This property is considered in designing internal combustion engines, boilers and turbines.

12. Fatigue. When a material is subjected to repeated stresses, it fails at stresses below the yield point stresses. Such type of failure of a material is known as ***fatigue**. The failure is caused by means of a progressive crack formation which are usually fine and of microscopic size. This property is considered in designing shafts, connecting rods, springs, gears, etc.

13. Hardness. It is a very important property of the metals and has a wide variety of meanings. It embraces many different properties such as resistance to wear, scratching, deformation and machinability etc. It also means the ability of a metal to cut another metal. The hardness is usually expressed in numbers which are dependent on the method of making the test. The hardness of a metal may be determined by the following tests:

(a) Brinell hardness test,

(b) Rockwell hardness test,

(c) Vickers hardness (also called Diamond Pyramid) test, and

(d) Shore scleroscope.

Steel

It is an alloy of iron and carbon, with carbon content up to a maximum of 1.5%. The carbon occurs in the form of iron carbide, because of its ability to increase the hardness and strength of the steel. Other elements *e.g.* silicon, sulphur, phosphorus and manganese are also present to greater or lesser amount to impart certain desired properties to it. Most of the steel produced now-a-days is ***plain carbon steel*** or simply ***carbon steel.*** A carbon steel is defined as a steel which has its properties mainly due to its carbon content and does not contain more than 0.5% of silicon and 1.5% of manganese.

The plain carbon steels varying from 0.06% carbon to 1.5% carbon are divided into the following types depending upon the carbon content.

1. Dead mild steel — up to 0.15% carbon

1. ***Silicon.*** The amount of silicon in the finished steel usually ranges from 0.05 to 0.30%. Silicon is added in low carbon steels to prevent them from becoming porous. It removes the gases and oxides, prevent blow holes and thereby makes the steel tougher and harder.

2. ***Sulphur.*** It occurs in steel either as iron sulphide or manganese sulphide. Iron sulphide because of its low melting point produces red shortness, whereas manganese sulphide does not affect so much. Therefore, manganese sulphide is less objectionable in steel than iron sulphide.

3. ***Manganese.*** It serves as a valuable deoxidising and purifying agent in steel. Manganese also combines with sulphur and thereby decreases the harmful effect of this element remaining in the steel. When used in ordinary low carbon steels, manganese makes the metal ductile and of good bending qualities. In high speed steels, it is used to toughen the metal and to increase its critical temperature.

4. ***Phosphorus.*** It makes the steel brittle. It also produces cold shortness in steel. In low carbon steels, it raises the yield point and improves the resistance to atmospheric corrosion. The sum of carbon and phosphorus usually does not exceed 0.25%.

References:

1. Machine Design - V.Bandari .
2. Machine Design – R.S. Khurmi
3. Design Data hand Book - S MD Jalaludin.

4. Assembling

Interchangeability

The term interchangeability is normally employed for the mass production of identical items within the prescribed limits of sizes. A little consideration will show that in order to maintain the sizes of the part within a close degree of accuracy, a lot of time is required. But even then there will be small variations. If the variations are within certain limits, all parts of equivalent size will be equally fit for operating in machines and mechanisms. Therefore, certain variations are recognized and allowed in the sizes of the mating parts to give the required fitting. This facilitates to select at random from a large number of parts for an assembly and results in a considerable saving in the cost of production.

In order to control the size of finished part, with due allowance for error, for interchangeable parts is called ***limit system***. It may be noted that when an assembly is made of two parts, the part which enters into the other, is known as ***enveloped surface*** (or **shaft** for cylindrical part) and the other in which one enters is called ***enveloping surface*** (or **hole** for cylindrical part). The term ***shaft*** refers not only to the diameter of a circular shaft, but it is also used to designate any external dimension of a part. The term ***hole*** refers not only to the diameter of a circular hole, but it is also used to designate any internal dimension of a part.

Important Terms used in Limit System

The following terms used in limit system (or interchangeable system) are important from the subject point of view:

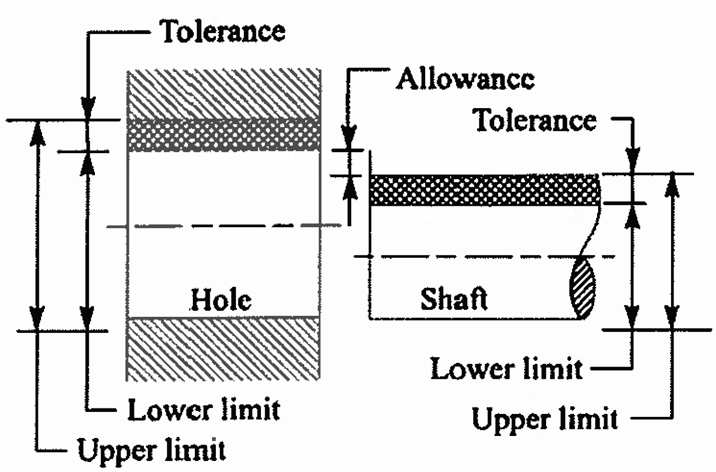


Fig. Limits of sizes.

allowance or type of fit. When the tolerance is allowed on both sides of the nominal size, e.g. $20^{+0.002}_{-0.002}$, then it is said to be **bilateral system of tolerance.** In this case + 0.002 is the upper limit and – 0.002 is the lower limit.

7. Tolerance zone. It is the zone between the maximum and minimum limit size.

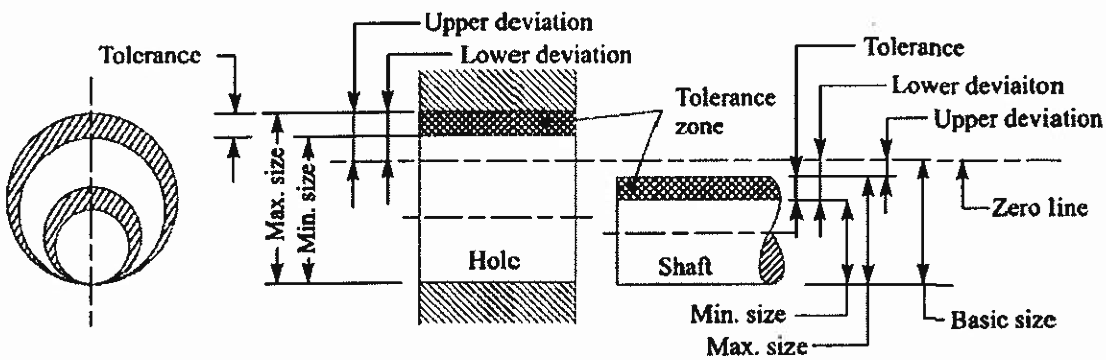


Fig. Tolerance Zone

8. Zero line. It is a straight line corresponding to the basic size. The deviations are measured from this line. The positive and negative deviations are shown above and below the zero line respectively.

9. Upper deviation. It is the algebraic difference between the maximum size and the basic size. The upper deviation of a hole is represented by a symbol ES (Ecart Superior) and of a shaft, it is represented by es.

10. Lower deviation. It is the algebraic difference between the minimum size and the basic size. The lower deviation of a hole is represented by a symbol EI (Ecart Inferior) and of a shaft, it is represented by ei.

11. Actual deviation. It is the algebraic difference between an actual size and the corresponding basic size.

12. Mean deviation. It is the arithmetical mean between the upper and lower deviations.

interference is the arithmetical difference between the sizes of the hole and the shaft, before assembly. The difference must be ***negative.***

Types of Fits

According to Indian standards, the fits are classified into the following three groups:

1. ***Clearance fit.*** In this type of fit, the size limits for mating parts are so selected that clearance between them always occur, as shown in Fig. (*a*). It may be noted that in a clearance fit, the tolerance zone of the hole is entirely above the tolerance zone of the shaft. In a clearance fit, the difference between the minimum size of the hole and the maximum size of the shaft is known as ***minimum clearance*** whereas the difference between the maximum size of the hole and minimum size of the shaft is called ***maximum clearance*** as shown in Fig. (*a*). The clearance fits may be slide fit, easy sliding fit, running fit, slack running fit and loose running fit.

2. ***Interference fit.*** In this type of fit, the size limits for the mating parts are so selected that interference between them always occur, as shown in Fig. (*b*). It may be noted that in an interference fit, the tolerance zone of the hole is entirely below the tolerance zone of the shaft. In an interference fit, the difference between the maximum size of the hole and the minimum size of the shaft is known as ***minimum interference***, whereas the difference between the minimum size of the hole and the maximum size of the shaft is called ***maximum interference***, as shown in Fig. (*b*).

The interference fits may be shrink fit, heavy drive fit and light drive fit.

3. ***Transition fit.*** In this type of fit, the size limits for the mating parts are so selected that either a clearance or interference may occur depending upon the actual size of the mating parts, as shown in Fig. (*c*). It may be noted that in a transition fit, the tolerance zones of hole and shaft overlap. The transition fits may be force fit, tight fit and push fit.

Basis of Limit System

The following are two bases of limit system:

1. ***Hole basis system.*** When the hole is kept as a constant member (*i.e.* when the lower deviation of the hole is zero) and different fits are obtained by varying the shaft size, as shown in Fig. (*a*), then the limit system is said to be on a hole basis.

Problem-1:

The dimensions of the mating parts, according to basic hole system, are given as follows:
Hole : 25.00 mm S aft : 24.97 mm
25.02 mm 24.95 mm
Find the hole tolerance, shaft tolerance and allowance.

Solution. Given : Lower limit of hole = 25 mm ; Upper limit of hole = 25.02 mm ; Upper limit of shaft = 24.97 mm ; Lower limit of shaft = 24.95 mm

Hole tolerance

We know that hole tolerance

= Upper limit of hole – Lower limit of hole

= 25.02 – 25 = 0.02 mm **Ans.**

Shaft tolerance

We know that shaft tolerance

= Upper limit of shaft – Lower limit of shaft

= 24.97 – 24.95 = 0.02 mm **Ans.**

Allowance

We know that allowance

= Lower limit of hole – Upper limit of shaft

= 25.00 – 24.97 = 0.03 mm **Ans.**

Problem-2:

Calculate the tolerances, fundamental deviations and limits of sizes for the shaft designated as 40 H8 / f7.

Solution. Given: Shaft designation = 40 $H8 / f7$

The shaft designation 40 $H8 / f7$ means that the basic size is 40 mm and the tolerance grade for the hole is 8 (*i.e.* $IT\,8$) and for the shaft is 7 (*i.e.* $IT\,7$).

Tolerances

Since 40 mm lies in the diameter steps of 30 to 50 mm, therefore the geometric mean diameter,

$$D = \sqrt{30 \times 50} = 38.73 \text{ mm}$$

We know that standard tolerance unit,

$$i = 0.45\sqrt[3]{D} + 0.001\,D$$
$$= 0.45\sqrt[3]{38.73} + 0.001 \times 38.73$$
$$= 0.45 \times 3.38 + 0.03873 = 1.559\,73 \text{ or } 1.56 \text{ microns}$$
$$= 1.56 \times 0.001 = 0.001\,56 \text{ mm} \qquad \ldots(\because 1 \text{ micron} = 0.001 \text{ mm})$$

From Table 3.2, we find that standard tolerance for the hole of grade 8 ($IT\,8$)

$$= 25\,i = 25 \times 0.001\,56 = 0.039 \text{ mm } \textbf{Ans.}$$

and standard tolerance for the shaft of grade 7 ($IT\,7$)

$$= 16\,i = 16 \times 0.001\,56 = 0.025 \text{ mm } \textbf{Ans.}$$

$\therefore$ Lower deviation for shaft g,

$$ei = es - IT = -0.01 - 0.03 = -0.04 \text{ mm}$$

We know that lower limit for hole

$= \text{Basic size} = 75 \text{ mm}$

Upper limit for hole $=$ Lower limit for hole + Tolerance for hole

$= 75 + 0.046 = 75.046 \text{ mm}$

Upper limit for shaft $=$ Lower limit for hole $-$ Upper deviation for shaft

...($\because$ Shaft g lies below zero line)

$= 75 - 0.01 = 74.99 \text{ mm}$

and lower limit for shaft $=$ Upper limit for shaft $-$ Tolerance for shaft

$= 74.99 - 0.03 = 74.96 \text{ mm}$

We know that maximum clearance

$=$ Upper limit for hole $-$ Lower limit for shaft

$= 75.046 - 74.96 = 0.086$ mm **Ans.**

and minimum clearance $=$ Lower limit for hole $-$ Upper limit for shaft

$= 75 - 74.99 = 0.01$ mm **Ans.**

as shown in Fig. (*b*). A little consideration will show that due to the tensile load, there will be a decrease in cross-sectional area and an increase in length of the body. The ratio of the increase in length to the original length is known as ***tensile strain.***

Let P = Axial tensile force acting on the body,

A = Cross-sectional area of the body,

l = Original length, and

δl = Increase in length.

Then □Tensile stress, $\sigma_t = P/A$

and tensile strain, $\varepsilon_t = \delta l / l$

Young's Modulus or Modulus of Elasticity

Hooke's law* states that when a material is loaded within elastic limit, the stress is directly proportional to strain, *i.e.*

$$\sigma \propto \varepsilon \qquad \text{or} \qquad \sigma = E.\varepsilon$$

$$E = \frac{\sigma}{\varepsilon} = \frac{P \times l}{A \times \delta l}$$

where E is a constant of proportionality known as ***Young's modulus*** or ***modulus of elasticity.*** In S.I. units, it is usually expressed in GPa *i.e.* GN/m^2 or kN/mm^2. It may be noted that Hooke's law holds good for tension as well as compression.

The following table shows the values of modulus of elasticity or Young's modulus (E) for the materials commonly used in engineering practice.

Values of E for the commonly used engineering materials.

Material	*Modulus of elasticity (E) in GPai.e. GN/m^2 for kN/mm^2*
Steel and Nickel	200 to 220
Wrought iron	190 to 200
Cast iron	100 to 160
Copper	90 to 110
Brass	80 to 90
Aluminium	60 to 80
Timber	10

rivet (or when the shearing takes place at Two cross-sections of the rivet), then the rivets are said to be in *double shear*. In su h a case, the area resisting the shear off the rivet,

$$A = 2 \times \frac{\pi}{4} \times d^2 \qquad \text{(For double shear)}$$

and shear stress on the rivet cross-section.

$$\tau = \frac{P}{A} = \frac{P}{2 \times \frac{\pi}{4} \times d^2} = \frac{2P}{\pi d^2}$$

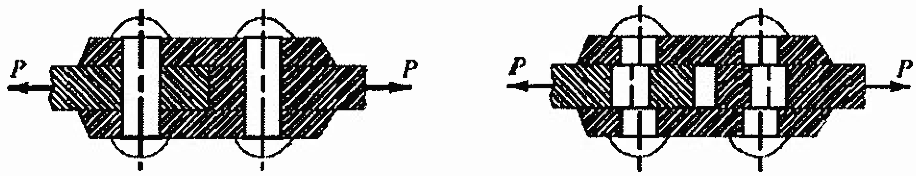


Fig. Double shearing of a riveted joint.

Notes:

1. All lap joints and single cover butt joints are in single shear, while the butt joints with double cover plates are in double shear.

2. In case of shear, the area invo ved is parallel to the external force applied.

3. When the holes are to be punched or drilled in the metal plates, then the tools used to perform the operations must overcome the ultimate shearing resistance of the material to be cut. If a hole of diameter '*d*' is to be punched in a metal plate of thickness '*t*', then the area to be sheared,

$$A = \pi d \times t$$

And the maximum shear resistance of the tool or the force required to punch a hole,

$$P = A \times \tau_u = \pi d \times t \times \tau_u$$

Where σ_u= Ultimate shear strength of the material of the plate.

Shear Modulus or Modulus of Rigidity

It has been found experimentally that within the elastic limit, the shear stress is directly proportional to shear strain. Mathematically

$$\tau \propto \phi \qquad \text{or} \quad \tau = C \,.\, \phi \quad \text{or} \quad \tau / \phi = C$$

4.18 Poisson's Ratio

It has been found experimentally that when a body is stressed within elastic limit, the lateral strain bears a constant ratio to the linear strain, Mathematically,

$$\frac{\text{Lateral Strain}}{\text{Linear Strain}} = \text{Constant}$$

This constant is known as **Poisson's ratio** and is denoted by $1/m$ or μ.

Following are the values of Poisson's ratio for some of the materials commonly used in engineering practice.

Values of Poisson's ratio for commonly used materials

S.No.	*Material*	*Poisson 's ratio (1/m or μ)*
1	**Steel**	0.25 to 0.33
2	Cast iron	0.23 to 0.27
3	Copper	0.31 to 0.34
4	Brass	0.32 to 0.42
5	Aluminium	0.32 to 0.36
6	Concrete	0.08 to 0.18
7	Rubber	0.45 to 0.50

Volumetric Strain

When a body is subjected to a system of forces, it undergoes some changes in its dimensions. In other words, the volume of the body is changed. The ratio of the change in volume to the original volume is known as ***volumetric strain.*** Mathematically, volumetric strain,

$$\varepsilon_v = \delta V / V$$

Where δV = Change in volume, and V = Original volume

Notes : 1. Volumetric strain of a rectangular body subjected to an axial force is given as

$$\varepsilon_v = \frac{\delta V}{V} = \varepsilon\left(1 - \frac{2}{m}\right); \text{ where } \varepsilon = \text{Linear strain.}$$

2. Volumetric strain of a rectangular body subjected to three mutually perpendicular forces is given by

Problem:

A steel bar 2.4 m long and 30 mm square is elongated by a load of 500 kN. If poisson's ratio is 0.25, find the increase in volume. Take E = 0.2 × 10^6 N/mm^2.

Solution. Given : l = 2.4 m = 2400 mm ; A = 30 × 30 = 900 mm^2 ; P = 500 kN = 500 × 10^3 N ; $1/m$ = 0.25 ; E = 0.2 × 10^6 N/mm^2

Let δV = Increase in volume.

We know that volume of the rod,

$$V = \text{Area} \times \text{length} = 900 \times 2400 = 2160 \times 10^3 \text{ mm}^3$$

and Young's modulus,

$$E = \frac{\text{Stress}}{\text{Strain}} = \frac{P/A}{\varepsilon}$$

$$\therefore \quad \varepsilon = \frac{P}{A.E} = \frac{500 \times 10^3}{900 \times 0.2 \times 10^6} = 2.8 \times 10^{-3}$$

We know that volumetric strain,

$$\frac{\delta V}{V} = \varepsilon\left(1 - \frac{2}{m}\right) = 2.8 \times 10^{-3}\,(1 - 2 \times 0.25) = 1.4 \times 10^3$$

$$\therefore \quad \delta V = V \times 1.4 \times 10^{-3} = 2160 \times 10^3 \times 1.4 \times 10^{-3} = 3024 \text{ mm}^3 \textbf{ Ans.}$$

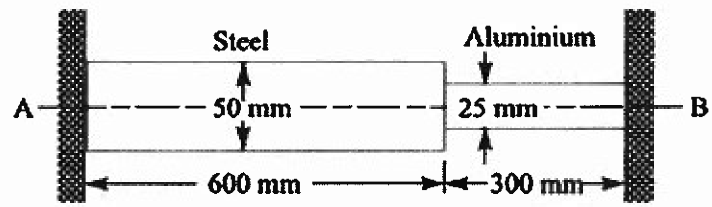
$$\sigma = E.\varepsilon = \frac{E\,(D-d)}{d}$$

Problem:

A composite bar made of aluminum and steel is held between the supports as shown in Fig. The bars are stress free at a temperature of 37°C. What will be the stress in the two bars when the temperature is 20°C, if (a) the supports are unyielding; and (b) the supports yield and come nearer to each other by 0.10 mm?

It can be assumed that the change of temperature is uniform all along the length of the bar.

Take Es = 210 GPa ; Ea = 74 GPa ; $\alpha_s = 11.7 \times 10^{-6}$ / °C ; and $\alpha_a = 23.4 \times 10^{-6}$ / °C.



Solution. Given : $t_1 = 37°C$; $t_2 = 20°C$; $E_s = 210$ GPa $= 210 \times 10^9$ N/m^2 ; $E_a = 74$ GPa $= 74 \times 10^9$ N/m^2 ; $\alpha_s = 11.7 \times 10^{-6}$/ °C ; $\alpha_a = 23.4 \times 10^{-6}$/ °C , $d_s = 50$ mm $= 0.05$ m ; $d_a = 25$ mm $= 0.025$ m ; $l_s = 600$ mm $= 0.6$ m ; $l_a = 300$ mm $= 0.3$ m

Let us assume that the right support at B is removed and the bar is allowed to contract freely due to the fall in temperature. We know that the fall in temperature,

$$t = t_1 - t_2 = 37 - 20 = 17°C$$

∴ Contraction in steel bar

$$= \alpha_s \cdot l_s \cdot t = 11.7 \times 10^{-6} \times 600 \times 17 = 0.12 \text{ mm}$$

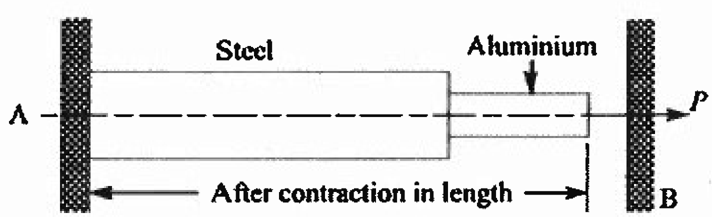
and contraction in aluminium bar

$$= \alpha_a \cdot l_a \cdot t = 23.4 \times 10^{-6} \times 300 \times 17 = 0.12 \text{ mm}$$

$$\text{Total contraction} = 0.12 + 0.12 = 0.24 \text{ mm} = 0.24 \times 10^{-3} \text{ m}$$

It may be noted that even after this contraction (*i.e.* 0.24 mm) in length, the bar is still stress free as the right hand end was assumed free.

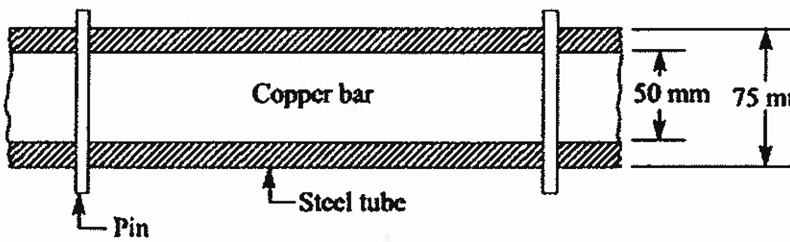
Let an axial force P is applied to the right end till this end is brought in contact with the right hand support at B, as shown in Fig.



raised by 50°C. Take E_s = 210 GN/m^2 ; E_c = 105 GN/m^2 ; α_s = 11.5 × 10^{-6}/°C and α_c = 17 × 10^{-6}/°C.

Solution. Given: d_c = 50 mm ; d_{se} = 75 mm ; d_{si} = 50 mm ; d_p = 18 mm = 0.018 m ; t = 50°C; E_s = 210 GN/m^2 = 210 × 10^9 N/m^2 ; E_c = 105 GN/m^2 = 105 × 10^9 N/m^2 ; α_s = 11.5 × 10^{-6}/°C ; α_c = 17 × 10^{-6}/°C

The copper bar in a steel tube is shown in Fig. 4.18.



We know that cross-sectional area of the copper bar,

$$A_c = \frac{\pi}{4}(d_c)^2 = \frac{\pi}{4}(50)^2 = 1964 \text{ mm}^2 = 1964 \times 10^{-6} \text{ m}^2$$

and cross-sectional area of the steel tube,

$$A_s = \frac{\pi}{4}\left[(d_{se})^2 - (d_{si})^2\right] = \frac{\pi}{4}\left[(75)^2 - (50)^2\right] = 2455 \text{ mm}^2$$

$$= 2455 \times 10^{-6} \text{ m}^2$$

Let l = Length of the copper bar and steel tube.

We know that free expansion of copper bar

$$= \alpha_c \cdot l \cdot t = 17 \times 10^{-6} \times l \times 50 = 850 \times 10^{-6}\, l$$

and free expansion of steel tube

$$= \alpha_s \cdot l \cdot t = 11.5 \times 10^{-6} \times l \times 50 = 575 \times 10^{-6}\, l$$

∴ Difference in free expansion

$$= 850 \times 10^{-6}\, l - 575 \times 10^{-6}\, l = 275 \times 10^{-6}\, l \qquad ...(i)$$

Since the free expansion of the copper bar is more than the free expansion of the steel tube, therefore the copper bar is subjected to a compressive stress, while the steel tube is subjected to a tensile stress. Let a compressive force P newton on the copper bar opposes the extra expansion of the copper bar and an equal tensile force P on the steel tube pulls the steel tube so that the net effect of reduction in length of copper bar and the increase in length of steel tube equalizes the difference in free expansion of the two.

Therefore, Reduction in length of copper bar due to force P

$$= \frac{P \cdot l}{A_c \cdot E_c}$$

Impact Stress

Sometimes, machine members are subjected to the load with impact. The stress produced in the member due to the falling load is known as ***impact stress.*** Consider a bar carrying a load W at a height h and falling on the collar provided at the lower end, as shown in Fig.

Let A = Cross-sectional area of the bar,

E = Young's modulus of the material of the bar,

l = Length of the bar,

δl = Deformation of the bar,

P = Force at which the deflection δl is produced,

σ_i = Stress induced in the bar due to the application of impact load, and

h = Height through which the load falls.

We know that energy gained by the system in the form of strain energy

$$= \frac{1}{2} \times P \times \delta l$$

And potential energy lost by the weight

$$= W(h + \delta l)$$

Since the energy gained by the system is equal to the potential energy lost by the weight, therefore

$$\frac{1}{2} \times P \times \delta l = W\,(h + \delta l)$$

$$\frac{1}{2}\sigma_i \times A \times \frac{\sigma_i \times l}{E} = W\left(h + \frac{\sigma_i \times l}{E}\right) \qquad \ldots\left[\because P = \sigma_i \times A \text{ and } \delta l = \frac{\sigma_i \times l}{E}\right]$$

$$\therefore \qquad \frac{Al}{2E}(\sigma_i)^2 - \frac{Wl}{E}(\sigma_i) - Wh = 0$$

From this quadratic equation, we find that

$$\sigma_i = \frac{W}{A}\left(1 + \sqrt{1 + \frac{2h\,A\,E}{W\,l}}\right) \qquad \ldots \textbf{[Taking +ve sign for maximum value]}$$

When $h = 0$, then $\sigma_i = 2W/A$. This means that the stress in the bar when the load in applied suddenly is double of the stress induced due to gradually applied load.

Problem:

An unknown weight falls through 10 mm on a collar rigidly attached to the lower end of a vertical bar 3 m long and 600 mm2 in section. If the maximum instantaneous extension is known to be 2 mm, what is the corresponding stress and the value of unknown weight? Take E = 200 kN/mm^2.

modulus of resilience. It is an important property of a material and gives capacity of the material to bear impact or shocks. Mathematically, strain energy stored in a body due to tensile or compressive load or resilience,

$$U=\frac{\sigma^2 \times V}{2E}$$

And Modulus of resilience

$$=\frac{\sigma^2}{2E}$$

Where σ = Tensile or compressive stress,

V = Volume of the body, and

E = Young's modulus of the material of the body.

When a body is subjected to a shear load, then modulus of resilience (shear)

$$=\frac{\tau^2}{2C}$$

Where τ = Shear stress, and

C = Modulus of rigidity.

When the body is subjected to torsion, then modulus of resilience

$$=\frac{\tau^2}{4C}$$

Problem:

A wrought iron bar 50 mm in diameter and 2.5 m long transmits shock energy of 100 N-m. Find the maximum instantaneous stress and the elongation. Take E = 200 GN/m^2.

Solution. Given : d = 50 mm ; l = 2.5 m = 2500 mm ; U = 100 N-m = 100 × 10^3 N-mm ; E = 200 GN/m^2 = 200 × 10^3 N/mm^2

Maximum instantaneous stress

Let σ = Maximum instantaneous stress.

We know that volume of the bar,

$$V=\frac{\pi}{4}\times d^2 \times l=\frac{\pi}{4}(50)^2\times 2500 = 4.9\times 10^6 \text{ mm}^3$$

We also know that shock or strain energy stored in the body (U),

$$100\times 10^3 = \frac{\sigma^2 \times V}{2E}=\frac{\sigma^2\times 4.9\times 10^6}{2\times 200\times 10^3}=12.25\,\sigma^2$$

$$\therefore \quad \sigma^2 = 100\times 10^3 / 12.25 = 8163 \quad \text{or} \quad \sigma = 90.3 \text{ N/mm}^2 \textbf{ Ans.}$$

Torsional Shear Stress

When a machine member is subjected to the action of two equal and opposite couples acting in parallel planes (or torque or twisting moment), then the machine member is said to be subjected to ***torsion.*** The stress set up by torsion is known as ***torsional shear stress.*** It is zero at the centroidal axis and maximum at the outer surface. Consider a shaft fixed at one end and subjected to a torque (*T*) at the other end as shown in Fig. As a result of this torque, every cross-section of the shaft is subjected to torsional shear stress. We have discussed above that the torsional shear stress is zero at the centroidal axis and maximum at the outer surface. The maximum torsional shear stress at the outer surface of the shaft may be obtained from the following equation:

$$\frac{\tau}{r} = \frac{T}{J} = \frac{C \cdot \theta}{l} \qquad \text{..........(i)}$$

Where τ = Torsional shear stress induced at the outer surface of the shaft or maximum shear stress,

r = Radius of the shaft,

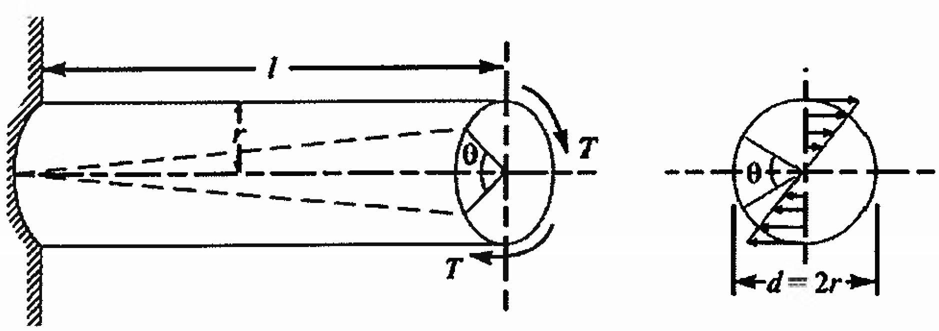
T = Torque or twisting moment,

J = Second moment of area of the section about its polar axis or polar moment of inertia,

C = Modulus of rigidity for the shaft material,

l = Length of the shaft, and

θ = Angle of twist in radians on a length l.



The above equation is known as ***torsion equation.*** It is based on the following assumptions:

1. The material of the shaft is uniform throughout.

2. The twist along the length of the shaft is uniform.

3. The normal cross-sections of the shaft, which were plane and circular before twist, remain plane and circular after twist.

Where T = Torque transmitted in N-m, and

ω = Angular speed in rad/s.

Problem:

A shaft is transmitting 100 kW at 160 r.p.m. Find a suitable diameter for the shaft, if the maximum torque transmitted exceeds the mean by 25%. Take maximum allowable shear stress as 70 MPa.

Solution. Given : $P = 100$ kW $= 100 \times 10^3$ W ; $N = 160$ r.p.m ; $T_{max} = 1.25\, T_{mean}$; $\tau = 70$ MPa $= 70$ N/mm^2

Let T_{mean} = Mean torque transmitted by the shaft in N-m, and

d = Diameter of the shaft in mm.

We know that the power transmitted (P),

$$100 \times 10^3 = \frac{2\,\pi\, N \,.\, T_{mean}}{60} = \frac{2\pi \times 160 \times T_{mean}}{60} = 16.76\, T_{mean}$$

$$\therefore \quad T_{mean} = 100 \times 10^3/16.76 = 5966.6 \text{ N-m}$$

and maximum torque transmitted,

$$T_{max} = 1.25 \times 5966.6 = 7458 \text{ N-m} = 7458 \times 10^3 \text{ N-mm}$$

We know that maximum torque (T_{max}),

$$7458 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 70 \times d^3 = 13.75\, d^3$$

$$\therefore \quad d^3 = 7458 \times 10^3/13.75 = 542.4 \times 10^3 \text{ or } d = 81.5 \text{ mm Ans.}$$

Bending Stress

In engineering practice, the machine parts of structural members may be subjected to static or dynamic loads which cause bending stress in the sections besides other types of stresses such as tensile, compressive and shearing stresses. Consider a straight beam subjected to a bending moment M as shown in Fig.

The following assumptions are usually made while deriving the bending formula.

1. The material of the beam is perfectly homogeneous (*i.e.* of the same material throughout) and isotropic (*i.e.* of equal elastic properties in all directions).

2. The material of the beam obeys Hooke's law.

3. The transverse sections (*i.e.* BC or GH) which were plane before bending remain plane after bending also.

4. Each layer of the beam is free to expand or contract, independently, of the layer, above or below it.

5. The Young's modulus (E) is the same in tension and compression.

6. The loads are applied in the plane of bending.

is $y = d/2$, where d is the diameter in case of circular section or depth in case of square or rectangular section.

3. In case of unsymmetrical sections such as L-section or T-section, the neutral axis does not pass through its geometrical centre. In such cases, first of all the centroid of the section is calculated and then the distance of the extreme fibres for both lower and upper side of the section is obtained. Out of these two values, the bigger value is used in bending equation.

Problem:

A beam of uniform rectangular cross-section is fixed at one end and carries an electric motor weighing 400 N at a distance of 300 mm from the fixed end. The maximum bending stress in the beam is 40 MPa. Find the width and depth of the beam, if depth is twice that of width.

Solution. Given: $W = 400$ N ; $L = 300$ mm ; $\sigma_b = 40$ MPa $= 40$ N/mm^2 ; $h = 2b$

The beam is shown in Fig. 5.7.

Let b = Width of the beam in mm, and

h = Depth of the beam in mm.

$\therefore$ Section modulus,

$$Z = \frac{b \cdot h^2}{6} = \frac{b\,(2b)^2}{6} = \frac{2\,b^3}{3}\ \text{mm}^3$$

Maximum bending moment (at the fixed end),

$$M = W.L = 400 \times 300 = 120 \times 10^3\ \text{N-mm}$$

We know that bending stress (σ_b),

$$40 = \frac{M}{Z} = \frac{120 \times 10^3 \times 3}{2\,b^3} = \frac{180 \times 10^3}{b^3}$$

$\therefore$ $b^3 = 180 \times 10^3/40 = 4.5 \times 10^3$ or $b = 16.5$ mm **Ans.**

and $h = 2b = 2 \times 16.5 = 33$ mm **Ans.**

Problem:

A cast iron pulley transmits 10 kW at 400 r.p.m. The diameter of the pulley is 1.2 metre and it has four straight arms of elliptical cross-section, in which the major axis is twice the minor axis. Determine the dimensions of the arm if the allowable bending stress is 15 MPa.

Solution. Given : $P = 10$ kW $= 10 \times 10^3$ W ; $N = 400$ r.p.m ; $D = 1.2$ m $= 1200$ mm or $R = 600$ mm ; $\sigma_b = 15$ MPa $= 15$ N/mm^2

Let T = Torque transmitted by the pulley.

We know that the power transmitted by the pulley (P),

Principal Stresses and Principal Planes

In the previous chapter, we have discussed about the direct tensile and compressive stress as well as simple shear. Also we have always referred the stress in a plane which is at right angles to the line of action of the force. But it has been observed that at any point in a strained material, there are three planes, mutually perpendicular to each other which carry direct stresses only and no shear stress. It may be noted that out of these three direct stresses, one will be maximum and the other will be minimum. These perpendicular planes which have no shear stress are known as ***principal planes*** and the direct stresses along these planes are known as ***principal stresses.*** The planes on which the maximum shear stress act are known as planes of maximum shear.

Determination of Principal Stresses for a Member Subjected to Bi-axial Stress

When a member is subjected to bi-axial stress (*i.e.* direct stress in two mutually perpendicular planes accompanied by a simple shear stress), then the normal and shear stresses are obtained as discussed below:

Consider a rectangular body *ABCD* of uniform cross-sectional area and unit thickness subjected to normal stresses σ_1 and σ_2 as shown in Fig. (*a*). In addition to these normal stresses, a shear stress τ also acts. It has been shown in books on '***Strength of Materials***' that the normal stress across any oblique section such as *EF* inclined at an angle θ with the direction of σ_2, as shown in Fig. (*a*), is given by

$$\sigma_t = \frac{\sigma_1 + \sigma_2}{2} + \frac{\sigma_1 + \sigma_2}{2} \cos 2\theta + \tau \sin 2\theta \qquad \ldots(i)$$

And tangential stress (*i.e.* shear stress) across the section *EF*,

$$\tau_1 = \frac{1}{2}(\sigma_1 - \sigma_2) \sin 2\theta - \tau \cos 2\theta \qquad \ldots(ii)$$

Since the planes of maximum and minimum normal stress (*i.e.* principal planes) have no shear stress, therefore the inclination of principal planes is obtained by equating $\tau_1 = 0$ in the above equation (*ii*), *i.e.*

$$\frac{1}{2}(\sigma_1 - \sigma_2) \sin 2\theta - \tau \cos 2\theta = 0$$

$$\tan 2\theta = \frac{2\tau}{\sigma_1 - \sigma_2} \qquad \ldots(iii)$$

The maximum and minimum principal stresses may now be obtained by substituting the values of sin 2θ and cos 2θ in equation *(i)*.

So, Maximum principal (or normal) stress,

$$\sigma_{t1} = \frac{\sigma_1 + \sigma_2}{2} + \frac{1}{2}\sqrt{(\sigma_1 - \sigma_2)^2 + 4\tau^2} \quad \text{...(iv)}$$

And minimum principal (or normal) stress,

$$\sigma_{t2} = \frac{\sigma_1 + \sigma_2}{2} - \frac{1}{2}\sqrt{(\sigma_1 - \sigma_2)^2 + 4\tau^2} \quad \text{...(v)}$$

The planes of maximum shear stress are at right angles to each other and are inclined at 45° to the principal planes. The maximum shear stress is given by ***one-half the algebraic difference between the principal stresses,*** *i.e.*

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} = \frac{1}{2}\sqrt{(\sigma_1 - \sigma_2)^2 + 4\tau^2} \quad \text{...(vi)}$$

Notes: 1. when a member is subjected to direct stress in one plane accompanied by a simple shear stress, then the principal stresses are obtained by substituting $\sigma_2 = 0$ in equation ***(iv)***, ***(v)*** and ***(vi)***.

$$\sigma_{t1} = \frac{\sigma_1}{2} + \frac{1}{2}\left[\sqrt{(\sigma_1)^2 + 4\tau^2}\right]$$

$$\sigma_{t2} = \frac{\sigma_1}{2} - \frac{1}{2}\left[\sqrt{(\sigma_1)^2 + 4\tau^2}\right]$$

$$\tau_{max} = \frac{1}{2}\left[\sqrt{(\sigma_1)^2 + 4\tau^2}\right]$$

2. In the above expression of $\sigma t2$, the value of $\frac{1}{2}\left[\sqrt{(\sigma_1)^2 + 4\tau^2}\right]$ is more than $\sigma_1/2$ Therefore the nature of σ_{t2} will be opposite to that of σ_{t1}, *i.e.* if σ_{t1} is tensile then σ_{t2} will be compressive and *vice-versa*.

Application of Principal Stresses in Designing Machine Members

There are many cases in practice, in which machine members are subjected to combined stresses due to simultaneous action of either tensile or compressive stresses combined with shear stresses. In many shafts such as propeller shafts, C-frames etc., there are direct tensile or compressive stresses due to the external force and shear stress due to torsion, which acts

Solution. Given : $W = 3$ kN $= 3000$ N ; $T = 1000$ N-m $= 1 \times 10^6$ N-mm ; $P = 15$ kN $= 15 \times 10^3$ N ; $d = 50$ mm; $x = 250$ mm

We know that cross-sectional area of the shaft,

$$A = \frac{\pi}{4} \times d^2$$

$$= \frac{\pi}{4} (50)^2 = 1964 \text{ mm}^2$$

$\therefore$ Tensile stress due to axial pulling at points A and B,

$$\sigma_o = \frac{P}{A} = \frac{15 \times 10^3}{1964} = 7.64 \text{ N/mm}^2 = 7.64 \text{ MPa}$$

Bending moment at points A and B,

$$M = W.x = 3000 \times 250 = 750 \times 10^3 \text{ N-mm}$$

Section modulus for the shaft,

$$Z = \frac{\pi}{32} \times d^3 = \frac{\pi}{32} (50)^3$$

$$= 12.27 \times 10^3 \text{ mm}^3$$

$\therefore$ Bending stress at points A and B,

$$\sigma_b = \frac{M}{Z} = \frac{750 \times 10^3}{12.27 \times 10^3}$$

$$= 61.1 \text{ N/mm}^2 = 61.1 \text{ MPa}$$

This bending stress is tensile at point A and compressive at point B.

$\therefore$ Resultant tensile stress at point A,

$$\sigma_A = \sigma_b + \sigma_o = 61.1 + 7.64$$
$$= 68.74 \text{ MPa}$$

and resultant compressive stress at point B,

$$\sigma_B = \sigma_b - \sigma_o = 61.1 - 7.64 = 53.46 \text{ MPa}$$

We know that the shear stress at points A and B due to the torque transmitted,

$$\tau = \frac{16\,T}{\pi\, d^3} = \frac{16 \times 1 \times 10^6}{\pi\,(50)^3} = 40.74 \text{ N/mm}^2 = 40.74 \text{ MPa} \qquad \ldots\left(\because\ T = \frac{\pi}{16} \times \tau \times d^3\right)$$

Factor of Safety

It is defined, in general, as the **ratio of the maximum stress to the working stress.** Mathematically,

Factor of safety = Maximum stress/ Working or design stress

In case of ductile materials *e.g.* mild steel, where the yield point is clearly defined, the factor of safety is based upon the yield point stress. In such cases,

Factor of safety = Yield point stress/ Working or design stress

In case of brittle materials *e.g.* cast iron, the yield point is not well defined as for ductile materials. Therefore, the factor of safety for brittle materials is based on ultimate stress.

Factor of safety = Ultimate stress/ Working or design stress

This relation may also be used for ductile materials.

The above relations for factor of safety are for static loading.

Design for strength and rigidity:

Design for strength:

All the concepts discussed so far and the problems done are strength based, i.e., there will be some permissible stress or strength and our task is to limit the stresses below the given permissible value and accordingly sizing the machine element.

Design for rigidity or stiffness:

It the ability to resist deformations under the action of external load. Along with strength, rigidity is also very important operating property of many machine components. Ex: helical and leaf springs, elastic elements in various instruments, shafts, bearings, toothed and worm gears and so on.

In many cases, this parameter of operating capacity proves to be most important and to ensure it the dimensions of the part have to be increased to such an extent that the actual induced stresses become much lower that the allowable ones. Rigidity also necessary to ensure that the mated parts and the machine as a whole operate effectively.

Forces subject the parts to elastic deformations: shafts are bent and twisted, bolts are stretched ect.,

1. When a shaft is deflected, its journals are misaligned in the bearings there by causing the uneven wear of the shells, heating and seizure in the sliding bearings.

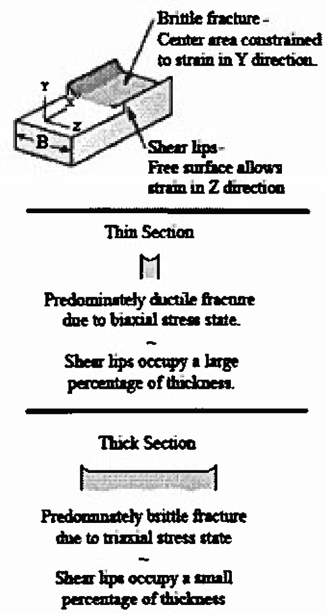
2. Deflections and angles of turn of shafts at the places where gears are fitted cause non-uniform load distribution over the length of the teeth.

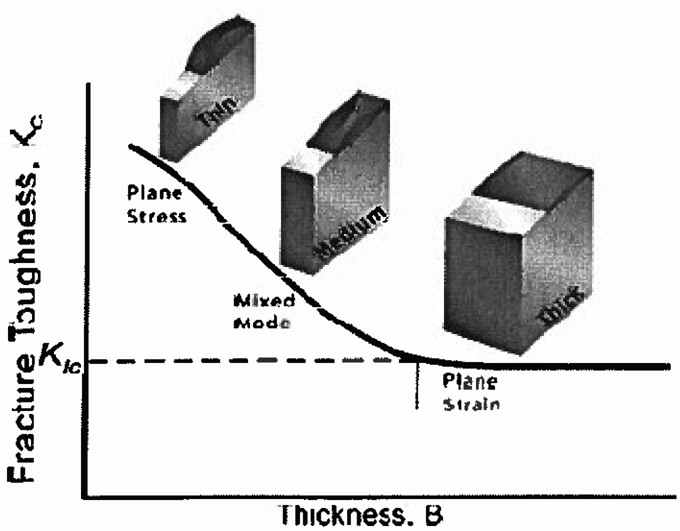
$$\sigma_{t2} = \frac{\sigma_1 + \sigma_2}{2} - \frac{1}{2}\sqrt{(\sigma_1 - \sigma_2)^2 + 4\,\tau^2}$$

$$\tau_{max} = \frac{\sigma_1 - \sigma_2}{2} = \frac{1}{2}\sqrt{(\sigma_1 - \sigma_2)^2 + 4\,\tau^2}$$

Role of Material Thickness

Specimens having standard proportions but different absolute size produce different values for K_I. This results because the stress states adjacent to the flaw changes with the specimen thickness (B) until the thickness exceeds some critical dimension. Once the thickness exceeds the critical dimension, the value of K_I becomes relatively constant and this value, K_{IC}, is a true material property which is called the plane-strain fracture toughness. The relationship between stress intensity, K_I, and fracture toughness, K_{IC}, is similar to the relationship between stress and tensile stress. The stress intensity, K_I, represents the level of "stress" at the tip of the crack and the fracture toughness, K_{IC}, is the highest value of stress intensity that a material under very specific (plane-strain) conditions that a material can withstand without fracture. As the stress intensity factor reaches the K_{IC} value, unstable fracture occurs. As with a material's other mechanical properties, K_{IC} is commonly reported in reference books and other sources.

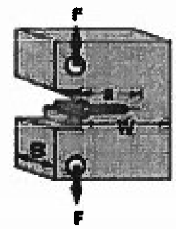




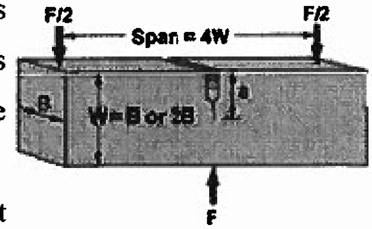
Plane Strain - a condition of a body in which the displacements of all points in the body are parallel to a given plane, and the values of theses displacements do not depend on the distance perpendicular to the plane

Plane Stress – a condition of a body in which the state of stress is such that two of the principal stresses are always parallel to a given plane and are constant in the normal direction.

When a material of unknown fracture toughness is tested, a specimen of full material section thickness is tested or the specimen is sized based on a prediction of the fracture toughness. If the fracture toughness value resulting from the test does not satisfy the requirement of the above equation, the test must be repeated using a thicker specimen. In addition to this thickness calculation, test specifications have several other requirements that must be met (such as the size of the shear lips) before a test can be said to have resulted in a K_{IC} value.



CT Specimen



SENB Specimen

When a test fails to meet the thickness and other test requirement that are in place to insure plane-strain condition, the fracture toughness values produced is given the designation K_C. Sometimes it is not possible to produce a specimen that meets the thickness requirement. For example when a relatively thin plate product with high toughness is being tested, it might not be possible to produce a thicker specimen with plain-strain conditions at the crack tip.

Plane-Stress and Transitional-Stress States

For cases where the plastic energy at the crack tip is not negligible, other fracture mechanics parameters, such as the J integral or R-curve, can be used to characterize a material. The toughness data produced by these other tests will be dependant on the thickness of the product tested and will not be a true material property. However, plane-strain conditions do not exist in all structural configurations and using K_{IC} values in the design of relatively thin areas may result in excess conservatism and a weight or cost penalty. In cases where the actual stress state is plane-stress or, more generally, some intermediate- or transitional-stress state, it is more appropriate to use J integral or R-curve data, which account for slow, stable fracture (ductile tearing) rather than rapid (brittle) fracture.

Uses of Plane-Strain Fracture Toughness

K_{IC} values are used to determine the critical crack length when a given stress is applied to a component.

$$\sigma_C \le \frac{K_{IC}}{Y\sqrt{\pi a}}$$

crack may develop under the action of repeated load and the crack will lead to failure of the member.

Stress Concentration due to Holes and Notches

Consider a plate with transverse elliptical hole and subjected to a tensile load as shown in Fig.1(a). We see from the stress-distribution that the stress at the point away from the hole is practically uniform and the maximum stress will be induced at the edge of the hole. The maximum stress is given by

$$\sigma_{max} = \sigma\left(1 + \frac{2a}{b}\right)$$

And the theoretical stress concentration factor,

$$K_t = \frac{\sigma_{max}}{\sigma} = \left(1 + \frac{2a}{r}\right)$$

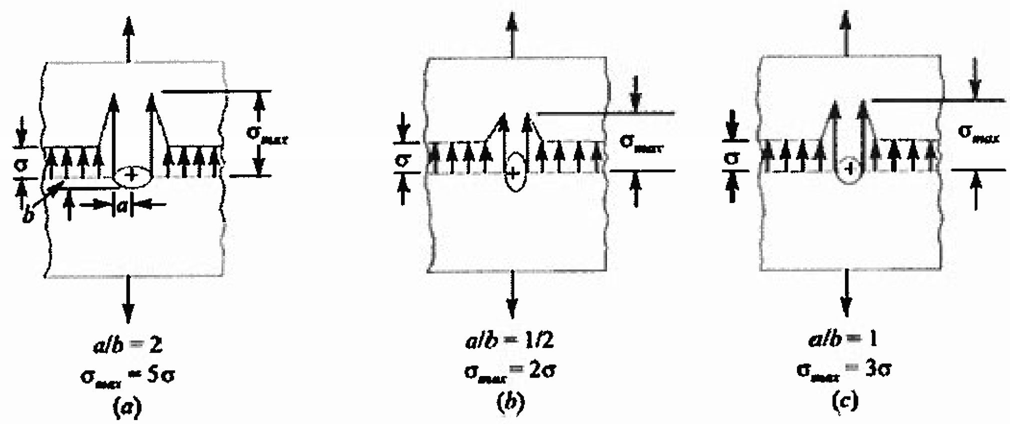


Fig.1. Stress concentration due to holes.

The stress concentration in the notched tension member, as shown in Fig. 2, is influenced by the depth a of the notch and radius r at the bottom of the notch. The maximum stress, which applies to members having notches that are small in comparison with the width of the plate, may be obtained by the following equation,

$$\sigma_{max} = \sigma\left(1 + \frac{2a}{r}\right)$$

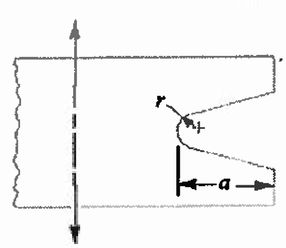


Fig.2. Stress concentration due to notches.

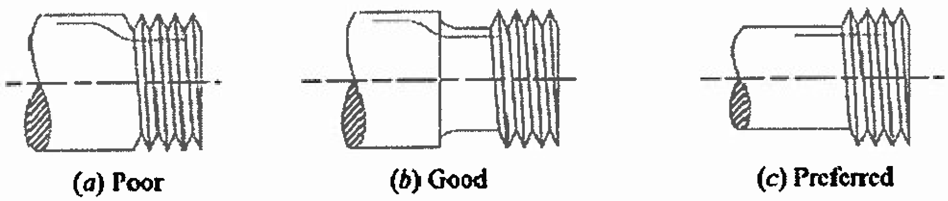
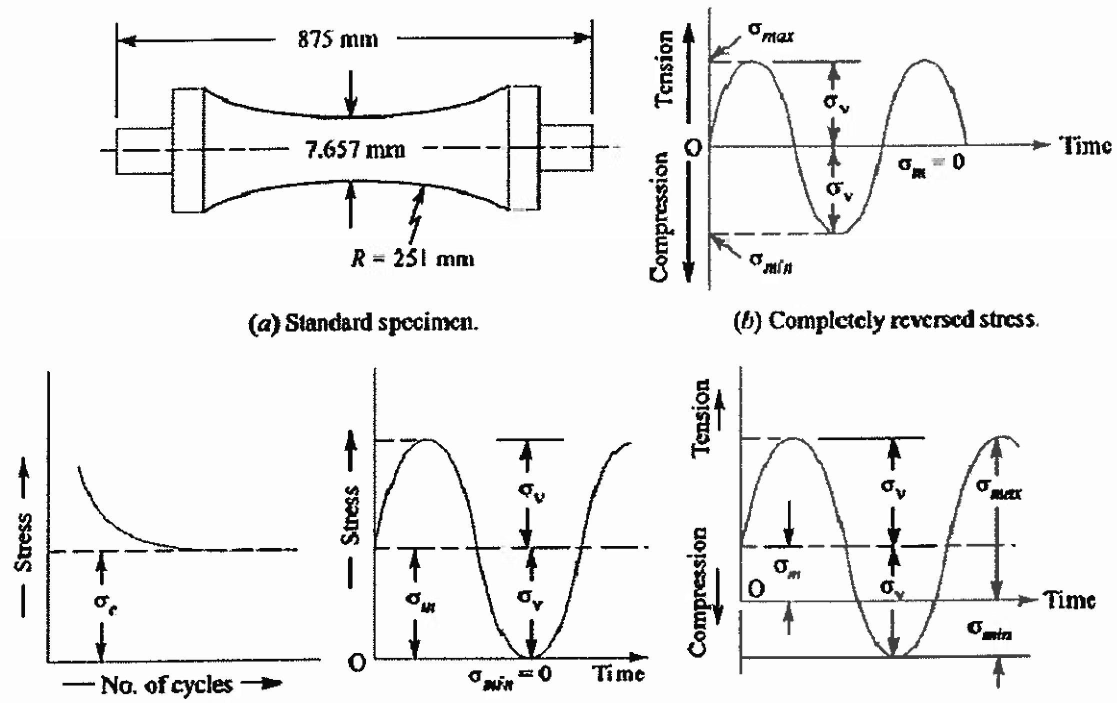


Fig. Reducing stress concentration in cylindrical members with holes



(*a*) Standard specimen. (*b*) Completely reversed stress.

(*c*) Endurance or fatigue limit. (*d*) Repeated stress. (*e*) Fluctuating stress.

Fig.2. Time-stress diagrams.

In order to study the effect of fatigue of a material, a rotating mirror beam method is used. In this method, a standard mirror polished specimen, as shown in Fig.2 (*a*), is rotated in a fatigue testing machine while the specimen is loaded in bending. As the specimen rotates, the bending stress at the upper fibres varies from maximum compressive to maximum tensile while the bending stress at the lower fibres varies from maximum tensile to maximum compressive. In other words, the specimen is subjected to a completely reversed stress cycle. This is represented by a time-stress diagram as shown in Fig.2 (*b*). A record is kept of the number of cycles required to produce failure at a given stress, and the results are plotted in stress-cycle curve as shown in Fig.2 (*c*). A little consideration will show that if the stress is kept below a certain value as shown by dotted line in Fig.2 (*c*), the material will not fail whatever may be the number of cycles. This stress, as represented by dotted line, is known as ***endurance*** or ***fatigue limit*** (σe) It is defined as maximum value of the completely reversed bending stress which a polished standard specimen can withstand without failure, for infinite number of cycles (usually 107 cycles).

It may be noted that the term endurance limit is used for reversed bending only while for other types of loading, the term ***endurance strength*** may be used when referring the

K_s = Load correction factor for the reversed torsional or shear load. Its value may be taken as 0.55 for ductile materials and 0.8 for brittle materials.

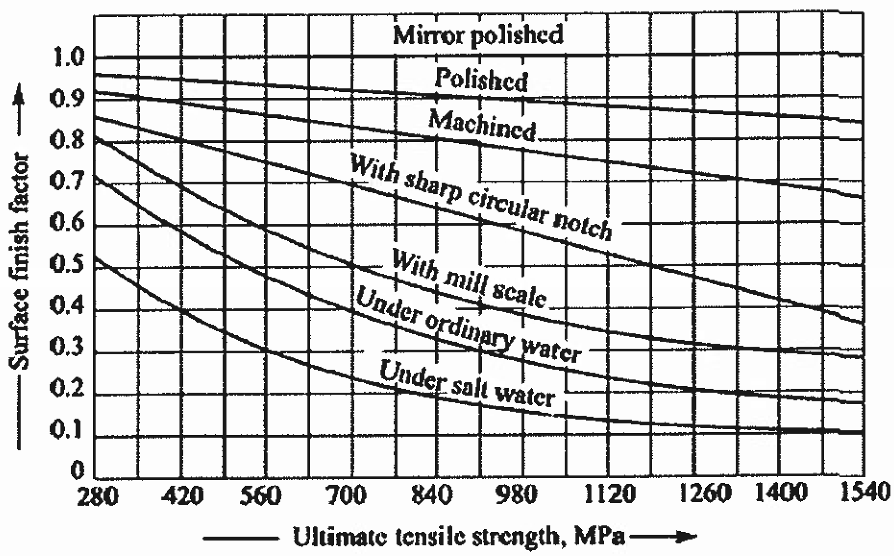
$\therefore$ **Endurance limit for reversed bending load,** $\sigma_{eb} = \sigma_e K_b = \sigma_e$

Endurance limit for reversed axial load, $\sigma_{ea} = \sigma_e K_a$

and endurance limit for reversed torsional or shear load, $\tau_e = \sigma_e K_s$

Effect of Surface Finish on Endurance Limit—Surface Finish Factor

When a machine member is subjected to variable loads, the endurance limit of the material for that member depends upon the surface conditions. Fig. shows the values of surface finish factor for the various surface conditions and ultimate tensile strength.



When the surface finish factor is known, then the endurance limit for the material of the machine member may be obtained by multiplying the endurance limit and the surface finish factor. We see that for a mirror polished material, the surface finish factor is unity. In other words, the endurance limit for mirror polished material is maximum and it goes on reducing due to surface condition.

Let K_{sur} = Surface finish factor.

Then, Endurance limit,

$$\sigma_{e1} = \sigma_{eb}.K_{sur} = \sigma_e.K_b.K_{sur} = \sigma_e.K_{sur} \quad ...(\because K_b = 1)$$

...(For reversed bending load)

$$= \sigma_{ea}.K_{sur} = \sigma_e.K_a.K_{sur}$$ ...(For reversed axial load)

$$= \tau_e.K_{sur} = \sigma_e.K_s.K_{sur}$$ ...(For reversed torsional or shear load)

For steel, $\sigma_e = 0.5\ \sigma_u$;
For cast steel, $\sigma_e = 0.4\ \sigma_u$;
For cast iron, $\sigma_e = 0.35\ \sigma_u$;
For non-ferrous metals and alloys, $\sigma_e = 0.3\ \sigma_u$

Factor of Safety for Fatigue Loading

When a component is subjected to fatigue loading, the endurance limit is the criterion for failure. Therefore, the factor of safety should be based on endurance limit. Mathematically,

$$\text{Factor of safety } (F.S.) = \frac{\text{Endurance limit stress}}{\text{Design or working stress}} = \frac{\sigma_e}{\sigma_d}$$

For steel, $\sigma_e = 0.8 \text{ to } 0.9\ \sigma_y$

σ_e = Endurance limit stress for completely reversed stress cycle, and

σ_y = Yield point stress.

Fatigue Stress Concentration Factor

When a machine member is subjected to cyclic or fatigue loading, the value of fatigue stress concentration factor shall be applied instead of theoretical stress concentration factor. Since the determination of fatigue stress concentration factor is not an easy task, therefore from experimental tests it is defined as

Fatigue stress concentration factor,

$$K_f = \frac{\text{Endurance limit without stress concentration}}{\text{Endurance limit with stress concentration}}$$

Notch Sensitivity

In cyclic loading, the effect of the notch or the fillet is usually less than predicted by the use of the theoretical factors as discussed before. The difference depends upon the stress gradient in the region of the stress concentration and on the hardness of the material. The term ***notch sensitivity*** is applied to this behaviour. It may be defined as the degree to which the theoretical effect of stress concentration is actually reached. The stress gradient depends mainly on the radius of the notch, hole or fillet and on the grain size of the material. Since the extensive data for estimating the notch sensitivity factor (q) is not available, therefore the curves, as shown in Fig., may be used for determining the values of q for two steals. When the notch sensitivity factor q is used in cyclic loading, then fatigue stress concentration factor may be obtained from the following relations:

$$q = \frac{K_f - 1}{K_t - 1}$$

Or

Problem: Determine the thickness of a 120 mm wide uniform plate for safe continuous operation if the plate is to be subjected to a tensile load that has a maximum value of 250 kN and a minimum value of 100 kN. The properties of the plate material are as follows: Endurance limit stress = 225 MPa, and Yield point stress = 300 MPa. The factor of safety based on yield point may be taken as 1.5.

Let t = Thickness of the plate in mm.

$\therefore$ Area, $A = b \times t = 120\,t \text{ mm}^2$

We know that mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{250+100}{2} = 175 \text{ kN} = 175 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{175 \times 10^3}{120t} \text{ N/mm}^2$$

$$\text{Variable load, } W_v = \frac{W_{max} - W_{min}}{2} = \frac{250-100}{2} = 75 \text{ kN} = 75 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Variable stress, } \sigma_v = \frac{W_v}{A} = \frac{75 \times 10^3}{120t} \text{ N/mm}^2$$

According to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e}$$

$$\frac{1}{1.5} = \frac{175 \times 10^3}{120t \times 300} + \frac{75 \times 10^3}{120t \times 225} = \frac{4.86}{t} + \frac{2.78}{t} = \frac{7.64}{t}$$

$\therefore$ $t = 7.64 \times 1.5 = 11.46$ say 11.5 mm **Ans.**

Problem:

Determine the diameter of a circular rod made of ductile material with a fatigue strength (complete stress reversal), σ_e = 265 MPa and a tensile yield strength of 350 MPa. The member is subjected to a varying axial load from $W_{min} = -300 \times 10^3$ N to $W_{max} = 700 \times 10^3$ N and has a stress concentration factor = 1.8. Use factor of safety as 2.0.

Let d = Diameter of the circular rod in mm.

$$\therefore \quad \text{Area, } A = \frac{\pi}{4} \times d^2 = 0.7854\, d^2 \text{ mm}^2$$

We know that the mean or average load,

$$W_m = \frac{W_{max} + W_{min}}{2} = \frac{700 \times 10^3 + (-300 \times 10^3)}{2} = 200 \times 10^3 \text{ N}$$

$$\therefore \quad \text{Mean stress, } \sigma_m = \frac{W_m}{A} = \frac{200 \times 10^3}{0.7854\, d^2} = \frac{254.6 \times 10^3}{d^2} \text{ N/mm}^2$$

and variable bending stress,

$$\sigma_v = \frac{M_v}{Z} = \frac{1875 \times 10^3}{0.0982 d^3} = \frac{19.1 \times 10^6}{d^3} \text{ N/mm}^2$$

We know that according to Goodman's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_u} + \frac{\sigma_v \times K_f}{\sigma_e \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 650} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \quad \text{...(Taking } K_f = 1\text{)}$$

$$= \frac{68 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{139 \times 10^3}{d^3}$$

$\therefore$ $d^3 = 139 \times 10^3 \times 1.5 = 209 \times 10^3$ or $d = 59.3$ mm

and according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_e \times K_{sur} \times K_{sz}}$$

$$\frac{1}{1.5} = \frac{44.5 \times 10^6}{d^3 \times 500} + \frac{19.1 \times 10^6 \times 1}{d^3 \times 350 \times 0.9 \times 0.85} \quad \text{...(Taking } K_f = 1\text{)}$$

$$= \frac{89 \times 10^3}{d^3} + \frac{71 \times 10^3}{d^3} = \frac{160 \times 10^3}{d^3}$$

$\therefore$ $d^3 = 160 \times 10^3 \times 1.5 = 240 \times 10^3$ or $d = 62.1$ mm

Taking larger of the two values, we have $d = 62.1$ mm Ans.

Problem:

A 50 mm diameter shaft is made from carbon steel having ultimate tensile strength of 630 MPa. It is subjected to a torque which fluctuates between 2000 N-m to – 800 N-m. Using Soderberg method, calculate the factor of safety. Assume suitable values for any other data needed.

Solution. Given: $d = 50$ mm ; $\sigma_u = 630$ MPa $= 630$ N/mm^2 ; $T_{max} = 2000$ N-m ; $T_{min} = -800$ N-m

We know that the mean or average torque,

$$T_m = \frac{T_{max} + T_{min}}{2} = \frac{2000 + (-800)}{2} = 600 \text{ N-m} = 600 \times 10^3 \text{ N-mm}$$

∴ Mean or average shear stress,

$$\tau_m = \frac{16\, T_m}{\pi d^3} = \frac{16 \times 600 \times 10^3}{\pi (50)^3} = 24.4 \text{ N/mm}^2 \qquad \ldots \left(\because\ T = \frac{\pi}{16} \times \tau \times d^3\right)$$

Variable torque,

$$T_v = \frac{T_{max} - T_{min}}{2} = \frac{2000 - (-800)}{2} = 1400 \text{ N-m} = 1400 \times 10^3 \text{ N-mm}$$

$$\therefore \text{ Variable shear stress, } \tau_v = \frac{16\, T_v}{\pi d^3} = \frac{16 \times 1400 \times 10^3}{\pi (50)^3} = 57 \text{ N/mm}^2$$

Since the endurance limit in reversed bending (σ_e) is taken as one-half the ultimate tensile strength (*i.e.* $\sigma_e = 0.5\, \sigma_u$) and the endurance limit in shear (τ_e) is taken as $0.55\, \sigma_e$, therefore

$$\tau_e = 0.55\, \sigma_e = 0.55 \times 0.5\, \sigma_u = 0.275\, \sigma_u$$
$$= 0.275 \times 630 = 173.25 \text{ N/mm}^2$$

Assume the yield stress (σ_y) for carbon steel in reversed bending as 510 N/mm^2, surface finish factor (K_{sur}) as 0.87, size factor (K_{sz}) as 0.85 and fatigue stress concentration factor (K_{fs}) as 1.

Since the yield stress in shear (τ_y) for shear loading is taken as one-half the yield stress in reversed bending (σ_y), therefore

$$\tau_y = 0.5\, \sigma_y = 0.5 \times 510 = 255 \text{ N/mm}^2$$

Let $F.S.$ = Factor of safety.

We know that according to Soderberg's formula.

$$\frac{1}{F.S.} = \frac{\tau_m}{\tau_y} + \frac{\tau_v \times K_{fs}}{\tau_e \times K_{sur} \times K_{sz}} = \frac{24.4}{255} + \frac{57 \times 1}{173.25 \times 0.87 \times 0.85}$$
$$= 0.096 + 0.445 = 0.541$$

$$\therefore \quad F.S. = 1/0.541 = 1.85 \quad \textbf{Ans.}$$

Problem:

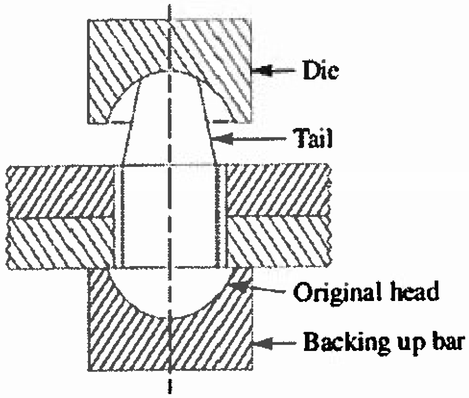
and according to Soderberg's formula,

$$\frac{1}{F.S.} = \frac{\sigma_m}{\sigma_y} + \frac{\sigma_v \times K_f}{\sigma_e \times K_{sur} \times K_{sz}}$$

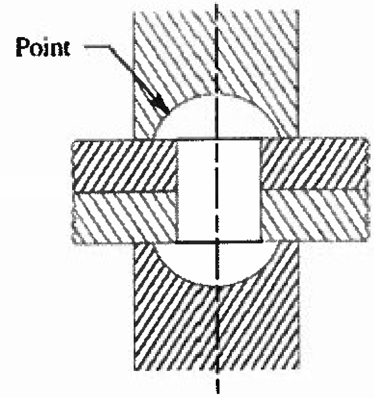
$$\frac{1}{1.3} = \frac{0.0147P}{500} + \frac{0.0088P \times 1}{330 \times 0.9 \times 0.85} = \frac{29.4P}{10^6} + \frac{34.8P}{10^6} = \frac{64.2P}{10^6}$$

$$\therefore \quad P = \frac{1}{1.3} \times \frac{10^6}{64.2} = 11\,982 \text{ N} = 11.982 \text{ kN}$$

From the above, we find that maximum value of $P = 13.785$ kN Ans.



(a) Initial position.



(b) Final position.

Fig. Methods of riveting.

The plates are drilled together and then separated to remove any burrs or chips so as to have a tight flush joint between the plates. A cold rivet or a red hot rivet is introduced into the plates and the **point** (i.e. second head) is then formed. When a cold rivet is used, the process is known as **cold riveting** and when a hot rivet is used, the process is known as **hot riveting.** The cold riveting process is used for structural joints while hot riveting is used to make leak proof joints.

The riveting may be done by hand or by a riveting machine. In hand riveting, the original rivet head is backed up by a hammer or heavy bar and then the die or set, as shown in Fig.(a), is placed against the end to be headed and the blows are applied by a hammer. This causes the shank to expand thus filling the hole and the tail is converted into a **point** as shown in Fig.(b). As the rivet cools, it tends to contract. The lateral contraction will be slight, but there will be a longitudinal tension introduced in the rivet which holds the plates firmly together.

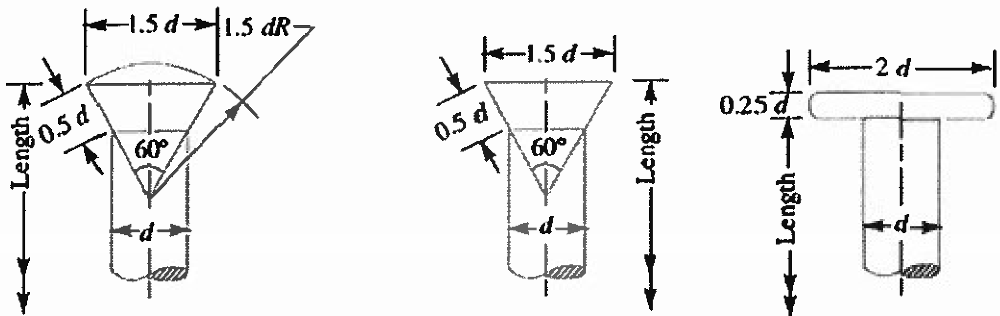
In machine riveting, the die is a part of the hammer which is operated by air, hydraulic or steam pressure.

Notes:

1. For steel rivets up to 12 mm diameter, the cold riveting process may be used while for larger diameter rivets, hot riveting process is used.

2. In case of long rivets, only the tail is heated and not the whole shank.

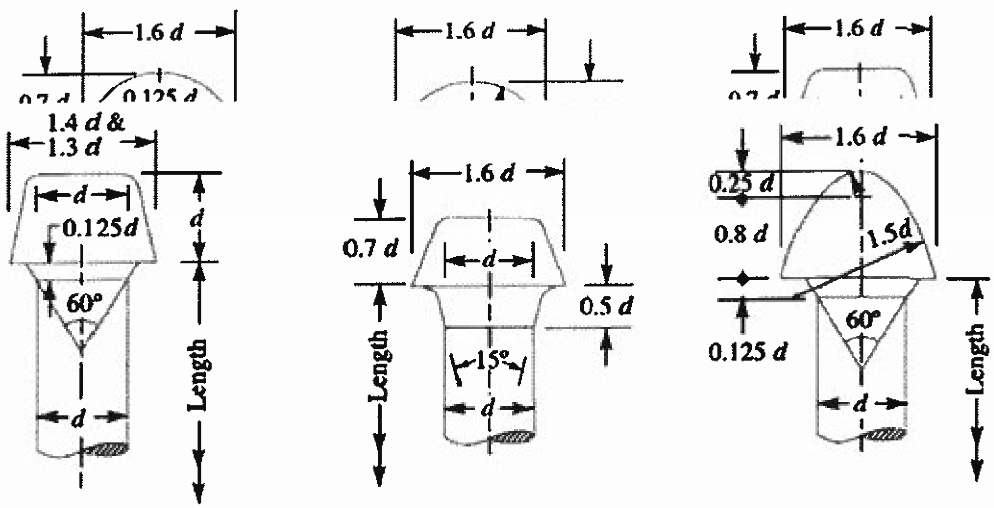
Types of Rivet Heads



(*d*) Round counter sunk head 60°. (*e*) Flat counter sunk head 60°. (*f*) Flat head.

Fig. Rivet heads for general purposes (from 12 mm to 48 mm diameter)

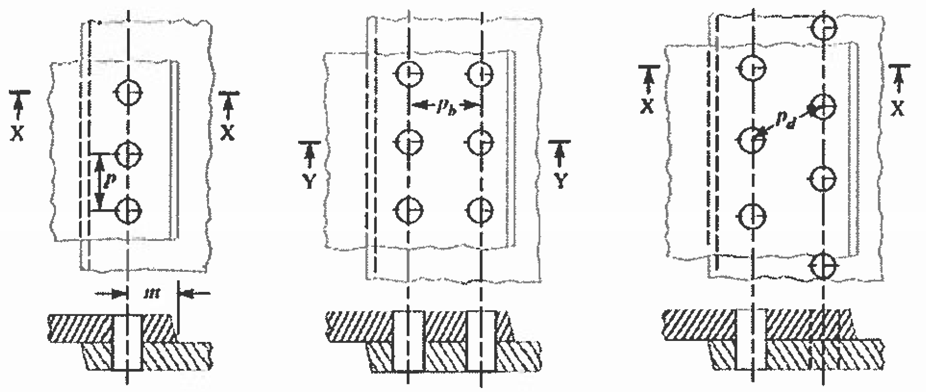
3. Rivet heads for boiler work (from 12 mm to 48 mm diameter, as shown in Fig.



1.4 d for rivets under 24 mm. (e) Pan head with tapered neck. (f) Steeple head.

Types of Riveted Joints

Following are the two types of riveted joints, depending upon the way in which the plates are connected.



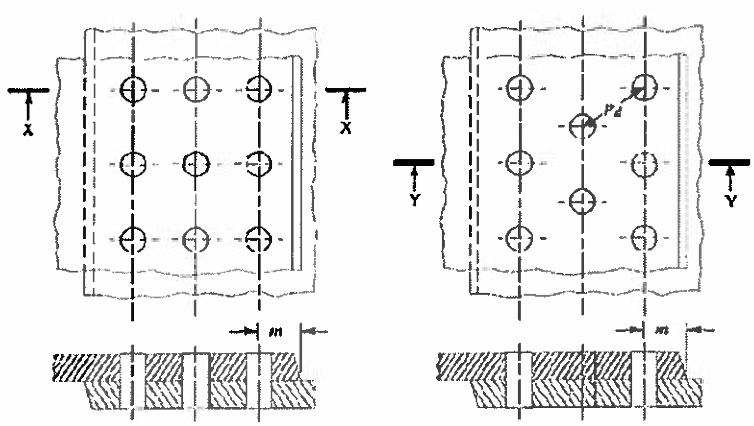
(a) Single riveted lap joint. (b) Double riveted lap joint (Chain riveting). (c) Double riveted lap Joint (Zig-zag riveting).

Fig. Single and double riveted lap joints.

Similarly the joints may be **triple riveted** or **quadruple riveted.**

Notes: 1. when the rivets in the various rows are opposite to each other, as shown in Fig. (b), then the joint is said to be **chain riveted.** On the other hand, if the rivets in the adjacent rows are staggered in such a way that every rivet is in the middle of the two rivets of the opposite row as shown in Fig. (c), then the joint is said to be **zig-zag riveted.**

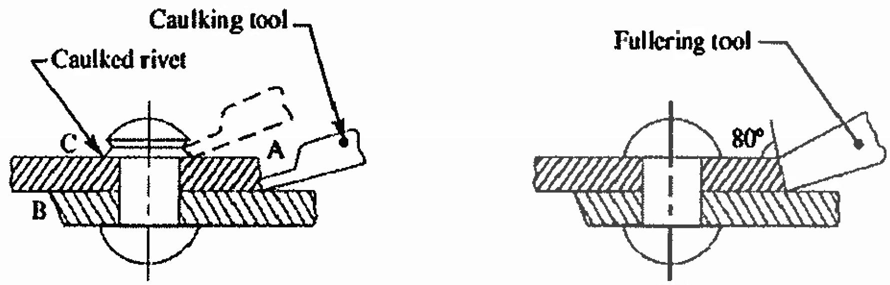
2. Since the plates overlap in lap joints, therefore the force P, P acting on the plates are not in the same straight line but they are at a distance equal to the thickness of the plate. These forces will form a couple which may bend the joint. Hence the lap joints may be used only where small loads are to be transmitted. On the other hand, the forces P, P in a butt joint act in the same straight line, therefore there will be no couple. Hence the butt joints are used where heavy loads are to be transmitted.



(a) Chain riveting. (b) Zig-zag riveting.

Fig. 9.7. Triple riveted lap joint.

turned down with a caulking tool to make a joint steam tight. A great care is taken to prevent injury to the plate below the tool.



(*a*) Caulking. (*b*) Fullering.

Fig.2. Caulking and fullering.

A more satisfactory way of making the joints staunch is known as **fullering** which has largely superseded caulking. In this case, a fullering tool with a thickness at the end equal to that of the plate is used in such a way that the greatest pressure due to the blows occur near the joint, giving a clean finish, with less risk of damaging the plate. A fullering process is shown in Fig. (b).

Failures of a Riveted Joint

A riveted joint may fail in the following ways:

1. Tearing of the plate at an edge. A joint may fail due to tearing of the plate at an edge as shown in Fig.3. This can be avoided by keeping the margin, m = 1.5d, where d is the diameter of the rivet hole.

2. Tearing of the plate across a row of rivets. Due to the tensile stresses in the main plates, the main plate or cover plates may tear off across a row of rivets as shown in Fig. In such cases, we consider only one pitch length of the plate, since every rivet is responsible for that much length of the plate only.

The resistance offered by the plate against tearing is known as **tearing resistance** or **tearing strength** or **tearing value** of the plate.

Let *p* = Pitch of the rivets,

d = Diameter of the rivet hole,

t = Thickness of the plate, and

σ_t = Permissible tensile stress for the plate material.

Let d = Diameter of the rivet hole,

τ = Safe permissible shear stress for the rivet material, and

n = Number of rivets per pitch length.

We know that shearing area,

$$A_s = \frac{\pi}{4} \times d^2 \quad \text{...(In single shear)}$$

$$= 2 \times \frac{\pi}{4} \times d^2 \quad \text{... (Theoretically, in double shear)}$$

$$= 1.875 \times \frac{\pi}{4} \times d^2 \quad \text{... (In double shear, according to Indian Boiler Regulations)}$$

∴ Shearing resistance or pull required to shear off the rivet per pitch length,

$$P_s = n \times \frac{\pi}{4} \times d^2 \times \tau \quad \text{... (In single shear)}$$

$$= n \times 2 \times \frac{\pi}{4} \times d^2 \times \tau \quad \text{... (Theoretically, in double shear)}$$

As we discussed earlier, when the shearing takes place at one cross-section of the rivet, then the rivets are said to be in **single shear.** Similarly, when the shearing takes place at two cross-sections of the rivet, then the rivets are said to be in **double shear.**

$$= n \times 1.875 \times \frac{\pi}{4} \times d^2 \times \tau \quad \text{... (In double shear, according to Indian Boiler Regulations)}$$

When the shearing resistance (P_s) is greater than the applied load (P) per pitch length, then this type of failure will occur.

4. Crushing of the plate or rivets. Sometimes, the rivets do not actually shear off under the tensile stress, but are crushed as shown in Fig. Due to this, the rivet hole becomes of an oval shape and hence the joint becomes loose. The failure of rivets in such a manner is also known as **bearing failure.** The area which resists this action is the projected area of the hole or rivet on diametric plane.

The resistance offered by a rivet to be crushed is known as **crushing resistance** or **crushing strength** or **bearing value** of the rivet.

Let d = Diameter of the rivet hole,

t = Thickness of the plate,

σ_c = Safe permissible crushing stress for the rivet or plate material, and

Strength of the un-riveted or solid plate per pitch length,

$$P = p \cdot t \cdot \sigma_t$$

$\therefore$ Efficiency of the riveted joint,

$$\eta = \frac{\text{Least of } P_t,\ P_s \text{ and} P_c}{p \times t \times \sigma_t}$$

Where p = Pitch of the rivets,

t = Thickness of the plate, and

σ_t = Permissible tensile stress of the plate material.

Find the efficiency of the following riveted joints:

1. Single riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 50 mm. 2. Double riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 65 mm. Assume Permissible tensile stress in plate = 120 MPa Permissible shearing stress in rivets = 90 MPa Permissible crushing stress in rivets = 180 MPa.

Solution. Given : $t = 6$ mm ; $d = 20$ mm ; $\sigma_t = 120$ MPa $= 120$ N/mm^2 ; $\tau = 90$ MPa $= 90$ N/mm^2 ; $\sigma_c = 180$ MPa $= 180$ N/mm^2

1. *Efficiency of the first joint*

Pitch, $p = 50$ mm ...(Given)

First of all, let us find the tearing resistance of the plate, shearing and crushing resistances of the rivets.

(i) Tearing resistance of the plate

We know that the tearing resistance of the plate per pitch length,

$$P_t = (p - d)\, t \times \sigma_t = (50 - 20)\, 6 \times 120 = 21\,600 \text{ N}$$

(ii) Shearing resistance of the rivet

Since the joint is a single riveted lap joint, therefore the strength of one rivet in single shear is taken. We know that shearing resistance of one rivet,

$$P_s = \frac{\pi}{4} \times d^2 \times \tau = \frac{\pi}{4}\, (20)^2\, 90 = 28\,278 \text{ N}$$

(iii) Crushing resistance of the rivet

Since the joint is a single riveted, therefore strength of one rivet is taken. We know that crushing resistance of one rivet,

$$P_c = d \times t \times \sigma_c = 20 \times 6 \times 180 = 21\,600 \text{ N}$$

$\therefore$ Strength of the joint

$$= \text{Least of } P_t, P_s \text{ and } P_c = 21\,600 \text{ N}$$

We know that strength of the unriveted or solid plate,

$$P = p \times t \times \sigma_t = 50 \times 6 \times 120 = 36\,000 \text{ N}$$

$\therefore$ Efficiency of the joint,

$$\eta = \frac{\text{Least of } P_t, P_s \text{ and } P_c}{P} = \frac{21\,600}{36\,000} = 0.60 \text{ or } 60\% \quad \textbf{Ans.}$$

2. *Efficiency of the second joint*

Pitch, $p = 65$ mm ...(Given)

(i) Tearing resistance of the plate,

We know that the tearing resistance of the plate per pitch length,

$$P_t = (p - d)\, t \times \sigma_t = (65 - 20)\, 6 \times 120 = 32\,400 \text{ N}$$

Design of boiler joints according to IBR

Design of Boiler Joints

The boiler has a longitudinal joint as well as circumferential joint. The ***longitudinal joint*** is used to join the ends of the plate to get the required diameter of a boiler. For this purpose, a butt joint with two cover plates is used. The ***circumferential joint*** is used to get the required length of the boiler. For this purpose, a lap joint with one ring overlapping the other alternately is used.

Since a boiler is made up of number of rings, therefore the longitudinal joints are staggered for convenience of connecting rings at places where both longitudinal and circumferential joints occur.

Design of Longitudinal Butt Joint for a Boiler

According to Indian Boiler Regulations (I.B.R), the following procedure should be adopted for the design of longitudinal butt joint for a boiler.

1. *Thickness of boiler shell.* First of all, the thickness of the boiler shell is determined by using the thin cylindrical formula, *i.e.*

$$t = \frac{P.D}{2\,\sigma_t \times \eta_l} + 1 \text{ mm as corrosion allowance}$$

Where t = Thickness of the boiler shell,

P = Steam pressure in boiler,

D = Internal diameter of boiler shell,

σ_t = Permissible tensile stress, and

η_l = Efficiency of the longitudinal joint.

The following points may be noted:

(a) The thickness of the boiler shell should not be less than 7 mm.

(b) The efficiency of the joint may be taken from the following table.

Indian Boiler Regulations (I.B.R.) allows a maximum efficiency of 85% for the best joint.

(c) According to I.B.R., the factor of safety should not be less than 4.

2. *Diameter of rivets.* After finding out the thickness of the boiler shell (t), the diameter of the rivet hole (d) may be determined by using Unwin's empirical formula,

i.e. $d = 6\,t$ (when t is greater than 8 mm)

But if the thickness of plate is less than 8 mm, then the diameter of the rivet hole may be calculated by equating the shearing resistance of the rivets to crushing resistance. In no case,

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

For double butt straps of equal width having every alternate rivet in the outer rows being omitted.

(c) For unequal width of butt straps, the thicknesses of butt strap are

$t_1 = 0.75\, t$, for wide strap on the inside, and

$t_1 = 0.625\, t$, for narrow strap on the outside.

6. *Margin.* The margin (m) is taken as $1.5\, d$.

Note: The above procedure may also be applied to ordinary riveted joints.

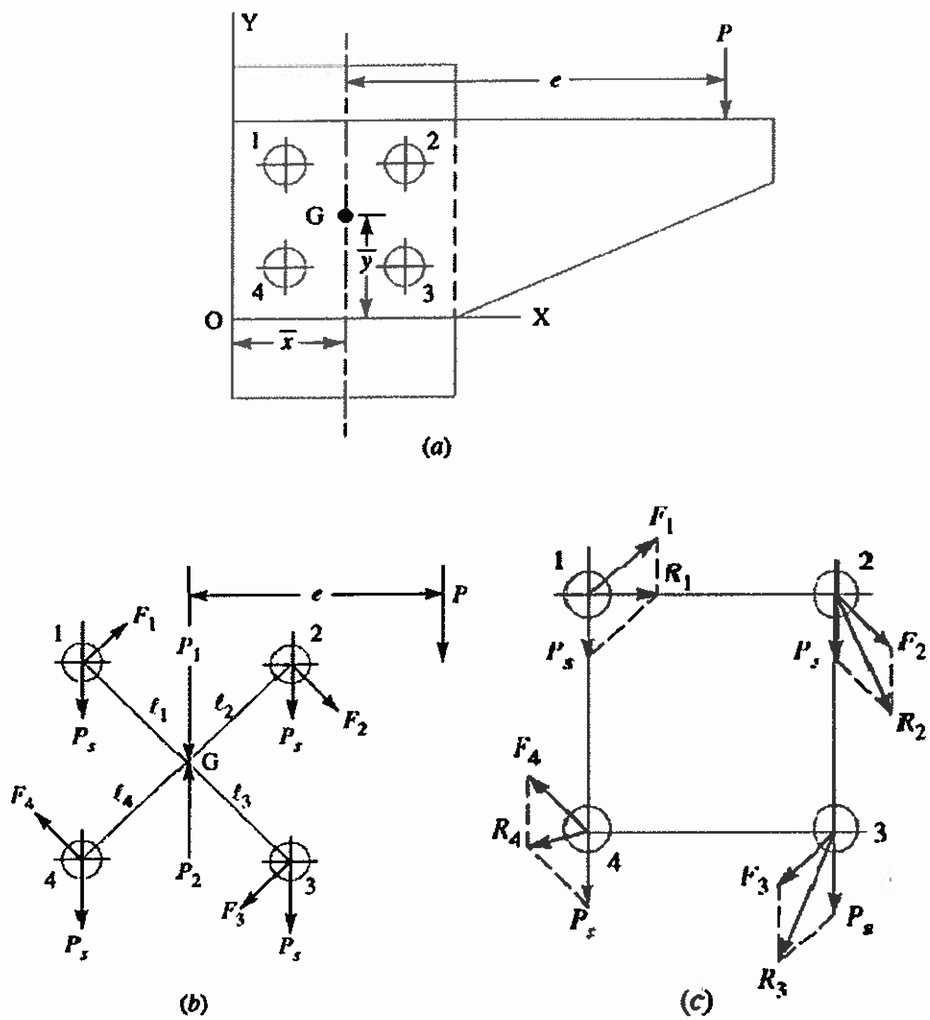


Fig. 1. Eccentric loaded riveted joint.

2. Introduce two forces *P*1 and *P*2 at the centre of gravity '*G*' of the rivet system. These forces are equal and opposite to *P* as shown in Fig.(*b*).

3. Assuming that all the rivets are of the same size, the effect of $P_1 = P$ is to produce direct shear load on each rivet of equal magnitude. Therefore, direct shear load on each rivet,

$$P_s = \frac{P}{n} \text{ acting parallel to the load P,}$$

4. The effect of $P_2 = P$ is to produce a turning moment of magnitude $P \times e$ which tends to rotate the joint about the centre of gravity '*G*' of the rivet system in a clockwise direction. Due to the turning moment, secondary shear load on each rivet is produced. In order to find the secondary shear load, the following two assumptions are made:

$$R = \sqrt{(P_s)^2 + F^2 + 2Ps \times F \times \cos\theta}$$

Where θ = Angle between the primary or direct shear load (P_s)

And secondary shear load (F).

When the secondary shear load on each rivet is equal, then the heavily loaded rivet will be one in which the included angle between the direct shear load and secondary shear load is minimum. The maximum loaded rivet becomes the critical one for determining the strength of the riveted joint. Knowing the permissible shear stress (τ), the diameter of the rivet hole may be obtained by using the relation,

$$\text{Maximum resultant shear load } (R) = \frac{\pi}{4} \times d^2 \times \tau$$

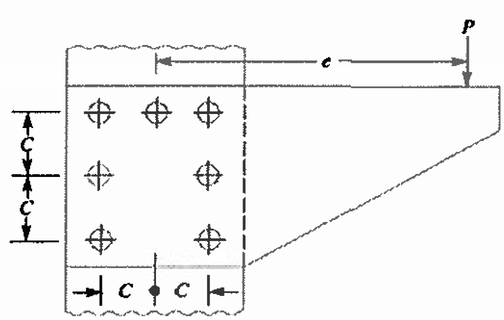
From DDB, the standard diameter of the rivet hole (d) and the rivet diameter may be specified

Notes : 1. In the solution of a problem, the primary and shear loads may be laid off approximately to scale and generally the rivet having the maximum resultant shear load will be apparent by inspection. The values of the load for that rivet may then be calculated.

2. When the thickness of the plate is given, then the diameter of the rivet hole may be checked against crushing.

3. When the eccentric load P is inclined at some angle, then the same procedure as discussed above may be followed to find the size of rivet.

Problem: An eccentrically loaded lap riveted joint is to be designed for a steel bracket as shown in Fig. 2. The bracket plate is 25 mm thick. All rivets are to be of the same size. Load on the bracket, P = 50 kN ; rivet spacing, C = 100 mm; load arm, e = 400 mm. Permissible shear stress is 65 MPa and crushing stress is 120 MPa. Determine the size of the rivets to be used for the joint.



$$P_s = \frac{P}{n} = \frac{50 \times 10^3}{7} = 7143 \text{ N}$$

The direct shear load acts parallel to the direction of load P *i.e.* vertically downward as shown in Fig. 2. Turning moment produced by the load P due to eccentricity (e)

$$= P \times e = 50 \times 103 \times 400 = 20 \times 10^6 \text{ N-mm}$$

This turning moment is resisted by seven rivets as shown in Fig.2.

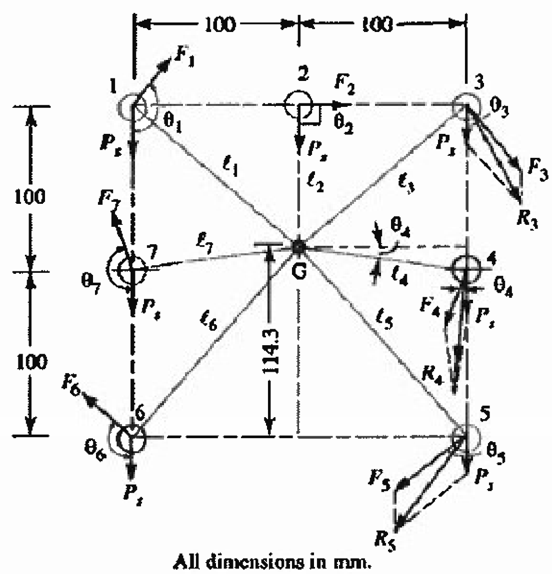


Fig. 3

Let F_1, F_2, F_3, F_4, F_5, F_6 and F_7 be the secondary shear load on the rivets 1, 2, 3, 4, 5, 6 and 7 placed at distances l_1, l_2, l_3, l_4, l_5, l_6 and l_7 respectively from the centre of gravity of the rivet system as shown in Fig. 3.

From the geometry of the figure, we find that

$$l_1 = l_3 = \sqrt{(100)^2 + (200 - 114.3)^2} = 131.7 \text{ mm}$$

$$l_2 = 200 - 114.3 = 85.7 \text{ mm}$$

$$l_4 = l_7 = \sqrt{(100)^2 + (114.3 - 100)^2} = 101 \text{ mm}$$

$$l_5 = l_6 = \sqrt{(100)^2 + (114.3)^2} = 152 \text{ mm}$$

Now equating the turning moment due to eccentricity of the load to the resisting moment of the rivets, we have

$$R_4 = \sqrt{(P_s)^2 + (F_4)^2 + 2\ P_s \times F_4 \times \cos\theta_4}$$

$$= \sqrt{(7143)^2 + (18\ 593)^2 + 2 \times 7143 \times 18\ 593 \times 0.99} = 25\ 684\ \text{N}$$

And resultant shear load on rivet 5,

$$R_5 = \sqrt{(P_s)^2 + (F_5)^2 + 2\ P_s \times F_5 \times \cos\theta_5}$$

$$= \sqrt{(7143)^2 + (27\ 981)^2 + 2 \times 7143 \times 27\ 981 \times 0.658} = 33\ 121\ \text{N}$$

The resultant shear load may be determined graphically, as shown in Fig.3.
From above we see that the maximum resultant shear load is on rivet 5. If d is the diameter of rivet hole, then maximum resultant shear load (R_5),

$$33\ 121 = \frac{\pi}{4} \times d^2 \times \tau = \frac{\pi}{4} \times d^2 \times 65 = 51\ d^2$$
$$d^2 = 33\ 121 / 51 = 649.4 \quad \text{or} \quad d = 25.5\ \text{mm}$$

From DDB, we see that according the standard diameter of the rivet hole (d) is 25.5 mm and the corresponding diameter of rivet is 24 mm.
Let us now check the joint for crushing stress. We know that

$$\text{Crushing stress} = \frac{\text{Max. load}}{\text{Crushing area}} = \frac{R_5}{d \times t} = \frac{33\ 121}{25.5 \times 25}$$
$$= 51.95\ \text{N/mm}^2 = 51.95\ \text{MPa}$$

Since this stress is well below the given crushing stress of 120 MPa, therefore the design is satisfactory.

1. Since there is an uneven heating and cooling during fabrication, therefore the members may get distorted or additional stresses may develop.

2. It requires a highly skilled labour and supervision.

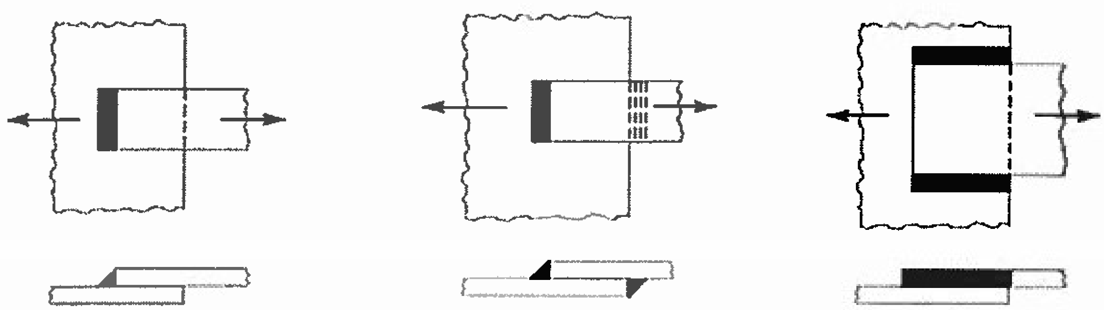
3. Since no provision is kept for expansion and contraction in the frame, therefore there is a possibility of cracks developing in it.

4. The inspection of welding work is more difficult than riveting work.

Types of Welded Joints

Following two types of welded joints are important from the subject point of view:

1. Lap joint or fillet joint, and **2.** Butt joint.



(*a*) **Single transverse.** (*b*) **Double transverse.** (c) **Parallel fillet.**

Fig.1. Types of Lab and Butt Joints

Lap Joint

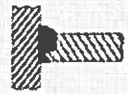
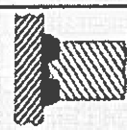
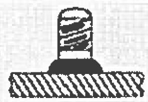
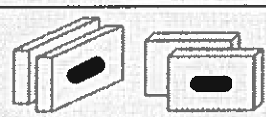
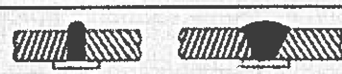
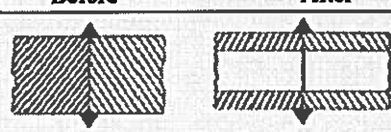
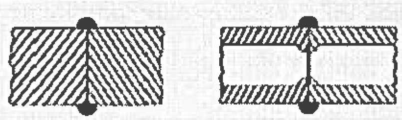
The lap joint or the fillet joint is obtained by overlapping the plates and then welding the edges of the plates. The cross-section of the fillet is approximately triangular. The fillet joints may be

1. Single transverse fillet, **2.** Double transverse fillet and **3.** Parallel fillet joints.

The fillet joints are shown in Fig.1. A single transverse fillet joint has the disadvantage that the edge of the plate which is not welded can buckle or warp out of shape.

Butt Joint

The butt joint is obtained by placing the plates edge to edge as shown in Fig.2. In butt welds, the plate edges do not require beveling if the thickness of plate is less than 5 mm. On the other hand, if the plate thickness is 5 mm to 12.5 mm, the edges should be beveled to V or U-groove on both sides.

S. No.	*Form of weld*	*Sectional representation*	*Symbol*
9.	Single-*J* butt		
10.	Double-*J* butt		
11.	Bead (edge or seal)		
12.	Stud		⊥
13.	Sealing run		○
14.	Spot		
15.	Seam		
16.	Mashed seam		
17.	Plug		
18.	Backing strip		=
19.	Stitch		
20.	Projection		△
21.	Flash		
22.	Butt resistance or pressure (upset)		|

Elements of a welding symbol

Elements of a Welding Symbol

A welding symbol consists of the following eight elements:

1. Reference line, **2.** Arrow,

3. Basic weld symbols, **4.** Dimensions and other data,

5. Supplementary symbols, **6.** Finish symbols,

7. Tail, and **8.** Specification, process or other references.

Standard Location of Elements of a Welding Symbol

The arrow points to the location of weld, the basic symbols with dimensions are located on one or both sides of reference line. The specification if any is placed in the tail of arrow. Fig. 1. shows the standard locations of welding symbols represented on drawing.

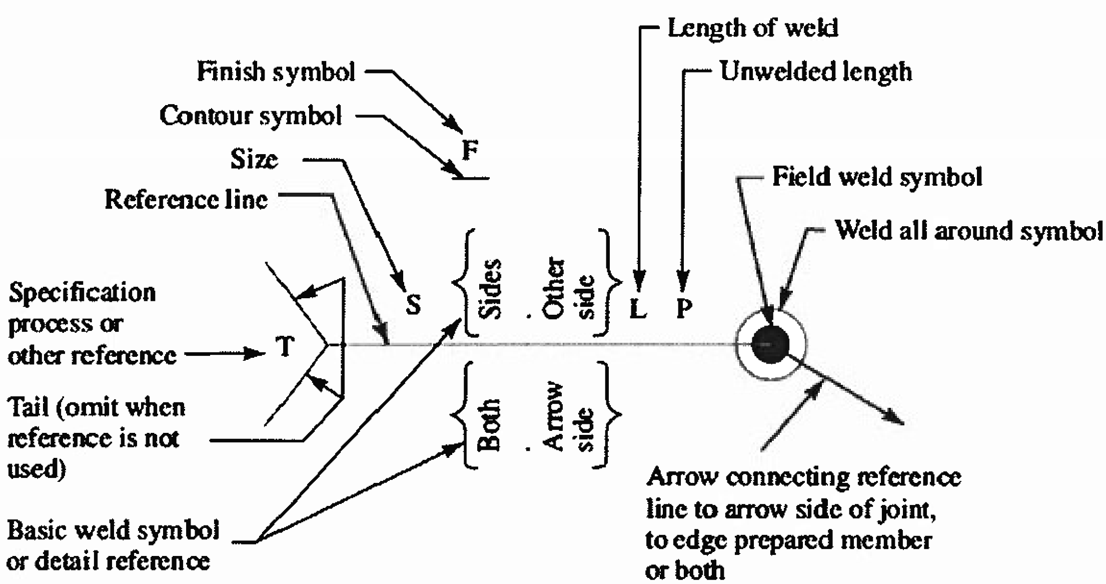


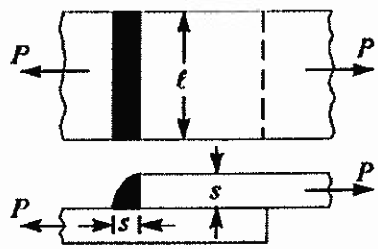
Fig.1 Standard location of weld symbols.

Some of the examples of welding symbols represented on drawing are shown in the following table.

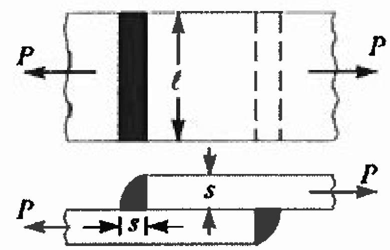
Contents: Design of Welded Joints

Strength of Transverse Fillet Welded Joints

We have already discussed that the fillet or lap joint is obtained by overlapping the plates and then welding the edges of the plates. The transverse fillet welds are designed for tensile strength. Let us consider a singl and double transverse fillet welds as shown in Fig. 1(*a*) and (*b*) respectively.



(*a*) Single transverse fillet weld.



(*b*) Double transverse fillet weld.

Fig.1 Transverse fillet welds.

The length of each side is known as ***leg*** or ***size of the weld*** and the perpendicular distance of the hypotenuse from the intersection of legs (*i.e. BD*) is known as ***throat thickness.*** The minimum area of the weld is obtained at the throat *BD*, which is given by the product of the throat thickness and length of weld.

Let t = Throat thickness (BD),

s = Leg or size of weld,

= Thickness of plate, and

l = Length of weld,

From Fig.2, we find that the throat thickness,

$$t = s \times \sin 45° = 0.707\, s$$

Therefore, Minimum area of the weld or throat area,

$$A = \text{Throat thickness} \times \text{Length of weld}$$
$$= t \times l = 0.707\, s \times l$$

If σ_t is the allowable tensile stress for the weld metal, then the tensile strength of the joint for single fillet weld,

$$P = \text{Throat area} \times \text{Allowable tensile stress} = 0.707\, s \times l \times \sigma_t$$

And tensile strength of the joint for double fillet weld,

$$P = 2 \times 0.707\, s \times l \times \sigma_t = 1.414\, s \times l \times \sigma_t$$

Solution. Given: *Width = 100 mm ; Thickness = 10 mm ; P = 80 kN = 80×10^3 N ; τ = 55 MPa = 55 N/mm^2

Let l = Length of weld, and

s = Size of weld = Plate thickness = 10 mm ... (Given)

We know that maximum load which the plates can carry for double parallel fillet weld (P),

$$80 \times 10^3 = 1.414 \times s \times l \times \tau = 1.414 \times 10 \times l \times 55 = 778\, l$$

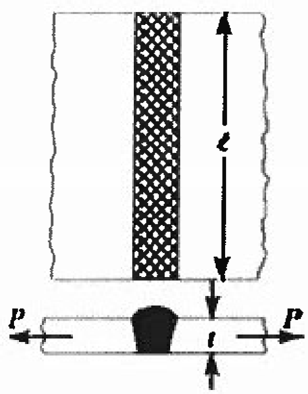
$$\therefore \quad l = 80 \times 10^3 / 778 = 103 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

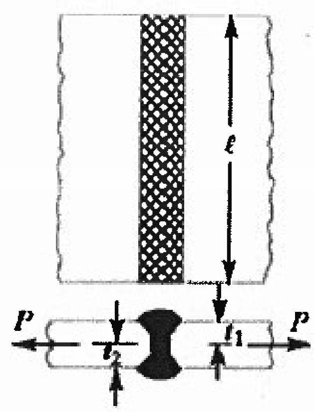
$$l = 103 + 12.5 = 115.5 \text{ mm} \quad \textbf{Ans.}$$

Strength of Butt Joints

The butt joints are designed for tension or compression. Consider a single V-butt joint as shown in Fig. 4(a).



(a) Single V-butt joint.



(b) Double V-butt joint.

Fig.4. Butt Joints

In case of butt joint, the length of leg or size of weld is equal to the throat thickness which is equal to thickness of plates. Therefore, Tensile strength of the butt joint (single-V or square butt joint),

$$P = t \times l \times \sigma_t$$

Where l = Length of weld. It is generally equal to the width of plate. And tensile strength for double-V butt joint as shown in Fig. 4(b) is given by

$$P = (t_1 + t_2)\, l \times \sigma_t$$

Where t_1 = Throat thickness at the top, and

t_2 = Throat thickness at t e bottom.

Problem:

A plate 100 mm wide and 12.5 mm thick is to be welded to another plate by means of parallel fillet welds. The plates are subjected to a load of 50 kN. Find the length of the weld so that the maximum stress does not exceed 56 MPa. Consider the joint first under static loading and then under fatigue loading.

Solution. Given: *Width = 100 mm ; Thickness = 12.5 mm ; $P = 50$ kN $= 50 \times 10^3$ N ; $\tau = 56$ MPa $= 56$ N/mm^2

Length of weld for static loading

Let l = Length of weld, and

s = Size of weld = Plate thickness

= 12.5 mm ... (Given)

We know that the maximum load which the plates can carry for double parallel fillet welds (P),

$$50 \times 10^3 = 1.414\, s \times l \times \tau$$

$$= 1.414 \times 12.5 \times l \times 56 = 990\, l$$

$$\therefore \quad l = 50 \times 10^3 / 990 = 50.5 \text{ mm}$$

Adding 12.5 mm for starting and stopping of weld run, we have

$$l = 50.5 + 12.5 = 63 \text{ mm} \quad \textbf{Ans.}$$

Length of weld for fatigue loading

From Table 10.6, we find that the stress concentration factor for parallel fillet welding is 2.7.

$\therefore$ Permissible shear stress,

$$\tau = 56 / 2.7 = 20.74 \text{ N/mm}^2$$

We know that the maximum load which the plates can carry for double parallel fillet welds (P),

$$50 \times 10^3 = 1.414\, s \times l \times \tau = 1.414 \times 12.5 \times l \times 20.74 = 367\, l$$

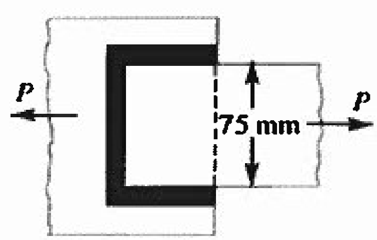
$$\therefore \quad l = 50 \times 10^3 / 367 = 136.2 \text{ mm}$$

Adding 12.5 for starting and stopping of weld run, we have

$$l = 136.2 + 12.5 = 148.7 \text{ mm} \quad \textbf{Ans.}$$

Problem:

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



Contents: Special fillet welded joints

Special Cases of Fillet Welded Joints

The following cases of fillet welded joints are important from the subject point of view.

1. *Circular fillet weld subjected to torsion.* Consider a circular rod connected to a rigid plate by a fillet weld as shown in Fig. 1.

Let d = Diameter of rod,

r = Radius of rod,

T = Torque acting on the rod,

s = Size (or leg) of weld,

t = Throat thickness,

J = Polar moment of inertia of the weld section $= \dfrac{\pi t d^3}{4}$

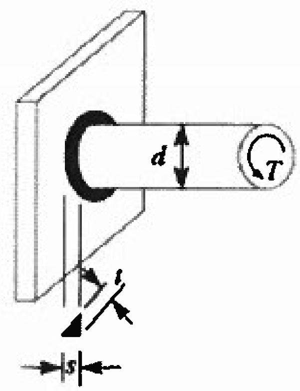


Fig. 1. Circular fillet weld subjected to torsion.

We know that shear stress for the material,

$$\tau = \frac{T.r}{J} = \frac{T \times d/2}{J}$$

$$= \frac{T \times d/2}{\pi t d^3/4} = \frac{2T}{\pi t d^2}$$

This shear stress occurs in a horizontal plane along a leg of the fillet weld. The maximum shear occurs on the throat of weld which is inclined at 45° to the horizontal plane.

Length of throat, $t = s \sin 45° = 0.707\, s$ and maximum shear stress,

$$\tau_{max} = \frac{2T}{\pi \times 0.707\, s \times d^2} = \frac{2.83\, T}{\pi\, s\, d^2}$$

2. Circular fillet weld subjected to bending moment. Consider a circular rod connected to a rigid plate by a fillet weld as shown in Fig.2.

Let d = Diameter of rod,

M = Bending moment acting on the rod,

s = Size (or leg) of weld,

t = Throat thickness,

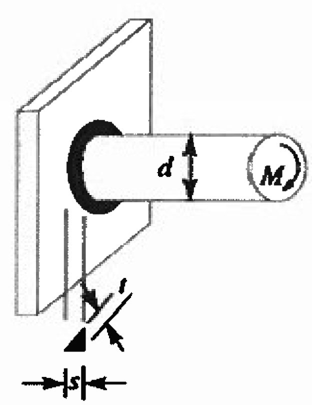


Fig.2.Circular fillet weld subjected to Bending moment.

The maximum shear stress occurs at the throat and is given by

$$\tau_{max} = \frac{3T}{0.707\, s \times l^2} = \frac{4.242\, T}{s \times l^2}$$

Section modulus of the weld metal through the throat,

$$Z = \frac{t \times l^2}{6} \times 2 \qquad \text{...(For both sides weld)}$$

$$= \frac{0.707 s \times l^2}{6} \times 2 = \frac{s \times l^2}{4.242}$$

Bending moment, $M = P \times e$

$$\therefore \text{ Bending stress, } \sigma_b = \frac{M}{Z} = \frac{P \times e \times 4.242}{s \times l^2} = \frac{4.242\, P \times e}{s \times l^2}$$

We know that the maximum normal stress,

$$\sigma_{t(max)} = \frac{1}{2}\sigma_b + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

And maximum shear stress,

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

Case 2

When a welded joint is loaded eccentrically as shown in Fig.2, the following two types of the stresses are induced:

1. Direct or primary shear stress, and

2. Shear stress due to turning moment.

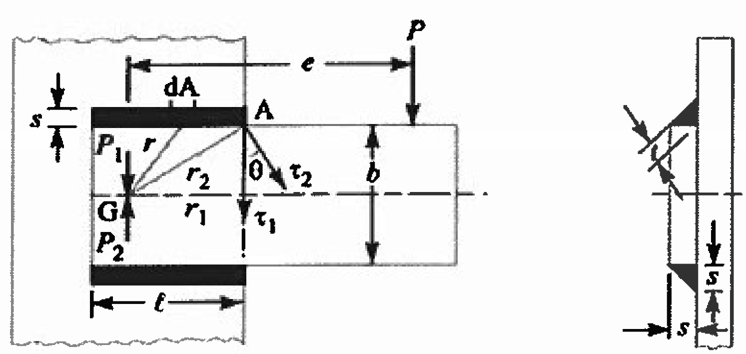


Fig.2 eccentrically loaded welded joint.

$$T = P \times e = \int \frac{\tau_2}{r_2} \times dA \times r^2 = \frac{\tau_2}{r_2} \int dA \times r^2$$

$$= \frac{\tau_2}{r_2} \times J \qquad \left(\because J = \int dA \times r^2\right)$$

Where J = Polar moment of inertia of the throat area about G.

□Shear stress due to the turning moment i.e. secondary shear stress,

$$\tau_2 = \frac{T \times r_2}{J} = \frac{P \times e \times r_2}{J}$$

In order to find the resultant stress, the primary and secondary shear stresses are combined vectorially.

Resultant shear stress at A,

$$\tau_A = \sqrt{(\tau_1)^2 + (\tau_2)^2 + 2\tau_1 \times \tau_2 \times \cos\theta}$$

θ = Angle between τ_1 and τ_2, and

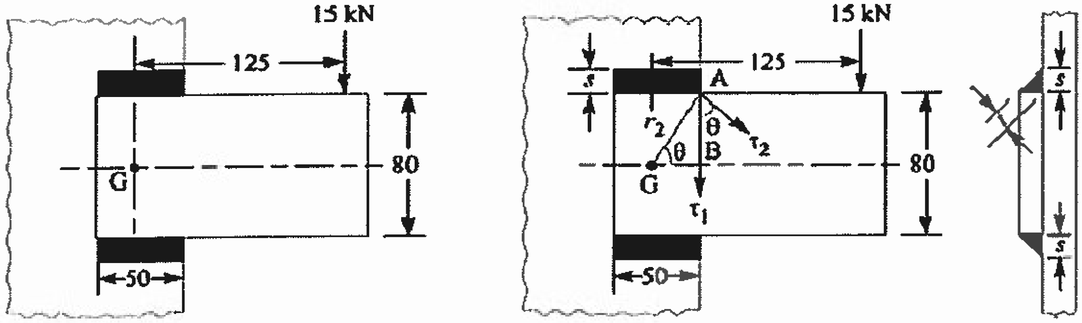
$$\cos\theta = r_1 / r_2$$

∴ Direct or primary shear stress,

$$\tau_1 = \frac{P}{A} = \frac{15 \times 10^3}{70.7\, s} = \frac{212}{s} \text{ N/mm}^2$$

$$J = \frac{t l\,(3b^2 + l^2)}{6} = \frac{0.707\, s \times 50\,[3\,(80)^2 + (50)^2]}{6} \text{ mm}^4$$

$= 127\,850\, s \text{ mm}^4$ ... ($\because t = 0.707\, s$)



All dimensions in mm.

∴ Maximum radius of the weld,

$$r_2 = \sqrt{(AB)^2 + (BG)^2} = \sqrt{(40)^2 + (25)^2} = 47 \text{ mm}$$

Shear stress due to the turning moment *i.e.* secondary shear stress,

$$\tau_2 = \frac{P \times e \times r_2}{J} = \frac{15 \times 10^3 \times 125 \times 47}{127\,850\, s} = \frac{689.3}{s} \text{ N/mm}^2$$

and $$\cos\theta = \frac{r_1}{r_2} = \frac{25}{47} = 0.532$$

We know that resultant shear stress,

$$\tau = \sqrt{(\tau_1)^2 + (\tau_2)^2 + 2\,\tau_1 \times \tau_2 \cos\theta}$$

$$80 = \sqrt{\left(\frac{212}{s}\right)^2 + \left(\frac{689.3}{s}\right)^2 + 2 \times \frac{212}{s} \times \frac{689.3}{s} \times 0.532} = \frac{822}{s}$$

∴ $s = 822 / 80 = 10.3$ mm Ans.

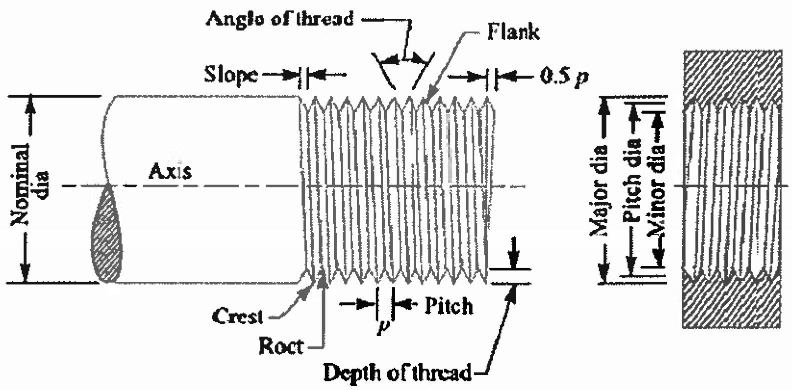


Fig.1 Terms used in screw threads

1. Major diameter. It is the largest diameter of an external or internal screw thread. The screw is specified by this diameter. It is also known as **outside** or **nominal diameter.**

2. Minor diameter. It is the smallest diameter of an external or internal screw thread. It is also known as **core** or **root diameter.**

3. Pitch diameter. It is the diameter of an imaginary cylinder, on a cylindrical screw thread, the surface of which would pass through the thread at such points as to make equal the width of the thread and the width of the spaces between the threads. It is also called an **effective diameter.** In a nut and bolt assembly, it is the diameter at which the ridges on the bolt are in complete touch with the ridges of the corresponding nut.

4. Pitch. It is the distance from a point on one thread to the corresponding point on the next. This is measured in an axial direction between corresponding points in the same axial plane. Mathematically,

$$\text{Pitch} = \frac{1}{\text{No. of threads per unit length of screw}}$$

5. Lead. It is the distance between two corresponding points on the same helix. It may also be defined as the distance which a screw thread advances axially in one rotation of the nut. Lead is equal to the pitch in case of single start threads, it is twice the pitch in double start, thrice the pitch in triple start and so on.

6. Crest. It is the top surface of the thread.

7. Root. It is the bottom surface created by the two adjacent flanks of the thread.

8. Depth of thread. It is the per endicular distance between the crest and root.

9. Flank. It is the surface joining the crest and root.

Stresses in Screwed Fastening due to Static Loading

The following stresses in screwed fastening due to static loading are important from the subject point of view:

1. Internal stresses due to screwing up forces,

2. Stresses due to external forces, and

3. Stress due to combination of stresses at (1) and (2).

Initial Stresses due to Screwing up Forces

The following stresses are induced in a bolt, screw or stud when it is screwed up tightly.

1. Tensile stress due to stretching of bolt. Since none of the above mentioned stresses are accurately determined, therefore bolts are designed on the basis of direct tensile stress with a large factor of safety in order to account for the indeterminate stresses. The initial tension in a bolt, based on experiments, may be found by the relation

$$P_i = 2840 \, d \text{ N}$$

Where Pi = Initial tension in a bolt, and

d = Nominal diameter of bolt, in mm.

The above relation is used for making a joint fluid tight like steam engine cylinder cover joints etc. When the joint is not required as tight as fluid-tight joint, then the initial tension in a bolt may be reduced to half of the above value. In such cases

$$P_i = 1420 \, d \text{ N}$$

The small diameter bolts may fail during tightening, therefore bolts of smaller diameter (less than M 16 or M 18) are not permitted in making fluid tight joints. If the bolt is not initially stressed, then the maximum safe axial load which may be applied to it, is given by

P = Permissible stress × Cross-sectional area at bottom of the thread

$$\textbf{Stress area} = \frac{\pi}{4}\left(\frac{d_p + d_c}{2}\right)^2$$

Where d_p = Pitch diameter, and

d_c = Core or minor diameter.

Stresses due to External Forces

The following stresses are induced in a bolt when it is subjected to an external load.

1. Tensile stress. The bolts, studs and screws usually carry a load in the direction of the bolt axis which induces a tensile stress in the bolt.

Let d_c = Root or core diameter of the thread, and

These stresses should not exceed the safe permissible values of stresses.

a/(1+ *a)* (*i.e. K*) for various type of joints are shown in the following table. The designer thus has control over the influence on the resultant load on a bolt by proportioning the sizes of the connected parts and bolts and by specifying initial tension in the bolt.

Values of K for various types of joints.

Type of joint	$K = \frac{a}{1+a}$
Metal to metal joint with through bolts	**0.00 to 0.10**
Hard copper gasket with long through bolts	**0.25 to 0.50**
Soft copper gasket with long through bolts	**0.50 to 0.75**
Soft packing with through bolts	**0.75 to 1.00**
Soft packing with studs	**1.00**

Design of Cylinder Covers

The cylinder covers may be secured by means of bolts or studs, but studs are preferred. The possible arrangement of securin the cover with bolts and studs is shown in Fig. 2 (*a*) and (*b*) respectively. The bolts or studs, cylinder cover plate and cylinder flange may be designed as discussed below:

1. Design of bolts or studs

In order to find the size and number of bolts or studs, the following procedure may be adopted.

Let D = Diameter of the cylinder,

p = Pressure in the cylinder,

d_c = Core diameter of the bolts or studs,

n = Number of bolts or studs, and

σ_{tb} = Permissible tensile stress for the bolt or stud material.

We know that upward force acting on the cylinder cover,

$$P = \frac{\pi}{4} (D^2)\, p \qquad \ldots(i)$$

This force is resisted by n number of bolts or studs provided on the cover.

Resisting force offered by n number of bolts or studs,

$$P = \frac{\pi}{4} (d_c)^2\, \sigma_{tb} \times n \qquad \ldots(ii)$$

From equations (*i*) and (*ii*), we have

The tightness of the joint also depends upon the circumferential pitch of the bolts or studs. The circumferential pitch should be between 20 d_l and 30 d_l , where d_l is the diameter of the hole in mm for bolt or stud. The pitch circle diameter (D_p) is usually taken as $D + 2t + 3d_l$ and outside diameter of the cover is kept as

$$D_0 = Dp + 3d_l = D + 2t + 6d_1$$

where t = Thickness of the cylinder wall.

2. Design of cylinder cover plate

The thickness of the cylinder cover plate (t_1) and the thickness of the cylinder flange (t_2) may be determined as discussed below:

Let us consider the semi-cover plate as shown in Fig. 3. The internal pressure in the cylinder tries to lift the cylinder cover while the bolts or studs try to retain it in its position. But the centres of pressure of these two loads do not coincide. Hence, the cover plate is subjected to bending stress. The point X is the centre of pressure for bolt load and the point Y is the centre of internal pressure.

We know that the bending moment at A-A,

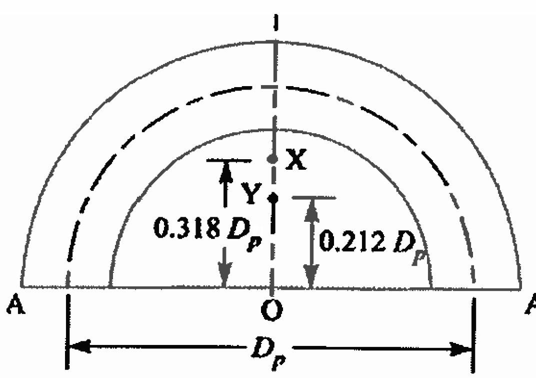


Fig.3

$$M = \frac{\text{Total bolt load}}{2}(OX - OY) = \frac{P}{2}(0.318\ D_P - 0.212\ D_P)$$

$$= \frac{P}{2} \times 0.106\ D_p = 0.053\ P \times D_p$$

$$Z = \frac{1}{6} w\,(t_1)^2$$

Where w = Width of plate

= Outside dia. of cover plate – 2 × dia. of bolt hole

$= D_0 - 2d_1$

Knowing the tensile stress for the cover plate material, the value of t_1 may be determined by using the bending equation,

$$Z = \frac{1}{6} w (t_2)^2$$

Knowing the tensile stress for the cylinder flange material, the value of t_2 may be obtained by using the bending equation *i.e.* $\sigma_t = M / Z$.

Problem:

A mild steel cover plate is to be designed for an inspection hole in the shell of a pressure vessel. The hole is 120 mm in diameter and the pressure inside the vessel is 6 N/mm^2. Design the cover plate along with the bolts. Assume allowable tensile stress for mild steel as 60 MPa and for bolt material as 40 MPa.

Solution. Given : D = 120 mm or r = 60 mm ; p = 6 N/mm^2 ; σ_t = 60 MPa = 60 N/mm^2 ; σ_{tb} = 40 MPa = 40 N/mm^2

First for all, let us find the thickness of the pressure vessel. According to Lame's equation, thickness of the pressure vessel,

$$t = r\left[\sqrt{\frac{\sigma_t + p}{\sigma_t - p}} - 1\right] = 60\left[\sqrt{\frac{60+6}{60-6}} - 1\right] = 6 \text{ mm}$$

Let us adopt $t = 10$ mm

Design of bolts

Let d = Nominal diameter of the bolts.

d_c = Core diameter of the bolts. and

n = Number of bolts.

We know that the total upward force acting on the cover plate (or on the bolts),

$$P = \frac{\pi}{4}(D)^2\, p = \frac{\pi}{4}(120)^2 6 = 67\,867 \text{ N} \qquad ...(i)$$

Let the nominal diameter of the bolt is 24 mm. From Table 11.1 (coarse series), we find that the corresponding core diameter (d_c) of the bolt is 20.32 mm.

∴ Resisting force offered by n number of bolts.

$$P = \frac{\pi}{4}(d_c)^2\, \sigma_{tb} \times n = \frac{\pi}{4}(20.32)^2\, 40 \times n = 67\,867 \text{ N} = 12\,973\, n \text{ N} \qquad ...(ii)$$

From equations (*i*) and (*ii*), we get

$$n = 67\,867 / 12\,973 = 5.23 \text{ say } 6$$

Taking the diameter of the bolt hole (d_1) as 25 mm, we have pitch circle diameter of bolts.

$$D_p = D + 2t + 3d_1 = 120 + 2 \times 10 + 3 \times 25 = 215 \text{ mm}$$

∴ Circumferential pitch of the bolts

$$= \frac{\pi \times D_p}{n} = \frac{\pi \times 215}{6} = 112.6 \text{ mm}$$

We know that for a leak proof joint, the circumferential pitch of the bolts should lie between $20\sqrt{d_1}$ to $30\sqrt{d_1}$, where d_1 is the diameter of the bolt hole in mm.

∴ Minimum circumferential pitch of the bolts

$$= 20\sqrt{d_1} = 20\sqrt{25} = 100 \text{ mm}$$

... (Since, there are two bolts each at distance of L_1 and L_2)

Also the moment due to load W about the tilting edge

$$= W.L \quad \text{... } (ii)$$

From equations (*i*) and (*ii*), we have

$$W.L = 2w(L_1)^2 + 2w(L_2)^2 \quad \text{or} \quad w = \frac{W.L}{2[(L_1)^2 + (L_2)^2]} \quad \text{...}(iii)$$

It may be noted that the most heavily loaded bolts are those which are situated at the greatest distance from the tilting edge. In the case discussed above, the bolts at distance $L2$ are heavily loaded.

So, Tensile load on each bolt at distance L_2,

$$W_{t2} = W_2 = w.L_2 = \frac{W.L.L_2}{2[(L_1)^2 + (L_2)^2]} \quad \text{... [From equation } (iii)]$$

And the total tensile load on the most heavily loaded bolt,

$$W_t = W_{t1} + W_{t2} \quad \text{... } (iv)$$

If d_c is the core diameter of the olt and σt is the tensile stress for the bolt material, then total tensile load,

$$W_t = \frac{\pi}{4}(d_c)^2 \sigma_t \quad \text{...}(v)$$

From equations (*iv*) and (*v*), the value of dc may be obtained.

Problem:

A bracket, as shown in Fig.1, supports a load of 30 kN. Determine the size of bolts, if the maximum allowable tensile stress in the bolt material is 60 MPa. The distances are: L_1 = 80 mm, L_2 = 250 mm, and L = 500 mm.

Solution. Given : W = 30 kN ; σ_t = 60 MPa = 60 N/mm^2 ; L_1 = 80 mm ; L_2 = 250 mm ; L = 500 mm

We know that the direct tensile load carried by each bolt,

$$W_{t1} = \frac{W}{n} = \frac{30}{4} = 7.5 \text{ kN}$$

and load in a bolt per unit distance,

$$w = \frac{W.L}{2[(L_1)^2 + (L_2)^2]} = \frac{30 \times 500}{2[(80)^2 + (250)^2]} = 0.109 \text{ kN/mm}$$

Eccentric Load Acting Perpendicular to the Axis of Bolts

A wall bracket carrying an eccentric load perpendicular to the axis of the bolts is shown in Fig.2.

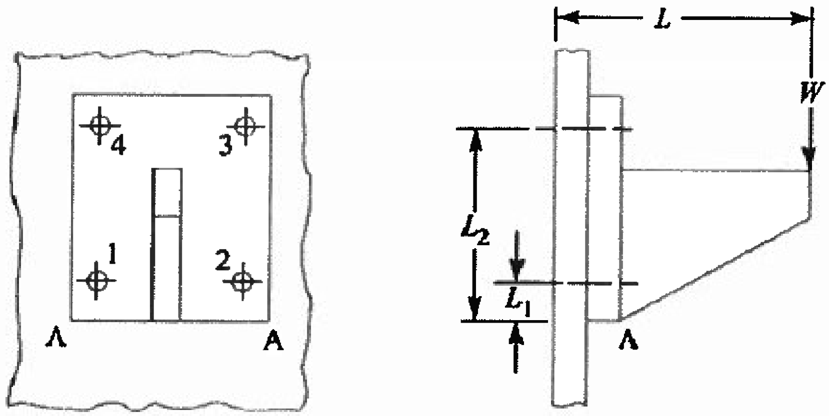


Fig. 2. Eccentric load perpendicular to the axis of bolts.

In this case, the bolts are subjected to direct shearing load which is equally shared by all the bolts. Therefore direct shear load on each bolts,

$W_s = W/n$, where n is number of bolts.

A little consideration will show that the eccentric load W will try to tilt the bracket in the clockwise direction about the edge A-A. As discussed earlier, the bolts will be subjected to tensile stress due to the turning moment. The maximum tensile load on a heavily loaded bolt (W_t) may be obtained in the similar manner as discussed in the previous article. In this case, bolts 3 and 4 are heavily loaded.

Maximum tensile load on bolt 3 or 4,

$$W_{t2} = W_t = \frac{W.L.L_2}{2\left[(L_1)^2 + (L_2)^2\right]}$$

When the bolts are subjected to shear as well as tensile loads, then the equivalent loads may be determined by the following relations:

Equivalent tensile load,

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

And equivalent shear load,

$$W_{se} = \frac{1}{2}\left[\sqrt{(W_t)^2 + 4(W_s)^2}\right]$$

Knowing the value of equivalent loads, the size of the bolt may be determined for the given allowable stresses.

Size of the bolt

Let d_c = Core diameter of the bolt.

We know that the equivalent tensile load (W_{te}).

$$7490 = \frac{\pi}{4}(d_c)^2\sigma_t = \frac{\pi}{4}(d_c)^2 84 = 66\,(d_c)^2$$

$\therefore$ $(d_c)^2 = 7490 / 66 = 113.5$ or $d_c = 10.65$ mm

From Table 11.1 (coarse series), the standard core diameter is 11.546 mm and the corresponding size of the bolt is M 14. **Ans.**

Cross-section of the arm of the bracket

Let t and b = Thickness and depth of arm of the bracket respectively.

$\therefore$ Section modulus,

$$Z = \frac{1}{6}\,t.b^2$$

Assume that the arm of the bracket extends upto the face of the steel column. This assumption gives stronger section for the arm of the bracket.

$\therefore$ Maximum bending moment on the bracket,

$$M = 12 \times 10^3 \times 400 = 4.8 \times 10^6 \text{ N-mm}$$

We know that the bending (tensile) stress (σ_t),

$$84 = \frac{M}{Z} = \frac{4.8 \times 10^6 \times 6}{t.b^2} = \frac{28.8 \times 10^6}{t.b^2}$$

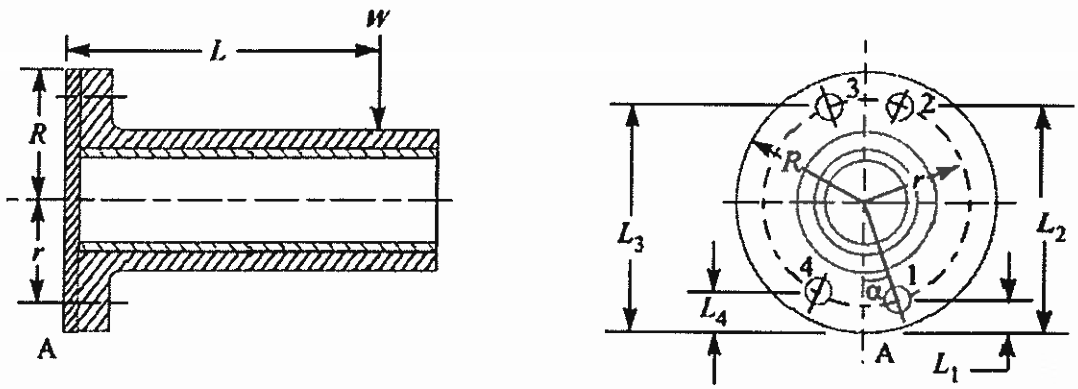
$\therefore$ $t.b^2 = 28.8 \times 10^6 / 84 = 343 \times 10^3$ or $t = 343 \times 10^3 / b^2$

Assuming depth of arm of the bracket, $b = 250$ mm, we have

$$t = 343 \times 10^3 / (250)^2 = 5.5 \text{ mm } \textbf{Ans.}$$

Eccentric Load on a Bracket with Circular Base

Sometimes the base of a bracket is made circular as in case of a flanged bearing of a heavy machine tool and pillar crane etc. Consider a round flange bearing of a machine tool having four bolts as shown in Fig. 1.



(a) (b)

Fig.1. Ecce tric load on a bracket with circular base.

Let R = Radius of the column flange,

r = Radius of the bolt pitch circle,

w = Load per bolt per unit distance from the tilting edge,

L = Distance of the load from the tilting edge, and

L_1, L_2, L_3, and L_4 = Distance of bolt centers from the tilting edge A.

As discussed in the previous article, equating the external moment $W \times L$ to the sum of the resisting moments of all the bolts, we have,

$$WL = w\left[(L_1)^2 + (L_2)^2 + (L_3)^2 + (L_4)^2\right]$$

$$\therefore \quad w = \frac{W.L}{(L_1)^2 + (L_2)^2 + (L_3)^2 + (L_4)^2} \quad \ldots(i)$$

Now from the geometry of the Fig. 1(*b*), we find that

$$L_1 = R - r\cos\alpha \; L_2 = R + r\sin\alpha$$

$$L_3 = R + r\cos\alpha \text{ and } L_4 = R - r\sin\alpha$$

Substituting these values in equation (*i*), we get

$$w = \frac{W.L}{4R^2 + 2r^2}$$

Load in the bolt situated at 1 = $w.L_1$ =

$$\frac{W.L.L_1}{4R^2 + 2r^2} = \frac{W.L(R - r\cos\alpha)}{4R^2 + 2r^2}$$

This load will be maximum when $\cos\alpha$ is minimum *i.e.* when $\cos\alpha = -1$ or $\alpha = 180°$.

$$W_t = \frac{2WL}{n}\left[\frac{R + r\cos\left(\frac{180}{n}\right)}{2R^2 + r^2}\right]$$

$$= \frac{2 \times 400 \times 10^3 \times 250}{4}\left[\frac{325 + 250\cos\left(\frac{180}{4}\right)}{2(325)^2 + (250)^2}\right] = 91\,643 \text{ N}$$

We also know that maximum load on the bolt (W_t),

$$91\,643 = \frac{\pi}{4}(d_c)^2 \sigma_t = \frac{\pi}{4}(d_c)^2\, 60 = 47.13\,(d_c)^2$$

$$\therefore \quad (d_c)^2 = 91\,643 / 47.13 = 1945 \quad \text{or} \quad d_c = 44 \text{ mm}$$

From DDB, we find that the standard core diameter of the bolt is 45.795 mm and

Diameter D for the arm of the bracket

The section of the arm having D as the diameter is subjected to bending moment as well as twisting moment. We know that bending moment,

$$M = 13\,500 \times (300 - 25) = 3712.5 \times 10^3 \text{ N-mm}$$

and twisting moment, $T = 13\,500 \times 250 = 3375 \times 10^3$ N-mm

$\therefore$ Equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(3712.5 \times 10^3)^2 + (3375 \times 10^3)^2} \text{ N-mm}$$
$$= 5017 \times 10^3 \text{ N-mm}$$

We know that equivalent twisting moment (T_e),

$$5017 \times 10^3 = \frac{\pi}{16} \times \tau \times D^3 = \frac{\pi}{16} \times 65 \times D^3 = 12.76\, D^3$$

$\therefore$ $D^3 = 5017 \times 10^3 / 12.76 = 393 \times 10^3$

or $D = 73.24$ say 75 mm **Ans.**

Diameter (d) for the arm of the bracket

The section of the arm having d as the diameter is subjected to bending moment only. We know that bending moment,

$$M = 13\,500 \left(250 - \frac{75}{2}\right) = 2868.8 \times 10^3 \text{ N-mm}$$

and section modulus, $Z = \frac{\pi}{32} \times d^3 = 0.0982\, d^3$

We know that bending (tensile) stress (σ_t),

$$110 = \frac{M}{Z} = \frac{2868.8 \times 10^3}{0.0982\, d^3} = \frac{29.2 \times 10^6}{d^3}$$

$\therefore$ $d^3 = 29.2 \times 10^6 / 110 = 265.5 \times 10^3$ or $d = 64.3$ say 65 mm **Ans.**

Tensile load on each top bolt

Due to the eccentric load W, the bracket has a tendency to tilt about the edge E–E, as shown in Fig. 11.46.

Let w = Load on each bolt per mm distance from the tilting edge due to the tilting effect of the bracket.

Since there are two bolts each at distance L_1 and L_2 as shown in Fig. 11.46, therefore total moment of the load on the bolts about the tilting edge E–E

$$= 2\,(w.L_1)\,L_1 + 2(w.L_2)\,L_2 = 2w\,[(L_1)^2 + (L_2)^2]$$
$$= 2w\,[(37.5)^2 + (237.5)^2] = 115\,625\; w \text{ N-mm} \quad ...(i)$$

...($\because L_1 = 37.5$ mm and $L_2 = 237.5$ mm)

and turning moment of the load about the tilting edge

$$= W.L = 13\,500 \times 300 = 4050 \times 10^3 \text{ N-mm} \quad ...(ii)$$

From equations (*i*) and (*ii*), we have

$$w = 4050 \times 10^3 / 115\,625 = 35.03 \text{ N/mm}$$

$\therefore$ Tensile load on each top bolt

$$= w.L_2 = 35.03 \times 237.5 = 8320 \text{ N } \textbf{Ans.}$$

$$= \sqrt{(W_{s1})^2 + (W_{s2})^2 + 2\,W_{s1} \times W_{s2} \times \cos 45^\circ}$$

$$= \sqrt{(3375)^2 + (5967)^2 + 2 \times 3375 \times 5967 \times 0.7071} = 8687 \text{ N } \textbf{Ans.}$$

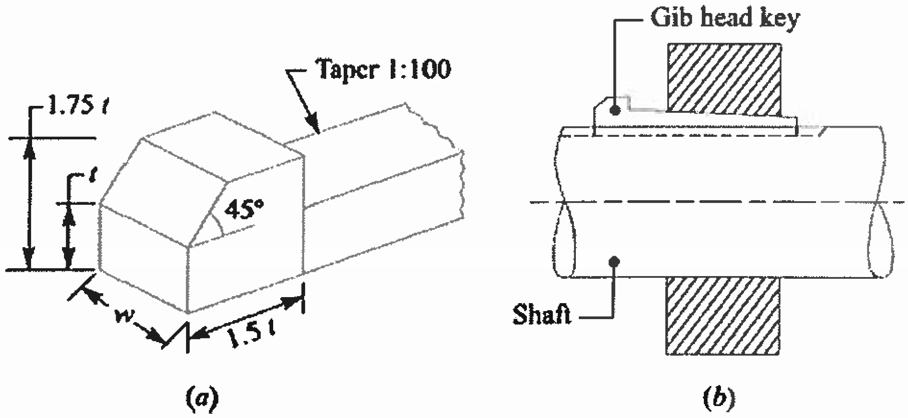


Fig. Gib head key and its use.

The usual proportions of the gib head key are:

Width, $w = d / 4$; and thickness at large end, $t = 2w / 3 = d / 6$

5. Feather key. A key attached to one member of a pair and which permits relative axial movement is known as **feather key.** It is a special type of parallel key which transmits a turning moment and also permits axial movement. It is fastened either to the shaft or hub, the key being a sliding fit in the key way of the moving piece.

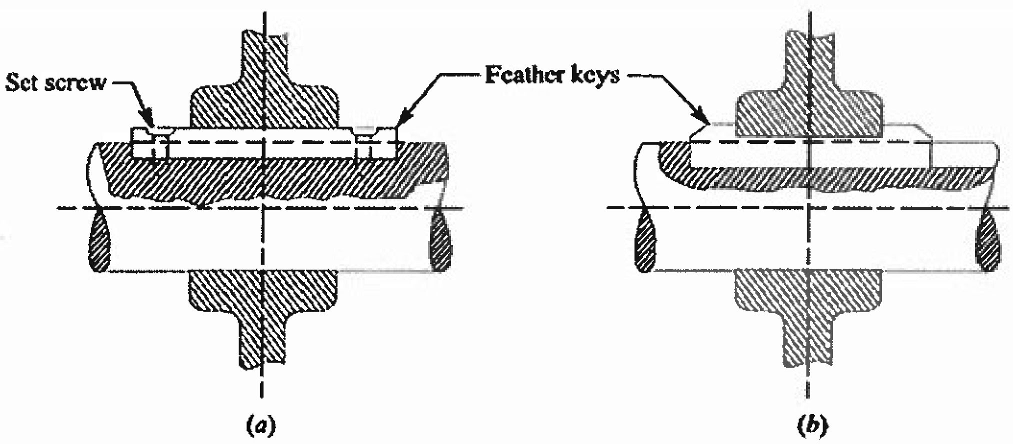


Fig. Feather Keys

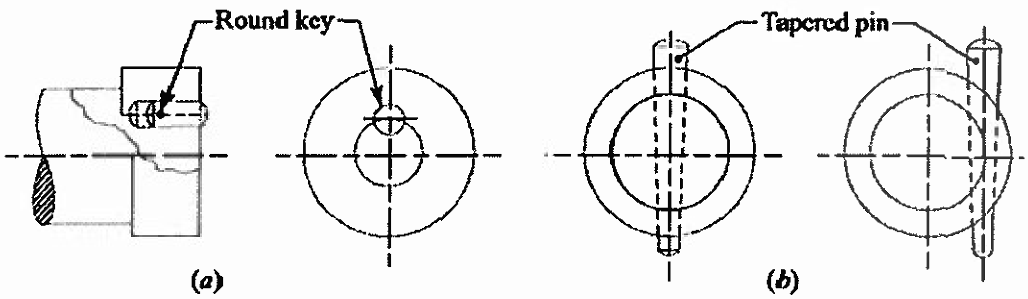
6. Woodruff key. The woodruff key is an easily adjustable key. It is a piece from a cylindrical disc having segmental cross-section in front view as shown in Fig. A woodruff key is capable of tilting in a recess milled out in the shaft by a cutter having the same curvature as the disc from which the key is made. This key is largely used in machine tool and automobile construction.

Tangent Keys

The tangent keys are fitted in pair at right angles as shown in Fig. Each key is to withstand torsion in one direction only. These are used in large heavy duty shafts.

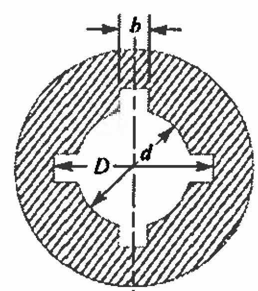
Round Keys

The round keys, as shown in Fig. (a) are circular in section and fit into holes drilled partly in the shaft and partly in the hub. They have the advantage that their keyways may be drilled and reamed after the mating parts have been assembled. Round keys are usually considered to be most appropriate for low power drives.



Splines

Sometimes, keys are made inte ral with the shaft which fits in the keyways broached in the hub. Such shafts are known as **splined shafts** as shown in Fig. These shafts usually have four, six, ten or sixteen splines. The splined shafts are relatively stronger than shafts having a single keyway.



$D = 1.25\,d$ and $b = 0.25\,D$

Stresses in Keys:

Forces acting on a Sunk Key

When a key is used in transmitting torque from a shaft to a rotor or hub, the following two types of forces act on the key:

1. Forces (F1) due to fit of the key in its keyway, as in a tight fitting straight key or in a tapered key driven in place. These forces produce compressive stresses in the key which are difficult to determine in magnitude.

2. Forces (F) due to the torque transmitted by the shaft. These forces produce shearing and compressive (or crushing) stresses in the key.

The forces acting on a key for a clockwise torque being transmitted from a shaft to a hub are shown in Fig.

$$l \times w \times \tau \times \frac{d}{2} - l \times \frac{t}{2} \times \sigma_c \times \frac{d}{2}$$

Or

$$\frac{w}{t} - \frac{\sigma_c}{2\tau}$$

The permissible crushing stress for the usual key material is at least twice the permissible shearing stress. Therefore from the above equation, we have w = t. In other words, a square key is equally strong in shearing and crushing.

In order to find the length of the key to transmit full power of the shaft, the shearing strength of the key is equal to the torsional shear strength of the shaft. We know that the shearing strength of key,

$$T = l \times w \times \tau \times \frac{d}{2}$$

And torsional shear strength of the shaft,

$$T = \frac{\pi}{16} \times \tau_1 \times d^3$$

From the above

$$i \times w \times \tau \times \frac{d}{2} = \frac{\pi}{16} \times \tau_1 \times d^3$$

$$l = \frac{\pi}{8} \times \frac{\tau_1 d^2}{w \times \tau} = \frac{\pi d}{2} \times \frac{\tau_1}{\tau} = 1.571\, d \times \frac{\tau_1}{\tau}$$

When the key material is same as that of the shaft, then $\tau = \tau_1$. So, $l = 1.571\ d$.

d = Diameter of the rods,

d_1 = Outside diameter of socket,

d_2 = Diameter of spigot or inside diameter of socket,

d_3 = Outside diameter of spigot collar,

t_1= Thickness of spigot collar,

d_4 = Diameter of socket collar,

c = Thickness of socket collar,

b = Mean width of cotter,

t = Thickness of cotter,

l = Length of cotter,

a = Distance from the end of the slot to the end of rod,

σ_t = Permissible tensile stress for the rods material,

τ = Permissible shear stress for the cotter material, and

σ_c = Permissible crushing stress for the cotter material.

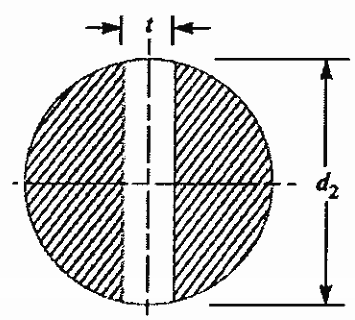
The dimensions for a socket and spigot cotter joint may be obtained by considering the various modes of failure as discussed below:

1. Failure of the rods in tension

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

From this equation, diameter of the rods (d) may be determined.

2. Failure of spigot in tension across the weakest section (or slot)



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

From this equation, the diameter of spigot or inside diameter of socket (d_2) may be determined. In actual practice, the thickness of cotter is usually taken as d_2 / 4.

3. Failure of the rod or cotter in crushing

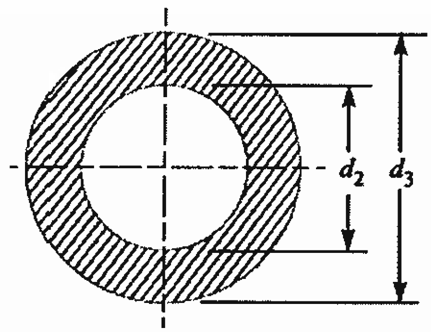
$$P = d_2 \times t \times \sigma_c$$

8. Failure of rod end in shear

$$P = 2\,a \times d_2 \times \tau$$

From this equation, the distance from the end of the slot to the end of the rod (a) may be obtained.

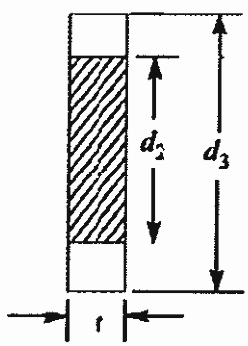
9. Failure of spigot collar in crushing



$$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right] \sigma_c$$

From this equation, the diameter of the spigot collar (d_3) may be obtained.

10. Failure of the spigot collar in shearing



$$P = \pi\, d_2 \times t_1 \times \tau$$

From this equation, the thickness of spigot collar (t_1) may be obtained.

11. Failure of cotter in bending

The maximum bending moment occurs at the centre of the cotter and is given by

Problem:

Design and draw a cotter joint to support a load varying from 30 kN in compression to 30 kN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied sta ically. Tensile stress = compressive stress = 50 MPa ; shear stress = 35 MPa and crushing stress = 90 MPa.

Solution. Given : $P = 30\text{ kN} = 30 \times 10^3\text{ N}$; $\sigma_t = 50\text{ MPa} = 50\text{ N/mm}^2$; $\tau = 35\text{ MPa} = 35\text{ N/mm}^2$; $\sigma_c = 90\text{ MPa} = 90\text{ N/mm}^2$

1. *Diameter of the rods*

Let d = Diameter of the rods.

Considering the failure of the rod in tension. We know that load (P),

$$30 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 50 = 39.3\, d^2$$

$\therefore$ $d^2 = 30 \times 10^3 / 39.3 = 763$ or $d = 27.6$ say 28 mm **Ans.**

2. *Diameter of spigot and thickness of cotter*

Let d_2 = Diameter of spigot or inside diameter of socket, and

t = Thickness of cotter. It may be taken as $d_2 / 4$.

Considering the failure of spigot in tension across the weakest section. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]50 = 26.8\,(d_2)^2$$

$\therefore$ $(d_2)^2 = 30 \times 10^3 / 26.8 = 1119.4$ or $d_2 = 33.4$ say 34 mm

and thickness of cotter, $t = \dfrac{d_2}{4} = \dfrac{34}{4} = 8.5$ mm

Let us now check the induced crushing stress. We know that load (P),

$$30 \times 10^3 = d_2 \times t \times \sigma_c = 34 \times 8.5 \times \sigma_c = 289\,\sigma_c$$

$\therefore$ $\sigma_c = 30 \times 10^3 / 289 = 103.8\text{ N/mm}^2$

Since this value of σ_c is more than the given value of $\sigma_c = 90\text{ N/mm}^2$, therefore the dimensions $d_2 = 34$ mm and $t = 8.5$ mm are not safe. Now let us find the values of d_2 and t by substituting the value of $\sigma_c = 90\text{ N/mm}^2$ in the above expression, *i.e.*

$$30 \times 10^3 = d_2 \times \frac{d_2}{4} \times 90 = 22.5\,(d_2)^2$$

$\therefore$ $(d_2)^2 = 30 \times 10^3 / 22.5 = 1333$ or $d_2 = 36.5$ say 40 mm **Ans.**

and $t = d_2 / 4 = 40 / 4 = 10$ mm **Ans.**

3. *Outside diameter of socket*

Let d_1 = Outside diameter of socket.

Considering the failure of the socket in tension across the slot. We know that load (P),

$$30 \times 10^3 = \left[\frac{\pi}{4}\left\{(d_1)^2 - (d_2)^2\right\} - (d_1 - d_2)\,t\right]\sigma_t$$

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

$$30 \times 10^3/50 = 0.7854\,(d_1)^2 - 1256.6 - 10\,d_1 + 400$$

9. Thickness of spigot collar

Let t_1 = Thickness of spigot collar.

Considering the failure of spigot collar in shearing. We know that load (P),

$$30 \times 10^3 = \pi \, d_2 \times t_1 \times \tau = \pi \times 40 \times t_1 \times 35 = 4400 \, t_1$$

$\therefore$ $t_1 = 30 \times 10^3 / 4400 = 6.8$ say 8 mm **Ans.**

10. The length of cotter (l) is taken as 4 d.

$\therefore$ $l = 4\,d = 4 \times 28 = 112$ mm **Ans.**

11. The dimension e is taken as 1.2 d.

$\therefore$ $e = 1.2 \times 28 = 33.6$ say 34 mm **Ans.**

t = Thickness of cotter,

l = Length of cotter,

b = Width of cotter,

a = Distance of the rod end from the beginning to the cotter hole (inside the sleeve end),

c = Distance of the rod end from its end to the cotter hole,

σ_t, τ and σ_c = Permissible tensile, shear and crushing stresses respectively for the material of the rods and cotter.

The dimensions for a sleeve and cotter joint may be obtained by considering the various modes of failure as discussed below:

1. Failure of the rods in tension

The rods may fail in tension due to the tensile load P. We know that

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

From this equation, diameter of the rods (d) may be obtained.

2. Failure of the rod in tension across the weakest section (i.e. slot)

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

From this equation, the diameter of enlarged end of the rod (d_2) may be obtained. The thickness of cotter is usually taken as $d_2 / 4$.

3. Failure of the rod or cotter in crushing

$$P = d_2 \times t \times \sigma_c$$

From this equation, the induced crushing stress may be checked.

4. Failure of sleeve in tension across the slot

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

From this equation, the outside diameter of sleeve (d_1) may be obtained.

5. Failure of cotter in shear

$$P = 2b \times t \times \tau$$

From this equation, width of cotter (b) may be determined.

6. Failure of rod end in shear

$$P = 2\,a \times d_2 \times \tau$$

From this equation, distance (a) may be determined.

Problem:

Design a sleeve and cotter joint to resist a tensile load of 60 kN. All parts of the joint are made of the same material with the following allowable stresses: σ_t = 60 MPa ; τ = 70 MPa ; and σ_c = 125 MPa.

Solution. Given : $P = 60$ kN $= 60 \times 10^3$ N ; $\sigma_t = 60$ MPa $= 60$ N/mm^2 ; $\tau = 70$ MPa $= 70$ N/mm^2 ; $\sigma_c = 125$ MPa $= 125$ N/mm^2

1. *Diameter of the rods*

Let d = Diameter of the rods.

Considering the failure of the rods in tension. We know that load (P),

$$60 \times 10^3 = \frac{\pi}{4} \times d^2 \times \sigma_t = \frac{\pi}{4} \times d^2 \times 60 = 47.13\, d^2$$

$$\therefore \quad d^2 = 60 \times 10^3 / 47.13 = 1273 \quad \text{or} \quad d = 35.7 \text{ say } 36 \text{ mm Ans.}$$

2. *Diameter of enlarged end of rod and thickness of cotter*

Let d_2 = Diameter of enlarged end of rod, and

t = Thickness of cotter. It may be taken as $d_2/4$.

Considering the failure of the rod in tension across the weakest section (*i.e.* slot). We know that load (P),

$$60 \times 10^3 = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times \frac{d_2}{4}\right]60 = 32.13\,(d_2)^2$$

$$\therefore \quad (d_2)^2 = 60 \times 10^3 / 32.13 = 1867 \quad \text{or} \quad d_2 = 43.2 \text{ say } 44 \text{ mm Ans.}$$

and thickness of cotter,

$$t = \frac{d_2}{4} = \frac{44}{4} = 11 \text{ mm Ans.}$$

Let us now check the induced crushing stress in the rod or cotter. We know that load (P),

$$60 \times 10^3 = d_2 \times t \times \sigma_c = 44 \times 11 \times \sigma_c = 484\,\sigma_c$$

$$\therefore \quad \sigma_c = 60 \times 10^3 / 484 = 124 \text{ N/mm}^2$$

Since the induced crushing stress is less than the given value of 125 N/mm^2, therefore the dimensions d_2 and t are within safe limits.

3. *Outside diameter of sleeve*

Let d_1 = Outside diameter of sleeve.

Considering the failure of sleeve in tension across the slot. We know that load (P)

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

$$= \left[\frac{\pi}{4}[(d_1)^2 - (44)^2] - (d_1 - 44)\,11\right]60$$

Gib and Cotter Joint

This joint is generally used to connect two rods of square or rectangular section. To make the joint; one end of the rod is formed into a U-fork, into which, the end of the other rod fits-in. When a cotter is driven-in, the friction between the cotter and straps of the U-fork, causes the straps open. This is prevented by the use of a gib.

A gib is also a wedge shaped piece of rectangular cross-section with two rectangular projections, called lugs. One side of the gib is tapered and the other straight. The tapered side of the gib bears against the tapered side of the cotter such that the outer edges of the cotter and gib as a unit are parallel. This facilitates making of slots with parallel edges, unlike the tapered edges in case of ordinary cotter joint. The gib also provides larger surface for the cotter to slide on. For making the joint, the gib is placed in position first, and then the cotter is driven-in.

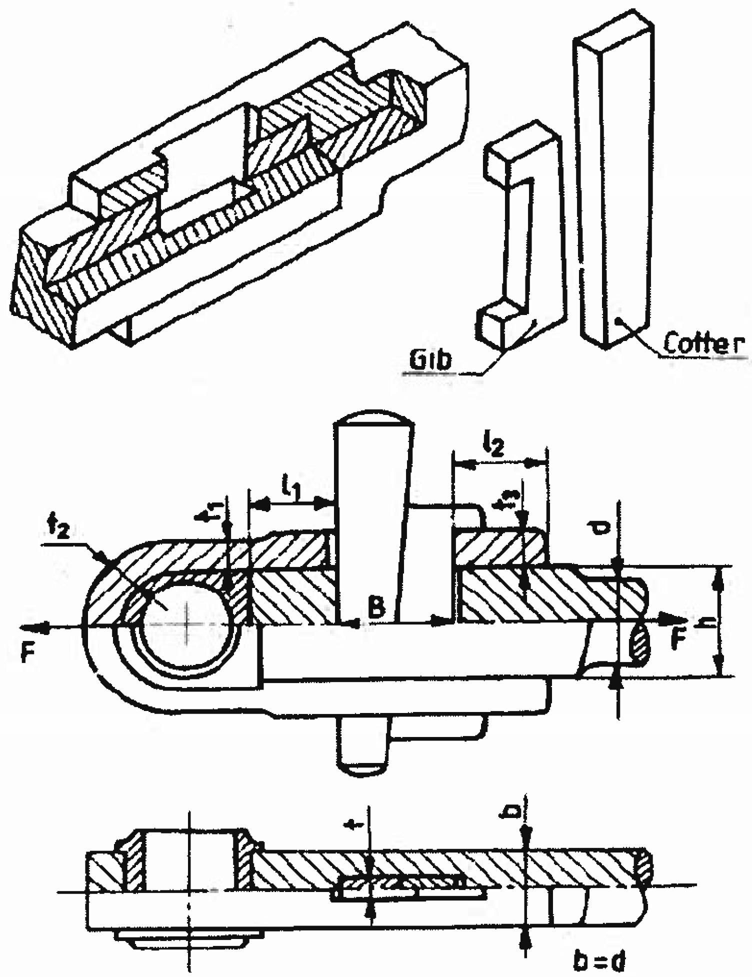


Fig. Gib and cotter Joint

Let F be the maximum tensile or compressive force in the connecting rod, and

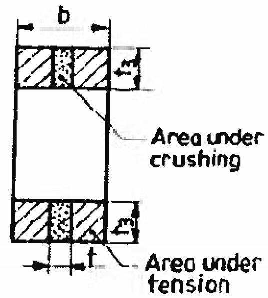


Fig.3

$$F = 2bt_3 - 2tt_3 = 2t_3(b - t)\sigma_t$$

The thickness, t2 may be taken as (1.15 to 1.5) t], and

Thickness of the cotter, t = b/4.

5. Crushing between the rod and cotter (Fig.1)

$$F = h\,t\,\sigma_c \text{ ; and } h = 2t_3$$

6. Crushing between the strap and gib(Fig.3)

$$F = 2\,t\,t_3\,\sigma_c$$

7. Shear failure of the rod end. It is under double shear (Fig.4).

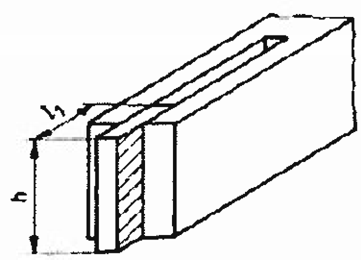


Fig.4

$$F = 2l_1 h\tau$$

8. Shear failure of the strap end. It is under double shear (Fig.5).

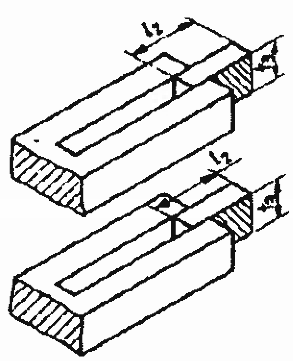


Fig.5

$$F = 4\,l_2\,t_3\tau$$

9. Shear failure of the cotter and gib. It is under double shear.

$$F=2Bt\tau$$

The following proportions for the widths of the cotter and gib may be followed:

or $(d_3)^2 - 21\, d_3 - 3670 = 0$

$$\therefore \quad d_3 = \frac{21 + \sqrt{(21)^2 + 4 \times 3670}}{2} = \frac{21 \pm 123}{2} = 72 \text{ mm} \quad \text{...(Taking +ve sign)}$$

Let us now check the induced crushing stress in the socket. We know that load (P),

$$70\,695 = (d_3 - d_2)\, t \times \sigma_c = (72 - 55)\, 16.5 \times \sigma_c = 280.5\, \sigma_c$$

$$\therefore \quad \sigma_c = 70\,695 / 280.5 = 252 \text{ N/mm}^2$$

Since the induced crushing is greater than the permissible value of 84 N/mm², therefore let us find the value of d_3 by substituting $\sigma_c = 84$ N/mm² in the above expression, *i.e.*

$$70\,695 = (d_3 - 55)\, 16.5 \times 84 = (d_3 - 55)\, 1386$$

$$\therefore \quad d_3 - 55 = 70\,695 / 1386 = 51$$

or $$d_3 = 55 + 51 = 106 \text{ mm Ans.}$$

We know the tapered length of the piston rod,

$$L = 2.2\, d_2 = 2.2 \times 55 = 121 \text{ mm Ans.}$$

Assuming the taper of the piston rod as 1 in 20, therefore the diameter of the parallel part of the piston rod,

$$d = d_2 + \frac{L}{2} \times \frac{1}{20} = 55 + \frac{121}{2} \times \frac{1}{20} = 58 \text{ mm Ans.}$$

and diameter of the piston rod at the tapered end,

$$d_1 = d_2 - \frac{L}{2} \times \frac{1}{20} = 55 - \frac{121}{2} \times \frac{1}{20} = 52 \text{ mm Ans.}$$

$$F \square \frac{\square d^2}{4} \square$$

2. Tension failure of the eye (fig.1)

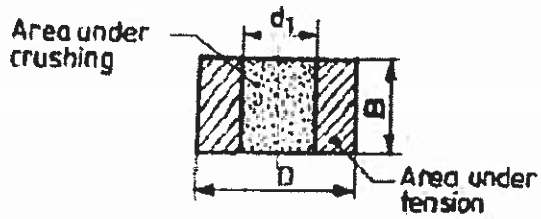


Fig.1

$$F = (D-d_1)\, B\, \sigma_t$$

3. Tension failure of the fork (fig.2)

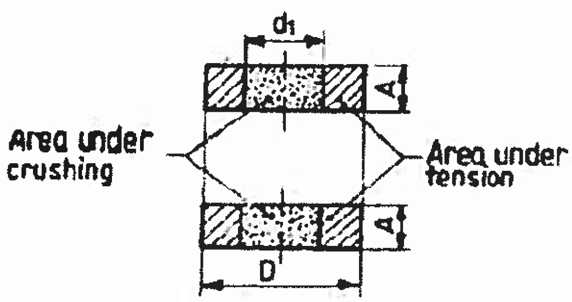


Fig.2

$$F = 2\,(D - d_1)\, A\, \sigma_t$$

4. Shear failure of the eye (Fig.3)

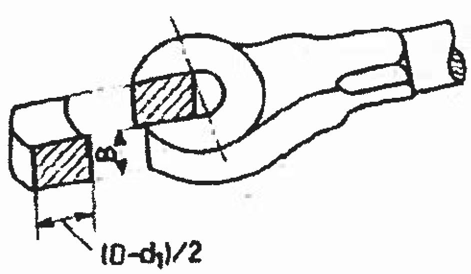


Fig.3

$$F = (D-d_1)\, B\, \tau$$

5. Shear failure of the fork (Fig.4)

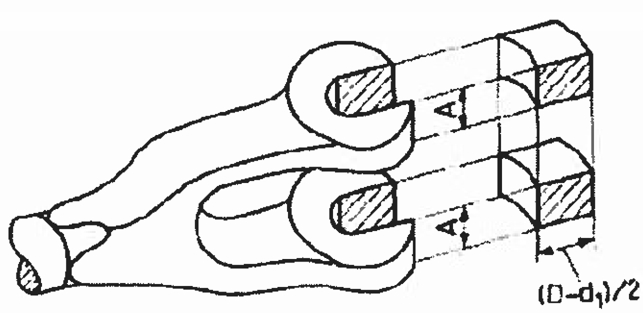


Fig.4

$$F = 2\,(D-d_1)\, A\, \tau$$

$\therefore \quad \tau = 150 \times 10^3 / 4248 = 35.3 \text{ N/mm}^2 = 35.3 \text{ MPa}$

3. ***Failure of the single eye or rod end in tension***

The single eye or rod end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \sigma_t = (104 - 52)\, 65 \times \sigma_t = 3380\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

4. ***Failure of the single eye or rod end in shearing***

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, t \times \tau = (104 - 52)\, 65 \times \tau = 3380\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

5. ***Failure of the single eye or rod end in crushing***

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times t \times \sigma_c = 52 \times 65 \times \sigma_c = 3380\, \sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 3380 = 44.4 \text{ N/mm}^2 = 44.4 \text{ MPa}$$

6. ***Failure of the forked end in tension***

The forked end may fail in tension due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \sigma_t = (104 - 52)\, 2 \times 40 \times \sigma_t = 4160\, \sigma_t$$

$$\therefore \quad \sigma_t = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

7. ***Failure of the forked end in shear***

The forked end may fail in shearing due to the load. We know that load (P),

$$150 \times 10^3 = (d_2 - d_1)\, 2\, t_1 \times \tau = (104 - 52)\, 2 \times 40 \times \tau = 4160\, \tau$$

$$\therefore \quad \tau = 150 \times 10^3 / 4160 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

8. ***Failure of the forked end in crushing***

The forked end may fail in crushing due to the load. We know that load (P),

$$150 \times 10^3 = d_1 \times 2\, t_1 \times \sigma_c = 52 \times 2 \times 40 \times \sigma_c = 4160\, \sigma_c$$

$$\therefore \quad \sigma_c = 150 \times 10^3 / 4180 = 36 \text{ N/mm}^2 = 36 \text{ MPa}$$

From above, we see that the induced stresses are less than the given design stresses, therefore the joint is safe.

In designing shafts on the basis of strength, the following cases may be considered:

(a) Shafts subjected to twisting moment or torque only,

(b) Shafts subjected to bending moment only,

(c) Shafts subjected to combined twisting and bending moments, and

(d) Shafts subjected to axial loads in addition to combined torsional and bending loads.

Shafts Subjected to Twisting Moment Only

a) Solid shaft:

When the shaft is subjected to a twisting moment (or torque) only, then the diameter of the shaft may be obtained by using the torsion equation. We know that

$$\frac{T}{J} = \frac{\tau}{r}$$

Where T = Twisting moment (or torque) acting upon the shaft,

J = Polar moment of inertia of the shaft about the axis of rotation,

τ = Torsional shear stress, and

r = Distance from neutral axis to the outer most fibre

$= d / 2$; where d is the diameter of the shaft.

We know that for round solid shaft, polar moment of inertia,

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

Then we get,

$$T = \frac{\pi d^3}{16} \tau$$

From this equation, diameter of the solid shaft (d) may be obtained.

b) Hollow Shaft:

We also know that for hollow shaft, polar moment of inertia,

$$J = \frac{\pi}{32}\left[(d_0)^4 - (d_i)^4\right]$$

Where d_o and d_i = Outside and inside diameter of the shaft, and $r = d_0 / 2$.

Substituting these values in equation (*i*), we have

$$\frac{T}{\frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right]} = \frac{\tau}{\frac{d_o}{2}} \quad \text{or} \quad T = \frac{\pi}{16} \times \tau \left[\frac{(d_o)^4 - (d_i)^4}{d_o}\right]$$

Let k = Ratio of inside diameter and outside diameter of the shaft $= d_i / d_o$

Now the equation (***iii***) may be written as

We know that for a round solid shaft, moment of inertia,

$$I = \frac{\pi}{64} \times d^4 \quad \text{and} \quad y = \frac{d}{2}$$

Substituting these values in equation

$$\frac{M}{\frac{\pi}{64} \times d^4} = \frac{\sigma_b}{\frac{d}{2}} \quad \text{or} \quad M = \frac{\pi}{32} \times \sigma_b \times d^3$$

From this equation, diameter of the solid shaft (d) may be obtained.

b) Hollow Shaft:

We also know that for a hollow shaft, moment of inertia,

$$I = \frac{\pi}{64}\left[(d_o)^4 - (d_i)^4\right] = \frac{\pi}{64}(d_o)^4 (1 - k^4) \qquad \text{...(where } k = d_i / d_o\text{)}$$

And $y = d_o / 2$

Again substituting these values in equation, we have

$$\frac{M}{\frac{\pi}{64}(d_o)^4 (1 - k^4)} = \frac{\sigma_b}{\frac{d_o}{2}} \quad \text{or} \quad M = \frac{\pi}{32} \times \sigma_b \,(d_o)^3 (1 - k^4)$$

From this equation, the outside diameter of the shaft (do) may be obtained.

Shafts Subjected to Combined Twisting Moment and Bending Moment

When the shaft is subjected to combined twisting moment and bending moment, then the shaft must be designed on the basis of the two moments simultaneously. Various theories have been suggested to account for the elastic failure of the materials when they are subjected to various types of combined stresses. The following two theories are important from the subject point of view:

1. Maximum shear stress theory or Guest's theory. It is used for ductile materials such as mild steel.

2. Maximum normal stress theory or Rankine's theory. It is used for brittle materials such as cast iron.

Let τ = Shear stress induced due to twisting moment, and

σ_b = Bending stress (tensile or compressive) induced due to bending moment.

a) Solid Shaft:

According to maximum shear stress theory, the maximum shear stress in the shaft,

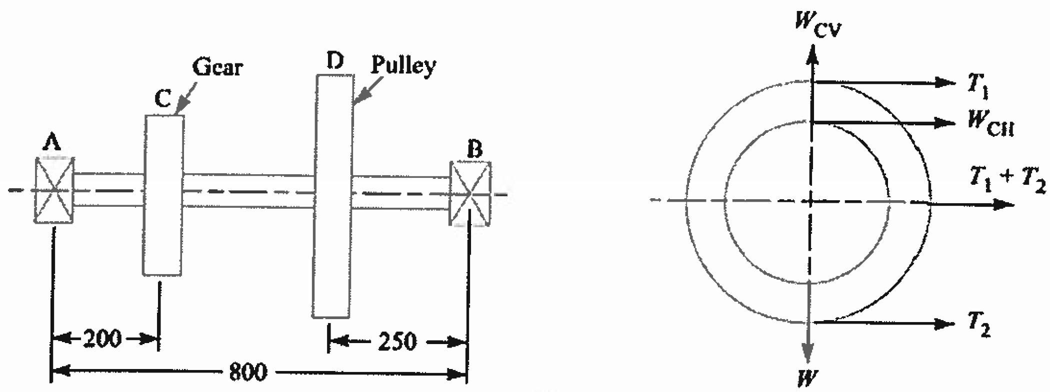
$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

In case of a hollow shaft, the equations *(ii)* and *(v)* may be written as

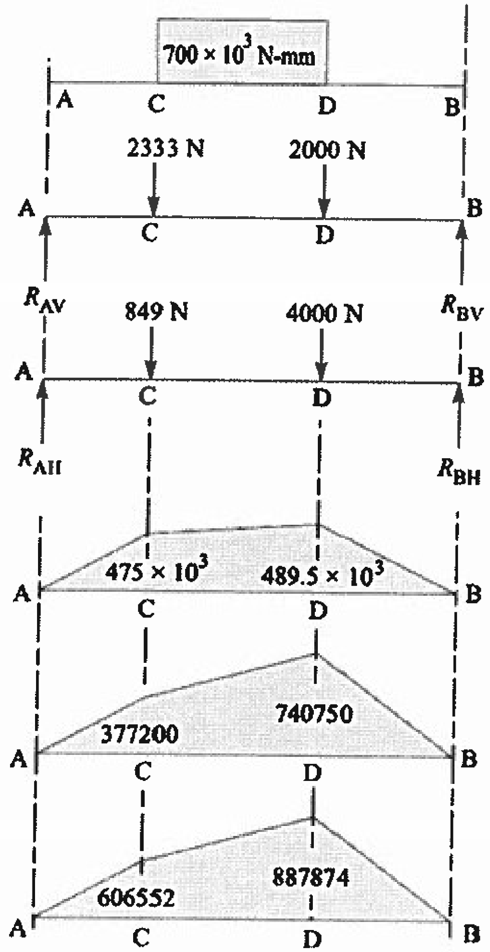
$$T_e = \sqrt{M^2 + T^2} = \frac{\pi}{16} \times \tau \, (d_o)^3 \, (1 - k^4)$$

$$M_e = \frac{1}{2}\left(M + \sqrt{M^2 + T^2}\right) - \frac{\pi}{32} \times \sigma_b \, (d_o)^3 \, (1 - k^4)$$

It is suggested that diameter of the shaft may be obtained by using both the theories and the larger of the two values is adopted.



All dimensions in mm.
(*a*) Space diagram.



(*b*) Torque diagram.

(*c*) Vertical load diagram.

(*d*) Horizontal load diagram.

(*e*) Vertical B.M. diagram.

(*f*) Horizontal B.M. diagram.

(*g*) Resultant B.M. diagarm.

∴ Horizontal load acting on the shaft at D,

$$W_{DH} = T_1 + T_2 = 3000 + 1000 = 4000 \text{ N}$$

and vertical load acting on the shaft at D,

$$W_{DV} = W = 2000 \text{ N}$$

Diameter of the shaft

Let d = Diameter of the shaft.

We know that the equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(887\,874)^2 + (700 \times 10^3)^2} = 1131 \times 10^3 \text{ N-mm}$$

We also know that equivalent twisting moment (T_e),

$$1131 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\, d^3$$

$$\therefore \quad d^3 = 1131 \times 10^3 / 7.86 = 144 \times 10^3 \quad \text{or} \quad d = 52.4 \text{ say } 55 \text{ mm Ans.}$$

Problem:

A steel solid shaft transmitting 15 kW at 200 r.p.m. is supported on two bearings 750 mm apart and has two gears keyed to it. The pinion having 30 teeth of 5 mm module is located 100 mm to the left of the right hand bearing and delivers power horizontally to the right. The gear having 100 teeth of 5 mm module is located 150 mm to the right of the left hand bearing and receives power in a vertical direction from below. Using an allowable stress of 54 MPa in shear, determine the diameter of the shaft.

Solution. Given : P = 15 kW = 15 × 10^3 W ; N = 200 r.p.m. ; AB = 750 mm ; T_D = 30 ; m_D = 5 mm ; BD = 100 mm ; T_C = 100 ; m_C = 5 mm ; AC = 150 mm ; τ = 54 MPa = 54 N/mm^2

The space diagram of the shaft is shown in Fig. 14.8 (*a*).

We know that the torque transmitted by the shaft,

$$T = \frac{P \times 60}{2\pi N} = \frac{15 \times 10^3 \times 60}{2\pi \times 200} = 716 \text{ N-m} = 716 \times 10^3 \text{ N-mm}$$

The torque diagram is shown in Fig. 14.8 (*b*).

We know that diameter of gear

= No. of teeth on the gear × module

∴ Radius of gear C,

$$R_C = \frac{T_C \times m_C}{2} = \frac{100 \times 5}{2} = 250 \text{ mm}$$

and radius of pinion D,

$$R_D = \frac{T_D \times m_D}{2} = \frac{30 \times 5}{2} = 75 \text{ mm}$$

Assuming that the torque at C and D is same (*i.e.* 716 × 10^3 N-mm), therefore tangential force on the gear C, acting downward,

$$F_{tC} = \frac{T}{R_C} = \frac{716 \times 10^3}{250} = 2870 \text{ N}$$

and tangential force on the pinion D, acting horizontally,

$$F_{tD} = \frac{T}{R_D} = \frac{716 \times 10^3}{75} = 9550 \text{ N}$$

$\therefore \quad R_{BV} = 2870 \times 150 / 750 = 574$ N

and $R_{AV} = 2870 - 574 = 2296$ N

We know that B.M. at A and B,

$$M_{AV} = M_{BV} = 0$$

B.M. at C, $\quad M_{CV} = R_{AV} \times 150 = 2296 \times 150 = 344\ 400$ N-mm

B.M. at D, $\quad M_{DV} = R_{BV} \times 100 = 574 \times 100 = 57\ 400$ N-mm

The B.M. diagram for vertical loading is shown in Fig. 14.8 (*e*).

Now considering horizontal loading at D. Let R_{AH} and R_{BH} be the reactions at the bearings A and B respectively. We know that

$$R_{AH} + R_{BH} = 9550 \text{ N}$$

Taking moments about A, we get

$$R_{BH} \times 750 = 9550\ (750 - 100) = 9550 \times 650$$

$\therefore \quad R_{BH} = 9550 \times 650 / 750 = 8277$ N

and $\quad R_{AH} = 9550 - 8277 = 1273$ N

We know that B.M. at A and B,

$$M_{AH} = M_{BH} = 0$$

B.M. at C, $\quad M_{CH} = R_{AH} \times 150 = 1273 \times 150 = 190\ 950$ N-mm

B.M. at D, $\quad M_{DH} = R_{BH} \times 100 = 8277 \times 100 = 827\ 700$ N-mm

The B.M. diagram for horizontal loading is shown in Fig. 14.8 (*f*).

We know that resultant B.M. at C,

$$M_C = \sqrt{(M_{CV})^2 + (M_{CH})^2} = \sqrt{(344\ 400)^2 + (190\ 950)^2}$$
$$= 393\ 790 \text{ N-mm}$$

and resultant B.M. at D,

$$M_D = \sqrt{(M_{DV})^2 + (M_{DH})^2} = \sqrt{(57\ 400)^2 + (827\ 700)^2}$$
$$= 829\ 690 \text{ N-mm}$$

The resultant B.M. diagram is shown in Fig. 14.8 (*g*). We see that the bending moment is maximum at D.

$\therefore$ Maximum bending moment,

$$M = M_D = 829\ 690 \text{ N-mm}$$

Let $\quad d$ = Diameter of the shaft.

We know that the equivalent twisting moment,

$$T_e = \sqrt{M^2 + T^2} = \sqrt{(829\ 690)^2 + (716 \times 10^3)^2} = 1096 \times 10^3 \text{ N-mm}$$

We also know that equivalent twisting moment (T_e),

$$1096 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 54 \times d^3 = 10.6\ d^3$$

$\therefore \quad d^3 = 1096 \times 10^3 / 10.6 = 103.4 \times 10^3$

or $\quad d = 47$ say 50 mm **Ans.**

$$= \frac{\alpha \times 4F}{\pi (d_o)^2 (1 - k^2)}$$

The value of column factor (α) for compressive loads* may be obtained from the following relation :

Column factor,

$$\alpha = \frac{1}{1 - 0.0044 (L/K)}$$

This expression is used when the slenderness ratio (L / K) is less than 115. When the slenderness ratio (L / K) is more than 115, then the value of column factor may be obtained from the following relation:

Column factor, α

$$\alpha = \frac{\sigma_y (L/K)^2}{C\pi^2 E}$$

Where L = Length of shaft between the bearings,

K = Least radius of gyration,

σ_y = Compressive yield point stress of shaft material, and

C = Coefficient in Euler's formula depending upon the end conditions.

The following are the different values of C depending upon the end conditions.

C = 1, for hinged ends,

= 2.25, for fixed ends,

= 1.6, for ends that are partly restrained as in bearings.

In general, for a hollow shaft subjected to fluctuating torsional and bending load, along with an axial load, the equations for equivalent twisting moment (T_e) and equivalent bending moment (M_e) may be written as

$$T_e = \sqrt{\left[K_m \times M + \frac{\alpha F d_o (1 + k^2)}{8}\right]^2 + (K_t \times T)^2}$$

$$= \frac{\pi}{16} \times \tau (d_o)^3 (1 - k^4)$$

$$M_e = \frac{1}{2}\left[K_m \times M + \frac{\alpha F d_o (1 + k^2)}{8} + \sqrt{\left\{K_m \times M + \frac{\alpha F d_o (1 + k^2)}{8}\right\}^2 + (K_t \times T)^2}\right]$$

$$= \frac{\pi}{32} \times \sigma_b (d_o)^3 (1 - k^4)$$

It may be noted that for a solid shaft, k = 0 and d_0 = d. When the shaft carries no axial load, then F = 0 and when the shaft carries axial tensile load, then α = 1.

Problem:

A hollow shaft is subjected to a maximum torque of 1.5 kN-m and a maximum bending moment of 3 kN-m. It is subjected, at the same time, to an axial load of 10 kN. Assume that the load is applied gradually and the ratio of the inner diameter to the outer diameter is 0.5. If the outer diameter of the shaft is 80 mm, find the shear stress induced in the shaft.

Solution. Given: T = 1.5 kN-m = 1.5×10^3 N-m ; M = 3 kN-m = 3×10^3 N-m ;

F = 10 kN = 10×10^3 N ; $k = d_i / d_o = 0.5$; d_o = 80 mm = 0.08 m

Let τ = Shear stress induced in the shaft.

Since the load is applied gradually, therefore from DDB, we find that $K_m = 1.5$; and $K_t = 1.0$

We know that the equivalent twisting moment for a hollow shaft,

$$T_e = \sqrt{\left[K_m \times M + \frac{\alpha F d_o (1 + k^2)}{8}\right]^2 + (K_t \times T)^2}$$

$$= \sqrt{\left[1.5 \times 3 \times 10^3 + \frac{1 \times 10 \times 10^3 \times 0.08 (1 + 0.5^2)}{8}\right]^2 + (1 \times 1.5 \times 10^3)^2}$$

$$= \sqrt{(4500 + 125)^2 + (1500)^2} = 4862 \text{ N-m} = 4862 \times 10^3 \text{ N-mm}$$

We also know that the equivalent twisting moment for a hollow shaft (T_e),

$$4862 \times 10^3 = \frac{\pi}{16} \times \tau (d_o)^3 (1 - k^4) = \frac{\pi}{16} \times \tau (80)^3 (1 - 0.5^4) = 94\,260\,\tau$$

$$\therefore \quad \tau = 4862 \times 10^3 / 94\,260 = 51.6 \text{ N/mm}^2 = 51.6 \text{ MPa } \textbf{Ans.}$$

Problem:

A hollow shaft of 0.5 m outside diameter and 0.3 m inside diameter is used to drive a propeller of a marine vessel. The shaft is mounted on bearings 6 metre apart and it transmits 5600 kW at 150 r.p.m. The maximum axial propeller thrust is 500 kN and the shaft weighs 70 kN.

Determine:

1. The maximum shear stress developed in the shaft, and
2. The angular twist between the bearings.

2. ***Angular twist between the bearings***

Let θ = Angular twist between the bearings in radians.

We know that the polar moment of inertia for a hollow shaft,

$$J = \frac{\pi}{32}[(d_o)^4 - (d_i)^4] = \frac{\pi}{32}[(0.5)^4 - (0.3)^4] = 0.005\ 34\ \text{m}^4$$

From the torsion equation,

$$\frac{T}{J} = \frac{G \times \theta}{L}, \text{ we have}$$

$$\theta = \frac{T \times L}{G \times J} = \frac{356\ 460 \times 6}{84 \times 10^9 \times 0.00\ 534} = 0.0048 \text{ rad}$$

... (Taking G = 84 GPa = 84×10^9 N/m^2)

$$= 0.0048 \times \frac{180}{\pi} = 0.275° \textbf{ Ans.}$$

Use of internal and external circlips, Gaskets and seals

Problems:

Compare the weight, strength and stiffness of a hollow shaft of the same external diameter as that of solid shaft. The inside diameter of the hollow shaft being half the external diameter. Both the shafts have the same material and length.

Solution. Given : $d_o = d$; $d_i = d_o / 2$ or $k = d_i / d_o = 1/2 = 0.5$

Comparison of weight

We know that weight of a hollow shaft,

$$W_H = \text{Cross-sectional area} \times \text{Length} \times \text{Density}$$

$$= \frac{\pi}{4}\left[(d_o)^2 - (d_i)^2\right] \times \text{Length} \times \text{Density} \qquad ...(i)$$

and weight of the solid shaft,

$$W_S = \frac{\pi}{4} \times d^2 \times \text{Length} \times \text{Density} \qquad ...(ii)$$

Since both the shafts have the same material and length, therefore by dividing equation (*i*) by equation (*ii*), we get

$$\frac{W_H}{W_S} = \frac{(d_o)^2 - (d_i)^2}{d^2} = \frac{(d_o)^2 - (d_i)^2}{(d_o)^2} \qquad ...(\because d = d_o)$$

$$= 1 - \frac{(d_i)^2}{(d_o)_2} = 1 - k^2 = 1 - (0.5)^2 = 0.75 \textbf{ Ans.}$$

Comparison of strength

We know that strength of the hollow shaft,

$$T_H = \frac{\pi}{16} \times \tau\,(d_o)^3\,(1 - k^4) \qquad ...(iii)$$

and strength of the solid shaft,

$$T_S = \frac{\pi}{16} \times \tau \times d^3 \qquad ...(iv)$$

Dividing equation (*iii*) by equation (*iv*), we get

$$\frac{T_H}{T_S} = \frac{(d_o)^3\,(1 - k^4)}{d^3} = \frac{(d_o)^3\,(1 - k^4)}{(d_o)^3} = 1 - k^4 \qquad ...(\because d = d_o)$$

$$= 1 - (0.5)^4 = 0.9375 \textbf{ Ans.}$$

Comparison of stiffness

We know that stiffness

$$= \frac{T}{\theta} = \frac{G \times J}{L}$$

$\therefore$ Stiffness of a hollow shaft,

$$S_H = \frac{G}{L} \times \frac{\pi}{32}\left[(d_o)^4 - (d_i)^4\right] \qquad ...(v)$$

and stiffness of a solid shaft,

Shaft Coupling

Shafts are usually available up to 7 meters length due to inconvenience in transport. In order to have a greater length, it becomes necessary to join two or more pieces of the shaft by means of a coupling.

Shaft couplings are used in machinery for several purposes, the most common of which are the following:

1. To provide for the connection of shafts of units those are manufactured separately such as a motor and generator and to provide for disconnection for repairs or alternations.

2. To provide for misalignment of the shafts or to introduce mechanical flexibility.

3. To reduce the transmission of shock loads from one shaft to another.

4. To introduce protection against overloads.

5. It should have no projecting parts.

Types of Shafts Couplings

Shaft couplings are divided into two main groups as follows:

1. Rigid coupling. It is used to connect two shafts which are perfectly aligned. Following types of rigid coupling are important from the subject point of view:

(a) Sleeve or muff coupling.

(b) Clamp or split-muff or compression coupling, and

(c) Flange coupling.

2. Flexible coupling. It is used to connect two shafts having both lateral and angular misalignment. Following types of flexible coupling are important from the subject point of view:

(a) Bushed pin type coupling,

(b) Universal coupling, and

(c) Oldham coupling.

Sleeve or Muff-coupling

It is the simplest type of rigid coupling, made of cast iron. It consists of a hollow cylinder whose inner diameter is the same as that of the shaft. It is fitted over the ends of the two shafts by means of a gib head key, as shown in Fig. The power is transmitted from one shaft to the other shaft by means of a key and a sleeve. It is, therefore, necessary that all the elements must be strong enough to transmit the torque. The usual proportions of a cast iron sleeve coupling are as follows:

Outer diameter of the sleeve, $D = 2d + 13$ mm

Note: The depth of the keyway in each of the shafts to be connected should be exactly the same and the diameters should also be same. If these conditions are not satisfied, then the key will be bedded on one shaft while in the other it will be loose. In order to prevent this, the key is made in two parts which may be driven from the same end for each shaft or they may be driven from opposite ends.

Since the crushing stress for the key material is twice the shearing stress, therefore a square key may be used.

Then, Thickness of key, t = w = 18 mm **Ans.**

We know that length of key in each shaft,

l = L / 2 = 195 / 2 = 97.5 mm **Ans.**

Let us now check the induced shear and crushing stresses in the key. First of all, let us consider shearing of the key. We know that torque transmitted (T),

$$1100 \times 10^3 = l \times w \times \tau_s \times \frac{d}{2} = 97.5 \times 18 \times \tau_s \times \frac{55}{2} = 48.2 \times 10^3 \, \tau_s$$

$$\tau_s = 1100 \times 10^3 / 48.2 \times 10^3 = 22.8 \text{ N/mm}^2$$

Now considering crushing of the key. We know that torque transmitted (*T*),

$$1100 \times 10^3 = l \times \frac{t}{2} \times \sigma_{cs} \times \frac{d}{2} = 97.5 \times \frac{18}{2} \times \sigma_{cs} \times \frac{55}{2} = 24.1 \times 10^3 \, \sigma_{cs}$$

$$\sigma_{cs} - 1100 \times 10^3 / 24.1 \times 10^3 - 45.6 \text{ N/mm}^2$$

Since the induced shear and crushing stresses are less than the permissible stresses, therefore the design of key is safe.

Clamp or Compression Coupling or split muff coupling

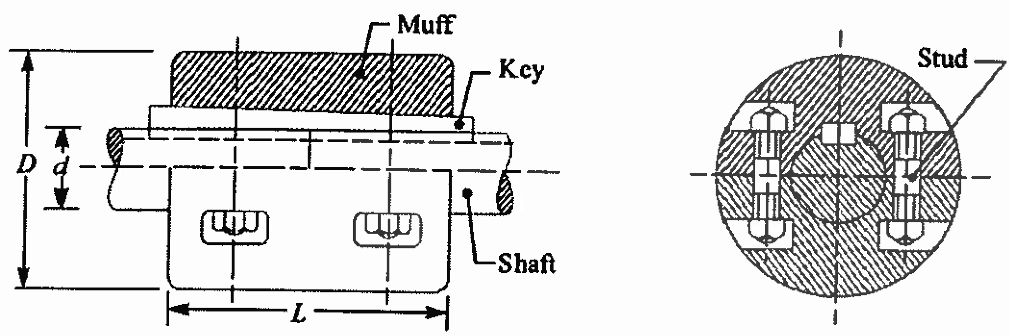
It is also known as **split muff coupling.** In this case, the muff or sleeve is made into two halves and are bolted together as shown in Fig. The halves of the muff are made of cast iron. The shaft ends are made to a butt each other and a single key is fitted directly in the keyways of both the shafts. One-half of the muff is fixed from below and the other half is placed from above. Both the halves are held together by means of mild steel studs or bolts and nuts. The number of bolts may be two, four or six. The nuts are recessed into the bodies of the muff castings. This coupling may be used for heavy duty and moderate speeds. The advantage of this coupling is that the position of the shafts need not be changed for assembling or disassembling of the coupling. The usual proportions of the muff for the clamp or compression coupling are:

Diameter of the muff or sleeve, $D = 2d + 13$ mm

Length of the muff or sleeve, $L = 3.5\ d$

Where d = Diameter of the shaft.

In the clamp or compression coupling, the power is transmitted from one shaft to the other by means of key and the friction between the muff and shaft. In designing this type of coupling, the following procedure may be adopted.



1. Design of muff and key

The muff and key are designed in the similar way as discussed in muff coupling.

2. Design of clamping bolts

Let T = Torque transmitted by the shaft,

d = Diameter of shaft,

d_b = Root or effective diameter of bolt,

n = Number of bolts,

σ_t = Permissible tensile stress for bolt material,

μ = Coefficient of friction between the muff and shaft, and

L = Length of muff.

1. Unprotected type flange coupling. In an unprotected type flange coupling, as shown in Fig.1, each shaft is keyed to the boss of a flange with a counter sunk key and the flanges are coupled together by means of bolts. Generally, three, four or six bolts are used. The keys are staggered at right angle along the circumference of the shafts in order to divide the weakening effect caused by keyways.

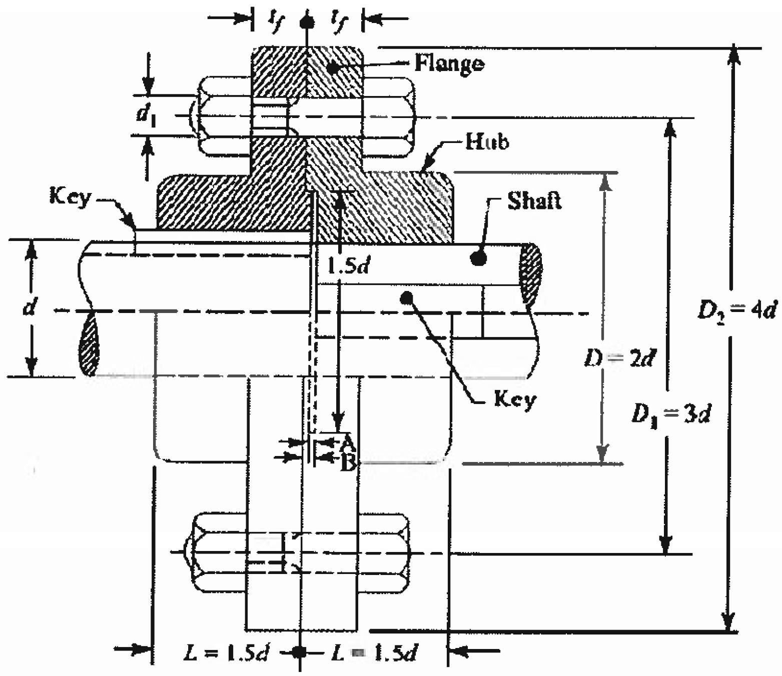


Fig.1 Unprotected Type Flange Coupling.

The usual proportions for an unprotected type cast iron flange couplings, as shown in Fig.1, are as follows:

If d is the diameter of the shaft or inner diameter of the hub, then Outside diameter of hub,

$D = 2\,d$

Length of hub, $L = 1.5\,d$

Pitch circle diameter of bolts, $D_1 = 3d$

Outside diameter of flange,

$D_2 = D_1 + (D_1 - D) = 2\,D_1 - D = 4\,d$

Thickness of flange, $t_f = 0.5\,d$

Number of bolts = 3, for d upto 40 mm

= 4, for d upto 100 mm

= 6, for d upto 180 mm

The flanges are held together by means of tapered headless bolts, numbering from four to twelve depending upon the diameter of shaft. The other proportions for the marine type flange coupling are taken as follows:

Thickness of flange = d / 3

Taper of bolt = 1 in 20 to 1 in 40

Pitch circle diameter of bolts, D_1 = 1.6 d

Outside diameter of flange, D_2 = 2.2 d

Design of Flange Coupling

Consider a flange coupling as shown in Fig.1 and Fig.2.

Let d = Diameter of shaft or inner diameter of hub,

D = Outer diameter of hub,

D_1 = Nominal or outside diameter of bolt,

D_1 = Diameter of bolt circle,

n = Number of bolts,

t_f = Thickness of flange,

τ_s, τ_b and τ_k = Allowable shear stress for shaft, bolt and key material respectively

τ_c = Allowable shear stress for the flange material i.e. cast iron,

σ_{cb}, and σ_{ck} = Allowable crushing stress for bolt and key material respectively.

The flange coupling is designed as discussed below:

1. Design for hub

The hub is designed by considering it as a hollow shaft, transmitting the same torque (T) as that of a solid shaft.

$$T = \frac{\pi}{16} \times \tau_c \left(\frac{D^4 - d^4}{D} \right)$$

The outer diameter of hub is usually taken as twice the diameter of shaft. Therefore from the above relation, the induced shearing stress in the hub may be checked.

The length of hub (L) is taken as 1.5 d.

2. Design for key

The key is designed with usual proportions and then checked for shearing and crushing stresses. The material of key is usually the same as that of shaft. The length of key is taken equal to the length of hub.

3. Design for flange

Problem: Design a cast iron protective type flange coupling to transmit 15 kW at 900 r.p.m. from an electric motor to a compressor. The service factor may be assumed as 1.35. The following permissible stresses may be used :

Shear stress for shaft, bolt and key material = 40 MPa

Crushing stress for bolt and key = 80 MPa

Shear stress for cast iron = 8 MPa

Draw a neat sketch of the coupling.

Solution. Given: P = 15 kW = 15 × 103 W; N = 900 r.p.m. ; Service factor = 1.35 ; $\tau_s = \tau_b = \tau_k$ = 40 MPa = 40 N/mm^2 ; $\sigma_{cb} = \sigma_{ck}$ = 80 MPa = 80 N/mm^2 ; τ_c = 8 MPa = 8 N/mm^2.

The protective type flange coupling is designed as discussed below:

1. Design for hub

First of all, let us find the diameter of the shaft (d). We know that the torque transmitted by the shaft,

$$T = \frac{P \times 60}{2\pi N} = \frac{15 \times 10^3 \times 60}{2\pi \times 900} = 159.13 \text{ N-m}$$

Since the service factor is 1.35, therefore the maximum torque transmitted by the shaft, T_{max} = 1.35 × 159.13 = 215 N-m = 215 × 103 N-mm

We know that the torque transmitted by the shaft (T),

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 40 \times d^3 = 7.86\, d^3$$

$$d^3 = 215 \times 10^3 / 7.86 = 27.4 \times 10^3 \quad \text{or} \quad d = 30.1 \text{ say } 35 \text{ mm Ans.}$$

We know that outer diameter of the hub,

D = 2d = 2 × 35 = 70 mm Ans.

And length of hub, L = 1.5 d = 1.5 × 35 = 52.5 mm Ans.

Let us now check the induced shear stress for the hub material which is cast iron. Considering the hub as a hollow shaft. We know that the maximum torque transmitted (T_{max}).

$$215 \times 10^3 = \frac{\pi}{16} \times \tau_c \left[\frac{D^4 - d^4}{D}\right] = \frac{\pi}{16} \times \tau_c \left[\frac{(70)^4 - (35)^4}{70}\right] = 63\,147\, \tau_c$$

Then, τ_c = 215 × 103/63 147 = 3.4 N/mm2 = 3.4 MPa

Since the induced shear stress for the hub material (i.e. cast iron) is less than the permissible value of 8 MPa, therefore the design of hub is safe.

2. Design for key

The bolts are subjected to shear stress due to the torque transmitted. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = \frac{\pi}{4}(d_1)^2\ \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4}(d_1)^2\ 40 \times 3 \times \frac{105}{2} = 4950\ (d_1)^2$$

$$(d_1)^2 = 215 \times 103/4950 = 43.43 \text{ or } d_1 = 6.6 \text{ mm}$$

Assuming coarse threads, the nearest standard size of bolt is M 8. Ans.

Other proportions of the flange are taken as follows:

Outer diameter of the flange,

$D_2 = 4\ d = 4 \times 35 = 140$ mm Ans.

Thickness of the protective circumferential flange,

$t_p = 0.25\ d = 0.25 \times 35 = 8.75$ say 10 mm Ans.

In designing the bushed-pin flexible coupling, the proportions of the rigid type flange coupling are modified. The main modification is to reduce the bearing pressure on the rubber or leather bushes and it should not exceed 0.5 N/mm2. In order to keep the low bearing pressure, the pitch circle diameter and the pin size is increased.

Let l = Length of bush in the flange,

D_2 = Diameter of bush,

P_b = Bearing pressure on the bush or pin,

n = Number of pins, and

D_1 = Diameter of pitch circle of the pins.

We know that bearing load acting on each pin,

$W = p_b \times d_2 \times l$

Then, Total bearing load on the bush or pins

$= W \times n = p_b \times d_2 \times l \times n$

And the torque transmitted by the coupling,

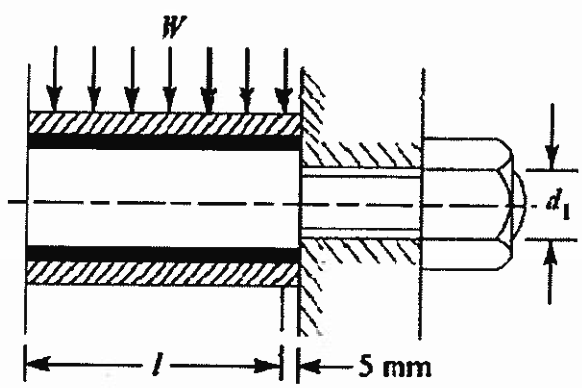
$$T = W \times n\left(\frac{D_1}{2}\right) = p_b \times d_2 \times l \times n\left(\frac{D_1}{2}\right)$$

The threaded portion of the pin in the right hand flange should be a tapping fit in the coupling hole to avoid bending stresses.

The threaded length of the pin should be as small as possible so that the direct shear stress can be taken by the unthreaded neck.

Direct shear stress due to pure torsion in the coupling halves,

$$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2}$$



Since the pin and the rubber or leather bush is not rigidly held in the left hand flange, therefore the tangential load (W) at the enlarged portion will exert a bending action on the pin as shown in Fig. The bush portion of the pin acts as a cantilever beam of length l. Assuming a uniform distribution of the load W along the bush, the maximum bending moment on the pin,

Let l = Length of the bush in the flange.

We know that the bearing load acting on each pin,

$W = p_b \times d_2 \times l = 0.8 \times 40 \times l = 32\,l$ N

And the maximum torque transmitted by the coupling (T_{max}),

$$382 \times 10^3 = W \times n \times \frac{D_1}{2} = 32\,l \times 6 \times \frac{132}{2} = 12\,672\,l$$

$l = 382 \times 103/12\,672 = 30.1$ say 32 mm

And $W = 32\,l = 32 \times 32 = 1024$ N

So, Direct stress due to pure torsion in the coupling halves,

$$\tau = \frac{W}{\frac{\pi}{4}(d_1)^2} = \frac{1024}{\frac{\pi}{4}(20)^2} = 3.26 \text{ N/mm}^2$$

Since the pin and the rubber bush are not rigidly held in the left hand flange, therefore the tangential load (W) at the enlarged portion will exert a bending action on the pin. Assuming a uniform distribution of load (W) along the bush, the maximum bending moment on the pin,

$$M = W\left(\frac{l}{2} + 5\right) = 1024\left(\frac{32}{2} + 5\right) = 21\,504 \text{ N-mm}$$

$$Z = \frac{\pi}{32}(d_1)^3 = \frac{\pi}{32}(20)^3 = 785.5 \text{ mm}^3$$

$$\sigma = \frac{M}{Z} = \frac{21\,504}{785.5} = 27.4 \text{ N/mm}^2$$

Maximum principal stress

$$= \frac{1}{2}\left[\sigma + \sqrt{\sigma^2 + 4\tau^2}\right] = \frac{1}{2}\left[27.4 + \sqrt{(27.4)^2 + 4\,(3.26)^2}\right]$$

$$= 13.7 + 14.1 = 27.8 \text{ N/mm}^2$$

And maximum shear stress

$$= \frac{1}{2}\left[\sqrt{\sigma^2 + 4\tau^2}\right] = \frac{1}{2}\left[\sqrt{(27.4)^2 + 4\,(3.26)^2}\right] = 14.1 \text{ N/mm}^2$$

Since the maximum principal stress and maximum shear stress are within limits, therefore the design is safe.

2. Design for hub

We know that the outer diameter of the hub, We know that the maximum torque transmitted (T_{max}),

$$382 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (80)^2}{2} \times \tau_c \times 20 = 201 \times 10^3 \tau_c$$

τ_c =382 × 103 / 201 × 103 = 1.9 N/mm2 = 1.9 MPa

Since the induced shear stress in the flange of cast iron is less than 15 MPa, therefore the design of flange is safe.

2. Design for key

Since the crushing stress for the key material is twice its shear stress (i.e. $\sigma_{ck} = 2\tau_k$), therefore a square key may be used. From DDB, we find that for a shaft of 35 mm diameter,

Width of key, w = 12 mm Ans.

And thickness of key, t = w = 12 mm Ans.

The length of key (l) is taken equal to the length of hub.

Then, l = L = 52.5 mm Ans.

Let us now check the induced stresses in the key by considering it in shearing and crushing.

Considering the key in shearing. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = l \times w \times \tau_k \times \frac{d}{2} = 52.5 \times 12 \times \tau_k \times \frac{35}{2} = 11\ 025\ \tau_k$$

Then, τ_k = 215 × 103/11 025 = 19.5 N/mm2 = 19.5 MPa

Considering the key in crushing. We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = l \times \frac{t}{2} \times \sigma_{ck} \times \frac{d}{2} = 52.5 \times \frac{12}{2} \times \sigma_{ck} \times \frac{35}{2} = 5512.5\ \sigma_{ck}$$

Σ_{ck} = 215 × 103/ 5512.5 = 39 N/mm^2 = 39 MPa.

Since the induced shear and crushing stresses in the key are less than the permissible stresses, therefore the design for key is safe.

3. Design for flange

The thickness of flange (t_f) is taken as 0.5 d.

Then, t_f = 0.5 d = 0.5 × 35 = 17.5 mm Ans.

Let us now check the induced shearing stress in the flange by considering the flange at the junction of the hub in shear.

We know that the maximum torque transmitted (T_{max}),

$$215 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi (70)^2}{2} \times \tau_c \times 17.5 = 134\ 713\ \tau_c$$

τ_c = 215 × 103/134 713 = 1.6 N/mm2 = 1.6 MPa

Since the induced shear stress in the flange is less than 8 MPa, therefore the design of flange is safe.

4. Design for bolts

Let d_1 = Nominal diameter of bolts.

Since the diameter of the shaft is 35 mm, therefore let us take the number of bolts,

n = 3 and pitch circle diameter of bolts,

Problem:

Two 35 mm shafts are connected by a flanged coupling. The flanges are fitted with 6 bolts on 125 mm bolt circle. The shafts transmit a torque of 800 N-m at 350 r.p.m. For the safe stresses mentioned below, calculate 1. Diameter of bolts; 2. Thickness of flanges; 3. Key dimensions ; 4. Hub length; and 5. Power transmitted. Safe shear stress for shaft material = 63 MPa Safe stress for bolt material = 56 MPa Safe stress for cast iron coupling = 10 MPa Safe stress for key material = 46 MPa

Solution. Given: d = 35 mm; n = 6; D1 = 125 mm; T = 800 N-m = 800 × 103 N-mm; N = 350 r.p.m.; τs = 63 MPa = 63 N/mm^2; τ_b = 56 MPa = 56 N/mm^2 ; τ_c = 10 MPa = 10 N/mm^2 ; τ_k = 46 MPa = 46 N/mm^2.

1. Diameter of bolts

Let d_1 = Nominal or outside diameter of bolt. We know that the torque transmitted (T),

$$800 \times 10^3 = \frac{\pi}{4}(d_1)^2\,\tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4}(d_1)^2\,56 \times 6 \times \frac{125}{2} = 16\,495\,(d_1)^2$$

$(d_1)^2 = 800 \times 10^3 / 16\,495 = 48.5$ or $d_1 =$ 6.96 say 8 mm **Ans.**

2. Thickness of flanges

Let t_f = Thickness of flanges.

We know that the torque transmitted (T),

$$800 \times 10^3 = \frac{\pi D^2}{2} \times \tau_c \times t_f = \frac{\pi\,(2 \times 35)^2}{2} \times 10 \times t_f = 76\,980\,t_f \quad \ldots (\because D = 2d)$$

t_f = 800 × 103 / 76 980 = 10.4 say 12 mm **Ans.**

3. Key dimensions

From Table 13.1, we find that the proportions of key for a 35 mm diameter shaft are:

Width of key, w = 12 mm **Ans.**

And thickness of key, t = 8 mm**Ans.**

The length of key (l) is taken equal to the length of hub (L).

l = L = 1.5 d = 1.5 × 35 = 52.5 mm

Let us now check the induced shear stress in the key. We know that the torque transmitted (T),

$$T = \frac{P \times 60}{2\pi N} = \frac{3 \times 10^6 \times 60}{2\pi \times 100} = 286 \times 10^3 \text{ N-m} = 286 \times 10^6 \text{ N-mm}$$

We also know that torque transmitted by the shaft (T),

$$286 \times 10^6 = \frac{\pi}{16} \times \tau_s \times d^3 = \frac{\pi}{16} \times 60 \times d^3 = 11.78\, d^3$$

$$d_3 = 286 \times 10^6 / 11.78 = 24.3 \times 10^6$$

or $d = 2.89 \times 10^2 = 289$ say 300 mm **Ans.**

2. Diameter of bolts

Let d_1 = Nominal diameter of bolts.

The bolts are subjected to shear stress due to the torque transmitted. We know that torque transmitted (T),

$$286 \times 10^6 = \frac{\pi}{4}(d_1)^2 \tau_b \times n \times \frac{D_1}{2} = \frac{\pi}{4} \times (d_1)^2\, 60 \times 8 \times \frac{1.6 \times 300}{2}$$

$= 90\,490\,(d_1)^2$ ... (Since $D_1 = 1.6$ d)

So, $(d_1)^2 = 286 \times 10^6 / 90\,490 = 3160$ or $d_1 = 56.2$ mm

Assuming coarse threads, the standard diameter of the bolt is 60 mm (M 60). The taper on the bolt may be taken from 1 in 20 to 1 in 40. **Ans.**

3. Thickness of flange

The thickness of flange (t_f) is taken as d / 3.

So, tf = d / 3 = 300/3 = 100 mm **Ans.**

Let us now check the induced shear stress in the flange by considering the flange at the junction of the shaft in shear. We know that the torque transmitted (T),

$$286 \times 10^6 = \frac{\pi d^2}{2} \times \tau_s \times t_f = \frac{\pi (300)^2}{2} \times \tau_s \times 100 = 14.14 \times 10^6\, \tau_s$$

$\tau_s = 286 \times 10^6 / 14.14 \times 10^6 = 20.2$ N/mm^2 = 20.2 MPa

Since the induced shear stress in the *flange is less than the permissible shear stress of 60 MPa, therefore the thickness of flange (tf = 100 mm) is safe.

4. Diameter of flange

The diameter of flange (D_2) is taken as 2.2 d.

So, $D_2 = 2.2$ d $= 2.2 \times 300 = 660$ mm **Ans.**

Question Bank

UNIT -I

1. What are the factors to be considered for the selection of materials for the design of machine elements? Discuss.
2. Enumerate the most commonly used engineering materials and state at least one important property and one application of each.
3. Define the following properties of a material : (i) Ductility, (ii) Toughness, (iii) Hardness, and (iv) Creep.
4. Distinguish clearly amongst cast iron, wrought iron and steel regarding their constituents and properties
5. Define plain carbon steel. How it is designated according to Indian standards?
6. Give the composition of 35 Mn 2 Mo 45 steel. List its main uses
7. Enumerate the various manufacturing methods of machine parts which a designer should know.
8. What are fits and tolerances? How are they designated?
9. Write short notes on the following : (a) Interchangeability; (b) Tolerance; (c) Allowance; and (d) Fits.
10. State briefly unilateral system of tolerances covering the points of definition, application and advantages over the bilateral system.
11. What is meant by 'hole basis system' and 'shaft basis system'? Which one is preferred and why?
12. Discuss the Indian standard system of limits and fits.
13. Define the terms load , stress and strain. Discuss the various types of stresses and strain.
14. What do you mean by factor of safety?
15. What is meant by working stress and how it is calculated from the ultimate stress or yield stress of a material? What will be the factor of safety in each case for different types of loading
16. Define the following : (a) Poisson's ratio, (b) Volumetric strain, and (c) Bulk modulus
17. Write short notes on : (a) Resilience (b) Proof resilience, and (c) Modulus of resilience
18. Derive a relation for the shear stress developed in a shaft, when it is subjected to torsion.
19. State the assumptions made in deriving a bending formula.
20. Prove the relation: $M/I = \sigma/y = E/R$
21. Write short note on maximum shear stress theory verses maximum strain energy theory.

UNIT -II

1. Explain the following terms in connection with design of machine members subjected to variable loads: (a) Endurance limit, (b) Size factor, (c) Surface finish factor, and (d) Notch sensitivity.
2. What is meant by endurance strength of a material? How do the size and surface condition of a component and type of load affect such strength?
3. Write a note on the influence of various factors of the endurance limit of a ductile material.
4. What is meant by `stress concentration'? How do you take it into consideration in case of a component subjected to dynamic loading?
5. Illustrate how the stress concentration in a component can be reduced.
6. Explain how the factor of safety is determined under steady and varying loading by different methods.

UNIT -IV

1. What is a cotter joint? Explain with the help of a neat sketch, how a cotter joint is made.
2. What are the applications of a cotteredjoint?
3. Discuss the design procedure of spigot and socket cotter joint.
4. Why gibs are used in a cotter joint? Explain with the help of a neat sketch the use of single and double gib.
5. Describe the design procedure of a gib and cotter joint.
6. Distinguish between cotter joint and knuckle joint.
7. Sketch two views of a knuckle joint and write the equations showing the strength of joint for the most probable modes of failure.
8. Explain the purpose of a turn buckle. Describe its design procedure.
9. What is a key? State its function.
10. How are the keys classified? Draw neat sketches of different types of keys and state their applications.
11. What are the considerations in the design of dimensions of formed and parallel key having rectangular cross-section?
12. Write short note on the splined shaft covering the points of application, different types and method of manufacture.
13. What is the effect of keyway cut into the shaft?

UNIT -V

1. Discuss the function of a coupling. Give at least three practical applications.
2. Describe, with the help of neat sketches, the types of various shaft couplings mentioning the uses of each type.
3. How does the working of a clamp coupling differ from that of a muff coupling? Explain.
4. Sketch a protective type flange coupling and indicate there on its leading dimensions for shaft size of 'd'.
5. What are flexible couplings and what are their applications? Illustrate your answer with suitable examples and sketches.
6. Write short note on universal coupling.
7. Why are two universal joints often used when there is angular misalignment between two shafts?
8. How the shaft is designed when it is subjected to twisting moment only?
9. Define equivalent twisting moment and equivalent bending moment. State when these two terms are used in design of shafts.
10. When the shaft is subjected to fluctuating loads, what will be the equivalent twisting moment and equivalent bending moment ?
11. What do you understand by torsional rigidity and lateral rigidity.
12. A hollow shaft has greater strength and stiffness than solid shaft of equal weight. Explain.
13. Under what circumstances are hollow shafts preferred over solid shafts ? Give any two examples where hollow shafts are used. How are they generally manufactured ?

Assignment NO.1

1. Write a brief note on different phases of design
2. Define fatigue and endurance limit?
3. What are the general considerations in the design of machine elements
4. An electric motor weighing 500N is mounted on a short cantilever beam of uniform rectangular cross section. The weight of motor acts at a distance of 300mm from the support. The depth of the section is twice the width. Determine the cross section of the beam. The allowable stress in the beam is 40N/mm2
5. State and explain theories of failures under static load?
6. The load on a bolt consists of an axial pull of 10kN together with a transverse shear force of 5kN. Find the diameter of bolt required according to i). Maximum principal stress theory; ii). Maximum shear stress theory; iii). Maximum principal strain theory; iv). Maximum strain energy theory; and v). Maximum distortion energy theory. Take permissible tensile stress at elastic limit = 100MPa and Poisson's ratio = 0.3.
7. A 50 mm diameter shaft is made from carbon steel having ultimate tensile strength of 630MPa. It is subjected to a torque which fluctuates between 2000 N-m to – 800 Nm. Using Soderberg method, calculate the factor of safety. Assume suitable values for any other data needed.
8. Define the term 'factor of safety'. Explain the influence of stress raiser on impact strength.
9. What do you understand by the terms riveted joint and welded joints
10. An automobile leaf spring is subjected to cyclic stress such that the average stress is 150Mpa, variable stress is 350Mpa; the material properties are; ultimate strength = 400Mpa; yield strength = 350Mpa; endurance limit = 270Mpa; estimate the factor of safety using Goodman method and Soderberg method?
11. A double riveted lap joint is made between 15-mm thick plates. The rivet diameter and pitch are 25 mm and 75 mm respectively. If the ultimate stresses are 400 MPa in tension, 320 MPa in shear and 640 MPa in crushing, find the minimum force per pitch which will rupture the joint. If the above joint is subjected to a load such that the factor of safety is 2, find out the actual stresses developed in the plates and the rivets.
12. Enumerate the list of failures of riveted joints.
13. A shaft is made of steel, ultimate tensile strength 700 MPa and yield point 420 MPa is subjected to a torque varying from 200N m anti-clockwise to 600 N m clockwise. Calculate the diameter of the shaft if the factor of safety is 2 and it is based on the yield point and the endurance strength in shear.

Assignment NO.2

1. List out the various types of stresses induced in shafts
2. What is a Gib? What is the uniqueness of Gib headed cotter joint
3. Distinguish between rigid and flexible couplings.
4. Design and draw a muff coupling to transmit 50 HP at 120 rpm. The shaft and key are made of the same material having allowable shear stress of 30N/mm2and compressor stress of 80N/mm2. The flange is made, as cast Iron with allowable shear stress is 15N/mm2
5. Design a cotter joint to support a load varying from 30KN in compression to 30KN in tension. The material used is carbon steel for which the following allowable stresses may be used. The load is applied statically. Tensile stress = compressive stress = 50MPa; shear stress = 35MPa and crushing shear stress = 90MPa.
6. Design a shaft to transmit power from an electric motor to a lathe head stock through a pulley by means of a belt drive. The pulley weighs 200N and is located at 300mm from the centre of the bearing. The diameter of the pulley is 200mm and the maximum power transmitted is 1KW at 120rpm. The angle of lap of the belt is $180^0$and coefficient of friction betweenthe belt and the pulley is 0.3. The shock and fatigue factors for bending and twisting are 1.5 and 2.0 respectively. The allowable shear stress in the shaft may be taken as 35MPa
7. The head of an air compressor cylinder is attached by eight bolts. The cylinder bore diameter is 80 mm and the maximum working pressure is limited to 3 MPa. Determine the diameter of bolt, when a copper-asbestos is used between the head and the cylinder and an initial compressive load of 5 KN is required on the gasket for a leak proof joint. Ultimate tensile strength, Yield strength and endurance limit for material are 400 MPa, 340 MPa and 200 MPa respectively. A factor of safety of three is desired and the stress-concentration factor of 2.84 may be assumed.
8. Design a socket and spigot type of cotter joint to sustain an axial load of 100kN. The material selected for the joint has the following design stresses. σt = 120 MPa, σc = 160 MPa and σpb = 60 MPa
9. Design a bushed-pin type flexible coupling for connecting a motor shaft to a pump shaft, with the following service conditions: Power to be transmitted =40kW Speed of the motor shaft =1000rpm Diameter of motor and pump shafts =45mm Bearing pressure on the rubber bush =0.7N/mm2 Allowable stress in the pins =60MPa
10. A 350mm diameter solid shaft is used to drive the propeller of a marine vessel. It is necessary to reduce the weight of the shaft by 80%. What would be the dimensions of a hollow shaft made of the same material as the solid shaft?

Tutorial Problems

UNIT -I

1. A steel rod of 25 mm diameter is fitted inside a brass tube of 25 mm internal diameter and 375 mm external diameter. The projecting ends of the steel rod are provided with nuts and washers. The nuts are tightened up so as to produce a pull of 5 kN in the rod. The compound is then placed in a lathe and the brass is turned down to 4 mm thickness. Calculate the stresses in the two materials.
2. A steel rod of 20 mm diameter passes centrally through a copper tube of external diameter 40 mm and internal diameter 20 mm. The tube is closed at each end with the help of rigid washers (of negligible thickness) which are screwed by the nuts. The nuts are tightened until the compressive load on the copper tube is 50 kN. Determine the stresses in the rod and the tube, when the temperature of whole assembly falls by 50°C. Take Es = 200 GPa ;Ec = 100 GPa ; αs = 12 × 10–6/°C and αc = 18 × 106/°C.
3. The following results were obtained in a tensile test on a mild steel specimen of original diameter 20 mm and gauge length 40 mm. Load at limit of proportionality = 80 kN Extension at 80 kN load = 0.048 mm Load at yield point = 85 kN Maximum load = 150 kN When the two parts were fitted together after being broken, the length between gauge length was found to be 55.6 mm and the diameter at the neck was 15.8 mm. Calculate Young's modulus, yield stress, ultimate tensile stress, percentage elongation and percentage reduction in area.
4. A steel shaft 50 mm diameter and 500 mm long is subjected to a twisting moment of 1100 N-m, the total angle of twist being 0.6°. Find the maximum shearing stress developed in the shzaft and modulus of rigidity.
5. Design a suitable diameter for a circular shaft required to transmit 90 kW at 180 r.p.m. The shear stress in the shaft is not to exceed 70 MPa and the maximum torque exceeds the mean by 40%. Also find the angle of twist in a length of 2 metres. Take C = 90 GPa.
6. Compare the weights of equal lengths of hollow shaft and solid shaft to transmit a given torque for the same maximum shear stress. The material for both the shafts is same and inside diameter is 2/3 of outside diameter in case of hollow shaft.
7. A rotating shaft of 16 mm diameter is made of plain carbon steel. It is subjected to axial load of 5000 N, a steady torque of 50 N-m and maximum bending moment of 75 N-m. Calculate the factor of safety available based on 1. Maximum normal stress theory; and 2. Maximum shear stress theory. Assume yield strength as 400 MPa for plain carbon steel. If all other data remaining same, what maximum yield strength of shaft material would be necessary using factor of safety of 1.686 and maximum distortion energy theory of failure. Comment on the result you get.
8. A cast iron pulley transmits 20 kW at 300 r.p.m. The diameter of the pulley is 550 mm and has four straight arms of elliptical cross-section in which the major axis is twice the minor axis. Find the dimensions of the arm, if the allowable bending stress is 15 MPa.

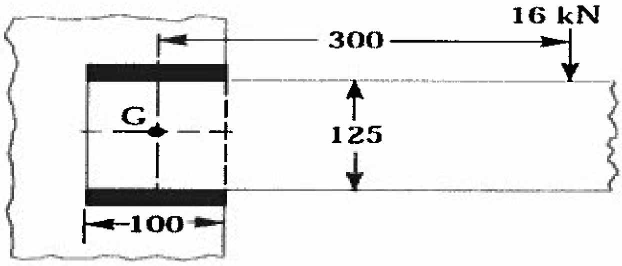
UNIT -II

1. A rectangular plate 50 mm × 10 mm with a hole 10 mm diameter is subjected to an axial load of 10 kN. Taking stress concentration into account, find the maximum stress induced. [Ans. 50 MPa] .
2. A stepped shaft has maximum diameter 45 mm and minimum diameter 30 mm. The fillet radius is 6 mm. If the shaft is subjected to an axial load of 10 kN, find the maximum stress induced, taking stress concentration into account
3. Determine the diameter of a tensile member of a circular cross-section. The following data is given : Maximum tensile load = 10 kN; Maximum compressive load = 5 kN; Ultimate tensile strength = 600 MPa; Yield point = 380 MPa; Endurance limit = 290 MPa; Factor of safety = 4; Stress concentration factor = 2.2
4. Find the diameter of a shaft made of 37 Mn 2 steel having the ultimate tensile strength as 600 MPa and yield stress as 440 MPa. The shaft is subjected to completely reversed axial load of 200 kN. Neglect stress concentration factor and assume surface finish factor as 0.8. The factor of safety may be taken as 1.5.
5. A cold drawn steel rod of circular cross-section is subjected to a variable bending moment of 565 Nm to 1130 N-m as the axial load varies from 4500 N to 13 500 N. The maximum bending moment occurs at the same instant that the axial load is maximum. Determine the required diameter of the rod for a factor of safety 2. Neglect any stress concentration and column effect. Assume the following values: Ultimate strength = 550 MPa Yield strength = 470 MPa Size factor = 0.85 Surface finish factor = 0.89 Correction factors = 1.0 for bending = 0.7 for axial load The endurance limit in reversed bending may be taken as one-half the ultimate strength
6. A steel shaft is subjected to completely reversed bending moment of 800 N-m and a cyclic twisting moment of 500 N-m which varies over a range of ± 40%. Determine the diameter of shaft if a reduction factor of 1.2 is applied to the variable component of bending stress and shearing stress. Assume(a) that the maximum bending and shearing stresses are in phase; (b) that the tensile yield point is the limiting stress for steady state component; (c) that the maximum shear strength theory can be applied; and (d) that the Goodman relation is valid. Take the following material properties: Yield strength = 500 MPa ; Ultimate strength = 800 MPa ; Endurance limit = ± 400 MPa

UNIT -III

1. A double riveted lap joint with chain riveting is to be made for joining two plates 10 mm thick. The allowable stresses are :σt = 60 MPa ; τ = 50 MPa and σc = 80 MPa. Find the rivet diameter, pitch of rivets and distance between rows of rivets. Also find the efficiency of the joint.
2. A double riveted butt joint, in which the pitch of the rivets in the outer rows is twice that in the inner rows, connects two 16 mm thick plates with two cover plates each 12 mm thick. The diameter of rivets is 22 mm. Determine the pitches of the rivets in the two rows if the working stresses are not to exceed the following limits:Tensile stress in plates = 100 MPa ; Shear stress in rivets = 75 MPa; and bearing stress in rivets and plates = 150 MPa. Make a fully dimensioned sketch of the joint by showing at least two views.
3. Design a triple riveted longitudinal double strap butt joint with unequal straps for a boiler. The inside diameter of the longest course of the drum is 1.3 metres. The joint is to be

mm long and parallel to the axis of the angle.



All dimensions in mm.

UNIT -IV

1. Two rod ends of a pump are joined by means of a cotter and spigot and socket at the ends. Design the joint for an axial load of 100 kN which alternately changes from tensile to compressive. The allowable stresses for the material used are 50 MPa in tension, 40 MPa in shear and 100 MPa in crushing
2. Two mild steel rods 40 mm diameter are to be connected by a cotter joint. The thickness of the cotter is 12 mm. Calculate the dimensions of the joint, if the maximum permissible stresses are: 46 MPa in tension ; 35 MPa in shear and 70 MPa in crushing.
3. Design a knuckle joint to connect two mild steel bars under a tensile load of 25 kN. The allowable stresses are 65 MPa in tension, 50 MPa in shear and 83 MPa in crushing.
4. A knuckle joint is required to withstand a tensile load of 25 kN. Design the joint if the permissible stresses are :σt = 56 MPa ; τ = 40 MPa and σc = 70 MPa.
5. A shaft 80 mm diameter transmits power at maximum shear stress of 63 MPa. Find the length of a 20 mm wide key required to mount a pulley on the shaft so that the stress in the key does not exceed 42 MPa.
6. Design the rectangular key for a shaft of 50 mm diameter. The shearing and crushing stresses for the key material are 42 MPa and 70 MPa.

UNIT -V

1. A hollow steel shaft transmits 600 kW at 500 r.p.m. The maximum shear stress is 62.4 MPa. Find the outside and inside diameter of the shaft, if the outer diameter is twice of inside diameter, assuming that the maximum torque is 20% greater than the mean torque.
2. A motor car shaft consists of a steel tube 30 mm internal diameter and 4 mm thick. The engine develops 10 kW at 2000 r.p.m. Find the maximum shear stress in the tube when the power is transmitted through a 4 : 1 gearing
3. Two 400 mm diameter pulleys are keyed to a simply supported shaft 500 mm apart. Each pulley is 100 mm from its support and has horizontal belts, tension ratio being 2.5. If the shear stress is to be limited to 80 MPa while transmitting 45 kW at 900 r.p.m., find the shaft diameter if it is to be used for the input-output belts being on the same or opposite sides.
4. Determine the diameter of hollow shaft having inside diameter 0.5 times the outside diameter. The permissible shear stress is limited to 200 MPa. The shaft carries a 900 mm diameter cast iron pulley. This pulley is driven by another pulley mounted on the shaft placed below it. The belt ends are parallel and vertical. The ratio of tensions in the belt is 3. The pulley on the hollow shaft weighs 800 N and overhangs the nearest bearing by 250 mm. The pulley is to transmit 35 kW at 400 r.p.m.

Code No: RT31033 **R13** SET - 1

III B.Tech I Semester Supplementary Examinations, October/November-2019

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**

PART –A (22 Marks)

1	a)	Define a fit and a tolerance.	[3M]
	b)	What is a fatigue stress concentration factor?	[4M]
	c)	What do understand about a bolt of uniform strength?	[4M]
	d)	Draw a sketch of triple riveted double cover butt joint with zig - zag type of riveting.	[4M]
	e)	What is the effect of keyway cut in to the shaft?	[3M]
	f)	Write short note on leaf springs.	[4M]

PART –B (48 Marks)

2	a)	A cantilever of span 500 mm carries a vertical download load of 6 KN at free end. Assume yield value of 350 MPa and factor of safety of 3. Find the economical section for the cantilever among: i) a circular cross section of diameter 'd', ii) rectangular section of depth 'h' and width 't' with $-$ = 2.	[4M]
	b)	State and explain various theories of failure under static loading.	[8M]
	c)	Find the diameter of shaft required to transmit 60 kW at 150 rpm if the maximum torque is likely to exceed the mean torque by 25% for a maximum permissible torsional shear stress of 60 N/mm^2. Also find the angle of twist for a length of 2.5 meters. Take G = 80 GPa.	[4M]
3	a)	Explain the types of fluctuating stresses.	[3M]
	b)	A machine component is subjected to a flexural stress which fluctuates between +300 MN/m^2 and –150 MN/m^2. Determine the value of minimum ultimate strength according to: i). Modified Goodman relation; and ii). Soderberg relation. Take yield strength = 0.55 Ultimate strength; Endurance strength = 0.5 Ultimate strength; and factor of safety = 2.	[8M]
	c)	A steel link having a rectangular section is subjected to a repeated axial load of 50,000 N with a medium shock. Determine the section if the endurance limit be 250 MPa with a design factor 1.5. Take side ratio as 2:1. Size factor may be taken as 0.85 and surface finish factor as 0.88.	[5M]
4	a)	An air compressor cylinder of effective diameter 300 mm is subjected to air pressure of 1.5N/mm^2. The cylinder head is connected by means of 8 bolts, having yield strength of 350 N/mm^2 and endurance limit of 240 N/mm^2. The bolts are tightened with an initial pre load force of 1.5 times that of the external force. A copper gasket is used to make the joint leak proof. Assume stress concentration factor of 2.5 and factor of safety of 2. Determine the required size of the bolt.	[4M]

Code No: RT31033 **R13** SET-1

III B. Tech I Semester Regular Examinations November - 2015
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

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PART –A

1 a) Enumerate the various phases of design. [3M]
b) Explain preferred numbers and their significance. [4M]
c) Describe the causes of stress concentration. [4M]
d) Explain modified Goodman's line. [4M]
e) Draw a sketch of triple riveted double cover butt joint with zig-zag type of riveting. [3M]
f) Discuss the stresses in Helical Springs of circular wire. [4M]

PART -B

2 a) Explain the manufacturing considerations in design. [4M]
b) State and explain various theories of failure under static loading. [8M]
c) Find the diameter of shaft required to transmit 60 kW at 150 rpm if the maximum torque is likely to exceed the mean torque by 25% for a maximum permissible torsional shear stress of 60 N/mm^2. Also find the angle of twist for a length of 2.5 meters. Take G = 80 GPa. [4M]

3 a) Explain the types of fluctuating stresses. [4M]
b) A hot rolled steel shaft is subjected to a torsional moment that varies from +350 Nm to -115 Nm and an applied bending moment at a critical section varies from 445 Nm to 225 Nm. The shaft is of uniform cross section. Determine the required shaft diameter. The material has an ultimate strength of 550 MPa and yield strength of 410 MPa. Take the endurance limit as half the ultimate strength, factor of safety of 2, size factor of 0.85 and a surface finish factor of 0.62. (Using Goodman's Line). [12M]

4 a) What is the difference between caulking and fullering? Explain with the help of neat sketches. [4M]
b) Design a triple riveted longitudinal double strap butt joint with unequal straps for a boiler. The inside diameter of the drum is 1.3 meters. The joint is to be designed for a steam pressure of 2.4 N/mm^2. The working stresses to be used are σ_t=77N/mm^2, τ=62 N/mm^2; σ_c=120 N/mm^2. Assume the efficiency of the joint as 81 %. [12M]

5 a) Explain different types of keys. [4M]
b) Design a cotter joint to connect two mild steel rods for a pull of 30 kN. The maximum permissible stresses are 55N/mm^2 in tension, 40N/mm^2 in shear and 70 N/mm^2 in crushing. Draw a neat sketch of the joint. [12M]

Code No: RT31033 **R13** SET-2

III B. Tech I Semester Regular Examinations November - 2015
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

*****

PART –A

1 a) List the mechanical properties of materials. [3M]
b) Describe the methods to determine stress concentration factors. [4M]
c) Explain the endurance limit modifying factors. [4M]
d) Explain the design procedure for eccentric loaded welded joints. [4M]
e) What is the purpose of shaft coupling? [3M]
f) Explain the stresses in helical springs of circular wire. [4M]

PART -B

2 a) Find the diameter of shaft required to transmit 60 kW at 150 rpm if the maximum torque is likely to exceed the mean torque by 25% for a maximum permissible torsional shear stress of 60 N/mm^2. Also find the angle of twist for a length of 2.5 meters. Take G = 80 GPa. [6M]
b) A bolt is subjected to a direct tensile load of 20 kN and a shear load of 15 kN. Suggest the suitable size of bolt according to various theories of elastic failure, if the yield stress in simple tension is 360 MPa. A factor of safety of 3.5 should be used. Take Poisson's ratio as 0.25. [10M]

3 a) Explain the causes of stress concentration. [4M]
b) A circular cross section cantilever beam having length 130 mm. subjected to a cyclic transverse load of varying form -150 N to 350 N, FOS is 2, theoretical stress concentration factor is 1.4, notch sensitivity factor is 0.9, ultimate strength is 540 MPa, yield strength is 320 MPa. Size correction factor is 0.85. Endurance limit is 275 MPa, surface correction factor is 0.9 and notch sensitivity factor is 0.9. Determine the diameter of the beam by (i) Goodman method and (ii) Soderberg method. [12M]

4 a) Explain the bolts of uniform strength. [4M]
b) A steam engine of effective diameter 300 mm is subjected to a steam pressure of 1.5 N/mm^2. The cylinder head is connected by 8 bolts having yield point 330 N/mm^2 and endurance limit at 240 N/mm^2. The bolts are tightened with an initial preload of 1.5 times the steam load. Assume a factor of safety 2. Find the size of bolt required the stiffness factor for copper gasket may be taken as 0.5. [12M]

Code No: RT31033 **R13** SET-3

III B. Tech I Semester Regular Examinations November - 2015
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

*****

PART –A

1 a) Define fits and their significance. [3M]
b) Explain how the factor of safety is adopted in designing machine elements varies with the nature and type of load imposed on them. [4M]
c) Describe fatigue stress concentration factor. [4M]
d) Explain endurance strength and fatigue strength. [4M]
e) Explain the function of key and a cotter. [3M]
f) Explain the construction of leaf spring. [4M]

PART -B

2 a) Discuss various theories of failure. [8M]
b) Find the diameter of shaft required to transmit 60 kW at 150 rpm if the maximum torque is likely to exceed the mean torque by 25% for a maximum permissible torsional shear stress of 60 N/mm^2. Also find the angle of twist for a length of 2.5 meters. Take G = 80 GPa. [8M]

3 a) Explain the factors that affect the fatigue strength. [6M]
b) A machine member is made of plain carbon steel of ultimate strength 650 N/mm^2 and endurance limit of 300 N/mm^2. The member is subjected to a fluctuating torsional moment which varies from -200 Nm to 400 Nm. Design the member using (i) modified Goodman's equation and (ii) Soderberg equation. [10M]

4 a) Explain with sketches the different types of failures and efficiencies of the riveted joints. [6M]
b) Two MS tie bars for a bridge structure are to be joined by means of a butt joint with double straps. The thickness of the tie bar is 12 mm and carries a load of 400 kN. Design the joint completely taking allowable stresses as 100 MPa in tension, 70 MPa in shear and 150 MPa in compression. [10M]

5 a) Discuss the advantages and disadvantages of riveted, bolted and welded joints. [6M]
b) Design a cotter joint of socket and spigot type which is subjected to a pull and push of 50 kN. All the parts of the joint are made of the same material with the permissible stress as 70 MPa in tension, 100 MPa in compression and 40 MPa in shear. [10M]

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Code No: RT31033 **R13** SET-4

III B. Tech I Semester Regular Examinations November - 2015
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

*****

PART –A

1 a) Enumerate the most commonly used Engineering materials. [3M]
b) What is the significance of preferred numbers? [4M]
c) Describe the stress concentration factor and its significance. [4M]
d) Differentiate the terms bolt, screw and stud. [3M]
e) List advantages of bolted joints over welded joints. [4M]
f) What is nipping in a leaf spring? Discuss its role. [4M]

PART -B

2 a) A shaft is required to transmit 1 MW power at 240 rpm. The shaft must not twist more than 1^{o} on a length of 15 diameters. If the modulus of rigidity for material of the shaft is 80 GPa, find the diameter of the shaft and shear stress induced. [6M]
b) A bolt is subjected to a direct tensile load of 20 kN and a shear load of 15 kN. Suggest the suitable size of bolt according to various theories of elastic feature, if the yield stress in simple tension is 360 MPa. A factor of safety of 3.5 should be used. Take Poisson's ratio as 0.25. [10M]

3 a) Explain the factors that affect the fatigue strength. [6M]
b) A machine member is made of plain carbon steel of ultimate strength 650 N/mm^2 and endurance limit of 300 N/mm^2. If the member is subjected to a fluctuating torsional moment which varies from -200 N-m to 400 N-m. Design the member using (i) modified Goodman's equation and (ii) Soderberg equation. [10M]

4 a) Explain briefly design procedure for circumference lap joint for a boiler. [6M]
b) Design a triple riveted longitudinal butt joint with unequal cover plates for a boiler seam. The diameter of the boiler is 2 m and the internal pressure is 2 MPa. The working stresses are 70 MPa in tension, 50 MPa in shear and 120 MPa in compression and the required efficiency of the joint is 80%. [10M]

5 Two tie rods are to be connected by means of a sleeve and two steel cotters. The rods are subjected to a tensile load of 40kN. Design the joint using the permissible stress in tension as 60MPa, in shear as 50MPa and in crushing as 120MPa. Draw a neat sketch and show all the dimensions. [16M]

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Code No: RT31033 **R13** SET-1

III B. Tech I Semester Supplementary Examinations, May - 2016
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)
*****

PART –A

1 a) Define any four theories of failure. [4M]
b) Draw S-N Curve and mark all salient points. [4M]
c) How do you obtain a bolt of uniform strength? [4M]
d) Write notes on Types of keys. [3M]
e) Draw split coupling. [4M]
f) What are the functions of springs? [3M]

PART -B

2 a) What are the general considerations in the design of machine elements? [4M]
b) A cast iron pulley transmits 10 KW at 400 rpm. The diameter of the pulley is 1.2meter and it has four straight arms of elliptical cross section. In which the major axis is twice the minor axis. Determine the dimensions of the arm if the allowable bending stress is 15 MPa. [8M]
c) Explain simple stresses. [4M]

3 a) Explain Goodman failure theory. [3M]
b) A circular bar of 0.5 m length is supported freely at its two ends. It is actedupon by a central concentrated cyclic load having a minimum value of 20 kN and a maximum value of 50 kN. Determine the diameter of bar by taking a factor of safety of 1.5, size factor of 0.85, surface finish factor of 0.9. The material properties of bar is given by: Ultimate strength of 650 MPa, Yield strength of 500 MPa and Endurance strength of 350 MPa. [8M]
c) Draw S-N curve for mild steel and explain its significance. [5M]

4 a) How the strength of transverse fillet weld is evaluated? [4M]

Code No: RT31033 **R13** SET-1

III B. Tech I Semester Supplementary Examinations, May - 2018

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

PART –A

1 a) Write about preferred numbers? [3M]
b) How will you reduce stress concentration in shaft with keyway? [4M]
c) Write the advantages and limitations of bolted joints? [4M]
d) Write the applications of spigot and socket joint? [4M]
e) What is the importance of split muff couplings? [3M]
f) List the classification of springs? [4M]

PART -B

2 a) Explain the manufacturing considerations in design? [8M]
b) How do you understand failure? Explain the various theories of failure? [8M]

3 a) A torsion bar spring has a solid round 20 mm diameter section which blends smoothly at each end with a larger splined section. It is subjected to a completely reversed nominal torsional stress of 210 MN/m^2. Stress concentration is negligible, and the surfaces are machined. Estimate the fatigue life corresponding to each of the following materials : [12M]
i) steel= 250 HB,
ii) Cast iron Su= 350 MN/m^2.
b) Describe the estimation of endurance strength? [4M]

4 a) How is the allowable stress calculated for a riveted joint subjected to alternating type of load? [6M]
b) The end of a receiver, cylindrical in shape is closed by a lap joint using rivets. The maximum pressure in the receiver is 1 MPa. The axial length of the receiver is limited to 2 m while its storing capacity is 2 m^3. Design the suitable lap joint giving a neat sketch. The permissible stresses in shear and crushing of rivets may be taken as 30 MPa and 70 MPa. The permissible tensile stress for the plate material is 80 MPa. [10M]

Code No: RT31033 **R13** SET-1

III B. Tech I Semester Regular/Supplementary Examinations October/November - 2016
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)
*****

PART –A

1 a) What are the general considerations in designing a machine component? [3M]
b) Explain the modified Goodman diagram for torsional shear stresses. [4M]
c) Explain the various ways in which a riveted joint may fail. [4M]
d) Draw any three keys with neat sketches. [3M]
e) Write the design procedure for muff coupling. [4M]
f) Write briefly about the helical torsion springs with a neat sketch. [4M]

PART -B

2 a) Explain briefly the design considerations of welded assemblies. [6M]
b) Explain briefly the various theory of failures. [10M]

3 a) Explain the modified Goodman diagram for bending stresses. [6M]
b) A transmission shaft of cold drawn steel 27Mn2 (S_{ut} = 500 N/mm^2 and S_{yt} = 30N/mm^2) is subjected to a fluctuating torque which varies from -100 N-m to +400 N-m. The factor of safety is 2 and the expected reliability is 0%. Neglecting the effect of stress concentration, determine the diameter of the shaft. Assume the distortion energy theory of failure. [10M]

4 a) Explain the design procedure for the eccentrically loaded bolted joint. [6M]
b) Design a double riveted butt joint with two cover plates for the longitudinal seam of a boiler shell 1.5 m in diameter subjected to a steam pressure of 0.95 N/mm^2. Assume joint efficiency as 75%, allowable tensile stress in the plate 90MPa, compressive stress 140 MPa and shear stress in the rivet is 56 MPa. [10M]

5 a) A shaft, 40 mm in diameter is transmitting 35 KW power at 300 rpm by means of Kennedy keys of 10X10 mm cross section. The keys are made of steel 45C8 (S_{yt} = S_{yc} = 380 N/mm^2) and the factor of safety is 3.Determine the required length of the keys. [6M]
b) Design a sleeve and cotter joint to resist a tensile load of 60 KN. All parts of the joint are made of the same material with the following allowable stresses. σ_t = 60 MPa , τ = 70 MPa and σ_c =125 MPa. [10M]

III B. Tech I Semester Regular/Supplementary Examinations October/November - 2016
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)
*****

PART –A

1 a) What is Standardization? Give examples of Indian Standards for engineering materials. [3M]
b) Explain briefly about the causes of stress concentration. [4M]
c) Discuss about the bolts of uniform strength. [4M]
d) Explain briefly about the design of flat and square keys. [3M]
e) What are the differences between Rigid couplings and Flexible couplings? [4M]
f) Explain briefly about the stresses and deflection in Coaxial springs. [4M]

PART -B

2 a) What are the manufacturing considerations in the design of Castings? [6M]
b) A manufacturer is interested to start his business with five different models of tractors ranging from 7.5 to 75 KW capacities. Specify power capacities of models. There is an expansion plan to further increase the number of models from five to nine to fulfill the requirements of the farmers. Specify the power capacities of additional models. [10M]

3 a) Explain briefly about Soderberg and Goodman lines with neat sketches. [6M]
b) A circular bar of 500 mm length is supported freely at its two ends. It is acted upon by a central concentrated cyclic load having a minimum value of 20 KN and a maximum value of 50 KN. Determine the diameter of bar by taking a factor of safety of 1.5, size effect of 0.85, surface finish factor of 0.9.The material properties of bar are given by : Ultimate strength of 650 MPa, yield strength of 500 MPa and endurance strength of 350 MPa. [10M]

4 a) Explain the design procedure for the socket and spigot joint. [8M]
b) A circular steel bar 50 mm diameter and 200 mm long is welded perpendicularly to a steel plate to form a cantilever to be loaded with 5KN at the free end. Determine the size of the weld, assuming the allowable stress in the weld is 100 MPa. [8M]

5 a) Explain briefly about the design of shafts subjected to combined bending and torsion. [6M]

Code No: RT31033 **R13** SET-3

III B. Tech I Semester Regular/Supplementary Examinations October/November - 2016
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

*****

PART –A

1 a) Explain about the different types of fits. [3M]
b) Explain about static strength design based on fracture toughness. [4M]
c) Draw the S-N curve for ferrous and nonferrous components. [4M]
d) Discuss briefly about the bolts of uniform strength. [3M]
e) Explain briefly about the design of flat and square keys. [4M]
f) Explain briefly about the stresses and deflection in helical compression springs. [4M]

PART -B

2 a) Explain briefly about the preferred numbers. [6M]
b) A cantilever cold drawn steel bar 20 mm diameter and 100 mm length is loaded by a transverse force of 0.55 kN, an axial load of 8 kN and a torque of 30 Nm. The yield tensile and compressive strength are 165 MPa and 190MPa.Compute factor of safety based on Maximum shear stress theory and Maximum distortion energy theory. [10M]

3 a) Explain briefly the design of welded joints subjected to twisting moment and the bending moment. [6M]
b) A circular shaft, 75 mm in diameter, is welded to the support by means of a circumferential fillet weld. It is subjected to a torsional moment of 3000 N-m. Determine the size of the weld, if the maximum shear stress in the weld is not to exceed 70 N/mm^2. [10M]

4 a) Design a Knuckle joint to transmit 150 kN. The design stresses may be taken as 75 MPa in tension, 60 MPa in shear and 150 MPa in compression. [10M]
b) Explain briefly a design of shafts subjected to combined bending and torsion. [6M]

5 a) Draw the Gerber curve, Goodman and Soderberg lines with neat sketchand explain its significance. [6M]
b) A solid circular shaft made of steel Fe620 (S_{ut} = 620 N/mm^2 and S_{yt} = 380 N/mm^2) is subjected to an alternating torsional moment, that varies from -200N-m to +400 N-m. The shaft is ground and the expected reliability is 90%. Neglecting the stress concentration, Calculate the shaft diameter for infinite life. The factor of safety is 2.Use the distortion energy theory of failure. [10M]

Code No: RT31033 **R13** SET-4

III B. Tech I Semester Regular/Supplementary Examinations October/November - 2016
DESIGN OF MACHINE MEMBERS – I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

*****

PART –A

1 a) Discuss the factors influencing the selection of materials for machine elements. [3M]
b) Define the term stress concentration and What are the causes of stress concentration? [4M]
c) List out the various failures of the riveted joint and how do you classify the riveted joints? [4M]
d) How do you design the solid and hollow shafts based on strength and rigidity? [3M]
e) What are the requirements of a good Coupling? [4M]
f) How do you design the helical compression springs for fatigue loading? [4M]

PART -B

2 a) Explain briefly about the torsional and bending stresses in the design of machine elements. [4M]
b) A cylindrical shaft made of steel of yield strength 700 MPa is subjected to static loads consisting of bending moment 10kN-m and a torsional moment of 30kN-m. Determine the diameter of shaft using all theories of failure and assuming a factor of safety of 2. Take E = 210 GPa and Poisson's ratio = 0.25. [12M]

3 a) Estimate the factors that affect the fatigue strength. [6M]
b) A simply supported beam has a point load at the centre which fluctuates from a value F to 4F. Length of beam is 500 mm and cross section is circular with a diameter of 60 mm. Ultimate, yield stresses are 700 MPa and 500 MPa respectively. Endurance limit in reverse bending is 330 MPa. Factor of safety desired is 1.3.Assume size factor 0.83, Surface finish factor 0.9, reliability factor 1.0.Find the maximum value of F. [10M]

4 a) What are the advantages and disadvantages of welded joints? [6M]
b) A 65 mm diameter solid shaft is to be welded to a flat plate by a fillet weld around the circumference of the shaft. Determine the size of the weld if the torque on the shaft is 3 kNm and the allowable shear stress in the weld is 70 MPa. [10M]

5 a) Write the design procedure for Jib and Cotter joint for square rods. [6M]
b) A mild steel shaft transmits 20 KW at 200 rpm. It carries a central load of 900 N and is simply supported between the bearings 2.5 m apart. Determine the size of the shaft, if the allowable shear stress is 42 MPa and the maximum tensile or compressive stress is not to exceed 56 MPa. What size of the shaft will be required, if it is subjected to gradually applied loads? [10M]

Code No: RT31033 **R13** SET-1

III B. Tech I Semester Regular/Supplementary Examinations, October/November - 2017

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

PART –A

1 a) Discuss in detail the factors which govern the selection of material for a machine component? [3M]

b) Explain the salient features of the maximum principal stress theory and indicate under what conditions such a theory is useful? [4M]

c) Explain the effect of the following factors on the type of fatigue failure [4M]
i) Range of imposed stress
ii) Surface treatment

d) Describe the purpose of gib in cotter joint? What are the applications of cotter joints? [4M]

e) Write a short note on universal coupling? [3M]

f) Define a Spring? What is the purpose of mechanical springs? [4M]

PART -B

2 a) A steel shaft 35 mm in diameter and 1.2 m long held rigidly at one end has a hand wheel 500 mm in diameter keyed to the other end. The modulus of rigidity of steel is 80 GPa [4M]
i) What load applied to tangent to the rim of the wheel produce a torsional shear of 60MPa?
ii) How many degrees will the wheel turn when this load is applied?

b) Derive a relation for the shear stress developed in a shaft, when it is subjected to torsion. [12M]

3 a) Explain the influence of stress concentration in the design of machine elements? What are the principal causes of stress concentration? Explain with suitable sketches? [8M]

b) Explain the significance of Goodman's line, Soderberg line and modified Goodman line in design of members subjected to reversal of stresses? [8M]

4 A bracket is riveted to a column by 6 rivets (A,B,C,D,E and F) of equal size as shown in Figure 3. The centres of rivets A,B,C are on the same vertical line and the centres of E,F are on the another vertical line. The centres of B, D are on the same horizontal line. The centres of A, E are on one horizontal line and the centres of C,F are on another horizontal line. The vertical distance between A,B and B,C are 75 mm and 75 mm respectively. The horizontal distance between B,D and C,F are 75 mm and 150 mm respectively. It carries a load of 100 KN at a horizontal distance of 250 mm from the central line of rivet D. If the maximum shear stress in the rivet is limited to 63 MPa, find the diameter of the rivet. [16M]

Code No: RT31033 **R13** SET-2

III B. Tech I Semester Regular/Supplementary Examinations, October/November - 2017

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

PART –A

1 a) Write a brief note on different phases of design. [3M]

b) Discuss various general considerations that are taken into account while designing a machine element. [4M]

c) Explain about the Maximum Normal Stress, Maximum Shear Stress and Maximum Distortion Energy Theories. [4M]

d) Explain the following methods of reducing stress concentration. [4M]
i) Removal of undesired material
ii) Added grooves

e) Discuss the effect of keys and key ways on the strength of the shaft. [3M]

f) Write a short note on universal coupling. [4M]

PART -B

2 a) Discuss the stress and stain relation. Draw a neat sketch of stress-strain diagram and explain various stress points. [8M]

b) Explain the influence of stress raiser on impact strength and Explain the term 'factor of safety'? [8M]

3 A cold drawn steel rod of circular cross-section is subjected to a variable bending moment of 565 N-m to 1130 N-m as the axial load varies from 4500 N to 13500N. The maximum bending moment occurs at the same instant that the axial load is maximum. Determine the required diameter of the rod for a factor of safety 2. Neglect any stress concentration and column effect. Assuming ultimate strength =550 Mpa, yield strength =470Mpa, size factor=0.85, surface finish factor = 0.89. Correlation factors = 0.1 for bending and 0.7 for axial load. The endurance limit is reversed bending may be taken as one half the ultimate strength. [16M]

4 a) Explain the method of determining the size of the bolt when the bracket carries an eccentric load perpendicular to the axis of the bolt. [6M]

b) The cylinder head of a steam engine is subjected to a steam pressure of $0.7 N/mm^2$. It is held in position by means of 12 bolts. A soft copper gasket is used to make the joint leak-proof. The effective diameter of cylinder is 300mm. Find the size of the bolt so that the stress in the bolts is not to exceed 100 MPa. [10M]

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Code No: RT31033 **R13** SET-3

III B. Tech I Semester Regular/Supplementary Examinations, October/November - 2017

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

PART –A

1 a) How do you classify materials for engineering use? [3M]
b) Write a note on important non-metallic materials of construction in engineering practice? [4M]
c) State and explain any two theories of failure [4M]
d) What are the principal causes of stress concentration? [4M]
e) Enumerate the different types of riveted joints. [3M]
f) What is a coupling? Classify shaft couplings? [4M]

PART -B

2 A pulley is keyed to a shaft midway between two anti-friction bearings. The bending moment of the pulley varies from 150 N-m to 450 N-m as torsional moment of the shaft varies from 50 N-m to 150 N-m. The frequency of variation of the loads is the same as the shaft speed. The shaft is made of cold drawn steel having an ultimate strength of 550 MPa and yield strength of 310 MPa. Determine the required diameter for an indefinite life. The stress concentration factor for the key way in bending and torsion may be taken as 1.6 and 1.3 respectively. Use a design factor of 1.8, size factor 0.85 and surface correction factor 0.88. Use the data for torsion, [16M]
Size correction factor = 0.6 and
The nominal design torsion stress = 0.6 Yield point in tension.

3 A steel rod is subjected to a reversed axial load of 180 kN. Find the diameter of the rod for a factor of safety of 2. Neglect column action. The material has an ultimate tensile strength of 1070 Mpa and yield strength of 910 Mpa. The endurance limit is reversed bending may be assumed to be one half of the ultimate tensile strength. The correction factors are as follows. Load factor = 0.7; surface finish factor = 0.8 [16M]
Size factor = 0.85; stress concentration factor = 1.

4 A bracket is supported by means of four rivets of same size as shown in Fig. [16M]
Determine the diameter of the rivet if the maximum shear stress is 140Mpa.
(16M)

Code No: RT31033 **R13** SET - 4

III B. Tech I Semester Regular/Supplementary Examinations, October/November - 2017

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

PART –A

1 a) What are the steps to be followed while designing a machine element? [3M]

b) Define the following properties of a material: [4M]
i) Ductility
ii) Toughness
iii) Hardness and
iv) Creep.

c) What is factor of safety? List the important factors that influence the magnitude of factor of safety? [4M]

d) Explain the following methods of reducing stress concentration [4M]
i) Drilled holes
ii) Using large fillet radius
iii) Added grooves

e) What do you understand by the term riveted joint? Explain the necessity of such a joint. [3M]

f) Write short notes on leaf springs? [4M]

PART-B

2 a) What are alloy steels? Discuss the effect of adding different alloying elements in steel? [4M]

b) A rotating shaft carries a 18 KN pulley at the center of a 0.75 m simply Supported span. The average torque is 230 N m. Assume the torque range to be 10 % of the average torque. The material has yield point of 770 MPa and the endurance limit of 450MPa. Determine the required diameter of the shaft based on [12M]
i) Maximum stress theory and
ii) Distortion energy theory.
Stress concentration factor may be taken as 1.5 and a factor of safety 2.

3 a) A 50mm diameter steel shaft is supported on bearings 1.5m apart and carries a fly wheel weighing 'W'. The allowable bending stress for the shaft material and the maximum deflection are limited to 100MPa and 2 mm respectively. The young's modulus for the shaft material is 210GPa. Determine the Maximum permissible weight of the flywheel. [6M]

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Code No: RT31033 **R13** SET-1

III B. Tech I Semester Supplementary Examinations, October/November - 2018

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

PART –A

1	a)	Write about types of fits?	[3M]
	b)	Differentiate the theoretical stress concentration factor and fatigue stress concentration factor.	[4M]
	c)	What do you mean by efficiency of riveted joint?	[3M]
	d)	Write the applications of sleeve and cotter joint?	[4M]
	e)	What is the importance of muff couplings?	[4M]
	f)	Write the applications of helical torsion springs?	[4M]

PART -B

2	a)	Explain the design considerations for the selection of Engineering Materials and their properties?	[8M]
	b)	Explain the concept of stiffness in tension, bending, torsion and combined situations?	[8M]
3	a)	Describe the modified Goodman's line theory for designing the components subjected to fatigue loads?	[6M]
	b)	A thin wall cylindrical pressure vessel of mean diameter of 60 cm is subjected to internal pressure varying from 0 to 40 MPa. Find the required thickness of the pressure vessel based on yield point of 400 MPa, endurance limit of 22 Mpa, and a factor of safety of 3. Use Soderberg criterion of failure.	[10M]
4	a)	What forms of rivet heads are used in boiler construction?	[4M]
	b)	A triple riveted lap joint is to be made between 6 mm plates. If the safe working stresses are ft= 84 MPa, fs= 60 MPa and fc= 120 MPa, calculate the rivet diameter, rivet pitch and distance between rows of rivets for the joint. Zig-zag riveting is to be used. State how the joint will fail.	[12M]
5	a)	A machinery shaft is subjected to torsion only. The bearings are 2.50 metre apart. The shaft transmits 190 kW at 220 rev/min. Allow a shear stress of 45 MPa after an allowance for keyways. i) Calculate the shaft diameter for steady loading and ii) Calculate the shaft diameter if the load is suddenly applied with minor shocks.	[12M]
	b)	Write the stresses in keys?	[4M]

Code No: R1622034 **R16** SET-1

II B. Tech II Semester Supplementary Examinations, November - 2019
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

1. a) Define principal stresses and principal planes? (2M)
   b) Define fatigue and endurance limit? (2M)
   c) Explain about methods of riveting? (3M)
   d) What are the elements of a welding symbol? (2M)
   e) Explain about general procedure in machine design? (3M)
   f) Define the terms used in compression springs? (2M)

PART -B

2. a) State and explain theories of failures under static load? (7M)
   b) The load on a bolt consists of an axial pull of 10kN together with a transverse shear force of 5kN. Find the diameter of bolt required according to (7M)
   i). Maximum principal stress theory; ii). Maximum shear stress theory; iii). Maximum principal strain theory; iv). Maximum strain energy theory; and v). Maximum distortion energy theory.
   Take permissible tensile stress at elastic limit = 100MPa and Poisson's ratio = 0.3.

3. a) Write short notes on the influence of various factors of the endurance limit of a ductile material? (7M)
   b) A 50 mm diameter shaft is made from carbon steel having ultimate tensile strength of 630MPa. It is subjected to a torque which fluctuates between 2000 N-m to – 800 Nm. Using Soderberg method, calculate the factor of safety. Assume suitable values for any other data needed. (7M)

4. a) What is an eccentric loaded welded joint? Discuss the procedure for designing such a joint. (7M)
   b) Obtain an expression for total load on a bolt in a bolted joint with gasket. (7M)

5. a) Design a knuckle joint to transmit 150kN. The design stresses may be taken as 75MPa in tension, 60MPa in shear and 150MPa in compression. (7M)
   b) A mild steel shaft transmits 20 kW at 200r.p.m. It carries a central load of 900N and is simply supported between the bearings 2.5metres apart. Determine the size of the shaft, if the allowable shear stress is 42MPa and the maximum tensile or compressive stress is not to exceed 56MPa. What size of the shaft will be required, if it is subjected to gradually applied loads? (7M)

Code No: R1622034 **R16** SET-1

II B. Tech II Semester Regular Examinations, April - 2018
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

1. a) Define factor of safety? (2M)
   b) Define endurance limit? (2M)
   c) What do you understand by the terms riveted joint and welded joints? (3M)
   d) List out the various types of stresses induced in shafts (3M)
   e) Write the applications of rigid couplings? (2M)
   f) Write about co-axial springs? (2M)

PART -B

2. a) Explain about manufacturing considerations in design, tolerances and fits? (7M)
   b) Describe various theories of failure? (7M)

3. a) Write about the design for fluctuating stresses? (5M)
   b) An automobile leaf spring is subjected to cyclic stress such that the average stress is 150Mpa, variable stress is 350Mpa; the material properties are; ultimate strength = 400Mpa; yield strength = 350Mpa; endurance limit = 270Mpa; estimate the factor of safety using Goodman method and Soderberg method? (9M)

4. a) A double riveted lap joint is made between 15-mm thick plates. The rivet diameter and pitch are 25 mm and 75 mm respectively. If the ultimate stresses are 400 MPa in tension, 320 MPa in shear and 640 MPa in crushing, find the minimum force per pitch which will rupture the joint. If the above joint is subjected to a load such that the factor of safety is 2, find out the actual stresses developed in the plates and the rivets. (10M)
   b) Enumerate the list of failures of riveted joints. (4M)

5. a) Design a cotter joint to withstand an axial load varying from 48kN in tension to 48kN in compression. The allowable for the steel used in the joint are 60Mpa in tension; 75Mpa in crushing; 48Mpa in shear (7M)
   b) A 350mm diameter solid shaft is used to drive the propeller of a marine vessel. It is necessary to reduce the weight of the shaft by 80%. What would be the dimensions of a hollow shaft made of the same material as the solid shaft? (7M)

Code No: R1622034 **R16** SET-2

II B. Tech II Semester Regular Examinations, April - 2018
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

1. a) Write the BIS codes of steels in brief? (3M)
   b) Define notch sensitivity? (2M)
   c) Write the necessity of riveted and bolted joint? (2M)
   d) What are the applications of socket and spigot joints? (3M)
   e) Write the applications of flexible couplings? (2M)
   f) Write the energy storage capacity of springs? (2M)

PART -B

2. a) Explain the general considerations in the design of engineering materials and their properties? (7M)
   b) A shaft is designed based on maximum distortion energy theory with a factor of safety of 2.0. The material used is 30C8 steel with a yield stress of 310MPa. It is subjected to an axial load of 40kN. Determine the maximum torque capacity. Diameter of the shaft is 20 mm. (7M)

3. a) Explain Goodman's method to calculate the safe values of fluctuating stress. For what materials it is applicable? (7M)
   b) A simply supported beam has a concentrated load at the center, which fluctuates from a value of P to 4 P. The span of the beam is 0.5 m and its cross-section is circular with a diameter of 0.06 m. Taking for the beam material an ultimate stress of 700 MPa, a yield stress of 500 MPa, endurance limit of 330 MPa for reversed bending, and a factor of safety of 1.3, calculate the maximum value of P. Take a size factor of 0.85 and a surface finish factor of 0.9. (7M)

4. a) A triple riveted lap joint with zig-zag riveting is to be designed to connect two plates of 6 mm thickness. Determine the diameter of the rivet, pitch of rivets and distance between the rows of the rivets. Indicate how the joint will fail. Also, find the efficiency of the joint. The permissible stresses are 120MPa in tension, 100MPa in shear and 150MPa in crushing. (10M)
   b) What are the advantages and limitations of welding over riveting? (4M)

Code No: R1622034 **R16** SET-3

II B. Tech II Semester Regular Examinations, April - 2018

DESIGN OF MACHINE MEMBERS-I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**

2. Answer **ALL** the question in **Part-A**

3. Answer any **FOUR** Questions from **Part-B**

PART –A

1. a) Write about types of tolerances? (2M)
   b) Define endurance strength? (2M)
   c) Write the applications of welded joints? (3M)
   d) Write the stresses in cotter joint? (3M)
   e) Write the purpose of flange couplings? (2M)
   f) Write the basic phenomenon of surge in springs. (2M)

PART -B

2. A machine element is loaded so that $\sigma 1$ = 120 Mpa, σ_2 = 70Mpa, σ_3 = −90Mpa the material has a maximum yield strength in tension and compression of 360Mpa. Find the factor of safety for each of the following theories. i) Maximum Normal stress theory ii) Maximum Shear stress theory iii) Distribution energy theory. (14M)

3. a) Determine the size of a piston rod subjected to a total load of having cyclic fluctuations from 150 kN in compression to 25 kN in tension. The endurance limit is 360 MPa and yield strength is 400 MPa. Take impact factor = 1.25, factor of safety = 1.5, surface finish factor = 0.88 and stress concentration factor = 2.25. (10M)
   b) Explain the methods to reduce stress concentrations? (4M)

4. A double riveted, double strap bolt joint is used to join 30mm thick plates. (14M) The pitch of the rivets in the outer row is twice that of the inner row. Zigzag riveting is to be employed with the following working stresses; τ = 63Mpa and σ_t = 84Mpa. Calculate rivet diameter, rivet pitches in the inner and outer rows and the thickness of the butt straps.

5. a) A 10kW power is transmitted at 800 rpm, from a motor shaft, through a key, to a machine shaft by a means of a pulley and a belt. Design the key. Take the allowable shear stress and crushing stress are 45MPa and 100Mpa. (6M)
   b) A shaft and a key are made of the same material and the key width is 1/4 of the shaft diameter. Consider shear only, determine the minimum length of the key in terms of the shaft diameter. The shearing strength of the key material is 60% of its crushing strength. Determine the thickness of the key to make the key equally strong in shear and crushing. (8M)

Code No: R1622034 **R16** SET-4

II B. Tech II Semester Regular Examinations, April - 2018
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

1\. a) Write about types of fits? (3M)
b) Define theoretical stress concentration factor? (2M)
c) Write the applications of bolted joint? (2M)
d) Write the stresses in keys? (3M)
e) Write the purpose of muff couplings? (2M)
f) How does surge in springs eliminated? (2M)

PART -B

2\. A cylindrical shaft made of steel yield strength 800Mpa is subjected to static loads bending moment 20kN-m and twisting moment 30N-m. Calculate the diameter of the shaft using Normal stress theory and Von Mises theory. Assume factor of safety is 2. (14M)

3\. a) Explain the effect of the following factors on the type of fatigue failure. (4M)
i) Type of material
ii) Stress distribution

b) A shaft made of steel having ultimate tensile strength of 700 MPa and yield point 420 MPa is subjected to a torque of 2000 N- m clockwise to 600 N- m anti-clockwise. Calculate the diameter of the shaft if the factor of safety is 2 and it is based on the yield point and the endurance strength in shear (10M)

4\. a) Two plates 16 mm thick are joined by a double riveted lap joint. The pitch of each row of rivets is 90 mm. The rivets are 25 mm in diameter. The permissible stresses are 140 MPa in tension, 80 MPa in shear and 160 MPa in crushing. Find the efficiency of the joint. (10M)
b) Enumerate the different types of riveted joints. (4M)

5\. Design a sleeve and cotter joint to resist a tensile load of 70kN. All parts of the joints are made of the same material with the following allowable stresses: σ_t = 60Mpa ; τ = 70Mpa ; σ_c = 125Mpa (14M)

Code No: R1622034 **R16** SET-1

II B. Tech II Semester Regular/ Supplementary Examinations, April/May - 2019
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

1. a) List out at least 6-important mechanical properties of engineering materials. (3M)
   b) Define the term endurance limit? Show diagrammatically. (2M)
   c) Identify different rivets configurations or systems? (2M)
   d) What is a Gib? What is the uniqueness of Gib headed cotter joint. (3M)
   e) What is a shaft coupling? What are its principal functions? (2M)
   f) Write the statement of leaf spring of uniform strength. (2M)

PART -B

2. a) Select a suitable materials for the manufacture of the following:
   i) Machine tool spindle ii) Valve spring
   iii) Condenser tubes iv) Connecting rod (4M)
   b) Define the term 'factor of safety'. Explain the influence of stress raiser on impact strength. (10M)
3. a) Draw the stress strain diagrams of
   i) Ductile materials and
   ii) Brittle materials (4M)
   b) A shaft is made of steel, ultimate tensile strength 700 MPa and yield point 420 MPa is subjected to a torque varying from 200N m anti-clockwise to 600 N m clockwise. Calculate the diameter of the shaft if the factor of safety is 2 and it is based on the yield point and the endurance strength in shear. (10M)
4. a) What do you understand by the term riveted joint? Explain the necessity of such a joint. (4M)
   b) The head of an air compressor cylinder is attached by eight bolts. The cylinder bore diameter is 80 mm and the maximum working pressure is limited to 3 MPa. Determine the diameter of bolt, when a copper-asbestos is used between the head and the cylinder and an initial compressive load of 5 KN is required on the gasket for a leak proof joint. Ultimate tensile strength, Yield strength and endurance limit for material are 400 MPa, 340 MPa and 200 MPa respectively. A factor of safety of three is desired and the stress-concentration factor of 2.84 may be assumed. (10M)
5. a) Discuss the effect of keys on the strength of the shaft. (4M)
   b) Design a socket and spigot type of cotter joint to sustain an axial load of 100kN. The material selected for the joint has the following design stresses. σ_t = 120 MPa, σ_c = 160 MPa and σ_{pb} = 60 MPa (10M)

Code No: R1622034 **R16** SET-2

II B. Tech II Semester Regular/ Supplementary Examinations, April/May - 2019
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B**)
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

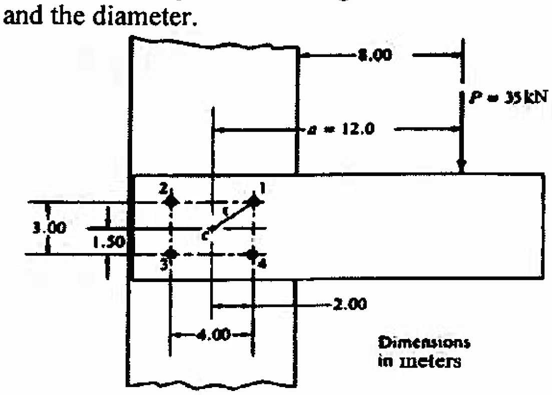
1. a) What is the relation between tolerances and fits? (3M)
b) Define the term fatigue? (2M)
c) Classify various welded joints? (2M)
d) Distinguish the Keyed and Cotter joins? (3M)
e) Classify the shaft couplings. (2M)
f) Distinguish helical and torsional springs. (2M)

PART -B

2. a) What are alloy steels? Discuss the effect of adding different alloying elements in steel? (4M)
b) A weight of 1400 N is dropped on to a collar at the lower end of a vertical steel shaft of 3m long and 25 mm. diameter, calculate the height of drop if the maximum instantaneous stress produced is not to exceed 120 MPa. Take E = 2.1×10^5 MPa. (10M)

3. a) Explain the effect of the following factors on the type of fatigue failure. (4M)
i) Strain rate ii) Type of material iii) Manner of loading
b) A steel connecting rod is subjected to a completely reversed axial load of 1,600 MPa. Suggest the suitable diameter of the rod using a factor of safety 2. The ultimate tensile strength of the material is 1,100 MPa and yield strength 930 MPa. Neglect column action and the effect of stress concentration. (10M)

4. a) What do you understand by the term riveted joint? Explain the necessity of such a joint. (4M)
b) For the bracket in Figure assume that the total force P is 35kN and the distance a is 12 m. Design the bolted joint, including the location and number of bolts, the material, and the diameter. (10M)



Code No: R1622034 **R16** SET-3

II B. Tech II Semester Regular/ Supplementary Examinations, April/May - 2019
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

1. a) List standard fundamental equations for Axial, bending and torsional loading. (3M)
   b) Define the term notch sensitivity? What is fatigue stress concentration factor and notch sensitivity? (3M)
   c) List different types of riveted joints. (2M)
   d) How do you classify keys for mechanical systems? (2M)
   e) How do you define rigid shaft couplings? Name them. (2M)
   f) What is energy storage capacity of a spring? (2M)

PART -B

2. a) State the alloying elements added to steel and the effects they produce in steel. (4M)
   b) A copper bar 50 mm in diameter is placed with a steel tube 75 mm external diameter and 50 mm internal diameter of exactly the same length. The two pieces are rigidly fixed together by two pins 18 mm in diameter, one at each end passing through the bar and tube. Calculate the stress induced in the copper bar, steel tube and pins if the temperature of the combination is raised by 50%. Take Es = $210 GN/m^2$; Ec = $105 GN/m^2$ $\alpha s = 11.5 \times 10{-}6/^oC$ and $\alpha c = 17 \times 10{-}6/^oC$. (10M)

3. a) Explain the Soderberg method for combination of stresses. (4M)
   b) A steel rod is subjected to a reversed axial load of 180 kN. Find the diameter of the rod for a factor of safty of 2. Neglect column action. The material has an ultimate tensile strength of 1070 Mpa and yield strength of 910 Mpa. The endurance limit is reversed bending may be assumed to be one half of the ultimate tensile strength. The correction factors are as follows. (10M)
   Load factor = 0.7; Surface finish factor = 0.8
   Size factor = 0.85; Stress concentration factor = 1.

4. a) Explain the method of determining the size of the bolt when the bracket carries an eccentric load parallel to the axis of the bolt. (4M)
   b) Determine the safe tensile load for bolts of M 20 and M 36. Assume that the bolts are not initially stressed and take the safe tensile stress as 200 MPa. (10M)

5. a) Design a cotter joint to withstand an axial load varying from 45kN in tension to 45 kN in compression. The allowable for the steel used in the joint are 60Mpa in tension; 70 Mpa in crushing; 45 Mpa in shear. (4M)
   b) A shaft running at 400rpm transmits 10kW. Assuming allowable shear stress in shaft is 40Mpa, find the diameter of the shaft. (10M)

|''|''||''||''''|||||

Code No: R1622034 **R16** SET-4

II B. Tech II Semester Regular/ Supplementary Examinations, April/May - 2019
DESIGN OF MACHINE MEMBERS-I
(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answer **ALL** the question in **Part-A**
3. Answer any **FOUR** Questions from **Part-B**

PART –A

1. a) Write the statement of Rankine's theory of failure. (3M)
   b) Define the term stress concentration and draw some critical regions to stress concentration. (3M)
   c) What are locking devices in bolted joints? (2M)
   d) What is a Key? Mention various functions of the keys. (2M)
   e) How do you define flexible shaft couplings? Name them. (2M)
   f) Write the formula for deflection of a helical spring. (2M)

PART -B

2. a) Enumerate the most commonly used engineering materials? Briefly explain shear stress and shear strain? (4M)
   b) A steel saw blade 1 mm thick is bent into an arc of a circle of 50 cm radius. Determine the flexural stresses induced and the bending moment required to bend the blade which is 15 mm wide. Take $E = 2.1\text{X}10^5$ N/mm^2 (10M)
3. a) A load on a bolt consists of an axial pull of 10 k N together with a transverse shear force of 5 kN. Find the diameter of bolt required according to (14M)
   i) Maximum principal stress theory
   ii) Maximum shear stress theory
   iii) Max-principal strain theory
   iv) Maximum strain energy theory.
   Take permissible tensile stress at elastic limit=100 Mpa and Poisson's ratio=0.3.
4. a) Discuss on bolts of uniform strength giving examples of practical applications of such bolts. (4M)
   b) A double riveted lap joint is made between 15-mm thick plates. The rivet diameter and pitch are 25 mm and 75 mm respectively. If the ultimate stresses are 400 MPa in tension, 320 MPa in shear and 640 MPa in crushing, find the minimum force per pitch which will rupture the joint. If the above joint is subjected to a load such that the factor of safety is two, find out the actual stresses developed in the plates and the rivets. (10M)
5. a) Derive suitable equations in terms of torque, cross section of key for same shaft and key material. (4M)
   b) A feather key is 12mm wide and is to transmit 700N-m torque from a 400mm diameter shaft. The steel key has an allowable stress in tension and compression of 120Mpa and an allowable shear stress of 575Mpa. Determine the required length of key. If the key dimensions are reversed as 9mm wide and 12mm deep, what would have been the required length of key for same load and material? (10M)

Code No: RT31033 **R13** SET - 1

III B. Tech I Semester Supplementary Examinations, May- 2019

DESIGN OF MACHINE MEMBERS – I

(Mechanical Engineering)

Time: 3 hours Max. Marks: 70

Note: 1. Question Paper consists of two parts (**Part-A** and **Part-B)**
2. Answering the question in **Part-A** is compulsory
3. Answer any **THREE** Questions from **Part-B**
(Data books may be allowed)

PART –A

1	a)	Write a brief note on different phases of design.	[3M]
	b)	Explain the following method of stress concentration i) Drilled holes, ii) Using large fillet radius, iii) Added grooves.	[4M]
	c)	Distinguish the riveted and the bolted joints?	[4M]
	d)	List advantages of bolted joints over welded joints.	[4M]
	e)	Write notes on Types of keys.	[3M]
	f)	Define spring? What is the purpose of mechanical springs?	[4M]

PART -B

2	a)	What are alloy steels? Discuss the effect of adding different alloying elements in steel?	[4M]
	b)	An electric motor weighing 500N is mounted on a short cantilever beam of uniform rectangular cross section. The weight of motor acts at a distance of 300mm from the support. The depth of the section is twice the width. Determine the cross section of the beam. The allowable stress in the beam is 40N/mm^2.	[8M]
	c)	What are the general considerations in the design of machine elements?	[4M]
3	a)	Explain the effect of the following factors on the type of fatigue failure i) Type of material ii) Surface treatment iii) Range of imposed stress	[8M]
	b)	A leaf spring in an automobile is subjected to cyclical stresses. The average stress = 150 MPa, variable stress = 50 MPa, Ultimate stress = 630 MPa, Yield point stress = 350 MPa and endurance limit = 150 MPa. Estimate under what factor of safety the spring is working, by Goodman and Soderberg formulae.	[8M]
4	a)	How the strength of transverse fillet weld is evaluated?	[4M]
	b)	The cylinder head of a steam engine is subjected to a steam pressure of 0.7N/mm^2. It is held in position by means of 12 bolts. A soft copper gas-ket is used to make the joint leak-proof. The effective diameter of cylinder 300mm. Find the size of the bolt so that the stress in the bolts is not to exceed 100 MPa.	[12M]
5	a)	Design a shaft to transmit power from an electric motor to a lathe head stock through a pulley by means of a belt drive. The pulley weighs 200N and is located at 300mm from the centre of the bearing. The diameter of the pulley is 200mm and the maximum power transmitted is 1KW at 120rpm. The angle of lap of the belt is 180° and coefficient of friction betweenthe belt and the pulley is 0.3. The shock and fatigue factors for bending and twisting are 1.5 and 2.0 respectively. The allowable shear stress in the shaft may be taken as 35MPa.	[6M]

Code No: 135AG Set No: 1

NAWAB SHAH ALAM KHAN COLLEGE OF ENGINEERING & TECHNOLOGY

New Malakpet, Hyderabad-500024

III-B.TECH I-SEM MID-I EXAMINATION SEPT - 2019

BRANCH: ME

SUBJECT: DMM-1 TIME: 10:00 A.M TO11.00A.M

I Answer any two ofthefollowing 2x5=10 marks

Q.NO	QUESTIONS	BLOOMS LEVEL
Q1)	a) List out the factors to be considered for selection of materials for the design of machineelement. b) Discuss the failure theories related to ductilematerials.	L3
Q2)	The load on a bolt consist of an axial pull of 10KN together with a transverse shear force of 5KN.Find diameter of bolt required using two different theories of failure.	L4
Q3)	Explain stress concentration and Notch sensitivity.	L3
Q4)	Determine the diameter of a circular rod made of ductile material with fatigue strength (complete stress reversal) σ_e=265MPA and tensile yield strengthofσ_y=350MPA.Thememberissubjectedtoavaryingaxialloadfrom W_{min}=-300KNtoW_{max}=700KNandhasastressconcentrationfactor1.8usefactorof safety as2	L4

10. Notch sensitivity (q) is obtained by thefollowingrelation, [ ]

a) $(k_f +1)/(k_t+1)$ b) $(k_f -1)/(k_t-1)$ c) $(k_t +1)/(k_f+1)$ d) $(k_t -1)/ (k_f-1)$

II Fill in the blanks

11. The maximum energy stored in a body due to external loading up to the elastic limit is called__________________

12. The stress produced in the member due to the falling load is knownas ____________

13. Rankine's theory of failure isapplicablefor_______________type of materials.

14. The yield point in staticloadingis_______________as compared to fatigueloading.

15. The irregularities in the stress distribution caused by abrupt changes of form iscalled ______________

16. At the neutral axis of a beam, the shear stressis ________________

17. for a flat plate with a circular hole is subjected to tensile force, then its theoretical stress concentration factoris ___________

18. Fatigue failure isdefinedas________________ under cyclicloading.

19. Strain energy per unit volumeis ____________

20 .Maximum principal stress theory holdsgoodfor ___________ materials.

Code No: 135AG   Set No: 1

NAWABSHAH ALAMKHAN COLLEGE OF ENGINEERING AND TECHNOLOGY

III B.TECH I SEM., II MID-TERM EXAMINATIONS, SEPTEMBER 2019

Objective Exam

Name:________________________ **Hall TicketNo.** | | | | | | | | | | | |

Answer all questions .All questions carry equal marks. Time: 20 minutes Max.marks:10

I Choose the correctalternative

1. According to unwins formula,the relation between diameter of rivet holeand thickness [ ]

a)d=t   b)d=1.6t½

c) d=2t   d) d=t6t½

2. The strength of solid plate per pitch length is equal to

a)$p*d*\sigma_t$ b) $p*t*\sigma_t$ c) $(p-t)d*\sigma_t$ d) $(p-d)t*\sigma_t$ [ ]

3. The center to center distance between two consecutive rivets in arrow is called [ ]

a) marginb)pitch   c)back pitch   d) diagonal pitch

4. In butt welded joint the size of the weld is equal to [ ]

a) throat of weld   b) 2xthroat of weld

c)throat of weld$^{1/2}$ d)0.5xthroat of weld

5. At what angle the plane of max shear occurs,for parallel load on a fillet weld having [ ]

a) 30   b) 45

c) 22.5   d) 60

6. Which of the following steel key is usually strong in failure by shear and crushing [ ]

a) wood ruff   b) kennedy   c)sunk   d) saddle

7. the type of stress develop in the key is/are

a)shear stress alone   b)bearing stress alone c)both shear and bearing   d)shearing bearing and bending

8. a cotter joint is used to transmit [ ]

a)axial tensile load   b)axial compressive load.

C)combine axial and twisting load d)axial tensile or compressive load

Cont.....2

REFERENCE BOOKS:

1. Design of Machine Elements / V. M. Faires / Macmillan
2. Design of Machine Elements-I / Annaiah, M.H / New Age

TEXT BOOKS:

1. Design of Machine Elements / V. Bhandari / Mc Graw Hill
2. Machine Design / Jindal / Pearson

Student List

S.NO	H.T. NO	NAME OF STUDENT /SUBJECTS
1	17RT1A0301	ABDULMANNAN BAIG
2	17RT1A0307	ARAFAT
3	17RT1A0308	BILAL MOHAMMED ATEEQ
4	17RT1A0311	HAMED BIN TAHER HARHARA
5	17RT1A0312	ISMAIL PASHA
6	17RT1A0319	MIRZA AFROZ BAIG
7	17RT1A0320	MIRZA AMAIR BAIG
8	17RT1A0321	MIRZA FARHAN BAIG
9	17RT1A0326	MOHAMMED ABDUL HADI
10	17RT1A0327	MOHAMMED ABDUL JALEEL
11	17RT1A0328	MD ABDUL RAHMAN ALEEM
12	17RT1A0329	MOHAMMED ADNAN HUSSAIN
13	17RT1A0330	MOHAMMED ABDUL WAJID
14	17RT1A0331	MOHAMMED ABDUL WASAY
15	17RT1A0332	MOHD ABDULLAH GHORI
16	17RT1A0333	MOHD ABIDULLAH ANSARI
17	17RT1A0334	MOHAMMED ABRAR HASSAN
18	17RT1A0336	MOHAMMED ASAD AHMED
19	17RT1A0338	MOHAMMED AZIZUDDIN
20	17RT1A0341	MOHAMMED HYDER AHMED
21	17RT1A0342	MOHAMMED ILIYAAS AKBAR
22	17RT1A0343	MOHAMMED IMRAN
23	17RT1A0346	MOHAMMED JUNAID
24	17RT1A0344	MOHAMMED INZAMAMUDDIN
25	17RT1A0348	MOHAMMED KHADER JILANI
26	17RT1A0351	MOHD MUHEEBUDDIN ASLAM
27	17RT1A0356	MOHAMMEDSHAHBAZ HUSSAIN
28	17RT1A0357	MOHD SHAHER YAR KHAN
29	17RT1A0360	MOHAMMED TAJ
30	17RT1A0361	MOHAMMED VASIUDDIN

31	17RT1A0362	MOHAMMED YASEER
32	17RT1A0366	MOHD ABDUL QAVI
33	17RT1A0367	MOHD ABDUL RAHMAN
34	17RT1A0368	MOHD ABDUL RAHMAN
35	17RT1A0370	MOHD ARBAZ
36	17RT1A0374	MOHD FAISAL HUSSAIN
37	17RT1A0377	M.IMRANUDDIN
38	17RT1A0378	MOHD KHAJA
39	17RT1A0379	MOHD KHALEEL UR RAHEMAN
40	17RT1A0382	MD OMAIR AHMED
41	17RT1A0383	MOHD PARVEZ
42	17RT1A0386	MOHD SULEMAN UDDIN ALI KHAN
43	17RT1A0389	MUSAIB MOHIUDDIN
44	17RT1A03A0	SHAIK SAMI UR RAHMAN
45	17RT1A03A7	SYED FARDEEN ALI
46	17RT1A03B5	SYED MOHD LATEEF
47	17RT1A03B6	SYED NAIYYER HUSSAIN
48	17RT1A03B8	SYED NOOR MOHAMMED
49	18RT5A0301	ADIL MOHAMMED SAIFUL ISLAM
50	18RT5A0306	KAMA NAVEEN
51	18RT5A0307	KHALEEL AHMED
52	18RT5A0309	MD AIJAZ UDDIN
53	18RT5A0310	M A MUNAWAR
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55	18RT5A0312	MOHAMMAD ABDUL RAHMAN
56	18RT5A0314	MOHAMMAD SUFIYAN OUSAF
57	18RT5A0318	MOHAMMED SHOAIB HUSSAIN
58	18RT5A0319	MOHAMMED SHOAIB KHAN
59	18RT5A0323	MOHD ASEEM UDDIN
60	18RT5A0326	MOHD SADIQ
61	18RT5A0328	SHAHEDA MAHREEN
62	17RT1A0381	MOHD NADEEM

63	17RT1A0391	SALAH MOHD SOHAIL
64	17RT1A0393	SHAIK ABDUL OBAID
64	17RT1A0394	SHAIK ABDUL WASI
65	17RT1A0395	SHAIK ASHRAF ALI
66	17RT1A03A2	SK MOHAMMED NAIF UDDIN
67	17RT1A03A5	SYED ALTAF UDDIN
68	17RT1A03A6	SYED ESA GIBRAN
69	17RT1A03A8	SYED HUSSAIN AHMED
70	17RT1A03B4	SYED MOHD NADEEM
71	17RT1A03C0	SYED SUFIYAN MOHAMMED
72	17RT1A03C1	SYED TALIB AZAM
73	18RT5A0302	AHMED ABDUL HAQUE
74	18RT5A0303	CHEGONDI SIVALINGA RAJU
75	18RT5A0304	HABEEB AHMED
76	18RT5A0305	IBRAHIM BIN HASAN MOHAMMADI
77	18RT5A0308	M A WASEEM
78	18RT5A0313	MOHAMMAD FARDEEN FARAZ
79	18RT5A0315	MOHAMMED ABDUL KHALEEL
80	18RT5A0316	MOHAMMED ABDUL RAHMAN
81	18RT5A0317	MOHAMMED IBRAHIM
82	18RT5A0322	MOHD AMIR
83	18RT5A0324	MOHD AZMATH QUADRI
84	18RT5A0325	MOHD MOIZ UDDIN
85	18RT5A0327	MOHD SHOAIB ABBAS
86	18RT5A0329	SHAIK AWAIS
87	18RT5A0330	SHAIK BASHEER
88	18RT5A0331	SHAIK UMAR SHARIEF

HALL TICKET NO	MID-1	SL/FL
17RT1A0307	14	SL
17RT1A0312	19	SL
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17RT1A0301	24	FL
17RT1A0308	21	FL
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18RT5A0323	20	FL
18RT5A0326	20	FL

NAWAB SHAH ALAM KHAN COLLEGE OF ENGINEERING AND TECHNOLOGY, JNTUH Hyderabad
DEPARTMENT OF MECHANICAL ENGINEERING

B.Tech. III YEAR, I SEM - ATTAINMENT CALCULATIONS - Academic Year: 2019-20

Subject: DMM-I Subject Code: C311 Faculty: FAZAL MOHAMMED

S.No.	Hall Ticket No.	MID-1											MID-2											MID		SEE
		ASG-1 (2.5M)	ASG-2 (2.5 M)	Quiz-1 (10 M)		Q 1 (5 M)	Q 2 (5 M)	BEST OF Q1&Q2	Q 3 (5 M)	Q 4 (5 M)	BEST OF Q3&Q4	Mid-1 TOTAL (25 M)	ASG-3 (2.5M)	ASG-4 (2.5 M)	Quiz-2 (20 M)		Q 1 (5 M)	Q 2 (5 M)	BEST OF Q1&Q2	Q 3 (5 M)	Q 4 (5 M)	BEST OF Q3&Q4	Mid-2 TOTAL (25 M)	Average MID (25 M)	TOTAL Marks (100 M)	End Exam (75 M)
		CO1	CO2	CO1	CO2	CO1	CO1	CO1	CO2	CO2	CO2		CO3	CO4	CO3	CO4	CO3	CO3	CO3	CO4	CO4	CO4				
1	17RT1A0301	2.5	2.5	4	5	5		5			5	24	2.5	2.5	5	4	5	5	5	5		5	24	24	59	35
2	17RT1A0307	2.5	2.5	4	4	1		1			0	14	1	1	5	4	5		5	5		5	21	18	39	22
3	17RT1A0308	2.5	2.5	5	4	3		3	4		4	21	1.5	1.5	5	4	5		5	2		2	19	20	39	19
4	17RT1A0311	2.5	2.5	5	4	5		5		4	4	23	1	1	5	4	5		5		5	5	21	22	49	27
5	17RT1A0312	2.5	2.5	5	4		5	5			0	19	1	1	5	5	5		5	5		5	22	21	39	19
6	17RT1A0319	2.5	2.5	3	2	5		5	5		5	20	1	1	5	5	5		5	5		5	22	21	49	28
7	17RT1A0320	2.5	2.5	3	3	5		5	4		4	20	1	1	5	5	5		5	3		3	20	20	49	29
8	17RT1A0321	2.5	2.5	4	3	2		2			0	14	1	1	5	5	5		5	5		5	22	18	39	21
9	17RT1A0326	2.5	2.5	4	4	2		2			0	15	1	1	5	5	5		5	5		5	22	19	39	21
10	17RT1A0327	2.5	2.5	5	4	3		3	4		4	21	1.5	1.5	5	5	5		5	3		3	21	21	49	28
11	17RT1A0328	2.5	2.5	5	4		3	3			0	17	1	1	4	4	4		4			0	14	16	39	24
12	17RT1A0329	2.5	2.5	5	4	0		0			0	14	1.5	1.5	5	4	5		5	2		2	19	17	39	23
13	17RT1A0330	2.5	2.5	4	3	4		4	3		3	19	1	1	5	4	5		5	4		4	20	20	49	30
14	17RT1A0331	2.5	2.5	3	3	3		3			0	14	2.5	2.5	2	1	5		5	1		1	14	14	49	35
15	17RT1A0332	2.5	2.5	3	4	3		3		3	3	18	0	0	5	2	5		5		4	4	16	17	59	42
16	17RT1A0333	2.5	2.5	4	4	4		4	4		4	21	1	1	2		5		5	5		5	14	18	39	22
17	17RT1A0334	2.5	2.5	4	3		3	3	3		3	18	0	0	5	5	5		5	1		1	16	17	39	22
18	17RT1A0336	2	2	5	4	1		1			0	14	0	0	5	4	5		5		3	3	17	16	39	24
19	17RT1A0338	2.5	2.5	5	4	4		4	4		4	22	1.5	1.5	5	5		5	5	3		3	21	22	39	18
20	17RT1A0341	2.5	2.5	3	3	4		4		4	4	19	1	1	5	5	4		4	5		5	21	20	59	39
21	17RT1A0342	2	2	5	4	3		3			0	16	0	0	5	4	5		5	4		4	18	17	39	22
22	17RT1A0343	2.5	2.5	5	4	4		4	2		2	20	1	1	5	4	5		5	5		5	21	21	39	19
23	17RT1A0344	2.5	2.5	2	2	4		4	4		4	17	0	0	5	4	5		5		3	3	17	17	49	32
24	17RT1A0346	2.5	2.5	4	4	5		5		4	4	22	1.5	1.5	5	5	5		5	5		5	23	23	59	37
25	17RT1A0348	2.5	2.5	3	3	3		3			0	14	0	0	5	4	5		5	5		5	19	17	39	23
26	17RT1A0351	2.5	2.5	4	4	4		4	4		4	21	1.5	1.5	5	5	5		5	3		3	21	21	39	18
27	17RT1A0356	2.5	2.5	5	4	4		4		3	3	21	0	0	5	4	4		4	5		5	18	20	39	20
28	17RT1A0357	2.5	2.5	5	4	4		4		3	3	21	1	1	5	5	5		5		5	5	22	22	49	28
29	17RT1A0360	2.5	2.5	5	4	5		5	3		3	22	1.5	1.5	5	5	5		5		5	5	23	23	39	17
30	17RT1A0361	2.5	2.5	4	4	4		4	5		5	22	1	1	5	5	5		5	2		2	19	21	39	19
31	17RT1A0362	2.5	2.5	5	4	4		4		3	3	21	1	1	5	5	5		5	5		5	22	22	39	18
32	17RT1A0366	2.5	2.5	5	4	4		4	2		2	20	0	0	5	5	3		3	4		4	17	19	59	41
33	17RT1A0367	2.5	2.5	4	4	4		4	2		2	19	1	1	5	5	5		5	5		5	22	21	49	29
34	17RT1A0368	2.5	2.5	4	3	4		4	2		2	18	0	0	5	5	5		5	4		4	19	19	39	21
35	17RT1A0370	2.5	2.5	5	4	0		0			0	14	0	0	5	4	4		4	5		5	18	16	39	23
36	17RT1A0374	2.5	2.5	5	4	5		5			0	19	1	1	5	4	5		5	5		5	21	20	39	19
37	17RT1A0377	2	2	3	2	5		5			0	14	1.5	1.5	4	5	5		5		5	5	22	18	49	31
38	17RT1A0378	2.5	2.5	2	2			0	5		5	14	0	0	5	5	5		5	5		5	20	17	39	22
39	17RT1A0379	2.5	2.5	5	4	5		5	4		4	23	2.5	2.5	5	4	5		5	5		5	24	24	69	46
40	17RT1A0381	2.5	2.5	5	5			0			0	15	2.5	2.5	4	3	5		5	2		2	19	17	39	22
41	17RT1A0382	2.5	2.5	5	4			0			0	14	2.5	2.5	4	4	5		5	5		5	23	19	39	21
42	17RT1A0383	2.5	2.5	4	4	5		5			0	18	2.5	2.5	4	3	5		5	4		4	21	20	49	30
43	17RT1A0386	2.5	2.5	4	4	5		5			0	18	2.5	2.5	4	4	5		5	5		5	23	21	49	29
44	17RT1A0389	2.5	2.5	5	5			0	5		5	20	2.5	2.5	5	4	5		5		4	4	23	22	69	48
45	17RT1A0391	2.5	2.5	5	4			0			0	14	2.5	2.5	4	4	5		5	4		4	22	18	39	21
46	17RT1A0393	2.5	2.5	5	5			0			0	15	2.5	2.5	4	3	5		5	1		1	18	17	49	33
47	17RT1A0394	2.5	2.5	5	4			0			0	14	2.5	2.5	3	3	3		3	3		3	17	16	39	24
48	17RT1A0395	2.5	2.5	4	3	4		4			0	16	2.5	2.5	3	4	5		5	1		1	18	17	49	32
49	17RT1A03A0	2.5	2.5	5	4	4		4			0	18	2.5	2.5	4	4	5		5	4		4	22	20	59	39
50	17RT1A03A2	2.5	2.5	5	4			0			0	14	2.5	2.5	4	4	5		5	4		4	22	18	39	21
51	17RT1A03A5	2.5	2.5	5	4			0			0	14	2.5	2.5	4	3		2	2	5		5	19	17	49	33
52	17RT1A03A6	2.5	2.5	5	5		5	5			0	20	2.5	2.5	4	4	5		5	5		5	23	22	39	18
53	17RT1A03A7	2.5	2.5	4	4	1		1			0	14	2.5	2.5	4	3	5		5	5		5	22	18	59	41
54	17RT1A03A8	2.5	2.5	5	4			0			0	14	2.5	2.5	3	4	5		5		3	3	20	17	39	22
55	17RT1A03B4	2.5	2.5	5	5			0			0	15	2.5	2.5	4	3	5		5	2		2	19	17	39	22
56	17RT1A03B5	2.5	2.5	5	4	2		2			0	16	2.5	2.5	4	4	5		5	4		4	22	19	59	40
57	17RT1A03B6	2.5	2.5	5	5	5		5	2		2	22	2.5	2.5	3	4	5		5	4		4	21	22	39	18
58	17RT1A03B8	2.5	2.5	5	4	3		3			0	17	2.5	2.5	4	4	5		5	4		4	22	20	39	20
59	17RT1A03C1	2.5	2.5	4	4			0	4		4	17	2.5	2.5	4	3	5		5		3	3	20	19	39	21
60	18RT5A0301	2.5	2.5	5	4	5		5	4		4	23	2.5	2.5	5	4	5		5	5		5	24	24	59	36
61	18RT5A0302	2.5	2.5	5	4	5		5			0	19	2.5	2.5	4	2	5		5			0	16	18	49	32
62	18RT5A0303	2.5	2.5	5	5			0			0	15	2.5	2.5	3	3	4		4			0	15	15	49	34
63	18RT5A0304	2.5	2.5	5	3		3	3			0	16	2.5	2.5	3	4	5		5			0	17	17	49	33
64	18RT5A0305	2.5	2.5	5	4	5		5			0	19	2.5	2.5	3	3	4		4			0	15	17	49	32
65	18RT5A0306	2.5	2.5	4	5	5		5			0	19	2.5	2.5	4	3	5		5			0	17	18	39	21
66	18RT5A0307	2.5	2.5	4	4		5	5			0	18	2.5	2.5	4	4	5		5	4		4	22	20	49	29
67	18RT5A0308	2.5	2.5	5	5	1		1			0	16	2.5	2.5	3	3	3		3			0	14	15	49	34
68	18RT5A0309	2.5	2.5	5	5	5		5			0	20	2.5	2.5	3	3	5		5			0	16	18	49	31
69	18RT5A0310	2.5	2.5	4	4	5		5	1		1	19	2.5	2.5	4	2	4		4			0	15	17	59	42
70	18RT5A0311	2.5	2.5	4	4	5		5			0	18	2.5	2.5	4	3	5		5			0	17	18	50	42
71	18RT5A0312	2.5	2.5	4	4	5		5			0	18	2.5	2.5	3	3	3		3			0	14	16	49	33
72	18RT5A0313	2.5	2.5	5	4			0			0	14	2.5	2.5	4	4	5		5		3	3	21	18	39	22
73	18RT5A0314	2.5	2.5	5	5	5		5	1		1	21	2.5	2.5	4	5	5		5	4		4	23	22	59	37
74	18RT5A0315	2.5	2.5	5	5			0			0	15	2.5	2.5	3	4	3		3			0	15	15	49	34
75	18RT5A0316	2.5	2.5	5	4			0			0	14	2.5	2.5	4	3	2		2			0	14	14	39	25
76	18RT5A0317	2.5	2.5	5	4	2		2			0	16	2.5	2.5	3	3	3		3			0	14	15	49	34
77	18RT5A0318	2.5	2.5	5	5	5		5	3		3	23	2.5	2.5	5	4	5		5	5		5	24	24	69	46
78	18RT5A0319	2.5	2.5	5	5			0			0	15	2.5	2.5	4	4	5		5	4		4	22	19	59	41
79	18RT5A0322	2.5	2.5	5	4	2		2			0	16	2.5	2.5	5	4	5		5		5	5	24	20	39	19
80	18RT5A0323	2.5	2.5	5	5			0	5		5	20	2.5	2.5	4	4	5		5		3	3	21	21	49	29
81	18RT5A0324	2.5	2.5	5	5	4		4			0	19	2.5	2.5	4	4	5		5	2		2	20	20	49	30
82	18RT5A0325	2.5	2.5	5	4	4		4			0	18	2.5	2.5	4	3	4		4			0	16	17	49	32
83	18RT5A0326	2.5	2.5	5	5	5		5			0	20	2.5	2.5	4	5	5		5	4		4	23	22	39	18
84	18RT5A0327	2.5	2.5	5	5			0		4	4	19	2.5	2.5	4	4	5		5	3		3	21	20	49	29
85	18RT5A0328	2.5	2.5	5	5	5		5	3		3	23	2.5	2.5	5	4	5		5	5		5	24	24	59	36
86	18RT5A0329	2.5	2.5	5	4			0			0	14	2.5	2.5	4	3	5		5		2	2	19	17	49	33

87	18RT5A0330	2.5	2.5	5	4	5		5			0	19	2.5	2.5	4	3	4		4			0	16	18	39	22
88	18RT5A0331	2.5	2.5	5	4	3		9			0	17	2.5	2.5	3	4	3		3			0	15	16	39	23
Average Marks		2.48	2.48	4.52	4.02	3.79	4.00	2.94	3.46	3.50	1.42	17.9	1.81	1.81	4.26	3.92	4.70	4.00	4.67	3.93	3.79	3.15	19.6	18.72	46.61	27.9

CIE (Mid Exam) CO Wise Percentage

COURSE OUTCOME	CO Wise Sum	CO Wise Percentage %
CO1	9.95	79.59
CO2	7.93	63.41
CO3	10.74	85.91
CO4	8.87	70.99
Average	9.37	74.98

CIE - CO Wise Sum Formula

CO1 = ASG(CO1) + Q1(CO1) + BestOfQ2&Q3(CO1)
CO2 = ASG(CO2) + Q1(CO2) + BestOfQ4&Q5(CO2)
CO3 = ASG(CO3) + Q1(CO3) + BestOfQ2&Q3(CO3)
CO4 = ASG(CO4) + Q1(CO4) + BestOfQ4&Q5(CO4)

CIE - CO Wise Percentage

CO1 % = (CO1 SUM/total CO1 Marks(12.5))*100
CO2 % = (CO2 SUM/total CO2 Marks(12.5))*100
CO3 % = (CO3 SUM/total CO3 Marks(12.5))*100
CO4 % = (CO4 SUM/total CO4 Marks(12.5))*100

Average Marks	27.89
Student Count >Avg	45
Total Students	83
Percentage	54.2169

SEE (End Exam) CO Wise Percentage

CO1-CO4	27.89	54.22

SEE - CO Wise Percentage

CO1-CO4 = End Exam Avg Marks

SEE - CO Wise Percentage

CO1-CO4 % = (End Exam Avg Marks/75)*100

CO ATTAINMENT	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
CO1	80	3	54.2	2	2.25
CO2	63	2	54.2	2	2
CO3	86	3	54.2	2	2.25
CO4	71	3	54.2	2	2.25
Average					2.19

INTERNAL EXAM ATTAINMENT LEVEL SCALE

Attainment Levels	0	<=49
	1	50-59
	2	60-69
	3	>=70

EXTERNAL EXAM / FINAL ATTAINMENT LEVEL SCALE

Attainment Levels	0	<=39
	1	40-49
	2	50-59
	3	>=60

Direct Attainment %

CO1=(CO1IntAtn*0.25+CO1ExtAtn*0.75)
CO2=(CO2IntAtn*0.25+CO2ExtAtn*0.75)
CO3=(CO3IntAtn*0.25+CO3ExtAtn*0.75)
CO4=(CO4IntAtn*0.25+CO4ExtAtn*0.75)

CO-PO Matrix

Course	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2	PSO3	[illegible]
CO1	3	2	3	3	2	3	2		1	0	1	3	2	1	3	2.25
CO2	3	2	3	3	2	3	2		1	0	1	3	2	1	3	2
CO3	3	2	3	3	2	3	2		1	0	1	3	2	1	3	2.25
CO4	3	2	3	3	2	3	2		1	0	1	3	2	2	3	2.25
Average	3	2	3	3	2	3	2	0	1	0	1	3	2	1	3	2.1875

Final Attainment %

CO1 = (DIRECT ATTAINMENT*0.8) + (INDIRECT ATTAINMENT*0.2)
CO2 = (DIRECT ATTAINMENT*0.8) + (INDIRECT ATTAINMENT*0.2)
CO3 = (DIRECT ATTAINMENT*0.8) + (INDIRECT ATTAINMENT*0.2)
CO4 = (DIRECT ATTAINMENT*0.8) + (INDIRECT ATTAINMENT*0.2)

Course PO Attainments

	PO1	PO2	PO3	PO4	PO5	PO6	PO7	PO8	PO9	PO10	PO11	PO12	PSO1	PSO2	PSO3
[illegible]	2.1875	1.4583	2.1875	2.1875	1.4583	2.1875	1.4583	0	0.7292	0	0.72917	2.1875	1.4583	0.7292	2.1875
[illegible]	2.1875	1.8229	2.1875	2.1875	1.8229	2.1875	1.8229	1.09375	1.4583	1.0938	1.45833	2.1875	1.8229	1.4583	2.1875
[illegible]	2.1875	1.5313	2.1875	2.1875	1.5313	2.1875	1.5313	0.2188	0.875	0.2188	0.875	2.1875	1.5313	0.875	2.1875

PO ATTAINMENTS

DIRECT ATTAINMENT (PO1)= (Average of PO1*Average of CO Direct Attainment)/3
Similar for PO2 TO PO12 & PSO1 TO PSO3
INDIRECT ATTAINMENT (PO1) = (Average of PO1*Average of CO Direct Attainment)/2
Similar for PO2-PO12 & PSO1 TO PSO3
FINAL ATTAINMENT = (DIR ATNM-PO1)*0.8 + (INDIR ATNM-PO1)*0.2