GOVT. POLYTECHNIC , BHADRAK

Design of Machine Elements

(Th- 02)

(As per the 2019-20 syllabus of the SCTE&VT, Bhubaneswar, Odisha)

Fifth Semester

Mechanical Engg.

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DESIGN OF MACHINE ELEMENTS

CHAPTER-WISE DISTRIBUTION OF PERIODS & MARKS

Sl. No.	Chapter No.	Topics	Periods asper syllabus	Periods actually needed	Expected marks for IA	Expected marks For End Sem
1	1	INTRODUCTION	**12**	**14**	**04**	**16**
2	2	DESIGN OF FASTENING ELEMENTS	**12**	**14**	**08**	**12**
3	3	DESIGN OF SHAFTAND KEYS	**12**	**14**	**08**	**12**
4	4	DESIGN OF COUPLING	**12**	**12**		**20**
5	5	DESIGN OF CLOSEDCOIL HELICAL SPRING	**12**	**12**		**20**
Total			**60**	**66**	**20**	**80**

Chapter No.: 01

INTRODUCTION

Scope of Syllabus:

Introduction to Machine Design and classify it
Different mechanical engineering materials used in design with their uses and properties
Define working stress, yield stress, ultimate stress and factor of safety and stress-strain curve for MS and CI
Modes of failure
State the factors governing the design of machine elements
Describe design procedure

1.0 MACHINE DESIGN:

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.

In designing a machine component, it is necessary to have a good knowledge of many subjects such as Mathematics, Engg Mechanics, Strength of Materials, Theory of Machines, Production Technology and Engg Drawing.

CLASSIFICATION OF MACHINE DESIGN:

1) **Empirical Design**: The design based on past experience and existing practice is known as the Empirical Design. In this design, various dimensions are set out as a proportion of certain main dimension e.g. the various dimensions of nut and bolt are given as proportions of the bolt diameter.

2) **Rational Design**: It is purely a mathematical design and it is based upon principles of engineering mechanics, engineering material, etc. In this design, the different relations developed or studied in the strength of material are used to decide the different dimensions of a product.

3) **Adaptive Design or Redesign**: This work is concerned with adaptation of existing designs. The final outcome doesn't differ much from the initial product. Much of designer's work consists of redesign.

4) **Developed Design**: It's also started from an existing design. The final outcome may quiet differ from the initial product.

5) **New Design** *or* **Inventive Design** *or* **Creative Design**: These designs are created from a scratch through the application of scientific laws, technical ability and creative thinking. If a problem arises that suggests a machine for the solution and no suitable machine exists, a designer will have to create it.

1.2 DIFFERENT MECHANICAL ENGINEERING MATERIALS USED IN MACHINE DESIGN (with their properties and uses):

- **CAST IRON**:

It is an alloy of iron and carbon. The carbon content in the CI varies from **1.7% to 4.5%**. It also contains small amount of Silicon, Manganese, Phosphorous and Sulphur.

Properties:

- It is a brittle material
- Low cost
- Good casting characteristics
- High compressive strength
- High wear resistance
- Excellent machinability

Uses:

- Machine tool bodies
- Automotive cylinder blocks, heads, housings, fly-wheels, etc.
- Gears, rolls for rolling mill and centrifugally cast products.
- Parts of agricultural machinery, Pipe fittings, door hinges, locks, etc.
- Hubs of wagon wheels, brake supports, etc.

- ## WROUGHT IRON:

It is the purest iron which contains 99.5% to 99.9% of iron. It also contains carbon, silicon, sulphur, phosphorus and slag.

Properties:

- It is a tough, malleable and ductile material
- It cannot stand sudden and excessive shocks.
- It can be easily forged or welded.

Uses:

- Chains
- Crane hooks
- Railway couplings
- Water and steam pipes, etc.

- ## STEEL (OR PLAIN CARBON STEEL/ CARBON STEEL):

It is an alloy of iron and carbon, with carbon content up to a maximum of 1.5%.the plain carbon steels are divided into the following types depending upon the carbon content. (i) Low carbon steel or Mild steel – 0.15% to 0.45% C (ii) Medium carbon steel – 0.45% to 0.80% C (iii) High carbon steel – 0.80 to 1.5% C.

Properties:

- Low carbon steel – low cost, low strength and good weld ability.
- Medium carbon steel – good toughness and ductility, relatively good strength.
- High carbon steel – screw driver, hammer, wrench, punch, spring, knives, razors, etc.

Uses:

- Low carbon steel – chain, pipe, wire, etc.
- Medium carbon steel – rolls, axles, screws, crankshafts, etc.
- High carbon steel – high strength, hardness and wear resistance.

- ## ALLOY STEEL:

It may be defined as a steel to which elements other than carbon are added in sufficient amount to produce an improvement in properties. The alloying is done for specific purposes to increase wearing resistance, corrosion resistance and to improve electrical and magnetic properties which cannot be obtained in plain carbon steel. The chief alloying elements are nickel, chromium, molybdenum, cobalt, vanadium, manganese, silicon and tungsten.

Properties:

- Nickel steel – high strength and toughness.
- Chrome steel – hardness with high strength.
- Tungsten steel – high strength, hardness, wear resistance, etc.
- Vanadium steel – high tensile strength, fine grain structure, etc.

Uses:

- Boiler tubes, valves for IC engines
- Crank shafts, axle and gears.
- Balls, rollers, races of bearings.
- Cutting tools, dies, taps and permanent magnet.
- Spring, pin, and many drop forged parts.

- ## ALUMINIUM AND ITS ALLOYS:

Aluminium may be alloyed with one or more other elements like copper, magnesium, manganese, silicon and nickel. The addition of alloying elements converts the weak and soft metal into hard and strong metal, while still remaining its light weight.

Properties:

- **Duralumin** (Al 95% + Cu 4% + Mn 0.5% + Mg 0.5%) – high tensile strength and light weight.
- **Y alloy** – high strength and machinability.
- **Magnalium** – light weight and good mechanical properties.

Uses:

- **Duralumin** – connecting rod, bar, pulley, rivets, etc.
- **Y alloy** – cylinder heads and pistons of IC engine.
- **Magnalium** – air craft and automobile components.

- ## COPPER AND ITS ALLOYS:

Properties:

- **Copper - zinc alloy (Brass)** – resistant to atmospheric corrosion and can be easily soldered.
- **Copper - tin alloy (Bronze)**– corrosion resistant, hard, resists surface wear, very easily shaped or rolled into wires, rods and sheets.

Uses:

- **Brass** – wire, tube, valve, plumbing fittings, automobile fittings, etc.
- **Bronze** – bearing, worm wheels, nuts for machine lead screw, tanks, cams, bushings, condenser bolts, etc.

- ## GUN METAL:

It is an alloy of copper, tin and zinc.

Properties:

- The metal is very strong and resistant to corrosion by water and atmosphere.

Uses:

- Casting guns, boiler fittings, bushes, bearings, etc.

- ## NON–METALLIC MATERIALS:

Properties:

- **Plastics** – high resistant to corrosion and have a high dimensional stability.
- **Rubber** – it resists abrasion, heat, strong alkalis and fairly strong acids.
- **Leather** – very flexible and can withstand wear.
- **Ferrodo** – friction material.

Uses:

- **Plastics** – aeroplane and automobile parts, safety glasses, laminated gears, pulleys, self-lubricating bearings, etc.
- **Rubber** – power transmission belting, etc.
- **Leather** – power transmission belting, washer, etc.
- **Ferrodo** – friction lining for clutches and brakes, etc.

1.3 WORKING STRESS, YIELD STRESS, ULTIMATE STRESS, FACTOR OF SAFETY AND STRESS – STRAIN CURVE FOR MS & CI:

Working stress:

When designing machine parts, it is desirable to keep the stress lower than the maximum stress or ultimate stress at which failure of the material takes place. This stress is known as working stress (or design stress/ safe stress/ allowable stress).

Yield stress:

It is the stress at the instant, when the strain in the body increases without any further increase in load. This is the maximum allowable stress that a body can bear without going into permanent deformation.

Ultimate stress:

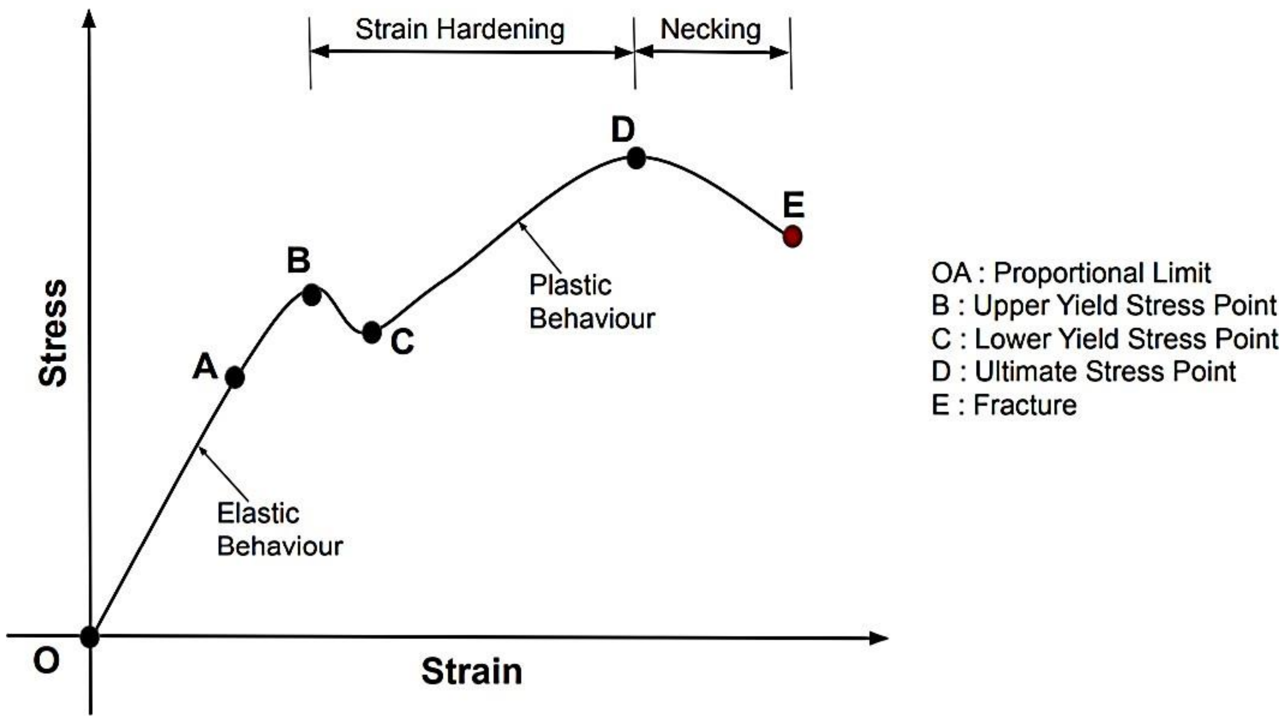
It is the maximum value of stress that a material can resist. It may be defined as the largest stress obtained by dividing the largest value of the load reached in a test to the original cross-sectional area of the test piece.

Factor of Safety:

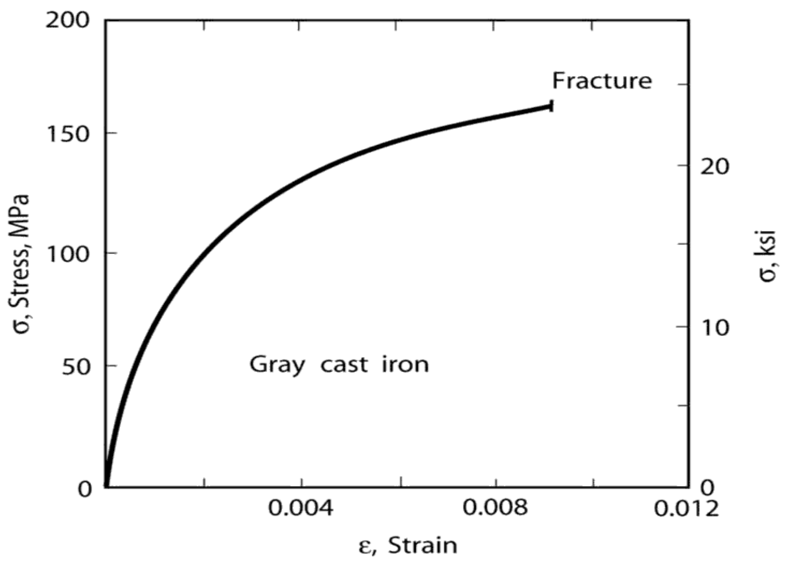
In general, it is defined as the ratio of maximum stress to the working stress. Mathematically,

$$f_s = \frac{Maximum\ Stress}{Working\ Stress}$$

Stress – Strain curve for Mild Steel (MS):



Stress – Strain curve for Cast Iron (CI):



MODES OF FAILURE:

There are three different modes of failure:

- Failure by elastic deflection
- Failure by general yielding
- Failure by fracture

Failure by Elastic Deflection:

In applications like transmission shaft supporting gears, the maximum force acting on the shaft, without affecting its performance, is limited by the permissible elastic deflection. Lateral or torsional rigidity is considered as the criterion of design in such cases.

Failure by General Yielding:

A mechanical component made of ductile material loses its engineering usefulness due to a large amount of plastic deformation after the yield point stress is reached. Considerable portion of the component is subjected to plastic deformation, called general yielding.

Failure by Fracture:

Components made of brittle material cease to function satisfactorily because of the sudden fracture without any plastic deformation.

FACTORS GOVERNING THE DESIGN OF MACHINE ELEMENTS:

The factors governing the design of machine elements are as follows:

- ***Type of Load and Stresses caused by the Load:*** The load, on a machine component, may act in several ways due to which the internal stresses are set up.
- ***Motion of the parts or kinematics of the machine:*** The successful operation of any machine depends largely upon the simplest arrangements of the parts, which will give the required motion.
- ***Selection of materials:*** Every Machine Design Engineer should have a thorough knowledge of the properties of material and their behaviour under different working conditions.
- ***Form and size of the parts:*** In order to design any machine part for form and size, it is necessary to know the forces which the part must sustain. The smallest practicable cross-section may be used, but it may be checked that the stresses induced in the designed cross-section are reasonably safe.
- ***Frictional resistance and lubrication***: There is always a loss of power due to frictional resistance. Hence, careful attention must be given to the matter of lubrication of all surfaces which moves in contact with others.

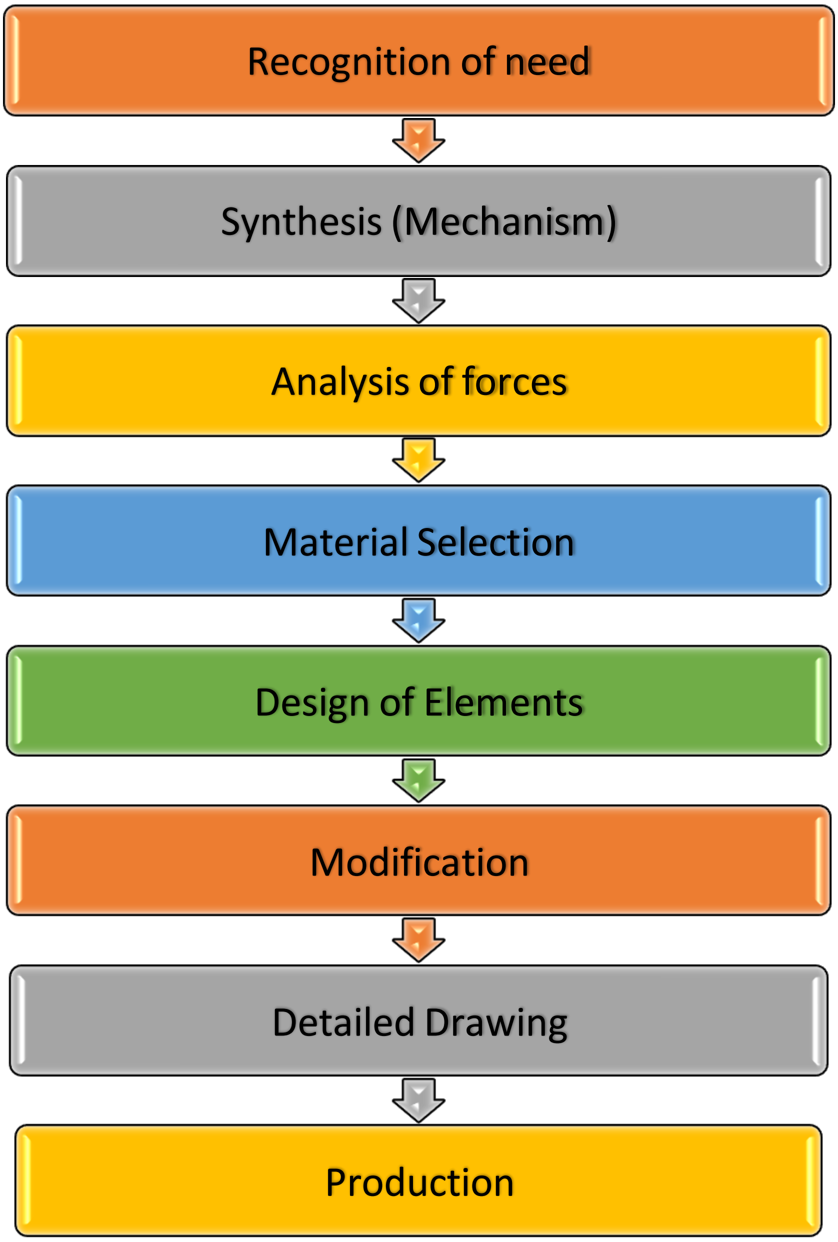
- ❖ ***Convenient and economical features***: The operating feature *of* the machine should be carefully studied. The starting, controlling and stopping levers should be located on the basis of convenient handling.
- ❖ ***Use of standard parts***: The uses of standard parts are closely related to the cost of machine. Because the cost of standard parts is only a fraction of the cost of similar parts made to order.
- ❖ ***Safety of operation:*** A Machine Designer should always provide a safety device for the safety of the operator.
- ❖ ***Workshop facilities:*** A Design Engineer should be familiar with the limitation of his Employer's Workshop.
- ❖ ***Number of machines to be manufactured:*** If only a few articles are to be made, extra expenses are not justified unless the machine is large or of some special design.
- ❖ ***Cost of construction:*** The aim of the Design Engineer under all conditions should be to reduce the manufacturing cost to the minimum.
- ❖ ***Assembling:*** The final location of any machine is important. Hence, The Design Engineer must anticipate the exact location and the local facilities for erection.

DESIGN PROCEDURE:

In designing machine component, there is no rigid rule. The problem may be attempted in several ways. However, the general procedure to solve a design problem is as follows:

- ❖ ***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.
- ❖ ***Synthesis (Mechanisms)***: Select the possible mechanism or group of mechanisms which will give the desired motion.
- ❖ ***Analysis of forces***: Find the forces acting on each member of the machine and the energy transmitted by each member.
- ❖ ***Material selection***: Select the material best suited for each member of the machine.

❖ ***Design of elements (Size and Stresses)***: Find the size of each member of the machine by considering the forces acting on the member and the permissible stresses for the material used.



❖ ***Modification***: Modify the size of the member to agree with the past experience and judgement to facilitate manufacture. The modification may also be necessary to reduce overall cost.

❖ ***Detailed drawing***: Draw the detailed drawing of each component and assembly of the machine with complete specification.

❖ ***Production***: The component, as per the drawing, is manufactured in the Workshop.

POSSIBLE SHORT TYPE QUESTION WITH ANSWER:

1. ***Define Machine design.***

Ans: 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.

2. ***Define working stress or design stress.***

Ans: When designing machine parts, it is desirable to keep the stress lower than the maximum stress or ultimate stress at which failure of the material takes place. This stress is known as working stress (or design stress/ safe stress/ allowable stress).

3. ***Define ultimate stress.***

Ans: It is the maximum value of stress that a material can resist. It may be defined as the largest stress obtained by dividing the largest value of the load reached in a test to the original cross-sectional area of the test piece.

4. ***Define yield stress.***

Ans: It is the stress at the instant, when the strain in the body increases without any further increase in load. This is the maximum allowable stress that a body can bear without going into permanent deformation.

5. ***Define factor of safety.***

Ans: In general, it is defined as the ratio of maximum stress to the working stress. Mathematically,

$$f_s = \frac{Maximum\ Stress}{Working\ Stress}$$

6. ***State the modes of failure of material.***

Ans: The different modes of failure of material are elastic deflection, general yielding and fracture.

POSSIBLE LONG TYPE QUESTION:

1. *Explain the factors affecting design of machine elements.*
2. *Describe design procedure with flow chart.*
3. *State the properties and uses of different mechanical engineering materials.*

Chapter No.: 02

DESIGN OF FASTENING ELEMENTS

Scope of Syllabus:

Joints and their classification
State types of welded joints
State advantages of welded joint over other joints **2.4***Design of welded joints for eccentric loads and numerical***2.5***State types of riveted joints and rivets*
Describe failure of riveted joints
Determination of strength and efficiency of riveted joints
Design of riveted joints
Numerical

JOINTS AND THEIR CLASSIFICATION:

Mechanical joints are used to connect the parts (or elements or links) of a mechanism or machine. These joints are ***temporary*** or ***permanent*** depending on whether the connection needs to be removed frequently or not removed at all.

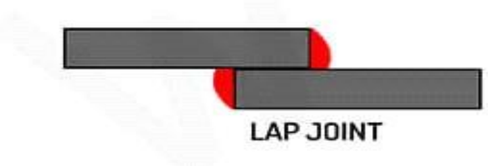
Temporary Joint	Permanent Joint
Temporary joints allow easy dismantling of assembled components without breaking them.	Permanent joints don't allow dismantling of assembled components without rupturing them.
Temporary joints are not necessarily leak-proof.	Permanent joints are usually leak-proof.
Strength of temporary joint is comparatively less.	Strength of permanent joint is high. Usually, joint strength is same with that of the components.
It facilitates fast, easy and cost-efficient inspection. No destructive testing is required for inspection of joints.	As permanent joints cannot be disassembled easily, so inspection is difficult and costly. Often destructive testing is carried out, which damages the assembled structures.
Repair and replacement are also easy.	Repair and replacement are difficult and costly.
Temporary joints are suitable where frequent separation of assembled components is required.	Permanent joints are suitable for such applications where separation is usually not desired in the service life.
Examples of various temporary joints: • Fasteners • Press fit • Cotter joints • Knuckle joints, etc.	Examples of various permanent joints: • Welding • Brazing and soldering • Riveting • Adhesive joining (mostly) • Coupling, etc.

WELDED JOINTS AND ITS TYPES:

A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without application of pressure and a filler material.

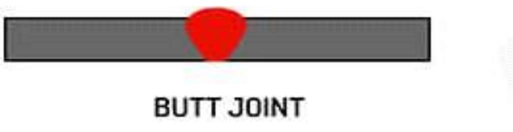
Following two types of welded joints are important from the subject point of view:

1. **Lap Joint or Fillet Joint:**



It is obtained by overlapping the plates and then welding the edges of the plates.

2. **Butt Joint:**



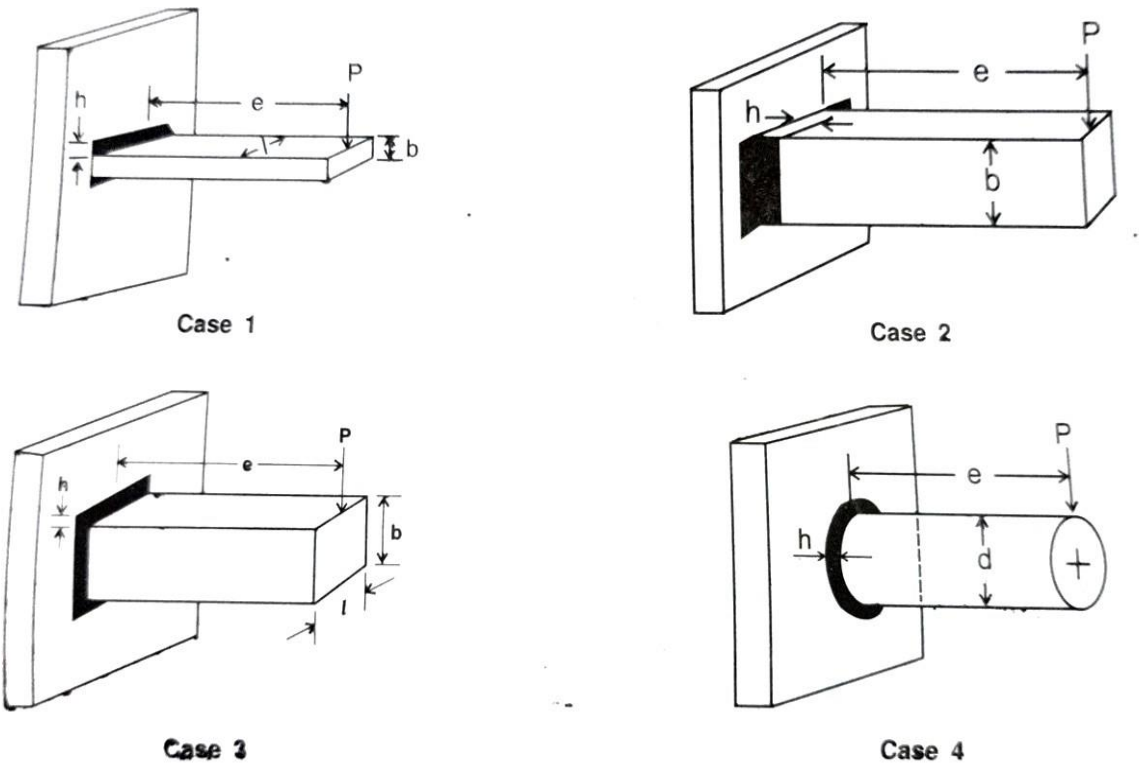
It is obtained by placing the plates edge to edge and then welding the edges.

ADVANTAGES OF WELDED JOINTS OVER RIVETED JOINTS:

- The welded structures are usually light in weight compared to riveted structures. This is due to the reason, that in welding, gussets or other connecting components are not used.
- The welded joints provide high efficiency, which is not possible in the case of riveted joints.
- Alterations and additions can be made easily in the existing structures.
- Welded structures are smooth in appearance; therefore, it looks pleasing.
- A welded joint has a great strength. Often a welded joint has the strength of the parent metal itself.
- It is easily possible to weld any part of a structure at any point. But riveting requires enough clearance.
- The process of making welding joints takes less time than the riveted joints.
- Shape like cylindrical steel pipes can be easily welded. But they are difficulty for riveting.
- The welding provides very rigid joints. This is in line with the modern trend of providing rigid frames.
- In welded connections, the tension members are not weakened as in the case of riveted joints.

DESIGN OF WELDED JOINTS FOR ECCENTRIC LOADING:

(Ref: Design Data Hand Book by S. Md. Jalaludeen)



Maximum normal stress due to eccentric load

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Maximum shear stress due to eccentric load

$$r_{max} = \frac{1}{2}[\sqrt{\sigma_b^2 + 4r^2}]$$

Where

For Case- 1

$$\sigma_b = \frac{1.414P.e}{h\,b\,l}$$

$$r = \frac{0.707P}{h\,l}$$

For Case- 2

$$\sigma_b = \frac{4.242P.e}{h\,b^2}$$

$$r = \frac{0.707P}{h\,b}$$

For Case- 3

$$\sigma_b = \frac{4.242P.e}{h\,(3lb + b^2)}$$

$$r = \frac{0.707P}{h\,(l + b)}$$

For Case- 4

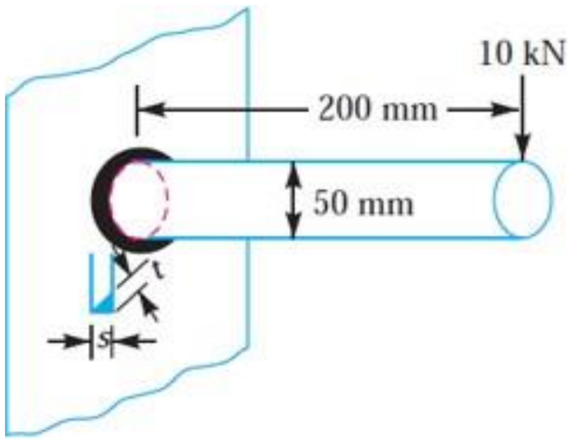
$$\sigma_b = \frac{5.66P.e}{\pi d^2 h}$$

$$r = \frac{1.414P}{\pi\,d\,h}$$

NUMERICALS ON WELDED JOINT:

Problem No. 01.

A 50 mm diameter and 200 mm long solid shaft is welded to a flat plate as shown in the figure below. If the size of the weld is 15 mm, find the maximum normal and shear stress in the weld.



Solution:

Data Given:

The given figure matches with figure of **case-4** of our *Data Book*. So,

P = 10 kN = 10 X 10^3; e = 200 mm; d = 50 mm; h = 15 mm

We know that for case-4,

Bending stress, $\sigma_b = \frac{5.66P.e}{\pi d^2 h} = \frac{5.66 \times 10 \times 10^3 \times 200}{\pi \times 50^2 \times 15}$ = 96.087 N/mm^2

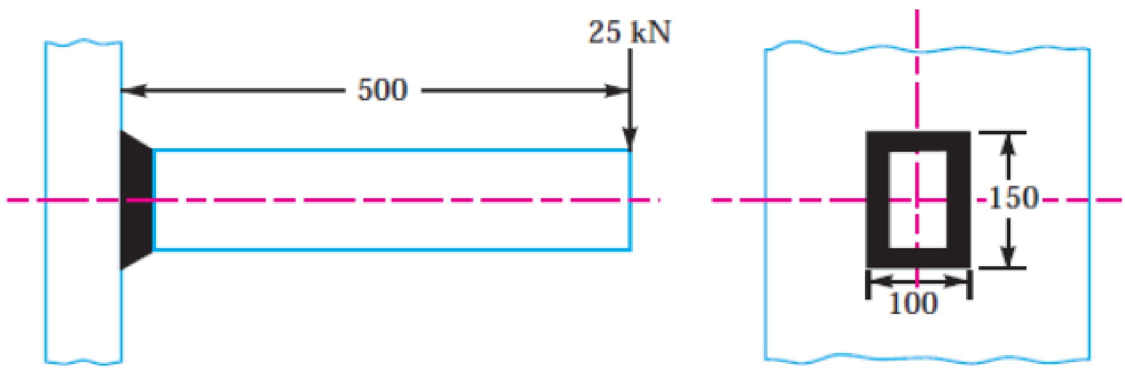
And, maximum shear stress,

= 48.4 N/mm^2 = **48.4 MPa (Ans)**

Problem No. 02.

A rectangular cross section bar is welded to a support by means of fillet welds as shown in the figure.

Determine the size of the welds, if the permissible shear stress in the weld is limited to 75 MPa.



Solution:

Data Given:

The given figure matches with figure of **case-3** of our *Data Book*. So,

P = 25 kN = 25 X 10^3; e = 500 mm; b = 150 mm; l = 100 mm; r_{max} = 75 MPa = 75 N/mm^2

We know that for case-3,

Bending stress, $\sigma_b = \frac{4.242\ P.e}{h\ (3lb + b^2)} = \frac{4.242 \times 25 \times 10^3 \times 500}{h\ (3 \times 100 \times 150 + 150^2)} = \frac{785.56}{h}$ N/mm^2

And, shear stress, $r = \frac{0.707P}{h(l+b)} = \frac{0.707 \times 25 \times 10^3}{h\ (100+150)} = \frac{70.7}{h}$ N/mm^2

We know that, maximum shear stress,

$r_{max} = \frac{1}{2}[\sqrt{\sigma_b^2 + 4r^2}]$ = 75

$or,\ \frac{1}{2}\left[\sqrt{\left(\frac{785.56}{h}\right)^2 + 4\left(\frac{70.7}{h}\right)^2}\right]$ = 75

$or,\ \frac{399.2}{h}$ = 75

Or, h = 399.2 / 75 = **5.32 mm (Ans)**

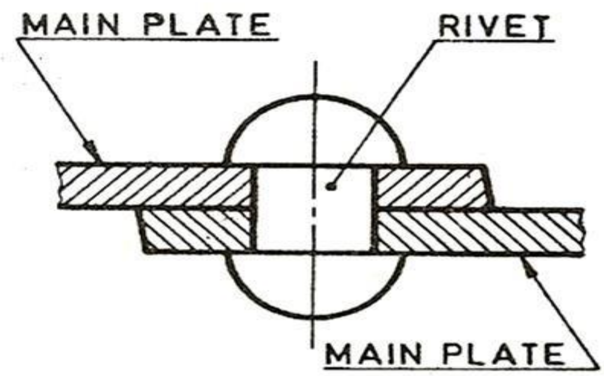
RIVETED JOINTS AND ITS TYPES:

Riveted joint is a permanent joint with mainly two components (parts to be joined) which are held together by a *rivet* with the head at top and tail at the bottom.

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

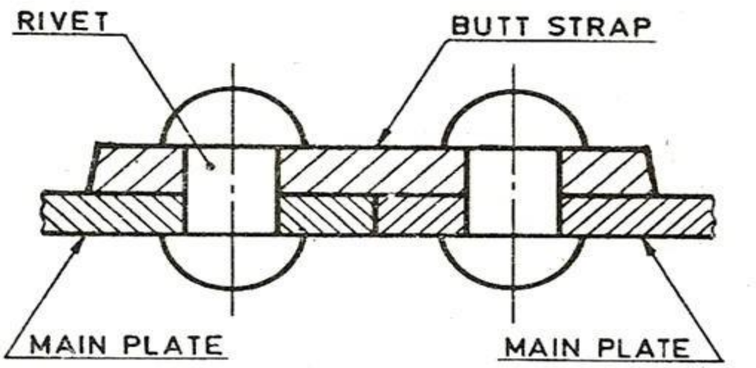
- Lap joints
- Butt joints

Lap Joint:



A lap joint is that in which one plate overlaps the other and the two plates are then riveted together.

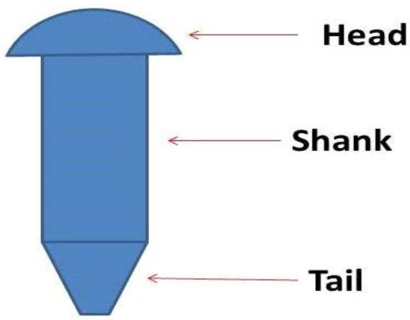
Butt Joint:



A butt joint is that in which the main plates are kept in alignment butting or touching each other and a cover plate or strap is placed either on one side or on both sides of the main plates. The cover plate is then riveted together with the main plates.

RIVET AND ITS TYPES:

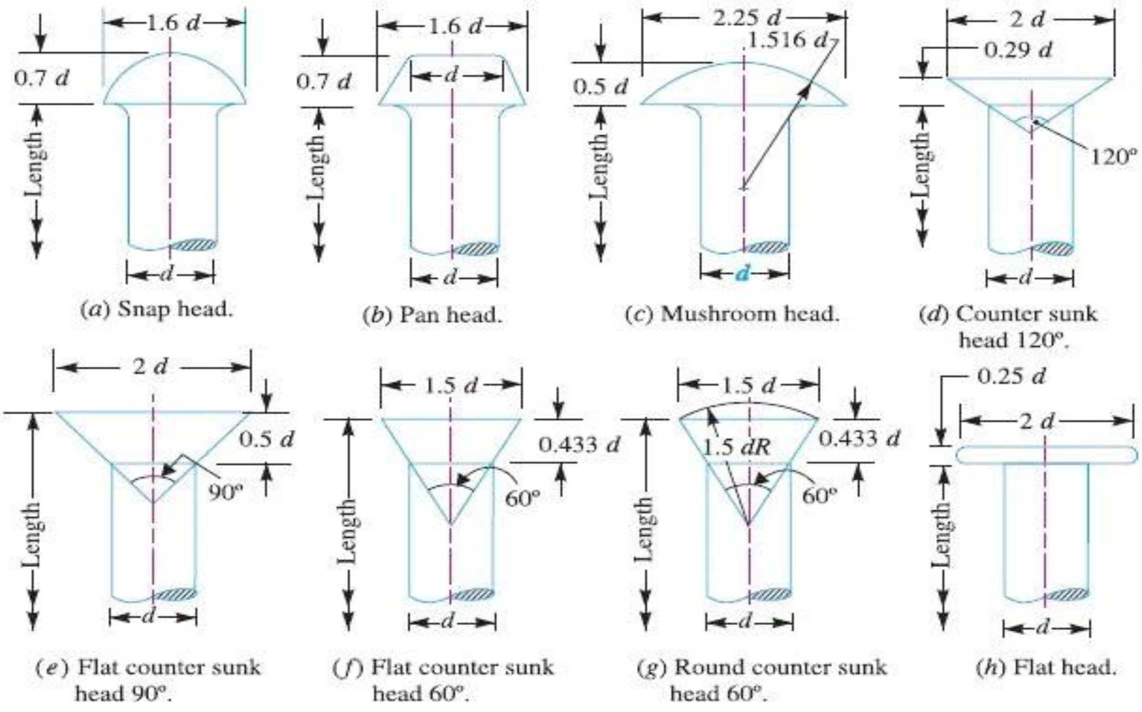
A rivet is a short cylindrical bar with a head integral to it.



Rivet

According to Indian Standard Specifications, the rivets (or rivet heads) are classified into following types:

- Snap head rivet
- Pan head rivet
- Mushroom head rivet
- Counter sunk head rivet
- Flat counter sunk (90^0) rivet
- Flat counter sunk (60^0) rivet
- Round counter sunk (60^0) rivet
- Flat head rivet

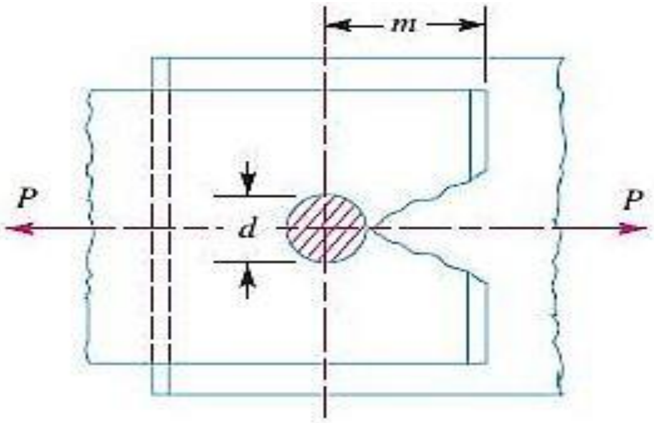


FAILURE OF RIVETED JOINTS:

A riveted joint may fail in the following ways:

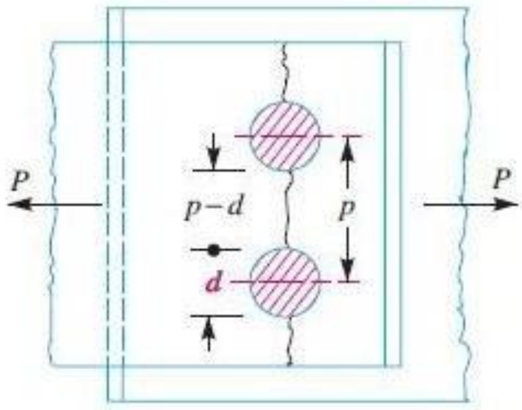
- ***Tearing of the plate at an edge***:

  A joint may fail due to the tearing of the plate at an edge.
  This can be avoided by keeping the margin, *m = 1.5d*
  where *d* is the diameter of the rivet hole.



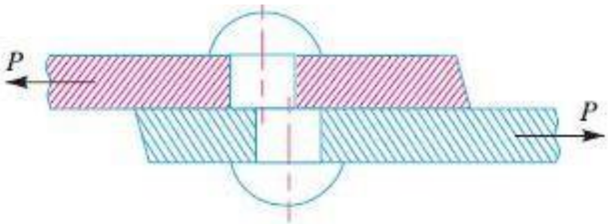
- ***Tearing of the plate across the row of rivet***:

  Due to the tensile stresses in the main plates, the main plate or cover plates may tear off across a row of rivets.

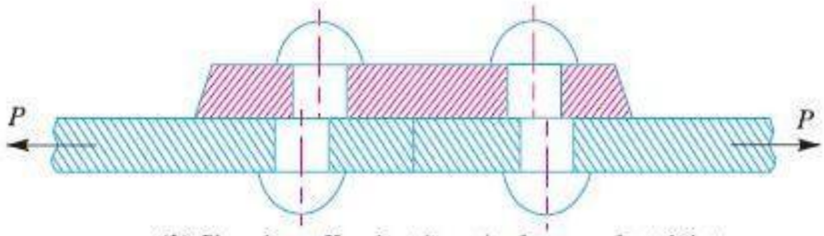


When the tearing resistance or tearing strength (P_t) is greater than the applied load (P) per pitch length, then this type of failure will not occur.

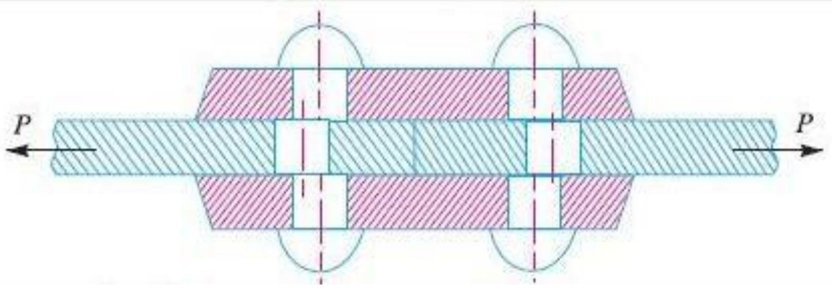
- ***Shearing of the rivets***: The plates which are connected by the rivets exert tensile stress on the rivets, and if the rivets are unable to resist the stress, they are sheared off.



(*a*) Shearing off a rivet in a lap joint.



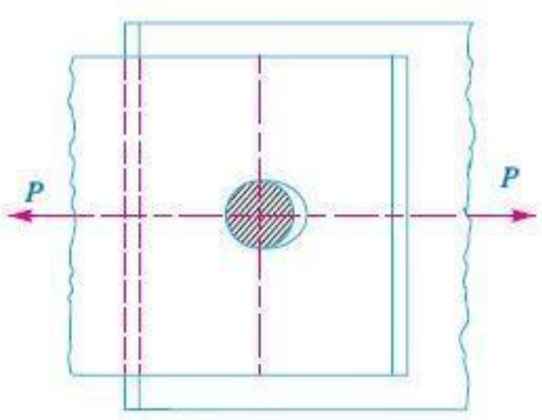
(*b*) Shearing off a rivet in a single cover butt joint.



When the shearing resistance or shearing strength (P_s) is greater than the applied load (P) per pitch length this type of failure will not occur.

- ***Crushing of the plate or rivet***:

Sometimes, the rivets do not actually shear off under the tensile stress, but are crushed. Due to this, the rivet hole becomes of an oval and hence the joint becomes loose. It is also known as ***Bearing Failure***.



When the crushing resistance or crushing strength (P_c) is greater than the applied load (P) per pitch length this type of failure will not occur.

STRENGTH AND EFFICIENCY OF RIVETED JOINT:

(*Ref: Design Data Hand Book by S. Md. Jalaludeen*)

The strength of a Riveted Joint:

It may be defined as the maximum force, which it can transmit, without causing it to fail.

Different strengths of riveted joint:

- Tearing strength of perforated plate, $F_t = (p - d)t.\sigma_t$
- Shearing strength of rivets, $F_s = n.\frac{\pi}{4} d^2 r$
- Crushing strength of rivets, $F_c = n\ d\ t\ \sigma_c$
- Strength of the unriveted plate, $F = p\ t\ \sigma_t$

Efficiency of a Riveted Joint:

The efficiency of a riveted joint is defined as the ratio of the strength of riveted joint to the strength of the unriveted or solid plate.

Mathematically,

Efficiency of the riveted joint,

$$\eta = \frac{Least\ of\ F_t, F_s, F_c}{F}$$

DESIGN OF RIVETED JOINTS FOR PRESSURE VESSEL/ BOILER:

Assumptions in designing Boiler joints:

- The load on the joint is equally shared by all the rivets.
- The tensile stress is equally distributed over the section of metal between the rivets.
- The shearing stress in all the rivets is uniform.
- The crushing stress is uniform.
- There is no bending stress in the rivets.
- The holes into which the rivets are driven do not weaken the member.
- The rivet fills the hole after it is driven.
- The friction between the surfaces of the plate is neglected.

Design Particulars:

(*Refer Design Data Hand Book by S. Md. Jalaludeen*)

- Thickness of boiler plate, $t = \frac{PD}{2\,\sigma_t\,\eta_l}$
- Diameter of rivet for $t > 8$ mm (for $t < 8$ mm, obtain d by equating the shear and crushing strength of rivets), **Unwin's formula**: $d' = 6.05\sqrt{t}$
- Minimum pitch, $p_{min} = (2.25\,d)$
- Maximum pitch, $p_{max} = C.t + 41$ (for the value of C , refer ***table 10.7***)
- Pitch of rivets (p) can be obtained by equating the tearing and shearing strength of the rivet. The value of p must be lies in between p_{max} and p_{min}.
- Distance between the rows of rivets or back pitch, $p_b = 2d$ (for chain riveting) or $p_b = 0.33p + 0.67d$ (for *zig-zag* riveting).
- Thickness of cover plate, $t_1 = 1.125\,t$ (for single cover) or $t_1 = 0.625\,t$ (for double cover of equal widths).
- Margin, $= 1.5\,d$.
- Find $F_t\,, F_s\ and\ F_c$ and then Efficiency of the joint.

NUMERICALS ON RIVETED JOINT:

Problem No. 01.

Find the efficiency of the single riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 50 mm. Take:

Permissible tensile stress in plate = 120 MPa

Permissible shearing stress in rivets = 90 MPa

Permissible crushing stress in rivet = 180 MPa.

Solution:

Data Given:

n = 1; t = 6 mm; d = 20 mm; p = 50 mm; $\sigma_t = 120\ MPa = 120\ N/mm^2$; $\tau = 90\ MPa = 90\ N/mm^2$; $\sigma_c = 180\ MPa = 180\ N/mm^2$

Tearing resistance of the plate, $F_t = (p - d)t.\sigma_t$ = (50 – 20) 6 X 120 = 21 600 N

Shearing resistance of the rivet, $F_s = n.\frac{\pi}{4} d^2 \tau$ = 1 X $\frac{\pi}{4}$ X 20^2 X 90 = 28 278 N

Crushing resistance of the rivet, $F_c = n\ d\ t\ \sigma_c$ = 1 X 20 X 6 X 180 = 21 600 N

Therefore, strength of the joint = Least of the above resistances = 21 600 N

Again, we know that strength of the unriveted plate, $F = p\ t\ \sigma_t$ = 50 X 6 X 120 = 36 000 N

Then, efficiency of the joint, $\eta = \frac{Least\ of\ F_t, F_s, F_c}{F}$ = $\frac{21\ 600}{36\ 000}$ = **0.60 or 60% (Ans)**

Problem No. 02.

Find the efficiency of the double riveted lap joint of 6 mm plates with 20 mm diameter rivets having a pitch of 65 mm. Take:

Permissible tensile stress in plate = 120 MPa

Permissible shearing stress in rivets = 90 MPa

Permissible crushing stress in rivet = 180 MPa.

Solution:

Data Given:

n = 2; t = 6 mm; d = 20 mm; p = 65 mm; $\sigma_t = 120\ MPa = 120\ N/mm^2$; $r = 90\ MPa = 90\ N/mm^2$; $\sigma_c = 180\ MPa = 180\ N/mm^2$

Tearing resistance of the plate, $F_t = (p - d)t.\sigma_t$ = (65 – 20) 6 X 120 = 32 400 N

Shearing resistance of the rivet, $F_s = n.\frac{\pi}{4}\ d^2\ r$ = 2 X $\frac{\pi}{4}$ X 20^2 X 90 = 56 556 N

Crushing resistance of the rivet, $F_c = n\ d\ t\ \sigma_c$ = 2 X 20 X 6 X 180 = 43 200 N

Therefore, strength of the joint = Least of the above resistances = 32 400 N

Again, we know that strength of the unriveted plate, $F = p\ t\ \sigma_t$ = 65 X 6 X 120 = 46 800 N

Then, efficiency of the joint, $\eta = \frac{Least\ of\ F_t, F_s, F_c}{F}$ = $\frac{32\ 400}{46\ 800}$ = **0.692 or 69% (Ans)**

Problem No. 03.

A double riveted lap joint with zig-zag riveting is to be designed for 13 mm thick plates. Assume

σ_t = 80 MPa; r = 60 MPa; σ_c = 120 MPa.

State how the joint will fail and find the efficiency of the joint.

Solution:

Data Given:

n = 2; t = 13 mm; $\sigma_t = 80\ MPa = 80\ N/mm^2$; $r = 60\ MPa = 60\ N/mm^2$; $\sigma_c = 120\ MPa = 120\ N/mm^2$

Diameter of the rivet:

Since the diameter of the plate is greater than 8 mm, therefore diameter of the rivet, $d' = 6.05\ \sqrt{t} = 6.05$ X $\sqrt{13}$ = 21.81 mm

From table 10.5, we find that the standard size of the diameter of rivet is 22 mm and the corresponding diameter of the rivet hole, **d = 23 mm (Ans)**

Pitch of rivets:

We know that the pitch of the rivets may be calculated by equating the tearing and shearing resistances.

Tearing resistance of the plate,

$F_t = (p - d)t.\sigma_t$ = (p – 23) 13 X 80 = (p – 23) X 1040 N

Shearing resistance of the rivet, $F_s = n.\frac{\pi}{4} d^2 r$ = 2 X $\frac{\pi}{4}$ X 23^2 X 60 = 49 864 N

Then, (p – 23) X 1040 = 49 864

Or, p = 48 + 23 = 71 mm **(Ans)**

The maximum pitch, $p_{max} = C.t + 41$ = 2.62 X 13 + 41 = 75.06 mm (taking C = 2.62 mm from ***Table No- 10.7***)

The minimum pitch, $p_{min} = (2.25\ d)$ = 2.25 X 23 = 51.75 mm

Since, the calculated pitch lies between the maximum pitch and the minimum pitch, therefore we shall adopt **p = 71 mm** **(Ans)**

Distance between the rows of rivets:

We know that, for zig-zag riveting, $p_b = 0.33p + 0.67d$ = 0.33 X 71 + 0.67 X 23

= 38.8 mm or **40 mm** **(Ans)**

Margin:

Margin, m = 1.5 d = 1.5 X 23 = 34.5 mm or **35 mm** **(Ans)**

Failure of the joint:

We know that,

Tearing resistance of the plate, $F_t = (p - d)t.\sigma_t$ = (71 – 23) 13 X 80 = 49 920 N

Shearing resistance of the rivet, $F_s = n.\frac{\pi}{4} d^2 r$ = 2 X $\frac{\pi}{4}$ X 23^2 X 60 = 49 864 N

Crushing resistance of the rivet, $F_c = n\ d\ t\ \sigma_c$ = 2 X 23 X 13 X 120 = 71 760 N

Then, strength of the joint = Least of the above resistances = 49 864 N, therefore **the joint will fail due to shearing of the rivets.** **(Ans)**

Efficiency the joint:

We know that the strength of the unriveted plate, $F = p\ t\ \sigma_t$ = 71 X 23 X 80 = 73 840 N

Then, efficiency of the joint, $\eta = \frac{F_s}{F} = \frac{49\ 864}{73\ 840}$ = **0.675 or 68%** **(Ans)**

Problem No. 04.

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 90 MPa; compressive stress 140 MPa; and shear stress in the rivet 56 MPa.

Solution:

Data Given:

n = 2; D = 1.5 m = 1500 mm; P = 0.95 N/mm²; η_l = 75% = 0.75; $\sigma_t = 90\ MPa = 90\ N/mm^2$; $r = 56\ MPa = 56\ N/mm^2$; $\sigma_c = 140\ MPa = 140\ N/mm^2$

Thickness of boiler plate:

We know that, thickness of boiler plate, $t = \frac{PD}{2\ \sigma_t\, \eta}$ = $\frac{0.95 \times 1500}{2 \times 90 \times 0.75}$ = 10.55 mm **or 12 mm (Ans)**

Diameter of the rivet:

Since the diameter of the plate is greater than 8 mm, therefore diameter of the rivet, $d' = 6.05\sqrt{t}$ = 6.05 X $\sqrt{12}$ = 20.95 mm

From table 10.5, we find that the standard size of the diameter of rivet is 20 mm and the corresponding diameter of the rivet hole, **d = 21 mm (Ans)**

Pitch of rivets:

We know that the pitch of the rivets may be calculated by equating the tearing and shearing resistances.

Tearing resistance of the plate, $F_t = (p - d)t.\sigma_t$ = (p – 21) 12 X 90 = (p – 21) X 1080 N

Shearing resistance of the rivet, $F_s = 1.875 \times n.\frac{\pi}{4} d^2\, r$ = 1.875 X 2 X $\frac{\pi}{4}$ X 21^2 X 56 = 72 745 N (*assuming that the rivets in double shear are 1.875 times stronger than in single shear*)

Then, (p – 21) X 1080 = 72 745

Or, p = 67.35 + 21 = 88.35 or 90mm **(Ans)**

The maximum pitch, $p_{max} = C.t + 41$ = 3.50 X 12 + 41 = 83 mm (taking C = 3.50 mm from ***Table No- 10.7***)

The minimum pitch, $p_{min} = (2.25\ d)$ = 2.25 X 21 = 47.25 mm

Since, the calculated pitch is greater than the maximum pitch, therefore we shall adopt pitch of the rivets, **p = p_{max} = 83 mm (Ans)**

Distance between the rows of rivets:

We know that, for zig-zag riveting, $p_b = 0.33p + 0.67d$ = 0.33 X 83 + 0.67 X 21

= 41.46 mm or **42 mm (Ans)**

Thickness of cover plate:

Taking equal widths for double cover, t_1 =0.625 t = 0.625 X 12 = **7.5 mm (Ans)**

Margin:

Margin, m = 1.5 d = 1.5 X 21 = 31.5 mm or **32 mm (Ans)**

Efficiency of the designed joint:

We know that,

Tearing resistance of the plate, $F_t = (p-d)t.\sigma_t$ = (84 – 21) 12 X 90 = 68 040 N

Shearing resistance of the rivet, $F_s = 1.875 \times n.\frac{\pi}{4} d^2\, r$ = 1.875 X 2 X $\frac{\pi}{4}$ X 21^2 X 56 = 72 745 N

Crushing resistance of the rivet, $F_c = n\ d\ t\ \sigma_c$ = 2 X 21 X 12 X 140 = 70 560 N

Then, strength of the joint = Least of the above resistances = 68 040 N.

Again, the strength of the unriveted plate, $F = p\ t\ \sigma_t$ = 84 X 21 X 90 = 90 720 N

Then, efficiency of the joint, $\eta = \frac{F_t}{F}$ = $\frac{68\ 040}{90\ 720}$ = **0.75 or 75% (Ans)**

Since the efficiency of the designed joint is equal to the given efficiency of 75%, therefore the ***design is satisfactory***.

POSSIBLE SHORT TYPE QUESTION WITH ANSWER:

1. *Define welded joint.*

Ans: A welded joint is a permanent joint which is obtained by the fusion of the edges of the two parts to be joined together, with or without application of pressure and a filler material.

2. *State the advantages of welded joint over riveted joint.*

Ans:

a) The welded structures are usually light in weight compared to riveted structures. This is due to the reason, that in welding, gussets or other connecting components are not used.

b) The welded joints provide high efficiency, which is not possible in the case of riveted joints.

c) Alterations and additions can be made easily in the existing structures.

d) Welded structures are smooth in appearance; therefore, it looks pleasing.

3. *Define strength of riveted joint.*

Ans: Strength of a riveted joint may be defined as the maximum force, which it can transmit, without causing it to fail. In other words, it is the least value of tearing strength, shearing strength and crushing strength of rivet.

4. *Define efficiency of riveted joint.*

Ans: The efficiency of a riveted joint is defined as the ratio of the strength of riveted joint to the strength of the unriveted or solid plate.

Mathematically,

Efficiency of the riveted joint, $\eta = \frac{Least\ of\ F_t, F_s, F_c}{F}$.

5. *State different modes of failure of riveted joint.*

Ans: Different modes of failure are:

1) Tearing of the plate at an edge
2) Tearing of the plate across a row of rivet
3) Shearing of the rivets
4) Crushing of the plate or rivet.

POSSIBLE LONG TYPE QUESTION:

1. *State the differences between permanent and temporary joints.*
2. *State the advantages of welded joint over riveted joint.*
3. *Describe failure of riveted joints with neat sketch.*
4. *State different types of rivets.*
5. *Numerical from design of welded and riveted joints.*

<u>Chapter No.: 03</u>

<u>DESIGN OF SHAFTS & KEYS</u>

<u>Scope of Syllabus</u>:

State function of shafts.

state materials for shaft.

Design of solid and hollow shaft to transmit a given power at given rpm based on strength and rigidity.

State standard size of shaft.

3.10 *Numerical from design of shafts.*

State function of keys, types of keys & material of keys

Describe failure of key, effect of key way

Design rectangular sunk key considering its failure against shear & crushing

Design rectangular sunk key by using empirical relation for given diameter of shaft

State specification of parallel key, gib-head key, taper key as per I.S.

Numerical from design of keys.

FUNCTION OF SHAFT:

A shaft is a rotating machine element which is used to transmit power from one place to another. In order to transfer power from one shaft to another, the various members such as pulleys, gears, etc are mounted on it.

In other words, we may say that a shaft is used for the transmission of twisting moment (torque) and bending moment.

<u>Little Further</u>:

a. Shafts are solid in cross section but, sometimes hollow shafts are also used.
b. An ***axle***, though similar in shape to the shaft, is a stationary machine element and is used for the transmission of bending moment only.
c. A ***spindle*** is a short shaft that imparts motion either to a cutting tool (e.g. drill press spindle) or to a work piece (e.g. lathe spindles).

MATERIALS USED FOR SHAFTS: (*Refer* *Table No.- 5.2*)

The material used for ordinary shafts is carbon steel of grades 40 C 8, 45 C 8, 50 C 4 and 50 C 12. When a shaft of high strength is required, then an alloy steel such as nickel, nickel-chromium or chrome-vanadium steel is used.

DESIGN OF SOLID AND HOLLOW SHAFT TO TRANSMIT A GIVEN POWER AND AT GIVEN RPM BASED ON STRENGTH:

(Refer Design Data Hand Book)

In designing shafts on the basis of strength, the following cases may be considered:

a. Shafts subjected to torque only
b. Shafts subjected to bending moment only
c. Shafts subjected to combined torque and bending moment

Torque transmitted by the shaft:

$T = \frac{6 \times 10^7 \times P}{2 \pi N}$; where *P in kW, N in rpm and T in N-mm*

Shafts subjected to torque only:

Solid shaft:

Diameter, $d = \left[\frac{16\,T}{\pi\,c}\right]^{1/3}$

Hollow shaft:

Outside Diameter, $d_0 = \left[\frac{16\,T}{\pi\,c\,(1-k^4)}\right]^{1/3}$ and $d_i = k\,d_0$

Shafts subjected to bending moment only:

Solid shaft:

Diameter, $d = \left[\frac{32\,M}{\pi\,\sigma_b}\right]^{1/3}$

Hollow shaft:

Outside Diameter, $d_0 = \left[\frac{32\,M}{\pi\,\sigma_b\,(1-k^4)}\right]^{1/3}$ and $d_i = k\,d_0$

Shafts subjected to combined torque and bending moment:

Solid shaft:

Diameter, $d = \left[\frac{32\,M_e}{\pi\,\sigma_b}\right]^{1/3}$ or $d = \left[\frac{16\,T_e}{\pi\,c}\right]^{1/3}$

Where, $T_e = \sqrt{M^2 + T^2}$ and $M_e = \frac{1}{2}[M + \sqrt{M^2 + T^2}]$

Hollow shaft:

Outside Diameter, $d_0 = [\frac{32\, M_e}{\pi\, \sigma_b\, (1-k^4)}]^{1/3}$ or $d_0 = [\frac{16\, T_e}{\pi\, c\, (1-k^4)}]^{1/3}$ and $d_i = k\, d_0$

Where, $T_e = \sqrt{M^2 + T^2}$ and $M_e = \frac{1}{2}[M + \sqrt{M^2 + T^2}]$

STANDARD SIZE OF SHAFT: (Refer Table No.- 5.5)

Diameter:

6, 8, 10, 12, 14, 16, 18, 20, 22, 25, 28, 32, 36, 40, 45, 50, 56, 63, 71, 80, 90, 100, 110, 125, 140, 160, 180, 200, 220, 240, 260, 280, 300, 320, 340, 360, 380, 400, 420, 440, 450, 480, 500, 530, 560, 600.

Length:

5m, 6m and 7m.

3.10 NUMERICAL FROM DESIGN OF SHAFT:

Problem No. 01.

A line shaft rotating at 200 rpm is to transmit 20 kW. The shaft may be assumed to be made of mild steel with an allowable shear stress of 42 MPa. Determine the diameter of the shaft, neglecting the bending moment on the shaft.

Solution:

Data Given:

N = 200 rpm; P = 20 kW; $r = 42\ MPa$ = 42 N/mm^2

We know that, torque transmitted by the shaft

$T = \frac{6 \times 10^7 \times P}{2\, \pi\, N} = \frac{6 \times 10^7 \times 20}{2\, \pi\, 200}$ = 955 X 10^3 N-mm

Then diameter of shaft, $d = [\frac{16\, T}{\pi\, c}]^{1/3} = [\frac{16 \times 955 \times 10^3}{\pi \times 42}]^{1/3}$ = 48.7 mm

From *Table 5.5*, the *standard diameter of shaft* is **50 mm. (Ans)**

Problem No. 02.

A solid shaft is transmitting 1 MW at 240 rpm. Determine the diameter of the shaft if the maximum torque transmitted exceeds the mean torque by 20%. Take the maximum allowable shear stress as 60 MPa.

Solution:

Data Given:

N = 240 rpm; P = 1 MW = 10^3 kW; $r = 60\ MPa$ = 60 N/mm^2; $T_{max} = 1.2\ T_{mean}$

We know that, mean torque transmitted by the shaft

$$T_{mean} = \frac{6 \times 10^7 \times P}{2\,\pi\,N} = \frac{6 \times 10^7 \times 10^3}{2\,\pi\,240} = 39784 \text{ X } 10^3 \text{ N-mm}$$

So, maximum torque transmitted,

$T_{max} = 1.2\ T_{mean} = 1.2 \text{ X } 39784 \text{ X } 10^3 = 47741 \text{ X } 10^3$ N-mm

Therefore, diameter of the shaft

$$d = \left[\frac{16\ T_{max}}{\pi\ c}\right]^{1/3} = \left[\frac{16 \times 47741 \times 10^3}{\pi \times 60}\right]^{1/3} = 159.4 \text{ mm}$$

From *Table 5.5*, the *standard diameter of shaft* is **160 mm. (Ans)**

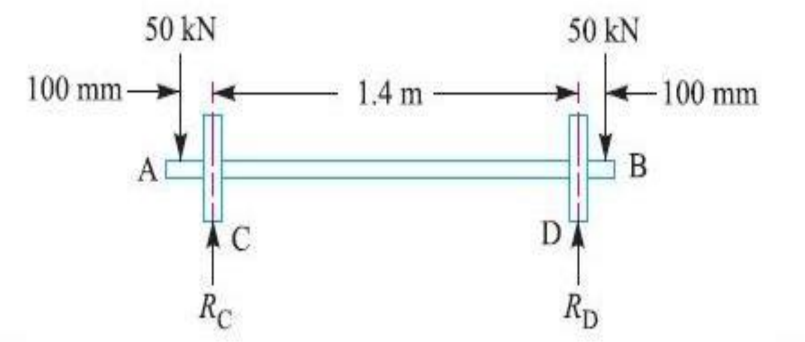
Problem No. 03.

A pair of wheels of a railway wagon carries a load of 50 kN on each axle box, acting at a distance of 100 mm outside the wheel base. The gauge the rails is 1.4 m. find the diameter of the axle between the wheels, if the stress is not to exceed 100 MPa.

Solution:

Data Given:

W = 50 kN = 50 X 10^3 N; L =100 mm; x = 1.4 m; σ_b = 100 MPa = 100 N/mm^2



A little consideration will show that the maximum bending moment acts on the wheel at C and D. so the maximum bending moment, M = W.L = 50 X 10^3 X 100 = 5 X 10^6 N-mm.

Therefore, diameter of the shaft

$$d = \left[\frac{32\ M}{\pi\ \sigma_b}\right]^{1/3} = \left[\frac{32 \times 5 \times 10^6}{\pi\ 100}\right]^{1/3} = 79.8 \text{ mm}$$

From *Table 5.5*, the *standard diameter of shaft* is **80 mm. (Ans)**

Problem No. 04.

A solid circular shaft is subjected to a bending moment of 3000 N-m and a torque of 10 000 N-m. the shaft is made up of 45 C 8 steel having ultimate tensile stress of 700 MPa and an ultimate shear stress of 500 MPa. Assuming a factor of safety as 6, determine the diameter of the shaft.

Solution:

Data Given:

M = 3 000 N-m = 3 X 10^6 N-mm; T =10 000 N-m = 10 X 10^6 N-mm; σ_{bu}= 700 MPa = 700 N/mm^2; r_u = 500 MPa = 500 N/mm^2; f_s= 6

We know that, $\sigma_b = \frac{\sigma_{bu}}{f_s} = \frac{700}{6}$ = 116.67 N/mm^2

And, $r = \frac{c_u}{f_s} = \frac{500}{6}$ = 83.33 N/mm^2

Considering T_e:

$T_e = \sqrt{M^2 + T^2} = \sqrt{(3 \times 10^6)^2 + (10 \times 10^6)^2}$ = 10.44 X 10^6 N-mm

$d = \left[\frac{16\,T_e}{\pi\, c}\right]^{1/3} = \left[\frac{16 \times 10.44 \times 10^6}{\pi \times 83.33}\right]^{1/3}$ = 86 mm

Considering M_e:

$M_e = \frac{1}{2}[M + \sqrt{M^2 + T^2}] = \frac{1}{2}[3 \times 10^6 + 10.44 \times 10^6]$ = 6.72 X 10^6 N-mm

$d = \left[\frac{32\,M_e}{\pi\,\sigma_b}\right]^{1/3} = \left[\frac{32 \times 6.72 \times 10^6}{\pi \times 116.7}\right]^{1/3}$ = 83.7 mm

Taking the larger of the two values, we have d = 86 mm

From *Table 5.5*, the *standard diameter of shaft* is **90 mm.** **(Ans)**

3.3 DESIGN OF SOLID AND HOLLOW SHAFT TO TRANSMIT A GIVEN POWER AND AT GIVEN RPM BASED ON RIGIDITY: (Refer Design Data Hand Book)

Equation of Torsional Rigidity: $\frac{T}{J} = \frac{G\,\theta}{l}$

or J $= \frac{T\,l}{G\,\theta}$

Where, T = Twisting moment or torque in N-mm

J = Polar moment of Inertia in mm^4

$= \frac{\pi}{32} d^4$ (for solid shaft)

$= \frac{\pi}{32} (d_o^4 - d_i^4)$ (for hollow shaft)

G = Modulus of Rigidity or Shear Modulus in N/mm^2

θ = Angle of twist in radians or torsional deflection

l = Length of shaft in mm

Problem No. 05.

A steel spindle transmits 4 kW at 800 rpm. The angular deflection should not exceed 0.25^0 per meter of the spindle. If the modulus of rigidity for the material of the spindle is 84 GPa, find the diameter of the spindle and the shear stress induced in the spindle.

Solution:

Data Given:

P = 4 kW; N = 800 rpm; θ = 0.25^0 = 0.25 X π / 180 = 0.0044 rad; l = 1 m = 1000 mm;

G = 84 GPa = 84 X 10^3 N/mm^2

We know that, torque transmitted by the shaft

$$T = \frac{6 \times 10^7 \times P}{2\,\pi\, N} = \frac{6 \times 10^7 \times 4}{2\,\pi\, 800} = 47\ 740 \text{ N-mm}$$

We also know that,

$$J = \frac{T\, l}{G\, \theta}$$

Or, $\frac{\pi}{32} d^4 = \frac{T\, l}{G\, \theta} = \frac{47\ 740 \times 1000}{84 \times 10^3 \times 0.0044} = 129\ 167$

Or, $d^4 = 129\ 167 \text{ X } 32/\pi = 1.3 \text{ X } 10^6$

Or, d = 33.87 mm

From *Table 5.5*, the *standard diameter of shaft* is **36 mm.** **(Ans)**

FUNCTION OF KEYS, TYPES OF KEYS & MATERIAL OF KEY:

Function of Key:

A key is a piece of mild steel inserted between the shaft and hub of the pulley to connect these together in order to prevent relative motion between them.

Key is a machine element which is used to connect the transmission shaft to rotating machine elements like pulley, gear, sprocket or flywheel.

Keys are always used as temporary fastenings and are subjected to crushing and shearing stresses.

Types of Key: (Ref. – Data Book)

Following types of keys are important from the subject point of view:

1. Sunk key (Rectangular/ square sunk key, parallel sunk key, gib head key, Feather key, Woodruff key)
2. Saddle key (Flat/ Hollow saddle key)
3. Tangent key
4. Round key
5. Splines

Material of Key:

The material used for key should be same as that of shaft. (already taught in ***Article No. 3.2***)

Keys are made of ductile materials. Commonly used materials for a key are hardened and tempered steel of grades C30, C35, C40, C50 and 55Mn75 etc. Brass and stainless keys are used in corrosive environment. Factor of safety of 3 to 4 is generally taken on yield strength.

FAILURE OF KEY AND EFFECT OF KEY WAY:

Failure of Key: (Ref. Design Data Hand Book)

Torque transmitted by the sunk key,

$T = l\,\mathrm{w}\,r\,d/2$ (Considering shear failure)

$T = l\,t\,\sigma_c\,d/4$ (Considering crushing failure)

Effect of Key-way:

The key way cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corners of the key way and reduction in the cross-sectional area of the shaft.

In other words, the torsional strength of the shaft is reduced.

The following relation for the weakening factor due to key way on shafts (***Ref. Data Book Equation No 5.23***)

$$C = 1.0 - 0.2\left(\frac{\mathrm{w}}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

DESIGN OF RECTANGULAR SUNK CONSIDERING SHEAR & CRUSHING FAILURE:

Problem No. 06.

Design a rectangular sunk key for a shaft of 50 mm diameter. The shearing and crushing stresses for the key material are 42 MPa and 70 MPa.

Solution:

Data Given:

d = 50 mm; r = 42 MPa = 42 N/mm^2; σ_c = 70 MPa = 70 N/mm^2

From Data Book (table No 6.1), we find that for a shaft of 50 mm diameter,

Width of key, w = 16 mm and thickness of key, t = 10 mm **(Ans)**

Considering shearing of the key:

We know that, torque transmitted by the shaft, $T = l\ w\ r\ d/2$

Or, $\frac{\pi}{16} r d^3 = lwrd/2$

Or, $\frac{\pi}{16} \times 42 \times 50^3 = l \times 16 \times 42 \times 50/2$

Or, 1.03 × 10^6 = $l \times 16{,}800$

Or, l =1.03 × 10^6/16,800 = 61.31 mm

Considering crushing of the key:

We know that, torque transmitted by the shaft, $T = l\ \frac{t}{2}\ \sigma_c\ d/2$

Or, $\frac{\pi}{16} r d^3 = l\ \frac{t}{2}\ \sigma_c\ d/2$

Or, $\frac{\pi}{16} \times 42 \times 50^3 = l \times \frac{10}{2} \times 70 \times 50/2$

Or, 1.03 × 10^6 = $l \times 8750$

Or, l = 1.03 × 10^6/8750 = 117.7 mm

Taking larger of the two values, we have length of the key,

$l = 117.7 \cong 120\ mm$ **(Ans)**

DESIGN OF RECTANGULAR SUNK KEY BY USING EMPIRICAL RELATION FOR GIVEN DIAMETER OF SHAFT: (Ref. Data Book)

Empirical relations of rectangular sunk key:

Width of the key	w = d/4
Thickness of the key	t = 2w/3 = d/6
Length of the key	l = 1.6 d

Problem No. 07.

Design a rectangular sunk key for a shaft of 50 mm diameter, using Empirical relation.

Solution:

Data Given: d = 50 mm

Empirical relations of rectangular sunk key:

Width of the key	w = d/4 = 50/4 = 12.5 = 14 mm
Thickness of the key	t = 2w/3 = d/6 =50/6 = 8.33 = 9 mm
Length of the key	l = 1.6 d = 1.6 × 50 = 80 mm

STATE SPECIFICATIONS OF PARALLEL KEY, GIB-HEAD KEY,TAPER KEY AS PER I.S.: (Ref. Data Book)

Parallel Key	Width of key, w = 0.25 d + 2 mm Thickness of key, t = 0.66 w
Gib–head Key	Width of key, w = 0.25 d + 2 mm Thickness of key, t = 0.66 w Standard Taper = 1 : 100 Height of gib-head, h = 1.75 t Width of gib-head, W = 1.5 t
Taper Key	Width of key, w = 0.25 d + 2 mm Thickness of key, t = 0.66 w Standard Taper = 1 : 100

POSSIBLE SHORT TYPE QUESTION WITH ANSWER:

1. ***State function of shaft.***

Ans: A shaft is a rotating machine element which is used to transmit power from one place to another. In order to transfer power from one shaft to another, the various members such as pulleys, gears, etc are mounted on it.

In other words, we may say that a shaft is used for the transmission of twisting moment (torque) and bending moment.

2. ***Differentiate between shaft and axle.***

Ans: A shaft is used for the transmission of twisting moment (torque) and bending moment. Whereas an ***axle***, though similar in shape to the shaft, is a stationary machine element and is used for the transmission of bending moment only.

3. ***State the materials used for the shaft.***

Ans: The material used for ordinary shafts is carbon steel of grades 40 C 8, 45 C 8, 50 C 4 and 50 C 12. When a shaft of high strength is required, then an alloy steel such as nickel, nickel-chromium or chrome-vanadium steel is used.

4. ***State the function of key.***

Ans: A key is a piece of mild steel inserted between the shaft and hub of the pulley to connect these together in order to prevent relative motion between them.

Keys are always used as temporary fastenings and are subjected to crushing and shearing stresses.

5. ***Define effect of key-way.***

Ans: The key way cut into the shaft reduces the load carrying capacity of the shaft. This is due to the stress concentration near the corners of the key way and reduction in the cross-sectional area of the shaft.

In other words, the torsional strength of the shaft is reduced.

6. ***What do you understand by splines*?**

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

POSSIBLE LONG TYPE QUESTION:

1. *Numerical from Design of shaft.*
2. *Numerical from Design of key.*

Chapter No.: 04

DESIGN OF COUPLING

Scope of Syllabus:

Design of Shaft Coupling

Requirements of a good shaft coupling

Types of Coupling.

Design of Sleeve or Muff-Coupling.

Design of Clamp or Compression Coupling.

Solve simple numerical on above.

SHAFT COUPLING:

Shafts are usually available up to 7 metres 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.

Little Further:

A coupling is termed as a device used to make permanent or semi-permanent connection where as a clutch permits rapid connection or disconnection at the will of the operator.

REQUIREMENTS OF A GOOD SHAFT COUPLING:

A good shaft coupling should have the following requirements:

a. It should be easy to connect or disconnect.
b. It should transmit the full power from one shaft to the other shaft without losses.
c. It should hold the shafts in perfect alignment.
d. It should reduce the transmission of shock loads from one shaft to another shaft.
e. It should have no projecting parts.

TYPES OF COUPLING: (Refer Data Book)

Shaft couplings are divided into two main groups as follows:

1. Rigid coupling
2. Flexible coupling

Rigid Couplings

Rigid Couplings are used to connect two shafts which are perfectly aligned. These are simple and inexpensive.

Rigid Couplings are of following types:

1. Sleeve or Muff Coupling
2. Clamp or Split-muff or Compression Coupling
3. Flange Coupling

Flexible Couplings

Flexible couplings are used to connect two shafts having lateral or angular misalignment. Flexible elements provided in flexible coupling absorb shocks and vibrations.

Flexible Couplings are of following types:

1. Bushed pin type Coupling
2. Universal Coupling
3. Oldham Coupling

DESIGN OF SLEEVE OR MUFF COUPLING: (Refer Data Book)

It is the simplest type of rigid coupling, made of CI. 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.

Ref. Fig No. 7.1 and approximate proportions in page no. 7.2 & 7.3.

Problem No. 01.

Design a muff coupling which is used to connect two steel shafts transmitting 40 kW at 350 rpm. The material for the shafts and the key is plain carbon steel for which allowable shear and crushing stresses may be taken as 40 MPa and 80 MPa respectively. The material for the muff is cast iron for which the allowable shear stress may be assumed as 15 MPa.

Solution:

Data Given:

P = 40 kW; N = 350 rpm; r_1= 40 N/mm^2; σ_{c1} = 80 N/mm^2; r_2 = 15 N/mm^2

1. ***Design for shaft:***

We know that, torque transmitted by the shaft

$T = \frac{60 \times 10^6 \times P}{2\,\pi\,N} = \frac{60 \times 10^6 \times 40}{2\,\pi\,350} = 1100 \text{ X } 10^3$ N-mm

Then diameter of shaft, $d = [\frac{16\,T}{\pi\,c_1}]^{1/3} = [\frac{16 \times 1100 \times 10^3}{\pi \times 40}]^{1/3}$ = 52 mm

From *Table 5.5*, the *standard diameter of shaft* is **56 mm. (Ans)**

2. Design for sleeve:

Outer diameter of muff or sleeve, D = 2d = 2 × 56 = **112 mm (Ans)**

Length of muff, L = 3d = 3 × 56 = **168 mm (Ans)**

Checking the induced stress:

We know that, torque transmitted by the coupling, $T = \frac{\pi}{16}\, r_2\, D^3\,(1 - k^4)$

Or, 1100 X $10^3 = \frac{\pi}{16} \times r_2 \times 112^3\,(1 - 0.5^4) = 258616\, r_2$

Or, r_2 = 258616/1100 X 10^3 = 0.235 MPa

Since the induced shear stress in the muff is less than the permissible shear stress of 15 MPa, therefore the *design of muff is safe*.

3. Design for key:

Width of the key, w = d/4 = 56/4 = **14 mm (Ans)**

Thickness of the key, t = w = **14 mm (Ans)** (since crushing stress = twice of shear stress)

Length of key, l = 1.5 d = 1.5 X 56 = **84 mm (Ans)**

No. of keys, n = **2 (Ans)**

Checking the induced shear and crushing stresses in the key:

Considering shearing of the key:

We know that torque transmitted by the shaft,

T = $l\, w\, r_1\, d/2$

Or, 1100 X 10^3 = 84 X 14 X r_1 X 28

Or, r_1 = 33.4 N/mm^2

Considering crushing of the key:

We know that torque transmitted by the shaft,

T = $l\ t\ \sigma_{c1}\ d/4$

Or, 1100 X 10^3 = 84 X 14 X σ_{c1} X 14

Or, σ_{c1} = 66.8 N/mm^2

Since the induced shear and crushing stresses are less than the permissible stresses, therefore the ***design of key is safe***.

DESIGN OF CLAMP OR SPLIT MUFF COUPLING OR COMPRESSION COUPLING:
(Refer Design Data Hand Book)

In this case, the muff or sleeve is made into two halves and are bolted together. This coupling may be used for heavy duty and moderate speeds.

Ref. Fig No. 7.2 and approximate proportions in page no. 7.4.

Problem No. 02.

Design a clamp coupling to transmit 30 kW at 100 rpm. The allowable shear stress for the shaft and key is 40 MPa and the number of bolts connecting the two halves are six. The permissible tensile stress for the bolts is 70 MPa. The co-efficient of friction between the muff and the shaft surface may be taken as 0.3.

Solution:

Data Given:

P = 30 kW; N = 100 rpm; r_1= 40 N/mm^2; n = 6; σ_{t1} = 70 N/mm^2; μ = 0.3

1. ***Design for shaft:***

We know that, torque transmitted by the shaft

$T = \frac{60 \times 10^6 \times P}{2\,\pi\, N} = \frac{60 \times 10^6 \times 30}{2\,\pi\, 100} =$ 2865 X 10^3 N-mm

Then diameter of shaft, $d = [\frac{16\ T}{\pi\ c_1}]^{1/3} = [\frac{16 \times 2865 \times 10^3}{\pi \times 40}]^{1/3}$ = 71.4 mm

From *Table 5.5*, the *standard diameter of shaft* is **80 mm. (Ans)**

2. ***Design for muff:***

Outer diameter of muff, D = 2.5 d = 2.5 × 80 = **200 mm (Ans)**

Length of muff, L = 3.5 d = 3.5 × 80 = **280 mm** **(Ans)**

3. *Design for key:*

From ***Table 6.1***, the width and thickness of the key for a shaft diameter of 80 mm, ***w = 22 mm*** and ***t = 14 mm*** **(Ans)**

Length of key, l = L = 3.5 d = **280 mm** **(Ans)**

No. of keys, n' = **1** **(Ans)**

4. *Design of bolts:*

Torque transmitted by shaft, T = $\frac{\pi^2}{16}\,\mu\, d_c^2\, \sigma_{t1}\, n\, d$

Or, 2865 X 10^3 = $\frac{\pi^2}{16} \times 0.3 \times d_c^2 \times 70 \times 6 \times 80$

Or, d_c = 22.2 mm

From ***Table 9.9***, we find that the standard core diameter of the bolt for coarse series is *25.706 mm* and the nominal diameter of the bolt is ***30 mm*** (***M30***) **(Ans).**

POSSIBLE SHORT TYPE QUESTION WITH ANSWER:

1. *Define necessity of shaft coupling.*

Ans:
(i) To provide for misalignment of the shafts or to introduce mechanical flexibility.
(ii) To reduce the transmission of shock loads from one shaft to another.

2. *State the requirements of a good shaft coupling.*

Ans: A good shaft coupling should have the following requirements:

(i) It should be easy to connect or disconnect.

(ii) It should transmit the full power from one shaft to the other shaft without losses.

3. *Classify shaft coupling.*

Ans:
Shafts couplings are divided into two main groups: (a) Rigid coupling and (b) Flexible coupling.

POSSIBLE LONG TYPE QUESTION:

1. *Numerical from Design of sleeve or muff coupling.*
2. *Numerical from Design of clamp or compression coupling.*

Chapter No.: 05

DESIGN OF CLOSED COIL HELICAL SPRING

Scope of Syllabus:

Materials used for helical spring

Standard size spring wire. (SWG)

Terms used in compression spring.

Stress in helical spring of a circular wire.

Deflection of helical spring of circular wire.

Surge in spring.

Simple Numerical

SPRING:

A spring may be defined as an elastic body, whose function is to distort when loaded and to recover its original shape when the load is removed.

***Applications*:**

1. *To absorb or control energy due to either shock or vibration as in car spring, etc.*
2. *To apply forces as in brakes, clutches, etc.*
3. *To measure forces as in spring balances.*
4. *To store energy as in watches, toys, etc.*
5. *To control motion by maintaining contact between two elements as in cams and followers.*

5.1 MATERIALS USED FOR HELICAL SPRING: (Refer Data Book)

Refer **Table No. 13.2.**

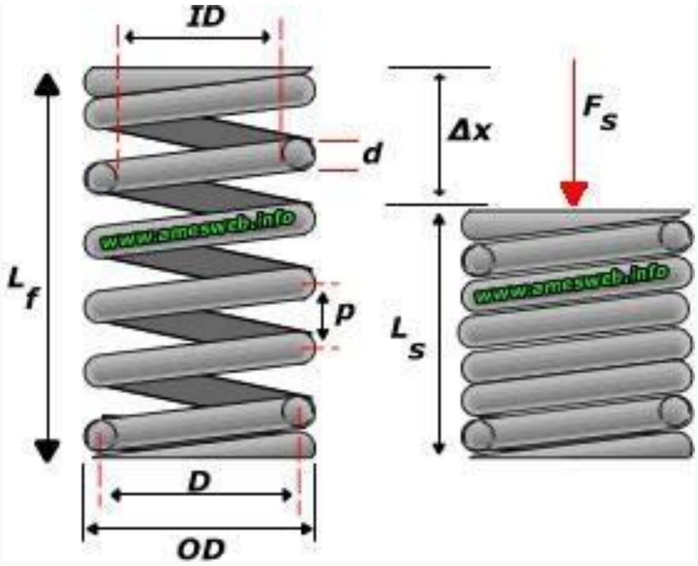
5.2 STANDARD SIZE SPRING WIRE (SWG): (Refer Data Book)

Refer **Table No. 13.5.**

5.3 TERMS USED IN COMPRESSION SPRING:

1. ***Solid length***: Length of the spring when it is compressed so that the coils come in contact with each other.

Mathematically, $L_s = N\ d$.



5. ***Pitch of the Coil***: Axial distance between adjacent coils of the spring in uncompressed state.

Mathematically, $p = \frac{L_f}{N-1}$

5.4 STRESS IN HELICAL SPRING OF CIRCULAR WIRE: (Refer Data Book)

Ref. Equation No. 13.1.

5.5 DEFLECTION OF HELICAL SPRING OF CIRCULAR WIRE: (Refer Data Book)

Ref. Equation No. 13.2.

SURGE IN SPRING:

When the natural frequency of vibrations of the spring coincides with the frequency of external periodic force, which acts on it, resonance occurs. In this state, the spring is subjected to a wave of successive compressions of coils that travels from one end to the other and back. This type of vibratory motion is called ‘surge’ of spring.

The surge in springs may be eliminated by using the following methods:

a. *The spring is provided with friction dampers on central coils. This prevents propagation of surge wave.*
b. *By using springs of high natural frequency.*

Problem No. 01.

A compression coil spring made of an alloy steel is having the following specifications:

Mean diameter of the coil = 50 mm; wire diameter = 5 mm; number of active coils = 20.

If this spring is subjected to an axial load of 500 N; calculate the maximum shear stress to which the spring material is subjected.

Solution:

Data Given:

D = 50 mm; d = 5 mm; n = 20; P = 500 N

We know that the spring index, C = D/d = 50/5 = 10

Wahl's shear stress factor, $K = \frac{4C-1}{4C+1} + \frac{0.615}{C}$

Or, $K = \frac{4\times10-1}{4\times10+1} + \frac{0.615}{10}$ = 0.9512 + 0.0615 = 1.0127

Maximum shear stress induced in the spring, $r = \frac{8\,K\,P\,C}{\pi\,d^2}$

Or, $r = \frac{8\times1.017\times500\times10}{\pi\times 5^2}$ = ***517.75 mm* (Ans)**

POSSIBLE SHORT TYPE QUESTION WITH ANSWER:

1. ***State the applications of spring.***

Ans:

→ *To absorb or control energy due to either shock or vibration as in car spring, etc.*
→ *To apply forces as in brakes, clutches, etc.*
→ *To measure forces as in spring balances.*
→ *To store energy as in watches, toys, etc.*

2. ***Define solid length of spring.***

Ans:

Length of the spring when it is compressed so that the coils come in contact with each other.

Mathematically, $L_s = N\,d$.

3. ***Define Spring constant.***

Ans:

Force required to produce unit deflection in the spring.

Mathematically, $k = \frac{P}{\delta} = \frac{G\,d}{8\,C^3\,n}$

POSSIBLE LONG TYPE QUESTION:

1. *Numerical from Design of closed coil helical spring.*
2. *Write short notes on surge in spring.*
