GOVT. POLYTECHNIC, ANGUL

LECTURE NOTES WITH QUESTION BANK

(Mechanical Department)

COURSE: -DESIGN OF MACHINE ELEMENTS

(ASSESMENT YEAR-2021-22)

CODE- Th-2

Semester-5th

Prepared By:

Mrs. L. Swain

Lect. (Mechanical Engg.)

Govt. Polytechnic, Angul

Instructional Objectives

At the end of this lesson, the students should have the knowledge of

- Basic concept of design in general.
- Concept of machine design and their types.
- Factors to be considered in machine design.

1.1.1 Introduction

Design is essentially a decision-making process. If we have a problem, we need to design a solution. In other words, to design is to formulate a plan to satisfy a particular need and to create something with a physical reality. Consider for an example, design of a chair. A number of factors need be considered first:

(a) The purpose for which the chair is to be designed such as whether it is to be used as an easy chair, an office chair or to accompany a dining table.

(b) Whether the chair is to be designed for a grown up person or a child.

(c) Material for the chair, its strength and cost need to be determined.

(d) Finally, the aesthetics of the designed chair.

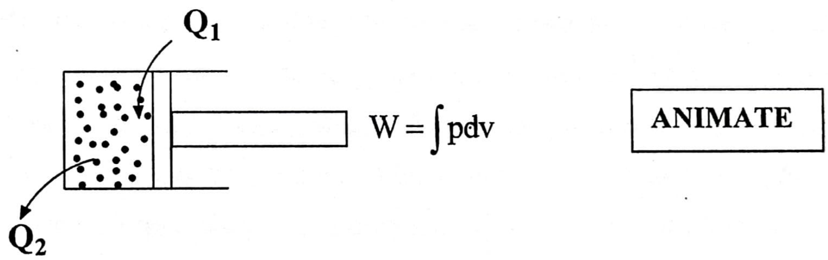
Almost everyone is involved in design, in one way or the other, in our daily lives because problems are posed and they need to be solved.

1.1.2 Basic concept of machine design

Decision making comes in every stage of design. Consider two cars of different makes. They may both be reasonable cars and serve the same purpose but the designs are different. The designers consider different factors and come to certain conclusions leading to an optimum design. Market survey gives an indication of what people want. Existing norms play an important role. Once a critical decision is made, the rest of the design features follow. For example, once we decide the engine capacity, the shape and size, then the subsequent course of the design would follow. A bad decision leads to a bad design and a bad product.

Design may be for different products and with the present specialization and knowledge bank, we have a long list of design disciplines e.g. ship design, building design, process design, bridge design, clothing or fashion design and so on.

Here we are concerned with machine design. We now define a machine as a combination of resisting bodies with successfully constrained relative motions which is used to transform other forms of energy into mechanical energy or transmit and modify available energy to do some useful work. If it converts heat into mechanical energy we then call it a heat engine. This is illustrated in figure-1.1.2.1.



***1.1.2.1**A- Conversion of heat to mechanical energy in a piston cylinder arrangement.*

In many cases however, the machines receive mechanical energy and modify it so that a specific task is carried out, for example a hoist, a bicycle or a hand-winch.

This modification or transformation of energy requires a number of machine elements, some small and some large. Machine design involves primarily designing these elements so that they may transmit the forces safely and perform their task successfully. Consider the following simple mechanisms:

(a) Hand winch (b) Small press operated by a power screw..

In each one of these mechanisms some useful work is being obtained with certain combinations of a number of machine parts. Designing these mechanisms would involve firstly designing these elements and then assembling them in order.

CLIPPING

1.1.2.1V *Introduction to machine design*

1.1.3 Types of design

There may be several types of design such as

Adaptive design

This is based on existing design, for example, standard products or systems adopted for a new application. Conveyor belts, control system of machines and mechanisms or haulage systems are some of the examples where existing design systems are adapted for a particular use.

Developmental design

Here we start with an existing design but finally a modified design is obtained. A new model of a car is a typical example of a developmental design .

New design

This type of design is an entirely new one but based on existing scientific principles. No scientific invention is involved but requires creative thinking to solve a problem. Examples of this type of design may include designing a small vehicle for transportation of men and material on board a ship or in a desert. Some research activity may be necessary.

1.1.4 Types of design based on methods

Rational design

This is based on determining the stresses and strains of components and thereby deciding their dimensions.

Empirical design

This is based on empirical formulae which in turn is based on experience and experiments. For example, when we tighten a nut on a bolt the force exerted or the stresses induced cannot be determined exactly but experience shows that the tightening force may be given by P=284d where, d is the bolt diameter in mm and P is the applied force in kg. There is no mathematical backing of this equation but it is based on observations and experience.

Industrial design

These are based on industrial considerations and norms viz. market survey, external look, production facilities, low cost, use of existing standard products.

1.1.5 Factors to be considered in machine design

There are many factors to be considered while attacking a design problem. In many cases these are a common sense approach to solving a problem. Some of these factors are as follows:

(a) What device or mechanism to be used? This would decide the relative arrangement of the constituent elements.

(b) Material

(c) Forces on the elements

(d) Size, shape and space requirements. The final weight of the product is also a major concern.

(e) The method of manufacturing the components and their assembly.

(f) How will it operate?

(g) Reliability and safety aspects

(h) Inspectibility

(i) Maintenance, cost and aesthetics of the designed product.

What device or mechanism to be used- This is best judged by understanding the problem thoroughly. Sometimes a particular function can be achieved by a number of means or by using different mechanisms and the designer has to decide which one is most effective under the circumstances. A rough design or layout diagram may be made to crystallize the thoughts regarding the relative arrangement of the elements.

Material- This is a very important aspect of any design. A wrong choice of material may lead to failure, over or undersized product or expensive items. The choice of materials is thus dependent on suitable properties of the material for each component, their suitability of fabrication or manufacture and the cost.

Load- The external loads cause internal stresses in the elements and these stresses must be determined accurately since these will be used in determining the component size. Loading may be due to:

i) Energy transmission by a machine member.
ii) Dead weight.
iii) Inertial forces.
iv) Thermal effects.
v) Frictional forces.

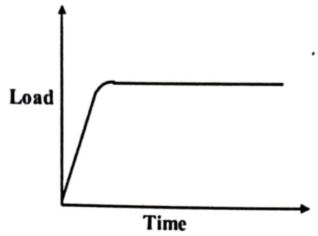
In other ways loads may be classified as:

i) Static load- Does not change in magnitude and direction and normally increases gradually to a steady value.

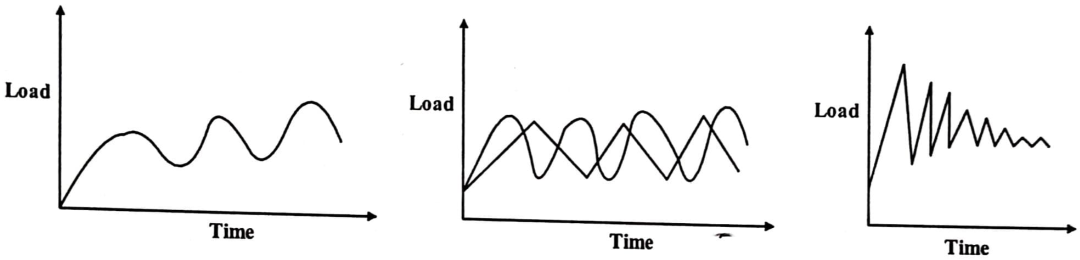
ii) Dynamic load- a) changes in magnitude- for e.g. traffic of varying weight passing a bridge.

b) changes in direction- for e.g. load on piston rod of a double acting cylinder.

The nature of these loads are shown in figure-1.1.5.1.



Static load



Dynamic Loading

1.1.5.1F *The nature of static and dynamic load*

Vibration and shock loading are types of dynamic loading.

Size, shape, space requirements and weight- Preliminary analysis would give an approximate size but if a standard element is to be chosen, the next larger size must be taken. Shapes of standard elements are known but for non-standard element, shapes and space requirements must depend on available space in a particular machine assembly. A scale layout drawing is often useful to arrive at an initial shape and size. Weight is important depending on application. For example, an aircraft must always be made light. This means that the material chosen must have the required strength yet it must be light. Similar arguments apply to choice of material for ships and there too light materials are to be chosen. Portable equipment must be made light.

Manufacture

Care must always be taken to ensure that the designed elements may be manufactured with ease, within the available facilities and at low cost.

How will it operate

In the final stage of the design a designer must ensure that the machine may be operated with ease. In many power operated machines it is simply a matter of pressing a knob or switch to start the machine. However in many other cases, a sequence of operations is to be specified. This sequence must not be complicated and the operations should not require excessive force. Consider the starting, accelerating and stopping a scooter or a car. With time tested design considerations, the sequences have been made user-friendly and as in any other product, these products too go through continuous innovation and development.

Reliability and safety

Reliability is an important factor in any design. A designed machine should work effectively and reliably. The probability that an element or a machine will not fail in use is called reliability. Reliability lies between $0 \leq R < 1$. To ensure this, every detail should be examined. Possible overloading, wear of elements, excessive heat generation and other such detrimental factors must be avoided. There is no single answer for this but an overall safe design approach and care at every stage of design would result in a reliable machine.

Safety has become a matter of paramount importance these days in design. Machines must be designed to serve mankind, not to harm it. Industrial regulations ensure that the manufacturer is liable for any damage or harm arising out of a defective product. Use of a factor of safety only in design does not ensure its overall reliability.

Maintenance, cost and aesthetics

Maintenance and safety are often interlinked. Good maintenance ensures good running condition of machinery. Often a regular maintenance schedule is maintained and a thorough check up of moving and loaded parts is carried out to avoid catastrophic failures. Low friction and wear is maintained by proper lubrication. This is a major aspect of design since wherever there are moving parts, friction and wear are inevitable. High friction leads to increased loss of energy. Wear of machine parts leads to loss of material and premature failure.

Cost and aesthetics are essential considerations for product design. Cost is essentially related to the choice of materials which in turn depends on the stresses developed in a given condition. Although in many cases aesthetic considerations are not essential aspects of machine design, ergonomic aspects must be taken into considerations.

1.1.6Problems with Answers

Q.1: Define machine design.

A.1: A machine is a combination of several machine elements arranged to work together as a whole to accomplish specific purposes. Machine design involves designing the elements and arranging them optimally to obtain some useful work.

Q.2: What is an adaptive design?

A.2: Adaptive design is based on an existing design adapted for a new system or application, for example, design of a new model of passenger car.

Q.3: Suggest briefly the steps to be followed by a designer.

A.3: Machine design requires a thorough knowledge of engineering science in its totality along with a clear decision making capability. Every designer follows his own methodology based on experience and analysis. However, the main steps to

be followed in general are :

- Define the problem.
- Make preliminary design decisions.

- Make design sketches.
- Carry out design analysis and optimization.
- Design the elements for strength and durability.
- Prepare documentations to be followed for manufacture.

Q.4: Discuss 'factor of safety ' in view of the reliability in machine design.

A.4: Reliability of a designed machine is concerned with the proper functioning of the elements and the machine as a whole so that the machine does not fail in use within its designed life. There is no single answer to this and an overall safe design approach at every stage of the design is needed. Use of factor of safety in designing the elements is to optimize the design to avoid over-design for reliability.

1.1.7Summary of this Lesson

The lesson essentially discusses the basic concept of design in general leading to the concept of machine design which involves primarily designing the elements. Different types of design and the factors to be considered have been discussed in detail.

UNIT - I
MACHINE DESIGN

INTRODUCTION :-

→ Design is basically a decision making process i.e. if we have a problem then we need a solution.

→ In other word, Design is a particular machine element, it should satisfy some sort of condition to term them physically reality.

Ex:- If we design a chair a number of factors is to be considered for design.

(i) The purpose of the chair, whether it is used as an easy chair, office chair or for used to accompanied dinning table.

(ii) Material of the chair, it strength and cost to be taken into consideration.

(iii) Whether the chair is design for old generation people, young generation people or for child.

(iv) Aesthetics (Beautiful design of component) of the design chair.

BASIC CONCEPTS OF MACHINE DESIGN :-

→ As design is a decision making process at every state we require a specific design.

→ Different design gives separate components basing on specific procedure of design.

→ Now we are consent with machine design so we have to design each part of the machine element in such a way that they combinely from a successful design.

CLASSIFICATION OF DESIGN :-

→ Machine design is classified into the following types :-

(I) Adaptive design

(II) Developmental design

(III) New design

(iv) Rational design

(v) Empirical design

(VI) Industrial design

(1) ADAPTIVE DESIGN :-

→ The design is based on the existing design.
Ex:- standard production or system adapted it from a new design.

→ In this type of design there is a minimum change in the existing design.

Ex:- Conveyor belt control system of machine.

→ Mechanisms used for lift elevator etc.

(2) DEVELOPMENTAL DESIGN :-

→ Here, we start with an existing design, then after applying certain method a modified design is often.
Ex:- New model of a car is typical example of developmetal design.

(3) NEW DESIGN :-

→ This type of design is new but based on existing scientifical principle.

Ex:- Design of small vehicle for transportation.

→ In this type of design some research activity needed.

(4) RATIONAL DESIGN :-

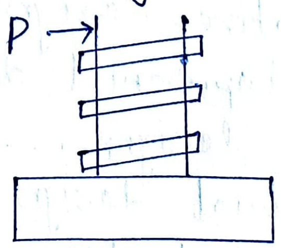
→ This is based on determining stress and strain of a particular component in such a way that the dimension can found out from these formula.

(5) EMPIRICAL DESIGN :-

→ This is based on empirical formula which is term based on experience and experiment.

Ex:- If we tightened a nut on the bolt the force exerted or the stresses induced can not be determine exactly.

→ But the experience shows the tightening force in the nut bolt assembly.



$P = 284\, d$

d = Bolt dia

P = tightening force

→ There is no mathematical basing of this equation but it is based on observation and experience.

(6) INDUSTRIAL DESIGN :-

→ These are based on Industrial observation and consideration such as market survey.

→ External Loop, production facilities, no cost use of existing standing production.

DEFINITION OF MACHINE DESIGN :-

Machine design is defined as a use of scientific principle, technical information and imagination in the description of a machine or a mechanical system to perform specification with maximum economy and efficiency.

* Different engineering material used in design with their uses :-

INTRODUCTION :-

→ Selection of a proper material for machine component is an important steps in the process of machine design.

→ Tensile test is the simplest test to determine various mechanical properties such as tensile & compressive strength, ductility, percentage elongation. This test carried out in a material called as mildsteel.

→ Hence mildsteel is used as an engineering material to design and test certain properties of the material.

→ Similarly cast iron, plain carbon steel, Free cutting steel are used as engineering materials which has certain mechanical properties such as ductility, strength, malleability, toughness, hardness etc. Hence they are used in machine design.

→ Cast iron, plain carbon steel, free carbon steel, alloy steel forms a major group of ferrous material. Hence it is widely used for the design of machine components.

→ The suitability of a steel improved by heat treatment of the steel.
→ There is also another type of material known as non-ferrous material.

NON-FERROUS MATERIAL :-

It includes aluminium alloys, copper alloys etc. and the non-metal group includes ceramics, plastic and FRP (Fibre Reinforced plastic).
→ Selection of this material depends upon following factors,
(i) Availability
(ii) cost
(iii) Mechanical properties
(iv) Manufacturing consideration
→ Materials play an important role not only in design but also in human civilisation because the civilisation it self gives the name, age after using ~~start~~ sort on material i.e. stone age, bronze age, iron age etc.

SOME OF IMPORTANT MATERIALS :-

1. CAST IRON :-

→ Cast iron refers to the family of materials i.e. having iron and carbon is the main composition.
→ By definition cast iron is defined as an alloy of carbon and iron containing more than 2% of carbon.
→ It contains, carbon = 3 - 4 %
silicon = 1 - 3%
Magnesium = 0.1 - 0.3 %
sulphur = upto 1%
phosphorus = upto 1%
iron = Remaining 90%

→ From the design consideration cast iron is used for the following advantages :-

(i) Because it is available in large quantity.

(ii) It is also produced in a mass scale.

(iii) The components made up of cast iron can be given any complex shape.

(iv) Cast iron has high compressive strength.

(v) Cast iron has high toughness i.e. if it is able to damp vibration.

(vi) Cast iron has more resistance to wear.

2. PLAIN CARBON STEEL :-

Depending ~~upon~~ on percentage of carbon plain carbon steel is of 3 types :-

(i) Low carbon steel

(ii) Medium carbon steel

(iii) High carbon steel

(i) LOW CARBON STEEL :-

→ Low carbon steel contains carbon which is less than 0.3% of carbon.

→ Low carbon steel is otherwise known as mild steel or soft steel.

→ Low carbon steel are easily machined and welded.

(ii) MEDIUM CARBON STEEL :-

→ The steel which is having 0.3% - 0.5% of carbon is known as medium carbon steel.

→ It is popularly known as machinery steel.

→ It is stronger and tougher than the low carbon steel.

(iii) HIGH CARBON STEEL :-

→ The high carbon steel contains carbon percentage more than 0.5%. It is also known as hard steel or tool steel.

→ It is readily respond to the heat treatment.

→ It is having high strength combined with hardness.

3. FREE CUTTING STEEL :-

→ The steel of this group contains carbon steel and manganese steel with small percentage of sulphur.

→ Due to addition of sulpher the machinability of the steel improved.

→ Again machinability is defined as the ease with which the component can be machined.

→ The machinability depends upon 3 types :-

(i) the amount of chip formation

(ii) the surface finish

(iii) ability to achieve economical tool life.

MECHANICAL PROPERTIES OF ENGINEERING MATERIAL :-

1. STRENGTH :-

→ Strength is defined as the ability of material to resist the deformation without rupture.

→ Strength is measured by different quantities.

→ Depending on the amount of external load strength is 3 types :-

(i) tensile strength

(ii) compressive strength

(iii) shear strength

(i) TENSILE STRENGTH :-

Tensile strength is defined as the ability of the material to resist external and load causing tensile stress without failure.

ii) COMPRESIVE STRENGTH :-

Compressive strength is defined as the ability of the material to resist the external load causing compressive stress without failure.

iii) SHEAR STRENGTH :-

Shear strength is defined as the ability of the material to resist external load causing shear stress without failure.

2. ELASTICITY :-

Elasticity is defined as the ability of the material to regain it's original shape and size after deformation when external load is removed.

3. PLASTICITY :-

Plasticity is defined as the ability of material to retain the deformation produced under the external load on a permanent basis after removal of external load.

4. STIFFNESS/RIGIDITY :-

It is defined as the ability of material to resist a deformation under the action of external load.

5. RESILLIENCE :-

→ Resillience is defined as the ability of material to absorbed energy when it is deform elastically and to release the energy when it is unloaded.

→ A resillience material absorbes the energy within the elastic limit.

Ex :- Spring limit.

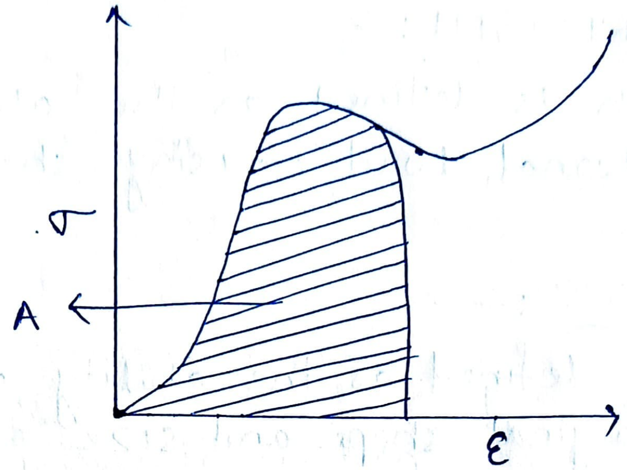
6. TOUGHNESS :-

→ It is defined as the ability of the material to absorb energy up to the fracture limit.

→ This property is essential for the machine component subjected to impact load. Ex :- Charpy & Izod test.

7. MODULUS OF RESILLIENCE :-

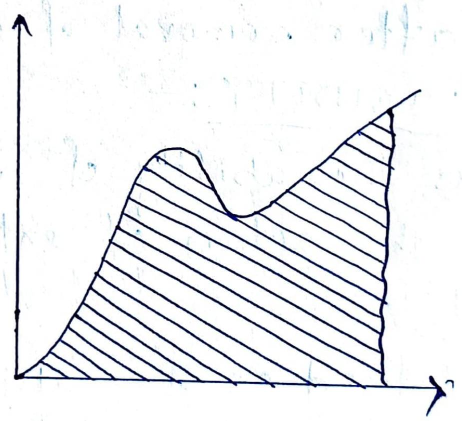
Modulus of resillience is defined as the area below the stress strain curve up to the elastic limit in a tension test.



$$A = \frac{1}{2} \sigma \varepsilon$$

8. MODULUS OF TOUGHNESS :-

Modulus of toughness is defined as the area below the stress strain curve up to the fracture limit.



9. MALLEABILITY :-

Malleability is defined as the ability of material to deform to a greater extend when a compressive load is applied.

10. DUCTILITY :-

It is defined as the ability of the material to deform to a greater extend before the sign of the crank when it subjected in tensile force.

11. BRITTLENESS :-

→ It is defined as the ability of the material which shows negligible plastic deformation before the fracture takes place.

→ Brittleness is opposite to that of ductility.

12. HARDNESS :-

Hardness is defined as the resistance of the material to indentation or penetration or permanent deformation.

DIFFERENT TERMS RELATED TO STRESS :-

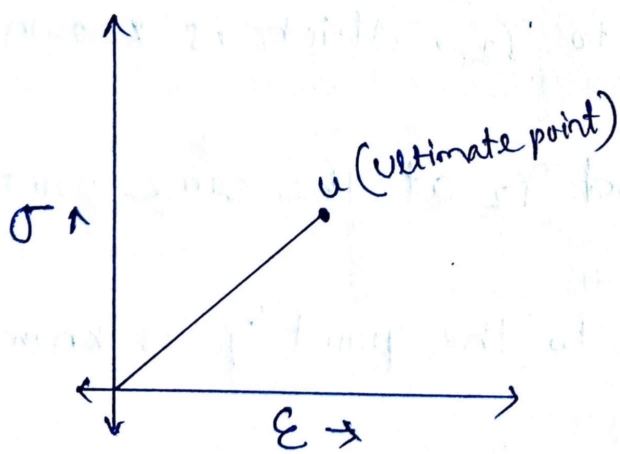
1. WORKING STRESS/SAFE STRESS/ALLOWABLE STRESS :-

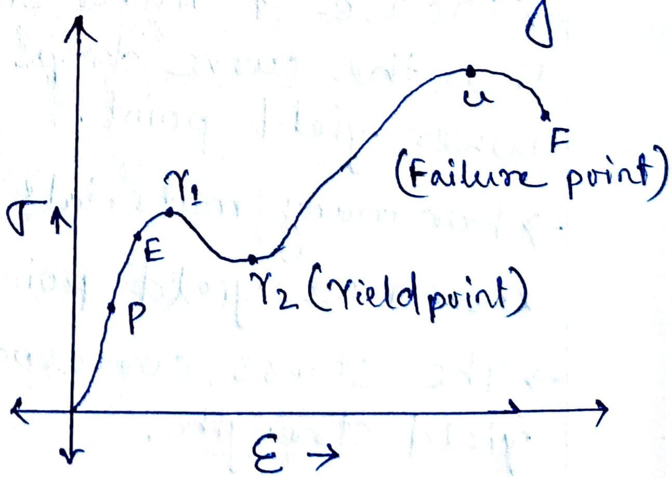
Working stress is defined as the safe stress taken within the elastic limit of the material. For brittle material working stress is equal to ultimate tensile strength divided by Factor of safety.

For brittle material; working stress $= \dfrac{\text{Ultimate strength}}{\text{Factor of safety (FOS)}}$

For ductile material which is having well defined Yield point.

For ductile material; working stress $= \dfrac{\text{Yield strength}}{\text{Factor of safety}}$





P = Proportional limit

E = Elastic point

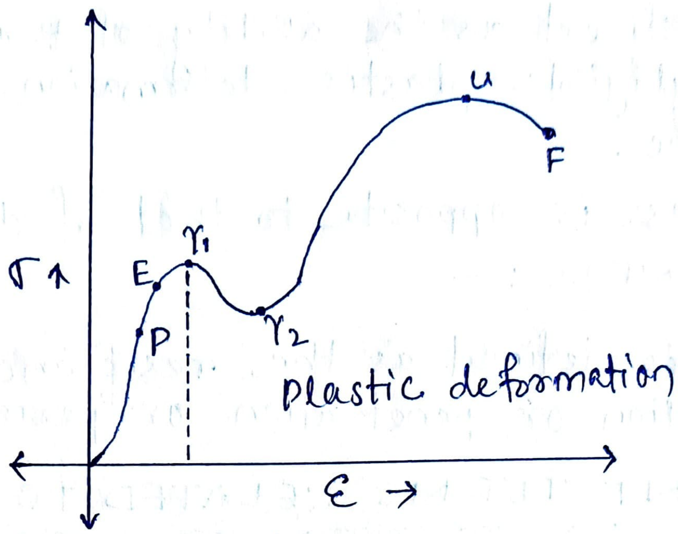
Y_1 = Upper yield point/limit

Y_2 = Lower yield point/limit

u = Ultimate point

F = Failure point

2. YIELD STRESS :-



→ When the specimen is stressed beyond the elastic point, then plastic deformation start and this material starts yielding.

→ During this stage it is not possible to recover the initial shape and size after the remoral of load.

→ It is seen from the diagram that beyond the elastic limit the strain increases at a faster rate up to the γ_1 i.e. upper yield point.

→ In other words there is an appriciable amount of increase in strain without increase in stress.

→ Incase of mild steel if there is a reduction of Load then the curve drops down to γ_2, which is known as lower yield point.

→ For many materials γ_1 and γ_2 at the same points i.e. known as yield point 'γ'.

→ The stress corresponding to the point 'γ' is known as yield strength.

3. ULTIMATE STRENGTH :-

→ If we refer to the stress strain diagram of the ductile material as shown in the figure.

→ The plastic deformation starts after the yield point.

→ The material become stronger because of strain hardening therefore higher and higher amount of load is required to deform the material.

→ After applying high amount of load, so the corresponding stress reaches a maximum value which is given by point 'u'.

→ The maximum stress corresponding to this ultimate point is known as ultimate point.

or,

Ultimate stress can defined as the maximum stress that can be reached in a tensile test.

FACTOR OF SAFETY (FOS) :-

Factor of safety is defined as the ratio between ultimate strength to the working strength for the brittle material and yield strength to the working strength for the ductile material.

Brittle material, $F.O.S = \frac{\text{Ultimate strength}}{\text{Working stress}}$

Ductile material, $F.O.S = \frac{\text{Yield strength}}{\text{Working stress}}$

Q-1 Find out the factor of safety for a mild steel specimen which has ductile strength of 350 MPa and working stress is 275 N/mm^2.

Solution:- Given data,

Working strength/stress = 275 N/mm^2 = 275 MPa

Ductile/Yield strength = 350 MPa

∴ Factor of safety (FOS) = $\frac{\text{Yield strength}}{\text{Working stress}}$

$= \frac{350}{275}$

$= 1.273$

FACTORS GOVERNING DESIGN OF MACHINE ELEMENTS :-

The following factors affect the design of machine elements.

① cost
② High output and efficiency
③ strength
④ stiffness / Rigidity
⑤ Wear resistance
⑥ Operational safety
⑦ Ease of assembly and disassembly
⑧ Light weight, reliability and durability

COST :-

→ Cost is an important factor for any type of machine design.

→ The machine design is the one which helps to get the finished product with highest possible quality and the lowest cost.

HIGH OUTPUT AND EFFICIENCY :-

→ Morden machines are using morden technology in order to improve the traditional machines.

→ These morden machines consume low power and give high power output that is why these machines are highly efficient than the traditional machines.

→ So high output and efficiency play an important role in the design of machines.

STRENGTH :-

Ability of material to sustain deformation without failure strength is up to 3 types :-

(i) Tensile strength - Ability of a material to sustain tensile load.

(ii) Compressive strength - Ability of a material to sustain compressive load.

(iii) Shear strength - Ability of a material to sustain shear Load without failure.

STIFFNESS / RIGIDITY :-

Stiffness is the resistance to deformation after the application of Load.

$$P = K\delta \ , \ K = \frac{P}{\delta}$$

WEAR RESISTANCE :-

The material used for design of machines should be able to wear and tear. - Resistance to wear.

OPERATIONAL SAFETY :-

During the design process safety is the most important criteria for every designer. That is why safety precaution is to be maintain for a good and economic design.

EASE OF ASSEMBLY AND DISASSEMBLY :-

A designer is responsible for making the machine components in such a way that its assembly and disassembly can be done easily.

LIGHT-WEIGHT, RELIABILITY & DURABILITY :-

Every designer wants to design the machine components in such a way that it will be easy to handle, the design is durable and also reliable.

DESIGN PROCEDURE OF MACHINE ELEMENT :-

Specify function of element

↓

Determine forces acting on machine element

↓

Select suitable material for machine element

↓

Determine failure mode for machine element

↓

Determine Geometric dimensions of each machine element

↓

Modify dimensions for assembly & manufacture & check design at critical cross-section.

PREPARE WORKING POINT ELEMENT :-

→ The design of machine element is the complete procedure of machine design.

→ The following steps which are represented above are illustrated as follows.

① SPECIFICATION FUNCTION :-

Function of each machine element is describe and discussed. For example :- Bearing function is to support the rotating shaft and confind its motion.

② DETERMINATION OF FORCES :-

The external and the internal forces act on the machine element as follows,

i, The extend force due to the energy, power and the torque transmitted by the machine part.

ii, Force due to the frictional resistance.

iii, Centrifugal force, Force of inertia etc.

③ SELECTION OF SUITABLE MATERIAL :-

Four basic factors are considered for the material selection

i, Availability

ii, Cost

iii, Mechanical properties

iv, Manufacturing consideration.

FAILURE MODE :-

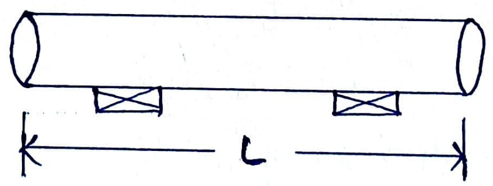
Different failure criterias are there, but from them 3 failure factors are most important.

(i) Failure by plastic deflection
(ii) Failure by General Yielding
(iii) Failure by Fracture

→ In applications like transmission shaft which is used to support the gears, the maximum force is found out by the elastic deflection.

→ So incase of these deflection reaches a value of 0.001 - 0.003 times of the span length of the shaft, then the shaft fails by elastic deflection.

Ex:- Transmission shaft



$\delta > (0.001 - 0.003)$

Let, $L = 2m$

then, $\delta > (0.002 - 0.006) \times L\,m$

→ Incase of the machine element made of ductile material failure mode will be yielding.

→ But if the components made up of brittle material then the deformation is because of the plastic deformation.

DESIGN OF FASTENING ELEMENTS

RIVETED JOINT :-

Riveted joints are used in the mechanical assembly for permanent connection. There are mainly 2 types of joint in the mechanical assembly.

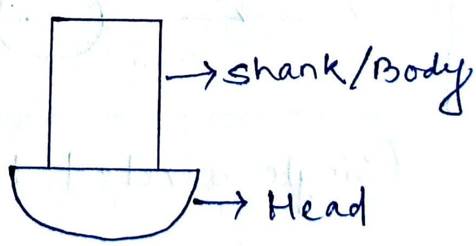
(1) Permanent joint
(2) Separable or detachable

→ The joints which can't be disassemble without damaging the assemble parts.

→ Riveted and welded joints are coming under permanent joint catagory.

→ Separable joints are the joints which permit assembly and disassembly without damaging the total joint.

→ A rivet consist of a rivet head with a cylindrical shank as shown in the figure.



(Rivet)

→ The head is formed on the shank by a metal forming process known as upsetting.

→ The rivet is inserted in the poles of the part which is to be assembled.

ADVANTAGES OF USING RIVETED JOINT :-

(i) It is used to joint different material.
(ii) It is used to joint different thickness plates.
(iii) There is no temperature effect.
(iv) It is having more strength and leak proof.
(v) It is having good damping capacity.

TYPES OF RIVETED JOINTS :-

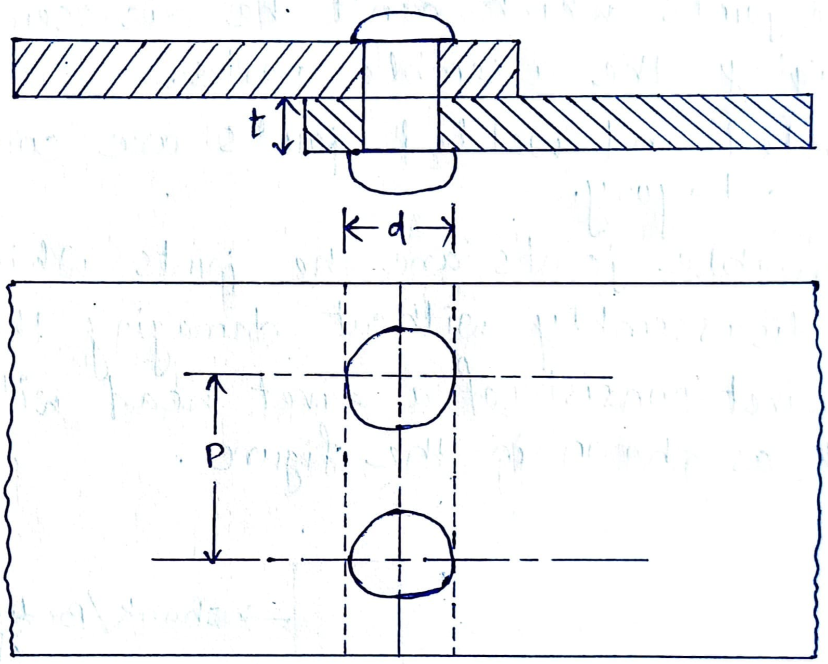
Riveted joints are classified into 2 groups mainly ;

(1) Lap joint

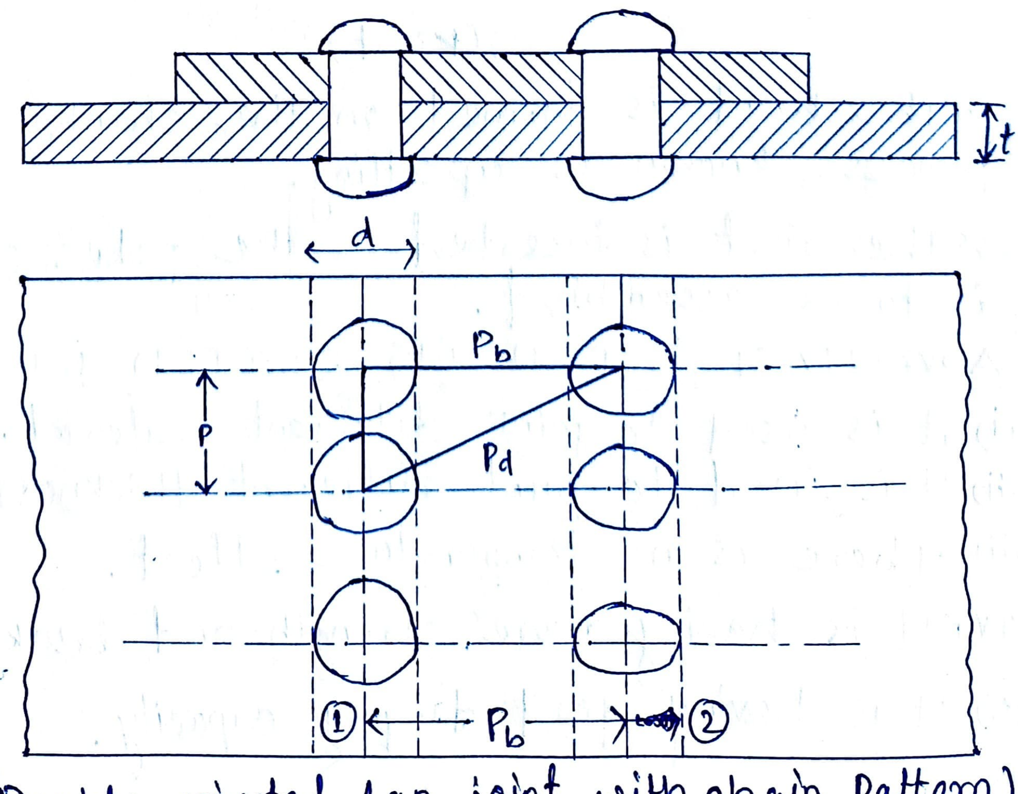
(2) Butt joint

LAP JOINT :-

Lap joint consist of two over lapping plates which are held together by one or more numbers rows of rivets.



(Single riveted lap joint)



(Double riveted lap joint with chain Pattern)

$\therefore P_d = \sqrt{P^2 + P_b^2}$

where, $P \rightarrow$ Pitch

$P_b \rightarrow$ Back pitch

$P_d \rightarrow$ Diagonal pitch

PITCH :-

Distance between the two consicutive rivet centres in a row is known as pitch (P).

GAUGE LINE :-

The line joining the rivet centers in a row is known as Gauge line.

BACK PITCH :-

The distance between two consicutive gauge line is known as Back pitch. (P_b)

DIAGONAL PITCH :-

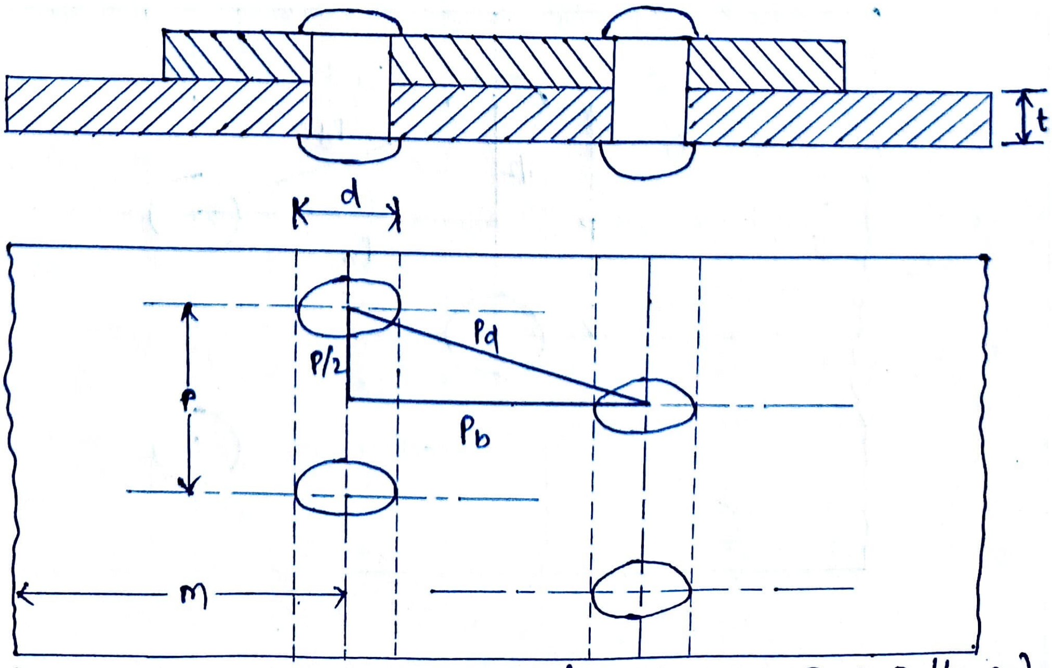
The distance between the centers of a rivet in adjacen rows of zig-zag center of a rivet is known a diagonal pitch.

$$P_d = \sqrt{P^2 + P_b^2}$$

Where, $P_d \rightarrow$ Diagonal pitch

$P \rightarrow$ Pitch

$P_b \rightarrow$ Back pitch



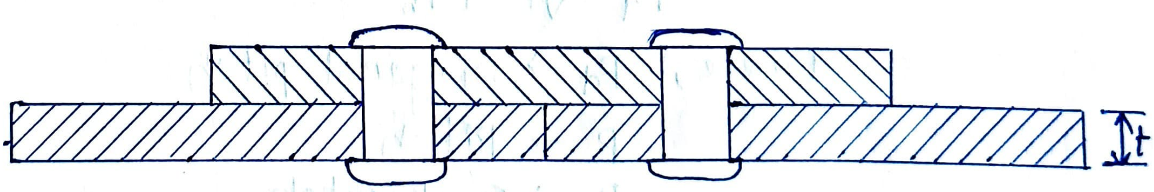
(Double riveted lap joint with Zig-Zag Pattern) diagonal pitch, $P_d = \sqrt{(P/2)^2 + (P_b)^2}$

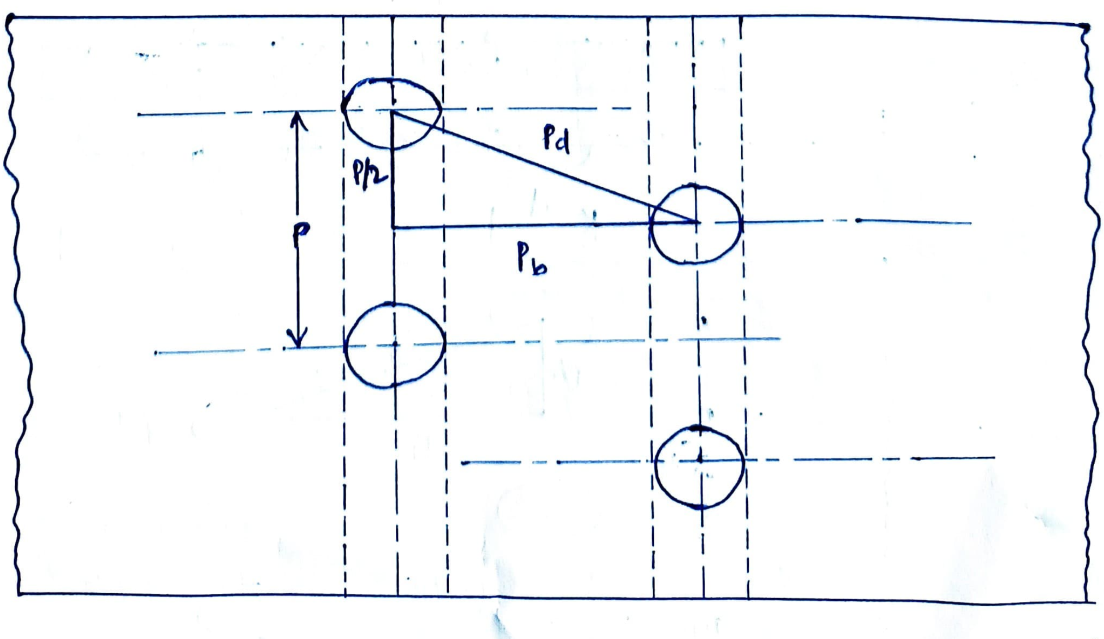
MARGINE :-

Rivet center to the edge of the plate this gap is known as margine.
Margine is always greater than equal to 1.5 times of diameter. ($M \geq 1.5d$)

BUTT JOINT :-

→ when two plates are kept in alignment with each other in the same plain and strap or coverplate is placed over these plates and riveted to each other.
→ Placing the two plates which are to be joined together is called butting. This joint is known as butt joint.
→ Depending on the number of rows of rivets the butt joints are classified on single row butt joint and double row butt joint.
→ Depending on the number of straps the butt joint is classified into 2 types ;- Single strap butt joint and double strap butt joint.





STRENGTH EQUATIONS :-

→ The strength of the riveted joint is defined as the force that the joint can withstand without causing the failure.

→ When the operating force exceeds the designed force then the failure occurs.

→ There are mainly 3 types of failures in the riveted joint :-

1. Shearing failure
2. Tearing failure
3. Crushing failure

SHEARING FAILURE :-

→ The shearing failure occurs in the lap joint and the butt joint due to the application of the shear force.

→ In the lap joint there is only single shear force occurs but in the butt joint double shear occurs.

The strength equation written as, $P_s = \frac{\pi}{4} d^2 \times \tau$

Where, P_s → shearing resistance of the rivet per pitch length.

D → The shank diameter of the rivet

τ → The shearing strength of the rivet

The is another formula for strength equation i.e.

$$P_s = n \times \frac{\pi}{4} d^2 \times \tau$$

Where, n = Number of rows of rivets

→ The values of 'n' differs from single riveted to triple riveted joint i.e. $n = 1$, for single riveted lap joint.

$n = 2$, for double riveted lap joint

$n = 3$, for triple riveted lap joint and so on

→ Incase of double strap single riveted butt joint the rivets are subjected to double shear hence, the shearing resistance will be, $P_s = 2 \times \frac{\pi}{4} d^2 \times \tau$

TEARING FAILURE :-

OR,

TENSILE strength of the plates between the rivets :-

The tensile failure of the plate between 2 consicutive rivets in a row is given by,

$$P_t = (P-d)t\,\sigma_t$$

Where, $P_t \rightarrow$ tensile force acting on the riveted joint

$P \rightarrow$ Pitch of the rivet

$d \rightarrow$ Diameter of the rivet

$t \rightarrow$ thickness of the plate

$\sigma_t \rightarrow$ tensile strength of the plate

CRUSHING FAILURE :-

The crushing failure within a riveted joint occurs when the compressive stress between the shank of the rivet and the plate exceeds the yield stress of the plate during compression.

It is denoted as P_c.

$$P_c = ndt\,\sigma_c$$

Where, $P_c \rightarrow$ the crushing force per unit pitch length

$n \rightarrow$ the number of rows of rivets per pitch length

$d \rightarrow$ shank diameter

$t \rightarrow$ thickness of the plate

$\sigma_c \rightarrow$ Crushing strength of the plate material

STRENGTH OF AN UNRIVETED JOINT :-

$$p' = p \times t \times \sigma_t$$

EFFICIENCY OF THE RIVETED JOINT :-

$$\eta = \frac{\text{Least of } P_s,\ P_t \,\&\, P_c}{p'}$$

UNWIN'S FORMULA :-

If thickness of the plate is greater than 8mm then, diameter of the rivet shank is,

$t > 8mm$ then

$d = 6\sqrt{t}$ mm

Q-1 A double riveted lap joint with chain riveting is to be design for a 13mm thick plate. Find the efficiency of the joint if,

$\sigma_t = 80\ N/mm^2$

$\tau = 60\ N/mm^2$

$\sigma_c = 120\ N/mm^2$

Solution :- Given data,

In a double riveted lap joint,

$n = 2$

$t = 13$ mm

$\sigma_t = 80\ N/mm^2$

$\tau = 60\ N/mm^2$

$\sigma_c = 120\ N/mm^2$

(i) $d = 6\sqrt{t}$

$= 6 \times \sqrt{13}$

$= 21.63$ mm $\simeq 22$ mm is the standard diameter according to design data hand book by S.MD. Jallaludeen

(ii) Pitch of the rivet :-

In order to find pitch, first we need P_t so,

P_t = Least of P_s / P_c

$\Rightarrow (P-d)t\,\sigma_t$ = Least of P_s/P_c

$\therefore P_s = n \times \frac{\pi}{4} d^2 \times \tau$

$= 2 \times \frac{\pi}{4} \times (22)^2 \times 60$

$= 45615.92\ N \simeq 45616\ N$

$\therefore P_c = n d t \sigma_c$

$= 2 \times 22 \times 13 \times 120$

$= 68640\ N$

So now least value is P_s so,

$P_t = P_s$

$P_t = P_s$

$\Rightarrow (P-d)t\,\sigma_t = 45616$ N

$\Rightarrow (P-22)\times 13\times 80 = 45616$

$\Rightarrow P = \frac{45616}{13\times 80} + 22$

$= 65.86$ mm or 66 mm

So now, $P_t = (P-d)t\,\sigma_t$

$= (66-22)\times 13\times 80$

$= 45760$ N

(iii) $\therefore \eta = \frac{\text{least of } P_s, P_c \,\&\, P_t}{P'}$

$= \frac{P_s}{P'}$

$= \frac{45616}{P t \sigma_t}$

$= \frac{45616}{66\times 13\times 80} \times 100$

$= 66.4\%$

Q-2 A single riveted lap joint is connected to 2 similar plates, the thickness of plate is 15 mm. If the diameter of the plate rivet is found by using unwin's formula, then design the riveted joint and find the efficiency of the joint. If all the stress are given,

$\sigma_t = 200$ MPa

$\tau = 100$ MPa

$\sigma_c = 150$ MPa

Solution :— Given data,

single riveted lap joint

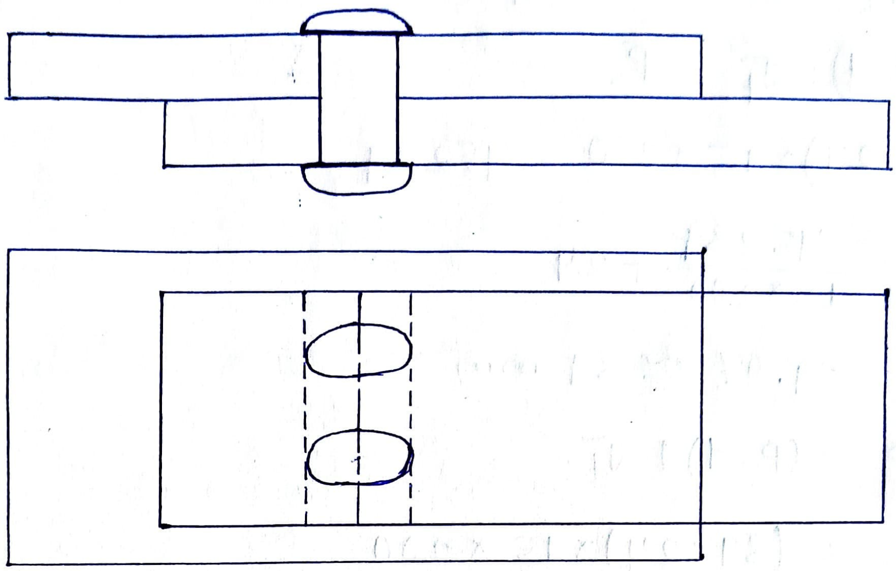
$t = 15$ mm

$\sigma_t = 200$ MPa

$\tau = 100$ MPa

$\sigma_c = 150$ MPa

Step-1



Step-2

Force calculation by determining geometrical dimensions of the riveted joint.

(i) Diameter of rivet :-

According to unwin's formula $t > 8$ mm so,

$$d = 6\sqrt{t}$$
$$= 6 \times \sqrt{15}$$
$$= 23.23 \text{ mm} \simeq 24 \text{ mm}$$

[∵ According to table No-10.5 in page-10.19 in design data hand book by S. MD. Jalaludeen]

(ii) Pitch of rivet :-

P_t = least of P_s/P_c so,

$$P_s = n \times \frac{\pi}{4} d^2 \times \tau$$
$$= 1 \times \frac{\pi}{4} \times (24)^2 \times 100$$
$$= 45238.93 \text{ N} \simeq 45239 \text{ N}$$

$$P_c = n d t \sigma_c$$
$$= 1 \times 24 \times 15 \times 150$$
$$= 54000 \text{ N}$$

$\therefore P_t$ = Least of P_s / P_c

$\Rightarrow (P-d)\, t\, \sigma_t = P_s$

$\Rightarrow (P-24) \times 15 \times 200 = 45239$

$\Rightarrow P = \dfrac{45239}{15 \times 200} + 24$

$\Rightarrow P = 39.07 \simeq 39$ mm

$\therefore P_t = (P-d)\, t\, \sigma_t$

$= (39-24) \times 15 \times 200$

$= 45000$ N

(iii) Efficiency of the riveted joint :-

$$\eta = \frac{\text{Least of } P_s, P_c \,\&\, P_t}{P'}$$

$$= \frac{P_t}{P t \sigma_t}$$

$$= \frac{(P-d)\cancel{t\,\sigma_t}}{P\,\cancel{t\,\sigma_t}}$$

$$= \frac{39-24}{39}$$

$$= \frac{15}{39} \times 100$$

$= 38\%$

Design of longitudinal butt joint for pressure vessel or boiler shell :-

The design of riveted joint for a boiler shell or pressure vessel consist of 2 types of designs of the joint :-

(i) Longitudinal butt joint

(ii) Circumferential lap joint

LONGITUDINAL BUTT JOINT :-

→ This type of butt joint is very common for the design of the pressure vessel.

→ The plate of the boiler shell is bend to form of a ring and their edges are joint to form a longitudinal butt joint.

→ This longitudinal butt joint is commonly a double, triple riveted butt joint.

→ This butt joint makes a ring like structure from the steel plate.

→ Boiler and pressure vessels are the cylindrical vessels, they are subjected to circumferential and longitudinal tensile stresses.

→ It can be proved that the circumferential stress is 2 times the longitudinal stress.

$$\sigma_H = \frac{PD}{2t}, \quad \sigma_L = \frac{PD}{4t}$$

→ Therefore longitudinal butt joint is stronger than circumferential lap joint and that is why longitudinal butt joint is used for pressure vessel.

→ Pressure vessels are subjected to steam pressure so it should be withstand the steam pressure in such a way that their should be no leakage of air or steam.

STEP-1

THICKNESS :-

Thickness of the plate of cylinder wall.

$$t = \frac{PD}{2\sigma_t \eta} + CA \text{ (corrosion allowance)}$$

Corrosion allowance → CA is defined as an additional material thickness which is added to withstand the internal pressure of the steam.

STEP-2

DIAMETER OF RIVETS :-

→ According to Indian Boiler Regulation Act (IBR), their is no specific formula to calculate the rivet diameter.

→ Hence an empirical relation is given by the design engineers in order to find the diameter of the rivet.

i) If thickness of the plate is greater than 8mm then, we can find the diameter of the rivet can be found by using 'unwin's formular, i.e. $t > 8mm$

$$d = 6\sqrt{t}$$

ii) If the diameter $t < 8mm$, is equating the shearing resistance of the rivets to $P_s = P_c$.

STEP-3

PITCH OF THE RIVETS :-

$$P_t = (P-d)t\,\sigma_t$$

$$P_t = \text{Least of } P_s/P_c$$

$$P_{min} = (2.25 \text{ to } 2.5)d$$

$$P_{max} = Ct + 41.28$$ [from design data hand book page-10.20 by S. MD. Jalaludeen]

Q-3 A steam boiler is to be design for a working fluid pressure of 250 KPa with internal diameter of 1.6 m. The size of the rivet to be made is 34.5 mm. The permisible shear stress of the material is 60MPa. Then find out the number of rivets required for the circumferential joint.

Solution:- Given data,

$$P_f = 250 \text{ KPa} = 250 \text{ N/m}^2$$

$$D = 1.6 \text{ m}$$

$$d = 34.5 \text{ mm}$$

$$\tau = 60 \text{ MPa}$$

$$\therefore \frac{\pi}{4} D^2 \times P_f = N \times \frac{\pi}{4} d^2 \times \tau$$

$$\Rightarrow \frac{\pi}{4} \times (1.6)^2 \times 250 = N \times \frac{\pi}{4} \times (34.5)^2 \times 60$$

$\Rightarrow N = \dfrac{502654.82}{56089.20}$

$= 8.96 \simeq 9$

$\therefore$ Number of rivets required is 9.

Q-4 Design the longitudinal joint for a 1.5 m diameter steam boiler to carry a steam pressure of 2.5 N/mm^2. The ultimate strength of the boiler plate is 420 MPa. Take the joint efficiency as 80%, If possible find out all the dimension and assume a suitable factor of safety.

Solution:- Given data,

$P = 2.5\ N/mm^2$

$D = 1.25\ m = 1250\ mm$

$\sigma_t = 420\ MPa$

$\sigma_c = 650\ MPa$

$\tau = 300\ MPa$

$\eta = 80\% = 0.8$

Assuming a factor of safety to 5 we get following allowable stresses :-

$\sigma_t = \dfrac{420}{5} = 84\ N/mm^2$

$\sigma_c = \dfrac{650}{5} = 130\ N/mm^2$

$\tau = \dfrac{300}{5} = 60\ N/mm^2$

(i) According to page 10.4 of design data hand book by S. MD. Jalaludeen for boiler structural joints, the thickness of plate, $t = \dfrac{PD}{2\sigma_t \eta} + 1\ mm$ [Assuming CA is 1]

$\Rightarrow t = \dfrac{2.5 \times 1250}{2 \times 84 \times 0.8} + 1$

$= 24.3 \simeq 25\ mm$

(ii) Diameter of rivet (d) :- [$\because t > 8\ mm$ so unwin's formula]

$d = 6\sqrt{t}$

$= 6 \times \sqrt{25} = 30\ mm$

According to page 10.5 of 10.02 of design data hand book by S.MD. Jalaludeen the standard shaft dia of the riveted joint of.

(iii) Pitch of rivets :-

P_t = least of P_s/P_c

According to page 10.7 of design data hand book by S.MD.Jalaludeen for longitudinal joint.

$$P_s = (n_1 + 1.875\,n_2)\frac{\pi}{4}d^2 \tau$$

$$= (1 + 1.875 \times 4) \times \frac{\pi}{4} \times (30)^2 \times 60$$

$$= 360497.75 \text{ N}$$

$$P_c = n d t \sigma_c$$

$$= 5 \times 30 \times 25 \times 130$$

$$= 48700 \text{ N}$$

So, $P_t = P_s$

$$\Rightarrow (P-d)\,t\,\sigma_t = 360497.75$$

$$\Rightarrow (P-30) \times 25 \times 84 = 360497.75$$

$$\Rightarrow P = \frac{360497.75}{25 \times 84} + 30$$

$$= 201.66 \text{ mm}$$

According to Indian boiler regulation Act (IBR) for boiler and axially loaded structural joint of 10.4 page no of design data hand book by S.MD. Jalaludeen the maximum permisible pitch, $P_{max} = Ct + 41$.

Assuming it is a double strap butt joint $C = 6$, According to table no - 10.6, recommended longitudinal joints for pressure vessels of design data handbook by S.MD. Jalaludeen then,

$$P_{max} = Ct + 41$$

$$= 6 \times 25 + 41 \text{ mm}$$

$$= 191 \text{ mm}$$

(iv) Distance between 2 rows of rivets or back pitch :-
According to Table no-10.8, characteristics of strong ~~structural~~ structural joints of page no-10.21 of design data hand book by S.MD. Jalaludeen, Double riveted butt joint with double unequal straps the back pitch.

$$P_b = 2 \text{ to } 2.5d \simeq 2.3d$$

$$= 2.3 \times 30$$

$$= 69 \text{ mm}$$

(v) Margine (m) $= 1.5d$
$= 1.5 \times 30$
$= 45$ mm

[According to table No-10.8 of design data hand book by S.MD. Jalaludeen]

(vi) Thickness of cover plate for double cover of equal widths,

$$t_1 = 0.625 \times t$$

$$= 0.625 \times 25$$

$$= 15.625 \text{ mm}$$

Q-5 Design a double riveted butt joint with two cover plates for the longitudinal joint of a boiler shell which is 1.5m diameter and subjected to a steam pressure of 0.75 N/mm². Assume joint efficiency 75%. Allowable tensile stress 90 MPa, Crushing stress 140 MPa, shear stress 56 MPa.

Solution:- Given data,

Double riveted longitudinal butt joint.

$$n = 2$$

$$D = 1.5\text{m} = 1500 \text{ mm}$$

$$P = 0.95 \text{ N/mm}^2$$

$$\eta = 75\% = 0.75$$

$$\sigma_t = 90 \text{ MPa}$$

$$\sigma_c = 140 \text{ MPa}$$

$$\tau = 56 \text{ MPa}$$

(i) Thickness of boiler shell plate :-

We know that thickness of boiler plate

$$t = \frac{PD}{2\sigma_t \eta} + 1\text{ mm} \quad [\text{assuming corrosion allowance is } 1]$$

$$= \frac{0.95 \times 1500}{2 \times 90 \times 0.75} + 1$$

$$= 11.6 \simeq 12 \text{ mm}$$

(ii) Diameter of the rivet :-

If the thickness of rivet $t > 8$ mm then,

$$d = 6\sqrt{t}$$

$$= 6 \times \sqrt{12}$$

$$= 20.78$$

According to table no-10.5 standard size of the rivets of page no-10.19

the nominal diameter of rivet $d = 22$ mm &

the diameter of rivet hole $d' = 23$ mm

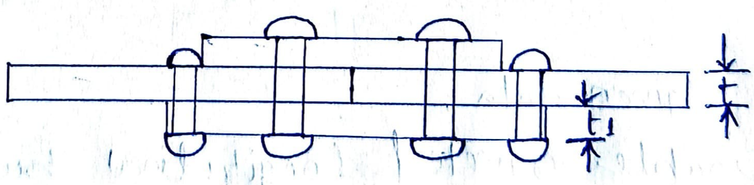
of design data hand book by S.MD. Jalaludeen.

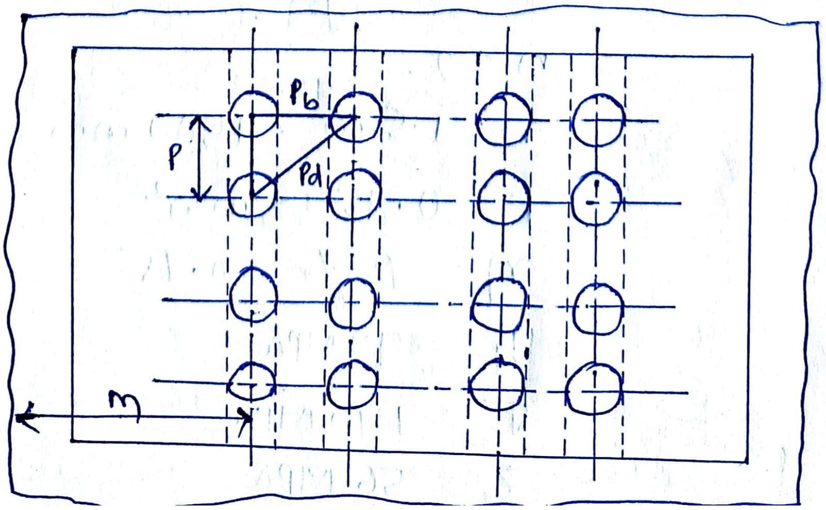
(iii) Pitch of rivets :-

P_t = least of P_s / P_c

$P_s = n \times 1.875 \times \frac{\pi}{4} d^2 \times \tau$ (where, n = no. of rivets per pitch)

$P_c = n \times d \times t \times \sigma_t$





To find out P_s for double riveted double strap butt joint as shown in the figure. There are two rivets per pitch length i.e. $n = 2$ and the rivets are in double shear according to IBR (Indian Boiler Regulation Act) double shear is 1.875 times stronger than the single shear. Therefore shearing strength of all the rivets will be;

$$P_s = n \times 1.875 \frac{\pi}{4} d^2 \times \tau$$

$$= 2 \times 1.875 \times \frac{\pi}{4} \times (22)^2 \times 56 \text{ N}$$

$$= 79827.86 \text{ N}$$

$$P_c = n d t \sigma_c$$

$$= 2 \times 22 \times 12 \times 140 \text{ N}$$

$$= 73920 \text{ N}$$

So $P_c < P_s$ then,

$$P_t = P_c$$

$$\Rightarrow (P-d) t \times \sigma_t = 73920 \text{ N}$$

$$\Rightarrow (P-22) \times 12 \times 90 = 73920$$

$$\Rightarrow P = \frac{73920}{12 \times 90} + 22$$

$$= 90.44 \text{ mm}$$

$$\therefore P_{max} = Ct + 41$$

$$= 3.5 \times 12 + 41$$

$$= 83 \text{ mm}$$

[C = 3.5, according to table 10.7, boiler code factor C of page 10.20 of design data hand book by S.MD Jalaludeen]

$P > P_{max}$ so,

$$P = P_{max} = 83 \text{ mm}$$

$$\therefore P_t = (P-d) t \times \sigma_t$$

$$= (83-22) \times 12 \times 90$$

$$= 65880 \text{ N}$$

(iv) Back Pitch :-

$P_b \geq 2d$

So $P_b = 2d$

$= 2 \times 22$

$= 44$ mm

(v) Thickness of the butt strap or cover plate (t_1)

$\therefore t_n = 0.625 t \left(\frac{P-d}{P-2d}\right)$

$= 0.625 t \left(\frac{83-22}{83-2\times 22}\right)$

$= 11.73$ mm

(vi) Margine :-

$m = 1.5d$

$= 1.5 \times 22$

$= 33$ mm

(vii) Efficiency of joint :-

$$\eta = \frac{\text{Least of } P_s, P_c \,\&\, P_t}{P'}$$

$$= \frac{P_t}{P'}$$

$$= \frac{(P-d) t \,\sigma_t}{P \times t \times \sigma_t}$$

$$= \frac{83-22}{83}$$

$$= 0.734$$

$$= 73.4\%$$

CIRCUMFERENCIAL LAP JOINT OF A BOILER :-

Design

step 1 & 2

The thickness of the shell and the diameter of the rivet :-

The thickness of the boiler shell and the diameter of the rivet is found on the similar manner as incase of Longitudinal joint of a boiler.

$$① \; t = \frac{PD}{2\sigma_t \eta} + 1$$

$$② \; d = 6\sqrt{t} \quad (\text{If } t > 8mm) \text{ - unwin's formula}$$

step-3

Calculation of no. of rivets :- (n)

Since it is a lap joint so the rivets are in single shear so the shearing resistance will be,

$$P_s = n \times \frac{\pi}{4} d^2 \times \tau \text{ ——— } (I)$$

where, n = total no. of rivets

Since the boiler shell is subjected to a fluid pressure of with a boiler diameter (D) so the shearing Load acting on the boiler is ;

$$P_s' = \frac{\pi}{4} D^2 \times P \text{ ——— } (II)$$

Now under equilibrium condition we can equites equ^n (I) & equation (II) i.e.,

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

$$\Rightarrow n = \left(\frac{D}{d}\right)^2 \times \frac{P}{\tau}$$

step-4

Pitch of the rivets (P_1) :-

Pitch can be foundout by considering the efficiency of the circumferencial joint.

$$\eta = \frac{P_1 - d}{P_1}$$

step-5

No. of rows of rivets :-

This can be found out by, No. of rows = $\frac{\text{Total no. of rivets}}{\text{No. of rivets in one row}}$

$$\Rightarrow \text{no. of rivets in one row} = \frac{\pi(D+t)}{P_1}$$

step-6

Margine :-

Refer data book and Margine (M) = 1.5 d mm for all riveted joint.

Q-6 A steam boiler is to be design for a working pressure of 2.5 N/mm^2, with an inside diameter of 1.6 m. Give the design calculation step wise. Assuming it is a circumferenceal lap joint. Consider the following working stress as follows.

$$\sigma_t = 75 \text{ MPa}, \tau = 60 \text{ MPa}, \sigma_c = 125 \text{ MPa}$$

Assume rivet hole diameter for the design calculation.

Solution :- Given data;

$P = 2.5 \ N/mm^2$

$D = 1.6 \text{ m}$

$= 1600 \text{ mm}$

Assume, $\eta = 1$

① Thickness of the boiler shell,

$$t = \frac{PD}{2\sigma_t \eta} + 1$$

$$= \frac{2.5 \times 1600}{2 \times 75 \times 1} + 1$$

$= 27.6 \text{ mm} \simeq 28 \text{ mm}$

② Diameter of the rivets :-

If $t > 8 \text{mm}$, $d = 6\sqrt{t}$

$= 6 \times \sqrt{28}$

$= 31.74 \simeq 34.5 \text{ mm}$ (Rivet hole dia)

[According to table 10.5 of the standard size of the rivets of design data handbook by S. MD. Jalaludeen]

③ No. of rivets :-

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

$$= \left(\frac{1600}{34.5}\right)^2 \times \frac{2.5}{60}$$

$= 89.61 \simeq 90$ nos.

④ Pitch of rivets :-

Assumption - let us consider the circumferential joint is double riveted lap joint. Hence the no. of rivets per row is divided between two consicutive rows so,

$$\text{no. of rivet in one row } (n_1) = \frac{90}{2} = 45 \text{ nos.}$$

$$\therefore \text{No. of rivets in one row } (n_1) = \frac{\pi(D+t)}{P_1}$$

$$\Rightarrow P_1 = \frac{\pi(1600+28)}{45}$$

$$= 113.65 \simeq 120 \text{ mm}$$

⑤ Efficiency, $\eta = \frac{P_1 - d}{P_1}$

$$= \frac{120 - 34.5}{120}$$

$$= 0.7125 = 71.25\ \%$$

⑥ Margine :-

$m = 1.5\, d$

$= 1.5 \times 34.5$

$= 51.75$ mm

DESIGN OF WELDING JOINT :-

→ A welded joint is a permanent joint which is obtain by the fusion of the edges of the 2 parts to be joint.

→ Welding is the joining of the parts by heating to a particular temperature with or without application of forces.

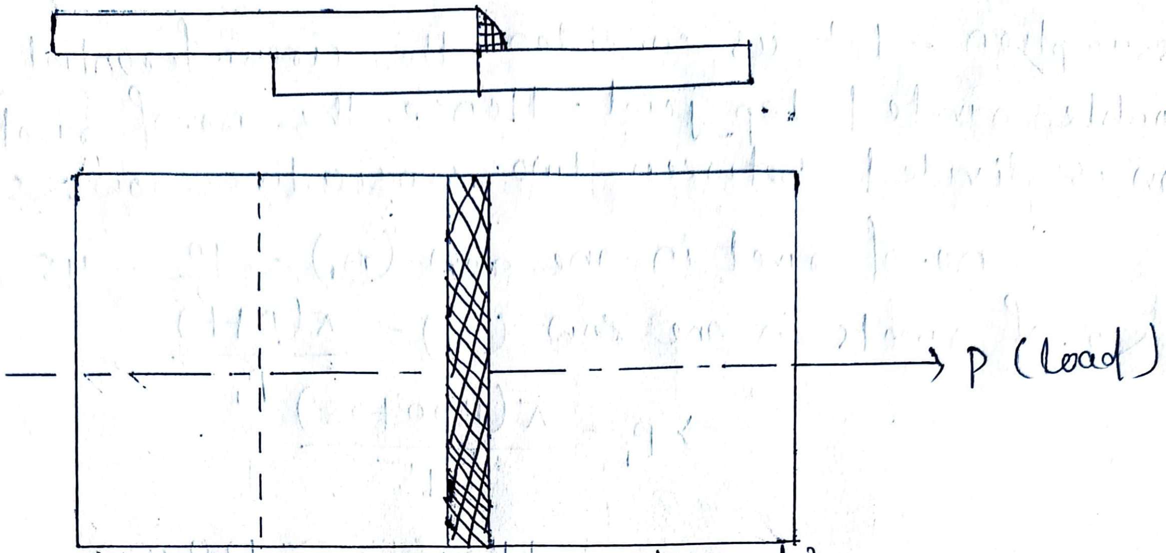
TYPES OF WELDED JOINT :-

Welded joint is two types :-

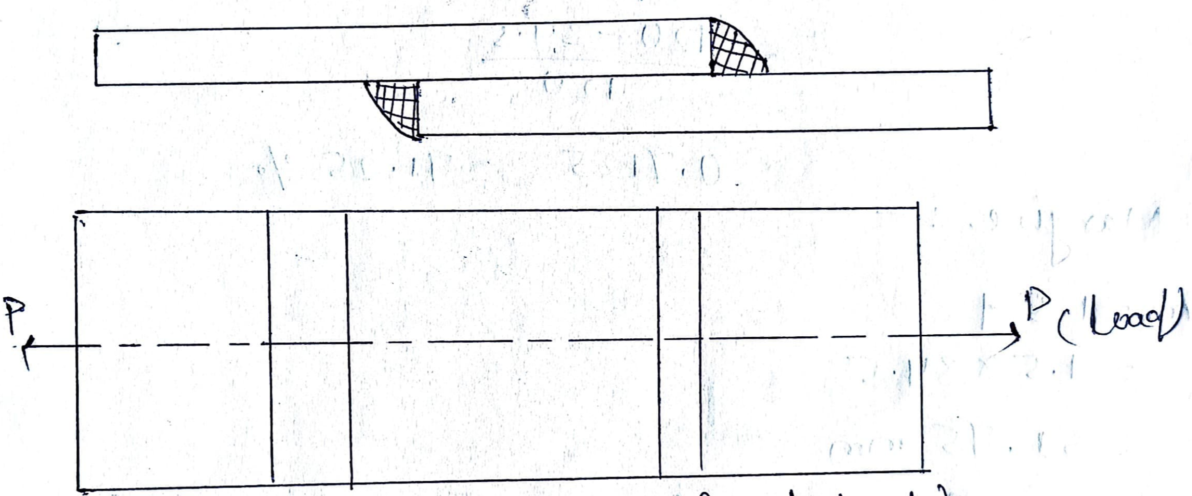
1) Lap joint / Fillet joint → (a) Parallel fillet joint
   - → Single
   - → Double

   (b) Transverse fillet joint

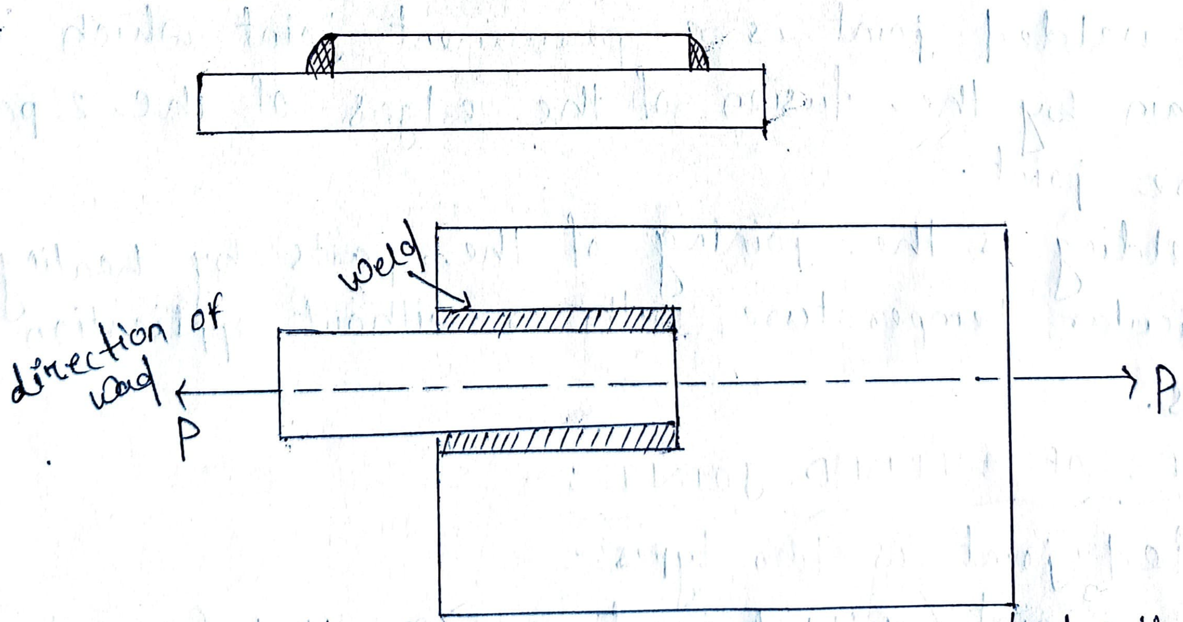
2) Butt joint.



(Single transeverse fillet joint)



(Double transverse fillet joint)



(Double parallel fillet weld joint)

The lap joint otherwisely known as fillet joint. It is a joint between two over lapping plates and the fillet weld is a triangular cross-section jointing two surfaces at right angle to each other.

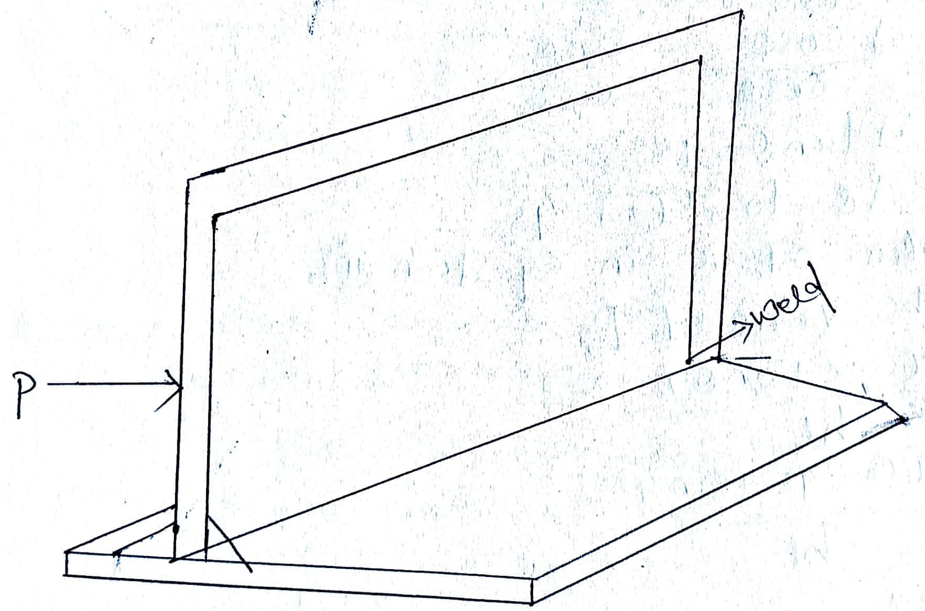
There are 2 types of fillet weld joints:-

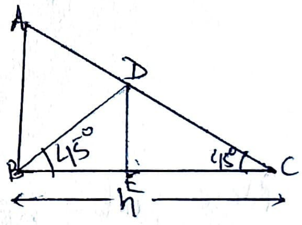
① parallel fillet weld joint.

② Transverse fillet weld joint, the load direction and the weld direction are the same.

But in the transverse fillet weld joint the load direction and the weld direction are perpendicular to each other.

Strength of parallel fillet weld Joint





where'

$h \rightarrow$ size of weld / leg of weld.

$t \rightarrow$ Throat thickness.

$P \rightarrow$ Strength of parallel fillet weld.

$\tau \rightarrow$ Shear stress

$BD = BE + BC$

$h = t(\cos\theta + \sin\theta)$

and $BC = BE + EC$

$= t(\cos\theta + \sin\theta)$

$h = t(\cos\theta + \sin\theta)$

$\Rightarrow t = \dfrac{h}{\cos\theta + \sin\theta}$ ———— (i)

$\tau = \dfrac{P}{tl}$ ———— (ii)

Now put the value of equation (i) in equation (ii), we get the following expression,

$$\tau = \frac{P}{\left(\dfrac{h}{\cos\theta+\sin\theta}\right) \times l}$$

$$= \frac{P(\cos\theta+\sin\theta)}{hl} \quad \text{———— (iii)}$$

for Maximum shear stress; $\dfrac{d\tau}{d\theta} = 0$

$$\Rightarrow \frac{P}{hl}(-\sin\theta + \cos\theta) = 0$$

$$\Rightarrow \cos\theta = \sin\theta$$

$$\Rightarrow \frac{\cos\theta}{\cos\theta} = \frac{\sin\theta}{\cos\theta}$$

$$\Rightarrow \tan\theta = 1$$

$$\Rightarrow \theta = \tan^{-1}(1) = 45$$

Now put the value of θ in Equation (iii), Now we get the value of P;

$$\tau_{max} = \frac{P(\cos\theta + \sin\theta)}{hl}$$

$$\Rightarrow \tau_{max} = \frac{P(\cos 45 + \sin 45)}{hl}$$

$$\Rightarrow P = \frac{hl\,\tau_{max}}{\frac{\sqrt{2}}{2} + \frac{\sqrt{2}}{2}}$$

$= \dfrac{hl\,\tau_{max}}{\sqrt{2}} = 0.707\, hl\tau \rightarrow$ Single parallel fillet weld joint

$$\Rightarrow \boxed{\begin{aligned} P &= 2 \times 0.707\, hl\tau \\ &= 1.414\, hl\tau \end{aligned}}$$

$\rightarrow$ This is the Expression for Double Prallel fillet weld joint.

STRENGTH OF SINGLE TRANSVERSE FILLET WELD JOINT:-

$$P = 0.707\, h\, l\, \sigma_t$$

where, σ_t = Permisible tensile stress for the fillet weld joint in Newton per m^2.

→ If there are 2 welds of equal length then the strength of double transverse weld joint, then,

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

STRENGTH OF BUTT JOINT:-

$$\boxed{P = h\, l\, \sigma_t}$$

(Butt Joint)

where, P = the tensile force on the plate.
σ_t = tensile stress in the weld.
h = throat thickness / size of the weld.
l - Length of the weld.

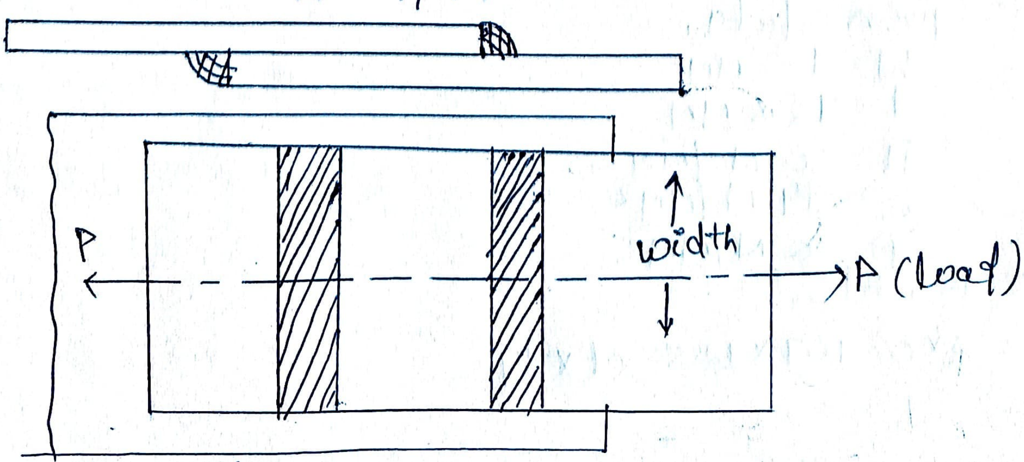
EFFICIENCY OF THE WELDED JOINT:-

$$\boxed{P = h\, l\, \sigma_t\, \eta}$$

(If there is no specific efficiency then let it be 1)

Q-7 A steel plate 100 mm in width and 10 mm thickness is welded to another steel plate by means of a double transverse fillet weld joint. The Maximum tensile should not exceed 110 N/MM², then find out the tensile force acting on the plate and then find the length of the weld.

Solution :- Given data,
t = 10 mm
width (w) = 100 mm
σ_t = 110 N/mm²



$\therefore \sigma_t = \frac{P}{A}$

$\Rightarrow P = \sigma_t \times A$

$= 110 \times (w \times t)$

$= 110 \times (100 \times 10)$

$= 11000 N$

$= 110 kN$

$\therefore P = 1.414\, h\, l\, \sigma_t$

$\Rightarrow 110 \times 10^3 = 1.414 \times 10 \times L \times 110$

$\Rightarrow l = 70.72$ mm

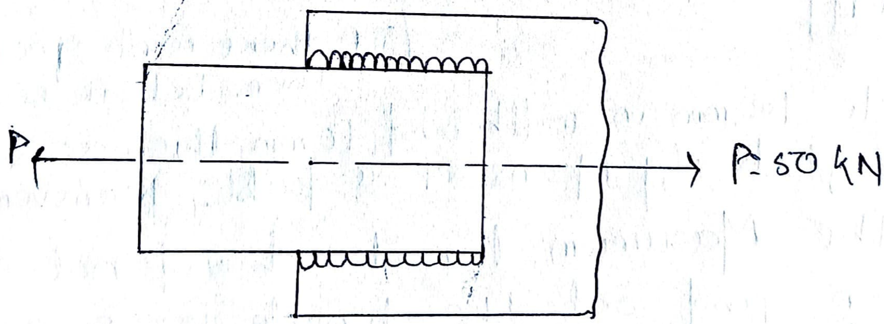
Adding allowance of 15MM in the length of the weld for Starting and stoping of the weld the length will be,

$l' = l + 15$

$= 70.72 + 15$

$= 85.72$ MM

Q-8
A steel plate 100mm in width and 12.5mm thick is welded to another steel plate by means of double parallel fillet weld joint as shown in fig.



$\sigma_t = 50 N/MM^2$, $\tau = 94 N/MM^2$

Find out the required length of the weld if the tensile stresses is acting on the weld is $50 N/MM^2$ and the shear stress is $94 N/MM^2$

Solution:- Given data,

$w = 100$ MM

$t = 12.5$ MM

$\sigma_t = 50 N/MM^2$

$\tau = 94 N/MM^2$

$P = 50 N/MM^2$

$\therefore P = 2 \times 0.707\, h\, l\, \tau$

$\Rightarrow 50 \times 10^3 = 2 \times 0.707 \times 12.5 \times L \times 94$

2) $l = \frac{50 \times 10^3}{1.414 \times 12.5 \times 94}$

$= 30.09$ mm

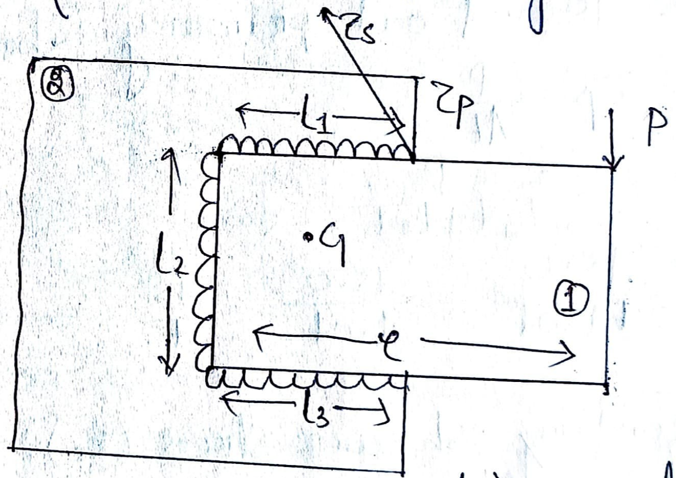
$l' = l + 15$

$= 30.09 + 15$

$= 45.09$ MM

WELDED JOINT SUBJECTED TO ECCENTRIC LOADING:-

A welded joint May have 2 types of fillet joints i.e. Parallel fillet joint and transverse fillet joints. They combinely form a fillet joint which is subjected to eccentric Loading and eccentric loading means the Load which is acting some distance away from the centre and the distance from the centre to the point of application of load is known as eccentricity or eccentrical distance.



e → eccentricity.

$x_G = \frac{\Sigma x_i l_i}{\Sigma l_i}$

$y_G = \frac{\Sigma y_i l_i}{\Sigma l_i}$

$G = (x_G, y_G)$

$\tau_P = \frac{P}{At}$

$\tau_s = \frac{Mr}{J}$

$\tau_R = \sqrt{\tau_P^2 + \tau_s^2 + 2\tau_P \tau_s \cos\theta}$

$At = l_1 t + l_2 t + l_3 t$

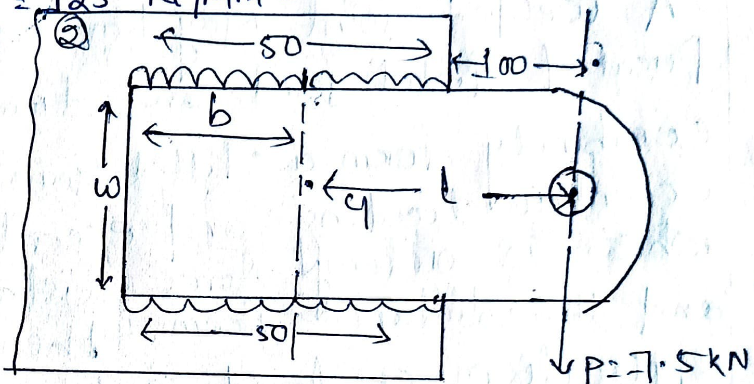
Q-9 A welded connecting as shown in the figure is subjected to an eccentric force of 7.5 kN. Determine the size of the weld if the permissible shear stree is $100 N/MM^2$.

Soleetion:- Given data,

$P = 7.5 kN = 7500 N$

$\tau = 100 N/MM^2$

$e = 100 + 25 = 125 N/MM^2$



Step-1 To find out primary shear stress (τ_p):-

$$\tau_p = \frac{P}{At}$$

$$= \frac{P}{L_1 t + L_2 t}$$

$$= \frac{7500}{50t + 50t} = \frac{7500}{100t} = \frac{75}{t} \quad \text{——— (1)}$$

Step-2

Secondary shear stress (τ_s):-

$$\tau_s = \frac{M r_1}{J}$$

Inorder to find τ_s, we need,

$\therefore M = P.e$

$= 7500 \times 125$

$= 937500$ N-MM

$$r_1 = \sqrt{25^2 + 25^2}$$

$= 35.36$ MM

$$J = tl\left(\frac{l^2 + 3b^2}{6}\right)$$

$$= tl\left(\frac{l^2}{6} + \frac{b^2}{2}\right)$$

$$= 50t\left(\frac{50^2}{6} + \frac{25^2}{2}\right)$$

$= 36.4583 \times 10^3 t$ (According to table no 11.8 polar Moment of inertia in torsion of design data hand book by S.MD Jalaludeen)

$= 36458.33\, t$

$\therefore \tau_s = \frac{M r_1}{J}$

$= \frac{937500 \times 35.36}{36458.33t}$

$\Rightarrow \tau_s = \frac{909.25}{t}$ ——— (11)

Step-3

Size of the weld:-

$\therefore \tau_p + \tau_s = \tau$

$\Rightarrow \frac{75}{t} + \frac{909.25}{t} = 100 \ N/mm^2$

$\Rightarrow \frac{75 + 909.25}{t} = 100$

$\Rightarrow t = \frac{75 + 909.25}{100}$

$= 9.84 \ mm$

ADVANTAGES OF WELDED JOINT OVER ALL OTHER JOINTS:-

A welded Joint can be a substitute for a riveted joint because of the following advantages:-

(i) Riveted joint required additional cover plate. A large no. of rivets but in welding joint there is no such requirements hence the welded assembly are light in weight.

(ii) Due to the elemination of these components the cost of the welded assembly is cheaper as that of the riveted joint.

(iii) The design of the welded assembly can be easily and economically modified.

(iv) Welded joints are tight and leak proof.

(v) The production time for the welded joint is very less.

(vi) The welded joints are smooth and pleasent in appearance.

(vii) The strength of the welded joint is very high.

(viii) The welded assemblies are very easily machined as compared to the casting.

SHAFT:-

Shaft helps in power transmission from one place to another place and it is the most important machine elements used in the transmission of power.

Function of Shaft:-

→ The transmission shaft or simply shaft are used in every rotating machine element to transmit rotary motion are torque and power from one location to another location.

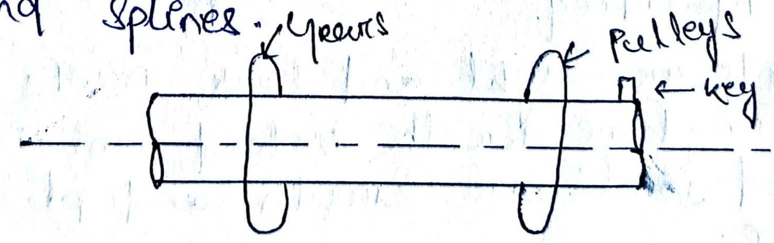
→ It is based on the formula.

$$\boxed{P = \frac{2\pi NT}{60} \text{ k.W}}$$

→ A Shaft is a rotating machine element usually circular in cross-section hence it can transmit a large amount of power.

→ Power transmitting elements such as gears, sprockets, pulleys are mounted over it.

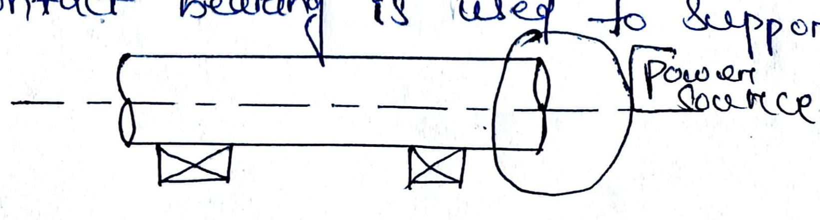
→ This machine element attach to the shaft by means of keys and splines.



COUPLING:-

→ Couplings are attached to the shaft by means of machine element to connect in to the power source.

→ Coupling contact bearing is used to support the shaft.



→ When the shaft is subjected to rotary Motion two types of Moment can come into existance.

(I) Twisting Moment (T)
(II) Bending Moment (M)

→ As shaft is an important Machine element for the power transmission, hence it's design is also very Much important.

→ Hence design of shaft depends on different important factors.

(i) Strength.
(ii) Rigidity.
(iii) Stiffness.

→ The speed of the shaft is an important Criteria to transmit power from one location to another. hence critical speed of the shaft play a significant role in the dynamic Condition.

→ The shaft are given specific name depending on it's typical application which involve transmission of Motion power and torque.

Ex:- Axle, Spindle, counter shaft, line shaft.

→ There are basically but their but their name changes according to it's application.

Axle:-

→ Axle is a type of shaft which only supports the rotating Machine element such as wheels, rope, drum.

→ Axle is only design for bending Moment.

SPINDLE:-

Spindle is a short shaft which support all type's of Machine tools Mounted.

MATERIAL OF THE SHAFT:-

→ The Material for the shaft are Medium carbon steel which ranges from 0.3% to 0.5%.

Ex:- Most commonly used shaft 30C8, & 40C8

→ This types of Material having higher strength. other types of Material is used for Shaft construction.

Ex:- Alloy steel.
Steel Made of Nickel.
Alloys Made from Nickel-Chromium.

→ Alloy steel are costly than the plan carbon steel but their high strength, high hardness, thoughness than the plan carbon steel hence alloy steels are mainly used for commercial ~~pep~~ purpose.

→ Commercial Shaft is also be Made from low carbon Steel.

DESIGN HOLLOW & SOLID SHAFT:-

SHAFT DESIGN BASED ON STRENGTH CRITERIA:-

Torsion equation — $\frac{T}{IP} = \frac{\tau}{r}$

$$\tau = \frac{T(D/2)}{\frac{\pi}{32} D^4} = \frac{32TD}{2\pi D^4}$$

$$\boxed{\tau = \frac{16T}{\pi D^3} \qquad T = \frac{\pi}{16} \tau D^3}$$

for solid shaft.

Where, T = Twisting moment.
IP = Polar moment of.
τ = Shear stress.
r = Radius
IP = 2XI

DESIGN HOLLOW SHAFT BASED ON SHEAR STRENGTH CRITERIA:-

$$\frac{T}{IP} = \frac{\tau}{r} \qquad \tau = \frac{Tr}{IP}$$

$$\Rightarrow \tau = \frac{T(D/2)}{\frac{\pi}{32}(D^4 - d^4)} = \frac{32TD}{2\pi(D^4 - d^4)}$$

$$\Rightarrow \tau = \frac{32TD}{2\pi(D^4 - d^4)}$$

$$\Rightarrow \tau = \frac{16TD}{\pi D^4 \left\{1 - \left(\frac{d}{D}\right)^4\right\}}$$

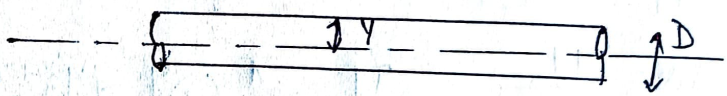
$$\Rightarrow \tau = \frac{16TD}{\pi D^4(1-k^4)}$$

$$\Rightarrow \tau = \frac{16T}{\pi D^3(1-k^4)}$$ → for hollow shaft.

$$\Rightarrow T = \frac{\pi}{16}\tau D^3(1-k^4)$$

DESIGN OF HOLLOW SHAFT BASED ON BENDING STRENGTH CRITERIA :-

$$\frac{\sigma_b}{y} = \frac{M}{I}$$



$$\Rightarrow \sigma_b = \frac{My}{I}$$

$$= \frac{MD/2}{\frac{\pi}{64}(D^4 - d^4)}$$

$$\Rightarrow \sigma_y = \frac{32M}{\pi D^3(1-k^4)} \qquad \left(\because k = \frac{d}{D}\right)$$

$$\Rightarrow M = \frac{\pi}{32}\sigma_d \pi D^3(1-k^4)$$

★ Bending Moment equation :- $\boxed{\frac{\sigma_b}{y} = \frac{M}{I} = \frac{E}{R}}$

★ Torsion equation :- $\boxed{\frac{T}{I_P} = \frac{\tau}{r} = \frac{G\theta}{L}}$

DESIGN OF SOLID SHAFT BASED ON BENDING STRENGTH CRITERIA :-

$$\frac{\sigma_b}{y} = \frac{M}{I}$$

$$\Rightarrow \sigma_b = \frac{My}{I}$$

$$= \frac{MD/2}{\frac{\pi}{64} \times D^4}$$

where, bending strength, $\sigma_b = \frac{32m}{\pi D^3}$

Solid shaft, $M = \frac{\pi}{32}\sigma_b d^3$

DESIGN OF THE SOLID & HOLLOW SHAFT BASED ON TORSIONAL & RIGIDITY CRITERIA :-

→ A torsional transmission shaft is said to be right on the basic of torsional rigidity. If it doesn't twist too much by the application of the torque then it is said to rigid.

→ Similarly the shaft is said to be based on the lateral rigidity. If it doesn't deflect under the action of the bending moment.

$$\frac{T}{I_P} = \frac{G\theta}{L} \qquad I_P = \frac{\pi}{32} D^4 \rightarrow \text{Solid shaft}$$

$$\Rightarrow \theta = \frac{TL}{G I_P} \qquad I_P = \frac{\pi}{32}(D^4 - d^4) \rightarrow \text{Hollow shaft.}$$

DESIGN OF SOLID & HOLLOW SHAFT BASED ON LATERAL RIGIDITY & CRITERIA :-

This design based on lateral rigidity remain same as the previous as the permissible angle of twist varies from 2° to 5° depending upon the twisting moment application.

Q-1 A propeller shaft is required to transmit a power 45 kW running at 500 rpm. It is a hollow shaft having inside dia 0.6 times the outer diameter. It is made up of plane carbon steel the permisible shear stress is 84 MPa. Then calculate inside & outer diameter.

Solution

Given, $P = 45\text{ kW} = 45 \times 10^3 \text{ W}$

$N = 500 \text{ rpm}$

$d/D = 0.6$

$\tau = 84 \text{ MPa}$

$$P = \frac{2\pi N T}{60}$$

$$\Rightarrow T = \frac{60P}{2\pi N}$$

$$= \frac{60 \times 45 \times 10^3}{2\pi \times 500}$$

$$= 859.436 \text{ N·m}$$

$$\tau = \frac{16T}{\pi D^3(1-k^4)}$$

$$\Rightarrow D^3 = \frac{16 \times 859.436}{\pi \times 84 \times (1-(0.6)^4)}$$

$$\Rightarrow D = \sqrt[3]{59.86} = 23.2m$$

$$\therefore \frac{d}{D} = 0.6$$

$$\Rightarrow \frac{d}{23.2} = 0.6$$

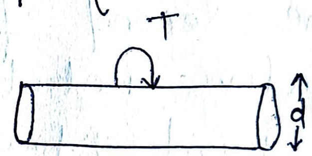
$$\Rightarrow d = 0.6 \times 23.2 = 13.92m$$

Q-2 A Solid shaft is transmitting power of 50 kw. which is rotating at 400 rpm. find out the diameter of the shaft If the ultimate shear stress is $60 N/mm^2$ & fos is 2.

Solution:- $P = 50$ kW

$N = 400$ rpm

$\tau_{ut} = 60 N/mm^2$



$$\tau_{per} = \frac{16T}{\pi d^3}$$

$$\Rightarrow \tau_{per} = \frac{\tau_{ut}}{FoS}$$

$$= \frac{60}{2} = 30 MPa$$

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

$$\Rightarrow P = \frac{2 \times \pi \times 400 \times T}{60}$$

$$\Rightarrow T = \frac{60 \times 50 \times 10^3}{3 \times \pi \times 400}$$

$= 11936$ Nmm

$= 1193.6 \times 10^3$ Nmm

$$\tau_{per} = \frac{16T}{\pi d^3}$$

$$\Rightarrow 30 = \frac{16 \times 1193.6 \times 10^3}{\pi \times d^3}$$

$$\Rightarrow d^3 = \frac{16 \times 1193.6 \times 10^3}{\pi \times 30} = 3978.6$$

$$\Rightarrow d = \sqrt[3]{3978.6}$$

$= 189.2$ mm.

KEY

FUNCTIONS:-

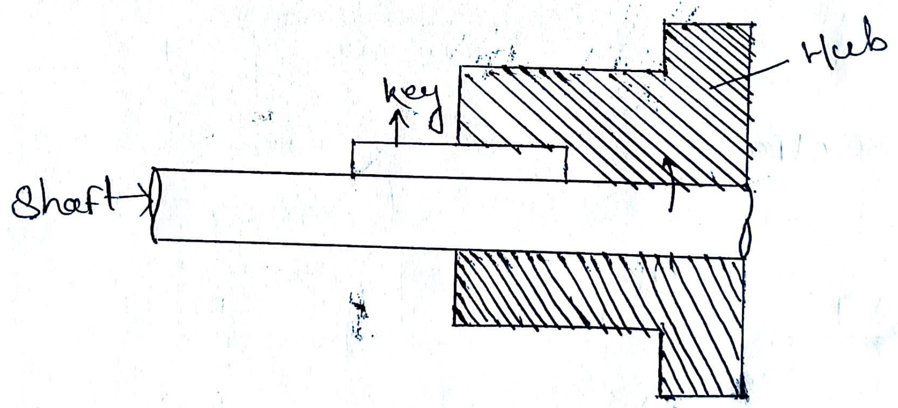
(i) A key is defined as a Machine element which is used Connect the transmission shaft to the rotating Machine element such as pulleys, gears, Sprockets, wheels.

ii) A keyed joint consist of shaft, hub and key.

iii) There are 2 Majore function of the key, they are as follows;

a) The primary function of the key is to transmit the torque from the shaft to the hub of Mating Part and vice-versa.

b) The Second function of the key is to present the relative rotational Motion between the shaft and the joint Machine element like gear or pulleys, in Most cases key also prevents axial Motion between the two elements.



TYPES OF KEYS:-

keys are divided into the following types:-

1. Sunk key.
2. Saddle key.
3. Flat key.
4. Tangent key.

SUNK KEY:-

→ The sunk key are provided half in the keyway of the shaft and half in the keyway of the hub.

→ The sunk keys are of following types:-

a) RECTANGULAR SUNK KEY:-

The Rectangular sunk key is having standard division of width thickness.

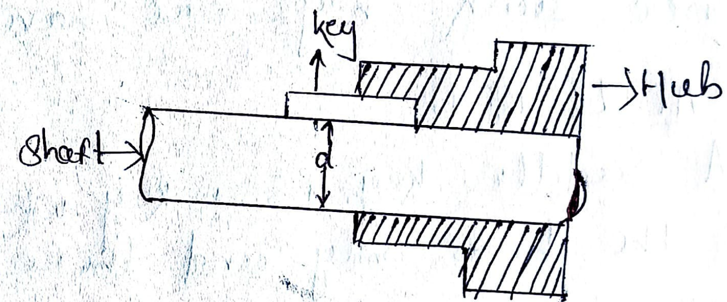
$$W = \frac{d}{4}; \quad t = \frac{2w}{3} = \frac{2}{3} \times \frac{d}{4} = \frac{d}{6}$$

$$\therefore W = \frac{d}{4}; \quad t = \frac{d}{6}$$

where, W = width.

d = Shaft dia.

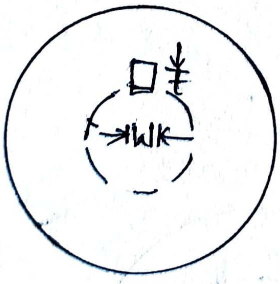
(Taper in 1:100 key)

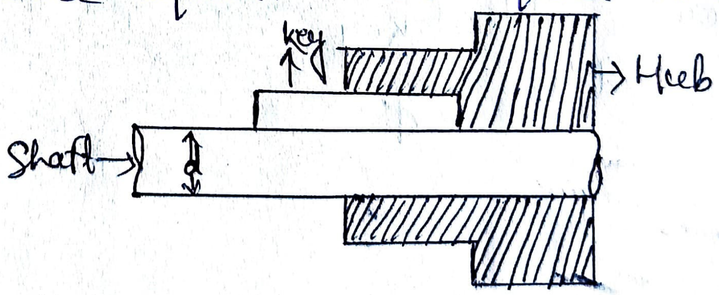


b) Square sunk key:-

The square sunk key is the one in which the width and the thickness are equal, which equal to $\frac{d}{4}$

$$\therefore W = t = \frac{d}{4}$$





c) PARALLEL SUNK KEY:-

→ The parallel sunk key can be of 2 types depending one it's cross-section i.e. it can be rectangular or square cross-section.

→ It is having uniform width and thickness through out, but it is taper less key.

→ Sunk key are play an important role in the design of all the Machine elements because it is used in heavy duty application.

→ In Sunk key the power is transmitted due to the Shear resistance of the key. Hence the relative Motion between the shaft and the key is also prevented by the shear resistance of the key.

→ Sunk keys with rectangular or Square cross-section is also called as flat key and flat keys are More Stable in all the type of Machine tool application hence Sunk keys are having industrial importants.

d) <u>SADDLE KEY</u>:-

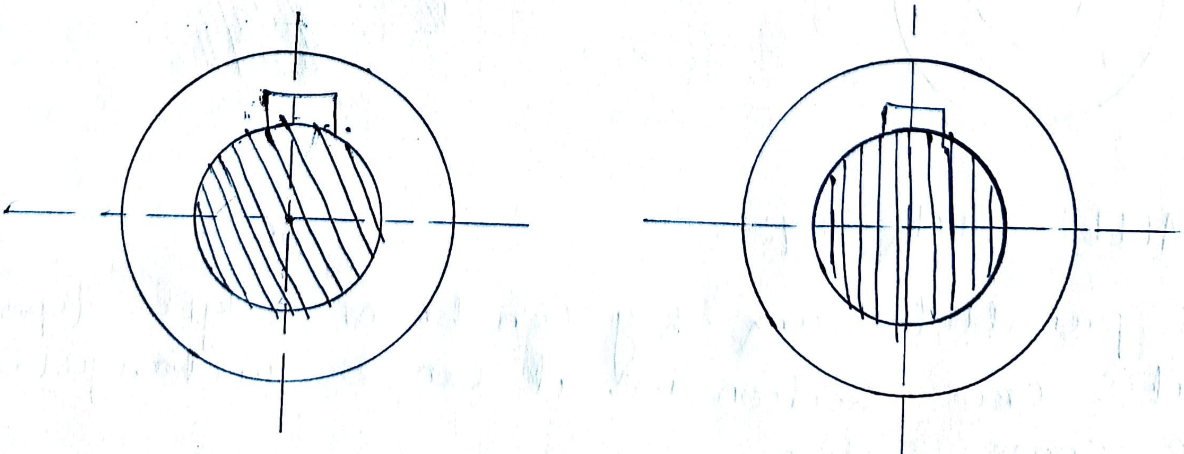
→ A Saddle key is a key which fits into the key way of the hub only and there is no keyway in the shaft

→ It also plays an important role in the industrial applications.

There are two types of Saddle keys.

① Hallow Saddle key.

② Flat Saddle key.



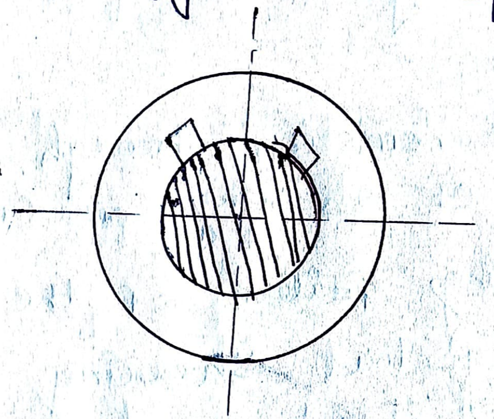
-> In the hollow saddle key the bottom surface slightly concave and in the flat saddle key the bottom surface is perfectly flat to accomodate the key.

-> It is used in the light duty application or low power transmission activities.

-> As there is no keyway in the shaft hence cost of the saddle key is less as compare to sunk key.

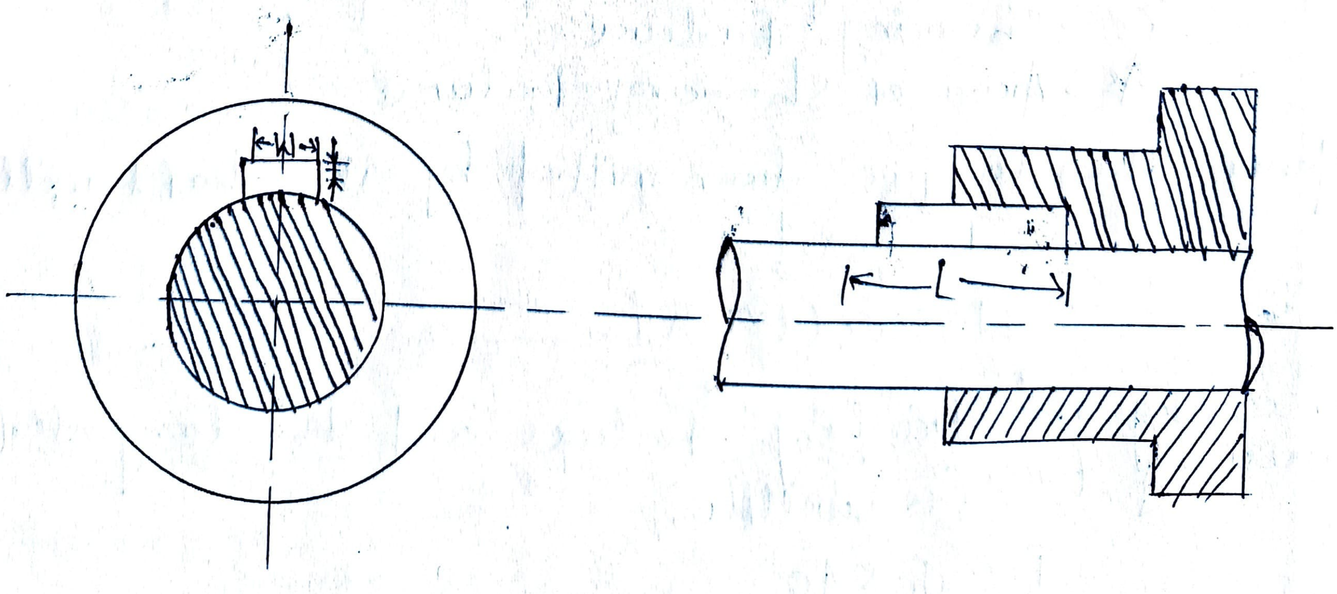
e) TANGENT KEY:-

The tangent keys are fitted at right angles but in pair.



Each key is able to withstand the torsion in one direction only and there used in heavy duty shafts.

DESIGN OF A RECTANGULAR SUNK KEY :-



Let us consider a power transmitting shaft which Connected to the hub of the Mating part by Means of a rectangular sunk key as shown in the fig.

Let 'T' is to be the torque transmitted by the shaft

F → The tangential force acting on the key at the circumference.

d → diameter of the shaft.

W → Width of the key.

L → Length of the key.

t → Thickness of the key.

τ & σ_c → Shearing and crushing stresses for the material of the key.

→ Due to the power transmission through the shaft the keyway fail because of shearing or crushing.

→ If we see the shearing failure of the key we have to calculate the tangential shearing force at the circumference of the shaft so tangential shearing Forse. (F_s)

$$F_s = \tau_s \times A_s$$
$$= \tau_s \times (L \times W)$$

where, F_s → Shearing force.

τ_s → Shearing failure.

A_s → Area of shearing failure.

Hence the torque transmitted by the shaft will be

$$T = F_s \times \frac{d}{2}$$

$$T = \tau_s (l \times w) \times \frac{d}{2} \quad (1)$$

Considering the crushing failure and the tangential crushing force (F_s) will be,

$$F_c = \sigma_c \times A_c$$
$$= \sigma_c \times \left(l \times \frac{t}{2}\right)$$

Hence the torque transmitted by the shaft due to the crushing failure will be,

$$T = F_c \times \frac{d}{2}$$

$$T = \sigma_c \times \left(l \times \frac{t}{2}\right) \frac{d}{2} \qquad \text{—— (ii)}$$

→ As the key is equally strong in shearing and the crusing failure hence we can equate the equation

① & equation ⑪

Equating equating ① & (ii) we will get

$$\sigma_c \times \left(\cancel{l} \times \frac{t}{2}\right) \times \frac{\cancel{d}}{2} = \tau_s (\cancel{l} \times w) \times \frac{\cancel{d}}{\cancel{2}}$$

$$= \sigma_c \times \frac{t}{2} = \tau_s \times w$$

$$\Rightarrow \boxed{\frac{\sigma_c}{\tau_s} = \frac{2w}{t}}$$

$$\Rightarrow \boxed{\frac{\tau_s}{\sigma_c} = \frac{t}{2w}}$$

Important case :

In case of a square key width = thickness $= \frac{d}{4}$

Hence, $\boxed{\frac{\sigma_c}{\tau_s} = 2} \Rightarrow \boxed{\sigma_c = 2\tau_s}$

(a) A 15 K.W and 960 rpm motor has a mild steel shaft of 40 mm dia and the extension being 75 mm. The permisiable shear & crushing stress for the mild steel key is 56 mpa & 112 mpa. Design the Key way in the motor shaft extension. chods the shear strength of the key against. The normal strength of the shaft.

Given: $P = 15\ K.W = 15 \times 10^3\ W$

$N = 960$ rpm $\qquad \sigma_c = 112$ mpa

$D = 40$ mm $\qquad L = 75$ mm

$\tau = 56$ mpa

$$P = \frac{2\pi N T}{60}$$

$$\Rightarrow \frac{P \times 60}{2\pi N} = T$$

$$\Rightarrow 15 \times 10^3 = \frac{2\pi \times 960 \times T}{60}$$

$$\Rightarrow T = \frac{15 \times 10^3 \times 60}{2 \times \pi \times 960}$$

$$= 149.2\ N.m$$

$$= 149.2 \times 10^3\ N\text{-}mm$$

considering the shearing failure of the key

$$T = F_s \times \frac{d}{2}$$

$$= \tau_s \times A_s \times \frac{d}{2}$$

$$= \tau_s \times (L \times w) \frac{d}{2}$$

$$\Rightarrow 149 \times 10^3 = 56 \ (75 \times w) \times \frac{40}{2}$$

$$\Rightarrow w = \frac{149 \times 10^3}{56 \times 75 \times 20}$$

$$= 1.77 \text{ mm} \simeq 1.8 \text{ mm}$$

$$w = \frac{d}{4} = \frac{40}{4} = 10 \text{ mm}$$

As the width is very very small so it cannot be considered for suitable design because for economical degin of the key way width w should atleast $\frac{d}{4}$ i.e $w = \frac{d}{4} = \frac{40}{4} = 10$ mm.

V.I.M.P

If we compare both the permisable shear stress and the crushing stress then we will final that

$$\boxed{\sigma_c = 2\tau}$$

This is only possible case if w = t i.e the sunk key is square key. so w = t = 10 mm so, we considered the design of the key.

According to HF more the sheaft strength factor.

$$e = 1.02\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)$$

$$= 1.02\left(\frac{10}{40}\right) - 1.1\left(\frac{5}{40}\right) = 0.8125$$

Strength of solid shaft with key way

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

$$= \frac{\pi}{16} \times 56 \times (40)^3 \times 0.8125$$

$$= 571.7 \times 10^3 \text{ N·mm}$$

strength of the key considering shear failure.

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

$$T = 75 \times 10 \times 56 \times \frac{40}{2}$$

$$= 840 \times 10^3 \text{ N-mm}$$

Shear strength of key ÷

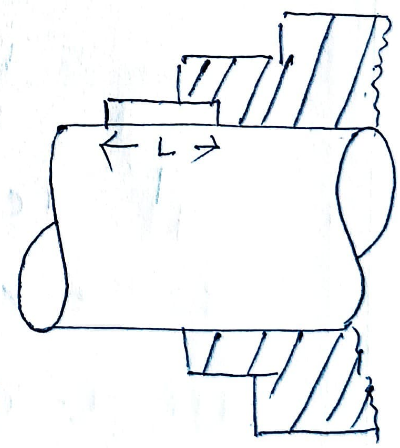
Normal strength of the shaft $= \dfrac{840 \times 10^3}{571.6 \times 10^3}$

$= 1.76 \simeq 1.5$ times

so from the design analysis of the key and the shaft we found that the strength of the key is 1.5 times the normal strength of the shaft it means the key design is stranger than the shaft degien.

(Q.) A rectangular sunk key is to be designed for a shaft having a diameter of 25 mm. then find out the tangential force due to the shear stress of 40 mpa acting on it and also find the torque due to the tangential shear force.

Given :-



$D = 25\ m$

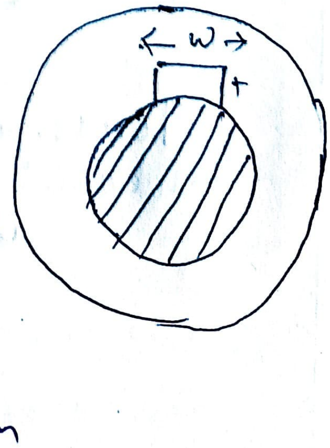
$\tau = 40\ M.p.a$

$w = \frac{d}{4} = \frac{25}{4} = 6.25\ mm$

$t = \frac{d}{6} = \frac{25}{6} = 4.16\ mm$

$L = 15\ mm$

$F_c = \tau_s\ (l \times w)$



$= 40\ (15 \times 6.25)$

$= 40 \times 93.75$

$= 3750\ N$

$T = 3750 \times \frac{25}{2} = 46875\ Nmm$

Now equating eq^{n} (i) and eq^{n} (ii) we will get the length of the key due to shearing failure :

$$16800 l = 1.03 \times 10^6$$

$$l = \frac{1.03 \times 10^6}{16800}$$

$$l = 61.3 \text{ mm}$$

Now considering failure of the key

$$T = \left(l \times \frac{t}{2}\right) \sigma_c \times \frac{d}{2}$$

$$= (l \times 5) \times 70 \times 25$$

$$= 8750 \, l \text{ N mm} \quad \text{———— (iii)}$$

Equating eq^{n} (iii) & (ii) we will get the length of the key baseng on the crushing failure .

$$8750 l = 1.03 \times 10^6$$

$$l = \frac{1.03 \times 10^6}{8750}$$

$$= 117.71 \text{ mm} \simeq 120 \text{ mm} .$$

So,

From the two length of the key which is found out considering the shearing and crushing failure of the key the larger length of the key is to be taken for the design of the because it is safe.

so length of the key is 120 mm,

Design a rectangular sunk key using an empiried relation for the given diameter of the shaft :

→ The design of rectangular sunk key using an empired relation for a given dia of the shaft avoides graphical representation and it is only formula based which gives a lelatoon with the dia of shaft and it also an imp. relation to findout dimension of the key. It uses a concept known as effect of keyway during the design of rectangular sunk key.

Effect of key way ÷

→ A little considertation will show that the keyway cut in the shaft reduces the load carrying capacity of the shaft. This is due to the formation of sharp edges in the keyway which increases the stress conncentration near the corner of the keyway and also results in reduction of cross-sectional area of the shaft.

so, this weakening effect of the shaft is represented in the form a equation which is based on the experimental results found by H.F Moore.

$$\boxed{e = 1 - 0.2\left(\frac{w}{d}\right) - 1.1\left(\frac{h}{d}\right)}$$

where,

e → shaft strength factor it is the ratio between the strength of the shaft with keyway and the strength of th of the shaft without keyway.

w → width of keyway

h → Depth of the keyway, $h = t/2$

d → Diameter of the shaft

t = thickness of the key.

→ It is usually assum that the strength of the keyed shaft is 75 % of solid shaft. which is some but higher then the value found out using this relation.

→ In case the key way is very long and the key is of sliding type then the angle of twist is increased in the ratio of k_θ which is given by the relation as follows.

$$\boxed{K\theta = 1 + 0.4\left(\frac{w}{d}\right) + 0.7\left(\frac{h}{d}\right)}$$

A hollow shaft having an outer dia of 60 mm and inner dia. of 40 mm is subjected to a twisting moment of 30 KNm. If the modulus of rigidity for a 1.5 m length shaft is 25 Mpa find out the angel of twist of hollow circular shaft by useing the rigidity criteria.

Ans:-

$d = 40\,mm$, $D = 60\,mm$

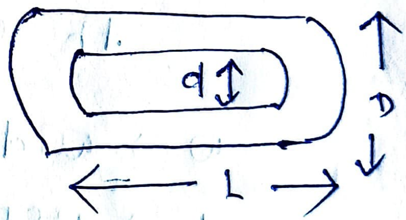
$L = 1.5\,m = 1500\,mm$

$G = 25\,M.p.a$

$\theta = ?$

$T = 30\,KNm$

$= 30 \times 10^6\,Nmm$



For the design of shaft using rigidity criteria.

$$\frac{T}{I_p} = \frac{G\theta}{L}$$

$$\Rightarrow \theta = \frac{TL}{G I_p}$$

$$I_p = \frac{\pi}{32}\left\{(60)^4 - (40)^4\right\}$$

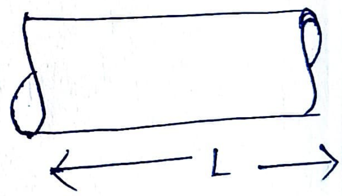
$$= 1021017.61\,mm^4$$

$$\theta = \frac{30 \times 10^6 \times 1500}{\cancel{196.349}\ 25 \times 1021017.61}$$

$= 1762.94$ radian

Q. A solid circular shaft of having 20 mm dia is subjected to a twisting moment of 30 K.N.M. If angle of twist for the shaft is 5.5° then find out the modulus of rigidity of shaft of using rigidity criterie for a length of shaft of 2.5 m.

Ans:- $L = 2.5\ m = 2500\ mm$



$d = 20\ mm$

$T = 30\ K.N\,m$

$= 30 \times 10^6\ Nmm$

$G = ?$

$\theta = 5.5^\circ = \frac{\pi}{180} \times 5.5\ rad = 0.0959\ rad$

$G = \frac{TL}{\theta I_p}$ $\qquad I_p = \frac{\pi}{32} d^4 = \frac{\pi}{32}(20)^4 = 15707.963\ mm^4$

$$= \frac{30 \times 10^6 \times 2500}{0.0959 \times 15707.96}$$

$= 49$ mpa.

Q. A solid shaft is required to transmit a torque 15 KN-m. find the necessary diameter of the shaft, if the allowable shear stress is 60 N/mm²

Given,

$T = 15 \text{ KN.m}$

$= 15 \times 10^6 \text{ N.mm}$

$\tau = 60 \text{ N/mm}^2$

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

$15 \times 10^6 = \frac{\pi}{16} \times 60 \times d^3$

$d^3 = 1273344.65$

$d = \sqrt[3]{1273344.65}$

$= 108.38 \text{ mm} \simeq$

$= 110 \text{ mm}$

Q. Find the dia of a solid steel shaft to transmit 20 K.W at 200 rpm. The ultimate shear stress for the steel may taken as 360 mpa and a factor of saftey as 8. If a hollow shaft is to be used in place of the solid shaft. Find the inside and outside dia. when ratio of inside to outside diameter is 0.5

Given:

$P = 20\ k.W = 20 \times 10^3\ W$

$K = \frac{d}{D} = 0.5$

$N = 200$ rpm

$\tau_u = 360$ Mpa $= 360$ N/mme

F.S $= 8$

we know that allowable shear stress

$$\tau = \frac{\tau_u}{F.S} = \frac{360}{8} = 45\ N/mm^2$$

$$T = \frac{P \times 60}{2\pi N} = \frac{20 \times 10^3 \times 60}{2\pi \times 200}$$

$= 955$ N-m

$= 955 \times 10^3$ N-mm

Torque transmitted by the solid shaft (T)

Dia of solid shaft

$$d^3 = 955 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3$$

$d = 47.6\ mm \simeq$

$d = 50\ mm$

Dia of hollow :-

Torque transmitted

$$955 \times 10^3 = \frac{\pi}{16} \times 2 \times D^3 (1-K^4)$$

$$= \frac{\pi}{16} \times 45 \times D^3 \{(1-(0.5)^4\}$$

$$D^3 = \frac{955 \times 10^3}{8.3} = 115060.241$$

$$D = 48.6 \simeq 50 \text{ mm}$$

$$d = 25 \text{ mm}$$

UNIT-4

DESIGN OF COUPLING

Introduction:-

→ shafts are usually 7m in length due to inconvenient in transport in order to have a greater length of the shaft it becomes necessary to point two or more pieces of shaft by means of a coupling.

→ shaft coupling are used in mechining for several process and the common and Imp of which are the following below.

(i) shaft coupling are used to provide a connection for the shaft of the unit that are manufacture for motor and genetor to provide for conn. and dis connection of shaft.

(ii) To provide for miss-allienment of shaft or to riduced mech. flexbility?.

(iii) To reduce the transmition of sock load form one shaft to another shaft.

(iv) To introduce protection against over load.

(v) The shaft couple shouldn't have any projecting parts because projecting part will disturb the assembly of shaft.

Requirement of good shaft coupling :-

→ A good shaft coupling sould have followings requirement :

(i) The coupling should be easy to connect or disconnect.

(ii) It should transmit the full power form one shaft to another shaft without any usee.

(iii) The coupling should hold the shaft in perfect element.

(iv) It should reduce the shock load form one shaft to another shaft.

(v) It should not have any projecting part.

Types of shaft coupling :-

(i) Rigid coupling

(ii) Flexiable coupling

(i) Rigid coupling :- The rigid coupling is used to connect two shaft which are perfect allinged.

→ There are three types of rigid coupling from the subject point of view.

a. Sleeve | Muff coupling

b. compression | clamp | split-muff

c. Flange coupling

(ii) Flexiable coupling :-

→ It is used to connect two shaft having both lateral and angular miss-allinement.

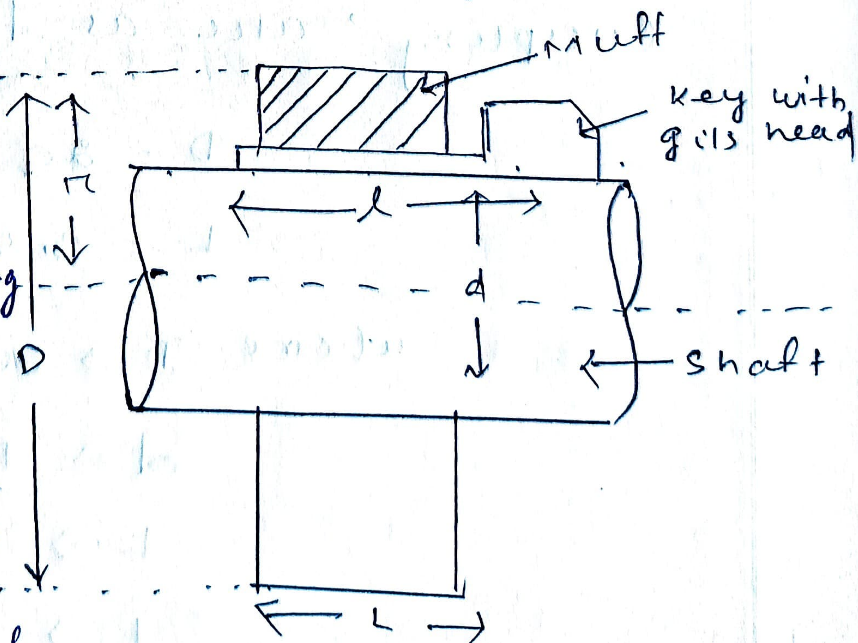
→ It is also three types depending on its application.

(a) Bushed pin

(b) Universal

(c) Oldham's

1.Q **Design of sleeve | Muff coupling :-**



→ Sleeve or muff coupling is the most simplest type of rigid coupling which is used to connect two shafts which are in perfect allinment.

→ It is made up of cast iron.

→ It consist hollow cylinder having inner diameter equal to diameter of shaft.

→ It is fitted over the ends of the two shaft by means of a gib head key (as show in fig) Hence the power is transmitted from one shaft to another shaft by means of that key & the sleeve.

→ It is therefore necessary that all the element in this coupling should be strong enough to transmit the torque through it.

→ The usual propertion of the cast iron sleeve coupling are as follows:

$$D = 2d + 13\ mm$$

$$L = 3.5d$$

where D → outer dia of sleeve

d → Dia of shaft

L → Length of sleeve

l → length of the coupling key

① Design of sleeve :-

→ The sleeve is to be design by considering it a hollow shaft. We know that

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

where T = Torque transmitted by the coupling

τ = permissible shear stress for the material of sleeve which is cast iron.

The safe value of permisible shear stress for cast iron may be taken as 14 M.pa. Now ~~we~~ we have to design sleeve on the basic of strength eqn of hollow shaft.

$$\Rightarrow \frac{T}{\frac{\pi}{32}(D^4 - d^4)} = \frac{\tau}{D/2} \qquad \left\{ r = \frac{D}{2} \right\}$$

$$\Rightarrow T = \frac{\pi}{16} \tau \left(D^4 - d^4 / D \right) \qquad \left\{ k = \frac{d}{D} \right\}$$

$$= \frac{\pi}{16} \tau D^3 (1 - k^4)$$

Design of key :

→ The key for the copling is to be design in same way as discoussion for the rectangular sunk key design.

→ The length of the copuling key should be equal to half of the length of the sleeve i.e 3.5d

$$l = \frac{L}{2} = \frac{3.5d}{2}$$

Now considering the crushing and shearing failure the torque transmission through the key is given by the following expression.

$$T = (l \times w)\, \tau \times \frac{d}{2} \quad \text{———— (i)}$$

$$T = \left(l \times \frac{t}{2}\right) \times \sigma_c \times \frac{d}{2} \quad \text{———— (ii)}$$

Q. Design and make a net dimensional sketch of a muff coupling which is used to connect two steel shaft transmitting 40 K.W at 350 rpm. The material for the shaft and key is plain carbon steel for which allowable shear and crushing stress may be taken as 40 MPa and 80 M.Pa respectivly. The material for the muff is cast iron for which the allowable shear stress may be assumed as 15 mpa.

Given :-

$P = 40\ K.W = 40 \times 10^3\ W$

$W = 350\ rpm$

Key $\begin{cases} \tau = 40.\ Mpa \\ \sigma_c = 80\ M.p.a \end{cases}$

Muff $\{ \tau = 15.\ Mpa$

$P = \frac{2\pi NT}{60}$, $T = \frac{P \times 60}{2\pi N}$

$= \frac{40 \times 10^3 \times 60}{2\pi \times 350}$

$= 1091.34\ N.m$

$= 109.1 \times 10^4\ N.mm$

For solid shaft :-

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

$109.1 \times 10^4 = \frac{\pi}{16} 40 d^3$

$d^3 = \frac{109.1 \times 10^4}{7.8539} = 138911.87$

$d = 51.79 mm \simeq 52$

Design of sleeve :

we know that the outer dia of sleeve is

$$D = 2 \times 52 + 13$$

$$= 117 \text{ mm}$$

Length of sleeve $L = 3.5d = 3.5 \times 52$

$$= 182 \text{ mm}.$$

Let us know check the permissible shear stress muff.

Let τ_m = Induced shear stress in the muff since the muff is consider to be a hollow shaft hence "τ_m" be found out from the following

Expression :

$$T = \frac{\pi}{16} \tau_m \left(\frac{D^4 - d^4}{D} \right)$$

$$109.1 \times 10^4 = \frac{\pi}{16} \tau_m \left(\frac{(117)^4 - (52)^4}{117} \right)$$

$$\tau_m = \frac{109.1 \times 10^4}{302205.615} = 3.61 \text{ Mpa}$$

Hence design of muff is safe as the design shear stress value for the muff is lower then the permisible shear stress value of the muff.

Design of key :-

We know that the length of coupling key

$$l = \frac{L}{2} = \frac{182}{2} = 91 \text{ mm}$$

For 52 mm standard dia of shaft width $w = 16$ mm & thickness 10 mm to age 6 : 12 by design data book S.M.d Jalaludin.

Now let us check the shear stress & crushing stress of the key.

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

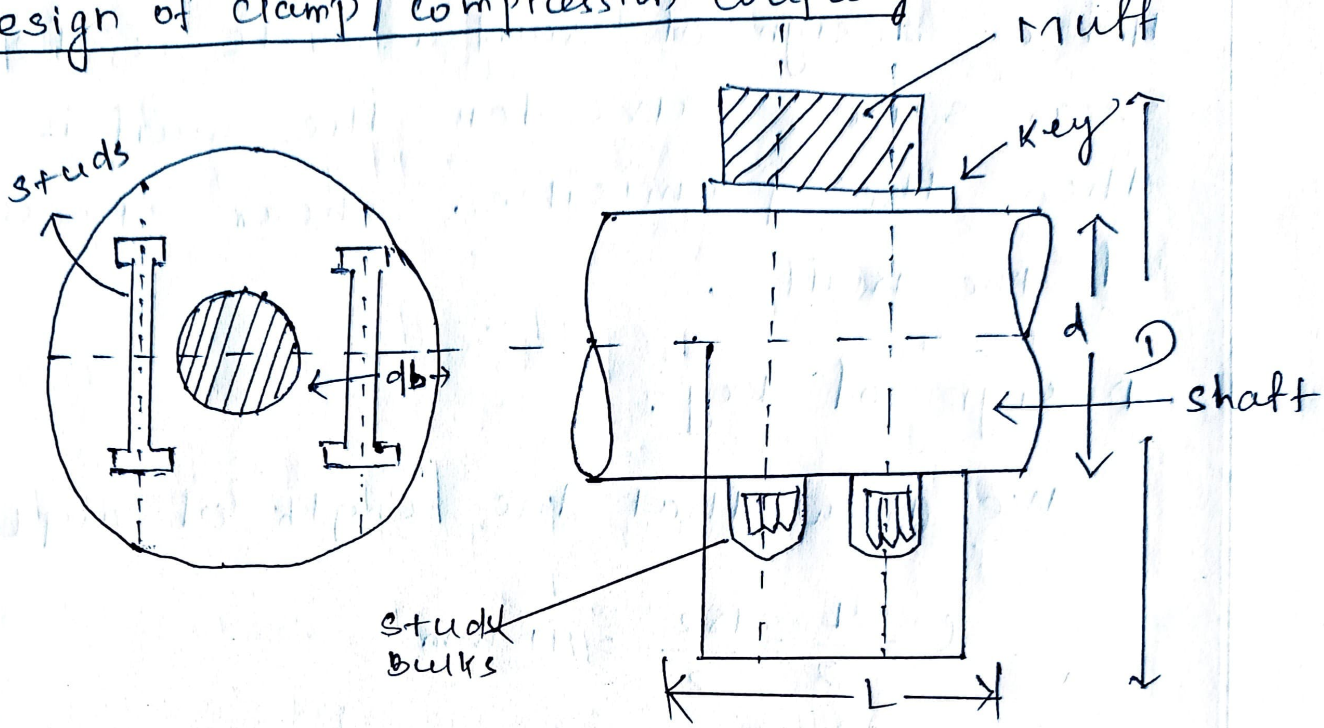
$$109.1 \times 10^4 = (91 \times 16) \times \tau \times \frac{52}{2}$$

$$\tau = \frac{109.1 \times 10^4}{91 \times 16 \times 26} = 28.81 \text{ MPa}$$

$$\tau = 28.8 \text{ M.Pa} < 40 \text{ M.pa}$$

Key design is safe.

Design of clamp/ compression coupling



→ It also a rigid type of coupling and known as clamp or compression coupling. It is also known as split muff coupling.

→ In this case the muff/ sleeve is made into two half and are bolted together as shown in fig.

→ The half of the muff are made up of cast iron the shaft ends to be point together are attached by means of this coupling. A single keyway is made in both the shaft. One half of the muff is fixed from bellow and other half of muff is fixed from the above. Both these half are of muff is fixed from the above.

Both these half are held together by means of studs or bolts. A number of bolt can varry from two to six depending upon assembly. this type of coupling are used for heavy duty and moderate speed. Advantages of use this coupling is that the position of the shaft can't be change for assembly and dis assembly. The usual properton of this split muff coupling are

$$D = 2d + 13\,mm$$

$$L = 3.5d$$

Design of muff / key :-

The design of muff and the key are done in the similar manner as discussed for split muff coupling.

Design of coupling bolts :-

T = Torque transmitted by the shaft

d = Dia of shaft

d_b - Root or effective dia of bolt

n - no. of bolts

σ_t = permissible tensile stress for the bolt material,

μ = co-efficient of friction bet(n) the muff & shaft

Force exerted by each bolt is

$$\sigma_t = \frac{F}{A_b}$$

$$F = \sigma_t \times \frac{\pi}{4} (d_b)^2$$

Force exerted by each bolt is

$$\sigma_t = \frac{F}{A_b}$$

$$F = \sigma_t \times \frac{\pi}{4} (d_b)^2$$

Force exerted by the bolt on each side of shaft :-

$$F = \sigma_t \times \frac{\pi}{4} (d_b)^2 \times \frac{n}{2}$$

Let p be the pressure on the shaft and muff surface due to this force. Hence the pr. is uniformly disturbted through the projected area Hence the pr. is uniformly distributed through the projected area Hence the pressure can be found out by

$$p = \frac{F}{A_p} = \frac{F}{\frac{1}{2} L \times d}$$

$$p = \frac{\sigma_t \times \frac{\pi}{4} (d_b)^2 \times \frac{n}{2}}{\frac{1}{2} L \times d}$$

Frictional force between each shaft and muff

$$F' = \mu \times p \times A$$

$$= \mu \times \frac{\sigma_t \times \frac{\pi}{4}(d_b)^2 \times \frac{n}{2}}{\frac{1}{2} \cancel{l} \times \cancel{d}} \times \frac{1}{2}\pi \cancel{d} \times \cancel{l}$$

$$= F' = \mu \frac{\pi^2}{8} \sigma_t (d_b)^2 n$$

Torque transmitted by the coupling:-

$$T = F' \times \frac{d}{2}$$

$$= \mu \frac{\pi^2}{8} \sigma_t (d_b)^2 n \times \frac{d}{2}$$

$$\boxed{T = \frac{\pi^2}{16} \mu \sigma_t n (d_b)^2 \times d}$$

Q. Design a clamp coupling to transmit 30 K.W at 100 rpm. The allowable shear stress for the shaft and key material is 40 M.p.a and no of bolts connecting the to halves are six. The permisibble tensile stress for the bolts is 70 M.p.a. The co-efficients of friction between the muff and the shaft surface may be taken as 0.3 respectively.

Given data

$P = 30$ K.W $= 30 \times 10^3$ W, $N = 100$ rpm

$\tau = 40$ Mpa $= 40$ N/mm², $n = 6$

$\sigma_t = 70$ Mpa $= 70$ N/mm², $\mu = 0.3$

(i) Design for shaft (d):

Let d = Diameter of shaft

we know that the torque transmitted by the shaft.

$$T = \frac{P \times 60}{2\pi N}$$

$$= \frac{30 \times 10^3 \times 60}{2\pi \times 100} = 2865 \text{ N.m}$$

$$= 2865 \times 10^3 \text{ N.mm}$$

We also known that the torque transmitted by the shaft (T).

$$2865 \times 10^3 = \frac{\pi}{16} \times \tau \times d^3 = \frac{\pi}{16} \times 40 \times d^3$$

$$= 7.86\, d^3$$

$$d^3 = \frac{2865 \times 10^3}{7.86} = 71.62 \text{ mm} \cong 75 \text{ mm}$$

(ii) Design for muff:

we know that diameter of muff

$$D = 2d + 13 \text{ mm}$$

$$= 2 \times 75 + 13 = 163$$

length of the muff

$$L = 3.5d = 3.5 \times 75$$
$$= 262.5 \text{ mm}$$

(iii) Design for key :-

The width and thickness of the key for a shaft diameter of 75 mm are as follows.

width of key, $w = 22$ mm

thickness of key, $t = 14$ mm.

length of key = total length of muff
$= 262.5$ mm.

(iv) Design for bolts :-

Let d_b = Root or core diameter of bolt

we know torque transmitted (T)

$$2865 \times 10^3 = \frac{\pi^2}{16} \times \mu (d_b)^2 \sigma_t \times n \times d$$

$$= \frac{\pi^2}{16} \times 0.3 \times (d_b)^2 \times 70 \times 6 \times 75$$

$$= 5829.23 (d_b)^2$$

$$d_3 = 22.1 \text{ mm}.$$

Nominal diameter of the bolt is 27 mm.

Design a closed coil helical spring :

spring : A spring is defined as an elastic body, whose function is to deform with the application of the load and recovered to the original shape after load is removed.

Some important application of spring :

→ To absorbed or control the energy due to shock or variation. coil spring, vibration depends applied force in case of break or clutches etc.

→ To measure the forces :

$F = kx$ K = spring constant

$\boxed{K = F/x}$ x = Distance

F = Force

→ To store energy as in the case of watches and toyes etc.

→ To control the motion.

Material of the spring ÷

→ The material of the spring should have following properties.

(I) High fitigue strength
(II) High ductility
(III) High resillence
(iv) creap resistance (Temp resistance)

The materials required foo the spring largly depend upon the service for which they are used ÷

Ex ÷ (i) severe service
(ii) Average service
(iii) Lilglet service

① severe service ÷

severe service means rapid contineous loading in which the ratio between minimum load and maximum load will be half and less than half.

Mathmetically ÷

$$\frac{P_{min}}{P_{max}} = \frac{1}{2} \text{ or } < \frac{1}{2}$$

Ex ÷ Automotive value spring

(ii) Average service :-

It includes the same stress range as in case of severe service but only inter midiate oporetoun.

Ex :- Governer spring, Automobile supersion spring.

(iii) Hight service :-

It include spring subjected to statet load or load that very frequently.

Ex :- safety value spring.

Spring are made from the following important :-

(i) Oil tempered (carbon steel)

(0.6% - 0.7)% carbon

(0.6% - 1%) Mn

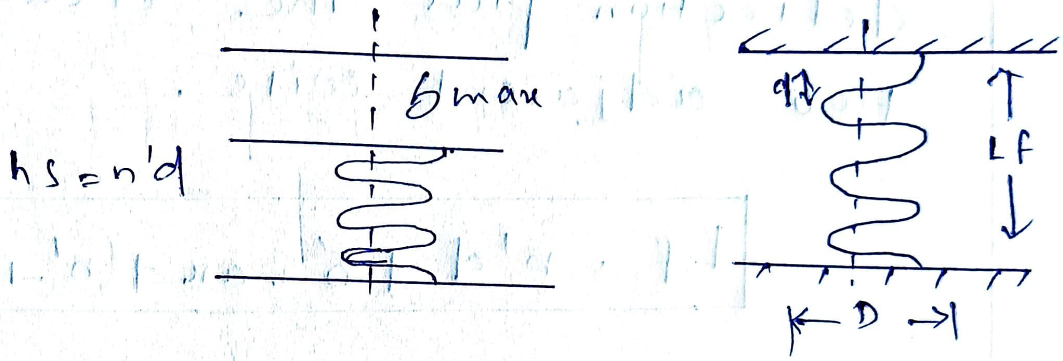
(ii) spring can also made of non-ferrous material.

Ex :- Aluminium, Brass, non metal, Phosphorous, Sulphur, Berillium etc.

Standard wire size :

SWG	Dia (mm)
7/0	12.70
6/0	11.785
5/0	10.973

Terms used in the spring :



Solid length (h_s)

When the compression spring is compressed by the application of load until the coils from contact to each other ; the spring is said to be solid length.

i.e : $\boxed{h_s = n'd}$

n' = no of coils

d = dia. of wire

Free length (L_f) :-

the free length of a compression spring as shown in fig is the length of the spring in the free or unloaded condition.

→ Free length is otherwise can be equal to the solid length plus the maximum deflection plus the clearance betn two adjacent coils.

$$\boxed{L_f = n'd + \delta_{max} + (n'-1) \times 1\,mm}$$

3/ Spring Index (C) :-

Spring index can be defined as the ratio between the mean dia of the coil to the dia of the wire.

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

4/ Spring rate (K) spring stiffness or spring constant :-

It is defined as the load required per unit deflection of the spring

i.e :- $\boxed{K = \frac{F}{x} = \frac{F}{\delta}}$

5/ <u>Pitch (p)</u>

$$\boxed{p = \frac{L_f}{n' - 1}}$$

→ pitch of the coil is defined as the axial distance between adjacent coils in an uncomposed state.

STRESSES IN HELICAL SPRINGS OF CIRCULAR WIREL :

Consider a helical compression spring made of circular wire & subjected to an axial load W as shown in fig.

Let D = Mean Dia of the spring coil,

d = wire dia.

n = No. of active coils

G = Modulus of rigidity for the spring material.

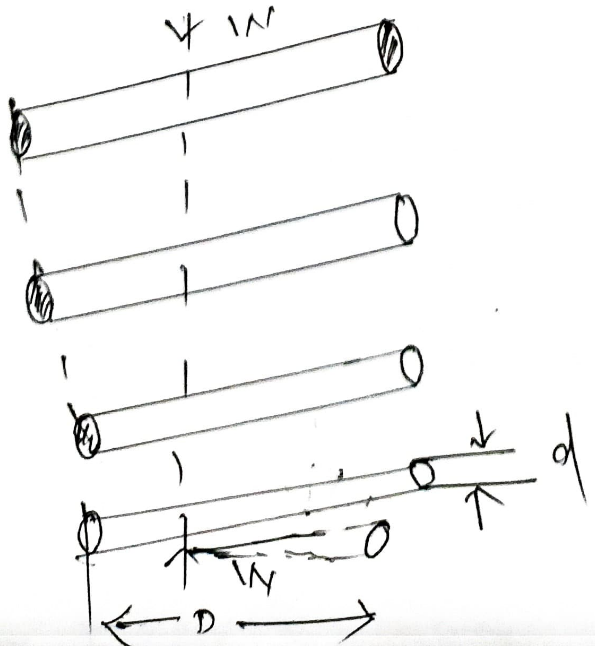
W = Axial load on the spring

τ = Maximum shear stress induced in the wire.

c = spring Index

p = pitch of the coils

δ = Deflection of the spring as a result of axial load 'W'



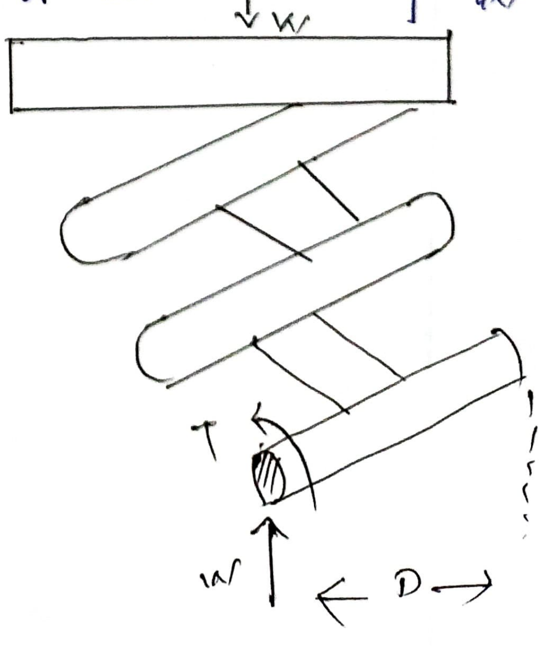


Fig-1 Axially loaded helical spring

Fig-2 : FBD showing the spring wire is s.t torsional shear and direct shear.

Let us consider a part of the compression spring as shown in fig (2)

Load W tends to rotate the wire due to the twisting moment (T) set up in the wire. Thus torsional shear stress is induced in the wire.

We know that,

$$T = W \times \frac{D}{2} = \frac{\pi}{16} \times \tau_1 \times d^3$$

$$\Rightarrow \tau_1 = \frac{8WD}{\pi d^3}$$

In addition to this torsional shear stress (τ_1) induced in the wire, the following stresses also act on the wire.

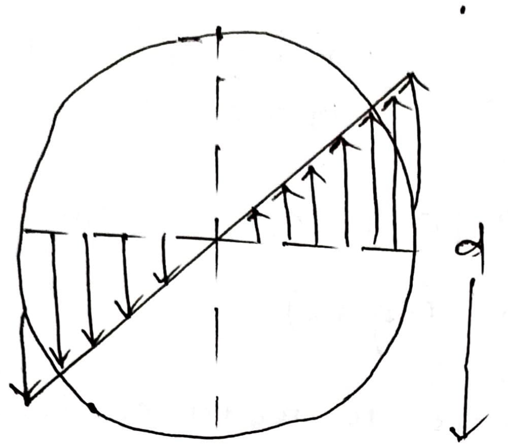
01. Direct shear stress due to load W.

02. Stress due to curvature of wire.

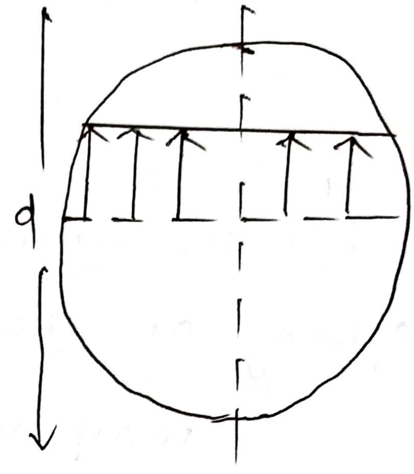
So, $$\tau_2 = \frac{\text{Load}}{\text{C/s Area of wire}}$$

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

$$\tau_2 = \frac{4W}{\pi d^2}$$



(a) Torsional shear stress diagram



(b) Direct shear stress diagram.

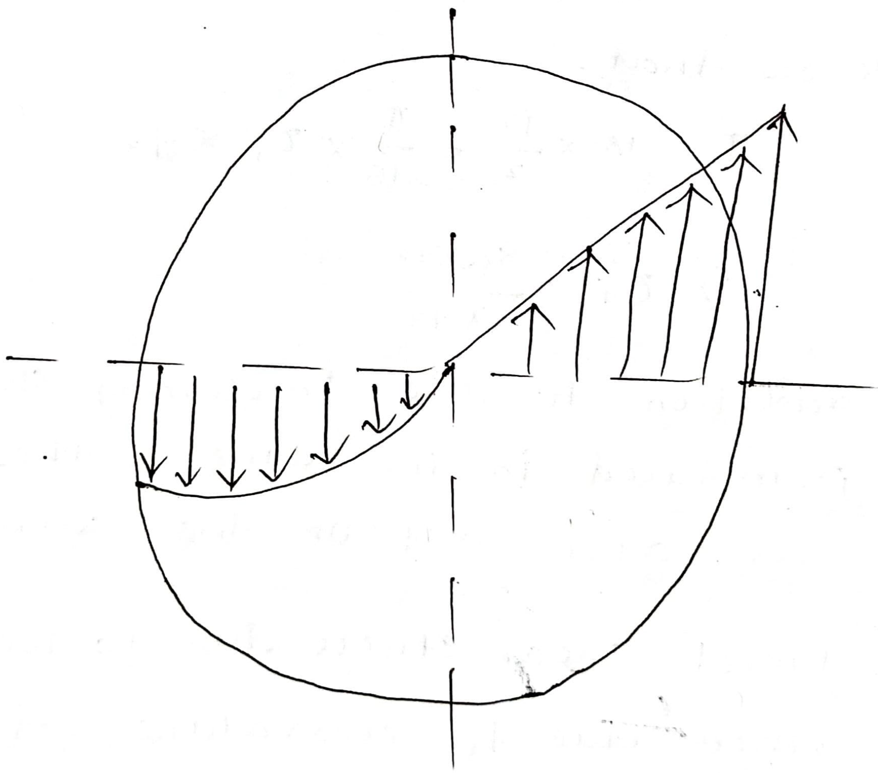


fig :① (Resultant torsional shear, direct shear & curvature shear stress digram)

We know that

Resceltant shear stress induced in the wire.

$$\tau = \tau_1 \pm \tau_2$$

$$= \frac{8WD}{\pi d^3} \pm \frac{4W}{\pi d^2}$$

+ve sign used for Inner edge of the wire.

+ve sign used for outer edge of the wine.

Since the stress is maximum of the inner edge of the wine, therefore

Maximum shear stress induced in the wire.

$$= \tau_1 + \tau_2$$

$$= \frac{8WD}{\pi d_3} + \frac{4W}{\pi d^2}$$

$$= \frac{8WD}{\pi d^3}\left(1 + \frac{d}{2D}\right)$$

$$= \frac{8WD}{\pi d^3}\left(1+\frac{1}{2C}\right) \quad \left(\because C = \frac{D}{d}\right)$$

$$= \frac{8WD}{\pi d^3} \times K_s$$

$$\boxed{\tau_{max} = K_s \times \frac{8WD}{\pi d^3}}$$

where $K_s \rightarrow$ shear stress factor $= 1 + \frac{1}{2C}$

If we consider the effects of both direct shear as well as curvature of the wire, a wahl's stress factor (K) introduced by A.M. Wahl may be used.

Resultant stress diagram is shown in fig. (1)

i.e :-

$$\boxed{\begin{aligned} \tau_{max} &= K \times \frac{8WD}{\pi d^3} \\ K &= \frac{4C-1}{4C-4} + \frac{0.615}{C} \end{aligned}}$$

DEFLECTION OF HELICAL SPRINGS OF CIRCULAR WIREL ÷

Axial deflection of the spring (s)

$$s = \theta \times \frac{D}{2}$$

we know that

$$\frac{T}{I_p} = \frac{\tau}{(D/2)} = \frac{G\theta}{L}$$

$$\Rightarrow \theta = \frac{TL}{GI_p} \quad \text{------------ (1)}$$

where $I_p \rightarrow$ polar moment of Inertia of the spring wire.

$$I_p = \frac{\pi}{32} d^4$$

d = Dia of spring wire.

G = Modulus of Rigidity, for the material of spring wire

substituting value of I_p in the above eq (1); we may get the following expression.

$$\theta = \frac{TL}{I_p G}$$

$$= \frac{\left(W \times \frac{D}{2}\right)(2\pi R n)}{\frac{\pi}{32} d^4 G}$$

$$= \frac{\frac{W \times D}{2} \times (\pi D n)}{\frac{\pi}{32} d^4 G}$$

$$= \frac{W \times \cancel{\pi} D^2 n}{2} \times \frac{32}{\cancel{\pi} d^4 G}$$

$$\boxed{\theta = \frac{16 W D^2 n}{G d^4}} \text{ ——— (11)}$$

Substituting the value of θ in the eq$^{(7)}$

we get,

$$\delta = \theta \times \frac{D}{2}$$

ie,

$$\delta = \theta \times \frac{D}{2}$$

$$= \frac{16 W D^2 n}{G d^4} \times \frac{D}{2}$$

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

$$= \frac{8Wn}{Gd} \times \left(\frac{D}{d}\right)^3$$

$$= \frac{8Wn}{Gd} \times (C)^3$$

$$\boxed{\delta = \frac{8WC^3 n}{Gd}}$$

$$\Rightarrow \boxed{K = \frac{W}{\delta} = \frac{Gd}{8C^3 n} = \frac{Gd^4}{8D^3 n}}$$

where

K = stiffness of spring / spring rate

δ = Deflection of spring