LECTURE NOTES ON

DESIGNOFMACHINEELEMENTS 4th SEMESTER

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Introduction to the Design Process

Accreditation Board for Engineering and Technology (ABET)

Definition of Design

- ***Engineering design*** is the process of devising a system, component, or process to meet desired needs.
- It is a decision-making process (often iterative), in which the engineering sciences and mathematics are applied to convert resources optimally to meet a stated objective.
- Among the fundamental elements of the design process are the establishment of objectives and criteria, synthesis, analysis, construction, testing and evaluation.

Joseph Shigley (*Mechanical Engineering Design*)

Definition of Design

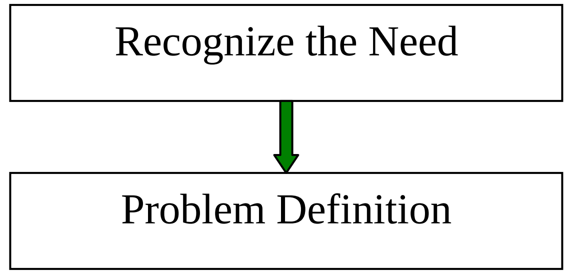
- ***Mechanical design*** means the design of components and systems of a mechanical nature—machines, products, structures, devices and instruments.
- For the most part mechanical design uses mathematics, materials, and the engineering-mechanics sciences.
- Additionally, it uses engineering graphics and the ability to communicate verbally to clearly express your ideas.
- ***Mechanical engineering design*** includes all mechanical design, but it is a broader study because it includes all the disciplines of mechanical engineering, such as the thermal fluids and heat transfer sciences too.
- Aside from the fundamental sciences which are required, the first studies in *mechanical engineering design* are in *mechanical design*, and that is the approach taken in this course.

Steps of the Design Process

1. Recognize the Need

- The first step is to establish the ultimate purpose of the project. Often, this is in the form of a general statement of the client's dissatisfaction with a current situation.
- example – "There is too much damage to bumpers in low-speed collisions."
- This is a general statement that does not comment on the design approach to the problem. It does not say that the bumper should be stronger or more flexible.
- Recognition and phrasing of the need are often very creative acts because the need may only be a sensing that something is not right. For this reason, sensitive people are generally more creative.
- example – the need to do something about a food packaging machine may be indicated by the noise level, variation in package weights, or by slight but perceptible variations in the quality of the packaging.

2. Problem Definition



- This is one of the most critical steps of the design process.
- There is an iteration between the definition of the problem and the recognition of need. Often the true problem is not what it first seems.
- The problem definition is more specific than recognizing the need. For instance, if the need is for cleaner air, the problem might be that of reducing the dust discharge from power-plant stacks, or reducing the quantity of irritants from automotive exhausts, or means for quickly extinguishing forest fires.
- The problem definition must include all the specifications for the thing that is to be designed. **Anything which limits the designer's freedom of choice is a specification.**
- It is imperative to write a formal problem statement which expresses what the design is to accomplish

  include:
  - objectives and goals
    (musts, must nots; wants, don't wants)
  - constraints
  - criteria used to evaluate the design

- Example: Mobile Vehicle

  Design a vehicle which can maneuver in an indoor environment. The vehicle will be operated via remote control and must be able to:

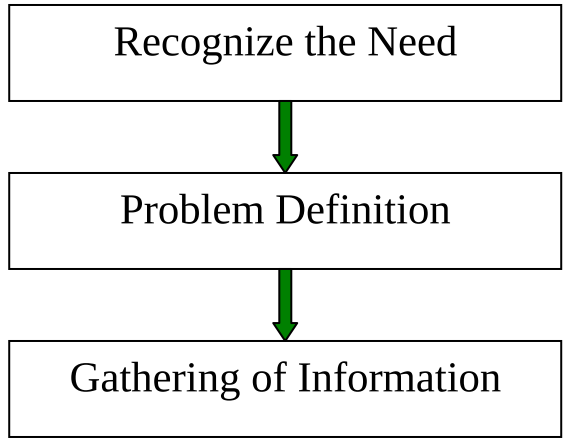
1) Travel up to a speed of 7 ft/sec on a flat, horizontal, dry, bare concrete surface.
2) Climb 5” high stairs at speeds up to 2 ft/sec.
3) Carry a payload of at least 20 lbs.
4) Fit through doorways.
5) Cross obstacles up to 20" high and up to 24" across within 20 seconds.
6) Climb a slope of up to 30 degrees and cross side slopes up to 20 degrees.
7) Rotate with zero turning radius.
8) Travel in any direction.
9) Total vehicle weight should be less than 275 lbs.

- Design considerations (in no particular order)

strength	**cost**	flexibility
reliability	safety	**control**
thermal properties	**weight**	**stiffness**
corrosion	life	surface finish
wear	noise	lubrication
friction	**styling**	maintenance
ergonomics	**shape**	**volume**
utility	**size**	liability
manufacturability	**speed**	feedrate
	ectetera	

Note: Design considerations in bold might be pertinent to the design project in EML2322L.

3. Gathering of Information



- Often, either no information is easily found, or there is an abundance of information
- Never-ending process for the best design engineers
- Info sources:

  textbooks
  trade journals & magazines
  technical reports from government sponsored R&D
  company catalogs, web pages and technical personnel
  handbooks
  company reports
  patents
  people

- Problems in gathering information:

  LAZINESS

  Where to find it?

  How to get it?

  How **accurate & credible** is the information?

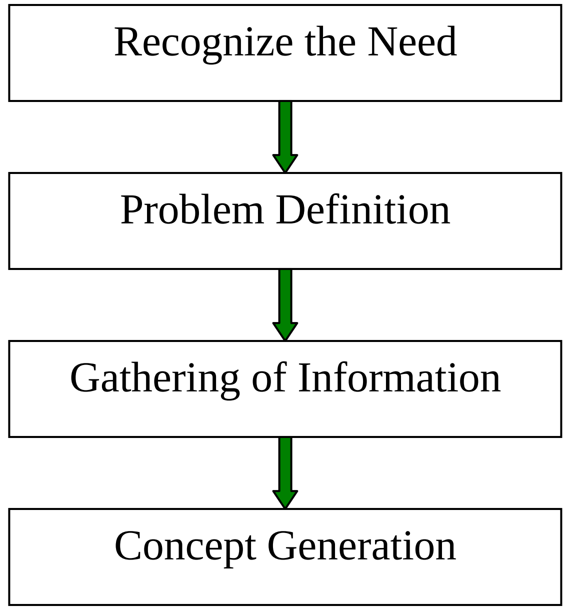
  How should the information be interpreted for my needs?

  When do I have enough information?

  What decisions result from the information?

  PLAGIARISM (integrity = giving others credit for their ideas)

4. Concept Generation



- This is the most creative part of the design process.
- Store ideas in a design notebook.
- Some approaches to concept generation:

  - **adaptation**

    a solution of a problem in one field is applied to a similar problem in another field (wine press → printing press → pistol grip)

  - **analogy**

    obstacle avoidance similar to potential fields

  - area thinking

    improve an existing product by concentrating on one of its important characteristics (cost, performance, function, appearance, safety, etc.)

  - **brainstorming**

    group of people who are familiar with the general nature of the problem;
    everyone says what comes to mind
    rules: (1) no judgements; (2) the more unconventional the better; (3) the more ideas the better

  - involvement

    visualize yourself as being part of the mechanism

- functional synthesis

  divide the system into subunits
  describe each subunit by a complete list of functional requirements
  list all the ways the functional requirements of each subunit can be realized
  study all combinations of partial solutions

Can Opener

Part	Function	Realization
Subunit 1	1. Separate metal	1. Shearing
		2. Tearing
		3. Fatigue
		4. Melting
		5. Drawing thin
		6. Chemical erosion
Subunit 2	1. Apply power	1. Hand
		2. Electric motor
		3. Hot wire
		4. Hydraulic motor
		5. Flame
		6. Chemical reaction
		7. Mechanical vibration
		8. Laser
	2. Position	1. Bring can to opener
		2. Bring opener to can
		3. Have opener built on can

- try inversion

    try reversing the ordering of things; i.e. an inversion is produced with an electric motor by holding the rotor stationary and permitting the field windings to rotate

- change the normal position and character of things

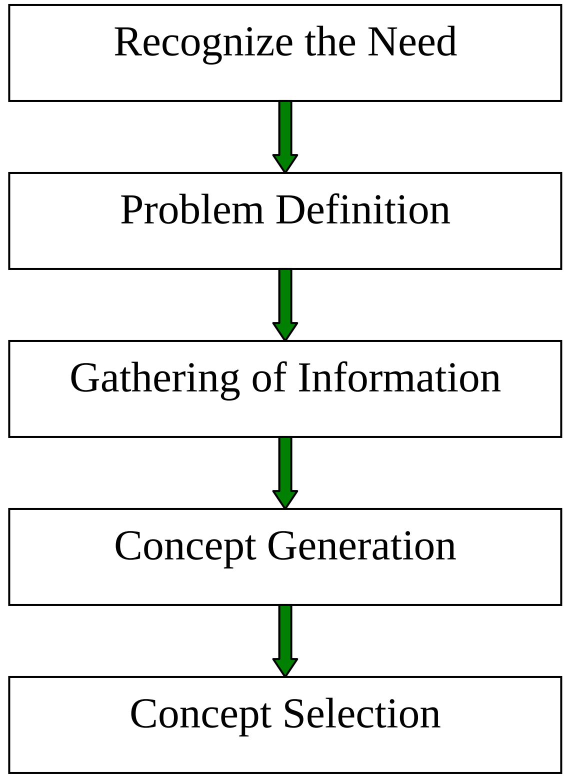
    if it operates horizontally, try operating it vertically. If it's round, try making it square. For example, doors hinged at top or bottom, a horizontal drill press, etc.

- **talk it over**

    If the designer has followed these suggestions thus far, he/she is now quite familiar with the problem. Many solutions have been found but none are quite satisfying. Having worked to this point, the designer's mind is in a receptive condition and will instantly recognize a solution. The problem is to bridge a gap between two groups of ideas—to make an association of ideas. It is generally conceded that this association occurs by pure chance. This event is most likely to occur when the problem is being discussed with another person or group of persons.

5. Concept Selection

- form **decision matrix** to unbiasedly evaluate different ideas based on a weighted set of objectives the design team decides are important for the solving the problem



Decision matrix for the design of a crane hook

CRANE HOOK			Welded Plates			Riveted Plates			Cast Hook		
Objective	Weighting Factor	Parameter	Mag.	Score	Value	Mag.	Score	Value	Mag.	Score	Value
Material Cost	0.10	$	2500	8.8	0.9	2500	8.8	0.9	2200	10.0	1.0
Manufacturing Cost	0.20	$	1500	8.0	1.6	1200	10.0	2.0	2400	5.0	1.0
Manufacturing Time	0.10	hours	40	6.3	0.6	25	10.0	1.0	50	5.0	0.5
Durability	0.15	experience	great	10	1.5	good	8	1.2	good	8	1.2
Reliability	0.30	experience	good	8	2.4	great	10	3.0	okay	6	1.8
Repairability	0.15	experience	good	8	1.2	great	10	1.5	fair	4	0.6
Overall value					**8.2**			**9.6**			**6.1**

Qualitative Score Assignments:	
great	10
good	8
okay	6
fair	4
poor	2

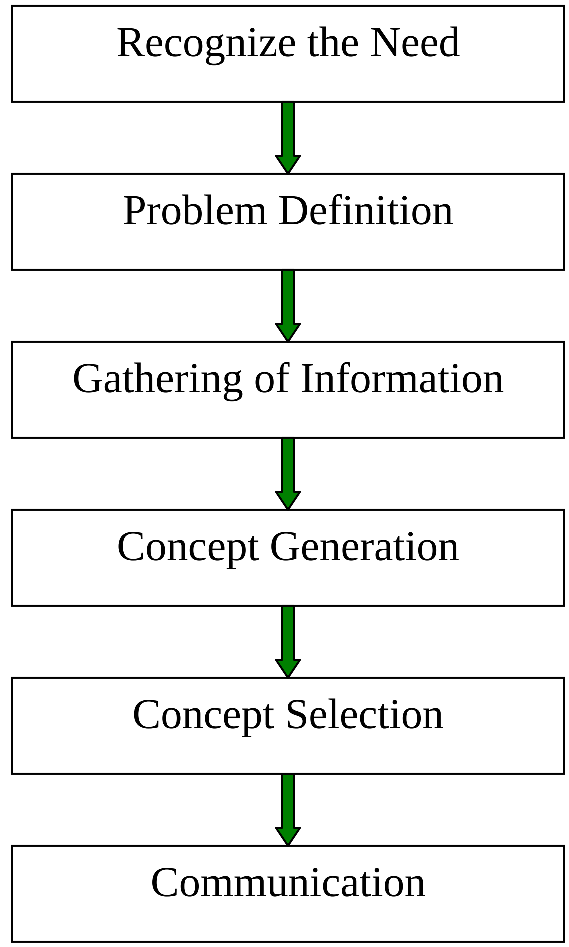
[CRANE PHOTO]

[CRANE HOOK PHOTO]

[BAD DAY AT WORK FOR CRANE OPERATOR PHOTO]

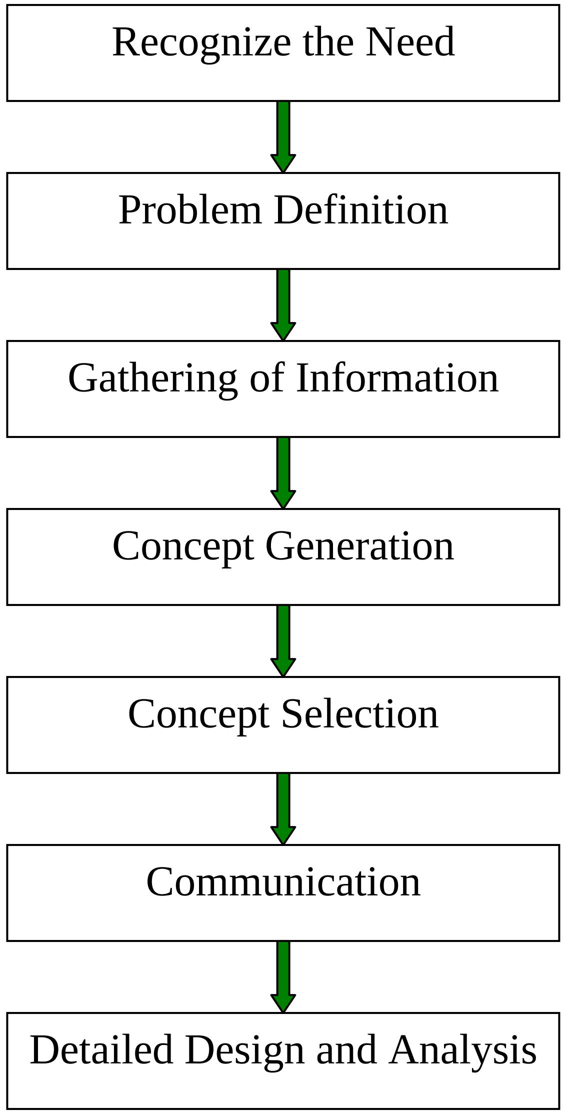
[CLICK HERE FOR THE SAME EXAMPLE WITH PRINTABLE (AS OPPOSED TO DIGITAL) COMMENTS]

6. Communication of the Design



- The purpose of the design is to satisfy the needs of the client.
- Designer must provide oral presentations and written design reports.
- Continuous communication is important in order to avoid surprises.
- Many great designs and inventions have been lost simply because the originator was unable or unwilling to explain his/her accomplishments to others.
- There are only 3 forms of communication available to us: written, oral and graphical. The successful engineer will be technically competent and versatile in all three. Competency only comes from practice.
- Ability in writing can be acquired by writing letters, reports, memos, and papers. It doesn't matter whether the articles are published or reviewed—the practice is the important thing. Ability in speaking can be obtained in educational, fraternal, civic, church and professional activities. To acquire drawing ability, pencil sketching should be employed to illustrate every idea possible. **CAD work should complement this, not replace it.**
- Importance of sketches, drawings, visual aids, computer graphics and models in the communications process.

7. Detailed Design and Analysis



- The principal goal of your engineering studies is to enable you to create mathematical models which accurately simulate the real physical world.
- All real physical systems are complex. Creating a mathematical model of the system means we are simplifying the system to the point that it can be analyzed. The terms *rigid body* and *concentrated force* are examples. The rule in making such *assumptions*, is that, in creating the model, the model must be meaningful—i.e. a good and appropriate model given the design constraints involved.
- The nature of the problem, its economics, the computational facilities available and the ability and working time of the engineer, all play a key role in the formulation of the model.
- **The designer’s time investment typically increases exponentially with regard to model accuracy.**

8. Prototype Development and Testing

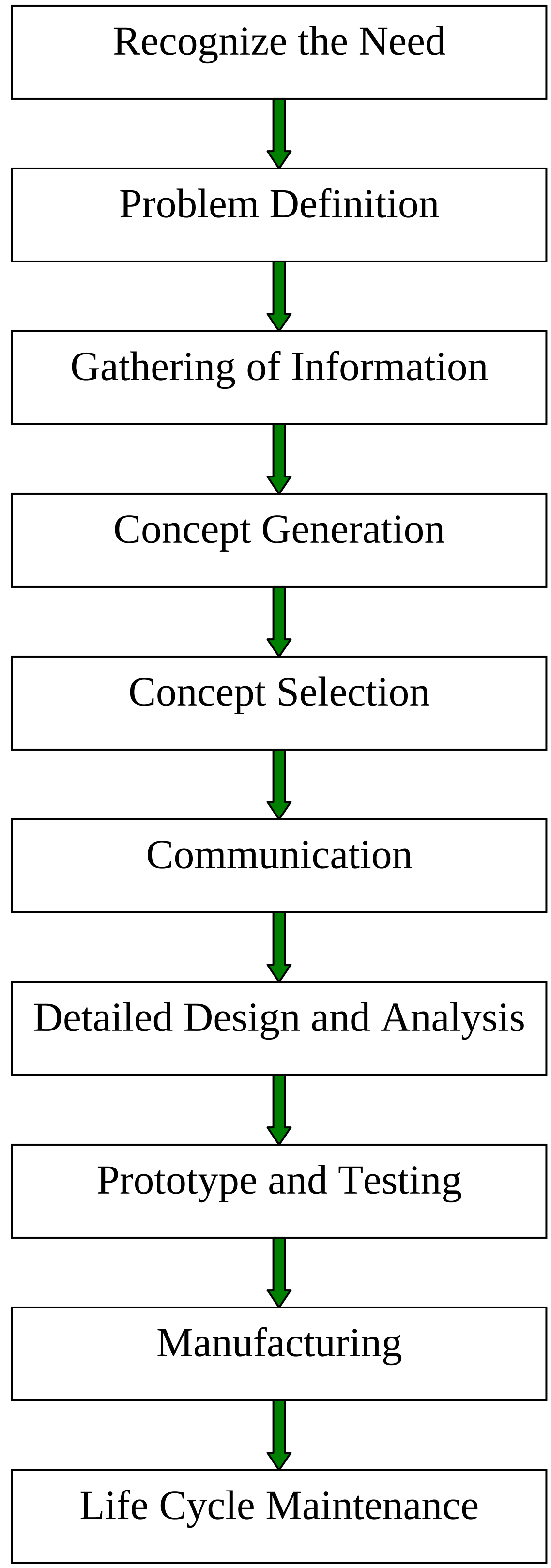
- Initial exposure in EML2322 lab and design project.

9. Manufacturing

- Initial exposure in EML4321 course.

10. Life Cycle Maintenance

- Learned from experience and industry standards.



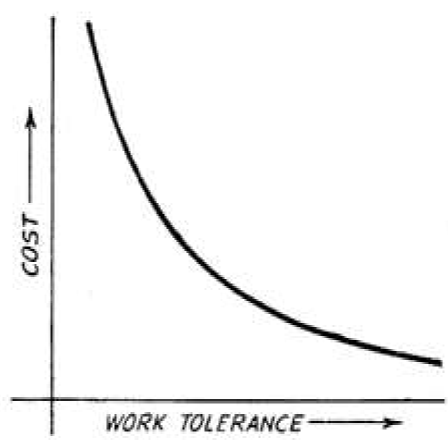
SYSTEM OF LIMITS, FITS, TOLERANCE AND GAUGING

INTRODUCTION:

It is well known fact that no two things in the nature can be identical, they may be found to be closely similar. This is true of production of component parts in engineering also. We know that every process is a combination of three elements, man, machine and material. A change in any one of these will constitute a change in the process. All these elements are subjected to inherent and characteristic variations.

Generally, in engineering, any component manufactured is required to fit or to match with some other component.

If a machine is under control, i.e. no assignable causes of variation exist, and then the resultant frequency distribution of dimension produced will be roughly in the form of normal curve, i.e. 99.7% parts will be within $\pm 3\sigma$ limits of means setting



The value of σ depends upon the machine used to produce a component. If value of σ has to be used reduced, then precision machines have to be used produces the component having less variation in dimensions. It is thus important to note that the cost of production keeps on increasing tremendously for very precise tolerance as shown in above **fig**, as the tolerance approaches zero, the task of achieving it becomes enormous and finally impossible .in general, tolerance vs. fabrication cost is hyperbolic curve.

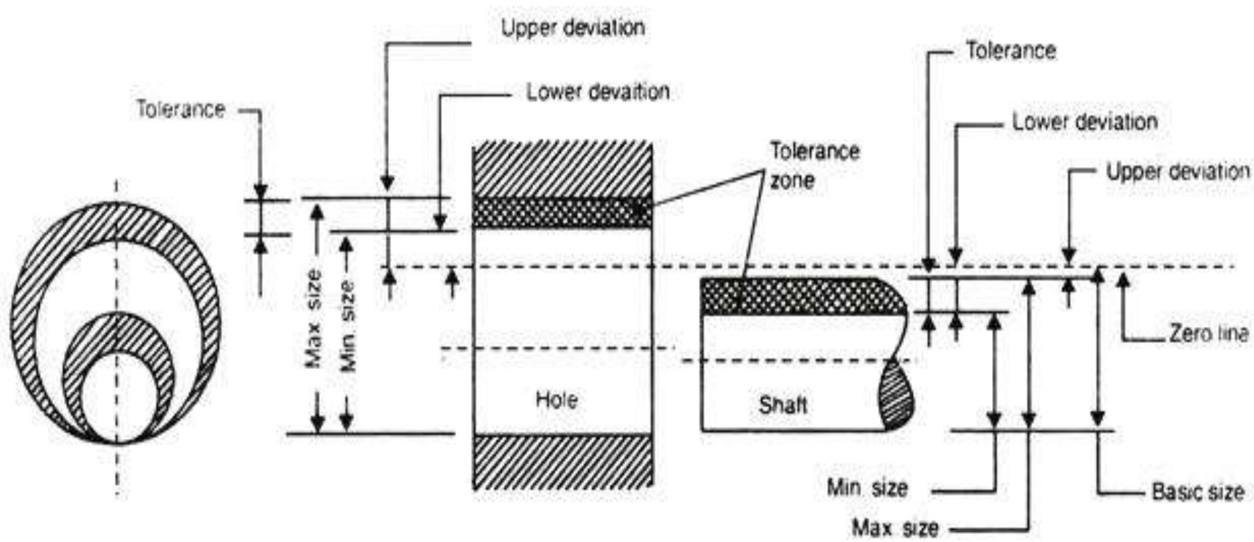
➢ LIMITS:

The maximum and minimum permissible sizes within which the actual size of a component lies are called limits.

- Limits are fixed with reference to the basic size of that dimension.
- Upper limit (The high limit) for that dimension is the largest size permitted and the low limit is the smallest size permitted for that dimension.

TERMINOLOGY

The terminology used in fits and tolerances is shown in Fig below. The important terms are



Basic size: It is the exact theoretical size arrived at by design. It is also called nominal size.

Actual size: The size of a part as may be found by measurement.

Maximum limit of size: The greater of the two limits of size.

Minimum limit of size: The smaller of the two limits of size.

Allowance: It is an intentional difference between maximum material limits of mating parts. It is a minimum clearance or maximum interference between mating parts.

Deviation: The algebraic difference between a size (actual, maximum, etc.) and the corresponding basic size.

Actual deviation: The algebraic difference between the actual size and the corresponding basic size.

Upper deviation: The algebraic difference between the maximum limit of size and the corresponding basic size.

Upper deviation of hole = ES (& art Superior)

Upper deviation of shaft = es

Lower deviation: The algebraic difference between the minimum limit of size and the corresponding basic size.

Lower deviation of hole = El (Ecart Inferior)

Lower deviation of shaft = ei

Upper deviation Lower deviation + Tolerance

Zero line: It is the line of zero deviation and represents the basic size.

Tolerance zone: It is the zone bounded by the two limits of size of the parts and defined by its magnitude, i.e. tolerance and by its position in relation to the zero line.

Fundamental deviation: That one of the two deviations which is conveniently chosen to define the position of the tolerance zone in relation to zero line, as shown in fig below.

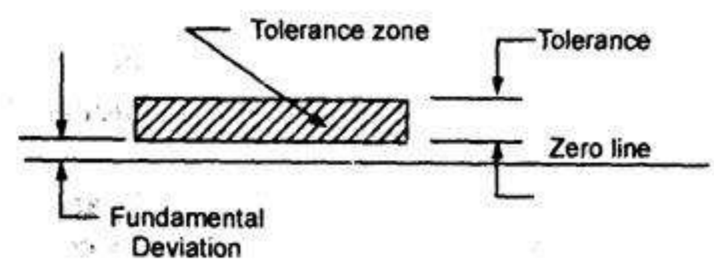


Fig: Disposition of fundamental deviation and tolerance zone with respect to the zero line

Basic shaft: A shaft whose upper deviation is zero.

Basic hole: A hole whose, lower deviation of zero.

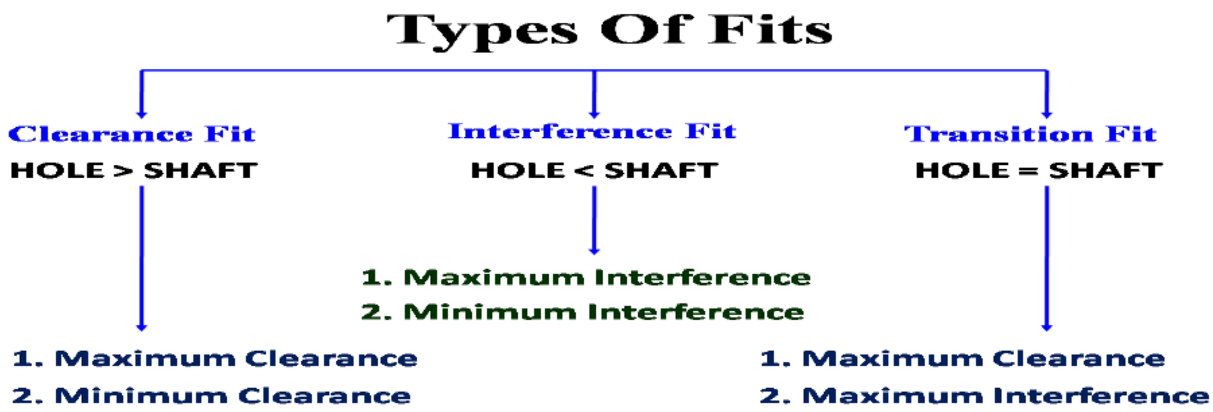
Clearance: It is the positive difference between the hole size and the shaft size.

Maximum clearance: The positive difference between the maximum size of a hole and the minimum size of a shaft.

Minimum clearance: The positive difference between the minimum size of a hole and the maximum size of a shaft.

➢ FITS

When two parts are to be assembled, the relation resulting from the difference between their sizes before assembly is called a fit. A fit may be defined as the degree of tightness and looseness between two mating parts.



(i) Clearance Fit:

This means there is a gap between the two mating parts. Let's see the following schematic representation of clearance fit. The diameter of the shaft is smaller than the diameter of the hole. There is a clearance between the shaft and the hole. Hence the shaft can easily slide into the hole.

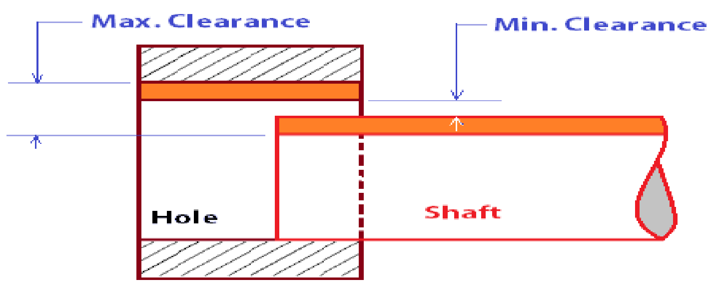


Figure: Clearance fit

In clearance fit the difference between the maximum size of the hole and the minimum size of the shaft is known as the **Maximum clearance** and the difference between the minimum size of the hole and the maximum size of the shaft is known as the **Minimum clearance**.

Clearance fit can be sub-classified as follows:

Loose Fit: It is used between those mating parts where no precision is required. It provides minimum allowance and is used on loose pulleys, agricultural machineries etc.

Running Fit: For a running fit, the dimension of shaft should be smaller enough to maintain a film of oil for lubrication. It is used in bearing pair etc. An allowance 0.025 mm per 25 mm of diameter of boring may be used.

Slide Fit or Medium Fit: It is used on those mating parts where great precision is required. It provides medium allowance and is used in tool slides, slide valve, automobile parts, etc.

EXAMPLE:

Question: A spindle slides freely in a bush. The basic size of the fit is 50 $x10^{-3}$ mm. If the tolerances quoted are 0 +62 for the holes and -80 +180 for the shaft, find the upper limit and lower limit of the shaft and the minimum clearance.

Solution: Tolerances are given in units of one thousandth of millimeter, so the upper limit of the hole will be 50.062 mm and lower limit for the hole is the same as the basic size of 50.000 mm.

The shaft upper limit will be $(50.000 – 0.080) \times 10^{-3} = 49.92x10^{-3}$ m

The shaft lower limit will be $(50.000 – 0.180) \times 10^{-3} = 49.82x10^{-3}$ m

The minimum clearance or allowance is $(50.000 – 49.920)\ \ 10^{-3} = 8x10^{-3}$ mm

(ii) Interference Fit:

There is no gap between the faces and there will be an intersecting of material will occur. In the following schematic representation of the Interference fit. The diameter of the shaft is larger than the hole diameter. There will be the intersection of two mating components will be occurred. Hence the shaft will need additional force to fit into the hole.

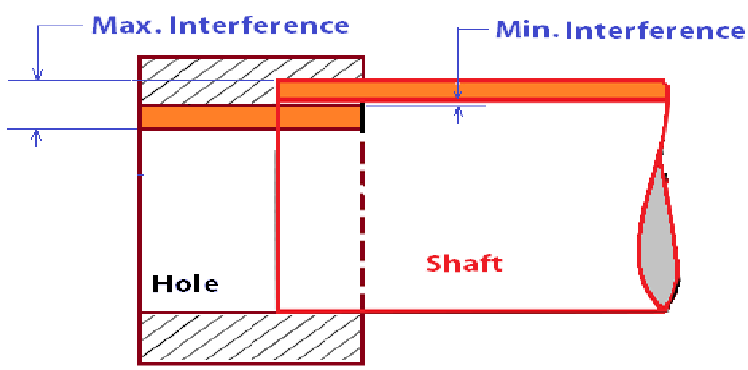


Figure: Interference Fit

In Interference fit the difference between the maximum size of the shaft and the minimum size of the hole is known as the **Maximum Interference** and the difference between the minimum size of the shaft and the maximum size of the hole is known as the **Minimum Interference**.

The interference fit can be sub-classified as follows:

Shrink Fit or Heavy Force Fit: It refers to maximum negative allowance. In assembly of the hole and the shaft, the hole is expanded by heating and then rapidly cooled in its position. It is used in fitting of rims etc.

Medium Force Fit: These fits have medium negative allowance. Considerable pressure is required to assemble the hole and the shaft. It is used in car wheels, armature of dynamos etc.

Tight Fit or Press Fit: One part can be assembled into the other with a hand hammer or by light pressure. A slight negative allowance exists between two mating parts (more than wringing fit). It gives a semi-permanent fit and is used on a keyed pulley and shaft, rocker arm, etc.

EXAMPLE

A dowel pin is required to be inserted in a base. For this application H 7 fit for hole and a p 6 fit for the shaft are chosen. The tolerance quoted is 0 +25 for the hole and 26 +42 for the shaft. Find the upper and lower limits of the hole and also dowel pin, and the maximum interference between dowel pin and the hole. The basic size of the fit is $50 \text{x} 10^{-3}$ m.

Solution:

The upper limit for the hole will be $(50.000 + 0.025) \text{ x } 10^{-3} = 50.025\text{x}10^{-3}$m

The lower limit for the hole will be $(50.000 + 0)\ 50.000 \text{ x } 10^{-3} = 50\text{x}10^{-3}$ m

The upper limit for dowel pin will be $(50.000 + 0.042) \text{ x } 10^{-3} = 50.042\text{x}10^{-3}$ m

The lower limit for dowel pin will be $(50.000 + 0.026) \text{x}10^{-3} = 50.026 \text{ x } 10^{-3}$ mm

The maximum interference between dowel pin and the hole is

$(50.042 – 50.000) \text{ x } 10^{-3} = 0.042\text{x } 10^{-3}$ m $= 42\text{x}10^{-6}$ m

(iii) Transition Fit:

Transition fit is neither loose nor tight as like clearance fit and interference fit. The tolerance zones of the shaft and the hole will be overlapped between the interference and clearance fits. See the following schematic representation of the transition fit.

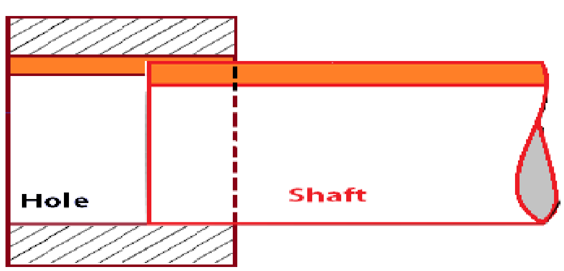


Figure: Transition Fit

Transition fit can be sub-classified as follows:

Push Fit: It refers to zero allowance and a light pressure (10 cating dowels, pins, etc.) is required in assembling the hole and the shaft. The moving parts show least vibration with this type of fit. It is also known as snug fit.

Force Fit or Shrink Fit: A force fit is used when the two mating parts are to be rigidly fixed so that one cannot move without the other. It either requires high pressure to force the shaft into the hole or the hole to be expanded by heating. It is used in railway wheels, etc.

Wringing Fit: A slight negative allowance exists between two mating parts in wringing fit. It requires pressure to force the shaft into the hole and gives a light assembly. It is used in fixing keys, pins, etc.

EXAMPLE:

For a particular application, an H 7 fit has been selected for the hole and a K 6 fit for the shaft. The tolerance quoted are 0 +25 for the hole and 12 +18 for the shaft. Find the upper limit and lower limit for the hole and also for bush. The basic size of fit is 50x10– 3 m.

Solution:

The upper limit for the hole will be $(50.000 + 0.025) \times 10^{-3} = 50.025 \times 10^{-3}$ m

The lower limit for the hole will be $(50.000 + 0) \times 10^{-3} = 50.000 \times 10^{-3}$ m

The upper limit for the bush will be $(50.000 + 0.018) \times 10^{-3} = 50.018 \times 10^{-3}$ m

The lower limit for the bush will be $(50.000 + 0.002) \times 10^{-3} = 50.002 \times 10^{-3}$m

SYSTEMS OF FITS:

A fit system is the systems of standard allowance to suit specific range of basic size. If these standard allowances are selected properly and assigned in mating parts ensures specific classes of fit.

There are two systems of fit for obtaining clearance, interference or transition fit. These are:

1. Hole basis system
2. Shaft basis system

1. **Hole Basis System:**

In the hole basis system, the size of the hole is kept constant and shaft sizes are varied to obtain various types of fits. In this system, lower deviation of hole is zero, i.e. the low limit of hole is same as basic size. The high limit of the hole and the two limits of size for the shaft are then varied to give desired type of fit. The hole basis system is commonly used because it is more convenient to make correct holes of fixed sizes, since the standard drills, taps, reamers and branches etc. are available for producing holes and their sizes are not adjustable. On the other hand, size of the shaft produced by turning, grinding, etc. can be very easily varied.

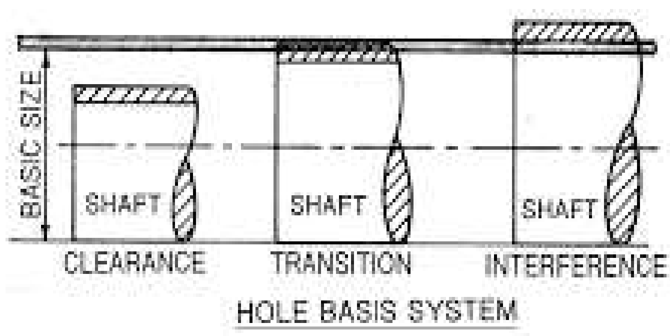


Fig: Hole basis system

2. **Shaft Basis System:**

In the shaft basis system, the size of the shaft is kept constant and different fits are obtained by varying the size of the hole. Shaft basis system is used when the ground bars or drawn bars are readily available. These bars do not require further machining and fits are obtained by varying the sizes of the hole. In this system, the upper deviation (fundamental deviation) of shaft is zero, i.e. the high limit of the shaft is same as basic size and the various fits are obtained by varying the low limit of shaft and both the limits of the hole.

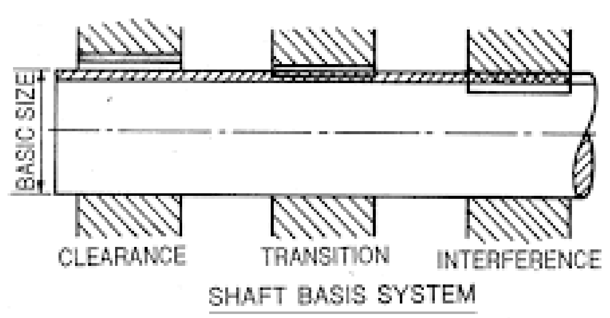


Fig: Shaft Basis System

DIFFERENCE BETWEEN HOLE BASIS & SHAFT BASIS SYSTEM:

BASIS SYSTEM	SHAFT SYSTEM
1. Size of hole whose lower deviation is zero (H.hole) is assumed as the basic size.	1. Size of shaft whose upper deviation is zero (h-shaft) assumed as basic size.
2. Limits on the hole are kept constant and those of shafts are varied to obtain desired type of fit.	2. Limits on the shaft are kept constant and those of holes are varied to have necessary fit.
3. Hole basis system is preferred in mass production because it is convenient and less costly to make a hole of correct size due to availability of standard drills reamers.	3. This system is not suitable for mass production because it is inconvenient time consuming and costly to make a hole of any size w.r to field shaft size so as to obtain required fit.
4. It is much more easy to vary the shaft sizes according to the fit required.	4. It is rather difficult to vary the hole sizes according to the fit required.
5. It required less amount of capital and storage space for roofs needed to produce shaft of different sizes.	5. It needs large amount of capital and storage space for large numbers of tools required to produce holes of different sizes.
6. Changing of shafts can be easily and conveniently done with suitable gap Gauges	6. Being internal measurement gauging of holes can’t be easily and conveniently done.

TOLERANCES:

Tolerance is a permissible limit and variation in dimensions or in physical parameters. It is possible to achieve dimensions and physical parameter exactly but it is very time consuming and economically unjustified or costly. It is quite often not necessary to achieve exact dimensions, in such functions or cases some permissible variation or tolerance is given.

The permissible variation in size or dimension is tolerance. The difference between the upper limit (high limit) and the lower limit of a dimension represents the margin for variation to workmanship, and is called a tolerance zone.

Tolerance can also be defined as the amount by which the job is allowed to go away from accuracy and perfectness without causing any functional trouble, when assembled with its mating part and put into actual service.

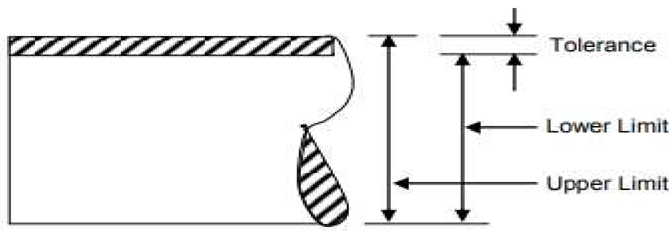


Fig: Tolerance

There are two ways of writing tolerances:

(a) Unilateral tolerance

(b) Bilateral tolerance.

Unilateral Tolerance:

In this system, the dimension of a part is allowed to vary only on one side of the basic size, i.e. tolerance lies only on one side of the basic size either above or below it (As shown in fig).

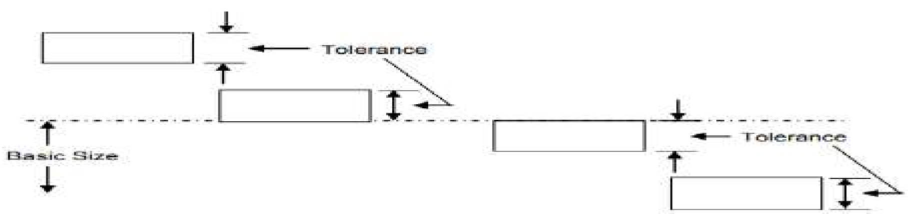


Fig: unilateral Tolerance

Examples of unilateral tolerance are :

$$25^{+0.02}_{+0.01},\ 25^{-0.02}_{-0.01},\ 25^{-0.01}_{-0.02},\ 25^{+0.0}_{-0.02}\ \text{etc.}$$

Unilateral system is preferred in interchangeable manufacture, especially when precision fits are required, because

(a) it is easy and simple to determine deviations,

(b) another advantage of this system is that „Go" Gauge ends can be standardized as the holes of different tolerance grades have the same lower limit and all the shafts have same upper limit, and

(c) This form of tolerance greatly assists the operator, when machining of mating parts. The operator machines to the upper limit of shaft (lower limit for hole) knowing full well that he still has some margin left for machining before the parts are rejected.

Bilateral Tolerance:

In this system, the dimension of the part is allowed to vary on both the sides of the basic size, i.e. the limits of tolerance lie on either side of the basic size, but may not be necessarily equally disposing about it.

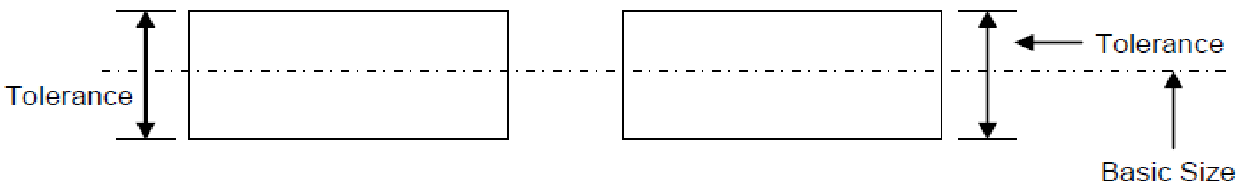


Fig: Bilateral Tolerance

Examples of bilateral tolerance are :

$$25^{\pm 0.02},\ 25^{+0.02}_{-0.01}\ \text{etc.}$$

In this system, it is not possible to retain the same fit when tolerance is varied and the basic size of one or both of the mating parts are to be varied. This system is used in mass production when machine setting is done for the basic size.

EXAMPLE

A 50 mm diameter shaft is made to rotate in the bush. The tolerances for both shaft and bush are 0.050 mm. determine the dimension of the shaft and bush to give a maximum clearance of 0.075 mm with the hole basis system.

Solution: In the hole basis system, lower deviation of hole is zero, therefore low limit of hole = 50 mm.

High limit of hole = Low limit + Tolerance

= 50.00 + 0.050

= 50.050 mm = 50.050 x 10^{-3} m

High limit of shaft = Low limit of hole – Allowance

= 50.00 – 0.075

= 49.925 mm = 49.925 x 10^{-3} m

Low limit of the shaft = High limit – Tolerance

= 49.925 – 0.050

= 49.875 mm = 49.875 x 10^{-3} m

The dimension of the system is shown in Figure

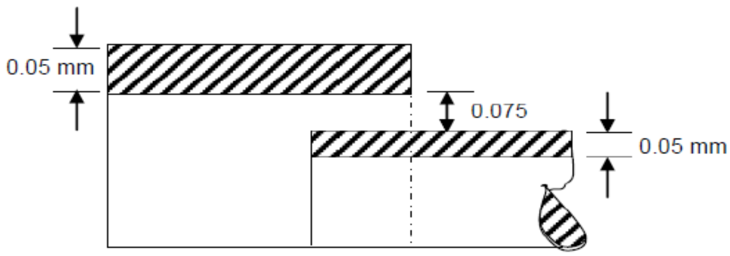


Fig: Shaft with Bush

GEOMETRICAL TOLERANCES

Geometric means geometric forms such as a plane, cylinder, square, etc. Geometrical features are: flatness, straightness, squareness etc. Geometrical tolerances refer to the shape of the surfaces (tolerance of form) as well as the relative location of one feature to another (tolerance of position). These tolerances are specified by special symbols (refer Tables 1 and 2).

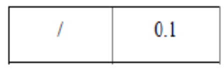
Table 1 : Symbol Specifying the Shape (Tolerance of Form)

Types of Error		Symbol
Briefly	Interpretation	
Flatness	Deviation from a flat Surface	—
Straighness	Deviation from a straight line	—
Cylindricality	Deviation from true cylinder	/O/
Circularity or roundness	Deviation from true circle	O
Accuracy of any surface	—	⊂

Table 2 : Symbol Specifying the Relative Location (Tolerance of Position)

Types of Error		Symbol
Briefly	Interpretation	
Parallelism	Lack of parallelism	//
Squareness and Perpendicularity	Lack of squareness	⊥
Concentricity	Lack of concentricity	⊚
Symmetry	Lack of symmetry	⌯

Geometrical tolerances are specified for geometrical features, in addition to linear tolerances. Data about the tolerances on the shape and location of surfaces are indicated on drawings in a rectangular box divided into two or three parts. For example "Lack parallelism between two surfaces is within 0.1 mm" can be written as

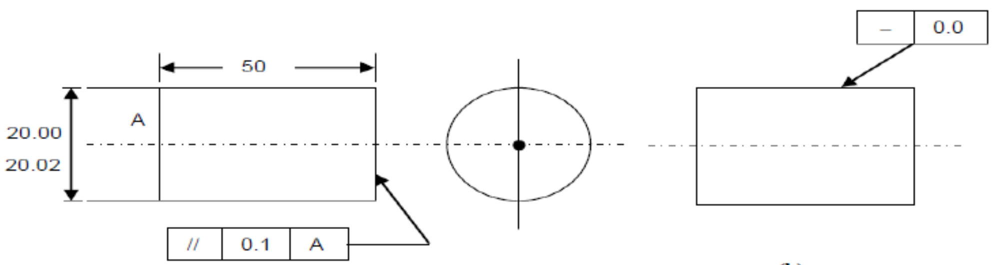


Examples of geometrical tolerances are given below :

Parallelism (Figure (a))

It indicates the requirement, "Surface A is parallel to opposite face within 0.1 mm".

Straightness (Figure (b))



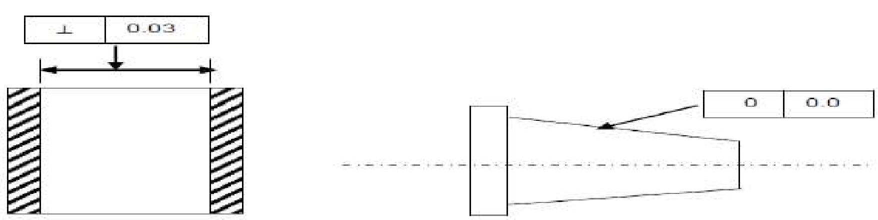
(a) (b)

It indicates the requirement, "Straight within 0.02 mm".

Squareness (Figure (c))

It indicates the requirement, "Square within 0.03 mm total".

Flatness (Figure (d))



(c) (d)

It indicates the requirement, "Flat within 0.002 mm total".

Roundness (Figure (e))

(e)

It indicates the requirement, "Taper round within 0.01 mm".

GAUGES:

Limit Gauges:

Two sets of limit gauges are necessary for checking the size of various parts. There are two gauges: Go limit gauge, and Not Go limit gauge.

1. **Go Limit:** The Go limit applied to that of the two limits of size corresponds to the maximum material condition, i.e. (1) an upper limit of a shaft, and (ii) the lower limit of a hole. This is checked by the Go gauge.

2. **Not Go Limit**: The Not Go limit applied to that of the two limits of size corresponds to the minimum material condition, i.e. (1) lower limit of a shaft, and (ii) the upper limit of a hole. This is checked by the Not Go gauge.

The types are:

1. Plug Gauge

3. Snap Gauge

4. Ring Gauge

1. Plug Gauge:

A plug gauge is a cylindrical type of gauge, used to check the accuracy of holes. The plug gauge checks whether the whole diameter is within specified tolerance or not. The 'Go' plug gauge is the size of the low limit of the hole while the 'Not-Go' plug gauge corresponds to the high limit of the hole.

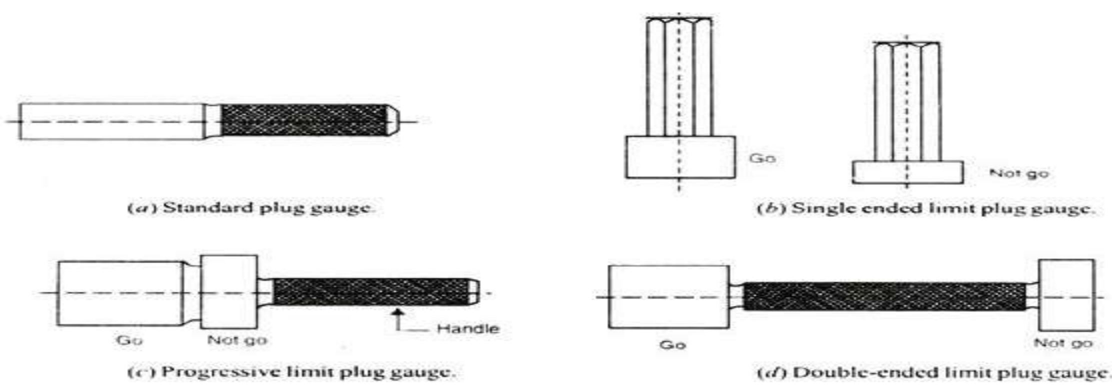


Fig: Types of Plug gauges

It should engage the hole to be checked without using pressure and should be able to stand in the hole without falling.

Snap Gauge:

A snap gauge is a U-Shaped frame having jaws, used to check the accuracy of shafts and male members. The snap gauge checks whether the shaft diameter is within specified tolerances or not.

The 'Go' snap gauge is the size of the high (maximum) limit of the shaft while the 'Not-Go' snap gauge corresponds to the low (minimum) limit of the shaft.

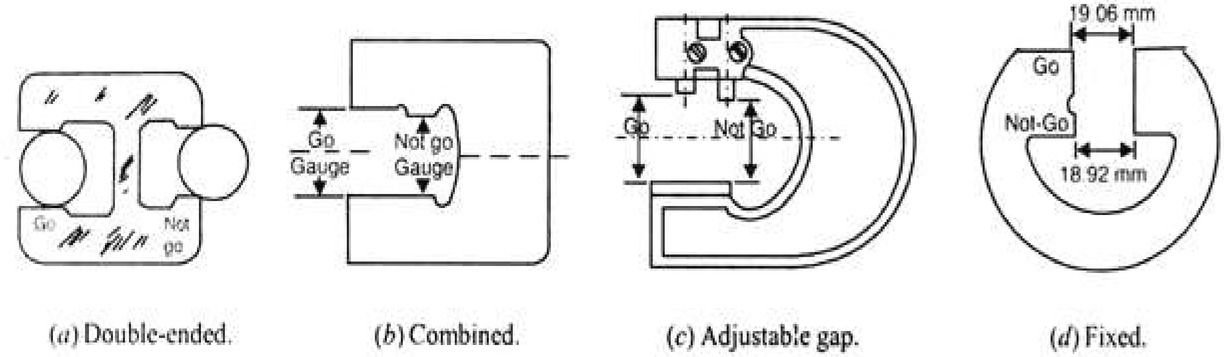


Fig: Types of Snap gauges

Ring Gauge:

A ring gauge is in the form of a ring, used to check the shafts and male members. The "Go' and 'Not Go' members may be separate or in a single ring. The opening or hole in the Go gauge is larger than that in the Not-Go gauge.

A ring gauge with both members combined in one ring is shown in figure (a):

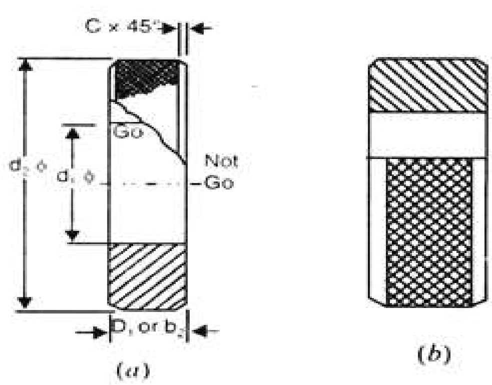


Fig: Plain ring gauge

PROPERTIES OF GAUGE MATERIAL:

The material for limit gauges should meet most of the following requirements:

(i) Optimal Hardness:

This is primary and most important property of gauge material. It is concerned with high durability, resistance to wear, and resistance to damage in use.

(ii) Stability of Dimensions:

The material should have high stability of dimensions to preserve size and form.

(iii) Proper Workability:

Proper workability, especially in manufacturing processes like grinding and polishing, to obtain required accuracy.

(iv) Wear and Corrosion Resistance:

The material should have high resistance to mechanical wear and corrosion.

(v) Low Coefficient of Linear Expansion:

The material should have low coefficient of linear expansion to avoid temperature and heating effect.

(vi) Uniformity of Structure:

The structure of gauge material should be uniform for better accuracy.

TAYLOR'S PRINCIPLE OF GAUGE DESIGN

The Taylor's Principle of gauge design gives two statements which are discussed here:

Statement 1:

The "Go" gauge should always be so designed that it will cover the maximum metal condition (MMC), whereas a "NOT-GO" gauge will cover the minimum (least) metal condition (LMC) of a feature, whether external or internal.

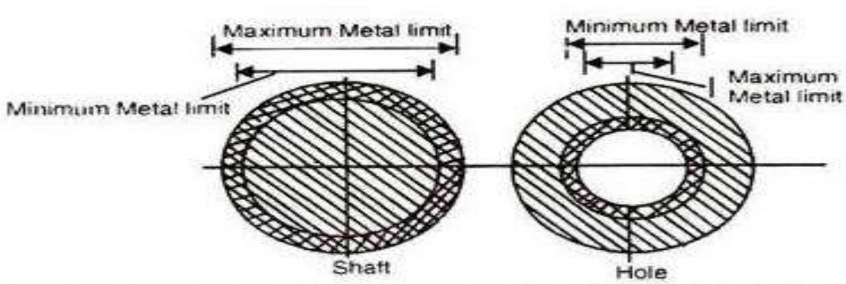


Fig. 1 . Maximum and minimum metal condition for Taylor's principle.

Statement 2:
The "Go" gauge should always be so designed that it will cover as many dimensions as possible in a single operation, whereas the "NOT-GO" gauge will cover only one dimension.
Means a Go plug gauge should have a full circular section and be of full length of the hole being checked as in shown figure 2:

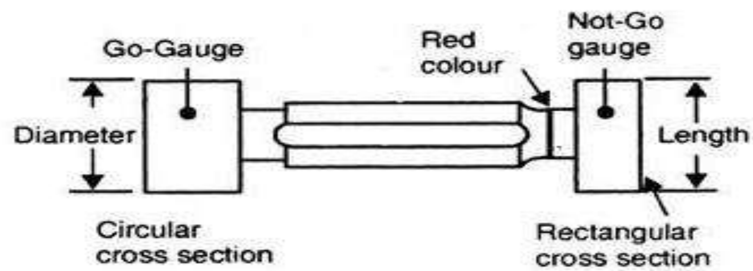
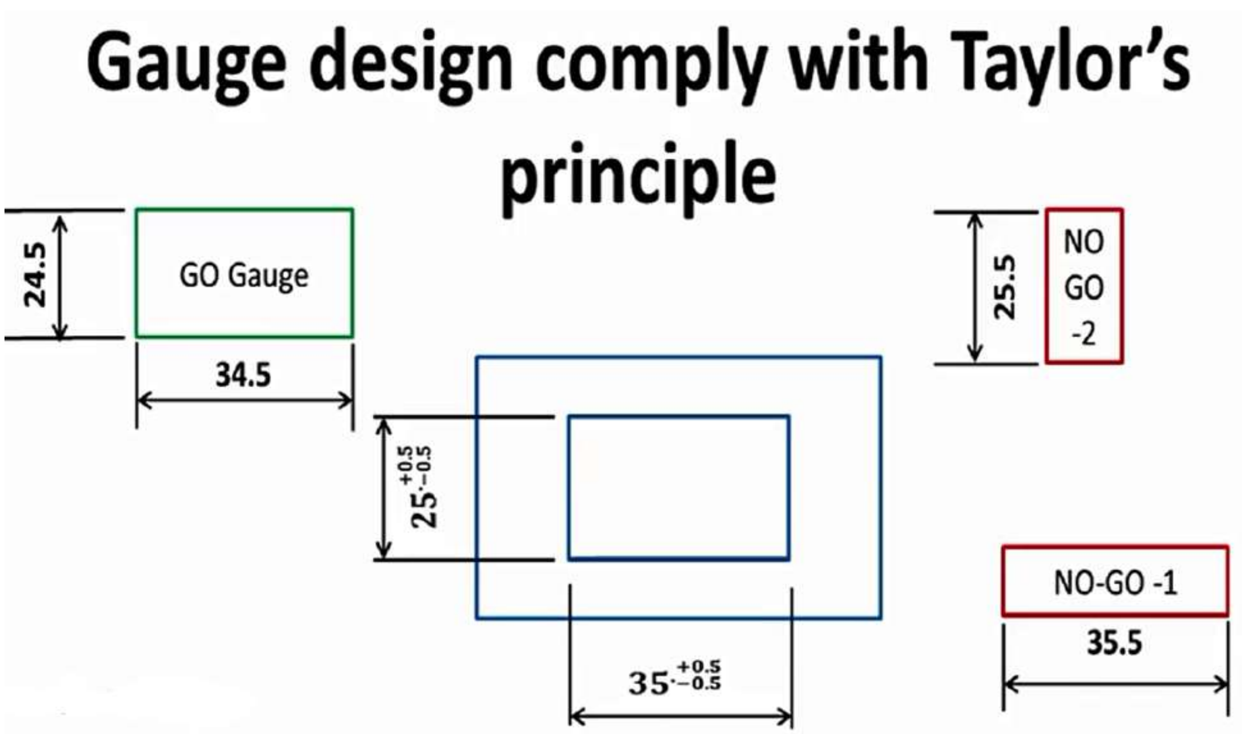


Fig. 2. Go and Not-Go gauges for hole (Taylor's principle).

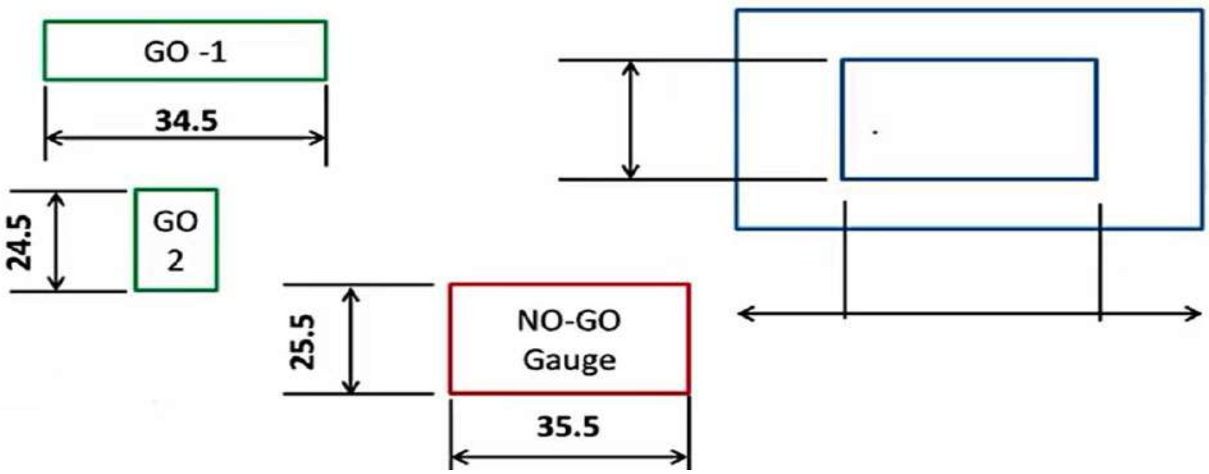
OR

i.e. According to Taylor's principle, the GO gauge should be made for maximum material limit and it has to incorporate as many dimensions as possible to inspect in one pass and NO-GO gauge can be made for minimum material limit and separate NO GO gauge should be made for each separate dimension.

Example:



Gauge design not comply with Taylor's principle

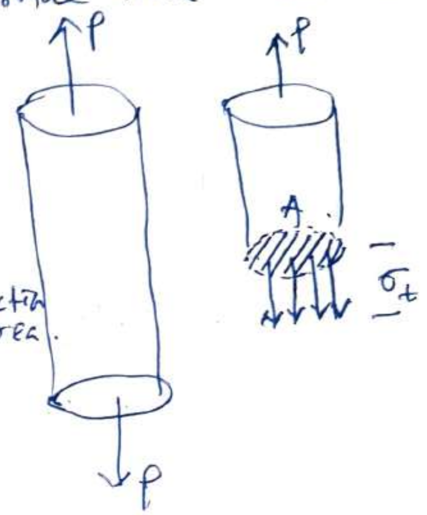


__Stress__: The internal resisting force per unit area of the component is called stress.

Stress (– tensile
       (– Compressive

If the fibres of the component tend to elongate due to the external force then it is tensile stress.

If the fibres tend to shorten due to external force then the stresses are called compressive stress.



$$\sigma_t = \frac{P}{A} \quad \text{unit MPa / N/m}^2$$

σ_t = tensile stress. P = external force A = cross sectional area.

__Strain__ : It is the deformation per unit length.

$$\varepsilon = \frac{\delta}{l}$$

δ = elongation of rod, l = original length of rod.

Hooke's Law : Stress is directly proportional to strain

$$\sigma_t \propto \varepsilon$$

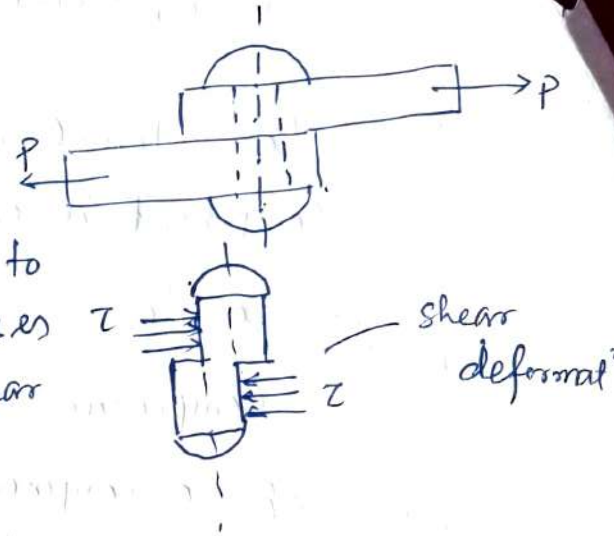
$$\sigma_t = E\,\varepsilon$$

where E = constant of proportionality which is also called as Young's modulus or Modulus of elasticity.

②

Shear stress

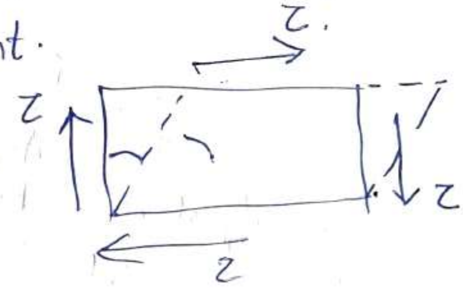
When the external force acting on a component tends to slide the adjacent planes with respect to each other, the resulting stresses on these planes are called shear stress.



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

A = cross sectional area of the rivet.

Shear strain (γ) is defined as the change in the right angle of a shear element.



with in elastic limit the stress strain relationship is given as

$$\tau = G\gamma.$$

where G is the constant of proportionality known as shear modulus or modulus of rigidity

Relation ship between { modulus of elasticity. modulus of rigidity Poisson's Ratio

$$E = 2G(1+\mu)$$

where μ = Poisson's ratio.

It is the ratio of strain in the lateral direction to that in the axial direction.

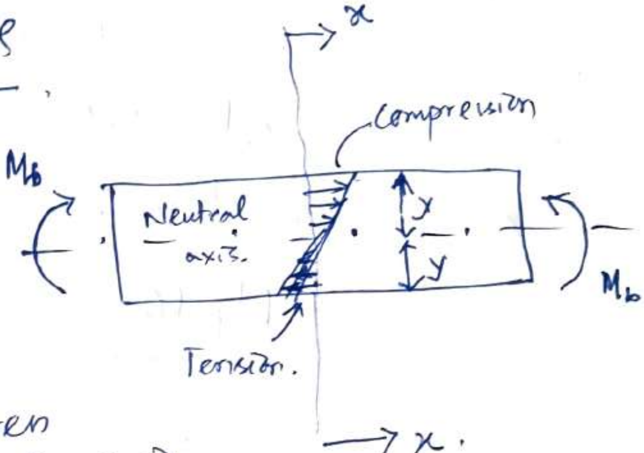
Permissible shear stress

$$\tau = \frac{S_{sy}}{f(s)}$$

S_{sy} : yield strength in shear.

Stresses due to Bending

If a beam is subjected to a bending moment M_b then there is a combination of tensile stress on one side of neutral axis and bending stress on the other side of neutral axis



(bending of a thick Leather belt)
- cracks on outer surface
& folds in inner surface.

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

σ_b = bending stress at a distance y from the neutral axis

M_b = Applied bending moment

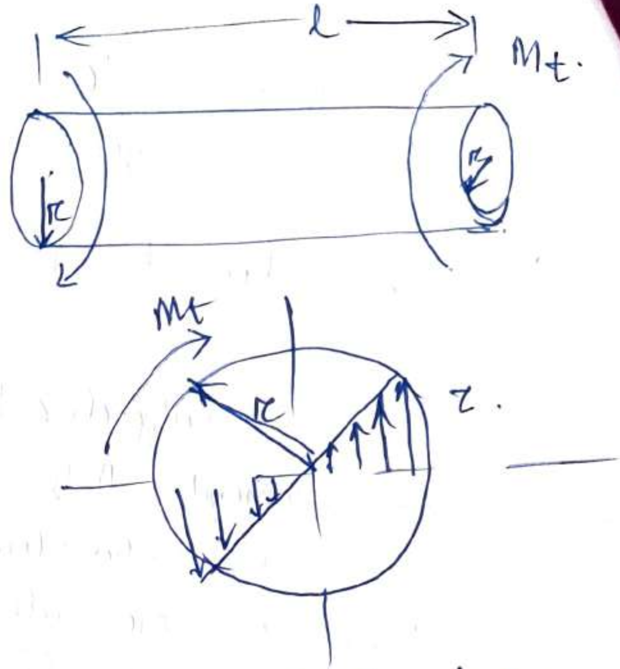
I = Moment of Inertia of the cross section about neutral axis.

The distribution of stress is Linear & the stress is proportional to the distance from neutral axis.

Rectangular crosssecn : $I = \frac{bd^3}{12}$, Circular crosssecn : $I = \frac{\pi d^4}{64}$

Stresses due to Torsional Moment

If a transmission shaft is subjected to external torque; the internal stresses which are induced to resist the action of twist are called torsional shear stress:



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

- M_t = applied torque. r = radial distance.

J = Polar moment of inertia of the crosssection about the axis of rotation

for a solid circular cylinder $J = \frac{\pi d^4}{32}$.

→ stress is maximum at the outer fiber & zero at the axis of rotation.

- the angle of twist is given by.

$$\theta = \frac{M_t l}{J G}$$

θ = angle of twist

G = modulus of rigidity.

Types of Load.

Load on Machine Element

- Steady/ Dead Load
- Variable or Live Load
- Shock or Sudden Load Impact Loading,
- Cyclic ~~Impact~~ Load - Repeative Load

① A Steady / Dead Load : If a load does not change in magnitude & direction

Static Load.

* do not change with time.
* It neither shift nor move from its original position.

It is a force which is gradually applied to a mechanical component but does not change in magnitude or direction with respect to time. Ex: Force due to gravity.

② Variable / Live Load : If a load varies continiously or frequently

Dynamic Load :— It is a force which changes in magnitude as well as direction with respect to time. ex: vehicle moving on a bridge.

③ Shock / Impact Load : It is the collision of one component in motion with a second component which may be either in motion or at rest.

→ It is rapidly applied to a machine component.

→ The time of application is of very short span.

Ex: Driving a nail on wall by hammer, brakes, shock absorber.

(6)

Modes of failure.

A mechanical component may fail or unable to perform its function satisfactorily due to the following

(a) failure by elastic deflection

(b) failure by general yielding.

(c) failure by fracture.

In applications like transmission shaft supporting gears the maximum force acting on the shaft without affecting its performance is limited by the permissible elastic deflection.

- Lateral / torsional rigidity is considered as design criteria -

and stresses ~~in duced~~ induced in the components due to such deflections are not significant enough as yielding/fracture

<u>Yielding</u>: due to excessive inelastic deformation/ plastic deformation the machine parts may not be suitable for performing its function.

↳ applicable for ductile materials

<u>fracture</u>: If a machine component cease to function satisfactorily because of the sudden fracture without any plastic deformation.

↳ applicable for brittle materials.

Factor of Safety.

While designing a component; it is necessary to provide sufficient reserve strength in case of an accident.

This is achieved by taking a suitable factor of safety (f_s)

$$f_s = \frac{\text{failure Stress}}{\text{Allowable stress.}} \quad \text{or} \quad \frac{\text{failure Load}}{\text{Working Load.}}$$

The allowable stress is a stress value which is used in design to determine the dimension of the component.

for ductile material $\quad \sigma = \dfrac{S_{yt}}{f_s}$

for brittle Material $\quad \sigma = \dfrac{S_{ut}}{f_s}$

S_{yt} = yield strength.

S_{ut} = ultimate tensile strength.

The magnitude of factor of safety depends upon.

- Effect of failure
- Type of Load
- Degree of accuracy in force Analysis
- Material of Component
- Reliability of Component
- Cost of Component.
- Testing of M/c element
- Service conditions.
- Quality of Manufacture.

(8) For finding the quantitative values of factor of safety the following things are to be considered.

1. Ultimate tensile strength
2. Yield strength.
3. Endurance Limit. (for external fluctuating stresses).
4. Pitting (surface fatigue failure) — gears, cam follower, rail wheel
5. Buckling: sudden large lateral deflection.

Theories of failure.

Theories of failure provides a relationship between the strength of machine component subjected to complex state of stress.

For example: 1) A power screw is subjected to torsional moment as well as axial force

2.) An overhang crank is subjected to combined bending and torsional moments

Many theories has been proposed to explain the theories of failure

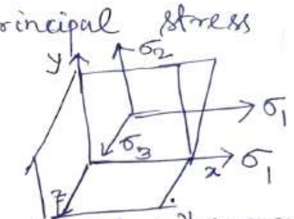
1. Maximum principal Stress theory (Rankine's Theory).
2. Maximum Shear Stress Theory (Coulomb, Tresca & Guest's Theory)
3. Maximum Strain Theory (St. Venant's Theory).
4. Maximum Total Strain Energy Theory (Haigh's Theory).
5. Distortion Energy Theory (Huber Von Mises & Hencky Theory).

(1) Maximum Principal Stress Theory. (Rankine Theory).

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

If $\sigma_1, \sigma_2, \sigma_3$ are three principal stress at a point on the component and

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



then according to this theory; failure occurs whenever

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

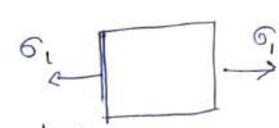
(ductile) (Brittle)

This theory mostly used for brittle materials.

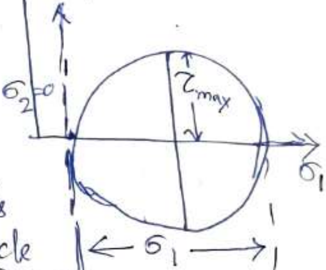
The dimension of compound is determined by using factor of safety $\boxed{\sigma_1 = \frac{S_{yt}}{(fs)}}$ or $\boxed{\sigma_1 = \frac{S_{ut}}{(fs)}}$

(2) Maximum Shear Stress Theory. (Coulomb, Tresca & Guest Theory).

This theory states that; the failure of a mechanical component subjected to bi-axial / tri-axial stresses occures when the maximum shear stress at any point in the component becomes equal to the maximum shear stress in the standard specimen of the tension test when yielding starts.



As in tension test the specimen is subjected to uniaxial stress (σ_1) only so $\sigma_2 = 0$
so from the Mohr's Circle diagram



Mohr's Circle Diagram

$$\boxed{\tau_{max} = \frac{\sigma_1}{2}.} = \frac{S_{yt}}{2}$$ when the specimen starts yielding

Therefore maximum shear stress theory predicts that the

Suppose $\sigma_1, \sigma_2, \sigma_3$ are three principal stresses at a point on the component; then the shear stress on three different planes will be

$$\tau_{12} = \frac{\sigma_1 - \sigma_2}{2}, \quad \tau_{23} = \frac{\sigma_2 - \sigma_3}{2}, \quad \tau_{31} = \frac{\sigma_3 - \sigma_1}{2}$$

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

Considering factor of safety

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

or $$(\sigma_1 - \sigma_2) = \frac{S_{yt}}{(fs)} \,/\, (\sigma_2 - \sigma_3) = \frac{S_{yt}}{(fs)} \,/\, (\sigma_3 - \sigma_1) = \frac{S_{yt}}{(fs)}$$

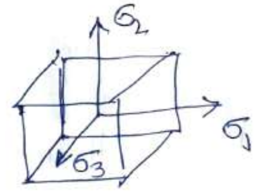
The above relationships are used to determine the dimensions of the component; which are ductile materials like transmission shaft.

Distortion Energy Theory.

(Huber. von Mises & Hencky's Theory)

This theory states that the failure of the mechanical component subjected to bi-axial or triaxial stresses occures when the strain energy of distortion per unit volume at any point in the component becomes equal to the strain energy of distortion per unit volume in the standard specimen of tension test when yielding starts.

If a unit cube subjected to three principal stresses $\sigma_1, \sigma_2, \sigma_3$, the total strain energy 'U' of the cube is given by



$$U = \frac{1}{2}\sigma_1\epsilon_1 + \frac{1}{2}\sigma_2\epsilon_2 + \frac{1}{2}\sigma_3\epsilon_3$$

Also
$$\epsilon_1 = \frac{1}{E}[\sigma_1 - \mu(\sigma_2+\sigma_3)]$$
$$\epsilon_2 = \frac{1}{E}[\sigma_2 - \mu(\sigma_1+\sigma_3)]$$
$$\epsilon_3 = \frac{1}{E}[\sigma_3 - \mu(\sigma_2+\sigma_1)]$$

For a component subjected to bi-axial stresses.

$$\frac{S_{yt}}{(fs)} = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2}$$

or $\boxed{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2 = \left(\frac{S_{yt}}{fs}\right)^2}$. ductile material

Maximum Principal Strain Theory. (St. Venant's theory).

This theory states that failure may occur when the maximum principal strain is equal to the maximum strain at the elastic limit in simple tension test.

$$\frac{1}{E}(\sigma_1 - \mu\sigma_2) = \frac{\sigma_{yt}}{E}$$

σ_{yt} = Yield point stress in tension as determined from simple tension test

$$(\sigma_1 - \mu\sigma_2) = \frac{\sigma_{yt}}{} = \frac{S_{yt}}{(fs)}$$

(12) **Maximum Strain Energy Theory. (Haigh's Theory)**

This theory states that; failure may occur when the maximum strain energy per unit volume is equal to the strain energy per unit volume at the elastic limit in simple tension test.

$$\boxed{\sigma_1^2 + \sigma_2^2 - 2\mu\sigma_1\sigma_2 = \sigma_{yt}^2 = \left(\frac{S_{yt}}{(fs)}\right)^2}$$

$$U_1 = \frac{1}{2E}\left[\sigma_1^2 + \sigma_2^2 - 2\mu\sigma_1\sigma_2\right].$$

$$U_2 = \frac{1}{2E}\left[\frac{S_{yt}}{(fs)}\right]^2 = \frac{1}{2E}(\sigma_{yt})^2$$

It may be used for ductile material.

Relationship of factor of safety & Theory of failure

Sl.No.	Theory of failure	factor of Safety
1.	Maximum principal Stress	σ_{yt}/σ_1
2.	Maximum shear stress	$\sigma_{yt}/\sigma_1 - \sigma_2$
3.	Maximum strain	$\sigma_{yt}/(\sigma_1 - \mu\sigma_2)$
4.	Maximum strain energy	$\sigma_{yt}/\sqrt{\sigma_1^2 + \sigma_2^2 - 2\mu\sigma_1\sigma_2}$
5.	Distortion Energy	$\sigma_{yt}/\sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2}$

Selection & Use of failure Theories.

- Ductile material have same tensile as well as compressive strength.
- In Max^m shear stress theory & distortion energy theory it is assumed that yield strength in tension is equal to yield strength in compression.
- Therefore; maximum shear stress theory & distortion energy theory are used for ductile material.
- Maximum principal stress theory is proper choice for brittle materials; for ductile material the choice of theory depends on the level of accuracy required and the degree of computational difficulty the designer is ready to face.
- for ductile material; the most accurate way to design is to use distortion energy theory of failure and the easiest way to design is to apply maximum shear stress theory.

Q. A bolt is subjected to an axial pull of 10 KN and a transverse shear force of 5 KN. The yield strength of the bolt material is 300 MPa. considering a factor of safety of 2.5; determine the diameter of bolt using

(i) Maximum normal stress theory
(ii) Maximum shear stress theory
(iii) Maximum principal strain theory,

take poisson's ratio as 0.25

(14) Data Given: $P = 10\,kN$

$Q = 5\,kN$

$\sigma_y = 300\,MPa$ d = diameter of bolt ??

$f.s. = 2.5$

$\mu = 0.25$

Direct tensile strength in the bolt, $\sigma_t = \frac{4P}{\pi d^2} = \frac{4 \times 10 \times 1000}{\pi d^2}$

$$= \frac{12732.4}{d^2}\ MPa$$

Transverse shear stress in the bolt $\tau = \frac{4Q}{\pi d^2} = \frac{4 \times 5 \times 1000}{\pi d^2}$

$$= \frac{6366.2}{d^2}\ MPa$$

Let σ_1 & σ_2 are the maximum & minimum principal stresses.

$$\boxed{\sigma_1, \sigma_2 = \frac{1}{2}\left[\sigma_t \pm \sqrt{\sigma_t^2 + 4\tau^2}\right]}$$

$$\sigma_1 = \frac{1}{2}\left[\frac{12732.4}{d^2} + \sqrt{\left(\frac{12732.4}{d^2}\right)^2 + 4\left(\frac{6366.2}{d^2}\right)^2}\right]$$

$$= \frac{1}{2d^2}\left[12732.4 + \sqrt{(12732.4)^2 + 4(6366.2)^2}\right]$$

$$= \frac{15369.4}{d^2}\ MPa$$

Similarly $\sigma_2 = \frac{1}{2d^2}\left[12732.4 - \sqrt{(12732.4)^2 + 4(6366.2)^2}\right]$

$$= -\frac{2637}{d^2}\ MPa.$$

Allowable stress/
Permissible stress $= \sigma = \frac{S_{yt}}{fs} = \frac{300}{2.5} = 120$ MPa

(i) According to maximum normal stress theory.

$$\sigma_1 = \sigma$$

$$\frac{15369.4}{d^2} = 120 \Rightarrow d = 11.32 \text{ mm.}$$

(ii) Maximum shear stress Theory.

$$\sigma_1 - \sigma_2 = \sigma = \frac{S_{yt}}{(fs)}$$

$$\frac{15369.4}{d^2} - \left(-\frac{2637}{d^2}\right) = 120$$

$$d = 12.25 \text{ mm.}$$

(iii) Maximum principal strain theory.

$$\sigma_1 - \mu\sigma_2 = \sigma$$

$$\frac{15369.4}{d^2} - 0.25\left(-\frac{2637}{d^2}\right) = 120$$

$$d = 11.58 \text{ mm}$$

Q. At a critical section in a shaft; the following stresses are induced

Bending stress = 60 MPa.

Torsional shear stress = 40 MPa.

Determine the factor of safety according to (i) maximum normal stress theory (ii) maximum shear stress theory (iii) maximum distortion energy theory. The proportional limit in a simple tension test is found to be 300 MPa. Take poisson's ratio as 0.3

(u) Given Data:

$$\sigma_t = \sigma_c = 60\,\text{Mpa}$$

$$\tau = 40\,\text{Mpa}$$

$$\mu = 0.3$$

$$\sigma_y = 300\,\text{Mpa.}$$

Let σ_1 & σ_2 are maximum principal stress.

$$\sigma_1, \sigma_2 = \frac{1}{2}\left[\sigma_t \pm \sqrt{\sigma_t^2 + 4\tau^2}\right]$$

$$\sigma_1 = \frac{1}{2}\left[60 + \sqrt{60^2 + 4\times 40^2}\right] = 80\,\text{Mpa.}$$

Similarly $\sigma_2 = \frac{1}{2}\left[60 - \sqrt{60^2 + 4\times 40^2}\right] = -20\,\text{Mpa}$

(i) <u>Maximum normal ~~Shear~~ stress theory</u>:

$$f_s = \text{factor of safety} = \frac{S_{yt}/\sigma_{yt}}{\sigma_1} = \frac{300}{80} = 3.75$$

(ii) <u>Maximum shear stress theory</u>:

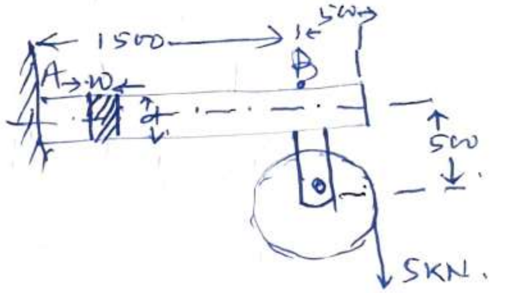
$$\text{factor of safety }(n/f_s) = \frac{\sigma_y}{\sigma_1 - \sigma_2} = \frac{300}{80-(-20)} = 3.$$

(iii) <u>Maximum ~~Principal~~ Distortion Energy theory.</u>

$$\text{factor of safety }(f_s) = \frac{\sigma_y}{\sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1\sigma_2}} = ??$$

$$\frac{300}{\sqrt{80^2 + (-20)^2 - (80\times -20)}} =$$

A Cantilever beam of rectangular cross section is used to support a pulley as shown in fig. The tension in the wire rope is 5KN. The beam is made up of cast iron FG200. and the factor of safety is 2.5. The ratio of depth to width of the cross section is 2. Determine the dimensions of the cross section of the beam.

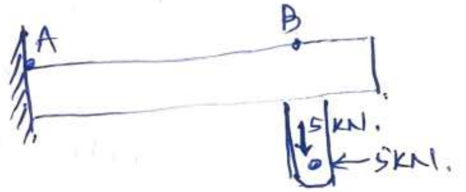


Data given $P = 5KN.$

$S_{ut} = 200\ N/mm^2$

$(fs) = 2.5$

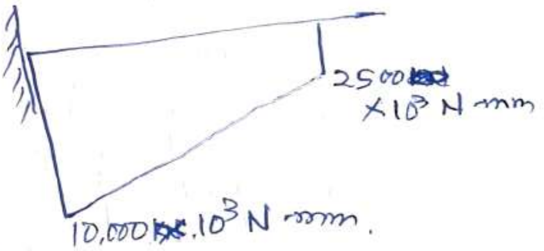
$d/w = 2.$



Step I. calculation of permissible bending stress.

$$\sigma_b = \frac{S_{ut}}{(fs)} = \frac{200}{2.5} = 80N/mm^2$$

Step II Calculation of bending moment.



$(M_b)_{at\ B} = 5000 \times 500 = 2500 \times 10^3\ N-mm$

$(M_b)_{at\ A} = 5000 \times 500 + 5000 \times 1500 = 10{,}000 \times 10^3\ N{\cdot}mm.$

Step-III Calculation of dimension of cross section.

The maximum bending stress is at 'A'.

for this point $y = d/2 = w$, $I = \frac{1}{12} w (2w)^3 = \frac{1}{12} bd^3 = \frac{2}{3} w^4\ mm^4$

$$\sigma_b = \frac{M_b y}{I} \quad \text{or} \quad 80 = \frac{(10000 \times 10^3) w}{(2/3\, w^4)}$$

$\Rightarrow w = 57.24\ mm \simeq 60mm$

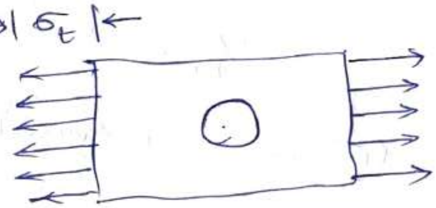
$d = 2w = 120mm.$

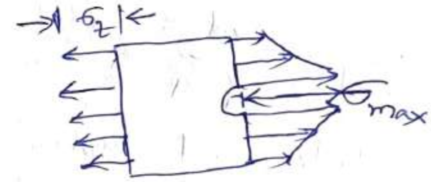
Stress Concentration

i) The three elementary equations for determining stress during designing machine elements are $\sigma_t = \frac{P}{A}$, $\sigma_b = \frac{M_b y}{I}$, $\tau = \frac{M_t r}{J}$ while there is no discontinuities in the cross-section.

ii) But in actual practice discontinuities & abrupt changes in cross section are unavoidable due to oil holes, grooves, keyways & splines, screw threads and shoulders.

iii) If a plate with a small circular hole is subjected to tensile stress; it can be observed that there is a sudden rise in the magnitude of stresses in the vicinity of the hole.





This localized stress is far greater than the stress ~~developed~~ obtained by the elementary equation.

So. Stress Concentration is defined as the localization of high stresses due to the irregularities present in the component & abrupt changes of the cross section.

Stress Concentration factor: (K_t)

$$K_t = \frac{\text{Highest value of actual stress near discontinuity.}}{\text{Nominal stress obtained by elementary equations for minimum cross section.}}$$

So $$K_t = \frac{\sigma_{max}}{\sigma_0} = \frac{\tau_{max}}{\tau_0}$$

Subscript 't' denotes theoretical.

Causes of Stress Concentration.

(i) Variation in properties of material

If the material is nonhomogeneous due to

* Internal cracks & flaws like blow holes
* Cavities in welds
* Air holes in steel components.
* Non metallic or foreign inclusion.

(ii) Load Application

If concentrated load is applied over a very small area like

* Contact between meshing teeth of gear.
* Contact between CAM & follower
* Contact between balls & races of bearing.
* " " rail & wheel.
* " " crane hook & chain.

(iii) Abrupt changes in section

In order to mount gears, sprockets, pulleys & Ball bearings on a transmission shaft steps are cut on the shaft & shoulders are provided from Assembly considerations. This changes in the crosssection result in stress concentration in the component.

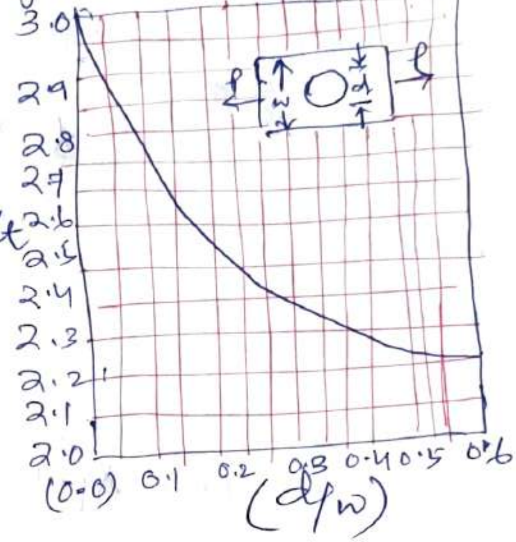
(iv) Discontinuities in the component.

Certain features of machine components such as oil holes or oil grooves, keyways and splines as well as Screw threads results in discontinuities. which is the cause of stress concentration.

(V) Machining Scratches:

Machining scratches, stamp marks, or inspection marks are surface irregularities which causes stress Concentration.

Stress Concentration factors are determined by two methods i.e. mathmatical methods based on theory of elasticity & experimental methods like photo elasticity.



The chart for the stress concentration factor for a rectangular plate with a transverse hole loaded in tension/ compression is shown in figure.

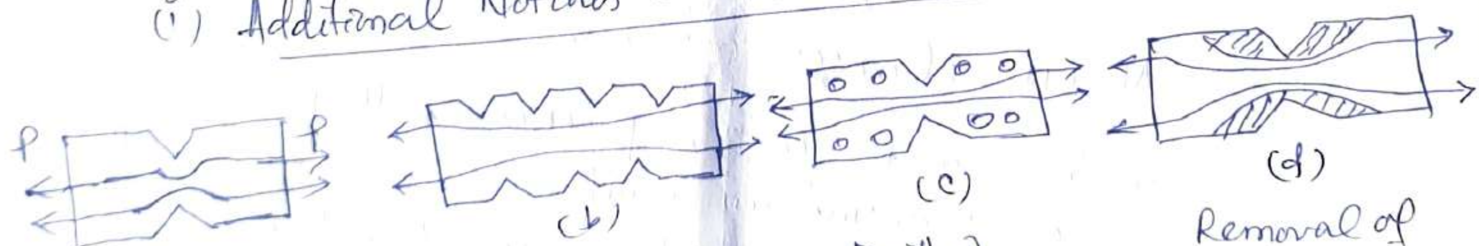
The nominal stress $\sigma_o = \dfrac{P}{(w-d)t}$

t = plate thickness

Reduction of Stress Concentration

Although it is not possible to completely eliminate the effect of stress concentration; there are methods to reduce stress concentrations. This is achieved by providing specific geometric shapes to the component.

(i) Additional Notches & holes in Tension Member.



(a) flat plate with 'V' notch subjected to tensile stress.

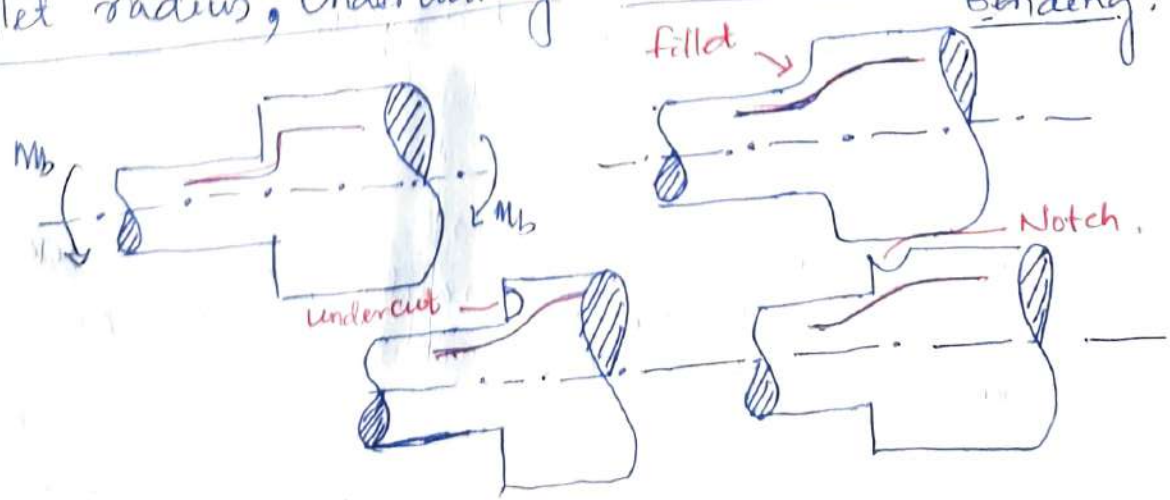
(b) Multiple notches

(c) Drilled holes

(d) Removal of material

methods for reduction of severity of stress concentration.

(ii) fillet radius, Undercutting and Notch for Member in Bending.

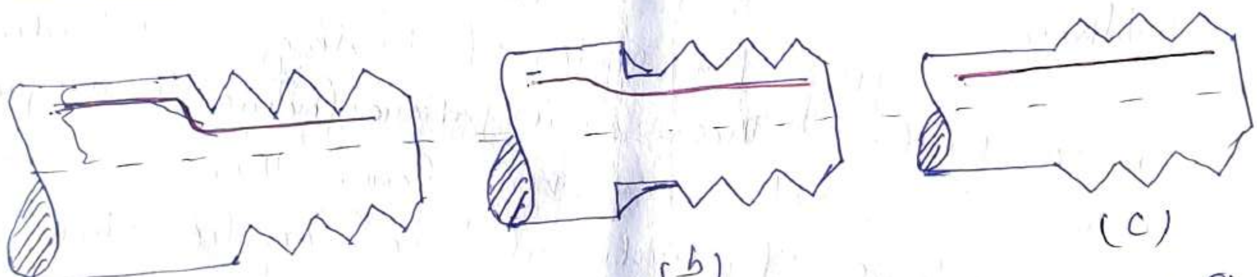


(iii) Drilling Additional holes for Shaft.

A transmission shaft with a keyway results in a discontinuity and ultimately develope stress concentration. which in turn reduces ~~stress concentrations~~ torsional shear strength.

There fore in addition to giving fillet radius at the inner corners of key way; drilling two symmetrical holes on the either sides of keyway will be a better option.

(iv) Reduction of Stress Concentration in Threaded Members.



(a) (b) (C)

It is observed that the force flow line bents as it passes from the shank portion to the threaded portion of a ~~Screw thread~~. Threaded member; which results in stress concentration.

This can be minimized by providing a small undercut in between the shank and the threaded portion or by reducing the shank diameter.

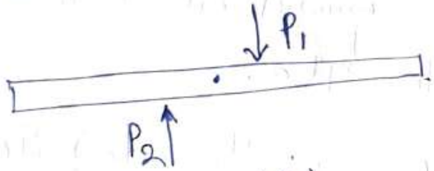
fa

6

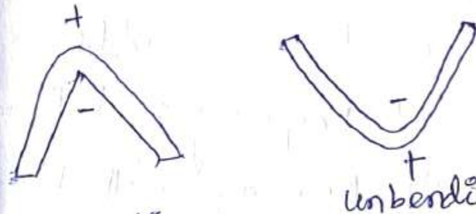
fatigue Failure

In case of repeatitive / cyclic loading ~~..~~ it has been observed that material fails under fluctuating stresses which is lower than ~~even~~ ultimate tensile strength or even lower than the yield strength sometimes.

This phenomenon of decreased resistance of the material to the fluctuating stresses is the main characteristic of fatigue failure.



(a)



Bending   unbending.

In childhood if we want to cut a wire into two parts. then we have to apply few cycles of bending & unbending at the required point. This is a fatigue failure & magnitude of stress required to fracture is very low. there is a decreased resistance of material to cyclic stress.

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

failure due to
- Static Load – Considerable plastic flow before fracture.
- fatigue – It begins with a crack at some point of material.

Endurance Limit

The fatigue or endurance limit of a material is defined as the maximum amplitude of completely reversed stress that the standard specimen can sustain for an unlimited number of cycles with out fatigue failure.

- Since fatigue test can not be conducted for infinite/unlimited number of cycles; 10^6 cycles is considered as sufficient number of cycles to define the endurance limit.

- Fatigue Life is defined as the number of stress cycles that the standard specimen can complete during the test before the apearance of the first fatigue crack.

Notch Sensitivity

It is observed that the actual reduction in the endurance limit of a material due to stress concentration is less than the amount indicated by the theoretical stress concentration factor K_t.

So K_t is applicable for homogeneous, isotropic & elastic material

And K_f = fatigue Stress concentration factor = $\dfrac{\text{Endurance limit of Notch free specimen.}}{\text{Endurance limit of notched specimen.}}$

The notch sensitivity is defined as the susceptibility of a material to overcome to the ~~dang~~ damaging effect of stress raising notches in fatigue loading.

$$q = \frac{\text{Increase of actual Stress over nominal Stress}}{\text{Increase of theoretical stress over nominal stress.}}$$

σ_0 = nominal stress as obtained by elementary equation

actual stress = $K_f \sigma_0$

thoretical stress = $K_t \sigma_0$

Increase of actual stress over nominal stress

$$= (K_f \sigma_0 - \sigma_0)$$

Increase of theoretical stress over nominal stress

$$= (K_t \sigma_0 - \sigma_0)$$

There fore $q = \cancel{K_f \sigma_0} - \frac{K_f \sigma_0 - \sigma_0}{K_t \sigma_0 - \sigma_0} = \frac{K_f - 1}{K_t - 1}$

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

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

There are three types of cyclic stresses.

* Fluctuating / alternating stress
* Repeated stress
* Reversed stress.

σ_{max} = Maximum stress

σ_{min} = Minimum stress.

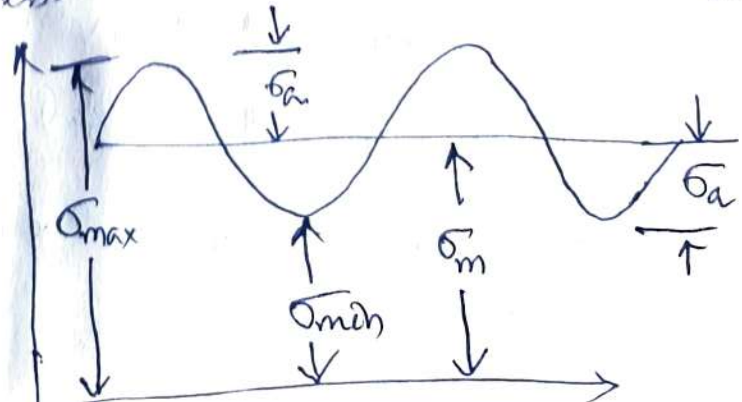
Mean stress.

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

stress amplitude

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

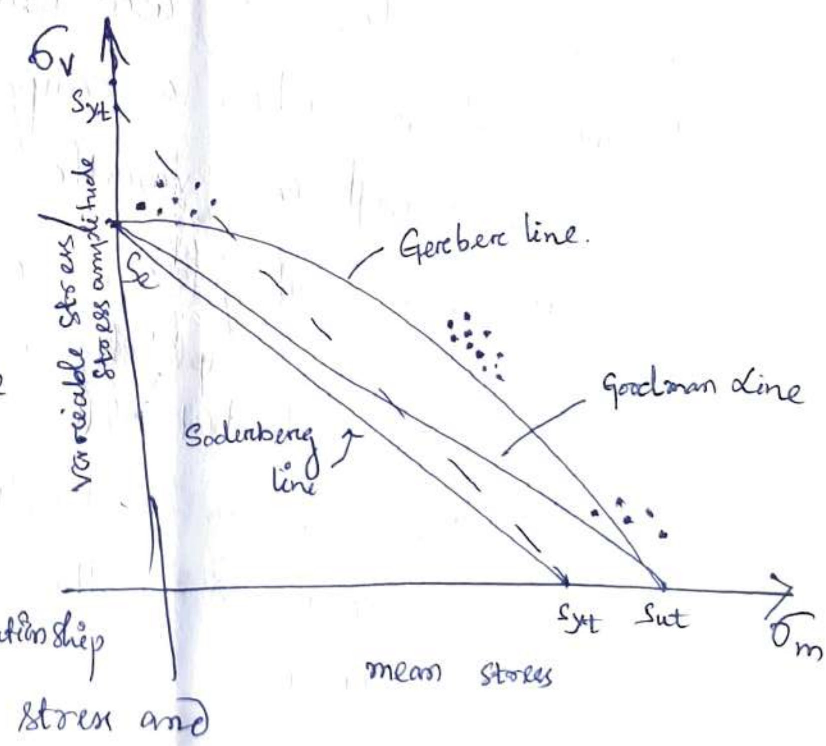
$= \sigma_v$ variable stress.



The magnitude of mean stress (σ_m) and variable stress (σ_a) or (σ_v) depends upon the magnitude of maximum & minimum force acting on the component.

→ when the stress amplitude / variable stress is zero; the load is purely static & criteria of failure is S_{ut} or S_{yt}. This is plotted on the abscissa.

→ When the mean stress is zero, the stress is completely reversing and criterion of failure is the endurance limit S_e. this is plotted on the ordinate.



Gerber Method

- A parabolic curve joining S_e on the ordinate & S_{ut} on the abscissa is called as the Gerber line

- It gives the relationship between the variable stress and mean stress for ductile material.

- According to Gerber the parabolic relationship for fluctuating stress is given by.

$$\left(\frac{\sigma_m}{\sigma_u}\right)^2 + \frac{\sigma_v}{\sigma_e} = 1 \quad \text{or} \quad \left(\frac{\sigma_m}{S_{ut}}\right)^2 + \frac{\sigma_a}{S_e} = 1$$

6

Goodman Method

i. → Generally the test data for ductile material fall closer to the Gerber parabola; but because of the possible scatter of the test points (results) a straight line relation is usually preferred.

→ A straight line joining S_e on the ordinate and S_{ut} on the abscissa is called the Goodman line.

→ It may be used for both ductile & brittle material.

→ This Goodman line is used widely as the criterion of fatigue failure when a component is subjected to mean as well as variable stress, because.

(i) It is safe from design consideration as it is completely inside the failure points of test data.

(ii) The equation of straight line is simple as compared to the equation of a parabolic curve.

The equation for Goodman Line is given as.

$$\frac{\sigma_m}{\sigma_{ut}} + \frac{\sigma_v}{\sigma_e} = \frac{1}{fs.}$$

$$\text{or} \quad \frac{\sigma_m}{S_{ut}} + \frac{\sigma_a}{S_e} = \frac{1}{fos}$$

Soderberg Line:

- Soderberg proposed a straight line (failure stress line) connecting σ_e or S_e with σ_y or S_{yt} for the design.

- So a straight line joining S_e on the ordinate & S_{yt} on the abscissa is called as Soderberg line.

- Equation of Soderberg line is given by.

$$\frac{\sigma_m}{S_{yt}} + \frac{\sigma_a}{S_e} = \frac{1}{fs}$$

or $$\frac{\sigma_m}{\sigma_y} + \frac{\sigma_v}{\sigma_e} = \frac{1}{fs}$$

Riveted Joints.

i. Joints used in mechanical assemblies are classified into two groups
- Permanent
- Separable.

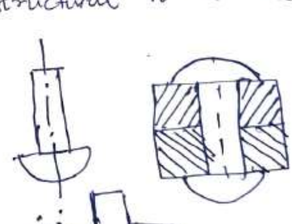
<u>Permanent joints</u>: are those joints which can not be disassembled without damaging the assembled parts.

example Riveted & welded joints.

<u>Separable Joints</u>: are those joints are those joints which permits disassembly and reassembly without damaging the assembled parts.

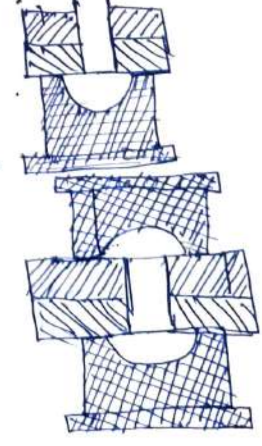
example Bolted joints, cotter joints, screw fasteners, Keys, couplings.

Riveted joints are widely used for making permanent joints in engineering applications like boilers, pressure vessels, reservoirs, ships, trusses, frames, cranes, structural works like bridges & roof trusses.



<u>Rivet</u>: It consists of a cylindrical shank with a head at one end as shown in figure.

- Rivet is specified by its shank diameter
- Mild steel, wrought iron, copper & Aluminium is used for riveting.



<u>Riveting</u>: It is the process of forming a riveted joint.

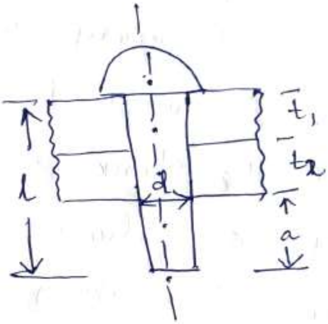
The rivet head is formed on the shank by an upsetting process in a machine called an automatic header.

- first the rivet is inserted in the holes of the parts to be assembled. and the head is firmly held against the back up bar.
- Then in the riveting process the protruding end of the shank is upset by hammer blows to form the closing head.
- In rivet terminology the closing head is called as point.
- Two methods for riveting
  - Hand Reveting (by hammering)
  - Machine riveting. (by die press with the help of pneumatic/hydraulic pressing)
- On the basis of temperature of the shank riveting method is classified as
  - Hot riveting
  - cold riveting.

Hot Reveting	Cold Reveting
- Rivet shank is heated upto 1000°c to 1100°c then hammered when it becomes red hot.	- No such heating.
- shank of the rivet is subjected to tensile stress	- shank mainly subjected to shear stress.
- Carried out for steel rivets with dia more than 10 mm	- Less than 10 mm shank made up of steel, brass, copper & Aluminium.

Types of Rivet heads

(i) Snap head Rivet
(ii) Pan head Rivet
(iii) Counter sunk head rivet
(iv) Cone head
(v) flat head



Length of rivet shank

$$l = (t_1 + t_2) + a$$

t_1, t_2 are thickness of shank

a = Length of the shank portion necessary to form the closing head.

$a = 0.7d$ to $1.3d$

d = diameter of shank of rivet.

Types of Riveted Joints.

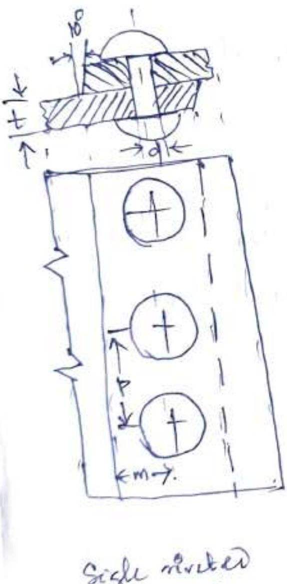
Classified into two groups
- Lap joint
- Butt joint.

Lap joint consists of two overlapping plates, which are held together by one or more rows of rivets.

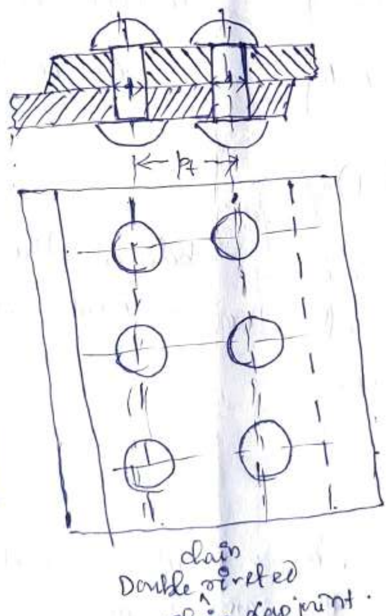
So depending upon the number of rows; the lap joints are further classified into.
- Single riveted Lap joint
- Double riveted Lap joint
- Tripple riveted Lap joint.

In double riveted lap joints, the rivets can be arranged in a
- Chain pattern
- Zig-Zag pattern.

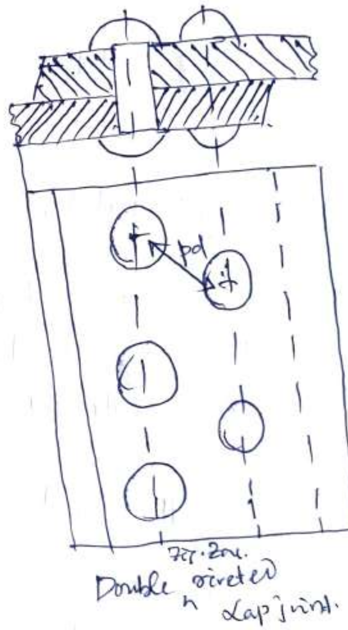
- In chain riveted lap joint; the rivets are arranged in such a way that rivets in different rows are located opposite to each other.
- In Zig-Zag riveted joint the rivets are arranged in such a way that every rivet in a row is located in the middle of the two rivets in the adjacent row.



Single riveted lap joint



Double riveted chain lap joint.



Double zig-zag riveted lap joint.

Terms used in riveted joint

Pitch: (p) The pitch of the rivet is defined as the distance between the centre of one rivet to the centre of adjacent rivet.

$p = 3d$ d = shank dia.

Margin (m) : the margin is the distone between the edge of the plate to the centreline of rivets in the nearest row.

$$m = 1.5\, d.$$

Transeverse Pitch (P_t): (Also called as back pitch / row pitch)

It is the distance between two consecutive rows of rivets in the same plate.

$$p_t = 0.8\, p \quad \text{for chain riveting.}$$
$$= 0.6\, p \quad \text{for zig zag riveting.}$$

Diagonal pitch (p_d):

It is the distance between the centre of one rivet to the centre of the adjecent rivet located in the adjecent row.

Butt Joints

- It consists of two plates which are kept in alignment against each other in the same plane and a strap or cover plate is placed over these plates and riveted to each plate.
- Depending upon the number of rows of rivets in each plate the butt joints are classified as
  - Single row butt joint
  - Double row butt joint
-

Advantage of Butt joint over lap joint

The line of action of the force acting on two plates joined by butt joint lies in the same plane. therefore there is no bending moment on the joint and no warping of the plates.

But butt joint is costly as separate/additional strap plates are required.

Diamond / Lozenge Joints.

Typical riveted joint used in construction work such as bridges, trusses & cranes. Here plate of smaller width is required for joint so often called as economical joint also.

Revet Material

- Mild steel. - Carbon - 0.23%, Sulphur - 0.05%, Phosphorus = 0.05% } (max)

  for Joining ferrous material

- for joining non. ferrous material

  Copper, brass, bronze, Aluminium alloy.

Failure of riveted Joints

Strength of riveted joints is defined as the force that the joint can withstand without causing failure. If the force exceeds the permissible limit failure may occur in the form of.

(i) Shear failure of the rivet.

(ii) Tensile failure of the plate between rivets.

(iii) crushing failure of the plate.

Shear strength of the rivet

If the rivet is in single shear.

$$P_s = \frac{\pi}{4} d^2 \tau = \text{shear resistance of rivet per pitch length.}$$

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

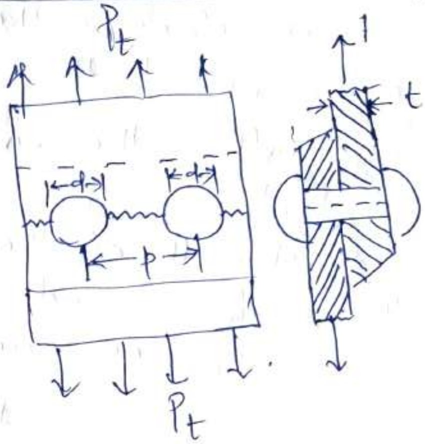
for double riveted joint $n = 2$. tripple $n = 3$.

In case of double strap single riveted butt joint.

$$P_s = 2\left[\frac{\pi}{4} d^2 \tau n\right]$$

1 Tensile strength of plate between rivets.

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



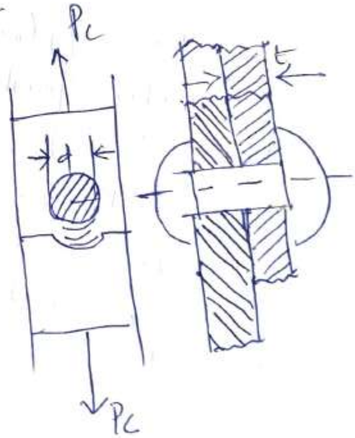
P_t = tensile resistance of plate per pitch length.

p = pitch of rivets.

t = thickness of plate

σ_t = permissible tensile stress of plate material

Crushing strength of Plate



This type of failure occurs when the compressive stress between the shank of the rivet & the plate exceeds the yield stress in compression. This results in elongating the rivet hole in plate & loosening of the joint.

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

P_c = crushing resistance of plate per pitch length.

σ_c = permissible compressive stress of the plate material.

Efficiency of Riveted Joints

- It is defined as the ratio of the strength of riveted joint to the strength of unriveted solid plate.
- The strength of the riveted joint is the lowest value of P_s, P_t, P_c
- The strength of the solid plate of width equal to the pitch p and thickness 't' subjected to tensile stress σ_t is given by

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

Therefore efficiency is given by $\eta = \dfrac{\text{Lowest of } P_s, P_t, \& P_c}{P}$

Caulking & Fullering.

→ In the applications like pressure vessels & boilers, the riveted joint should be leakproof & fluid tight.

→ Caulking & fullering are the processes used to obtain such leakproof riveted joints.

→ The caulking process is applied to the edges of the plate in a lap joint & the edges of the strap plate in a butt joint.

→ The edges are first beveled to approximately 70' to 75°

& caulking tool is hammered on the edges. as in fig. with the help of hand hammer or pneumatic / hydraulic hammer.

→ Head of the rivet is also hammered.

→ The blows of the caulking tool closes the surface asperities and cracks ~~between~~ on the contacting surfaces between the plates and also between the rivet & plate.

→ Great care must be excercised to prevent injury to plates.

Fullering Process :

⇒ It is similar to the caulking process except for the shape of the tool. Th

⇒ The width of the fullering tool is equal to the thickness of the plate being hammered.

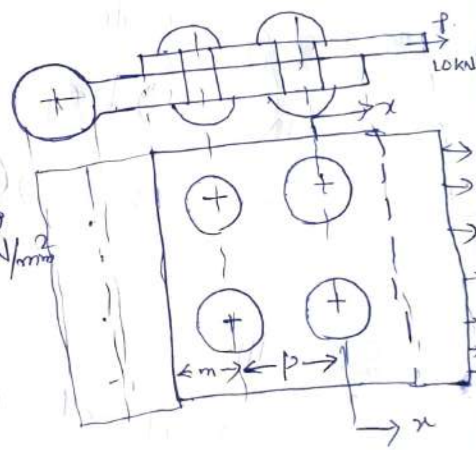
⇒ The blows of the fullering tool results in simultane un pressure on the entire edge of plate.

Example (Q) A brake band attached to the hinge by means of a riveted joint is as shown in fig. Determine the size of the rivets needed for the load of 10 KN. Also determine the width of the band. The permissible stress for the band, and rivets in tension, shear & compression are 80, 60, & 120 N/mm² respectively. Assume.

margin (m) = 1.5 d.

transverse pitch (P_t) = P.

find the pitch of the rivets.



Data Given P = 10 KN t = 3 mm σ_t = 80 N/mm²

τ = 60 N/mm² σ_c = 120 N/mm²

Step-1 Diameter of rivet.

There are four rivets in the lap joint ; which are subjected to single shear. From shear consideration.

$$4\left[\frac{\pi}{4}d^2\tau\right] = P \quad \text{or} \quad 4\left[\frac{\pi}{4}d^2(60)\right] = 10\times10^3$$

$$\Rightarrow d = 7.28 \approx 8 \text{ mm}.$$

From crushing Consideration.

$$4\,d t \sigma_c = P \quad \text{or} \quad 4(d)3(120) = 10\times10^3$$

$$d = 6.94 \text{ or } 7 \text{ mm}$$

from 'a' & 'b' it is observed that shearing becomes the criterion for failure.

$$\text{So } d = 8 \text{ mm}.$$

Step II

Width of the band.

Considering tensile strength of plate along the sec. 'xx'.

$$(w - 2d)\, t\, \sigma_t = P$$

$$(w - 2\times8)(3)(80) = 10\times10^3$$

$$\Rightarrow w = 57.67 \text{ or } 60 \text{ mm}$$

Step III

Pitch of rivets.

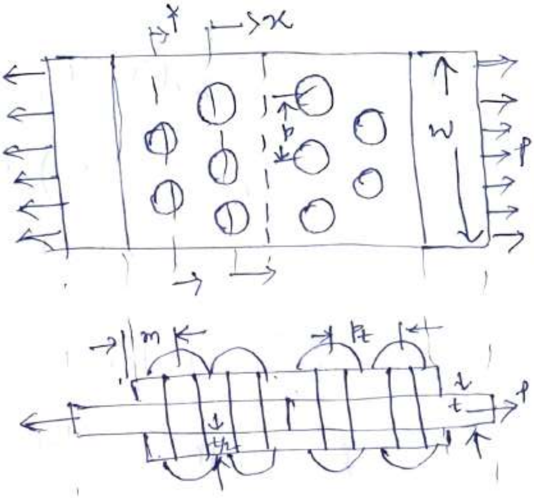
$$m = 1.5d = 1.5\times(8) = 12 \text{ or } 15 \text{ mm}.$$

$$p + 2m = w \quad \text{or} \quad p + 2(15) = 60$$

$$p = 30 \text{ mm}. = p_t.$$

Q. Two flat plates subjected to a tensile force P are connected together by means of double strap butt joint as shown in fig.

The force P is 250 KN. and the width of the plate 'w' is 200mm. The rivets and plates are made up of the same steel. The permissible stresses in tension compression & shear are 70, 100 & 60 N/mm² respectively.



Calculate.

(i) The diameter of the rivets.

(ii) Thickness of plates

(iii) Dimensions of the seam i.e. p, p_t, m

(iv) Efficiency of the boiler joint.

Given data.

$P = 250$ KN, $w = 200$ mm, $\sigma_c = 100$ N/mm²

$\tau = 60$ N/mm²

$\sigma_t = 70$ N/mm²

Step. I.

Diameter of rivets.

There are 5 rivets subjected to double shear.

$$P_s = 2\left[\frac{\pi}{4} d^2 \tau n\right] = 250 \times 10^3$$

$$\Rightarrow 2 \times \left[\frac{\pi}{4} d^2 (60)(5)\right] = 250 \times 10^3.$$

$$\Rightarrow d = 23.03 \simeq 25 \text{ mm}.$$

(N/mm²)

Step II Thickness of the plate.

Let the failure occurs in tension at the two hole section without affecting the three hole section:

then $(w-2d)t\,\sigma_t = P$

$$(200 - 2\times 25)(t)(70) = 250\times 10^3$$

$$\Rightarrow t = 23.81 \simeq 25\ mm.$$

Step - III Pitch of the rivet.

The pitch of the rivet is given by

$$p = \frac{\text{width of the plate}}{\text{no of rivets}} = \frac{200}{3} \simeq 66.67\ mm \simeq 65\ mm.$$

then $m = 1.5d = 1.5(25) = 37.5 \simeq 40\ mm$

$P_t = 0.6p = 0.6(65) = 39 \simeq 40\ mm.$

Step iv Efficiency of joint

$$P_s = 2\left[\frac{\pi}{4}d^2\tau n\right] = 2\left[\frac{\pi}{4}(25)^2(60)(5)\right] = 294524.31\ N.$$

$$P_t = (w-2d)t\sigma_t = (200 - 2\times 25)(25)(70) = 262500\ N.$$

$$P_c = dt\sigma_c n = (25)(25)(100)(5) = 312500\ N.$$

The lowest among them is 262500

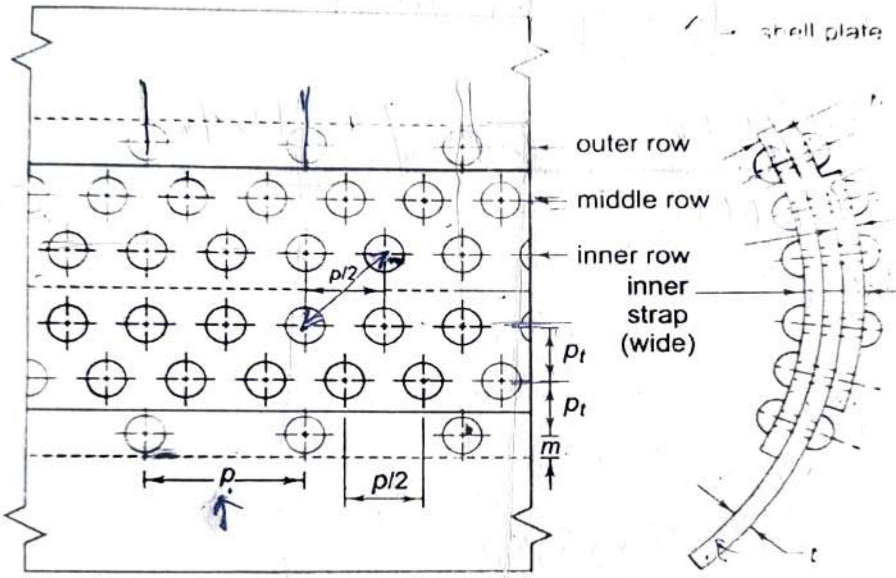
The strength of solid plate is given by

$$P = wt\sigma_t = (200)(25)(70) = 350000 N.$$

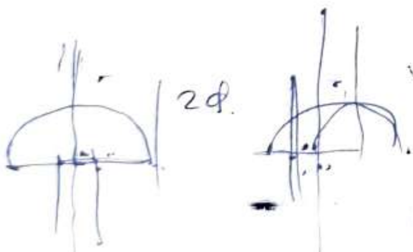
Therefore the efficiency of the plate is given by

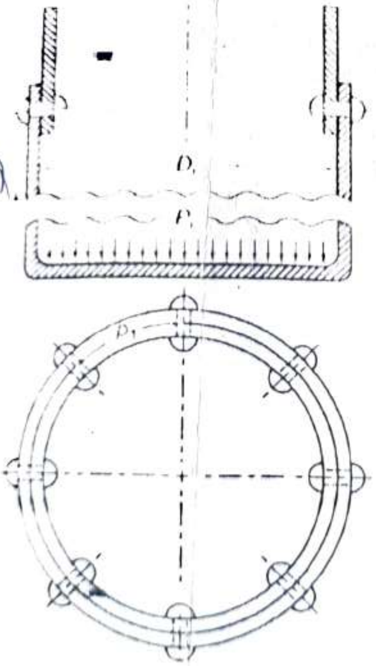
$$\eta = \frac{262500}{350000} = 0.75 / 75\%.$$

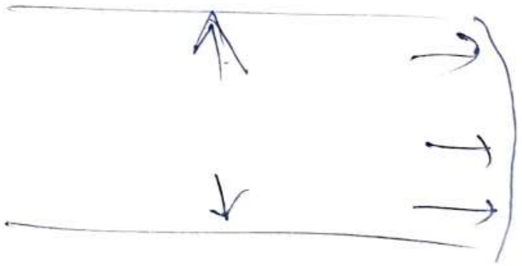
Boiler Joints. (Riveted)

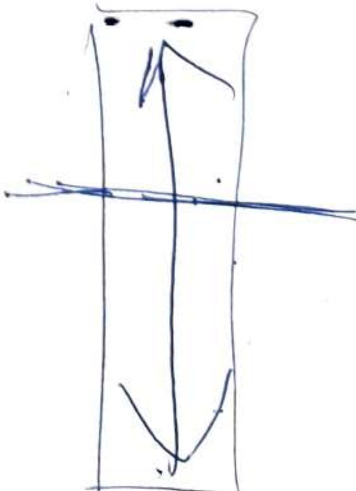


Longitudinal Butt Joint









Circumferential Lap Joint

Boiler Joints.

For a cylindrical boiler shell there are two types of riveted joints
- Longitudinal Butt joint.
- Circumferential Lap joint.

* First the plate of the boiler shell is bent to form the ring & two edges of the plate are joined by a longitudinal butt joint.
* This longitudinal butt joint is usually a double-strap tripple riveted butt joint.
* This Longitudinal joint makes a ring from the steel plate.

- But to get the required length of the boiler shell; one ring is joined with another with the help of circumferencial joint.
* For this one ring is kept overlapping over the adjecent ring. and the two rings are joint by circumferential Lap joint.

According to Indian boiler regulation act the boiler shells must with stand steam pressure & also prevent leakage. During this boiler shell is subjected to circumferential & longitudinal tensile stress; from which circumfential stress is twice the longitudinal stress.

$\sigma_t = \frac{S_{ut}}{fs}$

Design Consideration.

(i) Thickness of Boiler shell.

$$\boxed{t = \frac{P_i D_i}{2\sigma_t}}$$

t = thickness of cylindrical wall. (mm)
D_i = innerdiameter of cylinder (mm)
P_i = Internal pressure (N/mm^2)
σ_t = permissible tensile stress, (N/mm^2)

If we consider the efficiency of the boiler's joint

$$t = \frac{P_i D_i}{2 \sigma_t \eta}$$

where η = efficiency of riveted joint = $\frac{\text{strength of riveted joint}}{\text{strength of solid plate.}}$

In due course of time wall of boiler shell may be subjected to thinning due to corrosion. So suitable corrosion allowance (CA) is given. (1.5 to 2 mm).

It is the additional metal thickness over & above that required to withstand internal pressure.

$$t = \frac{P_i D_i}{2 \sigma_t \eta} + CA.$$

(ii) Diameter of Rivet

Although Indian boiler regulation won't specify the formula for calculating the rivet diameter, we can follow the empirical relationship. i.e.

(a) when the thickness of plate > 8mm.
then $\boxed{d = 6\sqrt{t}}$ } Unwin's formula.

(b) If plate thickness < 8mm
then diameter of rivet is obtained by equating shear resistance of rivets to crushing resistance

(c) In no case the dia of rivet is < plate thickness.

(iii) Diameter of rivet hole

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

↑ dia of rivet.

(iv) Pitch of Rivet.

$$p = \frac{(n_1 + 1.875 n_2)\pi d^2 \tau}{4 t \sigma_t} + d.$$

where n_1 = no of rivets subjected to single shear per pitch length.
(as the outer strap is smaller than the inner strap)
n_2 = no of rivets subjected to double shear per pitch length.

According to Indian Boiler Regulation.

(i) pitch of the rivets should not be less than (2d) to enable the forming of rivet head

$$p_{min} = 2d$$

(ii) In order to provide leakproof joint; the maximum pitch is given by.

$$p_{max} = Ct + 41.28$$

value of 'c' is in table.

(V) Transverse pitch (p_t)

For zigzag riveting: $p_t = 0.33p + 0.67d$.

For chain riveting: $p_t = 2d$

If * more than one rows of rivets
* equal no. of rivets in each row

* But if the no of rivets in the outer row is One half of the inner row then

$$p_t = 0.2p + 1.15d.$$

* The minimum distance between the rows in which there are full number of rivets is given by

$$p_t = 0.165p + 0.67d.$$

(vi) <u>Margin</u>

It is the distance between the centre of the rivet hole from the edge of the plate.

$$m = 1.5d.$$

(vii) <u>Thickness of strap</u>

* According to IBR; when the straps are of unequal width & in which every alternate rivet in the outer row is omitted.

$$t_1 = 0.75t \quad \text{(for wide strap)}.$$

$$t_1 = 0.625t \quad \text{(for narrow strap)}.$$

* when straps are of equal width & in which every alternate rivet in the outer row is omitted

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

Circumferential Lap Joint.

- It is used for connecting different cylindrical rings together & form the boiler shell.
- Usually this joint is also used to connect the end covers with the cylindrical shell.

Design Considerations

(i) Thickness of cylindrical shell:

$$t = \frac{P_i D_i}{2 \sigma_t \eta} + CA.$$

(ii) Diameter of rivet:

$$(d)_{t > 8mm} = 6\sqrt{t}$$

(iii) Number of rivets:

$$n = \frac{P}{\left(\frac{\pi}{4} d^2 \tau\right)} = \frac{\left[\frac{\pi}{4} D_i^2\right] P_i}{\left[\frac{\pi}{4} d^2 \tau\right]} = \left(\frac{D_i}{d}\right)^2 \left(\frac{P_i}{\tau}\right)$$

(iv) Pitch of rivet:

$$\eta = \frac{p_1 - d}{p_1}$$

where η_1 = efficiency of the circumferential joint

p_1 = pitch of rivet

d = Diameter of rivet

$$\left(\begin{array}{l} p_{min} = 2d \\ p_{max} = Ct + 41.28 \end{array} \right)$$

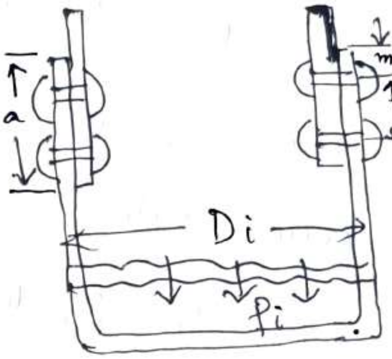
(v) Number of rivets in one rows:

$$n_1 = \frac{\pi (D_i + t)}{p_1}$$

$$\text{Number of rows} : \frac{\text{total Number of rivets in joint}}{\text{number of rivets in one row}}$$

Transverse Pitch.

In a double riveted circumferential lap joint, for a cylindrical pressure vessel; the transverse pitch (p_t) is the distance between two rows of rivets.



The overlap of plate is denoted by 'a'

which is given as $a = p_t + 2m$

m = margin

$p_t = 0.33p + 0.67d$ = Zig Zag riveting

$p_t = 2d$ Chain riveting

$m = 1.5d$

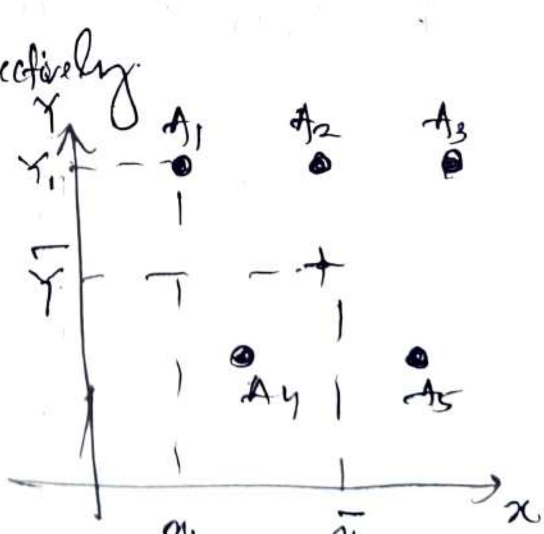
Eccentrically Loaded riveted joints:

In structural connections a group of rivets is frequently employed. Let $A_1, A_2, \ldots A_5$ are the ~~group of~~ area of the rivets at co-ordinates $(x_1 y_1), (x_2, y_2), \ldots (x_5 y_5)$ respectively.

G. is the centre of gravity.

Location of center of gravity.

$(\bar{x}, \bar{y})$

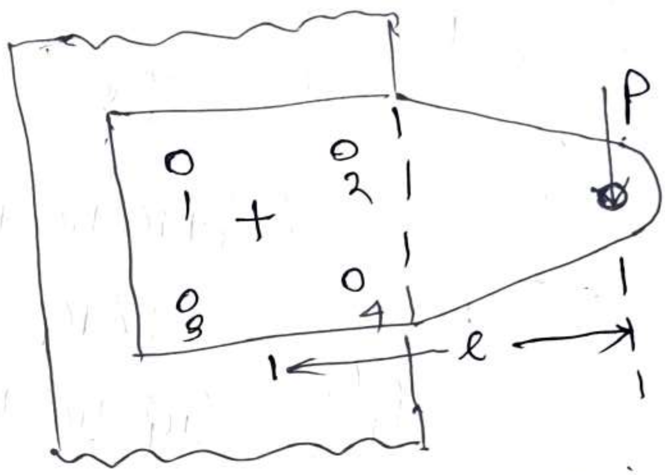


$$\bar{x} = \frac{A_1 x_1 + A_2 x_2 + \cdots + A_5 x_5}{A_1 + A_2 + A_3 + \cdots + A_5}$$

$$\bar{x} = \frac{\sum A_i x_i}{\sum A_i}$$

Similarly $\quad \bar{y} = \dfrac{\sum A_i y_i}{\sum A_i}$

An eccentrically loaded rivetted connection is shown in fig.



The eccentricity of the external force P is 'e'.

This eccentric force can be considered as equivalent to an imaginary force P at the centre of gravity and a moment $(P \times e)$ about the same point.

P_1', P_2', P_3' & P_4' are the primary shear forces.

which can be given as.

$$P_1' = P_2' = P_3' = P_4' = \frac{P}{\text{No of bolts}}.$$

The moment $(P \times e)$ about the center of gravity results in secondary shear forces, P_1'', P_2'', P_3'', P_4''.

If r_1, r_2, r_3 & r_4 are the radial distances of the rivets centers from the center of gravity then

$$\boxed{P \times e = P_1'' r_1 + P_2'' r_2 + P_3'' r_3 + P_4'' r_4}$$

It is assumed that; the secondary shear force at any ~~bolt~~ rivet is proportional to the distance from the center of gravity.

$P_1'' = C r_1$; $P_2'' = C r_2$; $P_3'' = C r_3$; $P_4'' = C r_4$.

constant of proportionality

So $$\boxed{C = \frac{P e}{(r_1^2 + r_2^2 + r_3^2 + r_4^2)}}$$

Therefore $$\boxed{P_1'' = \frac{P e r_1}{(r_1^2 + r_2^2 + r_3^2 + r_4^2)}}$$ and so on.

The primary & secondary shear forces are vectorically added to get the resultant shear force.

denoted by P_1, P_2, P_3, P_4.

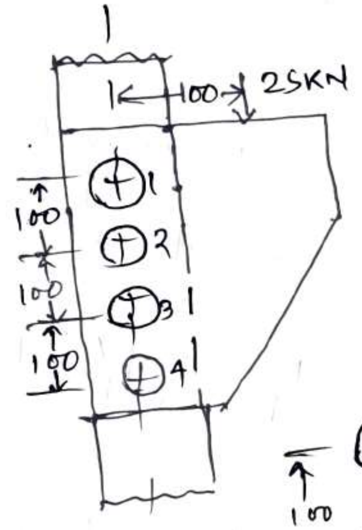
Let P_4 & P_2 are the maximum shear force

Equating maximum shear force P_2 & P_4 to the shear strength of rivet

$$\boxed{P_2 = P_4 = \left[\frac{\pi}{4} d^2\right] \tau}$$

The above equation is used to find out the diameter of rivet.

Quest A bracket; attached to a vertical column by means of four identical rivets ; is subjected to an eccentric force of 25 kN as shown in fig. Determine the diameter of rivets; if the permissible shear stress is 60 N/mm².



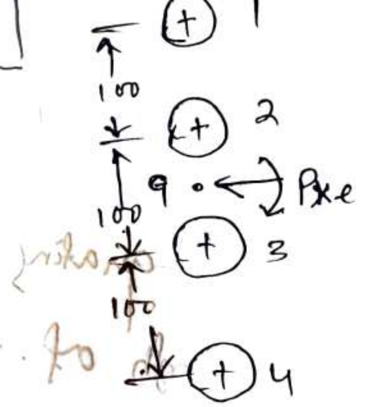
Given : $P = 25 \text{ kN}$

$e = 100 \text{ mm}$

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

(i) Primary shear force

$$P_1' = P_2' = P_3' = P_4' = \frac{P}{4} = \frac{25 \times 10^3}{4} = 6250 \text{ N}$$



(ii) Secondary shear force

By symmetry ~~secondary~~ centre of gravity of ~~...~~ four rivets is located midway between the centre of rivets between 2 & 3.

$r_1 = r_4 = 150$

$r_2 = r_3 = 50$.

$$C = \frac{Pe}{(r_1^2 + r_2^2 + r_3^2 + r_4^2)} = \frac{(25 \times 10^3)(100)}{(150)^2 + (50)^2 + (50)^2 + (150)^2} = 50$$

$P_1'' = P_4'' = Cr_1 = 50 \times 150 = 7500 \text{ N}.$

$P_2'' = P_3'' = Cr_2 = (50)(50) = 2500 \text{ N}.$

(iii) Resultant Shear force

As rivets 1 & 4 are located at the farthest distance from the centre of gravity are subjected to maximum shear force:

As the primary & secondary shear forces acting on rivets 1 & 4 are right angle to each other.

$$P_1 = \sqrt{P_1'^2 + P_1''^2} = \sqrt{(6250)^2 + (7500)^2}$$

$$= 9762.81 \text{ N.}$$

) Diameter of rivets:

Equating the resultant shear force to the shear strength of rivet;

$$P_1 = \frac{\pi}{4} d^2 \tau \quad \text{or} \quad 9762.81 = \frac{\pi}{4} d^2 (60)$$

$$d = 14.39 \simeq 15 \text{ mm.}$$

Welded Joints

Welding can be defined as a process of joining metallic Parts by heating to a suietable temperature with or without pressure.

Advantages of welded joints over riveted joints.

* Riveted joints requires additional cover plates / straps which increases the weight but since there are no such additional parts welding assembly results in light weight construction
* Also Cost of welding assembly is less.
* welded assemblies are tight & leak proof as compared to riveted joints.
* Production time is less.
* Due to formation of holes in riveted joints ; it results in stress concentration.
* strength of welded joints are high.
* wleded structures are smooth & pleasant appearance.
* Alternation / modifications can be easily possible with the existing structure.

Disadvantage.

- welded joints may subjected to residual stresses due to non-uniform heating of parts being joined.
- This localized thermal stresses may result from uneven heating & cooling during fusion.

This can be avoided by

* Pre heating the weld area
* Stress releaving the weld area by using proper heat treatment process like normalizing / Annealing.
* Hand peening : Hammering the weld along length when it is hot.

6

Welded joints are divided into two groups.

- Butt joints
- fillet joints.

Butt Joints.

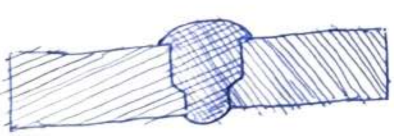
Welded joint between two components lying apparently in the same plane is called as butt joint.

Types:

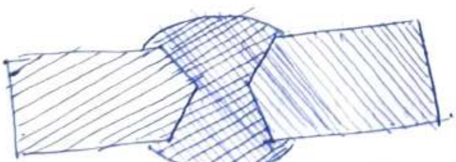
(i) **Square Butt joint:** If the thickness of plate is less than 5mm. it is not necessary to bevel the edges and the edges remained square in shape w.r.to the plates.

(ii) **V-butt joint:** When the thickness of plate is between 5 to 25mm The edges are beveled to form a 'V' shape before welding. therefore the joint is called a 'V' joint

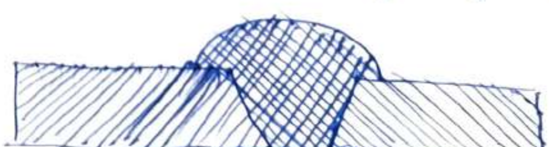
(iii) **Single 'U' joint:** When the thickness of plate is more than 20 mm; the edges of the two plates are machined to form a 'U' shape.



(iv) **Double 'V' joint:** When the thickness of plate is more than 30 mm; a double welded 'V'-joint is used.



(v) **'V' joint with Backing Strip:** When the welding is done from one side; a backing strip is used to avoid the leakage of the molten metal on the other side.



Fillet Joint

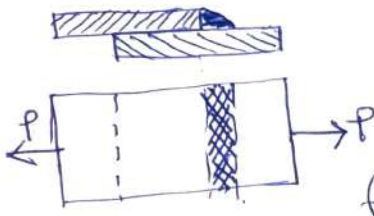
A fillet joint also called as Lap joint is a joint between two overlapping plates or components.

A fillet weld consists of an approximately triangular cross-section joining two surfaces at right angle to each other.

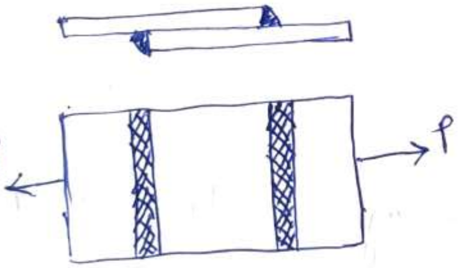
Two types of fillet jointe
- Transverse.
- Parallel.

Transverse fillet Joint: If the direction of weld is perpendicular to the direction of force acting on the joint.

- Single transverse fillet joint
- Double transverse fillet joint

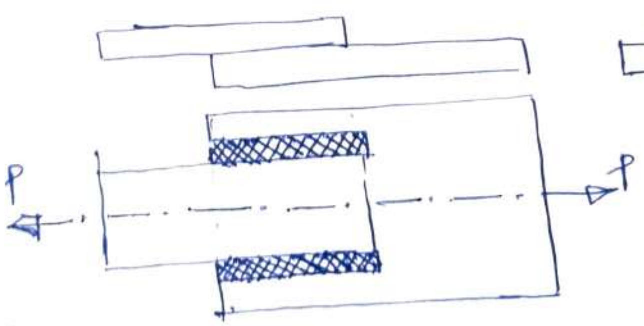


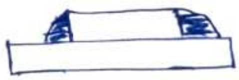
Single transverse fillet weld.



Double transverse fillet weld

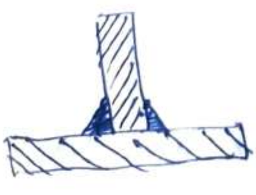
Parallel / Longitudinal: If the direction of weld is parallel to the direction of force acting on the joint



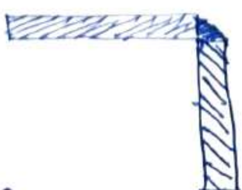


Double Parallel fillet weld.

Apart from these there are some specific types of welded joints.



tee-Joint

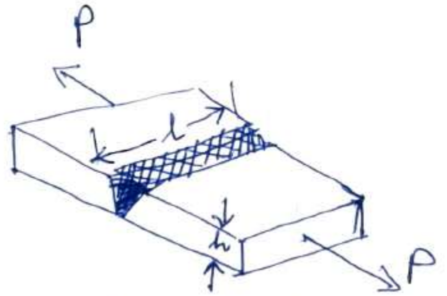


Corner Joint

Edge Joint.

Strength of Butt weld

If The butt weld joint subjected to tensile force 'P' then the average tensile stress σ_t =



Butt weld in tension

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

where σ_t = tensile stress in the weld (N/mm²)
P = tensile force on the plates. (N)
h = throat of the butt weld (mm)
l = Length of the weld (mm).

If the throat thickness is equal to the plate thickness 't' we can write

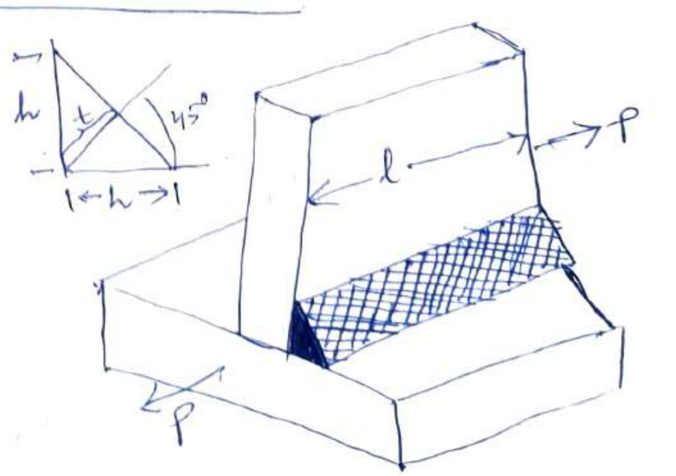
$$\boxed{P = \sigma_t\, t\, l}$$

If we consider the efficiency of the welded joint then

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

Strength of parallel fillet weld.

As shown in fig; a parallel fillet weld is subjected to tensile force P.



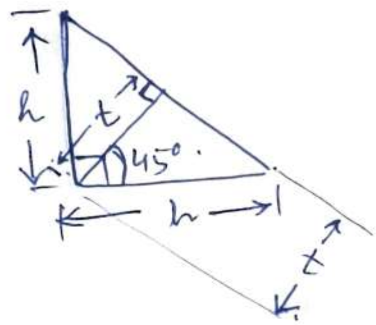
There are two terms

leg = h

throat = t.

the size of the (fillet) weld is specified by the leg length.

h = plate thickness = leg length.



The throat is the minimum cross-section of the weld located at 45° to the leg dimension.

therefore $t = h(\cos 45°)$

$t = 0.707\,h$.

failure of the fillet weld occurs due to shear along the minimum cross-section at the throat.

Cross sectional area at the throat $= (t\,l) = (0.707\,h\,l)$

So $$\boxed{\tau = \frac{P}{0.707\,h\,l}}$$

So the strength equation for parallel fillet ~~well~~ weld is given by

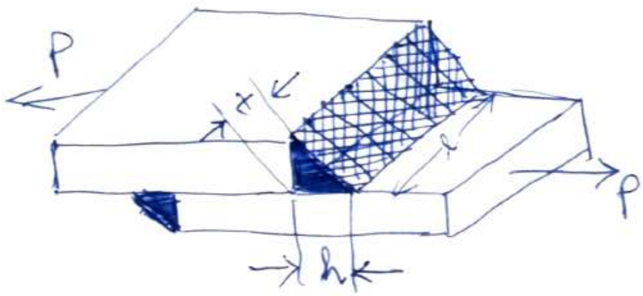
$$P = 0.707\,h\,l\,\tau.$$

If there are two welds on two sides of equal length

then $$P = 2(0.707)\,h\,l\,\tau = 1.414\,h\,l\,\tau.$$

Note: while determining the required length of the weld 15 mm should be added to the length of each weld. to allow starting & stopping of weld run.

Strength of Transverse fillet weld.



A transverse fillet weld subjected to a tensile force 'P' is shown in figure.

These transverse fillet welds are subjected to tensile stresses. And the failure due to this tensile stress will occour at throat section.

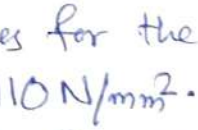
$$\sigma_t = \frac{P}{tl} = \frac{P}{0.707\,hl}.$$

or $P = 0.707\,hl\sigma_t$. for single fillet

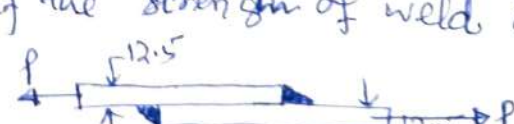
$P = 1.414\,hl\,\sigma_t$ for double parallel fillet.

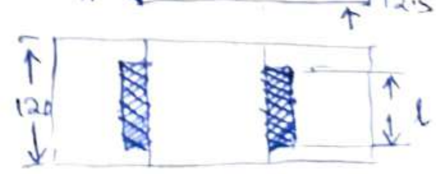
NB:

The inclination plane where maximum shear stress is induced is 67.5^0 to the leg dimension.



Q. Two steel plates; 120 mm wide and 12.5 mm thick are joined together by means of double transverse fillet welds as shown in fig. The maximum tensile stress for the plates and the welding material should not exceeds 110 N/mm². find the required length of the weld. if the strength of weld is equal to the strength of the plates.





Given Data:

width of the plate = 120mm

$t = 12.5\,mm$.

$\sigma_t = 110\,N/mm^2$.

$h = 12.5\,mm$ (for welds).

Step - I

Tensile force on Plates

As the plates are subjected to tensile stresses the maximum tensile force acting on the plates is given by

$$P = w\,t\,\sigma_t = 120 \times 12.5 \times 110 = 165000\,N.$$

Step II

Length of the weld.

In this case it is a double transverse fillet weld hence from the equation.

$$P = 1.414\, h\, l\, \sigma_t = 165000$$

$$\Rightarrow 1.414\,(12.5)\,l\,(110) = 165000.$$

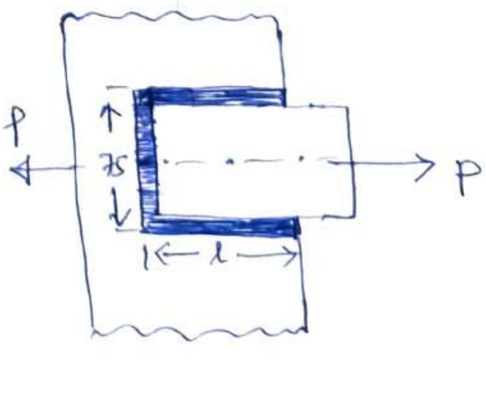
$$\Rightarrow l = 84.87\ mm.$$

Adding 15 mm for starting & stopping of weld run the required length of the weld will be

$$l = 84.87 + 15 = 99.87 \simeq 100\,mm.$$

Q. A Plate, 75 mm wide & 10 mm thick is joined with another steel plate by means of single transverse & double parallel fillet welds as shown in fig.

The joint is subjected to a maximum tensile force of 55 KN. The permissible tensile & shear stresses in the weld material are 70 & 50 N/mm² respectively. Determine the required length of each parallel fillet weld.



Solution:

Given $P = 55 KN$, $\tau = 50 N/mm^2$ $\sigma_t = 70 N/mm^2$

$t = h = 10 mm$.

Step - I. Strength of transverse & parallel fillet weld

(a) Strength of the transverse fillet weld. =

$$P_1 = 0.707\, h\, l\, \sigma_t = 0.707\,(10)(75)(70)$$

$$= 37117.5 \text{ N.}$$

(b) Strength of the double parallel fillet weld =

$$P_2 = 1.414\, h\, l\, \tau = 1.414\,(10)(l)(50)$$

$$= 707 \times l \text{ N.}$$

Step - II Length of parallel fillet weld

The total strength of the joint should be 55 KN.

So $$37117.5 + 707 \times (l) = 55 \times 10^3$$

$$\Rightarrow l = 25.29 \text{ mm.}$$

Adding 15 mm for starting & stopping of the weld run

$$l = 25.29 + 15 = 40.29 \simeq 45 \text{ mm}$$

Axially Loaded Unsymmetrical Welded Joints.

In certain applications; unsymmetrical sections such as angle or T are welded to the steel plates or the beams.

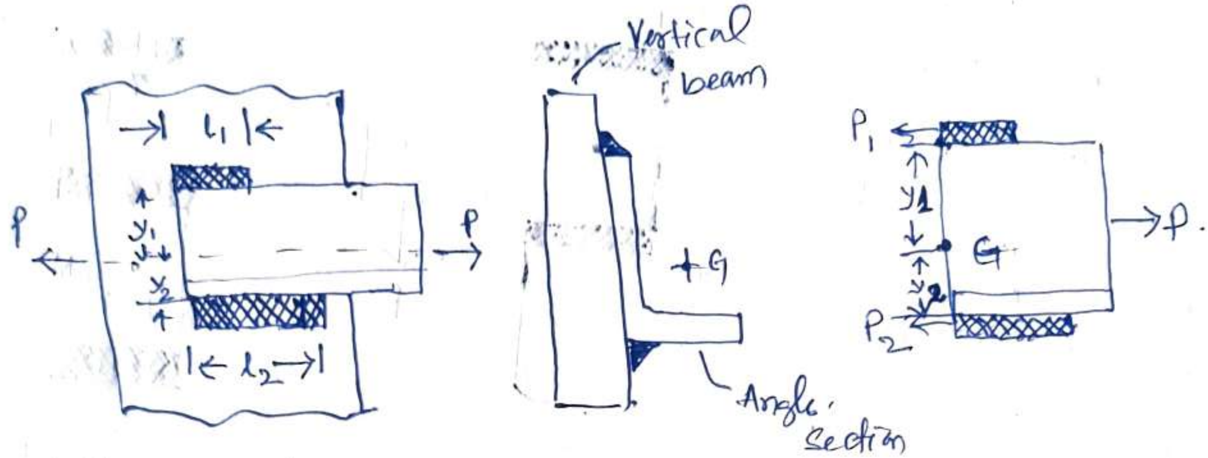


figure shows an angle section welded to a vertical beam by means of two parallel fillet welds 1 & 2.

G is the centre of gravity of angle section.

suppose P_1 & P_2 are the resisting forces set up in the welds 1 & 2 respectively. The we can write

$$P_1 = 0.707\, h\, l_1\, \tau$$

$$P_2 = 0.707\, h\, l_2\, \tau$$

$P = P_1 + P_2$; & Moment of forces about centre of gravity will be zero.

$$P_1 y_1 = P_2 y_2.$$

$$(0.707\, h\, l_1\, \tau)\, y_1 = (0.707\, h\, l_2\, \tau)\, y_2$$

$$\Rightarrow \quad l_1 y_1 = l_2 y_2$$

& $l_1 + l_2$ = Total length of weld = $l = \dfrac{P}{0.707 h \tau}$

from this l_1 & l_2 can be calculated.

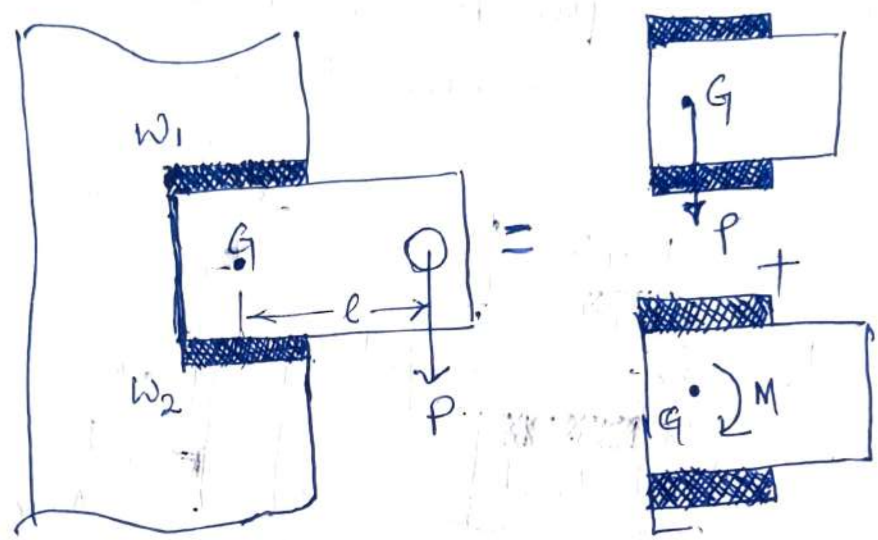
Eccentric Loading in the plane of welds.

w_1 & w_2 = two fillet welds

P = eccentric force

G = centre of Gravity

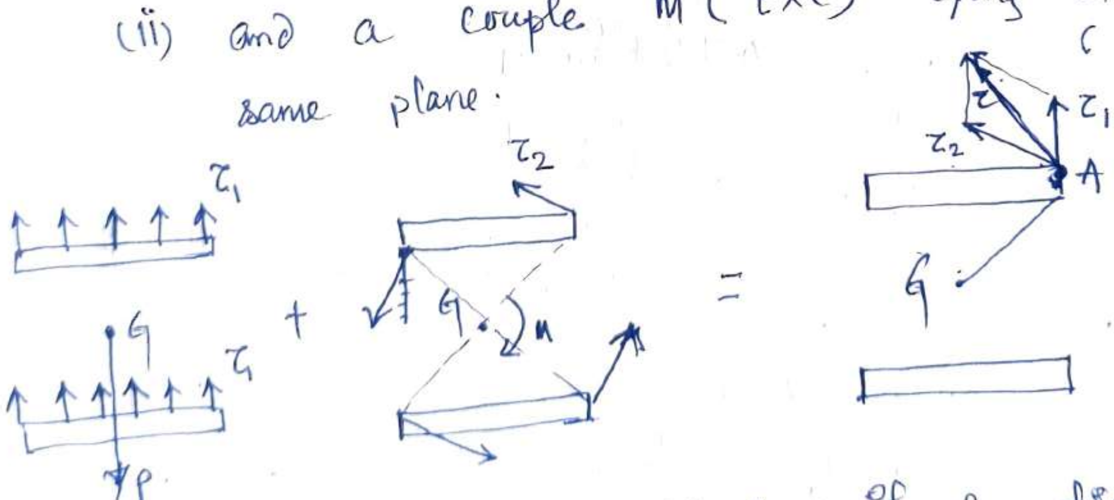
e = eccentricity between the CG & line of action of force P.



The design of welded joint subjected to an eccentric loading 'P' at a distance of 'e' from centre of gravity can be replaced by

(i) by an equal & similarly directed force P acting through the CG

(ii) and a couple M (P×e) lying in the same plane.



τ_1 = Primary shear stress & it is uniformly distributed over the throat area of all welds. $= P/A$.

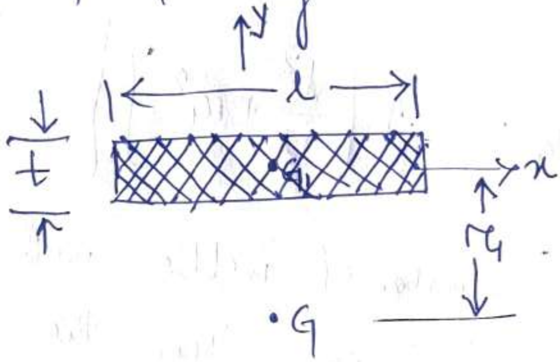
τ_2 = Secondary shear stress due to couple 'M' in the throat area of welds $= \frac{Mr}{J}$

where r = distance of a point in the weld from G

J = polar moment of inertia of all welds about G

So the secondary shear stress at any point of weld about 'G' is proportional to the distance from the CG. which is maximum at point A. which is farthest ~~appart~~. apart.

The resultant shear stress at any point is obtained by vector addition of primary & secondary shear stress.



weld length = l

throat thickness = t.

The moment of inertia about CG; G_1 is given by

$$I_{xx} = \frac{l t^3}{12} \quad \& \quad I_{yy} = \frac{t l^3}{12}.$$

since t is very small compared to 'l'

I_{xx} is negligible as compared to I_{yy}

$$J_{G_1} = I_{xx} + I_{yy} = I_{yy}$$

$$J_{G_1} = \frac{t l^3}{12} = \frac{(t l) l^2}{12} = \frac{A l^2}{12}$$

Where 'A' = throat area

J_{G_1} = Polar moment of inertia of the weld about its centre of gravity.

Then the polar moment of inertia about an axis passing through G is determined by parallel axis theorem.

$$J_G = J_{G_1} + A r_G^2 .$$

where r_G = distance between G & G_1

So $J_G = \frac{A l^2}{12} + A r_G^2 = A\left[\frac{l^2}{12} + r_1^2\right]$

$$\Rightarrow \boxed{J_G = A\left[\frac{l^2}{12} + r_G^2\right]}$$

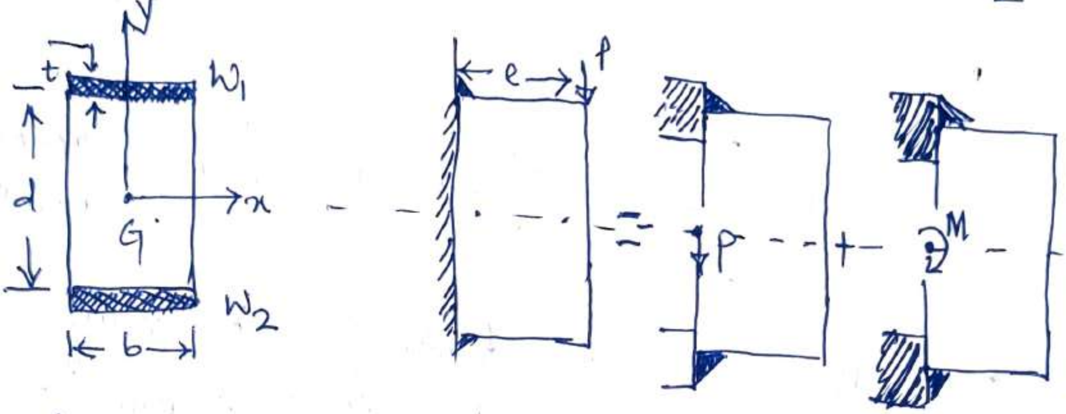
When there are number of welds with polar moment of inertias $J_1, J_2, \ldots\ldots$ then the resultant polar moment of inertia will be

$$J = J_1 + J_2 + \ldots$$

So from $\tau_2 = \frac{Mr}{J}$ we can find out the secondary shear stress.

Welded Joint subjected to Bending Moment

A cantilever beam of rectangular cross-section is welded to a support by means of two fillet welds W_1 & W_2



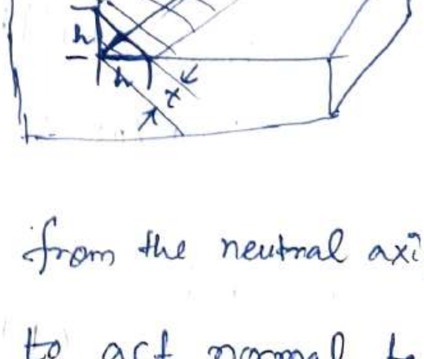
According to applied mechanics the eccentric force 'P' can be replaced by an equal & similarly directed force P acting through the plane of welds along with a couple $(M_b = P \times e)$ as shown in fig.

The force 'P' cause primary shear stress τ_1 through the plane of welds which is given by

$$\tau_1 = \frac{P}{A}$$

where A = throat area of weld

$A = t \times l$.



The moment M_b causes bending stress in the welds

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

I = Moment of inertia of all welds based on the throat area

y = distance of the point of weld from the neutral axis.

The bending stress are assumed to act normal to the throat area.