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CHAPTER

# 3

# Manufacturing Considerations in Machine Design



## 3.1 Introduction

In the previous chapter, we have only discussed about the composition, properties and uses of various materials used in Mechanical Engineering. We shall now discuss in this chapter a few of the manufacturing processes, limits and fits, etc.

## 3.2 Manufacturing Processes

The knowledge of manufacturing processes is of great importance for a design engineer. The following are the various manufacturing processes used in Mechanical Engineering.

**1.** ***Primary shaping processes*.** The processes used for the preliminary shaping of the machine component are known as primary shaping processes. The common operations used for this process are casting, forging, extruding, rolling, drawing, bending, shearing, spinning, powder metal forming, squeezing, etc.



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**2.** ***Machining processes.*** The processes used for giving final shape to the machine component, according to planned dimensions are known as machining processes. The common operations used for this process are turning, planning, shaping, drilling, boring, reaming, sawing, broaching, milling, grinding, hobbing, etc.

**3.** ***Surface finishing processes.*** The processes used to provide a good surface finish for the machine component are known as surface finishing processes. The common operations used for this process are polishing, buffing, honing, lapping, abrasive belt grinding, barrel tumbling, electroplating, superfinishing, sheradizing, etc.

**4.** ***Joining processes.*** The processes used for joining machine components are known as joining processes. The common operations used for this process are welding, riveting, soldering, brazing, screw fastening, pressing, sintering, etc.

**5.** ***Processes effecting change in properties.*** These processes are used to impart certain specific properties to the machine components so as to make them suitable for particular operations or uses. Such processes are heat treatment, hot-working, cold-working and shot peening.

To discuss in detail all these processes is beyond the scope of this book, but a few of them which are important from the subject point of view will be discussed in the following pages.

## 3.3 Casting

It is one of the most important manufacturing process used in Mechanical Engineering. The castings are obtained by remelting of ingots* in a cupola or some other foundry furnace and then pouring this molten metal into metal or sand moulds. The various important casting processes are as follows:

**1.** ***Sand mould casting.*** The casting produced by pouring molten metal in sand mould is called sand mould casting. It is particularly used for parts of larger sizes.

**2.** ***Permanent mould casting.*** The casting produced by pouring molten metal in a metallic mould is called permanent mould casting. It is used for casting aluminium pistons, electric iron parts, cooking utensils, gears, etc. The permanent mould castings have the following advantages:

*1. Shaping the Sand : A wooden pattern cut to the shape of one half of the casting is positioned in an iron box and surrounded by tightly packed moist sand.*

*2. Ready for the Metal : After the wooden patterns have been removed, the two halves of the mould are clamped together. Molten iron is poured into opening called the runner.*

* Most of the metals used in industry are obtained from ores. These ores are subjected to suitable reducing or refining process which gives the metal in a molten form. This molten metal is poured into moulds to give commercial castings, called **ingots.**

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(*a*) It has more favourable fine grained structure.

(*b*) The dimensions may be obtained with close tolerances.

(*c*) The holes up to 6.35 mm diameter may be easily cast with metal cores.

**3.** ***Slush casting.*** It is a special application of permanent metal mould casting. This method is used for production of hollow castings without the use of cores.

**4.** ***Die casting.*** The casting produced by forcing molten metal under pressure into a permanent metal mould (known as die) is called die casting. A die is usually made in two halves and when closed it forms a cavity similar to the casting desired. One half of the die that remains stationary is known as ***cover die*** and the other movable half is called ***ejector die.*** The die casting method is mostly used for castings of non-ferrous metals of comparatively low fusion temperature. This process is cheaper and quicker than permanent or sand mould casting. Most of the automobile parts like fuel pump, carburettor bodies, horn, heaters, wipers, brackets, steering wheels, hubs and crank cases are made with this process. Following are the advantages and disadvantages of die casting :

*Aluminium die casting component*

***Advantages***

(*a*) The production rate is high, ranging up to 700 castings per hour.

(*b*) It gives better surface smoothness.

(*c*) The dimensions may be obtained within tolerances.

(*d*) The die retains its trueness and life for longer periods. For example, the life of a die for zinc base castings is upto one million castings, for copper base alloys upto 75 000 castings and for aluminium base alloys upto 500 000 castings.

*Sand Casting*

*Investment Casting*

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(*e*) It requires less floor area for equivalent production by other casting methods.
(*f*) By die casting, thin and complex shapes can be easily produced.
(*g*) The holes up to 0.8 mm can be cast.

***Disadvantages***

(*a*) The die casting units are costly.
(*b*) Only non-ferrous alloys are casted more economically.
(*c*) It requires special skill for maintenance and operation of a die casting machine.

**5.** ***Centrifugal casting.*** The casting produced by a process in which molten metal is poured and allowed to solidify while the mould is kept revolving, is known as centrifugal casting. The metal thus poured is subjected to centrifugal force due to which it flows in the mould cavities. This results in the production of high density castings with promoted directional solidification. The examples of centrifugal castings are pipes, cylinder liners and sleeves, rolls, bushes, bearings, gears, flywheels, gun barrels, piston rings, brake drums, etc.

## 3.4 Casting Design

An engineer must know how to design the castings so that they can effectively and efficiently render the desired service and can be produced easily and economically. In order to design a casting, the following factors must be taken into consideration :

1. The function to be performed by the casting,
2. Soundness of the casting,
3. Strength of the casting,
4. Ease in its production,
5. Consideration for safety, and
6. Economy in production.

In order to meet these requirements, a design engineer should have a thorough knowledge of production methods including pattern making, moulding, core making, melting and pouring, etc. The best designs will be achieved only when one is able to make a proper selection out of the various available methods. However, a few rules for designing castings are given below to serve as a guide:

1. The sharp corners and frequent use of fillets should be avoided in order to avoid concentration of stresses.
2. All sections in a casting should be designed of uniform thickness, as far as possible. If, however, variation is unavoidable, it should be done gradually.
3. An abrupt change of an extremely thick section into a very thin section should always be avoided.
4. The casting should be designed as simple as possible, but with a good appearance.
5. Large flat surfaces on the casting should be avoided because it is difficult to obtain true surfaces on large castings.
6. In designing a casting, the various allowances must be provided in making a pattern.
7. The ability to withstand contraction stresses of some members of the casting may be improved by providing the curved shapes *e.g.*, the arms of pulleys and wheels.
8. The stiffening members such as webs and ribs used on a casting should be minimum possible in number, as they may give rise to various defects like hot tears and shrinkage, etc.
9. The casting should be designed in such a way that it will require a simpler pattern and its moulding is easier.
10. In order to design cores for casting, due consideration should be given to provide them adequate support in the mould.

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11. The deep and narrow pockets in the casting should invariably be avoided to reduce cleaning costs.
12. The use of metal inserts in the casting should be kept minimum.
13. The markings such as names or numbers, etc., should never be provided on vertical surfaces because they provide a hindrance in the withdrawl of pattern.
14. A tolerance of ± 1.6 mm on small castings (below 300 mm) should be provided. In case more dimensional accuracy is desired, a tolerance of ± 0.8 mm may be provided.

## 3.5 Forging

It is the process of heating a metal to a desired temperature in order to acquire sufficient plasticity, followed by operations like hammering, bending and pressing, etc. to give it a desired shape. The various forging processes are :

1. Smith forging or hand forging
2. Power forging,
3. Machine forging or upset forging, and
4. Drop forging or stamping

The **smith** or **hand forging** is done by means of hand tools and it is usually employed for small jobs. When the forging is done by means of power hammers, it is then known as ***power forging***. It is used for medium size and large articles requiring very heavy blows. The **machine forging** is done by means of forging machines. The **drop forging** is carried out with the help of drop hammers and is particularly suitable for mass production of identical parts. The forging process has the following advantages :

1. It refines the structure of the metal.
2. It renders the metal stronger by setting the direction of grains.
3. It effects considerable saving in time, labour and material as compared to the production of a similar item by cutting from a solid stock and then shaping it.
4. The reasonable degree of accuracy may be obtained by forging.
5. The forgings may be welded.

It may be noted that wrought iron and various types of steels and steel alloys are the common raw material for forging work. Low carbon steels respond better to forging work than the high carbon steels. The common non-ferrous metals and alloys used in forging work are brass, bronze, copper, aluminium and magnesium alloys. The following table shows the temperature ranges for forging some common metals.

**Table 3.1. Temperature ranges for forging.**

| *Material* | *Forging temperature (°C)* | *Material* | *Forging temperature (°C)* |
|---|---|---|---|
| Wrought iron | 900 – 1300 | Stainless steel | 940 – 1180 |
| Mild steel | 750 – 1300 | Aluminium and magnesium alloys | 350 – 500 |
| Medium carbon steel | 750 – 1250 | | |
| High carbon and alloy steel | 800 – 1150 | Copper, brass and bronze | 600 – 950 |

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## 3.6 Forging Design

In designing a forging, the following points should always be considered.

1. The forged components should ultimately be able to achieve a radial flow of grains or fibres.
2. The forgings which are likely to carry flash, such as drop and press forgings, should preferably have the parting line in such a way that the same will divide them in two equal halves.
3. The parting line of a forging should lie, as far as possible, in one plane.
4. Sufficient draft on surfaces should be provided to facilitate easy removal of forgings from dies.
5. The sharp corners should always be avoided in order to prevent concentration of stress and to facilitate ease in forging.
6. The pockets and recesses in forgings should be minimum in order to avoid increased die wear.
7. The ribs should not be high and thin.
8. Too thin sections should be avoided to facilitate easy flow of metal.

## 3.7 Mechanical Working of Metals

The mechanical working of metals is defined as an intentional deformation of metals plastically under the action of externally applied forces.

The mechanical working of metal is described as hot working and cold working depending upon whether the metal is worked above or below the recrystallisation temperature. The metal is subjected to mechanical working for the following purposes :

1. To reduce the original block or ingot into desired shapes,
2. To refine grain size, and
3. To control the direction of flow lines.

## 3.8 Hot Working

The working of metals above the *recrystallisation temperature is called ***hot working.*** This temperature should not be too high to reach the solidus temperature, otherwise the metal will burn and become unsuitable for use. The hot working of metals has the following advantages and disadvantages :

***Advantages***

1. The porosity of the metal is largely eliminated.
2. The grain structure of the metal is refined.
3. The impurities like slag are squeezed into fibres and distributed throughout the metal.
4. The mechanical properties such as toughness, ductility, percentage elongation, percentage reduction in area, and resistance to shock and vibration are improved due to the refinement of grains.

***Disadvantages***

1. It requires expensive tools.
2. It produces poor surface finish, due to the rapid oxidation and scale formation on the metal surface.
3. Due to the poor surface finish, close tolerance cannot be maintained.

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* The temperature at which the new grains are formed in the metal is known as **recrystallisation temperature.**

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## 3.9 Hot Working Processes

The various *hot working processes are described as below :

**1. *Hot rolling.*** The hot rolling process is the most rapid method of converting large sections into desired shapes. It consists of passing the hot ingot through two rolls rotating in opposite directions at the same speed. The space between the rolls is adjusted to conform to the desired thickness of the rolled section. The rolls, thus, squeeze the passing ingot to reduce its cross-section and increase its length. The forming of bars, plates, sheets, rails, angles, I-beam and other structural sections are made by hot rolling.

*Hot Rolling : When steel is heated until it glows bright red, it becomes soft enough to form into elabrate shapes.*

**2. *Hot forging.*** It consists of heating the metal to plastic state and then the pressure is applied to form it into desired shapes and sizes. The pressure applied in this is not continuous as for hot rolling, but intermittent. The pressure may be applied by hand hammers, power hammers or by forging machines.

**3. *Hot spinning.*** It consists of heating the metal to forging temperature and then forming it into the desired shape on a spinning lathe. The parts of circular cross-section which are symmetrical about the axis of rotation, are made by this process.

**4. *Hot extrusion.*** It consists of pressing a metal inside a chamber to force it out by high pressure through an orifice which is shaped to provide the desired form of the finished part. Most commercial metals and their alloys such as steel, copper, aluminium and nickel are directly extruded at elevated temperatures. The rods, tubes, structural shapes, flooring strips and lead covered cables, etc., are the typical products of extrusion.

**5. *Hot drawing or cupping.*** It is mostly used for the production of thick walled seamless tubes and cylinders. It is usually performed in two stages. The first stage consists of drawing a cup out of a hot circular plate with the help of a die and punch. The second stage consists of reheating the drawn cup and drawing it further to the desired length having the required wall thickness. The second drawing operation is performed through a number of dies, which are arranged in a descending order of their diameters, so that the reduction of wall thickness is gradual in various stages.

*Cold Rolled Steel : Many modern products are made from easily shaped sheet metal.*

**6. *Hot piercing.*** This process is used for the manufacture of seamless tubes. In its operation, the heated cylindrical billets of steel are passed between two conical shaped rolls operating in the same direction. A mandrel is provided between these rolls which assist in piercing and controls the size of the hole, as the billet is forced over it.

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* For complete details, please refer to Authors' popular book **'A Text Book of Workshop Technology'.**

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## 3.10 Cold Working

The working of metals below their recrystallisation temperature is known as ***cold working.*** Most of the cold working processes are performed at room temperature. The cold working distorts the grain structure and does not provide an appreciable reduction in size. It requires much higher pressures than hot working. The extent to which a metal can be cold worked depends upon its ductility. The higher the ductility of the metal, the more it can be cold worked. During cold working, severe stresses known as residual stresses are set up. Since the presence of these stresses is undesirable, therefore, a suitable heat treatment may be employed to neutralise the effect of these stresses. The cold working is usually used as finishing operation, following the shaping of the metal by hot working. It also increases tensile strength, yield strength and hardness of steel but lowers its ductility. The increase in hardness due to cold working is called ***work-hardening.***

In general, cold working produces the following effects :

1. The stresses are set up in the metal which remain in the metal, unless they are removed by subsequent heat treatment.
2. A distortion of the grain structure is created.
3. The strength and hardness of the metal are increased with a corresponding loss in ductility.
4. The recrystalline temperature for steel is increased.
5. The surface finish is improved.
6. The close dimensional tolerance can be maintained.

## 3.11 Cold Working Processes

The various cold working processes are discussed below:

**1.** ***Cold rolling.*** It is generally employed for bars of all shapes, rods, sheets and strips, in order to provide a smooth and bright surface finish. It is also used to finish the hot rolled components to close tolerances and improve their toughness and hardness. The hot rolled articles are first immersed in an acid to remove the scale and washed in water, and then dried. This process of cleaning the articles is known as ***pickling.*** These cleaned articles are then passed through rolling mills. The rolling mills are similar to that used in hot rolling.

***Gallium arsenide (GaAs)*** *is now being manufactured as an alternative to silicon for microchips. This combination of elements is a semiconductor like silicon, but is electronically faster and therefore better for microprocessors.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

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**2.** ***Cold forging.*** The cold forging is also called ***swaging.*** During this method of cold working, the metal is allowed to flow in some pre-determined shape according to the design of dies, by a compressive force or impact. It is widely used in forming ductile metals. Following are the three, commonly used cold forging processes :

(*a*) ***Sizing.*** It is the simplest form of cold forging. It is the operation of slightly compressing a forging, casting or steel assembly to obtain close tolerance and a flat surface. The metal is confined only in a vertical direction.

(*b*) ***Cold heading.*** This process is extensively used for making bolts, rivets and other similar headed parts. This is usually done on a cold header machine. Since the cold header is made from unheated material, therefore, the equipment must be able to withstand the high pressures that develop. The rod is fed to the machine where it is cut off and moved into the header die. The operation may be either single or double and upon completion, the part is ejected from the dies.

After making the bolt head, the threads are produced on a thread rolling machine. This is also a cold working process. The process consists of pressing the blank between two rotating rolls which have the thread form cut in their surface.

(*c*) ***Rotary swaging.*** This method is used for reducing the diameter of round bars and tubes by rotating dies which open and close rapidly on the work. The end of rod is tapered or reduced in size by a combination of pressure and impact.

**3.** ***Cold spinning.*** The process of cold spinning is similar to hot spinning except that the metal is worked at room temperature. The process of cold spinning is best suited for aluminium and other soft metals. The commonly used spun articles out of aluminum and its alloys are processing kettles, cooking utensils, liquid containers, and light reflectors, etc.

**4.** ***Cold extrusion.*** The principle of cold extrusion is exactly similar to hot extrusion. The most common cold extrusion process is ***impact extrusion.*** The operation of cold extrusion is performed with the help of a punch and die. The work material is placed in position into a die and struck from top

***Making microchips*** *demands extreme control over chemical components. The layers of conducting and insulating materials that are laid down on the surface of a silicon chip may be only a few atoms thick yet must perform to the highest specifications. Great care has to be taken in their manufacture (right), and each chip is checked by test probes to ensure it performs correctly.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

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by a punch operating at high pressure and speed. The metal flows up along the surface of the punch forming a cup-shaped component. When the punch moves up, compressed air is used to separate the component from the punch. The thickness of the side wall is determined by the amount of clearance between the punch and die. The process of impact extrusion is limited to soft and ductile materials such as lead, tin, aluminium, zinc and some of their alloys. The various items of daily use such as tubes for shaving creams and tooth pastes and such other thin walled products are made by impact extrusion.

**5.** ***Cold drawing.*** It is generally employed for bars, rods, wires, etc. The important cold drawing processes are as follows:

(*a*) ***Bar or rod drawing.*** In bar drawing, the hot drawn bars or rods from the mills are first pickled, washed and coated to prevent oxidation. A draw bench, is employed for cold drawing. One end of the bar is reduced in diameter by the swaging operation to permit it to enter a drawing die. This end of bar is inserted through the die and gripped by the jaws of the carriage fastened to the chain of the draw bench. The length of bars which can be drawn is limited by the maximum travel of the carriage, which may be from 15 metres to 30 metres. A high surface finish and dimensional accuracy is obtained by cold drawing. The products may be used directly without requiring any machining.

(*b*) ***Wire drawing.*** In wire drawing, the rolled bars from the mills are first pickled, washed and coated to prevent oxidation. They are then passed through several dies of decreasing diameter to provide the desired reduction in size. The dies are usually made of carbide materials.

(*c*) ***Tube drawing***. The tube drawing is similar to bar drawing and in most cases it is accomplished with the use of a draw bench.

**6.** ***Cold bending.*** The bars, wires, tubes, structural shapes and sheet metal may be bent to many shapes in cold condition through dies. A little consideration will show that when the metal is bend beyond the elastic limit, the inside of the bend will be under compression while the outside will be under tension. The stretching of the metal on the outside makes the stock thinner. Usually, a flat strip of metal is bend by ***roll forming***. The materials commonly used for roll forming are carbon steel, stainless steel, bronze, copper, brass, zinc and aluminium. Some of its products are metal windows, screen frame parts, bicycle wheel rims, trolley rails, etc. Most of the tubing is now-a-days are roll formed in cold conditions and then welded by resistance welding.

**7.** ***Cold peening.*** This process is used to improve the fatigue resistance of the metal by setting up compressive stresses in its surface. This is done by blasting or hurling a rain of small shot at high velocity against the surface to be peened. The shot peening is done by air blast or by some mechanical means. As the shot strikes, small indentations are produced, causing a slight plastic flow of the surface metal to a depth of a few hundreds of a centimetre. This stretching of the outer fibres is resisted by those underneath, which tend to return them to their original length, thus producing an outer layer having a compressive stress while those below are in tension. In addition, the surface is slightly hardened and strengthened by the cold working operation.

## 3.12 Interchangeability

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

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large number of parts for an assembly and results in a considerable saving in the cost of production. In order to control the size of finished part, with due allowance for error, for interchangeable parts is called ***limit system.***

It may be noted that when an assembly is made of two parts, the part which enters into the other, is known as ***enveloped surface*** (or **shaft** for cylindrical part) and the other in which one enters is called ***enveloping surface*** (or **hole** for cylindrical part).

**Notes: 1.** The term ***shaft*** refers not only to the diameter of a circular shaft, but it is also used to designate any external dimension of a part.

**2.** The term ***hole*** refers not only to the diameter of a circular hole, but it is also used to designate any internal dimension of a part.

## 3.13 Important Terms used in Limit System

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

**1. *Nominal size.*** It is the size of a part specified in the drawing as a matter of convenience.

**2. *Basic size.*** It is the size of a part to which all limits of variation (*i.e.* tolerances) are applied to arrive at final dimensioning of the mating parts. The nominal or basic size of a part is often the same.

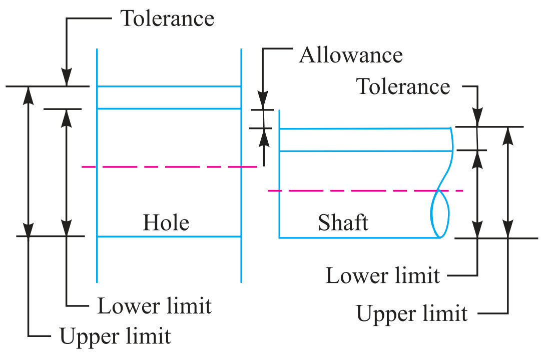


**Fig. 3.1.** Limits of sizes.

**3. *Actual size.*** It is the actual measured dimension of the part. The difference between the basic size and the actual size should not exceed a certain limit, otherwise it will interfere with the interchangeability of the mating parts.

**4. *Limits of sizes.*** There are two extreme permissible sizes for a dimension of the part as shown in Fig. 3.1. The largest permissible size for a dimension of the part is called ***upper*** or ***high*** or ***maximum limit,*** whereas the smallest size of the part is known as ***lower*** or ***minimum limit.***

**5. *Allowance.*** It is the difference between the basic dimensions of the mating parts. The allowance may be ***positive*** or ***negative***. When the shaft size is less than the hole size, then the allowance is ***positive*** and when the shaft size is greater than the hole size, then the allowance is ***negative***.

**6. *Tolerance.*** It is the difference between the upper limit and lower limit of a dimension. In other words, it is the maximum permissible variation in a dimension. The tolerance may be ***unilateral*** or ***bilateral***. When all the tolerance is allowed on one side of the nominal size, *e.g.* $20^{+0.000}_{-0.004}$ , then it is said to be ***unilateral system of tolerance***. The unilateral system is mostly used in industries as it permits changing the tolerance value while still retaining the same allowance or type of fit.


(*a*) Unilateral tolerance. (*b*) Bilateral tolerance.

**Fig. 3.2.** Method of assigning tolerances.

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

The method of assigning unilateral and bilateral tolerance is shown in Fig. 3.2 (*a*) and (*b*) respectively.

**7.** ***Tolerance zone.*** It is the zone between the maximum and minimum limit size, as shown in Fig. 3.3.

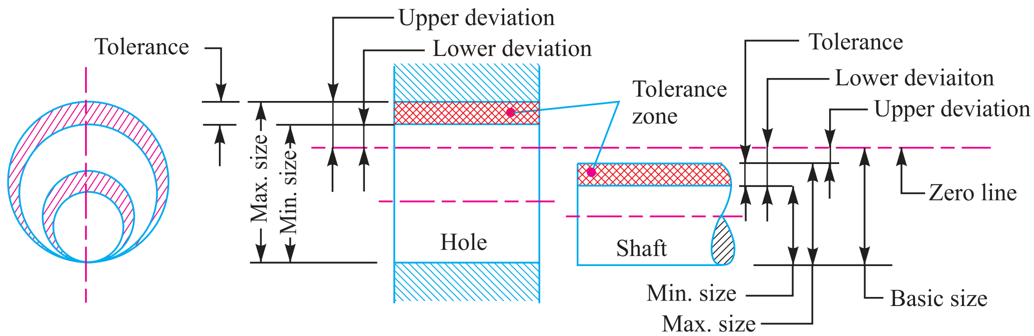


**Fig. 3.3.** Tolerance zone.

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

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

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

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

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

**13.** ***Fundamental deviation.*** It is one of the two deviations which is conventionally chosen to define the position of the tolerance zone in relation to zero line, as shown in Fig. 3.4.

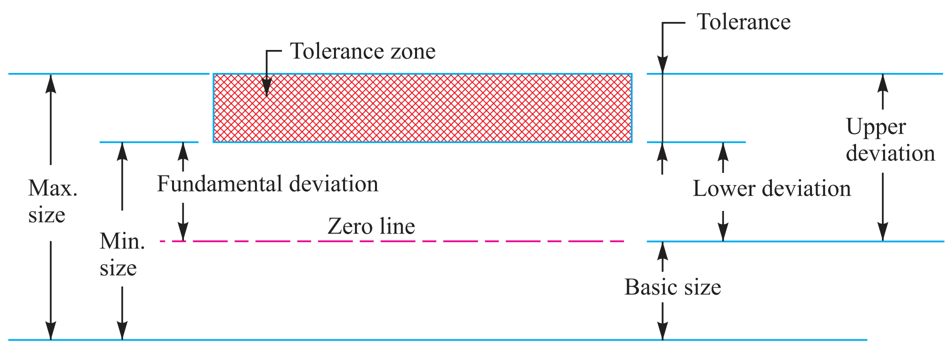


**Fig. 3.4.** Fundamental deviation.

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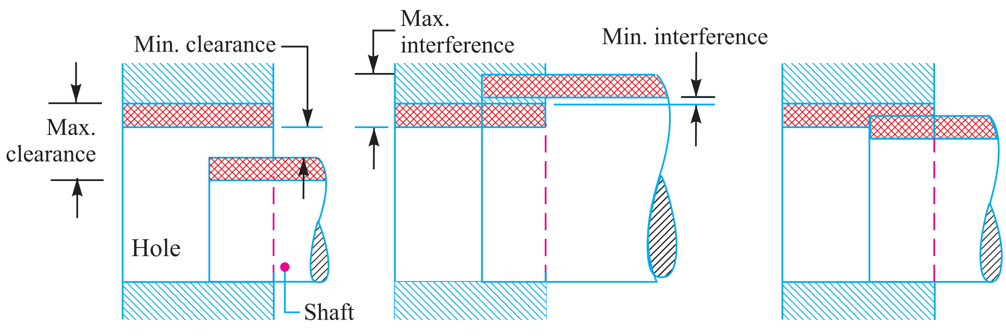
## 3.14 Fits

The degree of tightness or looseness between the two mating parts is known as a *fit* of the parts. The nature of fit is characterised by the presence and size of clearance and interference.

The ***clearance*** is the amount by which the actual size of the shaft is less than the actual size of the mating hole in an assembly as shown in Fig. 3.5 (*a*). In other words, the clearance is the difference between the sizes of the hole and the shaft before assembly. The difference must be ***positive.***




(*a*) Clearance fit. (*b*) Interference fit. (*c*) Transition fit.

**Fig. 3.5.** Types of fits.

The ***interference*** is the amount by which the actual size of a shaft is larger than the actual finished size of the mating hole in an assembly as shown in Fig. 3.5 (*b*). In other words, the interference is the arithmetical difference between the sizes of the hole and the shaft, before assembly. The difference must be ***negative.***

## 3.15 Types of Fits

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

**1. *Clearance fit.*** In this type of fit, the size limits for mating parts are so selected that clearance between them always occur, as shown in Fig. 3.5 (*a*). It may be noted that in a clearance fit, the tolerance zone of the hole is entirely above the tolerance zone of the shaft.

In a clearance fit, the difference between the minimum size of the hole and the maximum size of the shaft is known as ***minimum clearance*** whereas the difference between the maximum size of the hole and minimum size of the shaft is called ***maximum clearance*** as shown in Fig. 3.5 (*a*).

***A Jet Engine :*** *In a jet engine, fuel is mixed with air, compressed, burnt, and exhausted in one smooth, continuous process. There are no pistons shuttling back and forth to slow it down.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

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The clearance fits may be slide fit, easy sliding fit, running fit, slack running fit and loose running fit.

**2.** ***Interference fit.*** In this type of fit, the size limits for the mating parts are so selected that interference between them always occur, as shown in Fig. 3.5 (*b*). It may be noted that in an interference fit, the tolerance zone of the hole is entirely below the tolerance zone of the shaft.

In an interference fit, the difference between the maximum size of the hole and the minimum size of the shaft is known as ***minimum interference,*** whereas the difference between the minimum size of the hole and the maximum size of the shaft is called ***maximum interference,*** as shown in Fig. 3.5 (*b*).

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

**3.** ***Transition fit.*** In this type of fit, the size limits for the mating parts are so selected that either a clearance or interference may occur depending upon the actual size of the mating parts, as shown in Fig. 3.5 (*c*). It may be noted that in a transition fit, the tolerance zones of hole and shaft overlap.

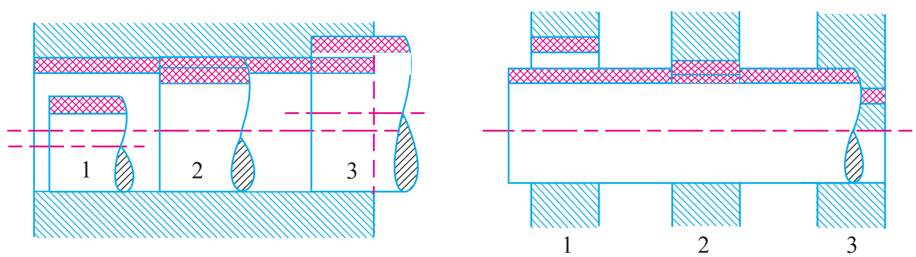
The transition fits may be force fit, tight fit and push fit.

## 3.16 Basis of Limit System

The following are two bases of limit system:

**1.** ***Hole basis system.*** When the hole is kept as a constant member (*i.e.* when the lower deviation of the hole is zero) and different fits are obtained by varying the shaft size, as shown in Fig. 3.6 (*a*), then the limit system is said to be on a hole basis.

**2.** ***Shaft basis system.*** When the shaft is kept as a constant member (*i.e.* when the upper deviation of the shaft is zero) and different fits are obtained by varying the hole size, as shown in Fig. 3.6 (*b*), then the limit system is said to be on a shaft basis.

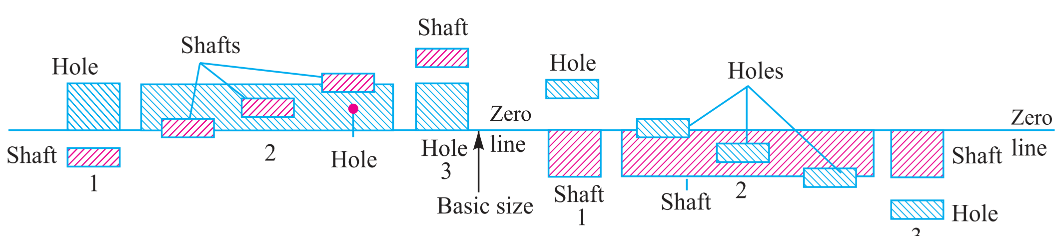



1. Clearance fit. 2. Transition fit. 3. Interference fit.

(*a*) Hole basis system. (*b*) Shaft basis system.

**Fig. 3.6.** Bases of limit system.

The hole basis and shaft basis system may also be shown as in Fig. 3.7, with respect to the zero line.




1. Clearance fit. 2. Transition fit. 3. Interference fit.

(*a*) Hole basis system. (*b*) Shaft basis system.

**Fig. 3.7.** Bases of limit system.

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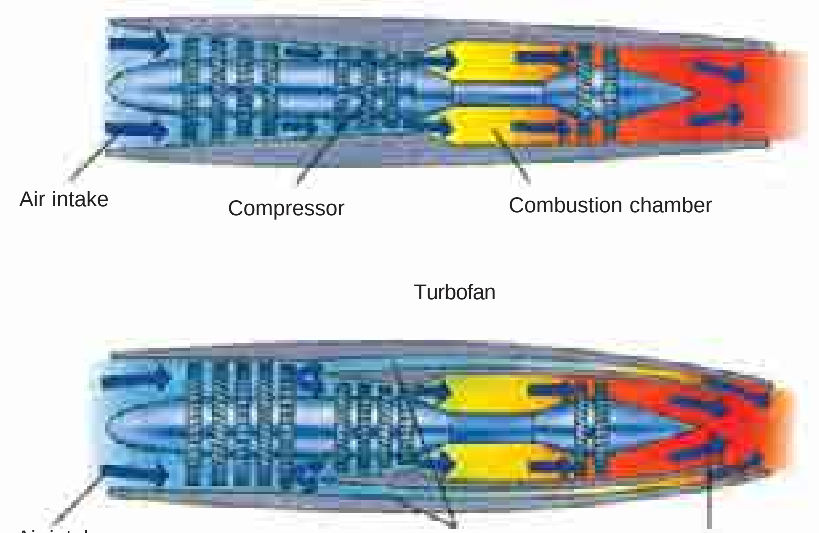
It may be noted that from the manufacturing point of view, a hole basis system is always preferred. This is because the holes are usually produced and finished by standard tooling like drill, reamers, etc., whose size is not adjustable easily. On the other hand, the size of the shaft (which is to go into the hole) can be easily adjusted and is obtained by turning or grinding operations.




Turbofan engines are quieter and more efficient than simple turbojet engines. Turbofans drive air around the combustion engine as well as through it.

Note : This picture is given as additional information and is not a direct example of the current chapter.

## 3.17 Indian Standard System of Limits and Fits

According to Indian standard [IS : 919 (Part I)-1993], the system of limits and fits comprises 18 grades of fundamental tolerances *i.e.* grades of accuracy of manufacture and 25 types of fundamental deviations indicated by letter symbols for both holes and shafts (capital letter *A* to *ZC* for holes and small letters *a* to *zc* for shafts) in diameter steps ranging from 1 to 500 mm. A unilateral hole basis system is recommended but if necessary a unilateral or bilateral shaft basis system may also be used. The 18 tolerance grades are designated as IT 01, IT 0 and IT 1 to IT 16. These are called ***standard tolerances***. The standard tolerances for grades IT 5 to IT 7 are determined in terms of standard tolerance unit (*i*) in microns, where

$i$ (microns) $= 0.45 \sqrt[3]{D} + 0.001 D$, where $D$ is the size or geometric mean diameter in mm.

The following table shows the relative magnitude for grades between IT 5 and IT 16.

**Table 3.2. Relative magnitude of tolerance grades.**

| *Tolerance grade* | IT 5 | IT 6 | IT 7 | IT 8 | IT 9 | IT 10 | IT 11 | IT 12 | IT 13 | IT 14 | IT 15 | IT 16 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| *Magnitude* | 7 *i* | 10 *i* | 16 *i* | 25 *i* | 40 *i* | 64 *i* | 100 *i* | 160 *i* | 250 *i* | 400 *i* | 640 *i* | 1000 *i* |

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The values of standard tolerances corresponding to grades IT 01, IT 0 and IT 1 are as given below:

For IT 01, $i$ (microns) $= 0.3 + 0.008\, D$,

For IT 0, $i$ (microns) $= 0.5 + 0.012\, D$, and

For IT 1, $i$ (microns) $= 0.8 + 0.020\, D$,

where $D$ is the size or geometric mean diameter in mm.

The tolerance values of grades IT 2 to IT 4 are scaled approximately geometrically between IT 1 and IT 5. The fundamental tolerances of grades IT 01, IT 0 and IT 1 to IT 16 for diameter steps ranging from 1 to 500 mm are given in Table 3.3. The manufacturing processes capable of producing the particular IT grades of work are shown in Table 3.4.

The alphabetical representation of fundamental deviations for basic shaft and basic hole system is shown in Fig. 3.8.

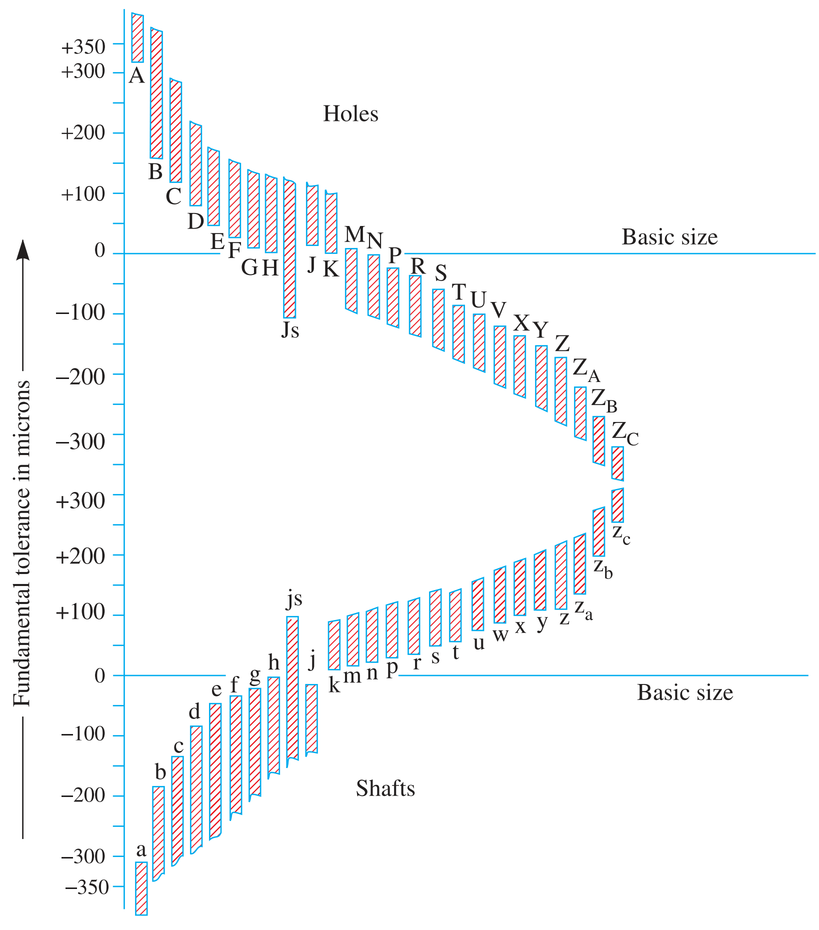


**Fig. 3.8.** Fundamental deviations for shafts and holes.

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**Table 3.3. Fundamental tolerances of grades IT01, IT0 and IT1 to IT16, according to IS : 919 (Part I) – 1993.**

| *Basic size (Diameter steps) in mm* | | *Standard tolerance grades, in micron (1 micron = 0.001 mm)* | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | IT01 | IT0 | IT1 | IT2 | IT3 | IT4 | IT5 | IT6 | IT7 | IT8 | IT9 | IT10 | IT11 | IT12 | IT13 | IT14 | IT15 | IT16 |
| Over<br>To and inc. | 1<br>3 | 0.3 | 0.5 | 0.8 | 1.2 | 2 | 3 | 4 | 6 | 10 | 14 | 25 | 40 | 60 | 100 | 140 | 250 | 400 | 600 |
| Over<br>To and inc. | 3<br>6 | 0.4 | 0.6 | 1 | 1.5 | 2.5 | 4 | 5 | 8 | 12 | 18 | 30 | 48 | 75 | 120 | 180 | 300 | 480 | 750 |
| Over<br>To and inc. | 6<br>10 | 0.4 | 0.6 | 1 | 1.5 | 2.5 | 4 | 6 | 9 | 15 | 22 | 36 | 58 | 90 | 150 | 220 | 360 | 580 | 900 |
| Over<br>To and inc. | 10<br>18 | 0.5 | 0.8 | 1.2 | 2 | 3 | 5 | 8 | 11 | 18 | 27 | 43 | 70 | 110 | 180 | 270 | 430 | 700 | 1100 |
| Over<br>To and inc. | 18<br>30 | 0.6 | 1 | 1.5 | 2.5 | 4 | 6 | 9 | 13 | 21 | 33 | 52 | 84 | 130 | 210 | 330 | 520 | 840 | 1300 |
| Over<br>To and inc. | 30<br>50 | 0.6 | 1 | 1.5 | 2.5 | 4 | 7 | 11 | 16 | 25 | 39 | 62 | 100 | 160 | 250 | 390 | 620 | 1000 | 1600 |
| Over<br>To and inc. | 50<br>80 | 0.8 | 1.2 | 2 | 3 | 5 | 8 | 13 | 19 | 30 | 46 | 74 | 120 | 190 | 300 | 460 | 740 | 1200 | 1900 |
| Over<br>To and inc. | 80<br>120 | 1 | 1.5 | 2.5 | 4 | 6 | 10 | 15 | 22 | 35 | 54 | 87 | 140 | 220 | 350 | 540 | 870 | 1400 | 2200 |
| Over<br>To and inc. | 120<br>180 | 1.2 | 2 | 3.5 | 5 | 8 | 12 | 18 | 25 | 40 | 63 | 100 | 160 | 250 | 400 | 630 | 1000 | 1600 | 2500 |
| Over<br>To and inc. | 180<br>250 | 2 | 3 | 4.5 | 7 | 10 | 14 | 20 | 29 | 46 | 72 | 115 | 185 | 290 | 460 | 720 | 1150 | 1850 | 2900 |
| Over<br>To and inc. | 250<br>315 | 2.5 | 4 | 6 | 8 | 12 | 16 | 23 | 32 | 52 | 81 | 130 | 210 | 320 | 520 | 810 | 1300 | 2100 | 3200 |
| Over<br>To and inc. | 315<br>400 | 3 | 5 | 7 | 9 | 13 | 18 | 25 | 36 | 57 | 89 | 140 | 230 | 360 | 570 | 890 | 1400 | 2300 | 3800 |
| Over<br>To and inc. | 400<br>500 | 4 | 6 | 8 | 10 | 15 | 20 | 27 | 40 | 63 | 97 | 155 | 250 | 400 | 630 | 970 | 1550 | 2500 | 4000 |

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**Table 3.4. Manufacturing processes and IT grades produced.**

| S.No. | Manufacturing process | IT grade produced | S.No. | Manufacturing process | IT grade produced |
|---|---|---|---|---|---|
| 1. | Lapping | 4 and 5 | 9. | Extrusion | 8 to 10 |
| 2. | Honing | 4 and 5 | 10 | Boring | 8 to 13 |
| 3. | Cylindrical grinding | 5 to 7 | 11. | Milling | 10 to 13 |
| 4. | Surface grinding | 5 to 8 | 12. | Planing and shaping | 10 to 13 |
| 5. | Broaching | 5 to 8 | 13. | Drilling | 10 to 13 |
| 6. | Reaming | 6 to 10 | 14. | Die casting | 12 to 14 |
| 7. | Turning | 7 to 13 | 15. | Sand casting | 14 to 16 |
| 8. | Hot rolling | 8 to 10 | 16. | Forging | 14 to 16 |

For hole, $H$ stands for a dimension whose lower deviation refers to the basic size. The hole $H$ for which the lower deviation is zero is called a ***basic hole.*** Similarly, for shafts, $h$ stands for a dimension whose upper deviation refers to the basic size. The shaft $h$ for which the upper deviation is zero is called a ***basic shaft.***

*This view along the deck of a liquefied natural gas (LNG) carrier shows the tops of its large, insulated steel tanks. The tanks contain liquefied gas at-162°C.*

A fit is designated by its basic size followed by symbols representing the limits of each of its two components, the hole being quoted first. For example, 100 $H6/g5$ means basic size is 100 mm and the tolerance grade for the hole is 6 and for the shaft is 5. Some of the fits commonly used in engineering practice, for holes and shafts are shown in Tables 3.5 and 3.6 respectively according to IS : 2709 – 1982 (Reaffirmed 1993).

**Table 3.5. Commonly used fits for holes according to IS : 2709 – 1982 (Reaffirmed 1993).**

| *Type of fit* | *Class of shaft* | *With holes* | | | | *Remarks and uses* |
|---|---|---|---|---|---|---|
| | | *H6* | *H7* | *H8* | *H11* | |
| Clearance fit | *a* | — | — | — | *a*11 | Large clearance fit and widely used. |
| | *b* | — | — | — | *b*11 | |
| | *c* | — | *c*8 | **c* 9 | *c* 11 | Slack running fit. |
| | *d* | — | *d*8 | **d* 8<br>*d* 9, *d*10 | *d* 11 | Loose running fit—used for plummer block bearings and loose pulleys. |
| | *e* | *e*7 | *e*8 | **e* 8-*e* 9 | — | Easy running fit—used for properly lubricated bearings requiring appreciable clearance. In the finer grades, it may be used on large electric motor and turbogenerator bearings according to the working condition. |
| | *f* | **f* 6 | *f* 7 | **f* 8 | — | Normal running fit—widely used for grease lubricated or oil lubricated bearings where no substantial temperature differences are encountered—Typical applications are gear box shaft bearings and the bearings of small electric motors, pumps, etc. |
| | *g* | **g* 5 | **g* 6 | *g* 7 | — | Close running fit or sliding fit—Also fine spigot and location fit—used for bearings for accurate link work and for piston and slide valves. |
| | *h* | **h* 5 | **h* 6 | **h* 7–*h* 8 | **h*11 | Precision sliding fit. Also fine spigot and location fit—widely used for non-running parts. |
| Transition fit | *j* | **j*5 | **j*6 | **j*7 | — | Push fit for very accurate location with easy assembly and dismantling—Typical applications are coupling, spigots and recesses, gear rings clamped to steel hubs, etc. |
| | *k* | **k* 5 | **k* 6 | *k* 7 | — | True transition fit (light keying fit)—used for keyed shaft, non-running locked pins, etc. |
| | *m* | **m* 5 | **m* 6 | *m* 7 | — | Medium keying fit. |
| | *n* | *n* 5 | **n* 6 | *n*7 | — | Heavy keying fit—used for tight assembly of mating parts. |

* Second preference fits.

| *Type of fit* | *Class of shaft* | *With holes* | | | | *Remarks and uses* |
|---|---|---|---|---|---|---|
| | | *H6* | *H7* | *H8* | *H11* | |
| Interference fit | *p* | *p*5 | **p*6 | — | — | Light press fit with easy dismantling for non-ferrous parts. Standard press fit with easy dismantling for ferrous and non-ferrous parts assembly. |
| | *r* | *r*5 | **r*6 | — | — | Medium drive fit with easy dismantling for ferrous parts assembly. Light drive fit with easy dismantling for non-ferrous parts assembly. |
| | *s* | *s*5 | **s*6 | *s*7 | — | Heavy drive fit on ferrous parts for permanent or semi-permanent assembly. Standard press fit for non-ferrous parts. |
| | *t* | *t*5 | *t*6 | **t*7 | — | Force fit on ferrous parts for permanent assembly. |
| | *u* | *u*5 | *u*6 | **u*7 | — | Heavy force fit or shrink fit. |
| | *v*, *x* | — | — | — | — | Very large interference fits — not recommended for use |
| | *y*, *z* | — | — | — | — | |

## Table 3.6. Commonly used fits for shafts according to IS : 2709 – 1982 (Reaffirmed 1993).

| *Type of fit* | *Class of hole* | *With shafts* | | | | | | *Remarks and uses* |
|---|---|---|---|---|---|---|---|---|
| | | **h*5 | *h*6 | *h*7 | **h*8 | *h*9 | *h*11 | |
| Clearance fit | *A* | — | — | — | — | — | *A*11 | Large clearance fit and widely used. |
| | *B* | — | — | — | — | — | *B*11 | |
| | *C* | — | — | — | — | — | *C*11 | Slack running fit. |
| | *D* | — | **D*9 | — | *D*10 | *D*10 | **D*11 | Loose running fit. |
| | *E* | — | **E*8 | — | *E*8* | *E*9 | — | Easy running fit. |
| | *F* | — | **F*7 | — | *F*8 | **F*8 | — | Normal running fit. |
| | *G* | **G*6 | *G*7 | — | — | — | — | Close running fit or sliding fit, also spigot and location fit. |
| | *H* | **H*6 | *H*7 | *H*8 | *H*8 | *H*8, *H*9 | *H*11 | Precision sliding fit. Also fine spigot and location fit. |
| | *Js* | **Js*6 | *Js*7 | **Js*8 | — | — | — | Push fit for very accurate location with easy assembly and disassembly. |

* Second preference fits.

| *Type of fit* | *Class of hole* | *With shafts* | | | | | | *Remarks and uses* |
|---|---|---|---|---|---|---|---|---|
| | | *h5 | h6 | h7 | *h8 | h9 | h11 | |
| Transition fit | *K* | **K*6 | *K*7 | **K*8 | — | — | — | Light keying fit (true transition) for keyed shafts, non-running locked pins, etc. |
| | *M* | **M*6 | **M*7 | **M*8 | — | — | — | Medium keying fit. |
| | *N* | **N*6 | *N*7 | **N*8 | — | — | — | Heavy keying fit (for tight assembly of mating surfaces). |
| Interference fit | *P* | **P*6 | *P*7 | — | — | — | — | Light press fit with easy dismantling for non-ferrous parts. Standard press fit with easy dismantling for ferrous and non-ferrous parts assembly. |
| | *R* | **R*6 | *R*7 | — | — | — | — | Medium drive fit with easy dismantling for ferrous parts assembly. Light drive fit with easy dismantling for non-ferrous parts assembly. |
| | *S* | **S*6 | *S*7 | — | — | — | — | Heavy drive fit for ferrous parts permanent or semi- permanent assembly, standard press fit for non-ferrous parts. |
| | *T* | **T*6 | *T*7 | — | — | — | — | Force fit on ferrous parts for permanent assembly. |

## 3.18 Calculation of Fundamental Deviation for Shafts

We have already discussed that for holes, the upper deviation is denoted by *ES* and the lower deviation by *EI*. Similarly for shafts, the upper deviation is represented by *es* and the lower deviation by *ei*. According to Indian standards, for each letter symbol, the magnitude and sign for one of the two deviations (*i.e.* either upper or lower deviation), which is known as fundamental deviation, have been determined by means of formulae given in Table 3.7. The other deviation may be calculated by using the absolute value of the standard tolerance (*IT*) from the following relation:

$$ei = es - IT \quad \text{or} \quad es = ei + IT$$

It may be noted for shafts *a* to *h*, the upper deviations (*es*) are considered whereas for shafts *j* to *Zc*, the lower deviation (*ei*) is to be considered.

*Computer simulation of stresses on a jet engine blades.*

* Second preference fits.

The fundamental deviation for Indian standard shafts for diameter steps from 1 to 200 mm may be taken directly from Table 3.10 (page 76).

**Table 3.7. Formulae for fundamental shaft deviations.**

| *Upper deviation (es)* | | *Lower deviation (ei)* | |
|---|---|---|---|
| *Shaft designation* | *In microns (for D in mm)* | *Shaft designation* | *In microns (for D in mm)* |
| *a* | = – (265 + 1.3 $D$) | $J$ 5 to $j$ 8 | No formula |
| | for $D \leq 120$ | $k$ 4 to $k$ 7 | = + 0.6 $\sqrt[3]{D}$ |
| | = – 3.5 $D$ | | |
| | for $D > 120$ | $k$ for grades | = 0 |
| | | ≤ 3 and ≤ 8 | |
| *b* | = – (140 + 0.85 $D$) | *m* | = + ($IT$ 7 – $IT$ 6) |
| | for $D \leq 160$ | | |
| | = – 1.8 $D$ | *n* | = + 5 $(D)^{0.34}$ |
| | for $D > 160$ | *p* | = + $IT$ 7 + 0 to 5 |
| *c* | = – 52 $(D)^{0.2}$ | *r* | = Geometric mean of values of *ei* |
| | for $D \leq 40$ | | for shaft *p* and *s* |
| | = – (95 + 0.8 $D$) | *s* | = + ($IT$ 8 + 1 to 4) for $D \leq 50$ |
| | for $D > 40$ | | = + ($IT$ 7 + 0.4 $D$) for $D > 50$ |
| *d* | = – 16 $(D)^{0.44}$ | *t* | = + ($IT$ 7 + 0.63 $D$) |
| *e* | = – 11 $(D)^{0.41}$ | *u* | = + ($IT$ 7 + $D$) |
| *f* | = – 5.5 $(D)^{0.41}$ | *v* | = + ($IT$ 7 + 1.25 $D$) |
| | | *x* | = + ($IT$ 7 + 1.6 $D$) |
| *g* | = – 2.5 $(D)^{0.34}$ | *y* | = + ($IT$ 7 + 2 $D$) |
| | | *z* | = + ($IT$ 7 + 2.5 $D$) |
| *h* | = 0 | *za* | = + ($IT$ 8 + 3.15 $D$) |
| | | *zb* | = + ($IT$ 9 + 4 $D$) |
| | | *zc* | = + ($IT$ 10 + 5 $D$) |

For *js*, the two deviations are equal to ± $IT/2$.

## 3.19 Calculation of Fundamental Deviation for Holes

The fundamental deviation for holes for those of the corresponding shafts, are derived by using the rule as given in Table 3.8.

### Table 3.8. Rules for fundamental deviation for holes.

<table>
<tr><td colspan="3">All deviation except those below</td><td>General rule<br>Hole limits are identical with the shaft limits of the same symbol (letter and grade) but disposed on the other side of the zero line.<br>EI = Upper deviation es of the shaft of the same letter symbol but of opposite sign.</td></tr>
<tr><td rowspan="2">For sizes above 3 mm</td><td>N</td><td>9 and coarser grades</td><td>ES = 0</td></tr>
<tr><td>J, K, M and N<br><br>P to ZC</td><td>Upto grade 8 inclusive<br><br>upto grade 7 inclusive</td><td>Special rule<br>ES = Lower deviation ei of the shaft of the same letter symbol but one grade finer and of opposite sign increased by the difference between the tolerances of the two grades in question.</td></tr>
</table>

The fundamental deviation for Indian standard holes for diameter steps from 1 to 200 mm may be taken directly from the following table.

### Table 3.9. Indian standard 'H' Hole
### Limits for H5 to H13 over the range 1 to 200 mm as per IS : 919 (Part II) -1993.

| *Diameter steps in mm* | | *Deviations in micron (1 micron = 0.001 mm)* | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|
| | | H5 | H6 | H7 | H8 | H9 | H10 | H11 | H12 | H13 | H5 – H13 |
| Over | To | High + | High + | High + | High + | High + | High + | High + | High + | High + | Low |
| 1 | 3 | 5 | 7 | 9 | 14 | 25 | 40 | 60 | 90 | 140 | 0 |
| 3 | 6 | 5 | 8 | 12 | 18 | 30 | 48 | 75 | 120 | 180 | 0 |
| 6 | 10 | 6 | 9 | 15 | 22 | 36 | 58 | 90 | 150 | 220 | 0 |
| 10<br>14 | 14<br>18 | 8 | 11 | 18 | 27 | 43 | 70 | 110 | 180 | 270 | 0 |
| 18<br>24 | 24<br>30 | 9 | 13 | 21 | 33 | 52 | 84 | 130 | 210 | 330 | 0 |
| 30<br>40 | 40<br>50 | 11 | 16 | 25 | 39 | 62 | 100 | 160 | 250 | 460 | 0 |
| 50<br>65 | 65<br>80 | 13 | 19 | 30 | 46 | 74 | 120 | 190 | 300 | 390 | 0 |
| 80<br>100 | 100<br>120 | 15 | 22 | 35 | 54 | 87 | 140 | 220 | 350 | 540 | 0 |
| 120<br>140 | 140<br>160 | 18 | 25 | 40 | 63 | 100 | 160 | 250 | 400 | 630 | 0 |
| 160<br>180 | 180<br>200 | 20 | 29 | 46 | 72 | 115 | 185 | 290 | 460 | 720 | 0 |

**Table 3.10. Indian standard shafts for common use as per IS : 919 (Part II)–1993.**

| *Shaft* | *Limit* | *Values of deviations in microns for diameter steps 1 to 200 mm (1 micron = 0.001 mm)* | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 1 to 3 | 3 to 6 | 6 to 10 | 10 to 14 | 14 to 18 | 18 to 24 | 24 to 30 | 30 to 40 | 40 to 50 | 50 to 65 | 65 to 80 | 80 to 100 | 100 to 120 | 120 to 140 | 140 to 160 | 160 to 180 | 180 to 200 |
| *s*6 | High+ | 22 | 37 | 32 | 39 | 39 | 48 | 48 | 59 | 59 | 72 | 78 | 93 | 101 | 117 | 125 | 133 | 151 |
| *s*7 | High+ | 24 | 31 | 38 | 46 | 46 | 56 | 66 | 68 | 68 | 83 | 89 | 106 | 114 | 132 | 140 | 148 | 168 |
| *s*6 & *s*7 | Low+ | 15 | 19 | 23 | 28 | 28 | 35 | 35 | 43 | 43 | 53 | 59 | 71 | 79 | 92 | 100 | 108 | 122 |
| *p*9 | High+ | 16 | 20 | 24 | 29 | 29 | 35 | 35 | 42 | 42 | 51 | 51 | 59 | 59 | 68 | 68 | 68 | 79 |
| | Low+ | 9 | 12 | 15 | 18 | 18 | 22 | 22 | 26 | 26 | 32 | 32 | 37 | 37 | 43 | 43 | 43 | 50 |
| *k*6 | High+ | — | — | 10 | 12 | 12 | 15 | 15 | 18 | 18 | 21 | 21 | 25 | 25 | 28 | 28 | 28 | 33 |
| *k*7 | High+ | — | — | 16 | 19 | 19 | 23 | 23 | 27 | 27 | 32 | 32 | 38 | 38 | 43 | 43 | 43 | 50 |
| *k*6 & *k*7 | Low+ | — | — | 1 | 1 | 1 | 2 | 2 | 2 | 2 | 2 | 2 | 3 | 3 | 3 | 3 | 3 | 4 |
| | High+ | 7 | 9 | 10 | 12 | 12 | 13 | 13 | 15 | 15 | 18 | 18 | 20 | 20 | 22 | 22 | 22 | 25 |
| | Low– | 2 | 3 | 5 | 6 | 6 | 8 | 8 | 10 | 10 | 12 | 12 | 15 | 15 | 18 | 18 | 18 | 21 |
| *h*6 | High– | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |

| Shaft | Limit | Values of deviations in microns for diameter steps 1 to 200 mm (1 micron = 0.001 mm) | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | 1 to 3 | 3 to 6 | 6 to 10 | 10 to 14 | 14 to 18 | 18 to 24 | 24 to 30 | 30 to 40 | 40 to 50 | 50 to 65 | 65 to 80 | 80 to 100 | 100 to 120 | 120 to 140 | 140 to 160 | 160 to 180 | 180 to 200 |
| *h*7 | High– | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| *h*6 | Low– | 7 | 8 | 9 | 11 | 11 | 13 | 13 | 16 | 16 | 19 | 19 | 22 | 22 | 25 | 25 | 26 | 29 |
| & | | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & |
| *h*7 | | 9 | 12 | 15 | 18 | 18 | 21 | 21 | 25 | 25 | 30 | 30 | 35 | 35 | 40 | 40 | 40 | 46 |
| *g* 6 | High– | 3 | 4 | 5 | 6 | 6 | 7 | 7 | 9 | 9 | 10 | 10 | 12 | 12 | 14 | 14 | 14 | 15 |
| | Low– | 10 | 12 | 14 | 17 | 17 | 20 | 20 | 25 | 25 | 29 | 29 | 34 | 34 | 39 | 39 | 39 | 44 |
| *f* 7 | High– | 7 | 10 | 13 | 16 | 16 | 20 | 20 | 25 | 25 | 30 | 30 | 36 | 36 | 43 | 43 | 43 | 50 |
| *f* 8 | High– | 7 | 10 | 13 | 16 | 16 | 20 | 20 | 25 | 25 | 30 | 30 | 36 | 36 | 43 | 43 | 43 | 50 |
| *f* 7 | Low– | 16 | 22 | 28 | 34 | 34 | 41 | 41 | 50 | 50 | 60 | 60 | 71 | 71 | 83 | 83 | 83 | 96 |
| & | | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & |
| *f* 8 | | 21 | 28 | 35 | 43 | 43 | 53 | 53 | 64 | 64 | 76 | 76 | 90 | 90 | 106 | 106 | 106 | 122 |
| *e* 8 | High– | 14 | 20 | 25 | 32 | 32 | 40 | 40 | 50 | 50 | 60 | 60 | 72 | 72 | 85 | 85 | 85 | 100 |
| *e* 9 | High– | 14 | 20 | 25 | 32 | 32 | 40 | 40 | 50 | 50 | 60 | 60 | 72 | 72 | 85 | 85 | 85 | 100 |
| *e* 8 | Low– | 28 | 38 | 47 | 59 | 58 | 73 | 73 | 89 | 89 | 106 | 106 | 126 | 126 | 148 | 148 | 148 | 172 |
| & | | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & | & |
| *e* 9 | | 39 | 50 | 61 | 75 | 75 | 92 | 92 | 112 | 112 | 134 | 134 | 158 | 158 | 185 | 185 | 185 | 215 |
| *d* 9 | High– | 20 | 30 | 40 | 50 | 50 | 65 | 65 | 80 | 80 | 100 | 100 | 120 | 120 | 145 | 145 | 145 | 170 |
| | Low– | 45 | 60 | 76 | 93 | 93 | 117 | 117 | 142 | 142 | 174 | 174 | 207 | 207 | 245 | 245 | 245 | 285 |
| *c* 9 | High– | 60 | 70 | 80 | 95 | 95 | 110 | 110 | 120 | 130 | 140 | 150 | 170 | 180 | 200 | 210 | 230 | 240 |
| | Low– | 85 | 100 | 116 | 138 | 138 | 162 | 162 | 182 | 192 | 214 | 224 | 257 | 267 | 300 | 310 | 330 | 355 |
| *b* 9 | High– | 140 | 140 | 150 | 150 | 150 | 160 | 160 | 170 | 180 | 190 | 200 | 220 | 240 | 260 | 280 | 310 | 340 |
| | Low– | 165 | 170 | 186 | 193 | 193 | 212 | 212 | 232 | 242 | 264 | 274 | 307 | 327 | 360 | 380 | 410 | 545 |

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**Example 3.1.** *The dimensions of the mating parts, according to basic hole system, are given as follows :*

| | |
|---|---|
| *Hole : 25.00 mm* | *Shaft : 24.97 mm* |
| *25.02 mm* | *24.95 mm* |

*Find the hole tolerance, shaft tolerance and allowance.*

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

***Hole tolerance***

We know that hole tolerance

= Upper limit of hole – Lower limit of hole

= 25.02 – 25 = 0.02 mm **Ans.**

***Shaft tolerance***

We know that shaft tolerance

= Upper limit of shaft – Lower limit of shaft

= 24.97 – 24.95 = 0.02 mm **Ans.**

***Allowance***

We know that allowance

= Lower limit of hole – Upper limit of shaft

= 25.00 – 24.97 = 0.03 mm **Ans.**

**Example 3.2.** *Calculate the tolerances, fundamental deviations and limits of sizes for the shaft designated as 40 H8 / f7.*

**Solution.** Given: Shaft designation = 40 $H8$ / $f$ 7

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

***Tolerances***

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

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

We know that standard tolerance unit,

$$i = 0.45 \sqrt[3]{D} + 0.001\, D$$

$$= 0.45 \sqrt[3]{38.73} + 0.001 \times 38.73$$

$$= 0.45 \times 3.38 + 0.03873 = 1.559\ 73 \quad \text{or} \quad 1.56 \text{ microns}$$

$$= 1.56 \times 0.001 = 0.001\ 56 \text{ mm} \qquad \text{...}(\because 1 \text{ micron} = 0.001 \text{ mm})$$

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

$= 25\, i = 25 \times 0.001\ 56 = 0.039$ mm **Ans.**

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

$= 16\, i = 16 \times 0.001\ 56 = 0.025$ mm **Ans.**

**Note :** The value of $IT$ 8 and $IT$ 7 may be directly seen from Table 3.3.

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*Fundamental deviation*

We know that fundamental deviation (lower deviation) for hole $H$,

$$EI = 0$$

From Table 3.7, we find that fundamental deviation (upper deviation) for shaft $f$,

$$es = -5.5\,(D)^{0.41}$$
$$= -5.5\,(38.73)^{0.41} = -24.63 \text{ or } -25 \text{ microns}$$
$$= -25 \times 0.001 = -0.025 \text{ mm } \textbf{Ans.}$$

∴ Fundamental deviation (lower deviation) for shaft $f$,

$$ei = es - IT = -0.025 - 0.025 = -0.050 \text{ mm } \textbf{Ans.}$$

The –ve sign indicates that fundamental deviation lies below the zero line.

*Limits of sizes*

We know that lower limit for hole

= Basic size = 40 mm **Ans.**

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

= 40 + 0.039 = 40.039 mm **Ans.**

Upper limit for shaft = Lower limit for hole or Basic size – Fundamental deviation (upper deviation) ...(∵ Shaft $f$ lies below the zero line)

= 40 – 0.025 = 39.975 mm **Ans.**

and lower limit for shaft = Upper limit for shaft – Tolerance for shaft

= 39.975 – 0.025 = 39.95 mm **Ans.**

**Example 3.3.** *Give the dimensions for the hole and shaft for the following:*

(*a*) *A 12 mm electric motor sleeve bearing;*

(*b*) *A medium force fit on a 200 mm shaft; and*

(*c*) *A 50 mm sleeve bearing on the elevating mechanism of a road grader.*

**Solution.**

***(a) Dimensions for the hole and shaft for a 12 mm electric motor sleeve bearing***

From Table 3.5, we find that for an electric motor sleeve bearing, a shaft $e$ 8 should be used with $H$ 8 hole.

Since 12 mm size lies in the diameter steps of 10 to 18 mm, therefore the geometric mean diameter,

$$D = \sqrt{10 \times 18} = 13.4 \text{ mm}$$

We know that standard tolerance unit,

$$i = 0.45\sqrt[3]{D} + 0.001\,D$$
$$= 0.45\sqrt[3]{13.4} + 0.001 \times 13.4 = 1.07 + 0.0134 = 1.0834 \text{ microns}$$

∴ *Standard tolerance for shaft and hole of grade 8 ($IT$ 8)

= 25 $i$ ...(From Table 3.2)

= 25 × 1.0834 = 27 microns

= 27 × 0.001 = 0.027 mm ...(∵ 1 micron = 0.001 mm)

From Table 3.7, we find that upper deviation for shaft '$e$',

$$es = -11(D)^{0.41} = -11\,(13.4)^{0.41} = -32 \text{ microns}$$
$$= -32 \times 0.001 = -0.032 \text{ mm}$$

* The tolerance values may be taken directly from Table 3.3.

We know that lower deviation for shaft '*e*',

$$ei = es - IT = -0.032 - 0.027 = -0.059 \text{ mm}$$

∴ Dimensions for the hole (*H* 8)

$$= 12^{+0.027}_{+0.000} \quad \textbf{Ans.}$$

and dimension for the shaft (*e* 8)

$$= 12^{-0.032}_{-0.059} \quad \textbf{Ans.}$$

**(*b*) *Dimensions for the hole and shaft for a medium force fit on a 200 mm shaft***

From Table 3.5, we find that shaft *r* 6 with hole *H* 7 gives the desired fit.

Since 200 mm lies in the diameter steps of 180 mm to 250 mm, therefore the geometric mean diameter,

$$D = \sqrt{180 \times 250} = 212 \text{ mm}$$

We know that standard tolerance unit,

$$i = 0.45\sqrt[3]{D} + 0.001\,D$$

$$= 0.45\sqrt[3]{212} + 0.001 \times 212 = 2.68 + 0.212 = 2.892 \text{ microns}$$

∴ Standard tolerance for the shaft of grade 6 (*IT*6) from Table 3.2

$$= 10\,i = 10 \times 2.892 = 28.92 \text{ microns}$$

$$= 28.92 \times 0.001 = 0.02892 \text{ or } 0.029 \text{ mm}$$

and standard tolerance for the hole of grade 7 (*IT* 7)

$$= 16\,i = 16 \times 2.892 = 46 \text{ microns}$$

$$= 46 \times 0.001 = 0.046 \text{ mm}$$

We know that lower deviation for shaft '*r*' from Table 3.7

$$ei = \frac{1}{2}\left[(IT\,7 + 0.4\,D) + (IT\,7 + 0 \text{ to } 5)\right]$$

$$= \frac{1}{2}\left[(46 + 0.4 \times 212) + (46 + 3)\right] = 90 \text{ microns}$$

$$= 90 \times 0.001 = 0.09 \text{ mm}$$

and upper deviation for the shaft *r*,

$$es = ei + IT = 0.09 + 0.029 = 0.119 \text{ mm}$$

∴ Dimension for the hole *H* 7

$$= 200^{+0.046}_{+0.00} \quad \textbf{Ans.}$$

and dimension for the shaft *r* 6

$$= 200^{+0.119}_{+0.09} \quad \textbf{Ans.}$$

**(*c*) *Dimensions for the hole and shaft for a 50 mm sleeve bearing on the elevating mechanism of a road grader***

From Table 3.5, we find that for a sleeve bearing, a loose running fit will be suitable and a shaft *d* 9 should be used with hole *H* 8.

Since 50 mm size lies in the diameter steps of 30 to 50 mm or 50 to 80 mm, therefore the geometric mean diameter,

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

We know that standard tolerance unit,

$$i = 0.45\sqrt[3]{D} + 0.001\,D$$

$$= 0.45\sqrt[3]{38.73} + 0.001 \times 38.73$$

$$= 1.522 + 0.03873 = 1.56073 \quad \text{or} \quad 1.56 \text{ microns}$$

∴ Standard tolerance for the shaft of grade 9 (*IT* 9) from Table 3.2

$$= 40\,i = 40 \times 1.56 = 62.4 \text{ microns}$$

$$= 62.4 \times 0.001 = 0.0624 \quad \text{or} \quad 0.062 \text{ mm}$$

and standard tolerance for the hole of grade 8 (*IT* 8)

$$= 25\,i = 25 \times 1.56 = 39 \text{ microns}$$

$$= 39 \times 0.001 = 0.039 \text{ mm}$$

We know that upper deviation for the shaft *d*, from Table 3.7

$$es = -16\,(D)^{0.44} = -16\,(38.73)^{0.44} = -80 \text{ microns}$$

$$= -80 \times 0.001 = -0.08 \text{ mm}$$

and lower deviation for the shaft *d*,

$$ei = es - IT = -0.08 - 0.062 = -0.142 \text{ mm}$$

∴ Dimension for the hole *H* 8

$$= 50^{+0.039}_{+0.000} \quad \textbf{Ans.}$$

and dimension for the shaft *d* 9

$$= 50^{-0.08}_{-0.142} \quad \textbf{Ans.}$$

**Example 3.4.** *A journal of nominal or basic size of 75 mm runs in a bearing with close running fit. Find the limits of shaft and bearing. What is the maximum and minimum clearance?*

**Solution.** Given: Nominal or basic size = 75 mm

From Table 3.5, we find that the close running fit is represented by *H* 8/*g* 7, *i.e.* a shaft *g* 7 should be used with *H* 8 hole.

Since 75 mm lies in the diameter steps of 50 to 80 mm, therefore the geometric mean diameter,

$$D = \sqrt{50 \times 80} = 63 \text{ mm}$$

We know that standard tolerance unit,

$$i = 0.45\sqrt[3]{D} + 0.001\,D = 0.45\sqrt[3]{63} + 0.001 \times 63$$

$$= 1.79 + 0.063 = 1.853 \text{ micron}$$

$$= 1.853 \times 0.001 = 0.001\,853 \text{ mm}$$

∴ Standard tolerance for hole '*H*' of grade 8 (*IT* 8)

$$= 25\,i = 25 \times 0.001\,853 = 0.046 \text{ mm}$$

and standard tolerance for shaft '*g*' of grade 7 (*IT* 7)

$$= 16\,i = 16 \times 0.001\,853 = 0.03 \text{ mm}$$

From Table 3.7, we find that upper deviation for shaft *g*,

$$es = -2.5\,(D)^{0.34} = -2.5\,(63)^{0.34} = -10 \text{ micron}$$

$$= -10 \times 0.001 = -0.01 \text{ mm}$$

∴ Lower deviation for shaft $g$,

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

We know that lower limit for hole

= Basic size = 75 mm

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

= 75 + 0.046 = 75.046 mm

Upper limit for shaft = Lower limit for hole – Upper deviation for shaft

...(∵ Shaft $g$ lies below zero line)

= 75 – 0.01 = 74.99 mm

and lower limit for shaft = Upper limit for shaft – Tolerance for shaft

= 74.99 – 0.03 = 74.96 mm

We know that maximum clearance

= Upper limit for hole – Lower limit for shaft

= 75.046 – 74.96 = 0.086 mm **Ans.**

and minimum clearance = Lower limit for hole – Upper limit for shaft

= 75 – 74.99 = 0.01 mm **Ans.**

## 3.20 Surface Roughness and its Measurement

A little consideration will show that surfaces produced by different machining operations (*e.g.* turning, milling, shaping, planing, grinding and superfinishing) are of different characteristics. They show marked variations when compared with each other. The variation is judged by the degree of smoothness. A surface produced by superfinishing is the smoothest, while that by planing is the roughest. In the assembly of two mating parts, it becomes absolutely necessary to describe the surface finish in quantitative terms which is measure of micro-irregularities of the surface and expressed in microns. In order to prevent stress concentrations and proper functioning, it may be necessary to avoid or to have certain surface roughness.

There are many ways of expressing the surface roughness numerically, but the following two methods are commonly used :

1. Centre line average method (briefly known as CLA method), and
2. Root mean square method (briefly known as RMS method).

The ***centre line average method*** is defined as the average value of the ordinates between the surface and the mean line, measured on both sides of it. According to Indian standards, the surface finish is measured in terms of 'CLA' value and it is denoted by $Ra$.


***Landing Gear :*** *When an aircraft comes in to land, it has to lose a lot of energy in a very short time. the landing gear deals with this and prevents disaster. First, mechanical or liquid springs absorb energy rapidly by being compressed. As the springs relax, this energy will be released again, but in a slow controlled manner in a damper-the second energy absorber. Finally, the tyres absorb energy, getting hot in the process.*

$$\text{CLA value or } Ra \text{ (in microns)} = \frac{y_1 + y_2 + y_3 + \ldots y_n}{n}$$

where, $y_1, y_2, \ldots y_n$ are the ordinates measured on both sides of the mean line and $n$ are the number of ordinates.

The ***root mean square method*** is defined as the square root of the arithmetic mean of the squares of the ordinates. Mathematically,

$$\text{R.M.S. value (in microns)} = \sqrt{\frac{y_1^{\,2} + y_2^{\,2} + y_3^{\,2} + \ldots y_n^{\,2}}{n}}$$

According to Indian standards, following symbols are used to denote the various degrees of surface roughness :

| Symbol | Surface roughness (*Ra*) in microns |
|---|---|
| ∇ | 8 to 25 |
| ∇ ∇ | 1.6 to 8 |
| ∇ ∇ ∇ | 0.025 to 1.6 |
| ∇ ∇ ∇ ∇ | Less than 0.025 |

The following table shows the range of surface roughness that can be produced by various manufacturing processes.

**Table 3.11. Range of surface roughness.**

| *S.No.* | *Manufacturing process* | *Surface roughness in microns* | *S.No.* | *Manufacturing process* | *Surface roughness in microns* |
|---|---|---|---|---|---|
| 1. | Lapping | 0.012 to 0.016 | 9 | Extrusion | 0.16 to 5 |
| 2. | Honing | 0.025 to 0.40 | 10. | Boring | 0.40 to 6.3 |
| 3. | Cylindrical grinding | 0.063 to 5 | 11. | Milling | 0.32 to 25 |
| 4. | Surface grinding | 0.063 to 5 | 12. | Planing and shaping | 1.6 to 25 |
| 5. | Broaching | 0.40 to 3.2 | 13. | Drilling | 1.6 to 20 |
| 6. | Reaming | 0.40 to 3.2 | 14. | Sand casting | 5 to 50 |
| 7. | Turning | 0.32 to 25 | 15. | Die casting | 0.80 to 3.20 |
| 8. | Hot rolling | 2.5 to 50 | 16. | Forging | 1.60 to 2.5 |

## 3.21 Preferred Numbers

When a machine is to be made in several sizes with different powers or capacities, it is necessary to decide what capacities will cover a certain range efficiently with minimum number of sizes. It has been shown by experience that a certain range can be covered efficiently when it follows a geometrical progression with a constant ratio. The preferred numbers are the conventionally rounded off values derived from geometric series including the integral powers of 10 and having as common ratio of the following factors:

$$\sqrt[5]{10},\ \sqrt[10]{10},\ \sqrt[20]{10} \text{ and } \sqrt[40]{10}$$

These ratios are approximately equal to 1.58, 1.26, 1.12 and 1.06. The series of preferred numbers are designated as *R5, R10, R20 and R40 respectively. These four series are called ***basic series.*** The other series called ***derived series*** may be obtained by simply multiplying or dividing the basic sizes by 10, 100, etc. The preferred numbers in the series R5 are 1, 1.6, 2.5, 4.0 and 6.3. Table 3.12 shows basic series of preferred numbers according to IS : 1076 (Part I) – 1985 (Reaffirmed 1990).

* The symbol R is used as a tribute to Captain Charles Renard, the first man to use preferred numbers.

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**Notes : 1.** The standard sizes (in mm) for wrought metal products are shown in Table 3.13 according to IS : 1136 – 1990. The standard G.P. series used correspond to R10, R20 and R40.

**2.** The hoisting capacities (in tonnes) of cranes are in R10 series, while the hydraulic cylinder diameters are in R40 series and hydraulic cylinder capacities are in R5 series.

**3.** The basic thickness of sheet metals and diameter of wires are based on R10, R20 and R40 series. Wire diameter of helical springs are in R20 series.

**Table 3.12. Preferred numbers of the basic series, according to IS : 1076 (Part I)–1985 (Reaffirmed 1990).**

| *Basic series* | *Preferred numbers* |
|---|---|
| R5 | 1.00, 1.60, 2.50, 4.00, 6.30, 10.00 |
| R10 | 1.00, 1.25, 1.60, 2.00, 2.50, 3.15, 4.00, 5.00, 6.30, 8.00, 10.00 |
| R20 | 1.00, 1.12, 1.25, 1.40, 1.60, 1.80, 2.00, 2.24, 2.50, 2.80, 3.15, 3.55, 4.00, 4.50, 5.00, 5.60, 6.30, 7.10, 8.00, 9.00, 10.00 |
| R40 | 1.00, 1.06, 1.12, 1.18, 1.25, 1.32, 1.40, 1.50, 1.60, 1.70, 1.80, 1.90, 2.00, 2.12, 2.24, 2.36, 2.50, 2.65, 2.80, 3.00, 3.15, 3.35, 3.55, 3.75, 4.00, 4.25, 4.50, 4.75, 5.00, 5.30, 5.60, 6.00, 6.30, 6.70, 7.10, 7.50, 8.00, 8.50, 9.00, 9.50, 10.00 |

**Table 3.13. Preferred sizes for wrought metal products according to IS : 1136 – 1990.**

| *Size range* | *Preferred sizes (mm)* |
|---|---|
| 0.01 – 0.10 mm | 0.02, 0.025, 0.030, 0.04, 0.05, 0.06, 0.08 and 0.10 |
| 0.10 – 1 mm | 0.10, 0.11, 0.12, 0.14, 0.16, 0.18, 0.20, 0.22, 0.25, 0.28, 0.30, 0.32, 0.35, 0.36, 0.40, 0.45, 0.50, 0.55, 0.60, 0.63, 0.70, 0.80, 0.90 and 1 |
| 1 – 10 mm | 1, 1.1, 1.2, 1.4, 1.5, 1.6, 1.8, 2.22, 2.5, 2.8, 3, 3.2, 3.5, 3.6, 4, 4.5, 5, 5.5, 5.6, 6, 6.3, 7, 8, 9 and 10 |
| 10 – 100 mm | 10 to 25 (in steps of 1 mm), 28, 30, 32, 34, 35, 36, 38, 40, 42, 44, 45, 46, 48,50, 52, 53, 55, 56, 58, 60, 62, 63, 65, 67, 68, 70, 71, 72, 75, 78, 80, 82, 85, 88, 90, 92, 95, 98 and 100 |
| 100 – 1000 mm | 100 to 200 (in steps of 5 mm), 200 to 310 (in steps of 10 mm), 315, 320, 330, 340, 350, 355, 360, 370, 375, 380 to 500 (in steps of 10 mm), 520, 530,550, 560, 580, 600, 630, 650, 670, 700, 710 and 750 – 1000 (in steps of 50 mm) |
| 1000 – 10 000 mm | 1000, 1100, 1200, 1250, 1400, 1500, 1600, 1800, 2000, 2200, 2500, 2800, 3000, 3200, 3500, 3600, 4000, 4500, 5000, 5500, 5600, 6000, 6300, 7000, 7100, 8000, 9000 and 10 000 |

## EXERCISES

1. A journal of basic size of 75 mm rotates in a bearing. The tolerance for both the shaft and bearing is 0.075 mm and the required allowance is 0.10 mm. Find the dimensions of the shaft and the bearing bore. **[Ans. For shaft : 74.90 mm, 74.825 mm ; For hole : 75.075 mm, 75 mm]**
2. A medium force fit on a 75 mm shaft requires a hole tolerance and shaft tolerance each equal to 0.225 mm and average interference of 0.0375 mm. Find the hole and shaft dimensions. **[Ans. 75 mm, 75.225 mm ; 75.2625 mm, 75.4875 mm]**

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3. Calculate the tolerances, fundamental deviations and limits of size for hole and shaft in the following cases of fits :

(*a*) 25 $H$ 8 / $d$ 9; and (*b*) 60 $H$ 7 / $m$ 6

**[Ans. (*a*) 0.033 mm, 0.052 mm; 0, – 0.064 mm, – 0.116 mm; 25 mm, 25.033 mm, 24.936 mm, 24.884 mm (*b*) 0.03 mm, 0.019 mm; 0.011 mm, – 0.008 mm; 60 mm, 60.03 mm, 59.989 mm, 59.97 mm]**

4. Find the extreme diameters of shaft and hole for a transition fit $H7/n6$, if the nominal or basic diameter is 12 mm. What is the value of clearance and interference?

**[Ans. 12.023 mm, 12.018 mm; 0.006 mm, – 0.023 mm]**

5. A gear has to be shrunk on a shaft of basic size 120 mm. An interference fit $H7/u6$ is being selected. Determine the minimum and maximum diameter of the shaft and interference.

**[Ans. 120.144 mm, 120.166 mm; 0.109 mm, 0.166 mm]**

## QUESTIONS

1. Enumerate the various manufacturing methods of machine parts which a designer should know.
2. Explain briefly the different casting processes.
3. Write a brief note on the design of castings?
4. State and illustrate two principal design rules for casting design.
5. List the main advantages of forged components.
6. What are the salient features used in the design of forgings? Explain.
7. What do you understand by 'hot working' and 'cold working' processes? Explain with examples.
8. State the advantages and disadvantages of hot working of metals. Discuss any two hot working processes.
9. What do you understand by cold working of metals? Describe briefly the various cold working processes.
10. What are fits and tolerances? How are they designated?
11. What do you understand by the nominal size and basic size?
12. Write short notes on the following :

    (*a*) Interchangeability; (*b*) Tolerance; (*c*) Allowance; and (*d*) Fits.
13. What is the difference in the type of assembly generally used in running fits and interference fits?
14. State briefly unilateral system of tolerances covering the points of definition, application and advantages over the bilateral system.
15. What is meant by 'hole basis system' and 'shaft basis system'? Which one is preferred and why?
16. Discuss the Indian standard system of limits and fits.
17. What are the commonly used fits according to Indian standards?
18. What do you understand by preferred numbers? Explain fully.

## OBJECTIVE TYPE QUESTIONS

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

   (*a*) permanent mould casting (*b*) slush casting
   (*c*) die casting (*d*) centrifugal casting
2. The metal is subjected to mechanical working for

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

3. The temperature at which the new grains are formed in the metal is called
   (*a*) lower critical temperature (*b*) upper critical temperature
   (*c*) eutectic temperature (*d*) recrystallisation temperature
4. The hot working of metals is carried out
   (*a*) at the recrystallisation temperature (*b*) below the recrystallisation temperature
   (*c*) above the recrystallisation temperature (*d*) at any temperature
5. During hot working of metals
   (*a*) porosity of the metal is largely eliminated
   (*b*) grain structure of the metal is refined
   (*c*) mechanical properties are improved due to refinement of grains
   (*d*) all of the above
6. The parts of circular cross-section which are symmetrical about the axis of rotation are made by
   (*a*) hot forging (*b*) hot spinning
   (*c*) hot extrusion (*d*) hot drawing
7. The cold working of metals is carried out ............. the recrystallisation temperature.
   (*a*) above (*b*) below
8. The process extensively used for making bolts and nuts is
   (*a*) hot piercing (*b*) extrusion
   (*c*) cold peening (*d*) cold heading
9. In a unilateral system of tolerance, the tolerance is allowed on
   (*a*) one side of the actual size (*b*) one side of the nominal size
   (*c*) both sides of the actual size (*d*) both sides of the nominal size
10. The algebraic difference between the maximum limit and the basic size is called
    (*a*) actual deviation (*b*) upper deviation
    (*c*) lower deviation (*d*) fundamental deviation
11. A basic shaft is one whose
    (*a*) lower deviation is zero (*b*) upper deviation is zero
    (*c*) lower and upper deviations are zero (*d*) none of these
12. A basic hole is one whose
    (*a*) lower deviation is zero (*b*) upper deviation is zero
    (*c*) lower and upper deviations are zero (*d*) none of these
13. According to Indian standard specifications, 100 $H\,6\,/\,g\,5$ means that the
    (*a*) actual size is 100 mm
    (*b*) basic size is 100 mm
    (*c*) difference between the actual size and basic size is 100 mm
    (*d*) none of the above
14. According to Indian standards, total number of tolerance grades are
    (*a*) 8 (*b*) 12
    (*c*) 18 (*d*) 20
15. According to Indian standard specification, 100 $H6/g5$ means that
    (*a*) tolerance grade for the hole is 6 and for the shaft is 5
    (*b*) tolerance grade for the shaft is 6 and for the hole is 5
    (*c*) tolerance grade for the shaft is 4 to 8 and for the hole is 3 to 7
    (*d*) tolerance grade for the hole is 4 to 8 and for the shaft is 3 to 7

## ANSWERS

| | | | | |
|---|---|---|---|---|
| **1.** (*c*) | **2.** (*d*) | **3.** (*d*) | **4.** (*c*) | **5.** (*d*) |
| **6.** (*b*) | **7.** (*b*) | **8.** (*d*) | **9.** (*b*) | **10.** (*b*) |
| **11.** (*b*) | **12.** (*a*) | **13.** (*b*) | **14.** (*c*) | **15.** (*a*) |

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CHAPTER

# 4

# Simple Stresses in Machine Parts



## 4.1 Introduction

In engineering practice, the machine parts are subjected to various forces which may be due to either one or more of the following:

1. Energy transmitted,
2. Weight of machine,
3. Frictional resistances,
4. Inertia of reciprocating parts,
5. Change of temperature, and
6. Lack of balance of moving parts.

The different forces acting on a machine part produces various types of stresses, which will be discussed in this chapter.

## 4.2 Load

It is defined as **any external force acting upon a machine part.** The following four types of the load are important from the subject point of view:



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**1. *Dead or steady load.*** A load is said to be a dead or steady load, when it does not change in magnitude or direction.

**2. *Live or variable load.*** A load is said to be a live or variable load, when it changes continually.

**3. *Suddenly applied or shock loads*.** A load is said to be a suddenly applied or shock load, when it is suddenly applied or removed.

**4. *Impact load.*** A load is said to be an impact load, when it is applied with some initial velocity.

**Note:** A machine part resists a dead load more easily than a live load and a live load more easily than a shock load.

## 4.3 Stress

When some external system of forces or loads act on a body, the internal forces (equal and opposite) are set up at various sections of the body, which resist the external forces. This internal force per unit area at any section of the body is known as ***unit stress*** or simply a ***stress***. It is denoted by a Greek letter sigma ($\sigma$). Mathematically,

$$\text{Stress, } \sigma = P/A$$

where $P$ = Force or load acting on a body, and

$A$ = Cross-sectional area of the body.

In S.I. units, the stress is usually expressed in Pascal (Pa) such that 1 Pa = 1 N/m$^2$. In actual practice, we use bigger units of stress *i.e.* megapascal (MPa) and gigapascal (GPa), such that

$$1 \text{ MPa} = 1 \times 10^6 \text{ N/m}^2 = 1 \text{ N/mm}^2$$

and

$$1 \text{ GPa} = 1 \times 10^9 \text{ N/m}^2 = 1 \text{ kN/mm}^2$$

## 4.4 Strain

When a system of forces or loads act on a body, it undergoes some deformation. This deformation per unit length is known as ***unit strain*** or simply a ***strain.*** It is denoted by a Greek letter epsilon ($\varepsilon$). Mathematically,

$$\text{Strain, } \varepsilon = \delta l / l \quad \text{or} \quad \delta l = \varepsilon . l$$

where $\delta l$ = Change in length of the body, and

$l$ = Original length of the body.

## 4.5 Tensile Stress and Strain

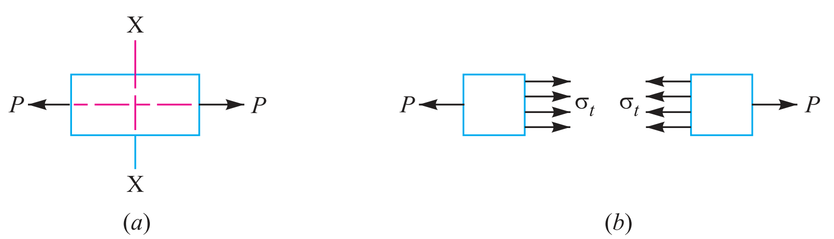


**Fig. 4.1.** Tensile stress and strain.

When a body is subjected to two equal and opposite axial pulls $P$ (also called tensile load) as shown in Fig. 4.1 (*a*), then the stress induced at any section of the body is known as ***tensile stress*** as shown in Fig. 4.1 (*b*). A little consideration will show that due to the tensile load, there will be a decrease in cross-sectional area and an increase in length of the body. The ratio of the increase in length to the original length is known as ***tensile strain.***

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Let $P$ = Axial tensile force acting on the body,

$A$ = Cross-sectional area of the body,

$l$ = Original length, and

$\delta l$ = Increase in length.

$\therefore$ Tensile stress, $\sigma_t = P/A$

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

*Shock absorber of a motorcycle absorbs stresses.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

## 4.6 Compressive Stress and Strain

When a body is subjected to two equal and opposite axial pushes $P$ (also called compressive load) as shown in Fig. 4.2 (*a*), then the stress induced at any section of the body is known as ***compressive stress*** as shown in Fig. 4.2 (*b*). A little consideration will show that due to the compressive load, there will be an increase in cross-sectional area and a decrease in length of the body. The ratio of the decrease in length to the original length is known as ***compressive strain.***

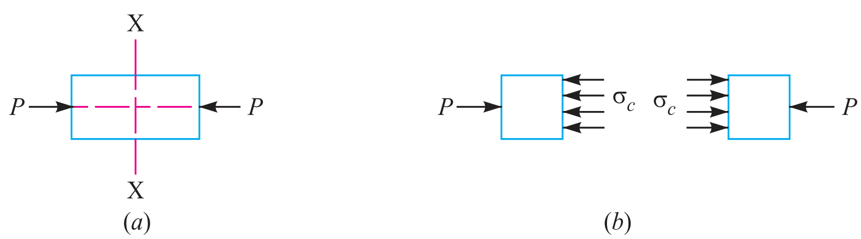


**Fig. 4.2.** Compressive stress and strain.

Let $P$ = Axial compressive force acting on the body,

$A$ = Cross-sectional area of the body,

$l$ = Original length, and

$\delta l$ = Decrease in length.

$\therefore$ Compressive stress, $\sigma_c = P/A$

and compressive strain, $\varepsilon_c = \delta l / l$

**Note :** In case of tension or compression, the area involved is at right angles to the external force applied.

## 4.7 Young's Modulus or Modulus of Elasticity

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

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

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

* It is named after Robert Hooke, who first established it by experiments in 1678.

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

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

**Table 4.1. Values of E for the commonly used engineering materials.**

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

**Example 4.1.** *A coil chain of a crane required to carry a maximum load of 50 kN, is shown in Fig. 4.3.*

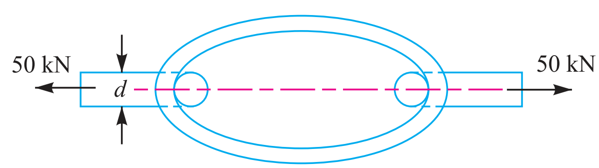


**Fig. 4.3**

*Find the diameter of the link stock, if the permissible tensile stress in the link material is not to exceed 75 MPa.*

**Solution.** Given : $P = 50 \text{ kN} = 50 \times 10^3 \text{ N}$ ; $\sigma_t = 75 \text{ MPa} = 75 \text{ N/mm}^2$

Let $d$ = Diameter of the link stock in mm.

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

We know that the maximum load ($P$),

$$50 \times 10^3 = \sigma_t . A = 75 \times 0.7854\, d^2 = 58.9\, d^2$$

$$\therefore \quad d^2 = 50 \times 10^3 / 58.9 = 850 \quad \text{or} \quad d = 29.13 \text{ say } 30 \text{ mm}$$ **Ans.**

**Example 4.2.** *A cast iron link, as shown in Fig. 4.4, is required to transmit a steady tensile load of 45 kN. Find the tensile stress induced in the link material at sections A-A and B-B.*

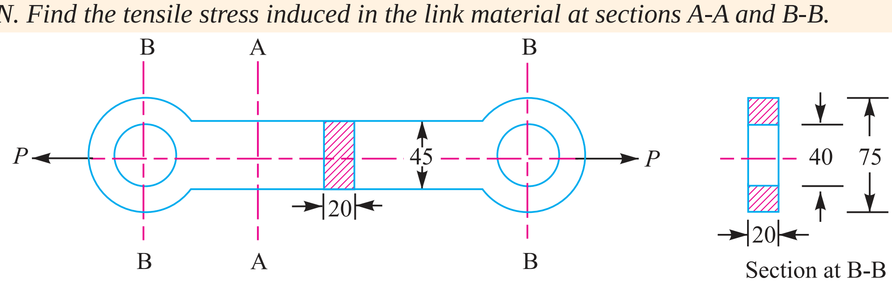


**Fig. 4.4.** All dimensions in mm.

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**Solution.** Given : $P = 45 \text{ kN} = 45 \times 10^3 \text{ N}$

***Tensile stress induced at section A-A***

We know that the cross-sectional area of link at section *A-A*,

$$A_1 = 45 \times 20 = 900 \text{ mm}^2$$

∴ Tensile stress induced at section *A-A*,

$$\sigma_{t1} = \frac{P}{A_1} = \frac{45 \times 10^3}{900} = 50 \text{ N/mm}^2 = 50 \text{ MPa} \quad \textbf{Ans.}$$

***Tensile stress induced at section B-B***

We know that the cross-sectional area of link at section B-B,

$$A_2 = 20\,(75 - 40) = 700 \text{ mm}^2$$

∴ Tensile stress induced at section *B-B*,

$$\sigma_{t2} = \frac{P}{A_2} = \frac{45 \times 10^3}{700} = 64.3 \text{ N/mm}^2 = 64.3 \text{ MPa} \quad \textbf{Ans.}$$

**Example 4.3.** *A hydraulic press exerts a total load of 3.5 MN. This load is carried by two steel rods, supporting the upper head of the press. If the safe stress is 85 MPa and E = 210 kN/mm², find : 1. diameter of the rods, and 2. extension in each rod in a length of 2.5 m.*

**Solution.** Given : $P = 3.5 \text{ MN} = 3.5 \times 10^6 \text{ N}$ ; $\sigma_t = 85 \text{ MPa} = 85 \text{ N/mm}^2$ ; $E = 210 \text{ kN/mm}^2 = 210 \times 10^3 \text{ N/mm}^2$ ; $l = 2.5 \text{ m} = 2.5 \times 10^3 \text{ mm}$

***1. Diameter of the rods***

Let $d$ = Diameter of the rods in mm.

∴ Area, $A = \frac{\pi}{4} \times d^2 = 0.7854\, d^2$

Since the load $P$ is carried by two rods, therefore load carried by each rod,

$$P_1 = \frac{P}{2} = \frac{3.5 \times 10^6}{2} = 1.75 \times 10^6 \text{ N}$$

We know that load carried by each rod ($P_1$),

$$1.75 \times 10^6 = \sigma_t \,.\, A = 85 \times 0.7854\, d^2 = 66.76\, d^2$$

∴ $d^2 = 1.75 \times 10^6/66.76 = 26\,213$ or $d = 162$ mm **Ans.**

***2. Extension in each rod***

Let $\delta l$ = Extension in each rod.

We know that Young's modulus ($E$),

$$210 \times 10^3 = \frac{P_1 \times l}{A \times \delta l} = \frac{\sigma_t \times l}{\delta l} = \frac{85 \times 2.5 \times 10^3}{\delta l} = \frac{212.5 \times 10^3}{\delta l} \quad \ldots \left( \because \frac{P_1}{A} = \sigma_t \right)$$

∴ $\delta l = 212.5 \times 10^3/(210 \times 10^3) = 1.012$ mm **Ans.**

**Example 4.4.** *A rectangular base plate is fixed at each of its four corners by a 20 mm diameter bolt and nut as shown in Fig. 4.5. The plate rests on washers of 22 mm internal diameter and 50 mm external diameter. Copper washers which are placed between the nut and the plate are of 22 mm internal diameter and 44 mm external diameter.*

*If the base plate carries a load of 120 kN (including self-weight, which is equally distributed on the four corners), calculate the stress on the lower washers before the nuts are tightened.*

*What could be the stress in the upper and lower washers, when the nuts are tightened so as to produce a tension of 5 kN on each bolt?*

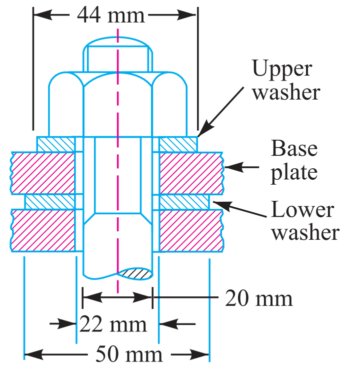


**Fig. 4.5**

**Solution.** Given : $d$ = 20 mm ; $d_1$ = 22 mm ; $d_2$ = 50 mm ; $d_3$ = 22 mm ; $d_4$ = 44 mm ; $P_1$ = 120 kN ; $P_2$ = 5 kN

***Stress on the lower washers before the nuts are tightened***

We know that area of lower washers,

$$A_1 = \frac{\pi}{4}\left[(d_2)^2 - (d_1)^2\right] = \frac{\pi}{4}\left[(50)^2 - (22)^2\right] = 1583 \text{ mm}^2$$

and area of upper washers,

$$A_2 = \frac{\pi}{4}\left[(d_4)^2 - (d_3)^2\right] = \frac{\pi}{4}\left[(44)^2 - (22)^2\right] = 1140 \text{ mm}^2$$

Since the load of 120 kN on the four washers is equally distributed, therefore load on each lower washer before the nuts are tightened,

$$P_1 = \frac{120}{4} = 30 \text{ kN} = 30\,000 \text{ N}$$

We know that stress on the lower washers before the nuts are tightened,

$$\sigma_{c1} = \frac{P_1}{A_1} = \frac{30\,000}{1583} = 18.95 \text{ N/mm}^2 = 18.95 \text{ MPa} \quad \textbf{Ans.}$$

***Stress on the upper washers when the nuts are tightened***

Tension on each bolt when the nut is tightened,

$$P_2 = 5 \text{ kN} = 5000 \text{ N}$$

∴ Stress on the upper washers when the nut is tightened,

$$\sigma_{c2} = \frac{P_2}{A_2} = \frac{5000}{1140} = 4.38 \text{ N/mm}^2 = 4.38 \text{ MPa} \quad \textbf{Ans.}$$

***Stress on the lower washers when the nuts are tightened***

We know that the stress on the lower washers when the nuts are tightened,

$$\sigma_{c3} = \frac{P_1 + P_2}{A_1} = \frac{30\,000 + 5000}{1583} = 22.11 \text{ N/mm}^2 = 22.11 \text{ MPa} \; \textbf{Ans.}$$

**Example 4.5.** *The piston rod of a steam engine is 50 mm in diameter and 600 mm long. The diameter of the piston is 400 mm and the maximum steam pressure is 0.9 N/mm². Find the compression of the piston rod if the Young's modulus for the material of the piston rod is 210 kN/mm².*

**Solution.** Given : $d$ = 50 mm ; $l$ = 600 mm ; $D$ = 400 mm ; $p$ = 0.9 N/mm$^2$ ; $E$ = 210 kN/mm$^2$ = 210 × 10$^3$ N/mm$^2$

Let $\delta l$ = Compression of the piston rod.

We know that cross-sectional area of piston,

$$= \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(400)^2 = 125\,680 \text{ mm}^2$$

∴ Maximum load acting on the piston due to steam,

$$P = \text{Cross-sectional area of piston} \times \text{Steam pressure}$$
$$= 125\,680 \times 0.9 = 113\,110 \text{ N}$$

We also know that cross-sectional area of piston rod,

$$A = \frac{\pi}{4} \times d^2 = \frac{\pi}{4} (50)^2$$

$$= 1964 \text{ mm}^2$$

and Young's modulus ($E$),

$$210 \times 10^3 = \frac{P \times l}{A \times \delta l}$$

$$= \frac{113\,110 \times 600}{1964 \times \delta l} = \frac{34\,555}{\delta l}$$

$$\therefore \quad \delta l = 34\,555 / (210 \times 10^3)$$

$$= 0.165 \text{ mm}$$ **Ans.**

*This picture shows a jet engine being tested for bearing high stresses.*

## 4.8 Shear Stress and Strain

When a body is subjected to two equal and opposite forces acting tangentially across the resisting section, as a result of which the body tends to shear off the section, then the stress induced is called ***shear stress.***

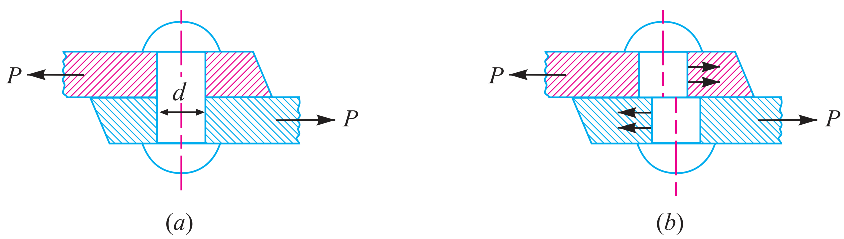


**Fig. 4.6.** Single shearing of a riveted joint.

The corresponding strain is known as ***shear strain*** and it is measured by the angular deformation accompanying the shear stress. The shear stress and shear strain are denoted by the Greek letters tau ($\tau$) and phi ($\phi$) respectively. Mathematically,

$$\text{Shear stress, } \tau = \frac{\text{Tangential force}}{\text{Resisting area}}$$

Consider a body consisting of two plates connected by a rivet as shown in Fig. 4.6 (*a*). In this case, the tangential force $P$ tends to shear off the rivet at one cross-section as shown in Fig. 4.6 (*b*). It may be noted that when the tangential force is resisted by one cross-section of the rivet (or when shearing takes place at one cross-section of the rivet), then the rivets are said to be in ***single shear.*** In such a case, the area resisting the shear off the rivet,

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

and shear stress on the rivet cross-section,

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

Now let us consider two plates connected by the two cover plates as shown in Fig. 4.7 (*a*). In this case, the tangential force $P$ tends to shear off the rivet at two cross-sections as shown in Fig. 4.7 (*b*). It may be noted that when the tangential force is resisted by two cross-sections of the rivet (or when the shearing takes place at two cross-sections of the rivet), then the rivets are said to be in ***double shear.*** In such a case, the area resisting the shear off the rivet,

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

and shear stress on the rivet cross-section,

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

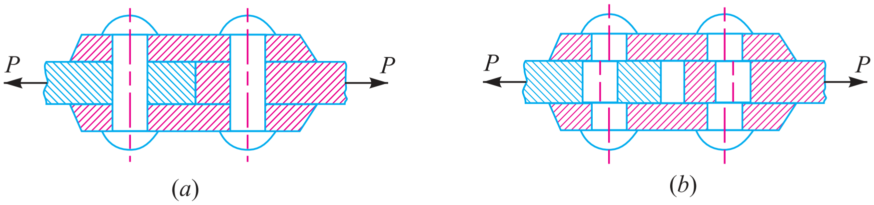


**Fig. 4.7.** Double shearing of a riveted joint.

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

**2.** In case of shear, the area involved is parallel to the external force applied.

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

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

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

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

where $\tau_u$ = Ultimate shear strength of the material of the plate.

## 4.9 Shear Modulus or Modulus of Rigidity

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

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

where $\tau$ = Shear stress,

$\phi$ = Shear strain, and

$C$ = Constant of proportionality, known as shear modulus or modulus of rigidity. It is also denoted by *N* or *G*.

The following table shows the values of modulus of rigidity (*C*) for the materials in every day use:

**Table 4.2. Values of *C* for the commonly used materials.**

| *Material* | *Modulus of rigidity (C) in GPa i.e. GN/m² or kN/mm²* |
|---|---|
| Steel | 80 to 100 |
| Wrought iron | 80 to 90 |
| Cast iron | 40 to 50 |
| Copper | 30 to 50 |
| Brass | 30 to 50 |
| Timber | 10 |

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**Example 4.6.** *Calculate the force required to punch a circular blank of 60 mm diameter in a plate of 5 mm thick. The ultimate shear stress of the plate is 350 N/mm².*

**Solution.** Given: $d = 60$ mm ; $t = 5$ mm ; $\tau_u = 350$ N/mm²

We know that area under shear,

$$A = \pi\, d \times \tau = \pi \times 60 \times 5 = 942.6 \text{ mm}^2$$

and force required to punch a hole,

$$P = A \times \tau_u = 942.6 \times 350 = 329\,910 \text{ N} = 329.91 \text{ kN} \quad \textbf{Ans.}$$

**Example 4.7.** *A pull of 80 kN is transmitted from a bar X to the bar Y through a pin as shown in Fig. 4.8.*

*If the maximum permissible tensile stress in the bars is 100 N/mm² and the permissible shear stress in the pin is 80 N/mm², find the diameter of bars and of the pin.*

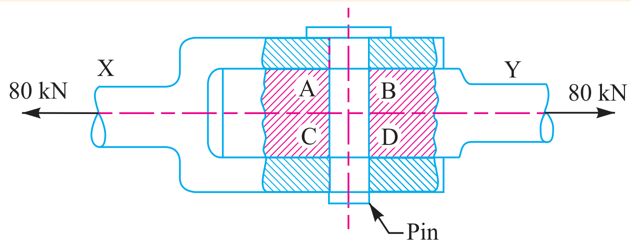


**Fig. 4.8**

**Solution.** Given : $P = 80$ kN $= 80 \times 10^3$ N; $\sigma_t = 100$ N/mm² ; $\tau = 80$ N/mm²

***Diameter of the bars***

Let $D_b$ = Diameter of the bars in mm.

$$\therefore \text{ Area, } A_b = \frac{\pi}{4} (D_b)^2 = 0.7854 (D_b)^2$$

We know that permissible tensile stress in the bar ($\sigma_t$),

$$100 = \frac{P}{A_b} = \frac{80 \times 10^3}{0.7854 (D_b)^2} = \frac{101\,846}{(D_b)^2}$$

$$\therefore \quad (D_b)^2 = 101\,846 / 100 = 1018.46$$

or $D_b = 32$ mm **Ans.**


*High force injection moulding machine.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

***Diameter of the pin***

Let $D_p$ = Diameter of the pin in mm.

Since the tensile load $P$ tends to shear off the pin at two sections *i.e.* at $AB$ and $CD$, therefore the pin is in double shear.

∴ Resisting area,

$$A_p = 2 \times \frac{\pi}{4} (D_p)^2 = 1.571 (D_p)^2$$

We know that permissible shear stress in the pin ($\tau$),

$$80 = \frac{P}{A_p} = \frac{80 \times 10^3}{1.571 (D_p)^2} = \frac{50.9 \times 10^3}{(D_p)^2}$$

$$\therefore \quad (D_p)^2 = 50.9 \times 10^3/80 = 636.5 \text{ or } D_p = 25.2 \text{ mm} \quad \textbf{Ans.}$$

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## 4.10 Bearing Stress

A localised compressive stress at the surface of contact between two members of a machine part, that are relatively at rest is known as ***bearing stress*** or ***crushing stress*.** The bearing stress is taken into account in the design of riveted joints, cotter joints, knuckle joints, etc. Let us consider a riveted joint subjected to a load $P$ as shown in Fig. 4.9. In such a case, the bearing stress or crushing stress (stress at the surface of contact between the rivet and a plate),

$$\sigma_b \text{ (or } \sigma_c\text{)} = \frac{P}{d.t.n}$$

where $d$ = Diameter of the rivet,

$t$ = Thickness of the plate,

$d.t$ = Projected area of the rivet, and

$n$ = Number of rivets per pitch length in bearing or crushing.

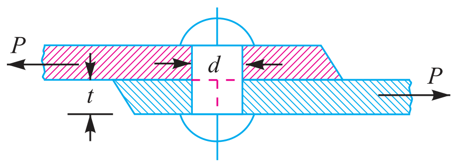
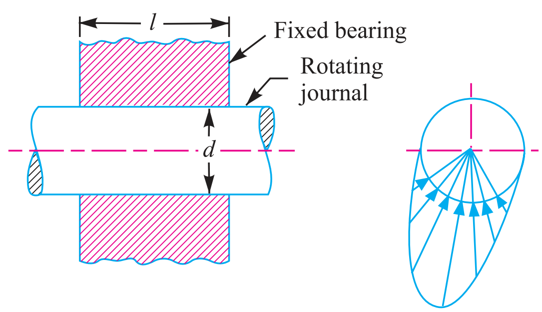



**Fig. 4.9.** Bearing stress in a riveted joint.




(*a*) Journal supported in a bearing.

(*b*) Distribution of bearing pressure.

**Fig. 4.10.** Bearing pressure in a journal supported in a bearing.

It may be noted that the local compression which exists at the surface of contact between two members of a machine part that are in relative motion, is called ***bearing pressure*** (not the bearing stress). This term is commonly used in the design of a journal supported in a bearing, pins for levers, crank pins, clutch lining, etc. Let us consider a journal rotating in a fixed bearing as shown in Fig. 4.10 (*a*). The journal exerts a bearing pressure on the curved surfaces of the brasses immediately below it. The distribution of this bearing pressure will not be uniform, but it will be in accordance with the shape of the surfaces in contact and deformation characteristics of the two materials. The distribution of bearing pressure will be similar to that as shown in Fig. 4.10 (*b*). Since the actual bearing pressure is difficult to determine, therefore the average bearing pressure is usually calculated by dividing the load to the projected area of the curved surfaces in contact. Thus, the average bearing pressure for a journal supported in a bearing is given by

$$p_b = \frac{P}{l.d}$$

where $p_b$ = Average bearing pressure,

$P$ = Radial load on the journal,

$l$ = Length of the journal in contact, and

$d$ = Diameter of the journal.

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**Example 4.8.** *Two plates 16 mm thick are joined by a double riveted lap joint as shown in Fig. 4.11. The rivets are 25 mm in diameter.*

*Find the crushing stress induced between the plates and the rivet, if the maximum tensile load on the joint is 48 kN.*

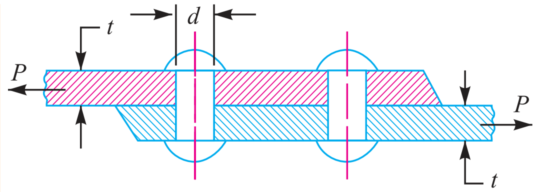


**Fig. 4.11**

**Solution.** Given : $t$ = 16 mm ; $d$ = 25 mm ; $P$ = 48 kN = 48 × $10^3$ N

Since the joint is double riveted, therefore, strength of two rivets in bearing (or crushing) is taken. We know that crushing stress induced between the plates and the rivets,

$$\sigma_c = \frac{P}{d.t.n} = \frac{48 \times 10^3}{25 \times 16 \times 2} = 60 \text{ N/mm}^2 \textbf{ Ans.}$$

**Example 4.9.** *A journal 25 mm in diameter supported in sliding bearings has a maximum end reaction of 2500 N. Assuming an allowable bearing pressure of 5 N/mm$^2$, find the length of the sliding bearing.*

**Solution.** Given : $d$ = 25 mm ; $P$ = 2500 N ; $p_b$ = 5 N/mm$^2$

Let $\quad l$ = Length of the sliding bearing in mm.

We know that the projected area of the bearing,

$$A = l \times d = l \times 25 = 25\, l \text{ mm}^2$$

∴ Bearing pressure ( $p_b$),

$$5 = \frac{P}{A} = \frac{2500}{25\, l} = \frac{100}{l} \quad \text{or} \quad l = \frac{100}{5} = 20 \text{ mm } \textbf{Ans.}$$

## 4.11 Stress-strain Diagram

In designing various parts of a machine, it is necessary to know how the material will function in service. For this, certain characteristics or properties of the material should be known. The mechanical properties mostly used in mechanical engineering practice are commonly determined from a standard tensile test. This test consists of gradually loading a standard specimen of a material and noting the corresponding values of load and elongation until the specimen fractures. The load is applied and measured by a testing machine. The stress is determined by dividing the load values by the original cross-sectional area of the specimen. The elongation is measured by determining the amounts that two reference points on the specimen are moved apart by the action of the machine. The original distance between the two reference points is known as ***gauge length.*** The strain is determined by dividing the elongation values by the gauge length.

*In addition to bearing the stresses, some machine parts are made of stainless steel to make them corrosion resistant.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

The values of the stress and corresponding strain are used to draw the stress-strain diagram of the material tested. A stress-strain diagram for a mild steel under tensile test is shown in Fig. 4.12 ($a$). The various properties of the material are discussed below :

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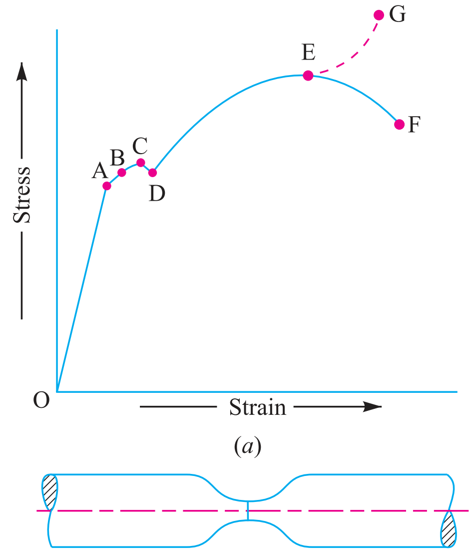
**1.** ***Proportional limit.*** We see from the diagram that from point *O* to *A* is a straight line, which represents that the stress is proportional to strain. Beyond point *A*, the curve slightly deviates from the straight line. It is thus obvious, that Hooke's law holds good up to point *A* and it is known as ***proportional limit***. It is defined as that stress at which the stress-strain curve begins to deviate from the straight line.

**2.** ***Elastic limit.*** It may be noted that even if the load is increased beyond point *A* upto the point *B*, the material will regain its shape and size when the load is removed. This means that the material has elastic properties up to the point *B*. This point is known as ***elastic limit.*** It is defined as the stress developed in the material without any permanent set.




(*a*)

(*b*) Shape of specimen after elongation.

**Fig. 4.12.** Stress-strain diagram for a mild steel.

**Note:** Since the above two limits are very close to each other, therefore, for all practical purposes these are taken to be equal.

**3.** ***Yield point.*** If the material is stressed beyond point *B*, the plastic stage will reach *i.e.* on the removal of the load, the material will not be able to recover its original size and shape. A little consideration will show that beyond point *B*, the strain increases at a faster rate with any increase in the stress until the point *C* is reached. At this point, the material yields before the load and there is an appreciable strain without any increase in stress. In case of mild steel, it will be seen that a small load drops to *D*, immediately after yielding commences. Hence there are two yield points *C* and *D*. The points *C* and *D* are called the ***upper*** and ***lower yield points*** respectively. The stress corresponding to yield point is known as ***yield point stress.***

**4.** ***Ultimate stress.*** At *D*, the specimen regains some strength and higher values of stresses are required for higher strains, than those between *A* and *D*. The stress (or load) goes on increasing till the

*A crane used on a ship.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

point $E$ is reached. The gradual increase in the strain (or length) of the specimen is followed with the uniform reduction of its cross-sectional area. The work done, during stretching the specimen, is transformed largely into heat and the specimen becomes hot. At $E$, the stress, which attains its maximum value is known as ***ultimate stress***. It is defined as the largest stress obtained by dividing the largest value of the load reached in a test to the original cross-sectional area of the test piece.

A recovery truck with crane.

Note : This picture is given as additional information and is not a direct example of the current chapter.

**5.** ***Breaking stress.*** After the specimen has reached the ultimate stress, a neck is formed, which decreases the cross-sectional area of the specimen, as shown in Fig. 4.12 (*b*). A little consideration will show that the stress (or load) necessary to break away the specimen, is less than the maximum stress. The stress is, therefore, reduced until the specimen breaks away at point *F*. The stress corresponding to point *F* is known as ***breaking stress.***

**Note :** The breaking stress (*i.e.* stress at *F* which is less than at *E*) appears to be somewhat misleading. As the formation of a neck takes place at *E* which reduces the cross-sectional area, it causes the specimen suddenly to fail at *F*. If for each value of the strain between *E* and *F*, the tensile load is divided by the reduced cross-sectional area at the narrowest part of the neck, then the true stress-strain curve will follow the dotted line *EG*. However, it is an established practice, to calculate strains on the basis of original cross-sectional area of the specimen.

**6.** ***Percentage reduction in area.*** It is the difference between the original cross-sectional area and cross-sectional area at the neck (*i.e.* where the fracture takes place). This difference is expressed as percentage of the original cross-sectional area.

Let $A$ = Original cross-sectional area, and

$a$ = Cross-sectional area at the neck.

Then reduction in area $= A - a$

and percentage reduction in area $= \dfrac{A - a}{A} \times 100$

**7.** ***Percentage elongation.*** It is the percentage increase in the standard gauge length (*i.e.* original length) obtained by measuring the fractured specimen after bringing the broken parts together.

Let $l$ = Gauge length or original length, and

$L$ = Length of specimen after fracture or final length.

$\therefore$ Elongation $= L - l$

and percentage elongation $= \dfrac{L - l}{l} \times 100$

**Note :** The percentage elongation gives a measure of ductility of the metal under test. The amount of local extensions depends upon the material and also on the transverse dimensions of the test piece. Since the specimens are to be made from bars, strips, sheets, wires, forgings, castings, etc., therefore it is not possible to make all specimens of one standard size. Since the dimensions of the specimen influence the result, therefore some standard means of comparison of results are necessary.

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As a result of series of experiments, Barba estabilished a law that in tension, similar test pieces deform similarly and two test pieces are said to be similar if they have the same value of $\frac{l}{\sqrt{A}}$, where $l$ is the gauge length and $A$ is the cross-sectional area. A little consideration will show that the same material will give the same percentage elongation and percentage reduction in area.

It has been found experimentally by Unwin that the general extension (up to the maximum load) is proportional to the gauge length of the test piece and that the local extension (from maximum load to the breaking load) is proportional to the square root of the cross-sectional area. According to Unwin's formula, the increase in length,

$$\delta l = b.l + C\sqrt{A}$$

and percentage elongation

$$= \frac{\delta l}{l} \times 100$$

where $l$ = Gauge length,

$A$ = Cross-sectional area, and

$b$ and $C$ = Constants depending upon the quality of the material.

The values of $b$ and $C$ are determined by finding the values of $\delta l$ for two test pieces of known length ($l$) and area ($A$).

**Example 4.10.** *A mild steel rod of 12 mm diameter was tested for tensile strength with the gauge length of 60 mm. Following observations were recorded :*

*Final length = 80 mm; Final diameter = 7 mm; Yield load = 3.4 kN and Ultimate load = 6.1 kN.*

*Calculate : 1. yield stress, 2. ultimate tensile stress, 3. percentage reduction in area, and 4. percentage elongation.*

**Solution.** Given : $D$ = 12 mm ; $l$ = 60 mm ; $L$ = 80 mm ; $d$ = 7 mm ; $W_y$ = 3.4 kN = 3400 N; $W_u$ = 6.1 kN = 6100 N

We know that original area of the rod,

$$A = \frac{\pi}{4} \times D^2 = \frac{\pi}{4}(12)^2 = 113 \text{ mm}^2$$

and final area of the rod,

$$a = \frac{\pi}{4} \times d^2 = \frac{\pi}{4}(7)^2 = 38.5 \text{ mm}^2$$

**1. *Yield stress***

We know that yield stress

$$= \frac{W_y}{A} = \frac{3400}{113} = 30.1 \text{ N/mm}^2 = 30.1 \text{ MPa} \quad \textbf{Ans.}$$

**2. *Ultimate tensile stress***

We know the ultimate tensile stress

$$= \frac{W_u}{A} = \frac{6100}{113} = 54 \text{ N/mm}^2 = 54 \text{ MPa} \quad \textbf{Ans.}$$

**3. *Percentage reduction in area***

We know that percentage reduction in area

$$= \frac{A - a}{A} = \frac{113 - 38.5}{113} = 0.66 \text{ or } 66\% \quad \textbf{Ans.}$$

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*4. Percentage elongation*

We know that percentage elongation

$$= \frac{L - l}{L} = \frac{80 - 60}{80} = 0.25 \text{ or } 25\% \textbf{ Ans.}$$

## 4.12 Working Stress

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

**Note :** By failure it is not meant actual breaking of the material. Some machine parts are said to fail when they have plastic deformation set in them, and they no more perform their function satisfactory.

## 4.13 Factor of Safety

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

$$\text{Factor of safety} = \frac{\text{Maximum stress}}{\text{Working or design stress}}$$

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

$$\text{Factor of safety} = \frac{\text{Yield point stress}}{\text{Working or design stress}}$$

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

$$\therefore \quad \text{Factor of safety} = \frac{\text{Ultimate stress}}{\text{Working or design stress}}$$

This relation may also be used for ductile materials.

**Note:** The above relations for factor of safety are for static loading.

## 4.14 Selection of Factor of Safety

The selection of a proper factor of safety to be used in designing any machine component depends upon a number of considerations, such as the material, mode of manufacture, type of stress, general service conditions and shape of the parts. Before selecting a proper factor of safety, a design engineer should consider the following points :

1. The reliability of the properties of the material and change of these properties during service ;
2. The reliability of test results and accuracy of application of these results to actual machine parts ;
3. The reliability of applied load ;
4. The certainty as to exact mode of failure ;
5. The extent of simplifying assumptions ;
6. The extent of localised stresses ;
7. The extent of initial stresses set up during manufacture ;
8. The extent of loss of life if failure occurs ; and
9. The extent of loss of property if failure occurs.

Each of the above factors must be carefully considered and evaluated. The high factor of safety results in unnecessary risk of failure. The values of factor of safety based on ultimate strength for different materials and type of load are given in the following table:

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Table 4.3. Values of factor of safety.

| *Material* | *Steady load* | *Live load* | *Shock load* |
|---|---|---|---|
| Cast iron | 5 to 6 | 8 to 12 | 16 to 20 |
| Wrought iron | 4 | 7 | 10 to 15 |
| Steel | 4 | 8 | 12 to 16 |
| Soft materials and alloys | 6 | 9 | 15 |
| Leather | 9 | 12 | 15 |
| Timber | 7 | 10 to 15 | 20 |

## 4.15 Stresses in Composite Bars

A composite bar may be defined as a bar made up of two or more different materials, joined together, in such a manner that the system extends or contracts as one unit, equally, when subjected to tension or compression. In case of composite bars, the following points should be kept in view:

1. The extension or contraction of the bar being equal, the strain *i.e.* deformation per unit length is also equal.
2. The total external load on the bar is equal to the sum of the loads carried by different materials.

Consider a composite bar made up of two different materials as shown in Fig. 4.13.

Let $P_1$ = Load carried by bar 1,

$A_1$ = Cross-sectional area of bar 1,

$\sigma_1$ = Stress produced in bar 1,

$E_1$ = Young's modulus of bar 1,

$P_2, A_2, \sigma_2, E_2$ = Corresponding values of bar 2,

$P$ = Total load on the composite bar,

$l$ = Length of the composite bar, and

$\delta l$ = Elongation of the composite bar.

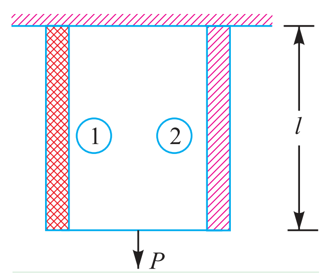


**Fig. 4.13.** Stresses in composite bars.

We know that $P = P_1 + P_2$ ...(*i*)

Stress in bar 1, $\sigma_1 = \dfrac{P_1}{A_1}$

and strain in bar 1, $\varepsilon = \dfrac{\sigma_1}{E_1} = \dfrac{P_1}{A_1 . E_1}$

∴ Elongation of bar 1,

$$\delta l_1 = \frac{P_1 . l}{A_1 . E_1}$$

Similarly, elongation of bar 2,

$$\delta l_2 = \frac{P_2 . l}{A_2 . E_2}$$

Since $\delta l_1 = \delta l_2$

*A Material handling system*

Note : This picture is given as additional information and is not a direct example of the current chapter.

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Therefore, $$\frac{P_1.l}{A_1.E_1} = \frac{P_2.l}{A_2.E_2} \quad \text{or} \quad P_1 = P_2 \times \frac{A_1.E_1}{A_2.E_2} \quad \text{...}(ii)$$

But $$P = P_1 + P_2 = P_2 \times \frac{A_1.E_1}{A_2.E_2} + P_2 = P_2\left(\frac{A_1.E_1}{A_2.E_2} + 1\right)$$

$$= P_2\left(\frac{A_1.E_1 + A_2.E_2}{A_2.E_2}\right)$$

or $$P_2 = P \times \frac{A_2.E_2}{A_1.E_1 + A_2.E_2} \quad \text{...}(iii)$$

Similarly $$P_1 = P \times \frac{A_1.E_1}{A_1.E_1 + A_2.E_2} \quad \text{...[From equation } (ii)] \quad \text{...}(iv)$$

We know that

$$\frac{P_1.l}{A_1.E_1} = \frac{P_2.l}{A_2.E_2}$$

$\therefore$ $$\frac{\sigma_1}{E_1} = \frac{\sigma_2}{E_2}$$

or $$\sigma_1 = \frac{E_1}{E_2} \times \sigma_2 \quad \text{...}(v)$$

Similarly, $$\sigma_2 = \frac{E_2}{E_1} \times \sigma_1 \quad \text{...}(vi)$$

From the above equations, we can find out the stresses produced in the different bars. We also know that

$$P = P_1 + P_2 = \sigma_1.A_1 + \sigma_2.A_2$$

From this equation, we can also find out the stresses produced in different bars.

**Note :** The ratio $E_1 / E_2$ is known as ***modular ratio*** of the two materials.

**Example 4.11.** *A bar 3 m long is made of two bars, one of copper having E = 105 GN/m² and the other of steel having E = 210 GN/m². Each bar is 25 mm broad and 12.5 mm thick. This compound bar is stretched by a load of 50 kN. Find the increase in length of the compound bar and the stress produced in the steel and copper. The length of copper as well as of steel bar is 3 m each.*

**Solution.** Given : $l_c = l_s = 3 \text{ m} = 3 \times 10^3$ mm ; $E_c = 105$ GN/m² $= 105$ kN/mm² ; $E_s = 210$ GN/m² $= 210$ kN/mm² ; $b = 25$ mm ; $t = 12.5$ mm ; $P = 50$ kN

***Increase in length of the compound bar***

Let $\delta l$ = Increase in length of the compound bar.

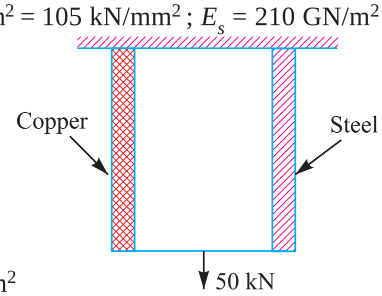


**Fig. 4.14**

The compound bar is shown in Fig. 4.14. We know that cross-sectional area of each bar,

$$A_c = A_s = b \times t = 25 \times 12.5 = 312.5 \text{ mm}^2$$

∴ Load shared by the copper bar,

$$P_c = P \times \frac{A_c \cdot E_c}{A_c \cdot E_c + A_s \cdot E_s} = P \times \frac{E_c}{E_c + E_s} \qquad \text{... } (\because A_c = A_s)$$

$$= 50 \times \frac{105}{105 + 210} = 16.67 \text{ kN}$$

and load shared by the steel bar,

$$P_s = P - P_c = 50 - 16.67 = 33.33 \text{ kN}$$

Since the elongation of both the bars is equal, therefore

$$\delta l = \frac{P_c \cdot l_c}{A_c \cdot E_c} = \frac{P_s \cdot l_s}{A_s \cdot E_s} = \frac{16.67 \times 3 \times 10^3}{312.5 \times 105} = 1.52 \text{ mm}$$ **Ans.**

***Stress produced in the steel and copper bar***

We know that stress produced in the steel bar,

$$\sigma_s = \frac{E_s}{E_c} \times \sigma_c = \frac{210}{105} \times \sigma_c = 2\,\sigma_c$$

and total load, $P = P_s + P_c = \sigma_s.A_s + \sigma_c.A_c$

∴ $50 = 2\,\sigma_c \times 312.5 + \sigma_c \times 312.5 = 937.5\,\sigma_c$

or $\sigma_c = 50 / 937.5 = 0.053 \text{ kN/mm}^2 = 53 \text{ N/mm}^2 = 53 \text{ MPa}$ **Ans.**

and $\sigma_s = 2\,\sigma_c = 2 \times 53 = 106 \text{ N/mm}^2 = 106 \text{ MPa}$ **Ans.**

**Example 4.12.** *A central steel rod 18 mm diameter passes through a copper tube 24 mm inside and 40 mm outside diameter, as shown in Fig. 4.15. It is provided with nuts and washers at each end. The nuts are tightened until a stress of 10 MPa is set up in the steel.*

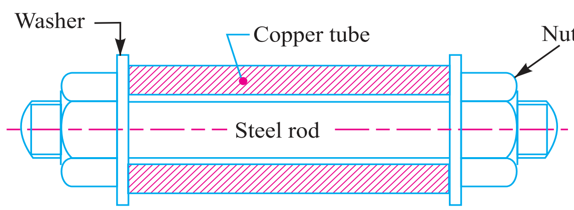


**Fig. 4.15**

*The whole assembly is then placed in a lathe and turned along half the length of the tube removing the copper to a depth of 1.5 mm. Calculate the stress now existing in the steel. Take $E_s = 2E_c$.*

**Solution.** Given : $d_s = 18$ mm ; $d_{c1} = 24$ mm ; $d_{c2} = 40$ mm ; $\sigma_s = 10$ MPa $= 10$ N/mm$^2$

We know that cross-sectional area of steel rod,

$$A_s = \frac{\pi}{4}(d_s)^2 = \frac{\pi}{4}(18)^2 = 254.5 \text{ mm}^2$$

and cross-sectional area of copper tube,

$$A_c = \frac{\pi}{4}\left[(d_{c2})^2 - (d_{c1})^2\right] = \frac{\pi}{4}\left[(40)^2 - (24)^2\right] = 804.4 \text{ mm}^2$$

We know that when the nuts are tightened on the tube, the steel rod will be under tension and the copper tube in compression.

Let $\sigma_c$ = Stress in the copper tube.

Since the tensile load on the steel rod is equal to the compressive load on the copper tube, therefore

$$\sigma_s \times A_s = \sigma_c \times A_c$$

$$10 \times 254.5 = \sigma_c \times 804.4$$

$$\therefore \quad \sigma_c = \frac{10 \times 254.5}{804.4} = 3.16 \text{ N/mm}^2$$

When the copper tube is reduced in the area for half of its length, then outside diameter of copper tube,

$$= 40 - 2 \times 1.5 = 37 \text{ mm}$$

∴ Cross-sectional area of the half length of copper tube,

$$A_{c1} = \frac{\pi}{4}(37^2 - 24^2) = 623 \text{ mm}^2$$

The cross-sectional area of the other half remains same. If $A_{c2}$ be the area of the remainder, then

$$A_{c2} = A_c = 804.4 \text{ mm}^2$$

Let $\sigma_{c1}$ = Compressive stress in the reduced section,

$\sigma_{c2}$ = Compressive stress in the remainder, and

$\sigma_{s1}$ = Stress in the rod after turning.

Since the load on the copper tube is equal to the load on the steel rod, therefore

$$A_{c1} \times \sigma_{c1} = A_{c2} \times \sigma_{c2} = A_s \times \sigma_{s1}$$

$$\therefore \quad \sigma_{c1} = \frac{A_s}{A_{c1}} \times \sigma_{s1} = \frac{254.5}{623} \times \sigma_{s1} = 0.41\, \sigma_{s1} \qquad \text{...}(i)$$

and

$$\sigma_{c2} = \frac{A_s}{A_{c2}} \times \sigma_{s1} = \frac{254.5}{804.4} \times \sigma_{s1} = 0.32\, \sigma_{s1} \qquad \text{...}(ii)$$

Let $\delta l$ = Change in length of the steel rod before and after turning,

$l$ = Length of the steel rod and copper tube between nuts,

$\delta l_1$ = Change in length of the reduced section (*i.e.* $l/2$) before and after turning, and

$\delta l_2$ = Change in length of the remainder section (*i.e.* $l/2$) before and after turning.

Since $\delta l = \delta l_1 + \delta l_2$

$$\therefore \quad \frac{\sigma_s - \sigma_{s1}}{E_s} \times l = \frac{\sigma_{c1} - \sigma_c}{E_c} \times \frac{l}{2} + \frac{\sigma_{c2} - \sigma_c}{E_c} \times \frac{l}{2}$$

or

$$\frac{10 - \sigma_{s1}}{2E_c} = \frac{0.41\, \sigma_{s1} - 3.16}{2E_c} + \frac{0.32\, \sigma_{s1} - 3.16}{2E_c} \qquad \text{...(Cancelling } l \text{ throughout)}$$

$$\therefore \quad \sigma_{s1} = 9.43 \text{ N/mm}^2 = 9.43 \text{ MPa}$$ **Ans.**

## 4.16 Stresses due to Change in Temperature—Thermal Stresses

Whenever there is some increase or decrease in the temperature of a body, it causes the body to expand or contract. A little consideration will show that if the body is allowed to expand or contract freely, with the rise or fall of the temperature, no stresses are induced in the body. But, if the deformation of the body is prevented, some stresses are induced in the body. Such stresses are known as ***thermal stresses.***

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Let $l$ = Original length of the body,

$t$ = Rise or fall of temperature, and

$\alpha$ = Coefficient of thermal expansion,

$\therefore$ Increase or decrease in length,

$$\delta l = l.\,\alpha.t$$

If the ends of the body are fixed to rigid supports, so that its expansion is prevented, then compressive strain induced in the body,

$$\varepsilon_c = \frac{\delta l}{l} = \frac{l.\alpha.t}{l} = \alpha.t$$

$\therefore$ Thermal stress, $\sigma_{th} = \varepsilon_c.E = \alpha.t.E$

**Notes : 1.** When a body is composed of two or different materials having different coefficient of thermal expansions, then due to the rise in temperature, the material with higher coefficient of thermal expansion will be subjected to compressive stress whereas the material with low coefficient of expansion will be subjected to tensile stress.

**2.** When a thin tyre is shrunk on to a wheel of diameter $D$, its internal diameter $d$ is a little less than the wheel diameter. When the tyre is heated, its circumferance $\pi\, d$ will increase to $\pi\, D$. In this condition, it is slipped on to the wheel. When it cools, it wants to return to its original circumference $\pi\, d$, but the wheel if it is assumed to be rigid, prevents it from doing so.

$$\therefore \quad \text{Strain, } \varepsilon = \frac{\pi D - \pi d}{\pi d} = \frac{D - d}{d}$$

This strain is known as ***circumferential*** or ***hoop strain.***

$\therefore$ Circumferential or hoop stress,

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

Steel tyres of a locomotive.

**Example 4.13.** *A thin steel tyre is shrunk on to a locomotive wheel of 1.2 m diameter. Find the internal diameter of the tyre if after shrinking on, the hoop stress in the tyre is 100 MPa. Assume $E$ = 200 kN/mm$^2$. Find also the least temperature to which the tyre must be heated above that of the wheel before it could be slipped on. The coefficient of linear expansion for the tyre is $6.5 \times 10^{-6}$ per °C.*

**Solution.** Given : $D$ = 1.2 m = 1200 mm ; $\sigma$ = 100 MPa = 100 N/mm$^2$ ; $E$ = 200 kN/mm$^2$ = $200 \times 10^3$ N/mm$^2$ ; $\alpha = 6.5 \times 10^{-6}$ per °C

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***Internal diameter of the tyre***

Let $d$ = Internal diameter of the tyre.

We know that hoop stress ($\sigma$),

$$100 = \frac{E\,(D-d)}{d} = \frac{200 \times 10^3\,(D-d)}{d}$$

$$\therefore \quad \frac{D-d}{d} = \frac{100}{200 \times 10^3} = \frac{1}{2 \times 10^3} \quad \text{...}(i)$$

$$\frac{D}{d} = 1 + \frac{1}{2 \times 10^3} = 1.0005$$

$$\therefore \quad d = \frac{D}{1.0005} = \frac{1200}{1.0005} = 1199.4 \text{ mm} = 1.1994 \text{ m} \quad \textbf{Ans.}$$

***Least temperature to which the tyre must be heated***

Let $t$ = Least temperature to which the tyre must be heated.

We know that

$$\pi D = \pi d + \pi d\,.\,\alpha.t = \pi d\,(1 + \alpha.t)$$

$$\alpha.t = \frac{\pi D}{\pi d} - 1 = \frac{D-d}{d} = \frac{1}{2 \times 10^3} \quad \text{...[From equation } (i)]$$

$$\therefore \quad t = \frac{1}{\alpha \times 2 \times 10^3} = \frac{1}{6.5 \times 10^{-6} \times 2 \times 10^3} = 77°\text{C} \quad \textbf{Ans.}$$

**Example 4.14.** *A composite bar made of aluminium and steel is held between the supports as shown in Fig. 4.16. The bars are stress free at a temperature of 37°C. What will be the stress in the two bars when the temperature is 20°C, if (a) the supports are unyielding; and (b) the supports yield and come nearer to each other by 0.10 mm?*

*It can be assumed that the change of temperature is uniform all along the length of the bar. Take $E_s$ = 210 GPa ; $E_a$ = 74 GPa ; $\alpha_s = 11.7 \times 10^{-6}$ / °C ; and $\alpha_a = 23.4 \times 10^{-6}$ / °C.*

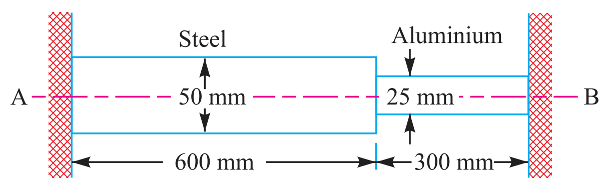


**Fig. 4.16**

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

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

$$t = t_1 - t_2 = 37 - 20 = 17°\text{C}$$

∴ Contraction in steel bar

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

and contraction in aluminium bar

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

Total contraction = 0.12 + 0.12 = 0.24 mm = 0.24 × $10^{-3}$ m

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

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Let an axial force $P$ is applied to the right end till this end is brought in contact with the right hand support at $B$, as shown in Fig. 4.17.

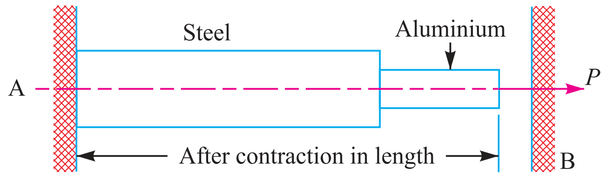


**Fig. 4.17**

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

$$A_s = \frac{\pi}{4}(d_s)^2 = \frac{\pi}{4}(0.05)^2 = 1.964 \times 10^{-3} \text{ m}^2$$

and cross-sectional area of the aluminium bar,

$$A_a = \frac{\pi}{4}(d_a)^2 = \frac{\pi}{4}(0.025)^2 = 0.491 \times 10^{-3} \text{ m}^2$$

We know that elongation of the steel bar,

$$\delta l_s = \frac{P \times l_s}{A_s \times E_s} = \frac{P \times 0.6}{1.964 \times 10^{-3} \times 210 \times 10^9} = \frac{0.6P}{412.44 \times 10^6} \text{ m}$$

$$= 1.455 \times 10^{-9} P \text{ m}$$

and elongation of the aluminium bar,

$$\delta l_a = \frac{P \times l_a}{A_a \times E_a} = \frac{P \times 0.3}{0.491 \times 10^{-3} \times 74 \times 10^9} = \frac{0.3P}{36.334 \times 10^6} \text{ m}$$

$$= 8.257 \times 10^{-9} P \text{ m}$$

∴ Total elongation, $\delta l = \delta l_s + \delta l_a$

$$= 1.455 \times 10^{-9} P + 8.257 \times 10^{-9} P = 9.712 \times 10^{-9} P \text{ m}$$

Let $\sigma_s$ = Stress in the steel bar, and

$\sigma_a$ = Stress in the aluminium bar.

***(a) When the supports are unyielding***

When the supports are unyielding, the total contraction is equated to the total elongation, *i.e.*

$$0.24 \times 10^{-3} = 9.712 \times 10^{-9} P \quad \text{or} \quad P = 24\,712 \text{ N}$$

∴ Stress in the steel bar,

$$\sigma_s = P/A_s = 24\,712 / (1.964 \times 10^{-3}) = 12\,582 \times 10^3 \text{ N/m}^2$$

$= 12.582$ MPa **Ans.**

and stress in the aluminium bar,

$$\sigma_a = P/A_a = 24\,712 / (0.491 \times 10^{-3}) = 50\,328 \times 10^3 \text{ N/m}^2$$

$= 50.328$ MPa **Ans.**

***(b) When the supports yield by 0.1 mm***

When the supports yield and come nearer to each other by 0.10 mm, the net contraction in length

$$= 0.24 - 0.1 = 0.14 \text{ mm} = 0.14 \times 10^{-3} \text{ m}$$

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Equating this net contraction to the total elongation, we have

$$0.14 \times 10^{-3} = 9.712 \times 10^{-9}\, P \quad \text{or} \quad P = 14\,415 \text{ N}$$

∴ Stress in the steel bar,

$$\sigma_s = P/A_s = 14\,415 / (1.964 \times 10^{-3}) = 7340 \times 10^3 \text{ N/m}^2$$
$$= 7.34 \text{ MPa}$$ **Ans.**

and stress in the aluminium bar,

$$\sigma_a = P/A_a = 14\,415 / (0.491 \times 10^{-3}) = 29\,360 \times 10^3 \text{ N/m}^2$$
$$= 29.36 \text{ MPa}$$ **Ans.**

**Example 4.15.** *A copper bar 50 mm in diameter is placed within a steel tube 75 mm external diameter and 50 mm internal diameter of exactly the same length. The two pieces are rigidly fixed together by two pins 18 mm in diameter, one at each end passing through the bar and tube. Calculate the stress induced in the copper bar, steel tube and pins if the temperature of the combination is raised by 50°C. Take $E_s$ = 210 GN/m$^2$ ; $E_c$ = 105 GN/m$^2$ ; $\alpha_s$ = 11.5 × 10$^{-6}$/°C and $\alpha_c$ = 17 × 10$^{-6}$/°C.*

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

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

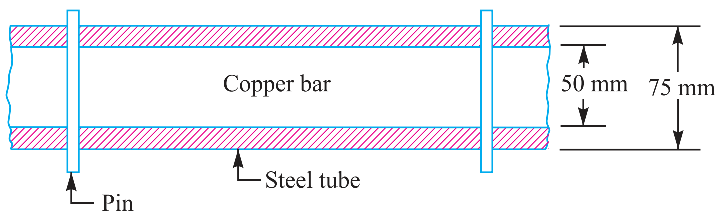


**Fig. 4.18**

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

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

and cross-sectional area of the steel tube,

$$A_s = \frac{\pi}{4}\left[(d_{se})^2 - (d_{si})^2\right] = \frac{\pi}{4}\left[(75)^2 - (50)^2\right] = 2455 \text{ mm}^2$$
$$= 2455 \times 10^{-6} \text{ m}^2$$

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

We know that free expansion of copper bar

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

and free expansion of steel tube

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

∴ Difference in free expansion

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

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Since the free expansion of the copper bar is more than the free expansion of the steel tube, therefore the copper bar is subjected to a *compressive stress, while the steel tube is subjected to a tensile stress.

Let a compressive force $P$ newton on the copper bar opposes the extra expansion of the copper bar and an equal tensile force $P$ on the steel tube pulls the steel tube so that the net effect of reduction in length of copper bar and the increase in length of steel tube equalises the difference in free expansion of the two.

Main wheels on the undercarraige of an airliner. Air plane landing gears and wheels need to bear high stresses and shocks.

Note : This picture is given as additional information and is not a direct example of the current chapter.

∴ Reduction in length of copper bar due to force $P$

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

$$= \frac{P.l}{1964 \times 10^{-6} \times 105 \times 10^{9}} = \frac{P.l}{206.22 \times 10^{6}} \text{ m}$$

and increase in length of steel bar due to force $P$

$$= \frac{P.l}{A_s . E_s} = \frac{P.l}{2455 \times 10^{-6} \times 210 \times 10^{9}} = \frac{P.l}{515.55 \times 10^{6}} \text{ m}$$

$$\therefore \text{Net effect in length} = \frac{P.l}{206.22 \times 10^{6}} + \frac{P.l}{515.55 \times 10^{6}}$$

$$= 4.85 \times 10^{-9} P.l + 1.94 \times 10^{-9} P.l = 6.79 \times 10^{-9} P.l$$

Equating this net effect in length to the difference in free expansion, we have

$$6.79 \times 10^{-9} P.l = 275 \times 10^{-6}\, l \quad \text{or} \quad P = 40\,500 \text{ N}$$

***Stress induced in the copper bar, steel tube and pins***

We know that stress induced in the copper bar,

$$\sigma_c = P / A_c = 40\,500 / (1964 \times 10^{-6}) = 20.62 \times 10^{6} \text{ N/m}^2 = 20.62 \text{ MPa } \textbf{Ans.}$$

Stress induced in the steel tube,

$$\sigma_s = P / A_s = 40\,500 / (2455 \times 10^{-6}) = 16.5 \times 10^{6} \text{ N / m}^2 = 16.5 \text{ MPa } \textbf{Ans.}$$

* In other words, we can also say that since the coefficient of thermal expansion for copper ($\alpha_c$) is more than the coefficient of thermal expansion for steel ($\alpha_s$), therefore the copper bar will be subjected to compressive stress and the steel tube will be subjected to tensile stress.

and shear stress induced in the pins,

$$\tau_p = \frac{P}{2\,A_p} = \frac{40\,500}{2 \times \frac{\pi}{4}(0.018)^2} = 79.57 \times 10^6 \text{ N/m}^2 = 79.57 \text{ MPa} \quad \textbf{Ans.}$$

...( $\because$ The pin is in double shear )

## 4.17 Linear and Lateral Strain

Consider a circular bar of diameter *d* and length *l*, subjected to a tensile force *P* as shown in Fig. 4.19 (*a*).

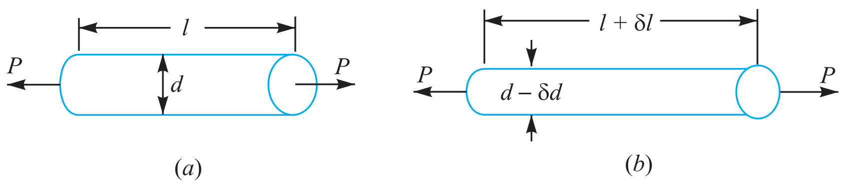


**Fig. 4.19.** Linear and lateral strain.

A little consideration will show that due to tensile force, the length of the bar increases by an amount $\delta l$ and the diameter decreases by an amount $\delta d$, as shown in Fig. 4.19 (*b*). Similarly, if the bar is subjected to a compressive force, the length of bar will decrease which will be followed by increase in diameter.

It is thus obvious, that every direct stress is accompanied by a strain in its own direction which is known as ***linear strain*** and an opposite kind of strain in every direction, at right angles to it, is known as ***lateral strain.***

## 4.18 Poisson's Ratio

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

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

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

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

**Table 4.4. Values of Poisson's ratio for commonly used materials.**

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

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## 4.19 Volumetric Strain

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

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

where $\delta V$ = Change in volume, and $V$ = Original volume.

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

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

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

$$\varepsilon_v = \varepsilon_x + \varepsilon_y + \varepsilon_z$$

where $\varepsilon_x$, $\varepsilon_y$ and $\varepsilon_z$ are the strains in the directions *x*-axis, *y*-axis and *z*-axis respectively.

## 4.20 Bulk Modulus

When a body is subjected to three mutually perpendicular stresses, of equal intensity, then the ratio of the direct stress to the corresponding volumetric strain is known as ***bulk modulus.*** It is usually denoted by *K*. Mathematically, bulk modulus,

$$K = \frac{\text{Direct stress}}{\text{Volumetric strain}} = \frac{\sigma}{\delta V / V}$$

## 4.21 Relation Between Bulk Modulus and Young's Modulus

The bulk modulus (*K*) and Young's modulus (*E*) are related by the following relation,

$$K = \frac{m.E}{3\,(m - 2)} = \frac{E}{3\,(1 - 2\mu)}$$

## 4.22 Relation Between Young's Modulus and Modulus of Rigidity

The Young's modulus (*E*) and modulus of rigidity (*G*) are related by the following relation,

$$G = \frac{m.E}{2\,(m + 1)} = \frac{E}{2\,(1 + \mu)}$$

**Example 4.16.** *A mild steel rod supports a tensile load of 50 kN. If the stress in the rod is limited to 100 MPa, find the size of the rod when the cross-section is 1. circular, 2. square, and 3. rectangular with width = 3 × thickness.*

**Solution.** Given : $P$ = 50 kN = $50 \times 10^3$ N ; $\sigma_t$ = 100 MPa = 100 N/mm$^2$

**1.** ***Size of the rod when it is circular***

Let $d$ = Diameter of the rod in mm.

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

We know that tensile load ($P$),

$$50 \times 10^3 = \sigma_t \times A = 100 \times 0.7854\, d^2 = 78.54\, d^2$$

$\therefore$ $d^2 = 50 \times 10^3 / 78.54 = 636.6$ or $d = 25.23$ mm **Ans.**

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*2. Size of the rod when it is square*

Let $x$ = Each side of the square rod in mm.

$\therefore$ Area, $A = x \times x = x^2$

We know that tensile load ($P$),

$$50 \times 10^3 = \sigma_t \times A = 100 \times x^2$$

$$\therefore \quad x^2 = 50 \times 10^3/100 = 500 \quad \text{or} \quad x = 22.4 \text{ mm } \textbf{Ans.}$$

*3. Size of the rod when it is rectangular*

Let $t$ = Thickness of the rod in mm, and

$b$ = Width of the rod in mm = $3\,t$ ...(Given)

$\therefore$ Area, $A = b \times t = 3t \times t = 3\,t^2$

We know that tensile load ($P$),

$$50 \times 10^3 = \sigma_t \times A = 100 \times 3\,t^2 = 300\,t^2$$

$$\therefore \quad t^2 = 50 \times 10^3 / 300 = 166.7 \quad \text{or} \quad t = 12.9 \text{ mm } \textbf{Ans.}$$

and $b = 3t = 3 \times 12.9 = 38.7$ mm **Ans.**

**Example 4.17.** *A steel bar 2.4 m long and 30 mm square is elongated by a load of 500 kN. If poisson's ratio is 0.25, find the increase in volume. Take E = 0.2 × $10^6$ N/mm$^2$.*

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

Let $\delta V$ = Increase in volume.

We know that volume of the rod,

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

and Young's modulus,

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

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

We know that volumetric strain,

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

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

## 4.23 Impact Stress

Sometimes, machine members are subjected to the load with impact. The stress produced in the member due to the falling load is known as ***impact stress.***

Consider a bar carrying a load $W$ at a height $h$ and falling on the collar provided at the lower end, as shown in Fig. 4.20.

Let $A$ = Cross-sectional area of the bar,

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

$l$ = Length of the bar,

$\delta l$ = Deformation of the bar,

$P$ = Force at which the deflection $\delta l$ is produced,

$\sigma_i$ = Stress induced in the bar due to the application of impact load, and

$h$ = Height through which the load falls.

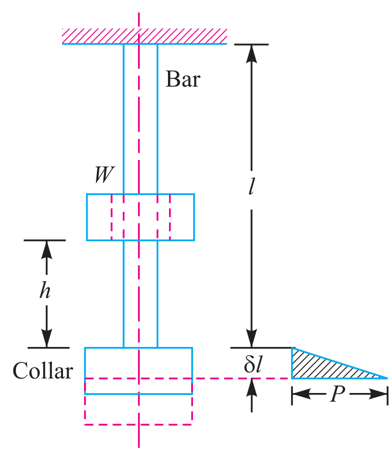


**Fig. 4.20.** Impact stress.

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

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

and potential energy lost by the weight

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

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

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

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

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

From this quadratic equation, we find that

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

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

**Example 4.18.** *An unknown weight falls through 10 mm on a collar rigidly attached to the lower end of a vertical bar 3 m long and 600 mm² in section. If the maximum instantaneous extension is known to be 2 mm, what is the corresponding stress and the value of unknown weight? Take $E$ = 200 kN/mm².*

**Solution.** Given : $h$ = 10 mm ; $l$ = 3 m = 3000 mm ; $A$ = 600 mm² ; $\delta l$ = 2 mm ; $E$ = 200 kN/mm² = 200 × 10³ N/mm²

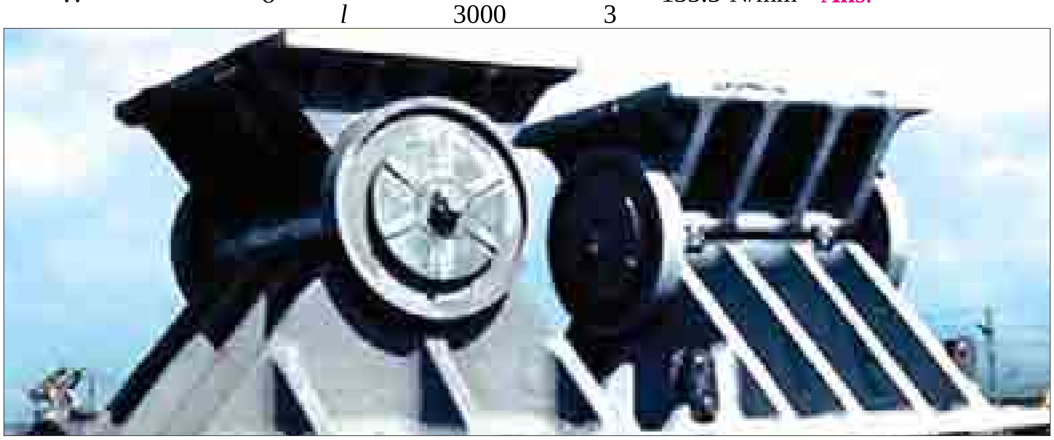
***Stress in the bar***

Let $\sigma$ = Stress in the bar.

We know that Young's modulus,

$$E = \frac{\text{Stress}}{\text{Strain}} = \frac{\sigma}{\varepsilon} = \frac{\sigma . l}{\delta l}$$

$$\therefore \quad \sigma = \frac{E . \delta l}{l} = \frac{200 \times 10^3 \times 2}{3000} = \frac{400}{3} = 133.3 \text{ N/mm}^2 \quad \textbf{Ans.}$$



*These bridge shoes are made to bear high compressive stresses.*

Note : This picture is given as additional information and is not a direct example of the current chapter.

*Value of the unknown weight*

Let $W$ = Value of the unknown weight.

We know that
$$\sigma = \frac{W}{A}\left[1 + \sqrt{1 + \frac{2hAE}{Wl}}\right]$$

$$\frac{400}{3} = \frac{W}{600}\left[1 + \sqrt{1 + \frac{2 \times 10 \times 600 \times 200 \times 10^3}{W \times 3000}}\right]$$

$$\frac{400 \times 600}{3\,W} = 1 + \sqrt{1 + \frac{800\,000}{W}}$$

$$\frac{80\,000}{W} - 1 = \sqrt{1 + \frac{800\,000}{W}}$$

Squaring both sides,

$$\frac{6400 \times 10^6}{W^2} + 1 - \frac{160\,000}{W} = 1 + \frac{800\,000}{W}$$

$$\frac{6400 \times 10^2}{W} - 16 = 80 \quad \text{or} \quad \frac{6400 \times 10^2}{W} = 96$$

∴ $W = 6400 \times 10^2 / 96 = 6666.7$ N **Ans.**

## 4.24 Resilience

When a body is loaded within elastic limit, it changes its dimensions and on the removal of the load, it regains its original dimensions. So long as it remains loaded, it has stored energy in itself. On removing the load, the energy stored is given off as in the case of a spring. This energy, which is absorbed in a body when strained within elastic limit, is known as ***strain energy***. The strain energy is always capable of doing some work.

The strain energy stored in a body due to external loading, within elastic limit, is known as ***resilience*** and the maximum energy which can be stored in a body up to the elastic limit is called ***proof resilience***. The proof resilience per unit volume of a material is known as ***modulus of resilience***. It is an important property of a material and gives capacity of the material to bear impact or shocks. Mathematically, strain energy stored in a body due to tensile or compressive load or resilience,

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

and Modulus of resilience $= \dfrac{\sigma^2}{2E}$

where $\sigma$ = Tensile or compressive stress,

$V$ = Volume of the body, and

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

**Notes : 1.** When a body is subjected to a shear load, then modulus of resilience (shear)

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

where $\tau$ = Shear stress, and

$C$ = Modulus of rigidity.

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

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

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**Example 4.19.** *A wrought iron bar 50 mm in diameter and 2.5 m long transmits a shock energy of 100 N-m. Find the maximum instantaneous stress and the elongation. Take E = 200 GN/m².*

**Solution.** Given : $d$ = 50 mm ; $l$ = 2.5 m = 2500 mm ; $U$ = 100 N-m = 100 × $10^3$ N-mm ; $E$ = 200 GN/m² = 200 × $10^3$ N/mm²

***Maximum instantaneous stress***

Let $\sigma$ = Maximum instantaneous stress.

We know that volume of the bar,

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

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

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

$\therefore$ $\sigma^2 = 100 \times 10^3 / 12.25 = 8163$ or $\sigma = 90.3$ N/mm² **Ans.**

***Elongation produced***

Let $\delta l$ = Elongation produced.

We know that Young's modulus,

$$E = \frac{\text{Stress}}{\text{Strain}} = \frac{\sigma}{\varepsilon} = \frac{\sigma}{\delta l / l}$$

$$\therefore \quad \delta l = \frac{\sigma \times l}{E} = \frac{90.3 \times 2500}{200 \times 10^3} = 1.13 \text{ mm} \quad \textbf{Ans.}$$

*A double-decker train.*

## EXERCISES

1. A reciprocating steam engine connecting rod is subjected to a maximum load of 65 kN. Find the diameter of the connecting rod at its thinnest part, if the permissible tensile stress is 35 N/mm². **[Ans. 50 mm]**

2. The maximum tension in the lower link of a Porter governor is 580 N and the maximum stress in the link is 30 N/mm². If the link is of circular cross-section, determine its diameter. **[Ans. 5 mm]**

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3. A wrought iron rod is under a compressive load of 350 kN. If the permissible stress for the material is 52.5 N/mm$^2$, calculate the diameter of the rod. **[Ans. 95 mm]**
4. A load of 5 kN is to be raised by means of a steel wire. Find the minimum diameter required, if the stress in the wire is not to exceed 100 N/mm$^2$. **[Ans. 8 mm]**
5. A square tie bar 20 mm × 20 mm in section carries a load. It is attached to a bracket by means of 6 bolts. Calculate the diameter of the bolt if the maximum stress in the tie bar is 150 N/mm$^2$ and in the bolts is 75 N/mm$^2$. **[Ans. 13 mm]**
6. The diameter of a piston of the steam engine is 300 mm and the maximum steam pressure is 0.7 N/mm$^2$. If the maximum permissible compressive stress for the piston rod material is 40 N/mm$^2$, find the size of the piston rod. **[Ans. 40 mm]**
7. Two circular rods of 50 mm diameter are connected by a knuckle joint, as shown in Fig. 4.21, by a pin of 40 mm in diameter. If a pull of 120 kN acts at each end, find the tensile stress in the rod and shear stress in the pin. **[Ans. 61 N/mm$^2$; 48 N/mm$^2$]**

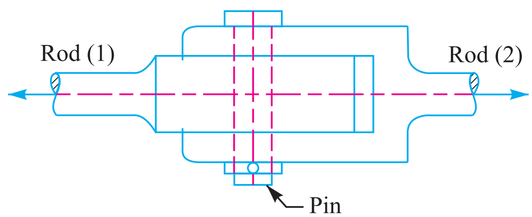


**Fig. 4.21**

8. Find the minimum size of a hole that can be punched in a 20 mm thick mild steel plate having an ultimate shear strength of 300 N/mm$^2$. The maximum permissible compressive stress in the punch material is 1200 N/mm$^2$. **[Ans. 20 mm]**
9. The crankpin of an engine sustains a maximum load of 35 kN due to steam pressure. If the allowable bearing pressure is 7 N/mm$^2$, find the dimensions of the pin. Assume the length of the pin equal to 1.2 times the diameter of the pin. **[Ans. 64.5 mm; 80 mm]**
10. The following results were obtained in a tensile test on a mild steel specimen of original diameter 20 mm and gauge length 40 mm.

| | | |
|---|---|---|
| Load at limit of proportionality | = | 80 kN |
| Extension at 80 kN load | = | 0.048 mm |
| Load at yield point | = | 85 kN |
| Maximum load | = | 150 kN |

When the two parts were fitted together after being broken, the length between gauge length was found to be 55.6 mm and the diameter at the neck was 15.8 mm.

Calculate Young's modulus, yield stress, ultimate tensile stress, percentage elongation and percentage reduction in area. **[Ans. 213 kN/mm$^2$; 270 N/mm$^2$; 478 N/mm$^2$; 39%; 38%]**

11. A steel rod of 25 mm diameter is fitted inside a brass tube of 25 mm internal diameter and 375 mm external diameter. The projecting ends of the steel rod are provided with nuts and washers. The nuts are tightened up so as to produce a pull of 5 kN in the rod. The compound is then placed in a lathe and the brass is turned down to 4 mm thickness. Calculate the stresses in the two materials. **[Ans. 7 N/mm$^2$, 7.8 N/mm$^2$]**

**12.** A composite bar made up of aluminium bar and steel bar, is firmly held between two unyielding supports as shown in Fig. 4.22.

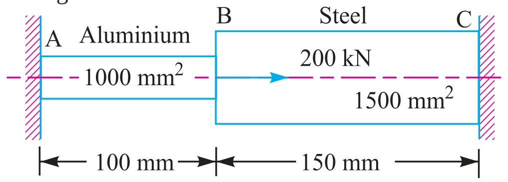


**Fig. 4.22**

An axial load of 200 kN is applied at *B* at 47°C. Find the stresses in each material, when the temperature is 97°C. Take $E_a$ = 70 GPa ; $E_s$ = 210 GPa ; $\alpha_a = 24 \times 10^{-6}$/°C and $\alpha_s = 12 \times 10^{6}$/°C.

**[Ans. 60.3 MPa; 173.5 MPa]**

**13.** A steel rod of 20 mm diameter passes centrally through a copper tube of external diameter 40 mm and internal diameter 20 mm. The tube is closed at each end with the help of rigid washers (of negligible thickness) which are screwed by the nuts. The nuts are tightened until the compressive load on the copper tube is 50 kN. Determine the stresses in the rod and the tube, when the temperature of whole assembly falls by 50°C. Take $E_s$ = 200 GPa ; $E_c$ = 100 GPa ; $\alpha_s = 12 \times 10^{-6}$/°C and $\alpha_c = 18 \times 10^{6}$/°C.

**[Ans. 99.6 MPa; 19.8 MPa]**

**14.** A bar of 2 m length, 20 mm breadth and 15 mm thickness is subjected to a tensile load of 30 kN. Find the final volume of the bar, if the Poisson's ratio is 0.25 and Young's modulus is 200 GN/m$^2$.

**[Ans. 600 150 mm$^3$]**

**15.** A bar of 12 mm diameter gets stretched by 3 mm under a steady load of 8 kN. What stress would be produced in the bar by a weight of 800 N, which falls through 80 mm before commencing the stretching of the rod, which is initially unstressed. Take $E$ = 200 kN/mm$^2$.

**[Ans. 170.6 N/mm$^2$]**

## QUESTIONS

1. Define the terms load , stress and strain. Discuss the various types of stresses and strain.
2. What is the difference between modulus of elasticity and modulus of rigidity?
3. Explain clearly the bearing stress developed at the area of contact between two members.
4. What useful informations are obtained from the tensile test of a ductile material?
5. What do you mean by factor of safety?
6. List the important factors that influence the magnitude of factor of safety.
7. What is meant by working stress and how it is calculated from the ultimate stress or yield stress of a material? What will be the factor of safety in each case for different types of loading?
8. Describe the procedure for finding out the stresses in a composite bar.
9. Explain the difference between linear and lateral strain.
10. Define the following :
    (*a*) Poisson's ratio, (*b*) Volumetric strain, and (*c*) Bulk modulus
11. Derive an expression for the impact stress induced due to a falling load.
12. Write short notes on :
    (*a*) Resilience (*b*) Proof resilience, and (*c*) Modulus of resilience

## OBJECTIVE TYPE QUESTIONS

1. Hooke's law holds good upto
   (*a*) yield point (*b*) elastic limit
   (*c*) plastic limit (*d*) breaking point
2. The ratio of linear stress to linear strain is called
   (*a*) Modulus of elasticity (*b*) Modulus of rigidity
   (*c*) Bulk modulus (*d*) Poisson's ratio

3. The modulus of elasticity for mild steel is approximately equal to
   (*a*) 80 kN/mm$^2$ (*b*) 100 kN/mm$^2$
   (*c*) 110 kN/mm$^2$ (*d*) 210 kN/mm$^2$
4. When the material is loaded within elastic limit, then the stress is ......... to strain.
   (*a*) equal (*b*) directly proportional (*c*) inversely proportional
5. When a hole of diameter '*d*' is punched in a metal of thickness '*t*', then the force required to punch a hole is equal to
   (*a*) $d.t.\tau_u$ (*b*) $\pi\, d.t.\tau_u$
   (*c*) $\frac{\pi}{4} \times d^2\ \tau_u$ (*d*) $\frac{\pi}{4} \times d^2.t.\tau_u$

   where $\tau_u$ = Ultimate shear strength of the material of the plate.
6. The ratio of the ultimate stress to the design stress is known as
   (*a*) elastic limit (*b*) strain
   (*c*) factor of safety (*d*) bulk modulus
7. The factor of safety for steel and for steady load is
   (*a*) 2 (*b*) 4
   (*c*) 6 (*d*) 8
8. An aluminium member is designed based on
   (*a*) yield stress (*b*) elastic limit stress
   (*c*) proof stress (*d*) ultimate stress
9. In a body, a thermal stress is one which arises because of the existence of
   (*a*) latent heat (*b*) temperature gradient
   (*c*) total heat (*d*) specific heat
10. A localised compressive stress at the area of contact between two members is known as
    (*a*) tensile stress (*b*) bending stress
    (*c*) bearing stress (*d*) shear stress
11. The Poisson's ratio for steel varies from
    (*a*) 0.21 to 0.25 (*b*) 0.25 to 0.33
    (*c*) 0.33 to 0.38 (*d*) 0.38 to 0.45
12. The stress in the bar when load is applied suddenly is ............. as compared to the stress induced due to gradually applied load.
    (*a*) same (*b*) double
    (*c*) three times (*d*) four times
13. The energy stored in a body when strained within elastic limit is known as
    (*a*) resilience (*b*) proof resilience
    (*c*) strain energy (*d*) impact energy
14. The maximum energy that can be stored in a body due to external loading upto the elastic limit is called
    (*a*) resilience (*b*) proof resilience
    (*c*) strain energy (*d*) modulus of resilience
15. The strain energy stored in a body, when suddenly loaded, is .............. the strain energy stored when same load is applied gradually.
    (*a*) equal to (*b*) one-half
    (*c*) twice (*d*) four times

## ANSWERS

| | | | | |
|---|---|---|---|---|
| **1.** (*b*) | **2.** (*a*) | **3.** (*d*) | **4.** (*b*) | **5.** (*b*) |
| **6.** (*c*) | **7.** (*b*) | **8.** (*a*) | **9.** (*b*) | **10.** (*c*) |
| **11.** (*b*) | **12.** (*b*) | **13.** (*c*) | **14.** (*b*) | **15.** (*d*) |

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