MODULE 1

DEFINITION OF DESIGN:

"Design is a process of formulating a plan for achieving a product."

Mechanical engineering design involves the application of the principles of science and technology in the creation of product and the consideration of various factors that affects its production and use The basic purpose of machine is to transmit or modify the available energy from the source.

To produce this effect machine requires the inclusion of a certain series of links and mechanism.

The study of geometrical arrangement of elements, the analysis of the forces involved and subsequent design of elements so that they will not fail in operation is the subject matter of design of machine elements.

Thus, machine design or mechanical engineering design may be defined as the practical application of mechanics of machinery (mechanics which is involved in construction and operation of machinery) to the design and construction of machines .

The machine design may be classified as:

1. **Adaptive design** : Adaptive design is the type of design in which designer adapt existing design, and designer only makes miner changes in the existing design of a product. It does not requires much skills and knowledge.

2. **Development design**: In this type of design, the designer modifies and improves the existing design. The final design of a product differ markedly from the original design. It requires considerable skills and knowledge to modify and improve the existing design.

3. **New design :** This design needs technical ability and creative thinking. New design may be further classified as .

a) Rational(analytical) design : is based on the application of principles of mechanics. It involves the analysis of forces and formulation of mathematical equations. It requires creative thinking, research abilities and technical skills

(b) Empirical design- depends on empirical formulae based on the practice and past experience.

(c) Industrial design-based on the production aspects to manufacture a designed component.

(d) Optimum design. It is the best design achieved by minimising undesirable effects.

(e) System design: design of any mechanical system comprising a large number of elements.

(f)Element design : design of individual components like piston, connecting rod etc.

(g) Computer aided design. This involves the use of computers in the design processes.

DESIGN FACTORS:

The following factors are to be considered while designing a machine element.

1. Load
2. Selection of materials
3. Shape, size and quantity
4.Durability and Reliability
5.Life
6. Mechanism
7 Manufacturing processes
8. Effect of environment
9. Cost
10. Safety

DESIGN STRESS AND WORKING STRESS:

The Design stress : is the stress value which is used in mathematical calculation of the required size of the machine member.

Selection of proper value for design stress is important in creating (designing) a new component, ana a suitable value is arrived by evaluating the strength under service conditions.

The working stress : as distinguished from the design stress, is the stress actually occurring under operating conditions. The working load per unit area of cross-section of the part gives the working stress.

FACTOR OF SAFETY :

In order to determine the size of component, the maximum load that s expected to carry must be known. To prevent failure of a machine part, the design stress or allowable stress, at all the times, must be below the elastic limit and the working stress (stress induced in actually loading) should not exceed allowable or permissible stress.

Factor of safety : It is the ratio of ultimate stress to the design Stress

Factor of Safety,

FS=(Ultimate stress/
Design (allowable) stress)

it is a measure of how much is the component is safe in its working .

DESIGN PROCEDURES

The procedure for designing a machine element usually involve the following steps

(1) State the problem:

Make a complete statement of the problem in the form of data which indicates the nature of problem and purpose of design

(2) Study the mechanism (synthesis)

Select suitable mechanism or group of mechanisms to Give desired motion

(3)Analyse the problem

Study the energy transmitted by elements and the forces acting on each member The results of analysis of forces should be presented in the form of sketches showing the magnitude direction and point of application.

(4) Select the mechanism and material :

Select the best possible mechanism which will give the desired motion with minimum friction and wear Select the suitable material considering the part size, load and manufacturing facilities.

(5) Determine the size of part:

Select the proper value ef factor of safety to find the allowable stress. The selection is based on material, nature of load and working conditions:

Determine the size of part so as to perform its function satisfactorily

(6) Modify the design:

Revise the size to agree with previous experience and the recommendations of the various committees. The revision is based on manufacture, functional and assembly requirements

(7) Make final drawings :

Transfer the design ideas in the form of the drawings and include complete specification of the product

The design steps are illustrated in Fig. 1.9.

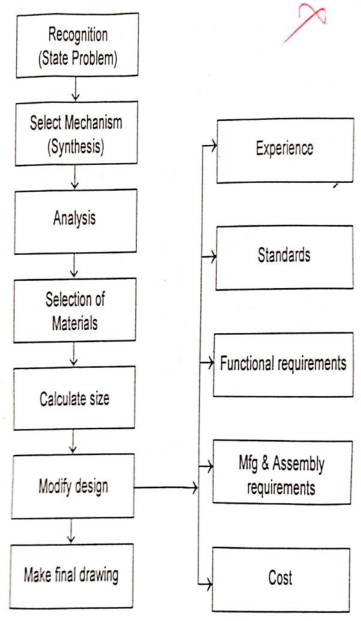


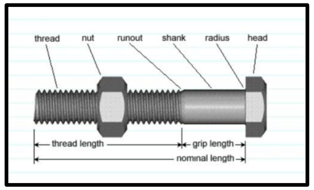
Fig. 1.9

DEFINITIONS: BOLT & NUT

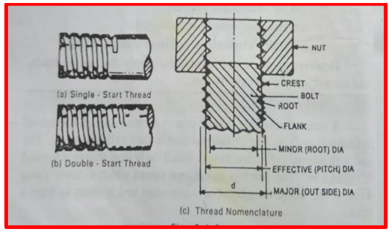
A screw thread is a helical groove formed around a external cylindrical surface. The cylindrical part is then called a **screw**.

The part with internal cylindrical hole in which the helical groove is formed is called a **nut**.

The **bolt** has an external thread which extends along a part of the shank from one end while other end is upset to form a head.



Thread Nomenclature



The major diameter (d) is the largest diameter and determines the nominal size of the thread. The **minor diameter** or root diameter or core diameter (dc)is the smallest diameter of the thread.

For V-threads

$$dc = d - 1.732\,p$$

For square threads

$$dc = d - p$$

where, p = pitch of thread up

The effective diameter or pitch diameter is the diameter where the width of the tooth is equal to the space between successive teeth.

The pitch (p) is the distance from one point on a screw thread to the corresponding point on the next thread measured parallel to the axis. It is reciprocal to number of threads per unit length of screw

Hence pitch, $p=1/T$

where T-Number of threads per unit length.

The **lead** is the distance advanced by a nut on a bolt in one revolution. For a single start thread the lead will be equal to the pitch ,

$$L = p$$

where, L = ***lead*** in mm pitch in mm
p = pitch in mm

With a double start thread, the lead is equal to twice the pitch:

$$L=2p$$

In general, for 'n start thread, the lead is equal to 'n' times the pitch

$$L=np$$

Crest is the top surface joining the two sides of threads.

Flank is the surface joining the crest and root.

Root is the bottom surface joining the two sides of adjacent threads

Thread angle is the included angle between the banks of adjacent threads

Depth is the perpendicular distance between the root and crest of the thread.

DESIGNATION OF I.S THREADS:

The Indian Standard screw threads are designated by the letter M followed by the nominal diameter and pitch, separated by the sign X

A fine screw shall be designated as:

M 8 X 1.25

M → Symbol for metric thread
8 → Nominal diameter, mm
1.25→ Pitch, mm

BOLTS OF UNIFORM STRENGTH

In an ordinary bolt, the effect of impulsive forces is concentrated at root area which is weakest part of the bolt. These weak sections being of small lengths tend to fracture, when the stress is high.

Bolts used as fastenings of connecting rod ends frequently fail by fracturing across the threads. Such bolts are made of uniform strength by
1.**Reducing the area of shank equal to the root area of the screw**. **(Fig b)**

With this modification, a greater force is permissible due to uniform distribution of stress and thus, the bolt becomes stronger and lighter.

The area of the shank is reduced to root area by
2**. Drilling a hole down to the threaded end from the bolt head.(Fig a)**

ig. 2.5 shows the various forms of bolts of uniform strength.

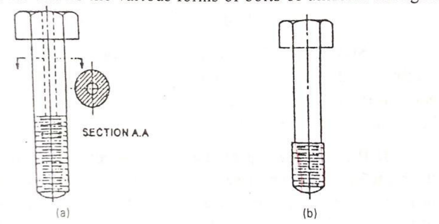


Fig. 2.5 UNIFORM STRENGTH BOLTS

d = Major diameter of thread ,mm
d_c = Minor diameter of thread, mm
d_h = diameter of drilled hole,

$$\boxed{d_h = \sqrt{d^2 - d_c^2}}$$

STRESSES IN SCREW FASTENERS

The following stresses are induced in screw fasteners with a static loading.

1. Initial stresses induced by tightening the nut or screw,

2. Stresses induced by external loading

3. Stresses due to combination of tightening and external loading

1. **Stresses due to initialTightening** :

The following stresses are induced in a screw fastener when it is tightened

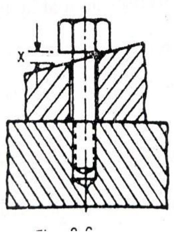
(a)The tensile stress due to the stretching of bolt. The tensile force due to initial tightening of a screw or nut is given by

Load , **$P = 2840\ d$**

P – Tensile load on screw thread, Newton
d - Major diameter, mm

b) **Bending stress:** If the surface of component is not perpendicular to the axis of bolt (Fig), then the bolt will be subjected to bending and the bending stress is given by the relation,

$$\sigma_b = \frac{x.E}{2l}$$



X= Difference in height between the extreme corners of the nut ,mm

E = Modulus of elasticity of bolt material.

l =Length of bolt, mm

2. **Stresses due to External Forces:**

The following stresses are induced in a bolt due to external load,

(a) Tensile stress
(b) Shear stress,
(c) Combined tensile and shear stresses

(a) **Tensile stress** is induced in a bolt due to the force tending to separate the parts in the direction of bolt axis. If 'n' bolts resist the tensile load ,

External load (P)

$$d_c = \sqrt{\frac{4P}{\pi\sigma_t.n}}$$

dc = Minor diameter, mm
P = Tensile load , N
σt = Tensile stress ,N/mm2
n = No : of Bolts

The nominal diameter may be obtained from standard table corresponding to d or by using the equation,

$d_c = 0.84\ d$, where d = Major diameter,mm

(b) **Shear stress** : is induced in a bolt which tends to prevent the relative motion between the parts in the direction perpendicular to bolt axis. Under these conditions, the shear load acts at cross-section corresponds to **nominal diameter**. If 'n' bolts resists shear load, Ps. Then

$$d = \sqrt{\frac{4\ P_s}{\pi\tau.n}}$$

d = nominal diameter, mm
Ps = shear load , N
τ = shear stress ,N/mm2
n = No : of Bolts

3. **Combined tension and shear stresses** are induced in a bolt when the load acts at an angle to the axis. Under such conditions, the diameter or bolt is obtained separately for tensile and shear load. A diameter slightly larger than that required may be adopted and checked for the following principal stresses.

The max. principal shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{\sigma_t^2 + 4\tau^2}, \quad \text{and}$$

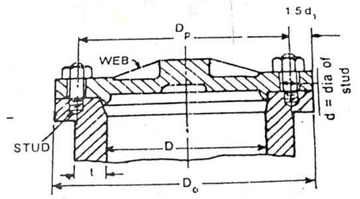
The max. principal tensile (normal) stress

$$\tau_{t,max} = \frac{1}{2}\left\{\sigma_t + \sqrt{\sigma_t^2 + 4\tau^2},\right\}$$

Joints for cylinder covers

The cylinder covers are secured with the help of bolts or studs. Studs are preferred to bolts. The use of studs prevent stripping of the thread and provides enough frictional resistance against turning when the nut is unscrewed.

Moreover, the studs are particularly convenient for positioning the covers. The possible arrangement with bolts (Fig) and studs



The size of bolts may be obtained by :

Equating **: " Pressure force act on the cylinder cover = Resistance offered by the bolts"**

$$\frac{\pi D^2}{4}.p = \frac{\pi d_c^2}{4}.\sigma_t.n$$

D = diameter of the cylinder,mm
P = Pressure inside the cylinder, N/mm2
dc = Minor diameter of studs or bolts ,mm
σt = Tensile stress ,N/mm2
n = No : of Bolts or studs

STRESSES DUE TO COMBINED LOAD :

Bolted joints with gasket and without gasket (i.e. there is metal to metal contact) are shown in Fig. (a) and (b) respectively.

- The purpose of gasket is to provide a leak proof joint.
- In ground joints the surfaces are very precisely machined to prevent leakage,
- consequently these joints are quite costly and adopted for very high pressures.

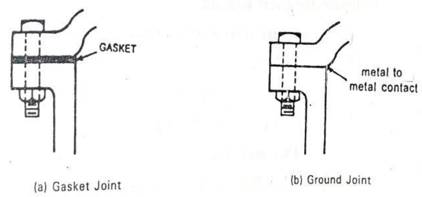


Fig. 2.13 TYPES OF SCREW JOINT

Resultant load due to initial tightening & External loading is given by,

$P = P_1 + KP_2$

P_1 = Initial load due to tightening ,N
P_2 =External load due to tightening, N
K = Stiffness constant

K= $\left[\frac{a}{1+a}\right]$; a =Ratio of elasticity = $\frac{Kb}{Kc}$

Kb = stiffness of bolt material
Kc =Stiffness of connected material

(i) When no gasket is used as in ground joints, the value of k = 0;

(ii) For gasket joints, the values of k are shown in Table

Type of joints	k
Metal to metal joint	0.00
Lead gasket with studs	0.1
Hard copper gasket with long through bolts	0.25
Soft copper gasket with long through bolts	0.5
Soft packing with through bolts	0.9
Soft packing with studs	1.0

DESIGN OF NUT

Stripping the screw thread in a nut is a common case of failure, and suitable thickness/height of a nut or number of threads in engagement may be obtained by considering the shear resistance at the bottom of threads (i.e., at major diameter for nut, and root diameter of bolt).

- In actual practice,**The height of the nut is equal to the diameter of screw.** However, **if the nut is made of weaker material than bolt, then the height of the nut is greater than the diameter of the screw** .
- **For Gun metal nut**, the height should be equal to 1.5 d.
- **For Cast iron nut**, the height of the nut is 2 d.

Proportions for Hexagonal Nut :

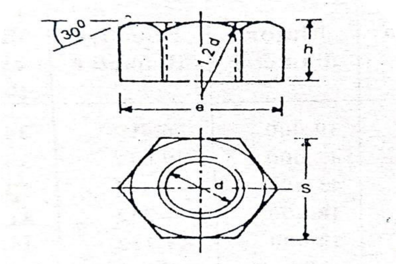


Fig. 2.15 HEXAGONAL NUT

The hexagonal nut is shown in Fig. 2.13. The sizes of nut in terms of nominal diameter, d are;

1. Size across flat, S = 1.25 d + 6 mm
2. Size across corner, e = 1.155 x S
3. Height of the nut, h = 0.8 d to d
4. Chamfer angle, = 30°

Standard size of screw thread

M 1.6 ,1.8 , 2 , 2.2, 2.5,3,3.5,4,4.5,5,6,7,8,9,10,11,12,14,16,18,20, 22,24,27,30,33,36,39,42,45,48,52,56,60,64,68,

KEYS

"A key is machine element used to prevent relative motion between parts such as gears and shafts, In other words, it is a device to transmit torque from shat to mating gear or wheel or from gear or pulley to shaft."

- Key is inserted between a shaft and hub in the recess made parallel to the axis of shaft.
- The recess in a shaft or hub to accommodate key is **called key way.**
- Most keys are made of steel and large number of types is available.
- The selection of key depends on power requirements, nature of fit, stability and cost

TYPES OF KEYS:

Keys are generally classified into two types.

1. **Sunk keys** : which require key way in both the shaft and hub

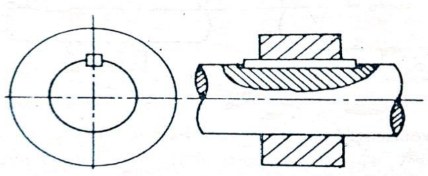
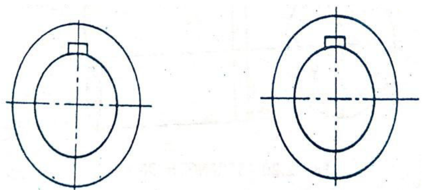


Fig. 2.18 (a) Square Key

2. **Saddle keys** : which require a key way in the hub only.



(a) Hollow Saddle Key (b) Flat Saddle Key

Fig. 2.19 SADDLE KEY

Types of Sunk Keys :

1. **Square and rectangular keys**: These are strongest and most widely used keys. Square or flat keys may be of uniform cross-section or they may be slightly tapered (1 in 100) on their thickness.

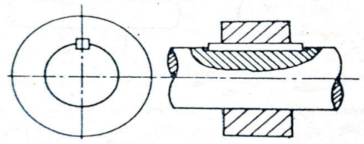


Fig. 2.18 (a) Square Key

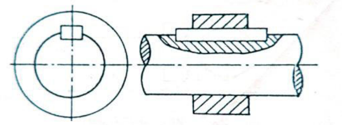


Fig. 2.18 (b) Rectangular Key

2. **Gib Head Key :**
It is a taper rectangular key fitted witg gib head to facilitate easy withdrawal .the taper is on the top of the key.

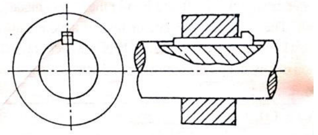


Fig. 2.18 (c) Gib Headed Key

3. **Feather Key :**
Feather key is used when relative axial motion is required between the components while transmitting the torque. It is fixed into shaft by small cap screws as in Fig. 2.18 (d). For double headed feather key (sliding feather key), the key way must be extended to the end of the shaft so that the gear or pulley can be slide on with the key in position

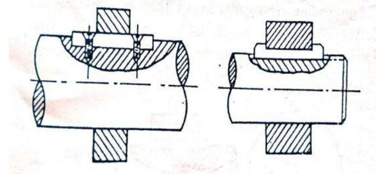


Fig. 2.18 (d) Feather Fig. 2.18 (e) Double Headed Feath

4. **Woodruff key:** This is semi-circular in shape and fits a similar shaped key way. This key is particularly useful on tapered shafts as it is self-aligning.

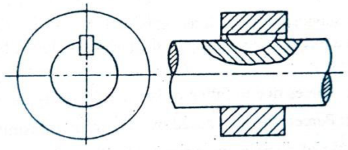


Fig. 2.18 (f) Woodruff Key

5. **Round key**: It is a circular pin and does not requires the accurate key way. It is driven into the hole, drilled partly on shaft and partly in the hub as shown in Fig. 2.18 (g). It is used for light and medium loads.

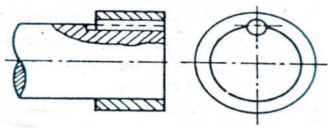


Fig. 2.18 (g) Round Key

6. **Tangent Key:** Tangent keys are used for heavy duty and are fitted in pair at angles as in Fig. 2.18 (h). Each key is withstand torque in one direction only.

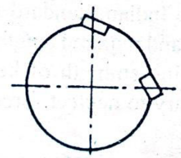


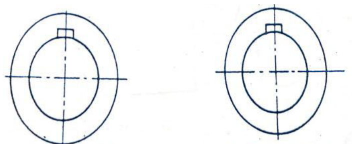
Fig. 2.18 (h) Tangent Key

Saddle Keys:

The two forms of saddle keys are shown in Fig. 2.19.

(a) Hollow saddle key: This type requires key way in the hub and the underside of the key is curved to suit the shaft. Key is tapered and transmits torque by friction alone. Therefore it is suitable for light loads.

(b) Flat saddle key : This type requires a key way in hub and corresponding 'flat' on the shaft. It is also used for light loads, but it has a greater driving than saddle key.



(a) Hollow Saddle Key (b) Flat Saddle Key

Fig. 2.19 SADDLE KEY

Design of Sunk Key

Keys support parts such as gears and pulleys on shafts which are subjected to a torque. The stresses in the key are induced by the following forces :

1. Forces due to fitting of key in a key way, and

2. Forces that are caused by the torque transmitted.

Because of complexity, an exact analysis of the stresses usually cannot be made. It is assumed that the entire torque is carried by a tangential force located at the shaft surface and the force along the length of the key is uniform.

The keys are fail due to shearing and crushing. The designer usually confirm to Indian Standard recommendations on the width and thickness of the key and required length is calculated by considering the shear strength and crushing strength of key. While considering the strength of key, it is customary to neglect forces due to fit of key in a key way.

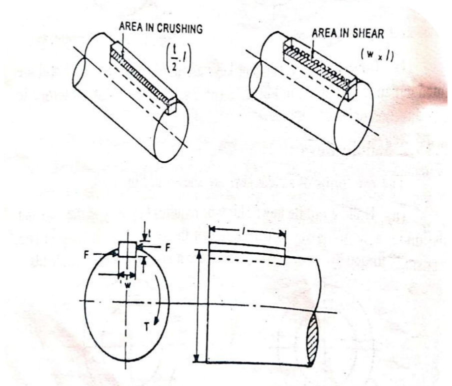


Fig. 2.20 STRENGTH OF KEY

1. Induced shear stress in key :

$$\tau = \frac{2T}{w\,l\,D}$$

2. Induced crushing stress in key :

$$\sigma c = \frac{4T}{t\,\,l\,D}$$

T = Torque to be transmitted, Nmm

τ = shear stress induced in key, N/mm²

σc = Compression stress induced in key, N/mm2

D = diameter of shaft, mm
w = width of key, mm
t = thickness of key, mm
l = length of key, mm

Proportions of Rectangular Sunk key

Width of key,	**w = D/4**
Thickness of key,	**t = D/6**
Length of key,	**l = 1.571 D**

POWER SCREW

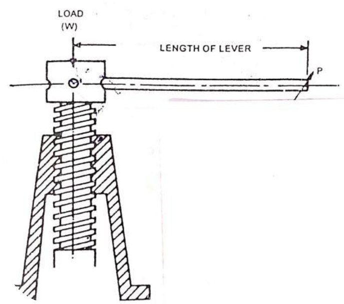
The various forms of screws are employed to fasten the parts together such a manner that the parts can be disconnected readily. i.e. They are used as temporary fasteners Certain forms of screw threads are **suitable transmission of power. " The power screws are used to convert rotary motion into linear motion."** These screws are widely used in presses, clamps, vices, testing machines, lathes and many other machine tools to move the parts (slides) against resisting force.

Power screw consists of screw and nut, and torque is applied to one of these elements causing it to rotate and move either itself or other element in an axial direction. In most power screws, the screw rotates in bearings while the nut has axial motion against the resisting force.

SCREW JACK:

The very large masses can be raised by pushing them up inclined plane having a shallow gradient The screw of a jack can be read an inclined plane wrapped around a cylindrical core. The general form of a screw jack is shown in below figure. It consists of a vertical screw and nut (body).

The load rest on the screw head and the nut forms the body of screw jack. The load resting on the head of the screw is regarded in the same way as the block on inclined plane The effort is applied at the end of lever or handle fitted to the head. When the screw is rotated through one turn, the load is moved by a height equal to the lead which is equal to the pitch in case of single start thread.



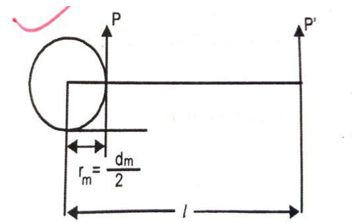
$$\boxed{\tan\alpha = \frac{p}{\pi\, d_m}}$$

For multistand thread,

$$\tan\alpha = \frac{n.p}{\pi d_m}, \text{ Where } n = \text{number of starts}$$
$$= 2 \text{ for double start}$$
$$= 1 \text{ for single start}$$

W – Load applied on head
P – Effort applied at the end of lever
l – Length of handle
p – Pitch of screw
d_m - mean diameter of screw
$\propto$ - Helix angle
n - No: of starts on thread
Ø - Angle of Friction

Effort Required to lift the load:



1. Circumferential effort (Pc) at the circumference of the screw to lift the load

$$\boxed{P_c = W.\tan(\propto + \emptyset)}$$

2. The effort required at the end of the lever of length "l" for lift the load: (**P'**)

$$P_c * r_m = P' * l$$

Or

$$P' = \left(\frac{Pc * rm}{l}\right)$$

Effort Required to Lower the load:

1. Circumferential effort (P'_c) at the circumference of the screw to lift the load

$$\boxed{P'_c\ W.\tan(\phi - \alpha)}$$

2. The effort required at the end of the lever of length "l" for lift the load: (P'_1)

$$P'_c \cdot r_m = P'_1 * l$$

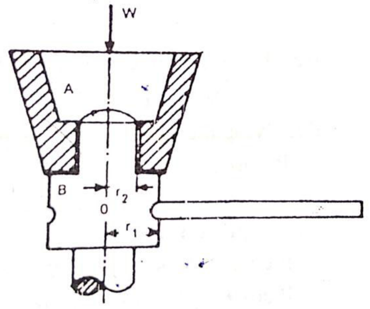
Or

$$P'_1 = \left(\frac{P'_c * rm}{l}\right)$$

Load does not rotate with screw:

To prevent the rotation of load, the screw head is made in two parts as shown in below figure. The part B is rotated by lever and part A does not rotate but simply moves up or down as the screw rotates. There will be sliding friction between A and B.

Let r_1 - External radii
r_2- Internal radii of bearing surfaces
μ -coefficient of friction between the bearing surfaces.



$$\text{Frictional torque} = \mu_1 W \left(\frac{r_1 + r_2}{2}\right)$$

Torque at the end of lever to lift load

$$P'.l = P.r_m + \mu_1 W \left(\frac{r_1 + r_2}{2}\right)$$

Efficiency of screw thread

Definition : "Efficiency is the ratio of ideal torque required to move the load to the actual torque"

$$\text{Efficiency} = \frac{\text{Ideal torque}}{\text{Actual torque}}$$

$$= \frac{W.r \tan\alpha}{W.r \tan(\alpha + \phi)}$$

$$\eta = \frac{\tan\alpha}{\tan(\alpha + \phi)}$$

SELF LOCKING AND OVERHAULING

Self locking condition: (on lowering)

"If $\phi > \alpha$, the load will not move downwards without the application of any torque".
Ie; $P_c = W.\tan(\phi - \alpha)$, $\phi > \alpha$

Effort will be a + ve value, that means
Torque is Positive
We should apply some torque to lower the load. Such condition is called **Self locking**.

Overhauling condition: (on lowering)

"If $\alpha > \phi$, the load will start moving downwards without the application of any torque".
Ie; $P_c = W.\tan(\phi - \alpha)$, $\alpha > \phi$
Effort will be a - ve value, that means
Torque is Negative
We should not apply any torque to lower the load. Such condition is called **Overhauling.**

Maximum efficiency of Screw Thread:

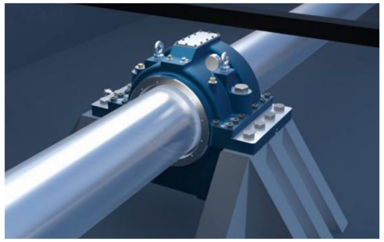
The maximum efficiency of a screw thread is given by ,

$$\boxed{\varphi = \frac{\tan\alpha}{\tan(\alpha + \phi)}}$$

MODULE 2

SHAFTS

- Shafts are a form the important elements of machines and power transmission devices.
- They are usually made in circular section and could be either solid or hollow, and are supported on bearings.
- They carry rotating parts like gears and pulleys, and are subjected to torque due to power transmission and bending moment due to weight of the pulleys and gears.



AXLE :
It is a rod or spindle (either fixed or rotating) passing through the centre of a wheel or group of wheels.

SPINDLE : a short shaft in a machine.

CLASSIFICATION OF SHAFTS:

Shafts involved in power transmission may be classified as

Transmission shafts: **Transmission shafts are used to transmit power between source and the machines using the power. They include line shafts, jack shaft and counter shaft.**

(a) **Line shafts**: Line shaft is a long continuous shaft which receives power from the source and distribute to different machines.

(b) **Jack shafts**: is directly connected to the source of power and from which other shafts are driven.

(c) **Counter shafts** receives power from line shaft and transmits to a machine.

Machine shafts: **Machine shafts are incorporated within the machines, such as crank shaft.**

Materials for shaft should possess the following requirements.

- **High strength**
- **Good wear resistance**
- **Low notch sensitivity**
- **Ability to be heat treated**
- **Good machinability**

  - All commercial shafts are usually made of carbon steel having 0.25 to 0.4 percent carbon.
  - For high strength requirements an alloy steel, such as nickel, nickel-chromium or nickel-vanadium steel, is used
  - Shafts are formed either by hot rolling or cold rolling processes.

Design Factors:

Factors considered while designing a shaft

1. Material and heat treatment
2. Strength to resist torque and bending
3. Stiffness to resist deflection
4. Weight and space limitation, and
5. Stress concentration.

Torsion Equation:

$$\frac{T}{J} = \frac{\tau}{R} = \frac{G\,\theta}{L}$$

T = Torque transmitted by the shaft, Nmm
J =Polar moment of inertia, mm^4
τ =Shear stress induced, N/mm^2
R= Radius of shaft, mm
θ = Angle of twist, radians
G = Modulus of rigidity, N/mm^2
L = Length of shaft, mm

Design of Shafts Subjected to Torsion:

The design of shafts subjected to torsion is usually based on the formula for torsional strength which is given as

Torque, $$T = \tau\left(\frac{J}{R}\right)$$

Where, R= Radius of shaft , mm

J = Polar moment of inertia, mm^4

For Solid shaft , Js

$$J_s = \frac{\pi\, D4}{32}$$

For Hollow shaft , J_h

$$J_h = \frac{\pi}{32}\left(D_o^4 - d_i^4\right)$$

Do – External diameter of shaft, mm
di - Internal diameter of shaft , mm

Polar Section Modulus, Z_p

For Solid shaft, Z_{ps}

$$Z_{ps} = \frac{\pi\, D3}{16}$$

For Hollow shaft, Zp_h

$$Zp_h = \frac{\pi}{16\,D}\left(D_o^4 - d_i^4\right)$$

- **Power transmitted by shaft:**

$$\text{Power, } P = \frac{2\pi NT}{60}$$

Where,

P - Power transmitted in, watt
N – Speed in ,rpm
T - Torque transmitted in, Nm

- **Percentage of saving of material :**

% saving of Material ,

$$= \frac{D^2 - (Do^2 - di^2)}{D^2} \times 100$$

Service Factor:

Ratio of maximum torque to the mean torque;

$$\frac{T_{max}}{T_{mean}}$$

Design of shaft based on Rigidity:

In many cases the sizes of shafts are determined on the basis of rigidity. Two kinds of rigidity must be considered, torsional rigidity and lateral rigidity.

Torsional rigidity: The angle of twist or torsional deflection, θ in radians, may be obtained by using the relation,

$$\frac{T}{J} = \frac{G\theta}{L} \quad \text{or} \quad \theta = \frac{TL}{JG}$$

Where, T = torque on shaft, Nmm
L = Length of shaft ,mm
G = Modulus of rigidity , N /mm^2
J = Polar moment of inertia ,mm^4

Limiting the torsional deflection is very important because shaft with excessive deflection cannot be used for certain purpose and a more rigid section must be chosen. As per the specifications the angle of twist must not exceed 1° for a length of shaft equal to 20 times the diameter.

Torsional stiffness: The torsional stiffness of a shaft is defined as the amount of torque required to twist the shaft through 1 radians.

$$\text{Torsional stiffness, } S = \frac{T}{\theta} = \frac{GJ}{L}$$

The strength of the shaft is measured by the amount of torque it can transmit.

Lateral deflection: The lateral (transverse) deflection of a shaft when used in a machine is often more important than its strength. As a result of large deflection, there will be an excessive amount of wear on the bearings and it must be within permissible limit to maintain proper bearing clearance of gear teeth alignment.

The general formula for lateral deflection is given by the relation,

Lateral deflection, y = a constant x $\frac{WL3}{EI}$

Where , W = load on shaft , N
L = Supported length of shaft , mm
E = Modulus of Elasticity , N/mm^2
I = Moment of inertia ,mm^4

The constant depends on manner of load distribution and supporting conditions. For simple support with the concentrated load the mid-span, the above expression reduces to

$$y = \frac{PL3}{48\,EI}$$

And for one end support with concentrated load at free end,

$$y = \frac{PL3}{3\,EI}$$

In practice, the deflection is limited to 0.84 mm per metre length of shaft.

Conversion of Degree to Radian:

$$1^\circ = \frac{\pi}{180}\ \text{radian}$$

Comparison of Hollow shaft and Solid shaft In terms of Weight, Strength, and Stiffness:

1. Comparison of Weight :

- For same material and same length
(Shear stress and length are same for both shafts)
- Then weight is proportional to square of the diameters

$$\frac{W_H}{W_S} = \frac{D_o^2 - d_i^2}{D^2} = (1-K^2)$$

Where , Do = outer diameter of hollow shaft,mm
di = inner diameter of hollow shaft ,mm
D= diameter of the solid shaft, mm .

Diameteral Ratio , $K = \frac{di}{Do}$

2. Comparison of Strength:

Torque , $T = \tau\, Zp$

- For the same material , Torque is proportional to Polar section Modulus,(Zp)

$$\frac{T_H}{T_S} = \frac{\frac{\pi D_o^3}{16}(1-K^4)}{\frac{\pi D^3}{16}} = (1-K^4)$$

3. Comparison of Stiffness:

Stiffness, S = $\frac{T}{\theta} = \frac{GJ}{L}$

- For the same material and length ,stiffness is proportional to Polar Moment of inertia,(J)

$$\frac{S_H}{S_S} = \frac{\frac{\pi(D_o^4 - d_i^4)}{32}}{\frac{\pi}{32}D^4} = (1-K^4)$$

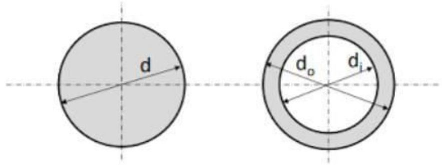


Figure Q3

Do =do= outer diameter of hollow shaft,mm
di = inner diameter of hollow shaft ,mm
D=d= diameter of the solid shaft, mm .

COUPLINGS

Coupling is a device for connecting the ends of two shafts together. Some of the reasons for joining the shafts are:

1. to increase the length of shafts to accomplish a special purpose
2. to transmit power from one shaft to other shaft.
3. to reduce the vibrations.
4. to provide mechanical flexibility, and
5. to permit misalignment of shafts.

Couplings are inexpensive and will withstand rough use. The requirements of good couplings are,

1. must transmit full torque of shaft
2. should permit easy connections and disconnection of the shafts
3. should protect against overload, and
4. should provide safety in operation.

TYPES OF COUPLINGS:

Couplings are of two types

1. Rigid couplings

2. Flexible couplings

Rigid couplings are used for the shafts whose axes are collinear.
These couplings do not permit any misalignment of shafts. The common forms of rigid couplings are

1. Muff coupling
2. Flange coupling, and
3. Clamp (split-muff)

Flexible couplings are used for the shafts whose axes are not collinear. These coupling permit misalignment of shafts, and possess flexibility and resilience. The common forms of flexible couplings are.

1. Oldham coupling: permits small lateral misalignment.
2. Universal coupling: permits small angular misalignment.
3. Bushed pin type coupling: absorbs shocks and permits small amount of angular and lateral misalignment.

Design of Muff coupling (sleeve coupling)

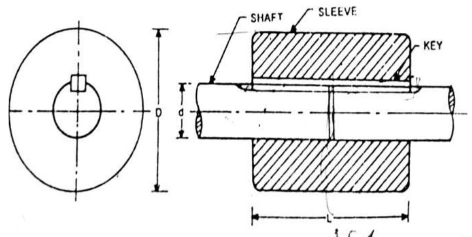


Fig. 5.1 MUFF COUPLING

- The muff or sleeve coupling is a simple form of rigid coupling and used to connect collinear shafts. It consists a muff made of cast iron. The muff is connected over the end of the shaft by means of sunk key as in above figure (5.1).

Sl no;	Parameter	Equation
1	Internal diameter of Muff (di)	di = d
2	Outer diameter of Muff (Do)	Do= 2d +3 ,mm
3	Length of Muff (L)	3.5 d

Design of Muff coupling involves:

1. Design of shaft (already discussed in this chapter)
2. Design of key (discussed in first module)
3. Design of Muff

- Muff is considered as Hollow shaft transmitting the torque.
- The shear stress induced in a Muff is calculated by this equation.

Induced Shear stress, τ_{muff}

$$\tau_{muff} = \tau = \frac{16\,T\,D_o}{\pi\,(D_o^4 - d_i^4)}$$

Where , T = Torque transmitted ,Nmm
Do = outer diameter of Muff, mm
di= inner diameter of Muff (shaft dia),mm

- **The dimension of muff is generally computed by empirical relation for induced stress.**

Design of Flange Coupling

- Flange couplings are widely used for heavy power transmission at low speeds.
- The flange coupling consists of two hubs keyed to the two shafts as in the Fig. 5.2 (a).
- The hubs extend into flanges whose faces are brought and held together by a series of bolts arranged concentrically about the shaft so that their axes are parallel to the shaft axis.
- To ensure correct alignment, one of the hubs has a circular projection (A) which fits into a corresponding depression in the other hub.
- The bolts transmit torque from one flange to the other flange, and then to shaft.
- Design of flange coupling includes the design of hubs, flanges, key and bolts.

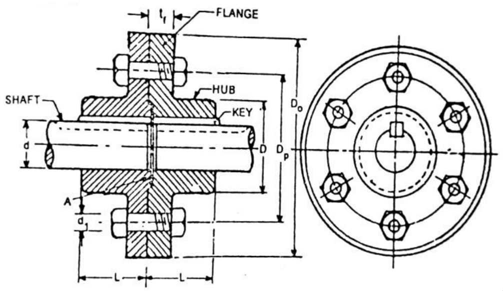


Fig. 5.2 (a) Flange Coupling

Sl No	Parameter	Empirical relation
1	Thickness of flange	$t_f = 0.5d$
2	Diameter of the shaft	d = 1d
3	Length of Hub	L = 1.5 d
4	Diameter of Hub	D = 2d
5	Bolt circle diameter	$D_p = 3d$
6	Outer diameter of Hub	Do = 4d
7	Nominal diameter of bolt	d1
8	Allowable shear stress for shaft , key and bolt material	τ_{shaft} τ_{key} τ_{bolt}
9	Allowable shear stress for Hub and Flange material	τ_{Hub}, τ_{flange}
10	Allowable crushing stress for bolt and key material	σc_{bolt} σc_{key}

Design of Flange Coupling involves

Sl No	Design part	Induced stress equation
1	**Design of shaft:**	
	Shear stress of shaft	$\tau_{shaft} = \frac{T}{Zp}$
	Design of key :	
2	Shear stress	$\tau_{key} = \frac{2T}{wld}$
	Crushing Stress	$\sigma c_{key} = \frac{4T}{tld}$
3	**Design of Hub:**	
	Shear stress of Hub	$\tau_{Hub} = \frac{16\,D\,T}{\pi(D^4 - d^4)}$
4	**Design of Flange:**	
	Shear stress of Flange	$\tau_{flange} = \frac{2\,T}{\pi\,D^2 t_f}$
5	**Design of Bolt :**	
	Shear stress of bolt	$\tau_{bolt} = \frac{8\,T}{\pi\,d1^2 n\,D_p}$
	Crushing stress of bolt	$\sigma c_{bolt} = \frac{2\,T}{d1.tf.n.Dp}$

Where , n = No : of bolts used to fasten the coupling

$$n = \frac{4}{15} d + 3$$

Where, d = Diameter of shaft in centi meter ,"cm"

For a Safe Design for your Machine Element / part:

- **The induced Shear stress values and the induced Crushing stress value should be less than the Permissible shear stress value and crushing stress value of the material you have chosen.**

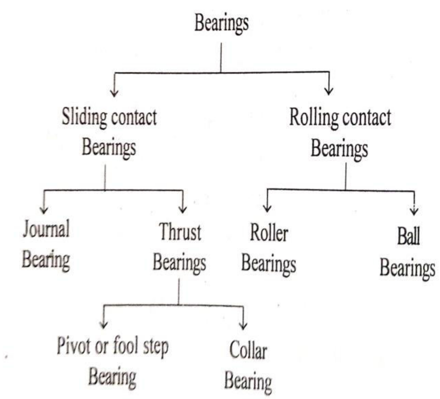
MODULE – 3

BEARINGS:

- Definition: A machine element which supports another element and at the same time constrains the relative motion of the part with minimum friction is called a **Bearing**.
- The portion of the shaft supported by the bearing is known as the journal.
- The common applications of bearings include automobile engines, aircraft engines, shafting are in workshops, many power transmission devices.

Fig: Ball bearing

CLASSIFICATION OF BEARING:



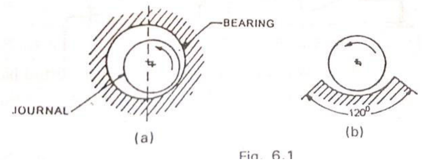
Bearings may be classified as given below.

1. According to the load application:

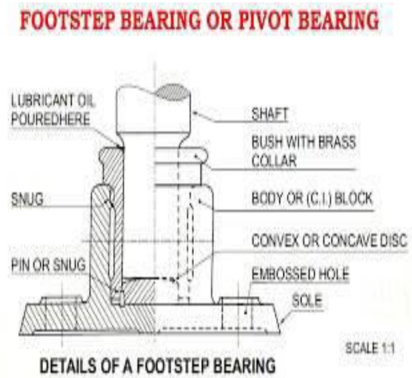
(a) Radial bearings (b) Thrust bearings (C) Guide bearings.

(a) Radial bearing: load acts radial to the axis of the shaft. They are also referred as Journal bearings. When the angle of contact of the bearing with the journal is 360° then it is called a **full journal bearing.**

When the angle of content of the bearing with the journal is 120 degree then the bearing is called **partial journal bearing.**



(b) Thrust bearing: carries a load collinear to the axis of the shaft. If the shaft is vertical and terminates in the bearing, then it is called pivot or foot step bearing. If the shaft continues through the bearing it is called a collar bearing. The shaft may be with single collar or many collars.



(c) Guides bearing: guides the motion of the machine member without specific regard to the direction of load applications.

2. According to the nature of contact:

(a) Sliding contact bearing (Plain bearing)

(b) Rolling contact bearing (Anti-friction bearing)

SLIDING CONTACT BEARINGS: Bearings that do not use rollers or balls as load supporting material are termed as sliding bearings. In this bearing the surfaces make sliding contact. To minimize the friction these surfaces are usually separated by a film of lubrication.

Depending on the direction of load on bearing surfaces, sliding contact bearings are further classified as:

1. Journal bearings 2. Thrust bearings

Journal Bearings: Journal bearings support the radial (transverse) load (i.e., load acts normal to the shaft axis). The following types of journal bearings are more commonly used.

(a). Bushed bearing (b). Plummer block or Pedestal bearing.

a. Bushed bearing: Bushed bearing consists of a cast iron block and a bush made of brass or gun metal. The base plate is provided with the holes for fixing the bearing in position. At the top of the bearing an oil hole is provided to facilitate lubrication of the shaft and the bush.

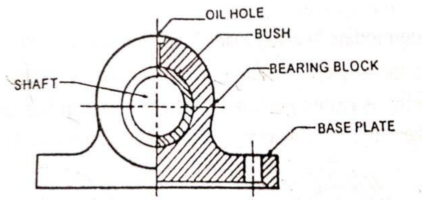


Fig. 6.2 Bushed Bearing

b. Plummer block or Pedestal bearing: The Plummer block or pedestal bearings are used to support the long shafts at several intermediate points. Pedestal bearing consists of a pedestal (block), a cap and a split bush. The shaft is placed between the bushes and the cap is then bolted down on to the block as shown in Fig. 6.3. Flanges provided at either end of the bush prevent its axial movement and snug prevents the rotation of the bush inside the housing.

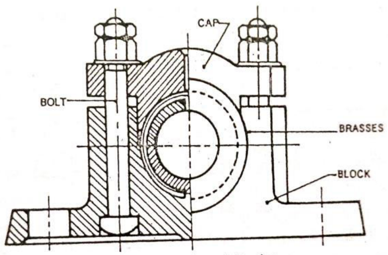


Fig. 6.3 Pedestal Bearing

2. Thrust bearings

- Sliding bearings desired to carry a axial load are called **thrust bearings**.
- The axis of the shaft may be vertical or horizontal. If the shaft is vertical and terminates in the bearing then it is called **a pivot or foot step bearing.**

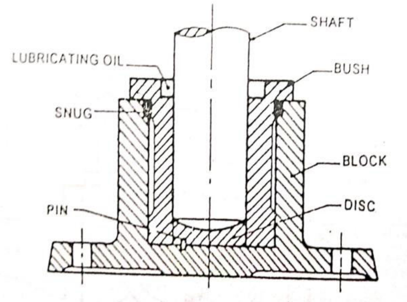


Fig. 6.5 Foot Step (Pivot) Bearing

- If the shaft continues through the bearing then it is called a **collar bearing.** The shaft may be with single collar or many collars. The various forms of thrust bearings are shown in Fig. 6.4**.** Collar bearing: A collar bearing is shown in Fig. 6.6. In this case the shaft passes through a bearing while bearing take up the axial loads on shaft.

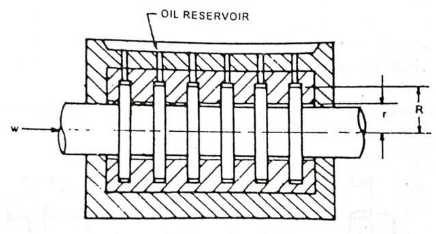


Fig. 6.6 COLLAR BEARING

BEARING MATERIALS:

The materials selected for sliding bearings should posses following properties:

1. Low-coefficient of friction
2. Wear and corrosion resistance
3. High thermal conductivity
4. Ability to withstand bearing pressure
5. Should possess strength and rigidity.

ROLLING CONTACT BEARINGS:

The object of rolling contact bearing is to minimize the friction by substituting pure rolling motion for sliding motion. Since the rolling friction is much less than the sliding friction, rolling contact bearings are called anti-friction bearings

The chief advantages of rolling contact bearing over sliding contact bearing are given below:

Advantages:

Can be adopted for combined radial and axial loads without any complications

1. More compact design
2. Low starting friction
3. Easier to provide lubrication
4. Reliable in service
5. Accurate alignment of parts can be maintained

Disadvantages:

1. Generally more expensive
2. Worn out of contact surfaces causes noise
3. Design of bearing housing is complicated
4. Require good quality of lubricant

Types of Rolling contact bearing:

1. Ball bearings:

- Consists of balls, positioned between hardened steel races.
- The balls are retained in position by cage (separator). Races and balls are made of high carbon chromium steel while the cage is usually made of brass (or any light alloy having low coefficient of friction).

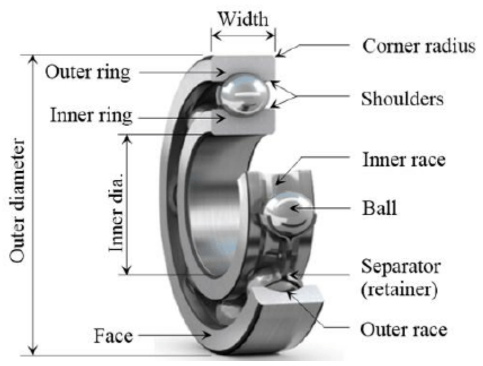


Fig: Ball bearing

2. Roller bearings :

- Consists of rollers, positioned between races with the help of cage or separator.
- The rollers may be in the form of cylinders or of frustums of cones.
- The important feature of these bearings is that the line contact is maintained between a roller and the races.
- The advantage of this type of bearing is the large ratio of load.

Fig: Roller bearing

3. **Needle bearing**: It does not require a cage or retainer. These bearings are used to carry heavy loads with an oscillating motion.

Fig: Needle bearing

DESIGN OF SIMPLE (SOLID) JOURNAL BEARING:

Solid bearings are mostly used for low-speed shafts subjected to light loads. It is a block of material which has desirable properties and a hole equal to the diameter of journal. The bearing may be required to operate continuously under full load for long periods.

The design of bearing involves finding sufficient bearing area which is equal to the product of diameter and length of journal.

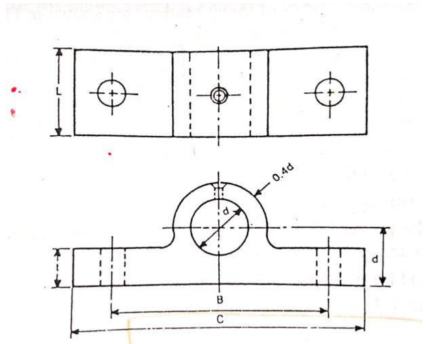
Bearing area, A = d*l = $\frac{W}{Pb}$

W - Load on bearing

Pb - bearing pressure, then

d - Diameter of journal (bearing diameter)

l- Length of journal (bearing length)



- Length of base , c = 3.5d +40 mm
- Width of base, L = d to 2d
- Thickness of base, t = 0.5 d + 6 mm
- Distance between Centers of bolts , B = 2.5 d + 30 mm
- d = diameter of the hole , mm

FRICTION IN JOURNAL BEARING:

(a) Flat Pivot (Foot step bearing)

Load on Bearing, W = p.πR^2

P – Bearing Pressure, N/mm^2
R –radius of shaft, mm

(b) Collar Bearing :

Load on Bearing, W = n.p.$\pi(R^2 - r^2)$

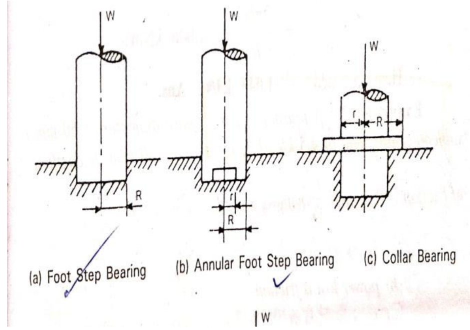
P – Bearing Pressure, N/mm^2
N – No: of collars,
r –inner radius, mm
R- Outer radius, mm

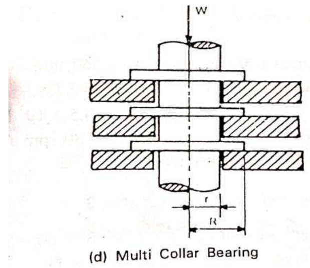
Sl No	Bearing type	Frictional torque (T_f) , Nm
1	Journal bearing	$Tf = \mu W \frac{d}{2}$
2	Flat Pivot bearing	$Tf = \frac{2}{3}\mu W R$
3	Collar Bearing	$Tf = \frac{2}{3}\mu W \frac{R^3 - r^3}{R^2 - r^2}$

µ - Coefficient of Friction.

N – Speed of Journal, rpm

Power Lost due to friction,
$$P = \frac{2\pi N T f}{60}, \text{ W}$$





Power lost due to Friction (W) equal to

Heat generated on bearing (J/s)

COEFFICIENT OF FRICTION AND BEARING CHARACTERISTIC NUMBER (BEARING MODULE)

The loss of power due to friction depends on coefficient of friction The variation of coefficient of friction in lubricated bearings vary with bearing characteristic number as shown in Fig. below

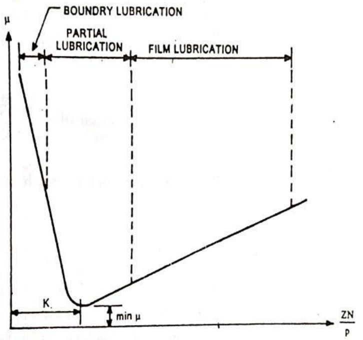


Fig. 6.13 RELATION BETWEEN μ AND $\frac{ZN}{p}$

The graph shows the relationship between coefficient of friction (μ) with respect to bearing characteristic number ($\frac{ZN}{P}$)

Analyze from right to left, the value of ($\frac{ZN}{P}$) decreases also (μ) decreases.

in film lubrication region: there is a thick layer of lubricant between shaft and bearing surface, so there is a minimum chance for the contact between shaft on the bearing surface and the journal bearing will be safe in operation

In partial lubrication region: the value of (μ) is minimum then the corresponding ($\frac{ZN}{P}$) value is called ***bearing module***, the designer have to design the bearing at this value of ($\frac{ZN}{P}$) but if , Z or N increases then (μ) also increases .in this region only a partial layer of lubrication is given to the journal.

In boundary lubrication region: a thin layer of lubrication is applied to the journal the value of (μ) increase exponentially according to the $\frac{ZN}{P}$ value changes.

NB: Safe to design a journal bearing in film lubrication region by suitably selecting the value of Z,N,and P.

The coefficient of friction is the function of:

$$\frac{ZN}{P}, \frac{d}{c} \text{ and } \frac{l}{d}$$

The term $\frac{ZN}{P}$ is **called Bearing characteristic number** in which,

Z = Absolute viscosity, N = speed of the journal,

P = bearing pressure

If the bearing pressure is in kg/cm, absolute viscosity is in centipose and speed in RPM, then the coefficient of friction is given by the equation:

(McKee equation)

$$\mu = \frac{33}{10^{10}} * \frac{ZN}{P} * \frac{d}{c} + K$$

d = diameter of journal

c = clearance

l= length of journal

K= correction factor for leakage

The correction factor depends on $\frac{l}{d}$ ratio.

Sommerfeld Number: Sommerfeld number is also used in the design of journal bearing,

$$\text{SFN} = \frac{ZN}{P} * \left(\frac{d}{c}\right)^2 = 14.3 \times 10^6$$

CAM

A cam is a machine element which drives an other element through a specified motion by direct contact. The cam and follower constitute a higher pair, because they maintain a line contact between the contacting surfaces,

Cam Mechanism:A cam and follower mechanism is shown in figure below basic links of cam mechanism are:

1. Cam
2. Follower
3. Frame

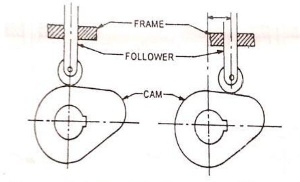


Fig. 7.1 CAM MECHANISM

- Cam rotates about an axis perpendicular its plane.
- Its profile imparts a desired motion to a follower which bears against the cam edge.
- Flower is kept in contact with the cam profile by gravity (for small accelerations) or by a spring.
- Frame is used to support the cam and to guide the follower

TYPES OF CAM

The cams are designed such that their profile will impart desired motion to the follower. They are classified as:

1. Wedge cam
2. Radial or disc cam
   a. Tangent cam, and
   b. Circular cam.
3. Cylindrical cam

1. **Wedge cam**: In wedge cam the reciprocating (translatory) motion of the cam is transformed into reciprocating (Fig 7,2.1 (a)) or oscillating (Fig. 721 (b))motion of the follower.

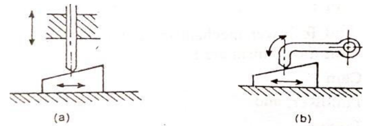


Fig. 7.2.1 Wedge cam

2. **Radial cam**: In radial or disc cams the follower reciprocates in a plane right angle to the axis of the cam. A radial cam having straight flank and circular nose is called tangent cam (Fig. 7.2.2 (a-b)). A circular cam (Fig. 7.2.2 c) has circular flank and circular nose.

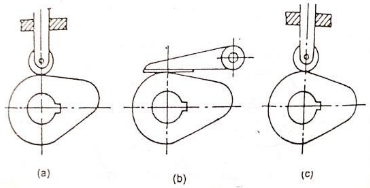


Fig. 7.2.2 Radial Cam

3. **Cylindrical cam**: In cylindrical cams the follower reciprocates or oscillates in a plane parallel to the axis of the cam. Fig. 7.2.3 shows the cylindrical cam.

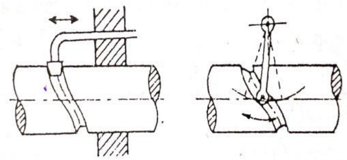


Fig. 7.2:3

TYPES OF FOLLOWERS: The three basic types of followers are;

1. Flat (mush room) follower
2. Roller follower
3. Knife edge follower

- Flat followers [Fig. 7.3 (a)] are generally used for slow moving cams
- Roller followers [Fig. 7.3 (b)] are for high speed, and can transmit high forces.
- The knife edge follower [Fig. 7.3 (c)] is not in common use because of high rate of wear.

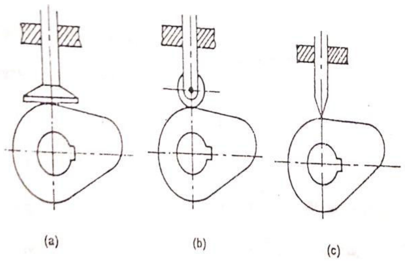
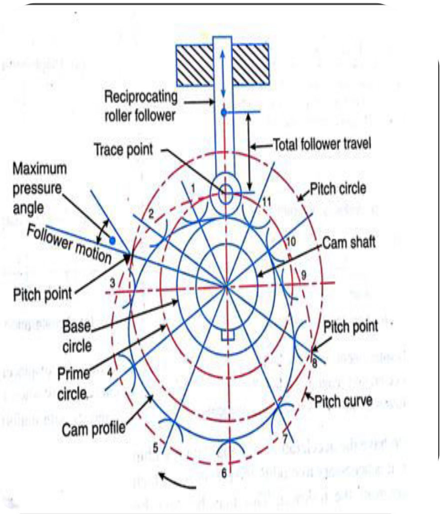


Fig. 7.3 Types of Followers

CAM Terminology:



1. **Cam profile**: It is the outline of the disc cam on which the follower is always in contact during its motion.
2. **Base circle**: The circle drawn with minimum radius from the cam centre to the cam profile.
3. **Cam angles**: The angle through which a cam rotates to displace the follower through a definite distance is called cam angle.
   a. The angle of cam rotation during the rise of follower is called *<u>angle of ascent.</u>*
   b. During the descent of follower, the angle turned by cam is <u>angle of descent.</u>
   c. The angle of cam rotation during which the follower is stationary called <u>dwell angle.</u>
4. **Trace point**. It is the reference point on follower and is used to draw pitch curve.
5. **Pressure angle**: It is defined as the angle between the direction of follower and normal to the pitch curve. This angle is limited to 30°.
6. **Pitch point**: The point on pitch curve which indicates the maximum angle is called pitch point. The circle drawn from cam centre through pitch point is called pitch circle.
7. **Prime circle**: It is the smallest circle drawn with centre on the cam centre and with minimum radius from cam centre to the pitch curve.
8. **Lift or stoke**: The travel of the follower from its lowest position to the top most position (i.e., max. travel of follower) is called lift or stoke.
9. **Dwell:** The stationary period of follower during a part of cam rotation is called dwell.

MOTION OF THE FOLOWERS

DISPLACEMENT DIAGRAMS: Displacement diagrams are used to analyze the movement of the follower relative to the rotation of the cam. Cams can be designed to produce the following motions of follower.

1. Uniform velocity (UV)
2. Simple Harmonic Motion (SHM)
3. Uniform acceleration and retardation.

UNIFORM VELOCITY DISPLACEMENT DIAGRAM

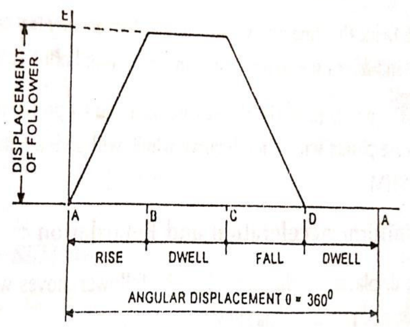
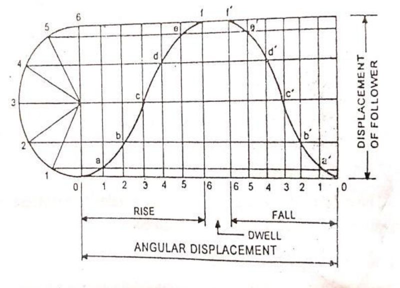


Fig. 7.5

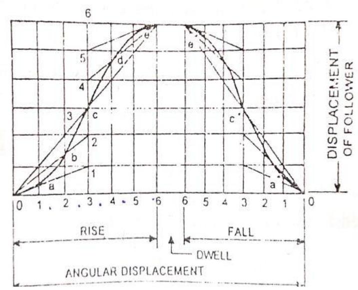
- The displacement diagram when the follower moves with uniform velocity is shown in Fig above.
- The ordinate (y-axis) represents the follower movement and the base (x-axis) represents the angular displacement of cam or time interval.
- As the cam rotates about its axis, the follower moves from A to E during the period AB.
- The follower then remains stationary during BC, and moves inwards (returns) during the period CD.
- . A second dwell period occurs during DA and the cycle is repeated for the next revolution.
-

SIMPLE HARMONIC MOTION (SHM)



- The displacement diagram for Simple Harmonic Motion of follower is shown in Fig. above.
- The following procedure may be followed to construct such a diagram.
  1. Draw a semi circle of diameter equal to the lift of the follower.
  2. Divide this semi-circle into any convenient number of parts (say 6).
  3. Divide the time interval or angular displacement of cam rotation during lift and fall of the follower into the same number of equal parts.
  4. The points (a-a', b-b!.. etc) are obtained by projecting the lines and join these points with smooth curve which will give the displacement curve for SHM.

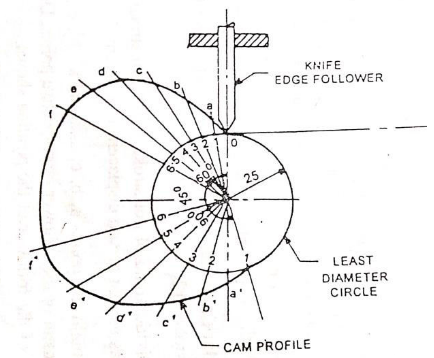
UNIFORM ACCELARATION AND RETARDATION:



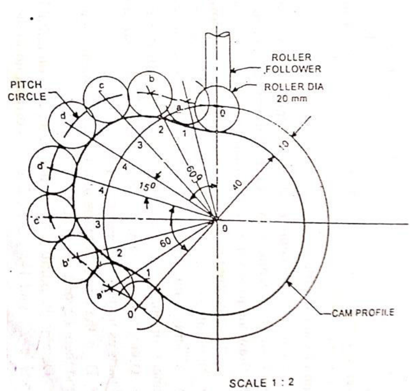
- The displacement diagram when the follower moves with uniform acceleration and retardation is shown in Fig. below.
- Divide the lift into any number of equal parts (say 6)
- Divide the angular displacement of cam for rise and fall of the follower into same number of parts as that are on lift.
- The points (a, b....e and a', b....e!) are obtained as shown in Fig. 7.7. Join these points to obtain a smooth curve which is parabolic.

CAM Profile Drawing:

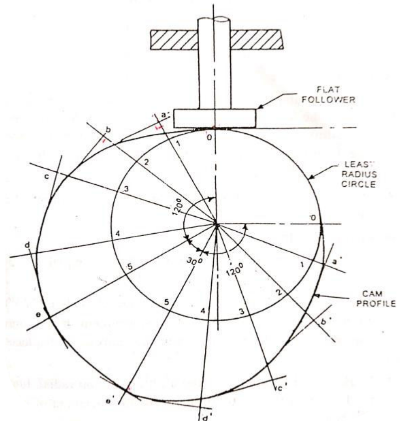
1. Knife edge follower



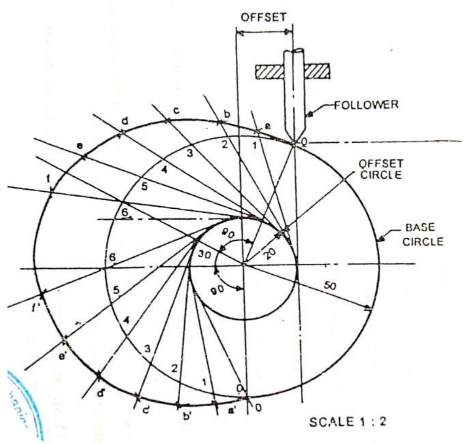
2. Roller follower:



3. Flat follower



4. Offset , knife edge follower:



GOVERNERS AND FLYWHEELS

- Whenever load change on the engine, the speed of output shaft will also changes accordingly.
- It is necessary to control the mean speed of engine at varying loads over a long period.
- **Governor is a device for automatic control of speed by controlling the supply of working fluid. The mean speed of the engine decreases as load increases and it is the purpose of governor to supply energy (working fluid) to suite the new load**.
- At lower loads the governor reduces the energy supply. Thus governor regulates the supply of working fluid according to load.
- In steam engine governing may be done either by throttling the steam or by varying the point of cut-off.
- In petrol engine, the governor manipulates the throttle valve. In diesel engine, it manipulates the fuel pump.

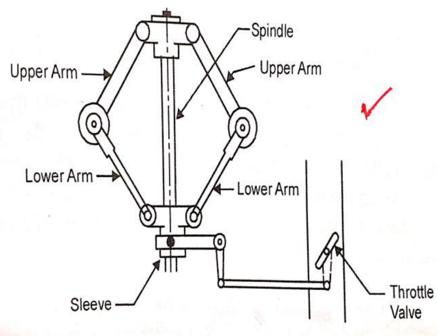


Fig. 8.1 ACTION OF SIMPLE GOVERNOR

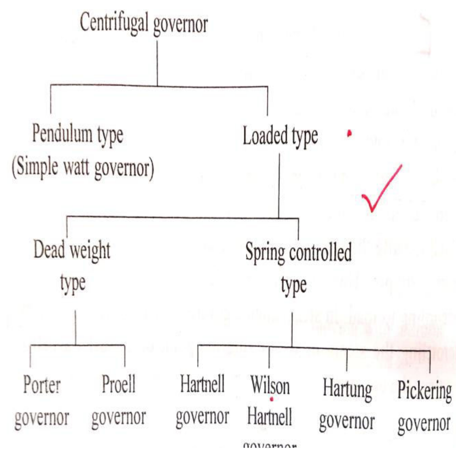
Types of Governors

Governors are broadly classified into two types

1. Centrifugal governor
2. Inertia governor.

Centrifugal governor:

- In centrifugal governor balls are caused to revolve about the axis of a shaft which is driven by engine crank shaft.
- The governor shaft (or spindle) is connected to crank shaft by suitable gearing.
- The revolution of balls gives rise to centrifugal force acting radially outwards.
- This centrifugal force which is balanced by controlling force (provided by dead weight or spring) plays the major part in the governor action.



PORTER GOVERNOR :

- Porter governor which is a dead weight type of centrifugal governor.
- Two balls are suspended by upper arms to the upper end of the spindle.
- Two links (lower arms) connect the balls to sleeve which can move up and down.
- The position of the sleeve depends on the speed of the spindle.
- The sleeve is connected to the engine throttle valve by bell crank lever.
- The speed of rotation of balls increases as the load on the engine decreases and the governor balls fly outwards (i.e., sleeve rise) until the centrifugal force is just balanced by the inward controlling force provided by dead weight.
- Conversely, if the speed of rotation decreases due to an increased load on the engine, the governor balls will move inward (i.e., sleeve descend) until the centrifugal force is again balanced by the controlling force.
- The movement of sleeve is transmitted by bell crank lever to throttle valve which controls the energy (working fluid) supplied to the engine.
- The upward movement of sleeve reduces the valve opening to reduce the energy supply and downward movement of sleeve increases the valve opening to increase the energy supply.

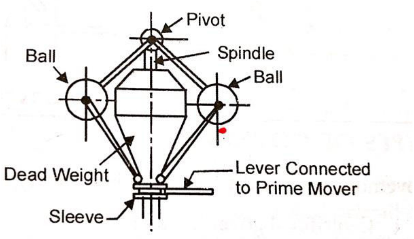


Fig. 8.2 PORTER GOVERNOR

INERTIA GOVERNOR:

- In inertia governor the governing mechanism is operated by the inertia force caused by an angular acceleration or retardation of the governor shaft.
- The amount of the displacement of governor balls caused by the inertia forces is controlled by springs.
- This governor is most sensitive because of rapid response to the change of load and the displacement of balls is determined by the rate of change of speed of rotation rather than the actual change of speed of rotation.
- However, it presents a practical difficulty of arranging for complete balance of the revolving parts of the governor. Therefore the centrifugal governors are widely used.

Definition :

(a) Height of a governor: The vertical distance from the centre of the ball to the point where the axes of the arms intersect on the spindle axis is called height of a governor. In case of offset pivot, the arms extended to intersect the spindle axis.

(b) Equilibrium speed: The speed at which the sleeve does not tends to move upward or downward is called equilibrium speed. At equilibrium speed governor balls, arms etc are in complete equilibrium.

(c) **Mean equilibrium speed**: It is the speed at the mean position of sleeve or balls.

(d) **Maximum and minimum equilibrium speed** : The speeds at maximum and minimum radius of the balls, without tending to move either way are called maximum and minimum speeds respectively.

(e) **Sleeve lift:** The vertical distance which the sleeve moves due to change equilibrium speed is called sleeve lift

(f) **Sensitiveness:** The governor is said to be sensitive when it readily responds to a small variation of speed. The difference between maximum and minimum speeds of a governor is called *range of speed*. The sensitiveness of a governor is defined as the ratio of the range of speed to the mean speed of governor

(g) **Stability**: A governor is said to be stable if there is only one radius of rotation of governor balls for each speed within the working range of governor ie, for each speed within the working range there is only one radius of rotation for equilibrium

(h) **Isochronism**: A governor is said to be Isochrous if it runs at a particular speed for any position of sleeve (i.e., for all radii of rotation of balls) within the working range. Thus. A governor is Isochronous if the equilibrium speed is same for all radii of rotation.

(i) **Hunting**: The condition at which the speed of the engine controlled by governor fluctuates continuously above and below the mean speed is called hunting.

THE FLYWHEEL:

- Some power sources produces energy during a small portion of the cycle and it is necessary to give out uniform energy throughout cyclic operation.
- **A flywheel is a device which acts as reservoir of energy.**
- It absorbs energy during the period when excess energy is supplied.
- The stored energy may be redistributed when the energy supplied is not sufficient for the load on the engine.
- Thus, the function of flywheel is to reduce the fluctuation of energy and to make the flow of energy uniform.

COEFFICIENT OF FLUCTUATION OF SPEED

- The difference between the maximum and minimum angular speed of the flywheel is called the maximum fluctuation of speed of the flywheel.
- The ratio of the maximum fluctuation of speed to the mean speed of the flywheel is called the coefficient of fluctuation of speed.
- Thus, the **coefficient of fluctuation speed,** k is given by the equation,

$$\boxed{k_s = \frac{\omega 1 - \omega 2}{\omega}}$$

$\omega 1$- Maximum angular speed,rad/s

$\omega 2$ - Minimum angular speed,rad/s

ω - Mean angular speed = $\frac{\omega 1 + \omega 2}{2}$

COEFFICIENT OF FLUCTUATION OF ENERGY:

- The difference between the kinetic energy of the flywheel at its maximum speed and minimum speed is called maximum fluctuation of energy.
- The ratio of maximum fluctuation of energy to the network done per cycle is called **the Coefficient of Fluctuation of energy.**

  Given by : $$Ke = \frac{dE}{E}$$

dE – maximum fluctuation of energy in a cycle

E - net work done per cycle.

ENERGY STORED IN FLYWHEEL:

It is given by:

$$\text{KE of Flywheel} = \frac{Ke.E}{2\,Ks} = \frac{dE}{2\,Ks}$$

TURNING MOMEMT DIAGRAM:

- Torque on the engine crank shaft varies considerably through the cycle.
- **Turning moment diagram is the graph plotted against crank shaft torque and crank angle.**
- Over a complete cycle, the sum of the areas of the loops above and below the mean torque line are equal.
- Flywheel act as reservoir, absorbing energy as the speed increases and releasing it as the speed falls.

If the energy of the flywheel at A , is E , then

Energy of flywheel at B = E+a

Energy of flywheel at C = E+a-b

Energy of flywheel at D = E+a-b-c

Energy of flywheel at E = E+a-b-c-d = E

The energy at point E must be same as that of the energy at point A, since the is the repeated.

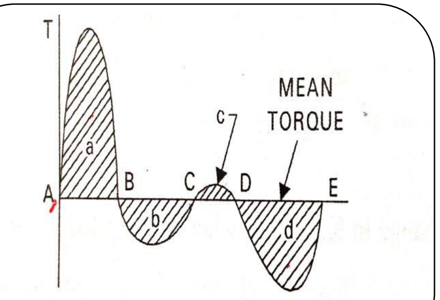


Fig. 8.5 TURNING MOMENT DIAGRAM