Machine Design

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Brakes and Clutches

Introduction to Brakes

A brake is a mechanical device whose function is used to absorb the kinetic energy of the system by means of friction.

Types of Brakes

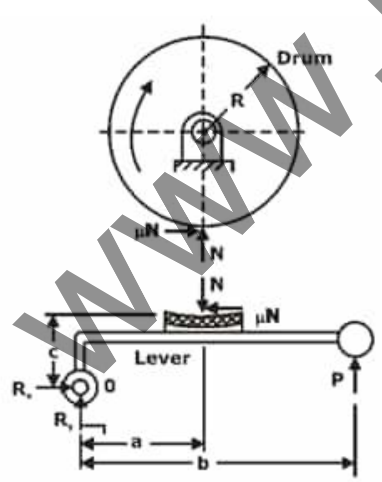
(i). Mechanical Brakes: The working of these brakes is based on mechanical means such as levers, springs and pedals.

(ii) Hydraulic and pneumatic brakes: Controlling action is due to fluid pressure such as oil pressure or air pressure.

(iii). Electrical brakes: Working is based on magnetic forces and this category of brakes includes magnetic particle brakes, hysteresis brakes and eddy – current brakes.

Block brake with short shoe brake: It is combination of a simple block and rotating drum.

As the block is pressed, the drum motion is retarded due to the friction between the drum and the block. The contact angle of the block and the brake drum is generally small. If the contact angle < 45°, then pressure distribution is uniform of the drum due to braking action.



The forces acting on the brake drum,

$$M_t = \mu NR$$

where:

M_t = braking – torque (N-mm)

R= radius of the brake drum (mm)

μ=coefficient of friction,

N= normal reaction (N),

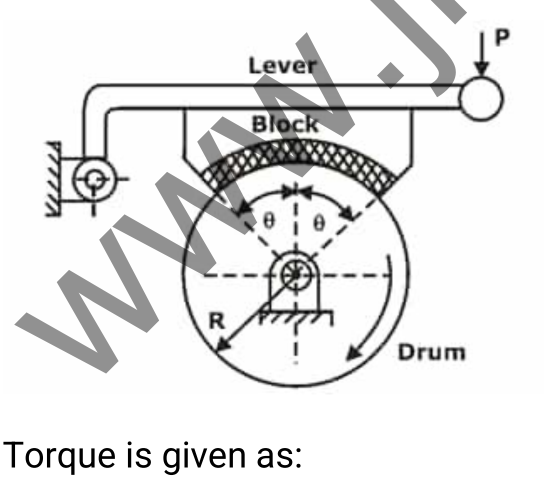
Based on the hinge pin location and coefficient of friction value, there are three different possibilities:

Case I: a > μc, The friction force helps to reduce the magnitude of the actuating force .

Case II: a = μc, The actuating force is zero which signifies that no external force is needed for the braking action and the brake is known a '**self – locking**' brake.

Case III: a < μc, The actuating force becomes negative. It is serious condition which results in uncontrolled braking and grabbing.

Block brake with long shoe:



Torque is given as:

$$M_t = \mu' NR$$

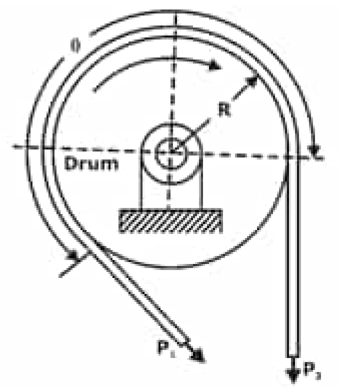
Where:

$$\mu' = \left[\frac{4\mu \sin\theta}{2\theta + \sin 2\theta}\right]$$

and, θ is the semi – cone angle.

Simple band brakes:

The steel band working is is same as that of a flat belt operation with zero velocity.



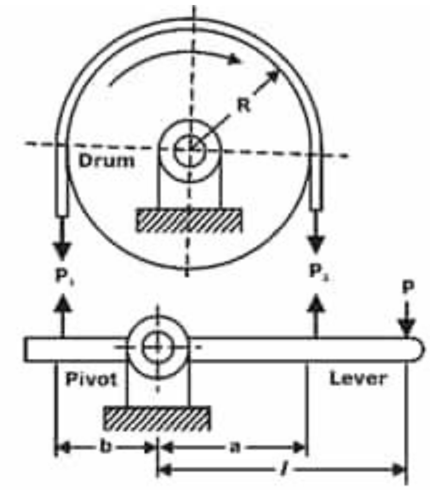
$$\frac{P_1}{P_2} = e^{\mu\theta}$$

Torque absorbed by the brake is given by:

$$M_t = (P_1 - P_2)R$$

Differential band brakes:

- When neither end of the steel band passes through the fulcrum of the actuating lever, the brake is called the differential band brake.
  The free – body diagram of forces acting on the band and the lever is shown in figure,



$$P = \frac{P_2\left(a - b \times e^{\mu\theta}\right)}{l}$$

For the self – locking condition:

$$\left(\frac{a}{b}\right) \le e^{\mu\theta}$$

Disk brakes: A disk brake is similar to a plate clutch with one small modification of replacement of one of the shafts by a fixed member.

Advantages of disk brakes

- Disk brake is simple to install and operate
- Disk brake has high torque transmitting capacity in small volume
- The brake can never become self – locking

Introduction of the clutch:

It is a mechanical device, which is used to connect or disconnect the source of power from the remaining parts of the power transmission system at operator's will.

Types of clutch:

(i). Positive Contact Clutches: Examples of such clutches are square jaw clutches, spiral jaw clutches and toothed clutches.

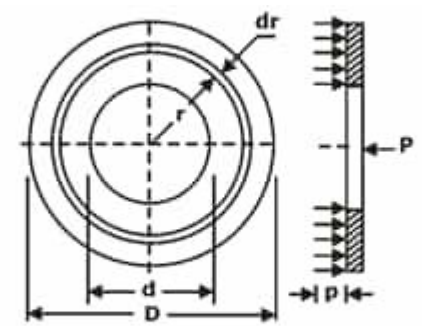
(ii). Friction Clutches: They include single and multi-plate clutches, cone clutches and centrifugal clutches.

(iii). Electromagnetic Clutches: Magnetic particle clutches, magnetic hysteresis clutches and eddy current clutches are examples of electromagnetic clutches.

(iv). Fluid Clutches: Hydraulic pressure is the means of the power transmission.

Torque transmitting capacity of single plate clutch:

A friction disk of a single plate clutch is shown in figure:



D= Outer diameter of friction disk (mm)

d = inner diameter of friction disk (mm)

P= intensity of pressure at radius (mm)

F = operating force (N)

M_t = torque transmitted by the clutch (N-mm)

$$M_t = 2\pi\mu \int_{d/2}^{D/2} pr^2 dr$$

The theories used to find the torque capacity of the clutch are Uniform pressure theory (UPT) and Uniform wear theory (UWT).

Uniform Pressure Theory (UPT): In case of new clutches employing a number of springs, the pressure remains constant over the entire surface area of the friction disk.

$$M_t = \frac{2\mu p}{3}\frac{(R^3 - r^3)}{(R^2 - r^2)}$$

Uniform wear theory (UWT): It is assumed that the wear is uniformly distributed over the entire surface area the friction disk. When the wear is uniform:

P.r = Constant

In this case, P is inversely proportional to r. Therefore, maximum pressure occurs at the inner radius while minimum pressure at the outer periphery.

$$M_t = \frac{\mu p}{2}.(R + r)$$

Note:

Number of disks = number of pairs of contacting surfaces + 1 = Z + 1

Suppose:

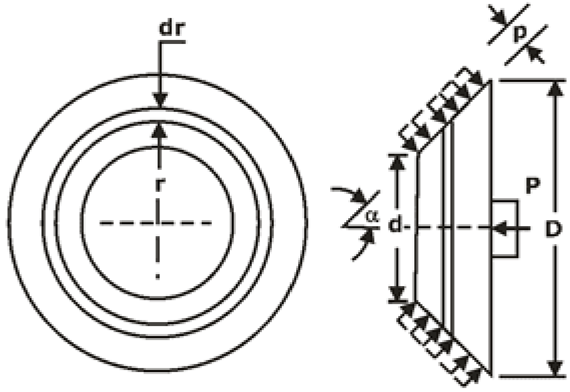
Z_1= number of disks on driving shaft

Z_2 = number of disks on driven shaft

$Z_1 + Z_2$ = number of pairs of contacting surfaces + 1

Number of pairs of contacting surfaces = $Z_1 + Z_2 - 1$

Cone clutches:



In case of uniform pressure theory:

$$M_t = \frac{\mu p}{3\sin\alpha}\frac{(D^3 - d^3)}{(D^2 - d^2)}$$

In case of uniform wear theory:

$$M_t = \frac{\mu p}{4\sin\alpha}(D + d)$$

where, α is the semi cone angle.

Centrifugal clutch: It consists of number of shoes on the inside of a rim of the pulley and the shoes outer surfaces are covered with a friction material. These shoes (move radially in guides) are held against the boss on the driving shaft by means of spring which causes a constant radially inward force.

the torque transmitting capacity of the clutch is given by,

$$M_t = \frac{\mu m r_g r_d z(\omega_2^2 - \omega_1^2)}{1000}$$

Notations used

r_d = radius of the drum (mm)

r_g = radius of the centre of gravity of the shoe in engaged position (mm)

m= mass of each shoe

P_{cf} = centrifugal force (N)

P_s = spring force (N)

z= number of shoes

ω_2 =running speed (rad/s)

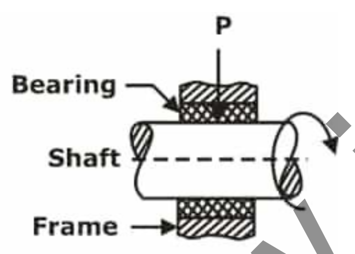
ω_1= speed at which engagement starts (rad/s).

Bearings

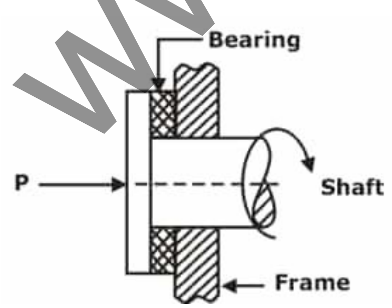
Bearings: It is a mechanical element that allows relative motion of the two machine elements such as the shaft and the housing resulting in minimum friction.

Types of bearings: Based on the direction of force that acts on them, bearings are classified into two categories – radial and thrust bearings.

Radial bearings: A radial bearing supports the load, which is perpendicular to the axis of the shaft.



Thrust bearings: Thrust bearing are used to support the loads acting along the axis of the shaft.



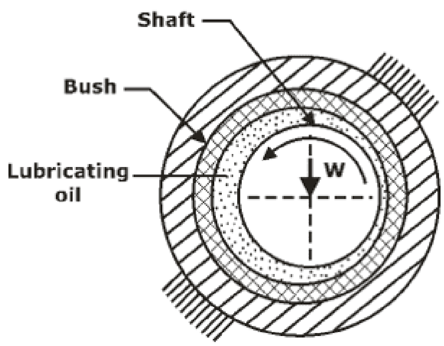
Based on the type of friction, bearings are classified into two categories:

(i). Sliding contact bearings

(ii). Rolling contact bearings

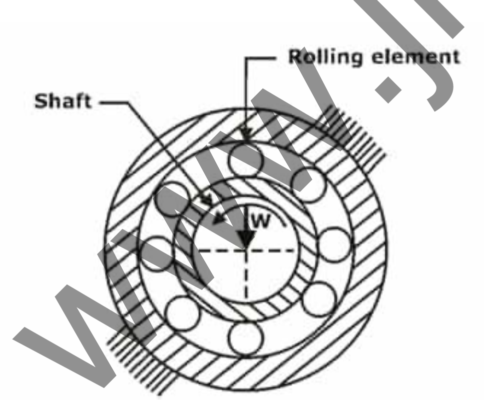
Sliding contact bearings (Plain bearings, Journal bearings or Sleeve bearings):

- Here, the shaft surface slides over the bush surface resulting in friction and wear. Two surfaces are separated by a film of lubricating oil to reduce the friction.
- The bearing material for bush is white metal or bronze.



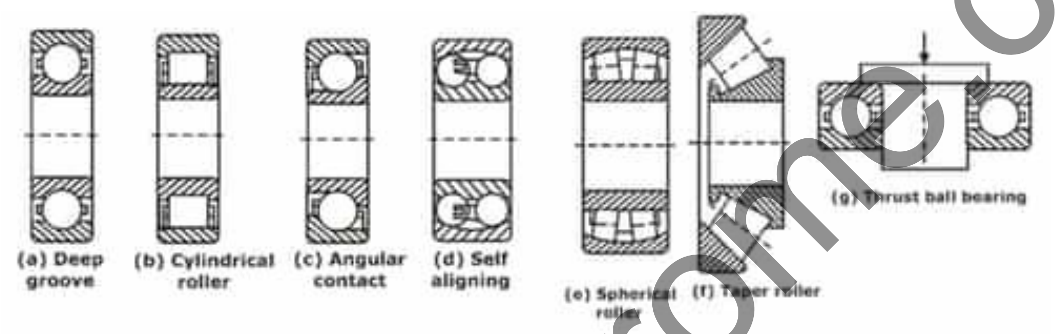
Rolling contact bearing(antifriction bearings):

- These bearings have the rolling elements (balls or rollers) between the surfaces having relative motion between them.
- Here, sliding friction is substituted by rolling friction.



- A rolling contact bearing is made inner and outer races, a rolling component such as ball, roller or needle and a cage. The function of cage is to hold rolling elements together and to provide evenly spaces for rolling elements to avoid clustering of them around the periphery of the shaft.

- Based on the rolling element, the bearings are divided as:
  (i). Ball bearings
  (ii). Cylindrical roller bearings
  (iii). Taper roller bearing
  (iv). Needle bearing.
- Based on the load direction, the bearings are also termed as radial bearing and thrust bearing.
- The types of rolling contact bearings, which are frequently used are shown in figure:



Static load carrying capacity:

- Static load is the load acting on the bearing in the stationary condition of the shaft.
- The static load carrying capacity of a bearing is the static load for which a total permanent deformation of balls and races (at the most heavily stressed point of contact) is equal to 0.0001 times of the ball diameter (D).

Dynamic load carrying capacity:

- It is expressed in terms of the number of revolutions (or hours of service at some given constant speed) completed by the bearing before the first evidence of fatigue crack in balls or races.
- It is based on the hypothesis that the inner race is rotating while the outer race is stationary.

Life of the bearing:

- The rating life of a group of an apparently identical bearings is the number of revolutions completed or exceeded before the first appearance of fatigue crack by the 90% bearings.
- The L_{10} life is the minimum life which 90% of the bearings will reach or exceed before fatigue failure.

Equivalent bearing load:

- In actual applications, two types of force i.e. radial and thrust, is acted in bearings. Thus, these two components acting on the bearing are converted into a single hypothetical load to fulfil the dynamic load carrying capacity conditions.
- It is the constant radial load in radial bearing (or thrust load in thrust bearings), the application of which to the bearing results in same life as the bearing will give under actual condition of forces.

Load life relationship:

The bearing life (L_{10}) in terms of the dynamic load carrying capacity (C) and the equivalent dynamic load (P), is given by:

$$L_{90} = \left(\frac{C}{P}\right)^{p}$$

where, L_{90} = bearing Rated life (million revolutions)

C = dynamic load carrying capacity (in N)

For ball bearings: p = 3 and for roller bearings: p = 10/3.

Bearing having a probability of survival other than 90%:
The reliability is defined as:

$$R = \frac{\text{No. of bearings which have successfully completed L million revolutions}}{\text{Total number of bearings under test}}$$

Formula:

$$\frac{L}{L_{10}} = \left[\frac{\log_e \frac{1}{R}}{\log_e \frac{1}{R_{90}}} \right]^{1/b}$$

where, R_{90}=0.9, and the values of a and b are constant.

Modes of lubrication in sliding contact bearings:

Lubrication is the science of friction reduction with application of a suitable substance called lubricant, between the rubbing surfaces of bodies having relative motion. The two modes of lubrication are thick – and thin film lubrication.

(i) Thick film lubrication: It is condition of lubrication in which two surfaces of the bearing in relative motion are completely separated by a film of fluid. Thick film lubrication is further has two groups:

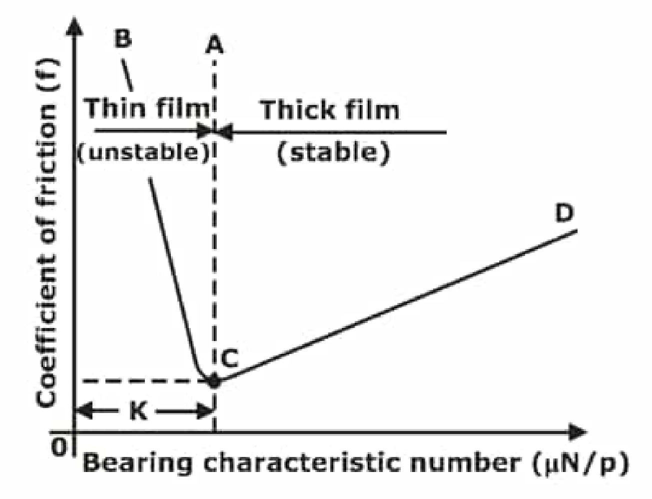
- **Hydrodynamic lubrication:** A system of lubrication where load – supporting fluid film is the result of the shape and relative motion of the sliding surfaces.
- **Hydrostatic lubrication:** Here, the load supporting fluid film, separating the two surfaces is created by an external source such as a pump by supplying sufficient fluid under pressure.

(ii) Thin film lubrication (boundary lubrication):

- It is a lubrication condition in which the lubricant film is relatively thin and partial metal to metal contact occurs.
- The lubrication in door hinges and machine tool slides is example of thin film lubrication.

Mckee's Investigation:

There is transition from thin film lubrication to thick film lubrication as the journal starts from rest and the speed increases. This transition can be better visualized by means of a curve called BCN curve



Bearing characteristic number (BCN)- It is a dimensionless group of parameters given by:

$$(BCN) = \left(\frac{\mu N}{p}\right)$$

where,

μ = absolute viscosity of the lubricant

N = journal speed

P: Unit bearing pressure (load per unit of projected area of bearing)

There are two different regions of the curve i.e. BC and CD

- For region BC, partial metal to metal contact occurs and there are partial patches of lubricant which is the state of thin film lubrication.
- The region CD shows a relatively thick film of lubricant and hydrodynamic lubrication occurs.
- Line AC divides these two modes of lubrication i.e. thin film zone (at left of AC) and the thick film zone (at right of AC).
- The coefficient of friction is minimum at C and the value of the bearing characteristic number corresponding to this minimum coefficient is known the bearing modulus (K).
- In order to avoid seizure, the bearing characteristic number ≥ 5 to 6 times of bearing modulus (K).

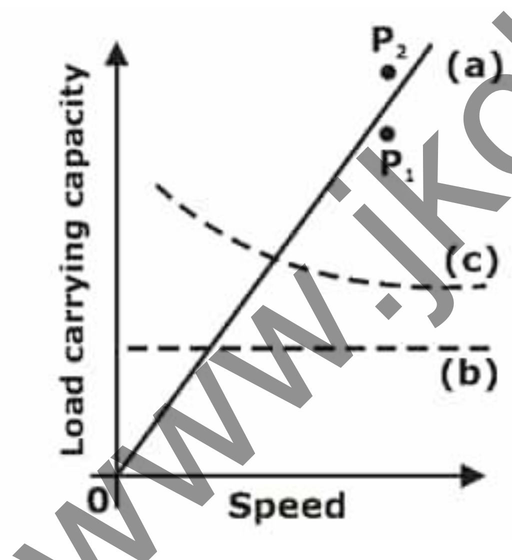
- If the bearing is subjected to fluctuating loads or impact conditions: bearing characteristic number ≥ 15 times of bearing modulus (K).
- The BCN curve is important because it defines the stability of hydrodynamic journal bearings and helps to visualize the transition from boundary lubrication to thick film lubrication.

Temperature rise of the lubricating oil:

- Viscosity of the lubricating oil result in the heat generation due to internal fluid friction. Thus, the frictional work is converted into heat resulting in lubricant temperature increase.

Comparison of sliding and Rolling contact bearings:

- The load carrying capacity of a hydrodynamic bearing is linearly proportional to speed. Point lying below this curve (i.e. point P_1) shows that the infinite life corresponding to this load-speed.
- When the load exceeds and reaching at point P_2 will result in breaking of fluid film and thus causing metal to metal contact.



- In hydrostatic bearings, the load capacity is independent of speed. For a given set of load and speed conditions, the rolling contact bearings result in finite life.
- Rolling contact bearings have poor damping capacity and thus they are vulnerable to shock loads. Hydrodynamic bearings are better suited for

these conditions, which occur in connecting rod and crankshaft applications.

- Rolling contact bearings have need of considerable radial space, while hydrodynamic bearings require more axial space.
- In case of rolling contact bearings, the axes of the journal and bearing are collinear. In hydrodynamic bearings, the journal motion is eccentric with respect to the bearing and the eccentricity is function of the load and speed.
- Due to metal-to-metal contact, rolling contact bearings generate more noise as compared to hydrodynamic bearings.
- The cost of hydrodynamic bearing is much more than that of rolling contact bearing due to additional accessories, like pump, filter and pipelines.
- The maintenance cost is more in hydrodynamic bearings while rolling contact bearings are cheaper.

Design of Springs, Shafts & Gears

Springs:

Spring is defined as an elastic machine element, which deflects under the action of the load and returns to its original shape when the load is removed.

Basic Functions of Spring

- Cushioning, absorbing, or controlling of energy due to shock and vibration. example: Car
- Springs or railway buffers
- To control energy, springs-supports and vibration dampers.
- Measuring forces, spring balances, gauges.
- Storing of energy, in clocks or starters, the clock has spiral type of spring which is wound to coil and then the stored energy helps gradual recoil of the spring when in operation.

Materials Used for Spring: One of the important considerations in spring design is the choice of the spring material. Some of the common spring materials are given below.

- **Hard-drawn wire:** This is cold drawn, cheapest spring steel. Normally used for low stress and static load. The material is not suitable at subzero temperatures or at temperatures above 120°C.

- **Oil-tempered wire:** It is a cold drawn, quenched, tempered, and general purpose spring steel. However, it is not suitable for fatigue or sudden loads, at subzero temperatures and at temperatures above 180°C. When we go for highly stressed conditions then alloy steels are useful.

- **Chrome Vanadium:** This alloy spring steel is used for high stress conditions and at high temperature up to 220°C. It is good for fatigue resistance and long endurance for shock and impact loads.

- **Chrome Silicon:** This material can be used for highly stressed springs. It offers excellent service for long life, shock loading and for temperature up to 250°C.

- **Music wire:** This spring material is most widely used for small springs. It is the toughest and has highest tensile strength and can withstand repeated loading at high stresses. However, it cannot be used at subzero temperatures or at temperatures above 120°C. Normally when we talk about springs we will find that the music wire is a common choice for springs.

- **Stainless steel:** Widely used alloy spring materials.

- **Phosphor Bronze / Spring Brass:** It has good corrosion resistance and electrical conductivity. That's the reason it is commonly used for contacts in electrical switches. Spring brass can be used at subzero temperatures.

Helical spring:

Springs are classified according to their shape.

- The spring can be a helical coil of a wire, a piece of stamping or a flat wound-up strip
- The most popular type of spring is helical spring
- The helical spring is made from a wire, usually of usually of circular cross-section, which is bent in the form of a helix

- There are two types of helical springs – compression spring and extension spring.

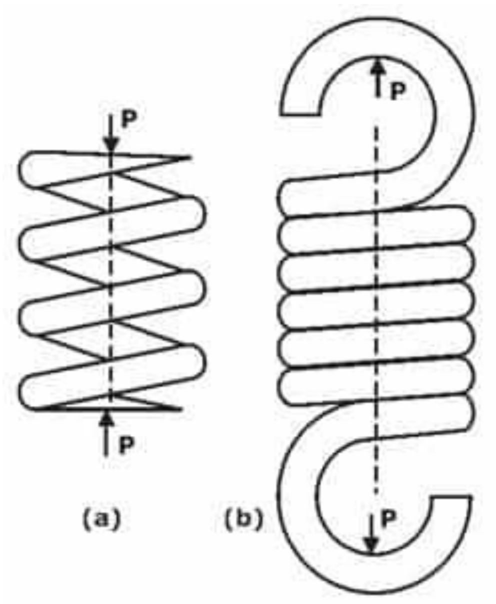


Fig. (a) Helical Compression spring, (b) Helical Extension spring

TERMINOLOGY OF HELICAL SPRING:

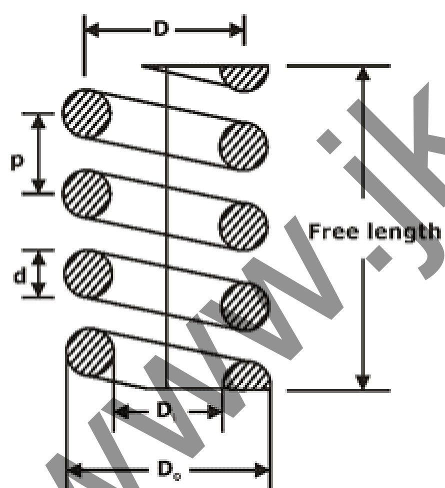


Fig.: Anatomy of helical spring

d = wire diameter of spring (mm),

D_i = inside diameter of spring coil (mm),

D_o=Outside diameter of spring coil (mm),

D=mean coil diameter (mm).

Therefore,

$$D = \frac{D_i + D_o}{2}$$

Spring index (C) = Ratio of mean coil diameter to wire diameter,

SOLID LENGTH:

Axial length of the spring which is so compressed that the adjacent coils touch each other. The spring is completely compressed and no further compression is possible.

Solid Length = Nd

where, N = total number of coils,

COMPRESSED LENGTH:

Axial length of the spring, which is subjected to maximum compressive force.

When the spring is subjected to maximum force, there should be some gap or clearance between the adjacent coils.

The gap is essential to prevent clashing of the coil.

Total gap = (N – 1) x gap between adjacent coils.

FREE LENGTH:

Axial length of an unloaded helical compression spring. In this case, no external force acts on the spring.

STIFFNESS OF THE SPRING:

Force required to produce unit deflection, where, P = axial spring force (N)

δ= axial deflection of the spring corresponding to the force (mm)

Spring Index

- Spring index is defined as the ratio of the mean coil diameter to the wire diameter of the spring and denoted as "C".

C = D/d = Mean coil diameter / Wire diameter

Shear stress correction factor is defined as,

$$K_s = \left(1 + \frac{0.5d}{D}\right) = \left(1 + \frac{0.5}{C}\right)$$

Wahl correction factor is defined as,

$$K = \frac{4C - 1}{4C - 4} + \frac{0.615}{C}$$

Resultant shear stress in the spring wire,

$$\tau = K\left(\frac{8PD}{\pi d^3}\right)$$

Axial deflection 'δ' of the spring:

$$\delta = \frac{8PD^3N}{Gd^4}$$

Stiffness of spring:

$$k = \frac{P}{\delta}$$

$$k = \frac{Gd^4}{8D^3N}$$.

Design of Shaft:

- A shaft is a machine component used to transmit rotary motion or torque. Shafts transmit torque using the gears, belts, pulleys etc.
- Shafts are generally subjected to bending moment, torsion and axial force or a combination of these three.

Axle: A stationary member used as support for rotating elements such as wheels, idler gears, etc.

Spindle: A short shaft or axle (e.g., the headstock spindle of a lathe).

Line shaft: A shaft having connection with a prime mover and transmitting power, to one or many machines, is called the line shaft.

Jackshaft: A short shaft connecting a prime mover with a line shaft, is called the jackshaft.

Flexible shaft: It is a connection used for power transmission between two members having shafts axes at an angle with each other.

Shapes

- Most shafts are round but they can come in many different shapes including square and octagonal. Keys and notches can also result in some unique shapes.

Hollow Shafts Vs Solid Shafts

- Hollow shafts are lighter than solid shafts of comparable strength but are more expensive to manufacture.
- Thus, hollow shafts are used primarily for applications where weight is critical. Example: rear-wheel drive cars propeller shafts should be lightweight to handle speeds within the operating range of the vehicle.

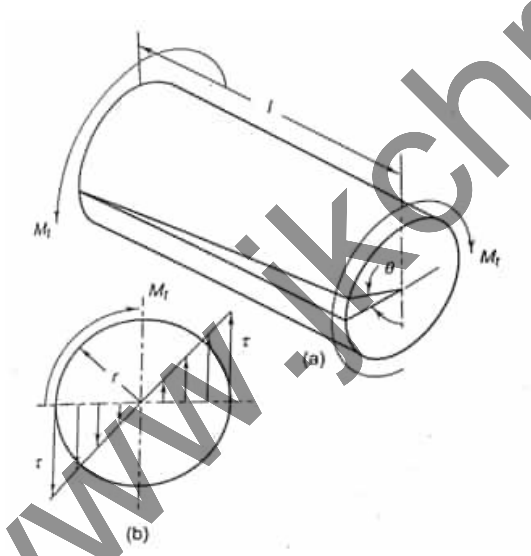
Shaft Design: Shafts are designed based on either the **strength or rigidity or both**.

Design based on Strength: The design in this method ensures that stresses in the shaft does not exceed the material yield stress.

Based on Rigidity: It is based on the principle that shaft deflection due to bending and maximum twist (due to torsion) is within the permissible limits.

Shaft Equations: The following equations given below are general equations and modifying factors such as loading factors, pulsating power source factors, safety factors, and stress concentration factors can be used as per application.

Torsional shear stresses: A machine component acted upon with two equal and opposite couples acting in parallel planes is called under the torsion. The induced internal stresses to resist the twist, are called torsional shear stresses.



The torsion equation is given as :

$$\frac{\tau}{r} = \frac{M_t}{J} = \frac{G\theta}{L}$$

Basic equations in torsion:

- Solid round shaft:

Hollow round shaft: $$\tau = \frac{16M_tD_o}{\pi(D_o^4 - D_i^4)}$$

Strength criteria:

The strength criteria uses the first two terms of torsion equation and design is done on basis that stress induced in the shaft must not exceed the strength of material of shaft.

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

Rigidity criteria: The rigidity criteria uses the last two terms of torsion equation and design is done on the basis that maximum angular twist must exceed a certain value.

$$\theta = \frac{TL}{GJ}$$

Bending stresses in shafts:

A beam or a member is said to be under pure bending when it is subjected to two equal and opposite couples in a plane along the longitudinal axis of the beam (i.e. bending couples) in such a way that magnitude of bending moment remains constant throughout the length of the beam.

ANALYSIS OF BENDING EQUATION:

$$\frac{\sigma_b}{y} = \frac{M_R}{I_{N.A.}} = \frac{E}{R}$$

$$Z_{N.A} \uparrow \implies (\sigma_b)_{max} \downarrow$$

Chances of failure decrease.

- Cross-section which has higher section modulus is best suitable under bending [Bending Stress should be minimum]

DESIGN OF GEARS:

Gears are defined as toothed wheels which transmit power and motion from one shaft to another by means of successive engagement of teeth.

CLASSIFICATION OF GEARS:

Gears are broadly classified into three groups, viz.,

- Spur Gears
- Helical Gears
- Bevel Gears

The torque transmitted by the gears is given by,

$$M_t = \frac{60 \times 10^6 \times P}{2\pi N} \text{ N-mm}$$

where,

P = Power transmitted by gears (kW)

N=speed of rotation (rpm),

The tangential component (P_t),

$$P_t = \frac{2M_t}{d}$$

The radial component,

$$P_r = P_t \tan \alpha$$

The resultant force P_N is given by,

$$P_N = \frac{P_t}{\cos \alpha}$$

BEAM STRENGTH OF GEAR DESIGN:

Lewis equation is considered as the basic equation in the design of gears.

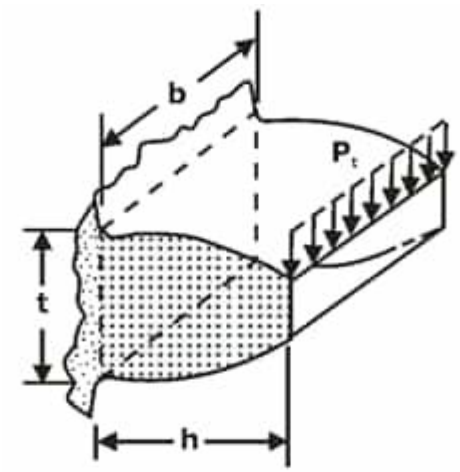
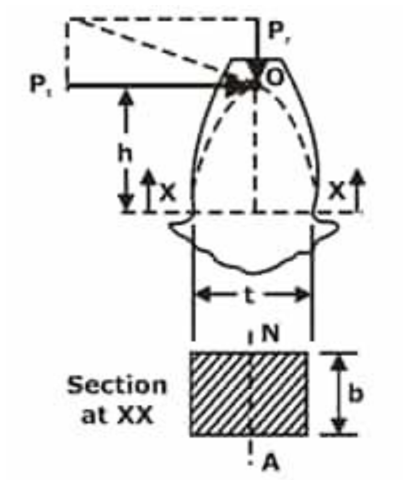


Fig.: Gear tooth as cantilever beam

The Lewis equation is based on the following assumptions:

- The effect of the radial component, which induces compressive stresses, is neglected
- It is assumed that the tangential component is uniformly distributed over the face width of the gear
- The effect of stress concentration is neglected
- It is assumed that at any time, only one pair of teeth is in contact and takes the total load

The cross-section of the tooth varies from the free end to the fixed end. Therefore, a parabola is constructed within the tooth profile.



The equation is written as,

$$P_t = mb\sigma_b Y$$

Y is called the Lewis form factor,

$$Y = \left(\frac{t^2}{6hm}\right)$$

When the stress reaches the permissible magnitude of bending stresses, the corresponding force P_t is called the beam strength.

Therefore, the beam strength (S_b) is the maximum value of the tangential force that the tooth can transmit without bending failure.

$$S_b = mb\sigma_b Y$$

This equation is known as Lewis equation.

It is observed that m and b are same for pinion as well as for gear.

- When the different materials are used, the product of decides the weaker between the pinion and gear.
- The Lewis form factor is always less for a pinion compared with gear.
- When the same material is used for the pinion and gear, the pinion is always weaker than the gear.

The service factor C_s is defined as,

$$C_s = \frac{\text{maximum torque}}{\text{rated torque}}$$

The effective load between two meshing teeth is given by,

$$P_{eff} = \frac{C_s P_t}{C_v}$$

In order to avoid failure of gear tooth due to bending,

$S_b > P_{eff}$

WEAR STRENGTH OF GEAR DESIGN:

In order to avoid this type of failure, the proportions of gear tooth and surface properties, such as surface hardness should be selected in such a way that the wear strength of the gear tooth is more than the effective load between the meshing teeth.

$$P_t = bQd'_pK$$

where,

Q ratio factor is defined as,

$$Q = \frac{2z_g}{z_g + z_p}$$

K is a load-stress factor is defined as,

$$K = \frac{\sigma_c^2 \sin\alpha \left(\frac{1}{E_1} + \frac{1}{E_2}\right)}{1.4}$$

d'p=diameter of pinion,

b= Width of gear,

When the tangential force is increased, the contact stress also increases.

Pitting occurs when the contact stress reaches the magnitude of the surface endurance strength. The corresponding value of P_t is called wear strength. Therefore, the wear strength is the maximum value of the tangential force that the tooth can transmit without pitting failure.

Replacing P_t by S_w,

- $$S_w = bQd'_pK$$

  S_w=wear strength of the gear tooth(N).

- The expression for the load-stress factor K can be simplified when both the gears are made of steel with a 20° pressure angle,

$$K = 0.16\left(\frac{BHN}{100}\right)^2$$

Couplings, Cotters, Knuckle & Belt Drives

Power transmission by belts:

- Belt drive is combination of two pulleys on each shaft with an endless belt wrapped around them having some initial tension.
- Power is transmitted from the driver pulley to the belt and from the belt to the driven pulley with the help of friction.
  - Friction between belt and pulley surface limits the maximum power that can be transmitted. Once limiting value is exceeded, slipping of belt occurs.
  - Rectangular cross-section belts are known as **flat belts** while V-belts have trapezoidal cross-section.
  - **Flat belts** are used to transmit the moderate amount of power between shafts less than 6m apart.
  - **V-belts** are used along with pulleys having a similar cross-section as that of the belt. To further increase the power transmission capacity, multiple V-belt system (having more than one V belt & grooves in the system) is used

(i). There are two types of drives—rigid and flexible.

(ii). Gear drives are called rigid or non-flexible drives where there is direct contact between the driving and driven shafts through the gears.

While in flexible drives, there is an intermediate link such as belt, rope or chain between the driving and driven shafts. Due to link being flexible, the drives are termed as the 'flexible' drives. Thus, Belt drives are the flexible drives.

Advantages and disadvantages of Belt drives:

Advantages: Belt drives offer the following advantages compared with other types of drives:

(i). They can be used for power transmission between the axes of driving and driven shafts having considerable distances between them.

(ii). The belt drive operation is smooth and silent.

(iii). Simple design and low initial cost.

Disadvantages:

(i). Have large dimensions and thus occupying space.

(ii). Due to belt slip, the velocity ratio is not constant.

(iii). They impose heavy loads on shafts and bearings.

CLASSIFICATION OF THE BELTS:

(i). Flat belts

(ii). V-belts

TYPES OF FLAT BELT DRIVE:

(i). Open belt drive (O.B.D.):

OBD ⇒ Direction of rotation are same ⇒ like internal gear

(ii). Cross belt drive (C.B.D.):

CBD ⇒ Direction of rotation are opposite = like external gear

(iii). Compound belt drive:

Compound ⇒ To obtain higher speed reduction (compound gear train).

Open belt drive (O.B.D.):

Let D_1 = Diameter of driven pulley

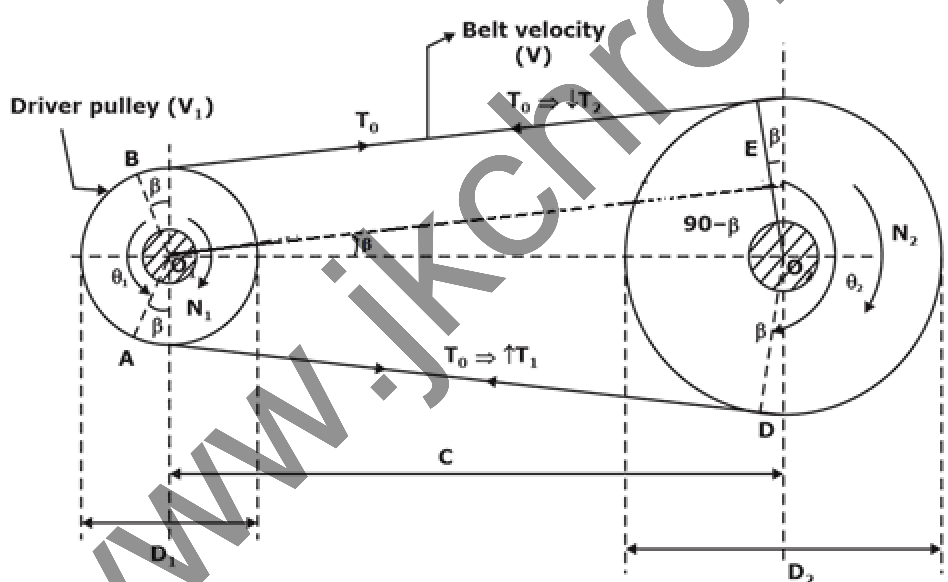
D_2 = Diameter of driven pulley or follower pulley

C = Centre distance between two parallel shafts

V_1 = Linear velocity of driven pulley

V = Linear velocity of belt

V_2 = Linear velocity of driven pulley,



$$\beta = \sin^{-1}\left[\frac{D_2 - D_1}{2C}\right] \times \frac{\pi}{180}$$

$$L_{OBD} = 2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_2 - D_1)^2}{4c}$$

⇒ This drive is not suitable for smaller center distance it should be medium.

Due to centrifugal force:

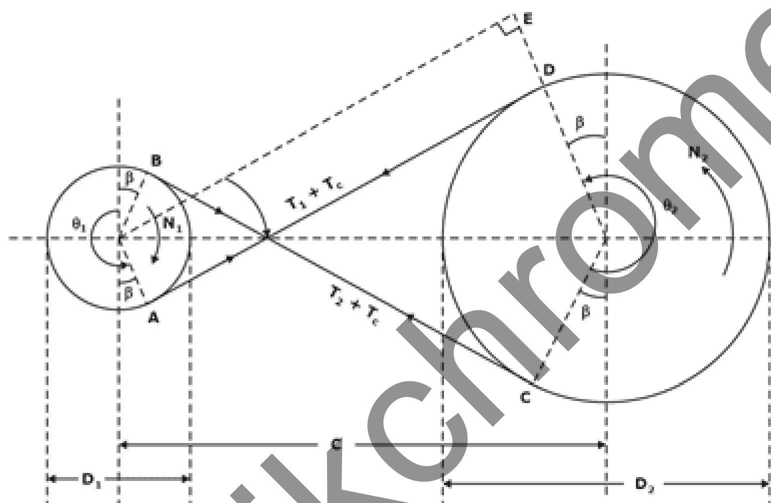
Total tension in tight side = $T_1 + T_C$

Total tension in slack side = $T_2 + T_C$

$T_1 + T_C \leq T_{max}$

$$T_{max.} = \sigma_{Per} \times A = \sigma_{Per} \times b \times t$$

Cross belt drive:



$$\beta = \sin^{-1}\left[\frac{D_2 + D_1}{2C}\right] \times \frac{\pi}{180}$$

Length of the belt:

$$L_{CBD} = 2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_2 + D_1)^2}{4C}$$

Comparison between open belt drive & cross belt drive:

Parameter	Open belt drive	Cross belt drive
1. Direction of rotation of pulleys	Same direction	Opposite direction
2. Centre distance	Suitable for medium center distance $C \geq C_{Min}$	Suitable for both smaller & Medium center distance
3. Idler pulleys	Are required when $C \leq C_{Min}$	Are not required
4. β	$\sin^{-1}\left(\frac{D_2 - D_1}{2C}\right) \times \frac{\pi}{180}$	$\sin^{-1}\left(\frac{D_2 + D_1}{2C}\right) \times \frac{\pi}{180}$
5. Angle of contact	$\theta_1 = \pi - 2\beta$ $\theta_2 = \pi + 2\beta$ or $2\pi - \theta_1$	$\theta_1 = \theta_2 = \pi + 2\beta$
6. Power transmission capacity	Less	More
7. Service life	More	Less
8. Length of the belt	$2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_2 - D_1)^2}{4C}$	$2C + \frac{\pi}{2}(D_1 + D_2) + \frac{(D_2 + D_1)^2}{4C}$

$$\text{Velocity ratio} = \frac{N_2}{N_1} = \frac{\text{Speed of driver pulley}}{\text{Speed of driver pulley}}$$

If slip and belt thickness is taking into consideration.

Then,

$$V.R = \frac{N_2}{N_1} = \left(\frac{D_1 + t}{D_2 + t}\right)\left(1 - \frac{S}{100}\right)$$

Initial tension (T_o):

- Initial tension is the tension develop in the belt when it is in the stationary cond.
- It is provided in the belt by taking a length of the belt less than actual required length.
- In presence of initial tension, power transmission capacity of belt drive increases hence it is useful.

$$T_o = \frac{T_1 + T_2}{2}$$

Centrifugal Tension (Tc):

Centrifugal tension is the addition tension develop in the belt in presence of centrifugal force acting on the belt.

$T_{max} = T_1 + T_c$

CONDITION FOR MAXIMUM POWER TRANSMISSION (Pmax):

(i).

$$T_c = \frac{T_{max}}{3}$$

(ii). $T_1 = 2T_c$

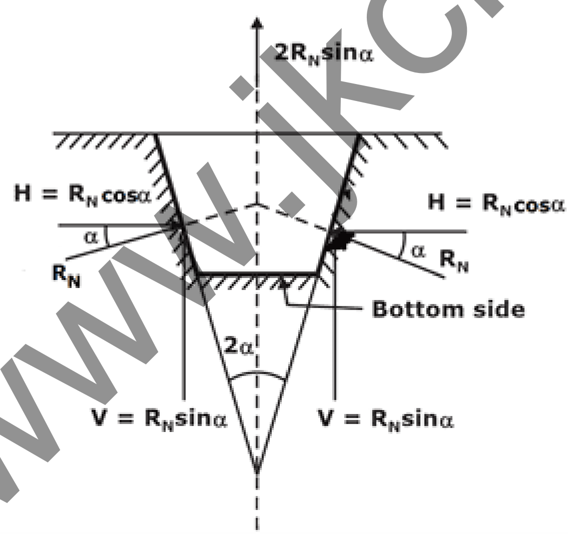
(iii).

$$V_{max} = \sqrt{\frac{T_{max}}{3m}}$$

V-BELT DRIVE:

V-belt are used to transmit power between two parallel shaft which are at smaller centre distance and rotating in same direction.

Cross section = Trapezoidal cross section



Ration of belt tensions:

$$\frac{T_1}{T_2} = e^{\frac{\mu\theta}{\sin\alpha}}$$

$$\left(\frac{T_1}{T_2}\right)_{V-belt} > \left(\frac{T_1}{T_2}\right)_{flat\ belt}$$

Practical application:

(i). air compressor

(ii). Automobile radiator fans

(iii). M/c tools like shaper, milling M/c etc.

Effect of creep:

(i). Creep results in the less angular velocity of the driven pulley than that calculated by considering the ratio of diameters of pulleys.

(ii). The efficiency reduction by 1 to 2% due to creep.

Knuckle joint:

- Knuckle joint is used to connected two rods whose axes either coincide or intersect and lie in one plane.
- The knuckle joint is used to transmit axial tensile force. The constructions of this joint permits limited angular movement between rods, about the axis of the pin.

Applications of knuckle joints are as follows:

- Joints in value mechanism of a reciprocating engine.
- Fulcrum for the levers.
- Joints between the links of a bicycle chain.

Advantages of Knuckle joints:

- The joint is simple to design and manufacture.

- Few parts in the knuckle joint reduces cost and improves reliability.
- The assembly or dismantling of the parts of a knuckle joint is quick and simple.

Cotter Joint:

- It is used to connect two co-axial rods subjected to either axial tensile' or axial compressive force.
- It is not suitable to connect rotating shafts and transmitting torque.

Applications of cotter joint are as follows: Its applications include:

(i). Piston rod and crosshead joint in a steam engine.

(ii). the piston rod and the tail or pump rod joint.

Couplings:

These are mechanical components used for connecting two rotating shafts and transmitting torque from one shaft to the other shaft.

Example: connection between the electric motor output shaft and input shaft of a hydraulic pump.

Types of Shafts Couplings

Rigid Couplings:

Perfectly aligned shafts are connected using the couplings. Couplings are simple and inexpensive.

Rigid Couplings are of following types:

- Sleeve or Muff Coupling
- Clamp or Split-muff or Compression Coupling
- Flange Coupling

Flexible Couplings

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

Flexible Couplings are of the following types:

- Bushed pin type Coupling
- Oldham Coupling
- Universal Coupling