Short Notes on Machine Design

Static Load

- A static load is a mechanical force applied slowly to an assembly or object.Load does not change in magnitude and direction and normally increases gradually to a steady value
- This force is often applied to engineering structures on which peoples' safety depends on because engineers need to know the maximum force a structure can support before it will collapse.

Dynamic load

- A dynamic load, results when loading conditions change with time. Load may change in magnitude for example, traffic of varying weight passing a bridge.
- Load may change in direction, for example, load on piston rod of a double acting cylinder. Vibration and shock are types of dynamic loading.

Factor of safety (F.O.S):

- The ratio of ultimate to allowable load or stress is known as factor of safety i.e. The factor of safety can be defined as the ratio of the material strength or failure stress to the allowable or working stress.
- The factor of safety must be always greater than unity. It is easier to refer to the ratio of stresses since this applies to material properties.

F.O.S = failure stress / working or allowable stress

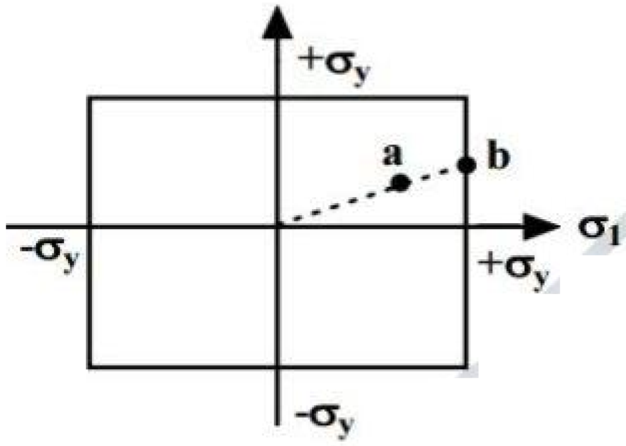
Static Failure Theories

Maximum Principal Stress Theory (Rankine Theory):

- The principal stresses ***σ1 (maximum principal stress)***, ***σ2 (minimum principal stress)*** or **σ3** exceeds the yield stress, yielding would occur.
- For two dimensional loading situation for a ductile material where tensile and compressive yield stress are nearly of same magnitude:

$$\sigma_1 = \pm\, \sigma_y$$

$$\sigma_2 = \pm\sigma_y$$



- Yielding occurs when the state of stress is at the boundary of the rectangle.

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

- If ε1 and ε2 are maximum and minimum principal strains corresponding to σ1 and σ2, in the limiting case:

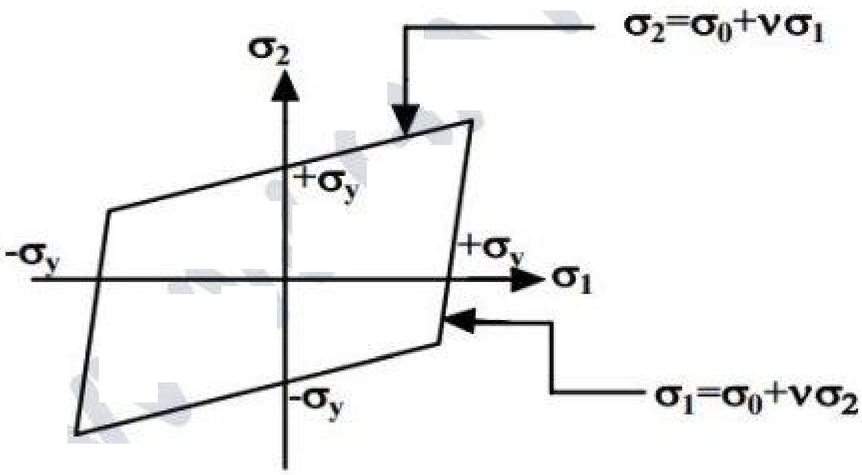
$$\varepsilon_1 = \frac{1}{E}(\sigma_1 - \nu\sigma_2) \qquad |\sigma_1| \geq |\sigma_2|$$

$$\varepsilon_2 = \frac{1}{E}(\sigma_2 - \nu\sigma_1) \qquad |\sigma_2| \geq |\sigma_1|$$

$$E\varepsilon_1 = \sigma_1 - \nu\sigma_2 = \pm\sigma_0$$

$$E\varepsilon_2 = \sigma_2 - \nu\sigma_1 = \pm\sigma_0$$

- Boundary of a yield surface in Maximum Strain Energy Theory is given below



Maximum Shear Stress Theory (Tresca Theory):

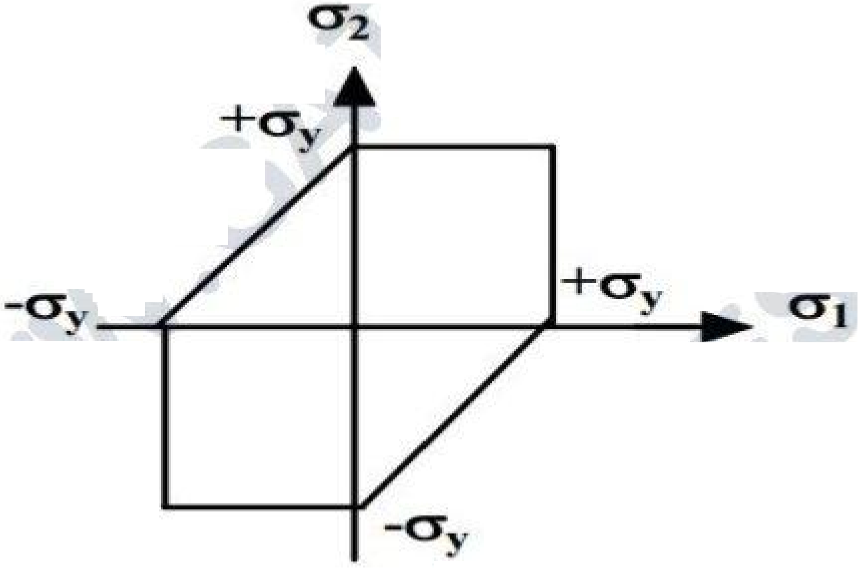
- At the tensile yield point σ2= σ3 = 0 and thus maximum shear stress is σy/2.

$$\sigma_1 - \sigma_2 = \pm\sigma_y$$

$$\sigma_2 - \sigma_3 = \pm\sigma_y$$

$$\sigma_3 - \sigma_1 = \pm\sigma_y$$

- Yield surface corresponding to maximum shear stress theory in biaxial stress situation is given below :



Maximum strain energy theory (Beltrami's theory):

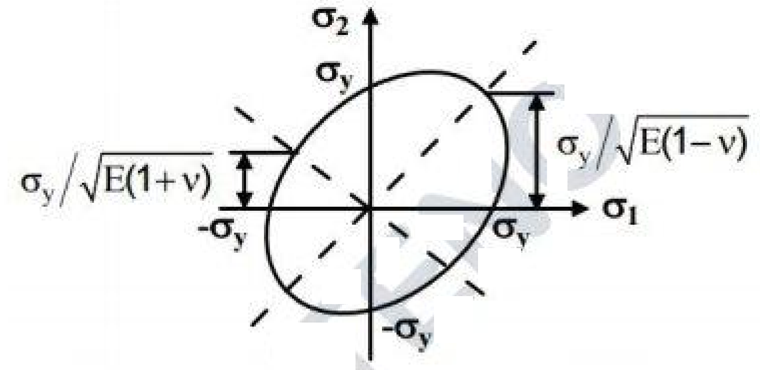
- Failure would occur when the total strain energy absorbed at a point per unit volume exceeds the strain energy absorbed per unit volume at the tensile yield point.

$$\frac{1}{2}\left(\sigma_1\varepsilon_1 + \sigma_2\varepsilon_2 + \sigma_3\varepsilon_3\right) = \frac{1}{2}\sigma_y\varepsilon_y$$

$$\sigma_1^2 + \sigma_2^2 + \sigma_3^2 - 2\upsilon\left(\sigma_1\sigma_2 + \sigma_2\sigma_3 + \sigma_3\sigma_1\right) = \sigma_y^2$$

$$\left(\frac{\sigma_1}{\sigma_y}\right)^2 + \left(\frac{\sigma_2}{\sigma_y}\right)^2 - 2\nu\left(\frac{\sigma_1\sigma_2}{\sigma_y^{\,2}}\right) = 1$$

- Above equation results in Elliptical yield surface which can be viewed as:



Distortion energy theory (Von Mises yield criterion):

- Yielding would occur when total distortion energy absorbed per unit volume due to applied loads exceeds the distortion energy absorbed per unit volume at the tensile yield point. Total strain energy E_T and strain energy for volume change E_V can be given as:

$$E_T = \frac{1}{2}\left(\sigma_1\varepsilon_1 + \sigma_2\varepsilon_2 + \sigma_3\varepsilon_3\right) \text{ and } E_V = \frac{3}{2}\sigma_{av}\varepsilon_{av}$$

$$E_d = E_T - E_V = \frac{2(1+\nu)}{6E}\left(\sigma_1^{\,2} + \sigma_2^{\,2} + \sigma_3^{\,2} - \sigma_1\sigma_2 - \sigma_2\sigma_3 - \sigma_3\sigma_1\right)$$

At the tensile yield point, $\sigma 1 = \sigma y$, $\sigma 2 = \sigma 3 = 0$ which gives,

$$E_{dy} = \frac{2(1+\nu)}{6E}\sigma_y^{\,2}$$

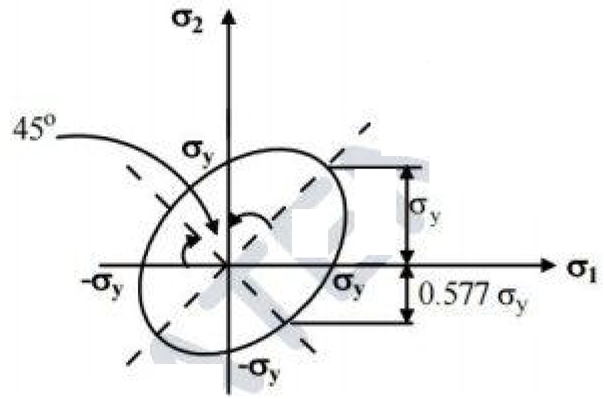
The failure criterion is thus obtained by equating Ed and Edy , which gives

$$\left(\sigma_1 - \sigma_2\right)^2 + \left(\sigma_2 - \sigma_3\right)^2 + \left(\sigma_3 - \sigma_1\right)^2 = 2\sigma_y^2$$

In a 2-D situation if $\sigma 3 = 0$, so the equation reduces to,

$$\left(\frac{\sigma_1}{\sigma_y}\right)^2 + \left(\frac{\sigma_2}{\sigma_y}\right)^2 - \left(\frac{\sigma_1}{\sigma_y}\right)\left(\frac{\sigma_2}{\sigma_y}\right) = 1$$

- This is an equation of ellipse and yield equation is an ellipse.
- This theory is widely accepted for ductile materials



Cotter and Knuckle Joints

A cotter joint is a temporary fastening and is used to connect rigidly two co-axial road or bars which are subjected to axial tensile or compressive forces.

Socket and Spigot Cotter Joints

In a socket and spigot cotter joint, one end of the rods is provided with a socket type of end as shown in figure and the other end of the rod is inserted into a socket. The end of the rod which goes into a socket is also called spigot.

Failures in Socket and Spigot Cotter Joints

Failure Cases	Tensile Force
Failure of the rod in tension	$P = \frac{\pi}{4} \times d^2 \times \sigma_t$
Failure of spigot in tension across the weakest section	$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$
Failure of the rod or cotter in crushing	$P = d_2 \times t \times \sigma_t$
Failure of the socket in tension across the slot	$P = \left\{\frac{\pi}{4}\left[(d_1)^2 - (d_2)^2\right] - (d_1 - d_2)\,t\right\}\sigma_t$
Failure of cotter in shear	$P = 2b \times t \times \tau$
Failure of the socket collar in crushing	$P = (d_4 - d_2)\,t \times \sigma_c$
Failure of rod end in shear	$P = 2(d_4 - d_2)\,c \times \tau$
Failure of spigot collar in crushing	$P = 2a \times d_2 \times \tau$
Failure of the spigot collar in shearing	$P = \frac{\pi}{4}\left[(d_3)^2 - (d_2)^2\right]\sigma_c$
	$P = \pi d_2 \times t_1 \times \tau$

Failures in Sleeve and Cotter Joints

Failure Cases	Tensile Force
Failure of the rod in tension	$P = \frac{\pi}{4} \times d^2 \times \sigma_t$
Failure of rod in tension across the weakest section	$P = \left[\frac{\pi}{4}(d_2)^2 - d_2 \times t\right]\sigma_t$
Failure of the rod or cotter in crushing	$P = d_2 \times t \times \sigma_t$
Failure of sleeve in tension across the slot	$P = \left\{\frac{\pi}{4}\left[(d_1)^2 - (d_2)^2\right] - (d_1 - d_2)\,t\right\}\sigma_t$
Failure of cotter in shear	$P = 2b \times t \times \tau$
Failure of rod end in shear	$P = 2a \times d_2 \times \tau$
Failure of sleeve end in shearing	$P = 2(d_1 - d_2)\,c \times \tau$

Knuckle Joint

- It is used to connect two rods whose axis either coincide or intersect and lie in one plane.
- This joint generally found in the link of a cycle chain tie rod joint for roof truss, valve rod joint with eccentric rod tension link in bridge structure, lever and rod connection of various types.
- It is sometimes also called forked pin joint.

Failures in Knuckle Joint

Failure Cases	Tensile Force
Failure of the solid rod in tension	$P = \frac{\pi}{4} \times d^2 \times \sigma_t$
Failure of the knuckle pin in shear	$P = 2\frac{\pi}{4}(d_2)^2 \tau$
Failure of the single eye or rod end in tension	$P = (d_2 - d_1)\, t \times \sigma_t$
Failure of the single eye or rod end in shearing	$P = (d_2 - d_1)\, t \times \tau$
Failure of forked end in tension	$P = (d_2 - d_1)\, 2t_1 \times \sigma_t$
Failure of the forked end in crushing	$P = d_1 \times 2t_1 \times \sigma_c$

- To connect the transmission shaft to rotating machine elements like pulley, gear, sprocket or flywheel.
- Cotter and knuckle joints are not used for connect

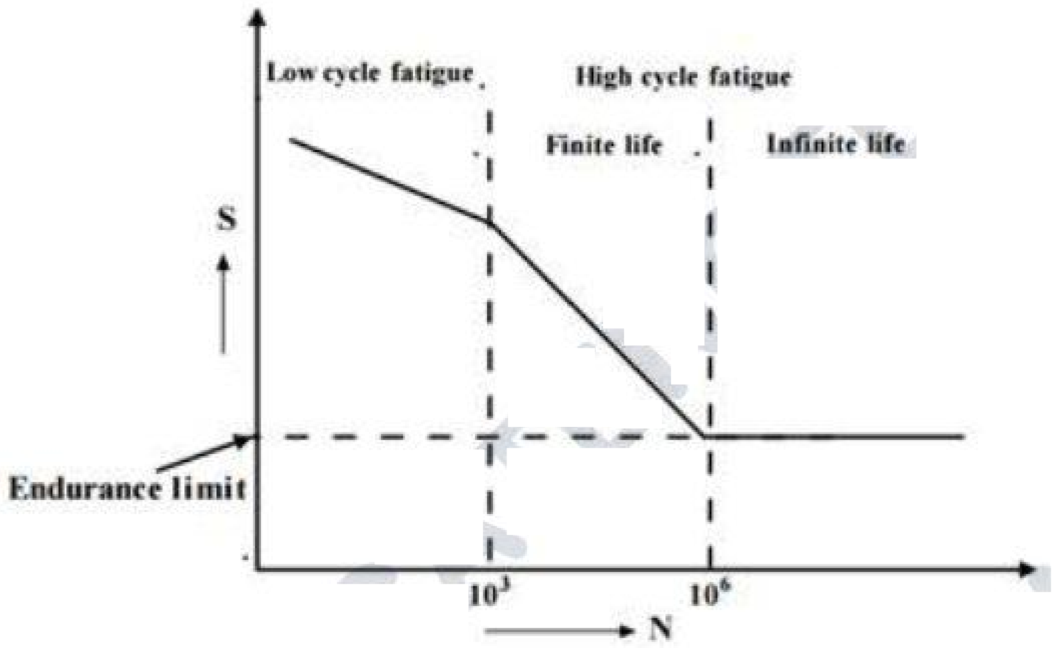
Fatigue

- Fatigue loading is primarily the type of loading which causes cyclic variations in the applied stress or strain on a component.
- Variable loading due to: -Change in the magnitude of applied load Example: punching or shearing operations- -Change in direction of load application Example: a connecting rod -Change in point of load application Example: a rotating shaft.

Fatigue Failure:

- Machine elements subjected to fluctuating stresses usually fail at stress levels much below their ultimate strength and in many cases below the yield point of the material too.
- These failures occur due to very large number of stress cycle and are known as fatigue failure.
- Fatigue failures are influenced by
    - Nature and magnitude of the stress cycle

- Endurance limit
- Stress concentration
- Surface characteristics



Riveted joints:

- There are two basic components of riveted joints:
  - Rivets
  - Two or more plates.
- The popular materials for the rivets are: *Steel, Brass, Aluminium* & *Copper* as per the requirement of the application for fluid tight joints the steel rivets are used

Welded Joints

- It is a permanent joint.
- When the two parts are joined by heating to a suitable temperature with or without application of pressure.

Welding Processes

- **Fusion Welding**
- **Thermit Welding**
- **Gas Welding**
- **Electric Arc Welding**
- **Forge Welding**

Types of Welded Joints:

Lap Joint or Fillet Joint

- In lap joint, overlapping the plate and welding the edge of the plates takes place in welding process.
- The strength of different types of fillet joint can be given according to their welding process as

- **Shear strength in parallel fillet weld,**

$$\tau = \frac{P}{0.707\, hl} \text{ or } P = 0.707\, hl\, \tau$$

where, *P* = Tensile force on the plates

h = Leg of the weld

l = Length of the weld

τ = Permissible shear stress

- For double parallel fillet weld,

$$P = 1.414\, h/\tau$$

- **Strength of Transverse Fillet Weld**
  - *p* = Throat area Allowable tensile stress
  - $= 0.707\, s \times l \times \sigma_t$
- For double transverse fillet joint

$$P = 1.414\, hl\sigma_t$$

Special Cases of Fillet Welded Joint

- **Circular Fillet Weld Subjected to Torsion**
  - Shear stress $\tau = \frac{2T}{\pi t d^2}$

or $$\tau_{max} = \frac{2.83T}{\pi h d^2}$$

where, *T* = Torque acting or rod

h = size of weld

t = Throat thickness

Circular Fillet Weld Subjected to Bending Moment

- **Bending stress:**

$$\sigma_b = \frac{4M}{\pi t d^2}$$

$$\sigma_{b\,(max)} = \frac{5.66\,M}{\pi h d^2}$$

- **Long Fillet Weld Subjected to Torsion**
  - Shear stress: $\tau = \frac{3T}{tl^2}$

$$\tau_{max} = \frac{4.242\,T}{hl^2}$$

Butt Joint

- **Strength of Butt Joint**
  - For single V – butt joint,

$$P = t \times l \times \sigma_t$$

  - For double V-butt joint,

$$P = (t_1 + t_2) l \times \sigma_t$$

Eccentric Loaded Welded Joints

When the shear and bending stresses are simultaneously present in a joint.

- Maximum normal stress

$$\sigma_{t_{max}} = \frac{\sigma_b}{2} + \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

- Maximum shear stress

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma_b)^2 + 4\tau^2}$$

- Direct or primary shear stress

$$\tau_1 = \frac{\text{Load}}{\text{Throat area}} = \frac{\text{P}}{\text{2tl}}$$

$$\tau_1 = \frac{P}{1.414\ hl}$$

Strength of Bolted Joint

- Maximum tensile stress in the bolt

$$\sigma_t = \frac{P}{\left(\frac{\pi}{4} d_c^2\right)}$$

where, d_c = Core diameter

$$\frac{\sigma_{yt}}{f_s} = \frac{P}{\frac{\pi}{4} d_c^2}$$

Torque Requirement for Bolt Tightening

$$M_t = \frac{P_i d_m}{2} \left(\frac{\mu \sec \theta + \tan \alpha}{1 - \mu \sec \theta \tan \alpha} \right)$$

where,

Pi = Pretension in bolt, $d_m = 0.9\ d$

d = Nominal diameter

For ISO metric screw thread ϑ = 30°

For ISO metric α = 25°

Eccentric Load on Bracket with Circular Base

- If there are n number of bolts, then load in a bolt

$$w_{b_1} = \frac{2wL\ (R - r\ \cos\alpha)}{n\ (2R^2 + r^2)}$$

- In above case when $n = 4$

$$w_b = \frac{w.L\,(R - \alpha\cos\alpha)}{2\,(2R^2 + r^2)}$$

Maximum load in bolt

$$(w_{b_1})_{max} = \frac{2wL}{n}\left(\frac{R+r}{2R^2+r^2}\right)$$

where, cos α = -1

Factor of Safety (FOS) in Bolted Joints

- It is defined as the ratio of failure stress to allowable stress.

$$\text{FOS} = \frac{\text{Failure stress}}{\text{Allowable stress}}$$

- For ductile material,

$$\text{FOS} = \frac{S_{yt}}{\sigma}$$

- For brittle material,

$$\text{FOS} = \frac{S_{ut}}{\sigma}$$

where, Syt = Yield strength of component material

Sut = Ultimate tensile stress of components material

σ = Allowable stress.

Stress concentration Factor

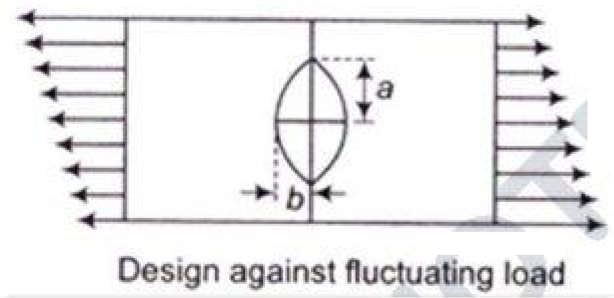
- It is defined as the ratio of highest value of actual stress near discontinuity to nominal stress obtained by elementary equations for minimum cross-section. It is denoted by k_t.

$$k_t = \frac{\text{Highest value of actual stress near discontinuity}}{\text{Nominal stress obtained by elementary equations}}$$

$$or\ k_t = \frac{\sigma_{max}}{\sigma_0} = \frac{\tau_{max}}{\tau_0}$$

where, σ_0, τ_0 = Nominal stresses

- The magnitude of stress concentration factor depends upon the geometry of the component.
- In this case, k_t = 1+2(a/b)



Design against fluctuating load

where,

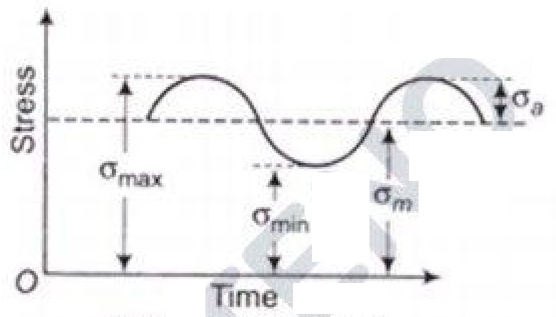
a = Semi-axis of ellipse perpendicular to the direction of load

b = Semi-axis of ellipse parallel to the direction of load

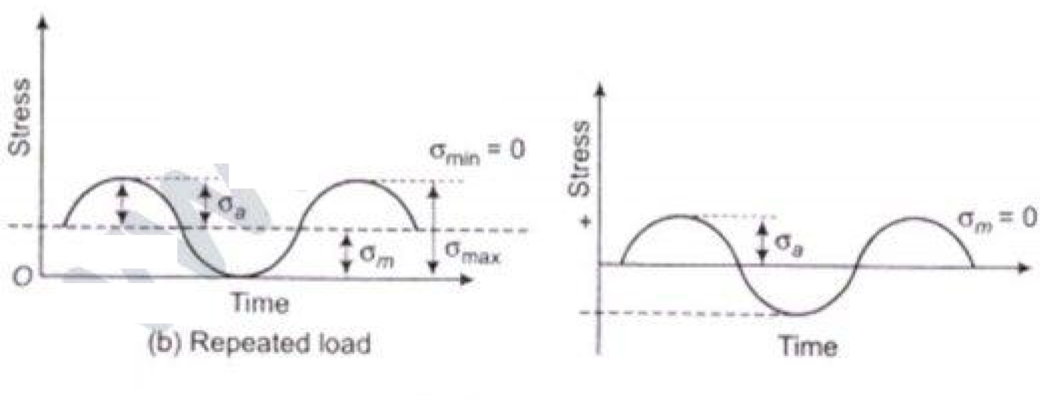
- If *b* = 0 then, hole is like as very sharp crack then, $k_t = \infty$
- If *a* = *b* then, hole becomes a circular hole then, k_t = 1+2=3

Fluctuating Load

- It is defined as the load, of which magnitude and direction both changes with respect to time.



(a) Fluctuating load



(b) Repeated load

σ_m = Mass stress, σ_a = Stress amplitude

- Mass stress and stress amplitude

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

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

- For repeated stress, $\sigma_{min} = 0, \sigma_m = \frac{\sigma_{max}}{2}$ and $\sigma_a = \frac{\sigma_{max}}{2}$
- For reversed stress, $\sigma_m = \frac{1}{2}(\sigma_{max} - \sigma_{min}) = 0$

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

Gears

Gear can be defined as the mechanical element used for transmitted power and rotary motion from one shaft to another by means of progressive engagement of projections called teeth.

Classification of Gears

- Spur Gear
- Helical Gear
- Bevel Gear
- Worm Gear

Spur Gear

In spur gears, teeth are cut parallel to axis of the gear.

- Circular pitch

$$P = \frac{\pi d}{z}$$

- Diametrical pitch

$$P = \frac{z}{d}$$

- Module

$$m = \frac{d}{z}$$

- Torque transmitted by gear

$$M_t = \frac{60 \times 10^6 (kW)}{2\pi n}$$

- **Dynamic load or incremental dynamic load**

$$P_d = \frac{21v(ceb + P_t)}{21v + \sqrt{ceb + P_t}}$$

Where, v = Pitch line velocity

c = Deformation factor

b = Face width of tooth

P_t = Tangential force due to rated torque. e = Sum of errors between two meshing teeth

- **Estimation of module based on beam strength**

$$m = \left[\frac{60 \times 10^6}{\pi} \left\{ \frac{(kw) c_s (f_s)}{z\, n c_v \left(\frac{b}{m}\right)\left(\frac{S_u}{3}\right) Y} \right\} \right]^{-1/3}$$

Where, c_s = Service factor,

c_v = Velocity factor

f_s = Factor of safety,

n = Speed (rpm)

- **Estimation of module based on wear strength**

$$m = \left[\frac{60 \times 10^6}{\pi} \left\{ \frac{(kw) c_s (f_s)}{z_p^2 n_p c_v \left(\frac{b}{m}\right) pK} \right\} \right]^{-1/3}$$

Helical Gear

- The teeth of helical gear cut in the form of helix or an angle on the pitch cylinder.

$$P_n = \frac{P}{\cos\psi}$$

Where, P_n = Normal diametrical pitch

P = Transverse diametrical pitch

Ψ = Helix angle

m_n = m cos ψ

m_n = Normal module

m = transverse module

- Axial pitch

$$P_a = \frac{P}{\tan\psi}$$

- Pitch circular diameter

$$d = \frac{zm_n}{\cos\psi}$$

- Tooth proportions
  - Addendum $h_a = m_n$
  - Dedendum hf = 1.25 m_n
  - Clearance c = 0.25 m_n

- Addendum circle diameter $d_a = d + 2h_a$ or

$$d_a = \frac{z\,m_n}{\cos\psi} + 2m_n$$

- Dedendum circle diameter

$$d_f = \frac{z\,m_n}{\cos\psi} + 2.5m_n$$

- Component of tooth forces

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

$$P_r = P_t \left[\frac{\tan \alpha_n}{\cos \psi} \right]$$

$$P_a = P_t \tan \psi$$

- Beam strength of helical gear

$$\mathbf{S_b = m_n\, b\, \sigma_b\, Y}$$

Where, *m* = Module,

σ_b = Permissible bending stress

y = Lawis form factor

- Dynamic load or incremental dynamic load P_d

$$P_d = \frac{21v\,(ceb \cos^2 \psi + P_t) \cos \psi}{21v + \sqrt{(ceb \cos^2 \psi + P_t)}}$$

Where, *e* = Sum of errors,

C = Deformation factor

- Wear strength of helical gear

$$S_w = \frac{bQ\, d_p K}{\cos^2 \psi}$$

Herringbone Gear

- In order to avoid an axial thrust on the shaft and the bearings, the double helical gears or Herringbone gears are used.

Bevel Gears

- Use to transmit power between two intersecting shafts.
- High speed high power transmission.

Classification of Bevel Gear

- Mitre Gear: When two bevel gears are mounted on shafts that are intersecting at right angle.
- Crown Gear: In pair of bevel gear, when one of the gear has a pitch angle of 90º.
- Internal Bevel Gear: When the teeth of bevel gear are cut on the inside of the pitch.
- Skew Bevel Gear: Mounted on non-parallel and non-intersecting shafts. It constant of straight teeth.

- Hypoid Gear: Similar to skew bevel gear, non-parallel and non-intersecting shafts. It consists of curved teeth.
- Zerol Gear: Sprial bevel gear with zero spiral angle.
- Force Gear: Consists of a spur or helical pinion meshing with a conjugate gear or disk form.

- **Beam strength of bevel gear**

$$S_b = mb\sigma_b Y\left[1 - \frac{b}{A_o}\right]$$

Where, $\left[1 - \frac{b}{A_o}\right] =$ bevel factor.

- **Wear strength of bevel gears**

$$S_w = \frac{0.75\, b\, Q\, D_p K}{\cos\gamma}$$

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

Bearing

- A bearing is a mechanical element that permits relative motion between two components or parts, such as the shaft and housing, with minimum friction.

Plain Bearings (Sliding Contact Bearings)

- A plain bearing is any bearing that works by sliding action, with or without lubricant. This group encompasses essentially all types other than rolling-element bearings.[i]

Journal or Sleeve Bearings

- These are cylindrical or ring-shaped bearings designed to carry radial loads.
- The simplest and most widely used types of sleeve bearings are cast-bronze and porous-bronze (powdered-metal) cylindrical bearings.

Thrust Bearings

- This type of bearing differs from a sleeve bearing in that loads are supported axially rather than radially which is shown in the following figure. Thin, disk like thrust bearings are called *thrust washers.*

Bearing Materials

- **Babbitts**

- **Bronzes and Copper Alloys**
- **Aluminium**
- **Porous Metals**
- **Plastics**

Anti Friction Bearings

- *Ball, roller, and needle bearings* are classified as antifriction bearings since friction has been reduced to a minimum.

Bearing Loads

- **Radial Load**
    - Loads acting perpendicular to the axis of the bearing are called *radial loads.* Although radial bearings are designed primarily for straight radial service, they will withstand considerable thrust loads when deep ball tracks in the raceway are used.
- **Thrust Load**
    - Loads applied parallel to the axis of the bearing are called *thrust loads.* Thrust bearings are not designed to carry radial loads.

Ball Bearings

- Angular-contact bearings are used for combined radial and thrust loads and where precise shaft location is needed. Uses of the other two types are described by their names: radial bearings for radial loads and thrust bearings for thrust loads (See the following figure).

Radial Bearings

- *Deep-groove* bearings are the most widely used ball bearings. In addition to radial loads, they can carry substantial thrust loads at high speeds, in either direction.
- *Self-aligning* bearings come in two types: internal and external. In internal bearings, the outer-ring ball groove is ground as a spherical surface.Externally self-aligning bearings have a spherical surface on the outside of the outer ring, which matches a concave spherical housing.
- *Double-row, deep-groove* bearings embody the same principle of design as single-row bearings. Double-row bearings can be used where high radial and thrust rigidity is needed and space is limited.
- *Angular-contact thrust* bearings can support a heavy thrust load in one direction combined with a moderate radial load.

Thrust Bearings

- *Flat-race* bearings consist of a pair of flat washers separated by the ball complement and a shaft-piloted retainer, so load capacity is limited. Contact stresses are high, and torque resistance is low.
- *One-directional, grooved-race* bearings have grooved races very similar to those found in radial bearings.

- *Two-directional, groove-race* bearings consist of two stationary races, one rotating race, and two ball complements.

Roller Bearing (Rolling Contact Bearings)

- The principal types of roller bearings are *cylindrical, needle, tapered,* and *spherical.*
- They have higher load capacities than ball bearings of the same size and are widely used in heavy-duty, moderate-speed applications..

Cylindrical Bearings

- Cylindrical roller bearings have high radial capacity and provide accurate guidance to the rollers. Their low friction permits operation at high speed, and thrust loads of some magnitude can be carried through the flange-roller end contacts.

Needle Bearings

- Needle bearings are roller bearings with rollers that have high length-to-diameter ratios. Compared with other roller bearings, needle bearings have much smaller rollers for a given bore size.
- *Loose-needle* bearings are simply a full complement of needles in the annular space between two hardened machine components, which form the bearing raceways. They provide an effective and inexpensive bearing assembly with moderate speed capability, but they are sensitive to misalignment.
- *Caged assemblies* are simply a roller complement with a retainer, placed between two hardened machine elements that act as raceways. Their speed capability is about 3 times higher than that of loose-needle bearings, but the smaller complement of needles reduces load capacity for the caged assemblies.
- *Thrust bearings* are caged bearings with rollers assembled like the spokes of a wheel in a wafer like retainer.

Tapered Bearings

- Tapered roller bearings are widely used in roll-neck applications in rolling mills, transmissions, gear reducers, geared shafting, steering mechanisms, and machine-tool spindles. Where speeds are low, grease lubrication suffices, but high speeds demand oil lubrication, and very high speeds demand special lubricating arrangements.

Spherical Bearings

- Spherical roller bearings offer an unequaled combination of high load capacity, high tolerance to shock loads, and self-aligning ability, but they are speed-limited.
- *Single-row* bearings are the most widely used tapered roller bearings. They have a high radial capacity and a thrust capacity about 60 percent of radial capacity.
- *Two-row* bearings can replace two single-row bearings mounted back-to-back or face-to-face when the required capacity exceeds that of a single-row bearing.

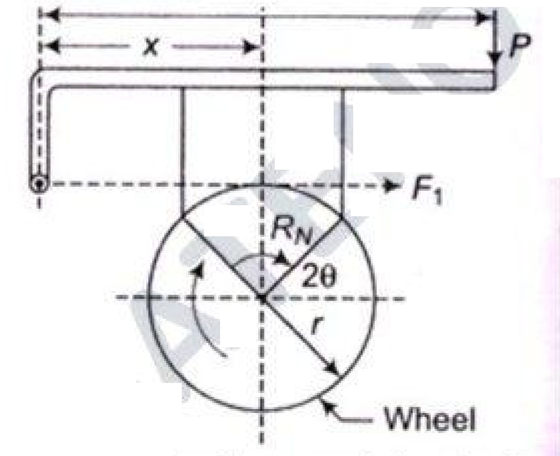
Brake

- A brake is a device by means of which artificial frictional resistance is applied to a moving machine member, in order to retard or shop the motion of a machine.

- The most commonly brakes use friction to convert kinetic energy into beat, though other methods of energy conversion may be employed.

Single Block or Shoe Brake

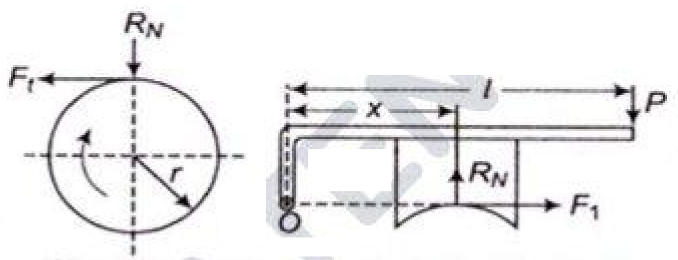
- It consists of a block or shoe which is passed against the rim of revolving brake wheel drum. The block is made of a softer material than the rim of the wheel.
- If the angle of contact is less than 60º then, it may be assumed that normal pressure or force between the block and the wheel is uniform.



A schematic diagram of shoe brake

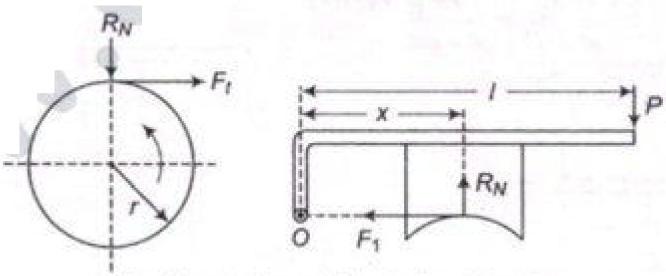
Case I: When the line of action of tangential braking force passes through the fulcrum O of the lever.

- If wheel is rotating in clockwise direction then, Free Body Diagram (FBD) of wheel and block is



Wheel is rotating anti-clockwise direction

- If wheel is rotating in anticlockwise direction then, FBD of wheel and block is



Wheel is rotating anti-clockwise direction

- Braking force

$$\mu R_N = \frac{\mu P l}{x}$$

- Braking torque

$$T_B = \frac{\mu P l r}{x}$$

- When wheel is rotating in anticlockwise direction then, the braking torque is same as above

$$T_B = \frac{\mu P l r}{x}$$

Case II: When the line of acting of the tangential braking force (F_t) passes through a distance a below the fulcrum *O*. Then, there are two cases:

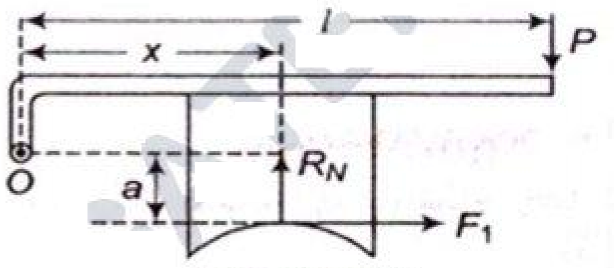
For Clockwise:

- Braking force

$$F_t = \mu R_N = \frac{\mu P l}{x + \mu a}$$

- Braking torque

$$T_B = \frac{\mu P l r}{x + \mu a}$$



FBD of block

For Anti Clockwise:

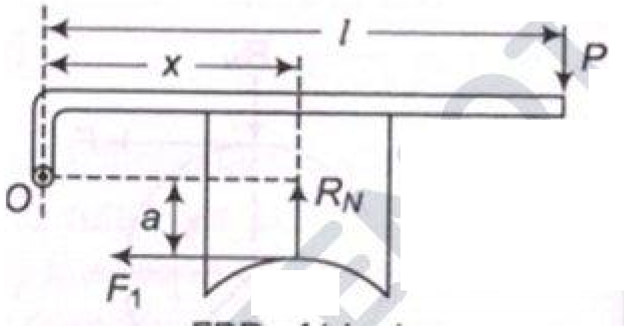
- Braking force

$$F_t = \frac{\mu P l}{x - \mu a}$$

- Braking torque

$$T_B = \frac{\mu P l r}{x - \mu a}$$

(as $T_B = F_t \times r$)



FBD of block

Case III: When the line of action of tangential braking force (F_t) passes through a distance 'a' above the fulcrum O.

For clockwise,

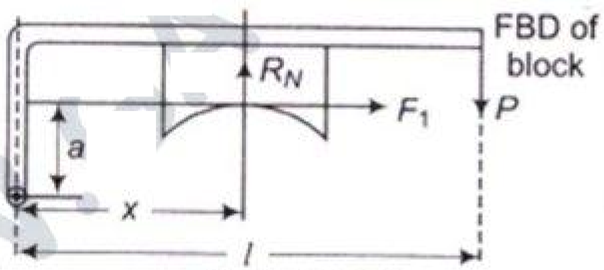
- Braking force

$$F_t = \frac{\mu P l}{x - \mu a}$$

- Braking torque

$$T_B = \frac{\mu P l r}{x - \mu a}$$

($T_B = F_t \times r$)



FBD of block

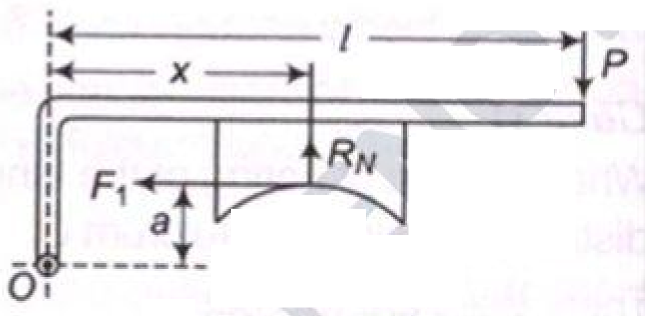
For Anti- Clockwise

- Braking force

$$F_t = \frac{\mu Pl}{x + \mu a}$$

- Braking torque

$$T_B = \frac{\mu Plr}{x - \mu a}$$



FBD of block

- When the frictional force helps to apply the brakes then, such type of brakes are said to self-energizing brakes.
- When P is negative or equal to zero then, these are known as self-locking brakes.

Simple Band Brake

- A band brake consists of a flexible band of leather, one or more ropes, or steel lined with friction material, which embraces a part of the circumference of the drum is called simple band brake.

We know,

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

$$(\theta = 360^\circ - \theta')$$

$$or\ 2.3\log\left(\frac{T_1}{T_2}\right) = \mu\,(360^\circ - \theta') = \mu\theta$$

- Braking force on the drum = $(T_1 - T_2)$
- Braking torque on the drum $(T_B) = (T_1 - T_2) \times r$
- When wheel rotates in the clockwise direction and taking moment about fulcrum O

$$Pl = T_1 \times b$$

$$T_1 = \frac{Pl}{b}$$

- For anticlockwise rotation of the drum $Pl = T_2 b$

$$T_2 = \frac{Pl}{b}$$

where, b = Perpendicular distance from O to the line of action T_1 or T_2

l = Length of the lever from the fulcrum

$T_1 = \sigma t \; wt$

w = Width of the band

t = Thickness of the band

σ_t = Permissible stress in the band.

Clutch

- A clutch is a mechanical device that provides for the transmission of power (and therefore usually motion) from one component (the driving member) to another (the driven member) when engaged, but can be disengaged.

Friction Clutch

- The friction clutch is used to transmit power of shafts and machines which must be started and stopped frequently.
- Friction surfaces of a clutch remain in contact to each other by applying an axial thrust or load w.

Considering Uniform Pressure

- The uniform pressure p can be evaluated as.

$$p = \frac{w}{\pi(r_1^2 - r_2^2)}$$

- **Total frictional torque** given in this case,

$$T = \frac{2}{3}\mu w \left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right) = mw R_m$$

where, Rm = Mean radius of friction surfaces

$$R_m = \frac{2}{3}\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right)$$

Considering Uniform Wear

- Total frictional torque acting on clutch

$$T = \frac{1}{2}\mu w (r_1 + r_2) = \mu w R_m$$

where R_m= Mean radius of friction surfaces $= \frac{r_1 - r_2}{2}$

- In uniform wear theory, Maximum pressure acts at the inner radius and minimum pressure acts at the outer radius.

$$p_{max} \times r_2 = c, \; p_{min} \times r_1 = c$$

$$\frac{p_{max}}{p_{min}} = \frac{r_1}{r_2}$$

- Average pressure on the friction surfaces

$$p_{av} = \frac{w}{\pi (r_1^2 - r_2^2)}$$

Multiple Disc Clutch

- Number of pairs of contact surfaces

$$n = n_1 + n_2 - 1$$

where, n_1 = Number of discs on the driving shaft

n_2 = Number of discs on the driven shaft

- **Total frictional torque** acting on the frictional surface

$$T = \eta \mu w R_m$$

where,

$$R_m = \frac{2}{3}\left(\frac{r_1^3 - r_2^3}{r_1^2 - r_2^2}\right) \text{ [in case of uniform pressure]}$$

$$= \frac{1}{2}(r_1 + r_2) \text{ [in case of uniform wear]}$$

where, r_1 and r_2 are outer and inner radii of the friction plates.

Cone Clutch

- In cone clutch, driver is keyed to the driving shaft by a sunk key and has an inside conical surface or face which exactly fits into the outside conical surface of the driven.

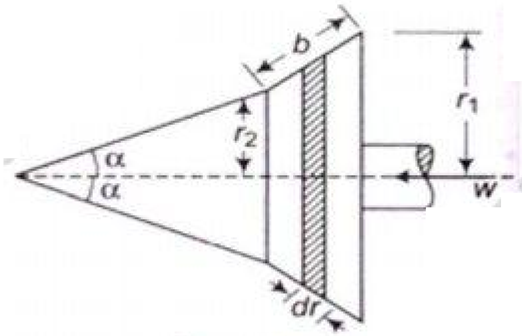
- Total torque on the clutch,

$$T = \frac{2}{3} \mu w \operatorname{cosec} \left[\frac{r_1^3 - r_2^3}{r_1^2 - r_1^2} \right]$$ (for uniform pressure)

$$= \frac{1}{2} \mu w \operatorname{cosec} \alpha \left(r_1 + r_2 \right)$$ (for uniform wear)

α = Semi angle of cone or face angle of the cone

$$w_n = \frac{w}{\sin \alpha}$$



Cone clutch

where, *w* = Axial load or thrust

- Axial force required for engaging the clutch

$$= w_n(\sin \alpha + \mu \cos \alpha)$$

- Axial force required to disengaged the clutch

$$w_d = w_n (\mu \cos \alpha - \sin \alpha)$$

- If face width b and mean radius of cone clutch is R_m.

Then,

$$R_m = \left(\frac{r_1 - r_2}{2} \right)$$

$$R_n = \frac{w}{2\pi R_m\, b\sin\alpha}, T = 2\pi\mu.P_n R_m b$$

- Power transmitted by clutch $= \frac{2\pi NT}{60}$

Centrifugal Clutch

- Centrifugal force acting on each shoe at running speed

$$p_c = m\,\omega^2 r$$

Where,

$$\omega = \left(\frac{2\pi N}{60}\right)$$

- Friction force acting on each shoe = $\mu(P_c - P_s)$

The direction of force is perpendicular to the radius of the rim pulley.

- Frictional torque on each shoe = $\mu(P_c - P_s) \times R$
- Total torque transmitted = Number of shoes $\mu(P_c - P_s)R$

$$= n\mu(P_c - P_s)R$$

Arc = Angle (in radian) × Radiua l = θR

Where, area of contact = *lb*

- Force exerted on each shoe = **plb**

$$P_c - P_s = lbp$$

Where, *l* = Contact length of the shoe

b = Width of the shoe

p = Pressure intensity on shoe

q = Angle made by shoe at the centre of spider in radian

R = Contact radius of shoe = inside radius of the rim of the pulley